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ENGINEERING DESIGN HANDBOOK

AUTOMOTIVE SERIES

THE AUTOMOTIVE ASSEMBLY

HEADQUARTERS, U. S. ARMY MATERIEL COMMAND

FEBRUARY 1965

HEADQUARTERS
UNITED STATES ARMY MATERIEL COMMAND
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*AMCP 706-355, The Automotive Assembly, forming part of the Automotive Series of the Army Materiel Command Engineering Design Handbook Series, is published for the information and guidance of all concerned.

(AMCRD)

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PREFACE

The Engineering Design Handbook Series of the Army Materiel Command is a coordinated series of handbooks containing basic information and fundamental data useful in the design and development of Army materiel and systems. The handbooks are authoritative reference books of practical information and quantitative facts helpful in the design and development of Army materiel so that it will meet the tactical and the technical needs of the Armed Forces.

The Automotive Assembly constitutes the first of a planned Automotive Series of handbooks concerned with the design of military automotive vehicles.

The design of satisfactory military vehicles, contrary to popular belief, does not happen as a by-product of normal progress in the civilian automotive industry. The military requirements must receive special consideration during design and development. The handbooks of the Automotive Series are a compilation of principles and data to supplement experience and education and assist engineers and designers in the development of military automotive equipment.

This handbook introduces the topics to be covered specifically by the succeeding handbooks and discusses these topics in their relationship to the automotive assembly as a whole. The scope of this handbook does not include design details of system elements, however, a certain amount of information more appropriate to the other volumes is included in this handbook so as to place it at the users' disposal as early as possible.

This work is a compilation of various data and design information gathered from numerous reports, publications and personal interviews, and its scope is necessarily limited to condensation and summary. For more complete information, the reader is referred to the References and the Bibliography appearing at the end of each chapter.

Titles and identifying numbers of the specifications, regulations and other official publications are given for the purpose of informing the user of the existence of these documents, however, he should make certain that he obtains editions which are current at the time of use.

This handbook was prepared by the Mechanics Research Division of the Illinois Institute of Technology Research Institute for the Engineering Handbook Office of Duke University, prime contractor to the U. S. Army Research Office-Durham. Technical supervision and guidance in this work was supplied by an ad hoc group with membership from the Major Subcommands of the Army Materiel Command. Chairman of the group was Daniel F. Smith of the Mobility Command.

This volume could not have been prepared without the excellent cooperation rendered by the Detroit Arsenal of the Mobility Command and the Development and Proof Services of the Test and Evaluation Command in providing reports, data and other information beneficial to the preparation of this handbook.

Appreciation is xpressed to the following civilian agencies also for assistance rendered i. this effort: Aircraft Armaments, Inc.; Allis-Chalmers Corporation; Americ. ı Ordnance Association; Bowen-McLaughlin-York, Inc.; Chrysler Defense E .gineering; Cleveland Ordnance Plant; Diamond "T" Company; Food Machinery and Chemical Corp.; Ford Motor Company, Special Military Vehicles Division; Fruehauf Trailer Company; General Motors Technical Center; International Harvester, Melrose Park Works; Mack Trucks, Inc.; Pacific Car and Foundry Company; Reo Division, White Motor Company; Townsend Engineered Products Company.

Elements of the U. S. Army Materiel Command having need for handbooks may submit requisitions or official requests directly to Publications and Reproduction Agency, Letterkenny Army Depot, Chambersburg, Pennsylvania 17201. Contractors should submit such requisitions or requests to their contracting officers.

Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office-Durham, Box CM, Duke Station, Durham, North Carolina 27706.

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CHAPTER 1

INTRODUCTION*

SECTION I GENERAL

The evolution of military vehicles has paralleled the evolution of civilian vehicles—starting with relatively simple designs and developing, as requirements and problem areas were explored, into the refined and highly complex machines present in the arsenals of today. Even now, this evolution of military vehicles is continuing, in response to new contingencies due to increased enemy capabilities, support requirements of new weapons, and a multitude of problems unique to the possible nuclear battlefield of the future.

During the evolution of military vehicles, many mistakes have been made and many lessons learned. These provide a valuable background from which the experienced designers of military vehicles can develop new concepts leading to vehicles with ever superior characteristics. As time progresses, however, these deans of military design retire, resign, or die; and, with their passing, the wealth of experience they have amassed is lost. New designers and engineers, handicapped by lack of this experience, are then required to carry forward the development of military vehicles. Furthermore, the actual outbreak of major hostilities usually intensifies development activities as the true enemy capabilities and our own deficiencies become more definitely known. This brings into the field of military design additional numbers of designers and contractors who have a limited background of experience in the requirements of military equipment. These designers and contractors engage in intense activities to determine the requirements that must be satisfied, to ascertain the capabilities and limits of standard equipment, and to become oriented, in general, in the policies and principles that govern the design of military equipment.

* Written by Rudolph J. Zastera of the Illinois Institute of Technology Research Institute, Chicago, Ill.

Unfortunately, the varied information sought after is not conveniently available. It exists in a multitude of Government publications, textbooks, reference manuals, technical reports, scientific documents, Army Regulations, Government specifications, miscellaneous Government directives, and in the minds of men. The complex task of gathering the needed information is compounded by the atmosphere of urgency that prevails during a time of national emergency and by the fact that the people who have the greatest need for this information, the neophyte design engineers, do not know what information is available nor where it can be obtained. This results in unavoidable mistakes, unnecessary delays, inefficiency, increased costs, and—worst of all—military equipment that falls short of the best that could be had if past and current records of accomplishment and technology were readily available and properly integrated.

In an effort to remedy this total situation, the Army Materiel Command supported a project for the purpose of developing a Series of Engineering Design Handbooks to consist of an integrated body of data covering the principles of materiel design. One of the major fields covered by this handbook is the design of military automotive vehicles. This major field is to be treated in a comprehensive series of handbooks referred to as the Automotive Series. The purpose of the Automotive Series is to guide designers and contractors by supplying them with a compilation of design principles, data, and information regarding the military requirements that must be satisfied in order for equipment to be suitable for operational use by the Army field forces and to embody satisfactory producibility and maintainability characteristics. In addition, this series of handbooks reflects the state of advancement of scientific and technical knowledge in the various fields concerned. Through periodic revision, the information given in these handbooks will be supplemented and kept current.

Summarized, the objectives of the Automotive Series of handbooks are to:

1. Provide a ready reference of design information to facilitate the development of new designs.
2. Provide a record of experience to forestall the duplication of past experiences and effort.
3. Provide a uniform approach to the design of military automotive vehicles.
4. Orient and guide design personnel and contractors in the principles and requirements relating to the design of military automotive vehicles.
5. Preserve knowledge now being lost when senior designers resign, retire, or die.
6. Present a packaged compilation of formulas, tables, values, and other information useful in military automotive design and not readily available in existing literature.

This handbook, entitled *The Automotive Assembly,* the first of the Automotive Series, serves as an introduction to the series. It contains a discussion of the various major elements, or systems, that make up the automotive assembly such as: the power plant, power train, steering system, suspension system, etc. The functions and characteristics of these major elements are described and their requirements, as related to the automotive assembly, are discussed. Design criteria, problem areas, and miscellaneous requirements that pertain to the design of the major elements themselves, or to the specific components that comprise these major elements, are treated in subsequent handbooks of the Automotive Series.

SECTION II DEFINITION

The term "automotive assembly" applies to a general category of mechanical land vehicles that contain a means of propulsion within themselves. They are usually considered to be either wheeled or track-laying vehicles or a combination of both; but, in the broad sense, this general category includes all types of walking, jumping, and crawling vehicles, as well as self-propelled sleds and various ground-effect—or air cushion-supported vehicles. These vehicles may have the ability to negotiate deep water barriers by either swimming on the surface, in which case they are amphibious automotive vehicles; by swimming submerged, in which case they are submarine automotive vehicles; or by propelling themselves along the bottom in the same manner that they are propelled on normal terrain, in which case they are described as having deep-water fording capabilities.

The military automotive assembly is usually designed to meet some primary function associated with the conduct of military operations. These are such functions as:

1. *To transport personnel*—either in close proximity to the enemy, as do armored, fully-tracked personnel carriers; in tactical areas not directly in contact with the enemy, as do tactical cargo trucks and ambulances; or in rear areas, communication zones, and in the zone of the interior, as do administrative vehicles and buses.
2. *To transport material*—either in close proximity to the enemy or in tactical situations not directly in contact with the enemy, as do armored, fully-tracked cargo carriers or the many, wheeled cargo trucks and truck-and-trailer combinations in use by the army.
3. *To serve as a prime mover*—the primary function of the various tractors and truck tractors used to tow artillery and trailers of all kinds.
4. *To provide armor protection*—the primary function of the early, World War I, tanks. In World War I, the successful employment of the machine gun pinned down the opposing armies in their labyrinths of trenches and barbed wire so effectively that the resulting stalemate resisted even the heaviest artillery concentrations. The first tanks were designed to give armored protection to the crew members while they endeavored to overcome the dug-in machine gun and barbed wire.

5. *To provide mobility for weapons systems*—the primary function of the self-propelled artillery or self-propelled antiaircraft type of automotive assembly or the self-propelled missile launcher of the modern army.
6. *To mount special-purpose equipment*—as in special shop trucks, truck-mounted radar units, mobile electronic warfare equipment, fire-fighting equipment, and others.
7. *Any combination of the foregoing categories.* Most military automotive assemblies have more than one specific function. This renders them versatile and achieves the greater economy of equipment necessary for the successful conduct of military operations. Thus, a tactical cargo carrier can easily serve as a personnel carrier and can readily provide mobility to rocket-type or recoilless-type weapons. By equipping the carrier with an appropriate pintle, it can serve as a prime mover as well. A combat tank provides armor protection for the crew, supplies mobility for elaborate weapons systems, is a tactical personnel carrier when it carries infantrymen piggy-back style, and with pintles and tow-bars may function as a prime mover. Further, by applying additional apparatus to standard military automotive assemblies, they may be equipped to perform various specialized, secondary functions. Thus, the addition of a bulldozer blade, appropriate actuating mechanisms, and controls, enables a standard battle tank to perform engineering missions requiring digging, ground leveling, or earth moving in close proximity to the enemy. The addition of other equipments may enable a standard vehicle to perform such secondary functions as: minefield breaching, placement of demolition charges under battlefield conditions, battlefield recovery of disabled vehicles, the placing and erecting of expedient bridges while under enemy fire, battlefield surveillance under conditions of poor visibility, and others.

In other words, the subject of this handbook encompasses all types of vehicles that are the responsibility of the Army Materiel Command with respect to design, development, manufacture, and service. This includes everything from motor scooters and motorcycles through the most highly developed combat tanks and ground-effect vehicles of the future. Trailers, vans, and special-purpose towed vehicles, although not self-propelled and, therefore, not automotive assemblies in the strictest sense of the definition, are also included, since many of the specifications and requirements covering the design of automotive vehicles also apply to these assemblies.

SECTION III PRINCIPAL ELEMENTS

The automotive assembly is considered to be comprised of a number of principal elements or systems. Each element, in turn, is comprised of a number of lesser components, each of which in itself is a subassembly of individual parts. This handbook devotes one chapter to the discussion of those requirements that apply to the overall vehicle assembly, such as physical limits, operational limits, environmental limits; and a separate chapter to a discussion of the components, characteristics, and design requirements of each of the principal elements. A list of the principal elements that comprise the automotive assembly, with a general discussion of their functions, follows.

1-1 THE POWER PLANT

The power plant is the integration of subassemblies and individual components required to convert the energy of some fuel source to a form useful to the vehicle. Thus, it includes not only the engine, or engines, but also the fuel systems, lubricating systems, cooling systems, exhaust systems, electrical systems, and all other necessary accessories. The engine need not be a reciprocating internal combustion type, although this type has been most favored for military vehicles. Rotary internal combustion types, external combustion types, steam types, turbine types, and others have been used and still are receiving attention. In any case, however, the function of the power plant is to convert the energy of the fuel into a form usable by the vehicle for propulsion, for the operation of weapons, and for the operation of accessories present on the vehicle. A detailed discussion of the various types of power plants appears as Chapter 7.

1-2 THE POWER TRAIN

The power train is the system of components that transmits the useful energy produced by the power plant from the output shaft of the power plant to its ultimate point of application, wheels or tracks for instance. It includes such components as: clutches, transmissions, transfer cases, drive shafts, differentials, axles, and brakes. A detailed discussion of the components that make up the power train appears as Chapter 8.

1-3 THE FRAME

The frame of an automotive assembly is the structure that supports the various components of the automotive assembly and maintains their spatial relationship. The frame provides strength and rigidity to the vehicle and enables it to carry the load placed upon it and to withstand the severe shocks, blows, twists, and vibrations to which it is subjected in operation. A detailed discussion of frame characteristics and design requirements appears as Chapter 9.

1-4 THE BODY OR HULL

The body or hull of an automotive assembly is that principal structure which houses the crew, passengers, or cargo; it is the major factor in giving the vehicle its characteristic appearance. The term "body" is usually applied to wheeled vehicles; "hull," to amphibious and tracked vehicles, especially the massively armored combat tank. When the hull of a vehicle is armored to withstand ballistic impact, it simultaneously achieves great structural rigidity and strength—so much, in fact, that the addition of a frame, whose purpose is rigidity and strength, is not necessary. For this reason, vehicles with hulls usually do not include a separate frame. The turret and cupola assemblies of tank-type vehicles are not considered part of the vehicle body or hull. A detailed discussion of bodies and hulls appears as Chapter 10.

1-5 THE SUSPENSION SYSTEM

The system of mechanical components between the vehicle frame or hull and the ground constitutes the suspension system. Its purpose is to provide a sprung or flexible support for the automotive assembly, while its function is to provide a smoother, more comfortable, ride, thereby allowing higher speeds, protecting delicate equipment that may be on board, and permitting greater stability and control of the vehicle. The main components of the suspension system are: the spring systems or torsion bars, shock absorbers, wheels and tires, road wheels and tracks, track supporting and tensioning components, load leveling systems, stabilizing equipment, and suspension lockout systems. A detailed discussion of these appears as Chapter 11.

1-6 THE STEERING SYSTEM

The steering system is that assembly of linkages and mechanical components which enables the vehicle operator to control the direction of the vehicle. For a front wheel-steered vehicle, control is effected by pivoting the front wheels on their supports in the direction that the vehicle is to travel; for a rear wheel-steered vehicle, by directing the rear wheels in a direction opposite. Steering of wheeled vehicles may also be accomplished by tilting the front wheels to obtain "camber steering." Control of tracked vehicles is effected by varying of speed of one track as compared with the speed of the other; tracked vehicles can also be steered by warping the tracks in the direction of steer. Articulated vehicles are sometimes steered by mechanically causing the leading sections to skew around at an angle to the original course and causing the trailing units to follow. A detailed discussion of various steering systems appears as Chapter 12.

1-7 THE ELECTRICAL SYSTEM

Electricity plays an important role in the modern automotive assembly—it powers engine starting and engine ignition and a multitude of accessory components. The chief users of electricity are: communications equipment, weapon traversing and elevating mechanisms, interior and exterior lights, a variety of heaters, and miscellaneous motors that operate sundry pumps, fans, and blowers. A detailed discussion of the electrical system appears as Chapter 13.

1-8 MISCELLANEOUS ELEMENTS

The miscellaneous elements of military automotive assemblies are those regular parts that cannot be classified under any of the foregoing categories—such as firefighting systems, intercommunications systems, heating systems, ventilating systems. A detailed discussion of miscellaneous elements appears as Chapter 14. Special or peculiar features often incorporated into military vehicles are also treated in this chapter.

CHAPTER 2

RESEARCH AND DEVELOPMENT, TEST AND EVALUATION, AND TYPE CLASSIFICATION

2-1 GENERAL

Effective design of military automotive materiel, and of all other military materiel, is dependent on research and development, test and evaluation, and other factors such as producibility, reliability and maintainability. This chapter presents information concerning research and development, and test and evaluation. It discusses the relationships among these functions and the responsibilities, objectives and requirements for their accomplishment. The closely related subject of type classification is also discussed. Particular consideration is given to those matters which are the concern of the Army Materiel Command (AMC). Nuclear developments, subject to special provisions, are not within the scope of this handbook.

Research and development extends from inception of ideas through investigation or discovery of re-research potentials, creation and testing of new or improved theories, techniques, processes, materials, or items; and evaluation and final acceptance or rejection for use by the Army.

2-2 RESEARCH POLICY

The Army's established policy is to conduct and support a broad program of basic and applied research with emphasis on that related to the needs of the Army. The chiefs of developing agencies and other appropriate commanders and chiefs are responsible for determination of applied research required to support their development programs; determination of appropriate areas of basic research and submitting recommendations to the Chief of Research and Development (CRD). New concepts and ideas from all individuals, units and agencies are encouraged.

2-3 MANAGEMENT OF RESEARCH AND DEVELOPMENT PROJECTS

Normally, chiefs of developing agencies manage research and development projects by using established organization structures and procedures; however, either Department of Army (DA) Headquarters or the chiefs of developing agencies may establish special operating procedures, to include project managers or materiel coordination groups, to expedite development of selected projects.

2-4 OBJECTIVES OF RESEARCH AND DEVELOPMENT

The ultimate objective of research and development is to develop weapons, equipment and techniques which are qualitatively superior to those of any potential enemy under all conditions of war.

Other objectives include the achievement of Qualitative Materiel Development Objectives (QMDO), and the development of materiel which satisfies Qualitative Materiel Requirements (QMR) or Small Development Requirements (SDR). Additional information on Army research and development and on organization and functions of DA and AMC will be found in Refs. 1-3.

2-5 QUALITATIVE MATERIEL DEVELOPMENT OBJECTIVES (QMDO)

A QMDO is a DA approved statement of a military need for development of new materiel, the feasibility of which cannot be determined sufficiently to permit the establishment of a qualitative materiel requirement. Broadly stated, it is a goal toward which research and component development efforts will be directed. For procedures prescribed for the establishment, modification or deletion of QMDO, see Ref. 4.

2-6 QUALITATIVE MATERIEL REQUIREMENTS (QMR)

The more significant Army requirements for new equipment or for major innovations or improvements are normally expressed as Qualitative Materiel Requirements. A QMR is a DA approved statement of a military need for a new item, system or assemblage, the development of which is believed feasible. The QMR is directed toward attainment of new or substantially improved materiel which will advance the Army's ability to accomplish its mission. It states the Army's major materiel needs in terms of military characteristics and priorities and relates materiel to the operational and organizational context in which it will be used. QMR's are stated at the earliest time after the need is recognized and feasibility of development has been determined.

Army staff responsibility for review, coordination, approval and modification of qualitative materiel requirements is vested in the Chief of Research and Development (CRD). He presents the coordinated QMR to the Materiel Requirements Review Committee for a final review to determine validity, requirements for a total feasibility study, intention to initiate a project and priority.

The Commanding General, Combat Developments Command (CDC) prepares and submits to DA Headquarters for approval all QMR for materiel to be used by units of the Army in the field. Using agencies prepare and submit to CDC proposed QMR for materiel to be used by units not of the Army in the field. The Commanding General, CDC reviews and comments on such proposed QMR insuring that they support established objectives and forwards them to DA Headquarters for approval. (Refs. 1, 5) A format for proposed QMR is shown in Ref. 1.

QMR's are published as separate documents. The summary of Section I of the QMR (Statement of Requirement) is placed in the Combat Development Objectives Guide (CDOG).

2-7 SMALL DEVELOPMENT REQUIREMENTS (SDR)

The SDR states a DA need for the development of equipment of proven feasibility which can be developed in a short time and which, because of low cost and simplicity of development, does not warrant the establishment of a QMR. Small development requirements are approved by DA Headquarters and published as Appendix E to CDOG. Recommendations for the establishment of SDR contain a brief description, purpose and cost of the development and a short justification of its need. Commanders of the Army elements of unified and special commands, Continental Army Command (CONARC), and other agencies submit recommended SDR to the Commanding General, CDC. The chief of the developing agency furnishes, upon request by CDC, its input to paragraph 3 of the SDR. Recommended SDR, in the format prescribed in Ref. 1, are submitted by the Commanding General, CDC to CRD for approval.

Major Army commanders, the chiefs of developing agencies and commanders of other interested commands and agencies review their portions of the SDR lists for validity, project status, priority and other information and submit recommendations to the Commanding General, CDC, who reviews and forwards them to DA Headquarters with comments or recommendations.

2-8 RESPONSIBILITIES

Research, development and design relating to automotive vehicles, and to a large majority of Army items, fall within the general responsibility of AMC. (Refs. 1, 3) specific responsibilities for these functions are delegated to appropriate Subcommands in accordance with their missions. Under such delegation the Mobility Command (MOCOM) (Ref. 6) is responsible for automotive vehicles except for combat vehicles, responsibility for which is delegated to the Weapons Command (WECOM). (Ref. 7)

The Test and Evaluation Command (TECOM), a major subordinate command of AMC, plans and conducts engineering and service tests of Army materiel for AMC; provides test and evaluation services and support to development agencies and project managers; and participates in the planning and preparation for troop tests involving Army materiel. (Ref. 8)

CRD exercises Army staff responsibility for planning, programming, coordinating and supervising all Army research, development, test and evaluation. (Refs. 1, 2)

The Deputy Chief of Staff for Military Operations (DCSOPS) has Army staff responsibility for overall staff supervision and coordination of combat developments and related policy in conjunction with research and development functions assigned to CRD. (Refs. 1, 4)

CDC directs combat development activities and submits recommendations to DA Headquarters for establishing materiel development objectives and for specific materiel requirements based on these objectives. (Ref. 5) The Air Defense Command (ADC) is responsible for submitting through CDC to DA Headquarters requirements for materiel to be used primarily in air defense of continental U.S.

Research, development, test and evaluation of assigned materiel are also performed by The Surgeon General and the Chief of Engineers (Ref. 1) and the Chief, Army Security Agency. (Ref. 9)

Responsibilities for reviewing appropriate research and development are assigned to DCSOPS, the Deputy Chief of Staff for Logistics (DCSLOG), the Assistant Chief of Staff for Intelligence, the Commanding General, Army Intelligence Center and the Chief Signal Officer. The Chief of Transportation is responsible for monitoring the research and development program for conformity with AR 705-8, i.e., the *Department of Defense Engineering for Transportability Program.* (Ref. 10)

The Army Research and Development Review Board, composed of representatives of Army General Staff agencies, reviews the research and development program to insure that it is in support of DA and DOD plans and guidance, and that it is a balanced effort.

The Materiel Requirements Review Committee, an Army staff committee, reviews each QMR before DA approval is granted, and conducts final review of total feasibility on selected projects before the decision to initiate a major end item development project.

2-9 INITIATION OF DEVELOPMENTS

The development process is initiated with the approval of a QMR or an SDR in response to a recognized development objective. The development process will be considered to have ended when the item is type classified as an adopted category or the project is canceled.

2-10 STUDIES AND EVALUATION

In addition to continuous study by all agencies to take advantage of the state-of-the-art, special studies may be conducted at any appropriate step during research and development by CDC, using or developing agencies, or as directed by DA Headquarters.

2-11 FEASIBILITY STUDIES

DCSOPS is responsible for determining total feasibility for each major research and development system. During review of the QMR, the Materiel Requirements Review Committee selects those for which a total feasibility study must be conducted. After selection, DCSOPS prepares the study. DA Headquarters directs project initiation, when so recommended by the Materiel Requirements Review Committee, after a review of the completed feasibility study and after considering the effects on all other programs and projects in the Army Long Range Capabilities Plan.

2-12 PRIORITIES

The Chief of Research and Development has Army staff responsibility for coordinating and approving priorities for research and development projects which normally will be the same as the priorities of the QMDO, QMR or SDR which they support. For description of priorities and the basis of assignments, see Ref. 1.

2-13 TEST AND EVALUATION

The process of development requires evaluation of the product by test to obtain performance data and to determine whether the product is satisfactory for its intended use. Army staff responsibility for planning, coordinating and supervising all materiel testing rests with CRD.

Materiel under development by AMC and its agencies is subjected to tests and evaluations, as indicated below, prior to type classification. Further information concerning these and other tests and evaluations will be found in Ref. 11.

Research Test. Test conducted during the research phase in order to confirm concepts and to further research projects and tasks. Responsibility rests with the agency assigned the research project or task.

Feasibility Test. The determination by a process of technical examination and study of the possibility of attainment of end item materiel development. Feasibility tests are the responsibility of the commodity command, project manager or separate installation or activity reporting directly to Headquarters, AMC, that is assigned the development task or project.

Engineer Design Test. Test conducted by or under the control of the design agency where the objective of the test is to determine inherent structural, electrical or other physical or chemical properties of construction materials, a component, subassembly, or prototype assembly, item or system, including the effect of environmental stresses on these properties. Responsibility for engineer design test is as indicated for feasibility test. Developing agencies assure participation by TECOM.

Research and Development Acceptance Test. A test conducted by the developing agency of an item or system designed and developed by a contractor, to insure that the specifications of the development contract have been fulfilled. Acceptance of the item or system for engineering testing is contingent on the research and development acceptance test. Responsibility for research and development acceptance tests is as indicated for feasibility tests.

Engineering Test. A test, using an engineering approach, conducted by or under the supervision of a separate test agency, not a part of the developing installation or activity concerned, where the objective of the test is to determine the technical performance and safety characteristics of an item or system and its associated tools and test equipment as described in the QMR, the technical characteristics, and as indicated by the particular design. The test is characterized by controlled conditions and the elimination of human errors in judgment, as far as possible, through the utilization of environmental chambers; physical measurement techniques; controlled laboratory, shop, and field trials; statistical methodology; and the use of personnel trained in the engineering or scientific fields. The engineering test provides data for use in further development and for determination as to the technical and maintenance suitability of the item or system for service test. TECOM is responsible for coodinating and establishing the test objectives, preparation and approval of the plan of test, conduct and report of the test, and evaluation and distribution as directed of the report of test.

Service Test. A test conducted under simulated or actual field conditions where the objective is to determine to what degree the item or system and its associated tools and test equipment perform the the mission as described in the QMR, and the suitability of the item or system and its maintenance package for use by the Army. This test is characterized by qualitative observations and judgment of selected military personnel having a background of field experience with the type of materiel undergoing test, with instrumentation limited to those measurements of characteristics of major operational significance. The test is conducted using soldiers representative of those who will operate and maintain the equipment in the field. The service test provides the basis for recommendations on type classification. Responsibility of TECOM is as indicated above for engineering tests.

Engineering/Service Test. This designation is given to engineering and service tests which have been either completely or partly integrated.

Check Test. A retest performed on a service test model of selected items to determine whether major deficiencies found in the service test have been corrected, these deficiencies being of such nature that the item was found unsuitable for type classification.

2-14 "IN-PROCESS" REVIEWS (IPR)

An IPR is a review of a materiel development project conducted at critical points of the development cycle for the purpose of evaluating the status of the project, accomplishing effective coordination, and facilitating proper and timely decisions bearing on the future course of the project to assure the materiel's ultimate acceptability for use by the Army. Ultimate acceptability is judged by evaluating the status of the project against the characteristics of the QMR and the appendix thereto. Particular emphasis is on eliminating unwarranted complexities and characteristics which are marginal or which make unnecessary provisions for safety, comfort, and ease of operation. A formal IPR is conducted by a conference among representatives of all agencies concerned. An informal IPR is normally conducted by correspondence, with the same participants as for a formal IPR. At the time of approval of a QMR, DA Headquarters states whether or not formal IPR's are required.

The IPR's are normally held at the following points in the development cycle and for the purposes indicated:

a. Technical Characteristics Review. Held upon receipt by the developing agency of the QMR or SDR and prior to finalizing the technical characteristics. The primary purpose of the review is to insure that the developer understands the requirement and has properly stated it in terms of technical characteristics.

b. Engineering Concept Review. Held upon completion of the engineering concept to insure that the contractor or in-house facility is not commencing a program that is beyond the state-of-the-art or contains too many high risk areas. It also assures that all feasible engineering approaches are being utilized.

c. Design Characteristics Review. Held upon completion of determination of the design characteristics and prior to release of the design for development. Appropriate consideration must be given to updating the QMR, if necessary, to avoid developing of hardware that does not fully satisfy the stated requirements.

d. Prototype Systems Review. Held after delivery of prototype development hardware.

e. Service Test Review. Held prior to commencement of the service test or combined service test/engineering test to insure that all aspects of the test program, both completed and to be conducted, thoroughly measure the ability of the materiel to meet the QMR.

Further information concerning IPR's may be found in References 1 and 12.

2-15 TYPE CLASSIFICATION (REF. 13)

For the purpose of recording the status of items of materiel from the standpoint of development and suitability for service use, items are type classified by appropriate Technical Committee action. (Ref. 14) Type classification and reclassification actions serve to obtain and record DA decisions on the current status of materiel relative to the Army supply system and to facilitate planning for orderly and economical phasing of items into, or out of, the supply system.

Items are authorized status in one of the following types:

a. Development Category.

(1) *Development Type.* Materiel being developed or tested to meet approved qualitative materiel requirements or small development requirements.

(2) *Limited Production Type.* An item under development, commercially available or available from other Government agencies, for which an urgent operational requirement exists and for which no other existing item is adequate; which appears to fulfill an approved qualitative materiel requirement or other DA approved requirements, and to be promising enough operationally to warrant initiating procurement and/or production for troop issue prior to completion of development and/or test or adoption as a standard item.

b. Adopted Category.

(1) *Standard Types.* The standard types designate the items that have been adopted as suitable for Army use (or other agencies when the Army is the supply agency); which are acceptable as assets to meet operational requirements; are authorized for inclusion in equipment authorization documents; and are described in published adopted item lists. There may be more than one standard type or more than one item of any specific standard type to fulfill the same requirement. Standard types are subdivided as follows:

(a) *Standard A.* The most advanced and satisfactory items currently available to fill operational requirements.

(b) *Standard B.* Items which have limited acceptability to fill operational requirements. These items are normally used and issued as substitutes for Standard A items.

(c) *Standard C.* Items which have only marginal acceptability for operating requirements, and are being forced out of the system as stocks of more acceptable items become adequate to meet requirements.

(2) *Limited Standard Type.* Items which are not acceptable for Army operational requirements and are not, therefore, counted as assets against operational requirements. Items in this category are limited to those which are useful in training or those which are being retained at the request of DCSLOG to meet peculiar requirements other than training.

(3) *Obsolete Category.* Items which are no longer acceptable for Army use.

REFERENCES

1. AR 705-5, *Army Research and Development.*
2. AR 10-5, *Organization and Functions, Department of the Army.*
3. AR 10-11, *Organization and Functions, U. S. Army Materiel Command.*
4. AR 71-1, *Army Combat Developments.*
5. AR 10-12, *Organization and Functions, U. S. Army Combat Developments Command.*
6. AMCR 10-20, *Mission and Major Functions of the U. S. Army Mobility Command.*
7. AMCR 10-21, *Mission and Major Functions of the U. S. Army Weapons Command.*
8. AMCR 10-24, *Mission and Major Functions of the U. S. Army Test and Evaluation Command.*
9. AR 10-22(C), *Organization and Functions, U. S. Army Security Agency* (U).
10. AR 705-8, *Department of Defense Engineering for Transportability Program.*
11. AMCR 70-7, *Test and Evaluation of Materiel.*
12. AMCR 70-5, *''In-Process'' Reviews of Materiel Development Projects.*
13. AR 700-20, *Type Classification of Materiel.*
14. AR 705-9, *Technical Committee Functions.*

CHAPTER 3

GENERAL REQUIREMENTS*

The process of designing any complex mechanism involves the careful planning, selection, development, proportioning, and arranging of the various components that comprise the overall assembly so that all requirements are satisfied. When this procedure is applied in designing military automotive vehicles, the impossibility of completely satisfying all requirements becomes quickly apparent. Many requirements are not compatible, for example: high ground clearance with low vehicle silhouette; maximum armor protection with minimum weight; maximum reliability, ruggedness, and crew comfort with minimum size, weight, and cost. This makes it necessary for the designer to evaluate the relative importance of the requirements specified and arrange them in a definite order of emphasis, in accordance with the missions intended for the vehicle being designed. He then proceeds to develop a concept of the vehicle, selects the various components that satisfy his requirements, and carefully organizes the available space and allowable weight for maximum utilization. He strives to satisfy all requirements, that can be satisfied practically and economically, and makes studied compromises, in areas where requirements are incompatible, based upon the order of emphasis he established. Since all requirements can seldom be completely satisfied, he endeavors to satisfy completely those that rank high in the order of emphasis, even, if necessary, at the expense of those less important. The resulting vehicle is, therefore, the best combination of interrelated compromises that were possible at the time.

* Written by Rudolph J. Zastera of the Illinois Institute of Technology Research Institute, Chicago, Ill.

SECTION I THE MILITARY ENVIRONMENT

Since this handbook is intended primarily for engineers who have a limited knowledge of the principles of military design, it is paramount that they understand, and fully appreciate, the rigors to which military vehicles are exposed and the gruelling punishment they are expected to survive with no impairment of their operation. The military environment is the most severe possible, from a vehicle operations viewpoint, with no civilian counterpart approaching it in severity. The most common error made by inexperienced military designers is their gross underestimation of this environment, or the assumption that the operating environment of road building or logging machinery is comparable to the military environment. Nothing could be farther from the truth.

The term "military environment" arouses visions of the battlefield—of bombardment, frantic movement, and chaos. A detailed, thoughtful examination of this scene reveals the impact these conditions have upon the requirements of the military vehicle. Since battles can be expected anywhere on the earth, military vehicles are required to have the capacity of operating in the frigid temperatures of the arctic and the intense heat of desert regions as well as in temperate zones. They must possess a high degree of off-the-road mobility in deep powdery arctic snows, in clinging, sucking swamp mud, over the drifting sands of desert and beach dune areas, over hard, rocky terrain, and also on paved roads. They must be sufficiently rugged to withstand the vibrations, shocks and violent twisting experienced during cross country travel over rough terrain, be capable of operating for long periods with very little or no maintenance, and be of minimum weight and size to facilitate airborne operations. In addition, they must be able to withstand the punishment attributable to drivers and crews that have had only limited training and who may be frequently suffering from extreme fatigue and fright. Furthermore, the use, or threatened use, of nuclear weapons foreseen in future warfare, makes it necessary to avoid concentrations of forces and equipment at bridge sites. This makes deep-water fording, or swimming, capabilities necessary for military vehicles to enable them to cross water barriers without approaches for launching or landing.

Both combat and tactical vehicles (see Chapter 4) are exposed to this environment, with the conditions being only slightly less stringent for the tactical vehicles. The capability of moving against the enemy (or of breaking contact during a retrograde movement) quickly and via the tactically most advantageous routes, despite an unfavorable terrain, is one of the prime requirements of combat vehicles. Tactical vehicles, too, must possess a high degree of cross country capability to enable them to support the operations effectively; but these vehicles can select, to a degree, routes that take advantage of more favorable terrain.

Commerical vehicles are unsatisfactory in combat operations as are also most commercial components, simply because the military environment is so much more severe than the conditions for which the commercial components were designed. The use of commercial components in tactical vehicles is more feasible, but they must be protected, in the military environment, by ancillary equipment. Experience has shown, however, that commercial vehicles without appropriate modifications are unsuitable when placed in tactical roles.

The engines of commercial vehicles in use on typical civilian hauling missions are readily expected to have service lives in excess of 100,000 miles. These same engines, when used in a military environment, fail in 2,000 miles of service. Medium truck engines used in the African campaign during World War II averaged 13,000 miles before they required rebuilding, while in the mountains of Italy engine life averaged only 5,000 miles. On the Red Ball Express in Europe, about 10,000 miles of service was realized from medium truck engines (Ref. 2). This disparity between the average service lives of vehicle engines used in civilian vs military operations is attributable to the military environment, a term that includes the effects of climate, terrain, and operator abuse.

These vehicles of World War II were operated predominantly on roads—not always first-class roads, to be sure, but better than the conditions encountered in off-the-road operations. The concept of future wars, envisioned by military planners, places maximum emphasis on the rapid development of troops and equipment. Task forces, comprised of widely scattered units, will be required to mass rapidly on any given target for a concerted strike and separate, with equal dispatch, back to their dispersed positions to minimize the threat of nuclear attack by the enemy.

Tactics of this kind are not feasible to a road-bound army because of the magnitude of the engineering effort that would be required to build and maintain an adequate road net to support such operations. Instead, vehicles will operate off-the-road, wherever necessary, to minimize the time required to make the strike and disperse and to refrain from being channelized into zones that are advantageous to the enemy. This requirement for off-the-road capability places even more stringent demands upon future vehicles than were placed upon their World War II counterparts.

Since the original military vehicles were adaptations of the then current commercial vehicles, it has been a popular belief that the development of military vehicles will take place as a by-product of the progress made in the civilian automotive industry. While it is true that certain components of civilian vehicles may be used to advantage in some military vehicles, and certain techniques of the civilian automotive industry may be applied, this belief is generally false. The proficiency of the highway engineer has had a great influence upon the development of the civilian automotive vehicle; and, were it not for the development of safe, smooth roads offering good traction and gentle slopes, civilian vehicles would be quite different today from what they are. In fact, it has been pointed out that, since early man made his first cart and found it difficult to propel through mud, soft sand, or rock-strewn terrain, he has been building roads to make the terrain compatible to his vehicle (Ref. 3). There is little evidence to indicate that he will not continue to develop the operational environment of his vehicles into ever smoother, faster, and safer roadways. Similarly, there is little doubt that the future design of civilian vehicles will continue to be influenced by improvements made in its operational environment.

The off-the-road requirements of the military vehicle preclude any efforts to improve the terrain over which the vehicle must travel. Instead, the need exists for extensive studies of the off-the-road environment to determine the principle governing the terrain-vehicle relationships. The development of military vehicles must be based upon these terrain-vehicle relationships and can conceivably lead to a morphology of vehicles radically different from those found in the civilian environment. It is obvious, therefore, that as the development of civilian vehicles becomes more and more dependent upon good roads, their suitability for military operations decreases. Thus, the development of vehicles compatible with the military environment cannot come about as a by-product of civilian vehicle development.

While it is true that certain segments of the automotive industry have developed off-the-road type vehicles for use in road construction, logging, and the oil fields, there is actually little in the operational environment of these vehicles that is common to the military environment. Typical road building machinery possesses only mediocre cross country capabilities. Access roads are required to permit the equipment to negotiate unfavorable terrain. The nature of the road bed being prepared, and upon which the road building equipment is operating, is such that it favors vehicle mobility. In fact, the entire purpose of the roadway in preparation is to improve the trafficability of the existing terrain to facilitate vehicular movement. The military vehicle, however, is required to traverse adverse terrain as that terrain actually exists, without the assistance of access roads or temporary delays while awaiting more favorable weather.

Certain logging and oil prospecting vehicles are designed to have off-the-road capabilities. The objectives of these vehicles, however, are so different as to make them almost totally unrelated to military vehicles, particularly in the area of size and weight. Furthermore, these vehicles have more freedom in the selection of their routes, and they do not operate in the same climate of urgency experienced by a combat vehicle.

Another, and perhaps the most important, factor that contributes to the severity of the military environment is the equipment operator (Refs. 4, 5, 6). The personnel selection process skims off the highest caliber men for leaders, combat crews, radio operator, and other skilled specialties leaving some of the least capable men to drive the vehicles, particularly the wheeled vehicles. They are generally young and immature, inexperienced, unreliable, irresponsible, and have a devil-may-care attitude toward their vehicles. At best, they have only a limited comprehension of the effect that operator neglect in recognizing and reporting minor problems has upon vehicle reliability and life, or the serious consequence of lubrication and cooling system neglect. Furthermore, they don't care —especially in units where vehicles are not assigned to specific drivers, but are assigned to a pool and driven by many drivers. Under these conditions, it is practically impossible to determine which driver was responsible for a particular neglect or abuse.

In contrast, the civilian vehicle driver employed by a large fleet operator is carefully selected. He is usually more mature, more experienced, and is generally more reliable and responsible than his military counterpart. Driving a vehicle is his means of earning a livelihood. He is fully aware of the importance of adequate maintenance, and realizes that deadlined equipment may mean loss of pay to him, and may even cost his job. The military operator has no such fears. In fact, deadlined equipment may even bring free time to the military driver.

Thus, driver abuse and neglect are major contributing factors to the severity of the military environment. A more judicious selection of driver personnel, along with improved driver training, would do a great deal to reduce these factors; but it is doubtful that they can be eliminated. Thus, they will remain as an additional problem for consideration by the vehicle designer.

With the advent of chemical, bateriological, and radiological warfare, the military environment is becoming exceedingly severe. This type of warfare requires additional protection for the personnel of combat and tactical vehicles in the form of sealed and protected personnel compartments, the use of recirculated air, and provisions for the personal requirements of the crew during extended periods of time. The sealing of personnel compartments and their insulation against nuclear radiations brings problems of providing effective vision outside the vehicle and requirements for remotely controlled weapons.

A new problem, stemming from the sealing and insulating of crew compartments against nuclear radiation on certain experimental vehicles. The insulating techniques employed on these vehicles screened out battle sounds. Subsequent evaluation tests revealed that a consciousness of the sounds about them are necessary to the efficient performance of combat crews. As a result, auxiliary equipment was needed to restore sound orientation.

These are certainly factors that are unique to the military environment. Other factors are: the need for protection against conventional weapons, and the requirements resulting from the long battle day specified for military vehicles. As applied to combat tanks the battle day specified is twenty-four hours long; while with respect to armored personnel carriers, the battle day is specified as three days long. In addition to being able to operate for these periods of time without refueling or maintenance, the vehicle must provide for all of the crew requirements during this period of time.

The required degree of protection from conventional weapons is specified in the military characteristics for each type of vehicle; it may be anything from no protection other than that afforded by the structural components of the vehicle, to the maximum practicable for protection against direct hits by large caliber high explosive and armor-piercing projectiles. The application of armor to a vehicle brings additional problems of weight, size, power requirements, ventilation, and vision. The length of the battle day imposes problems of fuel capacity, maintenance requirements, ammunition storage, and requirements of the crew.

In summary, the following salient features are enumerated as characteristics of the military environment:

1. High shock and vibration produced by off-the-road operations over rough terrain, airdrop operations, high explosive blast, and ballistic impact (see par. 3-4).
2. Extreme temperature ranges, extending from arctic to tropical (see par. 3-2.3).
3. Operations under conditions of extreme dust.
4. Operations in deep mud.
5. Operations in snow and on ice.
6. Amphibious operations in both fresh and sea water.
7. Operations under conditions conducive to corrosion and fungus growth.
8. Mountain operations involving extremely long, steep grades and side slopes (see par. 3-2.1).
9. Extended operations under conditions of low speed and high load.
10. Operator abuse in the form of overload, misuse, improper maintenance, and neglect.

SECTION II LIMITING DESIGN FACTORS

The design of military vehicles is governed by various restrictions that limit and control certain features of the completed vehicle. These limits and other factors affect dimensional, as well as operational, aspects of the vehicle and have been standardized to the point that many are included in Army Regulations. The need for these restrictions resulted from such considerations as: the need for unrestricted transportability of the vehicle by road, rail, air, and seagoing vessels; from the need for standardization to simplify supply and maintenance problems; from the need for tactical mobility under adverse conditions of terrain and climate; and from certain theoretical and empirical military requirements. Detailed, comprehensive limits are usually specified in the military characteristics for each developmental vehicle. Some of these factors that determine the general requirements for military automotive assemblies are discussed here to serve as guides in vehicle design.

3-1 PHYSICAL LIMITS

3-1.1 GENERAL

The policy of the Department of Defense with regard to the transportability of items of materiel is set forth in AR 705-8, *Department of Defense*

Engineering for Transportability Program, dated December 1959. It directs that "transportability (see Glossary) will be a major consideration when formulating the priority of characteristics to be considered in the design of any new item of material and equipment." Material and equipment being developed for use by the military departments must be of such gross weight and outside dimensions as will permit ready handling and movement by available transportation facilities. Special or unique arrangements of schedules, right-of-ways, clearances, or other operating conditions, will be undertaken only in exceptional cases and after first obtaining approval from the appropriate Transportability Agency.

The following Departmental agencies have been designated as "Transportability Agencies" to implement the *Department of Defense Engineering for Transportability Program*, AR 705-8.

1. Department of the Army: The Chief of Transportation, Washington, D. C.
2. Department of the Navy: Chief, Bureau of Supplies and Accounts, Washington, D. C.
3. Department of the Air Force: Headquarters, Air Research and Development Command, Washington, D. C.
4. U. S. Marine Corps: Commandant, Marine Corps, Washington, D. C.

The dimensions and weight of an item may be adjusted to suit the capabilities of specific modes of transportation when the item will require no further transportation under peacetime or mobilization conditions. In addition, due consideration must be given by designers to the following:

1. Many military items may be subjected to movement by several modes of transportation. The factors that govern the choice of mode, or modes, are mainly availability and capability of facilities, destination time requirements, operating conditions, and cost.
2. Combat and rough terrain equipment is subject to the normal transportation conditions, from manufacturer to off-the-road destinations, as apply to all other equipment.
3. Transportation systems in overseas areas are generally more restrictive than in the continental United States.
4. The first step toward rapid mobility of military forces is the proper design of equipment to meet the operational characteristics while adhering to the fundamentals of transportation capabilities.

When determining the maximum dimensions and weight of an automotive assembly, the designer must consider the limitations of the particular modes of transportation specified, or selected, for the equipment he is designing. These limitations are specified in the following regulatory documents:

1. The Highway Weight and Size Limitations, established by the Federal Aid Highway Act of 1956, a majority of the state laws of continental United States, and the physical limitations of highways in foreign countries to accommodate the potential volume and type of traffic anticipated.
2. The Outline Diagram of Approved Limited Clearances of the Association of American Railroads referred to in Car Service Rule 14, Section 2 (e), with weight limitations of individual carriers, shown in the current issue of the Railway Line Clearance publication for individual railroads of the United States, Canada, Mexico, and Cuba.
3. Diagram of the Berne International Rail Interchange Agreement with weight limitations applicable to the railroads of individual countries for all items that may require transportation by rail in foreign countries.
4. Loading and stowage limitations of ocean vessels, related factors, and Army and Navy procedures therefor.
5. Regulations of the Department of the Treasury (U.S. Coast Guard), the U.S. Army Mobility Command, and the Navy covering water transportation.
6. Regulations and instructions of the Army, Navy, Air Force, Marine Corps, Federal Aviation Agency, and Civil Aeronautics Board for loading cargo and combat aircraft.

For convenience to the designer, some of the main data has been extracted from these documents and is presented in the following subsections.

3-1.2 HIGHWAY TRANSPORTABILITY

3-1.2.1 Wheeled Vehicles

AR 705-8 prescribes maximum dimensions and weights for pneumatic-tired, highway- and off-the-road-type vehicles for unrestricted highway operations. These are as follows:

3-1.2.1.1 Width

Maximum overall width of a wheeled vehicle shall not exceed 96 in. No part of the vehicle, fixtures, or equipment that is attached or placed permanently upon the vehicle shall protrude beyond the outer face of the tires by more than 9 in. on either side of the vehicle.

3-1.2.1.2 Height

The maximum overall height of wheeled vehicles designed for highway operation in the continental United States shall not exceed 150 in. (12 ft 6 in.). The maximum permissible height for wheeled vehicles designed for overseas highway operations shall not exceed 132 in. (11 ft 0 in.).

3-1.2.1.3 Length

The maximum overall length of a wheeled vehicle comprised of a single, nonarticulated unit shall not exceed 35 ft. The maximum overall length of a wheeled vehicle comprised of an articulated double unit, such as a truck tractor coupled to a semitrailer or a truck tractor coupled to a full trailer, shall not exceed 50 ft. These dimensions exceed the statutory limitations for unrestricted highway movement in certain states in the continental United States. In these states, a special permit is required. Table 3-1 shows the size and weight limitations imposed by the various states on truck tractor-semitrailer combinations as of August 30, 1959.

3-1.2.1.4 Axle Loading

An axle load is defined as the total load transmitted to the road by all wheels whose centers are included between two parallel transverse vertical planes 40 in. apart, extending across the full width of the vehicle. The maximum axle load (subject to gross weight limitations) is 16,000 lb for axles spaced between 3½ ft and 7½ ft from the nearest adjacent axle in both the continental United States and overseas. For axles spaced more than 7½ ft from the nearest adjacent axle, the maximum permissible load is 18,000 lb, for vehicles designed for highway operation in the continental United States, and 16,000 lb, for vehicles designed for overseas operations.

3-1.2.1.5 Gross Weight

The maximum permissible gross weight of a wheeled vehicle designed for highway operations (subject to axle load limitations) shall not exceed 36,000 lb, for vehicles having a distance of 10 ft or less between the extreme front and rear axles, and shall increase by 850 lb for each additional foot of extreme axle spacing in excess of 10 ft to a maximum gross weight of 60,000 lb. This maximum gross weight exceeds the statutory limits of some states within the continental United States (see Table 3-1). For operations in these states, a special permit is required.

3-1.2.2 Tracked Vehicles

3-1.2.2.1 General Limits

AR 705-8 prescribes limiting dimensions for tracked vehicles in accordance with their gross weight. These limits are necessary for unrestricted operations on highways, in both the continental United States and in overseas areas, and are chiefly governed by bridge widths. Table 3-2 gives these limiting dimensions and weights. The values given for width of ground contact are the total widths of all ground contacting elements; e.g., for conventional vehicle having two tracks, the width of ground contact equals twice the width of one track.

It should be noted that the values given in Table 3-2 apply to tracked vehicles that are to be capable of unrestricted movement on highways and bridges. When vehicles that exceed these specifications are being designed, they must be approved by the appropriate departmental Transportability Agency (see par. 3-1.1).

3-1.2.2.2 Height and Ground Clearance

The *maximum height* of 132 in. given in Table 3-2 is based upon limits encountered in operations outside the continental United States. Because of the requirement to minimize the vehicle silhouette height to reduce vulnerability to enemy action, even this limit is seldom reached.

The *minimum height* of a tracked vehicle is greatly affected by the ground clearance specified in the military characteristics. A generous ground clearance reduces the danger of the vehicle bellying in soft ground, or upon an obstacle, and allows space for the use of belly escape hatches and provides clearance for the high flanges on the treads of military floating bridges. Currently, the minimum ground clearance is 17 in. with values up to 20-5/8 in. being attained (see par. 4-2).

TABLE 3-1 SIZE AND WEIGHT RESTRICTIONS BY STATES ON TRUCK TRACTOR-SEMITRAILER COMBINATIONS

State	Max. Height, ft-in.	Max. Length Semitrailer, ft	Max. Length of Truck Tractor-Semitrailer Combination, ft	Max. Gross Weight, lb
Alabama	12′-6″	35	50	64,650
Alaska	13′-0″	40	60	75,200
Arizona	13′-6″	40	65	68,000
Arkansas	13′-6″	35	50	56,000 P
California	13′-6″	40	60	68,000
Colorado	13′-6″	35	60	67,200
Connecticut	12′-6″	50	50	60,000
Delaware	12′-6″	40	50	60,000
Dist. of Columbia	12′-6″	40	50	63,890
Florida	12′-6″	40	50	66,450
Georgia	13′-6″	39.55	50	63,280
Hawaii	13′-0″	40	55	67,200
Idaho	14′-0″	35	60	68,000
Illinois	13′-6″	42	50	72,000
Indiana	13′-6″	36	50	72,000
Iowa	13′-6″	35	50	72,634
Kansas	13′-6″	35	50	63,890
Kentucky	13′-6″	40	50	59,640
Louisiana	12′-6″	35	50	64,000 P
Maine	12′-6″	50	50	60,000
Maryland	12′-6″	55	55	65,000
Massachusetts	NR	35	50	60,000
Michigan	13′-6″	40	55	68,000
Minnesota	13′-6″	40	50	72,500
Mississippi	12′-6″	35	50	55,650
Missouri	13′-6″	35	50	63,890
Montana	13′-6″	35	60	68,000
Nebraska	13′-6″	40	60	67,333
Nevada	NR	NR	NR	68,000
New Hampshire	13′-6″	35	50	66,400
New Jersey	13′-6″	35	50	60,000
New Mexico	13′-6″	40	65	75,600
New York	13′-0″	35	50	65,000
North Carolina	12′-6″	35	50	62,000
North Dakota	13′-6″	35	60	62,000
Ohio	13′-6″	40	50	72,000
Oklahoma	13′-6″	35	50	73,280
Oregon	13′-6″	40	60	68,000
Pennsylvania	12′-6″	35	50	60,000
Rhode Island	12′-6″	40	50	60,000
South Carolina	12′-6″	40	50	63,890
South Dakota	13′-0″	35	50	72,110
Tennessee	12′-6″	35	50	61,580
Texas	13′-6″	35	50	72,000
Utah	14′-0″	45	60	76,500
Vermont	12′-6″	40	50	60,000
Virginia	12′-6″	35	50	56,800
Washington	13′-6″	40	60	68,000
West Virginia	12′-6″	35	50	63,890
Wisconsin	13′-6″	40	50	68,000
Wyoming	13′-6″	40	60	72,110

LEGEND:

P Plus weight on front axle.

NR No restriction.

Reference: Through courtesy of Military Equipment Division Fruehauf Trailer Company.

TABLE 3-2 LIMITING DIMENSIONS AND WEIGHTS OF TRACKED VEHICLES FOR MOVEMENT ON HIGHWAYS AND BRIDGES (Ref. 7)

Maximum Gross Weight, lb	Width		Maximum Height, in.	Minimum Ground Contact	
	Maximum, in.	Minimum, in.		Length,* in.	Total Width, in.
8,000	96	None	132	32	20
16,000	96	78	132	55	24
24,000	96	80	132	73	27
32,000	96	84	132	87	30
40,000	120	96	132	98	33
48,000	120	100	132	107	36
60,000	120	100	132	132	37
80,000	120	112	132	144	45

* The maximum ground contact length for any vehicle is 180 in.

3-1.2.2.3 Gross and Distributed Weights

The maximum permissible gross weight of a tank is 160,000 lb (80 tons). This limitation is based upon the capacity of U.S. highway bridges of the heaviest classification (Ref. 8).

Minimum ground contact dimensions are given in Table 3-2. These are of importance as they affect the distribution of the gross weight. Two considerations affect the distribution of the gross weight of a tracked vehicle: one, the average ground pressure, determined as the quotient of the gross weight and the total ground contact area of the tracks; the other is the load distributed per linear foot of the length of ground contact. Ground pressures are regulated primarily to control floatation in cross country operations and to develop traction; while loading per foot of track length is regulated to control the effect of the vehicle's weight upon roads and bridges.

Ground pressures of heavy tracked vehicles are limited to 12.5 psi. Although pressures of 6 to 8 psi are considered more desirable, they are difficult to obtain in the design of larger vehicles. Certain light, amphibious and special-purpose tracked vehicles that require maximum floatation obtain ground pressures as low as 3 psi.

The distributed load per linear foot of ground contact is obtained by dividing the gross vehicle weight in pounds, GVW, by the length of the ground contact in feet, L_t, as seen in the side elevation. The maximum permissible distributed load is determined from the following (Ref. 8):

For values of GVW less than 60,000 lb:

$$\frac{GVW}{L_t} \leqq 3000 + 0.06\,(GVW - 8{,}000) \tag{3-1}$$

For values of GVW greater than 60,000 lb:

$$\frac{GVW}{L_t} \leqq \frac{20{,}000 \times GVW}{160{,}000 + GVW} \tag{3-2}$$

3-1.2.2.4 Vehicle Length

The length of the track at the region of ground contact is governed by the equations of weight distribution, the ground contact pressure, and by certain requirements for efficient steering. Requirements imposed by specified angles of approach and departure have an influence upon overall vehicle length. Equations 3-1 and 3-2 for maximum permissible distributed weight can be solved for L_t to determine the minimum permissible track length in ground contact. Thus, when GVW is less than 60,000 lb

$$L_t \text{ (inches)} \geqq \frac{12\,GVW}{3000 + 0.06\,(GVW - 8000)} \tag{3-3}$$

when GVW is greater than 60,000 lb

$$L_t \text{ (inches)} \geqq 96 + \frac{12\,GVW}{20{,}000} \tag{3-4}$$

The steering characteristics of a tracked vehicle are affected by the ratio of the track length in contact with the ground, L_t, to the tread, L_t/T, where T is the width between the track centers

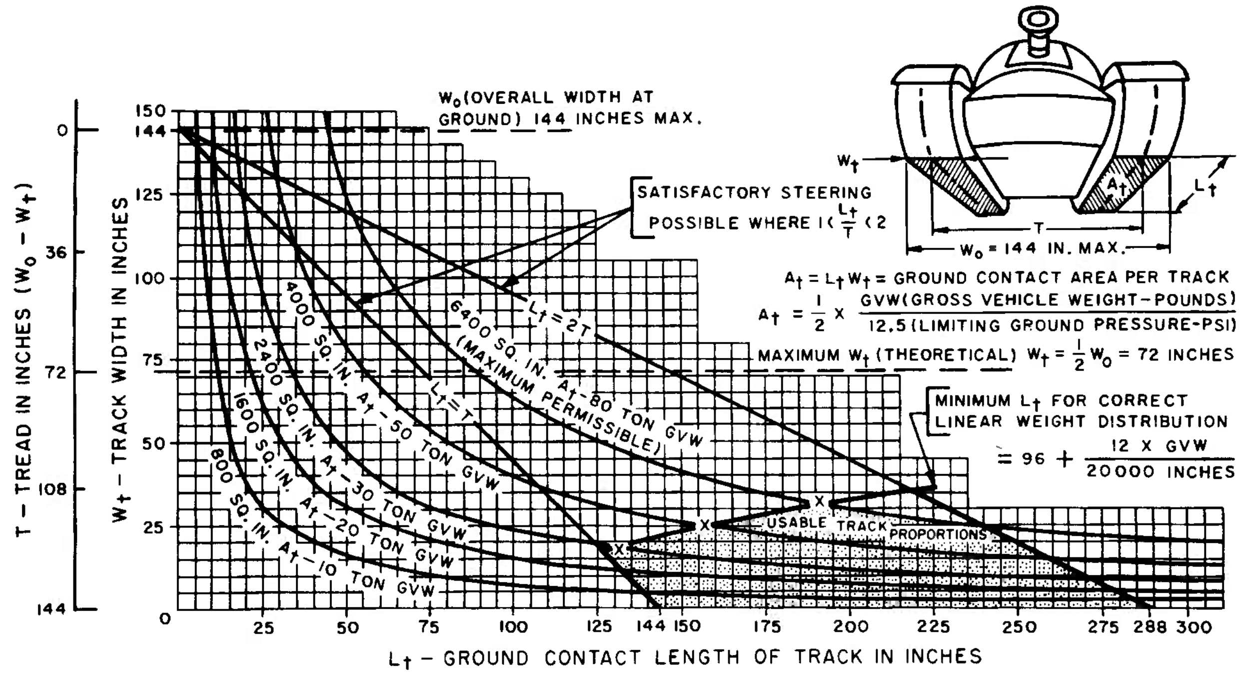


Figure 3-1. *Correlation of Physical Limits of a Tracked Vehicle (Ref. 8)*

(see Chapter 12). When this ratio becomes less than unity, that is, L_t is less than T, steering becomes relatively unstable. When this ratio approaches a value of 2, that is, L_t approaches a value of $2T$, steering imposes excessive power demands. Therefore, for satisfactory steering, the L_t/T ratio should be between 1.0 and 1.7. In actual practice, the values usually used are between 1.125 and 1.69 (Ref. 8).

3-1.2.2.5 Correlation of Physical Limits

When dimensional limits are applied simultaneously, a relatively narrow field of choice is available to the designer. This is illustrated in Fig. 3-1 which shows the track proportions that may be used lying in a relatively small zone. In the figure, the ground contact length of the track, L_t, is plotted against the track width, W_t, and against the tread, T. A theoretical maximum overall track width of 144 in. is used as a limit in order to establish $T = 0$ on the graph. Hyperbolas representing constant track areas are plotted on the graph using the equation $A_t = L_t W_t$ (where A_t is the ground contact area) for selected ground contact areas of 800, 1600, 2400, 4000, and 6400 sq. in. Based upon a limiting ground pressure of 12.5 psi, each hyperbola also represents a definite gross vehicle weight. The highest gross vehicle weight shown is 80 tons, the maximum permitted by regulations. Points of minimum track length are calculated, using Eqs. 3-3 and 3-4, and located on the hyperbolas. These points are connected to form a limiting minimum track length curve. Finally, the limiting steering curves ($L_t = T$ and $L_t = 2T$) are determined and drawn.

The usable track proportions are confined to the hatched area in the graph bounded by the limiting dimension curves. The chief significance of the usable proportions thus obtained lies in their effect on hull width. Decreasing the track width of a given vehicle gains only a small amount of space within the vehicle; because, the track width changes relatively little regardless of length, within the usable range of proportions.

One of the most critical dimensions on a combat tank is the turret ring diameter. This must be large enough to allow space for such major items as gun recoil, ammunition handling, personnel, and fire control equipment, yet its size is restricted by the overall vehicle width limits, the need for adequate track width, and the need for armor protection. Here, again, the designer is faced with the

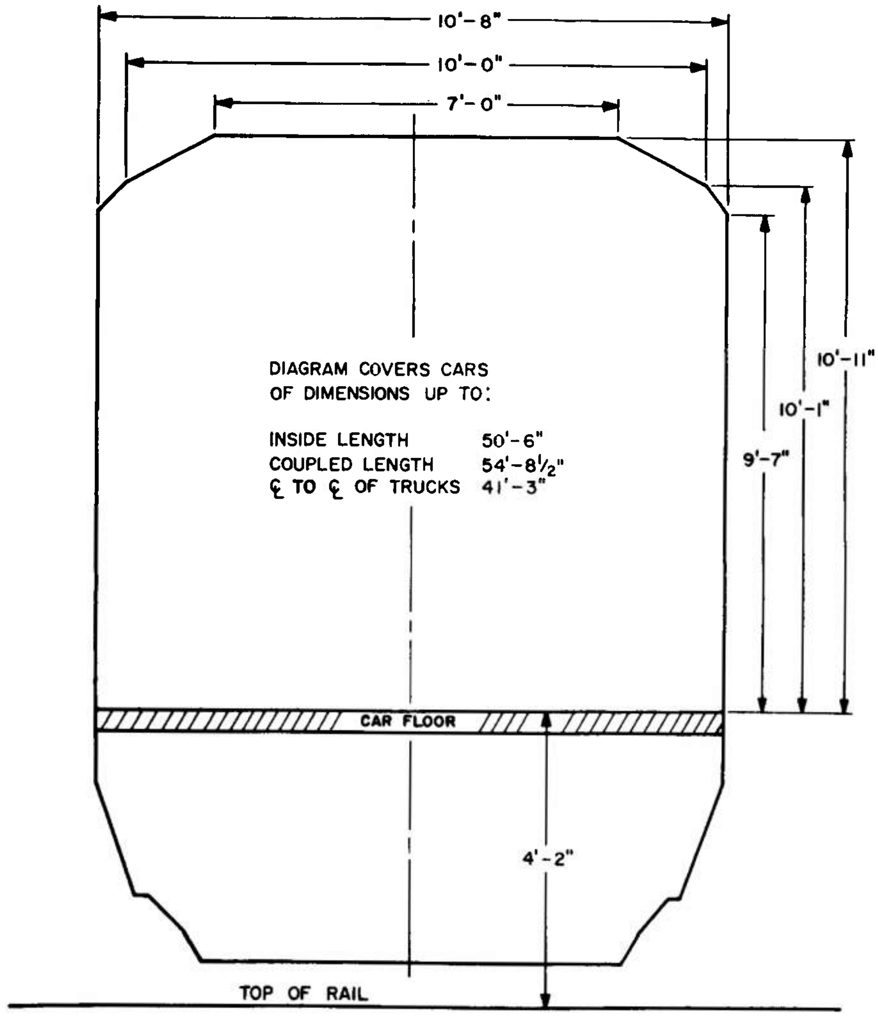


Figure 3-2. ***Outline Diagram of Approved Limited Clearances of the Association of American Railroads (Ref. 7)***

necessity to compromise, but his freedom to compromise is somewhat limited.

Turret rings currently being used on production vehicles are 85 in. inside diameter. Figure 3-1 shows that for a 50-ton vehicle, the designer can select a track width of from 15 to 25 in. For most efficient steering he would probably restrict his choice to track width between 22 and 25 in., letting considerations of ground contact length determine the final choice. Assuming that the 22 in. track width satisfies all requirements, the inside diameter of the turret ring plus the width of two tracks accounts for 129 in. of vehicle width. Additional width is needed for armor, track shrouding, width of turret bearing, and clearances. Thus, requirements to reduce the overall vehicle width present a serious problem to the designer.

3-1.3 RAIL TRANSPORTABILITY

In order to meet the requirements for transportability by railroad, railway lading clearances must be considered. Railway lading clearances are determined by bridges, tunnels, platforms, telephone and electric poles, and miscellaneous wayside structures. Data on all such clearances have

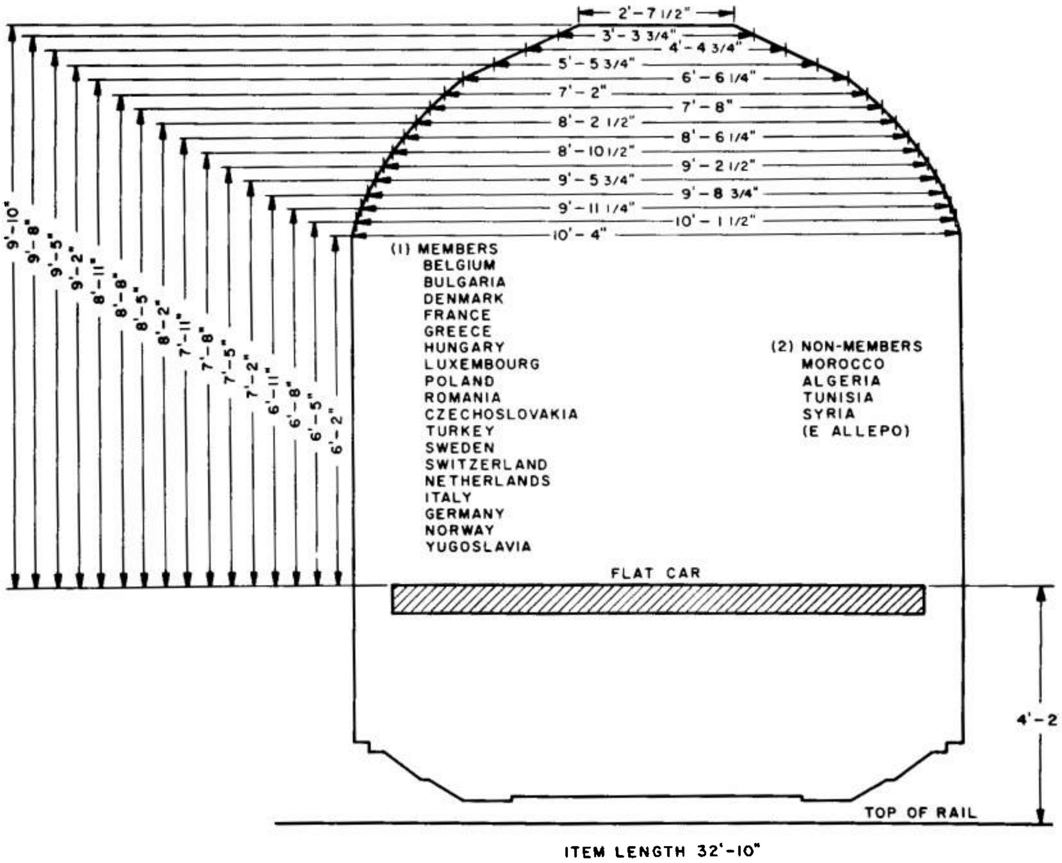
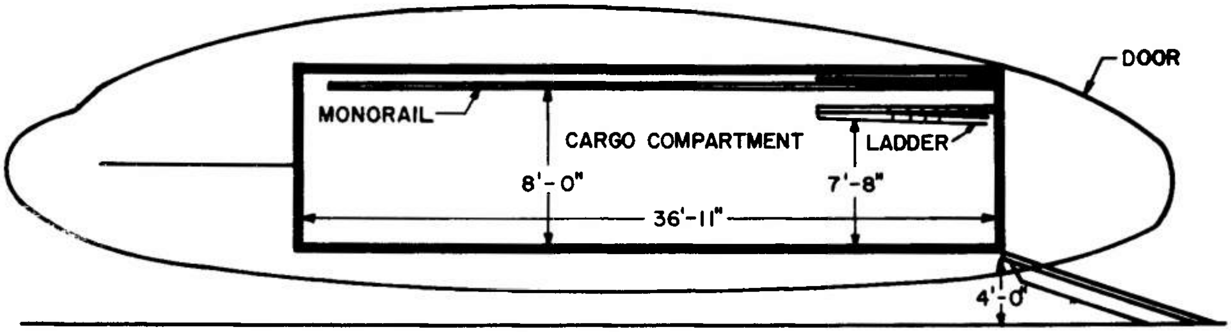


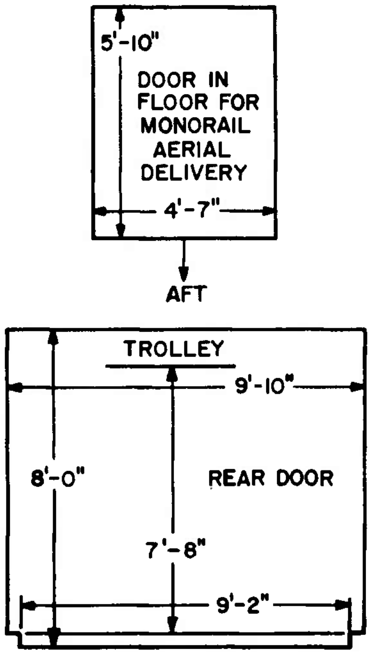
Figure 3-3. *Berne International Clearance Diagram (Ref. 7)*

been compiled and are presented in the form of dimensioned outline diagrams.

Since bridges settle, tunnel walls and roofs are subject to slippage, telephone and electric poles can lean, rock cuts are subject to slides, and the railroad track is subject to variations caused by springing and floatation actions, a safe distance factor or clearance is applied to all outlines published by the railroads. These clearance dimensions are not normally published nor are they standard between the railroad companies, but each railroad establishes its own safe distances. Railroad equipment and loads on railroad cars whose outlines exceed the limiting outline published by the railroads must be cleared by the superintendent of the railway lines prior to their acceptance. Figure 3-2 shows the outline diagram of the approved limited clearances published by the Association of American Railroads and supplies to all standard gauge unrestricted main lines in the continental United States. Figure 3-3 shows a comparable outline diagram that prescribes the minimum railway clearances required on railroads in overseas areas. It is the result of an international meeting held at Berne, Switzerland attended by the major countries of Europe. As a result, the diagram is often referred to as the Berne International Tunnel Diagram.

These diagrams indicate the maximum allowable cross section of a vehicle as loaded upon a railway car for shipping. For a vehicle of larger section, the maximum allowable cross section is the size to which the section must be reducible. An important factor in reducibility is the facility with which the vehicle can be reassembled in terms of time, tools, and skills required. It is current practice to use overhanging tracks on many types of tracked vehicles. By removing tracks, sprockets and tool boxes, vehicle widths can be reduced approximately 20 in., to the width across the outer faces of the road wheels. Further reducibility requires the disassembly of suspension elements and is not considered feasible.





DIMENSIONS:
MAIN COMPARTMENT:

HEIGHT (MAXIMUM)	- 8'-0"
HEIGHT (CLEAR UNDER TROLLEY)	- 7'-8"
LENGTH (OVER-ALL)	-36'-11"
WIDTH (MAXIMUM)	- 9'-10"
WIDTH (MINIMUM)	- 9'-2"

CAPACITY:

MAIN COMPARTMENT (VOLUME)	-3,150 CU. FT.
WITHOUT CARGO DOORS	-2,850 CU. FT.
MAIN COMPARTMENT (AREA)	- 353 SQ. FT.

REFERENCE:
(1) TECHNICAL ORDER IC-119B-9
(FORMERLY AN 01-115CC-9)
NAVAER 01-115CC-9
(2) STANDARD AIRCRAFT
CHARACTERISTICS BOOK

TYPICAL LOGISTICAL MISSION:
1000 NAUTICAL MILES (ONE WAY)
NORMAL WEIGHT 13,630 LBS

Figure 3-4. *C-119G Cargo Compartment Profile (Ref. 7)*

3-1.4 AIR TRANSPORTABILITY

When designing military vehicles for airlift, consideration must be given to the size, weight, and location of the vehicle's center of gravity, the size, location and configuration of the aircraft loading apertures, size and configuration of cargo compartments including limiting features that may prevent the full utilization of available space, the strength of the aircraft floor and loading ramp, and to the air transportability requirements specified in MIL-A-8421A (USAF) entitled, *Air Transportability Requirements, General Specifications for*. Each vehicle should be designed to be transportable in a maximum number of aircraft types to permit maximum utilization of available aircraft and to alleviate the possible shortage of a particular type. When designing vehicles for aerial delivery (airdrop), vertical and horizontal clearances in aircraft must be considered; and, in addition, the designer must make due allowances and adjustments for the weight and space taken up by the aerial delivery equipment.

Figures 3-4 through 3-11 show profile charts and miscellaneous data pertinent to the loading of various standard aircraft. This information is suitable for general guidance in the design of equipment to be loaded into aircraft and is not intended for operational purposes. Specific loading and per-

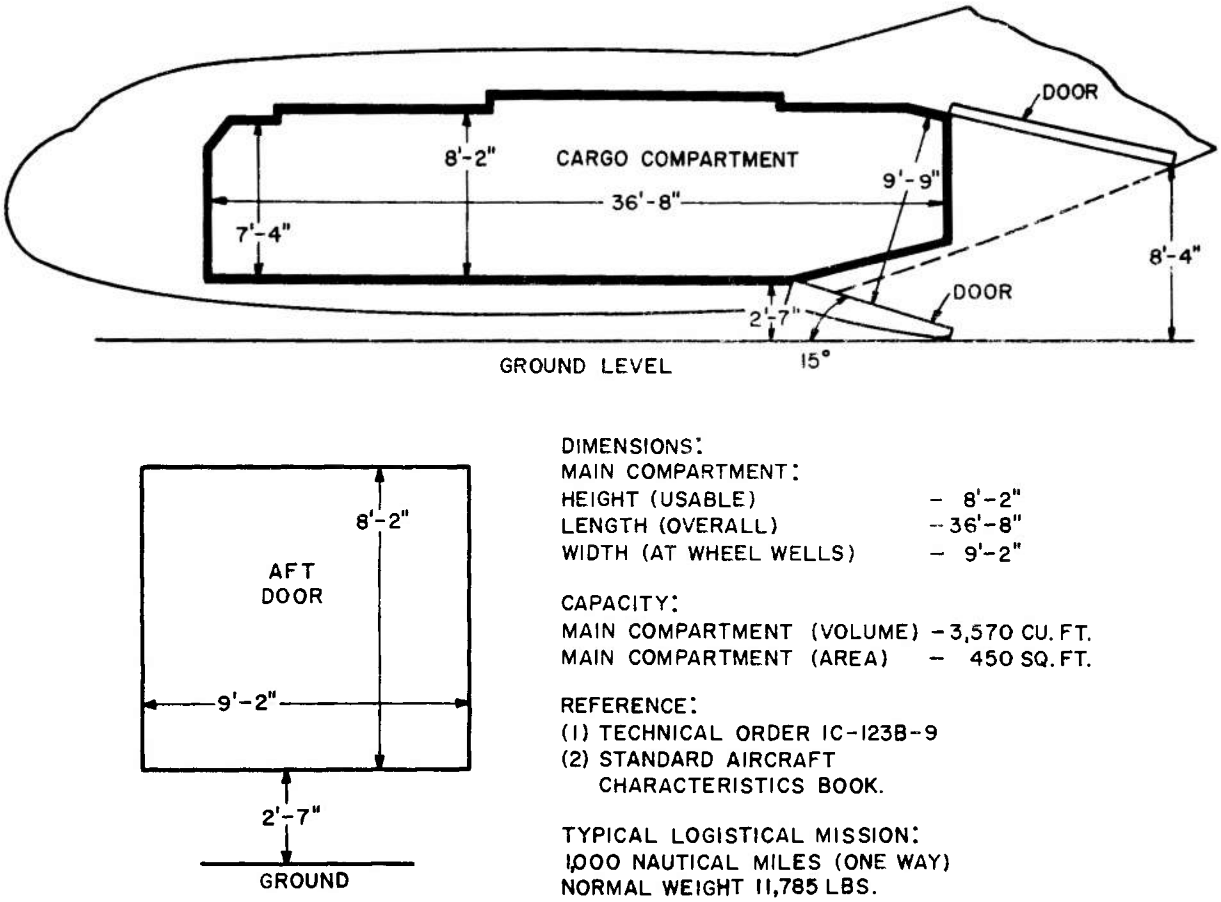


Figure 3-5. *C-123B Cargo Compartment Profile (Ref. 7)*

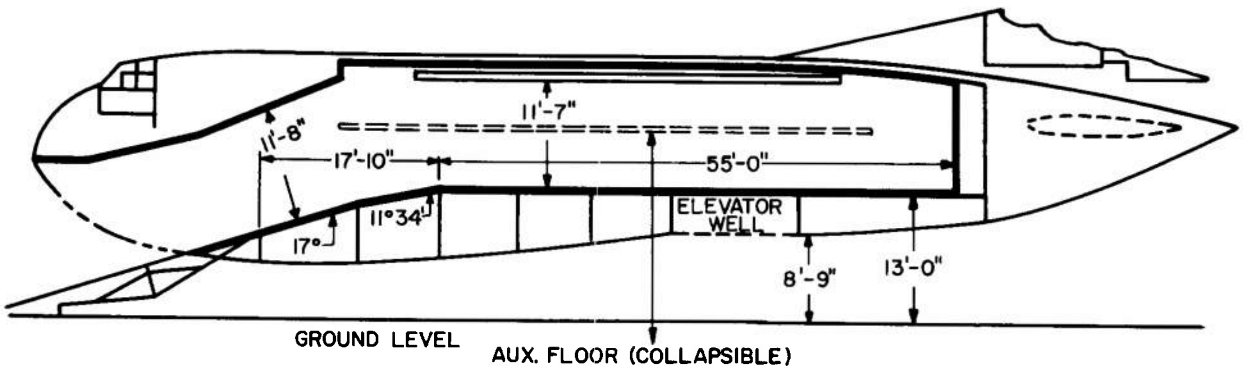
formance problems should be resolved by referring to the Technical Order applicable to the aircraft in question (e.g., Ref. 9). Special consideration must be given to the location, capacity, and type of tie-down fittings used, allowances for loading and unloading clearances, aisle space, access to loading controls and auxiliary equipment, and aircraft loading limitations. These factors limit the size and weight of the vehicles that can be transported. Additional data and guidance can be obtained from Refs. 10, 11, and 12.

3-2 OPERATIONAL LIMITS

Certain operational limits or levels of performance have been established for military vehicles as necessary to the successful accomplishment of future military missions. They are based upon experiences with vehicles of the past, in all parts of the world and under every conceivable environment, during peacetime field maneuvers as well as during wartime operations. These levels of performance are prescribed in various military directives and concern such factors as the capability of military vehicles to negotiate extreme grades and slopes, water barriers, and extreme climatic environments. Since these operational limits affect all military vehicles, they are presented here as guidance to the designer.

3-2.1 GRADES AND SIDE SLOPE PERFORMANCE (Ref. 25)

The capability of a vehicle to operate on grades and side slopes is referred to as the gradeability of the vehicle and is expressed as a percentage figure that represents the maximum grade that the vehicle can negotiate satisfactorily (see par. 5-2.2.5). Gradeability is of particular importance in military vehicles, since they are required to operate tactically without the benefit of roadways. An evaluation of the gradeability of a ve-



DIMENSIONS:

MAIN COMPARTMENT:
HEIGHT (USABLE) - 11'-7"
LENGTH (OVERALL) - 77'-0"
WIDTH (FLOOR LEVEL)-11'-4"

ELEVATOR WELL:
LENGTH - 13'-4"
WIDTH - 7'-8"
GROUND TO FUSELAGE - 13'-0"
ELEVATOR CAPACITY:
8,000 LBS PER HOIST (2)
16,000 LBS USING TWO HOISTS

RAMP: (SPLIT TYPE)
INCLINE - 17°
LOAD - 10,000 LBS PER SINGLE WHEEL EITHER RAMP
- 22,000 LBS PER SINGLE AXLE, BOTH RAMPS.

CAPACITY:
MAIN COMPARTMENT (VOLUME) - 10,000 CU. FT.
MAIN COMPARTMENT (AREA) - 1,652 SQ. FT.

REFERENCE:
(1) TECHNICAL ORDER 1C-124A-9 (FORMERLY 01-40NV-9) 1C-124C-9
(2) STANDARD AIRCRAFT CHARACTERISTICS BOOK.

TYPICAL LOGISTICAL MISSION:
1,000 NAUTICAL MILES (ONE WAY)

NORMAL WEIGHT -	C-124A	C-124C
	40,500	47,600

Figure 3-6. *C-124A or C Cargo Compartment Profile (Ref. 7)*

hicle provides a means of determining the adequacy of the power plant and power train as well as an assessment of the vehicle's tractive ability.

Army test specifications require that combat and tactical vehicles (see Chapter 4, Secs. I and II) be capable of negotiating a 60-percent grade of smooth, dry concrete in both forward and reverse gears, and that they be able to brake adequately on this same grade. In addition to this maximum gradeability test, vehicle performance is evaluated at the 60-percent grade and at lesser, more common, grades to determine the speeds at which the vehicle can climb.

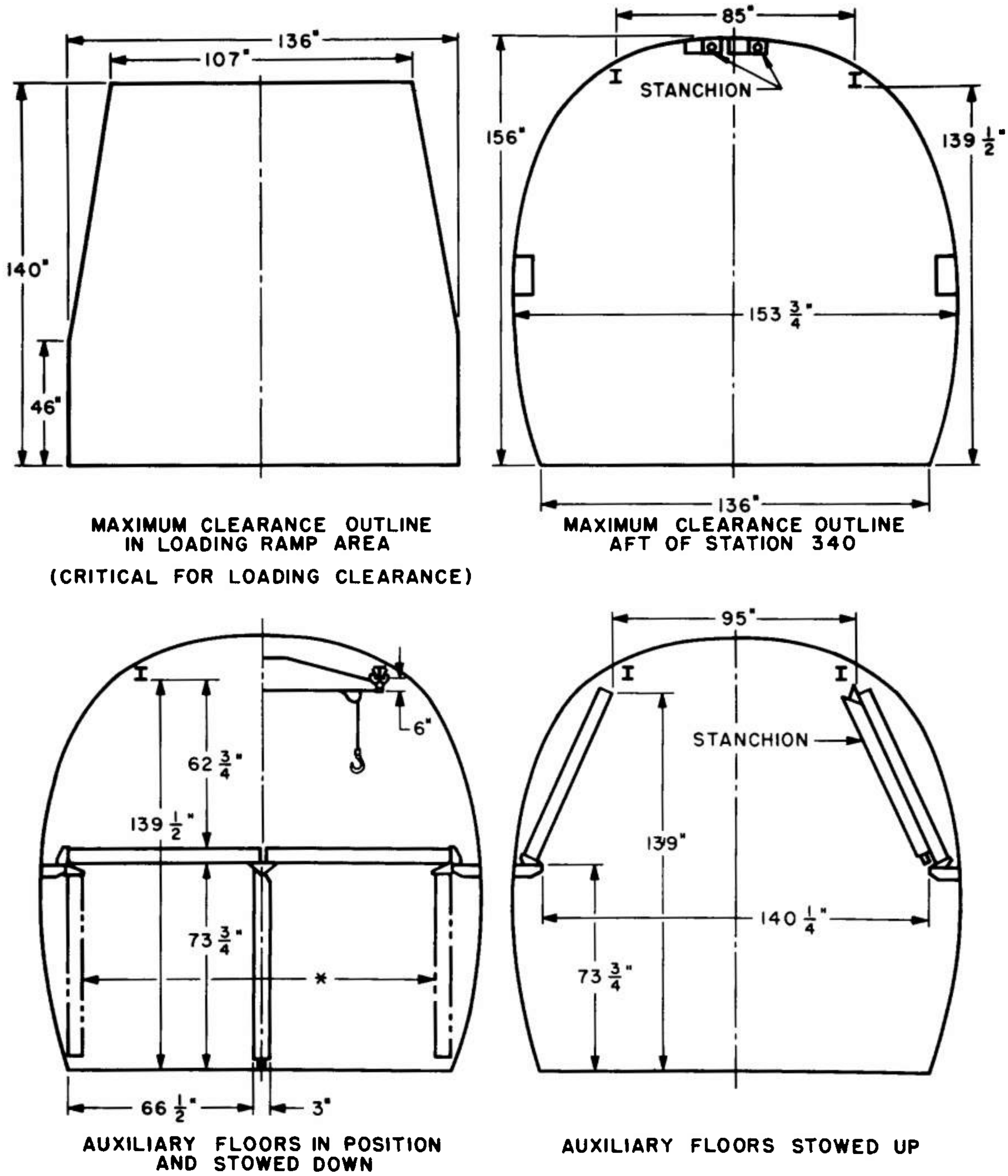
All vehicles must meet the gradeability and side slope requirements when loaded with their rated or combat loads. Wheeled vehicles with towed trailers are generally required to operate on maximum grades of 30 percent.

The artificial slopes used in testing military vehicles include 5-, 10-, and 15-percent grades each with black-top paving, a 20-percent gravel grade, and 30-, 40-, 50-, and 60-percent concrete grades. The sustained speed is determined by bringing the vehicle to a maximum slope speed from a standing start at the foot of the grade.

The braking system must be capable of stopping and holding the vehicle in both the forward and reverse directions on the maximum slope the vehicle was designed to ascend. When a towed load is prescribed for the vehicle, the brake tests are performed both with and without the towed load.

Engine performance, fuel consumption, fuel pressure, lubricating system performance, and cooling system performance are all evaluated during slope operations.

Requirements for side slope operation of military vehicles specify 20- to 40-percent gradients. In general, tactical vehicles are required to perform



*125" STA 340-540 (AF48-795 — AF49-250)
131" STA 540-1020 (AF48-795 — AF49-250)
131" ALL STATIONS (AF49-251 AND SUBS)

Figure 3-7. C-124 Fuselage Clearance Diagram (Ref.7)

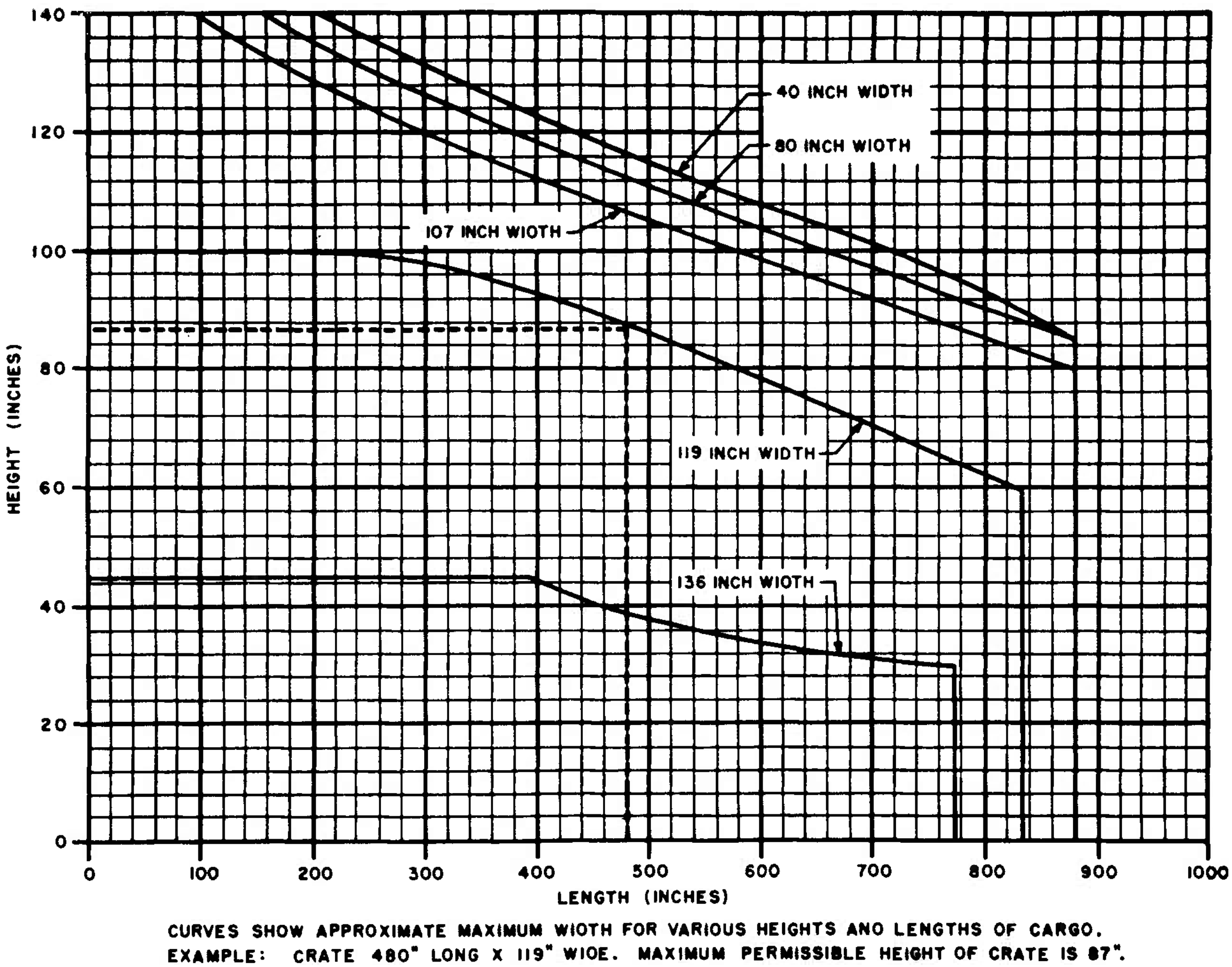


Figure 3-8. *C-124 Cargo Size Limits Chart (Nose Door Loadings) (Ref. 7)*

satisfactorily on side slopes of 20 percent; combat vehicles, 30 percent; and jeep-type vehicles, 40 percent. In addition to remaining under full control at the specified angles and at reasonable speeds, the main and auxiliary engines are operated at idling speed under all convenient electrical load for 5 min periods, first with the vehicle headed in one direction and then in the other. In addition to evaluating engine, suspension, and steering system performance under these side slope conditions, liquid containers are checked for overflow and bearings are checked for adequate lubrication.

3-2.2 WATER BARRIERS

3-2.2.1 Definitions

AR 705-2300-8 (Ref. 13) prescribes the water crossing requirements for future combat and tactical vehicles. It defines four general methods of crossing water barriers as follows:

a. *Shallow-fording.* The ability of a vehicle with its suspension in contact with the ground to negotiate a water obstacle of a specified depth without the use of special waterproofing kits.

b. *Deep-fording.* The ability of a vehicle with its suspension in contact with the ground to negotiate a water obstacle to a specified depth by the application of a special waterproofing kit.

c. *Floating.* The ability of a vehicle to negotiate water obstacles without being in contact with the bottom. Self-propulsion while in the water is not implied in this definition.

d. *Swimming.* The ability of a vehicle to negotiate a water obstacle by propelling itself across, without being in contact with the bottom.

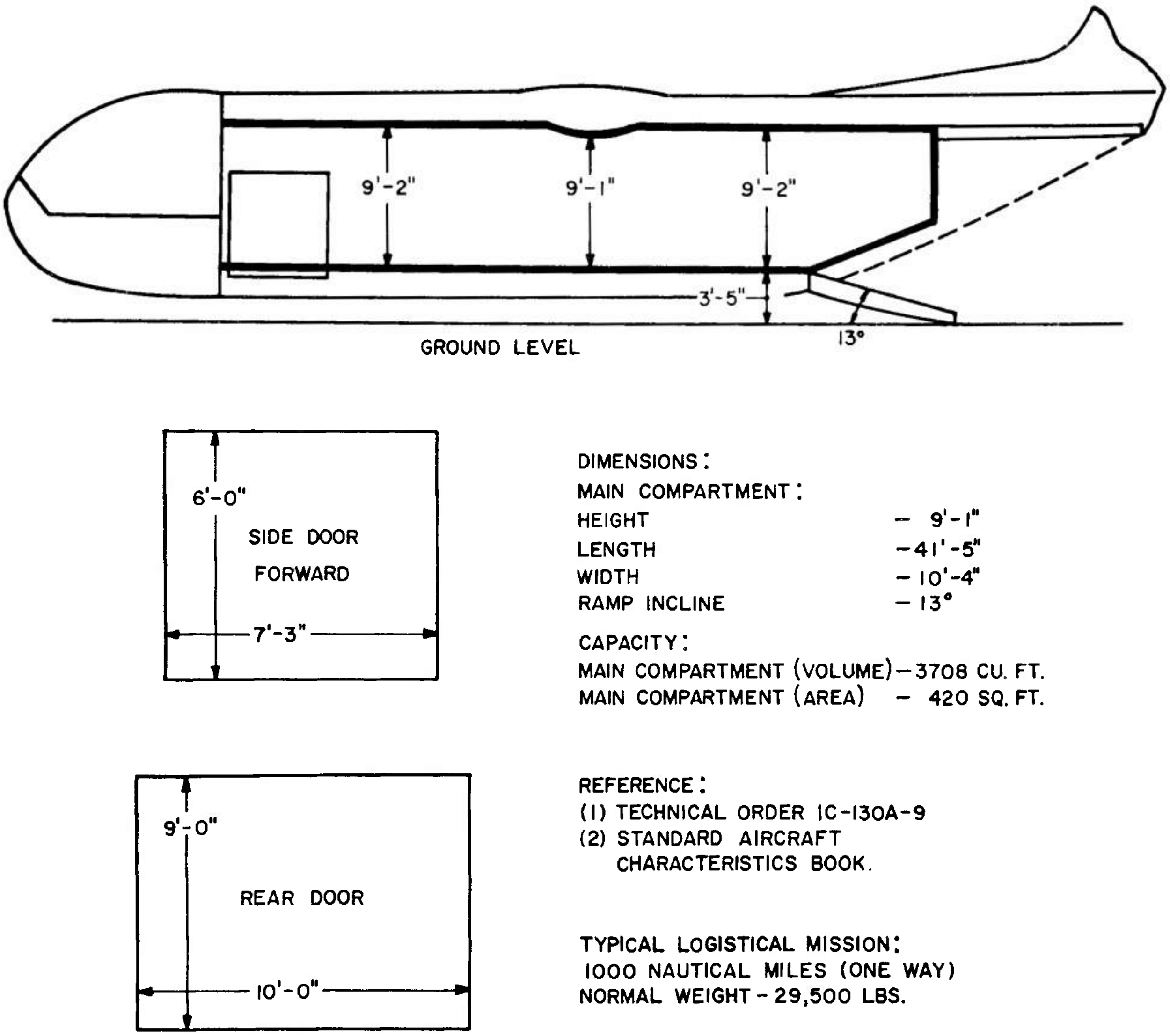


Figure 3-9. *C-130A Cargo Compartment Profile (Ref. 7)*

3-2.2.2 Capability Requirements

As a minimum requirement, all combat and tactical vehicles must be built with an inherent capacity of shallow-fording in fresh and salt waters to the maximum depth practicable but not less than 30 in., including sinkage depth and wave height, except for the following:

a. Tactical vehicles up to, and including, 1½-ton payload capacity must have capability to shallow-ford depths of not less than 20 in.

b. Tanks and other armored vehicles must have capability of shallow-fording depths of not less than 42 in.

The preferred water-crossing capability for combat and tactical vehicles is floating or swimming. Deep-fording kits are mandatory equipment for all combat and tactical vehicles that do not have one of these capabilities. These fording kits must provide the capability of deep-fording for at least 15 min. in fresh or salt water to the following depths, including both sinkage depth and wave height:

a. Fully enclosed armored vehicles, to the maximum depth practicable consistent with adequate freeboard. Freeboard is measured from

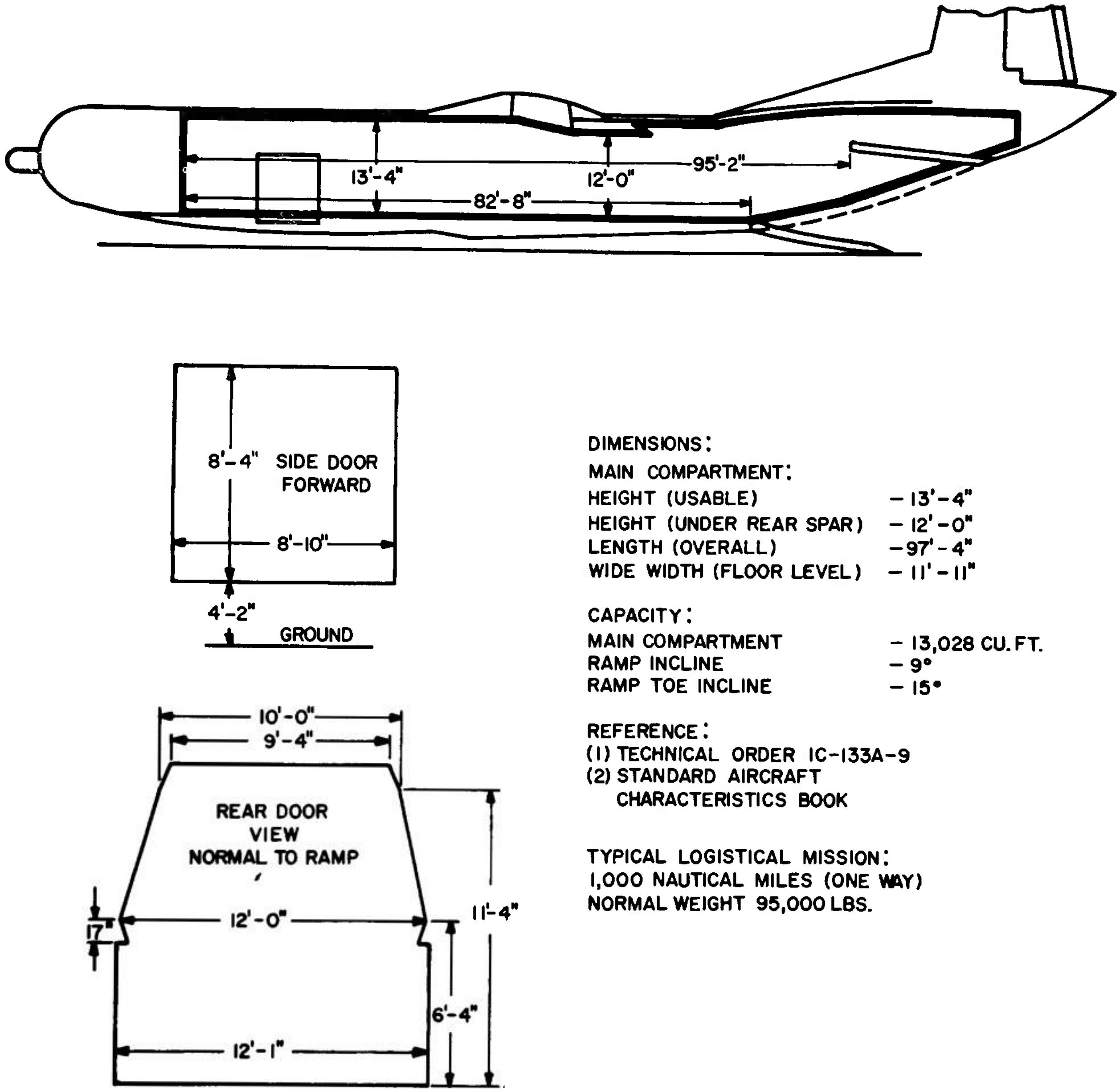


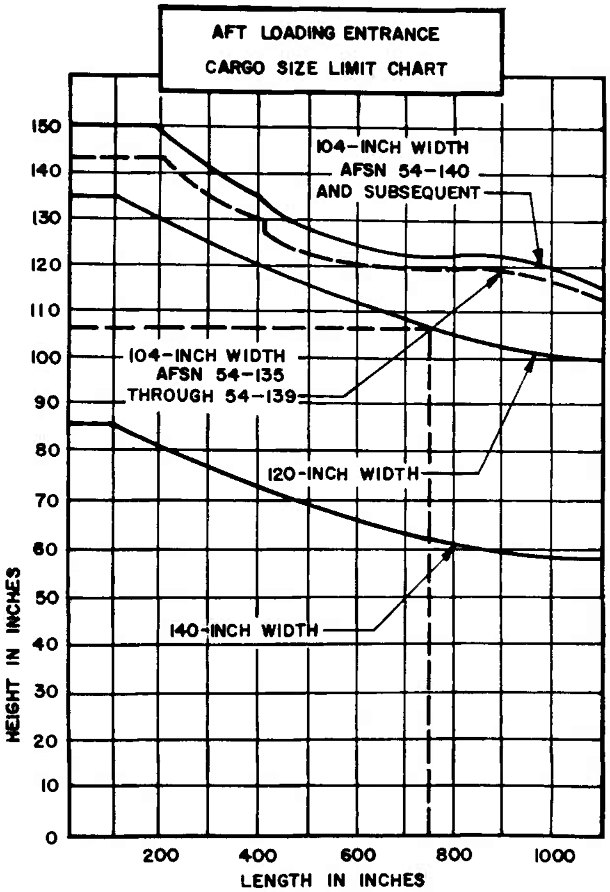
Figure 3-10. *C-133A Cargo Compartment Profile (Ref. 7)*

the top of the commander's hatch opening or turret.

b. All other vehicles, including trailers, 5 ft.

3-2.3 CLIMATIC ENVIRONMENT

The climatic environment anticipated for military vehicles operating in all parts of the world is set forth in considerable detail in AR 705-15 (Ref. 14). In general, this regulation divides operating conditions into three categories, namely, basic, extreme cold weather, and extreme hot weather. The *basic* conditions are those which may be expected to exist at certain times and places in the most densely populated portions of the world and where major military activities of the past have taken place. *Extreme cold weather* conditions are those encountered in the regions generally referred to as Arctic and subarctic; while the *extreme hot weather* conditions are those encountered in the hot, arid desert regions of the world where temperatures are higher than in the basic range and



AFT LOADING ENTRANCE
SAMPLE PROBLEM

GIVEN: A CRATE 107 INCHES HIGH AND 120 INCHES WIDE TO BE LOADED THROUGH THE AFT LOADING ENTRANCE.

PROBLEM: DETERMINE MAXIMUM ALLOWABLE LENGTH OF CRATE.

SOLUTION: ENTER LEFT SIDE OF CHART AT 107 INCHES ON HEIGHT SCALE. FOLLOW HORIZONTAL 107 INCHES LINE TO INTERSECTION WITH 120 INCH WIDTH CURVE. FOLLOW VERTICAL LINE FROM THIS INTERSECTION DOWN TO 750 INCHES ON LENGTH SCALE.

Figure 3-11. *C-133A Aft Loading Entrance Cargo Size-Limit Chart (Ref. 7)*

the region is generally characterized by its extremely low humidity.

3-2.3.1 Basic Operating Conditions

With the exception of certain sheltered and limited service items, all military combat and combat support materiel is required to be capable of satisfactory performance at all times under the basic operating conditions. These include:

a. Thermal stress imposed by ambient air temperatures ranging from −25°F, without benefit of solar radiation, to 115°F coupled with the impact of solar radiation at a rate of 360 Btu per sq ft per hr. At the lower end of the temperature range, due consideration must be given to the net heat loss due to longwave radiation because it may chill the equipment below the ambient air temperature. At the upper temperature range, maximum temperature and radiation conditions need to be considered for only four hours per day. At elevations above 3,000 ft, maximum design temperatures may be lowered to the values given in Table 3-3.

b. Absolute humidity as low as 0.1 grains per cu ft (corresponding to a dew point of −25°F), and as high as 13 grains per cu ft (corresponding to a dew point of 85°F). These high absolute humidities are likely only along the immediate coasts of the Red Sea, the Persian Gulf, and the Indian Ocean. For equipment that is likely to be used only in other areas, an absolute humidity of 11 grains per cu ft (corresponding to a dew point of 80°F) is applicable.

c. Relative humidity as low as 5 percent at temperatures of 115°F and as high as 100 percent at all temperatures from −25° to 85°F.

d. Rainfall characterized by two types of rainfall conditions which represent the extremes that

TABLE 3-3 MAXIMUM DESIGN TEMPERATURES FOR USE AT ELEVATIONS ABOVE 3,000 FEET

Elevation, ft	Maximum Design Temperature, °F
3,000	115
3,500	111
4,000	108
4,500	104
5,000	100
5,500	98
6,000	97
6,500	95
7,000	93
7,500	92
8,000	90

may be encountered, namely, continuous, moderately heavy, wind-driven rain; and brief, torrential downpours. Military equipment must be capable of satisfactory performance under each condition. Quantitatively, the first type of rainfall condition is characterized by 12 in. of rain falling at variable rates over a 12-hr period accompanied by winds as high as 40 mph. The second rainfall condition results when 7 in. of rain falls in one hour with as much as 2 in. falling in one 5-min period and a wind velocity of from 0 to 5 mph.

e. *Snowloads* on semipermanently installed equipment, on which snow can accumulate and is not usually removed between snowfalls, is 40 lb per sq ft. On temporary equipment, which is moved often and cleared of snow between storms, the snow load can be 20 lb per sq ft. On portable equipment, which may be moved daily, the extreme snow load is 10 lb per sq ft.

f. *Winds* of hurricane force occur in the mountains and in coastal areas, and military equipment designed for operation in these regions must be capable of withstanding maximum winds of 80 mph with gusts up to 120 mph. Most inland areas, however, do not experience winds in excess of 65 mph with gusts to 100 mph. The extreme winds associated with tornadoes are not considered in the selection of wind design criteria because it is not considered feasible to design for such extremes.

g. *Blowing snow, sand, and dust,* with their penetrative and abrasive effects, are capable of rendering military equipment inoperative. Quantitative design limits for equipment likely to be exposed to such blowing particles are specified in AR 705-15 as follows:

Blowing snow crystals 1 to 3 millimeters in diameter blowing at wind speeds as high as 40 mph at temperatures below 32°F.

Blowing sand particles of 0.18 to 0.30 millimeters in diameter blowing at 40 mph. Vertical distribution of the sand is such that few grains are more than 5 ft above the ground and approximately half the grains are less than 1 in. above the ground.

Blowing dust particles 0.0001 to 0.01 millimeters in diameter blowing at 15 mph. Dust may be distributed to high elevations and remain in the atmosphere for long periods of time.

3-2.3.2 Extreme Cold Weather Conditions

Military equipment destined for operations in the Arctic or subarctic regions falls into the extreme cold weather category. This equipment may be specially designed to meet this climatic extreme, or it can consist of equipment designed for the basic conditions and subsequently modified, or augmented with the installation of specially developed kits, to meet the required conditions.

The principal characteristic of this weather condition is that the low limit of the thermal stress is extended from the basic −25° to −65°F without benefit of solar radiation. The low limit of absolute humidity is extended to 0.1 grains per cu ft corresponding to a dew point of −65°F. Other limits are essentially the same as given for the basic conditions with the exception of the specifications with respect to rainfall. These are not applicable in the extended temperature range of −25° to −65°F.

3-2.3.3 Extreme Hot Weather Conditions

The specifications for this weather extreme are again essentially the same as the basic conditions with one major extension to the thermal stress. Here, the basic ambient air temperature range is extended from 115° to 125°F with solar radiation at the rate of 360 Btu per sq ft per hr. For other minor differences consult AR 705-15.

TABLE 3-4 LUBRICATING OILS, HYDRAULIC FLUIDS, AND GREASES USED IN MILITARY AUTOMOTIVE EQUIPMENT

	MILITARY SPECIFICATIONS
MIL-L-15019B	Lubricating Oil, Compounded
MIL-L-2104B	Lubricating Oil, Internal Combustion Engine, Heavy Duty
MIL-L-45199	Lubricating Oil, Internal Combustion Engine, High Output Diesel
MIL-L-10324A	Lubricating Oil, Gear, Subzero
MIL-O-10295	Oil, Engine, Subzero
MIL-O-6083A	Oil, Preservative, Hydraulic Equipment
MIL-H-5606A	Hydraulic Fluid, Petroleum Base, Aircraft and Ordnance
MIL-H-13919A	Hydraulic Fluid, Petroleum Base, Fire Control
MIL-H-13910	Hydraulic Fluid, Nonpetroleum Base, Automotive Brake, Arctic
MIL-G-3278	Grease, Aircraft and Instrument, Sealed Bearings
MIL-G-10924A	Grease, Automotive and Artillery
VV-G-632	Grease, Lubricating, Automotive and Industrial

3-2.3.4 Storage and Transit Conditions

All military materiel must be capable of safe storage and transportation without permanent impairment of its capabilities from the effects of extreme climatic conditions. Only temperature and humidity extremes are given in AR 705-15 for materiel in storage. Values for the other factors are the same as those given for basic operating conditions.

High temperature storage. The values given as design criteria for high temperature storage are air temperatures as high as 155°F for periods up to 4 hrs daily; no solar radiation; absolute humidity as high as 13 grains per cu ft. The materiel temperature under these conditions depends upon the thermal capacity and mass of the stored items.

Low temperature storage. Design criteria given for low temperature storage are: air ambient temperatures as low as -65°F for periods up to 3 days; no solar radiation. The net loss of heat through longwave radiation must also be considered.

3-2.4 ELECTRICAL SYSTEM

Specifications for the integral electrical systems of combat and tactical vehicles are contained in SR 705-325-1 (Ref.15). This regulation directs that all electrical systems of standard military vehicles, including electronic equipment intended for operation in military vehicles, be based upon a standardized nominal voltage of 24 v. DC. This includes the electrical systems on all towed loads such as trailers, gun carriages, and special-purpose towed equipment. Electronic equipment whose power requirements are such as to make necessary the use of auxiliary power plants may be exempt from this regulation. It is required, however, that the fuels and lubricants required by the auxiliary generating equipment be the same kind as is used by the main power plant and power train of the vehicle in which it is installed. Additional discussion of the electrical system is given in Chapter 13.

3-2.5 FUELS AND LUBRICANTS

The types of fuels used by military vehicles are dictated by the types that are readily available as well as by the requirements of the engine. In general, the fuels used are fundamentally the same as those used by commercial vehicles of comparable types. Since cruising ranges of vehicles are limited by the space available for carrying fuel, greater ranges, or smaller and lighter vehicles, result from the use of fuels that possess a higher heat content per gallon. In many operations, however, military vehicles are limited to the fuels that are readily available. This limitation is especially acute during a full-scale national emergency when both the armed forces and the civilian economy demand the same fuels. In order to alleviate this situation, engines are being developed that are less sensitive to the type of fuel with which they are supplied and also engines that are capable of utilizing, efficiently, several types of fuels. These are the multifuel engines that are gaining in popularity.

Fuels for military vehicles are classified into two general groups: the gasolines, and the Diesel fuels. The general category "gasoline" is defined as the fuel used in spark ignition types of internal combustion engines. In accordance with this definition, fuels such as kerosine, benzine, alcohol, propane, butane, and others qualify under the general heading of "gasoline." While these fuels may be delivered directly into the combustion chambers by injection instead of carburetors, if combustion is normally initiated by electrical ignition, or some similar means, the engine is considered a "gasoline engine" and the fuel a "gasoline" within the scope of this definition (Ref. 16). Detailed requirements for gasoline for use in military vehicles are given in Military Specification MIL-G-3056A, entitled *Gasoline, Automotive, Combat.*

Similarly, Diesel fuel is the broad category assigned to fuels used in compression-ignition internal combustion engines. Detailed specifications for Diesel fuels for use in military vehicles are given in Federal Specification VV-F-800, entitled *Fuel, Oil, Diesel.*

Lubricants used in military automotive vehicles include the engine oils, gear oils, preservative oil, hydraulic fluids and greases. These are supplied in various grades and types to cover the wide range of climatic conditions in which military equipment is expected to function (see par. 3-2.3). Detailed requirements for these are given in the Military Specifications listed in Table 3-4.

3-3 MAINTENANCE DOCTRINE*

3-3.1 GENERAL CONSIDERATIONS

Proper maintenance of mechanical equipment is required to achieve maximum service life. This is especially true of military automotive vehicles, which must be always ready to function under severe conditions. The more complex a mechanism, the greater the number of maintenance operations that are usually required. Modern military automotive equipment, although made initially rugged and reliable, cannot endure for long without adequate preventative maintenance. The Army Maintenance System is the end result of a long chain of logistic functions, and neglect during design and development may render maintenance extremely difficult and even impossible.

Military equipment should be so designed as to be capable of maintenance during severe military usage by means of readily available skills, tools, and supplies. The objective of maintenance engineering is to reduce the time during which equipment is denied to the user, and also to reduce the manpower, tools, equipment, and supplies that are required to perform competent maintenance. These objectives can be more readily attained if sufficient consideration is given during the equipment design phase to some of the following factors:

a. Components should be so designed and installed as to provide adequate working clearances and visibility for ease of servicing, adjusting, removal, and installation.

b. Service points for checking, replenishing, and draining of fuel, lubricant, hydraulic fluid, pneumatic pressures (including tires), coolant, electrolyte, etc., should be readily accessible and should incorporate features that facilitate these operations without being vulnerable to damage or contaminated.

c. Fuel tanks with capacities in excess of 50 gal should be capable of accepting fuel at the rate of 50 gal per min. Tanks having capacities of less than 50 gal should be capable of being filled within one min.

d. In general, all parts whose working surfaces are subject to wear or deterioration should be provided with appropriate means for lubrication. Exceptions to this are certain surfaces, such as tank-drive sprockets, on which lubricants are objectionable. Permanently lubricated assemblies and assemblies that require no lubrication should be used at all points where they can meet the rigorous requirements of the military environment and where their application is economically feasible. Porous, lubricant-impregnated bearings and rubber-bushed journals are examples of such devices.

e. Materials should be resistant to or protected against chemical and electrolytic corrosion brought about by atmospheric conditions and galvanic action between dissimilar metals in contact, and against normal wear and abrasion to the extent that such deterioration will not reduce the effectiveness of the equipment nor appreciably increase its maintenance require-

* For more complete coverage of maintenance engineering in military design the reader should consult Reference 27.

ments. Particular attention should be given to surfaces subject to wear and abrasion, such as, running boards, cabs, floor boards, and load decks, and to small, light parts that are vulnerable to corrosion, such as sheet metal items, screws, nuts, bolts, springs, retaining chains, and other thin-gage parts.

f. All electrical, pneumatic, hydraulic, and fuel systems should be resistant to corrosion and fungi and be protected against the entry of foreign matter. When selecting materials for these systems, careful consideration should be given to their compatibility with the fluids they will contain. Copper and high copper content alloys, for example, should not be used in contact with modern fuels. Copper has a catalytic effect on gasoline causing high gum deposits (Ref. 16).

g. Exposed surfaces should be shaped to avoid recesses that tend to collect and retain dirt, water, cleaning fluids, servicing fluids that may have been spilled or lost during operation, or foreign matter. Where such recesses cannot be avoided, suitable deflectors, closures, and drains should be provided.

h. Equipment should be designed to require a minimum number of periodic maintenance adjustments. Those maintenance adjustments that cannot be eliminated should be simplified to permit their accomplishment at the lowest practicable maintenance level.

i. To expedite the replacement of components, suitable aligning, piloting, guiding, lifting, and positioning features should be incorporated into the design.

j. The least possible number, sizes, and types of fastening devices should be used to minimize the number of operations and tools required for the removal and installation of components.

k. Maintenance operations should be capable of being performed by personnel wearing arctic clothing, including heavy gloves, to the maximum extent practicable.

l. Equipment should be designed to permit maintenance operations within a reasonable time after halting the vehicle, without danger to the maintenance personnel of being burned. Components which must be handled under these conditions, particularly heavy items that require a long time to cool, should be provided with handles, eyes, or other suitable devices.

m. The design of equipment should, to the maximum extent possible, permit maintenance adjustments with the standard tools issued with the vehicle.

n. Emphasis should be placed upon the use of a minimum number of line items of supply in maintenance, the use of standardized parts and hardware that are, in general used throughout the military supply system (Ref. 17), and the use of interchangeable parts and assemblies, particularly those incorporated into other equipment supported concurrently by direct support maintenance units.

3-3.2 MAINTENANCE CRITERIA

In January 1959, the Ordnance Corps issued a directive outlining the improvements expected in operational life expectancies of military vehicles during the next 10 years. This directive expresses the maintenance and design goals for future vehicles and is known as the *Maintenance Criteria* (Ref. 2). The ten-year period covered by the directive is divided into two phases and the maintenance criteria are expressed as follows:

a. "During Phase I (1960-1963), ground vehicles should have a 90-percent probability of completing the following mileages, in a military environment:

i. Wheeled, tactical vehicles: 10,000 miles without field maintenance and 20,000 miles without depot maintenance.

ii. Tracked vehicles: 2,000 miles without field maintenance and 4,000 miles without depot maintenance.

b. During Phase II (1964-1970), ground vehicles will have a 90-percent probability of accomplishing the following, in a military environment:

i. Wheeled, tactical vehicles: 25,000 miles without field or depot maintenance.

ii. Tracked vehicles: 5,000 miles without field or depot maintenance."

It is a recognized fact that the level of effort required to maintain current military vehicles is excessive. Improvements in future operational, organizational, and personnel activities will result

TABLE 3-5 TEST REQUIREMENTS FOR TRACKED VEHICLES (Ref. 18)

VEHICLE TYPES	COURSE AND MILEAGE				
			Cross Country		
	Paved	Gravel	Level	Hilly	Total
Tanks and Self-Propelled Weapons	1000	1000	2000	2000	6000
Armored Infantry Vehicles, Cargo Tractors (with towed loads), Vehicles with limited amphibious capabilities	2000	2000	2000	2000	8000
Modified Standard Vehicles incl. Heavy Recovery Vehicles and Wreckers	1000	1000	2000	2000	6000
Integral Flame Throwers in Tank-Type Vehicles	—	200	500	—	700
Armored Engineer Vehicles		250	250	500	1000
Amphibious Vehicles (LVT type)	1000	1000	1000	2000	5000

in only marginal improvements to the Army Maintenance System. Thus, the only prospect for a revolutionary reduction in the maintenance requirements of ground vehicles lies in design and development.

In order for a wheeled vehicle to achieve a 90-percent probability of surviving 20,000 miles, the vehicle will require an average design life of 34,000 miles. In addition, each of the six major components will need a 98-percent probability of surviving 20,000 miles.

3-3.3 DURABILITY AND RELIABILITY

These terms, "durability" and "reliability" are familiar to most designers, and yet there exists a certain amount of confusion with respect to their distinction, particularly as used in military design circles.

Durability is the term used to describe the ability of an object, device, or system of devices to render satisfactory performances over an extended period of time of continual operation when used in the service for which it was intended. It deals with the operational endurance of the item and is related to the time period during which satisfactory performance is obtained.

Reliability is the interaction of the durabilities of the individual components that constitute a particular assembly and the probabilities that each of these components will perform satisfactorily for the intended period under the operating conditions encountered. For a system comprised of several independent components, the overall reliability is the product of the individual reliabilities. For example, an assembly consisting of three components having a 90-percent reliability, each, will have an overall reliability of 72.9 percent. Similarly, 100 components with a 99-percent reliability, each, will have an overall reliability of only 36.5 percent. This illustrates the difficulty of obtaining a high degree of reliability with highly complex systems.

Reliability also includes the capacity of the device to perform its mission after sustaining the destruction or failure of specific components (Ref. 18). This facet of reliability is particularly important in evaluating combat vehicles. The capability to operate without individual components, such as certain road wheels, power supply, cooling fans, vision devices, or even one or more engine cylinders, may mean the successful accomplishment of the mission or the survival of the crew.

In response to the operational criteria specified in Ref. 2 for military vehicles (also given in

TABLE 3-6 TEST REQUIREMENTS FOR WHEELED TRANSPORT-TYPE VEHICLES (Ref. 21)

VEHICLE CLASSIFICATION	TYPE OF COURSE AND MILEAGE										
	Gravel	Belgian Block	Truck(a) Cross Country: Hilly	Truck(a) Cross Country: Level	Paved Roads(b)	Sand	Marsh and Swamp	Water Operations	Miles Per Cycle	No. of Cycles	Total Mileage
Military Trucks (c)	800	150	2000	1800	200	50	—	—	5000	4	20,000 (c)
Truck Bodies and Equipment (c)	500	50	250	250	2500	250	—	—	3800	3	11,400
Lightweight Low-Mileage Trucks (c)											
Sprung Types	250	50	125	125	1250	125	75	—	2000	2	4,000
Unsprung Types	550	50	600	600	350	100	250	—	2500	2	5,000
High Floatation Vehicles	500	250	1250	500	1000(e)	250	250	—	4000	1	4,000
Amphibians (c)	950 (f)	100 (f)	500	500	1650(e)	500(g)	—	175	4200	2	8,400
Fire Trucks	500	50	250	250	2500	250	—	—	3800	1	3,800
Commercial Trucks, Buses and Passenger Cars	750	50	—	—	4200	—	—	—	5000	7	35,000

LEGEND:

(a) 50% of all cross country mileage is run under muddy conditions.
(b) Includes incidental mileage between test courses and travel to other test locations.
(c) Vehicles that have two rated payloads are tested with payload applicable to course being run for first cycle. Thereafter, highway payloads are used for all courses.
(d) After the fourth cycle, additional high-speed paved mileage is accumulated for a total of 22,800 miles.
(e) May be reduced for vehicles considered nonroadable because of width or other factors.
(f) Run paved, gravel, Belgian block alternate laps, with 15 minutes in water after each lap for a total of 125 hours.
(g) Preferably ocean beach sand with 50-hours operation in salt water.

par. 3-3.2), a set of test requirements have been established to evaluate the reliability of military vehicles. These are shown in Tables 3-5 and 3-6.

3-4 HUMAN FACTORS*

Since the military vehicle is a complex man-machine system, it is not only logical, but imperative, that the vehicle designer give due consideration to the physiological and psychological limitations of the human being. Machines cannot operate without guidance and control by man. But in order to exercise efficient control, the operator and crew must be reasonably comfortable, have adequate perception of their situation, have sufficient space in which to perform their functions as crew members, and be provided with controls that are easily accessible, can be operated without strain, and that do not interfere with other operator functions. A considerable amount of data is available on the limitations and capabilities of the human operator, as well as those conditions that produce optimum performance. The vehicle designer, however, is subjected to certain inexorable military requirements which often preclude optimum solutions. He is, thus, forced to compromise the optimum solution for a less desirable one, but he must arrange his design to afford the crew certain minimum standards of comfort and ease of operation if they are to maintain satisfactory performances.

Numerous charts and tables exist which give pertinent body dimensions of human operators in various postures (Refs. 20, 21, 24, 26). When using this data, the designer must give due consideration to the clothing the operator will be wearing and to his equipment, which he may be wearing, be using, or for which storage space is required. In addition, a certain amount of "elbow room" is required within which the operator can exercise

* Basic human factors, applying to general military design, are treated in Reference 27.

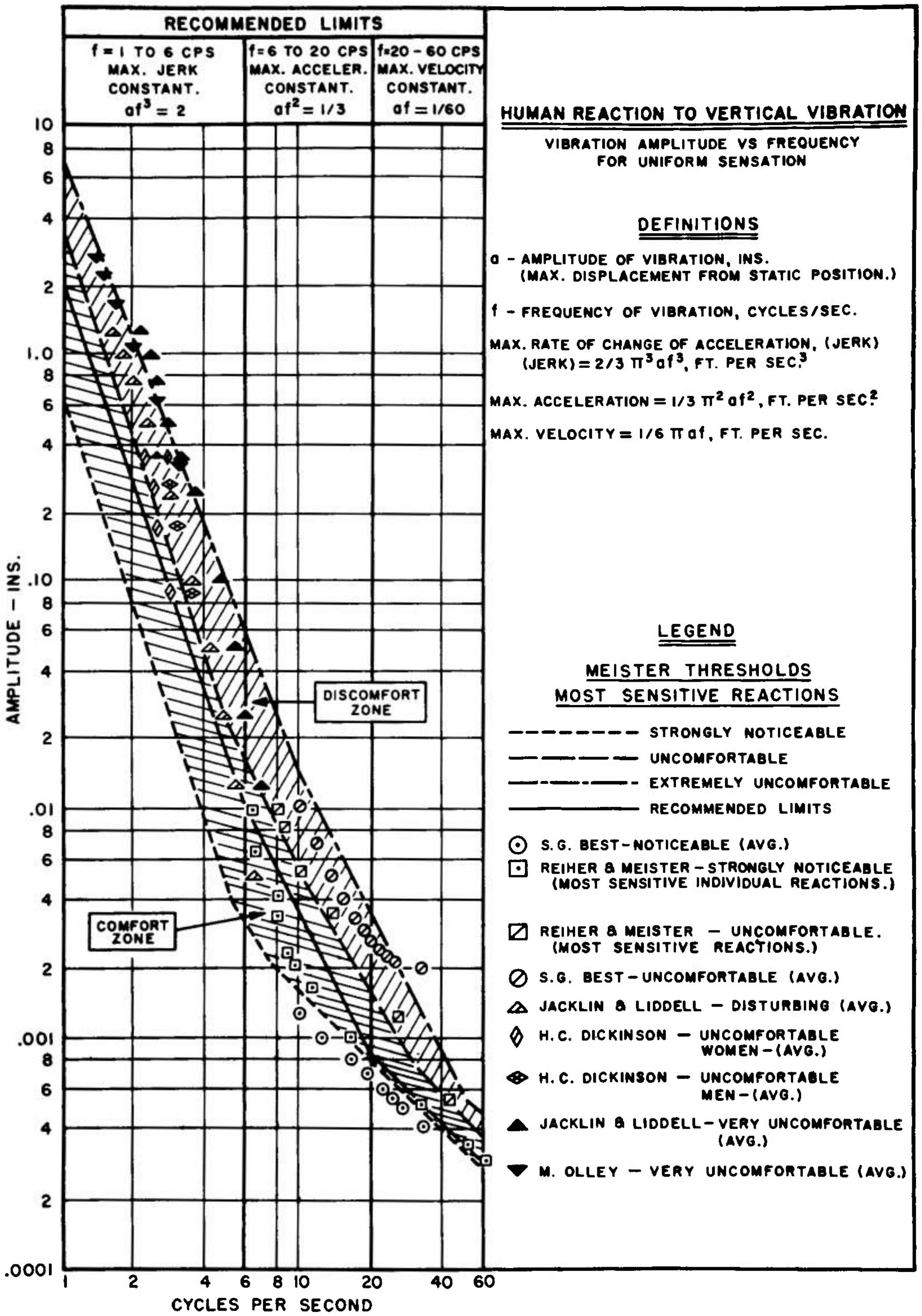
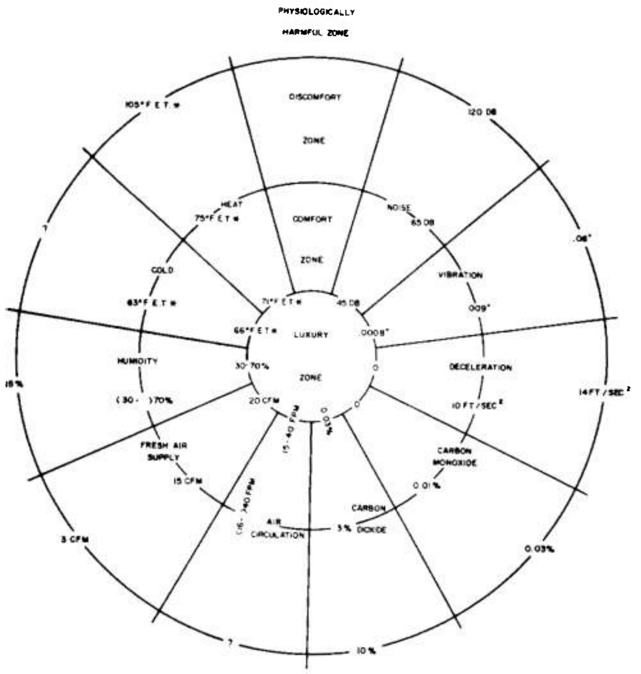


Figure 3-12. *Human Reactions to Vertical Vibration (Ref. 23)*



* INDICATES "EFFECTIVE TEMPERATURE" WHICH TAKES INTO ACCOUNT THE INTERRELATIONSHIP BETWEEN TEMPERATURE AND HUMIDITY

Figure 3-13. ***Various Comfort Criteria (Ref. 24)***

limited movement. If he cannot shift his position occasionally, he cannot relax; and, under these cramped conditions, he will become fatigued more rapidly.

Operational controls should be arranged in a manner that will maximize their efficiency of operation. The operating effort required by the controls should fall well within the physical force limits of the operator. All controls, including switches, should be capable of being operated both with and without the use of arctic mittens. Instruments should be located in an order of importance to the operator and should be so positioned and illuminated as to produce no ocular strain on the operator.

Adequate heating and ventilation must be provided (see Chapter 14). Normally, adequate heating requires about 180 Btu per hr per cu ft of crew space. Ventilation requirements of a closed vehicle are about 55 cfm per man, plus an additional 165 cfm per man during firing of normal tank armament where the toxic gases from the firing find their way into the crew compartment (Ref. 20).

The most serious operational factors affecting the crews of military vehicles are vibrations and shocks. When operating cross country over rough terrain, or on irregular road surfaces, the operator and crew are subjected to severe jostling, vibrations, and impacts. Under these conditions, the operator has difficulty in controlling the vehicle, his vision is impaired, and he experiences intense physical discomfort causing him to reduce the speed of his vehicle to an acceptable level. Even so, operator control, performance, and comfort are seriously compromised and hazards still exist.

The most common physiological effect produced by this environment is kidney irritation. It is believed that long-term exposure to this environment produces disabilities such as nephroptosis (a downward displacement of the kidneys which may incapacitate the individual for even light duties), ocular irritation, gastric upsets, hernia, and fracture or displacement of the invertebral disks (Ref. 20).

So serious is this shock and vibration problem, to both the operational capabilities of the vehicle as well as to the physical well-being of the crew, that maximum emphasis should be given during the vehicle design phase to means of ameliorating this condition. This can be accomplished by applying some of the following measures:

1. Improved vehicle suspension systems to minimize shocks and vibrations.
2. Improved seating and back support.
3. Full body restraint devices to prevent or limit involuntary movements of individual.
4. Restraining or limiting devices to prevent or restrict involuntary head movements.
5. Elimination or padding of hazardous projections, corners, and edges from the crew member's normal envelope of motion to reduce the possibility of injury.
6. Analysis of system during design, development, and prior to manufacture by competent human factors evaluators.

The reaction to vibrations, from a physical comfort viewpoint, is affected by the combination of the amplitude and frequency of the vibration. To maintain comfort when the frequency is increased, the amplitude of the vibration must be decreased. This is shown graphically in Fig. 3-12 from which comfort criteria can be selected.

Figure 3-13 shows graphically the comfort criteria for various factors to which the human is

TABLE 3-7 SHOCK AND VIBRATION DATA

Type of Operation	Part of Vehicle Considered	Shock (Accel. in g's)	VIBRATION Vertical g's	Vertical cps	Longitudinal g's	Longitudinal cps	Transverse g's	Transverse cps
High Speed on Hard Pavement	Hull		4	500	3.8	500	2.3	520
	Instr. Panel		2.6	300	1.8	400	2	350
	Eng. Mount.		12.5	450	15	900	14.1	650
	Generator		10.3	650	18.7	700	18	800
Medium Speed Off-the-Road	Hull		2.3	540	2	520	0.6	430
	Instr. Panel		1.2	120	1.3	120	0.9	120
	Eng. Mount.		11.4	500	18.7	900	11.3	850
	Generator		8	650	10	700	25.4	900
Low Speed Rough Terrain	Hull	8						
	Instr. Panel							
	Eng. Mount.		5	500	10.8	850	13	700
	Generator		11	550	10	750	14	900
	Axle (Semitrailer)		9.4	300	3.6	350		
					5.4	10	2.5	400
	Fifth Wheel Plate		36.4	150	21.9	400	12.2	100
	Cargo Bed Above Fifth Wheel		14.5	100	4.8	250	2.8	30
	Cargo Bed Above Axle (Semitrailer)		3	3			0.8	16
							0.4	90
	Side Wall of Van Semitrailer		2.4	20			3.3	400
Shipment by Truck	Vehicle Assembly	8	2	300	2	300	2	300
Shipment by Rail	Vehicle Assembly	20	2	70	2	70	2	70
Shipment by Fixed Wing Aircraft	*Vehicle Assembly*							
	Fwd.	8			5	300		
	Side	1.5					5	300
							0.25	10
	Vert. (up)	3	5	300				
	Aft.	1.5	0.5	10				
Shipment by Rotary Wing Aircraft	*Vehicle Assembly*							
	Fwd.	4						
	Side	1.5						
	Vert. (up)	2						
	Aft.	2						
Parachute Drop	Vehicle Assembly	16						
Ballistic Impact	Turret	20 (0.75″ ampl.)	50	1000			140	600
HE Blast	Turret and Hull	25 (1.0″ ampl.)						

REFERENCE:

Compiled from various sources including: AR 705-35, *Criteria for Air-Transportability and Air-Delivery of Material*, Feb. 1960; "Tracked Vehicles Design Practices Guide," Industrial Engineering Branch, Industrial Div., Ordnance Tank-Automotive Command, July 1958.

sensitive such as noise, vibration, deceleration, air pollution, humidity, and temperature. The chart is self-explanatory.

A compilation of quantitative data on shocks and vibrations normally experienced by military vehicles during various operating conditions is given in Table 3-7. These data are given here to serve as guides in the design of military vehicles.

SECTION III CONSIDERATIONS OF INTENDED USE

Certain design requirements arise from the ultimate purpose, or use, intended for the vehicle. The basic uses that pertain to military vehicles are discussed in this section, and the factors that require consideration by the designer are pointed out. Military vehicles are used most commonly for one or more of the following functions: (a) to carry personnel, (b) to carry material, (c) to provide armor protection, (d) to provide mobility for a weapons system, (e) to serve as a prime mover, (f) to serve as a towed vehicle, (g) to mount special-purpose equipment, (h) any combination of the foregoing uses. Typical examples of the major types of vehicles are given in Chapter 4.

3-5 TO CARRY PERSONNEL

Design requirements for vehicles whose function is to carry personnel relate directly to human factors. Sufficient space must be provided each individual for a minimum of comfort. When allocating space requirements, due allowances must be made for the equipment, if any, that will accompany each individual. Combat personnel moving forward in armored personnel carriers carry full field equipment, weapons, ammunition, rations, water, signal equipment, and medical supplies and equipment. These items add appreciably to the volume occupied by each individual and to his gross weight.

Doors, hatches, and companionways must be designed sufficiently large to accommodate a large man (95th percentile) complete with full field equipment attached to his person. Rifles, submachine guns, small rocket launchers, and similar individual weapons are usually carried slung across the soliders' backs to leave hands free to carry other equipment or supplies, to take advantage of hand holds when needed, and to help maintain balance. These slung weapons protrude considerably beyond the soldier's silhouette and are important items to be considered when allocating space for individuals. Appropriate steps or ladders must be provided to facilitate boarding the vehicle.

Wounded men may have to be assisted into and out of vehicles. This presents a design requirement of providing ample space for handling casualties into and out of vehicles. Casualties may also occur within the vehicle, making it necessary to care for them enroute. Provisions must be made for sufficient space for lying a wounded man down to ameliorate shock, to ease his suffering, and to give him medical assistance.

Adequate ventilation is an important consideration in personnel carriers, particularly in fully enclosed, armored and amphibious vehicles. Heating, too, is a consideration, especially in arctic regions, while adequate cooling is a problem in the tropics.

Seating and body restraint devices must be considered to protect the passengers from dangerous road shocks and vibrations, particularly in high-speed, cross country operations.

The passengers of a totally enclosed vehicle, particularly an amphibious vehicle, show signs of claustrophobia when deprived of all means of viewing their outside surroundings. To minimize anxiety and even hysteria, vision devices must be provided. Similarly, emergency escape hatches have a marked psychological effect upon the passengers in addition to their actual function of providing a means of exit.

Another requirement in the design of personnel carriers is that some means of communication be provided between the driver and passenger compartment. This may be through visual contact or by means of an intercom system. It permits the vehicle operator to alert the troops to impending dangers or other important circumstances, and it provides the passengers with a means of notifying the driver of emergencies that may occur in the passenger compartment.

3-6 TO CARRY MATERIAL

The primary requirements of vehicles designed to carry material are that the vehicle be structurally strong and that its design facilitates rapid loading and unloading. The material being loaded is often dropped, thrown, or dumped into the cargo compartment, thus, subjecting the cargo body to impact loads. Cargo bodies should be made of impact and abrasion resistance materials, and designed to withstand local impacts, such as produced by angular objects, without damage. Cargo discharge openings should be sufficiently large to permit expeditious unloading of cargo, and cargo compartments should be made easy to clean.

3-7 TO PROVIDE ARMOR PROTECTION

The application of armor to a vehicle increases its weight appreciably and decreases the available, useful space. A beneficial effect of armor on a vehicle is the additional strength and stiffness that it imparts to the vehicle if properly applied. As a result, combat tanks and tracked, amphibious vehicles are able to dispense with the frame that supports the body of conventional vehicles and utilize the rigid, armored hull to serve as a combination frame and body.

The undesirable weight factor associated with the application of armor to a vehicle can be reduced through its judicious use and through clever arrangement. A vehicle does not require the same degree of protection from all directions and, therefore, does not require the same thickness of armor in all areas. By clever use of design, maximum benefits can be gained by offering oblique planes to the projectile. This obliquity increases resistance to penetration in two ways: it presents a greater thickness of armor to the path of the projectile, and the oblique plane tends to deflect the projectile, or at least, to provide sufficient resistance to the missile to decrease its residual velocity below that required for penetration.

The probability of projectile impacts is an important factor in determining the required armor protection. This probability is controlled by (a) the relative size of the area under consideration with respect to the total projected area in a plane normal to the expected line of attack, (b) the location of the area in relation to the probable aiming point, and (c) the probability of exposure to enemy fire of the area when allowances are made for the protection and concealment provided by the terrain. Studies of these factors over a period of time have established the axiom that the heaviest attack will be against the front wall of the vehicle, and against the sides, top, floor, and rear, in that order.

3-8 TO PROVIDE MOBILITY FOR A WEAPONS SYSTEM

When the purpose of a vehicle is to provide mobility for a weapons system, the method of employing the weapon system must be determined. If the weapon is to go into action while mounted upon the vehicle, provisions must be made for absorbing the weapon recoil forces. If the weapon is to go into action while mounted on the ground, suitable means must be provided for mounting and demounting the weapon rapidly. If the weapon is to be fired from the vehicle while the vehicle is in motion, the vehicle suspension system may need to be designed for maximum stability to the weapon for greater accuracy of fire. In this situation, the suspension system will receive the impact of the weapon recoil. Some vehicle-weapon systems incorporate a suspension lockout feature which, in effect, provides a rigid suspension while firing.

3-9 TO SERVE AS A PRIME MOVER

Prime movers require a short wheelbase to facilitate maneuvering the towed load, a strong frame structure to withstand the pull of the towed load and to resist twisting or racking, and sufficient weight to develop maximum traction. When feasible, prime movers are equipped with a cargo body which not only gives them the capability of hauling cargo, but provides a means of supplying additional weight to the driving wheels for added traction. Truck tractor types of prime movers (see Chapter 4) support part of the weight of the semitrailer which gives them the axle loading they need to develop sufficient tractive effort.

Pintles and towing lugs should be located at least 30 in. above the ground and not more than 40 in. This is to facilitate maneuvering over rough, cross country terrain and to maintain the attitude of trailers nearly horizontal.

The spread between towing lugs should be between 30 and 60 in. A spread of 50 in. is preferred. Towing lugs should be located in such a manner that they will accept the tow bar without interference of any kind, and will permit the tow bar to swing 60° above and below the horizontal.

3-10 TO SERVE AS A TOWED VEHICLE

Towed vehicles have many of the same problems found in powered vehicles. Considerations of frame, suspension, steering, and braking are comparable. Semitrailers and two-wheeled trailers require some type of landing gear to support the front of the trailer when uncoupled from its prime mover. Of prime importance is the matching of towed vehicles to the prime mover. The coupling halves must match and must be of approximately the same height above the ground. Electrical systems of the two vehicles must be compatible (same voltage and type). Compatibility is also necessary between the two braking systems, for example, a prime mover equipped with air brakes cannot be coupled to a towed vehicle provided with vacuum brakes.

Towed vehicles are often required to have amphibious capabilities to permit being towed behind amphibious prime movers, even when water borne. This makes it necessary to design the body of the towed vehicle like the hull of a boat—waterproof; buoyant, even when fully loaded; stable when afloat, empty as well as when fully loaded; provided with sufficient freeboard when fully loaded to avoid swamping in the anticipated seas; and of minimum practicable hydrodynamic drag. Consideration must also be given to circumvent any tendency of the towed vehicle to dive beneath the surface, when being towed through water, or to yaw and make steering difficult. Also, it is advisable to provide suitable means, within the towing vehicle, of quickly uncoupling the towed vehicle under emergency conditions to safeguard the towing vehicle from being swamped.

3-11 TO MOUNT SPECIAL-PURPOSE EQUIPMENT

Automotive vehicles are often designed for the express purpose of providing mobility for special-purpose equipment. In these situations, the operating characteristics and requirements of the special-purpose equipment must be examined to determine the impact these will have upon the vehicle. For example, electronic warfare equipment, radar equipment, and electronic fire direction equipment, usually requires substantial amounts of electrical power and often of 400-cycle AC type. This power requirement is usually beyond the capability of the vehicle's electrical system making it necessary to install auxiliary electrical generating equipment. This auxiliary equipment may be driven by the main power plant of the vehicle or by an auxiliary engine. The relative efficiency and economy of both methods must be compared when deciding which course to take.

Some vehicles mounting special-purpose equipment may need to be mobile relatively few times during their operational life. For example, certain semipermanently installed radar detection equipment may require mobility only when moving to its operational location. Since it is semipermanently installed, it may seldom, if ever, require the mobility of the vehicle again. Under these conditions, it would be wasteful to provide a highly developed vehicle, capable of operating for long periods under adverse conditions, only to have it stand by in idleness. A vehicle for this type of application should be of a minimum design commensurate with its operational requirements; and a maximum number of its components, such as power plant, electrical system, and even structural components, should be given a dual role, i.e., a role as a vehicle component and adaptable to a role as a component of the special-purpose equipment (radar detection apparatus, etc.).

Other types of special-purpose equipment may subject the vehicle to extreme loads or overturning moments requiring reinforcement of highly stressed members, the installation of anchoring devices, or suspension lockout provisions. The development of most special-purpose vehicles usually is done by some military agency other than the Mobility Command and is based upon the use of standard components found on military vehicles and for which the Mobility Command is responsible. Usually, Mobility Command designers collaborate on such developments. Typical examples of such projects are Air Force and Navy crash trucks, ambulances, fire-fighting apparatus and combat bulldozers, wreckers and tank recovery vehicles, mechanized flame throwers, and various mobile launchers and service vehicles for rockets and guided missiles.

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CHAPTER 4

REPRESENTATIVE TYPES OF AUTOMOTIVE ASSEMBLIES*

Military vehicles are classified into three broad categories, namely, combat vehicles, tactical vehicles, and administrative vehicles. The administrative vehicles are largely standard commercial vehicles and of relatively minor interest to the military designer. The combat and tactical vehicles, however, are the total concern of the military vehicle designer and are, therefore, of prime importance to this book. These two general categories are further subdivided into various types. The many types that exist, coupled with their similarities in appearance and operating characteristics, make their identity and purpose confusing to the noninitiate, and often not too well understood by those who have associated with military vehicles for a long time. The purpose of this chapter, therefore, is to define the various types of vehicles, discuss their missions, describe their general characteristics, and present illustrations of representative types. By no means are all vehicles for which the Army Materiel Command is responsible included in this chapter, neither in the illustrations nor in the tabular data; nor are the examples cited necessarily the latest models of their type. Wherever illustrations of current models were readily available, they were used; but obsolescent models are included because their purpose is mainly to illustrate some representative type and not to show the latest version. A presentation of representative military vehicles, currently the responsibility of the Army Materiel Command, can be found in Ref. 1 cited at the end of this chapter.

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SECTION 1 COMBAT VEHICLES

4-1 GENERAL DISCUSSION

A combat vehicle is defined as a land or amphibious vehicle, with or without armor or armament, designed for specific functions in combat or battle. The installation of armor or armament onto other than combat vehicles does not alter their original classification (Ref. 2). Combat vehicles may be wheeled or track-laying; but, in all cases, they are designed to provide a high degree of mobility in cross country operations. Classic examples of combat vehicles are: tanks, self-propelled artillery, and armored cars. The majority of present combat vehicles are track-laying; however, research in the field of land locomotion has shown that full potential of the wheeled vehicle for cross country operations has not been exploited (Refs. 3, 4, and 5).

Because of their mission, combat vehicles are usually furnished with both armor and armament, although certain antitank vehicles are unarmored and depend upon their decreased silhouette and increased mobility for protection. There has been a continuing demand for increased firepower while maintaining or increasing the mobility of the vehicles. Currently, to help fill these demands, the trend is to decrease the vehicle weight by using lightweight armor. This reduction in weight also enables some vehicles, which were formerly too

TABLE 4-1 CHARACTERISTICS OF LIGHT TANKS (Ref. 6)

Nomenclature	Tank, Light M2A4	Tank, Light M3A3	Tank, Light M24	Tank, 76mm Gun T41E1
Weight, lb	25,608	30,900	39,500	51,232
Main Armament	37mm Gun	37mm Gun	75mm Gun	76mm Gun
Gun Control	Manual	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.
Secondary Armament	3 cal .30 MG	3 cal .30 MG	1 cal .50 MG 2 cal .30 MG	2 cal .50 MG
Engine	7 Cyl. Radial	7 Cyl. Radial	Twin Cadillac	AOS 895-1
Horsepower	250	250	280	500
Max. Speed, rpm	2,400	2,400	3,400	2,800
Horsepower/Ton	19.5	16.1	14.2	19.6
Transmission	Synchromesh	Synchromesh	Hydramatic	Cross Drive
Final Drive	Front	Front	Front	Rear
Suspension	Volute	Volute	Torsion Bar	Torsion Bar
Steering	Control Diff.	Control Diff.	Control Diff.	Cross Drive
Track Type	Rubber Block Steel	Rubber Block Steel	Rubber Bush. Steel	Steel w/Rubber Pad
Track Width, in.	7-5/8	7-5/8	16	21
Ground Pressure, psi	8.7	10.5	9.4	9.35
Dimensions, in. Length w/o Gun Width Height	 174½ 97¼ 98¼	 168 99½ 101	 198 112 87	 229 129 108-3/8
Turning Radius, ft	47	45	23	Pivot
Ground Clearance, in.	16½	13-7/16	17¾	17¼
Max. Speed, mph	30	31	34	46
Cruising Range, mi.	150	180	100	100
Max. Slope, %	45	45	60	60
Max. Tractive Effort, lb	13,000	14,000	24,000	44,000
Crew	3	4	5	4

heavy, to partake in airborne operations. To a large extent, the reduced weight of the new combat vehicles is due to an extensive use of aluminum in their construction, including aluminum armor. Combat vehicles must meet the most severe operational requirements specified for military vehicles.

Current combat vehicles are classified as:

a. Tanks

b. Self-propelled artillery

c. Combat reconnaissance vehicles

d. Miscellaneous self-propelled weapons

4-2 TANKS

A tank is a self-propelled, heavily armored vehicle designed for offensive combat in either atomic or nonatomic warfare. Current tanks are provided with fully enclosed, heavily armored, revolving turrets in which are mounted the primary weapons. These weapons are large caliber, high velocity, flat trajectory artillery pieces capable of defeating enemy tanks, neutralizing enemy bunkers, and providing effective artillery support to the infantry when needed. They are precision, direct-fire weapons intended primarily for use against enemy armor and other hard and medium point targets. Secondary armament, consisting of machine guns and light cannons, are mounted coaxially with the main armament and at advantageous locations in the hull. These are used against enemy personnel and other soft targets, and reduce the tank's vulnerability to close-in attack. A dual-purpose machine gun is mounted in a cupola atop the turret to provide the tank with a measure of antiaircraft protection and a means of firing on rooftops and into the upper windows of buildings when engaged in town or city fighting. The entire system is track mounted and powered by means of a large capacity, high performance power plant. This, coupled with a highly developed suspension system, provides the vehicle with a high degree of off-the-road mobility in all but the most difficult terrains. The addition of an effective fire control system, efficient communications equipment, and ample storage capacity for fuel and ammunition make the tank a formidable weapon system possessing great fire power, mobility, and armor protection for the crew.

In the past, the usual classification of tanks was light, medium or heavy depending upon the weight. But tank development has been such that the ratio of main gun size to weight of the vehicle has been constantly increasing, consequently, a classification according to gun size is sometimes used. Present policy calls for the development of a main battle tank, an airborne assault weapon, and a light tank along with the retention of the present family of light, medium and heavy tanks. On the basis of gross vehicle weight, tanks are classified as: (1) light tanks, 25 tons or less; (2) medium tanks, above 25 tons, up to 55 tons; and (3) heavy tanks, above 55 tons (Ref. 6).

Tables 4-1, 4-2 and 4-3 show the main characteristics of representative American tanks. Some of the tanks listed are obsolete but are included for purposes of comparison. Representative tanks are shown in Figs. 4-1 through 4-5.

4-3 SELF-PROPELLED ARTILLERY

Self-propelled artillery consists of artillery weapons permanently installed on vehicles to provide mobility. These weapons are fired from the vehicles. The artillery pieces may consist of cannon or launchers for rockets or guided missiles. The primary characteristic of artillery is its great firepower. It is capable of delivering atomic or nonatomic fires within a large area and on a wide front without change of position. It is capable of displacing quickly and delivering accurate fire on targets encountered under various conditions of visibility, weather, and terrain.

Self-propelled artillery is designed with an emphasis on mobility; it is of particular importance that the vehicle be able to reach an emplacement, fire its weapon and withdraw quickly to escape counterfire. To expedite this process, the vehicles are equipped with quick acting, automatic entrenching or emplacement devices. Obviously, weight seriously affects the mobility of these vehicles. Currently developed self-propelled artillery make liberal use of aluminum, even for armor protection, permitting the utilization of smaller engines and power train components.

There are basic differences between tanks and self-propelled artillery, although both use many of the same components, such as power plants, power trains, tracks and suspensions. Although they may both have the same caliber armament, there are usually fundamental differences between the weapons. The main differences between tanks and self-propelled artillery stem from the difference in the principal missions assigned to each.

TABLE 4-2 CHARACTERISTICS OF MEDIUM TANKS (Ref. 6)

Nomenclature	Tank, Medium M3A5	Tank, Medium M4A3	Tank, Medium T23E3	Tank, Medium M26	Tank, Medium M45	Tank, Medium M46A1	Tank, 90mm Gun T42	Tank, 90mm Gun M47	Tank, 90mm Gun M48A2	Tank, 105mm Gun M60
Weight, lb	62,240	71,145	79,390	92,000	92,500	97,000	73,500	97,200	105,000	102,000
Main Armament	75mm Gun 37mm Gun	76mm Gun	76mm Gun	90mm Gun	105mm How.	90mm Gun	90mm Gun	90mm Gun	90mm Gun	105mm Gun
Gun Control	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.	Hyd. Trav. Man. El.
Secondary Armament	3 cal .30 MG	1 cal .50 MG 2 cal .30 MG	1 cal .50 MG 2 cal .30 MG	1 cal .50 MG 2 cal .30 MG	1 cal .50 MG 2 cal .30 MG	1 cal .50 MG 2 cal .30 MG	2 cal .50 MG	2 cal .50 MG 1 cal .30 MG	2 cal .50 MG 1 cal .30 MG	1 cal .50 MG 1 cal .30 MG
Engine	Twin Diesel	Ford GAA	Ford GAN	Ford GAF	Ford GAF	AV 1790-5	AOS 895-3	AV 1790-5	AVI 1790-8	AVDS-1790-2
Horsepower	375	500	500	500	500	810	500	810	810	750
Max. Speed, rpm	2,100	2,600	2,600	2,600	2,600	2,800	2,800	2,800	2,800	2,400
Horsepower/Ton	12.	14.1	12.6	10.9	10.8	16.7	13.6	16.7	15.7	14.6
Transmission	Synchromesh	Synchromesh	Electric	Synchromesh	Torqmatic	Cross Drive	Cross Drive	Cross Drive	Cross Drive	Cross Drive
Final Drive	Front	Front	Rear	Rear	Rear	Rear	Rear	Rear	Rear	Rear
Suspension	Volute	Volute	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar
Steering	Control Diff.	Control Diff.	Electric	Control Diff.	Control Diff.	Cross Drive	Cross Drive	Cross Drive	Cross Drive	Cross Drive
Track Type	Rubber Bush. Steel	Bubber Block	Rubber Block	Steel or Rubber	Rubber Backed Steel	Rubber Backed Steel	Steel Center Drive	Rubber Backed Steel	Rubber Backed Steel	Rubber Backed Steel
Track Width, in.	16	16-9/16	19	23	23	23	24	23	24	28
Ground Pressure, psi	18.1	14.6	12.5	12.4	12.6	13.3	12.0	13.7	11.9	—
Dimensions, in. Length w/o Gun Width Height	 222 109 122½	 247 105 134-7/8	 244½ 128 109	 251½ 137 109	 251½ 138 109	 250¼ 138¼ 111	 232 140 109½	 250¼ 138¼ 116	 294¾ 143 121-5/8	 294 146 122½
Turning Radius, ft	33	31	20	31	31	Pivot	Pivot	Pivot	Pivot	Pivot
Ground Clearance, in.	17-1/8	17	17¾	17¾	17¼	18¾	14	19¼	16½	20-5/8
Max. Speed, mph	30	26	35	30	30	31.6	32	37	30	32
Cruising Range, mi.	100	100	100	110	110	80	90	100	160	250
Max. Slope, %	31	60	60	60	60	60	60	60	60	60
Max. Tractive Effort, lb	27,000	52,000	48,000	58,000	57,000	66,800	67,400	80,000	84,000	65,000
Crew	7	5	5	5	5	5	4	5	4	4

While both possess high mobility and firepower, the primary mission of the tank is to assault the enemy, including his tanks, shoot, and be shot at while in motion (Ref. 9). Tanks must, therefore, be heavily armored and have a weapons system designed primarily to deliver accurate, direct fire while in motion.

Self-propelled artillery, on the other hand, is basically artillery that has been provided with its own integral motive power which enables it to move quickly from place to place. The crew normally dismounts when the weapon goes into action. Thus, self-propelled artillery requires only light armor to protect the crew and weapon from fragments and nearby bursts. The power plant and driver can be located more advantageously, and the lighter armor results in more space within the vehicle allowing the use of a more efficient weapon and higher quality fire control equipment. While the gun of a tank is restricted in operation by the limited turret size, the artillery gun is not subject to such restrictions. It can, therefore, be of better design,

TABLE 4-3 CHARACTERISTICS OF HEAVY TANKS (Ref. 6)

Nomenclature	Tank, Super Heavy, T28	Tank, Heavy, T29	Tank, Heavy, T30	Tank, Heavy, T32	Tank, Heavy, T34	Tank, 120mm Gun, T43
Weight, lb	190,000	144,000	146,000	120,000	146,200	120,000
Main Armament	105mm Gun	105mm Gun	155mm Gun	90mm Gun	120mm Gun	120mm Gun
Gun Control	Manual	Hydraulic	Hydraulic	Hydraulic	Hydraulic	Hydraulic
Secondary Armament	1 cal .50 MG	3 cal .50 MG 1 cal .30 MG	2 cal .50 MG 1 cal .30 MG	1 cal .50 MG 2 cal .30 MG	3 cal .50 MG 1 cal .30 MG	2 cal .50 MG 1 cal .30 MG
Engine	Ford GAF	Ford GAC	Ford GAC	Ford GAC	AV 1790-3	AV 1790-5
Horsepower	500	770	770	770	810	810
Max. Speed, rpm	2,600	2,800	2,800	2,800	2,800	2,800
Horsepower/Ton	5.9	10.7	9.9	12.8	11.1	13.5
Transmission	Torqmatic	Cross Drive	Cross Drive	Cross Drive	Cross Drive	Cross Drive
Final Drive	Rear	Rear	Rear	Rear	Rear	Rear
Suspension	Horizontal Volute	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar
Steering	Control Diff.	Cross Drive	Cross Drive	Cross Drive	Cross Drive	Cross Drive
Track Type	Rubber Backed Steel	Rubber Bushed Steel	Steel or Rubber	Rubber Bushed Steel	Steel or Rubber	Steel or Rubber
Track Width, in.	39	28	28	28	23	28
Ground Pressure, psi	11.7	12.4	12.8	11.4	12.5	12.3
Dimensions, in. Length w/o Gun Width Height	 302½ 179 179	 291 149½ 126	 300 143 126	 278 147¼ 111	 300 149½ 126	 275 148 123¾
Turning Radius, ft	35.5	Pivot	Pivot	Pivot	Pivot	Pivot
Ground Clearance, in.	19	18¾	18¾	18¼	18¾	16
Max. Speed, mph	8	17	27	18.5	17	21
Cruising Range, mi.	100	75	100	75	70	100
Max. Slope, %	60	60	60	60	60	60
Max. Tractive Effort, lb	143,000	95,000	100,000	93,550	146,200	97,000
Crew	4	6	6	5	6	5

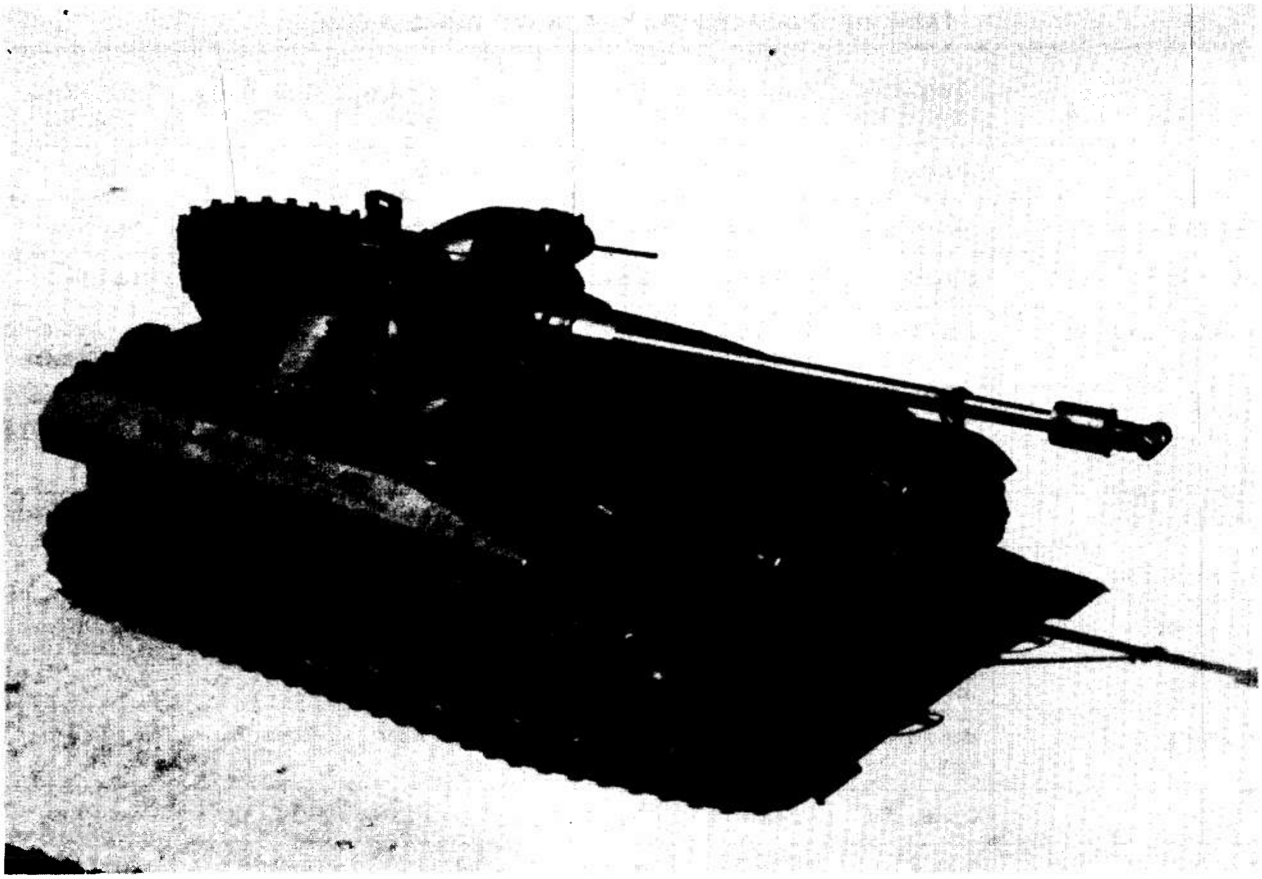


Figure 4-1. *Light Tank, T92, 76mm Gun—1959 (DA 61359)*

larger size, have longer recoil, and be easier to load. Some of the larger artillery weapons, which use separate loading ammunition, are hand loaded and rammed. The smaller weapons, using fixed ammunition, are loaded with automatic or semiautomatic devices. A tank turret is required to have full 360° traversing capability, while the range of vertical travel of the main weapon need not be very large. On the other hand, the traversing capability of an artillery gun is secondary to the importance of range or elevation due to the indirect-fire missions of the artillery weapon.

Self-propelled artillery of conventional design is comprised of two major types, howitzers and guns. Basic differences between the types are that howitzers, in comparison to guns of equal calibers, have shorter barrels with thinner walls, hence lighter weight, lower velocity, shorter recoil and less maximum range. Because of these characteristics the carriages can be designed to permit greater maximum angles of elevation, providing plunging fire to reach targets behind masks or on reverse slopes. Many of the present self-propelled guns and howitzers resemble tanks in outward appearance.

Relative newcomers to the artillery family are the tactical missiles. Self-propelled versions of these weapons consist of the missile, launcher, and the erection and control equipment all mounted upon a suitable automotive transporter. Wheeled, truck-type transporter-launchers are currently used for such missile systems as the Honest John, Lacrosse, and Corporal missiles, imparting considerable battlefield mobility to these weapons (Ref. 7).

The main characteristics of representative self-propelled artillery are listed in Table 4-4 and illustrated in Figs. 4-6 through 4-12.

4-4 COMBAT RECONNAISSANCE VEHICLES

A reconnaissance vehicle, in general, is any vehicle that is assigned a reconnaissance mission. However, the category of vehicles referred to here

Figure 4-2. *Light Tank, M41A1, 76mm Gun—1958 (DA 56349)*

Figure 4-3. *Medium Tank, M48A2, 90mm Gun—1958 (DA 56348)*

Figure 4-4. *Main Battle Tank, M60, 105mm Gun—1959 (DA 60146)*

as "combat reconnaissance vehicles" is comprised of those vehicles that have been designed specifically for reconnaissance missions in close proximity to the enemy and to execute security and counter-reconnaissance missions against him. They are provided with sufficient firepower for their defense and to give them the capability of performing missions of reconnaissance by fire (Ref. 10). Armor is provided to protect the crew and vehicle from counter-fire, but the amount of armor is compromised in preference to speed and mobility. Thus, speed and mobility are the prime characteristics of combat reconnaissance vehicles and are their main defense.

Light tanks, armored cars, and semitracked vehicles (half-tracks) are the principal combat reconnaissance vehicles. Half-tracks were fairly satisfactory in past operations, but they lacked the cross country capabilities of fully-tracked vehicles. They were uncomfortable to ride, difficult to steer, and lacked maneuverability. The front wheels had almost no effect upon steering when operating in mud or soft sand and transmitted a great deal of steering shock to the driver when operating over badly rutted or rough terrain.

Armored cars are wheeled vehicles (4-, 6-, and 8-wheeled) and, therefore, do not suffer from the steering and maneuverability difficulties experienced by the semitracked vehicles. When operating in mud, soft sand, and snow, they have been found inferior to fully-tracked vehicles; and are, therefore, limited in their operations to roads and fairly firm terrain. The current trend in the design of combat reconnaissance vehicles is toward high-speed, lightweight, fully-tracked, amphibious, air-transportable vehicles capable of operating efficiently in mud, soft sand, and snow as well as on rough terrain. The characteristics of the latest vehicles of this type are classified and, therefore, cannot be given in this book. Table 4-5 lists the characteristics of some typical armored cars and half-tracked reconnaissance vehicles used in the past. The characteristics of light tanks are given

Figure 4-5. Heavy Tank, M103A1, 120mm Gun—1958 (DA 55852)

in par. 4-3. Figure 4-13 shows the appearance of a typical reconnaissance vehicle.

4-5 MISCELLANEOUS SELF-PROPELLED WEAPONS

A number of self-propelled weapons are of particular interest either because of the type of armament they carry, the armament size in comparison with the vehicle weight, or the amount of armor carried. Some typical vehicles of this category are the "Ontos," the SPAT, and the Vigilante "B", described in the paragraphs following.

4-5.1 106mm SELF-PROPELLED, MULTIPLE RIFLE, M50 (ONTOS)

This is a relatively lightweight armored, assault vehicle weighing only 8.5 tons that is currently favored by the Marine Corps. It is a track-laying vehicle capable of high speeds, excellent maneuverability, and a high degree of cross country mobility on adverse terrain. Its armament consists of six 106mm recoilless rifles, mounted on a simple elevating structure that is common to all six rifles, four cal. 50 spotting rifles, and one cal. 30 machine gun. The weapons are fired from within the vehicle, thus giving the crew the advantages offered by the light armor. The vehicle is both air-transportable by military tactical aircraft and air-droppable. This vehicle is shown in Fig. 4-14 (Ref. 11).

4-5.2 90mm SELF-PROPELLED GUN, M56 (SPAT)

This vehicle, known as the SPAT (self-propelled antitank) or the "Scorpion," is an unarmored, track-laying, antitank weapon, weighing 7.5 tons, and designed for air transport and airdrop (see Fig. 4-7). It is equipped with a 90mm gun and carries a crew of three. It is highly maneuverable and has good mobility in off-the-road operations. A 200-hp air-cooled gasoline engine coupled to a crossdrive transmission gives the M56 a speed of 28 mph and a cruising range of 140 miles. It can safely negotiate a trench 48 in. wide, a vertical obstacle 30 in. high, and can climb a 60% slope (Refs. 11 and 12).

TABLE 4-4 CHARACTERISTICS OF SELF-PROPELLED ARTILLERY (Ref. 7)

Nomenclature	Gun Twin, 40mm, SP*, M42A1	Gun, 90mm, M56 (SPAT)*	Gun, 155mm, SP*, M53	Gun, 175mm, SP*, T235	Howitzer, 105mm, SP*, T98E1	Howitzer, 155mm, SP*, T99E1	Howitzer, 8" SP*, T108	Howitzer, 240mm, SP*, T92
Main Armament	Dual 40mm Gun, M2A1	90mm Gun	155mm Gun	175mm Gun	105mm How., T96E1	155mm How., T97	8-in How., T89	240mm How., M1
Ammunition, Rounds	480	29	20	——	102	30	10	——
Gun Control	Elec. Hydraulic	Manual	Hydraulic Servo	Hydraulic	Hydraulic	Hydraulic	Hydraulic Servo	——
Frontal Armor	½" at 56°	No Armor	¾" at 57°	——	½" at 55°-81°	½" at 55°-81°	¾" at 57°	1"
Engine	6 Cyl. Air Cooled Supercharged	6 Cyl. Air Cooled	12 Cyl. Air Cooled	AOI 628-3	6 Cyl. Air Cooled Supercharged	6 Cyl. Air Cooled Supercharged	12 Cyl. Air Cooled	Ford, GAF-C, V-8
Horsepower/Rpm	500/2800	207/3175	810/2800	370/	500/2800	500/2800	810/2800	500/2600
Horsepower/Ton	20.8	24.3	18.8	12.5	19.2	16.7	16.6	8
Transmission	Cross Drive	Cross Drive	Cross Drive	XTG-410	Cross Drive	Cross Drive	Cross Drive	Torqmatic
Suspension	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar	Torsion Bar
Track Width, in.	21	20 (Pneu Tires)	23	——	21	21	23	——
Fuel Capacity, gal	140	33	350	——	176	151.5	350	——
Weight, lb	48,000	15,400	94,100	59,200	52,500	60,000	97,000	127,500
Ground Pressure, psi	8.99	4.25	11.2	——	8.45	9.2	10.7	12
Dimensions, in Length (w/o gun) Width Height	 229 129 112½	 174 98 86½	 325 140 140	 —— —— ——	 215 128¼ 118	 240½ 123¾ 134	 325 140 134	 —— —— ——
Ground Clearance, in	17¼	12½	18½	——	17¼	19	18½	——
Max. Vert. Obstacle, in	28	18	42	——	30	36	42	——
Max. Trench Cross, in	72	60	95	——	72	72	96	——
Speed, mph	45	28	31	——	41	35	30.5	20
Cruising Range, mi	100	140	170	——	109	76	170	50
Turning Radius	Pivot	Pivot	Pivot	Pivot	Pivot	Pivot	Pivot	Pivot
Max. Slope, %	60	60	60	60	60	60	60	60
Max. Tractive, Effort, lb	40,000	14,250	60,000	——	53,500	42,050	60,000	105,000
Crew	6	3	6	5	5	5	6	8

* SP denotes self-propelled.

4-5.3 VIGILANTE "B"

The Vigilante "B," shown in Fig. 4-15, is a track-laying lightweight, lightly armored antiaircraft weapon designed specifically for defense against high-speed, low flying aircraft. It is amphibious, air-transportable in tactical aircraft, and airdroppable using conventional airdrop techniques. The main armament is a 37mm Gatling-type weapon, comprised of six separate barrels, individually loaded from an automatic, hydraulically operated, loading mechanism and capable of being fired in bursts of from 1 to 48 rounds at very high, antiaircraft cyclic rates. A push button selector allows the cyclic rate to be varied from the "very high rate," used for antiaircraft fire, to a "low rate" suitable for the weapon's secondary mission of providing antitank defense and close-in direct support fire for ground operations. Gun direction, when used in its antiaircraft role, is by an integrated pulse-Doppler radar-fire control system, mounted on the vehicle (Ref. 13).

The vehicle is 200 inches long, 108 inches wide,

Figure 4-6. *Twin 40mm Self-Propelled Gun, M42A1—1951 (APG A74324)*

and 82 inches high (with weapon and antenna in stowed position), has a gross weight of approximately 9.2 tons (airdrop weight of about 8.4 tons), a ground clearance of 14 in., and a ground pressure of about 5.3 psi. It is equipped with a flat track, torsion bar suspension, and a suspension lockout for greater stability when firing. A controlled differential provides steering and braking functions during high-speed operations on land, while a pivot steering system provides steering functions for water operations and precise maneuvering on land (Ref. 14).

SECTION II TACTICAL VEHICLES

4-6 GENERAL DISCUSSION

Tactical vehicles are generally defined as vehicles designed and manufactured specifically to meet the severe requirements imposed by combat and tactical operations in the field. Whereas combat vehicles are defined (par. 4-1) as vehicles designed to perform specific functions in combat, tactical vehicles are designed to support the tactical play of the operation. Tactics is that branch of the military art that deals with the arranging,

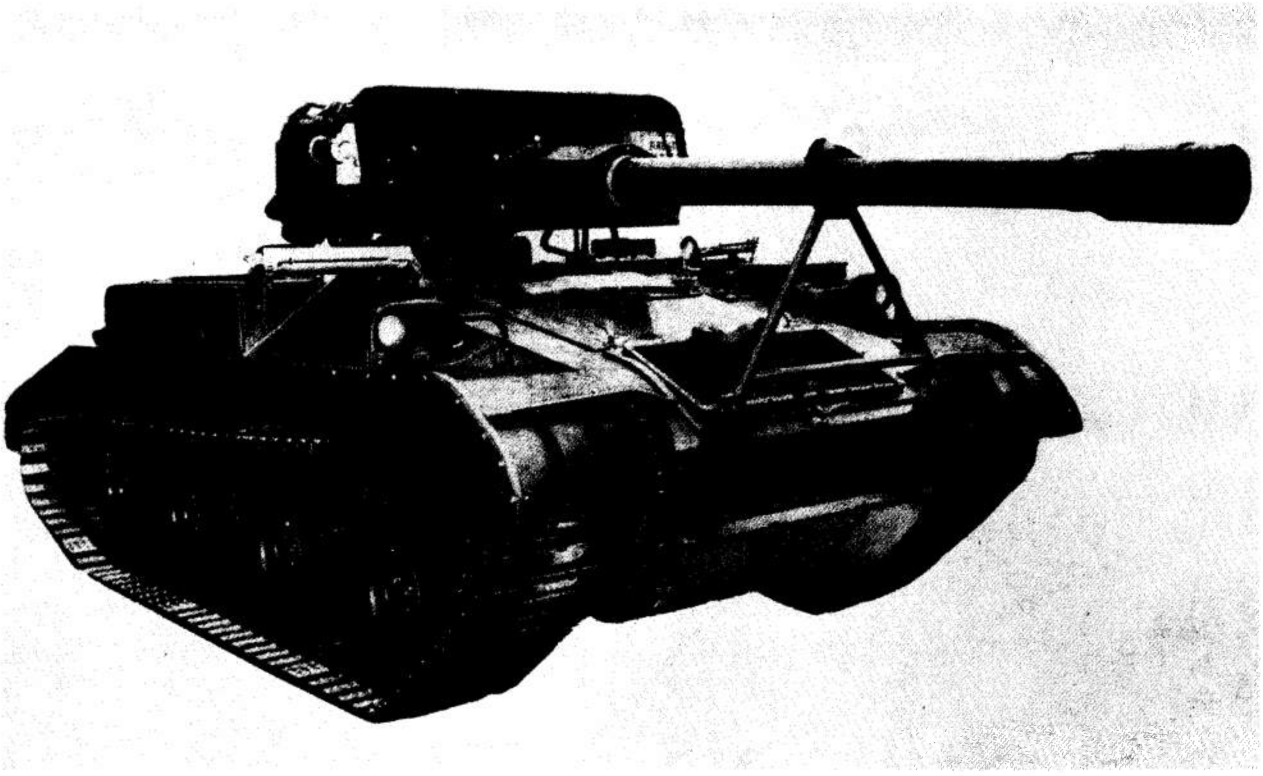


Figure 4-7. 90mm Self-Propelled Gun, M56 (SPAT)—1959 (DA 59401)

positioning, and maneuvering of the forces in contact, or near contact, with the enemy, and the maneuvering and positioning of material and supplies in support of the forces in contact, so as to attain an objective in a campaign or battle, to achieve some immediate advantage, or to ameliorate a disadvantage (Ref. 2). Since the function of tactical vehicles is to support the combat vehicles, it is imperative they have sufficient mobility to keep pace with the combat vehicles. Like combat vehicles, they are designed to have exacting military characteristics.

To meet the varied demands of the many tactical situations, different types of tactical vehicles are needed. Some are fully tracked for improved cross country mobility, while many are wheeled for greater speed over favorable terrain and for greater reliability and economy of manufacture. Many are designed for airborne operations, while amphibious capabilities make them mobile in spite of water barriers. As designers of military equipment strive for versatile vehicles, current tactical vehicles are fully tracked, lightly armored, amphibious, and lightweight for airborne operations.

4-7 TRACK-LAYING PERSONNEL AND CARGO CARRIERS

4-7.1 PERSONNEL CARRIERS

Since tactical operations relate specifically to the movement of troops and material, it is natural that tactical vehicles should take the form of personnel and cargo carriers. Their general characteristics vary in accordance with their primary mission. Vehicles intended primarily to add mobility to the infantry in the zone of combat are high-speed, armored, track-laying vehicles designed to provide protection from small arms fire and overhead artillery bursts. The interior of the armored body is usually heated and ventilated so that the personnel will travel in relative comfort. Large, quick-opening doors provide for rapid access and deployment. Light armament is sometimes mounted on the vehicle to provide air defense and firepower to support the deploying infantry. A pintle or towbar at the rear permits the vehicle to tow a cargo trailer, artillery weapon, or a disabled vehicle. When not carrying personnel, the vehicle is well suited for carrying ammunition and other cargo. Vehicles of this type are known variously

Figure 4-8. 155mm Self-Propelled Gun, M53—1952 (APG A77536)

as armored infantry vehicles, personnel carriers, armored utility vehicles, and tracked personnel carriers. Typical infantry-tracked carriers are shown in Figs. 4-16 through 4-18 while Fig. 4-39 shows the M113 personnel carrier compared to two current tractors.

4-7.2 CARGO CARRIERS

Cargo carriers and cargo tractors replace the old, slow moving, tracked prime movers that were formerly used to tow heavy artillery and heavy ammunition trailers. Since much of the artillery in the modern army is self-propelled, the need for artillery tractors is rapidly disappearing. The greatest need, however, exists for a highly mobile ammunition supply. The cargo tractors are, therefore, designed to meet this need. They are available in several sizes, each capable of carrying a considerable load of ammunition. Being fully tracked they have excellent cross country mobility. When operating in favorable terrain, they are capable of towing a cargo trailer with a capacity equal to, or in excess of, that possessed by the cargo carriers.

4-7.3 GENERAL CHARACTERISTICS

The personnel and cargo carriers transport up to 13 men and are usually armed with a machine gun. Since there are several sizes of carriers, their weights and dimensions can vary appreciably from vehicle to vehicle. There is also a wide variation of speeds and cruising ranges. The cruising

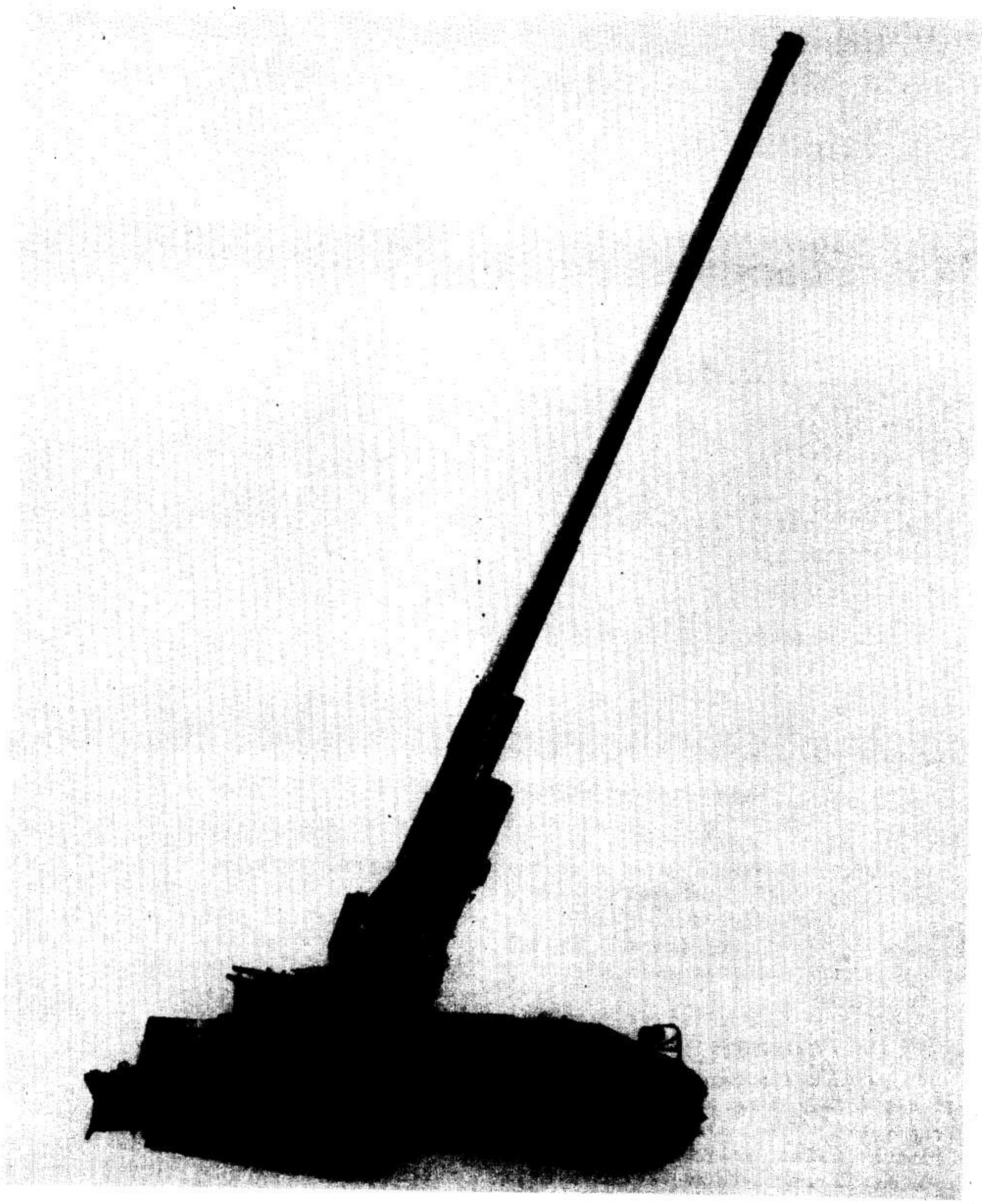


Figure 4-9. 175mm Self-Propelled Gun, T235E1—1960 (DA 64553)

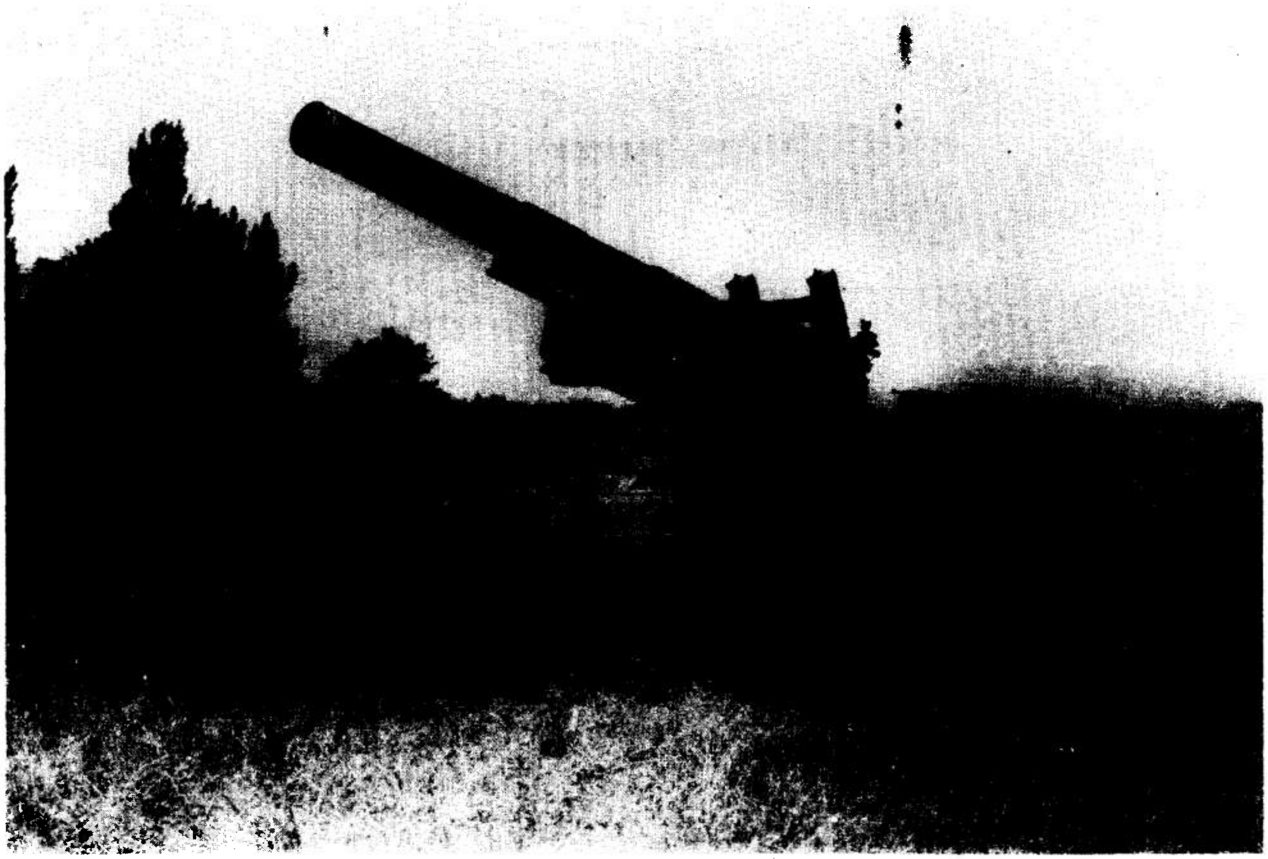


Figure 4-10. *8-Inch Self-Propelled Howitzer, T236—1958 (DA 56293)*

ranges are from 115 to 200 miles or more. Some carriers have an allowable governed speed up to 60 mph.

The design of these vehicles, both personnel and cargo, is based on the use of the latest concepts and components, where possible, of modern tank design. The engines, power trains, suspension systems, and tracks are quite similar to those of tanks of comparable size. The chassis and bodies are usually different because of the need for large uninterrupted interior spaces for personnel or cargo (Ref. 7).

Table 4-6 shows the characteristics of typical track-laying personnel and cargo carriers. Typical vehicles are shown in Figs. 4-16 through 4-19 and 4-24 through 4-26.

4-8 RECOVERY VEHICLES (FULL-TRACKED)

A full-tracked recovery vehicle is a self-propelled, armored vehicle, having boom and power winch equipment designed primarily to recover disabled tanks and other vehicles in combat areas. It may also be used for lifting engines, transmissions, and the like, during repair of disabled vehicles.

Recovery vehicles are now designed specifically for recovery missions whereas previously, they were merely converted tanks. However, many of their major components, such as tracks and suspension systems, power plants, and power trains, are standard tank components. They are usually equipped with fixed turrets and defensive armament, only, and utilize smoke grenades to screen recovery operations.

Table 4-7 shows some of the characteristics of a medium and a heavy recovery vehicle. The M51 heavy recovery vehicle is shown in Fig. 4-20.

4-9 AIRBORNE VEHICLES

The distinctive term *airborne* is given to a class of small, light vehicles to indicate their suitability for airborne tactical operations, and includes vehicles suitable for airdrop operations as

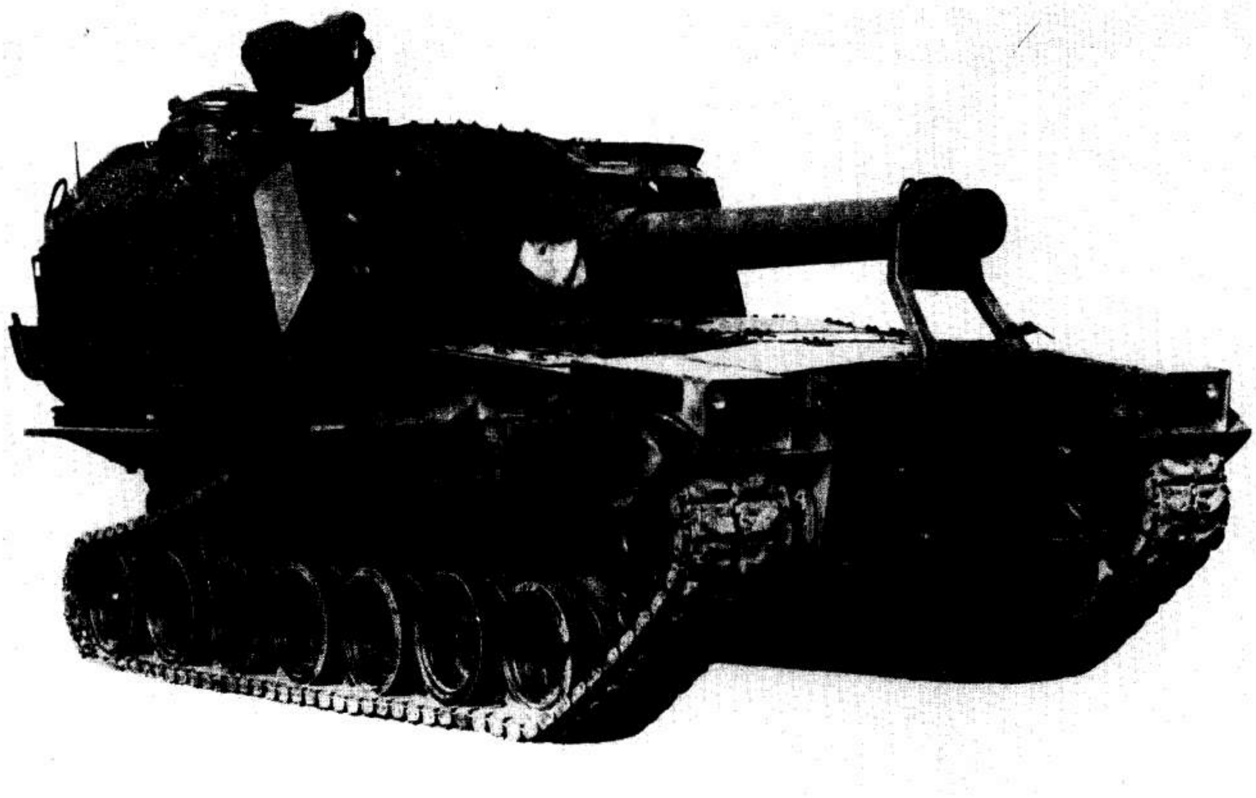


Figure 4-11. *8-Inch Self-Propelled Howitzer, M55—1959 (DA 59403)*

well as those suited only for air landed operations. The general requirements that must be satisfied by these vehicles are discussed in Chapter 3.

Early airborne operations used vehicles that were specially designed, or standard vehicles were modified, to fulfill some specific tactical purpose. The major difficulties encountered stemmed from the size and weight limitations imposed by both the aircraft and the airdrop equipment and technology of that day. The weight of early airborne vehicles was reduced almost to the point of flimsiness, while the size restrictions often made it necessary to partially disassemble the vehicle and even cut main frame members. This required that subsequent assembly and welding operations be conducted at the landing site, often under very unfavorable conditions. As military doctrine placed ever-increasing emphasis upon airborne operations, aircraft and airdrop equipment and techniques were developed to accommodate larger and heavier vehicles.

Paralleling the improvements in aircraft capacity were improvements in vehicle design. Highly efficient vehicles were designed to meet ever-decreasing weight and size specifications. The ultimate goal of this evolutionary trend will be reached when all military vehicles will have airborne capabilities.

Specifically, then, since airborne vehicles are basically those that can be satisfactorily accommodated aboard tactical aircraft, a particular vehicle that cannot qualify with present day aircraft may qualify in the future. Thus, airborne vehicles are rapidly becoming less of a specific type, and the airborne capability is becoming more of a general characteristic for the majority of automotive vehicles. The capacities of various aircraft used in airborne operations are given in Chapter 3. A partial list of representative vehicles that are suitable for airborne operations is given in Table 4-8.

4-10 AMPHIBIOUS VEHICLES

4-10.1 GENERAL DISCUSSION

An amphibious vehicle is one that is capable of operating satisfactorily on either land or water. It may be either a wheeled or track-laying vehicle, a walking vehicle, or have any other means, or

TABLE 4-5 CHARACTERISTICS OF REPRESENTATIVE RECONNAISSANCE VEHICLES (Ref. 7)

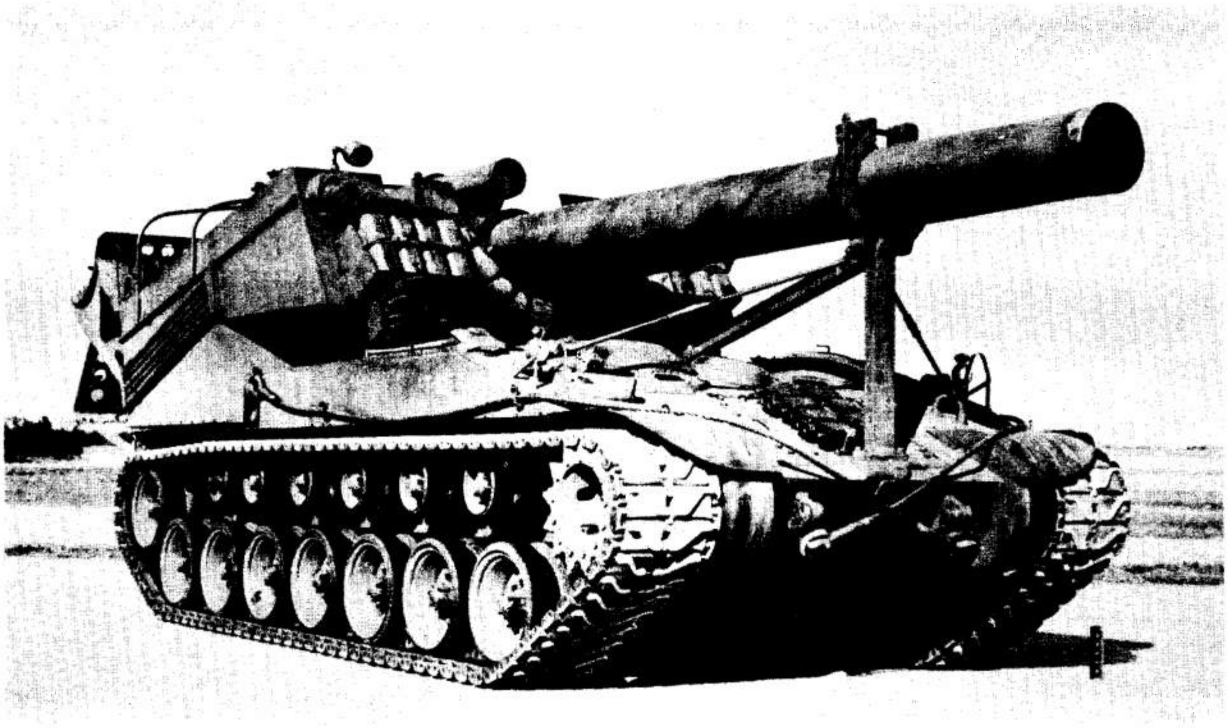
Nomenclature	Car, Armored, Light, M8	Car, Armored, Utility, M20	Car, Half-Track, M2A1	Carrier, Personnel, Half-Track, M3A1
Type	Wheeled, 6×6	Wheeled, 6×6	Half-Track	Half-Track
Loaded Weight, lb	17,200	15,650	19,600	20,500
Armament	37mm Gun 1 cal .30 MG 1 cal .50 MG	1 cal .50 MG	1 cal .50 MG 1 cal .30 MG	1 cal .50 MG 1 cal .30 MG
Engine	Hercules, JXD	Hercules, JXD	White, 160 AX	White, 160 AX
Horsepower/Rpm	86/2800	86/2800	127/3000	127/3000
Horsepower/Ton	10	11	12.9	12.4
Transmission	Sliding Gear	Sliding Gear	Constant Mesh	Constant Mesh
Track Ground Pressure, psi	——	——	11	11.6
Ground Clearance, in.	11½	11½	11-3/16	11-3/16
Dimensions, in. Length Width Height	 197 100 90	 197 100 91	 234¾ 87½ 100	 249-5/8 87½ 106
Max. Speed, mph	56	56	45	45
Max. Slope, %	60	60	60	60
Turning Radius, ft	28	28	29½	30
Cruising Range Loaded, mi.	250	250	210	210
Fording Depth, in.	32	32	32	32
Crew	4	6	10	8

combination of means, of propulsion. Since a wide variety of military vehicles qualify as amphibious under this definition, they are separated into three categories, namely: (a) floaters, (b) swimmers, and (c) true amphibians. First, however, a distinction must be made between amphibious vehicles and nonamphibious vehicles that have deep-water fording capabilities.

Fording is essentially a water crossing operation during which propulsion of the vehicle is achieved through contact with the ground beneath the body of water in the same manner that propulsion is achieved by the vehicle on dry land. While this makes it necessary for certain components to be waterproofed, provided with snorkeling devices, and otherwise equipped for submerged operations, it does not make the vehicle amphibious. The amphibian must be capable of swimming and maneuvering on the surface of the water and be otherwise adapted to water operations in addition to its ability to perform as a land vehicle. General requirements and design considerations, including amphibious capabilities, for military vehicles are discussed in Chapter 3.

4-10.1.1 Floaters

Floaters are conventional land vehicles made buoyant by the attachment of buoyancy devices. Early buoyancy devices consisted of collapsible rubber and fabric floats. Later ones were steel



240 MM HOWITZER MOTOR CARRIAGE T92

GENERAL

WEIGHT: (combat loaded) 127,500#   CREW: 8 men
WEIGHT: (less crew, stowage, & fuel) 123,000#
HORSEPOWER TO WEIGHT RATIO: 8 per ton
UNIT GROUND PRESSURE: 12 psi

ARMOR

Hull	Above Fender	Below Fender
FRONT	1"	
SIDES	½"	1"

VISION & SIGHTING EQUIPMENT
PERISCOPES: None
TELESCOPE: Panoramic M12 & Elbow M16AE1   QUANTITY: 1 ea.

ARMAMENT
PRIMARY: 240 mm Howitzer M1 in Mount T30
Traverse - 12° left and right; Elevation - 0° to 65°;
Location - rear
SECONDARY: cal. .30 carbines

AMMUNITION
900 rounds cal. .30

RUNNING GEAR
SUSPENSION: Torsion bar - M26 components

ENGINE
Ford GAF-C, V-8
FUEL: 80 octane gasoline
NET HORSEPOWER: 500 at 2600 rpm
COOLING SYSTEM: liquid
OIL CAPACITY: 32 quarts

POWER TRAIN
CLUTCH: Torque converter
TRANSMISSION: Torqmatic
NO. OF SPEEDS: 3 ranges forward, 1 reverse
STEERING MECHANISM: controlled differential (M26 components)

ELECTRICAL SYSTEM
MAIN GENERATOR: 3600 watts
AUXILIARY GENERATOR: Homelite, 1500 watts

COMMUNICATION
NO. OF INTERPHONE OUTLETS: 4

PERFORMANCE
MAX. TRACTIVE EFFORT: 105,000#
MAX. SPEED: 20 mph
CRUISING RANGE: 50 miles
GRADEABILITY: 60%

Figure 4-12. ***240mm Self-Propelled Howitzer, T92—1946 (APG A29392)***

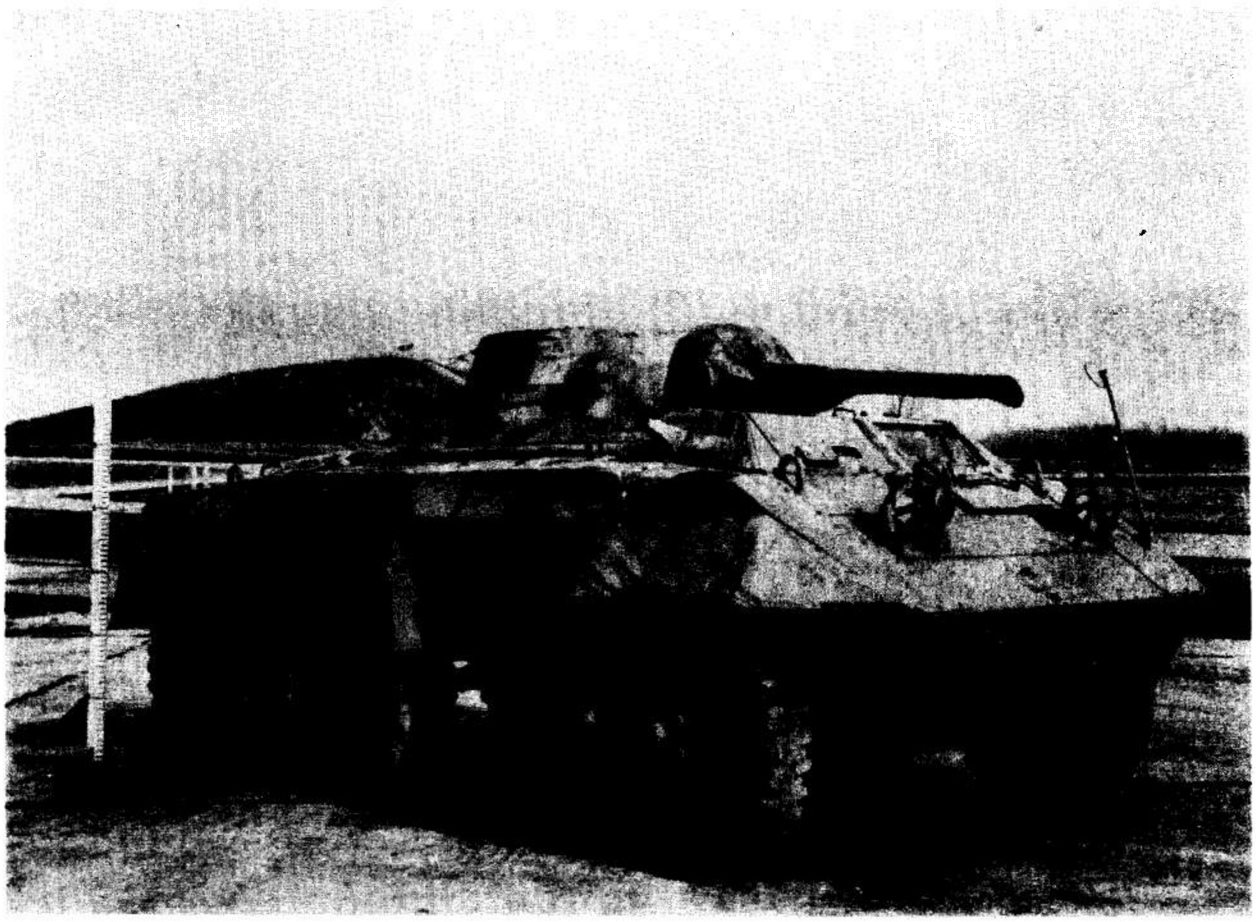


Figure 4-13. *Armored Car, M8E1—1945 (APG A20761)*

compartmented tanks that were attached to the sides and ends of the vehicle to be floated. Aluminum and other lightweight alloys replaced steel to reduce weight. The tanks were later filled with lightweight, closed cellular, foam plastic to maintain their buoyancy even when punctured by enemy fire (Ref. 15).

Even though the methods of attaching these floats to the vehicle are made as simple as possible, the entire procedure of making a large vehicle ready for an amphibious operation is slow and time-consuming. Immediately upon emerging from the water, the buoyant tanks are removed which requires additional time. With the tanks in place, the vehicles are extremely clumsy, since the tanks double the length and width of some vehicles and reduce the angles of approach and departure to as little as 15°. Floatation devices of this type weigh about 15% to 20% of the vehicle weight they are designed to support. Furthermore, floaters of the type just described, require some means of propulsion and steering while in the water. Propulsion is achieved by attaching auxiliary outboard motors to the assembly and steering is accomplished with rudders that are usually part of the float equipment.

4-10.1.2 Swimmers (Ref. 15)

Swimmers are a class of land vehicles that are specifically designed to have amphibious capabilities. The attachment of buoyancy devices and marine propulsion equipment to land vehicles, in order to transform them into floaters, is an amphibious expedient of a temporary nature. The swimmer requires no attachments or modifications to enable it to operate on the water. The vehicle body is fabricated as a watertight hull with sufficient buoyancy and stability to make the vehicle seaworthy on most inland waterways. Propulsion in the water is by means of the vehicle's wheels

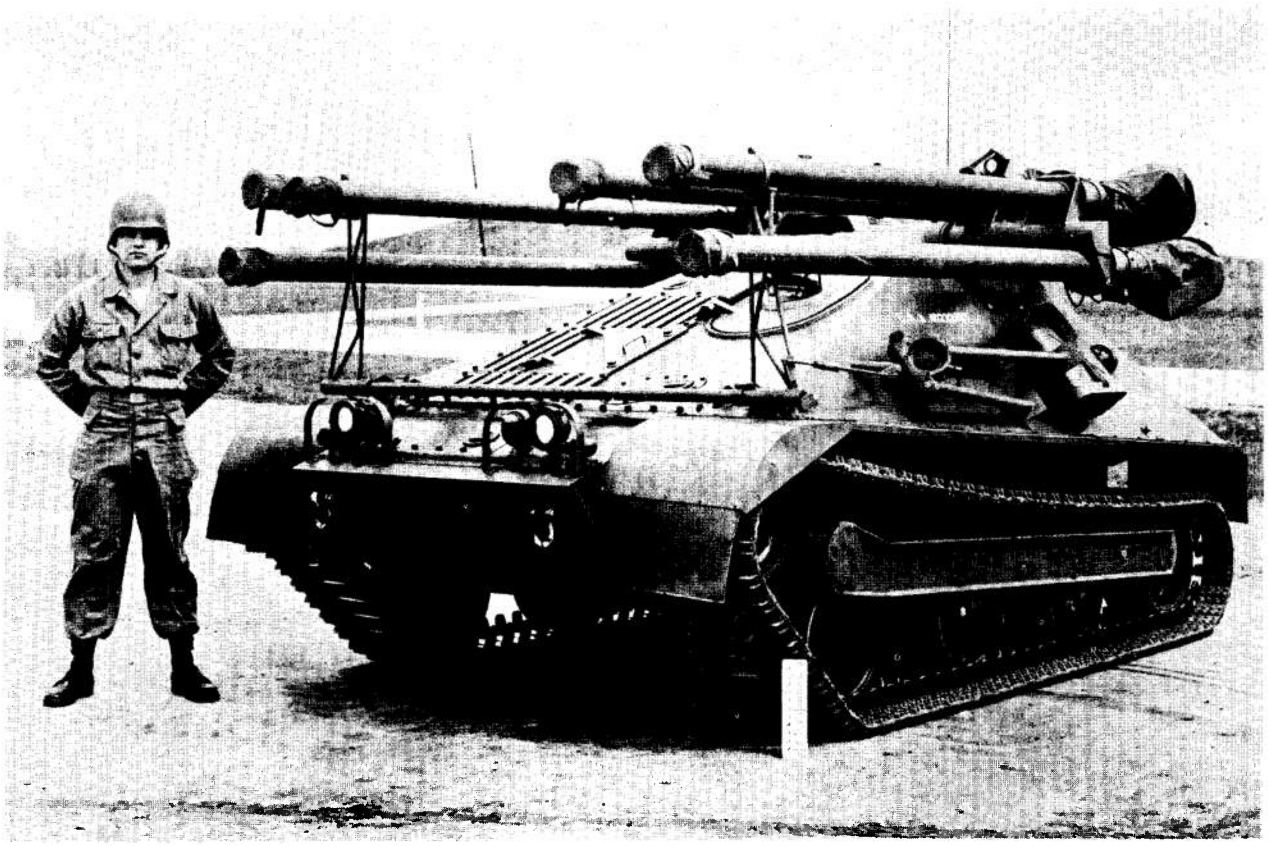


Figure 4-14. ***106mm Self-Propelled Multiple Rifle, M50 (ONTOS)—1953 (APG A88217)***

or tracks, or by means of a permanently mounted propeller. Walking vehicle concepts have been designed that utilize the feet for propulsion in water as well as on land. Hydrojet units are also used for water propulsion.

In general, the swimmers are far less clumsy than the floaters. They can enter the water immediately upon arriving at the water's edge; they have large angles of approach and departure, which reduce the problems of landing and launching; they are seaworthy; and their performance as land vehicles has not been greatly compromised by their marine requirements.

In their water performance, however, these swimmers do not rate particularly high when compared with a power boat of comparable capacity or horsepower. This is largely due to their inefficient hull shape, the resistance created by the protruding wheels or tracks, the water turbulence created by the projection of various suspension and power train components into the slip-stream under certain vehicles of this type, and due to the inherent inefficiency of wheels and tracks as marine propelling devices. Water speeds of amphibious vehicles in this category are below 8 mph with the average being about 3½ mph.

The complex problems of landing troops and equipment on foreign beaches led to the development of several different types of amphibians, particularly in the Pacific where the shoreline often consisted of sharp coral reefs, soft sandy beaches, or oozing mud flats, and the land side was tropical swamp and jungle. Furthermore, shore-to-shore operations versus ship-to-shore each contributed their specific demands. These varied requirements are filled by various tracked and wheeled amphibians. The tracked vehicles are generally referred to as LVT's (Landing Vehicle, Tracked), and nicknamed, Alligator, Water Buffalo, etc. Wheeled versions are generally called DUKW (pronounced duck) with the more recently developed being called Superduck, Drake, and BARC. Figures 4-16 through 4-18 and 4-24 through 4-28 show some of these vehicles.

The BARC is an overgrown version of the DUKW. It is 62 ft long, has four 10-ft diam.

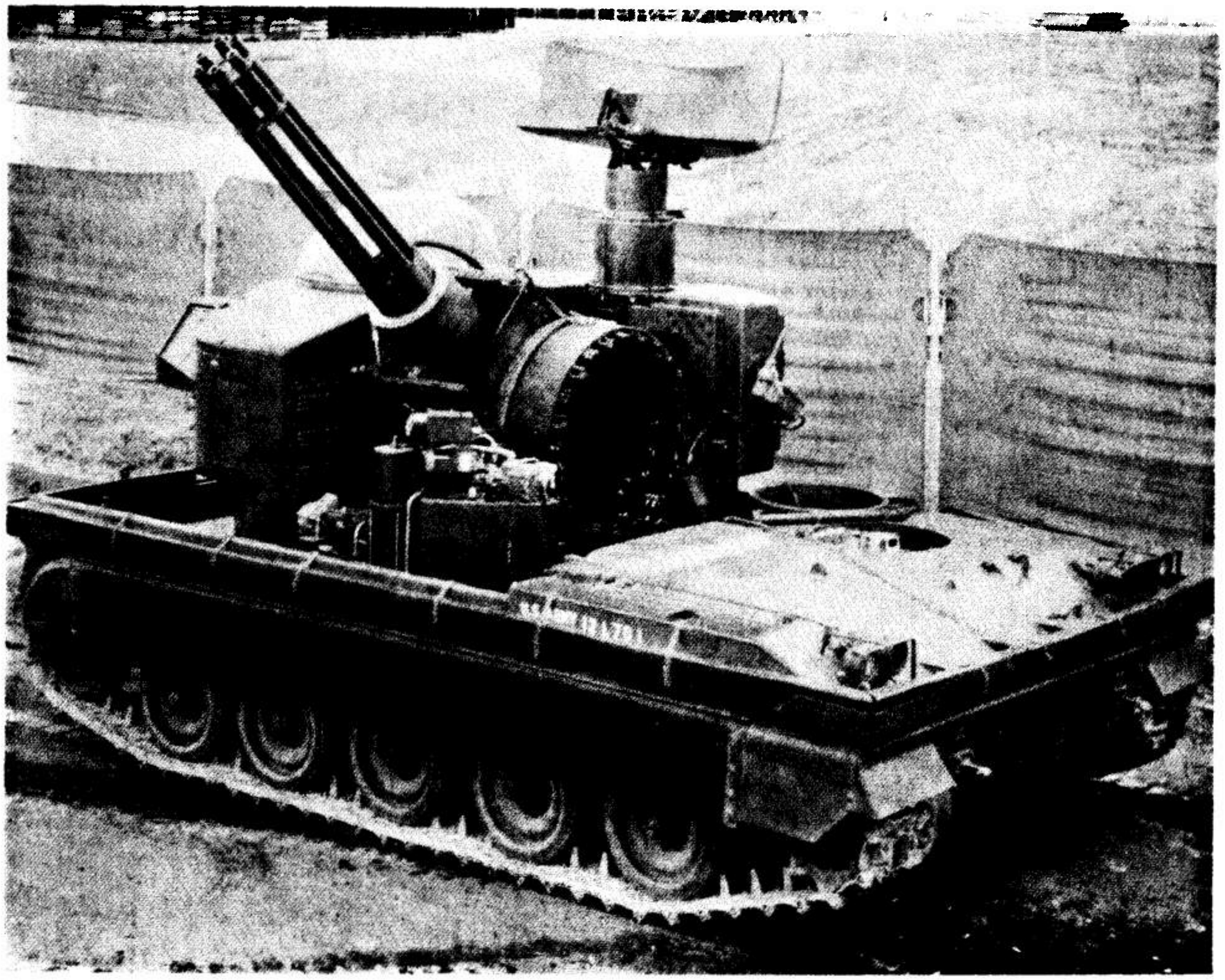


Figure 4-15. *Vigilante "B", 37mm Self-Propelled Antiaircraft Weapon Mounted on T249 S.P. Gun Chassis—1960 (Sperry Utah V-2830X 9-59)*

pneumatic-tired wheels and a bow that opens downward to provide a wide loading ramp. A 200-hp engine powers each wheel. This vehicle has a capacity of 100 tons which it can carry over water at 7 mph.

4-10.1.3 The True Amphibian

The concept of amphibious operations conducted during a nuclear war visualizes supply vessels lying offshore of an invasion beach with a minimum dispersal of 5 miles between vessels to minimize losses from enemy action and to prevent the creation of a profitable target for a nuclear weapon. With this much dispersion between vessels, and considering the large number of vessels required for a large scale amphibious operation, it has been estimated that the entire invasion force will occupy an area approximately the size of the state of New York, with some vessels as much as 350 miles from shore. Amphibious vehicles will travel from the shore to receive cargo from these vessels and transport it to the invasion beach, cross the beach, and continue on across country, or by way of road nets, if any are available, to widely dispersed supply dumps. It has been estimated that the average one-way trip, from shore to waiting ship, will be approximately 50 miles, or 100 miles for the water leg of the round trip. Because of these greatly increased water distances that the future amphibians will be required to travel, it is imperative that they be capable of high performance on water as well as on land.

Obviously, the floaters described in paragraph 4-10.1.1 cannot be considered for an operation of

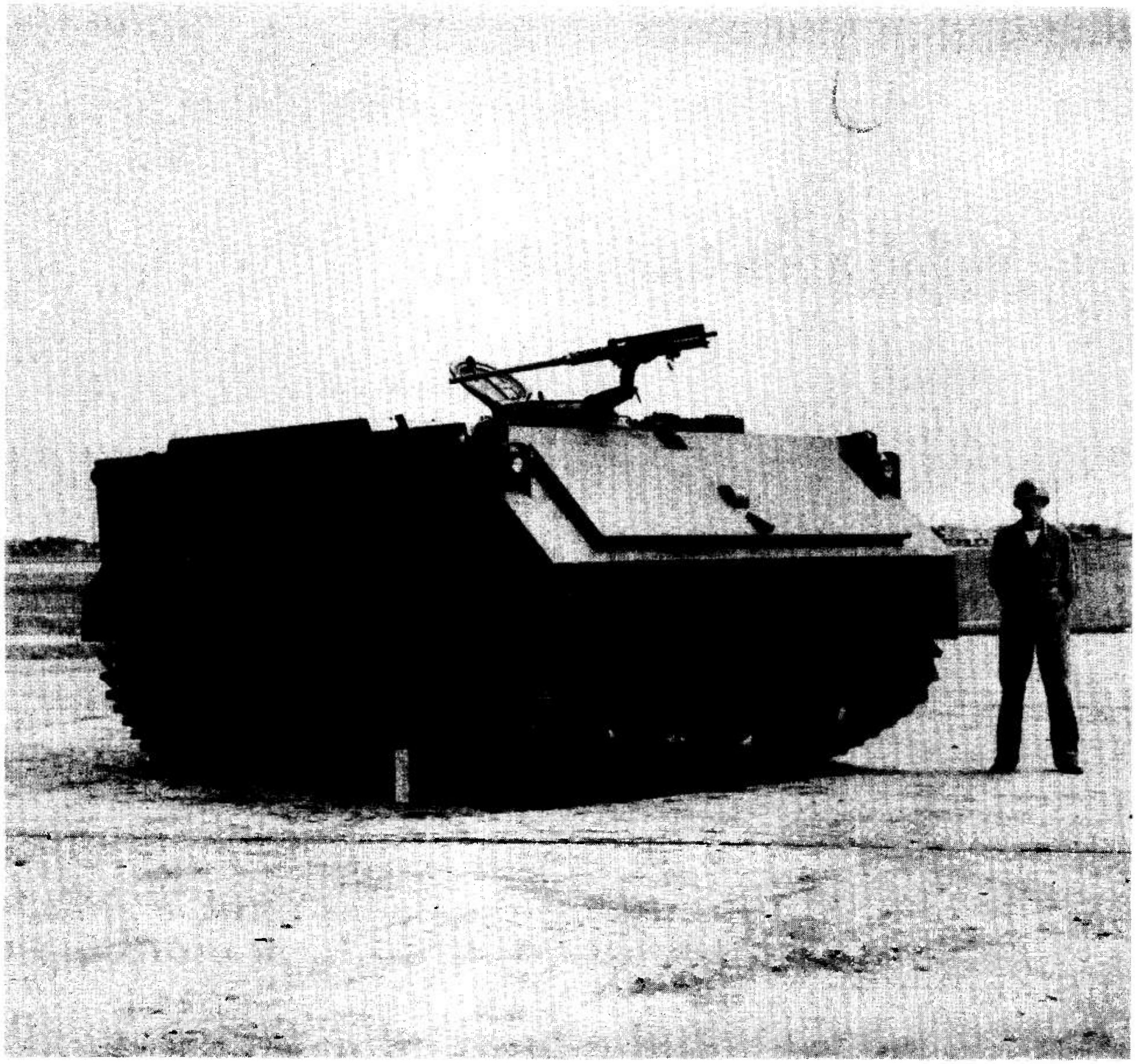


Figure 4-16. *Armored Infantry Vehicle, M59—1952 (APG A81804) (Armament, cal .50 machine gun. Land speed, 32 mph. Range, 120 miles. Crosses trench 66 in. wide. Climbs obstacle 18 in. high. Weight, 19 tons. Armored and amphibious.)*

this type. Their speed on water is much too slow, they lack sufficient water maneuverability, the conversion from land to water operation is intolerably time-consuming, and they are not sufficiently seaworthy for operating upon the open sea.

The swimmers, described in paragraph 4-10.1.2 are only somewhat better suited for this type of operation. They do not require the time-consuming version from land to water operation, and they do possess a higher degree of maneuverability, on relatively calm water, than do the floaters. Some swimmers, like the M113 and the LVTP5 (Figs. 4-17 and 4-18), are sufficiently seaworthy to operate in plunging surf, but their chief disadvantages are their slow speed and limited range (on water). Tidal currents in offshore areas vary in different parts of the world and at different times of the year, but currents of 5 to 13 knots are common; while rip tides, due to conflicting currents and shoals, are often much stronger and quite dangerous, particularly to underpowered, sluggish craft. Some swimmers are only capable of making 3-½ knots, in calm water, and none can do much more than 7 knots. Couple this speed capability

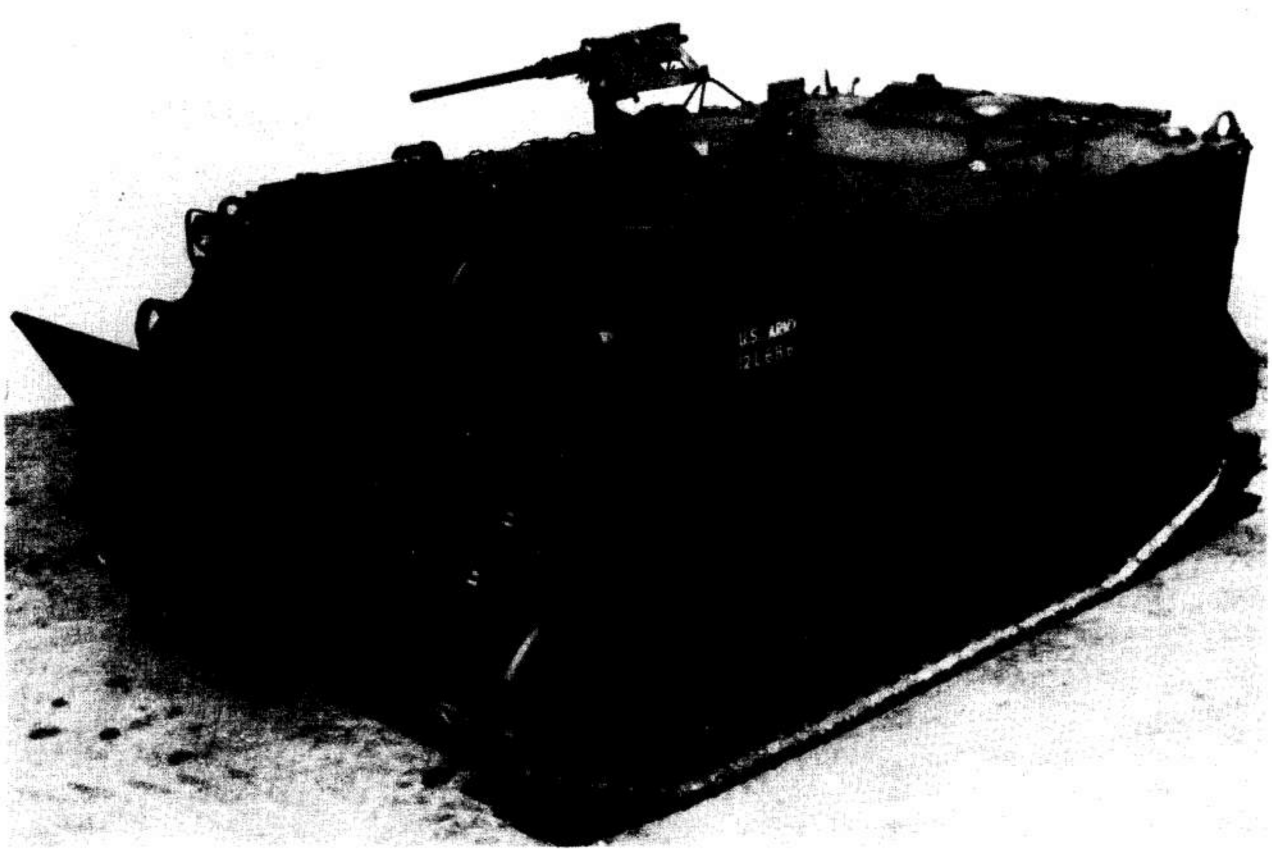


Figure 4-17. Armored Personnel Carrier, Full-Tracked, M113—1960 (DA 6482) (Armament, cal .50 machine gun. Capacity, 12 men plus driver. Weight combat loaded, 11.4 tons. Ground pressure, 7.25 psi. Ground clearance, 16 in. Crosses trench 66 in. wide. Climbs obstacle 24 in. high. Maximum gradeability, 60%. Cruising range on land, 200 miles. Land speed, 40 mph. Water speed, 3.7 mph. Dimensions: Length, 191-½ in., Width, 105-¾ in., Height, 86-½ in. Armored and amphibious.)

with an average range of about 50 miles and the unsuitability of the swimmer for the amphibious operations envisaged in the future becomes obvious.

The situation, therefore, requires the services of a third class of amphibious vehicles referred to here as "the true amphibians." These are vehicles that have marine capabilities comparable to those of high-performance water craft while retaining land capabilities equal to those of a purely land vehicle of comparable size. The hull is designed in strict accordance with all the principles of marine engineering. Wheels or tracks are retracted into wells in the hull for water operations (see Figs. 4-29 and 4-30). Designers of true amphibians are striving to achieve a maximum water speed of 25 to 35 mph. Two amphibians of this type are currently in the developmental stage. Each will be equipped with two engines, one a marine engine and the other a high-performance automotive engine. The one model will use an engine of 850 hp to give a water speed between 15 and 30 mph, and a 200- to 300-hp engine to meet the land requirements. The other model under development will use a 1250-hp engine to give the vehicle even greater water speed.

Another type of true amphibian, currently undergoing development, is the Flying Duck, shown in Fig. 4-31. This is an experimental version of a wheeled, amphibious, cargo carrier similar to that shown in Fig. 4-27 but equipped with hydrofoil wings beneath the hull. On water, the buoyant hull floats the cargo carrier, and the Flying Duck begins operation in much the same manner as a conventional boat. At approximately 5 mph, however, the action of the water flowing around the submerged hydrofoils develops a "lift," much in

Figure 4-18. *Landing Vehicle LVTP5—1960 (DA 64820) (Armament, machine guns. Speed, 27 mph on land, 7 mph on water. Land range, 180 miles. Crosses trench 102 in. wide. Climbs obstacle 36 in. high. Weight, 35 tons. Armored and amphibious.)*

the same manner as an airplane wing develops "lift" in a rapidly moving airstream. At 13 mph, this lift is sufficient to raise the entire hull out of the water. With the hull completely out of the water, more than 60% of the drag normally associated with boats is eliminated, thus, permitting a tremendous increase in speed.

The experimental Flying Duck, powered by an 860-hp gas turbine engine, attains a water speed of 50 mph with a gross load of 26,000 lb. At this speed, the hull is about 4 ft above the water while the hydrofoils travel about 30 in. below the surface. It can maintain level, high-speed flight in choppy seas of 4-ft high waves. For land operations, the hydrofoils are retracted and the vehicle proceeds on its wheels in the same manner as does a conventional 6×6 cargo truck (Ref. 16).

4-11 TRANSPORT VEHICLES

A transport vehicle is a wheeled vehicle primarily intended for personnel and cargo carrying in tactical situations, but excludes the combat vehicle (Ref. 2). It can also serve as a prime mover to tow a trailer or other towed equipment. Military transport vehicles are closely connected with commercial motor trucks, and are, therefore, one of the best known of modern machines. They are tactical vehicles in every sense, and, as such, are required to operate in forward combat zones, communication zones, and in the zone of the interior. The severe requirements imposed by operations in the combat zone bring about the greatest distinctions between the military transport vehicle and its commercial, civilian counterpart. As these requirements were better defined, these distinctions became greater, until the divergence between military and commercial transport vehicles is now an important consideration in the logistics of military preparedness as regards transport materiel.

Current trends in transport vehicle design (Refs. 16, 17, 18) include reduction of weight for

TABLE 4-6 CHARACTERISTICS OF TYPICAL TRACK-LAYING PERSONNEL AND CARGO CARRIERS (Ref. 7)

Nomenclature	Vehicle, Infantry, Armored, T18E1	Personnel Carrier, M113	Cargo Tractor, M8E2
Engine	6 Cyl AOS 895-4	Chrysler, V-8, A710B	6 Cyl AOS 895-3
Horsepower/Rpm	295/2660	215/4400	363/2800
Horsepower/Ton	15.5	21.5	14.1
Transmission	Cross Drive	TX-200-2X and Controlled Differential	Cross Drive
Suspension	Torsion Bar	Torsion Bar	Torsion Bar
Track Width, in.	21	15	21
Fuel Capacity, gal.	150	80	225
Weight, lb Empty Loaded	 38,100 42,000	 20,000 22,800	 51,700 66,700
Towing Capacity, lb	13,000	———	32,000
Dimensions, in. Length Width Height	 204½ 112 119¾	 191½ 105¾ 86½	 265⅛ 130½ 120
Cargo Space, in. Length Width	 11 passengers	 12 passengers	 150 100
Ground Pressure, psi	8.3	7.25	9.8
Ground Clearance, in.	17	16	19¼
Max. Vert. Obstacle, in.	18	24	23
Max. Trench Cross, in.	66	66	84
Speed, mph	46	40	40
Cruising Range, mi.	180	200	250
Turning Radius	Pivot	Pivot	Pivot
Max. Slope, %	60	60	60
Max. Tractive Effort, lb	32,000	———	40,000
Crew	1	1	2

airborne operation and increased amphibious capability. In part, the weight reduction is being accomplished through the use of aluminum instead of steel wherever possible. To increase the amphibious capability, many of the new vehicles are designed with sealed, unitized hulls. This enables vehicles to float in deep water. Propulsion in water is accomplished by driving the wheels, by auxiliary outboard motor, by hydrojets, or by propellers driven by the main power plant.

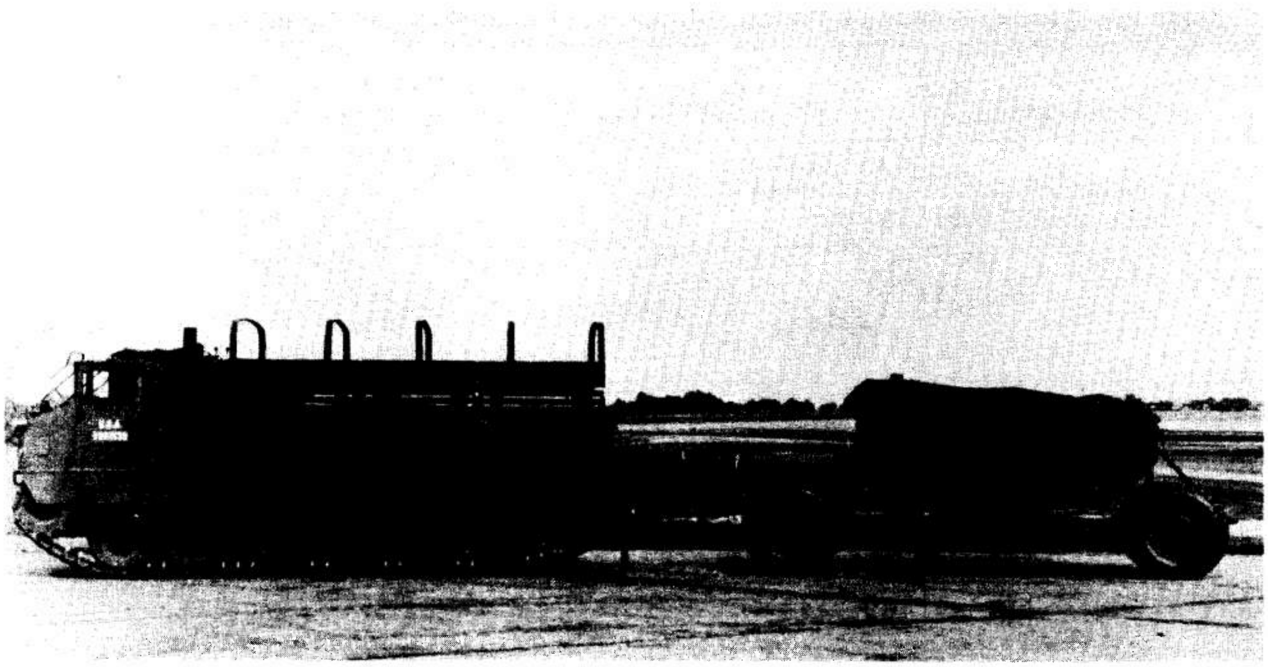


Figure 4-19. *Cargo Tractor, M8E2, Towing 75mm Gun, T83—1951 APG A72034)*

TABLE 4-7 CHARACTERISTICS OF TYPICAL RECOVERY VEHICLES

Nomenclature	Recovery Vehicle, M74	Recovery Vehicle, M51	Recovery Vehicle, T88
Armament	cal .30 and .50 MG	cal .50 MG	cal .50 MG
Engine, bhp/rpm	525/2800	980/2800	980/2800
Horsepower/Ton	11.2	16.35	17.5
Transmission	Cross Drive XT 1400-2A	Synchromesh Gear	Cross Drive XT 1410-2
Suspension	Volute Spring Bogie	Torsion Bar	Torsion Bar
Track Width, in.	23	28	28
Weight, lb	93,750	120,000	112,000
Ground Pressure, psi	13.6	12.2	10.5
Max. Vert. Obstacle, in.	24	31 w/Pintle 36 w/o Pintle	42
Max. Trench Crossing, in.	90	109	103
Speed, mph	21	30	30
Cruising Range, mi.	100	150	222
Max. Slope, %	60	60	60
Crew	4	4	4

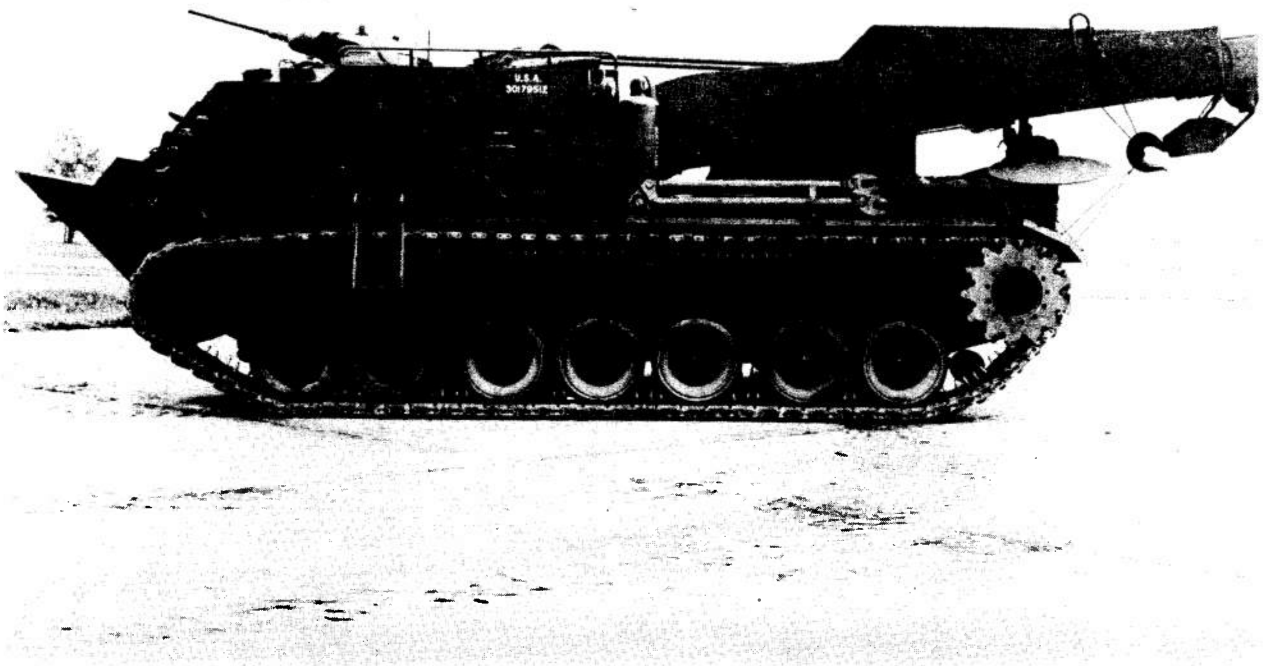


Figure 4-20. ***Heavy Recovery Vehicle, M51—1953 (APG A89031)***

In addition to the unitized hull, another new feature recently added to transport vehicles is independent wheel suspension for all wheels which reduces structural stresses by compensating for uneven terrain, improves stability and handling, increases traction on rough terrain, and permits greater speed.

A standard method of classifying military transport vehicles (trucks) is by the payload, by specifying the total number of wheels, and the number of wheels that are powered. The classification payload is the total weight of cargo and passengers, including crew, which can be safely imposed on the vehicle under cross country conditions. An on-highway payload is also frequently given. This is based on the maximum load capacity of the tires. Concerning the wheel designations, e.g., 6×6, 4×4, etc., the first figure indicates total number of wheels while the second indicates the total number of wheels that can be powered from the engine. Dual wheels are considered as one wheel.

Transport vehicles are attached to both service and combat organizations. The severe environmental and operational requirements of these vehicles dictate certain features. Traction is facili-

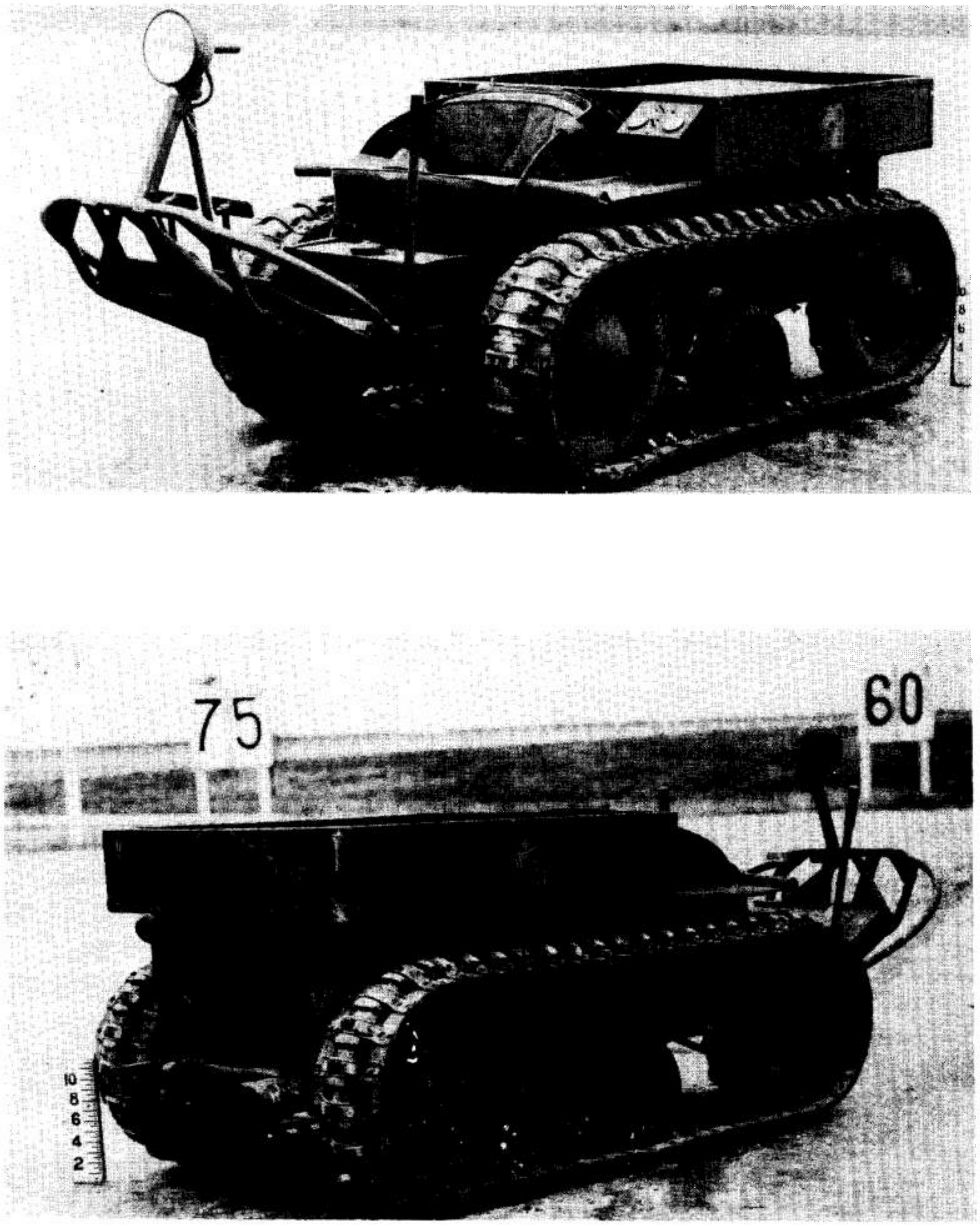


Figure 4-21. *Light Tractor, T37—1943 (APG 94747) (Left Front and Right Rear Views)*

tated in several ways, the most important is the all-wheel drive, giving traction to every wheel. Each vehicle has a front axle declutching device which permits the front-axle drive to be disengaged when operating on dry, hard-surfaced roads where the extra traction is not necessary. The tires are nondirectional mud and snow tread tires with a continuous center strip to provide smooth and long mileage operation on hard-surfaced roads and lugs to either side of the strip to grip soft terrain. In addition, trucks are provided with chains that can be attached to each wheel.

Some other features common to military transport vehicles include maximum ground clearance, minimum turning radius, high speed, and good braking action. Many have open cabs to permit antiaircraft defense or rapid egress in case of attack. In addition, the open cab provides improved ventilation during warm weather and better vision. Other features of these vehicles are adjustable

Figure 4-22. *Utility Truck, ¼-Ton, 4×4, M151—1954 (APG B561)*

windshields, radiators and headlight guards, towing hooks and pintles, special front and rear bumpers, and special spare-tire and fuel-tank mountings. Complete electrical suppression systems are standard on all transport vehicles (Ref. 19).

The XM410, shown in Fig. 4-33, is one of a new intermediate truck family which will also include four- and six-wheeled vehicles having 1- and 1½-ton capacities. It has an aluminum, integral body-frame hull-type construction that permits the vehicle to float in relatively calm waters even when fully loaded. Water propulsion is achieved by rotating the driving wheels, whereby a speed of about 3 knots can be obtained. For greater water speeds, an outboard motor is used. All wheels of this vehicle are independently suspended using torsion bars. All power train components, except the drive shafts to the wheels, are completely enclosed within the unitized hull. A novel feature of this vehicle is that the engine,

Figure 4-23. *Cargo Truck, ¾-Ton, 4×4, M37—1950 (APG A61333)*

transmission and transfer case are installed in reverse order, with the engine mounted behind the cab to facilitate the installation of future engines, such as multifuel types and gas turbines. The vehicle was designed expressly for lighter weight, better air-transportability, greater mobility, and greater versatility. It weighs approximately 5,000 lb less than the 2½-ton, 6×6 truck it is destined to replace, has approximately 30% greater payload-to-weight ratio, and approximately 50% better fuel mileage. The empty weight of this vehicle is 8,600 lb (Refs. 16, 17 and 18).

Military transport vehicles range in size from 1/4- to 10-ton capacity. A number of representative transport vehicles, along with some of their characteristics, are listed in Table 4-9. Representative vehicles are illustrated in Figs. 4-23, 4-32, and 4-33.

4-12 TRUCK TRACTORS, TRACTORS, AND TRANSPORTERS (Ref. 20)

4-12.1 GENERAL DISCUSSION

The nomenclature given to the vehicles discussed in this section are often confused and re-

TABLE 4-8 PARTIAL LIST OF REPRESENTATIVE AIRBORNE VEHICLES

Nomenclature	Illustration
Rifle, Multiple, 106mm, Self-Propelled, M50 (Ontos)	Fig. 4-14
Gun, 90mm, Self-Propelled, M56	Fig. 4-7
Armored Personnel Carrier, M113	Fig. 17
Tractor, Light, T37	Fig. 4-21
Tank, Light, T92	Fig. 4-1
¼-ton, 4×4, Utility Truck, M151	Fig. 4-22
¾-ton, 4×4 Cargo Truck, M37	Fig. 4-23

quire some definition. These definitions will be more easily remembered if the circumstances from which they arose are understood.

Prior to 1942, the Ordnance Corps was responsible for certain types of automotive vehicles, while the Quartermaster Corps was responsible for others. Most of the Ordnance Corps vehicles were of the track-laying type, and one particular group of these had the primary function of hauling towed-type artillery. This tracked artillery hauler was called a tractor.

The Quartermaster Corps, on the other hand, dealt almost exclusively with wheeled vehicles; and to it, the term "tractor" represented a short wheelbase truck whose primary function was to haul a semitrailer. In 1942, the Quartermaster Corps was relieved of its vehicle responsibilities, and these were transferred to the Ordnance Corps. To avoid confusion, the nomenclature of the wheeled semitrailer hauler was modified from "tractor" to "truck tractor," and the tracked artillery hauler retained its name of "tractor." Certain short-wheelbased, wheeled vehicles, equipped with the customary towing pintles, are used to tow both artillery and trailers. These are referred to as "prime movers." A few typical prime movers are listed in Table 4-10.

"Transporters" are composite vehicles consisting of one or more powered units connected to a "carrier" unit that is without power. The carrier unit is designed to carry some specialized load, such as a tank, a large gun, or a missile (see par. 4-12.4).

4-12.2 TRUCK TRACTORS

Truck tractors are a family of self-propelled, short-wheelbased, wheeled vehicles designed to tow, and partially support, a semitrailer by means of a fifth wheel type coupler. They utilize the same general components as do tactical vehicles of comparable size, but the capacities of these components (power plant, power train, suspension system, frame) are selected sufficiently large to accommodate the loads anticipated from the semitrailer. The major loads that act upon truck tractors are transmitted through the fifth wheel. Since the vehicle is designed to have proper weight distribution when pulling a loaded semitrailer, it has improper weight distribution when operating alone. Under these conditions, the vehicle will lack traction, because of the reduced weight on the rear wheels; and the front of the vehicle will be lower than the rear. This results in bad riding qualities and difficult steering when the truck tractor operates alone.

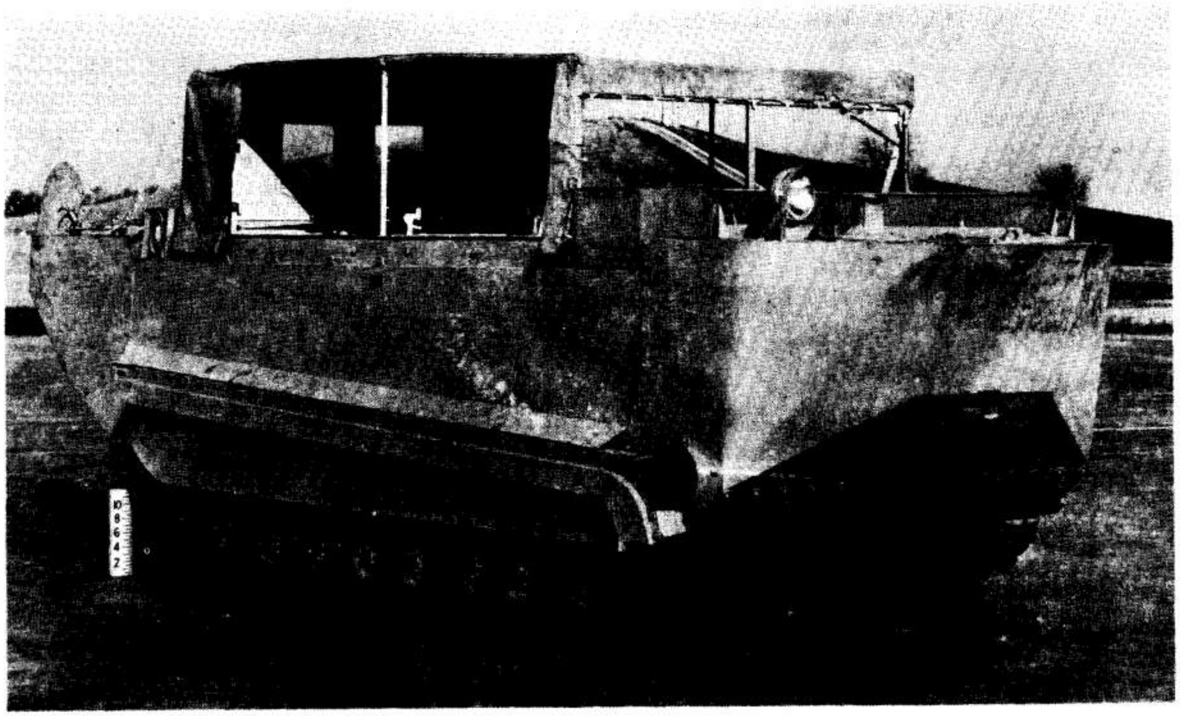
Truck tractors often become part of large transporters (par. 4-12.4) used to haul tanks, guns, or other heavy cargo. The cab is sometimes armored to give a degree of protection to the crew. Sometimes a short cargo space is provided between the cab and the fifth wheel which permits the truck tractor to haul ammunition while towing a gun or to improve its traction when operating without the semitrailer.

Some representative truck tractors and their characteristics are listed in Table 4-11. Figures 4-34 and 4-35 show two typical military truck tractors while Figs. 4-40 through 4-42 show other truck tractors coupled to carriers to form transporters for tanks and heavy guns.

4-12.3 TRACTORS

The general definition of a tractor defines it as an automotive vehicle designed for pulling or towing something; as, another vehicle, a sled, or some other load, by means of either a pintle hook or a fifth-wheel type coupling device (Ref. 2). In this respect, it can be either a wheeled or a tracked vehicle. Both types are used by the military forces and both are now the responsibility of the Army Materiel Command. In order to avoid confusion, the term "truck tractor" was introduced to apply to the wheeled tractor (see par. 4-12.2), and the term "tractor" when used alone generally refers to the track-laying variety.

Prior to World War II, military operations were not fully mechanized. Much of the mechanization that did exist consisted of tractor units



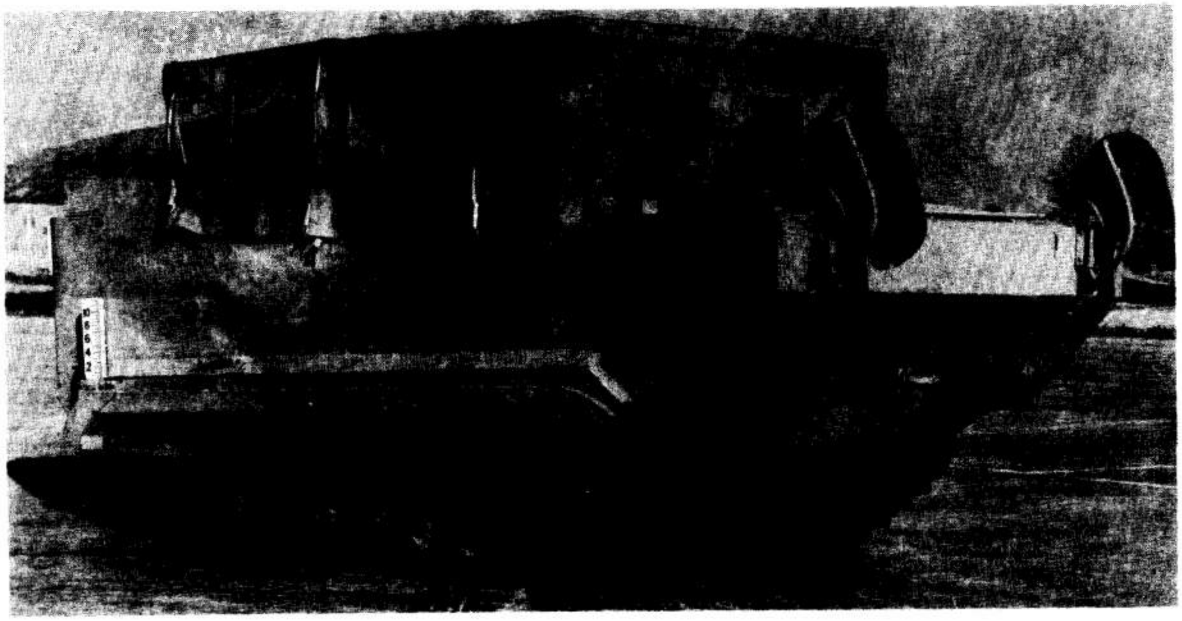


Figure 4-24. *Light Cargo Carrier, M29 (Amphibious Model)—1943 (APG 97030) (Gross weight, 5,971 lb. Ground pressure, 1.91 psi. Ground clearance, 11 in. Crosses trench 36 in. wide, Climbs obstacle 10 in. high. Maximum gradeability, 65%. Cruising range on land, 175 miles. Land speed, 36 mph. Dimensions: Length, 192 in., Width, 67-¼ in., Height, 71 in.)*

Figure 4-25. *Landing Vehicle, Tracked, Mark III—1945 (APG A23239A) (Armament, 1 cal. .50, 2 cal. .30 machine guns. Armor, ½" max., ¼" min. Crew, 2 to 7. Cruising range, 150 miles on land, 75 miles on water. Land speed, 21 mph. Net weight, 30,600 lb. Ground clearance, 19 in., Ground pressure, 8.75 psi. Dimensions: Length, 294 in., Width, 134 in., Height, 119 in.)*

which towed various materiel that had been modified for this purpose, from the designs originating in the days of animal-drawn transportation. Track-laying military tractors replaced the teams of horses and mules that formerly towed heavy artillery, ammunition trailers, and other heavy materiel in the forward zones. Tactical operations during World War II placed ever-increasing emphasis upon the importance of speed and a high degree of off-the-road mobility. This spurred the development of a family of high-speed tractors capable of speeds up to 40 mph. Table 4-12 lists some of the characteristics of a few representative tractors of this type. The M8E2 high-speed cargo tractor is shown in Fig. 4-19. Other typical tractors are shown in Figs. 4-36 through 4-39.

The T122 tractor, shown in Fig. 4-37, is armored, airdroppable, can ford 60 in. depths, and can be made sufficiently amphibious to cross inland waterways by the addition of a simple buoyancy curtain. As a cargo carrier it has a net weight of 13,300 lb, yet has a rated payload of 10,700 lb (Ref. 14). The XM474, shown in Fig. 4-39, is a basic chassis that was originally developed to provide cross country mobility for the Pershing missile but can be readily adapted to fulfill other requirements. It has a low silhouette, is airdroppable, and is unarmored. The net weight of this vehicle is only 11,900 lb, yet it has a rated payload of 12,000 lb (Ref. 21). These weights and payloads compare favorably with the M135 cargo truck (net weight, 12,000 lb; payload 5,350 lb) and the M54 cargo truck (net weight, 19,321 lb; payload, 10,350 lb). Figure 4-39 shows the XM474 with its two sister vehicles, the T122 high-speed tractor, and the M113 personnel carrier.

4-12.4 TRANSPORTERS

Transporters are composite vehicles consisting of one or more powered vehicles connected to a carrier. They are designed to carry specific equipment, such as tanks, heavy guns, and machinery. The carrier may be a trailer, a semitrailer, or a pallet suspended between two powered vehicles. An example of the pallet-type transporter is the T10 transporter for the 280mm "Atomic" Gun (Fig. 4-40). It consists of two independently powered and steered 4×4 units which pick up and carry the gun mount, recoil mechanism, and tube between them. The two units can operate independently of each other, but when carrying the gun the forward unit has control of throttle and brakes for both units. Each unit has an operator's cab for three crew members, with telephone communication between the front and rear cabs. High tensile, high carbon steels and aluminum are used extensively in this transporter to reduce weight and conserve steel. It has a gross weight in excess

Figure 4-26. Amphibious Cargo Carrier, T46 (M76, Otter)—1949 (APG A59497) (Armament, cal. .50 machine gun. Combat weight, 12,162 lb. Ground pressure, 2.1 psi. Ground clearance 16-¾ in. Crosses trench 60 in. wide. Climbs obstacle 18 in. high. Max. gradeability, 60%. Cruising range, on land 140 miles, on water 5 hrs. Land speed, 28 mph. Dimensions: Length, 188 in., Width, 98 in., Height, 108 in.)

of 85 tons and can transport the 50-ton, 280mm gun at 35 mph. With load, the T10 transporter is 84 ft, 2 in. long, yet can make right angle turns on roads only 28 ft wide. The angles of approach on the front and rear units are 32° and 29°, respectively.

Figures 4-41 and 4-42 show two typical transporters carrying tanks. The M15 transporter, shown in Fig. 4-41, is a semitrailer type, while the T8, shown in Fig. 4-42, is a pallet type similar to the T10, shown in Fig. 4-40. Although specifically developed to carry a heavy gun, the T10 can carry a heavy tank when equipped with a suitable pallet. Table 4-13 lists some semitrailers used as transporters.

4-13 TRAILERS AND SEMITRAILERS

"Trailers and semitrailers" is a category of either wheeled or tracked vehicles designed to carry material, supplies, or equipment and to be towed by a self-propelled motor vehicle. This section deals with all trailers and semitrailers except those used as part of transporters, which are discussed in a previous section.

Military trailers are classified into "Types," "Classes," and "Styles" (Ref. 22). Currently, there are two major types namely: Type I, Trailers and Type II, Semitrailers. The Type I trailers are divided into two classes (Classes 1 and 2), and Class 2 trailers are further subdivided into three styles (Styles a, b, and c).

4-13.1 TYPE I, CLASS 1, TRAILERS

Trailers belonging to this classification are known as "three-quarter trailers." Their distinguishing characteristics are their two-wheeled, single-axle construction and a design that balances about 85% of the load on this two-wheeled sus-

Figure 4-27. *Amphibious Truck, (DUKW) 2-½-Ton, 6×6—1954 (APG B593) (Gross weight, 10 tons. Speed: land 50 mph, water 6 mph. Cruising range: land, 240 miles, water 50 miles. Maximum gradeability 55%. Ground clearance, 11-½ in.)*

pension with only the remaining 15% supported by the towing vehicle. Three-quarter trailers are made in a variety of sizes and body styles ranging from general cargo and tank bodies (for fuel or water) to elaborate house trailers used by hospital units. When the load becomes appreciable for a single axle, a tandem axle is used. Towing is done by means of a tow-bar or tongue and lunette which attaches to the pintle of the towing vehicle.

4-13.2 TYPE I, CLASS 2, TRAILERS

This classification of trailers is applied to a category commonly known as "full trailers." Their distinguishing characteristics are the presence of supporting wheels at both ends and a towing tongue and lunette attached to the front truck which engages the pintle of the towing vehicle. Full trailers are made in a large variety of body

Figure 4-28. *Amphibious Truck, (Superduck) 4-Ton, 6×6, XM147E3—1958 (DA 54561)*

TABLE 4-9 REPRESENTATIVE TRANSPORT VEHICLES*

Model	Payload, tons		Weight, lb		Overall Dimensions			Axle Ground Clearance, in.	Engine				Maximum Speed, mph	Maximum Fording Depth, in.	Cruising Range, mi.	Transmission				Differential-Type	Suspension-Type	Service Brakes		Wheelbase, in.
	Cross Country	Highway	Empty	Fully Equipped with Cross Country Payload	Width, in.	Length, in.	Height, in.		Displacement, cu in.	Fuel Used	Cooling Medium	Max. Brake HP/rpm				Type	No. of Forward Speeds	No. of Reverse Speeds	Transfer Case			Actuated by	Type	
M38A1	3/8	5/8	2,665	3,865	60-5/8	138-5/8	73¾	8¾	134.2	G	Liq	70-4000	71	72	280	C	3	1	TS	C	SE	—	Hd	81
M37	¾	1	5,950[1]	7,800[1]	73½	189-3/8[1]	89¾	10¾	230.2	G	Liq	94-3200	55	72	225	C	4	1	TS	C	SE	—	Hd	112
M35	2½	5	12,880[1]	18,230[1]	96	274¾[1]	111-3/16	10-5/8	331.0	G	Liq	146-3400	58	72	300	C	5	1	TS	C	Bcp	Air	Hd	154
M211	2½	5	13,580[1]	18,930[1]	96	269¼[1]	1121/8	12	301.6	G	Liq	145-3400	58	72	300	Hy	8	2	[2]	C	Bcp	Air	Hd	156
M54	5	10	19,945[1]	30,295[1]	97	314¼[1]	116	11½	602.0	G	Liq	224-2800	53	72	214	C	5	1	TS	C	Bcp	Air	Hd	179
M125	10	15	32,550[1]	62,000	114	331½	129	15¾	844.0	G	Liq	297-2600	40	78	300	C	5	1	TS	—	Bcp	Air	Hd	181½
M274	½	½	925	1,925	49¾	118¼	49-1/8	8½	53.5	G	Air	15-3200	25	18	—	C	3	1	TS	N	N	Ma	Me	57
M422	3/8	½	1,700	2,550	61	107	59½	9¾	107.8	G	Air	55-3600	62	60	255	C	4	1	S	L	IE	Ma	Hd	65
M151	3/8	5/8	2,273	3,073	62¼	132	71	11½	141.5	G	Liq	71-3900	65	60	300	C	4	1	S	C	Co	—	Hd	85

ABBREVIATIONS

[1]—With winch
[2]—Two-speed reduction unit on transmission
Bcp—Bogie, constant paralleled arm (leaf springs)
C—Conventional
Co—Coil
G—Gasoline
Hd—Hydraulic
Hy—Hydramatic
IE—Independent elliptic
L—Locking
Liq—Liquid
Ma—Manual
Md—Multiple disk
Me—Mechanical
N—None
S—Single speed
SE—Semi-elliptic
TS—Two-speed

* SOURCE: *Automotive Industries*, Statistical Issue, Vol. 122, No. 6, March 15, 1960, pp. 150.

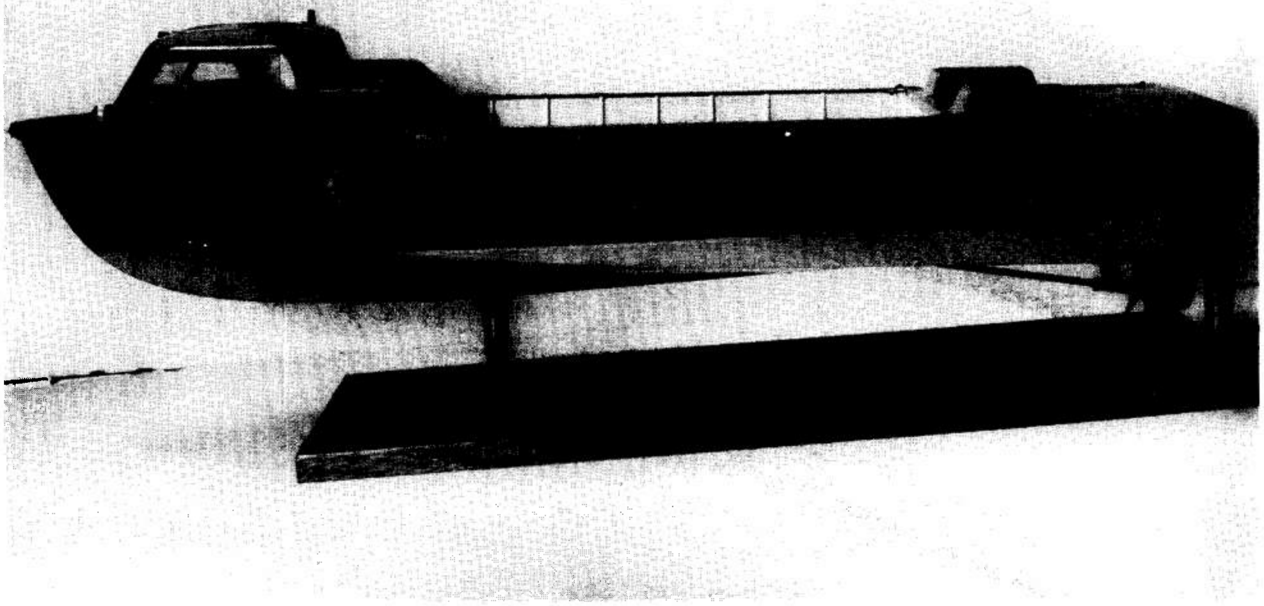


Figure 4-29. *High-Speed Amphibious Cargo Truck, 5-Ton, 4×4—1959 (DA 61562)*
(Wheels retracted for high speed water operation)

styles, in many sizes, and with both single and tandem axles.

4-13.2.1 Type I, Class 2, Style a

This classification is applied to a style of full trailer commonly known as "nonreversible," because it can be towed and steered by one end only. The front truck rotates about a fifth wheel assembly which has its lower ring mounted on top of the trailer truck and its upper ring mounted to the bottom side of the front chassis. Towing is accomplished through a lunette and towing tongue in the manner customary to full trailers.

4-13.2.2 Type I, Class 2, Style b

Trailers of this classification are full trailers that are reversible; that is, they may be towed and steered from either end. They are similar to the nonreversible trailers in construction and appearance except that both the front and rear trucks are mounted to the trailer chassis by means of fifth wheels. The towing tongue is detachable and may be attached to either truck. The trucks are equipped with lockout provisions which permit one truck to be locked against rotation when the other is connected to the towing vehicle.

4-13.2.3 Type I, Class 2, Style c

This classification applies to a semitrailer that has been converted into a full trailer by the addition of a dolly under the front end of the semitrailer to replace the truck tractor as the front-end support. A dolly is a short, two-wheeled trailer chassis with the lower ring of a fifth wheel assembly mounted on top of its frame. This lower ring engages the upper ring, mounted under the forepart of the semitrailer, to complete the converted semitrailer. Towing and steering is accomplished by means of a tongue and lunette assembly attached to the dolly.

4-13.3 TYPE II, SEMITRAILERS

This is the classification given to semitrailers. Semitrailers are defined as nonpowered vehicles having integral wheels at the rear, only, and de-

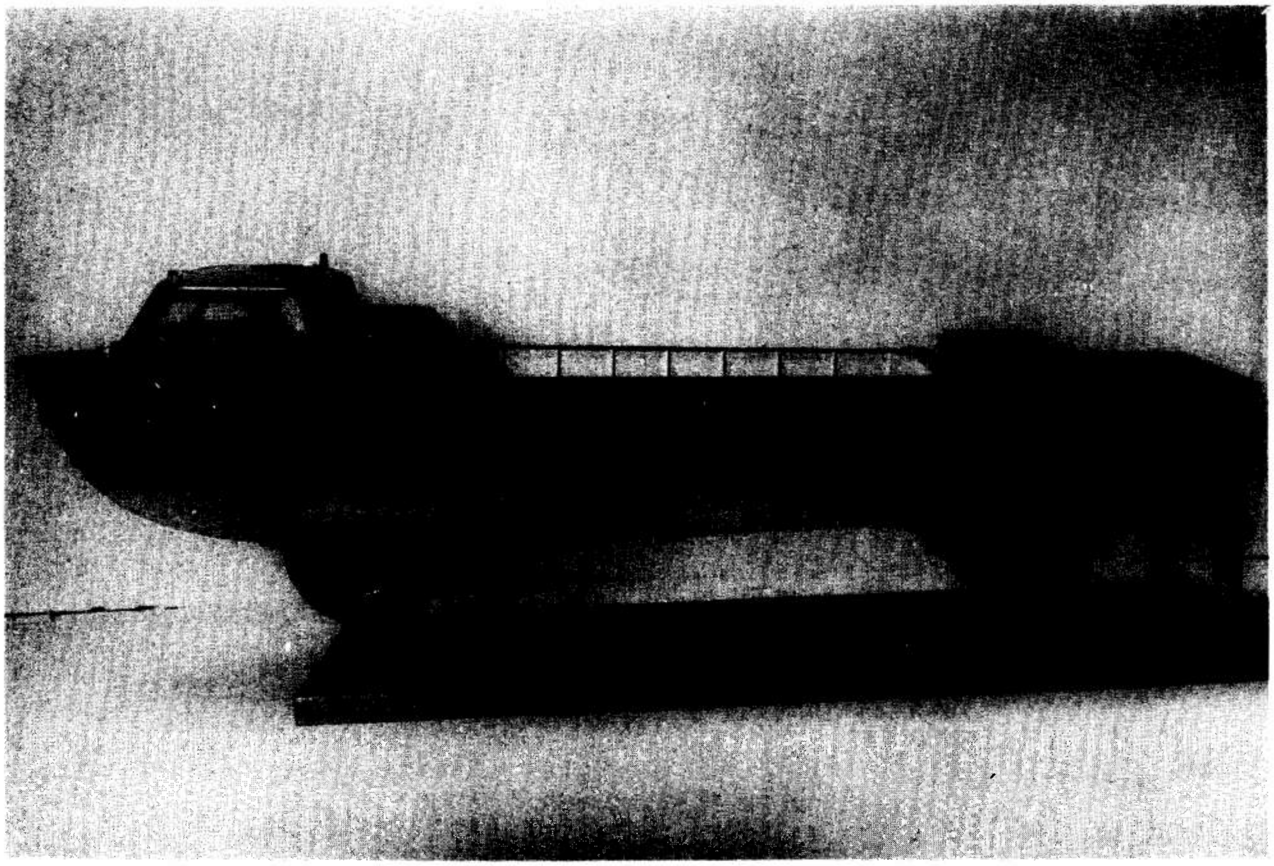


Figure 4-30. *High-Speed Amphibious Cargo Truck, 5-Ton, 4×4—1959 (DA 61561)*
(Wheels lowered for landing and land operation.)

signed to carry materiel, supplies, or equipment and to be towed by a self-propelled motor vehicle which also supports the front end by means of a fifth wheel coupling assembly. A landing gear under the vehicle supports the front end when the semitrailer is uncoupled from its prime mover and supports the vehicle when parked or while being loaded or unloaded. Most types of landing gear are retractible to prevent their being damaged while underway. Some types of landing gear are

TABLE 4-10 REPRESENTATIVE PRIME MOVERS (Ref. 20)

Model	Type	Wheel-base, in.	Tires	Net Weight, lb	Horse-power	Maximum Speed, mph	Use
M20	12T, 6×4	205	12.00×20 Dual	26,950	185	25	With 45T M9 Trailer Makes M19 Transporter
WWII T16	7½T, 6×6	185	12.00×24 Dual	30,000	159	32	For 155mm Gun
M125	10T, 6×6	171	14.00×24 Dual	32,550	270	43	For 75mm AA Gun
XM193	15T, 8×8	175	16.00×25 Dual	——	500	50	For 155mm Gun and 8-in. Howitzer

TABLE 4-11 REPRESENTATIVE TRUCK TRACTORS (Ref. 20)

Model	Type	Wheel-base, in.	Tires	Net Weight, lb	Horse-power	Maximum Speed, mph	Use
M26A1	12T, 6×6	173½	14.00×24 Dual	48,895	240	28	With M15 Semitrailer for 40T Transporter
T26E1	12T, 8×8	224½	14.00×24 Dual	53,825	500	—	With 60T Semitrailer
T28	8T, 6×6	168	12.00×24 Dual	36,265	290	—	With 25T Semitrailer
T46	25T, 6×6	186	18.00×29 Dual	66,100	500	—	With 75T Semitrailer
M275	2½T, 6×6	142	9.00×20 Dual	11,590	145	60	Like M58 Truck for 6T Semitrailer
M48	2½T, 6×6	154	9.00×20 Dual	11,430	145	60	Like M45 Truck for 6T Semitrailer
M221	2½T, 6×6	144	9.00×20 Dual	12,105	145	55	Like M209 Truck for 6T Semitrailer
M52	5T, 6×6	167	11.00×20 Dual	18,300	224	52	Like M61 Truck for 12T Semitrailer
XM123	10T, 6×6	171	14.00×24 Dual	27,600	270	43	Like XM121 Truck for 25T Semitrailer
XM194	15T, 8×8	175	16.00×25 Dual	4,700	500	50	Like XM192 for 25T Semitrailer
XM249 XM250	Front Unit Rear Unit	120 4×4	16.00×25 Dual	———	375	—	For T10 Transporters

equipped with steel wheels that permit moving the trailer when uncoupled from the towing vehicle.

4-13.4 GENERAL DISCUSSION (Ref. 23)

Both trailers and semitrailers may have various numbers and arrangements of wheels and may be stable or unstable when uncoupled from the prime mover or truck tractor. The design of these vehicles varies only slightly from their standard commerical counterparts. A semitrailer is easier to maneuver, steer, and brake than a full trailer of the same capacity. In the truck tractor-semitrailer unit, the weight of the semitrailer helps provide traction; whereas, in a prime mover-trailer unit, the weight of the prime mover alone provides the traction. The full trailer has an advantage over the semitrailer in that it can be more quickly coupled and uncoupled and is able to be moved independently, if necessary.

The advantages of using tractor trailer or semitrailer combinations instead of single units such as cargo trucks come from the flexibility and economy that their use can bring. Flexibility is achieved when one type of tractor or truck tractor may be used with several types of trailers or semitrailers. Economy is possible when one tractor is used with two or more semitrailers and the work is so scheduled as to have one, or more, trailers being loaded, or unloaded, while the tractor is hauling another. In this manner, expensive power equipment isn't standing idle while cargo is being discharged. Some disadvantages in the use of trailers and semitrailers in place of trucks, include the possibility of jack-knifing and uncoupling.

Figure 4-43 shows a typical two-wheeled cargo trailer. The appearance of military trailers and semitrailers does not differ materially from their civilian counterparts; therefore, profuse illustrations of these vehicles is unnecessary. A comprehensive discussion of trailers and semitrailers, particularly from the design standpoint, will be found in Ref. 22.

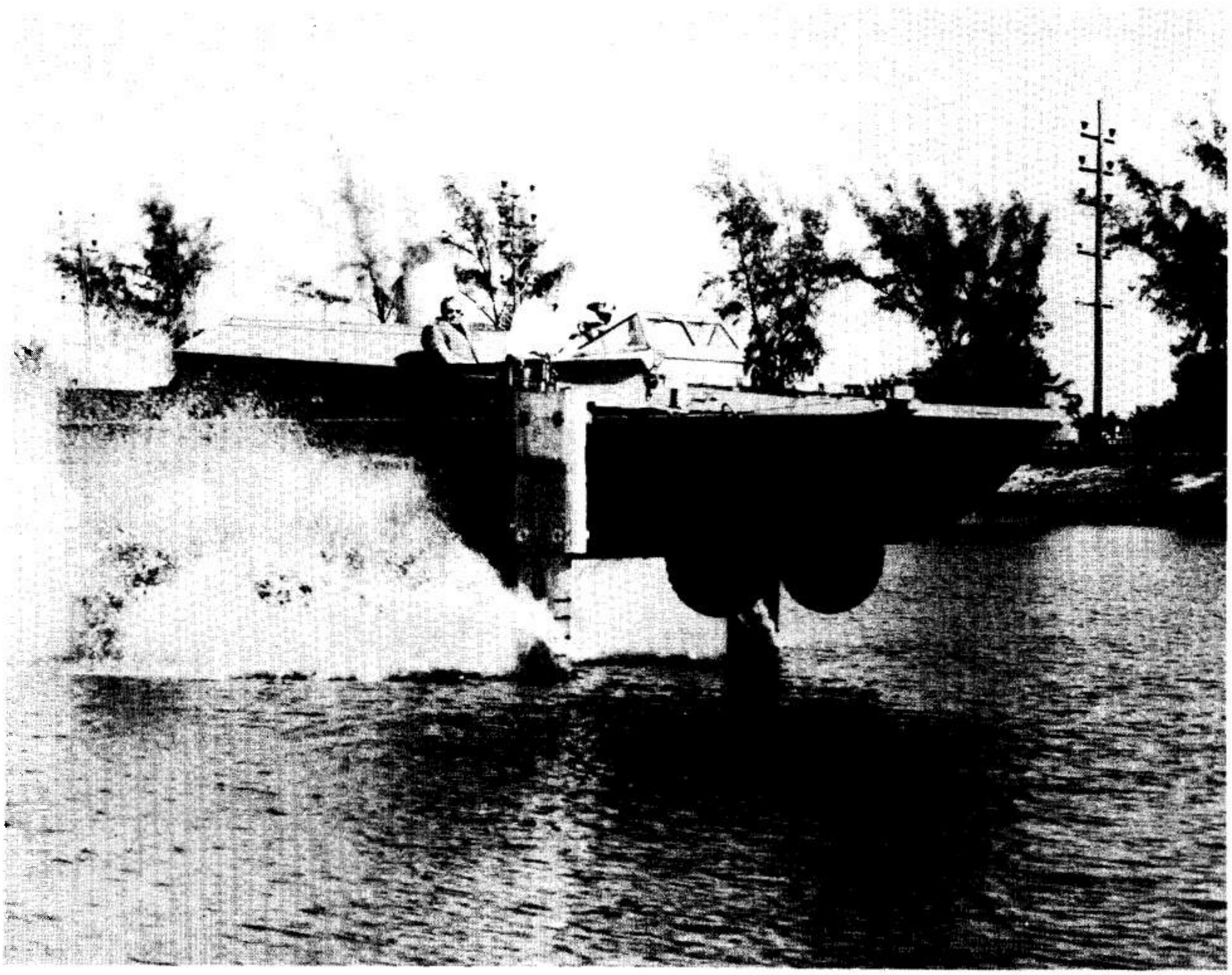


Figure 4-31. *The Flying Duck—Amphibious Cargo Truck-, 2½-Ton, 6×6*

SECTION III MISCELLANEOUS VEHICLES

Any structure that moves and is capable of carrying a load over land is considered a land vehicle. If it does not operate on rails and if it carries its own power plant, or if it is normally towed by a powered vehicle, it comes within the boundaries of this chapter. It need not even have wheels; it might be equipped with feet, or it might be a sled, or it might float a few inches above the ground on a cushion of air. Those vehicles that have a well-established design and a commonly accepted military use have been discussed in the preceding sections of this chapter. A wide range of varieties of vehicles remains, however, with which the vehicle designer is concerned. These are the miscellaneous vehicles which are divided into the following five categories:

1. Standard civilian vehicles.
2. Special equipment vehicles.
3. Special-purpose vehicles.
4. Miniature vehicles.
5. Novel types not yet evaluated or integrated into authorized tables of equipment.

4-14 STANDARD CIVILIAN VEHICLES

Many varieties of standard civilian vehicles, such as passenger cars, small and large trucks, and buses, are purchased by the military establishment for use in the zone of the interior to do the same tasks they would do in civilian life. These are

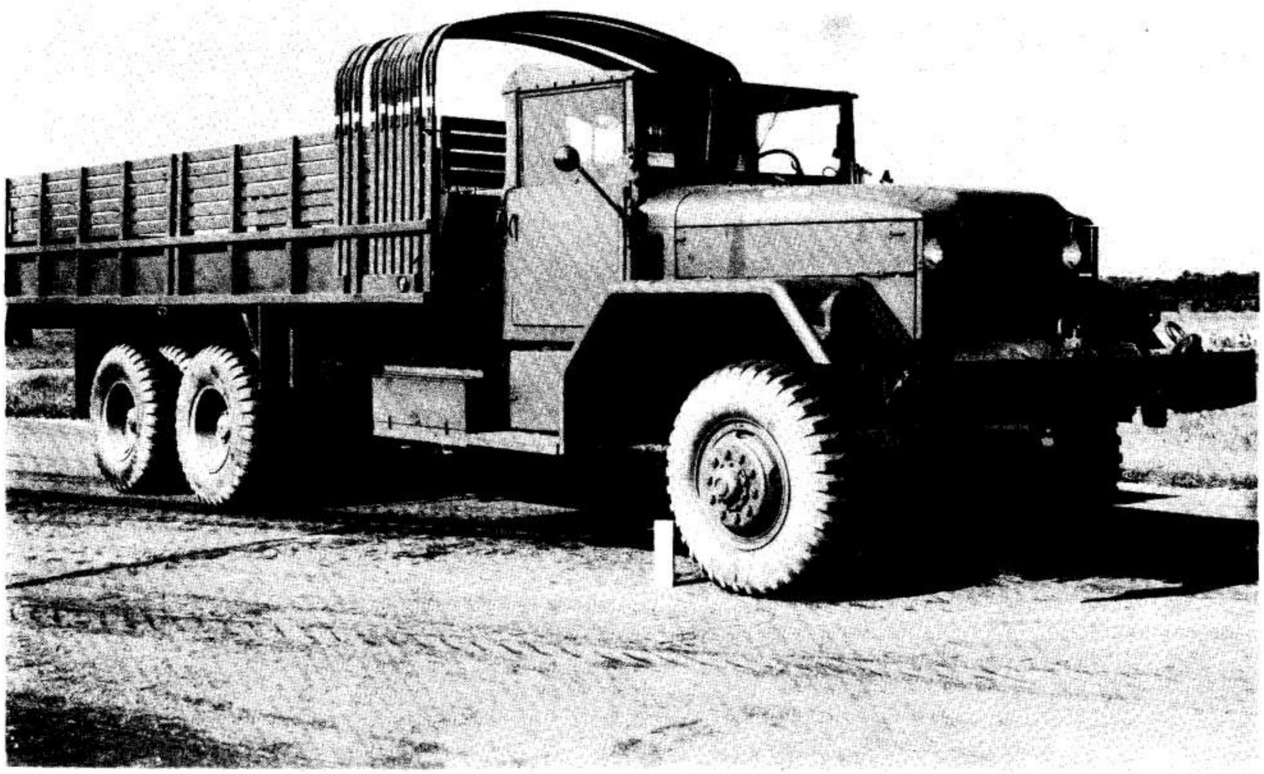


Figure 4-32. *Cargo Truck, 5-Ton, 6×6, M55—1956 (APG B19171)*

current models of several manufacturers; and, even though they are procured under appropriate military specifications, they are not tactical vehicles as they have not been designed to meet the severe military characteristics demanded of tactical vehicles. For the sake of uniformity, these vehicles are painted with the distinctive colors of the military service to which they are assigned (Army, Navy, Air Force). The engineering of these vehicles is well known. Some of these standard vehicles may have special-purpose bodies, such as oil tankers, garbage trucks, and buses. Details on many of these vehicles can be found in Ref. 1.

4-15 SPECIAL EQUIPMENT VEHICLES

These vehicles are comprised of a standard military-type chassis to which some minor modification was made or upon which a special body or equipment was mounted. The van body of the Ordnance Shop Truck is an example. Many vehicles of this type are required by other branches of the service. The dry cleaning and portable laundry vehicles, trailers for shoe repairs, and bath facilities are other examples. Although these special equipment vehicles are the responsibility of the specific service that operates it, the Army Materiel Command has the overall responsibility

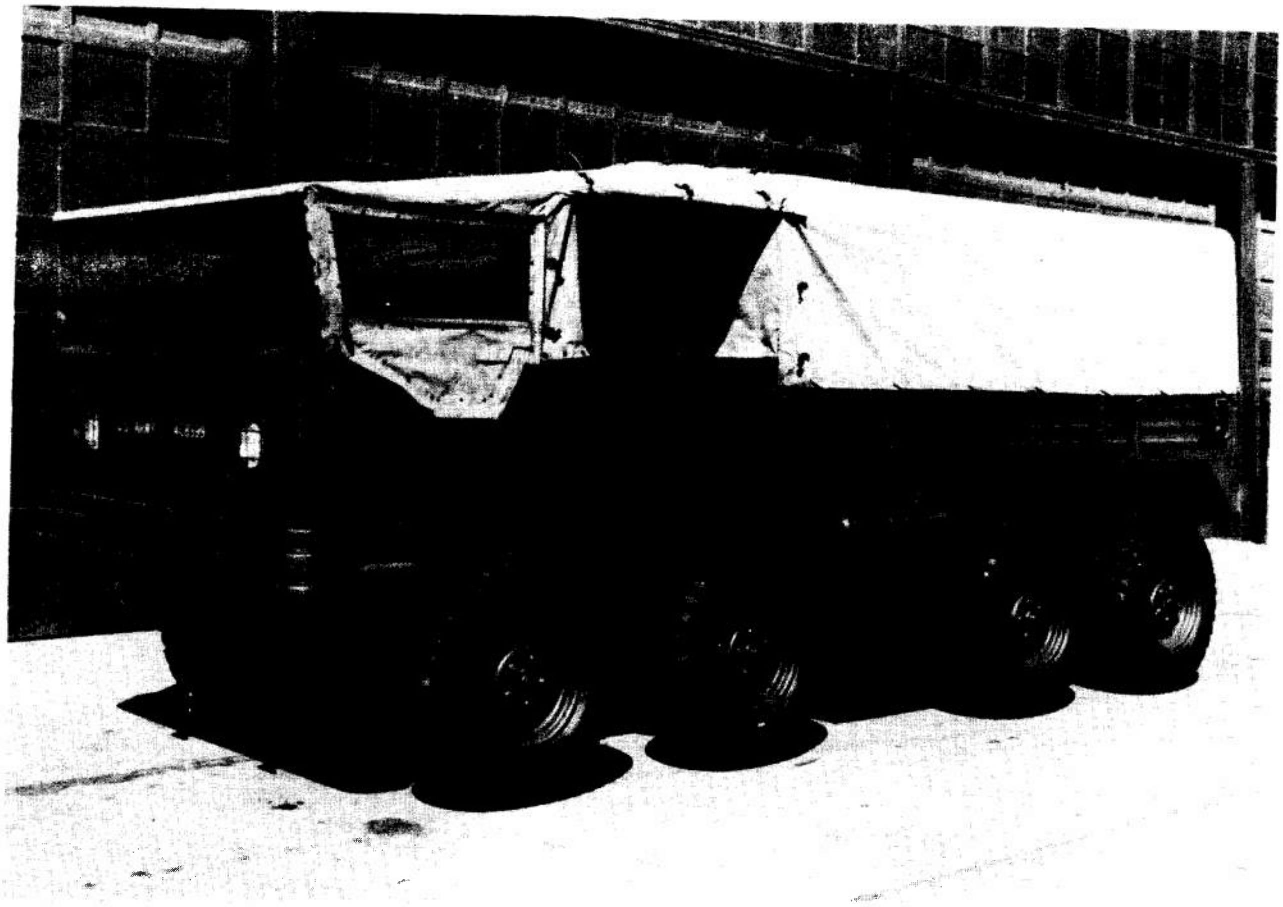


Figure 4-33. *Cargo Truck, 2-½-Ton, 8×8, XM410—1959 (DA 61192)*

TABLE 4-12 REPRESENTATIVE TRACTORS (TRACK-LAYING)

Model	Type	Gross Weight, lb	Horsepower	Towed Load, lb	Maximum Speed, mph	Use
M8E2	27 Ton	55,000	363	39,000	40	Tow artillery, transport personnel, cargo, ammunition
M5A1	13 Ton	30,405	207	20,300	30	Tow artillery, transport gun crew and ammunition
M4C	18 Ton	31,400	190	38,700	35	Tow artillery, transport gun crew and ammunition
M6	38 Ton	76,000	190	50,000	21	Tow artillery, transport gun crew and ammunition
M2	7 Ton	14,915	137	10,000	22	Air Force
T122	12 Ton	24,600	215	———	40	Tow artillery, transport cargo and ammunition
M85	20 Ton	41,000	533	33,000	37.6	Tow artillery, transport personnel, cargo or ammunition
XM474	12 Ton	24,000	215	———	40	Pershing missile, transport cargo or personnel

Figure 4-34. *Truck Tractor, 8-Ton, 6×6, T28E2—1948 (APG A53334)*

for the basic vehicle upon which the special equipment is mounted. The modifications required may result in difficulties with the vehicle center of gravity, power supply from the vehicle engine, electrical connections or electrical supply, clearances, load distribution, etc. Most combat ambulances belong in this group of special equipment vehicles, but ambulances used around base hospitals are usually standard civilian ambulances painted in keeping with the military vogue.

4-16 SPECIAL-PURPOSE VEHICLES

These vehicles are based on standard vehicle components but contain major modifications. Some of these vehicles are developed by agencies other than the Army Materiel Command. However, that Command is responsible for the basic components and usually will collaborate on such a development. Examples of vehicles in this category are Air Force and Navy crash trucks.

One of the most important groups of special-purpose vehicles are the wreckers developed by military designers for maintenance and recovery of military vehicles. These wreckers are usually developed from chassis components similar to those in the vehicles that the wrecker is expected to service. Figures 4-20 and 4-44 show typical examples of two general types.

Combat bulldozers are another important group of special-purpose vehicles. Other examples are flame throwers and portable launchers for missiles. Figure 4-45 shows an experimental special-purpose vehicle designed to explode antitank mines by the pressure of heavy rollers. The rollers were sufficiently rugged not to be damaged by the exploding mines. The system was successful until the enemy incorporated a special delay fuse in his mines so that the vehicle itself would detonate the mine.

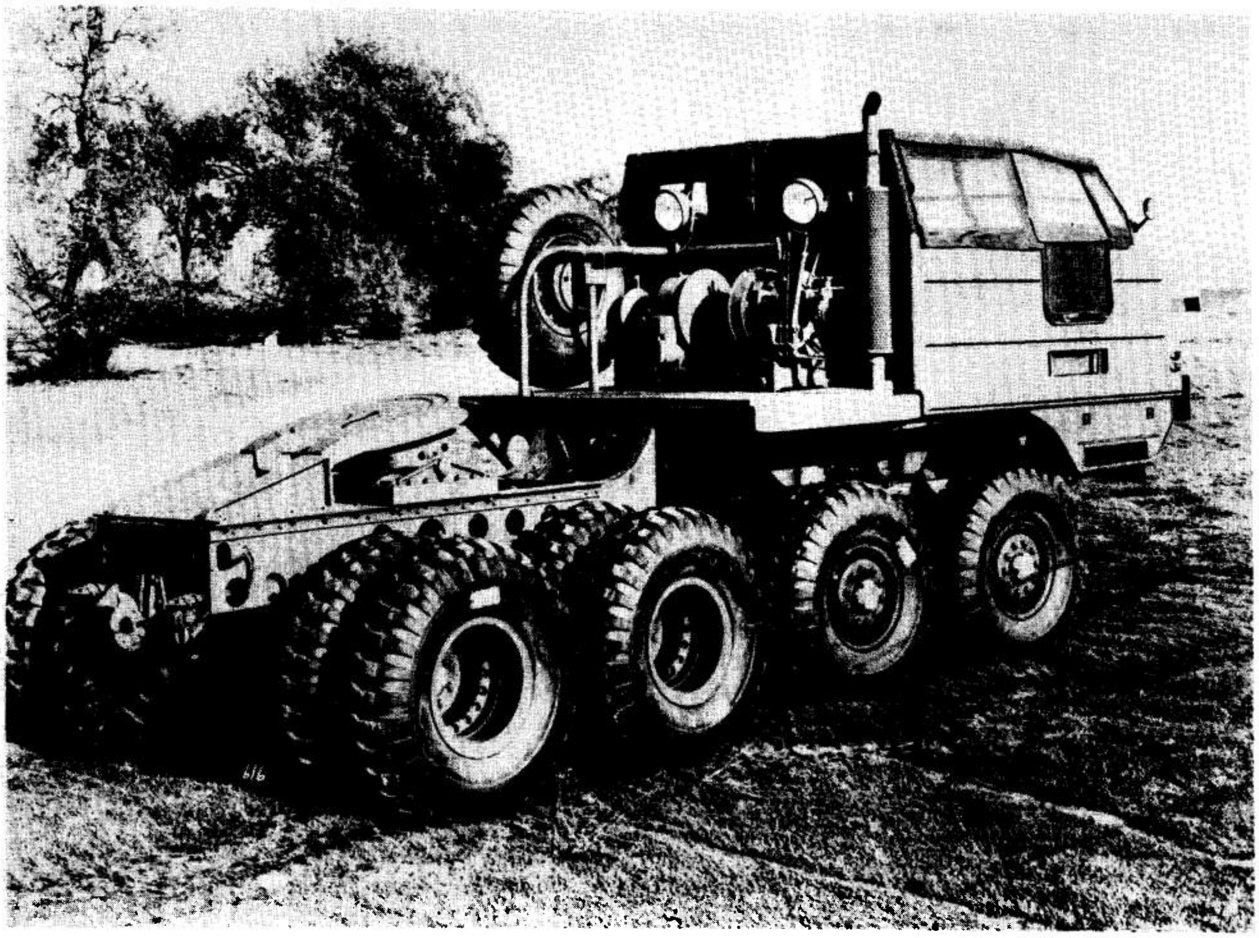


Figure 4-35. *Truck Tractor, 8-Ton, 8 ×8, T20—1945 (APG A22019)*

4-17 MINIATURE VEHICLES

The miniature vehicle category includes motorcycles, motor scooters, lift trucks, miniature tracklaying tractors, miniature one-man combat vehicles, and remotely controlled vehicles. Most of these, particularly the lift trucks and motor scooters, are adapted without modifications from civilian designs. Occasionally, however, special types are developed for military use. Size alone does not determine a vehicle for the miniature classification, but size and distance normally traveled are the true criteria. Lift trucks, used in loading cargo into planes, or logging trucks, which straddle loads in a timber yard, would qualify as big trucks if judged by their size; but, when judged by distances normally traveled, they fall into the miniature class. Some of these vehicles are quite special, as, for example, a vehicle for loading bombs into a plane. In general the responsibility for these vehicles rests with the using service; nevertheless, they are automotive vehicles in every sense and Army Materiel Command has an interest in them for this reason. There are some vehicles, however, for which the Army Materiel Command is directly responsible; for example, the T37 light tractor shown in Fig. 4-21. Furthermore, the Army Materiel Command is often called upon to assist in the preparation of specifications for miniature vehicles or to use its test facilities to evaluate existing designs.

4-18 NOVEL TYPES

As man's ingenuity is constantly directed toward the problems of mobility and locomotion, new concepts and unique designs are constantly appearing. Novel types of vehicles are constantly being tested and evaluated and the results of these tests lead to even further development. Most of this activity is directed toward a breakthrough to increase mobility over adverse terrain such as marshes, snow, soft sand, and mud. Many unique

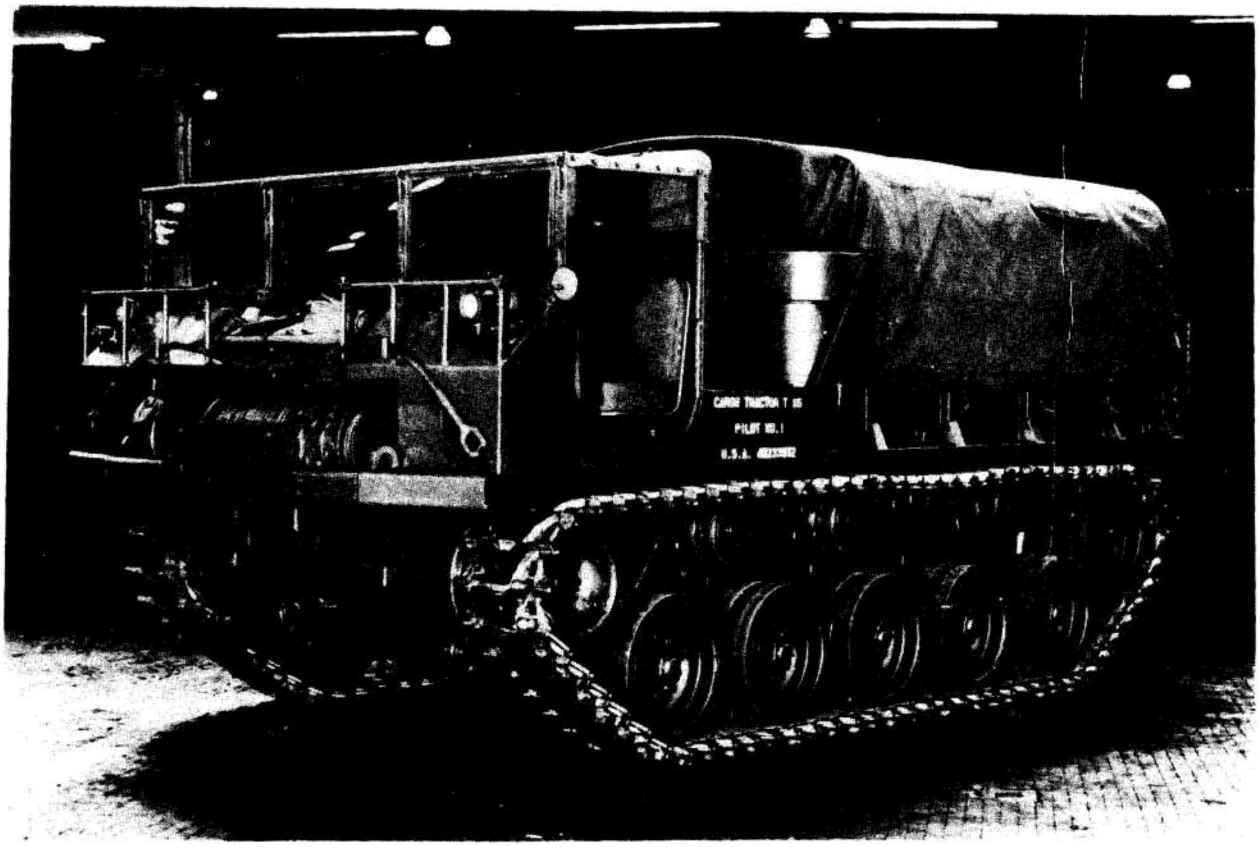


Figure 4-36. *Cargo Tractor, 20-Ton, Full-Tracked, M85—1954 (DA 44437)*

ground-contacting vehicles have been proposed, and, in some cases, prototypes have been tested. Thus, tests have been made on various walking vehicles, jumping vehicles, and even vehicles that employ an Archimedian screw principle to propel themselves across the ground. An extensive treatment of some of these unconventional vehicles can be found in Ref. 24.

4-18.1 THE SWAMP SKIPPER

Figure 4-46 shows an experimental vehicle, called the Swamp Skipper, that was designed for operations in extremely soft swamps and over snow. The huge metal wheels, 7 ft in diameter by 32 in. wide in the front and 36 in. wide in the rear, are hollow, to provide floatation on extremely soft terrain, and will even float the vehicle in water with its payload of 2,000 lb. When floating fully loaded, it has only 30 in. of draft. The vehicle performs well in mud, sand, and snow. It can cross open trenches 6½ ft wide if the edges of the trench are solid. Vertical obstacles 24 in. high are no problem. The high ground clearance (39 in.) enables it to pass over large obstacles by straddling them. Maximum speed of this vehicle is from 5 to 10 mph through muskeg and tundra. In rocky terrain the Swamp Skipper is forced to reduce its speed because of the severe and destructive jolting resulting from the lack of resiliency in the vehicle's suspension system (Ref. 24).

4-18.2 ROLLIGON VEHICLES (Ref. 24)

Various low-pressure air bags, resembling sausages, have been applied under vehicles to distribute the vehicle gross weight over a large ground area, thereby achieving floatation in marshy terrain or in soft snow. These air bags are commonly known as Rolligons. A typical Rolligon vehicle is shown in Fig. 4-47.

The Rolligon principle is somewhat unique in that the vehicle and payload weights are not carried on the axles of the air bags, as in the case in conventional wheeled vehicles. Instead, the weight is carried on a set of small diameter rollers lying on top of, and parallel to, the air bags. Thus, the load is transferred to the air bag in much the same

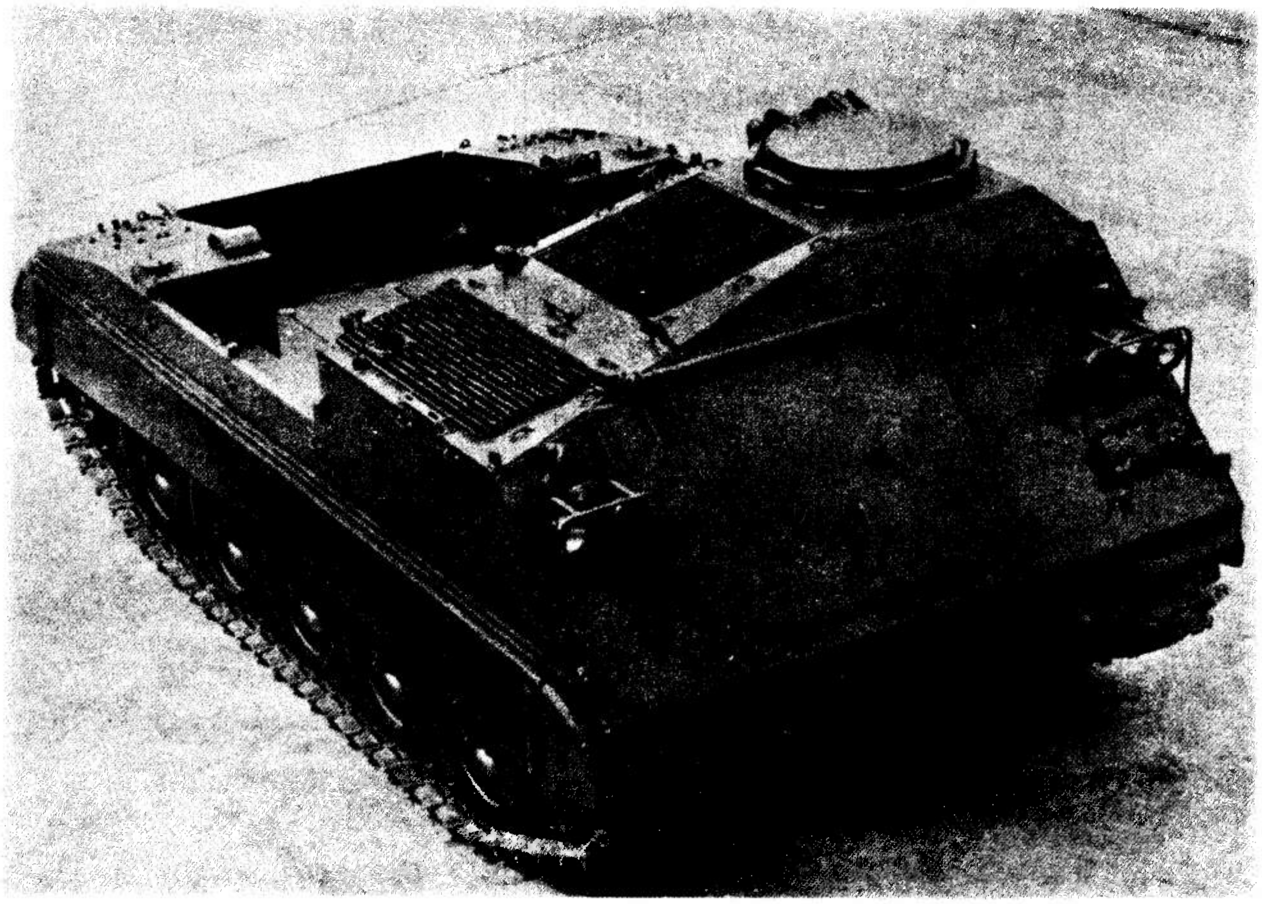


Figure 4-37. *High-Speed Tractor, T122 (FMCC ORD DIV. 13368)*

manner as load is transferred from the outer to the inner races of a roller bearing. Driving power is applied to the small rollers which drive the air bags through friction.

Rolligon vehicles of the type shown in Fig. 4-47 used air bags whose surfaces were relatively smooth. Mud, snow, and water reduced the frictional coefficient between the air bags and rollers, resulting in a poor transfer of power. Subsequent vehicles incorporated various molded designs into the tread of the air bags which meshed with matching designs on the surfaces of the rollers. This was an attempt at a more positive means of transferring power. None of these was satisfactory. An axle drive was later developed for Rolligon vehicles but this, too, had only limited success. Rolligon vehicles do exhibit low ground pressures and minimum penetration, even on dry, windblown snow. The low pressure air bags give an exceptionally smooth ride, despite the fact that there are no other springs or shock-absorber elements in the suspension system. In tests, however, this type of vehicle demonstrated poor tractive ability, particularly on snow and in mud, and even poorer durability of the air bags. The poor tractive ability exhibited by the Rolligon-type vehicles substantiates certain principles of land locomotion mechanics established by Bekker and others (Refs. 3 and 4) which maintain that greater tractive effort is developed by a footprint whose long axis is oriented parallel to the direction of vehicle travel than by a footprint of identical area and ground pressure, but oriented with its long axis at right angles to the direction of vehicle travel.

4-18.3 THE GROUND HOG

The Ground Hog vehicle is a test rig whose purpose was to compare the tractive ability of the spaced-link track with conventional closed-link tracks and to demonstrate the accuracy with which

Figure 4-38. *High-Speed Tractor, T122, Compared with 2-½-Ton, 6×6 Truck, M135, and 5-Ton, 6×6 Truck, M54 (FMCC ORD DIV 13380)*

performance can be predicted based upon laboratory tests. The Ground Hog (Fig. 4-48) is a bellyless vehicle that utilizes the suspension components and power train of the M29 Cargo Carrier (Weasel) (Fig. 4-24) with the addition of an extra bogie to increase the track length and an additional gear reduction between the differential axle and the driving sprockets to provide sufficient torque for maximum tractive effort. When operating in mud or deep snow, the vehicle sinks deeply until soil of sufficient shear strength for traction is encountered. The open track, however, presents a minimum frontal area to the mud or snow pack; and the vehicle, therefore, does not experience a high resistance to movement.

When tested in deep snow, the Ground Hog developed a maximum tractive effort equal to 84% of the gross vehicle weight, which was a greater tractive coefficient than was developed by any other of 18 different vehicles tested under the same conditions, and 2.4 times greater than was developed by the M29 Weasel with which it was specifically compared, because of its equivalent weight and similar suspension and power train. The slope-climbing ability of the Ground Hog in snow was also far superior to the other vehicles tested (Ref. 25). When tested in sand, the Ground Hog developed 50% more tractive effort than did the similarly weighted M29. In negotiating wet, soupy mud in a 4- to 5-ft deep basin, it was compared with three other vehicles: the standard M29 with 20-in. wide tracks fitted with 1-in. grousers, a modified M29 with tracks widened by 7-in. and grousers extended to 3 in., and the T46 (Otter) (Fig. 4-26).

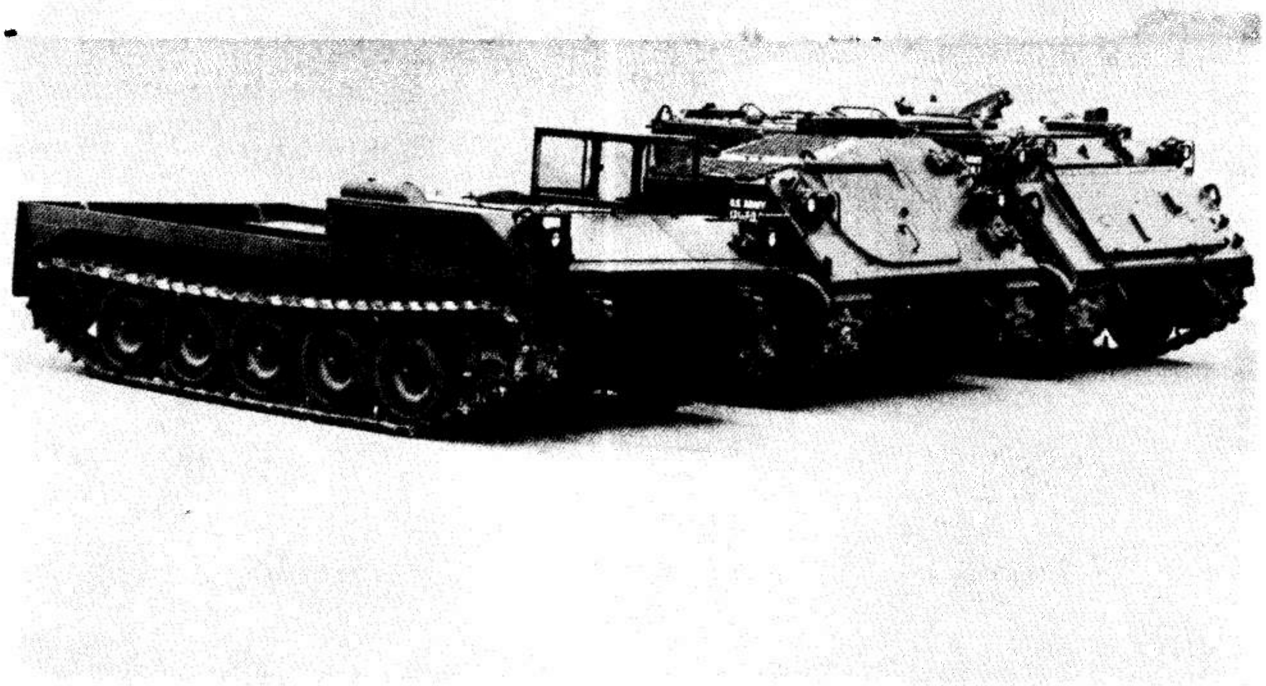


Figure 4-39. Left to Right, Missile Equipment Carrier, XM474, High-Speed Tractor, T122, Personnel Carrier, M113 (FMCC ORD DIV 13373)

All vehicles except the Ground Hog became immobilized, while this vehicle crossed the swamp with ease, practically without slip, and with the engine at only part-throttle with the transmission in high gear (Ref. 24).

The successful demonstration of the Ground Hog led to the development of the T60 test vehicle to proof-test the practicality of the spaced-link track for lightweight, cross country vehicles. This test vehicle was intended to establish the basis of design for prototypes of the T60 amphibious cargo carrier, a projected ½-ton cargo carrier for operating over water, soft marshy terrain, muskeg, sand, snow, ice, and tundra. When first tested by the manufacturer, the vehicle successfully scaled ridges of loose "gumbo" of 72.6% maximum slope, negotiated a pool of water about 3 ft deep, and towed a 13,625-lb tracked bulldozer, with its tracks locked and its dozer blade down, through approximately 18-in. deep mud (Ref. 24). In tests conducted at Aberdeen Proving Ground (Ref. 26), the vehicle operated successfully up a 60% slope with the transmission in high gear and the transfer case in low. In sand and mud, the vehicle successfully towed a 105mm howitzer weighing 3,900 lb; and it also pulled the M76 (T46) amphibious cargo carrier (Fig. 4-26), with the 105mm howitzer in tow, from the mud course when the M76 was unable to continue maneuvering. Operating in soft sand, the T60 test vehicle developed a drawbar pull equal to 73% of its gross weight. This exceeded the percentage drawbar pull of the M29 Weasel by 14% and that of the M76 Otter by 26%.

The major defects of the spaced-link track vehicles tested were poor maneuverability in deep snow and an appreciably greater resistance to tow-

TABLE 4-13 REPRESENTATIVE SEMITRAILERS USED AS TRANSPORTERS (Ref. 20)

Model	Nominal Capacity, tons	No. of Axles	No. of Wheels	Tire Size	Wheel-base,* in.	Notes
M9	45	3	14	8.25×15	187	Full Trailer
M15E1	45	2	8	14.00×24	342	Used with M26 truck tractor
T63	60	3	12	14.00×24	383	——
T67	100	6	24	11.00×20	390	Includes Dolly
T58	45	3	24	9.00×20	248	Axles are divided
T60E3	25	2	4	18.00×29	348	——
T79	100	38	10	1.00×29	549	Includes Dolly
T74	75	2	8	18.00×29	462	——
XM160	60	3	12	11.00×20	—	Used with XM194 truck tractor
XM161	60	2	4	14.00×24	—	Used with XM194 truck tractor
XM171	25	2	4	14.00×20	—	Used with XM123 truck tractor
XM173	25	3	12	9.00×20	—	Used with XM123 truck tractor
XM276	—	—	—	11.00×20	—	Powered rear axle

* Wheelbase on a semitrailer is the distance from the kingpin to the center of wheel axle or to the center of bogie assembly.

ing than comparable closed-track vehicles. The steering difficulties encountered when operating in deep snow were a major handicap which requires further study and development.

4-18.4 THE GOER VEHICLES (Ref. 27)

The term Goer has been applied to a relatively new family of vehicles which, allegedly, have such a high degree of mobility that they can literally operate anywhere—across rough terrain, through sand and thick mud, climb over a 30-in. vertical wall, and even swim in calm water while fully loaded. Figures 4-49 and 4-50 show two versions of this type of vehicle. The first has a cargo body with a payload capacity of 15 tons, while the second is a 5,000-gal tank truck. Principal characteristics of the Goer vehicles are their large-diameter low-pressure tires (29.5 × 25, 16-ply), wagon-type steering, and exoskeletal body construction. Exoskeletal design stresses the skin of the vehicle rather than a frame to which a body is added. This type design causes all of the material used in the vehicle to carry some portion of the total load, resulting in less dead weight and achieving inherent floatability. The vehicles were developed by a long-established producer of heavy, earth-moving machinery and were constructed largely from off-the-shelf components.

Each vehicle is comprised of two sections, somewhat resembling a truck tractor and semi-trailer combination. In the place of a fifth wheel, however, the juncture between the front and rear units has a ball-and-socket type mounting that permits a 20° lateral roll between the two units. Steering of the vehicle is accomplished by rotating the complete front section in a manner similar to that employed in animal-drawn wagons. This wagon steering is powered by an electric motor located at the pivot point between the tractor and cargo body. The front section can be swung 90° to either side from the straight-ahead position, thus, imparting a much shorter turning radius to the vehicle than is found on most military trucks.

The power plant of these vehicles is housed

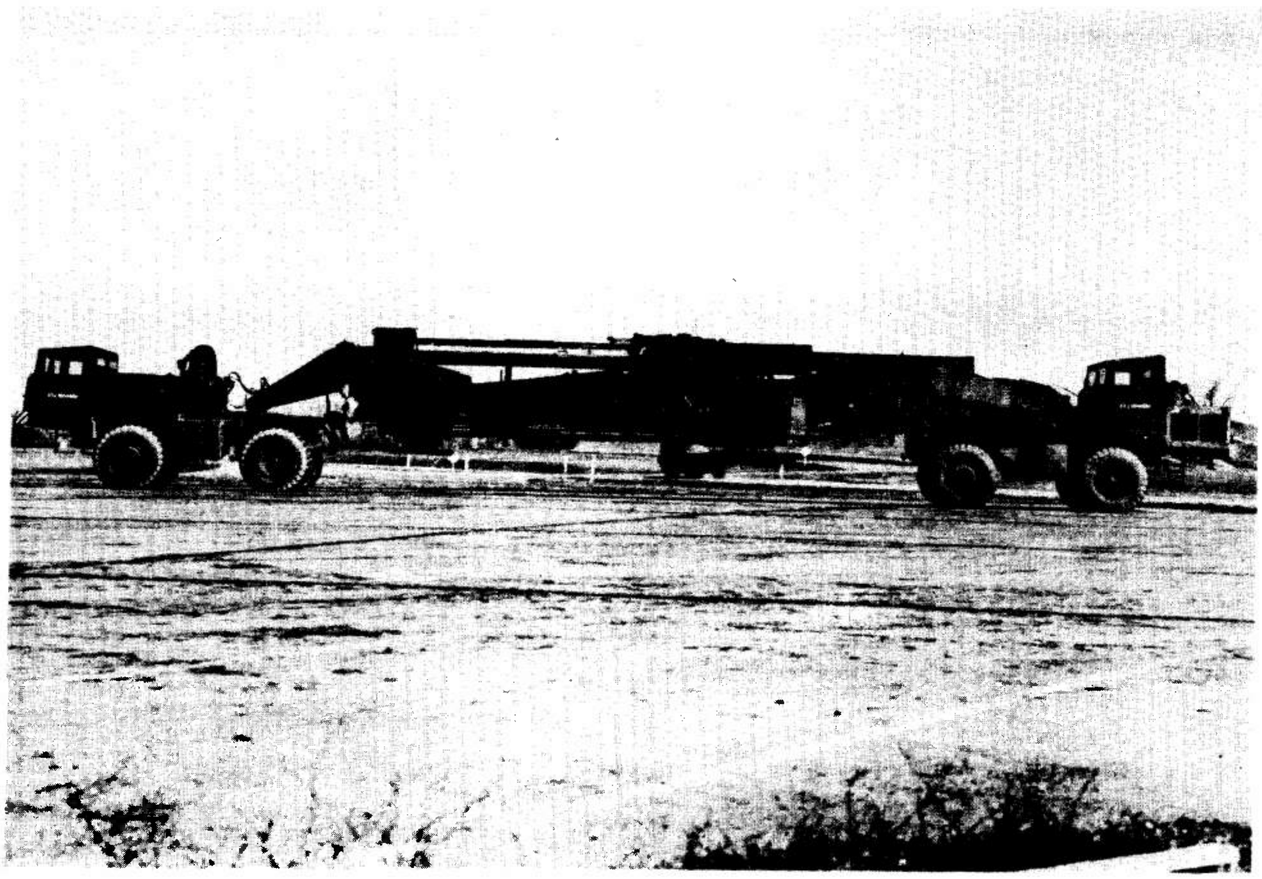


Figure 4-40. Heavy Artillery Transporter, T10—1953 (APG A85189)
(Coupled to T72 Gun Carriage with T131 Gun)

in the front unit. It consists of a 274-hp Diesel engine coupled to a three-phase, AC generator capable of a 518 v., 200-cycle output, and to a five-speed, air-actuated, mechanical transmission. The front wheels of the vehicle are driven mechanically through the transmission, while the rear wheels are driven electrically, with power supplied by the engine-driven AC generator, by electric motors mounted in each wheel. The rear wheels are driven only when the transmission is in first gear and in reverse gear low range. At all other times, both on land and in water, propulsion of the vehicle is by mechanical means, through the front wheels only. Multiple-disk type air-actuated brakes are supplied on all four wheels. In addition, dynamic braking is provided for use on long downhill grades. The electrical energy developed by the AC generator during braking is dissipated to the atmosphere in the form of heat by means of grids mounted on the tractor atop the transmission compartment.

The major dimensions, characteristics, and specifications of the XM437 Goer-type cargo truck are:

Curb weight	17 tons
Payload	15 tons
Length	36 ft, 2½ in.
Width	117 in.
Height	125 in.
Wheelbase	288 in.
Ground clearance	17 in. at axles, 30 in. between axles
Cruising range	300 miles
Maximum speed	32.5 mph on land, 3 mph on water.

4-18.5 GROUND-EFFECT VEHICLES

Another group of unique vehicles currently receiving considerable interest and undergoing extensive development in a variety of forms, is the ground-effect vehicles. Ground-effect vehicles, also known as surface-effect vehicles and air-cushion vehicles, operate in close proximity to the ground

Figure 4-41. *Tank Transporter, 40-Ton, M15, Loaded with Heavy Tank, T26E1—1944 (APG A15317) (Power unit is truck tractor, 12-Ton, 6×6, M26A1)*

or water and derive all or part of their support from a cushion of pressurized air. These vehicles operate with ground pressures of 0.5 to 1.0 psi. There are no standard vehicles in this category at the present time.

At least four basic methods are used to develop the required air cushion, these are:

a. *Plenum chamber*—Air is pumped into a plenum cavity formed between the underside of the vehicle and the ground. As the air pressure builds up, it causes the vehicle to rise allowing the air to escape into the atmosphere at the periphery of the plenum. The flow area at the periphery of the plenum adjusts itself until the weight of the vehicle is balanced by the force developed in the pressurized chamber.

b. *Peripheral jet*—In this system, the lift is developed by a momentum change of the air and the back pressure developed under the vehicle. Air is ejected downward through an annual jet or nozzle, located around the periphery of the machine, and is forced to flow horizontally when it strikes the ground. The reaction resulting from changing the direction of the velocity vector of the air together with the air pressure developed beneath the vehicle generate the lifting force. This system is sometimes called an air-curtain.

c. *Multiple curtain*—In this system, the peripheral jet is produced in stages by means of multiple fans. The use of multiple fans results in a reduction of the pumping power required. Like the single-stage, peripheral jet, the lifting force is generated by the momentum change of the airstream together with the air pressure developed under the vehicle.

d. *Levapad*—The levapad system is properly classified as an air-bearing system. Very small clearances are used between the vehicles and the supporting surface, and the pressures are

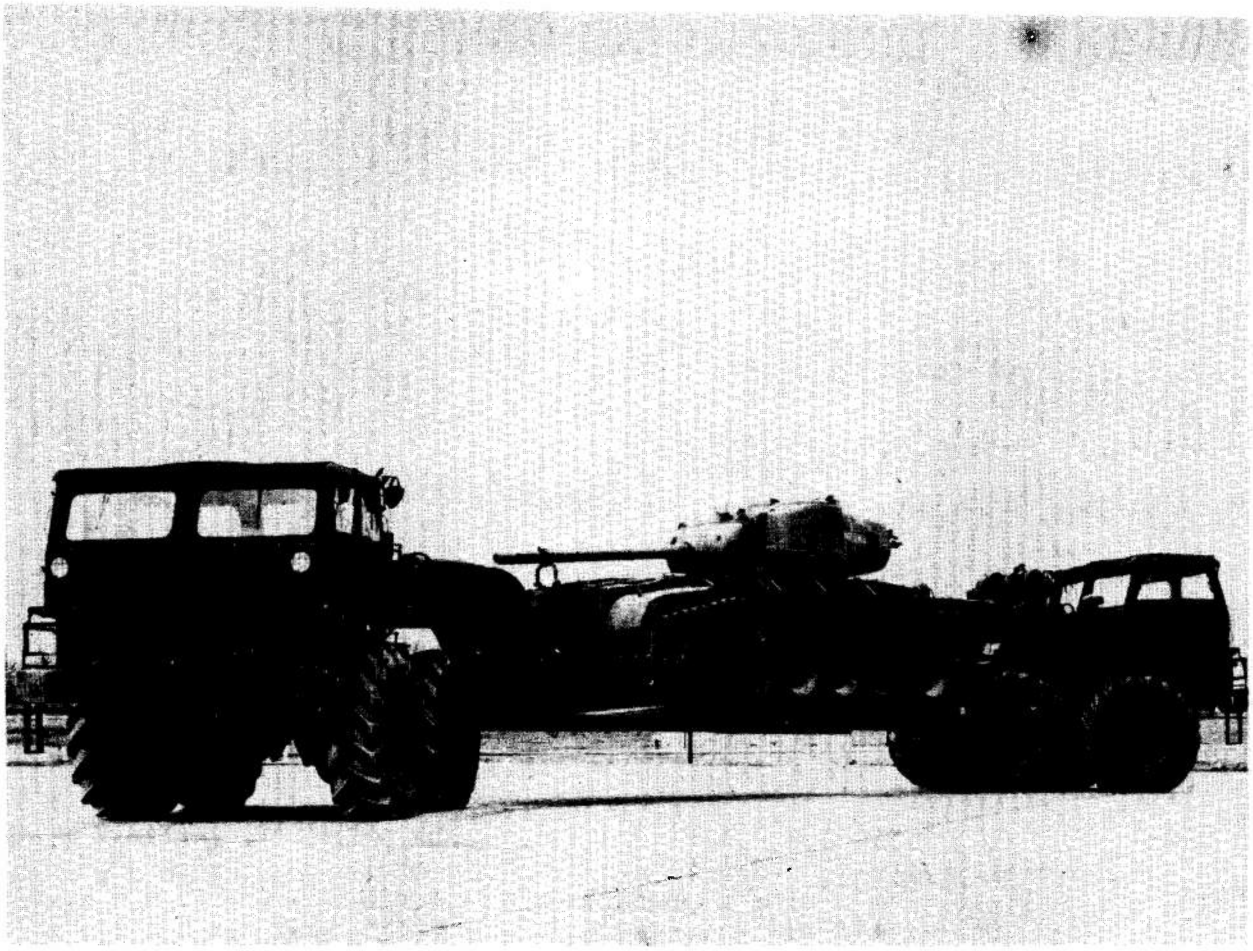


Figure 4-42. Heavy Transporter, T8 (Right Rear View)—1946 (APG A34969)

higher than those in other air-cushion systems (15 to 100 psi).

Some ground-effect vehicles maintain contact with the ground and are powered in a manner similar to conventional land vehicles.

The obvious advantage of ground-effect vehicles is their ability to operate over all types of terrain, including bodies of water. Sufficient clearance must be provided in order to pass over obstacles. One of the major problems associated with the development of ground-effect vehicles concerns the control. Propulsion and control are accomplished by means of air jets, or by adjusting louvers in the airpath to create a thrust in the desired direction. Ground slopes, and side slopes in particular, cause the vehicle to deviate from its prescribed course and drift downhill. Air jets or the redirection of the air by means of adjustable louvers are the chief means employed in controlling this downhill slipping. Ground-effect vehicles are very sensitive to shifting loads. Any shifting of the vehicle load will usually cause a change in the direction of motion. These problems, and others, are being considered in the current development programs. Additional information on these unique vehicles can be found in Refs. 28, 29, and 30. Figure 4-51 shows one type of experimental ground-effect vehicle.

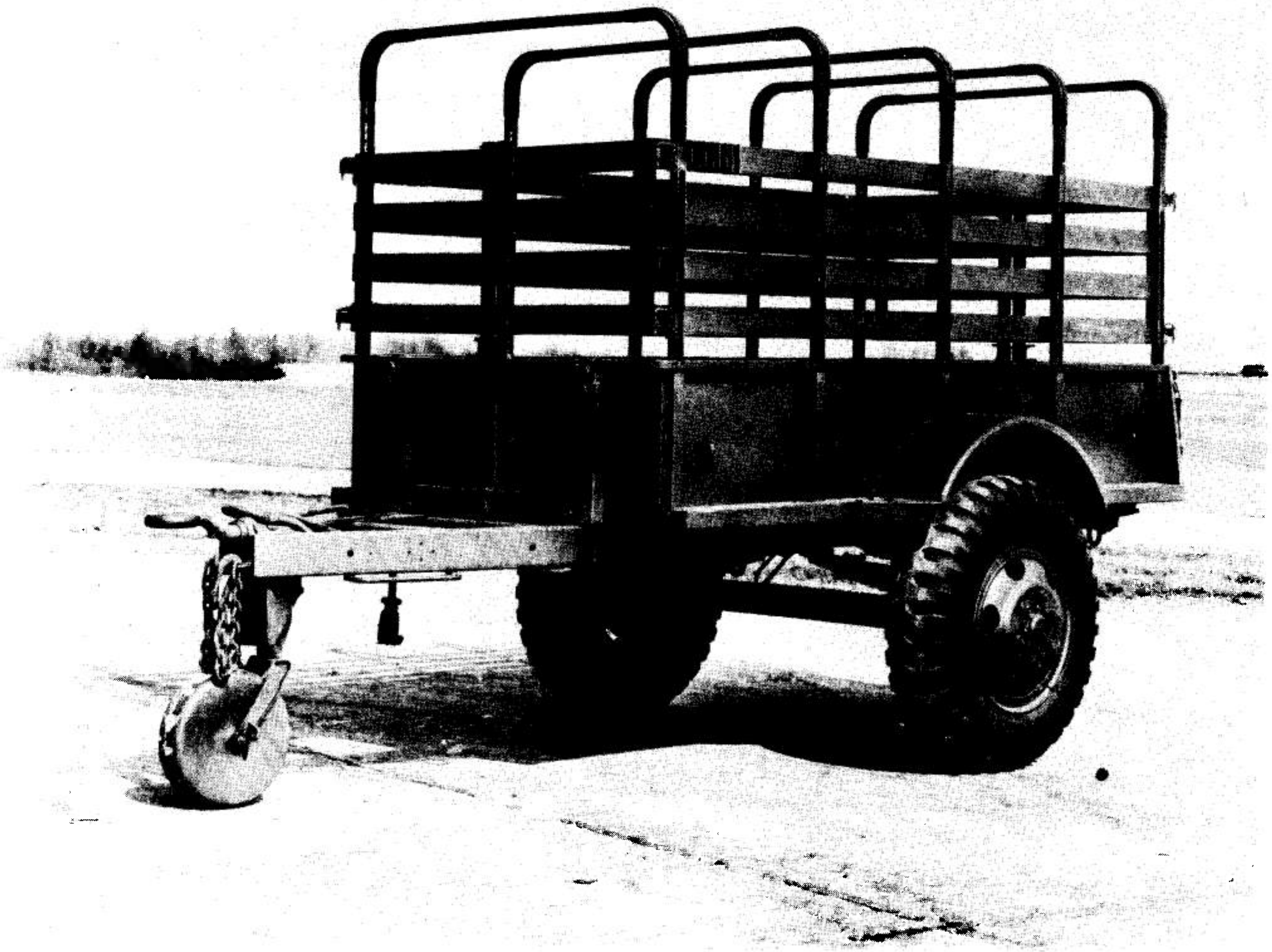


Figure 4-43. Cargo Trailer, 1-½-Ton, 2-Wheeled, XM105E1—1952 (APG A76934)

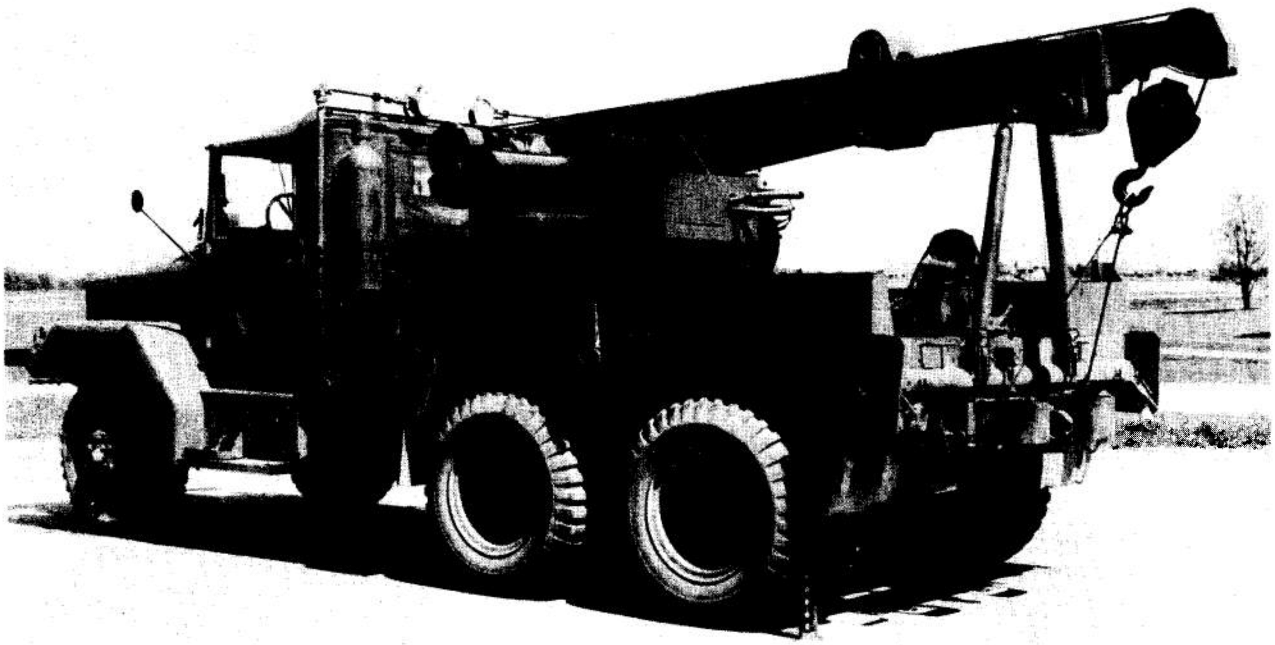


Figure 4-44. *Truck, Wrecker, 2-½-Ton, 6×6, M60—1952 (APG A78069)*

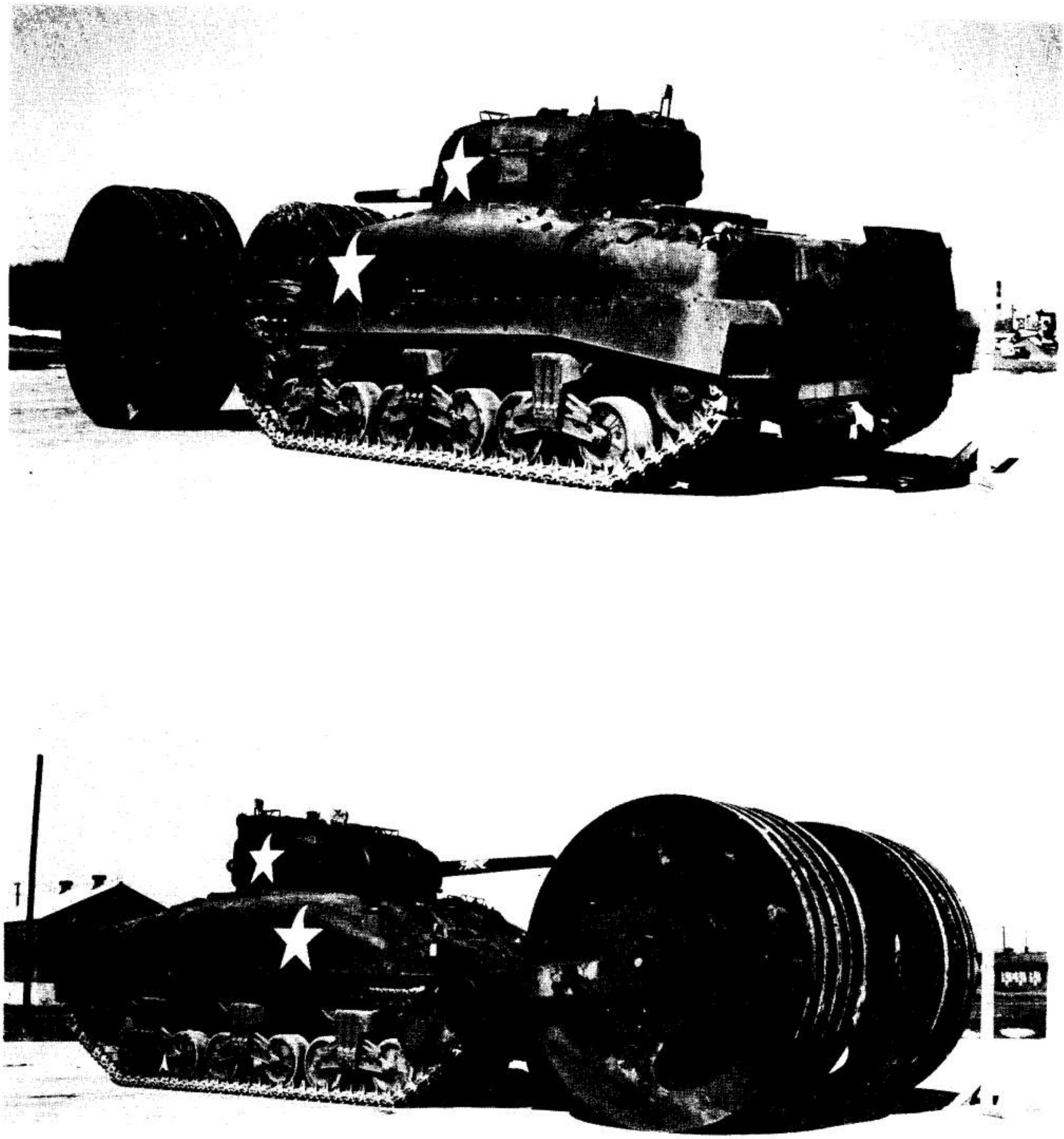


Figure 4-45. *Antitank Mine Exploder, T1E3—1944 (APG A921)*

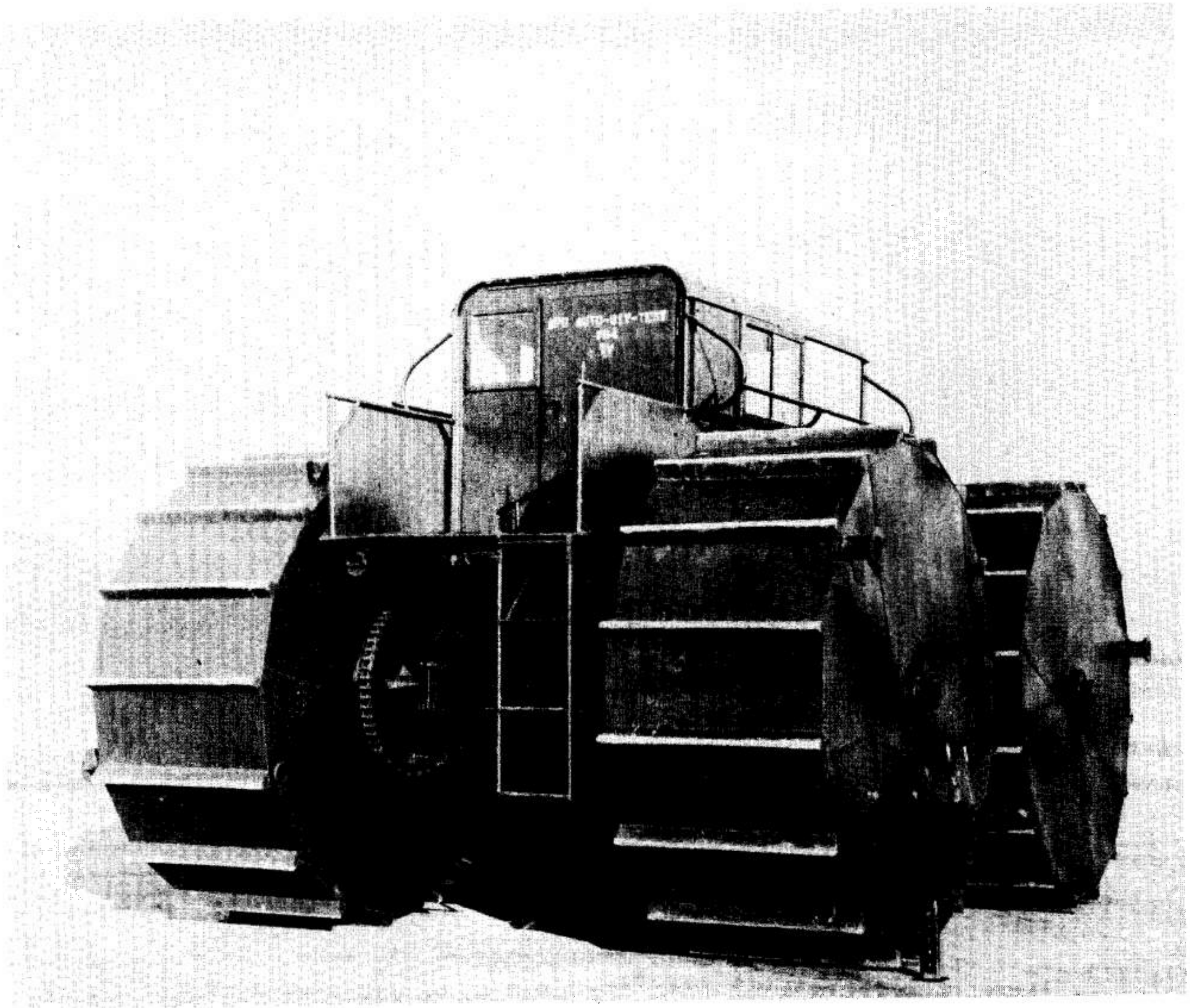


Figure 4-46. *Swamp Skipper, Model 5—1948 (Right Rear View) (APG A54835)*

Figure 4-47. *Rolligon-Equipped Cargo Carrier—1954 (APG A98799)*

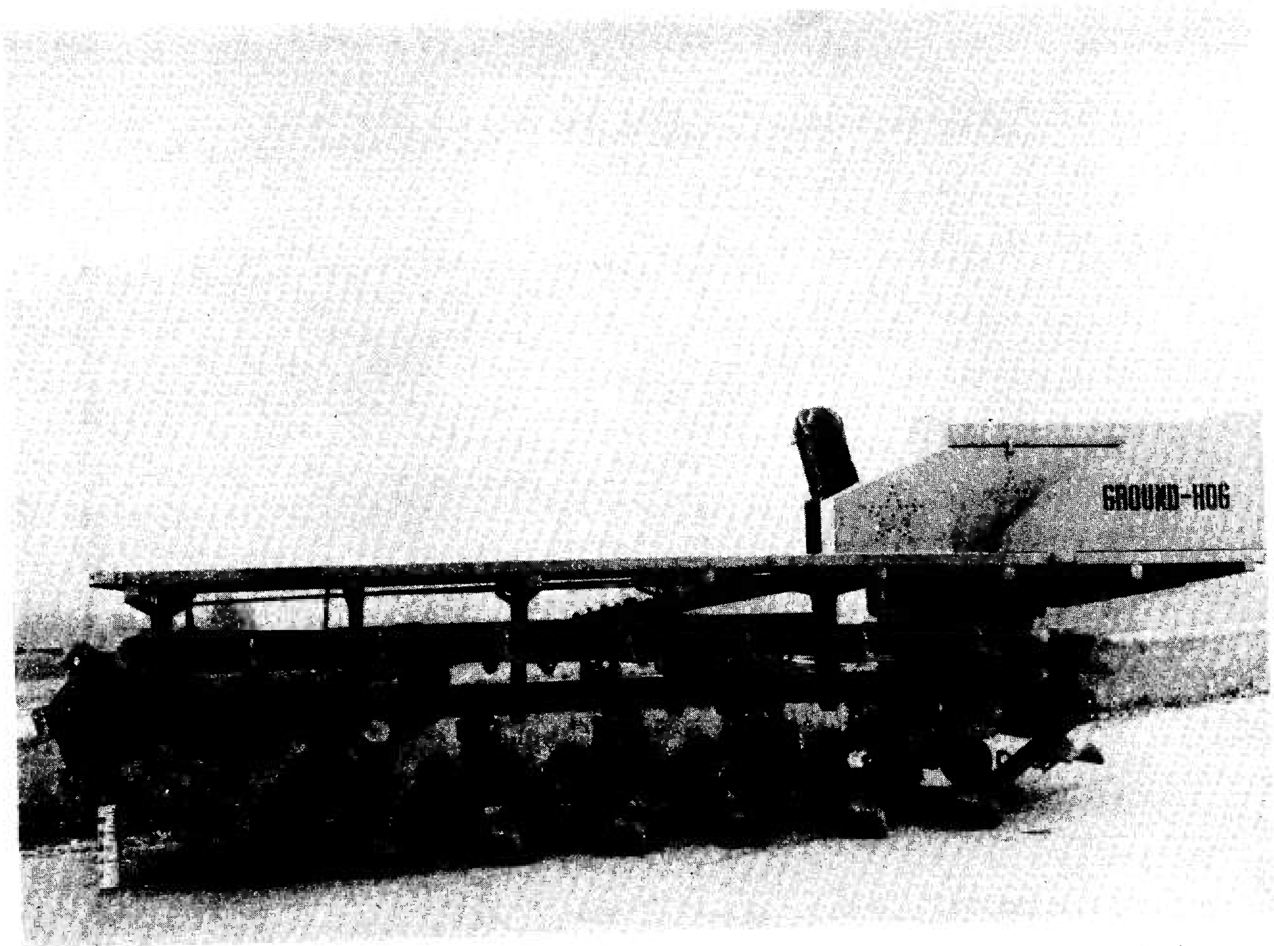


Figure 4-48. *Lightweight Cargo Carrier with Spaced-Link Track (Ground Hog)—1949 (APG A57413)*

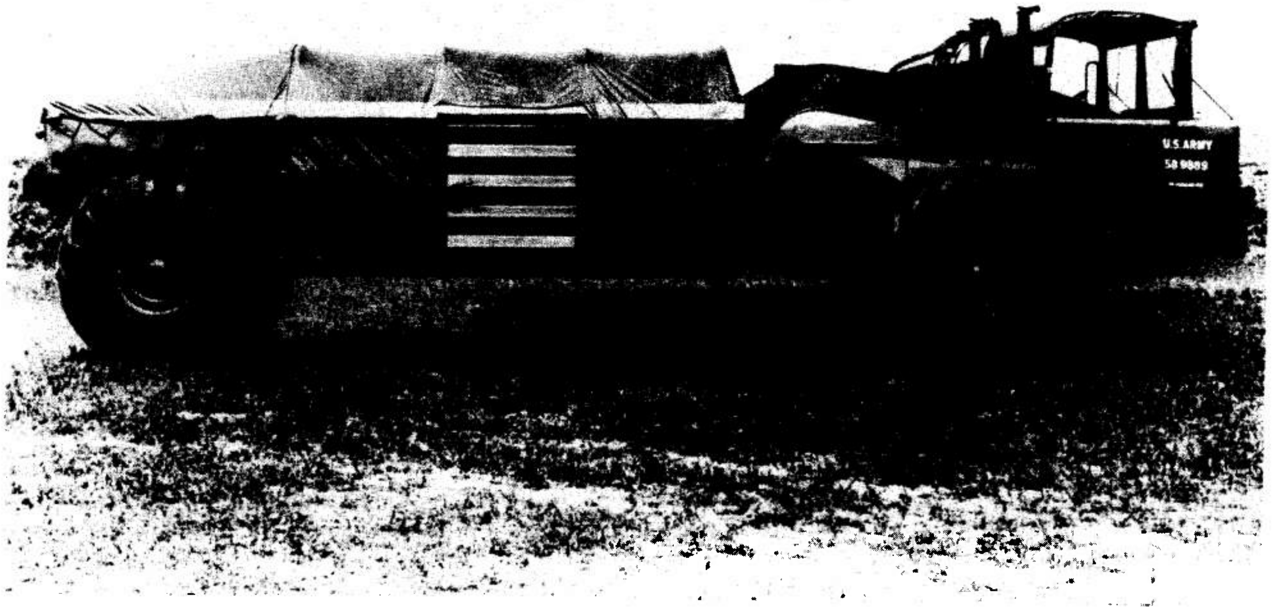


Figure 4-49. *Cargo Truck, High Mobility, 15-Ton, 4 × 4, XM437 (GOER)—1959 (DA 60516)*

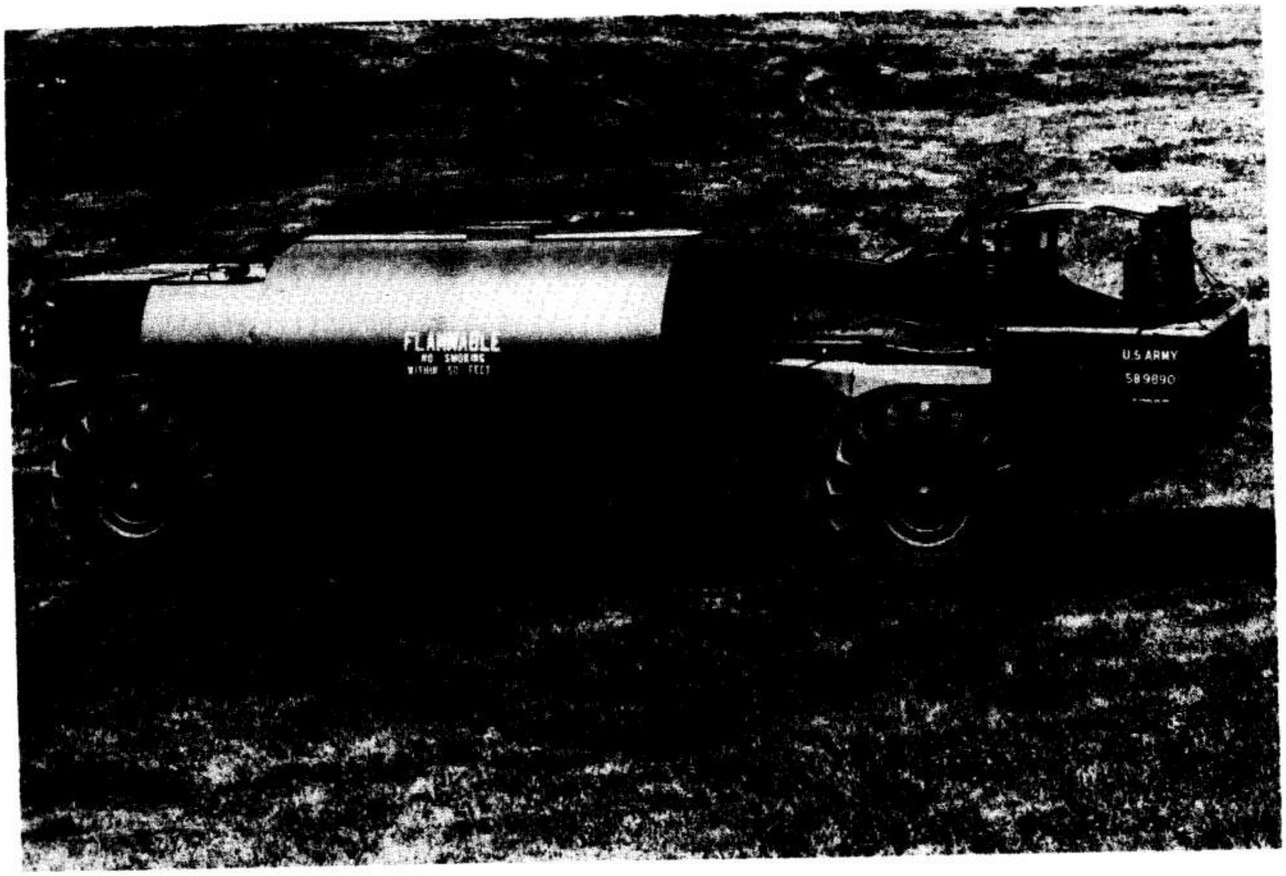


Figure 4-50. *Tank Truck, High Mobility, 15-Ton, 4 × 4, XM438 (GOER)—1959 (DA 60513)*

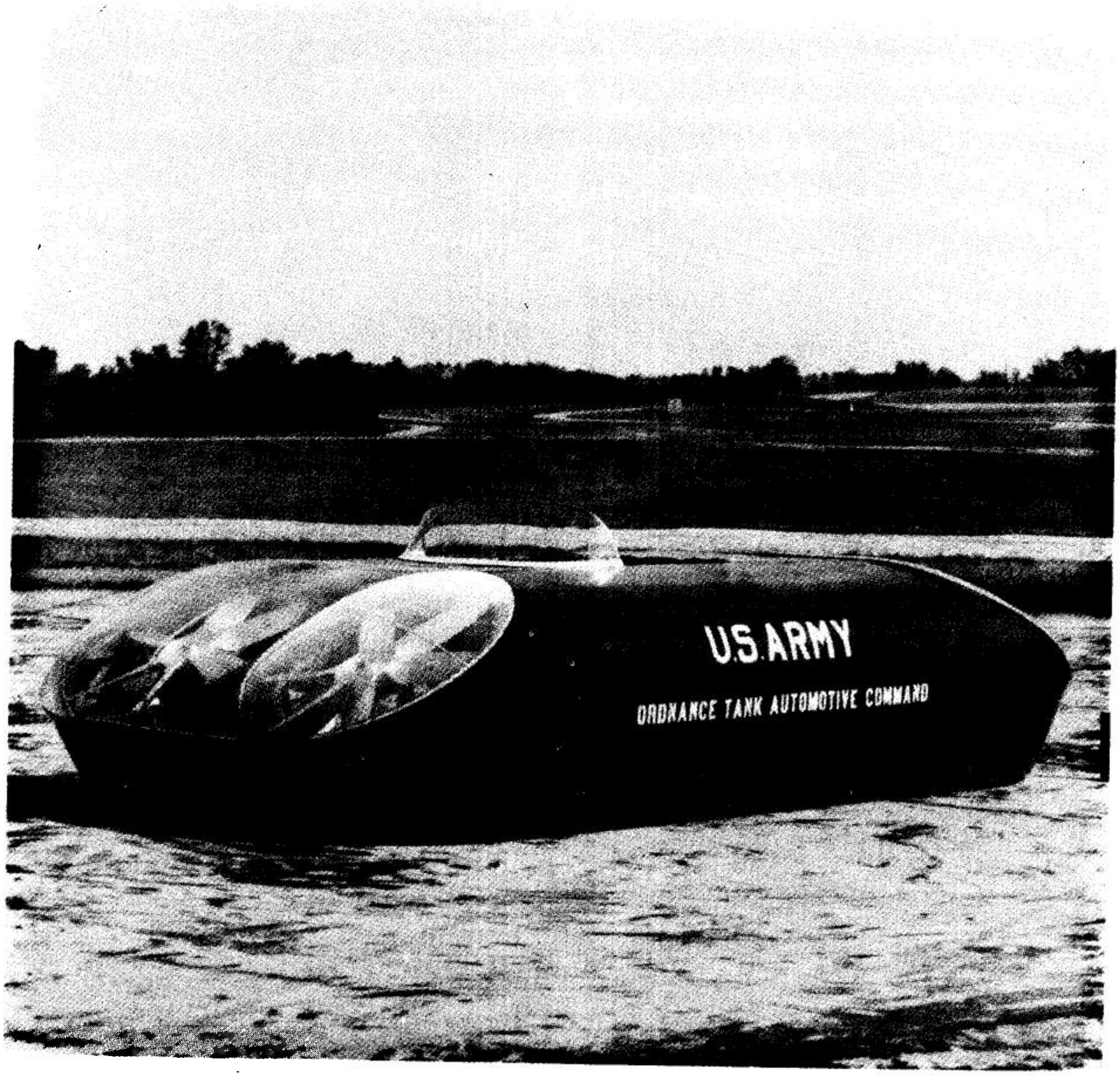


Figure 4-51. *Ground-Effect Vehicle, Pegasus I—1959 (APG 59P51C)*

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CHAPTER 5

THE FORCE SYSTEM*

Automotive assemblies are constantly subjected to a complex system of forces whose magnitude and orientation vary with time. This complex force system is comprised of forces that fall into one of two general categories: (a) those forces that can be readily determined by computations and simple measurements, and (b) those forces that cannot be readily calculated and require elaborate measuring procedures, complex equipment, and sophisticated mathematical techniques for their evaluation. For convenience in discussing these two categories, they are classified in this chapter as *determinate* and *indeterminate* forces.

* Written by Kenneth E. Hofer, Jr. and Rudolph J. Zastera of the Illinois Institute of Technology Research Institute, Chicago, Ill.

Typical determinate forces are those imposed by the weights of the various components and contents of the vehicle, those forces due to acceleration of the vehicle, and those to engine torque, braking torque, etc. Examples of indeterminate forces are forces resulting from shocks and vibrations encountered when the vehicle is traveling over rough terrain, when it is airdropped, or when it is subjected to high energy blast or ballistic impact. In order to simplify the discussion of the force system acting upon the automotive assembly, these two general categories of forces are treated separately.

SECTION I DETERMINATE FORCES

5-1 FORCES IMPOSED BY WEIGHT OF PARTS

5-1.1 VEHICLE AT REST ON LEVEL GROUND

Weight forces constitute the only force systems of importance when a vehicle is at rest on level ground. When the overall force system acting on a vehicle at rest is examined, it is observed that there are essentially two force resultants: W_T, the total weight of the vehicle, and $\int_A p(x,y)dA$, the base reaction force resultant (Fig. 5-1).

The point of application of W_T is at the center of gravity and does not change when the vehicle is tilted or placed in some other unfavorable position. The gross reaction, $\int_A p(x,y)dA$, must also remain constant, oppositely directed, and collinear with W_T. However, the distribution of $p(x,y)$ along the vehicle base, when the vehicle is tilted or is in some other unfavorable position, may differ substantially from that distribution in the rest position. Internally, that is to say, within the perimeter of the vehicle, the forces caused by the main structure of the vehicle are transmitted to the ground through a suspension system, consisting of springs, connecting rods, wheels, etc. This suspension system supports the frame which, in turn, supports the various components of the total load W_T. Each of these systems must be studied separately to ascertain their influence on the general spatial distribution of the various forces acting on the automotive assembly.

5-1.1.1 Forces Acting on the Frame

The frame is the base to which the body and the other units of the chassis are attached. The shape and construction of the frame depend upon the use for which the frame is intended; consequently, only an overall scheme of loading can be

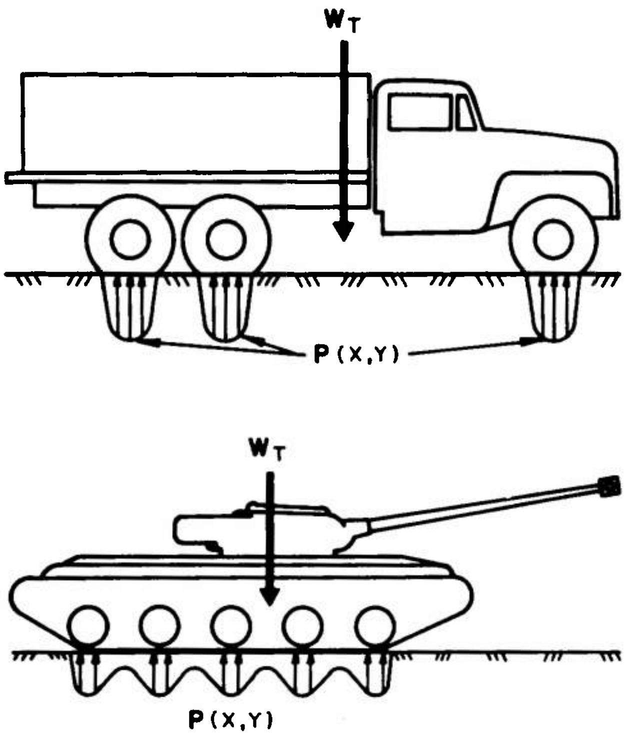


Figure 5-1. Force System Acting on a Vehicle at Rest on Level Ground (The gross base reaction, $\int_A p(x,y)dA = W_T$, where A=ground contact area and W_T=total vehicle weight.)

given here. In most wheeled vehicles, the frame is an element separate from the rest of the automotive assembly. In the case of tracked vehicles, amphibious vehicles, and certain cargo carriers, the hull serves as the frame. Most of the frames currently used in wheeled vehicles are simply scaled up from those used in ordinary passenger vehicles. If the vehicle is to be used as a prime mover, i.e., a towing mechanism, the longitudinal (lengthwise) members are reinforced as well as the rear members of the vehicle to prevent injurious distortions to the vehicle. In Ref. 1, some reinforced frames are discussed for their ability to sustain loads encountered during the vehicle motion.

The principal loads experienced by the frame when the vehicle is at rest on level ground are: the weight of the vehicle body, W_B; the weight of the power plant, W_e; the weight of the power train, W_P; the weight of the crew, W_M; weight of the equipment carried, W_q; and the weight of the cargo, W_c. Of these various load elements, the weight of the vehicle body, W_B, is the most appreciable, comprising from approximately 60% of the net vehicle weight, in the case of unarmored, tactical vehicles, to approximately 90% of the net vehicle weight in the case of armored, combat-type vehicles. When the frame is considered as an isolated free body, the reactions from the suspension system ($S_1, S_2, \ldots$) must also be included in the force system acting on the frame. In general, when the vehicle is at rest in a level position, it can be stated that the sum of the reaction forces is numerically equal to the sum of all the vehicle components supported by the frame plus the weight of the frame itself. In Fig. 5-2 a typical spatial force distribution is shown acting on a frame.

In towed vehicles, such as trailers or sleds, the principal load is the weight of the cargo rather than the weight of the vehicle body. Furthermore, the engine and crew weights are generally absent. This leads to a simplification of the effective loading on the vehicle frames. Sleds are an example of a vehicle in which the hull is the vehicle frame. Many of the typical cargo sleds in use are nothing more than ski-mounted pallets.

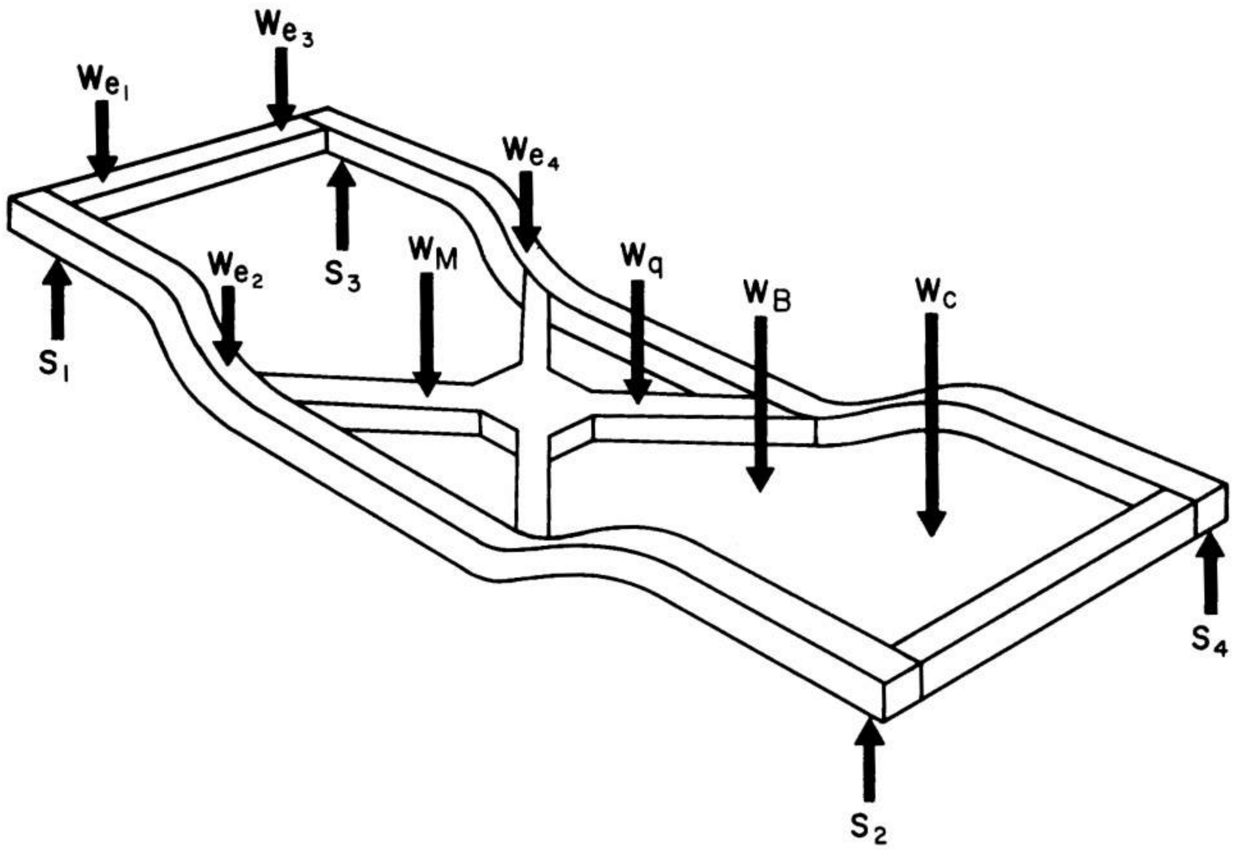
Each of the weight loading forces will now be discussed in slightly greater detail.

5-1.1.1.1 Power Plant Weight

The weight of the power plant acts through the center of gravity of the power plant and is given by the sum of the weights of the individual components. The power plant is mounted to the frame by a series of bolts and stabilizing elements. Each of these mounts experiences a certain portion of the total engine weight depending on its location relative to that of the center of gravity of the engine. When more than three mounts are used, they are nonsymmetrically distributed with respect to the mass of the power plant, the load on each individual mount can only be approximated. The use of shock-isolating type mounts for the power plant further complicates the force distribution from the power plant to the frame.

5-1.1.1.2 Dead Weight of Body

The body of a vehicle is defined as the passenger- or cargo-carrying portion of the vehicle. The body may be armored, which adds appreciably to its weight. In general, the weight of the body is one of the two largest force resultants acting on the frame when the vehicle is at rest. While the engine may be mounted either at the rear or forward part of the vehicle, consequently greatly af-



S_1, S_2, S_3, S_4	— SUSPENSION REACTIONS
W_{e_1}, W_{e_2}, W_{e_3}, W_{e_4}	— ENGINE LOADS
W_B	— BODY WEIGHT
W_C	— CARGO WEIGHT
W_M	— CREW WEIGHT
W_q	— EQUIPMENT WEIGHT

Figure 5-2. *Frame Loads on Four-Point Suspended Vehicle*

fecting the location of the W_e force vector, the body weight is generally fairly uniformly distributed over the vehicle. Hence W_B, the body force resultant, will be close to the geometric center of the frame (see Fig. 5-2).

5-1.1.1.3 Weight of Cargo

Since cargo weight may vary from less than 5% of the total vehicle weight (in heavy tanks) to more than 80% (in cargo sleds), it is important to examine this force resultant carefully. The total cargo resultant is the sum of the individual cargo components and is expressed by the symbol W_c as shown in Fig. 5-2.

Whatever the shape of the vehicle, certain principles are adhered to in the design of the frame for supporting the cargo. Reinforcing the frame members to sustain bending is essential. The design is arranged to place the cargo directly over the axles to minimize the bending moments induced in the frame. In Fig. 5-3 typical cargo weights are shown with the resulting base reactions.

5-1.1.1.4 Weight of Crew

Since the crew is generally an animate mass, the location of the crew weight force, W_M, is not fixed in time but varies within a relatively restricted area. In the rest position, the crew weights can be considered fixed at the locations ordinarily occupied by the crew when the vehicle is in motion. The crew weight is a relatively minor load factor in truck-type vehicles where the crew consists of only two men. In the case of personnel carriers, however, there may be fifteen men on board, each with his battle equipment. This increases the crew weight to a number well in excess of two tons and, therefore, well worthy of consideration.

5-1.1.1.5 Weight of Equipment

In making the force analysis, the force resultant, W_Q, due to the weight of equipment being carried on the vehicle, can be considered a portion of the cargo weight, W_c, if desired. One must bear in mind, however, that some of this equipment, such as the main and secondary armament, the turret, ground anchors, bulldozer blades, cranes, ammunition, and fuel, represents a considerable weight component and must not be neglected. Furthermore, the weight of this equipment materially affects the location of the vehicle center of gravity and thereby has an influence upon vehicle stability.

5-1.1.1.6 Suspension Reactions

The frame will experience all the force reactions, S_1, S_2, ... from the suspension system. These forces are directed vertically upward in a direction opposite to the imposed total weight force, W_T.

Since these forces are numerically equal to the forces acting on the suspension system itself, they are discussed in more detail in the next section.

5-1.1.2 Forces Acting on the Suspension System

The purpose of the vehicle suspension system is to support the total weight of the vehicle and insure efficient contact between the wheels or tracks and the ground under all operating conditions. Since this involves a multitude of complex requirements, many types of suspension systems have been developed. These are discussed in some detail in Chapter 11. The essential components of these suspension elements, however, are the spring elements, shock absorbers, axles, and wheels, in the case of the wheeled vehicles. In track-laying vehicles, the road wheels and tracks replace the axles and wheels.

When a vehicle is in motion, the forces acting upon the suspension system are quite complex and will be discussed later. At rest, however, particularly on level ground, the force analysis is straightforward and is resolved using the methods of statics. The total weight of the vehicle is resolved into the various components that act, through the frame, upon the spring elements. These, in turn, transmit the loads, through the axle elements, to the wheels, which, in turn, transfer them to the ground.

The suspension systems of track-laying vehicles are designed to distribute the weight of the vehicle over a large ground area in order to reduce sinkage of the vehicle in soft terrain and in an attempt to increase the traction of the vehicle. Various suspension systems for track-laying vehicles are discussed in Chapter 11. In these systems, the vehicle weight is distributed, in turn, through the spring elements, through various systems of links to a series of load-carrying wheels (road wheels), to the track, and then to the ground. By using a large number of road wheels placed as close together as possible, the load applied to each wheel is minimized. The tension in the track gives

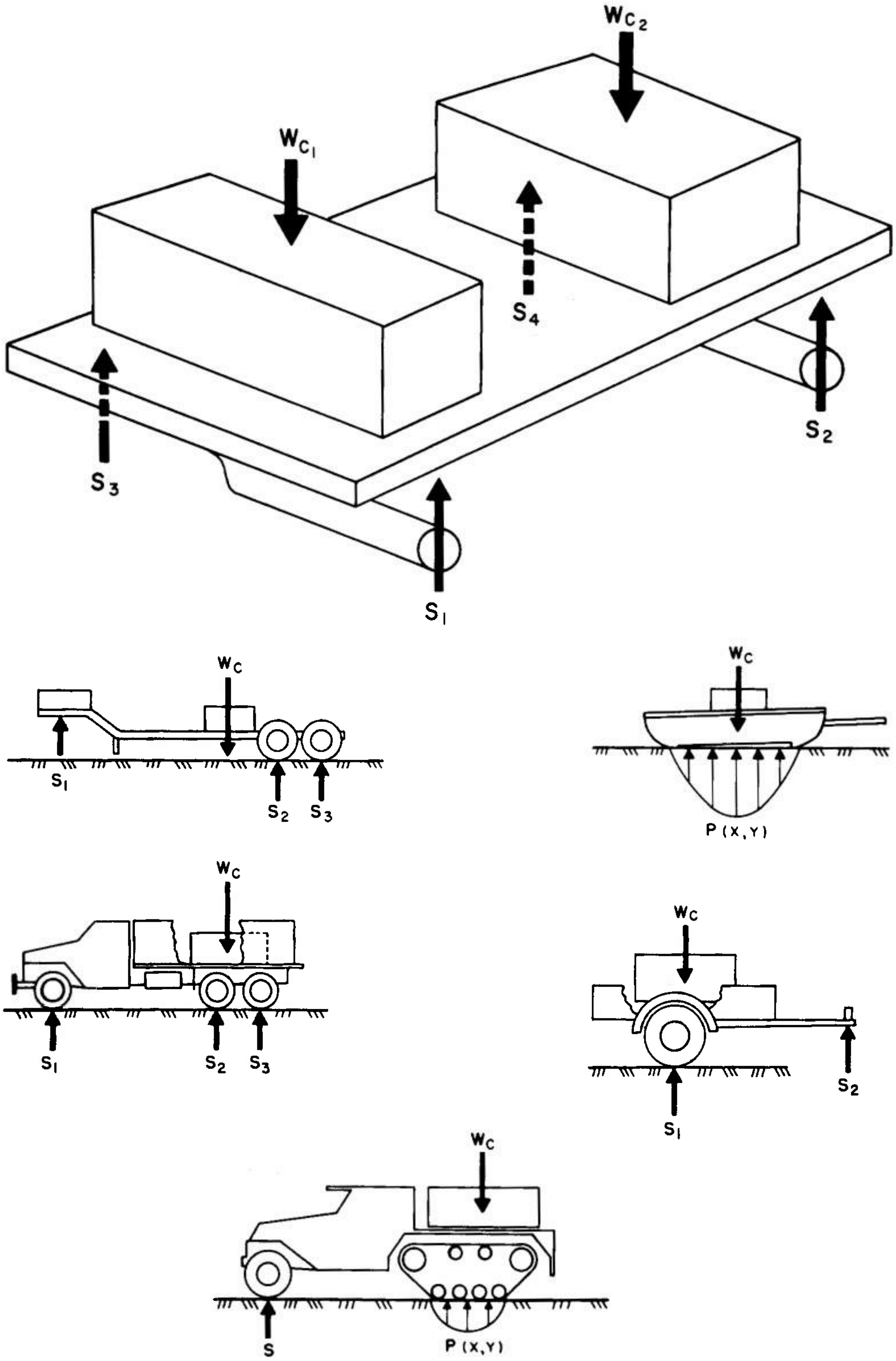


Figure 5-3. Cargo Weight Force Resultants

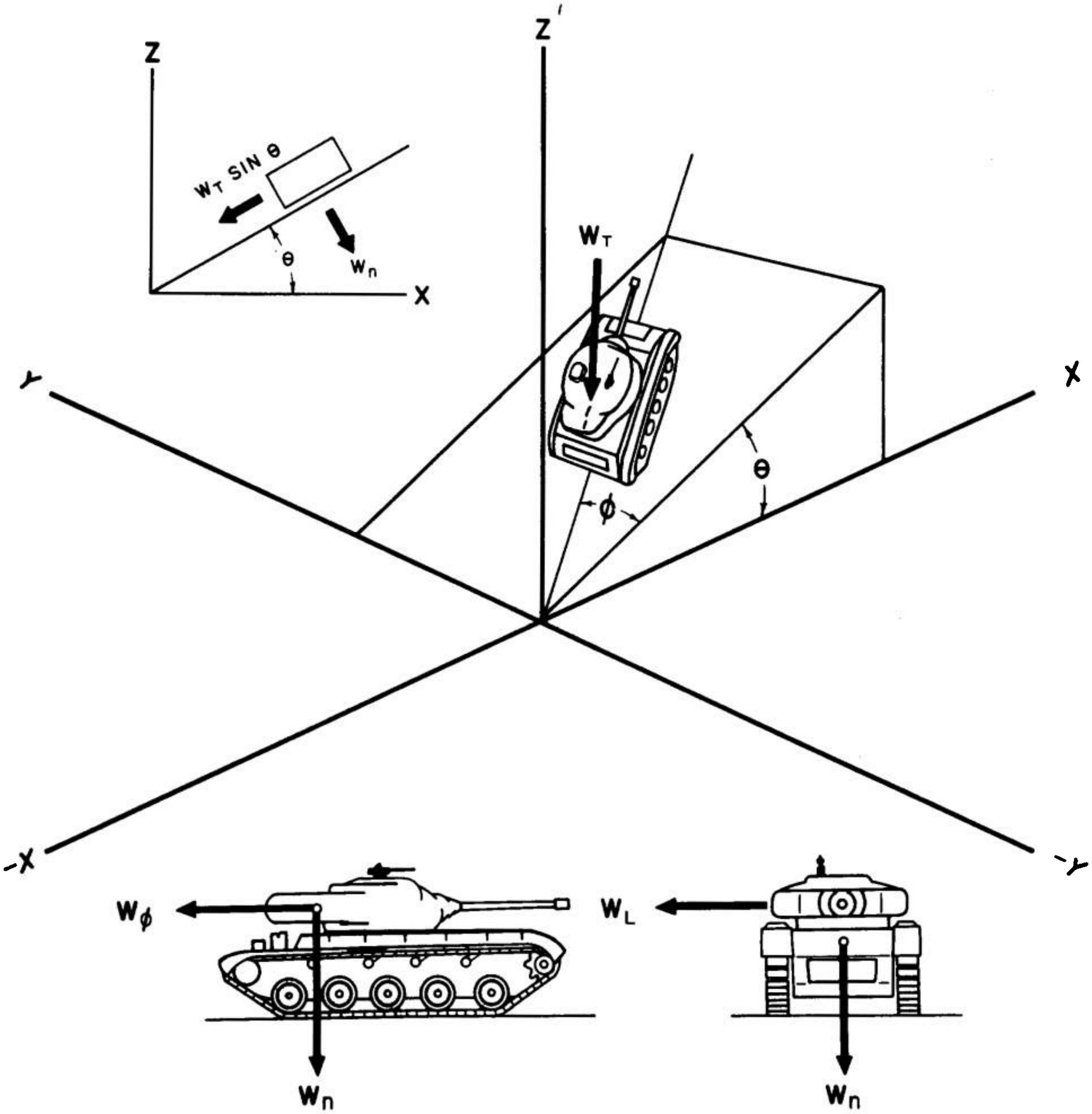


Figure 5-4. ***Static Forces Acting on Vehicle***

it a degree of longitudinal stiffness which permits the track to distribute the load more uniformly.

5-1.2 VEHICLE AT REST IN UNFAVORABLE POSITIONS

The force analysis of a vehicle at rest is carried out according to the ordinary principles of mechanics. The force picture changes quite radically, however, when one compares the situation of a vehicle at rest on level ground with that of the same vehicle at rest on an extreme slope, for example, or cantilevered over a bank, or supported upon two diagonally opposed points.

In order for the force analysis to be complete, the designer must carefully study the force system acting upon his specific vehicle and determine the most unfavorable position that it is likely to encounter. An unfavorable position may be defined as that position in which the vehicle encounters loads or stresses of considerable magnitude. A position which may be extremely unfavorable for one vehicle type may not be the most unfavorable for another type. Critical parameters are, of course, the number and location of contact points and the distribution of the mass of the vehicle. All vehicle positions considered in this section are experienced by the vehicle during operation under normal working conditions and do not include such unusual circumstances as when the vehicle is overturned or buried in mud or where the vehicle is in a position of unstable equilibrium, except for the case of two-point contact when it is assumed that overturning will not occur.

Consider a vehicle at rest on the side of a hill whose slope makes an angle Θ with the horizontal. Consider the vehicle oriented with its longitudinal axis at an angle φ with the X-Z plane as shown in Fig. 5-4. If W_T is the weight force acting on the tank then the components of this force are given by

$$W_n = W_T \cos \Theta \tag{5-1}$$

$$W_\phi = W_T \sin \Theta \cos \varphi \tag{5-2}$$

$$W_l = W_T \sin \Theta \sin \varphi \tag{5-3}$$

where

W_n is the component normal to surface of incline

W_ϕ is the component parallel to surface of incline and along axis of vehicle

W_l is the component normal to vehicle axis and parallel to incline

These components of the body weight force act upon the suspension system in a manner depending upon the distribution and location of the suspension points. The lateral component, W_l, will be the greatest when the vehicle is in a position where $\varphi = \pi/2$, as when the vehicle is traveling on the side of a hill parallel to the base. The suspension system should be reinforced in the longitudinal direction because the longitudinal force component, W_ϕ, provides a force component acting on the suspension in this direction. As a first approximation, neglecting the effect of soil sinkage, this component will be a maximum for $\varphi = 0$ and $\tan \Theta = \mu_0$ where μ_0 is the static coefficient of friction between the soil and the vehicle base.

The most severe loading condition, especially as it affects the vehicle frame and body, occurs when the vehicle is supported at two diagonally opposite points. This two-point contact is the most critical loading the vehicle must sustain. It is best to analyze the vehicle force system as if it were a static case (at rest) and disregard accelerations that reduce the weight on the two contact points. Impact forces which are incurred at this time can be calculated if the velocities are known, but a more realistic method is to use field data on similar vehicles and get some idea of the magnitude of these forces. (See Section II for a discussion of impact forces.) These serious loading conditions must be considered for their effects on vehicle parts. The condition of three-point support is quite common in military vehicles and requires specific attention from the designer.

Conditions of two- and three-point support are unfavorable in that they impose twisting loads upon the vehicle frame. Furthermore, the resulting forces on the springs are greater than when all base supports are in effect. The total upward reaction remains the same, since the downward forces do not change, hence the support reactions must increase to account for the loss of support previously in effect. Furthermore, the springs will feel additional downward force. The unsprung mass at the unsupported locations becomes an additional load for the loaded springs to support, and the weight of the unsprung mass produces a negative (downward) deflection of the spring elements at the unsupported locations. This should not present a problem as far as deflections of the spring elements are concerned, because they are designed to permit deflections both upward and downward from their normal level-standing position. It should be considered in the design, however, from the standpoint of the additional load that it places upon the vehicle frame.

For tracked vehicles, the condition of two-point support is more common since the tracks are continuous. Two-point contact can occur when the tracked vehicle is passing over the crest of a slope and the front and rear of the vehicle are free of contact with the ground.

In the past, and in some instances at present, combat-type, track-laying vehicles were heavily armored. This heavy armor made the vehicle frame so strong and rigid that there was very little need for an accurate force analysis of the vehicle. With the ever-increasing emphasis on lighter weight vehicles, made necessary by the advent of amphibious and airborne tactics, it has become paramount that the designer employ highly accurate, analytical methods so that every pound of material that is specified in the vehicle is justified.

5-1.3 SPECIAL CONDITIONS

5-1.3.1 Vehicle Being Hoisted

Vehicles generally must be transported rather than driven from their place of manufacture to the area where they will be used. During the course of this transportation, there almost always occurs a time when the vehicle is hoisted by a crane or other similar equipment, and during this time the vehicle undergoes some change in the distribution of the forces acting upon it.

Assuming the body remains in a level position during the course of such hoisting, the body weight, W_B, continues to act in the original location at the center of gravity of the vehicle. Furthermore, engine weight, W_e, will also be located in the same place as described previously. The suspension reactions $S_1, S_2, \ldots$, will, however, be changed greatly since they no longer support the weight of the vehicle. These reactions will now support the total weight of the wheels, axles, etc., W_w, or the tracks, half-tracks, or other similar equipment mounted below the frame and attached to the suspension element. There will also be forces directed vertically upward at the location of the supporting ropes or cables which will lift the vehicle. The sum total of these supporting forces, T_i, equals numerically the total vehicle body weight, W_T. In Fig. 5-5 the force system is shown acting on a typical automotive assembly during hoisting (Ref. 3). It is assumed here that the vehicle is either suspended stationary or is moving with a uniform velocity, so that acceleration forces do not add to the force field in existence.

5-1.3.2 Vehicle Suspended During Airdrop

When vehicles are airdropped during an airborne operation, they are first lashed securely onto a suitable platform which supports the vehicle while in the aircraft, facilitates the ejection of the vehicle from the aircraft, and has a major function in absorbing the shock of landing. After deployment of the parachute canopies, the parachute system is connected directly to the vehicle. Thus, the static loading on the vehicle during airdrop is very much the same as when the vehicle is being hoisted, except for the additional weight of the platform. This is added to the weight of the unsprung mass that must be supported by the suspension system during the descent. The shock loads experienced by the vehicle during an airdrop are quite severe. These are discussed in Chapter 3 and in par. 5-8.

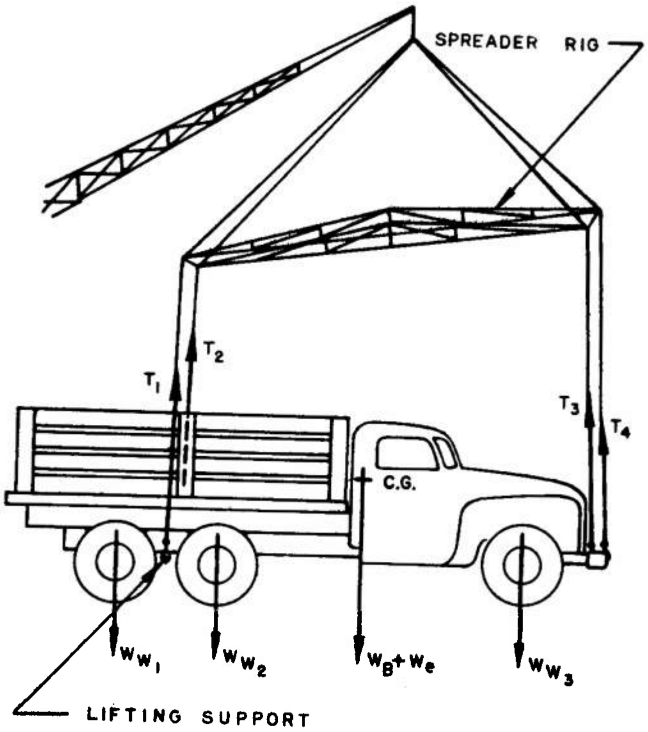


Figure 5-5. Force System Acting on an Automotive Assembly as Suspended During Hoisting

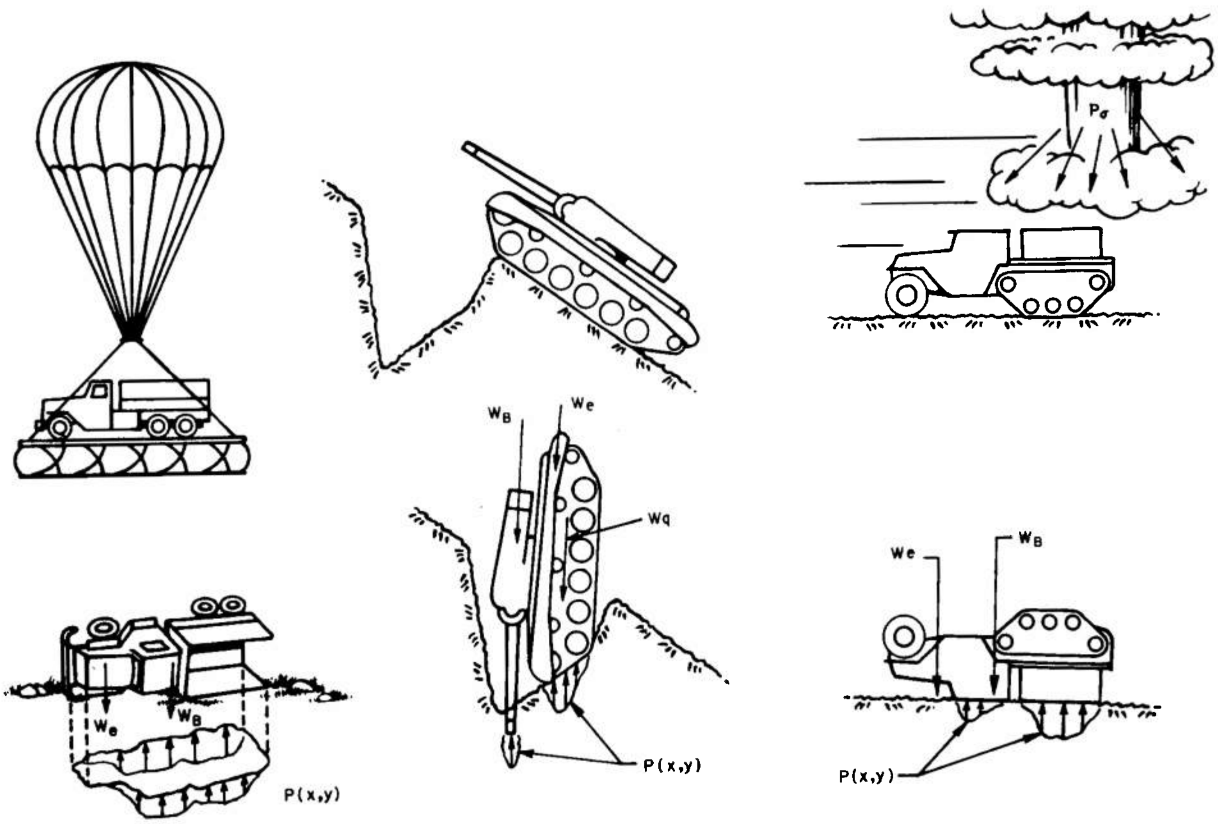


Figure 5-6. *Force Systems Acting on Vehicle (a) On side, (b) Upended, (c) Overturned*

5-1.3.3 Vehicle Overturned, Upended or on Side

Very often, espeically during tactical operations, a vehicle comes to rest in various unorthodox positions. Some of these are illustrated in Fig. 5-6. An airdropped vehicle may land on its side, end, or back and remain in this position until righted. A moving vehicle may encounter a ditch or a raised obstruction; and, in attempting to negotiate this obstacle, end up in one of these positions. Since a military vehicle is required to be operational after surviving such a mishap, it is necessary to study some of these vehicle positions to ascertain how the resulting loads imposed on the vehicle components affect these components.

5-1.3.3.1 Overturned Position

When a vehicle is overturned and comes to rest in the inverted position, the suspension system will not be seriously affected. In this position, the vehicle weight has, in effect, been removed from the suspension system. Similarly, the frame will generally have less forces acting on it than when in the upright position. The engine weight, W_e, acts in a direction opposite to its normal direction but still acts on the frame to which it is attached. The weight of the suspension system, W_s, is also reversed in the direction of action and now acts on the frame. The body or hull is generally loaded more severely by the total weight of the frame which in turn carries $W_e + W_s$. The equipment, cargo, and personnel weights, W_q, W_M, W_c, are eliminated in most cases.

5-1.3.3.2 Upended Position

When the vehicle comes to rest in an upended position, the suspension system is again not seriously affected. It will be subject to some load due to its own weight, and this load will be in a direction perpendicular to the direction of normal loading on the suspension, but none of this is severe. The frame experiences columnar loading by eccentric forces due to W_e, W_q, W_B, and W_s. The force, W_B, may be partially sustained by the ground if the vehicle happened to come to rest with part of the body in contact with the ground. The forces acting on the frame and body must be carefully studied under conditions of upending in order to prevent permanent damage due to excessive deflections. Note that excessive deformations of the frame, body, or hull will affect the operation of other vital components such as the gun traversing and elevating mechanism, fire control equipment, steering system, and automatic loading mechanisms.

5-1.3.3.3 Vehicle on Side

When the vehicle comes to rest on its side, the loading on the suspension system is again not severe. In fact, when a vehicle is properly designed to withstand the rigors of its anticipated service, it will survive any unorthodox position into which it may be placed. Minor damage, such as bent fenders, broken headlamps, or even a smashed radiator may result depending upon the conditions, but no serious damage to the major structure of the vehicle is anticipated. Under conditions of dynamic loading, the picture changes quite radically. Conditions of dynamic loading are discussed later in this chapter. It should be borne in mind that the entire discussion thus far in paragraph 5-1 deals with the vehicle at rest.

The results of atomic tests indicate, in general, that if the vehicle is not blast-damaged but has been simply overturned, righting the vehicle is sufficient to make the vehicle serviceable again in most cases.

5-2 FORCES ACTING ON VEHICLE IN MOTION

When a vehicle is in motion on level terrain, all of the forces that acted upon it while it was at rest continue to act unchanged. Additional forces also come to bear upon the vehicle. These arise from such factors as the tractive effort; resisting forces due to the soil, wind, or towed load; forces due to acceleration of the vehicle; and forces due to engine torques and braking torques.

5-2.1 GROSS TRACTIVE EFFORT

The gross tractive effort is defined as the maximum propelling force that can be developed by the ground-contacting elements of a vehicle on a given type of support. It is the total value before appropriate reductions are made for resistance resulting from sinkage of the wheels or tracks, resistance of the soil, mud, or snow against the underbelly of the vehicle or wind resistance.

5-2.1.1 Cross Country Operations (Refs. 4, 5, 6, 7, 8, 9, 10)

In cross country (off-the-road) operations, the maximum tractive effort is developed when the vehicle's ground-contacting elements (tracks, wheels, feet, etc.) penetrate the surface of the soil to get a firm grip upon the soil itself. Under these conditions, the maximum tractive effort is a function of the shear strength of the soil. When the track cleats (grousers) or the wheel tread do not penetrate the soil, the maximum tractive effort that can be developed is a function of the coefficient of friction between the surfaces in contact.

Since the maximum traction that can be developed is limited by the ultimate strength of the soil on which the vehicle is operating, the method of evaluating the gross traction of a vehicle, on a particular soil, is based on concepts from the field of soil mechanics. However, the present state of this specialized field of mechanics is such that only approximate solutions can be made. These approximate solutions, however, are acceptable. They err in that they neglect the complicated processes of soil consolidation and snow metamorphosis. This neglect is justified, since the time factor of the transient load that the vehicle places upon a particular segment of soil is, relatively, of too long duration to make the load a truly dynamic load, and yet is much too short to produce appreciable compaction. As has been demonstrated repeatedly, the results produced by the approximate methods now available are reasonably accurate.

The physical properties of soils are often described in terms of the "frictional constant" and the "cohesive constant" of the soil. The vertical element (grouser) of a tire tread or track of a moving vehicle develops a horizontal shearing force within the soil. This reaches a maximum value when the soil actually does shear. This maximum value is referred to as the "gross tractive effort."

In plastic or cohesive-type soils, such as wet clay or snow, the gross tractive effort remains constant for a given contact area regardless of the vertical load placed upon it, but varies with the contact area. Thus, for cohesive soils

$$H = Ac \qquad (5\text{-}4a)$$

where

H is the gross tractive effort, lb

A is the area of vehicle footprint, sq in.

c is the coefficient of cohesion of the soil, psi

However, in frictional-type soils, such as dry sand or extremely cold "sugar" snow, the gross tractive effort is found to be independent of contact area and is directly proportional to the vertical load, W. Thus, for frictional soils

$$H = W \tan \varphi \tag{5-4b}$$

where φ is the angle of internal friction of the soil.

Since most actual soils are neither purely cohesive nor purely frictional but a mixture of both types, the gross tractive effort can be calculated from the combined equation:

$$H = Ac + W \tan \varphi \tag{5-5}$$

Well-known long-established methods for determining the values of c and φ are available (Ref. 4). These values must be known before the tractive effort can be calculated. In general, the cohesion factor, c, can have any value from 0 to 3,000 lb per sq ft; although, for most soils, its value is below 1,000 lb per sq ft. The angle of internal friction, φ, varies from 0° to 20° for various clays and may reach values of 50° for some sands, under the proper conditions. Average sands have values of φ between 25° and 35°. A more detailed treatment of this subject, along with a description of apparatus and methods for determining c and φ, can be found in Refs. 5 and 6.

5-2.1.2 Paved Road Operations

The preceding subsection discussed the gross tractive effort developed by a vehicle in cross country operations, or similar operations where the vehicle ground-contacting elements can get a sufficiently firm grip on the ground to take advantage of the shear strength of the soil. Where this condition does not exist, as on extremely hard ground or on hard surfaced roads, the gross tractive effort is a function of the coefficient of static friction between the surfaces in contact. Thus

$$H = W\mu_0 \tag{5-6}$$

where

H is the gross tractive effort, lb

W is the total load on ground, lb

μ_0 is the coefficient of static friction

TABLE 5-1 REPRESENTATIVE VALUES OF COEFFICIENTS OF FRICTION FOR RUBBER TIRES IN VARIOUS PAVEMENTS (Ref. 15)

Type of Pavement	Coefficient of Static Friction, μ_o	Coefficient of Sliding Friction, μ_s
Asphalt or concrete (dry)	0.8 to 0.9	0.75
Asphalt (wet)	0.5 to 0.7	0.45 to 0.6
Concrete (wet)	0.8	0.7
Gravel	0.6	0.55
Earthen road (dry)	0.68	0.65
Earthen road (wet)	0.55	0.4 to 0.5
Snow (hard packed)	0.2	0.15
Ice or sleet	0.1	0.07

Table 5-1 lists some typical values of μ_0 for rubber tires on various pavements which can be used in calculating gross tractive effort.

A large number of factors influence the coefficient of friction. These include surface roughness, tire inflation pressure, tire construction, tread pattern, and speed of vehicle. For this reason, it is almost impossible to standardize coefficient-of-friction values. A value of μ_0 that will allow a sufficient margin of safety to accommodate inevitable side forces should be selected when calculating maximum tractive effort on braking forces. To provide this safety factor, values for sliding friction, μ_s, are used rather than the larger values of static friction, μ_0. Table 5-1 gives values for both μ_0 and μ_s.

Since vehicles are not designed to operate on a single type of road surface, a generalized coefficient of friction must be assumed. It is common to categorize vehicles into two general categories, namely, highway, and off-the-road vehicles. The values commonly used for μ_s for highway vehicles are between 0.6 and 0.7 when calculating tractive or braking effort. When calculating the strength of transmission parts and brake systems, the highest possible values of μ_0 are used. Good practice indicates a choice of μ_0 of 1.0, a value that has frequently been observed under favorable conditions. In off-the-road operations where a different traction principle is applied (par. 5-2.1.1), Eq.

5-6 can be written in terms of a hypothetical, equivalent coefficient of friction where

$$\mu_{eq} = \frac{Ac + W \tan \varphi}{W} \tag{5-7}$$

Under selected conditions, μ_{eq} values greater than 1.78 are conceivable.

5-2.2 RESISTING FORCES

5-2.2.1 Rolling Resistance Due to Sinkage (Refs. 5, 6, 7, 8, 9, 10, 11, 17)

When a vehicle moves over relatively soft terrain, a certain amount of sinkage of the wheels or tracks takes place. This produces a resistive force which acts on the wheel or track, and is due to the work required to compact the soil. If the sinkage becomes too great, this resistive force may become greater than the tractive effort, and the vehicle will stop. The basic equations for calculating the approximate sinkage of any rigid, uniformly-loaded ground-contacting area are

$$z = \left(\frac{p}{k}\right)^{1/n} \tag{5-8}$$

$$k = \left(\frac{k_c}{b} + k_\phi\right) \tag{5-9}$$

$$z = \left(\frac{p}{\frac{k_c}{b} + k_\phi}\right)^{1/n} \tag{5-10}$$

where

- z is the sinkage, in.
- p is the unit ground pressure under contact area, psi
- b is the smaller dimension of the ground contact area, in.
- k is a proportionality constant
- k_c is the cohesive modulus of soil deformation
- k_ϕ is the frictional modulus of soil deformation
- n is an exponent having relation to soil characteristics

If

$$p = \frac{F_s}{b_l}$$

where

- F_s is the load on one wheel or track, lb
- l is the length of ground contact area, in.

then

$$z = \left(\frac{F_s}{l(k_c + bk_\phi)}\right)^{1/n} \tag{5-11}$$

After the amount of sinkage is determined, the resistance to movement brought about by the compaction of the soil can be calculated by the following equation:

$$R_z = \frac{kbz^{n+1}}{n+1} \tag{5-12}$$

where R_z is the resistance to forward movement, lb. When written in terms of k_c and k_ϕ, Eq. 5-12 becomes

$$R_z = \frac{(k_c + bk_\phi)z^{n+1}}{n+1} \tag{5-13}$$

If Eq. 5-11 is substituted in Eq. 5-13, the resistance of a track or low pressure pneumatic tire to forward movement due to soil compaction is

$$R_z = \frac{1}{(n+1)(k_c + bk_\phi)^{1/n}}\left(\frac{F_s}{l}\right)^{\frac{n+1}{n}} \tag{5-14}$$

Equations 5-10 through 5-14 yield reasonably accurate results when applied to self-propelled vehicles having ground-contacting areas that can be considered rectangular, rigid, and uniformly loaded. They do not take into consideration rolling resistance due to the flexing of a tire carcass.

The resistance to rolling due to ground compaction in the case of a rigid wheel is

$$R_z = \left[\frac{1}{(3-n)^{\frac{2n+2}{2n+1}}(n+1)(k_c + bk_\phi)^{\frac{1}{2n+1}}}\right]\left(\frac{3F_s}{D^{1/2}}\right)^{\frac{2n+2}{2n+1}} \tag{5-15}$$

where D is the outside diameter of wheel, in.

The derivation of Eq. 5-15 gives due consideration to the curvature of the wheel, and to the fact that the wheel does not sink to the same depth as a flat plate, as described by Eq. 5-10, but to a depth given by the following approximate equation:

$$z=\left[\frac{3F_s}{(k_c+bk_\phi)(3-n)D^{1/2}}\right]^{\frac{2}{2n+1}} \tag{5-16}$$

Equations 5-10 through 5-16 yield reasonably accurate results when applied to medium types of soils such as those encountered in agricultural and road-building operations. The resistance encountered when operating in soil types yielding a very high sinkage, such as very loose soils resting upon a hard stratum, or extremely liquid muds, must be calculated according to another method (Ref. 5).

5-2.2.2 Rolling Resistance of Elastic Wheel on Rigid Surface

The case of the elastic wheel rolling on a rigid surface has been covered by many investigators. The case of the elastic wheel on a flexible surface is handled in par. 5-2.2.3.

Bekker (Ref. 5) and Spangler (Ref. 12) show the contact area of the elastic tire on rigid and flexible surfaces to be of an oval type such as shown in Fig. 5-7. The width of the contact zone may be determined by the equation

$$\frac{b'}{b}=\frac{\psi(\pi-\delta)-(\pi-\text{arc sin }\psi)\sin\delta}{\pi-\delta-\sin\delta} \tag{5-17}$$

where $\psi = b_r/b$ is the ratio of the hub width to free-wheel width. The relationship between the vertical load acting on the wheel, wheel dimensions, inflation pressure, and tire deflation is given by the following equation

$$F_s=(p_i+p_c)^{\frac{f2}{f+1}}\sqrt{2Dr-2f[(D/2)+r]} \tag{5-18}$$

where

- f is the tire deflation (cm) as shown in Fig. 5-7
- p_i is the tire inflation pressure, kg per sq cm
- p_c is the mean vertical pressure of the tire carcass, kg per squ cm
- D is the major diameter of tire, cm
- r is the radius of tire cross section, cm

No satisfactory analytical method exists at the present time for calculating the resistance to rolling due to deformation of the tire, R_d. A semiempirical formula developed by Kamm (Ref. 13) relates the rolling resistance to the tire inflation pressure, wheel load, and speed in the following way

$$R_d=5.1+\frac{5.5+0.00816F_s}{0.0703p_i}+\frac{8.5+7.05\times10^7\ F_sV^2}{0.0703p_i} \tag{5-19}$$

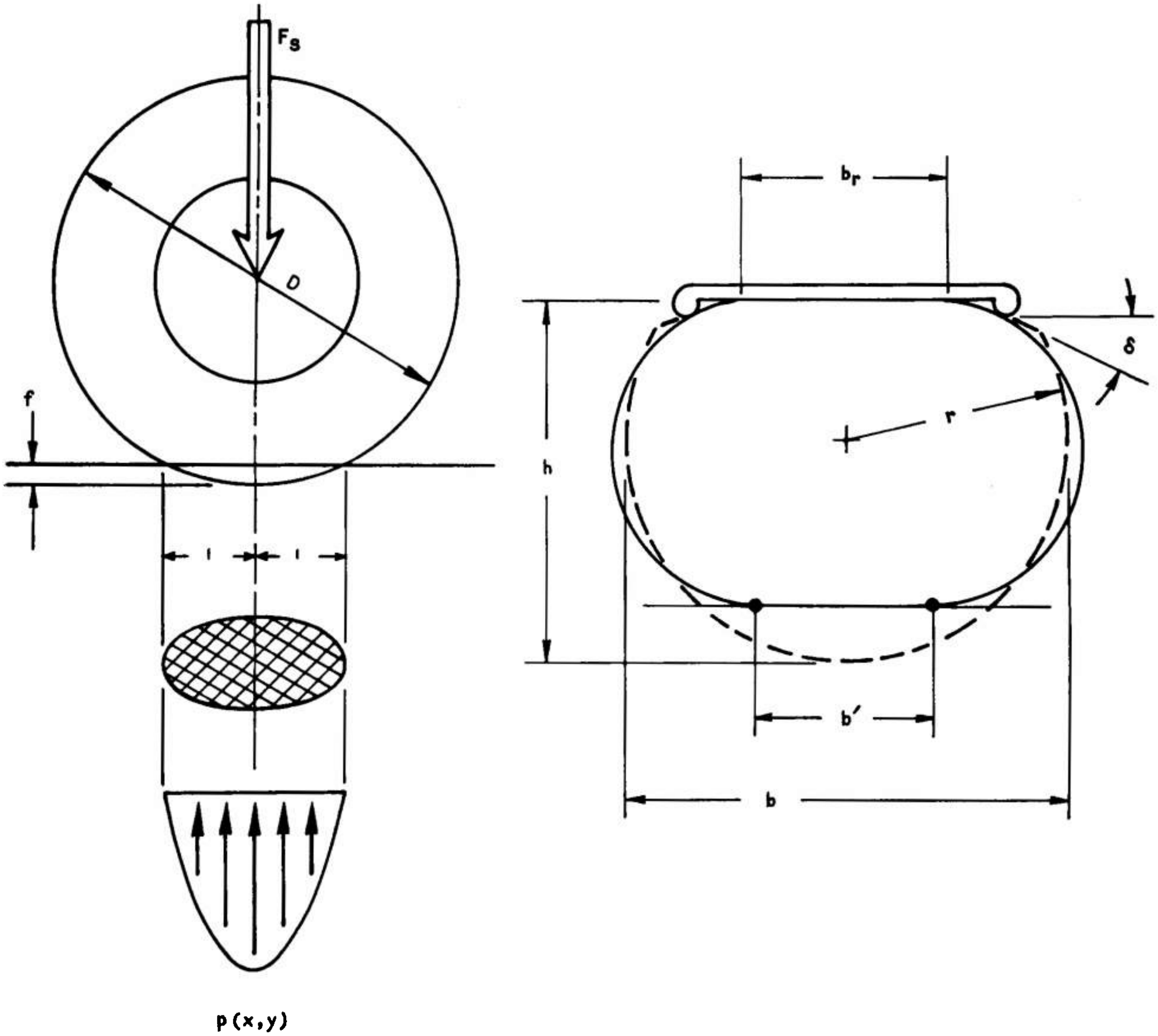
where

- R_d is the rolling resistance per wheel, lb
- F_s is the load on one wheel, lb
- V is the velocity of vehicle, mph
- p_i is the tire pressure, psi

Equation 5-19 was found to give reasonably accurate results when applied to conventional type, passenger car tires at speeds below 95 mph. Its application to unconventional tires, or to speeds in excess of 95 mph, has not been evaluated. Note that F_s is the load per wheel and not total vehicle weight. It the wheels are assumed equally loaded and total weight is used for F_s, then the first term (5.1) must be multiplied by the number of wheels supporting the vehicle.

Investigations performed by Hoerner (Ref. 14) on the rolling resistance of pneumatic tires on passenger cars operating in the range of their design loads showed this resistance to be approximately proportional to the load on the tire. Hoerner obtained satisfactory results from the equation

$$R_d=K_RW_T \tag{5-20}$$



p(x,y)

Figure 5-7. Deformation of an Elastic (Pneumatic) Tire on a Rigid Pavement

where

R_d is the rolling resistance, lb
W_T is the total weight of vehicle, lb
K_R is the rolling coefficient, lb per lb

The rolling coefficient, K_R, is the ratio of resistance to load, or pounds of resistance per pound of the vehicle weight. It depends primarily upon the tire inflation pressure, p_i, and, to a lesser extent, upon the speed of the vehicle. The following semiempirical equation can be used to determine K_R for use with Eq. 5-20.

$$K_R = 0.005 + \frac{0.15}{p_i} + \frac{0.000035V^2}{p_i} \tag{5-21}$$

where

V is the speed of vehicle, mph
p_i is the tire inflation pressure, psi

Equations 5-19 and 5-20 give comparable results when applied to passenger car tires operating at conventional inflation pressures ($\approx$ 25 to 35 psi).

Taborek (Ref. 15) gives the following equation for calculating the rolling resistance of pneumatic tires when operating on concrete surfaces. It treats the rolling resistance as a function of the speed, V, and the inflation pressure, p_i—as do Eqs. 5-19 and 5-20.

$$R_d = fW_T \tag{5-22}$$

$$f = f_o + 3.24 f_s \left(\frac{V}{100}\right)^{2.5} \tag{5-23}$$

where

R_d is the rolling resistance, lb

f is the coefficient of rolling resistance, lb per lb vehicle weight

f_o is the basic coefficient

f_s is the speed coefficient

The values for f_o and f_s are obtained from the curves shown in Fig. 5-8. Figure 5-9 illustrates the effect of the speed and inflation pressure upon the coefficient of rolling resistance, f. The three curves shown are a plot of Eq. 5-23 for three different inflation pressures. Plots of Eq. 5-21 will give similar curves.

5-2.2.3 Rolling Resistance of Elastic Wheel on Soft Ground (Refs. 5, 6, 7, 8, 9, 10, 11, 17)

The resistance encountered by a pneumatic tire in rolling on soft ground is comprised of two main components, namely: R_z, the resistance due to the compaction of the ground; and R_d, the resistance due to the deformation or flexure of the tire carcass. The total resistance being:

$$R = R_z + R_d \tag{5-24}$$

The stiffness of the tire carcass depends upon such structural characteristics of the tire as number and type of plies, thickness of wall, material used and direction and density of fabric. This stiffness factor is not negligible, and assists materially in supporting the wheel load. Thus, the ground pressure, p_g, beneath the tire is equal to the sum of the inflation pressure, p_i, and the ground pressure due to carcass stiffness, p_c, or

$$p_g = p_i + p_c \tag{5-25}$$

The ground pressure, p_g, can be easily determined for a given tire under a given load by dividing the wheel load, F_s, by the area of the tire print. By subtracting the inflation pressure, p_i, from p_g, the ground pressure due to carcass stiffness, p_c, can be determined.

The rolling resistance of a low-pressure pneumatic tire can be calculated from the equation

$$R = \frac{[b(p_i + p_c)]^{\frac{n+1}{n}}}{(k_c + bk_\phi)^{1/n}(n+1)} + \frac{F_s u}{p_i a} \tag{5-26}$$

where u and a are empirical coefficients that relate to the tire stiffness and must be evaluated experimentally for each tire. This can be done by rolling the tire on a hard surface under different loads and inflation pressures and measuring the resistance to rolling, which, in this instance, will be R_d.

When considering the rolling resistance of a pneumatic tire, one must decide whether to use Eq. 5-15, which applies to a rigid wheel, or Eq. 5-26, which applies to a low-pressure pneumatic tire. The distinction depends partly upon the magnitude of the inflation pressure, for with sufficient pressure a tire will obviously acquire the characteristics of a rigid wheel, but the distinction also depends upon the soil characteristics. When the ground is relatively strong and the tire is flattened in the contact area by the wheel load, the ground pressure, p_g equals the inflation pressure plus the pressure due to carcass stiffness, $p_g = p_i + p_c$, and the tire is considered a low-pressure tire and behaves in a fashion

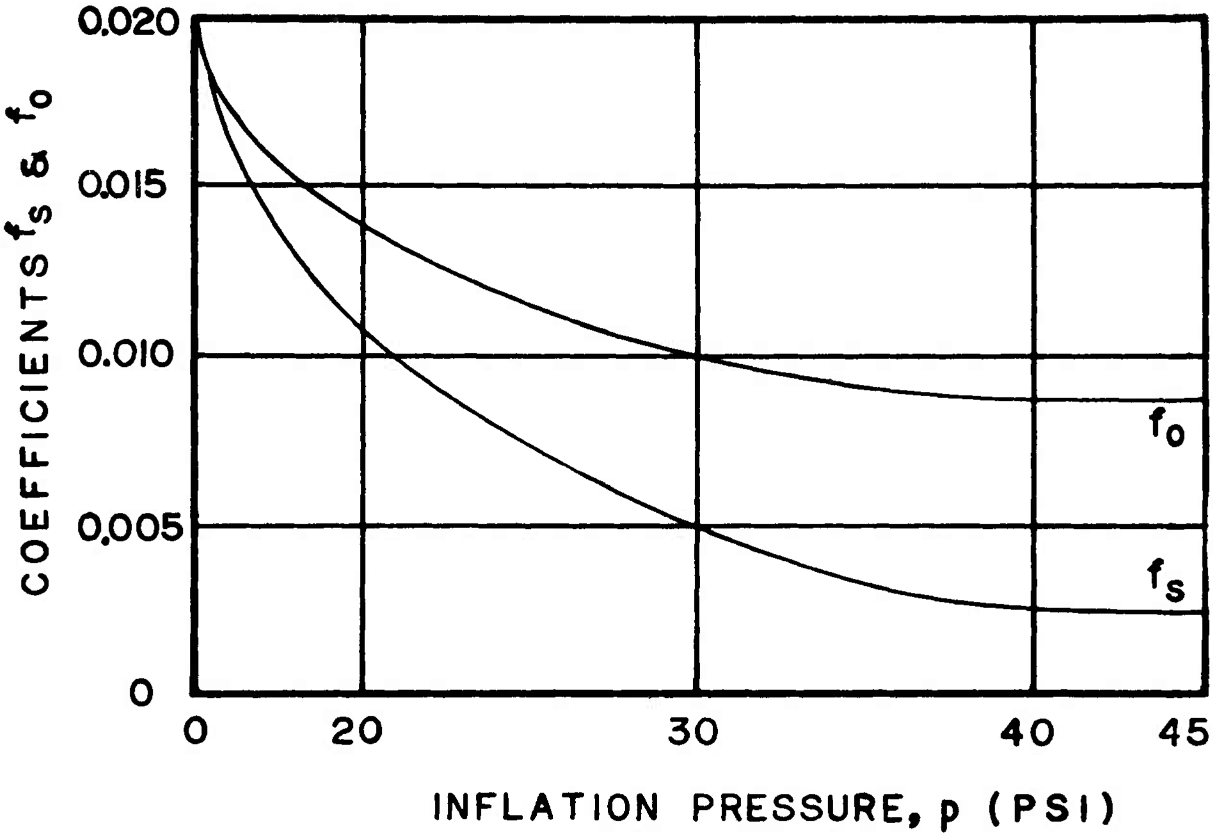


Figure 5-8. Coefficients f_s and f_o (From "Mechanics of Vehicles" by Jaroslav J. Taborek, Machine Design, July 25, 1957)

similar to a track. On the other hand, when the ground is relatively weak, the sum of p_i plus p_c will be greater than p_g and the tire will remain round and behave in a fashion similar to a rigid wheel. The critical inflation pressure, $(p_i)_c$, above which the tire behaves as a rigid wheel and below which it develops a flat ground-contact area, cannot be given as a definite value, since it depends upon the relationship between the wheel load, wheel-tire dimensions, the structural characteristics of the tire, and the characteristics of the soil in which the wheel is operating. This critical inflation pressure can be calculated using the following equation

$$(p_i)_c = \left[\frac{F_s(n+1)}{b\left[\frac{3F_s}{(3-n)bkD^{1/2}}\right]^{\frac{1}{2n+1}}}\right]\left\{D - \left[\frac{3F_s}{(3-n)bkD^{1/2}}\right]^{\frac{2}{2n+1}}\right\}^{-1/2} - p_c \tag{5-27}$$

where
$$k = \left(\frac{kc}{b} + k_\phi\right)$$

D is the outside diameter of wheel, in.

5-2.2.4 Aerodynamic Drag

Aerodynamic drag (Refs. 13, 14, 15, 16) or the resistance of the air surrounding the vehicle, is an important factor in the design of vehicles, particularly at high speeds. During the propulsion and acceleration phases of vehicle operation, it is a resistance that must be overcome by the power plant;

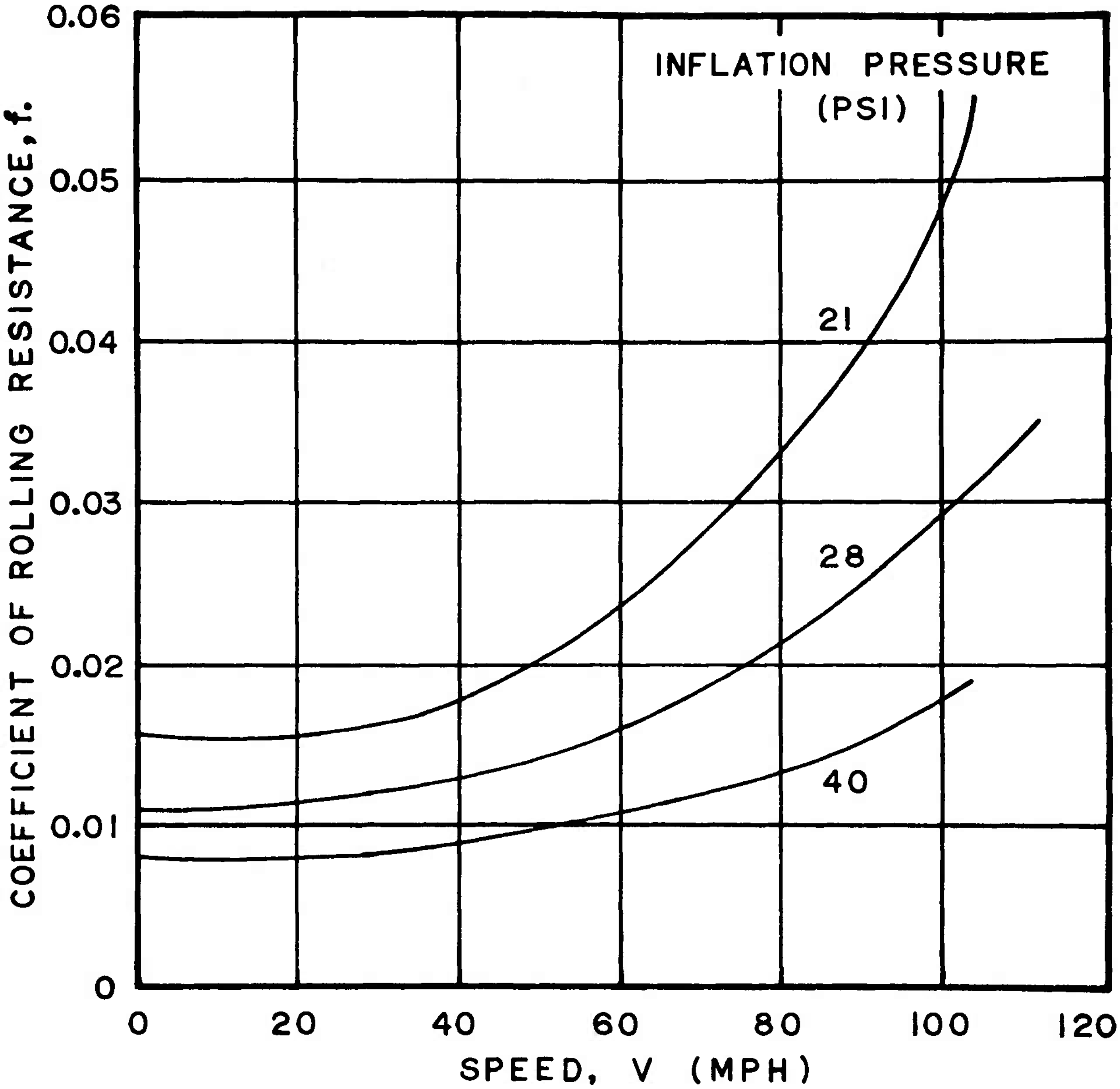


Figure 5-9. Coefficient of Rolling Resistance as a Function of Speed and Inflation Pressure (From "Mechanics of Vehicles" by Jaroslav J. Taborek, Machine Design, July 25, 1957)

while during deceleration of the vehicle, the air resistance serves as an additional braking force. The power requirements to overcome aerodynamic drag are discussed in Chapter 7. Aerodynamic drag, R_a, can be calculated using the following equation

$$R_a = C_D 1/2 \rho V^2 A_a \tag{5-28}$$

R_a is the aerodynamic drag, lb

ρ is the mass density of the air, slugs per cu ft

V is the relative velocity between vehicle and air in direction of motion, fps

A_a is the projected frontal area, sq ft

C_D is the drag coefficient

The drag coefficient, C_D, is a nondimensional fluid-dynamic coefficient related to the shape of the vehicle, roughness of the vehicle surface, and the relative number of proturberances extending into the air stream. The largest drag coefficients are obtained with (a) open convertible-type passenger cars, (b) blunt box-like shapes such as are common with van-type trucks and trailers, and (c) dump trucks, whose general shape is blunt and square, and whose open-topped cargo bodies create a great deal of turbulence. Aerodynamic drag can be appreciably reduced by such measures as rounding the front end of a blunt-nosed vehicle, rounding the lateral edges, or adding a tapering tail section to the vehicle. Rounding the rear end has practically no effect in reducing the aerodynamic drag.

The table compiled from works by Bekker (Ref. 5), Kamm (Ref. 13), Hoerner (Ref. 14), Taborek (Ref. 15), and others, gives some typical values of the drag coefficient, C_D, for representative vehicles.

TABLE 5-2 TYPICAL AERODYNAMIC DRAG COEFFICIENTS FOR REPRESENTATIVE VEHICLES

Type of Vehicle	Drag Coefficient, C_D
Extremely streamlined shape	0.13
Sphere	0.47
Square plate	1.2
Standard sedan cars	0.45 to 0.75
Open convertible passenger car	0.85 to 0.95
Van-type trailers	0.46 to 0.86
Buses	0.45 to 0.93
Trucks	0.8 to 2.1
Tractors and trailers	1.3 to 2.1

5-2.2.5 Grade Resistance

The weight, W_T, of a vehicle moving up a slope that makes an angle, φ, with the horizontal is resolved into two components, one normal and one parallel to the slope. The parallel component is directed downhill, and therefore constitutes a grade resistance, R_g, acting at the vehicle center of gravity, that the vehicle must overcome.

$$R_g = W_T \sin \varphi \tag{5-29}$$

Grades are customarily designated in percentage terms e.g., 20% grade. The designation represents the ratio of vertical distance to horizontal distance expressed as percentage. Thus, a 20% grade represents a rise of 20 ft in a horizontal distance of 100 ft, or an angle φ of approximately 11½°. Thus, the grade G can be expressed as

$$G = 100 \tan \varphi \tag{5-30}$$

For small angles, $\sin \varphi \approx \tan \varphi$, an equation for grade resistance becomes

$$R_g \approx W_T \tan \theta = \frac{W_T G}{100} \tag{5-31}$$

Equation 5-31 is usually commonly used in the design of civilian vehicles where steep slopes are not encountered. Grades of modern superhighways are kept below 6%, highways in mountain areas have uphill grades below 7%, and the steepest known slopes in high mountains do not exceed 32%. Such extreme slopes extend for only short distances. Even when applied to a 32% slope, Eq. 5-31 results in an acceptable error of 5% Military vehicles however are designed to negotiate 60% slopes. At these extreme slopes, Eq. 5-31 results in excessive error and should not be used. Equation 5-29 yields correct results.

5-2.2.6 Inertia Resistance

Inertia is the property of a mass to resist any effort to alter its existing state of motion, and becomes manifest as an opposing force (inertia force) that is proportional to the produce of the mass, m, and its time-rate-of-change of velocity (acceleration, a). An automotive vehicle encounters inertia resistance, R_i, when it is accelerated which it must overcome by additional power from the power plant. It again encounters inertia resistance when it is decelerated, which must be overcome with the braking system. The point of action of the inertia resistance is located at the center of gravity of the vehicle mass.

The total inertia resistance of an automotive vehicle consists of two parts, the inertia of the translating mass, R_{it}, and the inertia of the rotating parts, M_i. Thus,

$$R_i = R_{it} + M_i \tag{5-32}$$

The inertia resistance of the translating mass can be calculated by the basic equation

$$R_{it} = ma = \frac{W_T a}{g} \tag{5-33}$$

where

W_T is the total weight of vehicle, lb

a is the acceleration of vehicle, ft per sec per sec

g is the acceleration due to gravity, ft per sec per sec

Since all translatory motion of an automotive vehicle is coupled to the rotational motion of the wheels and rotating parts of the power train, any change in translatory velocity of the vehicle is accompanied by a simultaneous change of rotational velocity of these rotating components. These changes of rotational velocities (angular accelerations) give rise to resisting inertia torques, M_i. The basic equation for calculating torque is

$$M = I\alpha \tag{5-34}$$

where

I is the moment of inertia of the rotating parts, ft-lb-sec^2

α is the angular acceleration, radians per sec per sec

The total inertia torque of an automotive vehicle is the summation of the individual resisting torques of all of the rotating parts. This can be related to the drive axle by applying the appropriate ratio between the part under consideration and the drive axle

$$M_i = \Sigma M\zeta = \Sigma I\alpha\zeta \tag{5-35}$$

where ζ is the ratio between rotating part and drive axle.

If $\alpha = \alpha_d \zeta$ is substituted in Eq. 5-35 (where α_d is the angular acceleration of drive axle, radians per sec per sec), the resulting equation becomes

$$M_i = \alpha_d \Sigma I \zeta^2 \tag{5-36}$$

Equation 5-32 can be rewritten as a combination of Eqs. 5-33 and 5-36 to become

$$R_i = \frac{W_T a}{g} + \alpha_d \Sigma I \zeta^2 \tag{5-37}$$

In order to simplify the calculation of M_i, an equivalent mass, m_e, is determined which, when added to the translating mass, m, of the vehicle, will result in an effective inertia mass, m', that can be substituted into the basic equation for translatory inertia (Eq. 5-33) to obtain the total inertia resistance, R_i, or:

$$R_i = m' a \tag{5-38}$$

$$m' = m + m_e \tag{5-39}$$

$$m_e = \frac{\Sigma I \zeta^2}{r^2} \tag{5-40}$$

Equation 5-39 is often written as a product of the vehicle mass and a dimensionless mass factor, γ, which defines how many times the effective mass, m', exceeds the actual mass of the vehicle. Thus, Eq. 5-39 becomes

$$m' = m\gamma \tag{5-41}$$

and Eq. 5-38 becomes

$$R_i = \gamma m\, a \tag{5-42}$$

The mass factor, γ, can be calculated from the equation

$$\gamma = 1 + \frac{m_e}{m} = 1 + \left(\frac{\Sigma I_w}{mr^2} + \frac{\Sigma I \zeta^2}{mr^2} \right) \tag{5-43}$$

where

ΣI_w is the moment of inertia of wheels and other major components rotating at velocity of the wheels

ΣI is the moment of inertia of major components rotating at engine speed

Table 5-3 lists some average values of the mass factor, γ, for three types of wheeled vehicles operating at full load in different speed ranges. The values given can be used as a guide in preliminary calculations.

TABLE 5-3 AVERAGE VALUES OF MASS FACTOR, γ (Ref. 15)

Transmission Gear Setting	Type of Vehicle		
	Passenger Car (Large)	Passenger Car (Small)	Truck (Civilian)
Low	—	2.40	2.50
First	1.30	1 50	1.60
Second	1.14	1.20	1.20
High	1.09	1.11	1.09

5-2.2.7 Summary

Several observations which result from a comparison study of the various resisting forces acting upon a vehicle in motion are worth noting:

a. Rolling resistance and aerodynamic drag are always present when the vehicle is in motion. The gross tractive effort is used primarily to overcome these two resistances. The vehicle will be able to climb a grade (overcome grade resistance) or accelerate (overcome inertia) only when the gross tractive effort is in excess of that required to overcome rolling and air resistance.

b. Rolling resistance, grade resistance, and inertia resistance are proportional to the vehicle weight. Air resistance is the only resistance factor that is not directly associated with weight. Indirectly, however, even air resistance has a relationship to the vehicle weight, since the size of the frontal area and the shape of the vehicle affect the coefficient of air resistance, which influences aerodynamic drag. High capacity vehicles are usually designed for heavy loads and lower speeds and do not require aerodynamically clean body shapes.

c. Rolling resistance and aerodynamic drag are functions of the speed of the vehicle. Grade resistance and inertia resistance are independent of vehicle speed. Inertia resistance is proportional to the acceleration of the vehicle.

d. A comparison of the rolling resistance and areodynamic drag acting on lightweight passenger-type vehicles with heavy, cargo-types at equivalent speeds shows that, for both types of vehicles, the rolling resistance accounts for the greater proportion of the total resistance at low speeds, while the aerodynamic drag accounts for the greater proportion of the total resistance at high speeds. Since the heavy vehicle is designed to overcome a considerably greater total resistance than is the light passenger-type vehicle, the aerodynamic drag constitutes an appreciably smaller percentage of the total resistance acting on the heavy vehicle than is the case with light, passenger-type vehicles at the same speed, even though the body shape of the latter vehicle is aerodynamically superior. It has been shown (Ref. 15) that aerodynamic drag is practically negligible (less than 30% of rolling resistance) below 20 mph for passenger cars and below 35 mph for trucks. At approximately 35 mph for passenger cars and 55 mph for trucks, the aerodynamic drag will be equal to the rolling resistance; while at 70 mph, aerodynamic drag accounts for approximately 80% of the total resistance acting on the passenger car, but only 60% of that acting on the truck. This justifies emphasis placed upon the aerodynamic design of passenger car bodies.

5-2.3 DYNAMIC AXLE WEIGHT (Ref. 15)

The gross tractive effort that a vehicle is capable of developing is a function of a friction factor, related to the nature of the surfaces in contact, and an effective weight factor acting in a direction normal to the surfaces in contact (par. 5-2.1). When this weight factor is determined by actually measuring the load on each wheel, or by calculating the load on each wheel from the location of the vehicle center of gravity, the resulting wheel or axle loads apply only to the stationary vehicle. When the vehicle is in motion, it is acted upon by the various motion-resisting forces that have just been discussed. These forces produce moments that result in a weight shift toward one of the axles, with the ultimate result that axle loadings on a moving vehicle will be appreciably different from those acting on the same vehicle when stationary. This resulting effective axle load is termed "dynamic axle weight", W_b, and is the factor which ultimately determines the gross tractive effort developed by the vehicle. All other vehicle performance characteristics, such as acceleration, gradeability, speed, and drawbar pull, are directly dependent upon the gross tractive effort.

When calculating dynamic axle weights, it is customary to set up moment equations about the wheel-ground contact points. By doing this, all forces acting in the plane of the ground, such as the rolling resistance, tractive force, inertia resistance of rotating parts, are eliminated since they do not form a moment with respect to the wheel-ground contact point.

An analysis of the factors that influence dynamic axle weight shows the following relationships:

a. The normal-to-the-ground force component, W_n, of the total weight, W_T, of a vehicle ascending (or descending) a slope inclined at an angle, Θ, to the horizontal varies as the cosine of the slope angle, Θ; i.e.,

$$W_n = W_T \cos \Theta \qquad (5\text{-}44)$$

It is customary in design of civilian automotive vehicles, where grades seldom exceed 7° and never exceed 18°, to make the assumption that $\cos A \approx 1$. This permits the simplification

$$W_n \approx W_T \cos \Theta \approx W_T \qquad (5\text{-}45)$$

This simplification should be applied with discretion to the design of military vehicles which must be able to negotiate extremely steep grades. Equation 5-45 will result in an error of 14% when applied to a vehicle on a 60% grade.

b. The grade resistance, R_g, experienced by a vehicle on an incline produces a weight increase on the downhill axle, and a corresponding decrease on the uphill axle, proportional to the sine of the slope angle, Θ, and proportional to the height of the vehicle center of gravity.

c. The effect of aerodynamic drag, R_a, normally tends to increase the weight experienced by the rear axle by an amount proportional to the height of the center of pressure. At high speeds, however, the vehicle experiences aerodynamic lifting forces of unpredictable characteristics. These tend to decrease the loads experienced by the axles, particularly of the front axle. The shape of high-speed vehicles is often designed to create a downward component of the aerodynamic resistance force that will balance the lifting force and provide additions to the loads experienced by the axles.

d. Inertia resistance, R_i, acts at the center of mass of the vehicle and its effect is proportional to the height of the center of mass. During accelerated motion, the inertia resistance causes an increase in weight experienced by the rear axle with a corresponding decrease in weight on the front axle. During decelerated motion an opposite weight shifting occurs.

e. The drawbar pull acts on the pintle or towing bar and produces an increase to weight experienced by the rear axle proportional to the height of the drawbar action point.

5-2.4 VEHICLE BRAKING (Refs. 15, 16)

5-2.4.1 Braking Force

Figure 5-10 shows the forces that act upon a decelerating vehicle on a downhill grade. A complete analysis of the braking vehicle requires that all of the forces and moments related to the vehicle's state of motion be studied. All of these factors are not equally significant; some, as will be pointed out later, can be safely omitted.

The primary decelerating force is the braking force (B), which results from the frictional engagement of the brake shoes and brake drums when the vehicle brakes are applied. The following equation shows the basic relationship between the factors involved.

$$B = \frac{F_b \mu_b r_b}{r} \qquad (5\text{-}46)$$

where

B is the total braking force, lb

F_b is the total effective force between brake shoes and brake drums, lb

μ_b is the coefficient of sliding friction between brake shoe and drum

r_b is the radius of brake drum

r is the radius of the tire

The total braking force is the sum of the braking forces developed by all of the vehicle's axles, or

$$B = B_f + B_{r1} + B_{r2} \qquad (5\text{-}47)$$

where B_f represents the braking force developed by the front axle and B_{r1} and B_{r2}, the braking forces developed by the two rear axles.

Despite the maximum braking force that can be developed by the brake shoes and drums, the ultimate braking force for the entire vehicle depends upon the efficiency of the connection between the vehicle and the ground. The efficiency of this connection is governed by the same physical relationships that govern the gross tractive effort (par. 5-2.1). The maximum braking force is given by the expression

$$B_{max} = W_b\,(\mu_o + f) \qquad (5\text{-}48)$$

where

W_b is the total dynamic axle weights, lb

f is the coefficient of rolling resistance (see pars. 5-2.2.1 to 5-2.2.3, inclusive).

μ_o is the coefficient of static friction (see Table 5-1).

Equation 5-48 applies to the maximum braking force that can be developed by a vehicle operating on a hard, paved surface. It can be modified for application to a vehicle operating on soft ground by replacing μ_o with μ_{eq} (see Eq. 5-7 of par. 5-2.1.2) and by making the appropriate substitutions for rolling resistance due to sinkage and tire deformation as discussed in pars. 5-2.2.1 through 5-2.2.3.

If the wheels become locked during braking, the coefficient of static friction, μ_o, is replaced by the coefficient of sliding friction, μ_s, and the rolling resistance factor disappears from the equation. The maximum braking force under a locked wheels condition becomes

$$B_{LW} = W_b\,\mu_s \qquad (5\text{-}49)$$

5-2.4.2 Dynamic Weight Transfer

During the braking of a vehicle, the interaction of the braking forces and the inertia of the translating mass sets up a couple which increases the load acting upon the front axle and decreases the load acting upon the rear axle by a corresponding amount. This dynamic axle weight has a basic influence upon the performance of the brake system and upon the optimum distribution of the braking forces among the different axles of the vehicle. The relative sizes of the brakes are often varied for the different axles in order to equalize the braking forces developed by each axle. Furthermore, since the translatory inertia force acts at the vehicle center of gravity, which is separated from the axles by the flexible suspension system, this dynamic weight transfer produces an undesirable longitudinal instability of the vehicle body. Various stabilizing systems are used to maintain the vehicle body level. These serve their purpose but do not counteract the dynamic weight transfer on

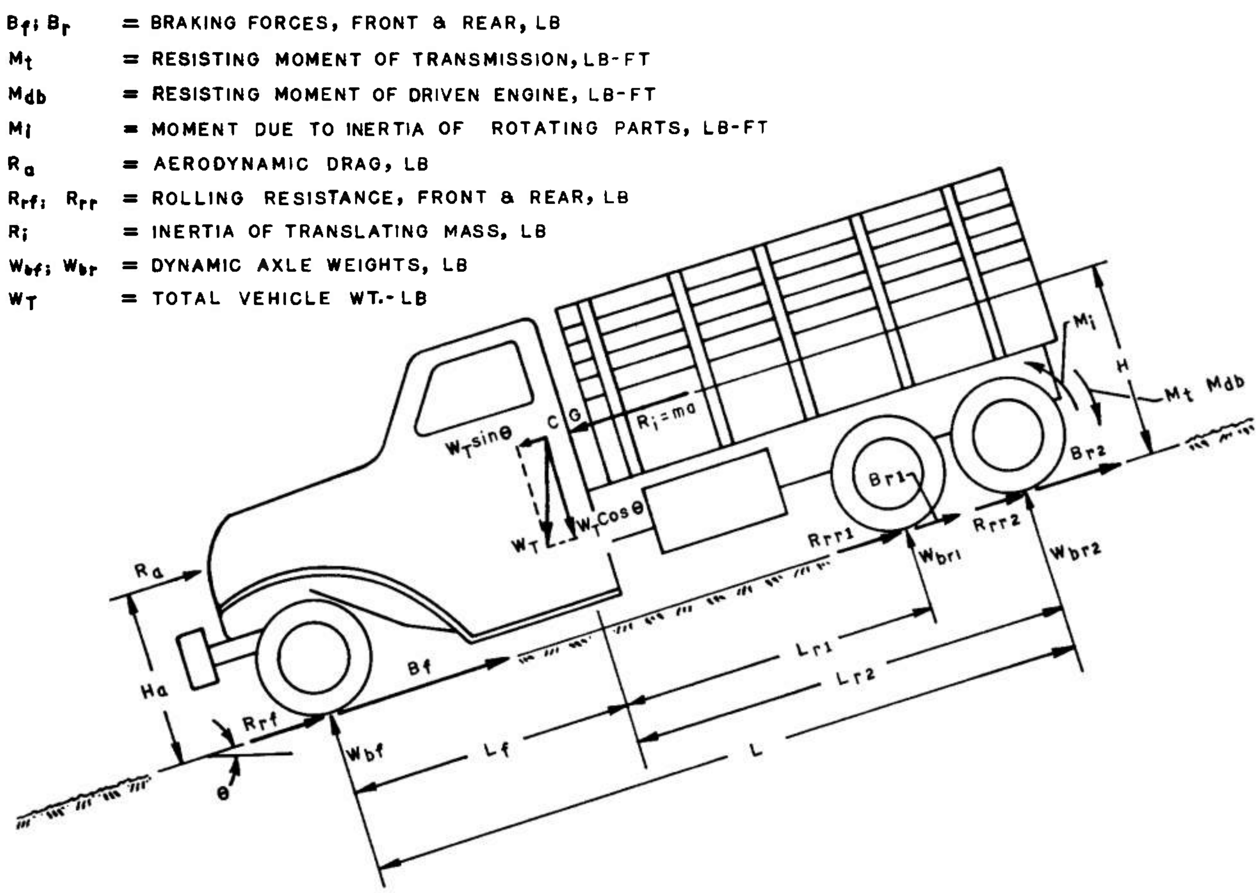


Figure 5-10. Force System Acting on an Automotive Vehicle Braking on a Downgrade (From "Mechanics of Vehicles" by Jaroslav J. Taborek, Machine Design, Nov. 14, 1957)

the axles. Dynamic weight transfer can be minimized by keeping the center of gravity of the sprung mass as low as possible.

5-2.4.3 Grade Effect

When calculating the braking force developed by a vehicle on a grade (Fig. 5-10), only the normal-to-ground, or cosine, component of the vehicle weight is significant. In civilian practice, it is customary to neglect this cosine factor, since cosine $\Theta \approx 1$ for small values of Θ. In the design of military vehicles, however, where 60% grade capabilities are specified, neglecting the cosine factor can result in appreciable error.

The sine component of the vehicle weight is the grade resistance, R_g, discussed in par. 5-2.2.5. Note that this is a motion-retarding force for a vehicle on an upgrade but is motion-supporting on a downgrade. Braking calculations should always treat the vehicle on a downgrade and the brake capacity designed to include the effect of the sine component.

5-2.4.4 Aerodynamic Drag

Aerodynamic drag, R_a, is discussed in pars. 5-2.2.4 and 5-2.2.7. Its influence as a decelerating force is small at the normal speeds of military vehicles. Furthermore, since it is proportional to the square of the velocity (Eq. 5-28), its effect decreases during the vehicle's deceleration. For these reasons the decelerating effect of aerodynamic drag is usually neglected in braking calculations. This simplification provides a safety factor, especially at high speeds where it is needed most.

5-2.4.5 Inertia

Inertia, discussed in par. 5-2.2.6, was treated primarily as a resistance to acceleration. In a braking vehicle, the effect of inertia is to impede braking. The equations given in par. 5-2.2.6 for calculating the inertia force are applicable to a braking vehicle with one important modification necessary to Eq. 5-41, namely, the value of the mass factor γ which relates to the inertia effects of the rotating parts. The value of γ_b used in braking calculations is different from the γ used in acceleration calculations in that γ_b should not include the inertia effects of the engine and flywheel, since the engine is often declutched during braking. This leaves only the wheels and transmission to he decelerated. Thus, Eq. 5-41 should be used in the following form in braking calculations to avoid error

$$R_i = \gamma_b mb \tag{5-50}$$

where

γ_b is the braking mass factor
m is the mass of vehicle, lb-sec^2 per ft
b is the deceleration, ft per sec per sec

In the design of conventional, civilian automotive vehicles, a value for γ_b of 1.04 is used in preliminary calculations. A corresponding value for military vehicles is not available at this time.

5-2.4.6 Transmission Resistance

The transmission resistance, M_t, is the torque required to overcome the frictional resistance of the gear trains, bearings, and miscellaneous shaft connections, plus the torque required to rotate the gears against the resistance of the oil in which they are immersed. When calculating the power required to propel and accelerate the vehicle, allowance is made for power lost to transmission resistance by applying a transmission-efficiency factor. When braking, the transmission resistance helps to dissipate some of the kinetic energy of the moving vehicle, and thus adds to the total braking effort. This additional braking effort is small in comparison to the primary braking effort developed by the braking system and is usually omitted from braking calculations. The omission has a bonus effect on the braking system.

5-2.4.7 Engine Braking

Under certain conditions, the vehicle is decelerated by throttling down the engine to its idling speed, causing a reduction or cessation of power to the drive wheels. Since the clutch normally remains engaged during this deceleration, a reverse flow of power takes place. The kinetic energy of the moving vehicle must overcome system friction, inertia, and cylinder compression forces that continue to act upon the pistons. This serves to dissipate an appreciable amount of the vehicle's kinetic energy and the engine thereby functions as a rather effective brake. The engine-braking effort is proportional to the speed at which the engine is driven, and, therefore, to the reduction ratio between the engine and wheels. For this reason, heavy vehicles on long downhill grades depend upon the braking power of their engines to aid in braking, and thereby reduce the amount of heat generated by the brake drums.

The torque required to drive the engine, M_e, is measured experimentally at the engine output shaft. By applying the reduction ratio, ζ, between the engine output shaft and the drive axle and the transmission-efficiency factor η, the effect of the engine-braking torque on the drive axle can be calculated as

$$M_{db} = \frac{M_e \zeta}{\eta} \tag{5-51}$$

5-2.5 CONSIDERATIONS IN THE DESIGN OF AMPHIBIOUS VEHICLES

The force system acting on an amphibious vehicle traveling through the water is essentially the same as that acting on any marine vessel. In detail, these forces include buoyancy, resistance, propulsion, weight, and steering forces and moments. Detailed methods for calculating these forces may be found in many books on marine architecture (Refs. 18, 19, 20, 21), therefore, only a brief discussion is included here.

5-2.5.1 Stability

Unless an amphibious vehicle can float rightside-up under all conditions of loading and in any sea that it is likely to encounter, it is useless as a military vehicle. An understanding of stability in water requires an understanding of the fundamentals of buoyancy.

Buoyancy is the term applied to the total upward force exerted upon an immersed or floating object by the fluid, and is equal to the weight of the volume of fluid that has been displaced by the immersed object. Obviously, for a floating object, this upward force must be equal to the total weight of the object; or, when applied to an amphibious vehicle,

$$F_B = W_T = V_D w \tag{5-52}$$

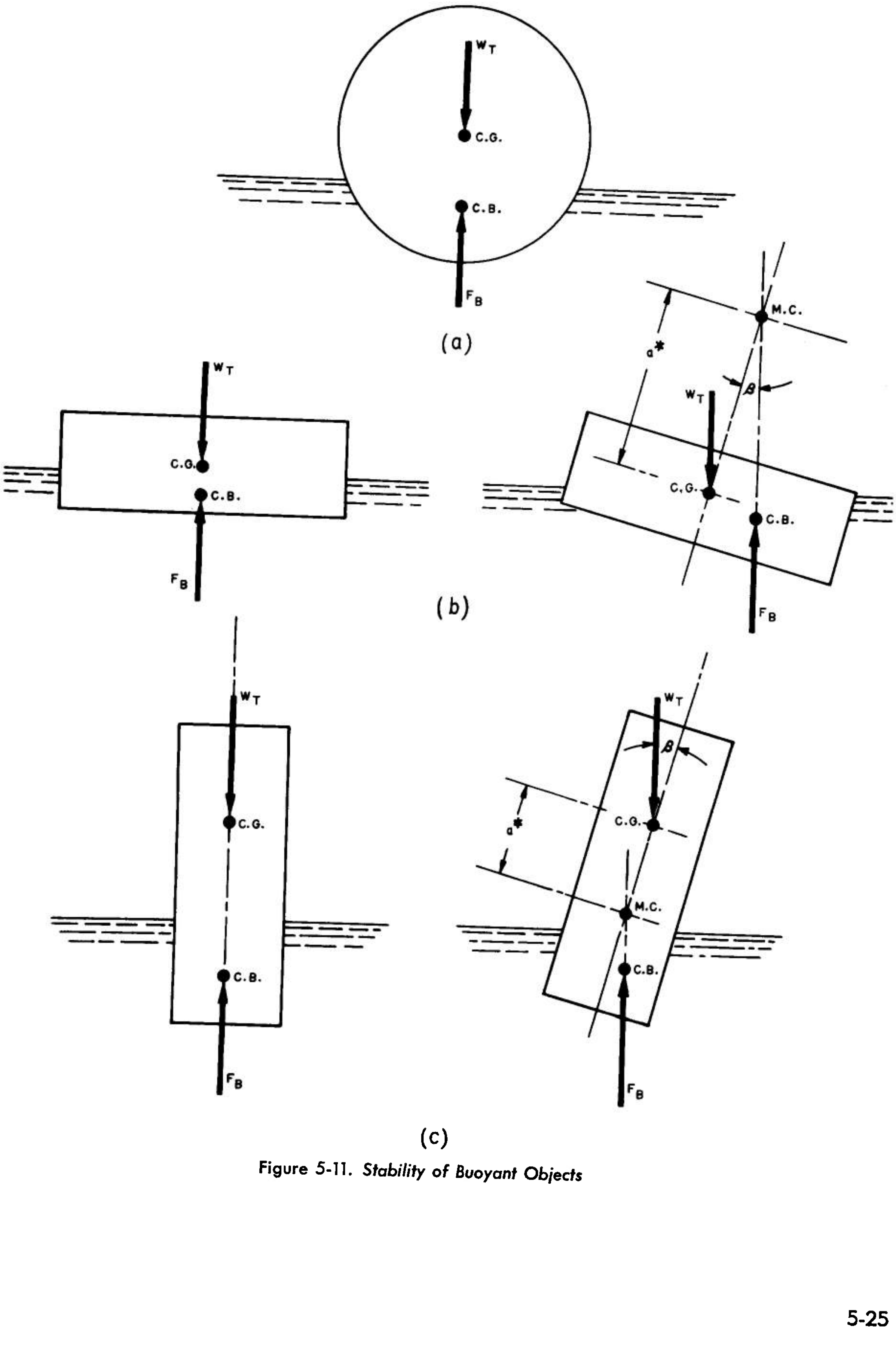


Figure 5-11. *Stability of Buoyant Objects*

where

F_B is the buoyant force
W_T is the total weight of vehicle
V_D is the volume of displaced liquid
w is the weight of unit volume of liquid

Equation 5-52 can be used to calculate the draft or depth of floatation of the vehicle.

The buoyancy results from the summation of buoyant pressure acting upon the wetted surface of the object. The point at which this resultant buoyant force acts is termed the center of buoyancy (C. B.) and is located at the center of gravity of the displaced volume of liquid (Fig. 5-11). Any tilting of the object that changes the shape or volume of the wetted area will cause the center of buoyancy to shift, both vertically and sideways. Thus, the center of buoyancy is not a fixed point, but the point about which the object balances at any given instant.

Stability, or the tendency of a floating object to remain upright, requires that a vertical line passed through the center of buoyancy must either (a) be in line with the center of gravity of the object, in which case the object is in equilibrium; or (b) it must cross the centerline of the object at some point (metacenter, M.C.) above the center of gravity, in which case, a couple is created that tends to right the tilted object. The centerline, as used here, is a vertical line that passes through the center of gravity when the body is in equilibrium. When the metacenter falls below the center of gravity, the action of the resultant couple is to overturn the object and the object is unstable. For relatively small angles of inclination (10° to 15°), the location of the metacenter is fairly constant. As the inclination increases, the metacenter moves toward the center of gravity. When the metacenter falls below the center of gravity, the vehicle will capsize.

These principles are illustrated in Fig. 5-11. If the cylindrical object shown at (a) is assumed to consist of a homogeneous material and floating as shown, its center of gravity (C. G.) is at its geometric center and the center of buoyancy (C. B.) is below the C. G. Since the C. G. and C. B. are in vertical alignment, the object is in equilibrium. When an external couple is introduced to rotate the object, the center of buoyancy does not shift, since neither the shape nor the volume of the wetted area is affected; and the center of buoyancy and center of gravity remain in vertical alignment. Since an object of this type will float in any position, it is said to be in neutral equilibrium.

Figure 5-11(b) illustrates an object that is in stable equilibrium. Note the location of the metacenter above the center of gravity and the couple produced by W_T and F_B tending to right the object. The distance between the center of gravity and the metacenter is called the metacentric height, a^*. Figure 5-11(c) illustrates an object whose geometry and weight distribution are such as to bring the metacenter below the center of gravity. In this case, the resultant couple tends to overturn the object. This is a condition of unstable equilibrium.

It is not necessary for the center of gravity to be below the center of buoyancy in order to have stability. Ships usually have the C. G. above the C. B. When the center of gravity is above the center of buoyancy, however, the vessel will also be in stable equilibrium when overturned. The center of buoyancy of a totally submerged object does not shift when the object tilts. Therefore, when a vehicle is designed for totally submerged operations, or which can become totally submerged, as an amphibious vehicle upon diving into the water, it must have its center of gravity below its center of buoyancy. Otherwise, the vehicle will turn completely over when totally submerged and will remain in the capsized position.

5-2.5.2 Rolling

Rolling of an amphibious vehicle in water is closely related to stability and is considered here for that reason. The mathematical theory of rolling was first investigated by Froude (Ref. 21) although several unsuccessful attempts were made earlier. Rolling is produced by wave action against the side of the vehicle, just as pitching is caused by waves passing from bow to stern.

The Froude theory of rolling first considers rolling in still water. Obviously, a vehicle cannot roll in still water, hence, some extraneous force must be used to start the roll. Once started it would continue to swing about a longitudinal axis near the center of gravity, C. G., of the vehicle. The period of rolling is the time in seconds for a complete roll cycle and is given by

$$\text{period} = T = 2\pi\sqrt{\frac{r^{*2}}{a^*g}} \qquad (5\text{-}53)$$

where r^* is the radius of gyration of the vehicle about the axis of rotation equal to

$$r^* = \sqrt{I^*/W_T} \tag{5-54}$$

I^* is the mass moment of inertia about the longitudinal axis and

a^* is the metacentric height (see Fig. 5-12)

This equation has only theoretical value since it is nearly impossible to calculate r^* with any accuracy.

When the metacentric height is small, T approaches infinity as a^* approaches zero. One of the experimentally deduced equations for I^* is given as

$$I^* = \frac{T^2\, W_T\, a^* \sin\beta}{4\pi^2} \tag{5-55}$$

This formula permits calculations of I^* for small a^*.

When waves are present, extraneous resistance forces are no longer necessary to begin rolling. When a wave approaches the vehicle broadside and passes beneath it, it introduces a transient condition into the relationship of vehicle center of gravity and center of buoyancy that causes the vehicle to heel to one side. As the disturbance passes, the vehicle continues to oscillate about its longitudinal axis. If the wave period is the same as the natural period of rolling of the vehicle, the inclination of the vehicle will double with each succeeding wave until the vehicle capsizes.

From wave theory, we find that the period and speed of the wave are proportional to the wave length, as given by

$$T_W = 0.442\sqrt{l_W} \tag{5-56a}$$

$$V_W = 1.341\sqrt{l_W} \tag{5-56b}$$

where

T_W is the wave period, sec

$_W$ is the wave length, ft

V_W is the wave speed, knots

5-2.5.3 Launching or Landing of Amphibious Vehicles (Ref. 5)

When an amphibious vehicle attempts to make a landing upon a steep bank, its front end makes contact with the bank at some point, A' (Fig. 5-12) and proceeds up the sloping bank to A''. As the front end makes contact, the vehicle receives partial support, S, at the point of contact, which produces

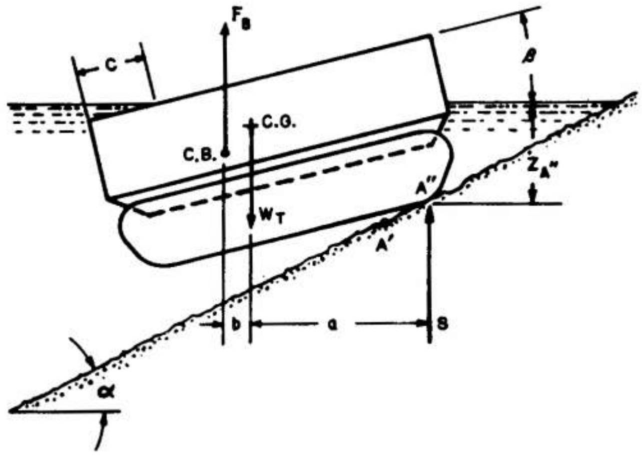


Figure 5-12. Force System Acting on an Amphibious Vehicle During Launching or Landing

a couple, Sa, about the center of gravity. This couple tilts the vehicle to an angle β, causing the stern to submerge deeper into the water. The change in geometry of the displaced volume causes the center of buoyancy to shift toward the stern creating a counter couple, F_Bb. While the vehicle remains in static equilibrium, the following two conditions must be satisfied at all times while the vehicle progresses up the bank

$$F_Bb = Sa \tag{5-57}$$

$$F_B + S = W_T \tag{5-58}$$

In extreme cases, the outboard end of the vehicle may submerge to such a degree that water will enter the ventilating system or cargo compartment, and the vehicle will be swamped. For this reason it is necessary to analyze the conditions of landing and launching.

The buoyancy, F_B, and its moment arm, b, are functions of both the depth $Z_{A''}$ of the point of contact and of the trim angle β. Similarly, the moment arm a is also a function of the trim angle β. The analysis of this problem is best done graphically.

A likely point of contact on the vehicle is selected for each contact depth $Z_{A''}$ that is being considered, and the buoyancy, F_B, and its moment arm, b, are calculated for several values of β to cover the likely range of equilibrium $\beta \leqq \alpha$. Since W_T is known, S is readily calculated, using Eq. 5-58 for each value of F_B (and hence of β). The moments, F_Bb and Sa, are calculated for each value of β and are each plotted as functions of β. The intersection of these two curves represents the value of β at which the vehicle will be in equilibrium.

A scale drawing can now be made of the vehicle partially supported at the assumed point of contact A'' and displaying the trim angle of equilibrium that has been just determined. The amount of submergence of the outboard end (such as dimension C in Fig. 5-12) can now be studied. This procedure can be repeated for a number of assumed contact points and a number of slopes, α, to develop continuous curves of desired information as the vehicle moves up (or down) the slope during a landing or launching operation.

5-3 FORCES RESULTING FROM THE OPERATION OF EQUIPMENT MOUNTED ON THE VEHICLE

When the various equipment mounted upon a vehicle is operated, a redistribution of forces takes place, and additional forces may be added to the force system already acting upon the vehicle. Figure 5-13 illustrates how the center of gravity of a vehicle is affected by elevating the main armament of traversing the turret. This must be considered in the analysis of the force system, as it affects not only the stability but the performance of the vehicle as well.

When loads are raised by crane-type equipment mounted on the vehicle, as in Fig. 5-14(a), the vehicle experiences a turning moment tending to upend the vehicle. This places an increased load upon the rear suspension system and a correspondingly lessened load upon the front suspension. The suspension system must, therefore, be designed to accommodate these increased loads. A similar redistribution of forces occurs when the crane is lifting a load at the side of the vehicle. In this case, unequal loading occurs on the two sides of the vehicle.

The upending moment produced by the operation of crane-type equipment mounted on the vehicle is resisted by a counter moment which is a product of the total sprung weight and the moment arm about which it acts. This establishes a limit on the lifting capacity of the crane. Some vehicles have suspension lockout provisions which, in effect, provide a rigid connection between the sprung and unsprung parts of the vehicle. This adds the weight of the unsprung mass to that of the sprung mass in resisting the upending moment.

Figure 5-14(b) shows a track-laying, combat vehicle on a side slope firing its weapon broadside. The violent overturning moment is quite obvious,

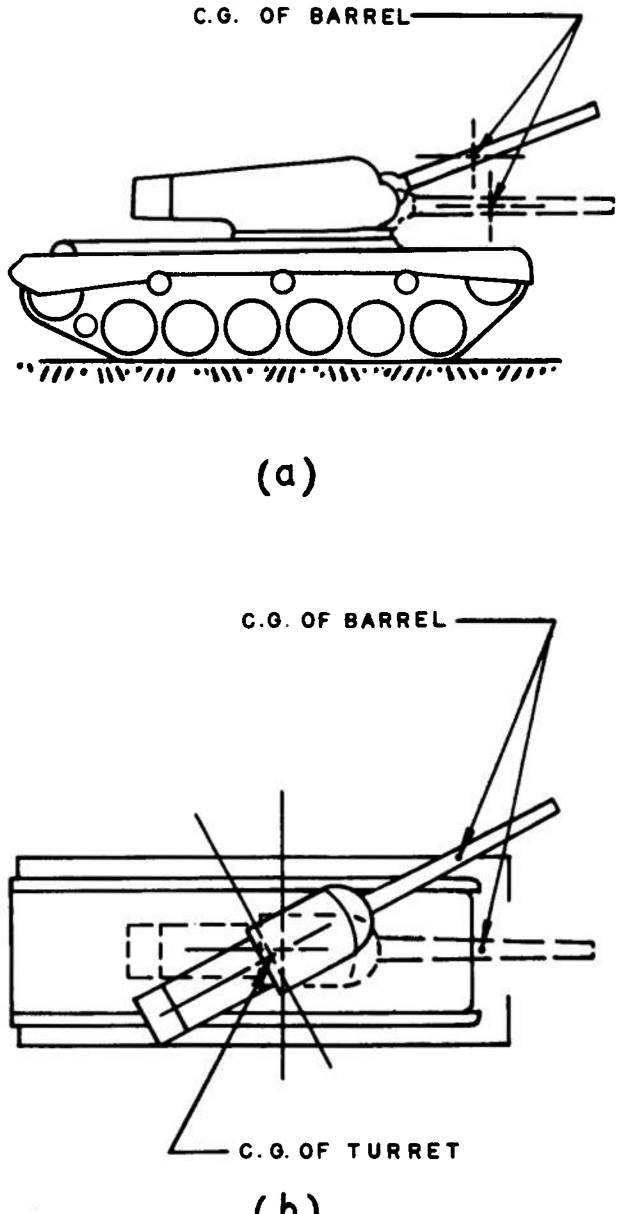


Figure 5-13. Change in Location of Center of Gravity of Vehicle Component Forces When (a) Elevating Main Armament, (b) Rotating Turret

and the side slope only aggravates the situation by causing a component of the vehicle weight to be added to the overturning moment produced by the weapon recoil. Here, too, suspension lockout provisions make the vehicle more stable by adding the weight of the unsprung mass to help resist overturning.

Figures 5-14(c) and 5-14(d) show other load conditions that are common in military vehicles.

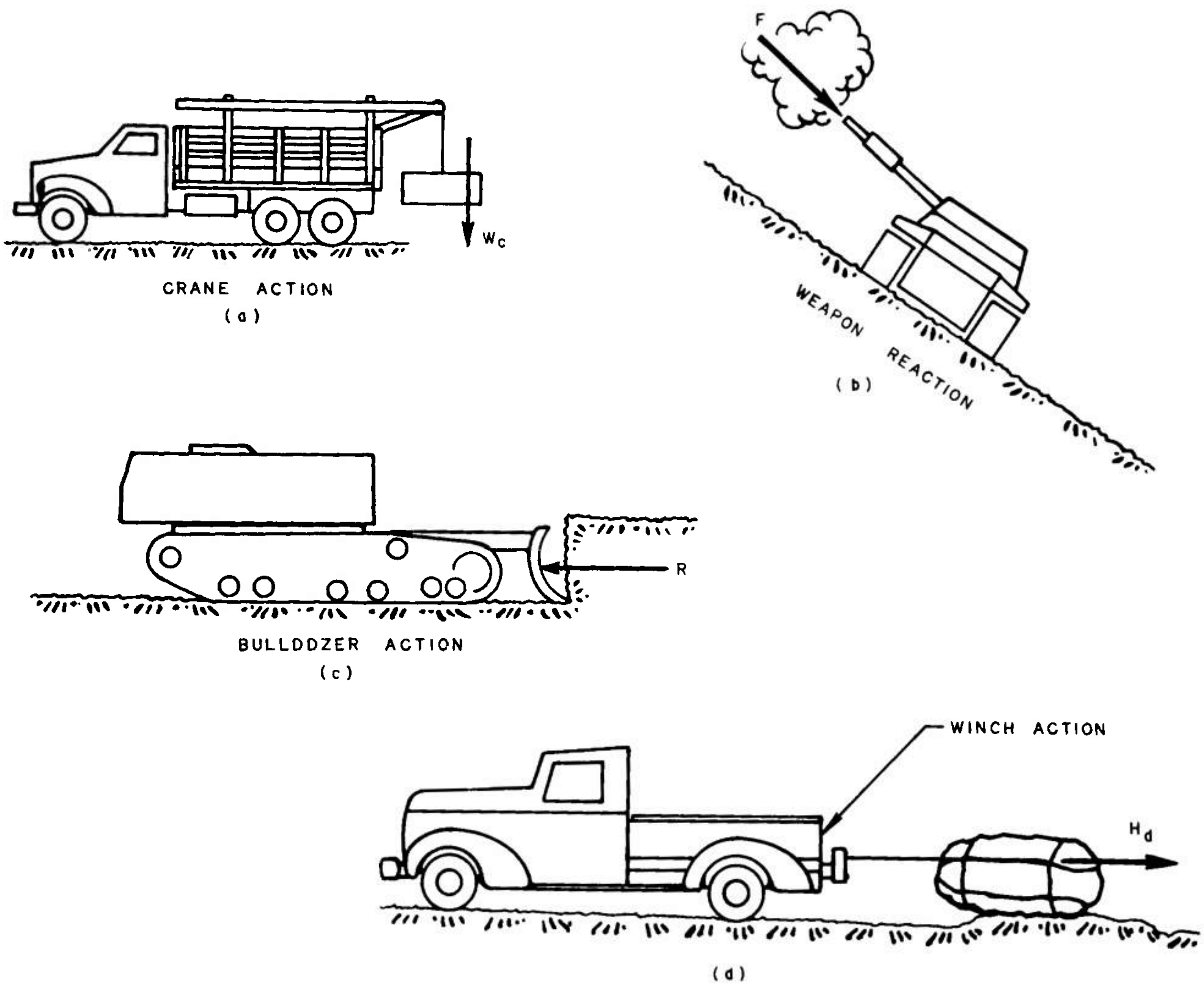


Figure 5-14. Force System Resulting from the Use of Equipment Mounted on Vehicle

In each case, a turning moment is produced that tends to upend the vehicle. There is little danger, however, of actually upending the vehicles, because the maximum reactions, R or H_d, acting upon the vehicle depends upon the maximum tractive effort that the vehicle can produce. This is limited by the shear strength of the soil, or by the coefficient of friction of the wheels against the pavement, to some value far below that needed to upend the vehicle. It should be noted, however, that the upending moment, produced by the reactions, R or H_d, affects the pressure distribution of the tracks or wheels upon the ground; so that one end of the vehicle will sink deeper into the ground while the sinkage of the other end will decrease. The first effect results in an increased rolling resistance that the vehicle must overcome, while the second effect results in a decreased tractive effort (see par. 5-2.1). The combined effect is a decrease in R or H_d proportional to the upending moment.

SECTION II INDETERMINATE FORCES

The second major category of forces that act upon the automotive assembly are those whose magnitude cannot be calculated readily, and that often require elaborate measuring procedures and sophisticated mathematical techniques for evaluation. This group of forces has been categorized in this chapter as "indeterminate forces" and consists of forces resulting from shocks and vibrations encountered by the vehicle during cross country operations over rough terrain, during airdrop operations, or as a result of ballistic impact or high-energy blast.

5-4 GENERAL DESIGN PROCEDURE

A rigorous method for evaluating the indeterminate forces during the design phase, and correctly relating them to the stresses experienced by the vehicle, is not known at the present time. The method generally employed by designers is to determine the acceleration produced by the shock force, and express this as a multiple of g, the acceleration due to gravity. This number, λ, is then applied as a multiplying factor to the mass under consideration to determine the magnitude of the shock force experienced by the member. Thus, the force, F_I, experienced by some component of weight, W_a, subjected to a shock of $\lambda_a g$ is considered to be

$$F_I = \frac{\lambda_a g W_a}{g} = \lambda_a W_a \tag{5-59}$$

A certain amount of caution is necessary to insure that the correct weight is used with the multiplier since components of different weights subjected to the same shock, λg, will experience shock forces proportional to their weights. For example, consider a fully-loaded vehicle hull weighing 25,000 lb that sustains an $8g$ shock as the vehicle passes over an obstruction at high speed. Based upon Eq. 5-59, the total of the suspension reactions acting on the hull will be 8 times 25,000 lb, or 200,000 lb. The force experienced by the mounting brackets supporting a 50-lb winch mounted on the hull, however, will be only 8 times 50, or 400 lb.

In the actual case, nothing is truly rigid, and a certain amount of attenuation takes place between the point of application of the shock and the point under consideration. In general, components mounted within a vehicle will experience shocks of lower intensity than will the hull, which will, in turn, experience less shock than will the axles or wheels, as a result of the attenuation produced by the suspension system.

While the relative flexibility of the various supporting members serves to attenuate the vehicle shocks before they reach the crew or delicate components mounted in the vehicle, it also allows these supporting members and the equipment they support to vibrate. The accelerations inherent in these vibrations are often far more severe than the shock which initiated the vibration. This is demonstrated in the following example:

Assume that a 20-g ballistic impact causes a tank turret to vibrate at its natural frequency of 600 cps with an initial amplitude of 0.75 in. The following classic equations describe the motion of a freely oscillating body

$$y = A \sin \omega t \tag{5-60}$$

$$\dot{y} = \omega A \cos \omega t \tag{5-61}$$

$$\ddot{y} = \omega^2 A \sin \omega t \tag{5-62}$$

where

ω is the angular frequency of vibration, radians per sec

t is the time, sec

y is the amplitude of vibration, in.

$\dot{y}$ is the velocity, ips

$\ddot{y}$ is the acceleration, in. per sec per sec

A is the maximum amplitude, in.

The acceleration reaches its maximum when $\sin \omega t = 1$, in which case

$$\ddot{y}_{max} = \omega^2 A \tag{5-63}$$

Substituting in Eq. 5-63

$$\ddot{y}_{max} = (2 \pi \times 600)^2 \times 0.75$$

$$\ddot{y}_{max} = 10.65 \times 10^6 \text{ in. per sec per sec}$$

$$\ddot{y}_{max} = 27{,}500 \text{ g}$$

This is a simplified treatment and serves merely to illustrate a point. In the actual case, the turret is not a freely vibrating body nor does it vibrate in a simple, sinusoidal manner. The actual vibration pattern is highly complex, being a summation of many different vibrations induced by the engine, the exhaust system, suspension system, roughness of the terrain, pumps, fans, air compressors, miscellaneous motors, and the firing of weapons, to name a few. Any additional vibrations, such as those imposed by blast or ballistic impact, are integrated into this already complex vibration pattern—where some specific vibrations are augmented while others are damped. This makes it virtually impossible to accurately predict the maximum acceleration that will be experienced by components of the automotive assembly. Therefore, extensive tests with existing vehicles operating in typical military environments must be performed and accelerations that actually occur on the various components and sections of the vehicle recorded.

Representative shock and vibration data recorded in actual tests are given in Chapter 3. The values given should not be interpreted as maximum values nor as the only values that can occur. They are presented here merely to give the designer a general concept of the shocks and vibrations that do occur in military automotive vehicles.

5-5 NEED FOR BETTER PROCEDURE

The procedure of using the g-multiple of the peak accelerations as a multiplier in determining the magnitude of the shock forces is a popular, but technically unsound, method. It results in vehicles capable of safely withstanding sustained loads many times greater than those normally experienced by the vehicle. This practice tends to produce overdesigned vehicles with their attendant excess weight and excess cost.

A second, undesirable aspect of overdesigning is the increased stiffness of the overdesigned members. This increased stiffness results in an increase in the natural frequency of the structure. Since the acceleration inherent in a vibrating mass is proportional to the square of its frequency (Eqs. 5-62 and 5-63), increasing the stiffness of a structural member can decrease its resistance to continuous vibration. The important consideration to be made in a shock or vibration loading is the force-time history of the peak conditions. A force that builds up to an intense peak and decays all in an extremely short time interval can be far less severe than a force which reaches a much lower peak, but whose time history covers a much longer interval. Similarly, intense peak forces of extremely short duration which occur at relatively long intervals can be less severe than moderate peaks that occur with rapid frequency. A thorough analysis of the problem must include the force-time history acting upon the vehicle. This must then be related to the mass, natural frequency, and physical dimensions of the members involved before a realistic computation of stresses can be made.

A great deal of work has been done to develop the basic principles of shocks and vibrations (Refs. 22, 23, 24, 25, 26, 27), but very little has been done to relate these principles to the stresses developed in structural members. The problem is complex, especially when dealing with composite vibrations of multiple frequencies in a six-degrees-of-freedom system, which is the usual case in military vehicles. The solution of specific problems of this type requires the assistance of experts in this field.

There is a need for the development of a general method—supplemented by tabular data, charts, and nomographs, and substantiated by experimental results—that could be applied by a design engineer to his specific design parameters, and that would result in a reasonably accurate stress analysis. Unfortunately, no such "golden method" exists at the present time. Designers must continue to use the inaccurate, design-penalizing, multiplier method. A general knowledge of the basic principles of shocks and vibrations, plus knowledge gained through experience, will develop an intuition within the designer with which he can temper the multiplying factors. This is the best that can be offered under the present state-of-the-art.

5-6 EFFECT OF SHOCKS AND VIBRATIONS

The general subject of shocks and vibrations is far too complex to be treated adequately in this handbook; furthermore, numerous excellent texts dealing with this subject exist (Refs. 22, 23, 24, 26, 27). An automotive vehicle, however, experiences certain effects as a result of shocks and vibrations that are usually overlooked by the designer, and it is only after field trials reveal deficiencies in the design that corrective modifications are made. Even then it is often not apparent, and therefore not realized, that the failure or malfunction is directly attributable to vibration or shock loading and could have been prevented had the designer been cognizant of the effects of vibrations upon vehicle components. For this reason, a discussion of some of the vibration effects is included in this chapter.

5-6.1 STRUCTURAL DAMAGE

The most familiar effect of shock and vibration loading is in their ability to produce structural failures, occasioned by the actual rupture or breaking of the structural material, or by producing such severe deflections in members as to strain them beyond their elastic limits and cause them to malfunction, or to become otherwise unsatisfactory, due to permanent structural deformation. The current practice is to apply so-called factors of safety to the known loads acting upon members in order to design additional strength into the structure for resisting shocks. A satisfactory method for determining a precise value for the factor of safety is not known. In selecting a factor of safety, the designer must draw upon his experience and his general knowledge of material characteristics, theory of failures, and accuracy in predicting the loads that will act upon his structure.

The term *factor of safety* is commonly used to denote the ratio of the ultimate strength of a material (or the yield strength) to the working stress used for the material in the design. The use of this term is rather unfortunate in that it misleadingly implies an idea of safety. For example, if a factor of safety of 4, based upon the ultimate strength of the material, is used in selecting a working stress for a ductile material, a structure designed on the basis of this working stress will not, in general, satisfactorily resist forces four times as great as those that produced the working stress. The characteristics of ductile materials are such that they will take large, permanent deformations at stresses considerably below their ultimate strength. Thus, the structure in the example will fail because of large permanent deformation of the load-carrying member long before the ultimate strength is reached, and the distribution of loads to the various members will be radically changed. The term *reduction factor* has been recommended (Ref. 28) as preferable to *factor of safety.*

The need for a margin of strength arises from the uncertainties of the exact physical properties of the materials used, the uncertainties of the true magnitude-time history of the loads to be resisted by the structural members, and the uncertainties in the methods used to calculate the stresses in the various members. In evaluating these uncertainties, additional factors to be considered include the seriousness of a failure of any specific member of the structure or machine and the impact of such a failure upon the whole; the extent of damage to property or human life of such a failure; the desired useful life of the assembly; the extent to which deterioration is likely to occur; the extent to which periodic inspections and maintenance can be conducted and their effect upon the margin of strength. The more these uncertainties can be reduced by analytical methods, tests, and experience the higher the working stress can be, and the higher the working stress, the lower the weight and cost of the completed vehicle.

5-6.2 FATIGUE

During operation, the structural members of automotive vehicles are constantly subjected to cycling stresses, that is, stresses which vary between minimum and maximum values. Complete reversals of stress, from tension to a compression of equal magnitude, are quite common. The cycling stresses are caused by cycling loads, as when a vehicle operates over rough terrain; or they occur in vibrating structural members that have been set vibrating by the roughness of the ground over which the vehicle is operating, by the pulsating of mechanical equipment and weapons mounted upon the vehicle, or by ballistic shocks.

These cycling stresses lead to a particular type of failure known as fatigue failure or failure by progressive fracture, which is very common in automotive vehicles. It results from a slippage occurring along certain crystallographic planes of the stressed material accompanied by local crystal fragmentation, rupturing the atomic bonds, and leading to the formation of submicroscopic cracks that soon develop into visible cracks. It starts at one or more points of high localized stress, such as an abrupt change in section or flaw in the metal, and gradually spreads by the continual rupturing of metal at the edges of the crack, as the stress is repeated, until the cross sectional area of the member is so greatly reduced that the member suddenly breaks. The chief characteristics of this type of failure are: (a) final failure occurs suddenly without a warning period of plastic deformation, and (b) failure occurs at the endurance limit for the material, a stress considerably lower than the yield point.

5-6.3 EFFECT ON SUSPENSION AND STEERING

In addition to the stress producing effects, shocks and vibrations also have definite effects upon the functioning of the major elements of the automotive assembly. The suspension system, for example, has a primary function of raising the top speed at which a vehicle can satisfactorily traverse rough terrain without detrimental or unacceptable shocks to vehicle and personnel and without experiencing unacceptable variations in tractive effort developed by the wheels or tracks. This is a more purposeful objective than merely to cushion the major portion of the vehicle and its personnel from terrain shocks.

In the performance of its specific function, the ground-contacting components of the suspension system (wheels, tracks, etc.) are required to follow the irregularities of the terrain, and are thereby subjected to violent and erratic vertical accelerations. Each acceleration produces a variation in the terrain loading; thus, a variation in the tractive effort of the vehicle.

Wheeled vehicles are equipped with pneumatic tires, and tracked vehicles with solid rubber tires on their road wheels, to help cushion the impacts between the wheels and the terrain, or between the road wheels and the track shoes. Since each wheel is suspended between its tire and the vehicle springs, the result is a vibratory system. When the combination of vehicle speed and spacing of ground irregularities are such that the wheels vibrate at their natural frequencies, a condition known as "wheel dance" occurs. During this condition, the wheels experience violent shock loading; they alternately pound and leave the ground, resulting in violent fluctuations in traction. Road wheels will dance upon the track shoes, which results in damage to the tires and imposes severe requirements upon the track-tensioning mechanism that may culminate in a thrown track. Secondary vibrations are transmitted through the shock absorbers to the main part of the vehicle. Chapter 11 discusses some of the factors that reduce wheel dance. Excessive shocks and vibrations cause shock absorbers to heat up, which affects their damping characteristics.

The steering system is often seriously affected by shocks and vibrations. Wheel dance, with its attendant loss of traction, results in a loss of steering control in both wheeled and tracked vehicles. Shocks create transient deflections in the steering linkage resulting in instability of control. The components of the steering system of a wheeled vehicle are particularly sensitive to damage from shock and fatigue loading.

5-6.4 EFFECT ON FRAME AND BODY

The main effect that shocks and vibrations have on the frame and body of an automotive vehicle, aside from the stress-related factors already discussed, has to do with the generation and amplification of noise. Noise is not only uncomfortable to occupants of a vehicle, but causes physical fatigue, irritability, and loss of efficiency in the crew, as well. In extreme cases, it may produce nausea. The large, often hollow, sections of the metal frame, together with the large, relatively flat areas of the vehicle body, often respond audibly to the multitude of vibrations occurring in the vehicle. The vehicle body, particularly the hull of an armored vehicle, acts as a gigantic sounding board, amplifying these noises until the result is sometimes intolerable.

5-6.5 EFFECT ON POWER TRAIN

The power train, too, is adversely affected by shocks and vibrations. Variations in tractive effort, brought about by variations in terrain loading as a result of the vertical accelerations experienced by the wheels, produce pulsations of torque that are felt throughout the power train. Although these torque pulsations do not produce reversals of stress in the power-train components, they do produce cycling stresses, which also lead to fatigue failures. Surface fatigue is commonly experienced in power-train components where highly concentrated compressive loading takes place between two rolling members, such as in ball bearings, cams, and gear teeth. The result of surface fatigue is a spalling or flaking of the surface material of one or both surfaces. This type of failure is also referred to as "fretting" and is caused by high shearing stresses below the surface of the material (Refs. 29, 30).

Wear of moving parts is also increased in an environment of high shock and vibration. This is largely due to the transient deformations of bearing surfaces and the transient misalignments of driving shafts caused by shock and vibrations. Lubricant seals are sometimes affected by vibrations which cause these to work loose and provide defective sealing. This may result in escape of the lubricant or the introduction of dirt or water into the system, or both, thus increasing wear.

5-6.6 EFFECT OF POWER PLANT

The irregularity of the ground over which the vehicle is traveling has, thus far, been considered the source of shocks and vibrations experienced by the vehicle. This, however, is not the only source. The power plant, miscellaneous powered equipment mounted in the vehicle, and the firing of weapons all add to the complex vibration of the vehicle. The operation of the main power plant can also be affected by vibrations (depending upon their frequency and amplitude). Carburetion, for example, is sometimes affected, resulting in an improper air-fuel ratio, a nonuniform distribution of air-fuel mixture among the various cylinders, or a nonuniform supply of fuel to the engine, resulting in a loss of power and power fluctuation. Magnetos, distributors, voltage regulators, and similar equipment whose operations involve the positioning of electrical contacts, are sensitive to shock and vibrations. An electrical contact made or broken at the improper time because of an external shock will affect the operation of the engine. Excessive, continuous vibration may cause arcing between contact points resulting in burned points and poor engine performance.

5-7 SHOCKS ENCOUNTERED DURING AIRDROP OPERATIONS (Ref. 31)

During airborne assault or air-supply operations, automotive vehicles are often airdropped into the target area to provide increased mobility to the committed forces. The objective of the aerial delivery system employed is to get the vehicle to the ground in the least possible time without damage to the vehicle, by applying the most economical and practical methods available. The smallest possible time factor is desirable for maximum drop accuracy, minimum dispersion of dropped loads, and minimum effect of wind drift. In addition, it minimizes the time during which the vehicle is vulnerable to enemy fire and decreases the time during which the enemy may pinpoint the drop area. The ideal aerial delivery system would not employ any deceleration device (parachute, retro rockets, etc.) but would absorb the landing shock to prevent damage to the vehicle. Unfortunately, the weight, bulk, and cost of such shock-absorbing devices prohibit their use for bulky or heavy cargoes. Present aerial delivery systems achieve an acceptable compromise by employing a parachute canopy in conjunction with shock-absorbing devices attached to the vehicle-carrying platform. The canopy has a stabilizing effect upon the air-dropped cargo which permits a system design requiring the absorption of impact shocks primarily in one direction. Present aerial delivery systems strive for the best possible compromise between a landing-impact absorbing system and a rate of descent low enough to hold the design of the shock-absorbing devices within reasonable weight, bulk and cost limits.

Most current aerial delivery systems attain a rate of descent of approximately 25 ft per second. This may be varied in accordance with the fragility of the cargo, the requirements of the tactical situation, and the efficiency of the ground-impact absorption system used. Ground-impact forces of up to 100 g have been measured on cargo platforms descending at approximately 30 ft per second. Wind drift can cause damage by making the parachute-supported vehicle roll or topple on impact with the ground. Antitoppling devices are often used on the cargo platforms to avoid this type of damage.

Two other shocks are experienced by the air-dropped vehicle during aerial delivery, namely, the snatch force and the opening shock.

5-7.1 THE SNATCH FORCE

Snatch force is defined as the force imposed upon the suspended load by the parachute to accelerate the parachute mass from its final velocity at the time of full line stretch (time of snatch) to the velocity of the suspended load. This force arises from the comparatively rapid deceleration of the deploying parachute in relation to the slow deceleration of the suspended load. This differential velocity must be reduced to zero.

The snatch force, fortunately, occurs at the instant the suspension lines are fully deployed and prior to the actual inflation of the canopy. Thus, the snatch force and parachute-opening shock are not additive, but follow one another rather closely. The peaks of these two shocks occur from 0.1 to 1.0 sec apart. At low aircraft speeds, and using less modern canopy designs, snatch forces do not exceed the opening shock. Opening shock can be reduced considerably by applying special reefing, venting, collapsing, and other techniques to the parachute design. Snatch forces, however, are much more difficult to control and will become the limiting factor in future aerial delivery operations.

The maximum snatch force can be calculated from the following set of equations

$$P = \left(\frac{W_c \, \Delta \, V^2 \, Z \, P_{max}}{g \, \epsilon_{max}} \right)^{1/2} \tag{5-64}$$

where

- P is the peak snatch force, lb
- W_c is the weight of canopy cloth area, lb (including weight of suspension lines across the cloth area but not including the weight of free length of suspension lines)
- Z is the number of suspension lines
- P_{max} is the breaking strength of suspension line, lb
- g is the acceleration due to gravity, ft per sec per sec

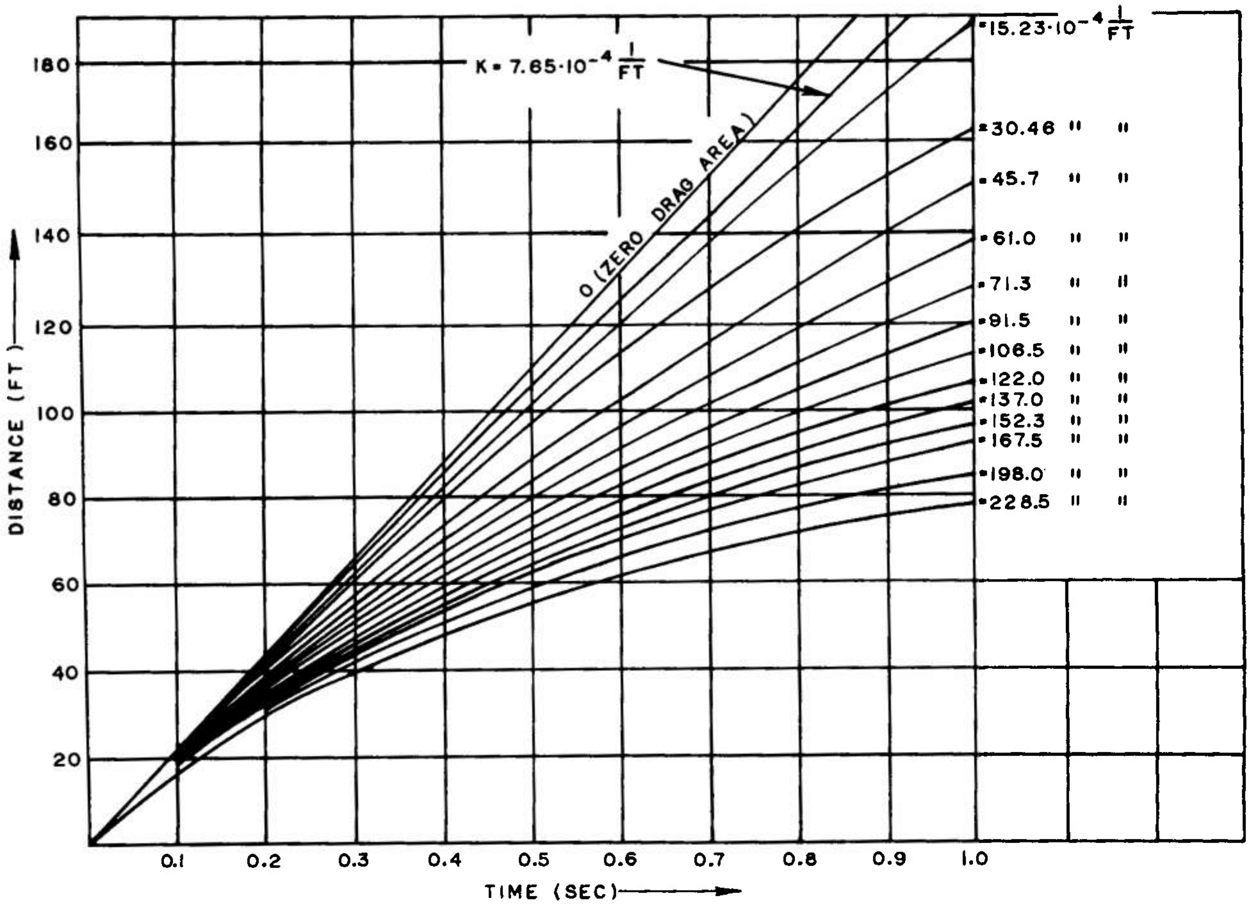


Figure 5-15. Distance of Travel of Bodies with Various Drag Loadings, Launched at 130 Knots

ε_{max} is the maximum elongation of suspension lines, ft

ΔV is the differential velocity between the center of gravity of the parachute canopy before and immediately after the snatch, fps

The ΔV term can be evaluated with the aid of the graphs shown in Figs. 5-15 through 5-19 and Eq. 5-65. The graphs show the time, t, required for two bodies with varying drag loading, K, to separate for a certain distance, d, provided that both have a launching speed of V_o and both are decelerated only by their individual drags. When the time, t, has been determined (from the appropriate graph), the differential velocity, ΔV, can be calculated by means of the following equations

$$\Delta V = V_o^2 \frac{tK_b(n-1)}{1+V_o\, tK_b(n+1)+V_o^2\, nK_b^2\, t^2} \tag{5-65}$$

$$K_b = \frac{C_{D_b}\, S_b\, \gamma}{2W_b} \tag{5-66}$$

$$K_p = \frac{C_{D_o}\, S_o\, \gamma}{2W_p} \tag{5-67}$$

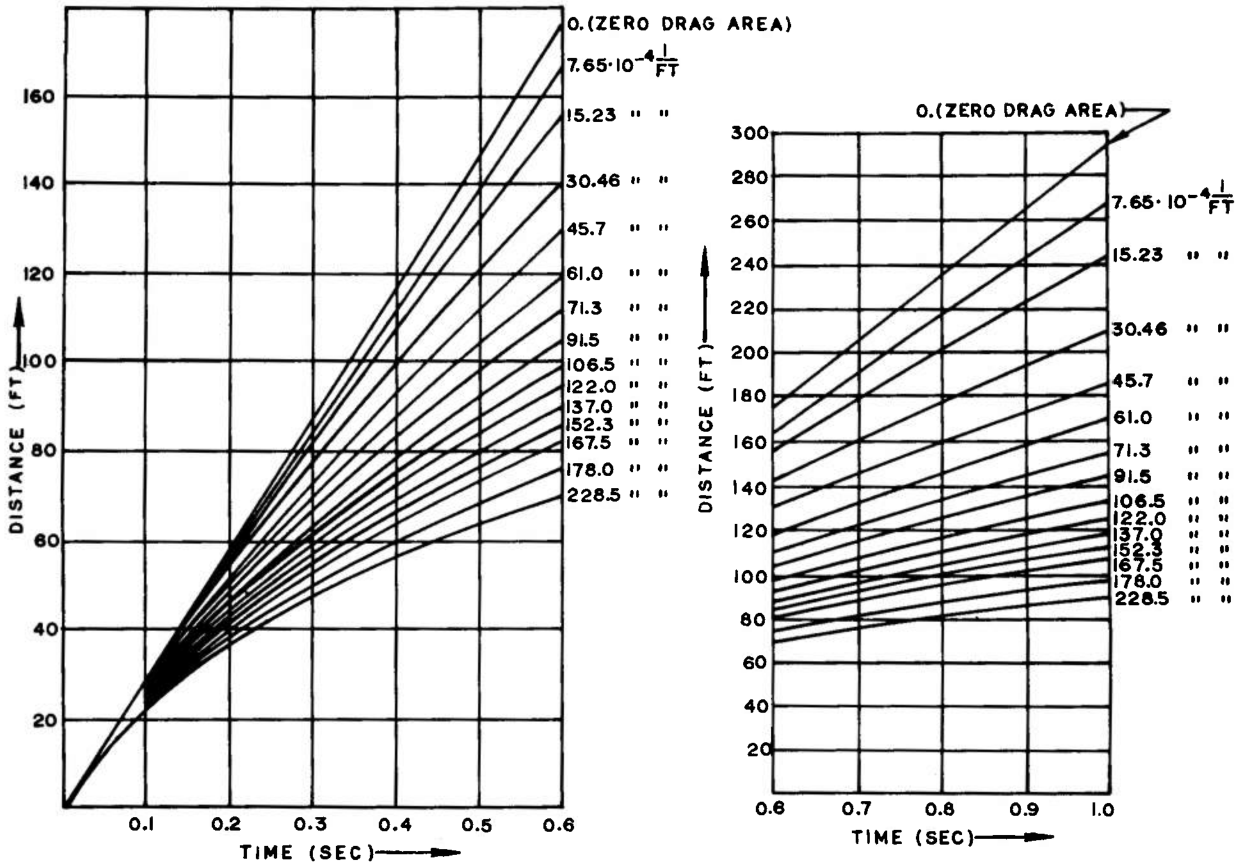


Figure 5-16. Distance of Travel of Bodies with Various Drag Loadings, Launched at 175 Knots

$$n = \frac{K_p}{K_b} = \frac{C_{D_o} S_o W_b}{C_{D_b} S_b W_p} \tag{5-68}$$

where

K_b ; K_p	are the loadings of the suspended load and of the uninflated parachute, respectively, ft^{-1}
C_{D_b} ; C_{D_o}	are the drag coefficients of the suspended load and of the uninflated parachute, respectively
S_b ; S_o	arc the aerodynamic areas of the suspended load and of the uninflated main parachute plus pilot chute, respcctively, sq ft
W_b ; W_p	are the weights of the suspended load and of the canopy cloth area including weight of external suspension lines, lb
V_o	is the launching speed, ft per sec (one knot = 1.68894 ft per sec)
d	is the distance from center of gravity of the canopy to the suspension point on the load, ft
γ	is the specific weight of air, lb per cu ft

This method of calculating the snatch force, P, and the differential velocity, ΔV, is based upon a few assumptions and simplifications. To the extent that has been assumed (Ref. 31), this method gives reasonably accurate results. This

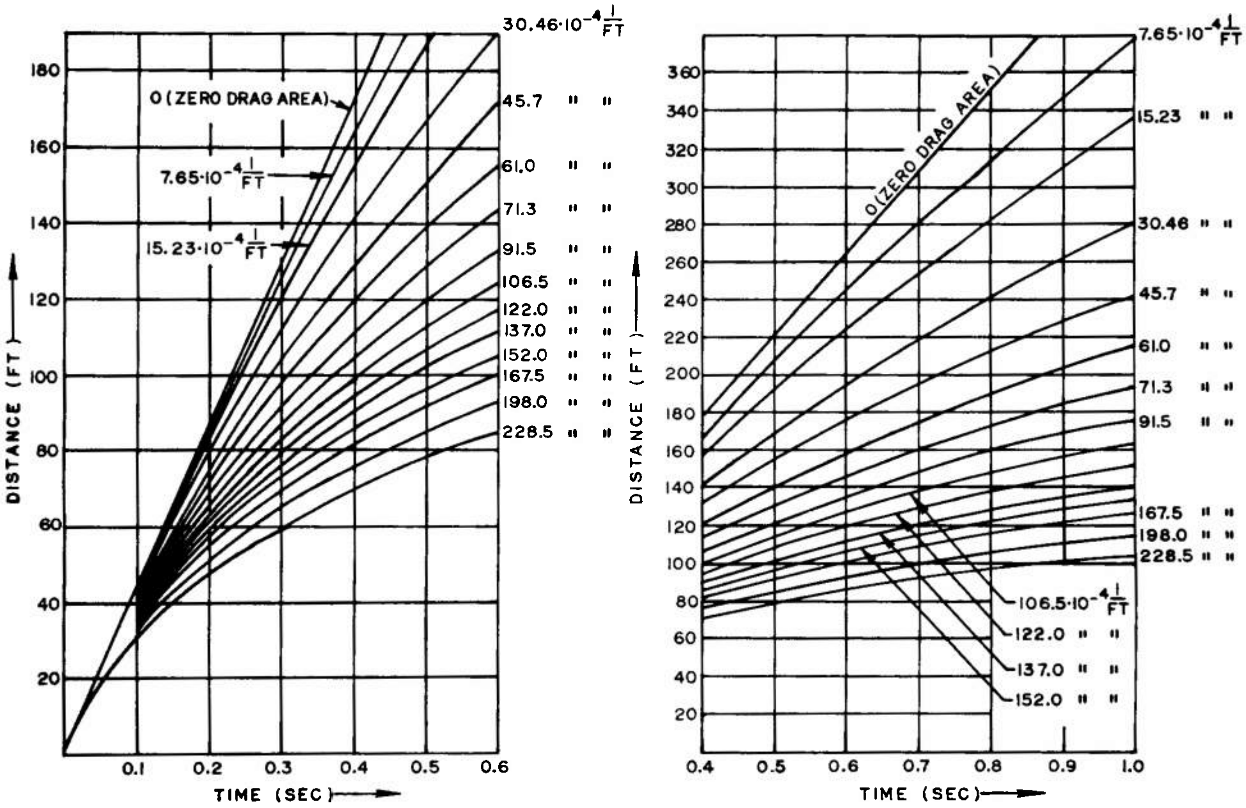


Figure 5-17. Distance of Travel of Bodies with Various Drag Loadings, Launched at 260 Knots

system is being continually studied, and it may become necessary to revise or to supplement some assumptions. Present assumptions of the drag of the uninflated parachute plus pilot chute, in particular, are not based upon measured data and cannot be considered definite. Such assumptions must continue until more accurate data is made available.

5-7.2 EXAMPLE OF SNATCH FORCE CALCULATION

Given the following conditions, the snatch force can be calculated

W_b	=	200 lb
W_p	=	10 lb (for parachute canopy)
W_c	=	7.5 lb (for canopy cloth area only)
$C_{D_b} S_b$	=	4 sq ft (for suspended load)
$C_{D_o} S_o$	=	3 sq ft (for uninflated parachute canopy plus pilot chute)
Z	=	24 suspension lines
P_{max}	=	550 lb, breaking strength of suspension lines
l_s	=	20 ft, length of suspension lines
ϵ_{max}	=	0.4×20 ft $= 8$ ft, maximum elongation of suspension lines (550-lb suspension lines have maximum elongation of 40%)
V_o	=	130 knots launching speed
γ	=	0.07651 lb per cu ft (specific weight of air at sea level)
d	=	21 ft from center of gravity of canopy to suspension point of load

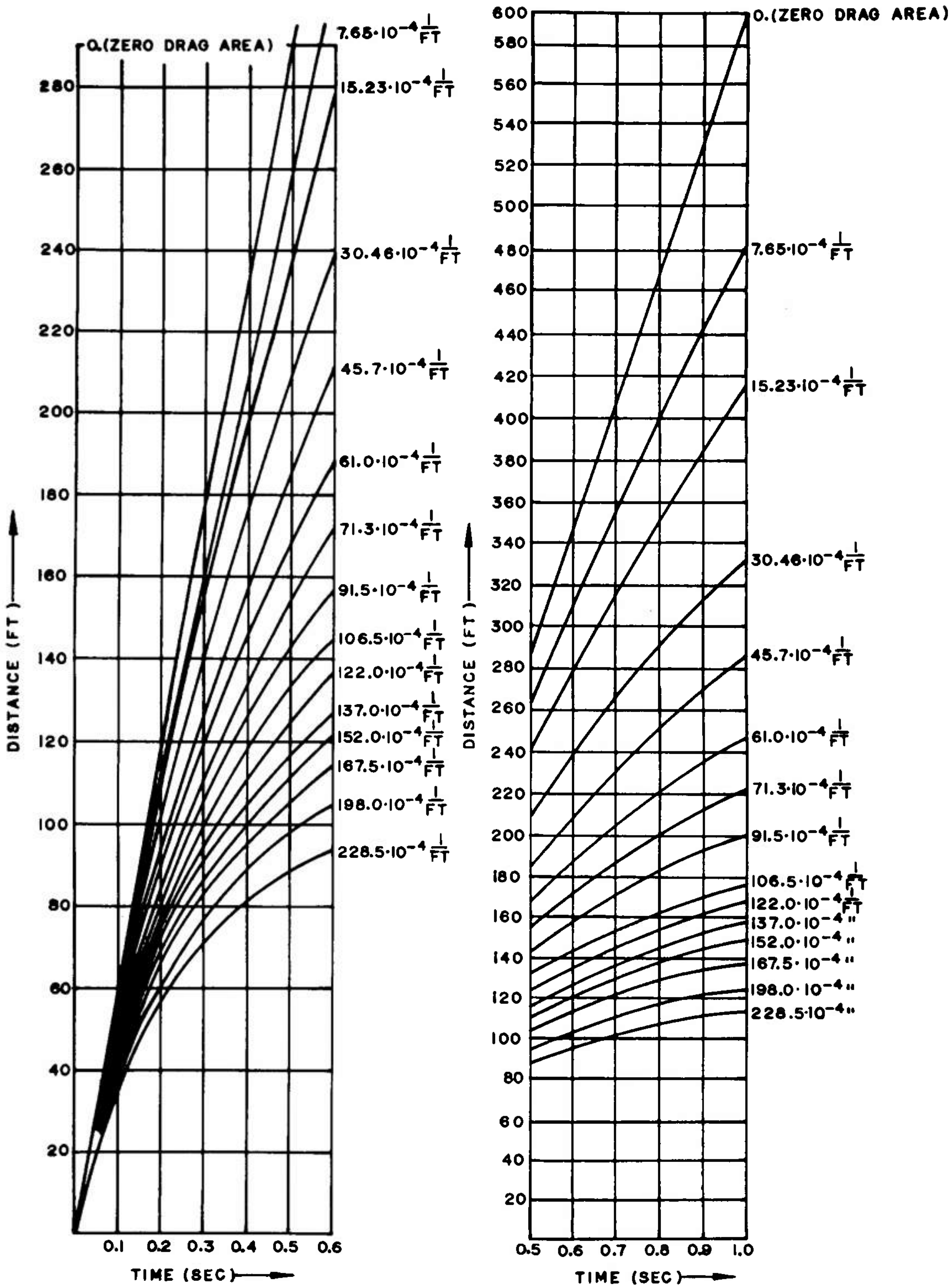


Figure 5-18. *Distance of Travel of Bodies with Various Drag Loadings, Launched at 350 Knots*

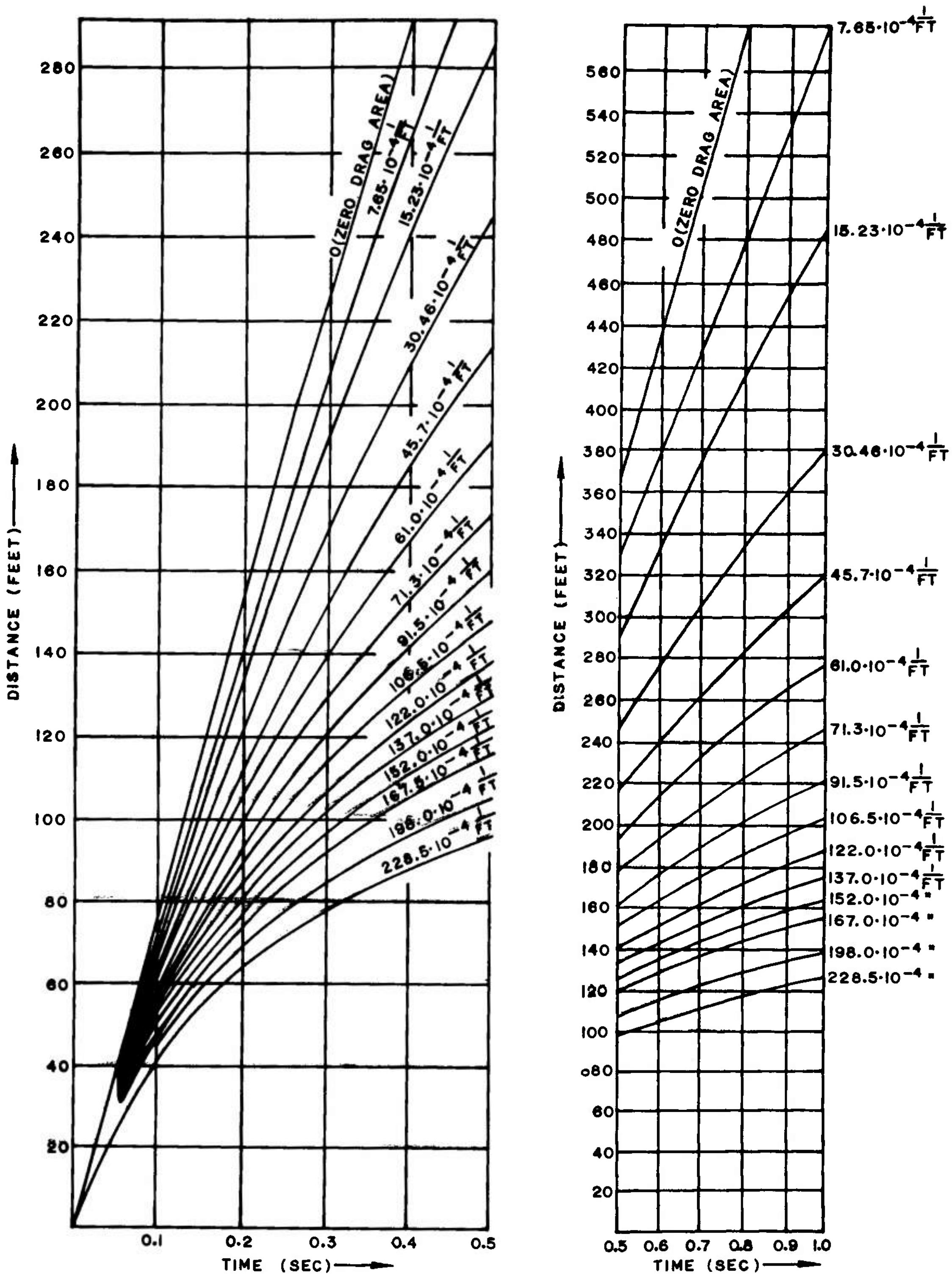


Figure 5-19. *Distance of Travel of Bodies with Various Drag Loadings, Launched at 435 Knots*

Equations 5-66, 5-67, and 5-68 yield the following specific values

$K_b = 7.65\times10^{-4}\ \text{ft}^{-1}$
$K_p = 115\times10^{-4}\ \text{ft}^{-1}$
$n = 15$

Figures 5-15 through 5-19 show the distance in feet that bodies with various drag loadings, K, will travel within a certain time, t. Each figure pertains to a different launching velocity. Since this example has specified a launching velocity of 130 knots, Fig. 5-15 applies. The difference in ordinate at any particular time between any pair of K lines represents the separation distance between two bodies having drag loadings represented by the K lines selected. Thus, the example requires the determination of the time, t, at which a suspended load of $K_b = 7.65 \times 10^{-4}$ and an uninflated parachute system of $K_p = 115 \times 10^{-4}$ will be separated by a disatnce, d, of 21 ft. If a line is drawn parallel to the K_b line (7.65×10^{-4}) and at a distance, measured along the ordinate, equal to 21 ft, the point along the abcissa at which it crosses the K_p (115×10^{-4}) line will be the required time, t. Since there is no curve shown for $K = 115 \times 10^{-4}$, it is necessary to extrapolate between curves $K = 122 \times 10^{-4}$ and $K = 106.5 \times 10^{-4}$. At $K = 122 \times 10^{-4}$, the value of t is determined as 0.35 sec while at $K = 106.5 \times 10^{-4}$ the value of $t = 0.375$ sec. The required value of t falls between these two values and is estimated at 0.36 sec. Substituting this value of t in Eq. 5-65, along with appropriate values of V_o (converted to feet per second), K_b, and n, the differential velocity is found to be

$\Delta V = 92$ fps

Having determined ΔV, its value is used in Eq. 5-64 and a snatch force of $P = 1{,}810$ lb is obtained. If the same load and parachute were dropped at 260 knots, the snatch force developed would be approximately 3,700 lb.

5-7.3 OPENING SHOCK

Modern aerial delivery systems incorporate special reefing, venting, collapsing, and other features in their design whose function is to attenuate the shock experienced by the system when the canopy is inflated during airdrop. Because of this successful development, opening shock is not generally of primary concern as it is usually attenuated to a value below the snatch force. Methods for the determination of opening shock by rigorous mathematical processes have not been satisfactorily developed. However, a method, based upon simplified theory, that yields results in satisfactory agreement with experimental data, is available.

The various stages of the parachute opening process are shown in Fig. 5-20. The term F_o is used to denote the actual force between the parachute and suspended load at the instant of full opening and can be calculated from the following equation

$$F_o = C_{D_o} S_o q_s \ x \ K \qquad (5\text{-}69)$$

where

$C_{D_o} S_o$	is the drag area of parachute, sq ft
q_s	is the impact pressure corresponding to velocity V_s at peak snatch force, psf
x, K	are dimensionless factors for various types of parachutes

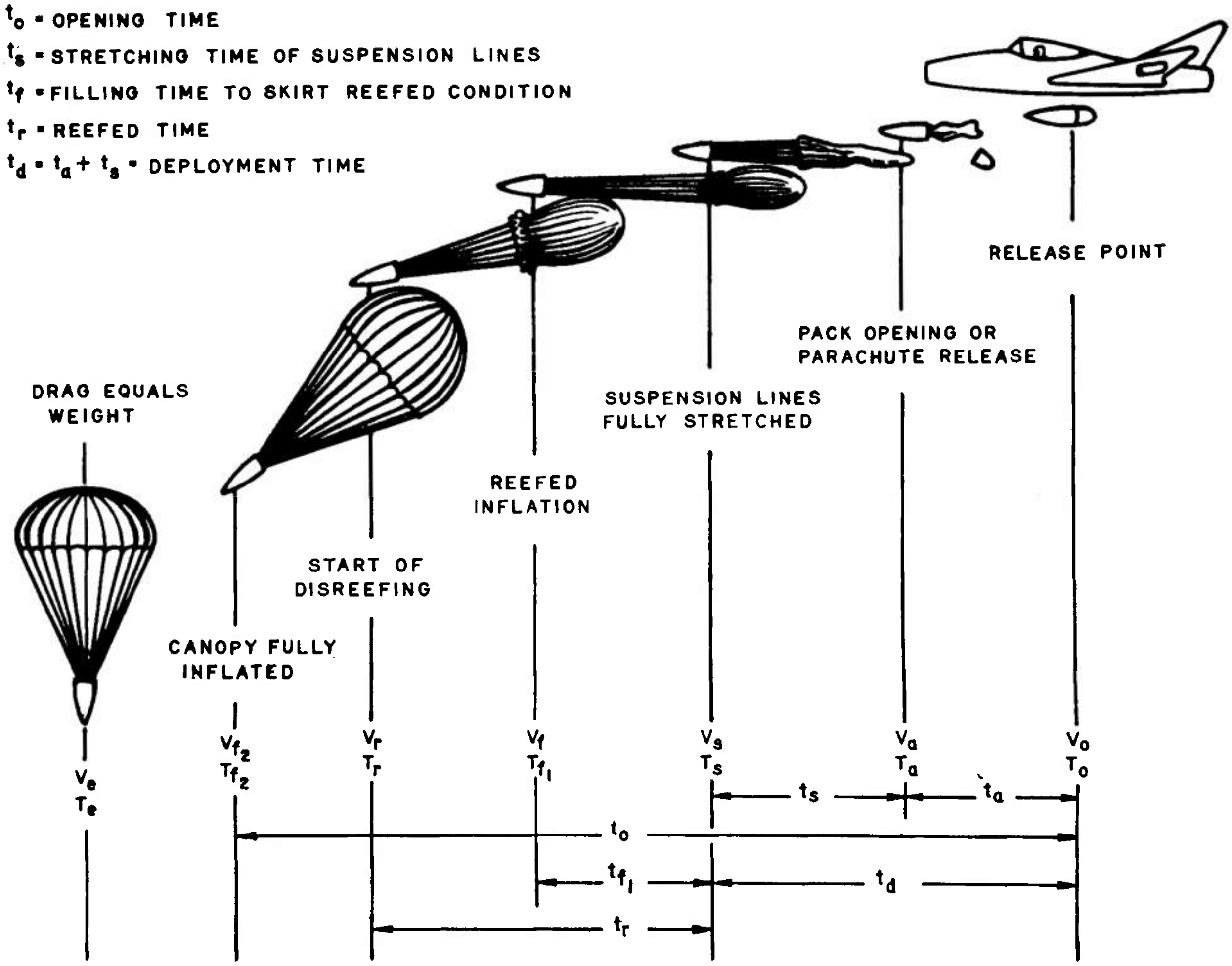


Figure 5-20. *Parachute-Opening Process*

The steps used in calculating opening shock are given below. The following information must be known initially

D_o	nominal diameter of canopy, ft
$C_{D_o} S_o$	drag area of parachute, sq ft
W_t	total weight, lb
$C_{D_b} S_b$	drag area of suspended load, sq ft
V_o	launching speed, fps
t_d	deployment time, sec (see Fig. 5-20)
x, K	dimensionless factors pertaining to type of parachute.

The speed decreases during the deployment time, t_d, from launching speed, V_o, to the velocity, V_s (fps), when the suspension lines are fully extended and the canopy inflation begins.

$$V_s = \frac{V_o}{1 + \dfrac{C_{D_b} S_b \, \delta \, g \, t_d \, V_o}{2W_t}} \tag{5-70}$$

where

δ is the mass density of air at given altitude, lb-sec^2 per ft^4

g is the acceleration due to gravity, ft per sec per sec

The impact pressure, q_s (lb per sq ft), corresponding to the velocity at snatch is

$$q_s = \frac{F_s}{C_{D_o} S_o} = \frac{C_{D_o} S_o \delta_o V_s^2}{2 C_{D_o} S_o} = \frac{\delta_o V_s^2}{2} \tag{5-71}$$

The filling time, t_f, or time required for the canopy to inflate, is determined next from

$$t_f = \frac{8D_o}{V_s^{0.9}} \cdot \frac{\delta}{\delta_o} \tag{5-72}$$

where

δ is the mass density of air at given altitude, lb-sec^2 per ft^4

δ_o is the mass density of air at sea level (0.00238 lb-sec^2 per ft^4)

The factor x in Eq. 5-69 compensates for the increasing drag area as the canopy inflates. Studies of canopy-opening processes show that the drag area increases approximately linearly with filling time; and, based upon this assumption, a dimensionless factor, A, was determined to define the approximate velocity decrease during filling time. Figure 5-21 shows the relationship between x and A and can be used in evaluating x. The factor, A, can be determined from the expression

$$A = \frac{2W_t}{C_{D_o} S_o V_s \delta t_f g} \tag{5-73}$$

The factor, K, in Eq. 5-69, can be selected from Table 5-4.

TABLE 5-4 *K* VALUES FOR TYPICAL PARACHUTE CANOPIES

Type of Parachute	Value of K
Solid, flat canopy	1.4
Ribbon and ring-slot canopy	1.0
Guide surface and rotofoil canopy	Data not sufficiently evaluated. Assume factor of 1.0

5-7.4 EXAMPLE OF OPENING-SHOCK CALCULATION

Assume a man carrying a standard parachute canopy with a diameter, D_o, of 82 ft, a drag area, $C_{D_o}S_o$ of 460 sq ft, a total suspended load, W_t, of 200 lb, and a load drag area, $C_{D_b}S_b$, of 4.5 sq ft is launched at sea level* at a speed, V_o, of 250 knots. The following calculations will yield the approximate value of the opening shock.

* Sea level conditions are assumed to simplify the calculations. At low altitudes, the ratio $\frac{P}{P_o}$ is approximately 1.

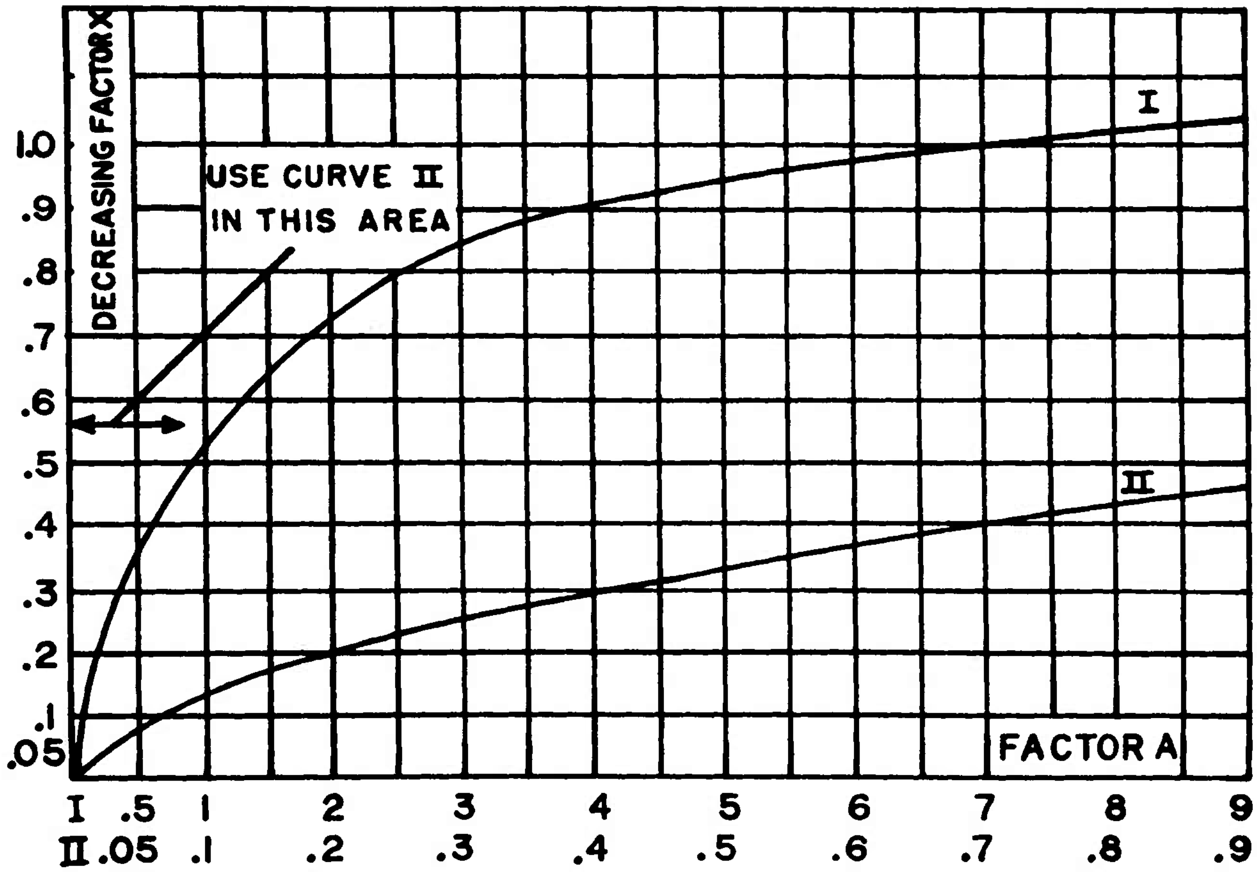


Figure 5-21. Opening Shock Decreasing Factor x Versus Factor A

The time of deployment, $t_d = t_a + t_s$ (Fig. 5-20), is assumed to be 0.7 sec. The speed decreases from V_o to V_s. From Eq. 5-70

$$V_s = \frac{250 \times 1.68894}{1 + \dfrac{4.5 \times 0.00238 \times 32.2 \times 0.7 \times 250 \times 1.68894}{2 \times 200}}$$

$$V_s = 337 \text{ fps}$$

Applying Eq. 5-71

$$q_s = \frac{0.00238 \times 337^2}{2} = 135 \text{ psf}$$

Equation 5-72 yields

$$t_f = \frac{8 \times 28}{337^{0.9}} (1) = 1.192 \text{ sec}$$

Factor A as determined by Eq. 5-73 is

$$A = \frac{2 \times 200}{460 \times 337 \times 0.00238 \times 1.192 \times 32.2}$$

$$A = 0.0283$$

The shock decreasing factor, x, corresponding to an A value of 0.0283 is found, from curve II of Fig. 5-21, to be 0.057.

The K factor for the type of parachute canopy being used in this example is 1.4.

The opening shock can now be determined by substituting the appropriate values just calculated in Eq. 5-69

$$F_o = 460 \times 135 \times 0.057 \times 1.4$$

5-8 SHOCKS FROM BALLISTIC IMPACTS

Shocks to military automotive vehicles caused by ballistic impact are of three general types: those produced by the kinetic energy of the projectile, as is the case with armor-piercing projectiles; those produced by high energy blast, as is the case with an exploding land mine; and those that are a combination of the preceding two, as in the case of impact by high explosive ammunition. The evaluation of the first type, the kinetic energy projectile, is rather simple. The assumption is usually made that all of the kinetic energy of the projectile is transmitted to the vehicle component that is hit. This energy, in ft-lb, is readily calculated by applying the classic equation

$$KE = \tfrac{1}{2} MV^2 \qquad (5\text{-}74)$$

where

M is the mass of projectile, lb-sec^2 per ft
V is the velocity of projectile at time of impact, fps

The peak force experienced by the member that received the impact can be determined by dividing the kinetic energy of the impact by the maximum deflection (in feet) that was momentarily produced by the blow.

The impact produced by high energy blast is not determined as simply. In general, orthodox military explosives develop a shock energy of approximately 200 ft-tons per pound of explosive upon detonation. This energy is released in all directions in the form of a pressure wave traveling at transonic velocity, producing impact stresses within the structural members in its path. Since this pressure energy moves outward in all directions like an expanding sphere, not all of it is effective against a vehicle. An analytical method for the accurate prediction of high energy blast effects upon vehicles is too complex for this book. Blast effects vary with the type of explosive used, the shape, size, and structure of the explosive casing, with the mass-related characteristics of the explosive, and other considerations. The best procedure at the present time is to base design calculations upon data obtained from field testing vehicles exposed to the specific high energy blasts under consideration.

In the case of shock produced by combination kinetic energy and high explosive ammunition, the resultant peak force is the sum of the forces produced by each of the separate effects. The peaks of the two component forces usually do not occur at the same instant, however, and hence the peak resultant force may occur at some time after the first peak of the component forces. More information on this subject is available (Refs. 32, 33, 34, 35, 36, 37 and 38).

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CHAPTER 6

THE PROPULSION SYSTEM CONCEPT (Refs. 1, 2, 3, 4)*

6-1 INTRODUCTION

The *propulsion system,* as the term is applied to the military automotive vehicle, encompasses that complete integration of related systems, major components, and accessories whose functions are concerned with the conversion of fuel energy to mechanical energy in a form suitable for propelling the vehicle. The propulsion system includes such major components as: the power plant with all of its accessories, the transmission system, transfer assemblies, final drives, driving axles, wheels, track assemblies, propellers, braking systems, and those parts of the suspension and steering systems that have a function in propelling the vehicle.

Since the conversion of the fuel energy to the form of energy required for propelling the vehicle (normally shaft torque at various speeds) can be accomplished by various means, the conversion system can be compared and evaluated on the basis of factors such as performance, fuel economy, bulk, weight, complexity, reliability, and total cost.

For the purpose of optimizing the vehicle power system, it is of utmost importance to compare and evaluate complete propulsion systems, rather than the separate components that make up the system. For example, the design or selection of an engine having the highest horsepower per pound ratio may not result in an acceptable or satisfactory propulsion system if it requires complex, bulky, and heavy power transmission equipment to convert the output torque characteristics of the power plant to a form compatible to the vehicle road load requirements. A seemingly inferior power plant, when evaluated on the basis of horsepower per pound, may have output torque characteristics that minimize the required transmission equipment, thus resulting in a superior propulsion system.

6-2 ENERGY CONVERSION AND PERFORMANCE REQUIREMENTS

The loads imposed on the power system of a tactical or combat vehicle during cross country operation fluctuate widely and, in general, are very severe. To meet the propulsion torque requirements under these operating conditions, the power source should, ideally, possess a high degree of flexibility, i.e., the power plant should be capable of producing a relatively high level of power throughout its speed range.

Assuming the source of mechanical power is rotary shaft power, the ideal source develops torque which varies inversely with speed to meet changing loading conditions. Deficiencies and limitations in the speed-torque relationship of the actual power source (power plant) must be met and compensated for by the power transmission system. In other words, it is the function of the power transmission system to extend the range of possible vehicle speeds and torques and minimize the developed power deficiencies of the various power plants.

6-3 POWER SOURCE CHARACTERISTICS

The power-speed (or torque-speed) relationships of various power producing units suitable for vehicular propulsion differ greatly, with some of them inherently superior to others for the intended purpose. For example, the torque-speed characteristics of conventional reciprocating piston engines are inferior to those produced by the reciprocating steam engine or the hydrostatic motor. These latter units are capable of producing a torque output inversely proportional to output shaft speed. High output torque is desirable for starting ve-

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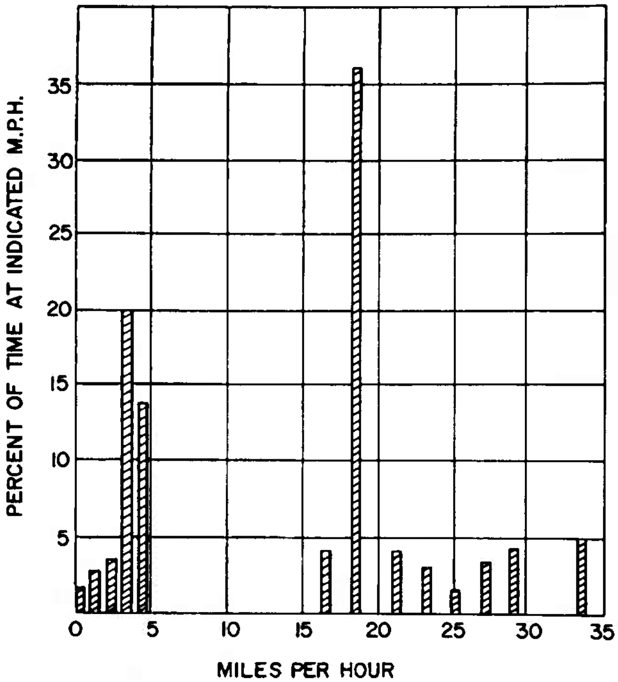

Figure 6-1. ***Percent of Vehicle Operating Life Spent at Different Speeds (Courtesy of Eng. Div., Detroit Arsenal)***

hicle from rest, for accelerating the vehicle to the desired speed, for hill climbing, and for negotiating adverse terrain.

Several important power source characteristics are directly related to the overall propulsion system and vehicle performance, and as such should be considered whenever propulsion systems are designed or evaluated. These include:

1. Maximum power (or torque) available at each speed within the useful operating range for: (a) short-time operation, and (b) continuous operation.
2. Range of speed and power over which satisfactory operation is possible or practicable.
3. Fuel consumption at all points within the expected range of operation.
4. Fuel consumption at preferred speeds (see Fig. 6-1).

The importance of evaluating and comparing complete propulsion systems can be emphasized by considering some currently popular methods of rating and comparing conventional piston engines used in all current standard military vehicles. Although many of the current military vehicle engines are supercharged; and although many of the following statements would apply to supercharged engines as well as unsupercharged engines, for the present purpose, the discussion should be considered as applying only to the units of the latter type.

A basic performance measure frequently considered is maximum brake horsepower per pound of engine weight (or inversely, weight per unit of power). This value indicates relative economy in the use of materials and may indicate factors such as thermal and volumetric efficiency (see par. 7-2). But, engines of the type under consideration, having low values of weight per unit power, frequently are high-speed units whose horsepower versus speed output characteristics include a sharply defined maximum. In other words, the maximum power is produced at high engine (and piston) speeds, and over a very narrow speed range. Since power is a function of both torque and speed, this peaked power output at high shaft speed indicates that maximum torque is developed at high engine speeds, and that it decreases rapidly with a decrease in engine speed. An engine having this characteristic lacks the flexibility desired for military vehicles and must be assisted by a complex transmission.

Another basis for rating power plants is engine volume per unit of power. This value indicates relative economy in the use of space, and, as such, is important from the military vehicle point of view. Although power plants possessing a low volume per unit of power may reflect efficient design, compactness may have been accomplished by resorting to high-speed engines having poor torque-speed characteristics.

A third basis for rating piston engines is horsepower per unit volume of piston displacement, e.g., horsepower per cubic inch. Since this standard of comparison is normally specified for maximum power conditions, it may be subjected to the same criticisms stated above, i.e., engines having relatively high values of horsepower per cubic inch may lack power at lower speeds.

Brake mean effective pressure (see par. 7-2) is frequently used as a standard for comparing engines. This factor reflects the volumetric efficiency, the brake thermal efficiency, and the fuel-air ratio (for a given fuel) at the rating point (see par. 7-2). Thus, it shows how well a piston engine uses its displacement (swept piston volume) to produce work. However, brake mean effective pressure, as is the case for torque which is directly proportional to the brake mean effective pressure, is normally specified for one point in the speed-power range, and as such, does not present a comparative picture of the speed-torque profile. An engine having a relatively high maximum value of brake mean effective pressure may have relatively poor torque characteristics at lower engine speeds. In general, an engine having relatively high and constant values of brake mean effective pressure throughout the lower portion of the useful speed range is desirable as a power source for military vehicles.

The performance ratio of power per unit piston area is a measure of effective use of the available piston area regardless of cylinder size. Since this ratio is proportional to the product of brake mean effective pressure and piston speed, and since piston speed is proportional to stroke and crankshaft speed, high values of power per unit piston area may be the result of high engine speeds. If a limiting piston speed is specified, this ratio reflects brake mean effective pressure, hence, torque developed within a given speed range.

A performance rating presented in Ref. 1 is horsepower per unit piston displacement per unit time, N_g. This relationship (also called "basic horsepower," hp per cubic inch per sec) may be expressed as

$$N_g = \frac{30 \times BHP}{D \times N} \tag{6-1}$$

where

BHP is brake horsepower
D is the piston displacement, cu in.
N is the engine speed, rpm.

Assuming that the same fuel is used in comparable engines, this performance rating indicates how well the swept piston volume is utilized to produce power by means of cylinder pressure rather than engine speed. Power plants having high values of horsepower per unit piston displacement per unit time produce high values of brake mean effective pressure at relatively low engine speeds, i.e., high values of torque at low engine speeds. Power plants having this characteristic are considered most suitable for military vehicle propulsion.

The above discussion points out some of the desirable characteristics for power plants as elements of the propulsion system, and also indicates the importance of the rating system used to evaluate the potential power plants.

Other power plants, such as the multiple-shaft gas turbine (considered most suited of the various gas turbine configurations for automotive propulsion systems), have torque-speed characteristics that are theoretically superior to piston engines for the intended purpose of vehicle propulsion; but other factors, such as the brake specific fuel consumption (see par. 7-2) throughout the useful speed range and variation in output, must be considered when the total propulsion system is designed or evaluated.

The power output of the multiple-shaft turbine is extremely sensitive to compressor inlet temperature, hence, to ambient temperature. The wide range of temperatures encountered in the military environment dictates that serious consideration should be given to this performance characteristic when propulsion systems are evaluated. Another unfavorable performance characteristic of the multiple-shaft gas turbine is its high specific fuel consumption at partial loads. Its performance, in this respect, is far superior to that of single-shaft turbine, however.

6-4 CHARACTERISTICS OF TORQUE CONVERSION SYSTEM

As previously stated, the most important function of the power transmission system (power train) of any military vehicle is to transform the power developed by the power plant into a form, in terms of shaft speed and torque, compatible to the requirements at the ground contacting elements. It follows that the degree of transformation required is directly proportional to the divergence between torque generated by the power source and the torque required at the wheels or tracks.

This problem can be emphasized and illustrated by considering the speed-ratio changing unit (transmission) of the power train required by a number of different power sources.

For vehicles utilizing conventional piston engines (spark-ignition and compression-ignition), a torque-multiplying unit (or speed-ratio changing unit) is essential. Ideally, an infinitely variable, automatically controlled unit is required to optimize both performance and fuel economy. The term *maximum performance* refers to condition of maximum horsepower that the propulsion system is capable of producing at the driving wheels or tracks. This occurs when the power plant is developing its maximum power and the ideal transmission is selecting the optimum speed ratio at every instant. Thus, for maximum performance, the engine operates at constant speed (full throttle maximum power) and the transmission constantly and continuously changes the speed ratio as vehicle speed changes.

Fuel economy is one of the most important factors related to the design, evaluation, or selection of propulsion systems. The function of the transmission as a means of optimizing fuel economy can be indicated by considering the fuel consumption characteristics of the conventional piston engine. In general, there is an ideal operating range, in terms of engine speed for any given vehicle power requirement, within which brake specific fuel consumption is minimized.

Figure 6-2 shows typical power-specific fuel consumption curves for a conventional piston engine. It is obvious that there is an optimum speed at which the engine should develop the required power to obtain minimum fuel consumption. Since it is the transmission reduction ratio that determines the engine speed for a given vehicle propulsion power requirement, the vehicle fuel economy will be determined by how well the transmission can regulate the engine to operate in the optimum range.

The ideal, infinitely variable transmission would continuously vary the reduction ratio under varying load requirements so that the engine would operate at its optimum point (minimum brake specific fuel consumption) under all conditions. These relationships (maximum performance and maximum economy as a function of the overall power system) are discussed more thoroughly in Chapter 8. In addition, some of the limitations of actual transmissions are reviewed in the same chapter. The ideal transmission has not been developed; existing systems for torque conversion succeed in varying degrees in accomplishing the desired results.

It is apparent, however, that the duties and complexity of the transmission (torque multiplying device) increase as the deficiencies of the power plant, with respect to producing the required propelling torque, increase. Although the previous statements emphasize the speed-torque characteristics of a potential power plant, the maximum power developed by the power plant also influences the requirements of the transmission. Vehicles having low power-to-weight ratios require very large reduction ratios to meet the road torque requirements for starting the vehicle, acceleration, and other high load operations, such as hill climbing. Yet, in order to utilize the engine most efficiently, the reduction ratio should change as the speed of the vehicle changes. Since the reduction ratio (speed ratio between the engine and the drive axles or sprockets) for a low powered vehicle must be very high, a large number of transmission elements, such as gears, are required to optimize the vehicle's performance. In general, the weights and sizes of the transmission units increase as the limitations of the power sources increase.

6-5 SUMMARY AND FUTURE DEVELOPMENTS (Ref. 6)

The preceding discussion could be expanded to include a comparison of operating characteristics of other systems, e.g., the volume per unit output of a gas turbine is about two-thirds to one-sixth as large as that of a conventional piston engine of similar power rating. It should be clear, however, that the interdependence of the various components within a given propulsion system is such that not only must a rational method be used to compare individual components, e.g., transmission efficiency throughout the design speed range, but total energy conversion systems should be compared rather than only the power generating components.

Three trends which will influence the choice of propulsion units available to the designer, may be noted in the development of power sources suitable, or potentially suitable, for automotive propulsion systems. The first is concerned with the development of conventional piston engines. Multifuel operation, variable pressure supercharging, and air-cooling of compression-ignition engines are examples in this category. A second trend is expressed in the effort to develop relatively new sources of mechanical power. The greatest effort in this area has been directed toward the gas turbine; however, the free piston engine, rotary piston engines, and the Stirling external combustion engine have received increased attention in recent years.

The third trend is expressed in the development of the various generators that are the sources

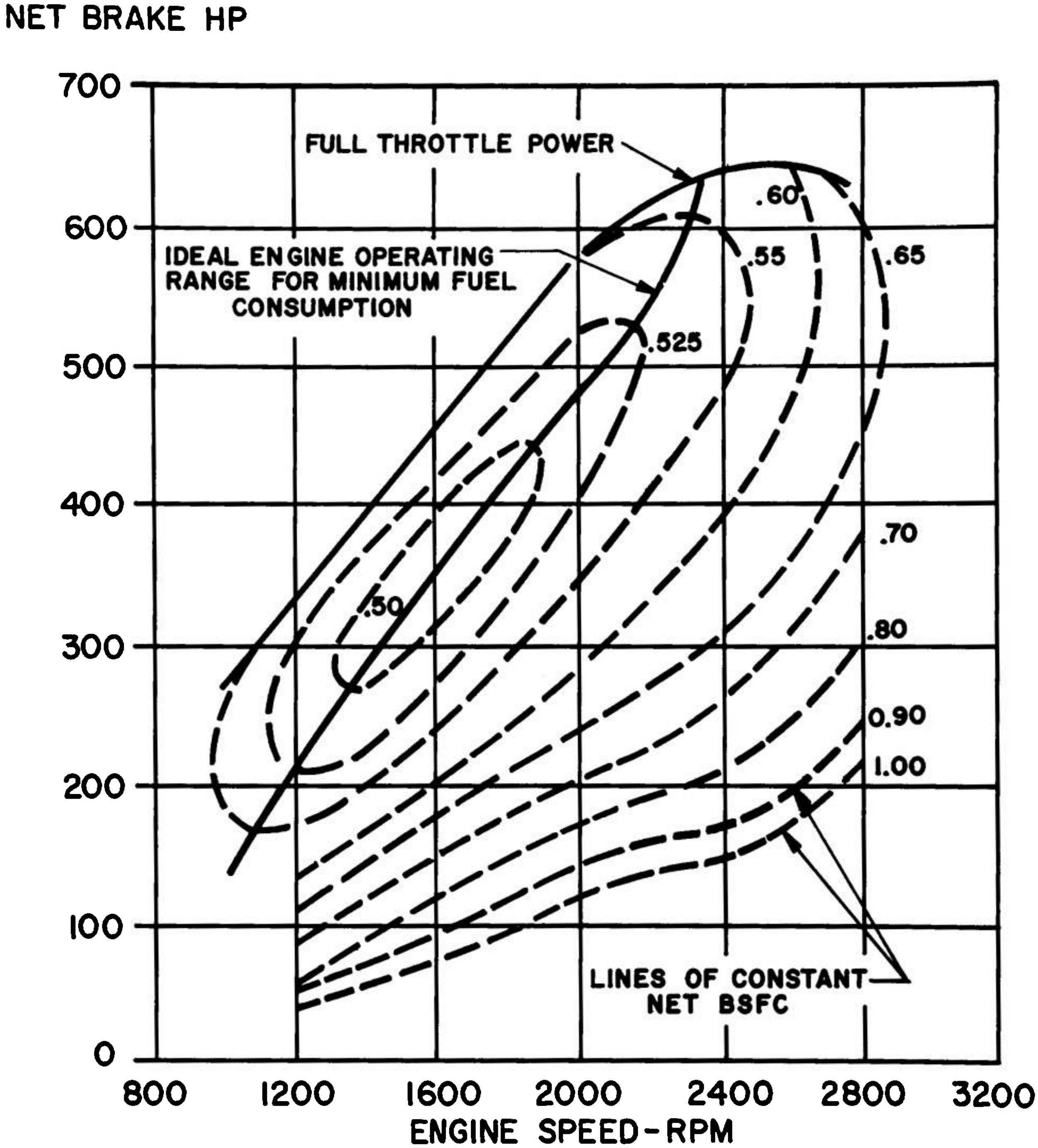

Figure 6-2. Current Development Engine Full- and Part-Throttle Power and Brake Specific Fuel Consumption

of direct electrical power. In this category are: fuel cells, thermoelectric generators, thermionic converters, magnetohydrodynamic generators, and solar cells. Fuel cells are considered extremely promising sources of power for vehicular propulsion. Used in conjunction with series-wound traction motors, they would provide the highly desirable, inversely proportional speed-torque relationship for propelling off-the-road vehicles. This electric drive system would compete with, and be compared with, other integrated systems including the electric drive system utilizing constant speed, gas turbine driven generators and suitable traction motors. In any event, the suitability of such new developments for incorporation in future vehicles, should be made on the basis of comparing overall propulsion unit capabilities rather than individual components of such units.

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CHAPTER 7

THE POWER PLANT*

SECTION I GENERAL DISCUSSION

The characteristics of military vehicles place great emphasis upon the requirements of the power plant. Prior to, and during, World War II power plants used in military vehicles were, with several exceptions, adaptations of existing commercial designs. The battlefield proving ground soon revealed the shortcomings of these standard commercial power plants and established the need for special military designs. The outstanding achievements of the war were possible due to the remarkable advances that had been made in the development of specialized power plants. At the present time, power plants for military vehicles consist of two general types: Basically standard commercial power plants modified to meet military specifications, and power plants specially designed for military applications. Logistical factors justify the continued use of the former types for all applications where they have been proven suitable, while experiences in past wars have proved that conditions do exist which justify the use of specialized military automotive power plants, particularly in large combat and tactical vehicles.

Demands upon the military power plant require it to be simultaneously a heavy-duty, low-speed, tractor engine capable of high lugging ability for superior tactical mobility, and a flexible, high-speed automotive engine capable of high vehicle speeds for strategic moves. It is required to provide maximum cruising range on a limited amount of fuel, either through superior fuel economy or by virtue of its small size, or, preferably, both. It must be capable of operating, without overheating, in ambient temperatures of 125°F even though housed in an armored engine compartment ventilated only through restrictive overhead grills and must operate equally well at temperatures down to −65°F. It must be capable of operating when completely submerged in either fresh or sea water, be extremely rugged, absolutely dependable, and require a minimum of corrective maintenance.

Unfortunately, these characteristics are not compatible with one another. Each is maximized only at the expense of others. High performance at low speed is obtained at the expense of top speed, power, and economy. Small power plant size and high specific output dictate high engine speeds. Ruggedness and reliability usually lead to increased weight, even when high quality materials are used. Compactness and accessibility for maintenance are, to a large extent, incompatible. High quality, lightweight materials are expensive and may be critical in supply during a national emergency. Thus, economic considerations introduce further incompatibility. Since production economy is proportional to quantity produced, standardization of a power plant to serve as many vehicles as possible is desirable. Standardization also brings about simplification of maintenance and the logistics of repair parts, but necessitates building into one power plant the characteristics desired for several diversified applications.

Since none of the desired characteristics can be maximized except at the expense of others, the design of the power plant becomes a series of compromises. A designer, upon undertaking the design of a new military power plant, is faced with the need to resolve many important questions. These fall into three general groups, namely: (a) questions pertaining to military policy and logistics, (b) questions pertaining to the relative importance of the desired characteristics, and (c) emergency or technical questions.

Questions in the first group deal with the quantity that will be needed, both in peace and war, the variety of types and sizes that will be necessary, the production facilities that are or will

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TABLE 7-1 RELATIVE EMPHASIS IN DESIGN*

Requirement	ORDER OF EMPHASIS	
	Military Power Plant	Commercial Power Plant
Reliability	1	5
Ability to Withstand Abuse	2	11
Ease of Maintenance	3	12
Space Occupied	4	6
Weight	5	8
Simplicity	6	9
Life	7	7
Economy of Operation	8	4
Investment for Production	9	10
Ease of Manufacture	10	3
Minimum Cost	11	1
Public Acceptance (saleability)	12	2

* This list is suggestive only, and subject to revision to make particular vehicles conform to requirements outlined by military characteristics.

be available, and the types and quantities of fuel that will be available. The answers to such questions are outside the scope of this book but have important bearing upon the design of the power plant.

The designer can improve his condition if the planning for a complete family of power plants to meet all predictable needs is done at the outset of the program and made known to him. He will then strive to make use of as many common details between the various engines as possible, such as a common size of piston or valve. The design of the tooling can also be aimed at the development of composite tools that can be adjusted for use in the manufacture of several different power plants. In this way he can use to advantage the economics of common design and interchangeable tooling. Care must be taken, however, not to stymie the development of future designs by this type of program.

The second general group of questions confronting the designer are those pertaining to the relative importance of the desired characteristics. In addition to reconciling the usually conflicting demands of the many agencies whom he is striving to please with his design, he has to consider such factors as cost, ease of manufacture, appearance, reliability, ease of maintenance, etc. Many of these are incompatible with each other. Greater reliability usually increases cost while ease of manufacture is often the opposite of ease of maintenance. The unskilled designer tends to strive for a happy compromise by averaging all of the design demands. If he succeeds in doing this he will produce a mediocre design that will satisfy no one. The skillful designer considers all of the demands, arranges them in a definite order of emphasis in accordance with the use for which the particular article is intended, and proceeds accordingly. This may result in some dissatisfied customers, but the majority will be satisfied.

A major difference exists, however, in the order of emphasis placed upon the requirements of a military versus a commercial power plant. The designer of a military power plant must change his entire point of view in this respect, because factors vital in a military application are of minor importance in civilian applications and the opposite applies to other factors. Table 7-1 shows some of the main factors considered in power plant design and a suggested order of emphasis as applied to both commercial and military applications.

The third general group of questions confronting the designer of military power plants includes the technical considerations. The answers to these questions must be made in light of the two previous groups. The bulk of this chapter is devoted to the technical considerations of automotive power plants, particularly military power plants, and is intended to provide the designer with a useful, general background in the subject to enable him to make rational decisions in the selection and design of power plants and auxiliary power plant components.

SECTION II ENGINES (Ref. 1)

Heat engines which power all current military automotive vehicles may be classified as external combustion or internal combustion engines. In the external combustion engine, the working fluid is entirely separated from the heat source; i.e., heat transfer is effected by means of a heat exchanger. In the internal combustion engine, the working fluid consists of the products of combustion of the fuel-air mixture itself. Individual engines within each of the above categories may be classified as (a) reciprocating, (b) rotary, (c) compound, or (d) thrust, depending on whether the working fluid acts directly on (a) pistons, (b) turbine blades, (c) pistons and blades, or (d) is subjected to a time rate of change of linear momentum. The following is a list of the common types of heat engines grouped according to their general classification:

General Class	Type	Mechanical Aspect
External Combustion	Steam engine	Reciprocating
	Steam turbine	Rotary
	Hot-air engine	Reciprocating
	Closed-cycle gas turbine	Rotary
Internal Combustion	Spark-ignition engine	Reciprocating
	Compression-ignition engine	Reciprocating
	Gas turbine	Rotary
	Thermal-jet engine	Rotary-Thrust
	Free-piston gasifier turbine	Compound
	Rocket-jet engine	Thrust

7-1 THERMODYNAMIC CYCLES (Refs. 3, 4, 5)

Although the internal combustion engine does not operate on a thermodynamic cycle, the concept of a cyclic process is useful in analyzing and comparing the various types of combustion engines. By applying idealized cycles and basic thermodynamic principles, hypothetical cycles may be generated for the various actual engine types. If air is assumed to be the working fluid in the hypothetical cycle, the term air-standard cycle is used. Air-standard cycle analyses are useful to compare combustion engine types, to study operating conditions, and to determine theoretical efficiencies. Actual efficiencies are always much lower than the air-standard efficiencies.

7-1.1 CARNOT CYCLE

The Carnot cycle is an idealized, nonflow cycle that is defined as the simplest and most efficient cycle working between two definite temperatures. In this cycle (Fig. 7-1) for a gas, the medium is compressed isentropically (reversible adiabatic process) from Point b to c, to the heat-addition temperature, T_A. The next process, cd, is heat addition, reversibly, and at constant temperature. Process da is an isentropic (reversible adiabatic) expansion to sink temperature, T_R. Finally, the gaseous medium is reversibly cooled at constant temperature, T_R, process ab, until the initial state is again reached.

Since the limiting cycle parameters for the Carnot cycle are the heat-addition and heat-rejection (sink) temperatures, the thermal efficiency of the cycle is given in terms of these temperatures:

$$\eta_t = 1 - \frac{T_R}{T_A} \tag{7-1}$$

where T_A and T_R are the maximum and minimum temperatures, respectively.

The Carnot cycle is the criterion or reference cycle for availability of heat added in heat-engine processes. No other cycle can be more efficient than the Carnot cycle for given operating temperature limits.

7-1.2 OTTO CYCLE

The ideal air-standard cycle for the spark-ignition gasoline engine is the Otto cycle or constant-

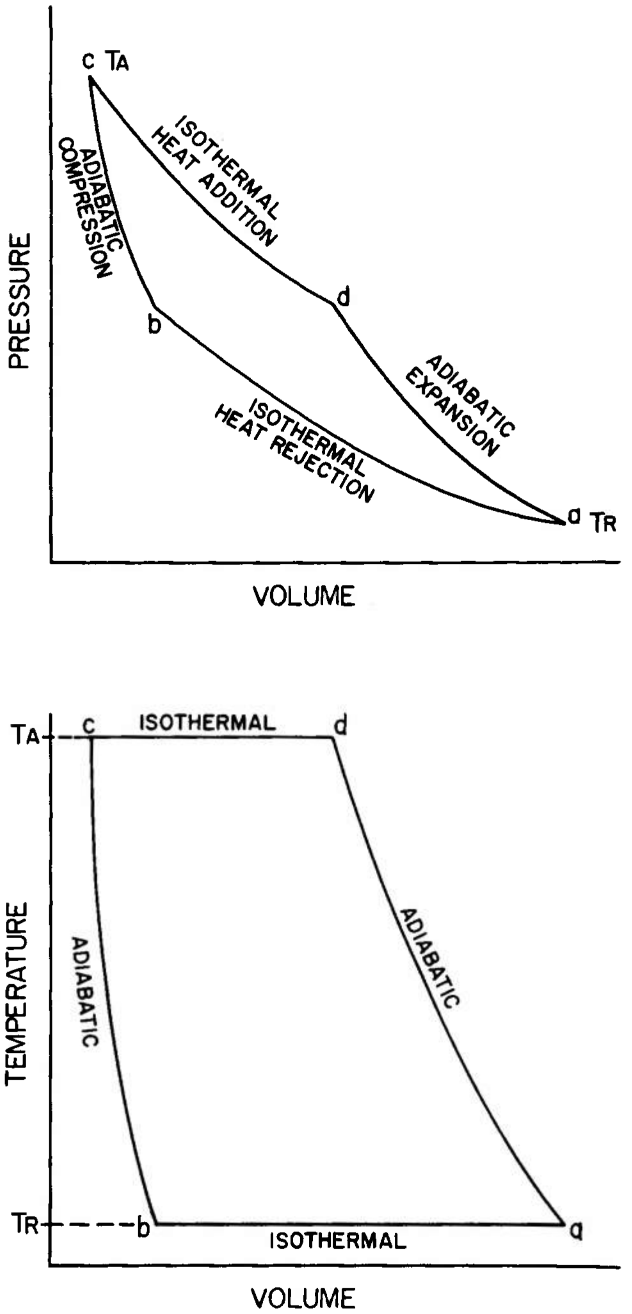


Figure 7-1. ***Carnot Cycle for a Gas***

volume cycle. In the Otto cycle, heat addition and heat rejection take place at constant volume, while compression and expansion are isentropic processes. Figure 7-2 shows the pressure-volume state diagram for the air-standard Otto cycle. The sequence of processes is as follows:

ab, isentropic compression
bc, constant-volume addition of heat

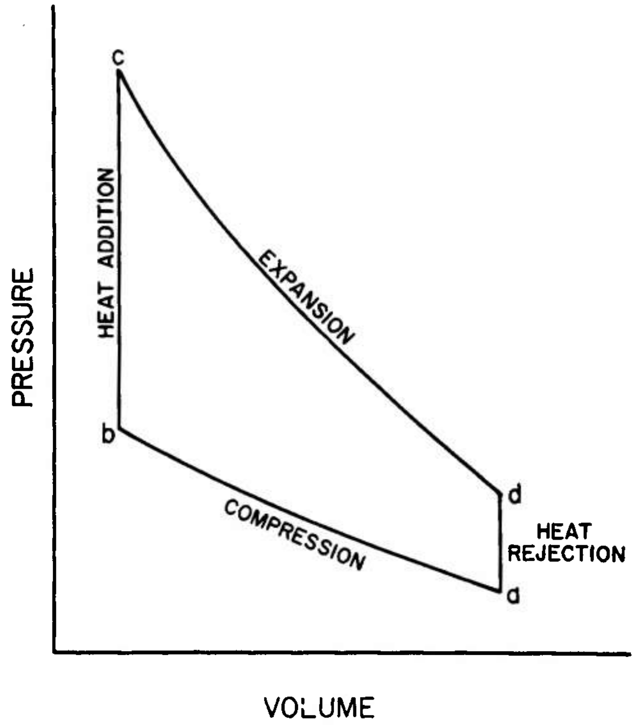


Figure 7-2. ***Air-Standard Otto Cycle***

cd, isentropic expansion
da, constant-volume rejection of heat

The thermal efficiency, η_t, of an ideal Otto cycle (Ref. 6) is:

$$\eta_t = 1 - \frac{1}{r_v^{k-1}} \qquad (7\text{-}2)$$

where r_v is the expansion ratio or the compression ratio and k is the ratio of the specific heats at constant pressure and constant volume of the gas used as the working medium.

Equation 7-2 shows that the thermal efficiency of the Otto cycle is a function of compression (expansion) ratio and the properties of the working medium. The value of k is not a constant; it varies not only with the kind of gas used, but also with temperature. If the relationship between thermal efficiency and compression ratio is plotted, assuming a constant value for k (Fig. 7-3), the following trend is observed: In the lower range of r_v, the gain in efficiency is much greater per increment of compression ratio increase than is the gain in efficiency per increment in the higher ranges of r_v. In other words, the rate of increase in efficiency with an increase in compression (expansion) ratio decreases as the compression ratio is increased.

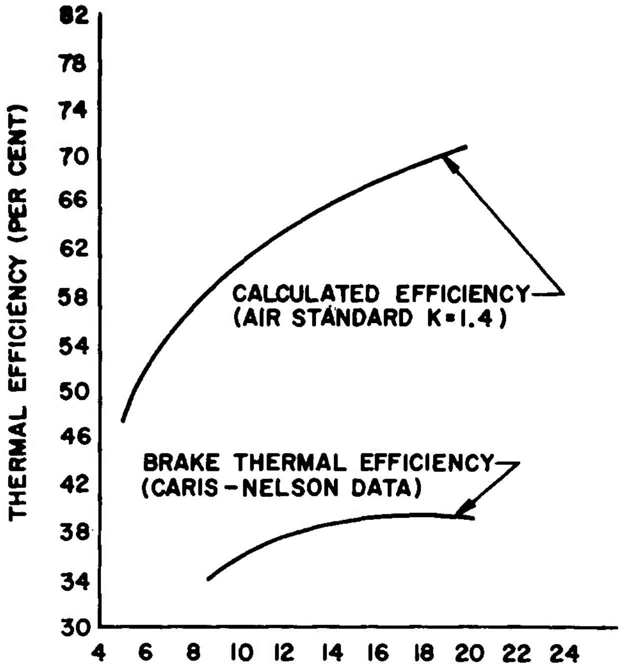


Figure 7-3. *Variation of Thermal Efficiency With Compression Ratio for the Otto Cycle*

The actual or brake thermal efficiency of an internal combustion engine is defined as:

$$\eta_t = \frac{\text{work output}}{\text{heat input}} \quad (7\text{-}3)$$

For a given compression ratio, the brake thermal efficiency of an engine, which includes friction, pumping losses, heat losses, and combustion phenomena, will be less than the thermal efficiency of the ideal cycle. The brake thermal efficiency of a typical engine, as shown in Fig. 7-3, does not vary with r_v in the same manner as the ideal cycle variation. The major factors affecting the brake thermal efficiency-compression ratio relationship are fuel characteristics and combustion phenomena (Ref. 7).

The Caris-Nelson experimental results show that, for a typical automobile gasoline engine, the brake thermal efficiency reaches a maximum at a compression ratio of 17:1. An analysis of the results led to the conclusion that the major factors causing the decrease in thermal efficiency at compression ratios above 17:1 are: delay in the completion of the combustion process, and chemical dissociation of the products of combustion. Other well-known limiting factors are pre-ignition and detonation, which are related to combustion chamber design and fuel characteristics (see par. 7-2.2.4).

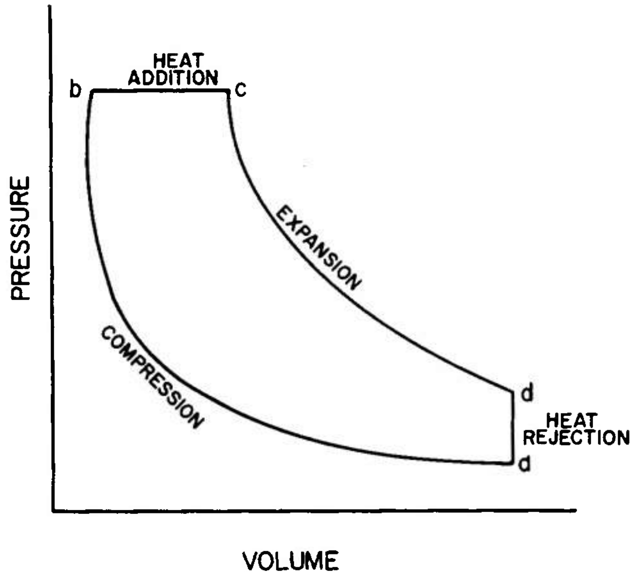


Figure 7-4. *Air-Standard Diesel Cycle*

7-1.3 DIESEL CYCLE

In the air-standard Diesel cycle, Fig. 7-4, heat is added at constant pressure: Process *bc* on the diagram. The processes for the ideal cycle are:

ab, isentropic compression
bc, constant-pressure addition of heat
cd, isentropic expansion
da, constant-volume rejection of heat.

The thermal efficiency, η_t, of an ideal Diesel cycle (Ref. 8) is:

$$\eta_t = 1 - \frac{1}{r_v^{\,k-1}}\left(\frac{L^k - 1}{k(L-1)}\right) \quad (7\text{-}4)$$

where

r_v is the compression ratio, $\frac{V_a}{V_b}$

k is the ratio of specific heat at constant pressure and specific heat at constant volume of the working medium, $\frac{C_p}{C_v}$

L is the cutoff or load ratio, $\frac{V_b}{V_c}$, (V_c is determined by the termination of the fuel injection process)

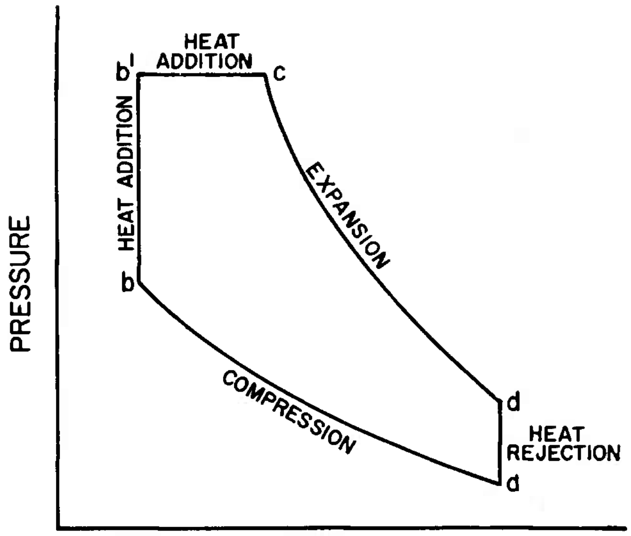


Figure 7-5. *Air-Standard Dual Cycle*

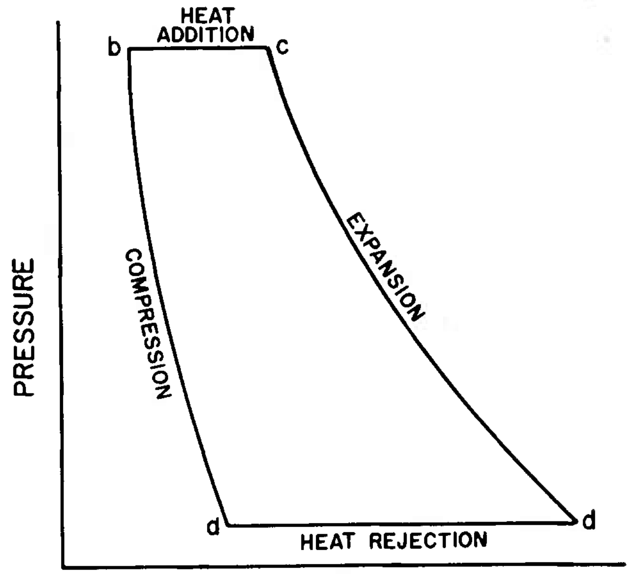


Figure 7-6. *Air-Standard Brayton Cycle*

Equation 7-4 shows that the thermal efficiency of the Diesel cycle depends on compression ratio and on the total heat input. The efficiency increases with an increase in compression ratio and decreases with an increase in heat input, i.e., with load. The ratio of heat supplied to heat rejected decreases as the load ratio increases. At normal loads, a Diesel engine is less efficient than a gasoline Otto cycle engine for a given compression ratio. However, since the Diesel engine compresses air only, the compression ratio with present fuels may be higher than that in an Otto engine. Under operating conditions, the thermal efficiencies of the Diesel and Otto engines are approximately equal.

7-1.4 DUAL CYCLE

Modern high-speed compression-ignition engines do not operate on a constant-pressure heat-addition cycle. Heat is supplied partly in a constant-volume process and partly in a constant-pressure process. This dual cycle is a result of the relationship between the time available for fuel injection and the time required for injection at high engine speeds. An ideal air-standard cycle based on the compound combustion process can be studied. The pressure-volume diagram for such a dual cycle is shown in Fig. 7-5, where the processes are:

ab, isentropic compression
bb', constant-volume addition of heat
b'c, constant-pressure addition of heat
cd, isentropic expansion
da, constant-volume rejection of heat

The air-standard thermal efficiency of the dual or mixed cycle (Ref. 9) is:

$$\eta_t = 1 - \frac{1}{r_v^{k-1}}\left(\frac{r_p L^k - 1}{r_p - 1 + k r_p (L-1)}\right) \qquad (7\text{-}5)$$

where

k is the specific heat ratio
r_v is the compression ratio (V_a/V_b)
L is the load or cutoff ratio $(V_c/V_{b'})$
r_p is constant-volume pressure ratio $(P_{b'}/P_b)$

For any given compression ratio, the air-standard thermal efficiency is between that of the Otto cycle and the Diesel cycle; the relative position is determined by the ratio of heat supplied at constant volume to that supplied at constant pressure.

7-1.5 BRAYTON CYCLE

The Brayton cycle is the basic air-standard for all modern gas turbine units. The continuous-combustion gas turbine can be represented by an idealized air-standard cycle as shown in Fig. 7-6, where the following processes apply:

ab, isentropic compression
bc, constant-pressure addition of heat

cd, isentropic expansion

da, constant-pressure rejection of heat

By using basic thermodynamic relations, the air-standard thermal efficiency of the Brayton cycle can be expressed as

$$\eta_t = 1 - \frac{1}{\left(r_p\right)^{\frac{k-1}{k}}} \qquad (7\text{-}6)$$

where k is the specific heat ratio and r_p is the pressure ratio

($r_p = p_{max}/p_{min}$ of the cycle).

Since Eq. 7-6 can also be written as

$$\eta_t = 1 - \frac{1}{(r_v)^{k-1}} \qquad (7\text{-}7)$$

where r_v is the adiabatic compression ratio, the air-standard efficiency expressions for the Otto and Brayton cycles are identical.

In the Diesel cycle, the addition of heat at constant pressure made the cycle less efficient than the Otto cycle, for a given compression ratio. As the efficiency equations show, the addition of heat at constant pressure in the Brayton cycle does not make the cycle less efficient than an Otto cycle operating at the same compression ratio.

Brake thermal efficiencies of typical gas turbines are usually lower than those of comparable spark-ignition or compression-ignition reciprocating engines. This can be explained on the basis of thermodynamic and material considerations. The thermal efficiency of an actual gas turbine is governed by the temperature requirements of the unit. The continuous high-temperature gas flow through the turbine (as opposed to the intermittent exposure to high-temperature gases experienced in reciprocating engines) combined with metallurgical factors limits the usable pressure ratio (Ref. 10).

7-1.6 STIRLING CYCLE

The ideal Stirling cycle, shown in Fig. 7-7, consists of the following processes:

ab, isothermal compression

bc, constant-volume addition of heat, at temperature T_A

cd, isothermal expansion

da, constant-volume rejection of heat, at temperature T_R

The Stirling cycle is less efficient than the Carnot cycle since the former includes constant-volume heat transfers, while the latter includes adiabatic processes between T_A and T_R. The Stirling cycle can be made as efficient as the Carnot cycle if a regenerative arrangement is used to transfer reversibly the heat from process *da* to process *bc,* so that all external heat transfer takes place at T_A and T_R.

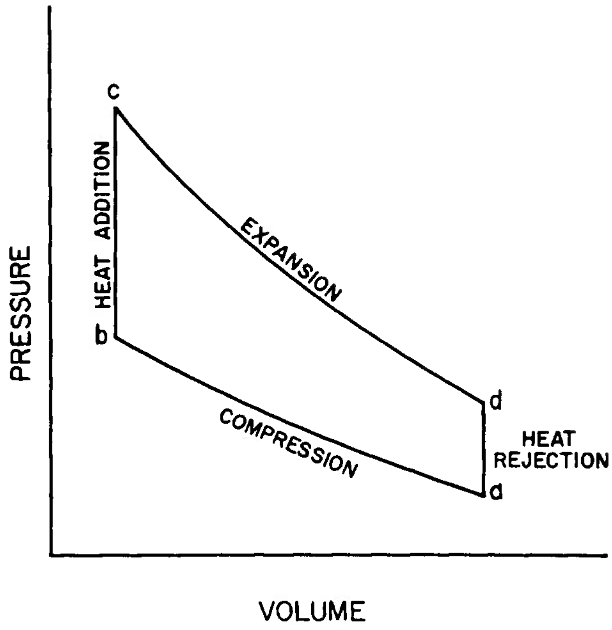


Figure 7-7. Air-Standard Stirling Cycle

Stirling cycle engines are hot-air or external combustion reciprocating piston engines. The fuel, combustion air, and the products of combustion do not enter the engine cylinder; the working medium is sealed within the working spaces of the engine. The working medium passes through a true thermodynamic cycle with heat transfer affected by heat exchanges within the system.

7-1.7 COMPARISON OF AIR-STANDARD CYCLES

With regard to thermal efficiency, the idealized air-standard cycles can be compared by selecting certain parameters. The present comparison will be limited to the Otto, Diesel, and mixed (dual) cycles (Ref. 3).

For a constant expansion ratio and a constant heat input, the thermal efficiency decreases in the following order:

a. Otto cycle
b. Mixed cycle (high-speed Diesel)
c. Diesel cycle.

For a constant heat input and a constant maximum pressure (in each cycle), the thermal efficiency decreases in the following order:

a. Diesel cycle
b. Mixed cycle
c. Otto cycle.

For a constant maximum pressure and a constant temperature, the thermal efficiency decreases in the following order:

a. Diesel cycle
b. Mixed cycle
c. Otto cycle.

The equations indicate that the thermal efficiencies of the Brayton and the Otto cycles are equal for a given expansion ratio, and that the Diesel cycle is less efficient than the Otto cycle at a given expansion ratio. The difference may be explained as follows (Ref. 11). In the Brayton cycle, the expansion ratio is constant for each increment of heat added, because the gases can expand to atmospheric pressure. In the Diesel engine, the expansion is limited by the piston to a pressure far above atmospheric.

7-2 RECIPROCATING INTERNAL COMBUSTION ENGINES

7-2.1 BASIC PERFORMANCE FACTORS AND RELATIONSHIPS

Some performance factors or characteristics that provide bases for comparative evaluation of different engines for any application are described briefly below.

The total horsepower developed within the cylinders of an engine is termed *indicated horsepower, ihp*. The part of indicated horsepower that does not appear as shaft or *brake horsepower, bhp*, but is expended to overcome friction of the mechanical parts and to effect the induction and exhaust processes, is called *friction horsepower, fhp*. The ratio of brake horsepower to indicated horsepower is the mechanical efficiency:

$$\eta_m = \frac{bhp}{ihp} \tag{7-8}$$

As shown in Fig. 7-8 friction horsepower, which is the power used to overcome the friction of the mechanical components and to pump the gases in and out of the engine, is a function of engine speed; thus mechanical efficiency, for a given engine, decreases with an increase in engine speed.

The *mean effective pressure, mep*, is defined as the theoretical constant pressure which, if exerted on the cylinder for the entire stroke, would produce the power actually produced by the varying cylinder pressures. The terms *brake mean effective pressure, bmep*, and *indicated mean effective pressure, imep*, are applicable, depending on whether brake horsepower or indicated horsepower is the reference factor.

The horsepower of a reciprocating engine can be calculated in terms of mean effective pressure by the following equation:

$$hp = \frac{p\ LAN}{33{,}000(12)} \left(\frac{n}{x}\right) \tag{7-9}$$

where

p is the mean effective pressure, psi
A is the piston face area, sq in
L is the length of stroke, in
N is the engine speed, rpm
x is the stroke factor: 2 for a four-stroke cycle engine and 1 for a two-stroke cycle engine
n is the number of cylinders in the engine

Torque is directly proportional to mean effective pressure and, as such, for a given engine, varies with volumetric efficiency over the speed range of the engine. Volumetric efficiency is defined as the ratio of the actual weight of air inducted into the engine to the theoretical weight of air necessary to fill the piston displacement volume under atmospheric conditions.

Specific fuel consumption, sfc, is an estimate of operating economy. Most commonly used is *brake specific fuel consumption, bsfc*, which is a measure of pounds of fuel per brake horsepower-hour. Brake thermal efficiency is the inverse of brake specific fuel consumption and is a measure of the engine's effectiveness in converting the heating value of the fuel into work. As shown in Fig. 7-8, the brake thermal efficiency increases with engine speed at full throttle, reaches a maximum, and then decreases with a further increase in engine speed. The variation of thermal efficiency with speed can be explained as follows: The increase in brake specific fuel consumption with a speed increase is the result of a decrease in the time available for heat loss to the cylinder walls (more of the energy of the fuel is converted into work). The decrease of thermal efficiency with speed increase is a result of the decrease in mechanical efficiency of the

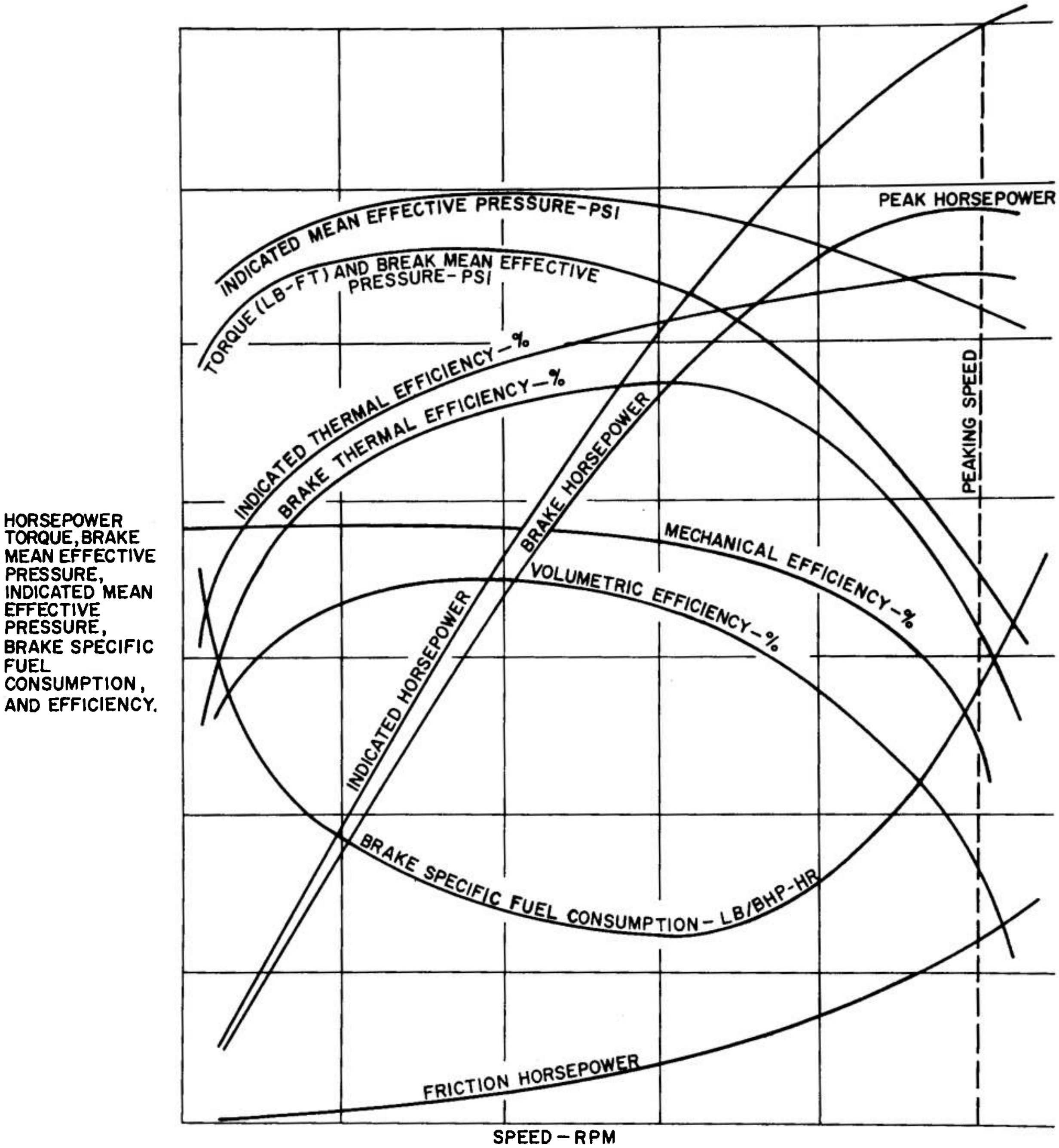


Figure 7-8. Idealized Performance Characteristics of a Typical Reciprocating Automotive Engine at Full-Throttle

engine; i.e., friction horsepower becomes the dominating factor.

For constant-speed engines, brake specific fuel consumption may be plotted against horsepower, Fig. 7-9. The point of maximum economy (minimum brake specific fuel consumption) for a typical constant-speed piston engine occurs at somewhat less than maximum horsepower. For a variable-speed engine, a number of the "fishhook" curves are plotted in Fig. 7-10 for various constant engine speeds. Enveloping curves show maximum-economy brake specific fuel consumption and full-throttle brake specific fuel consumption. Each point on the maximum-economy curve represents

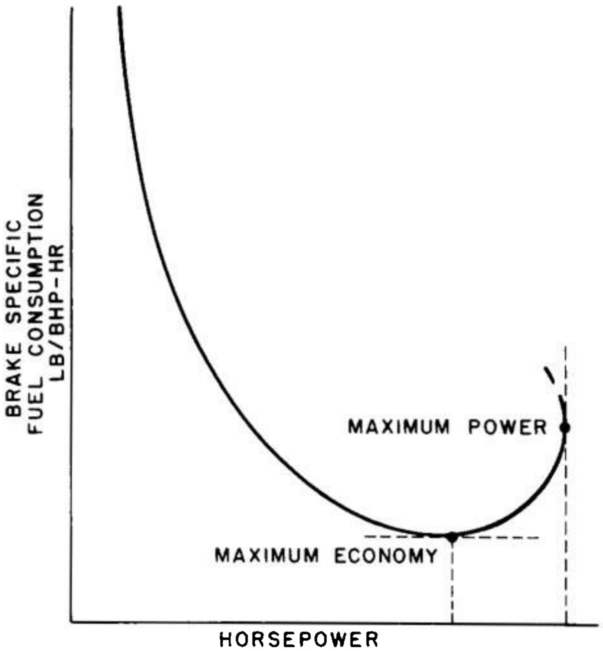


Figure 7-9. *Typical Fuel Consumption Curve for a Constant-Speed Reciprocating Engine*

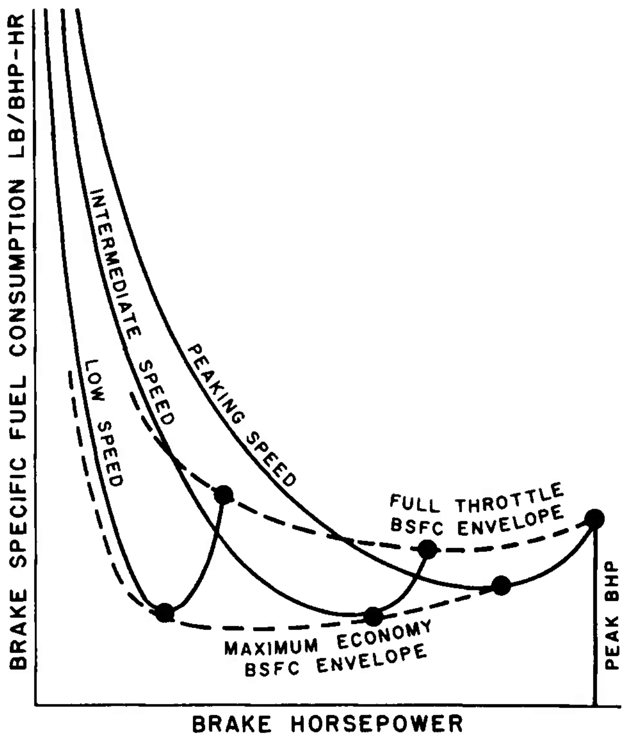


Figure 7-10. *Maximum-Economy Operating Conditions Indicated by Envelope of Constant-Speed Fuel Consumption Curves*

the most efficient engine speed for a given load condition. Figure 7-11 shows a three-dimensional plot of the specific fuel consumption-horsepower-engine speed relationship.

The term *air-fuel ratio* defines the weight ratio of air to fuel inducted into an engine. It is most commonly used with spark-ignition gasoline engines since a definite range of air-fuel ratio is required for the operation of a spark-ignition engine. Although a spark-ignition engine will operate on air-fuel mixtures as lean as 20:1 and as rich as 4:1 (Ref. 2), both power and specific fuel consumption vary with the air-fuel ratio, Fig. 7-12. The point of maximum economy (minimum brake specific fuel consumption) occurs at a ratio of about 16:1 with a small amount of excess air. The theoretical air-fuel ratio for complete combustion in an engine using gasoline as the fuel is 15.27:1. The ratio for maximum power is about 13:1; combustion is incomplete, but rate of burning is most rapid, and, thus, the greatest amount of heat is liberated near top dead center.

Another important design factor for reciprocating piston engines is *mean piston speed,* which can be calculated as follows:

$$S = 2LN \tag{7-10}$$

where

S is the mean piston speed, fpm
L is the stroke, ft
N is the engine speed, rpm

Current engines have piston speeds ranging from below 1,000 fpm to about 3,000 fpm. In the automotive field, 2,500 fpm is often considered the maximum desirable mean piston speed. As piston speeds increase excessively, friction losses become very high and mechanical efficiency decreases. Volumetric efficiency also decreases rapidly as piston speed becomes excessive. At extremely high piston speeds other detrimental effects are high rates of ring and cylinder wear and loss of oil control due to piston ring flutter.

7-2.2 MECHANICAL CYCLES

7-2.2.1 Four-Stroke Cycle

7-2.2.1.1 Spark-Ignition

Most spark-ignition engines operate on a four-stroke mechanical cycle. The sequence of operations is as follows:

a. Induction stroke. Admission of combustible

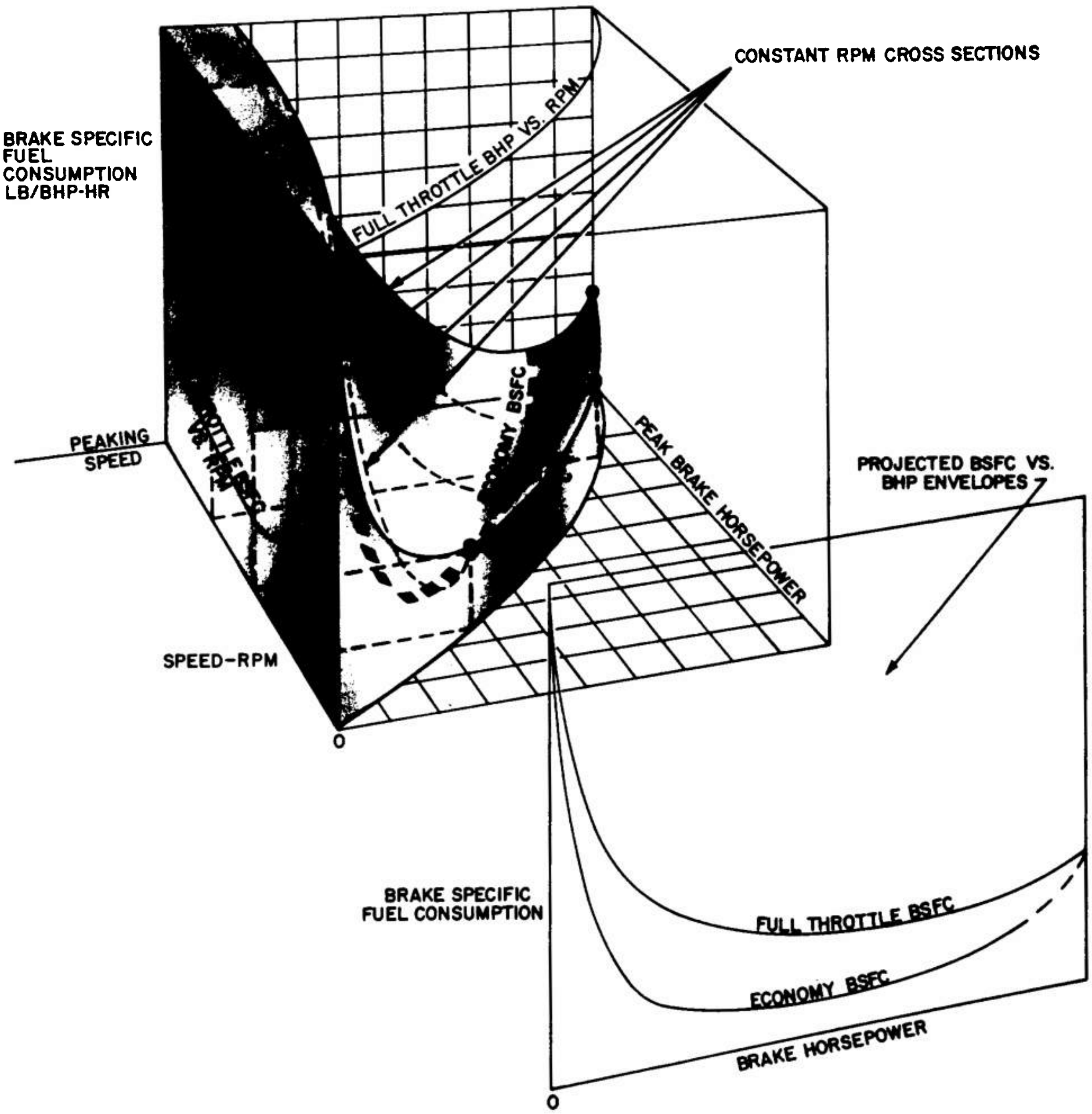


Figure 7-11. ***Three-Dimensional Plot of Specific Fuel Consumption-Brake Horsepower-Engine Speed Relationship (Ref. 2)***

mixture into the cylinder of the engine (intake valve open).

b. *Compression stroke.* Increase of temperature and pressure of mixture (both valves closed).

c. *Power stroke.* The compressed, homogeneous mixture of fuel and air is ignited and burns at approximately constant volume; expansion of high pressure and temperature gases drives the piston downward (both valves closed).

d. *Exhaust stroke.* Products of combustion are forced from the cylinder (exhaust valve open).

The sequence of operations of the actual spark-ignition engine differs from the ideal thermodynamic processes discussed previously. The actual compression stroke is not adiabatic since heat flows into the charge (air-fuel mixture) at the beginning

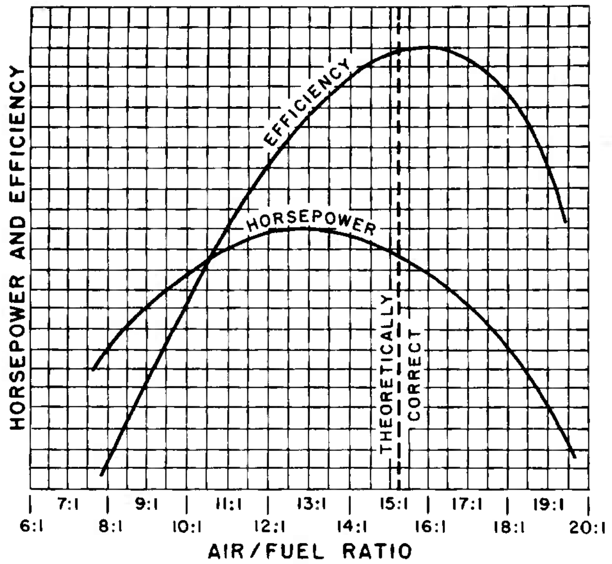


Figure 7-12. *Characteristic Variation of Horsepower and Thermal Efficiency with Air-Fuel Ratio*

of compression and out of the charge at the end of compression, the direction of flow being determined by the relative temperature between charge and cylinder wall. Heat addition does not occur in an actual engine at constant volume since the time required for combustion may permit in excess of 70° of crankshaft rotation (Ref. 11).

Ignition timing is usually adjusted so that combustion is initiated before the end of the compression stroke. This results in approximately constant-volume combustion and maximum pressures at the beginning of the expansion stroke. The intake and exhaust valves of a typical engine do not open and close at top dead center, tdc, and bottom dead center, bdc. The intake valve opens before the piston reaches top dead center on the exhaust stroke and closes after it leaves bottom dead center on the compression stroke. This timing is used to obtain maximum charging at some desirable speed. When the intake valve is opened early and closed late, it is fully opened when the induction stroke starts, and charging is enhanced by the kinetic energy of the incoming mixture.

The exhaust valve is opened before the end of the expansion or power stroke. Early opening allows the high-pressure products of combustion to escape from the cylinder before the piston begins the exhaust stroke. This timing results in a loss of part of the expansion energy of the gases, but this loss is more than compensated by the decrease in the amount of engine work required to force the exhaust gases from the cylinder (Ref. 12). The gain occurs at high engine speeds only; at lower speeds (part-throttle operation), the early opening of the exhaust valve will result in power losses.

Both valves may be open at the end of the exhaust stroke (or beginning of the induction stroke). This overlap will assist scavenging of the exhaust gases at high engine speed and full-throttle operation, but will cause exhaust gas dilution of the incoming charge under part-throttle conditions.

7-2.2.1.2 Compression-Ignition

The four-stroke compression-ignition or Diesel engine cycle processes are similar to the four-stroke spark-ignition processes except for the method of adding fuel and igniting the air-fuel mixture. The sequence of operations in current Diesel engines is as follows:

a. *Induction stroke.* Admission of air alone into the cylinder (intake valve open).

b. *Compression stroke.* Increase of temperature and pressure of the air charge; the temperature of the compressed charge exceeds the ignition point of the fuel (both valves closed).

c. *Power stroke.* Injection of fuel starts at beginning of the expansion stroke and continues at a rate such that constant-pressure combustion occurs during the first part of the power stroke; at a given time, the injection is stopped and the high-pressure and -temperature gases continue to force the piston downward (both valves closed).

d. *Exhaust stroke.* Products of combustion are forced from the cylinder (exhaust valve open).

The modern Diesel engine operates on a mixed cycle. The heat addition takes place partly in a constant-volume process and partly in a constant-pressure process. Fuel injection starts during the last part of the compression stroke, so that the first phase of combustion occurs at approximately constant volume. Injection continues during the first part of the expansion stroke, resulting in approximately constant-pressure combustion.

The actual valve timing (opening and closing of intake and exhaust valves with respect to piston position) of Diesel engines varies in a manner similar to that of spark-ignition engines, and the reasons for the timing are the same.

Speed and load control in the compression-ignition engine is accomplished by means of the fuel injection system. The fuel injection system for Diesel engines controls the beginning, rate, and duration of injection of fuel into the combustion chamber. Unlike the spark-ignition gasoline engine, the compression-ignition engine does not require a definite air-fuel ratio.

7-2.2.2 Two-Stroke Cycle

The two-stroke cycle may be applied to either spark-ignition engines (gasoline) or compression-ignition engines. The two-stroke cycle engine develops a power stroke per cylinder for each revolution of the crankshaft. The simplest two-stroke engines have the intake and exhaust ports in the cylinder wall and use crankcase compression to charge the cylinders. The sequence of operations is as follows:

a. *Compression stroke.* The temperature and pressure of the charge within the cylinder is raised; the charge can be an air-fuel mixture (spark-ignition engine) or air alone (compression-ignition engine); during this stroke, air is inducted into the crankcase (both ports closed).

b. *Power stroke.* The expansion stroke is initiated by spark-ignition of the air-fuel mixture, or by injecting fuel into the high-temperature air (both ports closed).

c. *Exhaust process.* Near the end of the power stroke, the piston uncovers a port or ports in the cylinder wall, and most of the products of combustion escape from the cylinder (exhaust port open).

d. *Charging process.* The intake port is uncovered immediately after the exhaust port is uncovered, and either an air-fuel mixture or air alone is forced into the cylinder (intake port open).

The process of emptying and refilling the cylinder after each power stroke is called *scavenging.* The type of scavenging system used for a given engine is important since volumetric efficiency varies appreciably with the various systems used.

If the intake and exhaust ports are located on opposite sides of the cylinder, the system is known as *cross scavenging.* Engines using cross scavenging have deflectors located on the pistons to prevent the incoming charge from passing directly across to the exhaust ports. Another type of port arrangement results in *loop scavenging.* For loop scavenging, the intake and exhaust ports are near each other, and the inlet air passes through a complete loop before reaching the exhaust ports. A third type of scavenging is called *through* or *uniflow scavenging.* For uniflow scavenging, exhaust valves are located in the head and inlet ports are located in the cylinder. The flow of gas is undirectional from intake port to exhaust port.

In the description of the sequence of operations, crankcase compression was assumed to effect cylinder scavenging. Crankcase compression is not used on modern high-specific-output multicylinder two-stroke engines. Rotary blowers (usually of the Roots type) are used for scavenging and supercharging. Supercharging is not possible with symmetrical cylinder port timing where each piston-controlled port closes the same number of crankshaft degrees after top dead center as it opened before top dead center. This causes the exhaust port to close after the inlet port closes making supercharging impossible. Engines using unsymmetrical timing, e.g., the uniflow engines, have exhaust valve timing that allows supercharging. Supercharging is especially suited to two-stroke cycle Diesel engines since air alone is used to scavenge the cylinder.

7-2.2.3 Comparison of Two-Stroke and Four-Stroke Engines (Ref. 13)

The basic differences between four-stroke and two-stroke cycle engines are (a) the method of discharging the products of combustion from the cylinder, and (b) the number of power strokes per revolution of the crank. In the two-stroke cycle engine, compressed air from a source external to the actual cylinder is used to scavenge the cylinder. The same process is effected by the piston in the four-stroke unit. The fact that the scavenging is accomplished while the piston is at or near the bottom of its stroke in a two-stroke unit allows each upward stroke to be a compression stroke and hence, a power stroke per cylinder for each revolution of the crank.

With crankcase compression scavenging, two-stroke engines have low volumetric efficiencies ranging from 30% to 50%, and piston speeds are usually limited to less than 1,000 fpm. A scavenging blower with a displacement 20% to 80% greater than the piston displacement improves the volumetric efficiency appreciably; however, it requires as much as 30% of the total horsepower developed by the engine. In addition, the use of a blower increases the complexity of the engine. The two-stroke engine develops from 50% to 80% greater specific horsepower (hp/cu in of piston displacement) than a comparable four-stroke engine.

Compared with a four-stroke engine of the same size and number of cylinders, the crankcase-compression-type, two-stroke engine has more uniform torque at any speed; but the developed torque tends to fall rapidly with a decrease in speed. Light-load operation of the crankcase-compression-type engine tends to be poor, and the brake specific fuel consumption is high. A multicylinder crankcase-compression-type, two-stroke engine requires a separately sealed crank compartment for each cylinder. On the other hand, the lack of poppet or rotary valve mechanisms makes this engine mechanically simple and of low initial cost.

The specific fuel consumption of carbureted two-stroke engines is relatively high, owing to the loss of fuel during the scavenging process. The pumping losses of two-stroke engines are generally higher than those of comparable four-stroke engines, hence, the mechanical efficiency of the latter is normally higher. Proper lubrication is difficult to achieve in the two-stroke crankcase-compression engine. Furthermore, piston temperatures normally run higher in two-stroke engines, and spark plug life is naturally shorter in terms of hours of operation.

Many of the stated difficulties are reduced or eliminated in the two-stroke compression-ignition engine. Present two-stroke compression-ignition engines use blowers to scavenge or, with proper valving, to supercharge the cylinders. Light-load operation is satisfactory since there is no loss of fuel during scavenging and scavenging pressure remains fairly constant. The thermal problems are also lessened in two-stroke compression-ignition engines, owing to lower cyclic temperatures.

7-2.2.4 Comparison of Spark-Ignition and Compression-Ignition Engines

In the spark-ignition engine, the air and fuel (gasoline) are inducted into the engine simultaneously in a ratio (air:fuel) suitable for combustion. As the air-fuel mixture is compressed, an appreciable amount of energy is released as a result of chemical reactions between the air and fuel (Ref. 14). The remainder of the energy of the air-fuel mixture is released after it is ignited by an electrical spark in an approximately constant-volume process. The fact that a combustible mixture is compressed in the spark-ignition engine permits the occurrence of either of two detrimental combustion phenomena. The first is usually termed *pre-ignition*. Pre-ignition occurs when the air-fuel charge is prematurely ignited (during the compression process) as a result of a localized high-temperature area within the cylinder. Since the direct heat losses are increased during pre-ignition, continued operation under these conditions can result in thermal stress failure of the engine.

The second phenomena is known as *detonation*. Detonation is a complex phenomena that is a function of many variables such as pressure, time, fuel-air ratio, fuel composition, residual products of combustion, and combustion chamber configuration. Detonation occurs when the normal progressive combustion process with a normal flame front is disturbed by a spurious secondary combustion process originating in the unburned portion of the charge. The secondary combustion process occurs almost instantaneously. The resulting pressure waves and heat transfer increases reduce efficiency and can lead to engine component failures. The tendency for detonation, in a given engine and fuel, increases with an increase in compression ratio; hence, detonation imposes a limit on compression ratio in practical engines.

In the compression-ignition engine, air alone is inducted into the engine and is compressed along with the residual gases. The magnitude of the compression ratio is such that the air charge reaches a temperature greater than the ignition temperature of the fuel. The fuel (usually some grade of Diesel fuel) is injected at a controlled rate after compression is completed. The rate of injection is timed so that a constant-pressure combustion process is approached (see par. 7-1.3). Pre-ignition cannot occur, and detonation does not occur. However, practical high-speed combustion-ignition engines encounter definite combustion problems. The first problem is based on the phenomena known as *ignition delay,* a result of injecting a solid fuel into a high-temperature oxidizing atmosphere. Ignition delay can be divided into a *physical delay* and a *chemical delay*. The physical delay is the time period required after injection begins, for the fuel to be atomized, vaporized, mixed with air, and raised to the chemical reaction temperature. The chemical delay is the time period required for the chemical reaction to reach a sufficiently high rate for combustion to occur. If a large portion or all of the fuel is injected during the ignition delay period, uncontrolled combustion occurs with an extremely rapid pressure rise that may cause knocking and abnormally high forces on the engine components. The second combustion problem of compression-ignition engines concerns the mixing of the fuel and air. Inadequate mixing may result in incomplete combustion and lowered thermal efficiency.

The spark-ignition and the compression-ignition engines can be compared by considering some of their performance factors:

a. Initial cost, terms of dollars per horsepower of spark-ignition engines is normally less than that of compression-ignition engines.

b. Spark-ignition engines normally have lower specific weights per horsepower than compression-ignition engines.

c. The cranking effort required for spark-ignition engines is usually less than that for similar compression-ignition engines.

d. The mechanical efficiency of spark-ignition engines is normally greater than that of compression-ignition engines due to the influence of higher friction losses in the latter engines (longer pistons, larger bearings, etc.). Although the part-throttle pumping losses are higher in the spark-ignition engine, these losses constitute only about 15% of the total mechanical losses, while mechanical friction contributes about 85% to the total losses.

e. The maximum mean effective pressure of spark-ignition engines is normally greater than the maximum mean effective pressure of comparable compression-ignition engines in similar environments. The ratio of maximum mean effective pressure of compression-ignition and spark-ignition engines ranges from 70% to 90%, depending upon the quality of combustion in the compression-ignition engine and the compression ratio used in the spark-ignition engine (Ref. 15). Spark-ignition engines normally develop more specific horsepower (hp/cu in. of piston displacement) than compression-ignition engines.

f. The overall brake specific fuel consumption of compression-ignition engines is normally lower than that of comparable spark-ignition engines.

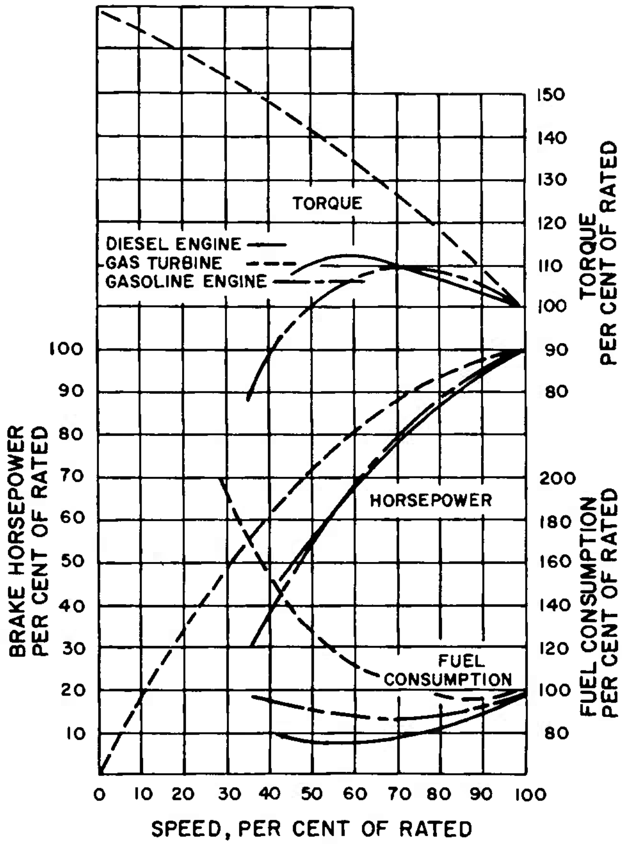


Figure 7-13. ***Performance Characteristics for Basic Power Plants***

The advantage of the compression-ignition engine increases as the load is reduced from a maximum. Under these conditions, the brake specific fuel consumption of the compression-ignition engine remains relatively constant, while for spark-ignition engines the brake specific fuel consumption increases appreciably as load decreases (part-throttle operation). Figure 7-13 shows these trends.

g. The lugging ability of compression-ignition engines is normally superior to that of spark-ignition, as shown by the torque curves of Fig. 7-13. This characteristic makes the compression-ignition engine better suited to military vehicles.

h. Compression-ignition engines are well suited to two-stroke operation, while carbureted spark-ignition engines suffer a loss of fuel through the exhaust ports during scavenging. These losses could be eliminated in a two-stroke spark-ignition engine by the use of fuel in-

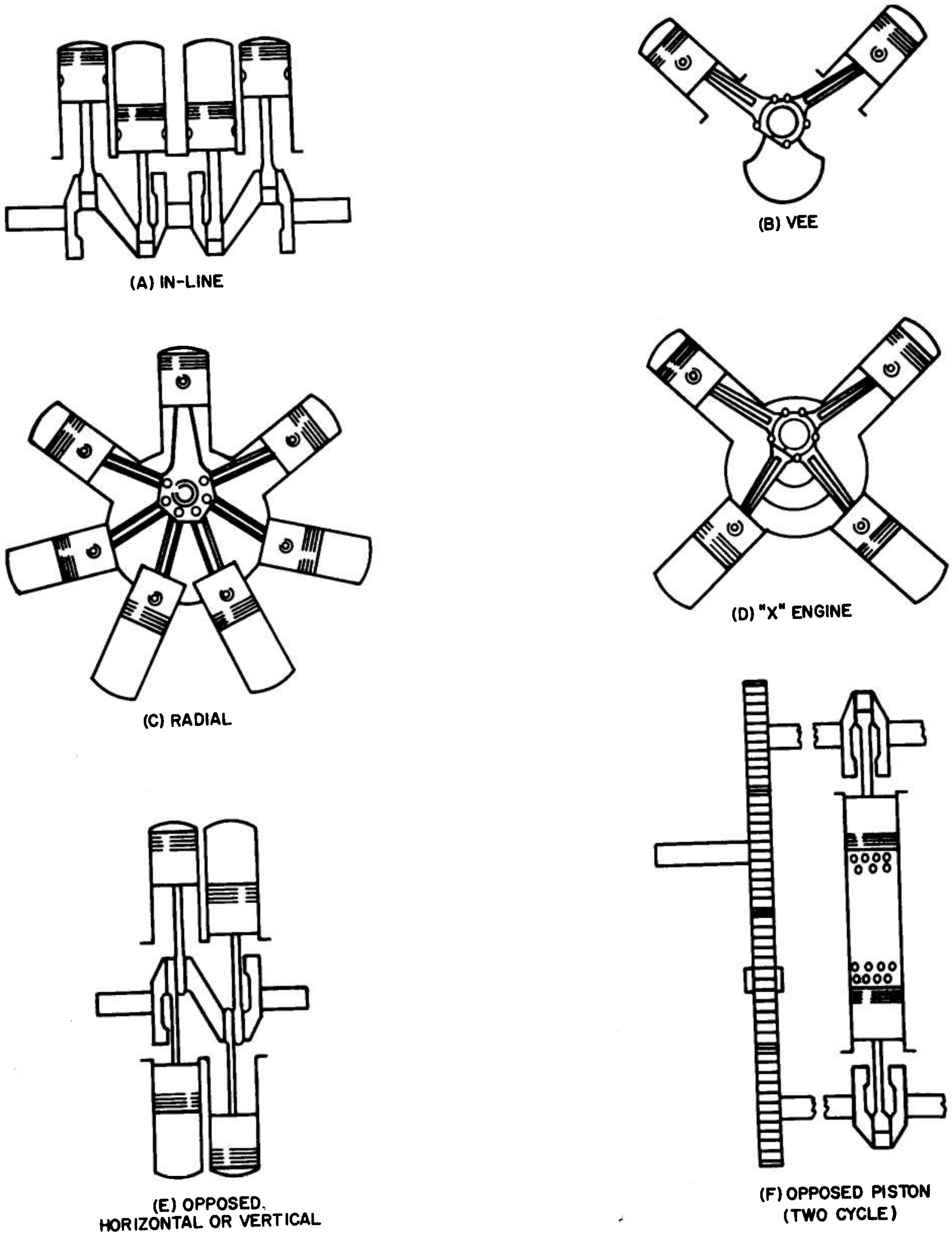


Figure 7-14. *Typical Cylinder Arrangements for Reciprocating Engines*

jection. At present, gasoline-injection systems have not reached the state of development of oil-injection systems.

i. The exhaust gases of the compression-ignition engine contain no carbon monoxide, except near full-load or overload conditions. The exhaust gases of spark-ignition engines normally contain carbon monoxide, and when combustion is poor, or the mixture is rich, the proportion of carbon monoxide increases.

j. The volatility of gasoline relative to Diesel fuel makes the former more hazardous to use and more difficult to store. This is an important consideration in military operations.

7-2.3 ENGINE CONFIGURATIONS

The arrangement of the cylinders in a multiple-cylinder reciprocating engine can take several forms. Factors such as compactness, rigidity, proportions, accessibility, ease of manufacture, and inherent dynamic balance should be considered when military engine designs are evaluated. Some of the most common types (configurations) of engines and their advantages and disadvantages are discussed below and illustrated in Fig. 7-14.

7-2.3.1 Inline Engines

Inline engines have their cylinders arranged in a single line on one side of the crankshaft and perpendicular to the crankshaft. Military automotive inline engines usually have two, four, six, or eight cylinders. Normally, the inline engine is installed in a vehicle with the cylinders vertical (with the crankshaft at the bottom); however, inline engines may be designed to operate inverted, horizontal, or at some position between vertical and horizontal (inclined). The major advantages of incline engines, relative to other configurations, are ease of manufacture and maintenance; the lack of angles simplifies these operations. The major disadvantages are the overall length of inline engines which may become excessive for a given displacement when compared to other types of engines, and the relatively long crankshaft of the inline engine which makes the torsional vibration problem more severe. Inline air-cooled engines usually require extensive shrouding to effect efficient and equal cooling of the cylinders. Other disadvantages are poor charge distribution, multiple carburetion necessary, and difficulty of cooling, particularly when liquid cooling is used.

7-2.3.2 V-Type Engines

The V-type engine configuration consists of two inline banks of cylinders arranged in a "V" about a common crankshaft (Fig. 7-14). V-type engines most commonly have two, four, six, eight, twelve, or sixteen cylinders. Engines with V-type configurations have comparatively short crankshafts, which are inherently more rigid than inline engine crankshafts for the same number and size of cylinders. The V-type engine usually is more efficient in the utilization of space than the inline engine. Cooling is usually easily accomplished in the V-type engine. Manufacturing and maintenance, however, are made more difficult by the angular construction of the V-type engine.

7-2.3.3 Horizontal-Opposed Engines

Horizontal-opposed engines have two inline banks located 180° from each other. Like the V-type engine, the horizontal-opposed engine has a relatively short, rigid crankcase and crankshaft. This type of engine has a relatively low vertical dimension, but is usually wider than other types of the same piston displacement. Horizontal-opposed engines are difficult to service while they are mounted in the vehicle, but once they are removed, servicing becomes very simple and two or more mechanics may work on the engine at the same time.

7-2.3.4 Radial Engines

Radial engines have their cylinders in one or two planes that are perpendicular to the axis of the crankshaft. Radial engines are built in single rows with three, five, seven, and nine cylinders, and in two rows with double the number of cylinders (usually 14 or 18). An odd number of cylinders per row is used to obtain uniform spacing of power strokes in the four-stroke engine. Radial engines have the shortest crankcases and crankshafts for a given piston displacement of all reciprocating engines. However, the diameter of these engines is relatively large, and the central position of the output shaft ill suits them to vehicle use. Most radial engines have been designed for aircraft and are normally air-cooled.

7-2.3.5 Barrel or Round Engines

Barrell or round engines have their cylinders parallel to the axis of the power shaft. In a barrel engine, the conventional crankshaft is replaced by

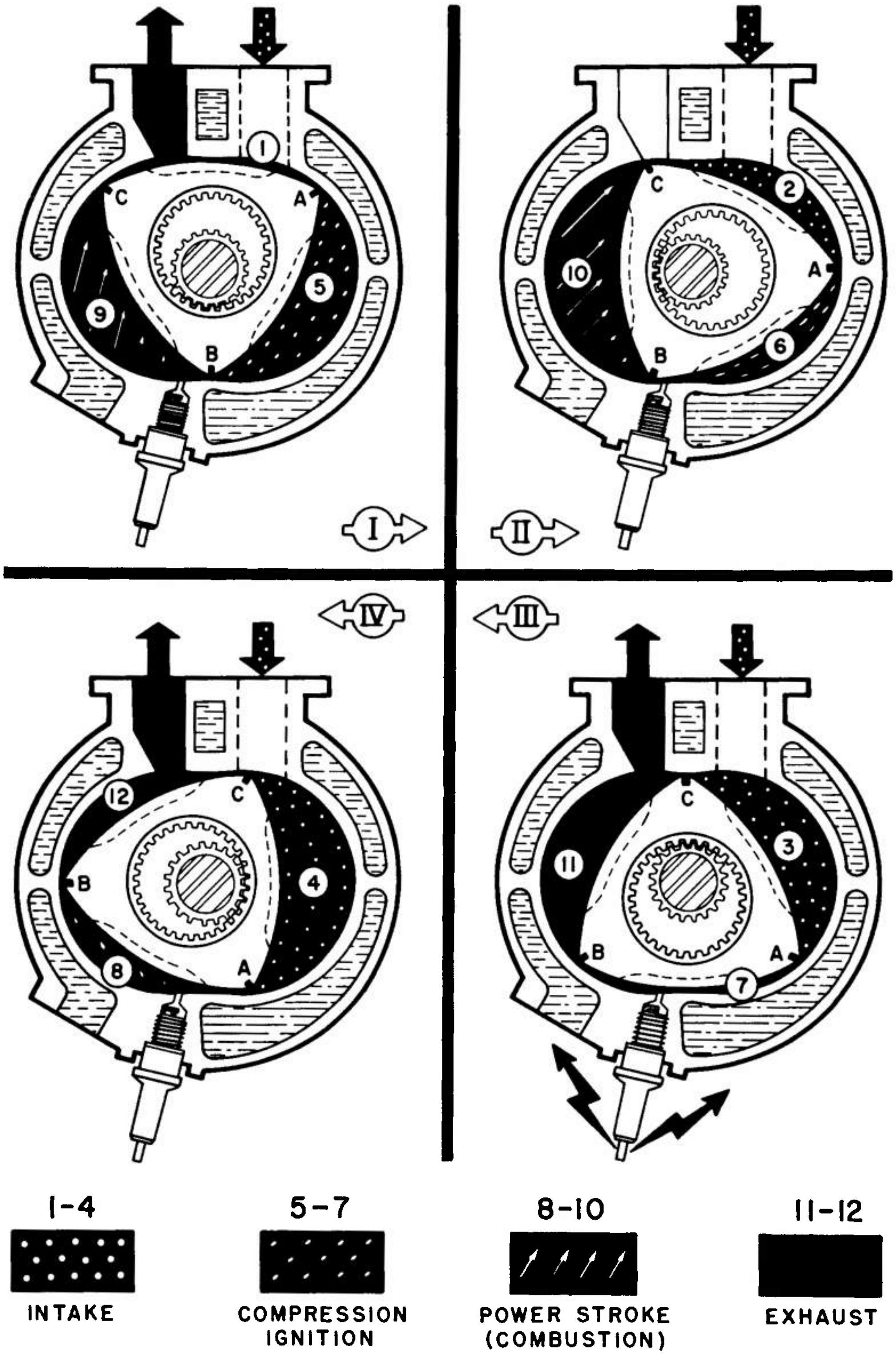


Figure 7-15. *Sequence of Operations in the Rotary-Piston Internal Combustion Engine (From "That Curtis-Wright Announcement—Germans Strip Mystery from Rotary Engine,"* Product Engineering, *Dec. 28, 1959)*

a swash plate or some type of cam and follower arrangement. The unbalanced centrifugal couple of a simple swash plate can be balanced by a piston inertial couple or a "folded" balanced swash plate can be used. Round engines have relatively small frontal area, but the overall length is large for the piston displacement.

7-2.3.6 Opposed-Piston Engines

The opposed-piston engine has two pistons in each cylinder, as shown in Fig. 7-14. This design is well suited to the two-stroke cycle. The upper piston controls the intake ports, while the lower piston controls the exhaust ports. Uniflow scavenging is obtained, and asymmetric valve port timing permits supercharging of the engine. High specific horsepower (hp/cu in of piston displacement) is possible with the opposed-piston engine. However, the design requires two crankshafts or long linkage, and the overall dimension in the direction of the cylinder axis is relatively large. Two important applications for this type of engine are as an air compressor and as a gasifier for a turbine.

7-2.3.7 Rotary-Piston Engines (Refs. 32, 33, 34)

Rotary-piston engines attempt to eliminate reciprocating masses and thus eliminate a source of high inertial forces present in all conventional reciprocating internal combustion engines. Figure 7-15 shows the basic components of an experimental engine of this type by means of a simplified transverse cross section of the engine with the crankshaft removed. The solid black circle in the center of each diagram represents the bore of the crankshaft support bearing (main bearing) at the far end of the engine. The rotor is triangular in shape with outward curving sides and a large, circular bore through its center. Internal gear teeth are cut concentric to the bore at one end of the rotor. These mesh with a stationary pinion mounted on the cover plate that carries the rear main bearing. The gear ratio between rotor and pinion is 3:2 in the experimental engine. The crankshaft has a main journal surface at each end and a large eccentric journal surface that fits the bore of the rotor. Counterweights (not shown in Fig. 7-15) are splined to the crankshaft to achieve perfect balance.

As the rotor is caused to rotate by the pressure of the expanding gases acting against the sides, each of the three sides goes through a complete cycle of intake, compression, power, and exhaust phases, resulting in three power impulses per revolution of the rotor. The gear ratio between rotor and fixed pinion causes the crankshaft to rotate at three times the speed of the rotor, resulting in one power impulse per each crankshaft rotation.

The torque developed by this type of engine decreases quite rapidly from the maximum value as engine speed decreases and reaches a value of approximately 75 ft-lb at 1500 rpm. However, it is believed that improved performance in this area (higher torque at low speeds) can be obtained, if desired, by proper porting configurations.

The rotary-piston engine was studied and evaluated by a group of experts (Ref. 32), and the following conclusions were reached:

a. Based on the models tested, the rotary-piston engine does not have any thermodynamic advantages over conventional Otto engines.

b. Engine speed is limited by the thermal loading factor.

c. Sealing-vane ware and casing wear is high at the mean peripheral speeds of the models: 8,000 fpm.

d. Engine performance is severely curtailed by low-octane fuels.

e. The present engine has a theoretical compression-ratio limit of 15:1; thus, it is not suitable for small Diesel engine designs.

f. Compared with conventional reciprocating engines, the rotary-piston engine is lighter, more compact, simpler, quieter, and better balanced.

g. There are positive indications that the rotary-piston engine under consideration would have improved life and performance in sizes larger than the test models.

The rotary-piston engine may have application in military vehicles since long life is not a major requirement for these vehicles.

7-2.4 TYPES OF ENGINES (Ref. 13)

7-2.4.1 Aircraft Engines

Reciprocating aircraft engines are usually four-stroke spark-ignition units. They may be liquid-cooled, but most of them are air-cooled since adequate cooling is easily accomplished and a minimum total weight is sought. Air-cooled aircraft engines require a cooling blower when they are installed in ground vehicles.

Aircraft engines have the lowest specific weight (lb/max bhp) of any internal combustion engine, and they have the lowest brake specific fuel consumption of any carbureted spark-ignition engine. Usually, aircraft engines employ a valve in-head design in order to achieve high mean effective pressures. Valve timing is such that high mean effective pressure is obtained at rated speed.

Aircraft engines must be compact, lightweight, and very reliable. The machining tolerances, the balance requirements, the materials and the assembly techniques of aircraft engines make them more expensive than automobile engines of similar output. Aircraft engines are frequently equipped with superchargers to compensate for the power loss due to an increase in altitude.

American aircraft engines are of various configurations. The smallest (40 bhp) are 2-cylinder air-cooled horizontally opposed engines. Other configurations are 4-, 6-, or 12-cylinder opposed; 6-cylinder vertical, inverted, or horizontal; 12-cylinder V-type; 5-, 7-, or 9-cylinder single-row radial; 14-, and 18-cylinder double-row radial; 28- and 36-cylinder four-row radial air-cooled; and 24-cylinder double V-type water-cooled.

7-2.4.2 Automobile Engines

Most of the currently produced automobile power plants are spark-ignition four-stroke gasoline engines. Automobile engines are built in various configurations: from 1-cylinder to 12-cylinder V-types. They may be air-cooled or liquid cooled; however, liquid-cooled engines predominate. Usually, the automobile engine is naturally aspirated as opposed to the supercharged aircraft engine. Like aircraft engines, valve in-head construction predominates in automobile engines. The great majority of American automobile engines are inline 6-cylinder or V-type 8-cylinder units.

Almost all of the current American automobile engines utilize a cast-iron engine block consisting of the cylinders, the upper part of the crankcase, and the water jacket. The detachable cylinder head or heads are also cast iron with integral water passages. Recent developments in this area include aluminum alloy blocks for water-cooled engines and aluminum alloy crankcases and cylinder heads for air-cooled engines.

7-2.4.3 Truck Engines

Truck engines may be either Otto or Diesel cycle units. The Otto cycle truck engines are similar to automobile engines, but, in general, the truck engines are of larger displacement, heavier, lower speed, and have lower stresses for longer life than the typical automobile engines. The Diesel truck engines are either two-stroke or four-stroke units. Displacement of the larger truck engines are in the 300- to 1,000-cu in range. Generally, they are water-cooled and normally aspirated.

Truck engines have valve timing designed to produce maximum torque at relatively low engine speeds. Also, the horsepower per cu in is generally less than that of automobile engines. Truck engines are generally constructed with 4 or 6 cylinders inline or 6, 8, 12 cylinders in V-type configuration.

7-2.4.4 Tractor Engines (Refs. 16, 17)

Tractor engines may be either Otto or Diesel cycle units. The former may use either liquid or gaseous fuel. Tractor engines are normally valve in-head water-cooled slow-speed engines. Piston speeds range from 1,000 to 1,300 fpm at maximum speed. Tractor engines are built as 2-cylinder horizontal inline, 3-cylinder vertical inline, 4-cylinder vertical inline, 6-cylinder vertical inline, and 8-cylinder V-type engines. Since tractor engine cylinders are subjected to relatively rapid wear, the cylinders are often replaceable.

An important characteristic of tractor engines is their good lugging ability. Lugging ability refers to capability of an engine to sustain and pull through temporary overloads. When an engine is momentarily overloaded, its speed decreases. If the output torque of the engine increases as the speed decreases, the engine has a favorable degree of lugging ability. The percentage of torque increase for a given engine is a measure of its lugging ability. In general, Diesel engines have better lugging ability than Otto engines.

Another factor of importance in the selection of tractor engines is the part-load specific fuel consumption. The brake specific fuel consumption of spark-ignition engines, in general, is greater than that of compression-ignition engines at part-load operation, owing to the high pumping losses of the former when throttled.

7-2.4.5 Marine Engines

Marine engines are built using either spark-ignition or compression-ignition. These engines may be either two-stroke or four-stroke designs. The smallest marine engines are two-stroke spark-ignition outboard units ranging in power from less

TABLE 7-2 NORMAL RANGE OF INTERNAL COMBUSTION ENGINE CHARACTERISTICS

CHARACTERISTICS	AUTOMOBILE ENGINES	AIRCRAFT ENGINES	TRUCK ENGINES			TRACTOR ENGINES			MARINE ENGINES, INBOARD‡			MARINE ENGINES, OUTBOARD
	Spark-Ignition 4-cycle (SI-4)	Spark-Ignition 4-cycle (SI-4)	Spark-Ignition 4-cycle (SI-4)	Compression-Ignition 4-cycle (CI-4)	Compression-Ignition 2-cycle (CI-2)	Spark-Ignition 4-cycle (SI-4)	Compression-Ignition 4-cycle (CI-4)	Compression-Ignition 2-cycle (CI-2)	Spark-Ignition 4-cycle (SI-4)	Compression-Ignition 4-cycle (CI-4)	Compression-Ignition 2-cycle (CI-2)	Spark-Ignition 4-cycle (SI-4)
Brake Horsepower (bhp) at Engine Speed (rpm)	80 at 4400 375 at 5000	65* at 3400 2300* at 2900	47 at 3200 368 at 2200	45 at 2400 420 at 2300	97 at 2800 504 at 2300	17 at 2500 600 at 1200	9.5 at 3000 730 at 1300	47 at 2000 675 at 2300	10 at 1100 350 at 2600	17 at 1500† 750 at 1200†	58 at 2200† 416 at 1800†	1.7 at 4000 80 at 6000
Piston Displacement, (cu in)	140-430	171-3350	124-1091	144-1197	159-851	59.5-3619	35-2493	142-1135	69-935	129-3619	159-1320	2.87-89.5
Brake Horsepower (bhp) per Piston Displacement (cu in)	0.46-0.91	0.38-1.02	0.25-0.64	0.22-0.44	0.51-0.71	0.127-0.66	0.16-0.52	0.44-0.73	0.10-0.47	0.09-0.24†	0.33-0.65†	0.5-1.06
Stroke to Bore Ratio, S/B	0.72-1.36	0.78-1.15	0.75-1.50	0.75-1.50	1.11-1.22	0.77-1.47	1.00-1.40	1.12-1.17	0.95-1.40	1.00-1.4	1.1-1.2	0.7-1.00
Average Piston Speed, S_p (fpm) at Max Brake Horsepower (bhp)	1750-3120	1390-3210	1800-2600	1000-1900	1600-2100	1000-2500	1000-2200	1670-2250	870-2760	1200-2100†	1000-1500†	900-2200
Compression Ratio, CR	8.00-10.5	6.00-8.7	5.4-8.5	12-20	17.00-18.00	4.5-8.5	12.0-22	17-22	4.0-7.5	11.4-17.0	16-18	4.5-9
Brake Mean Effective Pressure, bmep (psi)	96-144	131-259	90-124	80-130	70-80†	80-120	75-100	75-100	72-125	70-135	65-100†	55-75
Brake Specific Fuel Consumption, bsfc (lb/hp-hr)	0.5-0.6	0.4-0.6	0.45-0.62	0.47-0.57	...	0.46-0.89	0.44-0.7	...	0.5-0.7	0.32-0.45	...	...
Brake Thermal Efficiency, η_t (%)	17-23	22-33	20-30	19-28	...	15-30	20-30	...	18-24	31-35	...	...
Engine Weight (lb) per Max. Brake Horsepower (bhp)	3.5-7	1.09-2.62	7-14	10-18	10-17†	10-30	30-60	30-60	6-60	12-80	10-15	3-11

* Takeoff performance.
† At continuous horsepower ratings.
‡ Does not include the largest stationary-type marine engines.

Sources: L. S. Marks, Ed., *Mechanical Engineers' Handbook*, 6th Ed., McGraw-Hill Book Company, New York, New York, 1958.
Automotive Industries, Statistical Issue, Vol. 122, No. 6, March 15, 1960. (Ref. 35).

than 1 to 80 bhp. Two-stroke outboard marine engines normally use crankcase compression with either three ports or two ports and a rotary or reed-type crankcase inlet valve. Outboard engines are built as single-cylinder, inline, and V-type units.

Inboard marine engines exist in a wide range of power. Spark-ignition inboard engines range from less than 1 to more than 900 bhp; most of these are of the four-stroke cycle type. Compression-ignition inboard engines range from less than 50 bhp to more than 5,000 bhp. Both two- and four-stroke Diesel engines are used. Many of smaller inboard engines are modified automotive types (both spark-ignition and compression-ignition). Typical inboard gasoline engines (all four-stroke cycle) in this category are: inline, 4 or 6 cylinders; V-type, 8 or 12 cylinders.

Most automotive-type marine compression-ignition engines are single-acting four-stroke water-cooled engines. Both air-cooled and two-stroke cycle units also exist. The larger compression-ignition marine engines are stationary types. The type of engine varies with the type of drive used. For direct-drive, two-stroke cycle Diesel engines, either supercharged or normally aspirated, either single- or double-acting, are used in powers up to 14,000 bhp per engine. Two-stroke opposed-piston Diesel engines range to 8,000 bhp per engine. Four-stroke Diesel engines, either supercharged or normally aspirated, range to 6,000 bhp per unit. Diesel engines for either electric drive or geared drive are relatively small high-speed units.

7-2.4.6 Motorcycle Engines

Motorcycle engines generally are air-cooled four-stroke spark-ignition units of one or two cylinders. They may use either side valves or overhead valves. The two-cylinder engines are built-in inline, V-type, or horizontally opposed configurations. The V-type engines normally use an included angle of 42° or 45° between cylinders.

7-2.4.7 Comparison of Engine Types

Engines can be compared by considering a number of design and performance factors. The relative importance of these factors depends upon the type of engine and its proposed usage. Factors of greatest importance in the selection and design of military vehicle engines are:

a. Reliability
b. Economy (low specific fuel consumption)
c. Lugging ability
d. Compactness (small envelope)
e. Multifuel capability.

The selection of reciprocating internal combustion engines for military vehicles is ordinarily based on: (a) the type of vehicle, (b) the operational environment, and (c) the performance or operational requirements. In the present discussion, the complex relationship between the vehicle power system and the actual vehicle performance in a given environment is not considered.

Table 7-2 expresses the normal range of design and performance characteristics of current engines. A more complete listing of engine specifications appears annually in *Automotive Industries,* Statistical Issue. Figures 7-16 through 7-22 show typical reciprocating engines for military vehicles. Table 7-3 shows representative standard military vehicle engines.

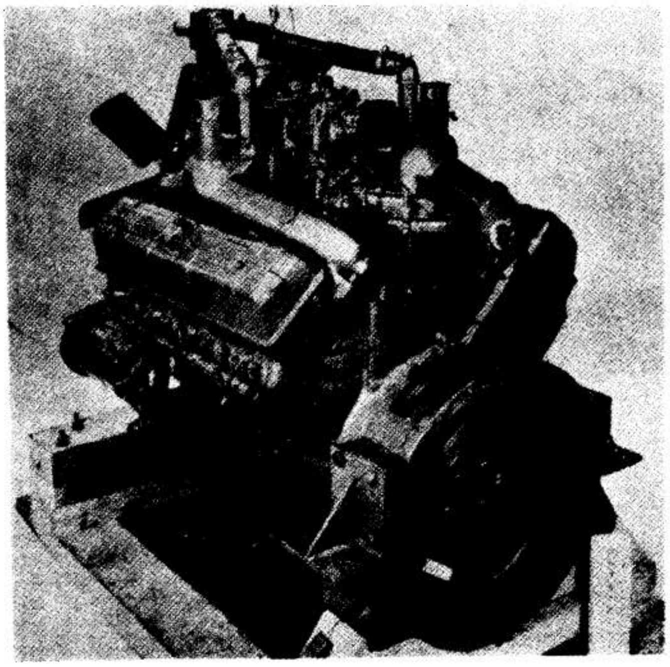
7-3 RECIPROCATING, EXTERNAL COMBUSTION ENGINES

7-3.1 STEAM ENGINES (Ref. 19)

Reciprocating steam engines are of minor importance at present as a source of power. Large reciprocating steam engines have been replaced by steam turbines. The relative efficiencies of reciprocating steam engines and steam turbines depend upon size and exhaust pressure range. The steam engine may be superior to the steam turbine in thermal efficiency in the smaller sizes (up to several hundred horsepower) and also in higher exhaust pressure ranges. In larger sizes and when low exhaust pressures are used, the turbine usually operates at higher thermal efficiency.

In the larger sizes, turbines have displaced reciprocating engines because the turbine has the advantages of less weight and bulk, less maintenance, simpler foundation requirements, and better speeds for driving electric generators and centrifugal pumps. The smallest steam turbines have been displaced by internal combustion engines and electric motors.

The major disadvantages of steam power plants for vehicles are related to the necessary boiler and burner assembly. The advantages of reciprocating steam engine propulsion for vehicles are high starting torque, silent operation, wide speed range with



ENGINE CHARACTERISTIC DATA

MODEL	TH-844	
MANUFACTURER	LE ROI	
NUMBER OF CYLINDERS	8	
CYLINDER ARRANGEMENT	90 Deg. V	
COOLING MEDIUM	LIQUID	
CYCLE	4	
FUEL	GASOLINE	
BORE	5.25 In.	
STROKE	4.875 In.	
DISPLACEMENT	844 Cu. In.	
COMPRESSION RATIO	6.7 To 1	
MAXIMUM GOVERNED SPEED	2600 RPM (FULL LOAD)	
RATED GROSS HORSEPOWER	290 at 2600 RPM	
NET HORSEPOWER	265 at 2600 RPM (INSTALLED)	
MAXIMUM GROSS TORQUE	657 Ft. Lbs. at 1600 RPM	
CRANKSHAFT ROTATION (DR. END)	COUNTER-CLOCKWISE	
LENGTH	46.625 In.	
WIDTH	35.5 In.	
HEIGHT	46 In.	
WEIGHT (DRY)	1900 Lb.	
OIL CAPACITY	4 Gal.	
IGNITION	DISTRIBUTOR HIGH TENSION	
VOLTAGE	24 VOLTS	
OCTANE REQUIREMENT	80 OCTANE	
INDUCTION SYSTEM	NATURALLY ASPIRATED	
FIRING ORDER	1-8-7-3-6-5-4-2	
ACCESSORY DRIVE RATIOS	COOLING FAN	1.0
	GENERATOR	1.5
	DISTRIBUTOR	.5
	GOVERNOR	.5
	STARTER	12.15
	FUEL PUMP	ELECTRIC
	AIR COMPRESSOR	.8
	WATER PUMP	1.5
GENERATOR OUTPUT	25 Amp.	
GROSS SPECIFIC FUEL CONSUMPTION	.564 Lb/BHP/Hr at 1800 RPM	
IDLE FUEL CONSUMPTION	2.35 Gal/Hr at 420 RPM	
OIL (GRADE)	SAE 30	
GOVERNOR (MODEL)	ZENITH	
FUEL PUMP (MODEL)	AC-1539614	
DISTRIBUTOR (MODEL)	DELCO 1110618	
CARBURETOR (MODEL)	ZENITH 29D14RZ	
SPARK PLUGS (SIZE & TYPE)	14 MM CHAMPION J8	

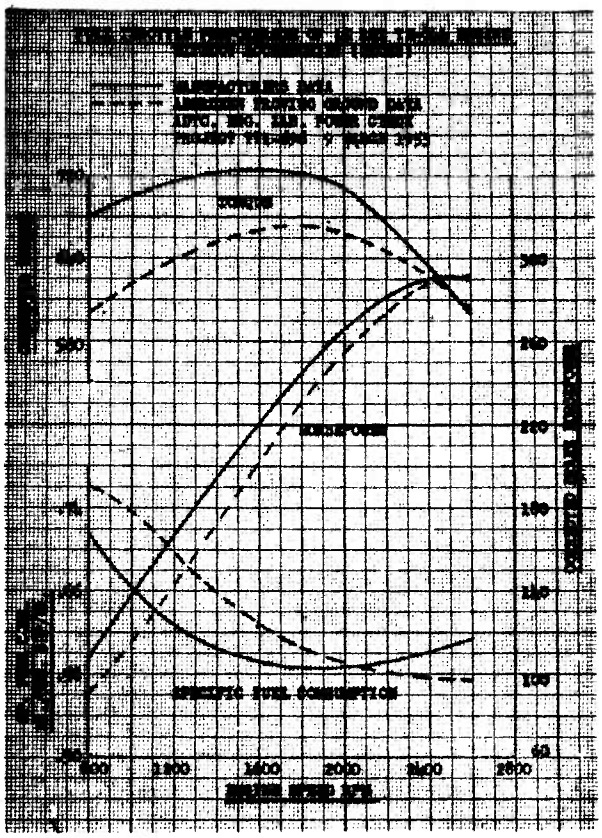


Figure 7-16. *Engine, Le Roi, TH-844*

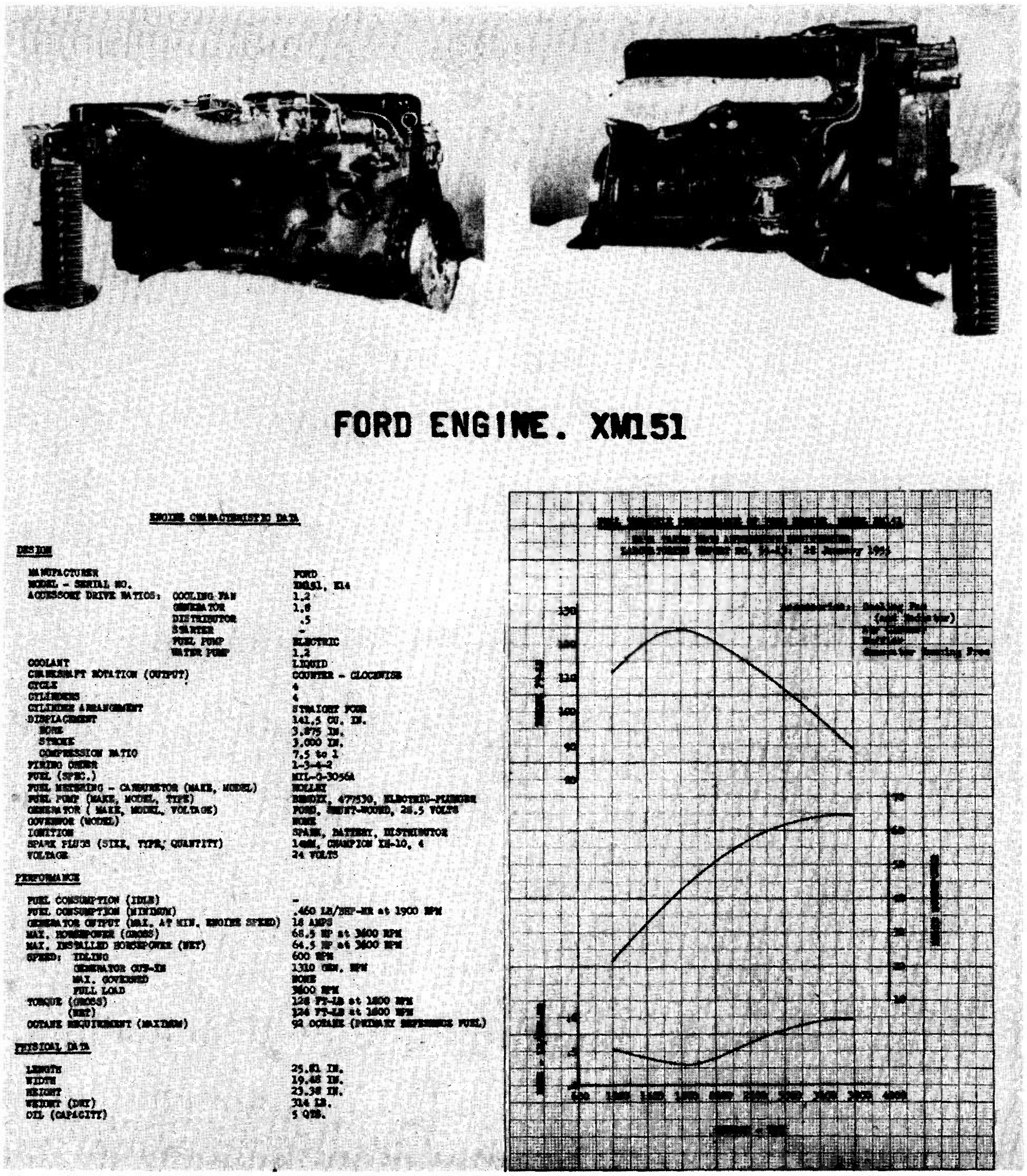
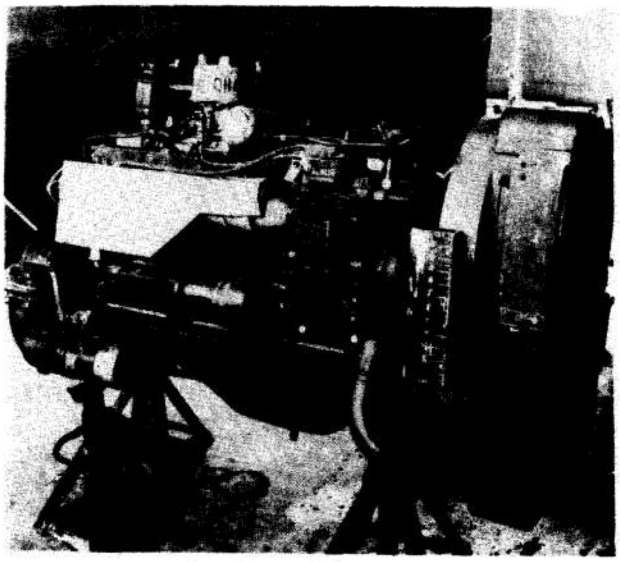


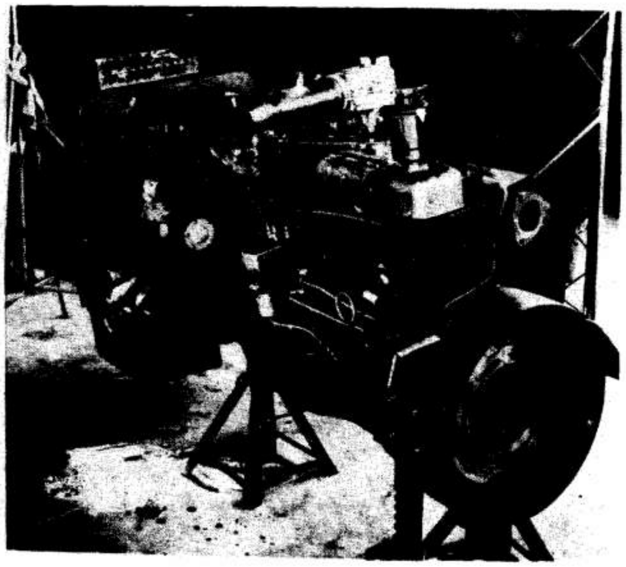
Figure 7-17. Ford Engine, XM151

a minimum transmission, and ease of obtaining double action in the cylinders.

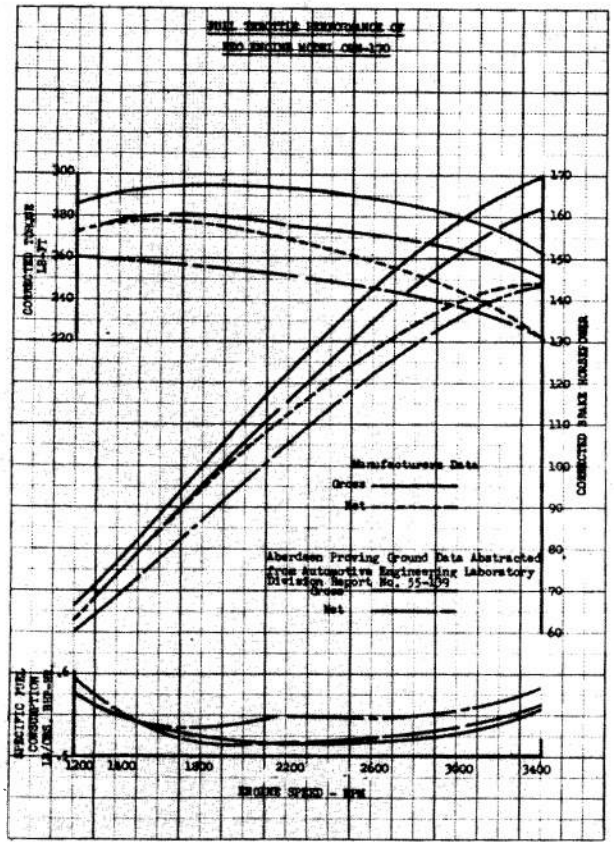
7-3.2 STIRLING-CYCLE ENGINES

The Stirling engine, which is currently under development as a potential power plant for military vehicles, differs from conventional spark-ignition and compression-ignition engines in two major aspects: It is a closed-cycle external-combustion engine having the working medium sealed within the active spaces of the engine. The fuel, the combustion air, and the products of combustion never





ENGINE, REO, MODEL OHM-170

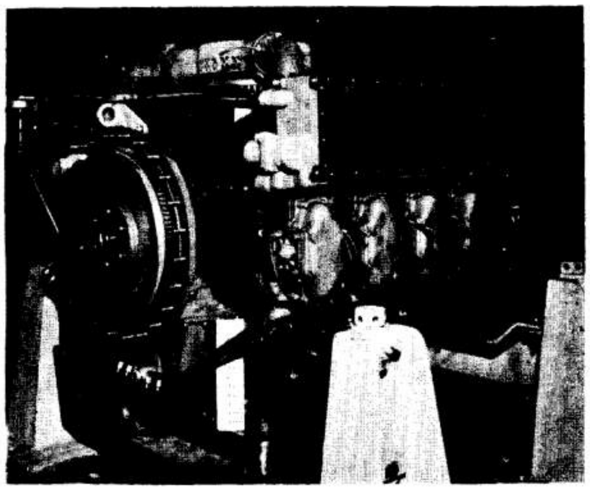


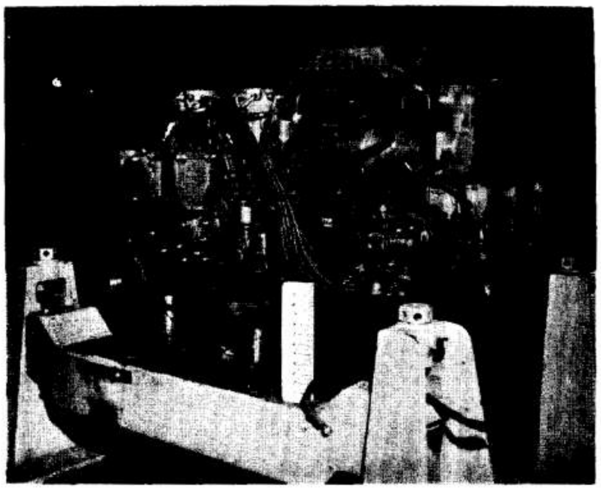
ENGINE CHARACTERISTIC DATA

DESIGN	
MANUFACTURER	REO
MODEL, SERIAL NO.	OHM-170 S.N. 3
ACCESSORY DRIVE RATIOS	
COOLING FAN	1.11
COMPRESSOR	0.568-OLD BELT 0.62-NEW BELT
GENERATOR	2.03
DISTRIBUTOR	0.5
STARTER	17.5
FUEL PUMP	ELECTRIC
HYDRAULIC STEERING PUMP	1.4
WATER PUMP	1.11
COOLANT	WATER 24 qt.
CRANKSHAFT ROTATION	COUNTER CLOCKWISE
CYCLE	4
CYLINDERS	6
CYLINDER ARRANGEMENT	IN-LINE
DISPLACEMENT	331 in.^{3}
BORE	4.125 in.
STROKE	4.125 in.
COMPRESSION RATIO	7.565:1
FIRING ORDER	1-5-3-6-2-4
FUEL SPECIFICATION	MIL-G-3056A
FUEL METERING - CARBURETOR	HOLLEY MODEL 885FFG
DISTRIBUTOR	DELCO REMY MODEL 1111556
FUEL PUMP - MAKE, MODEL, TYPE	ECLIPSE MODEL 1117495
GENERATOR - MAKE, MODEL, TYPE	DELCO-REMY
STARTER	DELCO-REMY MODEL 1108575
GOVERNOR MODEL	HOLLEY 1176
COMPRESSOR	BENDIX-WESTINGHOUSE TYPE 2E 7¼ RI
IGNITION	DIST. GAP - 0.022"
SPARK PLUGS - SIZE, TYPE, QUANTITY	14 MM. AR 58, 6
VOLTAGE	24
AIR CLEANER	DONALDSON TYPE 9 "E"
PERFORMANCE	
FUEL CONSUMPTION (IDLE) PPH	6
FUEL CONSUMPTION (MAX) PPH	85
FUEL CONSUMPTION (MIN BSFC)NET-LB/OBS-BHP	0.535 at 1800
GENERATOR OUTPUT (MAX AT MIN ENG SPEED)	25 amps
MAXIMUM HORSEPOWER GROSS	162
MAXIMUM HORSEPOWER (NET INSTALLED)	144
SPEED IDLING	400
GENERATOR CUT-IN	
MAXIMUM GOVERNED	3400
FULL LOAD	3400
TORQUE GROSS - LB-FT	281 at 1600
TORQUE NET - LB-FT	260 - 1200
OCTANE REQUIREMENT, MAXIMUM	83/92.5 SRF at 1400 RPM
MAXIMUM AIR FLOW (NET)	1038 LB/HR
PHYSICAL DATA	
LENGTH	
WIDTH	
HEIGHT	
WEIGHT - W/RADIATOR & 9 QT. OIL	1266 lb.
OIL CAPACITY	9 qt.

Figure 7-18. *Engine, Reo, Model OHM-170*

enter the engine cylinder. The working medium passes through the thermodynamic cycle repeatedly. Heat of combustion is transferred to the working fluid by means of a heat exchanger and is rejected from the fluid by means of a second exchanger. Modern Stirling engines utilize regenerators, the purpose of which is to store the heat removed from the working fluid during the constant-volume cool-





ENGINE, ORDNANCE, AIR-COOLED, AOI-1195-5X

ENGINE CHARACTERISTIC DATA

MODEL		AOI-1195-5X
MANUFACTURER		CONTINENTAL
NUMBER OF CYLINDERS		8
CYLINDER ARRANGEMENT		HORIZONTAL-OPPOSED
INDUCTION SYSTEM		NATURALLY ASPIRATED
COOLING MEDIUM		AIR
CYCLE		FOUR
FUEL		GASOLINE
OIL (GRADE)		SEASONAL
BORE		5.75
STROKE		5.75
DISPLACEMENT		1195
COMPRESSION RATIO		6.25
MAXIMUM GOVERNED SPEED		2800 RPM
RATED GROSS HORSEPOWER		560 BHP at 2800 RPM
NET HORSEPOWER		437 BHP at 2750 RPM
FUEL CONSUMPTION		20 LB/HR at 650 RPM
MINIMUM SPECIFIC - GROSS		.460 LB/BHP-HR at 2200 RPM
- NET		.536 LB/BHP-HR at 1600 RPM
RATED GROSS TORQUE		1160 FT-LB at 1800 RPM
OCTANE REQUIREMENT		81/89 at 2400 RPM
CRANKSHAFT ROTATION		CLOCKWISE
LENGTH		48 IN.
WIDTH		51.6 IN.
HEIGHT		35 IN.
WEIGHT (DRY)		2200 LB.
IGNITION		MAGNETO
VOLTAGE		24 VOLTS
FIRING ORDER		1R-6L-3R-8L-2L-5R-4L-7R
ACCESSORY DRIVE RATIOS:	COOLING FANS	1.37
	GENERATOR	2.52
	STARTER	10.0
	MAGNETOS	0.5
	TACHOMETER	0.5
	FUEL PUMP	0.5
	POWER TAKE-OFF	1.0
	GOVERNOR	1.0
	OIL PUMP	1.17
	INJECTOR	1.00
GENERATOR OUTPUT		300 AMPS
GOVERNOR (MODEL)		NOVI - 55305B
FUEL PUMP (MODEL)		TITAN - 4101-B-6B-C
FUEL INJECTOR (MODEL)		SIMMONDS - SU1963
GENERATOR (MODEL)		JACK & HEINTZ - G-22
STARTER (MODEL)		LEECE-NEVILLE - 21899
MAGNETO (MODEL)		BENDIX - L-13463-1
SPARK PLUGS (SIZE & TYPE)		TAC-4

This data is for a preproduction engine. Changes may be expected which will affect details but not general overall design or performance.

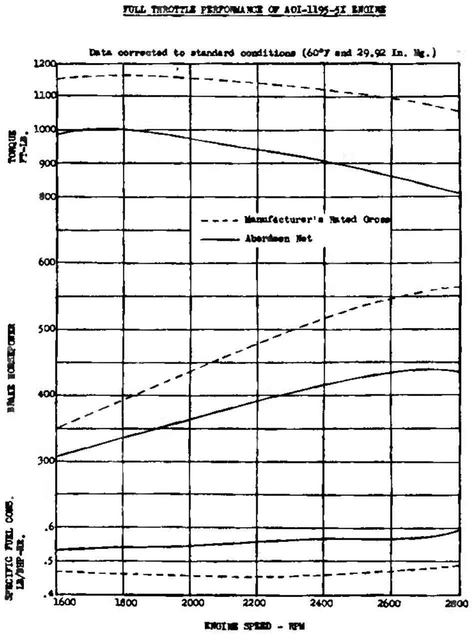


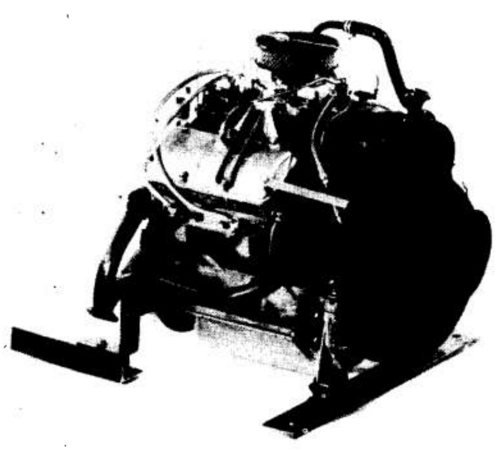
Figure 7-19. *Engine, Military, Air-Cooled, AOI-1195-5X*

ing process and return it to the fluid during the constant-volume heating process (Fig. 7-7).

The Stirling engine is a reciprocating piston engine consisting of the following basic components:

1. The heater—A combustion chamber with a fuel injection system and a heat exchanger.
2. The cooler—A heat exchanger that is continuously supplied with a coolant.
3. The power piston—The means of compressing the working fluid and of transferring the expansion work of the gas to the output shaft.
4. The displacer piston—Used to control the movement of the working fluid through the heater, cooler, and regenerator.
5. The regenerator—Extracts heat from and returns it to the working fluid.

The two pistons operate out-of-phase within the same cylinder. By proper timing, it is possible to expand and compress the fluid by means of the



AMERICAN MOTORS AV-108-3 ENGINE

ENGINE CHARACTERISTIC DATA

DESIGN	
MANUFACTURER	AMERICAN MOTORS CORP.
MODEL, SERIAL NO.	AV-108-3, S. N. 35
ACCESSORY DRIVE RATIOS	
COOLING FAN	1.92:1
GENERATOR	2:1
IGNITOR	0.5:1
FUEL PUMP	0.5:1
STARTER	-
COOLANT	AIR
CRANKSHAFT ROTATION - FROM FLYWHEEL	COUNTER CLOCKWISE
CYCLE	FOUR
CYLINDERS	FOUR
CYLINDER ARRANGEMENT	90°V
DISPLACEMENT	108 In.^{3}
BORE	3.25 In.
STROKE	3.25 In.
COMPRESSION RATIO	7.5:1
FIRING ORDER	1-3-4-2 1L, 2L, 2R, 1R
FUEL SPECIFICATION	MIL-G-3056A
FUEL METERING - CARBURETOR	CARTER AS
IGNITOR	MOD. IAU-EO 14428 UT
FUEL PUMP	AC
GENERATOR	AUTO-LITE MOD. EO 14434
LUBE OIL COOLER	McCORD TYPE TUBE & FIN
OIL PUMP	GEROTOR 5.25 Gpm at 3600 Rpm
STARTER	AUTO-LITE MOD. MCZ4001 UT
GOVERNOR	NONE
COMPRESSOR	NONE
IGNITION	IGNITOR-BREAKER GAP 0.016 In.
SPARK PLUG	14MM AR75 OR ORD-2 0.030" GAP 4 REQ'D
VOLTAGE	24
AIR CLEANER	FRAM PLEATED, PAPER
PERFORMANCE	
FUEL CONSUMPTION - IDLE - PPH	2.4
FUEL CONSUMPTION - MAX. - PPH	34.0
FUEL CONSUMPTION - MIN. - BSC at WOT(2800 Rpm)	0.605 LB/Obs. BHP
GENERATOR OUTPUT	25 Amp
MAX. NET BRAKE HORSEPOWER	53 at 3600 Rpm
MAX. NET TORQUE	89 Lb-Ft. at 2400 Rpm
IDLE SPEED	650 Rpm
GENERATOR CUT-IN SPEED	-
MAX. RATED FULL LOAD SPEED	4000
OCTANE REQUIREMENT	79/86.5 SRP at 2400 Rpm
MAX. AIR CONSUMPTION	405 Lb/Hr.
PHYSICAL DATA	
LENGTH	23 13/32 In.
WIDTH	24 1/16 In.
HEIGHT	25 13/16 In.
WEIGHT	247 Lb. Dry
OIL CAPACITY	4 Qt.

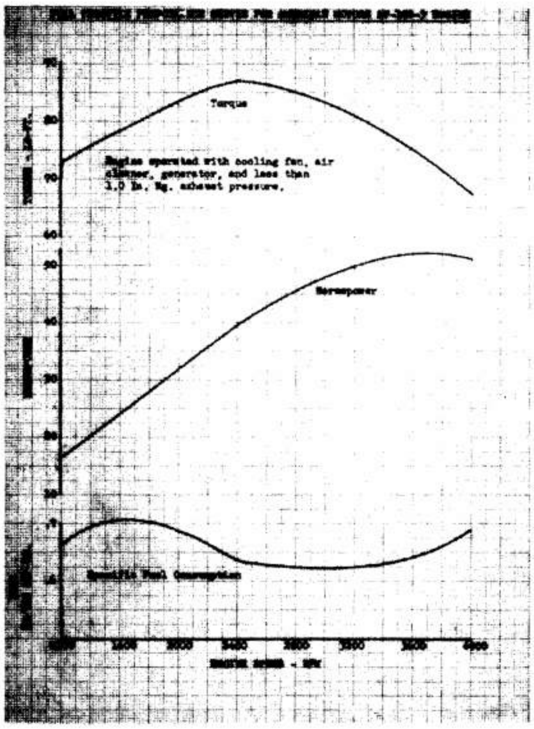
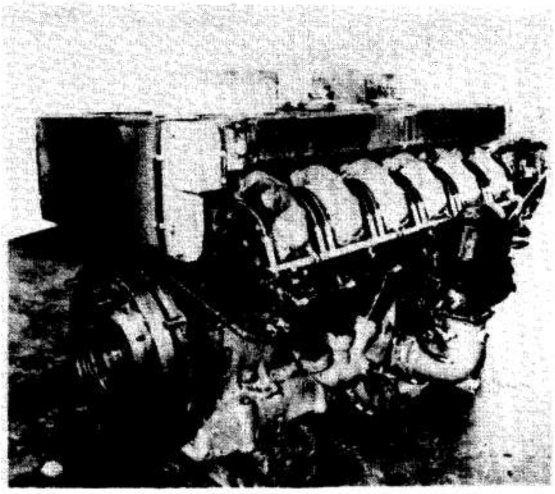


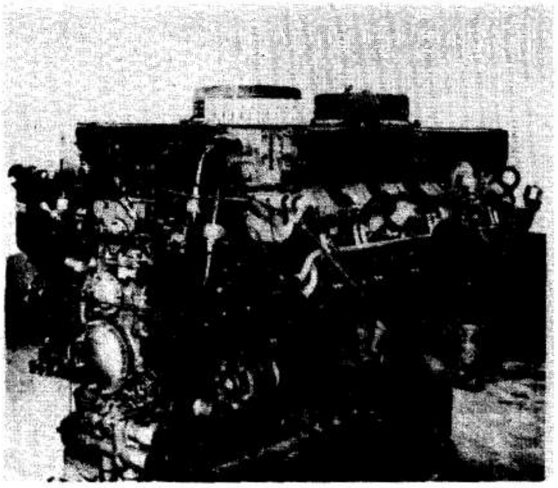
Figure 7-20. *American Motors AV-108-3 Engine*

power piston and to effect the heating and cooling of the fluid by means of the displacer piston. A complete description of the reciprocating Stirling engine is given in Ref. 18.

Inherently, the Stirling engine with regeneration has high thermal efficiency, but it must handle large quantities of working fluid to attain specific outputs (bhp/cu in) comparable to spark-ignition and compression-ignition engines. The mass of the working fluid, for a given engine, can be increased by maintaining a high mean pressure in the active spaces. The power output of a Stirling engine can be regulated by changing the mass of the working fluid passing through the thermodynamic cycle, while keeping the heater at constant temperature.

The Stirling external combustion engine is a





ENGINE, CONTINENTAL, AVSI-1790-6 No. 261

ENGINE CHARACTERISTIC DATA

DESIGN

MANUFACTURE	CONTINENTAL
MODEL - SERIAL NO.	AVSI-1790-6 #251
ACCESSORY DRIVE RATIOS	
COOLING FAN	9.00
TACHOMETER	0.50
GENERATOR	2.56
MAGNETOS	0.50
POWER TAKE-OFF	1.00
GOVERNOR	1.11
CAMSHAFTS	0.50
STARTER	1.51
FUEL PUMP	0.86
SUPERCHARGERS	7.86
OIL PUMP	1.70
INJECTORS	1.00
COOLANT	AIR
CRANKSHAFT ROTATION (DR. END)	COUNTER-CLOCKWISE
CYCLES	4
CYLINDERS	12
CYLINDER ARRANGEMENT	UPRIGHT 90° VEE
DISPLACEMENT	1790 CU. IN.
BORE	5.75 IN.
STROKE	5.75 IN.
COMPRESSION RATIO	6.5 : 1
FIRING ORDER	1R, 2L, 5R, 4L, 3R, 1L, 6R, 5L, 2R, 3L, 4R, 6L
FUEL (SPEC.)	MIL-G-3056A (83-91 OCTANE)
FUEL METERING	INJECTION
FUEL PUMP	TITAN H2x3-15
GENERATOR	BENDIX-ECLIPSE 30E00-3A
GOVERNOR	NOVI 54470A
IGNITION	4 BENDIX-SCINTILLA'S MAGNETOS S6LN-32
SPARK PLUGS (24)	14 MM CHAMPION TAC-2 AUTOLITE AER-28 AC-WR-49-L BG-RB-897-S
VOLTAGE	24 VOLTS

PERFORMANCE

FUEL CONSUMPTION (IDLE)	33 LB/HR at 550 RPM
FUEL CONSUMPTION (SPECIFIC)	.51 LB/OBS BHP-HR. at 1800 RPM
GENERATOR OUTPUT	300 AMP at 850 RPM
MAXIMUM GROSS HP	1020 at 2800 RPM
MAXIMUM NET HP	779 at 2600 RPM
OCTANE REQUIREMENT	81/89 at 2200 RPM
SPEED: IDLING	650 RPM
MAXIMUM GOVERNED	2950 RPM (NO LOAD)
FULL LOAD	2800-2840 RPM
TORQUE: GROSS	1758 AT 2100 RPM
NET	1938 at 2400 RPM

PHYSICAL DATA

LENGTH	72.83 IN.
WIDTH	56.74 IN.
HEIGHT	45.68 IN.
WEIGHT (WITH ACCESSORIES)	3050#
OIL	16 GALLONS

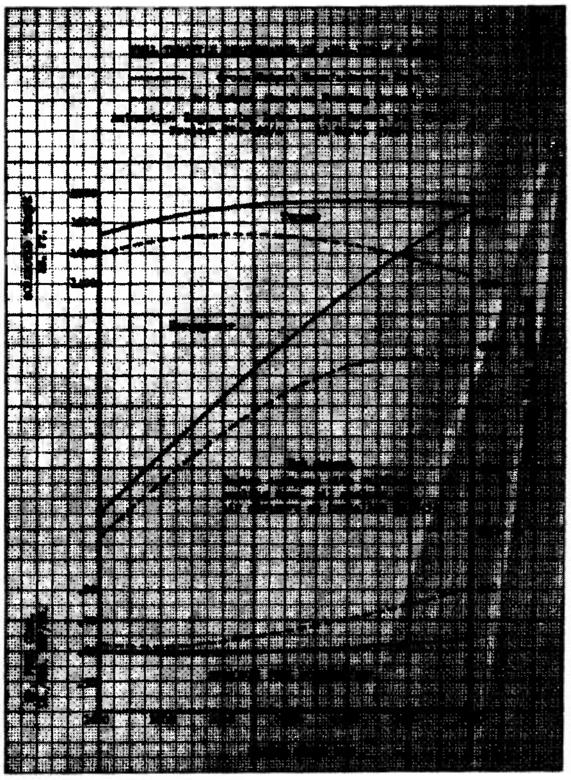
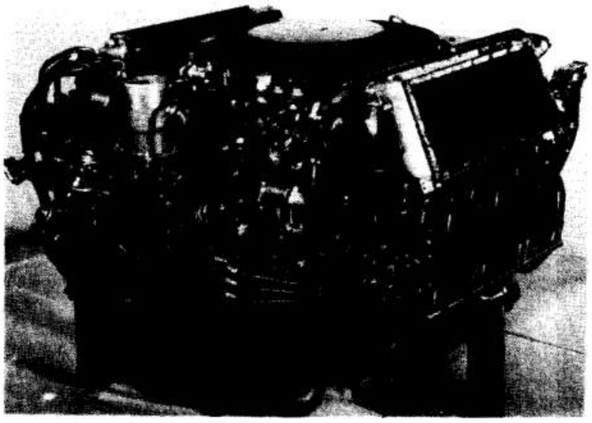
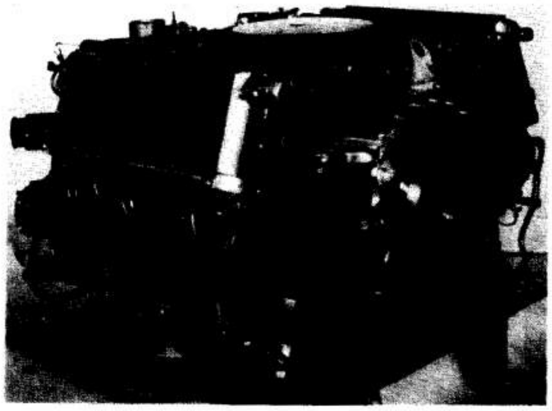


Figure 7-21. *Engine, Continental, AVSI-1790-6*

true multifuel engine; a wide range of fuels may be used. Current developmental Stirling engines are silent compared to more conventional internal combustion engines; the most evident noise is produced by the timing gears on the crankshafts. Current developmental Stirling engines use hydrogen as the working fluid. Since hydrogen has lower viscosity and higher heat capacity than air, it reduces the pumping losses associated with air and increases engine efficiency.





ENGINE, ORDNANCE, AIR-COOLED, A01-895-4A

ENGINE CHARACTERISTIC DATA

MODEL		AOI-895-4A
MANUFACTURER		CONTINENTAL
NUMBER OF CYLINDERS		6
CYLINDER ARRANGEMENT		HORIZONTALLY OPPOSED
COOLING MEDIUM		AIR
CYCLE		4
FUEL		GASOLINE
BORE		5.75 In.
STROKE		5.75 In.
DISPLACEMENT		895 Cu. In.
COMPRESSION RATIO		6.5 to 1
MAXIMUM GOVERNED SPEED		2950 RPM (NO LOAD)
RATED GROSS HORSEPOWER		420 at 2800 RPM
NET HORSEPOWER (INSTALLED)		310 at 2800 RPM
MAXIMUM GROSS TORQUE		840 Ft. Lbs. at 2200 RPM
CRANKSHAFT ROTATION (DR. END)		COUNTER-CLOCKWISE
LENGTH		47 In.
WIDTH		54 In.
HEIGHT		37 In.
WEIGHT (DRY)		
OIL CAPACITY		14 Gal.
IGNITION		DUAL MAGNETOS
VOLTAGE		24 VOLTS
OCTANE REQUIREMENT		80/87 OCTANE (SHF)
INDUCTION SYSTEM		NATURALLY ASPIRATED
FIRING ORDER		1-6-3-2-5-4
ACCESSORY DRIVE RATIOS:	COOLING FANS	1.44
	GENERATOR	2.60
	STARTER	0.91
	MAGNETOS	0.50
	TACHOMETER	0.50
	FUEL PUMP	0.50
	POWER TAKE-OFF	1.00
	GOVERNOR	1.06
GENERATOR OUTPUT		150 Amps, 28.5 VOLTS
FUEL CONSUMPTION, IDLE		18 Lb/Hr at 700 RPM
MINIMUM SPECIFIC - GROSS		.495 Lb/BHP/Hr at 2300 RPM
- NET		.536 Lb/BHP/Hr at 1900 RPM
OIL (GRADE)		SEASONAL
GOVERNOR (MODEL)		NOVI
FUEL PUMP (MODEL)		TITAN - G6N 203/4
GENERATOR (MODEL)		BENDIX - 14806-N
STARTER (MODEL)		BENDIX
MAGNETO (MODEL)		BENDIX - S6L N-32
FUEL INJECTOR (MODEL)		SIMMONDS - SU15-F
SPARK PLUGS (SIZE & TYPE)		14 MM - AC - RESISTOR

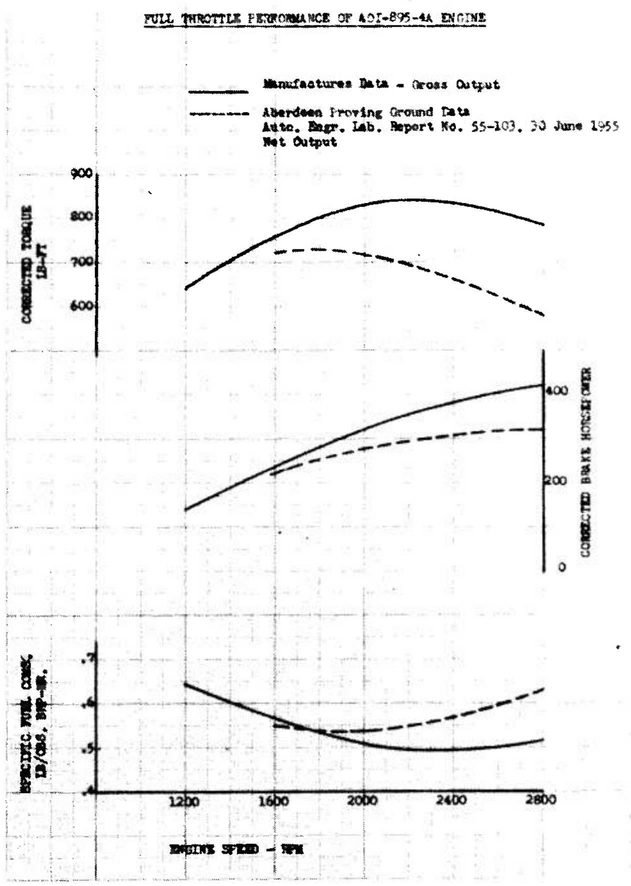


Figure 7-22. *Engine, Military, Air-Cooled, AOI-895-4A*

Some of the disadvantages of the Stirling engine are:

1. Most of the heat rejected from the Stirling engine cycle is rejected to the cooler; very little leaves through the exhaust gases as in conventional internal combustion engines. Therefore, the cooling system of a Stirling engine would be larger than that of a comparable internal combustion unit.
2. A blower must be used to force air through

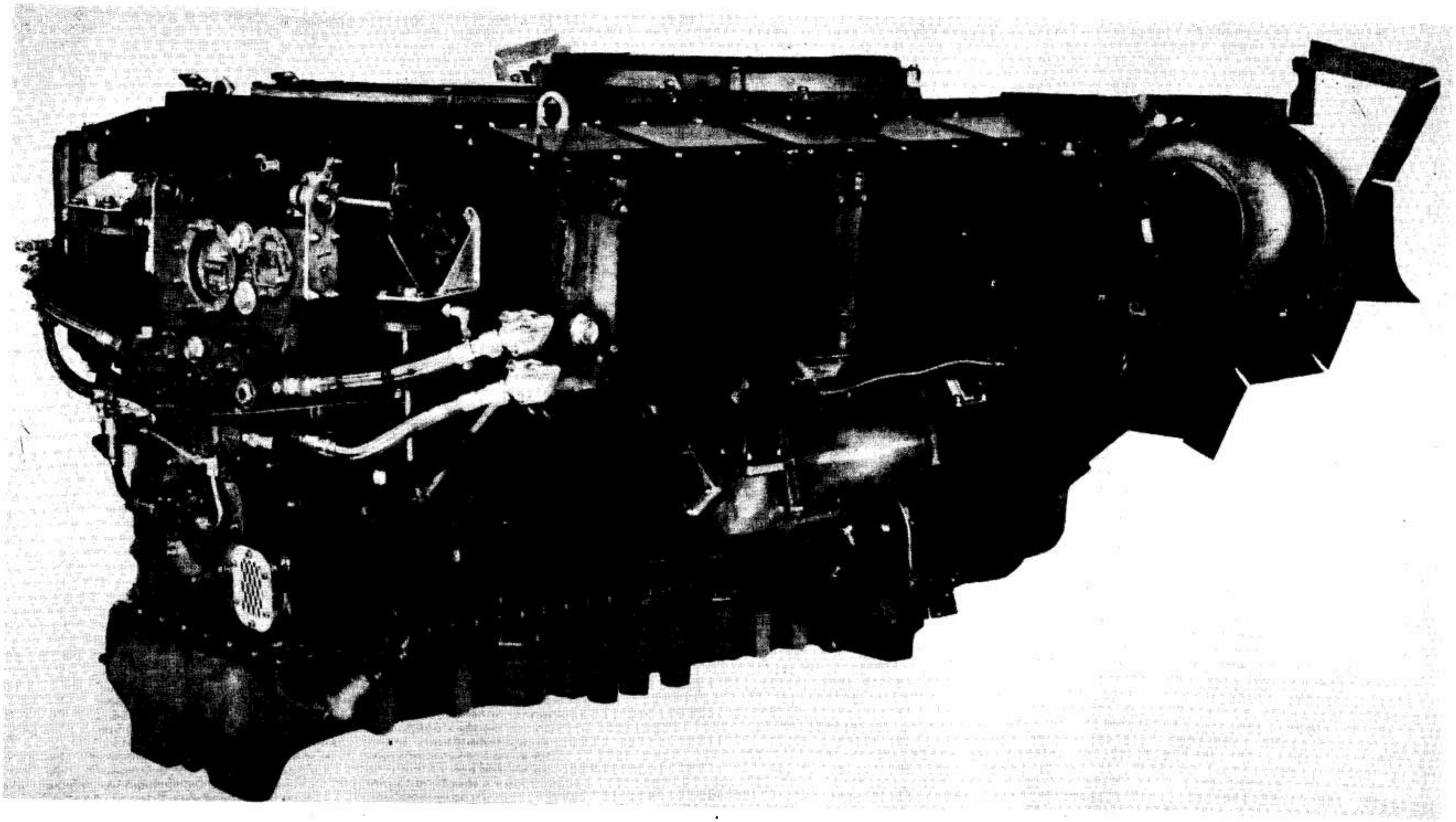


Figure 7-23. Engine, Military, Air-Cooled, Diesel, AVDS-1790-2 (Courtesy Continental Motors Corp., Military Division)

TABLE 7-3 REPRESENTATIVE STANDARD MILITARY VEHICLE ENGINES

Part No.	Bhp at Rpm	Cool-ant	Type	Induc-tion	Ign.	No. Cyl.	Displace-ment, cu in	Length, in	Width, in	Height, in	Weight, lb
GASOLINE ENGINES											
*CE-605127	5.24 3600	A	IL	C	M	2	—	16.00	16.00	15.56	43
*8328393	70 4000	L	IL	C	D	4	134.2				499
*7032647	94 3200	L	IL	C	D	6	230.2				675
8717486	127 3200	A	O	I	M	4	268	29.18	37.00	30.64	565‡
*7411599	145 3400	L	IL	C	D	6	301.6				785
*7538638	146 3400	L	IL	C	D	6	331				1038
*8327037	224 2800	L	IL	C	D	6	602				1920
*8333610	297 2600	L	90° V	C	D	8	844				1950
7388574	375 2800	A	O	C	M	6	895	49.24	50.72	36.96	1856*
7355952	810 2800	A	90° V	C	M	12	1790	68.00	58.80	39.84	2200†
875265(M60)	1020 2800	A	90° V	SI	M	12	1790				
DIESEL ENGINES											
	365 2800	A	90° V	SI	—	8	750	55.62	39.31	34.91	2340
K8125265	700 2400	A	90° V	SI	—	12	1790	66.56	59.60	44.21	4200

LEGEND

A—Air-Cooled
I—Fuel Injected
S—Supercharged
L—Liquid-Cooled
O—Horizontal—Opposed
M—Magnetic
C—Carburetor
D—Distributor
IL—Inline

* Includes engine and transmission oil-coolers and cooling system and all accessories except generator.

† Includes all engine accessories except generator and oil-coolers.

‡ Includes cooling fan and all accessories except generator.

the pre-heater (if any) and into the com-bustion chamber. The blower constitutes a parasitic power loss and could be a source of noise.

3. Cost estimates based on existing Stirling engines indicate that production Stirling engines will cost appreciably more than other modern reciprocating engines, primarily because of the complexity of the heat exchangers and the drive and timing mechanisms.

7-4 TURBINE ENGINES

Turbine engines are essentially devices that transform fluid energy into mechanical work by changing the momentum of a flowing fluid. In general, turbine power plants can be classified as internal combustion and external combustion types. In the former, the working fluid includes the products of combustion, while, in the latter, the working fluid is heated by means of a high-temperature source used in conjunction with a heat exchanger, which physically prevents the intermixing of the products of combustion and the working medium.

7-4.1 GAS TURBINES

The present discussion will be limited to multiple-shaft turbines of: (1) open-cycle type, and (2) closed-cycle type. Multiple-shaft turbines have a separate output turbine in series or parallel with one or more turbines driving the compressor. Performance studies of gas turbine types indicate that multiple-shaft gas turbine arrangements are best suited to the requirements of automotive vehicles.

7-4.1.1 Open-Cycle Gas Turbines

Open-cycle multiple-shaft turbines (internal combustion turbines) consist of four basic components: a compressor, a compressor turbine, a combustion chamber, and a power turbine. Figure 7-24 shows the arrangement of these components for an open-cycle unit. There is no mechanical connection between the compressor and power turbines. Figure 7-24 also includes a regenerator, as frequently used on developmental automotive gas turbines; Fig. 7-25 shows the component arrangement for a nonregenerative multiple-shaft turbine. Regenerators (heat exchangers), which transfer heat from the turbine exhaust to the air leaving the compressor, increase thermal efficiency, but at the penalty of increased weight, bulk, and cost of the power plant. Regenerators must have low pressure drops and be capable of withstanding large temperature changes. Rotary regenerators offering low weight, compactness, and good performance are

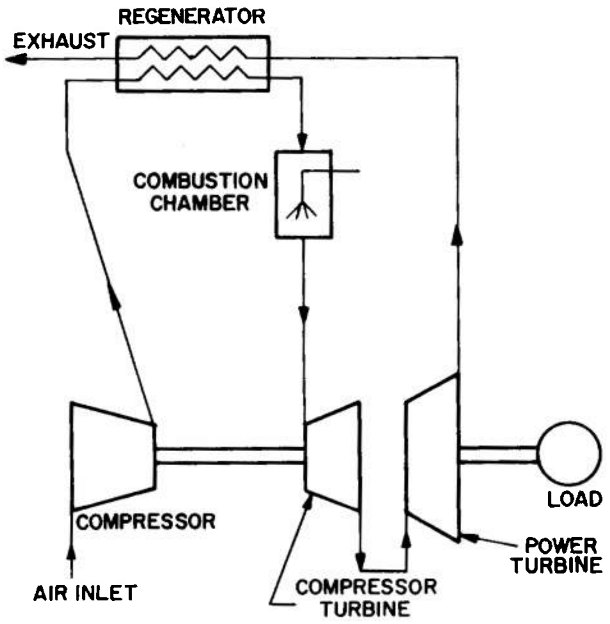


Figure 7-24. *Schematic Diagram of Regenerative Open-Cycle Turbine*

being developed, but sealing remains a problem.

The performance characteristics of open-cycle multiple-shaft turbines are well suited to vehicle traction applications, owing to the rising torque curve in the overload range (Fig. 7-13). A major advantage of the multiple-shaft turbine over single-shaft turbines is the wide load-speed range of the former. The multiple-shaft gas turbine accomplishes variable load operation primarily through changes in air flow and pressure ratio, with moderate changes in turbine inlet temperature. This characteristic permits gas turbines to be designed for relatively good part-load efficiency and the capability of producing full-load at high altitudes and high ambient temperatures. The gas turbine power plant is much more sensitive to changes in ambient temperature and changes in altitude than is the reciprocating internal combustion engine. Figure 7-26 shows typical curves for power variation with variable ambient air temperature for a reciprocating gasoline engine and a gas turbine.

The open-cycle multiple-shaft turbine may be compared with conventional reciprocating engines for military vehicles as follows:

Advantages:

a. The speed-torque characteristics of the turbine are better suited to vehicle propulsion under most circumstances.

b. The wide range of operation and the rising torque curve as turbine speed decreases should simplify the transmission system requirements.

c. The bulk and weight per horsepower of the turbine are much less than those of comparable reciprocating engines, especially if a regenerator is not used.

d. A wide variety of fuels may be used in turbines; gas turbines are true multifuel engines that are not sensitive to octane or cetane ratings.

e. Gas turbines have excellent cold-weather starting characteristics.

f. Gas turbines are much simpler mechanically than conventional reciprocating engines.

g. The absence of reciprocating mechanisms in a turbine contributes to smooth, vibrationless operation.

h. The cooling system of a gas turbine can be much smaller than that of a comparable reciprocating engine since the turbine rejects a greater proportion of the total heat to its exhaust.

Disadvantages:

a. The multiple-shaft gas turbine is slow in response to large and sudden changes in load. If the unit is running at light load, the compressor and gasifier turbine are operating at relatively low speed. The turbine speed will be governed by the load at a given compressor speed. If the load on the output shaft increases suddenly, the power turbine will begin to slow down. The governor will increase the fuel flow to the combustion chamber. Full torque for acceleration or correction of the power turbine speed is not immediately available because it takes time for the gasifier to accelerate to the required speed. The acceleration rate of the gasifier is limited by the maximum permissible turbine inlet temperature. This lag can be reduced by limiting the minimum permissible speed of the compressor; however, overall fuel consumption would then tend to increase.

b. In general, the brake specific fuel consumption of gas turbines is higher than that of comparable reciprocating engines. Further development of turbines is required to bring the brake specific fuel consumption to the level of gasoline and Diesel reciprocating engines.

c. Turbines require large inlet and exhaust ducts.

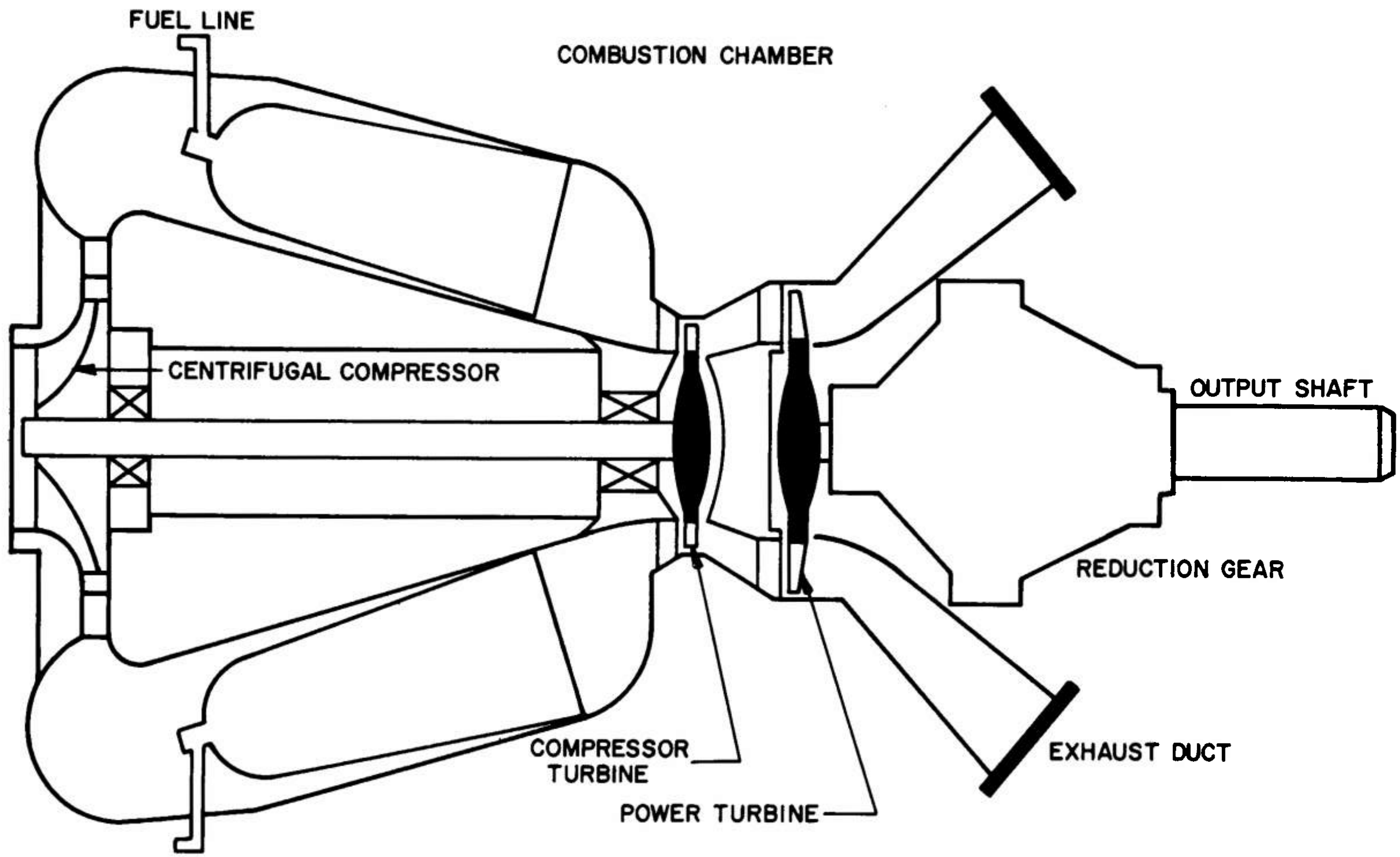


Figure 7-25. Component Arrangement for a Typical Multiple-Shaft Nonregenerative Gas Turbine

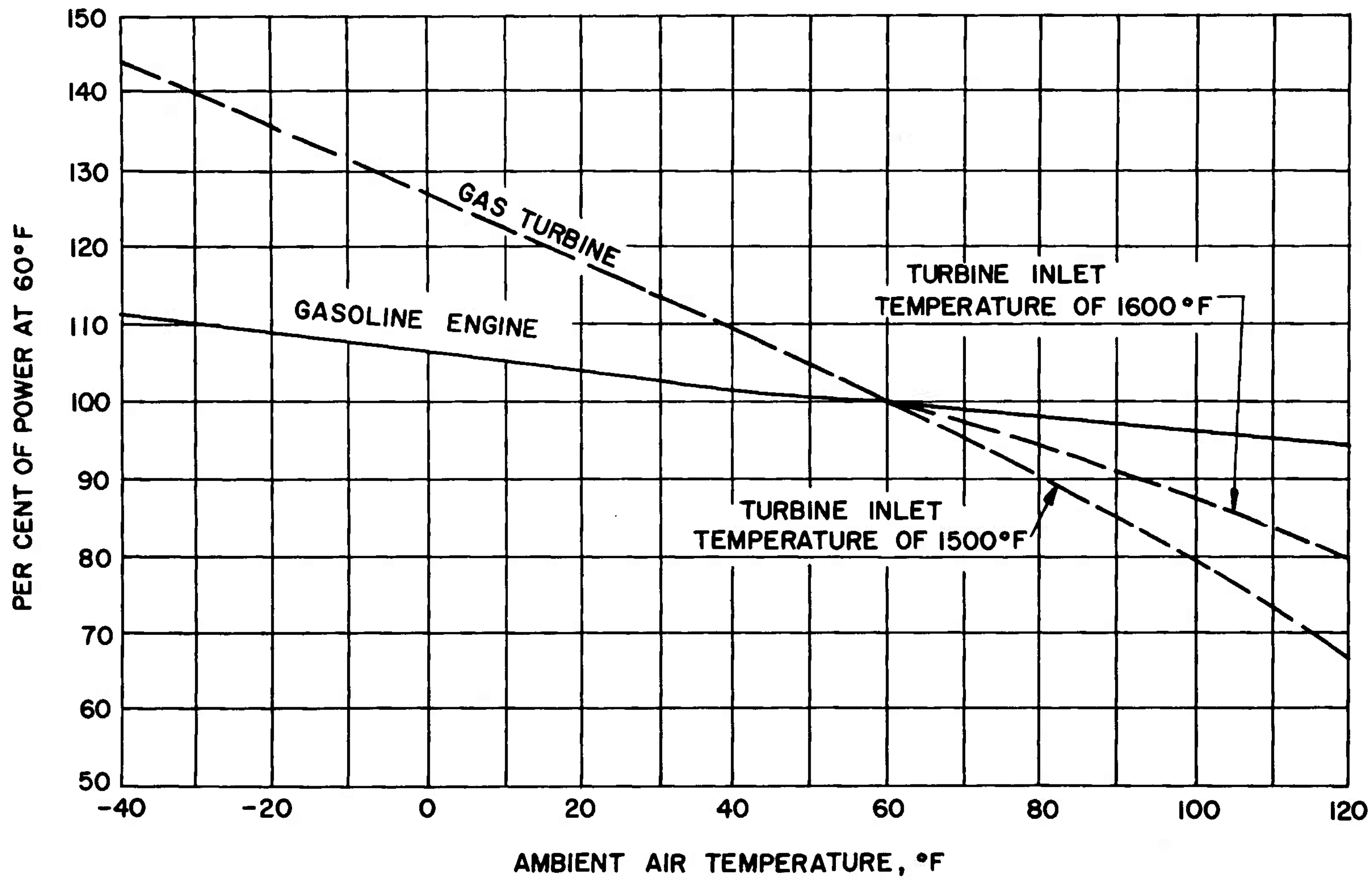


Figure 7-26. *Power Variations Versus Ambient Temperature for Typical Gasoline Engines and Multiple-Shaft Gas Turbines*

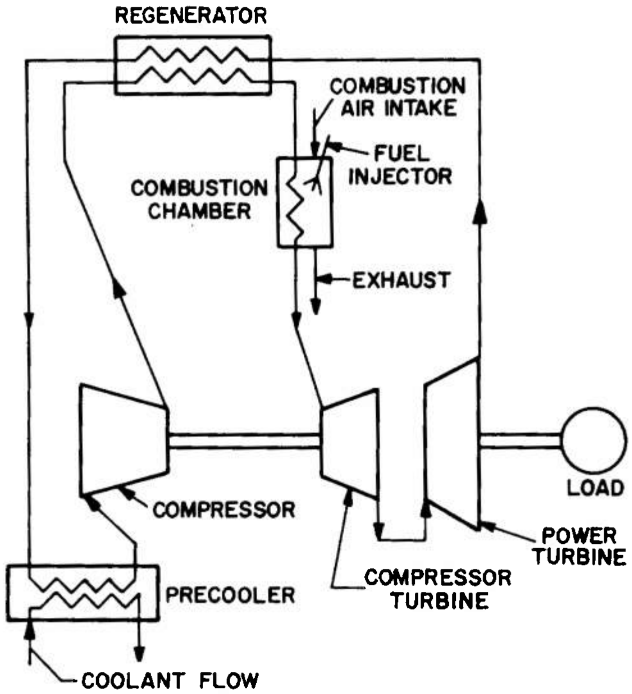


Figure 7-27. Schematic Diagram of Regenerative Closed-Cycle Turbine

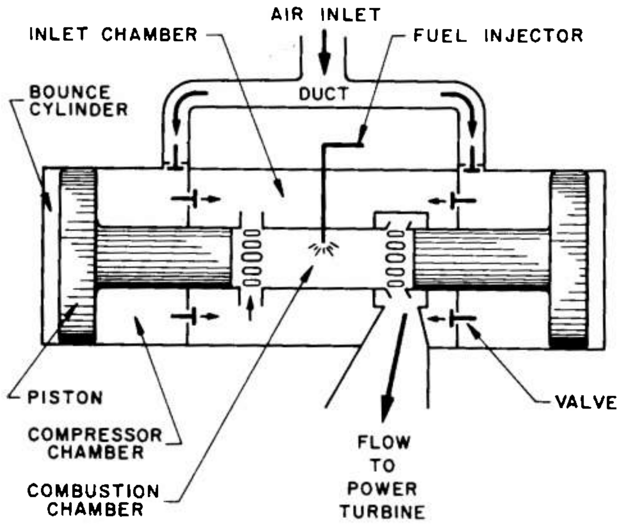


Figure 7-28. Schematic of Free-Piston Gasifier Turbine

The high mass flow rate of turbines dictates an air filtering system having a minimum pressure drop to maintain efficiency.

d. Thermal efficiencies of current gas turbines are limited by turbine inlet temperatures. The reciprocating piston engine can operate with very high combustion temperatures (heat addition temperatures) since the duration of the combustion is very short. The continuous combustion temperature of the gas turbine is the constant operating temperature of nozzles, passageways, and blazing. Material limitations impose a maximum turbine inlet temperature range of 1,000° to 1,800°F on current gas turbines.

e. The cost of current gas turbines is relatively high. Developmental costs, the use of special alloys, balancing requirements, high-quality reduction gearing, and special bearings add to the total cost.

7-4.1.2 Closed-Cycle Gas Turbines

In the closed-cycle gas turbine (Fig. 7-27), the working fluid is continuously recycled within a sealed system. The closed-cycle turbine requires a closed heater (heat is transferred to the working fluid through walls) and a precooler to reduce the temperature of the working medium before recompression. The working fluid can be pressurized. The increased density of the gas makes possible the use of a smaller compressor and turbine than possible for similar units in an open-cycle plant, for a given output.

In a closed-cycle unit, a wide range of load can be handled, with slight change in speed and efficiency, by varying the pressure at the compressor inlet. The clean working fluid is less detrimental to the components of the plant than are the gases acting on similar components in an open-cycle plant.

The major disadvantages of the closed-cycle unit are: (a) the heater is relatively large and expensive, and (b) a precooler is essential for practicable efficiencies.

7-4.2 FREE-PISTON GASIFIER-TURBINE

The free-piston power plant consists of a free-piston compressor-gas generator and a turbine. The gas generator or gasifier, supplies high-temperature gas to the power turbine. Figure 7-28 shows a diagrammatic section of a typical free-piston power plant.

All current units of this type are two-stroke cycle compression-ignition engines. The operating cycle may be described as follows: Combustion is initiated when the opposed pistons are in the extreme inward position. As the opposed pistons are forced apart by the high-pressure combustion gases, air is compressed in the bounce chambers. The pistons are stopped by the bounce chamber pressure and then forced inward again. As the pistons move outward, one of them uncovers the exhaust ports in the cylinder wall, allowing the high-pressure gases to escape. These escaping gases act directly on the power turbine. The opposite piston, traveling outward, uncovers the intake ports and admits air under pressure, scavenging and charging the cylinder. The pressure within the bounce chamber returns the pistons to start the cycle again. The inlet air is stored under pressure in a chamber charged by the pistons acting in the compressor cylinder. Fuel injection is timed as in a conventional Diesel engine. An external linage or rack is commonly used to synchronize the motion of the pistons.

The free-piston engine has the following advantages over conventional Diesel engines of the same piston area:

a. Free-piston engines inherently adjust the compression ratio to the fuel requirements. The tolerance of low-ignition-quality fuel is apparently due to the self-adjusting compression ratio.

b. The opposed-piston units are in perfect dynamic balance; thus, the vibration control problem is simplified.

c. The use of a turbine instead of a crankshaft and connecting rods may save some weight and will simplify lubrication.

d. The weight- and space-saving features of small cylinders may be utilized by placing a number of gasifiers in parallel with a single turbine.

e. The speed-torque characteristics of a free-piston engine allow the employment of a simpler transmission than necessary with a conventional reciprocating piston engine, for similar performance.

Some of the disadvantages of the free-piston engine are:

a. Fuel economy at maximum rated output of current free-piston units is poorer than that of other comparable internal combustion engines.

b. Part-load fuel economy (brake specific fuel consumption) lb/bhp-hr is still poorer relative to other engines. The narrow range of the frequency of oscillation of the free pistons causes some waste of air from the compressor at light loads. (The physical system is such that the frequency of oscillation of the pistons during idling is greater than 50% of maximum frequency. This results in high friction losses during idling or part-load operation.)

c. The turbine and necessary reduction gearing is relatively expensive, and service life is not determined at present.

An alternate compound engine design utilizes opposed pistons and compression-ignition in conjunction with two crankshafts. The power from the crankshafts is used solely for scavenging and cooling.

7-4.3 STEAM TURBINES (Ref. 13)

Steam turbines as a means of propulsion are confined to marine vessels, at present. A comparison between reciprocating internal combustion engines and steam turbine power plants results in the following partial list of advantages for each type of power source.

Reciprocating internal combustion engines have:

a. Higher maximum efficiencies.

b. Simpler cooling systems.

c. Higher output (hp) to weight or bulk ratio for units under 10,000 hp.

Steam-turbine power plants have:

a. A much greater range of fuels.

b. No reciprocating parts, hence greater freedom from vibration.

c. A weight and bulk advantage for very large units up to 200,000 hp or more.

The requirements of minimum weight and size per unit output, in the 50- to 1,000-bhp range, have enhanced the position of the internal combustion engine as the primary military vehicle power plant.

7-5 THRUST ENGINES (Refs. 13, 19, 37, 38)

Thrust engines are characterized by a rapid ejection of a working fluid from within the body of the vehicle. The forward thrust is produced by the reaction to the time rate-of-change of linear momentum of the ejected fluid. Any type of fluid can form the fluid jet that is discharged. Only two types, however, have been found suitable for propelling vehicles through the atmosphere; these are the thermal-jet and the chemical rocket jet. The thermal-jet is comprised of highly heated, compressed atmospheric air admixed with the products of combustion produced during the burning of the fuel in the air to obtain the desired temperature. Thermal-jet engines are divided into three basic types: turbojets, pulsejets, and the ramjets. The chemical rocket jet is comprised of large quantities of high-temperature, high-pressure gases produced by fuel and oxidizing chemicals reacting chemically in a suitable device. No atmospheric air is required. The equipment wherein the chemical reaction takes place, including the exhaust nozzles and controls, is called the rocket motor or the rocket engine in accordance with its utilization of solid or liquid propellant. The term "engine" is also used sometimes in a generic sense to include motors and engines.

Thrust engines are not employed currently to power military land or amphibious vehicles, but a general discussion of them is included in this work in the event that they should be needed in the future.

7-5.1 BASIC PERFORMANCE RELATIONSHIPS

The basic performance relationships for jet engines can be determined by use of the momentum change principle applied to a steady-state flowing fluid.

The force exerted by the flowing stream, or jet, is termed *thrust, T*. Thus, for a thermal-jet engine,

$$T = \frac{m_o V_e - m_i V}{g} = \frac{m_i}{g}(V_e - V) + \frac{m_f}{g} V_e \quad (7\text{-}11)$$

where

- m_o is the mass rate of outflow of exhaust gases, slug/sec
- m_i is the mass rate of inflow of air, slug/sec
- m_f is the mass rate of flow of fuel, slug/sec
- V_e is the exit relative velocity of the fluid, fps
- V is the inlet velocity of the fluid, fps
- g is the acceleration due to gravity, ft/sec^2

The rocket engine carries its own fuel and oxidizer, hence does not induct air: $m_i = 0$. Therefore, for the rocket engine,

$$T = \frac{m_f}{g} V_e \quad (7\text{-}12)$$

In this analysis, the pressure thrust is neglected since it normally is small compared to the momentum thrust. Pressure thrust results from excess static exit pressure (relative to ambient pressure) in the tailpipe of a thrust-propelled vehicle. Another way of eliminating pressure thrust from consideration is to assume that the exit velocity of the exhaust gases of the jet-propelled vehicle is achieved by expanding to the ambient pressure.

Thrust power is defined as:

$$\text{thrust power} = TV \quad (7\text{-}13)$$

Leaving losses occur in jet propulsion as a result of the residual kinetic energy in the exhaust gases. Since the exhaust gases have the absolute velocity of $(V_e - V)$, leaving loss is written as

$$\text{leaving loss} = \frac{m_e}{g}(V_e - V)^2 \quad (7\text{-}14)$$

Total power generated in the engine is termed propulsion power.

Therefore,

$$\text{propulsion power} = \text{thrust power} + \text{leaving loss.}$$

Propulsion power can be found by adding appropriate equations, giving

$$\text{propulsion power (thermal-jet)} =$$

$$\frac{1}{2g}(m_o V_e^2 + m_f V^2 - m_i V^2) \quad (7\text{-}15)$$

$$\text{propulsion power (rocket)} =$$

$$\frac{m_f}{2g}(V_e^2 + V^2) \quad (7\text{-}16)$$

The ideal propulsion efficiency, N_p, of the engines under consideration can be written as

$$N_p = \frac{\text{thrust power}}{\text{propulsion power}} \quad (7\text{-}17)$$

The thermal efficiency, N_{th}, is defined as

$$N_{th} = \frac{\text{propulsion power}}{\text{energy supply rate}} \quad (7\text{-}18)$$

The energy supply rate equals $m_f Q$, where Q is the heat of combustion (btu/lb).

The overall efficiency, N, is

$$N = N_p N_{th} \quad (7\text{-}19)$$

Specific fuel consumption, *sfc*, for jet propulsion engines, may be based on thrust or thrust horsepower, T_{hp},

$$T_{sfc} = \frac{\text{specific fuel rate}}{T} \quad (7\text{-}20)$$

$$T_{hp\ sfc} = \frac{\text{specific fuel rate}}{T_{hp}} \quad (7\text{-}21)$$

7-5.2 THERMAL-JET ENGINES

Thermal-jet engines are characterized by their use of atmospheric air, which is inducted, compressed, and raised to the desired temperature by the combustion of fuel injection into the air. In all of the thermal-jet engines to be considered, the induction of air and the ejection of gases takes place in the direction of the line-of-flight, i.e., axial to the vehicle. Thermal-jet engines may be classified as three basic types: (a) ramjet engines, (b) pulsejet engines, and (c) turbojet engines.

7-5.2.1 Ramjet Engines

Ramjet engines are normally designed to operate at supersonic speeds. This engine consists of three basic components: (a) a diffusion system (b) a combustion chamber, and (c) an exhaust nozzle.

The diffusion system has a supersonic stage, and a subsonic stage. Under operating conditions, the supersonic air stream is decelerated by the first-stage (supersonic) diffuser to approximately Mach 1 at the entrance to the second-stage (subsonic) diffuser. The supersonic diffusion process results in an increase in pressure within the engine. The subsonic diffuser reduces the velocity further (to about Mach 0.2) with a corresponding increase in the pressure of the air stream. The air stream then moves into the combustion chamber where its temperature is raised. Both the specific volume and the mass flow rate of the fluid increase as it passes through the combustion chamber and the exhaust nozzle. The increase in the momentum of the fluid gives reaction force or thrust. The ramjet engine must be accelerated to a minimum flight speed by an external force in order to develop the required minimum operating pressures.

7-5.2.2 Pulsejet Engines

The pulsejet engine is an intermittent flow unit. The basic components are: (a) the inlet diffuser, (b) the inlet valve bank, (c) the combustion chamber, (d) the fuel injection system, (e) the ignition system, and (f) the exhaust nozzle-tailpipe.

Like the ramjet engine, the combustion air is forced into the pulsejet engine by ram pressure during flight. However, a flapper valve assembly limits the flow to a downstream direction only. When the fuel-air mixture is ignited in the combustion chamber, the increase in pressure in the chamber forces the valves closed, and the combustion gases are discharged through the tailpipe with a velocity greater than that of the entering air. The ejection of the gases causes a decrease in the combustion chamber pressure to a valve below that in the entrance diffuser. This causes the one-way valves to open and admits a new charge to the chamber. The cycle repeats itself with a frequency that depends on the physical dimensions of the unit.

7-5.2.3 Turbojet Engines

Turbojet engines are designed to operate at both subsonic and supersonic speeds. At present, turbojet engines are the only thermal-jet engines used for standard piloted aircraft. Turbojet engines are well suited to propelling vehicles in the 500- to 1,800-mph range. In the upper part of the range, the characteristics of the turbojet engine tend to change to those of the ramjet; at speeds above approximately 1,800 mph, pure ramjet engines are advantageous.

The basic components of the turbojet engine are: (a) the diffuser, (b) the air compressor (axial flow or centrifugal), (c) the combustion chamber, (d) the compressor turbine, (e) the tailpipe and nozzle, and (f) the afterburner (optional).

7-5.2.4 Rocket Engines

Chemical rocket engines may be classified as: (a) liquid-propellant rocket engines, and (b) solid-propellant rocket motors. In each type, a fuel and an oxidizer react in a combustion chamber and pass through a suitable nozzle to produce the high-speed gaseous jet required for propulsion.

7-6 UNIQUE ENGINES

7-6.1 FUEL CELLS (Refs. 20, 21)

The fuel cell may be defined as "an electrochemical device in which part of the energy derived from a chemical reaction, maintained by a continuous supply of chemical reactants, is converted to electrical energy" (Ref. 20).

The fuel cells convert chemical energy directly into electrical energy, as opposed to conventional methods of electrical power generation in which the chemical energy is transformed into heat, which, in turn, is transformed into mechanical energy, and, finally, into electrical energy. The elimination of the heat engine, by the fuel cell process, avoids the Carnot cycle efficiency limitation imposed by thermodynamic considerations.

A brief explanation of some of the terms and efficiencies applying to fuel cells follows:

a. A *fuel cell power pack* (or battery) consists of one or more fuel cells and of the auxiliary equipment necessary for the production of electrical energy. Auxiliary equipment includes pumps, heat exchangers, storage reservoirs, chemical regenerators, and control equipment.

b. The *primary fuel* is the material that is continuously supplied to the fuel cell system to maintain operation.

c. The *current density* of a fuel cell is the electrical current developed per unit area of the electrode.

d. The *elcctrode area* is the projected area of the electrodes in contact with the electrolyte.

e. The (a) *power per unit weight* and the (b) *power per unit volume* of a fuel cell power pack is the ratio of the electrical power delivered, under the stated operating conditions, to the weight (a) or volume (b) of the fuel cell system when the primary fuel and its control and storage apparatus are not considered (e.g., watts per pound or per cubic foot).

f. The (a) *encrgy per unit weight* and the (b) *energy per unit volume* of a fuel cell power pack is the ratio of the total electrical energy developed, under the stated operating conditions, to the weight (a) or volume (b) of the cell power pack, for the rated fuel capacity (e.g., watt-hour per pound or per cubic foot).

g. The *ideal efficiency* of a fuel cell is the ratio, expressed in percent, of the free energy (electrical energy) to the heat of reaction of the cell reactants, when the process is isothermal and reversible at constant pressure.

h. The *voltage efficiency* of a fuel cell is the ratio, expressed in percent, of the terminal voltage of the cell, under the stated operating conditions, to the calculated reversible voltage under the same conditions.

i. The *fuel utilization efficiency* of a fuel cell is the ratio, expressed in percent, of the quantity of electricity (e.g., Coulombs) through the cell terminals, under the stated operating conditions, to the calculated quantity of electricity as determined by the rate of consumption of the reactants.

j. The *energy efficiency* of a fuel cell is the ratio, expressed in percent, of the usable electrical energy at the cell terminals, under the stated operating conditions, to the maximum available energy. Energy efficiency is equal to the product of the voltage and fuel utilization efficiencies.

k. The *overall efficiency* of a fuel cell power pack is the percentage of the heat of reaction of the primary fuel that is converted to useful electrical energy at the cell terminals under the stated operating conditions.

Fuel cells may be classified into five main types: (a) hydrogen-oxygen, (b) molten salt electrolyte, (c) redox, (d) regenerative, and (e) consumable electrode. The basic components of a fuel cell are: the fuels, the electrodes, and the electrolyte.

In all types of fuel cells, at least two fuels are required. These are known as oxidants or reductants, i.e., oxidizers or reducers. In some cells, e.g., the hydrogen-oxygen cell, the required chemical fuels are supplied directly and continuously, while in other types of fuel cells, the chemical fuels are produced within the cell by a continuous external energy source, i.e., heat. Some of the chemical fuels currently used are: Oxidants: oxygen, air, chlorine; Reductants: hydrogen, carbon monoxide, methane, ethane, natural gas, coal, formaldehyde, alcohol, zinc, magnesium. The fuel cell functions by chemical formation of a neutral product from charged ions, e.g., electrons are formed when hydrogen and hydroxyl ions combine.

Because the voltage of each cell is about one volt, series and parallel connections may be required to produce the required voltages. Table 7-4 shows the performance characteristics of typical fuel cells.

As compared with other sources of power, some of the advantages and disadvantages of fuel cells are:

Advantages:

a. The overall efficiency is very high; efficiencies of 80% have been attained in experimental systems.

b. Fuel cells can be operated on a variety of fuels; in some cases, low-grade fuels are used.

c. There are no moving mechanical parts; vibration, noise, or objectional waste products and maintenance are negligible.

TABLE 7-4 PERFORMANCE OF TYPICAL CELLS

Type of Cell	OPERATING CONDITIONS		POWER OUTPUT					
	Temperature, °C	Pressure, atm	Voltage, v	Current Density, amp/sq ft	Per Unit Weight, watt/lb	Per Unit Volume, watt/cu ft	Life	Efficiency, %
Hydrogen-oxygen	50- 60	1- 5	0.95-0.6	90-450	2-3	200-300	1000 hr	65-70
Hydrogen-oxygen	200-240	40-53	1.0-0.6	30-1000	15-20	2000	1500 hr	80 max
Molten Salt	700-800	1	0.96-0.54	30-1000	...	100-1000	several hr	65 max
Molten Salt	500-730	1	0.84-0.5	10-70	...	800	6 months	70 max
Redox	80- 85	1	0.8	40	8	300-460	1 week	80 max
Redox	20- 90	1	0.62-0.29	9-46	...	...	...	60 max
Heat Regenerative	450*	near 1	0.4 -0.36	150 max	2	90	...	10-12
Consumable Electrode	45	1	1.76-1.21	25-150	...	2.8	1 year	...
Consumable Electrode	Room	1	1.95	15	...	...	12 hr	...

* Regeneration temperature—850°C.

REFERENCE: "Status Report: Army Rounds Up Fuel-Cell Progress" *Machine Design* February 18, 1960.

d. The energy-to-weight and energy-to-volume ratios are extremely high for operating periods of long duration (150 hours).

e. The cells operate in a wide range of environmental conditions.

Disadvantages:

a. Power output is limited to direct current.

b. A large amount of auxiliary equipment is required for some systems.

c. Large quantities of energy-per-unit-weight cannot be generated in a short time (10 hours). Silver-zinc batteries are much superior in this respect.

7-6.2 RESPONSIVE ENGINES

The speed-torque relationship of the conventional reciprocating piston engine is not well suited to vehicle requirements. As shown in Fig. 7-13, the reciprocating engine does not develop appreciable torque in the lower speed ranges and develops maximum torque at only one engine speed.

The torque requirements (at the wheel or track) of a vehicle vary with the performance requirements and the environment of operation. In general, maximum torque should be available to put the vehicle in motion, to accelerate it at lower vehicle speeds, and to negotiate grades and obstacles. To match the torque developed by the engine to the torque requirements at the road wheel, a transmission or torque multiplier is used. It is clear that the complexity of the transmission system increases as the torque deficiency of the engine increases (see Chapter 8).

A current research program is aimed at the development of responsive engines. A fully responsive engine would develop the exact combination of speed and torque required at the road without an intermediate transmission.

SECTION III AUXILIARY COMPONENTS

7-7 FUEL SYSTEM

All components of the fuel system must be designed to operate satisfactorily with the applicable fuels: Group 1—gasoline engines: gasoline shall conform to Specification MIL-G-3056 or to 80-octane grade of Specification VV-M-561; Group 2—Diesel engines: fuel shall conform to Specification MIL-F-16884 or VV-F-800, under the applicable conditions stated in Par. 7-14.

The fuel system (MIL-E-13129(ORD)) shall include, if applicable, a fuel pump, fuel strainer and filter unit, a carburetor or fuel injection system, and necessary manifolding, fuel piping, and fittings. A suitable priming system to facilitate starting the engine at ambient temperatures down to −65°F shall be incorporated. If the application of heat is necessary to maintain proper engine operation, the heat applied to the fuel system shall not raise the temperature of the fuel to the flash point. The fuel system shall be so designed and installed as to eliminate vapor lock at elevated temperatures, to the maximum practical extent.

7-7.1 FUEL TANKS

The fuel tank or tanks of military vehicles may be positioned at any convenient location. Factors such as vulnerability, effective space utilization, and convenience of servicing should be considered when the type and location of fuel tanks are specified. Since all military automotive engines are equipped with fuel pumps, the elevation of the tanks may be less than that of the carburetor. The advantage of gravity feed to the carburetor in the event of a fuel pump failure, is usually offset by the high fuel tank location required.

A prime consideration in fuel tank construction is that it be resistant to corrosion. A variety of materials are used to accomplish this purpose, including fiber glass and plastics; although the most common is sheet steel having a corrosion resistant plating. The inlet for filling the tank must allow a flow rate of 50 gpm. Provisions should be made to prevent spilled fuel from flowing into the engine compartment, and the inlet should have ballistic protection. The outlet pipe for the tank should be located so that sediment from the bottom of the tank is not drawn into the fuel line. A depth of at least ½ inch should be allowed for the sediment trap. A drain plug should be provided for draining and cleaning purposes.

Fuel tanks normally contain baffles to prevent the fuel from surging or splashing when the vehicle is in motion and to strengthen the tank. The baffles are notched or perforated to allow a free flow of fuel through the compartments. Fuel tanks must be vented. Venting ports may be incorporated in the inlet cap or may be provided through a special line from the top of the tank to a common vent chamber for chasis components.

Self-sealing tanks are used on some vehicles. On these, rubber latex is placed between inner and outer walls. The latex tends to seal small holes caused by bullets or fragments. Fuel tanks normally contain electrical sensing units, which, with the proper gage, indicate the fuel level in the tank. Many military vehicles have more than one tank. These tanks are interconnected and each has a fuel shutoff valve.

7-7.2 FUEL FILTERS

At least one fuel filter should be provided in the fuel system. The fuel filter should be placed between the fuel tank and the carburetor, and located to provide ease of maintenance and to minimize fire hazard when servicing.

The sediment bowl type of filter is often used for military engines. In this filter, the fuel enters the sealed bowl or sediment chamber and passes out through a screening element suspended in the bowl. Gravitational force causes dirt and water to settle in the bottom of the chamber. One type uses a series of laminated closely spaced disks as the screening element; others use mesh or a porous solid as the element.

7-7.3 FUEL PUMPS

Fuel pumps for internal combustion engines can be classified as positive-displacement pumps or nonpositive-displacement pumps.

The pump most frequently used on gasoline engines is of the spring-loaded diaphragm nonpositive type. These pumps are commonly actuated mechanically by the engine. In addition to mechanically actuated pumps, electromagnetic diaphragm-type and electromagnetic plunger-type fuel pumps are available. The electromagnetic fuel pump can be mounted in or near the main fuel tank. This arrangement minimizes vapor lock in the fuel lines.

A second type of electric pump is the rotary pump. Rotary fuel pumps may be of the gear type, impeller type, or sliding-vane type. Rotary fuel pumps having externally powered shafts are difficult to seal against gasoline leakage, and, therefore, are not well suited to spark-ignition engines.

Diesel engines normally use fuel pumps to supply the injection pumps. Diesel fuel supply pumps are normally of the gear, plunger, or vane types.

All military engine fuel pumps must be waterproof, easily accessible, and mounted so that the outlet line can be disconnected and a pressure gage installed to check operating pressure. Diaphragms, if employed, must be capable of withstanding the effects of aromatic-type fuels.

7-7.4 CARBURETORS

The purpose of a carburetor is to meter and atomize the fuel and to mix it with the air inducted into the engine. Engines require different air-fuel ratios under various conditions of load, e.g., a rela-

TABLE 7-5 STANDARD CARBURETORS FOR MILITARY ENGINES

Part No.	Type	Venturi Tubes	Venturi Size	Use	Application
7966561	Dn-Draft, w/o Governor	1	1.124	Main Engine	17 HP Engine
7372509	Dn-Draft, w/o Governor	1	1-1/32	Main Engine	60 HP Engine
8329774	Dn-Draft, w/o Governor	1	1-1/4	Main Engine	70 HP Engine
7001053	Dn-Draft, w/ Governor	1	1-11/32	Main Engine	94 HP Engine
7411781	Dn-Draft, w/ Governor	2	1-5/32	Main Engine	145 HP Engine
7368643	Dn-Draft, w/ Governor	2	1-5/32	Main Engine	146 HP Engine
7368717	Dn-Draft, w/ Governor (For slave unit)	2	1-5/32	Main Engine	146 HP Engine (Vehicle w/ winch)
7375469	Dn-Draft, w/ Governor	2	1-1/2	Main Engine	224 HP Engine
8327282	Dn-Draft, w/ Governor	2	1-1/2	Main Engine	224 HP Engine (Vehicle w/ winch)
8333232	Dn-Draft, w/ Governor	2	1-1/2	Main Engine	297 HP Engine
7416587	Dn-Draft	2	1-1/2	Main Engine	500 HP Engine
7521189	Dn-Draft	2	1-27/32	Main Engine	810 HP Engine
8680542	Dn-Draft	2	1-1/2	Main Engine	810 HP Engine

tively rich mixture is required for idling and low loads, medium-load operation is satisfactory with a somewhat leaner mixture, and full-load operation (3/4- to full-throttle) demands a mixture falling between the other two. Table 7-5 lists standard carburetors for military engines.

Carburetors for military vehicles must be (a) sealed against dust, (b) properly shielded for underwater operation, (c) arranged so that fuel and air adjustments are not required for ordinary day-to-day operation, and (d) designed to minimize the possibility of flooding. In addition, carburetors should be arranged to provide optimum fuel vaporization during cold-weather operation by utilization of exhaust-gas heat. A choke or priming pump must be provided to facilitate starting the engine.

7-7.5 SUPERCHARGERS (Refs. 13, 26, 27)

A supercharger is a compressor or blower used to increase the density of the charge supplied to the reciprocating engine. Superchargers for automotive use may be classified as positive displacement, centrifugal, or axial flow. Within the positive displacement category are the Roots type and the rotary sliding vane type. The types most commonly used are Roots blowers and centrifugal blowers.

Like all positive displacement pumps, the delivery rate of a Roots blower is proportional to speed. The leakage is approximately proportional to the square root of the pressure difference and is independent of speed. This type of supercharger is desirable for variable-speed engines where high torque is required at various speeds.

The delivery rate of a centrifugal supercharger is proportional to the square of the impeller speed. When maximum impeller speeds and maximum pressures are selected for a given installation, the supercharge ratio is very small at low impeller speeds. Hence, if a fixed speed ratio between engine and impeller exists, the increase in charge density will be very slight. Centrifugal superchargers are well suited to aircraft engines and to automotive engines where a power increase at high engine speeds is desired.

Superchargers may be installed singly, in parallel, or in stages (series), with cooling between stages or at the outlets. The carburetor may be located before or after the supercharger or between the stages.

Superchargers may be driven by mechanical connection with the output shaft (directly or through gears) or by exhaust gas turbines. The exhaust gas-driven units are called turbochargers and are normally used with centrifugal blowers. All of the energy of the exhaust stream cannot be normally used, therefore, a waste gate is provided that allows some of the gases to bypass the turbine when necessary.

Spark-ignition engines having maximum compression ratios for available fuels cannot be supercharged without the introduction of combustion knock. Satisfactory supercharging of these engines requires a decrease in compression ratio, an enriched mixture, increased cylinder cooling, or an increase in fuel octane rating. Supercharging a compression-ignition engine increases the maximum cylinder pressure. If this pressure is at the upper limit for the engine, supercharging must be accompanied by lowering the compression ratio or by changing the fuel-injection timing.

7-7.6 FUEL INJECTORS

Fuel injection for military vehicle engines (Ref. 22) is limited to solid fuel injection. Fuel injection can be used for either compression-ignition or spark-ignition engines.

Compression-ignition engines normally are equipped with direct fuel injection systems, i.e., the fuel is injected directly into the combustion chamber rather than into the manifold or intake port areas. The basic components of this system are: a high pressure pump, fuel lines, and nozzles. These components meter the fuel, control the beginning and duration of injection, atomize the fuel, and distribute it within the combustion chamber. Some of these components are combined in the unit injector system. Descriptions of injection systems for compression-ignition engines can be found in TM 9-8000, *Principles of Automotive Vehicles.*

Although fuel injection is widely used on Diesel engines in all fields, its application to spark-ignition engines is limited. Neglecting injection-type carburetors, spark-ignition systems inject fuel into: (a) the supercharger, (b) the intake manifold, (c) the intake valve ports, or (d) the combustion chambers. The proper injection of fuel in spark-ignition engines is more difficult than in compression-ignition engines, owing to the fact that the former system must meet the definite air-fuel ratio requirements of the engine.

The three methods most frequently considered for spark-ignition engine injection systems are: (a) intermittent port injection, (b) continuous port injection, and (c) direct cylinder injection during the intake stroke.

Direct injection during the intake stroke is highly desirable for four-stroke cycle engines. The turbulence and long mixing time result in complete fuel evaporation. The evaporation process results in lowered temperatures and pressures of the charge and an increase in the mass of fluid inducted into the cylinder. The gain in output due to the cylinder vaporization process can be as high as 10% over a similar engine with manifold vaporization. Other factors contributing to an increased volumetric efficiency are the elimination of carburetor pressure losses and the elimination of manifold heating. The improved volumetric efficiency of the direct-injected engine results in lower brake specific fuel consumption at maximum power and a lower minimum value. The major disadvantages of direct injection are that the injector nozzles are exposed to high temperatures and pressures and that careful nozzle design is required to avoid crankcase dilution and high cylinder wear when the engine is cold-started.

Intermittent port injection consists of fuel discharge into the intake port of each cylinder during the intake process only. With this system, the disadvantages of direct injection are avoided. The gain in volumetric efficiency (over a similar, carbureted engine) is less than the direct injection gain; however, the difference is in the order of 3%.

A continuous port injection system injects a metered amount of fuel into the intake valve area. Injection continues during the time the valve is closed. Much of the fuel will evaporate in the port rather than in the cylinder. The gain in volumetric efficiency is less than for the other injection systems. Furthermore, provisions must be made to prevent or minimize the tendency for the vapor of a given cylinder to pass into an adjoining cylinder.

TABLE 7-6 STANDARD AIR CLEANERS FOR MILITARY VEHICLE ENGINES

Part No.	Flow Rate, cfm	Type	Application
7355937	29	Oil Bath	1 Pt. (Aux. Generator)
7966572	32	Oil Bath	1 Qt. (Wheeled Vehicles)
7047962	115	Oil Bath	1 Qt. (Wheeled Vehicles)
8329642	115	Oil Bath	1 Qt. (Wheeled Vehicles)
6582072	140	Oil Bath	1 Qt. (Wheeled Vehicles)
7041784	245	Oil Bath	1 Qt. (Wheeled Vehicles)
7539101	245	Oil Bath	1 Qt. (Wheeled Vehicles)
8331801	245	Oil Bath	1 Qt. (Wheeled Vehicles)
8340994	245	Oil Bath	1 Qt. (Wheeled Vehicles)
7971317	350	Oil Bath	1-1/2 Qt. (Wheeled Vehicles)
7405725	410	Oil Bath	2 Qt. (Track Vehicles)
7522668	410	Oil Bath	2 Qt. (Track Vehicles)
7969648	410	Oil Bath	2 Qt. (Track Vehicles)
7982615	410	Oil Bath	2 Qt. (Track Vehicles)
7982959	410	Oil Bath	2 Qt. (Track Vehicles)
8338000	410	Oil Bath	2 Qt. (Track Vehicles)
8379250	410	Oil Bath	2 Qt. (Track Vehicles)
8332775	550	Oil Bath	2 Qt. (Wheeled Vehicles)
7987761	570	Oil Bath	2-1/2 Qt. (Track Vehicles)
8340033	570	Oil Bath	2-1/2 Qt. (Track Vehicles)
10865550	—	Dry Type	(Alum Const) M113
8376527	570	Oil Bath	2-1/2 Qt. (Track Vehicles)
7351787	615	Oil Bath	3 Qt. (Track Vehicles), Right Assembly
7351788	615	Oil Bath	3 Qt. (Track Vehicles), Left Assembly
7364751	615	Oil Bath	3 Qt. (Track Vehicles), Left Assembly
7364752	615	Oil Bath	3 Qt. (Track Vehicles), Right Assembly
8382383	615	Oil Bath	3 Qt. (Track Vehicles), Right Assembly
8382384	615	Oil Bath	3 Qt. (Track Vehicles), Left Assembly

A comparison between fuel injection and carburetion follows. In addition to the advantages listed above for injection, the following apply: Fuel injection results in more uniform distribution of fuel between the cylinders of multicylinder engines if more than one cylinder depends on one carburetor. The more uniform air-fuel ratios between cylinders eliminate combustion knock due to "lean running" cylinders. The major disadvantages of fuel injection when compared to carburetion are increased cost, increased complexity, and increased weight.

7-7.7 AIR CLEANERS AND PRECLEANERS

The air induction system of military engines includes air cleaners, manifolds, ducts from cleaners to carburetors, air heating devices, and intake grills. Table 7-6 lists standard air cleaner assemblies for military vehicles.

The importance of high-efficiency air cleaners for military engines cannot be overemphasized. Tests have shown that engine life is a function of air cleaner efficiency. A recent report (Ref. 23) on air cleaners for military vehicles includes a case in which engine life varied from 45 hr with an air cleaner efficiency of 95% to 180 hr with an air cleaner efficiency of 99%. Information and knowledge of the effect of dust particle sizes and concentrations is somewhat limited. However, experience has shown that for vehicles operating in extremely dusty environments, e.g., deserts, a minimum cleaner efficiency approaching 99% is desirable.

Specifications covering the general requirements of air cleaners for internal combustion engines are given in MIL-A-13488. Some of the factors covered are:

a. Minimum Air Flow Restriction

Any current air cleaner experiences a pressure drop across the unit in use. This pressure drop must not exceed a specified value.

b. Dust Capacity

This term refers to the change in restriction (pressure drop) with time, under existing operating conditions. Quantitative values for airflow, dust size, and concentration, and allowable pressure drop are specified for military engine air cleaners.

c. Air Capacity and Angle Operation

Air cleaners for military vehicles must operate satisfactorily when they are tipped from their normal positions. Compression-ignition engine air cleaners must be able to operate at 150% of rated flow when tipped at an angle of 30° from the vertical in any direction. Spark-ignition engine air cleaners must be able to operate at 110% of rated flow under similar tipped conditions. Oil bath cleaners operated under these conditions must not lose oil for a minimum period of 15 min.

In addition to these requirements, air cleaners must be able to withstand shock and backfire, be free of leakage, readily serviceable, and suitable for the attachment of deep-water fording extensions.

Some of the basic types of air cleaners for internal combustion engines are described in the following paragraphs.

The oil bath air cleaner depends upon inertial forces and an oil-soaked gauze to clean the air. The air enters the cleaner and flows downward vertically until it strikes the oil in the reservoir. The air then reverses its direction of flow 180° and passes through the gauze element before flowing into the air inlet of the engine. Oil bath filters have a maximum efficiency of about 98%, high dust capacity, and a low, gradually increasing restriction. They are relatively large for a given capacity.

The felt element dry-type air-cleaner uses a corrugated, cylindrical felt element. All of the air passes through the element before entering the engine air inlet. This filter has a very high efficiency of 99.7% and a low initial (when clean) restriction. It is about the same size as an oil bath air cleaner of the same capacity, but has the disadvantage of low dust capacity, i.e., the pressure drop across the filter increases rapidly when in use.

Paper element air cleaners are similar in design to felt element cleaners. The paper element is corrugated to provide maximum surface area for the chosen outside diameter. Like the felt element cleaner, the paper element cleaner is very efficient (99.8%) and has a low initial restriction, however, the dust capacity is low and the element is vulnerable and nonserviceable. There is no size advantage over oil bath cleaners.

The water bath air cleaner depends upon the reverse flow of air and cascading water to remove dirt from air. This bath cleaner has a low initial restriction and good dust capacity. Its disadvantages are complexity, low efficiency, large size for a given capacity, e.g., about four times as large as an oil bath cleaner of the same flow capacity, and the water supply required.

The recirculating oil bath air cleaner is generally similar to the standard oil bath unit in operating principle, except that the stationary oil pool in the standard unit is replaced by an oil spray. This type of cleaner has a high efficiency, a long service life, and a constant low restriction. It is slightly larger than an oil bath cleaner and requires an oil storage tank and powered pump. Furthermore, the oil pull-over is considerable when the dust capacity of the cleaner is reached.

The electrostatic air cleaner removes dust from the air passing through it by first ionizing the dirt particles and then attracting the charged particles to an oppositely charged grid. Electrostatic cleaners have very high efficiency, but are very large and require an external power source.

The oil-wetted flocked-screen air cleaner directs the airflow through flocked screening, which may be installed in several stages with varying densities. The screening material is lightly coated with oil to increase its ability to retain dust particles. Oil-wetted filters have low initial restriction and high efficiency. They have a low dust capacity and a very short service life.

The inertia air cleaner utilizes the principle of conservation of linear momentum to deposit dust particles when the air stream is diverted. Inertia type cleaners are very compact and can be designed so that they never require servicing, i.e., they may have a dust bleed-off. Their restriction is very low and their dust capacity is very high.

The disadvantage of inertia cleaners is their low efficiency (about 83% maximum).

Precleaners are devices used in conjunction with standard air cleaners to increase the service life of the latter. Development work shows that systems using self-cleaning inertia-type precleaners in series with dry-type felt element cleaners have considerable merit. Although the service life is appreciably increased when felt element cleaners are used in conjunction with precleaners, a self-cleaning dry element would be highly desirable.

A problem encountered during desert testing of air cleaning systems was the induction of large amounts of dust into the engine via the clean air ducting. Small cracks in flexible metal tubing, leakage of rubber hoses, and leakage at clamps have resulted in engine failures during actual desert tests. A further problem concerns the high degree of vulnerability or rubber hoses used in the ducting system.

7-7.8 INTAKE MANIFOLDS

The purpose of the intake mainfold is to distribute the air or air-fuel mixture uniformily to each of the cylinders and to assist in the vaporization of the fuel. The problems of manifold design are most severe when a large number of cylinders are charged through a single carburetor. Careful design is required to prevent unequal flow, localized condensation of fuel, and power losses due to overheating of the charge.

The downdraft manifold is most frequently employed with current automotive engines and is commonly designed for a mean gas velocity of 50 fps in the riser between the carburetor and the main branch of the mainfold, at an average piston speed of 708 fpm. Maximum mean gas velocity is about 250 fps at an average piston speed of 3500 fpm.

7-8 EXHAUST SYSTEM

7-8.1 EXHAUST MANIFOLDS

The purpose of the exhaust manifold is to collect and carry the products of combustion from the cylinders with a minimum back pressure. The pulsating flow in the manifold results in pressure variations that may improve scavenging or may result in increased back pressure for one or more cylinders. Since the frequency of the pressure variations, for a given engine and exhaust system, varies with engine speed, the exhaust manifold should be designed to operate efficiently over the expected speed range of the engine. The entire system, comprised of the manifold, exhaust pipe, muffler, and tailpipe, affects the efficiency of combustion gas evacuation from the cylinders.

7-8.2 MUFFLERS

Exhaust mufflers reduce engine noise by reducing the exhaust-gas velocity and either absorbing the sound waves or cancelling them by interference. Efficient mufflers reduce the exhaust noise to an acceptable level with a minimum back pressure. An increase of 1 psi in back pressure will cause a power loss of about 2½%. The volumetric capacity of a muffler should be from six to eight times the piston displacement.

Straight-through absorption mufflers depend mainly upon the ability of porous material, e.g., steel wool or Fiberglas, to absorb the sound waves. Interference mufflers break the waves into parts that are later brought together out-of-phase and hence tend to cancel each other.

The high operating temperature and the corrosive nature of the exhaust gases result in a relatively short service life for mufflers. Recently, ceramic-coated (inside and outside) mufflers have been developed for the purpose of extending muffler service life. The major difference between commercial and military exhaust systems is that the latter must have fordability features.

7-8.3 EXHAUST COOLERS AND DEFLECTORS

Exhaust coolers are devices designed to reduce the temperature of the combustion gases before discharge from the vehicle and thus to minimize the effectiveness of heat detecting devices. The most common exhaust coolers use the principle of air-bleed cooling. The exhaust gases are diluted by mixing them with atmospheric air before discharge.

Exhaust deflectors are used to minimize "torching" of the exhaust gases. The torching effect is highly undesirable because it provides direct visual detection.

7-9 COOLING SYSTEM

The function of the cooling system is to maintain the various engine components at temperature levels conducive to long life and proper functioning. For example, cylinders must be cooled and maintained at a specified temperature, to prevent thermal stress failure, to maintain a film of lubricant on the walls, and to ensure proper clearances between moving parts. The pistons, cylinder heads, and valves are cooled to prevent thermal stress failure, to maintain proper clearances, and to prevent combustion knock. Bearings must be kept at a suitable temperature and lubricants must be cooled to maintain the desired viscosity. Conventional internal combustion engines use either liquid- or air-cooling systems which may be supplemented by oil-cooling systems.

The heat rejected to the cooling system of a reciprocating piston engine ranges from 15% to 20% of the energy input (heating value of the fuel) for large compression-ignition engines; for automotive engines at higher loads, the heat rejected to the cooling system ranges from 20% to 35% of the input, and may run as high as 40% at light loads. In terms of output, the heat loss to the cooling system ranges from 40% to 50% of the brake horsepower output at the flywheel for large compression-ignition engines, and from 100% to 150% of the brake output for automotive engines.

7-9.1 LIQUID COOLING

Water is normally used as the cooling medium in liquid-cooled engines, although ethylene glycol can be used when high jacket temperatures are desired.

Water-cooled military engines have integral water jackets around the cylinders and heads. The jacket and a radiator (heat exchanger) form a closed system. A pump increases circulation, and a thermostat normally facilitates engine warm-up for cold-engine starting.

Water jacket design must be carefully executed so that adequate cooling is maintained for the various components. For example, the exhaust valves are exposed to the high temperatures of combustion and the hot gases during exhaustion, while the inlet valves are cooled by the induction process. Therefore, exhaust valve cooling should be given priority over inlet valve cooling in engine design.

7-9.2 AIR COOLING

Air-cooled engines depend on a direct transfer of heat from the high-temperature components to the atmospheric air as opposed to liquid-cooled engines, which may be considered indirectly air-cooled. Factors such as the material, shape, and temperature of the cooled body, i.e., power plant and factors such as the thermal conductivity, viscosity, velocity, density, and specific heat of the cooling medium, enter into the heat transfer considerations.

For engines of the same power, the quantity of air (cu ft/min) required for cooling will vary inversely with engine temperature for a constant ambient temperature. Since the design operating temperature of an air-cooled engine is approximately three times as great as the design temperature of a water-cooled unit, theoretically, the air requirement of the former should be about one-third that of the latter. However, the more efficient water jacket-radiator system reduces the difference to approximately one-half.

Air-cooled engines normally possess a heat exchanger to cool the lubricating oil. The percentage of the total engine cooling accomplished in this manner may be considerable. In fact, without an oil cooler, many engines could not be operated continuously under military conditions.

7-9.3 COMPARISON OF LIQUID COOLING AND AIR COOLING

Current military engine specifications include both liquid- and air-cooled power plants.

The advantages of liquid-cooled engines are:

1. The vast majority of automotive engines built in the United States have been water-cooled.
2. The cost of liquid-cooled engines is somewhat lower than comparable air-cooled engines.
3. The one-piece construction of the cylinders and the crankcase is inherently rigid.
4. Liquid-cooled cylinders tend to operate at a more uniform temperature.
5. The lower operating temperatures of the liquid-cooled engine result in higher volumetric efficiencies and higher specific outputs for the liquid-cooled engines.

The advantages of air-cooled engines are:

1. The envelope (volume) for the combination engine and cooling system is less than that for liquid-cooled engine of the same output.
2. Air-cooling eliminates the need for coolant and antifreeze supply.
3. There is definite evidence (Ref. 25) that the quicker warm-up of air-cooled engines reduces cylinder bore wear.
4. The increased cost and complexity of individual removable cylinders is offset by the ease

TABLE 7-7 STANDARD 24 VOLT, DC STARTER ASSEMBLIES

Part No.	No Load		Rpm	Volts	Max. Amps	Torque ft-lb	Application
	Volts	Amps					
7355782	23.6	19.5	2400	20.8	168	20.0	1/4-Ton, M38 Series, 60-hp Engine
8329740	23.6	19.5	2400	20.8	168	20.0	1/4-Ton, M38 Series, 72-hp Engine
7355783	23.6	19.5	2400	20.8	168	20.0	3/4-Ton, M37 Series, 94-hp Engine
7762618	23.8	20	3700	19.8	200	20.0	2-1/2-Ton, M44 Series, 146-hp Engine
7350454	23.7	35	4100	6	212	19.5	2-1/2-Ton, M133 Series, 145-hp Engine
7389561	23.7	35	4000	6	200	19.5	5-Ton, M41 Series, 224-hp Engine
8360017	23.7	35	4100	6	212	19.5	10-Ton, M125 Series, 386-hp Engine
8360051	23.7	35	4100	6	212	19.5	Special-Purpose Vehicle (M59) 145-hp Engine
8360084	23.8	20	3700	19.8	200	20.0	Special-Purpose Vehicle (M50)
8365476	23.7	35	4000	6	200	19.5	Track Vehicle, 500-, 810-, 1020-hp Engine

TABLE 7-8 STANDARD MAGNETO ASSEMBLIES

Part No.	Description	Application
7372995	Single Ignition, High Tension Wound Armature, Shielded, 4 Pole	4-cyl. Engine
7974214	Single Ignition, High Tension Wound Armature, Shielded, 6 Pole (Optional with 7974215)	6- and 12-cyl. Engine
7974215	Single Ignition, High Tension Wound Armature, Shielded, 6 Pole (Optional with 7974214)	6- and 12-cyl. Engine
7402753	Single Ignition, High Tension, Shielded	Aux. Generator (All)

of replacement of these units in case of damage.

5. The higher cylinder wall temperature of air-cooled engines results in a slightly higher thermal efficiency.

7-9.4 VAPOR COOLING

Vapor- or evaporative-cooling systems are those in which a liquid coolant is evaporated by the heat of the surface to be cooled. In this system, the coolant is used more efficiently than in a conventional liquid-cooling system. The latent heat of vaporization is used to transfer heat from the engine. The vapor-cooling system has the advantage of being smaller than a conventional water-cooled system of the same cooling capacity. However, the vapor-cooling system is more complicated than the conventional system.

7-9.5 FANS, PUMPS, AND RADIATORS

The design of these units is well covered in the literature of fluid mechanics and heat transfer. Therefore, the following comments are brief and general.

Water-cooled engines normally employ axial fans mounted between the radiator and the engine. Flow is through the radiator and past the engine. For some vehicles, the fan is required for adequate cooling only during idling and low-speed operation.

Air-cooled engines normally employ centrifugal fans. These are used in conjunction with shrouds and baffles to ensure uniform and proper cooling of the various components. Military vehicles with air-cooled engines do not depend on motion-induced flow as a major source of cooling.

A centrifugal pump is normally included in the circulatory system of liquid-cooled engines. This pump is usually on a common shaft with the fan, and both are crankshaft-driven by means of a V-belt. Centrifugal pumps have several advantages when applied to cooling systems: (a) they have a high capacity-size ratio, (b) they are inexpensive, (c) they are not easily clogged, and (d) they per-

TABLE 7-9 STANDARD 24 VOLT, DC GENERATOR ASSEMBLIES

Part No.	Max. Output		Min. Speed Rated Load, rpm	Max. Opr. Speed, rpm	Type	Drive	Application
	Volts	Amps					
7355736	28.5	25.0	1700	6000	Enclosed	Belt	2-1/2-Ton, M44 and M133 Series, 146-hp Engines 5-Ton, M41 Series, 224-hp Engines 10-Ton, M125 Series, 386-Engines
7374750	28.5	25.0	1800	5800	Open	Belt	3/4-Ton, M37 Series, 92-hp Engines
7524310	28.5	25.0	1800	5800	Enclosed	Belt	2-1/2-Ton, M44 and M133 Series, 146-hp Engines 5-Ton, M41 Series, 224-hp Engines 10-Ton, M125 Series, 386-hp Engines
7539520	28.5	25.0	1700	6000	Enclosed	Belt	2-1/2-Ton, M44 and M133 Series, 146-hp Engines 5-Ton, M41 Series, 224-hp Engines
7966489	28.5	25.0	1700	6000	Enclosed	Belt	2-1/2-Ton, M44 and M133 Series, 146-hp Engines 5-Ton, M41 Series, 224-hp Engines 10-Ton, M125 Series, 386-hp Engines
8673350	28.5	25.0	1800	5800	Open	Belt	1/4-Ton, M38 Series, 60-hp Engines
8673353	28.5	25.0	1700	6000	Enclosed	Belt	1/4-Ton, M38 Series, 72-hp Engines
8724633	28.5	150	2500	6000	Enclosed	Spline	Track Vehicles, 500-hp Engines

mit a limited circulation by thermosiphon action when the engine is not running.

The radiators or heat exchangers used for automotive vehicles consist of an upper and lower tank, a filler inlet and cap, and a core. The core consists of a large number of passageways for the coolant. The passageways are finned to provide large surface areas to enhance the heat transfer process. The flow through the radiator is normally from the top to the bottom. Upper and lower radiator tanks are connected to the engine's water jacket.

7-4 ELECTRICAL SYSTEM

The electrical system for a spark-ignition engine has two major functions, ignition and starting. For a compression-ignition engine, starting is the only function. Starting may also be accomplished nonelectrically. Continuous operation and repeated engine starting require equipment to generate and to store electrical energy for the spark-ignition engine. Since the input speed and the load of the electric system varies considerably, regulation of the generator output is required. Control of such factors as ignition voltage is a matter of the design specifications of the various components. The various components of the starting system are described in Chapter 13, the present discussion will be limited to the ignition system for spark-ignition engines.

The basic components in a battery-ignition system are: the battery, ignition coil, breaker points, condenser, distributor rotor, spark advance mechanism, and sparkplugs. These items and their functions are described in detail in TM 9-8000, *Principles of Automotive Vehicles.* The battery-ignition system depends on a constant source of external energy; i.e., a battery or generator supplies the electrical power. The timing and distribution functions and operation of the breaker points are accomplished by mechanical power from the engine.

The magneto-ignition system differs from the battery-ignition system in that the magneto is a generator of electrical energy. Therefore, the required high voltages can be supplied at the sparkplug by applying mechanical power only to the magneto system. The magneto generator produces an alternating current in a low-tension coil (the armature winding). Breaker points and a condenser in the low-tension circuit interrupt the current in a manner similar to that of the battery-ignition distributor. The current interruption and reversal result in an induced current in the high-tension circuit. Distribution and timing of the voltages to the various sparkplugs are accomplished mechanically within the magneto assembly.

Since the primary voltage of the magneto depends on the speed of rotation of the armature and since the magneto is normally powered by the engine, starting difficulties may be experienced because of insufficient voltage at the sparkplugs. To alleviate the starting problem, a hand-cranked booster magneto, a vibrating coil, or an impulse starting system is used.

For a battery-ignition system, the secondary voltage increases as engine speed is increased from idle, reaches a maximum, and then drops with engine speed over most of the speed range. In the magneto system, the generated primary voltage increases with engine speed, i.e., magneto speed. Therefore, the voltage at the plugs increases with engine speed. High compression ratios, supercharging, and high engine speeds, are factors that emphasize the advantages of magneto- over battery-ignition systems. But, battery-ignition systems are less expensive than comparable magneto systems. The extensive use of electric starting motors that require a battery-generating system makes economical and convenient the use of the battery-ignition system. However, the high degree of reliability of magneto-ignition systems makes them ideal for aircraft and severe military applications, e.g., tanks. Table 7-7 and Table 7-8 list standard starter and magneto assemblies. Table 7-9 lists standard generator assemblies for military vehicles.

SECTION IV REQUIREMENTS OF THE POWER PLANT

7-11 POWER REQUIREMENTS

7-11.1 TRACTIVE POWER REQUIREMENTS

7-11.1.1 Basic Relationships

The power requirements of automotive vehicles assuming propulsion through ground reaction only (no thrust propulsion) can be analyzed on the basis of the maximum force attainable by the wheels or tracks acting on the ground, i.e., the gross tractive effort, H. When the motion resisting forces are subtracted from the gross tractive effort, the effective propelling force remains. Gross tractive effort for vehicles operating under cross country conditions, i.e., in soils and snows, is discussed in Chapter 5 of this handbook.

Classical analyses exist for vehicles operating on relatively firm ground, such as concrete roads, gravel roads, and ice. They utilize the concepts of road-adhesion coefficient and rolling resistance coefficient and are basically friction oriented; i.e., the propelling force is derived as a function of the friction, μ, developed between the wheel or track and the ground. The gross tractive effort or maximum transferable force, P_{max}, is expressed as

$$P_{max} = W\mu \tag{7-22}$$

where W is the gross vehicle weight in pounds and μ is the road-adhesion coefficient.

The effective tractive effort or drawbar pull, P_d, can be expressed as

$$P_d = P_{max} - R_r - R_g \tag{7-23}$$

where R_r is the rolling resistance in pounds and R_g is the horizontal grade resistance in pounds. The rolling resistance factor is a complex function of the characteristics of the wheel or track and the ground. Chapter 5 discusses the rolling resistance of wheels and tracks on various surfaces. The value R_r can be written Wf_r, where the factor f_r is called the coefficient of rolling resistance. The horizontal grade resistance can be expressed as

$$R_g = W \sin (\pm\Theta) \qquad (7\text{-}24)$$

where Θ is the angle of inclination of the grade in degrees. Summarizing, the total tractive effort or drawbar pull is

$$P_d = W\left(\mu - f_r - \sin (\pm \Theta)\right) \qquad (7\text{-}25)$$

A more thorough discussion of the resistances encountered by a vehicle in cross country operation is given in par. 5-2. That discussion considers also the internal characteristics of the soil, e.g., cohesion, granular friction, while the above discussion considers only the interface relationship between the wheel or track and the ground.

Engine torque figured on the basis of gross tractive effort must always equal or exceed the tractive demands of the wheels or tracks. In other words, the vehicle concerned should be "traction limited" rather than "power limited." The power requirements of military vehicles are partly based on the criterion of gross tractive effort; this criterion is used for calculating drawbar pull and gradeability only. For all current military vehicles, then, maximum vehicle speed and maximum acceleration are power limited.

Grade, G, is customarily expressed as a ratio in percent of vertical rise to horizontal distance traveled:

$$G = \frac{y}{x} (100) = 100 \tan \Theta \qquad (7\text{-}26)$$

In the field of civilian automotive design, where grades rarely exceed 12% (7°) and the steepest mountain roads known only reach 32% (18°) for short distances, common practice is to simplify Eqs. 7-24 and 7-25 by replacing sin ($\pm \Theta$) with G:

$$R_g = \frac{WG}{100} \qquad (7\text{-}27)$$

$$P_d = W\left(\mu - f_r - \frac{G}{100}\right) \qquad (7\text{-}28)$$

This simplification is based upon the assumption that, for small values of Θ, $\sin \Theta \approx \tan \Theta$. Equations 7-27 and 7-28 should be applied with caution to the design of military vehicles as they result in serious errors at steep slopes.

7-11.1.2 Torque-Weight Ratio

The longitudinal slope specifications for all military vehicles establish a required torque-weight ratio for a given vehicle. This ratio may be expressed as

$$\frac{T}{W} = \frac{\sin \Theta \, r}{R_g e} \qquad (7\text{-}29)$$

where

- T is the torque at the clutch, ft-lb
- Θ is the angle of inclination of the grade, deg
- r is the effective radius of the wheel, ft
- R_g is the overall ratio (speed of final drive wheel divided by speed of engine)
- e is the power train efficiency

The specification or selection of a rate, e.g., the velocity of the vehicle, determines the required power at the clutch.

The power-gross vehicle weight ratios of typical military vehicles can be utilized as a reference to estimate the performance potentiality of similar proposed vehicles. The value of the estimate is directly related to the degree of similarity of the vehicles under consideration. Factors such as engine type, transmission type, track of wheel type, and ground pressure enter into the performance pattern. Table 7-10 lists some current military vehicles and their power-weight characteristics. Several commercial vehicles are included for comparison.

7-11.1.3 Drawbar and Brake Horsepower

If the net tractive effort is assumed to be available to produce a specified drawbar pull, the horsepower required to produce this effort, $(hp)_d$, can be determined from the maximum desired vehicle speed:

$$(hp)_d = \frac{P_d V}{375} \qquad (7\text{-}30)$$

where P_d is the drawbar pull in pounds and V is the vehicle speed in miles per hour. The required brake horsepower at the clutch, $(bhp)_{cl}$, for the given conditions is

$$(bhp)_{cl} = \frac{(hp)_d}{e} \qquad (7\text{-}31)$$

where e is the overall power train efficiency expressed as a decimal.

TABLE 7-10 POWER-GROSS VEHICLE WEIGHT RATIOS FOR REPRESENTATIVE VEHICLES

VEHICLE	Gross Vehicle Weight, lb	Horsepower	Horsepower/Ton
Personnel Carriers			
AIV-M75	42,000	330	15.7
AIV-M59	42,000	260	12.3
AIV-M113	21,000	205	19.5
Civilian Automobile	3,662*	179*	97.81*
Transporters			
Army GOER	65,000	274	8.4
Dump Truck, Commercial	60,000 to 125,000	218 to 375	6.0 to 7.5
Dozers			
Crawler Tractors, Class 4, w/Dozer Equipment	35,000	135	7.7
New Combat Vehicles			
M60 Medium Tank	102,000	650	12.7
T195 Self-Prop.-Artillery	42,000	375	17.8
T235 Self-Prop.-Artillery	59,000	375	12.7
T95 Medium Tank	88,000	500	11.3

* Average values for 1960 American automobiles.

7-11.1.4 Speed

The maximum speed of military vehicles on hard, level terrain (ice not included) is power limited; i.e., the total motion resisting forces, rolling resistance and air resistance, primarily, never reach a level where they are equal to the gross tractive force that the vehicle-ground combination is capable of developing. The maximum speed for a given vehicle is a function of power available, motion resisting forces, and the power plant-power train speed ratio.

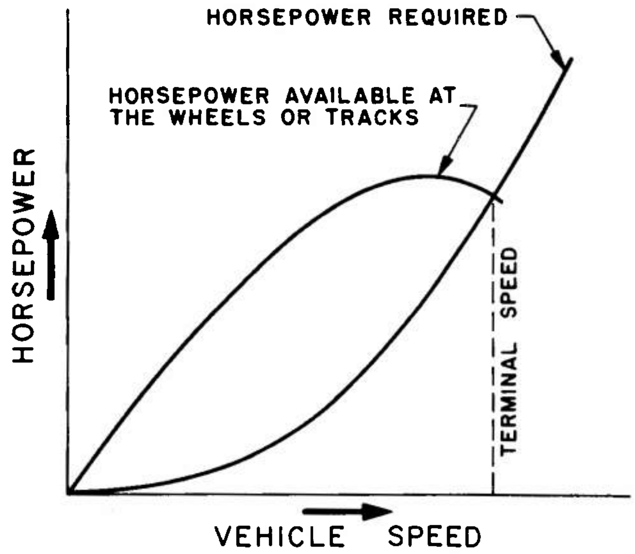


Figure 7-29. Terminal Speed Determination for Automotive Vehicles

Figure 7-29 shows typical curves of power available and power required for an automotive vehicle. Equations for computing the curves appear in par. 7-11.1.6. The intersection of the curves represents the terminal speed of the vehicle. The difference in ordinate between the two curves represents the power available for acceleration. The power-available curve shown represents the performance curve at one particular power plant-power train speed ratio for a particular power plant. For a wheeled vehicle, this ratio is the engine speed-wheel speed ratio; for a tracked vehicle, this ratio is the engine speed-speed of track driving sprocket ratio. If the power train ratio is changed, the power-available curve will shift to the right or left; i.e., the point of maximum horsepower will move to the right or left. Increasing the ratio will shift the maximum point to the left, resulting in decreased terminal speed with increased power for acceleration over the new speed range.

The maximum cruising speed of a military vehicle should be based on the manufacturer's

TABLE 7-11 TYPICAL MEAN PISTON SPEEDS OF AMERICAN ENGINES

Type	Piston Speed, fpm
Spark-Ignition Engines	
Passenger Automobile	2,800
Truck and Bus	2,500
Industrial and Tractor	2,000
Compression-Ignition Engines	
Automotive—two-stroke	1,800
Automotive—four-stroke	2,000
Large Marine and Stationary	1,400
Highly Supercharged Engines	2,500

recommended safe continuous operating speed of the power plant, or a power plant should be selected that bears a manufacturer's continuous duty rating adequate for the maximum vehicle cruising speed. The safe continuous operating speed of the power plant varies with the type and size of the unit. For conventional reciprocating piston engines, the operating speed can be related to average piston speed. Table 7-11 shows the average piston speed at maximum power rating for some typical, current power plants. For a reasonable service life, these piston speeds should not be exceeded during extended cruising of the vehicle.

The minimum vehicle speed for extended periods of time, under stated conditions, is determined by the minimum speed requirements of the engine. For reciprocating engines, the minimum crankshaft speed must be approximately 500 rpm for continuing operation. This minimum speed, which may range to 1000 rpm in supercharged engines, is termed the idling speed. In order to move the vehicle, the engine speed must be above idling speed. Another factor that may influence the minimum vehicle speed for extended periods of time is the engine cooling capacity. Since the cooling capacity of the engine may depend on motion-induced flow and cooling fan speed, low vehicle and engine speeds may result in insufficient cooling when ambient temperatures are high. Military engines normally have cooling systems of greater capacity than comparable civilian engines to compensate for the greater amount of extended low-speed operation of the military vehicle.

7-11.1.5 Acceleration

To accelerate an automotive vehicle, forces must be applied to change the linear velocity of the vehicle and the angular velocity of the rotating power and drive components.

The basic equation expressing the relationship between force, F, and acceleration, a, for a given mass is

$$F = \frac{W}{g} a \tag{7-32}$$

where W is the weight of the body in pounds and g is the gravitational constant in feet per second per second.

The force required at the wheels or tracks to accelerate a vehicle can be determined by totaling the gross weight of the vehicle and the equivalent weight of the rotating parts of the power-drive assembly and applying Eq. 7-32.

The equivalent weight, w, in pounds, of the rotating parts can be expressed as

$$w = A_r^2 A_t^2 J_e + J_a/r^2 \tag{7-33}$$

where

- A_r is the rear axle ratio (driven/driving)
- A_t is the transmission system ratio (driven/driving)
- J_e is the moment of inertia of all parts rotating with the crankshaft of the engine, lb-ft^2
- J_a is the moment of inertia of all parts rotating with the axle, lb-ft^2
- r is the effective radius of the wheel, ft

The force at the wheel necessary to accelerate the vehicle can be expressed as

$$F = \frac{(W + w)\, a}{g} \tag{7-34}$$

where W is the gross vehicle weight in pounds. The horsepower at the clutch (or equivalent point) required to accelerate the vehicle can be expressed as

$$(hp)_{cl} = \frac{(W + w)\, a\, V}{375\, g\, e} \tag{7-35}$$

where V is the speed of the vehicle in miles per hour and e is the drive train efficiency expressed as a decimal.

7-11.1.6 Air Resistance

A body moving through air encounters a motion resisting force. This force is termed air resistance or aerodynamic drag. The air resistance, R_a, lb, varies with the density of the air, the shape of the moving body, and the relative velocity of the body and the air

$$R_a = \frac{C_D \rho A V_R^2}{2g} \quad (7\text{-}36)$$

where

- ρ is the air density, lb/cu ft
- A is the projected area of the body in the direction of motion, sq ft
- V_R is the velocity of the body relative to the air, fps
- C_D is a dimensionless coefficient of air resistance, also called the drag coefficient
- g is the gravitational constant, ft/sec^2

The drag coefficient is influenced by three factors: the aerodynamic shape, the outside surfaces or "skin" of the body, and the airflow through the body (if any). Experimental values of C_D for various vehicle types are determined by wind-tunnel testing of models and actual vehicles. Some values of C_D for typical vehicles at SAE standard air conditions are given in par. 5-2.5. Another factor that influences the total drag coefficient is the density of the air. Therefore, the value of ρ should be determined for given barometric and temperature conditions.

The total horsepower at the clutch to overcome air resistance can be expressed as

$$(bhp)_{cl} = \frac{R_a V}{375\,e} \quad (7\text{-}37)$$

where V is the vehicle speed in miles per hour and e is the drive line efficiency expressed as a decimal.

7-11.1.7 Slope Requirements (Ref. 36)

Military tactical vehicles are required (MIL-E-13929) to operate satisfactorily on longitudinal slopes and on lateral (side) slopes. . . . (Ref. 36 and also par. 3-2.1). Some of the engine components that may require special attention, in order to meet these specifications, are: (a) the intake of the lubricating oil pump, (b) the carburetors, (c) the cooling system, and (d) the lubricating system, *per se*. The engine should be tested to see that it can provide sufficient power for the vehicle to climb the maximum longitudinal slope and for proper continuous operation in the four extreme tilted positions. The requirement for lateral slope operation does not influence the total power required for the vehicle.

The power required to move a vehicle up a given longitudinal slope at a given velocity can be calculated. If the calculated power required at the wheels or tracks is available, the "gradeability" of the vehicle will depend solely on the tractive effort. The power required to move a vehicle up a slope can be expressed as in terms of the brake horsepower at the wheels, $(bhp)_{wh}$, as

$$(bhp)_{wh} = \frac{GWV}{(100)\ (375)} + (hp)_r + (hp)_a \quad (7\text{-}38)$$

where

- G is the grade expressed as percent
- W is the gross vehicle weight, lb
- V is the vehicle velocity, mph
- $(hp)_r$ is the horsepower required to overcome rolling resistance
- $(hp)_a$ is the horsepower required to overcome air resistance

The last two terms in Eq. 7-38 can be rewritten as:

$$(hp)_r \quad \frac{R_r V}{375} \quad (7\text{-}39)$$

$$(hp)_a \quad \frac{R_a V}{375} \quad (7\text{-}40)$$

Methods for evaluating R_r and R_a are given in Chapter 5, Section I, paragraph 5-2. By substituting Eqs. 7-39 and 7-40 into Eq. 7-38, the following is obtained:

$$(bhp)_{wh} = \frac{V}{375}\left[\frac{GW}{100} + R_r + R_a\right] \quad (7\text{-}42)$$

Finally, brake horsepower at the clutch (or equivalent point) is

$$(bhp)_{cl} = \frac{V}{375\,e}\left[\frac{GW}{100} + R_r + R_a\right] \quad (7\text{-}42)$$

where e is the drive line efficiency expressed as a decimal. As slopes approach the maximum specified for the vehicle, the possible vehicle speed becomes very low; hence, the air resistance may be neglected.

Although no vehicle speeds are specified for slope climbing, Eq. 7-42 shows that, for a given vehicle, the velocity of the vehicle is inversely proportional to the percent grade. Another consideration is the reduction ratio between the engine and the wheel or track driving sprocket. The reduction ratio must permit the engine to develop its maximum rated output while the vehicle travels at or below the selected slope velocity.

7-11.1.8 Climatic Conditions (Ref. 36)

7-11.1.8.1 Temperature, Barometric Pressure, and Water Vapor Pressure

The power developed by reciprocating-piston or gas turbine engines varies with the temperature, the barometric pressure, and the water vapor pressure of the ambient atmosphere. The correction factors vary with the type of engine, e.g., normally aspirated versus supercharged, gasoline engines versus diesel engines, reciprocating engines versus turbine.

To compare the performance of engines, the observed output must be converted to some standard conditions. The Society of Automotive Engineers chose dry air at 60°F and a barometric pressure of 29.92 in. of Hg as standard conditions and a semi-empirical equation for correcting the output of naturally aspirated and blower-scavenged spark-ignition engines to these conditions. Military specification MIL-E-13929 utilizes this data and gives the correction equation in the form:

$$\text{Corrected value} = \text{observed value} \times \frac{29.92}{B\text{-}E}\sqrt{\frac{T}{520}} \tag{7-43}$$

where

- B is the corrected barometric pressure at test location, in. Hg
- E is the water vapor pressure, in. Hg
- T is the absolute temperature of the intake air, °R

Equation 7-43 can be used to correct the brake horsepower, output torque, and brake mean effective pressure. It shows how these characteristics decrease as: (a) barometric pressure decreases, (b) ambient temperature increases, and (c) water vapor pressure increases. Equation 7-43 should not be used, however, to correct the specific fuel consumption values because, presumably, the engine was supplied with the correct amount of fuel to burn the air that entered the engine.

Compression-ignition engines do not use all of the air present in their cylinders because of the short time available for combustion. Some compression-ignition engines begin to smoke with a relatively large amount of unused air in their cylinders, while more efficient engines successfully burn more of the air present. An equation for correcting the output of a compression-ignition engine for different atmospheric conditions would be similar to Eq. 7-43 but would have to be varied for each engine to account for the amounts of excess air present when smoke became visible in the exhaust. Military specification MIL-E-13929, however, directs that the performance of compression-ignition engines be corrected to standard conditions of dry air at 90°F and a sea level barometer of 30.212 in. Hg for purposes of evaluation by applying the same basic equation used for spark-ignition engines. The difference between the two equations stems from different standard conditions upon which the correction is based. Thus, the correction equation for naturally aspirated and blower-scavenged compression-ignition engines is given as:

$$\text{Corrected value} = \text{observed value} \times \frac{30.212}{B\text{-}E}\sqrt{\frac{T}{550}} \tag{7-44}$$

where the symbols are the same as for Eq. 7-43. The corrected atmospheric pressure for dry air, B-E, is usually only slightly different from the uncorrected barometric pressure, B, therefore, the correction for vapor pressure, E, can be neglected except in instances of extremely high humidity or temperature.

Gas turbine output varies directly with ambient temperature. Figure 7-26 shows power variation versus ambient temperature variations for a typical gasoline engine and a typical gas turbine. Both units vary because the mass flow rate of the inducted air varies directly with temperature. However, the turbine is more sensitive, with respect to power output, than the gasoline engine.

7-11.1.8.2 Dust and Dirt

The power output of an engine can be reduced by atmospheric dust and by loose dirt and fine gravel. A dust-clogged air cleaner or dirt-clogged induction air-intake grills will reduce the volumetric efficiency, hence, the power of an engine.

TABLE 7-12 VEHICLE PERFORMANCE EQUATIONS*

Forces, Torques and Horsepower necessary to:	At The Wheel			At The Clutch		
	Force, lb	Torque, lb-ft	Horsepower, lb-ft/min	Force, lb	Torque, lb-ft	Horsepower, lb-ft/min
1. Overcome air resistance	$K_1 A V^2$	$K_1 A V^2 r$	$\frac{K_1 A V^3}{375}$	$\frac{K_1 A V^2 r}{A_r A_t r_c e}$	$\frac{K_1 A V^2 r}{A_r A_t e}$	$\frac{K_1 A V^3}{375 e}$
2. Overcome rolling resistance	$K_2 W$	$K_2 Wr$	$\frac{K_2 WV}{375}$	$\frac{K_2 Wr}{A_r A_t r_c e}$	$\frac{K_2 Wr}{A_r A_t e}$	$\frac{K_2 WV}{375 e}$
3. Ascend grade	$W \sin \theta$	$Wr \sin \theta$	$\frac{Wr\ V \sin \theta}{375}$	$\frac{Wr \sin \theta}{A_r A_t r_c e}$	$\frac{Wr \sin \theta}{A_r A_t e}$	$\frac{Wr\ V \sin \theta}{375 e}$
4. Accelerate vehicle	$\frac{W+w}{g} a$	$\frac{W+w}{g} a r$	$\frac{(W+w) a V}{375 g}$	$\frac{(W+w) a r}{A_r A_t r_c eg}$	$\frac{(W+w) a r}{A_r A_t e g}$	$\frac{(W+w) a V}{375 e g}$

LEGEND:

A Frontal area, ft²

A_t Transmission ratio in a given gear (driven/driving)

A_r Rear axle ratio (driven/driving)

a Acceleration, ft/sec²

e Approximate driveline efficiency

Wheeled Vehicles

=0.90 for direct drive

=0.85 for overall ratio of 12

=0.80 for overall ratio of 20

Tracked Vehicles

=0.76 for high range full load

=0.72 for low range full load

g =32.2 ft/sec²

J_a Moment of inertia of all parts rotating with axle, lb-ft²

J_e Moment of inertia of all parts rotating with crankshaft, lb-ft²

K_1‡ Air resistance coefficient

=0.000156 for extremely streamlined shape

=0.00054 to 0.0009 for standard sedan automobiles

=0.00102 to 0.00114 for open convertible automobiles with flat windshields

=0.00055 to 0.00103 for trailers, van type (various shapes)

=0.00054 to 0.00112 for buses

=0.00096 to 0.00252 for trucks

=0.00156 to 0.00252 for tractor-trailer combinations

K_2 Rolling resistance coefficient, see Chapter 5, par. 5-2.

r Effective radius of wheel, ft

r_c Effective radius of clutch, ft

W Gross vehicle weight, lb

w Equivalent weight of rotating parts of driveline, $(A_r^2 \times A_t^2 \times J_e + J_a)/r^2$

V Speed relative to air, mph

θ Angle of grade, deg.

* Based on publications of the Society of Automotive Engineers.

‡ K_1 values based on publications of M. G. Bekker, W. Kamm, and S. F. Horner.

Clogged exhaust systems or grills will have the same effect on engine power. Blocked radiators can result in excessive engine temperatures with consequent loss of power. Finally, the cylinder wall and piston ring wear due to inducted dust will eventually cause a loss in compression pressure and will permit gas "blow-by." These conditions reduce the power output of an engine.

7-11.1.9 Power Losses and Efficiencies

The determination of the power plant requirements for any automotive vehicle must include an analysis of the power losses of the various propulsion power transmitting systems and of the engine accessory group. The engine accessory losses may be dropped from consideration if the net or output shaft horsepower of the engine is chosen as a reference value. Of course, if an engine is to power auxiliary equipment, the auxiliary power requirements must be considered in reference to the net power plant output requirements.

The overall drive train losses from the engine output shaft to the ground may be divided into transmission system losses and wheel and tire losses for wheeled vehicle or transmission system losses and track and suspension system losses for tracked vehicles. The transmission system losses occur between the engine and the axle (wheeled vehicle) or the engine and the drive sprocket (tracked vehicle).

The rolling resistance or resistance to propulsion of wheeled vehicles under various conditions

TABLE 7-13 SUMMARY OF VEHICLE TRANSMISSION SYSTEM EFFICIENCIES DURING FULL-THROTTLE OPERATION OVER PREPARED ROADS

Vehicle	Slope, %	Road Speed, mph	Engine Speed, rpm	Transmission Efficiency, %
M41A1 No. 806	0	6.6	....	76.0*
	20	5.9	2320	64.0
	30	4.9	2240	65.0
	40	3.3	2180	57.0
	50	2.6	2150	53.5
	60	1.7	2170	38.5
M48A1 No. 117	0	5.0	....	76.0*
	20	5.8	2275	72.1
	30	4.7	2220	76.0
	40	3.2	2180	66.7
	50	2.3	2150	57.1
	60	1.7	2165	48.3

* Maximum value: attained in low range of transmission.

is discussed in Chapter 5. Typical driveline efficiencies of wheeled vehicles with gear transmissions are included in Table 7-12, which summarizes vehicle performance equations for wheeled vehicles.

Information concerning power losses (efficiencies) of tanks or other tracked vehicles is relatively scarce. Reference 31 reports the results of extensive tests to determine the sprocket horsepower characteristics of several tanks under a variety of conditions. Naturally, the efficiency of any power transmitting system will vary with the type of mechanical system. Furthermore, the power losses may vary with parameters such as vehicle speed and throttle setting.

Table 7-13 is a summary of transmission efficiencies during operation over prepared roads and slopes, for two track-laying vehicles equipped with the following engines, transmission and suspension components:

Vehicle M41A1 No. 806

Tank, 76mm Gun; Engine: Model AOS-895-3; Transmission: Model CD-500-3; Track: T91E3 with Rubber Pads; Vehicle Weight: 47,500 lb.

Vehicle M48A1 No. 117

Tank, 90mm Gun; Engine: Model AV-1790-5B; Transmission: Model CD-850-4A; Track: T97E1; Vehicle Weight: 95,500 lb.

Figures 7-30 and 7-31 show power losses for the M48A1 No. 117 vehicle in low- and high-transmission ranges. Table 7-13 and Figs. 7-30 and 7-31 are presented as a guide only; for additional information on the method of determining efficiencies and power losses of drive train components, see Ref. 31.

7-11.2 NONTRACTIVE POWER REQUIREMENTS

The power plant of a military vehicle must first supply the power for tractive effort and then possibly for miscellaneous equipment operated from the main power plant. The miscellaneous or auxiliary equipment may include electrical generators, hydraulic pumps, winches, ventilating equipment, and compressors. Some of the electrical equipment is discussed in Chapter 13.

In vehicles having only one engine, although all of the auxiliary units may not be operated at the same time, a brake horsepower at the clutch (or equivalent location) based on maximum tractive effort requirements plus the auxiliary equipment requirements will provide a margin of safety during normal operation.

Tanks and other large military vehicles often depend on an auxiliary engine to supply nontractive power. Auxiliary engines charge the main batteries for starting the main plant and operate

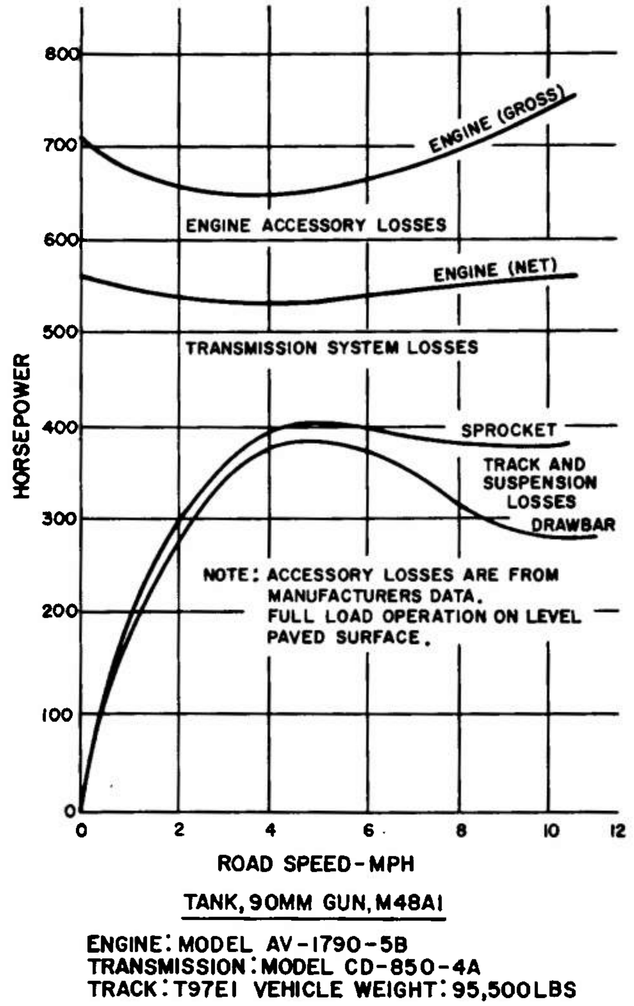


Figure 7-30. *Low-Range Power Losses*

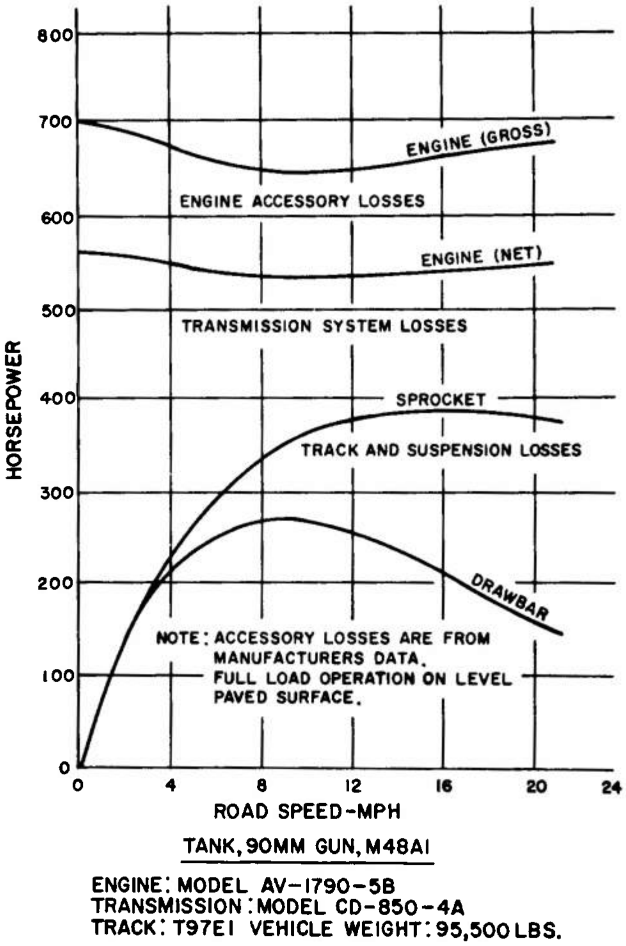


Figure 7-31. *High-Range Power Losses*

the auxiliary equipment when the main plant is stopped. Auxiliary engines supply power for radio equipment, traversing and elevating guns, radar equipment, lighting, heating, and ventilation. An independent auxiliary power system can be used to conserve fuel and to minimize noise and heat rejected while lying in concealment. The auxiliary power plant adds complexity to the vehicle and, theoretically, reduces the reliability of the vehicle. This accounts for the efforts devoted to the task of improving the main power plants on some intermediate-sized vehicles so that they can exclusively supply power with good economy.

7-12 OVERALL DIMENSIONS OF THE POWER PLANT

The overall dimensions of a proposed or existing power plant should be considered in reference to the following factors:

a. *Vehicle Silhouette*

In general, the silhouette of military vehicles should be as low and as small as practicable. Engines presenting relatively large vertical dimensions, such as vertical opposed-piston units, are not as desirable as some of the other types. The single-crank, horizontally opposed-piston engines and conventional inline engines operating in a horizontal position have minimum vertical dimensions.

b. *Engine Compartment Armor*

Since the total weight of the engine compartment armor increases as the compartment size increases, the engine envelope (cu ft) should be kept as small as practicable with a minimum number of projecting elements.

c. *Space Available*

The battle-day requirements impose a certain fuel tank capacity on vehicles. A minimum engine envelope will assist in achieving the required fuel tank capacity.

d. Size and Arrangement of Components

For reasons stated above, the size and arrangements of components should be considered carefully. When the component sizes are known, alternate arrangements within the hull or body should be studied, so that the space available is put to best usage.

e. Cooling Requirements

Air-cooling systems are usually more compact than liquid-cooling systems. However, the radiator for a liquid-cooled engine may be located some distance from the engine itself and may thus increase the efficiency of space utilization. (Some of the heat-transfer problems of these systems are discussed in par. 7-9).

f. Maintenance Requirements

Compact power plants may be more difficult to service while in the vehicle because of the minimum space between components, but the compact unit will generally be easier to remove from the vehicle for servicing.

g. Anti-IR and Noise Detection Measures

The total space requirements for a power plant must include the mufflers and the exhaust gas coolers.

h. Anti-CBR Measures

CBR agents are not expected to have any serious effects on the short-time operation of power plants. However, the special filters required for the passenger compartments will accentuate the need for more and more compact power plants.

7-13 LOCATION OF THE POWER PLANT

The location of the power plant within the vehicle should be considered in reference to the following factors:

a. Vehicle Silhouette

The location of the power plant should be based on the "lowest and smallest possible silhouette concept." To the extent practicable, the power plant envelope should be arranged with respect to other envelopes, e.g., the crew compartment, so that the vehicle silhouette is of minimum height.

b. Location of Crew

The power plant may be located in any position with respect to the crew compartment as long as adequate shielding is provided to reduce the engine noise and heat to an acceptable level. One disadvantage of locating the power plant at the opposite end of the vehicle from the final drive in tank-like vehicles is that the drive shaft may pass through the crew compartment.

c. Vehicle Stability

The stability and performance of an automotive vehicle are affected by the weight distribution on the wheels or track (Refs. 28 and 29). In addition to static load distribution, the dynamic load distribution under various operating conditions should be studied for proposed designs. The vehicle stability is also affected by the location of the center of gravity with respect to the ground; the location of the engine will, of course, affect this magnitude.

d. Location of Functional Equipment

The location of the power plant should be determined by the above considerations and the desirable locations of the functional equipment such as (a) armament, (b) ammunition, (c) cargo compartment, and (d) special purpose equipment.

7-14 INTENDED USE OF THE VEHICLE

The selection or design of automotive power plants will depend on the intended use of the vehicle. The following discussion covers some of the factors to be considered with reference to usage.

7-14.1 TYPE OF ENGINE

A *tactical engine,* or Type I engine (MIL-E-13929(ORD)), is defined as an engine used in a vehicle having exacting military characteristics and designed primarily for use by forces in the field engaged in combat or tactical operations, or to provide direct logistic support by service elements to such forces engaged in combat or tactical operation, or for the training of troops for these operations. An *administrative engine,* or Type II engine, is defined as an engine for a vehicle primarily of commercial design, intended for use at bases, depots, air stations, and other shore establishments where specialized use and road and climatic conditions are not such as to require a tactical vehicle.

The selection of specific power plants should be based on how well the various existing and proposed engine-transmission system assemblies meet the performance requirements. The characteristics of the engine and the power train must be considered since currently used power plants are both limited and deficient in their speed-torque relationship relative to vehicle performance requirements. Another factor of importance in the selection of military automotive engines is the type of fuel required; until multiple-fuel engines are standard equipment, this problem will exist.

7-14.2 OPERATIONAL REQUIREMENTS

Unless otherwise specified, military automotive engines must meet the following operational requirements (also see Chapter 3).

The engines, with accessories operating, shall be capable of satisfactory performance (MIL-E-13939(ORD)) in any ambient air temperature up to + 125°F, with full impact of solar radiation for at least 4 hr, and down to −25°F, when exposed at least three days without benefit of solar radiation. Engines shall be capable of satisfactory performance with the aid of approved winterization equipment, in ambient air temperatures down to −65°F. The engines shall not be damaged when stored in any ambient air temperature from +160°F, for periods of 4 hr per day down to −80° F for periods of at least 3-days duration. Unless otherwise specified, engines shall start successfully in ambient air temperatures down to −25°F without cold-starting aids. With the aid of approved winterization equipment, Type I engines shall start successfully at all temperatures down to −65°F.

Unless otherwise specified, engines shall be capable of satisfactory performance at any elevation from sea level to 12,000 ft. Engines shall be capable of developing and maintaining the rated net continuous horsepower and speed specified in the detail specification, at an elevation of 5,000 ft and an ambient air temperature of 107°F.

Figure 7-32 shows a chart of environmental envelopes of barometric pressures and ambient temperatures within which military engines must operate. The solidly outlined envelope represents the basic operational limits, while the envelope outlined in broken lines represents the possible extreme limits. The environmental limits indicated by this chart were substantiated by actual measurements taken during military operations for Ordnance testing at various locations (Ref. 31).

The engines shall be capable of satisfactory performance under severe humidity conditions such as 100% relative humidity at 85°F (AR 705-15).

Unless otherwise specified, Type I engines shall operate continuously and satisfactorily on longitudinal slopes from 0% to 60%, and on lateral slopes from 0% to 36%.

Type I engines shall start successfully and be capable of satisfactory performance up to the normal maximum fording depth specified in the end product specification. When equipped with the fording kit specified in the end product specification, Type I engines shall start successfully and be capable of satisfactory performance when completely submerged in fresh or salt water.

Engines must operate satisfactorily under the specified dust and dirt conditions when using air and oil cleaners meeting these specifications.

7-15 COMPARISON OF MILITARY AND COMMERCIAL POWER PLANTS

The various differences existing between military and commercial power plants reflect the different service requirements of military and commercial engines. Some functional and design characteristics of military power plants are:

a. Engines for military vehicles must be completely waterproof and must be capable of operating submerged in either fresh or sea water with a minimum of additional fording equipment. For example, auxiliary intake and exhaust stacks may be required, but totally enclosed ignition systems and totally enclosed crankcase breathers must be provided on the basic engine.

b. Military engines must come equipped with automatic cooling fan declutching equipment to permit underwater operation; however, the cooling water pump must not be disconnected from its source of power.

c. The use of V-belts should be minimized because they perform unsatisfactorily in the extreme dirt and low-temperature operating conditions that are common to the military environment.

d. Military engines require more efficient oil, air, and fuel filters than similar components of commercial power plants.

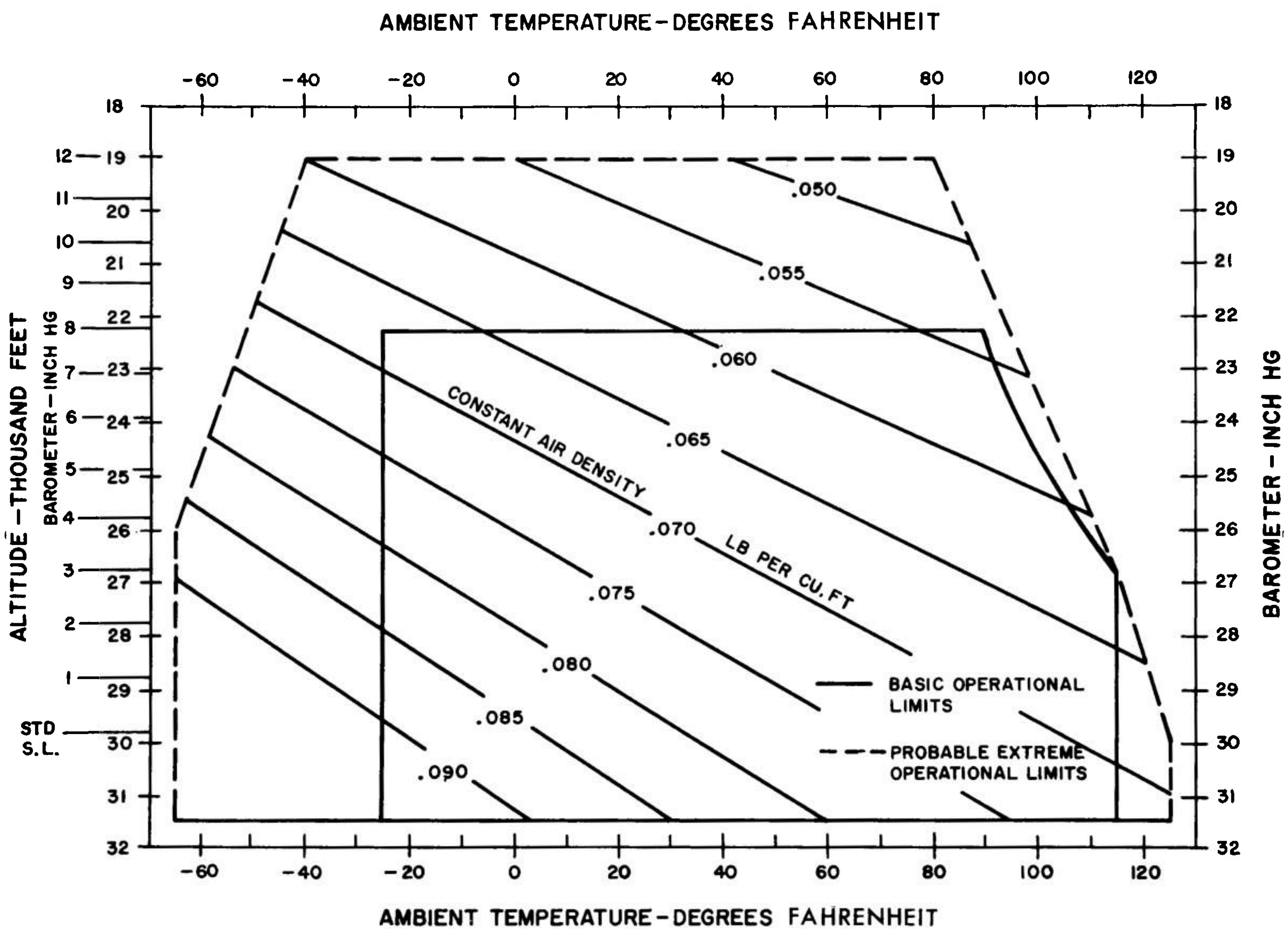


Figure 7-32. *Environmental Limits for Automotive Engines**

* *Taken from* Testing Engine in Simulated Atmospheric Environment, *H. T. Cline, Automotive Engine Lab., APG., April 1960.*

e. Twenty-four volt electrical systems are mandatory on all military engines.

f. The generator capacity of military engines is much greater than the capacity of standard commercial engine generators.

g. The electrical system, including the generator, of military engines must be fully suppressed to prevent interference in electronic equipment.

h. Both inertial engagement and positive engagement electric starters are used in military vehicles. However, the latter type is preferred and may be specified for certain vehicles.

i. All electrical control devices must meet the requirements as to waterproofing, fungusproofing, 24 v, and must be suppressed to prevent electronic interference.

j. Military combat operations do not permit servicing power plants at frequent intervals. For this reason, the power plant must be designed with increased oil and cooling capacities and greater reliability of accessory items, such as sparkplugs and ignition system points, compared to commercial engines, to minimize servicing.

k. Military engines should have lower specific weight (lb/bhp), and a smaller envelope (cu ft) per unit output than comparable commercial engines.

l. Military operations require that power plants operate for extended periods of time while on inclines, both longitudinal and lateral; these conditions are not normally encountered in civilian environments.

m. Military engines must meet more severe slope requirements.

n. Military engines must be designed for reliable operation in much greater climatic extremes than those normally encountered in civilian environments.

Some of the important categories to be considered when selecting military power plants are: reliability, fuel consumption, weight, bulk, manufacturing cost, type of fuel required, storability, maintainability, and durability. A brief discussion of each of these items follows.

7-15.1 RELIABILITY

In general, military engines must be more reliable than comparable commercial engines, especially in severe environments. Military engines are expected to be reliable under conditions of (a) extreme dust; (b) extreme temperature range, $-65°$ to $+125°F$; (c) complete water submergence; (d) rough usage because of operator abuse or military necessity. The greatest increase in reliability of military engines, relative to commercial units, is a result of using better quality components, such as ignition systems, air cleaners, generators, etc.

7-15.2 FUEL CONSUMPTION

Minimum fuel consumption is a main objective for both commercial and military operations. Commercial vehicle operators are interested in minimum operating costs, while military operators are interested in maximum vehicle range for the fuel carried in the vehicle. The emphasis on minimum brake specific fuel consumption over the widely varying range of engine operation encountered in military operations has led to the effort to introduce compression-ignition engines in combat vehicles. An overall reduction of about 40% in fuel consumption is possible by using compression-ignition engines rather than spark-ignition units. However, the initial cost and the specific weight (lb/bhp) of compression-ignition engines are higher than those of spark-ignition engines.

7-15.3 WEIGHT AND BULK

The current emphasis on air transportability has accentuated the need for lighter weight power plants. The trend toward lighter, more efficient military vehicles also emphasizes the need for lighter weight, higher performance power units. Both of these factors, air transportability and high performance, affect the size or bulk requirements of military engines also. Many military engines require armor protection, the surface area of which increases at a faster rate than does the volume or envelope of the engine. Thus, a small envelope is desired to keep the total weight of the engine armor to a minimum.

7-15.4 COST

The manufacturing cost of military engines should be held to a minimum without sacrificing performance, reliability, and durability. Maximum effort must be made at all times to produce the very best military engine that human ingenuity can contrive, at any expense! The second best engine may result in the second best vehicle, which may lead to the second best army on the battlefield —and the second best army can only meet with defeat. Thus, monetary economy at the expense of the highest possible performance is a false economy when it results in defeat on the battlefield.

7-15.5 STORABILITY

Commercial engines, normally, are not subjected to long storage. Military engines are frequently stored for long periods. The main problem encountered during storage of military engines is corrosion of cylinder bores. Special cylinder and ring materials or treatment can be resorted to but only at prohibitive increase in costs. At present, cylinder bores are "fogged" with a rust-preventive oil.

7-15.6 MAINTAINABILITY

Accessibility is given higher priority in design in the case of military vehicle engines than in commercial vehicle engines of similar types. Tank engines are designed for maximum accessibility from the top of the engine. When the provision of top accessibility to engine auxiliaries is impractical, units possessing a high degree of reliability are chosen.

7-15.7 DESIGN LIFE

The relationship between developed power and power plant life can be investigated by considering the performance characteristics of specific power output. For the present purpose, specific output can be given two definitions: (a) brake horsepower per cubic inch of piston displacement and (b) brake horsepower per pound of engine weight.

High specific outputs in either sense of the term tend to produce high operating stresses. These stresses can be classified as load-induced or temperature-induced. Another important aspect of the stress problem concerns the change in mechanical properties of materials as a function of temperature. In general, the strength of the materials used for conventional engines decreases with an increase in temperature. Therefore, with a given stress pattern, the service life of a part may be severely reduced by increasing the temperature of the materials.

In terms of brake horsepower per cubic inch of piston displacement, high values of specific output are attained by developing high values of brake mean effective pressure or high engine speeds. Brake mean effective pressure is increased by increasing the volumetric efficiency of the cylinder (through valve timing and location, supercharging, induction and exhaust tuning, etc.) and by increasing the quantity of fuel burned per unit of time.

An increase in output by means of an increase in brake mean effective pressure, for a given engine, will result in higher load-induced and temperature-induced stresses and a reduced factor of safety for affected parts. Increasing the brake horsepower per cubic inch of displacement by means of increasing the engine speed increases the operating stresses by increasing the inertial loads. The operating temperatures of the parts may increase also, owing to the reduced time period for cooling between combustion events.

In terms of brake horsepower per unit weight, the specific output may be increased, in general, by three means: (a) using higher working stresses, (b) using components designed for uniform stress distributions, and (c) using materials with higher strength-weight ratios.

If the working stresses of an engine are increased, the service life tends to decrease and the reliability decreases. However, in many instances the working stresses of a component are lower than necessary for satisfactory service; a lack of knowledge of the loadings and the effects of the loadings results in over-designed parts.

The weights of most current reciprocating engines (excluding aircraft engines) could be substantially reduced with no decrease in service life or reliability if all the components were designed for uniform strength; i.e., the unit stress would be constant throughout the component. This approach would require thorough design-development programs and a re-evaluation of manufacturing costs.

Finally, the specific output of current engine designs could be increased with no decrease in service life if materials having higher strength-to-weight ratios were used wherever practicable.

We may conclude that increasing the specific power output (in either sense) of present reciprocating engines tends to decrease service life unless compensating measures are used.

The specific output of present multiple-shaft gas turbines, based on brake horsepower per unit weight or unit volume, is limited by material limitations. The high, continuous operating temperature of the compressor and the power turbines influences the design stresses.

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CHAPTER 8

THE POWER TRAIN*

SECTION I GENERAL DISCUSSION

8-1 SCOPE

This chapter contains a discussion of the characteristics and requirements of power trains for military vehicles. It includes a discussion of various components of the power train system and the factors that should be considered in the determination of power train requirements for both wheeled and track-laying vehicles.

The power train, or drive train, of an automotive vehicle is a system of components that transmits the useful energy produced by the power plant from the output shaft of the power plant to its ultimate point of application, the wheels or tracks. Included are components such as clutches, transmissions, transfer cases, drive shafts, differentials, axles, and brakes. The power train includes transmission systems for powered auxiliary equipment in addition to that for the main propulsion system.

8-2 BASIC POWER TRAINS FOR WHEELED VEHICLES

Typical elements of the power train for a wheeled vehicle are shown in Fig. 8-1. Brief descriptions of individual elements of the power train are carried out in this paragraph. Detailed data regarding operation of each element are included in Sections II through VIII of this chapter.

8-2.1 CLUTCH

The clutch serves as a control element in the power train. By means of the clutch the operator can disconnect the engine from the remainder of the power train. This feature is essential in starting the engine, in allowing the vehicle to remain motionless while the engine is running, and to permit a gradual engagement of the power source to the power train. In addition, the clutch may be required to permit gear ratio changes in the transmission while the vehicle is in motion.

8-2.2 TRANSMISSION

The basic purpose of the transmission in an automotive vehicle power train is twofold. It provides the necessary engine torque multiplication required to propel the vehicle under a variety of road and load conditions, and it permits reserve motion of the vehicle by reversing the direction of rotation of the driving axle shafts.

8-2.3 TRANSFER CASE OR ASSEMBLY

The purpose of the transfer case is to divide the power output of the engine so that it may be transferred to both front and rear drive shafts of all-wheel-drive vehicles and to the propeller shaft of amphibious vehicles. It allows sufficient lateral displacement in the position of the front propeller shaft to permit it to pass to one side of the crankcase of the engine rather than beneath it.

8-2.4 UNIVERSAL JOINTS

Because relative angular motion occurs between the different elements of the power train, universal joints must be placed in the shafting connecting these elements.

8-2.5 SLIP JOINTS

Because relative displacement occurs between the different elements of the power train, slip joints must be placed in the shafting connecting these elements.

8-2.6 PROPELLER SHAFT

The propeller shaft ordinarily transmits the output torque of the vehicle transmission to the final drive unit of the power train.

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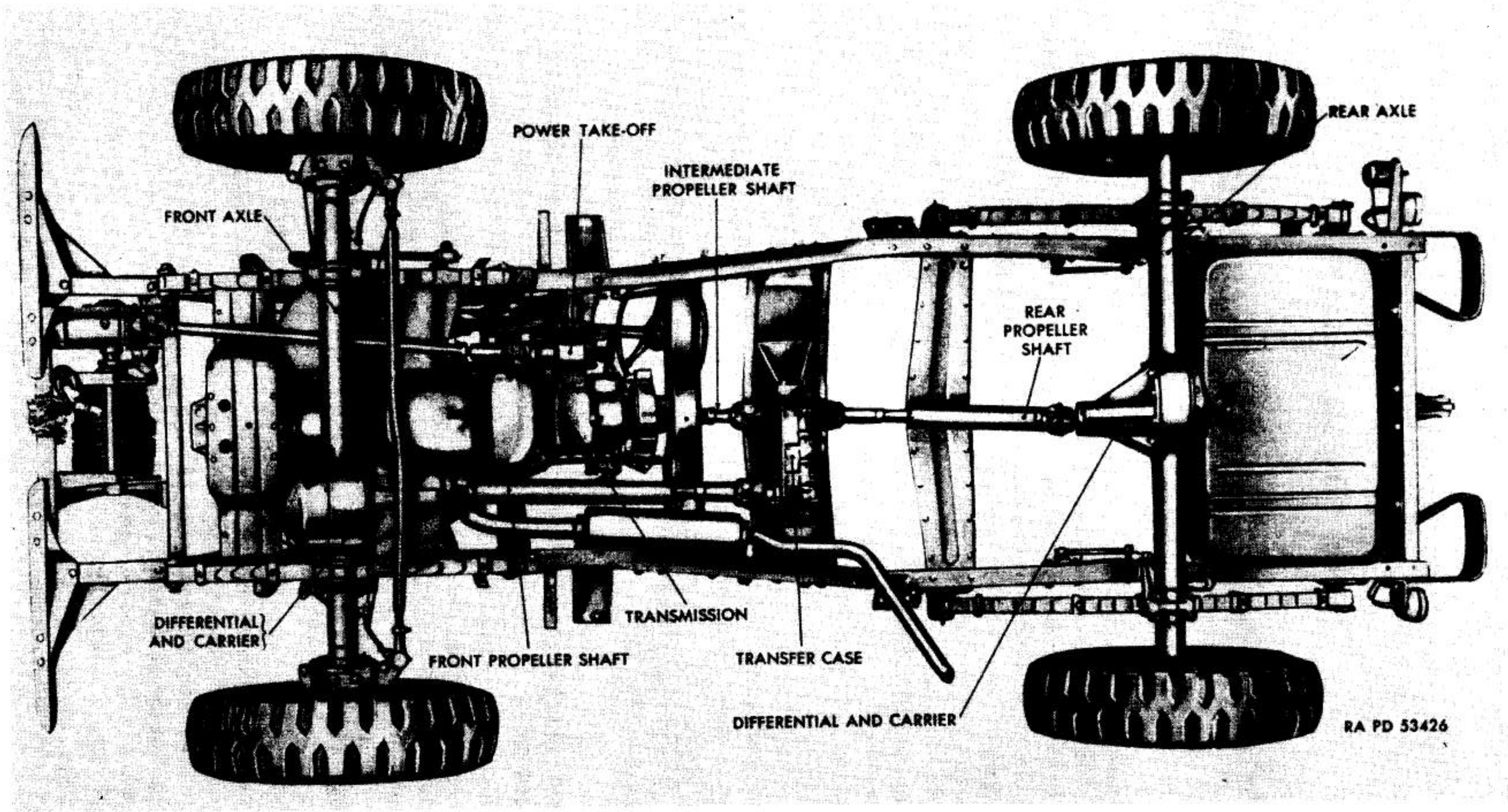


Figure 8-1. *Typical Power Train for A Four-Wheel Drive Vehicle*

8-2.7 DIFFERENTIAL

When a vehicle traverses a curve, the outer wheels, which travel on a curve of greater radius than the inner wheels, must travel a greater distance. If this is to be accomplished without wheel slippage, the rotational velocity of the outer wheel must be greater than that of the inner wheel. The function of the differential is to produce this difference in rotational velocities between inner and outer wheels.

8-2.8 FINAL DRIVE UNIT

The final drive unit is the element of the power train located between the propeller shaft and the differential. Its function is to modify the torque output of the propeller shaft to a form suitable for use at the driving axles. The torque is changed in magnitude and direction by means of reduction gearing. Angularity relations between propeller shaft and driving axle determine final drive unit configuration.

8-2.9 AXLE ASSEMBLY

The term *axle assembly*, as used herein, includes the driving axle, which includes the axle shaft, the axle shaft housing, and the torque-resisting element or elements. When a driving torque is applied to the axle shaft by means of the final drive, an equal and opposite torque or reaction, tending to turn the axle housing in the reverse direction, arises. The tractive effort of the wheels on the road, which propels the vehicle forward or backward, is also exerted on the axle housing. Torque-resisting members must be incorporated in the axle-positioning linkage to transmit this reaction to the vehicle frame.

8-2.10 BRAKES

Brakes are those elements of the power train which are used to retard or arrest vehicle motion.

8-3 BASIC POWER TRAINS FOR TRACK-LAYING VEHICLES

Full-track vehicles are usually steered by changing the speed of one track relative to that of the other. As a result, the steering system of the full-tracked vehicle is incorporated in the power train.

8-3.1 CLUTCH-BRAKE STEERING

In some full-tracked vehicles, a system of clutches and brakes is used in the final drive to each sprocket. By engaging and disengaging the clutches, and by varying the braking action to each sprocket, the relative speed of the tracks can be controlled by the operator. As discussed elsewhere, this system is not entirely satisfactory for military vehicles.

8-3.2 CONTROLLED DIFFERENTIAL STEERING

In many modern military track-laying vehicles, steering is accomplished by means of the controlled differential system. Such differential systems employ planetary gear trains by means of which power is applied to both tracks at all times and their relative speeds regulated for steering control. The cross-drive transmission, one form of controlled differential used on recent tanks, utilizes a system of planetary gear trains as in other controlled differentials, and, in addition, contains a torque converter.

SECTION II CLUTCHES AND COUPLINGS

8-4 CLUTCHES

Clutches are used to transmit rotary motion from one shaft to another, while permitting engagement or disengagement of the shafts during rotation of one or both members. Normally, these shafts have a common axis of rotation.

There are two general types of clutches: those which provide only positive engagement and those which are capable of gradual engagement. The positive engagement clutch is either fully engaged or disengaged, while the gradual engagement clutch can be engaged to any degree between nonengagement and complete engagement. Gradual engagement clutches include the common friction type, hydraulic, magnetic, and others. Detailed data are available in Refs. 6 to 19.

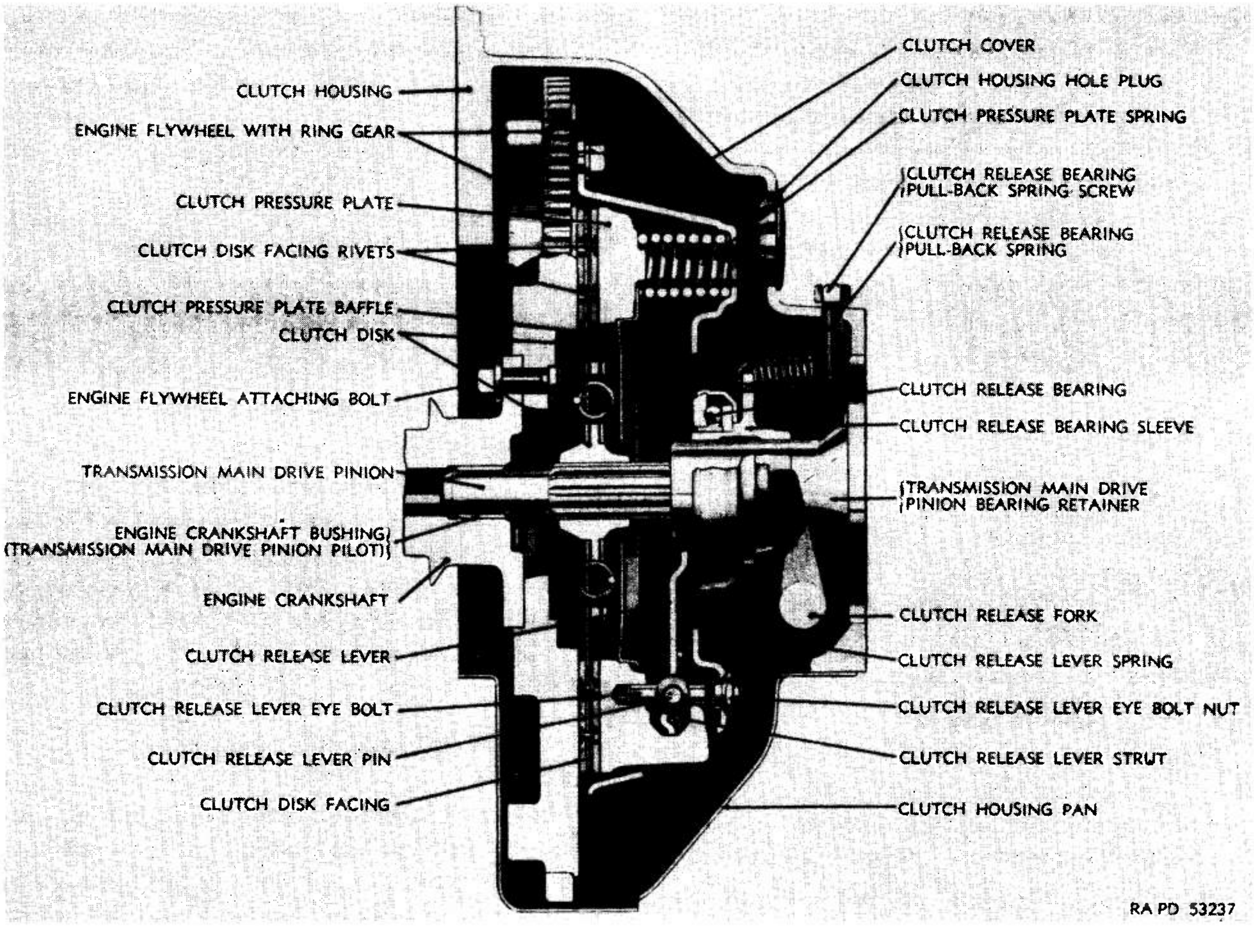


Figure 8-2. *Plate Clutch—Cross Section View*

8-4.1 FRICTION CLUTCHES

8-4.1.1 Clutch Classification

Three types of friction clutches have been used in automotive applications: the single-plate, multiple-plate, and cone configurations. The latter type is no longer in use and is not discussed here.

Single-plate clutches are generally used on light- and medium-weight vehicles, while multiple-plate clutches may be used on heavier vehicles.

8-4.1.2 Clutch Elements

The basic elements of the friction clutch are discussed below by reference to the single-plate clutch.

The principal parts of a clutch, Fig. 8-2, include the driving members, which are fastened to the output shaft of the power unit; the driven members, which are fastened to the input shaft of the transmission; and the operating members, which include a spring or springs and the linkage required to apply and release the pressure to maintain the driving and driven members in contact.

The driving members usually consist of two machined, flat, cast iron plates. Cast iron is employed because the embedded graphite provides some lubrication during slippage. The rear face of the engine flywheel, and a comparatively heavy flat ring, known as the pressure plate, comprise the driven members which are bolted together. The pressure plate, together with several operating members, are contained in a common housing.

The disk-shaped driven member is free to slide on the splined clutch shaft, and drives the shaft through these splines. The spring-steel clutch disk is usually formed into a single, flat disk or into a number of flat segments. Frictional facings are attached to each side of the disk by means of copper rivets or suitable bonding agents.

In order to obtain smooth clutch engagement, the driven disk is made flexible. In one design, the

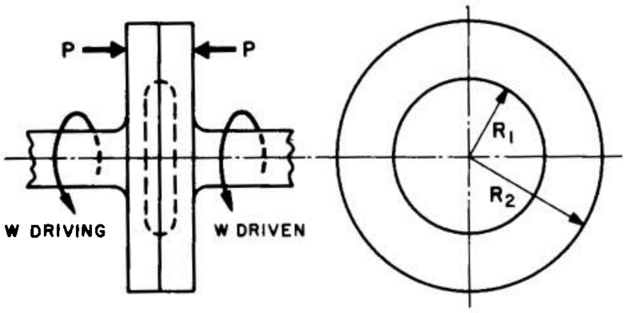


Figure 8-3. *Elements of a Friction Clutch*

driven disk is dished to permit its inner and outer edges to make initial contact with the driving members as the members approach. Under increasing spring pressure, the disk is flattened and contact area increases. In another design, steel segments, attached to the driven disk and slightly twisted, make initial contact with the driving member over a small area. As clutch spring pressure increases, the segments are flattened and increase the contact area.

The driven member of the clutch is usually provided with a flexible center which absorbs the torsional vibrations of the crankshaft and prevents their transmission to the power train. The flexible center usually contains a number of steel compression springs located between the hub and the steel disk. Under load, these springs permit the disk to rotate slightly with respect to the hub.

8-4.1.3 Clutch Torque Capacity

8-4.1.3.1 Plate Clutches

Both single- and multiple-disk clutches are commonly used in automotive service. Figure 8-3 shows the elements of a simple friction plate clutch. Two design approaches are used for plate clutches: one is based on the assumption of constant pressure over the clutch face; the other, that wear is constant over the clutch face.

In the first approach it is assumed that the unit pressure, p, is uniform over the entire frictional surface and the coefficient of friction, μ, between the contacting surfaces is constant. The coefficient of friction for a given combination of materials will vary with contact pressure and sliding velocity. In the normal automotive application, during steady-state operation of the fully engaged clutch, slippage can be neglected. For a more rigorous treatment of this problem, see Ref. 5. The steady-state torque, T (in-lb), which can be transmitted is then

$$T = \frac{2\mu\ (r_2^3 - r_1^3)\ P}{3\ (r_2^2 - r_1^2)} \tag{8-1}$$

where

μ is the static coefficient of friction

r_2 is the outer radius of the contacting surface, in

r_1 is the inner radius of the contacting surface, in

P is the total axial force on the clutch members, lb

For a clutch having n pairs of frictional surfaces in contact, the torque, T_n, which can be transmitted is

$$T_n = T \cdot n \tag{8-2}$$

If it is assumed that wear is uniform, that wear is proportional to the product of pressure and rubbing speed (tangential velocity) and the coefficient of friction is constant, the steady-state torque, T, which can be transmitted is

$$T = \tfrac{1}{2}\mu\ (r_2 + r_1)\ P \tag{8-3}$$

For a clutch having n pairs of frictional surfaces, the torque which can be transmitted is given by Eq. 8-2.

Equations 8-1 through 8-3 indicate the capacity of a plate clutch is a direct function of the coefficient of friction, the contact area, and the normal force between the contacting surfaces. More conservative results are obtained by use of the uniform wear approach.

8-4.1.3.2 Cone Clutches

The cone clutch utilizes mating conical surfaces to transmit torque. Normal force, for a given engagement force, is greater between conical surfaces than between parallel surfaces. The relation between forces in the cone clutch, shown in Fig.

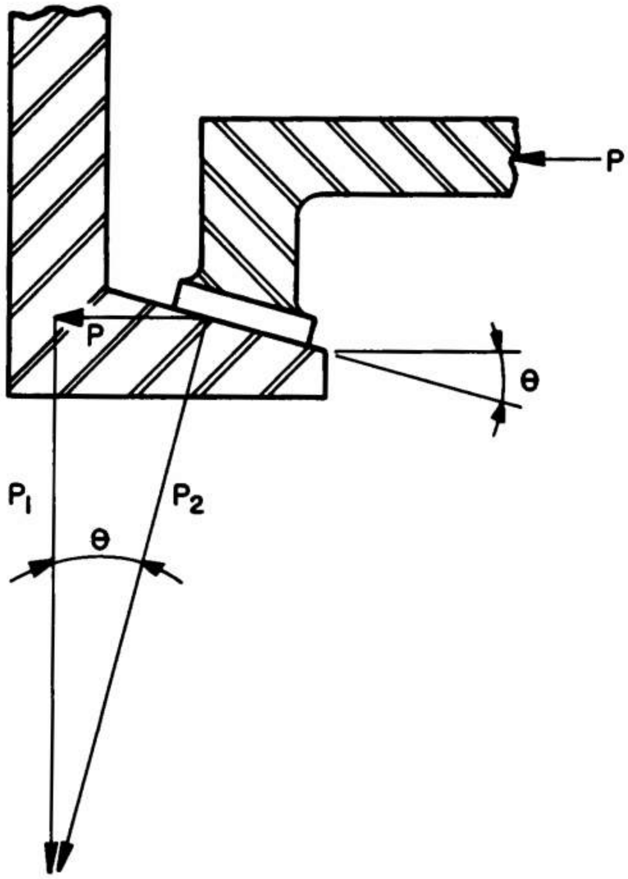


Figure 8-4. *Force Diagram, Cone Clutch*

8-4, where P is the engagement force and P_1, the normal force, is

$$P_1 = \frac{P}{\sin \vartheta} = P csc\ \vartheta \qquad (8\text{-}4)$$

The cone angle, ϑ, is usually selected such that $P_1 = 5\ P$.

To avoid binding the conical surfaces, and to reduce sensitivity to wear, ϑ should not be less than about 20°.

Clutch wear, which is dependent upon both contact pressure and slip velocity, is high in the cone clutch because of the high contact pressures involved. In addition, the physical configuration of a cone clutch is generally more complicated than the plate clutch, which leads to difficulties in maintenance and servicing.

8-4.1.4 Friction Surfaces

A material suitable for use as a clutch facing must meet certain requirements including:

1. Its coefficient of friction must be high.
2. It must be relatively unaffected by such agents as salt water, petroleum-based liquids and road contaminants.
3. It must be highly resistant to wear, commensurate with other factors.
4. It must retain its properties at elevated temperatures.

Present military requirements dictate the use of either a woven type or molded (composition) type of friction lining. The woven lining employs asbestos filler woven into cloth impregnated with bonding material. The molded or composition lining is pressure-formed from asbestos fibers mixed with a bonding agent. Metallic threads are sometimes added to increase the wearing properties. Bonding agents include: petroleum-based vegetable gums, rubber, and synthetic resins. The latter two offer the most promise in military applications.

Friction coefficients vary from approximately 0.3 to 0.6, with the rubber bases exhibiting the higher coefficient. Temperatures up to and including 450°F appear to cause no major change in friction coefficients for materials presently used in military vehicles. For some materials, however, the friction coefficient increases as the temperature is increased due to the exudation of the bonding material.

Surface pressures in contemporary clutch designs are on the order of 15 to 50 psi, well below the compressive strength of asbestos, which is approximately 500 psi.

8-4.1.5 Wet Clutches

A wet clutch is similar to a dry friction clutch in operation; however, oil is supplied to the friction surfaces for cooling purposes. Wet clutches are almost exclusively multiple-disk clutches, few of them being used in American military vehicles. Heat absorbed from the clutch is dissipated by passing the oil through a heat exchanger.

8-4.2 MAGNETIC CLUTCHES

The use of magnetic clutches on military vehicles has been confined to auxiliary drives. Conventional magnetic disk or cone clutches utilize solenoids or magnets to supply the force for engagement by means of a system of links, or they may force the friction surface together by magnetic means. Such clutches have proven less rugged than mechanically-operated types.

In the magnetic particle clutch, a newer type of magnetic clutch, two iron plates are separated by an air gap. Oil containing a suspension of extremely fine iron particles flows through this gap. When the gap between the facing plates is magnetized, the particles in the oil are polarized, causing the oil to act as a solid and the clutch is engaged.

8-4.3 EDDY CURRENT CLUTCHES

The eddy current clutch utilizes the drag produced by electrical eddy currents for clutch action. Eddy current clutches have the same disadvantages as magnetic clutches when considered for the military vehicle application.

8-4.4 SUMMARY

The factors which influence the choice of a clutch include input torque, rotative speed, available space, service requirements, and frequency of operation. Torque capacity may be evaluated by use of Eqs. 8-1 to 8-3.

High rotative speeds require the use of balanced clutches. Clutches which are in frequent operation should have a small travel, simple engaging and disengaging mechanisms, and large heat-dissipating areas.

Clutches are ordinarily used in the power train of the vehicle as described above. In addition, clutches may be used in the steering system of a track-laying vehicle, gun drive systems, or in auxiliary power takeoffs.

In general, for a given clutch face diameter, a multiple-plate clutch will have a higher torque capacity than a comparable single-plate clutch. The rate of engagement of the former is normally slower than that of the latter.

8-5 FLUID COUPLINGS

All current hydrodynamic drives may be classified as either fluid couplings or hydraulic torque converters. Operation of the fluid coupling or basic hydrodynamic drive is based on a transfer of kinetic energy of the working fluid as it accelerates in an impeller (driving member) and decelerates in a turbine (driven member). Torque transmitted is proportional to mass flow rate and the change in fluid velocity in each member.

8-5.1 OPERATING CHARACTERISTICS

8-5.1.1 Velocity Relations

A fluid coupling, consisting of an impeller with radial vanes, or blades, and a similar matching turbine, is represented schematically in Figs. 8-5 and 8-6. The working fluid is contained within a closed chamber, most generally a hollow toroid, which also houses the turbine and impeller blading. The velocity of the fluid particles in a fluid coupling is shown in Fig. 8-7 where U represents the tangential component of the velocity of a fluid particle, F, its radial component, and V, the resultant velocity.

The magnitude of the velocity of the fluid particle at Point B is essentially the same as it is at A. Its direction, however, is changed, as shown in Fig. 8-7. The variation of the velocity components under a number of operating conditions is discussed below.

8-5.1.2 Modes of Operation

The coupling may operate in any of three possible modes or conditions:

(1) *No slip.* The impeller and turbine rotate at the same angular velocity. Such operation occurs at no-load.

(2) *One hundred percent slip.* The impeller rotates at any speed; the turbine is stalled. When the impeller is at full rated speed and the turbine is stalled, the coupling operates at maximum torque capacity.

(3) *Intermediate condition of slip.* Some slip occurs; turbine angular velocity is less than that of the impeller.

With the coupling operating in the first mode, i.e., at no slip, no transfer of kinetic energy from driving to driven member occurs. No power is available at the output shaft; and, theoretically, no power is required from the prime mover. In practice, however, a small amount of power is required to overcome bearing friction. Under this condition the radial velocity component, F, Fig. 8-7, is zero, and the tangential velocity component, U, equals the resultant velocity, V. Fluid motion is, therefore, purely rotational.

When the coupling is operating in the second condition, i.e., at 100% slip, the relative velocity between impeller and turbine is a maximum; and maximum kinetic energy transfer, as well as maximum fluid flow, occurs. In this case, the radial component of velocity, F, is a maximum.

With the coupling operating in the third condition, i.e., at an intermediate condition of slip, an intermediate value of relative velocity between impeller and turbine exists. As a result, the fluid

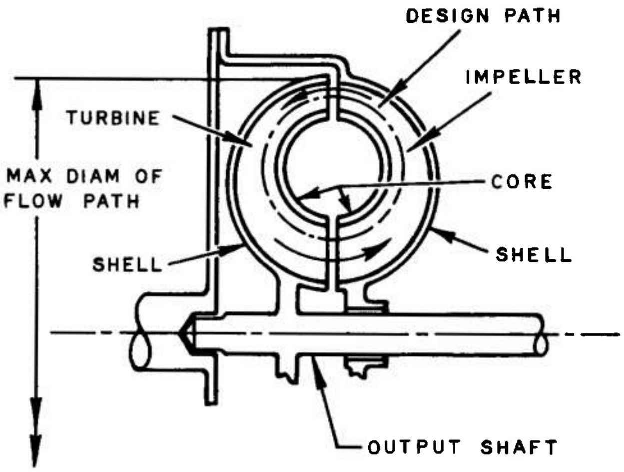


Figure 8-5. *Schematic Representation of Fluid Coupling (From "Fluid Couplings" by W. B. Gibson, Machine Design, March, 31, 1960)*

pressure in the impeller is higher than that in the turbine. Fluid flows from the outer portion of the impeller into the turbine, then radially inward and back into the impeller near its center of rotation. Relative magnitudes of the velocity components during operation in the intermediate condition are shown in Fig. 8-7.

8-5.2 FLUID COUPLING PERFORMANCE

Figure 8-8 shows torque absorbed and transmitted to the turbine as a function of speed ratio; N_2 (turbine rpm)/N_1 (impeller rpm) for a typical fluid coupling. Data are presented for a constant input rpm of 1700. Torque, T, absorbed at any other input speed, N (rpm), is given by $T = T_0 \ (N/1700)_2$. Torque absorbed is a maximum at a speed ratio of zero and drops to zero at a speed ratio of unity. Efficiency rises linearly with speed ratio until this ratio approaches unity, at which time efficiency drops to zero. Ordinarily, couplings are designed to operate at about 3% or 4% slip (the point of maximum efficiency) when the prime mover is operating at its design speed. Point *B* (Fig. 8-8) represents such a point: the coupling operating at a speed ratio of 0.96 and an efficiency of 96%.

When the load torque required increases, the fluid coupling turbine slows, causing the coupling to operate at a lower speed ratio. Because more torque is now absorbed in the clutch, prime mover output speeds drops. If the prime mover is operating at a speed greater than that for maximum torque, torque to the coupling rises. In any case,

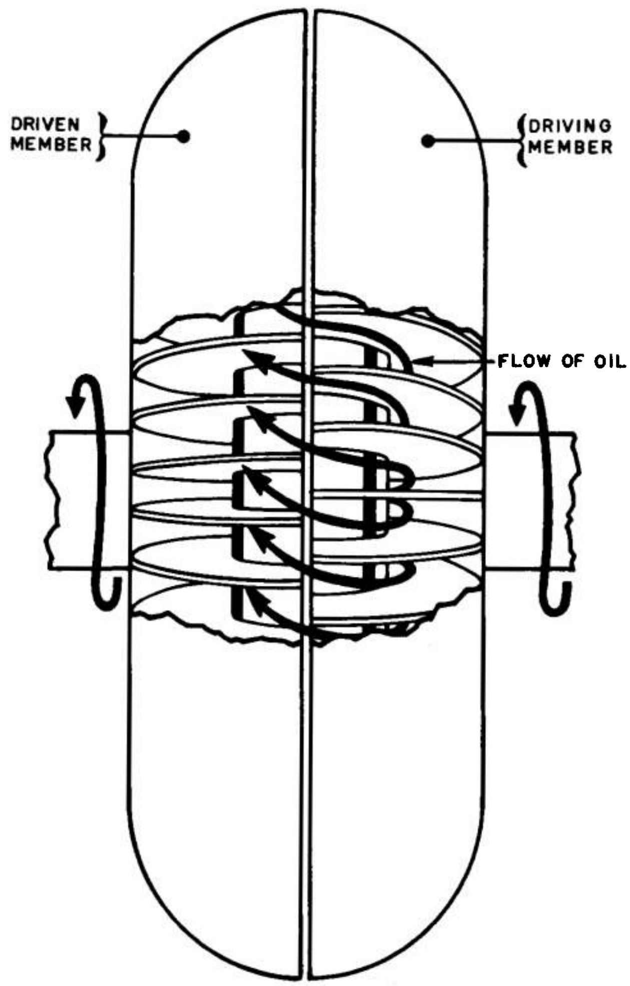


Figure 8-6. *Fluid Coupling, Path of Working Fluid*

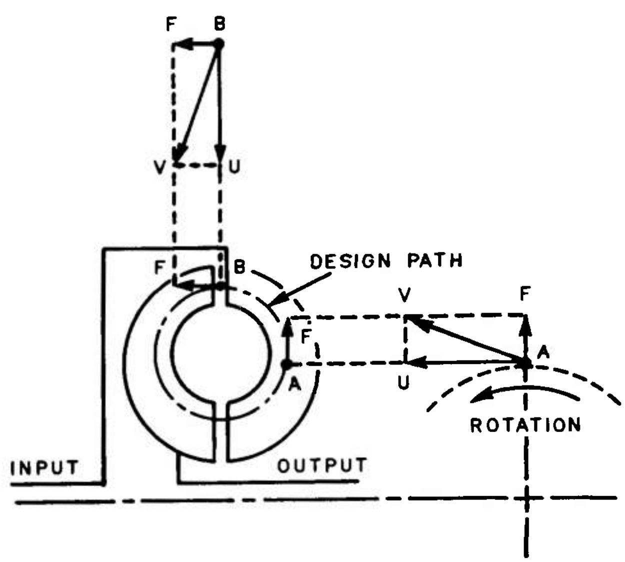


Figure 8-7. *Velocity of Fluid Particles in a Fluid Coupling (From "Fluid Couplings" by W. B. Gibson, Machine Design, March 31, 1960)*

because impeller speed drops, the speed ratio of the coupling rises and torque absorbed drops. This process continues until the power train attains a new state of balance, with the fluid coupling operating at the proper condition of slip to supply load torque requirements.

Torque capacity of a fluid coupling depends on the mean diameter of the fluid chamber, and the rate of fluid transfer from the impeller to the turbine. The rate of fluid transfer, in turn, is determined by the shape of the chamber, configuration of the passages, and resistance to fluid circulation of all parts of the hydraulic circuit.

8-5.3 FLUID COUPLING APPLICATIONS

The fluid coupling may be used either with a conventional clutch and transmission or, as is more frequent, as part of an automatic transmission, in which case no clutch is required.

Because a fluid coupling will slip if the torque demand rises suddenly, such couplings are placed in series with friction clutches to protect both the clutch and power source from overloads.

To eliminate the losses caused by slip in a fluid coupling when the coupling is operating at, or near, its design speed, a so-called fluid-friction clutch may be used. In this system, a spring-loaded, centrifugal-actuated friction clutch, mounted in parallel with the fluid coupling, is employed. Above a predetermined speed level, usually the point of maximum efficiency, this clutch engages and slip drops to zero. At lower speeds, the operation of the fluid coupling remains unaffected.

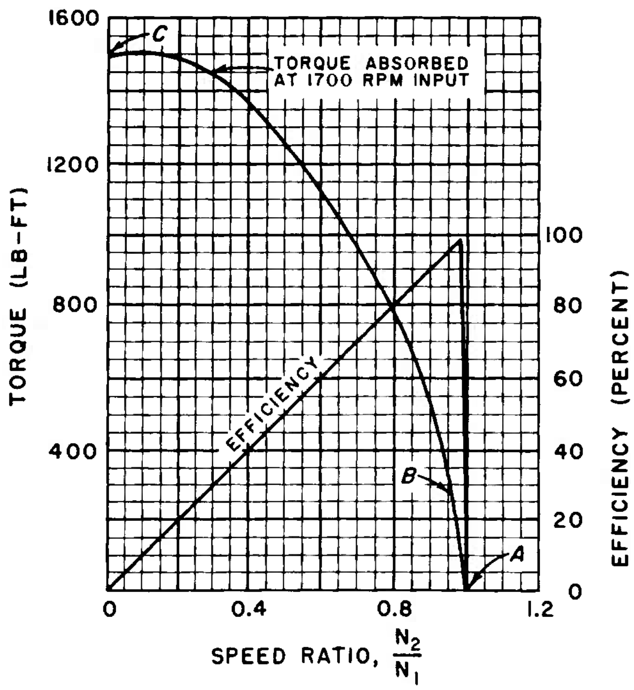


Figure 8-8. Torque Absorbed and Efficiency as Functions of Fluid Coupling Speed Ratio (From "Fluid Couplings," by W. B. Gibson, Machine Design, March 31, 1960)

SECTION III TRANSMISSIONS

8-6 FUNCTION OF THE AUTOMOTIVE TRANSMISSION

The primary function of the transmission is to provide a means of varying the speed ratio between power source and tractive elements of the vehicle. The transmission may be manually or automatically operated, and it may be mechanical, hydráulic, or electrical, or a combination of these in nature.

The reciprocating piston internal combustion engine is used in all current standard military vehicles. Engines of this type have certain performance or operating characteristics which are not ideally suited to vehicle propulsion. The most important of these are:

1. The inherent power-torque relationship of an engine of this type is unfavorable for efficient vehicle propulsion. Figures 8-9 and 8-10 indicate the variation of power and torque with engine speed for an ideal power plant and for a typical spark-ignition engine, both operating at full throttle. The ideal power plant for vehicle propulsion would provide a constant power output throughout its entire usable speed range (the level of this power output would be varied to suit the performance needs). Torque output for such a power plant would decrease hyperbolically with increasing engine speed. A power plant of this type, coupled directly to the drive axles or sprockets,

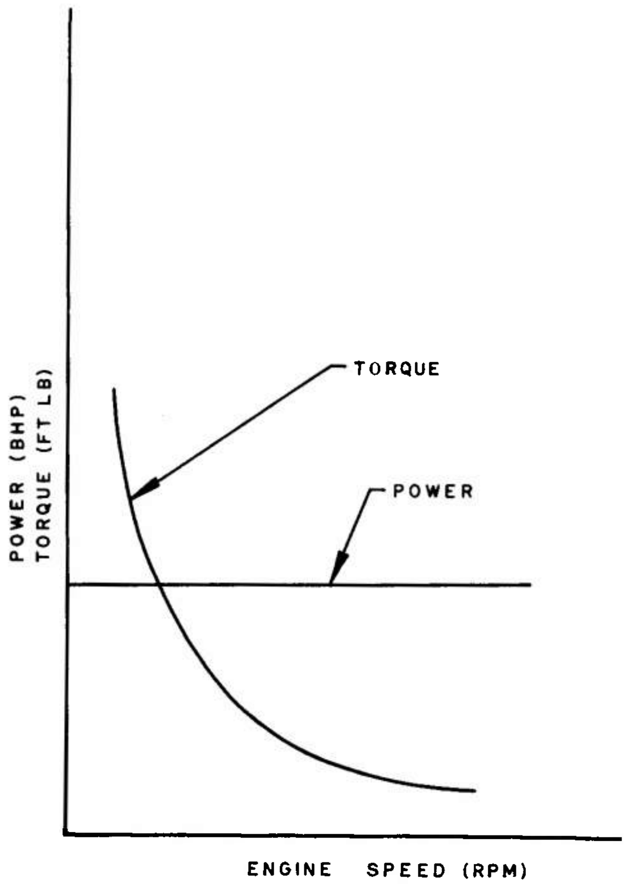


Figure 8-9. ***Power-Torque Relationship for Ideal Vehicle Power Plant***

would provide peak torque to these elements at the lowest vehicle (and power plant) speeds when it is most needed for vehicle starting, acceleration, and grade performance. For any predetermined power level up to the maximum available from the engine, the road horsepower at the wheels or tracks would be constant throughout the vehicle speed range.

The power and torque developed by the conventional power plant, at full throttle, varies with engine speed. If such an engine were to be coupled directly to the drive axles or sprockets, the torque would be relatively low at engine (and vehicle) speeds, and maximum torque at the axles or sprockets would occur at some intermediate speed. The road horsepower at the wheels or tracks would vary with vehicle speed, in a manner similar to variation in horsepower delivered to the transmission of existing vehicles.

2. The specific fuel consumption of conventional reciprocating engines varies with engine speed

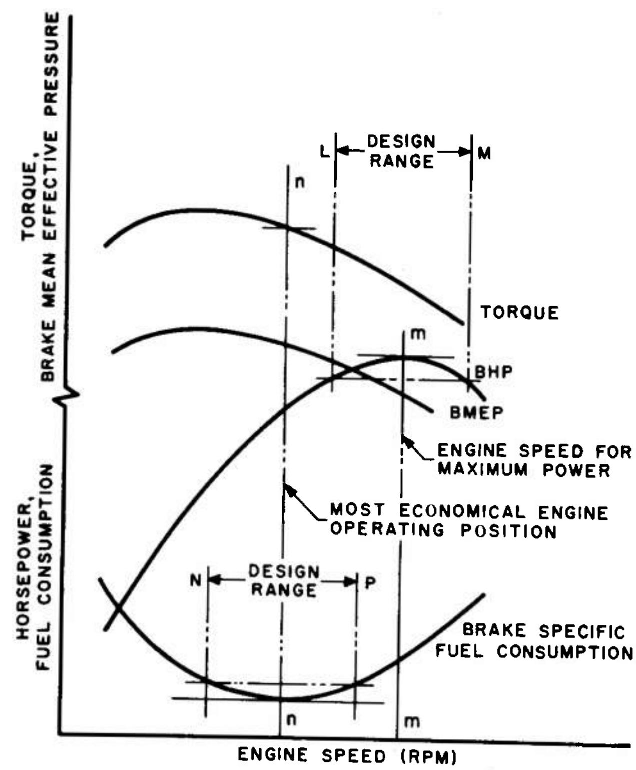


Figure 8-10. ***Typical Engine Performance Curves for Spark-Ignition Engine—Full-Throttle***

and load. For maximum fuel economy, the engine must produce the required power at a specific speed. Direct coupling between the engine and the wheels or tracks will not permit this.

3. The rotation of the output shaft for engines of this type is unidirectional.

The limitations and deficiencies of the conventional engines used to propel vehicles are partially alleviated by a transmission mechanism which can change the speed ratio between the engine and the drive axles or sprockets and which can reverse the direction of rotation of the power plant input shaft. In such a case, speed ratio between the engine and the drive axles or sprockets is adjusted, within limits set by the transmission, to produce the desired results in terms of vehicle performance or operating economy.

Transmissions used in track-laying vehicles, in addition to the stated functions, incorporate the function of controlling the relative speeds of the separate tracks for the purpose of vehicle steering.

Some of the other requirements applying to all military vehicle transmissions are:

1. The transmission and auxiliary components must be readily serviceable.
2. All transmissions must have a high degree of reliability and must be able to withstand operator abuse.
3. The transmission should have a relatively high mechanical efficiency.
4. Transmissions should be as light in weight and as compact as is practicable.

In summary, the purpose of the automotive transmission is to transform power from the form in which it is produced to the form required for efficacious use. Detailed data regarding the automotive transmission are given in Refs. 20 to 48.

8-7 VEHICLE PERFORMANCE AS A FUNCTION OF THE POWER PLANT-TRANSMISSION SYSTEM

8-7.1 BASIC CONSIDERATIONS

Vehicle performance factors which are determined by the power plant-transmission system are: (1) tractive effort, (2) power plant speed-torque characteristics, (3) torque multiplication of the transmission, and (4) power train efficiencies.

The concept of *gross tractive effort* is defined in par. 5-2 of Chapter 5 as the maximum propelling force that can be developed by the ground-contacting elements on a given supporting medium. It is the total propelling force before appropriate reductions are made for resisting forces.

Assuming adequately designed ground-contacting elements, the maximum tractive force that can be developed, for a given vehicle operating on a given supporting medium, is limited by the ultimate strength of the ground material or the interface coefficient of friction between ground material and ground-contacting elements of the vehicle. For vehicles operating under cross country conditions, in various soils and snows, the ultimate strength criterion applies; for vehicles operating on hard pavement, or in some cases, ice, the interface coefficient of friction criterion applies.

In order to develop the *limiting tractive force*, a sufficient torque potential is required at the ground-contacting elements; and the power plant must be capable of producing this torque with the given torque multiplying system.

Since it is the function of the power plant and power train combination to provide the torque required for propulsion of the vehicle under a variety of operational conditions at various vehicle speeds, the functional requirements of the transmission increases as the deficiencies of the power plant and the severity of the operational requirements increase.

Power train efficiencies, at various loads and speeds, influence vehicle performance, since efficiency during any period of operation determines the percentage of engine power available for propulsion.

Current military vehicles use either a multiple-gear transmission, having a limited number of fixed gear ratios, any one of which can be selected and incorporated into the drive system, or a hydrodynamic transmission composed of a hydraulic torque converter and an epicyclic gear set. Each of these transmissions have inherent deficiencies as torque multiplication units for military vehicle propulsion. The following discussion of a typical multiple-gear transmission indicates the purpose, advantages, and limitations of current units. Hydrodynamic transmissions are discussed in par. 8-9, Fluid Transmissions.

8-7.2 PERFORMANCE ANALYSIS

The present discussion applies to a vehicle having a conventional reciprocating piston engine and a multiple fixed-ratio gear transmission. It is assumed that the drive wheels or tracks are positively coupled to the power plant through reduction gearing whose ratio can be changed in several steps. Since a definite speed ratio exists between the engine output shaft and the drive axles or sprockets in any of the several gear combinations, the torque developed at the ground-contacting components will differ from the engine torque, at any instant, by a constant factor which expresses speed ratio and efficiency of the power train.

The definite speed ratio between engine and ground-contacting elements, e.g., the wheels, permits conversion of the engine speed into a theoretical vehicle speed, in any gear ratio, by multiplying engine speed by a suitable constant. Actual vehicle speed will differ from the theoretical speed by a slippage factor—the slippage occurring between wheels or tracks and ground. In the following discussion, the condition of no slippage is considered.

If the torque-speed (or horsepower speed) curve at full throttle is available for a given engine (Fig. 8-10), the torque developed at the wheels or tracks in each of the gears can be calculated for the vehicle speed range. Both engine torque and wheel torque depend on the loading at the wheel. In the present discussion, it is assumed that the loading is such that the maximum possible torque can be developed throughout the speed range. These torque values can be converted to tractive forces by dividing them by the effective radius of the wheel or track. These relationships, for a three-speed fixed ratio transmission, are shown in Fig. 8-11. The tractive effort (force) curves for low, second, and high gears are shown for a vehicle operating at full throttle.

Superimposed on these tractive effort curves of Fig. 8-11 are curves of grade resistance, curves G to G_m, which represent resistances for various positive grades. Grade resistance, which can be a negative value, must be added to the rolling and air resistances to obtain total vehicle resistance at a selected vehicle speed.

If a vehicle, traveling at a particular speed in a given gear, encounters an increase in grade, for example, from G_1 to G_2 in Fig. 8-11 (from Point r to Point s), vehicle speed decreases until the available tractive effort and the new motion-resisting forces are equal. If the grade increases further, the vehicle speed will decrease until the forces are in balance; for example, at Point p. This process would continue until the total resistance exceeds the available tractive force available in high gear. In order to negotiate a grade such as G_4, it would be necessary to increase the speed ratio between the engine and the wheels or tracks. (This would be accomplished by shifting to second gear, in this case, and operating at Point v.)

Since the difference between tractive effort, in any of the gear ratios of the transmission, and total resistance to motion represents drawbar pull, or excess propelling force, for negotiation of grades, vehicle acceleration, or for towing, the function and desirability of a multiple ratio transmission is evident.

Maximum performance, i.e., maximum torque at the road wheels, at a given road speed, is obtained when the engines (at full-throttle) operates at the speed at which it produces maximum power and the transmission ratio is chosen so as to provide the correct speed at the road wheels. If maximum performance is to be attained at any road speed, an infinitely variable ratio transmission is required. Tractive effort versus speed for such a

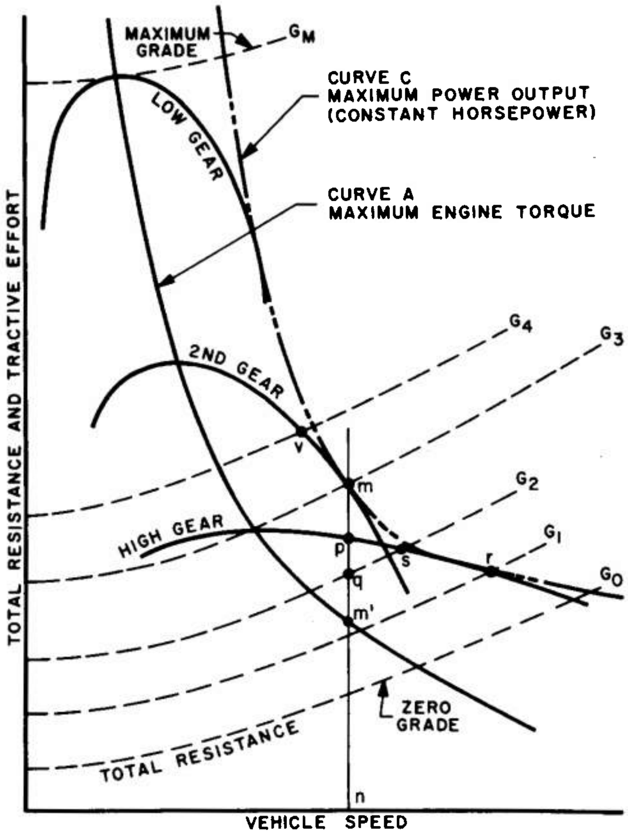


Figure 8-11. ***Performance Diagram—Limited Fixed Ratio Transmission***

transmission is shown by Curve C of Fig. 8-11 which passes through the point of maximum power for each of the curves of the fixed ratio transmission.

Curve A represents the curve of tractive effort versus vehicle speed for an infinitely variable transmission that would provide proper road speed when the engine was operating at maximum torque. For a given vehicle speed, tractive effort for the maximum power-based transmission is greater than that for the maximum torque-based transmission, being represented, for example, at vehicle speed n, Fig. 8-11, by Point m as compared to Point m'.

Maximum economy for a given load is obtained when the prime mover speed and throttle condition chosen will result in minimum brake specific fuel consumption, bsfc. At constant engine speed, brake specific fuel consumption drops with increasing load until a point of maximum economy is attained. As the load is further increased, brake specific fuel consumption rises. At higher values of constant engine speed, the point of minimum brake specific fuel consumption occurs at higher horsepower. For any given load horsepower, therefore, there is an engine speed at which the brake specific fuel consumption is minimum. If the engine is to operate at maximum economy for a given road speed, or load horsepower, the transmission ratio must be such as to require the engine to operate at the speed for which the brake specific fuel consumption is a minimum. An infinitely variable ratio transmission is again required. The control criterion, in this case, is the selection of a ratio to produce minimum brake specific fuel consumption. In the previous case, the criterion was the selection of a ratio to produce maximum performance. In the maximum economy case, a vehicle would travel up a given slope at whatever constant speed was selected by the driver, provided the engine could produce the required horsepower. In the maximum performance case, the vehicle would accelerate up a given slope until the entire engine output was required to overcome tractive resistance, after which the vehicle would travel upward at the maximum possible speed.

The ideal transmission would be an infinitely variable ratio unit capable of automatically selecting the optimum reduction ratio, i.e., maximum performance or economy. This ideal is not realizable at present; current transmissions are either limited fixed-ratio gear types or hydrodynamic torque converters which are discussed in par. 8-9.

The selection of proper gear ratios for a limited multiple-ratio transmission and conventional final drive assembly depends on a knowledge of the power-speed characteristics of the power plant, physical dimensions of the vehicle (such as the effective wheel radii), drive train efficiencies, and motion resistance-speed characteristics of the vehicle. The final drive alone frequently determines the minimum total reduction ratio between power plant and drive axles of a wheeled vehicle since the ratio through the transmission in high gear is normally unity. If motion resistance for the expected speed range is expressed in terms of horsepower, and maximum possible road horsepower is determined from engine and drive train characteristics, a potential maximum velocity of the vehicle is established. The high-gear reduction ratio is selected such that the engine will develop maximum road horsepower (brake horsepower developed at the ground) at the calculated maximum vehicle speed. First-gear, i.e., the gear ratio that produces the maximum reduction in speed between engine and drive axle (sprocket), is selected such that the maximum slope requirements are met. Intermediate ratios of the transmission, ideally, would form a geometric progression, so that the same speed range of the engine horsepower curve would be used in each gear if changes were properly made.

Actual multiple-ratio gear transmissions for military vehicles usually have ratios which differ from the theoretical geometrical ratios to maximize tractive effort over a selected speed range or to limit the number of intermediate gear ratios required.

The number of discrete gear ratios required in the transmission to approach a hyperbolic tractive effort output (torque versus speed) will increase as the engine power versus speed curve deviates from constant power. Engines that are characterized by horsepower versus speed curves with sharply defined maxima require a large number of discrete ratios for high performance, since the engine speed range, over which high power output may be obtained, is relatively small.

8-8 GEAR TRANSMISSIONS

Three basic types of gear transmissions are commonly used in automotive vehicles. These include: (1) the sliding-gear transmission, (2) the constant-mesh transmission, and (3) the planetary gear transmission. Any of these may be incorporated in a track-laying vehicle transmission; however, the present discussion (par. 8-8) is limited to wheeled-vehicle transmissions. Track-laying vehicle transmissions are discussed in par. 8-9 of this chapter.

8-8.1 SLIDING-GEAR TRANSMISSION

In the usual sliding-gear transmission, gear ratios are selected by sliding spur gears into or out of mesh. Some of these gears are splined so that they can be moved axially along the shaft upon which they are mounted.

Two basic types of sliding-gear transmissions are available. In the progressive type, it is necessary to pass through the ratios in a definite order. The selective type, however, permits the selection of any gear ratio in any order. The progressive type is limited to motorcycles and similar vehicles, while the selective type is used in larger vehicles. The term sliding-gear transmission, as used in the present discussion, refers to the selective type.

For illustrative purposes, a sliding-gear transmission having three speeds forward and a reverse will be examined. Three shafts are present in this case: the input shaft, the countershaft, and the output shaft. The main or output shaft and the input shaft are mounted coaxially, but both rotate independently. The centerline of the countershaft is parallel to these. A constant mesh is maintained between a drive pinion on the input shaft and a drive gear on the countershaft. Different gear ratios are obtained by meshing different combinations of gears, free to slide on the main or output shaft, with fixed pinions on the countershaft. The main shaft is not engaged with the countershaft in the neutral position. Two different sets of gears engage the countershaft with the main shaft in low and in second gear. In high or drive gear, the main or output shaft is coupled directly to the input shaft by means of a sliding spline connector; hence, the gear ratio between the engine and the propeller shaft is 1:1, and the countershaft transmits no power. In reverse, a gear, which is free to slide on the main shaft, is placed in mesh with an auxiliary reversing gear which, in turn, is in mesh with a pinion on the countershaft. This causes the output shaft to rotate in an opposite direction.

The vehicle operator controls the position of the sliding gears on the main shaft by means of a pivoting gearshaft lever.

8-8.2 CONSTANT-MESH TRANSMISSION

Sliding-gear transmissions generally use stub-tooth gears for easy engagement; consequently, the transmission is generally noisy when operating in the intermediate speed range. For most current military vehicles, this transmission has been superseded by the constant-mesh type. The constant-mesh transmission retains the three-shaft arrangement of the sliding-gear type; however, the gears on the main shaft are no longer free to slide axially, but can rotate freely.

A clutch gear splined to the main shaft, and with external teeth which can mesh with corresponding internal teeth on the gears which are free to rotate on the main shaft, can move axially along the main shaft. Different speed ratios are obtained by shifting this gear into engagement with different internal gears. Helical gears are usually used to provide smoother and more quiet operation. Meshing between the rotating gears which contain the internal teeth and the nonrotating clutch gear results in tooth-clashing. A

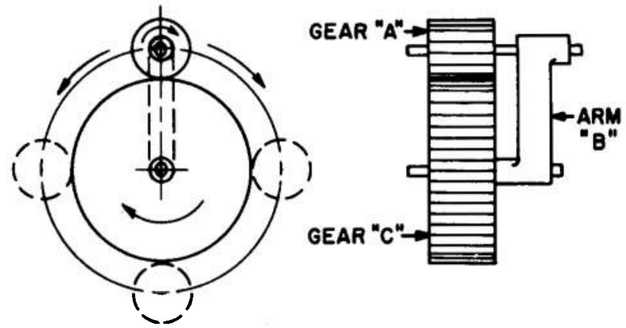


Figure 8-12. ***Schematic—Epicyclic Gear Train***

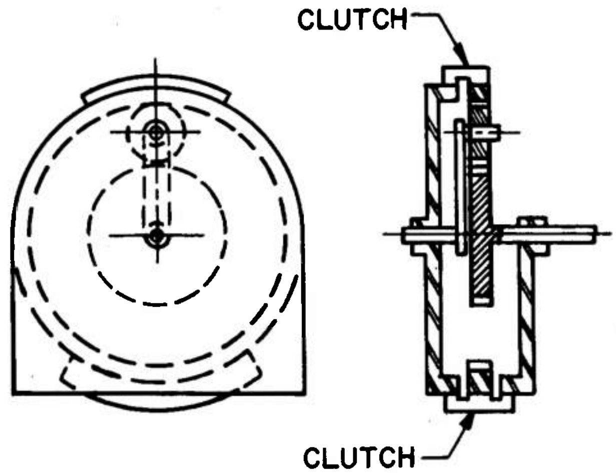


Figure 8-13. ***Schematic—Internal Epicyclic Gear Train***

given transmission may include both constant-mesh and sliding-gear elements. If sliding gears are used in this manner, they are normally used on first (starting) gear and reverse gear only.

8-8.3 EPICYCLIC TRANSMISSION

Epicyclic (planetary) gear trains are combinations of gears in which some or all of the gears undergo a compound motion consisting of rotation about an axis which in turn is moving on a circular path. Typical epicyclic gear sets are shown in Figs. 8-12 and 8-13.

In current military vehicles, epicyclic gear trains are employed in hydrodynamic transmissions and in track-laying vehicle transmissions. A brief discussion of such units is included in par. 8-9; comprehensive discussions are included in the references.

Several planetary gear sets are normally used in vehicle transmissions to achieve the desired number of gear ratios. These transmissions, within limits, automatically adjust the torque produced

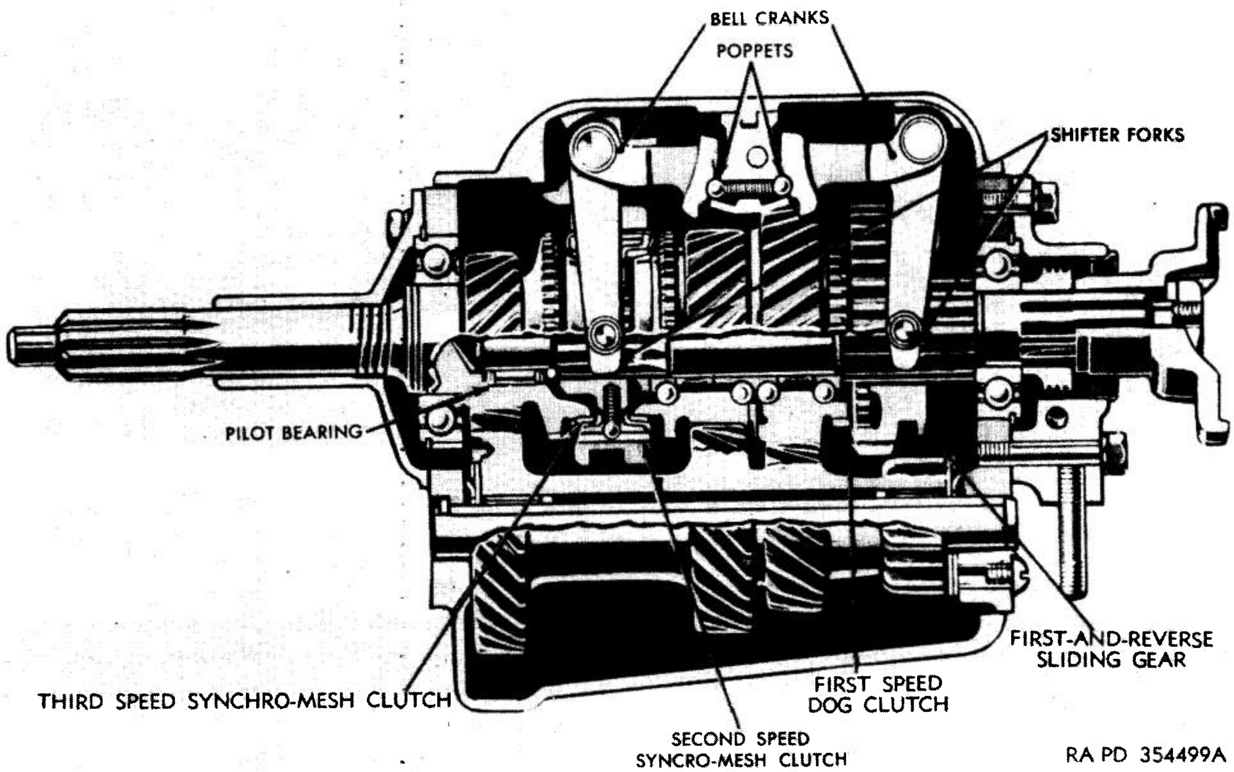


Figure 8-14. *Synchromesh Transmission*

by the engine to meet the torque requirements at the wheels.

8-8.4 SYNCHROMESH TRANSMISSION

The synchromesh constant-mesh transmission, Fig. 8-14, permits gears to be engaged without clashing by synchronizing the speeds of mating parts before engagement. It employs a combination friction and positive dog clutch to permit engagement of gears to the transmission main shaft. The friction clutch, often a cone type, is first engaged to bring the driven member to the speed of the drive before the dog clutch engages. This process is accomplished in one continuous operation when the operator declutches and moves the gear shift lever.

8-9 FLUID TRANSMISSIONS

Fluid transmissions may be divided into two classes, hydrodynamic and hydrostatic. In the hydrodynamic drive, transfer of power occurs from the transfer of kinetic energy of fluid in motion. In the hydrostatic drive, static pressure of the fluid is the primary means by which power is transferred.

Hydrostatic drives most generally employ positive displacement hydraulic pumps and motors, while hydrodynamic drives utilize fluid couplings and torque converters. The fluid coupling has been discussed in the section dealing with clutches and couplings.

8-9.1 HYDRODYNAMIC TRANSMISSIONS

8-9.1.1 Torque Converters

The hydraulic torque converter provides a continuously varying torque multiplication, within limits. Unlike the fluid coupling, which transmits torque without multiplication, the output torque of a converter, for a given value of input torque, varies with the speed ratio of turbine and impeller shafts. The output torque versus output speed characteristics of a torque converter transmission, for a power plant operating at constant power, closely approaches the torque-speed demands of an automotive vehicle. The torque developed at the wheels (or tracks) by the torque converter is maximum when the vehicle is stopped (output shaft stalled) and diminishes with increasing vehicle speed.

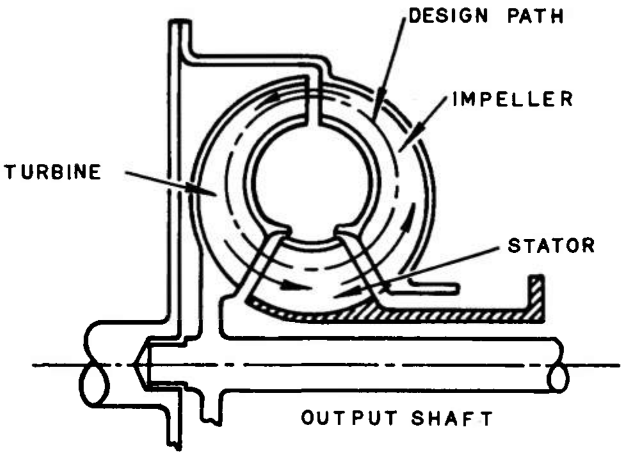


Figure 8-15. ***Single-Phase, Single-Stage Torque Converter (From "Torque Converters" by W. B. Gibson and R. W. Bachmann, Machine Design, April 14, 1960)***

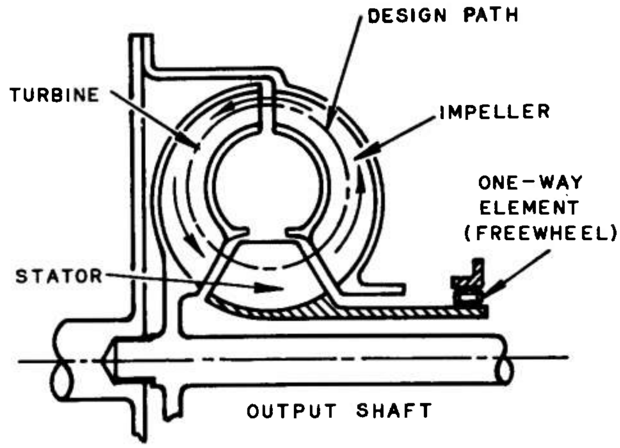


Figure 8-16. ***Two-Phase, Single-Stage Torque Converter (From "Torque Converters" by W. B. Gibson and R. W. Bachmann, Machine Design, April 14, 1960)***

The torque converter is similar to a fluid coupling in construction; however, an additional element is present. The torque converter consists of three principal elements, including an impeller, turbine, and a fixed reaction element (stator). The stator, located between the impeller and turbine, produces a change in the magnitude of the torque being transmitted. As previously mentioned, a fluid coupling has no stator, hence, no torque change occurs.

Figure 8-15 illustrates a single-phase, single-stage torque converter. The fluid is forced, by the impeller, through the turbine and thence through the stator blading. The change in direction of the fluid velocity introduced by the stator elements results in a change in momentum of the fluid. As the fluid circulates through a complete circuit, the total change in angular momentum of the fluid must be zero for equilibrium conditions (no acceleration or deceleration).

The torque applied to the fluid within the torque converter consists of T_P, impeller or pump torque; T_R, reaction torque of stator; and T_T, turbine reaction torque. For steady-state conditions, $T_P + T_R = T_T$. In a fluid coupling, $T_R = 0$ and $T_P = T_T$.

Converter torque characteristics are determined by the shape of the fluid circuit, the position of the various elements within the fluid circuit, and the number and shape of the blades in each of the working members.

Torque converters are classified by the number of reaction members or stages they contain and by their number of phases. A single-phase unit is one which can act only as a converter. A two-phase unit is one in which the reaction member of a single-stage converter is coupled to the converter frame by a one-way overrunning clutch which allows the reaction member to turn freely in one direction. When the reaction member is turning freely, a two-phase converter acts as a fluid coupling; hence it may act either as a converter or a fluid coupling. A polyphase torque converter has four or more functional elements. Several torque converters are shown schematically in Figs. 8-15 through 8-18.

Because the reaction element in a torque converter redirects the fluid back into the impeller with a minimum loss of momentum, the momentum imparted by the impeller need not be as great as that of a fluid coupling of equal size. The torque absorption capacity of a converter, therefore, will be less than that of a fluid coupling of equal size at a given speed ratio. Maximum torque occurs at stall.

If the load on a converter is increased, the turbine slows, decreasing the speed ratio (output over input) and increasing the torque absorbed. Because the curve of torque absorbed versus speed ratio is flatter for the converter than for the coupling, the torque absorbed by the converter as the speed ratio drops is less than for the coupling. Accordingly, the drop in speed of the engine is less than it is with a fluid coupling, and the engine can operate closer to its point of maximum

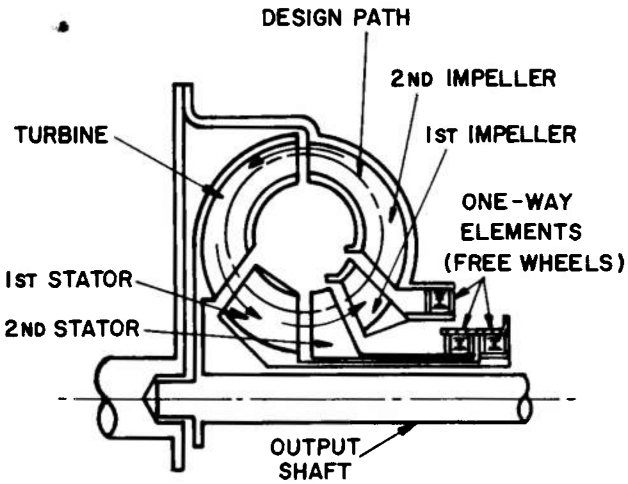


Figure 8-17. Polyphase Single-Stage Torque Converter (From "Torque Converters" by W. B. Gibson and R. W. Bachmann, Machine Design, April 14, 1960)

power, resulting in a higher tractive effort. Tractive effort to the driving elements remains high over a greater portion of the turbine output speed range for the converter. This feature permits a reduction in the number of gear ratios in combination gear-converter systems.

8-9.1.1.1 Torque Converter Performance Characteristics

The parameters generally used in the evaluation of converter performance are discussed in this section. *Primary* torque is the engine output torque which can be absorbed by the impeller at a given ratio. Primary torque is a function of input speed. *Speed ratio* is the ratio of output (turbine) speed to input (impeller) speed. *Efficiency,* η_c, is the ratio of output power, P_T, to input power, P_I, times 100. Alternately, η_c, may be written as

$$\eta_c = \frac{T_T\,N_T}{T_I\,N_I} \tag{8-5}$$

where T_T, N_T are the turbine torque and speed, and T_I, N_I are the corresponding values for the impeller.

Torque ratio is the ratio of output torque to input torque. *Stall torque ratio,* the torque ratio when the turbine shaft is stalled, is a function of input speed. Torque ratio at any point other than stall equals the ratio of efficiency to speed ratio.

Utility ratio is the ratio N_2/N_1 where N_1 and N_2 are the lowest and the highest speed ratios for which the efficiency exceeds 70%, Points A and B on Fig. 8-19. The 70% figure is chosen arbitrarily.

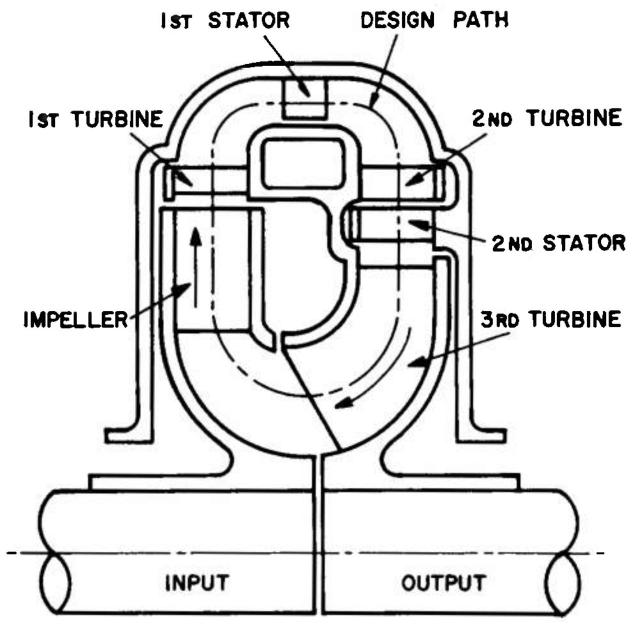


Figure 8-18. Single-Phase, Three-Stage Torque Converter (From "Torque Converters" by W. B. Gibson and R. W. Bachmann, Machine Design, April 14, 1960)

This ratio is indicative of the range over which the transmission may operate while dissipating 30% or less of the engine horsepower. The utility ratio is useful in determining the number of gear ratios which are required in a proposed transmission.

A comprehensive discussion of the automotive application of the torque converter is given in the references. The following paragraphs are a brief introduction to this subject.

8-9.1.1.2 The Function of Torque Converters in Automotive Power Trains

Typical characteristics of a multistage, single-phase torque converter are shown in Fig. 8-20. In this figure curves of output torque and efficiency as a function of output speed at different, constant prime mover speeds, N_{ii}, are shown. The converter is reasonably efficient over only a small range of output speeds; losses go directly into heating the working fluid. Because excessive heating may change the properties of the working fluid, oil coolers are often provided to keep the temperature at a reasonable level.

As shown in Fig. 8-20, the efficiency of the typical torque converter drops rapidly at the lowest and highest speed ratios of the operating range. If a converter of this type is used as the trans-

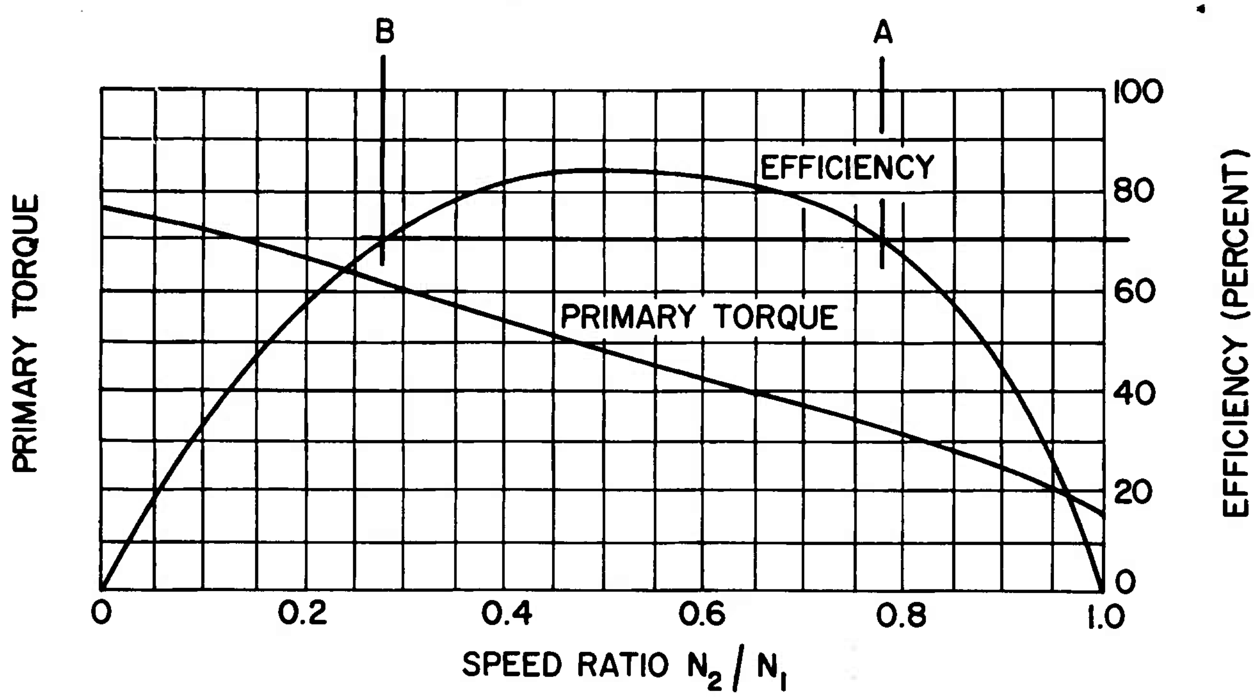


Figure 8-19. *Torque Converter Performance Characteristics (From "Torque Converters" by W. B. Gibson and R. W. Bachmann, Machine Design, April 14, 1960)*

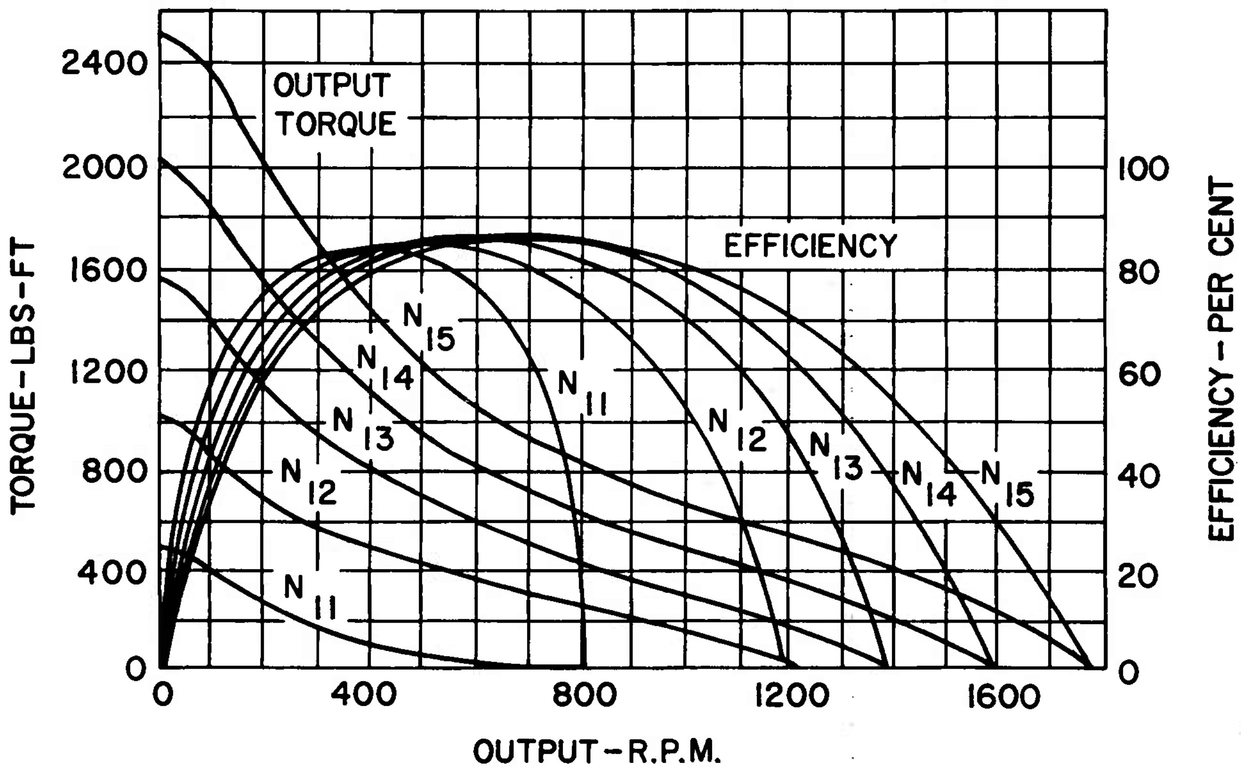


Figure 8-20. *Performance Characteristics of a Multistage, Single-Phase Torque Converter*

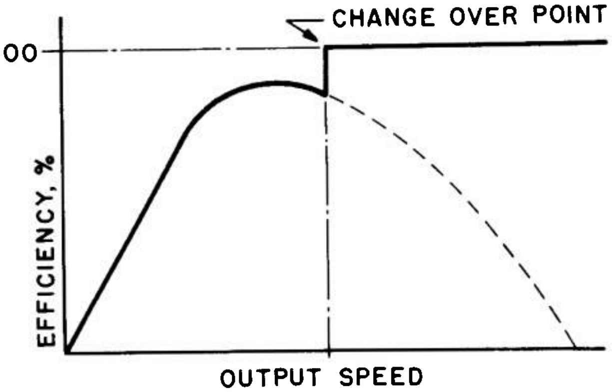


Figure 8-21. ***Efficiency Characteristics of a Direct Drive Torque Converter***

mission of a military vehicle, the severe operating requirements, e.g., slow vehicle speeds and high tractive effort for long periods, cause the torque converter to operate in a range of low efficiency, i.e., low speed ratio. For this reason, and to increase torque multiplication, the reduction range of a torque converter is usually extended by a mechanical transmission system, e.g., sets of planetary gears. To obtain higher efficiencies at higher speed ratios, the torque converter may be converted to direct drive, or the converter may be changed to operate as a fluid coupling when operating at these ratios.

8-9.1.1.3 Direct Drive Adapter

A double-faced clutch between engine and the torque converter provides the necessary control to switch the torque converter to a direct drive. By shifting the clutch, the engine can be connected either to the torque converter impeller or directly to the propeller shaft.

When the clutch engages the impeller, power reaches the propeller shaft through the converter. When the clutch engages the propeller shaft directly, the converter is bypassed. Because the converter is connected to the propeller shaft through a one-way overrunning clutch, the converter is entirely disconnected from the power train during direct operation. Engagement of the clutch can be controlled either manually or automatically. Automatic actuation is initiated by a centrifugal governor driven from the output of the torque converter. Figure 8-21 shows the characteristics of this converter arrangement.

8-9.1.1.4 Function of the Two-Phase Converter

A two-phase converter maintains a high efficiency at high speeds by converting to fluid-coupling operation at such speeds. The impeller is attached to the engine output shaft, the turbine to the converter output shaft. The reaction members, or stators, mounted on one-way overrunning clutches, are constrained from backward rotation but are free to rotate in a forward direction.

When the torque on the turbine is greater than the engine torque ($T_T/T_I > 1$), the torque reaction on the stator causes it to remain stationary. If the torque on the turbine falls below that of the engine ($T_T/T_I < 1$), the torque reaction on the stator causes it to rotate freely in the forward direction. The torque converter then acts as a fluid coupling and efficiencies over 95%, at high-speed ratios, are attained. The change from converter to coupling occurs automatically whenever the torque transmitted attains a 1:1 ratio. Part-throttle performance for this type of converter is good. Figure 8-22 represents the torque ratio and efficiency characteristics of this arrangement.

Many arrangements of the basic elements within a torque converter are possible. Complete discussions on the practical and theoretical considerations of these systems are given in the references.

8-9.1.1.5 Torque Converter Combinations

In some units a multiphase torque converter is used in combination with epicyclic gearing. These units, by use of a combination of the properties of the converter and of gearing, provide a variable torque which is inversely proportional to the output or drive shaft angular velocity. At low forward speeds, high torque is provided, while at high speeds, the entire unit acts as a fluid coupling.

These units are used extensively in automotive passenger and heavier vehicles including some tank-like units. The previously cited references provide a detailed discussion of these units.

8-9.1.1.6 Automatic Transmissions

The previously discussed sliding-gear and constant-mesh transmissions are normally manually operated, i.e., the operator selects the combination of gears to produce the reduction ratio required. The suitability of the reduction ratio selected to the operating requirements is a matter of operator judgment.

Automatic transmissions, currently in use in the United States, are of two basic types: (1) the fluid coupling-planetary gear transmission and (2) the torque converter-planetary gear transmission. Both types incorporate complex epicyclic gear trains and control systems for automatically matching the tractive effort to the operating requirements within the limitations of the hydraulic and mechanical systems of the automatic transmission. The factors of vehicle speed, load, and throttle position serve as control parameters for the control system of the transmission. To minimize the complexity of the system, some degree of manual control is frequently incorporated into the system. Thus, an operator may be able to select a low or high range for conditions requiring either maximum tractive effort or a high level of economy.

Descriptions of various automatic transmissions and their methods of operation are given in the references.

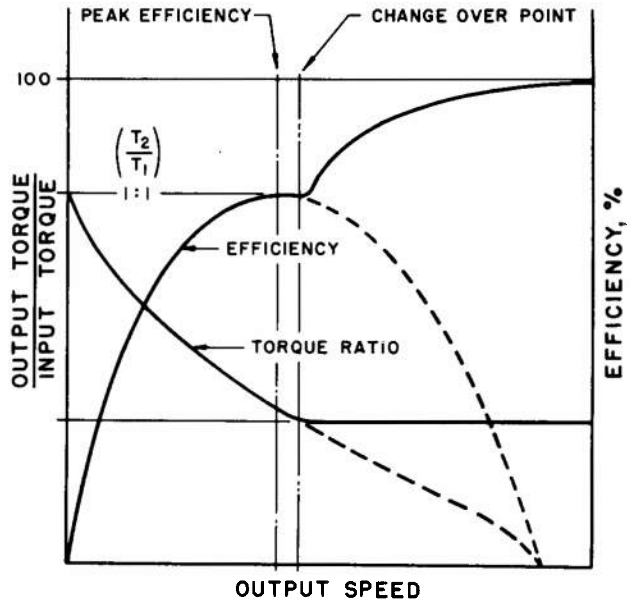


Figure 8-22. ***Performance Characteristics of a Multistage, Two-Phase Torque Converter***

8-9.2 HYDROSTATIC TRANSMISSIONS

Essential elements of a hydrostatic drive are shown in Fig. 8-23. The output shaft of the power source is coupled to the rotor shaft of a positive displacement hydraulic pump. The pump supplies one or more positive displacement hydraulic motors. These motors (or motor) are, in turn, mechanically connected to the propulsive elements of the vehicle.

At the present time, the only applications of hydrostatic drives to military vehicles have been experimental in nature. The possibilities, inherent in their use, will undoubtedly result in additional applications, such as self-propelled weapons, auxiliary-propelled artillery, and lightweight, off-the-road vehicles. The exact configuration of a given hydrostatic drive would depend upon the application; however, general principles may be discussed below.

A system using a variable displacement pump, fixed displacement motors, and suitable control valving is shown schematically in Fig. 8-23. An internal combustion engine drives a variable-displacement, pressure-compensated, positive-displacement hydraulic pump through a gear reducer. The gear reducer is necessary because the optimum operating speed of a piston engine is greater than the corresponding optimum operating speed for the pump. Operational speeds for vane-type hydraulic pumps and motors are approximately 2000 rpm. Speed variation at the wheel is obtained by varying the output flow rate of the pump. The need for clutches is eliminated because the pump output can be reduced to zero. Forward and reverse flow to the motors is obtained by means of a manually-controlled, 3-position, 4-way valve. With the control valve in the central position, a solid fluid lock is formed and an effective parking brake is realized.

The circuit modification shown as dotted lines in Fig. 8-23 will provide for braking if the operating valve is placed in the central position. Fluid from the output of the driving motors passes through a variable restriction, V, and suffers a pressure drop. The pressure at the outlet port of the driving motors, now acting as pumps, rises. The increased torque on the motor shaft slows the motor and the vehicle. Makeup driving fluid is obtained from the sump through check valve, R.

The output of each motor drives the vehicle tractive element through a planetary gear train. The planetary gear train acts as a gear reducer

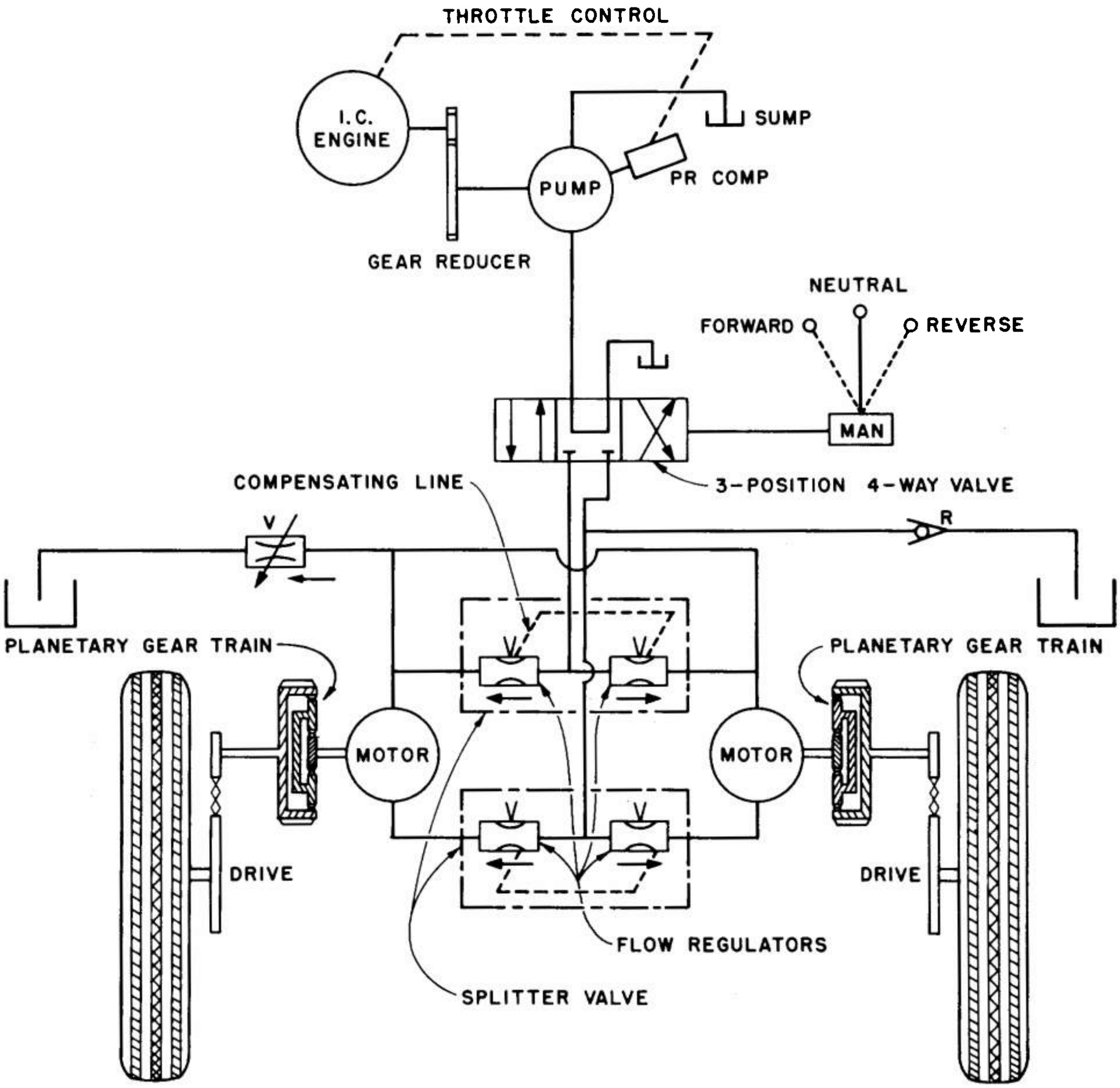


Figure 8-23. ***Hydrostatic Drive System***

(equivalent to a final drive unit), as a clutch, allowing disengagement of the transmission, and to assist in the regenerative braking of the vehicle.

Other possible configurations for the hydrostatic drive exist. A constant displacement pump with variable displacement motors at each tractive element offers improved control of each individual wheel for maneuvering and steering. The use of duel hydraulic motors at each tractive element provides dual speed ranges at each drive point similar to that available from a two-position transmission.

The most commonly proposed systems utilize at least one motor for each tractive element of the vehicle. If more than one motor is used, flow regulators must be employed in the hydraulic circuit to ensure equal flow to each motor. If the vehicle is a track-laying vehicle, these flow regulators can provide the means for steering the vehicle. By increasing flow to the motor of one track and decreasing it to the motor of the other, a difference in track speed is attained, causing the vehicle to move in a curved path.

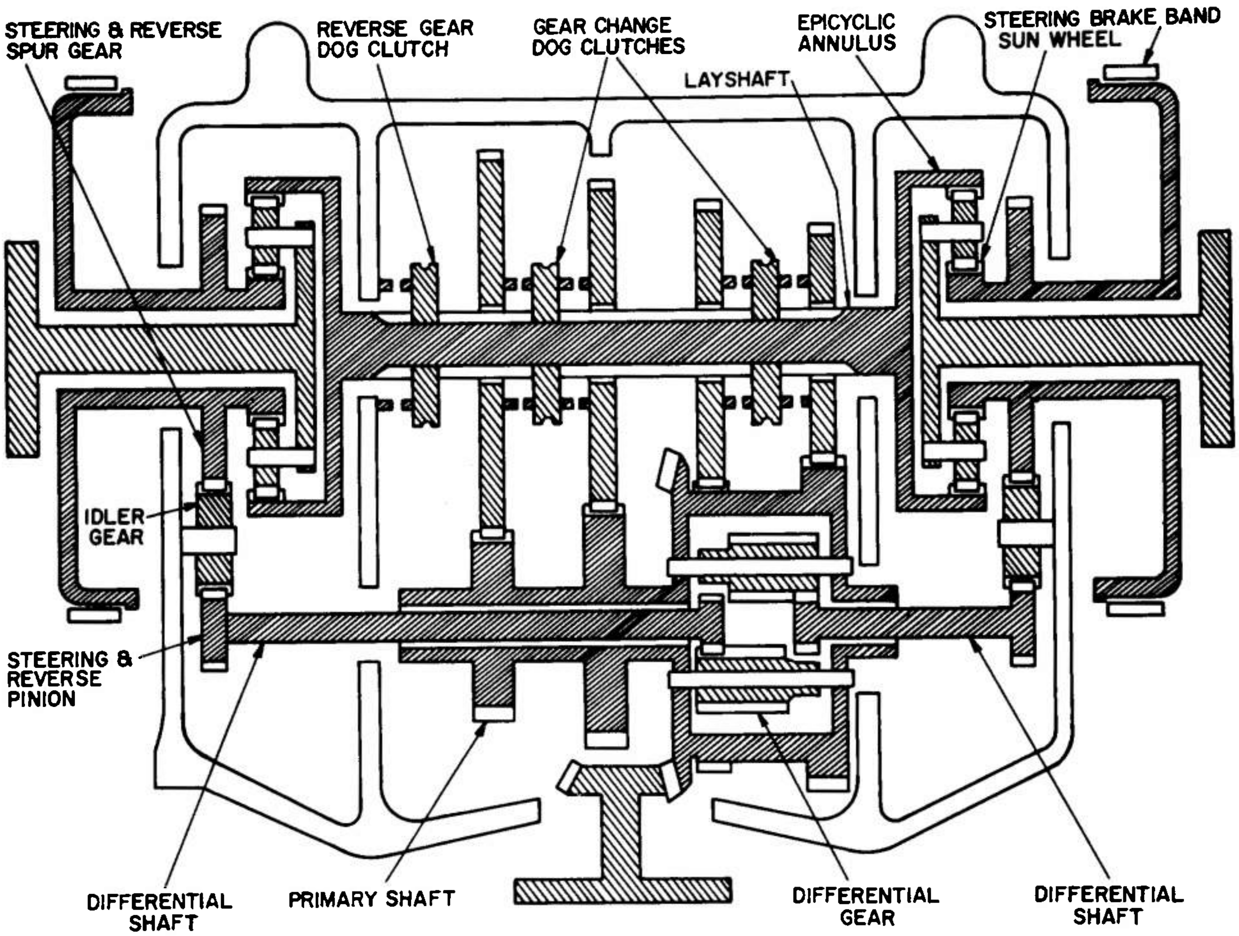


Figure 8-24. *Merritt-Brown Cross-Drive Transmission*

Differential action may be obtained by varying the flow rates to driving motors on either side of a nonsteerable axle in a wheeled vehicle.

8-9.3 TRANSMISSIONS FOR TRACK-LAYING VEHICLES

The transmission for track-laying vehicles must be capable of: (1) varying the gear ratio between engine and sprocket to meet varying operating conditions, (2) providing the required power at the sprocket at the most economical engine speed, (3) reversing the direction of sprocket rotation under power, and (4) regulating the relative speed of the tracks for vehicle steering.

Some typical basic design requirements of tank transmissions are: (1) they may be required to transmit the torque necessary to operate vehicles weighing 60 tons or more; (2) they must operate satisfactorily in ambient temperatures ranging from −65°F to +125°F; (3) they must provide the torque multiplication to permit the vehicle to climb a 60% grade at 2.5 mph; and (4) they must be simple to operate and must not require great effort to operate.

The steering efficiency of a track-laying vehicle transmission varies with the design of the mechanism; for example, the simple clutch-brake system is satisfactory when the length/tread ratio is less than 1.3:1. For greater length/tread ratios, the clutch-brake system is inadequate and regenerative steering must be used.

8-9.3.1 Cross-Drive Transmission

The cross-drive transmission, mounted crosswise in the vehicle, is composed of a hydraulic torque converter, an epicyclic gear train giving two speeds forward and one in reverse, and hydraulically controlled planetary gear sets for steering.

A typical planetary gear set for steering is shown in Fig. 8-24. The transmission illustrated regulates the relative speed of the tracks by com-

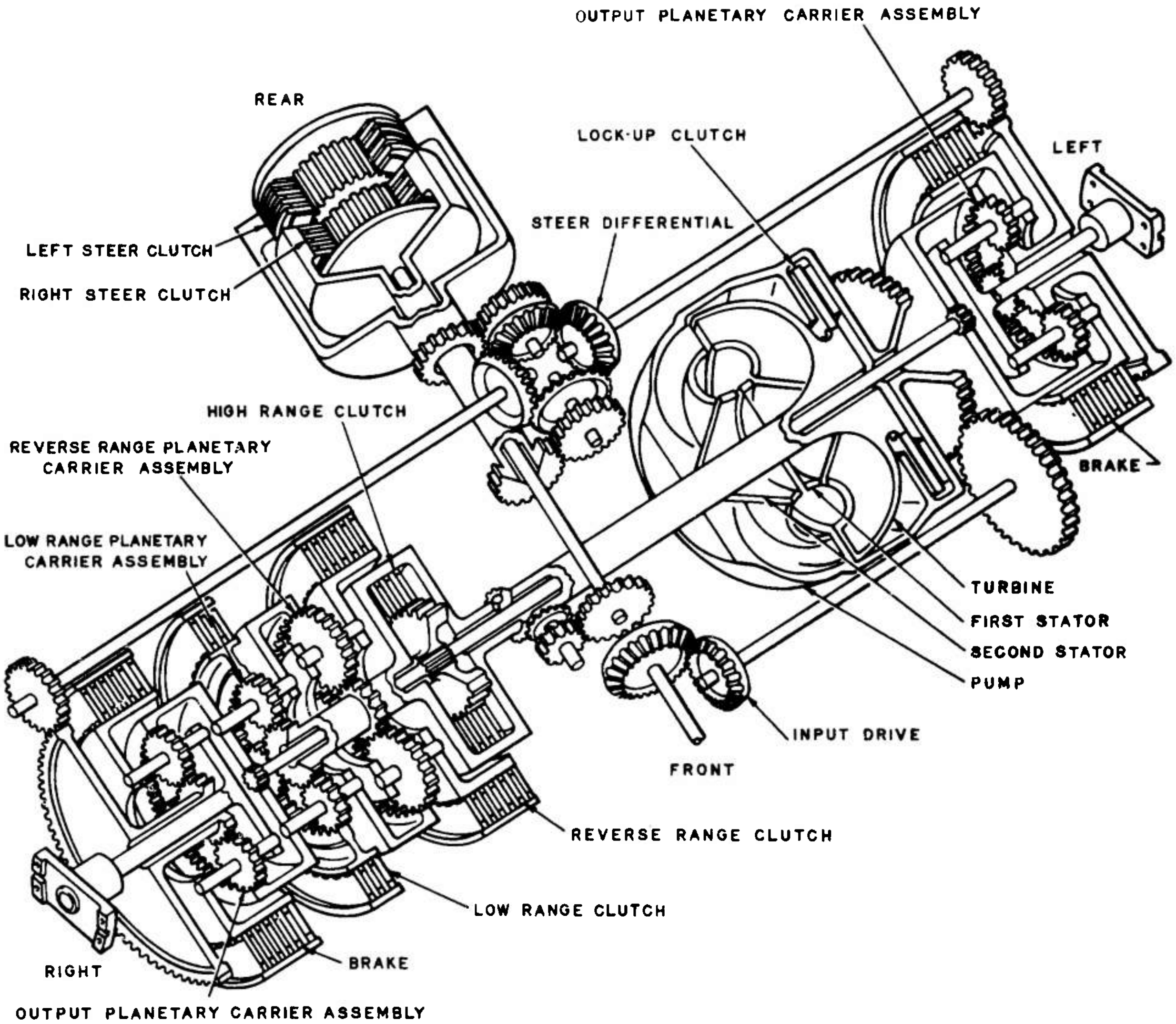


Figure 8-25. *Cross-Drive Transmission, CD-500*

bining the output speed of a cross-connected differential through a planetary gear train at each sprocket drive. The degree of turn is controlled by a steering brake on each planetary gear and by the transmission speed ratio. This transmission, therefore, provides a geared turning radius for each transmission speed. The transmission also provides regenerative steering and provides pivoting about a vertical centerline with the transmission in neutral.

Another cross-drive transmission designed for use with engines of 550 bhp to 850 bhp consists of a single-stage polyphase unit. Two planetary gear sets coupled to the torque converter provide high, low, and reverse speeds. The gear sets are coupled to the steering differential which drives two output planetary gear sets. Steering brakes are multiplate frictional.

The power path through a CD-850 cross-drive transmission is in part mechanical and in part hydraulic. The mechanical power path passes through a steering differential on the torque converter input (impeller) shaft, and through two steering drive gear shafts, to the output planetary gear sets. In these output gears, it is combined with the torque converter output.

In low range, the percentage of power through the mechanical power path varies from 60% to

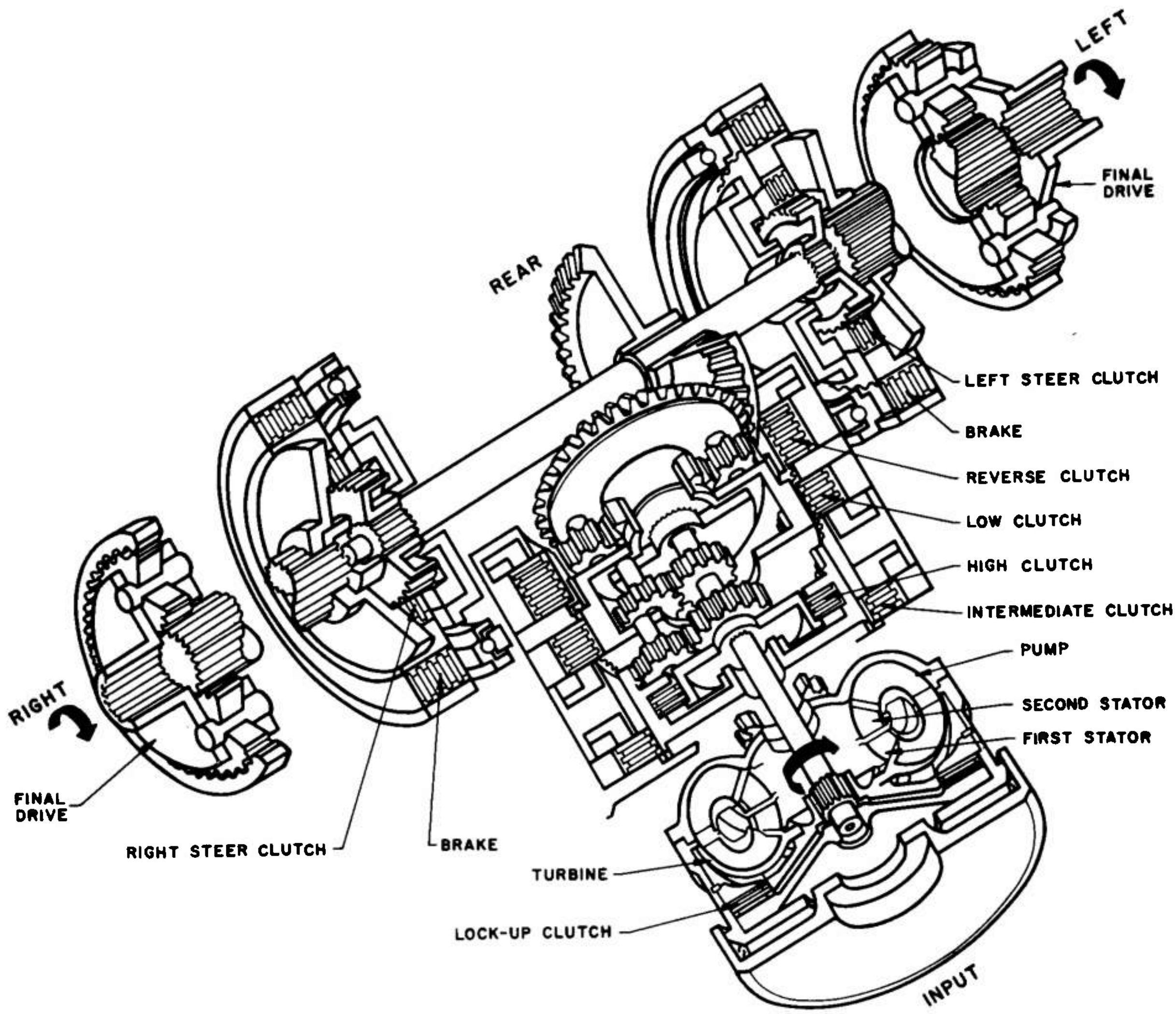


Figure 8-26. *Transmission, XT-500*

15% with increasing vehicle speed. In the high range, the power through the mechanical power path varies between 29.5% and 5.8%.

Cross-drive transmissions for 375-500 bhp engines, Fig. 8-25, incorporate a lockup clutch which, at predetermined speed in high gear, locks the converter members together. The split torque power path of the larger units is not employed in these smaller systems. Here the entire engine output passes through the torque converter.

The use of a cross-drive transmission permits the combination of the power train elements, engine, cooling system, and transmission into a single-unit power package, with consequent savings in weight and size.

8-9.3.2 XT Series Transmission

The XT series, Fig. 8-26, represents a recent development in steering transmission for track-laying military vehicles. This transmission has considerably fewer components than the cross-drive transmission and can be produced at less cost. The XT series transmission also provides a more effective middle-speed range performance with attendant increases in fuel economy.

A major advantage in the XT series over the cross-drive transmission lies in the interchangeability of components. The major transmission components of the smaller XT series units, used in lightweight track-layers, are interchangeable with those of the TX transmission for wheeled vehicles.

The XT series transmission is composed of a single-stage polyphase torque converter, a lockup clutch, and a reverse planetary transmission providing three speeds forward and a reverse. The converter lockup clutch of this transmission operates in both the high and intermediate speed ranges. It has throttle-velocity control, in which lockup is governed by a combination of engine manifold vacuum and transmission governor speed. The lockup clutch engages automatically at low vehicle speeds and low torque requirements. It is automatically disengaged as the torque demand increases, or the throttle is opened to permit torque converter operation.

Three types of steering systems are being developed for the XT series transmissions. A clutch-brake steering system and a geared steering system are under development for vehicles below 40 tons, while a controlled planetary system, producing regenerative steering, is under development for larger vehicles.

8-9.3.3 Electric Transmission

Electric drives for track-laying vehicles are still in the experimental state. Such drives have a number of desirable features for vehicle application. Here a generator, driven by an internal combustion engine, provides power to operate electric motors which drive each track. Infinitely variable drive and steering ratios are available. The internal combustion engine runs continuously at maximum power or maximum economy.

Electric drives were first tried in French WW I light tanks and, later, in very heavy assault tanks of the late 1920's. These latter vehicles saw service in World War II. The drives were successful, but extremely bulky. Drives for both light and heavy American tanks have been tested, but none is in operational use.

8-9.3.3.1 Electrogear System

The electrogear system is a differential transmission wherein electrical and mechanical power are combined by means of a planetary gear train. During starting, and during other high torque-demand periods, all of the prime mover power is converted to electric energy and then reconverted to mechanical energy at the driving elements. Under normal operating conditions, somewhere between 5% and 30% of the engine power is converted to electrical energy.

8-9.3.3.2 General Electric System

The General Electric system consists of a main generator coupled to the engine and two track motors, one at either track, electrically connected in series. A small amplidyne exciter generator is used with each track motor and with the main generator. Operation of the track motors is controlled by amplidynes which, in turn, are manually controlled by means of rheostats.

In this system, steering is accomplished by varying the field strength of one track motor with respect to that of the other. In a tight turn, the field current of one motor may be reversed, which makes it act as a generator and supply current to the opposite track motor to produce regenerative steering.

Braking is accomplished by reversing the field current of both track motors, causing them both to act as generators. Output of the generators is dissipated in a resistor cooled by the engine exhaust.

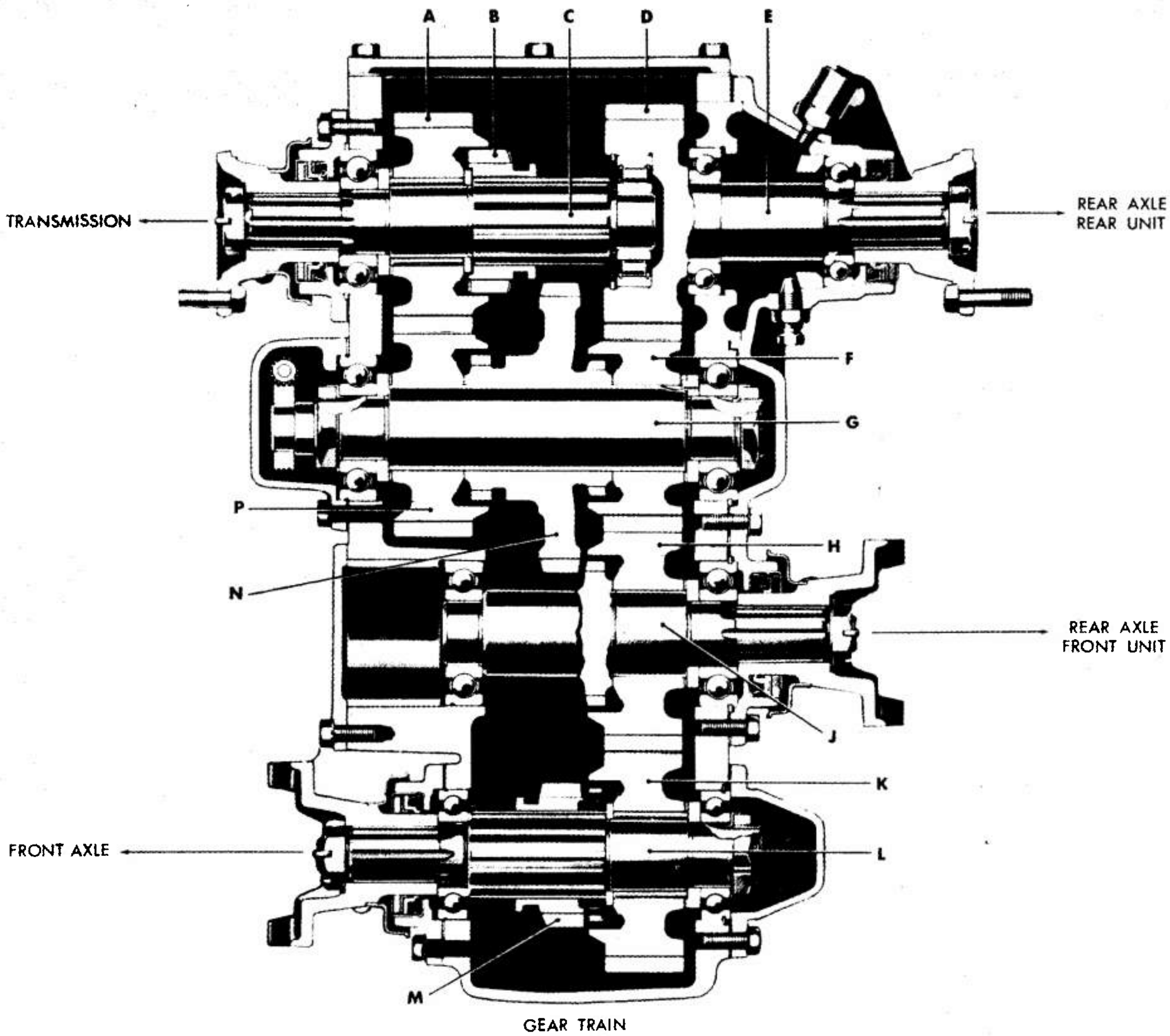
8-9.3.3.3 General

The excessive weight of electric drives as compared with standard drives has resulted in the abandonment of most experimental programs. The potential of this concept is high, however, and design studies continue. The proposed use of the gas turbine as a power source appears to offer a new opportunity for the electric drive. The electric transmission system is well suited to handle the relatively small speed range in which the turbine is reasonably efficient. Electrical components would be smaller because of the high rotational speed of the turbine.

SECTION IV TRANSFER CASES

8-10 PURPOSE AND OPERATION

The transfer assembly is an auxiliary gear train which enables engine power to be divided or transferred to more than one axle. The transfer case permits the forward propeller shaft to be placed to one side of the vehicle centerline, thus passing to one side of the engine crankcase, resulting in a more favorable ground clearance. The transfer unit is essentially a two-speed transmission, which provides low and direct drives. Figure



A—MAINSHAFT CONSTANT MESH GEAR
B—MAINSHAFT SLIDING GEAR
C—MAINSHAFT
D—REAR AXLE (REAR UNIT) DRIVE GEAR
E—REAR AXLE (REAR UNIT) DRIVE GEAR ASSEMBLY
F—IDLER SHAFT CONSTANT MESH GEAR
G—IDLER SHAFT
H—DRIVE SHAFT CONSTANT MESH GEAR
J—REAR AXLE (FRONT UNIT) DRIVE SHAFT
K—DRIVE SHAFT CONSTANT MESH GEAR
L—FRONT AXLE DRIVE SHAFT
M—DRIVE SHAFT SLIDING GEAR
N—IDLER SHAFT LOW SPEED GEAR
P—IDLER SHAFT CONSTANT MESH GEAR

RA PD 183950

Figure 8-27. ***Transfer Case Assembly, Cross Section***

8-27 shows typical transfer unit for a dual rear wheel vehicle. A number of transfer units are shown in Fig. 8-28.

The typical transfer case may be operated in either of two modes. In one, the front wheels are driven; in the other, they are not. In addition, the transfer case may provide either of two drive speeds, high or low. The transfer from high to low speed is accomplished by manually shifting a gear on the main drive shaft from engagement with the drive (high) gear to engagement with the low-speed pinion. Front-wheel drive can be engaged by shifting a sliding gear or a dog clutch into engagement with a driven gear on the front-wheel drive shaft. The sliding gears and dog clutches are positively driven by means of splines.

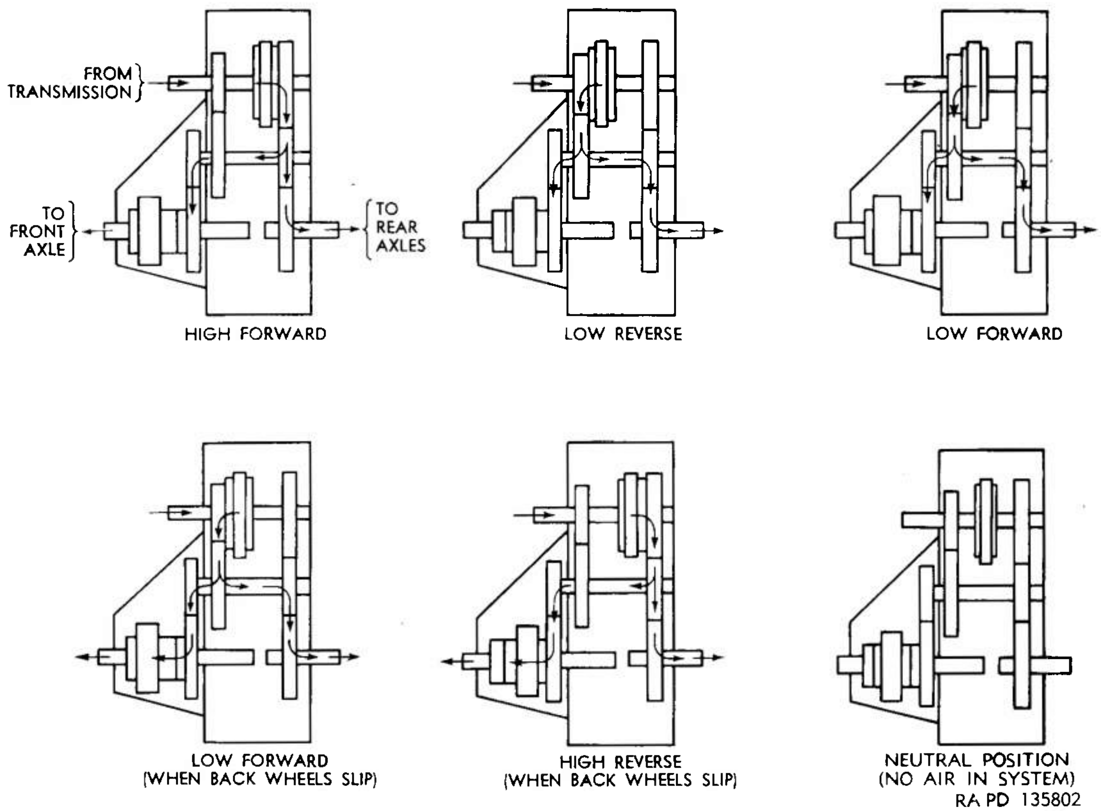


Figure 8-28. ***Transfer Case, Power Train Diagrams***

8-11 TRANSFER UNITS WITH OVERRUNNING SPRAG CLUTCHES

Transfer units may contain one or more overrunning sprag clutch units on the front output shaft. The transfer unit is designed to drive the front axle at a slightly slower speed than the rear axle. During normal operation, when both front and rear wheels turn at the same speed, only the rear wheels drive. However, if the rear wheels should lose traction and begin to slip, they tend to turn faster than the front wheels. At this point, the sprag clutch engages and the front wheels drive. Two types of sprag-clutch-equipped transfer assemblies are in common use, the single and the double sprag-clutch units. Both types function in a similar manner during normal vehicle operations in a forward direction, but their functions differ slightly when the vehicle is operating in reverse. Since the overrunning feature of a sprag clutch permits power to be transmitted in one direction only, the double sprag-clutch unit incorporates a second sprag clutch to transmit power to the front wheels during reverse operations.

8-11.1 SINGLE SPRAG-CLUTCH UNIT

The outer race of the sprag clutch is attached to the driven gear of the front wheels, while the inner race is attached to that output shaft of the transfer case which leads to the front wheels. During normal operation, the outer race of the sprag unit turns more slowly than the inner race. The sprags are thus free and the entire unit acts as an overrunning clutch. However, when the rear wheels lose traction, their rotational speed increases. The outer race tends to turn faster than the inner, causing the sprags to wedge between the inner and outer races, and drive power is provided to the front wheels. In reverse, the sprag clutch is locked out by a manually-actuated, reverse shift collar. Power to front wheels is supplied by a positive drive gear train.

8-11.2 DOUBLE SPRAG-CLUTCH UNIT

The operation of the double sprag clutch differs only in reverse. A second sprag clutch provides for engagement of the front wheel drive when the vehicle is operating in reverse.

8-11.3 GENERAL

Engagement and disengagement of the sprag units of both single and double sprag clutches can be accomplished pneumatically. This system provides a parking lockout feature when the transmission lever is in reverse.

SECTION V DRIVE SHAFT ASSEMBLY

8-12 PROPELLER SHAFT

Generally the propeller shaft (drive shaft) in an automotive vehicle transmits power from the transmission to the differential. Power trains have been built, however, with the propeller shaft between the power source and the transmission. In amphibious vehicles the propeller shaft has a second function, that of transmitting power to the screw propeller.

The propeller shaft is generally circular in cross section, being either tubular or solid depending upon the design criterion used for the particular application in question. A tubular configuration represents the most efficient means for transmitting torque for a given weight of material. For equal outside diameters and materials, a solid shaft is stronger and more rigid than a hollow shaft.

For a shaft undergoing a torsion, T (in-lb), the shear stress, S_s (psi), is given by

$$S_s = \frac{Tr}{J} \qquad (8\text{-}6)$$

where r is the shaft radius (in), and J, the shaft polar moment (in^4). For a solid shaft, $J = \frac{\pi r^4}{2}$, while for tubular shafts $J = \frac{\pi}{2} (r_o^4 - r_i^4)$ where r_o, r_i are the outer and inner radii of the shaft.

The torque, T (in-lb), being transmitted by a shaft rotating at N rpm while transmitting a power level, HP, is

$$T = \frac{63{,}025}{HP \times N} \qquad (8\text{-}7)$$

In the foregoing equations, the shaft is assumed to be subject only to a torsional loading. In practice, all propeller shafts are subject to both bending and torsion. Because the loads applied to the shaft are fluctuating, service factors must be introduced into design equations. Several standards for shafting design have been formulated, including codes by both the American Society of Mechanical Engineers (ASME) and the Westinghouse Company.

The ASME Code for the Design of Transmission Shafting, B17c-1927, considers both combined loading and the effects of load fluctuation. The ASME Code, based on the maximum shear theory of failure, indicates the allowable stress to be given by

$$(S_s)_{max} = \frac{0.5\ S_{yp}}{FS} \qquad (8\text{-}8)$$

where the stress existing is given by

$$(S_s)_{max} = \frac{r_o}{J} (K_m M)^2 + (K_t T)^2 \qquad (8\text{-}9)$$

where

S_{yp} = yield point stress, in. tension, psi
FS = factor of safety
r_o = the outside radius of the shaft, in
J = polar moment of inertia of the shaft, in^4
M = bending moment, in-lb
T = applied torque, in-lb
K_m = bending moment combined shock and fatigue factor
K_t = corresponding factor for applied torque

Values of K_m and K_t are given in Table 8-1.

The ASME Code recommends the allowable working stress, $(S_s)_{max}$, obtained by means of Eq. 8-8 be less than 30 percent of the elastic limit in tension and not more than 18 percent of the ultimate tensile strength. If keyways or fillets are present to produce stress concentration, it is recommended that the allowable stress given by Eq. 8-8 be reduced 25 percent, and where failure of the shaft would be a serious matter, an additional 25 percent reduction in allowable stress is proposed.

The Westinghouse Code is also based on the maximum shear theory of failure. Values of K_m and K_t are found by use of equations that depend upon physical properties of the shaft material, nature of load fluctuation, and stress concentra-

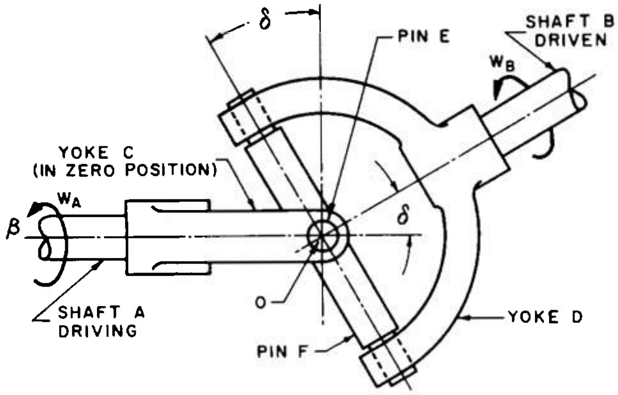


Figure 8-29. ***Schematic Diagram of a Hooke's Joint***

tion. Applied torque and the bending moment are divided into average and range components, and the range components are multiplied by appropriate stress concentration factors. Detailed data regarding propeller shaft design are presented in Refs. 5 and 44-52.

8-13 UNIVERSAL JOINTS

A universal joint is a connection between two shafts whose axes intersect at some angle less than 90°, which angle can change with time during rotation.

8-13.1 HOOKE'S JOINT

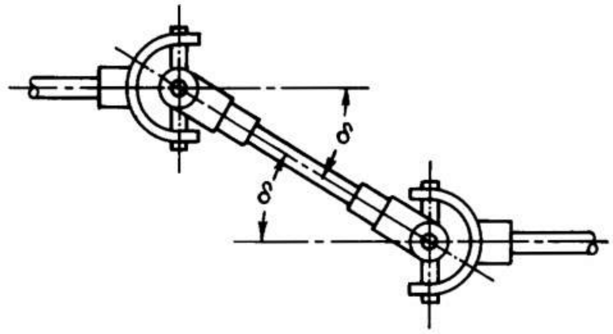
The Hooke's joint, or cross type universal joint, is commonly used in automotive drive shaft (propeller shaft) assemblies. Figure 8-29 illustrates such a universal joint.

Shaft *A* is the input or driving shaft, shaft *B* is the output or driven shaft. The motion of shaft *A* is transmitted through yoke *C*, cross pivot pins *E* and *F*, and yoke *D* to shaft *B*. The ratio of instantaneous velocities of *B*, ω_B rad per sec to that of *A*, ω_A rad per sec, is given by

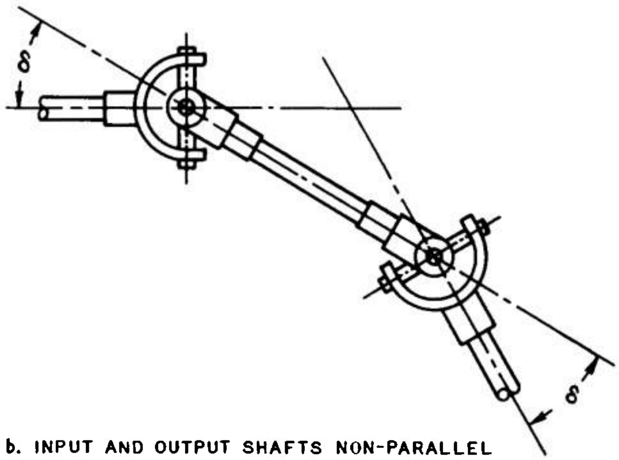
$$\frac{\omega_B}{\omega_A} = \frac{\cos\delta}{1-\sin^2\beta\,\sin^2\delta} \tag{8-10}$$

where β is the angular displacement of shaft *A* as shown in Fig. 8-29. For constant, δ, ω_B/ω_A is a maximum at $\beta = 90, 270°$, and a minimum at 0, 180°. If the input revolutions per minute are held constant, the output revolutions per minute vary with β.

To avoid output rpm variations, two Hooke's joints, Fig. 8-30, are employed. The angles between shafts, δ, must be equal to obtain a constant



a. INPUT AND OUTPUT SHAFTS PARALLEL



b. INPUT AND OUTPUT SHAFTS NON-PARALLEL

Figure 8-30. ***Double Hooke's Joint***

output rpm. The efficiencies of double Hooke's joints for various shaft angles are shown in Fig. 8-31.

8-13.2 THE CONSTANT VELOCITY UNIVERSAL JOINT

Speed fluctuations through a Hooke's joint may be quite large if the shaft angle, δ, is large. If this angle remains small, however, little speed variation is experienced. For automotive drive shafts (propeller shafts) which normally use a pair of Hooke's joints in series, speed variation is very small.

In front wheel drive vehicles where the wheels may be cramped up to 30° in steering, velocity fluctuations can be serious. In this application, constant velocity universal joints are used exclusively. At the present time, three constant velocity universal joints are commonly used in military

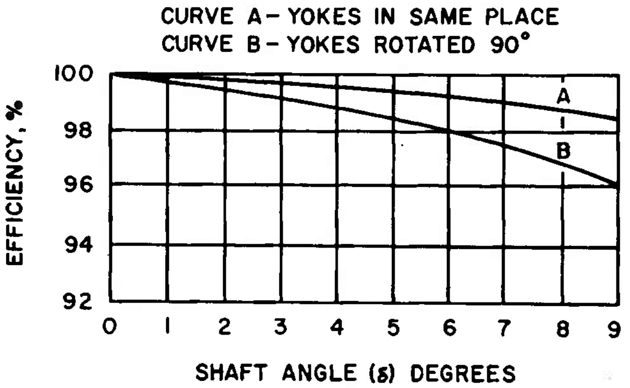


Figure 8-31. *Efficiencies of Double Universal Joints*

vehicles. These include the Rzeppa joint, the Bendix-Weiss joint, and the Tracta joint.

8-13.2.1 The Rzeppa Joint

In the Rzeppa universal joint, Fig. 8-32, motion is transmitted by hardened steel balls rolling in grooved race ways. Constant velocity is achieved by virtue of the ball groove geometry, which maintains the driving balls and cage in a predetermined relative position at all times. The construction of the ball groove is such that the balls and their cage are compelled to lie in a plane which bisects the angle between the driving and driven shaft regardless of the shaft angle. Because the balls remain in this bisecting plane at all times, constant output angular velocity is achieved.

8-13.2.2 The Bendix-Weiss Joint

The Bendix-Weiss joint, Fig. 8-33, utilizes balls for driving contact. Construction differs from that of the Rzeppa joint in that the balls are tightly fitted between the two halves of the coupling, no cage being required. The center ball, which rotates on a pin inserted on the outer race, serves as a locking medium for the four other balls. Driving contact remains in the plane which bisects the angle between the two shafts; rolling friction between the balls and the universal joint housing locates the balls.

Action of the Bendix-Weiss joint is similar to that of a differential. With the inner race stationary, and the outer race movable, the balls act as the middle member in a simple three-member differential in which motion of the balls is proportional to the relative motion of the two races. The balls move half the distance that the outer race moves. Since the motion of the outer race is proportional to shaft angle, the balls lie in a plane which bisects this angle. Consequently the points of drive for the joint also lie in the plane defined by the bisector of the shaft angle, and constant velocity transmission is accomplished.

8-13.2.3 The Tracta Joint

The Tracta joint, Fig. 8-34, consists of one universal joint within another. The points of driving contact are at the outer portions of the joint. The joint consists of four major elements

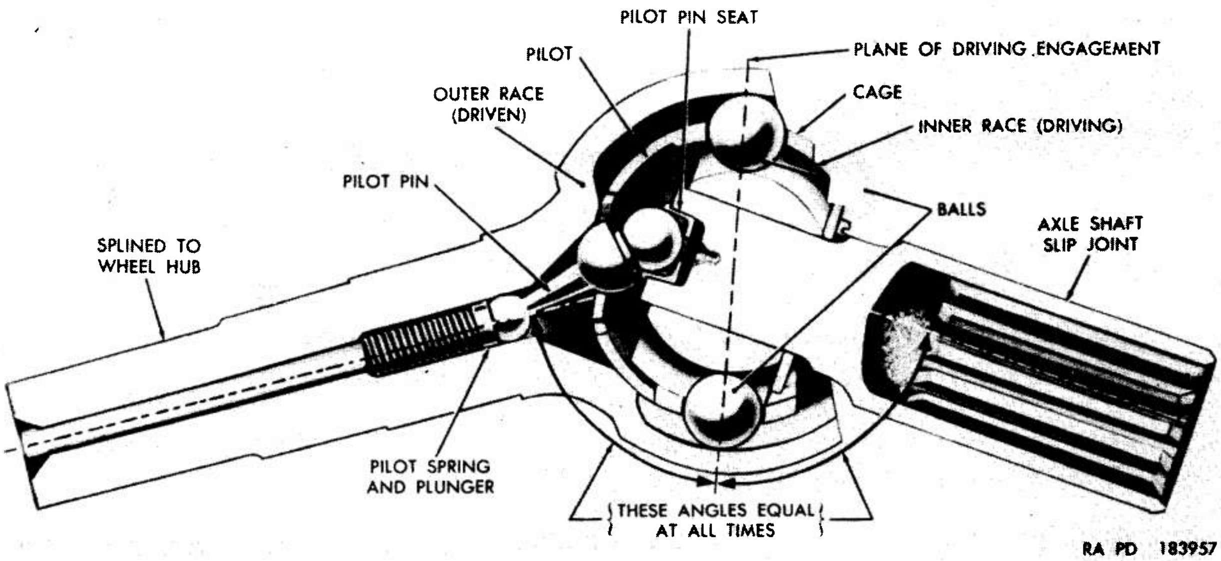


Figure 8-32. *Rzeppa Constant Velocity Universal Joint—Cross Sectional View*

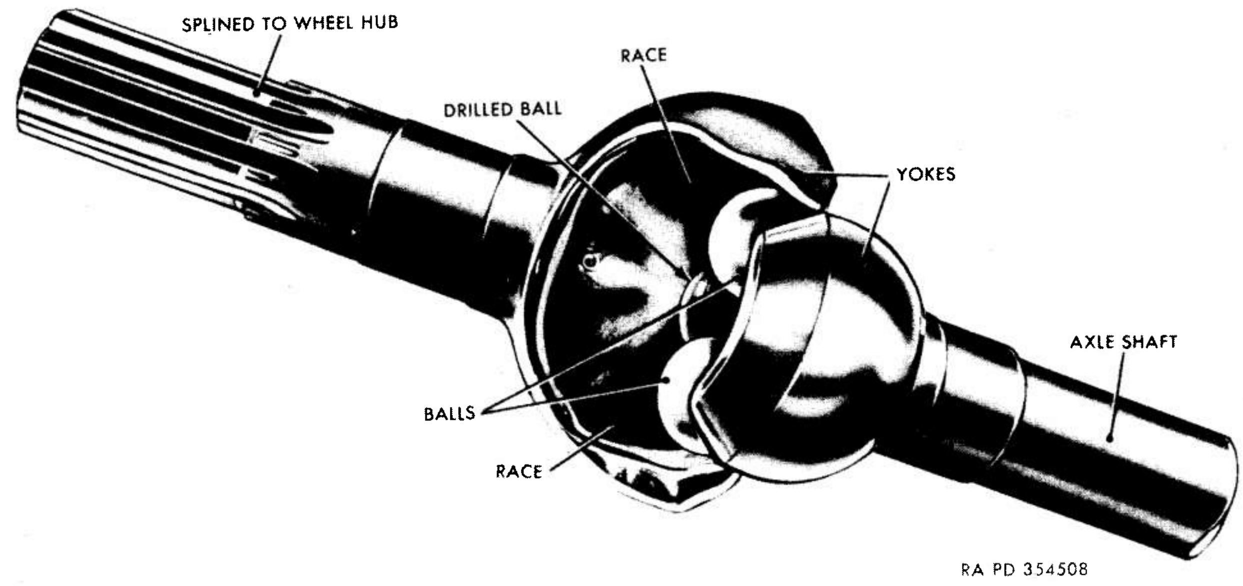


Figure 8-33. *Bendix-Weiss Constant Velocity Universal Joint—Assembled View*

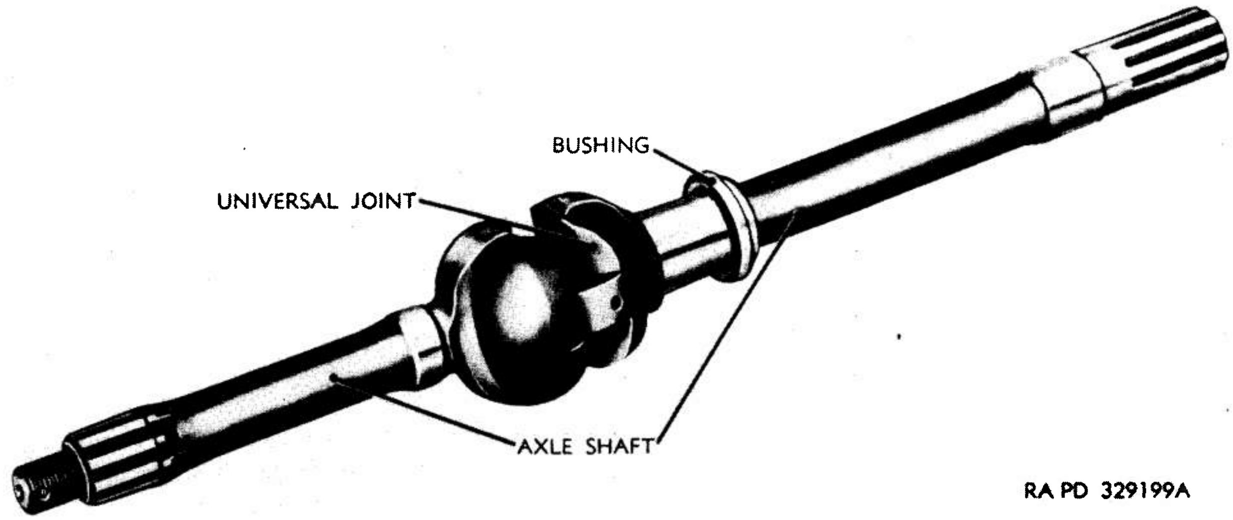


Figure 8-34. *Tracta Constant Velocity Universal Joint—Assembled View*

including a driven forked shaft, a driving forked shaft, a slotted connector, and a pronged connector.

The complete joint, including both connectors, floats between the two forks. Movement can occur between the individual connectors in a direction perpendicular to that in which motion is permitted by the forks. The points of driving contact translate as the entire joint rotates. Therefore they remain in a plane which bisects the angle between shafts, and constant velocity is achieved.

8-14 OTHER COUPLINGS

In addition to the true universal joints mentioned above, other flexible couplings are employed in automotive vehicles. The number and variety of these couplings preclude their being studied herein. They are discussed in the References.

Flexing of the springs causes the rear axle housing to move longitudinally with respect to the vehicle. Provision is made, therefore, to change the length of the propeller shaft by means of slip joints.

8-15 SPLINED SLIP JOINTS

A slip joint ordinarily consists of a male-female spline assembly, a grease seal, and a lubrication fitting. The male portion of the assembly is integral with the propeller shaft, while the female portion is fixed to the universal joint directly behind the transmission or transfer case. Design data for sliding splines (involute type) is given in the references.

TABLE 8-1 SHOCK AND FATIGUE FACTORS, ASME SHAFTING CODE

NATURE OF LOADING	VALUES FOR	
	K_m	K_t
Stationary shafts		
Gradually applied load	1.0	1.0
Suddenly applied load	1.5-2.0	1.5-2.0
Rotating shafts		
Gradually applied or steady load	1.5	1.0
Suddenly applied loads, minor shocks only	1.5-2.0	1.0-1.5
Suddenly applied loads, heavy shocks	2.0-3.0	1.5-3.0

SECTION VI DIFFERENTIALS

8-16 INTRODUCTION

The purpose of differential mechanisms in automotive power trains is to provide for the differences in the speed of rotation of a pair of wheels as a vehicle rounds a corner or travels over uneven ground. In the case of track-laying vehicles, the differential aids in turning the vehicle. Detail data regarding automotive differentials is presented in Refs. 53-56.

8-17 PRINCIPLES OF OPERATION

The action of a differential can be demonstrated by allowing a cylinder to roll between two parallel surfaces which move relative to one another. The displacement of the cylinder equals one-half the vector sum of the relative displacement of the two surfaces.

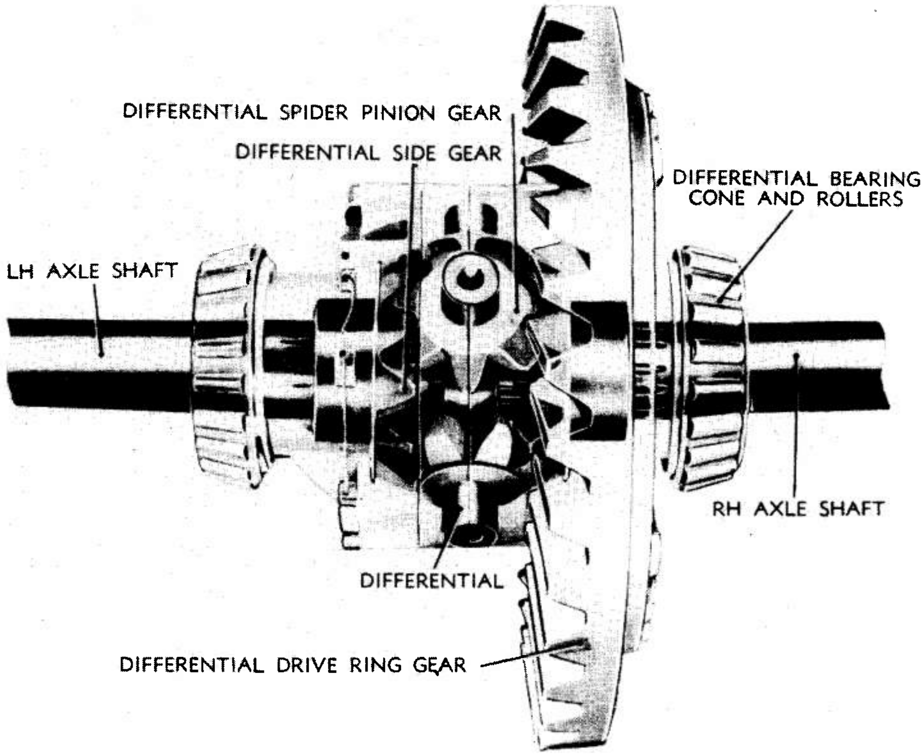
If gears are substituted for the cylinder and surfaces, the angular velocity, ω_C, of the center gear equals one-half the vector sum of the angular velocities ω_A, ω_B of the two outer gears. This relation can be expressed as

$$\omega_C = \frac{\omega_A \overrightarrow{+} \omega_B}{2} \qquad (8\text{-}11)$$

Differentials can be electrical, hydraulic, pneumatic, or magnetic in nature. Differentials commonly used in automotive power trains are mechanical gear differentials. Mechanical gear differentials are either of the bevel gear or the spur gear types. Differentials based on bevel gear types are generally used in automotive design.

8-17.1 BEVEL GEAR DIFFERENTIAL

Figure 8-35 shows a typical bevel gear differential used in wheeled automotive vehicles. Torque is transmitted through the differential drive pinion to the differential drive ring gear and to the differential case which is fastened to the ring gear. The differential spider, which is mounted within the differential case, rotates with the case. Mounted on the spider (free to turn) are four differential spider pinions (bevel gears) that mesh with the side gears which are, in turn, splined to the axles.



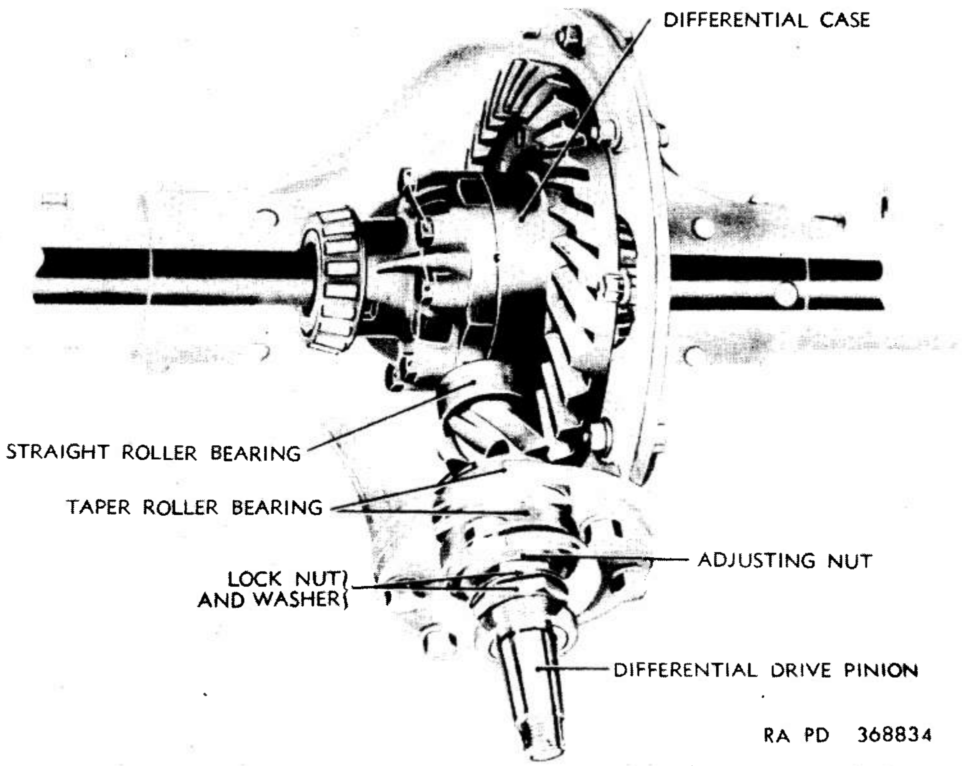


Figure 8-35. *Conventional Differential*

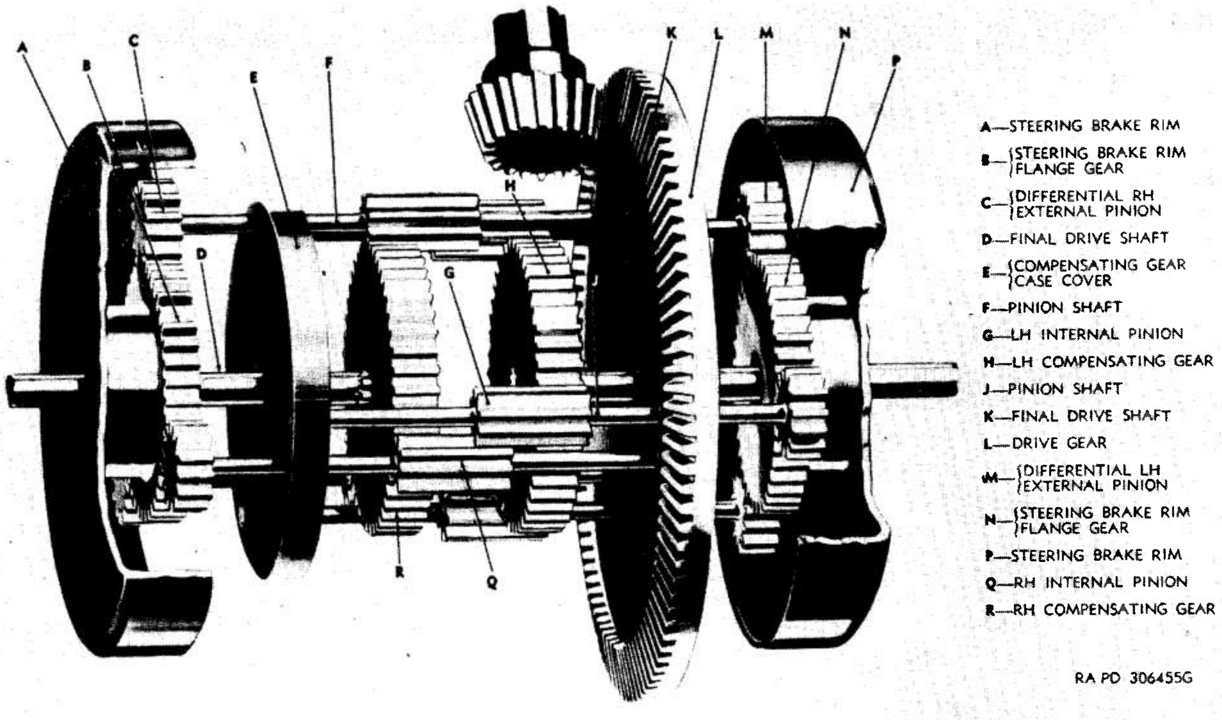


Figure 8-36. ***Controlled Differential—Schematic View***

The gears are spiral bevel gears which, because of their inherent advantages, have superseded straight bevel gears in the automatic differential application.

When both axles (wheels) rotate at the same speed in the same direction, there is no relative motion between the side gears and the spider pinion gears; both axle shafts rotate at the speed of the differential drive ring gear. When the wheels rotate at different speeds (during a turning maneuver or when operating on uneven terrain) the differential spider pinion gears rotate on their own axes while transmitting the system torque to the side gears and axles.

The functional relationship between input and output velocities can be expressed by rewriting Eq. 8-11 as

$$2\omega_C = \omega_A + \omega_B = C \tag{8-12}$$

where C is constant.

Assuming ω_C is the angular velocity of the differential spider and ω_A and ω_B are angular velocities of the side gears, Eq. 8-12 indicates an increase or decrease in angular velocity of one side gear will result in a decrease or increase in angular velocity of the other side gear. This compensating feature allows the wheels to turn at different speeds while transmitting equal torques.

8-17.2 SPUR GEAR DIFFERENTIAL

Figure 8-36 illustrates a typical spur gear differential which is an integral part of a controlled differential steering system. In the spur gear differential shown, the internal pinions, *G*, are functionally equivalent to the spider, and the compensating gears, *H*, are functionally equivalent to the side gears of the conventional unit.

The relationships expressed by Eq. 8-12 also apply to spur gear differentials.

The application of spur gear differentials to military vehicles is limited to steering transmissions for track-laying vehicles.

8-17.3 TORQUE TRANSMISSION

Under ideal conditions (friction absent), the torque transmitted to each of the side gears of a bevel gear differential is the same and the magnitude of this torque equals one-half the torque applied to the input spider gear. In the spur gear differential, the input torque also divides evenly, with equal torques going to each of the side gears.

Because friction is present, torques transmitted

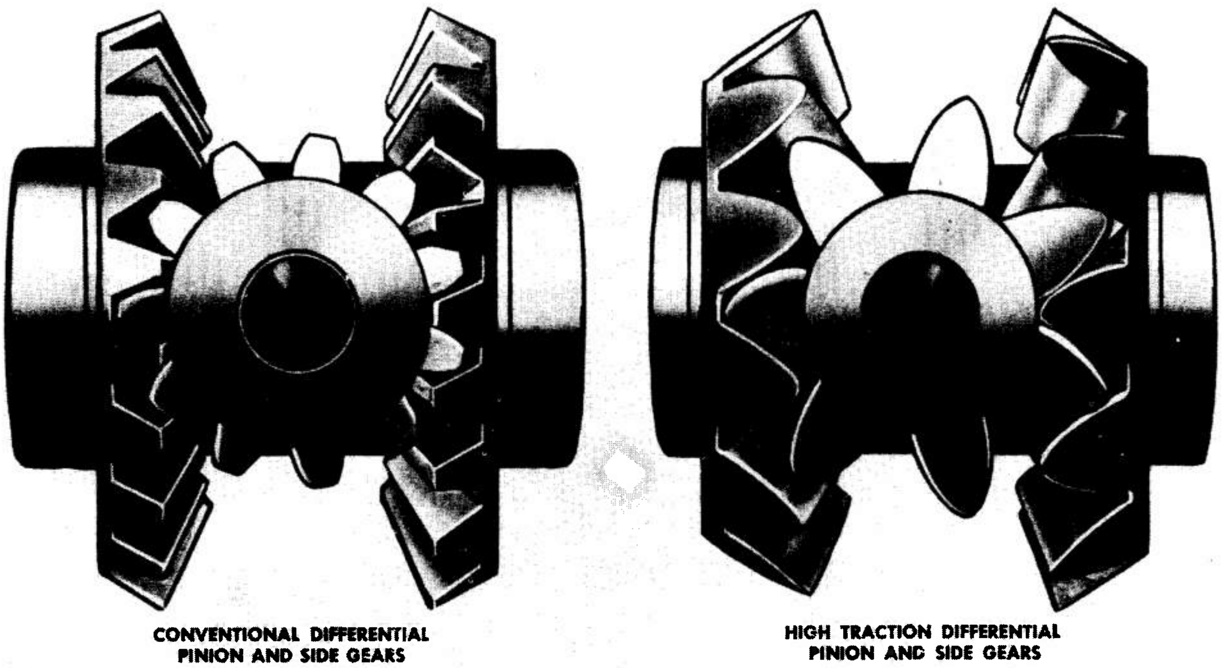


Figure 8-37. Comparison of High Traction Differential Gears with Conventional Differential Gears

to the drive shafts are not quite equal. This difference in torque is usually neglected and it is assumed torque division is equal.

The near-equal division of torque characteristic of conventional differentials is a disadvantage under certain circumstances. If one driving wheel loses traction, the torque developed at that wheel, drops to a low value. This causes the torque delivered to the opposite wheel to drop also. If a condition of 100% slip exists at one wheel, the opposite wheel (and the vehicle) will remain stationary, because the small amount of torque transmitted as a result of internal friction is not sufficient to cause motion.

In order to overcome this problem, some differentials are designed so as to develop a large amount of internal friction. A greater amount of torque can be transmitted to the nonspinning wheel with such a differential.

Another means of overcoming this difficulty consists of locking the differential out of the power train whenever desired. With the differential locked out, all torque is directed to the wheel that has traction. The typical differential lock acts by coupling one or both side gears to the differential case by means of dog clutches. In some cases, one or more pinions are constrained from rotating about their own axes.

8-17.4 HIGH TRACTION DIFFERENTIAL

The high traction differential resembles a conventional differential in outward appearance and operation but uses an unconventional form of gear tooth on the differential pinions and side gears (see Fig. 8-37). The successive teeth of these modified pinions contact the mating side gears at different radial distances from the pinion centers producing the effect of a variable length lever arm. Whenever rear wheel loading is such that there is relative motion between pinions and side gears, input torque is unevenly divided between the wheels. The differential pinion moves to a new position so that both wheels rotate at the same speed. Relative motion between pinion and side gears ceases, and torque to each wheel is the same.

The high traction differential is useful in starting the vehicle in cases where one wheel loses most of its traction while the other wheel has traction. It will not function where one wheel loses complete traction.

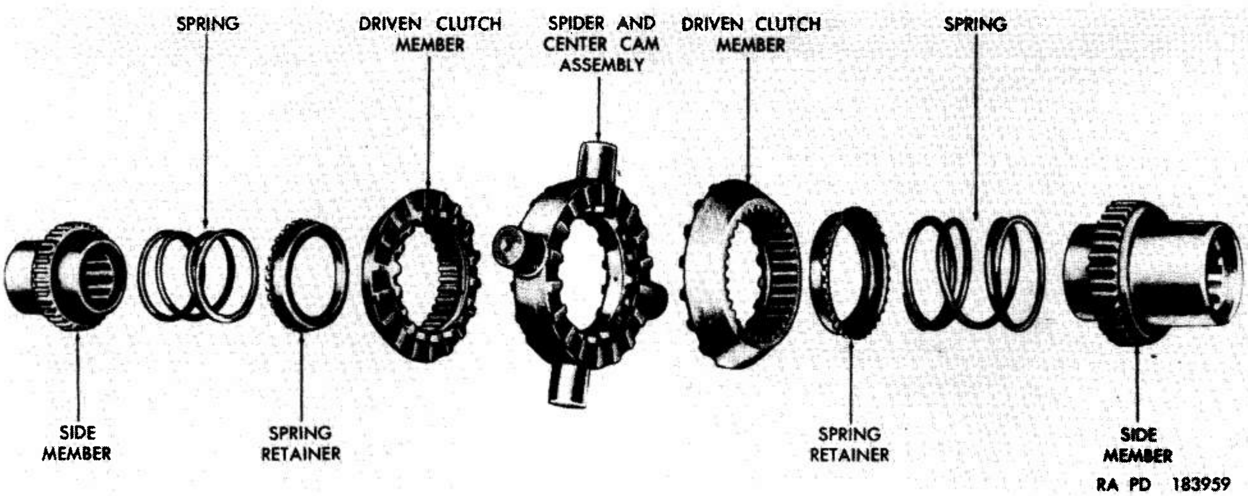


Figure 8-38. *No-Spin Differential—Disassembled View*

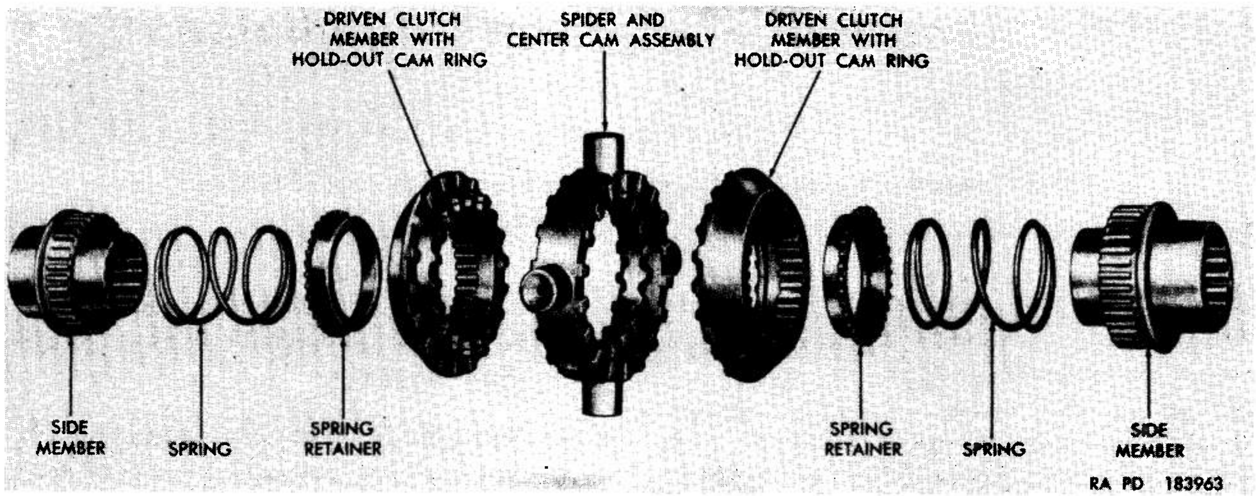


Figure 8-39. *Silent Type No-Spin Differential—Disassembled View*

8-17.5 NO-SPIN DIFFERENTIAL

A no-spin differential, Fig. 8-38, a combination of gear, clutch, and cam, delivers full torque to the tractive wheel by means of an automatic lockout. The conventional spider assembly is replaced by a spider and cam-ring assembly. Dog clutches, which replace the side gears, are coupled to the driving axles by splined side members. The dog clutches are held in engagement with the cam-ring assembly by spring pressure.

Under ordinary circumstances, drive is through the cam ring. Both wheels are powered and travel at the same angular velocity. When the vehicle negotiates a curve, the angular velocity of the outside wheel increases, causing the corresponding dog clutch to disengage from the serrated cam ring. Clutch action is similar to a slipping or overrunning ratchet and pawl. With the dog clutch disengaged, no torque is applied to the axle involved, and all torque is delivered to the wheel which has traction. When the slipping wheel slows, the dog clutch re-engages.

The silent no-spin differential, Fig. 8-39, is a modification of the no-spin differential. Each dog clutch plate is fitted with a holdout cam ring. When a difference in speed causes a dog clutch plate to be cammed out of engagement, the plate is restrained by the holdout cam. The ratchet and pawl action occurring in the no-spin differential is eliminated.

For vehicles with a tandem driving-axle unit, or multiple axles, a no-spin differential may be incorporated in the transfer case between output shafts to prevent loss of tractive effort from one

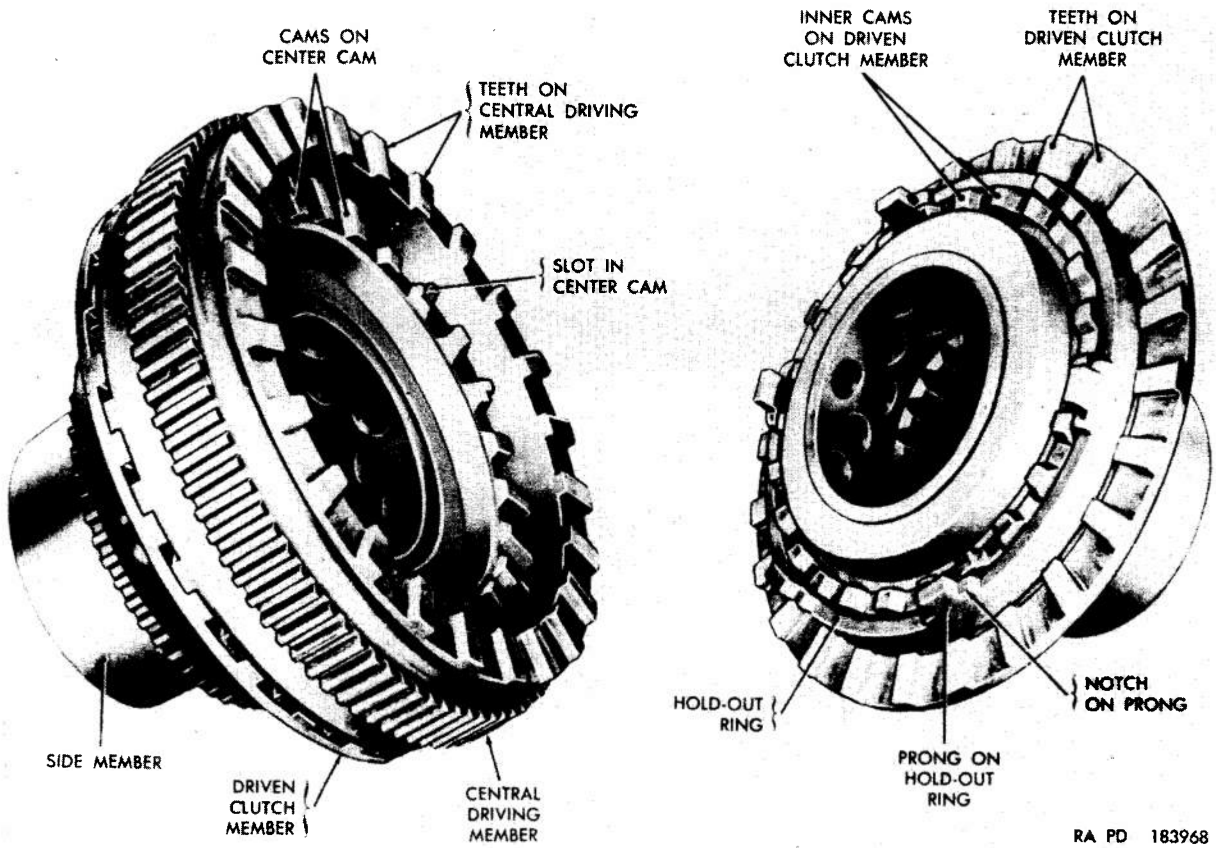


Figure 8-40. *No-Spin Overrunning Clutch—Partially Disassembled View*

set of wheels. Trapped inter-axle torque, which tends to reduce total tractive effort, is reduced.

8-17.6 CONTROLLED DIFFERENTIAL

A schematic view of a spur-gear controlled differential used on track-laying vehicles, to provide a means for steering, is shown in Fig. 8-36. The controlled differential consists of a pair of spur-gear-differential assemblies joined by a common differential carrier. The components on each side of the differential include a sun gear, compensating gears, and external and internal pinions.

Torque from the propeller shaft is delivered from the differential drive pinion and drive gear to the differential carrier. From the compensating gears it passes to the final drive shafts. The brake drums and the sun gears are connected to these final drive shafts.

If one brake is actuated, the attendant final drive shaft slows, slowing the attached sun gear and causing the internal pinions, attached to the differential carrier, to advance around the sun gear. This action results in increasing the speed, with respect to the differential carrier, of the opposing compensating gear. The tracks move at different speeds, causing the vehicle to turn about the slower moving track.

Because the controlled differential balances (equalizes) the torque developed at the output shafts (because the vector sum of the angular velocities of the output shafts is constant), braking one of the output shafts will result in a transfer of power from this shaft to the opposite output shaft for constant input power. Since the torque of both output shafts remains equal, this increase in power available will result in an increase in speed of the nonbraked shaft.

The controlled differential has replaced the older clutch brake steering system primarily because it affords reduced impact loading on final drive and tracks. In the modern track-laying vehicle, however, the controlled differential is being replaced by the cross-drive transmission. This latter steering transmission is discussed elsewhere in this chapter.

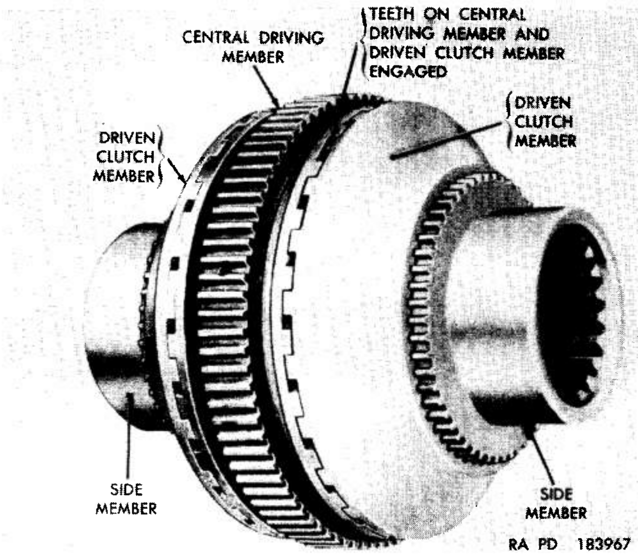


Figure 8-41. *No-Spin Overrunning Clutch*

8-17.7 NO-SPIN OVERRUNNING CLUTCH

The no-spin overrunning clutch, Figs. 8-40 and 8-41, although not a differential, approximates the action of a no-spin differential in an automotive power train. It is used in the transfer assembly of some multiwheeled drive vehicles to increase the traction potential of the vehicle.

The clutch consists of a central driving member, driven clutch plates, holdout cams, and side members. The central driving member receives power from the driving pinion and transmits it through driven clutch plates to side members which carry output shafts. One output shaft transmits power to the rear wheels while the other output shaft transmits power to the front wheels. The front wheels are not driven during normal running. If, however, the rear wheels begin to slip and the angular velocity of the front wheels becomes less than that of the rear wheels, power is supplied to the front wheels.

When the rear wheels slip, the angular velocity relative to the front clutch plate of the central driving member increases. Relative motion between the central driving member and the forward clutch plate causes them to engage and drive is supplied to the front wheels. When the rear wheels turn at the same speed as the front wheels, the latter disengage from the power source by means of the no-spin overrunning clutch.

SECTION VII AXLE ASSEMBLIES

8-18 INTRODUCTION

The rear axle assembly as defined in this chapter includes the final drive unit as well as the rear axle assembly. The functions of the rear axle assembly include:

1. Changing the direction of rotation from that of the propeller shaft to that of the driving axles.
2. Providing for a fixed reduction between propeller shaft and driving axles.
3. Acting as torque and thrust reaction systems for wheel loadings (conventional axle).
4. Providing for the driving of the rear wheels.

Detailed data regarding rear axle assembly design are presented in Refs. 57-60.

8-19 FINAL DRIVE

The final drive unit changes the direction of shaft rotation and provides a fixed speed reduction. All final drives, in general use, are gear

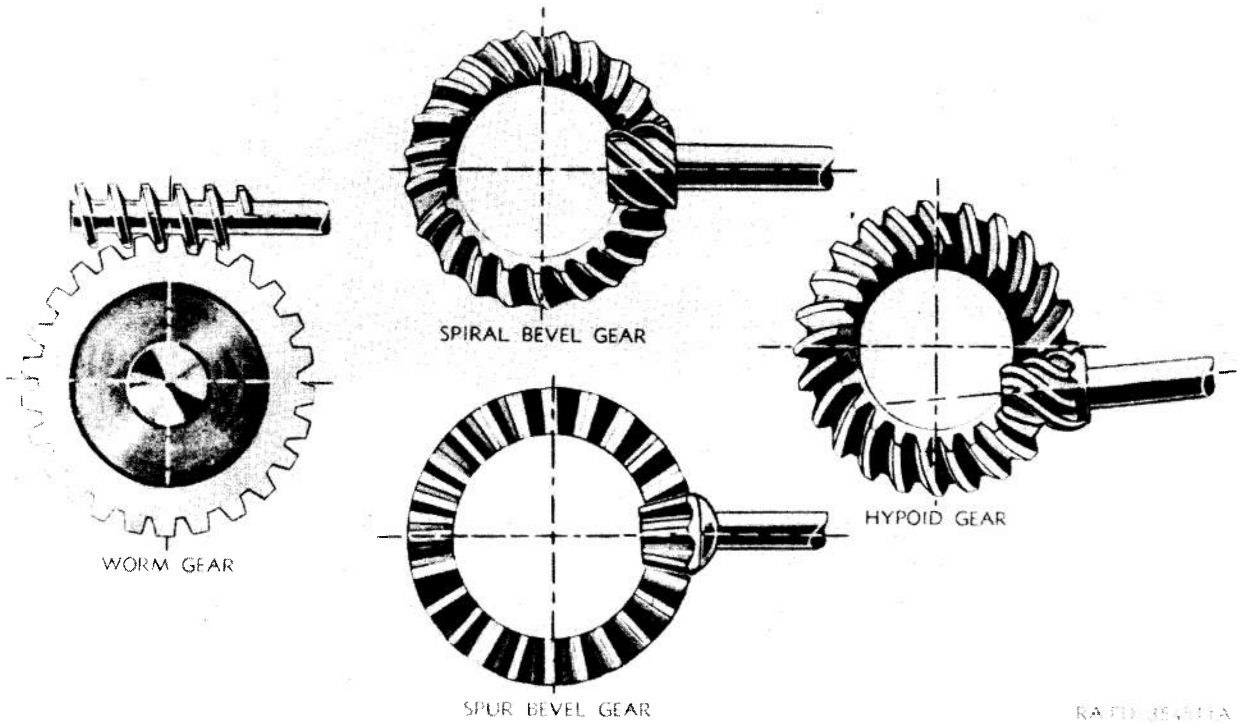


Figure 8-42. ***Final Drive Gears***

types. Spur gears are seldom used because of noise problems.

Most commonly, the final drive unit consists of a pair of spiral bevel gears, a pinion on the propeller shaft and a drive gear mounted in the differential case. These bevel gears may be straight or spiral bevels or hypoid. Spiral bevel gears are the most commonly used in general military automotive vehicles and hypoid gears in passenger cars and light trucks. The use of hypoid gears permits the drive pinion to be placed below the center of the drive gear. As a result, the propeller shaft may be lowered and greater body room obtained within the vehicle. Hypoid gears operate at a low noise level.

The simple, straight-bevel-gear final drive, Fig. 8-42, has several disadvantages. Mounting and location tolerances are critical. The pinion axis of rotation cannot be lower than the gear axis of rotation. Bearing problems may arise because of location limitations imposed by restrictions in gear placement. Speed and load-carrying capacities are somewhat less than other types of bevel gears.

Zerol bevel gears are distinguished by a curved rather than straight tooth face, in the case of ordinary bevel gears. Zerol gears are finished by grinding, and tooth accuracy is high. They have a slight overlapping action, and, consequently, run more quietly than straight bevel gears.

Spiral bevel gears are similar to zerol gears. They differ in the greater angle the face of the tooth makes with the axis of the gear. This larger angle results in greater overlap than occurs with the zerol gear. Consequently, the spiral bevel gear is quieter in action, and has higher load carrying capacity. A disadvantage of the spiral bevel gear is the much higher thrust load imposed on the supporting bearings, which tends to decrease bearing life.

Worm gears are used extensively in the final drive of medium and large trucks because they permit a large speed reduction or torque increase in a single reduction. The helical tooth is the most commonly used gear tooth form.

Worm and gear sets are mounted so that their shafts are nonintersecting, the angle between shafts being 90°. The worm bearings sustain a high thrust load, while worm gear bearings sustain a high radial load and a low thrust load.

All worm gear sets in common usage are either single enveloping or double enveloping. In a single enveloping gear set, the worm gear envelopes the worm, while in a double enveloping set both members of the gear set envelop each other, the

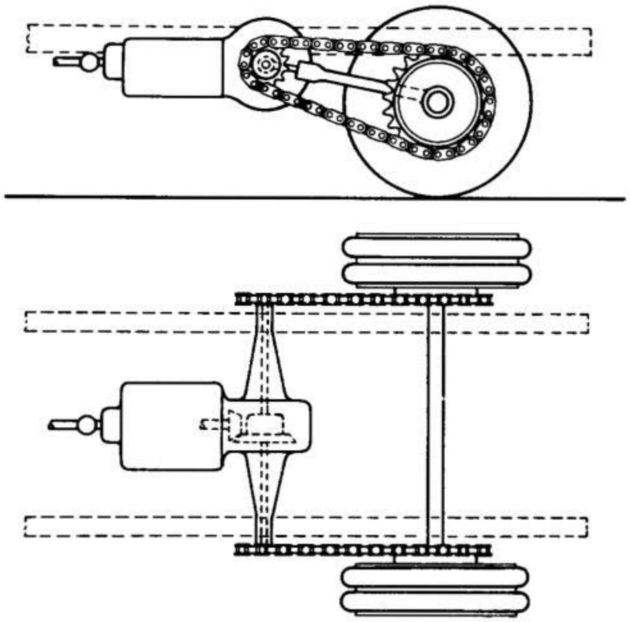


Figure 8-43. *Chain Drive (From The Motor Vehicle, by K. Newton and W. Steeds, Chilton Co., Philadelphia, Pa.)*

worm being hourglass in outline. This mutual envelopment results in a higher load-carrying capacity. The effective gear ratio can be increased by use of multiple pitch threads on the worm.

Internal gears have been used because they permit a large speed reduction in a small space. Drives, utilizing a solid, nondriving axle for wheel support, in which a jack shaft is employed to drive pinions within wheel-mounted internal gears, have been employed.

A final drive system, once used on almost all heavy vehicles, is the chain drive (double or single). It is rarely used for main propulsive power transmission at the present time, although some auxiliary drives or power takeoffs are chain driven. A typical double chain drive, illustrated in Fig. 8-43, employs a solid nondriving dead axle.

8-20 THE REAR AXLE

The rear axle, exclusive of the torque and thrust resisting systems, is a shaft of circular cross section. The function of the axle assembly element is to support a part of the vehicle weight and to either provide a bearing point for driving wheel rotation (dead axle) or to transmit power to the driving wheel in conjunction with a housing (live axle).

Four types of live axles are in common usage on military vehicles. They are distinguished by the means in which the axle shafts are connected and the loads they must sustain.

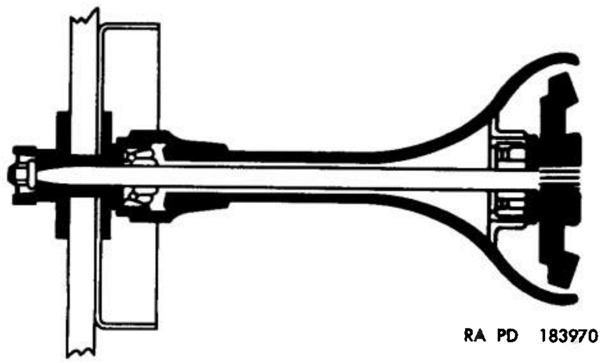


Figure 8-44. *Semifloating Rear Axle*

8-20.1 PLAIN REAR AXLE (NONFLOATING)

The plain or nonfloating rear axle is obsolete. In this system, the rear axle shaft is supported, within a housing, by bearings at either end. This shaft carries the full wheel load at one end and part of the differential case load at the other. All road-induced forces are taken by the axle shaft itself, as are the forces generated by the operation of the differential.

8-20.2 THE SEMIFLOATING REAR AXLE

This form of the live rear axle, illustrated in Fig. 8-44, is commonly used on passenger and light commercial vehicles. In the semifloating unit, the differential case is supported by bearings in the differential carrier. This relieves the inner ends of the axle shafts of the weight of the differential case. The outer end of the semifloating rear axle supports the vehicle and as such is subject to road-induced stresses caused by turning and skidding.

8-20.3 THE THREE-QUARTER FLOATING REAR AXLE

The three-quarter floating axle, Fig. 8-45, is a modified semifloating axle. The differential case is supported by the differential carrier; however, the shaft housing, rather than the axle shaft, carries the vertical wheel loads. Because the wheel is keyed to its outer end, the axle shaft must take lateral road loads. Some passenger vehicles use this construction.

8-20.4 THE FULL FLOATING REAR AXLE

The full floating axle, Fig. 8-46, is used on most heavy-duty trucks, on many wheeled off-the-

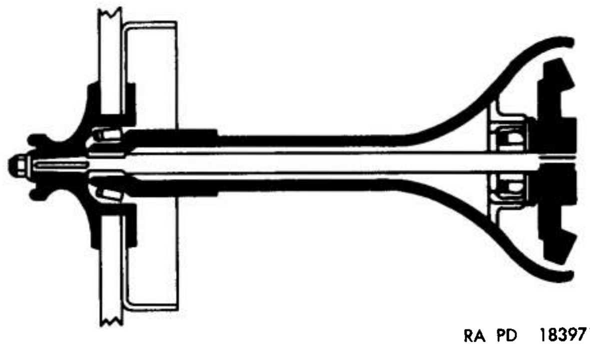


Figure 8-45. *Three-Quarter Floating Rear Axle*

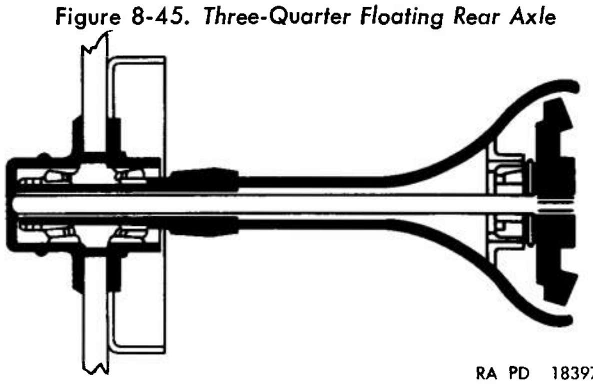


Figure 8-46. *Full Floating Rear Axle*

road vehicles, and on almost all military multi-wheel-drive trucks. This rear axle is similar to the three-quarter floating design; however, an additional outboard bearing supports the wheel end of the axle shaft. The axle tube or case carries the bearings, the axle shaft being nonrigidly connected to the wheel by means of a member such as a splined clutch. The axle shaft transmits only torque, all road-induced loads being sustained by the axle housing. With the full floating rear axle, axle shafts can be removed without removing the wheels.

8-20.4.1 The Double Reduction Rear Axle

On heavy-duty trucks and off-the-road vehicles, the full floating rear axle is frequently a double reduction type. An initial reduction is effected ahead of the differential by means of a spiral bevel gear set, and a second reduction is effected at the differential by means of a spur gear set. Output from the second reduction goes to the differential carrier, then to the axle shaft. The double reduction axle results in increased torque at the rear axle.

8-20.4.2 The Dual-Ratio Rear Axle

In many truck applications and in some passenger vehicles, a dual-ratio rear axle assembly is used, Fig. 8-47. A manual control is customarily provided for selecting either of two rear axle ratios.

The dual ratio system serves as an auxiliary transmission providing the driver with greater flexibility in dealing with varying road conditions. The dual ratio element is a planetary gear system, located between the differential ring gear and the differential case. From the propeller shaft, the drive is transmitted through the differential drive pinion and ring gear, the planetary gear train, the differential and to the driving wheels.

In heavy-duty trucks and special wheeled off-the-road vehicles, where maximum flexibility and high torque are required, double reduction and dual-ratio rear axle assemblies are combined in a single unit.

8-21 MULTIWHEELED DRIVES

Tactical wheeled vehicles normally have multiwheel drives. Many of the heavier vehicles of this category are equipped with four rear drive wheels in order to increase traction and to increase the load rating of the vehicle. Dual wheels are normally used with this arrangement.

Several rear wheel suspensions are used with four rear wheel drive systems. Of these, the bogie suspension described in Chapter 11 is most commonly used. Wheeled vehicles having bogie suspension normally use one of two different systems to distribute power to the individual axles. The dual rear axles may be driven by independent propeller shafts (Fig. 8-48) from a transfer assembly. With this system, the propeller shaft is divided into three parts, a short center section passing through bearings mounted on the forward rear axle. The transfer assembly may contain an interaxle differential.

An alternate arrangement is the tandem drive (Fig. 8-49). The tandem drive employs a double reduction axle and a single propeller shaft from the transmission transfer assembly to the forward rear-axle drive pinion, and another short interaxle propeller shaft between the two axle assemblies. Ordinarily, no interaxle differential is used, but a power divider is incorporated into the system. The power divider supplies a differential action when required.

8-21.1 FRONT WHEEL DRIVES

In multidrive wheeled military vehicles, the front wheels may be driven through a driving axle

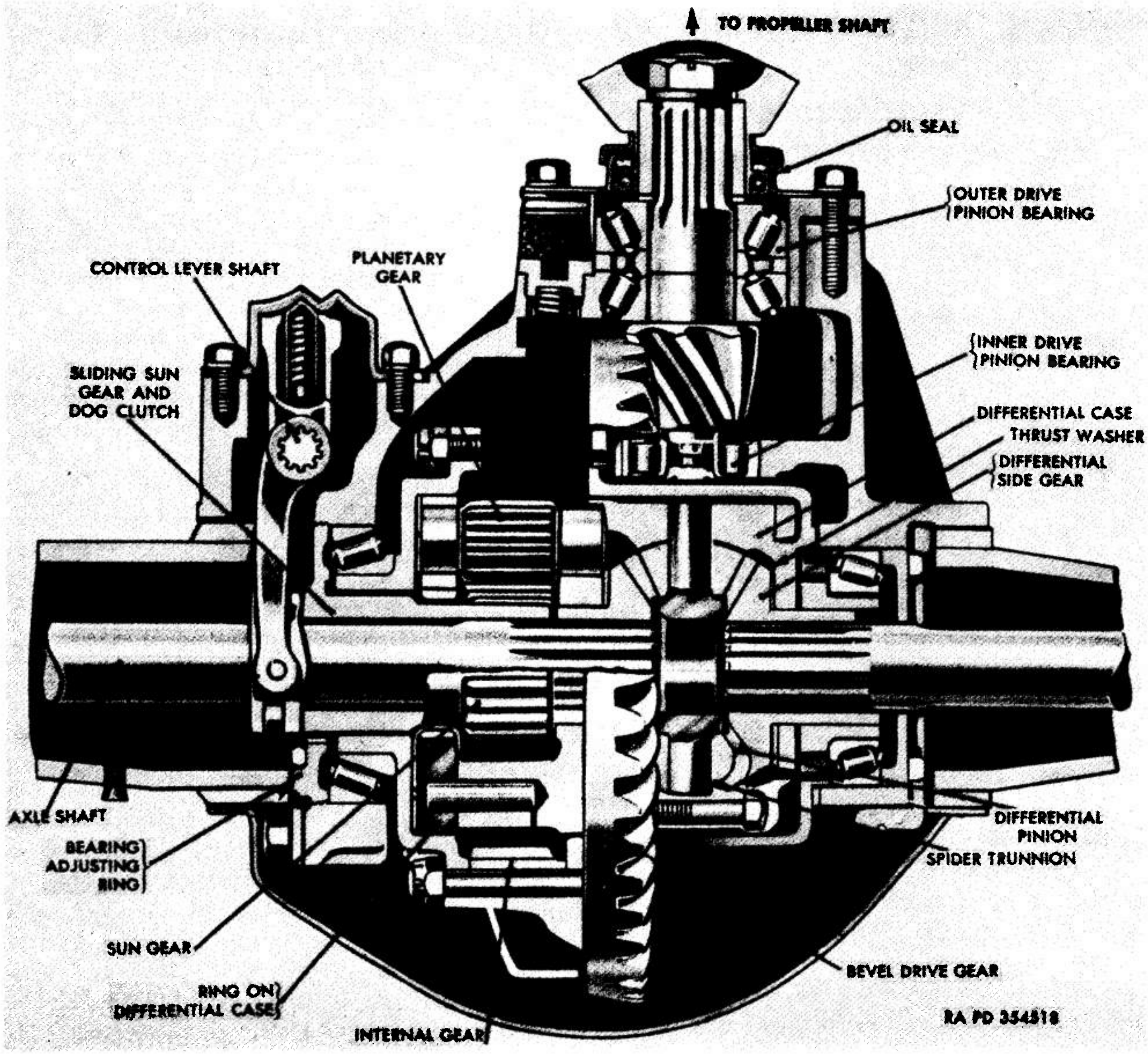


Figure 8-47. *Dual-Ratio Rear Axle*

assembly whose configuration is similar to the driving rear axle. The axle assembly, usually of the full floating type, can be either single or double reduction. The system is generally of the simple Hotchkiss type.

Because the front axle must include provision for steering, and the front wheels must turn on pivots or steering knuckles, the wheels are usually driven by axle shafts with universal joints that are concentric with the steering knuckle pivot centerline. Figure 8-50 shows the arrangement of a typical live front axle assembly.

An alternate front wheel drive of limited usage utilizes a spiral bevel pinion affixed to the end of the axle drive shaft. The pinion drives the lower member of a double bevel gear set fastened to the lower end of the steering knuckle. The upper member of the double bevel meshes with a bevel integral with the wheel hub. When the wheels are turned in steering, the bevel on the wheel hub precesses about the bevel on the steering knuckle.

8-21.2 INTERWHEEL DIFFERENTIAL

A recent development in front-wheel drives is the utilization of dual front wheels having an interwheel spur-gear differential between each pair

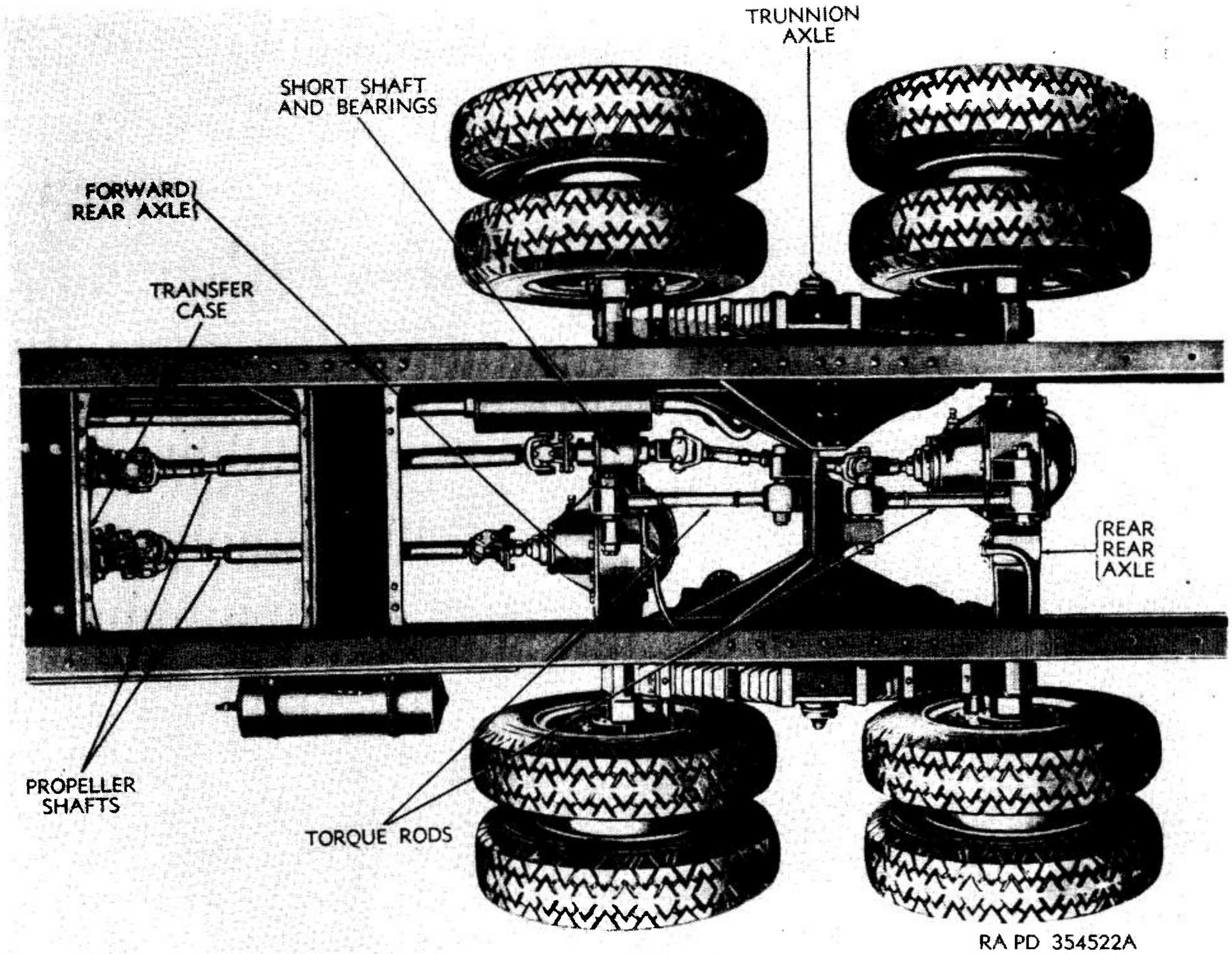


Figure 8-48. *Dual Rear Axle Drive with Independent Propeller Shafts*

of dual wheels to make the wheels easily steerable. Each wheel can be individually braked.

Interwheel differentials are used for front nondriving (dead) axles and driving (live) rear axles to provide differential action in dual wheel applications.

8-22 TORQUE AND THRUST REACTION SYSTEMS

A powered axle assembly and associated suspension system must incorporate provision to withstand torque reaction and propulsive thrust of the vehicle. In addition to these forces, lateral forces and other externally induced forces on the vehicle are transmitted to the suspension-axle assembly. Finally, the axle assembly is subject to the weight of the vehicle and any load carried by the vehicle.

Torque reactions act in two planes, which are: (1) perpendicular to the input shaft of the final drive, and (2) perpendicular to the output shafts (e.g., axles).

In general, the resistive elements are rigid or semirigid members located between the axle housing and the drive frame. They are arranged to withstand the torques and forces imposed and yet allow the desired wheel movements.

A number of mechanical configurations are in use to provide the necessary resistance to forces imposed on the axle system. Leaf springs alone or with torque arms are frequently used for this purpose on military vehicles. One of the most commonly used drive systems is the Hotchkiss drive, Fig. 8-51.

In the Hotchkiss type rear axle assembly, the rear springs of the vehicle act as torque and thrust members in addition to carrying the vehicle weight. The springs are semielliptical, rigidly attached to the axle at their center and connected to the ve-

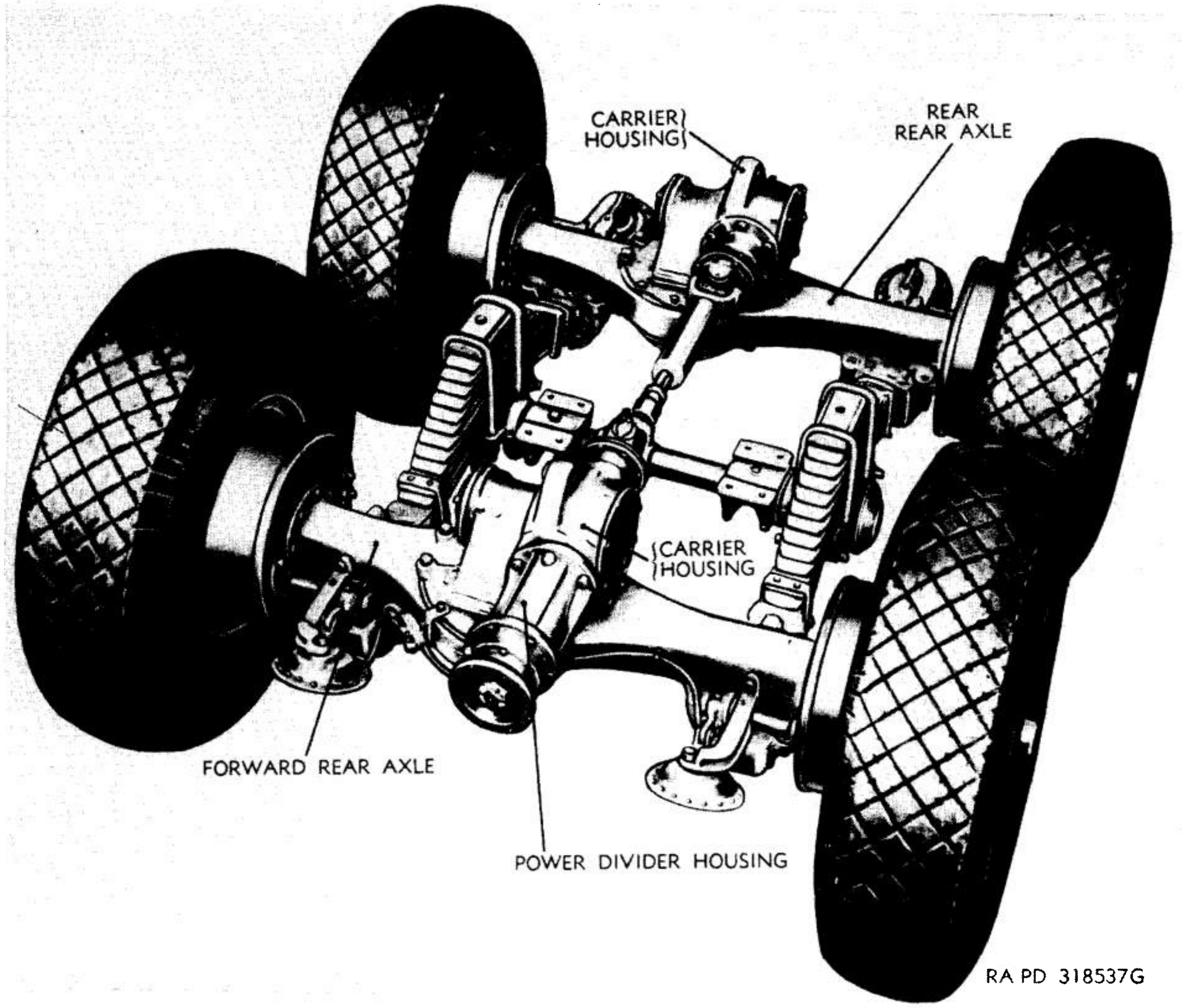


Figure 8-49. *Tandem Dual-Rear Axle*

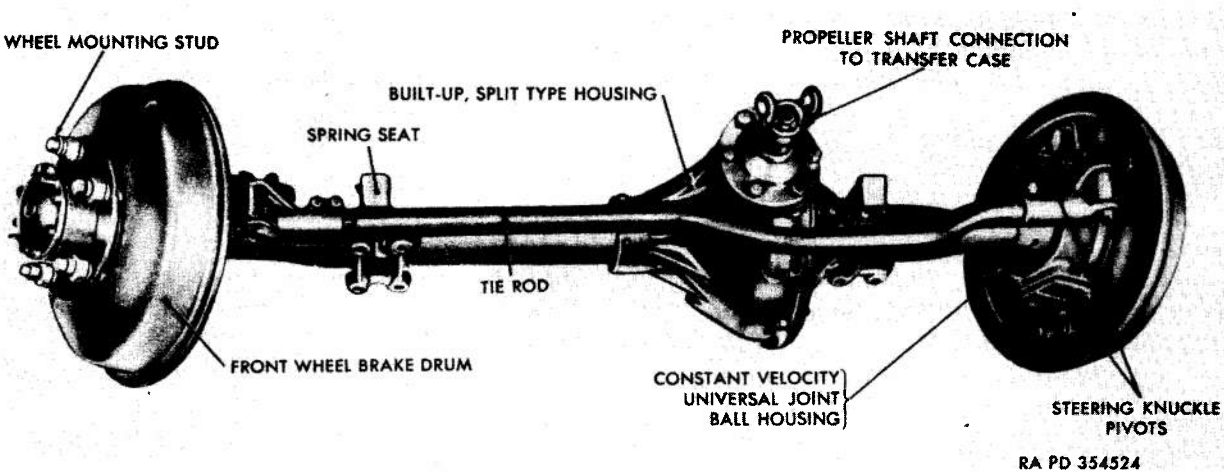


Figure 8-50. *Front Driving Axle Assembly*

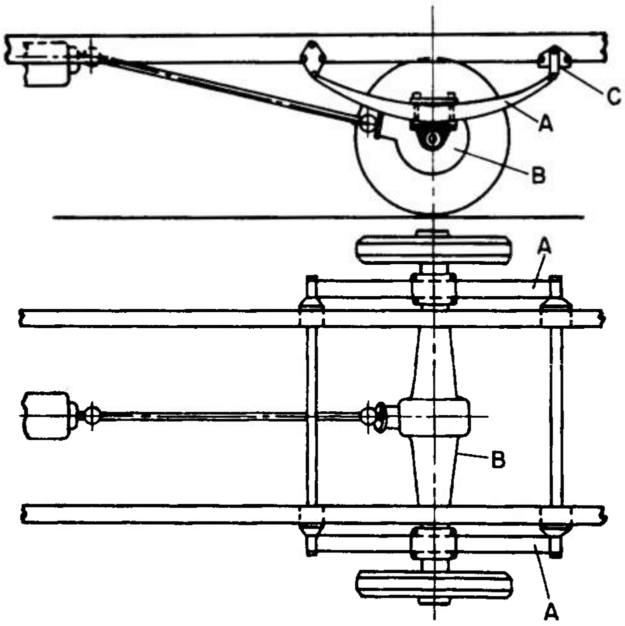


Figure 8-51. *Hotchkiss Drive (From* The Motor Vehicle, *by K. Newton and W. Steeds, Chilton Co., Philadelphia, Pa.)*

hicle at either end. Their forward ends are attached to the frame by fixed pivots, their rear ends by shackles. Driving thrust is transmitted to the frame via the fixed pivot; torque reactions are resisted by the springs. Flexing of the springs requires a double universal joint in the propeller shaft. Road-induced lateral reactions are transmitted to the frame through both the front and rear attachments of the spring members. Connections between the spring and frame may be incorporated to reduce twisting of the frame.

If the springs are stiffened to resist torque reactions without undue twisting, they become too stiff for optimum suspension springing. To overcome this difficulty, a modified Hotchkiss drive is often used in conjunction with torque resisting members, i.e., linkages that permit vertical movement yet absorb the torque reactions.

A further modification of the Hotchkiss drive is made, in which a separate member is incorporated to resist both torque reaction and driving thrust. Under this system, the springs are required only to resist nonpropulsive road-induced forces and support the weight of the vehicle.

Other systems use pivoted rods or, in some cases, a central tube surrounding the propeller shaft to resist the torque reactions. These systems are described in Chapter 11.

8-23 FINAL DRIVE FOR TRACK-LAYING VEHICLES

The final drive for track-laying vehicles is similar in physical configuration to final drives in wheeled vehicles. At their inner ends, driving axles are splined to the compensating gears of the differential, or are connected to them by the propeller shaft. At their opposite ends, the driving axles are coupled, by means of gears, to the driving sprockets which drive the tracks.

SECTION VIII BRAKES

8-24 INTRODUCTION

Brakes are members of the power train which provide a means for reducing vehicle speed, for stopping, or in some instances, turning a moving vehicle. Because the brakes must bring the vehicle to a controlled stop in the shortest possible distance in an emergency, they must be capable of applying large retarding forces. On long downgrades, the brakes must dissipate large quantities of heat which may be generated during prolonged braking, without experiencing an excessive temperature rise.

A driving wheel may be braked directly by means of a drum on the wheel or indirectly by means of a drum attached to the drive shaft, the main output shaft of the transmission, or the input shaft of the final drive unit. Because of the gear reduction in the final drive unit, brakes acting at a point in the power train ahead of the final drive unit are more effective than similar brakes acting at the wheels themselves. Detailed data regarding brake design are presented in Refs. 61-66.

8-25 GENERAL THEORY OF RETARDATION

The major factors entering into the dynamics of braking are:

1. The *primary motion-resisting force* originating in the frictional engagement between brake elements.

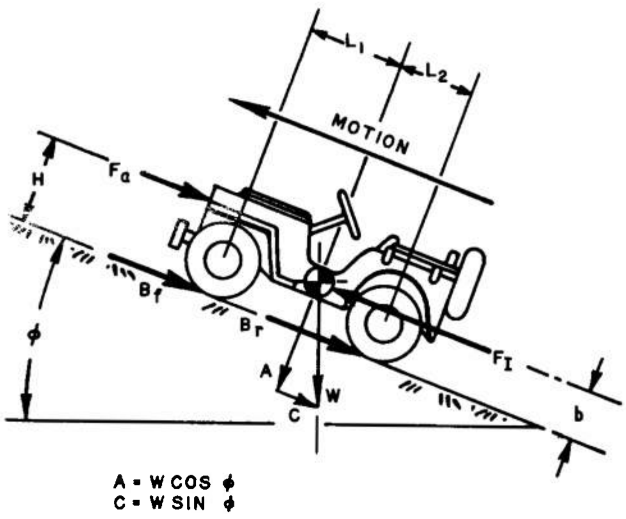


Figure 8-52. *Grade Effect on a Vehicle*

2. The *dynamic braking effect* resulting from the dynamic weight transfer on the axles. The distribution of the vehicle weight between axles influences the limits of braking performance.
3. The *grade effect,* as shown in Fig. 8-52, influences braking. Effective vehicle weight and the grade resistance are functions of the slope on which a vehicle is operating.
4. The *rolling resistance,* a motion-opposing force that aids the deceleration of a vehicle.
5. The *air resistance,* also a motion-opposing force and as such, aids braking.
6. The *inertial forces* due to linear deceleration of total vehicle mass and angular deceleration of rotating parts.
7. The *power train resistance,* resulting from the inherent resistance torque of the system.
8. The *engine braking effect,* resulting from the inherent resistance torque of the engine.

The general equation of decelerated motion may be written as

$$B = b\,(m\gamma_b) - \Sigma F \qquad (8\text{-}13)$$

where

B is the total braking force required, lb
b is the deceleration, ft per sec^2
m is the mass of vehicle, lb per ft per sec^2
γ_b is the inertia mass factor introduced by rotating parts
ΣF is the algebraic summation of all resistive forces other than the braking force, lb

The stopping distance, S, neglecting air resistance, can be expressed as

$$S = \frac{\gamma_b m}{B + \Sigma F}\left[\frac{V_i - V_f}{2}\right] \text{ ft} \qquad (8\text{-}14)$$

where

V_i and V_f are initial and final vehicle velocities, fps.

8-26 HEAT DISSIPATION

During braking, kinetic energy of the vehicle is converted to thermal energy within the brakes. The brake mechanism must dissipate this heat to the surrounding atmosphere to avoid excessive brake temperatures.

The rate at which brake temperature rises is dependent upon the rate of energy absorption by the brake, heat transfer coefficients of the brake materials, and mass of the heated parts. In automotive service, temperatures of 400° to 500°F are encountered.

The energy absorbed per unit time, E $\frac{\text{in-lb}}{\text{sec}}$ is

$$E = \mu p_{n_{(ave)}} A_c V_T \qquad (8\text{-}15)$$

where

μ = the coefficient of friction between the moving and stationary elements of the brakes
$p_{n_{(ave)}}$ = the average normal pressure on the friction surfaces, psi
A_c = the area of contact, in^2
V_T = the sliding velocity, in/sec

This energy input must not exceed the brake energy dissipating capacity.

Table 8-2 (see page 8-55) lists values for μ and p_n. Values of μ represent average conditions and do not consider changes which occur with changing slip velocity.

8-27 BRAKE MECHANISM

Friction brakes compose the majority of the automotive brakes used in the United States. They may be classified as either shoe-and-drum types or disk types.

Shoe-and-drum brakes can be external-contracting or internal-expanding. The latter is more common. Either may be applied to the wheels or to a point in the power train. The former is more representative of automotive practice.

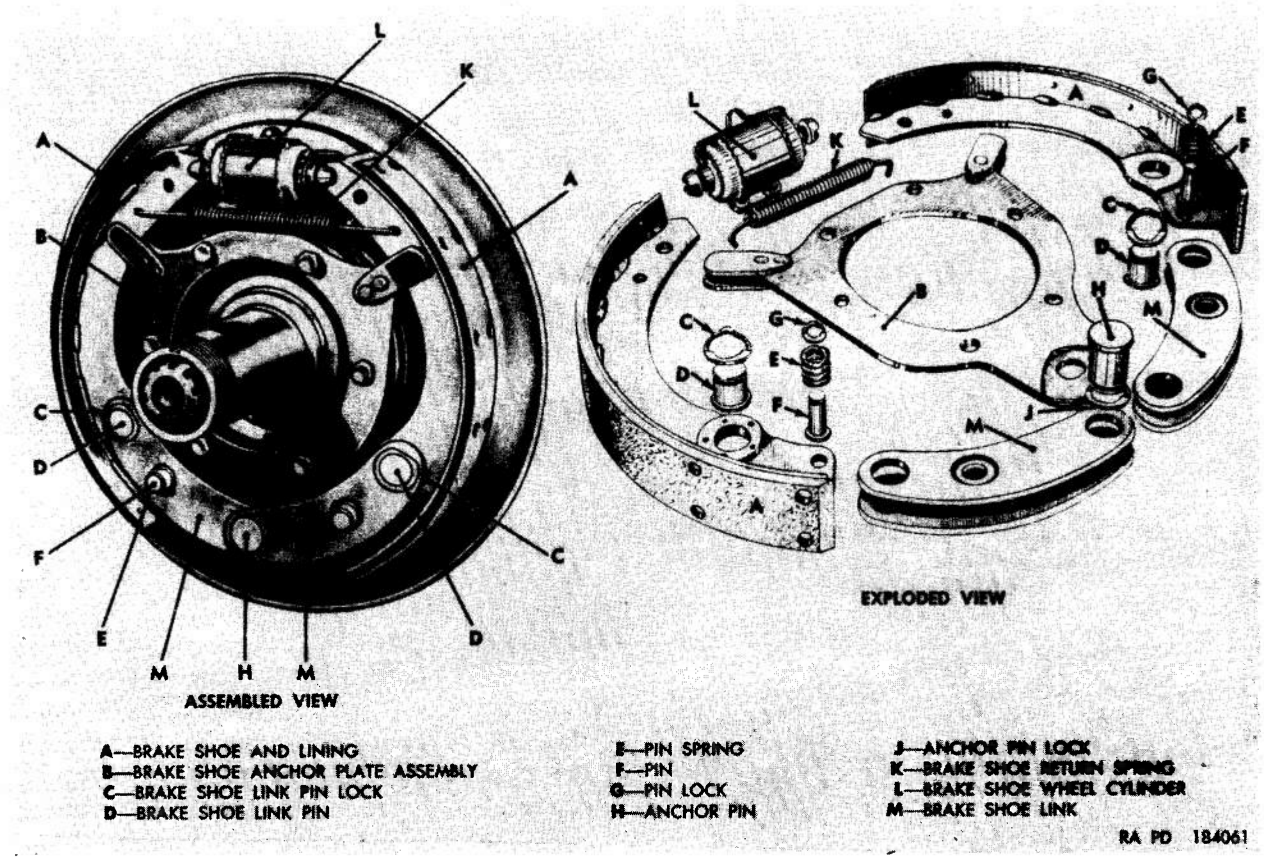


Figure 8-53. *Internal-Expanding Automotive Brake*

8-27.1 EXTERNAL-CONTRACTING BRAKES

In external-contracting brakes, the brake shoes are applied against the outside of the drum.

A cover arrangement must be provided to avoid exposure of the contacting elements to foreign matter whose entrance would cause rapid brake wear, especially in wheel brakes. External braking of this type offers no advantages over internal braking, and has several serious drawbacks. It is, therefore, seldom used in the automotive applications.

A common application for the external-contracting brake is the propeller shaft brake. The physical configuration of the power train at this point lends itself to a band-type version of this brake.

A common application of the external-contracting brake is the hand or parking brake. Disk type brakes have also been used in this application.

8-27.2 INTERNAL-EXPANDING BRAKES

The nonrotating member may be placed inside of the rotating drum to take advantage of the protection afforded by the brake drum. This brake, Fig. 8-53, is known as an internal-expanding brake. The shoe is forced outward to engage the drum. The internal brake is used on modern automotive vehicles because of its compact, economical, and trouble-free construction. Brake shoes and brake operating mechanism may be mounted on a backing plate or brake shield, which fits against the open end of the brake drum and protects the braking surfaces from foreign matter.

8-27 DISK BRAKES

In the disk brake, disks or pressure plates are forced against the faces of a rotating plate to provide braking action. The primary disk arrangements have been known and used for at least 40 years but have only recently been rediscovered by the automotive industry. Modern disk brakes are self-energizing or nonself-energizing. In the self-energizing type, frictional forces assist in the application of the brakes.

8-27.3.1 Nonself-Energizing Disk Brakes

Figures 8-54 and 8-55 illustrate the principle of the nonself-energizing disk brake. A pair of

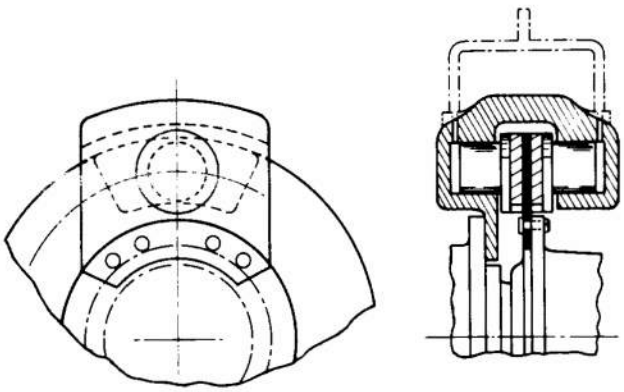


Figure 8-54. *Nonself-Energizing Disk Brake (From The Motor Vehicle, by K. Newton and W. Steeds, Chilton Co., Philadelphia, Pa.)*

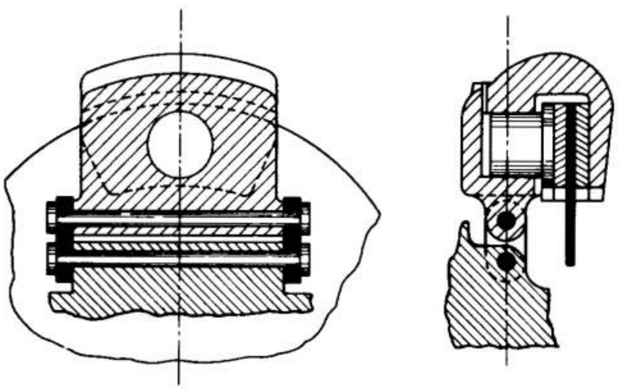


Figure 8-55. *Nonself-Energizing Disk Brake (From The Motor Vehicle, by K. Newton and W. Steeds, Chilton Co., Philadelphia, Pa.)*

friction elements, arranged in a C-shaped manner, straddle a single circular friction plate. This plate is attached to the wheel assembly and rotates with it. The pressure pads may be actuated by hydraulic pressure, or by electromagnetic means. Pads pressing against the disk create a friction torque which opposes motion of the disk. Friction torque, T_f (in-lb), is

$$T_f = 2\mu p A R_o \tag{8-16}$$

where

μ = coefficient of friction between the pads and the disk

p = fluid pressure within the cylinder, psi

A = area of piston and pad, in^2

R_o = distance from the wheel axis to the effective center of the friction pads, in

The friction coefficient, μ, is a function of sliding velocity. It may be expressed as

$$\mu = \mu_o - mV_T \tag{8-17}$$

where

μ_o is the static coefficient of friction

m is the slope of the friction vs velocity curve

$$m = -\frac{\Delta\mu}{\Delta V_T}$$

V_T is the sliding velocity, in per sec

The area of contact, A, is relatively small, thus the pressure can be assumed constant across it. The friction torque may then be expressed as

$$T_f = (2\mu_o - mV_T)\ pAR_o \tag{8-18}$$

Because of the C-clamp configuration of the pressure pads, the applied force, pA, is balanced in the direction perpendicular to the clutch face. The radial force, $[\mu_o - mV_T]\ pA$, is unbalanced, and must be resisted by the wheel bearings. To eliminate this problem, the radial forces are made to balance by locating a second pair of pressure pads diametrically opposite the first pair. This configuration requires more space than the internal-expanding shoe and drum type. Configurations which attempt to reduce the space requirement are discussed in the references.

8-27.3.2 Self-Energizing Disk Brakes

A typical self-energizing disk brake is shown in Fig. 8-56. Two flat plates, D and E, with segmented friction lining, C, attached to their faces, form the pressure elements. One plate, D, keyed by means of G, to the axle, F, can move axially. The other, E, is free to rotate. The plates, separated by a number of balls, H, which ride in matching recesses, are forced apart by a pair of hydraulic or mechanical actuators during braking.

Upon actuation during braking, plate E, makes contact with brake drum, A, which rotates with the wheel. Rotation of plate, E, causes balls, H, to move along ramps which form the sides of their recesses. The plates move farther apart and the barking force increases. The slope of the ramp surface governs the amount of self-energizing produced by the brake.

Figure 8-57 indicates another form of self-energizing disk brake. Both pressure plates, keyed to the axle, are free to move axially and angularly. Upon actuation by means of a hydraulic piston and cylinder, the two pressure plates, Fig. 8-57, rotate

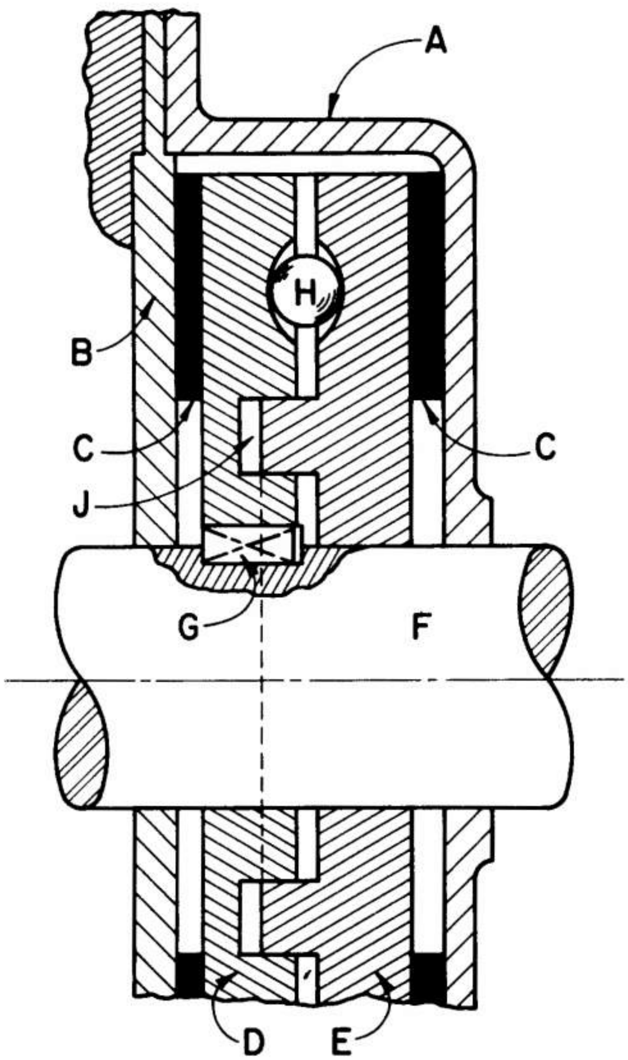


Figure 8-56. *Self-Energizing Disk Brake (From* The Motor Vehicle, *by K. Newton and W. Steeds, Chilton Co., Philadelphia, Pa.)*

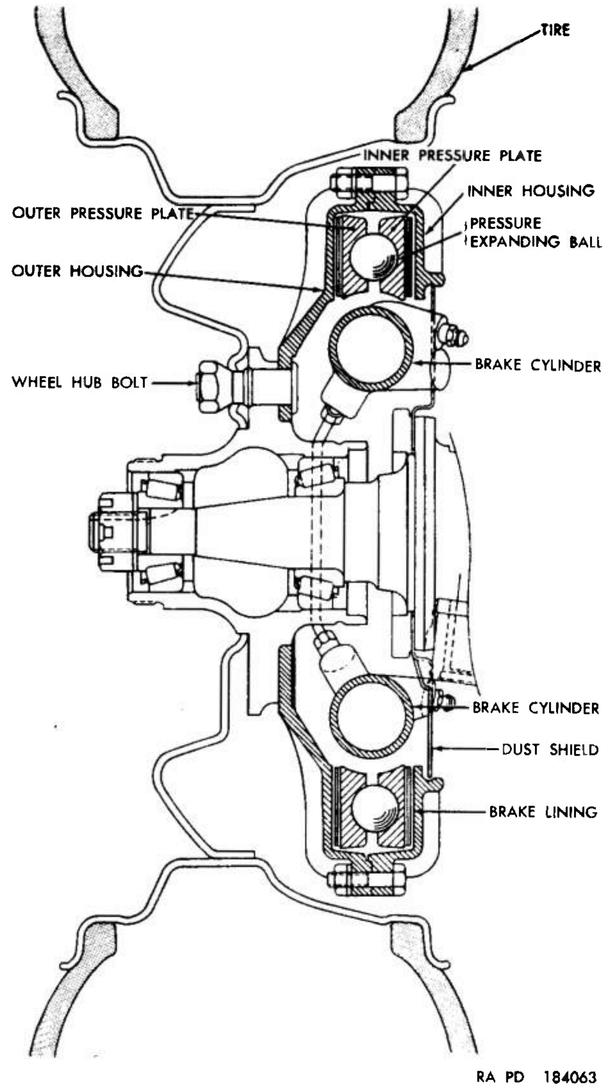


Figure 8-57. *Disk Brake—Sectional View*

slightly in opposite directions. The balls move along ramps producing axial displacement of the plates, causing engagement with the moving drum. After contact, self-energizing provides additional force.

The self-energizing disk brake requires less actuating pressure to produce given braking than does the nonself-energizing type. The second of the two types of self-energizing brakes discussed requires lower actuating pressures, because wedging action of the balls results in increased actuation pressure. The slope of the ramps incorporated influences the degree of self-energizing. There is a limiting value of this angle above which the brake will lock when the pressure plates make contact with the mating surface.

8-28 THE PARKING BRAKE

The function of a parking brake is to retain a vehicle on grades equal to the maximum grades that the vehicle can traverse. Parking brakes for military vehicles are normally mechanically actuated, hence, they are independent of the main hydraulic or air system.

Contracting band brakes and disk transmission brakes are in common usage on military vehicles as parking brakes. These brakes are usually located at the rear of the transmission or transfer case and act on a propeller shaft.

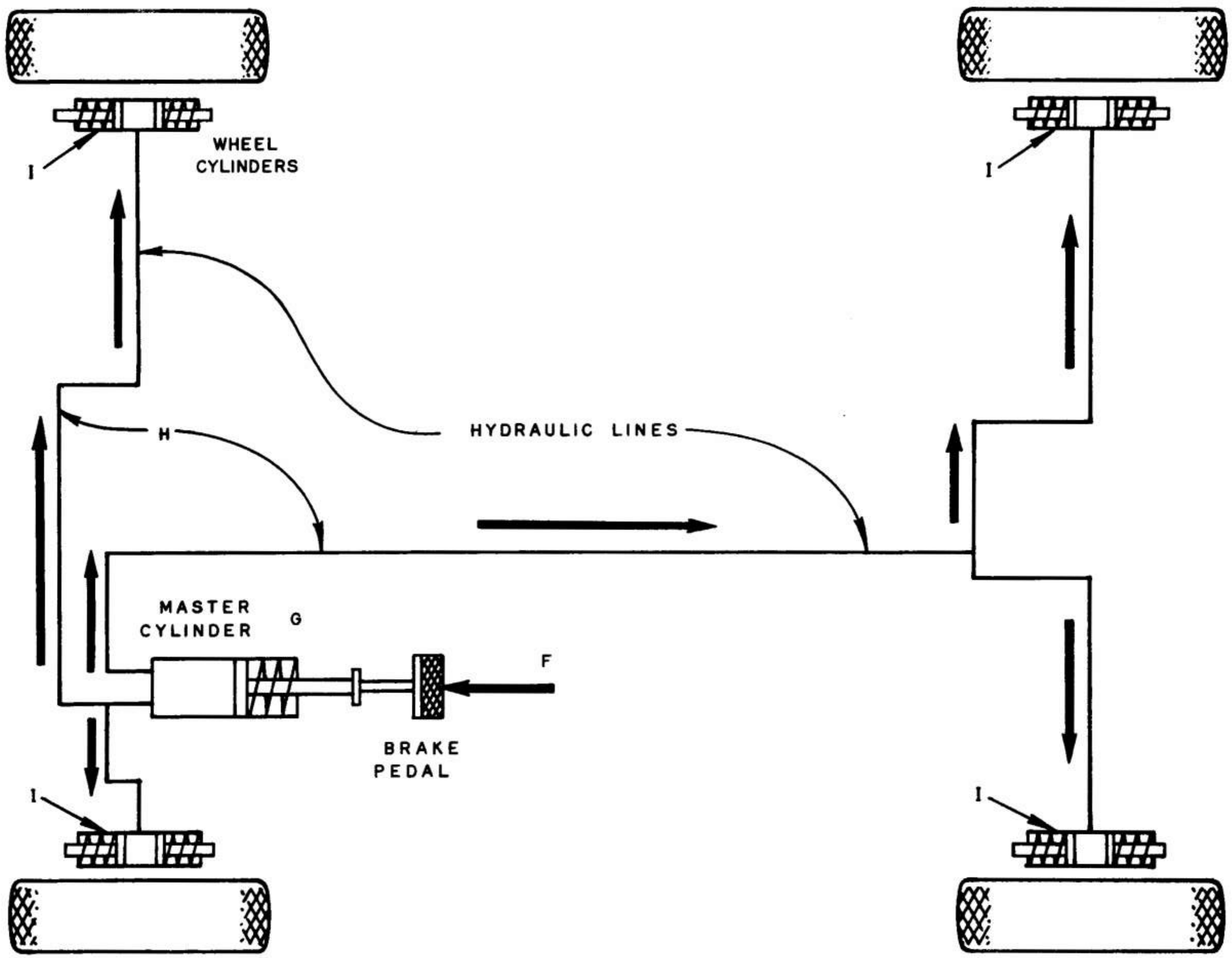


Figure 8-58. *Elements of a Hydraulic Actuation System, Manual Brakes*

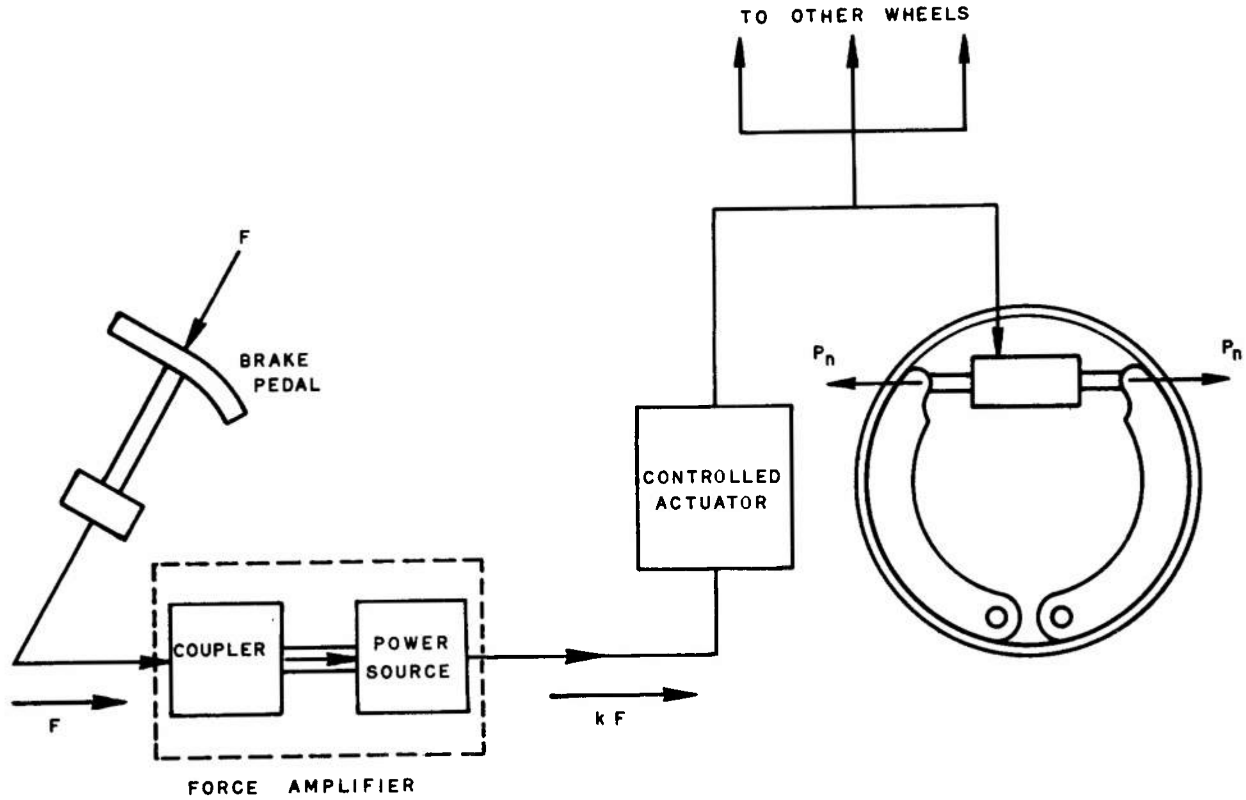


Figure 8-59. Elements of a Power-Boosted Braking System

Transmission and transfer assembly brakes, or propeller shaft brakes, are theoretically more efficient than wheel brakes, since the braking effort is multiplied by the final-drive ratio, and theoretically, the braking action is perfectly equalized through the differential.

Propeller shaft brakes, however, in contrast to wheel brakes, put a severe strain on the power transmission system. Furthermore, propeller shaft brakes permit differential action to occur, and, hence, variations in the braking forces at the wheels can take place in an actual system.

8-29 BRAKE ACTUATION SYSTEMS

There are three classes of actuation systems: manual, power-boosted manual, and full power actuation.

8-29.1 MANUAL ACTUATION

In the manual actuation system, braking force is obtained by multiplying the force exerted by the operator. The manual pedal (or lever) effort is transmitted to the brake mechanism mechanically or hydraulically.

8-29.1.1 Mechanical Actuation

Manual *mechanical* actuation systems are obsolete except for motorcycles (and similar vehicles) and for auxiliary (parking) brake systems. In the past, rods have been used throughout the mechanical actuation systems. Current systems employ flexible steel cables enclosed in flexible sheaths in conjunction with a system of rods. The flexible cables are located at the wheels so that the tension of the system is not affected by wheel motion. All of these systems have means of adjusting the lengths of the elements and in some cases, the system includes an equalizer. The purpose of the equalizer is to equalize the force exerted on the actual brake elements of the various wheels. The major disadvantage of a pure mechanical actuation system is the difficulty in maintaining an equal pressure on all brakes, even when equalizers are used. The manual-hydraulic system has replaced the pure mechanical system.

8-29.1.2 Hydraulic Actuation

Manual-hydraulic actuation is used extensively on vehicles weighing up to 3 tons, including civil-

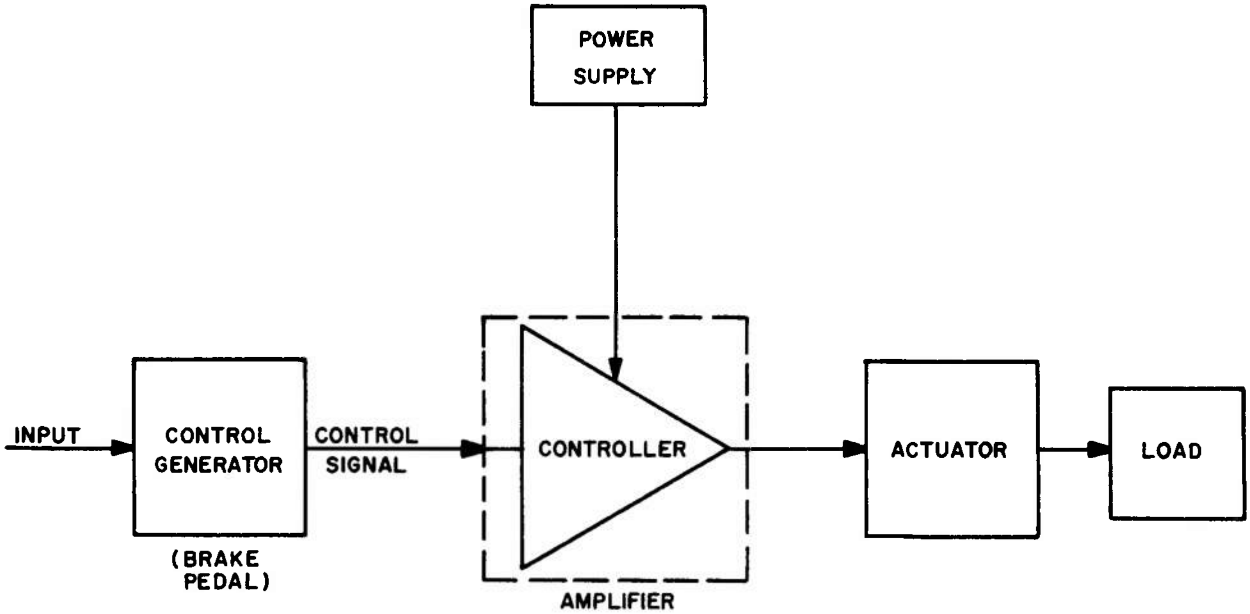


Figure 8-60. *Block Diagram of a Power Operated Brake Actuation System*

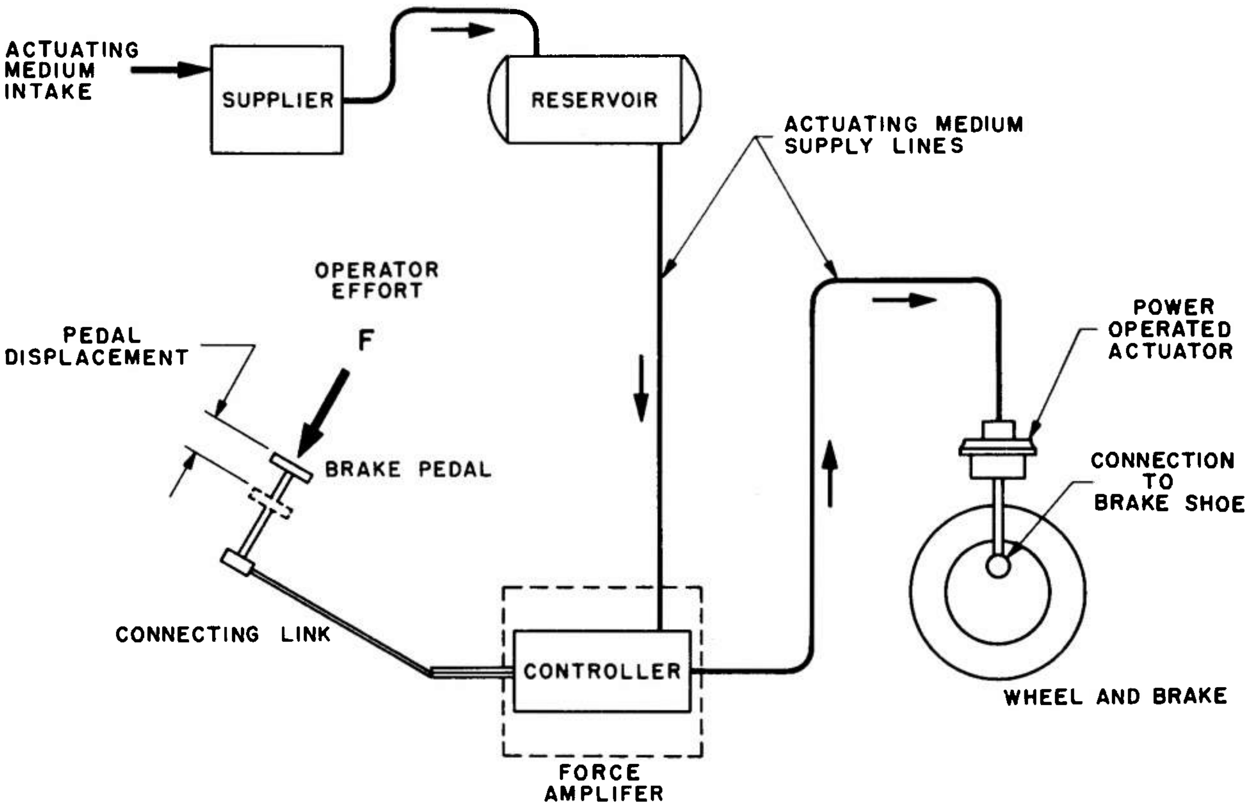


Figure 8-61. *Elements of Power Operated Brake Actuation System*

ian passenger vehicles and light military vehicles.

In the manual-hydraulic brake actuation system, hydraulic pressure provides the necessary actuation force. Figure 8-58 illustrates the elements of a manual-hydraulic actuation system.

When the foot pedal, *F*, is depressed, the piston of master cylinder, *G*, is displaced. Fluid flows through the hydraulic lines, *H*, to the wheel cylinders, *I*. At the wheel cylinder, the fluid displaces two opposing pistons. Rod ends of the op-

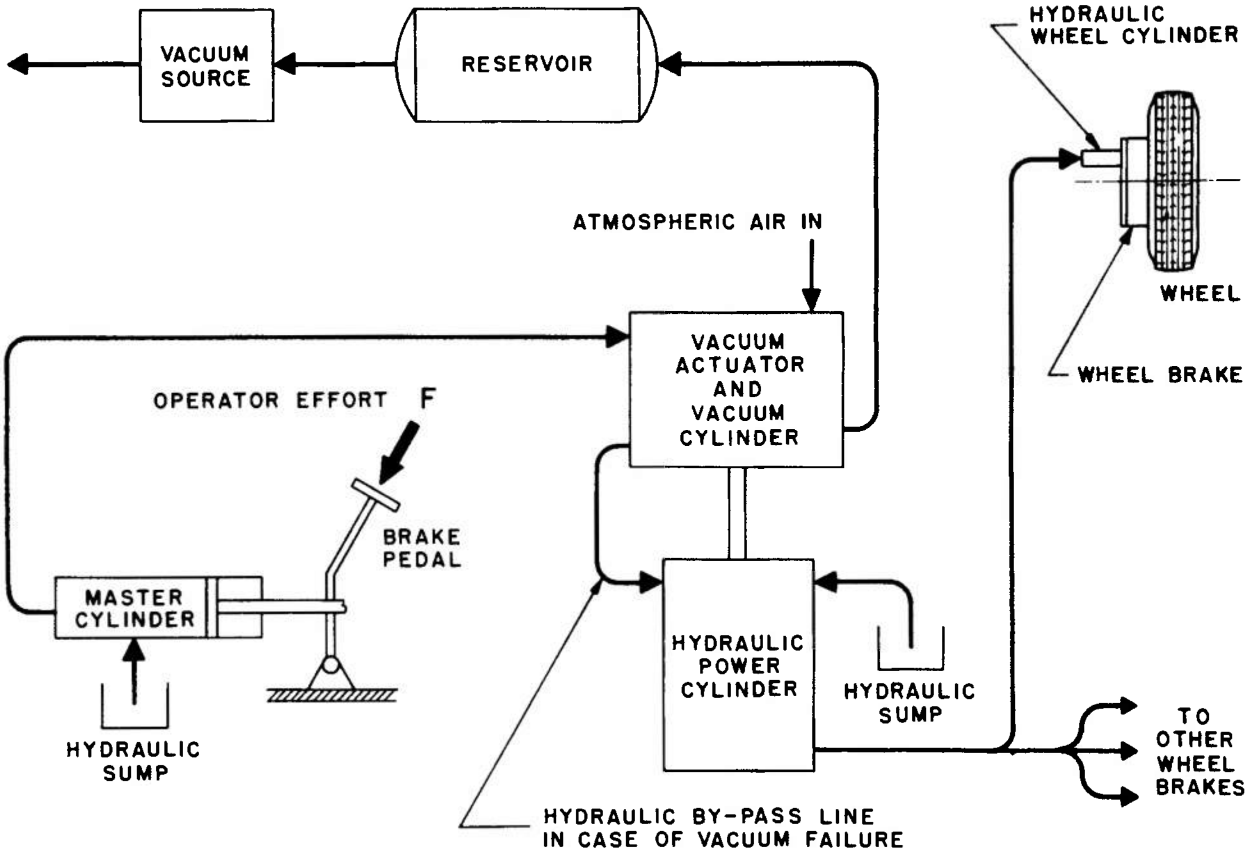


Figure 8-62. *Schematic Drawing of Vacuum-Hydraulic Actuation System*

posing pistons contact the brake shoes (not shown) and force them against the drum. A mechanical hand brake linkage may be provided to provide an additional means of braking.

8-29.2 POWER-BOOSTED ACTUATION

When gross vehicle weight of a vehicle is in the range of 3 to 6 tons, or when small off-the-road vehicles are subjected to severe loads, operator pedal efforts may be assisted by means of a servo system.

Basic elements of a power-boosted system are shown in Fig. 8-59. A force, F, applied to the brake pedal, is amplified at the amplifier to a value of KF, where K is greater than unity. In the amplifier, pedal effort is coupled to an auxiliary power source, which produces amplification in direct proportion to the pedal effort. The source of this auxiliary power may be vehicle momentum, vacuum, or hydraulic fluid or air under pressure, as discussed in the references.

8-29.3 POWER OPERATED SYSTEMS

Power operated actuation for braking is almost universal on large automotive vehicles above 6 tons, and in off-the-road vehicles subjected to severe service requirements. Figures 8-60 and 8-61 illustrate the basic elements of a power operated brake actuation system.

In the power-boosted system, the operator effort at the pedal is transmitted to the brake shoe in amplified form, hence, a force at the pedal is required. In the power operated actuation system, the only function of the brake pedal is one of control.

Displacement of the brake pedal causes a proportional displacement in the controller, which in turn causes a brake actuation proportional to the pedal displacement. Operator effort is determined solely by the friction and any preload present in the linkage between the pedal and the brake valve.

In some systems, a pressure, proportional to the brake actuation pressure, is impressed on a cylinder which opposes brake pedal motion. The

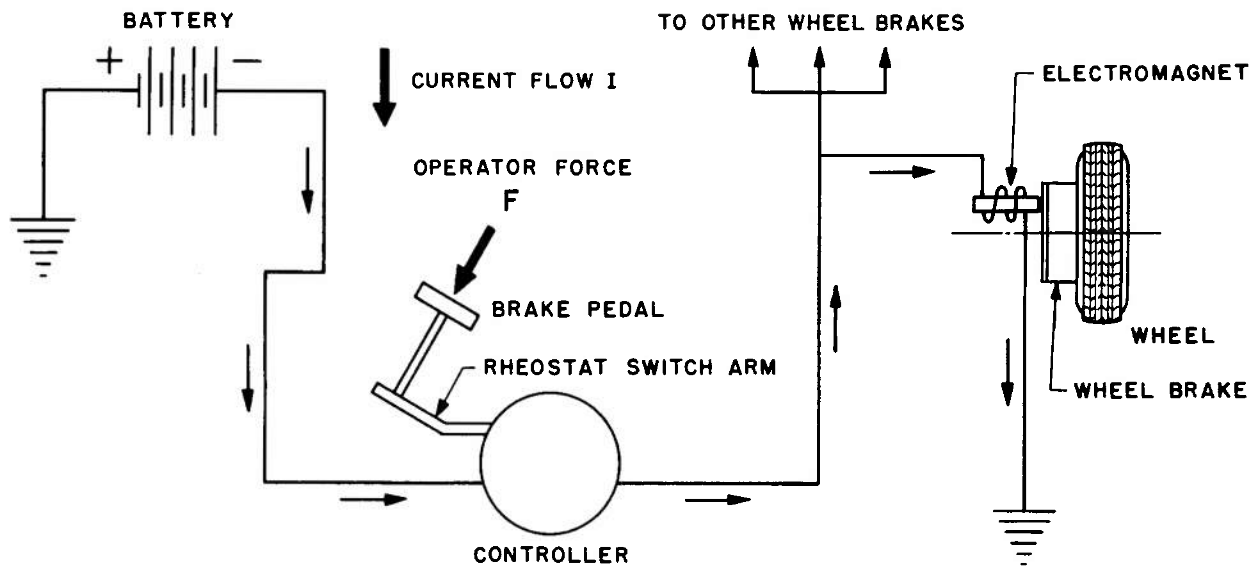


Figure 8-63. *Schematic of Typical Electric Brake System*

attempt is made, in this instance, to provide an artificial feel or road sense to the operator.

Details of operation and specific configurations vary depending on the actuating mechanism used. The means commonly employed in actuation include compressed air, hydraulic fluid under pressure, and vacuum.

Dual control media such as compressed air-hydraulic fluid, and vacuum-hydraulic fluid are also in use.

8-29.3.1 Air-Hydraulic Power Actuation System

The air-hydraulic system utilizes hydraulic brake actuation. Hydraulic pressure is obtained by means of an air actuated hydraulic cylinder. Actuation of the brake cylinder is by means of valves controlled by hydraulic pressure from a master cylinder, which in turn is controlled by the brake pedal.

8-29.3.2 Vacuum-Hydraulic Power Actuation System

The vacuum-hydraulic combination actuation system combines a vacuum servo with the standard master cylinder of the hydraulic actuation system. Figure 8-62 shows the elements of such a system, wherein hydraulic pressure, generated in the master cylinder, is the control input. This fluid under pressure controls a vacuum actuator, which, in turn, operates a vacuum cylinder in series with a hydraulic cylinder. Hydraulic fluid from the hydraulic cylinder enters the wheel cylinder which operate the brakes. The force applied to the brakes is proportional to the pressure in the master cylinder and, hence, to operator effort. Even though failure occurs in the vacuum system, pressure generated in the master cylinder will be felt at the wheel cylinders. This fail-safe feature sometimes results in the vacuum-hydraulic system being classified as a servo rather than a power operated system.

8-30 ELECTRIC BRAKE SYSTEM

Electric brakes are utilized as primary braking for large vehicles or as auxiliary brakes for other vehicles. The brake is powered from the storage battery. In the system of Fig. 8-63, the controller or rheostat controls the current to the wheel brakes in proportion to pedal depression. The wheel brake armature, revolving with the brake drum, is kept in contact with an electromagnet by means of flat springs. When the brake pedal is depressed, the electromagnet, which can travel through a limited arc when it is coupled to the armature, attracts the armature. As the magnet travels through this arc, it turns, engaging a cam lever. The cam lever actuates a quick rise cam which forces the brake shoe against the drum.

If the vehicle is not in motion when current is supplied to the brake, braking does not occur. Should the vehicle begin to move, braking occurs.

TABLE 8-2 FRICTION COEFFICIENTS AND ALLOWABLE PRESSURES FOR BRAKE MATERIALS

MATERIALS IN CONTACT	FRICTION COEFFICIENT			ALLOWABLE PRESSURE, psi
	Dry	Greasy	Lubricated	
Cast iron on cast iron	0.2-0.15	0.10-0.06	0.10-0.05	150-250
Bronze on cast iron	. . .	0.10-0.05	0.10-0.05	80-120
Steel on cast iron	0.30-0.20	0.12-0.07	0.10-0.06	120-200
Wood on cast iron	0.25-0.20	0.12-0.08	. . .	60-90
Fiber on metal	. . .	0.20-0.10	. . .	10-30
Cork on metal	0.35	0.30-0.25	0.25-0.22	8-15
Leather on metal	0.5-0.3	0.20-0.15	0.15-0.12	10-30
Wire asbestos on metal	0.5-0.35	0.30-0.25	0.25-0.20	40-80
Asbestos blocks on metal	0.48-0.40	0.30-0.25	. . .	40-160
Asbestos on metal, short action	. . .	. . .	0.25-0.20	200-300
Metal on cast iron, short action	. . .	. . .	0.10-0.05	200-300

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CHAPTER 9

THE FRAME*

SECTION I GENERAL DISCUSSION

The automotive vehicle frame is an assembly made of stamped or rolled metal structural members which performs or contributes to the performance of three basic functions. First, the frame supports such chassis components as the engine, suspension members, driveline elements, and exhaust systems and maintains the proper alignment and relationship between them. Secondly, the frame, in conjunction with the body, resists or absorbs the dynamic loads caused by torque reactions. Finally, the frame provides a base or foundation for the passenger and cargo compartments.

Civilian passenger vehicle frames are designed primarily for rigidity since structural stiffness is important to riding quality and vehicle control. When a high degree of torsional rigidity is achieved within these frames, the stresses in the members are relatively low. The bodies of passenger vehicles contribute to the overall stiffness in varying degrees, but generally both frame and body are required to obtain the desired rigidity. Several types of frames have become standard for civilian vehicle use (Refs. 2 and 4).

Truck frames are designed primarily for strength and durability. Most truck frames currently used in the United States are of the ladder type (Fig. 9-1), having straight channel side members of varying depth and a number of transverse cross members. This type of frame has very low torsional rigidity, and in off-the-road operations is permitted to deflect appreciably, thus tending to conform to the terrain contour. Development activity shows that more rigid frames and improved suspension systems would improve the riding characteristics of trucks. Since truck cabs and cargo compartments are flexibly mounted, the frame provides virtually all vehicle rigidity and strength.

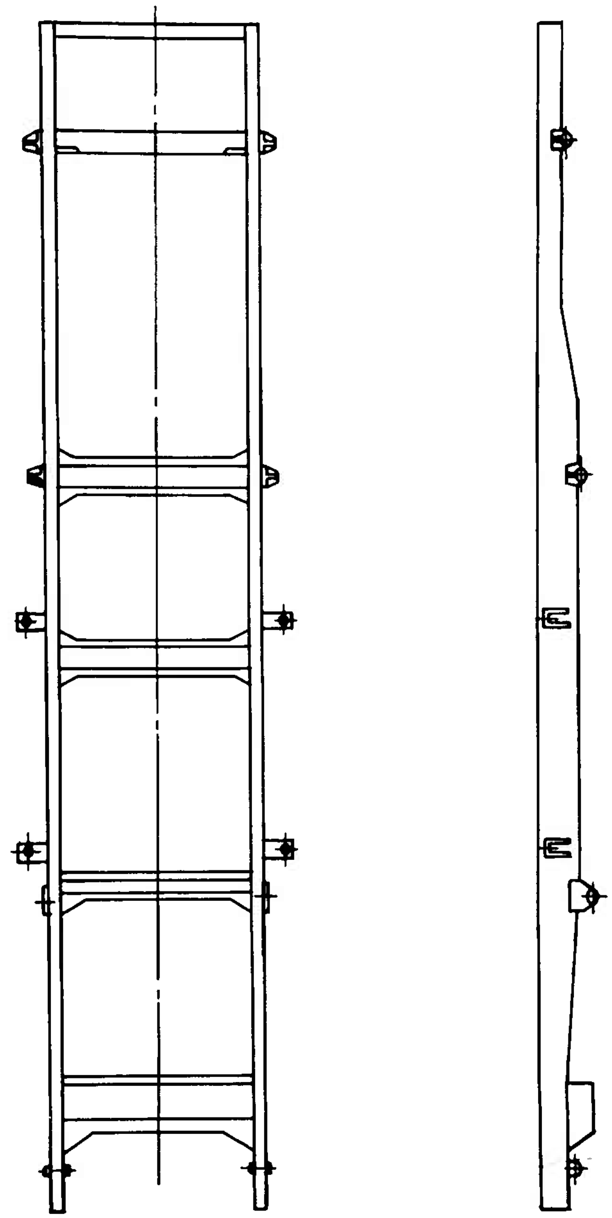

Figure 9-1. *Truck-Type Frame*

* Written by Nicholas R. Rome of the Illinois Institute of Technology Research Institute, Chicago, Ill.

SECTION II FRAME CONSTRUCTION

9-1 MATERIALS

Steel used in automotive frames varies with the vehicle size and capacity. Frames for civilian passenger cars and lightweight trucks, which often require extensive metal forming, are made of low-carbon SAE-1010 or SAE-1015 steel. Medium-duty truck frames use steels of somewhat greater strength, SAE-1015 or SAE-1020. Heavy-duty frames frequently use heat-treated high-manganese SAE-1027 or SAE-8620 steel or high-strength, low-alloy SAE-950 steel to reduce weight. Aluminum is occasionally used in heavy-duty truck frames to further reduce weight. Unitized construction combines the body and frame into one unit. Low-carbon steel or aluminum-alloy sheet stock is usually used for unitized bodies.

9-2 FRAME ELEMENTS AND JOINTS

Frames for conventional military vehicles consist of two longitudinal or side members, cross members, gussets, and various mounting brackets. Components are riveted or welded together to form an approximate rectangle. Both longitudinal members and the cross members that join the longitudinal members may be of various cross section, e.g., tubular, channel or I-beam. In general, cylindrical tubular members have the greatest torsional rigidity while I-beams have the greatest beam strength for given weights and lengths.

9-2.1 SIDE MEMBERS

The side members in truck frames, often called side rails, are usually parallel to each other at standardized SAE widths (34 in, maximum tolerance + 5/8-in, − 0 in) to permit the mounting of standard transmissions, transfer assemblies, axles, and other units. Kickups, used in the side members of civilian passenger vehicles to lower the center of gravity of the vehicle and provide adequate room for wheel and axle displacement, are not normally used in truck frames.

Side members are usually fabricated of pressed steel shapes, although rolled-steel shapes are used on very heavy trucks. Rolled sections usually have better physical properties than pressed sections; but, because they are necessarily of the same cross section throughout their length while loads vary along the length, the use of rolled sections leads to a waste of material and an excess frame weight.

The pressed longitudinal members of truck frames are usually channel sections with the depth of the section decreasing toward the ends. The channel section is often converted to a box section for part of its length by welding additional plates across the channel opening.

Beam strength of the frame is dependent on the side and cross members and is directly affected by any offsets in the horizontal plane, i.e., relatively abrupt changes in frame width. A horizontal offset in the beam introduces torsional stresses in addition to the bending and shear stresses within the member and should be avoided.

Transverse or cross members do not affect the longitudinal beam strength except as the joint affects the strength of the side members. The transverse beam strength of a frame, however, depends entirely upon the strength of the cross members. The X-type cross member contributes to the overall beam strength both longitudinally and transversely. As in every structure, localized failures can occur in frames when stresses are concentrated. Therefore, the side members must be of sufficient overall strength and should be free of actual or potential stress raisers.

9-2.2 CROSS MEMBERS

Frame cross members function to: (a) locate and maintain the alignment of the side rails, (b) provide a support or mounting base for various chassis components, (c) increase the torsional and longitudinal rigidity of the frame as a whole, and (d) provide lateral beam strength and, in some cases, adds to the longitudinal beam strength. Cross members can have various sections. Tubular members have been used; although, currently, cross members are flanged channel sections made of pressed steel.

To a large extent, the type and location of the cross members determine the overall torsional rigidity of the frame. Since the side members are distorted when the entire frame is distorted by torsional loading, they also contribute to the overall torsional stiffness.

Two basic types of cross members, the trans-

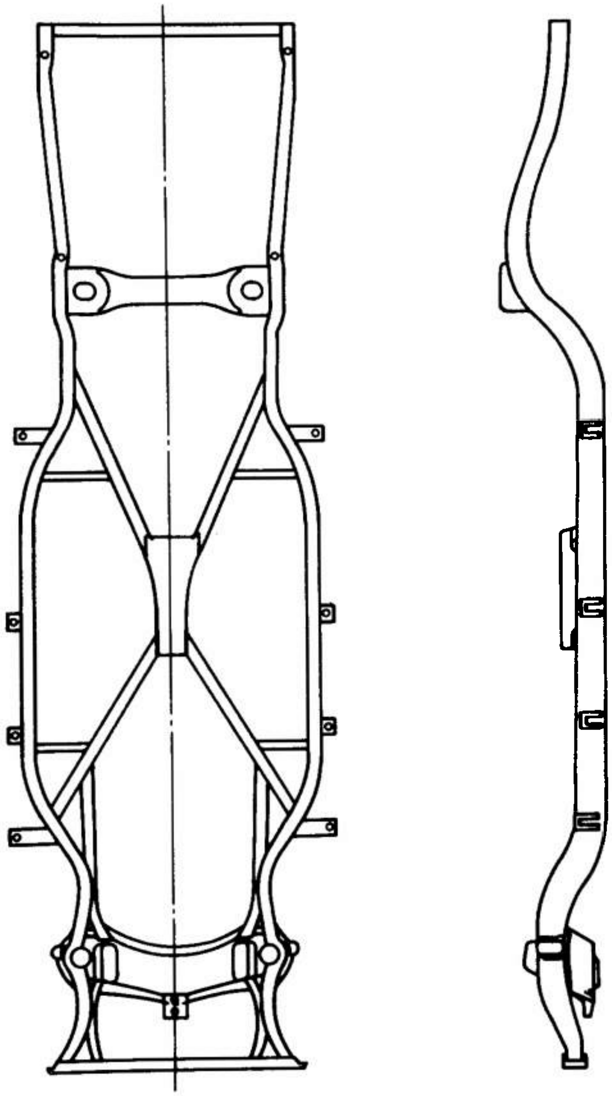

Figure 9-2. *X-Cross Member-Type Frame*

verse and the X-type, are used in military vehicles. Transverse (perpendicular to the frame) members may be used throughout the vehicle, or the intermediate cross member may be of the X-type. Figure 9-2 shows a typical X-type frame.

The ladder-type frame (parallel side members and transverse cross members as shown in Fig. 9-1) is usually used for large civilian trucks and military vehicles. Because it is inherently less rigid in torsion than some other designs and can be made more rigid only by adding more cross members, a ladder-type frame tends to be heavy for a given degree of stiffness. Another factor leading to a large number of cross members, hence weight increase, is localized twisting moments produced by the suspension components mounted on the outside of the narrow ladder frame. On the other hand, the ladder-type frame is simple to manufacture and has a high degree of adaptability in that one standardized frame can be used for a number of body styles.

The X-member is highly developed. Current practice extends the X-member over at least 1/3 of the frame length. As the X-member is extended longitudinally, its contribution to the total beam strength increases. For a given moment of inertia, the X-type frame is much more rigid than the diamond-, cross-, K-, or plain-frame. A comparison between a plain- (without cross members) frame and an X-member frame with an equal length of sides, load, modulus of elasticity, and identical moments of inertia, shows that the X-member frame would be over 50 times more rigid in torsion. The X-member frame, or some variation of it, is most commonly used for civilian passenger vehicles.

Other types of frames have been developed for use in passenger automobiles. Examples of these are the truss frame and the pressed-steel platform frame. The truss frame consists of a lattice structure formed by welding tubular members together. This structure is so designed that it forms a body and chassis frame; i.e., body panels and chassis components are fastened to the one-piece welded, three-dimensional structure. The pressed-steel platform frame is a one-piece unit to which the upper body is bolted or welded.

9-2.3 JOINTS

Both welding and riveting techniques are used to fabricate side and cross members. Two channels may be welded together to form a box section to be used as a side member. Frames have been completely welded, but riveted construction is more common. Rivets are used to join side and cross members and to fasten gussets and brackets into position. Both hot and cold riveting techniques are used. Hot riveting results in a high force which draws the two members together. The frictional force between the two parts adds to the shear resistance of the rivets. Cold riveting is also used extensively in frame construction. By this technique the shear loads are transmitted immediately and directly to the rivets. Bolts are sometimes used to fasten brackets or supports to frames but are seldom used to fabricate the frame *per se*.

To separate a vehicle into halves for transportation convenience, the frame side members are sometimes cut at the rear of the cab, and each section is fitted with flanges adequate to allow the two parts to be bolted together. Since this joint or connection will usually be at or near the point of maximum bending stress in the frame, it must be carefully designed to ensure adequate strength.

9-3 UNITIZED CONSTRUCTION

Unitized body, as applied to automotive vehicles, describes the type of construction in which a separate frame and body do not exist. In unitized construction, the entire frame-body structure is designed to support the expected beam and torsional load; while in the conventional frame plus body construction, the frame is the major load-bearing element and the body contributes secondarily to the total strength and rigidity of the combination. (This statement is true of trucks, only. In a typical passenger automobile, the frame supplies about 37% of the torsional rigidity and 34% of the bending rigidity.)

Unitized body construction is used extensively in civilian automobile fabrication; but, at present, is limited to several small military vehicles. The general method of fabricating unitized bodies is to weld together large panels that have been preformed to their final shape by large presses. Military and civilian agencies are conducting extensive research on unitized construction for large trucks. Both aluminum-alloy and steel bodies are undergoing testing and development. Compared to separate frame designs, unitized construction shows the following advantages and disadvantages:

(a) For given strength and deflection specifications and a given material, a well-designed unitized vehicle will weigh less than a comparable vehicle with separate frame and body.

(b) Unitized construction lends itself to uniform-stress design more readily than does separate frame designs. The large number of individual structural members of the unitized body makes it more practical to match the structure with the actual or anticipated loads.

(c) Unitized construction reduces the amount of vibration present in vehicles.

(d) Some of the current unitized vehicles are designed so that the engine, transmission units, drive shafts, brakes and parts of the axles are within the hull and thus protected from dirt, mud and water.

(e) A rigorous analysis may be required to achieve a high degree of weight reduction for unitized structures. Improperly designed unitized vehicles may weigh more than similar vehicles having separate frames.

(f) Ground and drive train noise transmitted to the crew compartment is greater in unitized structures. In the body plus frame design, the body can be mounted on rubber pads which act as sound barriers.

(g) The separate-frame vehicles have the advantage of interchangeability of body types. For example, a basic truck chassis can be used to mount cargo or van bodies.

SECTION III DESIGN CONSIDERATIONS

9-4 TYPE OF SERVICE

The required strength and configuration of an automotive vehicle frame depend on the type of service for which the vehicle is intended. Stresses and resulting strains are induced in a frame by static loads and dynamic forces. The stresses in the frame resulting from the static loads, such as the weights of the components and the payload, are determined by the classical methods of structural mechanics. Some of the stresses resulting from dynamic forces are relatively easy to calculate. These are the stresses induced by the acceleration or deceleration of the entire vehicle by the engine or brakes. Complex dynamic stresses (see Chapter 5, Section II) are induced in the frame members when a vehicle travels over rough ground or is subjected to high-energy blast or projectile impacts.

The magnitude and distribution of the load imposed on a vehicle varies with the type of service. The maximum payload distribution of a personnel carrier will be approximately uniform over the passenger compartment of the carrier. However, due to the average physical dimensions and weights of human beings, a 50% overload may be approached in a vehicle as the 2-1/2-ton truck, when the number of passengers exceeds the rated number for the vehicle. When coupled with the operator abuse, a factor to which military vehicles are usually subjected, this overload possibility becomes a major factor to be considered by the frame designer.

Cargo carriers are also subject to overloading. The stresses in the cargo carrier frame can be higher than those in a similar personnel-carrying vehicle with the same percentage of overload. Such would be the case if the cargo payload were concentrated in a small area of the cargo compartment. Dump truck frames are subjected to concentrated loadings at the dump-body pivot point and the actuating-cylinder pivot point during unloading. When a standard, stationary-body truck frame is used on a dump truck, special reinforcing members are needed to support the concentrated loads. Frames of prime movers and trailers must be designed to withstand, without failure or unacceptable deformations, the stresses that result from towing as well as those previously discussed.

Weapon systems and special-purpose equipment mounted on framed vehicles may produce variable loading due to the elevation and rotation of the mechanism and shocks caused by firing.

9-5 STRESS CONSIDERATIONS

The maximum stresses normally induced in frame members are due to the dynamic loads caused by road conditions and various other impacts. In current design practice, a number of factors are considered in selecting the structural members for a given frame and vehicle.

The bending moment and the shearing forces on a frame, caused by the static loads and the braking reaction forces, are studied by standard bending moment and shearing diagrams. The basic static load diagram is studied, both with and without the superposition of the braking reaction forces, since the vehicle may experience both conditions. The braking reaction forces may increase or decrease the stresses on the frame.

The design stress or allowable stress of the various frame members is determined by dividing the applicable material property, e.g., yield strength, and fatigue strength, by a factor of safety. The factor of safety is based on the uncertainties related to the design of the member. These uncertainties refer to such factors as magnitude and kind of operating loads, the material characteristics, fabrications stresses, and the validity of the assumptions upon which the theories used in the analyses are based. The design stresses and the loading profile for the frame permits the selection of the section moduli for the various members.

The proposed design must meet the torsional rigidity specifications as well as the strength (stress) specifications. Any design based on the strength criterion must be subjected to a deflection analysis. A frame must be designed on the basis of stiffness or rigidity and subsequently analyzed for stresses in the various members. In structural members (beams) used in frames under transverse loading, the deflection varies directly with the stress and inversely with the modulus of elasticity of the material used.

Both stress and deflection vary inversely with rectangular moment of inertia of the given beam. These values may be varied for a beam or member of given length, material, and loading by changing the rectangular moment of inertia of the member. Deflection is inversely proportional to the modulus of elasticity of the material used in a given member. Since the modulus of elasticity varies very little with the type of steel or its heat-treatment, attempts to change the rigidity of a steel member by changing the type of steel are not effective. When materials such as aluminum are used for frame members in place of steel, the lower modulus of elasticity of aluminum must be compensated for by an increase in section modulus if the original rigidity is to be retained.

9-6 MISCELLANEOUS CONSIDERATIONS

9-6.1 STABILITY OF THE VEHICLE

A vehicle frame is sometimes designed with an arch or kickup over each axle. This arrangement permits a lower center of gravity for the vehicle than that allowed by a straight (in the horizontal plane) frame. The height of the center of gravity has a direct influence on vehicle stability. The effect of an increase in the height of the center of gravity for a given wheeled or tracked vehicle is outlined as follows:

(a) The sprung mass will roll more (tilt about its longitudinal axis from its static horizontal position) for a given side force, e.g., centrifugal force. The increased roll for a given force results from the increase in the moment arm extending from the roll center to the center of gravity.

(b) The overturning moment experienced on a side slope is increased.

(c) The weight transfer between front and rear axles, which occurs during acceleration and braking, is increased.

Deflection of the frame also may affect the stability of a vehicle. Modern vehicles with independently sprung wheels depend on the geometry of the suspension linkages to control the camber angles of the wheels during bump and rebound. The designer, by using the proper linkages, can control the camber angle for the full path of the wheel. A frame with a relatively large degree of flexibility will permit unpredictable changes to occur in camber and castor angles. These changes affect the stability and control of the vehicle. Current vehicles have more rigid frames than earlier vehicles and superior suspension systems to reduce the impact loadings for given road conditions.

9-6.2 USAGE

Intended usage is the prime consideration in design of a vehicle frame. Special-purpose vehicles may not require the same degree of rigidity and strength as is required of tactical vehicles. The vehicle that is not subjected to severe off-the-road operations may gain in economy and reduction of weight by an adjustment of the safety factors.

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CHAPTER 10

THE BODY OR HULL*

SECTION I GENERAL DISCUSSION

10-1 DEFINITION

The body or hull of an automotive assembly consists of crew, passenger, and cargo compartments, and compartments for various components of the vehicle such as the engine. The compartments may be integral or separately mounted on a frame.

In the past, the term *body* was applied primarily to wheeled vehicles, and the term *hull* was applied to the body of amphibious and tracked vehicles, especially the massive tank. However, recent technical literature terms the lightweight unitized body of a wheeled or tracked vehicle the hull.

Like all other components used in military vehicles, bodies and hulls are designed to be mass-produced. Standard components are used whenever possible and readily available materials are specified. The turret and cupola assemblies of tank-type vehicles are not considered as part of the vehicle body or hull and therefore are not discussed in this chapter.

10-2 HULLS OF TYPICAL VEHICLES

10-2.1 HULLS OF TANKS AND ARMORED CARS

Tank hulls may be (1) welded assemblies of armor plates, (2) welded assemblies of armor plates and armor castings, or (3) one-piece armor castings. The hull serves as the frame, the crew compartment, and the equipment compartment. The hull also serves as a base for the turret and cupola in the case of tanks (Fig. 10-1). Removable or hinged sections are provided for the installation and servicing of components such as the power plant and to provide access for personnel. Various provisions, such as mounting holes and brackets, are made for fastening suspension members, guns, vision devices and other equipment to the hull. Towing and hoisting provisions are incorporated in the hull design.

The interior of the hull is normally divided into an engine compartment and a fighting compartment by a lateral bulkhead which also strengthens the hull assembly and seals the compartments against the passage of gases and liquids. The hull of an armored car is constructed in a similar manner, although it normally is made of lighter weight plate.

10-2.2 HULLS OF CARGO CARRIERS AND PERSONNEL CARRIERS

The hulls used for unitized cargo and personnel carriers may or may not be armored. The current unarmored vehicles have hulls consisting of sheet steel or aluminum alloy. Both welding and riveting are used in the fabrication of these lightweight unarmored vehicles. Armored carriers are currently constructed of steel or aluminum-alloy plates. The plate thickness for different areas of the vehicle varies depending on the strength and ballistic requirements. The weight reduction resulting from the use of aluminum-alloy hulls, in place of steel hulls, influences the performance and increases the air transportability of these vehicles. For example, the M113 aluminum-armored personnel carrier, which weighs 11 tons less than its 21-ton, steel-armored predecessor, the M59, has a range of 200 miles compared to the M59 vehicle range of 120 miles. The maximum speed of the M113 is increased from 32 to 40 miles per hour, and the fuel consumption rate reduced from 1.0 to 2.6 miles per gallon.

10-2.3 HULLS OF AMPHIBIOUS VEHICLES

The hulls of true amphibious vehicles, Figs. 4-29 and 4-30 of Chapter 4, differ fundamentally

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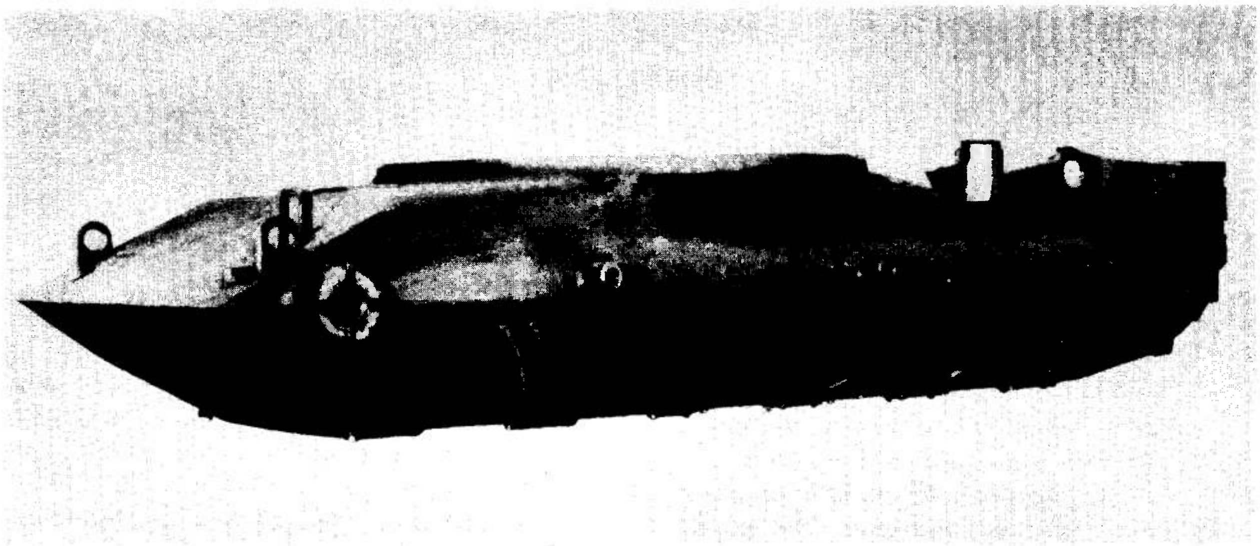

Figure 10-1. *Typical Tank Hull*

from the hulls of all other military automotive vehicles. True amphibious vehicles have hulls that are designed according to the principles of marine engineering. Factors such as stability, resistance, and power required for propulsion in water are of primary importance in the design of the hull. To increase the efficiency of these vehicles, i.e., to achieve satisfactory vehicle speed-propulsion power ratio, it is necessary to incorporate retracting wheels (or tracks) into the design and to use an efficient form of marine propulsion, such as propeller drive or hydro-jet drive.

The hulls of true amphibious vehicles are of welded steel or nonferrous alloy construction.

10-2.4 MATERIALS USED FOR HULLS

The hulls of standard military vehicles are currently fabricated of either steel or aluminum alloy. Experimental work is being conducted on the use of magnesium and titanium as materials for hulls and bodies.

Titanium and titanium alloys possess an unusual combination of properties. The alloys have, at room temperature, ultimate strengths ranging from 115,000 to 150,000 psi, yield strengths ranging from 100,000 to 140,000 psi, elongations ranging from 10% to 15% in 2 inches, and densities of approximately 0.16 lb per cu in. These values indicate that the strengths of titanium alloys are comparable to alloy steels, and their densities are only about 60% as great. In addition, the corrosion resistance of titanium alloys is superior to that of aluminum and stainless steel under most conditions, particularly when salt-water spray is present.

The major disadvantages of titanium have to do with its processing. Titanium melts at 3,150°F and at this temperature is extremely reactive chemically. It reacts rapidly with the atmosphere to form titanium nitride and the oxides of titanium. It also reacts, when molten, with carbon monoxide, carbon dioxide, and hydrogen. This behavior imposes penalties from the manufacturing point-of-view. When titanium is in the molten state, it must be kept in a vacuum or in an atmosphere of properly prepared inert gases such as argon or helium. Furthermore, molten titanium reacts to a varying but prohibitive degree with all known refractory materials. Finally, titanium has a maximum recommended operating temperature of about 1,000°F for extended service. At elevated temperatures (above 1,500°F) the surface of the metal absorbs oxygen and nitrogen from the atmosphere causing surface hardening which may be undesirable.

SECTION II TYPE OF SERVICE

10-3 TRANSPORTATION VEHICLES

Tactical transportation vehicles may either be wheeled or tracked. The functions of these vehicles are described in Chapter 4. The general requirements for an automotive assembly are discussed in Chapter 3. The following comments are intended as a review of some of the factors relative to body or hull design. Quantitative values are given in Chapter 3.

The present day emphasis on airborne and seaborne operations imposes new requirements on all tactical transportation vehicles. Vehicle size and gross weight are important considerations. Door sizes of planes and ramp openings of landing vehicles or barges place limits upon the overall dimensions of vehicles. Bodies and hulls must be designed so that with standard accessories in place, they can be loaded and unloaded from the intended carrier. The approach and departure angles of the various vehicles must be considered so that they can negotiate standard loading ramps.

Many current and proposed military vehicles are capable of both land and water operation. The problems of weight distribution and vehicle stability for both land and water operations must be considered. Problems encountered in launching or landing amphibious vehicles and the problem of buoyant stability are discussed in Chapter 5.

Since the body or hull of a vehicle may support all or part of the transported load, the weight, bulk, and distribution of the expected loads should be considered. Some of these factors are discussed in Chapter 9.

Towing and suspension loads in framed vehicles are normally transmitted to the frame itself. In unitized vehicles, these loads are transmitted to the hull and must be considered when a hull is analyzed for stresses.

10-4 COMBAT VEHICLES

10-4.1 GENERAL

Many of the factors discussed in the previous paragraph apply to combat vehicles also. However, when a vehicle is designed primarily for combat, the armament and armor of the vehicle greatly affect the hull design.

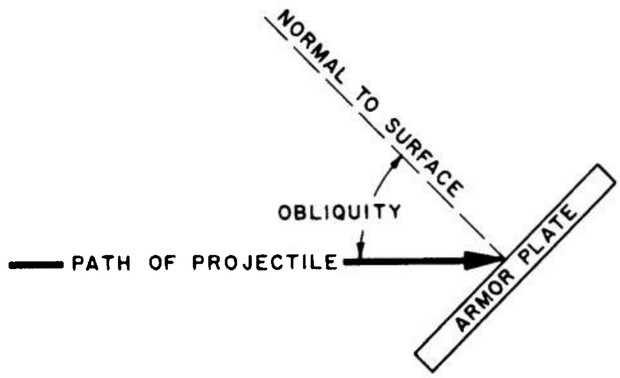

Figure 10-2. *Illustration of Obliquity*

The primary weapon of a tank is normally mounted in the turret. Although the turret is not considered a part of the hull, the hull must provide for mounting the turret. Vehicles without turrets may have provisions for mounting the weapons on the hull proper. Since the major weapons and their control systems normally occupy space within the vehicle, the hull should be designed accordingly.

A distinctive feature of most combat vehicles is the armor which may be applied to all or part of the vehicle. The design of armored hulls is a specialized field in which a knowledge of the effect of projectiles and explosives is required. The following comments indicate some of the areas and problems related to armor. One of the initial steps in the design of armored hulls should be a thorough study of the literature in this field. For a comprehensive study of armor and its applications, see Ref. 9.

Considerations in evaluating the degree of protection afforded to the personnel and the equipment of an armored vehicle may be classified under four headings.

a. Basic armor protection concerns the type, thickness and obliquity (the angle formed by the path of the projectile and the normal to the face of the armor at the point of contact, Fig. 10-2) that each armored surface presents to an attacking projectile. If these three factors are known, it is possible to predict at what minimum range and from what direction of attack a specific projectile can be defeated by an armored area.

b. Exterior design and fabrication covers items such as the strength and design of welded joints, resistance to blast of tracks and suspension system, vulnerability of vision devices, and the basic principles involved in providing the maximum protection against the most likely attacks.

c. Design of openings and movable components deals with the protection devices that are used to prevent projectiles and bullet splash from entering openings in the armor, and the design and location of exterior movable components to minimize the probability of ballistic immobilization.

d. Interior design concerns locating and shock mounting equipment to minimize the probability of damage by shock, blast or penetrating fragments or projectiles and with providing the maximum protection for the crew.

10-4.2 SPECIAL CONSIDERATIONS

One of the primary concepts related to the design of armored hulls is that of equalization of protection. Once the probability of projectile impact on a given area of the hull is established, the entire section should be provided with the same degree of protection to the extent practicable. Both obliquity and plate thickness must be considered. Equalization of protection and the probability of damage must be considered when the underside of the vehicle is designed. Since land mine detonation will usually occur beneath the front section of an armored vehicle, the front section of the underplate is made heavier than the rear section. In general, a greater degree of protection is provided for the crew than for components such as the engine. The silhouette and surface configuration of armored vehicles influence the vulnerability of the vehicle. Flat surfaces and convex surfaces are superior to concave surfaces or any surfaces which form a re-entrant angle. Convexity, formed by flat or curved plates, is striven for in the design of armored hulls. A projectile striking within a re-entrant angle may be deflected from a heavily armored section to a lightly armored section. It is also important to protect the junction of two movable sections (such as the junction between the hull and the turret) from direct and deflected impacts.

Bullet splash is the dispersion of finely divided or melted metal produced upon impact of a projectile with armor plate or other hard objects. These fragments travel at extremely high speeds

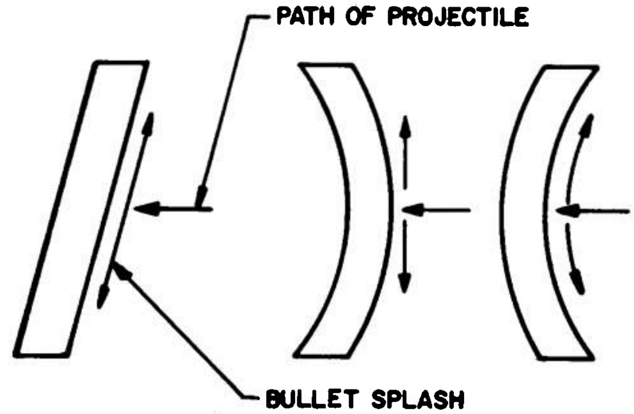

Figure 10-3. *Characteristic Patterns of Bullet Splash on Various Surfaces*

and are capable of injuring personnel or equipment. In general, bullet splash travels along a plane tangent to the armor at the point of impact. Consequently, the bullet splash from impacts on a convex surface will travel away from the surface of the armor; while splash from impacts against a flat or a concave surface will travel along the surface until it becomes convex or until the bullet splash is deflected by an irregularity in the surface (Fig. 10-3). Oblique impacts produce splash concentrated toward the direction of original flight (however, the splash, even under these conditions, will normally occur 360° around the impact area). Because bullet splash behaves as a high-velocity fluid, it can be turned in several directions and still cause damage. Three right-angle turns are considered the minimum number necessary to expend the harmful energy of bullet splash. Since bullet splash can pass through relatively small openings, careful attention must be given to its control at hatch covers, air vents, vision ports, gun shields, or anywhere else an opening occurs in the armor structure. Bullet splash is controlled in combat vehicles by means of baffles or traps. These deflect the splash and absorb its kinetic energy. In some cases, the splash is turned back along its original course by means of specially designed deflecting surfaces. An illustration of a splash trap is shown in Fig. 10-4.

Combat vehicles must include storage space for ammunition and fire control equipment. The location of the ammunition racks or storage bins within a combat vehicle presents a problem. It would be highly desirable to locate the ammunition racks in the lower section of the vehicle away from

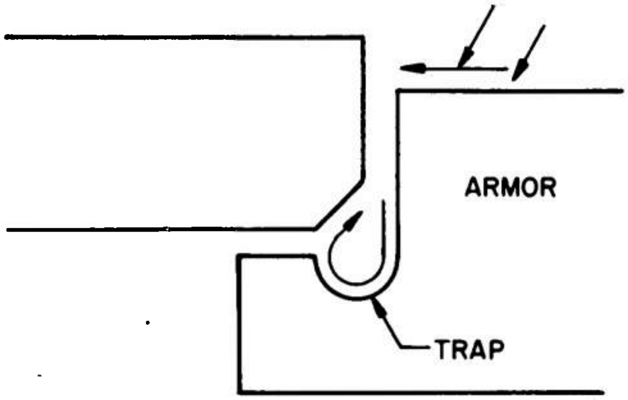

Figure 10-4. *Typical Splash Trap*

the crew compartment but ammunition must be readily available to the gun loader. This requirement necessitates that ammunition be stacked in the fighting compartment. This arrangement is not entirely satisfactory. An attempt to reduce the vulnerability of the stored ammunition is made by providing sufficient clearance between the racks and the wall and belly armor of the tank or other vehicles so that bulges in the armor will not contact the ammunition. Generally, the ammunition is stowed as low as possible in the hull.

10-4.3 SUMMARY OF ARMORED COMBAT VEHICLE REQUIREMENTS

The armored hull and related components, such as the turret, are results of design compromises. All factors must be weighed carefully and an acceptable design agreed upon.

In summary, the armored hull serves as the chassis and also has the function of affording protection against attack by various weapons. Therefore, considering the hull design principles, the designer must:

a. Apportion the armor so that the greatest thickness of hull armor is applied in front, and the next greatest protection is applied to the sides, roof, floor, and rear, in that order. Protection against mines dictates floor thickness.
b. Select the most effective and efficient obliquity of armor with respect to weight and ballistic properties, remembering the need for equalization of protection.
c. Design the front section to be as smooth as possible, eliminating all unnecessary joints, abrupt changes in obliquity and external equipment.
d. Avoid re-entrant angles.

10-5 ADMINISTRATIVE VEHICLES

Military vehicles used for administrative and technical services normally are either standard civilian vehicles, or slightly modified civilian vehicles. Standard civilian vehicles are highly developed vehicles designed for a specific purpose. The military vehicle designer is normally not required to design civilian-type vehicles, however, he must become familiar with the performance and intended use of these vehicles in order to evaluate and select standard civilian vehicles for military purposes. The type, size, and weight of material to be placed in the vehicle and the accessibility requirements of the material must be considered when administrative vehicles are evaluated or selected.

10-6 HUMAN ENGINEERING CONSIDERATIONS

Some of the human engineering factors that apply to military vehicles in general are discussed in Chapter 3. The present section is limited to several topics in the field of human engineering directly related to hull or body design: (a) the location of escape hatches, (b) temperature and ventilation considerations, (c) noise considerations, and (d) maintenance considerations.

10-6.1 ESCAPE HATCHES

Military vehicles are provided with various doors and hatches through which the crew and passengers gain entrance to the vehicle. In addition to these, tanks are usually provided with an escape hatch located in the underside (belly) of the vehicle. This exit is of particular value when the tank is on fire, as the flames and heat naturally reach upward, away from this escape route making escape possible from an otherwise hopeless situation. Tanks are often used to assist in the recovery of badly wounded troops from fire-swept battlefields. In this operation, the tank maneuvers into a position astride the helpless man, whereupon he is lifted into the safety of the armored hull through this belly hatch.

Experience with fully enclosed vehicles operating on water, as in amphibious operations, has shown that troops become apprehensive when they are deprived of means to visually observe their outside surroundings. Therefore, it is important that suitable viewports be incorporated into the hull design of fully enclosed vehicles. These must be accessible, not only to the driver, but to the crew

TABLE 10-1 NECESSARY VENTILATION RATE

	Oxygen Consumption per Person at Sea Level, cu ft per min	Ventilation Rate Per Person to Maintain Concentration of CO_2 Below 0.5 percent, cu ft per min			
		Sea Level	5,000 ft	10,000 ft	15,000 ft
At rest	0.008	1.2	1.4	1.7	2.1
Moderate Activity	0.028	3.9	4.7	6.7	6.9
Vigorous Activity	0.056	8.7	9.7	11.7	14.5

and passengers as well. Even though these ports permit only limited observation, they have a great influence upon relieving the nervous anxiety that is otherwise experienced by the occupants.

Turret baskets that have only one position access openings are also disliked by the personnel who use them. When the turret is rotated so that the exit opening is blocked, the turret crew experience a trapped feeling. To avoid this, turret baskets should be provided with as many access openings as other design considerations permit.

Fully enclosed amphibious vehicles of the M113 type (see Fig. 4-17) have passenger access through the large, downward opening door at the rear. During water-borne operations, this main access door is almost completely submerged. Opening it to affect an emergency escape while at sea would be immediately disasterous. Thus, escape hatches for emergency use must be provided in the top of the vehicle where they can be opened without danger of immediately swamping the vehicle.

Another condition that must be considered when locating escape hatches, particularly in an amphibious vehicle, is escape from a capsized vehicle. In spite of the stability and low center of gravity of amphibious vehicles (M113 type), it is possible for them to capsize, especially when entering the water from a high bank. Since their hulls are in stable equilibrium when capsized (see par. 5-2.5.1), they will not right themselves. Escape hatches should, therefore, be provided that will permit escape under these conditions.

10-6.2 TEMPERATURE AND AIR CONDITIONING

The temperature within a closed vehicle, such as a tank, will be affected by ambient conditions, the components or equipment of the vehicle (the operating power plant is a source of heat within the vehicle), and the personnel within the vehicle. The heating, cooling and ventilating specifications of a vehicle are based on the physiological requirements of the using personnel.

Sealed vehicles such as tanks or track-laying personnel carriers are equipped with ventilating systems which supply fresh air and remove the carbon dioxide, carbon monoxide, and other gases generated by the main or auxiliary power plants of the armament.

Sufficient oxygen is necessary to human life. The normal volumetric oxygen content air is approximately 21%; this may be reduced to 14% in enclosed compartments without harmful effect on personnel. The minimum volumetric content for normal breathing is 10%.

Table 10-1 shows the necessary ventilation rates for various degrees of activity at various altitudes.

Carbon dioxide will be generated in the closed personnel compartment even if no leakage from the engines or other equipment occurs. In the enclosed space occupied by personnel, the carbon dioxide content, by volume, should not be greater than 0.5%; 1% to 2% may not be noticeable but may reduce a person's efficiency. When more than 3% carbon dioxide is present, a slight effort in breathing is noticed. With between 5% and 10% carbon dioxide air-content, a person will breathe heavily and tire quickly. More than 10% of carbon dioxide may prove fatal if exposure is continuous.

The temperature and moisture content of the atmosphere influence the functioning of human beings. There are ranges of atmospheric temperature and humidity for human comfort and toler-

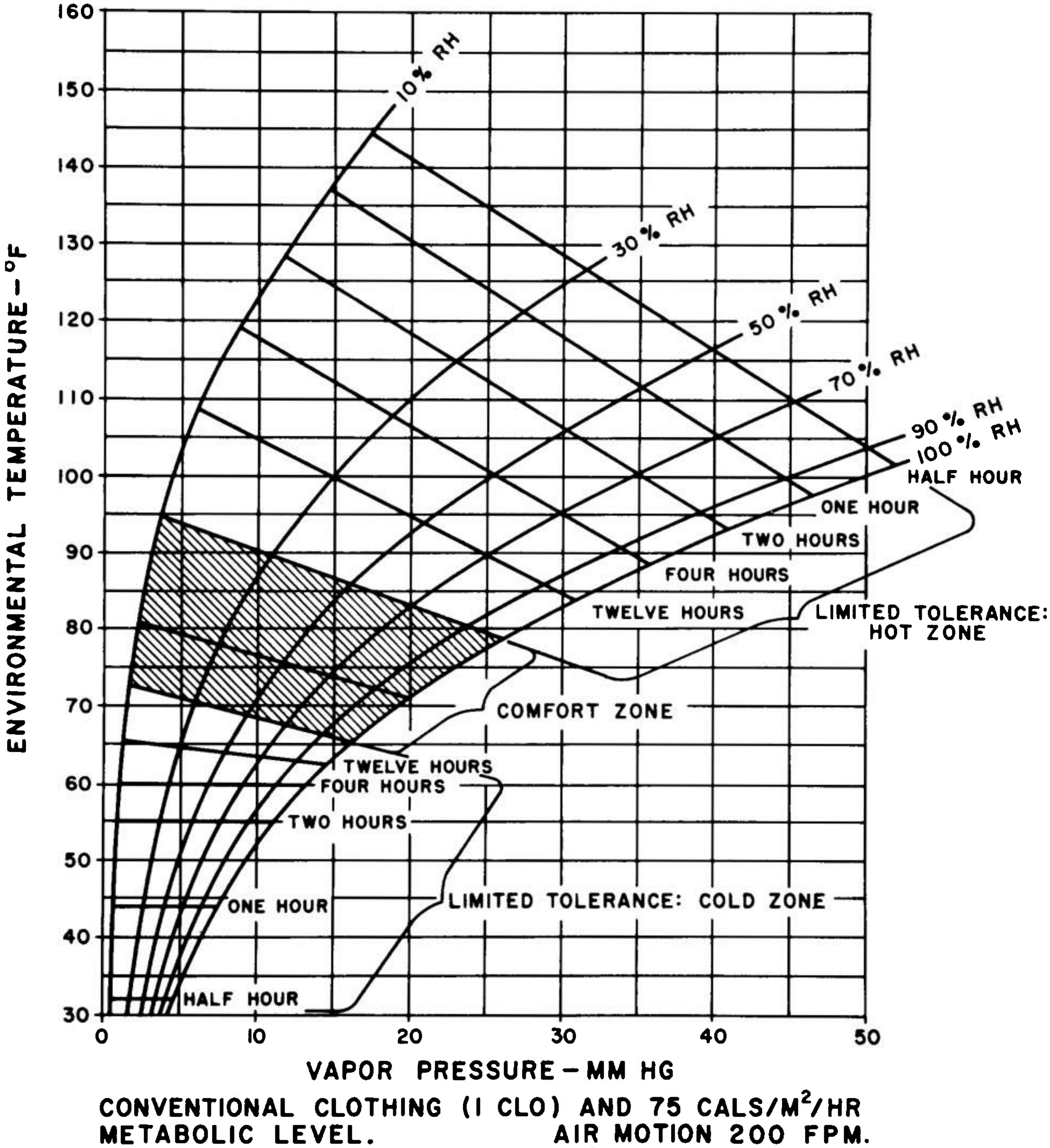

Figure 10-5. *Thermal Requirements for Tolerance and Comfort (Ref. 4)*

ance. The comfort/discomfort zone lies between psychological boundaries, while the tolerance zone lies between physiological boundaries. In addition to the temperature and humidity factors, the degree of air motion influences tolerance and comfort reactions. Figure 10-5 shows the thermal requirements for tolerance and comfort under specific conditions.

The unit "1 CLO" used in Fig. 10-5 is defined as the amount of insulation required to maintain in comfort a sitting-resting subject in an environment ventilated as 200 fpm at a temperature of 70°F and a humidity less than 50% and is approximately equal to a man's everyday clothing, or a heavy topcoat alone (Ref.4).

10-6.3 Noise Considerations

The problem of noise within closed vehicles such as armored personnel carriers became serious with the introduction of new construction methods, new materials, and higher vehicle speeds. It usually arises from the vibration of surfaces in contact with air. Body or hull panels may be a major source of noise in thin-skinned vehicles. Noise within the vehicle interferes with communication; and if the sound levels are high enough and continuous, the efficiency of the personnel is adversely affected. As in temperature and humidity considerations, there are psychological and physiological boundaries to noise tolerance. The reaction to a range of noise varies from physical discomfort to actual physiological damage. Paiu and illness may result from exposure to noise.

The amount of noise or sound that can be tolerated by the average person depends on several factors. The sound pressure level, the frequency, and the duration of the noise are some of the factors which determine the reaction to various sounds. In general, shrill, high-pitched, irregular sounds are judged less pleasant than low-pitched, regular sounds. The study of noise and its control are complex subjects which are well covered in the technical literature.

10-6.4 MAINTENANCE CONSIDERATIONS

Accessibility for maintenance should be considered when bodies or hulls of vehicles are designed. The internal components should be arranged so that they can be readily inspected, serviced, and if applicable, adjusted, without removing the component and with minimum disturbance to other parts. Removable or hinged access panels should be provided where required.

All bodies and hulls should be designed so that corrosion is minimized. All ledges, pockets, and crevices where dirt and moisture can collect should be eliminated. Drain plugs or valves should be provided to permit drainage of moisture from enclosed places.

SECTION III GENERAL FACTORS

10-7 MISCELLANEOUS EQUIPMENT

Tactical military vehicles must operate satisfactorily in a wide range of temperatures (−65° to 125°F). The present trend for arctic operation is to install high-output personnel heaters in the compartments, to replace the tarpaulin with quilted Fiberglas, and to insulate the floors.

Body and hull designers must consider the various kits which may be applied to the vehicle. Some of the units are: (a) deep-water fording kits, (b) winterization kits for personnel and power plants, (c) ground mine protection kits, and (d) armored cabs to replace standard cabs.

The type, size, and weight of the miscellaneous equipment, e.g., the kits described above, to be mounted in or on the body of a vehicle must be considered in the body or hull design process.

10-8 DOORS AND OPENINGS

Doors, hatches, windows, panels, inspection plates and all other openings in the bodies and hulls of military vehicles must be properly located and have adequate locking and sealing arrangements.

The openings should be of sufficient size to provide clearance for servicing and entering the vehicles when the personnel are dressed in arctic clothing. All removable doors and panels should be marked as to location and position to expedite replacement. Quick action fasteners should be provided where frequent removal is required. For doors and hatches, the locking arrangement should be such that the doors will not inadvertently open due to vibration or casual contact. The designer should consider making provisions to keep doors open positively when doors are intentionally opened. This is of utmost importance when heavy armored doors are used since these doors can cause fatal injuries if they accidentally fall.

The sealing of the various openings in a hull is a major development problem. Sealing the personnel doors for amphibious operations has not been satisfactorily accomplished in many vehicles. Personnel doors and other frequently used hatches must not require high forces to seal; and, furthermore, since they are frequently used, they must have a long cyclic life, e.g., they must not take a permanent set which would reduce their efficiency.

The problem of sealing a vehicle against chemical, bacteriological and radiological attack should be also considered. Effective seals for these purposes are currently under development.

Windows and sighting ports should provide maximum visibility with a minimum increase in the overall vulnerability.

Since there is an increasing demand for vehicles to operate on water as well as on land, the need for making the hull waterproof should be considered. Even though a waterproof hull is not an immediate requirement, the designer should, to the extent reasonable, arrange the design to facilitate making it waterproof at a later date when it may become a requirement. For example, suspension members and drive train members may extend through the hull and thus require adequate seals.

10-9 CREW AND EQUIPMENT

A military vehicle must be highly efficient in every respect including the utilization of space. Not only must the various automotive components, the armament, and the equipment carried compose a compact package, but the overall design must permit the crew members or operators to operate the equipment under the most severely anticipated conditions. Major considerations include floor layout, headroom, passageways, ilumination, interior and exterior doors, and protrusions within the vehicle.

In general, crew and passenger compartments should be designed so that the maximum degree of safety is present without sacrificing efficient military performance. For example, fire-resistant materials should be used for soundproofing and upholstering military vehicle interiors. Projections within a vehicle should be eliminated whenever possible or if they are necessary, they should be padded. Equipment should be securely mounted so that impacts on the vehicle will not cause the interior item to become a projectile.

Layout drawings, three-dimensional fractional-scaled models, and full-size mock-ups are frequently used as aids in optimizing space utilization in military vehicles.

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CHAPTER 11

THE SUSPENSION SYSTEM*

SECTION I GENERAL DISCUSSION

11-1 DEFINITION

11-1.1 GENERAL

The suspension system of a vehicle is that complex of mechanical, structural, pneumatic and hydraulic members which provides flexible support between the ground and frame, or ground and hull of the vehicle. Although some of its members must transmit power, it is not considered part of the power train. Dissimilar methods of locomotion and types of ground contact, required to meet the wide range of military requirements, result in major design variations of suspension components. The general definition, therefore, must be expanded to cover each type of vehicle.

11-1.2 WHEELED VEHICLES

The main components of the suspension system for wheeled vehicles are: (1) springs, (2) shock absorbers, (3) bogies, (4) axles, (5) wheels, and (6) tires (Ref. 1). All of the above components may not be incorporated in a given vehicle, and the properties of some may be modified or expanded to provide the function normally provided by some other component.

11-1.3 TRACKED VEHICLES

The major components constituting the suspension system for track-laying vehicles are: (1) springs, (2) shock absorbers, (3) road wheel arms with spindles, (4) road wheels, (5) idler wheels, (6) tensioning devices, and (7) tracks (Ref. 1).

11-1.4 SLEDS

Sleds are usually considered as not having a suspension system. In the strictest sense, however, the runners and their supports constitute the suspension. Some sleds also incorporate springs between the frame or body and the runners.

11-1.5 WALKING, RUNNING, JUMPING, AND LEAPING VEHICLES

Vehicles in this category are rather uncommon, and therefore, the classification of components that make up their suspension systems has not been rigidly established. It may be assumed, by applying the general definition of a suspension system, that their suspension systems consist of mechanisms, hydraulic devices, pneumatic devices, and ground-contacting elements.

11-1.6 SPRUNG MASS AND UNSPRUNG MASS

That portion of the vehicle which is supported by the main flexible elements (springs) of the suspension system is referred to as the *sprung mass*. Those parts not supported by the springs constitute the *unsprung mass*. Generally, the unsprung mass is comprised of the suspension components plus a portion of the power train. A proportionate part of the springs, swinging linkages, and other components attached to the frame is included in the sprung mass.

Considering the mobility of the military vehicle, it is advantageous to have a small unsprung mass. The lighter unsprung mass maintains more uniform contact with the terrain, thereby resulting in superior steering and traction characteristics. Impacts, resulting from traveling over irregular terrain, induce lower stresses in the suspension components, particularly the wheels and tires, when the unsprung mass is kept small. Undesirable effects on the ground that supports the vehicle are decreased with a lighter unsprung mass; and, within certain limits, the ride qualities of the vehicle are improved (Refs. 2, 3).

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11-2 PURPOSE

The primary purpose of a suspension system for the military vehicle is to improve mobility. It accomplishes this by isolating the vehicle and personnel from uncomfortable shocks and vibrations while maintaining terrain contact for efficient traction, all of which results in increased speed and maneuverability (Ref. 4).

Corollary purposes and functions of the suspension are (Refs. 1, 5, 6):

1. To support the vehicle body or hull, keeping it off the ground,
2. To distribute the vehicle weight equally over the ground contact area,
3. To provide the propulsive contact with the ground,
4. To transmit the propelling force between the ground and the frame or hull,
5. To transmit the driving and braking torque and to accommodate the torque reactions resulting therefrom,
6. To continuously make adjustments between the moving vehicle and the irregularities of the stationary terrain, thereby providing obstacle-climbing and ditch-crossing capabilities,
7. To prevent undue lateral and vertical deflections of the vehicle from its selected course, and thereby providing a stable gun platform,
8. To provide means for changing course at will,
9. To provide comfort and safety for the crew, thereby extending their endurance, and
10. To improve the reliability of other vehicle components by reducing shock and vibration.

11-3 REQUIREMENTS

Military suspension systems must be very rugged to endure the high impacts experienced when traveling rough terrain. They are normally designed to withstand a maximum impact loading of 8 G or more. The 8 G or more impact must be absorbed by the suspension so that a maximum of 3 G is transmitted to the sprung mass (Ref. 1).

The high degree of refinement in ride quality found in commercial vehicles is not required for military vehicles, but noise and high-frequency vibrations should be subdued to minimize the fatiguing of the operating personnel. The comfort and ability of the crew to function are limited by the vibration frequency, and amplitude of displacement of the body or hull. It is generally agreed that the bounce frequency should be maintained between 60 and 120 cpm. Below 60 cpm, the crew may experience motion sickness, and above 120 cpm, fatigue (Refs. 7, 8, 9).

The military suspension systems must provide for sufficient amplitude of movement of its ground-contacting members to allow for extreme surface irregularities. Common practice is to design suspension systems so that any wheel of a multi-wheeled vehicle will be capable of moving to any position from 12 in. above to 6 in. below its normal level standing position without increasing or decreasing the load it supports by more than 25% (Ref. 1). Furthermore, the military suspension must adjust to terrain conditions to provide the maximum floatation possible in difficult soil conditions. This requirement not only dictates uniform distribution of the vehicle weight over all the points of terrain contact, but also requires that the wheel or track oscillations be controlled to minimize the possibility of locally exceeding the strength of the soil (Refs. 3, 5). Improper design for these conditions may not only limit the vehicle speed, but may also result in bogging of the vehicle.

When operating under certain conditions, the military suspension may experience repeated high loadings continually. Materials for the suspension components, particularly the springs, must be carefully selected for their resistance to fatigue failure. The design, heat treatment and surface treatment must preclude the introduction of fatigue stresses. Consideration must be given to the endurance limit of the material in relation to the design loading and stress repetition expected during the service life of the vehicle (Ref. 1).

Travel over rough terrain requires that substantial quantities of energy be absorbed and dissipated by the shock absorbers. Due consideration must be made to prevent the overheating of these components to ensure their continued reliable functioning (Refs. 1, 10).

SECTION II SUSPENSIONS FOR WHEELED VEHICLES

11-4 GENERAL

Suspension systems for wheeled vehicles are classified into two categories: solid axle (conventional) and independent axle, according to the interrelationship of the wheels to each other. Each category is represented in current automotive vehicles by a great variety of design configurations that provide the functional objectives outlined in par. 11-2. Some of the designs have been highly refined to emphasize special characteristics, particularly in the sports car. The discussion in this text is oriented toward those designs currently employed on military vehicles, with reference to particular advanced designs that may have some potential applications on the military vehicle. A complete treatment of the numerous suspension designs and their special characteristics can be found in Refs. 2 and 7.

Although the suspension system is the impact-absorbing mechanism of the vehicle, its design is influenced to a great extent by the vehicle's power transmission and steering requirements.

11-5 SOLID AXLE SUSPENSION

11-5.1 GENERAL

The axle of the suspension is a cross support on which the wheels turn. Axles which provide only support for the vehicle weight are termed *dead axles* while those which also incorporate means for driving the wheel are called *live axles* (see Chapter 8, Section VII). The term live axle applies to the entire axle assembly, consisting of the housing which contains the drive gears, differential and power transmitting shafts. Until recently, wheeled military vehicles have employed the solid axle suspension, both front and rear, almost exclusively, because of the basic ruggedness, low manufacturing cost, and their wide use on heavy commercial vehicles.

11-5.2 DEAD FRONT AXLE

The dead front axle supports the vehicle weight and resists the torsional stresses that occur during braking. In order to resist these stresses, as well as those resulting from impacts experienced during traveling, and still maintain reasonable wheel alignment and directional stability, the dead front axle is usually attached to the frame by means of rather stiff semi-elliptic leaf springs (Ref. 7). These springs produce a harsh ride and limit the vehicle speed over irregular terrain. Softer springs may be employed, in which case the axle guidance and torque resistance must be taken by separate linkages or other mechanisms.

To permit steering, the dead front axles are equipped with pivoting wheel spindles. The axles are usually I-sections of drop-forged alloy steel. The unsprung mass may be reduced, and the torque-resistance properties improved, by using more expensive tubular axles of molybdenum steel (Ref. 5).

11-5.3 LIVE AXLE SUSPENSIONS

11-5.3.1 Hotchkiss Drive (Ref. 11)

The Hotchkiss drive is the conventional front and rear live axle suspension that was formerly employed in American military vehicles, to the exclusion of other types. In this drive, a propeller shaft (drive shaft) is employed with two universal joints and a slip joint, as shown in Fig. 11-1. The torque reaction, drive thrust, and alignment of the axle housings are resisted by the suspension springs.

The suspension springs are pivoted on brackets at their forward ends and shackled to the frame at their rear ends. The rear spring brackets are the point of application of the driving thrust to the frame. Because the suspension springs must resist the torque reaction of the drive, stiffer springs are required than are used with some of the other live axle suspensions. In addition, because the springs must transmit the driving thrust, they must be fairly flat. Because of these torque and force transmission requirements, the Hotchkiss drive has been criticized as providing inferior riding qualities, thereby limiting vehicle mobility. However, this drive system is in extensive use because of simplicity, low cost, and ruggedness.

An advantage of the Hotchkiss drive is that the flexible connection between axle and frame throws less strain on the driving mechanism than do other types. When sudden loads are applied, as in suddenly engaging the clutch, the axle housing

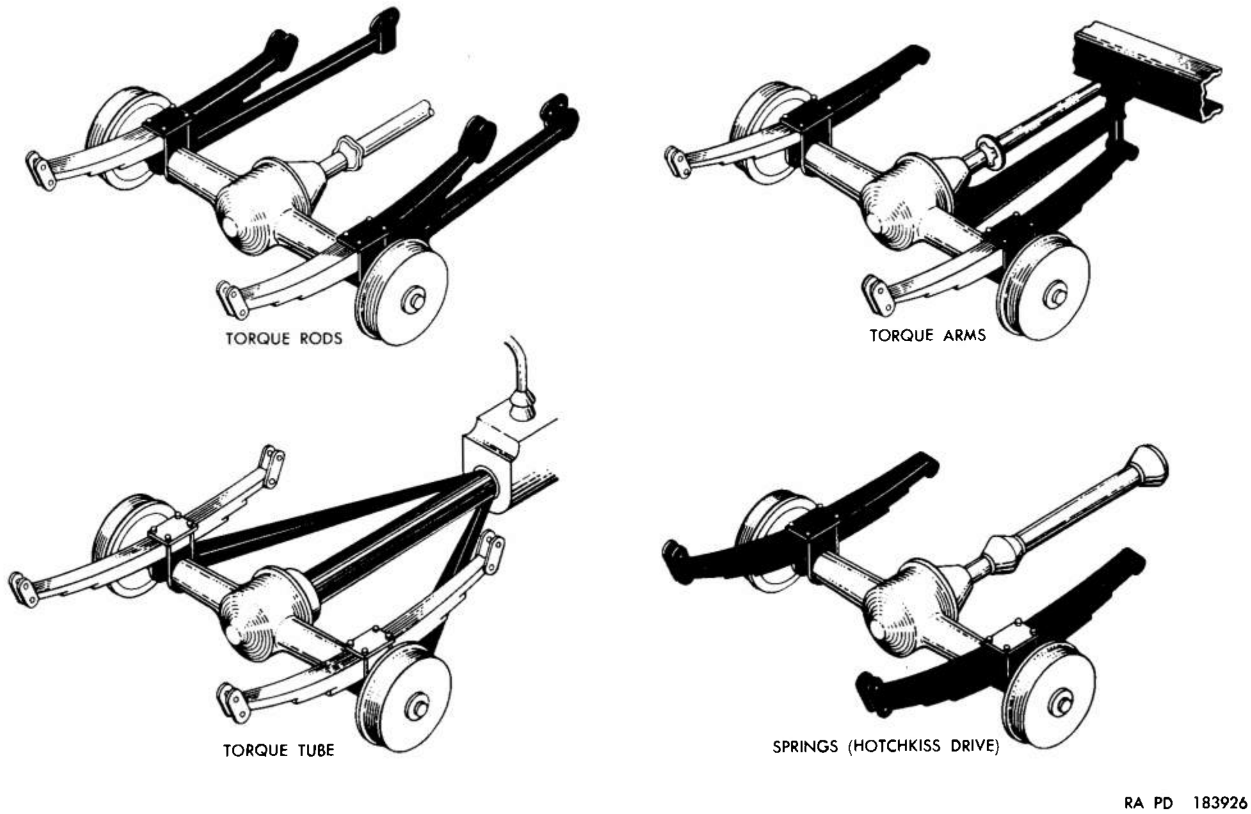

Figure 11-1. ***Types of Live Rear Axle Suspensions***

can rock about the drive shaft slightly, which cushions the shock transmitted through the driving mechanism and reduces the load between the teeth of the final driving mechanism.

11-5.3.2 Torque Tube Drive (Ref. 11)

The torque tube drive, while not common on heavy military vehicles, is used on a number of passenger and light commercial vehicles. In this type of drive, the propeller shaft (drive shaft) is housed in a steel tube, the torque tube (Fig. 11-1). The rear end of the torque tube is bolted rigidly to the rear axle housing by means of a flange, while its front end is connected to the transmission or a frame cross member by means of a ball-and-socket joint. One universal joint is used in the propeller shaft and is located at the ball-and-socket joint of the torque tube. A slip joint is placed in the propeller shaft to take up end play arising when the driven axle moves up and down. A center bearing is generally used to support the drive shaft in the torque tube.

Two suspension system radius rods are utilized to connect the outboard ends of the axle housing with the transmission end of the torque tube to maintain the axle housing aligned at right angles to the torque tube. The suspension springs are shackled at both ends.

In a torque tube drive, both the torque reaction and the driving thrust are resisted by the torque tube. Since the suspension springs do not resist the torque reaction and drive thrust, they can be made more flexible and thus, impart better riding qualities than, for example, a Hotchkiss drive.

In a torque tube drive, the driving thrust is applied to the frame at the engine mounting or at a frame cross member. In a torque rod or Hotchkiss drive, the force is applied at the suspension springs. Both the torque tube and the Hotchkiss drives are used in contemporary designs.

11-5.3.3 Torque Arm Drive (Ref. 11)

The torque arm drive is seldom employed. It consists of a solid or tubular arm, rigidly connected to the driving axle housing at its rear end and to a frame cross member, through a ball-and-socket joint or spring bracket, at its front end (see Fig.

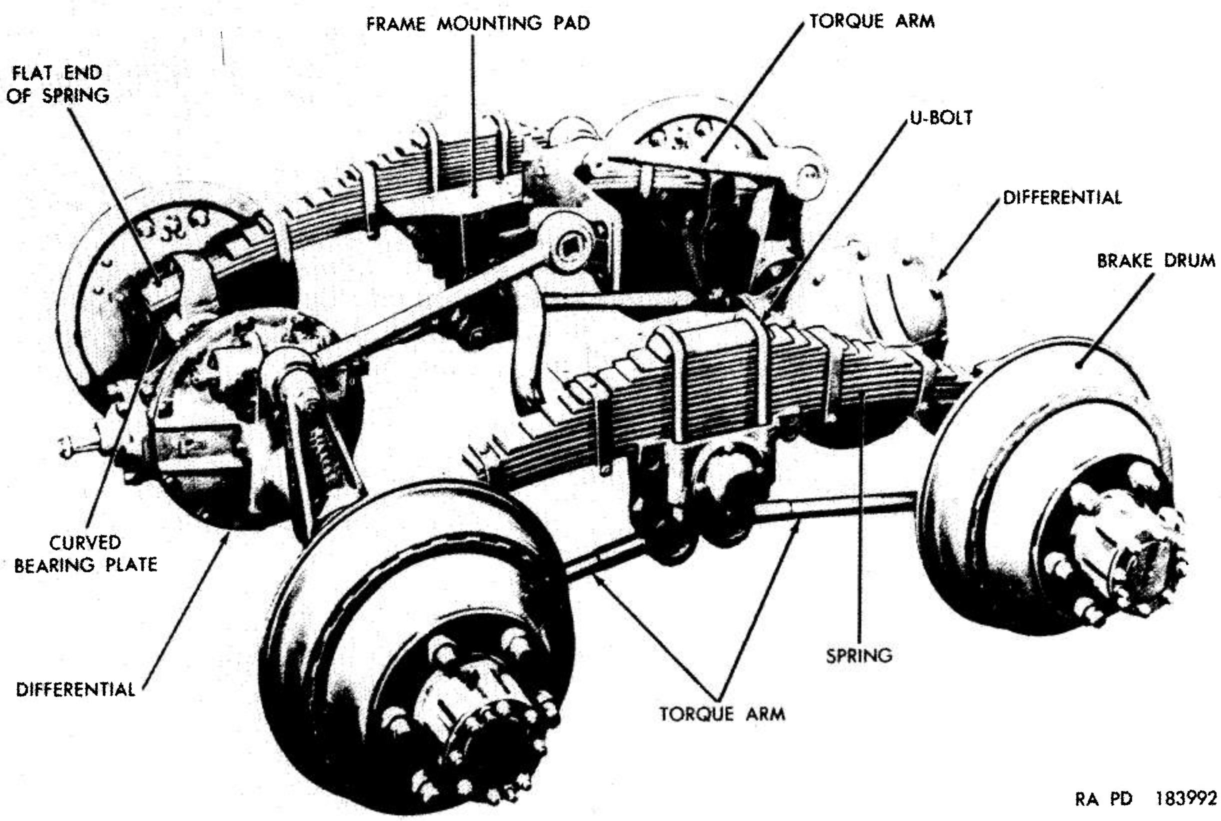

Figure 11-2. ***Bogie Suspension for Wheeled Vehicle***

11-1). An open propeller shaft is employed. The torque arm drive is similar to the torque tube drive, the principal difference being that it employs an open propeller shaft running parallel to the torque arm instead of a drive shaft housed within a torque tube.

11-5.3.4 Radius Rod Drive (Ref. 11)

In the radius rod drive (Fig. 11-1), two radius rods or torque rods are used to transmit the driving thrust to the frame and to maintain the alignment of the driving axle. The radius rods are connected to both the axle housing and to the frame by jointed connections which permit full vertical, and sometimes lateral, motion of the axle housing relative to the frame. The torque reaction is resisted by the suspension springs, as in the Hotchkiss drive. An open propeller shaft with two universal joints is usually employed with the radius rod drive. This type of drive is used to a very limited extent.

11-5.3.5 Bogie Suspension (Ref. 5)

Multiwheel suspensions are employed on heavy military vehicles to maintain the wheel and tire loadings within safe limits. The most common arrangement to accommodate the additional wheels is the bogie suspension. Bogies are generally used only at the rear; although, some unusual vehicles, such as the Teracruzer, have also employed them at the front (Ref. 12). In military vehicles, bogie suspensions are live axle systems, except, of course, when used on trailers.

The automotive bogie is a suspension assembly that enables tandem axles to function together as load-carrying and driving axles. The usual arrangement consists of tandem axles longitudinally interconnected by a pair of walking beams. These are joined, usually at their midpoints, by a single cross support (trunnion axle) which serves as the pivot point for the entire unit. Most frequently, leaf springs are employed as the primary flexible member, because they can also serve as the walking beam. A typical military bogie is shown in Fig. 11-2.

In the unit shown, suspension is by means of leaf springs which are fastened at their midpoints to a spring seat, which, in turn, is secured to the vehicle frame. The outer ends of the springs rest on hardened steel bearing plates on the tandem-axle housings. Both spring seats are mounted on spindles at the ends of the trunnion axle. Tapered roller bearings are incorporated into the spindles to allow them to rotate freely despite side thrusts. Torque arms are employed to prevent the driving and braking torques from producing a spring wind-up which would impose unequal axle loadings. These arms also maintain wheel alignment, since spring ends can float relative to the axle bearing plate. Because the springs (walking beams) are pivoted, they can distribute half of the load to each axle. As a result, the load is equally distributed over four wheels, allowing heavy vehicle loading without exceeding the safe tire loading of any one wheel.

The bogie suspension has good obstacle-climbing and ditch-crossing capabilities because the pivoting-walking beam adjusts to terrain irregularities in such a manner that uniform ground pressure and full traction are maintained within design limits. The effect is somewhat similar to that of employing a single, larger diameter wheel.

Bogies can be designed to allow various amounts of movement between suspension components and the vehicle superstructure. Common practice has established that bogie design should permit any one axle to rotate about the longitudinal axis of the vehicle to an amount of 30° without interference or damage to any part. Any wheel should be capable of moving to any position from 12 in. above to 6 in. below its normal level standing position (Ref. 1). Military characteristics for tactical vehicles may specify more severe requirements of this type. The more difficult these specifications the more places will the vehicle be able to go, but its cost will rise accordingly, also.

Since a substantial part of the vertical clearance between the wheel and frame is allowed for the walking beam displacement, the springs must be rather stiff, lest the axles prematurely bottom on the frame. The effect of the stiff springs is moderated somewhat by the pivoting feature. At low and moderate speeds, when one wheel of a bogie suspension is deflected vertically by an obstacle in the vehicle path, the spring pivots deflecting both ends, thereby reducing the shock transmitted to the sprung mass. At high speeds, this effect is not always fully realized, and the transmitted shock may be greater than that of an independently suspended wheel.

Consideration should be given in bogie design to minimize the resistance to steering. The tandem axle centers should be as close as tire diameters, plus reasonable clearances, permit. Some bogie suspensions provide a small amount of free play of the axles so that they may adjust during cornering to reduce the steering resistance. In bogies, where the semi-elliptic leaf springs are attached to the axles, a natural steering effect is produced by the spring deflections caused by the centrifugal force during cornering (Ref. 13).

11-6 INDEPENDENT SUSPENSIONS

11-6.1 GENERAL

The term *independent suspension* is applied to a method of vehicle suspension in which each wheel supports its share of the vehicle load without the intermediary axle (Ref. 14). Each wheel is free to oscillate independently of the other wheels. Either the front or rear wheels, or both may be independently suspended. They may be driven or free-wheeling. No one type of spring is peculiar to independent suspensions. Leaf, coil, torsion bar, torsi-elastic, and pneumatic springs have been used. A variety of mechanisms have been devised in implementing the independent suspension. It is beyond the scope of this handbook to describe all of them. Those used on military vehicles, and covered herein, are typical. The reader may consult Refs. 2 and 14 for a more thorough coverage of the subject.

11-6.2 FREE-WHEELING INDEPENDENT SUSPENSIONS

A typical independent front suspension for a light vehicle is shown in Fig. 11-3. In this design, each wheel is held in alignment by a pair of stiff control arms, commonly called parallel wishbones. The vehicle weight is transferred from the frame to the rigidly attached cross member, through the coil spring, to the lower wishbone. It may be noted that the control arms are of unequal lengths. The vehicle designer may achieve certain desirable suspension and cornering characteristics by carefully selecting the control arm lengths (Refs. 2, 14). The characteristics of this suspension, as compared to the solid front axle, are discussed in par. 11-6.4.

11-6.3 DRIVEN INDEPENDENT SUSPENSIONS

A driven independent suspension of the parallel wishbone is shown in Fig. 11-4. The engine

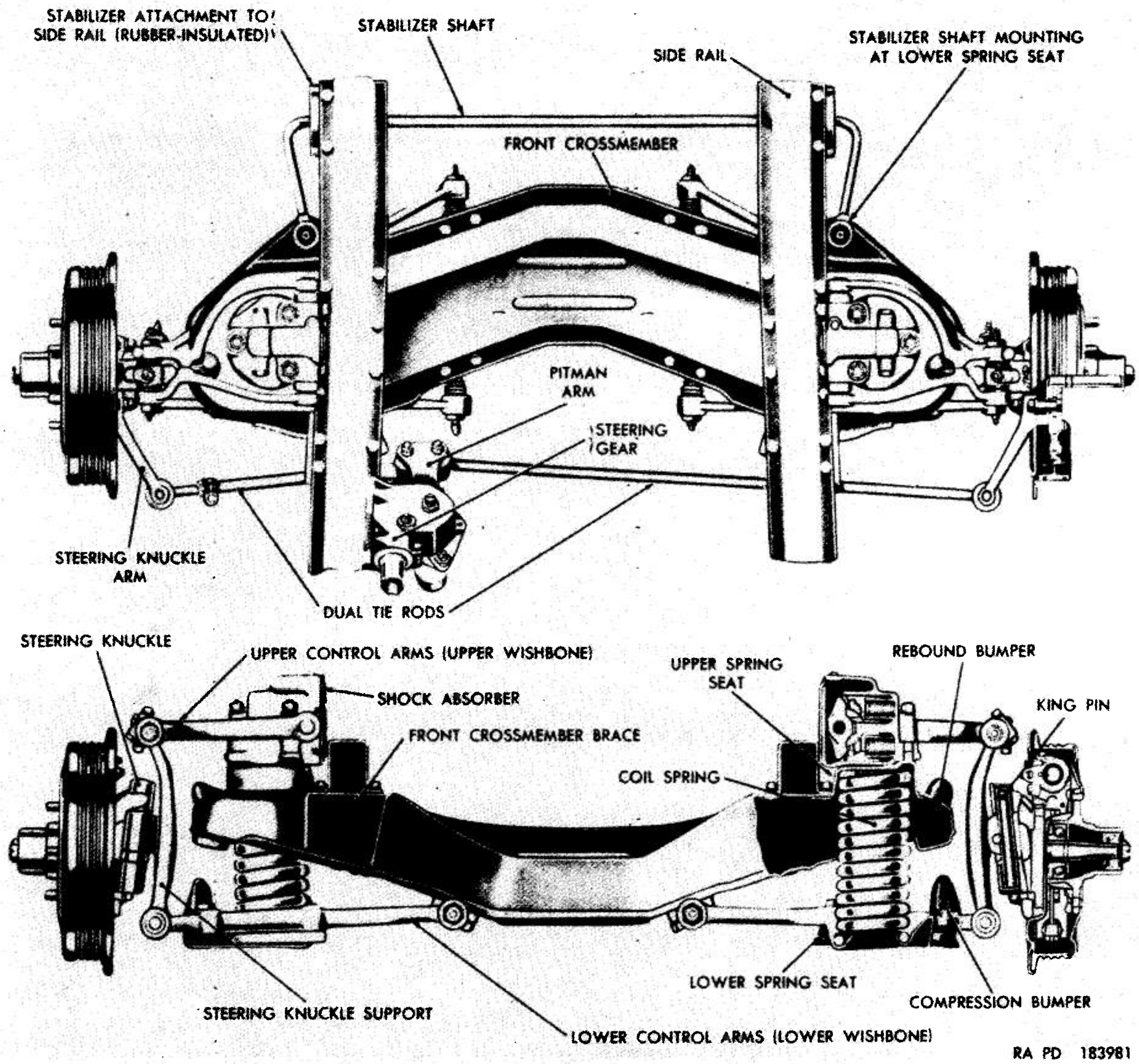

Figure 11-3. Front Axle Coil Spring Suspension

power is transmitted from the gear case, which is rigidly attached to the vehicle frame, through half shafts, to each wheel. The geometry of the guiding linkage dictates the use of two universal joints. Since the gear box is rigidly fastened to the frame, the driving torque reactions are taken through the frame and do not affect the suspension spring as they do with the Hotchkiss drive (see par. 11-5.3.1).

Figure 11-5 shows a parallel wishbone suspension for a heavy multiwheeled vehicle. This design is similar to that of Fig. 11-4 except that torsion bar springs are used in place of coil springs.

An independent rear suspension employing coil springs is shown in Fig. 11-6. In this design, the alignment of each wheel is maintained by a swinging-arm type of linkage. The arm consists of a rigid wishbone whose pivot axis is not necessarily parallel to the longitudinal axis of the vehicle. The wheel spindle is rigidly attached to the arm, which results in a tilting of the plane of the wheel, both longitudinally and vertically, when the suspension is displaced. The location of the swinging-arm pivot axis causes the rear wheels to produce a steering effort which improves the cornering characteristics of the vehicle. Two universal joints are necessary in the power shafts of this design, because the swinging-arm pivot axis is skewed with respect to the power shaft. The pivot axis does, however,

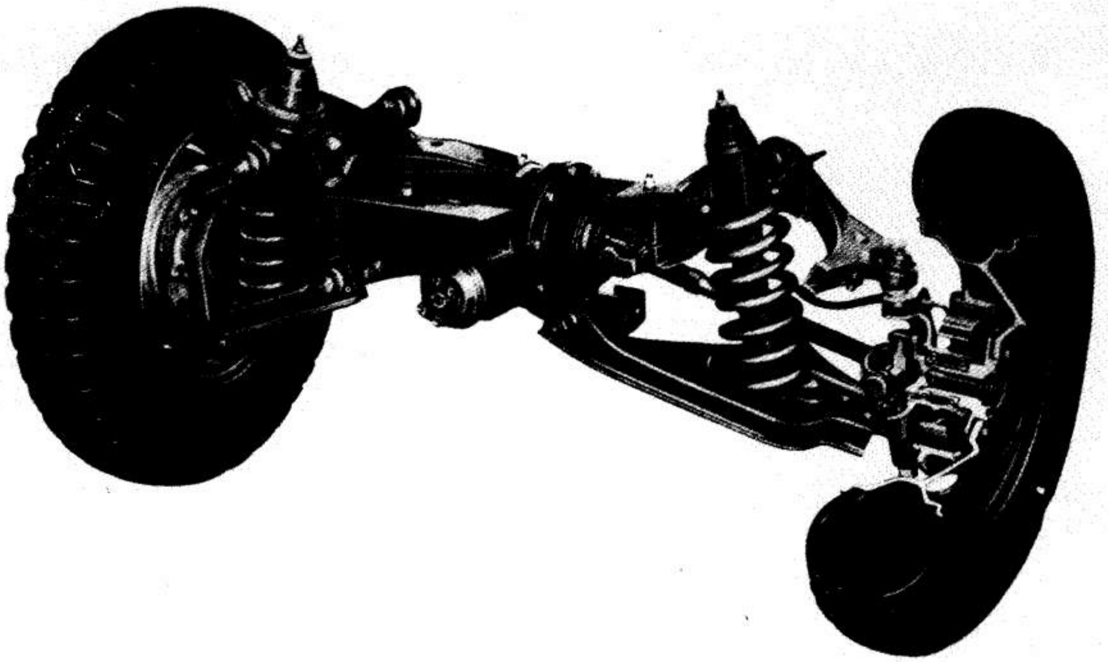

Figure 11-4. *Driven Parallel Wishbone Coil Spring Front Suspension*

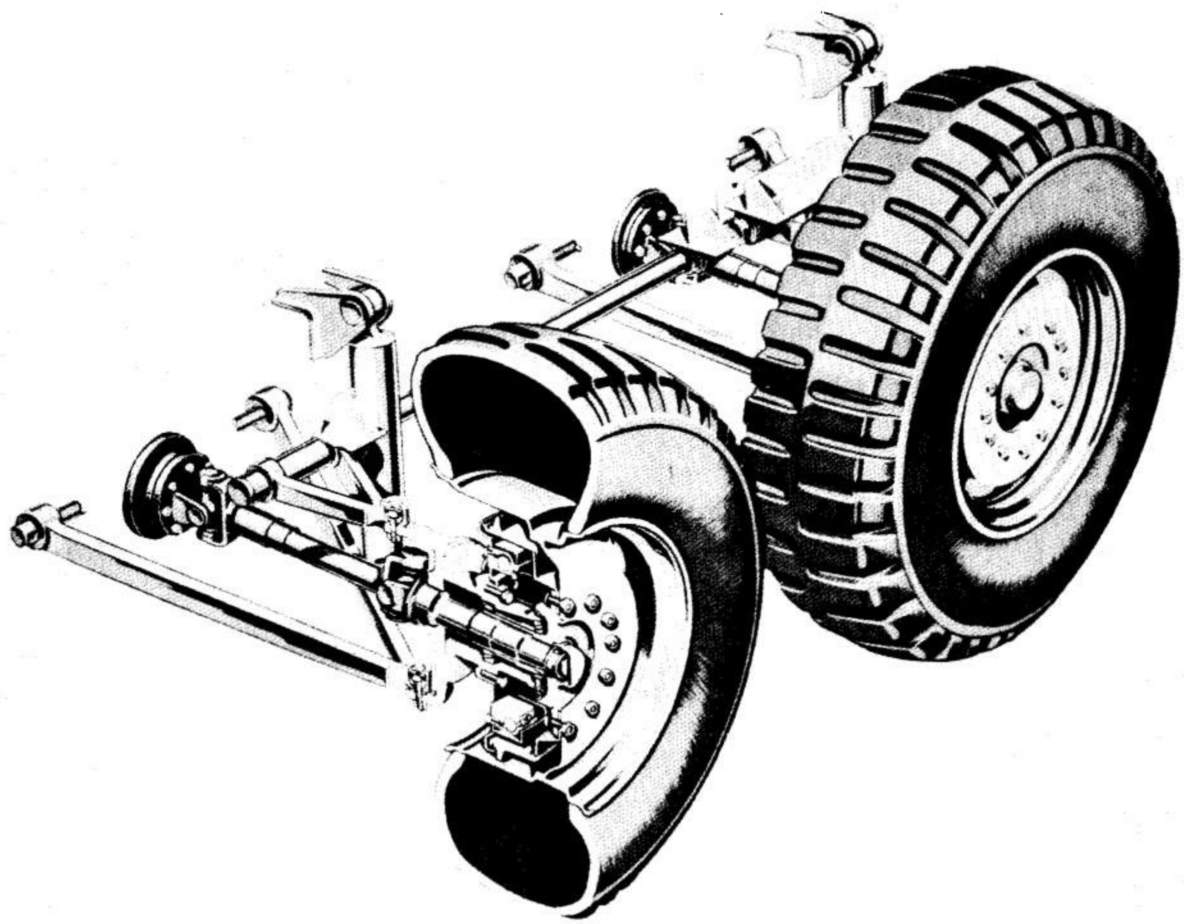

Figure 11-5. *Driven Parallel Wishbone Torsion Bar Suspension*

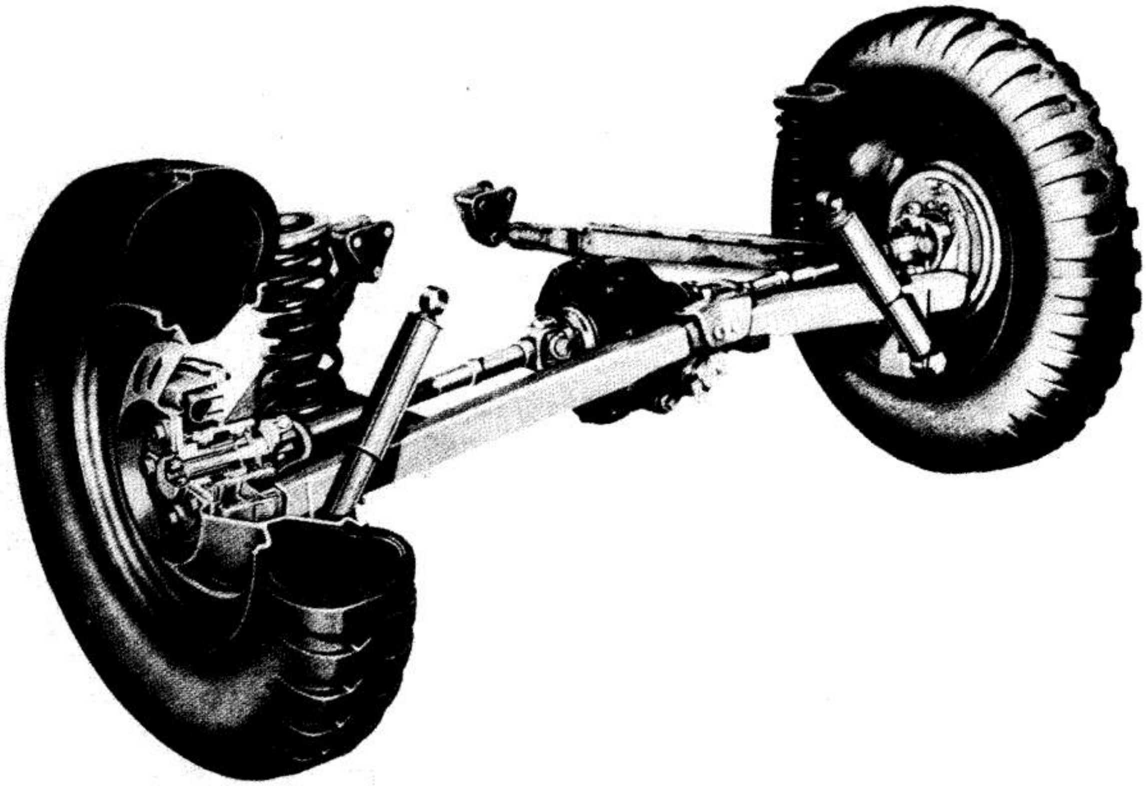

Figure 11-6. *Swinging Arm Independent Rear Suspension*

pass through the inboard universal joint to minimize the relative sliding motion of the splined coupling (Ref. 2).

The independent suspension types just discussed are representative of the types used on military vehicles to date. Many other designs have been developed in refining vehicle performance. The vehicle designer should become familiar with these designs and glean the features that may have a place in military suspensions. The future development of individual hydraulic or electric drives at each wheel will enable, and require, variations of these suspensions and the introduction of new types.

11-6.4 INDEPENDENT SUSPENSION CHARACTERISTICS

11-6.4.1 General

The development of the independent suspensions was motivated primarily to reduce the unsprung mass, thereby improving the handling and traction characteristics of high-speed road cars. However, the demand for greater speed and mobility for military vehicles warrants the use of independent suspensions on such vehicles. In addition to the improved performance associated with reduced unsprung mass, the use of independent suspensions increases the speed and mobility of the military vehicle by:

1. Reducing front end vibration (wheel shimmy and axle tramp),
2. Permitting the use of softer suspension springs,
3. Providing more ground clearance,
4. Permitting more optimum wheel spacing.

The means by which these advantages are affected are discussed in the following paragraphs.

11-6.4.2 Reduction of Unsprung Mass (Refs. 2, 5, 14)

When one of the wheels mounted on opposite ends of a rigid axle experiences an impact, the entire suspension unit is affected. The inertia of the large unsprung mass resists movement resulting in high forces imposed on the wheels, tires, bearings, etc. The ride quality is adversely affected by the subsequent wheel and axle disturbances, thereby limiting vehicle speed. Equally high forces are experienced by the terrain, which under marginal conditions, may fail and impair vehicle mobility.

The wheel suspension using pneumatic tires is a two-degree-of-freedom spring system consisting the sprung mass, the primary suspension spring, the unsprung mass, and the springiness of the tire. When the tire maintains contact with the ground, the two masses have separate and distinct natural frequencies of oscillation. When the tire leaves the ground, the unsprung mass responds to the action of the suspension spring at yet another natural frequency. It is this last frequency that determines the maximum speed at which the vehicle can negotiate terrain irregularities and still maintain tire contact. It can be demonstrated that smaller unsprung masses will maintain ground contact over irregular terrain at higher speeds than will larger masses by the following simple relationships:

From fundamental laws of vibration, disregarding damping

$$T_m = 2\pi\sqrt{\frac{m}{k}} \qquad \text{(11-1)}$$

$$T_M = 2\pi\sqrt{\frac{M}{k}} \qquad \text{(11-1a)}$$

where

- T_m is the period of free vibration of unsprung mass
- T_M is the period of free vibration of sprung mass
- m is the unsprung mass
- M is the sprung mass
- k is the spring constant

Then

$$\frac{T_m}{T_M} = \sqrt{\frac{m}{M}} \qquad \text{(11-2)}$$

$$T_m = T_M\sqrt{\frac{m}{M}} \qquad \text{(11-2a)}$$

It is seen that reducing the unsprung mass, m, in Eq. 11-2a results in a shorter free vibration period, T_m. Obviously, then, the shorter period permits the unsprung mass to follow a particular ground wave or terrain irregularity at a higher vehicle speed. Constant ground contact of the wheel is essential to provide directional stability, steering and traction. The ratio of the unsprung to the sprung mass, m/M, plus the stiffness of the spring, are the most influential factors in determining the suspension characteristics. The m/M ratio for solid axle suspensions is about one-quarter, and may be as low as one-eighth for independent suspensions.

11-6.4.3 Wheel Shimmy (Refs. 2, 5, 14, 15)

When one wheel of a solid axle suspension passes over an obstacle, the axle executes an angular movement in the vertical plane, and both wheels are simultaneously angularly displaced by the same amount. Since the rotating wheels act as gyroscopes, a forced gyroscopic precession occurs, particularly on the steerable wheels, tending to make them swing about their kingpins. When the axle again returns to the horizontal following a displacement, the gyroscopic forces are reversed, thereby, causing the wheels to oscillate, or shimmy about the kingpins. When the angular deflection of the axle occurs at a frequency near the natural frequency of torsional vibration of the vehicle body, the wheel shimmy is self-sustaining and can be removed only by a drastic reduction in speed. Another factor that tends to perpetuate the oscillation of the wheels is the self-aligning force produced by the kingpins caster angle.

Wheel shimmy resulting from terrain irregularity can be virtually eliminated by the use of independent front suspension. When one wheel of an independent suspension passes over an obstacle, both wheels will most likely be deflected, but not simultaneously nor in the same amount. With certain types of independent suspension, the plane of the wheel during vertical movement remains substantially parallel, thereby eliminating the gyroscopic effects which produce wheel shimmy.

11-6.4.4 Axle Tramp (Refs. 2, 15)

Axle tramp is a vibration of the axle in a vertical plane. It occurs in solid axle suspensions when negotiating irregular terrain and under certain conditions of braking. The mechanics of axle tramp, related to irregular terrain and wheel shimmy, are discussed in par. 11-6.4.3 above. It occurs during braking when the braking effect of the two front wheels is unequal and is a result of employing leaf springs to maintain wheel and axle alignment. When braking hard, with the typical solid axle suspension, the leaf springs wind up as a result of transmitting the braking reaction torque. As long as the braking effort of both wheels is equal, the windup of the springs is equal, and the wheel alignment is unaffected. If, however, one wheel loses its braking adhesion, the spring on this side unwinds and the axle rotates about a vertical axis so that it is no longer at right angles to the longitudinal centerline of the vehicle. A gyroscopic precession is produced which lifts the opposite wheel, thereby increasing the load on the first wheel. The first wheel then recovers braking effort and the second loses braking effort, reversing the force conditions. The cycle continues and is self-sustaining until the brakes are released. This axle tramp could be eliminated in solid axle suspensions by providing separate rigid guiding linkages that would relieve the springs of braking torques. Axle tramp cannot occur in independent suspensions because the wheels are not rigidly interconnected.

11-6.4.5 Spring Stiffness (Refs. 2, 7, 14, 15)

In the solid axle suspension, leaf springs are usually employed as the guidance members to maintain the wheel alignment against impacts and braking forces. For this reason, the springs are relatively stiff. The independent suspension usually provides for wheel alignment by means of separate rigid members (parallel wishbones, trailing arms, etc.). This permits the use of softer springs, resulting in lower shock transmission to the frame, greater wheel deflection to accommodate terrain irregularities, and a reduction of pitching.

The acceleration forces experienced by the occupants and the components mounted on the sprung mass, when a vehicle negotiates a terrain irregularity, are a function of the vehicle velocity, height and shape of the irregularity, and ratio of the spring stiffness to the sprung mass. A simplified equation for determining the acceleration is:

$$\ddot{x} = \frac{(\cos p_t - \cos n_t) h p^2 n^2}{2(n^2 - p^2)} \text{(Ref. 9)} \qquad (11\text{-}3)$$

where

- $\ddot{x}$ is the vertical acceleration of sprung mass
- p $= 2\pi\, v/l$
- v is the vehicle velocity
- h is the height of irregularity
- l is the length of irregularity
- t is the time from initial contact of irregularity
- n $= k/M$
- k is the spring constant
- M is the sprung mass

The springiness of the tire and the damping of the suspension spring are neglected in the above equation and a sinusoidal waveform of the terrain irregularity is assumed. An exact solution which includes those factors involves rather complex-simultaneous equations. Computer techniques are advised for the thorough analysis of the suspension so that many variables may be considered in selecting the spring for the optimum suspension characteristics.

Although it is not obvious from Eq. 11-3, the reduction of the spring constant (softening of the spring) results in a lower vertical acceleration of the sprung mass. This means that irregular terrains can be traveled at a higher speed before encountering intolerable accelerations. The designer is cautioned that softening the spring increases the period of free vibration of the system according to Eq. 11-1a and that the amplitude of vibration after passing over a hump may reach intolerable values. The frequency should be maintained between 60 and 130 cpm, and adequate damping during rebound should be provided to preclude the buildup of extreme vibratory excursions.

The ability of the tire to maintain ground contact, when negotiating a terrain depression or ripple, is governed by the amount of spring deflection produced by the static weight supported by the suspension spring. An approximate maximum depth, or height or ripple, for limiting road contact is given by

$$x = d\left(1 + \frac{m}{M}\right) \text{(Ref. 2)} \qquad (11\text{-}4)$$

where

- d is the initial static spring deflection
- m is the unsprung mass, and
- M is the sprung mass

It can be seen from Eq. 11-4 that a reduction of the unsprung mass results in a lower value of x. This in itself is a disadvantage of the independent suspension, since a lower unsprung mass is one of its inherent characteristics. However, the permissible use of softer springs offsets this disadvantage and the independent suspension is regarded as having superior ground-hugging capabilities.

Pitching is defined as the angular oscillation of the sprung mass of a vehicle about an axis within the wheelbase, parallel to the transverse axis (Ref. 7). It is induced by bumps and depressions in the roadway and is an undesirable motion that limits vehicle speed and mobility. To minimize pitching, the front and rear suspensions should have nearly equal frequencies, with the front springs slightly softer to compensate for the front wheels hitting a bump slightly ahead of the rear wheels. Because the solid axle suspension employs stiff front springs to maintain wheel alignment, the rear springs are softer, making the situation conducive to pitching. The independent suspension permits the use of softer springs all around, thereby reducing pitching.

It must be noted that softer springs may be employed with the solid axle suspension if separate rigid members are provided to maintain wheel alignment. However, this would increase the system complexity and manufacturing cost to that of the independent suspension without attaining all of the advantages of the latter system.

11-6.4.6 Improved Ground Clearance

The advantages of the independent suspension discussed above result primarily in greater vehicle speed. Suspension and drive component configurations, inherent to independent suspension designs, further enhance the performance of the military vehicle.

Mobility is improved with the independent suspension by increasing ground clearance between the wheels. In the solid axle suspension, the bevel gearbox and axle housing are located at the wheel center height. The bevel gearbox extends some distance below this line, and substantial jounce clearance is provided above. Traveling through plastic soils, over rutted, snow-covered, or rock-strewn terrain is limited by the minimum ground clearance. With the independent suspension, the bevel gearbox is rigidly attached to the frame or body, thereby providing substantially greater ground clearance.

11-6.4.7 Optimum Wheel Spacing (Ref. 16)

Heavy vehicles employ numerous wheels in tandem to improve traction and floatation characteristics. Solid axle, multiwheel suspensions employ a walking beam, the merits of which were discussed under bogie suspensions in par. 11-5.3.5. The bogie configuration dictates the wheel spacing. This spacing is not necessarily the optimum for equal loading of all of the vehicle wheels. The independent suspension allows more flexibility in locating the wheels along the length of the vehicle so that all of the wheels of a 6×6 or 8×8 vehicle are more nearly equally loaded. The equal loading results in more favorable locomobility over plastic soils. It is possible to interconnect the tandem, independently suspended wheels in a manner to obtain load-dividing characteristics, such as are inherent to a bogie suspension. However, this arrangement would have the effect of stiffening the springs, thereby resulting in reduced suspension performance at high speed. The load-dividing properties would of course be lost, once a wheel bottomed on the frame stops as a result of jounce.

Wheel spacing may be determined from a study of the dynamic forces and suspension system vibration characteristics. A mathematical model of the vehicle suspension system is set up on a computer, and the suspension characteristics are studied as the vehicle travels over assumed road irregularities of various types and sizes. A wheel spacing is then selected that will give the best ride characteristics within the limits imposed by other considerations.

11-6.5 INDEPENDENT SUSPENSION DISADVANTAGES

The disadvantages of independent suspensions for military vehicles are primarily those of cost and maintenance. The separate rigid members employed to maintain wheel alignment require more careful design, more expensive machining, and costlier bearings than the simple solid axle, leaf-spring suspension. The pivot bearings must be properly lubricated lest excessive clearances induce vibrations and disturbances in the steering and suspension systems. The independent suspension is not inherently as rugged as is the solid axle type. Continuing development of the independent suspension for military vehicles will minimize these disadvantages.

11-7 SUSPENSION SELECTION

The military vehicle designer is often faced with the problem of deciding which type of suspension to employ. He must consider all of the factors discussed in the foregoing plus additional factors relating to production schedules, availability, etc. A table similar to that shown in Table 11-1 is useful in making this judgment. In this table, the factors to be evaluated are listed on the left. The second column, totaling 100%, assigns relative values to the factors based upon their degree of importance to the particular vehicle program under consideration. In the remaining columns, each factor is assigned a value for each type

TABLE 11-1 COMPARISON OF SOLID AXLE AND INDEPENDENT AXLE SUSPENSION (Ref. 16)

		SOLID AXLE SUSPENSION			INDEPENDENT AXLE SUSPENSION		
		4×4	6×6	8×8	4×4	6×6	8×8
Evaluating Factor	Value of Factor	2 Single Axles	1 Single 1 Bogie	2 Bogies	All Independent Long and Short Arm, Torsion Arm		
Weight of complete suspension for vehicle using 2-1/2-ton military axle		1,539 lb	2,599 lb	3,690 lb	1,816 lb	2,732 lb	3,632 lb
Number of existing production parts used by % of total	25	75	75	75	50	50	50
% of interchangeability of pieces between each vehicle	15	75	100	75	100	100	100
Ease of maintenance based on 100% optimum	15	100	90	80	90	80	70
Vehicle concept adaptation based on 100% optimum (including weight advantage)	10	95	85	85	100	100	100
Development time based on 100% optimum	25	95	95	95	90	90	90
Mobility rating based on 100% optimum	5.	94	96	98	96	98	100
Steering accuracy and ease based on 100% optimum	2.5	85	85	85	100	100	100
Handling based on 100% optimum	2.5	85	85	85	100	100	100

RESULT: Solid Axle Suspension 25,918 points
Independent Axle Suspension 24,570 points

of suspension and each type of vehicle, based upon an optimum rating of 100%. These ratings are then multiplied by the values of each corresponding factor, given in the second column, to determine a point value for each. (These are not shown in Table 11-1). The points are then totaled for each suspension. In this way, a quantitative evaluation is obtained. It must be noted that the technically superior design is not necessarily the one most likely to meet the requirements of the program. The relative weights of each factor influence the decision. For example, in Fig. 11-7, the program emphasized existing production parts; consequently, this factor received a high value, thereby giving the solid axle suspension preference. If some of the other items, such as, handling, steering, and parts interchangeability, had been given greater values in relation to availability of production parts, the independent suspension might have been selected.

SECTION III SUSPENSIONS FOR TRACKED VEHICLES

11-8 INTRODUCTION

Considering the track as a prepared roadway which the tracked vehicle carries with it, the tracked suspension is functionally identical to, and design-wise not too dissimilar from, that for wheeled vehicles (Ref. 5). Its purpose is to increase vehicle mobility by improving floatation, traction, and speed. The role of the suspension cannot be overemphasized in the effort to increase cross country speed (Ref. 17). The inadequacies of the suspension are the factors which limit vehicle speed over rough terrain. Generally, the power plant and drive gear are capable of greater speeds over irregular terrain than the operating personnel can withstand. Severe jolting, bouncing and pitching of the vehicle cause the operator to reduce speed lest control be lost or injury result (Ref. 4).

The fundamental objective of tracked suspension design is to place between the track and hull a strong, simple, rugged, and easily maintained system of sufficient flexibility that will (1) minimize shocks and vibrations transmitted to the hull; (2) support the sprung mass in a stable manner; and (3) distribute the weight of the vehicle uniformly along the track (Ref. 18).

The design objective is realized by incorporating some, or all, of the following features into the design:

1. A substantial number of support points (road wheels).
2. A means for equalizing the load on different sections of the track (bogies).
3. A means for providing resilence (springs).
4. A means for damping the shocks and vibrations (shock absorbers).

To provide optimum floatation and traction, the ground supporting the tracked vehicle should be uniformly loaded. The tracks of high-speed tracked vehicles are flexible, possess little beam strength, and, in themselves, have but limited load distribution capabilities. The ground pressure is therefore, greatest directly under the road wheels and is the least about half-way between them. It is the peak ground pressure under the tracks that usually determines locomobility in marginal soil conditions and not the average ground pressure (Ref. 3). It follows, then, that the greater the number of support points, the less will be the variation in the ground pressure. An infinite number of road wheels would be desirable. Such a condition is obtained with the skid suspension, where the track is supported by a flexible, pneumatically-inflated bag (Ref. 19). Preliminary testing substantiates the validity of this concept. There are many difficult design problems associated with the skid suspension, and practical vehicles embodying this type of suspension are not likely in the immediate future.

To obtain a large number of support points with wheels, the wheels must either be small, resulting in inferior rolling characteristics on the track, or, if they are large, they must be staggered, resulting in a twisting of the track, difficult maintenance of the inboard road wheels, and additional areas subject to impaction by snow, mud and stones (Ref. 20).

Early military tracked vehicles employed a rigid suspension; i.e., the tracks engaged rollers that were rigidly mounted to the vehicle frame. The theory was that the great weight of the vehicle would iron out terrain bumps, and the vehicle frame would bridge holes and ditches. This design gave little consideration to soil mechanics and vibration isolation techniques. The resulting vehicles could travel at only very slow speeds and became easily stalled. It became obvious that the vehicle had to be flexibly supported on the track to permit greater locomobility and speed (Ref. 10).

Suspensions for tracked vehicles fall into two main categories, bogie suspension and independent suspension. Classifications within these two types is determined primarily by the kinds of springs employed and the manner in which they are applied.

11-9 BOGIE SUSPENSION

11-9.1 GENERAL

A bogie for a tracked vehicle (tank bogie) is a suspension assembly wherein a system of links, arms, and springs is interconnected in such a manner as to permit two or more road wheels to function together in tandem (Fig. 11-7). This type of suspension was used almost exclusively on early

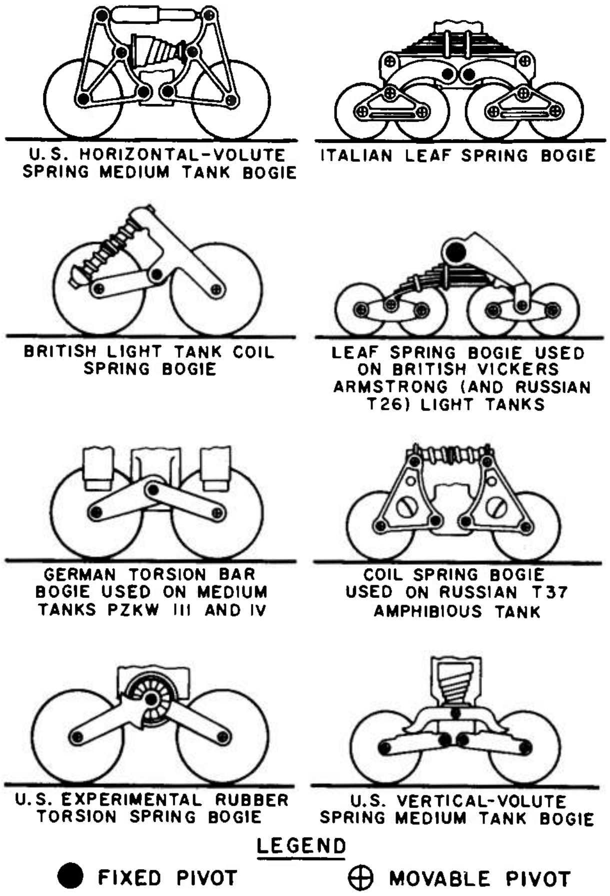

Figure 11-7. *Bogie Suspensions*

flexibly suspended tracked vehicles, and many design variations were developed. The bogie suspension provides a walking-beam effect that divides the load equally between a pair of tandem wheels. When one of the wheels is displaced, or is subjected to a track force, an equal force is reflected on the other wheel of the bogie unit. This arrangement permits the vehicle to negotiate substantial terrain irregularities without transmitting undue shocks to the sprung mass, and without varying excessively the load distribution or vehicle elevation (see Fig. 11-8). The bogie behaves in the manner intended at low speeds only. At speeds above 10 mph, the walking beam effect is lost and the entire bogie unit "pancakes," leaving but limited spring travel to reduce impacts to the sprung mass (Ref. 6).

Several of the more common bogie suspensions employed on tracked military vehicles are discussed in the following paragraphs.

11-9.2 VERTICAL VOLUTE SPRING SUSPENSION (Ref. 10)

Figure 11-9 shows the principal functional parts of a typical vertical volute spring suspension. Each of a pair of bogie wheels is mounted to one end of a pair of suspension arms that are pivoted at their other ends about pivot centers fixed relative to the vehicle hull. The two ends of a rigid suspension lever are forced against the top sides of both suspension arms by a vertical volute spring which bears on the midpoint of the suspension lever. The suspension lever is pivoted at its center on a trunnion which is guided by a vertical slot in the fixed structure. The suspension lever can rotate about its trunnion, and the trunnion can be displaced vertically, but it cannot be displaced horizontally because of the influence of the suspension lever guide on the guide trunnion. The vertical volute spring transmits the loads (vehicle weight and vertical accelerations) between the suspension lever and the hull.

When a load is applied to both road wheels, the suspension arms rotate about their pivots and the suspension lever trunnion slides upward in the vertical suspension lever guide, while a sliding motion occurs between the suspension arms and the suspension lever. The spring is compressed, thereby

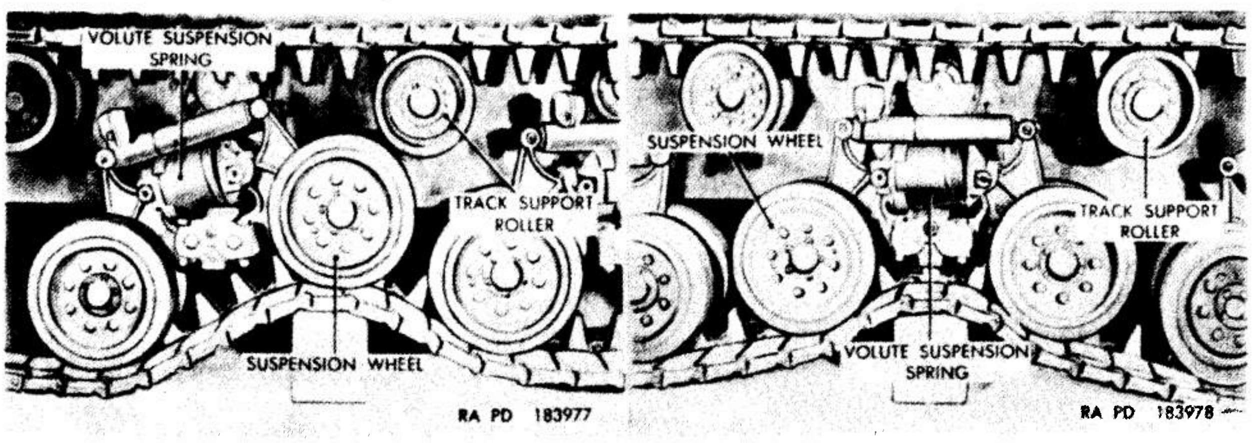

Figure 11-8. *Bogie-Suspended Track Negotiating Obstacle*

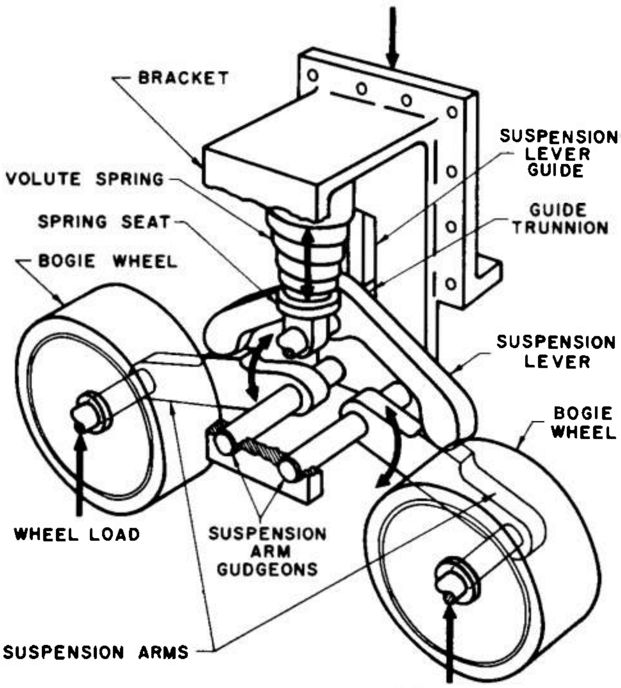

Figure 11-9. *Schematic Drawing of Vertical Volute Spring Bogie*

transmitting the applied load to the suspension attachment point on the hull. When only one wheel is displaced, such as when traveling over a terrain irregularity, the suspension lever tends to rotate as well as deflect vertically. The resulting wheel motion is a combination of angular and linear displacement that permits the track to conform to the irregularity with a minimum change in ground pressure, vehicle elevation, and track length.

The most serious disadvantage of the vertical volute spring suspension is that it cannot be satisfactorily damped. The numerous exposed sliding surfaces are subject to extreme variations of friction coefficients, thereby resulting in substantial changes in the inherent damping. When the surfaces are well lubricated, the vehicle may pitch and bounce excessively. When the surfaces are covered with soil, the ride is harsh because the impact isolation function is impaired.

11-9.3 HORIZONTAL VOLUTE SPRING SUSPENSION (Ref. 10)

The horizontal volute spring suspension shown in Fig. 11-10 consists of essentially the same types of components as are employed in the vertical volute spring suspension. The fundamental difference

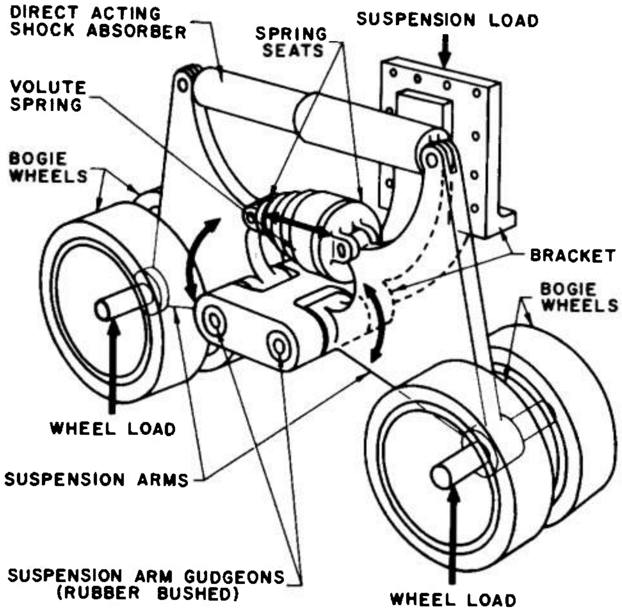

Figure 11-10. *Schematic Drawing of Horizontal Volute Spring Bogie*

in the design is the placement of the springs. In the horizontal volute spring suspension, the spring is located between bellcrank-like suspension arms, thereby avoiding the many exposed sliding surfaces associated with the vertical volute spring suspension. When a load is applied to the bogie wheels, the suspension arms pivot upward, transmitting the force directly to the spring, thereby compressing it. Vertical displacement of only one road wheel changes the spring load, thereby changing the load on the other wheel correspondingly. The balance of forces in the suspension mechanism results in a wheel motion that is a combination rocking motion and vertical deflection.

The design configuration of the horizontal volute spring suspension facilitates the installation of shock absorbers. The damping characteristics of the suspension system can be more accurately controlled because the inherent damping of the suspension mechanism is less subject to variations induced by environmental conditions.

11-10 INDEPENDENT SUSPENSIONS

11-10.1 GENERAL DISCUSSION

Independent suspensions for tracked vehicles resulted from the requirements for greater speed of military vehicles. Just as the independent suspensions are used with wheeled vehicles to avoid the high-speed shortcomings of the solid axle, so the independent suspension is employed on tracked

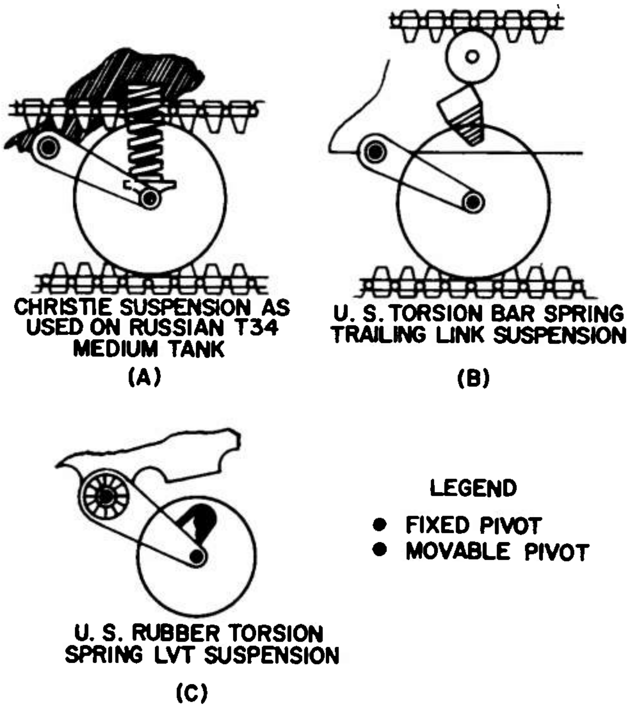

Figure 11-11. *Independent Trailing Arm Suspensions*

vehicles to overcome the high-speed inadequacies of the bogie suspension. The performance of the independent suspension at higher speed has been so superior to the bogie suspension that, currently, only independent suspensions are used on United States tanks (Ref. 10). The present emphasis on independent suspensions may influence the designer to overlook the merits of the bogie suspension. The bogie still offers superior performance at slow speeds (below 10 mph) (Ref. 8).

Figure 11-11 shows several typical independent suspensions for tracked vehicles. It will be noted that these are all of the trailing-arm type. The swinging arm, swing axle, and parallel wishbone types, so popular on wheeled vehicles, are not suited for use on tracked vehicles, primarily because of vertical displacement of a wheel on these suspensions results also in a transverse displacement (Ref. 8). The vertical guide type would be compatible with tracks. However, the trailing arm is such a simple and rugged mechanism, so well suited to track application, that the other types of mechanism have not been employed.

The bogie suspension derives its impact-absorption and obstacle-swallowing abilities from a combination of articulation of a tandem set of wheels plus a deflection of the suspension spring. Because a substantial amount of the wheel deflection is through articulation (rocking), the suspension spring is relatively stiff. At high speeds, when the lead wheel of a bogie encounters an obstacle, this wheel deflects (spring is compressed and bogie unit rotates) and the impact is satisfactorily isolated from the sprung mass. However, the time interval before the subsequent wheel encounters the obstacle is not sufficient to allow the spring to recover its static position. As a result, little deflection capacity remains in the spring and a severe impact is transmitted to the sprung mass when the second wheel encounters the obstacle (Ref. 6).

The independent suspension employs a separate elastic element for each road wheel which provides for the entire vertical displacement of the road wheel. The displacement of one road wheel does not substantially change the spring force of the remaining suspension springs. This is particularly true at high speed when the wheel may be deflected by the terrain irregularities so rapidly that the elevation of the sprung mass is unaffected. Since the suspension springs are not preloaded by the deflections of the preceding road wheels, optimum impact isolation capabilities are available at all wheels at all times. The resulting suspension is relatively soft, having a spring rate about one-quarter that of the volute spring bogie suspension. This is a disadvantage from the standpoint of pitching and bouncing, and damping is absolutely necessary to stabilize the motions of the sprung mass. Although the softer spring suspension requires dampers of less energy dissipation capacity, damping is more essential to its satisfactory performance. Table 11-2 compares the characteristics and merits of the independent suspension with the bogie suspension.

Independent suspensions are classified according to the types of spring they employ. The early systems used coil springs while current independent suspensions use torsion bars almost exclusively. A description of several types with characteristics, merits, and disadvantages is presented in the paragraphs that follow.

11-10.2 CHRISTIE SUSPENSION

The Christie suspension is characterized by its use of large, independently suspended road wheels which also serve to support the upper or return portion of the track. Today it is referred to as the flat track suspension. The early Christie

TABLE 11-2 CHARACTERISTICS AND QUALITIES OF BOGIE SUSPENSION VERSUS INDEPENDENT SUSPENSION FOR TRACKED VEHICLES

INDEPENDENT SUSPENSION	BOGIE SUSPENSION
1. The displacement of one or several road wheels does not substantially change the displacement of the remaining road wheels.	1. The maximum displacement of a single road wheel is greater than the maximum simultaneous displacement of all the road wheels of the entire bogie unit.
2. Changing the loading of one road wheel does not change the loading of the others while the hull is stationary. Redistribution of the loads on the road wheels is possible only by changing the position of the hull.	2. Changing the loading of one road wheel correspondingly changes the loading of the other wheels of the bogie unit.
3. In encountering a terrain irregularity, an impact is immediately received by only one road wheel.	3. In encountering a terrain irregularity, the impact is received by all of the road wheels of the particular bogie unit.
4. Decommission of one road wheel does not impair the effectiveness of the remaining road wheels except for the increased load.	4. Decommission of one road wheel impairs the functioning of the entire bogie unit.
5. Large diameter road wheels are usually employed; never less than 18 in.	5. Numerous small-diameter road wheels are used, improving weight distribution but reducing tire life.
6. Mechanically less complex than bogies, thereby facilitating maintenance. More protected from damage by enemy action and terrain debris.	6. More complex than independent; therefore, more subject to malfunction and difficult to maintain. More exposed to ballistic attack.
7. Usually softer suspension with higher reserve of energy absorption and road wheel deflection, compared with bogies. Dampers essential.	7. Usually harder suspension resulting in harsh jolting ride.
8. Recommended for high-speed (over 10 mph) operation because of superior impact absorption reserve.	8. Inferior for high-speed operation because of preloading of suspension spring by interconnection of road wheels in tandem. Superior for low-speed locomobility because interconnection equalizes load.
9. No inherent track tension adjustment to compensate for road wheel deflections.	9. Substantial inherent track tensioning compensation for wheel deflections.
10. Combined types offer certain functional advantages but not advised because of increased parts inventory requirement.	

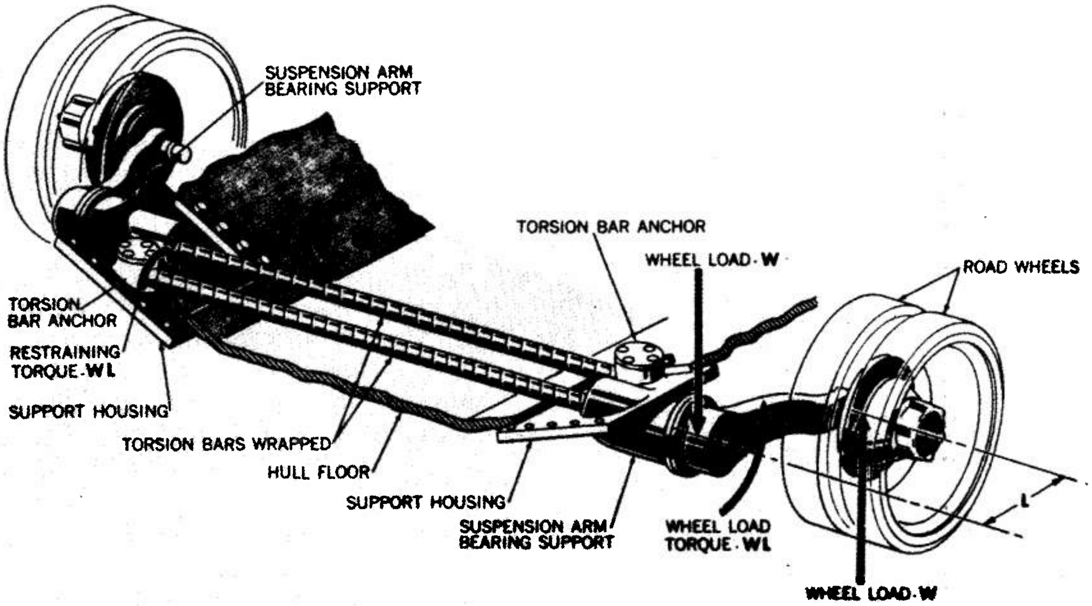

Figure 11-12. *Torsion Bar Suspension*

suspensions employed coil springs as their elastic elements. The road wheels were mounted on trailing arms, and the weight of the vehicle hull was transmitted through the coil springs bearing on the trailing arms. This suspension allowed about 14-in. deflection of the road wheels against the springs, thereby, providing improved high-speed impact isolation characteristics. The primary objection to the original suspension was that the coil springs occupied substantial volume, imposing a penalty on the width of the vehicle. Another disadvantage resulting from the use of large wheels was the limited number of support points possible, resulting in a considerable variation of ground pressure along the track. The load distribution along the track was improved by overlapping the wheels, and the spring space problem was eventually solved by the development of the torsion bar (Refs. 6, 10, 17).

11-10.3 TORSION BAR SUSPENSION (Refs. 1, 6, 10, 18)

The torsion bar was introduced to tracked vehicle suspensions to achieve independent suspension without increasing vehicle width. It is an efficient spring for its weight in terms of impact reduction. The adaptation of the torsion bar to tracked vehicle suspension involves design configurations and techniques peculiar to this type of spring. A typical torsion bar track suspension is shown in Fig. 11-12. The road wheels are mounted on trailing suspension arms which are not interconnected by springs, links, levers, etc. The crank-shaped arms are connected to the hull through antifriction bearing mountings which relieve the springs of all except the torsional loads. One end of a torsion bar is splined to the pivoting axle of each suspension arm. The other end of the bar is rigidly secured to the opposite side of the hull. Angular displacement of the suspension arm is resisted by the torsional spring force (moment) of the bar. Because the arms operate in sealed antifriction bearings, and because of the inherent lack of friction in the spring, little natural damping of the spring system is available. This condition makes the use of shock absorbers essential to minimize bouncing and pitching of the sprung mass. This is a rational design approach because the designer can select the shock absorbers for optimum damping with minimal interference to the impact absorption properties of the spring (Ref. 21). Shock absorbers are not usually installed on road wheels at or near the center of the vehicle, because, at this location, they would have little effect on reducing pitching, which usually takes place about a transverse axis close to the center of gravity of the vehicle. Volute springs or rubber bumpers are usually employed to limit the travel of the suspension in jounce.

The simple configuration of the torsion bar and the nature of its elastic operation contribute toward simple suspension design. It is readily adapted to the trailing arm mechanism which, in itself, is a simple and compact means of maintaining road-wheel alignment. The basic all-around simplicity permits the use of identical, easily installed and maintained components for most of the road wheels of the vehicle. Because the springs are located within the vehicle hull, they are protected against ballistic attack and terrain debris. However, the location of the torsion bars across the bottom of the hull increases the height of the vehicle if reasonable ground clearance and internal heights are maintained. The means of securing the stationary end of the torsion bar is well suited to applying mechanisms for varying the spring preload, thereby providing a means for adjusting the free standing position of the vehicle to best suit the operating conditions.

Torsion bars are durable. Comparable life data of commonly used suspensions are:

Torsion bars	—	4,000 miles
Horizontal Volute Spring	—	2,500 miles
Vertical Volute Spring	—	1,200 miles

11-11 TRACK-SUPPORTING AND TENSIONING COMPONENTS (Refs. 1, 6, 10, 18, 22)

The return run of the track must be properly supported to prevent excessive sag, and adequate track tension must be maintained to prevent track throwing. Excessive track tension increases power losses, while insufficient tension results in thrown tracks and in the track skipping over the sprocket teeth. Thus, the main function of track tensioning devices is to limit track tension to a practical value that minimizes power losses and still prevents track throwing.

On suspensions employing relatively small road wheels, separate small rollers or metal skids, secured to the vehicle hull, are provided to support, guide, and retain the upper run of the track. The number of supports provided depends upon the type and length of the track. The rollers may be rubber-covered to reduce noise and vibration. For tracked vehicles operating mostly in snow and in arctic regions, it is advisable to use laminated plastic or impregnated wooden skids to preclude the immobilization of the track by freezing. On flat track (Christie) suspensions, the tops of the relatively large road wheels support and help convey the upper run of the track.

As a tracked vehicle passes over obstacles and depressions that cause the road wheels to deflect, the perimetric length of the track envelope necessary to pass around all the road wheels and sprockets is caused to vary. Since the perimetric length of the track envelope in a given suspension is fixed, motions of the road wheels will result in track slack or excessive track tension, which interfere with the proper load distribution and impact isolation functions of the suspension. To prevent excessive variations in track tension, separate track tensioning mechanisms must be provided. These mechanisms must maintain proper track tension over a wide range of speeds and terrain irregularity.

Track tensioning is provided in modern tracked vehicles by two methods. One method is to spring-load a separate small idler roller. On rear-driven tracks this tensioning idler roller is usually located between the last road wheel and the drive sprocket. The spring force exerted through the roller onto the track maintains track tension and reduces track skipping between the rear road wheel and the drive sprocket. The track-tensioning idler roller located at this point also reduces the deflections of the rear road wheels that would otherwise be caused by sudden increases in track tension during acceleration. A rocking backwards of the vehicle still occurs due to the horizontal acceleration forces, but the tension-idler roller substantially reduces the pitching tendency of the vehicle during acceleration.

The other means of maintaining track tension during changes in the track contour is the compensating idler wheel. Compensating idler wheels are usually the same size and type as the road wheels and are mounted on either individual suspension arms or extensions of the front road wheel suspension arms. They are interconnected to the front road wheel suspension arms by a system of links in such a way that a vertical displacement of the road wheel produces a horizontal displacement of the compensating idler. The displacement of the compensating idler must be carefully considered. Usually it is designed so as not to take up all of the slack, lest excessive track tension be developed by deflection of the subsequent road wheels. It may be feasible, in a particular design, to combine the spring-loaded and compensating idlers into a single mechanism; i.e., mount a compensating idler roller on a resilient arm.

The mechanisms for maintaining track tension usually incorporate adjusting devices to compensate for track pin wear and also provide for ready release of the track tension to facilitate track installation and removal.

Although considerable progress has been made in the design and development of track-tensioning devices, track derailments still occur. The requirements for the tensioning devices are severe. They must allow for rapid road-wheel deflection without impairing the impact isolation characteristics of the suspension. Since the entire suspension is a complex spring-mass system, it is subject to the resonant vibrations attendant to those systems. Considerable analytical and design effort remains before a system that will function reliably at high speed over irregular terrain is achieved.

11-12 ROAD WHEELS

11-12.1 GENERAL

The tracked vehicle rolls on the bottom of its tracks by means of road wheels, sometimes referred to as bogie wheels, a carry-over from the period when only bogie suspensions were used. Considering the track as a prepared roadway, the design approach for the road wheel is similar to that for the wheeled vehicle; i.e., a vehicle of a certain weight will require a certain number of wheels of a particular load capacity. Although the design problem is alleviated somewhat by the fact that road wheels are not required to provide traction and floatation, the conditions associated with directional control and stability present challenging problems which demand the utmost in design ingenuity to provide even reasonable life of the road wheel.

11-12.2 ROAD WHEEL SIZE

Large-diameter road wheels are more desirable because they offer less rolling resistance and are less likely to become clogged by mud, snow, and stone impactions on the track. Generally, road wheel diameters are selected to be compatible with the limited space available for the suspension (Ref. 24). Bogie-type suspensions generally employ rather small-diameter road wheels, while independently-suspended road wheels are usually more than 18 in. in diameter (Ref. 8). The width of the road wheel is also limited by the space available. To facilitate the supply and maintenance problems, the sizes of some road wheels have been standardized for certain types of vehicles. For example, several of the current medium and heavy tanks employ the same 26 $\times$ 6 road wheels, each tank using a different number of wheels to keep the loading within the rated capacity of the wheels (Ref. 10). Similar standardizations are to be expected in other classes of tracked vehicles. The designer of new vehicles of improved capabilities should not limit his selection of road wheels to the standardized components, because new standards will be created if sufficient design advancemeut can be demonstrated.

11-12.3 ROAD WHEEL MATERIALS

For the most part, road wheels are steel, disk-type and of riveted or welded construction (Refs. 5, 10). Since vehicle weight, or rather the reduction of it, has become of such vital importance in the transportability and mobility of modern warfare equipment, much effort is being expended to develop lightweight road wheels. Cast and forged aluminum and magnesium road wheels are being developed, and some vehicles are successfully using them. Careful design and surface treatment are required to extend the fatigue life of these materials (Ref. 25). Experiments with high-strength, reinforced-plastic road wheels to achieve a substantial savings in weight are meeting with some success (Ref. 26).

11-12.4 BEARINGS

The bearings employed for the road wheels are the antifriction types and they must be selected to withstand the severe thrust and radial shock loads experienced. Adequate sealing against the severe environment is of the utmost importance. The usual automotive-type seal does not provide sufficient protection against the entrance of mud, dirt, water and snow into the bearing. It is well to provide some supplementary protection to inhibit the accumulation of foreign substances in the bearing area (Ref. 10).

11-12.5 ROAD WHEEL TIRES (Refs. 5, 10, 17, 18, 24)

11-12.5.1 General

Perhaps the most difficult design problems associated with road wheels concern the tires. Early tracked vehicles employed steel-tired road wheels. At the low speeds traveled by early tracked vehicles, the shocks, vibrations, and noise that were produced as the wheels rolled relative to the track were tolerable. The steel-tired wheels or rollers, bearing against the track guides, provided good directional control of the tracks. However, even at slow speeds, the wheels and tracks wore rapidly and required frequent replacement. At higher speeds, the impacts and resulting damage to wheels and tracks, were excessive. The application of solid rubber tires to the road wheels reduced the impacts considerably and permitted greater vehicle speeds while increasing the life of the tracks and wheels. Much effort has gone into the development of the solid rubber-tired road wheel to extend its capabilities, but shortcomings of the design still leave much to be desired in vehicle performance and tire life. Pneumatic tires provide improved impact isolation but other characteristics have limited their use on tracked military vehicles. They have been used successfully on certain lightweight vehicles, such as the M76 (Otter) and the M56 (Scorpion) (see Chapter 4).

11-12.5.2 Causes of Tire Failure

Two principal factors create the severe conditions which cause most of the road wheel tire failures. They are (1) track guide engagement, and (2) high tire loading. Each of these factors is discussed here to alert the designer to the difficulties, and possibly suggest some design solutions.

11-12.5.2.1 Track Guide Engagement

The alignment of the track relative to the vehicle is maintained by the driving sprocket, the idler and the road wheels. A certain amount of clearance is provided between the road wheels and the track guides, so that the road wheels seldom contact the track guides, as long as the tracked vehicle travels a straight course on level, smooth terrain. During turning movements, sideslope operations, and when traveling over irregular terrain, the road wheels experience very large lateral forces which they must transmit and resist. These lateral forces are transmitted through the track guide resulting in severe rubbing and scuffing of the tires. Since the track guide presents an interrupted surface to the road wheels, the road wheels undergo continuous impacts and they try to climb over the guide causing chunks of rubber to be gouged from the tires. Attempts have been made to extend the rims of the road wheels, so that the rims, rather than the tires, engage the track guides; but these experiments have not produced the desired results. Instead, the track guide becomes trapped between the tire and rim flange, creating even greater damage (Ref. 27). Furthermore, the rim restricts the lateral deflection of the tire, thereby, aggravating the other principal cause of road wheel tire failure—overload.

11-12.5.2.2 Tire Overload

The repeated distortion of the solid rubber tire under load results in heating of the tire. This is a result of the hysteresis, or energy consumption characteristics of the rubber. The heat generated is a function of the magnitude and cyclic rate of deflection, tire width and thickness, type of elastomer, and degree of lateral restraint. When the interior temperature of the tire is high enough, the rubber separates internally, and the tire actually explodes and blows out. Separation of the vulcanized bond at the rim may also occur. To minimize the occurrence of tire failure, the following three design requirements have been established (Ref. 10):

1. Tire width should not exceed 7 in. If the wheel loading and vehicle configuration are such that greater tire width is required, then two tire sections should be used. Limiting the tire width maintains the lateral shearing forces at the tire-rim junction at acceptable values.
2. The tire diameter should be as large as practicable. The load-carrying capacity of solid rubber tires varies as the 2.25th power of the outside diameter. This factor is a function of the internal heat build-up under rated load and vehicle speed.
3. Tread thickness should be limited to minimize tire blowout. At present, the tire thickness for medium and heavy tanks is 1-5/8 in. and for light tanks is 1-11/32 in. The additional flexibility of thicker tires is desirable to reduce the impacts; however, the greater flexing generates more internal heat, thereby causing blowouts.

Road wheel tire failure is caused by impact loading as well as by steady loading. For this reason, independent soft sprung suspensions, having a lighter unsprung mass, are more favorable to prolong tire life. Tire loadings of a vehicle in operation are greater on the front and rear road wheels due to acceleration forces, pitching, load redistribution when operating on slopes, firing of weapons, and operation of equipment mounted on vehicle (see Chapter 5).

Specifications regarding the solid rubber tires are covered in MIL-T-3100A. Tables in this specification show the tire load capacity for various diameters and vehicle speeds.

11-12.5.3 Pneumatic Road Wheel Tires (Refs. 17, 18, 24)

Because of their superior shock isolation characteristics, which allow greater vehicles speeds, pneumatic tires are just as desirable for use on tracked vehicles as on wheeled vehicles. However, pneumatic tires provide less lateral stability, are more vulnerable to ballistic attack, and generally require more space than solid tires. Despite these disadvantages, pneumatic road wheel tires are gaining in popularity, especially in lightweight vehicles. Many light tanks, light self-propelled antitank weapons, and amphibious support vehicles are now equipped with them. Pneumatic road wheel tires operate at higher inflation pressures and have thicker sidewalls than their wheeled vehicle counterparts.

SECTION IV SPRINGS

11-13 GENERAL

The spring is the primary elastic element in the suspension. As such, its purpose is to cushion the vehicle body or hull against the vertical acceleration resulting when the ground-contacting parts of the suspension negotiate terrain irregularities. It is beyond the scope of this handbook to present detailed spring design procedure. Information of this type may be found in the texts and articles listed in the References and Bibliography. It is rather the aim of this section to describe the characteristics of various spring types and explain what characteristics determine the application of certain types of springs to particular types of suspensions. Quite frequently, factors other than the impact isolation capabilities, dictate the type of spring to be used. The impact isolation properties are similar enough for most springs to achieve the desired reduction of vertical acceleration of the sprung mass, relative to the unsprung mass. Factors such as weight, cost, space availability, configuration, adaptability to general vehicle and suspension design, ruggedness, resistance to ballistic attack, damping requirment, etc., govern the selection of the spring type as much as the cushioning characteristics.

11-14 VEHICULAR SPRING TERMINOLOGY

11-14.1 GENERAL

Although detailed information regarding the design of springs will not be presented here, it is well to discuss a few of the more common terms associated with vehicular springs.

11-14.2 SPRING RATE

The spring rate is the ratio of the change in spring force to the change in spring deflection, i.e.,

$$k = \frac{F}{x} \tag{11-5}$$

where

k is the spring rate
F is the change in spring force
x is the change in spring deflection

Springs having a high value of k are termed hard or stiff springs while those with a low value of k are called soft springs. The spring rate for metal springs is a function of the dimensions of the spring and the physical properties of the spring material. Springs whose rates are independent of deflection are termed constant rate springs. The spring may be designed to have a spring rate that varies with the deflection of the spring. These are called variable rate springs.

11-14.3 SUSPENSION STIFFNESS

The flexibility or impact cushioning capabilities of a suspension depends not only on the spring rate, but also upon the magnitude of the mass supported by the spring. To achieve cushioning against impacts, large deflections of the unsprung mass relative to the sprung mass must be permitted. The effect of the sprung mass on suspension stiffness may be understood if one considers a spring, supporting no mass, installed in a suspension system. When the ground-contacting element of this suspension is accelerated upward, the entire spring, including the end normally connected to the sprung mass is accelerated upward at substantially the same high rate as the ground-contacting part. If the spring supports a mass, the inertia of the mass resists the acceleration and the spring is compressed, thereby resulting in a lower acceleration of the sprung mass than of the ground-contacting member. The amount of spring compression and reduction of acceleration is a function of the size of the mass. It can be seen then, that a spring that provides a soft suspension (large deflection) with one loading will produce a stiff suspension if the suspended mass is substantially reduced.

The degree of suspension stiffness is frequently associated with particular types of springs although the spring type itself has no direct bearing on the suspension stiffness. It is usually other requirements that dictate the suspension stiffness and certain other characteristics of particular springs make them more suited for use in suspension with a particular degree of stiffness.

Suspension stiffness is associated with the natural period or frequency of vibration of the spring-mass system by the relationships:

$$T=2\pi\sqrt{\frac{W}{kg}}\,;\, f=\frac{1}{T}=\frac{1}{2\pi}\sqrt{\frac{kg}{W}} \qquad (11\text{-}6)$$

where

- T is the spring period
- W is the weight of sprung mass
- k is the spring rate
- g is the acceleration due to force of gravity
- f is the frequency of oscillation

In par. 11-1 is was stated that it is desirable to maintain a natural frequency of 60 to 120 cpm (1-2 cps) for vehicle suspension. Once the weight that the suspension spring must support has been established, a spring rate is selected that will result in a natural frequency within the desired range. Employing springs with a higher rate will result in a higher frequency and stiffer suspension.

11-14.4 ENERGY STORING CAPACITY

The designer of military vehicles is frequently on the alert for means of producing the minimum weight device that will meet the functional requirements. A criterion for indicating the relative weights of different types of springs is the energy storing capacity per lb of spring material. Table 11-3 shows this factor for several springs commonly used in vehicle suspensions. The energy storing capacity per cu. in. of spring material is also given. These values must be used with caution because factors not apparent in the table may discount favorable energy storing characteristics. For example, the volume considered for Table 11-3 is that of the spring material only. It does not take into account volumes rendered useless for other equipment because of the spring configuration.

11-15 ACTION OF VEHICLE SPRINGS

In par. 11-3, it was stated that suspensions for military vehicles must be designed to withstand 8-G impacts without transmitting more than 3-G's to the sprung mass. The factors producing these impacts and the reasons for the above limits are discussed in this paragraph.

When a solidly suspended (no springs) vehicle with solid wheels negotiates a terrain obstacle, the vehicle body is accelerated upward at the same rate as the wheel axle. This acceleration is a function of the wheel diameter, vehicle speed and shape of the ground obstacle. After passing over the bump, the wheel and body are accelerated downward at a rate not exceeding that due to the earth's gravitational attraction. Because the body follows the ground irregularities, the speed of such a vehicle is limited to keep the vertical accelerations within the limits considered tolerable by the vehicle structure and occupants. These accelerations may be readily calculated provided the shape of the ground obstacle is expressed in simple mathematical terms.

When a spring separates the wheel and body of a vehicle, the vertical acceleration of the body is less than that of the wheel, when impacting an obstacle, because the inertia of the sprung mass causes the spring to deflect. However, after the wheel arrives at the top of the bump, the sprung mass will continue upward and then vibrate on the spring according to the spring-mass laws. The calculation of the accelerations of the sprung mass of a vehicle passing over a bump and its behavior afterward is quite complex, especially when the spring rate of the tire and the effects of damping are considered. The behavior of springs and suspensions of varying stiffness have been determined employing mathematical techniques based on classi-

TABLE 11-3 ENERGY-STORING CAPACITY OF SPRINGS

Type of Spring	Energy-Storing Capacity	
	in-lb per lb of Spring	in-lb per cu. in. of Spring Material
Leaf, equal length steel leaves	100-150	30-40
Leaf, optimum stepped leaves	300-450	85-125
Volute, steel	500-1000	140-280
Hydraulic	600-900	—
Helical, round steel wire	700-1000	200-280
Torsion Bar, steel	1000-1500	280-420
Torsion Bar, rubber	2000-4000	80-160

cal vibration formulas. The results of interest to the vehicle designer are:

1. A soft suspension cushions a bump more effectively than a hard suspension but requires greater body clearances for wheel travel and is more likely to oscillate severely after passing the bump. It is possible under certain conditions for the amplitude of oscillation to exceed the height of the ground bump that induced the oscillation. Damping is essential to maintain vehicle stability.
2. A hard suspension will produce a harsh ride but provides greater vehicle stability after passing over the bump. Most military vehicles employ rather stiff suspensions to minimize the amplitude and duration of the post-impact oscillations.

11-16 TYPES OF SPRINGS

11-16.1 GENERAL

The various characteristics of different types of springs have resulted in an association of certain types of suspensions according to suspension stiffness. For example, helical coil springs are usually associated with soft suspensions whereas leaf springs are frequently associated with hard suspensions. A discussion of the spring types and the characteristics creating the above impression is presented in the paragraphs that follow.

11-16.2 LEAF SPRINGS

Leaf springs are flat bar springs that are relatively thin in proportion to their length and width. Leaf springs for vehicles generally consist of laminations of several leaves of unequal length. Various techniques are used in fastening the laminations, and these produce definite effects upon the spring characteristics of the assembled spring. Most frequently the leaves are curved. When one such unit is used, it is referred to as a semi-elliptical spring. An elliptical spring consists of two of the semi-elliptical units connected in parallel. Laminated, semi-elliptical leaf springs were perhaps the most widely used type of springs for vehicle suspensions. They are characterized by high inter-leaf friction. Although this friction is useful to damp out suspension oscillations, it is frequently excessive and uncontrollable for optimum damping. The inter-leaf friction may be minimized by various design and lubrication techniques.

One of the reasons for the popularity of the leaf spring in vehicle suspensions is that its configuration allows the spring to serve multiple functions. Quite frequently, it is called upon to control front wheel alignment without the aid of rigid control members. The leaf springs of the bogie suspensions shown in Fig. 11-2 serve as structural members to provide "walking beam" action. The springs of some of the suspensions in Fig. 11-1 are required to transmit driving and braking torque, propulsive thrust, and engine torque reaction.

Leaf springs are normally associated with stiff suspensions. This impression is created because in providing the multiple functions, a relatively stiff member is required. In most cases, a softer leaf spring could be employed, in which case, separate rigid members would be required to maintain wheel alignment.

Failures of leaf springs in service are usually due to fatigue. The spring should be designed to eliminate all possible stress concentrations which may result in fatigue cracks. Frequently, the tension side of the spring leaves are shot-peened to improve their fatigue life.

Leaf springs are normally designed to have equal stresses in all of the leaves at maximum load and, thus, exhibit a varying spring rate. The change in effective moment arm with spring deflection introduces an additional variation in the spring rate. Greater variation in the spring rate is achieved by supporting the ends of the spring on curved surfaces so that the effective length of the spring is reduced as the spring deflects.

11-16.3 HELICAL COIL SPRINGS

Hellical coil springs consist of round, square or rectangular wire, wound in the form of a helix, and offer a resistance to a force applied along the coil axis. Most helical coil springs for vehicles are wound with a space between adjacent coils and are loaded in compression. A few vehicular installations employ coil springs in tension, in which case they may be closely wound. The coils may be so closely wound that a force is required to separate them.

Helical springs are characterized by their relative friction-free action. These springs are usually associated with soft vehicular suspensions. Because of the large wheel excursions required for a soft suspension and because the coil spring itself cannot effectively resist lateral forces, separate rigid control linkages are employed to maintain wheel alignment. The separate control linkages provide better wheel control than say a stiff leaf spring, thereby permitting more precise steering control.

Most vehicular helical coil springs are designed with a constant coil diameter and, hence, a constant spring rate. Variable rate characteristics may be achieved by locating the spring in the suspension linkage in such a manner that the effective moment arm varies with deflection. A true variable rate may be obtained by varying the coil diameter. The large diameter coils provide a lower rate.

Helical coil compression springs are used extensively on wheeled vehicles independent suspensions. Their physical size and configuration are readily adapted to certain types of suspension without sacrifice of usable space. However, their use of tracked vehicles has been less popular because of the reduction of the usable hull width resulting from the space consumed by the spring. Typical suspensions employing helical coil springs are shown in Figs. 11-3, 11-4 and 11-6.

11-16.4 VOLUTE SPRINGS

The usual compression volute spring for use on vehicles consists of a relatively wide, thin strip of steel wound to form a distorted spiral. The spring is loaded axially. The spring is usually wound so that adjacent coils rub, thereby producing frictional forces that tend to damp out oscillations. Although the volute spring can be designed to have a constant spring rate, it is normally a variable rate spring because of the varying diameter of the coils. Because of its spiral construction, efficient use is made of the space envelope it occupies and a relatively high force spring of this type consumes little functional volume.

This type of spring was widely used on United States tanks during WW II because its characteristics suited the stiff spring requirements of the bogie type suspension. Figures 11-9 and 11-10 show typical volute spring installation. Currently, volute springs are used as bumper or bottoming springs on soft suspension vehicles. The variable rate and compactness of this type of spring make it ideally suited for this application.

11-16.5 STEEL TORSION BAR SPRINGS (Refs. 51, 52, 53)

The torsion bar spring is a straight metal bar, usually cylindrical, secured at one end to the vehicle frame or hull and loaded in torsion by applying a moment perpendicular to its longitudinal axis by means of an arm secured to the other end. This type of spring is characterized by its friction-free operation. Suspensions employing torsion bars are usually relatively soft and require shock absorbers to minimize pitching and bouncing.

Great care must be exercised in the selection of material, heat treatment, forming and machining of torsion bars to maintain a high quality. Failures usually occur as a result of fatigue at stress concentration points. In addition to the rigid quality control, the fatigue resistance and energy storing capacity may be enhanced by shot-peening, presetting, and anticorrosion treatment. *Presetting* is a process whereby the torsion bar is angularly deflected beyond the yield point, in the direction of loading. This gives the finished torsion bar an allowable stress in the direction of loading greater than the yield stress of the material. However, the strength of the bar when twisted opposite to preset is below that of the original material.

The torsion bar, because of its configuration, allows the use of soft suspension in tracked vehicles without sacrificing usable interior hull width. However, some increase in vehicle height is required if reasonable ground clearances and interior hull heights are maintained. As used in current tracked vehicle suspensions, they are protected against ballistic attack by virtue of their location within the hull. The configuration of the torsion bars allows them to be employed in soft suspensions in such a manner that results in a lower unsprung mass than say the helical coil spring (see Fig. 11-12).

11-16.6 RUBBER SPRINGS (Ref. 54)

The use of rubber and other elastomers loaded in tension or compression, to serve as the principal elastic member, has not proven too successful in vehicular suspensions. However, contoured rubber blocks are frequently employed as bottoming or helper springs.

Rubber loaded in shear exhibits more favorable characteristics and several military, as well as commercial, vehicles have successfully employed suspensions that derive their elasticity this way. The rubber torsion spring generally consists of a metal shaft bonded to an annular layer of rubber which, in turn, is bonded to an outer concentric metal shell. Either the shaft or the shell is rigidly attached to the vehicle frame or hull and the outer metallic component is connected to the unsprung mass by an arm or linkage such that relative angular motion between the shaft and shell results when the unsprung mass is deflected. The spring action is derived from the twisting motion loading the rubber in annular shear.

Rubber torsion springs operate relatively friction-free and suspensions employing them require shock absorbers to damp undesirable oscillations. Rubber compounds with high hysteresis properties are available but are not suited for use as springs because of their poor elastic properties.

Referring to Table 11-3, it is seen that rubber torsion springs possess a favorable energy storing capacity for their weight. However, because rubber is so much less dense than steel, a rubber torsion unit occupies more volume than a comparable steel torsion spring. Another undesirable characteristic of rubber is that it is temperature-sensitive and becomes stiff and hard at low temperatures. Future improvements in elastomers and changes in vehicle requirements may permit greater use of this type of spring. In addition to the favorable energy-storage capacity, rubber torsion springs possess characteristics of basic design simplicity, freedom from noise and lubrication, ability to absorb shock from any direction and resistance to corrosion. Suspensions employing rubber torsion springs, incorporating a system for automatically adjusting the hull end of the unit to maintain a constant hull height for changes in vehicle loading, have been built.

11-16.7 HELICAL TORSION SPRINGS (Refs. 55, 56)

The helical torsion spring consists of a round, square or rectangular wire, wound in the form of a helix, loaded by applying a moment in a plane perpendicular to the coil axis. This type of spring should always be loaded in a direction which reduces the coil diameter. When "winding up" the helical torsion spring, the length is increased. Clearances must be provided to accommodate the increase in length so as not to interfere with the spring action. Most frequently, the helical torsion spring is supported on a rod or in a hole because it is laterally unstable and tends to buckle. Although this type of spring has found only limited use in the past, future trends and changes in vehicle design may require a spring possessing its characteristics.

11-16.8 CONED DISK SPRINGS

This type of spring is frequently referred to as a dished washer or Belleville spring. It consists essentially of an annular metal disk (washer), dished to a conical shape. Coned disk springs are used where space requirements necessitate high spring stresses and short range of motion, i.e., a high spring rate. They are frequently stacked in series to obtain a lower spring rate. They may be used on a vehicle that has no separate soft suspension but where it is desirable to somewhat isolate the body against severe impacts without employing a complex mechanism.

11-16.9 PNEUMATIC SPRINGS (Refs. 57, 58, 59)

A pneumatic or air spring for a vehicle suspension consists essentially of a fabric reinforced rubber bellows of one or two convolutions, sealed at each end by suitable mounting plates, pedestal or clearance chambers. Although the rubberized material deflects elastically when the spring is loaded, the principal source of springiness is derived from the compression of the entrapped air. These springs may be installed on vehicles so they are direct-acting or may be loaded by means of a trailing arm.

One of the characteristics of this type of spring is the variable spring rate. With a small clearance volume, the rate of change of the spring rate is quite pronounced. A factor affecting the spring rate of simple air springs at low loadings is the stiffness of the bellows wall. In some systems the bellows is only slightly bowed at low loading and contributes substantially to the spring stiffness. The more refined air springs employ a system of valves to maintain a midpoint vehicle level for variations in the static loading by controlling the air pressure within the bellows. Under these conditions the bellows is usually bowed and resistance to deflection is substantially reduced.

Air springs generally operate at 65 to 75 psi under static loading but have been used successfully under static load pressures of 2 to 100 psi. The bellows may be inflated to increase ground clearance, thereby improving passability in difficult terrain.

The action of the air spring is friction-free and provides inherent isolation from high frequency vibrations. This feature enables the use of lightweight, low fatigue resistance materials for the superstructure.

Some claim has been made that damping properties are built in the system. This claim is not substantiated by a practical installation and separate shock absorbers are usually employed. The air spring has but little lateral stability and suspensions employing it require separate rigid linkages. The future of air springs on military vehicles is doubtful owing to their vulnerability to ballistic attack and space consumption. As air springs improve, and vehicle design concepts change, it may be feasible to use these springs.

11-16.10 HYDRAULIC SPRINGS (Ref. 60)

The hydraulic spring consists of a sealed plunger or piston working in a highly finished cylinder, against an enclosed volume of liquid. Although the deflections of metal parts of the unit contribute to the springs resiliency, the principal source of elasticity is derived from the compressibility of the fluid. The maximum compressibility (% reduction of volume) of liquids currently available is about 12% for the limiting pressure of 20,000 psi. Recent experiments with these springs indicate that they have characteristics desirable for heavy vehicle suspensions. They are characterized by a very high spring rate. For lighter vehicles or where a softer suspension is desired, the effects of a lower spring rate may be derived by employing a lever system. The principal advantage of the hydraulic spring is its compactness. Although it has about the same energy-storing capacity for its weight as the helical coil spring, it occupies less functional space. Despite the high pressures and small fluid displacement associated with these springs, effective damping is possible.

The disadvantages of the hydraulic spring are associated with sealing the fluid for the high pressures experienced. The plunger and cylinder must be finely finished to function properly and provide proper seal action. The seals are also subject to leakage by wear heating.

11-16.11 HYDROPNEUMATIC SPRINGS

The hydropneumatic spring as used on vehicular suspensions consists of a closed volume of gas separated from the system's hydraulic fluid by a flexible diaphragm. The resiliency of the system is derived from the compressibility of the gas. The hydraulic portion of the system is employed to maintain ground clearances, body or hull leveling, and incorporates damping devices. Suspension lockout is readily accomplished with this system. Because the hydropneumatic system is sealed, malfunctions associated with atmosphere moisture are minimized.

SECTION V SHOCK ABSORBERS

11-17 GENERAL DISCUSSION

11-17.1 FUNCTION

The primary function of the shock absorber is to regulate the suspension spring rebound so that the primary vibrations are damped out, thereby permitting greater vehicle speeds and mobility. These benefits are achieved by virtue of reduced bouncing and pitching of the body or hull and reduced variations of the traction with the terrain (Ref. 4). Additional benefits derived from the use of shock absorbers are: (1) improved ride quality, (2) reduction of wheel dance, (3) prevention of excessive sidesway, (4) reduction of wheel shimmy, and (5) general improvement of the desirable vehicle traveling qualities, collectively termed *roadability* (Ref. 28).

11-17.2 RELATIONSHIP TO SPRINGS

The aim of any suspension system is to maintain contact between the ground and the wheels or tracks with a minimum of shock transmitted to the sprung mass. This effect is accomplished by means of relatively soft suspension springs, which serve to isolate the sprung mass from the motions of the ground-contacting components.

When a vehicle with an undamped suspension system passes over a vertical obstacle, the spring elements are suddenly compressed, and the spring-mass system is set to vibrating. The system will continue to vibrate according to the spring-mass vibration laws until the energy that was put into the spring by the impulse is dissipated by frictional and hysteresis losses. In an undamped system, the sprung mass will pitch and bounce violently, resulting in an uncomfortable, and possibly damaging, ride to the occupants and cargo. The effect is particularly severe when successive shocks are experienced at time intervals near the natural frequency of the system. In this case the amplitudes of succeeding oscillations become additive, and a reduction of vehicle velocity is required lest control of the vehicle be lost. Shock absorbers are incorporated into the suspension system to rapidly dissipate the energy of impact and, thereby, stabilize the vehicle.

The wheel of the suspension system is also subject to vibration. This wheel dance, as it is called, also limits the vehicle speed and mobility. Therefore, in order to preclude or minimize the undesirable spring-system characteristics, damping is required. The damping must be capable of dissipating the vibratory energy at a rate equal to the rate of energy input (Ref. 4). The vehicle speed is limited to fulfill the above requirements.

All suspension systems possess a certain inherent damping tendency which results from (1) friction of joints and bearings, (2) friction of guiding mechanisms, (3) interaction of spring components as in leaf and volute springs, and (4) hysteresis losses of resilient components such as rubber tires, grommets, etc. Seldom, if ever, are the inherent damping forces sufficient to reduce the vibrations to within acceptable limits for all traveling conditions. Also, the natural damping may not necessarily be of the type that contributes to optimum wheel control or to the desired body or hull motion. Therefore, rational design dictates that the inherent damping forces, which are frequently erratic and subject to variations with ambient conditions, be minimized and separate shock absorbers with controllable characteristics be employed (Ref. 21). The shock absorbers are normally installed so that their line of motion is parallel to the motion resulting from the suspension springs.

In the interest of clarity, it must be stated that the term *shock absorbers,* as applied to the above function, is a misnomer; for it is the spring that really reduces the shock, and the shock absorber damps the spring vibrations. Spring dampers would be a more exact name for these devices, but shock absorber is in such wide use that attempts to change the name would only cause confusion (Ref. 7). The shock absorber, as used in modern vehicle design, does dissipate the energy of the impact as heat. This energy comes from the vehicle engine, and it is estimated that from one to as much as eight hp may be lost through this means (Refs. 21, 29).

The use of rubber grommets at suspension system joints and for "bottoming" bumpers, and the use of snubbers which abruptly limit the rebound excursion, in a sense contribute to absorbing the shock. However, these devices are used primarily to prevent metal-to-metal impact, and are not treated as shock absorbers *per se.*

11-18 CLASSES OF SHOCK ABSORBERS

Shock absorbers are of two general classes; single-acting and double-acting. Those which check only spring rebound are termed *single-acting.* They are so designed, or attached to the suspension system in such a manner, that the damping force is not generated during spring deflection. Their main disadvantage is that they provide damping only part of the time, thereby imposing the requirement of stiffer springs in the system. Also, a slight preload on the suspension spring is experienced due to the shock absorber return spring. This has a tendency to disproportionately stiffen the suspension spring for mild terrain irregularities (Ref. 21).

Those shock absorbers that provide damping during spring deflection as well as during rebound are termed *double-acting.* They permit the use of softer suspension springs and allow optimum damping in both directions. In most cases, the damping force developed by the double-acting shock absorber during spring compression is much lower than is developed during rebound. This is highly desirable, because a high damping force during spring compression would have the same effect on impact isolation as would a very stiff spring; i.e., it would transmit the shock to the vehicle body, causing it to displace vertically. It is desirable not to impede the impact-isolating properties of the spring during its compression stroke, because there is no upper limit to the amount of acceleration that the vehicle body can experience when the ground-contacting elements pass over a vertical obstacle at high speed. For this reason, the larger damping force is exerted during rebound; i.e., when the spring is being extended from the position to which it was compressed by the vertical impact.

There is some disadvantage in this arrangement when the moving vehicle encounters a sudden depression in its path. The action of the spring elements, in this instance, is to extend so that the ground-contacting elements can maintain good contact with the ground. The high damping force exerted by the shock absorber impedes this action, resulting in a downward acceleration of the vehicle body (Ref. 2). This is an acceptable condition, however, because the downward acceleration of the sprung mass cannot exceed the acceleration of gravity since no other downward force is acting upon the sprung mass (Ref. 21).

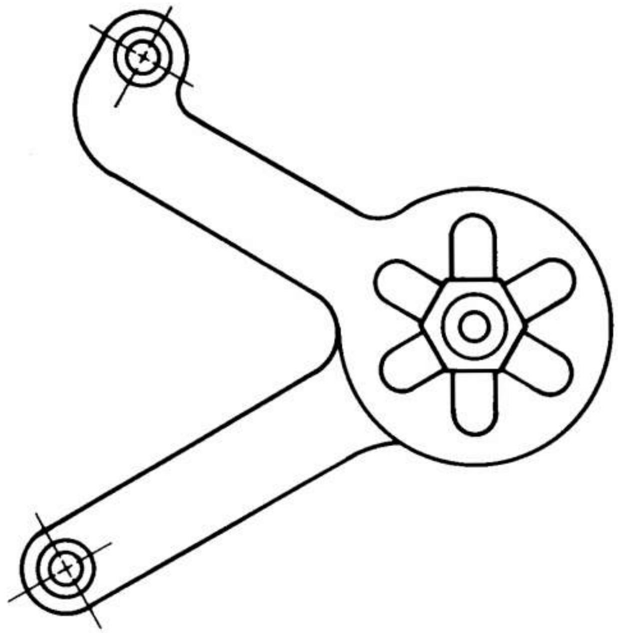

Figure 11-13. *Typical Mechanical Shock Absorber*

11-19 TYPES OF SHOCK ABSORBERS

11-19.1 GENERAL

Shock absorbers may be divided into two major types: (a) those that depend upon mechanical friction to generate a resistance, and (b) those that employ hydraulic fluid to create a damping force. A third general type, termed *dynamic absorber,* does not depend upon friction and is described separately in par. 11-19.4.

11-19.2 MECHANICAL SHOCK ABSORBERS

The damping force is generated, in a mechanical-type shock absorber, by the interaction of dry surfaces under load. The energy of the induced impulses is converted to heat by mechanical friction. A typical mechanical shock absorber is shown in Fig. 11-13. This design consists of a pair of arms, enlarged at one end into disks, and pivoted together at that end. The other ends are hinged, one to the vehicle frame, and the other to the unsprung mass. A disk of brake lining-like material is clamped between the pivoted ends. A clamping force is applied to the friction surfaces through a spider spring so as to minimize the reduction of damping force as the friction material wears. The characteristic force-velocity curve of a mechanical shock absorber is shown in Fig. 11-14. As indicated by the curve, the damping force is somewhat

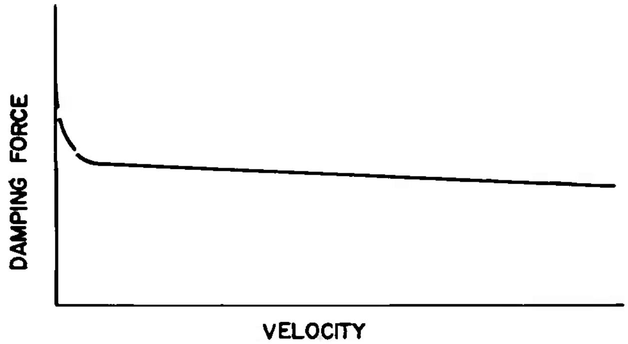

Figure 11-14. *Characteristic Curve of Simple Mechanical Shock Absorber*

independent of the velocity, once sliding begins. For most conditions experienced, there is a direct relationship between the amplitude of deflection, shock input, and relative velocity of sprung to unsprung components. It is possible, then, to design a mechanical shock absorber that will produce the desired damping for a given set of conditions. However, it is difficult to obtain the proper degree of damping for all conditions which the vehicle will experience (Refs. 4, 21). A mechanical shock absorber providing adequate damping at extreme deflections would be too harsh for average deflections.

An attempt has been made, in one design of a mechanical shock absorber, to overcome this limitation by producing a frictional force that is approximately proportional to the body displacement from the mean. This is accomplished by providing a camming action to increase the normal force on the friction elements as the suspension system travels from the midpoint (Ref. 30).

Another disadvantage of the mechanical shock absorber stems from the fact that static friction is substantially greater than sliding friction. This phenomenon contributes to a harsh ride, especially when operating in a terrain containing only moderate irregularities and with a vehicle equipped with double-acting shock absorbers. The shock must exceed the breakaway frictional force before the suspension spring will deflect. As soon as relative motion begins, the vehicle body experiences a vertical lurch due to the sudden decrease in resistance force.

Although friction damping characteristics are not suited to meet the full range of damping requirements, a small amount of friction damping is desirable to eliminate wheel dance so as to reduce the secondary vibrations induced into the spring mass. Friction, by itself, tends to induce secondary vibrations; but the small amount of friction needed to damp wheel dance causes less disturbance than results from undamped wheel dance (Ref. 4).

Mechanical shock absorbers were widely used in past years because manufacturing difficulties and lack of reliability of other types made them unpopular. Now they are seldom used because of the superior characteristics of the hydraulic shock absorber. A recent application of a mechanical shock absorber in a medium tank has proven successful (Ref. 10). This shock absorber derives its damping force from the friction of a brake lining-type material pressed against the inside surface of a steel tube. Although the ride quality over the entire speed range of the tank was inferior to that of similar tanks equipped with other types of shock absorbers, the increased durability of this new type warranted its selection.

11-19.3 HYDRAULIC SHOCK ABSORBERS

11-19.3.1 General

Hydraulic shock absorbers develop their damping force by one or more of the following characteristics of hydraulic fluids (Ref. 21):

a. *Viscous damping.* Characterized by a resisting force linearly proportional to the velocity. In practice it is found where there is relative motion between two well-lubricated surfaces, and where a viscous fluid is forced through a relatively long passage of small cross sectional area.

b. *Degenerate viscous damping.* Characterized by a resistance to motion that is proportional to a power of the velocity less than unity.

c. *Hydraulic damping.* Characterized by a resisting force proportional to the square of the velocity. This relationship is attained by forcing fluid of low viscosity through a sharp-edged orifice. Most hydraulic shock absorbers in use today operate on this principle.

Figure 11-15 shows the velocity-force relationship of the above hydraulic characteristics.

11-19.3.2 Description of Typical Hydraulic Shock Absorbers

11-19.3.2.1 Single-Acting, Cam-Operated Hydraulic Shock Absorber (Ref. 5)

A typical single-acting, cam-operated shock absorber is shown in Fig. 11-16. When the sprung

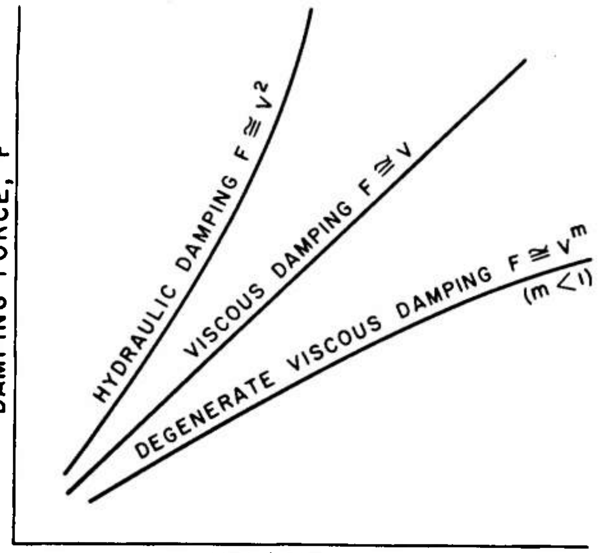

Figure 11-15. *Characteristic Curves of Hydraulic Shock Absorbers*

and unsprung masses of the suspension system move toward each other, the shock absorber arm rotates counterclockwise, moving the cam to the right, thereby permitting the piston spring to move the piston to the right. This causes the intake valve in the piston to open, allowing oil to flow from the reservoir into the increasing cylinder volume. Because the piston motion and oil flow are caused by the piston spring, the shock absorber has little effect upon the spring action of the vehicle. During rebound, the cam moves to the left, forcing the piston to the left against the oil in the cylinder. The intake valve closes, and the oil in the cylinder is forced out through the relief valve. The restricted passage of the oil through the relief valve orifice is the primary factor in generating the damping force during rebound.

11-19.3.2.2 Opposed-Cylinder, Double-Acting Shock Absorber (Ref. 5)

One type of double-acting hydraulic shock absorber is shown in Fig. 11-17. The two opposing pistons shown are rigidly inter-connected by a bracket at their reservoir ends, in which the cam rests. Each piston is equipped with a spring-loaded intake valve which permits the one-way flow of oil from the reservoir to the cylinder. When the suspension spring is compressed, the cam rotates counterclockwise, carrying both pistons to the right, thereby exerting a pressure on the oil in the compression cylinder. When the pressure is great enough, it forces the compression relief valve open, and oil flows through the valve into the rebound cylinder. At the same time, the intake valve in the rebound piston may open to permit flow of makeup oil from the reservoir into the rebound cylinder.

During the suspension spring rebound, both pistons are forced to the left by the cam and the oil is forced through the rebound relief valve, back to the compression cylinder. The resistance to flow of the orifices in the relief valves determines the spring damping force. The orifice in the rebound relief valve is much smaller than the compression relief valve orifice.

Some double-acting hydraulic shock absorbers incorporate a low pressure relief valve in series with the compression relief valve in an attempt to simulate the light friction damping desirable for suppressing wheel dance. However, the effect is inferior to friction damping because of the slower response to high-frequency reversals (Ref. 4).

11-19.3.2.3 Vane-Type Shock Absorber (Ref. 5)

The housing of the vane-type shock absorber, shown in Fig. 11-18, is divided into two working chambers by stationary partitions, each of which contains a check valve. The central shaft is connected to the unsprung mass through the arm and link, and has a pair of vanes attached to it which extend into each working chamber. As the suspension spring is compressed, the central shaft rotates, and the vanes develop a pressure in the chamber which causes oil to flow, unrestricted, through the opened check valves in the stationary partitions. On the rebound stroke, the vanes develop a pressure on the opposite side, closing the check valves. Since oil cannot flow through the check valves, it is forced through the needle valve in the center of the shaft, thereby producing a resistance to motion of the arm. The vane-type shock absorber is not widely used, primarily because of the high cost and poor reliability resulting from the difficulty in sealing this type.

11-19.3.2.4 Direct-Acting Shock Absorber (Refs. 5, 7)

A typical, direct-acting shock absorber is shown in Fig. 11-19. It consists of three concentric cylinders. The innermost of these contains a double-acting piston which divides the cylinder into upper and lower chambers. The annular volume between

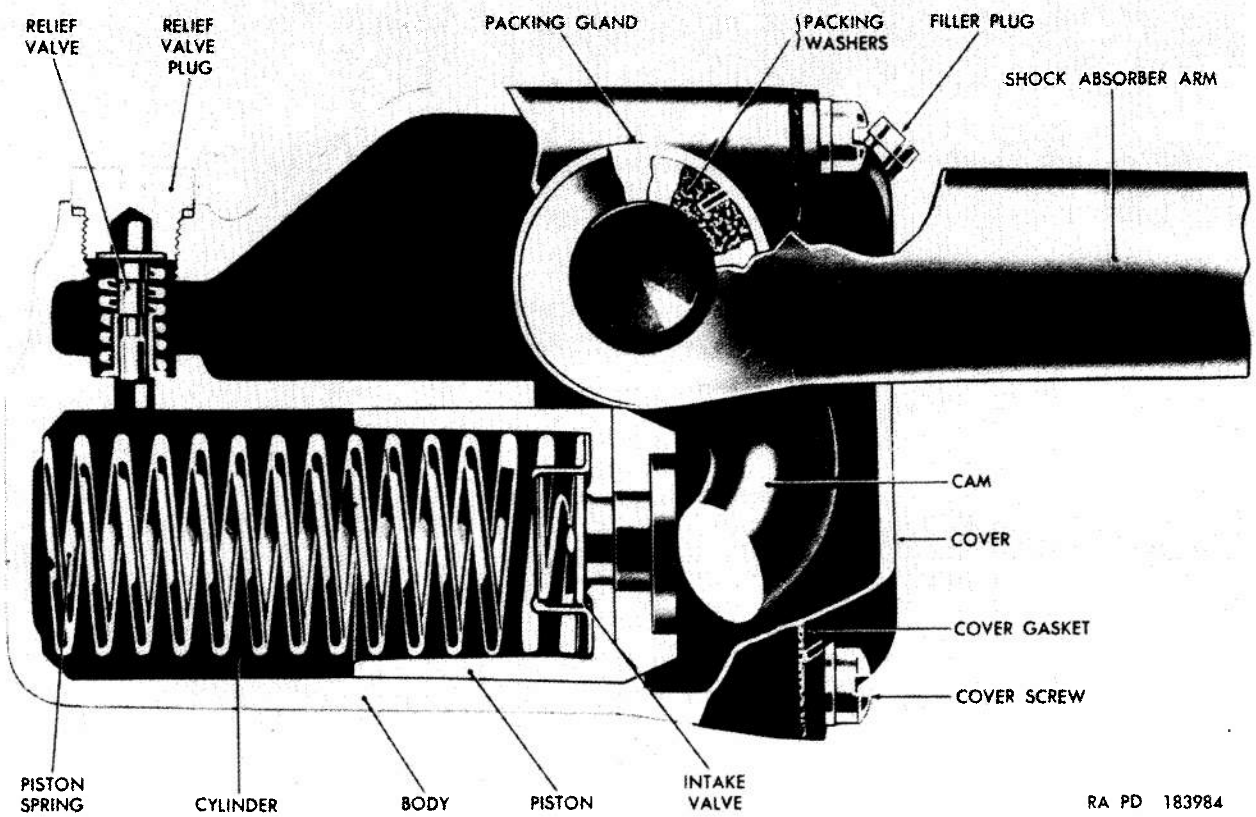

Figure 11-16. *Single-Acting Cam-Operated Piston Shock Absorber*

the first and second cylinders serves as a reservoir. The outer cylinder is a dust shield. Built into the pistons are compression and rebound orifices and check valves.

During compression of the suspension spring, the piston, usually connected to the sprung mass, moves downward, forcing oil through the compression orifices in the piston to the rod end of the cylinder. Because of the volume occupied by the piston rod, the volume change above the piston is not as rapid as that below; therefore, some of the oil is forced into the reservoir through the reservoir check valve. On the rebound stroke, oil is forced from the rod end to the piston end of the cylinder through the rebound orifice producing the damping force. Some oil flows from the reservoir to the lower chamber through the reservoir check valve to compensate for the differential volume change caused by the piston rod.

The configuration of the direct-acting shock absorber lends itself well to vehicular installation. The arm, link and bearings, associated with the other types, are eliminated. They are frequently installed obliquely to the vertical to achieve some sideways and roll control.

Because of the greater piston travel of direct-acting shock absorbers over vane- and cam-operated types for a given damping, their operating pressure is lower, thereby improving reliability by relieving the sealing problem. The pressures in the direct-acting types are about 400 psi, compared with 1500 psi for the other types (Ref. 21). The direct-acting shock absorber experiences a greater temperature rise than the other hydraulic types. This is because the bodies of the cam-operated and vane-types are in intimate contact with substantial structural members which improves their heat dissipation rates, whereas, the direct-acting type is attached to the vehicle by insulating rubber grommets and depends almost entirely upon the surrounding air for its heat dissipation (Ref. 21).

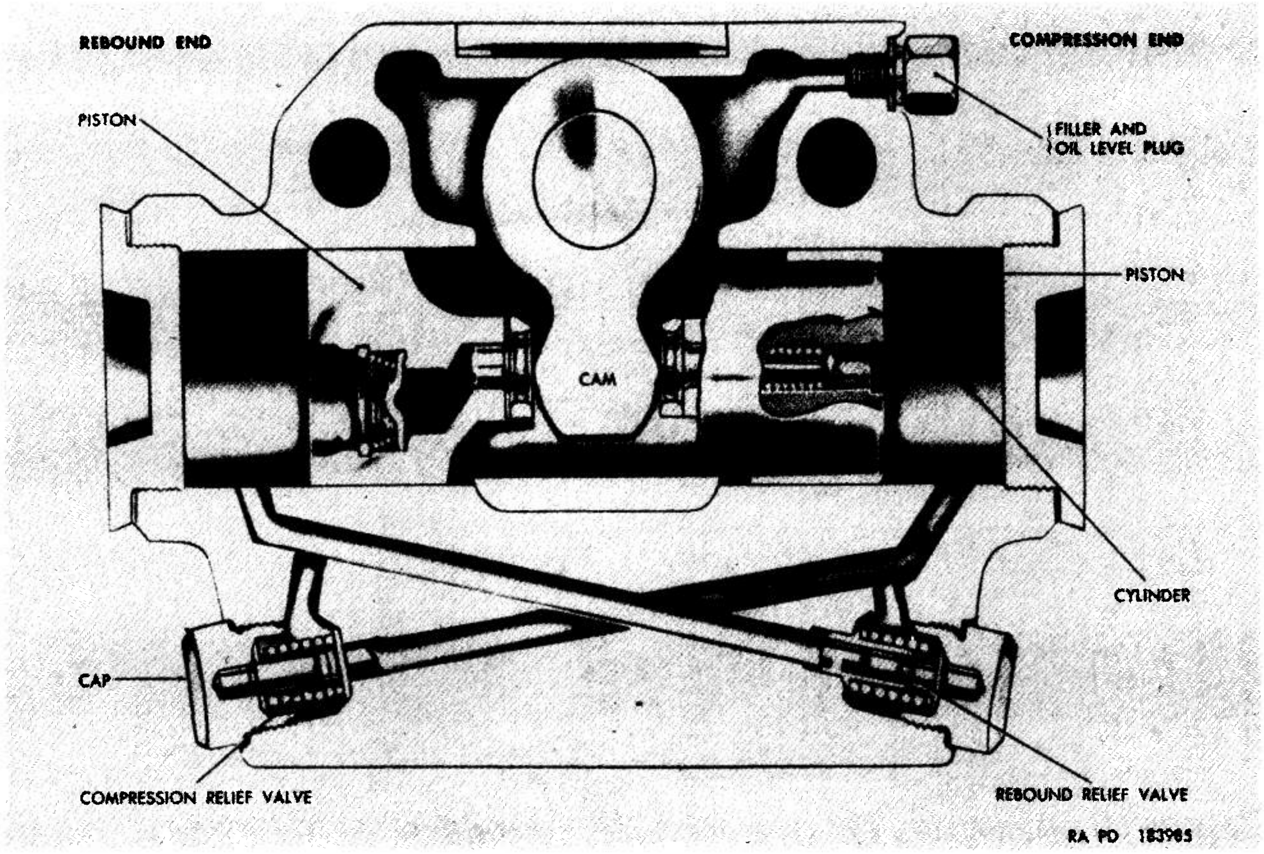

Figure 11-17. Double-Acting Cam-Operated Piston Shock Absorber (Opposed-Cylinder Type)

11-19.3.2.5 Inertia and Frequency-Controlled Shock Absorber

Except for the suppression of wheel dance, damping of the suspension spring is undesirable when the sprung mass does not experience acceleration. For this reason, it is desirable to vary the degree of damping of a particular suspension system to suit the various conditions. An attempt in this direction has been made by incorporating an inertia-sensitive valve into the hydraulic shock absorber body, in parallel with the rebound relief valve. This valve is normally open for average shock conditions. It is closed by a spring-suspended weight, when the vehicle body experiences a substantial acceleration during the rebound. This action automatically increases the damping momentarily (Refs. 4, 5, 21, 31). However, this feature is not widely employed because the added complexity of the shock absorber adversely affects its reliability and cost.

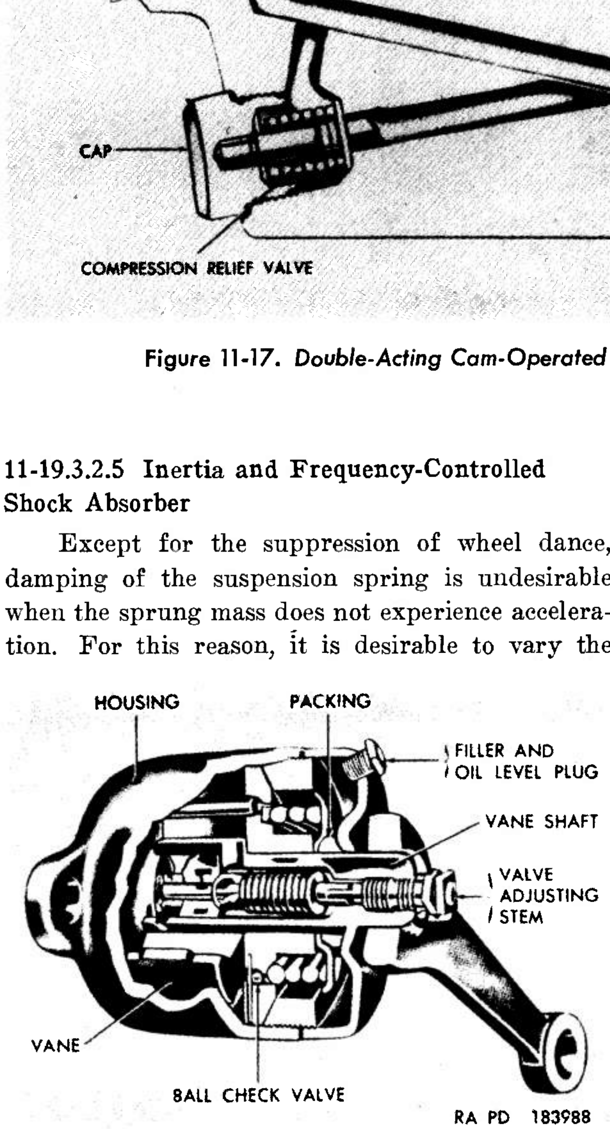

Figure 11-18. Double-Acting Vane-Type Shock Absorber

Another approach toward improving the damping depends upon the fact that both the free vibrations and the resonant vibrations occur at the same frequency. By incorporating a damping valve that is sensitive to the natural frequency of the system, the ideal is approached (Ref. 21). However, this idea has not yet been implemented by a practical design.

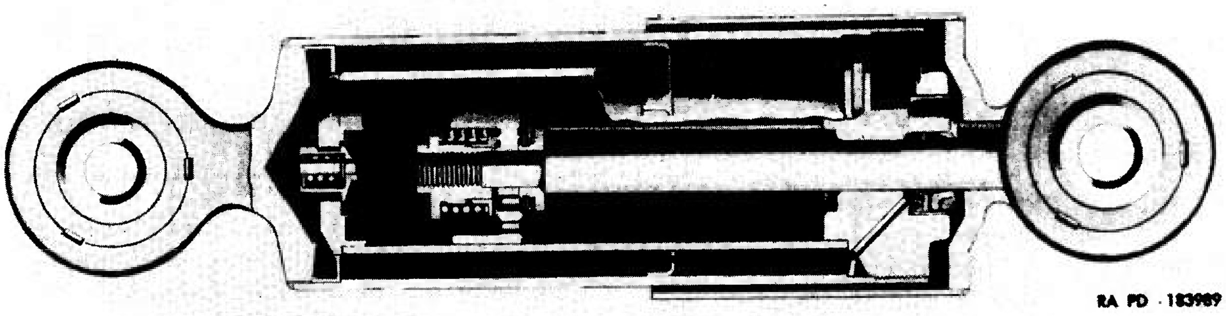

Figure 11-19. *Direct-Acting Shock Absorber*

11-19.4 DYNAMIC ABSORBERS (Ref. 29)

The shock absorbers discussed in the preceding paragraphs supply a damping force to the suspension spring by means of a frictional or hydraulic device. Another device for supplying a force to modify the suspension spring characteristics is the dynamic absorber. The dynamic absorber is a small spring-mass system mounted on the unsprung mass. It is tuned in such a manner that it vibrates in antiphase to, and therefore opposes, an exciting force of a specific frequency. The frequencies at which the absorber must be effective are: (1) the natural frequency of the sprung mass on its spring (pitch and bounce), and (2) the natural frequency of the wheel on its spring system (wheel dance). The small spring mass system usually has its own small frictional damper.

The advantages claimed for the dynamic absorbers are simplicity, isolation of the absorber from the sprung mass, and reduction of conventional damping which is power-consuming and subject to failure by overheating. The dynamic absorber does, however, increase the unsprung mass, thereby offsetting to some degree the reduction of high frequency vibrations gained by isolation.

The application of the system has been very limited. On the one vehicle that has used dynamic absorbers, a noticeable low-frequency bounce resulted. However, wheel dance was completely eliminated.

11-20 SELECTION AND DESIGN OF SHOCK ABSORBERS

The approach to the suspension spring damping problem has been developed more by experimental than theoretical means (Ref. 21). The assumptions made when selecting the design parameters for the suspension spring itself are so many and varied that they preclude the accurate prediction of the damping requirements. Given a vehicle of particular size and type, the shock absorber manufacturer may supply a group of shock absorbers that have proven successful on previously developed vehicles. After experimental evaluation on the new vehicle, adjustments and modifications can be made to best suit the requirements of the vehicle under consideration. However, the experimental approach should be guided by several simple principles resulting from experience. These principles are summarized below:

1. Techniques should be employed which reduce the natural damping of the suspension system to a minimum.
2. The vertical accelerations and changes in acceleration of the vehicle body relative to the ground are the factors that influence ride quality.
3. Damping is essential when operating near resonant frequencies and for eliminating free vibrations. It is objectionable for all other conditions.
4. Viscous damping tends to amplify terrain irregularities in proportion to their severity.
5. Hydraulic damping tends to amplify mild irregularities to a lesser extent than does viscous damping and to amplify severe irregularities to a greater extent.
6. Vehicles with relatively soft springs require less damping in terms of energy absorbed than those with hard springs; but damping is more essential to maintain the stability of the soft sprung vehicle.
7. A small amount of friction damping is desirable to eliminate wheel dance.

SECTION VI WHEELS AND TIRES

11-21 GENERAL DISCUSSION

11-21.1 FUNCTION

The primary function of wheels on military vehicles is to minimize resistance to vehicle motion while supporting the vehicles on the terrain. Although wheels possess certain shortcomings and present many problems to the military vehicle, they are the lightest, simplest, and most extensively employed method of accomplishing the above function.

In addition to the primary function, wheels and tires are required to accomplish the following secondary functions:

1. Transmit the driving and braking torques between the vehicle and the terrain
2. Provide a means of steering the vehicle
3. Reduce road shocks to the balance of the suspension system
4. Provide traction for the vehicle on all types of terrain
5. Absorb terrain irregularities
6. Provide lateral stability for the vehicle

11-21.2 DEVELOPMENT

Military wheels and tires of the past have generally been modifications of those that had been developed for commercial vehicles whose requirements are not as severe. The current trend is to design more specifically to meet the military requirements while still taking advantage of commerical developments and production capacity. Even here, the designer must consider the supply and maintenance problem, particularity in the forward area, and attempt to utilize standard wheels and tires wherever possible without degrading his design (see Table 11-4).

In recent years, the demands of the heavy construction industry for improved cross country locomotion have scored some notable achievements in wheeled vehicles. These vehicles, however, are too wide for general military use, and, except for the Goer concept, have not been adopted by the Army.

11-21.3 DESIGN REQUIREMENTS

While the selection of the wheel and tire is greatly influenced by traction and floatation requirements, the designer must also consider the compatibility to the general vehicle configuration and function. In general, a larger diameter wheel will provide greater mobility and better ride quality than a smaller diameter wheel (Ref. 3). Also, the greatest mobility is achieved when the vehicle is equipped with wheels of one diameter and one width and with the wheels equally loaded (Ref. 3). Dual tires, while providing greater floatation in soft terrain, offer greater resistance to motion than single tires of a larger diameter. Therefore, their use is discouraged for the military vehicle.

TABLE 11-4 REPRESENTATIVE SIZES OF TIRES IN CURRENT USE

Tire Size	Representative Vehicles
6.00-16	1/4-T, 4×4 (Models MB, GPW, C3-3A)
7.00-16	1/4-T, 4×4, M38
9.00-16	3/4-T, 4×4, M37 and M42
7.50-20	WW II, 1-1/2-T Vehicles
9.00-20	3/4-T, T53
	2-1/2-T, 6×6, M35
11.00-20	2-1/2-T, 6×6, M34 and M135
	5-T, M40
14.00-20	5-T, M41
14.00-24	40-T Tractor, M25
	10-T, XM125
16.00-25	Transporter, T10
	15-T, XM194
21.00-29	Transporter, T8E1

11-22 WHEELS AND RIMS

11-22.1 CONVENTIONAL STEEL WHEELS (Refs. 5, 24)

Military vehicle wheels in general use are pressed steel disks. The wheels are dished to bring the point of ground contact directly under the larger or inboard wheel bearing, and to permit the mounting of dual wheels. Lighter vehicles employ wheels with integral rims. Since integral rim wheels cannot be disassembled, the diameter between the flanges is reduced (drop center) to permit the mounting of the tire. The rims of larger wheels are usually permanently fastened to the wheel. To permit tire mounting, one flange is re-

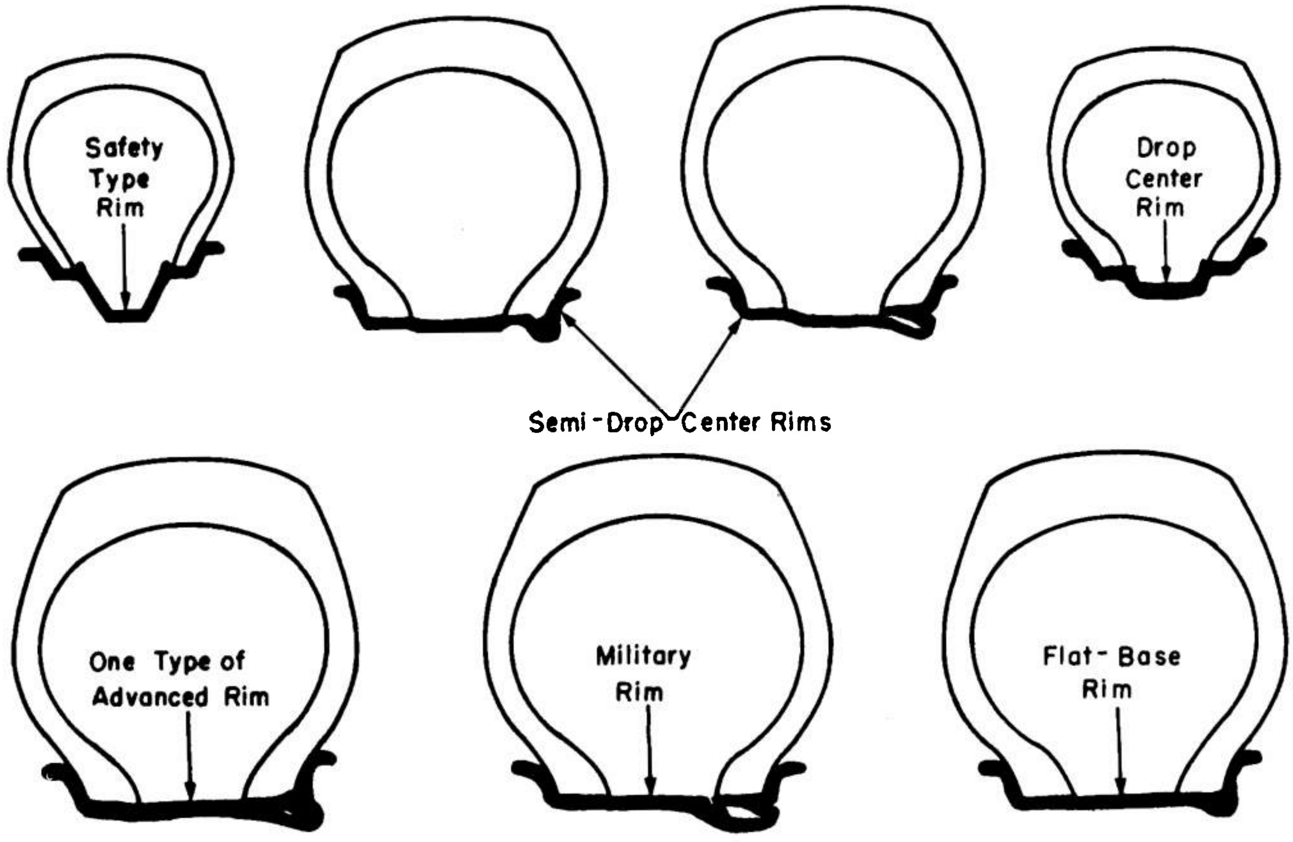

Figure 11-20. *Types of Rims*

movable. It is held in place by engaging a circumferential groove on the rim. Sizes and shapes of military rims have been standardized by the Ordnance Advisory Committee, working with the Manufacturers Tire and Rim Association. Some of the standard types are shown in Fig. 11-20.

11-22.2 COMBAT WHEELS (Ref. 5)

Wheels for the severe service encountered under tactical conditions are of the divided rim type (see Fig. 11-21). In this type, the two parts of the wheel, each incorporating part of the rim, are held together by bolts or studs. A detailed discussion of wheels for military vehicles can be found in Ref. 5.

11-22.3 LIGHTWEIGHT WHEELS

Although the majority of the wheels in current military service are of steel disk construction, the demand for lightweight vehicles has led to the development of wheels made of lightweight metals. Aluminum wheels are employed on some of the newest military vehicles at a considerable reduction of weight. Preliminary testing of magnesium wheels has yielded promising results (Refs. 32, 33). Fiberglas reinforced epoxy and polyester wheels have been proposed to reduce further the weight of the wheel (Ref. 34). This reduction of wheel weight is desirable, not only to lessen the total vehicle weight, but also because the smaller unsprung mass generally results in improved mobility. Wire wheels are light and resilient, but they do not equal the weight-strength ratio of aluminum disk wheels (Ref. 24). Furthermore, they are more vulnerable to damage by terrain debris and flying objects than are disk wheels.

11-22.4 UNUSUAL WHEELS

In order to achieve some function normally provided by other parts of the suspension system, unique wheel designs have been tried. In one such design, two rubber disks replaced part of the metal wheel disk. This was done in an attempt to improve the ride quality of a vehicle that has no separate suspension springs and depends solely upon the tires to absorb the road shock. While this

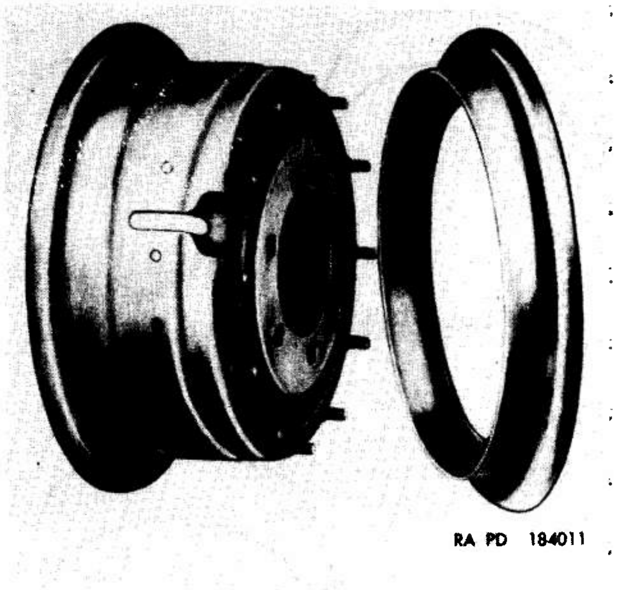

Figure 11-21. *Divided-Type Rim*

design did permit slightly higher speeds over rough terrain, lateral stability, steering, and durability were poor (Ref. 35). Another unusual wheel was the Martin wheel. It consisted of rubber spokes connected to a hickory-reinforced nonpneumatic rubber tire. The purpose was to eliminate some of the shortcomings and vulnerability of the pneumatic tire. This wheel, while comparing favorably with a pneumatic tire wheel in some respects, proved totally inadequate to cope with severe road impacts. The lateral stability, soft-soil performance, and curb-climbing ability were inferior to the pneumatic-tired wheel (Ref. 36).

11-23 TIRES

11-23.1 GENERAL

Present military wheeled vehicles are equipped almost exclusively with pneumatic tires. Metal or solid rubber tires, while extensively used on track suspensions, are seldom employed on wheeled vehicles (Ref. 24).

11-23.1.1 Tire Standards

Tire manufacturers have standardized tire sizes and types. The Ordnance Committee of the Tire and Rim Association has established additional standards for military tires. These standards are given in MS-35388 through MS-35392 (Ref. 39). The ply rating given in these tables is an index of the tire strength and does not necessarily represent the number of cord plies in the tire. A cross section of a typical pneumatic tire, giving the nomenclature of the various tire elements, is shown in Fig. 11-22.

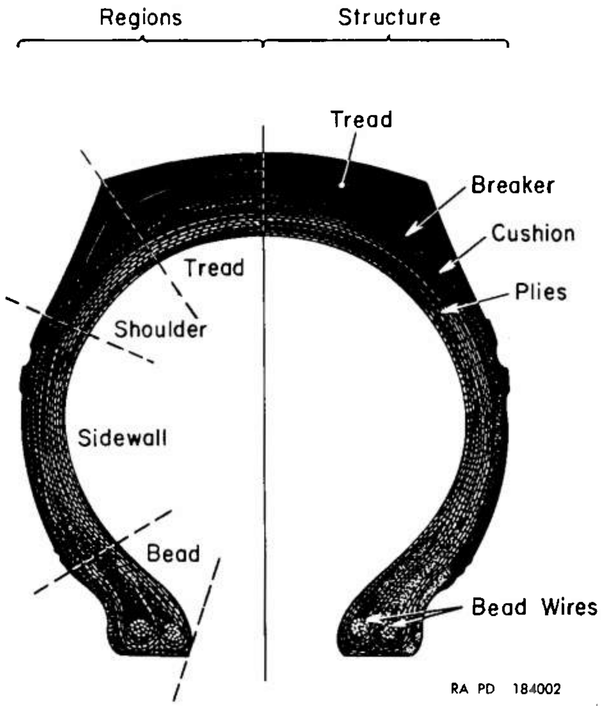

Figure 11-22. *Pneumatic Tire, Cross Sectional View*

11-23.2 PNEUMATIC TIRE CHARACTERISTICS

11-23.2.1 Advantages

The pneumatic tire is so widely used because it improves vehicle mobility by (1) reducing shocks and vibrations of the wheel, thereby permitting higher speeds, and (2) improving floatation and traction on soft soils, thereby extending cross country capabilities (Ref. 24). It is the dominating elastic factor in producing wheel dance, and its characteristics must be considered when damping the suspension system (Ref. 4). The pneumatic tire serves to some degree as an energy absorber, converting some of the impact energy to heat by the hysteresis of the rubber.

The improved floatation and traction characteristics of the pneumatic tire are derived from the substantial ground contact area developed under the deformed tire. The principles governing the wheel-soil interaction are not fully understood. Theories and empirical relationships have been established to aid the vehicle designer in providing optimum cross country performance. These are discussed in Chapter 5 and in Ref. 37.

11-23.2.2 Tire Selection Criteria (Ref. 24)

Military vehicles are generally designed for conservative loading, with the expectation that they will be extremely overloaded and abused. The tire must have the resiliency to withstand repeated shocks and overloads. Impact values as high as 8 G have been measured. A pneumatic tire will crush flat on the rim at 3 G. Resistance to wear, as encountered by commercial vehicles, is not as important for the military tire as are resistance to sharp stones, terrain abrasion, temperature extremes, shrapnel, bullets, and fire.

11-23.2.3 Abnormal Inflation

In order to function under overload conditions, tires are frequently inflated to above normal pressures. To gain greater floatation and traction, tires are sometimes operated at below-normal pressures. The variation of tire pressure to achieve optimum performance is sometimes facilitated by the use of a central inflation control system. This system permits the operator to change tire pressure while the vehicle is moving, thereby improving mobility. To be reliable, the air lines and control valves must be protected against damage by enemy action, rough terrain, and objects thrown up by the wheels. Devices to prevent the loss of air from the entire system, in the event of failure of one tire, must also be included (Refs. 24, 38).

11-23.2.4 Beadlocks (Ref. 5)

The tire is normally held tight against the rim flange by the air pressure. The friction at the mating surface is sufficient to transmit the driving and braking torque at normal inflation pressures. However, the under-inflated or deflated tire will slip or collapse when these loads are applied. To permit the emergency operation of under-inflated and deflated tires, beadlocks are employed. A beadlock is a ring, of channel cross section, that fits inside the tire, between the beads. It is slightly wider than the space between the beads so that a compression fit is obtained when the rim is assembled (Fig. 11-23). This clamping action, plus the heavy wall of the military tire, will support the vehicle load for a short time, thereby, permitting the vehicle to continue its mission. Although

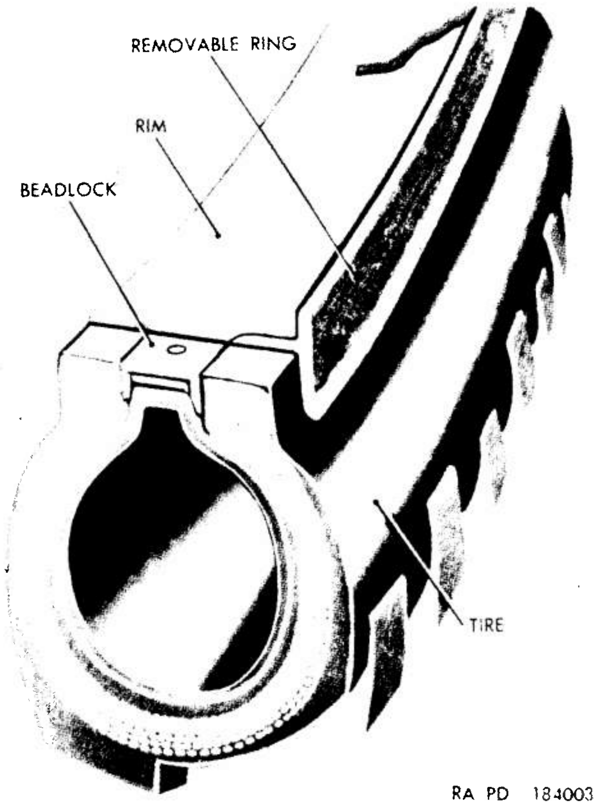

Figure 11-23. *Tire Installation with Beadlock*

the stiff wall impairs the impact-absorbing capabilities of the tire, the demands of tactical conditions warrant this design.

11-23.2.5 Tire Tread (Refs. 24, 41, 42)

To provide traction in plastic soils, military tire treads are designed with lugs or grousers. To be effective, the space between the grousers must remain clean, for once the tire becomes packed with mud or snow, the tractive ability is no greater than that of a smooth tire. Tread design has undergone considerable study and development, and experts do not agree as to the optimum design. Several typical tread designs are shown in Fig. 11-24. The heavy treads and grousers are essential for military tires, and they are used at the expense of the impact-absorbing capabilities of the tire.

11-23.2.6 Very Low Pressure Tires

Very low pressure tires, called Rolligons or terra tires, have been recently introduced for use on military vehicles to improve traction, floatation and bump envelopment. These tires have a high width-to-diameter ratio and operate at pressures

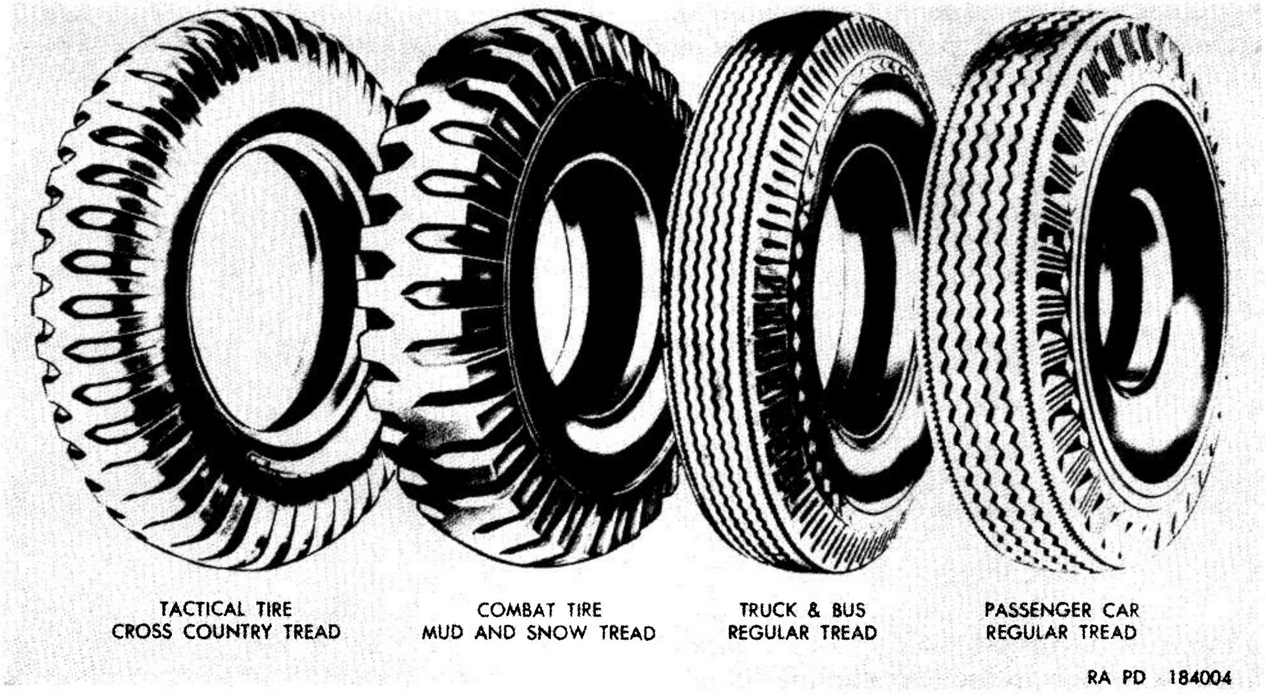

Figure 11-24. *Typical Military Tire Treads*

from 3 to 16 psi. The impact-absorbing capacity of these tires is great enough to eliminate separate suspension springs (Ref. 12). Vehicles employing these tires have not been in service long enough to fully evaluate the tire performance. Early operation of these vehicles has demonstrated a degree of improved off-the-road locomobility while permitting reasonable highway speeds. Their performance over rock-strewn terrain has been particularly successful, because the small terrain irregularities are "swallowed" by the tire, thereby transmitting but little shock to the vehicle. Although the very low pressure, wide tire has experienced some success, its configuration presents serious design disadvantages. The suspension and drive systems incorporating this tire approach the track-laying suspension in bulk and complexity, resulting in a vehicle substantially wider than that desirable for a wheeled military vehicle while not possessing the cross country mobility of a tracked vehicle (see par. 4-18.2 and Fig. 4-47).

Land locomotion studies indicate that a narrow, low-pressure tire is superior to a wide tire of the same diameter (Ref. 42). An application of tires in this category is found in the Goer family of military vehicles. These vehicles employ tires of the type used on earth-moving equipment. Their performance is better than that of the smaller tired vehicle, but is not as good as that of tracked vehicles. The tire size and configuration again result in a large vehicle not suited to all military operations (see par. 4-18.4).

11-23.2.7 Unusual Tires (Ref. 24)

Because the pneumatic tire is vulnerable to puncture by terrain conditions and enemy action, there exists a demand for a nondeflatable tire that will equal the pneumatic tire for impact-absorption and floatation. Many concepts, including solid rubber tires, elastic-spoked wheels, cellular elastomer tires, and wire bristle tires, have been proposed. None of these concepts has been successful. Modifications of the pneumatic tire, such as bulletproof tires, self-sealing tubes, slow-deflating tires, tubeless tires, and safety rims are used to some degree on combat vehicles. The greatest progress in tire performance has been made by employing improved elastomers and fibers. The goal in military tire design is to obtain a tire that will be (1) resilient and have good riding qualities; (2) have good traction; (3) remain on the rim when deflated; (4) run flat on the rim in emergencies for a substantial distance; and (5) be inexpensive.

11-24 TUBES AND FLAPS (Refs. 5, 41)

11-24.1 DESCRIPTION AND FUNCTION

The tube is a hollow torus employed to retain the air within the tire. Current tubes are made of butyl rubber, which is superior to natural rubber for this purpose.

11-24.2 TYPES OF TUBES

The three types of tubes in general use on military vehicles are:

1. Standard tubes, of single-layer rubber construction, for use with standard-type tires.
2. Combat tubes, of single-layer rubber construction, made smaller than standard tubes to fit inside the smaller air space of combat tires.
3. Bullet-resisting tubes, of heavy laminated construction, which automatically seal bullet punctures.

11-24.3 FLAPS

The tube is soft and pliable and, therefore, subject to damage by chafing, pinching, and puncture. To minimize the chafing and pinching of the tube against the rim or beadlock, a continuous strip of rubber, called a flap, is installed about the circumference of the rim.

SECTION VI TRACKS

11-25 INTRODUCTION

Much emphasis has been placed on the indispensability of the flexible suspension for high-speed tracked vehicle performance. However, it is the track itself that distinguishes the tracked vehicle and provides for its superior cross country performance. The concept of the tracked vehicle goes back to the 1770's and its historical development is well-covered in many texts and articles (Refs. 43, 44, 45). The concept is simple. Lay down a roadway, travel over it, pick it up again. The requirements for directional control, weight distribution, lateral stability, adaptability to various terrains, high speeds, reduction of weight, power losses, vibration, etc., complicate the problem for the track designer. So varied are the terrain and speed conditions under which tracked vehicles are expected to perform that generally separate design requirements are established for particular types of vehicles. Even within the same type of vehicles, the conflicting demands imposed by the various functions that the track must perform, often result in design compromises that may only partially fulfill some of the functions. The extent to which a particular function is fulfilled may depend to a large degree upon factors over which the track designer has no control, such as characteristics of the suspension system, the number and size of the road wheels, and the contour of the track. These factors are determined by the performance requirements of the vehicle, which, in turn, largely govern the type of track selected (Ref. 44).

11-26 TRACK FUNCTIONS (Refs. 44, 46, 47)

11-26.1 GENERAL

In adapting the track concept to practical vehicles, the track is called upon to provide some, or all, of the following functions:

a. Provide multiwheel drive from a single axle to ensure that the entire area of the ground supporting the vehicle is utilized to obtain tractive effort.

b. Increase the area of ground which is used in providing tractive effort.

c. Distribute the ground pressure over as large an area as possible to minimize sinkage.

d. Provide the equivalent of a ramp over holes to prevent individual wheels from dropping into them.

e. Produce a smooth path for road wheels.

f. Provide limited water propulsion.

In many modern vehicles, other requirements, particularly the need for speed and a spring suspension, have made compromises necessary. In the paragraphs below, the extent to which the various types of tracks meet the above functional requirements are discussed.

11-26.2 MULTIWHEEL DRIVE

The amount of tractive effort which can be exerted by a single wheel is limited by the propulsive force that can be developed by the action of the wheel against the soil. By increasing the number of wheels by which the tractive effort is transmitted, slippage is less likely. In the case of the tracked vehicle, the track transmits the drive from the final drive unit to all the wheels so that the entire weight of the vehicle contributes to the frictional force to make up the tractive effort.

The use of a track eliminates the complication involved in transmitting the drive to a wheel which is not only sprung but also steered.

11-26.3 IMPROVED ADHESION

The maximum tractive effort for adhesion which can be provided by a soil is dependent upon the load applied to it, as well as the area of the soil undergoing such loading. Therefore, more tractive effort is provided when the area, over which the adhesive forces act, is increased.

11-26.4 REDUCTION OF SINKAGE

The area over which the vehicle track makes contact with the ground is larger than the corresponding area between the road wheels and the ground. The provision of a track, however, does not ensure that the ground pressure at any point in contact with the ground has a constant value equal to the mean ground pressure.

In the case of a very flexible track, when traveling over hard ground, the only point of contact is immediately below the road wheel. On very soft ground, where the wheels sink to an appreciable depth, the ground pressure will be uniformly distributed if the track tension is high. It can be shown, however, that over 80% of the track ground contact area supports very little weight.

With the rigid girder track, which forms a rigid bridge between neighboring wheels, very good pressure distribution can be achieved on reasonably soft ground with little sinkage. Rolling resistance is low.

The flexible pin-pointed track with a long pitch produces an effect which is between the continuously flexible track and the rigid girder track. This track has been adopted to reduce rolling resistance in soft ground.

Rolling resistance is dependent upon the maximum ground pressure; hence, such resistance will be at a minimum when the maximum ground pressure is equal to the mean ground pressure.

11-26.5 IMPROVED OBSTACLE CROSSING

Obstacle-crossing performance by a wheeled vehicle may be improved by increasing the number of load-carrying wheels or the number of drive wheels. Performance is further improved by the addition of a track which will bridge the spaces between the wheels, and by the provision of front and rear idlers which provide the equivalent of extended ramps. The maximum height obstacle which can be surmounted by a vehicle is governed as much by the length of the vehicle as by the height of the front sprocket.

11-26.6 THRUST FOR WATER PROPULSION

In recent years considerable emphasis has been placed on the need for amphibious fighting machines. Water screws and hydrojets for providing water propulsion not only complicate the design and increase the weight of the vehicle, but are also deficient during operation in debris-laden water. For amphibious vehicles it is generally agreed that the tracks should provide thrust for propulsion in water.

11-27 TRACK DESIGN

11-27.1 GENERAL

In equipping a vehicle with a mobile roadway (track) that supplies the functions listed in par. 11-26, the designer must devise a mechanical system that will (1) be compatible with the basic vehicle, (2) properly utilize the available power, and (3) provide for directional control and lateral stability of the vehicle. The structural features and design configurations of tracks take various forms. However, all tracks have many features in common. Some of these are:

a. A surface which rests on the ground and gives support.

b. A surface which engages the ground to give adhesion by friction or by digging-in.

c. A wheel path upon which the road wheels may run.

d. Guide faces to keep the wheels on the tracks.

e. Drive surfaces to take the drive from the sprocket.

f. A hinge or other means of flexible interconnection.

g. A water grouser or vane for increased propulsive thrust in water (for amphibians).

11-27.2 GROUND-ENGAGING SURFACES

The track area must be sufficient to support the weight of the vehicle on the type of terrain over which operation is expected. Mean ground pressure for most light tanks is about 8 psi (pressure of a man's foot). Heavy tanks usually have higher ground pressures. A mean ground pressure of 12.5 psi has been established as the maximum advisable for heavy-tracked vehicles (Refs. 17, 46). High-mobility vehicles such as the Weasel exert a pressure on the ground of about 1.9 psi (Ref. 48). Special-purpose vehicles for use on soft snow have ground pressures as low as ½ psi (Ref. 49). In determining the track width necessary to obtain sufficient ground contact area, the track length must be considered. A long, narrow track contact area is desirable for traction, but the longer the track, the more difficult the steering becomes. For this reason, the length of the track print is limited to from 1.1 to 1.8 times the center-to-center lateral distance between the tracks (Ref. 17).

A ground-engaging surface which is completely flat will provide a certain measure of adhesion. A link incorporating a flat surface has been used. Generally, however, the ground-engaging surface has various types of grooves or recesses formed in it to improve adhesion. Stamping is commonly used to form projections on the track surface.

Recesses in links are liable to become filled with earth; accordingly, the majority of track links are constructed with one or more spuds formed into them to improve adhesion. A common form of ground-engaging surface is one employing a single transverse spud and a series of ribs standing out from the adjacent under-surface. The space between the ribs may be closed. In some tracks, each link is provided with two spuds. Rubber pads are frequently mounted on the ground-engaging surface to reduce impact and improve traction on hard surfaces.

Recent developments for improved locomobility in snow and plastic soils have resulted in the spaced-link track. In this design the grouser size and spacing are arranged to take maximum advantage of the strength properties of the soil while producing a minimum resistance to forward motion.

11-27.3 WHEEL PATHS AND GUIDE SURFACES (Refs. 23, 46, 47, 50)

To permit high vehicle speeds, the track configuration must present a smooth roadway for the road wheels. The width and number of paths are consistent with the dimension, types and arrangement of the road wheels employed. For example, dual road wheels require two paths along the track. Many of the current designs include rubber surfaces on the road wheel side of the track. This not only reduces vibration and noise, but also increases tire and track life and resists the adhesion of mud and snow by freezing. The track-block pitch and road wheel spacing must be related in such a manner that suspension vibrations will not be induced by simultaneous engagement of all the road wheels with track blocks. This design feature is particularly important on the band track and spaced-link track where substantial deflections between track bars are possible.

Lateral track forces, induced by steering and side slope operations, are transmitted from the track to the vehicle hull through the road wheels and suspension mechanism. The track is equipped with projecting guides that bear against the rotating road wheels so that alignment of the track relative to the vehicle is maintained. The shape of the guides may vary, but it must be such as will resist climbing of the guide by the road wheels and minimize the damage to the road wheel tires. The height of the guides is influenced by the drive sprocket hub diameter, root diameter, and idler sprocket outside diameter. The guides should nest with clearance when the track wraps around the sprockets. Although increased guide height is desirable to maintain track alignment, the guides should not be pointed, because pointed guides inflict severe tire damage when misguiding and track derailments do occur.

Tracks for dual road wheels usually employ a single center guide. Its width at the base is determined by the section necessary to safely carry the guiding stress. The sides of the guide are sloped somewhat, the angle being determined by the factors establishing the guide height.

Tracks for single road wheel suspensions employ two guide horns whose side angles are usually about 10° from the vertical. Steeper slopes scuff the walls of pneumatic road wheel tires and shallower slopes provide a ramp that facilitate road wheel climbing and derailment.

The track guides should be long enough and spaced close enough so that all road wheels are in contact with at least one guide at all times. Two guides in contact are preferred. On block and pin tracks, the track guides are usually separate steel castings, bolted to the track block assembly. On band tracks, they are most frequently an integral part of the track bar. In designs where the track guide engages a metal portion of the road wheel, it is advisable to make the track guides of softer material in order to minimize the wear on the road wheels.

11-27.4 DRIVE SURFACE FOR SPROCKET (Ref. 44)

The form of the track driving surface is determined by the type of sprocket and whether a single- or twin-sprocket drive is utilized. In some cases, the bosses, or trunnions, which engage the sprocket teeth are formed only on the upper part of the track link; while in other cases, the sprocket teeth engage holes in the link. The boss, or trunnion, may surround one of the pins or be between pins. Commonly, one of the pin bosses constitutes the trunnion and the sprocket tooth extends through an adjacent gap.

Roller-type sprockets require teeth on the track links to enter between the sprocket rollers to effect the drive. Rubber-covered sprockets require that the track bar pitch be greater than the sprocket pitch to facilitate disengagement of the sprocket teeth and minimize reverse bending of the track (Ref. 50).

11-27.5 WATER PROPULSION VANE

The usual track, designed only for land propulsion, does not generally provide sufficient thrust to propel a vehicle in water. Accordingly, when tracks are employed as the sole means of water propulsion, transverse vanes, or water grousers, are formed on the links to increase the effective area in water. The water grouser may be located above, below, or along side of the track pitch line, depending upon the types of road wheels employed and the type and amount of land operation expected. The water grouser is sometimes designed to serve also as the track center or side guide.

11-27.6 SUMMARY OF DESIGN OBJECTIVES (Ref. 46)

The following points may be considered as the major goals for the design of tracks:

a. Low ratio of track weight to vehicle weight

b. Low power loss in the track

c. High static and dynamic floatation

d. High traction values for extreme terrain conditions from hard pavements to mud, snow and ice

e. High degree of durability—1,000 miles for removable grouser pads and 2,000 miles for integral grousers

f. Resistance to track derailment

g. Minimum introduction of vibrations and wheel hop

h. Ease of maintenance and manufacture

i. High-speed performance

11-28 TYPES OF TRACKS

11-28.1 GENERAL

The designer's efforts to provide the optimum track has led to the development of various types, each type intended to best serve some particular track function. Tracks for military vehicles in current service are of two general types, the band track and the block and pin track. Within each type there are, of course, numerous design variations. This is especially true of the block and pin type which has been the most popular in the past. A third type, which is currently experiencing increased attention, is the spaced-link track. This concept may be adapted to either of the two general types. A description of the most frequently used designs is presented in the paragraphs that follow.

11-28.2 BLOCK AND PIN TRACKS

The most common of the block and pin tracks are:

a. Flexible pin-jointed, either dry or lubricated

b. Rubber-bushed type

c. Irreversible types—rigid girder and elastic girder types

11-28.2.1 Flexible Pin-Jointed Type (Ref. 44)

Most foreign tank tracks, at least through the WW II period, were of the flexible pin-jointed type. This track consists of a chain formed of a series of rigid links connected by hinged joints. These hinges are formed by passing a pin through interlocking links, or through adjacent rigid links. The links may be identical or may be of two different types, alternating around the track. This latter construction is referred to as a two-piece link.

Characteristics of pin-jointed tracks are strong-

TABLE 11-5 TRACK APPLICATIONS

Track Model	Description	Application	Track Width, in.	Track Pitch, in.	Shoe Weight, lb	Pin Size, in.
T72E1	Cast steel, single-pin	Lt Tk M24	16	5-1/2	26	.828
T80	Steel and rubber, double-pin	Med Tk M4 series with hor. sus.	23	6	56.5	1.25
T80E1	Steel and rubber, double-pin	Med Tk M26 and M46	23	6	55	1.25
T80E5	Steel and rubber, double-pin	Med Tk M4 series with hor. sus.	23	6	59	1.25
T80E6	Steel and rubber, double-pin (1-1/2″ grouser)	Med Tk M47, M46, and T42	23	6	57.5	1.25
T80E7	Steel and rubber, double-pin (1-1/2″ grouser and 2-1/2″ end connector)	Hvy Tk T43	28	6	60	1.25
T84	Rubber chevron, double-pin	Med Tk M4 series with hor. sus.	23	6	51.7	1.25
T84E1	Rubber chevron, double-pin	Med Tk M47, M26 and T42	23	6	50.6	1.25
T85E1	Rubber chevron, double-pin	Lt Tk M24	14	5-1/2	24	1.00
T91E3	Cast steel and rubber, single-pin, detachable rubber pad	Lt Tk T41E1	21	6	40	.875
T95	Forged steel and rubber, single-pin, detachable rubber pad	Med Tk T42	24	6	48.44	.875
T96	Steel and rubber, double-pin	Hvy Tk T43 and Med Tk M48	28	6-15/16	67.5	1.25
T97	Rubber chevron, double-pin	Hvy Tk T43 and Med Tk M48	28	6-15/16	59.4	1.25
T96E1	Steel and rubber, double-pin	Hvy Tk T43 and Med Tk M48	28	6-15/16	67.5	1.25
T97E1	Rubber chevron, double-pin	Med Tk T43 and Hvy Tk M48	28	6-15/16	59.47	1.25

ly dependent upon track pitch (distance between corresponding points of neighboring joints). Pitches of different tracks vary from about 1-3/4 in. to 10 in. These tracks bend into a polygon, rather than into a smooth curve, when deformed.

Flexible pin-jointed tracks may be subdivided into dry pin and lubricated types of track. In most modern tracks, the pin is without lubrication and friction arises as the metal surfaces move with respect to each other. Track power losses are high and this type of track wears out rapidly. In the past, designs involving lubricated and sealed pins have been produced; however, such systems are not in general use today because of the good service obtained from the rubber-bushed track (Ref. 47).

11-28.2.2 Rubber-Bushed Tracks (Refs. 44, 46, 47)

The rubber-bushed track is similar to the flexible pin-jointed type except that the pin is surrounded by a rubber bushing. The pivotal motion of the hinge takes place as a result of distortion of the rubber bushing rather than by sliding between two metal surfaces. The inner surface of the bushing is bonded to the pin while the outer surface is obliged to turn with the surrounding link.

Most of the block and pin type tracks in cur-

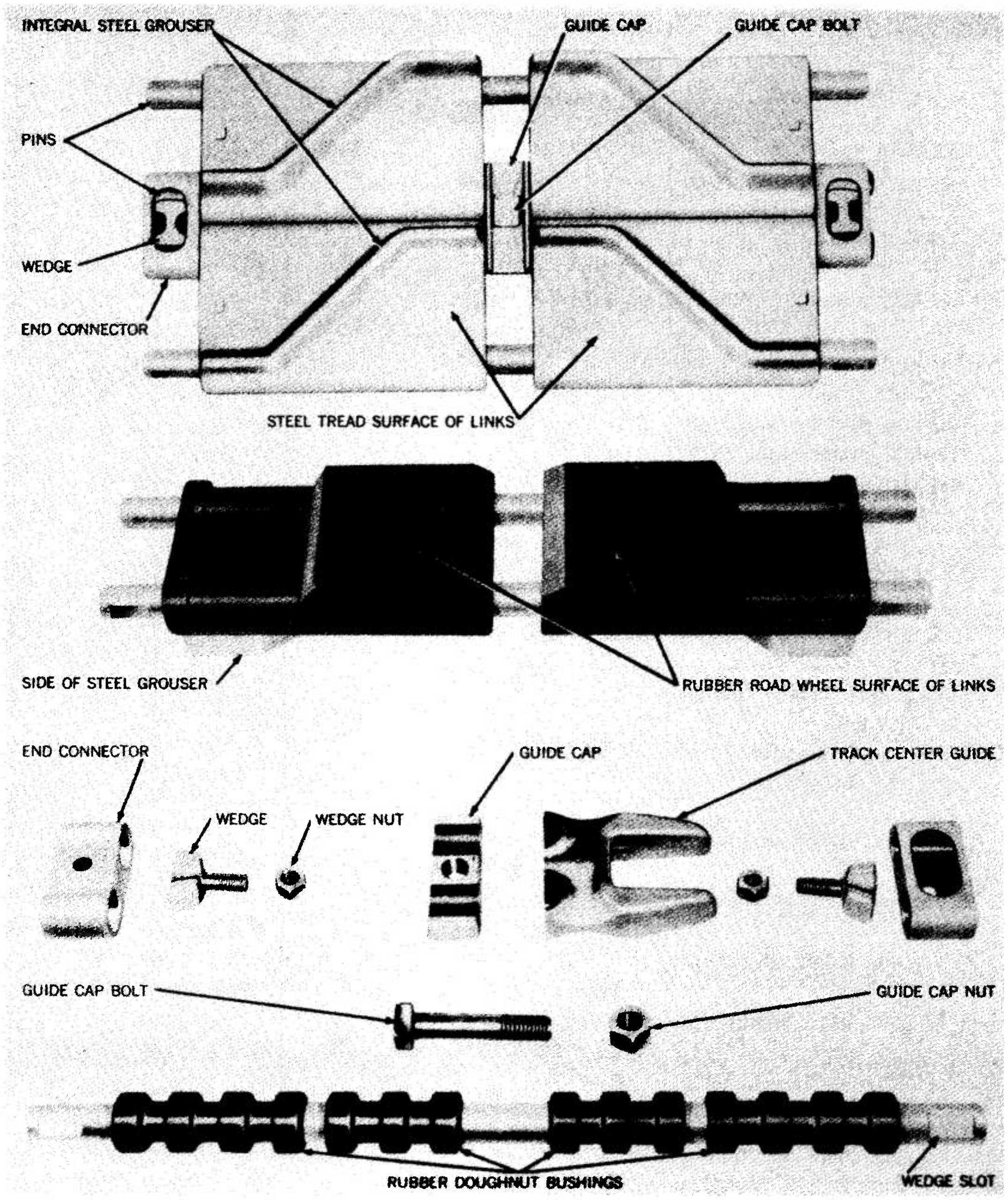

Figure 11-25. *Double-Pin Track Shoe Assembly (T96 Track)*

rent service on military vehicles employ the rubber-bushed pin. Table 11-5 lists several current track models and gives pertinent values concerning each track. Block and pin tracks employ either single- or double-pin construction at the hinge joint. Because of the external appearence of the frame for the double-pin track, they are referred to as binocular frames.

Figure 11-25 shows a typical double-pin track shoe assembly. The basic track unit is the link, which consists of a pair of track blocks assembled onto two rubber-bushed pins. The addition of the end connectors, wedges, and center guide completes the track shoe assembly. The road-wheel side of the blocks are rubber-covered. The cast steel blocks shown have grousers integrated into the ground-contacting side. Quite frequently, rubber pads are employed on this side to cushion the impacts of

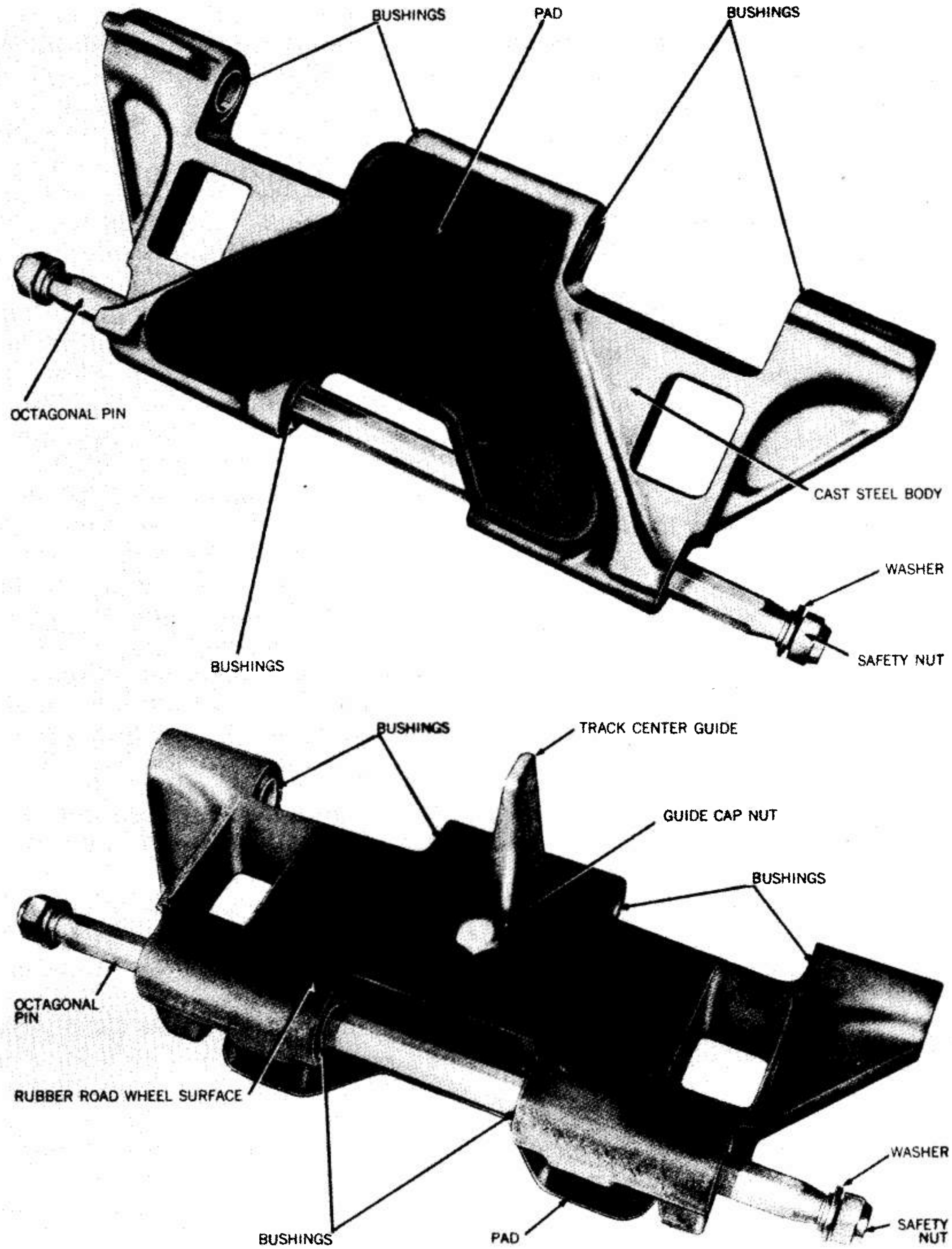

Figure 11-26. *Single-Pin Track Shoe Assembly (T91E3)*

the shoes contacting hard pavement. Sometimes the grouser is a raised portion of the rubber pad.

The links of the double-pin tracks are interconnected by end connectors which engage the track pins. They are secured to the pins by means of the wedges shown. The track center guide is a separate member fastened to the pins. It also serves to space the blocks along the pins.

A typical single-pin track link is shown in Fig. 11-26. Except for the single-pin arrangement, the general design features are similar to the double-pin track links. Component nomenclature is somewhat different in that the single-pin track has no blocks. The basic metal and rubber structure is called the body; the addition of the bushings completes the link, which is comparable to the link of the double-pin track. The link, pin, nuts, and washers make up the shoe assembly.

11-28.2.3 Block and Pin Track Component Design

11-28.2.3.1 Steel Blocks or Links

The durability of steel track links is a function of structural design, type of steel, and method of fabrication. Structural design is essentially a compromise between allowable track weight and structural strength. All-steel, rubber-bushed links generally have an average life of 3,000 miles (Ref. 46).

11-28.2.3.2 Rubber Surfaces for Track Blocks (Ref. 46)

The application of rubber to the tread and road-wheel surfaces of the track shoes has introduced several factors that the designer must consider. The four principal ones are:

a. Bonding of rubber to steel

b. Blowout of rubber by internal heat build-up

c. Excessive hysteresis resulting in power loss

d. Cutting, chipping, shelling out, and abrasion of the rubber

11-28.2.3.2.1 Rubber Bonding

The rubber parts are either vulcanized or cemented to the metal parts. The bond must be strong enough to prevent separation of the materials under the severe loading experienced. The deformation of the rubber produces a lateral stress on the bond.

11-28.2.3.2.2 Hysteresis

The hysteresis of rubber causes two undesirable conditions associated with tank track use. First, energy is lost in overcoming internal friction in flexing the rubber which increases the demands on the vehicle engine, thereby, limiting speed. Second, the friction causes internal heating of the rubber resulting in internal separation and blowout. The designer must specify rubber compounds possessing the lowest hysteresis characteristics available and the highest resistance to scuffing, chipping, chucking, and abrasion. The improvement of vehicle performance and the increase of road wheel tire life more than offsets the disadvantages normally associated with the use of rubber track components.

11-28.2.3.3 Track Pins and Bushings (Ref. 46)

Two basic types of track pins are used, cylindrical for double-pin tracks and octagonal for single-pin tracks. In the double-pin track, the doughnut shaped rubber bushings are bonded to the pin. Their outside diameters are larger than the bored holes in the track links. Upon installation, the bushings are compressed radially to permit their insertion and then allowed to expand radially until their outside diameters are compressed firmly within the hinge bore. Angular oscillations of the pin relative to the hinge bore produce torsional distortions of the rubber rather than sliding of parts. Pin retention is accomplished by means of the wedges.

The single-pin track employs bushings that are bonded into the hinge eyes. The bushings are equipped with octagonal sleeves which engage the octagonal pins, thereby, permitting angular movement through the deflections of the rubber. Retention of the single pin is by means of lock nuts or lock washers at both ends of the pin.

11-28.3 IRREVERSIBLE TYPES (ELASTIC GIRDER AND RIGID GIRDER TYPES) (Ref. 44)

All of the track types discussed previously were free to bend in either direction. In the rigid girder track, the links are so interlocked as to permit the track to form a convex, but not a concave, curve. The ground pressure under a short-pitched pin-jointed track, or a continuously flexible track, is greatest directly under each road wheel, because the track is relatively free to bow upward between them. A rigid girder track is designed to prevent such upward bowing, and the portion of the track between the road wheels remains in contact with the ground.

This type of track cannot be used with a sprung suspension system, which permits road wheels to displace vertically, since this would place an overload upon the unsupported girder when passing over hard obstacles. Its application has been restricted to very slow-moving vehicles, such as, lifeboat carriers and log trailers. It has been suggested that a rigid-girder tracked vehicle be built with the road wheels all mounted on a single

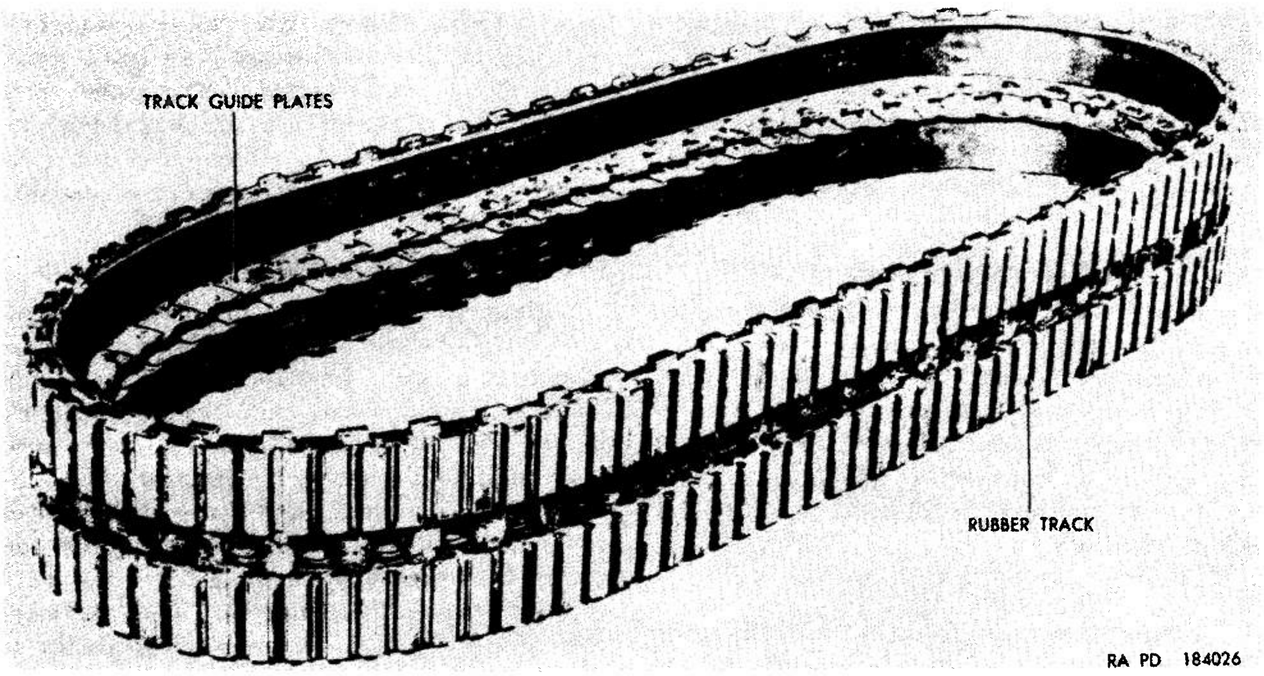

Figure 11-27. ***Rubber Band Track***

rigid unit with suspension springing between this unit and the vehicle hull.

The elastic girder track is similar in principle to the rigid girder track; however, in this case, the joints incorporate rubber blocks which tend to make the track assume a slightly convex curvature. Normal track loads cause this track catenary to flatten out so that ground pressure is more evenly distributed. A slight amount of flexibility is present which permits the track to adjust to rough, hard terrain.

11-28.4 CONTINUOUSLY FLEXIBLE OR ENDLESS BAND TYPE

11-28.4.1 General

This type of track consists of a continuous band which is flexible at any point along its length rather than a series of rigid links that are flexible only at their pin-connected joints. The band is usually formed of fabric or steel-reinforced rubber. Various types of transverse bars, blocks and grousers are added to this flexible band to impart lateral stiffness and to improve its traction. While these transverse elements detract somewhat from the longitudinal flexibility of the track, the overall flexibility is still greater than that of a block and pin track and the track is still considered a band track. The use of the band track has not been as widespread as the block and pin type; therefore, the development of this type is not as advanced. However, the recent emphasis for greater speed and air transportability of military vehicles makes the use of the band track more attractive.

There are three basic types of band tracks:

a. Flexible friction drive

b. Rubber band track

c. Band-block track

11-28.4.2 Flexible Friction Drive Track

The distinguishing characteristic of this track is that stiff lateral members are not employed. This track, like most band tracks, is usually made up of steel and fabric reinforced rubber. Since lateral stiffeners are not employed, the drive force is transmitted by means of V-groove-type drive wheels which engage raised V-belt-like rails on the track. This light flexible type of track has many desirable characteristics for achieving high floatation for light vehicles and cargo trailers. It is not used on combat-type tracked vehicles because of its strength limitations.

11-28.4.3 Rubber Band Track

This term applies to the type of reinforced rubber track used in the American half-track vehicles of the thirties and WW II. Lateral stiffness is obtained in this track by means of rigid cross members vulcanized into the rubber. The driving force is transmitted by the sprocket through these rigid cross members. In order to restrict the twisting of the track along its longitudinal axis, the track guide members are interlocked (see Fig. 11-27).

11-28.4.4 Band-Block Track (Ref. 50)

The band-block track has either rubber blocks or metal track bars bolted or riveted to the flexible, reinforced rubber bands. It is this type that has undergone the most development and is now referred to simply as the band track. Several of the more pertinent design details and functional advantages of this track are discussed in the following paragraphs.

11-28.4.4.1 Vehicle Speed (Ref. 50)

The block and pin track induces substantial vibrations into the suspension system and vehicle hull traveling at high speed. This is because the track bends into a polygon rather than into a smooth curve. The slapping of the track blocks as they strike the terrain produces high ground pressures and shortens the life of the track block. The polygon effect can be minimized by reducing the track link pitch, but this results in increased track weight and compounds the problems associated with track pins (Ref. 47). The continuously flexible band track with narrow track bars approaches the desired smooth bending curve with a reasonable track weight. It is the light weight that is considered by some to be the chief advantage of the band track. The low weight permits the use of very wide tracks to achieve the low ground pressures necessary for the high floatation vehicles.

Data obtained by towing 18- to 25-ton tracked vehicles, with their drive shafts removed, indicate that rolling resistance of band-tracked vehicles is comparable to other types up to 15 mph. At 15 to 30 mph, the rolling resistance of the band track is substantially lower than that of block and pin tracks.

The reduced rolling resistance combined with the superior vibration characteristics of the band track are the factors that permit higher vehicle speeds.

11-28.4.4.2 Sectional Construction

One of the chief disadvantages to the early band tracks was the endless belt type of construction (Ref. 47). Damage to a small section made it necessary to discard the entire band track. Since it was impractical to carry an entire spare track on the vehicle, the vehicle was completely immobilized. Techniques have been developed, however, for the sectional construction of band tracks to eliminate this disadvantage (Ref. 50).

11-28.4.4.3 Band Track Design (Ref. 50)

Figure 11-28 shows a typical band track assembly for a vehicle with a single-road-wheel type of suspension. Band track construction for a dual-road-wheel suspension is quite similar. Referring to the nomenclature on Fig. 11-28, the parallel cable reinforced bands are clamped between the cover plates and track bars by means of through bolts and self-locking nuts. The road wheels run on the track bars between the guides. The drive sprocket engages the center portion of the track bar. Track alignment is provided by means of steel track guides welded to the steel track bars. These guides straddle the road-wheel tires, idler wheel, and sprocket. Each track section is made to a convenient length for ease of handling and is connected to adjacent sections by connector plates. The joint is made between the track bars. Some designs provide for the joint to be made at the track bar.

Rubber pads are bonded into the "V" section of the stamped steel track bars. These inserts greatly reduce wear of the track bars and increase traction on hard surfaces. The rubber pads protrude one-eighth of an inch beyond the metal edge of the track bar. The pad continues to protrude during wear of the metal track bar. Although the track bars shown are steel stampings, aluminum and magnesium castings, forgings, and stampings have been successfully used with the band track on various types of vehicles.

Since it is desirable to have all the track sections of the same length, a design compromise between track bar spacing and total track length is necessary. The width and number of steel-reinforcing cables is a function of the vehicle size. Table 11-6 lists some typical track bands currently in production for vehicles up to 25 tons.

Design parameters and techniques that should be considered by the vehicle designer are given in Ref. 50. Two of the more important factors are:

a. Track bar spacing relative to road-wheel spacing so as not to induce severe disturbances in

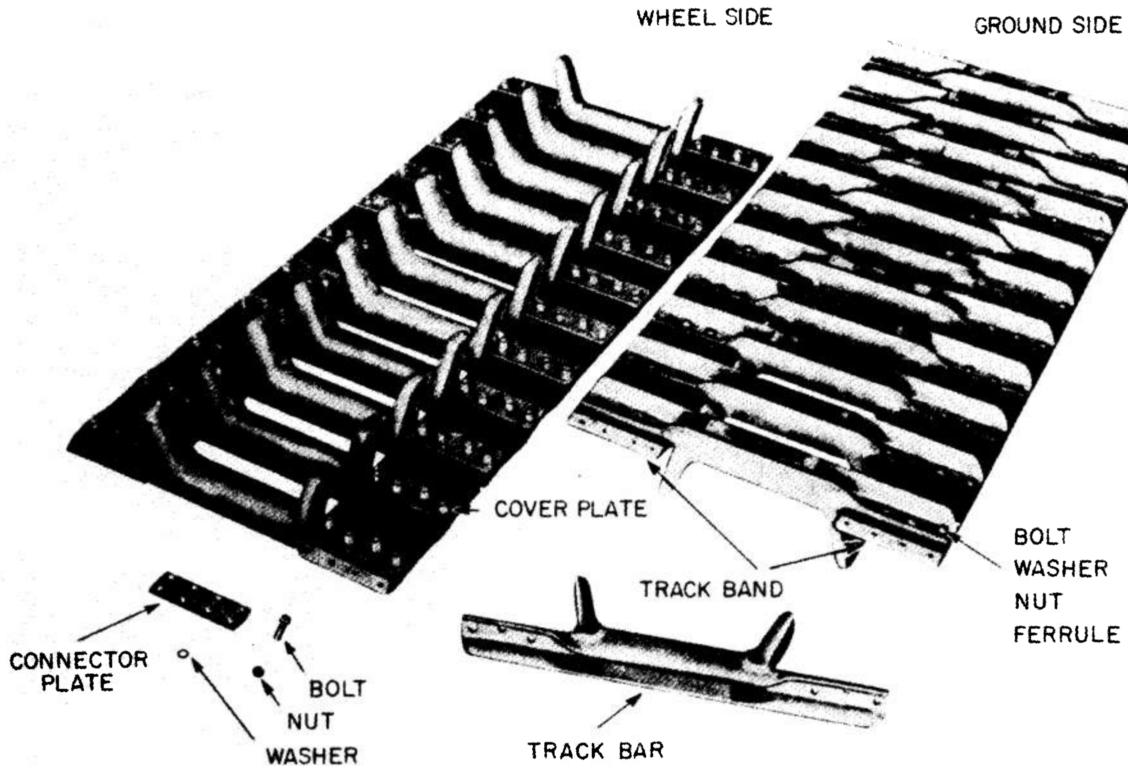

Figure 11-28. *Assembly of Band Track Section*

the suspension by simultaneous climbing and dropping of all the road wheels on the track bars.

b. Drive sprocket tooth pitch to facilitate track bar engagement and disengagement and thereby preclude reverse bending of the cable which reduces the fatigue life.

11-28.5 SPACED-LINK TRACK (Ref. 19)

A vehicle may negotiate a particular terrain provided (1) the soil will support the vehicle at a depth where major non-driving elements are clear of soil contact, and (2) the soil has shear strength sufficient to permit the vehicle to develop tractive effort greater than its resistance to motion. The spaced-link principle utilizes the principles of soil mechanics to develop the maximum shear strength of the soil, while, at the same time, producing a minimum of resistance to motion. A simplified explanation of this principle will aid the track designer in understanding how this is accomplished.

When a horizontal force is applied to a grouser embedded in the soil, shear forces are set up within the soil. These shear forces are distributed in characteristic patterns that curve first downward from the lower tip of the grouser and then upward to the surface of the ground. Failure of the soil occurs when the soil mass under shear begins to slide. Figure 11-29(a) shows the general shape of the soil shear curve. The actual shape depends upon various soil properties and loadings.

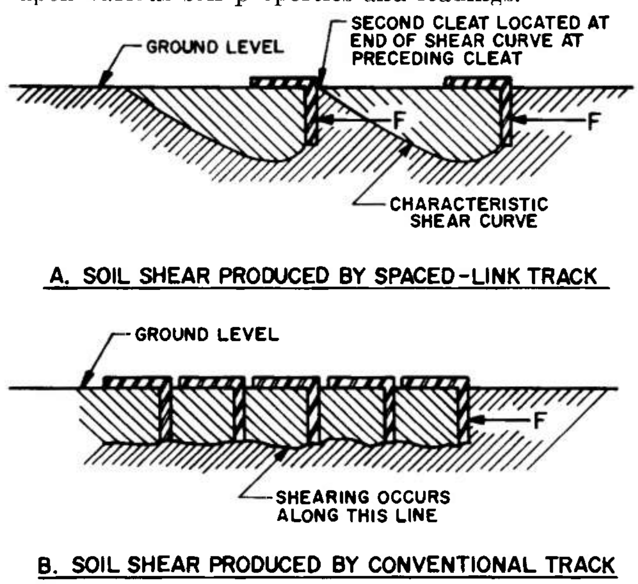

Figure 11-29. *Soil Shear Produced by Conventional and Spaced-Link Tracks*

TABLE 11-6 PRODUCTION AND EXPERIMENTAL TRACK BANDS

Part No.	Vehicle	Weight, tons	Nominal Length, in.	Width, in.	Thickness, in.	No. of Cables	Cable Type	Cable Size	Track Band Weight, lb	Swaged Cable Strength 3D—lb	Band Design Strength, lb	Mfg. Min Specified Strength, lb
PRODUCTION VEHICLES												
8740998-9	M-56	8	44.0	5.370	.430	14	7×7×7	5/32″	4.86	2,300	32,200	25,535
8727043	M-50	9	59.90	5.375	.430	11	7×19	5/32″	9.63	2,800	30,800	29,100
7976864	M-76	6	44.0	3.5	.360	7	7×7×7	5/32″	3.05	2,300	16,100	15,530
7977329	M-76	6	44.0	7.0	.360	14	7×7×7	5/32″	6.04	2,300	32,200	29,950
PILOT VEHICLES												
37858	T-116	4	30.50	4.0	.468	8	7×19	3/16″	3.5	3,775	26,864	25,790
T-700AS	T-107	3	58.37	3.5	.375	8	7×19	3/16″	5.1	3,775	26,864	25,790
EXPERIMENTAL BANDS—GM DYNAMOMETER TEST VEHICLES												
5-18919	TDTV	25	39.62	7.00	.770	8	7×7×7	11/32″	14.29	10,000	80,000	76,800
5-19141	TDTV	20	38.36	6.40	.590	11	7×7×7	1/4″	9.52	5,150	56,650	53,530
5-19142	TDTV	18	38.36	6.40	.590	11	7×19	1/4″	10.23	6,200	68,200	64,450
5-19391	TDTV	18	35.88	5.14	.710	7	7×19	5/16″	9.70	9,700	67,900	65,520
5-19392	TDTV	18	35.88	5.14	.710	7	7×19	5/16″	9.69	9,700	67,900	65,520

The spaced-link track is designed to subject a maximum soil mass to shear. The grousers are located along the length of the track at the points where the preceding shear curves meet the surface. When the grousers are closely spaced, they interfere with the full development of the shear curve, Fig. 11-29(b). Since this results in a much smaller mass under shear, the tractive force is greatly reduced.

Other advantages resulting from this design are (1) a reduction of track weight by the use of fewer grousers, and (2) a reduction of resistance to motion by permitting the soil free entrance at the forward end of the track through large openings.

Test vehicles were built to demonstrate the validity of this concept. The superior performance of one such vehicle, the Ground Hog, is described in par. 4-18.3.

The spaced-link principle may be implemented into practical design by either the block and pin or band track types of tracks. Since it is a relatively new concept, there exists considerable area for development to exploit its full potential. Undoubtedly, design shortcomings will become apparent as attempts are made to extend the capabilities of this principle. Many design problems will arise; for example, when high-speed operation on hard pavement is attempted. Because the vehicle equipped with spaced-link track can go in soft snow where other tracked vehicles cannot, new problems in steering have been discovered. Another area that will demand design ingenuity involves the adaptation of this type of vehicle to operation on rock-strewn terrain. The grousers and links must withstand impacts and abnormally high loadings in this type of operation and still be capable of developing the desired effect in soft terrain.

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CHAPTER 12

THE STEERING SYSTEM*

SECTION I GENERAL DISCUSSION

The behavior of the wheeled or tracked vehicle in response to steering is influenced by many factors. In this chapter the more important of these factors are introduced and the part each plays in determining the lateral motion of the vehicle as it travels along a curved path, examined.

The steering system of a given vehicle, in conjunction with the suspension system, determines the degree of control and stability present under any operating condition. The directional control system (steering system) must function as a maneuvering and course-keeping unit, while keeping manual effort and power losses at a minimum. Power loss is especially important with regard to the steering of tracked vehicles, since it may represent a relatively high percentage of the total power available; and as such, it may severely limit the steering capabilities of a vehicle in adverse terrain.

Some of the factors that influence the steering behavior of a vehicle are:

1. Static weight distribution
2. Geometry of the mechanical components
3. Operating characteristics of steering transmissions
4. Vehicle response to centrifugal forces, slope induced forces and drawbar forces
5. Vehicle-ground interaction

In the design of the steering system, attention must be directed towards the problems of adjustment and repair in the field. Steering of military vehicles is often complicated by their extreme size and weight as well as their operating conditions. Power assistance is normally included in the steering system of tracked vehicles and is necessary for the larger wheeled vehicles.

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SECTION II STEERING OF WHEELED VEHICLES (Ref. 16)

12-1 INTRODUCTION

Conventional wheeled vehicles are usually steered by either fifth-wheel steering or Ackermann steering (Fig. 12-1). Camber steering, in which the wheels are inclined from a vertical position has limited application in current vehicular development.

Although the fifth-wheel (wagon steer) method of steering was developed prior to the Ackermann system, the Ackermann system is used on all standard military vehicles at present. However, a powered fifth-wheel steering system (Goer vehicle) is currently under development and eight-wheeled vehicles have been built using fifth-wheel steering systems. Experimental vehicles with articulated steering have also been built that did not have the vertical steering axis located at the center of the steering axle.

Steering, in which the speed of the wheels on one side of the vehicle is varied in relation to that of the other, has been incorporated in an eight-wheeled experimental vehicle. A steering transmission or separate power plants for each side of the vehicle is required in this case. The geometry

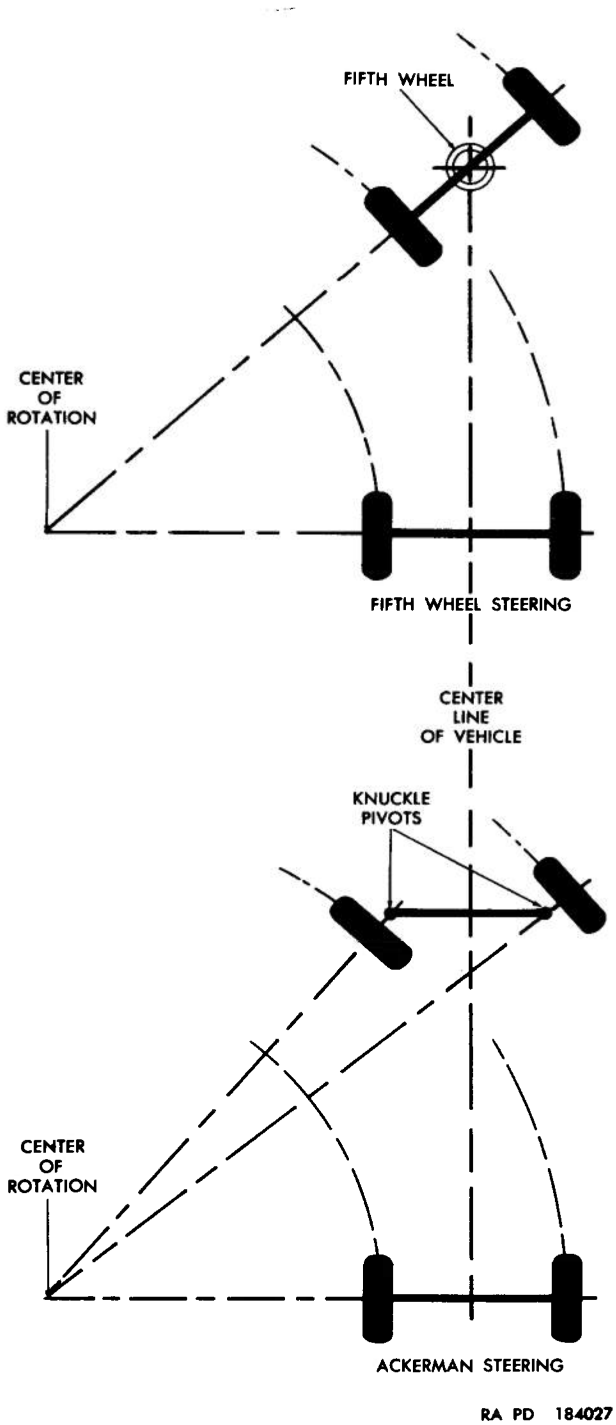


Figure 12-1. ***Basic Methods of Steering Wheeled Vehicles***

of this system is such that, during a turn, severe lateral sliding (tire scrubbing) occurs.

Regardless of the system used, steering of multiwheeled vehicles are compounded by the presence of multiple parallel axles. It can be shown that if such a vehicle is to turn without excessive lateral scrubbing of the tires, the extended projections of the axles must pass through a vertical line at the center of curvature of the vehicle path. This is not possible with parallel axles of tandem wheel assemblies.

Tractor-trailer units and articulated vehicles have steering characteristics different than those of single vehicles.

Other factors to be considered in the steering of wheeled vehicles are: tire slip-angle during a turn, effects of braking and traction, suspension geometry, load transfer arising from vehicle roll, and the effect of inertial and aerodynamic forces on the vehicle.

12-2 ACKERMANN STEERING

The divided axle steering system used in most current automotive vehicles was invented in 1817 by Lankensperger, a Munich carriage builder. Because the English patent was taken out in the name of Rudolph Ackermann, his English agent, this steering system has come to be known as the Ackermann steering system in English-speaking countries.

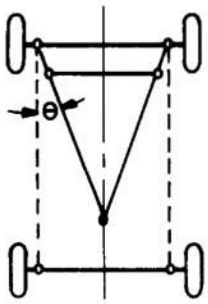
Charles Jeantaud, a French carriage builder refined this mechanism in 1878, by noting that the center lines of the axles, if extended, must meet in a common point if the vehicle is to turn a corner without slipping. His contribution consisted of inclining the steering arms toward each other in the plan view, (see Fig. 12-2) so that their center lines, if extended, would intersect at the center of the rear axle. Later investigations showed that the usual four-bar linkage could not satisfy the conditions of correct steering over the whole range of steering wheel motions. It has been shown that the correct point of intersection of the two steering arms lies some distance in front of the rear axle.

Correct Ackermann steering during a turn requires that each wheel turn about a point which is located on an extension of the rear axle center line (see Fig. 12-2). Accordingly, the relation between the front wheel steering angles, α for the inner wheel, and β for the outer wheel, is

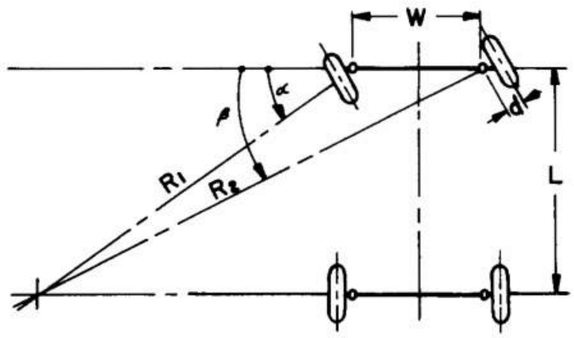
$$\cot\beta - \cot\alpha = \frac{w}{l} \qquad (12\text{-}1)$$

where w is the distance between the vehicle steering pivots and l, the wheelbase.

The minimum turning radius of a four-wheeled



(A) ACKERMANN-JEANTAUD STEERING LINKAGE



(B) STEERING ANGLES IN ACKERMANN STEERING

Figure 12-2. ***Ackermann Steering Relations***

vehicle having Ackermann steering is defined as the radius of curvature of the centerline of the path made by the outer front wheel when the vehicle is making its shortest turn.

The radius of curvature, R, is given by

$$R = R_2 + d \qquad (12\text{-}2)$$

where d is the length of the steering pivot arm and R_2 is the distance from the center of rotation to the steering arm pivot point (Fig. 12-2).

From the geometry of the figure, the minimum radius of curvature can be written as

$$R = \left[\left(\frac{l}{\sin \alpha}\right)^2 + w^2 + \frac{2wl}{\tan \alpha}\right]^{\frac{1}{2}} + d \qquad (12\text{-}3)$$

where α is the maximum angle through which the inner front wheel can be deflected from the straight-ahead position.

Because the first term in the bracket of Equation 12-3 is much greater than the remaining right-hand terms, it is common to designate the turning radius of the vehicle as

$$R = \frac{l}{\sin \alpha} \qquad (12\text{-}4)$$

The steering angles produced by a typical four-bar linkage may be examined with the aid of Fig. 12-3. The angle, Θ, represents the initial inward inclination of the steering arms from parallel. The angles, α and β, represent the steering angles introduced into the steering arms as the steering wheel is turned. All of terms used in the derivation below are defined in Fig. 12-3.

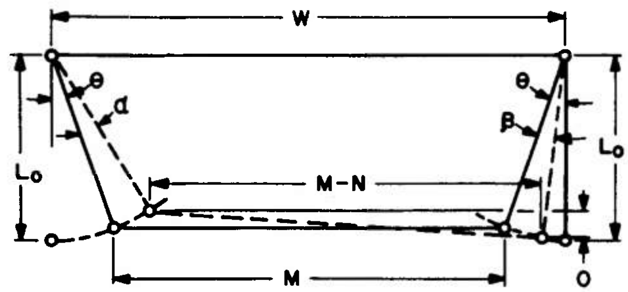


Figure 12-3. ***Geometric Relation Between Steering Angles of Front Wheels***

The geometric relation between the angles α and β is given by the following equations (Ref. 12):

$$\beta > \Theta \qquad (12\text{-}5a)$$
$$\sin(\Theta - \beta) = 2 \sin \Theta - \sin(\Theta + \alpha) - N/l_o$$
$$\beta < \Theta \qquad (12\text{-}5b)$$
$$\sin(\beta - \Theta) = (\Theta + \alpha) - 2 \sin \Theta - N/l_o$$

Steering arms extending to front

$$\beta > \Theta \qquad (12\text{-}5c)$$
$$\sin(\beta - \Theta) = (\Theta + \alpha) - 2 \sin \Theta - N/l_o$$
$$\beta < \Theta \qquad (12\text{-}5d)$$
$$\sin(\Theta - \beta) = 2 \sin \Theta - \sin(\Theta + \alpha) + N/l_o$$

where

$$N = M - (M^2 - O^2)^{1/2} \qquad (12\text{-}6)$$

and

$$O = l_o\,[\cos(\beta - \Theta) - \cos(\alpha - \Theta)] \qquad (12\text{-}7)$$

These equations are generally solved by approximate methods. In one such method, β is first solved in terms of α when N/l_o is assumed to be equal to zero. N is then found from Eq. 12-6 and substituted in Eqs. 12-5 to find a new value of β. Experience has shown that such a calculation determines β within 1 minute of arc, under the most extreme conditions, hence, this value is taken as the correct one. Graphical methods are often used to determine β as a function of α.

If the values of β produced by the steering linkage for given value of α are compared to the theoretical Ackermann values, it will be seen that they agree in the straight-ahead position and at one other value of α. For small angles, the angle of the outer wheel produced by the steering linkage is somewhat too large, while at larger angles this angle is too small. The optimum steering arm angle therefore depends on the maximum turning angle desired as well as the length of the track, wheelbase, and steering arm. Greater steering arm angles result in smaller values of excess outer wheel angle at lower values of α. However, they introduce greater errors at higher value of α. For greater ranges of motion, smaller values of steering arm angle are required.

12-3 STEERING GEARS

In the control of the vehicle, it is necessary to convert the rotational motion of the steering wheel to the linear motions of the steering linkage which, in turn, moves the front wheels. This conversion is carried out by the steering gear which is ordinarily fixed to the bottom of the steering column.

The steering gear (see Fig. 12-4) has two main functions. It converts rotary motion of the steering wheel into linear motion of the steering linkage and it also serves as a reduction gear between steering wheel motion and the front wheels. Typical gear ratios are 14 to 1 or higher and the total motion of the front wheels is about 80 degrees, for a light vehicle.

A number of types of steering gears have been or are being built. Early designs included worm-and-gear, worm-and-sector, and worm-and-nut types. Skew and bevel gears have also been used.

In the worm and gear designs, a worm fixed to the lower end of the steering column meshed with the worm gear which was mounted on the Pitman arm shaft. Because the rotation of the worm gear was limited to less than 90°, it was soon replaced by a gear sector.

The worm and nut arrangement employs a nut fitted to the worm. The nut is prevented from rotating and therefore moves axially when the steering wheel is turned. The motion of the nut is transferred to the Pitman shaft by means of a lever.

A disadvantage of the early worm and gear designs was the difficulty in adjusting for wear. A disadvantage of the worm and nut designs was the large surface contact between them which tended to make steering stiffer under cold weather conditions.

Present steering systems, which are evolved from these basic designs include; cam-and-roller, cam-and-lever, and worm-and-nut types. The first two of these are derived from the worm and sector type. In the place of the worm sector, the cam and roller type employs a roller attached to the Pitman arm shift, while the cam and lever type utilizes a follower in the place of the sector gear. Figures 12-5 through 12-9 show various steering gear configurations.

Another design, used abroad, consists of a rack and pinion mechanism. Here, rotational motion of the steering wheel is converted to linear motion by

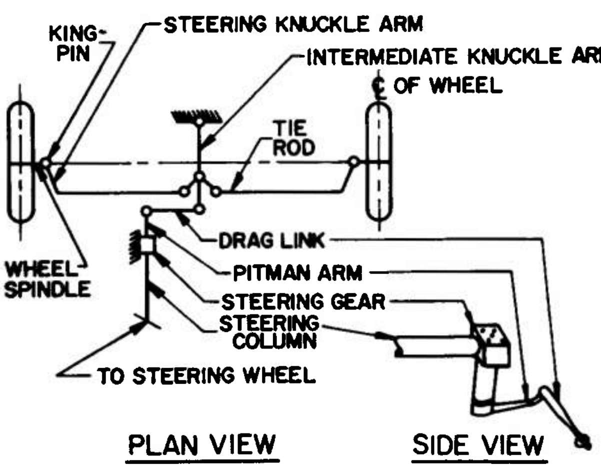
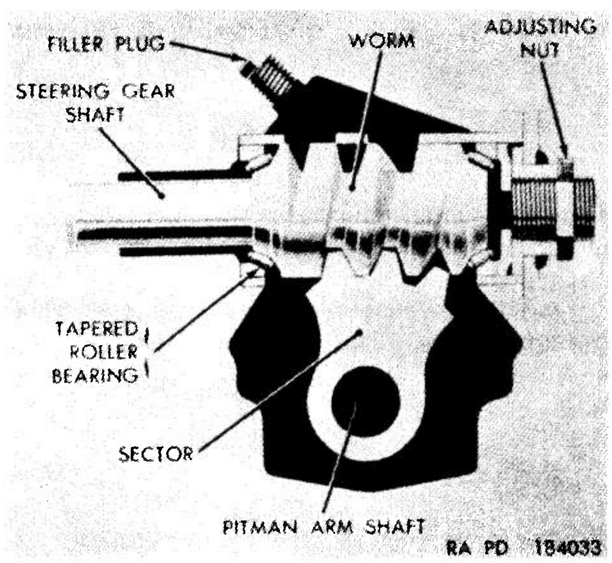


Figure 12-4. *Steering Linkage with Intermediate Knuckle Arm*



Fgiure 12-5. *Simple Worm-and-Sector Steering Gear*

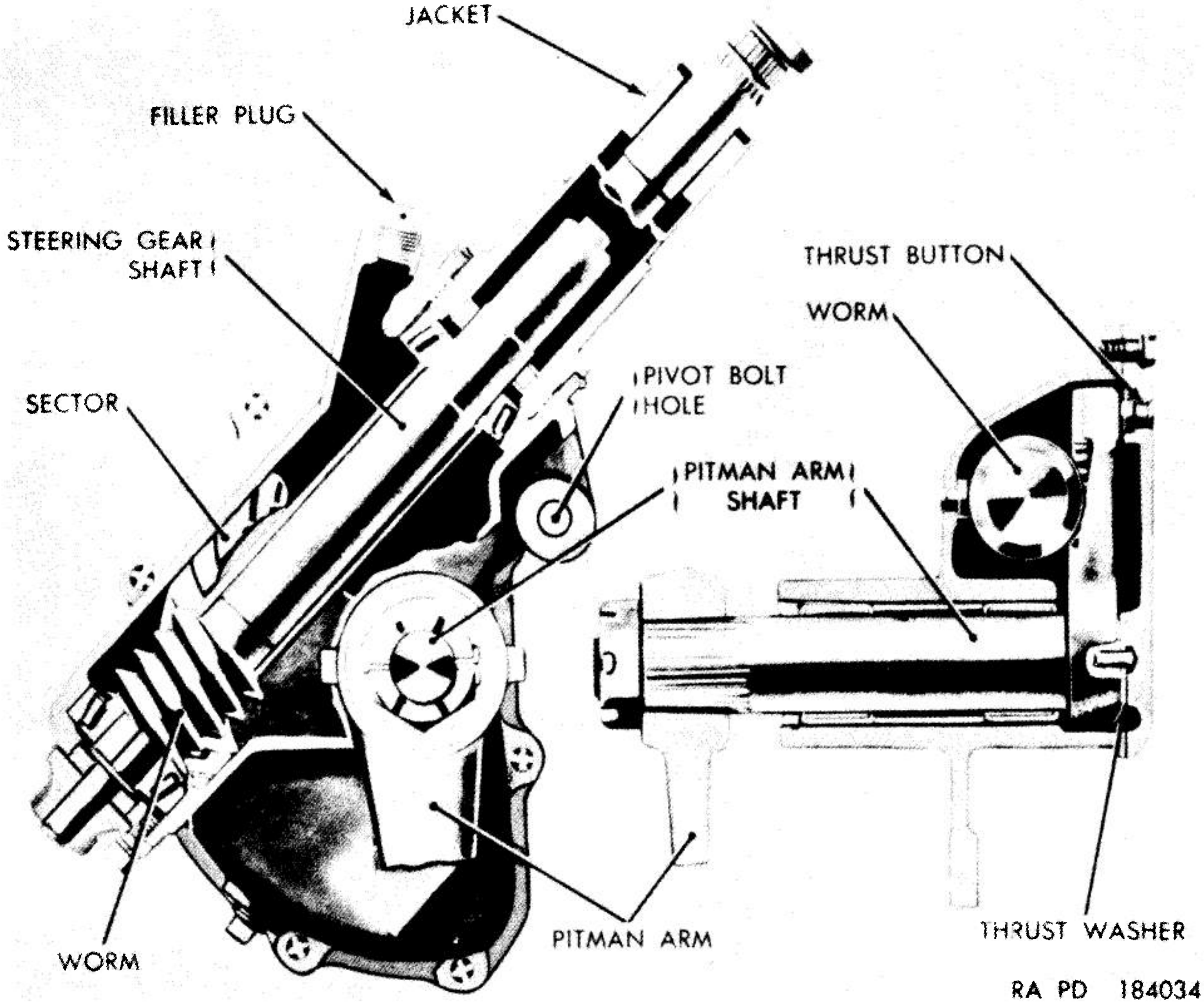


Figure 12-6. *Variation of Worm-and-Sector Steering Gear*

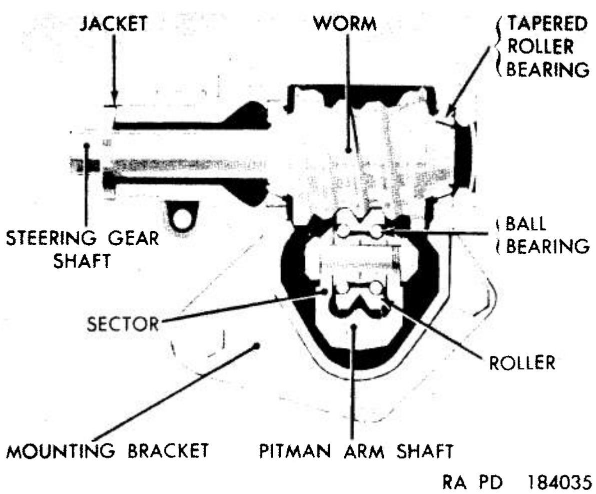


Figure 12-7. *Worm-and-Roller Steering Gear*

the meshing of a pinion on the steering column with a rack which slides transversely across the vehicle.

In the modern cam-and-roller type, the relative motion between them is entirely rolling action which results in reduced friction. The cam, which replaces the worm of the earlier design, has a helical thread and is of hourglass shape to ensure contact with the roller, which moves in an arc during its rotation. With the roller mounted in ball bearings, forward mechanical efficiency in the range 85 to 90 percent can result. In this design, the steering ratio is variable throughout the motion of the steering wheel if the lead of the worm is constant.

In the cam-and-lever system, the helical cam

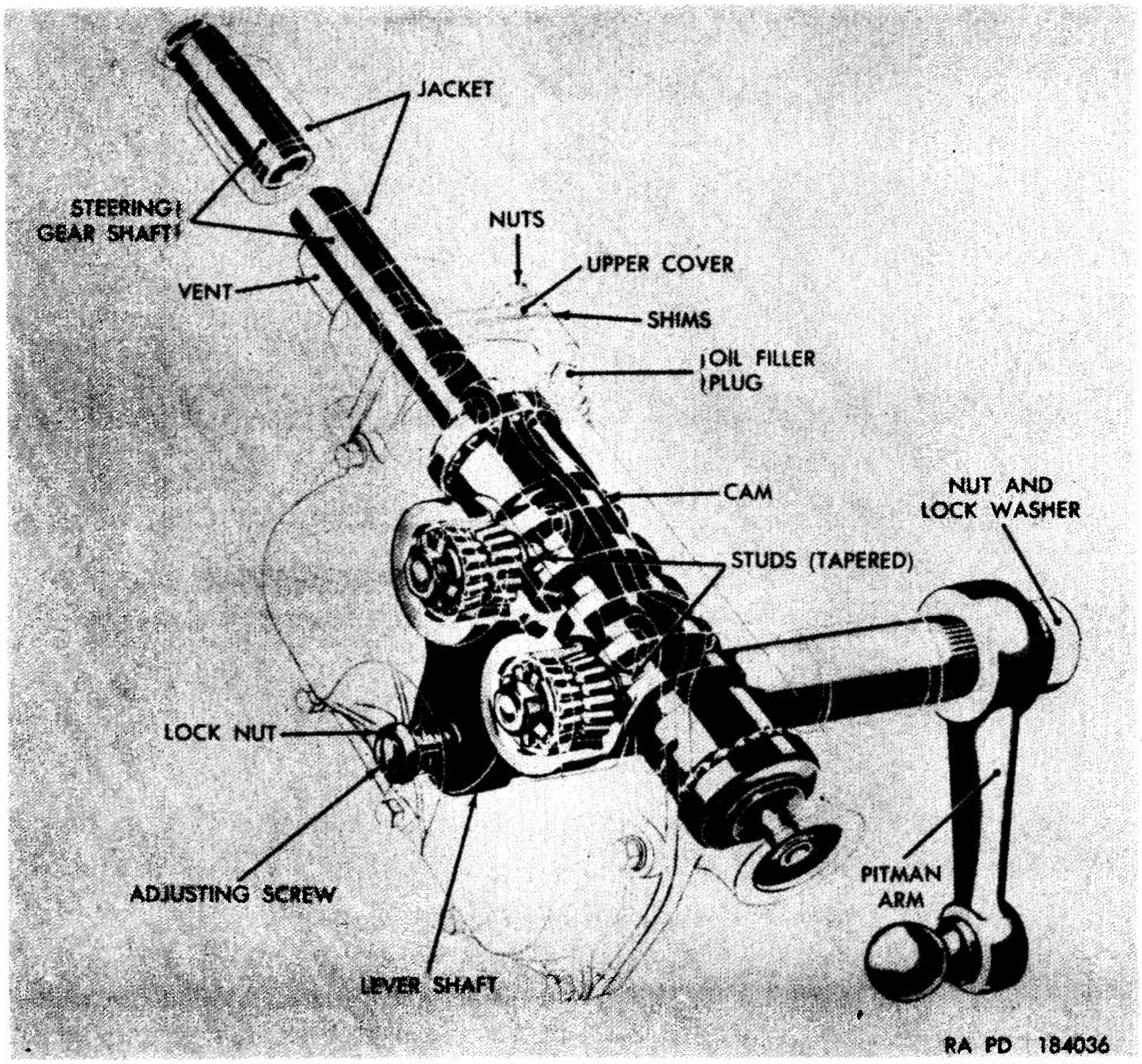
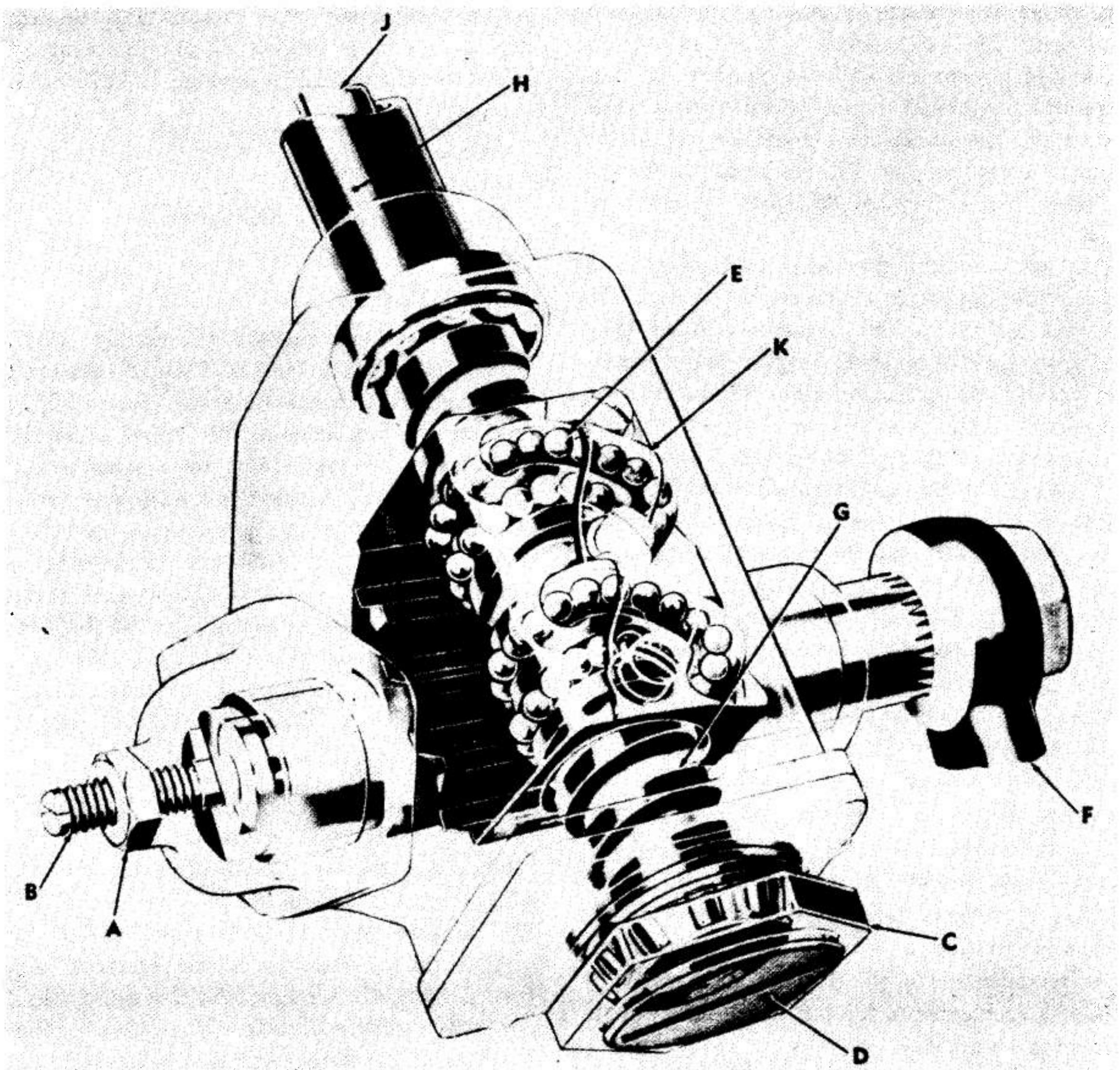


Figure 12-8. *Cam-and-Lever Steering Gear*

which replaces the worm of the earlier system is of cylindrical shape. The motion is transferred to the Pitman arm shaft by means of a follower attached to a lever which is fastened to the shaft. As the cam rotates, the follower is constrained to move in an arc. If the helix pitch is made variable, lower gearing can be provided around the center or straight ahead position. One example of such variability on a light vehicle design included a narrow band ($\pm 24°$) about the straight ahead direction in which 12 to 1 ratio existed, a linear increase in ratio to 14 to 1 at $\pm 50°$, and constant ratio thereafter. In the above design, the steering ratio dropped linearly when the steering gear rotation exceeded 144° to left. This was done to compensate for steering linkage geometry effects. The lower ratio provides for more responsive action in the range required at high speeds.

For a follower supported in roller bearings, forward mechanical efficiencies of 75 to 90 percent and reverse efficiencies of 66 to 82 percent are obtained. For an installation in which the follower



A—LASH ADJUSTER SCREW LOCKNUT
B—LASH ADJUSTER SCREW
C—WORM BEARING ADJUSTING SCREW LOCKNUT
D—WORM BEARING ADJUSTING SCREW
E—RECIRCULATING BALLS
F—PITMAN ARM
G—WORM
H—JACKET
J—STEERING GEAR SHAFT
K—BALL NUT

RA PD 184037

Figure 12-9. *Worm-and-Nut Steering Gear (Recirculating Ball-Type)*

was not free to rotate, these values were 52 to 58 percent forward and 29 to 37 percent in reverse.

The contemporary worm-and-nut type utilizes a recirculating ball type of nut to reduce friction. The helical track on the cam is semicircular, which form is matched by an internal helical track in the nut. Recirculating balls run in the resulting circular helical track and provide a rolling action between cam and nut. As the nut is restrained from turning, it moves along the cam as the cam rotates. The linear motion of the nut is transferred to the Pitman arm by means of an extension to that arm. The extension is fastened to the nut by means of a spherical socketed joint which permits relative rotational motion between them while transforming the linear motion of the nut to the rotational motion of the Pitman arm. A typical installation has a forward efficiency of 71 to 82 percent.

The worm-and-nut type can produce either a higher or lower gearing at the center position. In those cases where the nut is not restrained from rotation, the ratio is somewhat higher at the center. As an example, in one installation, the ratio drops progressively from about 14 to 1 at the straight ahead position to 12 to 1 at 40° of Pitman arm travel. When the nut is restrained from rotating, the ratio increases as the wheel is moved from the center position. In one example, this ratio increased from 17.6 to 1 at the straight ahead position to 26.2 to 1 at 35°. For a heavier vehicle, the latter type of curve is preferred to reduce the steering effort during low speed maneuvering when the wheels are displaced near the lock position.

In the United States, the cam-and-roller gear, the cam-and-lever gear, the worm-and-nut type are most frequently used. The cam-and-lever gear in some cases contains two followers which engage alternately to provide an improved ratio curve. One type of worm-and-nut gear contains a double set of recirculating balls and the motion of the nut is transferred to the Pitman arm shaft by means of a rack cut into the nut which meshes with a pinion on the shaft.

Rack and pinion gears are used on small passenger vehicles where high precision steering is required. Their use on larger vehicles is limited because of the conflicting space requirements of rack and engine. Because reverse efficiency is almost as high as forward efficiency, road reactions are transmitted backwards to the steering wheel, hence, accurate steering geometry and wheel balancing are required. Damping to increase friction is generally needed and helical gears are often employed to increase smoothness of operation. Forward mechanical efficiencies of about 65 percent and reverse efficiencies of 59 percent have been attained on recent units.

12-4 WHEEL ALIGNMENT

Wheel alignment is the process of keeping all of the interrelated parts of the unsprung mass properly adjusted. Five factors must be properly chosen to obtain proper wheel alignment. These include (see Figs. 12-10 through 12-12):

1. Toe-out
2. Caster
3. Camber
4. Kingpin (pivot) inclination
5. Toe-in

12-4.1 TOE-OUT

Toe-out describes the process by which the proper Ackermann effect is obtained. An examination of an Ackermann diagram (Fig. 12-2) shows that the steering arms in the typical 4-bar linkage arrangement do not travel over equal arcs. The inner wheel travels through a greater arc than does the outer wheel to permit it to travel about a smaller radius. The difference in steering angles between the two wheels is termed the toe-out. It is usually specified as the number of degrees over 20° which the inner wheel is turned when the outer wheel turns 20°. Figure 12-13 shows these relationships for a nonindependent front axle.

12-4.2 CASTER

Caster is the angle, measured in degrees, that the steering knuckle pivots are tilted forward or backward from the vertical, when viewed from the side. Caster is designated positive for a backward tilt. Caster tends to keep front wheels pointed straight ahead because as the castered wheel is turned from the straight ahead it raises the front end of the vehicle slightly. When the steering gear is released, the weight of the vehicle tends to force the wheels to the center position.

Caster can be affected by braking forces. As an example, for a solid axle supported on semi-elliptic springs, caster angle may change more than 3° in heavy braking. If the caster angle is chosen to give adequate caster during normal running, violent braking could result in an anticaster effect (Ref. 15).

Self-aligning torque of the tire has an important self-centering effect. Reference 15 has shown that at some slip angles, this torque can be equivalent to 10° of caster angle.

The increased rolling resistance of the outer tire in cornering acting at a point outside the

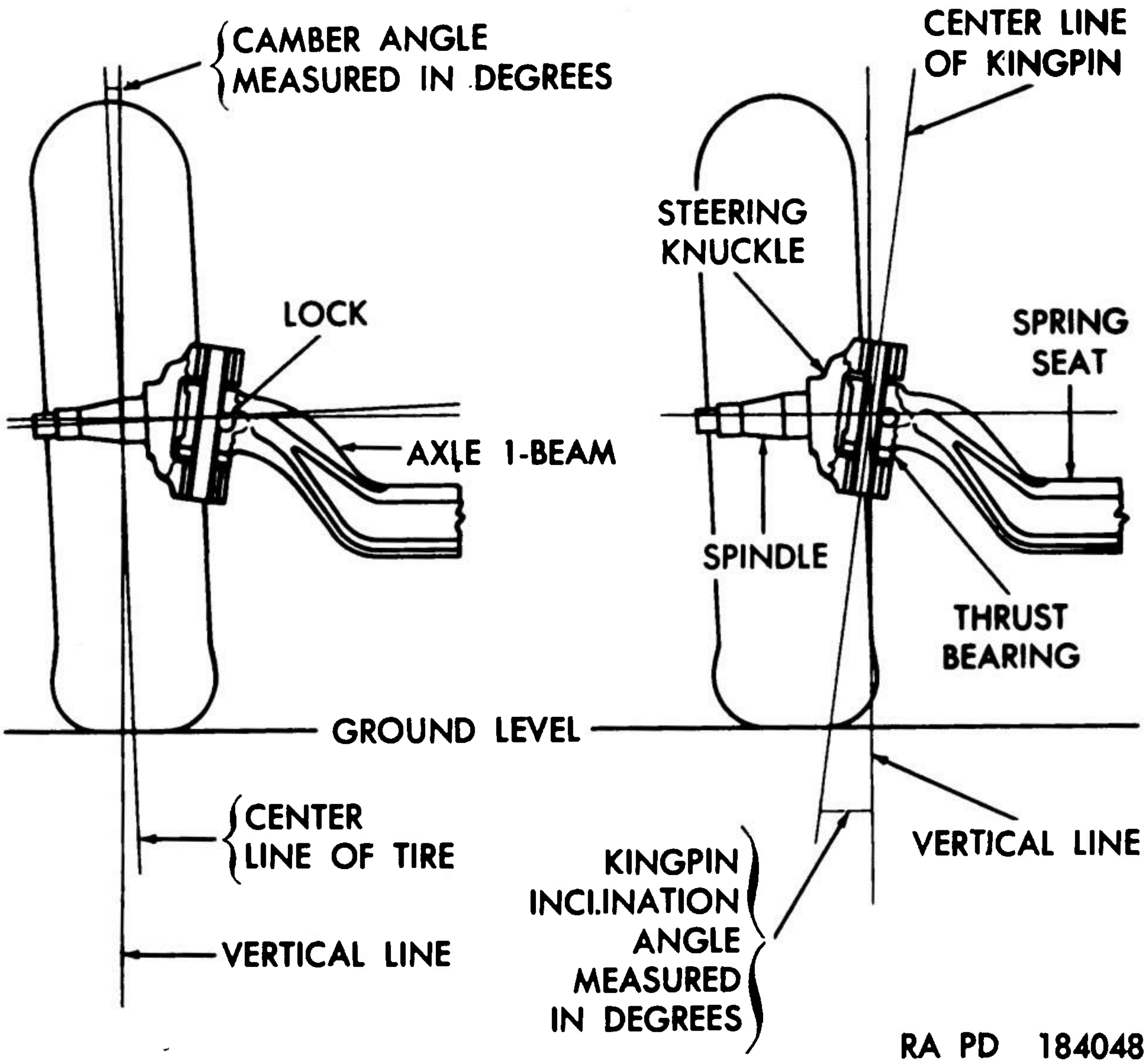


Figure 12-10. *Camber and Kingpin Inclination*

steering pivot axis also tends to straighten out the vehicle. Ordinarily, however, this force is less than that of the self-aligning torque.

12-4.3 CAMBER

Wheel camber is the angle between the center plane of the wheel and the vertical when viewed from the front of the vehicle. Camber is considered positive when the top of the wheel leans outward. In earlier years camber up to 3° was employed to permit the wheel to be vertical to highly crowned roads. In recent years, the use of flat roads and low pressure tires has led to a reduction in camber to about 1°. Under ideal conditions, zero camber would result in no wheel slippage; however, because of bearing clearances and axle deflection, a slight amount of camber must be provided. Excessive camber would result in side slip of the tire because of lateral forces on the tire tending to move it in the direction toward which it is leaning.

12-4.4 KINGPIN INCLINATION

Kingpin or pivot inclination is the amount in degrees that the steering knuckle pivots are tilted inward toward the center of the vehicle. This inclination results in a self-aligning torque as the steering wheel is turned because a rotation of the wheel raises the vehicle slightly.

The pivot axis meets the ground near the cen-

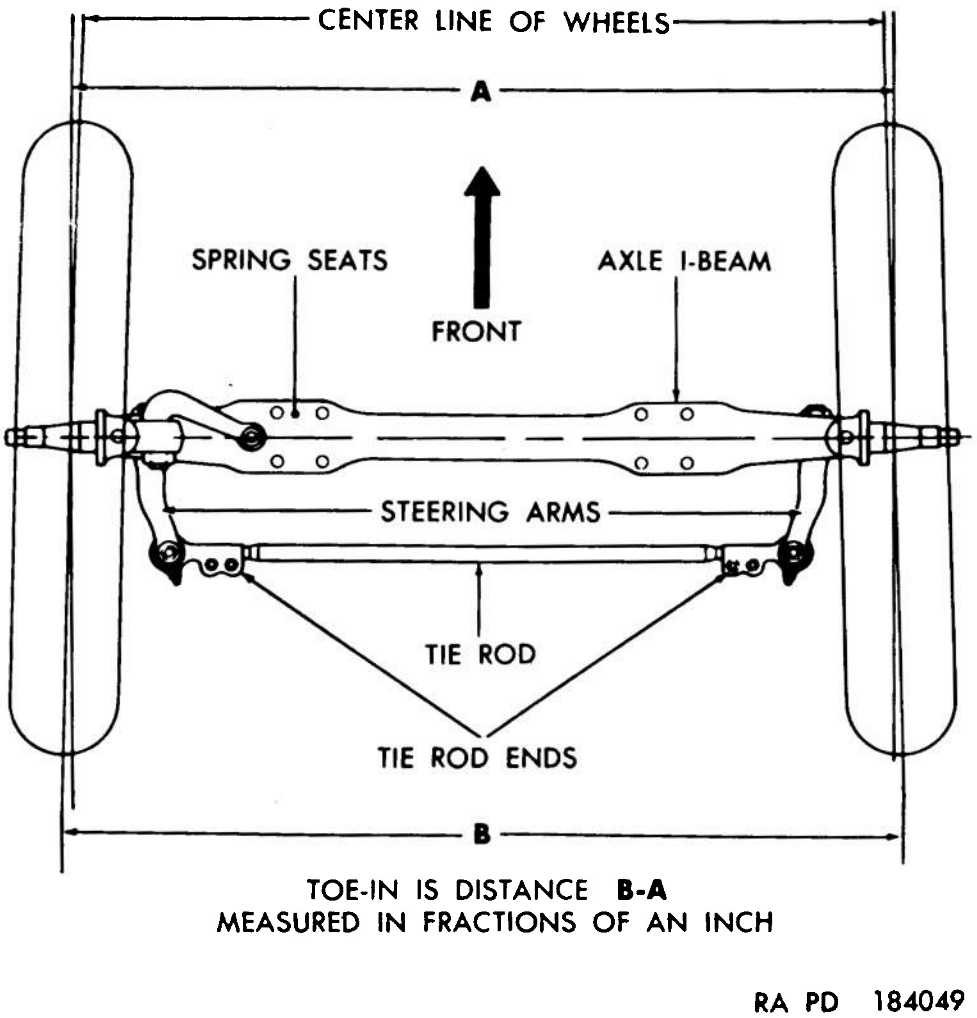


Figure 12-11. *Toe-in*

ter of the tire patch so that the lever arm of the tire forces about the pivot axis is less and steering moments are reduced. This distance, however, should not be to small or the tire will slide rather than roll during steering. Pivot inclination is generally in the range of 3° to 7°.

12-4.5 TOE-IN

Toe-in is the amount in inches that the wheels point in when viewed in plan. For a flat road, Ref. 14 indicates that zero toe-in should be maintained. However, such a condition is difficult to maintain.

Toe-in is used to balance the effect of camber. Generally, the front wheels are somewhat cambered and as a result tend to move in a curved path as the vehicle moves forward. There is a continual tendency for wheels which are oppositely cambered to slip away from each other. Toed-in wheels tend to slip toward each other and counteract this condition. By proper choice of camber and toe-in, tire wear, and the pull on the steering mechanism, are reduced.

12-5 STEERING GEOMETRY ERRORS

The path followed by the front wheels is influenced by many factors, some of which require that the theoretical Ackermann steering geometry be compromised. Complications introduced by independent front wheel suspension include the need for a double tie-, or track-rod, greater wheel travel, and the involved, nonplanar motion of the wheel hub.

If the path of the wheel hub, that is, the suspension geometry, does not agree with the steering geometry, the front wheels will turn as they are deflected upwards. This causes the vehicle to

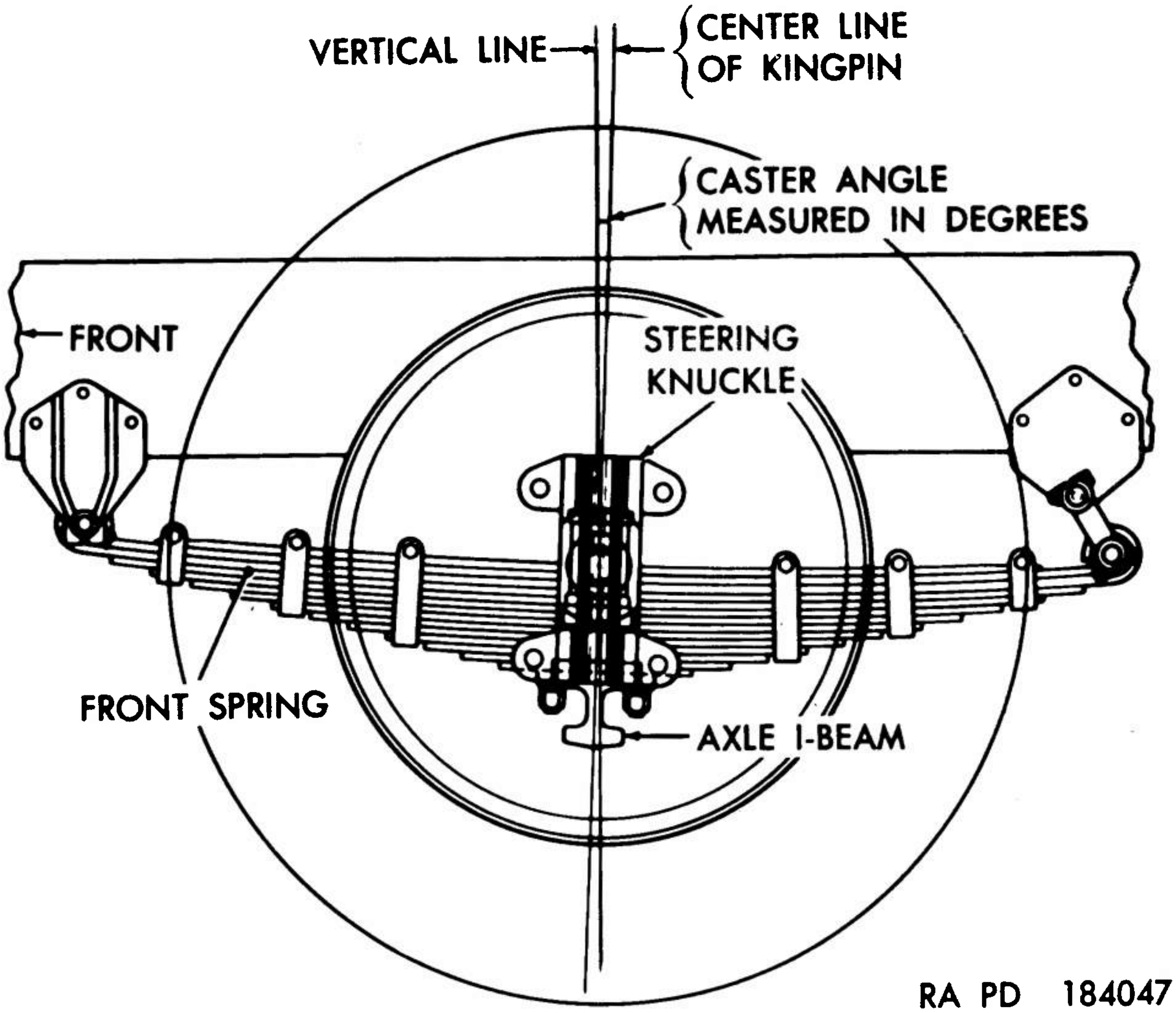


Figure 12-12. *Caster (Shown Positive)*

depart from its intended path unless corrected by the driver, and produces road shock at the steering wheel. Road shock, felt at the steering wheel, is in most cases, not the reflected direct shock which arises as a bump is traversed, but is, instead, the inertia (and gyroscopic) force of the wheel resulting when compromise between steering and suspension geometries attempts to change the plane of rotation of the wheel. Direct inertia and gyroscopic torques can also arise from the compromise suspension constraint of the wheel hub which results in a change in the plane of rotation during wheel motions.

In a wishbone- or linked-type front suspension employing equal parallel links, wheel motion is vertical for no roll of the sprung mass, hence, gyroscopic torque is eliminated. The path of any point on the kingpin of such a system is an arc whose radius is equal to the length of the links. Because of conflicting space requirements, it is not always possible to obtain the theoretical steering geometry, and compromises must be made.

A deviation from the theoretical value has been found to be of less importance when this deviation is in the length of a steering component rather than a deviation in its angular position. Also, the greater the radius about the steering pivot axis at which the steering arm acts, the less angular error introduced to the wheel by a given linear error in steering arm motion. The effects of such geometry errors on motion of the front wheels are shown in Fig. 12-14. In that figure, the correct steering link (reach rod) length is AO. Under deflected conditions AO moves to AO_A and no change

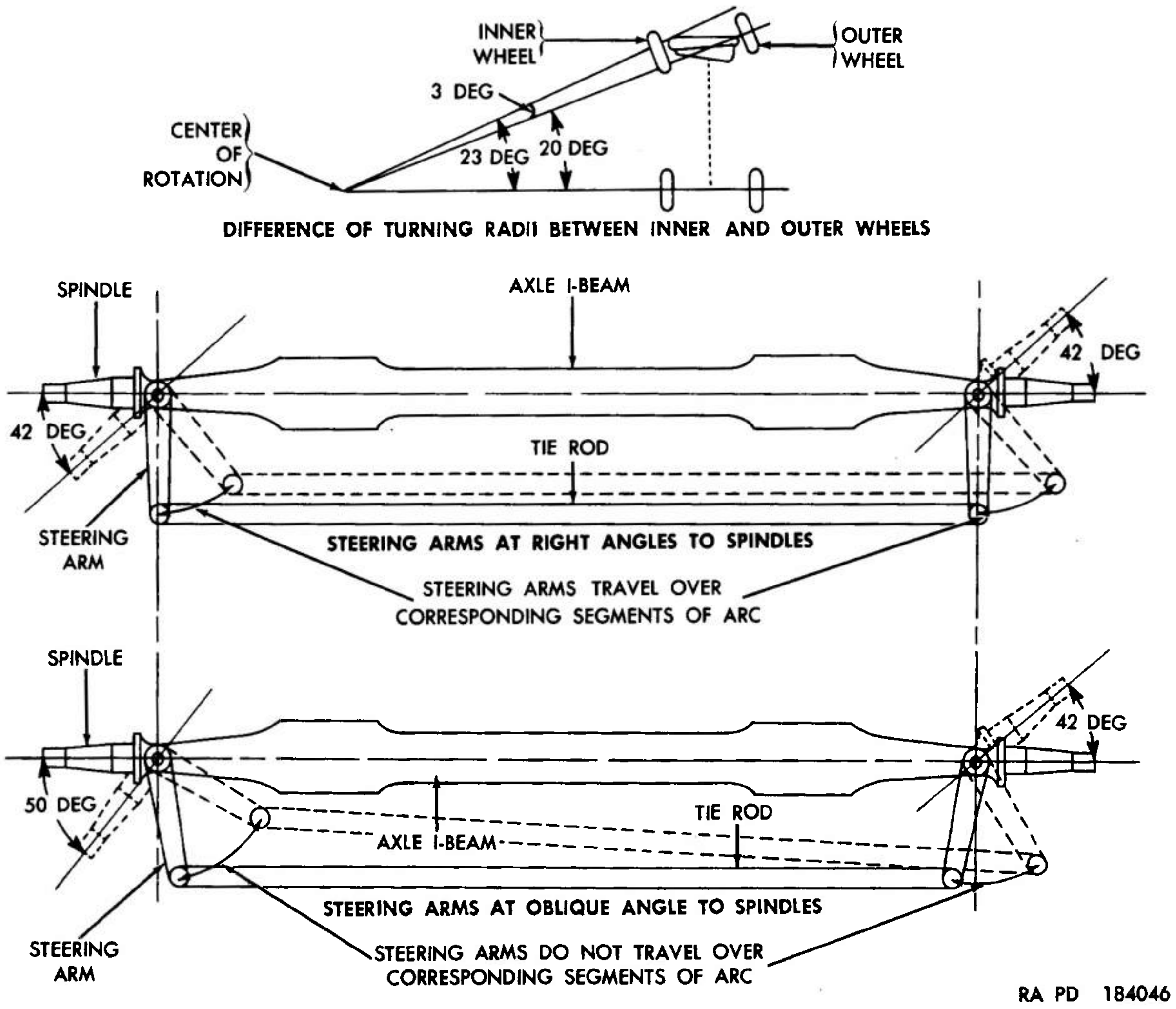


Figure 12-13. ***Steering Geometry for a Beam Front Axle***

in the direction of the front wheel occurs as shown in the plan view. A steering link of incorrect length, OB, is shown in the deflected position, BO_B. It will be noted that in this position, the end of the link is displaced inward from the correct position.

In the plan view it is seen that in order for the steering link to pass through the point OB, the steering knuckle arm must rotate through an angle, ε. Accordingly, the front wheel spindle and wheel must rotate through the same angle. A steering angle whose value is ε is therefore developed.

The most popular form of independent front suspension employs unequal length upper and lower links. For the case in which these links are originally parallel, the theoretically correct length for the steering connection may be established with the aid of Fig. 12-15.

In this figure, the horizontal displacement, of the upper and lower links for a vertical displacement, d, can be shown to be $d^2/2A$ and $d^2/2B$. To avoid scrubbing of the tire on the ground, the point of tire contact should have no horizontal motion. Because the wheel is a rigid body, the horizontal deflection of a point on the wheel must therefore be proportional to its distance above the ground, or

$$d^2/2A = K(h + s) \qquad (12\text{-}8)$$

$$d^2/2B = Kh \qquad (12\text{-}9)$$

and

$$\frac{h}{h + s} \quad \frac{A}{B} \qquad (12\text{-}10)$$

It should be noted that if scrubbing is permissible, the point of no horizontal motion may be selected above or below ground level.

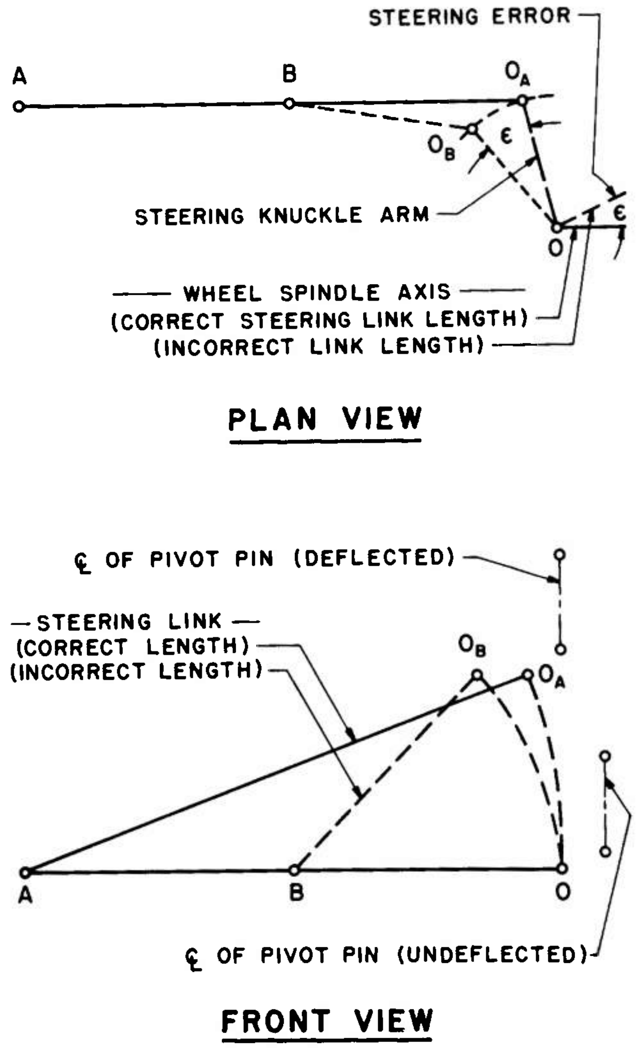


Figure 12-14. ***Effect of Steering Geometry Errors on Motion of the Front Wheels***

For a steering link of length, R, to be installed at a height y above the ground (or point of zero lateral motion), the relation connecting R and y is

$$Ry = A(h + s) \qquad (12\text{-}11)$$

which is a rectangular hyperbola asymptotic to the values $y = o$ and $R = o$.

The introduction of independent front suspension in light vehicles has been accompanied by a forward relocation of the engine to obtain increased passenger and cargo space. This relocation has influenced front suspension design. If the steering connection was kept below the engine it had to be fairly long as was shown in the preceding paragraph. With these long steering arms it was often possible to use only a two-piece cross steering tube without incurring too serious geometry errors. The

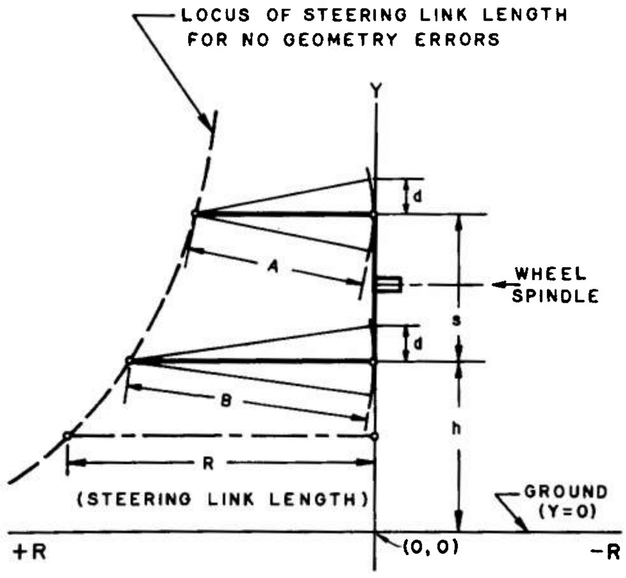


Figure 12-15. ***Steering Link Geometry***

central bell crank lever which acted as the inboard connection to the two sides of the steering arm was, in this case, not too far from the theoretical geometry.

With increasing demands for greater cargo space, the engine was moved further forward. It was now required that the steering arm be placed forward of the wheel centers. To satisfy safety reasons and because of easier Ackermann positioning, the steering arms were also raised above the wheel centers.

Because of the clearance gained between the wheel and the steering pivot axis (the steering pivot slants in at the top), the ball joint between steering knuckle and steering arm could be located farther outboard. This is an advantage because it permits a reasonable angle between the steering pivot arm and steering arm at full lock angle, α, (Fig. 12-16), a factor which aids in the return of the wheel to the center position. Because of the high steering arm location, the steering arm radius must be short, hence a three-piece steering arm or cross tube is required. This three-piece gear can be furnished by using two idler levers mounted on the frame and using a cross tube to link the inboard ends of the steering arm by a cross tube

To maintain good geometry, both direction and length of the steering arm must be correct. The direction of the steering arm is best maintained by canting the steering arm idler pivots so their axes

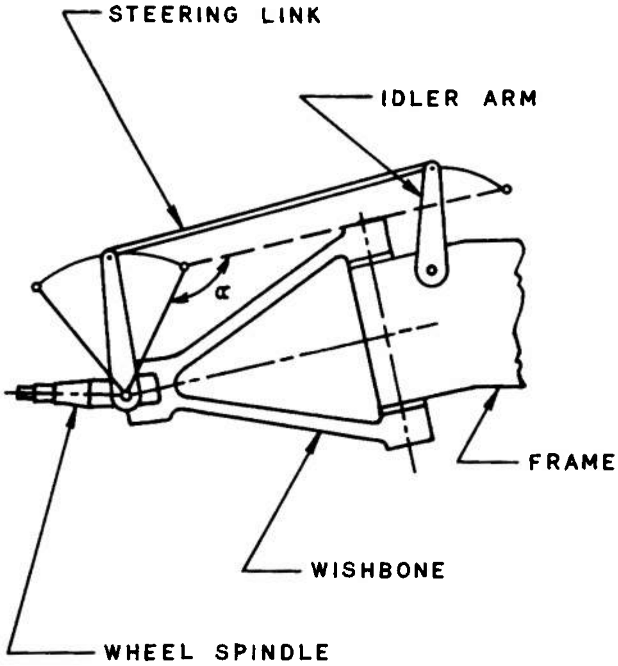


Figure 12-16. ***Plan View Showing Steering Arm Above Wheel Centers***

are parallel with the corresponding axes of the wheel steering pivots. However, because the drop or Pitman arm (when viewed in the side view), moves in a plane parallel to the steering column, road shocks are introduced when the wheel is turned from the center position.

These latter errors are avoided by incorporating two idler levers whose axis are parallel to the steering head, and linked by a cross tube. A separate connection is used to link the Pitman arm to the cross tube or to one idler.

12-6 STEERING WHEEL REACTION

Reactions at the steering wheel result directly or indirectly from road irregularities. If the suspension geometry is such that the plane of rotation of the front wheel changes during rebound, gyroscopic moments about the steering pivot axis will result which may be felt at the steering wheel. In Ref. 15, the gyroscopic reaction for a solid front axle and for a wishbone system, with initially parallel links, are compared. Sprung mass and unsprung mass are assumed to be the same in both cases as are spring and tire spring rates. For the solid axle, the inward tilt (camber) at 2 in. bump is 2° while at 2 in. rebound, the tilt is outward at 2°. For the well-designed parallel link system, these values are 0.5° inward tilt in both cases. In each case, no camber is present at the normal position. Maximum gyroscopic torque for the solid axle system is about 300 lb-ft while for the parallel link system it is only 100 lb-ft.

Gyroscopic torque is given by the equation

$$T = I\omega\Omega \qquad (12\text{-}12)$$

where I is the mass moment of inertia of the rotating mass (lb-ft-sec^2), ω, its angular velocity, and Ω (rad/sec), the rate at which the axis of rotation is being turned. For the solid axle, the maximum torque occurs at near zero spring deflection because at this point rotation of axle center line about its midpoint is most rapid. For the parallel link system, maximum torque occurs towards, but not at, the end of deflection, because up to a point, wheel tilt rate increases with increasing spring deflection. Gyroscopic torque for such a system is zero at the nondeflected case because at this point the wheel does not tilt for an infinitesimal spring deflection. It was found, in the reference, that the vehicle which employed the parallel link system was very free of road shock. To decrease road shock in a steering system, which introduces considerable road shock to the steering wheel, it is necessary to provide damping in the steering linkage.

12-7 STEERING PHENOMENA

12-7.1 LOW-SPEED WOBBLE

The rotation of the front wheels of the vehicle about the kingpin axis against the flexibility of the steering linkage is termed low-speed wobble. It has been found to depend on the caster angle of the front wheels, the weight on the front wheels, flexibility of the frame, and on vehicle velocity. The resulting vibratory motion is transmitted to the steering wheel.

The following conclusions regarding low-speed wobble have been established (Ref. 15):

a. Wobble increases with increasing caster because of the greater pneumatic trail (see Section 8).
b. Wobble is unstable unless damped.
c. A critical velocity exists; however, in practice this is not sharply defined.

In practice, it has been found that wobble decreases under wet conditions because of the reduced frictional forces which result in lower tire self-aligning torques.

Among the ways suggested to overcome low-speed wobble are (Ref. 15):

a. Make engine and mount system natural frequency coincident with wobble natural frequency.

b. Increase stiffness of steering linkage. In one case an increase from 720 lb/° to 1100 lb/° resulted in almost complete elimination.

c. Include sufficient damping to prevent an increase in magnitude of oscillation. The inclusion of 33 lb-ft of friction torque in one case ended wobble. The steering effort with vehicle wheels stationary was increased from 53 to 56.5 lbs.

Because of the variation in caster angle with suspension deflection, the self-aligning torque of the tires will vary. If, in an effort to reduce wobble, caster is reduced to a small value at some conditions of suspension deflection, caster centering action may be nearly absent. The final design to reduce wobble must consider all of the factors mentioned so as to avoid a steering system which is either too heavy or too sensitive.

12-7.2 HIGH-SPEED WOBBLE

High-speed wobble is usually encountered in vehicles having a solid front axle. It is a rocking of the front axle about a horizontal axis near its center. It is influenced by the tire and suspension springs, mass of the front axle, the gyroscopic torque of the front wheels, and velocity of the vehicle.

The frequency of this oscillation is increased by employing stiffer springs, higher tire pressures, and stiffening the frame. Its magnitude is reduced when caster is reduced, roll center is reduced, and it is increased when braking occurs. Damping is employed between frame and axle near the wheels to reduce oscillations.

For an independent wishbone system, both the bump and rebound motion of the wheel can result in an inward tilt of the top of the wheel, hence the disturbing frequency of wheel tilt is twice as great as the frequency of wheel hop. In the case where, for a wishbone system, the tilt is of opposite sign at either end of the travel, high-speed wobble can be made to occur. The wheel hop critical frequency is a function of wheel deflection, decreasing with increasing deflection.

For the case in which the sign of the tilt is the same at bump and rebound, wobble can occur only if the frame is sufficiently flexible. It has been found that high-speed wobble can occur on the road in a system in which the camber changes sign if the steering system is made sufficiently flexible to bring the natural flap frequency of the wheels into coincidence with the wheel hop frequency.

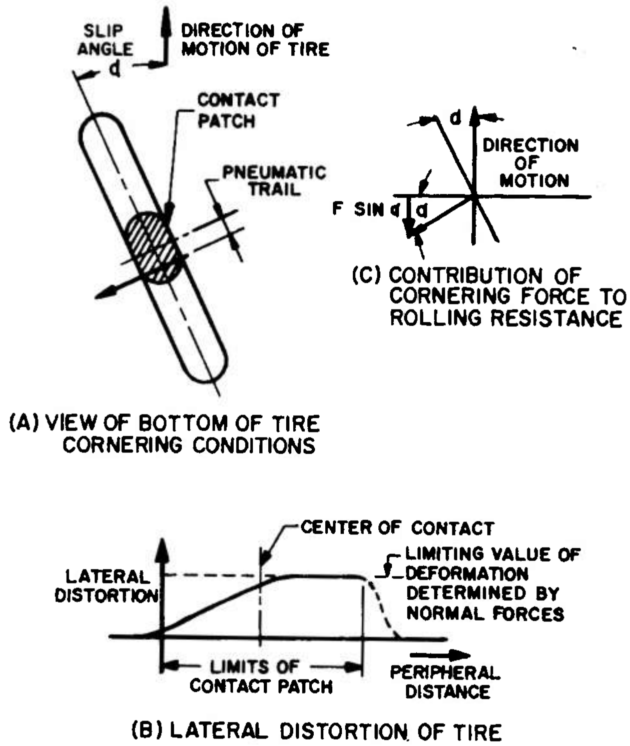


Figure 12-17. *Tire Characteristics During Cornering*

12-8 TIRE EFFECTS ON STEERING

When the center plane of a pneumatic tire is directed at an angle to its direction of motion (Fig. 12-17(a)), the lateral distortion of the tread contact with the road produces a force at right angles to this center plane. This force is called the cornering force, and the angle between the direction of motion and the axis of the tire is called the slip angle. As shown in Fig. 12-17(b), the lateral distortion of the center line of the tire increases with distance along the tire center line starting at the front of the contact patch.

Because the lateral forces in the rearward portion of the contact patch are generally greater than those in the forward portion, the line of action of the cornering force generally intersects the wheel center plane at a point behind a point immediately below the center of the wheel. This rearward displacement is referred to as the *pneumatic trail.* The moment of the cornering force about a vertical line through the center of the contact patch is called the self-aligning torque because it generally tends to lessen the slip angle.

When a loaded tire rolls, continuous distortion of the tread dissipates energy; the power involved is called the horsepower consumption of the tire. Power consumption may be related to the resistance of the motion of the tire, the so-called rolling resistance. This resistance is assumed to act along the line of, but in the opposite direction to, the line of motion. Accordingly, when a tire runs at a slip angle, a component of the cornering force will be added to the rolling resistance which results in an increased tire horsepower consumption.

The tire cornering force is the only force acting on a car which enables it to turn a corner or resist a side wind. Self-aligning torque provides a measure of the force required to steer the car, that is, it gives measure of the feel at the steering wheel.

12-9 MEANS BY WHICH CORNERING FORCE IS DEVELOPED

The cornering properties of tires have been examined by a number of investigators (Refs. 1 and 5). The detailed study of the development of cornering force requires an analysis of tire-to-ground contact. Figure 12-17(b) from Ref. 1, shows the lateral distortion of the tire tread as a function of longitudinal distance along the contact length. Distortion of the tire is the displacement from the center plane of the wheel. This distortion gives an accurate picture of the lateral force developed because under slip angle conditions, motions at the crown and at the shoulder ribs of the tire are approximately equal. This data is replotted for both a 3° and an 8° slip angle, Fig. 12-18, where lateral force as a function of distance along contact length is shown.

The resultant cornering force is proportional to the total area under the curve of Fig. 12-18. The resultant cornering force is seen to act at a point which is behind the midpoint of the contact area. The distance between the center of contact and the intersection of the line of action of the resultant with the wheel center plane is termed the pneumatic trail. As is evident from Fig. 12-18, it is large at low slip angles and decreases with increasing slip angle and cornering forces.

The reason for the variation in cornering force and self-aligning force may be examined by considering the curves of Fig. 12-17. If it is assumed that a limiting value of lateral friction force exists

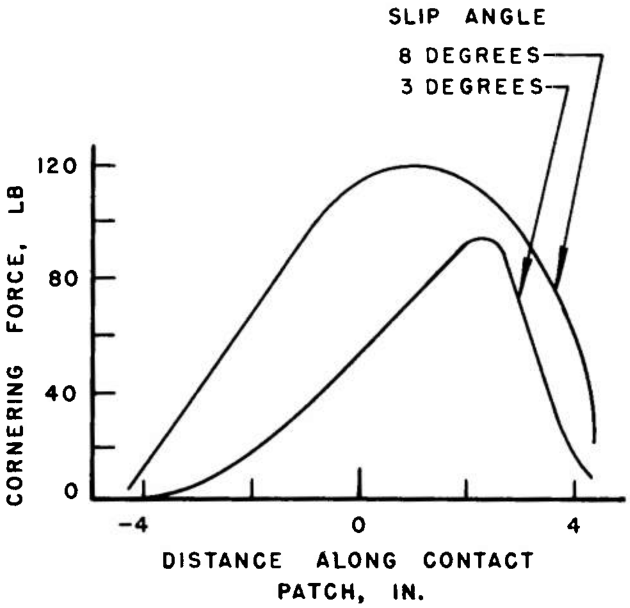


Figure 12-18. Cornering Force as a Function of Distance Along Contact Patch

which is determined by the contact pressure and friction coefficient, an upper limit on the lateral force per unit length of contact length is established as shown in Fig. 12-17. Lateral distortion of a point on the tire tread increases as it moves along the length of the contact path. The sideways force exerted by it on the ground and tire increase with its distortion. When this force reaches the limiting friction value, defined above, it remains at that value for the remainder of its motion along the contact length. It will be noted that the limiting frictional value drops as the end of the contact patch approaches because the tire is separating from the ground and its contact pressure is decreasing.

At low slip angles, the lateral displacement is such that the limiting frictional value is not attained until very late in the contact patch. Accordingly, the total cornering force remains low and the pneumatic trail is large. For high slip angles, the lateral displacement corresponding to the limiting frictional value is attained at an earlier point in the contact patch. The total cornering force is therefore high and because high values of lateral force are developed ahead of the center point of the contact patch, the pneumatic trail is decreased.

Experimental results from Ref. 1 show that the force system acting on the tire contact patch of a free rolling tire may be considered to be a verti-

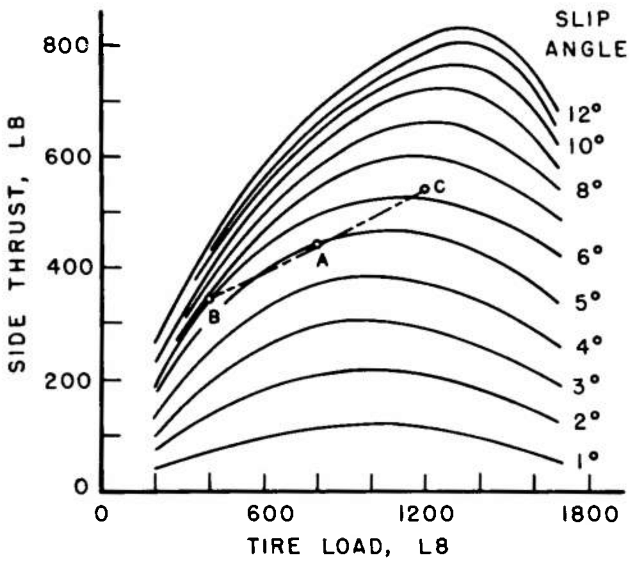


Figure 12-19. *Side Thrust vs Load 6.00 × 16 Tires at 29 lb/in²*

cal force acting at the center of the contact patch which force is comparable to be the effective upward force from the ground, and a force in the ground plane discussed in the following paragraph.

The resultant of the horizontal forces in the ground plane has a line of action to the rear of the midpoint of the contact point by a distance equal to the pneumatic trail. Its direction is mainly lateral, however, a slight backward component is present. This small backward component arises from tire rolling resistance or drag while the lateral component arises from cornering forces. The backward component is the same in magnitude as that developed in a freely rolling tire.

The variation of cornering force, self-aligning torque and horsepower consumption with slip angle and load was investigated in Ref. 5. In Fig. 12-19 is shown the variation of cornering force with tire loading for different slip angles. Cornering force for a given tire load increases with slip angle as described previously. Cornering force for given slip angle increases with tire loading until a maximum cornering force is developed. For further tire loading, cornering force decreases, as the frictional force in the contact patch changes from a static to slipping type.

Self-aligning torque, at constant slip angle, increases with tire loading. At constant load, self-aligning torque increases with slip angle up to a slip angle of 6°. Above this value, self-aligning torque decreases, and, if the angle is increased sufficiently, becomes negative. This angle of peak torque results as follows. At low slip angles, the line of action of the cornering forces lies near the rear of the contact patch. With increasing slip angle the resultant moves forward as discussed previously. Coincidently, the maximum cornering force developed approaches the limiting frictional value. Maximum self-aligning torque occurs at that slip angle at which the product of the magnitude of the resultant cornering force and its lever arm about the center point of the contact patch is a maximum.

Horsepower consumption increases with both load and slip angle. The increase with load results from increased tire deflections with a corresponding higher rolling resistance. The increase with slip angle results from the increased rolling resistance contributed by the component of the cornering force in the direction of tire motion, as shown in Fig. 12-17(b).

The effects of speed on cornering power, self-aligning torque and horsepower were also studied by the reference. Effects on all but the latter were negligible. Horsepower increased linearly with speed. This is explained by assuming a constant rolling resistance. Horsepower is then proportional to the product of speed and rolling resistance.

The effects of inflation pressure on the three tire characteristics are shown in Ref. 1. Cornering force was shown to vary linearly with inflation pressure for constant slip angle. This results because the lateral force required to distort the tire to a given slip angle is greater for tire having a greater inflation pressure.

Self-aligning torque decreases with inflation pressure. This results because the greater forces required to distort the tires laterally, for a given slip angle, cause the cornering force to attain the limiting frictional force near the front of the contact patch. Horsepower consumption at low slip angles, varies only slightly with inflation pressure because the decreased rolling resistance is offset by the rearward component of the increased cornering force. At high slip angles, horsepower consumption increases because of this latter component.

The effects of tire construction and configuration on tire cornering force, self-aligning torque, and horsepower consumption were studied in Ref. 1. The effect of increasing the rim width is to increase cornering force, to have very little (slightly decreasing) effect on self-aligning torque and to have negligible effect on horsepower consumption when the tire is run at a constant slip angle. The increased cornering force results from increased resistance to tread distortion.

Self-aligning torques decrease slightly because at higher cornering forces more of the contact patch must generate forces at the limiting frictional values. This can occur only in the forward part of patch and so the line of action of the resultant cornering force moves forward.

The effect of an increase in rim diameter is to increase cornering force, self-aligning torque and horsepower consumption.

The effect of tire section width on the tire characteristics is as follows. An increase in section width results in an increase in cornering force and self-aligning torque and in a negligible change in horsepower consumption when measured at the cause a greater value of limiting friction force can be developed by the tire as a result of its increased contact area. Even if the resultant of the total cornering force remains at the same distance behind the center of the contact patch, the higher magnitude of the resultant cornering force will result in an increased self-aligning torque.

The effect of an increase in number of plies on cornering force and horsepower is negligible. The self-aligning torque at a given slip angle is less for a 6-ply tire than for a 4-ply tire. The decrease in self-aligning torque with increasing number of plies may be explained by the same mechanism resulting from an increase in rim width. Further study, however, is required to explain why cornering force does not change with number of plies. A tentative explanation is based on the assumption that contact area decreases slightly with number of plies. Under such an assumption, the limiting friction value attainable decreases slightly hence cornering forces at the rear of the contact patch are less which in turn tends to decrease the self-aligning torque. Cornering forces at the front of the contact patch are increased somewhat by the greater resistance to lateral deformation resulting from an increased number of plies. Total cornering force, therefore, remains approximately constant.

12-10 REAR END STEERING EFFECTS

The lateral behavior of a vehicle is influenced by guidance from the rear wheels as well as that from the front wheels. Rear wheel steering, controlled by the driver, has been employed sparingly because it has been found unstable. Only at low speeds has it been found safe.

One of the causes for the unstable nature of rear wheel control results because the lateral force between the road and tire reverses during the steering maneuver. After a steerable rear tire is initially moved to its steering position, the lateral force on the tire is toward the outside of the turn. As the vehicle begins to turn, the tires begin to develop a cornering force by sliding outward. The lateral force on the tire is then directed inward toward the inside of the turn. The tire tread must roll under the rim during this reversal and it is this action which is felt by the driver.

When a conventional vehicle carries a heavy load behind the rear axle, a similar roll-under effect can occur. Turning the front wheels can result in a yawing motion about a point behind the rear axle in which the rear wheels move toward the center of curvature of the path the vehicle is starting to follow. As the vehicle begins to move along the curved path, the direction of lateral forces on the rear tires reverses.

Yawing of the conventionally steered vehicle is determined by the cornering forces of both the front and rear tires. Each pair of tires provides a lateral force to resist the side forces and doing so slips sideways. Even when the tire planes remain parallel to the longitudinal axis of the vehicle, the slip angles of the front and rear ends of the vehicle can be different. These differences give rise to the over or understeering condition.

The principal rear steering effect introduced by a conventional solid rear axle is that resulting when the center of the axle rotates slightly, in the horizontal plane, about a vertical axis through its midpoint. Such a rotation can result when the vehicle rolls, causing the rear suspension springs to deflect. If the point of attachment between axle and spring moves backward or forward as the vehicle rolls, the axle can rotate in the horizontal plane about its midpoint as shown in Fig. 12-20. Reference 4 states that the softer suspension systems now in use have aggravated this skewing more than is generally realized. In that reference, it is also pointed out that large lateral motions in the presence of side wind forces can often result from rear axle steering caused by roll effects, rather

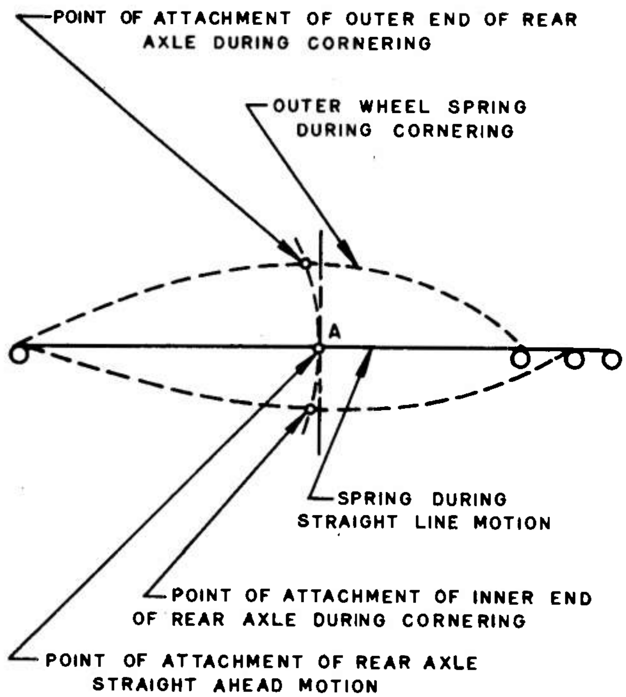


Figure 12-20. *Roll Steer Effects in Live Rear Axle Resulting from Spring Deflections (Asymmetrical Spring Eyes)*

than from the aerodynamic characteristics of the body.

The steering effects in the geometry of the rear suspension could be used in conjunction with known front end steering effects to obtain desired characteristics. As an example, a vehicle with a large understeer could be made responsive by using an oversteering rear axle. In this manner the effects of an oversteering front geometry could be masked by a geometrical roll understeer at the rear. In Ref. 4 it is pointed out that such compensating means do not avoid the inherent tire roll-under discussed above. It was the opinion of the reference that the rear axle should be maintained in as close alignment as possible and rear steering effects avoided.

The increasing use of lower spring rates has given rise to higher roll angles. For a roll angle to lateral acceleration ratio of 1.0 to 1.25°/g, the vehicle will roll to about 4° to 5° under hard cornering at 4g. This is approximately equal to the maximum desirable limit. As a result, as even softer suspensions are introduced, antiroll torsion bars will be required.

By incorporation of such an antiroll bar and by increasing the lateral distance between rear springs, the rear suspension can be stiffened in its roll resistance while maintaining its ride stiffness. Because comfort depends on both roll and ride qualities and upon the requirement that their natural frequencies be close, this means for reducing roll is somewhat limited in scope.

An alternative means of reducing roll is to reduce the roll producing couple. The magnitude of this couple is given by the product of the lateral disturbing force and the distance between its point of application and the roll axis of the sprung mass. The roll axis is defined as the straight line joining the front and rear roll centers. By raising the roll axis, the roll couple and the roll angle are reduced, hence a higher roll axis permits the employment of a reduced stiffness in roll without an increase in roll angle for a given lateral force.

Longitudinal radius arms, in conjunction with antiroll bars and proper geometry, can provide excellent control of the rigid rear axle.

12-11 ROLL ANGLES

The behavior of a vehicle as it travels along a curved path is influenced by the changes in tire loadings which result from vehicle roll. In this paragraph, the locations of the vehicle roll centers and their effect on tire loading is discussed.

The roll center of the spring mass in a given plane may be obtained as that point in the plane about which the sprung mass rotates. In general, the roll center height at either end of the vehicle is different, hence the line joining these points, ordinarily called the roll axis, is not horizontal.

12-11.1 ROLL CENTER LOCATION

The roll center location for a number of the more common front and rear suspension systems is shown in Figs. 12-21 and 12-22 from Ref. 2. As an example of the means used to locate the roll center, that for the independent type is examined.

Point 1 on the upper link (see Fig. 12-21(a)) may be considered to be a point on both the upper link and the wheel hence it is their instantaneous center. Points 4 and 5, the connecting points to the sprung mass of the upper and lower links, respectively, are also instantaneous centers, in this case, link-to-chassis. According to Kennedy's theorem, which deals with the instantaneous centers of three bodies moving with respect to each other, only three centers exist and these centers lie on a straight line. Accordingly, for the three body sys-

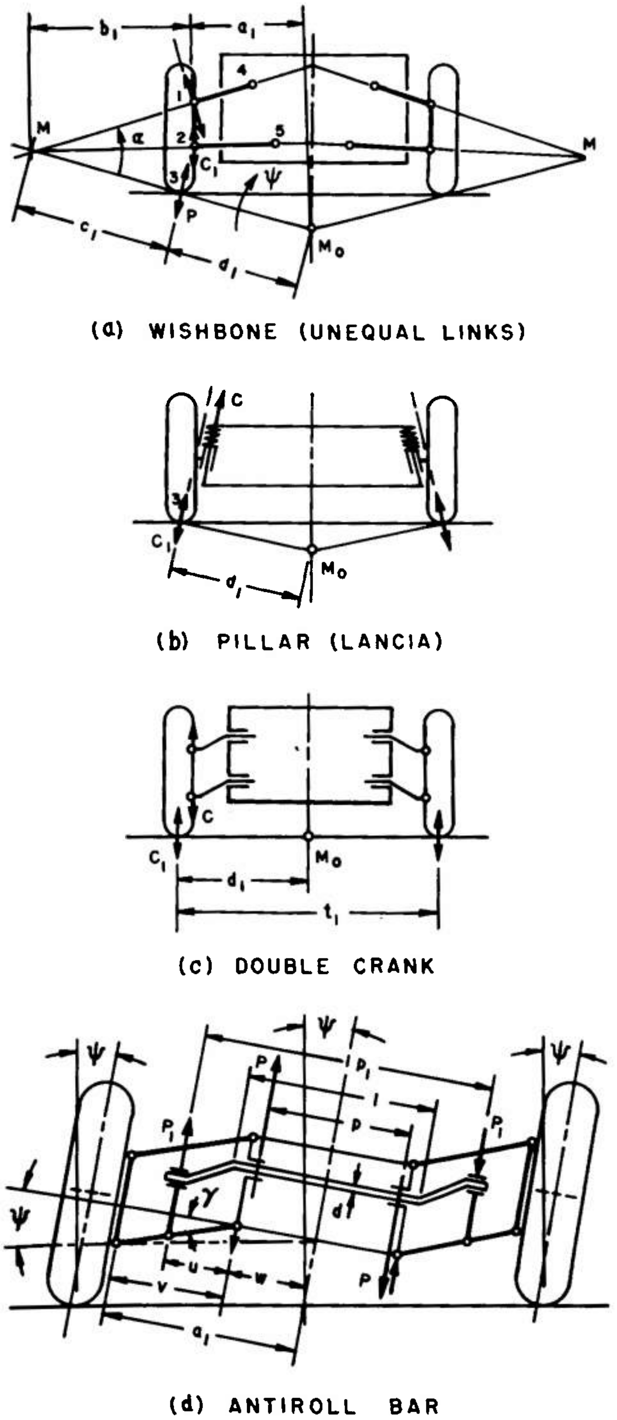


Figure 12-21. ***Roll Centers and Restoring Moments for Front Suspensions and Antiroll Bars***

tem composed of the sprung mass, the upper links, and the wheel, the third center must lie on the extension of the line connecting Points 1 and 4. Similarly, for the system composed of the sprung mass, lower link, and wheel, the remaining instantaneous center is located on an extension of the line 5-2. These extensions intersect at the point M which is the instantaneous center of wheel and spring mass.

Point 3 at the point of tire contact may be considered to be a point on the ground and on the tire, and hence an instantaneous center in the wheel, ground, sprung mass system. The three instantaneous centers of this system must lie on the line M-3 extended. Because of symmetry, the instantaneous center of sprung mass and ground, called the roll center, is located on the vehicle centerline. In the illustration shown, the roll center is shown below ground level, however, by a proper choice of suspension geometry, this point may be raised.

With horizontal parallel links, for example, the instantaneous center, M, lies at infinity and the line M-3-M_0 is horizontal. Accordingly, the roll center is at the road surface.

Wheel motion in the guided sleeve mounting type suspension (Fig. 12-21(b)) is parallel to the axis of the sleeve. In the arrangement shown, the roll center lies below the road surface.

In the trailing link or double crank suspension system (Fig. 12-21(c)) the wheel motion is vertical, hence the roll center lies at the road surface. In both of the above systems, the roll center is found by noting that wheel motion is perpendicular to the hypothetical link joining the roll center and the point of tire contact.

For the beam axle suspended by leaf springs (Fig. 12-22(c)) the roll center is at approximately the height of the spring anchorages. This location is influenced by the lateral flexibility of the spring mounting.

When a cross link is used for lateral location of a beam axle (Fig. 12-22(b)) the roll center is at the height of the attachment of the link to the sprung mass.

For the DeDion rear axle, the roll center lies at the point of the axle about which the sprung mass rotates (see Fig. 12-22(c)).

For the swing axle, the roll center is located along a line joining the point of tire contact, Point 3 of Fig. 12-22(d) and the swing axle joint, point M. The roll center, M_0, is therefore located above the swing axle center line when it is in the horizontal position.

Roll center height varies with spring deflection

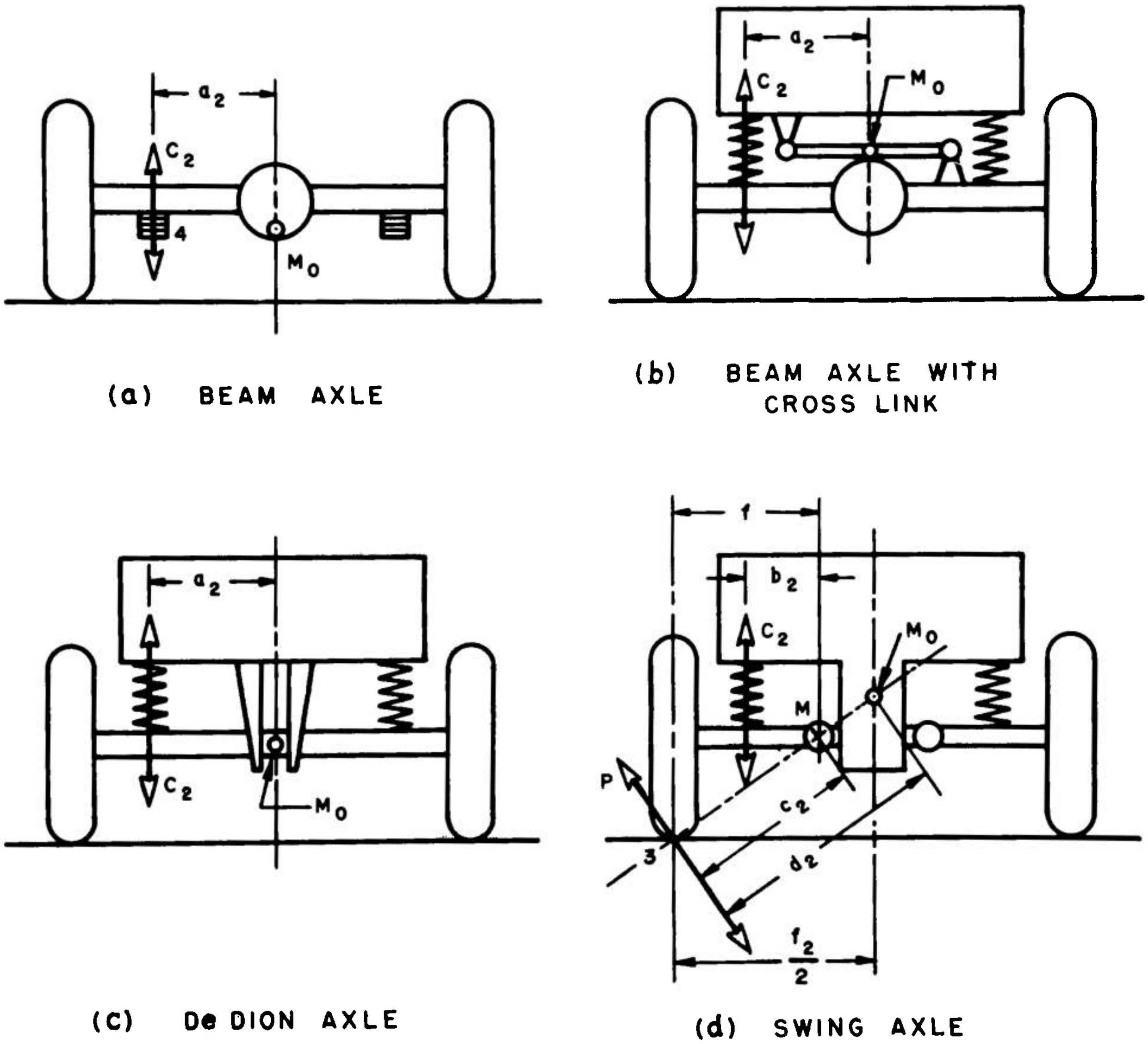


Figure 12-22. *Roll Centers and Restoring Moments for Rear Suspension Systems*

for suspensions employing links, the beam axle with lateral location link, and for the swing axle.

12-11.2 ANTIROLL BAR

In order to reduce the roll angle and to influence the dynamic wheel loads during cornering, an antiroll bar connecting opposite wheels is sometimes used. The function of the antiroll bar is to produce a restoring moment when the deflections of the springs are unequal. During cornering, the outboard springs are more highly loaded than the inboard springs. The transverse roll bar transfers some of the load from the outboard springs to the inboard spring thus reducing the roll angle for a given side force.

12-12 RESTORING ROLL MOMENTS

With the roll center located, it is possible to determine the value of the restoring moment exerted by the suspension system as the vehicle rolls. Tabular values of the restoring moments introduced by the more common suspension systems are shown in Table 12-1.

It will be noted that in every case, the restoring couple is proportioned to the roll angle or

$$M = K\varphi \tag{12-13}$$

where M is the restoring moment and φ is the roll angle of sprung mass, radians.

In some suspension systems, the effects of centrifugal force on the unsprung mass causes it to

TABLE 12-1 RESTORING ROLL MOMENTS INTRODUCED BY VARIOUS SUSPENSION SYSTEMS

System	Roll Moment*
Independent Linkage (Unequal Wishbones): (Fig. 12-21(a))	$2\left(\frac{b_1 d_1}{c_1}\right)^2 C_1 \phi$
Independent Linkage (Parallel Horizontal, Equal Wishbones)	$\frac{t_1^2}{2} C_1 \phi$
Guided Sleeve or Double Crank: (Fig. 12-21(c))	$2d_1^2 C \phi$
Swing Axle: (Fig. 12-22(d))	$\frac{1}{2}\left(\frac{b_2 t_2}{f}\right)^2 C_2 \phi$
Beam Axle: (Fig. 12-22(a))	$2 C_2 a_2^2 \phi$
Antiroll Bar: (Fig. 12-21(d)) Independent Linkage	$\frac{a_1 p_1 u}{v} C_3 \phi$
Antiroll Bar: Beam Axle	$\frac{p_1^2}{2} C_3$

* *Nomenclature:*

C_1, spring rate of lower suspension arm, lb/in. of motion of Pt. 2
C, spring rate of suspension spring, lb/in. of motion of Pt. 3
C_2, spring rate of suspension spring, lb/in. of motion of Pt. 4
C_3, spring rate of antiroll bar, lb/in.
ϕ, roll angle of sprung mass, rad

exert a roll couple on the sprung mass or to change the dynamic loading of the tires. These roll effects are summarized in Table 12-2. It should be noted that because the roll center is above the wheel center for the swing axle, it exerts a righting moment on the sprung mass. The centrifugal moment exerted by the beam axle is not exerted on the sprung mass; however, it does influence the dynamic wheel loads.

TABLE 12-2 ROLL MOMENTS OF UNSPRUNG MASS

System	Moment	Note
Independent	$\mu W_i r$	Tilting
Swing Axle	$-\mu W_2(n-r)$	Restoring
Beam Axle	$\mu W_2 r$	Influences wheel loading only

where μW_i is the centrifugal force acting, r is the wheel radius, and n, the height of the roll axis.

In general terms, from Table 12-1, the suspension system restoring moments may be expressed as

$$M_{Si} = K_{Si}\varphi \qquad (12\text{-}14)$$

where $i = 1$ for the front suspension, $i = 2$ for the rear suspension system, and $i = 3$ for the antiroll bar.

From Table 12-2 the unsprung mass roll moment, MT, may be expressed as

$$M_T = C_T\mu W_i = C_T\mu KW = K_T\mu W \qquad (12\text{-}15)$$

where

W_i is the weight of the unsprung mass under consideration, lb

W is the weight of the sprung mass under consideration, lb

μ is equal to $\frac{\omega^2 r}{g}$, lb

ω is the angular velocity of the (sprung or unsprung) mass, rad/sec

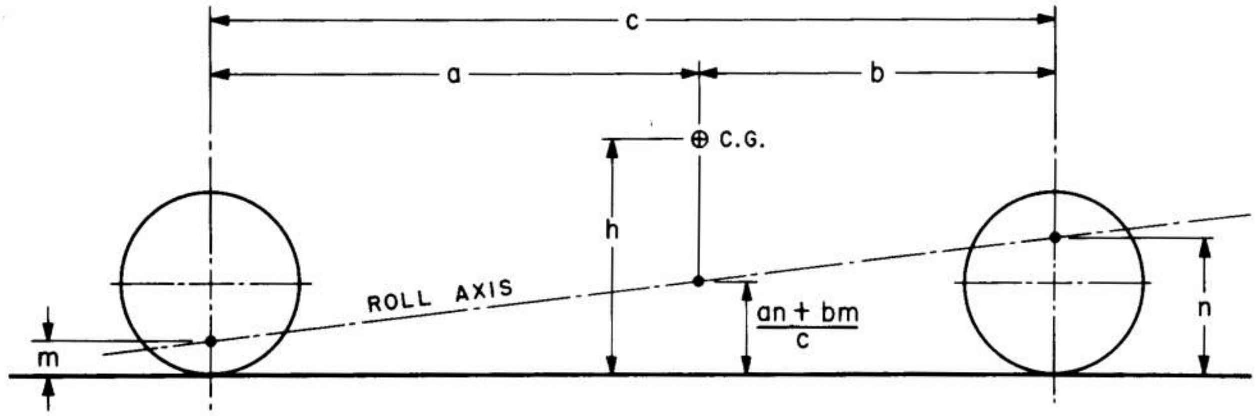


Figure 12-23. Roll Axis Location

r is the radius of curvature, ft

g is the gravitational constant, $\frac{\text{ft}}{\text{sec}^2}$

C, K are constants

The roll angle of the sprung mass for a given suspension system under the effects of a centrifugal cornering load is therefore

$$\phi = \frac{\mu W(K_c + K_T)}{K_{S1} + K_{S2} + K_{S3}} = \mu W K_R \qquad (12\text{-}16)$$

where $\mu W K_C$ is the centrifugal roll moment from the sprung mass.

The roll angle is therefore directly proportional to the centrifugal cornering load. It is assumed in this discussion that the tires and frame are rigid.

The roll angle of a given vehicle under stated conditions can be calculated by summing the moments about the roll axis, acting on the vehicle (Fig. 12-23). As an example, the equation to determine the roll angle, φ, of a vehicle with an independent front suspension having unequal wishbones and an antiroll bar and a beam axle at the rear would be determined as follows (referring to Tables 12-1 and 12-2 and Fig. 12-23):

$$\mu W\left[h\frac{an+bm}{C}\right] + \mu W_1 r = M_{S_1} + M_{S_2} + M_{S_3} \qquad (12\text{-}17)$$

where

$$M_{S_1} = 2\left(\frac{b_1 d_1}{c_1}\right)^2 C_1\phi, \text{ ft-lb}$$

$$M_{S_2} = 2C_2 a_2^2 \varphi, \text{ ft-lb}$$

$$M_{S_3} = \left(\frac{a_1 p_1 u}{v}\right) C_3\phi, \text{ ft-lb}$$

or

$$\mu\left\{W\left[h\frac{an+bm}{C}\right] + W_1 r\right\} = \left[2\left(\frac{b_1 d_1}{c_1}\right)^2 C_1 + 2a_2^2 C_2 + \left(\frac{a_1 p_1 u}{v}\right) C_3\right] \qquad (12\text{-}18)$$

Finally

$$\phi = \frac{\mu\left\{W\left[h\frac{an+bm}{C}\right] + W_1 r\right\}}{2\left(\frac{b_1 d_1}{c_1}\right)^2 C_1 + 2a_2^2 C_2 + \left(\frac{a_1 p_1 u}{v}\right) C_3}, \text{ rad} \qquad (12\text{-}19)$$

12-13 DYNAMIC WHEEL LOADS

Under the action of the roll moments introduced by centrifugal forces in cornering the loads imposed on the tires are changed. The outer tire is subjected to an incremental load increase, ΔF, (front wheel), or ΔR, (rear wheel) while the inner tire will be unloaded by an equal amount. To calculate these incremental loads it is first necessary to calculate the roll angle of the sprung mass as shown above.

Under steady-state cornering conditions, the change in tire loading caused by the roll moments acting is given by the equations below.

Front wheels:

$$\Delta F w = M_{S1} + M_{C1} + M_{S3} + M_{T1} \qquad (12\text{-}20)$$

Rear wheels:

$$\Delta R w = M_{S2} + M_{C2} + M_{S3} + M_{T2} \qquad (12\text{-}21)$$

where w is the vehicle track distance, ft

M_{S1}, M_{S2}, M_{S3} are the restoring motions of the front and rear suspensions, and the antiroll bar.

M_{T1}, M_{T2} are the tilting moments arising from centrifugal force on the front and rear unsprung mass.

M_{C1} is the roll moment upon the unsprung mass introduced by the centrifugal loading of the sprung mass acting at the front roll center (see Fig. 12-23)

$$M_{C1} = \mu \left(\frac{b}{c}\right) mW \tag{12-22}$$

M_{C2} is the roll moment upon the unsprung mass introduced by the centrifugal loading of the sprung mass acting at the rear roll center.

$$M_{C2} = \mu\left(\frac{a}{c}\right)nW \tag{12-23}$$

The load transfer at the front wheels is therefore

$$\Delta F = \frac{1}{w}\left[K_{S1}\phi + \mu\frac{b}{c}mW + K_{S3}\phi + K_T\mu W\right]$$

$$= \frac{1}{w}\left[(K_{S1}+K_{S3})\phi + \mu W(K_{L1})\right] \tag{12-24}$$

where

$$K_{L1} = \left(\left(\frac{b}{c}\right)m + K_T\right) \tag{12-25}$$

The load transfer at the rear wheels is

$$\Delta R = \frac{1}{w}\left[K_{S2}\phi + \mu\frac{a}{c}nW + K_{S3}\phi + K_T\mu W\right]$$

$$= \frac{1}{w}\left[(K_{S2}+K_{S3})\phi + \mu W K_{L2}\right] \tag{12-26}$$

where

$$K_{L2} = \left(\left(\frac{a}{c}\right)n + K_T\right) \tag{12-27}$$

With these equations, the load transfer for a given suspension system can be determined for a specified roll angle and centrifugal loading. In general, load transfer is found to be directly proportional to both angle and centrifugal loading and may be represented as

$$\Delta L = C_1\varphi + C_2\mu W \tag{12-28}$$

where C_1 and C_2 are constants.

As shown in the preceding section, for a given vehicle under stated conditions, the load transfer at the front and rear wheels can be determined by substituting the proper values for the constants in Eqs. 12-26 and 12-27.

If it is assumed in the determination of the slip angle that both tires have the same slip angle the slip angle under load transfer conditions is always greater than for the case in which the two tires are equally loaded.

The effects of load transfer are shown graphically in Fig. 12-19. When one pair of wheels carries a load of 1600 lb, each tire carries a load of 800 lb under conditions of no load transfer. Under these conditions and when a slip angle of 5°, Point *A*, is developed, each tire develops a lateral thrust of 440 lb. If, under rolling conditions, a weight transfer of 400 lb occurs, the inner tire carries 400 lb and the outer tire carries 1200 lb. Because the two tires must run at the same slip angle, the resulting slip angle is greater to develop the same cornering force. From Fig. 12-19, the new slip angle is 6.33°, with inner tire, Point *B*, supplying 340 lb of lateral thrust and the outer tire, Point *C*, supplying 540 lb. The total thrust is therefore 840 lb.

12-14 STABILITY MARGIN

To provide a convenient process for the estimation of vehicle stability, the term stability or static margin was introduced by Ref. 3. Stability margin is defined as the horizontal distance, expressed as a percentage of the wheelbase length, between the center of gravity of the vehicle and the center of reaction of the road forces.

The center of reaction of the road forces passes through a line designated as the neutral steer line. This line is defined as a line in the fore and aft center plane of the vehicle anywhere along which a lateral force may be applied without producing yawing of the vehicle. It is assumed the limit of adhesion of the tires is not exceeded.

Should the center of gravity of the vehicle be behind the neutral steer line, the vehicle will turn towards a lateral force applied at the center of gravity. Such a vehicle is considered to be unstable. In a stable vehicle, the neutral steer line is behind the center of gravity. This is the equivalent to understeer.

It is desirable that the vehicle be stable for case of the fixed steering wheel, however, because of the self-aligning torque of the front tires, there is no need for an excessive stability margin. Excessive stability margins result in greater, stabilizing motions which, in turn, slow vehicle response. Reference 3 recommends a maximum static margin of 4 to 6 percent.

12-15 AERODYNAMIC EFFECTS

The aerodynamic forces acting upon a moving vehicle may be resolved into three components par-

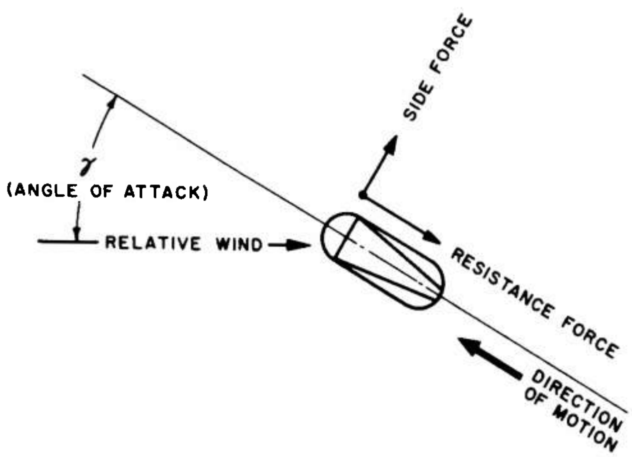


Figure 12-24. ***Aerodynamic Forces Acting on an Automotive Vehicle***

allel to the longitudinal, lateral, and vertical axes of the vehicle. These forces are referred to as the resistance or drag force, the side force, and the lift force, respectively. Aerodynamic forces are not of importance at lower speeds and can probably be neglected at speeds less than about 50 mph.

These force components do not act at the center of gravity of the car, hence they give rise to pitching, yawing, and rolling movements about this point. At higher speeds the lift and pitching moments generated may cause a weight transfer from front to rear wheels with a consequent change in the neutral steer line.

In the present discussion only the effects of the resistance force, the side force, and the yawing moment are examined. These have the greatest influence on the steering problem.

In Ref. 3 it is shown that for the usual case, where the longitudinal axis of the vehicle closely coincides with the direction of motion, the most convenient resolution of aerodynamic forces is along, and perpendicular to, the center line of the vehicle (Fig. 12-24). In that figure, the resultant aerodynamic force is shown acting at the center of pressure.

The *center of pressure* is defined as follows. When an area is subjected to a pressure distribution, a point exists in the area through which the entire force, due to the pressure, could be concentrated with the same external effect. If the pressure is uniformly distributed over the area, the center of pressure coincides with the centroid of the area.

If, for a given vehicle, the center of pressure is not colinear with the neutral steer line, an aerodynamic moment acts on the vehicle. The moment developed is termed the yawing moment.

Aerodynamic coefficients for aircraft are measured perpendicular to, and along, the direction of relative wind; while for automotive vehicles, the aerodynamic coefficients are measured perpendicular to, and along, the longitudinal axis of the vehicle. The resistance force, R, and the side force, S, are functions of the angle of attack, γ, for a given relative wind, Fig. 12-24. The resistance force has been discussed briefly in Chapter 5, so the present comments will be limited to the effect of these forces on vehicle control.

Since the hypothetical point of action of the resistance force, R, is at the center of pressure of the frontal area, and since this point is some distance, h_a, above the ground level the axial force on the vehicle due to the relative wind will result in dynamic axle reactions. The resistance force normally tends to increase the rear-axle effective weight in proportion to the height of the center of pressure. However, the determination of exact relationships is difficult at high vehicle speeds when lift forces appear. Such forces diminish the axle loadings, particularly on the front axle. To counteract this effect, the bodies of high speed vehicles are designed to create vertical force components that oppose the lifting forces. These considerations are not of great importance with present military land vehicles since the vehicle speeds are relatively low, and control problems related to the drag resistance are not serious.

The side forces acting on a vehicle, as a result of a relative wind at some angle, γ, are of importance for military vehicles since they affect the general stability of the vehicle and the degree of corrective action required by the operator.

The location of the center of pressure, as defined above, is significant to vehicle stability since the relative positions of the center of pressure, the center of gravity, and the neutral steer line determines vehicle response to side forces.

12-16 SUDDEN CHANGES IN WIND INTENSITY

In general, steady side winds should not greatly inconvenience the driver. However, the large responses which may occur as a result of sudden changes in wind forces will prove more difficult to control.

The behavior of a vehicle in the presence of a

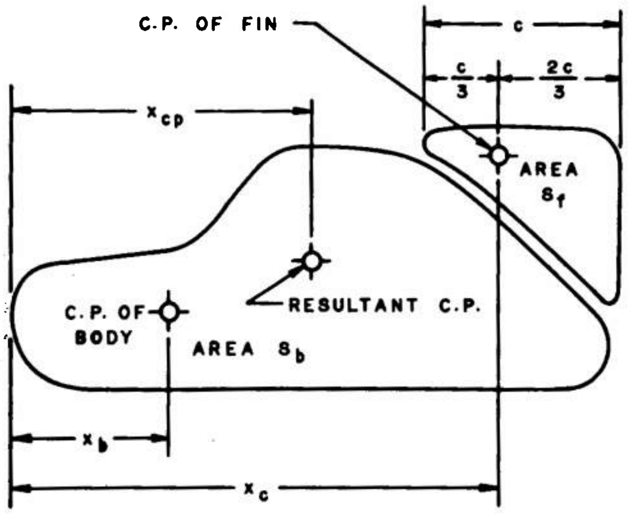


Figure 12-25. *Effect of Adding Fin on Center of Pressure Location*

side wind force depends upon its aerodynamic configuration. In the case of the unstreamlined vehicle, the center of pressure is normally close to the neutral steerline. In the case of the streamlined vehicle, the center of pressure is located near the front axle. For a streamlined and stabilized vehicle,* the center of pressure can be located behind the neutral steerline.

In the presence of a side wind force, the unstreamlined vehicle slips sideways away from the wind. The streamlined vehicle, because the wind force acts almost at the front axle, develops a greater slip angle at the front than at the rear tires. Accordingly, it begins to yaw at an increasing rate until the driver applies correction.

The stabilized streamlined vehicle slips sideways away from the wind. However, because the center of pressure is located behind the neutral steerline, the rear wheel slip angle is greater than the front wheel slip angle. As a result, the vehicle yaws so as to head into the wind and returns towards its original course. When the vehicle crosses its original course, the driver must apply correction.

In Ref. 3 the effect moving the center of gravity with respect to the center of pressure was examined. For a vehicle having a center of pressure at the center of the wheelbase tests were run with (a) 60 percent of vehicle weight on the front axle (center of pressure 10 percent of wheelbase behind neutral steer line), and (b) 70 percent of weight on front wheels.

In case (a), the vehicle turned gradually into the wind in about 1.25 sec. This was accompanied by very little downwind drift. In case (b), the vehicle turned violently into the wind.

Tests were also run to show the effect of wind velocity on vehicle motions. It was found that the time for the vehicle to yaw into a returning course was independent of wind velocity. Lateral drift, however, rose rapidly with wind velocity.

In the ideal case, the vehicle is completely unresponsive to side wind forces. In an actual design, the vehicle should be proportioned to produce comparatively slow reactions to side forces. In this manner, the driver has time to make corrections. Reference 3 has shown that the center of pressure should be located between 2 and 10 percent of the wheelbase behind the neutral steer line to permit the vehicle to turn slowly into the wind.

12-17 LATERAL EQUATIONS OF MOTION

In studying the lateral motion of the vehicle as it negotiates a turn, the influence of all of the effects described previously must be considered. These effects include: cornering forces and moments generated by the tires, aerodynamic forces and moments on the vehicle, rear and steering effects, change in tire rolling resistance due to roll, weight transfer effects due to roll, shock absorber forces, camber change effects and the momentum effects produced by the vehicle mass.

12-17.1 CORNERING FORCES

In the section dealing with the lateral force capability of the pneumatic tire it was shown that both cornering force and self-aligning force were functions of load, camber angle and slip angle.

Previously it was stated that if a tire runs at a given slip angle, both its cornering force and self-aligning torque will increase to a maximum value and then decrease as the load on the tire increases. As camber becomes more positive, both cornering power and self-aligning torque decrease.

12-17.2 SLIP ANGLE

The slip angles of the front and rear tires of a vehicle traveling at a velocity, V (ft/sec), while undergoing a lateral velocity, v (ft/sec), and a yaw angular velocity, r (rad/sec), is shown in Fig. 12-26.

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* Fins can be used to bring the center of pressure rearward as shown in Fig. 12-25.

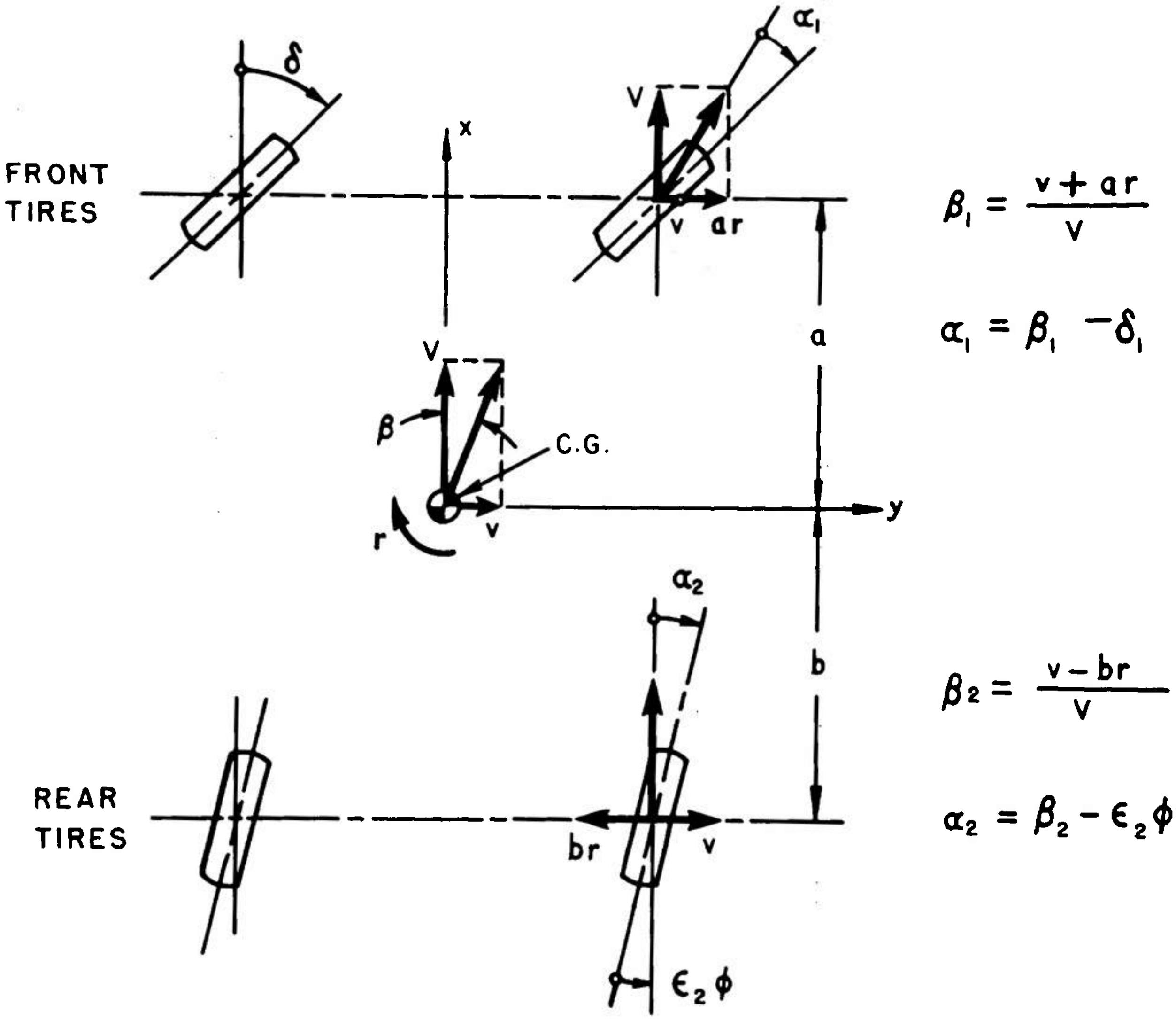


Figure 12-26. ***Slip Angle Relations in the Yawing Vehicle***

The slip angle of the vehicle center of gravity, assuming small angles of slip, is

$$\beta = \frac{v}{V} \qquad (12\text{-}29)$$

while for the front tires, assuming equal values of steering angle, the slip angle is

$$\alpha_1 = \beta_1 - \delta_1 = \beta + \frac{ar}{V} = \frac{v + ar}{V} \qquad (12\text{-}30)$$

For the rear tires, the slip angle is

$$\beta_2 = \frac{v - br}{V} \qquad (12\text{-}31)$$

If it is assumed that the rear roll steer effect, that is, the misalignment of the rear axle, is a linear function of the roll angle, the slip angle resulting from this effect is

$$\delta_2 = \varepsilon_2 \varphi \qquad (12\text{-}32)$$

Accordingly, under these conditions, the slip angle of the rear tires is

$$\alpha_2 = \frac{v - br}{V} - \varepsilon_2 \varphi \qquad (12\text{-}33)$$

12-17.3 AERODYNAMIC FORCES

The aerodynamic effects of the lateral motion of the vehicle arise from the side forces and the yawing moments introduced by the relative wind.

The side force acting on the vehicle is equal to

$$F_s = qAC_S \qquad (12\text{-}34)$$

where C_S is the side thrust coefficient which depends on the vehicle aerodynamic coefficients and

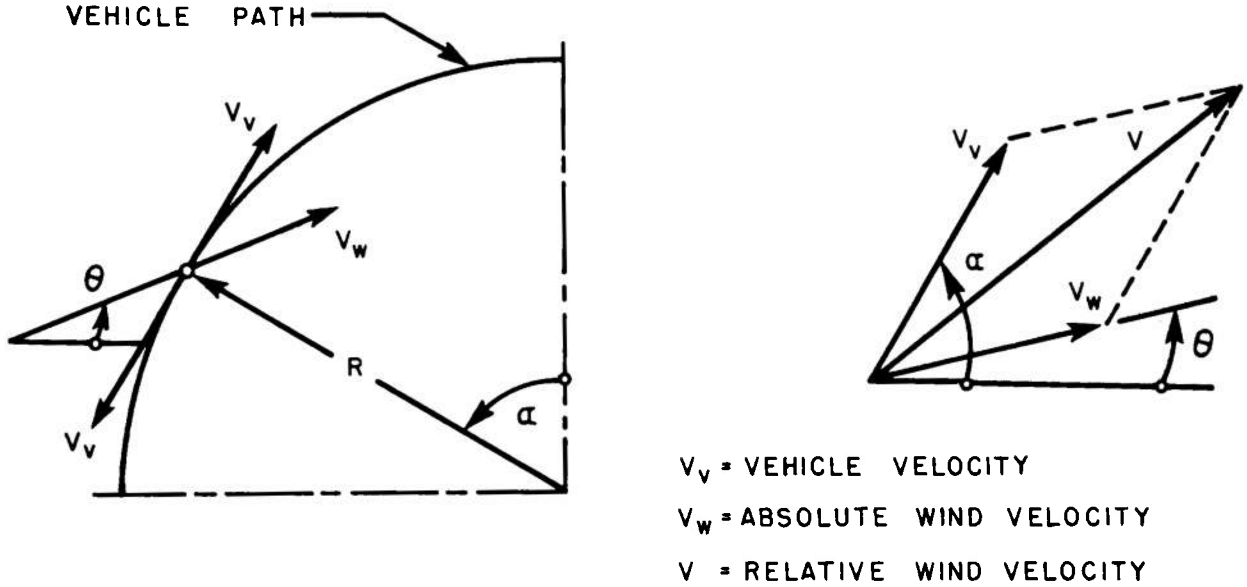


Figure 12-27. Wind Forces Acting on a Vehicle During a Steady-State Turning Motion

angle of attack, A is the projected frontal area of the car, ft, and q is the dynamic pressure acting, lb/ft[2].

The dynamic pressure is given by

$$q = 1/2\rho V^2 \tag{12-35}$$

where V is the relative velocity of the wind, and ρ is the mass density of the air, $\frac{\text{lb sec}}{4}$. The relative velocity of the wind is obtained from Fig. 12-27.

$$V^2 = V_v^2 + V_w^2 + 2V_v V_w \cos(\alpha - \Theta) \tag{12-36}$$

where V_v is the vehicle velocity, ft/sec, V_w, the absolute wind velocity, ft/sec, and $\alpha - \Theta$, the angle between the direction of the wind and vehicle.

The dynamic pressure is therefore

$$q = \tfrac{1}{2}\rho[V_v^2 + V_w^2 + 2V_v V_w \cos(\alpha - \Theta)] \tag{12-37}$$

and the side force acting

$$F_s = \tfrac{1}{2}\rho[V_v^2 + V_w^2 + 2V_v V_w \cos(\alpha - \Theta)]AC_S \tag{12-38}$$

where C_S is a function of the angle of attack.

The yaw moment acting is given by

$$M_y = F_s l \tag{12-39}$$

where l is the distance between the center of pressure and the neutral steer line, ft.

12.17.4 SHOCK ABSORBERS

The vehicle shock absorbers produce forces which resist vehicle roll moments. These forces are functions of vehicle roll velocity, hence, the shock absorber roll resisting force may be written as

$$F_{SA} = f_{SA}(\dot{\phi}) = C_{1A}\dot{\phi} + C_{2A}\dot{\phi}^2 + \ldots \tag{12-40}$$

where the constants C_{iA} are determined experimentally.

12-17.5 SUSPENSION SYSTEMS

In the section dealing with vehicle roll angles it was shown that the dynamic wheel loads introduced by the vehicle system are proportional to the roll angle. In addition dynamic loads are affected by centrifugal force effects introduced by both the sprung and unsprung masses.

The change in tire loads is therefore proportional to both roll angle and centrifugal force and may be written as

$$\Delta L = C_1\phi + C_2\mu W \tag{12-41}$$

or

$$\Delta L = C_1\phi + C_2\frac{mV^2}{R} = C_1\phi + C_2 mVr \tag{12-42}$$

where m is the vehicle mass, lb/ft/sec[2], V is the forward velocity, ft/sec, R is the radius of curvature, and r its yaw angular velocity, rad/sec.

12-17.6 TIRE ROLLING RESISTANCE

Tire rolling resistance is a function of both load and slip angle. Tire rolling resistance is given by:

$$F_T = f_T(W) + F_i \sin\beta_i \tag{12-43}$$

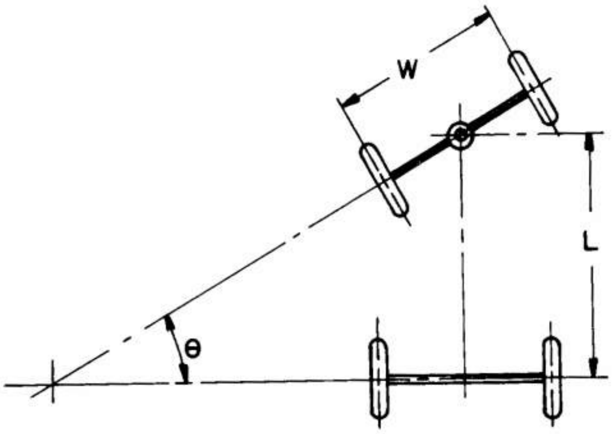


Figure 12-28. *Fifth-Wheel Steering Relations*

where β is the slip angle of the tire, degrees, F_i, its cornering force, lb, and f_T represents the functional relation. For small values of slip angle this equation may be written as

$$F_T = C_{1T}W + C_{2T}W^2 + \ldots F_i\beta_i \qquad (12\text{-}44)$$

where constants C_{iT} are obtained from tire data.

Because of the increased tire resistance resulting as load transfer occurs, a torque is applied about the vehicle yaw axis. The torque is given by

$$(F_{T2}-F_{T1})\frac{w}{2}=\left[C_{1T}(W_2-W_1) + C_{2T}(W_2^2-W_1^2)+\ldots\right] \qquad (12\text{-}45)$$

where F_{T2} is the rolling resistance of the more heavily loaded outer tire, F_{T1} that of the inner tire, w, the track length, ft, and $W_1 - W_2$, the load transfer effect. Load transfer is made up of both the suspension system effects and shock absorber forces.

12-18 FIFTH-WHEEL STEERING

Fifth-wheel steering is accomplished by pivoting an entire axle about a central pivot (Fig. 12-28). The wheels maintain their initial positions with respect to each other and the interconnecting axle during a turn. The theoretical center of rotation for a fifth-wheel steering system on a four-wheeled vehicle is found by extending the center lines of the axles until they intersect a common vertical axis. (The actual center of rotation will differ from the theoretical center because of the slip angles of the wheels.)

The theoretical radius of curvature, R, for the outermost wheel is given by

$$R = \frac{l}{\sin\Theta} + \frac{w}{2} \qquad (12\text{-}46)$$

where

- l is the wheelbase, ft
- Θ is the angle through which front axle is moved, degrees
- w is the track of front wheels, ft

Although fifth-wheel steering is most commonly limited to towed vehicles, such as four-wheeled trailers, this system has been utilized on the new high-mobility truck (Goer). Since a relatively large and heavy mass must be rotated during the steering of the Goer vehicle, a power assisted steering system is used. Normally, greater underbody clearance is required for the rotating elements of a fifth-wheel steering system than for a comparable Ackermann system.

Camber steering is based on the principle that a cambered wheel will tend to travel in a curved path whose theoretical center of rotation is at the point of intersection of the axial line and the ground. The actual center of rotation will deviate from the theoretical center owing to the deflection characteristics of the tire. By adjusting the camber of individual wheels on a common axle, a common center of rotation for these wheels can be achieved. However, each axle will have its own center of rotation, and some slippage (other than that arising during the generation of cornering power) must take place during the turning process.

12-19 THE STEERING OF MULTIWHEELED VEHICLES (Ref. 14)

12-19.1 SINGLE FRONT AXLE

In the present context, multiwheeled vehicles are defined as vehicles having more than two axles. Currently, the single front axle, six-wheeled vehicle is the most common multiwheeled military vehicle.

As stated previously, in order for a wheeled vehicle to turn a corner without excessive skidding (lateral slippage of the tires), the projection of all axles must intersect at a common point (the axis of rotation). For a six-wheeled vehicle, Fig. 12-29, it is obvious that the above condition cannot be met unless the rear axles are skewed with respect to each other. If it is assumed that the rear axles remain parallel during a turn, the exact location of the center of rotation for the rear wheels is indeterminate from geometric considerations, since its position will vary with tire loadings, tire pressures, tire tread design, and wheel camber. Be-

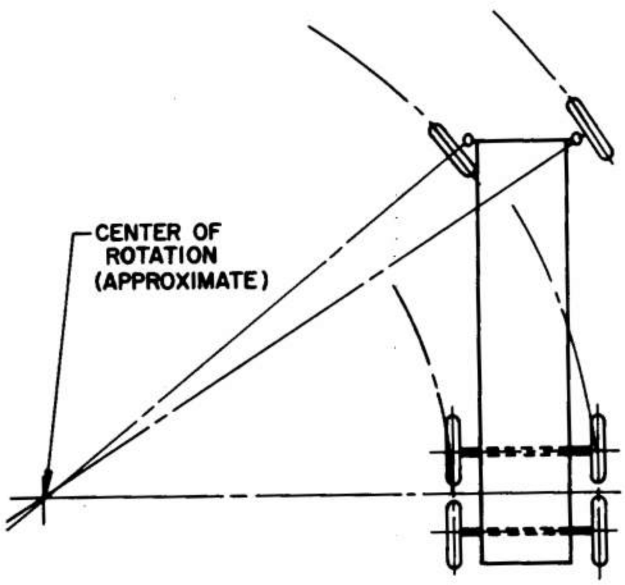


Figure 12-29. ***Turning-Radius Diagram of a Multiwheeled Vehicle***

cause it is probable that the axis of rotation of the rear wheels lies approximately midway between the axles, steering linkages are designed on this assumption.

Since the torque required to produce lateral skidding of the multiple axle rear wheels during a turn must be developed by means of the front wheels, the length of the wheelbase and the location of the center of gravity (percentage of weight on the front wheels) influence steering. Very short-wheelbased vehicles, or vehicles with a high percentage of the total weight on the rear wheels, may not steer properly because of the inability of the front wheels to generate the required torque. Dual rear tires also introduce slippage during turning since the two dual tires on a given side of the vehicle turn at the same angular speed while traveling along different radii of curvature. This slippage may be greater than that introduced by a pair of dual axles if the dual wheels are large.

With reference to the steering behavior, it is desirable to minimize the distance between the axles in multiwheeled vehicles. However, this distance is governed by tire size and the necessary clearance between tires. Some dual axle suspension systems are designed so that the roll induced load transfer during the turn causes a favorable realignment of the rear axles and consequently reduces the amount of wheel skidding which occurs. This action is essentially roll induced steering of the rear wheels.

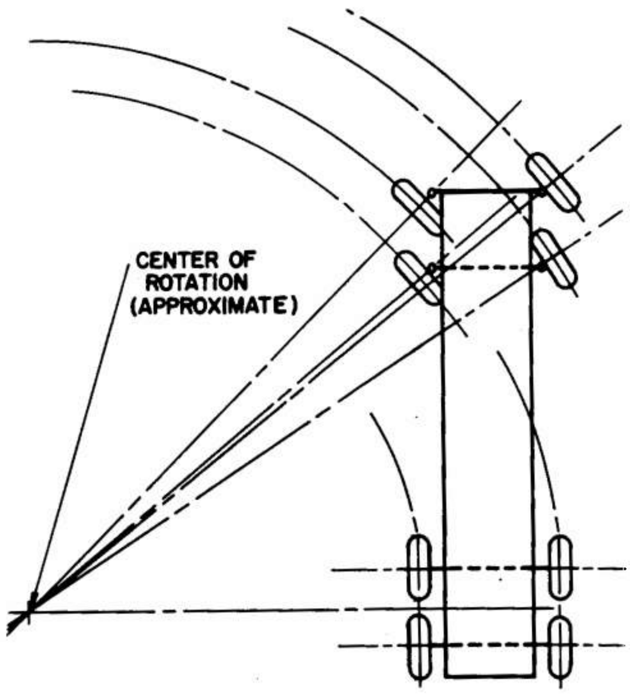


Figure 12-30. ***Turning-Radius Diagram of an Eight-Wheeled Vehicle***

12-19.2 DUAL FRONT AXLES

Eight-wheeled vehicles having dual front axles present increased steering problems when compared to vehicles having a single front axle because both pairs of front wheels must be turned. A number of eight-wheel arrangements have been constructed. In one arrangement the axles are consolidated in pairs by means of a bogie arrangement. The rear bogie is mounted in the normal manner while the front bogie constitutes a fifth-wheel arrangement. Steering is accomplished by turning the front bogie relative to the frame. The eight-wheeled trucks, T20 and T26, built at the close of WW II utilized this arrangement.

This method of steering introduces skidding of both front and rear wheels since projected centerlines of the individual axles do not pass through a common point or axis of rotation. However, an approximate center of rotation can be established, and vehicle turning behavior predicted with a high degree of accuracy.

A second method of steering an eight-wheeled vehicle utilizes a pair of conventional knuckle type (Ackermann) front axles, each with steering elements designed to produce true tracking as shown in Fig. 12-30. The steering linkage system is complicated by the requirement that all of the front

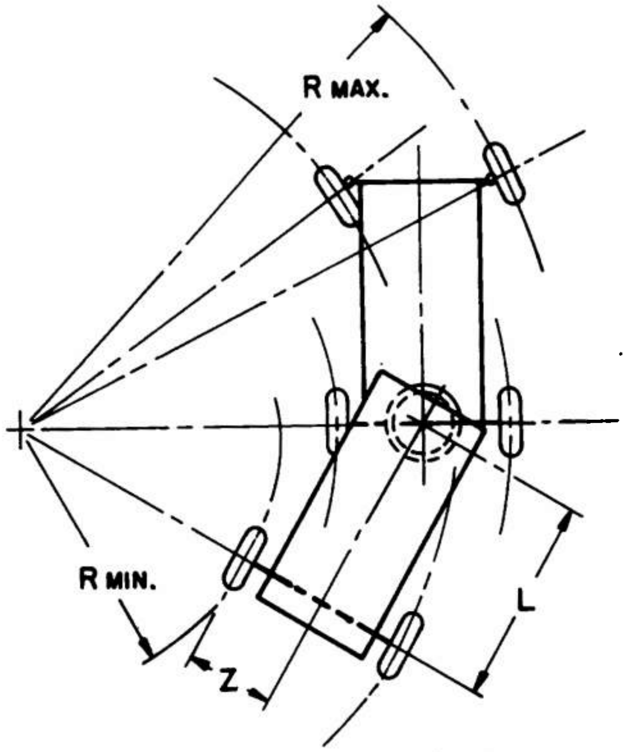


Figure 12-31. *Turning-Radius Diagram for Tractor and Trailer Steering*

wheels must turn through different angles as the vehicle negotiates a given curve.

A third method of steering eight-wheeled vehicles has been applied to several experimental vehicles. Here the relative speeds of the wheels on either side of the vehicle are varied. Since none of the axles is traveling about a vertical center of rotation, lateral sliding of all of the wheels occurs during turning. This method has the further disadvantage of requiring either a complex transmission or multiple power plants to permit variation in relative wheel speeds. A definite advantage of relative speed steering is that no provision must be made for steering motions of the plane of the wheel. Thus, it is not necessary to reduce the width of the hull in the vicinity of the wheels.

12-20 TRACTOR AND TRAILER STEERING

The tractor of a tractor-trailer combination will behave as a conventional vehicle during the turning process. Maximum turning radius is described by the outside front wheel; the rear inner wheel of the fifth-wheel mounted trailer will describe the minimum turning radius. The difference between maximum and minimum radii determines whether a given tractor-trailer combination can negotiate a turn on intersecting roads of given widths. The theoretical turning radii (Fig. 12-31) for a tractor-trailer vehicle can be determined by the following equations, (see Fig. 12-2). For the outer front wheel

$$R_{max} = \frac{l}{\sin \delta} \approx \frac{l}{\delta}$$

where

- R_{max} is the radius of curvature followed by the outer front wheel, ft
- l is the wheelbase, ft
- δ is the steering angle (average of the steering angles of the inner and outer wheels), deg

For the inner trailer wheel

$$R_{min} = \left(\left(\frac{l}{\sin \delta} \right)^2 - l^2 - L^2 \right)^{\frac{1}{2}} - z \qquad (12\text{-}47)$$

where

- L is the wheelbase of the trailer, ft
- z is the distance from the trailer centerline to the center of the inner wheel (one-half the trailer track distanee), ft

12-21 POWER STEERING AND POWER STEERING SYSTEMS (Refs. 12, 14)

As the tire size and loading increase, wheeled vehicles develop turning resistances of such magnitude that conventional manual steering systems become impracticable. The high reduction ratios necessary to keep manual effort within reasonable limits require the operator to turn the steering through large arcs when turning the wheels of the vehicle. Some form of power steering to aid the driver in steering heavier vehicles is therefore desirable. Cross country operations also accentuate the need for power steered vehicles. Power steering systems currently in use are booster-type installations, i.e., they act on some element of a mechanical steering system in such a manner that the manual force required to turn the wheels is decreased.

A power steering unit consists of two essential systems; one for generating and transmitting power, the other for controlling the power. The control unit consists of a servomechanism actuated by the application of a force on the steering wheel. When no force is applied to the steering wheel, which may be in any possible position, power assist must cease. It is desirable that the power steering system function so that a power failure will not prevent manual steering in the usual manner. Power steering systems that fail instantly when

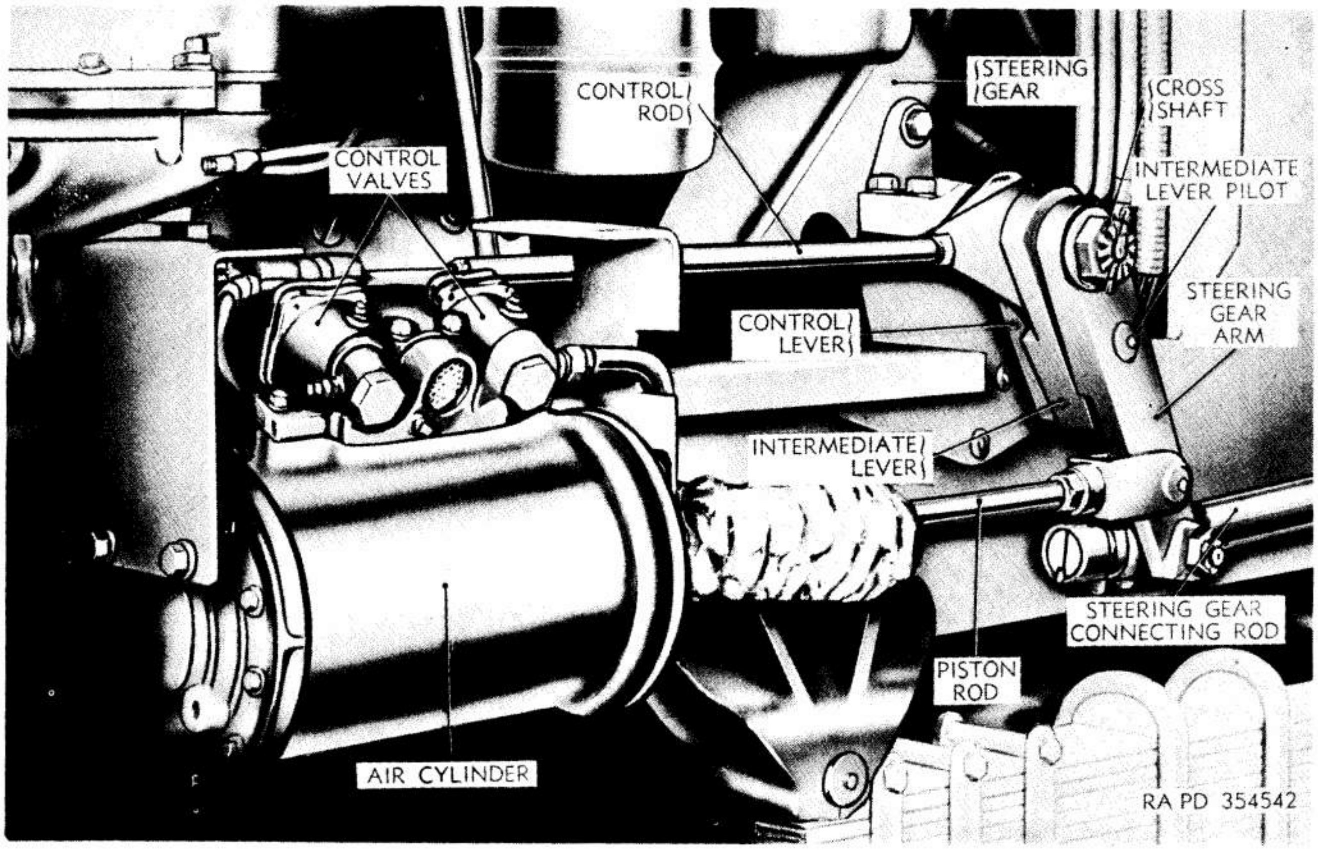


Figure 12-32. ***Typical Pneumatic Steering System***

the engine stalls are to be avoided, since the operator normally has only a light hold on the steering wheel and is unprepared to exert instantly the very much greater steering effort required.

Another problem related to power steering systems concerns wandering of the vehicle during straight-ahead motion. The steering wheel, when positioned for straight-ahead motion, has a small dead-band in which control is not exercised. The task of maintaining the vehicle on a steady course is performed by the power assist unit as described below.

12-21.1 PNEUMATIC STEERING

Power steering units operated by compressed air are available. Such units are well suited to vehicles having air brakes since a source of compressed air is already available. If the vehicle does not have an air braking system, an air compressor and a reservoir must be provided.

A compressed air system consists of three primary elements (other than the compressor and reservoir); a combination of control linkages mounted on the steering gear arm, two control valves, and an air cylinder containing a double-acting piston, Fig. 12-32. The control valves, mounted directly on the air cylinder, each admit air to one side of the cylinder.

The pressure delivered to either side of the cylinder piston is proportional to the force applied to the proper valve through the control rod. As long as a force, in a given direction, is applied to the steering wheel, the corresponding valve stays open. When the force on the steering wheel is removed and the steering wheel position remains constant, the control linkage shifts and shuts off the air supply. If the steering wheel is turned in the reverse direction, the control valves permit one side of the cylinder to exhaust while pressurizing the opposite side.

12-21.2 HYDRAULIC STEERING

Several hydraulic steering systems have been developed. Basically, these units consist of a double-acting hydraulic cylinder, a control valve, a hydraulic pump and control linkages. The double-acting cylinder and piston are incorporated in the steering mechanism in a manner similar to the pneumatic system. A hydraulic steering unit, there-

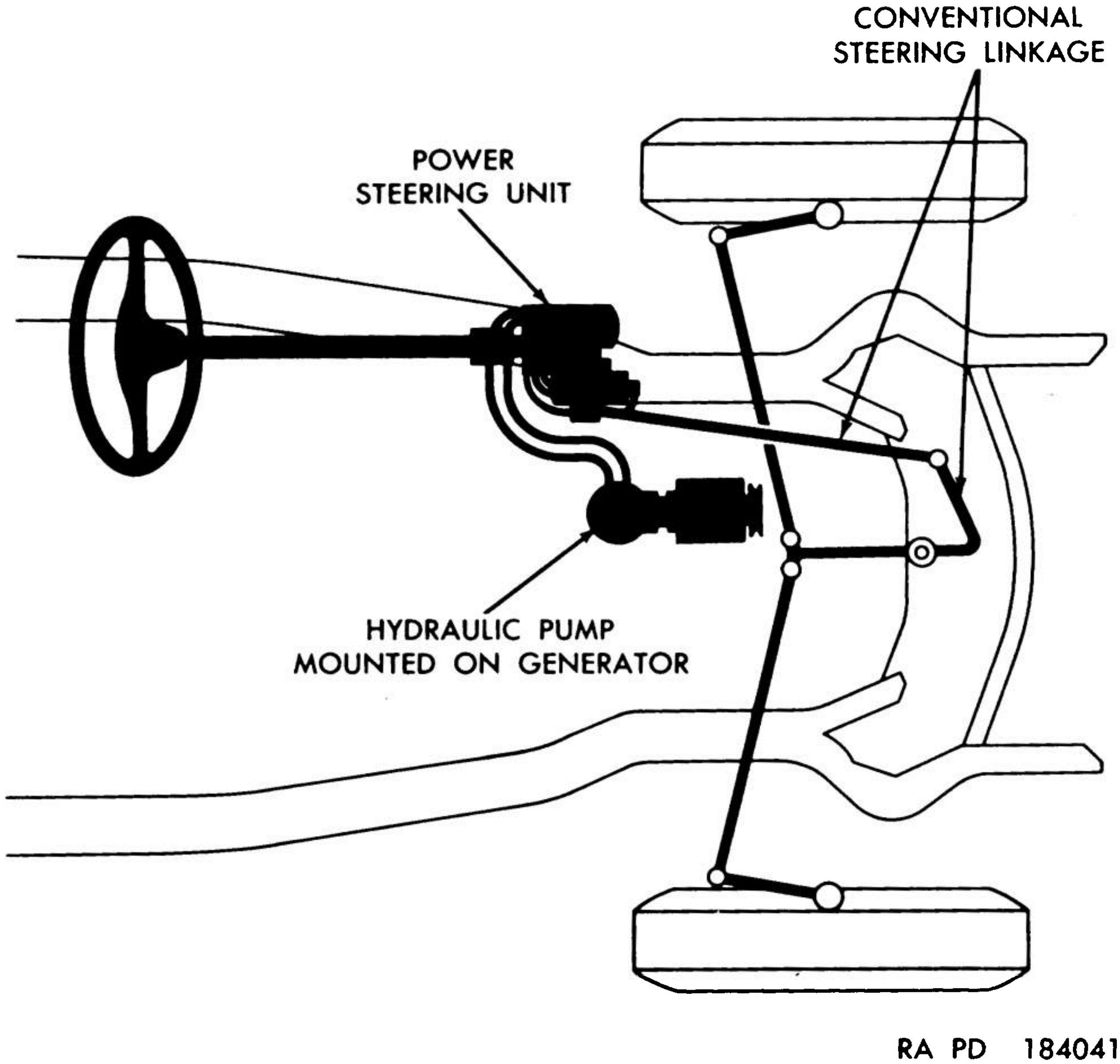


Figure 12-33. ***Location of a Hydraulic Power-Steering Unit***

fore, acts as a booster to reduce the manual force required at the steering wheel.

The control valve is actuated by motion of the steering wheel: as long as a force is applied to the steering wheel, a differential pressure exists in the double-acting cylinder. When the force on the wheel ceases, the pressure in the opposite ends of the cylinder becomes equal. A detailed explanation of the most commonly installed hydraulic steering system is found in Refs. 12 and 14. Figures 12-33 and 12-34 show a typical installation.

Hydraulic steering systems are normally designed so that a failure in the hydraulic system will not prevent manual steering of the vehicles. The inherent characteristics of the hydraulic power steering system are such that road shocks and extraneous lateral forces are resisted by the unit, thus, kick-back at the steering wheel is minimized. The major disadvantages of hydraulic power steering are, an increased cost over manual steering, a loss of "road-feel" greater power consumption (usually supplied by the main power plant), an increased maintenance. The control valve leakage must be avoided in a system operating at 800 to 1,000 psi.

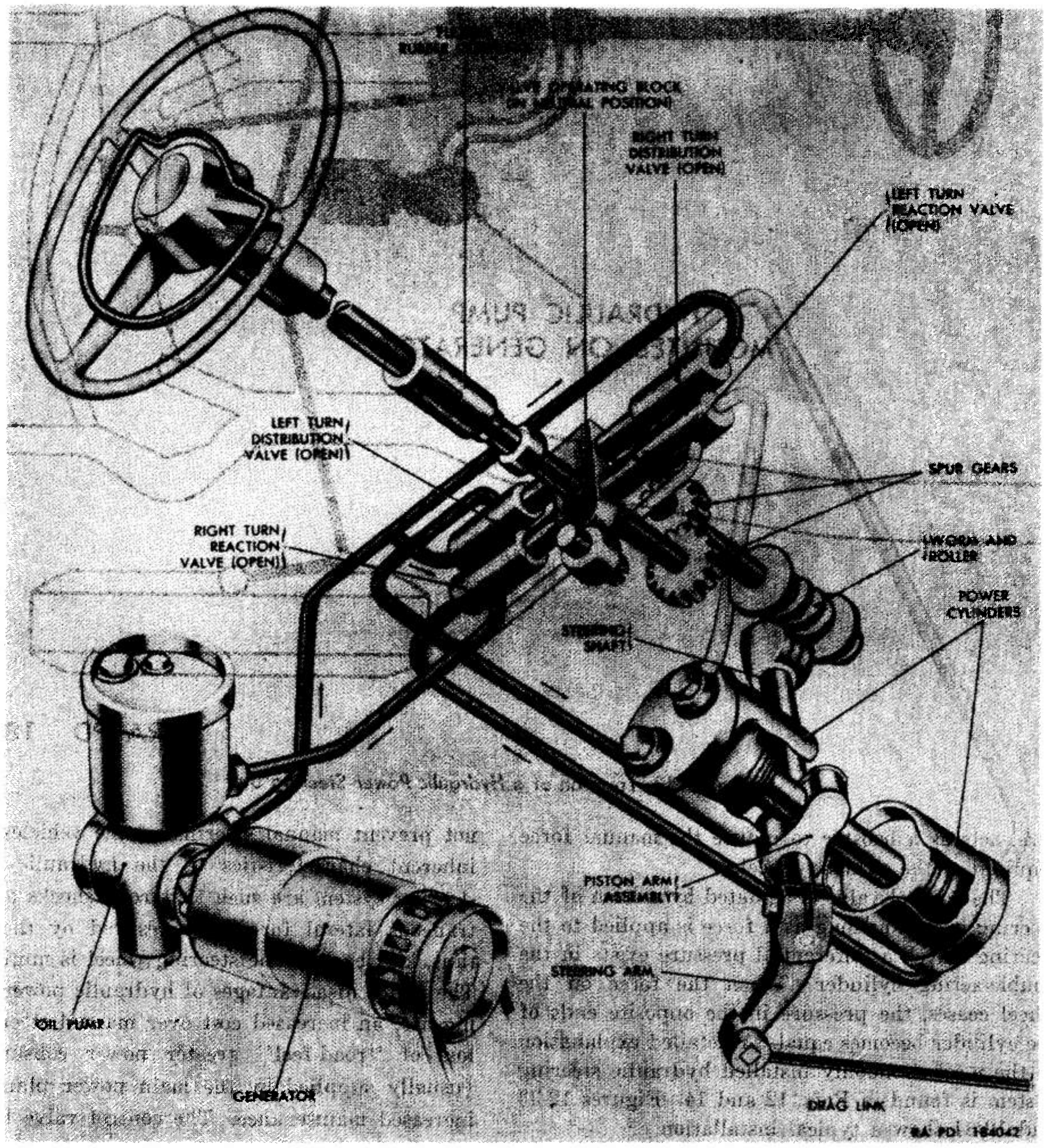


Figure 12-34. *Schematic View of a Hydraulic Power-Steering System*

SECTION III STEERING OF TRACKED VEHICLES (Refs. 17, 18, 19, 20)

From the point of view of steering, tracked vehicles can be divided into two groups: (1) those employing laterally flexible tracks, and (2) those employing laterally rigid tracks.

A laterally flexible track has two degrees of freedom and, as such, can be laid down on level ground in a curved path. A laterally rigid track, however, because it has only a single degree of freedom, must be laid down in a straight path.

The steering of a vehicle with laterally flexible tracks has been effected in one or two ways. In one method, a pivoted leading idler is used to lay each link on the ground at an angle in such a manner that its longitudinal centerline is directed at an angle, Θ, to the longitudinal centerline of the preceding link. In this manner, the track is laid on the ground in a circular path. If P is the pitch of the track links, the radius of curvature of the circular path, R, is

$$R = \frac{P}{2 \tan \frac{\Theta}{2}} \qquad (12\text{-}48)$$

Since the track on either side of the vehicle traverses a different radius of curvature during a turn, the inner steering idler is turned through a greater angle, with respect to the longitudinal axis of the vehicle, than is the outer idler. The amount of lateral flexibility required of the track is relatively small; for example, assuming a track pitch of 6 in. and a radius of curvature of 10 ft, the required angle, Θ, is 2° 52′.

With the pivoted leading idler system of flexible track steering, the road wheels must be able to conform to the track curvature; this introduces additional complications into the design of the overall system. Furthermore, a pivoted leading idler will not provide for steering in reverse; accordingly, some additional means is required to accomplish this maneuver.

Laterally flexible tracks are expensive to produce, wear more quickly than conventional tracks, and are more prone to come off the sprockets.

In a second method of flexible track steering, the idler, road wheels and sprockets are all turned through small angles and the track forced into a curved path.

Either system requires a differential mechanism to compensate for the difference in inner and outer track speeds which exists during the turn.

The main advantage of flexible track steering is that much less power is required during a turn than for comparable rigid track steering. Less damage to roads and lower stresses in the system should result. Flexible track steering appears best suited to light, high-speed vehicles (Ref. 17).

All current standard military tracked vehicles use laterally rigid tracks. The remaining discussion in the present chapter applies to this type of track.

The steering of a tracked vehicle having laterally rigid tracks is accomplished by controlling the relative speed of the tracks while applying sufficient power to the tracks to overcome steering induced losses. Since the tracks are relatively rigid in the lateral direction, turning is accomplished by sliding all parts of the track, except its lateral midline, transversely across the ground. When this sliding occurs, either or both tracks also slide longitudinally as power is applied to them. This transverse and longitudinal sliding represents a power loss and during low speeds may be the dominant factor in determining the power required to turn the tank. Some of the factors that effect the steering of track-laying vehicles are (Ref. 21):

a. The total weight and weight distribution of the vehicle.

b. The length-tread (L/T) ratio.

c. The longitudinal coefficient of traction.

d. The transverse coefficient of traction.

e. The relative loads carried by the two tracks during the turning process.

f. The unbalanced transverse frictional forces on the tracks caused by centrifugal force or by a side slope.

g. The resisting forces acting on the vehicle, which may be caused by a towed load, ground resistance, inertial forces, or grades.

h. The transverse flexibility of the tracks.

i. The radius of the desired turn.

j. The speed of the vehicle.

k. The design and performance characteristics of the steering system.

Some of these parameters are discussed briefly in the following paragraphs.

12-22 STEERING MECHANISMS FOR TRACKED VEHICLES (Ref. 20)

12-22.1 CLASSIFICATION OF STEERING SYSTEMS

Numerous mechanisms utilizing gear trains and clutches and/or brakes to control the various elements of the system have been devised for the purpose of steering track-laying vehicles. The classical systems are described in par. 12-23. Although there is considerable variation in the type and arrangement of their elements, their performance characteristics permit a classification into three basic types, A, B, and C, and a miscellaneous category. The basis for classification is the relation between power wasted in slipping the steering clutches and drive sprocket torques when the vehicle is negotiating a turn of larger radius than that for which it is geared. These relations are shown in Eqs. 12-50, 12-51 and 12-52. In a Type A system, waste power is proportional to the difference in outer and inner drive sprocket torques; in a Type B system, waste power is proportional to inner drive sprocket torque; in a Type C system, waste power is proportional to outer drive sprocket torque. Miscellaneous types are those which do not fall into these categories. The classification of commonly applied steering systems into these three basic types is shown at the end of par. 12-22.1.

If all other factors are equal, the amount of power wasted as a vehicle negotiates a turn of given radius, will be the same for all steering systems of the same type. The amount of power lost, W_i, will, under most operating conditions, be least for a Type B system and greatest for a Type C system or

$$W_B < W_A < W_C \qquad (12\text{-}49)$$

In addition to differing relations between power wasted and sprocket torques, the three systems exhibit differences in the manner in which the ratio of the transmission input speed, N_e, to mean velocity of the tracks, V, is affected during a turning maneuver. The variation of this ratio is shown in Table 12-3. Also shown in this table are ratios among the elements of the several systems which remain constant.

TABLE 12-3 BASIC CLASSES OF STEERING MECHANISMS FOR TRACKED VEHICLES (Ref. 20)

Type	N_e/V Ratio	Constant Ratios
A	Constant	N_e/N_s
B	Increases	N_e/N_o
C	Decreases	N_e/N_i

KEY:

N_e is the speed of transmission input shaft
V is the mean linear velocity of the tracks
N_o is the speed of the outer sprocket
N_i is the speed of the inner sprocket
N_s is the mean speed of the sprockets $\frac{N_o+N_i}{2}$

The most common steering systems for tracked vehicles are described in par. 12-23. These systems may be classified into the three basic types as follows:

Type A

Controlled differential
Merritt's geared differential
Back-geared differential
Braked differential
Double differential
Merritt's double differential
Triple differential (split torque propulsion)
Triple differential (regenerative torque propulsion)
Double differential (infinitely variable hydrostatic drive)

Type B

Geared
Merritt's geared
Clutch-brake
Gates
Multiple-ratio geared

Type C

Geared (inverse operation)

Miscellaneous Types

Independent propulsion (e.g., hydrostatic drive)

12-22.2 WASTED HORSEPOWER

The power losses for steering systems in each of three classes can be expressed by the following equations (Ref. 20):

$$\text{Type A: } Q_A = K_A \frac{N_s\,(T_o - T_i)}{5252} \qquad (12\text{-}50)$$

$$\text{Type B: } Q_B = K_B \frac{N_s\,T_i}{5252} \qquad (12\text{-}51)$$

$$\text{Type C: } Q_C = K_C \frac{N_s\,T_o}{5252} \qquad (12\text{-}52)$$

where

Q_A, Q_B, Q_C is the horsepower wasted at the slipping steering clutch

K_A energy ratio factor: $\dfrac{R-1}{R+1} - \dfrac{r+1}{r-1}$

K_B energy ratio factor: $\dfrac{R-1}{R} - \dfrac{(R)}{(R+1)}\dfrac{(r+1)}{(r-1)}$

K_C energy ratio factor: $(R-1) - (R+1)\dfrac{(r-1)}{(r+1)}$

N_s is the mean speed of sprockets, rpm

T_o is the torque of outer sprocket, lb-ft

T_i is the torque of inner sprocket, lb-ft

R is the maximum ratio of steer system

r is the steer ratio of the system with slipping clutch

12-22.3 REGENERATIVE AND NONREGENERATIVE STEERING SYSTEMS

A regenerative steering system for track-laying vehicles is a system in which the power developed by the inner track is transferred to the outer track. It is assumed the inner track is driving the sprocket at the beginning of the turn.

Whenever a steering clutch (or brake) is slipped to obtain a partial steer, energy is wasted in the clutch. Under some conditions, the amount of energy wasted exceeds the amount of energy fed back (regenerated) into the system.

Systems of Types A, B, or C, when operating with slipping steering clutches, may be regenerative under some conditions and nonregenerative under other conditions.

If the sprocket torques to the inner and outer sprockets are assumed to be equal and opposite during a partial steer, the energy relations are as described below. With Type B systems, the energy losses of the steering clutch may equal but will not exceed the inner track feedback energy. For Type A systems, the energy absorbed by the steering clutch can exceed the energy feedback of the inner track, but it can never be greater than twice the energy from the inner track. For Type C systems, the energy loss in the slipping clutch is greater than that of a Type A system. In addition, the steer ratio* for such a system cannot be extended to infinity as it can be in Types A and B steering systems.

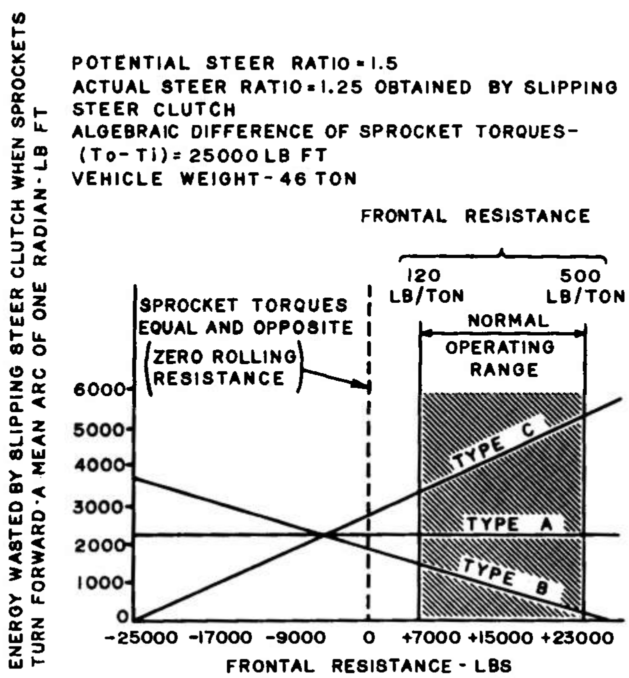


Figure 12-35. Energy Wasted of Types A, B, and C Steering Systems as Frontal Resistance Varies (Ref. 20)

* The steer ratio is the ratio of the velocities of the outer and inner tracks.

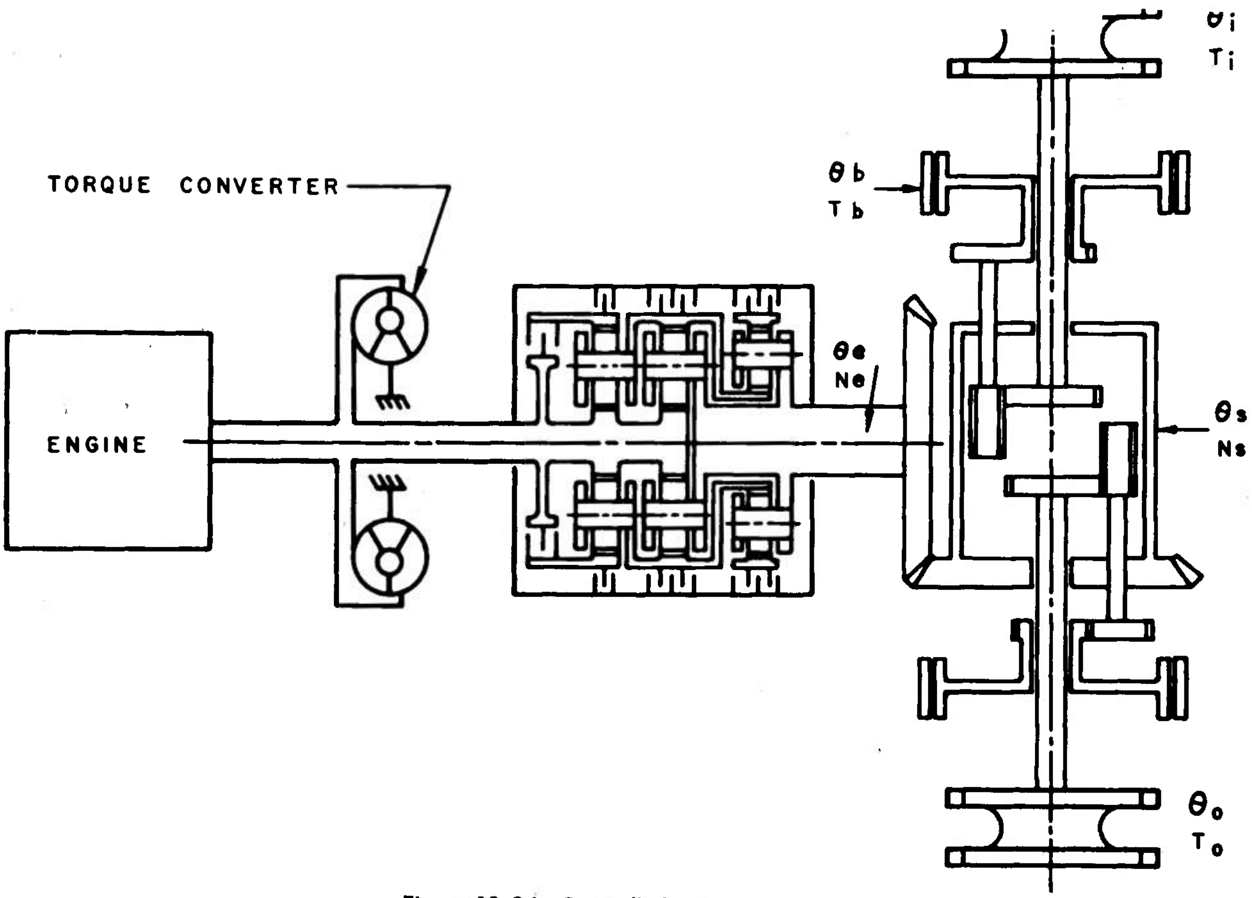


Figure 12-36. *Controlled Differential*

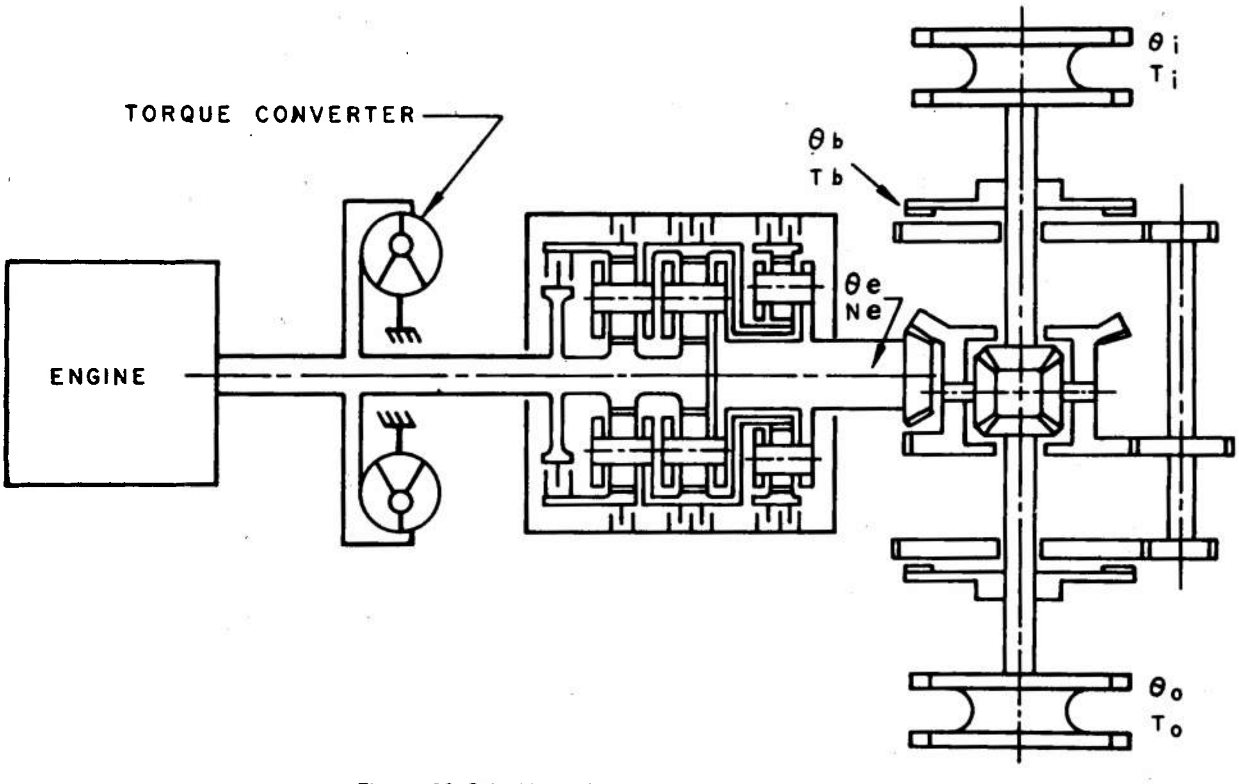


Figure 12-37. *Merritt's Geared Differential*

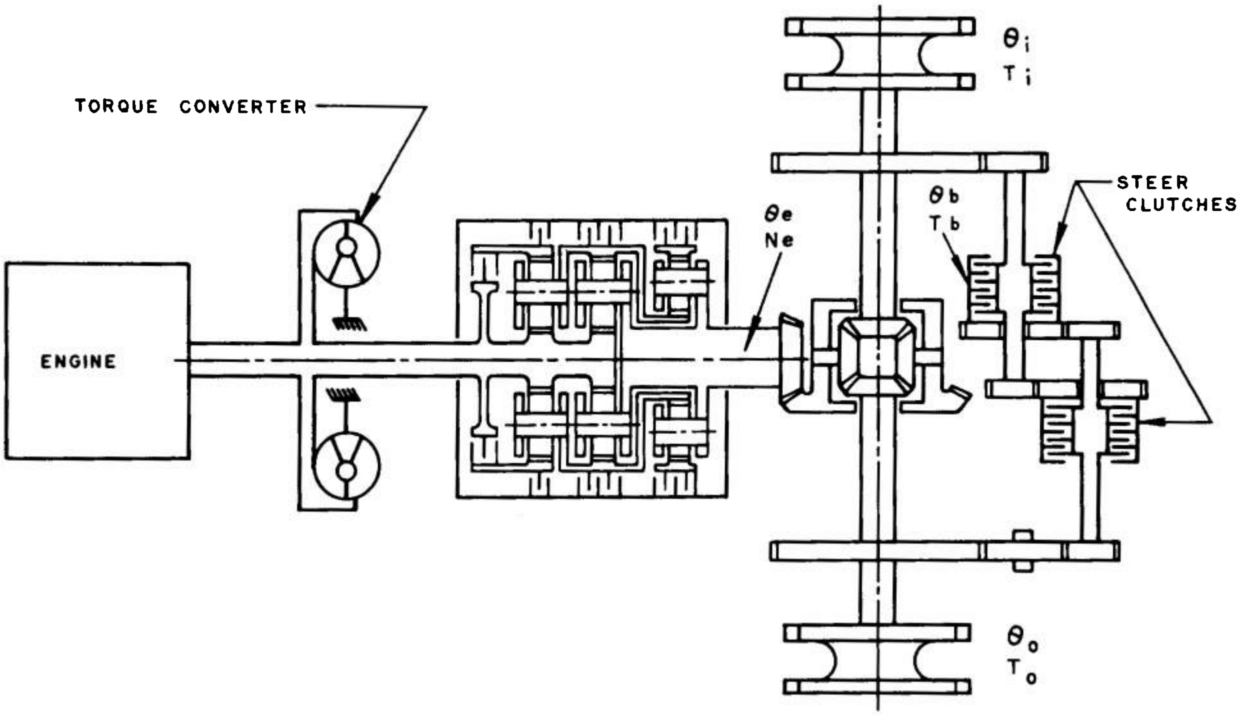


Figure 12-38. *Back-Geared Differential*

A practicable and efficient infinitely variable steering transmission would eliminate the energy or power loss under all turning conditions, and would be regenerative under all operating conditions.

The energy wasted by each of the basic types (A, B, and C), is shown in Fig. 12-35. The diagram applies to the case of a vehicle the size and weight of a T48 tank negotiating a radius of 60 ft by slipping the steer clutch of a steering system which is geared for a minimum radius turn of approximately 35 ft. Energy wasted (power lost) is a function of the frontal resistance (or motion resisting forces), i.e., rolling resistance, towing resistance, and grade resistance.

In general, when a turning track-laying vehicle encounters a resisting force opposing its forward motion, outer sprocket torque is greater than the inner sprocket torque. The resulting slewing couple on the vehicle is a function of the algebraic difference of the sprocket torques. From Eq. 12-50 it is seen that the horsepower wasted or clutch energy loss of the Type A group is proportional to the algebraic difference in inner and outer sprocket torques. Since the algebraic difference of the sprocket torques remains constant with a change in front resistance, the energy loss for a steering system of Type A is independent of front resistance.

For the Type B group, clutch energy loss, Eq. 12-51, is proportional to the inner sprocket torque. During a turn, the inner sprocket torque decreases as the frontal (longitudinal) resistance increases. As a result, clutch energy losses for a Type B system decrease with an increase in frontal resistance.

For steering systems in the Type C group, the clutch energy loss is proportional to the outer sprocket torque, Eq. 12-52 which increases as the frontal resistance increases, hence, the clutch energy loss increases with increased frontal resistance.

As shown in Fig. 12-35, the normal range of

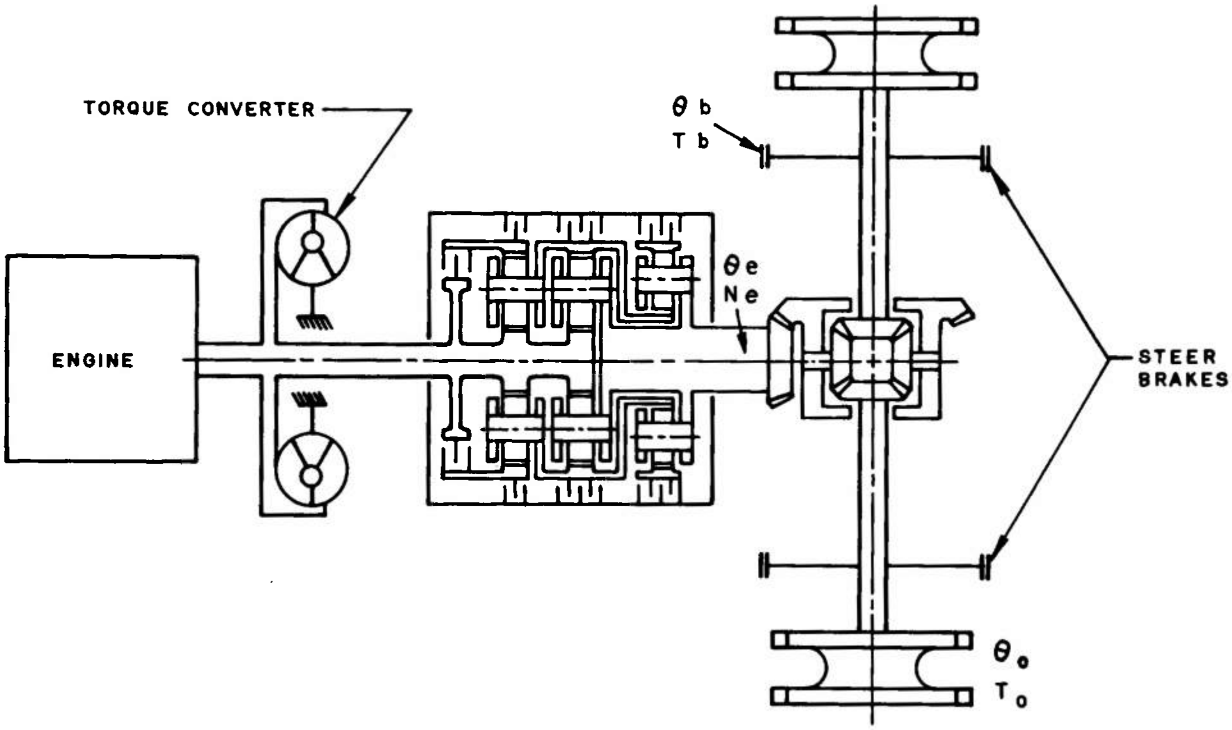


Figure 12-39. *Braked Differential*

operation for most operating tracked vehicles lies between 120 and 500 lb/ton of vehicle weight. In this range, Type B steering systems waste less energy than the other basic types.

12-23 DESCRIPTIONS OF STEERING SYSTEMS

12-23.1 TYPE A SYSTEMS

12-23.1.1 Controlled Differential

The controlled differential system, Fig. 12-36, is an epicyclic gear system in which a brake application results in a decrease in speed of one sprocket and an equivalent increase in that of the opposite sprocket. With the brake fully applied, the ratio of inner and outer sprocket speeds is fixed, regardless of vehicle speed. Controlled differential steering does not affect the mean propulsion ratio of the vehicle since the decrement in speed of one track is equaled by the increment in speed of the opposite track. In addition, steer ratio is not affected by gear ratio changes in the power train gearing. The system is regenerative.

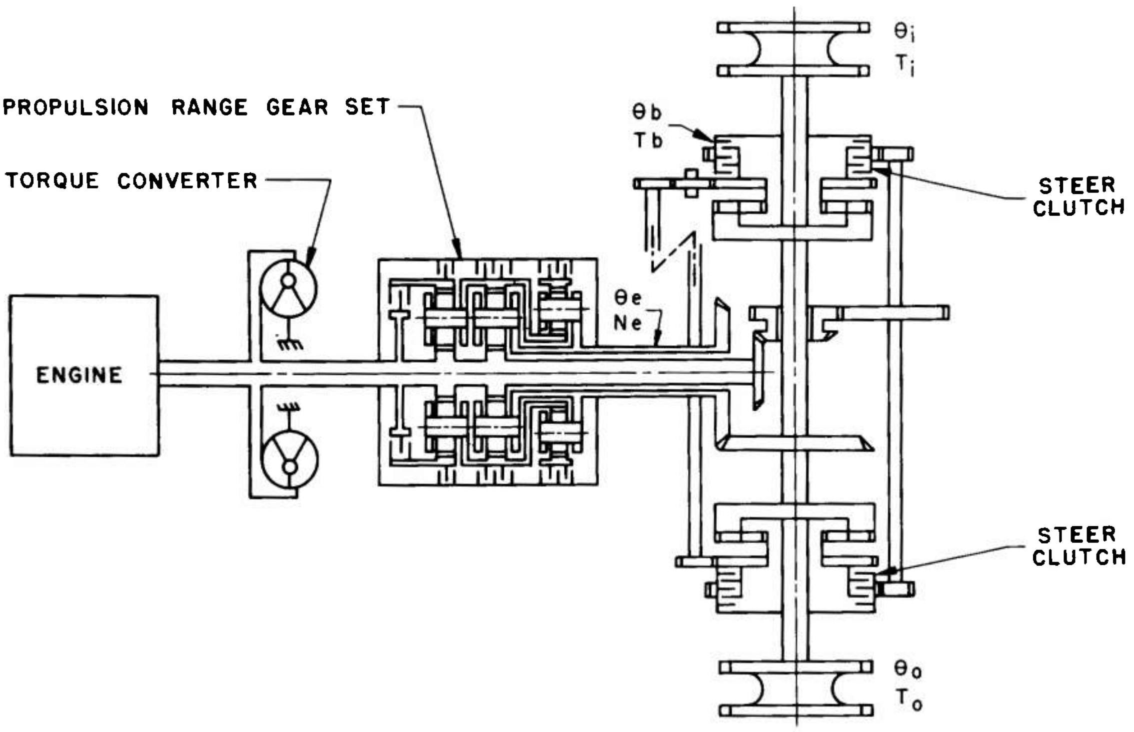
The controlled-differential steering system is dependent upon the stabilizing forces of ground contact to maintain straight-ahead steering stability. Considerable veering and a loss of steering control occurs during deceleration. No tendency towards true reverse steering occurs, however. A vehicle steered by a controlled differential system is not capable of making a pivot turn about its vertical centerline.

Two arrangements, which have characteristics identical to those of the controlled differential steer system are shown in Figs. 12-37 and 12-38. These are Merritt's geared differential and the back-geared differential steering systems, respectively.

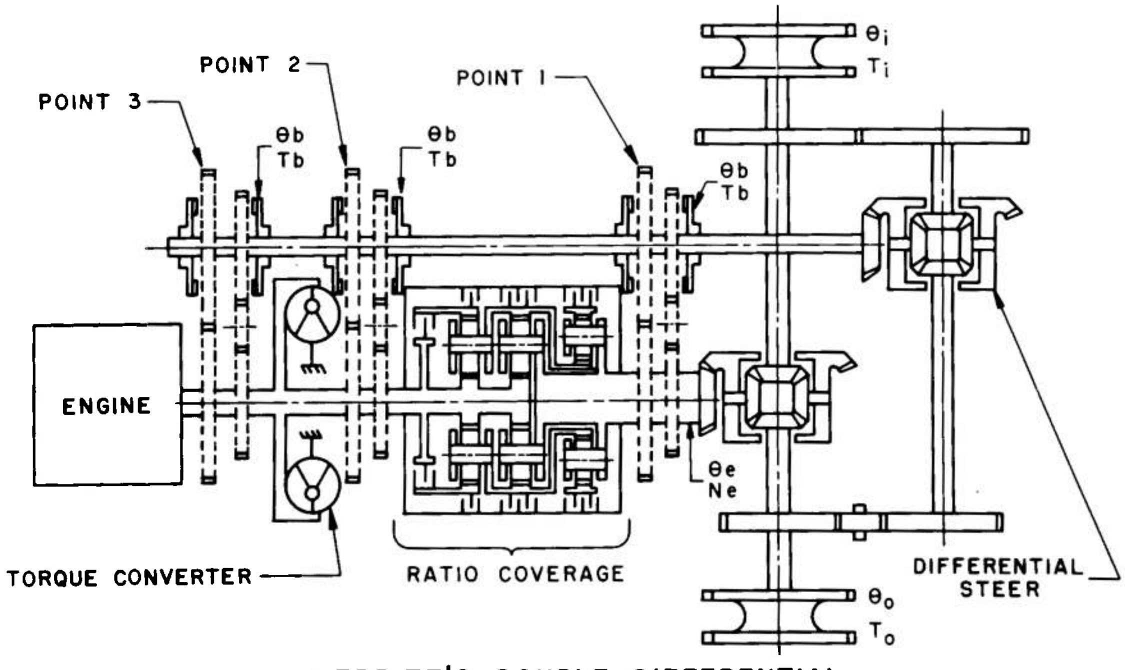
12-23.1.2 Braked Differential

The braked differential as shown in Fig. 12-39 consists of a simple differential-brake arrangement. The steer ratio or maximum velocity ratio between outer and inner tracks, V_o/V_I, can be made infinite; it is not affected as the power train gear ratio is changed. The braked differential obtains power for turning only from the outside track. It is not capable of producing a true pivot turn; instead, it causes the vehicle to turn about one track.

The braked differential is not considered satisfactory for military vehicles even though it has the advantages of simplicity, is continuous in operation, and produces a slewing moment proportional to the torque exerted by the steering brake. The disadvantage of this system results from the re-



PLAIN DOUBLE DIFFERENTIAL



MERRITT'S DOUBLE DIFFERENTIAL
WITH THREE LOCATIONS OF STEER DIFFERENTIAL DRIVE

Figure 12-40. *Double Differential Systems*

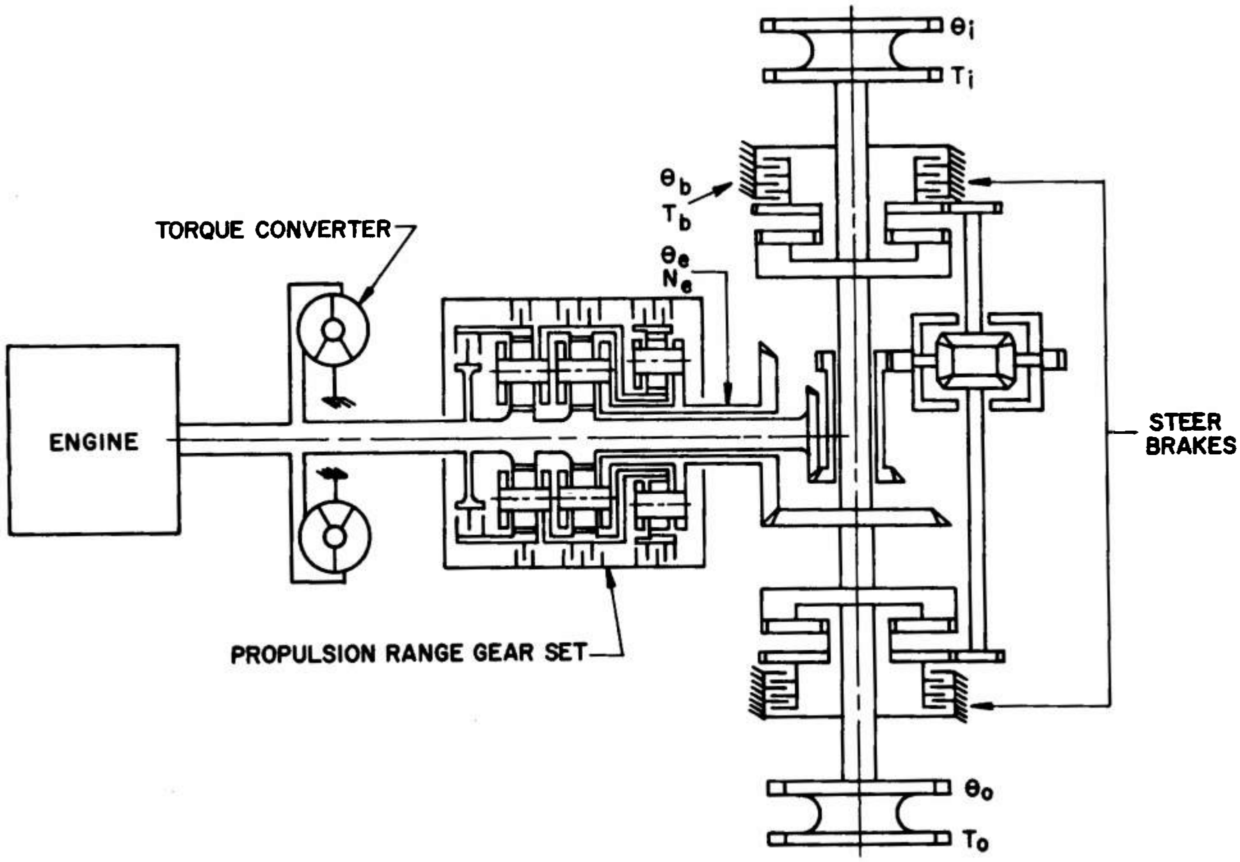


Figure 12-41. Triple Differential—Split Torque Propulsion

quirement that during a turn, the brake must absorb one-half the power of the engine plus that developed by the inner track. At higher speeds longitudinal skidding tends to make the vehicle uncontrollable. The desirable features of this transmission can be obtained by other mechanisms which are free of its undesirable characteristics.

12-23.1.3 Double Differential

The double differential system, Fig. 12-40, enforces a differential speed between the tracks by increasing the speed of one track and decreasing the speed of the opposite track a like amount. This system is regenerative and has no tendency to reverse steer. The mean propulsion ratio is not affected by this system during straight-ahead driving or turning. With the power train gear train in neutral, the system is capable of making a pivot turn. Since this steering mechanism obtains its power from a point in the power train ahead of the transmission gears, the steer ratio changes with changes in the transmission ratio. The normal direction of turn is reversed when the vehicle is driven backwards. This characteristic can be eliminated reversing the signal from the steering control when the transmission is placed in reverse.

Because the number of steering ratios equals the number of propulsion ratios, it is not necessary to depend on the braking system alone to achieve various turning radii. Horsepower losses, wear, and cooling problems are much less than those of a comparable controlled differential steering system.

12-23.1.4 Merritt's Double Differential

A double differential steering system shown in Fig. 12-40 is similar to that described by H. E. Merritt (Ref. 19). The steering differential may be driven from several points in the system. If it is driven from Point 1, the system has characteristics identical to those of the controlled differential illustrated in Fig. 12-36; if driven from Point 2, to those of the double differential (Fig. 12-40); and, if driven from Point 3, changes in the power train gear ratio will affect the steer ratio.

Experience has shown that driving from Point

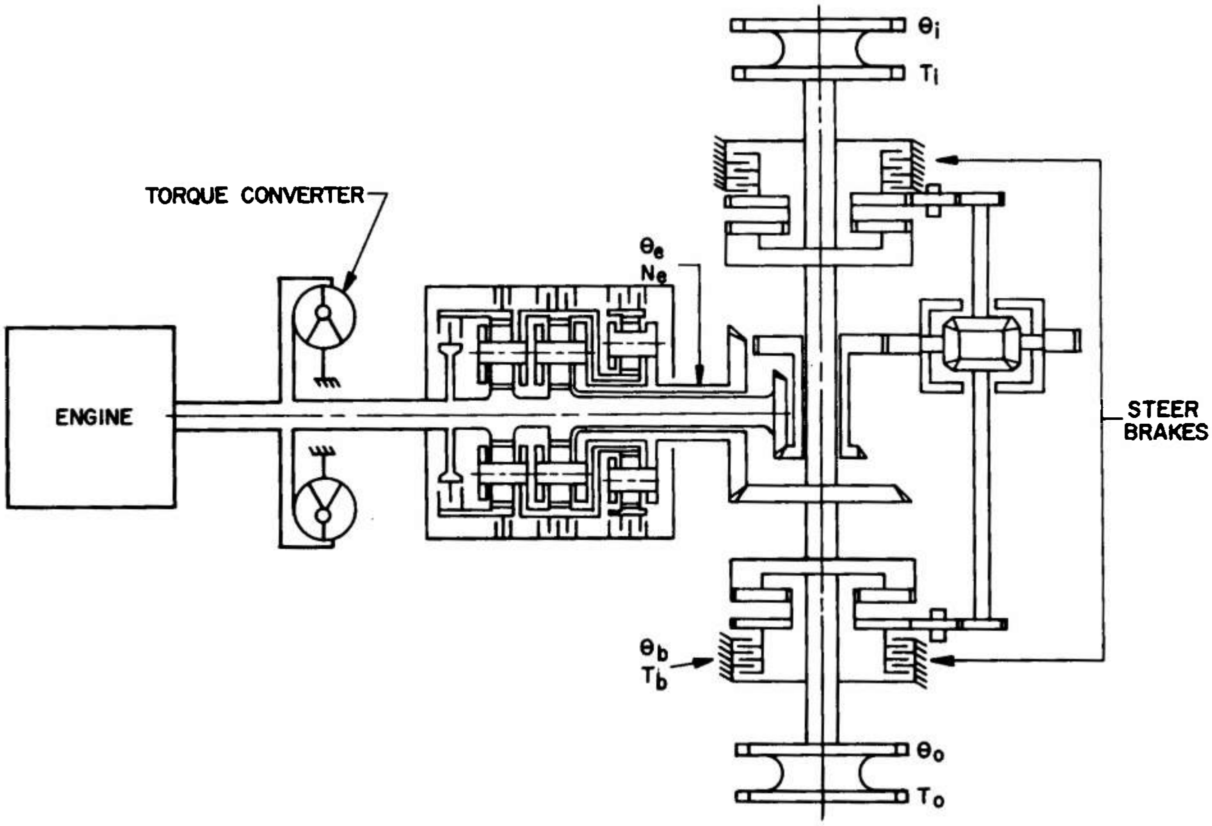


Figure 12-42. *Triple Differential—Regenerative Torque Propulsion*

3 may provide too great a range of steer ratios. Two unsatisfactory performance characteristics result: (1) turning radii become excessively large in the high ranges (especially on downgrades); (2) turning radii are smaller than necessary in the lower ranges (excessive power required to steer in the lower ranges).

12-23.1.5 Triple Differential

12-23.1.5.1 Split Torque Propulsion

The triple differential steering system, shown in Fig. 12-41, forms a split torque propulsion system, since a portion of the propulsion power is transmitted by the steering system gearing.

Turning is accomplished by increasing the speed of one track and decreasing the speed of the opposite track a like amount, hence, there is no change in the average overall vehicle propulsion ratio as a result of a turning maneuver. The overall effective propulsion ratio is, however, affected in this system by virtue of the split torque feature, since a portion of the total power is taken from ahead of the propulsion gear change set while the remaining portion passes through the gear change set. The effective propulsion ratio approaches the propulsion ratio of the steering system as the propulsion gear ratio approaches infinity.

This system is regenerative, is capable of making pivot turns, and has no tendency to reverse steer. The steer ratio changes in a manner similar to the double differential system, Fig. 12-40, when a change is made in the propulsion gear ratio.

12-23.1.5.2 Regenerative Torque Propulsion

The system shown in Fig. 12-42 is similar to the system described previously with the exception that the sun gears of the steering epicyclical gear sets turn in a direction opposite to that of the output shafts. This results in a recirculation of part of the output power back through the steering system to the propulsion gear set. A change in the steer ratio occurs when a change is made in the propulsion gear ratio.

12-23.1.6 Double Differential (Hydrostatic)

This system, shown in Fig. 12-43, consists of a double differential steering arrangement actuated

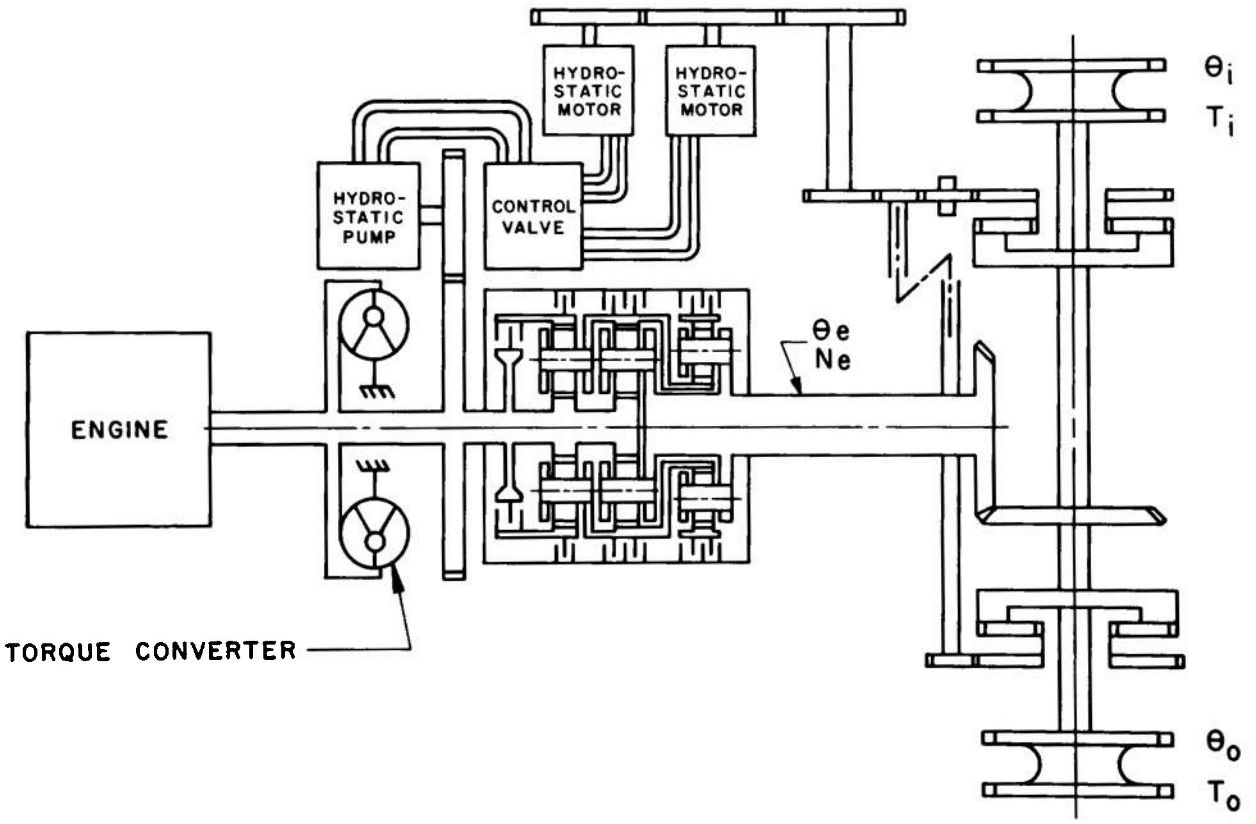


Figure 12-43. ***Double Differential***

by means of variable displacement hydrostatic pump and motor system which is driven from the input shaft of the propulsion gear set. The output of the hydrostatic motor is infinitely variable within its design range. Therefore, in theory, this system provides an infinite number of steer ratios, thus, eliminating the power lost by slipping clutches.

Other performance characteristics include regenerative steering and no tendency to reverse steer.

12-23.2 TYPE B SYSTEMS

12-23.2.1 Geared Steering System

The geared steering system shown in Fig. 12-44 consists of two, 2-speed gear sets (one for each track) interposed between the propulsion gear set and the sprockets. By changing the ratio of one of the gear sets, the relative speeds of the tracks are changed to effect a turn. During such a maneuver, the system is regenerative; however, by proper declutching and braking, this system functions as a clutch-brake system. In the latter case, it is not regenerative. Timing of the clutching and braking operations must be adjusted to prevent reverse steering when the tracks drive sprockets.

When a turn is affected by changing the ratio of one gear set, the outer track does not increase in speed. The mean propulsion ratio is therefore affected and the increment in torque required to make the turn is less than for the controlled double or triple differential systems.

A true pivot turn cannot be made with either mode of operation; however, a pivot about one track can be made only in the clutch-brake mode. With this system, the steer ratio is not affected by a change in the propulsion ratio or braking effort required and is not constant. It is a function of the grade encountered, increasing with negative grade and decreasing with positive grade.

Another geared steering system proposed by Merritt is shown in Fig. 12-45. This system has characteristics similar to the system shown in Fig. 12-46; however, it does not incorporate a clutch-brake.

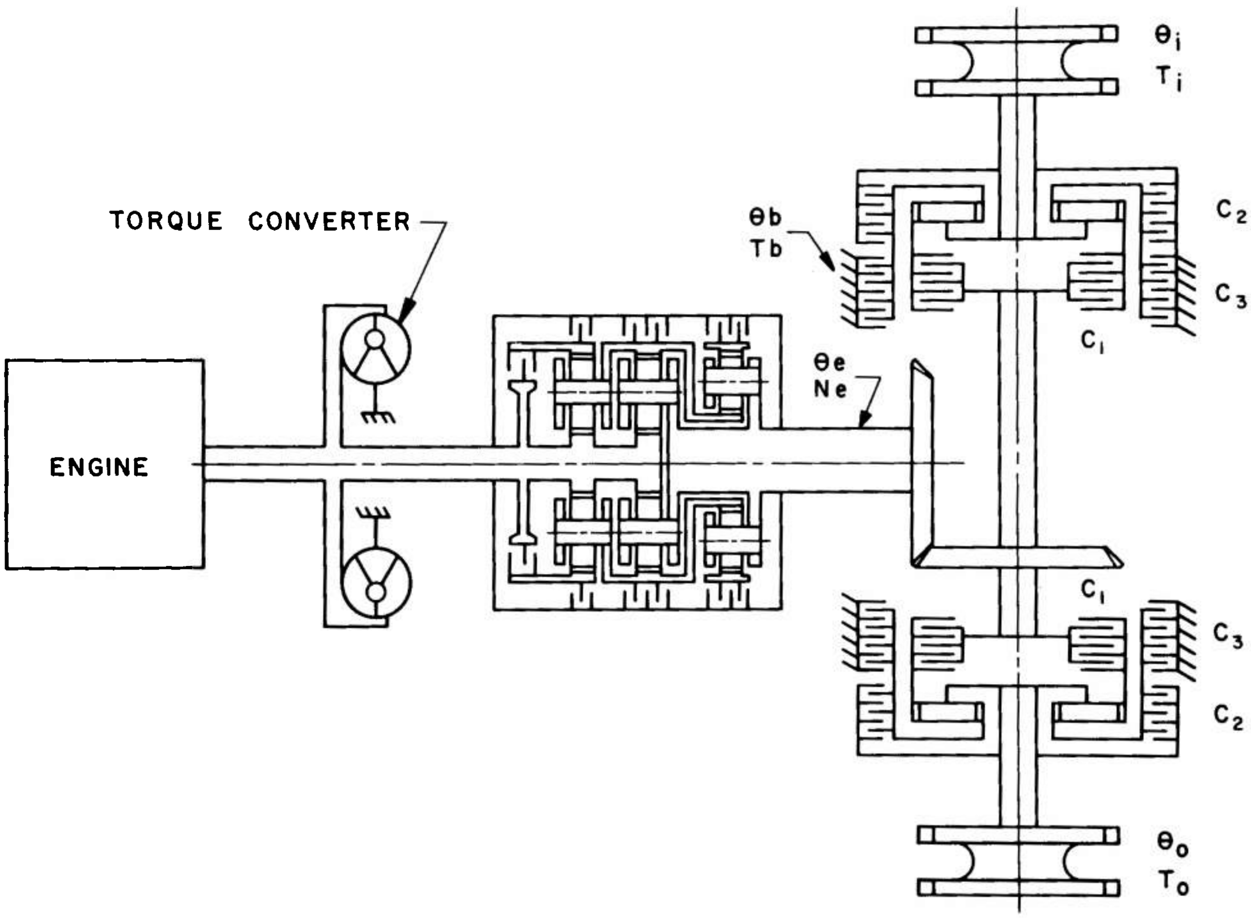


Figure 12-44. *Geared Steering*

12-23.2.2 Clutch-Brake System

The clutch-brake system, shown in Fig. 12-46, is the simplest steering system suitable for track-laying vehicles. Turning is accomplished by disconnecting the power train from one track and then applying a single brake to this track; concurrently, power is maintained or increased to the opposite track. The clutch-brake system is not regenerative under any circumstances. Because a regenerative steering system must be used if the track length to tread, L/T, ratio exceeds approximately 1.3, and because power losses are high whenever the clutches or brakes are slipped, the clutch-brake system has only limited application in military vehicles.

Although this system provides maximum straight-ahead stability and pivot turns about one track are possible, it is dangerously unstable, with respect to reverse steering, unless an interlocking clutch-brake control system is provided. Clutch-brake steering has the advantage that high draw-bar loads, positive grades, and high-rolling resistance assist the steering effort.

12-23.2.3 Gates Steering System

The Gates steering system is shown in Fig. 12-47. For straight-ahead operation, the drive clutches, C_d, are engaged and the steering clutch, C_s, is disengaged. To execute a turn, the drive clutch to the inner track is released and the steering clutch applied. This effects a ratio change to the inner track while ratio to the outer track remains the same. This system has performance characteristics that are identical to those of the geared steering system shown in Fig. 12-45.

12-23.2.4 Multiple Ratio Geared Steering System

The multiple ratio geared steering system, Fig. 12-48, provides a change in steer ratio for each ratio change in the propulsion gear set. During straight-ahead operation, clutches C_1 are normally engaged and clutches C_2 are released. To effect a

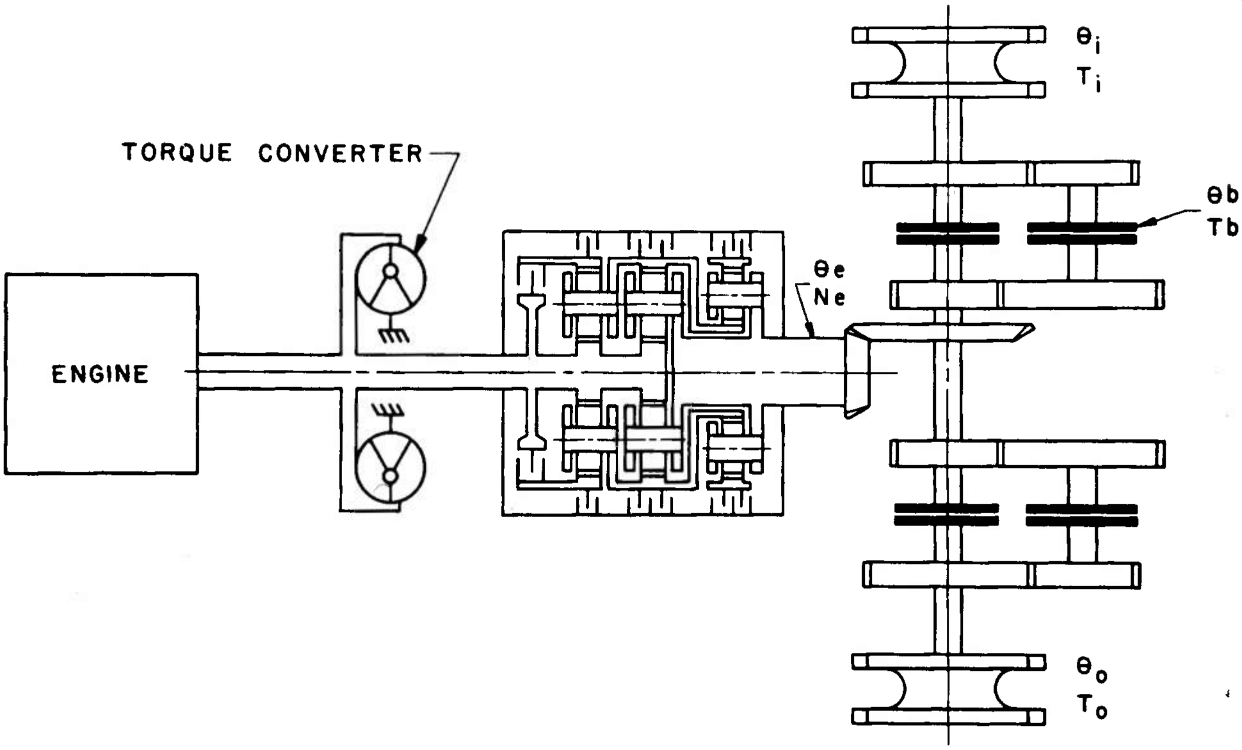


Figure 12-45. *Merritt's Geared Steering*

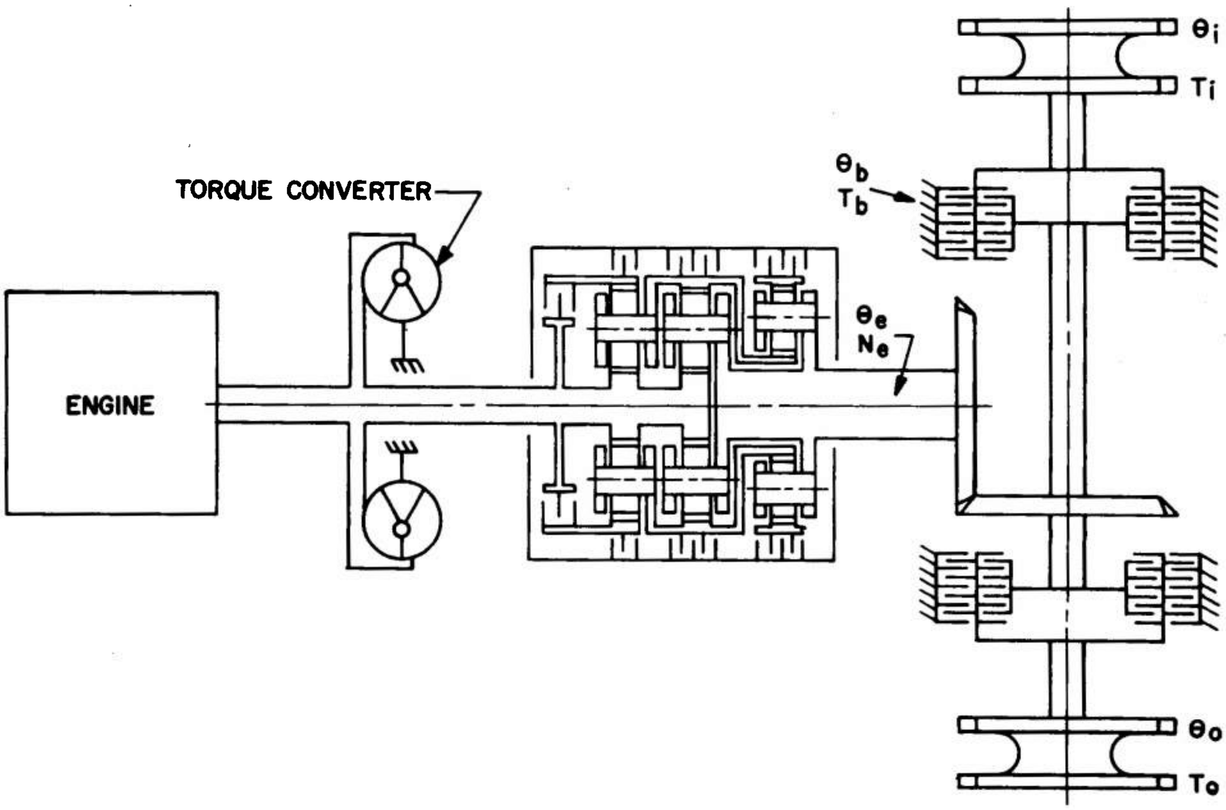


Figure 12-46. *Clutch-Brake System*

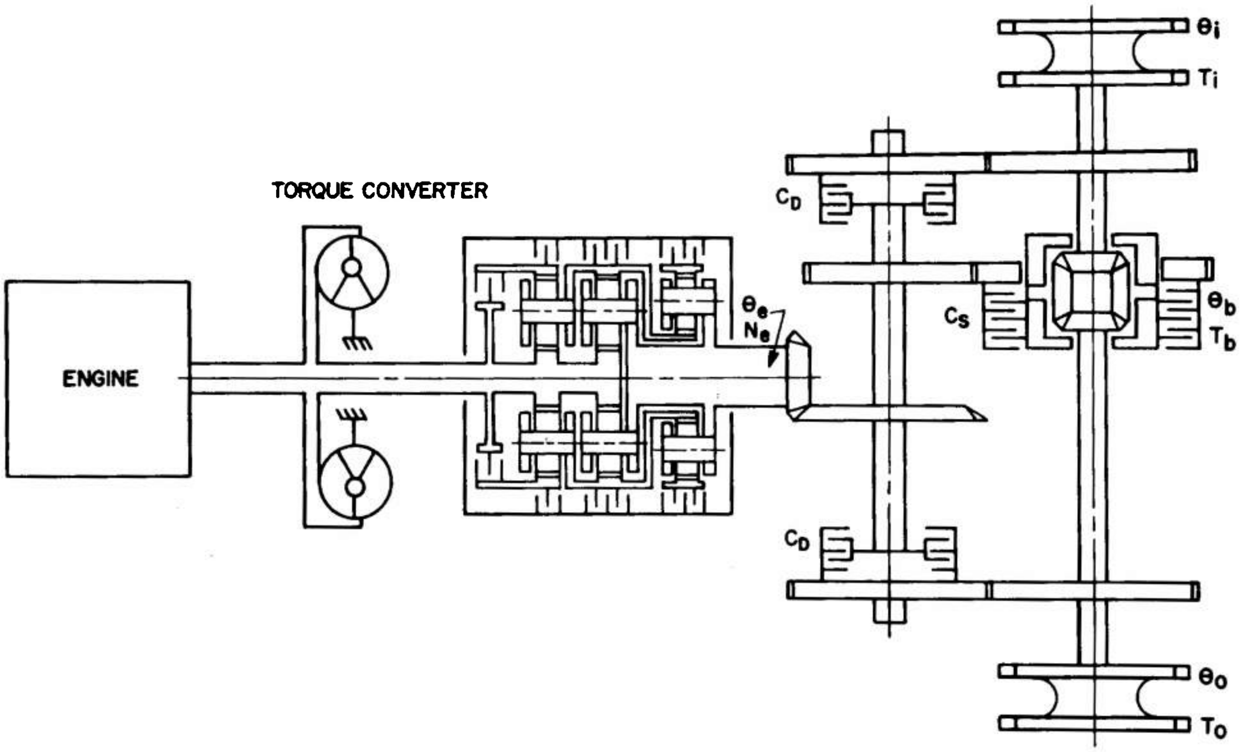


Figure 12-47. Gates Steering System

turn, clutch C_1 to the inner sprocket is released and the corresponding clutch C_2 is engaged. This results in a ratio change (downshift) at the inner sprocket.

12-23.3 TYPE C SYSTEMS

Geared Steering System (Inverse Operation)

This system, Fig. 12-49, is identical to the geared steering system, Fig. 12-44, except in the means by which it is operated. For straight-ahead operation, clutches C_1 and C_3 are engaged and clutch C_2 is disengaged. To effect a turn, clutch C_3 to the outer track, is released and corresponding clutch C_2 is engaged. The result is an upshift to the outer track while the clutch arrangement (and the gear ratio) for the inner track remains the same. With this system, power loss during a turn, is higher than for any other system described herein.

12-23.4 MISCELLANEOUS TYPES

12-23.4.1 Independent Propulsion Steering System

An independent propulsion steering system has a separately controllable source of power for each track. Various arrangements of this type, e.g., electric drive, hydrostatic drive, and multiple power plant, have been proposed or experimentally tested. An independent steer arrangement utilizing two power plants and separate propulsion range gear sets is shown in Fig. 12-50.

The overall performance of an independent propulsion steering system depends on the characteristics of the component units. The system shown in Fig. 12-50 has a fixed number of steer ratios and is nonregenerative; a conventional hydrostatic drive system would have an infinite number of steer ratios and could be made regenerative. Both systems can execute true pivot steering. The hydrostatic or a suitable electric drive system can effect total vehicle braking by means of reversed power flow or regeneration. The hydrostatic steering transmission is discussed in the following section.

12-23.4.2 Hydrostatic Steering Transmissions (Ref. 22)

The basic direct-acting hydrostatic steering transmission consists of a hydraulic pump (the source of pressure), hydraulic traction motors for each track, a fluid reservoir, and a flow control system.

The hydraulic traction motor is ideally suited

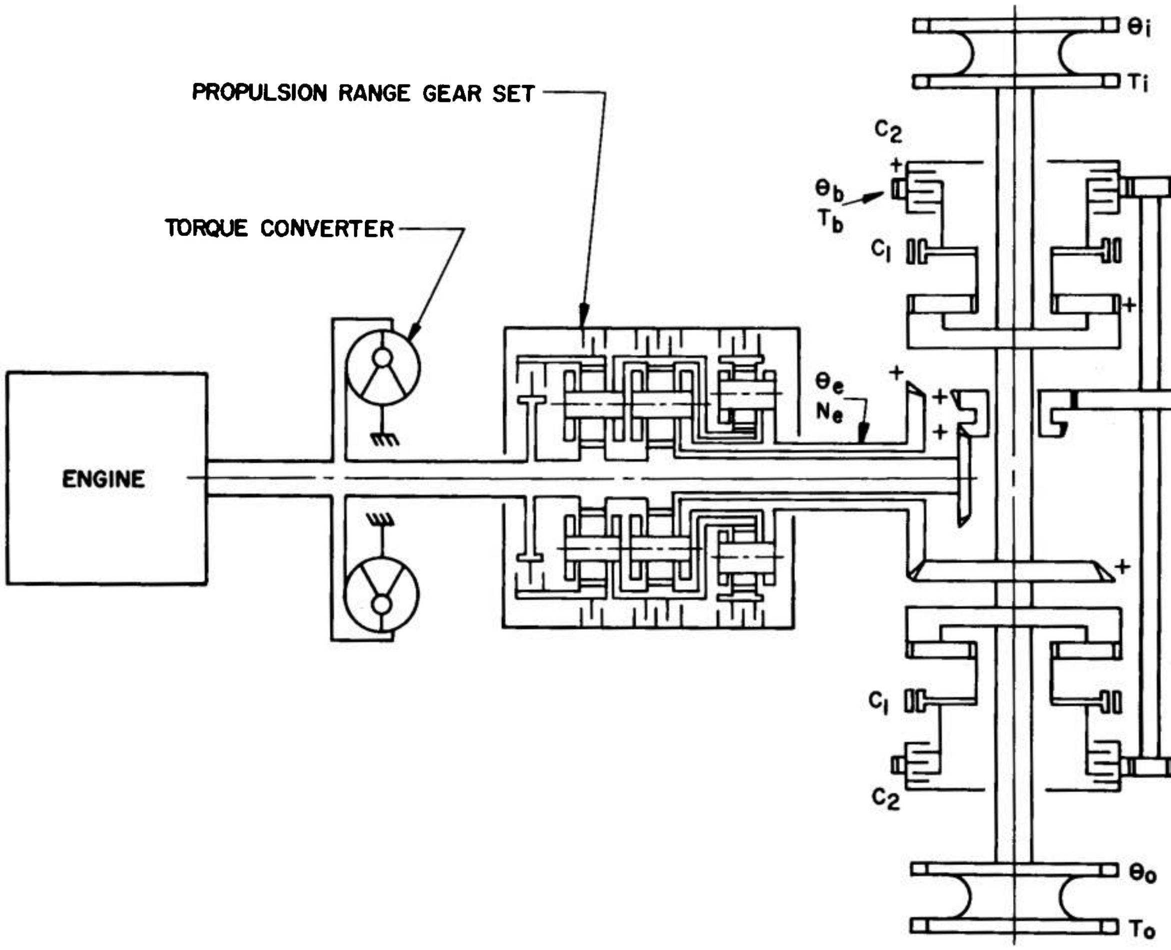


Figure 12-48. *Geared Steering with Multiple Ratios*

for vehicle propulsion since the output power, HP_{out}, is a function of the line pressure and the fluid flow to the motor both of which may be easily controlled. The functional relation is

$$HP_{out} = \frac{PQE_vE_t}{1715} \tag{12-53}$$

where

P is the differential pressure between the motor inlet and outlet ports, psi
Q is the fluid flow to the motor, gpm
E_v is the volumetric efficiency of the motor
E_t is the torque efficiency of the motor

It is possible, therefore, to achieve ideal vehicle speed-torque characteristics at the sprockets by maintaining line pressure and varying the flow to the pump.

The operating characteristics of a given system will depend on the type and characteristics of its individual elements. For example, if a variable displacement pump is used in conjunction with a fixed displacement motor, infinitely variable speed selection in either direction* is obtainable. This arrangement provides a constant output torque over a wide speed range.

Steering maneuvers with a pump and motor hydrostatic transmission are effected by changing the relative speeds of the tracks, no friction braking being required. The system provides true pivot turns (the opposing tracks move at the same speed in opposite directions) and has infinite turning radius capabilities. Regenerative steering could be obtained if the flow from the inner track motor is directed to the outer track motor at the early stage of the turn. No clutch or brake slippage occurs during a turn, therefore, power loss from these sources is not incurred.

* A means of changing the direction of flow through the motor must be provided.

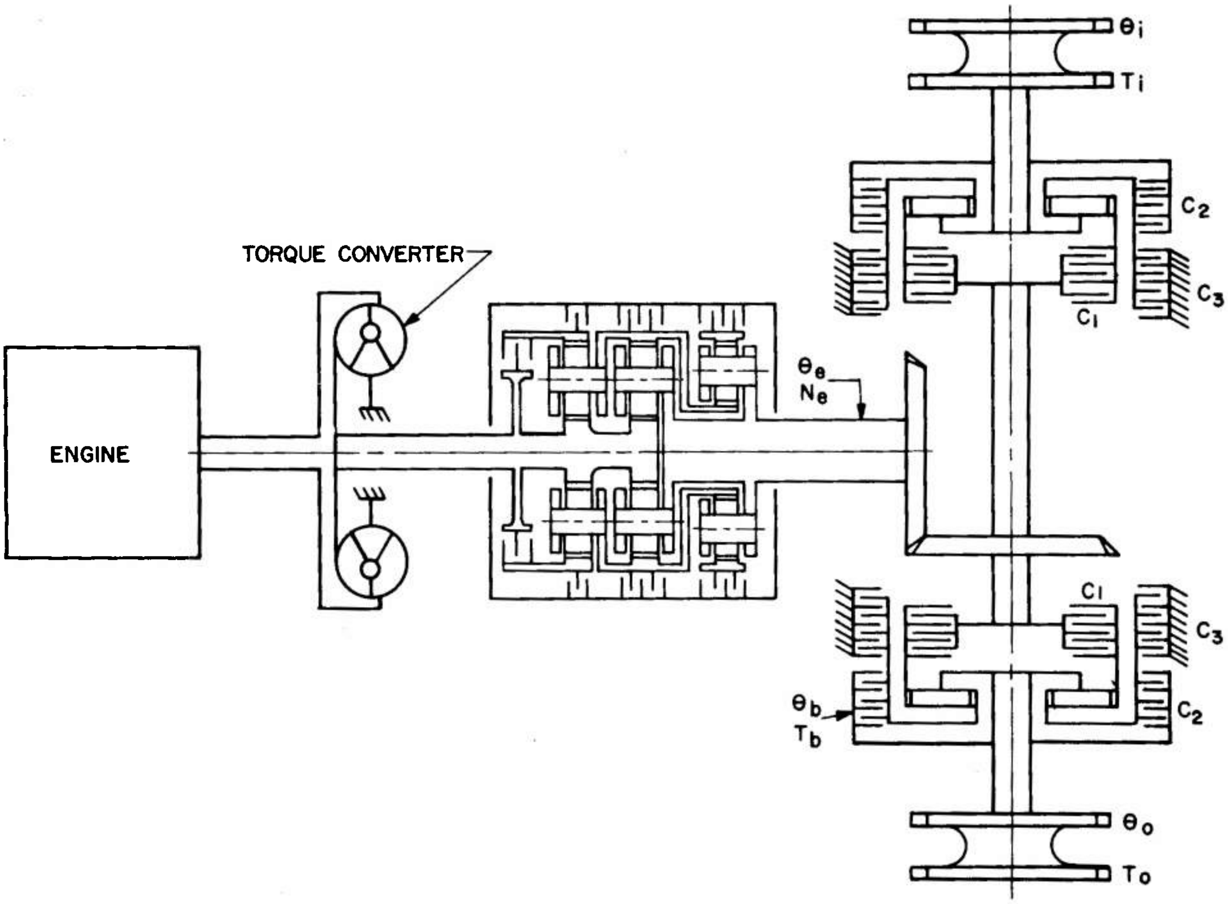


Figure 12-49. ***Geared Steering with Inverse Operation***

Braking can be incorporated into the pump and motor of a hydrostatic transmission by providing a means of slowing and reversing the motors.

Various combinations or arrangements of pumps and motors may be used for the propulsion-steering system of a track-laying vehicle. These include:

1. A single pump and two motors.
2. Single power plant, two pumps and two motors.
3. Separate power plant, two pumps and two motors.

A typical differential hydrostatic transmission is described in par. 12-23.4.2.

The major advantage of the differential hydrostatic transmission, in comparison with the direct pump-motor system, is its higher overall efficiency. If the components of the system are properly arranged, the losses in straight-ahead motion will be minimized since the flow work of the hydraulic system will be limited to counteracting hydraulic slippage losses, i.e., the power transfer is through the mechanical elements. Under these conditions, the overall efficiency can exceed 90%, as compared with the 80% to 85% attainable with a straight pump and motor. For the differential hydrostatic system efficiency will decrease as the flow work increases, approaching that of the pump-motor system as this quantity becomes large.

Another advantage of the hydrostatic differential transmission is that a pump of comparatively small displacement will provide wide range of output speeds. This arises because the output speed in the differential arrangement is determined by the direction of rotation of the hydraulic motor. Output speed in the straight pump and motor system is determined by the unidirectional speed range of the motor.

An advantage of the straight pump-motor system is its greater flexibility. The differential system can drive the track sprockets only in a single

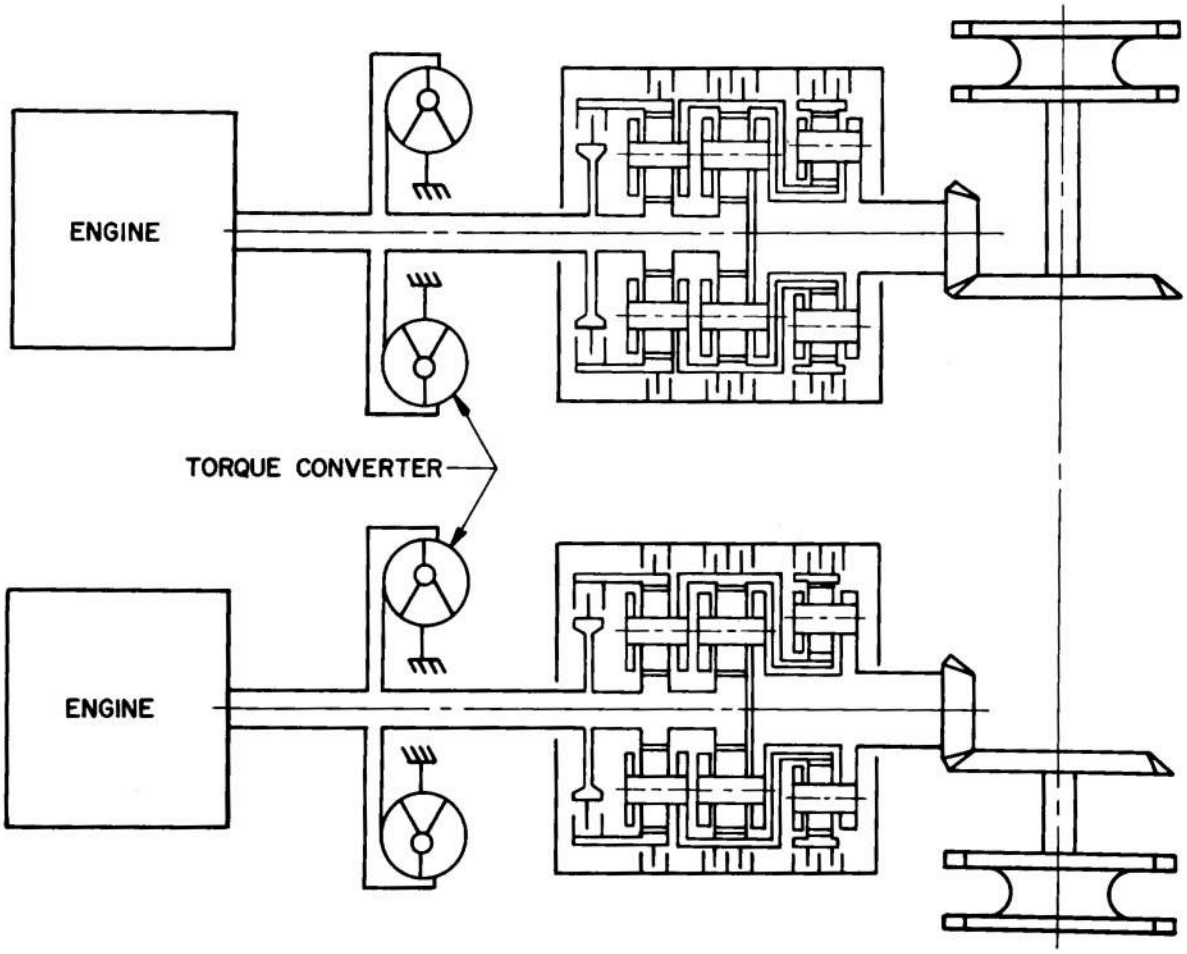


Figure 12-50. *Independent Propulsion*

direction for a given input shaft rotation. Also, the differential transmission does not have the same degree of freedom in the remote location of the various elements as does the pump and motor system.

12-24 ANALYTICAL CONCEPTS

The important parameters related to the steering requirements are listed in Section III of the present chapter. The steerability of track-laying vehicles and the response of a given steer mechanism depend, to a large extent, on the geometric configuration of the vehicle and the load distribution along the ground contact area.

A rigorous treatment of the mechanics of steering of tracked vehicles is beyond the scope of this handbook. The subject is thoroughly discussed in Refs. 17, 18, and 19. The present discussion describes a number of the fundamental relationships which influence steering behavior of tracked vehicles.

12-24.1 FORCE CONSIDERATIONS (Ref. 18)

Among the fundamental parameters related to the steering of a tracked vehicle, during a turn are the propelling forces required at each track. If a uniformly loaded vehicle moves in a curved path (Fig. 12-51), the propelling forces may be designated as F_o (outer track) and F_i (inner track), the motion resisting forces at each track as $R/2$, and the moment of resistance to rotation about the vehicle center of gravity as M_o. The equations of motion for the vehicle can be written by considering the total displacement of the vehicle to be made of a combination of linear and angular displacements. These become

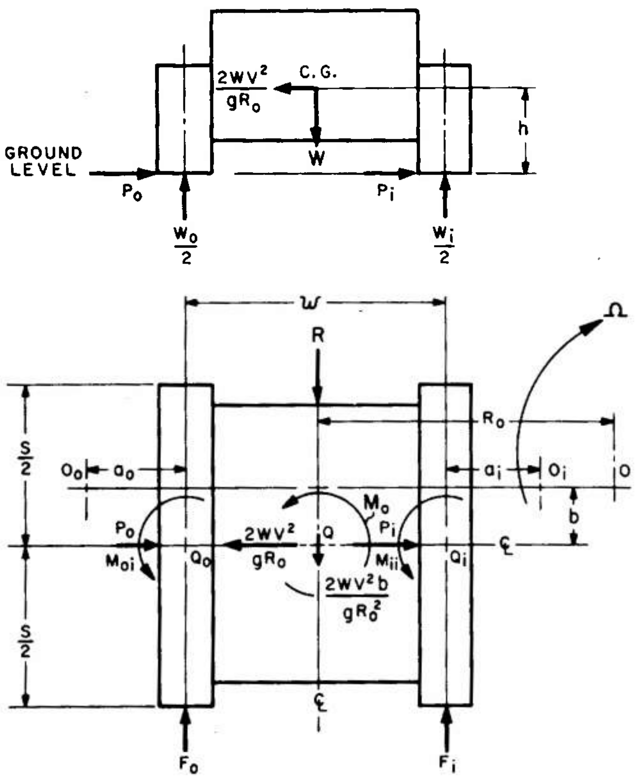


Figure 12-51. ***Dynamics of Tank During a Sustained Turn***

$$m\,\frac{d^2\,s}{dt^2} = F_i + F_o - R \tag{12-54}$$

$$I\,\frac{d^2\,\Theta}{dt^2} = \frac{w}{2}\,(F_o - F_i) - M_o \tag{12-55}$$

where

m is the mass of the vehicle, lb per ft per sec per sec

$\frac{d^2\,s}{dt^2}$ is the linear acceleration of the vehicle, ft radians per sec per sec

I is the moment of inertia of the vehicle about the center of rotation, lb-sec^2-ft

$\frac{d^2\,\Theta}{dt^2}$ is the angular acceleration of the vehicle, per sec per sec

w is the transverse distance between centerlines of the track, ft

In the case of a vehicle moving at constant velocity along a path of constant radius, the left hand terms of Eqs. 12-54 and 12-55 become zero, and as a result, the force equations become

$$F_i = \frac{R}{2} - \frac{M_o}{w} = f_o\left(\frac{W}{2}\right) - \frac{M_o}{w} \tag{12-56}$$

$$F_o = \frac{R}{2} + \frac{M_o}{w} = f_o\left(\frac{W}{2}\right) - \frac{M_o}{w} \tag{12-57}$$

where

R is the total rolling resistance of the vehicle, lb

$f_o\left(\frac{W}{2}\right)$ is the rolling resistance of one track expressed as function of total vehicle weight, W, lb

w is the distance between track centerlines, ft

These equations represent the general case under the stated conditions.

The term describing rolling resistance can be calculated by methods outlined in Chapter 5. The terms containing the turning resistance moment, M_o, must be evaluated by considering the load distribution on the tracks. If it is assumed that a track of length S supports a uniformly distributed load, $\frac{W}{2}$, and the coefficient of lateral friction between the track and ground is μ, the force equations become

$$F_i = f_o\left(\frac{W}{2}\right) - \mu\frac{WS}{4w} \tag{12-58}$$

$$F_o = f_o\left(\frac{W}{2}\right) + \mu\frac{WS}{4w} \tag{12-59}$$

If the ground pressure distribution of the vehicle is not uniform, the value of the moment, M_o, will change. Methods for calculating the moments under various loadings are presented in Ref. 18. As an example, if the distribution of ground pressure is triangular with the maximum pressure at the transverse centerline of the track at a distance, $S/2$, from the end of the track and zero at either end, the force equations become

$$F_i = f_o\left(\frac{W}{2}\right) - \mu\frac{WS}{6\,w} \tag{12-60}$$

$$F_o = f_o\left(\frac{W}{2}\right) + \mu\frac{WS}{6\,w} \tag{12-61}$$

In general, the moment of turning resistance, M_o, can be calculated if the load distribution is known and the center of ground pressure can be determined. If the load distribution is expressible in functional form, analytical integration methods may be used in its calculation. If the loading is not so expressible, graphical integration is required.

As shown by the previous equations, resistance to turning is a function of weight distribution on the tracks (ground pressure distribution). It is possible, therefore, to reduce the power required for steering by adjusting the distribution of ground pressure for a given vehicle. The equations also indicate that, for the stated conditions the outer track force is always larger than the inner track force, and that these forces are independent of the radius of turn and the vehicle speed. Experimental results (Ref. 20) indicate that the slewing force, defined as the algebraic difference of the outer and inner track, $F_o - F_i$, decreases as the radius of turn increases.

The actual mechanism of steering is more complex than that obtained by assuming a constant ground pressure. The center of ground pressure and hence the turning resistance moment, M_o, varies with external factors such as centrifugal forces, slope induced forces, and drawbar loadings.

The ratio*, L/T, of track length L to tread T influences the steering problem. The upper and lower limits of the L/T ratio for satisfactory steering and stability are determined by the interaction of tracks and ground (Ref. 23). If the ratio exceeds a value of approximately 1.7, the average vehicle operating on the most favorable ground surface will not steer without excessive slippage of the outer track. In addition, higher ratios impose excessive power demands on the vehicle. If, however, the ratio is less than 1, steering becomes relatively unstable and excessive use of the steering brakes becomes necessary for average vehicle and ground conditions. In actual practice, values between 1.125 and 1.69 are encountered.

When a track-laying vehicle turns on a side slope, α, a lateral force of magnitude, $W \sin \alpha$, acting parallel to the surface of side slope tends to move it down the slope. The total reaction force parallel to the ground surface developed by the track and ground will equal this force (assuming no side slippage occurs). Because these lateral reaction forces are proportional to the ground contacting forces which are perpendicular to the ground surface, they will be distributed along the tracks in accordance with the ground pressure distribution. These reaction forces act in the same plane as the turning resistance forces, therefore, they will cause the center of rotation to change with respect to the level ground position. If the vehicle turns up the slope, the combination of forces acting on the tracks will cause the center of rotation of the tracks to move toward the front of the vehicle. A downward turn on the slope will cause the center of rotation of the tracks to move toward the front of the rear of the vehicle. These changes result because the slope-induced loading is unidirectional while the motion (turning) induced loading depends on the direction of the turn. The center of rotation is determined from superposition of the load distribution profiles of these two effects. In addition, the rolling resistance of each track on the side slope changes owing to the weight transfer that occurs.

For turns carried out on slope by a vehicle having a uniform distribution of load on the tracks (Eqs. 12-58 and 12-59), the force equations become*:

Vehicle turning uphill:

$$F_i = \frac{Wf_o}{2}\left(1 - \frac{2h}{w}\tan\alpha\right) - \frac{W\mu S}{4}\left[1 - \left(\frac{\sin\alpha}{\mu}\right)^2\right] \quad (12\text{-}62)$$

$$F_o = \frac{Wf_o}{2}\left(1 + \frac{2h}{w}\tan\alpha\right) + \frac{W\mu S}{4}\left[1 - \left(\frac{\sin\alpha}{\mu}\right)^2\right] \quad (12\text{-}63)$$

Vehicle turning downhill:

$$F_i = \frac{Wf_o}{2}\left(1 + \frac{2h}{w}\tan\alpha\right) - \frac{W\mu S}{4}\left[1 + \left(\frac{\sin\alpha}{\mu}\right)^2\right] \quad (12\text{-}64)$$

$$F_o = \frac{Wf_o}{2}\left(1 - \frac{2h}{w}\tan\alpha\right) + \frac{W\mu S}{5}\left[1 + \left(\frac{\sin\alpha}{\mu}\right)^2\right] \quad (12\text{-}65)$$

where

h is the height of the center of gravity of the vehicle, ft

α is the angle of the side slope, deg.

(All other symbols have been defined previously.)

Equations 12-62, 12-63, 12-64 and 12-65 indicate that the effect of the side slope is to decrease the moment of resistance (slewing couple) when the vehicle turns uphill, and to increase the moment of resistance when the vehicle turns downhill.

Since the shift in the center of rotation is a direct function of the magnitude of the slope-induced side loading, a limiting value of the side loading exists for a given track length, beyond which the point of rotation will no longer be on the track, and the vehicle will not steer. For example, with uniform load distribution on the tracks, the following relationship holds:

* This ratio is designated as S/w in Eqs. 12-58, 12-59, 12-60 and 12-61.

* Comparable equations could be written for other track load distributions.

$$\sin \alpha \leq \mu \tag{12-66}$$

To prevent a sideward slip of a vehicle as it moves along a side slope, an identical relation must exist.

The centrifugal force, acting at the center of gravity of the turning vehicle, also affects the slewing forces, F_o and F_i, Fig. 12-51. As a result, transfer of load from the inner to the outer track occurs. Lateral reaction forces, P_o and P_i, are generated at the tracks. These lateral forces affect the location of the center of rotation of the tracks as do slope-induced forces. For forward motion of the vehicle, the resulting displacement, b, is always toward the front of the vehicle. If the centrifugal force, F_c, is given as

$$F_c = \frac{W\ V^2}{g\ R_o} V \tag{12-67}$$

where

- V is the linear speed of the center of gravity of the vehicle, ft per sec
- R_o is the radius of curvature of the path of the vehicle, ft

force equations, Eqs. 12-58 and 12-59, for a vehicle operating with a uniform pressure distribution in the static state on level ground become:

$$F_i = \frac{W}{2} f_o - \frac{hWV^2}{wgR_o} - \frac{W\mu S}{4w}\left[1 - \left(\frac{V^2}{gR_o\mu}\right)^2\right] \tag{12-68}$$

$$F_o = \frac{W}{2} f_o + \frac{hWV^2}{wgR_o} + \frac{W\mu S}{4w}\left[1 - \left(\frac{V^2}{gR_o\mu}\right)^2\right] \tag{12-69}$$

The longitudinal component of the centrifugal force acting on the vehicle is neglected in these equations. This longitudinal component of the centrifugal force (Fig. 12-51) is generated by the shift in the center of rotation of the vehicle caused by changes in the centers of rotation of the individual tracks. A discussion of the effects of this component is given in Ref. 17.

The distribution of forces on the tracks as a result of the action of centrifugal force is similar to a vehicle on a side slope turning uphill; the total rolling resistance does not change, however, the moment of turning resistance decreases.

When a tracked vehicle tows a trailer, the drawbar loading will affect the steering characteristics by changing the load distribution on the tracks. Due to the location of the towing hook, there is normally a shift of the center of ground pressure towards the rear of the vehicle. In addition, the drawbar force will change the lateral loading on the tracks. As shown in Ref. 18, the tractor-trailer unit will always require an increased turning moment to negotiate a turn; the required moment being a function of the angle between the axes of the vehicles during the turn.

12-24.2 TRACK VELOCITY CONSIDERATIONS

Longitudinal slippage of the tracks during the slewing process was not considered in the previous discussion. References 17 and 18 treat the case where both lateral and longitudinal track slip occurs during turning.

A consequence considering longitudinal slip is that the radius of curvature of the vehicle path can no longer be expressed as a simple function of the angular velocities of the two tracks and the distance between them, as is the case with a pair of wheels on a common axle containing a differential mechanism.

As indicated on Fig. 12-51, the hull turns about point O with an angular velocity, Ω, at a radius, R_c. The velocity of various points on the hull can be expressed as

Ω, R_c at the center of the vehicle

$\left(R_c + \frac{w}{2}\right) \Omega$, at the center of the outer track

$\left(R_c - \frac{w}{2}\right) \Omega$, at the center of the inner track

In addition, the individual tracks each have angular velocities of Ω about their respective instantaneous centers, O_o and O_i. The instantaneous centers of rotation are determined by the resultant slip velocities of the tracks, while the angular velocity of the tracks about their instantaneous centers is assumed identical to the angular velocity of the vehicle around the point O, i.e., Ω. The resulting instantaneous center of track rotation can be on either side of a track, depending on the direction of the resultant slip velocities of the tracks. Radii of slip, a_o and a_i, are called positive or negative, depending on whether the centers O_o and O_i are on the same or opposite sides of the track as the turning center O.

Based on the above considerations, the vehicle turning radius, R_c, can be expressed as

$$R_c = \frac{\frac{w}{2}\left(\frac{\omega_o}{\omega_i} + 1\right) + a_o - a_i\frac{\omega_o}{\omega_i}}{\frac{\omega_o}{\omega_i} - 1} \tag{12-70}$$

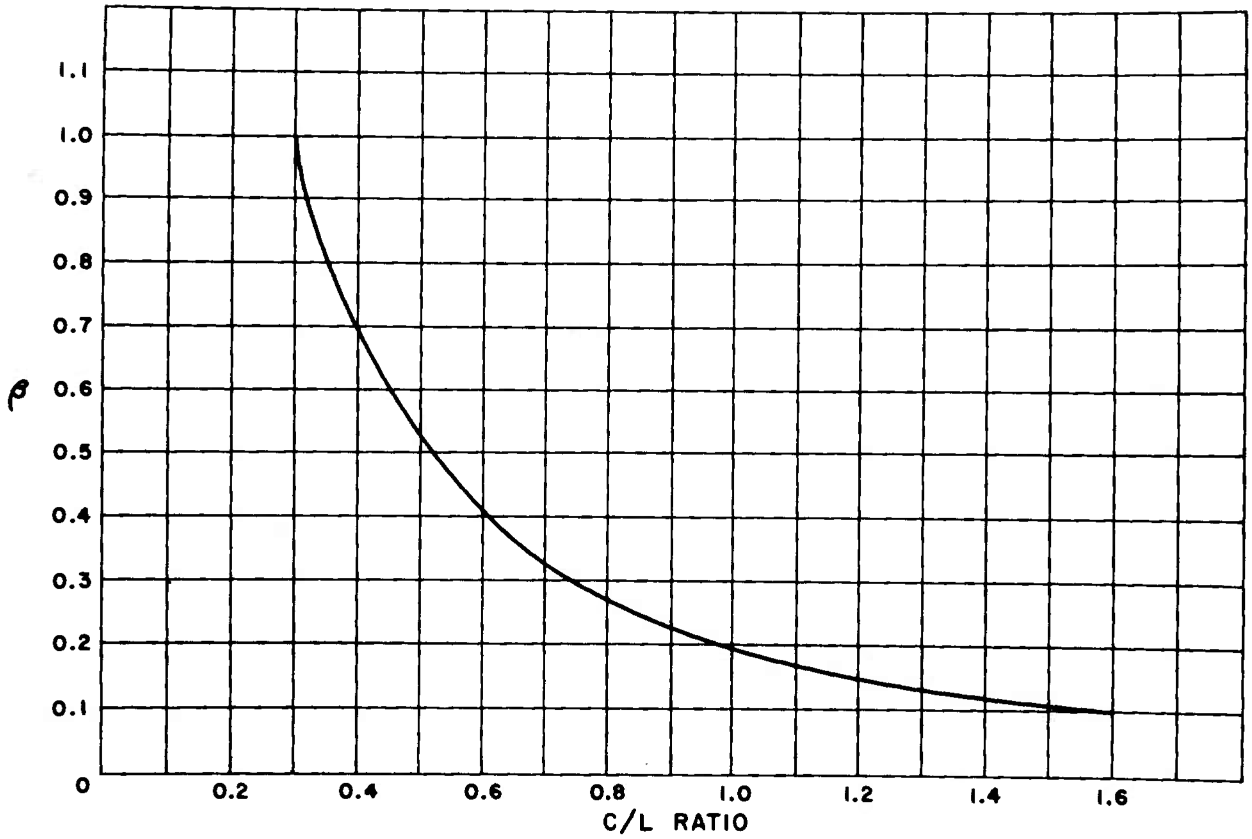


Figure 12-52. *Graph of β vs C/L (Ref. 19)*

where

w is the width between track centers, ft

ω_o is the angular velocity of the outer sprocket

ω_i is the angular velocity of the inner sprocket

a_o is the radius of slip of the outer track, ft

a_i is the radius of slip of the inner track, ft

The positions of the instantaneous centers for a given vehicle operating on a specific type of ground depend upon the longitudinal and transverse coefficients of traction of the system. Methods of predicting the magnitude and position of these centers are presented in Ref. 17.

An alternative equation predicting the turning radius of a tracked vehicle from Ref. 19 is

$$R_c = C\left(\frac{V_o+V_i}{V_o-V_i}\right)(1+\beta\ L/C) \qquad (12\text{-}71)$$

where

R_c is the radius of turn measured to the centerline of the vehicle, ft

C is one-half of the track center-to-center distance, ft

V_o is the outer track velocity, ft per sec

V_i is the inner track velocity, ft per sec

L is one-half of the length of track in contact with the ground, ft

β is the instantaneous center factor taken from Fig. 12-51.

Figure 12-52 shows the relationship between β and the C/L ratio.

12-24.3 STEERING OF ARTICULATED TRACKED VEHICLES (Refs. 18, 24, 25)

An articulated tracked vehicle consists of two powered track-laying units joined together at a hinge or pivot point (Fig. 12-53). Steering control is accomplished by applying torques about the hinge in order to yaw the two units with respect to each other. Unlike the usual or nonarticulated tracked vehicle, a differential track velocity is not utilized to effect the turn.

Investigations in the field of land locomotion (Refs. 18, 25) have shown that the most effective way to maximize the drawbar pull-weight, DP/W, ratio for a tracked vehicle, (assuming that the

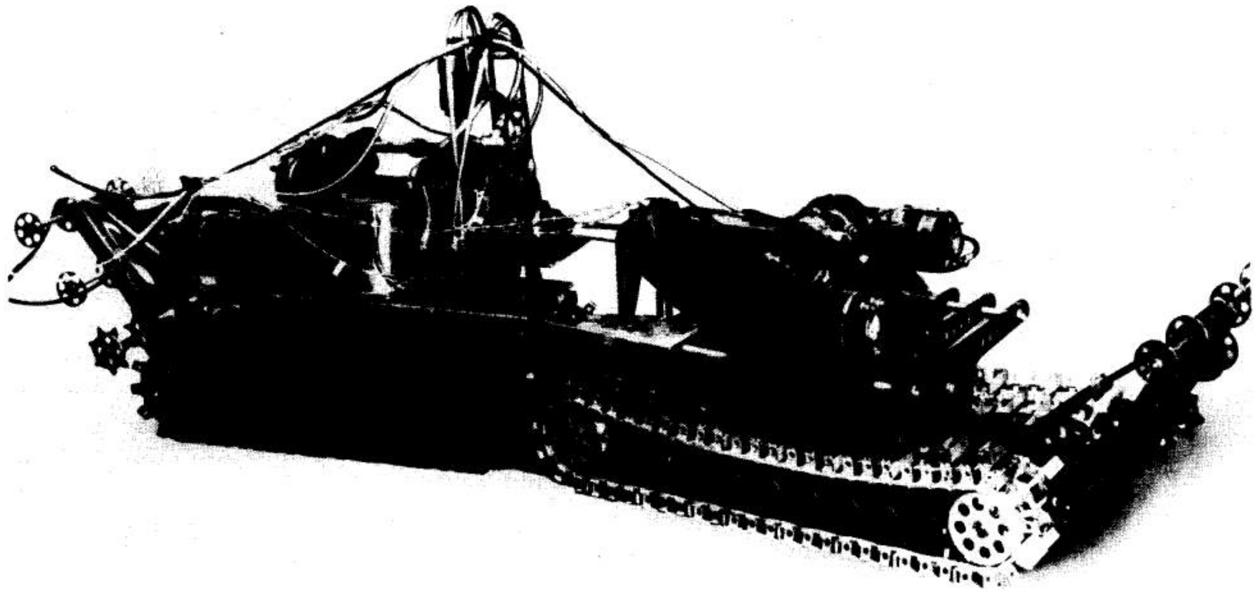


Figure 12-53. *Scale Model of Articulated Spaced-Link Tracked Vehicle (Ref. 25)*

total weight remains constant) is to reduce the ground pressure, p. Since the ground pressure is a direct function of the length of ground contact area of the track, L, and the track width, b, average ground pressure for a given vehicle weight can be reduced by increasing L, b, or both. Reference 25 indicates that to increase the DP/W ratio, it is much more effective to increase the track length L than the track width b. With the conventional tracked vehicle, however, steering characteristics are influenced by the L/T ratio. The following table expresses a qualitative evaluation of the effect of the L/T ratio on steering (Ref. 25):

Range of L/T Ratio	*Steering Characteristics*
1.0 to 1.2	Very good
1.2 to 1.4	Good
1.4 to 1.6	Average
1.6 to 1.8	Poor
1.8 or more	Very poor

The steering limitations imposed by the L/T ratio effects are among the factors that have led to proposed and actual experimental articulated tracked vehicles.

A complete analysis of the turning behavior of articulated track-laying vehicles is presented in Ref. 24. Some of the conclusions and recommendations resulting from this study are summarized below.

The steering response and behavior of articulated track-laying vehicles depend on many parameters. These include the magnitude of the centrifugal acceleration, the ratio of rear-section-to-front-section weights, and the rear-to-front-section length ratio, the effect of angle of articulation and the torque requirements to achieve or maintain various angles of articulation.

Analytical and experimental studies of vehicle behavior utilizing articulated steering at lateral accelerations to 0.4 g have produced the following conclusions:

1. Turning response is a linear function of articulation angle and track speed.
2. Rear-to-front-section length and weight ratios greater than unity result in oversteering behavior, i.e., the hinge moment requirements become negative as the path curvature (radius of turn) per unit articulation angle increases with increasing track speed.
3. Rear-to-front-section length and weight ratios less than unity result in understeering behavior, i.e., the hinge moment requirements become positive as the path curvature per unit articulation angle decreases with increasing track speed.
4. Rear-to-front-section length and weight ratios of unity (symmetrical vehicle) result in neutral steering as evidenced by constant steady-state path curvature with increasing track speed.
5. Symmetrical vehicles require minimum power for turning. (As in conventional tracked vehicles, the power required for the turning process exceeds that required for straight-ahead motion.)
6. Symmetrical vehicles require a minimum of steering system hinge moment.

As the vehicle yawing response increases with increasing articulation angle and/or track speed with resultant high values of centrifugal acceleration, the equations of motion developed for low values of lateral acceleration become inadequate to predict the vehicle's turning behavior. As the centrifugal acceleration exceeds approximately 0.4 g, the relationship between the yawing velocity (rad/sec) and the articulation angle becomes nonlinear. Furthermore, the functional relation becomes double-valued for a portion of the range. Analytical results indicate vehicle steering becomes unstable at some combination of track speed and articulation angle, i.e., at some yaw rate. If the vehicle can successfully negotiate the unstable region, and if the large power and steering torque requirements of high speed operation are met, a controlled high-speed skid steer with a relatively small turning radius and a large yaw rate is theoretically possible. To bring the vehicle out of the turn, the unstable region must again be traversed.

Since an oversteering vehicle can also encounter a sudden reversal of hinge torque requirements with increasing centrifugal acceleration, the steering system must have sufficient stiffness to maintain a constant articulation angle while passing through the point of discontinuity.

For low values of centrifugal acceleration, the following equation may be used to estimate the yaw rate

$$r \cong \frac{1}{l_1}\left(2\frac{1+{}^{\Gamma}/\gamma}{1+{}^{\Gamma}/\gamma}\right)^{\frac{1}{2}} V_t \tan \delta/2 \qquad (12\text{-}72)$$

where

r is the vehicle yaw rate, rad/sec

l_1 is the front section ground contact length, ft

Γ is the ratio of rear section to front section weight

γ is the ratio of rear section to front section ground contact

V_t is the longitudinal velocity vector of the entire vehicle, ft per sec

δ is the input articulation angle, rad.

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CHAPTER 13

THE ELECTRICAL SYSTEM*

The electrical system is an essential element of the automotive assembly. A malfunction of the electrical system will not only seriously impair the operation of the vehicle but may totally incapacitate the vehicle. When a malfunction occurs during combat, the inevitable result is the failure of the vehicle to accomplish its mission and the probable destruction of the vehicle and its crew by the enemy.

The functions of the electrical system are many and varied: it provides power to crank the primary power plant during starting; it provides surges of high-voltage current to the engine spark plugs to produce engine ignition; it powers miscellaneous electric motors that drive vital systems on the vehicle; it powers lights, heaters, fans, and blowers; it powers radio and radar equipment mounted on the vehicle; it powers electric, electronic, and infrared surveillance devices; it powers fire control equipment; and it powers various gages, instruments, and warning devices.

The electrical engineering that enters into the design of the electrical system of the modern military vehicle includes every branch of the electrical engineering profession. Not so many years ago, automotive electrical engineering was considered of only minor importance and not particularly difficult, as the electrical system on a vehicle consisted primarily of the engine ignition system, headlamps, and heater. The modern military vehicle, however, particularly a combat vehicle designed for aggressive assault and rapid exploitation, is a highly complex mechanism. All of the problems encountered in the design of the electrical system in a civilian vehicle are present in a military vehicle—plus many additional ones which relate to factors such as maximum durability and reliability under the extreme stresses of the military environment, and the need for waterproofing all components to meet the deep-fording and amphibious requirements imposed upon tactical and combat vehicles. Further, the requirement for compaction and weight reduction has led to miniaturization of components. Miniaturization, however, is not always compatible with the ruggedness dictated by the necessity for maximum durability. And so, the problem becomes more complex. A fairly detailed presentation of the fundamentals of automotive electrical systems can be found in TM 9-8000, *Principles of Automotive Vehicles.*

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SECTION I USES OF ELECTRICITY IN MILITARY VEHICLES

Military vehicles utilize electric power to operate many devices. The function of these devices may be classified into eight general groups, namely:

a. Production of mechanical power

b. Power transmission

c. Energy storage

d. Heating

e. Communication

f. Ignition
g. Firing of armament
h. Lighting.

13-1 PRODUCTION OF MECHANICAL POWER

This group of electrical power users includes various motor-operated devices, such as: engine starters, gun traversing and elevating systems, stabilizing equipment, winches, remote control devices, windshield wipers, and miscellaneous solenoid-actuated power devices.

13-1.1 STARTERS

An electric starting motor, commonly referred to as the starter, consists primarily of an electric motor plus various electric controls, gears, and clutch components integrated into a compact unit designed to crank an internal combustion engine until it starts to operate under its own power.

Starters used on military vehicles are 24-v. DC high-torque, low-resistance motors, which operate on electrical current supplied directly from the battery. Because of the limitations of size and battery capacity, a high-speed electric motor is used with a high gear reduction to obtain the torque necessary for starting the primary engine. Standard military starters develop approximately 19.5 to 20 ft-lb of torque and vary in rated speed from 2400 rpm (used with engines of less than 100 hp) to 4000 rpm (used with engines up to 1020 hp). Current requirements for these starters vary from 168 to 212 amp under conditions of sustained operation, with momentary surges upon first energizing the starter to 1.5 to 2 times these values; extreme cold weather may double these values. Table 7-7 shows the power requirements of specific standard military starters.

Since all tactical and combat vehicles are required to have deep-water fording capabilities, the starter assembly must be waterproof. Waterproofing is usually accomplished by completely sealing the starter assembly against the entrance of water. Cooling of the starter is then more difficult, but, because the starter operates intermittently, this is not a serious situation.

13-1.2 GUN TRAVERSING AND ELEVATING SYSTEMS (Refs. 3 and 4)

Three general types of traversing and elevating systems may control the laying of the main armament of a combat vehicle. These are the electrohydraulic system, the hydraulic system, and the electric system. Electric power is required in all three systems.

In the *electrohydraulic system,* electricity powers the gunner's and commander's controls, the pulsing relay circuit that operates the tracking motors, and the constant-speed electric motors that run the variable-displacement hydraulic pumps, which accomplish the actual traversing of the turret and elevating of the gun. The large number of electric motors in this system imposes a considerable power drain on the electric power supply.

The *hydraulic system* utilizes electric power merely to run a constant-speed electric motor that drives a hydraulic pump. By incorporating a hydraulic accumulator into the system, a source of constant hydraulic pressure of 1000 psi is maintained without the need for continuous operation of the motor. The turret can be traversed approximately 80° without motor pumping action.

The *electric system,* as its name would imply, requires electricity in almost all of its elements. Traversing of the turret is accomplished by means of an electric traversing motor that rotates the turret through an appropriate gear train. Elevation of the gun is accomplished by means of a hydraulic cylinder that receives hydraulic power from an electric motor-operated hydraulic pumping unit. All controls, both for the traversing and elevating mechanisms, are electrically operated. This system requires prodigious amounts of electrical power, but its redeeming characteristic is precise rate control in spite of variations in friction between the moving parts or when operating with the vehicle on a slope.

The electric system is very much favored for light and medium tanks, as is the hydraulic system. In the case of heavier tanks, however, where the turret weight attains appreciable proportions, the electric system becomes undesirable because of space requirements. In a large tank, a traversing motor of approximately 2-1/2 hp is necessary. A motor of this size not only drains considerable power, but also occupies considerable space, fighting space, within the vehicle.

Heat dissipation is a major problem associated with electrical equipment. High-torque loads placed on an electric motor cause increased currents to flow through the motor windings, resulting in production of heat. When the vehicle is operating over rough terrain or turning abruptly, the traversing mechanism experiences severe shocks. The electric motor is then called upon to instantaneously exert a momentary high torque to counter this shock loading and to prevent the turret from rotating. When electric motors are required to develop high torques at very low speeds, a considerable amount of heat is generated. Ventilation is employed to dissipate this heat, but the operation of the ventilating equipment also adds to the power drain.

13-1.3 STABILIZING EQUIPMENT

In order for a combat vehicle to fire its weapons accurately while moving, stabilizing equipment is needed to counteract the effects of vehicle movement on the positions of the gun and sighting telescopes. In effect, the gun and sighting telescopes remain fixed in space while the vehicle pitches, rolls, and slews. The stabilizing equipment senses the angular movements of the vehicle axes with gyroscopes and translates these movements into appropriate signals to the traversing and elevating mechanisms of the gun and sighting instruments, so as to maintain these elements on target. The electrical requirements of a gyroscopically stabilized telescope are not particularly severe. The electrical requirements of such a system, when applied to a heavy turret and gun, however, especially when the vehicle is operating on rough terrain or on the side of a slope, may reach 10 to 15 kw.

13-1.4 MISCELLANEOUS MOTOR-DRIVEN DEVICES

A number of miscellaneous electric motors are employed in military vehicles in addition to those already mentioned. These are very small motors, such as used in electric windshield wipers and small fans for cooling low-wattage electronic equipment, and large motors, such as used to drive large hydraulic pumps. Some are continuous-duty motors, while others operate intermittently. Most of these motors are of the DC type having a commutator and brushes. All are required to possess maximum reliability, for the performance of some function vital to the successful operation of the vehicle. Amphibious vehicles, and certain other vehicles that are equipped with a totally enclosed hull, require a bilge pump to remove water that may have entered during an amphibious or deep-fording operation. These bilge pumps are usually electrically driven pumps, which, due to their location, must be extremely rugged and completely waterproof.

13-2 POWER TRANSMISSIONS

Electrical power transmission devices utilize electric means to transfer mechanical power from one place to another. A typical example of a system of this type is an internal combustion engine driving a generator that supplies electricity to power traction motors that, in turn, drive the wheels of a vehicle.

Electrically driven tanks were used in World War II. Two types of electric drives were tested in the United States, namely, the electrogear system and the General Electric system. These are discussed in Chapter 8. Development of electric drives were discontinued largely because of the weight factor. The weight of the generators and motors exceeded the weight of the conventional transmissions and steering systems that they were designed to replace.

A certain amount of interest in electric drives still continues because of several desirable features that such drives would possess. The transmission of power from the engine to the wheels is simplified; infinitely variable drive and steering ratios are possible; and the system permits operation of the main power plant at its most efficient speeds. In view of the recent progress in the development of high-speed gas turbines, a turbo-electric drive may be possible that will resolve the weight barrier.

Another example of an electrical power transmission device used in military vehicles is the electromagnetic or eddy current clutch. Electricity establishes in this clutch an electromagnetic field between the driving and driven elements of the clutch, and the drag of the electrical eddy currents thus induced in the clutch elements produces driving action.

13-3 ENERGY STORAGE

Energy storage devices play an important role in military vehicles. The purpose of an energy storage device is to provide standby power with which vital components or systems can be energized for limited periods when the primary source of energy is inoperative; or, to serve as a reservoir between an energy source and an intermittent load, thereby assuring instant response without frequent or continuous operation of the energy source. The best example of electrical standby power is the storage battery, which is discussed below. Other examples of energy storage devices that use electricity indirectly are: hydraulic accumulators, which maintain constant pressure in hydraulic systems, and hydropneumatic spring systems. In these applications, constant-speed DC motors drive hydraulic pumps to maintain the system pressure between desired limits. Hydropneumatic spring systems are discussed in Chapter 11.

The most convenient means of storing standby electrical power for instant use is the storage battery. The battery is floated on the electric system and supplies the system with instant power any time the generator supply falls below the requirements of the vehicle. When generator output builds up again, automatic relays cut the generator in. Excess generator current recharges the battery.

Storage batteries for automotive vehicles are heavy, bulky, inefficient, and a constant source of maintenance trouble, especially in the arctic regions. All of the disadvantages notwithstanding, no better or more convenient means is available at the present time for obtaining a small supply of instantaneous power. The critical points, however, are the amount of power required and the time over which it must be applied. If the amount of power is large or the time is long, a battery is undesirable. On small vehicles, a large battery can usually perform all required functions. On large vehicles, where an auxiliary generator is supplied for other purposes, this generator also may be made capable of starting the main power plant; the battery, then, need not be very large. In recent years, however, batteries have been improved. The newer batteries are lighter in weight, of stronger construction, and have increased energy storing capacity.

Currently, five types of storage batteries are common: (a) lead-acid, (b) nickel-iron, (c) nickel-cadmium, (d) silver-zinc, and (e) silver-cadmium. All except the lead-acid type are alkaline and can be stored either with or without electric charge. The lead-acid type must be kept charged at all times because lead sulfate, which is formed during discharge, converts gradually to a form that impedes recharging. The following characteristics should be considered when selecting a battery for a particular application:

a. Battery potential, v.
b. Total capacity (current × time = amp-hr)
c. Energy density (ratio of energy to size or weight = w-hr/cu in. or lb)
d. Time to discharge and charge
e. Life in storage, cycling, and floating in circuit
f. Maintenance
g. Mechanical strength
h. Effects of altitude, vibrations, and temperature
i. Cost, initial and operational

Table 13-1 compares some of these characteristics for the five common battery types. Storage batteries for engine starting, lighting, and ignition in military vehicles must meet the following Government specifications: MIL-B-11188B(1), MIL-B-26509, and W-B-131f(3).

Safety must also be considered in choosing a battery. Nickel-cadmium cells can be charged and discharged at high rates without the formation of the corrosive fumes that are characteristic of a lead-acid cell. However, if a nickel-cadmium cell is discharged to a value of electrical potential well below one-half of its normal value, a reversal of chemical action can occur, accompanied by a release of hydrogen and oxygen. As these gases result in a highly flammable mixture, ventilation must be adequate. Batteries for military vehicles are made completely waterproof and are provided with breathers to permit submerged operations. Batteries for arctic operations are being developed with built-in heating devices to prevent the rapid loss of potential at low temperatures.

13-4 HEATING

The need for supplying heat to various components and compartments of the military vehicle places an additional load upon the electrical system, particularly during cold-weather operations. In addition to the heat required for crew comfort and windshield defrosters, heat must be supplied to various components of the fuel system to prevent ice or frost from clogging minute orifices. Diesel fuel must be heated during cold-weather operations to maintain it in a free-flowing state, and both gasoline and Diesel fuel must be heated to prevent the water in the fuel from freezing and obstructing fuel passages. Fuels are being currently tested, however, that do not require heating at −65°F. Air heaters preheat engine intake air; oil heaters maintain the desired viscosity of transmission oils; engine heaters decrease the break-

TABLE 13-1 TYPICAL CHARACTERISTICS OF FIVE COMMON BATTERY TYPES (Ref. 1)

CHARACTERISTICS	TYPE OF BATTERY				
	Lead-Acid	Nickel-Iron	Nickel-Cadmium	Silver-Zinc	Silver-Cadmium
Cell Potential, v					
Open Circuit	2.14	1.34	1.34	1.86	1.34
Discharging	2.1-1.46	1.3-0.75	1.3-0.75	1.55-1.1	1.3-0.8
Time to Discharge					
Fastest, min.	3-5	10	5	0.5	5
Average, hr	8	5	5	5	5
Slowest, days	3	3	3	90	90
Shelf Life, discharged (wet)	Not permitted	Decades	Years	Years	Years
Life in Operation					
Cycles*	10-400	100-3000	100-2000§ 25-1000#	100-300	500-3000
Float, yr†	Up to 14	8-20	8-14§ 4-8§	1-2	2-3
No. of Cells Req.**	24	40	40	38	45
Potential at Cutoff,‡ v/cell**	1.46	0.6	0.8	1	0.6
Power at Cutoff,‡ kw**	3.5	2.4	2.9	3.8	2.7
Approx. Battery Size, cu in**	6700	8500	5800	1700	2000
Approx. Battery Weight, lb**	500	450	400	125	160
Approx. Battery Cost, 1**	450	1500	2800	4900	5000

* One cycle is one complete discharge-recharge sequence.
† Float refers to emergency standby operation.
‡ Cutoff is minimum limit of useful drain on battery.
§ Pocketed plate construction.
\# Sintered plate construction.
** Based on 5-kw load powered by 50-v battery sets of 100-amp-hr rated capacity and 1/2-hr discharge life.

away torque of the engine and facilitate starting. Hydraulic systems are heated to prevent sluggish operation; pneumatic systems are heated to prevent the formation of ice; optical systems are heated to prevent fogging or frosting; and storage batteries are heated to maintain their capacity.

Once the main engine is started, heat from the engine exhaust replaces the function of some of these heaters. Before the engine is started, however, and to facilitate its starting, electric heaters are necessary—which is ironical, in view of the fact that soon after the engine begins to operate it generates excess heat that must be removed. Electronic equipment, hydraulic equipment, air compressors, and electric motors all generate heat when operating and create an appreciable cooling problem. These cooling requirements place another load upon the electrical system, but, fortunately, not an additional load, because heating and cooling are not required simultaneously by the same piece of equipment.

13-5 COMMUNICATIONS

Two types of communications equipment are found on military vehicles—intercommunication (intercom) systems and outside communication equipment. Intercom systems are usually required in combat vehicles to maintain contact between the crew members while they are manning their combat stations. In the 280mm gun carriage, an intercom system is used between the drivers at each end of the extremely long vehicle. Certain tactical vehicles, such as the armored personnel carrier, require an intercom system because of the high noise level within the vehicle. In general, intercom systems do not constitute a very heavy electrical load. Radio equipment for communication to stations outside of the vehicle, however, does constitute an appreciable load. Largely on account of this load, the 24-v system was adopted for military vehicles. This radio equipment includes radio receiving and transmitting equipment and radar and infrared surveillance equipment.

Another class of communication equipment in military vehicles is composed of the multitude of instruments and indicators. The function of most of these instrument systems is to sense various factors that affect the vehicle and its mission and to present them to the attention of the driver of the responsible crew members. Speeds, temperatures, pressures, operating characteristics of vehicle components, etc., comprise the factors that the instrument systems electrically measure and convey from the sensor to the indicating instruments. Some of these instruments are of the rheostat type, some are of the potentiometer type, and some employ combinations of synchros.

13-6 IGNITION

The ignition system consists of the ignition devices for starting and operating the main and auxiliary power plants. This system is discussed in detail in Chapter 7. Two general types of ignition systems are used in military vehicles: battery ignition and magneto ignition.

In battery ignition, the electric current for igniting the air-fuel mixture in the cylinders initially is supplied by the storage battery. When the engine comes up to speed, the generator supplies the ignition current. This system is common in small vehicles. In magneto ignition, the magneto replaces the functions of the battery, generator, ignition coil, and distributor. Some means of cranking the engine must be provided for the magneto ignition system; often, an impulse starter is chosen. Magneto ignition is common in large vehicles. Since magneto speed is not high during engine cranking, developing a hot spark may be difficult. An external source of high-tension current therefore is often incorporated into the system to facilitate starting. This external source can be a booster magneto or a high-tension coil that derives current for its primary windings from a storage battery. For increased reliability, particularly on large vehicles, dual ignition systems are common. Two complete ignition circuits, battery or magneto, are available to all cylinders, either one of which is capable of running the engine.

Because military vehicles are required to operate with their main power plant submerged, the ignition system must be completely waterproof. The distributor and ignition coil are sealed in a common housing, which must be adequately ventilated to prevent the condensation of moisture and the formation of harmful chemicals through ionization of the air within the housing caused by electrical sparking. This ventilation is normally accomplished by connecting the distributor housing to the intake manifold and, with a separate line, to the air cleaner. In this manner, clean air will be constantly drawn through the distributor by the engine vacuum.

13-7 FIRING OF ARMAMENT

An electrical circuit peculiar to military vehicles is that circuit concerned with the firing of weapons mounted upon the vehicle. Some guns are fired by electrically heating a thermal detonator; others, by a solenoid-operated percussion mechanism. The current required by these devices varies from about 4 amp, for electrically fired machine guns, to 25 amp, for some large guns. Safety relays incorporated into the firing circuits prevent accidental firing.

13-8 LIGHTING

The lighting system found on military vehicles consists of the following:

a. Headlights to illuminate the road ahead of the vehicle
b. Parking and side lights to indicate the location of the vehicle
c. Tail lights to indicate the rear of the vehicle
d. Instrument panel lights to illuminate the instruments
e. Body lights, such as dome and step lights, to light the interior of the vehicle
f. Special lights, such as spot lights, signal lights, blackout lights, and stop and backing lights
g. Wires and control switches to connect these lights and lamps to the current source

Military vehicles may or may not have numerous other lights, in addition, depending upon the type of vehicle and the equipment carried—for example, control lights, such as firing control indicators. Most special control lights utilize small-wattage lamps that draw very little current. A number of these small lamps operating at the same time, however, may draw 2 to 5 amp. For this reason, these must be considered in the design of the electrical system. All combat and tactical vehicles must be equipped, both fore and aft, with special low-intensity blackout lights to prevent observation by the enemy at night. A special switch is required in conjunction with the blackout lights to prevent accidental use of the regular service lights during blackout operations.

The lamps used in military vehicles are standard gas-filled incandescent lamps with tungsten filaments. The voltage of the lamps must correspond to the design voltage of the electrical system unless a resistor is inserted to lower the voltage to the lamps. Control lights are usually of a very low wattage because they primarily indicate a function, not illuminate an object. Lamps range in size from small 1/2-cp. instrument lamps to 50-cp. or more driving lights. These lamps are able to withstand the severe vibration and shocks to which a military vehicle is subjected. All lights on the exterior of military vehicles must be totally waterproof. Because of the low operating voltage of automotive lamps, the current requirement is high. A lamp having two filaments, one of 32 cp and the other of 21 cp, will draw 3.9 and 2.8 amp per filament, respectively. With this current requirement, lighting must be considered seriously in computing the power requirements of an automotive electrical system. The vehicle specification should also be checked as to the lighting requirements of the vehicle and the type of lamps to be supplied.

SECTION II FACTORS TO BE CONSIDERED IN DESIGNING THE ELECTRICAL SYSTEM

13-9 POWER AND VOLTAGE REQUIREMENTS

The electric power requirements vary greatly in different types of vehicles depending on their size and use. For the smallest transport vehicle, an 18- or 25-amp generator supplying 450 to 600 w. to the engine and other components may be sufficient. Larger combat vehicles equipped with many electrical devices may require as much as 10 to 15kw, however. This amount of power cannot be supplied by the electric power supply designed for the engine; therefore, an auxiliary power supply is required. The electrical devices or accessories that will be installed in or on the vehicle are listed in the vehicle specification. Table 13-2 lists average power requirements for various applications of electricity on military vehicles; the equipment manufacturer should be contacted for the specific power characteristics of his particular equipment. The power supply must accommodate the total amount of power required for all of the electrical units—although all of the electrical devices do not normally operate at the same time, it must be assumed that all of them may be required to do so.

The design of the electrical system begins with the establishment of a particular design voltage to operate the electrical devices. At the present time, a 24-v DC system is standard for all military vehicles. This voltage is prescribed by the Department of the Army in SR 705-325-1, *Research and Development of Materiel, Electrical Systems in Motor Vehicles.*

While a 24-v DC system is standard, further developments may result in changes to improve the efficiency of the electrical system. The electrical demands on military vehicles are so varied and unstable that constant research is conducted to find more economical and efficient methods for generating, distributing, and storing energy. The military facilities, aircraft companies, railroads, and independent research houses continually exchange ideas and experiences in this area for transportation vehicles of all types. One possible approach being considered is the generation of a primary voltage of alternating current at perhaps 110 v to be transformed or rectified as needed for various applications. This would be advantageous to the extent that it would be more economical in weight, size, and cost and would increase the efficiency of the system. The main disadvantage of using AC systems in automotive vehicles is the difficulty of frequency and voltage control. This difficulty arises because the main power plant of the vehicle operates at variable speeds and sometimes stands idle for long periods of time during which the need for electrical power continues.

13-12 CONNECTIONS AND CONNECTORS

An electrical connection or connector is more than a joint or a piece of hardware, especially on a military vehicle. The connection or connector represents a selected technique that must match the application, in other words, that must be capable of withstanding the rigors of military operation. The choice of any electrical connection or connector requires knowledge of operating voltage and current, necessary mechanical strength, wires and components used, and service and environment space available.

To facilitate field maintenance, separable, quick-acting connectors are used wherever practical in the installation of electrical components. In a combat zone, field maintenance personnel will not have the time or the facilities to repair a component while it is installed in a vehicle. The component must be removed and replaced. For this reason also, all circuits are numbered, with the same number for the same circuit in every type of vehicle, and all wires are identified with band markers.

Separable-type connections are subject to many possible variations in method, base material, and plating, as selected by the designer to fit the application. In general, because separable connections must withstand at least occasional insertion-removal, they constitute a compromise between electrical and mechanical performance. But, for military usage, at least the external connections of electrical components that are often replaced or repaired should be separable. Several patented separable connectors are commercially available.

Permanent-type connections that are meant to be installed and left alone are made with a high-pressure or solid metal joint that is stable both electrically and mechanically. The internal connections of a component will most often be permanent connections—formed thermally (welding, soldering), chemically (plating), or mechanically (eyelets, screws, wire nuts).

Etched or printed circuits may also be considered with respect to connections or connectors. These circuits are, however, not dependable at the present time for this application and are either throw-away items or nonfield-repairable items that eliminate the possibility of the handyman-type repairs that often keep a badly damaged vehicle moving.

13-13 WEATHERPROOFING

Environmental extremes for the design of military equipment are prescribed in AR 705-15 and are discussed in some detail in Chapter 3. This Army Regulation classifies environmental operating conditions into three categories, namely; *basic*, *extreme cold weather*, and *extreme hot weather* conditions. Generally, all military vehicles are required to function satisfactorily under the basic conditions. Equipment designed for use in the arctic and sub-arctic regions must be capable of satisfactory performance under conditions referred to as extreme cold weather, while equipment designed for use in the hot deserts of the world must meet the requirements of the extreme hot weather conditions. Temperature, humidity, and rainfall ranges are specified for each category.

High temperature and humidity conditions, as in the extreme hot weather category, impose severe hardships upon the electrical system. High temperatures increase the cooling requirements of certain electrical components, while high temperature and high humidity together stimulate fungus growth on electrical insulation. Fungus causes short circuits and deteriorates insulation. All electrical components must, therefore, be designed of materials that inhibit fungus growth or must be thoroughly treated with fungus-proof chemicals, lacquers, or varnishes.

The consideration of wind-driven rain makes imperative waterproofing of the electrical system, and fording requirements for vehicles are such that the entire vehicle will at times be completely submerged in either fresh or salt water and will be required to remain operative under these conditions. Although all components of the vehicle are not required to operate while submerged, none of them must be damaged as a result of the submersion. Critical parts must be enclosed in watertight housings to permit submersion, and provisions must be made for the removal of excess heat and the prevention of ionization of the atmosphere within the watertight housing, which may lead to a breakdown of the electrical insulation. Furthermore, the watertight enclosures should be ventilated to prevent undesirable condensation of moisture resulting from the sudden temperature change normally associated with immersion. This moisture condensation can cause short circuits, can jam contacts if the moisture freezes, and can cause equipment to deteriorate generally.

13-14 RADIO INTERFERENCE AND SUPPRESSION

13-14.1 CAUSES OF RADIO INTERFERENCE

Radio interference is electrical noise in the radio receiver that competes with incoming signals. The source of this electrical noise can be the vehicle in which the receiver is mounted or it can be a nearby vehicle. Any item of electrical equipment that produces a spark when it operates is a potential source of radio interference. This includes such items as spark plugs, circuit breakers, coils, generators, motors, voltage regulators, magnetos, and distributors. In addition, loose or dirty electrical connections may cause sparking, while the chafing of metal parts often produces static charges that interfere with radio reception.

The system of wires that interconnects the various components of the electrical system acts as an antenna to transmit radio interference. The radiating characteristic of this system causes the radiated energy to affect a wide band of frequencies on a radio receiver, with pronounced effects on certain frequencies. Not only is this undesirable from the standpoint of reception, but interference can be detected by sensitive electronic detectors and can disclose the location of the vehicle to the enemy.

13-14.1.1 Ignition Noise

The ignition circuit is designed specifically to produce surges of high-voltage current that are discharged through the spark plugs as short high-tension sparks. With each surge of current, a magnetic field is built up and collapsed with a rapidity dependent upon the speed of the engine. A capacitor placed across the breaker points increases the rate at which these surges can build up and collapse. These electromagnetic waves are picked up by the receiving set as a series of clicking sounds that vary in speed in intensity with the speed of the engine.

13-14.1.2 Generator Noise

Generaters while in operation exhibit some sparking between the brushes and the commutator segments. The sparking produced by a generator that is in good mechanical condition is of no consequence, as it does not cause radio interference. But, if this sparking is intensified because of mechanical defects in the generator, it may cause radio interference. The most common mechanical defects that cause excessive sparking are:

a. The contour of the contact surfaces of the brushes does not match the contour of the commutator.

b. The brushes are worn more than one-half of their original length.

c. The brush spring tension is incorrect.

d. The brushes jump because the commutator is worn out of round.

e. The generator is loaded in excess of rated capacity.

f. The commutator segments are burned or grooved and do not make good contact with the brushes.

g. The brushes jump because the insulation protrudes between the segments of the commutator.

h. Oil or carbon particles are accumulated around the commutator.

Generator noises in the radio equipment can be recognized by a roaring or whining that varies in pitch with the speed of the engine.

13-14.1.3 Body Noise

The chafing of various parts of the vehicle when it is moving or for a short time after it is stopped causes static charges of electricity to be induced and collected in the vehicle body. These charges are retained by the poorly grounded sections of the body until they build up sufficient potential to jump to a well-grounded section. Such static discharges constitute body noises, which appear as frying or snapping noises in the radio equipment while the vehicle is in motion. Body noise can sometimes be detected by moving loose parts and listening for scratching sounds in the receiver. Tightening the various bolts and nuts will eliminate some body noise.

13-14.2 SUPPRESSION OF RADIO INTERFERENCE

The suppression of radio interference is accomplished by the installation of special devices, such as resistor-suppressors, capacitors, and filters, and by the application of bonding and shielding techniques.

Resistor-suppressors reduce the intensity of electrical surges in high-tension components of the electrical system, such as the ignition circuit, and thereby reduce the interference from these sources.

Capacitors reduce the electrical surges caused by the sparking of generator brushes, voltage regulators, and gage contacts. These surges while not as intense as those of the ignition circuit are large enough to cause interference in a radio set. Resistor-suppressors cannot be inserted in these low-voltage circuits because their high resistance would affect circuit operation. Capacitors offer little resistance and are used successfully. One side of the capacitor is connected to the circuit, as close as possible to the source of the surges, and the other side is connected to ground. Thus, the capacitor grounds the interfering electrical surges without draining useful current from the system.

Filters are assemblies of low-resistance coils and capacitors. Their method of operation represents a combination of those of the resistor-suppressor and the capacitor. Filters are often used in generator circuits, voltage regulator circuits, and in low-voltage ignition circuits.

Bonding is the term applied to the technique of electrically connecting all metal components of a vehicle to each other and to the frame or hull of the vehicle, to provide an easy path to ground for static charges. Bonding is accomplished by the use of toothed lockwashers under all mounting screws and by the use of bonding straps between metal components. Maximum effort is made to achieve a low-resistance path to ground for the interfering electrical currents.

Shielding, the term applied to another method of suppressing electrical interference, employs a grounded metal shield to cover all wires that carry interfering voltages or electrical surges. Woven metal conduit is used where flexibility is required, while solid metal conduit is used elsewhere. Components that cause interference, such as spark plugs, ignition coils, distributors, and regulators, are enclosed in grounded metal boxes. The purpose of the shielding is not to reduce the intensity of the interfering suges, but to prevent their radiation. Filters and capacitors are still necessary in a fully shielded circuit to prevent the surges from traveling on the wires and effecting the radio through the power supply. And, these filters and capacitors must also be enclosed in grounded metal shielding boxes to prevent radiation.

13-15 STANDARD PARTS

All components of the electrical system must be capable of withstanding the vibrations and shocks encountered in a military vehicle. Unlike civilian vehicles, which usually traverse paved or smooth surfaces, the military vehicle is called upon to travel cross country. It is also expected to survive all weather conditions and be able to travel submerged as specified.

Standard, interchangeable parts shall be used wherever possible, to minimize the number of repair parts. This is important in a combat area where storage facilities are limited and procurement is difficult. A compilation of components and assemblies standardized by the Army Tank-Automotive Center is given in the ***Ordnance Corps*** *Tank-Automotive Components Directory.* Copies of specific sections can be obtained by applying to the Standards Section, Standardization Branch, Engineering Division, U.S. Army Tank-Automotive Center, Warren, Michigan.

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CHAPTER 14

MISCELLANEOUS EQUIPMENT*

SECTION I GENERAL DISCUSSION

The materiel discussed in the present chapter may be classified as: (1) standard equipment, or (2) supplementary equipment. Some of the individual items may be classified as standard equipment for one type of vehicle and as supplementary equipment for another type of vehicle. For example, a personnel compartment ventilating and heating system may be a standard installation in a tank, while, if applied to a conventional wheeled vehicle, it would be a special or supplementary system.

Various kits, such as winterization or fording kits, also fall into the miscellaneous equipment category. Although the various kits are not installed on a vehicle unless they are actually required, the vehicle designer must incorporate in his design the proper facilities to permit field installation of the equipment if the need arises.

The major groups of equipment under the miscellaneous classification are:

1. Ventilating, heating, and cooling systems
2. Winterization kits
3. Fording kits and floatation devices
4. Fire-fighting systems
5. Special equipment
6. Communication systems.

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SECTION II TYPES OF MISCELLANEOUS EQUIPMENT, THEIR CHARACTERISTICS AND REQUIREMENTS

14-1 VENTILATING, HEATING AND COOLING SYSTEMS

Enclosed military vehicles, such as tanks and armored personnel carriers, usually have heating and ventilating systems installed as standard equipment. These systems have two basic functions: (1) to control the temperature of the interior of the vehicle, and (2) to supply fresh air while expelling contaminated air. The contamination results from normal breathing of the crew, the toxic and irritating fumes given off by the weapons, and possibly fumes from the engines. The ventilating system must also be capable of reducing the dustiness of the personnel compartment air to an acceptable level when the vehicle is operated in dust-producing environments.

Combat experience has demonstrated the importance of adequate crew compartment ventilation. Failure of the ventilating system to maintain proper atmospheric conditions in the fighting compartment has resulted in the withdrawal of tanks from combat.

Crew compartment ventilation systems must not depend entirely on the main power plant, and must be capable of prolonged periods of operation when the vehicle is stationary with the main power plant not running.

14-1.1 CONTROL OF FUMES GENERATED DURING FIRING

An important aspect of the problem of ventilation of the fighting compartments of closed vehicles is the control of fumes generated during firing, since these fumes contain gases such as carbon monoxide and ammonia. The problem is very much simplified if these fumes are prevented, to the extent possible, from entering the fighting compartment. One way of accomplishing this is by incorporating a bore evacuator on the gun. Present bore evacuators are designed to use the pressure of the propellant gases to exhaust fumes through the muzzle.

Both *exhaust ventilation* (negative pressure) and *forced ventilation* (positive pressure) systems have been applied to tanks to remove the fumes. The exhaust ventilation system proved inadequate as the rate of fire increased. In general, the space limitations make it impractical to install exhaust fans of sufficient capacity in most vehicles.

A forced ventilation system of sufficient capacity can be used to remove the fumes and to meet the other ventilating needs. The main disadvantage is that it fails to provide positive fume control when the vehicle hatches are open. The problem of fume control is under constant study to find methods which are practicable and economical.

14-1.2 CONVENTIONAL SYSTEMS FOR THE CONTROL OF TEMPERATURE AND VENTILATION

The problem of maintaining a tolerable level of temperature and humidity within the crew compartment of fully enclosed vehicles is very important. Military vehicles and their crews are expected to operate satisfactorily in climates characterized by extremely high or low temperatures and relative humidities, with the possibility of high concentrations of dust or snow particles and strong winds (see Chapter 3). To maintain a reasonable level of human efficiency, it is necessary to provide ventilating, heating and cooling systems for vehicles of the type under consideration.

The current "Heating and Ventilating System for Combat Vehicles," as used in the latest tanks, has the following specifications and operating characteristics (Ref. 2). The system consists of a 240-cfm standard military heater, and a 600-cfm blower ducted to receive ambient air through air maze type filter inlets to the heater and blower. The air is distributed through ducts to the driver's cockpit and to the gun turret compartment at points located so as to expel the contaminated air efficiently through a mushroom-type outlet in the turret. The heater effects a complete change of air within the crew's quarters at 1-1/2-min intervals, and when the large blower is used in conjunction with the 240-cfm blower during the firing of the weapons, a complete change of air is effected at 25-sec intervals.

The 240-cfm heater can be manually controlled to supply heated air or ambient temperature air and is automatically activated during the firing of the guns by direct electrical connections to the various trigger switches. The 600-cfm blower operates in a similar manner. A time-delay relay switch, known as an "Agastat" (solenoid-activated pneumatically-timed switch) is included in the system. This switch causes the blowers to operate for a predetermined period of time after the firing ceases.

The ducts leading to and from the heater and blower are constructed of 14-gage aluminum and are insulated with 1/8-in. glass fiber cloth, MIL-114-OB. A removable filter unit is provided for the ambient air intake, and self-closing one-way gates are provided, as required, to control the air flow. A manually operated shutter is provided inside of the mushroom exhaust to permit the regulation of air pressure within the crew compartment.

Formerly, ambient air for the main power plant was induced through the crew compartment at a rate of 800 to 900 cfm. This arrangement made uniform heating within the tank very difficult, and was abandoned in favor of an ambient air inlet directly to the engine compartment.

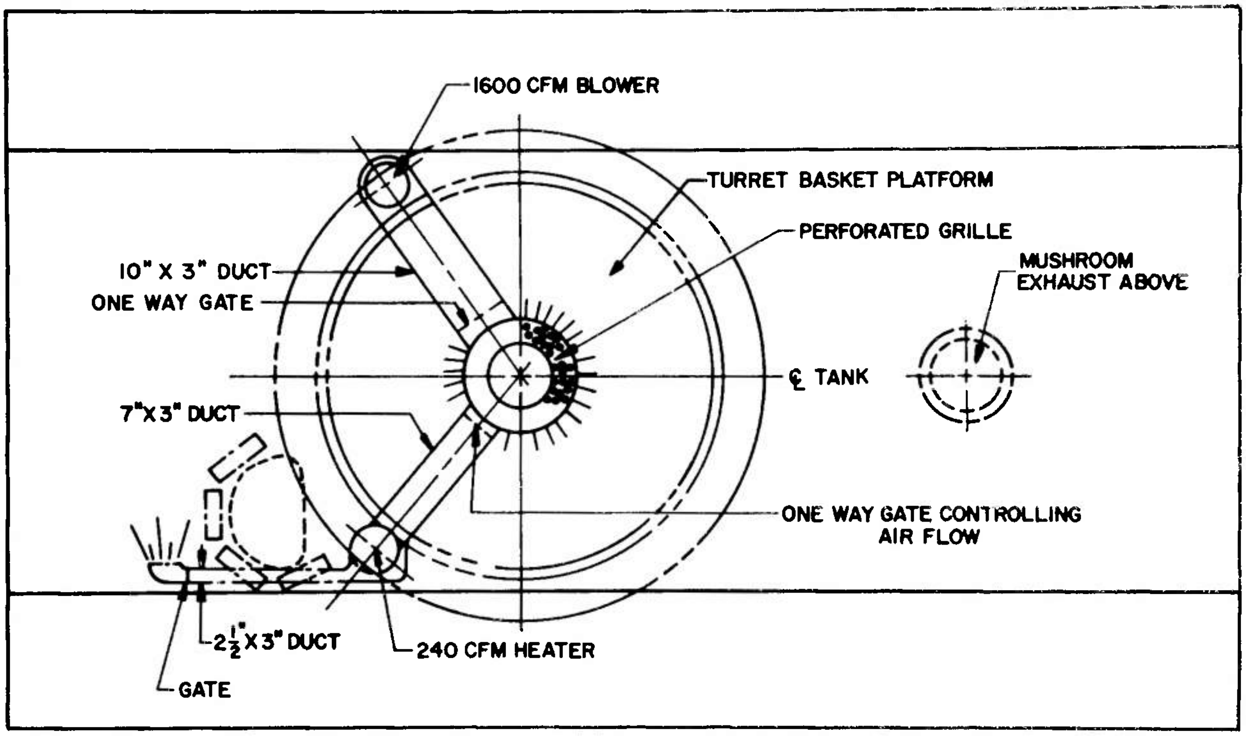
Special filters are under development for the purpose of permitting vehicles to operate in areas contaminated through chemical, bateriological or radiological actions.

Figures 14-1 through 14-3 show a typical tank heating and ventilation system (Ref. 2). Standard military heater assemblies include gasoline units and hot water units. These are installed as required in military vehicles.

14-1.3 AIR CONDITIONING SYSTEMS (Ref. 3)

Air conditioning, in the present context, may be defined as the simultaneous control of the temperature, humidity, motion, and distribution of the atmosphere within a closed passenger or crew compartment. The system described in the previous section is not capable of the cooling and humidity control functions of a true air conditioning system.

The heating and ventilating systems currently used in enclosed military vehicles have been great-



NOTES:
1.- DUCTS MADE OF 14 GAGE ALUMINUM.
2.- DUCTS INSULATED WITH GLASS FIBER CLOTH PER MIL-1140-B.
3.- AIR CLEANING FILTERS INSTALLED IN AIR INTAKE OPENING.
4.- AIR SHUTTER PROVIDED UNDER MUSHROOM EXHAUST.
5.- PROVIDE 1" ARMOR COVER AT AIR INTAKES

Figure 14-1. *Tank Heating and Ventilating System (Plan View)*

ly improved in recent years, yet they are not capable of providing a satisfactory crew compartment atmosphere under conditions of extreme ambient temperature. This is especially true when the ambient temperature exceeds 95°F. It has been demonstrated that, at ambient temperatures of 90°F or lower, with the sun shining from a clear sky, the wall temperature of a tank could exceed 140°F, and the air temperature within the closed unventilated tank could exceed 130°F (Ref. 3).

Various investigators have shown that there is a substantial decrease in both mental and physical efficiency of personnel when ambient temperatures exceed approximately 90°F. Furthermore, air velocities were found to be comfortable up to 100 fpm when the temperature did not exceed 95°F Above this temperature, air currents resulted in increased convective heat gain by the body, rather than any feeling of comforting coolness. Thus, at ambient temperatures above 95°F, ventilation of a closed vehicle with air that has not been cooled artificially could actually be a detriment rather than an advantage, except when the ventilator is used to remove fumes generated by firing. However, if the temperature within the compartment exceeds uncooled ventilated air, the resulting lowering of the temperature will be beneficial to the crew. Under these conditions, a copious supply of atmospheric air, distributed by a carefully designed duct system, would be required to maintain a reasonable level of crew efficiency.

The feasibility of using air conditioning systems in combat tanks was thoroughly studied and the results and recommendations were presented in Ref. 3.

In view of the limited amount of space available in modern combat vehicles, it is interesting and informative to consider the refrigeration requirements for a typical vehicle. Using the M47 medium tank as a basis for calculations, the following results were obtained. First, it was assumed that cooling would be attempted in the gross sense,

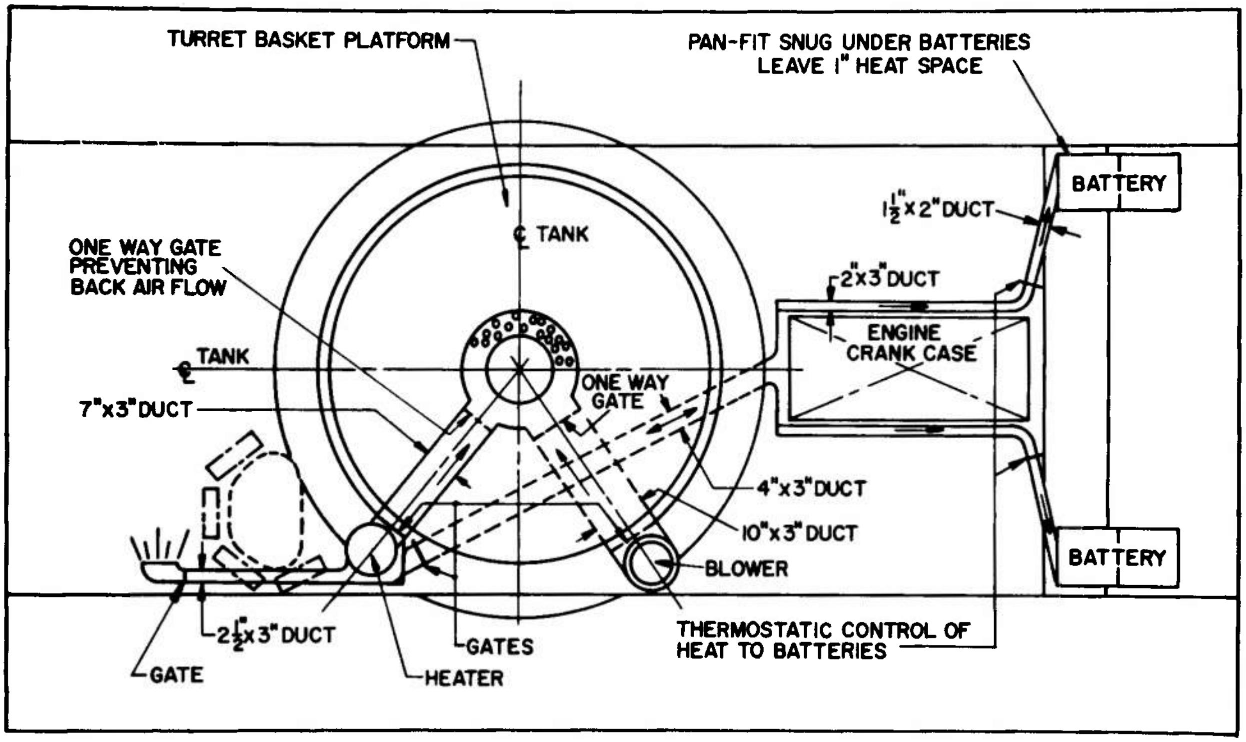


Figure 14-2. *Tank Heating and Ventilating System (Elevation View)*

NOTES:

1. ALL DUCTS TO BE MADE OF #14 GA. ALUMINUM.
2. ALL DUCTS TO BE INSULATED WITH 1/16" ASBESTOS CLOTH PER MIL-C-10316 AND GLASS FIBER CLOTH PER MIL-1140-B.
3. INSTALL AIR-CLEANING FILTER IN AIR INTAKE OPENING.
4. PROVIDE 1" BALLIST.C COVER AT AIR INTAKE.

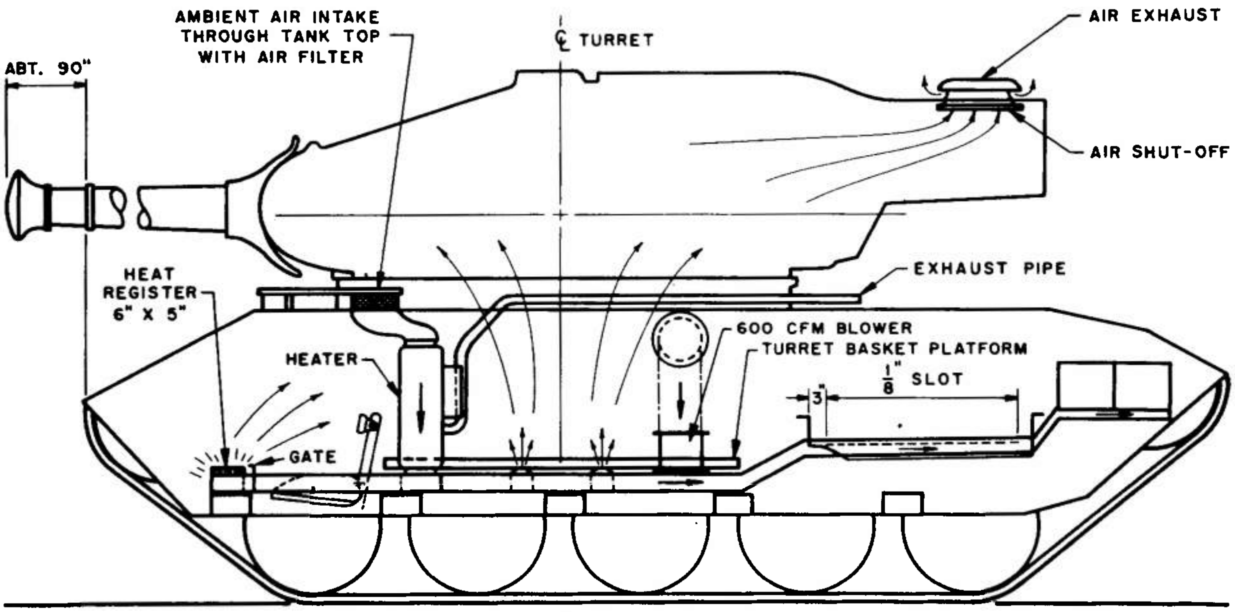


Figure 14-3. *Heating System for Crankcase and Batteries*

i.e., air, in sufficient quantities, would be cooled to a low-enough level and distributed throughout the compartment so that the air in the compartment as a whole would remain at a specified temperature. As a result of the heat-transfer studies, it has been determined that, in order to maintain a compartment temperature of 80°F when the ambient temperature is at 120°F, approximately 6-1/2 tons of refrigeration would be required (based on 300 cfm of air at the refrigerator exit). If 1/4 in. of insulation, having a thermal conductivity of 0.04 btu/(ft^2-hr°F/ft), is used to cover the compartment walls, the refrigeration requirement is reduced to 3-1/2 tons. The importance of insulation under the stated conditions is obvious.

If the same volume of air (300 cfm at 80°F) is considered, with reference to Fig. 14-4, the theoretical refrigeration required to lower the temperature from 120° to 80°F is about 1-1/10 tons. It can be concluded that since space is at a premium and since the comfort of each crew member is of primary interest, it may be most economical and practicable to bring the cooled air as directly as possible to each individual's position by means of a suitably insulated duct system. Insulation of the tank walls would still be desirable since the duct heat losses will depend on the overall compartment temperature.

Closely related to the cooling requirements are the heating requirements. Figure 14-4 can be used to determine the theoretical amount of heat that must be added to the air passing through the heater to raise its temperature the desired amount. For example, if 300 cfm of air is to be heated from −30° to +25°F at the heater exit, approximately 24,000 btu/hr must be added to the air.

These values expressing theoretical cooling and heating requirements, for the stated conditions, indicate that localized heating and cooling of the vehicle via ducting is most practicable.

14-1.4 CONCLUSIONS AND RECOMMENDATIONS

On the basis of research studies (Ref. 3), a number of conclusions and recommendations for incorporating satisfactory air-tempering systems in present and future combat vehicles have been presented. At the expense of some repetition, these can be stated as follows:

1. Among the most critical features that will affect the required capacity of the heating-cooling system are the duct system design and the proper use of insulation on both the duct walls and the main compartment walls. It is desirable to have adjustable (direction and volume) discharge points in the immediate vicinity of the crew members' stations. It would be advantageous to have small auxiliary vents available for playing heated or cooled air over important control levers. The application of insulation to the walls of the vehicle presents a difficult problem. Particular attention should be given to sprayable types as a possible satisfactory method of insulating.

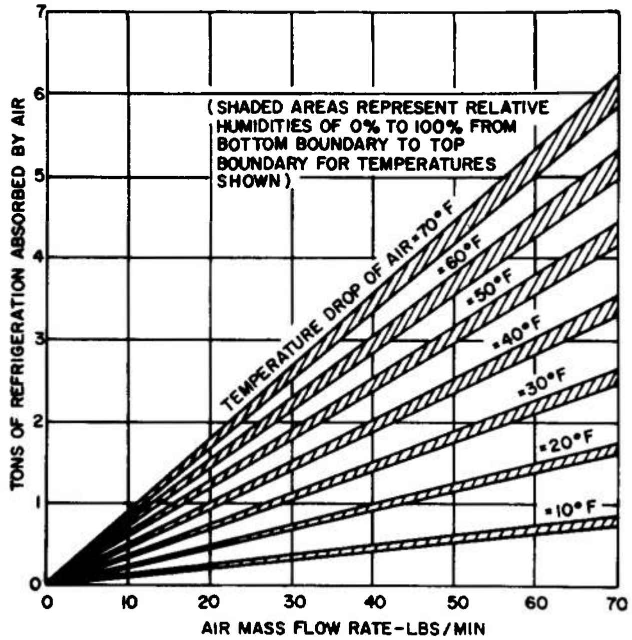


Figure 14-4. Refrigeration Absorption by Air as a Function of Mass Flow Rate, Temperature Drop, and Relative Humidity (Ref. 3)

2. The most feasible system for meeting the cooling requirements appears to be the vapor compression system using a positive displacement compressor, since the required capacities are far too small to use dynamic compression efficiently.
3. The method of combustion heating seems to be the most practical means of heating both the crew and engine compartments when the main engine is not running. An effective method of utilizing waste heat when the main power plant is running would result in increased fuel economy for the vehicle.
4. The refrigerator and heater should normally

use recirculated air, but provisions should be made for drawing in ambient air as required.

5. For the purpose of dispelling fumes, a high-capacity exhaust fan should be available. The flow of air through this fan should be from the weapon breech directly to an exterior discharge vent.
6. All controls for the air-tempering system, with the possible exception of the fume fan, should be manually operated.
7. Finally, the prolonged operation of the ventilating system equipment dictates components of the highest quality to minimize maintenance and failures.

14-2 FORDING KITS

The ability to operate in reasonable depths of water greatly enhances the mobility of tactical equipment; hence, all tactical vehicles must meet established fording requirements. Current requirements make a distinction between shallow and deep fording. The first is applied to standard tactical vehicles operating without the addition of special kits (although they may have factory-installed items, such as intake and exhaust extensions and waterproof ignition systems). The basic vehicle must be capable of fording a specified depth of water without any special preparation.

Deep-water fording, on the other hand, implies the usage of special equipment, usually installed in the field by the vehicle's crew prior to the fording operation. The deep-water fording kit may interfere, to some extent, with the normal functioning of the vehicle on land, but is easily and quickly removable immediately after use.

Important considerations in the design of fording kits are ease of installation, jettison ability, and a high degree of reliability.

Salt-water fording operations offer additional problems owing to the corrosive effect of the salt water. A detailed discussion of these problems is presented in Ref. 5.

14-2.1 SHALLOW-WATER FORDING

Combat vehicles are required to ford 42 in. of water at 3 mph (SR 705-125-10). All other tactical vehicles are required to ford 30 in. of water, except 1/4-ton types, which must be capable of fording 20-in. depths.

The preparation of tanks for shallow-water fording is relatively simple. The drain valves are closed and the escape hatch is secured. Openings subject to splash only are adequately protected by the sealing normally provided to exclude water during rainy weather operation. Drain valves can be opened from the inside of the tank after fording to drain any accumulated water.

14-2.2 DEEP-WATER FORDING

To meet the requirements of SR 705-125-10, all enclosed armored vehicles must be capable of fording water to the top of the turret after the proper kit is installed. All other vehicles (except trailed loads, which must operate submerged) are required to operate in five feet of water with the proper kits.

14-2.3 REQUIREMENTS OF DEEP-WATER FORDING KITS

The component parts of a deep-water fording kit must be simple and inexpensive. Immediately upon completion of fording, the vehicle must regain its original fire power and mobility. This post-fording requirement dictates that all parts of the kit that interfere with fire power or mobility be jettisonable. Because these parts will probably not be salvaged for reuse, economy is an important consideration. A kit should be simple to install because tactical situations often limit the time available for preparation. If elaborate and time-consuming preparations are necessary to install a kit, the success of the operation may be endangered.

Although the kits for various types of vehicles will differ, the basic design and preparation factors pertaining to them will be similar. The following list covers some of the most important factors related to deep-water fording.

1. Cooling fans must automatically disengage when the fan blades are submerged.
2. Water must not be allowed to enter any of the various transmissions, differentials, gear boxes or final drive assemblies, which are normally vented to the atmosphere. These vents must not be sealed prior to fording. If they are sealed, the sudden cooling of the unit upon submerging creates a temporary partial vacuum within the housing. The resultant pressure difference could cause serious water leakage into the housing (through the shaft seals). Therefore, some provision must be made to vent the various housings to the atmosphere. Usually, the simplest way to accomplish the

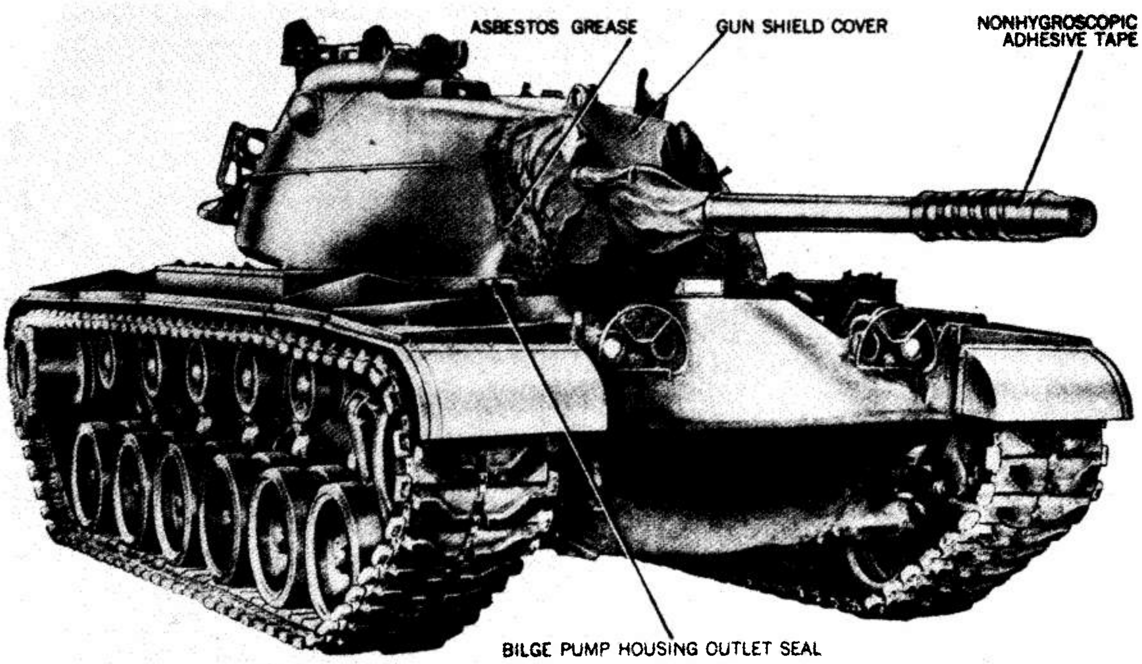


Figure 14-5. ***Right Front View of 90mm Gun Tank, T48, Prepared for Deep-Water Fording***

venting is to utilize tubing to extend the vents to the crew compartment or above the highest water level.

3. One or more exhaust stacks must be provided to allow engine exhaust gases to escape above the water level. The above-the-water discharge is necessary to prevent water from entering the engine in the event of a stall while submerged. For the same reason, the exhaust stacks must not leak with the engine running or stalled.
4. The fording kit must provide sealing and venting of the fuel tank or tanks.
5. A seal or cutoff for the hot-spot manifold is normally provided.
6. The main engine air intake must be above the water level or in the crew compartment, and must be adequately sealed. (Auxiliary engines must also have special intake and exhaust extensions.)
7. A bilge pump is supplied with standard deep-water fording kits for enclosed vehicles. Provisions for mounting the pump and discharging the water must be made on the various vehicles.
8. Tank turrets present a difficult but important sealing problem. A permanent-type hull-to-turret seal, to be installed at the time of manufacture, would be of value in excluding foreign material from the turret race during normal operation, as well as water during fording. In addition, the permanent-type seal would save time during the preparation of a tank for deep-water fording.
9. Openings in and around the main gunshield of a tank present a particularly difficult sealing problem. Since the gun must be capable of firing in all directions as soon as the tank comes out of the water, the gunshield sealing must be flexible enough to permit elevation and depression of the primary armament. At present, the most effective gunshield seal consists of a waterproof canvas cover. The edge of the cover is clamped tightly to the front face of the turret around the gunshield. The short tubular section of the cover is long enough to permit recoil of the gun. Individual pockets are provided in the cover to accommodate the coaxial machine gun muzzle and the sighting telescope. The waterproof canvas gunshield is not entirely satisfactory. Among the more important drawbacks are vulnerability

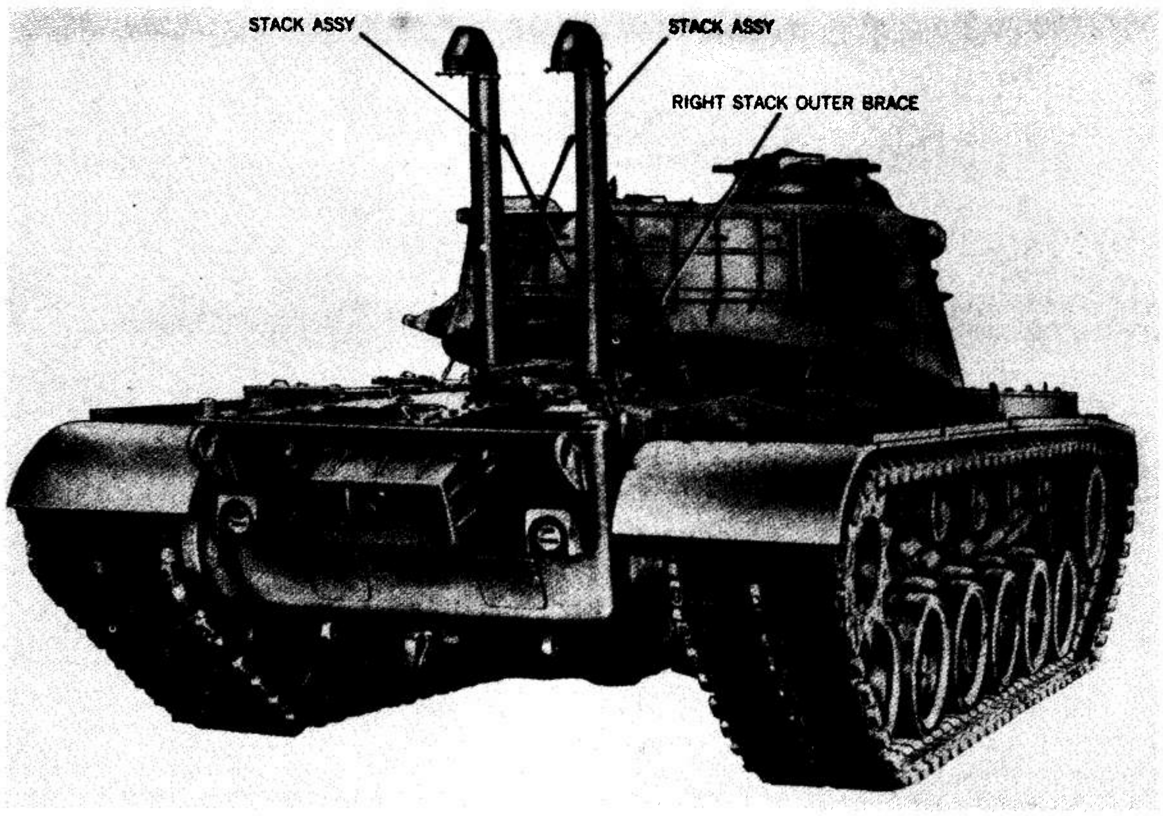


Figure 14-6. *Right Rear View of 90mm Gun Tank, T48, Prepared for Deep-Water Fording*

to ballistic attack and loss of sealing once the coaxial machine gun is fired. Since the covers have a relatively short life in the field, each fording kit includes a complete cover assembly. Development of permanent, built-in sealing would be a major improvement.

10. Miscellaneous openings are sealed with either asbestos grease or nonhygroscopic adhesive tape.

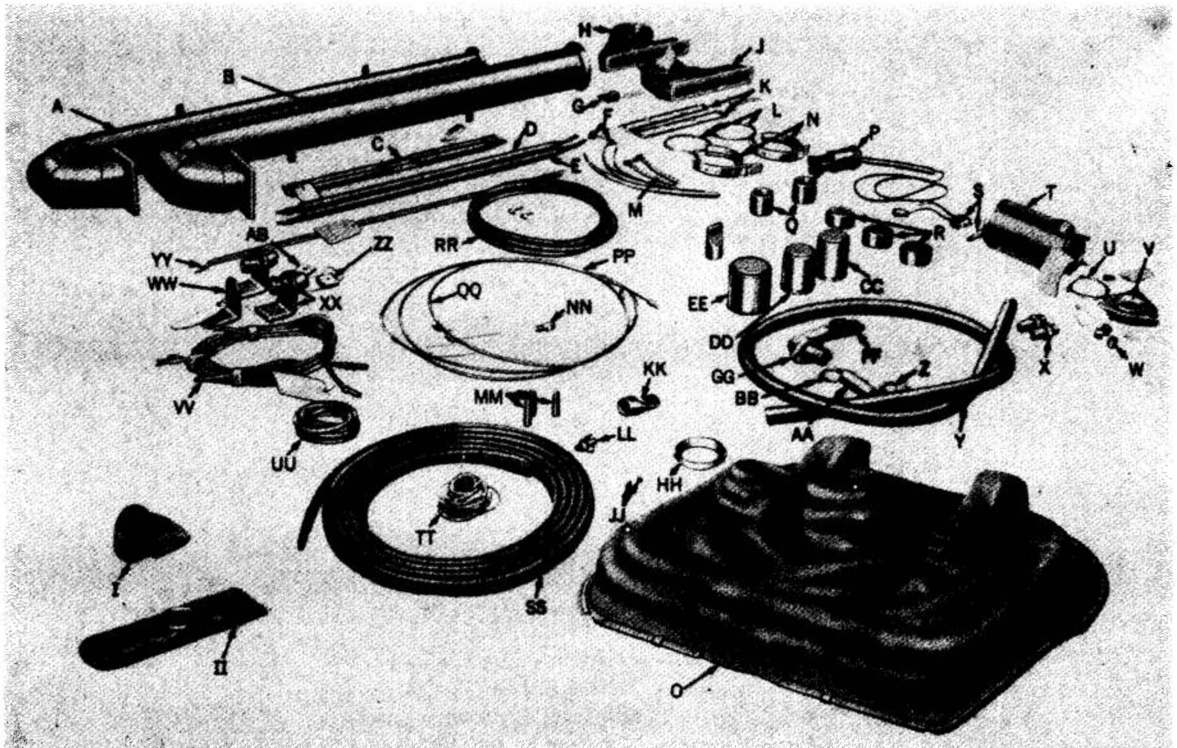
Figures 14-5 through 14-8 show deep-water fording equipment as applied to tanks.

14-3 FLOATATION DEVICES

During landing operations, troops and their equipment are particularly vulnerable to enemy fire until they have arrived on the beach and set up their equipment. To supplement naval and air support, the firing power of tanks and other armored vehicles was deemed desirable. The firing of the weapons of these vehicles could only be accomplished if the gun muzzle were above water. Floatation devices plus deep-water fording kits were necessary to float a tank high enough in the water to permit firing of the vehicle armament at all times. These two devices are also used to enable vehicles to negotiate deep rivers and lakes when better facilities are not available. A typical tank floatation device is shown in Fig. 14-9.

Experience gained from amphibious operations and numerous tests have established the following military requirements for successful floatation devices as applied to tanks:

1. Weight to be such that component parts can be handled by a double tank crew aided by the equipment normally assigned to the second echelon maintenance shops of tank batallions or similar installations. A minimum amount of special tools shall be required.
2. Length such that the tank can negotiate a 40% (22°) slope either below or above the surface of the water, with the floatation device attached and operable.



A	Right stack assembly
AA	Nipple
AB	Right stack outer brace anchor
B	Left stack assembly
BB	Bilge pump housing outlet jacket
C	Stack cross brace
CC	Water pump grease
D	Left stack outer brace
DD	Asbestos grease
E	Right stack outer brace
EE	Nonhygroscopic adhesive tape
F	Clamp
FF	Bilge pump housing outlet seal
G	Exhaust duct support
GG	Bilge pump outlet housing
H	Left duct assembly
HH	Auxiliary engine bellows gasket
I	Collar cap assembly
II	Machine gun cover assembly
J	Right duct assembly
JJ	Oil fan drive brush box plug
K	Gasket
KK	Auxiliary engine exhaust outlet seal
L	Exhaust stack gasket
LL	Hot spot control shut-off valve
M	Clamp
MM	Turret race drain plug
N	Clamp
NN	Final drive vent line tee assembly
O	Gun shield cover
P	Bilge pump harness
PP	Final drive breather tubing
Q	Fuel tank cap seal
R	Fire extinguisher horn cap
RR	Fuel tank vent hose
S	Switch assembly
SS	Turret-to-hull seal
T	Bilge pump and motor assembly
TT	Elastic shock absorber cord
U	Pump and base gasket
UU	Rangefinder blister cord
V	Bilge pump base
VV	Stack release cord assembly
WW	Fording cord guide
X	Bilge pump hose nipple
XX	Cord guide assembly
Y	Bilge pump hose
YY	Fuel tank vent tube
Z	Elbow
ZZ	Left stack outer brace anchor

Figure 14-7. Component Parts of a Typical Deep-Water Fording Kit for Tanks

3. Overall width when attached to the tank to be the minimum practicable.
4. Height such that all tank armament can be fired through a field of fire of 360° when the floatation device only is attached. The stacks of the fording kit will limit this field of fire.
5. When disassembled the floatation device must be capable of being transported in a standard wheeled or tracked cargo carrier or by air.
6. The floatation device is to consist of a metal frame (steel, titanium, or aluminum) assembly in which an approved plastic material, or an approved substantially equal material can be used to supply the required buoyancy to float the vehicle.
7. The plastic material is to have a density of three pounds per cubic foot and a minimum compressibility of 30 pounds per square inch. The plastic material must be resilient and shock-absorbing, waterproof, fire-resistant, chemical-resistant, vermin and fungus proof, and have a maximum heat shrinkage of one percent.

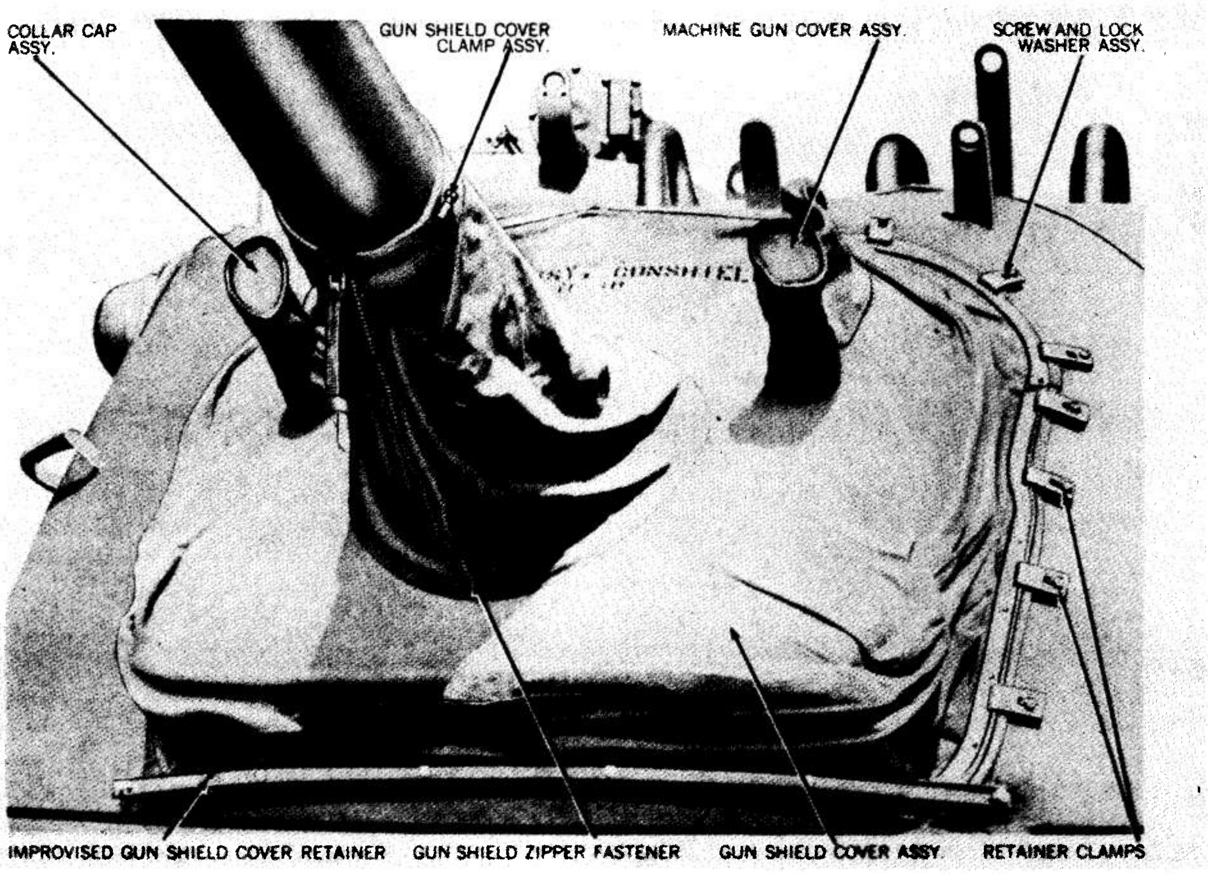


Figure 14-8. ***Gunshield Cover Assembly***

8. The metal frame structure supporting the floatation units to be made detachable from the tank by use of explosive pins, controlled and discharged by the vehicle operator.
9. Floatation blocks to be securely locked into four units by the metal structure, one each for the front, rear, and sides of the vehicle. Each unit to be discharged at will by the tank operator according to a predetermined sequence.
10. Floatation units must provide ground clearance, approximately 2 in. above that of the bottom of the hull of the tank, exclusive of bumpers.
11. Provisions shall be made for protection of the bottom and sides of floatation units to prevent or lessen damage to floatation blocks from obstructions.
12. Angle of approach or departure on front and rear floats to be not less than 20°.
13. Greatest degree of interchangeability of various parts of floatation units must be provided and maintained using standard and uniform size float blocks and metal structure.
14. Propulsion in the water to be by propellers on both sides of the tank, powered by power take-offs mounted on the drive sprockets with means of disconnecting the propeller drive during operation of the tank on land.
15. The floatation device must maintain floatation and proper trim of the tank in streams with current speed of 11 fps (7.5 mph).
16. The speed of the floatation device to be not less than 6 mph in smooth still water.
17. The floatation device must be sufficiently durable to withstand normal handling and cross country travel while attached to the tank or in the course of being transported by train, truck, or otherwise.
18. The floatation device must be designed to maintain floatation with the tank hull completely filled with water.
19. The floatation device must be capable of maintaining floatation of the tank after multiple punctures by caliber .50 and .30 bullets.

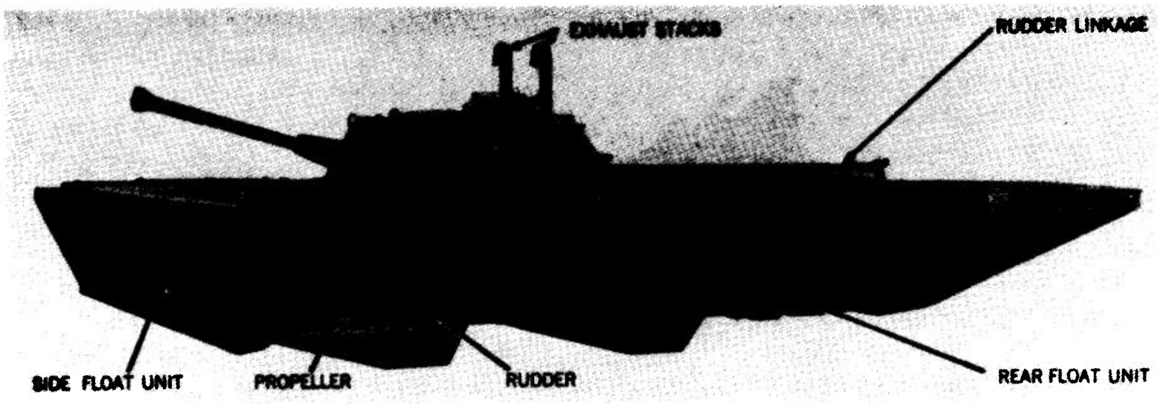


Figure 14-9. ***Floatation Device for Tracked Vehicle***

20. Jettisoning must be accomplished from inside the vehicle.
21. Steering must be accomplished from inside the tank.

Although the preceding requirements have been established for tanks, many of them would apply to floatation kits for wheeled vehicles.

14-4 PROTECTION AGAINST RADIATION

With development of thermonuclear weapons, an enclosed vehicle may be called upon to operate within a radioactive area. This type of operation is not possible at the present time, because no effective shielding against atomic radiation is in existence. Shieldings have been developed but they are bulky and do not lend themselves to vehicle shielding. Research for the development of economical and practical shielding is progressing constantly.

14-5 BULLDOZING KITS

In combat zones, it may be necessary to clear or grade a piece of land for a road or airstrip in a minimum amount of time. If there are not enough road graders or bulldozers in the area, a bulldozing attachment can be installed on tanks or large trucks. These attachments are not capable of lifting material into trucks as is a bulldozer, however, they are capable of clearing and leveling the terrain.

The vehicle designer will have to consider certain factors in designing a bulldozing attachment.

1. The weight of the bulldozing blade and attachments must be kept to a minimum. When the attachment is in place, the suspension loading of the vehicle is changed and the original vehicle balance may be upset. For this reason, the attachment does not lend itself readily to light vehicles. Aluminum blades have been tested, but the results of the tests are inconclusive as to the suitability of these lightweight units.
2. The blade of this attachment is usually hydraulically operated and provisions must be made for supplying hydraulic pressure to the cylinder. The hydraulic components must be carefully designed to avoid leakage and to minimize vulnerability.
3. An emergency blade-lifting device must be incorporated into the design so the blade can be raised if the hydraulic system fails.
4. The bulldozing attachments are normally designed to mount on the towing lugs of the vehicle. This arrangement is not entirely satisfactory; special mounting lugs would improve the overall operation of using the bulldozing equipment.
5. The bulldozing blade should not obstruct the visibility of the vehicle operator when it is in the raised position.

14-6 WINTERIZATION KITS (Refs. 1, 4)

Experience in World War II emphasized the need for vehicles capable of sustained fightability within any geographical area during any season of the year. In recognition of this need, AR 705-15 states that automotive material developed by the Army should be capable of acceptable performance throughout the ambient temperature range of 115°

to $-25°F$ with no aids or assistance other than standard accessories, and to $-65°F$ with employment of specialized aids in kit form.

Winterization kits are those appliances that are necessary to assure dependable vehicle starting and operation in the temperature range of $-25°$ to $-65°F$. The basic equipment and materials for extremely cold weather operation of vehicles are arctic-type fuels, lubricants and engine primers; high-capacity heating equipment for power plants and batteries, and personnel heaters.

Vehicles destined for operation in arctic environments must be prepared with specified fuels and lubricants for arctic operation (see Chapter 3). For vehicles so prepared, the major starting aids are the heaters used to preheat engines, batteries, and elements of the power train to facilitate starting; and, in the case of batteries, to keep the batteries at the proper temperature for continuous charging with the standard electrical generating system.

14-6.1 CLASSIFICATION OF HEATING METHODS

The techniques of applying heat to military vehicles to ensure starting in cold environments are the results of extensive testing and developmental efforts. Two methods have been developed: the standby-heat method and the quick-heat method.

The standby-heat method uses a comparatively small heater which operates continuously when the vehicle is idle. It must produce sufficient heat to compensate for losses and keep the power plant at a temperature high enough to ensure starting. For vehicles having engine displacements of 100 to 300 cu. in., 20,000 btu/hr, properly distributed, will maintain satisfactory temperatures at all desired points. When standby heat is used, the vehicle is always warm and ready to start. Heat is usually supplied to liquid-cooled plants by a thermosyphon system, thus avoiding pumps and fans that drain batteries. Heat can be supplied to batteries by hot water coils and, thereby, minimize the danger of overheating. Since space is usually at a premium, the relatively small size of standby heaters is a distinct advantage.

The quick-heat method, which is well adapted to the present air-cooled engines, provides a heater having sufficient capacity to start a cold engine in a short period of time. For current engines, starts in less than an hour require heaters producing from 30,000 to 100,000 btu/hr. Several design problems are presented by the quick-heat method. Among these are the prevention of damage to electrical equipment and the avoidance of heating the battery too rapidly. Conventional rubber-cased batteries cannot be heated faster than about $1°F$ per minute; supplying heat at a faster rate may damage the battery.

Quick heating eliminates the need for continuously heating equipment not in service. The life of the quick heater is greater and maintenance is less than in types requiring constant operation.

Both standby and quick heaters have advantages, and both are currently in use. There is a trend toward a combination of the two. This combination heater should be capable of bringing a thoroughly soaked power plant from $-65°F$ to a starting temperature in 45 to 60 minutes. The heater should be thermostatically controlled so that it can be used as a standby heater or a quick heater as desired.

14-6.2 COLD-STARTING KIT (SLAVE KIT)

The cold-starting kit (slave kit) M40 provides an auxiliary source of electrical energy and heat to aid in starting the engine and warming vital parts of the vehicle and vehicle batteries. It is provided with 6-, 12-, and 24-volt battery-boosting circuits which can be connected to the electrical systems of the vehicle or other pieces of equipment to faciliate starting. It also includes a gasoline-engine-driven generator for battery charging and a gasoline burner for supplying a large volume of heated air for use as a quick starting aid.

The high capacity of the heater and the auxiliary battery of the slave kit makes it possible to put even large tanks into operation in a very short time. The slave kit is usually transported on a light cargo truck (one kit per 25 vehicles). Using slave kits instead of starting aids may be very inconvenient when vehicles are operating in remote areas or when an entire unit must be ready to move in a short period of time.

14-6.3 ESSENTIAL EQUIPMENT AND MATERIAL FOR TANKS (Ref. 1)

Essential equipment and material for cold weather operation of tanks after three days' exposure include:

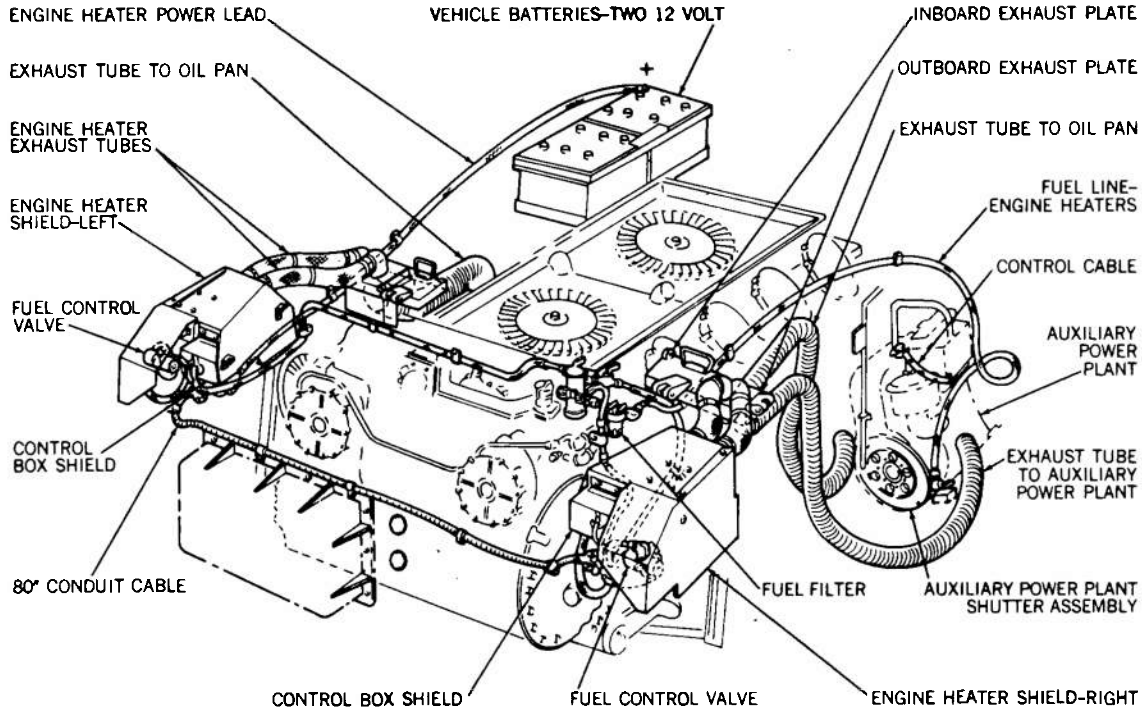


Figure 14-10. ***Typical Contaminated-Air Heating Installation***

1. At 40° to 0°F: personnel heaters
2. At 0° to −25°F: arctic-type fuels, lubricants, and engine primers; and personnel heaters. Heat for power plants and batteries may or may not be needed depending on equipment design.
3. At −25° to −40°F: arctic-type fuels, lubricants, engine primers and personnel heaters. Heat for power plant and batteries is needed also.
4. At −40° to −65°F: arctic-type fuels, lubricants, and engine primers; high-capacity heating equipment capable of raising temperatures of power plants and batteries to 0°F before starting; and personnel heaters. Heat to batteries must be continued after starting until these units reach 40°F to ensure recharging with normal voltage regulator settings. Heat to other components may have to be continued after starting to achieve efficient operating temperatures.

14-6.4 CONTAMINATED-AIR HEATERS FOR TANKS

Power plant heaters currently used as starting aids for tanks in the field are standardized units that employ attachments or kits designed especially for air cooled units. These arctic winterization kits are issued for use in regions outside the continental United States where the temperature during the coldest month of the year is −25°F or lower.

Contaminated-air heaters are designed with burners having high-excess air (low CO) to produce a large volume of comparatively low-temperature exhaust gases. The exhaust gases are ducted to critical points within the engine compartment. Usually, the air is discharged beneath the engine and allowed to rise up through the cylinder fins and the air cooling system of the engine. Figure 14-10 shows the schematic layout of a typical contaminated-air heating installation.

Every precaution must be taken to prevent carbon monoxide from entering the crew compartment. Batteries are enclosed in an insulated box through which the contaminated air circulates to heat the batteries rapidly. The system must be equipped with thermostatic valves to cut off the heat before batteries are damaged.

Standby contaminated-air heating of air-cooled engines requires no external fans or pumps and thus has a very low current drain. Quick-heat contaminated-air heating is rapid and efficient, but requires more power for the amount of heat produced. The main disadvantages of the use of contaminated air for heating are condensation and resultant corrosion, danger to personnel breathing the exhaust gases, and necessity for installing stainless steel ducts.

Radiation heating is a modification of the contaminated-air system. It carries the heater exhaust gases through ducts adjacent to the points to be heated. In this manner, the gases do not come in direct contact with the parts of the power plant. This eliminates the corrosion problem and need for the use of stainless steel parts on the engine.

A combination radiant and direct heating from exhaust gas systems could be designed to obtain high heating efficiency with protection for personnel and all delicate parts of the equipment. The cost and difficulty of installation of this type of system may be prohibitive.

14-7 DESERTIZING EQUIPMENT

Desertizing equipment is considered as those appliances and modifications that are necessary to vehicle equipment to assure satisfactory operation in the desert and other locations where extremely high temperatures (up to 125°F) are experienced. The main difficulties encountered in high-temperature operation of vehicles is the formation of a vapor lock in the fuel system, overheating of the engine, and, in desert areas, the induction of dust into the engine. With the improvement of gasoline quality and incorporation of certain additives to the gasoline, vapor lock is not a serious problem at the present time. To this date, desertizing kits have not been standardized; tests and research are being conducted along these lines. Some of the items that may be included in a desertizing kit are as follows:

1. Electric fuel pump to replace the standard fuel pump. The electric fuel pump tends to eliminate the possibility of vapor lock.
2. Radiator surge tank for liquid-cooled engines. The purpose of the surge tank is to catch the overflow of cooling liquid as it expands during hot weather operation.
3. Larger diameter cooling fan for air-cooled engines.
4. High-capacity air filters for both the engine and crew compartments.

14-8 FIRE-FIGHTING SYSTEMS

All military vehicles are equipped with hand-operated fire extinguishers; while some vehicles are equipped with an automatic central fire-fighting system, utilizing carbon dioxide. The carbon dioxide, under pressure, is piped to the various locations of the vehicle that are most vulnerable to fire. These pipes are terminated with fittings having low melting point seals. When a seal is overheated, it breaks, releasing a stream of carbon dioxide gas which smothers the fire. The specifications normally designate the type of fire-fighting equipment to be used for a particular vehicle.

14-9 SPECIAL EQUIPMENT

The special equipment classification is very broad, ranging from power takeoffs to entire vehicles. The following paragraphs present a brief description of several of the most common special accessories. A more comprehensive discussion is presented in Ref. 6.

14-9.1 POWER TAKEOFF

A power takeoff is an attachment for connecting the main power plant to power-driven auxiliary machinery when the use of such machinery is required. The power takeoff is usually attached to a transmission, auxiliary transmission, or a transfer case, and is provided with a means of engaging and disengaging the unit. A number of power takeoff assemblies are specified as standard military components. These vary from single-speed nonreversible units to three-speed reversible assemblies.

The main purpose of the power takeoff is to drive the winches that are standard equipment on tactical military vehicles. When applicable, the power takeoff also drives the pump for the hydraulic dump-truck mechanism. Figure 14-11 shows a typical power takeoff winch drive.

14-9.2 WINCHES (STANDARD VEHICLES)

The primary purpose of the winch on standard military vehicles is increased mobility by providing a means by which the vehicle may be able to pull itself or another vehicle out of adverse terrain. Expediency may dictate other uses for the winches in the field.

Standard military drum winches range from assemblies of 5,000-lb capacity, having 3/8 in. by 100 ft of cable, to assemblies of 90,000-lb capacity

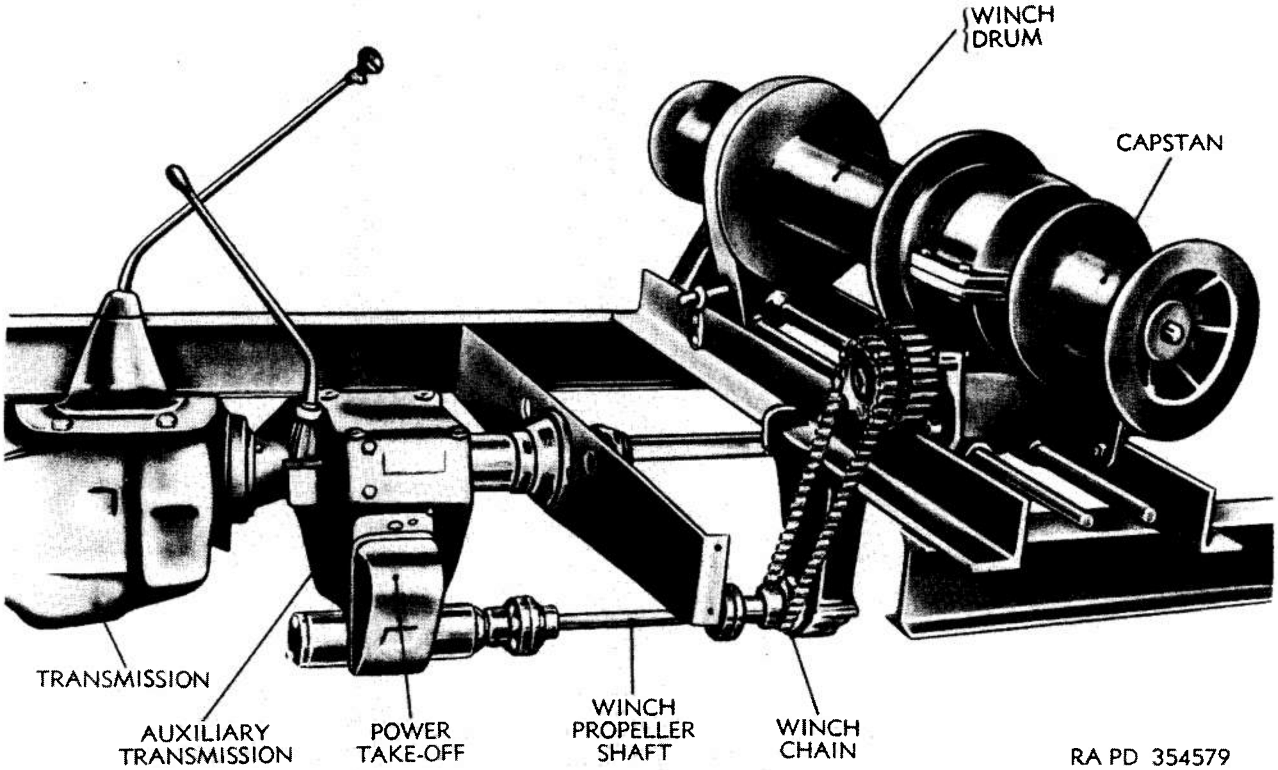


Figure 14-11. ***Auxiliary Transmission Power Takeoff and Winch Assembly***

having 1-1/4 in. by 200 ft of cable. The latter units are used on tracked recovery vehicles.

14-9.3 DUMP BODIES

Dump trucks are examples of special equipment vehicles. Dump bodies are raised and lowered by means of a hydraulic cylinder. A separate pump driven by a power takeoff supplies the required high-pressure hydraulic fluid. Figure 14-12 shows a typical dump truck in elevated position.

14-9.4 TIRE INFLATION SYSTEM

Certain military vehicles are equipped with a central tire pressure control system. The tires may be inflated or deflated, as required, to meet the various conditions encountered by the vehicle. For example, when operating on sand, the tire pressure can be reduced to increase floatation. When the vehicle reaches harder surfaces, the tires can be inflated to meet the new conditions.

Currently used tire inflation systems have a two-cylinder, water-cooled, self-lubricated air pump, with a capacity of 9 cfm mounted in the engine compartment and driven by the engine crankshaft. This pump is controlled by a governor which stops the pump when the maximum allowable pressure is attained in the air storage tank. The governor will also start the pump when the pressure in the tank falls below a prescribed limit. Air pressure is piped from the air storage tank to the inflation-deflation control assembly. From the control assembly, air is piped to the tire-inflating device located on each wheel hub. A safety valve must also be provided in the tire-inflation system to protect it in the event the governor does not shut off the pump at the desired pressure.

14-10 PROVISIONS FOR ON-VEHICLE MATERIEL

All tactical vehicles are supplied with equipment that, while not permanently installed in the vehicle, is essential to the successful execution of tactical missions. The type and amount of material in this classification will vary with the vehicle, and may include fighting items such as ammunition; repair parts, e.g., track shoes, tools for repairs and pioneer operations; emergency items, e.g., fire extinguishers; and personal items, e.g., rations.

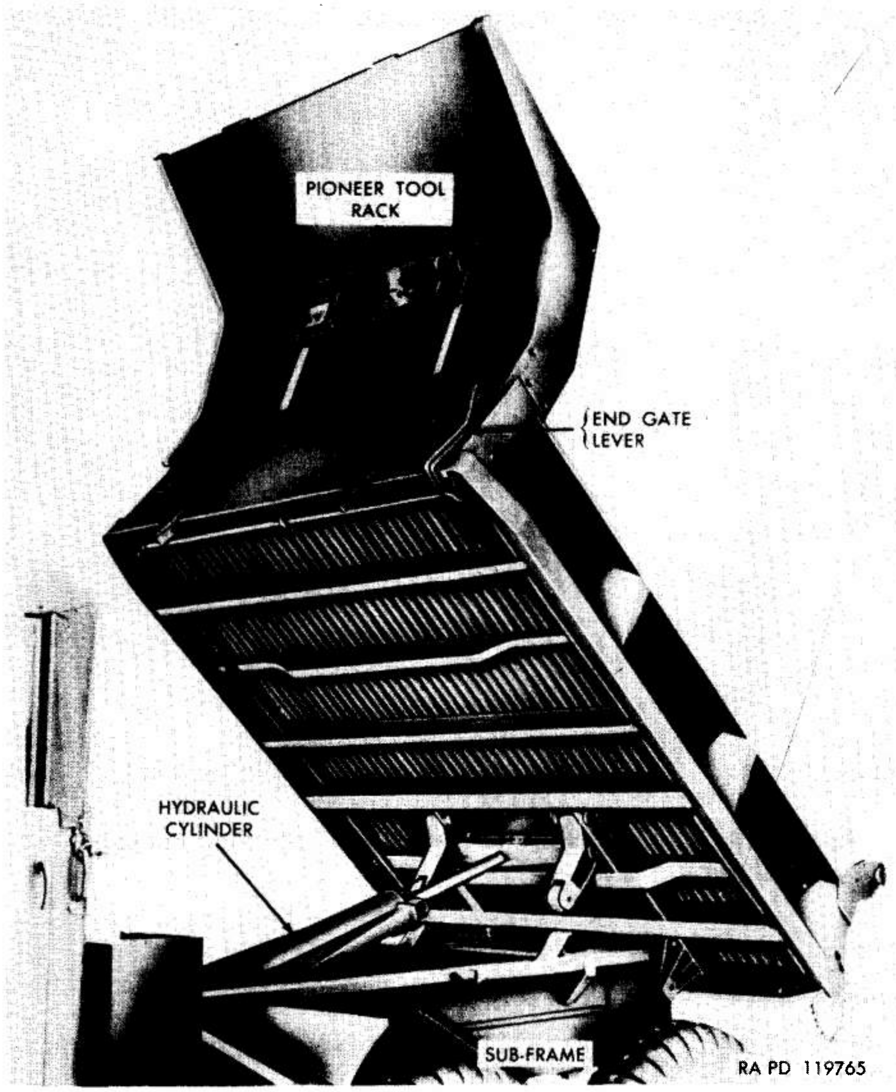


Figure 14-12. ***Dump Body in Raised Position***

The selection of on-vehicle materiel is made by the using services and the designer; however, provisions for locating and mounting the various items must have the continuing attention of the designers throughout the development of a new vehicle. All items required by the crew members during operations, such as ammunition, must be located to afford the maximum convenience of access. Fire control and signal items, likewise, must be within easy reach to facilitate prompt use when required.

14-11 COMMUNICATION EQUIPMENT

Communication equipment, including intercom systems, has been standardized by the Army Electronics Command. The basic radio sets can be installed and operated in tanks, trucks, armored personnel carriers, and other vehicles as required. Auxiliary equipment peculiar to the needs of a particular vehicle can be added to the basic radio set to fulfill the communication requirements of that vehicle.

Placement of the main components and the auxiliary items of the basic radio sets within the various vehicles depends on functional requirements, crew members position, and available space. Although the vehicle designer will not be required to design the communication equipment, he will have to incorporate in his design the provisions for the installation; and since this equipment is maintained on a unit replacement basis, removal and reinstallation should be easily accomplished.

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GLOSSARY

Ackermann steering. The standard system of steering in which the front wheels are mounted on pivoted knuckles and are interconnected by a linkage. During a turn, the inner wheel rotates through a larger angle than does the outer wheel.

adiabatic. Occurring without gain or loss of heat; a change of the properties, such as volume and pressure of the contents of an enclosure, without exchange of heat between the enclosure and its surroundings.

air cleaner, intake. A device designed to prevent foreign particles from entering the air intake system of an internal combustion engine or air compressor.

air resistance. The motion resisting force caused by the interaction of the air and a body. Air resistance is a function of the state of the air, the geometry of the body, and the relative velocity between the body and the air.

amphibious tractor. (amtrac) Vehicle used for the movement of troops and cargo from ship to shore in the assault phase of amphibious operations or for limited movement of troops and cargo over land or water.

angle of approach. The maximum angle of an incline onto which a vehicle can move from a horizontal plane without interference; as, for instance, from front bumpers.

angle of departure. The maximum angle of an incline from which a vehicle can move onto a horizontal plane without interference; as, for instance, from rear bumpers.

antiknock. A substance, such as iso-octane or tetraethyl lead, which may be added to gasoline, or used as a fuel itself, to prevent detonation in an engine cylinder.

articulated steering. The system of steering used by tracked or wheeled vehicles consisting of two or more powered units in which the turning maneuver is accomplished by yawing the units with respect to each other about a pivot system not located over an axle of either unit.

articulated vehicle. A tracked or wheeled vehicle consisting of two or more powered units.

automobile. A self-propelled, wheeled vehicle, generally commercially designed, for transporting less than ten (10) passengers on highways and/or roads. Excludes bicycles, motorcycles, and motor scooters.

automotive vehicles. A general category of mechanical land vehicles that contains means of propulsion within themselves. They are generally considered to be either wheeled or track-laying; but, in the broad sense, this category includes all types of walking and jumping vehicles as well as self-propelled sleds and various air-cushion supported vehicles. They may have the ability to negotiate deep water barriers by swimming on the surface, in which case they are amphibious automotive vehicles, or by swimming submerged, in which case they are submarine automotive vehicles.

axle assembly. A device suspended between and connecting opposite wheels which consists of the housing and driving differential mechanism. The assembly also supports the weight of the vehicle.

axle load. The total load transmitted to the road by all wheels whose centers are included between two parallel transverse vertical planes 40 inches apart, extending across the full width of the vehicle (from AR 705-8).

axle tramp. The sustained vibration of the axle of a solid axle suspension in a vertical plane.

band track. A continuously flexible track usually comprised of an endless band of rubber reinforced with steel cables.

binocular frames. Track units with webbed ends designed to contain the bushings and pins of double-pin tracks.

block and pin track. *See* **jointed track.**

body. *See* **hull.**

bogie wheel. *See* **road wheel.**

bottom roller. *See* **road wheel.**

brake fade. A temporary failure in a braking system due to excessive temperature.

brake specific fuel consumption. The amount of fuel used by an engine related to the brake horsepower output—lb of fuel per horsepower-hour.

bogie (tracked vehicles). A suspension assembly in which roadwheels (bogie wheels) are interconnected in tandem by a system of arms, walking beams, cranks, springs, etc., in such a manner that when one wheel experiences a vertical force or displacement, a corresponding change in loading or position is reflected in the other wheels of the bogie unit.

bogie (wheeled vehicles). A suspension assembly consisting of tandem axles, interconnected by walking beams which pivot vertically about a cross member (trunion axle). Also, a tandem axle assembly without a distinct walking beam but interconnected by a system of crank and links in such a manner that when an axle experiences a vertical force or displacement, a corresponding change in load or position is reflected in the other axle.

bogie wheel. *See* **bogie (tracked vehicles).**

bounce. The upward movement of the sprung mass of a vehicle, away from the unsprung mass, in response to suspension system disturbances. Cf. **jounce.**

bounce distance. The maximum upward travel of the sprung mass of a vehicle, away from the unsprung mass and measured from the free standing position, before further upward deflections of the suspension mechanism are rigidly restrained. Cf. **jounce distance.**

bullet splash. Minute metal particles or metal dust which is formed as a result of a projectile impact against armor.

camber. A setting of the front or rear wheels of a vehicle, closer together at the bottom than at the top.

carburetor. A mechanical device for atomizing and mixing a liquid fuel with air in correct proportions for combustion.

car, armored. A wheeled, self-propelled vehicle with protective armor plate designed for combat use and usually equipped with armament.

carrier, personnel. A self-propelled vehicle, sometimes armored, used for the transportation of troops and their equipment.

clutch, friction. A clutch which transmits motion or power from the driving to the driven member by the frictional resistance between the engaging surfaces.

combat vehicle. A land or amphibious vehicle, with or without armor or armament, designed for specific functions in combat or battle. The installation of armor or armament on vehicles other than combat vehicles does not change their original classification.

combustion chamber. The space between a piston and the cylinder head of a reciprocating engine at the end of the compression stroke. Combustion is initiated in this volume.

compound engine. A power plant that combines features of the reciprocating piston engine and the gas turbine. Examples are the turbosupercharged piston engine and the free-piston engine.

compression-ignition. In an internal combustion engine, ignition of the fuel produced by the temperature of the compressed gas within the cylinder.

compression-ignition engine. A type of reciprocating internal-combustion engine in which ignition of the injected fuel is caused by the temperature of the compressed air charge within the cylinder.

compression-pressure ratio. The ratio of the final pressure reached during compression divided by the pressure at the beginning of compression.

compression-volume ratio. The ratio of the volume at the beginning of compression to the volume at the end of compression in a piston-cylinder system. In the automotive field, the normal use of the term compression ratio is based on this relationship.

condual tire. A tire consisting of two tubes or carcasses, the major outside diameter of the smaller equal to and nested within the major inside diameter of the other. Each carcass is permitted a maximum deflection compatible with acceptable wear rates. The allowable deflection permitted in this type of tire is approximately double that of conventional tires. The larger deflection obtained leads to a long, thin contact area considered desirable from a soft-soil mobility viewpoint.

coned-disk spring (Belleville). Annular metal disk dished to a conical shape, loaded by a compressive force applied along the axis of the annulus.

constant-pressure combustion. Combustion of fuel in a cylinder at a rate slow enough so that there is no rise in cylinder pressure. The slow-speed air-injection Diesel is a constant-pressure combustion engine.

constant-volume combustion. Combustion in a cylinder while there is no change in clearance volume. All the energy of combustion goes to raise the cylinder pressure. The gasoline engine and many high-speed Diesels have constant-volume combustion, or operate on the Otto cycle.

cornering force. The force, in pounds, measured normal to the longitudinal plane of a wheel or track, which is exerted by the ground contacting area in resisting the centrifugal force developed when a vehicle moves in a nonlinear path.

cycle. A complete series of recurring values or events. Specifically, the series of actions an internal-combustion piston engine must perform to operate and deliver power. *See* **four-stroke-cycle engine; two-stroke-cycle engine.**

damping. Process of effecting a continued decrease in the amplitude of vibration of an oscillating component, generally accomplished through some type of friction.

deadline. To remove a vehicle or other piece of equipment from use for one of the following reasons: (1) Vehicle is inoperative due to damage, malfunctioning, or is undergoing necessary repairs. This does not include vehicles removed temporarily from use for routine maintenance and repairs that do not affect its combat capability. (2) Vehicle is unsafe. (3) Vehicle would be damaged by further use.

deep-fording. *See* **fording.**

detonation. The instantaneous and abnormal combustion of an unburned part of the fuel-air mixture in the cylinder of an engine.

diffuser. A device for diffusing a fluid. Specifically, a duct or vane designed to convey air into a manifold or combustion chamber while reducing its velocity and increasing its static pressure.

dissociation. Process by which a chemical combination breaks into simpler constituents. Dissociation of combustion products in a piston internal-combustion engine prevents the attainment of the theoretical temperature of complete combustion within the chamber.

drawbar pull. The amount of tractive effort developed by a vehicle in excess of motion resistance (net tractive effort).

drawbar pull-weight ratio. An index of the efficiency of a vehicle system similar in concept to the lift-drag ratio for an aircraft. The drawbar pull-weight ratio indicates the effort available for hill climbing, vehicle acceleration, load towing, etc.

durability. That characteristic, pertaining to an object, device, or system of devices, related to the period of time of satisfactory operation on a comparative basis. If two or more comparable items are subjected to the same operating conditions, the one that operates satisfactorily for the longest period of time is the more durable. Ability to withstand abuse is also a characteristic of a durable unit.

dynamic axle reaction. The motion-induced effective axle loading. Acceleration, braking, air resistance, and drawbar loads affect the axle loadings.

elastic girder track. A track in which adjacent links are interlocked by elastic components, such as rubber buffers, to limit reverse bending. Cf. **rigid girder track; flexible track.**

elastic wheel. A resilient wheel such as the pneumatic-tired wheel. The ground contact area of an elastic wheel on rigid ground is relatively large. Cf. **rigid wheel.**

engine. Any of various machines that convert energy in one form, as that of heat, into a form suited to a particular use, as that of torque, applied to a crankshaft or of kinetic flow directed into a jet stream.

engine, pulsejet. A combination-type power unit designed to exert thrust by receiving air through valves in its front and mixing this air with a continuous supply of metered fuel which is ignited. The expanding gases close the valves which causes the exhaust gases to leave through a tail pipe with the forward thrust reopening the valves and causing a repetition of the cycle.

engine, ramjet. A continuous mass flow power unit designed to exert thrust. The forward motion of the engine is used to compress atmospheric (ram compression) in the inlet diffuser. The compressed air is charged with a continuous spray of pressurized fuel, ignited and ejected at high velocities through the exit nozzle.

engine, turbojet. A continuous-combustion-type power unit designed to exert thrust. Prime physical characteristics of a turbojet engine include an air compressor, a fuel injection system, combustion chamber(s), a turbine to drive the compressor, or an exit nozzle to expel the hot gases rearward.

enthalpy. The sum of the internal and pressure energies of a substance or system; often called the total heat. Change in enthalpy is the amount of heat added to, or subtracted from, a substance or system in going from one state to another under constant pressure.

exhaust manifold. A collecting chamber through which the burnt gases from the various cylinders are discharged on their way through the exhaust pipe and through the muffler.

exoskeletal construction. A construction technique in which the body is a major stressed member. This is the principle of unit construction used by some automotive manufacturers and can result in a sizable reduction in vehicle weight.

expansion ratio. In jet propulsion the ratio of the nozzle exit section area to the nozzle throat area.

fifth wheel. Flat round steel plate, swivel-mounted on the frame siderails at the rear of a truck tractor used to couple a semitrailer to it. Part of a fifth-wheel assembly.

fifth-wheel assembly. A device designed for attaching a semitrailer to a truck tractor or dolly in such a way as to allow free rotation in a horizontal plane and yet prevent tipping.

fighting compartment. Portion of a fighting vehicle in which the occupants service and fire the principal armament. It occupies a portion of the hull and all of the turret, if any.

filter, oil. On automotive vehicles, a device whose primary function is to remove contaminating substances, such as dust and dirt, from the oil by passing it through a filtering element. It is generally designed with a bypass valve, which permits free circulation of the lubricating oil when the filter elements become clogged and retard oil movement.

flat track suspension system. A suspension system on a tracked vehicle wherein the track returns on the top surfaces of the road wheels without the use of supplementary support rollers.

flexible track. A track that can flex in either direction about a horizontal transversal axis. Cf. **rigid girder track; elastic girder track.**

floating. This is the ability of a vehicle to negotiate water obstacles without being in contact with the bottom. Self-propulsion while in the water is not implied in this definition.

fording. This is the ability of a vehicle with its suspension in contact with the ground to negotiate a water obstacle of a specific depth. **Shallow-fording** is fording without the use of special waterproofing kits, while **deep-fording** is fording of greater depths with the application of a special waterproofing kit.

four-stroke-cycle engine. An internal combustion, piston engine requiring four strokes of each piston to complete a cycle. Cf. **two-stroke-cycle engine.** This type of engine is often called a 'four-cycle engine'; consequently a misunderstanding of the work 'cycle' has arisen, some users of the term confusing 'cycle' with 'stroke'. The four piston strokes necessary to complete a cycle in the four-stroke-cycle engine are the intake stroke, compression stroke, power stroke, and exhaust stroke.

frame. A structure, separate from the body or hull, that supports the various components of the automotive assembly and maintains their spatial relationship. The frame provides strength and rigidity to the vehicle.

friction horsepower. The difference between indicated horsepower and brake horsepower, i.e., the horsepower used by an engine in overcoming the friction of moving parts, inducting air or air-fuel mixtures, expelling exhaust, driving oil and fuel pumps, and the like.

fuel-air ratio. The weight ratio of fuel to air as supplied to the combustion chamber of an engine.

fuel cell. An electrochemical device in which part of the energy, resulting from a chemical reaction that is maintained by a continuous supply of chemical reactants, is converted directly to electrical energy.

fuel injection. The forced introduction (in the form of a spray) of fuel or fuel and air into the intake system or directly into the combustion chambers of a piston engine. Fuel injection is necessary for compression-ignition engines and may be applied to spark-ignition engines.

full-track vehicle. Vehicle entirely supported, driven and steered by means of tracks.

gas turbine. A continuous combustion engine consisting primarily of a compressor, fuel injection system, combustion chamber, and a turbine to produce rotary shaft power. Two of the basic types are the open type and the closed type. In the open type, all of the working fluid (air and combustion products) passes through the plant but once. In the closed type the working fluid, which does not include the combustion products, is continuously recycled. Heat is transferred to the working fluid through the walls of a closed heater.

Goer type vehicle. A four-wheeled vehicle having the following combination of distinguishing features: large-diameter tires, exskeletal construction, powered-wagon-wheel steering, power to all wheels, and suspension system consisting of tires only.

gradeability. The slope-climbing ability of an automotive vehicle.

grade resistance. The motion-resisting force acting on a vehicle traveling up a grade. For a vehicle going down a grade the grade resistance force becomes negative.

gross tractive effort. The maximum propelling force that can be developed by the ground-contacting elements of a vehicle on a given type of support.

gross vehicle weight. The chassis or the hull weight, plus the weight of the entire body, fully equipped and serviced for operation, plus operating personnel.

ground contact area. The area of the ground contacting-element of a suspension system that is in contact with the ground and has a fuction in supporting the weight of the vehicle. On soft ground, it is assumed to be the product of the overall length and width of the area in contact, including all open spaces between components of the ground contacting element. On rigid surfaces, it is the actual area in contact with the ground, exclusive of the open areas.

ground pressure. The force exerted by a vehicle on the ground, usually expressed in pounds per square inch. Mean ground pressure equals the gross weight of the vehicle divided by the ground contact area in soft ground. Actual ground pressure can be obtained only from complex calculations that take into consideration the unequal wheel loading, flexibility, form, and dimensions of the ground-contacting element. The actual ground pressure is usually nonuniformly distributed beneath the ground-contacting element. Cf. **ground contact area.**

grouser (spud). A detachable or integral projection (often chevron shaped) on a track shoe, normal to the tread surface, provided for improved traction in off-the-road operation.

guide horn. *See* **track guide.**

half-track vehicle. A vehicle in which some wheels (usually the front steered wheels) run without tracks while the others run on tracks.

handling. The maneuvering and course-keeping characteristics of an automotive vehicle. Expressed by: (1) the ease and precision with which it is possible to steer the vehicle or achieve a desired path and with which this path is maintained, and (2) the control response and stability of the vehicle (stability is the ability to maintain a given state of equilibrium). A stable vehicle returns to its initial state of equilibrium after a disturbance has been removed or acquires a new equilibrium state if the disturbing force is held constant.

helical coil spring. Round, square or rectangular wire, wound in the form of a helix, offering a resistance to a force applied along the axis of the coils. When wound with space between coils, they may be loaded in compression. When the force is applied in a manner that separates the coils it is termed a helical tension spring.

helical torsion springs. Round, square or rectangular wire, wound in the form of a helix, offering a resistance to a moment applied in a plane perpendicular to the coil axis.

hot spot. An area within an intake manifold which receives heat from the engine exhaust (usually thermostatically controlled) and on which the fuel particles impinge while passing through the manifold. The purpose of the hot spot is to assist in the vaporization of the fuel.

Hotchkiss drive. In automotive vehicles, a method of drive by which the torque reaction is transmitted to the frame through the spring rather than through a torque tube or a torque arm.

hull. The body or hull of an automotive vehicle is the main structure which forms the passenger, cargo and component compartments. The term body is usually applied to wheeled vehicles, while the term *hull* is applied to amphibious and tracked vehicles.

hydraulic spring. A sealed plunger working in a highly finished cylinder, against an enclosed volume of liquid. The resiliency of the spring is derived from the compressibility of the liquid at high pressure.

hydropneumatic. Pertaining to, or operated by means of, a liquid and a gas; used with recoil and equilibrator mechanisms which provide variable absorption of energy or thrust.

hydropneumatic spring. A self-contained spring and shock absorbing unit comprised of an enclosed volume of gas and fluid separated from each other usually by a flexible diaphram or a piston. The system derives its elasticity from the compressibility of the gas, while the fluid provides system damping, vehicle leveling, and ground clearance control.

hydrospring. Pertaining to, or operated by means of, a liquid and springs; used with recoil and equilibrator mechanisms which provide variable absorption of energy or thrust.

idler. On track-laying vehicles, the wheel at the end of the vehicle opposite the driving sprocket, over which the track returns. It maintains track tension and reduces track skipping.

independent suspension. A system of arms, springs, wheels, etc., for elastically supporting the sprung mass of a vehicle, which permits the deflection of any one of the supporting wheels without substantially changing the load or position of the remaining wheels (or distinguished from solid axle or bogie suspension systems).

inertia resistance. As applied to an automotive vehicle, the resisting forces opposing the linear and angular accelerations of the various masses of the vehicle.

inline engine. An internal-combustion, reciprocating-piston engine in which the cylinders are arranged in a single straight row.

intake manifold. A device that distributes the air (with fuel injection) or the air-fuel mixture (with carburetion) to individual cylinders on a multicylinder engine.

isentropic process. A reversible adiabatic process. Cf. **adiabatic.**

jointed track (block and pin track). A track comprised of rigid links connected by joints at which flexing occurs. Cf. **band track.**

jounce. The downward movement of the sprung mass of a vehicle, toward the unsprung mass, in response to suspension system disturbances. Cf. **bounce.**

jounce distance. The maximum downward travel of the sprung mass of a vehicle, toward the unsprung mass, and measured from the free standing position. Cf. **bounce distance.**

leaf spring. A flat bar spring that is relatively thin in proportion to its length and width, designed to be loaded in bending. In vehicle suspensions, leaf springs usually are a lamination of several leaves of unequal lengths.

liquid-cooled engine. An engine that has a water jacket around the valve ports, combustion chambers, and cylinders and a radiator for dissipating the heat from the cooling liquid into the surrounding air. As a rule, liquid-cooled engines use a pump for circulating the cooling liquid.

L/T ratio. A steering ratio in which L represents the length of track in contact with the ground, and T represents the lateral distance between the centerlines of the tracks.

lunette. A towing ring in the trail plate or torque of a towed vehicle, such as a nonself-propelled gun carriage or trailer, used for attaching the towed vehicle to the prime mover or towing vehicle.

mean effective pressure. That theoretical constant pressure which, if exerted on a piston during a power stroke, would yield a net amount of work equal to the actual work output of the cycle.

mean ground pressure. *See* **ground pressure.**

mechanical efficiency. The external efficiency of an engine rated in horsepower, as an internal-combustion reciprocating engine, expressed as the ratio of brake horsepower to indicated horsepower.

military characteristics. Those characteristics of equipment found desirable or necessary to the performance of a military mission, either combat or noncombat. Military characteristics are prescribed by the using arms and usually form the basis of initiating development of a new item.

mobility. The competence of a vehicle to perform its mission as measured by its best average speed over a route representative of the terrain where it will operate.

neutral steer line. The line at which lateral forces applied to a automotive vehicle do not cause yawing.

neutral steering vehicle. A vehicle that inherently tends to maintain the radius of curvature as it travels in a curved path and is acted on by centrifugal force. Cf. **oversteering vehicle; understeering vehicle.**

octane number. A number assigned to a liquid fuel to designate its relative antiknock value in a reciprocating engine of the spark ignition type. The octane number is the percentage number of the iso-octane in a given fuel mixture of iso-octane and normal heptane that matches the fuel being tested in antiknock properties. The higher the octane number, the more compression the fuel can withstand without detonation.

opposed engine. An internal-combustion, reciprocating engine having pistons on opposite sides of the crankshaft.

Otto cycle. A reciprocating, internal-combustion engine cycle characterized by constant-volume combustion.

oversteering vehicle. A vehicle that inherently tends to decrease the radius of curvature as it travels in a curved path and is acted on by centrifugal force. Cf. **understeering vehicle; neutral steering vehicle.**

percent of slope. Angle of ascent or descent expressed as a percent; the number of units a slope rises, or falls, vertically in a horizontal distance of 100 identical units.

pin jointed track. A track of a track-laying vehicle in which the flexing occurs as angular oscillations of the journals about their pins, resulting in a sliding of the surfaces.

pintle assembly. A hook and latch assembly, usually mounted to the center rear of a vehicle, used in towing other vehicles. Some vehicles are provided with a pintle assembly at the front as well as at the rear.

piston displacement. The volume displaced by any or all of the pistons of a reciprocating engine during a specified number of strokes, usually one stroke per piston.

piston engine. A reciprocating engine, especially an internal-combustion reciprocating engine.

pitch. The angular displacement of a vehicle about an axis parallel to its lateral (horizontal) axis. Cf. **roll; yaw.**

pneumatic spring. A self-contained spring assembly that derives its spring action from the compressibility and elasticity of an enclosed gas.

power plant. The integration of subassemblies and individual components required to convert the energy of some fuel source to a form useful to the vehicle. Thus, it includes not only the basic engine, or engines, but also the fuel systems, lubricating systems, cooling systems, exhaust systems, electrical systems, and all other necessary accessories.

power train. The system of components that transmits the useful energy produced by the power plant to its ultimate point of application. It includes such components as clutches, transmissions, transfer cases, drive shafts, differentials, axles and brakes.

power train efficiency. The ratio, expressed as a percent, of the power input to the wheels or tracks of a vehicle over the power delivered to the output shaft of the driving engine (power input to transmission).

power-transmission system. A group of units transmitting power from the engine (power plant) to the wheels or tracks. It consists of clutch, transmission, propeller shafts, universal joints, differentials, and driving axle shafts.

preignition. The spontaneous and premature ignition of the mixture in the combustion chamber of a reciprocating engine, caused by an overheated part or spot in the chamber.

prime mover. In a contrivance of two or more moving parts, that unit considered to be the source, or principal source, of energy for movement, as with a tractor pulling a trailer.

radial engine. An engine with one or more stationary rows of cylinders arranged radially around a common crankshaft. In a more general sense, any engine having the cylinders arranged radially around the crankshaft.

ratio of specific heats. The ratio of specific heat at constant pressure to specific heat at constant volume.

re-entrant angle. Angle formed by surfaces of a vehicle such that a ballistic impact striking either surface may be richocheted against the other surface.

regenerative engine. An engine that utilizes the heat of combustion to preheat air or fuel entering the combustion or expansion chamber.

reliability. The probability of a device performing its purpose adequately for the period of time intended under the operating conditions encountered. For a system with independent components the overall reliability is based on the product of the individual reliabilities; e.g., three independent components with a 90% reliability each will have an overall reliability of $.9 \times .9 \times .9$ or 72.9%. Similarly, 100 components with a 99% reliability each will have an overall reliability of only 36.5%. Mechanical reliability as applied to military automotive equipment also includes the capacity of a vehicle to perform its mission after sustaining failure or destruction of specific components.

return roller. *See* **return wheel.**

return wheel (top roller) (return roller). One of a number of wheels that supports the top run (return run) of the track between the drive sprocket and idler of a track-laying vehicle.

reverse bending. Flexing of a track in a direction opposite to that assumed when passing around the sprockets.

reversible steering gear. A vehicle steering gear that transmits motion from the driving wheels to the steering wheel.

rigid girder track. A track in which adjacent links interlock to form a girder that is rigid in one direction, thus preventing reverse bending. Cf. **elastic girder track; flexible track.**

rigid wheel. A wheel that deforms a relatively negligible amount on a hard surface, and in the limiting case has a line ground contact pattern. A steel railway wheel is an example of a rigid wheel. Cf. **elastic wheel.**

ring gear. A gear cut on a ring-shaped rim. Specifically, in an automotive vehicle, the large gear in the differential that is driven by the propeller shaft pinion and transmits the power through the differential to the live axle.

road wheel (bogie wheel, bottom roller). One of a number of wheels which support the weight of a tracked vehicle and roll on the inside of the bottom run of the track.

roadability. A rating of the operating characteristics of an automotive vehicle, taken collectively, that define the quality of the vehicles traveling performance. Included in this total rating are such factors as ease of steering, gradeability, acceleration, road holding, suspension stiffness, rebound control, directional stability, braking characteristics, skidding characteristics, etc.

roll. The angular displacement of a vehicle about an axis parallel to the vehicle's longitudinal axis. Cf. **pitch; yaw.**

roll axis. *See* **roll center.**

roll center. The center about which a portion of the total sprung mass of a land vehicle rotates when a side force is imposed on the vehicle. The position of the roll center relative to the road surface depends on the type of suspension system used. The entire vehicle rotates about a roll axis which is generated by the positions of the major roll centers of the vehicle. For example, a four-wheeled vehicle will have a separate roll center for the front and rear suspension systems and these points will be on the roll axis.

rolling resistance. The motion-resisting force developed by the interaction of the wheels or tracks of a vehicle and the ground. When the rolling resistance is subtracted from the gross tractive effort the effective propelling force remains.

rubber-bushed track. A jointed track incorporating rubber bushings which permit flexing by annular shear, i.e., relative rotation between the inner and outer cylindrical surface of the rubber.

rubber torsion spring. A spring assembly generally consisting of a metal shaft bonded to an annular layer of rubber which is in turn bonded to an outer concentric metal shell. Spring action is derived by twisting the inner shell relative to the outer shell by applying a moment in a plane perpendicular to the shaft axis, thus loading the rubber in annular, or torsional, shear.

SAE horsepower formula. The standard (Society of Automotive Engineers) formula for computing the horsepower of gasoline engines for tax purposes is as follows:

$$\text{hp} = \frac{D^2 \times N}{2.5}$$

based on 1,000 feet per minute piston speed. D is the cylinder bore in inches, N the number of cylinders, and 2.5 a constant.

scavenging. *See* **supercharging.**

self-aligning torque. The horizontal torque exerted by a tire operating at a slip angle. The self-aligning torque is a function of slip angle and may be positive or negative.

semi-Diesel engine. A reciprocating internal-combustion engine of a type resembling the compression-ignition (Diesel) engine and using a heavy oil for fuel but employing a lower compression pressure than is customary in compression-ignition engines. Fuel ignition is accomplished by spraying the fuel, under pressure, against a hot (uncooled) surface or spot within the combustion chamber, or by the precombustion or supercompression of a portion of the change in a separate member or uncooled portion of the combustion chamber.

semitrailer. A nonpowered vehicle having integral wheels at the rear only, and designed to carry material, supplies, or equipment and to be towed by a self-propelled motor vehicle that also supports the front end, by means of a fifth-wheel coupling assembly. The front end can also be supported by a dolly that is provided with a fifth-wheel assembly, for coupling to the semitrailer, and a tongue and lunette, for coupling to the prime mover.

shaft horsepower. The horsepower delivered by an engine shaft. Usually the same as *brake horsepower.*

shallow-fording. *See* **fording.**

shimmy (wheel wobble). The vibratory oscillation of the steerable wheels of a vehicle about the kingpins.

shock absorber, direct action. A damper, either frictional or hydraulic, designed to dampen the shock of suddenly applied force and/or to control spring rebound and oscillation, usually attached to the vehicle frame, body or hull and connected to an axle, spring, spring support web or pad, or between suspension arms of track-laying vehicles.

shock absorber, lever action. A damper, either frictional or hydraulic, designed to dampen the shock of suddenly applied force and/or to control spring rebound and oscillation, usually attached to the frame of a vehicle, with the arm connected by a link or linkage to the axle or spring.

spark ignition. In an internal-combustion engine, ignition of the air-fuel mixture within the cylinder brought about by an electric spark.

specific fuel consumption. The amount of fuel used by an engine related to its power output usually given as lbs of fuel per horsepower-hour output. When based upon brake horsepower, it is given as brake specific fuel consumption (lbs of fuel per brake horsepower-hour) and when based upon indicated horsepower, it is referred to as indicated specific fuel consumption (lbs of fuel per indicated horsepower-hour).

specific horsepower. The power developed by an internal-combustion engine related to the total piston displacement. The units of specific horsepower are: hp per cubic inch of piston displacement.

sprung weight. Sprung weight is the total weight of all of the vehicle components that are supported by the vehicle spring system. This includes such major components as frame, body, power plant, transmission, clutch, cargo, etc. It does not include such items as wheels, tracks, axles, road wheels, etc. Cf. **unsprung weight;**

spud. *See* **grouser.**

square engine. An engine in which the stroke is equal to the diameter of the cylinder bore.

static-steering torque. The torque required to turn the wheels of a stationary vehicle. Actual turning center of a steered wheel is the intersection of the kingpin axis with the ground. The steering motion of the wheel around this point is a combination of sliding and pure rotation.

steering system. The assembly of linkages and components which enables the driver to control the direction of the vehicle. Wheeled vehicles are normally steered by rotating the axes of rotation of two or more wheels with respect to the longitudinal center line of the vehicle. While tracked vehicles are usually steered by varying the speed of the tracks with respect to each other.

supercharger. A compressor used to increase the volumetric efficiency or to assist the intake process of a piston internal-combustion engine. Cf. **supercharging.**

supercharging. In general, any assistance given to the intake process of a reciprocating internal-combustion engine by means of supplementary blower or compressor. It is usual to consider supercharging as a process which results in intake manifold pressures in excess of the ambient pressure, and to term the processes (with the blower) that do not increase the manifold pressure as scavenging.

suspension system. The mechanical linkages and the elastic members that provide a flexible support for the sprung components of a vehicle.

swimming. The ability of a vehicle to negotiate a water obstacle by propelling itself across, without being in contact with the bottom.

tactical vehicle. Any vehicle designed for field requirements in combat and tactical operations, or for training personnel for such operations.

tank, amphibious. Vehicle mounting a howitzer or cannon, capable of delivering direct fire from the water as well as ashore, and used in providing early artillery support in amphibious operations.

tank, combat, full-tracked. A self-propelled, heavily armored, vehicle having a fully inclosed revolving turret with one major weapon. It may mount one or more machine guns. Excludes self-propelled weapons.

tank, transporter. Special-purpose wheeled or tracked vehicle, or combination of vehicles, designed to transport tanks or other heavy vehicles over highway and natural terrain, and incorporating integral provisions for loading and unloading disabled vehicles without supplemental assistance.

toe-in. The degree (usually expressed in fractions of an inch) to which the forward part of the front wheels are closer together than the rear part, measured at hub height with the wheels in the normal 'straight ahead' position of the steering gear. Toe-in has the effect of counteracting the tendency of the wheels to roll outward or separate as a result of positive camber. Cf. **toe-out.**

toe-out. The outward inclination of the wheels at the front on turns due to setting the steering arms at an angle. Cf. **toe-in.**

thrust horsepower. The thrust of a jet engine or rocket expressed in terms of horsepower. Thrust is converted into horsepower by the following formula: thp equals thrust pounds times aircraft speed in miles per hour divided by 375.

top roller. *See* **return wheel.**

torque rod, tandem axle. A metal device designed to insure correct spacing and alignment of truck and trailer axles.

torque tube. In automotive vehicles, a tube that encloses the propeller shaft and is designed to resist propelling and braking reaction forces while maintaining the spacial relationships between the various interconnected units.

torsion bar spring. A straight bar spring, usually cylindrical, employed as the elastic member in one type of vehicle suspension. One end of the bar is secured in torsion to the vehicle frame or hull while the other end is supported by and free to rotate in a hull mounted bearing. Torsional loads are applied to the bar by means of an arm fastened to the free end and rotated in a plane perpendicular to the longitudinal axis of the torsion bar.

total heat. *See* **enthalpy.**

track. The continuous band or segmented chain upon which a tracked vehicle runs. Cf. **track-laying vehicle.**

track body. The basic structural unit of a track link. On double-pin tracks it is the track block. Cf. **track shoe assembly; track link.**

track guide. The track projections on the roadwheel side of a track that locate the roadwheels on the wheel path and transmit lateral forces between the track and roadwheels during steering and side slope operation.

track-laying vehicle. A vehicle that utilizes endless belts or tracks to distribute its gross load over the supporting ground to achieve more uniform ground pressure for improved traction and mobility on adverse soils.

track link. Each of the rigid units that are flexibly connected to form a jointed type track. On double-pin tracks it consists of two track blocks assembled with two track pins. On single-pin tracks it is the track body, with bushings but without pins. Cf. **track body; track shoe assembly.**

track pin. A pin that fits into track links to form the hinge about which flexing occurs in the jointed type track.

track shoe assembly. The assembly consisting of a track link, pins end connectors, center guides and bushings necessary to provide one complete unit of a jointed type track. Cf. **track body; track link.**

tractor. A track-laying vehicle designed to tow by means of a pintle hook or fifth-wheel coupling device.

tractor, cargo. Military track-laying vehicles designed to carry cargo, as well as to perform as a tractor.

trailer. A wheeled or tracked vehicle, nonpowered, with all or most of its weight supported by its own integral wheel or tracks, designed to carry materials, supplies, or equipment and to be towed by a self-propelled motor vehicle. Excludes **semitrailer,** which see.

transmission. A mechanism, included in the power train, the purpose of which is to provide a means of varying the speed ratio between the power source and the tractive elements of the vehicle and also to provide a means of reversing the direction of rotation of the power plant input shaft. Transmissions used in track-laying vehicles perform the stated functions and also permit changing the speed of one track relative to the other track.

transmission system efficiency. The ratio of the power developed at the drive axles or sprockets of a vehicle to the input power at the clutch or equivalent unit. The efficiency may vary with the type of transmission and for a given system may vary with load and vehicle speed.

transport vehicle. Vehicle primarily intended for personnel and cargo carrying. Excludes **combat vehicle,** which see.

transportability. The capability of item of military equipment to be transported efficiently and effectively via railways, highways, waterways, oceans, and airways, either by carrier, by being towed, or by self-propulsion.

truck, automotive. A self-propelled wheeled vehicle designed primarily to transport supplies and/or equipment and which may be used to tow trailers or other mobile equipment. Excludes truck tractor.

truck tractor. A short wheelbased wheeled vehicle designed to tow and partially support a semi-trailer through a fifth-wheel coupling device.

turbosupercharger. A supercharger that is driven by an exhaust-gas turbine. Cf. **supercharger.**

two-stroke-cycle engine. A reciprocating, internal-combustion engine that completes the events of a cycle in two strokes of the piston(s), i.e., one complete revolution of the crankshaft. Each upward stroke of the piston includes a compression event, and each downward stroke includes a combustion (power) event. Arrangements differ, but in general, exhaust valve(s) or port(s) are caused to open near the end of the power stroke, and the intake valve(s) or port(s) admit air or air-fuel mixture under pressure, thus eliminating separate exhaust and intake strokes used in four-stroke-cycle engine(s). Scavenging of exhaust gases and charging with fresh air or mixture, particularly at higher speeds, are accomplished by using a crankcase compression system or an auxiliary blower.

understeering vehicle. A vehicle that inherently tends to increase the radius of curvature as it travels in a curved path and is acted on by centrifugal force. Cf. **oversteering vehicle; neutral steering vehicle.**

unsprung weight. The total weight of all of the vehicle components that are not supported by the vehicle spring system. This includes such items as wheels, tracks, axles, road wheels, etc. Cf. **sprung weight.**

viscosity. In a liquid, the property of internal resistance caused by molecular attraction that makes the liquid resist flow.

viscosity index. A number given to a certain lubricating oil to indicate its performance, particularly as to change of viscosity with temperature variation, as compared with the average, of two groups of test oils.

volute spring. A form of conical compression spring usually made of flat spring stock and wound in a spiral helix with the successive coils telescoping into each other. It is characterized by its compactness, variable spring rate, and high friction damping. It is used as the spring element in certain bogie suspensions of tracked vehicles and as bottoming springs on vehicles with soft suspensions.

wagon steering. Steering of a vehicle consisting of one or more units by a single pivot system with the pivot point located over the front axle.

wheel dance (wheel hop). The vertical vibration of the unsprung mass of a suspension system occurring at the natural frequency of the spring-mass system, consisting of the primary spring elements, the unsprung mass, and the spring characteristics of the tires. The spring rate of the tire is the dominant elastic factor associated with wheel dance. Wheel dance is the principal source of secondary disturbances and vibrations of the sprung mass.

wheel slip angle. The angle between the direction of rolling and the actual direction of travel of a moving wheel under the influence of a side thrust. A tire will develop a cornering force only if it is operating at a slip angle. Cf. **cornering force.**

wheel wobble. *See* **shimmy.**

winterization. The process of converting equipment, especially by changes in accessories, instruments, or special installations, for use in cold or very cold weather, as in the Arctic.

winterization kit, vehicle. A group of items used to prepare a vehicle for efficient operation during cold weather. It contains one or more heater(s) and necessary parts to insulate and/or inclose all or a portion of the engine compartment and/or cab and/or body.

yaw. The angular displacement of a vehicle about an axis parallel to its normal (vertical) axis. Cf. **pitch; roll.**

X-engine. A multicylinder engine with the cylinder banks so arranged around the crankshaft that they resemble the letter 'X' when the engine is viewed from the end.

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