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This report summarizes findings of an electric drive study contract conducted over a 2 year period. The study objective was to obtain and generate data required for evaluating electric drive components and systems in application to military combat vehicle propulsion sytems. Major results were the development of three concepts using AC and DC motor systems.

The primary conclusion of the study was that electric drives can provide improved vehicle performance with respect to conventional mechanical systems.

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1.0. INTRODUCTION

This report, prepared by FMC Corporation for the U.S. Army Tank Automotive Command under Contract DAAE07-84-C-R017, describes Electric Drive Study activities which evaluate the application of current and near term electric drive components to military tracked combat vehicle propulsion systems.

1.1. Background

Electric drive military vehicles have been the subject of previous studies and prototype builds. Among these are:

- M113 Variable Frequency AC Electric Drive Test Bed (1969)
- M35 Electric Wheel Test Bed
- UHS (Ultra High Speed) Electric Vehicle Test Bed

Feasibility of electric drive systems was demonstrated during these early efforts; however, the controller components were inadequate for the electric system to compete with its mechanical counterpart in performance, size, weight, and efficiency. For example, the thyristors used in the controllers for the UHS were able to carry only 63% of the rated current so that motor stall torque was consequently reduced.

1.2. Current Status

Current electrical/electronic technologies and components show significant size and weight reductions and increased performance and efficiency over those integrated into previous electric drive vehicles. A controller today has 10-15 times less volume; an equivalent horsepower motor is 25% smaller. These facts contribute to the potential advantages of electric drives over conventional mechanical drives for military combat vehicles; these advantages are outlined below:

- Flexibility in interior arrangement and drive location due to:
  - electric wire as opposed to shafting and hydraulics
  - modularity of motor, alternator, and control packages
- Ease of auxiliary power integration due to:
  - compatibility with electrical requirements of most auxiliary systems.
  - ability to generate electrical energy in excess of auxiliary requirements
- Improved maneuverability and operation due to:
  - continuously variable ratio characteristics
  - simplified operator controls
  - reverse capability equal to forward

- o Improve signature due to low noise

- o Ease of integration with future technologies (vetronics, robotics, energy weapons due to:
  - electronics controller
  - modularity
  - potential multiple motors
  - ease of configuration of power system to meet specific mission profiles.

The potential gains in vehicle system performance, size weight, efficiency, and design these advanges offer, based on current and near-term technology, motivate the need for this study.

2.0. OBJECTIVES

2.1. Overall Objective

The overall objective was to obtain and generate appropriate data and to evaluate electric drive components and systems as applied to combat vehicles.

2.2. Contract Extension Objective

The second contract extension asked for analysis of key technology parameters and projection of trends to the year 2000.

2.3. Secondary Objectives.

Implied or derived objectives were developed, including the following:

- o Technology Survey. An interim report was written and published in January 1985; results of that report have since been updated and are provided in Appendix A of this report.

- o Concept Generation. This activity provided a variety of electric drive propulsion system concepts. These were fed into the concept screening process to determine the best three.

- o Concept Screen. This was a trade-off among generated concepts including a conventional mechanical transmission baseline.

- o Data Generation. This activity was to develop powertrain data at two levels of detail: a description of all concepts, and a detailed description including performance and operating characteristics of the three best concepts.

3.0. CONCLUSIONS

3.1. The Best Electric Drive Systems

The two best electric drive systems were the High Frequency AC System and the DC Homopolar System. These systems were nearly equal in size, weight, risk, and performance. This is a favorable situation because these alternatives provide fall back positions in the event of an unanticipated problem or unexpected technical advances.

3.2. The Best Configuration

Configuration I uses separate motors for each sprocket to provide both steering and propulsion. It was found to be the best configuration because it provides complete arrangement flexibility, excellent forward efficiency, adequate regeneration for effective steering, and minimum mechanical complexity.

3.3. System Advantages

Both electric approaches were found to be very competitive with the mechanical baselines. Electric drives have advantages in weight, volume, and space utilization, while the mechanical systems have an advantage in technical risk. The mechanical advantages appear to result from the more mature mechanical technology, and therefore we expect electric drives will advance faster than mechanical systems in these areas.

3.4. Additional Electric Drive Advantage

Electric drives have an additional advantage in vehicles that have substantial non-automotive electric power requirements.

3.5. Conclusions of Recent Phase

3.5.1. Earlier conclusions were confirmed.

3.5.2. Breakthroughs are possible in technology areas related to electric drive, but are not necessary to make electric drive systems attractive.

3.5.3. We believe the data indicates a technical revolution is begining in this area, and that electric drives will eventually dominate the field of combat vehicles. These findings lead us to project that electric drives will largely supplant mechanical systems in twenty years.

4.0. RECOMMENDATIONS

4.1. Prototype Development Program

A program should be initiated/continued to develop a prototype electric drive vehicle by about 1990. This would provide hardware confirmation of the claimed advantages of electric drives. Following is a program resulting in a prototype electric drive vehicle. The development process is summarized below by near and far term activities.

4.2. Near term (1987-1988).

- Determine electric drive system (configuration and motor) for detailed development

- Develop propulsion system specification

- Based on system specification
  - Conduct a detailed control system analysis to answer to operability and functional design issues and optimize controller strategy.
  - Refine system efficiency predictions and component performance for update of system specification.
  - Refine cooling design requirements based on motor performance and braking, environmental, and steering considerations.
  - Consolidate analyses for system specification update.

- Conduct peripheral types of trade-offs which impact the usefulness of electric drives in military applications:
  - Assess mission applications of electric drives to determine those with highest payoff. This would include assessing the most appropriate configuration and motor system for various applications. Included here would be an investigation of the impact of future battlefield requirements on electric drive systems; an example is assessing survivability against electromagnetic pulse (EMP), and determining the requirements to protect against EMP. .
  - Assess applicability of various prime power sources for an electric drive propulsion system. Potential sources for trade-off would include reciprocating diesel, rotary diesel, gas turbine, fuel cell, and Stirling engine.

4.3. Far Term (1989-1991).

- Perform detailed concept design which draws on the near term activities described above.

- o Construct a brassboard electric drive system to prove out control strategy and components, electrical components (motor, generator, feedback devices), and mechanical integration. A test program to prove operability in a tracked military combat vehicle environment would be part of the brassboard phase.

- o Construct a vehicle level prototpe to prove feasibility in terms of installation and operation in a tracked combat vehicle.

4.4. Recommendations from Recent Phase

4.4.1. FMC is pursuing IR&D programs in controller design and prototype development. Given adequate govenment support, a prototype electric drive vehicle should be operational by 1989.

4.4.2. We recommend conducting a thorough cost analysis of electric drive vehicles. This analysis should include acquisition, ownership, and life cycle costs.

4.4.3. We recommend development of a technical design integration package for electric drives. This would allow comparison of conceptual electrical and mechanical system vehicles.

4.4.4. We recommend initial development of the High Frequency AC electric drive system. Although the AC system and the DC Homopolar system scored very close in our evaluation, initial development of the AC system is preferable due to the evolution required in brush technology for the DC system.

5.0. DISCUSSION

5.1. Electric Drive Tutorial

The purpose of this section is to provide a brief comparison of electric and mechanical drives and to introduce electric operation and features.

5.1.1. Electric drive systems are among the earliest components used for automotive propulsion systems. During the early 1900s, a family of electric cars were developed both in the United States and Europe. The Baker Electric (1908) was one of the most successful and was known for its reliability and quiet operation. The car's primary limitation was lack of range due to the limited amount of energy stored in the car's battery system. Within a few years, the internal combustion (IC) engine appeared and quickly displaced the electric car, due to the ease and speed with which the car could be refueled. Higher vehicle speed and longer range were immediately available to meet the user needs and thus the IC engine became the standard for all automotive applications.

5.1.2. Electric drive systems have continued to be examined for automotive propulsion for specific applications. Among the major high power automotive systems in wide spread use at present are many types of large earth moving trucks which use a diesel engine/generator/motor drive train to replace a more conventional hydraulic torque converter/geared transmission. The railroad industry has also adopted electric drives (diesel engine/generator/ motor) for most rail propulsion systems.

5.1.3. System Comparisons. A comparison of the drive line for conventional and electric combat vehicle drive is shown in Figures 5-1. and 5-2. These figures depict only the components directly in the power path and do not show the peripheral components necessary to complete the system such as cooling or braking. Since the purpose of this study was to select and evaluate a component system which could replace a conventional hydrokinetic or hydromechanical transmission, the primary task was to evaluate candidate electric motors which could meet the output speed and torque requirements. The comparison of conventional and electric systems focused on physical characteristics such as space claim, volume, weight and performance required to meet vehicle speed and gradability requirement. A general discussion of these factors for conventional and electric transmission follows.

5.1.4. Conventional Drive Systems. Present mechanical transmissions use an integrated system of gearing, torque converters, clutches, and brakes to achieve multispeed capability and associated torque changes. In currently used transmissions, the power flow is a combination of power splits whereby the output power is delivered by the engine, gearing, and torque converters through appropriate lock-up clutches and brakes to provide for direct engine drive when performance allows for relatively constant vehicle speed. In some designs, the final gear

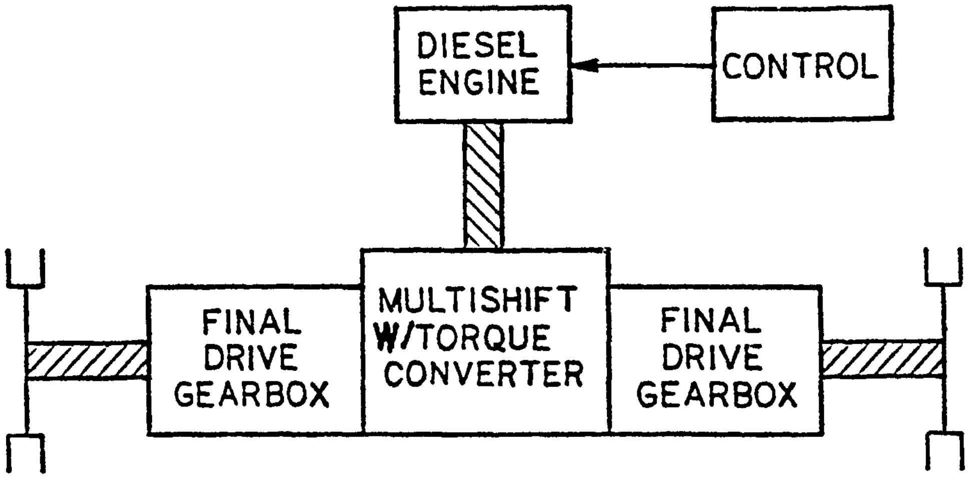


Figure 5-1. Conventional Combat Vehicle Drive.

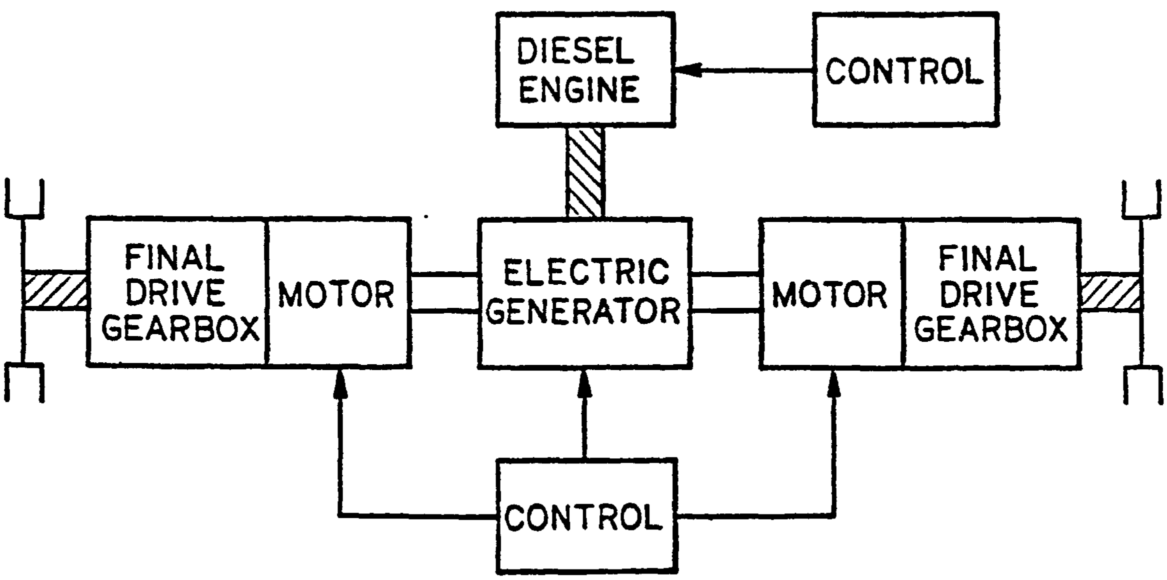


Figure 5-2. Electric Combat Vehicle Drive

reduction to the output sprocket is integrated with the transmission which allows for a common oil sump. This oil also provides lubrication and thermal control for the vehicle gears, brakes, and clutches.

5.1.4.1. Primary control for the conventional system is via the fuel flow valve. This is adjusted by the operator to produce the engine speed and torque required to satisfy the load requirements. Conventional systems respond to variations in engine speed and produce torque proportional to that speed. This interrelationship of speed/torque requires that the conventional transmission be able to sense any changes in load and adjust the ratio of input torque to output torque in a continuous manner so the vehicle performs the intended mission. In practice however, the continuous variability cannot be fully realized due to the finite ratios of the gear shift points. This point is shown by Figure 5-3. which compares drive system horsepower as a func-tion of vehicle speed. The fact that the mechanical trans-mission cannot optimally match the required load speed/torque pulls the engine away from its best load (speed/torque) point and thus detracts from the maximum available engine horsepower available. This figure shows the effect of the speed/torque mismatch on both engine developed horsepower (top curve) and output power available to the track (bottom curve).

5.1.4.2. Recognition of this fact is a primary point of system comparison when the attributes for electric drives are considered. Figure 5-3. also shows the power available to the electric drive (dotted lines) and the power delivered to the tracks. In this case, the power is smoothly delivered over the specified performance speed range. This difference in power availability has a direct impact on system performance, efficiency, and fuel economy.

5.1.5. Electric Drive Systems. Electric drive system components have matured in the past 25 years to a level which justifies their consideration as an alternative to the mechanical transmission. Although no clear choice is evident in selection of alternating or direct current for the primary power systems, it is evident that electric drive components can be selected for either system which yield equivalent performance for the vehicle application. Each type of power system has certain operating characteristics and may be configured in a variety of ways. Figure 5-4. is arranged as a two motor system (Configuration I). This configuration allows maximum utility of the drive motors since each is individually controlled for speed and torque. Primary power comes from the engine driven generator and is applied to the drive motors in a regulated manner determined by the controls. It is the method of controlling the power to the electric drive which distinguishes AC and DC electric drives.

5.1.5.1. AC Drive Systems. AC drive systems use either synchronous or induction motors as the primary component in the drive system. These motors respond to a change in frequency of the applied power to produce a change in speed of the motor. Figure 5-4. shows a typical AC drive

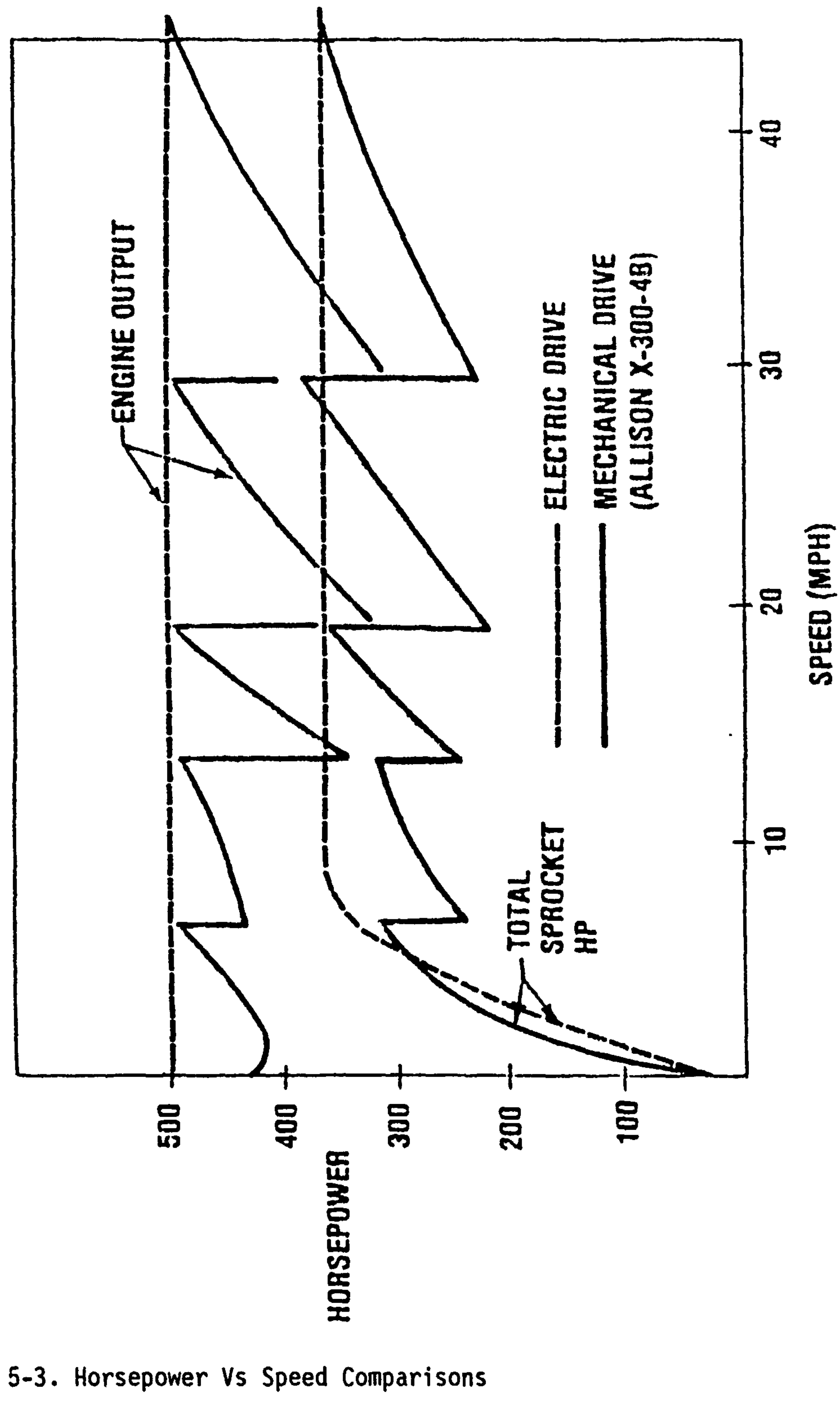


Figure 5-3. Horsepower Vs Speed Comparisons

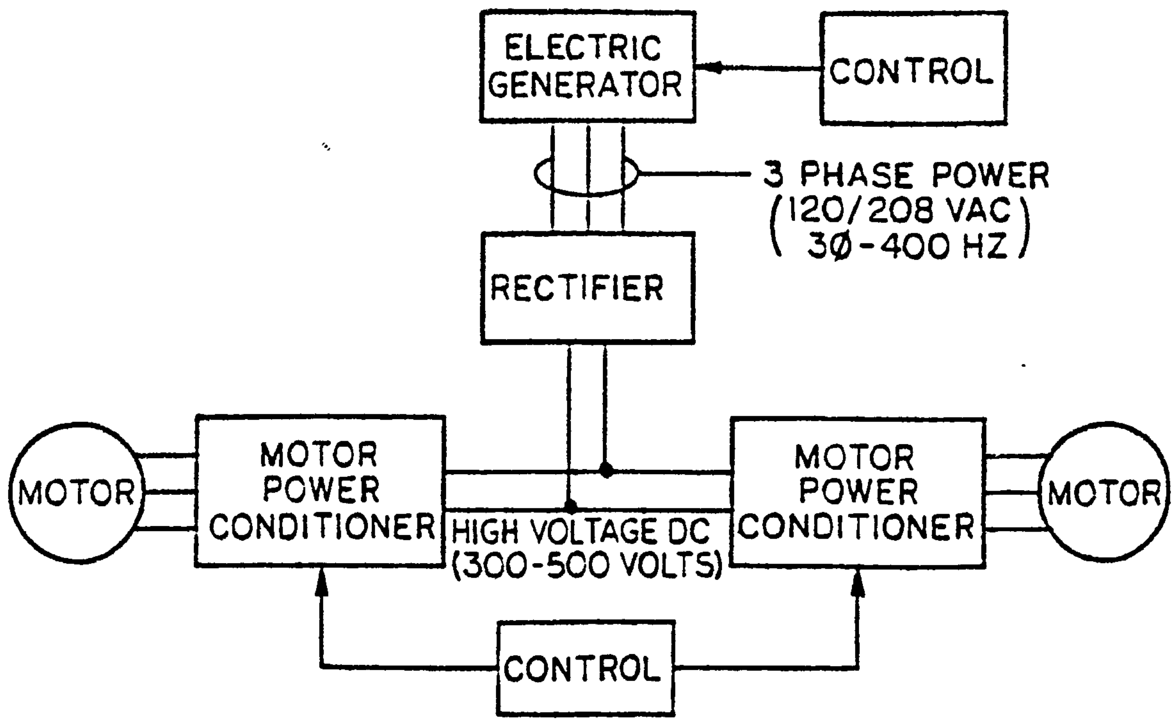


Figure 5-4. Typical AC Drive System

system usable with an induction motor. The controlling component in an AC drive is the motor power conditioner. This component changes the high voltage DC power to controlled AC power of selected voltage and frequency which is applied to the drive motors. Since the vehicle gains stored energy as it is accelerated, the motor power conditioner must also handle the acceleration power as well as the power regenerated to the electrical system when the vehicle is decelerated. It is this fact which is compared to the DC system.

5.1.5.1. DC Drive Systems. DC drive systems use either conventional, commutated brush motors or homopolar motors as the primary element in the drive system. These motors respond to a change in system voltage to produce a change in motor speed. Figure 5-5. shows a typical DC drive system usable with a homopolar motor. In this system, the motors and generator are directly connected together without the requirement for a series power conditioner for each motor. Thus, power can be both delivered to the motors or accepted from the motors (in the case of regeneration). DC machines, such as the homopolar can function as either motors or generator. This characteristic is desirable in that a simple control system can be used to command the drive system motors to perform independently as either motors or generators. This bilateral power flow is required when the vehicle executes coordinated turns or dynamic braking.

5.1.5.3. Electric Drive Control. AC system control is accomplished in a DC/AC inverter system. This inverter is generally a three phase configuration and consists of at least six power semiconductors per phase. Each power semiconductor is shunted by a power rectifier which provides a path for reverse power flow. Figure 5-6. shows a typical three phase inverter circuit having the six power semiconductors (Q1--Q6) and six reverse power rectifiers (D1--D6). By appropriately controlling the power semiconductors Q1--Q6, a quasi-sine wave of current is generated in the motor windings, thus producing a rotating torque. Motor reversing is caused by changing the switching sequence of the power semiconductors.

5.1.5.4. For the DC system control, a simple "H" switch circuit can be used and is shown in Figure 5-7. In this scheme, the power semiconductors are controlled to determine the direction of current in the exciter field. As shown, if Q1--Q3 are operated, the field will be excited from left to right. This would be the forward direction of travel. To reverse, power semiconductors Q2--Q4 would be operated, thus exciting the field from right to left. As shown in Figure 5-7., the control for the DC system is in parallel with the main power circuit. Thus, the control/ power conditioner does not have as great an impact on system reliability as in the AC system case where the motor power conditioners are in series with the propulsion motors.

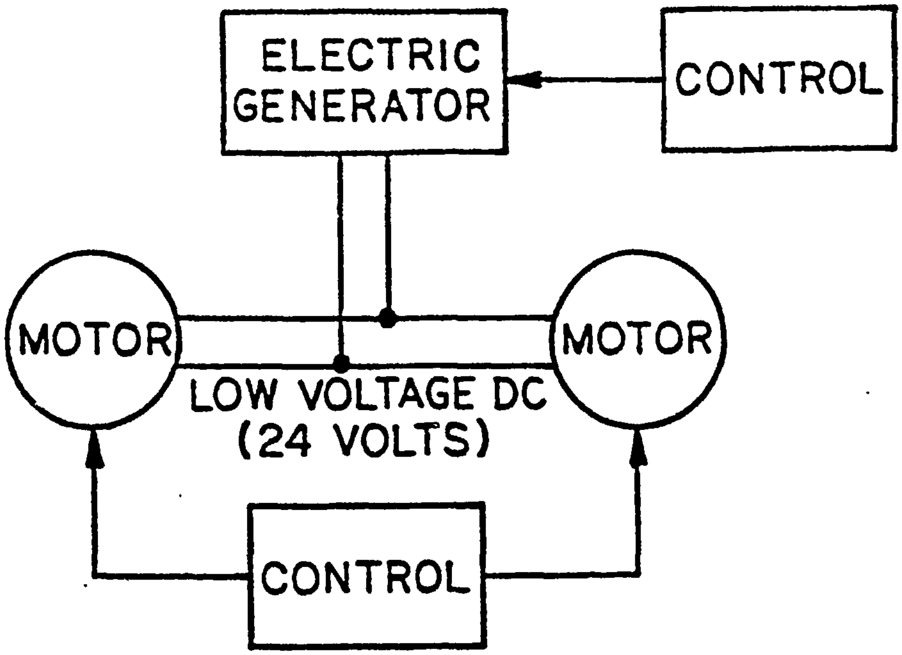


Figure 5-5. Typical DC Drive System

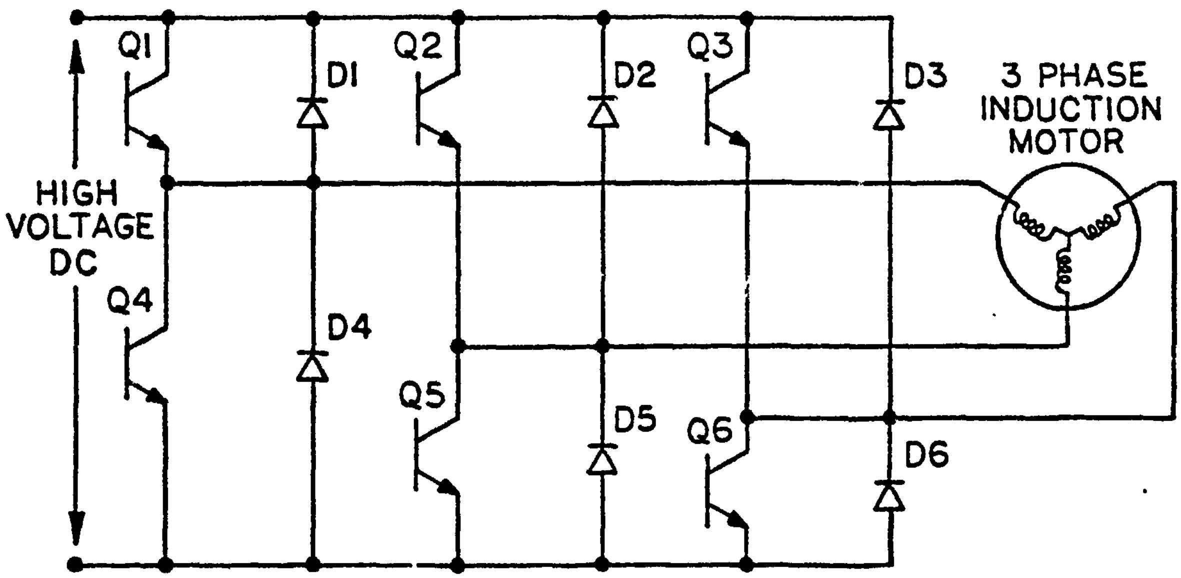


Figure 5-6. Typical Three-Phase DC/AC Inverter

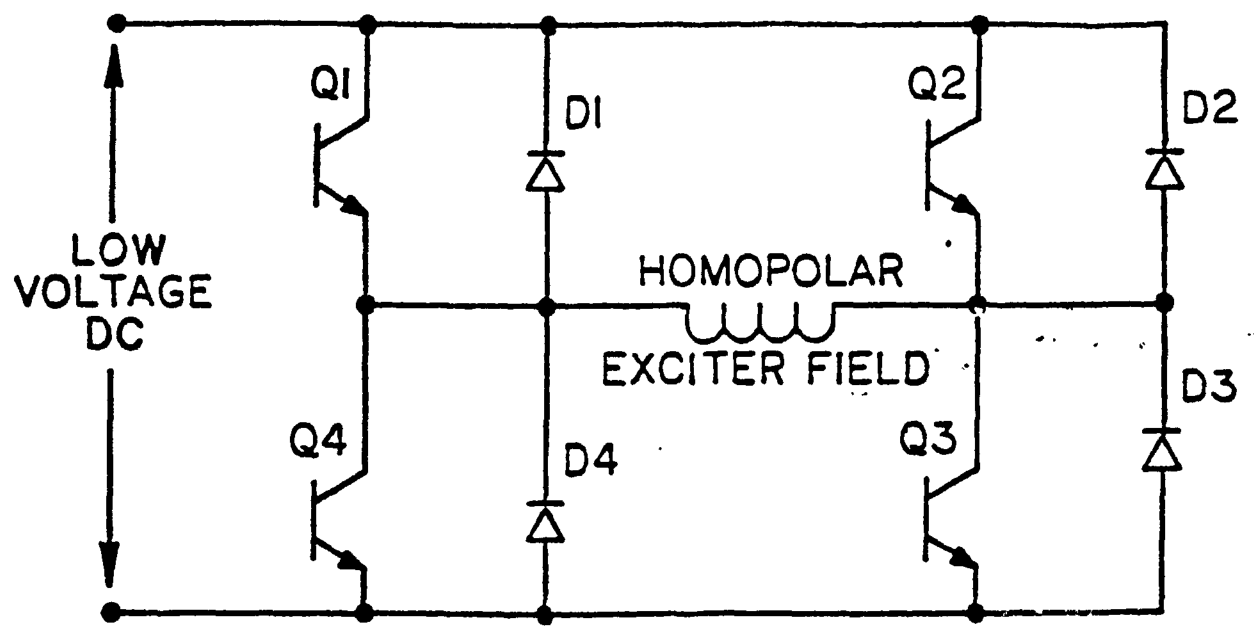


Figure 5-7. Typical Exciter Field Control

5.1.5.5 A major difference in controls strategy for the AC and DC drive systems results from consideration of the motor gain ratio. DC systems and some AC systems (e.g., synchronous) exhibit a characteristic where the total power of the motor may be controlled by a relatively small percentage of the total power. Motor gain ratio is defined as the ratio of power developed by the motor to power required to fully control the motor. In the case of the AC motor systems which operate from a DC voltage, a power inverter is required to handle the full power of the motor. In this example, the motor gain ratio would be effectively 1.0. In the DC drive system, the motor control is in parallel to the motor and handles only a percentage of the motor power. For a homopolar motor, the motor gain ratio can range to 20 to 40. Thus the power control can be reduced in weight and volume have reduced thermal loss, and contribute to improved system efficiency.

5.1.6. System Attributes. Conventional mechanical transmissions and the electric drive alternatives are compared at Table 5-1. in terms of attributes derived from design requirements or intended mission. Selected attributes are discussed in more detail in the paragraphs below.

5.1.6.1 Performance. Performance related characteristics of conventional mechanical transmissions and alternative electric drives are similar. The most prominent benefit in performance is the improvement in the power availability ratio inherent in the electric drive. Power density of the electric drive is improved over conventional mechanical transmissions due to significant weight and volume reduction. Drive efficiency is improved since parasitic losses are reduced. Electrical components have high overall efficiency and may be interconnected without severe penalty on the system efficiency.

5.1.6.2. Electric drive systems have an advantage over conventional mechanical transmission systems due to the independent control of speed and torque. In mechanical transmissions, torque is produced as a function of speed of the engine and selected gear ratios. This limits the availability of high torque at very low speeds approaching stall. Electric drives however, can produce full torque over the entire speed range of the selected motors. This fact improves vehicle performance in acceleration, gradability, and braking. The ability of the electric drive to absorb and dissipate large amounts of power is a direct benefit for vehicle control. The electric drive acts as a dynamic brake and can effectively operate from full vehicle speed down approaching zero speed. This fact reduces the need for full power service brakes and their associated heat generation and parasitic losses.

5.1.6.3. Vehicle Integration. Electric drive components offer flexibility in installation when compared to conventional mechanical transmissions. Due to power transmission via wire and cables, the electrical components can be placed in areas of the vehicle where space is available rather than being confined to the engine compartment.

Table 5-1. Comparable System Attributes

SYSTEM ATTRIBUTES	MECHANICAL DRIVE	ELECTRICAL DRIVE
PERFORMANCE	EXCELLENT	EXCELLENT
POWER AVAILABILITY	GOOD	BETTER
POWER DENSITY	MODERATE	GOOD
EFFICIENCY	GOOD	BETTER
WEIGHT/VOLUME	HIGH	MODERATE
FLEXIBILITY OF INSTALLATION	POOR	GOOD
MAINTENANCE	HIGH	LOW
STORABILITY	POOR	GOOD
COST OF OWNERSHIP	MODERATE	LOW
DEVELOPMENT RISK	LOW	MODERATE
SUPPLY EXTERNAL POWER	MINIMAL	YES
USE WITH ADVANCED WEAPONS (KEW, ETC)	NO	YES

5.1.6.4. Conventional mechanical transmissions are not readily interfaced to computer monitoring or electronic data bus systems. Although many present conventional transmissions have speed monitoring tachometers for gear range selection, they cannot be easily adjusted for control of speed and torque. Many vehicle missions can be improved in efficiency by having the ability to control vehicle speed and torque independently.

5.1.6.5. Electric drive systems can be configured to have independent control of motor speed and torque. The use of independent sensing components for voltage and current sets the levels of speed and torque developed by the drive motors when operated or controlled by a closed-loop regulating system. In this configuration, the operator contol loop can adjust the levels of the voltage and current monitoring system and allow the closed loop servo system error to cause the drive motors to respond to the set conditions. Computer monitoring systems can perform these functions easily.

5.1.6.6. Closed-loop, computer controlled command/monitoring functions can offer significant benefit to the operator/crew in advance vehicle systems. Electric drive systems, due to their technology compatibility, offer ease in integration of the computer control/command system.

5.1.7. Cost Factors. Electric drives have benefits in areas where high labor content directly affects cost of ownership. Areas such as maintenance and upkeep costs are reduced with electric drives due to reduction in wear-out parts and reduced labor content in all replacement parts. Electric drive vehicles may be stored for relatively long periods without system degradation. Risk associated with development is greater with electric vehicles primarily in the controls area than in conventional transmission vehicles. However, the risk is balanced by the ease of correction in electronic systems should it be necessary.

5.1.8. Electric System Special Applications. As vehicle utilization becomes a concern factor in tactical applications, increasing vehicle capabilities becomes necessary. Advanced, all-electric weapon systems with features such as kinetic energy weapon systems, active armor systems, and improved navigation and fire control systems all require increased amounts of electric power. This additional power would be readily available in an electric drive vehicle.

5.1.9. Electric drive technology is easily integrated to meet the needs of external power requirements. New systems are expected to be capable of supplying both AC and DC power for specific needs. Generally, high frequency AC systems are necessary for navigation and stabilization systems and can supply large subsystems such as fire control and range computers. DC power systems are better in the areas of kinetic weapons, active armor, and propulsion.

5.2. Program Overview.

5.2.1. Contract Performance Requirements. The following items are contract performance requirements that served as propulsion system evaluation criteria for the various drive systems (results are detailed in numbered paragraphs in this report):

- o Tractive effort vs speed (5.11.1., 5.11.3.)
- o Acceleration (5.11.4.)
- o Speed on grade (5.11.5.)
- o 60% grade startability (5.11.6.)
- o 7 RPM axis steer (5.11.7.)
- o Minimum turn radius vs. speed (5.11.8.)
- o Braking capability (5.11.9.)
  - Panic stop
  - 60% grade hold
  - 25 cycle test
  - Continuous 20 MPH down 15% grade

Each contract performance requirement was evaluated to ensure the recommended electric drives would provide satisfactory vehicle performance. A detailed analysis of each requirement was made using the methods detailed in Appendix B; results are summarized in the referenced paragraphs. Most requirements are fully met, some are exceeded and a few are only approached. The recommended electric drives generally outperform the mechanical baseline comparison transmissions.

5.2.2. Technology Survey. Early in the study, a technology survey was performed to examine current and near term electrical components and technologies with potential applications for electric drive systems. Figure 5-8. is a technology tree, showing the possible approaches to electric motors. The technology survey addressed both AC and DC systems including control schemes. The components and technologies were evaluated on the following:

- o Performance specifications
- o Size and weight
- o State of development
- o Technology selection criteria.

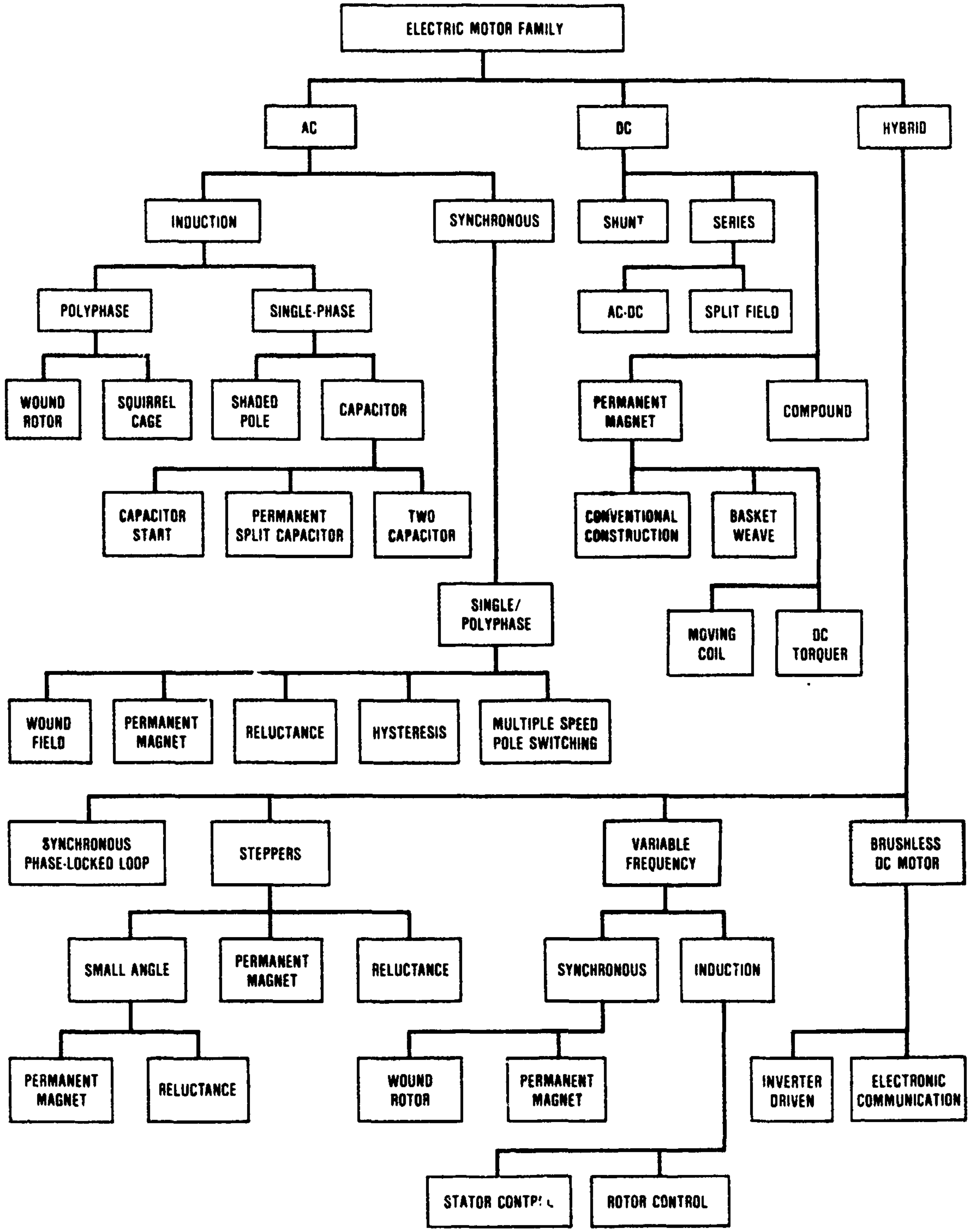


Figure 5-8. Technology Tree.

In addition, pictorial/schematic depictions of components along with a narrative on principles of operation were to be provided as a result of the technology survey. The technology survey was documented in a report approved by TACOM in January 1985. The updated survey is included herein as Appendix A. The technologies selected were used for the concept generation phase of the study.

5.2.3. Concept Descriptions. The purpose of the concept generation phase of the study was to generate electric drive propulsion concepts to attain an optimum system. The concepts generated were based on the technologies examined and selected during the technology survey. The following four configurations were specified for analysis as a minimum in the contract.

5.2.3.1. Configuration I consists of an engine driven alternator and individual motors driving directly or through gear boxes at each sprocket (Figure 5-9). Steering is accomplished by controlling the speed of the individual motors.

5.2.3.2. Configuration II, illustrated in Figure 5-10., consists of an engine driven alternator, with a single motor driving a propulsion cross-shaft and a single motor driving a steering cross-shaft.

5.2.3.3. Configuration III, Figure 5-11., consists of an engine driven alternator, with individual motors at each sprocket for propulsion and a single motor driving a cross-shaft for steering.

5.2.3.4. Configuration IV, Figure 5-12., consists of an electro-mechanical dual power path drive system analogous to a hydromechanical such as the CVX-650 transmission. An alternator and motor combination replaces the hydrostatic pumps and motors of the CVX-650 transmission. The concepts were refined as they were generated and then were evaluated as part of the concept screening phase of the study.

5.2.4. Concept Screening. The concepts developed during the concept generation phase were to be analysed and compared against one another using contract specified criteria. The three best concepts were derived from that concept screening and then compared with the mechanical baseline concepts for the 19.5 and 40.0 ton weight classifications.

5.2.4.1. The concept screening criteria are presented below in descending order of priority:

- o Performance
- o Total system volume and space utilization
- o Technical risk
- o Weight

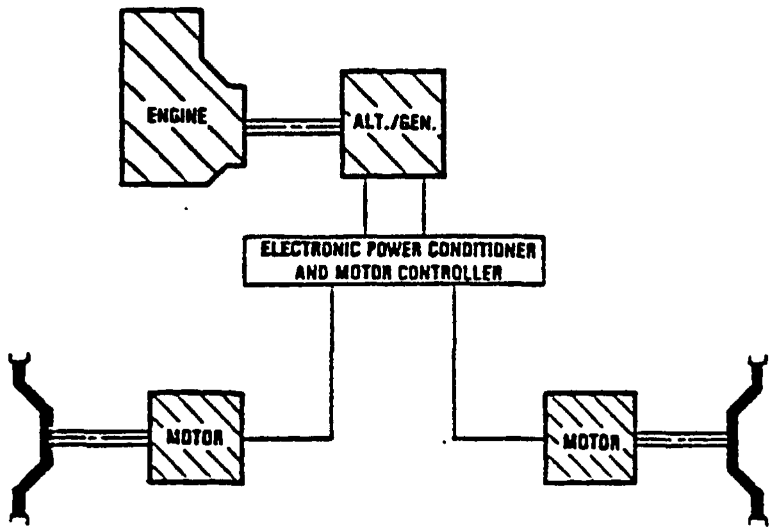


Figure 5-9. Configuration I

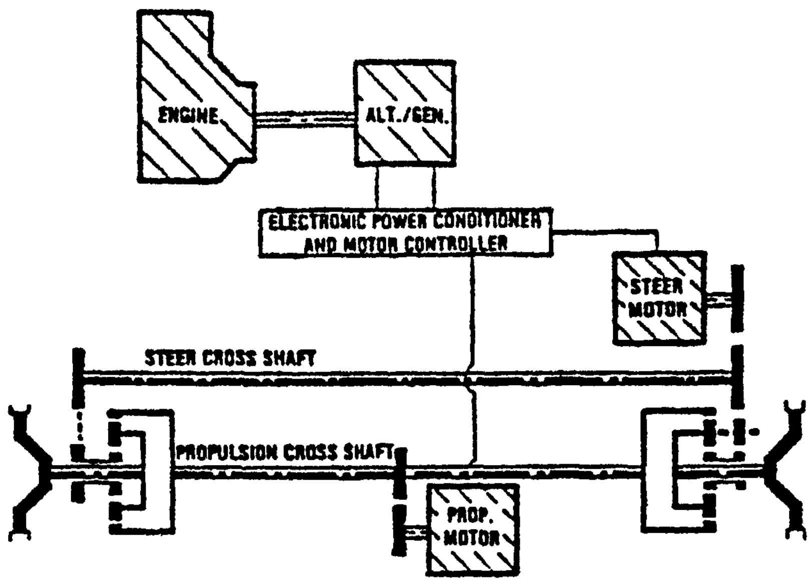


Figure 5-10. Configuration II

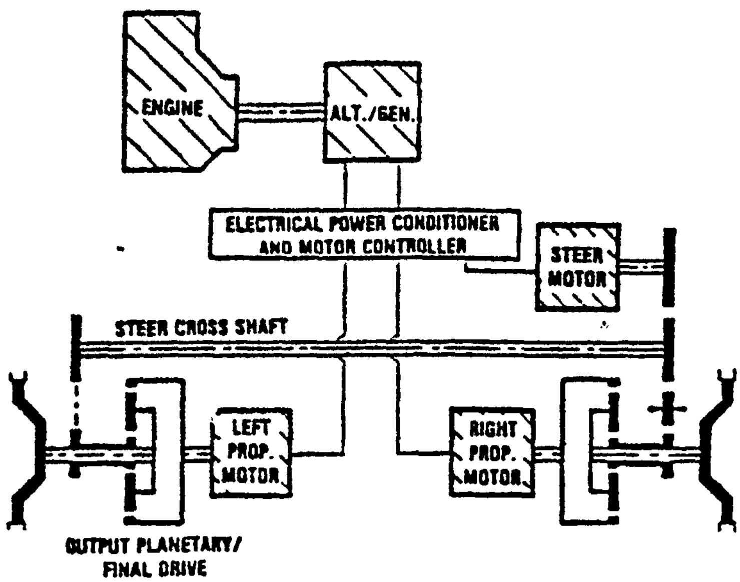


Figure 5-11. Configuration III

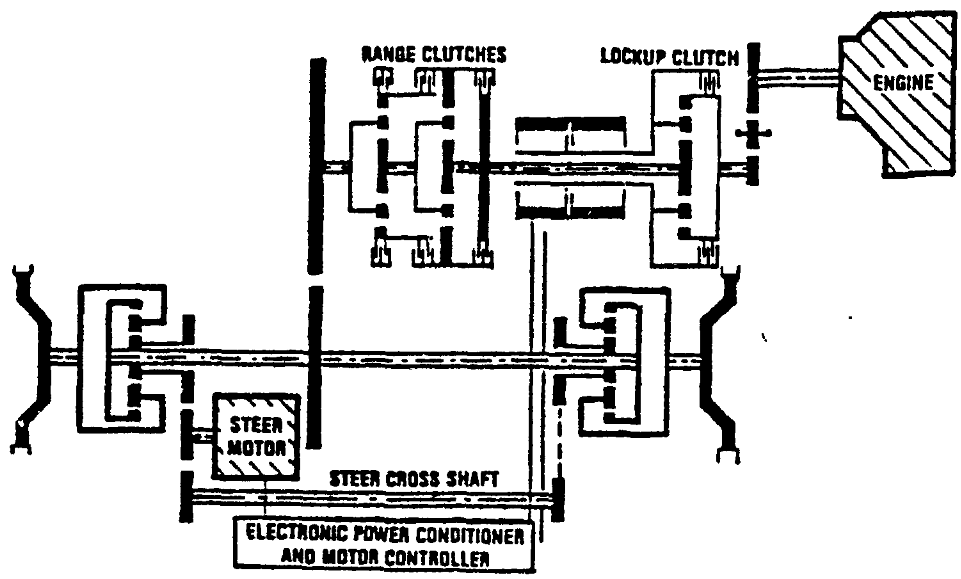


Figure 5-12. Configuration IV

- o Reliability and maintainability
- o Safety.

5.2.4.2. Other concepts were also reviewed. Any concept which failed to meet any of the above criteria was eliminated from the analysis. The best three electric drive concepts were compared with the mechanical baseline drive concept. The concept screening was the basis for the study conclusions and the future recommendations.

5.3. Concept Development

The Electric Drive concepts were generated using technologies that were selected in the technology survey. The methodology for the development of concepts involved a technology survey, a concept generation and refinement phase, and a concept screening phase.

The technology survey resulted in the selection of the best suited motor types for vehicle drive applications. The motors selected were then developed into vehicle drive concepts based on the viable contract specified configurations. Of the four configurations provided in the contract, Configurations I and II were found to be viable for vehicle drive systems. The concepts are complete drive systems, including cooling, gearing, and controllers.

The concepts were evaluated on a computer model to determine, with a high degree of confidence, the concept's performance characteristics. Refinement of the concepts continued in order to optimize the performance of the system. If a concept was unable to meet the contract specified performance levels, alternate concepts were generated based on the motor type. Continued failure to perform within the minimum requirements (i.e. the mechanical baseline characteristics) resulted in the non-pursuit of a concept and/or motor type for a configuration.

5.3.1. Concept Development And Descriptions.

5.3.1.1. Concepts fell into two distinct categories with Configuration I offering greatest design freedom and Configurations II, III, & IV constrained by the required mechanical interconnections. Evaluation of the electric drive systems found through the technology survey narrowed the field to two leading contenders, the AC Induction Motor system and the DC Homopolar system. Since size, weight, speeds and efficiencies were comparable, either system could be used interchangable in most concepts. For specific missions availability of a particular electrical source, high voltage AC from the induction system or high current DC from the homopolar system could be a deciding factor. Within the 19.5 ton to 40 ton vehicle size range, components for either system can be produced with current technology. More mature motor technology gave the induction system a slight edge in the final analysis, but it was considered a fortuitous circumstance to have two viable systems to assure a "fall back" position if unforseen problems eventually were to preclude use of one of the systems.

5.3.1.2. Evaluation of the contract configurations I, II, III and IV led to the selection of configuration I as the system that fullfilled the arrangement flexibility potential of electric drive. The high regeneration efficiency of configuration II provided sharper steering in the 3 to 7 mile per hour range, therefore this configuration should be considered in cases where the arrangement constraint of the steering cross shaft does not compromise vehicle system arrangement. Configuration III provides less efficient regeneration than Configuration II, yet had the same arrangement constraint due to the steering cross shaft. Configuration IV was eliminated due to the excessive mechanical and electrical control complexity.

5.3.2. Subsystem Design Optimization

Subsystems were designed for optimum performance within constraints of reasonable size and weight using design criteria that has evolved from extensive FMC experience with military combat vehicles. To produce prompt and effective results from a system integration study, like the concept development effort, it is essential to start with realistic initial sizing for the components of the many subsystems. FMC starts such efforts either by direct comparison with existing vehicles or by application of appropriate basic parameters. As an example, a preliminary radiator sizing would be made either by correcting the size of the existing radiator for changes in the engine horsepower and the heat rejection or it would be sized by applying curves of frontal area vs (fan HP/engine HP). When the concept has evolved to sufficient detail, a final cooling check is made with a computer model. Since the initial sizing methods generally produce good size approximations, only minor adjustments are required to achieve final optimization and the system trade-offs are not upset by the need for major redesign.

5.3.3. Cooling System Integration.

The integrated cooling system contains five basic cooling subsystems as follows:

- o Electronic controls loop (150 degrees F maximum)
- o Engine cooling loop (230 degrees F maximum)
- o Electric unit cooling loop (275 degrees F maximum)
- o Mechanical braking loop (325 degrees F maximum)
- o Dynamic brake grid (1200 degrees F maximum)

5.3.4. Cooling System Discussion.

5.3.4.1. In an environment where ambient temperature plus solar radiation can easily reach 135 degrees F, maintaining the 150 degree coolant temperature for the electronic controls is critical. To achieve this goal, cooling air is drawn through the electronic controls radiator immediately after entering the vehicle through the ballistic grills. Circulation rate of the turbine oil coolant will be maintained to limit electronic controls temperature differential to 5 degrees F and a margin of 15 degrees F with a maximum coolant temperature of 150 degrees F.

5.3.4.2. Air from the electronic cooling loop radiator passes directly to the electric component cooling loop radiator. Although the temperature gradients indicate the engine coolant loop should be next in the air circuit, the electric components loop radiator is placed in the more favorable position in the air circuit. This is because the turbine oil coolant used in this circuit is much more viscous than water. Even with turbulators in the radiator, it is difficult to achieve efficient heat transfer at the oil to metal interface for this circuit. Since the total heat rejection of the electronic and electrical cooling loops is relatively small, heat rise in the air circuit through these radiators will be relatively low, and air should be discharged from them at a still usable temperature in the order of 160 degrees F.

5.3.4.3. This air next enters the engine cooling radiator. The electronics cooling radiator, the electrical components cooling radiator and the engine cooling radiator are designed with the same core frontal dimensions so they can be mounted together in a single subassembly. Optimization of this system produced an engine cooling radiator of relatively small frontal area but quite thick (in the order of 7 1/2 inches). Such thickness produces substantial head loss in the air circuit, but in a circuit with the restrictive ballistic grills, the increase in fan requirements and resulting fan horsepower is only a small fraction of the total requirement and therefore acceptable.

5.3.4.4. The subassembly of the three radiators will be directly mounted to the engine/generator package. Self-sealing hydraulic disconnects will be provided in the electronics and electrical cooling loops. Since the engine/generator and engine coolant radiator are parts of the same subassembly, it will not be necessary to disconnect engine coolant lines before the power pack is pulled for service.

5.3.4.5. Air will next pass through the engine compartment where it will directly cool such items as hydraulic pumps, transfer cases, and auxiliary generator before it enters the cooling fan. The air will enter the fan at a temperature exceeding 200 degrees F when operating at maximum load and ambient conditions. Fan design corrections must be made to provide proper performance with this low density, expanded air. This will have negligible impact on fan horsepower, but it tends to drive fan sizes larger, and increase tip speeds and the resulting noise signature from the fan. Variable speed fan drives are used in all cases. Air circulation and resulting substantial parasitic load will be minimized when not required for full load maximum ambient conditions. Use of the series air flow circuit will necessitate multiple temperature sensors to assure adequate air flow for the critical cooling loop for various operation conditions.

5.3.4.6. In the case of dry disk brakes, the units have been sized to permit direct heat rejection to the engine compartment air. In the case of the wet disk brake system, the brakes will be cooled by the final drive lubricant (SAE #30W viscosity engine oil). This arrangement precludes contamination of the electrical and electronic loops with brake wear products. Extensive experience with mechanical drives shows that these wear products can be accomodated by a properly filtered lubricant circuit. Since the final drive gearing is highly efficient, combining its losses with the brake thermal loads produces a negligible increase in cooling requirements, but substantially simplifies the resultant cooling circuit design and sealing problems. The coolant/lubricant will be cooled by an oil to water heat exchanger that will reject the heat to the engine coolant circuit. This arrangement has the incidental advantage during operation in extremely cold environments of supplying engine heat to the final drive/brake assemblies to lower viscosity and improve lubrication while reducing losses.

5.3.4.7. The fan will discharge directly into the dynamic braking cooling grid. Even though the entering air temperature will at times exceed 200 degrees F, adequate temperature will exist because the grid can operate at very high temperatures. The size of the grid will be primarily determined by thermal signature limitations, as there are direct correlations between surface areas and operating temperatures.

5.3.5. Vehicle Electrical Systems

5.3.5.1. The vehicle contains two basic electric systems, the propulsions system and the auxiliary system. The auxiliary system is essentially similar to the system used in the conventional mechanically driven vehicle. Six maintenance free batteries provide electric power to the conventional engine starter. Power for lights, communication, navigation, and auxiliary drives (like turret drives or bilge pumps) are provided by the battery circuit. Batteries are charged by means of a conventional auxiliary generator in the case of DC homopolar drive systems or by rectified DC from the power conditioning system of the AC induction motor systems.

5.3.5.2. The components of the electrical propulsion systems could be located with substantially more freedom that comparable mechanical components. It was this freedom of arrangement to facilitate optimum new configurations for the electric drive vehicle concepts. This weight, size, speeds and other installation constraint data was obtained from the technology survey and from FMC design efforts. The induction motor configuration I provided the greatest degree of integration freedom. The other recommended systems were more constrained by installation requirements, but all were more flexible than the conventional mechanical drive.

5.3.5.3. The induction motor configuration I system had essentially unlimited installation flexibility because relatively low currents precluded excessive power losses, even when leads were long to reach remote components. Induction motor configuration II systems were constrained by the steering cross-shaft. Remote location of engine and drive sprockets was still feasible, but the steering cross-shaft conflicted with optimum engine location if engine and drive sprockets were at the same end of the vehicle. This cross-shaft also was a potential obstacle for troop carriers with a front engine, rear drive (for better balance) and a rear ramp for troops. The homopolar configuration I was more constrained than the induction motor concept because the high currents required heavy interunit conductors, limiting feasibility of placing engine and drive sprockets at opposite ends of the vehicle without encountering substantial power losses. In all systems, arrangement of the engine compartment was facilitated by flexibility of engine location. Engine crankshaft centerlines were not constrained to specific waterlines or alignment with mechanical transmission inputs. Engines could be placed in lengthwise, transverse or angular positions if desired to optimize packaging and access for maintenance.

5.3.6. Brake Systems

5.3.6.1. Two types of mechanical brake systems are used in the vehicle concepts. The induction motor systems use a dry ventilated disk and brake caliper on the end of the motorshafts. The homopolar systems use a wet multiple disk brake system that is built into the final drive.

5.3.6.2. The electric drive systems provide a regenerative braking capability that minimizes the loads imposed on the mecahnical braking system. The primary functions of the mechanical brakes are:

- o Provide a redundant emergency brake system.
- o Hold the vehicle on a 60% grade.
- o Supplement the dynamic brake capability at speeds below 5 MPH when dynamic capacity is limited.

5.3.6.3. These requirements are met with a relatively small brake because frequent usage is not anticipated for the emergency brake and energy inputs are very low at speeds below 5 MPH. The use of the dry disk brake on the induction motor is the preferred arrangent because it facilitates maintenance and permits direct air cooling. The close coupled arrangement of the homopolar motor/generator assemblies precludes mounting a dry disk on the motorshaft. Oil cooled multiple disk brakes are therefore used in an arrangement long proven in mechanical drives, but of smaller size because of the lower energy input.

5.3.6.4. The brakes are normally actuated by a hydraulic servo mechanism that is controlled by the main control microprocessor. This max imizes application of the electrodynamic braking system for best control and minimum maintenance. The mechanical brakes are actuated only at the below 5 MPH speeds where the dynamic brakes are normally inadequate. An independent mechanical actuation system is provided for emergency and parking functions.

5.3.7. Fuel Tanks

5.3.7.1. Fuel tanks are shown in all vehicles to achieve a 300 mile range on level, hard surfaced roadways. Initial analysis was based on the mechanical transmission because test data for many similar vehicles is available to support the accuracy of the prediction. Performance analysis of the vehicles showed comparable efficiencies for the electric drives and the mechanical transmissions, therefore the same amount of fuel should be required. All 19.5 ton and all 40 ton vehicles were provided with the same size fuel tanks.

5.3.7.2. In most 19.5 ton configurations, the fuel tanks are on the rear sponsons and in most 40 ton concepts the fuel tanks is mounted transversely across the hull in front of the engine, with auxiliary tanks on the sponsons. Appropriate bulkheads would be provided to isolate fuel and resulting fire hazard from the troops and engine compartment. Final optimization of the fuel tanks will depend on the mission of the chassis. The size, shape and location of individual tanks is quite flexible so that should be located to take full advantage of ballistic protection provided by the rest of the components.

5.3.8. Exhaust System

The exhaust system is designed to optimize vehicular performance by minimizing backpressure. This is done by short direct runs of the piping and by eliminating the muffler. Since both engines are turbocharged, it is anticipated that the turbochargers will provide adequate silencing for combat situations. Systems were arranged to minimize the possibility of recirculation of exhaust gasses through the engine cooling air circuit.

5.3.9. Vehicle Designs And Gearing

5.3.9.1. Experience has shown that torque at sprocket is based on tractive effort that can be supported between the track and ground, and not by the available input power. It is also known that allowable design values are the result of successful applications from which ground rules are established.

5.3.9.2. The current contract specifies tractive effort to be .7 GVW at each sprocket, which is a reasonable number used when applied with the following values:

- "K" Factor = 5000 (an index of the intensity of gear tooth load from the standpoint of surface durability) 17A
- Tooth unit load = 56,500 PSI (wt. Pd/F)
- Gear tooth bending stress = 120,000 PSI

These were the values used to establish gear drive component sizes.

5.4. Power Control and Conversion

5.4.1. In an electrically driven vehicle propulsion system there are four modes of internal energy conversion which are of potential interest. These include:

o Mechanical Energy to Alternating Current	ME	AC
o Mechanical Energy to Direct Current	ME	DC
o Alternating to Direct Current	AC	DC
o Direct to Alternating Current	DC	AC

Each of these has applications in at least one of the candidate systems which have been evaluated. The hardware implementation of each and the resulting impact on the system are considered in the following sections.

5.4.2. Mechanical Energy to Alternating Current Conversion.

5.4.2.1. An alternator is the conventional implementation for conversion of prime mover power to alternating current. In operation, a group of magnets is rotated by the prime mover past the stationary (stator) windings of the machine. The lines of magnetic flux which are cut by the stator windings generate a voltage across the terminals of the alternator, allowing electrical current and power to flow from the device. Since the level of generated voltage is directly proportional to the speed of the prime mover and the magnetic flux, adjustment of the electromagnetic excitation provided by the rotating field structure allows for control of the voltage. This control is not possible, however, if the magnetic flux is provided by permanent magnets.

5.4.2.2. Due to the high control gain between the field excitation and stator windings and the lack of high current brushes, a high speed three phase alternator is the most efficient means of making this conversion. A high speed machine is more efficient since the higher frequencies that are generated required less magnetic material than do lower frequencies. The use of three phases insures that the maximum amount of power will be produced, as compared to a single phase machine. Major losses in the alternator are restricted to mechanical friction and windage, and frequency related electrical losses in the magnetic material.

5.4.2.3. Regeneration directly back to the alternator is not possible. Typically the alternator feeds a bank of rectifiers to implement the AC to DC conversion. These devices are inherently one-way and will not allow reverse energy to flow. An electrical by-pass arrangement can be used to provide a reverse path, however, this involves using switching components which must be synchronized with the alternating stator fields and turned on appropriately. Essentially a complete bi-directional power bridge needs to be constructed to provide the regeneration function.

5.4.3. Mechanical Energy to Direct Current Conversion.

5.4.3.1. There are two major means of accomplishing this conversion. The alternator/rectifier approach (ME AC DC), and the DC generator (ME DC). The latter of these is the more direct and efficient approach. DC generators can take three basic forms; a commutated machine with either an electromagnetic or permanenet magnet field excitation, or a field excited homopolar machine. Each of the commutated machines requires a brush set to provide the mechanical rectification, whereas the homopolar machine only requires the brushes to provide a current path from the rotor. As with the alternator, the generator output voltage is dependent upon the speed of the prime mover. With either the electromagnetic field driven or the homopolar generator, control of the output voltage level is provided by the field. The output of the machine with permanent magnet field excitation cannot be directly controlled.

5.4.3.2. Mechanical losses in the DC generator are governed by friction, windage, and brush contact, and the electrical losses are those associated with resistive heating and brush contact potential. Overall efficiency on the DC generator is less than, but close to that of the AC machine.

5.4.3.3. DC machines have full regeneration capability. As the direction of power flow changes, the mode of operation makes a smooth transition from generation to motoring. Terminal polarity on the machine remains constant with only the direction of current reversing.

5.4.4. Alternating to Direct Current Conversions.

5.4.4.1. In the simplest manner, this conversion process is accomplished using and array of power rectifiers. These may be assembled in a variety of ways, however, the most common high power configuration is the three phase, full-wave bridge (see figure 5-13.). In this conversion process there is no control, so that the DC output of the bridge directly reflects the AC input level. A similar implementation of the three phase bridge using controllable power elements (i.e., silicon controlled rectifiers) does not allow for a degree of control over the output voltage. Such a "phase delay rectifier" arrangement is shown in figure 5.14.

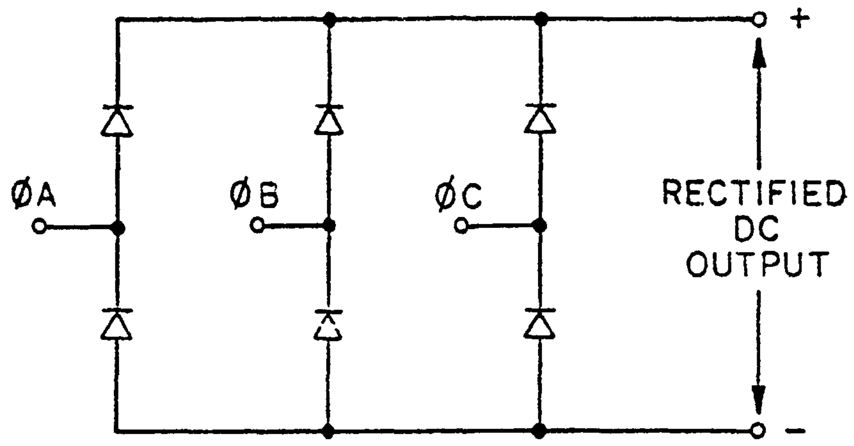


Figure 5-13. Three Phase Full-Wave Rectifier Bridge.

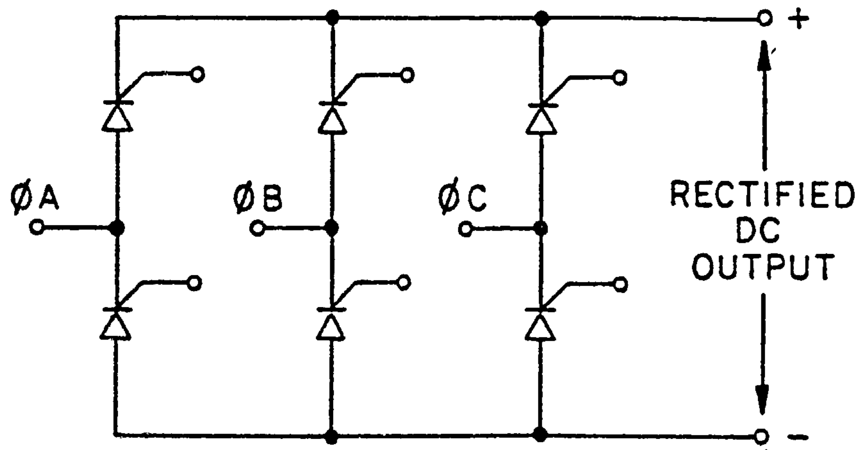


Figure 5-14. Phase Delay Rectifier.

5.4.4.2. The efficiency of the AC DC converter is governed by the forward voltage drop of the diodes when they are in a conducting mode. Power loss is the product of this voltage by the current drawn through the diodes. Since the voltage drop is a relatively fixed value, as the amplitude of AC input increases, the diode power loss is fixed (for a given load) and the net efficiency increases. For this reason it is advantageous to utilize the highest input voltage possible.

5.4.4.3. Regeneration is not directly possible with the common implementations of the AC DC coverter. Since the diodes are strictly one-way devices, power flow in the reverse direction is blocked. To allow regeneration to occur, active power elements must be placed in parallel with each diode rectifier and switched in synchronism with the phases of the AC input.

5.4.5. Direct to Alternating Current Conversions.

5.4.5.1. Conversion from a DC to AC is accomplished using an inverter. A common implementation of an inverter which uses power transistors as the active switching element is shown in figure 5-15. In operation, the DC bus is connected across the rails of the inverter and the active elements are appropriately switched to yield three phase AC waveforms at points A, B, and C. Depending on the control circuitry used, these waveforms may take the shape of trapezoidal flat-topped signals or true sinusoids. The waveform desired is dependent on the load which is driven. In form, the inverter described is identical to the construction of the motor controllers used to drive induction, synchronous, and brushless motors. The exact waveform required from the inverter is selected by the design of the motor.

5.4.5.2. Inverter efficiency is a function of the forward voltage drops of the solid-state elements, as well as of the switching losses associated with turning the semiconductors on and off. These switching losses can often equal or exceed those associated with the steady-state losses. For this reason, the losses associated with the inverter are often double that of the three phase rectifier (AC DC converter). The net efficiency of most high power inverters is greater than 95 percent.

5.4.5.3. Regeneration in the DC AC converter is accomplished by placing a three-phase rectifier bridge in parallel with the inverter power bridge as shown in figure 5-16. In this arrangement, bi-directional current flow is possible, although it is only controllable in the forward direction. Once the phase voltages exceed that of the DC bus, the reverse diodes will begin to conduct regardless of the state of the active switching elements.

5.4.6. System Aspects of Power Conversion .

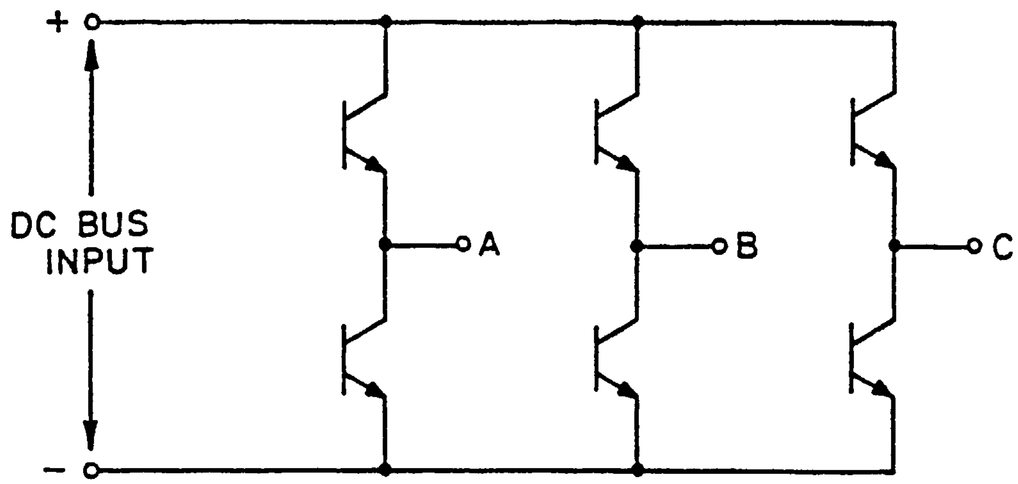


Figure 5-15. Three Phase Transistor Inverter.

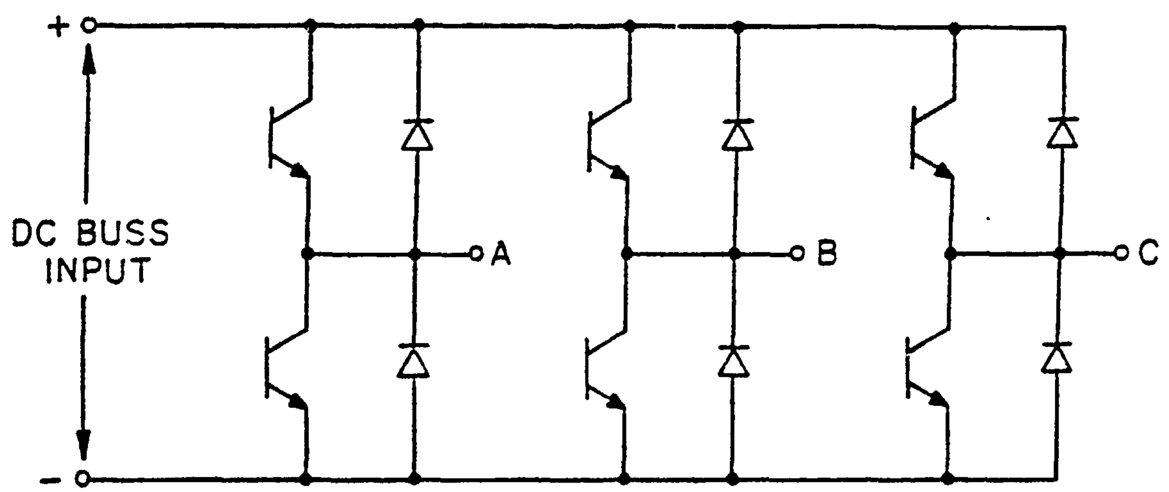


Figure 5-16. Three Phase Bidirectional Inverter.

5.4.6.1. In general, the fewer the number of power conversions that are required in a given system, the higher the efficiency and the reliability of that system. This assumes that each of the conversion blocks is roughly of the same efficiency. If the general power flows of the major candidate systems are mapped, the number of conversions involved yields an indication of the complexity, efficiency, and ultimately the reliability of each system.

- For the brushless DC, low and high frequency induction motor systems the power flow is given by; ME AC DC AC ME (prime mover to alternator to rectifiers to inverter to motor). Four conversions.

- For the homopolar and commutated DC motor (alternator driven) systems the power flow is given by; ME AC DC ME (prime mover to alternator to rectifiers to motor). Three conversions.

- For the homopolar (generator driven) system the power flow is given by; ME DC ME (prime mover to generator to motor). Two conversions.

5.4.6.2. It can be seen that the closer the system design approaches a purely DC power flow, the lower the number of energy conversions that are required.

5.5. Concept Descriptions

Paragraphs 5.5.1. through 5.5.3. describe the concepts developed for the 19.5 ton and 40.0 ton mechanical baseline. Paragraph 5.6. describes the performance characteristics of the concepts and their comparison to the contract requirements. The configurations that were eliminated, including the rationale for elimination, are described in Paragraph 5.7. and 5.8. In addition, alternate configurations and concepts that were analyzed are described in Paragraph 5.9.

5.5.1. Mechanical Baseline Concepts

The mechanical baseline concepts were generated to quantify the physical and performance characteristics for comparison with selected electric drive systems. To make a realistic comparison of electric drive vehicle concepts, mechanical baseline concepts were generated using contract specified diesel engines and hydro-kinetic transmissions for the 19.5 and 40 ton vehicles. All other propulsion system components were selected and sized based on the contract requirements for performance. The electric drive concepts were compared to the mechanical concepts and the summary of the comparison is contained in Paragraph 5.11.

5.5.2. 19.5 Ton Mechanical Baseline

5.5.2.1. The 19.5 ton mechanical baseline concept (Figure 5-17.) is configured with the contract specified engine and transmission along with state of the art propulsion system accessories (i.e., air cleaner, final drives, cooling system, etc.) to be compared with selected electric drive concepts based on the vehicle's physical layout and performance characteristics.

5.5.2.2. The Cummins VTA-903T engine was specified and utilized for the 19.5 ton mechanical baseline. The engine is an eight cylinder, 4 cycle, 903 cu. in., turbocharged, aftercooled diesel with a rating of 500 bhp at 2600 rpm.

5.5.2.3. The Detroit Diesel Allison X300-4A transmission is used in the 19.5 ton mechnical baseline concept. This transmission is the cross drive drive incorporating steering, braking and propulsion into one unit design. Its components consist of:

- o Torque Converter. The torque converter is directly connected to the engine by a flexplate drive.

- o Lockup Clutch. A hydraulicly actuated, automatic lockup clutch is employed to directly connect the converter pump and turbine at higher converter speed ratios. The time of lockup engagement is matched specifically to the engine-converter full throttle performance characteristics; however, lockup engagement is also modulated by throttle position for part throttle operation.

- o Bevel Gear Set. Power from the torque converter and lockup clutch passes through the bevel gear set where the rotational axis of the transmission input is changed from the longitudinal engine/converter plane to a plane perpendicular to the vehicle sprockets.

- o Range Planetary Pack. The range section consists of hydraulicly applied clutches and planetary gearing that provides four speeds forward and two speeds in reverse.

- o Hydrostatic Steer. Steer capability is independent of engine speed, which enables the vehicle to meet all steering requirements including "dead engine" steer. The steering hydrostatic unit is driven by the converter turbine through the bevel gear and spur gears and provides infinitely variable control. During steer, the variable displacement hydrostatic pump powers a fixed displacement motor in proportion to the command signal. The hydrostatic output drives the combining planetaries through differential gearing, causing equal acceleration and deceleration of the output shafts.

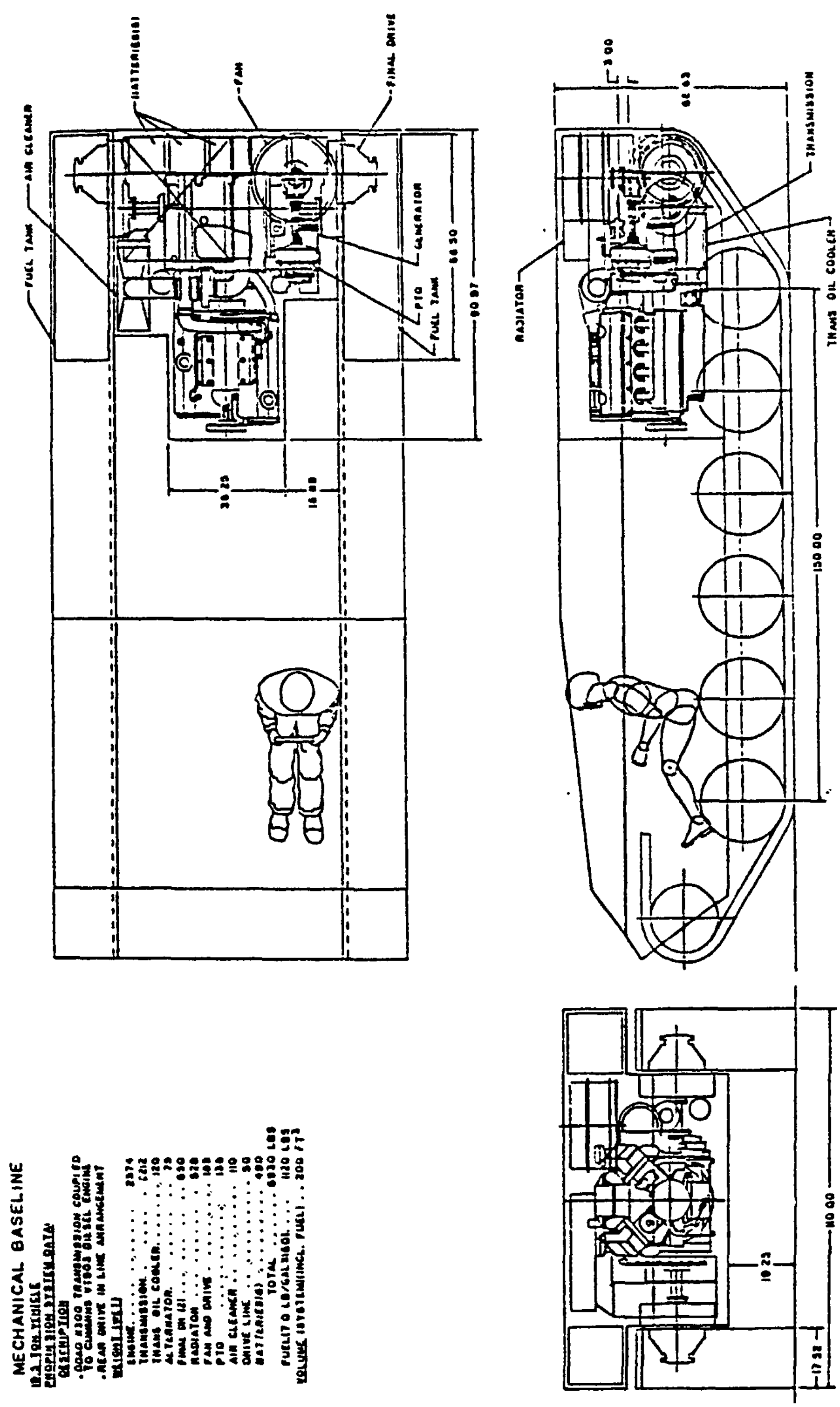


Figure 5-17. Mechanical Baseline Concept (19.5 Ton).

- Brakes. Mechanically applied brakes with hydraulic assist are incorporated in the X300 transmission. They are oil cooled, multiple-plate brakes. A "backup" mechanical brake system is used to provide additional braking capability for rapid or emergency stops as well as providing parking brake functions.

- Shift Control. A Detroit Diesel Allison commercially developed electric shift control in conjunction with the valve body provides the control for the transmission. This electro-hydraulic servo system is fully automatic with driver control of upshifts and fully inhibited to prevent engine overspeed. With the range selector in the automatic position, all range and lockup shifts are hydraulically actuated and occur automatically to provide the desired performance to meet all road load conditions.

5.5.2.4. The mechanical baseline final drives were selected and sized based on meeting the vehicle performance requirements of 45mph top vehicle speed and a maximum tractive effort per gross vehicle weight ratio 1.2. These final drives are single reduction spur gear design with a 3.86/1.00 fixed ratio.

5.5.2.5. The vehicle on board fuel capacity was determined using FMC's Vehicle Automotive Performance Program (VEHPER) and was based on a 300 mile range at an average vehicle speed of 25mph on hard paved ground (see Table 5-2. for computer output and range calculation). Since the fuel capacity required for the 24 hour mission profile was lower than that required for the 25 mph speed, the mission profile was not used in determining the on board fuel required (see Appendix B for mission profile fuel calculation).

5.5.2.6. The electrical system for the 19.5 ton mechanical baseline includes 6 nickel-cadmium batteries and a transmission driven 200 amp alternator. These components were selected to meet vehicle starting, lighting, and ignition requirements as well as silent watch conditions.

5.5.2.7. The air cleaner selected for the 19.5 ton mechanical baseline is similar in size and volume to the M2 vehicle air cleaner since both use the VTA-903 engine and require the same combustion airflow.

5.5.2.8. The exhaust system coupled to the VTA-903 engine is basically a straight pipe exhaust. Since the cooling fan and vehicle track constitute most to the ambient noise signature, a muffler was not used in any of the vehicle concepts.

5.5.2.9. The power pack cooling system components were selected and sized with the aid of FMC's vehicle automotive cooling computer program. Parametric studies were performed, varying cooling component size and

Table 5-2. Sample Operating Point and Fuel Economy

PREDICTED VEHICLE AUTOMOTIVE PERFORMANCE
PROPULSION SYSTEMS STAFF GROUP ORDNANCE ENGINEERING DIVISION

VEHICLE- TACOM ELECTRIC DRIVE; TYPE- TRACKED; GVW- 39000 LBS.; FAN DRIVE- FMC MODULATED
ENGINE- CUMMINS VTA-903T (500 HP); RADIATOR- YOUNG AAH-118 2-PASS COUNTER FLOW; CORE BLOCKAGE- 0.00 PERCENT
TRANSMISSION- ALLISON X300-4A (MOD); FAN- FMC 24 INCH AXIAL; MAX FAN POWER- 36.2 HP
GEAR RATIOS: TRANSFER CASE- 1.0000, DROPBOX- 1.0000, STEER DIFFERENTIAL- 1.0000, FINAL DRIVE- 3.8600
SPROCKET- 11. TEETH, 6.030 IN PITCH; AMBIENT-85.0° F, 29.00 IN. HG; SOLAR RADIATION ALLOWANCE-0.0° F;
****

VEHICLE OPERATING POINT

VEHICLE SPEED (MPH)	ROLLING RESISTANCE (LBS/TON)	PERCENT SLOPE	FRONTAL AREA (SQ FT)	DRAG COEFFICIENT (LBS/SQ FT/MPH**2)	TRACKTIVE EFFORT (LBS)
25.0	100.0	0.0	57.0	0.00240	2035.

*** PREDICTED FUEL ECONOMY ***

FUEL TYPE - DIESEL NO. 2
FUEL DENSITY - 7.000 LBS/GAL

GEAR	ENGINE SPEED (RPM)	LOAD FACTOR (PERCENT)	CONVERTER SPEED RATIO	CONVERTER TORQUE RATIO	BSFC (LBS/BHP-HP)	FUEL CONSUMPTION (GPH)
3LU	2197.	54.4	1.000	1.000	0.386	12.77

$$\frac{12.77 \text{ GAL/HR}}{25 \text{ MPH}} = 1.958 \text{ MPG}$$

FOR 300 MILE RANGE:

$$\frac{300 \text{ MILES}}{1.958 \text{ MPG}} = 153.21 \text{ GAL}$$

\+ 7.66 GAL (5% RESERVE)

\------

160.87 GAL TOTAL FUEL REQUIRED

fan power to aquire the optimum cooling system configuration (see Figure 5-18. for example). The following components were selected for use in the 19.5 ton mechanical baseline concept (Figure 5-19. illustrates the mechanical cooling system arrangement):

- o Radiator. The radiator for the 19.5 ton mechanical baseline provides air to water cooling for the engine along with water to oil cooling for the transmission (see transmission oil cooler). It is an 11 fin/in., 8 row, 2-pass counterflow heat exchanger similar to the M2 unit.

- o Fan and Drive. The FMC 24 inch axial flow fan driven by the M2 variable flow fluid coupling (VFFC) drive is installed for the mechanical baseline vehicle. It provides sufficient airflow to meet vehicle cooling requirements while conserving fan horse-power. The VFFC is mounted to transmission PTO along with the vehicle electrical alternator.

- o Transmission Oil Cooler. Coolant flows from the engine through a 6 inch diameter, 23 inch long, shell and tube heat exchanger to provide cooling for the X300 transmission oil. This unit is mounted next to the transmission in the horizontal position.

- o Ballistic Grille. Ballistic grilles impose a significant pressure penalty on the airflow of the cooling system and is therefore, in this instance, considered part of the cooling system. Grille sizing was estimated by using standard FMC practice and experience. M2 type ballistic grille bars were selected as represetative of typical combat vehicle grilles.

5.5.3. 40 Ton Mechanical Baseline. The 40 ton mechanical baseline concept (Figure 5-20.) was configured with the contract specified engine and transmission along with state of the art propulsion system accessories compatible with selected electric drive concepts. Details of the baseline vehicle's physical and performance characteristics can be found in section 5.11.

5.5.3.1. The Cummins LCR-V903T engine is used in the 40 ton mechanical baseline. It is an 8 cylinder, 4 cycle, 903 cubic inch, turbo-charged, turbocompound, aftercooled diesel with a rating of 1000 bhp at 3200 rpm. The turbocompound system is composed of an advanced high pressure ratio turbocharger with variable geometry and a low pressure power turbine geared to the engine crankshaft at a fixed ratio.

5.5.3.2. The Detroit Diesel Allison ATT-1064 transmission is used in the 40 ton mechanical baseline concept. This transmission is similar to that of the X300 used in the 19.5 ton mechanical baseline concept, except the ATT-1064 has six forward ranges where the X300 has only four.

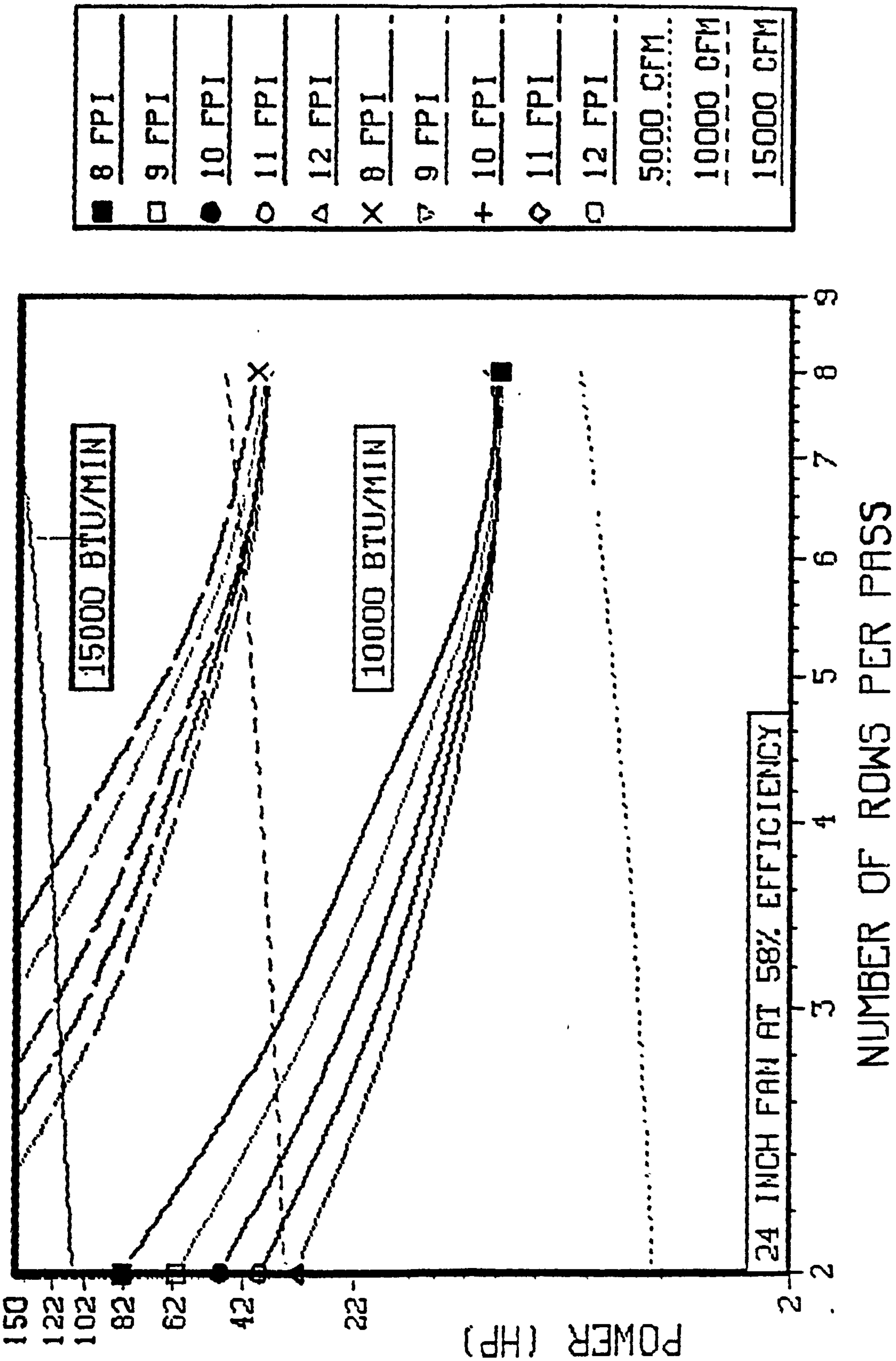


Figure 5-18. Radiator Sensitivity Analysis.

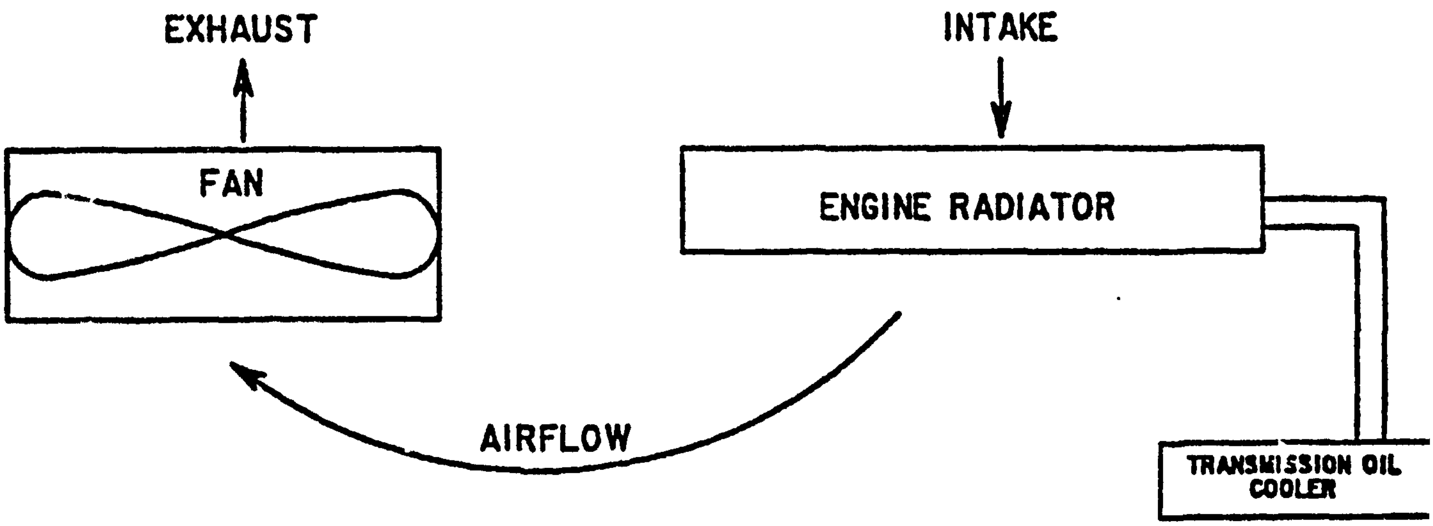


Figure 5-19. Mechanical Cooling System Arrangement.

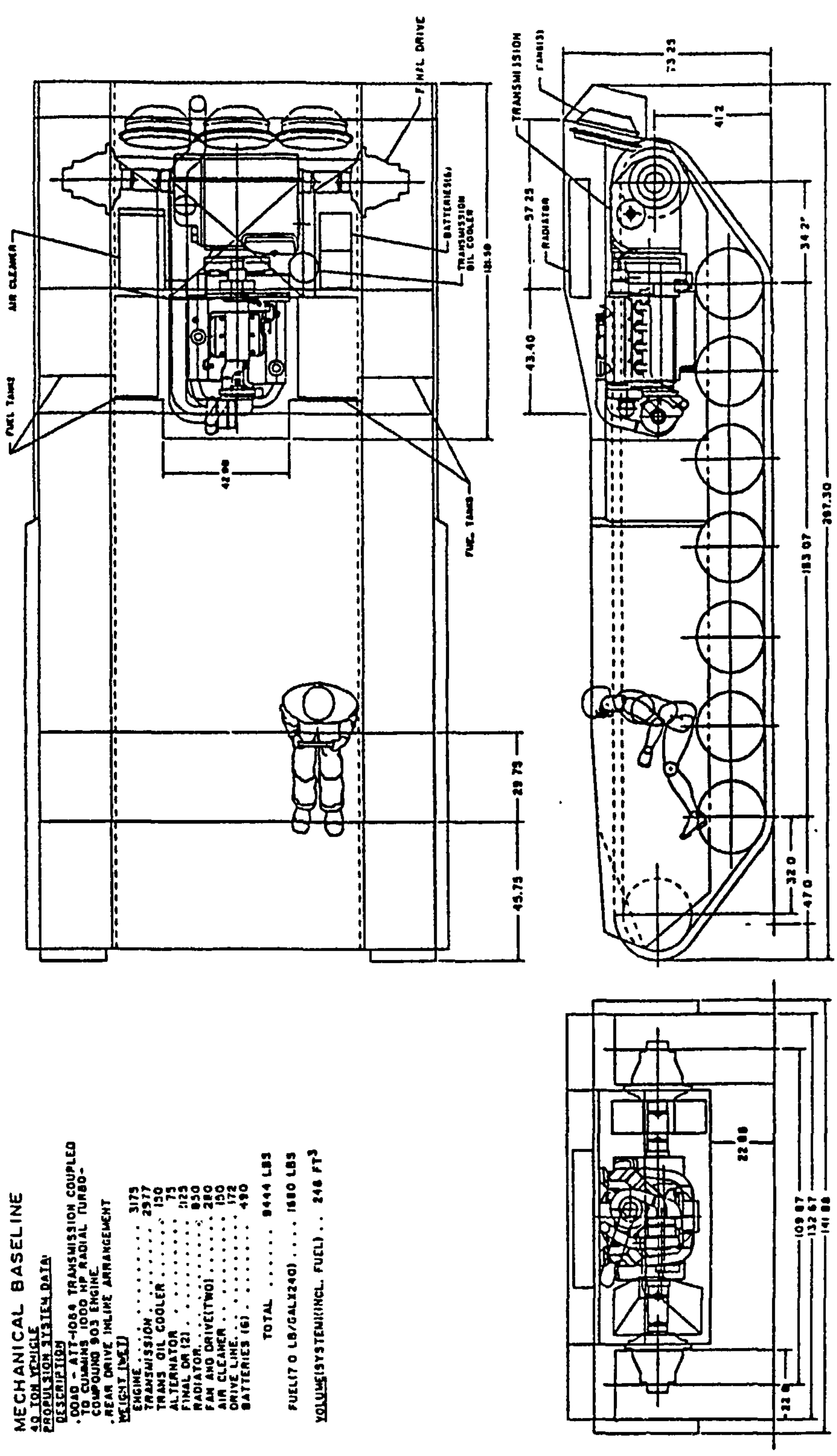


Figure 5-20. Mechanical Baseline Concept (40 Ton Vehicle)

5.5.3.3. The final drive selected for the 40 ton mechanical baseline vehicle is a double reduction spur gear design with roller bearings, spherical output bearings, and a face type output seal to exclude contamination. A 7.625:1.00 fixed ratio was selected to meet or exceed the vehicle performance requirements of 45mph top speed and a maximum tractive effort per gross vehicle weight ratio of 1.2.

5.5.3.4. As with the 19.5 ton mechanical baseline, the on-board fuel capacity for the 40 ton mechanical baseline was determined based on a 300 mile range at an average vehicle speed of 25mph on hard paved ground. Table 5-3. is a sample of the fuel consumption prediction from the vehicle automotive performance computer program.

5.5.3.5. To meet the vehicle starting, lighting, ignition and silent watch requirements, the 40 ton mechanical baseline is equipped with 6 nickel-cadmium batteries and a transmission driven 200 amp alternator.

5.5.3.6. The air cleaner selected for the 40 ton mechanical baseline is a self cleaning type which was sized using estimated airflow for the LCR-903 engine at 900HP.

5.5.3.7. As in the 19.5 ton mechanical baseline the 40 ton mechanical baseline is configured with a straight pipe exhaust. A muffler was not used in any of the 40 tone vehicle concepts.

5.5.3.8. The 40 ton mechanical baseline cooling was selected and sized using FMC's vehicle automotive cooling computer program (see Table 5-4. for program output). The following components make up the 40 ton baseline cooling system:

- Radiator. The radiator for the 40 ton mechanical baseline provides air to water cooling for the engine along with water to oil cooling for the hydrokinetic transmission. It is an 11 fin/in., 8 row, 2-pass couterflow heat exchanger.

- Fans and Drives. Three 20 inch mixed flow, hydraulically driven, Airscrew fans were concepted in the 40 ton mechanical baseline.

- Transmission Oil Cooler. Coolant flows from the engine through a shell and tube heat exchanger providing cooling for the ATT-1064 transmission. This unit is mounted next to the transmission in the vertical position.

- Ballistic Grilles. The ballistic grilles for the 40 ton mechanical baseline were sized using standard FMC practice and experience. As was in the 19.5 ton vehicle, M2 type ballistic grilles were selected as representative of typical combat vehicle grilles.

Table 5-3. Sample Operating Point and Fuel Economy

PREDICTED VEHICLE AUTOMOTIVE PERFORMANCE
PROPULSION SYSTEMS STAFF GROUP ORDNANCE ENGINEERING DIVISION FMC CORPORATION SAN JOSE, CA

VEHICLE- ELECTRIC DRIVE (40 TON); TYPE- TRACKED; GVW- 80000 LBS.; FAN DRIVE- HYDRAULIC
ENGINE- CUMMINS LCR V903TCPD (1000 HP); RADIATOR- YOUNG AAH-118 2-PASS COUNTER FLOW; CORE BLOCKAGE- 0.00
TRANSMISSION- ALLISON ATT-1064 (6 SPEED); FAN- 20 INCH MIXED FLOW; MAX FAN POWER- 100.0 HP
GEAR RATIOS: TRANSFER CASE- 1.0000, DROPBOX- 1.0000, STEER DIFFERENTIAL- 1.0000, FINAL DRIVE- 5.1000
SPROCKET- 11. TEETH, 7.625 IN PITCH; AMBIENT-85.0° F, 29.00 IN. HG; SOLAR RADIATION ALLOWANCE-0.0° F;
****

VEHICLE OPERATING POINT

VEHICLE SPEED (MPH)	ROLLING RESISTANCE (LBS/TON)	PERCENT SLOPE	FRONTAL AREA (SQ FT)	DRAG COEFFICIENT (LBS/SQ FT/MPH**2)	TRACKTIVE EFFORT (LBS)
25.0	100.0	0.0	68.3	0.00240	4102.

*** PREDICTED FUEL ECONOMY ***
FUEL TYPE - DIESEL NO. 2
FUEL DENSITY - 7.000 LBS/GAL

GEAR	ENGINE SPEED (RPM)	LOAD FACTOR (PERCENT)	CONVERTER SPEED RATIO	CONVERTER TORQUE RATIO	BSFC (LBS/BHP-HP)	FUEL CONSUMPTION (GPH)
4LU	3143.	44.9	1.000	1.000	0.396	25.17
5LU	2127.	54.7	1.000	1.000	0.344	19.26

$$\frac{19.26\ \text{GAL/HR}}{25\ \text{MPH}} = 1.298\ \text{MPG}$$

FOR 300 MILE RANGE:

$$\frac{300\ \text{MILES}}{1.298\ \text{MPG}} = 231.1\ \text{GAL}$$

+ 11.5 GAL (5% RESERVE)

242.7 GAL TOTAL FUEL REQUIRED

Table 5-4. Sample Cooling Point

PREDICTED VEHICLE COOLING SYSTEM PERFORMANCE
PROPULSION SYSTEMS STAFF GROUP   ORDNANCE ENGINEERING DIVISION   FMC CORPORATION   SAN JOSE, CA

VEHICLE- ELECTRIC DRIVE COMP;   TYPE- TRACKED;   GVW- 80000 LBS.;   FAN DRIVE- HYDRAULIC
ENGINE- CUMMINS LCR V903TCPO (1000 HP);   RADIATOR- YOUNG AAH-118 2-PASS COUNTER FLOW;   CORE BLOCKAGE- 0.00 PERCENT
TRANSMISSION- ALLISON ATT-1064 (TC-494 CONVERTER);   FAN- 3 EA 20 IN MIXED FLOW NOAAH FANS;   MAX FAN POWER- 102.0 HP
GEAR RATIOS: TRANSFER CASE- 1.0000, DROPBOX- 1.0000, STEER DIFFERENTIAL- 1.0000, FINAL DRIVE- 5.1000
SPROCKET- 11. TEETH, 7.625 IN PITCH, AMBIENT-120.0° F, 29.00 IN. HG; SOLAR RADIATION ALLOWANCE-0.0° F; COOLING POINT-0.700 TE/GVW
****

*** PREDICTED DATA ***

	POWER (HP)
ENGINE	
GHP AT 2581.3 RPM	869.6
BHP AFTER 3.50 PERCENT DERATE	839.1
POWER LOSS BEFORE TRANSMISSION	
INTAKE AND EXHAUST	8.4
FAN DRIVE	102.0
ELECTRICAL SYSTEM	4.7
HYDRAULIC SYSTEM	2.4
TRANSFER CASE	0.0
DISCONNECT CLUTCHES	0.0
TRANSMISSION	
INPUT POWER	721.6
OUTPUT POWER	557.4
POWER LOSS	164.2
POWER LOSS BEFORE SPROCKET	
DROPBOX	0.0
STEERING DIFFERENTIAL	0.0
FINAL DRIVE	16.7
AT THE SPROCKET	
OUTPUT POWER	540.7
TRACKTIVE EFFORT	56000. LBS
TE/GVW	0.700

VEHICLE SPEED 3.62 MPH

CONVERTER	
SPEED RATIO	0.6614
TORQUE RATIO	1.2846

SYSTEM CONFIGURATION (COMPONENT LOCATION)	
TRANSMISSION COOLER	2
RADIATOR	1
FAN	3

FAN OPERATING POINT		
STATIC PRESSURE	12.35	IN WG
VELOCITY PRESSURE	2.82	IN WG
AIRFLOW	24031.	CFM
INLET DENSITY	.05788	LBS/CU FT
BRAKE HORSEPOWER	102.0	HP
FAN SPEED		RPM

	SYSTEM STATIC PRESSURE LOSS (IN WG)
INLET GRILLE	1.16
ENGINE COMPARTMENT	6.15
RADIATOR	2.06
OUTLET GRILLE	2.98
RAM	0.0

	HEAT REJECTION RATE TO (BTU/MIN) COOLANT	AIR
ENGINE (27.00 BTU/M/BHP)	22656.5	6162.6
FAN DRIVE	0.0	0.0
HYDRAULIC SYSTEM	0.0	0.0
ELECTRICAL SYSTEM	0.0	0.0
DISCONNECT CLUTCHES	0.0	0.0
TRANSFER CASE	0.0	0.0
TRANSMISSION	6267.1	696.3
DROPBOX	0.0	0.0
STEER DIFFERENTIAL	0.0	0.0
EXHAUST SYSTEM (MUFFLER)		0.0
FAN HEAT OF COMPRESSION		4325.8
TOTAL	28923.7	11184.7

	TEMPERATURE (DEG F) INLET	OUTLET
COOLANT		
RADIATOR	235.46	208.78
FAN DRIVE COOLER	208.78	208.78
ENGINE	208.78	229.68
TRANSMISSION COOLER	229.68	235.46
AIR		
FAN	226.74	239.65
RADIATOR	120.00	206.28
OIL		
TRANSMISSION COOLER	290.30	252.18

RADIATOR HEAT REMOVAL CAPABILITY- 28923.7 BTU/MIN AT 135.5 GPM COOLANT, 18546. SCFM AIR, AND 115.46° F ITD

TRANSMISSION COOLER HEAT REMOVAL- 6267.1 BTU/MIN AT 135.5 GPM COOLANT FLOW, 44.4 GPM OILFLOW AND 60.62° F ITD

5.6. Configuration I and II Concept Descriptions

In this section Configuration I and II concepts are described in terms of the areas shown below:

- o Concept layout and packaging
- o Operation in terms of meeting contract vehicle performance and vehicle requirements:
  - Transmission operation
  - Steer system operation (pivot, engine failure, towing, full engine power)
  - Braking operation (capability, redundancy)
  - Electrical/hydraulic capability
  - Shock
  - Flow chart or other description of propulsion system control logic along with engine schedu-ing objectives
- o Component and subsystem descriptions
- o Propulsion and steer system schematic with power flow shown in bold line for all speed ranges or changes in power flow direction
- o Identification and discussion of unusual materials, production processes, technical risk and relative cost estimate.

5.6.1. Configuration I Arrangement and Packaging.

The final configuration concept arrangements were selected during the concept screening process discussed in section 5.10. A rear drive was used for the fundamental location in order to establish a basis for equal comparison. Vehicle mission profile usually determines the drive location, depending on whether it is a weapon or personnel carrier. An attempt was made to centralize the machinery components and fuel storage. The 19.5 ton concepts are illustrated at Figure 5-21. (high frequency induction motor) and Figure 5-22. (homopolar motor).

One concept of a split power pack is included at Figure 5-23. for the High Frequency Induction AC system which shows a front engine-alternator assembly with a rear motor/final drive. The split pack is a graphic representation which shows the versatility of the electric drive system. Electrical controls are shown in close proximity to their termination points. The 40 ton concepts are shown at Figures 5-24. (High Frequency

Induction Motor) and 5-25. (Homopolar Motor). All concepts with Configuration I arrangement are discussed below in terms of vehicle subsystem.

5.6.1.1. Vehicle Cooling. The prime consideration for component placement is function. The radiator was placed as far away from the fan as possible in the engine compartment in order to reduce dead air space. Generally the radiator is shown over the engine for maximum cooling.

The 19.5 ton electric vehicles use a 24 inch fan with a 30 HP direct drive 4400 rpm motor. The 40 ton vehicle cooling is somewhat unique because the three, 20 inch fans are hydraulically driven using one pump and individual motors.

5.6.1.2. Electrical Cooling. Components of the electrical system (motors, alternator/generator and controls) are cooled by a radiator placed in series with the vehicle main cooling radiator.

5.6.1.3. Regenerative Brake Cooling. A resistive grid is used in all of the electric drive systems to dissipate the heat generated during braking. Power regeneration, caused by electrical braking of the vehicle occurs when the power absorption exceeds the capability of the system. High heat can be generated but is of short duration. The brake grid is shown located in the exhaust fan plenum of all electric vehicle drawings.

Two other methods for regenerative brake cooling were also considered. An additional radiator could be added in the vehicle radiator intake grill or a completely separate cooling unit could be used.

5.6.1.4. Vehicle Electric. Vehicle electrical requirements were determined to be similar for both the 19.5 ton or 40 ton vehicle. Six maintenance-free batteries were provided for each vehicle. Battery charging is done on the homopolar concepts with an alternator which located and is driven by the engine speed increaser gear box between the engine and generator as shown on figure 5-26. Battery charging for the High Frequency Induction System is provided by the propulsion alternator (Figure 5-27.). "Central Control Unit" called out on each concept drawing is shown as space allocated for the estimated maximum control volume required for the system. Weight for the controls is included in the drawing tabulation.

5.6.1.5. Brakes. Two types of brakes are shown. Dry rotor and puck in the induction system is a totally external arrangement which picks up a gear box shaft extension on the low torque end of the gear train. Actuation can be mechanical or hydraulic or both, depending on preference and/or legal requirements.

HF INDUCTION AC SYSTEM

19.5 TON VEHICLE CONFIGURATION • I

PROPULSION SYSTEM DATA:

DESCRIPTION

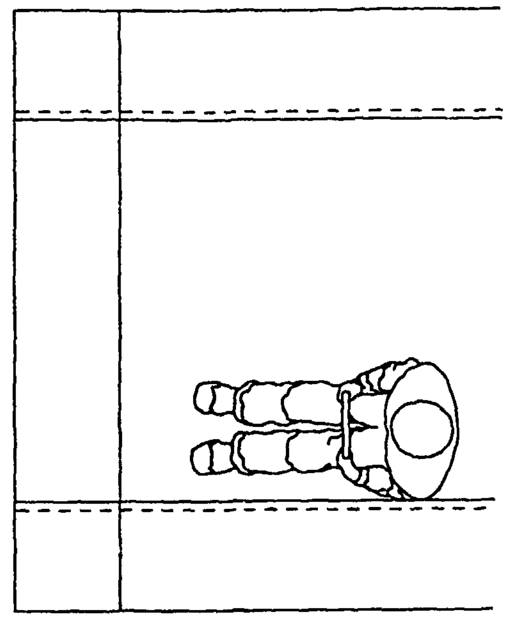
- INDUCTION MOTORS/FIXED RATIO FINAL DRIVES WITH DRY BRAKES
- ALTERNATOR COUPLED TO CUMMINS VT903 DIESEL ENGINE. THRU SPEED INCREASING GEAR BOX.
- REAR DRIVE TRANSVERSE ARRANGEMENT

WEIGHT (WET)

ENGINE	2374
ALTERNATOR	132
ALT/DRIVE	125
MOTOR (2)	600
FINAL DRIVES (2)	524
RADIATOR	536
FAN AND DRIVE	160
AIR CLEANER	110
BUS BARS	100
BATTERIES (6)	490
REGENERATION BRAKE GRID	120
CONTROLS	200
PARKING BRAKE	67
TOTAL	5538 LBS

FUEL (7.0 LB/GALX160) . . . 1120 LBS

VOLUME (SYSTEM)(INCL. FUEL) . . 164 FT^3



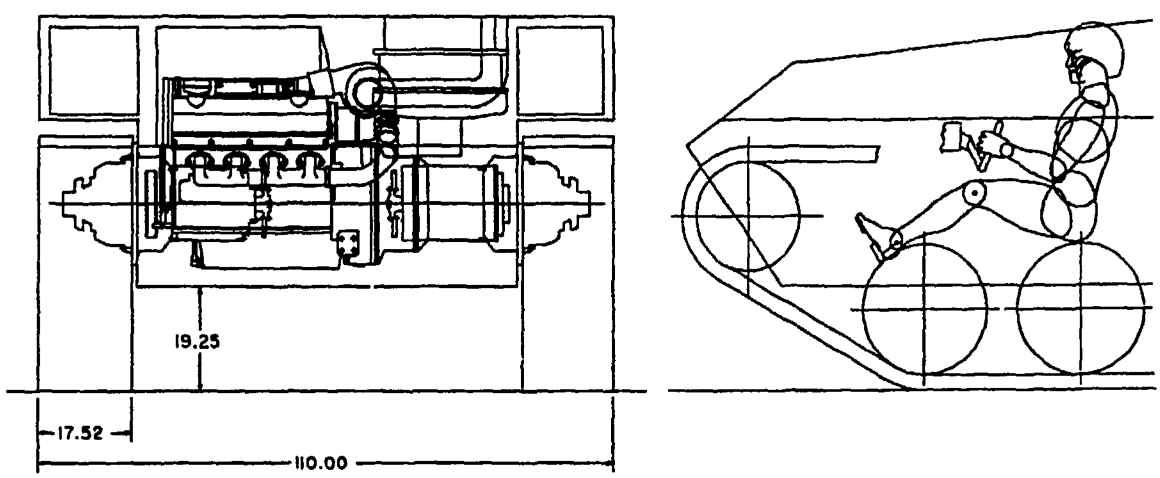


Figure 5-21. HF Induction AC System (19.5 Ton)

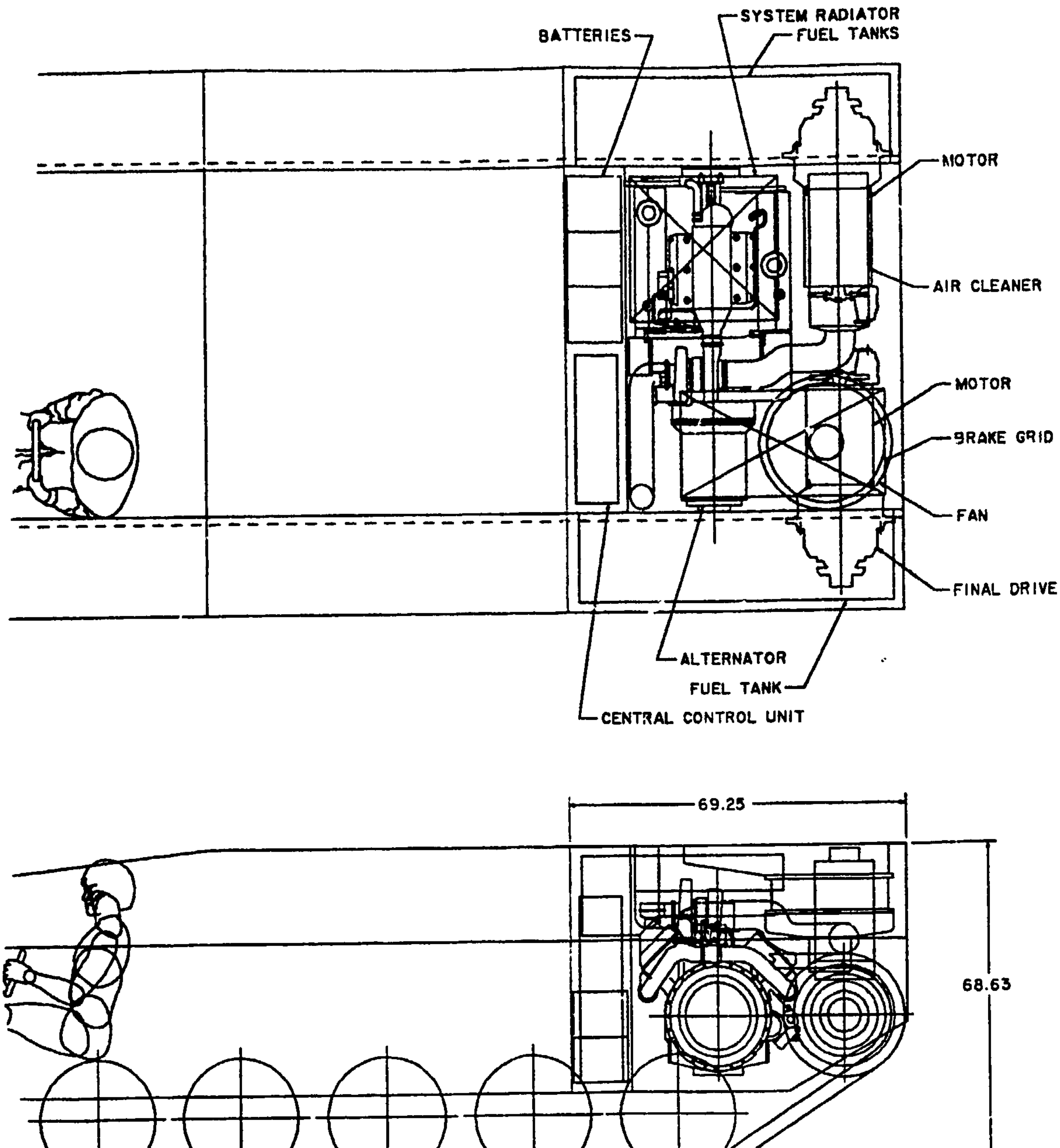


Figure 5-21. (Continued)

HOMOPOLAR SYSTEM

19.5 TON VEHICLE CONFIGURATION • I

PROPULSION SYSTEM DATA:

DESCRIPTION

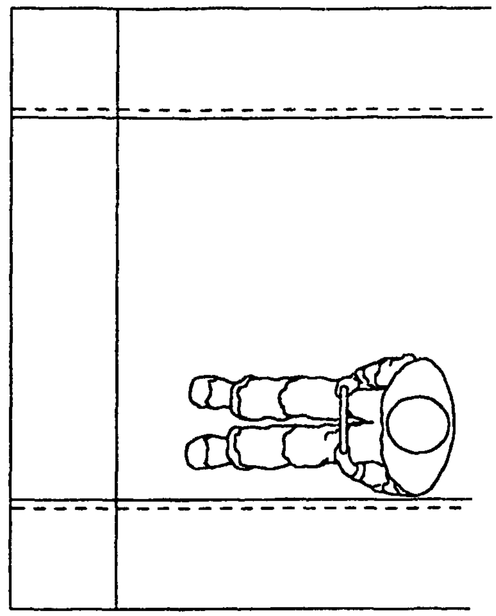
- INDUCTION MOTORS/FIXED RATIO FINAL DRIVES WITH WET BRAKES
- GENERATOR COUPLED TO CUMMINS VT903 DIESEL ENGINE. THRU SPEED INCREASING BEVEL GEAR BOX.
- REAR DRIVE 'T' IN-LINE ARRANGEMENT

WEIGHT (WET)

ENGINE	2374
GENERATOR	749
GEN/DRIVE	125
MOTOR (2)	749
FINAL DRIVES (2)	524
VEH ALTERNATOR	75
RADIATOR	536
FAN AND DRIVE	160
AIR CLEANER	110
BUS BARS	205
BATTERIES (6)	490
REGENERATION BRAKE GRID	120
CONTROLS	180
TOTAL	6397 LBS

FUEL (7.0 LB/GALX160) . . . 1120 LBS

VOLUME (SYSTEM)(INCL. FUEL) . . 149 FT^3



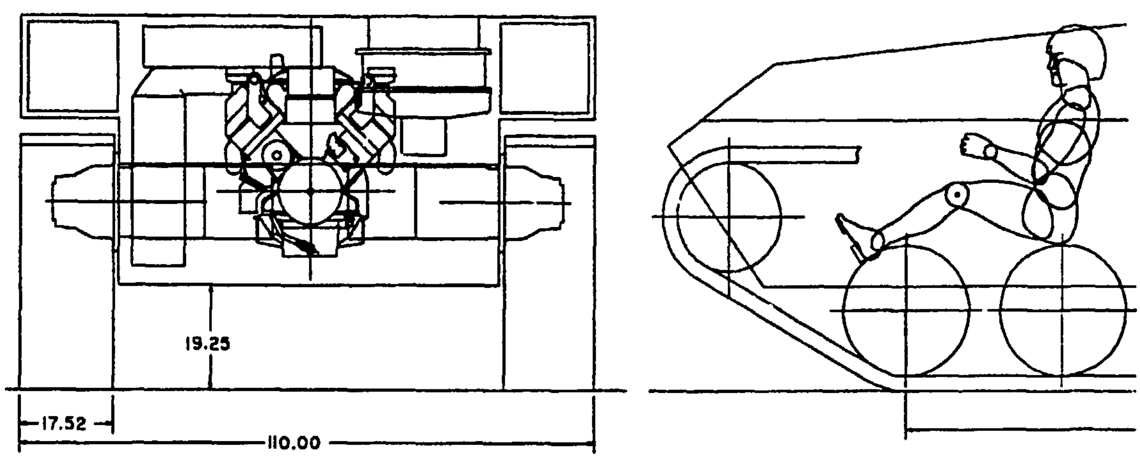


Figure 5-22. Homopolar (DC) System (19.5 Ton)

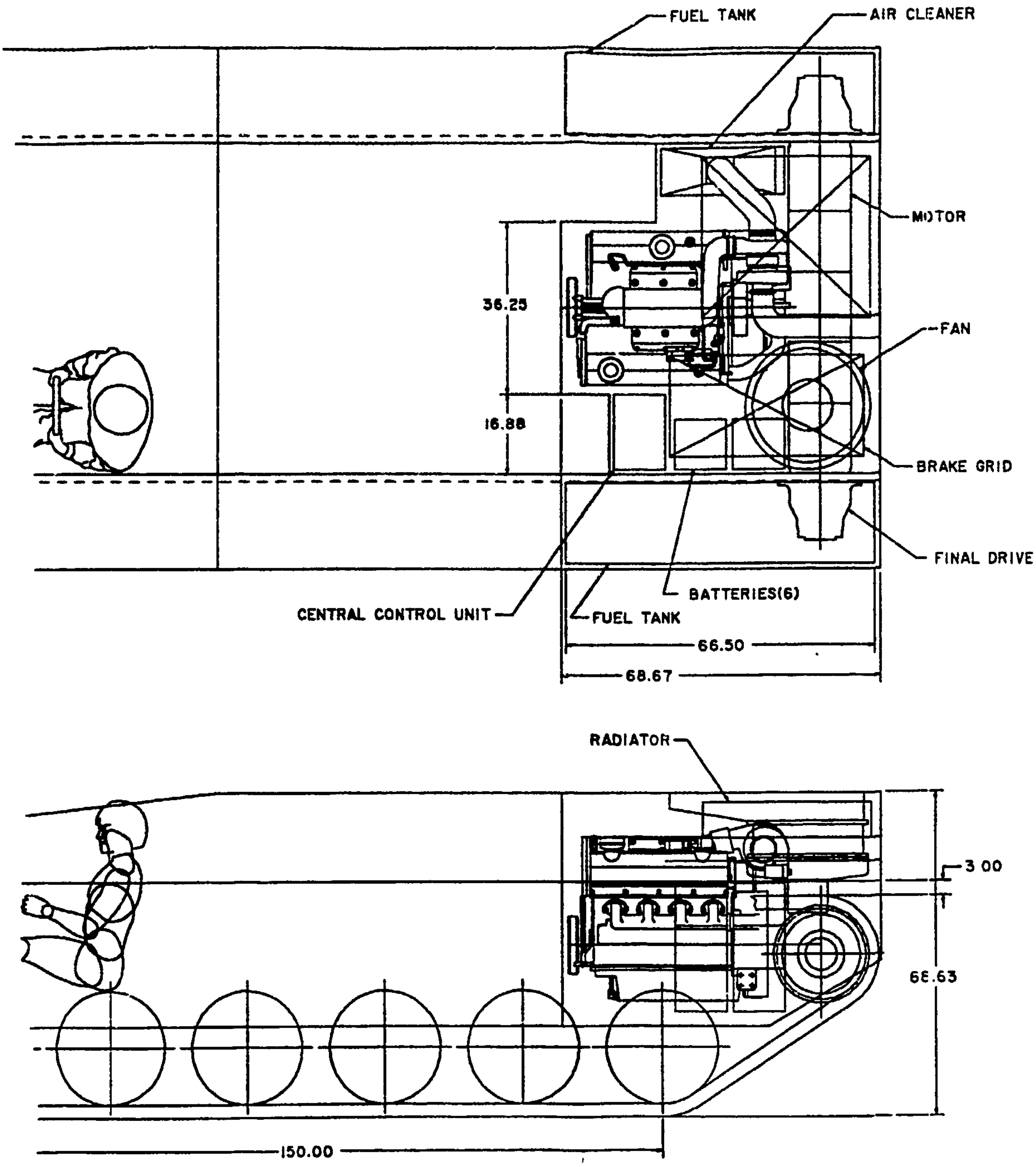


Figure 5-22. (Continued)

HIGH FREQUENCY INDUCTION AC

19.5 TON VEHICLE CONFIGURATION • I

PROPULSION SYSTEM DATA:

DESCRIPTION

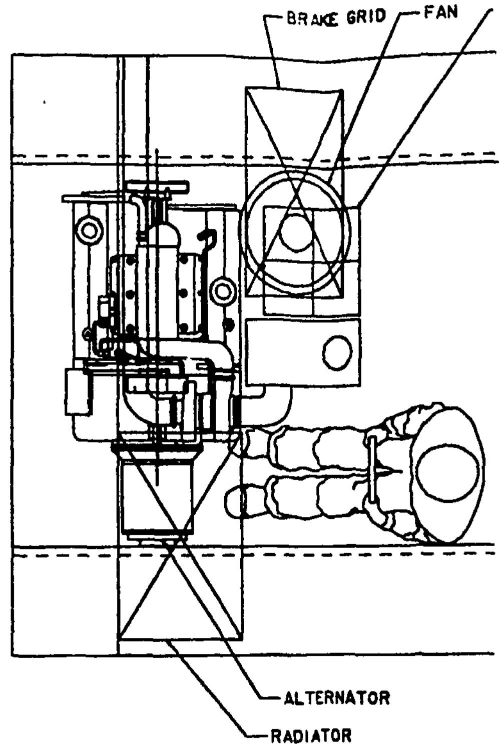
- HIGH FREQUENCY MOTORS/FIXED RATIO FINAL DRIVES.
- ALTERNATOR DIRECTLY COUPLED TO CUMMINS VT903 DIESEL ENGINE.
- REAR DRIVE SPLIT ARRANGEMENT

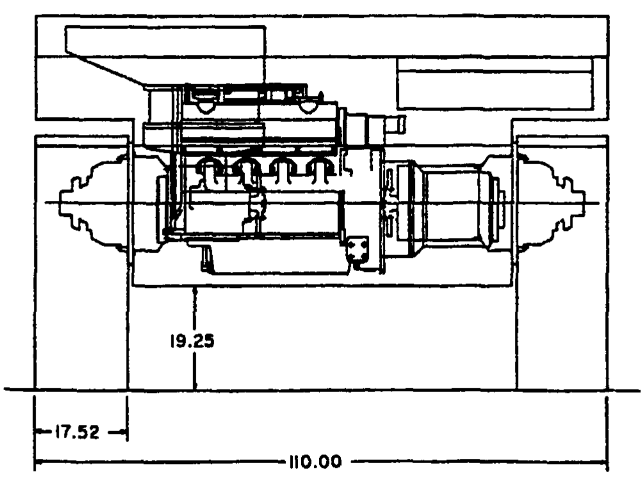
WEIGHT (WET)

ENGINE	2374
ALTERNATOR	132
ALTERNATOR DRIVE	125
MOTOR (2)	600
FINAL DRIVES (2)	524
RADIATOR	536
FAN AND DRIVE	160
AIR CLEANER	110
BUS BARS	100
BATTERIES (6)	490
REGENERATION BRAKE GRID	120
CONTROLS	200
PARKING BRAKE	67
TOTAL	5538 LBS

FUEL (7.0 LB/GALX160) . . . . 1120 LBS

VOLUME (SYSTEM) . . . 182 FT^3





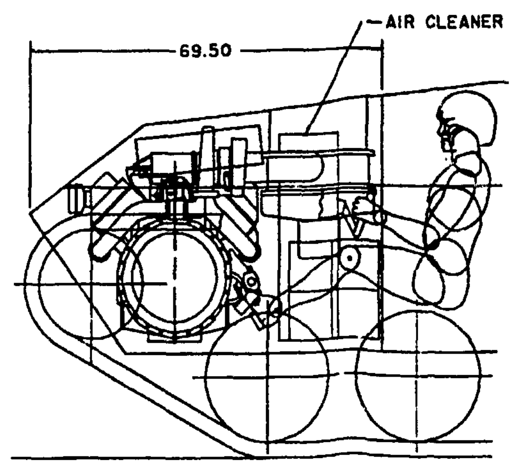


Figure 5-23. HF Induction AC System (19.5 Ton) (Split Power Pack)

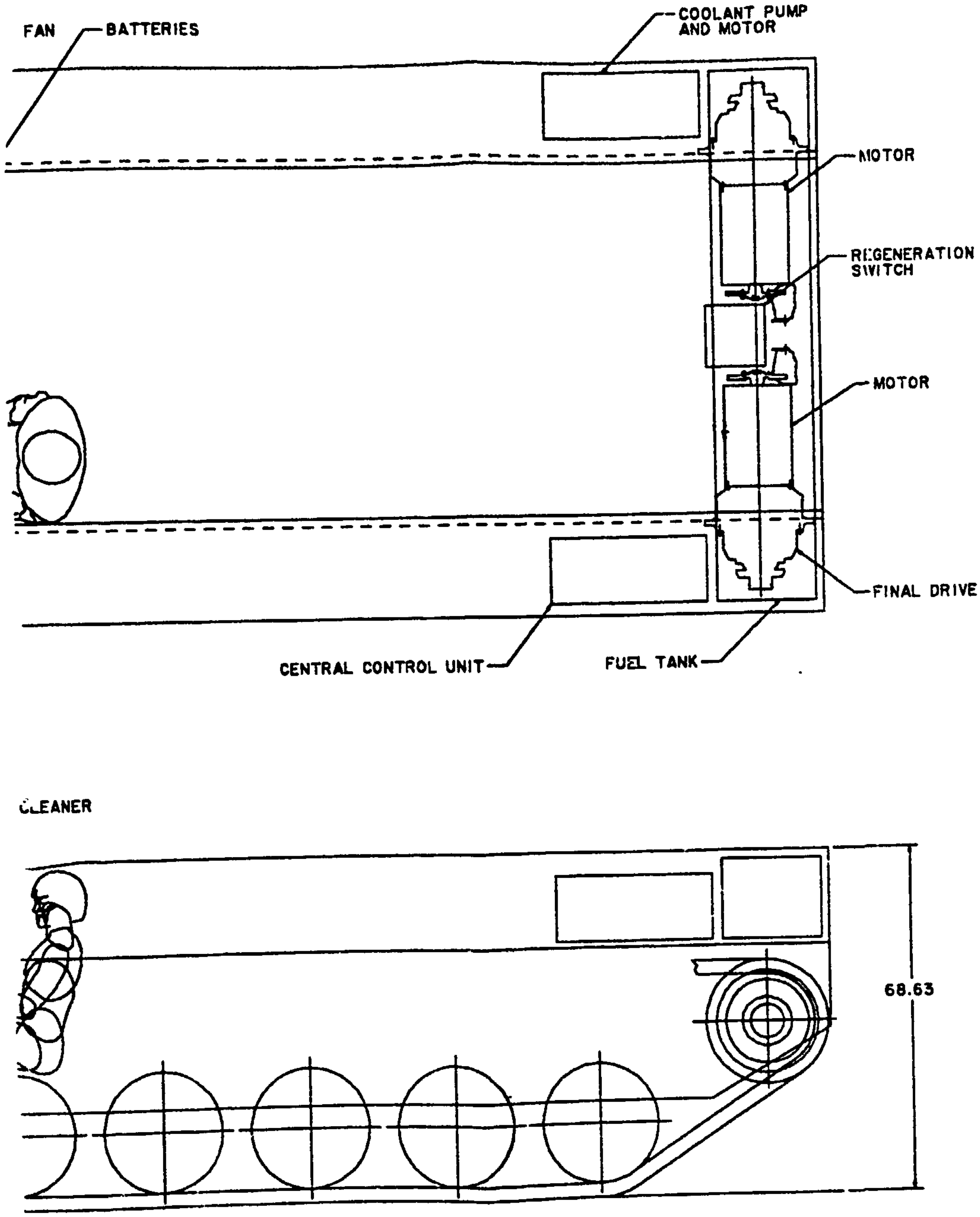


Figure 5-23. (Continued)

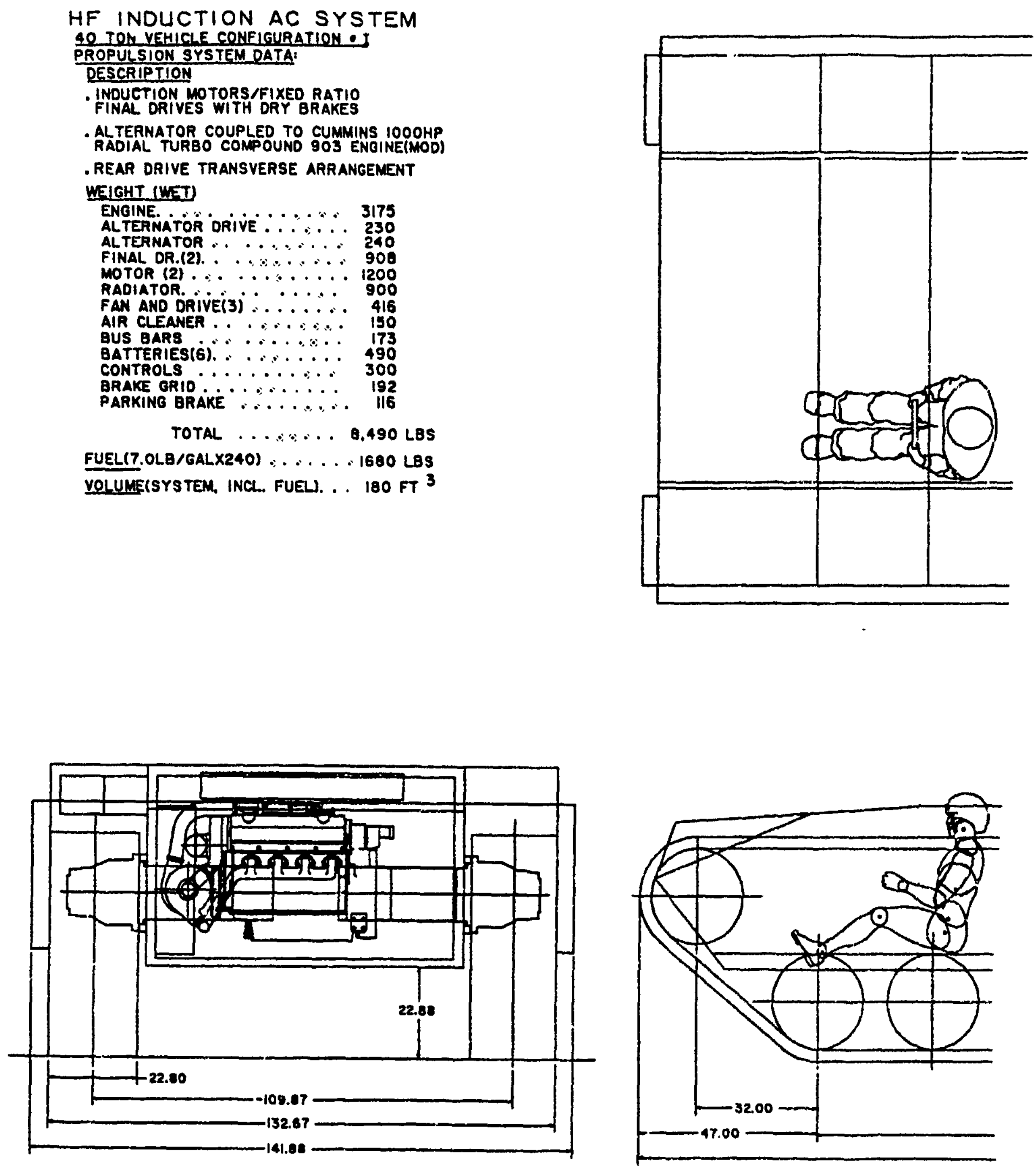


Figure 5-24. HF Induction AC System (40 Ton)

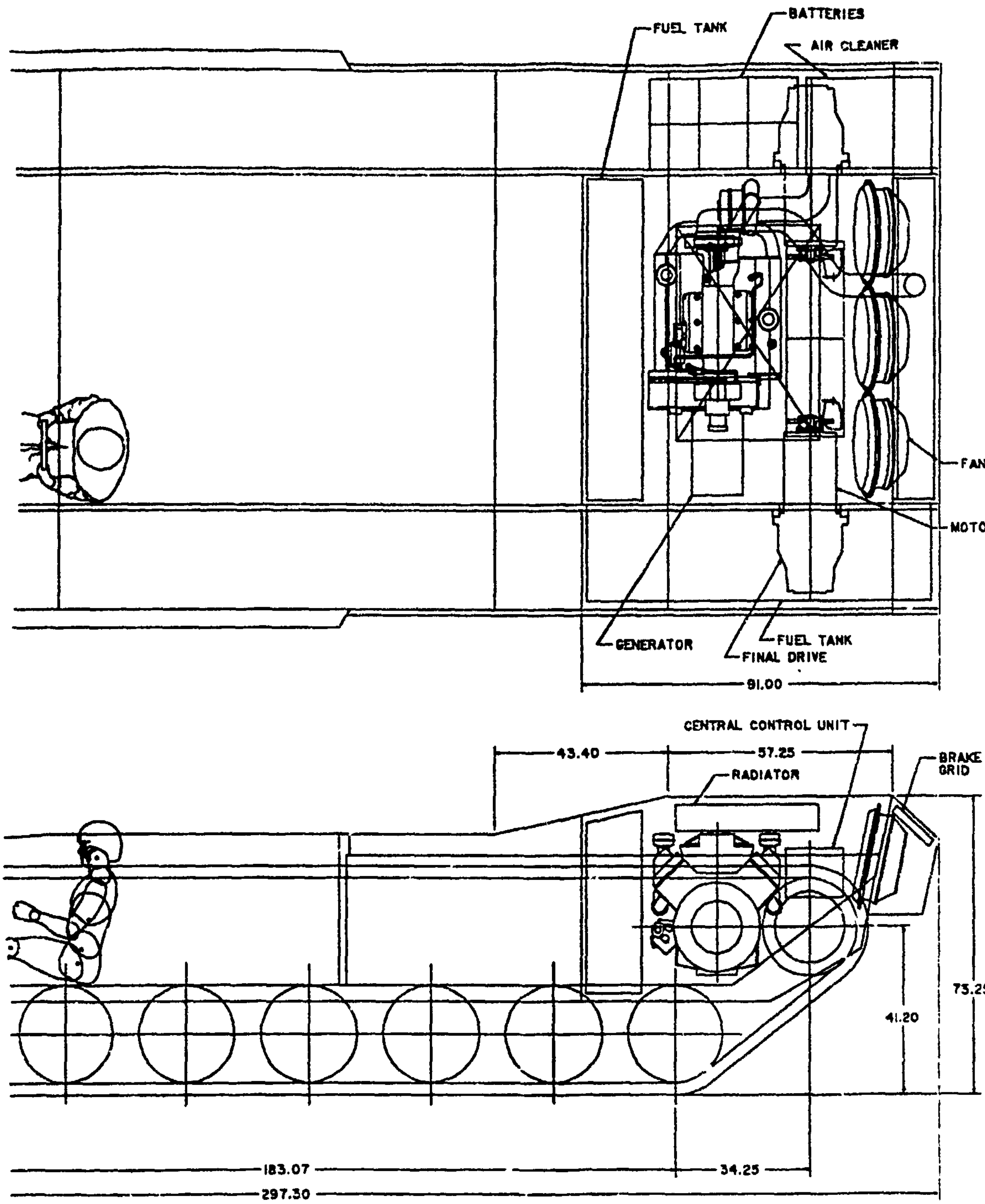


Figure 5-24. (Continued)

HOMOPOLAR SYSTEM

40 TON VEHICLE CONFIGURATION # I

PROPULSION SYSTEM DATA:

DESCRIPTION

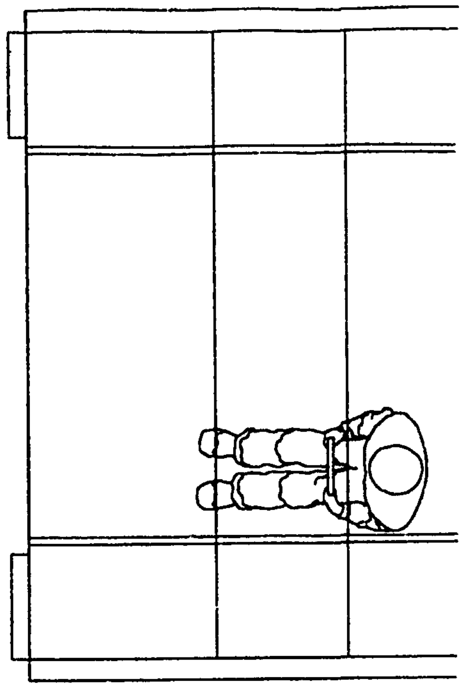
- HOMOPOLAR MOTORS/FIXED RATIO FINAL DRIVES WITH WET BRAKES
- GENERATOR COUPLED TO CUMMINS 1000HP RADIAL TURBO COMPOUND 903 ENGINE(MOD)
- REAR DRIVE TRANSVERSE ARRANGEMENT

WEIGHT (WET)

ENGINE	3175
GENERATOR DRIVE	230
GENERATOR	1123
FINAL DR.(2)	908
MOTOR (2)	1498
RADIATOR	900
FAN AND DRIVE(3)	416
DRIVE LINES	120
AIR CLEANER	150
BUS BARS	328
BATTERIES(6)	490
CONTROLS	270
BRAKE GRID	192
TOTAL	9,800 LBS

FUEL(7.0LB/GALX240) . . . . 1680 LBS

VOLUME(SYSTEM, INCL. FUEL). . . 180 FT^3



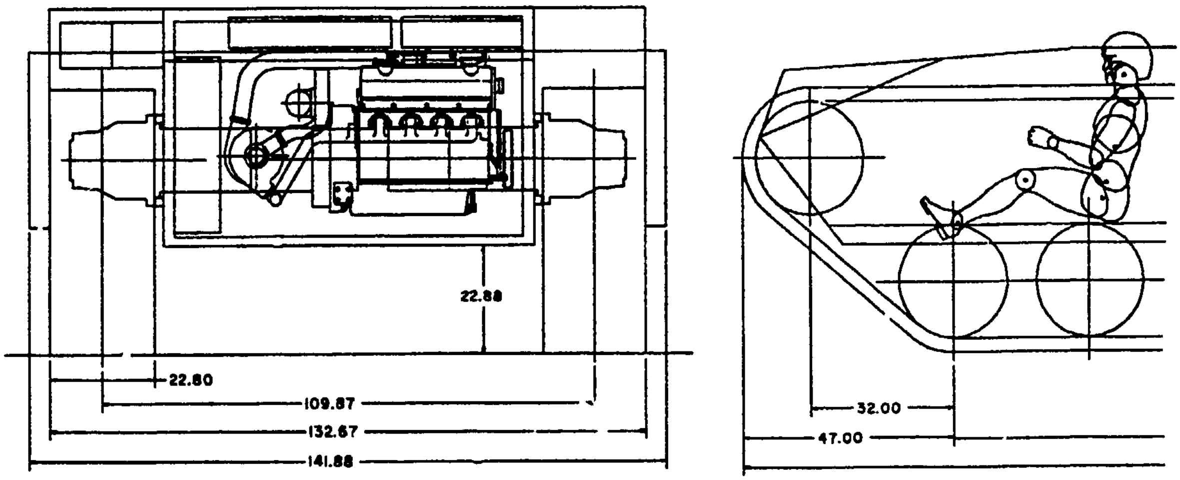


Figure 5-25. Homopolar (DC) System (40 Ton)

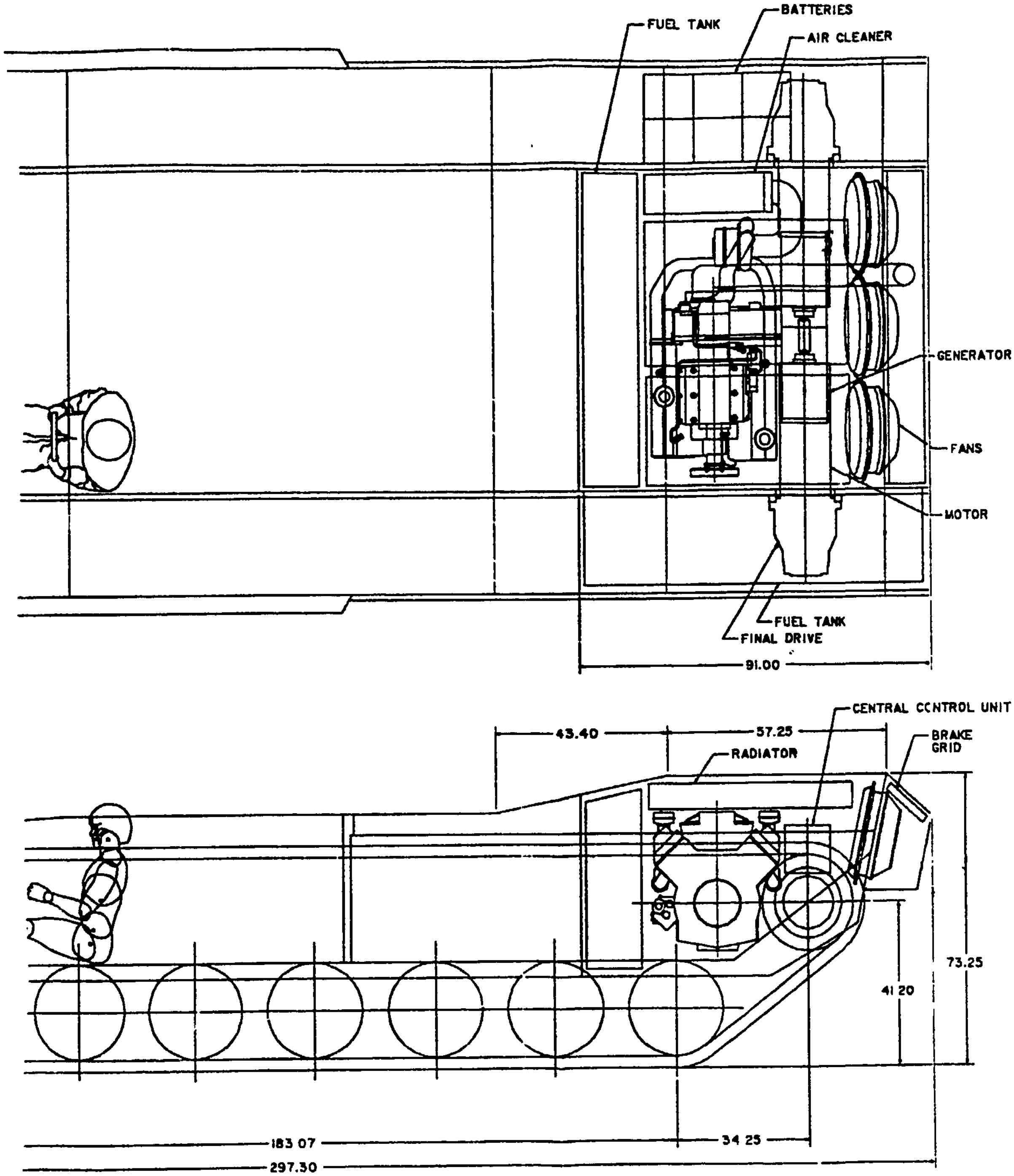


Figure 5-25. (Continued)

A speed increasing gear box is coupled directly to the engine to drive the high frequency induction alternator. The 5.38:1 ratio epicyclic gear train changes the 2600 RPM engine speed to 14,000 RPM at the alternator.

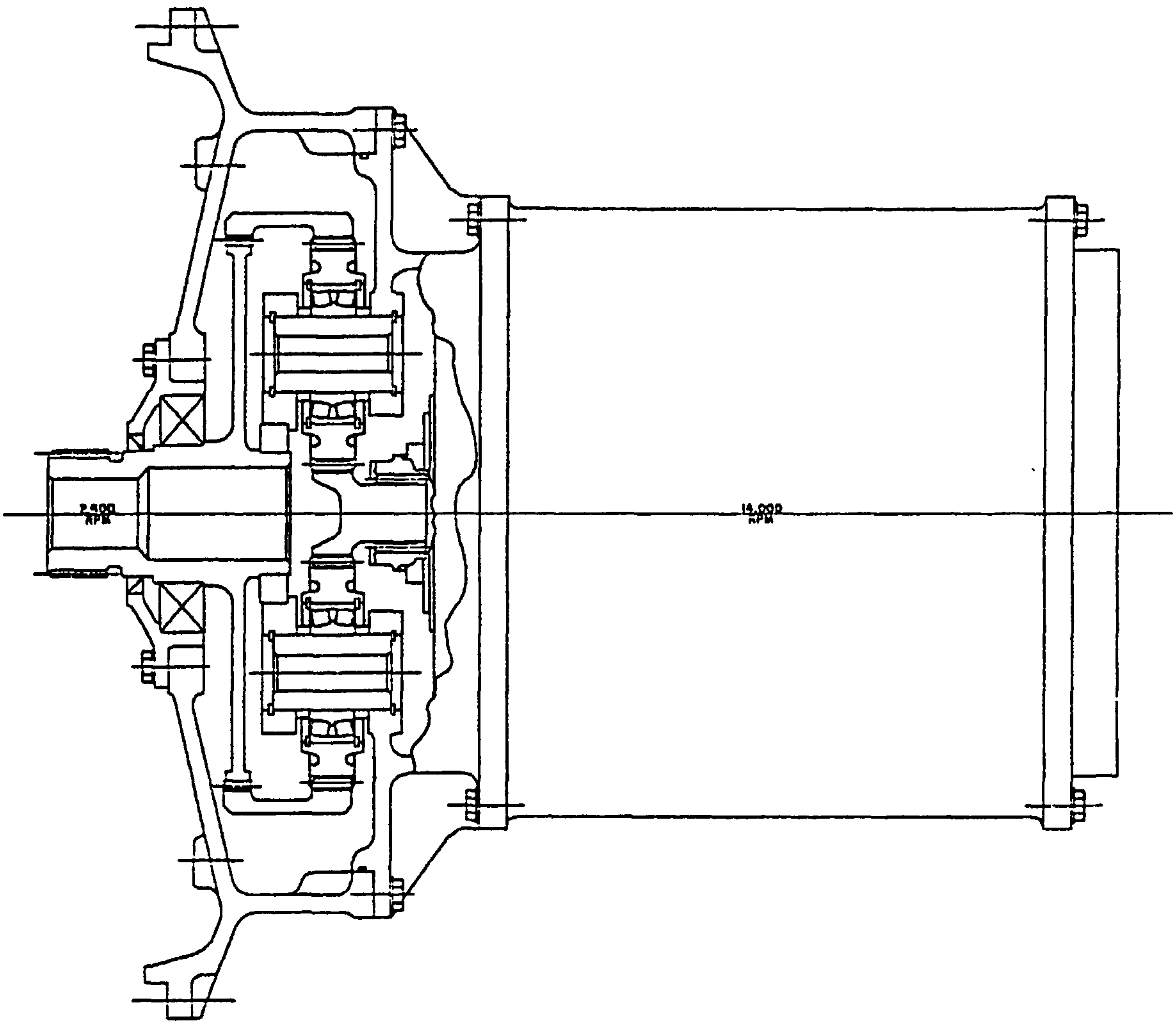


Figure 5-26. HF Induction AC Alternator Drive

The same speed increaser gearing is used for the generator drive as is shown for the induction system. A parallel shaft power take off is added to drive the vehicles electrical system alternator.

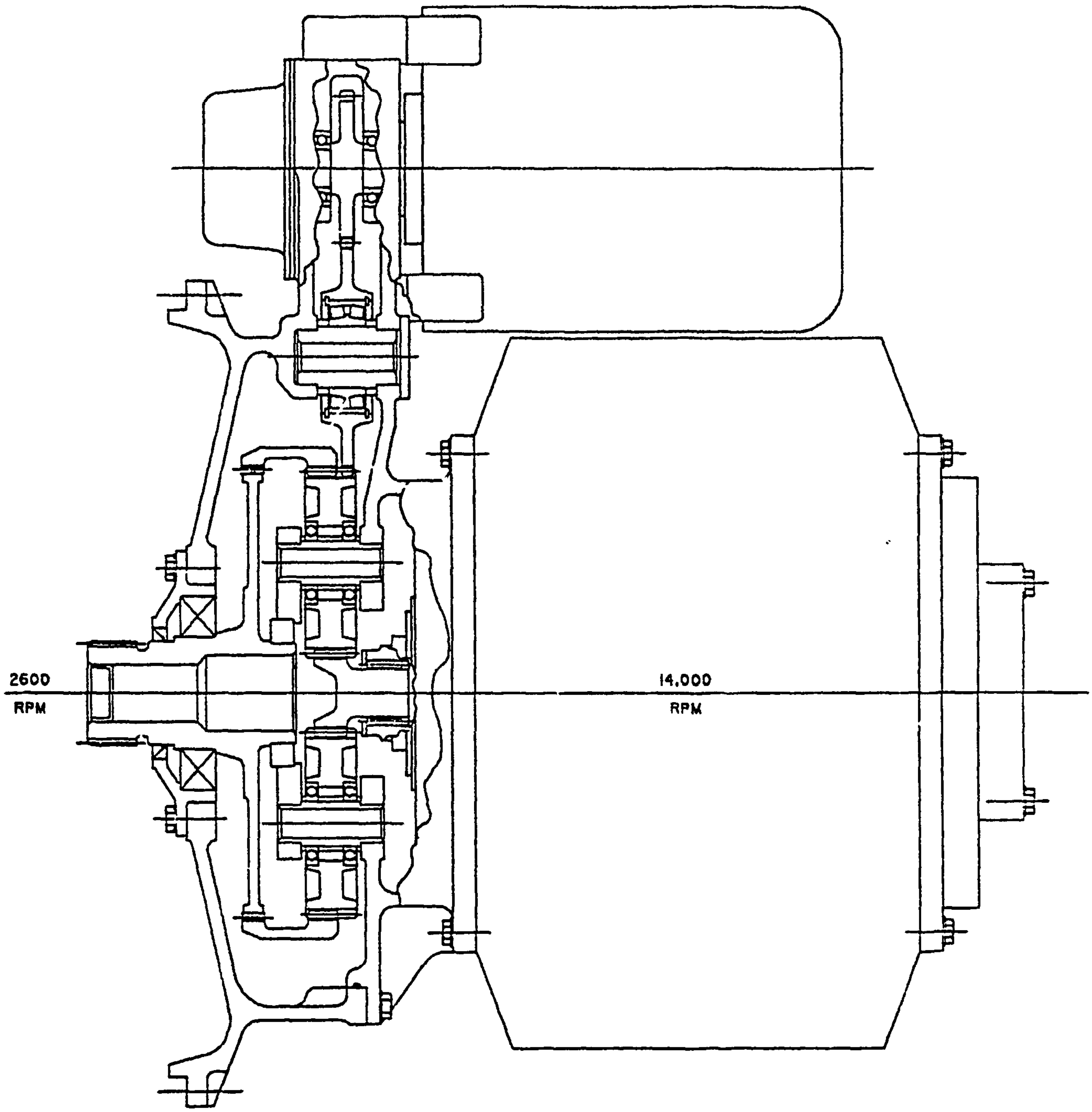


Figure 5-27. Homopolar Generator Drive

The homopolar system uses a wet brake located in the final drive, also on the low torque end of the gear train. Actuation is hydraulic and would be cooled by the gear box oil. Final Drives are dry sump, which requires filtered and cooled oil. Configuration of the homopolar systems eliminates the possibility of an external member for dry disk application.

5.6.1.6. Fuel Tanks. Fuel volume was based on the mechanical baseline vehicle requirements for a range of 300 miles. Since efficiency differences are unknown, all concept drawing tabulations show the same volume of fuel for a given vehicle weight. Fuel tanks for the 19.5 ton vehicle was not a problem because the basic shape of the vehicle has high sponsons which allow large regular volume shapes.

The 40 ton vehicle was more difficult because of the large volume required and the shape of the aft end of the vehicle. Sponsons were used as much as was practical, but the bulk of the fuel is stored in tanks located transversely across the hull.

5.6.1.7. Exhaust System. The exhaust muffler was deleted from the concept drawings because the size and shape of a muffler is difficult to package and was not germaine to the project goals. However, space for the other components in the engine exhaust system was provided in in order to remain objective. Effort was made to realistically separate the exhaust discharge from the air intake grill.

5.6.1.8. Final Drive Concept. Two types of final drive are shown for Configuration I. These are described below.

- o Split Power Flow. The split power flow design is shown in Figure 5-28. for the high frequency induction system gear box. Input from the motor is divided at the input gear set to apply torque directly to the output shaft at the input planet carrier and also to the ring gear driving the sun of the output planetary. Power from the output ring gear adds to the power from the input planetary to produce the total power at the sprocket. By this method the high torque normally applied to the low speed gears is reduced. This allows smaller rotating components than would be required if all of the torque were transmitted through each gear sets. Although the two ratios are not equally divided, the split is sufficient to cause an appreciable torque reduction in the low speed planetary. Engagement velocities are also reduced resulting in improved efficiency.

- o Simple Planetary. The second type gear box proposed is a more conventional planetary system which uses a triple gear reduction. Two speed reduction could have been used to give the required 21:1 ratio. However, three stages were

ELEMENT	A → D	D → S	A → S
RATIO	7.2105	1.7636	20.9269

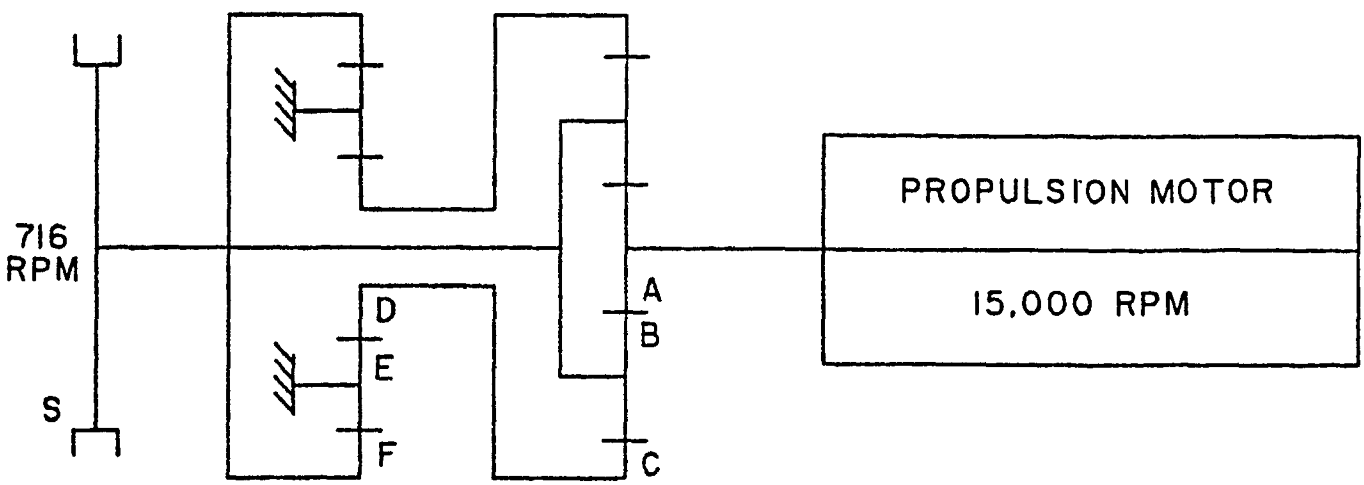


Figure 5-28. Final Drive with Split Power Flow (Configuration I)

selected because each ratio is then reduced which allows a better size proportion between the sun and ring gear and reduces the diameter of individual planets. Since system component loads vary with the number of planets, a large number of planets can reduce component size. Figure 5-29. shows the Homopolar Configuration I concept with triple reduction final drives in which the second stage uses three planets while the third stage has six. Torque varies directly with ratio so it can be seen that the high torque, low speed end of the system remains very small. Bearing loads are reasonable and gear face width is small because of many planets sharing the load.

5.6.1.9. Epicyclic Gear Systems. All of the gearing proposed in this study is of the epicyclic configuration. This is done for reasons of efficiency, weight and space. Packaging of this type of gearing is light when compared with a parallel shaft arrangement. Space is utilized more efficiently with all components on one axis. Ratio variations are flexible because the input to output elements can be easily changed and the number of stages is relatively easy to change. A single axis arrangement is also readily adaptable to direct mounting of the motor and/or alternator. If required, a parallel shaft or power take off can be adapted to this type of gearing as easily as on a parallel shaft gear box.

5.6.1.10. Sprocket Mounted Motor. It can be seen when comparing the sprocket mounted motor (Figure 5-30.) with the homopolar triple planetary final drive (Figure 5-31.) that further design study could develop an integrated motor/final drive (Figure 5-32.) which would enclose the final drive entirely within the sprocket carrier. This enhances the location versatilitry of the electric motors and increases useable vehicle interior space. A wet brake is included within the final drive to reduce the system complexity. It has been argued that the wet brake adds complexity, but that is only true if the cooling oil must be separated, filtered and cooled. Integration of the brake oil with final drive oil is possible if brake service is confined to parking and emergency only.

5.6.2. Configuration II Arrangement and Packaging.

The rationale for packaging components into concepts also applies to Configuration II arrangements. The concept for the 19.5 ton vehicle is shown at Figure 5-33.

Figure 5-34. shows the details of the drive design and sprocket carrier integration for the 19.5 ton Configuration II concept. Figure 5-35. is a schematic of the same gearing showing the ratios invloved to accommodate the 10000 rpm steer motor and the 15110 rpm propulsion motor.

ELEMENT	A → D	D → S	A → S
RATIO	7.2105	1.7636	20.9269

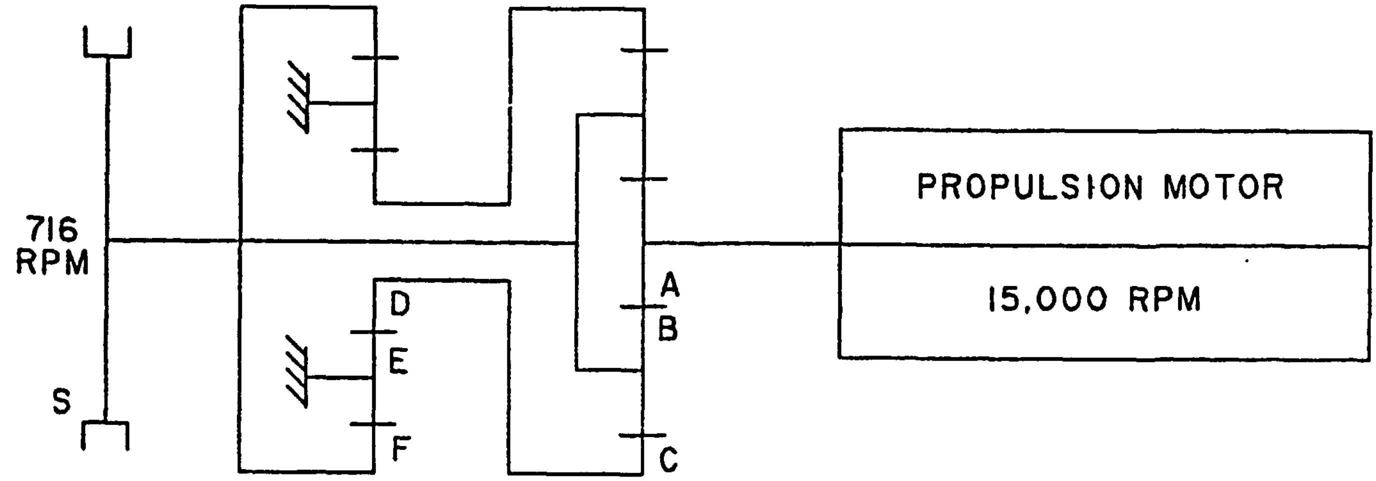


Figure 5-29. Final Drive Epicyclic System (Configuration I)

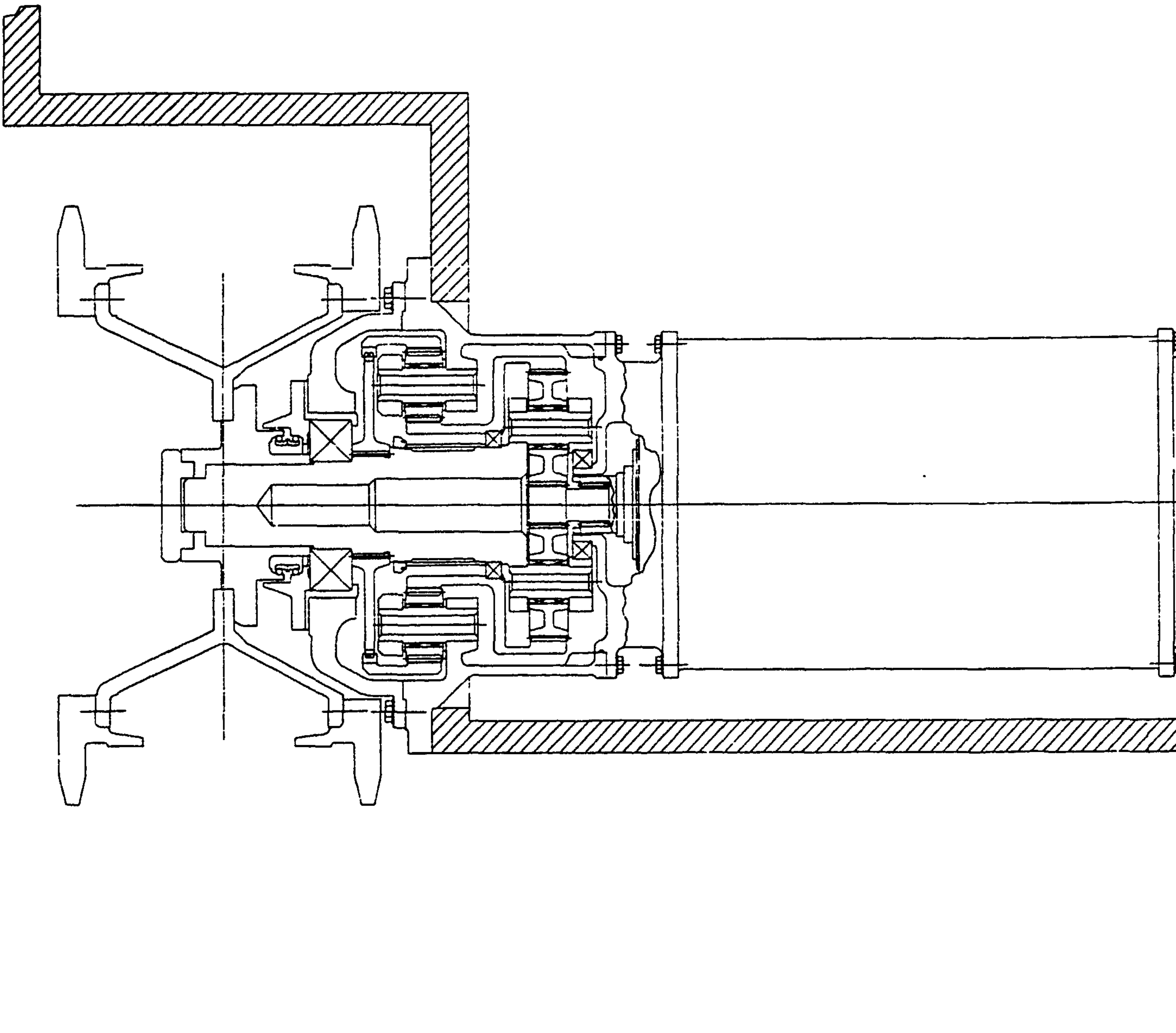


Figure 5-30. Sprocket Mounted Motor (19.4 Ton) (Configuration I)

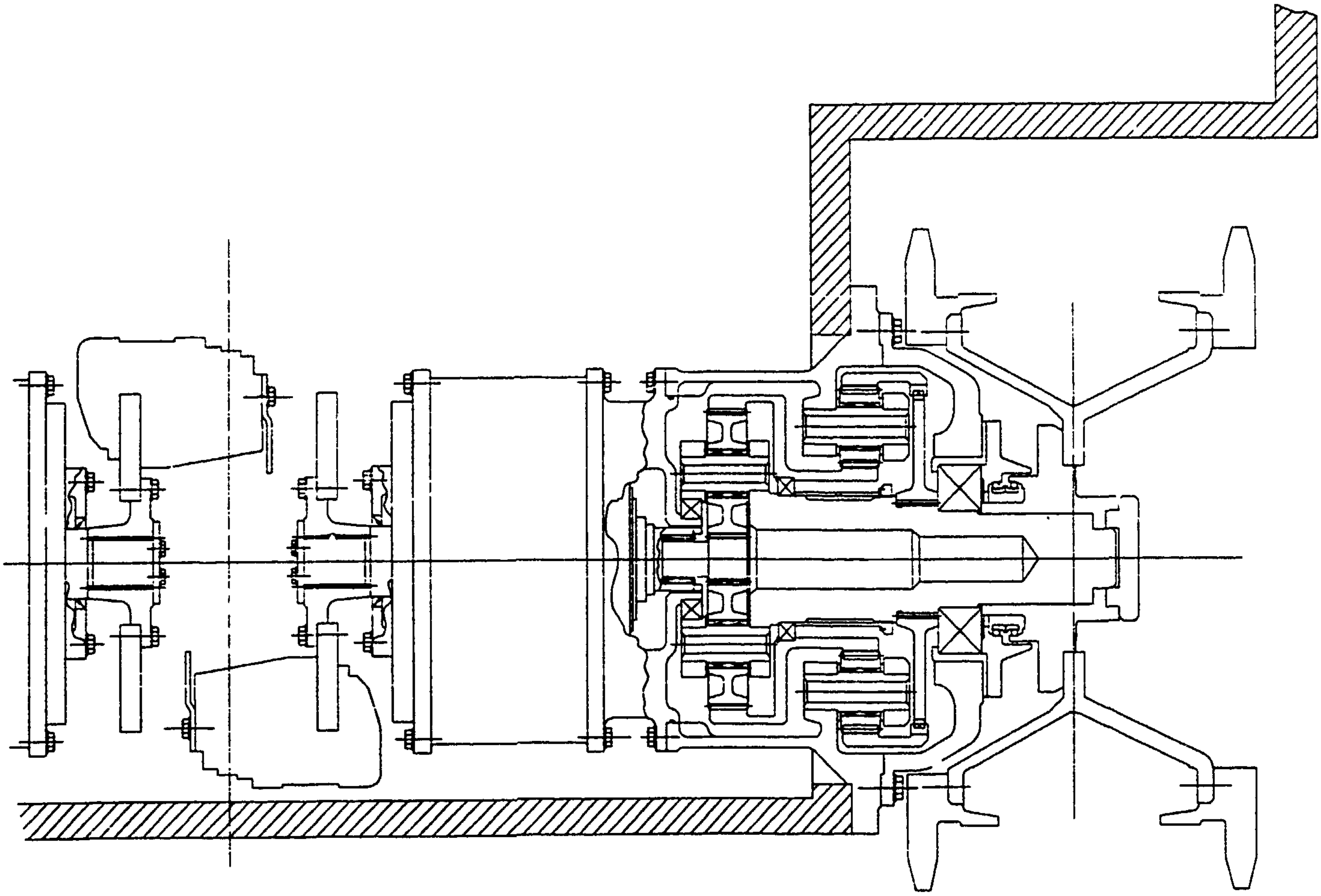


Figure 5-30. (Continued)

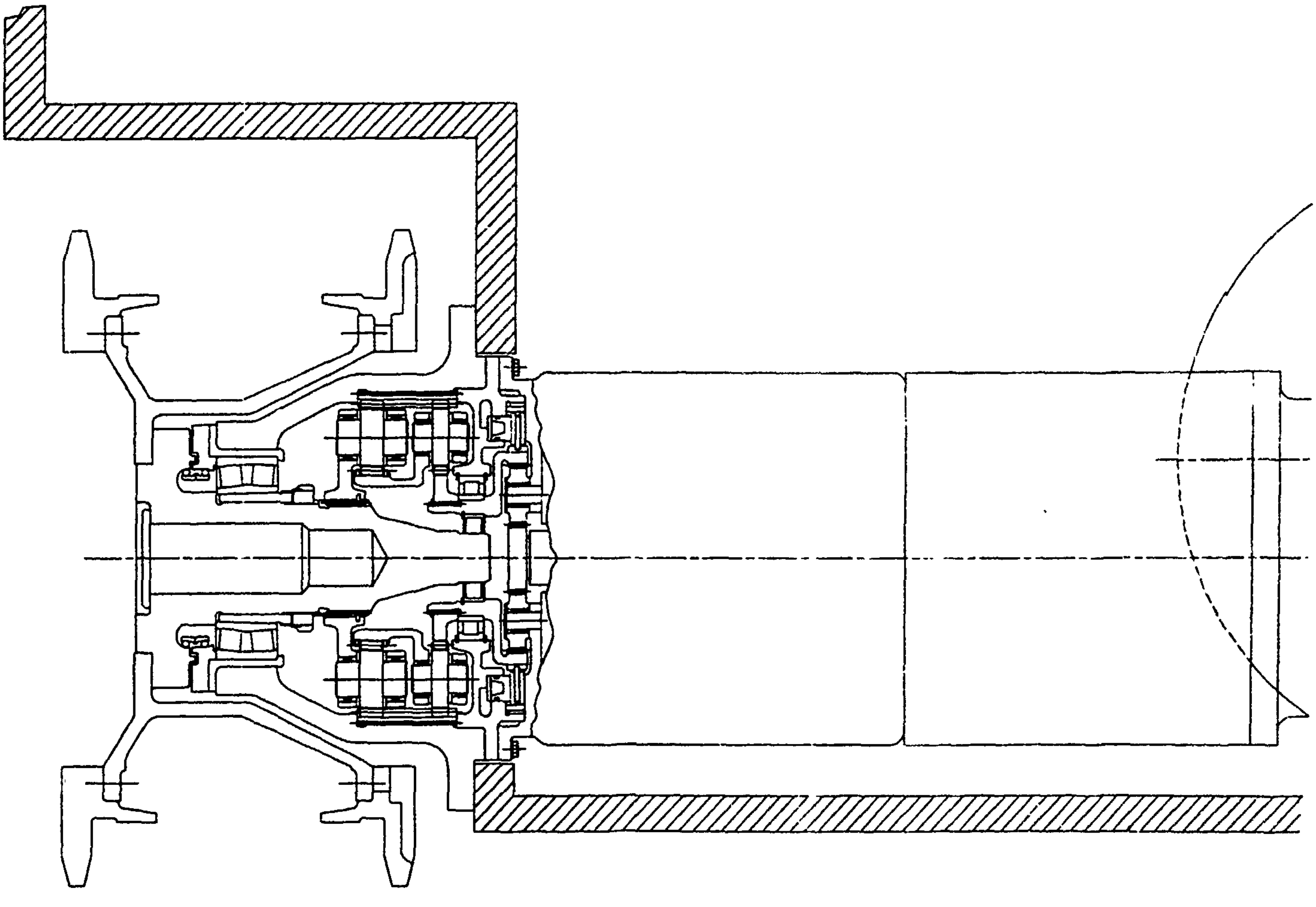


Figure 5-31. Triple Planetary Final Drive (19.5 Ton) (Homopolar)

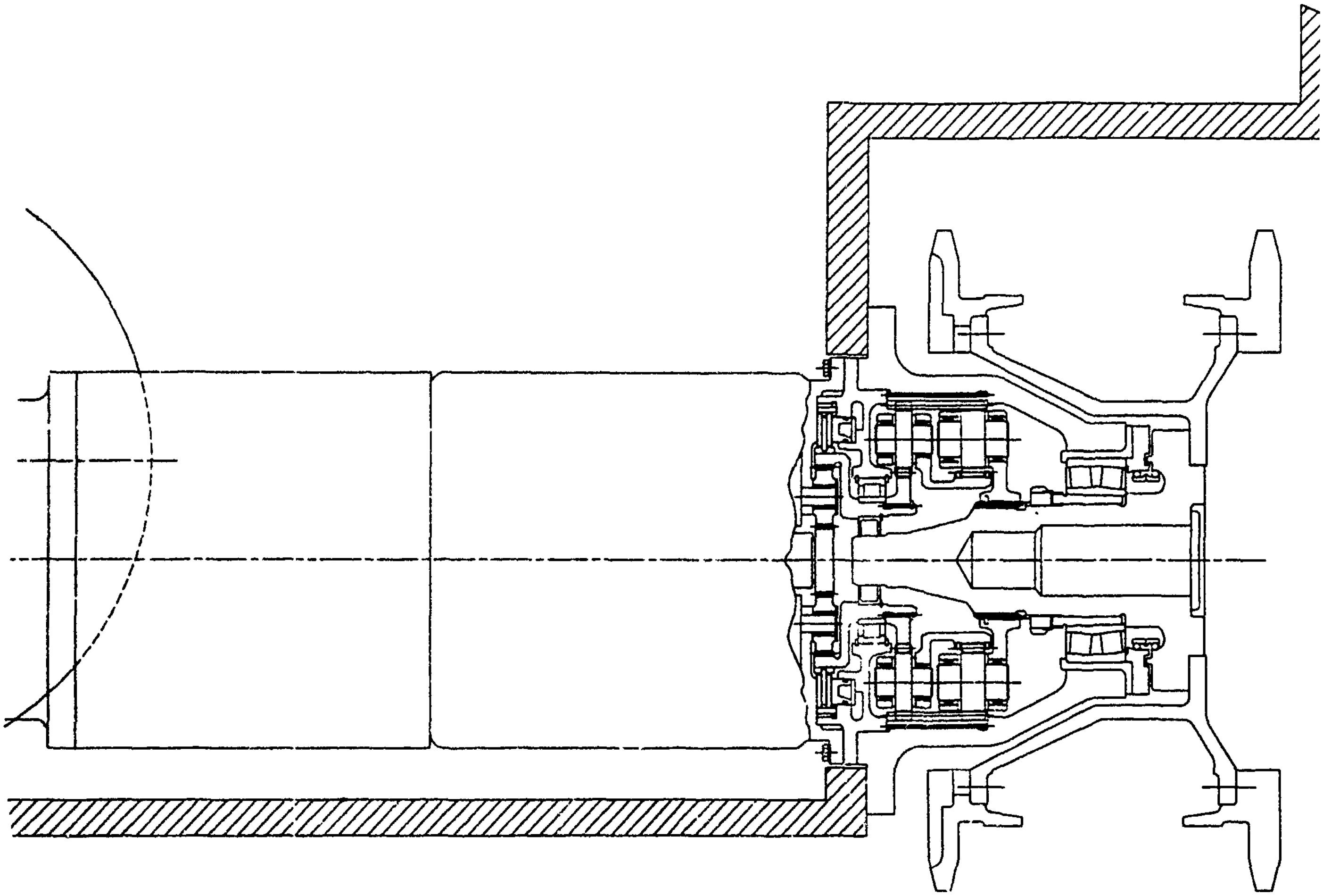


Figure 5-31. (Continued)

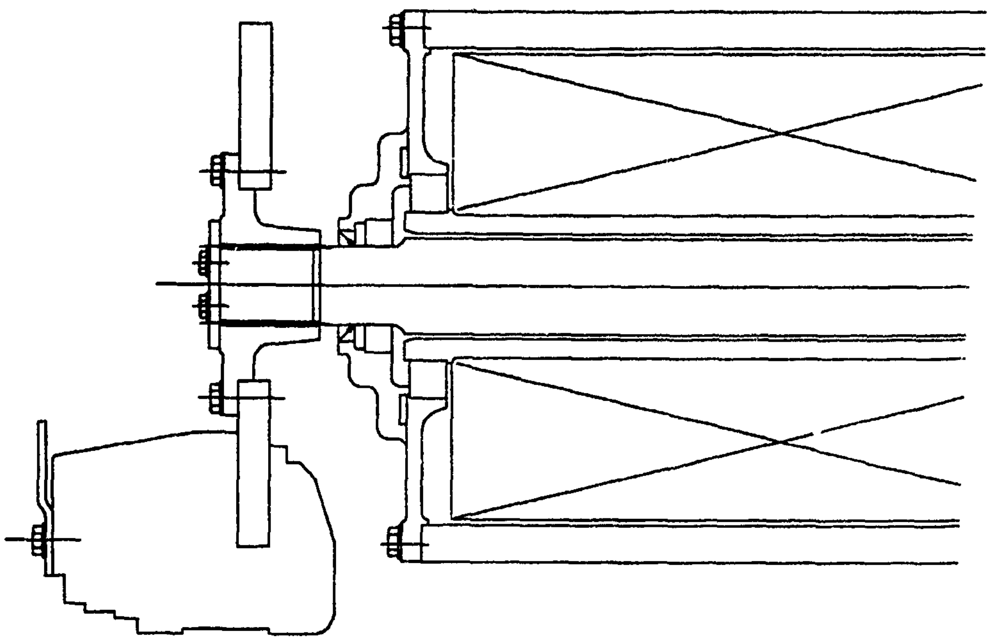


Figure 5-32. Integrated Motor/Final Drive

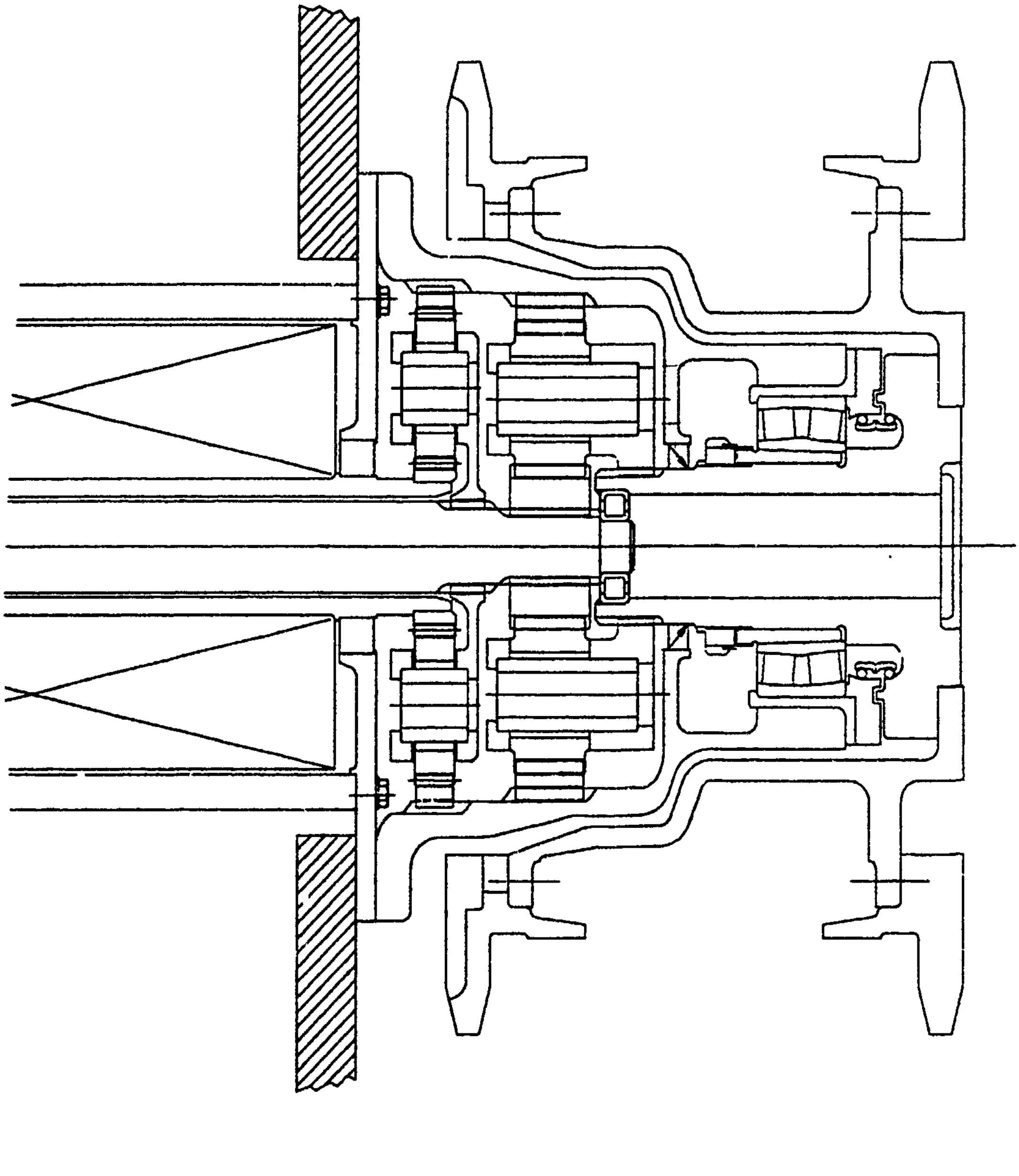


Figure 5-32. (Continued)

ELECTRICAL DRIVE SYSTEM

19.5 TON VEHICLE CONFIGURATION • II

PROPULSION SYSTEM DATA:

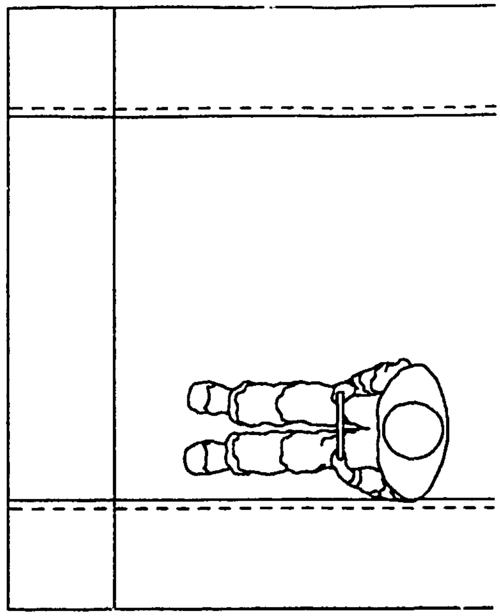
DESCRIPTION

- PROPULSION MOTOR AND STEERING MOTOR WITH SPEED REDUCING GEARS
- ALTERNATOR/GENERATOR COUPLED TO CUMMINS VT903 DIESEL ENGINE THRU SPEED INCREASING GEAR BOX.
- REAR DRIVE TRANSVERSE ARRANGEMENT

WEIGHT (WET)	HFIAC/HOPOL
ENGINE	2374
ALTERNATOR/GENERATOR	132/749
ALT/GEN/DRIVE	125
MOTOR (PROPULSION)	600/749
MOTOR(STEERING)	150/228
FINAL DRIVES (2)	160
PROPULSION/STEER GEARS	330
VEH ALTERNATOR	0/75
RADIATOR	536
FAN AND DRIVE	160
AIR CLEANER	110
BUS BARS	100/205
BATTERIES (6)	490
REGENERATION BRAKE GRID	120
CONTROLS	200/180
PARKING BRAKE	67
TOTAL	5654/6658 LBS

FUEL (7.0 LB/GALX160) . . . . 1120 LBS

VOLUME (SYSTEM)(INCL. FUEL) . . 139 FT^3



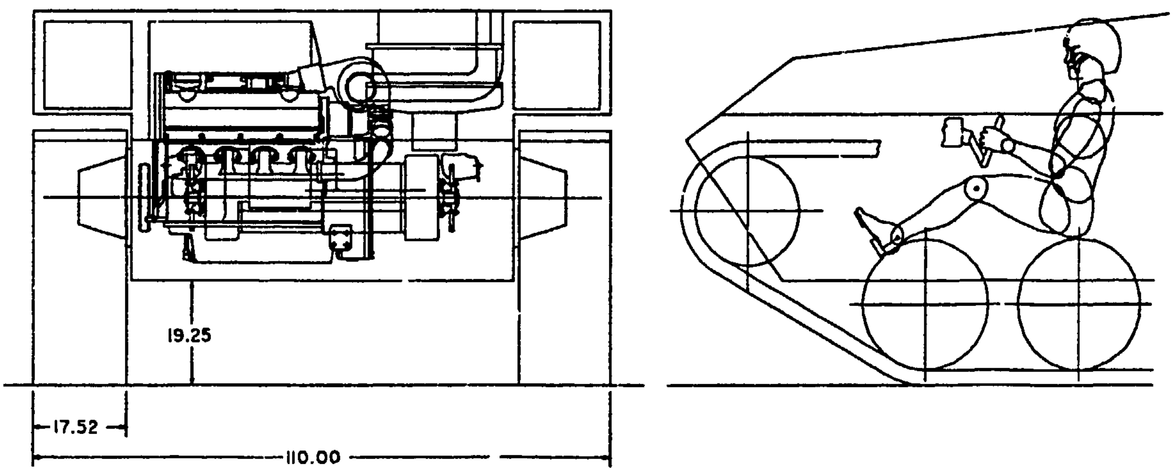


Figure 5-33. Configuration II, 19.5 Ton Vehicle

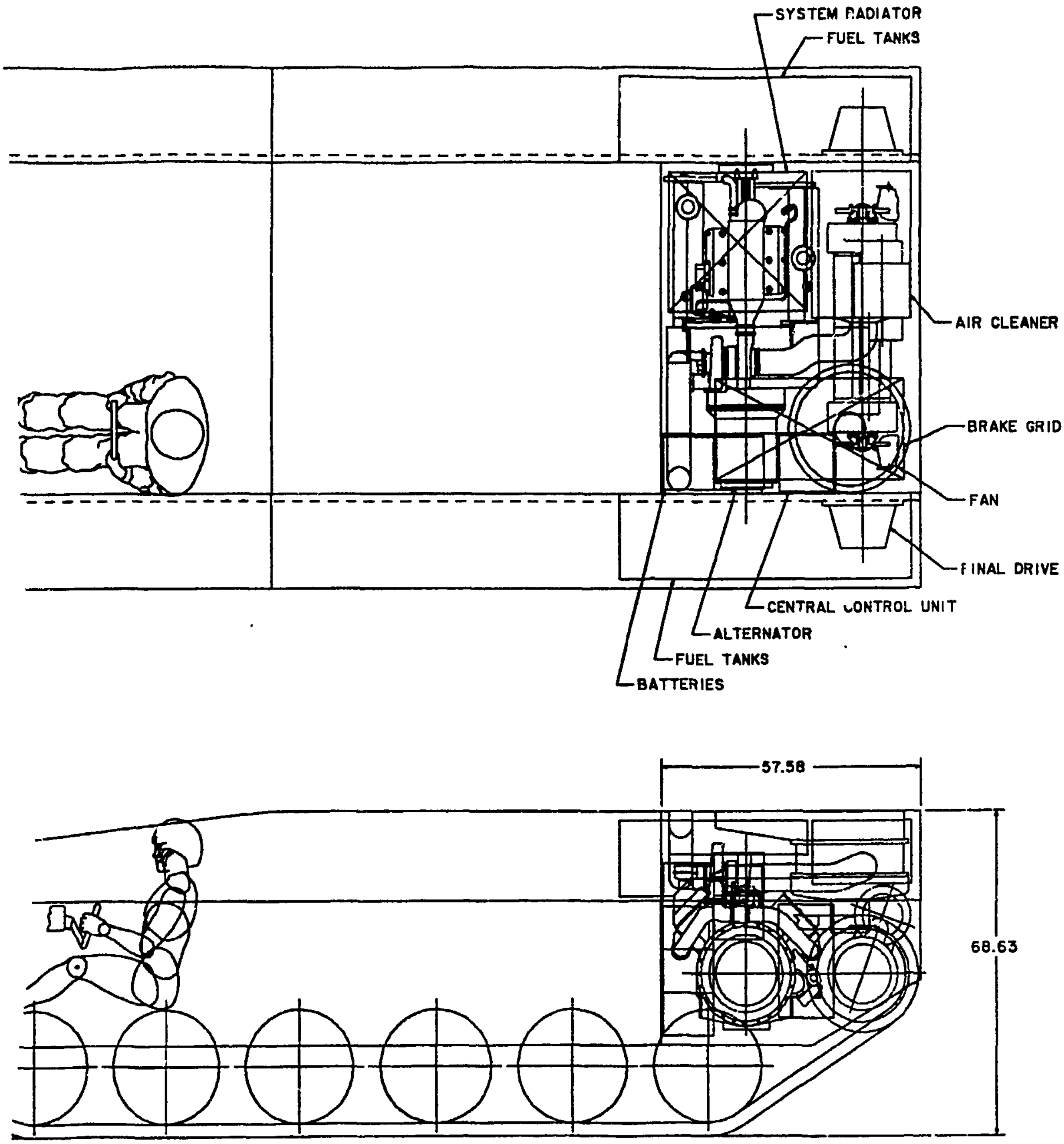


Figure 5-33. (Continued)

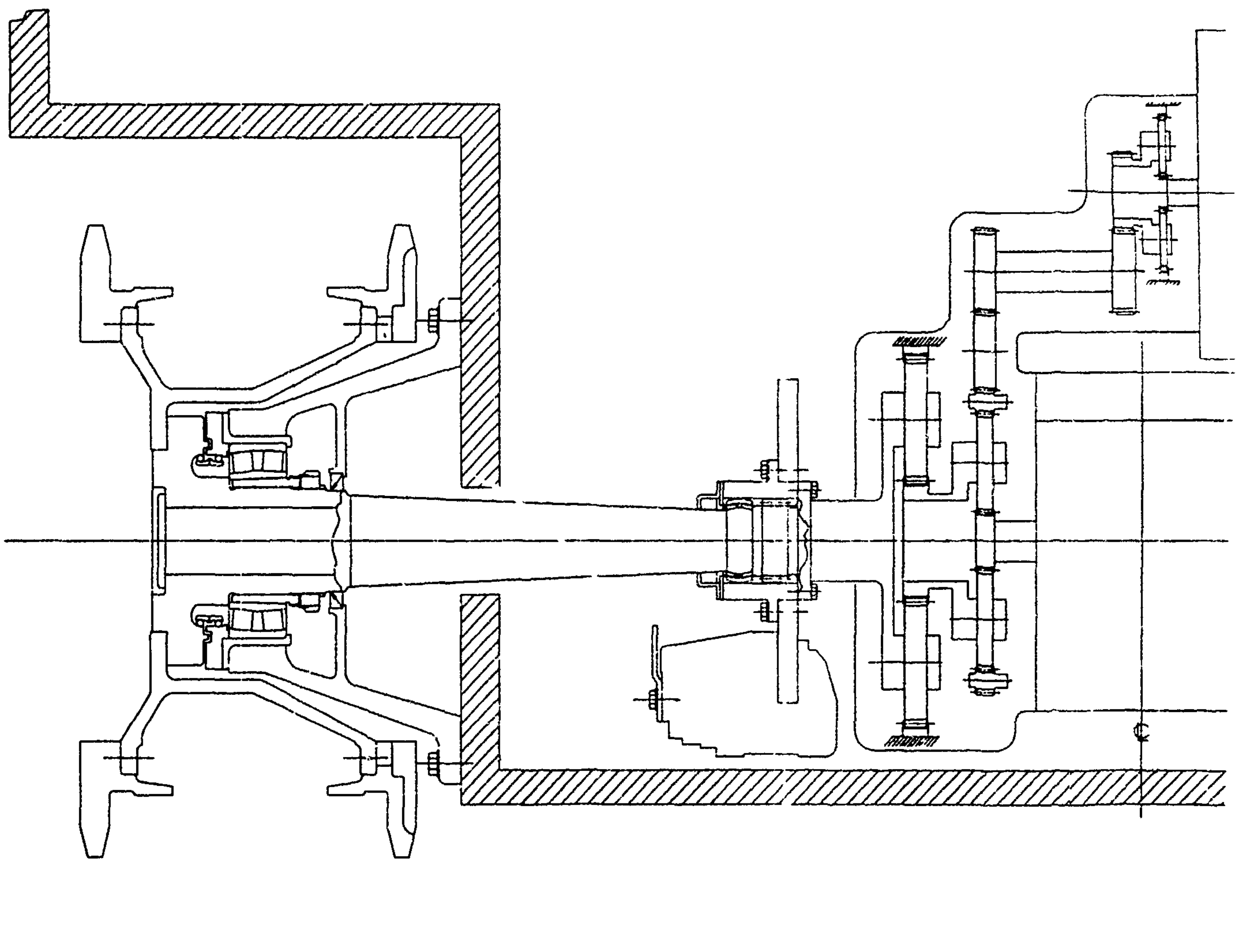


Figure 5-34. Drive and Sprocket Carrier Integration (19.5 Ton)
(Configuration II)

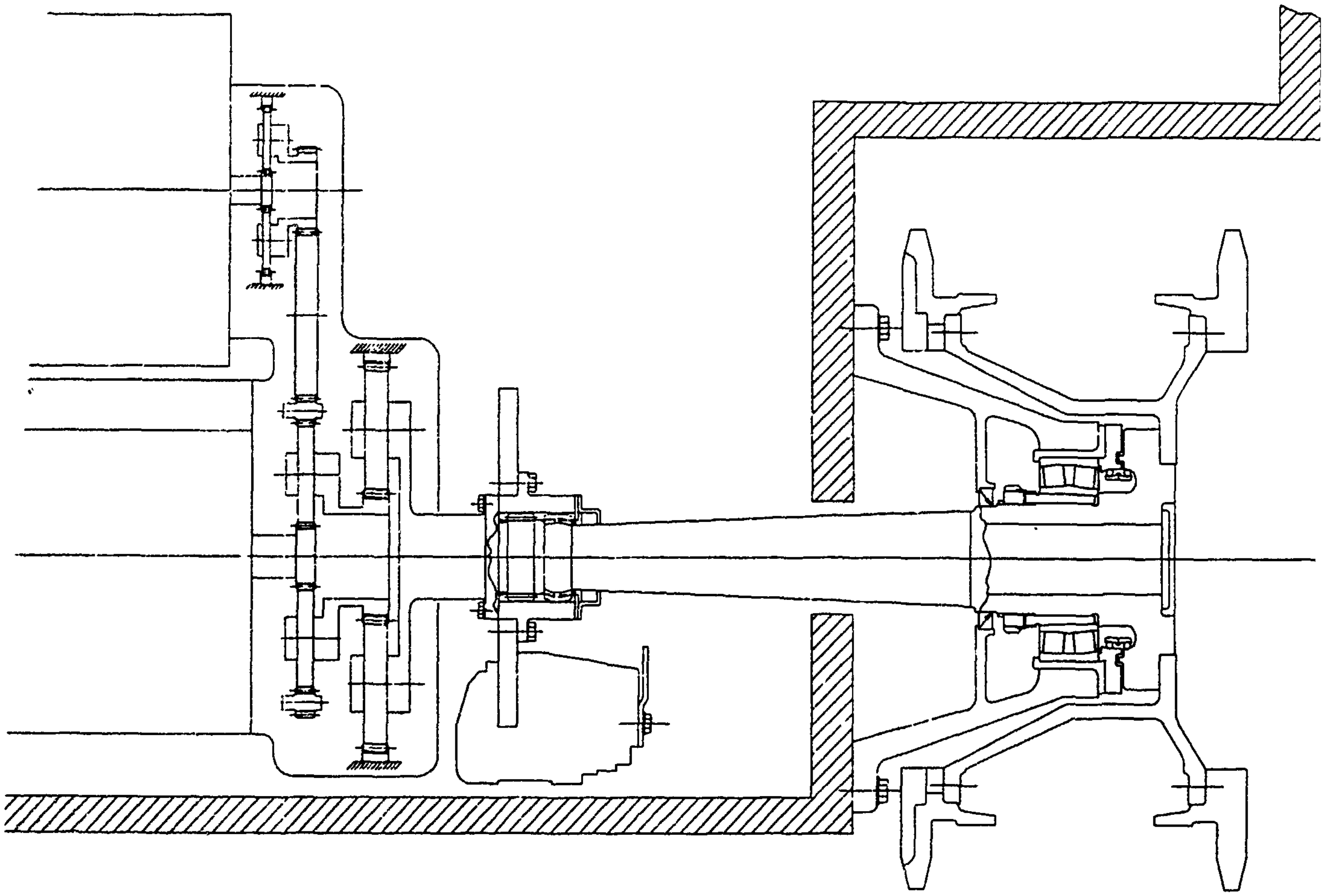


Figure 5-34. (Continued)

ELEMENT	A→D	D→S	A→S	G→K	K→L	L→D	D→S	G→S
RATIO	5.2727	4.000	21.092	5.2727	3.0276	1.234	4.000	99.616

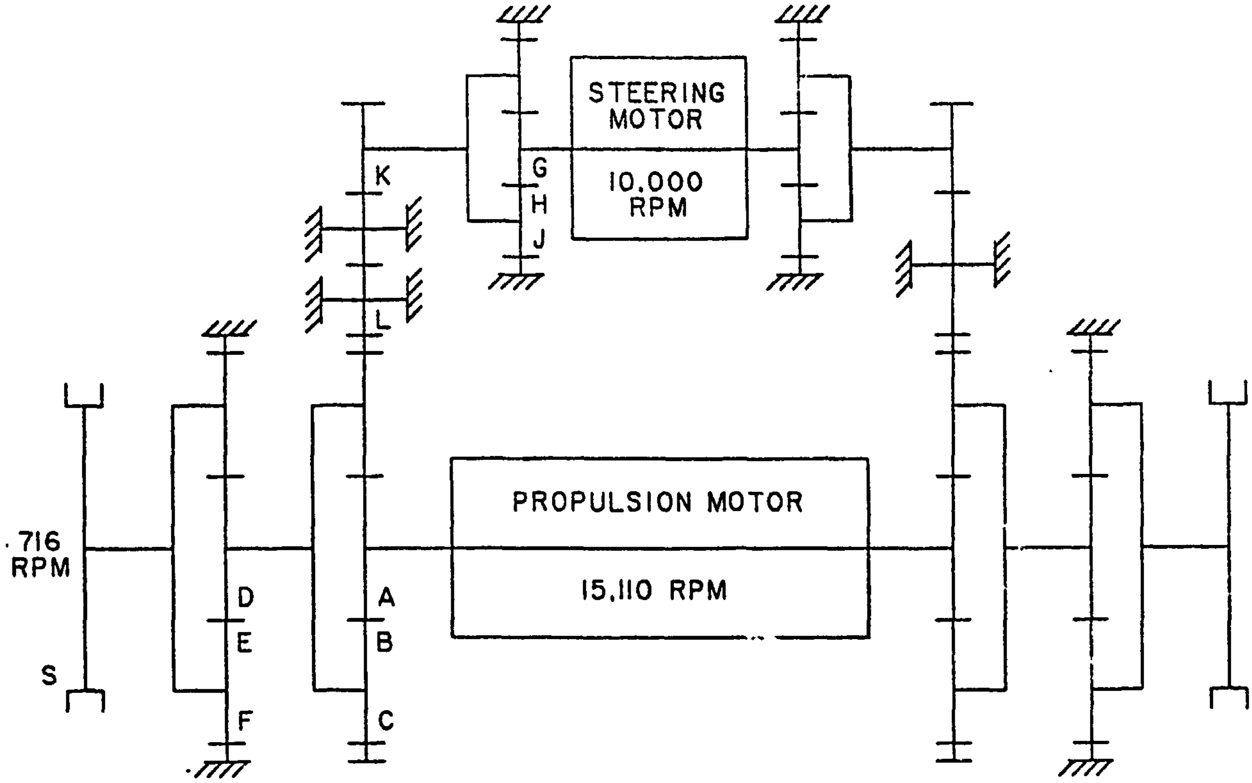


Figure 5-35. Propulsion and Steering Gear System (Configuration II)

5.6.3. Configuration I - Homopolar Motor System. The dual-independent homopolar drive shown in Figure 5-36. is the selected DC propulsion system for application in the Configuration I drive train. Presented in this figure are all the power path and control elements which are integral to the drive concept. Components not shown include those associated with the cooling system and peripheral prime mover accessories (i.e. fuel pump, radiator, etc.). This system uses two identical velocity feedback servo systems to provide a drive train for each track which is totally independent from the other within the constraints of total available vehicle power. Each primary power path receives its input from the diesel engine through a split output step-up gearbox. This drives each of the homopolar generators, whose armatures are connected to those of the respective drive motors on either leg of the system. Motor output power is coupled to each sprocket via a final drive gear reducer (A detailed description of homopolar machine operation is available at Appendix G as additional information).

5.6.3.1. Vehicle Control Concept. Control of each independent drive is achieved through the forward gain and feedback elements. Since the system is configured for sprocket velocity control, the sprocket feedback element is a tachometer. The system could be configured in a torque control mode, in which case a strain gauge type torque sensor would be substituted for the velocity transducer.

5.6.3.2. In operation, the velocity and turn (steer) commands are developed by the driver interface transducers (i.e., accelerator pedal and steering wheel) and are input to the velocity order generator. Differential velocity orders are generated for each track from these commands. When each velocity servo-loop receives its order, a determination is made as to whether or not that track is turning at the appropriate speed. This information is provided by the sprocket tachometer. If the commanded and the actual speeds are not the same (as in the case of an accele-ration or deceleration), the difference between the signals (the error) is amplified by the gain element (K) and passed through the stability compensation network to the coordinated field controller. This system element is responsible for determining the level of field excitation applied to each of the homopolar machine fields to most efficiently achieve the required speed/torque load point. The appropriate signals are applied to the field exciters which act as power amplifiers to drive the fields of the homopolar machines. As the homopolar motor responds to adjustments in its field and armature voltage (due to the generator field adjustment) the tachometer monitors and relays these changes back to close the servo-loop. While in this transient state, the system error is constantly being reduced by the servo-loop's response to the change in velocity order. After a given time period, which is unique to each servo-loop, the system error is reduced to a minimum steady-state value and the track speed matches the velocity order.

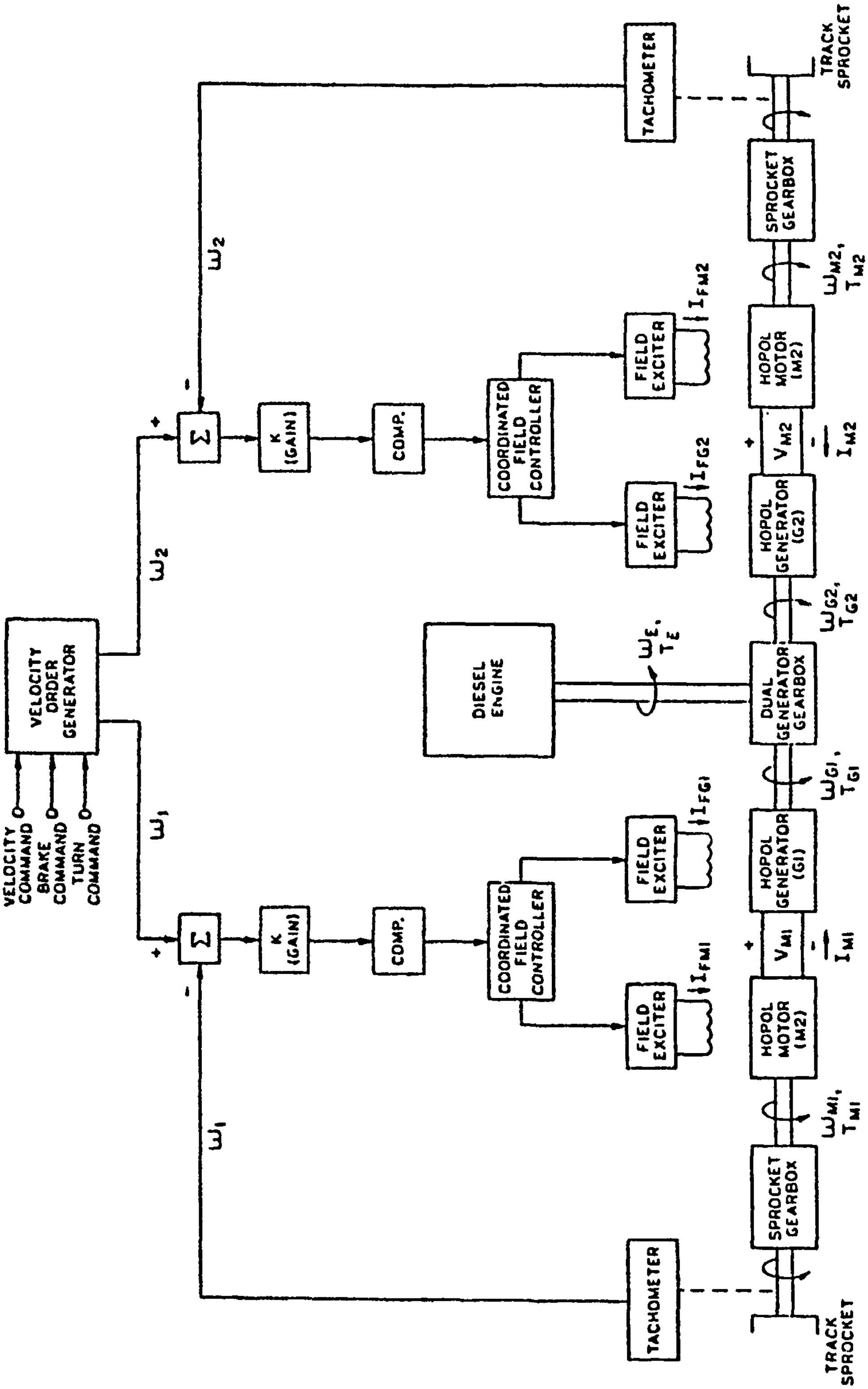


Figure 5-36. Dual Independent Homopolar Drive

5.6.3.3. System Attributes. Although the operation of the velocity servo in the dual independent drive is a common configuration, there are some unique attributes which this system can offer as a vehicle propulsive system:

- Independent Track Control. Each track control system is isolated from the other to the degree that there is adequate power available to each track from the prime mover. There is not interdependence as with either the parallel or series homopolar systems which utilize a common generator.

- Common Motor/Generator Design. Since each generator only needs to supply a single motor of equivalent power, each machine can be of a common design. This will simplify repairs and maintenance of the drive.

- System Voltage Compatible With Vehicle Operating Voltage. The inherent low voltage nature of the homopolar machine would allow for direct excitation of the drive motors from the vehicle battery bank in the event of a short term emergency propulsion requirement.

- Integrated Motor/Generator Design. Due to the common power requirements of the motor and generator, the resulting common size, and the need for two generators, each motor-generator pair can be integrated into a stand alone unit. This results in a smaller overall machine package and simplified maintenance.

- Flexible Engine Scheduling. Either constant or variable diesel engine speed can be accommodated by the system due to the closed-loop nature of the coordinated field controller. This presumes that sufficient power is available at the engine speed to overcome all system losses so that the required power can be delivered to the sprockets.

- Independent Speed/Torque Control. The flexibility of the homopolar machine field control characteristic coupled with the coordinated controller allows for the speed and torque of the machines to be controlled nearly independently of the loads imposed on the system.

- Dead Engine Steer Capability. Unlike any vehicle drive system which requires a series controller, the homopolar drive offers dead engine steer capability through excitation of the appropriate machine fields to execute the desired maneuver. The control is provided through the fields; and the propulsive power from either a downward grade or a towing vehicle.

- o Mechanical Regeneration Link. A mechnical link is provided between the track drives in this concept. During regenerative turns, downhill operation, or dynamic braking, power is transferred from the sprocket, across the electrical link and directly across the mechanical gearbox link to either the prime mover or opposite sprocket, depending on the maneuver.

- o Flexible Machine Field Excitation. Either solid state (semiconductor) or rotary (motor-generator set) field excitation may be used to control the fields of the homoplar machines.

- o Dead Engine Dynamic Brake Capability. As with the dead engine steer condition, the machine fields can be excited with stored energy from the vehicle operating system batteries, allowing the vehicle to be dynamically braked without the use of power from the prime power mover. All systems with a series controller require that the controller be energized to control the rate of vehicle braking.

5.6.3.4. Vehicle Drive Operational Considerations. The power flow which occurs in the dual-independent homopolar drive during typical vehicle manuevers is illustrated in Figures 5-37. through 5-42. In addition to straight ahead motion, five particular modes that involve a reversal or rerouting of power are considered:

- o Forward/reverse motion
- o Non-regenerative turning
- o Regenerative turning (single track)
- o Regenerative turning (dual track)
- o Neutral-axis-steer
- o Dynamic braking

5.6.3.5. Forward/Reverse Motion. Operation of the dual independent drive in a straight ahead, forward mode is accomplished by providing a selected velocity command and a net turn command equal to zero to the velocity order generator. Since there is no differential velocity commanded (turn command equal to zero) each track servo responds in the same manner to provide field excitation which yields equivalent track velocities. The power flow which occurs in such a maneuver is shown in Figure 5-37., which focuses on the system elements in the power path of the drive. Power flow direction is given by the accompanying arrows, and the relative magnitude by the length of each. In this mode of operation, both track power flows are identical.

5.6.3.6. Turning Maneuver. There are three power flow conditions that may result from a forward, or reverse vehicle turn. For each of these maneuvers the general response of the velocity servos is essentially the

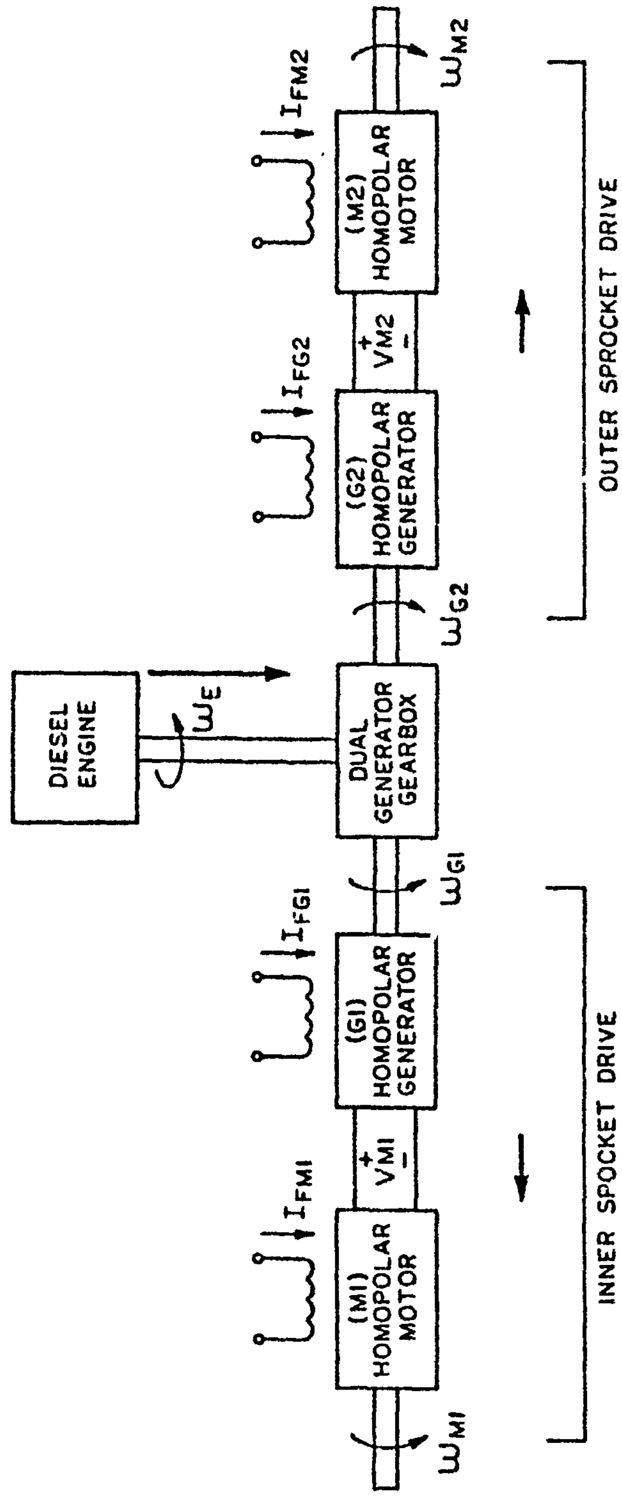


Figure 5-37. Dual Independent (DI) Homopolar Drive Power Flow Straight Ahead Motion

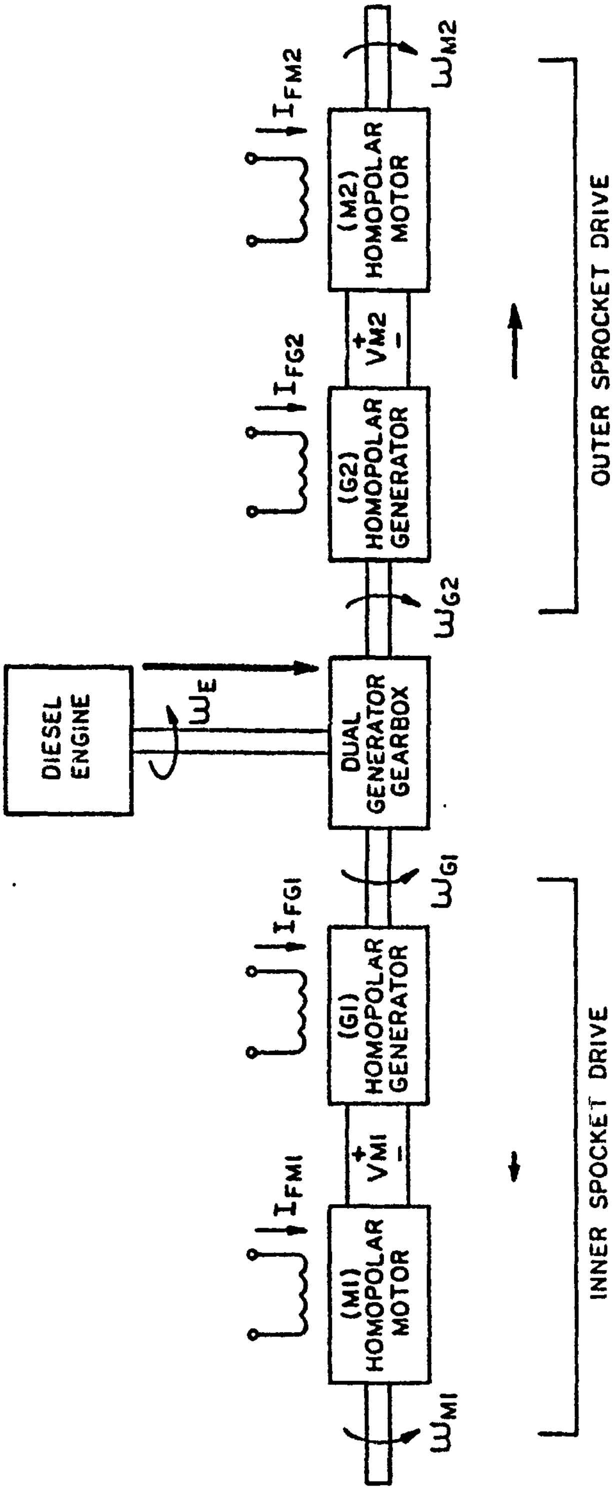


Figure 5-38. DI Homopolar Power Flow, Non-Regenerative Turns

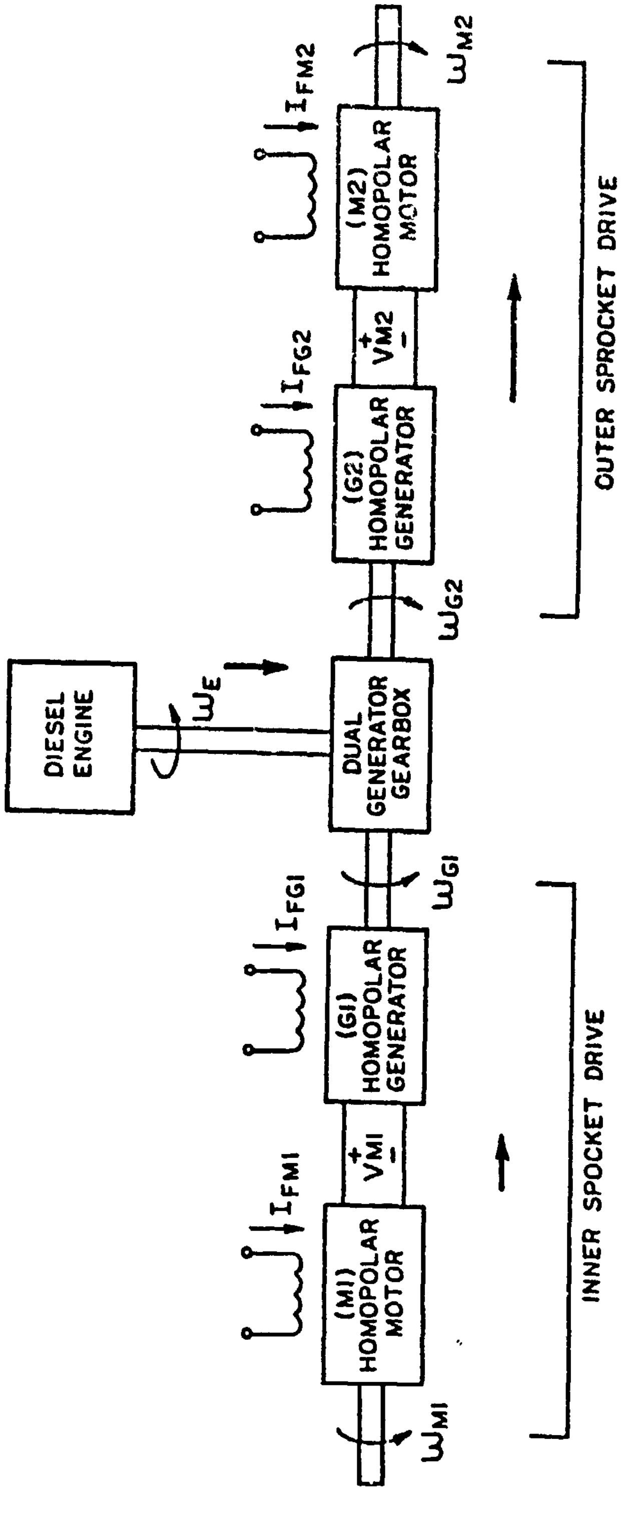


Figure 5-39. DI Homopolar Power Flow, Single Track Regenerative Turns

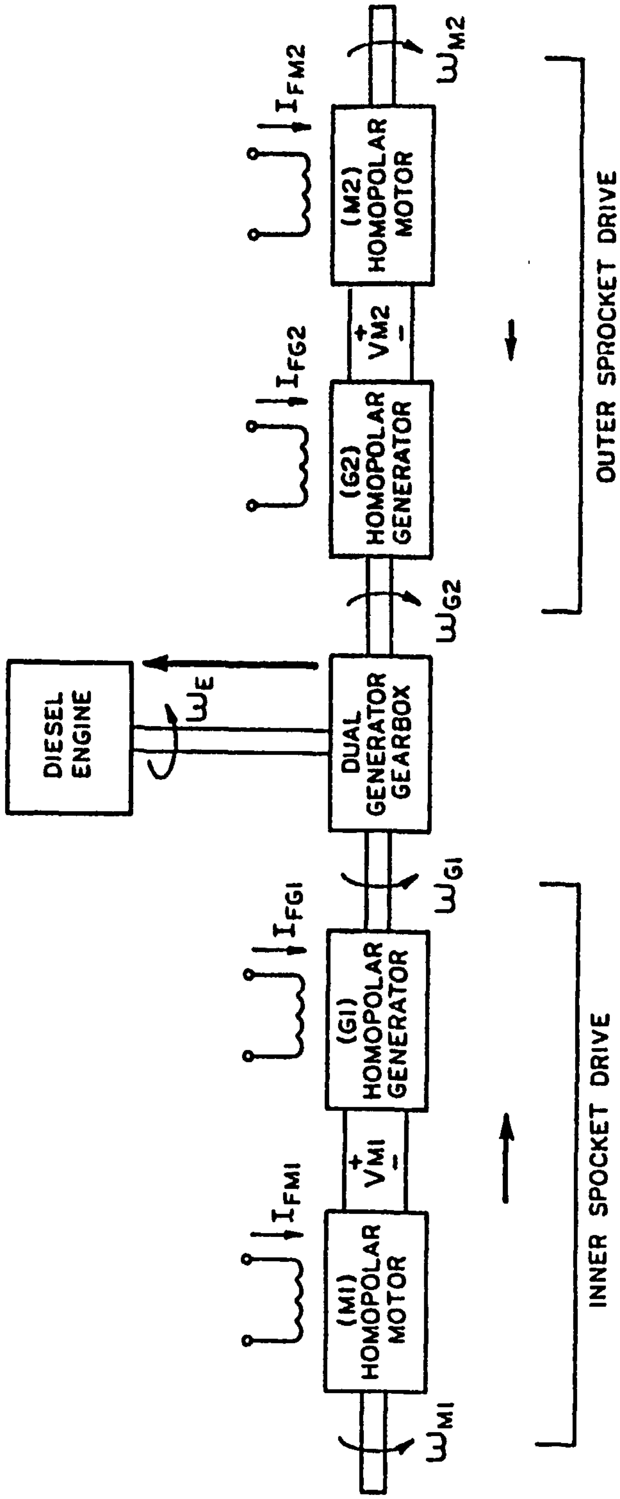


Figure 5-40. DI Homopolar Power Flow, Dual Track Regenerative Turns

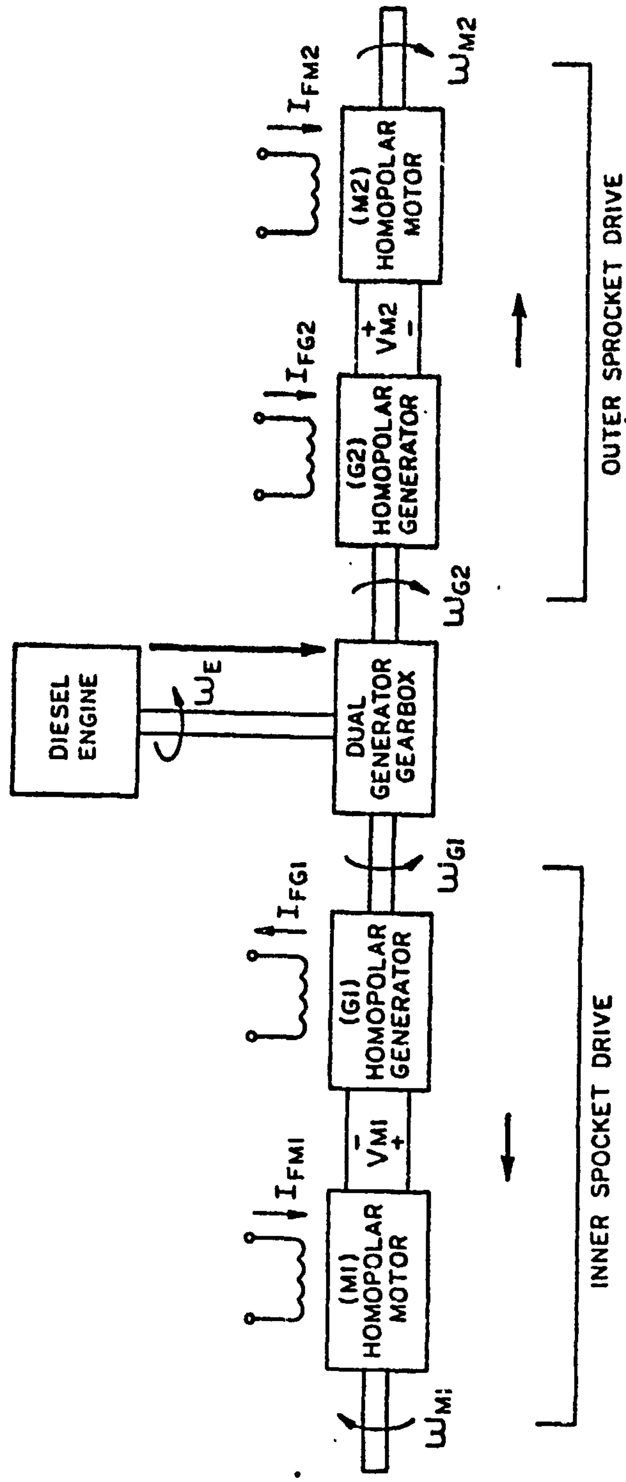


Figure 5-41. DI Homopolar Power Flow, Neutral Axis Steer

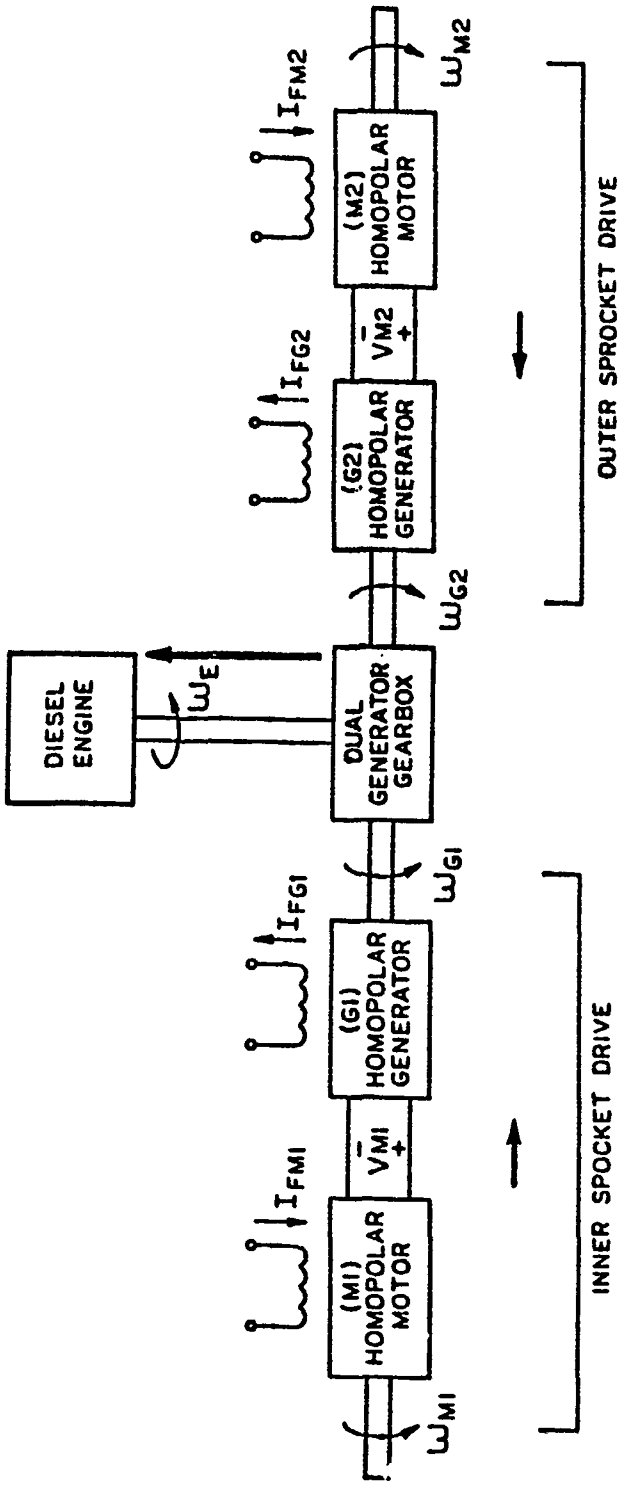


Figure 5-42. DI Homopolar Power Flow, Dynamic Braking

same. A velocity command which represents the average velocity of the two tracks is input to the velocity order generator, as well as a turn command which defines the differential velocity between the two tracks. Depending on a number of parameters, including velocity, grade of terrain, and vehicle configuration, one of the three power flow situations will occur. In all three cases, however, the polarity of machine field excitation will remain the same as for straight ahead, forward motion. Only the magnitude of the excitation will be different.

5.6.3.7. If conditions exist where the vehicle is executing a broad steer on a relatively level surface, a non-regenerative turn condition will occur. The resulting power flow is shown in Figure 5-38. Prime mover output power is split at the dual generator gearbox, and a net differential power flow out to each track will exist. The majority of the power is delivered to the outer track which must supply the higher velocity. If a tighter turn is executed over similar terrain, net power on the inner track will at some point go through zero and reverse. Under this condition, power from the inner track is transmitted through the homopolar machines, and across the mechanical gearbox link where it supplements the prime mover power supplied to the outer track. The power flow which results from this single track regenerative turn is given in Figure 5-39. Increasing the grade of the terrain sufficiently for either of the above maneuvers will result in a dual track fully regenerative turn. This situation is illustrated in Figure 5-40., where the net power to each track has reversed and is being regenerated through the dual gearbox to the prime mover. In such a case, the prime mover must be capable of absorbing this reverse energy, or it may be dumped to a resistive grid which would be switched across the electrical link between each pair of homopolar machines. Turning is not a requisite for power regeneration. As long as grade and vehicle conditions exist for a negative track power flow, regeneration will occur in turns as well as for forward, straight ahead motion.

5.6.3.8. Neutral-Axis Steer. A neutral-axis-steer maneuver requires that the velocity of each track be identical in magnitude, but in opposite direction. This is accomplished by supplying equal, but opposite velocity commands to each track's closed-loop servo. These commands result in one of the homopolar machines in each motor/generator pair having its field excited in reverse polarity from its mirror machine on the opposite track drive. Both motor and generator field excitations cannot be reversed, since the net effect would be to negate any change in rotational direction. Since the power delivered by each track to the ground is the same regardless of the direction of track rotation, the neutral-axis-steer power flow diagram given in Figure 5-41. is identical to that for the forward, straight ahead mode, with the exception of a reversed field excitation polarity.

5.6.3.9. Dynamic Braking. Dynamic braking is initiated when it is desired to slow the vehicle's present rate of speed by actively driving the sprocket motors in the reverse direction from their present rotation. It is accomplished by first initiating a brake command to the velocity order generator which supplies a reverse velocity order to each tracks servo-loop. Since both track tachometers maintain their present rotation, the system error which results causes the field excitation of each of the homopolar generators to reverse. This has the effect of driving the sprocket motors in reverse, causing a negative power flow from each track back through the dual gearbox to the prime mover. The power flow diagram is given in Figure 5-42. As with dual path regeneration, the power supplied to the system in excess of the reverse capacity of the prime mover must be dissipated in resistive power grids.

5.6.3.10. Vehicle System Power Audit. Homopolar machine efficiency is governed by both mechanical and electric losses as are all rotary electric machines. Friction and windage losses are the primary mechanical impacts and electrically, both the brushes, the field resistance, and the armature resistance are the major contributors. How these losses relate to affect the overall homopolar system efficiency is the subject of figure 5-43. In this figure, forward power flow through the system is examined at full sprocket motor speed and maximum output power. Power losses due to component inefficiencies are shown as takeoffs from the major power path elements. These system losses are further broken down into their constituent components.

- o Prime Mover Losses. The prime mover has been defined to have a gross power capacity of 500 HP. There are three primary types of losses: engine losses, motor cooling losses, and motor controller losses. Engine losses are those which are associated with the prime mover directly or as power take-offs for other than propulsion system use. These include inlet/exhaust losses, the power required to circulate the engine coolant, the alternator for charging the starting-lighting-ignition (SLI) system, and the power to drive the low level system controls. Motor cooling losses account for the power required from the engine to circulate coolant through each homopolar machine (both motors and generators). The final loss is associated with the field excitation of the homopolar machines. The particular numbers given in efficiency map reflect a homopolar machine gain of 25.

- o Generator/interface Gearbox Losses. The prime mover power remaining is passed to the homopolar generators through the generator gearbox. Losses associated with this box include the friction of surfaces acting against one another, and the viscosity losses of the gear lubricant. This system element is very efficient, transferring the great majority of its power through to the homopolar generator. Generator losses include those previously undentified, with other inefficiencies attributable to mechanical losses.

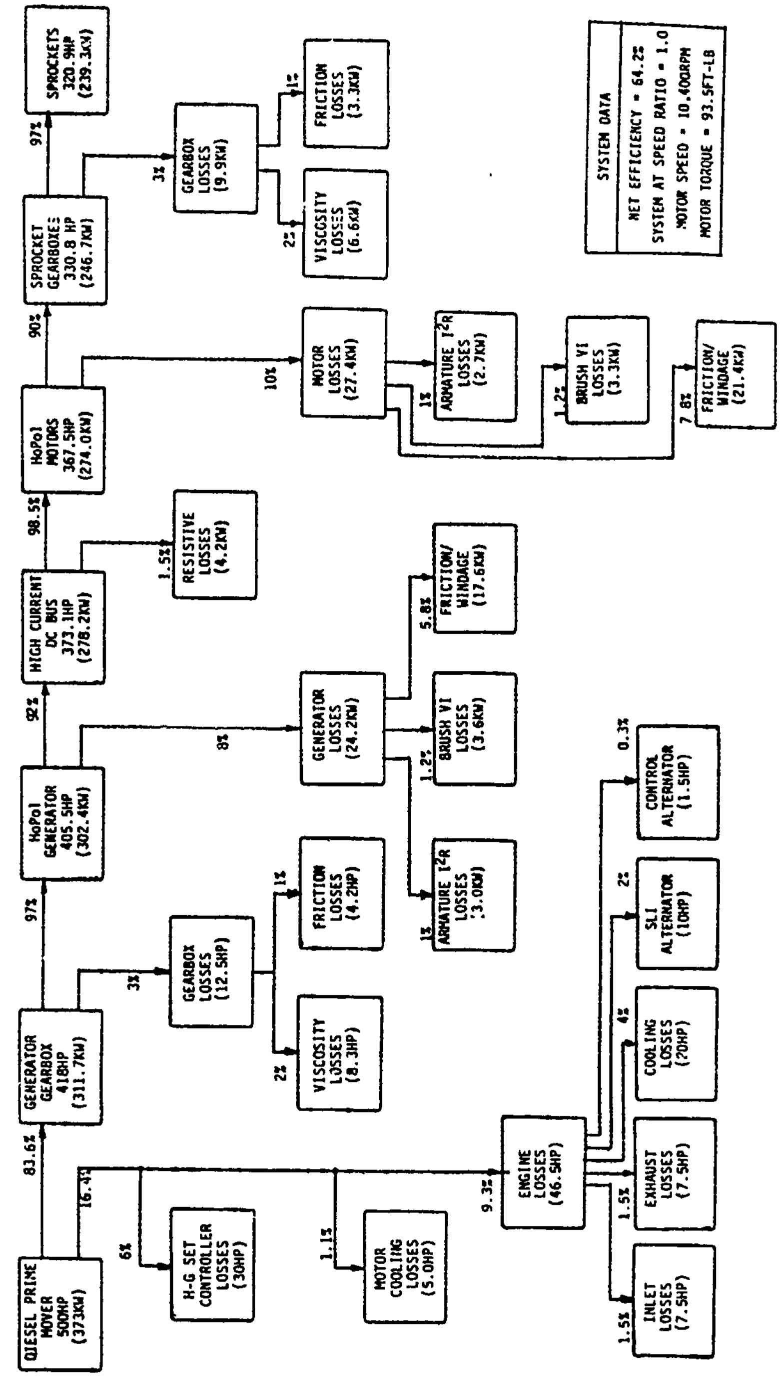


Figure 5-43. Homopolar Motor System Efficiency Map

- Motor/DC Bus Losses. A high-current DC bus forms the electrical link between the homopolar generator and motor. This link is very efficient, contributing less that 2 percent loss to the system. The homopolar motor to which it connects yields higher losses of the same type as the generator.

- Final Drive Losses. Final output power is transmitted from the homopolar motors to the track via the sprocket gearbox. This element exhibits roughly identical behavior as the generator gearing and its losses are proportionally the same. The net efficiency shown for the homopolar system at full power load from prime mover output to the sprockets is 64.2% based on the loss assumptions. Several refinements in motor design have shown the capability of raising this efficiency to approximately 70 percent. This is at a speed ratio of 1.0 (14000 rpm homopolar motor speed).

5.6.3.11. Homopolar Machine Interface/control

- Field Excitation/control. There are several options available to provide the field excitation for the homopolar machines. The most relevant include two varieties of solid-state power supply and a rotary power converter.

- Solid-State Excitation. Solid state power supplies/converters may be divided into two broad categories; linear and switching. Linear power supplies rely on the solid state device which is inserted in series between an unregulated DC source and the load. By selecting the operating point for the load and monitoring the load power, the solid state device is adjusted to take up the difference between the source and the load to maintain the selected operating point. The primary advantages to this type of supply are high reliability and extremely low EMI. A major disadvantage is the heat generated by the series pass semiconductor and the heat sink required to remove it. This is in contrast to a switching supply in which the pass element is alternately switched on and off. This yields a time averaged operating power for the load which correlates to that desired. Since semiconductors dissipate the least amount of power when turned fully on, the switching supply yields significantly lower losses that the linear supply. The EMI generated from switching, however, is the primary disadvantage. For high power (more than 5KW) requirements, the losses associated with the linear system cause the switching supply to be the logical choice for a solid state homopolar field exciter.

- Rotary Power Excitation. An alternative to the solid state field exciter is one of rotary design. Figure 5-44. illustrates a possible configuration for this converter which would be powered from the vehicles operatng system. As shown, a shunt wound or permanent magnet motor is used to turn an equivalent power DC generator with a separately excitable field. The motor is run at a constant velocity, such that the armature voltage developed by the generator is controlled by the excitation of its field. Since the field of the homopolar machine is connected to the armature of the control generator, the actual control power required to drive the field exciter is reduced by the product of the gain of the homopolar machine and the gain of the control generator. This reduction could be as high as 1,000, such that the control power for a 200 HP homopolar machine would be as little as 150W. Either a small linear or switching power supply could easily supply the required power. The chief advantage of the rotary field exciter is the use of the control generator gain to reduce the volume and weight required to control the homopolar system.

- As discussed in Appendix G, it is possible to use the homopolar machine over the complete "square" speed/torque characteristic. This presumes that adequate thermal cooling capability is provided to the machine. The cooling does not present a problem, but machine operation in this mode is not the most efficient. Since the machine can operate at the corner of the "square" characteristic it can be thermally sized to accommodate this maximum power. When in any region of the characteristic other than the corner, its full power capabilities are not being used. However, if each machine is operated along a constant power contour, as shown in figure 5-45., the maximum use is made of each machine. This results in the smallest and lightest machine possible.

- In the dual independent drive, the coordinated field controller determines at what speed point the sprocket motor must operate and the corresponding load point from the generator to deliver the required bus voltage. From this information, the necessary field excitation for each machine is determined and implemented through the exciter. In this way, neither machine must be sized to handle high torques at both low and high rotational velocities. The two machines work together in an integrated manner, utilizing variable field on both machines to realize the full potential of each. An optimized control strategy is then achieved.

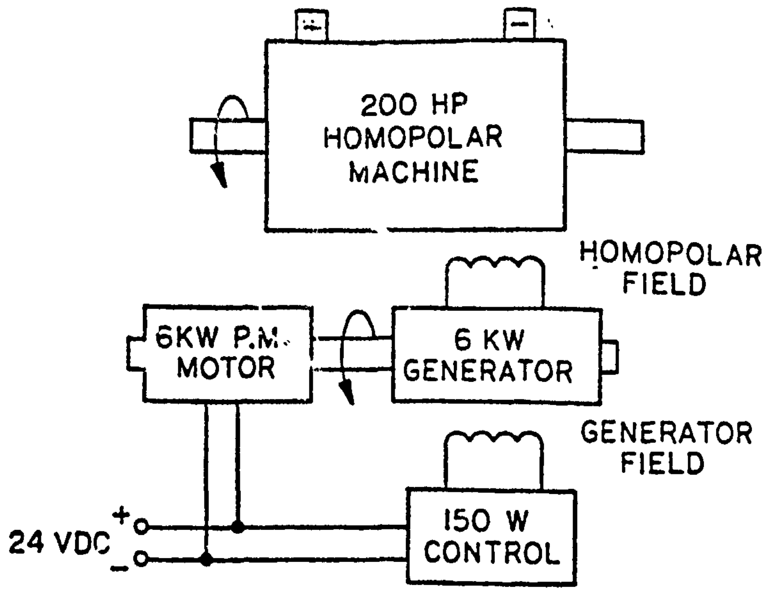


Figure 5-44. Potential Rotary Field Exciter Configuration

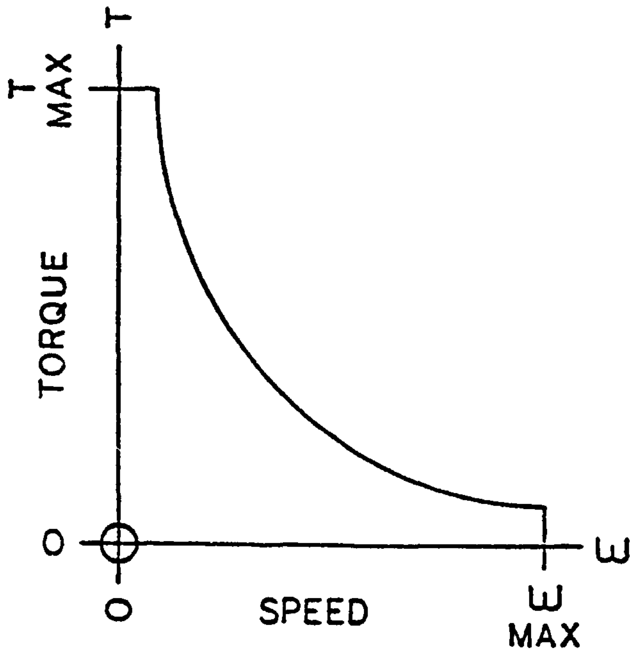


Figure 5-45. Homopolar Machine Constant Power Contour

5.6.3.12 Homopolar Machine Advancenents. There are several areas in which significant gains in the state of homopolar machines may be realized: brush technologies, gain enhancement, magnetic material, thermal design, and inertia reduction.

- Brush Technologies. Presently available homopolar machines use graphite composition (silver-graphite, copper-graphite) brushes as the current collection system. The primary limitation of these devices are the realizable current densities (Amps/in2) and the accelerated wear which results from very high rotary surface speeds. Work has been done in several areas to overcome these potential shortcomings. Liquid metal brushes, such as mercury (Hg) or a mixture of sodium and potassium (NaK), have also been used, with a number of experimental machines being built. A major problem associated with this approach is the current distribution at low rotor speeds since most systems relied on the centrifugal force of the rotor to evenly distribute the liquid metal around its periphery. A promising new development is the concept of metal fiber brushes, which involves the use of many brush-like fibers to contact the surface of the rotor drum and act as current collectors.

- Gain Enhancement. Since the control power required for the homopolar machine is directly dependent on its gain, it is advantageous to increase this parameter as much as possible. The bulk of this effort involves reduction of air gaps within the machine and the use of materials which exhibit both good magnetic as well as electrical properties. Innovative rotor design and current collector placement can also be used as techniques to increase machine gain. Future increases on the order of five times present machine gains appear to be achievable with current technologies.

- Magnetic Materials. Homopolar machines are inherently low voltage devices. If in a given machine size the amount of flux which is cut by the rotor sleeve can be increased, so can the generated voltage (Faraday's Law). Since the size of the machine is assumed to be fixed, the only means available to increase the magnetic flux is to use a material with a higher allowable flux density. This could entail utilizing newer magnetic material types, or possibly a combination of materials in various components of the machine. Since the volume of the rotor drum is presently the point of tightest flux constraint, it may be possible to fabricate a higher flux density rotor and maintain the balance of the material in the machine as iron.

- Thermal Design. Heat removal is of primary concern in the homopolar machine due to its influence on the power density. This device parameter is directly proportional to the amount of heat removal provided, such that as the capacity to remove the thermal energy increases, the power density also does so proportionally. Factors such as coolant type, system flow rate, radiator capacity, and machine design all contribute to the effectiveness of vehicle drive system heat removal. Machine thermal design is of course a significant aspect in system power density improvement. Techniques such as spray cooling, flooded rotor design, integral field coil cooling loops, and internal rotor cavities all contribute to yield a smaller, more efficient machine. At present, homopolar machines use many of the above methods, however, further advances could be made in both new heat removal techniques and the implementation of present methods.

- Inertia Reduction. Rotary machine inertia can be a primary contributor to overall system inertia in an electrically driven vehicle. It is therfore imperative to reduce this parameter as much as possible without compromising other critical device characteristics. The primary benefits of decreased inertia are greater system responsiveness and increased servo-loop performance. In a vehicle velocity servo-system this has direct impact on both vehicle maneuverability as well as on acceleration performance. Techniques to reduce homopolar machine inertia center largely on selective removal of rotor material; this will not impede the flow of magnetic flux, but will impact the rotor periphery where the inertial component is the greatest. Another potential solution involves maintaining a stationary magnetic rotor core, while the voltage generating sleeve structure is allowed to rotate. This technique has been shown to reduce the homopolar inertia to as little as 25 percent of present machine implementations.

5.6.4. Configuration I - Induction Motor System. An induction motor sprocket drive and control system was designed for the 19.5 ton vehicle. Much of the concept has already been designed and tested in a M113 vehicle modified for installation of an electric drive system.

The induction motor drive system produces a continuous tractive effort (TE) of 0.7 at low speeds and an intermittent TE of 1.2 at stall. The system was designed to: deliver 380 HP to the tracks at speeds between 5 and 45 MPH, provide 0.5g lateral steer capabilities up to 45 MPH, and electrically brake from 45 MPH to 0 MPH at 5g's. A block diagram of the control system is shown in Figure 5-46. The major elements of the propulsion system are shown in the block diagram (Accessory elements are omitted for clarity).

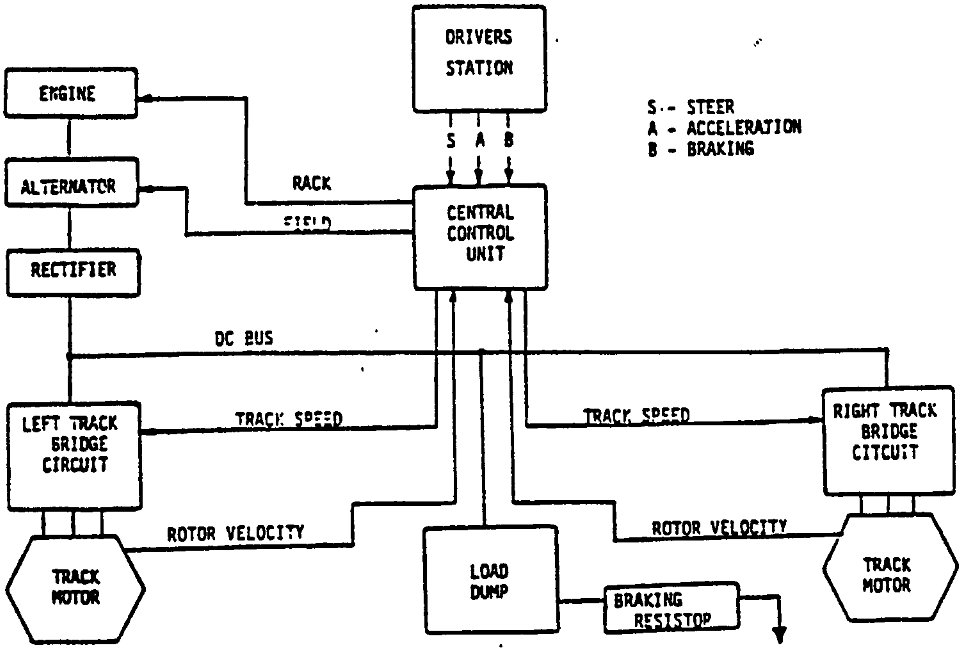


Figure 5-46. Control System Block Diagram

5.6.4.1. System Description. A brief description of the propulsion system is given below, and a detailed description of each major element follows. A complete electric drive vehicle concept is also described.

The driver inputs electronic steer, acceleration, and braking commands to a central control unit which controls the vehicle propulsion system. The control unit interprets commands from the driver and it controls the engine fuel, alternator field current and left and right track bridge circuits which supply power to each induction motor track drive. The engine is coupled to a high speed AC alternator which generates electrical power for the propulsion system. A rectifier converts variable AC to variable DC and each bridge circuit generates a 3-phase rotating voltage which is applied to each induction motor stator. A load dump circuit and braking resistor are provided to absorb vehicle braking energy.

5.6.4.2. Vehicle Control Concept. The central control unit provides the following functions: It sets the engine speed and the alternator field current, closes a velocity servo loop for each track motor, receives driver's commands and controls the above functions to make the vehicle manuver as commanded. This control unit adjusts the engine speed to provide the required power at the best fuel point. The alternator field current is set to provide the DC bus voltage required. The bus voltage may be reduced at low vehicle speeds.

The central control unit closes a velocity servo loop for each induction motor. It receives driver's commands and rotor velocity feedback. It then supplies the proper power to each track motor to force the rotor velocity to match the commanded velocity. A block diagram of the servo loop is shown in Figure 5-47. The induction motor is controlled by varying the frequency and current in the stator winding. To obtain a positive torque the stator field is rotated slightly faster that the rotor; to obtain a negative torque the stator field is rotated slightly slower than the rotor. The frequency difference between the stator and the rotor is called slip. When the slip is positive, the motor develops a positive torque and acts as a motor. When the slip is negative, the motor develops negative torque and acts as a generator and supplies power back to the DC bus. A unique characteristic of the induction motor system is that it will generate high voltages and high torques at very low speeds. This is particularly important for steering and braking at low track speeds.

As shown on the control system block diagram both bridge circuits are connected to a common DC bus. When a positive torque is required at each track power is supplied by the engine alternator (Figure 5-48.). When a negative torque is required at each track, power flows from each bridge circuit back to the DC bus. The power then flows from the DC bus to the load dump circuit and braking resistor where it is dissipated in the propulsion system cooling air stream (Figure 5-49.).

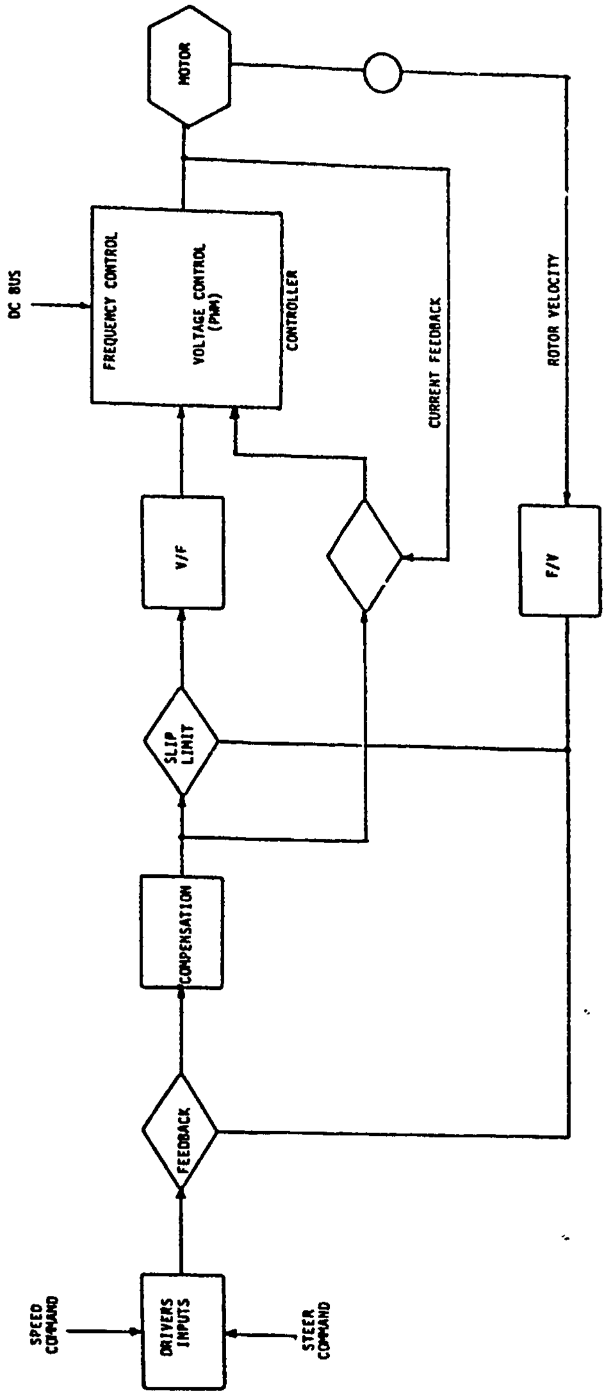


Figure 5-47. Track Motor Servo Loop

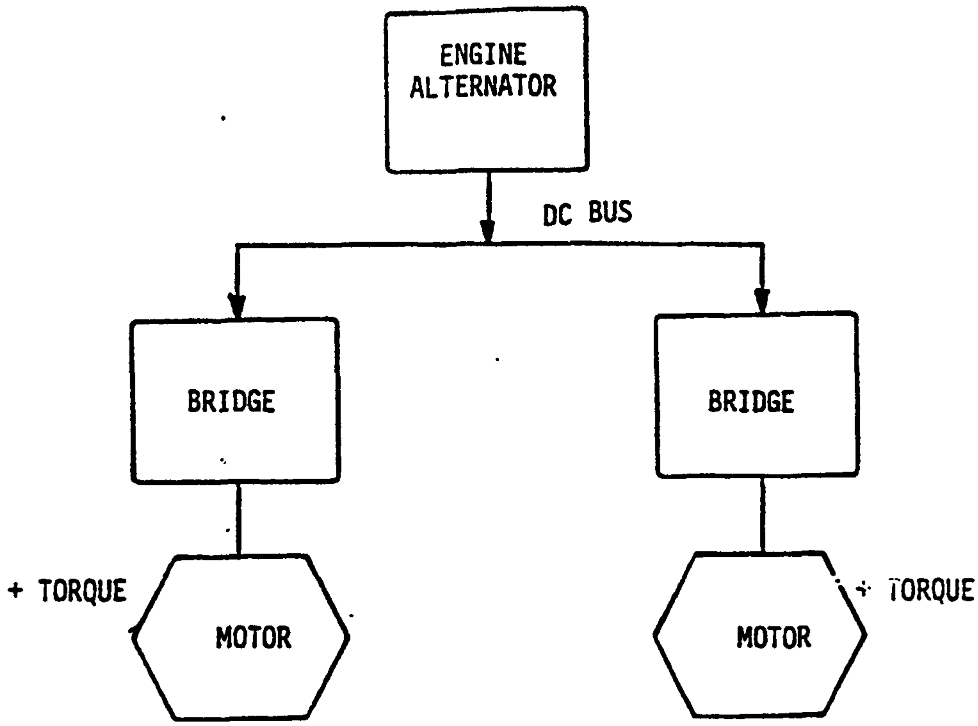


Figure 5-48. Power Flow Diagrams; Positive Torque at Motors

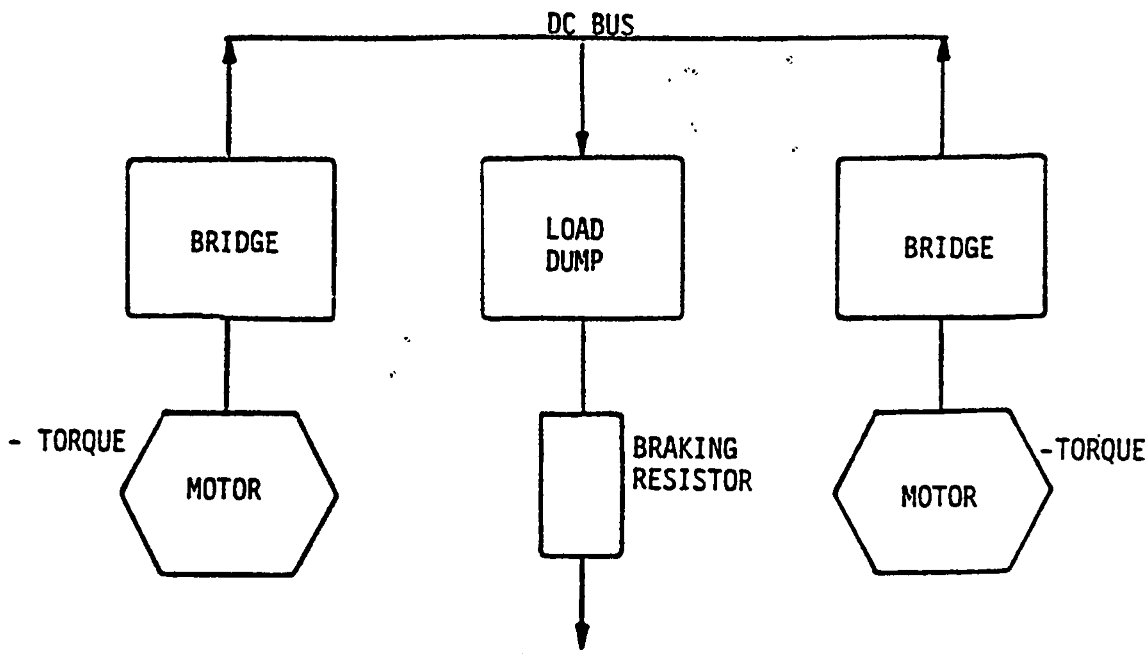


Figure 5-49. Power Flow Diagrams; Negative Torque at Motors

5.6.4.3. Induction Motors. Vendors were contacted and asked to design induction motors for the 19.5 ton vehicle. Motors with the following minimum charcteristics were specified: 15000 rpm, 600 ft lbs continuous torque to 2000 rpm, 200 HP continuous above 200 rpm, and 100 ft lbs of stall torque for 1 minute. To obtain minimum motor size speed range was increased or decreased and torques scaled accordingly; in addition cooling was optimized to achieve space claim objectives. Three induction motors were designed that would operate within the characteristics described above. Physical characteristics of these motors are outlined below:

- o 24 krpm - 9" Dia x 22" Long at 233 pounds
- o 18 krpm - 11.5" Dia x 20.5" Long at 350 pounds
- o 15 krpm - 12.5" Dia x 19.7" Long at 455 pounds

The efficiency of the 15 krpm motor over the speed range at 200 HP output is shown below and is representative of the efficiency of all three motors:

- o 95.1% at 15,000 rpm , (45 MPH)
- o 96.2% at 12,000 rpm
- o 96.4% at 6,000 rpm
- o 88 % at 1,700 rpm

The efficiency of the motor below 5 MPH at 600 ft lbs is:

- o 88% at 1,666 rpm, (5 MPH)
- o 86% at 1,000 rpm, (3 MPH)
- o 85% at 666 rpm, (2 MPH)
- o 68% at 333 rpm, (1 MPH)

The motor is basically conduction oil-cooled with a wet sump. Oil is circulated through the stator and through the rotor. The maximum cooling requirement occurs at 600 ft lbs of motor torque; 16 HP is lost in the stator and 10 HP is lost in the rotor. The induction motor is a 4 pole, 3 phase machine. Some of the detail construction data is presented below:

- o Magnetic Weight 284# (weight of stator and rotor)
- o Stator OD 11.75 inch

- o Stack Length 9.7 inch
- o Rotor Inertia 3.4 lb in sec2

A synchronous AC motor was also designed to meet the above requirements. The motor is an 8 pole, 3 phase machine. The size and weight is 12.75 inches OD x 22.5 inches long and weighs 285 pounds. The efficiency of the synchronous motor over the speed range at 200 HP is:

- o 92% at 15,000 rpm
- o 95% at 2,000 rpm

The efficiency of the motor below 5 MPH at 600 ft lbs is:

- o 94% , 1666 rpm (5 MPH)
- o 90% , 1000 rpm (3 MPH)
- o 85% , 666 rpm (2 MPH)
- o 70% , 333 rpm (1 MPH)

The synchronous motor is a 3 phase, 8 pole machine. The construction is:

- o Rotor OD - 9.33 inches
- o Stack length - 9.2 inches
- o Rotor material - 4140
- o Stator material - 3% silicon steel
- o Air gap length - .050 inches

Based on performance and weight/volume comparison the induction motor is used for the Configuration I concept. The induction motor continuous capability and the required capability are shown in Figure 5-50. The figure shows the continuous generating capability is the same as the motoring capability. The intermittent capability of the induction motor is several times the continuous rating and is only required to achieve a TE of 1.2 at stall.

5.6.4.4. Vehicle Drive Operational Considerations. The downhill braking requirement on a 15% slope at 20 MPH is only 208 HP total or 104 HP per motor. At 20 MPH each motor could generate about 650 HP continuously; well above this braking requirement. The problem of steering and braking at low speeds is of particular importance because it eliminates several motor types for Configuration I when a common DC bus is used.

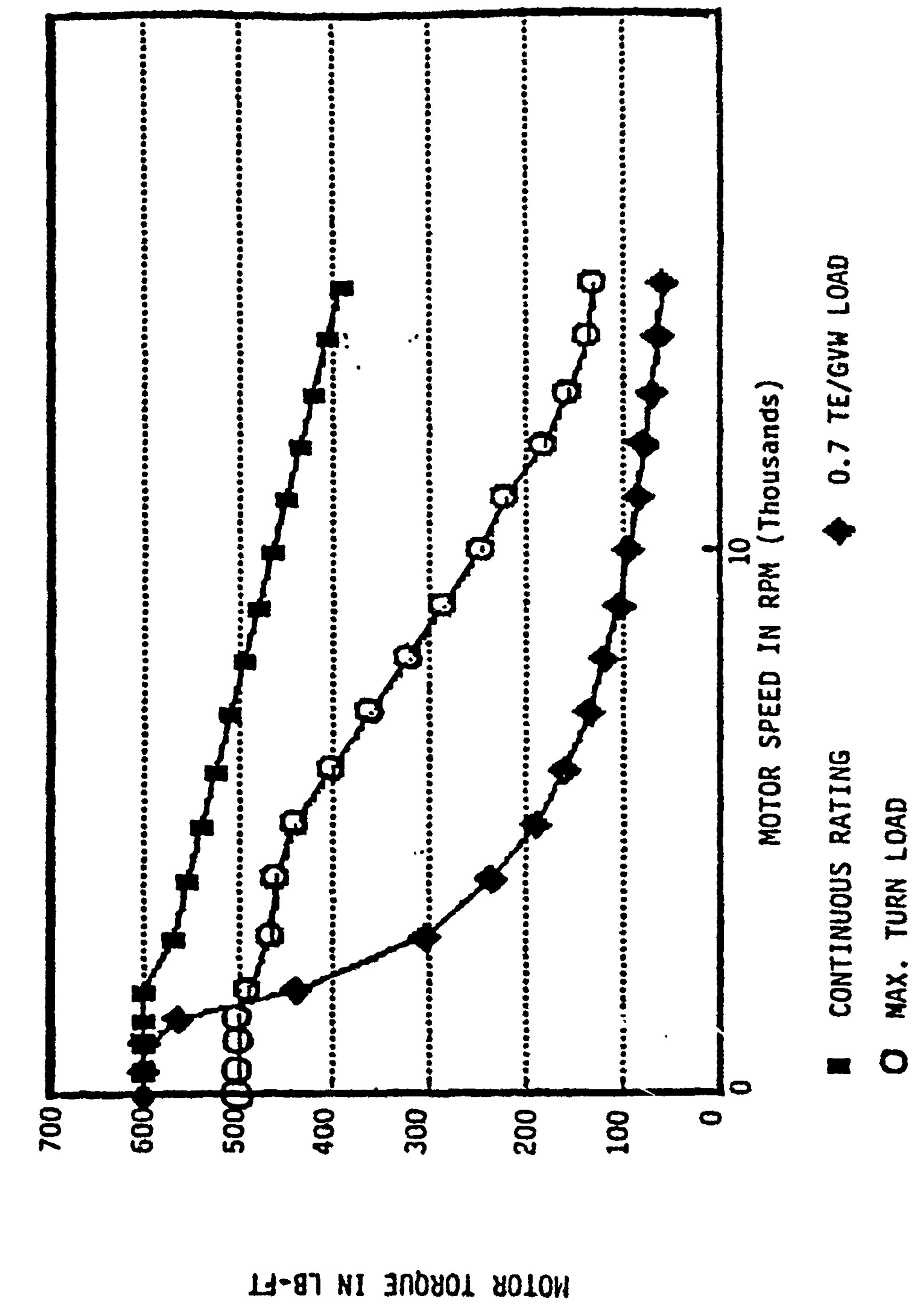


Figure 5-50. Induction Motor Continuous Capability

Examining the sprocket horsepower table below (Table 5-5.), the following speeds and torques are required simultaneously for steering.

Table 5-6. Sprocket Horsepower and Torque

VEHICLE SPEED (MPH)	INSIDE HP	INSIDE RPM	INSIDE TORQUE (FT LB)	OUTSIDE HP	OUTSIDE RPM	OUTSIDE TORQUE (FT LB)
1.0	3.9	-2.6	-7760	62	34	9475
2.5	7.7	-5.2	-7750	152	84	9466
5.0	-7.4	5	-7684	275	154	9403
7.5	-45	31	-7583	367	207	9306
10.0	-116	83	-7322	404	234	9050

- At 1.0 and 2.5 MPH both motors are motoring in opposite directions. At 5 MPH the inside track motor must generate 7.4 HP at only 5 RPM and the outside sprocket motor must deliver 275 HP at 154 rpm. If the inside sprocket motor could not generate the bus voltage required for the outside sprocket then it could be plugged to generate the negative torque at the low power level of 7.4 HP.

- At 7.5 MPH the inside sprocket motor must generate 45 HP at 31 rpm and the outside sprocket motor must deliver 367 HP at 207 rpm. Plugging the inside sprocket motor at 45 HP is not practical and will result in the motor overheating. The motor must be able to generate at the low speed of 31 rpm and it must generate the bus voltage required by the outside track motor.

- At 10 MPH the inside track motor must generate 116 HP at 83 rpm and the outside track motor must deliver 404 HP at 234 rpm. Again at the low speed of 83 rpm the inside track motor must generate the bus voltage rquied by the ouside track motor.

- The motor can be wound to operate at any bus voltage without affecting size or weight for a given HP. The inside track motor must generate the voltage required by the outside track motor to steer the vehicle correctly even though their speeds differ greatly.

- The voltage generated by the inside track must be the same or higher than the voltage required by the outside track at each vehicle speed for steering. Refer to Table 5-6. for typical inside and outside track requirements.

- The inside track motor will generate the bus voltage required down to 10 rpm. Below 10 rpm the motor will not generate the required bus voltage and it must be plugged to provide the negative sprocket torque. However, the power levels required below 10 rpm are low and the motor is in a safe operating area.

5.6.4.5. Alternator. The alternator designed for the 19.5 ton vehicle was sized to produce more power than the engine could deliver over the speed range so the engine could be always be throttled back for reduced power operation. It was designed to produce the following HP levels at the speeds shown: 100 HP @ 2500 rpm, 200 HP @ 5000 rpm, 300 HP @ 7500 rpm, and 600 HP @ 15,000 rpm. The alternator output capability is shown superimposed on the engine fuel map in Figure 5-51.

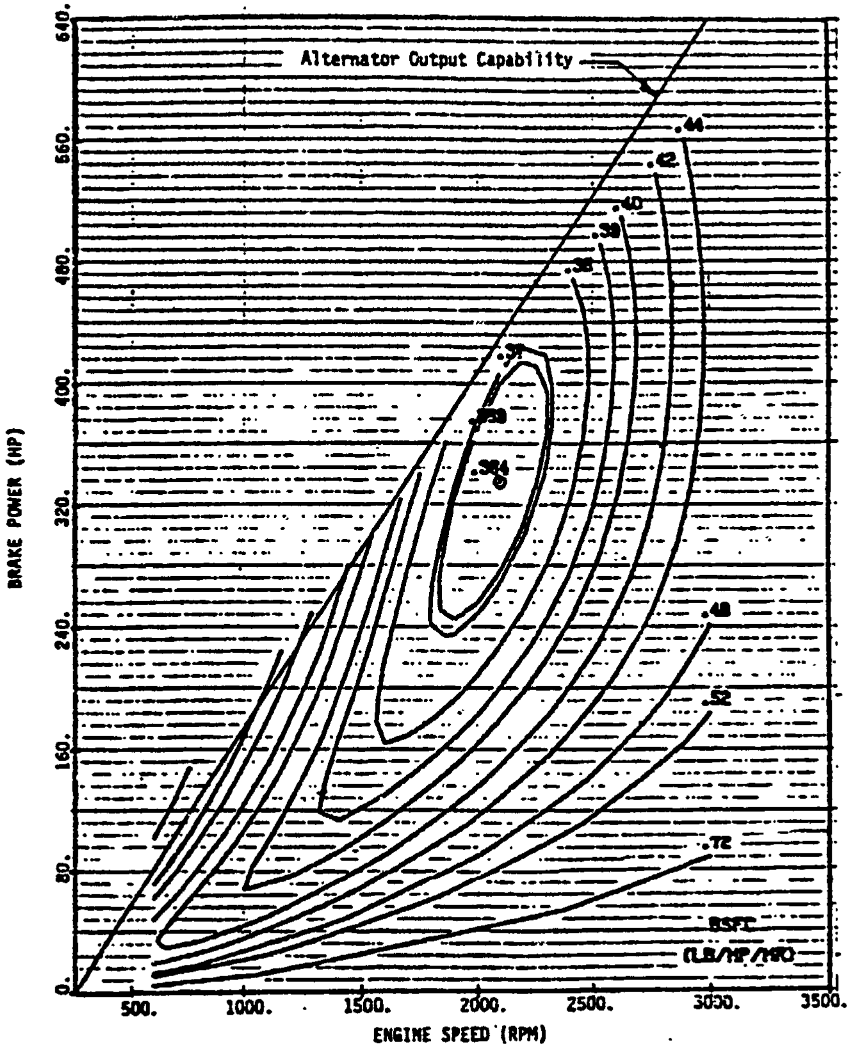


Figure 5-51. Cummins VT903 Engine Fuel Map

The voltage required at the outside track for steering varies with speed. The table below shows requirements for a system designed to operate with a 600 volts DC bus.

Table 5-6. Voltage and Current Requirements During Turns

	OUTSIDE TRACK			INSIDE TRACK		
SPEED (MPH)	VOLTAGE	CURRENT (AMPS)	RPM	VOLTAGE	CURRENT (AMPS)	RPM
1.0						-2
2.5	135	850	84	135		-5
5.0	247	847	154	247		5
7.5	331	842	207	331		31
25	554	682	428	554		367
45	600	440	773	600		699

5.6.4.6. Braking Circuit. The braking circuit is designed to connect the DC bus to the braking resistor when vehicle braking is required. A diagram of the braking circuit is shown in Figure 5-52. The circuit is arranged so the DC bus is connected to the braking resistor when the bus voltage rises above the reference voltage. This occurs when a negative slip is commanded at each induction motor. The braking resistor grid is located in the vehicle exhaust plenum or it could be a liquid cooled resistor located in the engine cooling system. A liquid cooled resistor was concepted to absorb the braking energy. It is 6 inches diameter by 21 inches long and weighs 75 pounds. The temperature rise in the cooling system is 11.5F after a stop from 37 mph assuming all the energy goes into the coolant. In practice much of it will go into the engine and radiator mass and the temperature rise will be less than 11.5F.

5.6.4.7. Bridge Circuits . Each bridge circuit contains six switching elements as shown in Figure 5-53. Diodes are connected around each switching element to provide current flyback paths and regenerative power flow paths. Terminals A, B, and C are connected to the induction motor stator windings. By switching the upper and lower elements sequentially a 3 phase variable frequency waveform is generated.

At present parallel power transistors are used as the switching elements and this is what is used in a currently operating electric drive M113. As larger devices become available the need for parallel elements is elminated and simpler bridge curcuit assemblies can be designed. Other new switching elements are being developed that have higher power handling capabilities. The progress in this field is dramatic and power handling capabilities are doubling every two years. Silicon Controlled Rectifiers(SCRs) can be used as the switching elements but they require additional turn off commutation circuitry and they can not be controlled as well as the power transistor.

The efficiency of the power transistor bridge curcuit is about 98% over the complete HP ranges. Cooling is accomplished by flowing oil through the control box heat sink.

5.6.4.8. Vehicle Concept. A complete electric drive vehicle concept was generated for the 19.5 ton vehicle including the vehicle accessory systems. A block diagram of the concept is shown in Figure 5-54.

- o An electric fan is used to provide air flow past two stacked radiators. The electrical radiator is used to cool oil for the electrical components. The engine radiator is used to cool engine coolant. Exhaust air flows past a brake grid.

- o Fan speed is controlled electronically by the fan control unit, which supplies 3 phase variable frequency AC to the induction motor fan drive. Power flow is from the DC bus to the fan motor and about 60 HP of braking energy can be used to increase the air flow during long downhill braking.

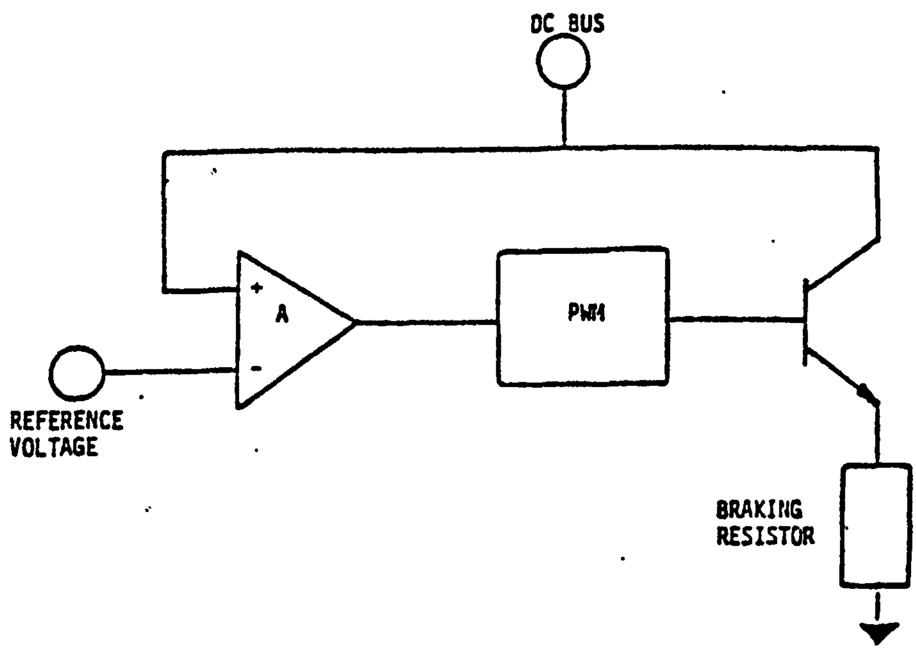


Figure 5-52. Braking Circuit Diagram

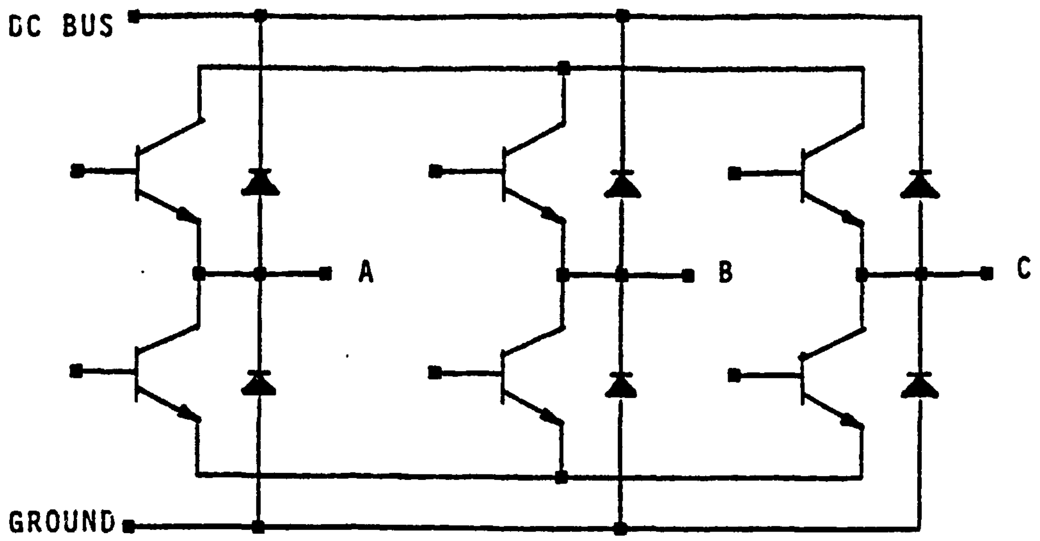


Figure 5-53. Bridge Circuit

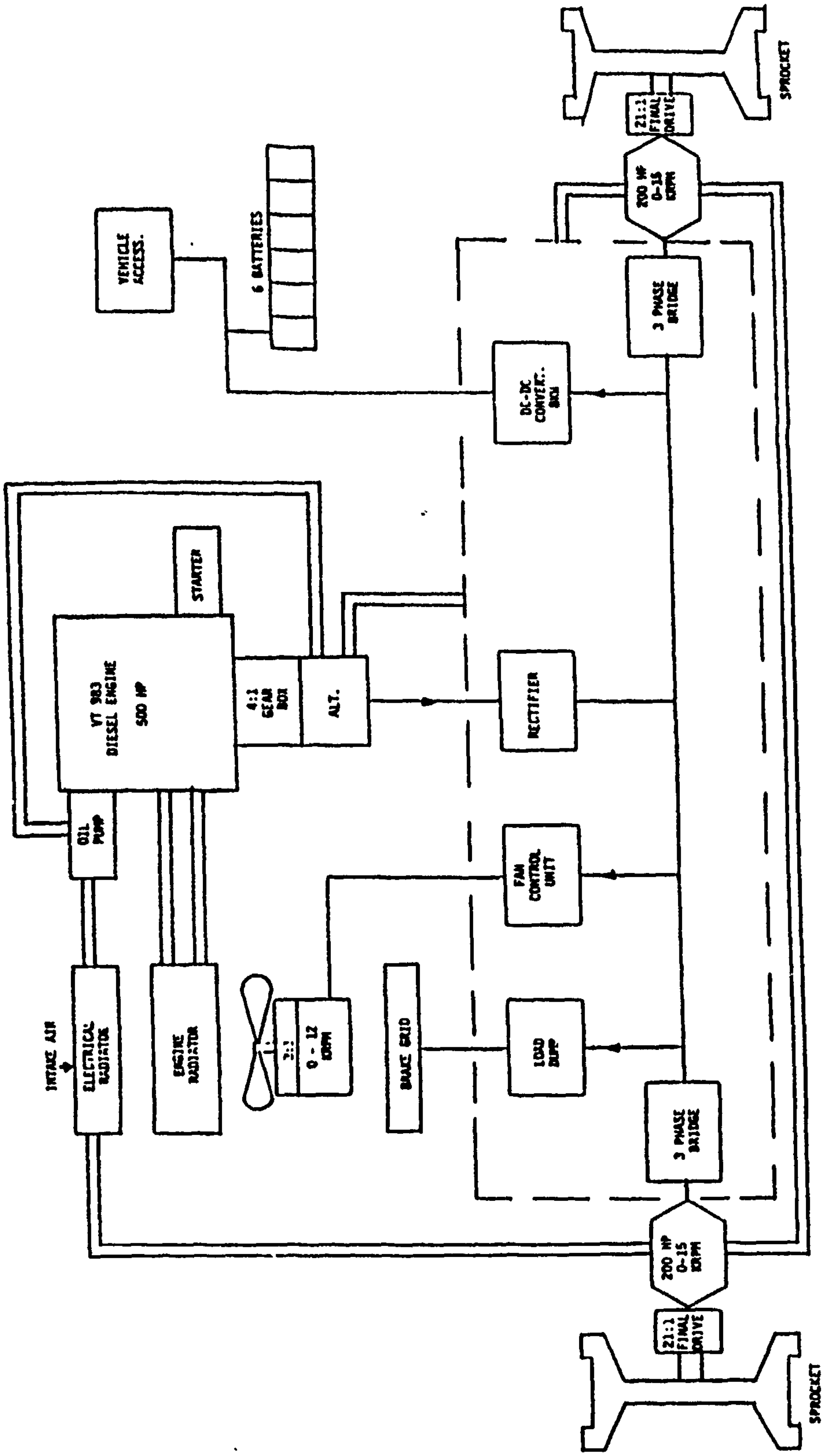


Figure 5-54. Induction Motor Concept (19.5 Ton) (Configuration I)

- o A DC-DC converter is used to convert high voltage DC to 28 volts DC to charge 6 vehicle batteries and supply other accessory loads. A cover actuated switch is used to automatically disconnect voltage when the access cover is removed.

- o All the power electronic circuitry is located on a common cooling plate in a closed control box; the cooling oil is pumped through the plate. An engine driven oil pump is used to circulate cooling oil to the electrical components.

5.6.5. Configuration II - Induction/Homopolar Motor Systems. The Configuration II electrical system is configured housing the propulsion and steer motor functions divided into separate components. In this configuration, the propulsion motor can be rated for the maximum speed/torque requirements dictated by the vehicle performance specifications without the additional burden of superimposed steer torques such as required by Configuration I drive system. The separate steer control motor is sized to provide the required differential velocity to the propulsion planetary or final drive gear set with the requirement of supplying extremely high stall torques for low differential speed turns such as pivot steer.

In practice, the steer motor can be sized at a ratio of 0.25 to 0.33 of the propulsion motor rating due to the discontinuous nature of steering maneuvers. Both the AC and DC selected systems are capable of meeting the requirements for the Configuration II drive.

5.6.6. AC Induction Motor System. The high frequency AC induction motor system previously described in section 5.6.4. is adapted for Configuration II. The major components are described as follows:

5.6.6.1. Propulsion Motor. The propulsion motor is base rated at 365 HP with a maximum speed of 15000 rpm and rated torque of 128 lb. ft. The motor is capable of operating in both forward and reverse direction. The motor is capable of full torque generation over the entire speed range of 0-15000rpm.

5.6.6.2. Steer Motor. The steer motor is a high torque, high speed motor having a torque rating of 4/1 over the rated torque. The wide range torque requirement allows the steer motor to be significantly reduced in size from propulsion motor. Although not required to provide torque in straight driving, the steer motor must supply full differential torques to each final drive when a turn is commanded. The motor is capable of rapid reverse operation, thus reducing the lag in correcting from a tight commanded turn.

5.6.6.3. Propulsion Motor Controller. The propulsion motor controller, rated at 325 KVA, supplies the propulsion motor with voltage and frequency dictated by the vehicle performance requirement. As described in Section 5.6.4., the controller converts the high voltage DC base voltage

(300-600 volts) to variable frequency AC voltage. The variable frequency AC controls the speed of the induction motor.

5.6.6.4. Steer Motor Controller. The steer motor controller, rated at approximately 100 KVA, drives the steer motor when a turn is commanded. This controller also derives its power from the high voltage DC bus and converts it to variable frequency AC.

5.6.6.5. All components of this system are equivalent to those described in the Configuration I induction drive section except for rated power. Thermal management is identical.

5.6.6.6. Since the total active power input peak is approximately 425 KVA, the system uses a 440 KVA alternator as the prime power source driven by the diesel engine. The alternator output is rectified to supply the 300-600 VDC for the power bus and the motor power controllers. Figure 5-55. illustrates proposed Configuration II system. In this block diagram, each motor is shown powered by individual DC to AC inverters. The inverters are variable frequency devices which provide the propulsion or steer motor with the commanded frequency specified by the vehicle mission. In practice, the two inverters can be packaged together and thus share the DC bus, associated protection mechanisms, and thermal management systems.

5.6.7. Homopolar Motor System. The application of homopolar machines to a Configuration II drive team requires two independent motor-generator systmes; a propulsive pair and a steering pair (see Figure 5-56.). The steering pair need only have 0.25 to 0.33 the power capabilities of the propulsion pair due to the discontinuous nature of steering maneuvers. Although each drive pair is electrically isolated, both homopolar generators are driven by the common prime mover and can be integrated into a common generator package with separate rotor (armature) circuits. The electrical circuits between machine armatures are connected for each homopolar pair, completing the power flow path of the drive. Control of each of the machines is then achieved through the excitation of the fields, with either solid-state or rotary controls.

5.6.7.1. Vehicle Control Concept. As in the dual independent drive scheme of Configuration I, the operation of each homopolar motor-generator pair is achieved through the coordinated field controller. This system element assures that maximum use is being made of the speed/torque characteristics of each machine by adjusting the field excitations to achieve the most optimum operating point for each. A velocity, or torque, feedback look is closed around each pair of the machines, such that independent control of propulsion and steer results. The system requires that both an average forward speed and a turn command be provided by the operator. These inputs are maintained separately, since no differential velocity commands need to be generated as in the Configuration I drive, and determine the response of the

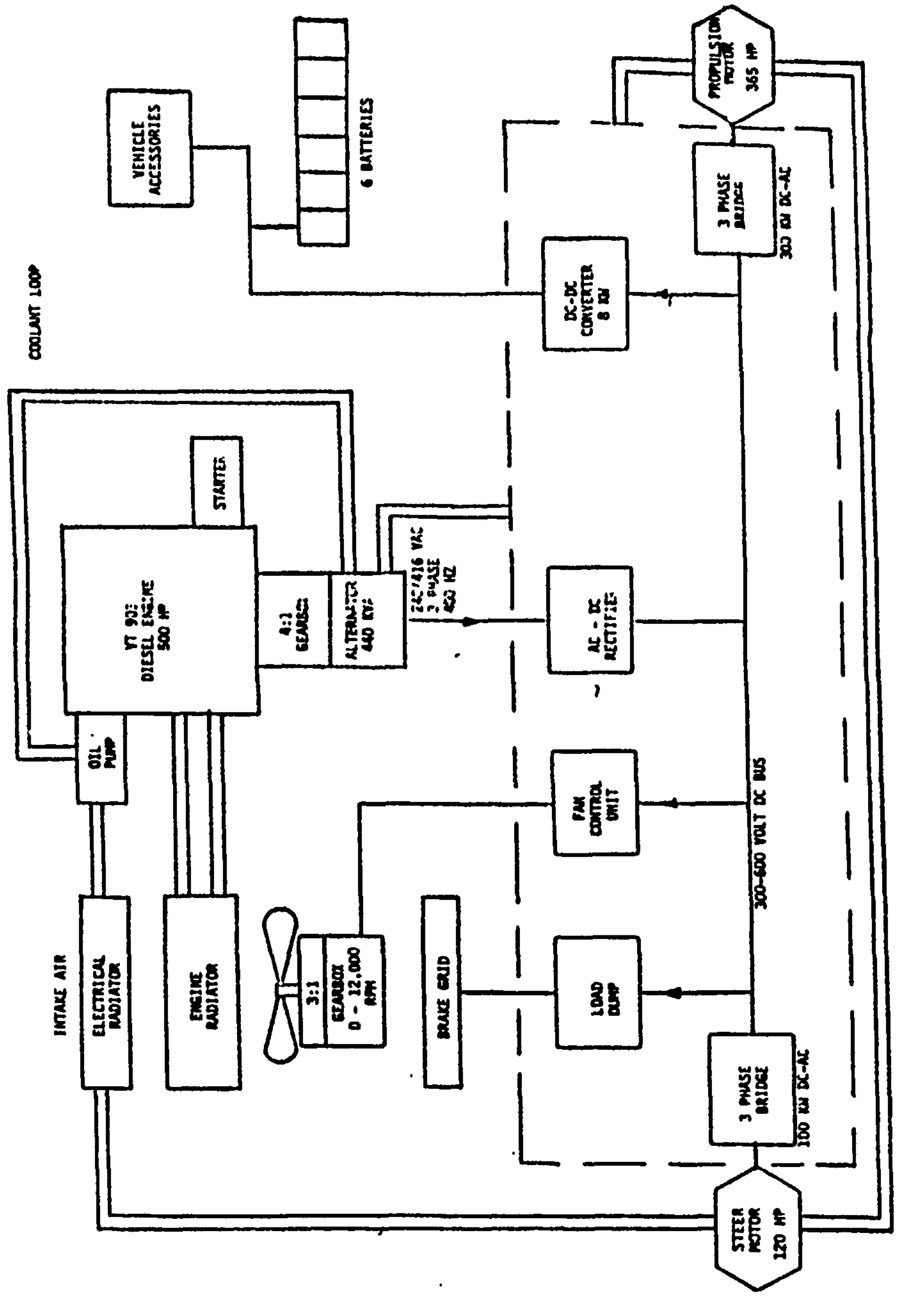


Figure 5-55. Configuration II; Induction Motor System

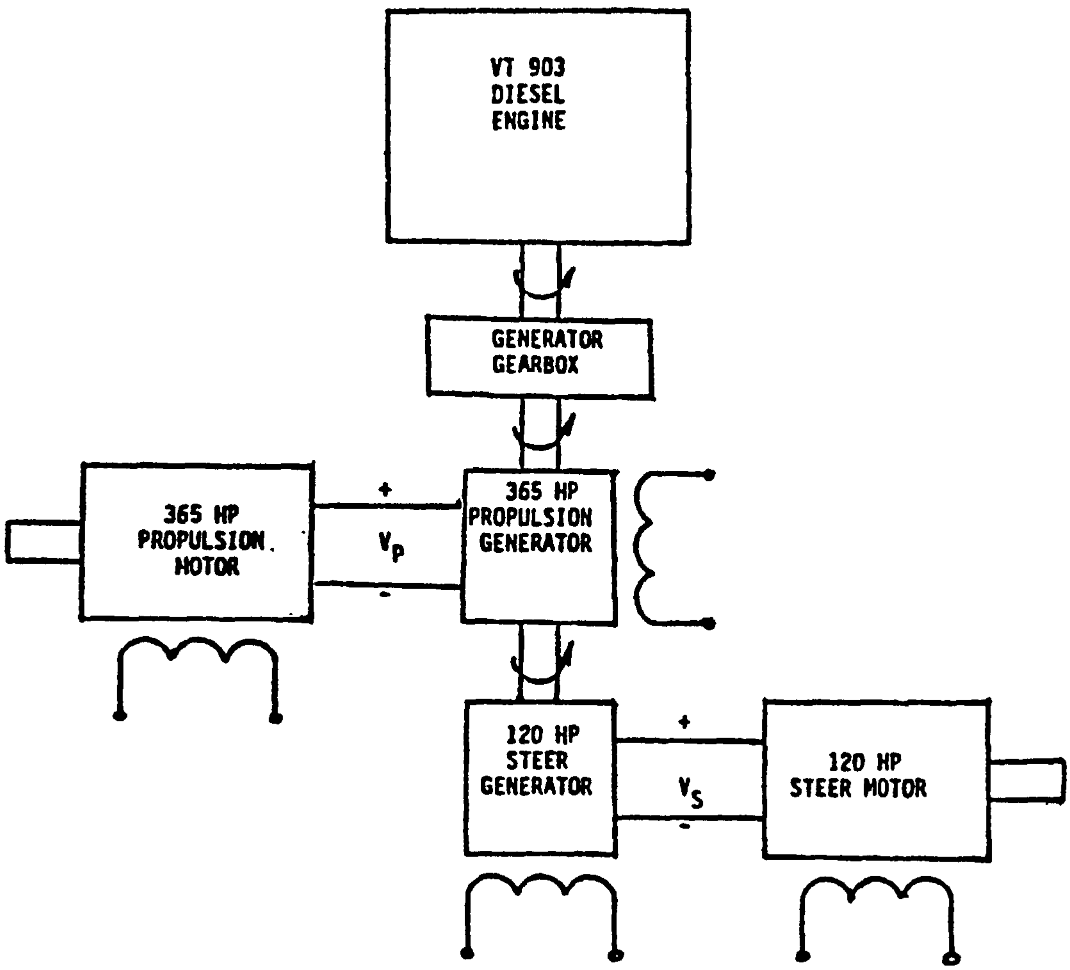


Figure 5-56. Configuration II; Homopolar Motor System

propulsion and the steer machine pairs respectively. Closed loop operation of each drive pair operates in a similar manner to that which is described for the Configuration I drive.

5.6.7.2. System Attributes . Advantages to vehicle performance which result from a Configuration II homopolar drive train are described below.

- Independent Velocity/Steer Control - The inherent separation of these two functions in the Configuration II drive provides advantages in terms of the ability to manually steer the vehicle under slow speed conditions, in the event the primary steer drive is incapacitated.

- System Voltage Compatible with Vehicle Operating Voltage - The inherent low voltage nature of the homopolar machine allows for direct excitation of the propulsion and steer motors from the vehicle battery bank in the event of a short term emergency propulsion requirement.

- Flexible Engine Scheduling - Either constant or variable diesel engine speed can be accomodated by the system due to the closed-loop nature of the coordinated field controller.

- Independent Speed/Torque Control - The flexibility of the homopolar machine field control characteristc coupled with the cooordinated controller allows for the speed and torque of the machines to be controlled nearly independently of the loads imposed on either the propulsion or the steering drive systems.

- Dead Engine Steer Capability - The homopolar drive offers dead engine steer capability by energizing the steering drive system. Since during this condition the homopolar generator which provides power to its corresponding motor would not be operational, motor armature power is required from the vehicle batteries.

- Mechanical Regeneration Link - All Configuratin II drives have the ability to transmit power from track to track with a hard mechanical connection. Neither the propulsion or the steer motors are actively involved in the transmission of the power from track to track by electrical means.

- Flexible Machine Field Excitation - Either solid-state (semiconductor) or rotary (motor-generator set) field excitation may be used to control the fields of the homopolar machines.

- Dead Engine Dynamic Brake Capability - As with the dead engine steer condition, the propulsion machine fields can be excited with stored energy from the vehicle operating system batteries, allowing the vehicle to be dynamically braked without the use of power from the prime mover.

5.6.7.3. Vehicle Drive Operational Considerations. Six specific vehicle maneuvers are explored below with regard to the reaction of the homopolar machines in Configuration II drive: forward/reverse operation, turning, non-regenerative turning, regenerative turning (single track), regenerative turning (dual track), neutral axis steer, and dynamic braking.

- Forward or reverse operation in the Configuration II homopolar drive only requires that the propulsion motor/generator fields be excited since the operator steer command will be zero. The closed-loop propulsion servo responds to the speed command by exciting the machine fields so that either the desired forward or reverse motion is achieved. The power flow is shown at Figure 5-57.

- Turning results in three possible power flow conditions. During normal steering, power is delivered proportionally to each track by the steer motor so that the net power flow to each track is positive (non-regenerative turn). As the turning radius is tightened, the steer motor is required to provide more power to achive the differential track velocity. The power flow is also affected; the inner track being driven by the ground rather than the propulsion drive. As this occurs, the power generated by the inner track is transmitted through the cross shaft where it reduces the load the propulsion drive must carry to drive the outer track (regenerative turn-single track). If grade conditions are such that both inner and outer tracks are being driven by the ground, then the propulsion drive motor and generator homopolar machines reverse roles and begin to transmit the generated power back across the electrical link, which connects the homopolar machines to the prime mover. In this condition, the prime mover absorbs part of the energy, the remainder being dumped to a resistive grid (regenerative turn-dual track). See Figure 5-58. for this power flow.

- Neutral axis steer places the highest demands on the steering drive elements in the Configuration II scheme. Since the net velocity, forward or reverse, of the vehicle is essentially zero, no power is commanded from the propulsion drive elements. All power required for the maneuver results from the operation turn command, and must be supplied by the steer system. See Figure 5-59. for the steering only power flow.

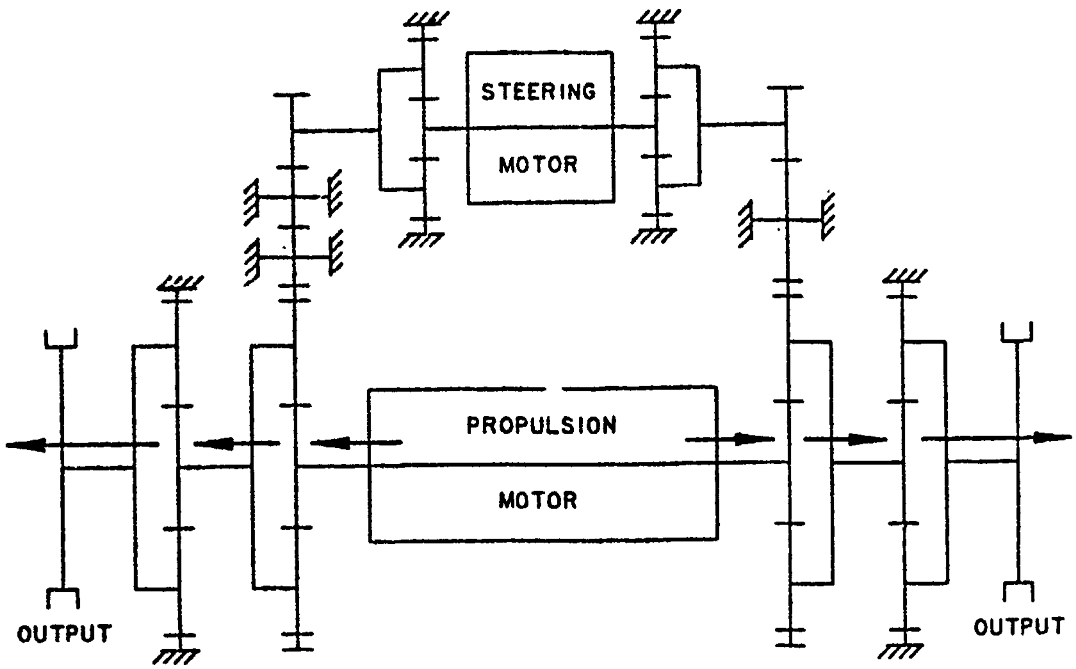


Figure 5-57. Power Flow for no Steering (Configuration II)

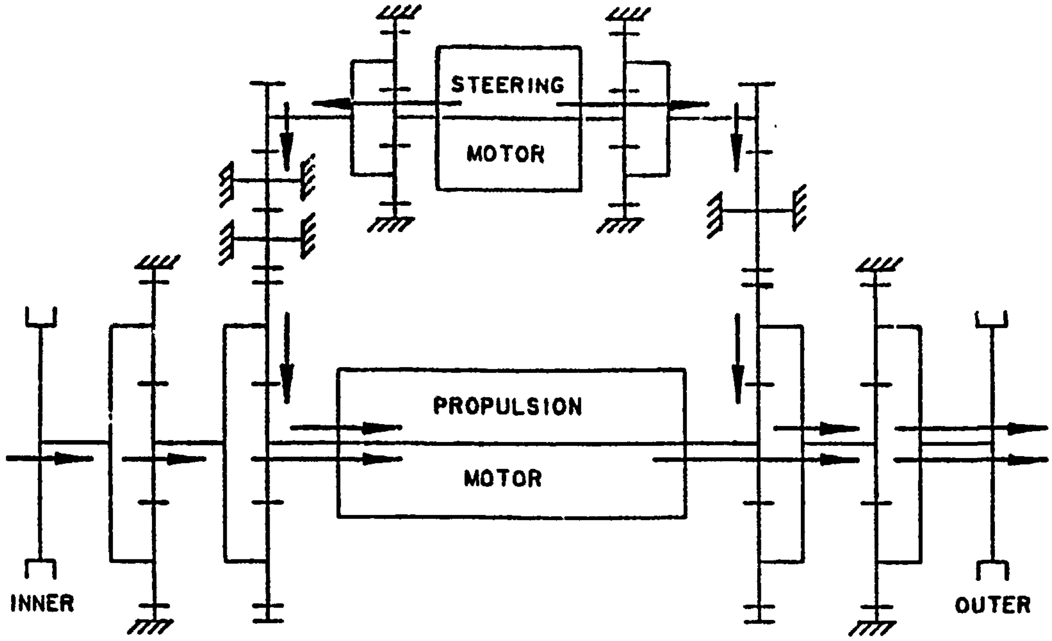


Figure 5-58. Power Flow for Propel and Steer (Configuration II)

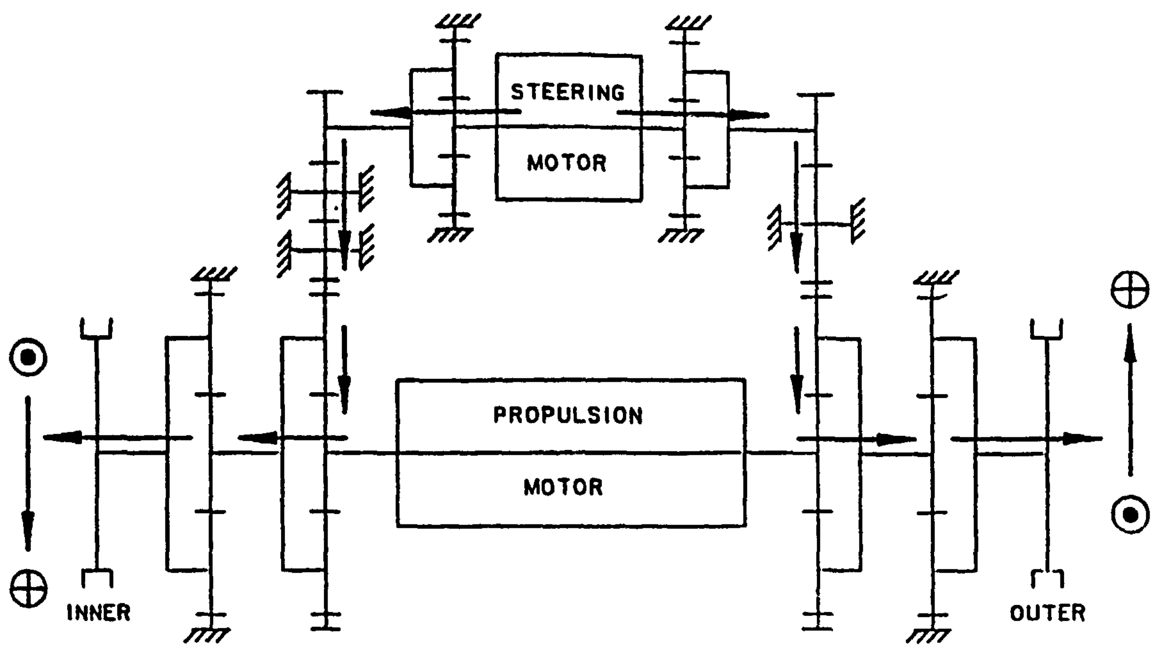


Figure 5-59. Power Flow - Steering Only (Configuration II)

- Dynamic braking is initiated by commanding a motion from the propulsion drive which is the opposite of that which the vehicle's tracks are experiencing. This is accomplished through an operator supplied brake command which reverses the appropriate homopolar machine field polarity. Since the propulsion motor-generator pair is now being driven, assuming a straight ahead braking motions, equal power flow from each track back through the prime mover. As with the dual track regeneration, the power supplied to the system in excess of the reverse capacity of the prime mover is dissipated in resistive power grids.

5.7. Configuration III

Configuration III consisted of an engine driven alternator and electric drive motors at each final drive. A steer system, consisting of a single electric motor driving a null shaft into combining planetary gearing at each final drive, was superimposed on the propulsion in the conventional way. A schematic of this arrangement is shown in Figure 5-60.

5.7.1. The following assumptions were made in generating this configuration for the original Electric Drive proposal effort:

- The propulsion motors would operate at identical rpm thus controlling the average vehicle speed.

- Differential track velocity required during steering maneuvers and some of the steer horsepower would be provided by the single steer motor and null shaft arrangement.

- Additional steer power requirements would be transmitted by regeneration from the inner to the outer tracks via the steering null shaft.

A detailed analysis proved the latter statement to be incorrect. It was found that when the propulsion and steer loads were such that the net power at the inner sprocket, of a turning vehicle, was into the final drive. This power then passes through the propulsion side of the differential gearing into the drive motors, and not through the steer null shaft as assumed.

5.7.2. Configuration III was judged therefore to offer no advantage over Configuration I and was in fact penalized by the added complexity of the steer drive mechanism.

A summary of the calculations leading to this conclusion are included in Appendix C.

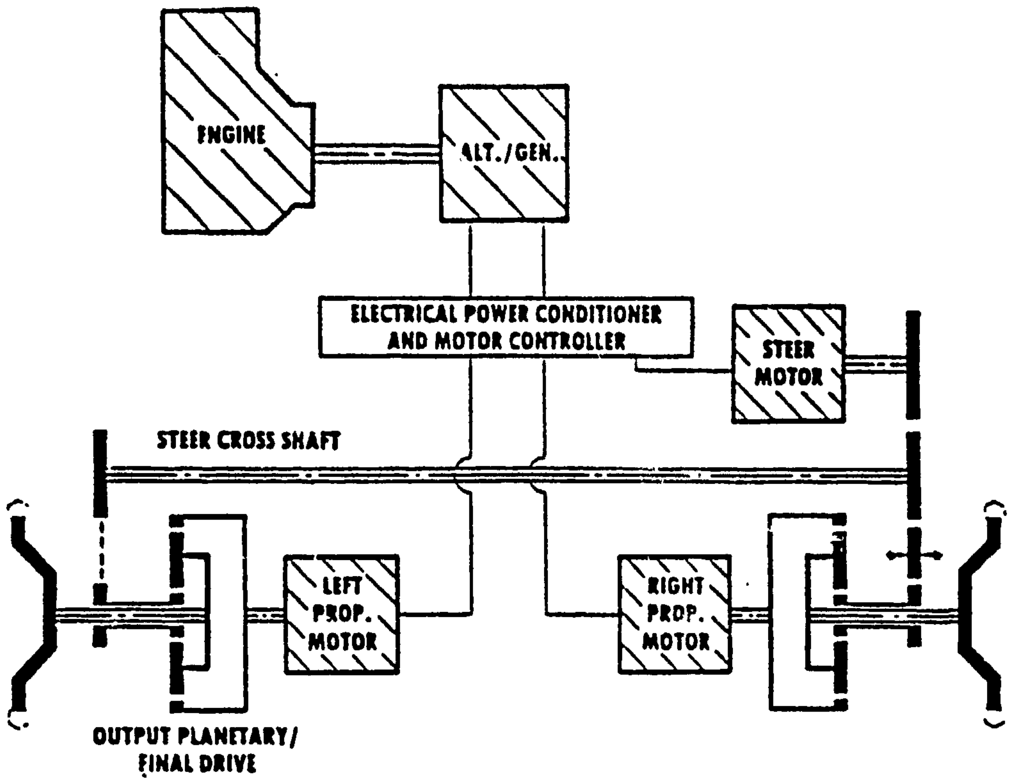


Figure 5-60. Configuration III Schematic

5.8. Configuration IV

5.8.1. Several attempts were made at combining electric drive systems with dual power path transmission arrangements analogous to known hydromechanical units. The Allison CVX 650, GE HMPT and conventional split power flow hydromechanical arrangements were reviewed. In all cases complex gear arrangements were required, and all except the GE HMPT required additional steer drives superimposed on the propulsion power path via combining planetary gearing.

5.8.2. Compared to the configuration I and II, described elsewhere in this report, the dual path arrangements appeared to offer no advantage. In addition, these arrangements would be larger, heavier, more complex and require more development time than the other configurations.

5.8.3. Customer supplied data on the CVX 650, Figure 5-61., was evaluated to gain knowledge of the operation of this transmission and to evaluate the proposed electrical conversion. The data provided was incomplete so that only the Reverse and 1st and 2nd ranges could be analyzed numerically. This analysis appeared sufficient to establish trends however, and power levels in 3rd range were estimated.

5.8.4. It was found that the range section of this transmission was a complex arrangement using both input and compound hydromechanical power splitting as the ranges are shifted. Both hydrostatic units are variable displacement. The rotation of the "motor" unit was reversible going from +3000 RPM to -3000 RPM while the "pump" varied from +1000 to +3000 RPM. The hydraulic power flow was found to be maximum in reverse when 175% of engine gross HP was recirculating through the hydrostatic units. In first range this hydraulic HP flow was 57% and in second approximately 25%. Maximum motor torque occurs when transmission output speed is zero and the motor is stalled with the pump running at 66% speed. To replace the hydrostatic drive in such a system the electrical system will need the same characteristics. The hydrostatic drive in the CVX 650 occupies a space approximately 10" long x 20" dia., an electrical drive operating at the same speed is estimated at 24" dia. x 27" long. Smaller electrical systems operating at high speeds could be used if suitable gearing arrangements could be devised, high ratio reduction gears and/or high pitch line velocity transfer gearing are anticipated in such systems. A separate electrically driven steer drive of similar size would also be required. This approach was rejected for the above reasons. The power flow analysis for the CVX 650 is included as part of Appendix D of this report.

5.8.5. Figure 5-62. is a representation of the GE HMPT transmission. The hydraulic units in this transmission operate at similar speeds and torques to those of the CVX 650. Conversion to electrical power utilizing small high speed devices would again require a long development time at considerable cost. In 1st range all power flow would be through the electrical units and the transmission performance would be

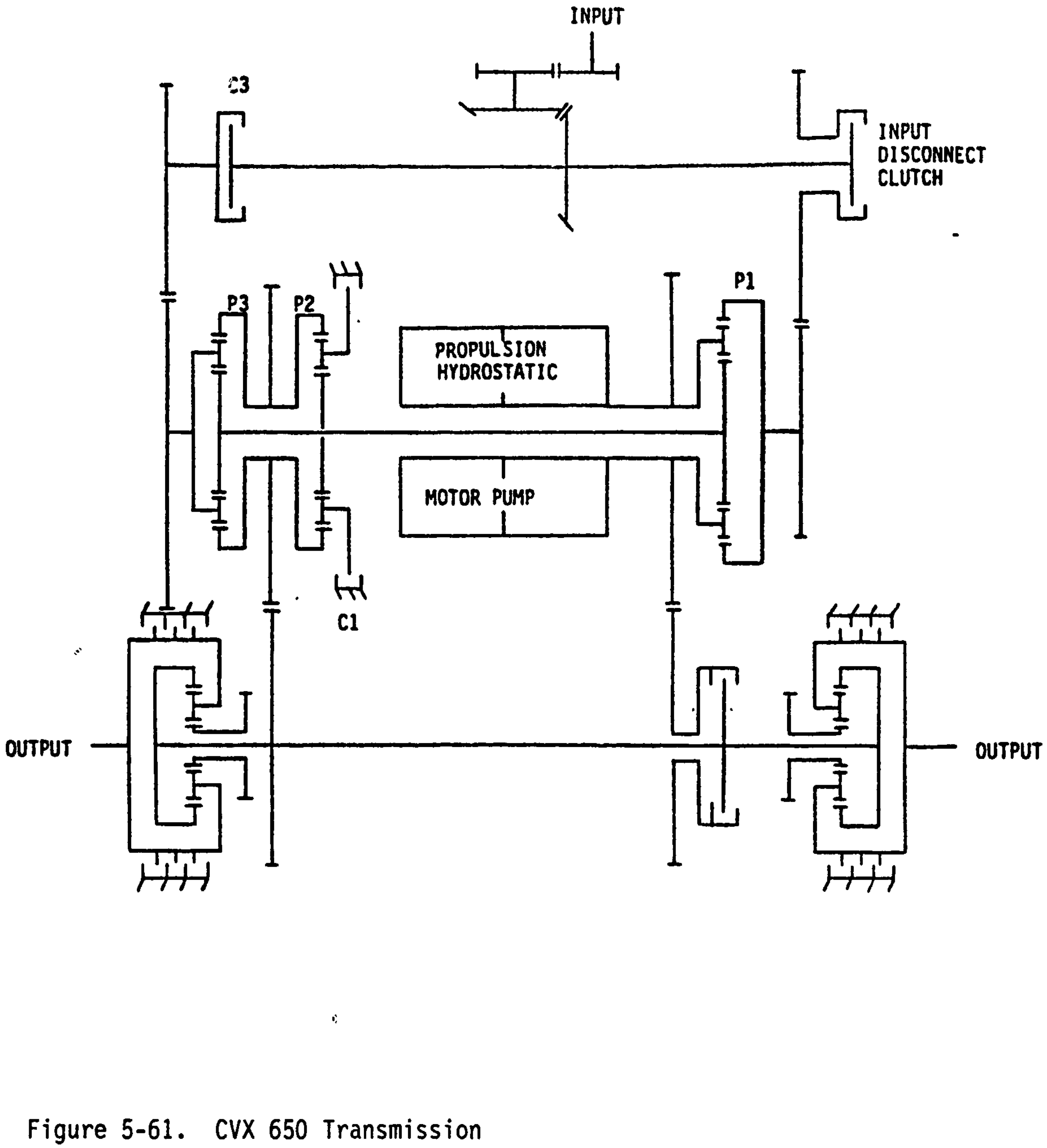


Figure 5-61. CVX 650 Transmission

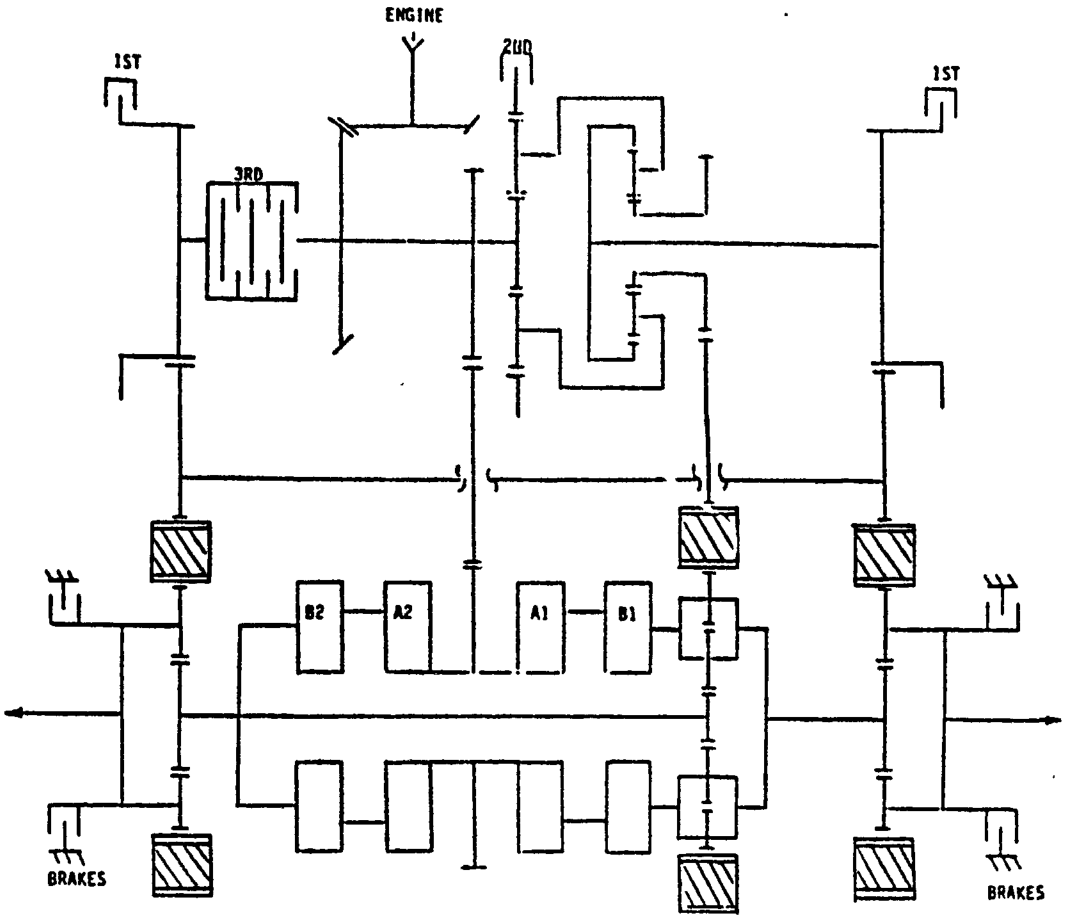


Figure 5-62. GE HMPT Transmission

the same as Configuration I and II. Electromechanical power splitting takes place in the 2nd and 3rd ranges thus the potential for some efficiency gains over Configuration I and II exists under some operating conditions. Here again electrical units of the same performance as the hydraulic units they replace would be larger and heavier, incorporating higher speed smaller units would require major revisions to the gear arrangement and again high reduction ratio and high pitch line velocity transfer gearing is anticipated. This approach was also rejected. Power flow analysis of the HMPT transmission is included as part of Appendix D.

5.8.6. Combination power shunt propulsion systems and conventional cross drive steer systems as shown in Figure 5-63. were reviewed. Because all configurations utilized the same steering system, the power shunt propulsion system was the key discriminator. It was found that these types of transmissions are well established, having been used and evaluated in various forms since the early 1900's. Studies published in 1960 by Block and Schneider, and in SAE Vol 68 concluded that although increasing peak efficiency split torque transmission degrade stall torque, regenerative transmissions on the other hand increased stall torque but suffered a reduction in peak efficiency. Neither transmission type was recommended for applications requiring maximum overall performance. For this study performance curves were developed for both split torque and regenerative drives with homopolar motors and generators. This data was compared with that of Configuration II. Since Configuration II was judged best for overall performance and mechanical simplicity and equal or less for system weight and space, these (Configuration IV) concepts were rejected. Schematics and power flow analysis for systems studied is included as Appendix D.

5.9. Alternate Configurations

The alternative configurations were evaluated and found generally to have some merit, but they do not have such outstanding characteristics that they should be added to the final evaluation matrix. Concepts for four alternative configurations were developed during the proposal effort. They were sufficiently different from the basic concepts that it was decided to include them as additional possibilities. These concepts are illustrated in Figures 5-64. and 5-65. Descriptions and conclusions for each alternate follow.

5.9.1. Configuration IIB. This configuration adds a pair of hydrodynamic couplings to the system in a manner similar to the Renk transission used in the Leopard II. In the Renk transmission, a pair of dump and fill couplings are used to help power the steering cross shaft. By transmitting part of the required power through the couplings, the size, weight, and cost of the continuously variable hydrostatic steering drive is significantly reduced. The couplings are relatively small and of a simple design. The objective of Configuration II was to similarly reduce the steering loads required from the electric steer motor, and thus reduce the component size in the electric steering drive.

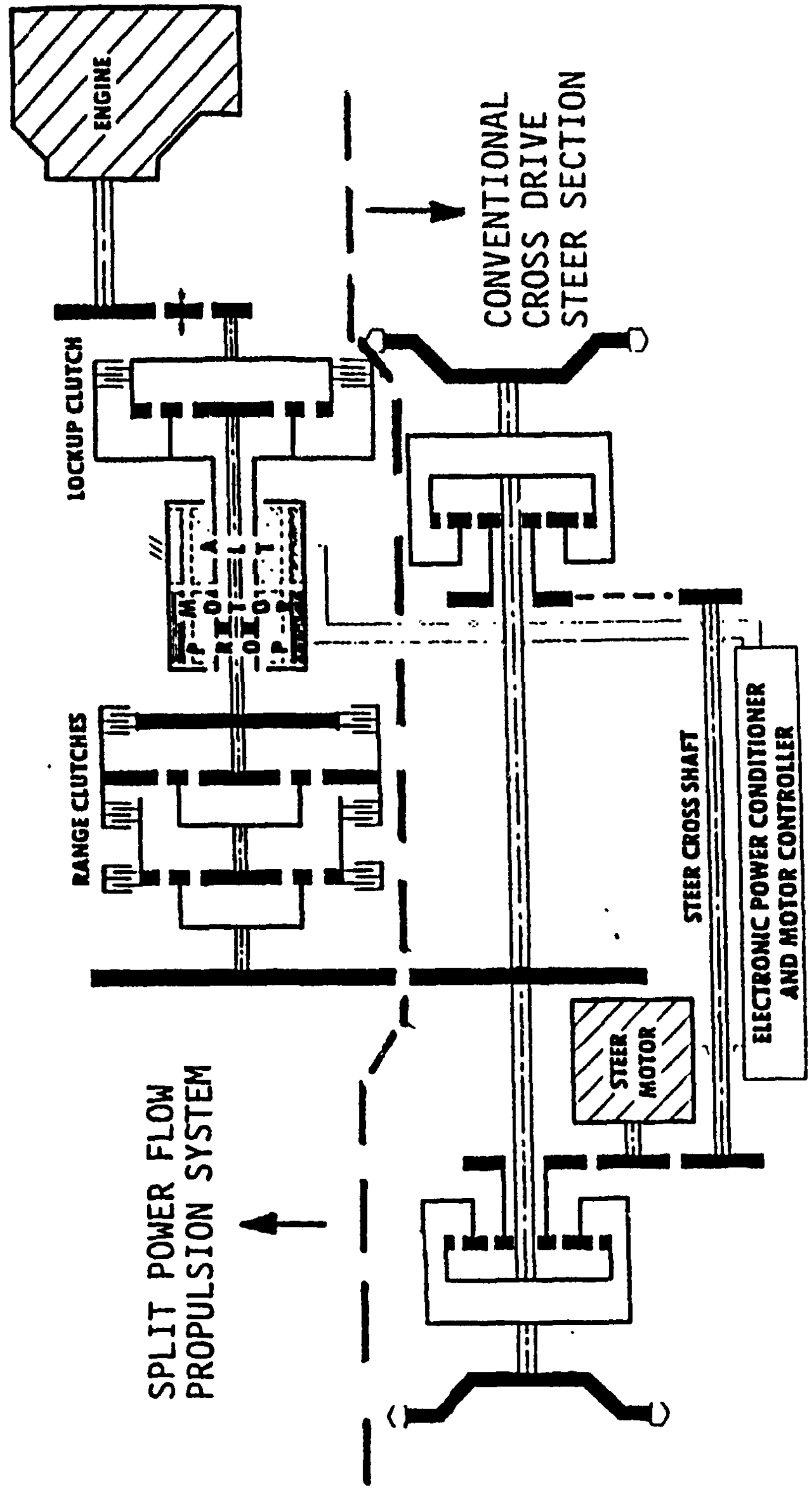
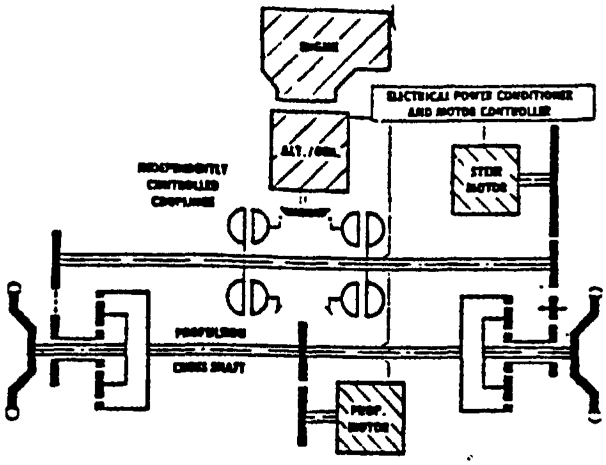
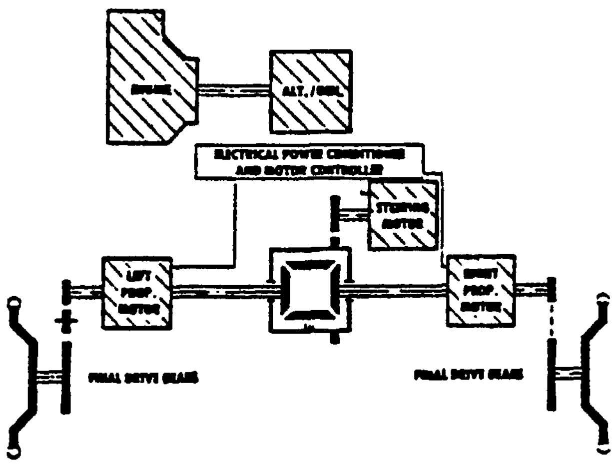


Figure 5-63. Split Power Flow Configuration

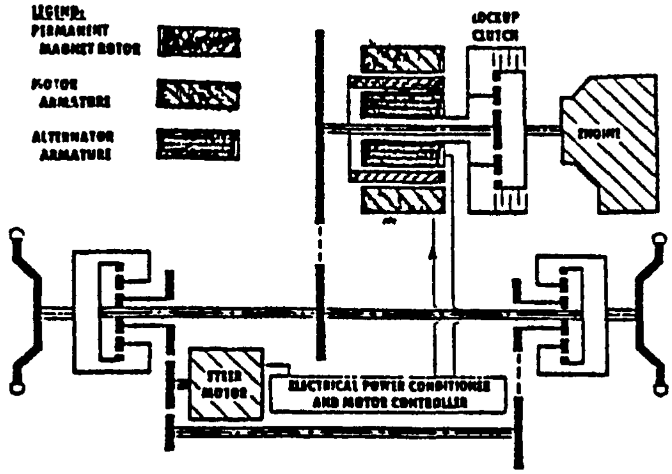


Configuration IIB

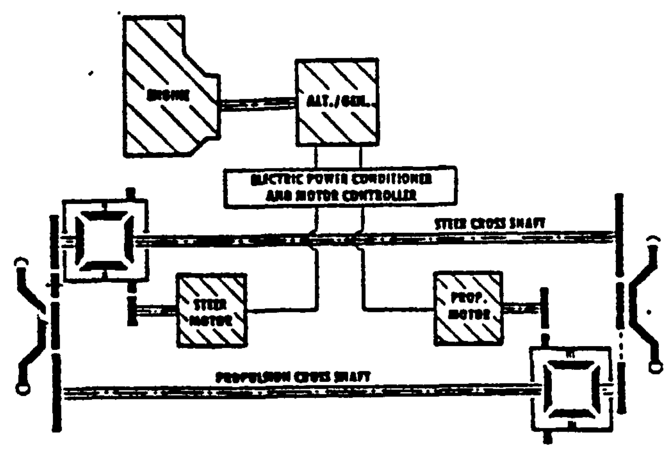


Configuration IIIB

Figure 5-64. Alternate Configurations (IIB & IIIB)



Configuration V



Configuration VI

Figure 5-65. Alternate Configurations (V & VI)

The system would add to overall complexity of both the controls and the mechanical design of the system. The tradeoff in space and weight appear to be marginal, as more components are needed to reduce the size of others. Since the system is basically a Configuration II, it was concluded that the system should be given a quantified evaluation if Configuration II ever reaches a design phase.

5.9.2. Configuration IIIB. This concept provides an efficient regeneration path between the motor armatures. Since the steer motor and the outside propulsion motor would share the loads during turns, there should be a moderate reduction in the size of the total electric drive system. The mechanical design of the system is very simple and compact. The problem of scheduling power to the three motors to simultaneously get effective steering and optimum load sharing to minimize motor loads is quite complex.

The complexity of the required analysis to confirm control feasibility appeared to be beyond the scope of this initial investigation. Since it has a number of similarities to Configuration III, a detailed analysis of this alternative should be made prior to design of a Configuration III system.

5.9.3. Configuration V. This configuration offers a potential improvement in efficiency, particularly in the higher speed ranges. The torque reaction from the rotating generator field is delivered to the output shaft by the generator armature, providing an extremely efficient power shunt.

The requirement for the special motor-generator unit is not consistent with the stated project schedule, and so the concept was deleted from the viable candidates list.

5.9.4. Configuration VI. This concept was functionally similar to Configuration II. The propulsion motors and steer motors in either system sustain similar loads under similar operational conditions. It offers an alternate mechanical arrangement that could be advantageous in some vehicles, and a potential for improved commonality of parts.

Since the concept does not aid in discriminating between various electric drive systems, additional study of this concept did not appear appropriate for this contract effort.

5.10. Concept Screening Description

The concept screening analysis determined that the three best Electric Drive concepts were (not in order of preference): Configuration I High Frequency Induction motor; Configuration I Homopolar motor; and Configuration II High Frequency Induction motor.

5.10.1. The concept screen used the criteria outlined below for the analysis, in order of contract specified priority:

- o Performance (.30)
- o Total system volume and space utilization (.25)
- o Technical risk (.20)
- o Weight (.12)
- o Reliability and maintainability (.08)
- o Safety (.05).

The numbers in the parentheses indicates the weighting factors which were established and clarified with TACOM during the initial phases of the study. Any concept which failed to meet the minimum requirements of at least one of the above criteria was eliminated from the analysis.

5.10.2. The technologies and concepts which were initially refined through concept screening are listed below:

- o DC Systems
  - Homopolar Motors
  - Conventional DC Motors (CDCM)
- o AC Systems
  - High Frequency Synchronous Motors (HFSM)
  - Low Frequency AC (LFAC) Motors
  - High Frequency Induction Motors
- o Hybrid Systems - Brushless DC Motors (BDCM).

5.10.3. Table 5-7. summarizes the concepts and technologies which were eliminated during the concept screen for Configuration I systems. Following is a summary description of these eliminated concepts.

5.10.3.1. The Conventional DC Motor (CDCM) was eliminated from further consideration due to performance related speed constraints, excessive volume, and excessive weight. The CDCM must be air cooled due to brush maintenance.

5.10.3.2. The Low Frequency AC (LFAC) Motor concept was eliminated because of excessive weight and volume. The LFAC motor was a commercial/industrial standard and was used as a benchmark to position the other motor candidate characteristics.

Table 5-7. Concepts Eliminated for Configuration I

REASON FOR ELIMINATION	CONCEPTS			
	BDCM	LFAC	HFSM	CDCM
PERFORMANCE	X		X	X
VOLUME		X		X
RISK	X			
WEIGHT		X		X
RAM - D				
SAFETY				

5.10.3.3. The High Frequency Synchronous Motor (HFSM) concept was eliminated because it would only be able to accomplish regeneration during steering through half the motor speed range. At motor speeds less than a half, regeneration would not be possible.

5.10.3.4. The Brushless DC Motor (BDCM) concept was eliminated due to the fact that a complex control system was required to accomplish regenerative steering. Regeneration would be possible as long as the bus voltage to the regenerating motor was reduced. The control system would have to capable of supplying a variable bus to the propulsion motors in order for regeneration to occur. The variable bus would make the BDCM a viable candidate for configuration I.

5.10.4. For Configuration II, any of the technologies selected during the technology survey would be applicable. The High Frequency Induction Motor concept offered a proven technology with minimal risk, while meeting the performance criteria and therefore was selected as a representative concept for the concept screen and comparison.

5.10.5. Table 5-8. shows a weighted trade-off matrix summarizing the comparison of the three selected concepts. The Configuration I High Frequency Induction Motor concept has the highest overall score, 9.87, based on a ten point must system for best overall concept for the criteria. The individual criteria scores were normalized by the weighting factor so that the best possible overall score would be 10.0 and the lowest overall score would be 0.0.

5.10.6. The analysis showed that for performance the three concepts met the contract performance requirements. For system volume and space utilization, the concepts had similar total volumes, with the Configuration I high frequency induction concept offering the most flexibility for component location and space utilization. The homopolar motor's flexibility would be equal to the high frequency induction motor concept for Configuration I if similar component arrangements were used.

5.10.7. The technical risk analysis showed all the systems are at risk for proof of principle hardware by 1987. The Homopolar motor incorporated advanced technologies relative to motor construction.The induction motor is a well established technology. The major risk associated with the induction motor is relative to the development of reliable high power semiconductors required for the bridge circuit. The Configuration II Induction motor concept has risk associated with the mechanical considerations of the system, specifically in the final drive gearing and the output shafting to the drive sprockets.

5.10.8. The induction motor offers the lowest overall weight total, but the differences between the concepts was minimal. The weight of the Configuration II concept may be higher than originally estimated based on the method of construction for the drive housing. This cannot be determined until a final design for build has been completed.

Table 5-8. Comparison Matrix of Selected Concepts (19.5 Ton)

	CONCEPTS 19.5 TON								
CRITERIA	SCORE WEIGHT	CONFIG. I HOMO POLAR MOTOR		CONFIG. I HI FREQ. INDUCTION MOTOR		CONFIG. II HI FREQ. INDUCTION MOTOR		BASELINE MECHANICAL	
PERFORMANCE	.30	10	3.0	10	3.0	10	3.0	9	2.7
SYSTEM VOLUME & SPACE UTILIZATION	.25	8	2.0	10	2.5	9	2.25	7	1.75
TECHNICAL RISK	.20	9	1.8	9	1.8	9	1.8	10	2.0
WEIGHT	.12	9	1.08	10	1.2	10	1.2	8	0.96
RELIABILITY / MAINTAINABILITY	.08	9	0.72	8	0.64	8	0.64	10	0.8
SAFETY	.05	9	0.45	8	0.40	8	0.40	10	0.5
TOTALS	1.0	54	9.05	55	9.54	54	9.29	54	8.71

5.10.9. Reliability and Maintainability (RAM-D) was the most subjective of the criteria to evaluate for the concepts. The homopolar motor technology has had the majority of the current research directed toward "burst" power generation. There is also some question relative to the reliability of high power semiconductors required for the induction motor controller. It was determined that the current research in Japan relative to high power semiconductors may reduce the reliability burden for the induction motor concepts. Better reliability and maintainability assessments will be possible after proof of principle hardware is constructed in follow on programs.

5.10.10. The induction motor technology was determined to have the highest safety risk based on the high system operating voltages (300-600 volts). The homopolar motor system operates on a 24 to 28 volt system which is compatible with military vehicle standards. Safety concerns would be minimized by a proper hazard analysis during the final design of the systems.

5.10.11. The mechanical baseline did not meet the contract specified performance requirements for the speed on grade requirement at 0.7 TE/GVW. It has minimal flexibility relative to component layout and space utilization and the highest overall system volume.

5.10.12. The mechanical baseline has the lowest technical risk of all the compared systems and the highest overall weight. It has good reliability and maintainability characteristics with the best overall safety considerations based on the currently applied hardware in the field.

5.10.13. The selected electric drive concepts were determined by the analysis to be an improvement over the mechanical baseline for the 19.5 ton vehicle with similar results for the 40 ton vehicle. Based on the criteria specified, the concept of an electric drive for main vehicle propulsion will give better performance than the current mechanical systems.

5.11. Vehicle Performance

The recommended electric drive systems effectively meet the specified performance criteria. The performance requirements of the specification have been divided into two groups: general performance factors and braking performance factors.

5.11.1. General Performance Factors.

- o Tractive Effort (TE) versus Speed. The requirements are plotted in Figure 5-66., and are shown numerically in Table 5-9. following.

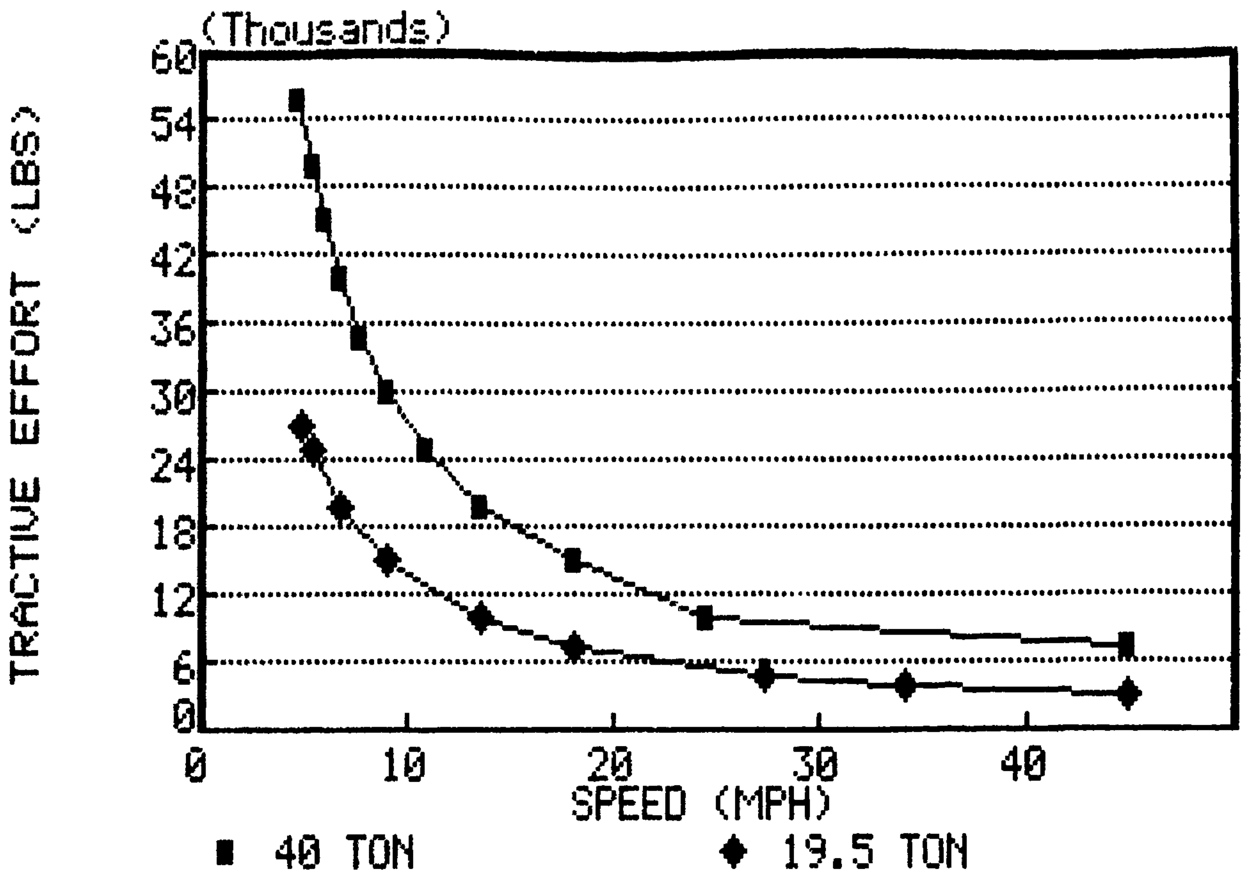


Figure 5-66. Tractive Effort Versus Speed

Table 5-9. Tractive Effort Versus Speed

40 Ton TE (Lbs)	19.5 Ton TE (Lbs)	Speed MPH
56,000	---	4.9
---	27,300	5.0
50,000	25,000	5.5
45,000	---	6.1
40,000	20,000	6.8
35,000	---	7.8
30,000	15,000	9.1
25,000	---	11.0
20,000	10,000	13.7
15,000	7,500	18.3
10,000	5,000	27.4
7,500	3,042	45.0

- o Acceleration: The acceleration goal was to achieve a speed of 20 mph within seven seconds, with no time delay allowed for throttle response.

- o Speed on grade: This requirement was provided in the form of a curve, which is reproduced in Figure 5-67.

- o Maximum Grade Startability: Ths vehicle must be capable of starting on a 60 percent grade.

- o Minimum Turn Radius Versus Speed: This was not a contract requirement, but turn radius versus speed was found in many cases to induce the critical loads in the drive systems. Data on this subject may be found in Appendix B.

5.11.2. Braking Performance Factors.

- o Deceleration rates: The vehicle is required to decelerate at 7 meters per second per second or more (peak) and 5 meters per second pers second or more (average) from maximum speed when operating on a level hard surfaced road.

- o Cycle test: The vehicle is required to make at least 25 stops at 3 minute intervals from 60 kilometers per hour, with a 5 meter per second per second deceleration rate.

- o Continuous braking: The vehicle must be capable of continuously maintaining a speed of 20 mph on a hard surfaced 15 percent downgrade.

- o Hold on grade: The vehicle brakes must be able to hold the vehicle on a hard surface 60 percent downgrade with the engine off.

The above performance factors will now be discussed in detail.

5.11.3. Tractive Effort Versus Speed. The selected electric drive systems provide tractive effort performance that approaches the specification goals and falls between these goals and the mechanical baseline transmissions. Evaluation of the tractive effort performance of the vehicle focused on three specific operating conditions. The first was the ability to develop a tractive effort at stall that was at least equal to 1.2 times the GVW. This load represents the maximum torque that can be developed without track slip, even under the most severe conditions. The second was the ability to continuously develop a tractive effort of 0.7 times the GVW. This value is typical for military vehicle cooling tests and simulates severe operation such as towing a disabled vehicle. The third was a specified curve that was equal to delivering 73 percent of the gross engine horsepower to the sprockets from 10 to 100 percent of full speed. This level of efficiency represents very good efficiency for conventional systems.

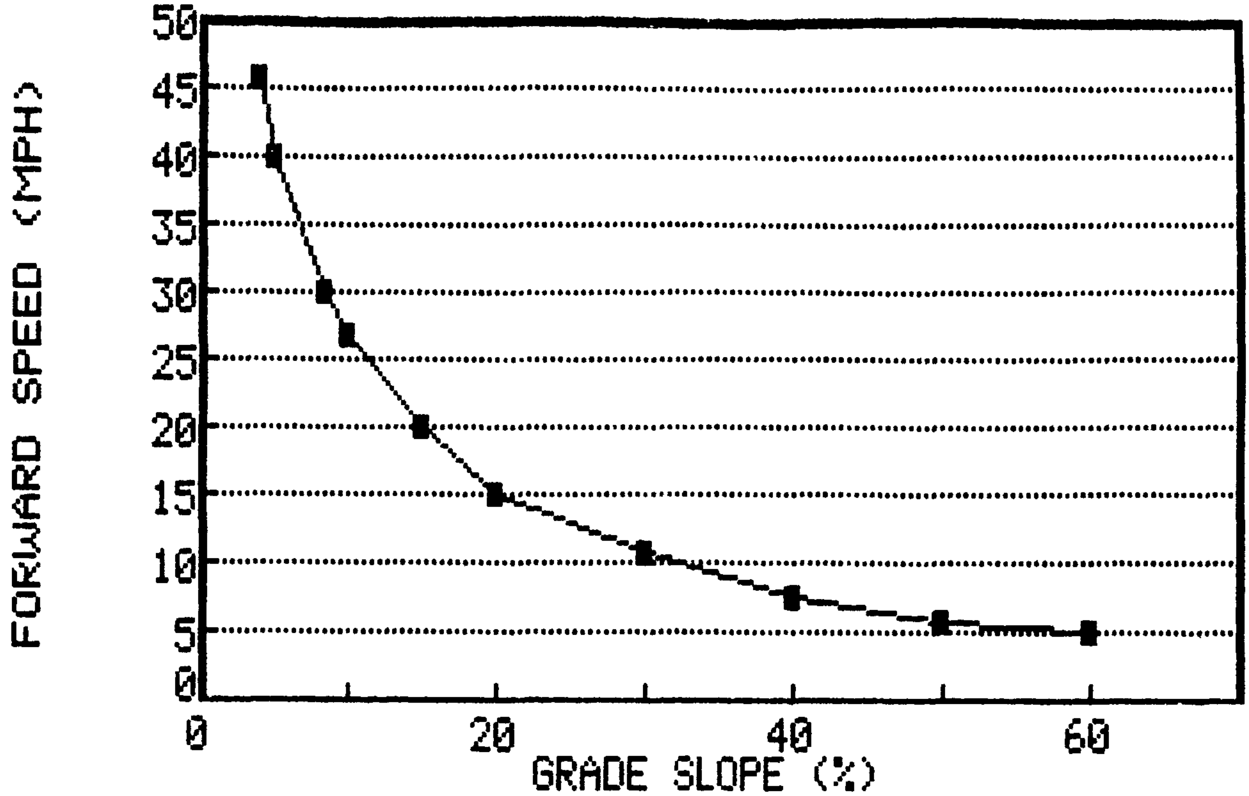


Figure 5-67. Sustained Speed Versus Grade

5.11.3.1. Performance predictions were made for the electric drive systems and for the x-300-4A and ATT-1064 mechanical baseline transmissions. The method of analysis is described in detail in Appendix B. The predictions for the recommended electric drives and the mechanical baselines are plotted against the performance goals in Figure 5-68. for the 19.5 Ton and Figure 5-69. for the 40 Ton vehicle.

5.11.3.2. The overall conclusion was that the recommended electric drive systems would deliver tractive performance that was comparable to the current mechnaical transmissions. The following conclusions were reached through evaluation of the plotted data:

- o All systems have been sized to meet the 1.2 TE/GVW requirement.
- o All recommended electric drives meet the 0.7 TE/GVW requirement, but at approximately 4 mph, slightly below the 5 mph target speed. All of the recommended electric drives outperform the mechanical baseline, which can achieve only 3 mph at the 0.7 TE/GVW condition.
- o All recommended electric drives closely approach the 73 percent efficiency curve at speeds from 4.5 to 45 mph. In general, they can achieve a speed about 1 mph slower that the target curve, but they outperform the mechanical drive by 1 to 7 mph.

5.11.4. Acceleration Performance.

5.11.4.1. All selected electric drive systems provide outstanding acceleration that exceeds both specified and mechanical baseline performance. Evaluation of acceleration performance of the electric drives was made by comparing their calculated performance with specification requirements and with the X-300-4A and the ATT-1064 mechanical transmission performance. The specification requires acceleration on hard level surface to 20 mph within 7 seconds, with no allowance for throttle delay.

5.11.4.2. Acceleration predictions were made for the electric drive vehicles and for the X-3004A and the ATT-1064 mechanical baseline transmissions by use of the analytical methods detailed in Appendix B. The results are compared with the specification requirements for the 19.5 ton vehicle in Figure 5-70. and the 40 ton vehicle in Figure 5-71.

5.11.4.3. The following conclusions were reached:

- o All recommended electric drive systems significantly exceed the specification requirements.

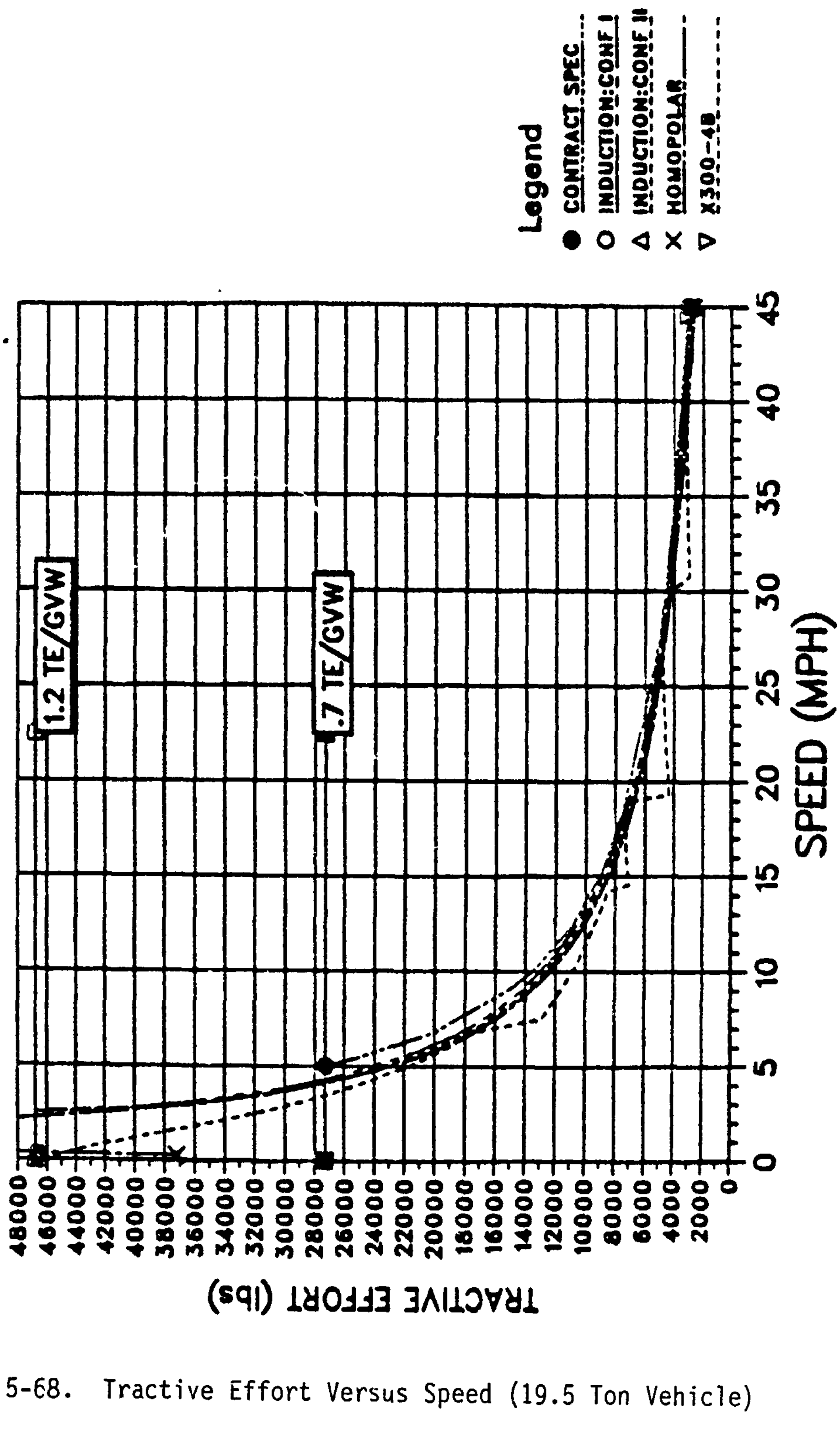


Figure 5-68. Tractive Effort Versus Speed (19.5 Ton Vehicle)

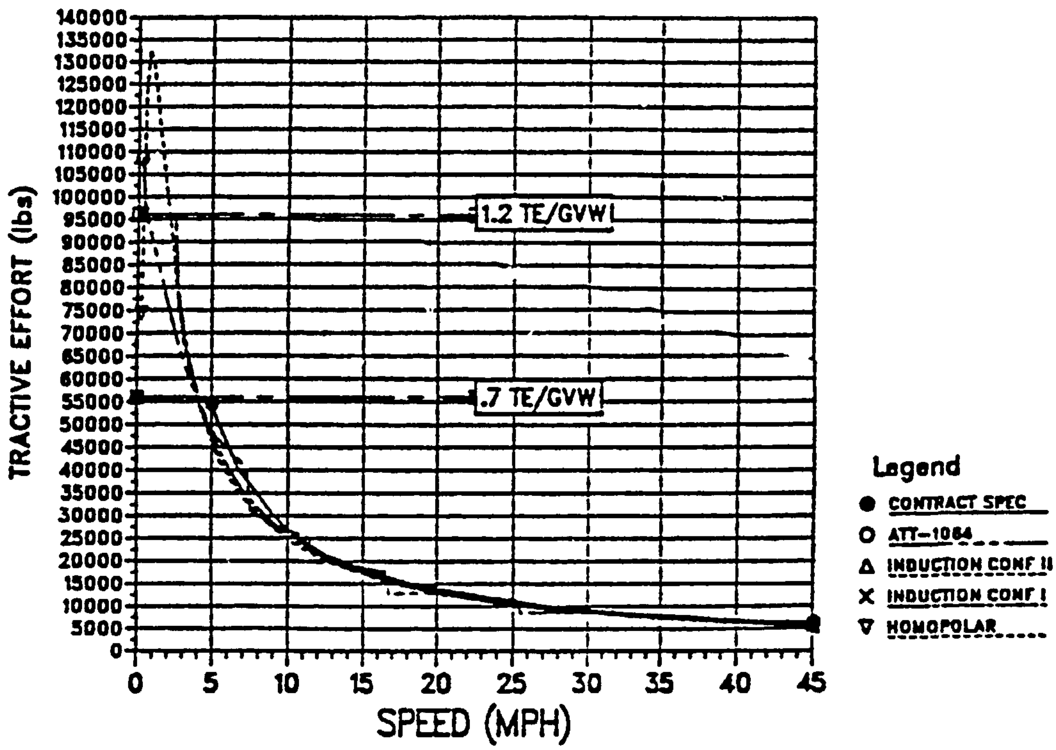


Figure 5-69. Tractive Effort Versus Speed (40 Ton Vehicle)

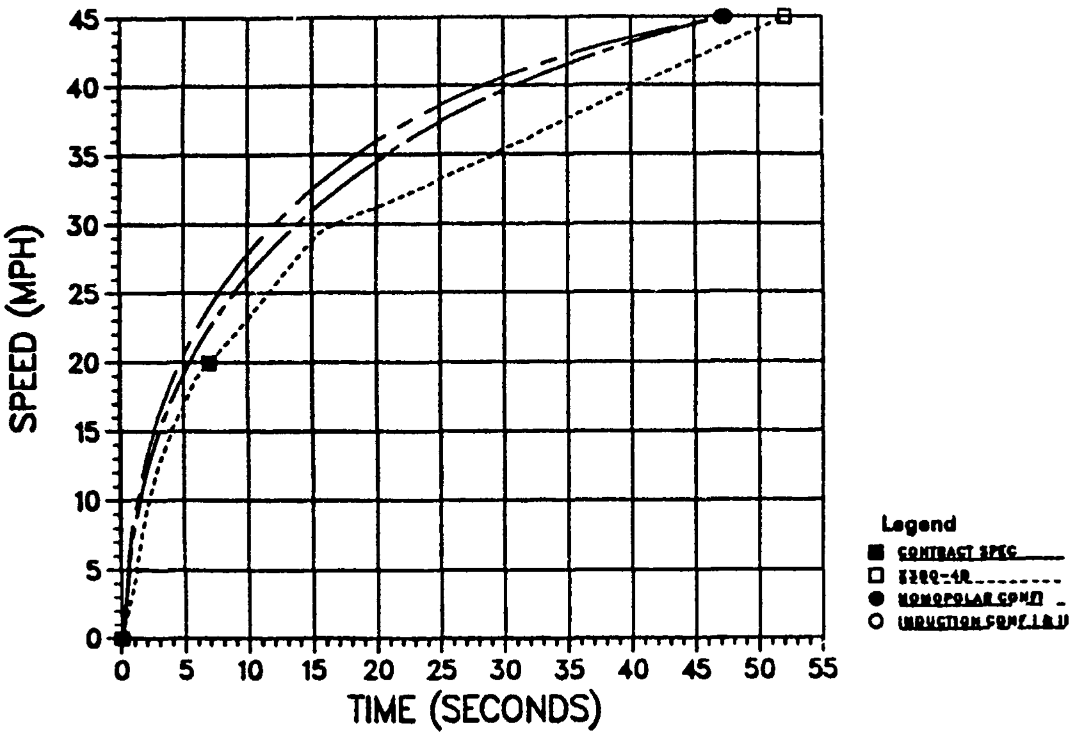


Figure 5-70. Acceleration Performance (19.5 Ton Vehicle)

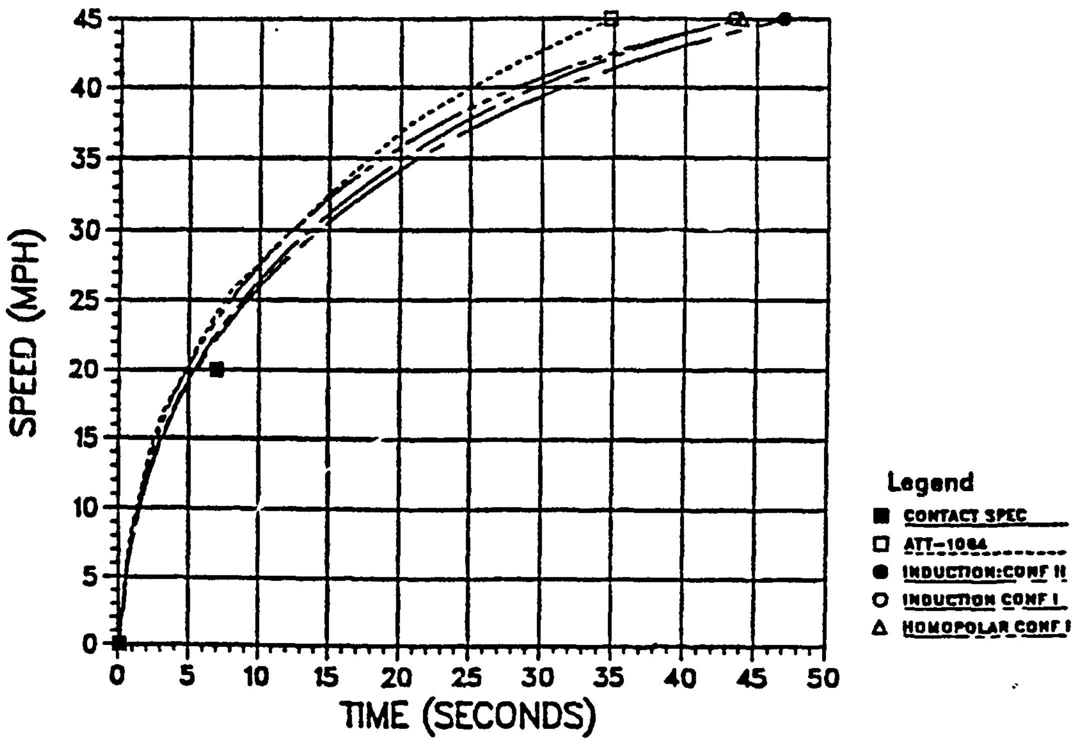


Figure 5-71. Acceleration Performance (40 Ton Vehicle)

- All recommended electric drive systems substantially outperform the mechanical baseline.

- There is greater difference in the acceleration performance of the various electric drives than was apparent in their available tractive effort versus speed characteristics.

5.11.4.4. The speed versus distance data for the electric drives and the mechanical baseline are plotted in accoerdance with contract requirements in Figure 5-72. (19.5 ton vehicle) and Figure 5-73. (40 ton vehicle).

5.11.4.5. The differences in performance between the electric drives are primarily the result of different armature inertias. Rotational kinetic energy was calculated for the subsystems (engine and accessories, generator or torque converter, motor or transmission, and track and suspension) and the results totaled. Figure 5-74. shows the results. The major difference is in the motors and transmissions, with the induction motor inertia substantially higher than the other systems. The induction motor rotor is a solid structure of magnitic material with integral current bars imbedded. In this case, the volume of iron is required to support the magnetic flux transformed to the rotor. In the homopolar motor, the rotor is required only to pass the field flux through the rotating sleeve. This allows for a significant reduction in non-essential magnetic material structure and the transfer of only the required flux to the stationary structural portion of the machine. There are minor differences in the generator and transmission inertias, with the homopolar system having the greatest inertia. To put these factors into proper perspective, while total rotational inertias are substantial, the effect is minor if the total vehicle inertia is considered (see Figure 5-75.). The differences in total inertia appear consistent with the differences in the acceleration performance of the various drive systems.

5.11.4.6. The major performance difference between the electric drive systems and the mechanical baseline results form the difference in the power input from the engine to the propulsion systems. In the case of the electric drives, control of the field currents permits variation of the generator input speed versus input torque characteristics. This is used to completely isolate engine loading from the vehicle operating conditions. It then becomes possible to match the engine load so operation at maximum horsepower RPM can be achieved at any vehicle speed. In the case of the mechanical transmissions, lock-up clutches are used to avoid converter losses in the highere speed ranges. The engine speed in each range then becomes a function of the vehicle speed. To avoid overspeeding the engine, it is necessary to match maximum horsepower engine speed to the maximum vehicle speed in that range. At any lower speed in each range, the engine speed is necessarily lower and engine output correspondingly reduced. This loss of available power degrades vehicle

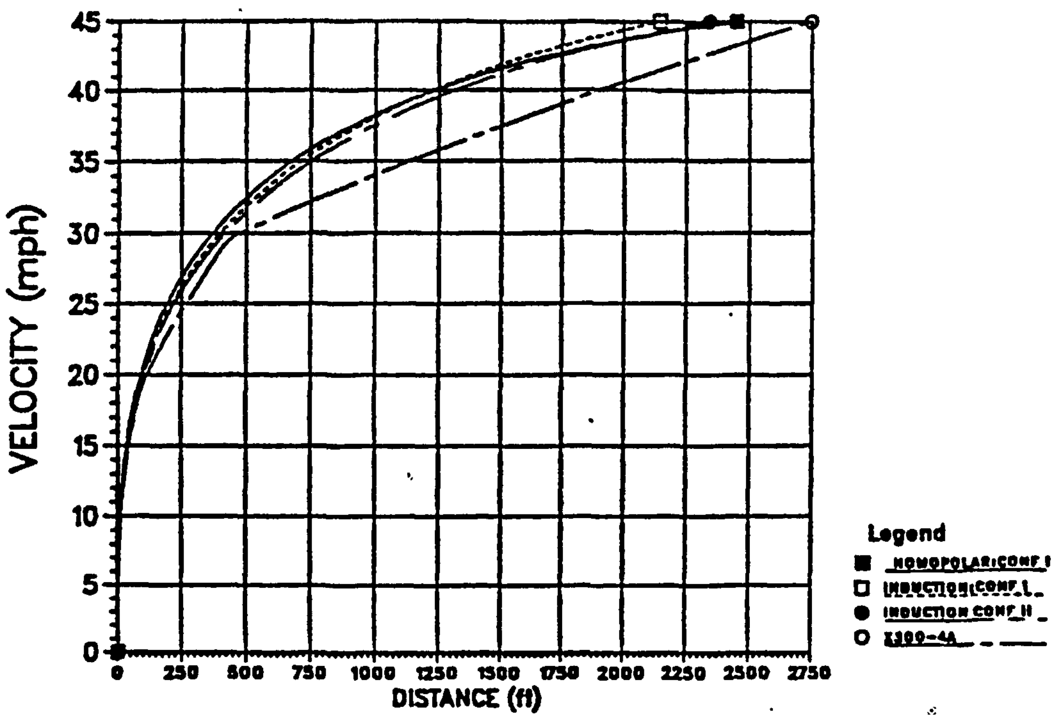


Figure 5-72. Speed Versus Distance (19.5 Ton Vehicle)

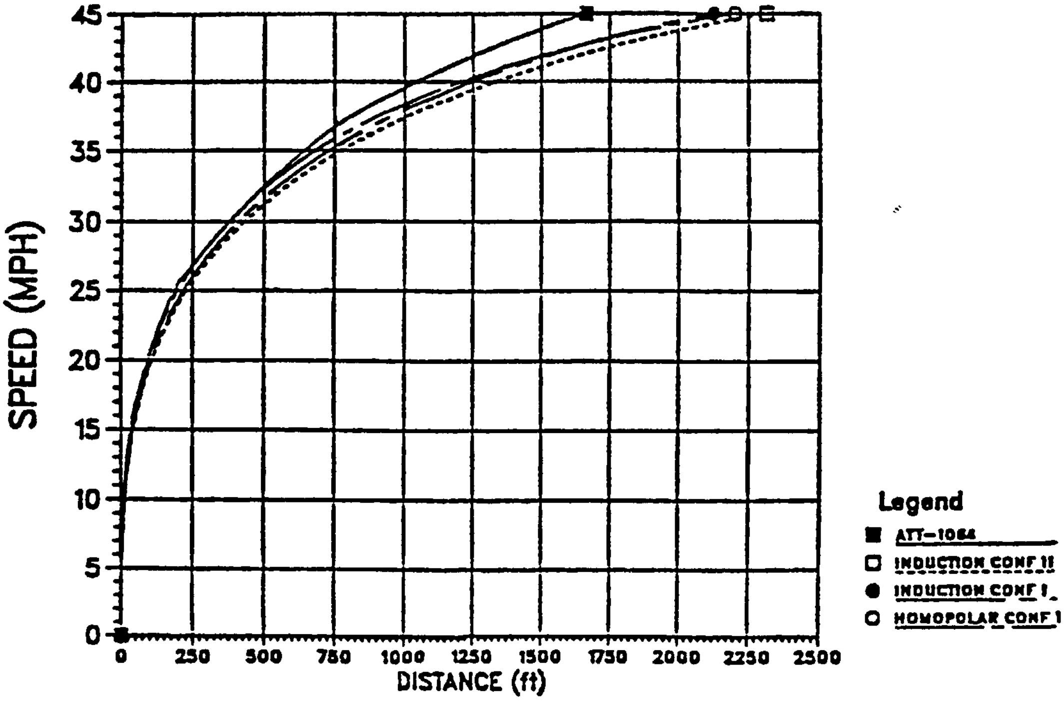


Figure 5-73. Speed Versus Distance (40 Ton Vehicle)

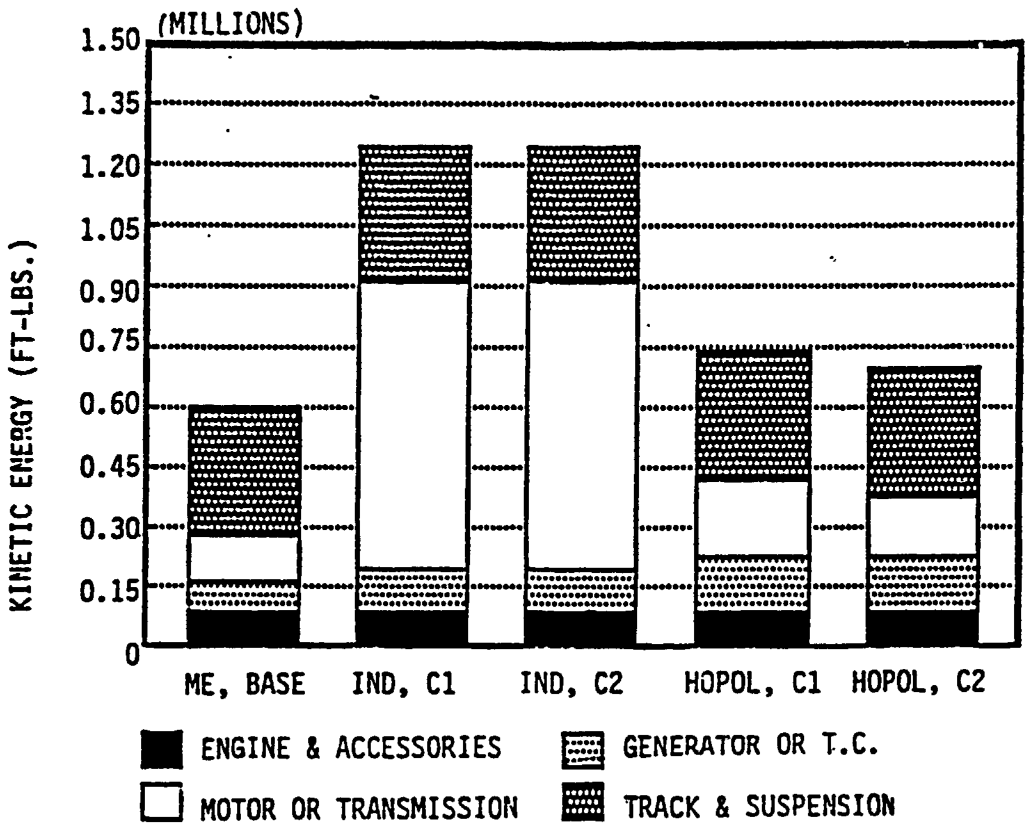


Figure 5-74. Kinetic Energy in Rotation (45 mph)

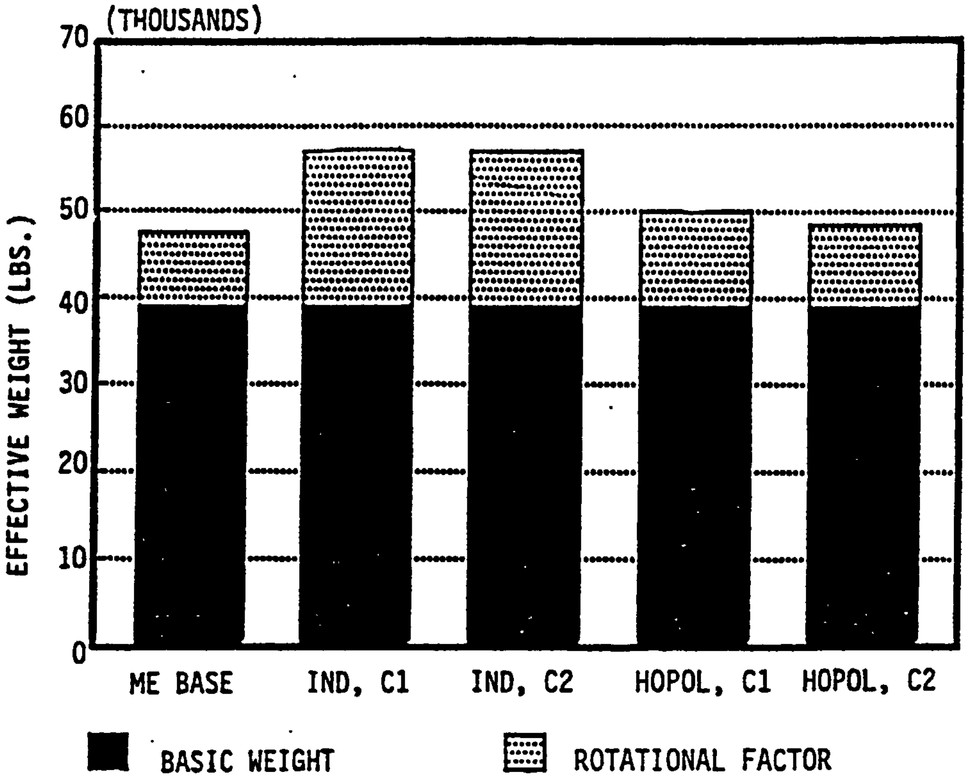


Figure 5-75. Impact of Rotational Inertia

performance, regardless of the fact that efficiency in the narrow sense is comparable to electric systems.

5.11.4.7. The combination of an efficiency curve that is broad, flat and reasonably high, combined with ideal engine speed regulation enables the electric drive systems to provide superior acceleration performance.

5.11.5. Speed on Grade.

5.11.5.1. The recommended electric drive systems provide satisfactory speeds on grade that approaches contract goals, substantially exceeding the 19.5 ton mechanical baseline performance and effectively equaling the 40 ton mechanical baseline performance. The speed on grade performance was calculated using data from the "Tractive Effort Versus Speed" curves (Figures 5-67. and 5-58.). At each grade, speed was increased to the point where the total resistance equaled the tractive effort. Factors included in the total resistance are rolling resistance, wind resistance, and grade resistance. Details of the analytical procedure are given in Appendix B.

5.11.5.2. The speed versus grade performance from the contract, from analysis of the recommended electric drives, and for the X-300-4A mechanical baseline transmission are plotted in Figure 5-76. for the 19.5 ton vehicle. Similar data for the 40 ton vehicle is shown in Figure 5-77. Evaluation of the results shows that performance of the several electric systems is very similar and closely approximates the target curve from the Statement of Work. The mechanical system provides lower performance, primarily because transmission input characterisatics do not enable the engine to produce full power at many vehicle speeds.

5.11.5.3. Evaluation criteria based on speed versus grade produce the same conclusions as the "Tractive Effort Versus Speed" curves. The recommended electric drives are able to provide satisfactory automotive performance.

5.11.6. Maximum Grade Startability. Good efficiency at low speeds enables the selected transmissions to provide positive starts on a 60 percent grade. The performance analysis for such starts followed the same basic principles as the other tractive effort and acceleration analysis, and is described in detail in Appendix B. The 60 percent grade resistance is GVW * sin(ATN(grade/100)), which in the case of the 19.5 ton vehicle is 22,016 pounds. Rolling resistance on a hard surface is assumed as 100 pounds per ton, which comes to 1950 pounds for the 19.5 ton vehicle. Air resistance at low speeds is negligible. These combined resistances require a TE/GVW of .615. Since there are other contract requirements for a momentary 1.2 TE/GVW and a continuous 0.7 TE/GVW, the 60 percent grade start is not the restrictive requirement.

Acceleration analysis was made for the 60 percent grade starts and the results are shown in Figure 5-78. These curves show positive starts that promptly reach grade limited speed.

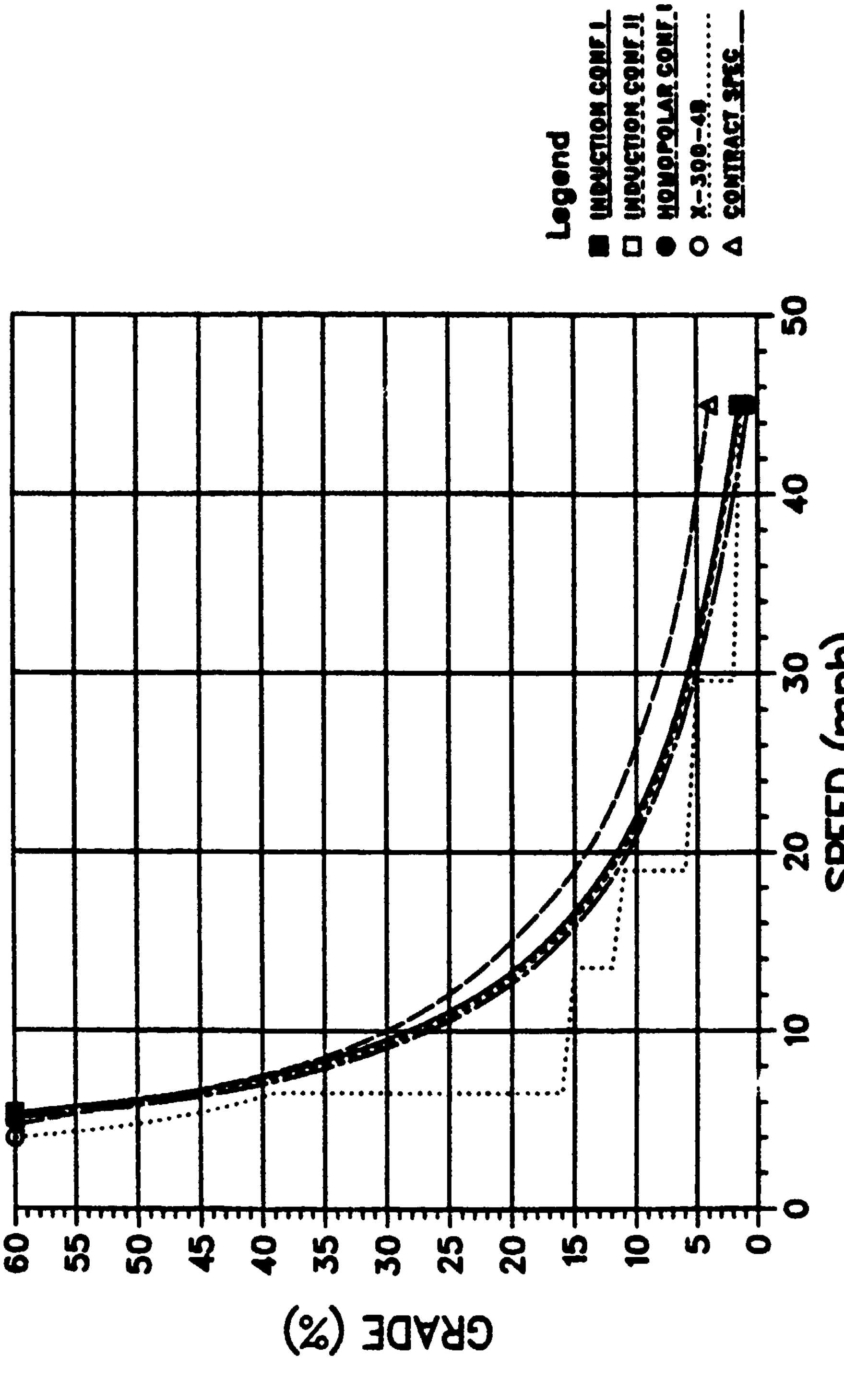


Figure 5-76. Speed Versus Grade (19.5 Ton Vehicle)

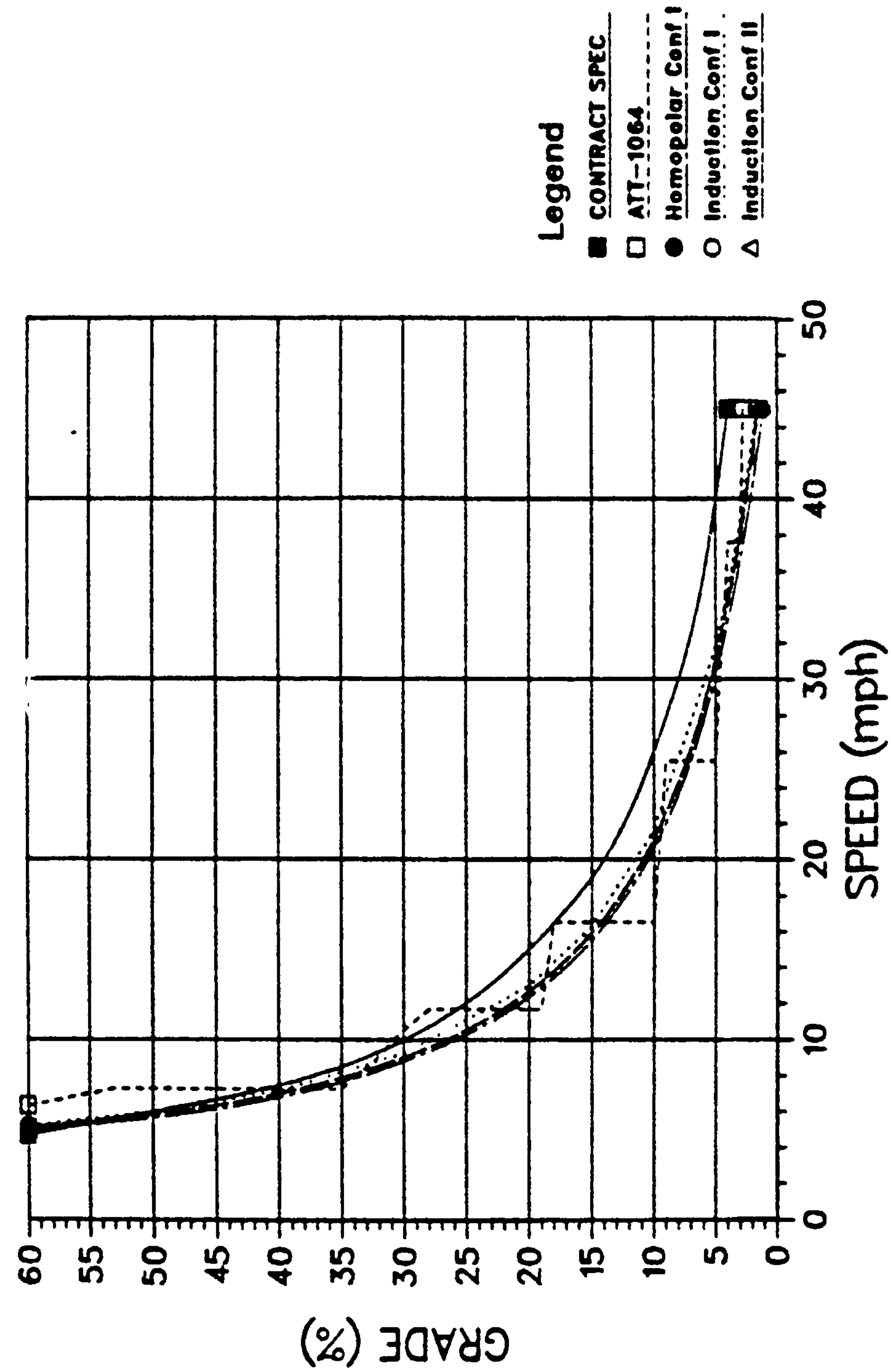
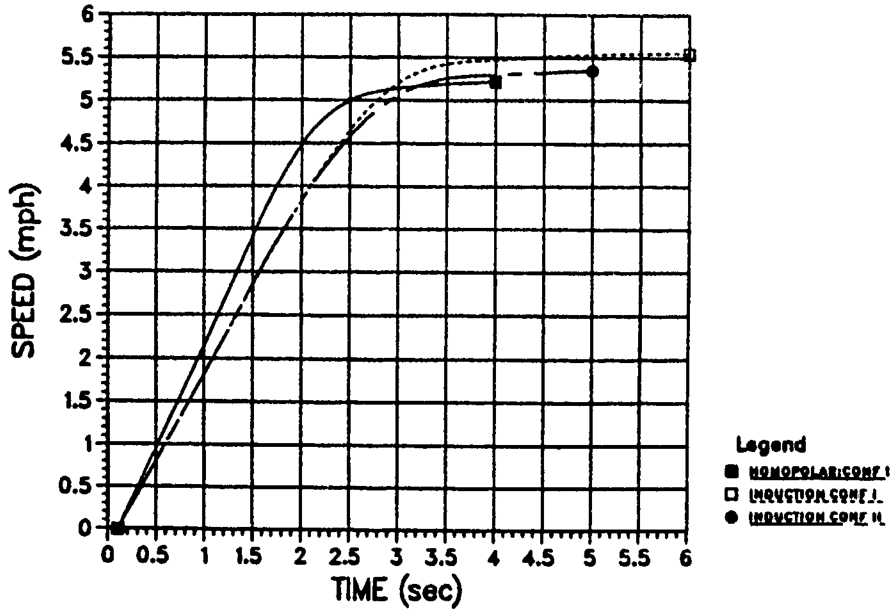


Figure 5-77. Speed Versus Grade (40 Ton Vehicle)



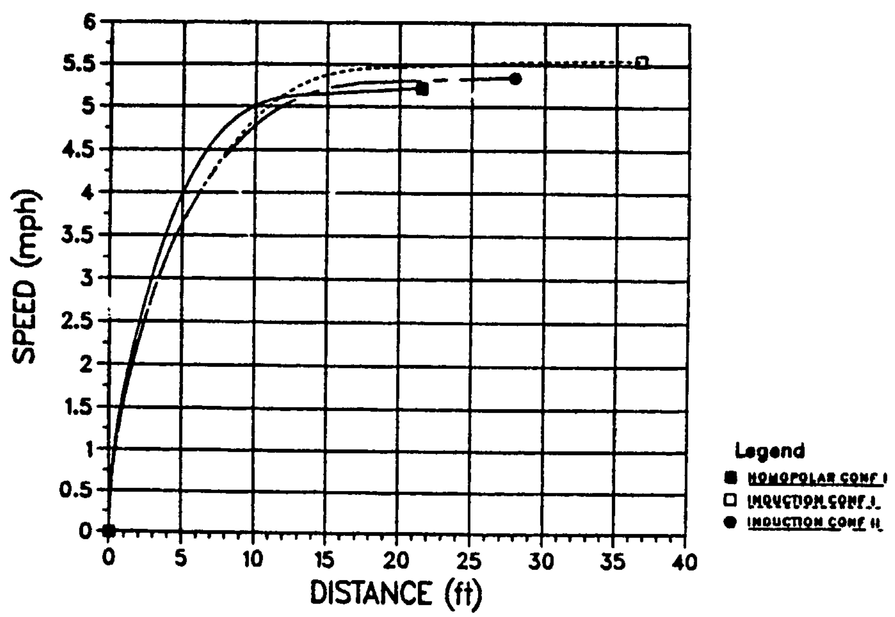


Figure 5-78. Acceleration and Speed on 60 Percent Grade

5.11.7. Axis Steer Capability.

5.11.7.1. The contract requirement to provide an axis steering capability of 7 revolutions per minute has been found to be easily achieved, and therefore has no significant impact on the system selection. Optimization of the several electric drive systems to meet other contract requirements has provided the low speed drive efficiencies and high motor torque capacities essentioal for superior axis steer performance. Typical sprocket torques for the various required operating modes are as follows:

- o Develop tractive effort of 1.2 GVW . . .20,586 lb-ft
- o Develop TE of 0.7 GVW @ 7 mph. . . . . .12,009 lb-ft
- o Maximum driving steer torque . . . . . . 9,819 lb-ft
- o Axis steer at 7 rpm. . . . . 9,917 lb-ft

5.11.7.2. The surplus of sprocket torque permits axis steer operation at 12 rpm, substantially exceedinig the contract goal.

5.11.8. Minimum Turn Radius Versus Speed.

5.11.8.1. All systems provide nearly identical performance at speeds greater than 15 mph, but at lower speeds the electric drive steering is superior to the X-300-4A mechanical baseline transmission. It is desireable to be able to achieve the smallest possible turn radius at every speed because the small radius represents the agility that makes a vehicle a difficult, evasive target. The analysis was based on a 0.5 G lateral acceleration limit, which was considered representative of aggressive, but not reckless driving for hard surfaces, where skidout can occur at less than 0.7 G.

5.11.8.2. All electric drive systems provided the desirable 0.5 G lateral acceleration at all speeds between 15 and 40 mph. Between 40 and 45 mph, the homopolar Configuration I becomes power limited and does not achieve the 0.5 G. This limitation does not appear critical because violent maneuvers are not frequently made at high speeds due to control and safety considerations. At lower speeds, Configuration II systems provide moderately better steering performance than Configuration I systems. This loss of Configuration I performance in the 5 mph range is caused by the need to hold the inside sprocket near zero rpm to perform the turn. In Configuration I, the sprocket speed and motor speed are directly related. The poor efficiency of motors near zero rpm causes losses reflected in the performance degredation.

5.11.8.3. Full range steering performance is shown in Figure 5-79. Figure 5-80. enlarges the curves below 15 mph so the differences are more clearly visible.

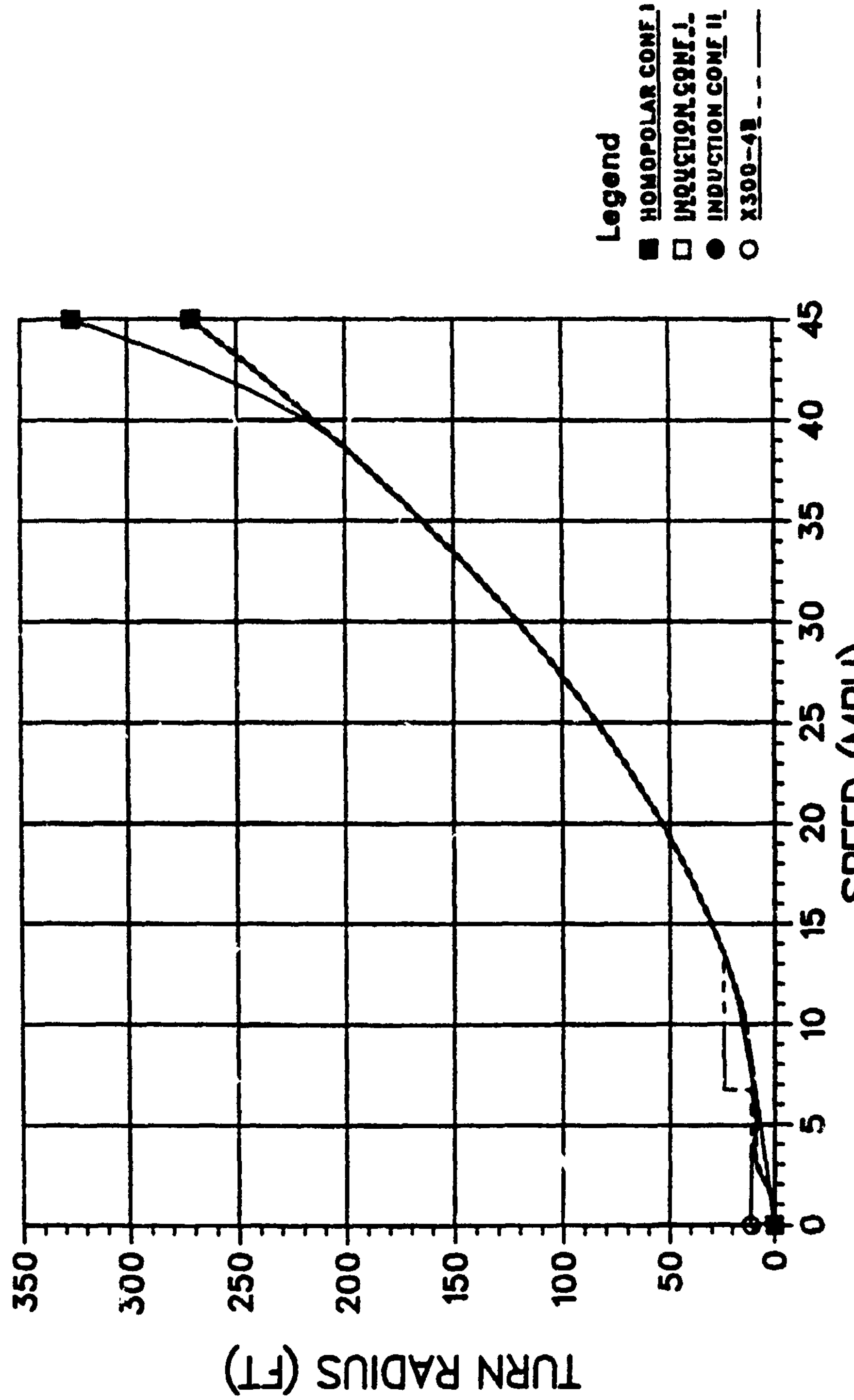


Figure 5-79. Minimum Turn Radius Versus Speed (>15 mph)

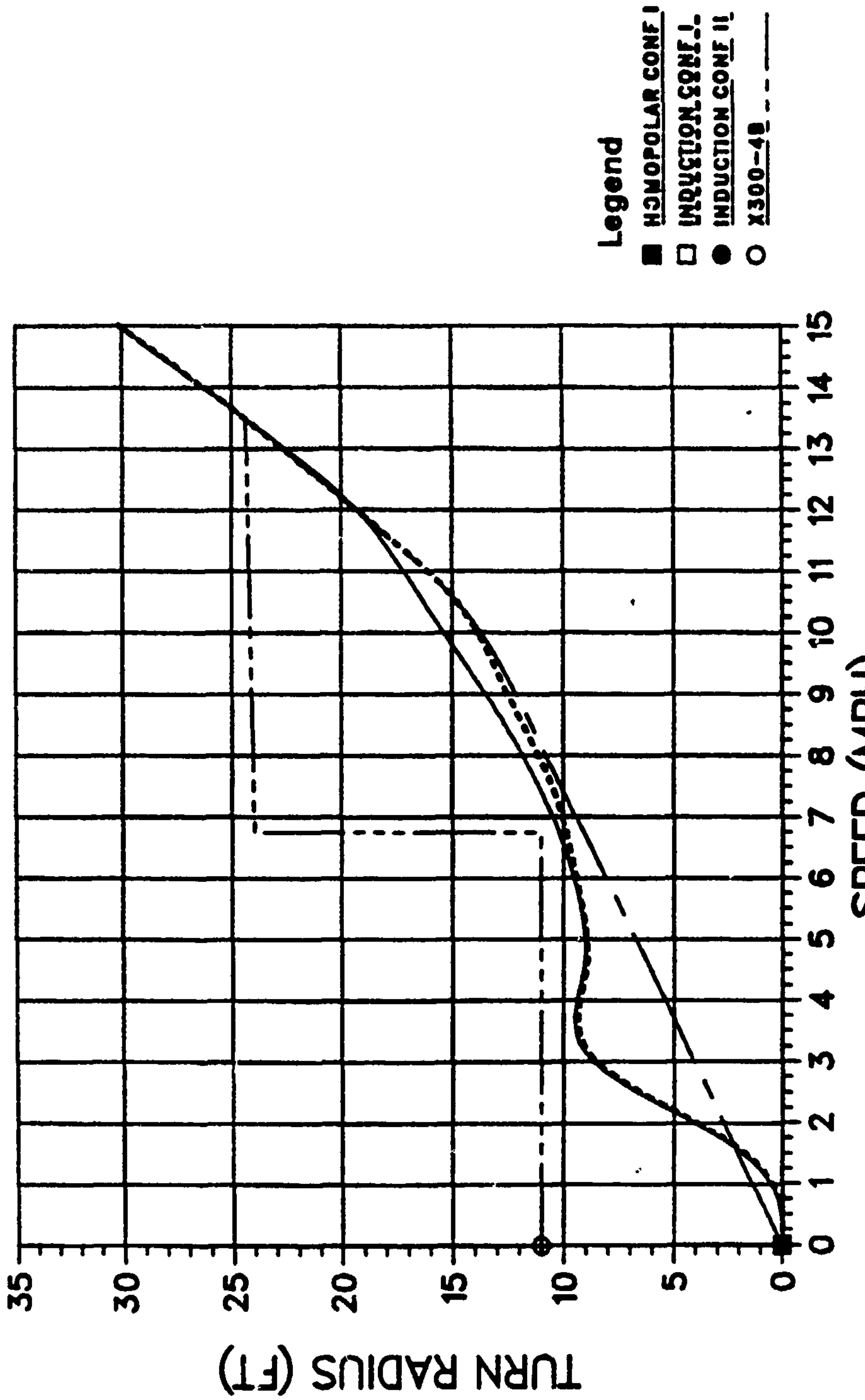


Figure 5-80. Minimum Turn Radius Versus Speed (<15 mph)

5.11.9. Brake Performance.

5.11.9.1. Electro-dynamic braking capability of the electric drive system has been combined with disk brakes to fully meet specifications with minimum space, weight, and cost. Brake studies and optimization were made within the constraints of the specified brake performance characteristics. These requirements were:

- Peak deceleration rate of 7 meters per second per second on level hard surface roadway.
- Average deceleration rate of 5 meters per second per second from top speed (45 mph) to fully stopped on level, hard surface roadway.
- Maintain a speed of 20 mph while continuously descending a 15 percent downgrade.
- Pass a 25 cycle durability test consisting of 5 meter per second per second stops from 60 kilometers per hour (37.7 mph) at three minute intervals.
- Hold the vehicle on a 60 percent grade with the engine off.

5.11.9.2. It was found that maximum utilization of electro-dynamic braking capabilities achieved these goals with minimum space, weight and cost. Existing components (motors, generators, and controls) form the core of the electro-dynamic braking system, with minor revisions to the low power controls and a resistance grid completing the installation. The recommended drive systems are capable of providing all required braking capabilities down to a speed of approximately 5 mph.

5.11.9.3. Below 5 mph, supplemental braking is provided by use of disk brakes. Since energy levels are low at such slow speeds, the size and weight of the disk mechanical brake systems are minimal. These mechanical brakes are sized to serve as an emergency brake in case of a failure of the electro-dynamic sytem. For most concepts the dry disk brake is recommended, however, in certain arrangements, access to the motor shaft for mounting the disk is limited, in which case a wet disk brake system would be built into the final drive gear case.

5.11.9.4. How the combination of dynamic braking with supplemental disk brakes fully meets the specified braking requirements is detailed in the following sections.

5.11.9.5. Brake analysis method. The analysis method compares the net brake load to brake capacity to assure that brake performance will be adequate for all operating conditions. It is based on the principle that the loads imposed on the brakes must be equal to the vehicle braking inputs, less the various losses. In the braking mode, a number of items that are normally losses absorb power and thus minimuze loads imposed on the brake system. Rolling resistance, wind resistance and final drive losses are typical of these factors. Braking load inputs to the vehicle system are energy from deceleration and downgrades. Table 5-10. summarizes the equations used in this analysis.

5.11.9.6. The calculated braking horsepower was checked against the combined braking capacity of the electro-dynamic and disk brake systems to be sure that all requirements could be fully met. All requirements above 5 mph were met by the electro-dynamic system. At lower speeds, use of the mechanical disk brake was required to achieve the desired performance.

5.11.9.7. Electro-dynamic brake capability. The main components for the electro-dynamic brake system consist of existing motors, generator, controls, fan, and cables. The added space and weight for braking purposes are therefore minimal. Added items would be limited to control additions and a resistance grid that would dissipate the energy to the engine coolant or directly to the air.

5.11.9.8. Multiple disk wet brakes can be sized to fully meet all brake requirements, but the resulting size, weight and power loss is undesirable. This type of brake is the standard braking system used on most tracked military vehicles. They combine generous surface area for good wear life with positive oil cooling to absorb high levels of energy without destructive temperature rise. Their ability to meet all normal braking requirements has been demonstrated by their many successful applications.

The disadvantages to such systems are in the areas of size, weight, and power loss. The compressive pressure that can be sustained by the brake linings is about 200 - 250 lbs per square inch to maintain satisfactory wear life. The friction coefficients running in the oil coolant are limited to about 0.07 to 0.12 for dynamic loads, and 0.12 to 0.15 for static loads. The need to hold on a 60 percent grade necessitates very high maximum torque capacity. To achieve this high torque with low surface pressures and friction coefficient requires many disks of large diameter.

The brakes also suffer from power loss due to viscous drag when they are released. Figure 5-81. shows the power loss for a system sized for the 19.5 ton vehicle. The data came from FMC tests that were conducted to optimize a brake system for minimum drag, therefore little additional drag reduction appears likely. The performance reduction is relatively minor except near top speed, where the gradability is reduced from 1.75 percent to 1.25 percent.

Wet disk brakes can provide a satisfactory system, but their drawbacks make them the system of choice only when effective dynamic braking can not be provided, or when the mechanical arrangement does not permit installation of other brake systems.

Table 5-10. Brake Analysis Equations

- Grade Horsepower:

  HPG = GVW * SIN(ATN (GRADE/100)) * MPH/375

- Acceleration Horsepower:

  HPA = GVW * ACC * MPH/375

- Rolling Resistance Horsepower:

  GVW/2000 *RR * MPH/375

- Wind Resistance Horsepower

  HPW = (1/391 * MPH * MPH * AF * CD) * MPH/375

- Sprocket Horsepower:

  HPS = HPG + HPA + HPR + HPW

- Braking Horsepower:

  HPB = EFG * HPS

Where:

HPG = Grade Horsepower
GVW = Gross Vehicle Weight (lbs)
GRADE = Grade (percent)
MPH = Miles Per Hour
HPA = Acceleration Horsepower
ACC = Acceleration (g's)
HPR = Rolling Resistance Horsepower
RR = Rolling Resistance (lbs per ton)
HPW = Windage Horsepower
AF = Frontal Area (square feet)
CD = Drag Coefficient for Vehicle
HPS = Sprocket Horsepower
HPB = Horsepower to be Absorbed by Brake System
EFG = Efficiency of Gear Train from Sprocket to Motor

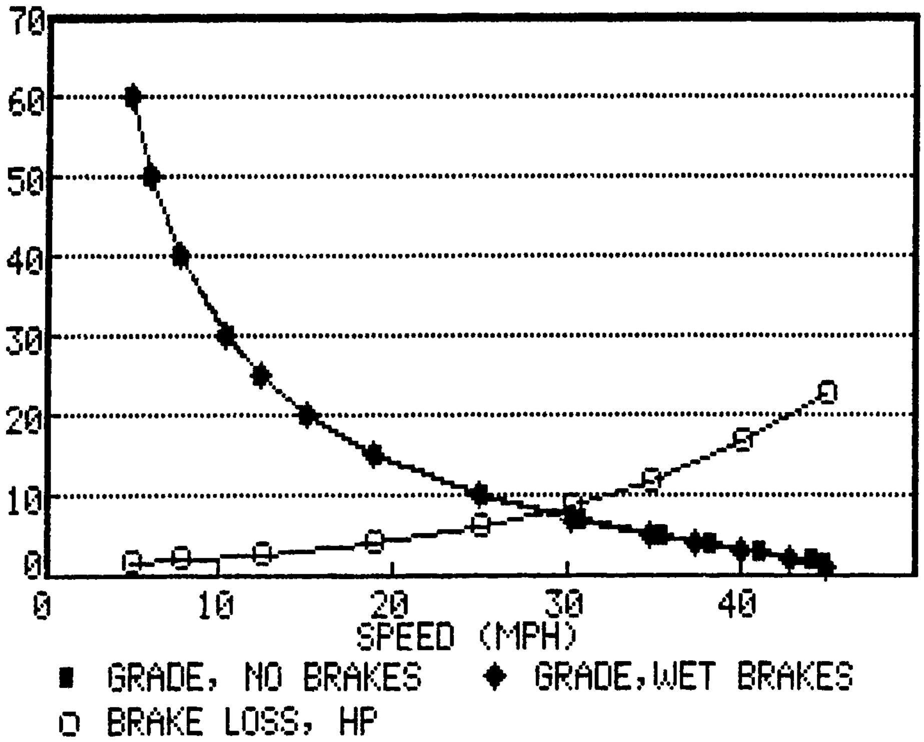


Figure 5-81. Wet Disk Brake Losses

5.11.9.9. Dry disk brake systems are small, light, and have minimum power loss, and they have adequate capacity to supplement dynamic brakes at low speeds, can provide 60 percent grade holding capability and can serve as an independent emergency brake.

For equal torque capacity, dry disk brakes are much smaller and lighter than wer ones because of the higher friction coefficient of dry surfaces, and because there is no coolant circulation system. The thermal capacity of the dry disk brake is more limited than the wet disk. Reduced size and weight limits the effective heat sink and results in a much more rapid temperature rise.

The dry disk brake appears ideal for use with the effective electrodynamic braking capabilities of the recommended electric drive systems. Where the mechanical arrangement of the vehicle permits, the dry disk brakes have been used in the vehicle concepts. In combination with the electyro-dynamic brakes they provide a braking system that fully meets the braking specification with minimum space and weight.

The electro-dynamic brake functions will be controlled by the same microprocessor that controls the electric drive. The disk brakes will normally be applied by a hydraulic servo unit controlled by the same microprocessor , but a direct mechanical override will be provided to the disk brake for redundancy in emergency operation.

5.11.9.10. Emergency stop. All configurations and weights easily meet the emergency stop requirements. The recommended electric drive systems offer a substantial overload capacity. The specified average deceleration rate of 5 meters per second per second limits deceleration time to about 4 seconds. All of the recommended electric drives can easily absorb six times their normal rating for this period of time. Since the systems are sized to compensate for any effects of the configurations, this factor does not affect the available emergency braking capacity.

5.11.9.11. Figures 5-82. and 5-83. illustrate that braking capacity exceeds the deceleration requirement at all specified operating speeds. The 7 meter per second per second reuirement is met at all speeds below 38 mph for the 19.5 ton vehicle and below 36 mph for the 40 ton vehicle.

5.11.9.12. Stop sequence test. All configurations and weights easily meet the specified stop sequence requirements. This test requires that 25 stops be made at three minute intervals from 60 kilometers per hour, with a 5 meter per second per second deceleration rate. Power input occurs during less than 2 percent of the cycle time, with the remaining time available to cool the brakes. Figure 5-84. illustrates the loads versus capacities for several cycles of this test. The high overload capacity of the electro-dynamic braking system is ideally suited to this duty cycle. The 25 cycle test is easily met and does not drive the selection of systems/components for the electric drive vehicle concepts.

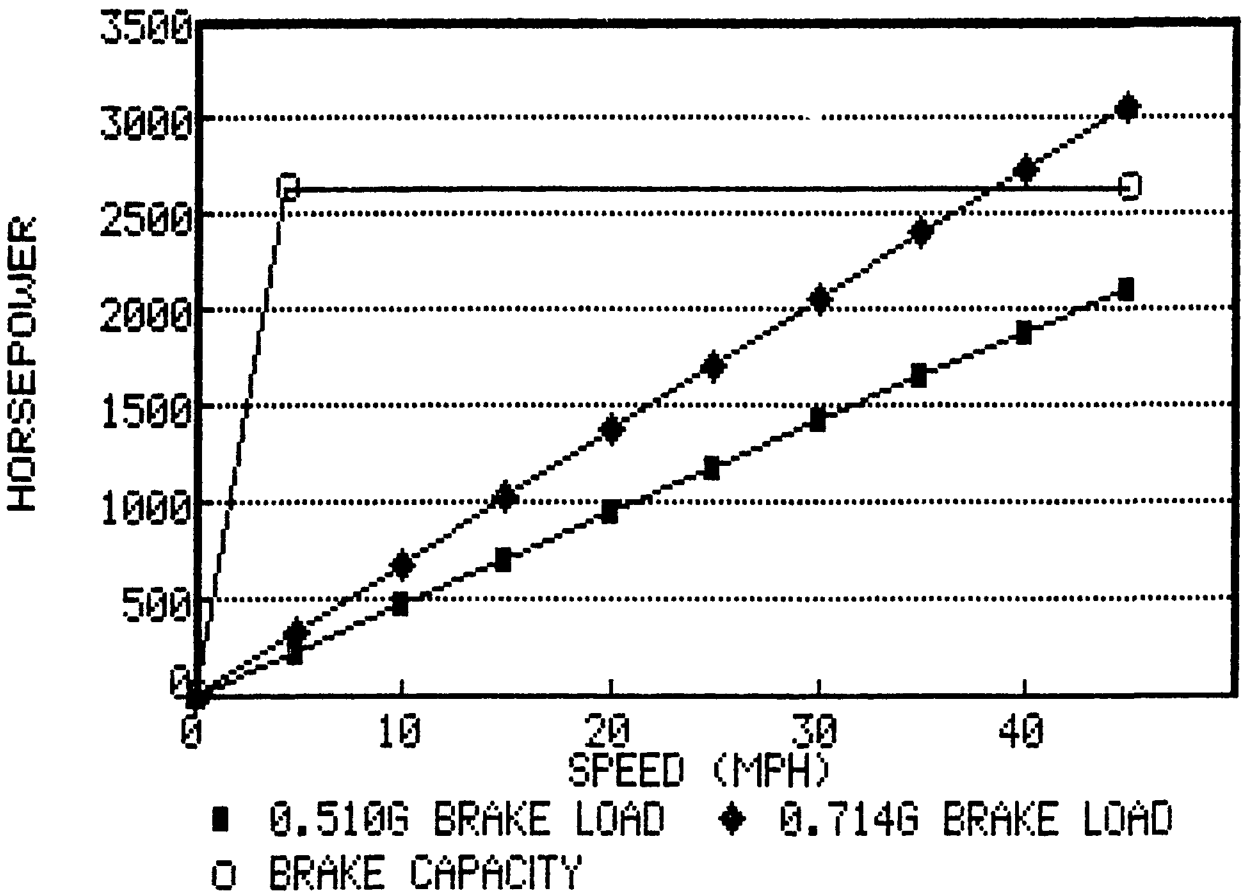


Figure 5-82. Panic Brake Load Versus Capacity (19.5 Ton Vehicle)

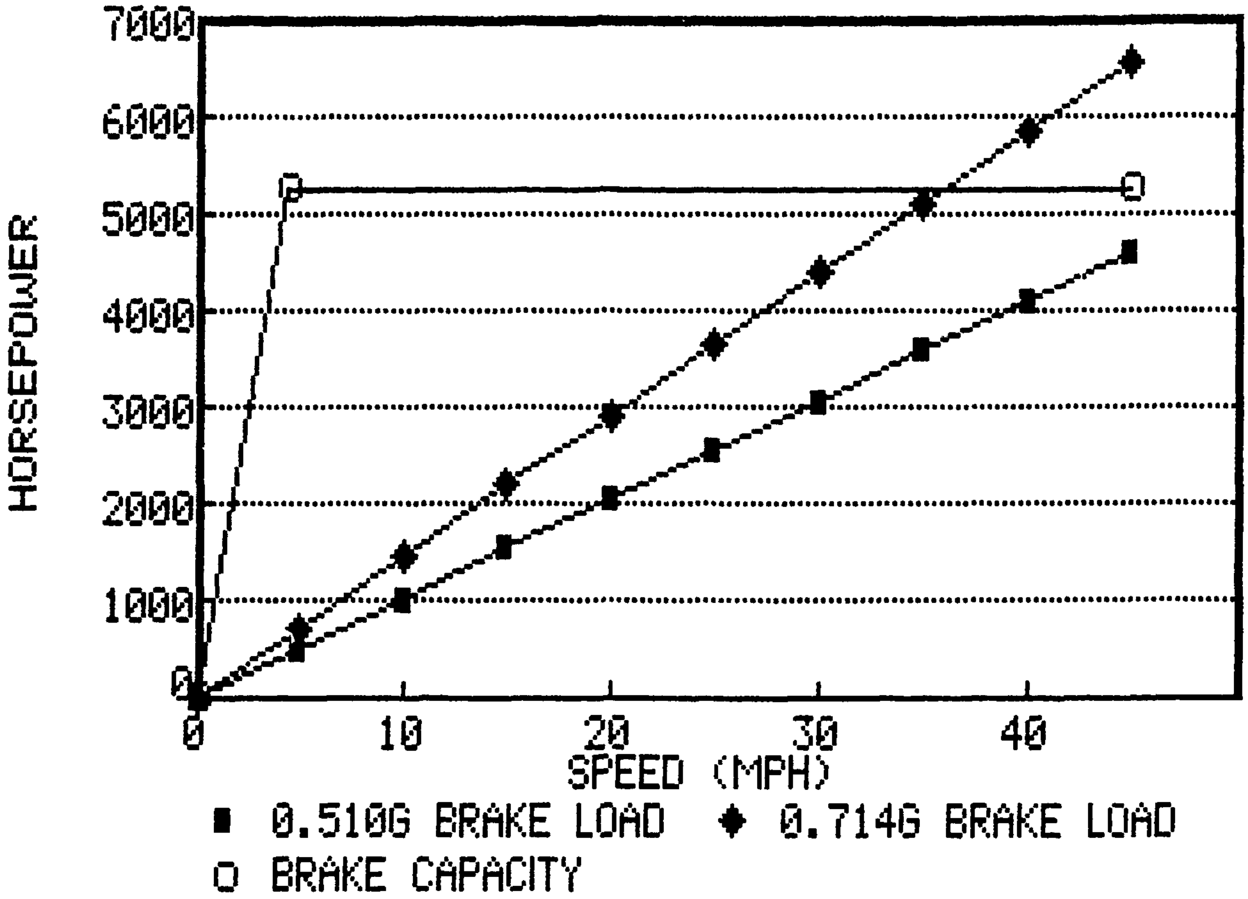


Figure 5-83. Panic Brake Load Versus Capacity (40 Ton Vehicle)

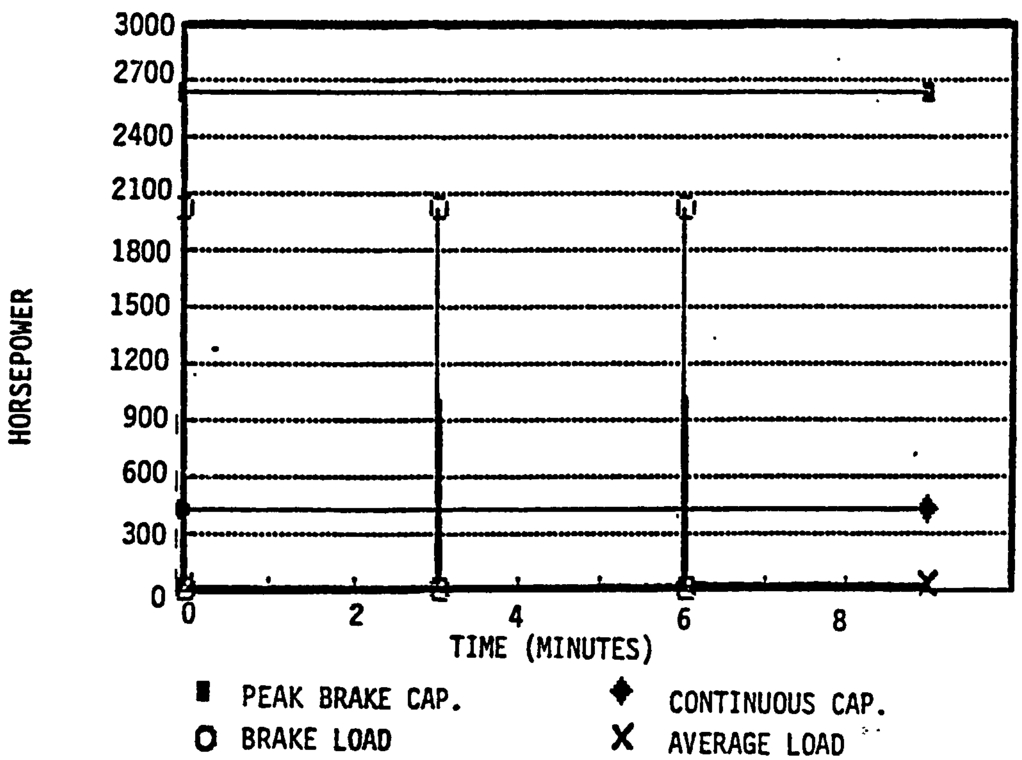


Figure 5-84. Horsepower Versus Time (25 Cycle Braking)

5.11.9.13. Continuous downhill braking. Electro-dynamic braking is adequate for a continuous downgrade of 15 percent over the full vehicle speed range; therefore the margin of safety is generous at the specified 20 mph. The size of the electric drive components has bveen determined to meet the vehicle performance specifications. For dynamic braking purposes the components of the recommended drive systems can absorb the same power as they can deliver for propulsion. The additional components, primarily the braking grid, were matched to the same power capacity to provide a balanced system.

5.11.9.14. Brake loads of the 19.5 ton vehicle were calculated for speeds up to 45 mph, and are plotted in figure 5-85. The capacity of the electro-dynamic braking system was plotted on the same chart, showing the reserve braking capacity over the range of speeds. The requirement to maintain 20 mph on a 15 percent downgrade is easily met with components that are sized to meet other requirements. Therefore, this requirement does not impact the vehicle concepts or their drive components.

5.11.9.15. Hold on 60 percent grade. Disk brakes provide ample torque to hold on a 60 percent grade. The torque required to hold at the sprockets is 15,941 lb-ft for the 19.5 ton vehicle, while the brake capacity is 24,000 lb-ft. The brake capacity for the 40 ton vehicle is approximately doubled to retain the same percent reserve capacity.

5.12. Parametric Study.

A parametric study of the technologies involved in developing electric drive systems for combat vehicles was added to the contract by Modification P00006. The primary goal of the study was to determine what advances, if any, were required to bring electric drives to maturity, and were projected for the near term. The characteristic technologies of the motors, alternators and controllers were targeted for examination. Specifically addressed were:

- Electrical
  - Brushes
  - Insulation
  - Semiconductors
  - Magnetic materials
  - Controls
  - Commutation

- Mechanical: Motor structures

- Thermal
  - Cooling techniques
  - Insulation

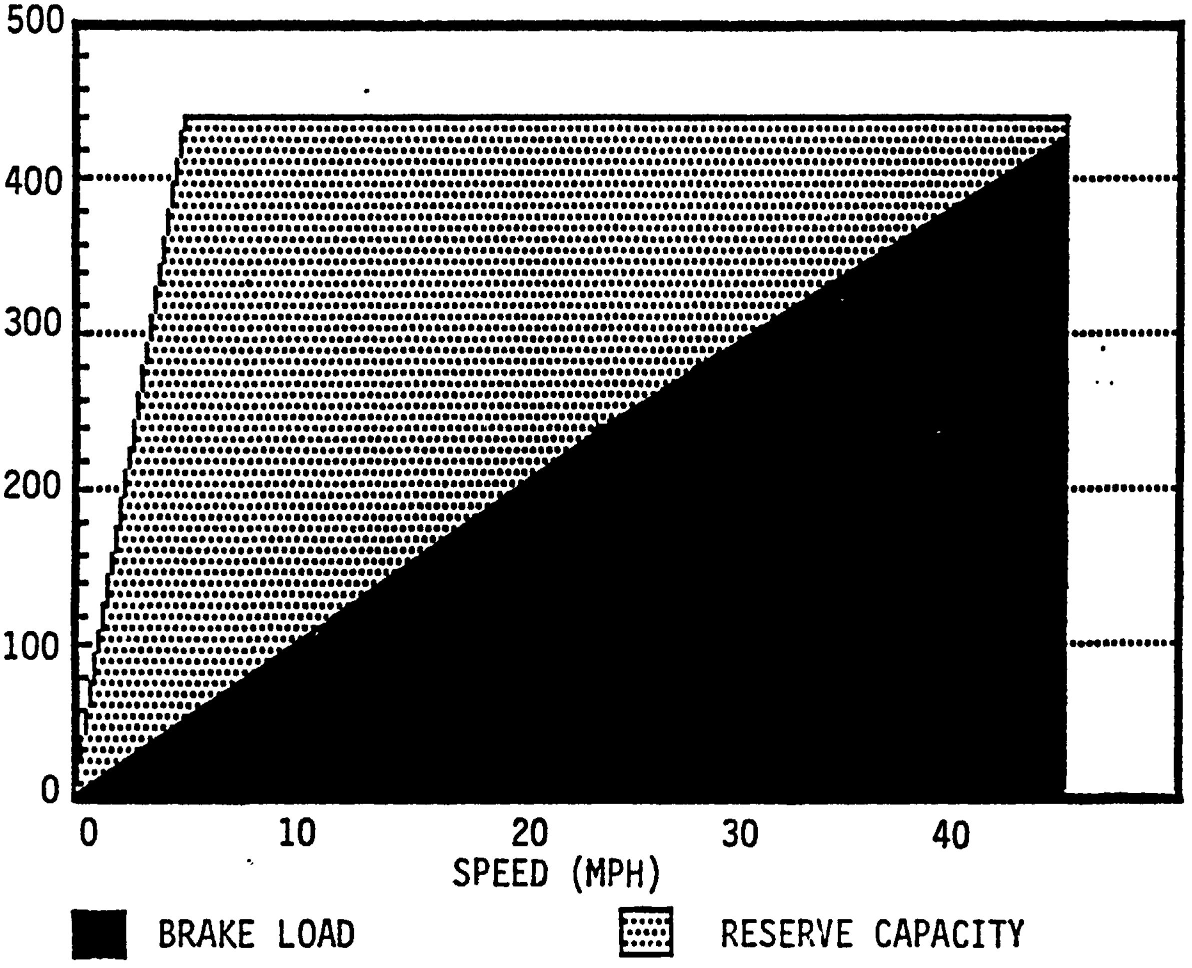


Figure 5-85. Brake Load & Capacity (15 percent Downgrade)

5.12.1. During the initial phases of the contract, a technology survey was conducted to examine and collect data relevant to electric drive systems. The technology survey results were used to establish the baseline for the parametric study.

5.12.2. The first activity accomplished for the parametric study was a parameter screening. Each of the above technologies was evaluated in two respects. First, determine whether there were any significant anticipated changes projected for the future; and second, given that an advance occurred in a particular area, would it have any significant impact on electric drive vehicles. The developed assessments were confirmed by surveying various members of the research community and the industry community who were contacted under the original technology survey. Results of the parameter screening are shown in Table 5-11.

5.12.3. All of the technologies were deemed to have a potential impact on the future of electric drive systems. Examination of projected advances in technology indicated that substantial changes were not likely in the areas of motor structures and insulation. The technology parameters in Table 5-11 are listed in descending order of significance to electric drive vehicles. This significance is a combined evaluation of the two factors: probability of technology advance, and potential impact of the advance. Based on these criteria, motor structures and electrical insulation were eliminated from further consideration.

5.12.4. After the completion of the parameter screening, past and present maturation trends were updated for the selected technologies. The technology survey, which served as the baseline, was completed in 1984 and did not analyze technology trends in depth. Current research was factored into the projections for future developments as part of the update. Graphical depicitons of the trends for each of the selected technologies were assembled. The trend lines represent past levels achieved, current levels, and future levels projected. Each of the technology assessments are in units suitable for representing its capabilities.

5.12.5. The rate of change projected for the technologies may best be described as evolutionary in nature. The changes are expected to improve the competitive position of electric drives compared to mechanical systems. Breakthroughs may be possible in some technology areas, particularly in power transistors and brushes. Breakthroughs would further improve electric drive advantages. but are not necessary in order to meet or exceed performance of existing mechanical systems.

5.12.6. Figures 5-86., 5-87., and 5-88. represent the impact of various cooling techniques on motor size, controller size and brush capacity respectively.

Table 5-11. Parameter Screening Results

	PARAMETER SCREENING SUMMARY		
PARAMETERS	POSSIBILITY OF SIGNIFICANT CHANGE TO TECHNOLOGY	POSSIBILITY OF IMPACT ON ELECTRIC DRIVE CONCEPTS	EXAMINE IN THE PARAMETRIC STUDY
1) CONTROLS	YES	YES	YES
2) SEMICONDUCTORS	YES	YES	YES
3) COMMUTATION	YES	YES	YES
4) COOLING TECHNIQUES	YES	YES	YES
5) BRUSHES	YES	YES	YES
6) MAGNETIC MATERIALS	YES	YES	YES
7) MOTOR STRUCTURES	NO	YES	NO
8) INSULATION	NO	YES	NO

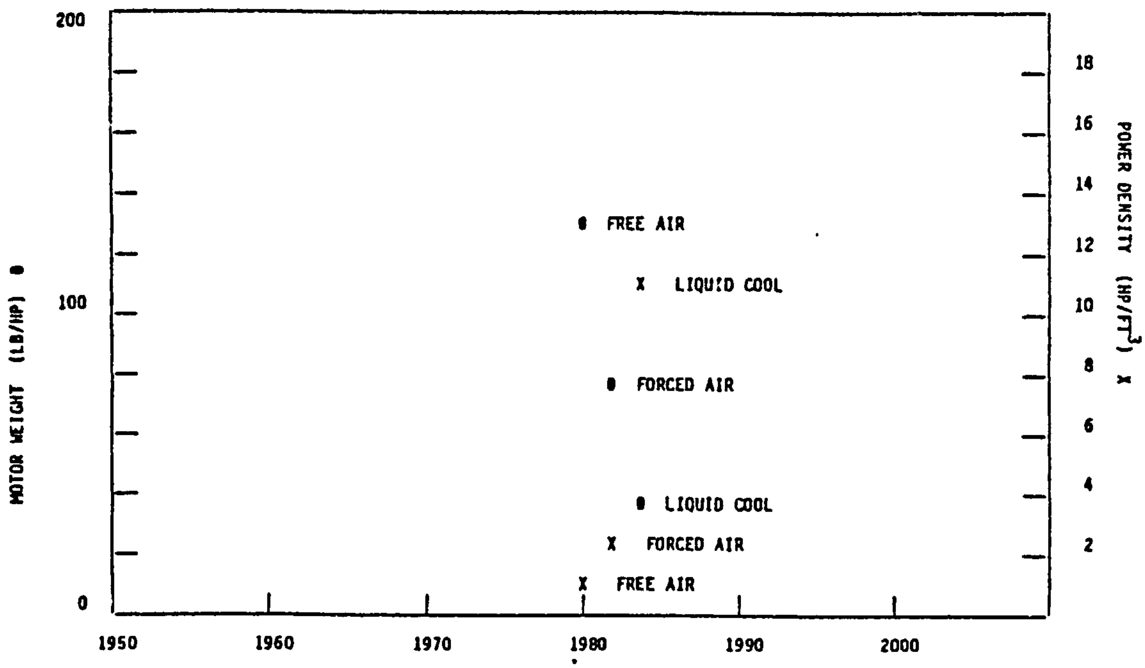


Figure 5-86. Motor Size Comparison Data

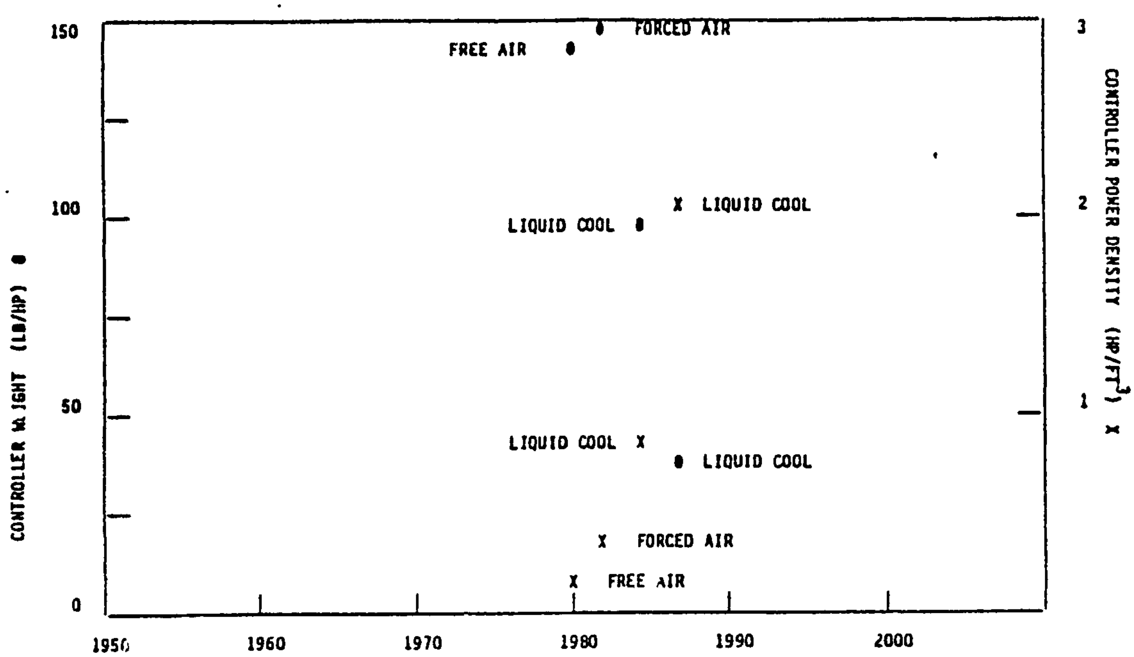


Figure 5-87. Controller Size Comparison Data

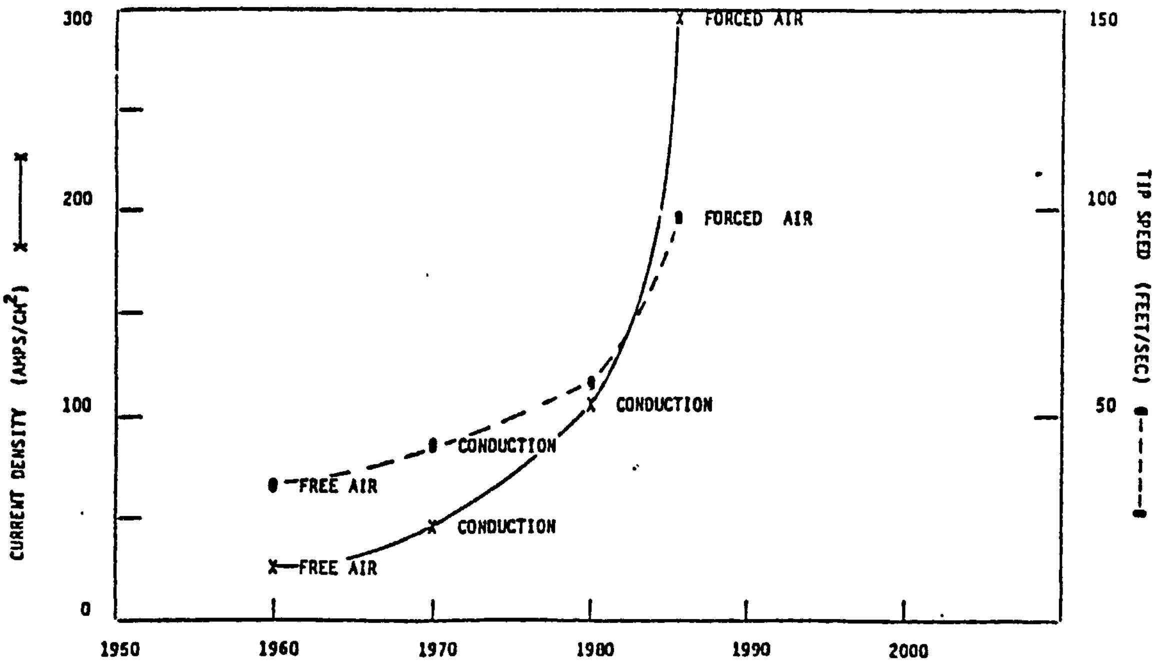


Figure 5-88. Brush Capacity Comparison Data (Graphite)

5.12.6.1. For motors and controllers, the advent of liquid cooling techniques have greatly reduced the weight of the systems while increasing the power density. Further advances are expected in the areas of high thermal limit liquids and heat pipes, which will have a favorable impact on both power density and weight.

5.12.6.2. The impact of cooling techniques on size and weight of motors and controllers is so pronounced, that trends based solely on time (years) were masked. For this reason we chose to present Figures 5-86. and 5-87. only for current technology, to clearly show the impact of the cooling approach.

5.12.7. Controllers will continue to be reduced in size by the projected increases in the power handling capabilities of semiconductor devices. Better methods of cooling and heat rejection will be required to support the smaller, more compact controllers of the future. The ability to sufficiently cool the high power devices is currently a limiting factor to both their performance capability and reliability.

5.12.8. Figure 5-88. (previous page) represents the capacity of graphite type brushes in terms of current density and tip speed, as affected by various cooling techniques. Advances in brush technology are primarily material related with the majority of the research and development funding being provided by the Air Force. Forced air techniques have yielded significant gains in current density and tip speed for graphite type brushes.

5.12.9. Figure 5-89. represents the trendline growth of the commutation capacity in volts/micro-second. Between 1965 and 1975, the average commutation capacity grew by a factor of four due to increased research in the technology. Current projections indicate steady growth at a rate consistent with past trends.

5.12.10. The magnetic materials performance data, Figure 5-90., shows a steady growth to current levels of performance. As magnetic materials improve, permanent magnet motors may beome competitive with induction motors. Further reductions in the hysteresis losses in the iron structure would improve the overall performance of high frequency AC motors. In addition, iron materials with higher permeability levels would improve the performance of DC motors.

5.12.11. The parametric study confirmed the general conclusions drawn in the earlier proram phases, and resulted in two new conclusions:

- Breakthroughs are possible in some of the critical technology areas (for example, power transistors and brushes), but are not necessary to make electric drive systems competitive with mechanical transmissions.

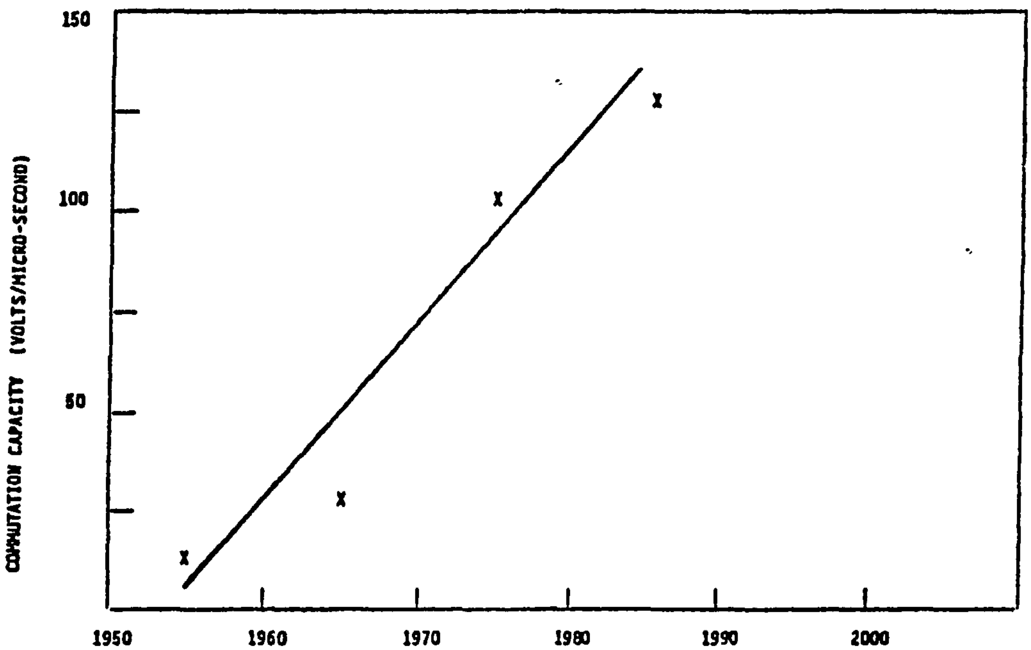


Figure 5-89. Commutation Capacity Comparison Data

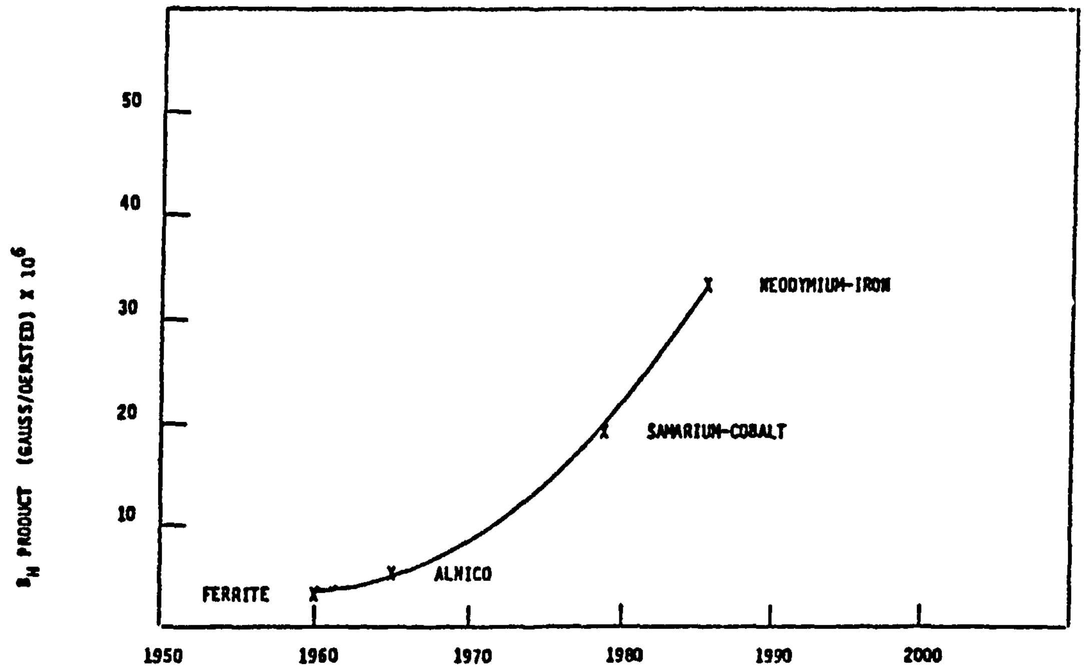


Figure 5-90. Magnetic Materials Performance

- Projected evolutionary improvements in all electric drive technology areas would continue to enhance the feasibility of the systems. It also indicates that a technical revolution is likely in the area of vehicle transmissions and drives. Electric drive systems may dominate the field within 20 years.