AD 420740

Tracked Amphibian Personnel and Cargo Carrier

Feasibility Study
under
Contract NObs-4463
for
Bureau of Ships

Proposed as
Marine Assault Carrier
(MAC)

Volume II of II

15 November 1961

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5.0 SUSPENSION SYSTEM

The new MAC suspension system incorporates a flat track, over-the-wheels track return, a system which made possible the use of a particularly compact road wheel and suspension arm assembly. See Figure 6-1.

The new track and suspension concept achieves greater simplicity and reliability, together with a good potential for reduction in maintenance requirements. In addition, the new system utilizes light weight materials, and is exceptional for its quietness, low rolling resistance, high tractive characteristics, efficient water propulsion, and resistance to track throwing.

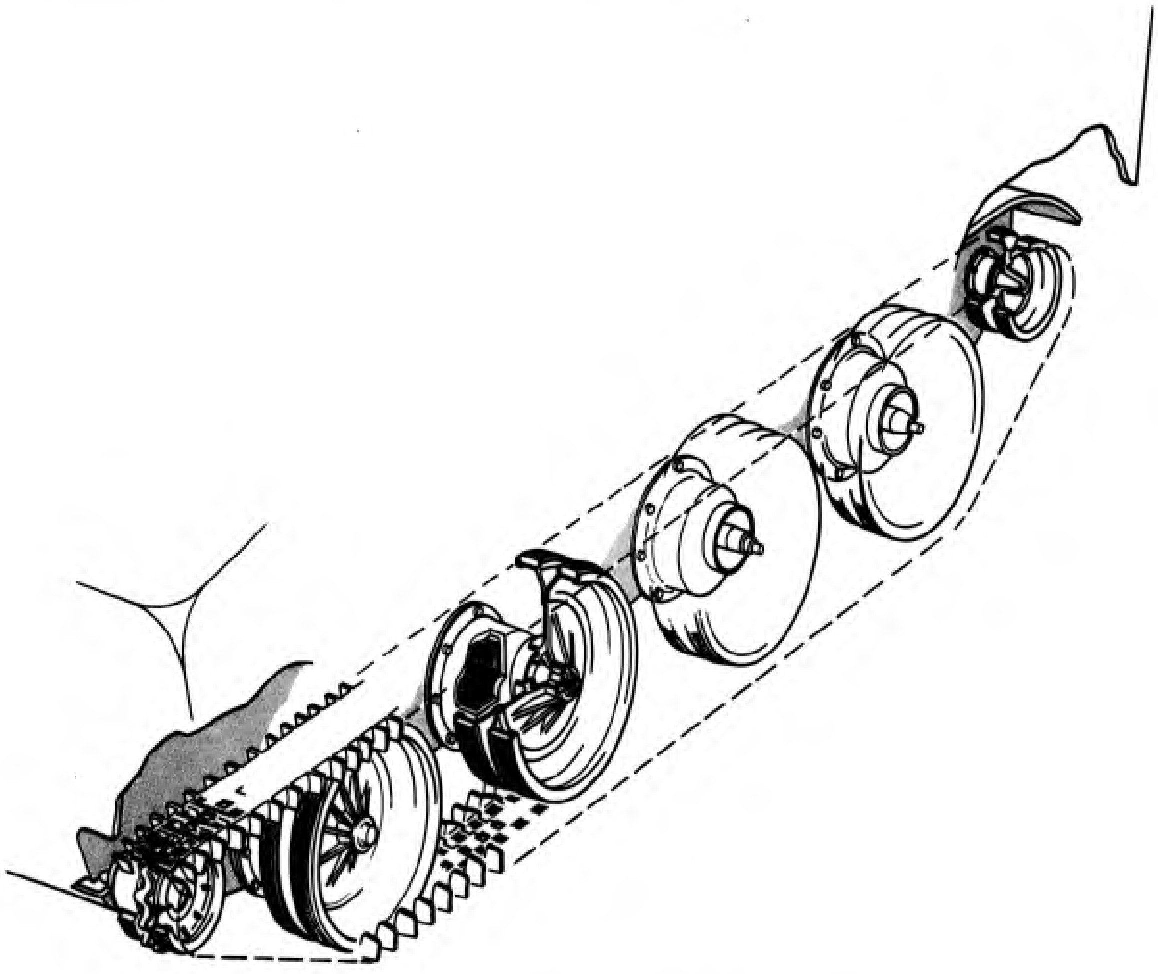


Figure 5-1. The New MAC Suspension System

Primary in its consideration for this particular vehicle application is its compact, self-contained design. Elimination of commonly used torsion components, such as cross-vehicle steel torsion bars and components which add weight and structural complications was considered an advantage, particularly in view of the limited angular deflection those designs could achieve within reasonable torsion bar lengths.

The design principles embodied in the new, compact, simplified suspension concept, and the design philosophy followed in selection of an exactly matched set of track system components intended to exploit its inherent advantages to the utmost, is covered in the remainder of this section.

5.1 Track Return

5.1.1 Underwater Track Return Over Roadwheels.

This system provides for the shortest length of track and fewest components, due to the dual purpose of the road wheels, resulting in a lower net vehicle weight. A lower vehicle silhouette is also possible since less space is required for the suspension system. This track return was selected for adaptation to the MAC vehicle for light weight and simplicity of design, reduced maintenance, reduced rolling resistance, and increased tire life due to the larger diameter road wheels. Greater vehicle mobility for a given track contact area results due to reduced concentrated loads under the large road wheel. Ground pressure is 4.1 psi for net vehicle weight.

5.1.2 Drive System.

The front drive sprocket does operate under dirt-free conditions, which is desirable. However, the total return length of chain from the rear wheel to the front sprocket is under constant tension. Further, the bearing loads on both the sprocket and return idler must operate under double loads during all forward operation.

All of these conditions demand heavier duty components and result in increased wear. In contrast, the rear sprocket drive has only that short section of track from the last road wheel to the sprocket under tension, and the sprocket bearing loads are reduced by one-half, with the front return idler under very light loads during normal forward operation. These factors influenced the choice of a rear drive system for the vehicle.

5.2 Road Wheel and Suspension Arm.

Design of the road wheel and suspension arm is handled as a unit since the assembly must be compatible with the type of suspension system and track selected. Therefore the road wheel must be of sufficient diameter to perform as the track return, incorporate wear surfaces where it guides with the track, and have good guiding characteristics to prevent or reduce track throwing tendencies. The suspension arm, to be compatible with the selected suspension system, is cantilever mounted to the hull due to the flat track return.

5.2.1 Road Wheel.

Investigation of plastic material for application to road wheel fabrication has led to the conclusion that such application is still in the experimental and developmental field. Its use in this vehicle has therefore been bypassed at this time even though further investigation is continuing. Construction of the road wheel (see Figure 5-2) is of aluminum with steel incorporated for the track guiding wear surface. This will provide the desired weight savings without sacrificing life.

The tire design is based on recently developed test tires evaluated on the LVTP5 vehicle by both the USMC Test and Experimental Unit, Camp Pendleton, California, and the Contractor. These tests proved the fully trapped tire design to be greatly superior to untrapped and modified trapped tires, the greatest contributing factor being the greater length of track guiding surface. Experience has proved that all track guiding surfaces must be steel. Therefore, a wear ring is attached to the flange to provide the desired wheel lift. However, the wear ring is not intended as a replaceable item since experience has proved re-tiring of rims to be impractical. Vehicle operation will result in wear of the circumferential surfaces of the aluminum flanges as well as the steel wear ring. However, the steel ring serves to control the rate of wear. Molding of tires to a rim requires a closely controlled machined surface to close the rubber mold to the part, thereby providing the means to mold the tire to the rim under pressure.

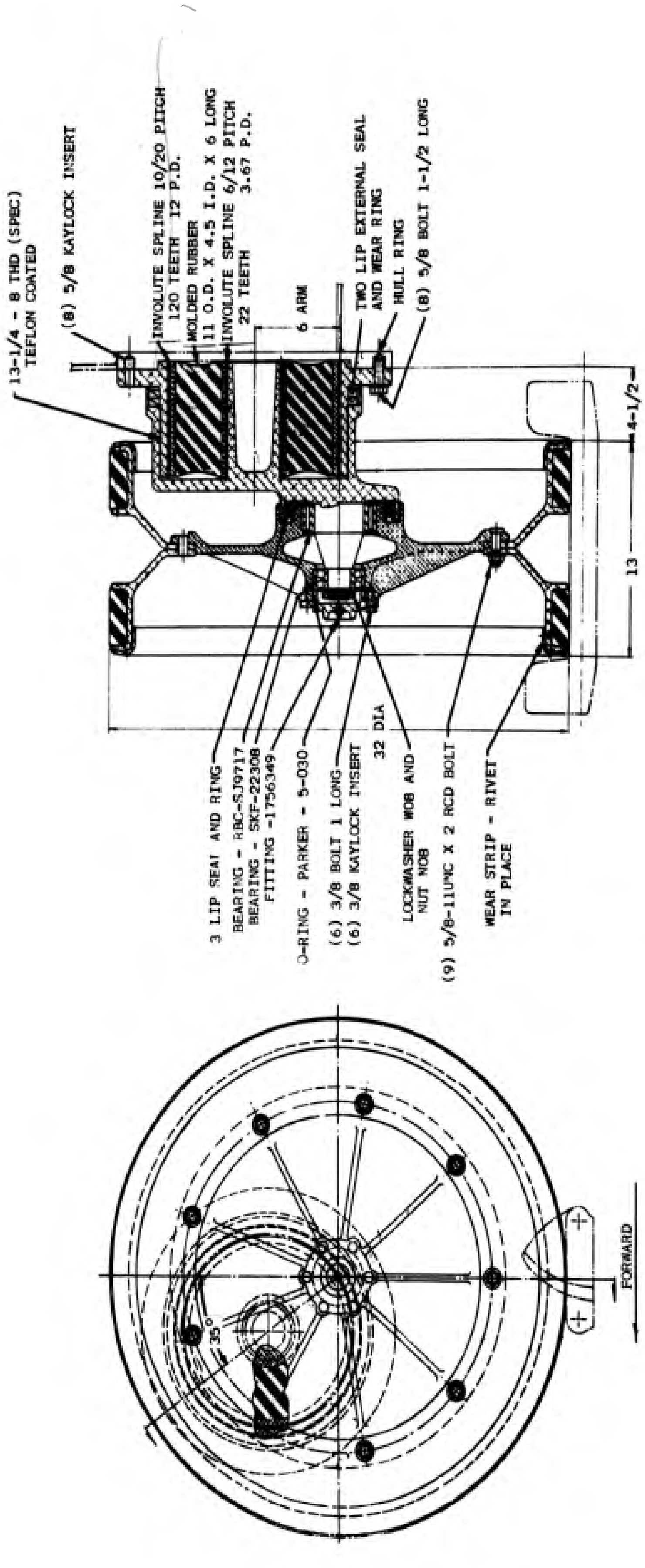


Figure 5-2. Suspension Arm and Road Wheel Assembly

The quality of bond, tire to rim, is a function of this pressure. Retiring of the rims requires considerable machining and a set of variously sized molds compatible with the extent of wear and degree of machining required to provide a true surface.

The tire and rim as a unit has been designed for easy replacement. Note that tires can be removed from the hub without exposing the bearings to contamination.

Recent Army Ordnance data points to success with oil-lubricated wheel bearings protected with face seals. However, with amphibious-type vehicles, experience has proven most successful the bearing arrangement shown in Figure 5-2. The bearings are grease lubricated and protected with triple lip seals. Reference: Test results of the full width rubber tired road wheels produced and tested under Contract NObs 3855-57-13 for the Department of Navy, Bureau of Ships. Since there are no wearing surfaces, the hub is machined from an aluminum casting to provide the greatest weight savings.

5.2.2 Suspension Arm.

The ability and speed of a vehicle to traverse cross country terrain is characterized by the efficiency of the suspension arm. The suspension arm is the means by which the hull, crew, and equipment are insulated from the irregularities of the ground. Therefore considerable effort has been expended toward designing an optimum suspension arm for the MAC.

The "over the road wheel" track return system with its necessarily large diameter road wheels predicates a cantilevered mounting of the road wheel to the hull. The suspension arm shown in Figure 5-3 has a six inch swing which provides ten inches of vertical road wheel travel. The arm is free to travel to the extreme vertical position with no interference. When in this position the arm is in tension and therefore provides its own stop. The short swing arm length provides extremely strong light weight construction. Reversal of the arm due to backing up has no effect on the arm as it again provides its own stop.

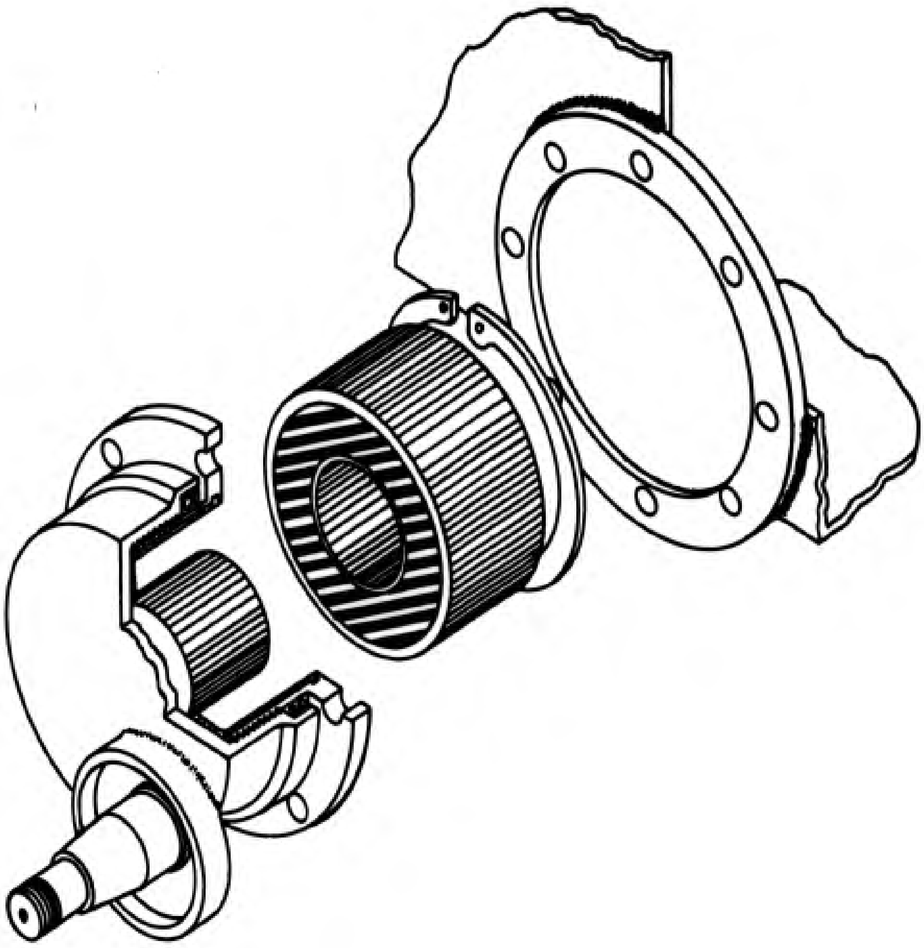


Figure 5-3. Torsion Suspension Arm

The arm is mounted to the hull by means of a threaded adapter. The threads will take all thrust and vertically imposed loads while permitting free rotation of the arm. The threaded adapter is protected from corrosion and dirt by seals. The threaded adapter is mounted to a hull mounting ring by means of cap screws. The hull mounting ring does not communicate to the hull interior, thus insuring watertight integrity.

Steel torsion bars and other types of steel springs were considered for this application. However, the high angular deflection required by the MAC suspension arm results in impractical torsion bar lengths. Therefore, a rubber torsion spring was considered more suitable for use. Further, a rubber torsion spring is self-snubbing, and requires no shock absorber as is usually incorporated with other types of steel springs.

Figure 5-3 shows a rubber torsilastic spring that provides ease of maintenance, compactness, self-snubbing, and light weight. The spring consists of two cylinders joined as a unit by bonded rubber. The outer cylinder is splined to the male threaded adapter and the inner cylinder is splined to the spring arm. Only torsion loads are imposed on the torsilastic spring due to the function of the threaded bushing. The spring is a floating member for easy assembly and maintenance. The male threads of the threaded adapter are permanently lubricated by means of a sprayed and baked coating of Teflon.

The compact suspension arm assembly allows it to be located within the periphery of the track to lessen its vulnerability to damage.

For specialized vehicles, such as the weapons carrier, the design can be readily adapted to receive a friction-type shock absorber with a lock-out system. The clutch and lock-out system would be located inside the vehicle and engaged to the arm by means of a splined shaft extending through the hull mounting adapter. This system is intended for special purpose vehicles only, since it would otherwise unnecessarily consume cargo space, add weight, increase costs, and provide further maintenance.

5.2.2.1 Torsilastic Spring Analysis.

Unsprung Weight

3/4 Track = 82P x 24.5 x .75 x 2 = 3014

1/2 Suspension Arm = 94 x 8 ÷ 2 = 376

Wheels and Hubs = 164 x 8 = 1312

Unsprung Weight = 4702 pounds

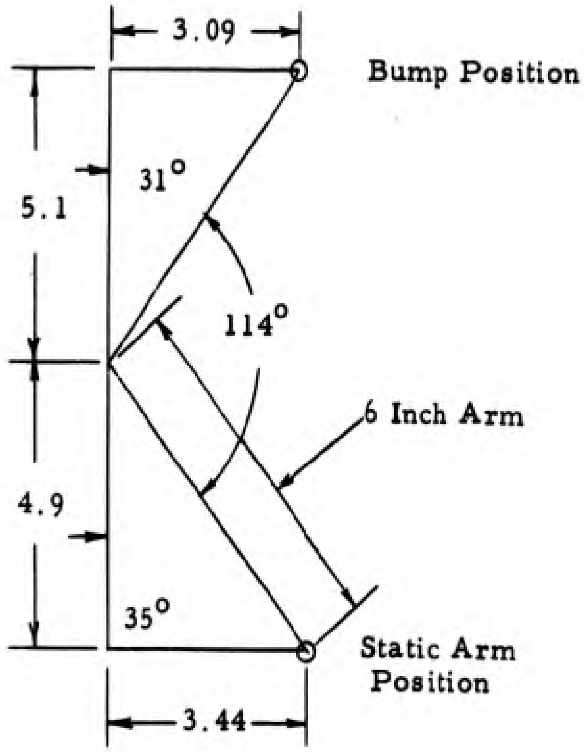
Sprung Weight = 35,000 GVW - 4702 = 30,298 pounds

Known: 6 inch arm: Static arm position = 35°; bump arm

Position = 31° from vertical (114° wind up)

Spring I.D. = 4.5 inch: Spring O.D. = 11.75 inches;

Length = 6.5 inches



Windup, static to bump = 114°
Vertical deflection = 10"

Wheel loading = 30298 ÷ 8 = 3787
Static torque = 3787 x 3.44 = 13027 lb. - in.

Static shear stress I.D. = $\frac{13027 \times 2}{4.5^2 \times 6.5}$ = 63 psi

Static shear stress O.D. = $\frac{13027 \times 2}{\times\, 11.75^2 \times 6.5}$ = 9.2 psi

From Stress Stain Curve 38-40 Durometer Compound:

S I.D. of 63 = 25° Windup

S O.D. of 9.2 = 3° Windup

Difference = 22°

Static Windup = 22° + 15% (permanent set) = 25.3°

Static Rate = 13027 ÷ 25.3 = 515 lb. - in. per degree

Dynamic Rate = 515 x 1.2 = 618 lb. - in. per degree

Bump Torque = 13027 + (114 x 618) = 83479 lb. - in.

Bump Stress I.D. = 84579 x 2 ÷ 413.5 = 400 psi

Percent of Wall = $\frac{(11.75 - 4.5)}{2}$ ÷ 4.5 = 80.5

Bump Force = 83479 ÷ 3.09 eff. arm = 27,000 lb. nearly

or 27,000 ÷ 3787 = 7.1 G's

5.3 Sprocket.

The severe wear imposed on the drive sprockets requires the use of steel. However, the hub is made of aluminum in the interest of weight reduction. The center drive sprocket is paired with support wheels to balance the track and to set the sprocket pitch (see Figure 5-4). The support wheels are of aluminum construction, rubber tired, and provided with a steel wear ring to guide the track. The hub, sprocket, and tire assembly can be removed for maintenance without exposing the bearings, or the sprocket can be serviced by first removing the outboard support wheel and tire.

Sprockets may have many teeth forms; they may be under-pitched or over-pitched, and they may even be dual-pitched. In addition, they may be arranged so that the pitch is maintained by a ring or tire rather than by the root diameter of the tooth. We have experimented with many types of sprockets and have found the following to be generally true:

1. A tooth form relieved at the trailing face will tend to prevent the track jumping a tooth because during steer, the track can move slightly ahead of the sprocket without perching on the land at the top of the teeth.

2. Over-pitching to any degree is desirable as long as the sprocket is the driving member. The added clearances necessary to an over-pitched sprocket provide good cleaning and reduced wear. However, when the sprocket is driven (during steer), climbing is excessive.

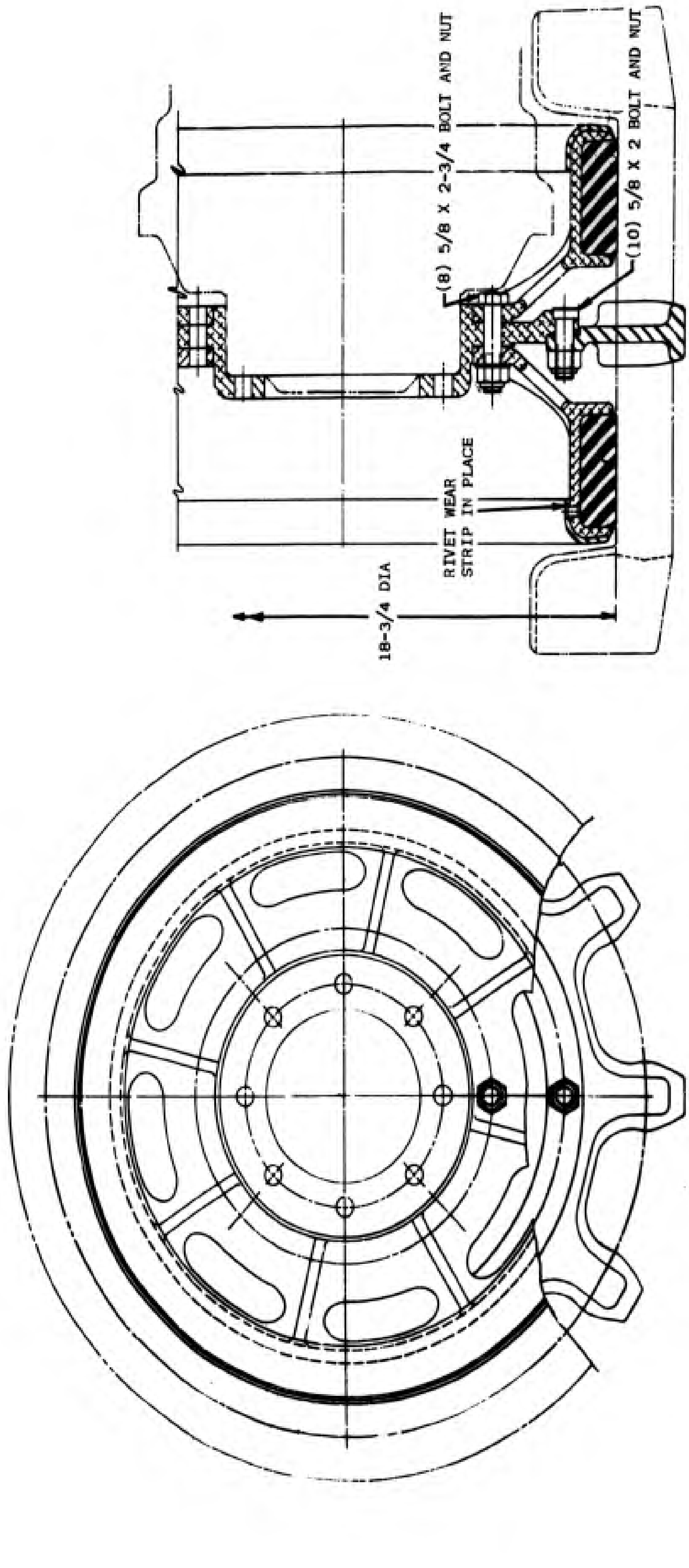


Figure 5-4. Sprocket

3. Under-pitching works well during steer, but the jamming action that occurs during all phases of operation results in rapid wear. This condition is further aggravated by track elongation.

4. Dual-pitch sprockets work well for both steering and forward operation, but the teeth are not symmetrical and demand special tooling to produce.

5. Sprockets that provide a ring or tire to maintain the pitch diameter reduce noise and vibration. They may easily be made self-cleaning and wear is greatly reduced.

The sprocket tooth form selected is relieved at the root diameter to provide a self-cleaning feature. The sprocket tooth will be slightly relieved at the trailing edge, and will be over-pitched just enough to provide for the normal elongation that usually occurs in any new track.

A rubber bushed sprocket hub designed to reduce noise and shock loads transmitted to the power train was successfully tested on the LVTP5 vehicle by the Contractor. However, since the rubber-tired support wheels adjacent to the single drive sprocket will provide some noise reduction and dampening of shock loads, incorporation of the special hub with its additional weight and the increased cost is not warranted for this application.

5.4 Idler.

Since the track compensating system has been incorporated in conjunction with the final drive, the idler (see Figure 5-5) becomes a tack-on member for simplicity. The idler tires are the same as the support tires used in conjunction with the sprocket. A ring spacer is provided to permit tire removal without exposing the bearing cavity. The hub is of aluminum construction, and the bearing arrangement is of the same principle employed on the road wheels. The hull mounting is also of aluminum construction and is blind bolted to the hull mounting ring to eliminate a source of water entering the hull.

5.5 Track Compensator.

In the operation of track laying vehicles, a basic problem has been encountered in inefficient track operation which results from incorrect track tension. Methods of controlling track tension under certain operating conditions have been developed, but no ideal compensator for controlling track tension under all operating conditions has been developed to date. The new design compensating system incorporated in the MAC Vehicle will insure optimum performance as follows:

1. Eliminate constant manual adjustment to compensate for track wear. Track would operate at optimum tension under all conditions. It would only be necessary to release the pressure in order to remove a track block when the total track elongation began to exceed the length of a pitch.

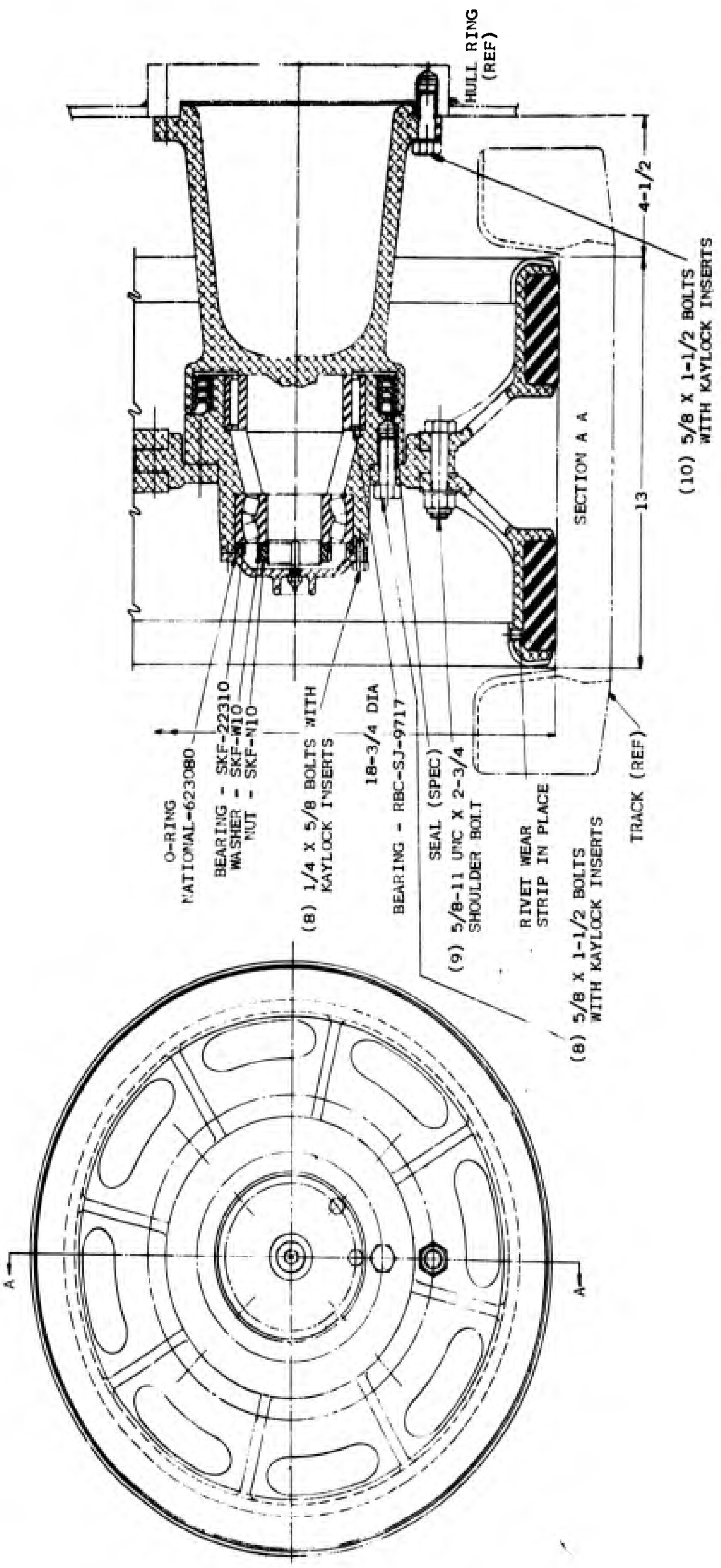


Figure 5-5. Idler Wheel

2. Provide for automatically moving the sprocket forward to eliminate slack track which ordinarily would occur when the track periphery changes because of added loads on rough terrain operation.

3. Hold the slack produced by acceleration to a minimum because of the tension applied through all positions of the sprocket.

4. Hold sprocket climb or jump to an absolute minimum during steer by preventing any slack from occurring at the sprocket.

5. Permit the track to climb the sprocket in emergency without producing dangerously high track tension. Therefore, whether the track is operating under running or steering tensions, the sprocket would move back and permit the track to climb with no increase in track tension.

The above results are provided by a dual force compensator (see Figure 5-6). During normal operation, a force of approximately 2,000 pounds tension is maintained on the track, and just prior to steer, a track tension of approximately 18,000 pounds is applied on the inside track. The outer or driven track would remain at 2,000 pounds. These forces are automatically controlled by the position of the steer control, and are applied by means of a bellcrank. In this instance, the unique final drive housing also serves as the bellcrank. A hydraulic system tied into the

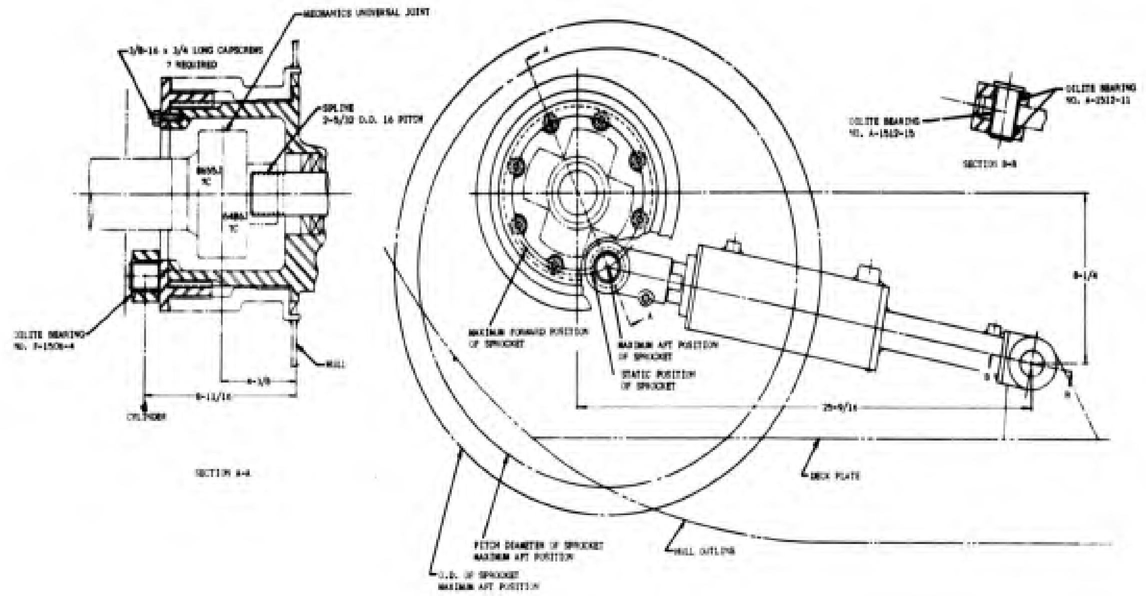


Figure 5-6. Track Compensator

steer control governs the application of "normal" or "steer" track tension forces. See Section 8 for description of the hydraulic control system.

5.6 Track.

5.6.1 Hinge Pin Tracks.

Hinged pin tracks are very stable in all planes, and when they are properly adjusted, it is virtually impossible to disengage them from the wheel guides. Horsepower losses are also low for a lubricated pin track, because it is not necessary to run them "tight" to obtain satisfactory performance. In actual vehicle operation the lubricated hinge pin track has been proven the best for efficiency. By using widely spaced pins in multiple shear, adequate strength can be provided with small diameter pins, the use of small pins in multiple shear, adequate strength can be provided with small diameter pins, the use of small pins making it easier to produce lightweight blocks. The great fault of hinged tracks has been pin wear. At the best, dry unsealed pins produce wear and galling common to clean metal in contact, but usually the bores become packed with abrasive materials and wear progresses at a prohibitive rate. Lubricated pins, oscillating in bores, or sleeve bearings of dissimilar metals have proved very satisfactory when adequate sealing is provided. Similar tracks have been tested on the LVTP5 family of vehicles and have proved to be good tracks with the exception of clevis wear that results in seal and consequent bushing failure.

A second version of the hinge pin track incorporates a threaded bushing and pin arrangement that offers merit over the lubricated pin design. The prime advantage of the threaded pin and bushing design is the absolute control of the clevis gap between mating blocks, which assures satisfactory operation of a compression type seal. This is accomplished by taking all thrust loads on the threads, thereby preventing clevis wear.

For this reason the suspension system will incorporate the threaded hinge pin track with compression seals. The design, (Figure 5-7) incorporates the following features:

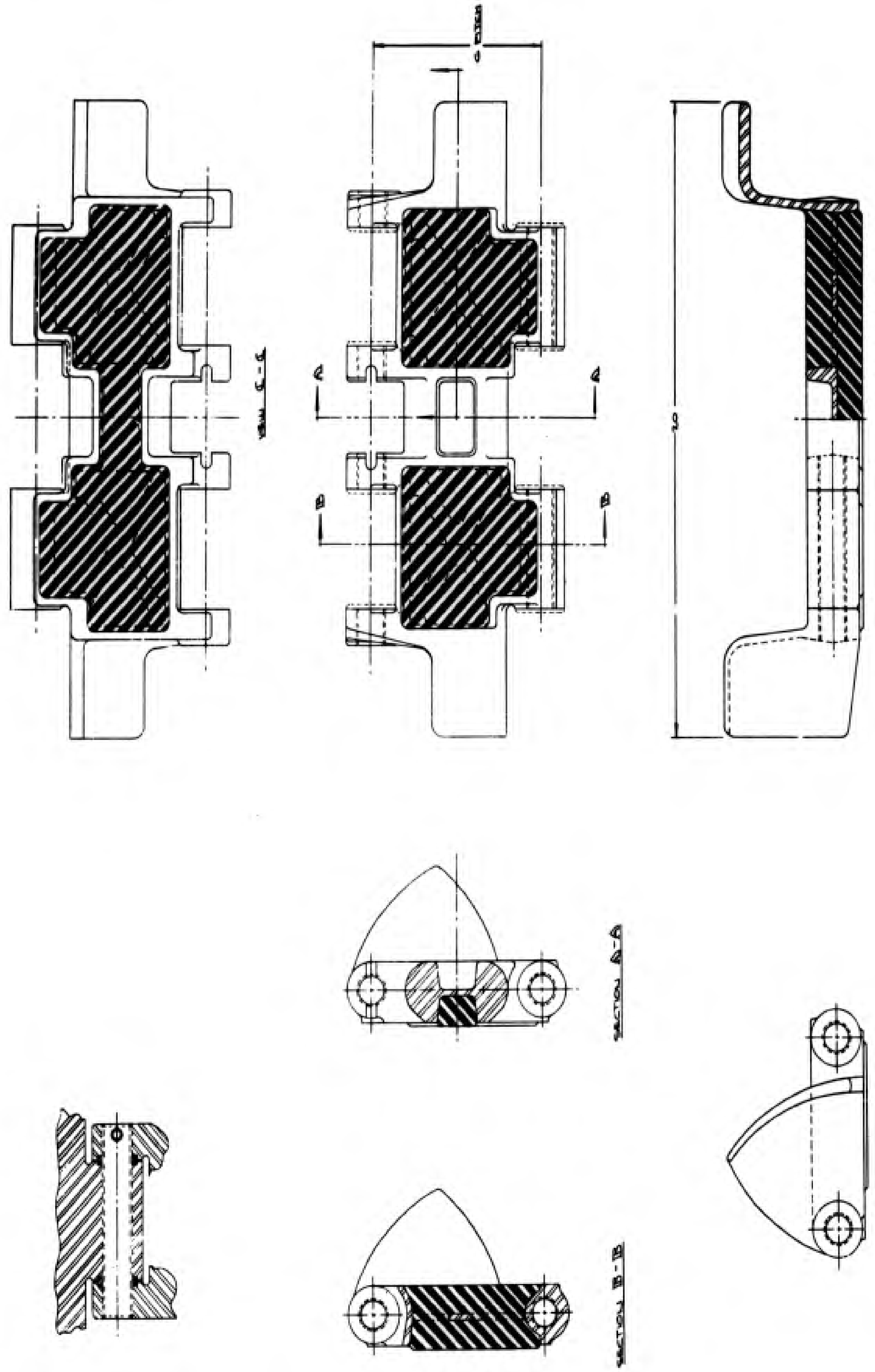


Figure 5-7 Track Block

1. Rubber molded-in pads are provided in the block which acts as a rubber coated wheel path. Arctic tests of the LVTP5 hinge pin track proved the desirability of a rubber path for the road wheel to prevent ice accumulation and consequent tire failure. It also follows that mud build up will be held to a minimum.

2. Water propulsion is provided by the end buckets, which are compatible with the rubber pads incorporated in the design. The end buckets not only provide water propulsion, but additionally provide an aggressive track which is in accordance with the latest tank ordnance practice.

3. Use of a center drive simplifies the incorporation of the rubber-covered wheel path as provided by the pads. The centrally located single sprocket eliminates the requirement for accurate indexing, as would be required with two sprockets.

4. The inboard ends of the grousers serve as the track guides. It is recognized that center guides are somewhat better than side guides. However, since the side buckets exist, they are made to serve a dual purpose, again in the interest of lightweight construction.

5. Investigation has been made of lightweight materials for the basic track block with steel inserts employed in the wear

areas. This approach would be highly experimental and require field evaluation and therefore is more adaptable to a track development program on a vehicle with known characteristics than on an untried vehicle. In light of this an all steel track block has been selected. The track block is presently shown as a casting; however, in the final design efforts will be directed toward a forged steel block to provide a still lighter weight track. Development of lightweight alloy tracks in Army Ordnance programs is in process and should be watched for possible future test on this vehicle.

6. The connector pin will be one inch in diameter having 14 threads to the inch with a minor thread root diameter of 0.9072 inch. The pin size is based on experience over the past years of LVTP5 operation using a one inch diameter unthreaded pin.

5.1.5.4 Track Analysis.

Width = 20": Pitch = 6:

Weight per pitch = 24.5 lb.

Weight per foot = 49 lb.

Number of pitches comprising one track = 82

Weight of one track = 2009 lb.

Ground pressure, empty vehicle, 2 inch penetration = 4.1 psi

Ground pressure, fully loaded vehicle, 2 inch penetration = 5.7 psi

5.1.5.5 Pin Stress.

One inch diameter pin threaded 14 threads to the inch

Thread root diameter = 0.907

Area = 0.646 sq. in.

Maximum Track Tension - 18,000 lb. (compensator controlled)

Force on pin = 18,000 ÷ 2 pins = 9,000 lb. ÷ 2 shear planes = 4,500 lbs.

Pin shear stress = 4,500 ÷ 0.646 = 6966 psi

Pin bending stress due to seal counter bores = 4,500x0.3125 ÷ 0.0731 = 19,260 psi

Experience shows that the lateral and torsional forces acting on the track as it lays over the ground will result in localized stress loads of approximately four times that of the 18,000 pound maximum tension loads. Therefore, the pin material will be SAE 4340 heat treated to provide a tensile yield of 194,000 psi. This will provide a realistic safety factor in shear of 4 to 1 and in bending of 2.5 to 1.

5.1.5.6 Block Stress.

The center drive sprocket results in a track block having the highest stress through its center. In one instance the load is applied by the sprocket tooth which is resisted by the connector pins and clevises. In the other instance the block becomes a beam supported at the ends with the load applied by a stone under the inner tire flange.

Material: Cast steel QQS 681b 4C3

Tensile 120,000 psi

Yield 100,000 psi

Elongation 14%

Brinell 311

Stress from sprocket drive = $\frac{Mb}{Z}$

$$\frac{18{,}000 \times 4.125 \text{ (Moment Arm)}}{2 \times 1.21} = 30{,}681 \text{ psi}$$

$$\text{Safety factor} = \frac{100{,}000}{30{,}681} = 3.3 \text{ to } 1$$

Stress from roadwheel equals $\frac{Wav}{\cancel{Z}I}$

$$= \frac{35{,}000 \times 7.75 \times 10}{8 \times 0.566 \times 20} = 29{,}945 \text{ psi}$$

$$\text{Safety factor} = \frac{100{,}000}{29{,}945} = 3.3 \text{ to } 1$$

5.7 Other Concepts Considered.

5.7.1 Airoll.

The Airoll system possesses extremely high mobility characteristics and extreme simplicity of design as the large footprint tires also perform as the suspension for the vehicle. Tests have proved that a length width ratio of nearly 1 to 1 is required for best steer. Therefore this system would be too space consuming in conjunction with the volume pay load requirements for the subject vehicle.

5.7.2 Overwater Track Return With Rollers.

This system can possess unlimited approach and departure angles with the grade climbing ability of the vehicle limited only by the aggresiveness of the track, cohesion of the soil, and stability of the vehicle itself. An aggressive track will also perform as an excellent ice breaker.

The disadvantages include a high vehicle silhouette, less than optimum water performance due to aeration of the water by entrance of track, increased net vehicle weight due to return rollers and in particular to the excessive length of track.

5.7.3 Underwater Track Return With Rollers.

This system permits a lower vehicle silhouette and less track aeration of the water than the overwater track return; however, the net vehicle weight includes unnecessary weight due to the return rollers and resultant extra length of track.

5.7.4 Band Tracks.

From the standpoint of weight, the band tracks, usually constructed of molded rubber and steel cables, would seem to be the logical track for this application. However, tracks of this type lack transverse torsional stability and have a short life span when compared to the rubber bushed and hinged pin tracks generally selected for use on medium and heavy tanks. Failure is usually caused by bending fatigue in the cables and/or rubber band.

5.7.5 Rubber Bushed Tracks.

The rubber bushed tracks have been developed to a point where bushings and pins are very dependable. In addition, the tracks are quiet and will absorb loads readily. The same flexibility that provides such desirable features also results in lack of transverse stability and in excessive track stretch, either of which will tend to cause "thrown tracks" and excessive sprocket climbing or jumping. It is also very difficult to produce lightweight tracks, because the rubber bushing and pin combination must be enclosed in a relatively large tube which becomes a part of the basic shoe or block.

6.0. ELECTRICAL AND ELECTRONIC SYSTEMS

6.1. General System Description.

The electrical system in the MAC will be required to furnish power for engine starting, instrumentation, warning system, accessories, communications, lighting and controls. The basic system will consist of a battery to furnish power during periods when the engine is not in operation; the dynamic device, a generator, to furnish power during periods when the engine is operating; a control system to control the power utilized by the components; and a distribution system to transmit the power from the sources to the points of utilization.

Monitoring of the engine, transmission, and accessory equipment is accomplished through the use of appropriate instruments and warning systems. These systems are designed utilizing proven principles as well as new developments in these fields to provide an accurate and highly reliable system of supplying vehicle operational information to the operator at all times.

All circuits in the vehicle are protected by automatic reset type circuit breakers to provide maximum dependability. The entire electrical system is designed to provide maximum reliability consistent with low weight and economy. Multiple sources are available for each component.

6.2. Basic Electrical Circuits.

The basic electrical system is a 24 volt DC system as utilized in the majority of military vehicles today. Figure 6-1 illustrates the schematic diagram of the

B-1	Scavenge Blower
B-2	Heater Blower
BT-1	Battery
BT-2	Battery
CB-1	Circuit Breaker 1 Amp.
CB-2	Circuit Breaker 3 Amps.
CB-3	Circuit Breaker 3 Amps.
CB-4	Circuit Breaker 1 Amp.
CB-5	Circuit Breaker 60 Amps.
CB-6	Circuit Breaker 6 Amps.
CB-7	Circuit Breaker 7 Amps.
CB-8	Circuit Breaker 60 Amps.
CB-9	Circuit Breaker 10 Amps.
CB-10	Circuit Breaker 15 Amps.
CB-11	Circuit Breaker 6 Amps.
CB-12	Circuit Breaker 10 Amps.
CB-13	Circuit Breaker 15 Amps.
CB-14	Circuit Breaker 10 Amps.
CB-15	Circuit Breaker 5 Amps.
CB-16	Circuit Breaker 1 Amp.
CR-1	Rectifier
CR-2	Rectifier
DS-1	Generator Warning Light (Red)
DS-2	Engine Low Oil Pressure Warning Light (Red)
DS-3	Engine High Temperature Warning Light (Red)
DS-4	Transmission Oil High Temperature Warning Light (Red)
DS-5	Transmission Low Oil Pressure Light (Red)
DS-6	Range Light
DS-7	Navigation Lights
DS-8	Service Stop and Turn Rear Port Light
DS-9	Service Turn Front Port Light
DS-10	Blackout Turn and Tail Port Light
DS-11	Blackout Marker and Turn Front Port Light
DS-12	Blackout Turn and Tail Starboard Light
DS-13	Blackout Marker and Turn Front Starboard Light
DS-14	Service Stop and Turn Rear Starboard Light
DS-15	Service Turn, Front Starboard Light
DS-16	Service Tail, Port Light
DS-17	Service Tail, Starboard Light
DS-18	Service Headlight, Port
DS-19	Service Headlight, Starboard
DS-20	Park Light, Port
DS-21	Park Light, Starboard
DS-22	Signal Search Light
DS-23	Blackout Stop Light
DS-24	1-R Head Light Low Beam, Port
DS-25	1-R Head Light Low Beam, Starboard
DS-26	1-R Head Light High Beam, Port
DS-27	1-R Head Light High Beam, Starboard
DS-28	Compartment Light
DS-29	Compartment Blackout Light (Red)
DS-30	Compartment Light
DS-31	Compartment Blackout Light (Red)
DS-32	Compartment Light
DS-33	Compartment Blackout Light (Red)
DS-34	Compartment Light
DS-35	Compartment Blackout Light (Red)
DS-36	Compartment Light
DS-37	Compartment Blackout Light (Red)
DS-38	Instrument Panel Light (Red)
DS-39	Compass Light (Red)
G-1	Generator
G-2	1-R High Voltage Amplifier

J-1	Slave Receptacle
J-2	Convenience Outlet
J-3	Convenience Outlet
J-4	Convenience Outlet
J-5	Convenience Outlet
J-6	Radio Outlet
K-1	Master Contactor Relay
K-2	Generator Warning Light Relay
K-3	Starter Contactor Relay
K-4	Bilge Pump Relay
K-5	Port Turn Light Relay
K-6	Starboard Turn Light Relay
K-7	Emergency Hydraulic Pump Relay
L-1	Fuel Shut-Off Solenoid (Engine Stop)
L-2	Port Track Compensating Solenoid
L-3	Starboard Track Compensating Solenoid
LS-1	Horn
M-1	Speedometer
M-2	Tachometer
M-3	Voltmeter
M-4	Port Fuel Level Gage
M-5	Starboard Fuel Level Gage
M-6	Speedometer Sending Unit
M-7	Tachometer Sending Unit
M-8	Port Fuel Gage Sending Unit
M-9	Starboard Fuel Gage Sending Unit
M-10	Turn Signal Light Flasher
MP-1	Bilge Pump
MP-2	Port Fuel Pump
MP-3	Starboard Fuel Pump
MP-4	Starter Motor
MP-5	Emergency Hydraulic Pumps
S-1	Master Switch
S-2	Bilge Pump Switch
S-3	Horn Switch
S-4	Scavenge Blower Switch
S-5	Heater Blower Switch
S-6	Emergency Hydraulic Pump Switch
S-7	Steer Switch
S-8	Engine Low Oil Pressure Switch
S-9	Engine High Temperature Switch
S-10	Transmission Oil High Temperature Switch
S-11	Transmission Low Oil Pressure Switch
S-12	Ignition Switch
S-13	Light Switch
S-14	Navigation Light Switch
S-15	Turn Signal Switch
S-16	Stop Light Switch
S-17	Compartment Light Switch
S-18	Compartment Light Switch
S-19	Compartment Light Switch
S-20	Compartment Light Switch
S-21	Compartment Light Switch
S-22	I-R Viewer Switch
S-23	I-R Lights Dimmer Switch
S-24	Signal Search Light Trigger Switch
V-1	Infrared Viewer
VR-1	Voltage Regulator

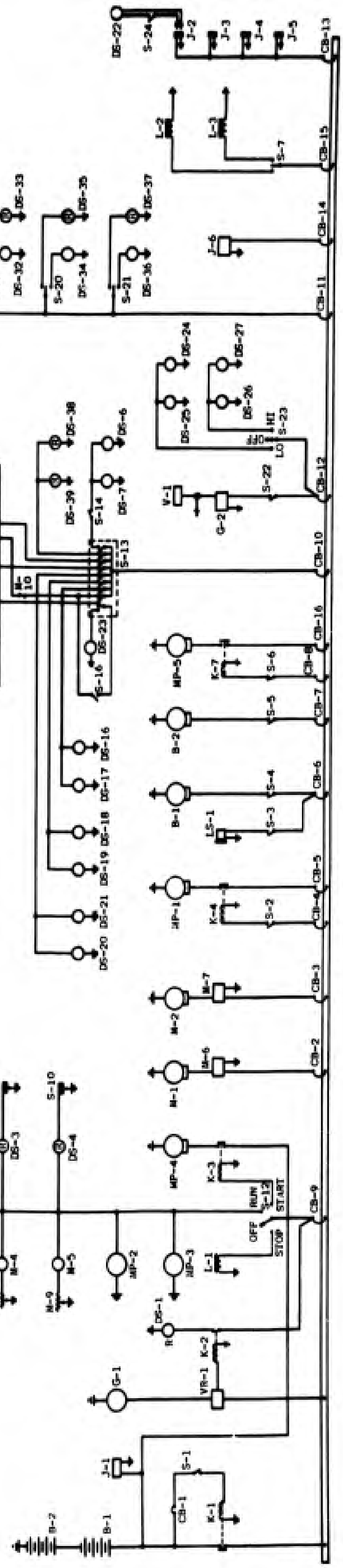


Figure 6-1. Electrical System Schematic

complete system. The voltage and distribution system utilized provides the greatest degree of flexibility and compatibility with other vehicles with which it will be operating. To provide the lightest weight and simplicity, power is supplied by a single wire to each electrical component with the bonded structural members completing the circuit to these components.

6.3. Static Power Supply.

Lead acid type batteries are utilized as the static source of power for the vehicle when the main engine is not operating. This supply is composed of two 12 volt batteries connected in series to provide a total capacity of 100 ampere hours at 24 volts. These batteries are standard military components similar to MS-35000-3. It has been established by test that this battery will supply a current of 50 amperes for 38 minutes at a temperature of -25°F. to a final voltage of 0.875 volts per cell. This capacity is such that sufficient power is available under all normal ambient operating conditions for engine starting purposes as well as accessory operations. Due to the inherent characteristics of the lead acid battery, capacity is reduced when operating in low ambient temperatures. The battery capacity is reinstated under these conditions simply by providing a source of heat to raise the battery temperature. The lead acid type battery was selected for this application due to initial and operating economy, as well as ease of service and logistic support.

6.4. Dynamic Power Supply.

An alternator is incorporated as the dynamic power source for the MAC. This is an AC generator whose output is rectified to provide the direct current necessary for compatibility to the system. The rectifiers are an integral part of the alternator which provides for greatest reliability as well as ease of cooling since the internal fan of the alternator is utilized for both machine and rectifier cooling. Utilization of this type of generator in lieu of a direct current generator is advantageous because power may be obtained during engine idle conditions, thus reducing the load on vehicle batteries. Further, this type of dynamic power source was chosen over a standard direct current generator to reduce maintenance problems and improve reliability since the commutation problem is eliminated and longer brush life may be expected when operating on slip rings rather than a commutator. The rectifiers are the small semi-conductor variety which are inherently more reliable.

The power analysis for this vehicle, as portrayed in Table 6-1, illustrates that the maximum possible load for continuous plus intermittent operation is 218.99 amperes. This condition occurs when the vehicle is operating at night under service conditions, in the water. The alternator capacity is then determined as follows:

From Table 6-1

Anticipated continuous current = 63.79A Maximum

Anticipated intermittent current - 111.20A Maximum

Assume 85 percent probability factor for continuous requirement

Probable continuous current = (63.79) (0.85)

= 54.22A.

Assume 50 percent probability factor for intermittent requirement

Probable intermittent current = (111.20) (0.5)

= 55.6A.

Total probable current = 55.6 + (63.79-54.22)

= 55.6 + 9.57

= 65.17A.

Maximum total (continuous + probable)

A_t = 54.22 + 65.17

= 119.39A.

Assume 60 percent demand factor

Q = (119.39) (0.6)

= 71.63A.

Based on the above, a 100 ampere alternator has been incorporated. The capacity of the alternator plus the capacity of batteries will satisfy the complete power requirements of the vehicle under all normal emergency conditions. The alternator selected is similar to ORD 7954720 with the exception that the selenium rectifiers are replaced by the silicon type.

Table 6-1. Electrical Power Analysis

Description	Current Requirements	Quantity	Water				Land			
			Service		Blackout		Service		Blackout	
			Continuous	Intermittent	Continuous	Intermittent	Continuous	Intermittent	Continuous	Intermittent
Infrared Equipment	5.5	1			3.5				3.5	
Headlights	2.86	2	5.72				5.72			
Service Stop & Turn Lights	1.15	4						4.60		
Service Taillights	.19	2	.38				.38			
Range Light	.42	1	.42							
Navigation Lights	.66	1	.66							
Panel & Compass Lights	.29	2	.58		.58		.58		.58	
Compartment Lights, Service	.75	5	3.75				3.75			
Compartment Lights, Blackout	.19	5			.95				.95	
Warning Lights	.29	5		1.45		1.45		1.45		1.45
Signal - Search Light	1.15	1		1.15				1.15		
Voltmeter	.04	1	.04		.04		.04		.04	
Fuel Gage	.08	2	.16		.16		.16		.16	
Speedometer	2.5	1	2.5		2.5		2.5		2.5	
Tachometer	2.5	1	2.5		2.5		2.5		2.5	
Radio	10.0	1	10.0		10.0		10.0		10.0	
1-R Headlights	1.91	2			3.82				3.82	
Bilge Pump	52.0	1		52.0		52.0				
Scavenge Blower	4.25	1								
Heater Blower	6.0	1	6.0		6.0		6.0		6.0	
Fuel Pump	2.5	2	5.0		5.0		5.0		5.0	
Relays	.35	6	.70	2.10	1.40	2.10	.35	1.75	1.05	1.75
Solenoids	1.5	3		4.5		4.5		4.5		4.5
Starter	750 Breakaway 550 Sustain			550.0		550.0		550.0		550.0
Battery Charging			25.0		25.0		25.0		25.0	
Blackout Turn & Taillights	.19	4			.76				.76	
Blackout Stop	.19	1				.19				.19
Service Park Lights	.19	2	.38				.38			
Emergency Hydraulic Pump	150A. Max. 50A. Ave. 30A. Idle	1		50.0		50.0		50.0		50.0
Sub Totals			63.79	111.2	62.21	110.24	62.36	63.45	61.86	57.89
Totals			174.99		172.45		125.81		119.75	

6.5. Auxiliary Power Source.

In some vehicular applications a standby source such as an auxiliary power unit (APU) is provided to supply electrical power without the necessity of operating the vehicle engine. This may also be utilized to supply additional power during periods when supplementary power is required. The power analysis of the vehicle indicates that this APU will not be required in the basic vehicle, but may be required in the specialized vehicles to permit operation of the specialized equipment without the necessity of operating the engine. When this power unit is required, it will operate on the same fuel and from the same fuel supply as utilized by the main engine. In addition, a slave receptacle is provided which is connected directly to the batteries to permit power transfer either to or from the vehicle should this necessity arise. In this manner one vehicle may supply power to another to aid in engine starting should the batteries of a particular vehicle become disabled. This high capacity receptacle also permits battery charging from an outside power source.

6.6. Distribution System.

Past experience has proven that a centralized distribution system is the most efficient method of power distribution in vehicles. Utilization of connectors at all equipment and cable terminations permits rapid installation and ease of vehicle servicing. A central distribution point contains the majority of control and protection equipment such as contactors, relays, and circuit breakers. Since the majority of circuits will originate or emanate from this point, electrical access to remotely located equipment for in-vehicle testing purposes is greatly facilitated. The connectors utilized on all cables and equipment are the standard AN type configurations. The entire system is designed to meet the waterproofing requirements of specification MIL-E-13856. Radio interference is reduced to acceptable levels in accordance with MIL-S-10379 by shielding and the use of feed-through type filters. Where applicable the wiring meets cable specification MIL-C-13486 and the remainder of the wiring meets specification MIL-W-5086 or other military specifications which may apply to a specific application as determined by environmental conditions.

6.7. Instrumentation.

All instruments, control switches and warning lights are mounted on an instrument panel located at the driver's station. These are arranged in such manner that the desired vehicle operational information is presented to the driver in the least confusing manner. This arrangement is illustrated in Figure 6-2. Operation of electrically actuated and remotely located equipment will be accomplished from this panel. Instruments are provided to monitor fuel level in both port and starboard fuel tanks, engine speed, vehicle speed, and battery generator voltage. An engine hour totalizer is included in the engine tachometer, and an odometer is provided in the vehicle speedometer to register statute miles traveled. The instruments utilized possess a movement of approximately 330 degrees to provide maximum accuracy. The fuel level sending unit is the submerged resistance type which permits complete sealing of this unit since

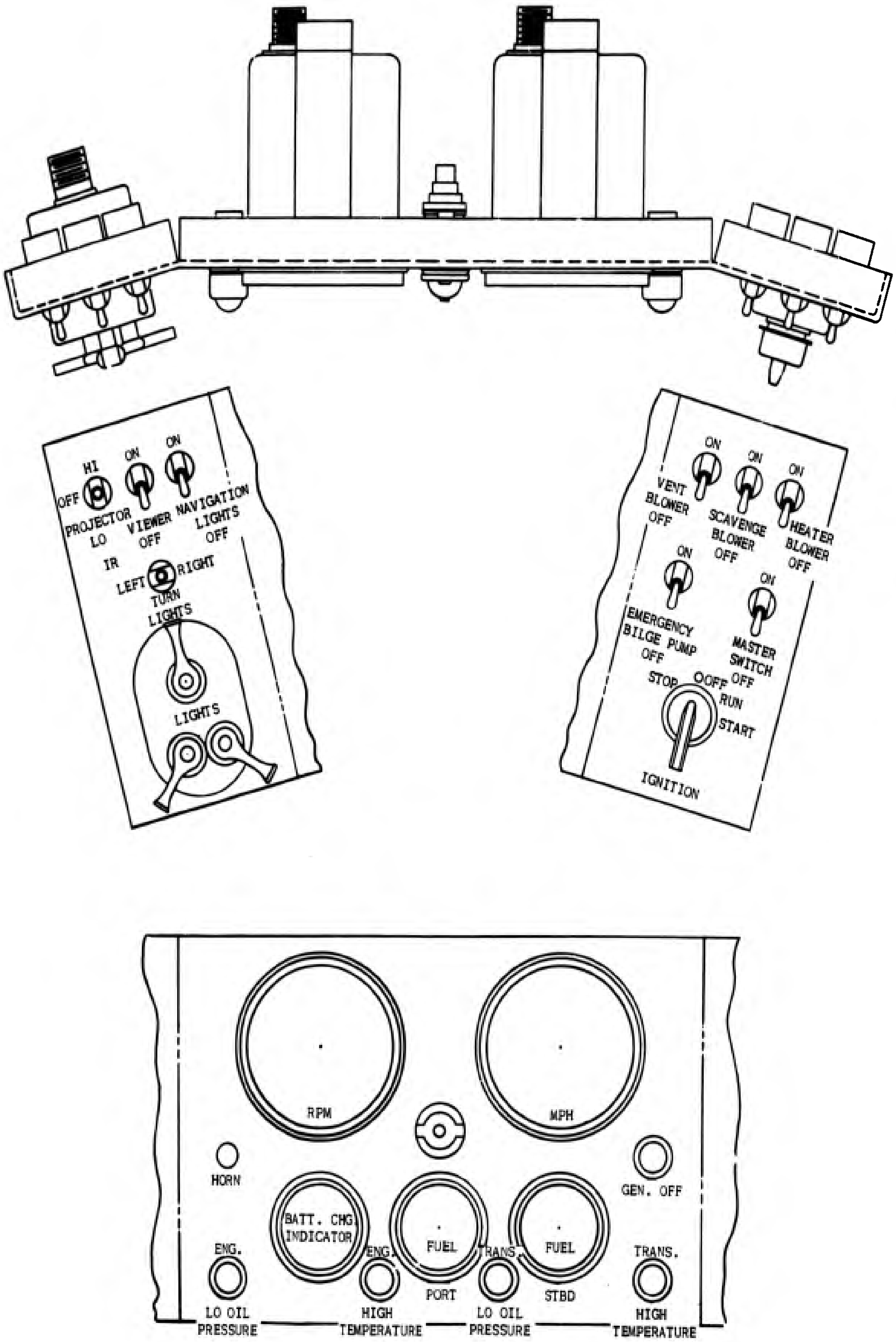


Figure 6-2. Instrument Control Panel

only a sealed wire must be brought out from the variable resistor to the waterproof connector on the outside.

Toggle type switches have been utilized to operate remotely controlled equipment since toggle position is most readily recognized by the driver. The lighting and engine control switches are the rotary type to provide contrast. The lighting switch includes a mechanical interlock to prevent inadvertent operation of the blackout, service and navigation lights. The engine control switch includes a momentary position for actuating the engine cranking motor.

6.8. Circuit Protection.

Thermally operated automatic reset type circuit breakers are incorporated to protect the circuits involved. These are located at the circuit origination point to provide maximum protection against overloads and faults. Short duration transient effects are minimized by utilizing the thermal type of circuit protection system.

6.9. Warning System.

Warning lights are utilized to indicate engine low oil pressure, engine coolant high temperature, transmission low oil pressure, transmission high oil temperature and generator off. These lights are located on the driver's instrument panel to provide a warning that the monitored systems have departed from the normal operating condition. The display is arranged to attract the driver's attention in the shortest practical time.

6.10. Lighting Systems.

Several lighting systems are provided on the vehicle for use under all operating conditions which may be encountered, utilizing Ordnance and Navy type lights. These include service lights, blackout lights, navigation lights, internal lights, and infrared viewing system. Service lights, both from and rear, are provided for night operation under noncombat conditions. These include headlights, tail-lights, turn and parking lights. To permit operation under blackout conditions, blackout type lights are provided both front and rear. These include marker lights at the front, marker lights at thc rear, and stop lights at the rear. For operation on waterways during nontactical conditions, removable navigation l lights are included for ships of this class. These are standard Navy lights consisting of a combination red and green light forward and a range light aft. A signaling search light is also incorporated to increase visibility, both land and water, and to signal other vehicles, boats, or troops if this is desired. This light is normally stowed inside the vehicle. Lighting of the crew compartment and engine compartment is accomplished through the use of combination service and blackout lights. Each light includes a selector switch with a mechanical interlock for selecting either the service or blackout portion of the light.

Under complete blackout conditions an infrared viewing system is provided for the driver. This system consists of two infrared projectors and an infrared viewing system which may be operated concurrently or independently to permit greatest flexibility. The viewer is normally stowed inside the vehicle and may be installed on the periscopes for viewing when the hatches are closed, or installed on a bracket on the outside of the driver's cupola for operation when this hatch may be opened. In this manner the infrared equipment may be utilized for driving under complete blackout conditions or the viewer may be utilized for observation purposes to detect any infrared radiation source.

6.11. Radio Equipment.

Provisions are included near the gunner's station to install the military AN/PRC-47 or the Collins Radio Company 618T single side band radio equipment. The installation of the AN/PRC-47 radio is shown on Figure 6-3. The 618T transceiver could also be installed in this area and the remaining equipment required for the 618T installation located near the upper deck on the starboard side of the vehicle. The antenna utilized is the fiberglass protected whip type, and is located forward of the main cargo opening on top of the vehicle.

6.12. Bilge Pump.

An electrically driven bilge pump is installed which provides a supplemental pump capacity of approximately 125 gallons per minute. Any infiltration of water may be removed from the bilges if it should be necessary to stop the engine while the vehicle is water-borne. Should the necessity arise to tow the vehicle in the water, additional power may be supplied to the vehicle for operation of the pump by means of the slave receptacle as discussed in

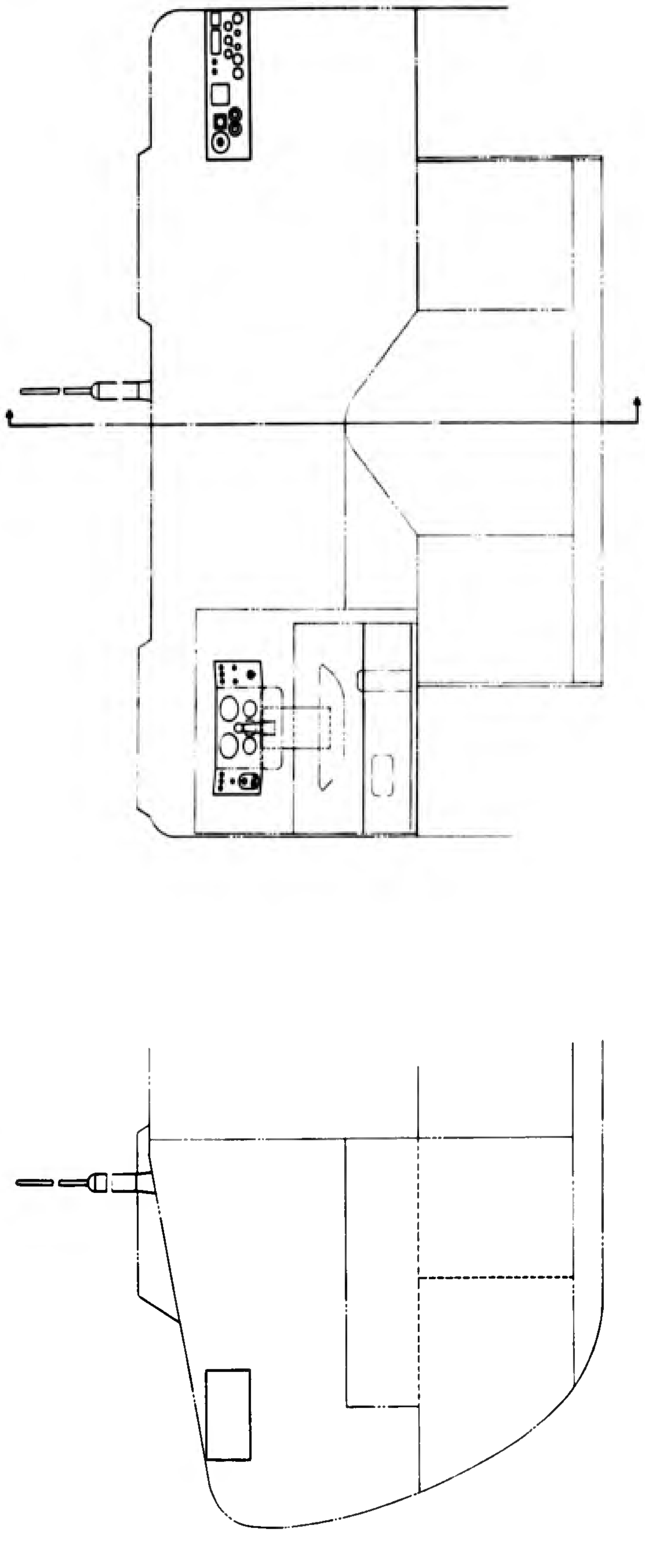


Figure 6-3. Radio Installation

paragraph 6.5. This pump is a centrifugal type cabable of supply a relatively high pressure against the total dynamic heads which may be encountered in the installation. The motor is a shunt wound type with lubrication provided to the motor and pump seals which permits continuous operation under either wet or dry conditions. This pump is the same as recently developed and incorporated in the LVTP5 family of vehicles.

6.13. Compass.

A compensated magnetic compass is located at the driver's station. Provisions are included for internal illumination to permit ease of visibility during hours of darkness or when all access hatches are closed.

7.0. HYDRAULIC SYSTEM

The accessory fluid power and control functions for the MAC are accomplished by a simple, reliable and trouble free hydraulic system. The basic hydraulic system, illustrated in Figure 7-1, is a closed center, power on demand type which is adaptable to other vehicles of this family group for fulfilling their specific power functions. The main hydraulic power source is an engine driven power take-off pump which is rated at 3,000 psi and 32 gpm at 2,400 rpm. This pump is a variable volume, pressure compensated, axial piston type. Higher pressure hydraulic systems reduce the size of the components, thus reducing weight and cost. The variable volume pump eliminates the valving required by a normal fixed displacement pump. This type of pump will give better actuation response time and will maintain a minimum heat or horsepower loss in the circuit because it will only allow fluid power on demand.

One pump supplies the necessary hydraulic fluid power to operate the ramp, bilge pumps, fan drive motors and track compensating cylinders. A priority valve at the outlet of the pump provides first priority to the track compensating cylinders, the fan drive motors and aspiration fan motor thereby giving maximum reliability to this circuit. However, since the track compensating circuit is a static circuit and its needs are intermittent and with the fan drive units using only 6.5 gpm the bilge circuit in effect will have continuous power flow. Pressure compensated flow control valves, incorporated in the circuits, limit the maximum speed of actuation of the bilge motors, fan motors and ramp cylinders.

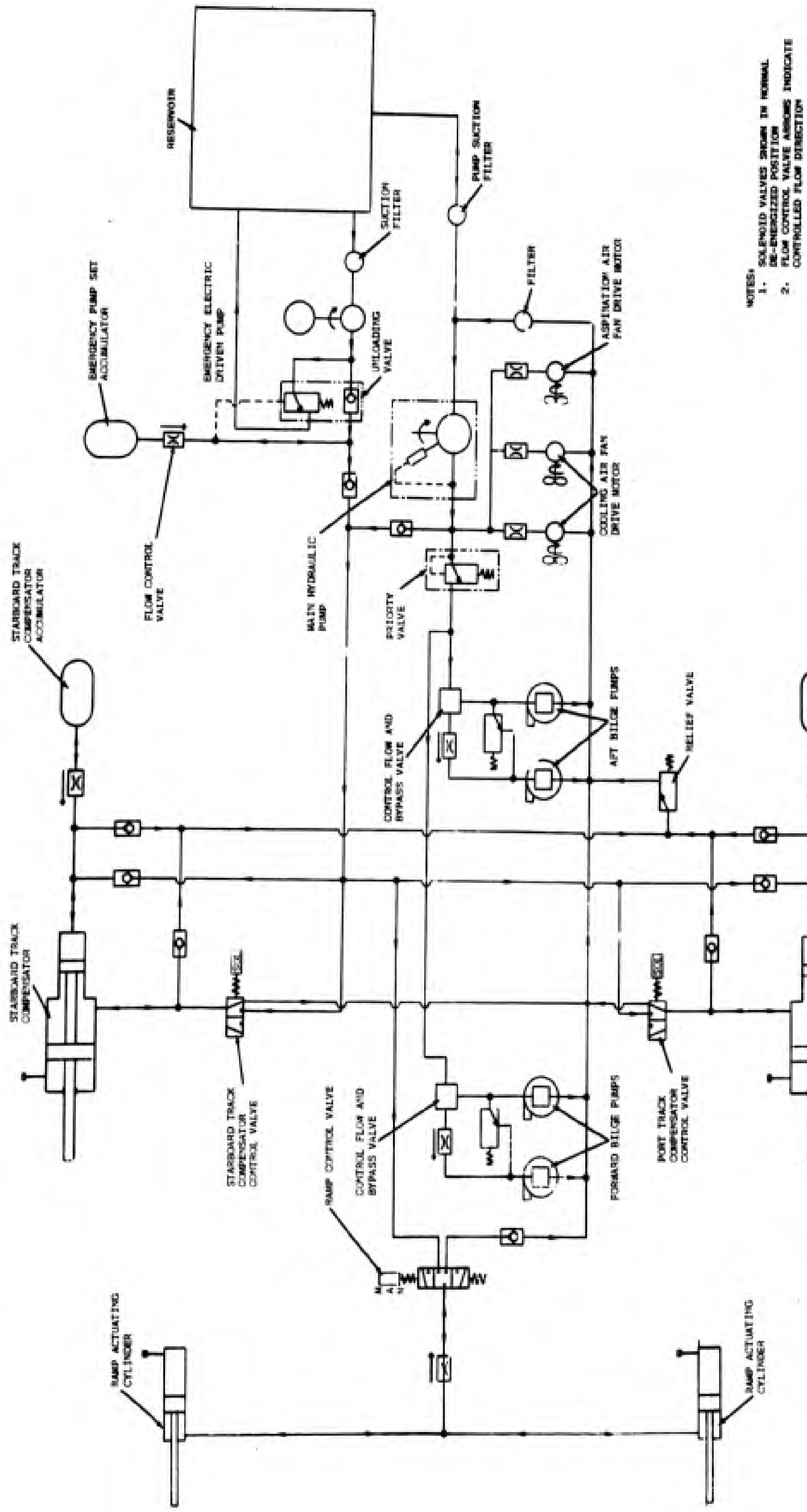


Figure 7-1. Hydraulic System Schematic

7.1. Bilge Circuit.

The four bilge pumps (Figure 7-2) run continuously. Each pump is rated at 300 gpm at a 10 foot head evacuating bilge fluid at 1,200 gallons per minute. At minimum engine speed, one forward and one aft pump have maximum pressure flow. The flow is automatically controlled by a priority bypass and flow control valve. As the engine speed increases, the pressure flow of the remaining aft and forward bilge pump increases to maximum capacity. This arrangement insures maximum bilge fluid evacuating capacity during idling at sea. The four bilge pumps, located at each corner of the vehicle below deck, are sized to evacuate entering sea water from the air intake louver openings during surfing and possible hull leakage through access openings. The bilge pumps are designed to a minimum height so they will fit under the floor deck. They are enclosed by wire screen to prevent damage to or rag clogging of the inlet impeller assembly of the units.

Piping from the bilge pumps terminates on the side of the hull at a maximum practical height from the hull bottom. Flapper valves at hull outlet prevent sea water entrance should these points be submerged (see detail on Figure 7-2). The pumps are of lightweight aluminum construction. Typical pumps have given well over 1,000 hours maintenance-free operation. Electrically driven emergency pump is described in Section 6-0.

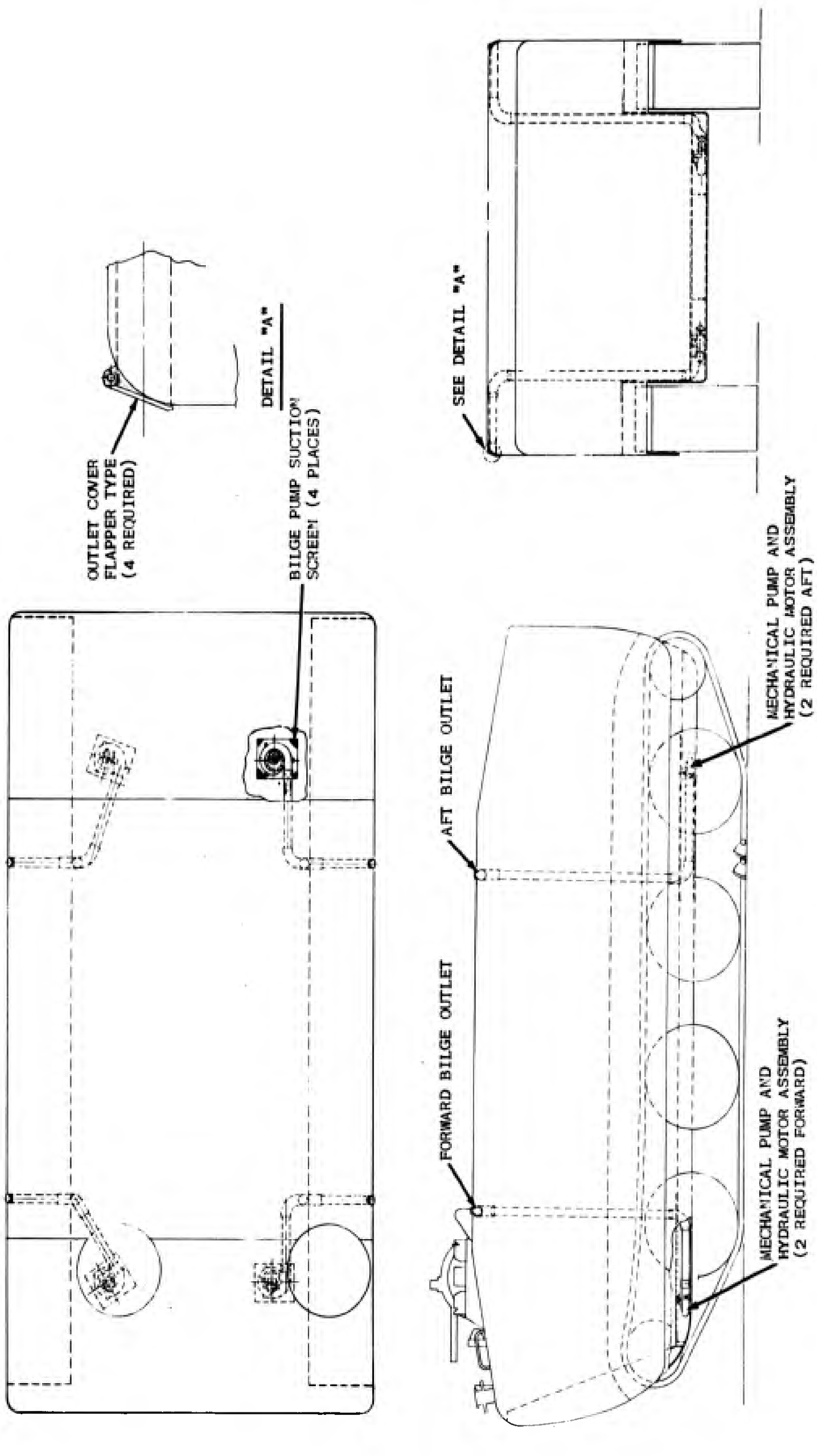


Figure 7-2. Bilge Pump Installation

7.2. Ramp Circuit.

The ramp circuit, Figure 7-3, is controlled by a manual four-way valve with three positions: "raise", "lower", and spring centered "hold" or normal position. This valve actuates the two single-acting ramp cylinders. Time response is controlled by one flow control valve which gives proper speed in lowering the ramp.

The cylinders actuate cables connected to the ramp for raising and lowering. The cable arrangement will prevent damage to the ramp cylinder mechanism when loading or unloading. The cylinders are located under the deck at the bow of the vehicle.

7.3. Track Compensator Circuit.

The track compensator cylinders are controlled by two solenoid operated three-way valves, one for each track steer function. These specially designed single acting cylinders incorporate two pistons of different sizes. The smaller piston in the cylinder is continuously pressurized with 3,000 psi system pressure which counteracts the track forces during straight ahead operation. However, during steer, the inner track must counteract extremely high track forces and during this steering the solenoid is actuated electrically from the driver's steering control stick pressurizing the large piston in the inside track compensating cylinder. This will give the desired large reacting force necessary to prevent track climb and also control maximum reacting force to prevent track damage. The same action takes place for opposite steer. The steering

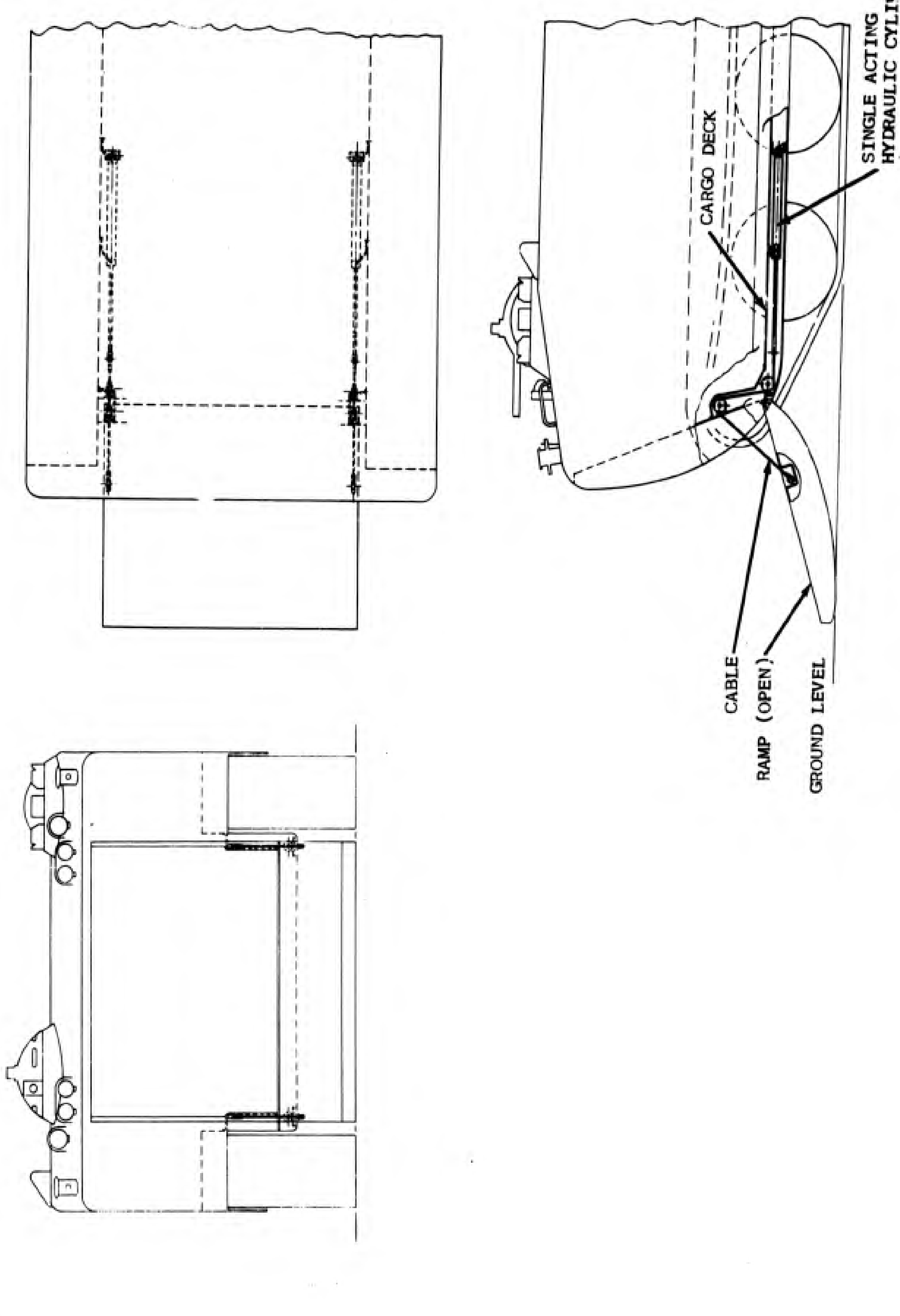


Figure 7-3. Ramp Circuit Schematic

control stick has a normal play before any actuation or track tension takes place and is similar to the play of the steering wheel in present automotive type vehicles.

Any sudden forces acting on the track such as objects encountered over rough terrain are dampened by direct-acting accumulators with controlled make up flow. This type accumulator has been operated and tested for several thousand hours without maintenance.

Check valves are incorporated in this circuit to give the desired flow direction and also to maintain system pressure at the small piston area of the track compensating cylinders in the event of failure of the main hydraulic power source.

7.4. Cooling Fans and Aspiration Fan Drive Circuit.

The three fan drive hydraulic motors (see schematic Figure 7-1) run continuously. The two cooling fan motors are rated at 3 gpm at 3,000 psi to drive the cooling fans at 2,200 rpm. The aspiration fan motor is rated at 0.5 gpm at 3,000 psi. The speed of the motors is automatically controlled by flow control valves. The fan motors are located in the engine compartment. The cooling fan motors could readily be adapted with a modulated thermo control if required by incorporation of one valve.

7.5. Emergency Power Pack Circuit.

An emergency power source (electrically driven hydraulic pump) is provided for emergency use in case the prime mover power source is not available. This unit will provide hydraulic power for ramp and track tensioner cylinders. It has sufficient capacity to operate one bilge pump in an emergency. Although the emergency operation of one fore or aft bilge pump is not shown in the circuit it may be readily incorporated by the addition of one solenoid operated valve. This power pack unit is a 24 volt electric motor driving at 3 gpm at 3,000 psi hydraulic pump. See battery capacity in Section 6.0. Although the capacity of the power pack unit is much less than that of the primary pump, it is adequate for emergency operation for the functions of the personnel carrier vehicle as well as additional low powered functions required by the other members of this vehicle family.

Hand pumps investigated for an emergency power source had too low a cubic inch displacement per stroke for achieving 3,000 psi working pressure. This would have resulted in extremely long time periods to complete a function cycle. It would be limited to cylinder functions only which would not provide sufficient working capacity needed for the auxiliary power functions of the family of vehicles.

7.6. Tubing, Fittings and Hose.

The tube fitting specified is MS (ER) flareless tube type covered by MIL-F-18280 which is made of lightweight aluminum. In order to reduce weight and achieve low maintenance, tubing specified will be 304 or 316 stainless steel. Pipe fittings, where used, will be NPTF (Dryseal thread). The use of Loctite on all threaded fittings will give maximum protection against leakage due to vehicle vibration and will assure trouble free, long-term operation of the vehicle. The use of "Diso-grin" O-rings on fittings will assure one common type O-ring for all applications. These O-rings have an indefinite storage life.

Hoses used will be to MIL-H-19606 (Ships) type B specifications. This type has the synthetic rubber covering to guard against hose deterioration and corrosion due to oil, gasoline, mildew and abrasion.

Piping sizes will be established to provide fluid velocities of 15 feet per second for pressure lines, 10 feet per second on common return lines and 4 to 6 feet per second on suction lines.

7.7. Other Components.

In this design, the contractor has used service tested and proven off-the-shelf items, or items that have been accepted by qualifying under a military specification.

Filters are of the micronic or self-cleaning, edge type featuring reusable cartridge with a built in differential relief valve.

Check valves are low pressure cracking type that are self-cleaning.

Solenoid valves are covered by MIL-E-13956 specification.

7.8. Reservoir.

The reservoir capacity is approximately 10 gallons in the closed loop hydraulic circuit. Since this hydraulic circuit is basically a power-on-demand type, very little heat will be generated and no additional cooling is required. The reservoir has a drain outlet, a fill inlet, a pressure relief cap with vacuum release, and a dip stick for oil level control.

7.9. Oil.

Oil specified is MIL-H-6083A type I, a stable viscosity oil which insures reliable performance during all-weather operation. However, an SAE 10 type oil could be incorporated to give satisfactory performance under limited temperature conditions.

8.0 FUEL SYSTEM

8.1 Type of System.

The fuel system for operation of the MAC engine, illustrated in Figure 8-1, is a pressurized system, with fuel being gravity fed to a booster pump which pressurizes the fuel to the engine pump.

A study of the relationship of the engine fuel pump to the fuel cells at the specified vehicle attitudes determined that the engine fuel pump did not have sufficient capacity. It could not lift the fuel at the maximum specified slopes. Therefore a pressurized fuel system was chosen to assure a positive head of fuel at the engine for the 70 percent forward and 60 percent side slope conditions of the vehicle.

8.2 Capacity.

A study of the fuel required to achieve the specified operational ranges, both land and water, led to the selection of two fuel cells of 117 gallon total capacity. This amount of fuel specifically provides 50 nautical miles of water operation and 301.5 miles at 25 mph on land.

.43 #/BHP-HR

1790 miles

8.3 Fueling.

Fuel fills are provided at each cell and are located on the top deck above each fuel cell. The fill tube is designed to extend above the deck and lock-in position to permit refueling at sea. The fuel fill cap is a pressure and vacuum release type. It is considered practical to utilize a

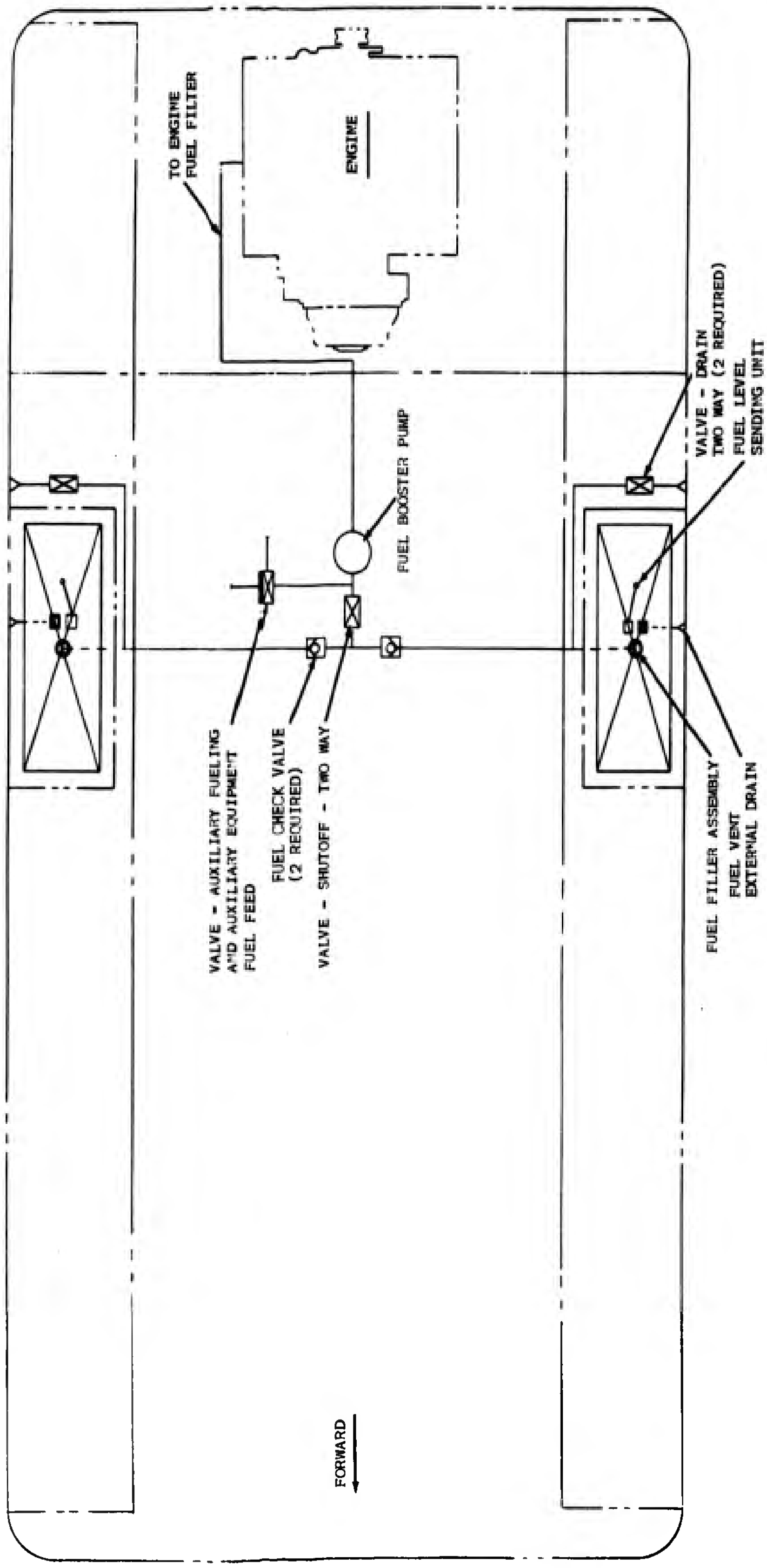


Figure 8-1. Fuel System Schematic

fuel time of 4 minutes for fueling the MAC to maximum capacity. It would be practical to gyrate the fuel into one cell at the rate of 30 gpm. Filling both cells simultaneously, the vehicle could be filled in 2 minutes.

8.4 Venting.

An overboard vent is provided for each cell and is designed to prevent sea water from entering the fuel cells. They are positioned on the cell to prevent fuel loss while negotiating the 70 and 60 percent slopes with the cells filled to capacity.

8.5 Auxiliary Fueling.

A three way valve incorporated in the fuel system permits use of a portable auxiliary fuel cell for engine operation during manufacturing and vehicle testing. The auxiliary fuel tank is used in place of the vehicle fuel cells, preventing accumulation of volatile gases in the vehicle fuel cells and system. This same three way valve is also used as a feed point to operate auxiliary equipment if necessary.

8.6 Piping Arrangement.

Check valves in the connecting piping between fuel cells prevent interchange of fuel during the various vehicle attitudes. A shut-off valve at the common feed point to the booster fuel pump allows quick maintenance of the pump. The piping from the fuel booster pump then terminates at the engine fuel filter. Drain piping at each cell directs the fuel outside the vehicle through a shut-off valve.

8.7 Fuel Level.

Each fuel cell liquid head is self leveling. This is accomplished by the check valves and interconnecting piping and only during vehicle operation. Each fuel cell has an electrical type sending unit actuated by a float. See electrical section 6.0 for further details on the sending unit and fuel gauge.

8 8 Piping, Valves.

All fittings are MS flareless tube fittings per specification MIL-F-18280 and all pipe connections are NPTF (dry seal) pipe threads. Tubing is 304 stainless steel and hose is a lightweight type conforming to specification MIL-H-19606A.

Rotary Teflon plug type valves are used in the system and do not require special maintenance lubrication to keep them operative. Check valves are low cracking pressure and self cleaning types.

8.9 Fuel Cell Construction, Installation and Data.

The fuel supply is stored in two fuel cells with a volume of 62.6 gallons each. Calculated unusable fuel amounts to 7 percent or 4.1 gallons per cell. This is based on 5 percent or 2.92 gallons allowed for thermal expansion and 2 percent or 1.18 gallons retained in the cell due to cell outlet fitting location on the cell. The net amount of usable fuel capacity of each cell is 58.5 gallons.

117 gal

The fuel cells are located on each side of the vehicle one above each track channel slightly aft of midship, symetrically opposite. This achieves a practical trim of vehicle regardless of fuel level.

Due to the location of the fuel cells they are vulnerable to small arms fire and shell fragments. For this reason a lightweight self sealing fuel cell constructed per MIL-T-5578B, type II, class A was selected.

The fuel cell cavity for a type II, class A tank requires support completely around the cell to provide adequate support for sealing and sufficient strength to withstand the stresses caused by the hydraulic surge of duel incident to gunfire. Also these supports are somewhat flexible to allow some movement under gunfire to prevent change from overloading support points. This is accomplished through the use of metal stiffeners spaced approximately six inches apart on the interior of the cavity over which plastic backing material is loosely attached to assist in supporting and aligning bullet wounds. The plastic backing material is procured under specification MIL-P-8045. The cell is supported in the cell cavity by two methods. One method is a system of flexible attaching points between the cavity and outer cell surfaces. These are located around the top edges and on the sides, midway between top and bottom surfaces. The other method is a system of hangers that are vulcanized on the cell and located in the same manner as mentioned above. Metal hangers are located on the cell supports in line with and between the cell hangers. Nylon cord per specification MIL-C-5040B type 2 is laced through these hangers and secures the cell in the cell cavity. To remove fuel spillage or leakage in the cell cavity as a result of gunfire a drain valve is provided. Port fuel tank installation is shown on Figure 8-2.

8.10 Fuels.

Some concern has existed in the ability to handle and store, various fuels such as JP-4 and JP-5 aboard vehicles of this type by conventional means and still retain a high level of overall safety to the parent vehicle and its occupants.

An investigation has been made by Ingersoll Kalamazoo Division with various fuel manufacturers to determine the degree of special care required to handle these fuels.

To provide a yardstick of comparison, the fuels in question were compared with gasoline since we have successfully designed adequate fuel systems to handle this fuel on this type of vehicle.

The results of this comparison indicate that the use of JP-4 or JP-5 fuels pose no added problem in handling or providing a safe and adequate system. Actually the contrary exists in that a system safe for gasoline is somewhat safer for the JP fuels.

The following information was obtained from Sinclair test data. An investigation was made as to what portions of air and fuel mixture were

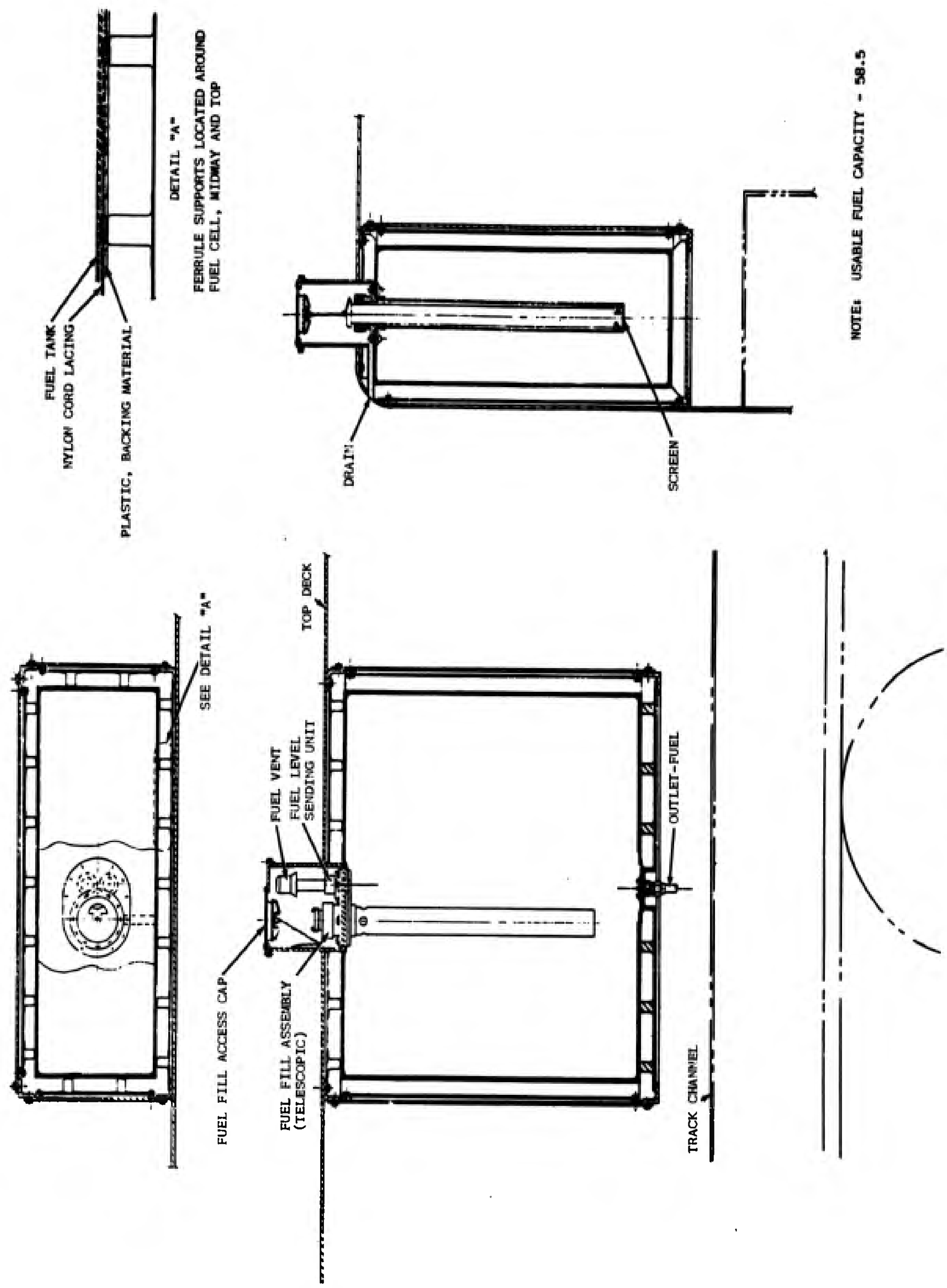


Figure 8-2. Fuel Tank Installation

required to ignite gasoline of 80 to 87 octane and JP-4 jet fuel.

A. Lean mixture consists of 16 parts of air to 1 part of gas.

B. Regular mixture consists of 12.5 to 13 parts air to 1 part of gas.

C. Rich mixture consists of 10 parts of air to 1 part of gas.

Approximately the same mixture is required with JP-4 fuel. However, to obtain proper ignition from this fuel a hot carburetor and manifold are required to retain the heavy end of the fuel and eliminate condensation in the carburetor and manifold.

A comparison of the Distillation Range, Reid Vapor Pressures, and Flash Points are included to substantiate the conclusion drawn. This data is taken from MIL specs as follows:

DATA COMPARISON BETWEEN JP-4 AND GASOLINE

	MIL-J-5624E JP-4	MIL-G-5572C Gasoline Reg. Grade
Distillation Range		
Fuel Evaporated 10% Min. at	------	167° F.
Fuel Evaporated 20% Min. at	290° F.	
Fuel Evaporated 50% Min. at	370° F.	221° F.
Fuel Evaporated 90% Min. at	470° F.	275° F.
Reid Vapor Pressure		
100° F. Min. psi	2.0	5.5
100° F. Max. psi	3.0	7.0
Flash Point		
	125° F. (Approx.) See Note	

Note:

Gasoline is a volatile hydrocarbon fuel and the flash point cannot be given at a definite temperature, because weather conditions play a very important part as to where this point may be. Fuel manufacturers inform us that a greater care is required to maintain a safe system with gasoline than is required to maintain a safe system with gasoline than is required with JP-4 fuel which is about the same as number 1 fuel oil.

TYPICAL SPECIFICATIONS FOR JP-4 JET FUEL

Gravity	56.0
Freezing Point, °F. Max.	-76
Flash Point, °F. (Approx.)	0
Aromatics, Volume, % Max.	25
Olefins, Volume, % Max.	5
Smoke Point, mm Max.	29
Smoke Volume Index	54
Burning Test, 16 hrs.	-
Existent Gum, mg/100 ml. max.	7
Accelerated Gum, mg/100 ml. max.	14
Sulfur, Total Weight, % Max.	0.4
Mercaptan Sulfur, Weight, % Max.	0.005
Heating Value, BTU Min.	18400
Pour Point, °F. Max.	-
RVP, psi	2 - 3
Viscosity @ -30 °F. Max.	-
Water Tolerance, ml. Max.	1
Color, Lovibond Max.	-
Corrosion, Copper Strip Max.	Pass
Distillation, °F.:	
Initial,	159
10% Min. Evap. @ Max.	180
50% Min. Evap. @ Max.	290
90% Min. Evap. @ Max.	370
Final Boiling Point, °F.	470
Residue, Volume %	1.5
Loss, Volume %	1.5
Oxidation Inhibitor Added	Option
Corrosion Inhibitor Added	Yes
Metal Deactivator Added	Option
Acidity	-
Viscosity, Kinematic	0.945
Thermal Stability	OK
Cetane Number	40 - 45

9 Parts JP-4 (2.7 RVP)
1 Part AMOLITE (5RVP) ---- 3 pounds

Note: These specifications were furnished by a JP-4 fuel manufacturer.

For added information, the following lists a general description of fuels on which multifuel engines should be capable of operating to meet military multifuel requirements.

1. Diesel Fuel Oil #2.

   (Military Specification -- VV-7-800)

   Regular Diesel fuel which meets requirements of all automotive high and medium speed diesels in areas where ambient temperatures are above 20° F.

2. Compression Ignition Engine Fuel (C.I.E.).

   (Military Specification -- F-45121A)

   Covers one grade of fuel used in research, development, proof-testing and qualification of compression-ignition engines.

3. Combat Gasoline.

   (Military Specification - G-3056)

   Covers combat grade gasoline suitable for use in all gasoline engines under all conditions of service.

Multifuel engines should operate efficiently on the above fuels and in addition on any of the following:

Diesel Oil #1 (Common commercial type of fuel oil)
JP #5 (Jet - meets special Navy requirements)
Kerosene (Select quality commercial)
JP #4 (Jet - single largest volume military fuel used today)
JP #3

---or any other fuel in the gravity range between Diesel Fuel #2 and combat gasoline or any mixture without engine adjustment.

9.0. VEHICLE SPECIALIZATION

9.1. Command Vehicle.

The basic vehicle concept of the personnel and cargo carrier can be readily adapted to the incorporation of the communications equipment and related components to fulfill the command vehicle requirements.

The cargo area and its easy access through the ramp and hatches provide an all around comfortable environment for the operating personnel.

The equipment arrangement and the over-all concept configuration is shown in Figure 9-1.

Radio equipment is provided to cover the frequency range for communications with other vehicles, aircraft, ships, artillery, and infantry installations. It is anticipated at this time that the following radio equipment will be utilized to accomplish the above:

AN/TRC-75	AN/GRC-5
AN/VRC-24	AN/PRC-10
AN/PRC-47	AN/PRC-38

These radios operate in the HF, VHF, UHF ranges with operation accomplished by AM, FM, and SSB.

Each radio operator will have his own operating location in the craft as shown. Also, by each operator rotating 180 degrees in his seat he will face all of the other operators and the vehicle commander seated around a common reference table. This arrangement provides additional flexibility and ease of command control.

To incorporate all the necessary command equipment requires only modification of the top deck to accept the antenna mounts and placement of the required brackets and supports to accept the radio gear, seats, and tables. Therefore, basic retention of all major vehicle components and systems is accomplished in the inherently flexible design concept of this vehicle.

9.2. Recovery Vehicle.

The over-all general arrangement of the recovery adaptation is shown in Figure 9.2 and includes equipment for towing, lifting and retrieving other vehicles. It also provides maintenance facilities for welding, battery charging, compressed air, and a power source for electric hand tools. The large unobstructed cargo area and wide ramp opening facilitate the incorporation of the necessary equipment to provide these functions.

Located on the front deck area a hydraulic powered crane boom and hoist is mounted. The traverse of the boom encompasses 360 degrees and ample reach is provided for the hoisting of loads up to 7,000 pounds. Swivel-type fairleaders for the retrieving line are mounted on the top deck at extreme fore and aft positions.

Weight distribution is such that excellent trim results and inherent stability is retained.

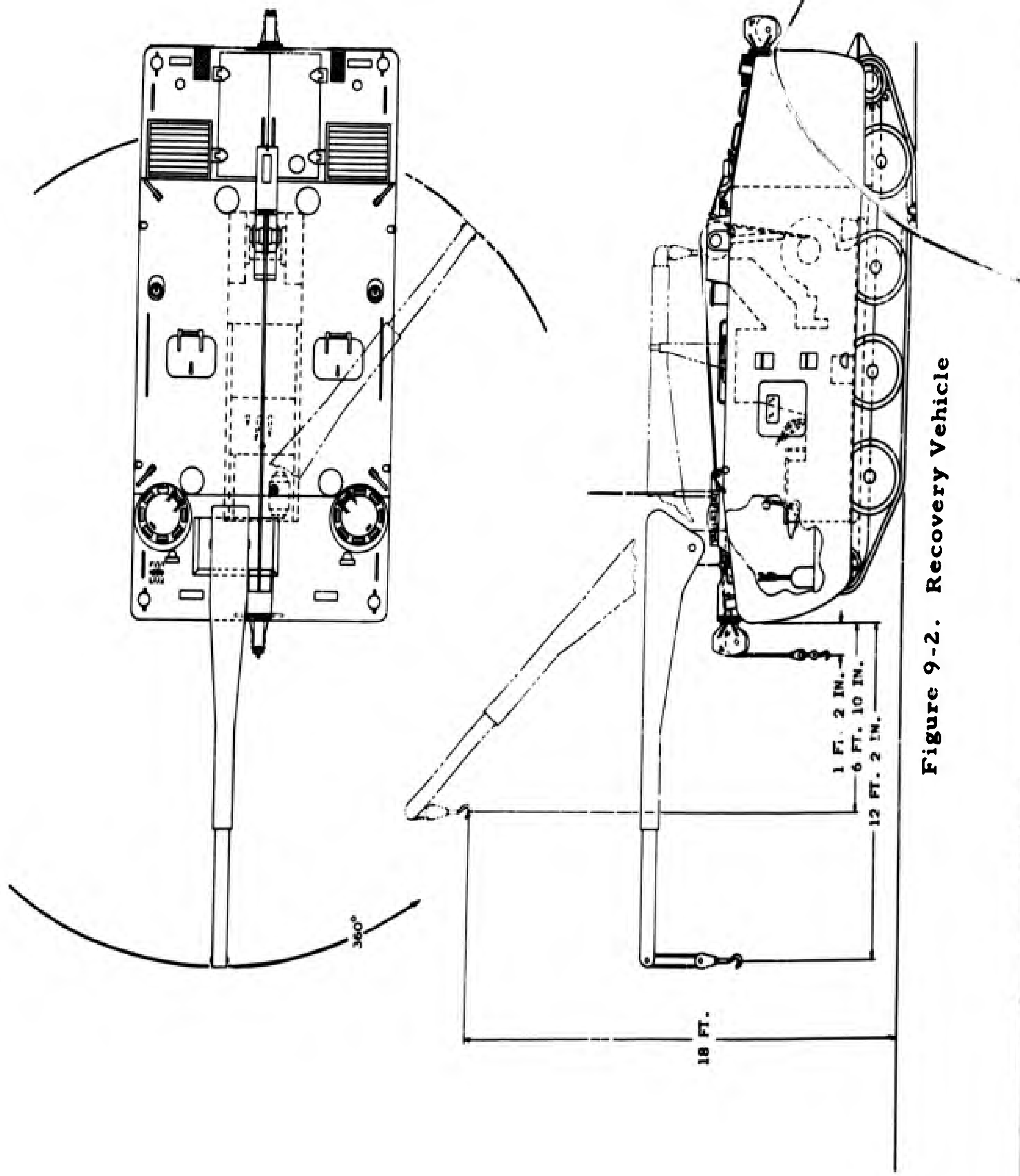


Figure 9-2. Recovery Vehicle

9.3. [illegible] Vehicle

The possibilities of installing both the Entac and Shillelagh system [illegible] MAC were investigated. However, since the Shillelagh and its installation are covered under the Artillery Vehicle description, only the Entac will be discussed here.

The Entac missile system with its lightweight and compact design can be easily installed on a multiple launching mechanism located in the cargo area of the basic vehicle. Sufficient space is available to carry the loaders, the gunner and approximately [illegible] rounds of stowed missile ammunition in addition to the missiles loaded in the launcher.

A deta.. [illegible] -cher and crew arrangement are not shown since an operational analysis and further stu[illegible] required to properly accomplish this, although from the standpoint of space and weight, no app[illegible] t problems have been noted.

Since no hydraulic or electrical power is required to actuate the system, other than the possibility of a power operated launcher elevating system and standby electrical power, the basic vehicle simplicity and reliability are retained.

9.4. Amphibian Field Artillery Weapon Vehicle.

Figure 9-3 shows the basic vehicle with a turret and 105 mm howitzer. The turret is located just forward of the vehicle center of buoyancy to provide adequate machinery clearance and simultaneously provide a level trim condition.

Here, again, the basic vehicle hull, power train, track and suspension system are retained and the equipment peculiar to this application is placed in the vehicle cargo area. It should be noted here that a lockout system may be required to provide a level gun platform during howitzer firing. If necessary, a system can be incorporated without excessive complication to the basic suspension. This has been covered briefly in Section 5.0.

Details of howitzer ammunition stowage racks are not shown, but sufficient space is available to stow a substantial amount of ammunition.

Placement of the radio antenna on the turret provides an interference free rotation of the turret and thereby minimizes the fire control problem.

The artillery vehicle is not limited to the application of the 105 mm howitzer and possibly the currently-under-development Shillelagh system may replace or supplement the 105 mm howitzer. In either case the Shillelagh 152 mm cartridge, missile, and barrel have been considered and preliminary investigation shows that it can readily be adapted to this craft. The over-all appearance should not be changed appreciably since a similar type of turret will still be required although barrel length will change.

Since this weapon system is still under development and since the general overall appearance and turret space requirements are not to different than the turret installation shown, a specific installation of the Shillelagh is not included in this study. However, it has been established that it can be installed on the basic craft without modifying the power train or track and suspension system.

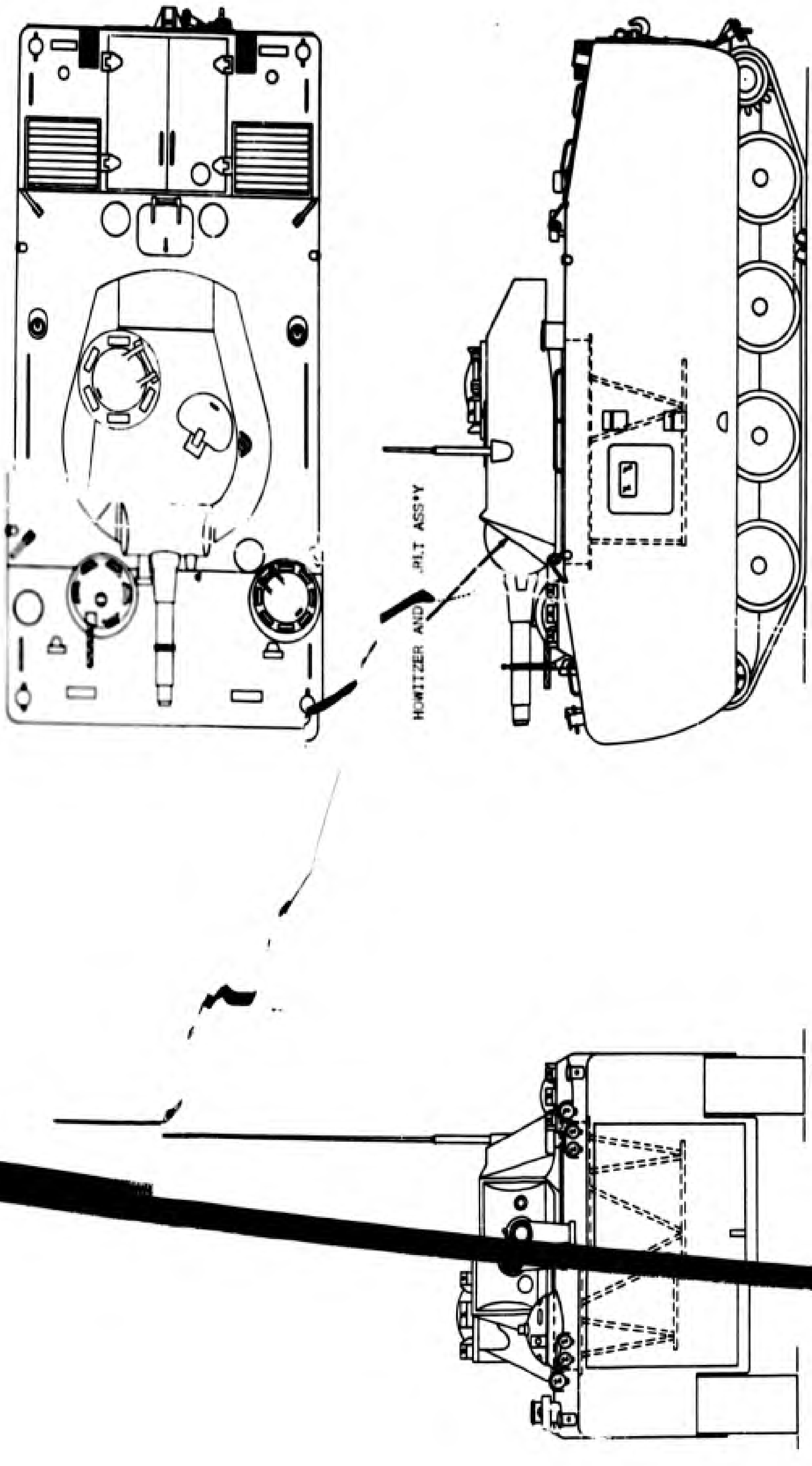


Figure 9-3. Field Artillery Weapon Vehicle

9.5. Amphibian Light Air Defense Weapon Vehicle.

Figure 9-4 shows the Mauler missile system pod installed in the cargo area of the parent vehicle.

To incorporate the entire Mauler system pod installation only limited modification of the cargo area superstructure is required. Since the pod itself includes its own power generating system, no modification in the power train, hydraulic, and electrical systems are required. Further, the basic vehicle retains the parent track, suspension system and controls.

A fuel connection between the main fuel tanks and the pod will be provided to enable sustained operation of the missile system. The diesel fuel used by the vehicle engine is compatible to the fuel requirements of the pod power generating turbine, thus enabling the interchange of fuel which results in flexible dependability of missile system operation.

From a weight and trim standpoint, this pod installation gross weight is very nearly identical to the cargo carrying capacity and. since the pod is installed in the cargo area, a level trim will result.

In general, it appears that the basic arrangement of our proposed vehicle concept is well adapted to accept this missile system.

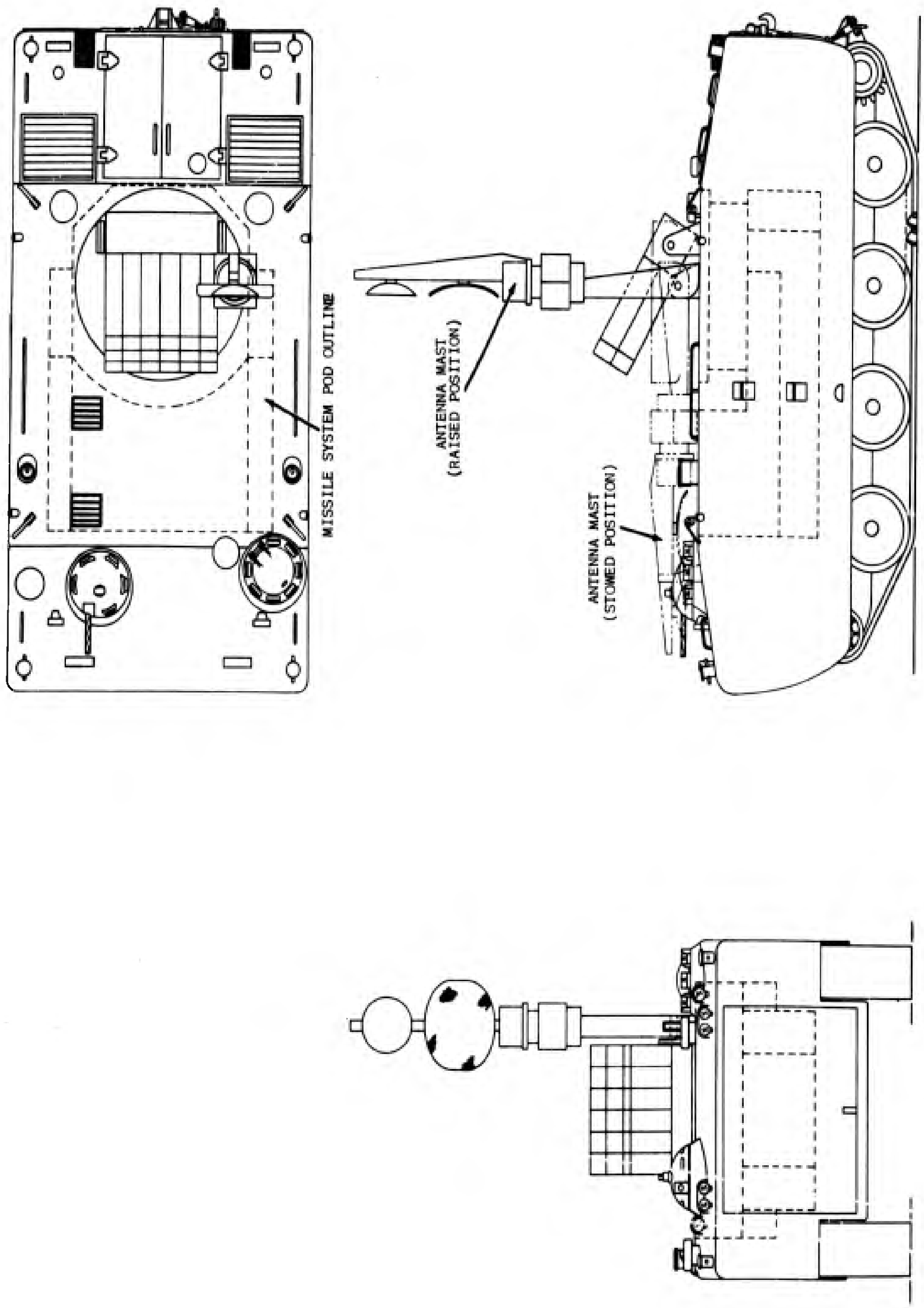


Figure 9-4. Light Air Defense Weapon Vehicle

9.6. Engineer Mine Clearance Vehicle.

The engineer vehicle requirements are (for purposes of this study) considered to parallel the LVTE1 which includes a mine excavator blade and a line charge pallet lifting mechanism and launcher. The general arrangement of this combination is shown on Figure 9-5.

Basically, the blade is designed to include sufficient displacement to provide a positive righting moment while water-borne. The pallets and launching mechanism are located in the cargo area of the vehicle.

It is recommended that, for this vehicle application, a lockout system be incorporated in the suspension system to provide an optimum excavating platform. The lockout system has been described in more detail in Section 5.0.

In general, this basic vehicle lends itself very well to the incorporation of the specific equipment required to perform the engineer vehicle requirements.

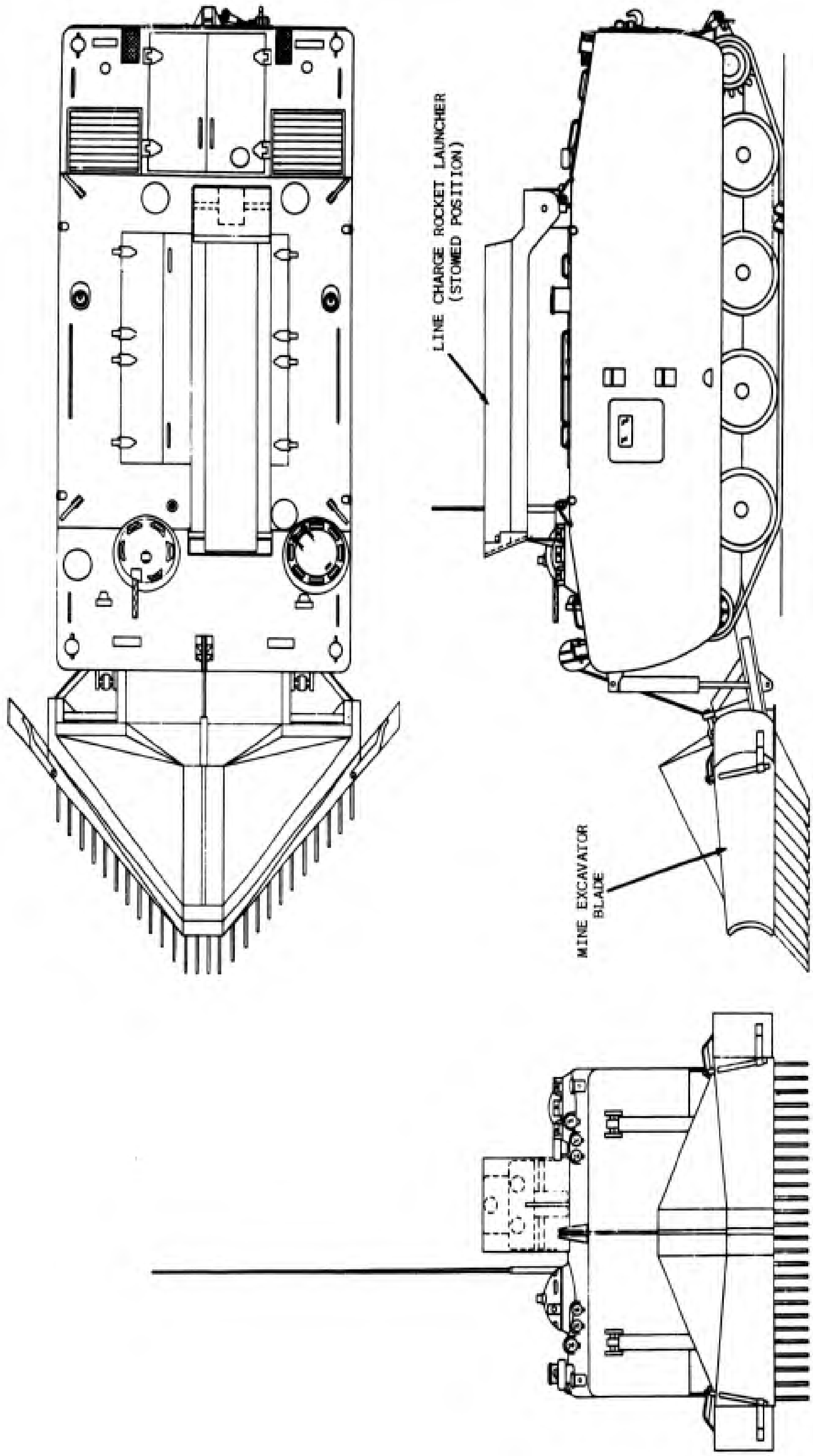


Figure 9-5. Engineer Mine Clearance Vehicle

10.0. ELECTRIC DRIVE SYSTEMS

10.1. General Drive System Characteristics.

The electric drive power system has been utilized in the past for many vehicular applications. One of the reasons for its wide spread use is that the series wound electric motor incorporated in the majority of vehicle drives to date has speed-torque characteristics which come close to satisfying the ideal requirements. The second reason is transmission flexibility. The action obtained is the same as placing a slipping clutch between the power source and the driving components. No gear shifting is required and the absence of clutches eliminates clutch wear problems. Inherent characteristics provide smooth, high accelerations which can be made constant to a fairly high speed, plus the ability to operate in the forward and reverse direction without providing special gearing. Performance demands of present day vehicles, however, virtually dictate the use of a reduction gear to amplify motor torque. In the final analysis then, the optimum vehicle drive will result from the ideal mating of an electric motor and a set or sets of gears.

10.2. Application to Track Laying Vehicles.

In the majority of cases since their inception, tracked vehicles have utilized a drive system which includes in each track a single drive sprocket either front or rear, to deliver power, an idler wheel or sprocket, and a number of bogie wheels to carry the load. With this drive arrangement, only one member in the system is supplying power for motivation. The remaining members are absorbing power in the process of returning the track to the driving member.

Probably the principal reason for utilizing a system such as this is the complexity which would be encountered in supplying power to the other wheels in the system through steering transmissions, drive shafts and reduction gear boxes which are the mainstay of power trains in tracked vehicles today.

In front sprocket drive systems as utilized in many vehicles, including the LVT3, and under dynamic conditions on land, slack or low tension track is encountered in the section of track between the sprocket and the first bogie wheel. The rear sprocket drive system is also utilized in many vehicles, including the LVTP5 family. The slack track in this drive system is encountered between the sprocket and the idler. With only one member in the system supplying power, the track tension under dynamic conditions appears to be unevenly distributed. The section of track on the output side of the sprocket will possess considerably less tension than that at the sprocket input. Track tension appears to increase at a given rate between the first bogie wheel and the sprocket input for vehicles with front sprocket drive, with minimum tension at the first bogie wheel. In vehicles with a rear sprocket drive, track tension appears to increase at a given rate between the idler and the sprocket input, with minimum tension at the idler.

Supplying power for motivation to the idler and possibly also to the bogie wheels, would tend to balance the system, with the result that better operating characteristics as well as increased propulsive efficiencies may be obtained. This is derived from the more even distribution of load which is obtained by placing the power transmitting equipment directly at the points of utilization, that is, the four corners of the vehicle.

Advancements in the state-of-the-art of electric drive systems indicate that electric power trains economically provide the same operating characteristics available with mechanical power trains plus many advantages which are unobtainable with mechanical drive lines. The flexibility of the electric power train will permit utilization of an electric drive motor at any point and in as many points as desired in the system without complexity.

The AC electric drive system proposed utilizes AC generators and AC induction type motors to provide a long life, low weight power train. No brushes or slip rings are required; therefore, the only parts subject to wear are the bearings. Utilization of this system precludes the use of special accessory equipment for speed control since motor speed is directly dependent on slip frequency, which in turn is controlled by the operator with an accelerator type control.

The electric motors are coupled directly to the final drives which are driving each sprocket. A motor and final drive assembly is located at each of the four corners of the vehicle which provides a more even distribution of weight, as well as improved operating characteristics.

10.3. General Description of System Types.

Many manufacturers have designed vehicle drive systems which are currently utilized in vehicles. The General Electric Company, Erie, Pennsylvania, manufactures a vehicle drive utilizing a DC series wound motor which, with its reduction gear, is mounted in the hub of a wheel, driving off the road earth moving equipment. This system is described in SAE paper 92T entitled, "Electric Drive for Off Highway Vehicles" by H. J. McLean and H. Vitt.

Westinghouse Electric Corporation manufactures a vehicle drive system incorporating AC squirrel cage induction type motors. This drive is incorporated on the GOER type vehicles currently in use.

The Jack & Heintz Company manufactures a vehicle drive system utilizing high speed, high frequency AC induction type motors. This drive system is currently being installed on an M-34 truck under Contract DA-33-019-ORD-3625 and is described in minutes of the Electric Drive System Meeting conducted at the Detroit Arsenal, Centerline, Michigan, on March 30, 1961.

The R. C. Ingersoll Research Center of Borg-Warner Corporation has conducted several studies of electrically driven vehicles. The results of their investigations are generally summarized in a paper entitled "Optimization of an Individually Motorized Wheel Drive for Ordnance Vehicles" prepared for the Ordnance Tank and Automotive Command, Detroit Arsenal, Centerline, Michigan.

R. G. LeTourneau, Inc. currently manufactures electric drive systems for many types of vehicles. The majority of these drive systems utilize a DC series wound motor which drives a reduction gear system with the motor and gearing mounted within the hub of the drive wheels.

The Curtiss-Wright Corporation is currently engaged in the design and manufacture of very high frequency AC electric systems. The generators of these systems operate in the 20,000 to 60,000 RPM speed range, which permits a very low weight to horsepower output machine.

10.4. Advantages and Disadvantages of AC and DC Systems.

The advantages of the DC system include an almost ideal speed-torque characteristic and the ability to be connected to the drive system with a minimum amount of reduction gearing required. On the other hand, since this is a low speed device, size and weight are relatively high. In addition, the commutators and brushes require a considerable amount of service.

The low frequency AC electric drive motors improve the service requirement problem since no brushes or commutators are required. Operating on relatively low frequencies again requires the use of slow speeds with the attendant high weight per horsepower.

The high frequency, high speed AC motor is also an induction type, but operates on a much higher frequency and speed, which in turn reduces the size and weight per horsepower. Since this is a self excited induction motor, no commutators or slip rings are required, which reduces maintenance problems to a minimum.

Considering the foregoing characteristics of the various types of electric drive systems, the high frequency high speed AC drive system has been selected for incorporation in the MAC vehicle because of low weight and low maintenance requirements.

10.5. Description of Type Proposed for Possible MAC Prototype.

The high speed, high frequency AC drive system as manufactured by Jack & Heintz, Inc. consists of three basic elements. These elements are the alternator, the static frequency changer, and the traction motor. A simplified block diagram for the MAC drive is illustrated in Figure 10-1.

The traction motors are three phase squirrel cage, induction type motors. These operate at 16,000 RPM and possess an operating efficiency of 89 percent. Speed reduction is accomplished through planetary gearing which is incorporated in conjunction with the traction motor, within the hub of the drive sprocket. Cooling is accomplished by an oil system which is connected to the remainder of the vehicle cooling system. Since this is an AC induction type motor, no brushes or slip rings are required, therefore the only wearing parts which will require servicing are the bearings.

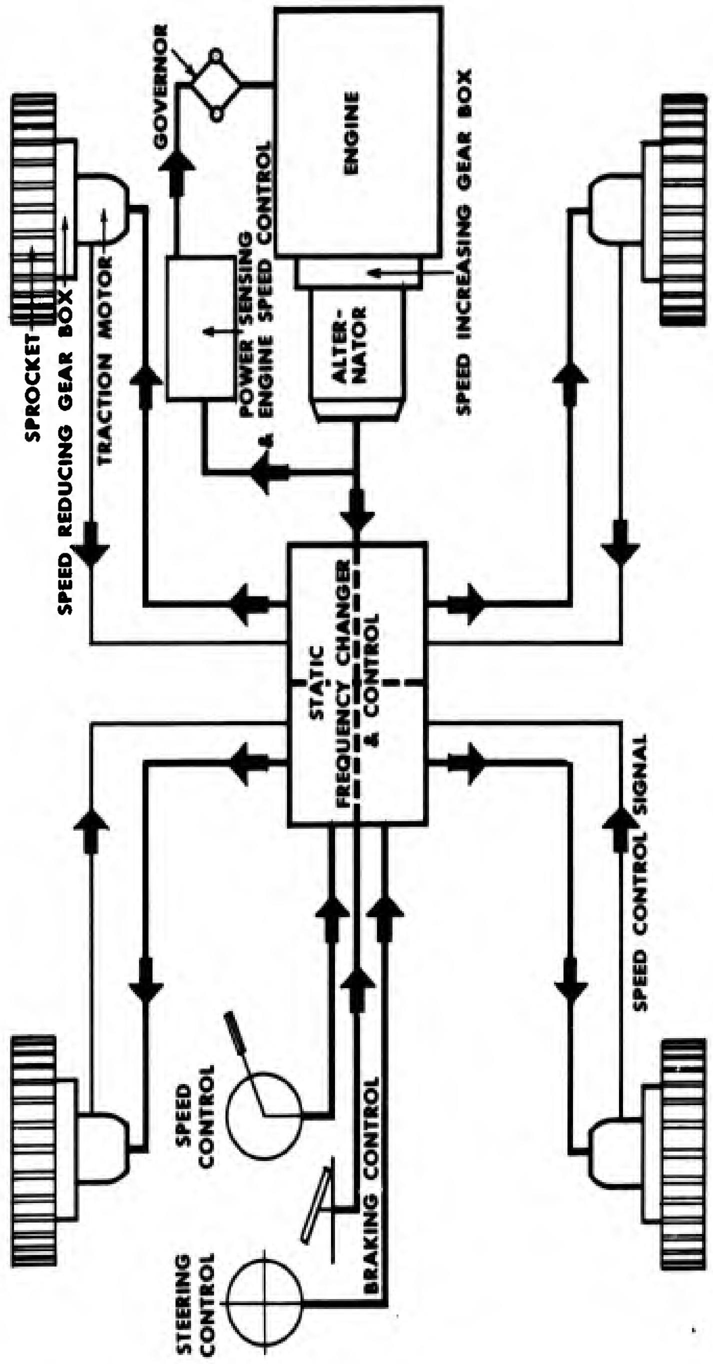


Figure 10-1. Block Diagram - MAC Electric Transmission

The alternators, which operate at an efficiency of 89 percent, are mounted directly to the engine through suitable step up gearing which increases the speed to 7500 RPM. This is a brushless alternator incorporating solid rotor construction and is illustrated in Figure 10-2. A conventional armature winding is located in the stator slots and magnetic excitation is supplied by stationary bobbin wound coils which are assembled into the end bells. The rotating member contains no windings and is comprised entirely of steel and other metallic parts. The only wearing components on the alternator are the bearings, and since no windings are present in the rotor, cooling requirements of this item are reduced. Stator cooling is accomplished by an oil cooling system.

The static frequency changer is fabricated of semiconductor elements which convert the variable frequency alternating current or voltage derived from the alternator, into a controlled frequency alternating current or voltage. It is capable of handling both real and reactive power flow in either direction, and operates at an efficiency of 96 percent. Figure 10-3 is a diagrammatical illustration of this device. It fabricates a controlled frequency wave from sections of the basic alternator output waves. In essence this device is comprised of a number of switches, usually six per unit, which are opened and closed at the proper times to produce a resultant wave form of the desired frequency. The fabrication process produces a jagged controlled frequency wave which is smoothed by means of integrally mounted chokes.

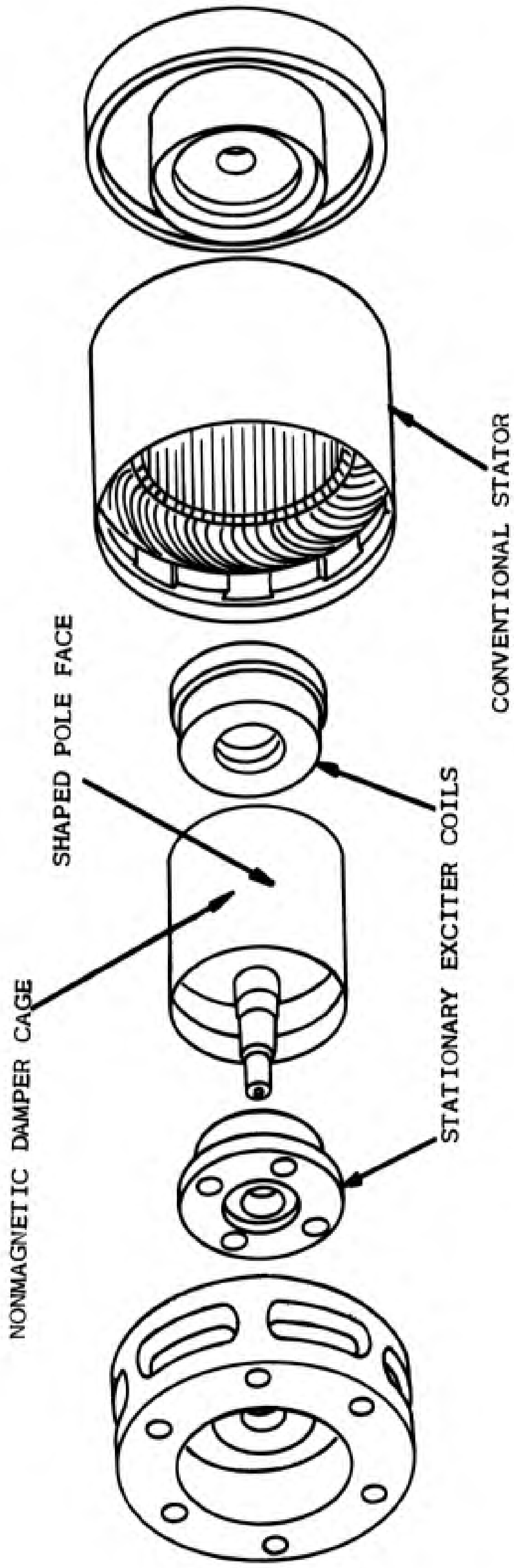


Figure 10-2. Solid Rotor Generator

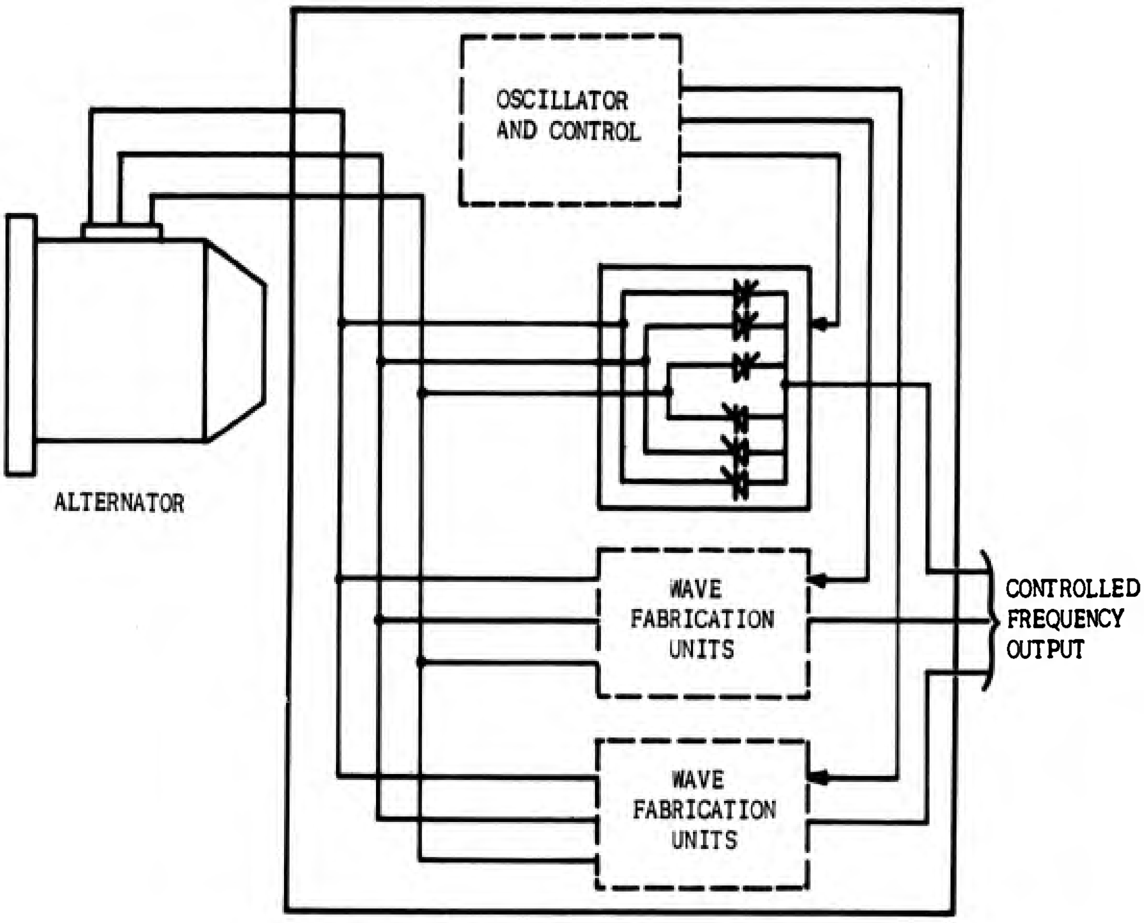


Figure 10-3. Static Frequency Changer

Figure 10-4 is an illustration of the fabricated wave. The frequency changer illustrated in Figure 10-3 consists of six groups of three phase half-wave circuits. One-half of these groups are referred to as positive groups and the other half are referred to as negative groups. The function of the positive groups is to carry current during the positive half cycle of the controlled frequency output wave, and the negative groups carry current during the negative cycle of this wave. The value of the controlled frequency is determined by the length of time both groups are conducting. This type frequency changer has been utilized for other applications under the synonym "cycloconverter." The recently developed silicon controlled rectifier is utilized to perform the switching function. Since this is a static device, an extensive operating life can be expected. A photograph of a relatively high capacity silicon controlled rectifier is shown in Figure 10-5. This component is switched on or rendered conducting, by means of a small pulse applied to the gate. Conduction ceases or the device is switched off, when the cathode current is removed by the controlling elements in the circuit. Actually this is the solid state equivalent of a thyratron or mercury arc rectifier.

The general method of operation of the static frequency changer is as follows: Considering the positive group of rectifiers shown in the wave fabrication detail of Figure 10-3 and assuming that the firing of these three rectifiers is delayed to such an extent that the average output voltage is zero, the zero point on the output voltage wave is established. If the delay in the firing of the rectifiers is now progressively decreased, the average voltage

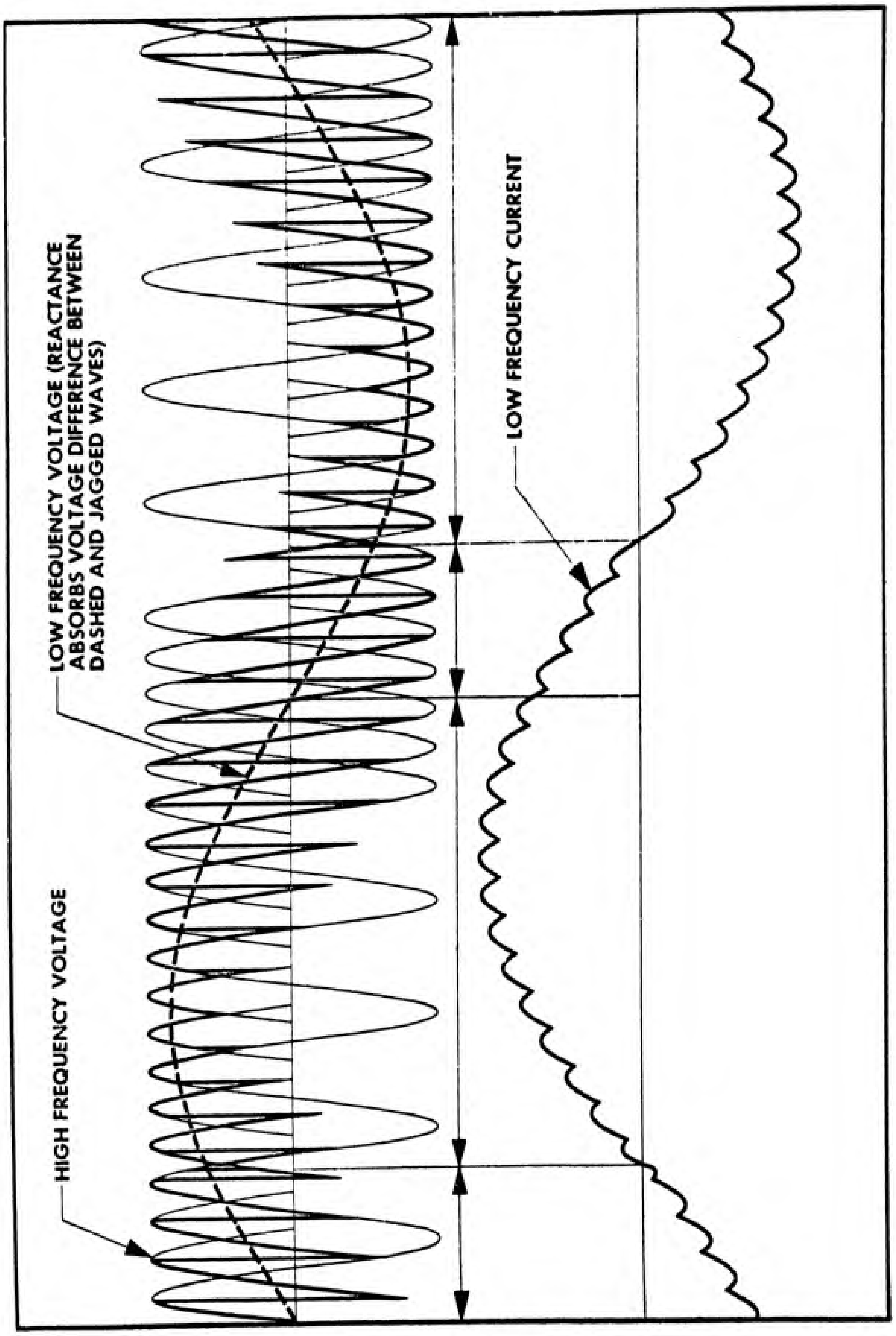


Figure 10-4. Three-Phase Half-Wave Frequency Changer

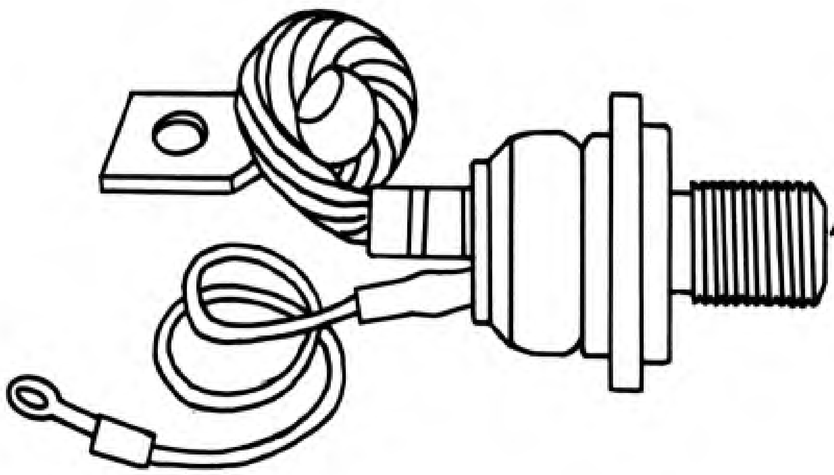


Figure 10-5. Silicon Controlled Rectifier

output of the group will progressively increase to a maximum value. This increase in output voltage is sinusoidal. Having reached the maximum of the controlled frequency voltage wave, the delay in firing of the rectifiers is progressively increased until the voltage output is reduced to zero again. At this point with a power factor of unity prevailing, the negative group of rectifiers in the same output phase commences operation and the negative half cycle of the controlled frequency voltage wave is fabricated.

Firing signals which control switching of the rectifiers are derived from an oscillator. Theoutput frequency is thus slaved to the oscillator frequency and a change in the output frequency of the changer is brought about by merely controlling the oscillator.

It is expected at this time that the static frequency changer will not require wave filtering components if only power to the induction type traction motors is required since the motor windings will act as filtering elements.

The basic motor control scheme is based on the fact that an induction motor is capable of delivering useful driving torque only if the applied frequency is slightly in excess of the equivalent rotational speed of the motor. This difference is normally referred to as slip. When the slip frequency is zero, the motor will deliver no useful torque. As the slip frequency is increased, the motor torque rapidly increases until the slip frequency reaches a given value which is determined by the design of the motor. Beyond this the useful torque of the motor returns towards zero.

The resistance and inductance of an induction motor are fixed values. Therefore it is necessary to maintain a predetermined relationship between voltage and frequency if the motor efficiency is to remain consistently high over a wide spread range. Since the voltage output of the frequency changer is a direct reflection of the voltage input to the frequency changer, the necessary voltage control is conveniently achieved by controlling the output voltage of the alternator as a function of the frequency applied to the motor.

In the control system, the command from the driver by means of an accelerator pedal, determines the degree of slip. True motor speed is determined by a signal derived directly from the traction motor. Since both of these signals are summed and the output frequency of the summation device utilized to control the output of the frequency changer, a torque controlled system, rather than a speed controlled system, is obtained. Vehicle speed control by engine drag braking or dynamic braking is accomplished with this system. By reducing the output frequency of the static frequency changer to a value below that of the equivalent speed of the rotor, the motors act as generators and the flow of power is reversed through the frequency changer. For engine drag braking, this power is directed to the alternator which now acts as a motor and tends to accelerate the engine. The torque required to overcome engine losses plus producing engine acceleration thus becomes braking torque for the vehicle. For dynamic braking, power generated by the motors due to sprocket rotation is converted into heat which is dissipated by the cooling system. The control for this braking system is also associated with the brake pedal normally utilized.

Steering is accomplished through control of the frequency changer. This places a differential frequency between the driving members of both tracks, thus permitting a differential in speed of each track. Regenerative steering is accomplished by a reduction of frequency to the motors of the inside track to a point which causes them to act as generators thus delivering additional power to the opposite track.

Alternator output frequency is not of primary interest for operation of the static frequency changer. A prime mover speed control loop is utilized to detect alternator output power to control prime mover speed. The output frequency of the alternator will vary directly with engine speed. However, other than in the alternator design parameters, which include consideration of the prime mover speed range, no alternator frequency control is required. The frequency of the alternator is in the order of 2000 cycles per second at a prime mover speed corresponding its maximum horsepower output. This concept provides an arrangement where no fixed relationship between vehicle speed and prime mover speed exists, permitting the engine to operate on the maximum economy profile of the fuel consumption curve for any given vehicular power requirement.

Figure 10-6 illustrates the performance data of a 200 HP class diesel engine plotted to show the speed-horsepower relationship under the condition of constant fuel flow. It may be seen that the horsepower output of the engine will vary over a considerable range for a given constant fuel flow. It may also be seen that for a given horsepower, as much as a 25 percent increase in fuel flow can be expected merely by permitting the speed to significantly depart from the maximum efficiency operating point.

The engine speed versus load program followed by the electric drive system is also illustrated in Figure 10-6. The engine characteristics are relatively flat in the maximum horsepower region, so attaining high fuel economy is not dependent on a highly precise governor.

10.6. Installation in MAC.

Figure 10-7 illustrates the high speed, high frequency drive system installed in the MAC vehicle. A motor-reduction gear assembly is installed in each of the four drive sprockets. Placing a power conversion system thusly at each of the four corners of the vehicle provides a more even distribution of weight as well as lighter power transmitting components since each drive sprocket need only produce 25 percent of the total required power.

The alternators are driven by a step-up gear box, and are mounted directly under the engine. Two alternators are utilized for this application due to the availability of existing components and tooling. These alternators are a

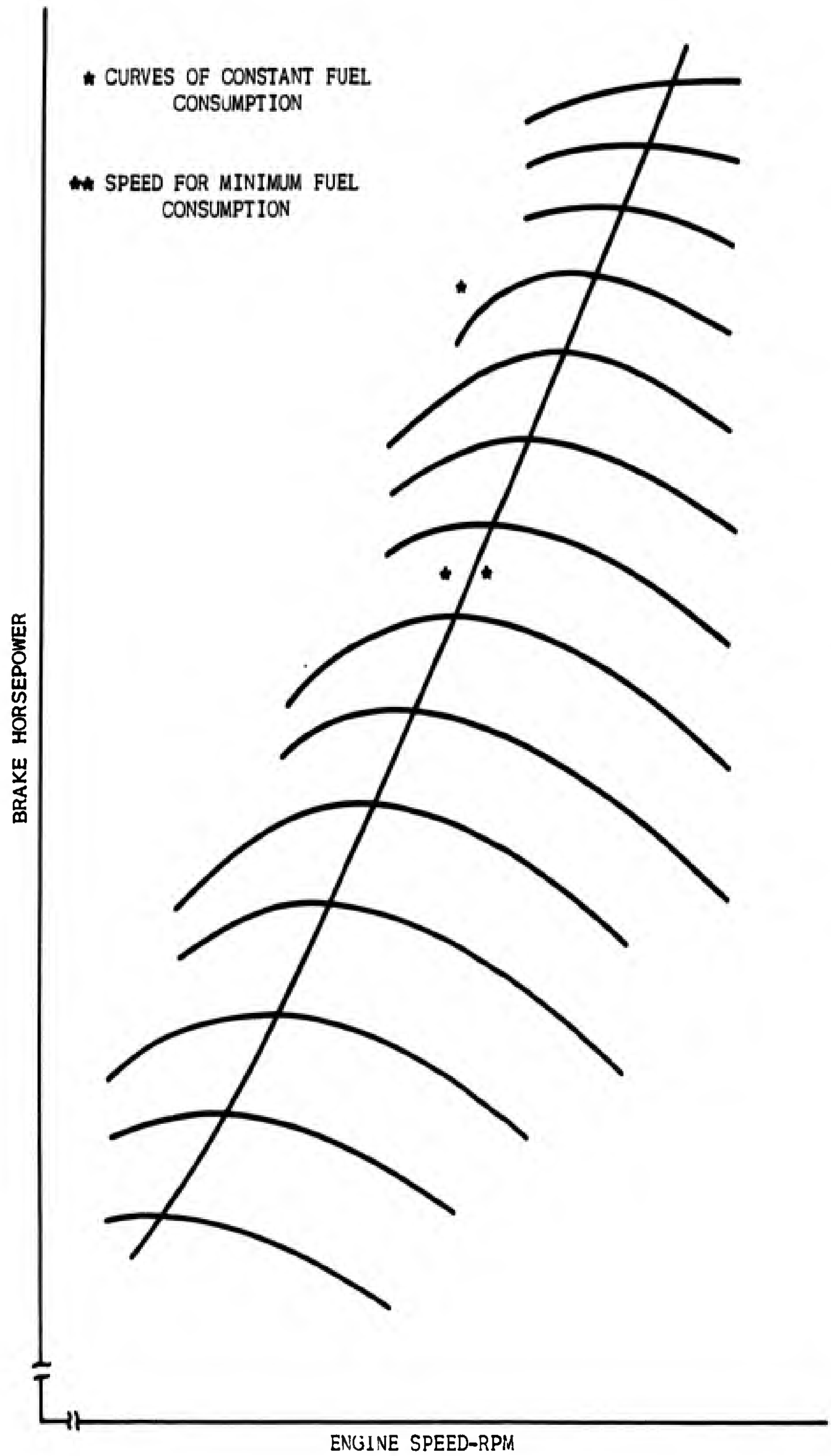


Figure 10-6. Fuel Consumption Curves

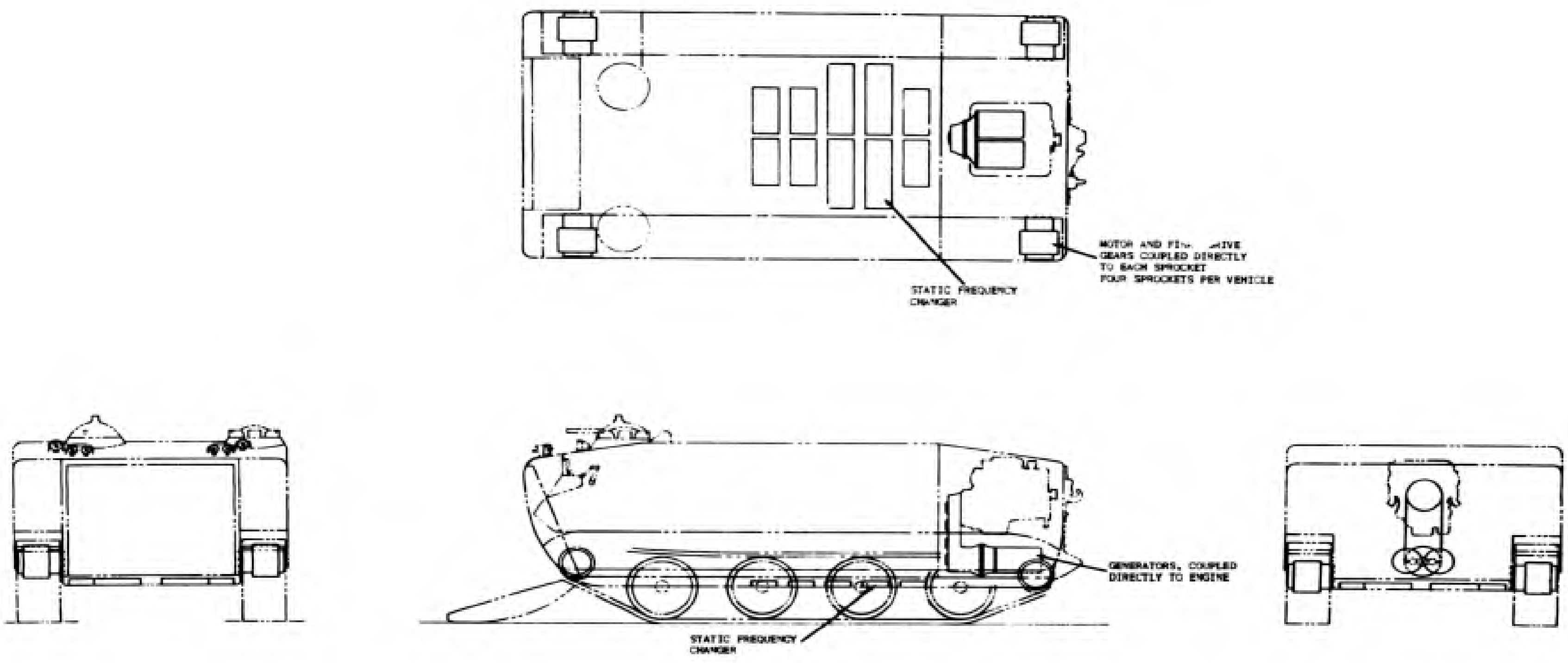


Figure 10-7. Electric Drive Layout in MAC

brushless version of the well-known salient pole machines. The generating portion of these machines is a salient pole alternator while the exciter portion has been changed from a DC generator to a rotating-armature alternator-rectifier combination. The output of the rotating armature is directed to a solid state rectifier arrangement which is assembled to the rotor. The DC output of the rectifier is thus available for salient pole excitation without resorting to the use of brushes in the system.

Control circuitry is provided to permit operation of the two alternators in parallel. They will, in effect, feed power into a common bus arrangement, with the controls assuring proper sharing of the load by both machines. Alternator specifications are as follows:

Speed (at maximum engine speed)	7500 RPM
Alternator frequency at 7500 RPM	2000 cps
Cooling	Oil
Diameter	14-1/2 in.
Length	21 in.
Weight	205 lbs.
Efficiency	89 percent

The drive motors are mounted in the sprocket hub in conjunction with the reduction gearing. Since these motors operate at 16,000 RPM and the required sprocket speed for a pitch diameter of 21.3 inches is 631 RPM, the gears provided will possess a ratio of 25.36:1. It is contemplated that planetary type gearing will be utilized to produce the ratios required. The oil required for lubrication will also provide cooling of the motor. The motor specifications are as follows:

Motor Speed (Maximum)	16,000 RPM
Cooling	Oil
Weight	160 lbs.
Efficiency	89 percent

The static frequency changer utilized will be mounted in the bilge area of the vehicle, below the deck. Since the components of this device consist mainly of semiconductors, packaging requirements are very flexible. For this reason the bilge area was selected for locating this device since the package can be designed to fit in presently unused space. It is anticipated at this time that supplemental cooling of the frequency changer will be unnecessary in the configurations illustrated. However, by providing supplementary oil cooling a more compact arrangement could be utilized. The specifications for this device are as follows:

Cooling	Conduction
Weight	420 lbs.
Efficiency	96 percent

10.7. Mobile Power Source.

Since the electric generating capacity of the electrically driven vehicle is of the order of 300 KVA the vehicle could become a mobile power source for supplying electrical loads in the field. To accomplish this, additional controls would be required to maintain a constant frequency output. However, the increase in the number of components required would be of a minor nature considering the equipment which would be necessary to supply power for special equipment in the field, if this equipment were separately provided.

10.8. Recommendations.

It is recommended that one of the prototype vehicles constructed be equipped with an electric transmission as described above.

Testing of this vehicle can then be accomplished concurrently with the other prototypes to provide an accurate comparison of vehicular performance.

11.0. HYDROSTATIC DRIVES

Since the time of World War I the steering of tracklaying vehicles has been studied in detail. Great improvements in track and suspension have taken place since that time, making possible higher road speeds and the utilization of higher engine horsepower. As the speed of tracklaying vehicles increases, the importance of vehicle controlability and more efficient power transmission becomes more evident, necessitating careful consideration of the problems of steering mechanism design. If the design of the ideal steering mechanism were simple and straightforward, and well defined by performance requirements, as well as being mechanically feasible, we could be sure that such a mechanism would have been brought into being by now, and that all tracklayers would now employ such a mechanism. Since there are now various steering systems in use, we must assume that the problem is not so simple, and that all the steering mechanisms are a compromise between what is needed and what is possible to provide economically.

Regardless of the type of power transmission system, or steering device, the tank is propelled by the action of the tracks on the ground, and to steer the vehicle, it can easily be seen that the inner track must move, relative to the hull, more slowly than the outer one. If the tank is moving in some curved path, the tracks, both inner and outer, are moving with the same angular velocity as the tank hull. This means that the tracks, as a whole, must skid angularly over the ground; but the track length on the ground may be long, and the vehicle weight and coefficient of friction may both be high, so that the forces required to deflect the vehicle from a straight path are large in comparison with those needed to propel it.

The motion of the vehicle in a curved path requires a slewing couple on the vehicle. This couple must be produced by a difference in the track forces. The action of a steering device is to superimpose upon the propulsive forces, additional forces which will cause the required slewing couple. In the case where a sustained turn is to be provided, the steering forces delivered to the tracks should be equal and opposite in direction so that no decrease in vehicle speed will be encountered. At the same time the inner track should be slowed down and the outer track speeded up by the same amount. If these requirements are not met, the tank will decelerate during the turn, which may be objectionable.

The theory shows, and is substantiated by test, that at the instant of starting a turn, the outer track must be provided with a very large positive force. The horsepower delivered to the outer track is very high at this point. At the same time, the inner track must be provided with a very large negative, or retarding force. It continues to move, however, in a positive direction. The inner track is then generating a large amount of negative horsepower. If there are no means possible for utilizing this horsepower, it must be absorbed by a brake and dissipated as waste heat. It is in the manner of handling the large track horsepowers that the various steering mechanisms differ.

Power delivery is a most important item and increases in scope with an increase in vehicle speed. Hence, the importance of low power losses during steering is less critical on a slow moving vehicle than in a fast moving one but important in all.

The ideal steering transmission would take all these varied forces, integrate them efficiently and force the track sprocket to respond to a given signal from the operator under all operating conditions. The hydrostatic transmission as proposed herein, is designed to meet this ideal steering transmission arrangement. This transmission, except for power transmission losses, will not dissipate power as it has no brakes for steering and therefore will deliver the maximum power to the track sprocket. Torque reversals due to reverse track loads would result in a circulation of power from one track sprocket to the other. This type of system is called a regenerative one.

A second important feature of this transmission is that the sprocket speeds are controlled independently of sprocket loads.

A third characteristic is that a continuous power train connection is made and this is the most important factor in eliminating reverse or uncontrolled steering as well as in maintaining mobility under difficult conditions of traction.

How close this proposed hydrostatic transmission approaches an ideal performance can be evaluated by examination of the control mechanisms. These details may be of extreme importance in the controllability of the transmission.

Although a simple, reliable production-type tracked vehicle should adhere to proved design, new concepts sometimes outmode older designs causing costly field obsolescence.

The tracked amphibian personnel assault cargo carrier vehicle transmission study and evaluation indicated that serious consideration should be given to the possible incorporation of a hydrostatic type of transmission drive.

11.1 The Hydrostatic Drive.

Basically, a hydrostatic transmission is a hydraulic circuit or system consisting of positive displacement pumps and positive displacement motors, with interconnecting lines or passages. The pumps or motors, or both, enable variable-volume delivery, which is usually accomplished by varying the swash plate angle. A control system is provided, which uses either or both hydraulic valves and cams to operate the swash plate. There are many possible configurations, the choice depending on the requirements of the operating conditions. Each vehicle must be evaluated individually for tractive effort, speed, efficiency, and cost.

11.2 Investigation of Hydrostatic Drives.

During the past several years, the contractor has investigated various types of hydrostatic drives. The following companies are among those actively engaged in testing their designs in off-the-road and military vehicles:

B. F. Goodrich Company

Clark Equipment Company

Sundstrand Aviation Division of Sundstrand Corp.

Staffa Works, Double A Division Distributor

Watertown Division of New York Air Brake Co.

Stratos Division, Fairchild Engine and Airplane Corp.

General Electric Company

The Denison Engineering Company

Pesco Division, Borg-Warner Corporation

Gar Wood Industries, Incorporated

The Marquardt Corporation

The Electric Products Company

Dynex, Incorporated

Lucas-Rotax, Limited

Ingersoll Kalamazoo Division, Borg-Warner Corporation

Although the above list shows only a sampling of companies actively engaged in the research, design and testing of components and systems for hydrostatic transmission drive systems, it does point out the intensity of effort being expended to achieve a satisfactory hydrostatic drive system.

11.3 Summary of Hydrostatic Drive Systems.

This summary covers all of the known currently feasible hydrostatic drive systems which are available and/or which have been proposed for use as a vehicle drive system. The drive systems listed are all equivalent to, and in many cases are superior to, currently used or proposed mechanical vehicle drive systems (trains) which handle comparable horsepower.

Because of the extremely versatile nature of the hydrostatic drive, no attempt has been made in this summary to list applications of the various drive systems to specific vehicles or types of vehicles.

11.4 Classification.

For purposes of this summary, hydrostatic drive systems have been divided into two major types: namely, integral drive system and split drive system. The types are defined as follows:

Integral drive systems are systems in which the hydraulic power pump(s) (the input) and the hydraulic drive motor(s) (the output) are contained in a common case or framework and the hydraulic fluid conduits are an integral part of the supporting case or framework. Drive system controls may be integral or separate or any combination of these arrangements.

Split drive systems are systems in which the hydraulic power pump(s) and the hydraulic drive motor(s) are contained in separate cases or housings. The pump(s) or motor(s) may be mounted remotely from one another. The units are connected together by separate conduits which carry the hydraulic fluid.

The hydraulic drive systems listed in the chart in Table 11-1 are further broken down into the classes A, B, C, etc., and arrangements 1 and 2, as outlined below:

Class A - Variable-displacement drive pumps and variable-displacement drive motors.

Class B - Variable-displacement drive pumps and fixed-displacement drive motors.

Class C - Fixed-displacement drive pumps and variable-displacement drive motors.

Class D - Fixed-displacement drive pumps and fixed-displacement drive motors.

Arrangement 1 - Single-ratio output shaft gear box.

Arrangement 2 - Multiple-ratio output shaft gear box.

11.5 Brief Remarks on Investigated Drives.

B. F. Goodrich (Barnes & Reinecke).

The drive unit has a four-speed range box between the hydraulic drive motor and output shaft. Speed range is changed by oil-actuated clutches. Hydraulic pumps and motors and gear box clutches are current production items.

Present status: An experimental test unit incorporating this system was supposed to have been built and tested by September 1961, but the current status of this test is unknown.

The maximum system operating pressure is 5000 psi.

Table 11-1. Summary of Hydrostatic Vehicle Drive Systems

	Manufacturer	HYDROSTATIC DRIVE MANUFACTURED																REMARKS
	TYPE	I								II								
	CLASS	A		B		C		D		A		B		C		D		
	ARR.	1	2	1	2	1	2	1	2	1	2	1	2	1	2	1	2	
1.	B. F. Goodrich (Barnes & Renecke)											O	A					See Item (1)
2.	Electric Products Company															A	O	See Item (2)
3.	Denison									A	O	A	O					See Item (3)
4.	Dynex, Incorporated													O	A			See Item (4)
5.	Gar Wood Industries									A	O							See Item (5)
6.	General Electric Company			A						A	O							See Item (6)
7.	Marquardt Corporation									A	O							See Item (7)
8.	New York Airbrake									A	O							See Item (8)
9.	Borg-Warner Corporation	A	O															See Item (9)
10.	Staffa											A	O					See Item (10)
11.	Stratos Division Fairchild	A	O									A	O					See Item (11)
12.	Sundstrand Aviation	A	O	A	O					A	O	A	O					See Item (12)
13.	Clark Equipment											O	A					See Item (13)
14.	I.K.D. Borg-Warner	O	A															See Item (14)

A - Indicates hardware currently in production or on which design is complete.

O - Indicates hardware that can be made available by modification of existing hardware or design.

Electric Products Company.

This drive utilizes fixed displacement units. They are arranged in a unique manner so that the characteristics of the drive system are the same as those of a system using variable displacement units through a range of 50% or more of the system capacity.

Hardware for this system is based on a design which has had many hours of use in service.

Present status: Prototypes of some components of this system have been built and are being tested. The completed system is still a paper study.

Maximum system operating pressure is 2000 psi.

Denison Engineering Company.

This company makes pumps and motors which are used by other manufacturers (B. F. Goodrich & Clark) in hydrostatic drives for vehicles. The company also produces hydrostatic drives for large industrial and military applications, such as gun turrets, variable speed drives, etc. but does not at the present time produce a drive suitable for vehicular use.

Equipment is available for system pressures to 5000 psi.

Dynex, Incorporated.

The current drive design utilizes two hydraulic drive motors which drive a two-speed range box between motors and output shaft. Power output is changed by staging hydraulic drive motors in various sequences with each other and with the two ratios of the range box, to provide six separate drive ranges. Hydraulic components are standard Dynex units or are based on existing Dynex designs.

Current drive unit was designed in conjunction with Westinghouse-LeTourneau.

Present status: Aside from the standard Dynex units incorporated in design, this drive system is still in the paper-study stage.

Maximum system operating pressure is 6000 psi.

Gar Wood Industries.

This system utilizes variable displacement hydraulic motors which drive the output shaft through a single reduction gear box. Hydraulic power for the drive motors is furnished by a variable displacement pump, driven by prime mover. The system can be arranged for dynamic braking if desired. System incorporates standard components which are in production, and it is available in a range of sizes. All central components for the system have been designed and built. Components are primarily aluminum for light weight.

Present status: An experimental test vehicle which uses this drive system has been built and tested. The test vehicle was demonstrated at Fort Belvoir, Virginia on September 20-21, 1961. Equipment is available for adaptation to tracked vehicle operation.

Maximum system operating pressure is 3500 psi.

General Electric Company.

This company's system utilizes variable displacement hydraulic motors which drive the output shaft through a single reduction gear. Hydraulic power for drive motors is furnished by a variable-volume pump driven by prime mover. System utilizes ball-piston pumps and motors instead of the axial-piston type used in other systems (except the Electric Products system already discussed). The ball-piston unit is capable of operating at very high speeds. Component design for this system is based on components used in aircraft hydrostatic transmissions. For a given horsepower capacity, components in this system are generally smaller and lighter than those in systems using axial pistons or other-type components.

Present status: Prototype components in the 200-300 horsepower range were to be built and tested by fall of 1961 but current status of the program is unknown.

Maximum system pressure is 2500 psi.

Marquardt Corporation.

Marquardt system utilizes variable displacement motors which are directly connected to the output shaft. Hydraulic drive motors are the radial-piston type unique design. A variable-displacement, axial-piston type furnishes hydraulic power to the drive motors.

The unique design of the hydraulic drive motor enables it to develop large torques in six relatively small packages. Design requirements make the weight of this system higher than that of other comparable units.

Present status: Small prototype drive units were to be produced by fall of 1961, but the current status of this design is unknown. Design studies of both system and components have been extensive, and several patents have been taken out on components. Completed system is still in paper study stage.

Maximum system operating pressure is 5000 psi.

New York Air Brake Co., Watertown Division.

This system utilizes variable- or fixed-displacement hydraulic motors which are powered by a variable-displacement pump. The system is generally furnished without any gearing on the hydraulic drive motors. System was first used to drive a commercial lift truck, and about two hundred such units a year are produced. In the lift truck application, the hydraulic drive motor furnishes power to a standard differential and axle arrangement.

Present status: Hydraulic pumps and motors are currently in production. Present pumps and motors are limited to fifty or sixty horsepower. Larger capacity units are reportedly in the design stage at the present time.

Maximum system operating pressure is 5000 psi.

Borg-Warner Corporation.

This drive system utilizes variable-displacement drive motors and variable-displacement power pumps. Units are designed specifically for application to tracked and wheeled vehicles and are intended to replace conventional gear and power shift drive systems.

Present status: Prototype units have been built and are presently undergoing extensive testing. Test results have been very good to date. Prototype units for vehicle installation can be ready in the near future if required. Production units will be available at a later date.

Maximum system pressure is 3000 psi.

Staffa.

Staffa has large fixed-displacement units. These units are of the radial-piston type, and drive motors are generally connected directly to the driven load without any gear reduction.

Present status: Units currently in production are low-speed, high-torque units. These units develop higher shaft torques for their operating speeds and pressures than any other units used in vehicle drive systems. These units are currently being used by other manufacturers in vehicle drive systems, but this company itself does not build a complete vehicle drive system.

Maximum system operating pressure is 3000 psi.

Stratos Division, Fairchild.

This system utilizes both fixed- and variable-displacement motors in various combinations.

Present status: This company has produced a type I prototype hydrostatic drive system for a truck and a tracked vehicle. It also has a design for a type II vehicle drive system, but this system is still in the paper study stage.

Maximum system operating pressure is 4500 psi.

Sundstrand Aviation Division.

This company uses hydrostatic drive systems utilizing both fixed- and variable-displacement pumps and motors in various designs of type I and type II drive systems. Applications of both types of drive systems have been made to both tracked and wheeled vehicles.

Present status: Components are currently in production for both type I and type II drives. Horsepower capability of production units is limited especially in the type II systems. Development work is being done on a new crosshead-type pump design, but units are not yet in production for high-horsepower components. Drive systems having 300 hp capability and utilizing the crosshead-type pump and motor have been proposed, but to date, no hardware in this class has been built.

Maximum system operating pressure is 6000 psi.

Clark Equipment Company.

This system utilizes fixed-displacement hydraulic motors which drive the output shaft through a two-speed range box. The range box utilizes oil-operated clutches to change the gear ratio.

Present status: As far as can be determined, no complete drive system has been proposed by this company, and design work has been done only on a drive unit. The design of the unit is very similar to that of the B.F. Goodrich unit, except for the number of gear ratios. To date, no hardware has been built, and the program is still in the paper stage.

Maximum system operating pressure is 5000 psi.

IKD, Borg-Warner Corporation.

This company has proposed a type I drive capable of handling 120 hp. The drive incorporates variable-displacement pumps and motors and a two-ratio range box at each of the two output shafts. This drive was designed for minimum weight (approximately 300 pounds dry) and for use in a track-laying vehicle.

Present status: This drive system has been proposed and is now under consideration by the military. Prototypes of the hydraulic pumps and motors intended for use in this drive system have been built and tested.

Maximum system operating pressure is 3000 psi.

Lucas - Rotax, Limited.

This company builds both fixed- and variable-displacement hydraulic pumps and motors. It does not, itself, market a complete hydraulic vehicle-drive system as a package, but does build components for such systems and will furnish consulting engineering service to system manufacturers.

It is reported that Lucas-Rotax has furnished pumps and motors for a four-sprocket-drive tracked vehicle built by one of the U.S. military agencies and now undergoing tests.

Lucas-Rotax is a Canadian Company. Its parent company, Lucas Industries, is located in England. Lucas-Rotax has built equipment for the U.S. Air Force and other U.S. government agencies.

Maximum system pressure is 5000 psi.

11.6 Order of Company Preference.

The order of preference was based on the following evaluations:

1. Years of experience in the pump and motor field.
2. Present-day technical progress and strength of organization.
3. Hardware availability within a year.
4. Production methods previously used.
5. Eagerness in accomplishing the work.
6. Relative cost of production units.

The following order of preference was based not only on presently known information, but also on the probability of what might be available in a year's time. Since the intensity of effort being expended in this field by various manufacturers is tremendous, there will be rapid changes from time to time in field leadership. Each manufacturer has definite possitilities as an alternate source during and after the prototype program.

Sundstrand Aviation Division of Sundstrand Corp. was given first preference because they have been continuously engaged in hydraulic research and development work for the government for more than 15 years. They have developed and applied successful hydrostatic drive systems. They have furnished the government with production-grade components that were reliable and reasonable in cost. They have shown eagerness and have given all necessary data to substantiate their claims. Although they proposed an integral transmission for this bid, they could just as readily have shown a split system, as Gar Wood Industries did. Their prototype cost is a little higher, but it is still fair. Their production costs would be lower than the rest of the companies.

Gar Wood Industries was given second preference, based on the working system they demonstrated to us and the government on a "Goer" type vehicle. However, their lack of eagerness and their reluctance in furnishing substantiating data will present difficulty when government drawings must be furnished by the vehicle manufacturer. To the best of our knowledge, Gar Wood is the only company that has available components today. The cost of their units is very reasonable. Further improvements of their units are necessary, however, for achievement of some degree of efficiency.

The combination of Lucas-Rotax, Limited, and Borg-Warner Corporation is indicated as third preference because Lucas-Rotax alone will furnish only pump and motor hardware, and not a complete hydrostatic system. Lucas-Rotax presently has motors and pumps available. The Roy C. Ingersoll Research Center of Borg-Warner has the technical facilities for an optimum control system. The Pesco and Wooster Divisions of Borg-Warner have excellent facilities for manufacturing components, and the Ingersoll Kalamazoo Division of Borg-Warner has excellent facilities and experience for assembly and installation relative to a vehicle. This combination could furnish a complete system at reasonable cost to the government.

General Electric Company was chosen fourth, primarily because of the high cost of their units. They are presently testing pump and motor units of sufficient size for this vehicle. They have shown eagerness in furnishing data to substantiate their claims, but these data were only paper calculations, and not results of actual tests. Their primary asset is low weight per horsepower and high speed range. However, they must hold their pressures down in order to achieve some degree of efficiency.

Watertown Division of New York Air Brake Company was chosen fifth, primarily because they have been actively engaged in producing a hydrostatic drive system, even though it was for vehicles of lower weight, speed, and torque characteristics. They indicate that they are working on larger units, but they have been extremely reluctant about furnishing substantiating data. Their present costs are high.

Dynex, Incorporated has been indicated as our sixth choice primarily because they are relatively new in the field. They did considerable work, however, on the Mobile Assault Bridge-Ferry bid proposal. The prices quoted were for development of pumps and motors. The insufficiency of data received to date prevents any conclusion on their providing a system, except with respect to development costs.

Based on the preferences we have given Sundstrand Aviation Division and Gar Wood Industries, these companies have submitted design data to substantiate their claims of being able to provide a tracked vehicle, as proposed, with a reliable hydrostatic drive system. In order to compare the essential merits of an integral versus a split-type system, the two proposals are presented in detail in paragraphs 11.7 and 11.8.

11.7 Sundstrand Aviation Division Proposal of Hydrostatic Transmission for LVT Vehicle.

Sundstrand has proposed an integral drive system in which the components, such as pumps, motors, conduits, and valves, are enclosed in a common case. However, this company has produced split drive systems and can just as readily furnish one for this vehicle, if it is desired. Their proposal is as follows:

> Note: The Sundstrand proposal discloses information and data in which the Sundstrand Corporation has proprietary rights. Such information and data is disclosed in confidence solely for the purpose of the proposal. It must not be disclosed to others or otherwise used to the detriment of the Sundstrand Corporation. Neither receipt nor possession of said information confers or transfers any right to reproduce, use or disclose, in whole or in any part, any such information and data for any other purposes, except with written permission of the Sundstrand Corporation.

11.7.1 Features.

The hydrostatic propulsion system is a new concept in automatic transmission systems for ground vehicles. With it, starting and stopping, forward and reverse, vehicle steering, infinitely variable torque control, and dynamic braking are possible with a minimum number of operator controls and without gear shifting and clutching. The hydrostatic transmission makes possible control, performance, and maneuverability never before attainable in ground vehicles. Other unique features of the hydrostatic transmision are as follows:

a) Allows full engine power to be delivered to the vehicle driving members throughout the entire speed range of the vehicle, with exceptionally high torque multiplication at low speeds.

b) Eliminates the necessity of drive axles and shafts, differentials, torque converters, clutches, and other mechanical components.

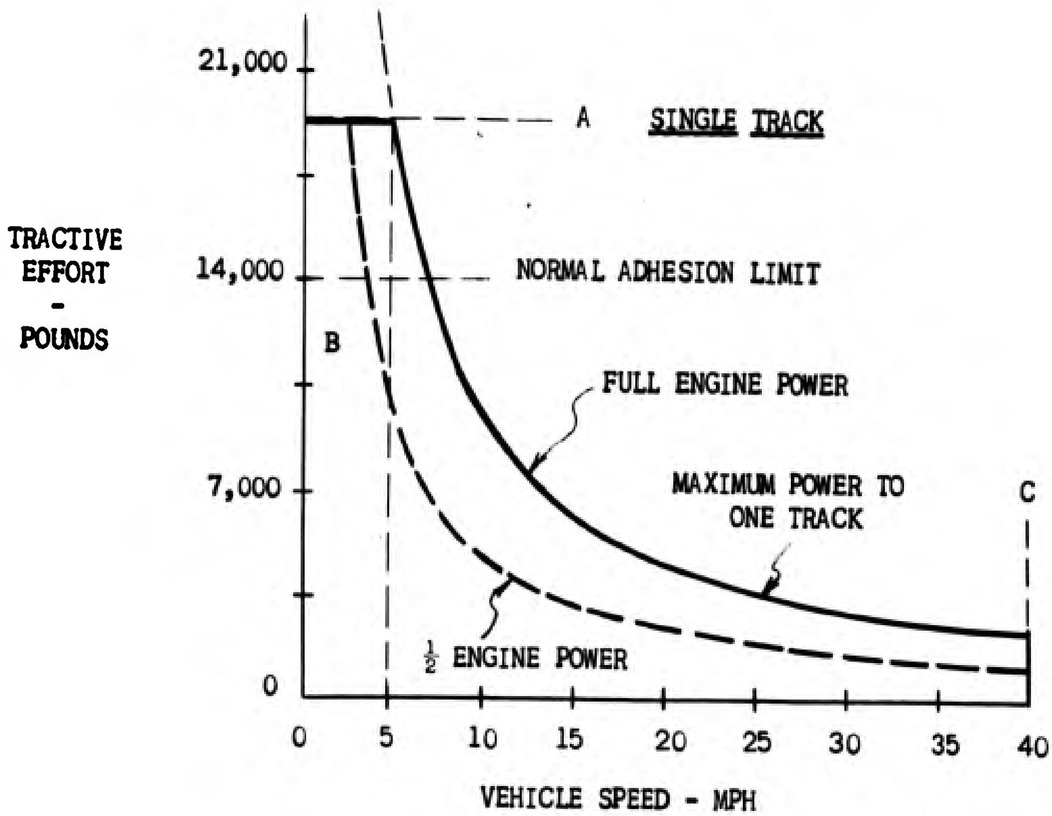
c) Automatically prevents engine overload and stall.

d) Automatically controls engine speed for maximum fuel economy for any given vehicle load condition.

11.7.2 Performance.

Under normal operating conditions and on normal terrain, the transmission will allow the vehicle to negotiate spin turns; i.e., one track will rotate in the opposite direction with respect to the other track. The vehicle is able to go through the complete speed range in both forward and reverse directions without the benefit of a change speed gearbox. Vehicle performance is shown graphicly in Graph 11-1.

The proposed hydrostatic transmission, by being able to utilize rated engine power throughout the vehicle speed range without "slipping", is able to



A = TRACTIVE EFFORT REQ'D FOR 60% SIDE SLOPE

B = TRACTIVE EFFORT REQ'D FOR 70% FORWARD SLOPE

C = MAXIMUM VEHICLE SPEED

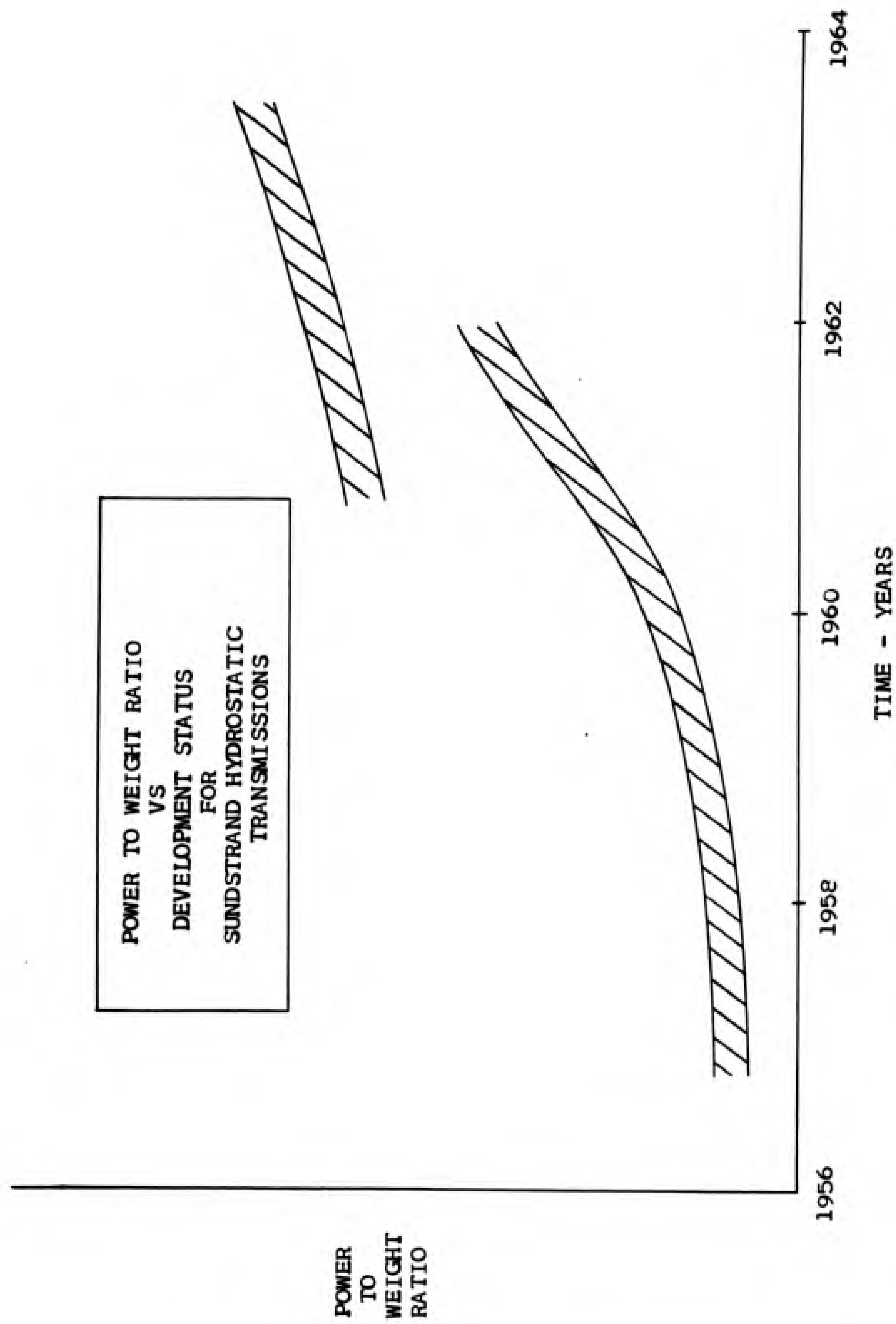
Graph 11-1. Performance Curve

operate with the engine running at its most efficient speed for any given load within the capability of the vehicle. Better engine efficiencies are possible when the engine is operating close to minimum output speed for any specified load. Such a feature can be easily incorporated with the hydrostatic transmission, but cannot be readily accomplished with a change speed gearbox. The overall fuel consumption of a vehicle that utilizes a torque converter or other "slip-type" transmission will not compare favorably with a vehicle equipped with a hydrostatic transmission because of the excessive fluid slipping and resultant power losses at low vehicle speeds.

11.7.3 Hydrostatic Development Potential.

The trend in development for Sundstrand in the ground propulsion hydrostatic transmission field is depicted by the bands shown in Graph 11-2 The reasons for the abrupt improvement of power-to-weight ratio for tracked vehicle transmissions are as follows:

The first transmissions were of a fixed ratio displacement design. Thus, the pump that was to supply the motor with hydraulic power had to be at least as large in displacement as the motor, in order for the transmission to be properly sized. Then, the variable displacement motor was developed. This milestone allowed the pump to be approximately one-fourth the size of the motor in displacement. The next significant milestone was the capability of the transmission to operate at a pressure almost double that of the first transmissions. The last large development improvement was the adoption of a single pump-multiple motor design in the hydrostatic transmission.



Graph 11-2. Hydrostatic Transmission Field

At approximately this same time, the Sundstrand research department brought forth a new design in hydraulic components. As noted in Figure 2, a tremendous breakthrough was realized. It is this type of hydraulic component that is being proposed for the hydrostatic propulsion system for the LVT vehicle. This new design is referred to as the cross head design and is in the initial stages of testing. This component design has a broad range of torque multiplication capabilities per weight. Because of the peculiar design of this type of component, it will be possible to achieve even higher pressures without increasing the over-all weight of the unit, as shown projected in Graph 11-2. The cross head design will compare favorably to any other means of transmitting power for ground vehicle propulsion. The crosshead stroke control arrangement is illustrated schematically in Figures 11-1 and 11-2.

11.7.4 Design Requirements of the Transmission.

The proposed hydrostatic transmission system is designed to meet the requirements specified by Ingersoll Kalamazoo Division, Borg-Warner Corp.

A. Vehicle Data.

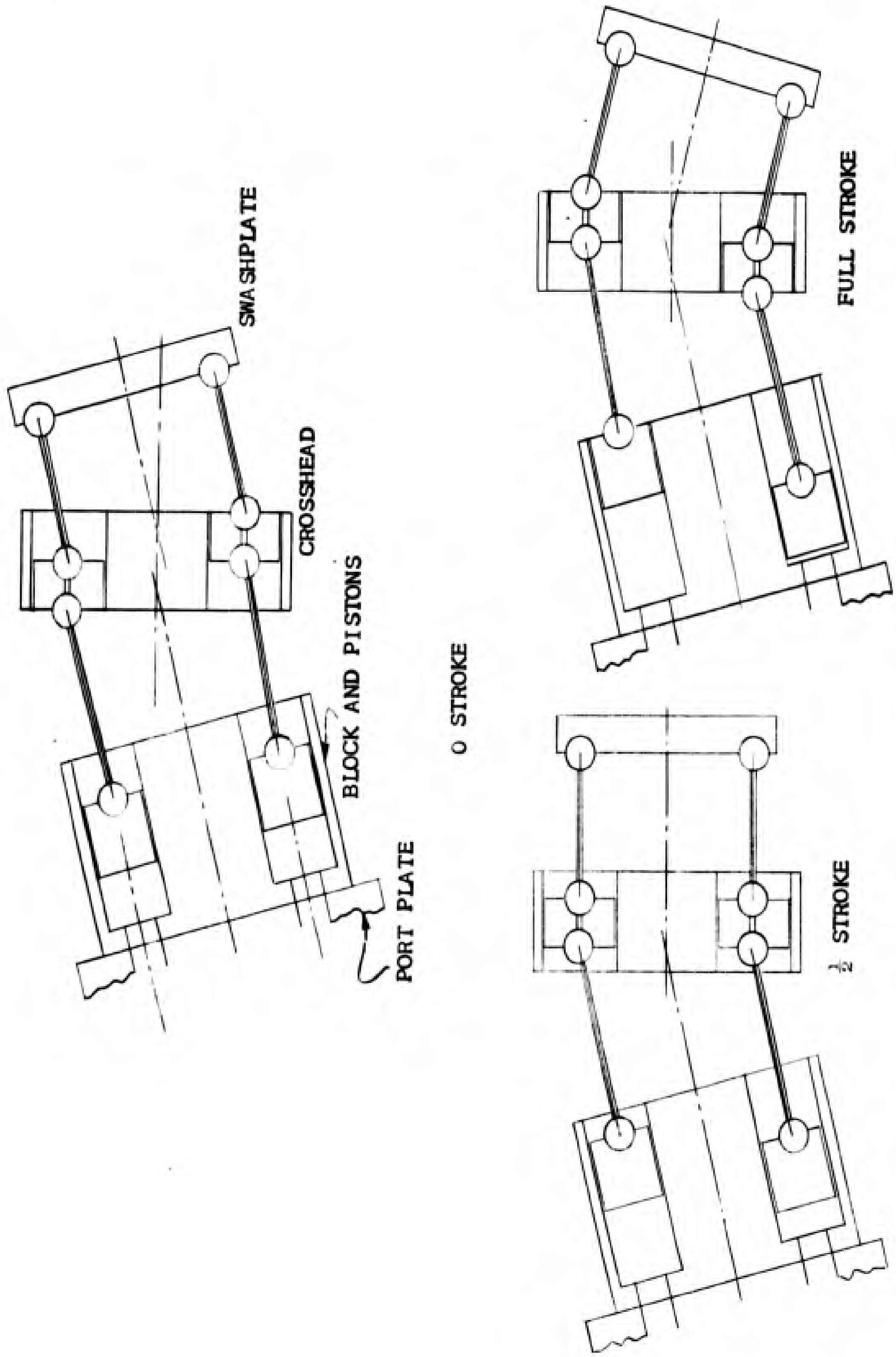
Type - lightweight, tracked vehicle (amphibious)

Weight - 35,000 pounds

Speed - maximum 40 mph on improved roads and
2.5 mph on 70% forward grade

Final drive - 3:1 reduction, furnished by Ingersoll

Sprocket torque (total two tracks) -
32,800 lb.-ft. @ 100% tractive effort
1,155 lb.-ft. @ 40 mph
19,500 lb.-ft. @ 2.5 mph on 70% grade



CROSSHEAD STROKES

Figure 11-1. Crosshead Strokes Control Arrangement

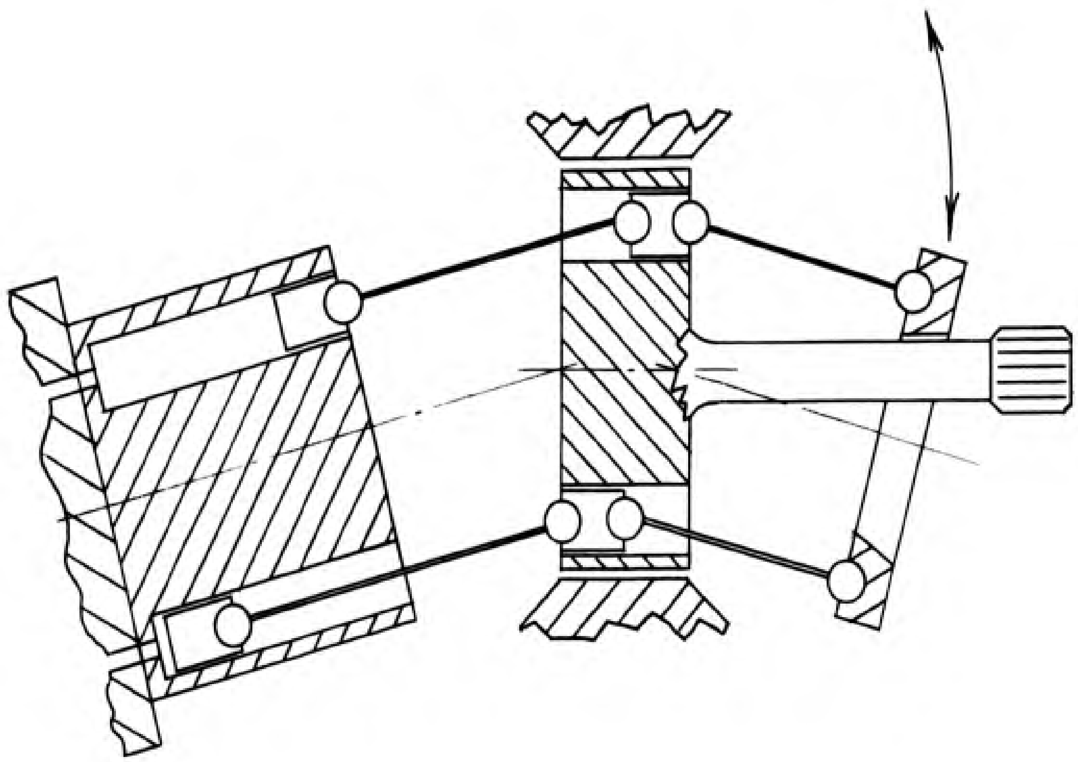


Figure 11-2. Crosshead Assembly

Sprocket speed
630 rpm @ 40 mph
39.4 rpm @ 2.5 mph

Sprocket power (total two tracks) -
140 hp @ 40 mph
146.5 hp @ 2.5 mph on 70% grade

Parking brake - furnished by Ingersoll

Water propulsion - by tracks

B. Engine Data.

Model - Cummins V8300 Diesel

Rating - 300 hp @ 3000 rpm (idle 550-800 rpm)

C. Transmission Parameters.

Controls -	Single stick for steering and direction. Accelerator pedal for torque and velocity. Brake pedal.
Steering -	15% differential in track speed @ 40 mph 25% differential in track speed @ slow speed
Braking -	Capable of stopping a 35,000-lb. vehicle 25 times from 40 mph @ 2 minute intervals, at an average deceleration rate of 12 ft./sec.^{2}. (See figure 19)
Fluid -	MIL-H-6083 or MIL-L-7808D preferred. SAE 10 acceptable.

11.7.5 System Equipment Description.

A. General Configuration.

The hydrostatic transmission system is schematically represented in Figures 11-2 and 11-4.

The transmission has the general physical characteristics of a "T" configuration; i.e., the power input provision is perpendicular to each of the two oppositely directed power output provisions. The proposed hydrostatic

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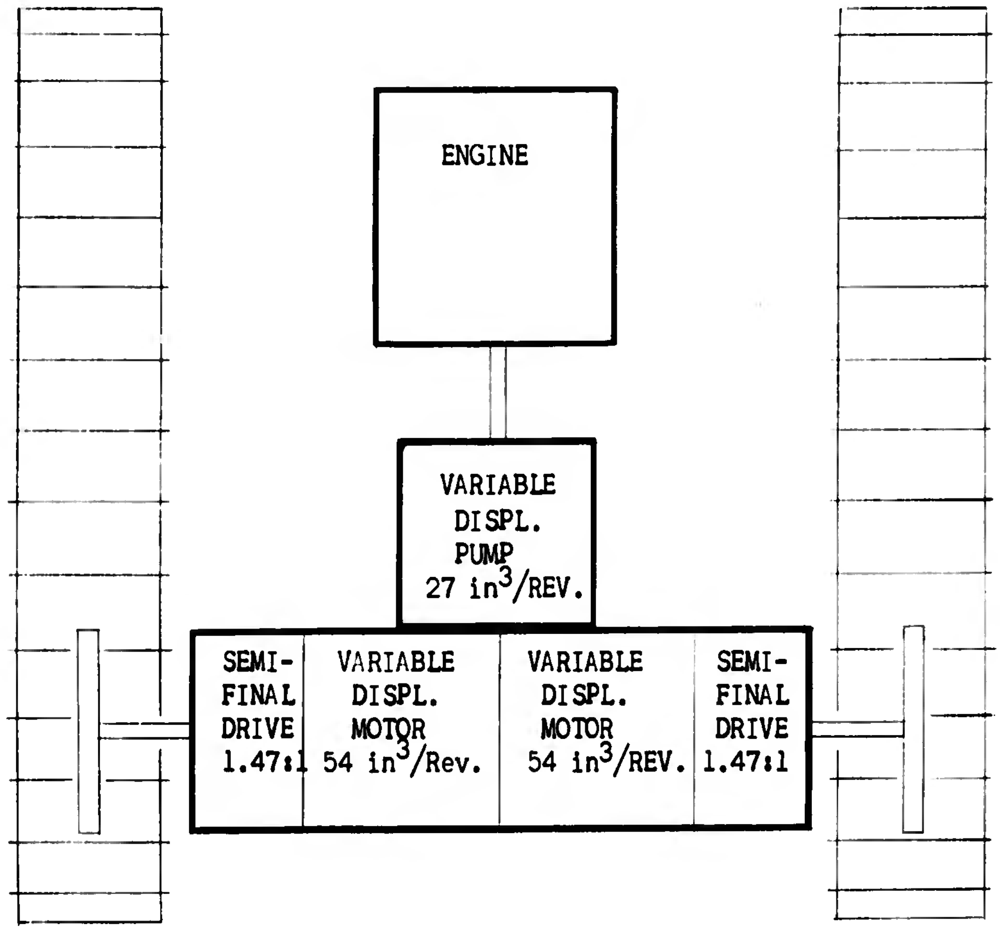
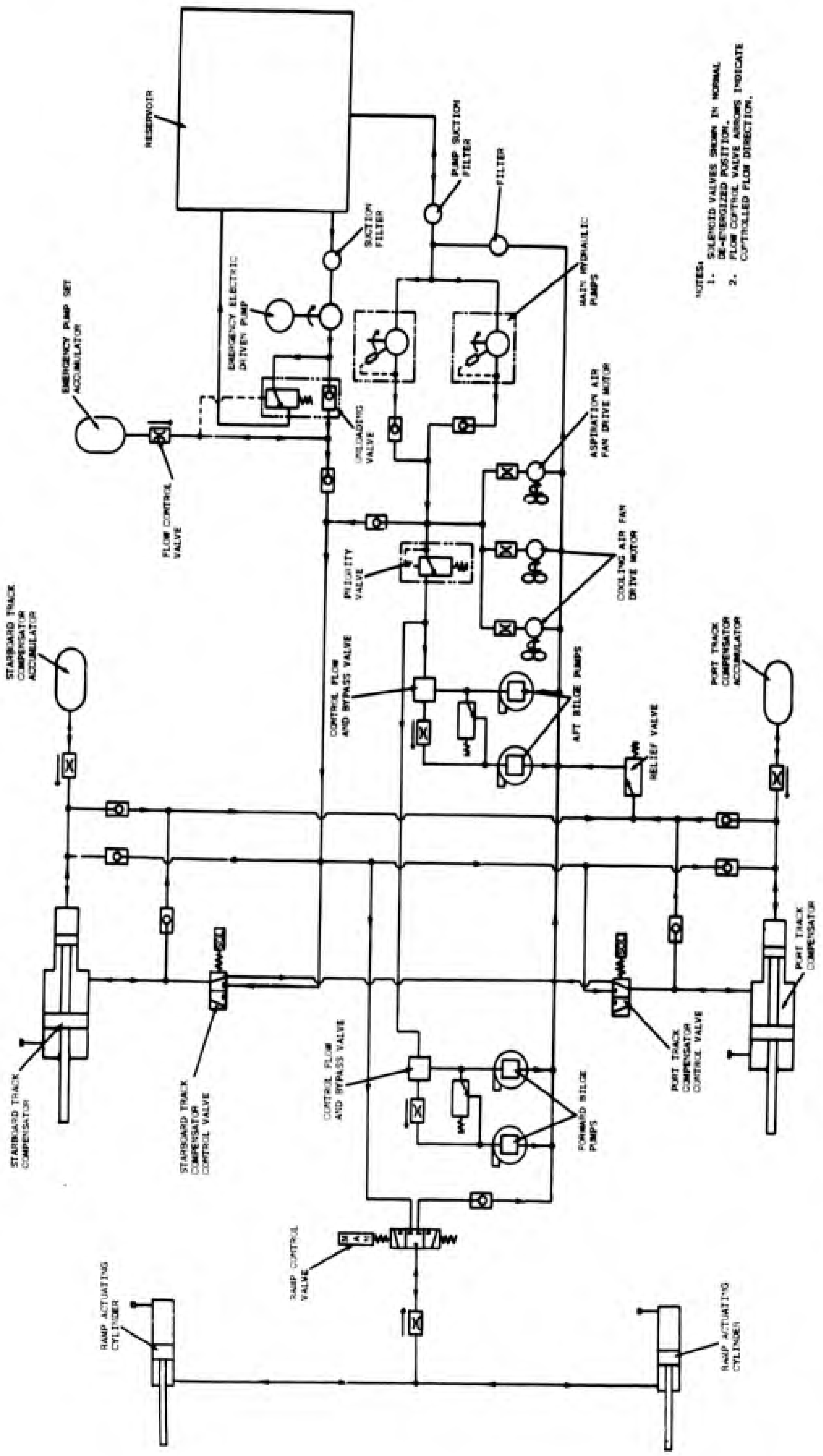


Figure 11-3. Hydrostatic Transmission System - Schematic



NOTES:
1. SOLENOID VALVES SHOWN IN NORMAL DE-ENERGIZED POSITION.
2. FLOW CONTROL VALVE ARROWS INDICATE CONTROLLED FLOW DIRECTION.

Figure 11-4. Schematic Diagram of Hydrostatic Transmission System transmission is composed of a hydraulic pump that couples to the power output pad of the engine or engine mounted gearbox. On the other end of the pump and through a mutual endcap manifold are connected two hydraulic motors "back to back". During vehicle propulsion, power is mechanically transmitted from the engine to the pump. The pump then converts mechanical power to hydraulic power, which is ported to the motors. The motors convert the hydraulic power back to mechanical rotating-shaft power. This power is directed to the final drive. In dynamic braking, the direction of power flow is reversed for each of the above described components.

The purpose of the endcap manifold, which is common to the three hydraulic units, is to internally port the hydraulic fluid between the hydraulic units and to house various control valves. Externally located on the basic transmission are the fluid reservoir, fluid filter, bypass valve, and certain transmission control valving and plumbing. Wherever practical, it is proposed to use aluminum as housing material.

Lubrication of the power train gearing will be accomplished by a wet sump, splash lube system deriving fluid from the hydraulic unit case leakage. Other metal bearing surfaces in the transmission will be lubricated by lube jets, as required. A scavenge pump will return the lubricating fluid from the sump to the transmission reservoir.

In general, the main functions of the proposed hydrostatic transmission are to control the propelling, steering and braking of the vehicle. The controls necessary for the operator to perform these functions are as follows:

1. Steering and Direction Device
2. Accelerator Device
3. Braking Device

A minimum of two indicator instruments for the transmission are recommended on the cab control panel of the vehicle; namely, a lamp indicator signal for transmission fluid temperature and a charge pressure gage.

B. System Components.

The transmission system components are shown in the transmission block diagram, Figure 11-4.

1. Variable Displacement Hydraulic Pump.

The proposed hydraulic pump unit is of an axial-piston, crosshead design and has a speed range of 0 to 3000 rpm. The unit is designed to vary its displacement from 0 to 27.0 cu. in. per rev. and has a maximum pressure capability of 6000 psi. The displacement is varied by a hydraulically controlled force multiplying piston which regulates the angular displacement of a swash plate. The pump unit is operated with a scavenged case and will receive a large amount of lubrication from normal fluid leakage. The pump housing will be of cast aluminum.

2. Variable Displacement Hydraulic Motor.

The proposed hydraulic motor units are of axial-piston, crosshead design and have a speed range of 0 to 2800 rpm. Each unit is designed to vary its displacement from 0 to 54.0 cu. in. per rev. and has a maximum pressure capability of 6000 psi. As with the pump unit, the displacement is varied by a hydraulically controlled force multiplying piston which regulates the angular displacement of a swash plate. The motor units are also operated with scavenged cases and will receive lubrication from normal fluid leakage. The motor housings will be of cast aluminum.

3. End Cap Housed Control Valving.

a. Charge Return Check Valves. These two check valves are located between the high and low pressure ports to allow the return of charge pressure flow to the main hydraulic system during all modes of transmission operation; i.e., power transmitting, power absorbing, vehicle towing, and push-to-start.

b. Working Pressure Relief and Neutral Valve. This device is also located between the high and low pressure ports to protect the hydraulic system from over-pressure conditions. It works by "short circuiting" flow from high to low pressure. This device is also used as a neutral valve to place the system in a short circuit condition during engine starting, engine idling, vehicle towing, and the push-to-start situation. The neutral valve is manually actuated by the operator.

c. Charge Return Valve. This device is located between the high and low pressure ports. It allows the extraction of return pressure flow from the main hydraulic system during all modes of transmission operation in order to accommodate the proper magnitude of fluid flow through the heat exchanger circuit.

d. Flow Reversing Valve. This device is located between the fluid ports leading to each motor so as to direct either high or low pressure flow into the two portings of each hydraulic motor. Reversing the high and low pressure flow to the motors during certain vehicle steering operations is manually controlled through the flow reversing pilots.

e. Flow Reversing Pilot. This hydraulic valve is located between each flow reversing valve and the operator mode selector valve. The purpose is to activate the flow reversing valves upon a signal from the steering mechanism.

4. Transmission Accessory Equipment.

a. Input-Driven Charge Pump. This device is a variable-displacement pump that provides a positive and proper quantity of fluid through the heat exchanger circuit of the transmission system. The charge pump is also used to replenish the fluid leakages of the hydraulic components from the reservoir to the main fluid lines. The displacement of the pump will be controlled by system pressure and engine speed.

b. Input-Driven Scavenge Pump. This variable-displacement pump provides the means of scavenging the inner cavities of the transmission housing of fluid leaked by components. The displacement of the pump will be controlled by system pressure and engine speed.

c. Charge Relief Valve. This device will be used to maintain an adequate back pressure to assure the filling of all hydraulic component cavities and thereby avoid fluid cavitation during transmission operation.

d. Output-Driven Charge Pumps. These two gear-type positive-displacement pumps are connected to the output shafts of the hydraulic motors. During normal transmission operation, they supplement the function of the input-driven charge pump discussed earlier. However, the more distinct purpose of these pumps is to control the displacement of the hydraulic motors. As vehicle speed increases, the output-driven charge pumps increase in speed to provide a control signal to the motor to decrease displacement at a selected rate. The charge pumps serve also to charge the system during vehicle towing and during push-to-start conditions.

e. Filter and By-Pass Valve. The filtering device will be located in the cooler and reservoir hydraulic circuit to remove contamination from the hydraulic fluid which passes through the elements. The by-pass valve will be an integral part of the element housing. The by-pass valve becomes activated at a specified pressure difference across the filter assembly, so the fluid can by-pass the element in the event the element becomes filled with contaminants. Normally, a 33-micron filter is used.

f. Reservoir. The reservoir tank will be supplied by Ingersoll. It should have a capacity of approximately eight gallons. The purpose of the reservoir is to store excess fluid during transmission shut down and to provide a temporary storage delay for fluid so that deaeration can take place while the transmission is operating.

5. Semi-Final Drive Gearing.

Reduction of hydraulic motor speed and increase of torque output is accomplished in this transmission by the semi-final drive gearing. A spur gear arrangement with 1.47:1 ratio is proposed for this transmission to achieve minimum weight and envelope and to match the 3:1 ratio of the final-drive being supplied by Ingersoll as part of the vehicle.

6. Engine Speed Control Equipment.

a. Engine Speed Control Cam and Linkage Mechanism. This system of mechanical devices and interconnecting linkage is to achieve proper control of the engine for optimum fuel economy. This system is located between the engine driven governor, the accelerator and brake valve, and the operator's accelerator device.

b. Engine Drive Governor. This rotating mechanical governor receives signals from the vehicle operator through the engine speed control cam and linkage and properly operates the engine for optimum fuel economy. The governor will be located on the vehicle engine. It will either be specified by or furnished by Sundstrand.

7. Vehicle Control Equipment.

a. Accelerator and Brake Valve. This device controls vehicle acceleration, velocity, and braking. It is located between the operator's accelerator device, the braking device, and the hydraulic pump displacement control piston. The valve receives signals from the operator via the accelerator device and brake pedal and relates them to the transmission and engine simultaneously.

b. Operating Mode Selector Valve. This manually operated device controls the forward and reverse modes of vehicle motion. The valve serves also to put the transmission into neutral by manual selection. It hydraulically controls the flow reversing pilots of the hydraulic motors to achieve the desired direction of vehicle travel.

c. Steering Mechanism Control. This system of mechanical devices and valving controls vehicle steering. This device is mechanically connected to the displacement control pistons of the hydraulic motors. Signals are received by the steering mechanism control from the operator through the vehicle steering device.

C. Operator Controls.

1. Cab-Mounted Operator Controls.

The proposed type and number of cab-mounted controls have been determined on the basis of mobility and safety. By incorporation of multiple-purpose controls, the number in the cab is established at three, as listed below. The controls are schematically illustrated in figures 11-5 and 11-6.

a. Steering and direction lever

b. Accelerator pedal

c. Service brake pedal

2. Steering and Direction Lever.

Steering of the vehicle will be accomplished by moving the lever to left or right. A maximum movement to either the left or right will cause the ve-

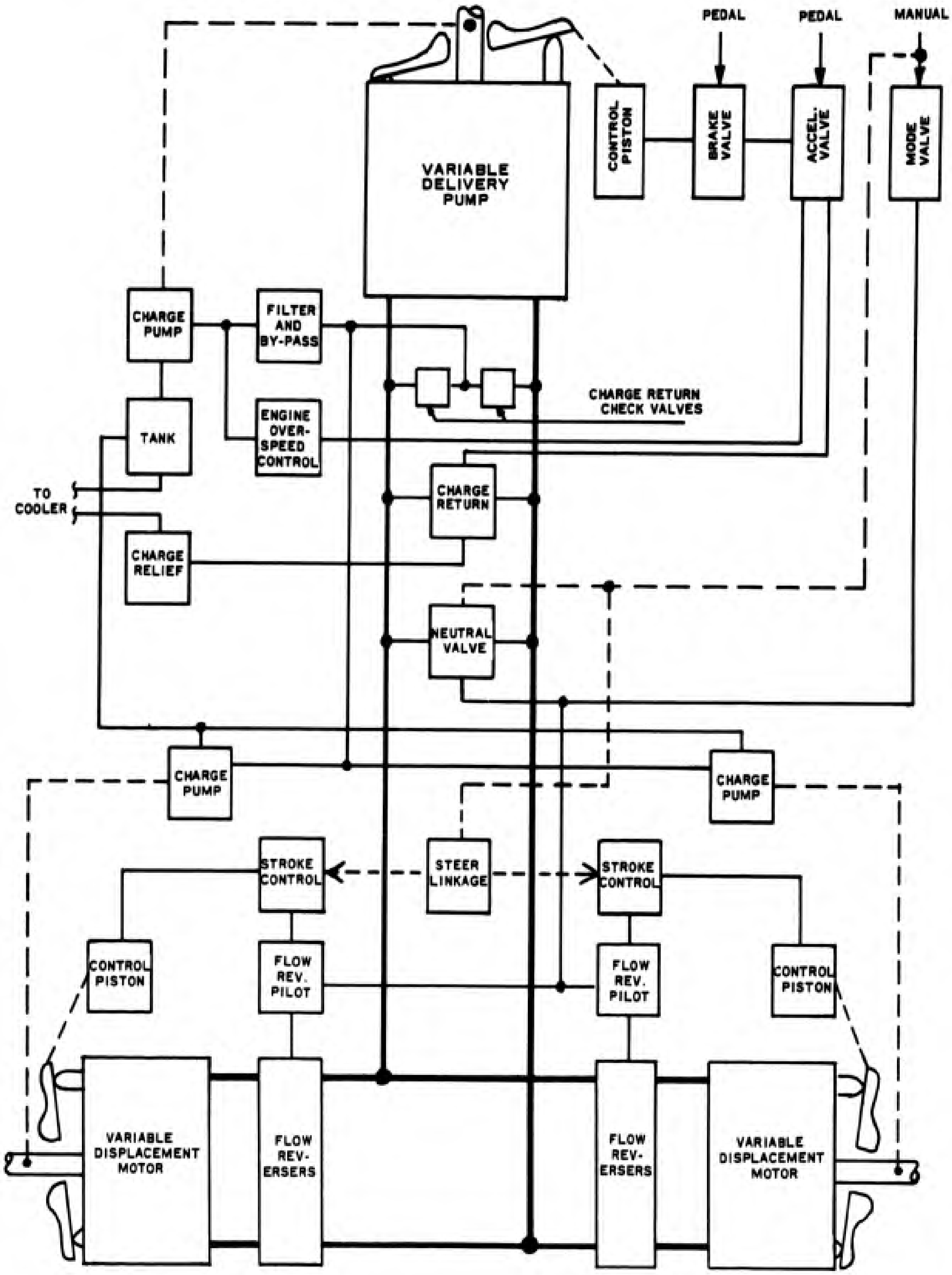


Figure 11-5. Control Block Diagram - HST System

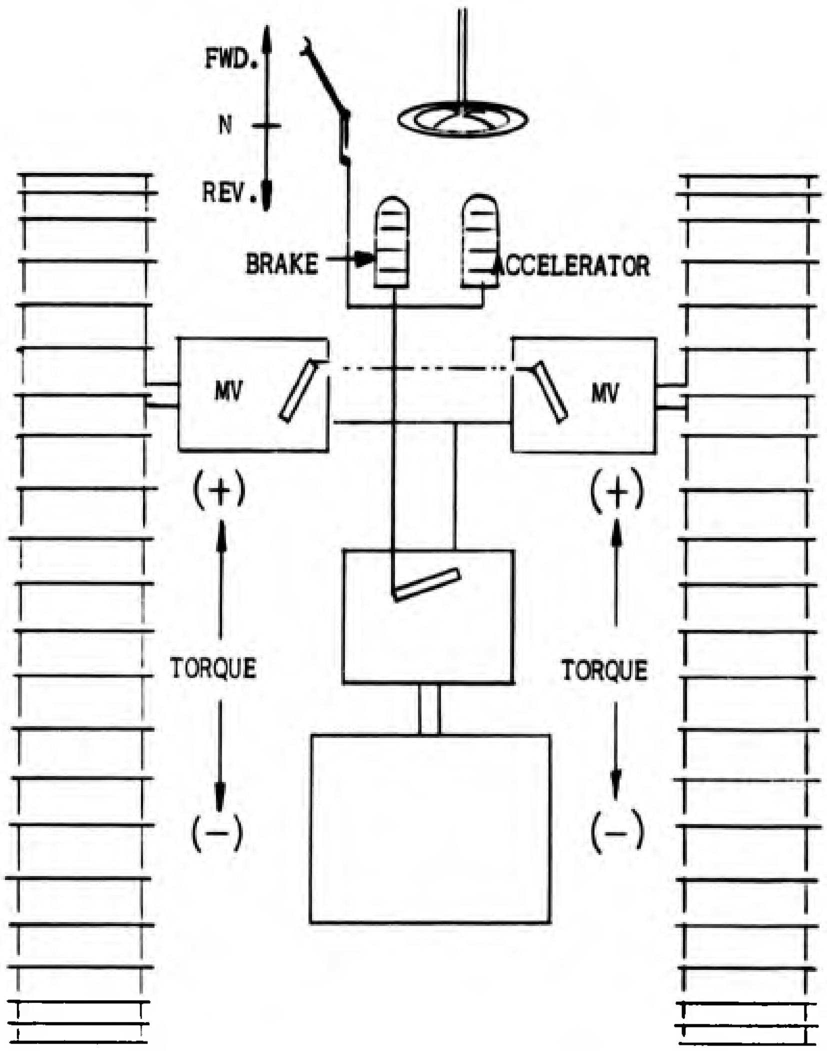


Figure 11-6. Block Diagram - One Pump, Two Motors

hicle to negotiate a spin turn. The power steering system is of a modulated-torque type to facilitate smooth steering control of the vehicle with minimum operator effort. Direction control is obtained by moving the lever forward for forward vehicle movement and reverse for reverse movement. With lever straight up, the transmission is in neutral.

3. Accelerator Pedal.

The accelerator proposed for this hydrostatic transmission will be a pedal which is similar to the normal truck accelerator. Depressing the accelerator will cause the vehicle to accelerate. The greater the accelerator movement, the greater is the torque applied to the tracks. Removal of the foot from the pedal causes the engine to return to the idle position and the pressure level in the transmission to be lowered to a nominally low amount of propelling torque.

4. Service Brake Pedal.

Depressing the brake pedal will actuate the vehicle service brakes. As with most standard brakes, the greater the pedal is displaced, the greater is the braking effort. The hydrostatic transmission system makes use of two methods of service braking: (1) raising the engine speed to make use of its frictional drag and (2) dissipating heat energy from vehicle momentum into the hydraulic system of the hydrostatic propulsion system. These two methods will be employed to operate consecutively in the order mentioned above. The engine is automatically protected from over speeding during vehicle braking by the engine overspeed control valve.

5. Instrument Panel.

Besides the operator's controls in the cab, it is proposed that two gauges be mounted on the control panel to monitor the operation of the transmission during vehicle driving and braking. One is a charge pressure gauge to indicate to the operator whether the system is correctly primed and whether cooling flow is properly circulating through the heat exchanger. The other is an oil temperature gauge to indicate whether the cooling system is functioning properly.

11.7.6 Operation and Control Features.

A. Vehicle Operation.

1. Velocity Control.

Speed of the vehicle is controlled by imposing a torque from the transmission. This is accomplished by a pressure compensated pump in conjunction with the two variable-displacement motors. The pump is connected directly to the engine. Its displacement is determined by the pressure desired in the system. The degree of pressure is determined by the position of the accelerator pedal. Depressing the accelerator places a spring force bias on the accelerator and braking valve which in turn ports oil to the displacement control piston of the pump. As a result, the desired pressure is created in the system. A continuous pressure signal is fed back from the system to the accelerator and braking valve. As the pressure approaches the desired magnitude, the accelerator and braking valve approaches its neutral position and begins to modulate. Since, in a hydraulic circuit of this type, the resistances down-stream from the pump determine the pressure level for a given flow, the flow must be varied in order to establish the pressure level desired by the operator. The resistances in the hydraulic circuit are the load torques on the output motors and the parasitic flow losses. The flow that the motors will require is determined by the speed of the vehicle and the displacement of the motors. A slight increase in flow to the motors will cause the pressure and the subsequent motor output torque to build up. A very small change in flow is all that is necessary to make a large change in the pressure level. This is because of the nearly incompressible nature of the hydraulic oil and the small change in leakage due to a change in oil pressure. Since the displacement change of the pump is very small for a change in pressure level, the torque build-up in the transmission can occur nearly instantaneously, but is easily controlled. The vehicle propulsion controls are illustrated in figures 11-7 and 11-8.

The value of torque output of a hydrostatic transmission is determined by the pressure drop across the motors and their respective displacements. In the proposed transmission, when the vehicle is moving straight ahead, the displacements of the motor are automatically adjusted to a specified value for any given vehicle speed. Consequently, the motor torque output is determined solely by the system pressure level, which, in turn, is set by the accelerator pedal. Therefore, the force the driver applies to the vehicle is determined by the position of the accelerator pedal.

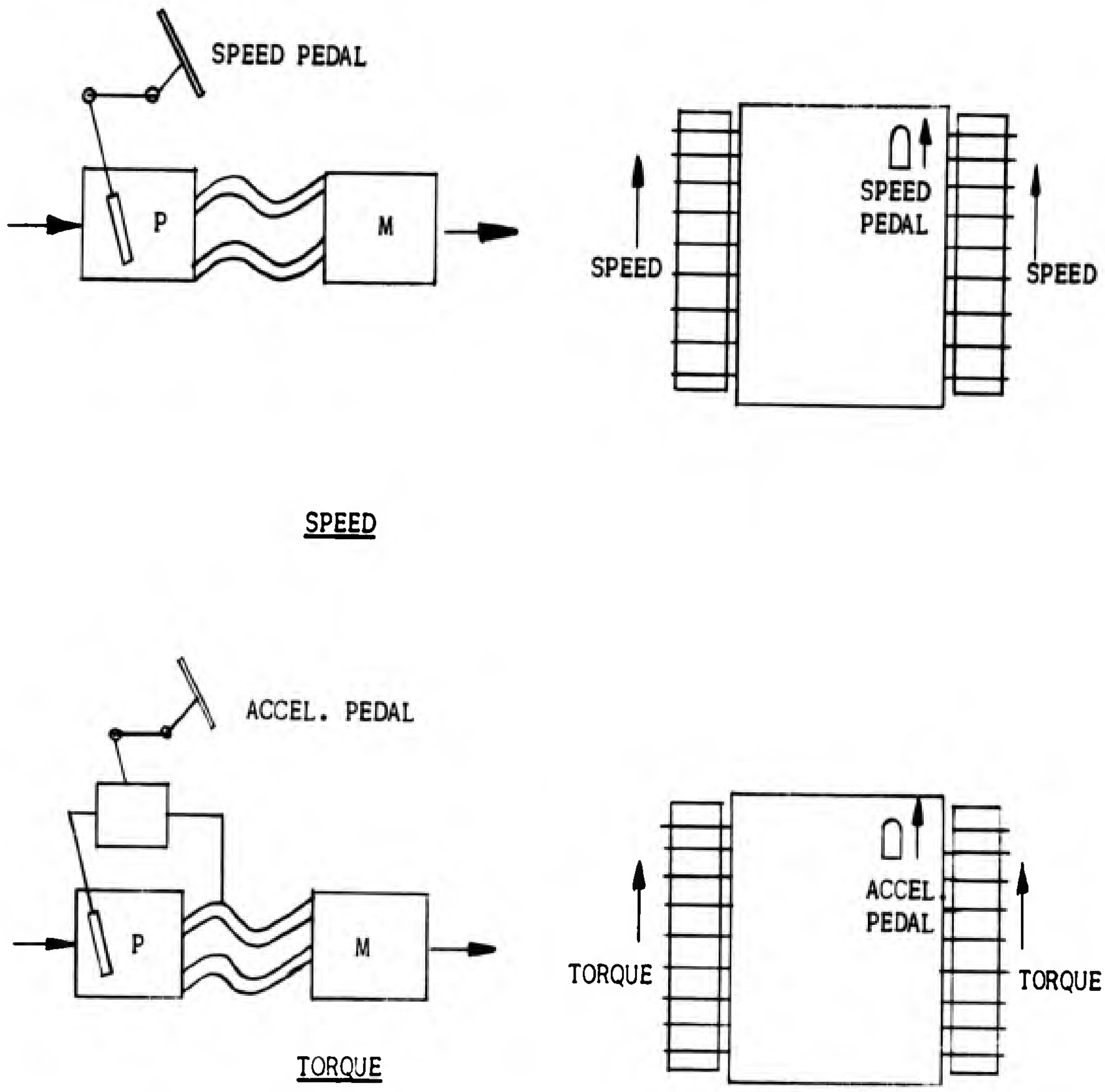


Figure 11-7. Propulsion Control

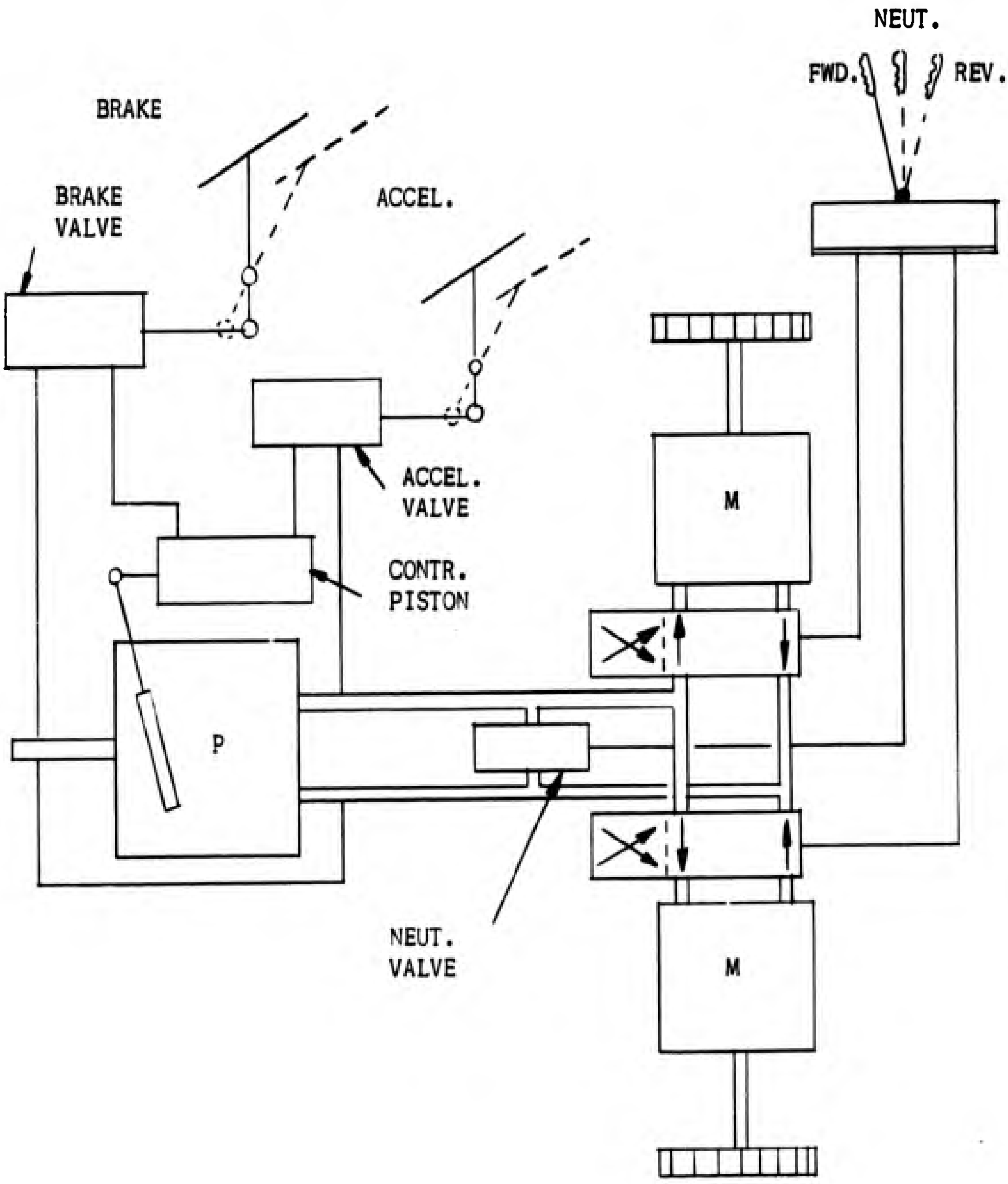


Figure 11-8. Block Diagram - Forward, Reverse, Brake Controls

Reversal of the vehicle's direction of travel is accomplished by moving the operating mode lever. This lever is mechanically connected to the three-way operating mode selector valve. The valve controls the direction of working pressure flow to the hydraulic motors. The rest of the transmission operates in an unchanged manner when the direction of vehicle travel is changed. Braking the vehicle is accomplished as before the power-output hydraulic units reverting to pumps and feeding power back to the hydraulic unit mounted on the engine gearbox.

2. Steering Control.

The method of steering most desirable for this transmission, from a standpoint of simplicity, is the modulated-torque type. This steering arrangement is shown in the block diagram in Figure 11-9. Vehicle steering forces required and steering force available are shown in Figure 11-10. Regenerative steering is accomplished because it is possible to impose infinitely variable torque from full forward torque to full reverse torque on each track independently. This is done by adjusting the motor displacements relative to each other. Since the pressure in the system is determined by the accelerator, variations of the motor displacements will have a direct effect on the torque applied to the tracks. If the torques can be modulated, smooth and uniform regenerative vehicle turns are possible.

The motors in this transmission have been designed so that they are unable to achieve negative displacement. This was done with the idea of reducing

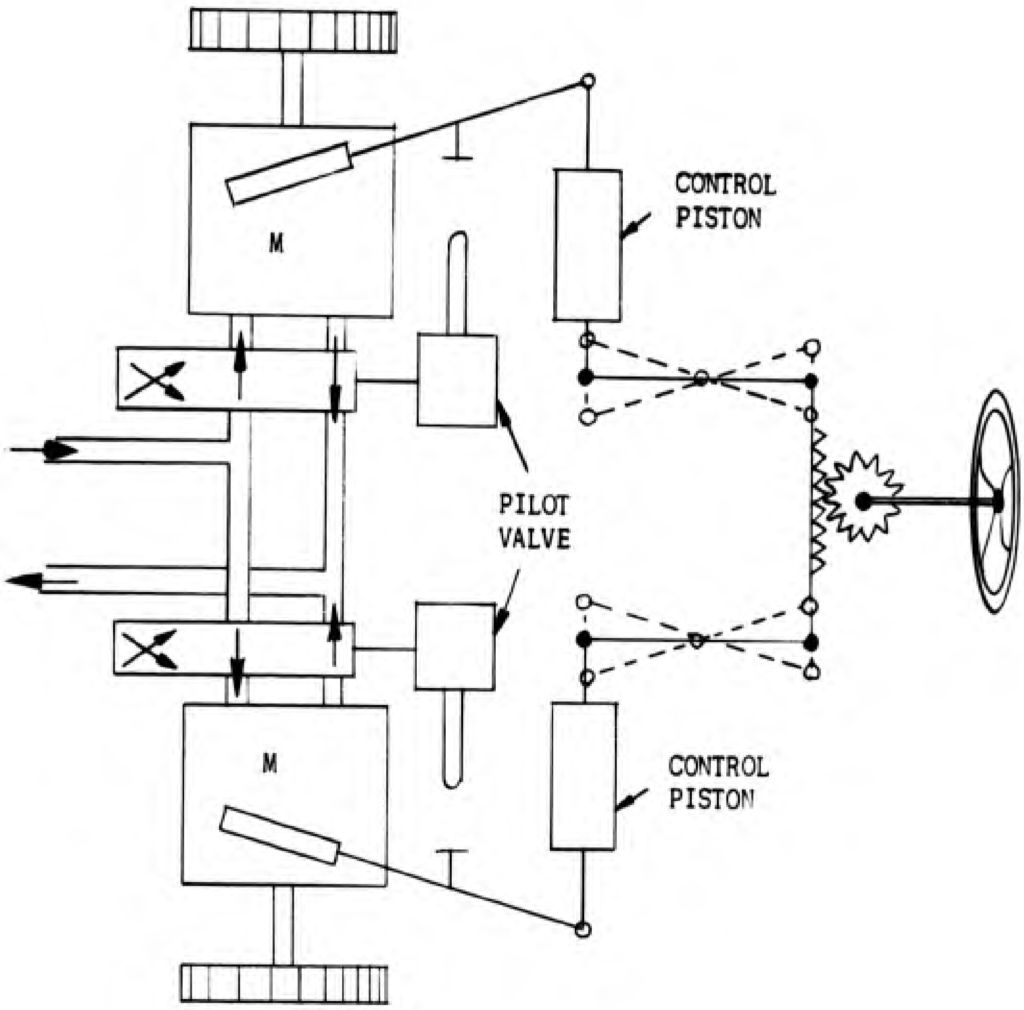


Figure 11-9. Modulated Torque Type Steering Transmission

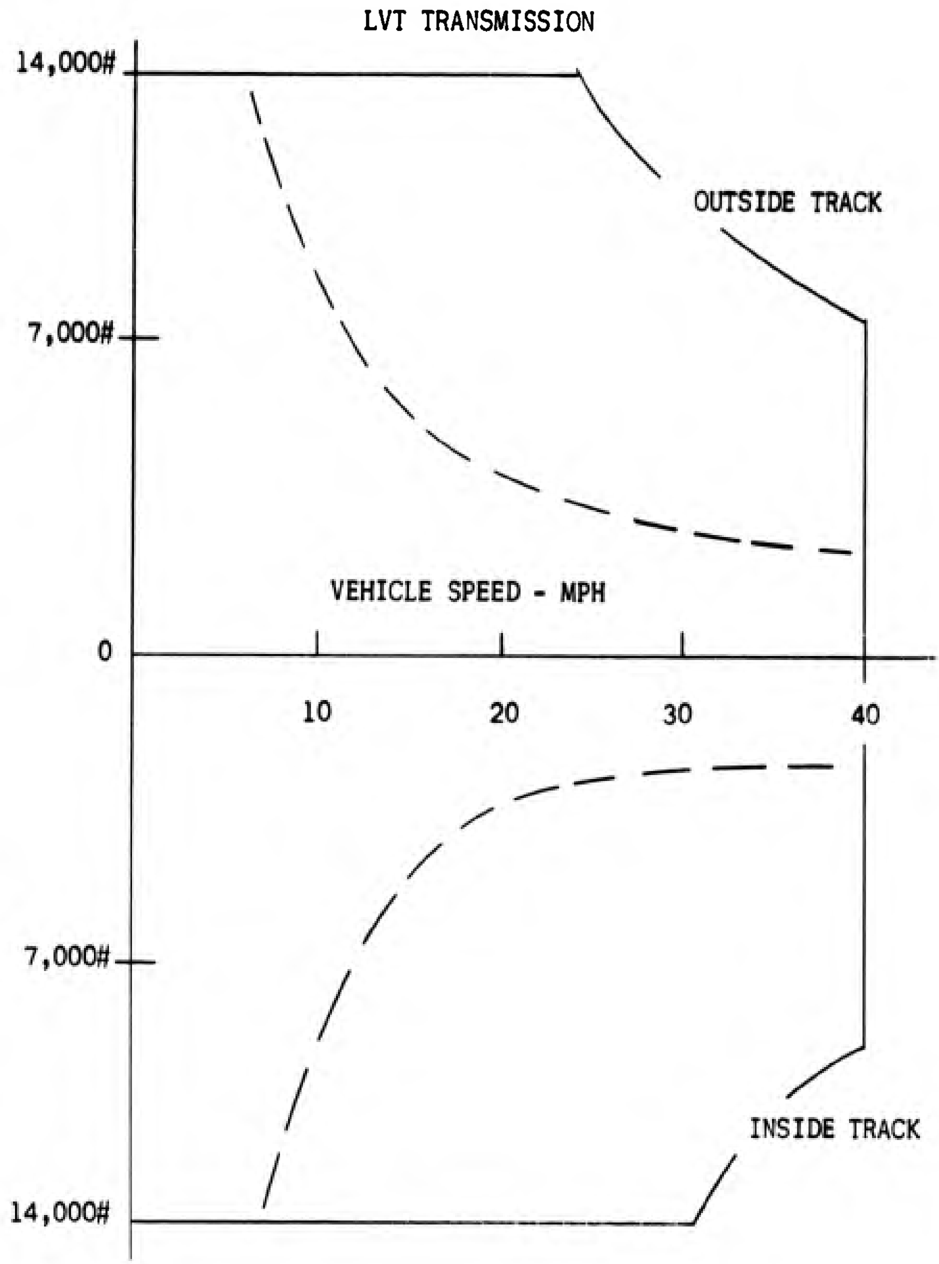


Figure 11-10. Steering Forces Block Diagram

their size as much as possible. In order to obtain the negative torques that are needed for live track and high speed steering, flow-reversing valves are incorporated. These valves reverse the flow to the motor each time the motor swashplate reaches the zero-stroke position. When the flow reversal occurs, the motor swashplate continues back into stroke to give the required negative torque. Since these motor swashplates are directly connected through linkages to the steering wheel, the recycling of the flow reverser will be automatic as the steering wheel is returned to the center position.

During the time of steering, the pressure-compensated pump will attempt to retain the pressure level demanded in the system by the accelerator pedal. It will do this by varying its stroke and will accomplish its purpose up to the point where one of the two motors goes to full displacement. Prior to this, an increase in one motor displacement will cause a commensurate decrease in the other motor displacement. As a result, the average flow to the motors is the same. Once one of the motors reaches full displacement, the pump displacement may increase and, in certain turns, may saturate. If this happens, the vehicle speed will decrease sufficiently to generate the pressure required for turning. For most turns, however, the average speed of the vehicle will remain constant.

The controls needed for modulated-torque steering are the two flow reversers, two pilot valves, and linkage to override lost motion between the displacement control pistons of the motors. Regenerative modulated-torque steering is both reliable and simple, and yet, because of its characteristics, will give a far superior performance than generally seen on present tracked vehicles.

Two other types of steering control are available with the proposed hydrostatic transmission. They will be discussed in part C of this section.

3. Braking.

The hydraulic units employed in this propulsion system are equally capable of being either pumps or motors. To reverse their roles, which is necessary in braking, the two fluid ports in the end cap manifold for each motor must reverse in high and low pressure exposure. Whenever the flow through the output units exceeds the flow from the engine driven unit, the track units become pumps and the engine-driven unit a motor. The pressure level of the system is determined by the relative speed and respective displacements of the track units and the engine mounted unit. The reversal of power flow from propulsion to braking is illustrated in figure 11-11. Figure 11-12 and 11-8 illustrate the general arrangement of the braking system and its cooling. Figure 11-19 shows braking force.

In vehicles with standard gearbox transmission, down-shifting may be employed to help brake on steep hills or to slow down gradually. Kinetic energy from the vehicle can be transmitted through the transmission to the engine where it is dissipated in the form of frictional drag horsepower. The hydrostatic transmission is an infinitely variable speed transmission in the driving direction as well as the braking direction. Consequently, there are steering is both reliable and simple, and yet, because of its characteristics, will give a far superior performance than generally seen on present tracked vehicles.

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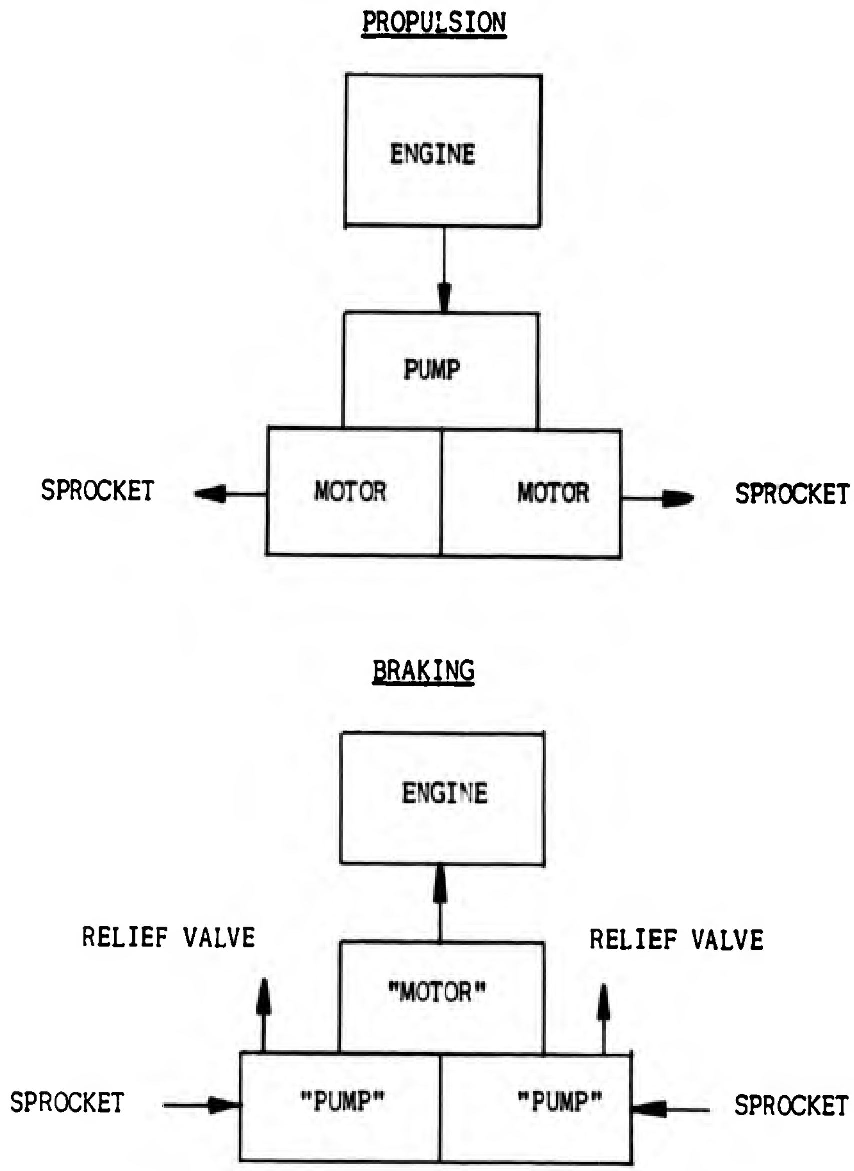


Figure 11-11. Braking Propulsion

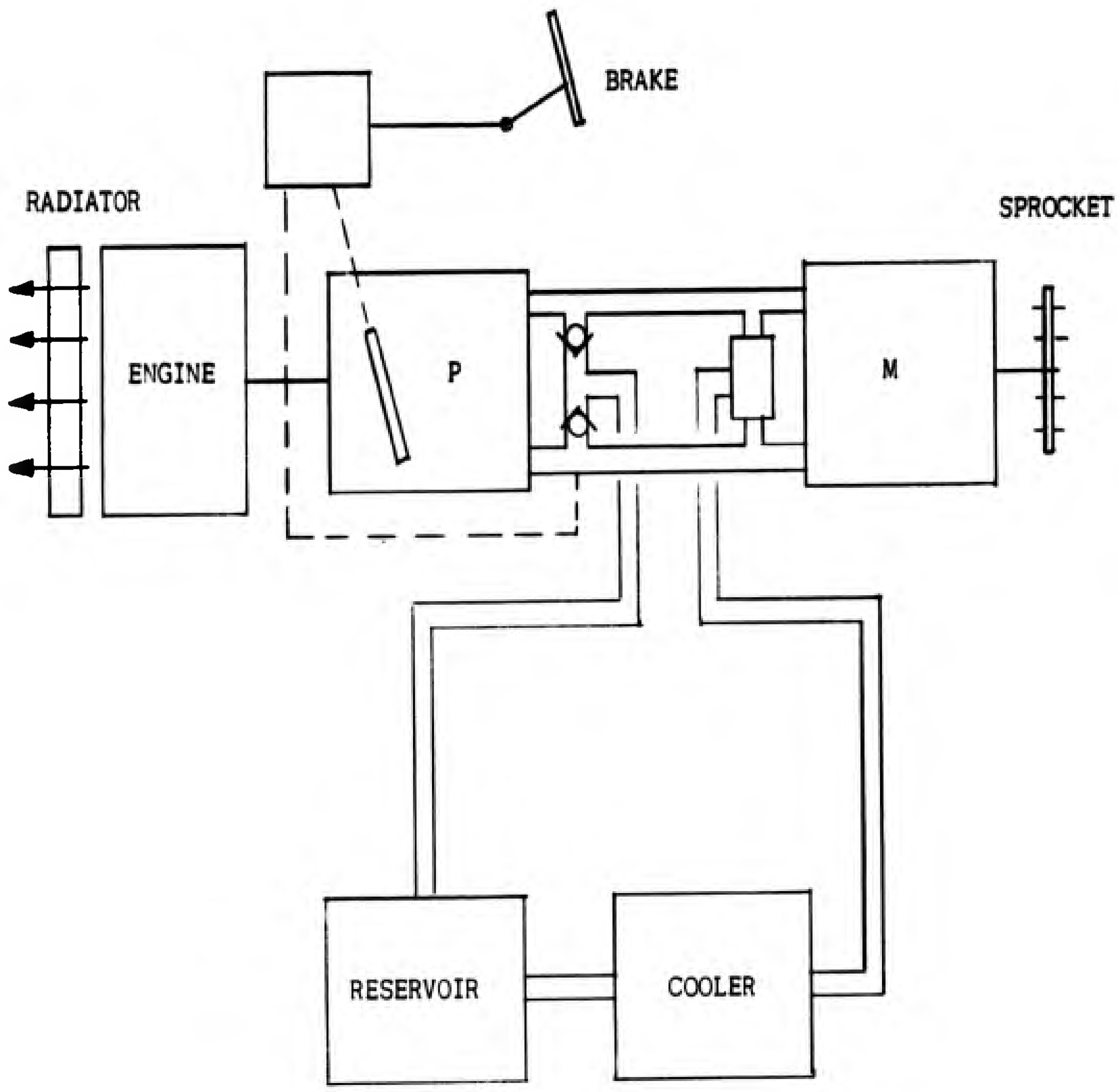


Figure 11-12. Braking Circuit

infinite down-shifting capabilities inherent in the transmission.

The transmission also brakes by the pumping motor units' dumping a portion of their flow over the working pressure relief valve, which is adjustable by virtue of the action performed by the engine overspeed control valve. The pressure setting of the relief valve determines the extent of braking. For the free gas turbine engine prime movers, braking will be accomplished solely by the output units pumping their flow over the adjustable relief valve.

The energy dissipated over the relief valve heats the hydraulic fluid which will be carried to the transmission cooler by the charge circuit. The cooler should adequately dissipate heat generated by engine braking. Brake cooling loads are thereby shared between the two cooling systems. However, since free turbine engines cannot be utilized for vehicle braking, the cooling system for the hydrostatic propulsion system would be considerably greater. The brake cooling system is shown in figures 11-12 and 11-13.

4. Vehicle Towing.

The vehicle can be towed either in the forward or reverse direction without the engine operating. This facilitates return of the vehicle after an engine or engine-pump failure or certain control malfunctions. For vehicle towing, the operating mode lever is placed in the neutral position. This makes a hydraulic short circuit in the system porting. As the vehicle is pulled, the output hydraulic units revert to pumps and attempt to motor the engine. The short circuit is accomplished by resetting the working pressure relief valve

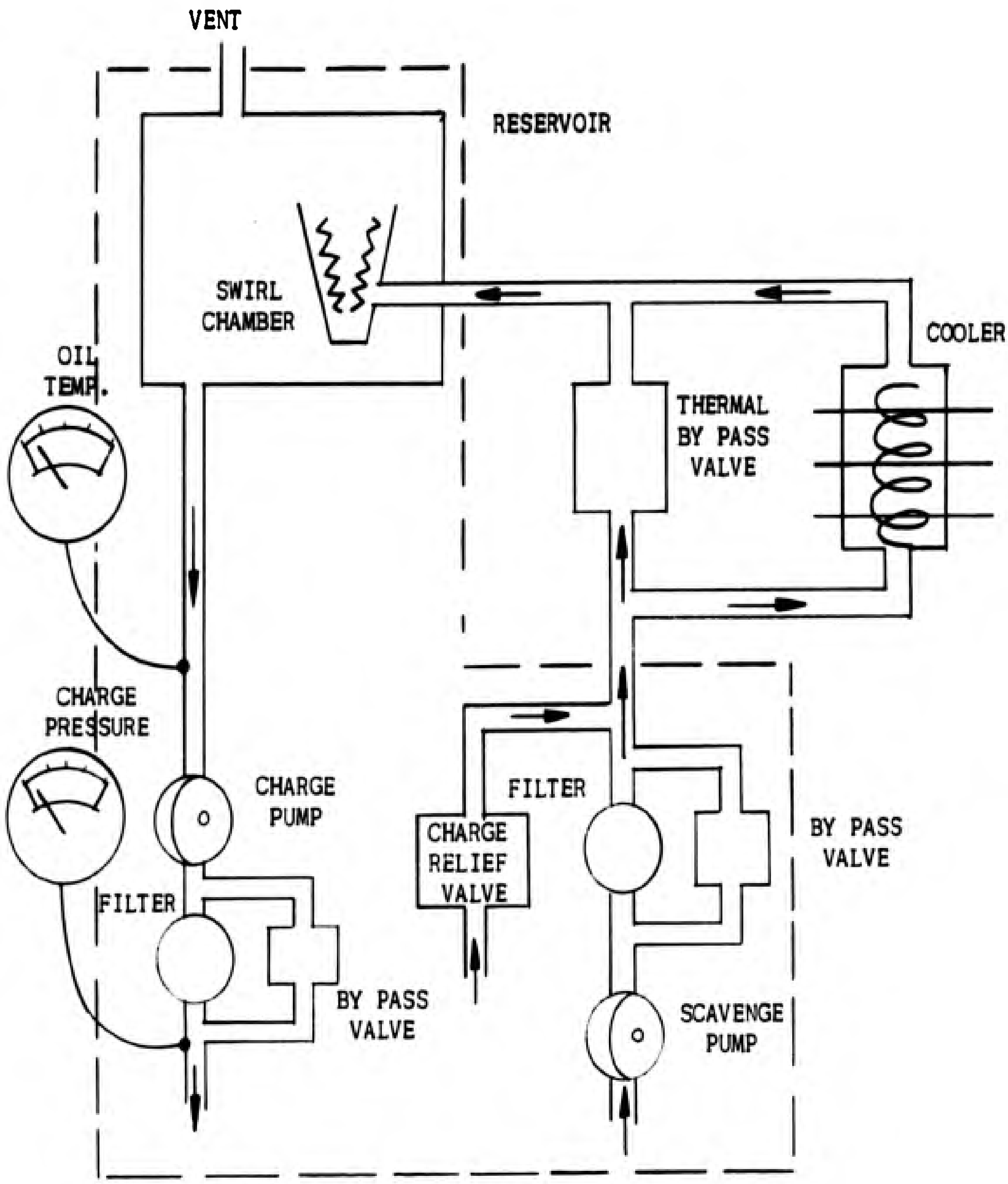


Figure 11-13. HST Cooling Oil Circuit

to a very low pressure setting, which prevents a pressure build-up and therefore enables a minimum towing force to be used. Each of the output units has a charge pump which serves as a replenishing pump as well as a cooling circuit pump. The vehicle can be towed at reasonable speeds for prolonged periods of time.

5. Push-to-Start Capability.

The vehicle can be started by pushing in either the forward or reverse direction. Placing the operating mode lever in the forward direction when vehicle is being pushed forward or placing the lever in the reverse position when it is being pushed in reverse will start engine cranking. Control of the amount of cranking torque applied to the engine is accomplished by the accelerator pedal. A moderate amount of pedal depression will produce substantial cranking torque. Because of the inherent high torque multiplication, the vehicle engine can be cranked at very low vehicle speeds.

B. Fuel Programming Control.

The object of the engine speed control is to operate the engine at the lowest speed commensurate with handling the load. This means the engine runs near full throttle over most of the vehicle's operating range to obtain the best possible specific fuel consumption. In addition to this, control is provided to fully utilize the engine inertia forces for transient power demands. The control will maintain a reasonable idle speed at zero and very light loads.

The essential components of the engine speed control system are the accelerator foot pedal, the engine-driven governor, the engine speed control program cam linkage mechanism, and the accelerator and braking valve. The foot pedal and its linkage are connected to the engine governor and the accelerator and braking valve through the engine speed cam. This arrangement is illustrated by the block diagram in Figure 11-14. Therefore, as the foot pedal is depressed, the engine governor is reset to a higher speed. The governor reset linkage is connected to a dash pot to give smooth operation. Simultaneously, the engine speed cam will advance to its appropriate position to serve as an index of engine speed.

When the speed governor is signaled by an increase of output torque, the linkage will produce a movement to increase the engine speed. The throttle linkage will move until it reaches its full open position, indicating the engine is now fully loaded for the particular speed setting. The control will always insure that the engine will be running at full throttle for any particular vehicle power requirement. The prime mover fuel program control is shown schematically in Figure 11-15.

C. Optional Types of Steering Control (see Fig. 11-16).

1. Difference Speed Steering.

Difference speed steering is a means of achieving a precise and positive vehicle turn independently of the torque being transmitted by the infinitely variable transmission.

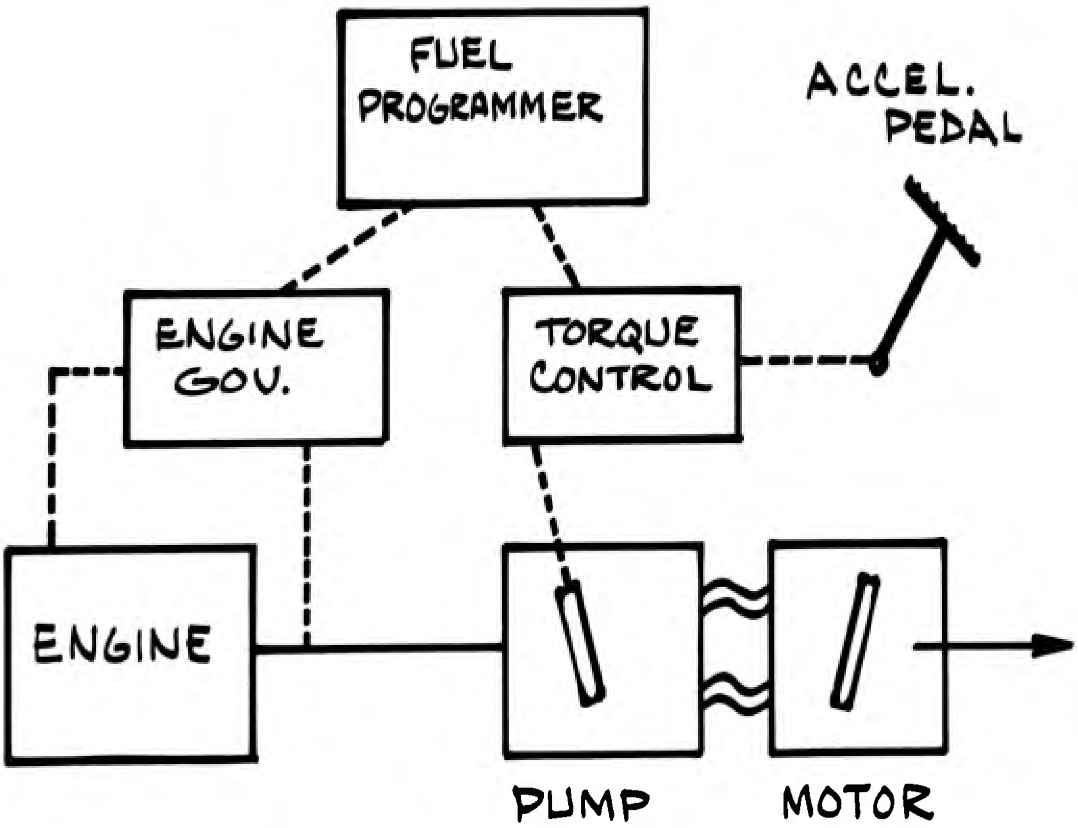


Figure 11-14. Engine Fuel Programming Block Diagram

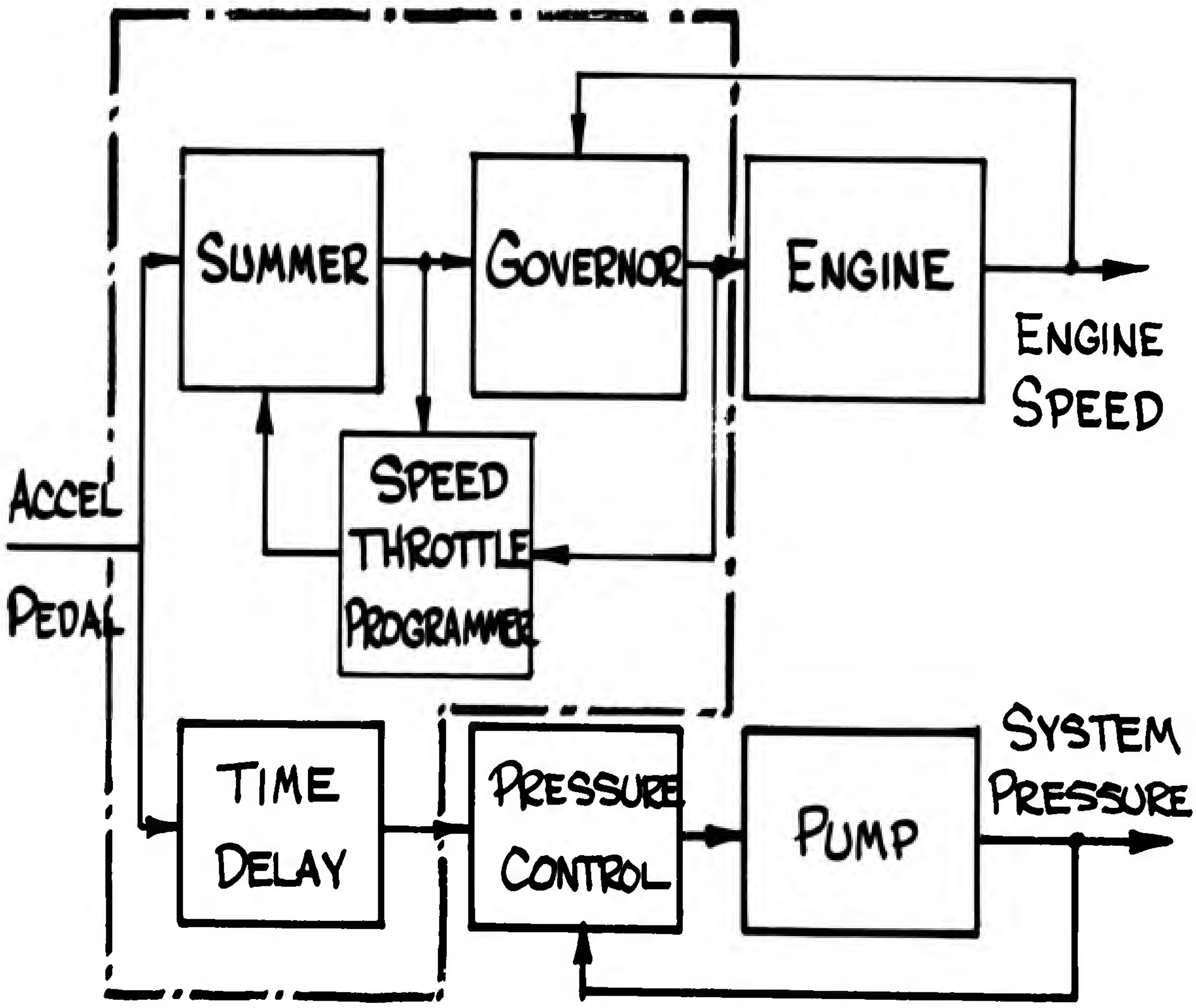


Figure 11-15. Fuel Program Control

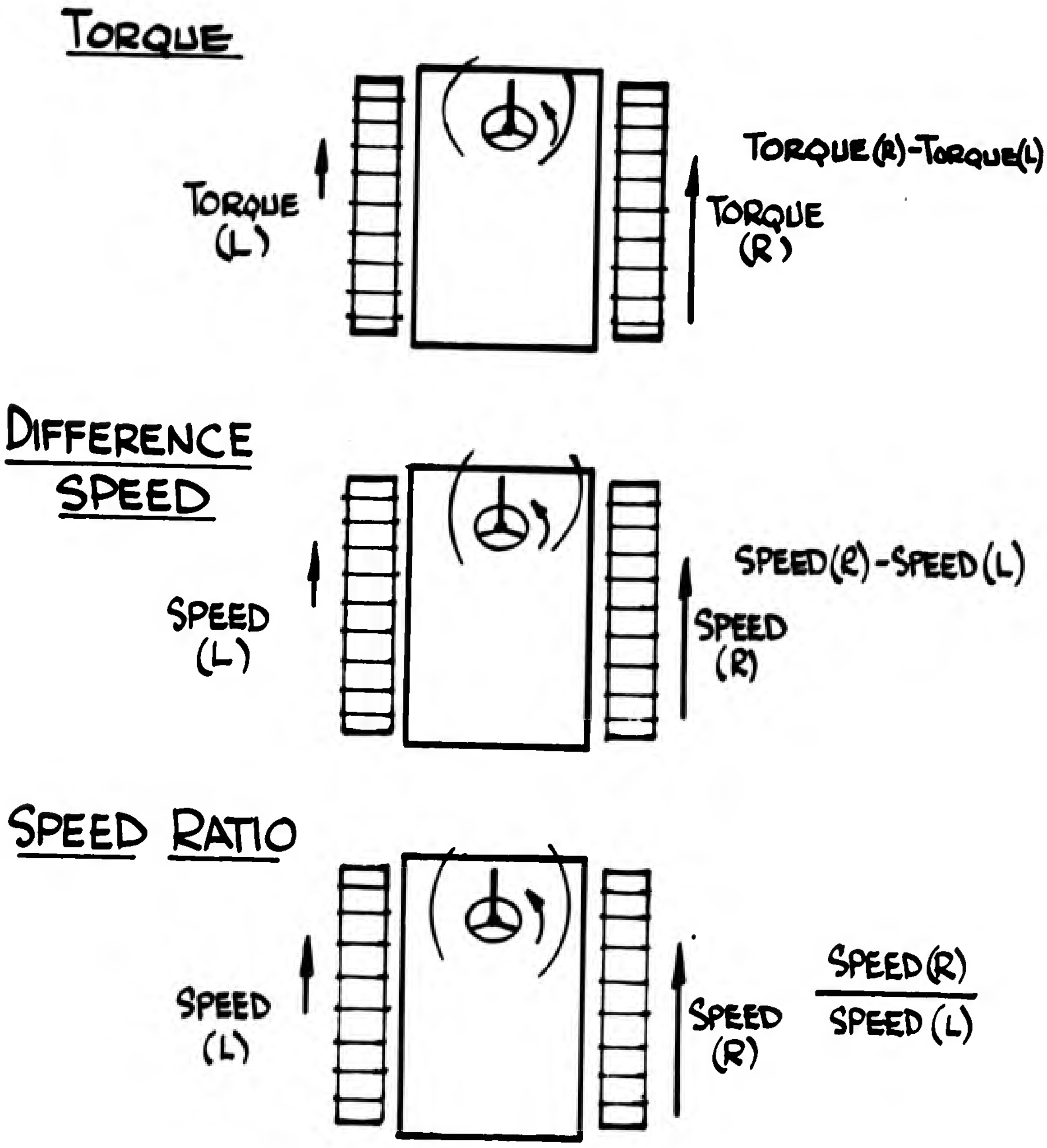


Figure 11-16. Steering Control

As shown in Figure 11-17, the difference speed steering control system receives signals from the steering wheel when it is turned by the vehicle operator. The control system, in turn, signals the control pistons of the hydraulic motors to properly change their respective displacements. This increases the torque and resulting velocity of one track while decreasing torque and velocity of the other. This control is effective throughout the full speed range of the vehicle.

If one hydraulic motor is at maximum displacement and therefore unable to increase with respect to the other, which is decreasing in stroke, "lost motion" linkages allow the one to continue decreasing to achieve the correct speed difference for the turn. In fact, it is possible for the hydraulic motor to attain a reverse torque and consequent reverse track motion with respect to the other track.

The displacement control pistons of the hydraulic motors receive their signals from two distinct sources; namely, the output-driven charge pump and the differential steering control system. The two signals are synchronized, in that difference steering governs if the vehicle is negotiating a turn, and the output-driven charge pump governs when vehicle is moving at infinite radius. The signals from the differential steering control system regulate the relative torques and velocity differences of the two tracks in negotiating turns, and the signals from the output-driven charge pump reduce the stroke of the motors during vehicle acceleration and high velocity operation.

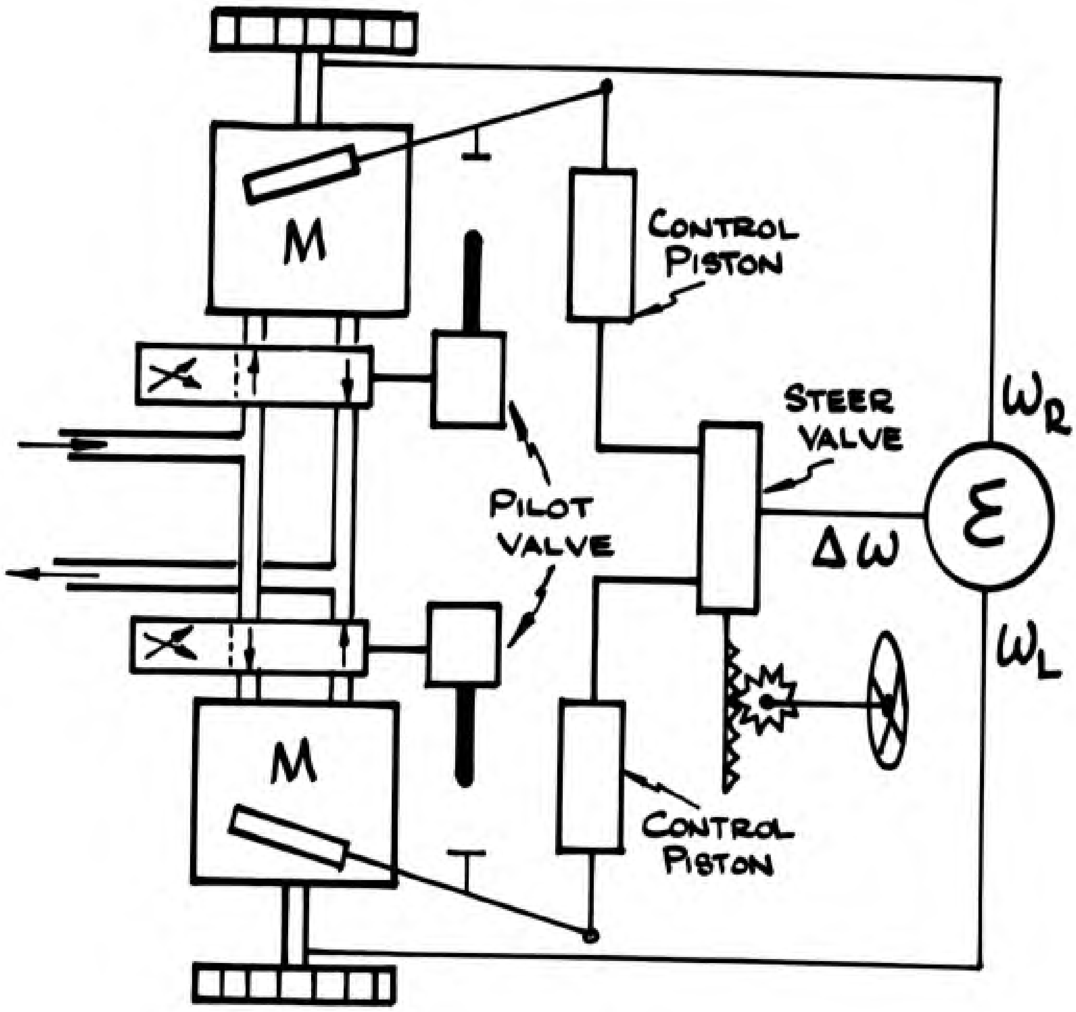


Figure 11-17. Block Diagram LVT Transmission Difference Speed Control Steering

2. Ratio Speed Steering.

Ratio-speed steering is a type of steering in which the position of the steering wheel will dictate the ratio of the speed of one track with respect to the other track. This would show itself in a given turn radius regardless of vehicle speed. It can be accomplished with a control system quite similar to that of difference speed steering. The control system is shown schematically in Figure 11-18. Ratio-speed steering would have certain advantages in vehicle remote control applications where positive prediction of steering would be required without the benefit of operator discretion.

D. System Protection Features.

1. System Overpressure Protection.

The proposed hydrostatic transmission system is protected against fluid overpressure by the charge pressure relief valve. As system pressure reaches the maximum limit of the pressure relief valve because of heavy vehicle loading conditions, the valve will open and pass fluid from the high to low pressure port. As long as the heavy-load condition exists, the relief valve will modulate to limit the system pressure to whatever the relief valve setting is.

2. Vehicle Overspeed Protection.

The transmission and engine are protected by a vehicle overspeed control. Vehicle speed is sensed by the output-driven charge pumps of the transmission. The displacements of the motor units are controlled by the signals

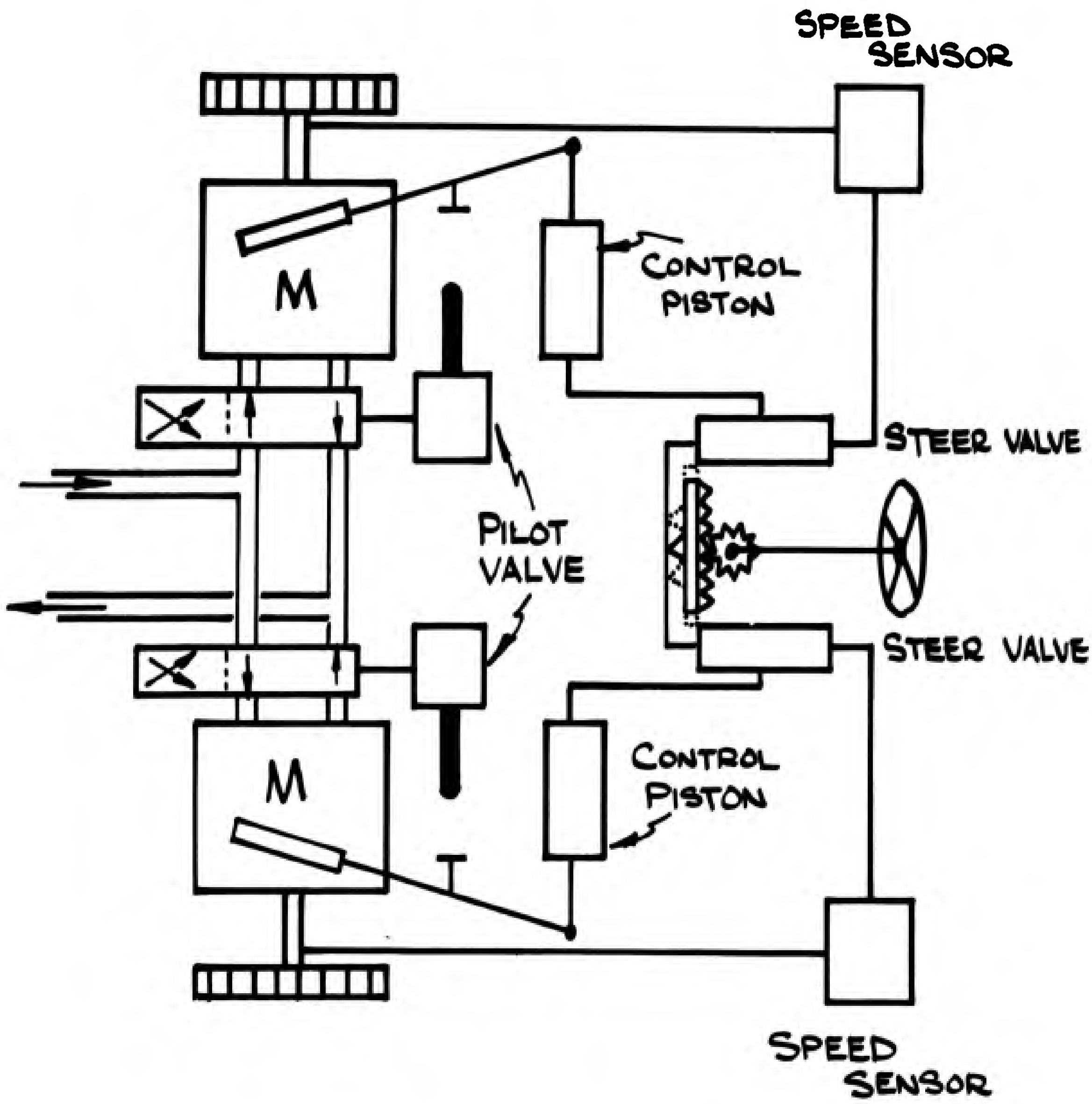


Figure 11-18. Block Diagram LVT Transmission Ratio Speed Control Steering

of vehicle speed received by the motor-driven charge pumps. As vehicle speed increases, the displacement of the motor units decreases. As vehicle speed increases beyond the maximum design speed of the vehicle, the charge pumps signal this condition and cause the motor displacements to increase. The vehicle will decrease in speed by the hydrostatic transmission's functioning as a brake until the vehicle decreases to maximum design speed.

3. Engine Overspeed Protection.

The engine is protected from overspeed conditions during vehicle braking by the engine overspeed control valve. As the engine speed increases over a specified value, the engine overspeed valve biases the control piston of the hydraulic pump to override the control effects of the operator which try to increase the displacement of the transmission hydraulic pump. This action will decrease the speed of the motoring pump and the directly coupled engine to the maximum value. If more braking capacity is required on the vehicle after the engine is at maximum speed and maximum frictional drag, the hydraulic fluid is directed through the working pressure relief valve for subsequent energy dissipation.

11.7.7 Component Construction.

A. Hydraulic Units (See Figure 11-3).

There are three major subassemblies in the crosshead designed hydraulic unit; namely, the swashplate, the crosshead assembly, and the piston port plate and block assembly. The important features of this type of design are the elimination of contact stress limits, the elimination of piston cocking problems, and the ability to vary the swashplate angle for displacement adjustment.

The cylinder block is angled at 18° relative to the driving shaft. The piston connecting rods, instead of entering fixed sockets in a swashplate, connect into sliding connecting links (sliders) which couple at the other end to a second set of ball-end connecting rods. These terminate in sockets on an adjustable angle swashplate. The sliders move back and forth in cylinders in a rotating crosshead which is connected to the driving shaft. The cylinder block is driven by small bevel gears from the crosshead, and the thrust plate on the swashplate is driven by the connecting rods in a manner to be described later.

Since the piston rods are in line with the pumping cylinders, negligible mechanical side loads occur, and the only torque needed to drive the block is that necessary to overcome friction and inertia forces.

All forces on the sliders acting normal to the driving shaft axis are directed into the crosshead, since the link can transmit no axial force to the crosshead except by friction. Therefore, the crosshead receives all the torque and transmits it to and from the shaft. The crosshead can be mounted in a simple double-row roller bearing set because it is subject neither to the large overturning moments normally found on driving shafts of angled hydraulic units nor to appreciable thrust loads. The small axial forces are transmitted to and from the drive shaft.

The thrust plate on the swashplate assembly slides on the stationary element with negligible mechanical loads because of hydrostatic bearing pockets at each rod end. These pockets are hydraulically connected to the block cylinders by holes through both connecting rods, the slider, and the piston. The hydrostatic bearing pockets are proportioned so as to balance all but a small fraction of the load transmitted into the sliding plate by each rod.

The light forces needed to rotate the thrust plate are transmitted from the crosshead through the connecting rods into a retainer plate fastened to the thrust plate. The retainer plate has partial sockets that hold the connecting rod ball ends loosely in place. It also has fingers that straddle the connecting rods midway along their length. These fingers have just enough clearance to compensate for the slightly elliptical motion of the rod. As the thrust plate tends to lag behind the crosshead, the rods engage the fingers on the retainer plate and drive the assembly in synchronism.

The non-rotating portion of the swashplate is mounted on offset trunnions so that the loads produce a moment of constant sign about the trunnion axis. A control cylinder of conventional design adjusts the swashplate through an angle of plus or minus 18^{o} from the vertical, resulting in displacement varying from 100% to zero.

In order to minimize leakage at the thrust plate, no tensile loads can be tolerated in the connecting rods. Therefore, a minimum mechanical closing force between the thrust plate and swashplate can be maintained. The piston and rod assemblies are maintained in compression on the discharge stroke by a pressure differential from the return port to the case. This differential is developed by means of the scavenge gear pump that takes its suction from the case of the hydraulic units.

The drive shaft is supported on a single angular contact ball bearing at one end and by its spline engagement with the rotating crosshead at the other. An angular contact bearing is required because of the large thrust force applied during proof pressure testing. During normal operation, the light thrust load due to case pressure is just enough to provide smooth bearing operation, minimizing noise and wear.

The cylinder block, due to the relatively small side forces applied to it, can be supported by a sleeve bearing at its center. The block is designed so that the cylinder bores can be sleeved if this proves necessary. However, one of the successful high temperature motor designs developed by Sundstrand utilizes a cast iron cylinder block, and the absence of roller bearings on the block of this motor permits use of cast iron for the whole block. If iron proves successful in this new application, sleeves can be eliminated.

The valve that the cylinder block rotates against is a separable plate. In this way the optimum material can be chosen for this function without compromising the design of the end cap. The valve plate is hydraulically balanced on both its faces and is held down by the cylinder block clamping spring and the hydraulic force unbalance between the cylinder block and the valving surface.

The inlet and discharge ports approach the valve plate ports in a tangential fashion so that the fluid undergoes a minimum change in direction as it enters and leaves the cylinders. Since the piston axial velocity is sinusoidal while the block rotates at constant speed, and since the piston velocity also varies with displacement, it is obvious that the proper vector relationship can exist during only one portion of the piston motion in each direction and for only one condition of speed and displacement. It has been found by test that a design optimized for maximum piston velocity markedly reduces noise and effects an improvement in performance.

In this design, where the hydraulic thrust loads can conveniently be dealt with by hydrostatic bearings, the only heavily loaded anti-friction bearings are the roller bearings on the rotating crosshead.

The approach used to overcome differential expansion problems in areas of critical sliding fits is to use the same materials, or materials with similar coefficients of expansion, on both sides of the fit. All valves are steel, sliding in steel sleeves, the sleeves being fitted into steel housings. Where different materials are needed for bearing purposes, thin-walled bushings are used, as in the cylinder bores. If proper proportioning is used in the bushings, the bushing material exerts a negligible effect on the cylinder's diameter as temperature changes. Since the block and piston are both of the same material, no problems from differential expansion occur unless differential temperatures exist.

B. Semi-Final Drive.

The integral semi-final drive which is located at each output provision of the transmission is a single spur gear set with 1.47:1 ratio. Lubrication of the gearing will be done by a combination of splash lube and forced lubrication. The source of the lubricant will be the hydrostatic transmission system fluid. (In all probability, this and the final drive will be one reduction unit.)

11.7.8 Technical Data.

A. Installation and System.

1. Transmission Mounting (Ref. Drawing SK-4807.)

   a. Power Input - Shaft coupling

   b. Power Output - Flange mounting

   c. Transmission - Three-point suspension

2. Transmission Weight

   a. Dry - 1048 lbs.

   b. Fluid - 48.5 lbs.

3. Transmission Envelope (Ref. Drawing SK-4807.)

4. Transmission Fluid Pressure

   a. Maximum Working Pressure - 6000 psi

b. Rated Working Pressure - 3000 psi

c. Back or Return Pressure - 250 psi

d. Maximum Δ Pressure - 5750 psi

5. Temperature Data

a. Ambient Temperature Range: -65° to 125° F

b. Fluid Temperature Range: -65° to 250° F

6. Heat Exchanger Requirements (Based Upon Braking Requirements)

a. Dissipate a maximum of 6500 BTU/min.

b. Cooling Circuit Flow: 40 gpm nominal
60 gpm maximum

c. Maximum allowable exit temperature of oil at Heat Exchanger - 200° F

d. Aerated oil - 20 gpm out of 60 gpm

e. Cooling Circuit Pressure - 50 psi nominal

7. Fluid Reservoir Assembly (furnished by Ingersoll)

a. Capacity - 8 gals.

b. Vented to atmosphere

8. Fluid Type - SAE #10

9. Working Pressure Relief Valve Setting - 6000 psi maximum

10. Return Pressure Relief Valve Setting - 250 psi minimum

11. Output-Driven Charge Pump

a. Rated Flow - 2 gpm each

b. Rated Discharge Pressure - 250 psi

12. Input-Driven Charge Pump

a. Rated Flow - 40 gpm

b. Rated Discharge Pressure - 250 psi

13. Input-Driven Scavenge Pump

a. Rated Flow - 60 gpm

b. Rated Discharge Pressure - 50 psi

14. Heat Rejection Data (Ref. Figures 11-20 and 11-21)

15. Transmission Efficiency (Ref. Figure 11-22)

B. Transmission Components.

1. Variable Displacement Pump

a. Manufacturer: Sundstrand Aviation Division

b. Displacement: Variable from 27.0 to 0 cu. in. per rev.

c. Outlet Pressure: 6000 psi maximum

d. Inlet Pressure: 250 psi minimum

e. Housing Material: Aluminum

f. Speed Range: 0 to 3000 rpm

g. Rotation: Single direction, once selected

h. Life: 1000 hours

i. Rated Flow Delivery: 172 gpm

2. Variable Displacement Motor

a. Manufacturer: Sundstrand Aviation Division

b. Displacement: Variable from 54.0 to 0 cu. in. per rev.

c. Outlet Pressure: 6000 psi maximum

d. Inlet Pressure: 250 psi minimum

e. Housing Material: Aluminum

f. Speed Range: 0 to 2800 rpm

g. Rotation: Reversible

h. Life: 1000 hours

i. Rated Torque Output: 57,600 in. lbs.

3. Semi-Final Drive Gearbox Assembly

a. Manufacturer: Sundstrand Aviation Division

b. Single step spur gear 1.47:1 ratio

C. Transmission Durability and Design Life.

1. Transmission will have the ability to complete a 24-hour battlefield day, which consists of 40% idle, 40% cross-country operation, and 20% secondary road operation, without power package malfunction.

2. Transmission durability shall be 5,000 miles, consisting of 3,300 miles of cross-country operation with equal amount of running at 50% and 100% of rated engine load at 5-7 mph and 1,700 miles of secondary road operation with equal amounts of 55% and 80% of rated load at 15-18 mph.

3. Transmission will be capable of 25 hours continuous operation at full engine load at maximum speed for water operation.

4. Transmission will be capable of operating one thousand hours between overhauls. Approximately 400 hours will be at idle.

D. Sustained Downgrade Braking.

The suggested cooling capacity of the hydrostatic transmission is based upon

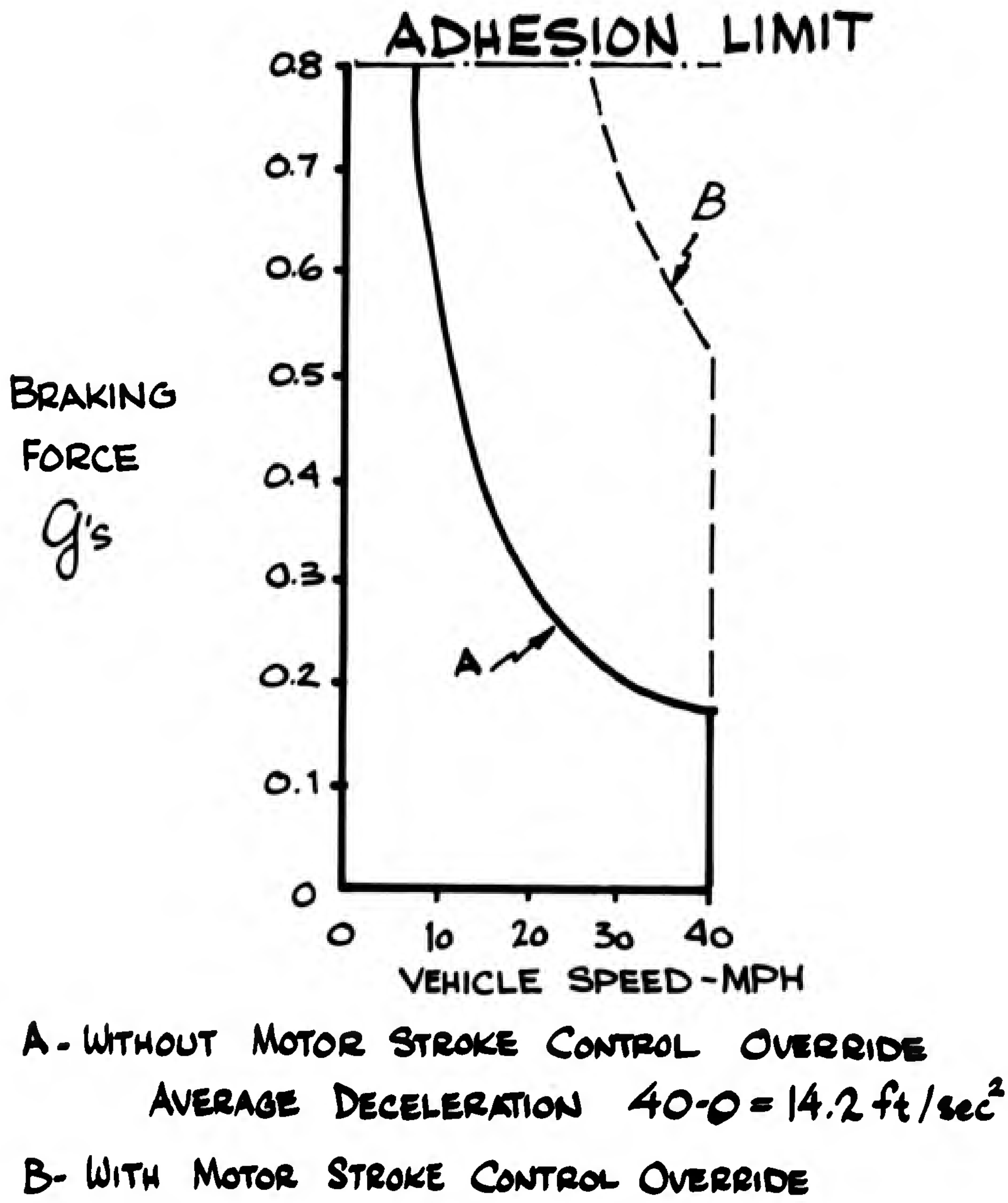


Figure 11-19. LVT Transmission Brake Force Curve

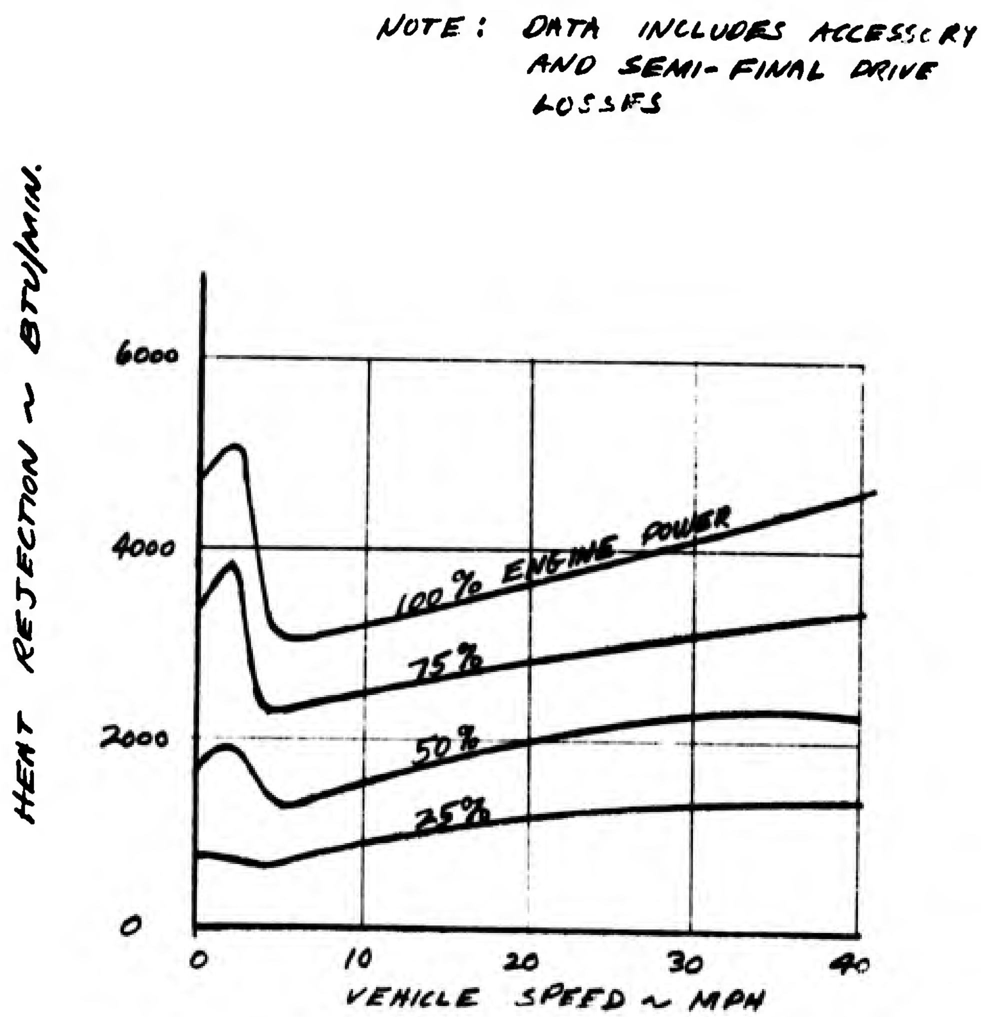


Figure 11-20. Hydrostatic Transmission Heat Rejection Vs. Vehicle Speed

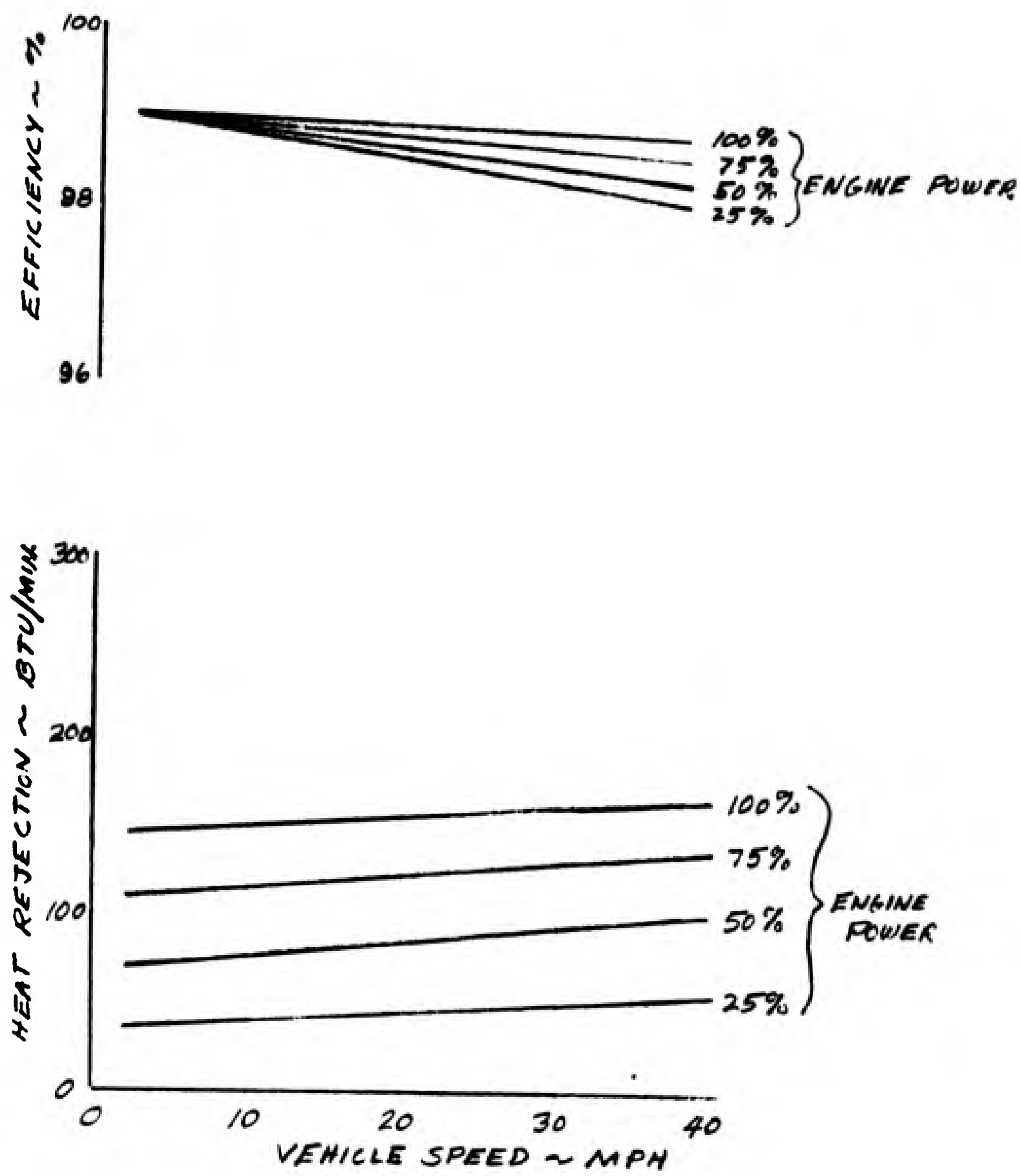


Figure 11-21. HST Semi-Final Drive Efficiency and Heat Rejection Vs. Vehicle Speed

repetitive braking as specified in the proposal. However, the heat exchanger will not be adequate for sustained downgrade braking of the vehicle at high speeds. With the 6500 BTU/min. heat exchanger, it will be possible to brake the vehicle at a speed of 5 mph down a 70% grade or 19 mph down a 35% grade without overheating the transmission fluid. If braking at higher velocities downgrade is required, the heat exchanger capacity must be increased.

11.8 Gar Wood Industries, Inc. Proposal of Hydrostatic Transmission for LVT Vehicle.

Gar Wood Industries has proposed a split-type hydrostatic drive system in which the pumps and motors are separated and connected by piping. Their proposed drive employs basic units which they presently have in operation on an operating "Goer"-type vehicle. The pumps and motors are the Variacs variable-volume piston design.

11.8.1 Component Outline.

A. Power Unit (Figs. 11-24. 11-25, 11-26, and 11-29).

1. Variable volume pumps - two, one to drive each track. They are controlled separately. Displacement is 7.24 cu. in.

2. Servo Unit - hydraulically activated and manually controlled. These units are power assists to control the volume output of the pumps.

3. Control Pump - power source for servos.

4. Supercharge Pump - provides make-up of leakage in transmission circuit.

5. Peaking Relief Valve - absorbs peak pressure surges in transmission circuit.

6. Input Gear Box - provides a splitter gear box and a 1.25:1 increaser for pumps.

7. Retardation Limiters - prevent overspeeding of engine during braking cycle.

B. Two Sprocket Drive Units (Figs. 11-25, 11-26, 11-27, and 11-28).

1. Variable Volume Motors - two, with 23.7 cu. in. displacement each.

2. Servo Unit - hydraulically activated and compensator controlled. It derives its power source from system pressure. A single unit controls both motors.

3. Compensator - rate build-up type. Regulates displacement by sensing pressure.

4. Out-Put Gear Box - provides a means of coupling the two motors together and providing a speed reduction.

11.8.2 Operation.

The pump controls are connected to a single stick with suitable linkage so that a forward motion on the stick strokes the servo to change the displacement of both pumps simultaneously. The same is true of a reverse motion on the stick. A side motion changes the relative relation of one pump to the other by advancing one pump control and retarding the other. This displaced relation of the pump controls increases as the side displacement of the stick is increased.

In the event the operator places the control in an advanced position such that more power would be required than the engine can produce, the override compensator automatically limits the pump's input torque, thereby preventing stalling of the engine.

Braking is accomplished by returning the stick to neutral. The motors become pumps, and the pumps become motors. Thus, dynamic braking is effected. In order to prevent overspeeding of the engine, retardation limiters are placed in the lines. These units allow free flow of oil in the drive and in the return line as long as the pressure is low. If the pressure in the return line exceeds an existing level, it is automatically throttled, and power is dissipated before it can react on the engine.

The motors are normally at their minimum displacement position and will move automatically to their maximum displacement for maximum torque delivery.

A combination parking and emergency brake is mounted on the output shaft of one motor in each sprocket drive. It is of the disc type and requires 800 psi to hold the vehicle, equivalent to the output torque.

Filtration requirements are for an intake of 160 gpm at 10 microns, preferable, but 25 microns is acceptable.

Reservoir capacity should be about 60 gallons.

The heat exchanger should be able to handle 80 hp continuously at 120°F over ambient temperature.

11.8.3 Circuit.

Note that the circuit has been set up to provide for the following (See Figure 11-23):

1. Tow starting.
2. Foot controlled brake.
3. Retardation limited in forward motion only.

By adding a small pump to one motor unit and connecting its output to the control circuit, a tow start can be effected, because the operator has control of the pump.

The brake valve will allow a conventional stop when required. This brake valve will be locked out during operation of the vehicle. The easiest way to do this is to latch the brake pedal when the pedal is depressed, and this latch can be kicked off when deceleration is required without feed-back to the engine.

Releasing the brake pedal will allow unit to free-wheel. Depressing the brake pedal will effect deceleration proportional to foot pressure.

11.8.4 Design and Performance Advantages.

Listed below are a few of the many advantages of a hydrostatically driven vehicle and the Gar Wood Ultra-Static drive in particular:

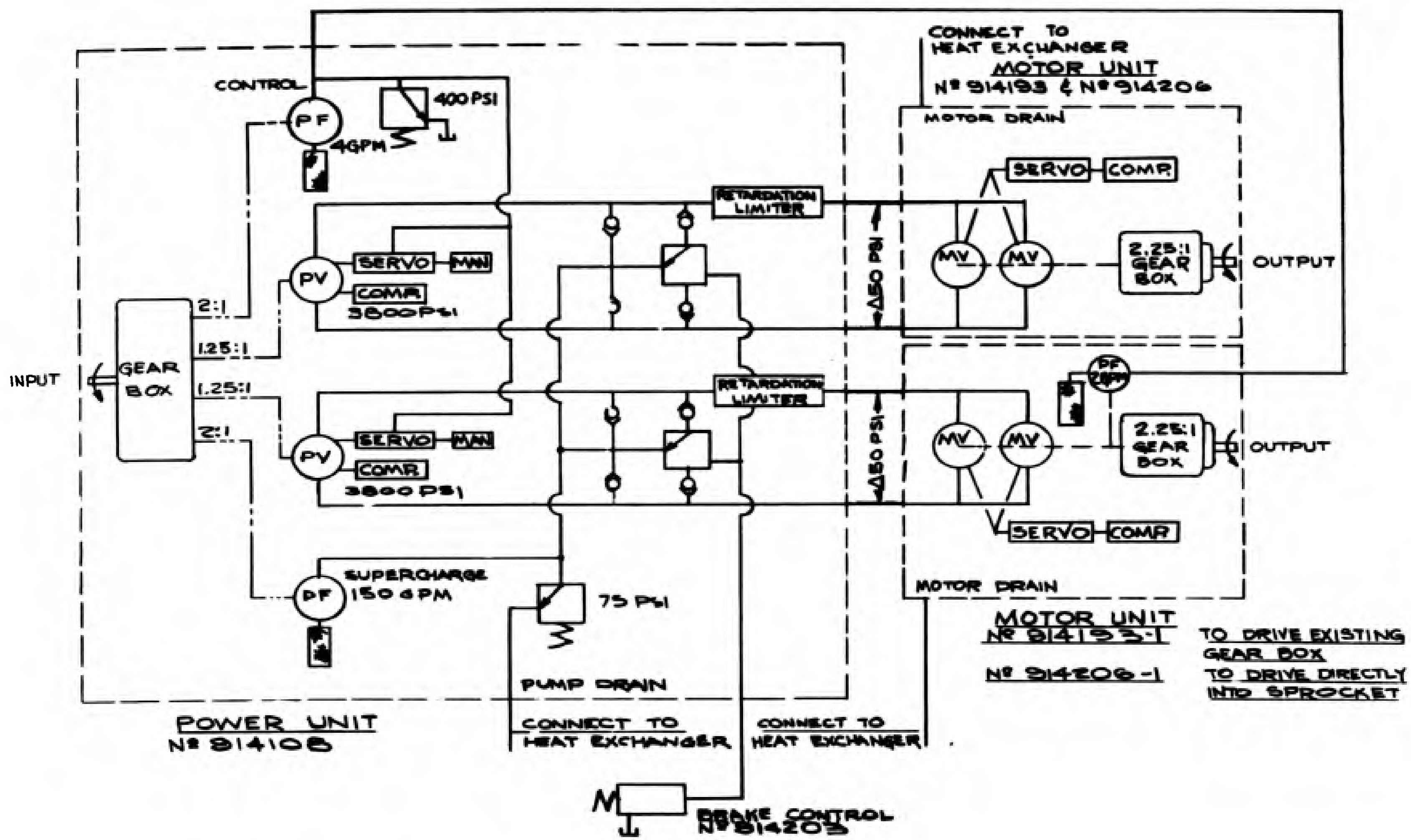


Figure 11-23. Hydrostatic Drive LVT Circuit Diagram

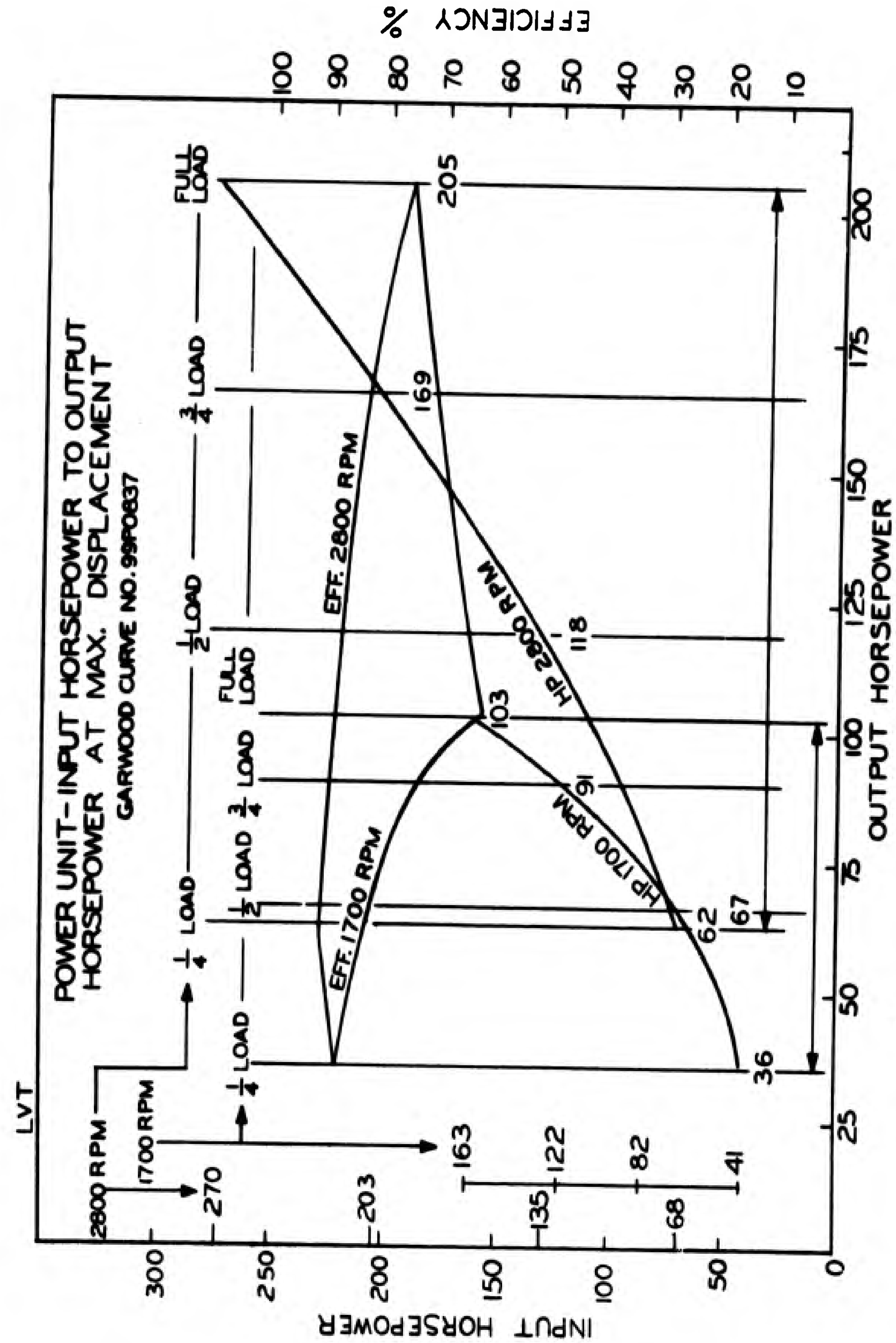


Figure 11-24. Power Unit - Input Horsepower to Output Horsepower at Maximum Displacement

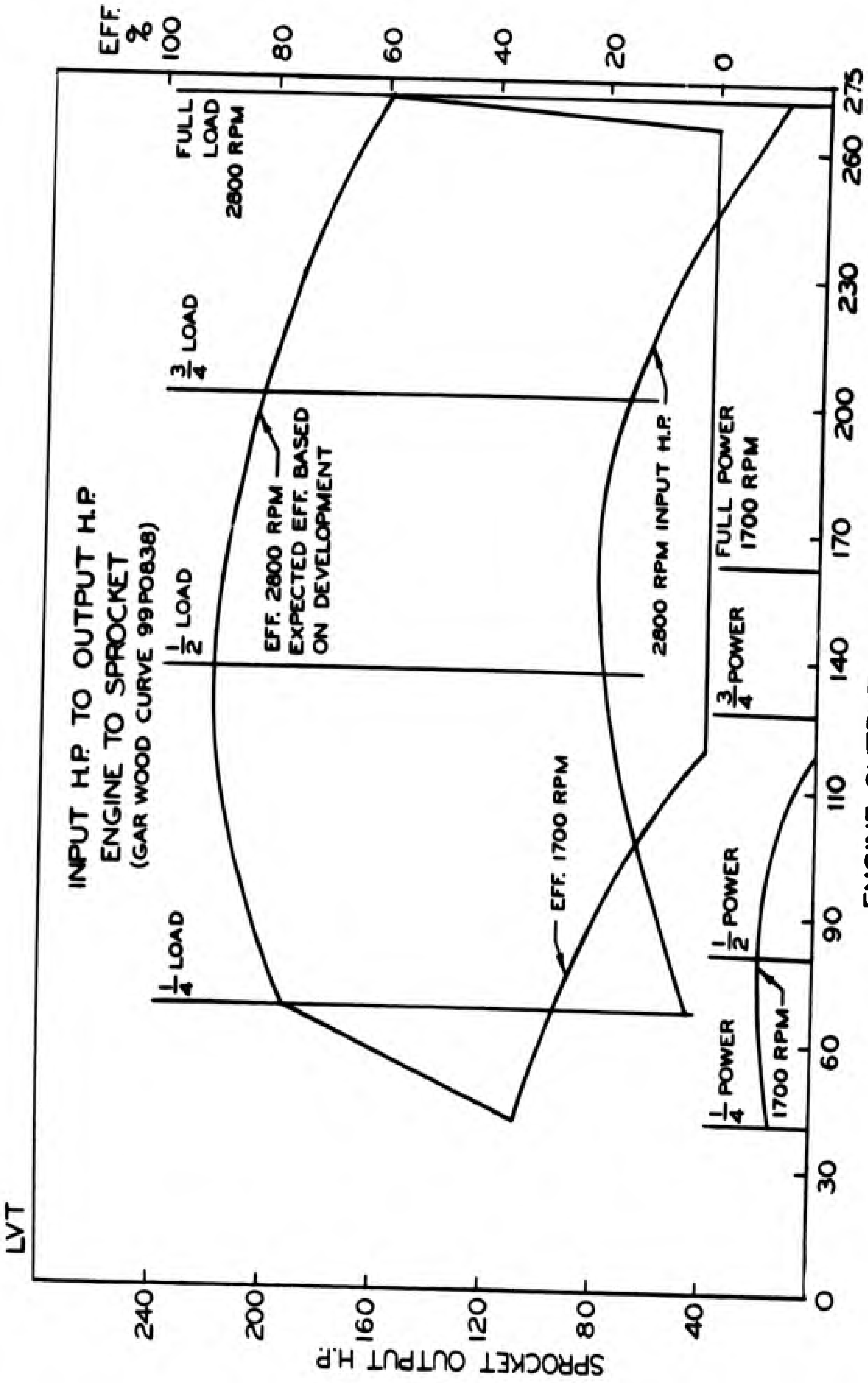


Figure 11-25. Input HP to OutPut HP, Engine to Sprocket

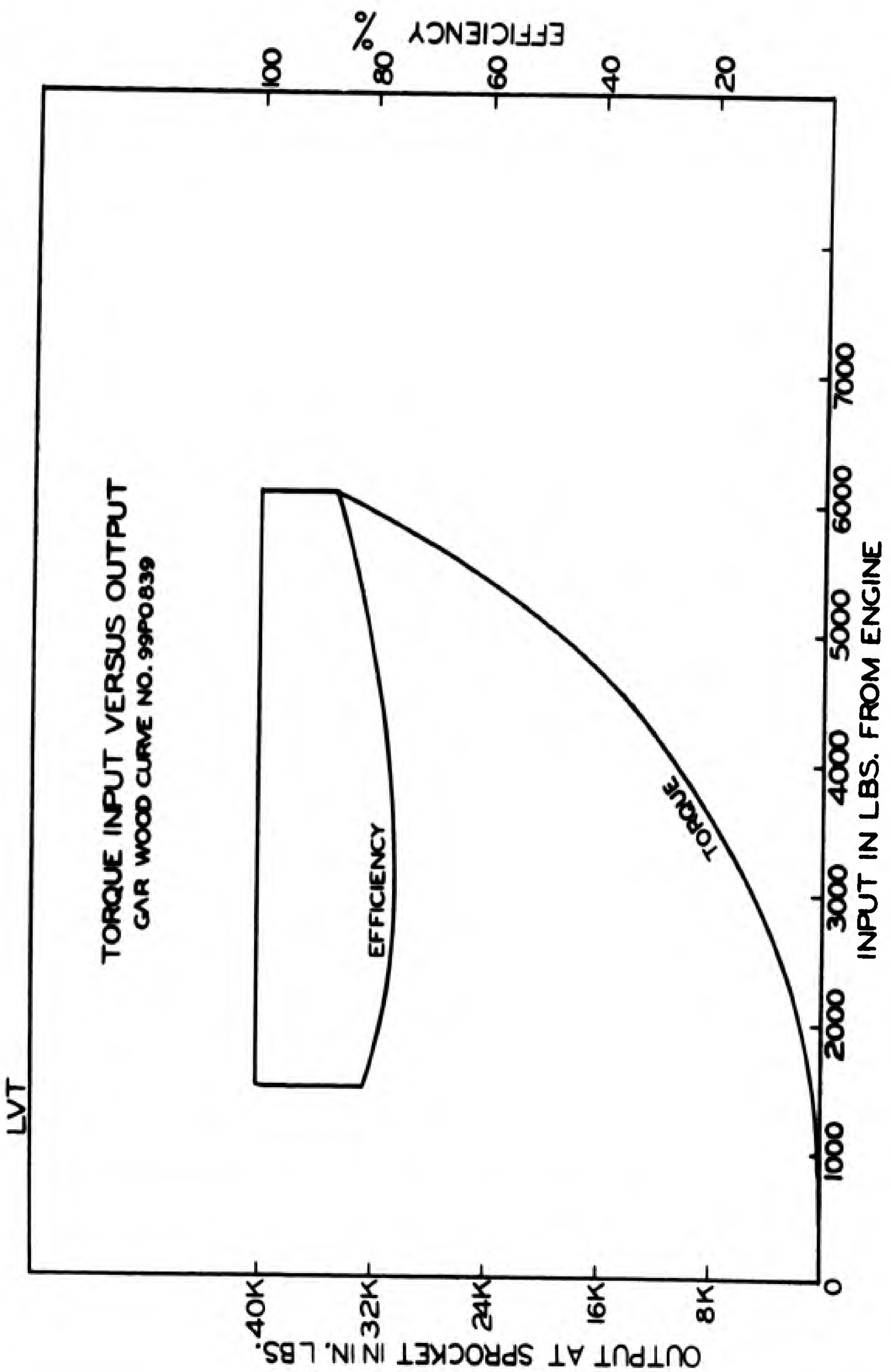


Figure 11-26. Torque Input Versus Output

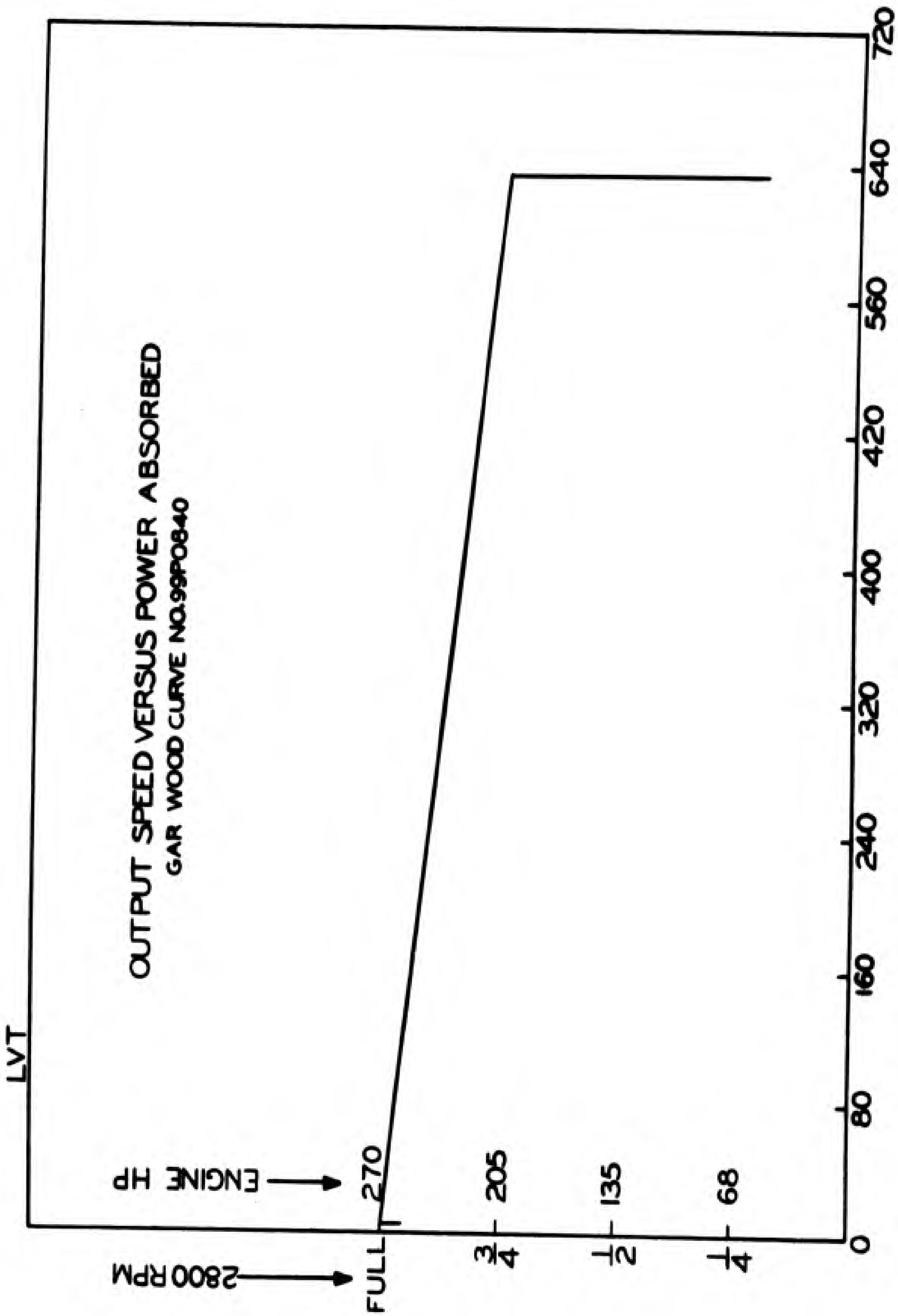


Figure 11-27. Output Speed Versus Power Absorbed

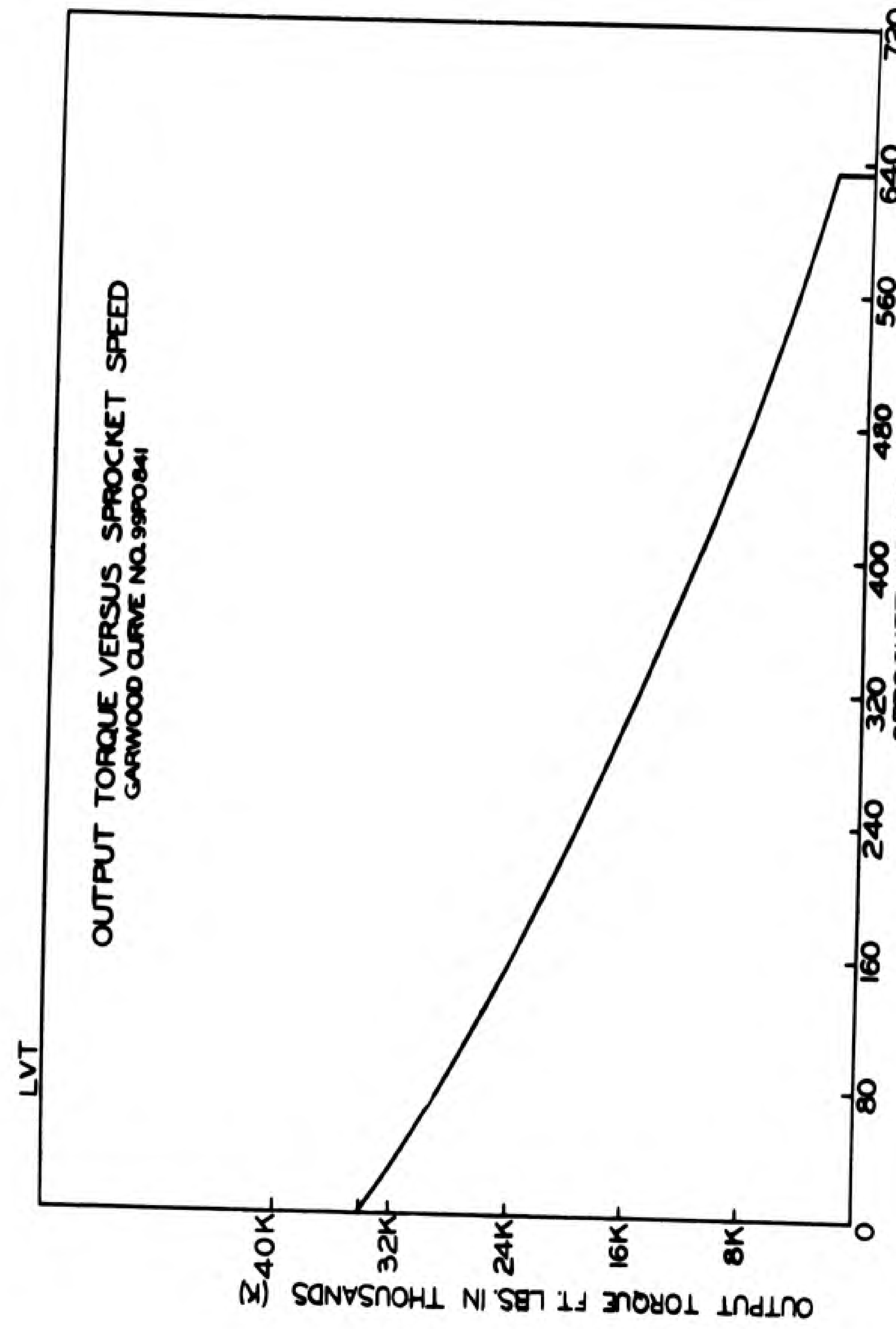


Figure 11-28. Output Torque Versus Sprocket Speed

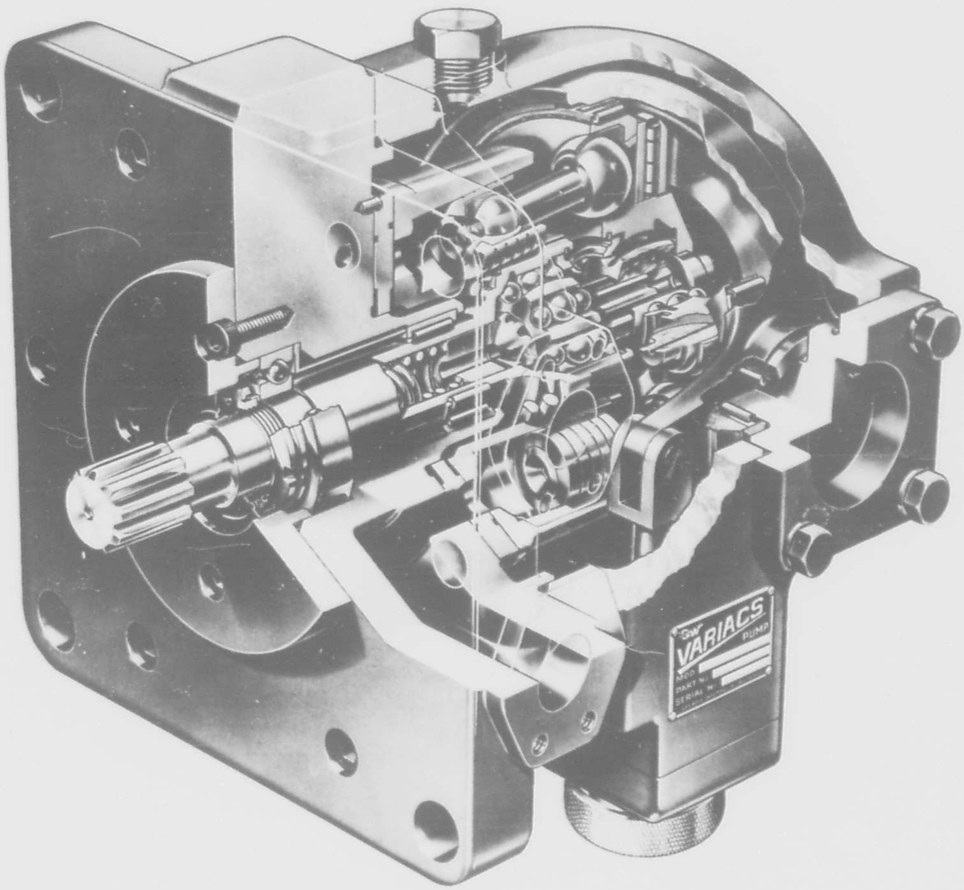


Figure 11-29. Gar Wood Variacs, Variable Volume Piston Pump

1. Design flexibility given almost absolute freedom of vehicle silhouette, location of center of gravity and payload, ground clearance, and placement of power source.

2. Parts interchangeability offers efficiencies and economies in maintenance and service parts inventories.

3. Full torque is available at all engine rpm.

4. The installation is simple and versatile. (Note in Figure 11-31 how little room the Ultra-Static drive requires in comparison with conventional drives in Figure 11-30.)

5. There are no drive lines or axles. The whole wheel drive unit is inside the tire rim, in space usually reserved for brakes and bearings. These are integral with the wheel drive units.

6. Servicing is simplified. Units can be replaced as cartridges.

7. The drive is completely stepless through a wide range of speeds and torques. This means power on the wheels with just the torque required to power the vehicle and its payload.

8. Since there is no axle, it provides better ground clearance in many vehicles.

9. Considerable weight savings are possible in the transmission package, plus additional savings in structure weight.

10. Space saving enables more payload to be carried or a smaller vehicle to be utilized.

11. Vehicle retardation can be selected as required.

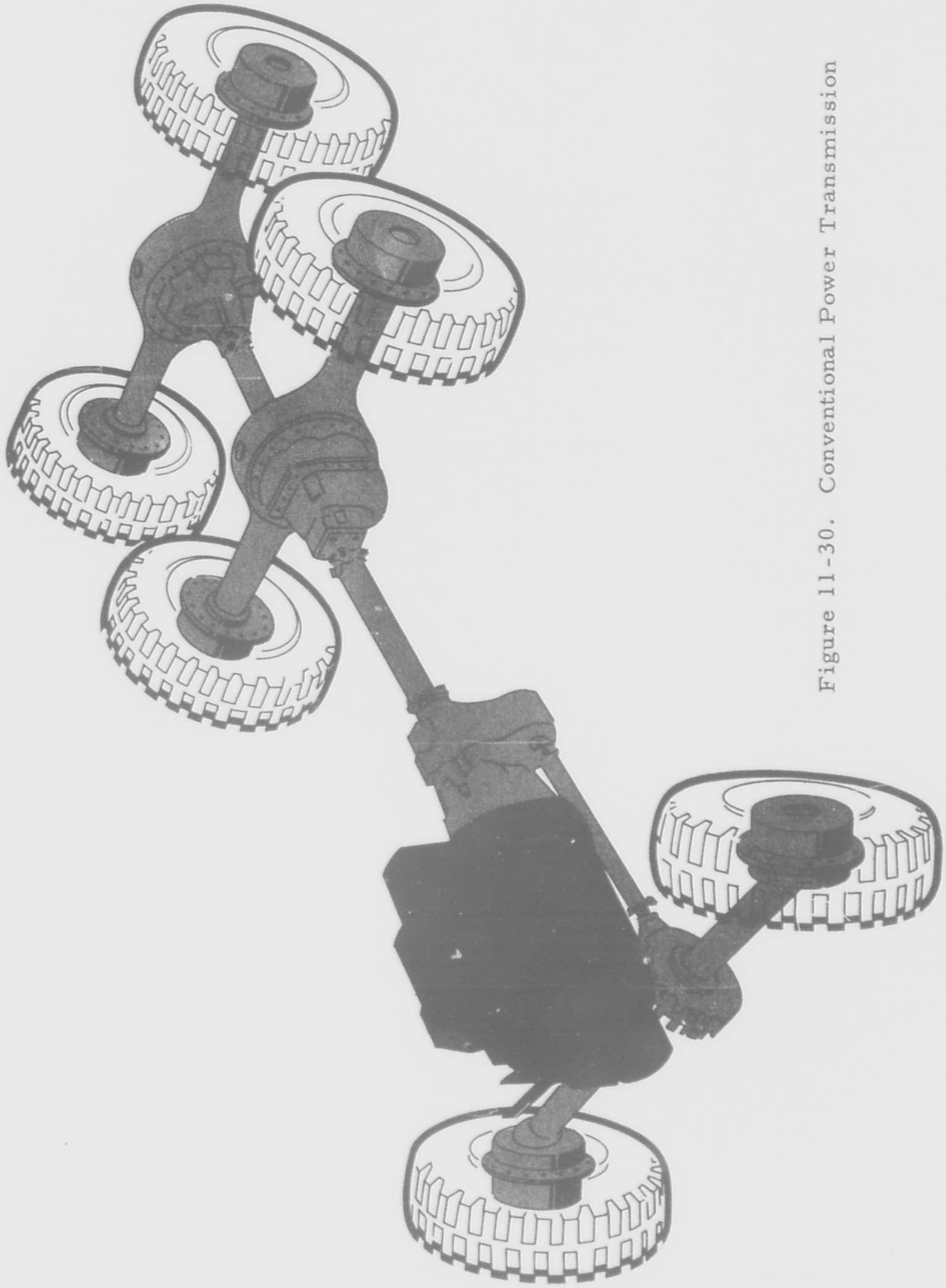


Figure 11-30. Conventional Power Transmission

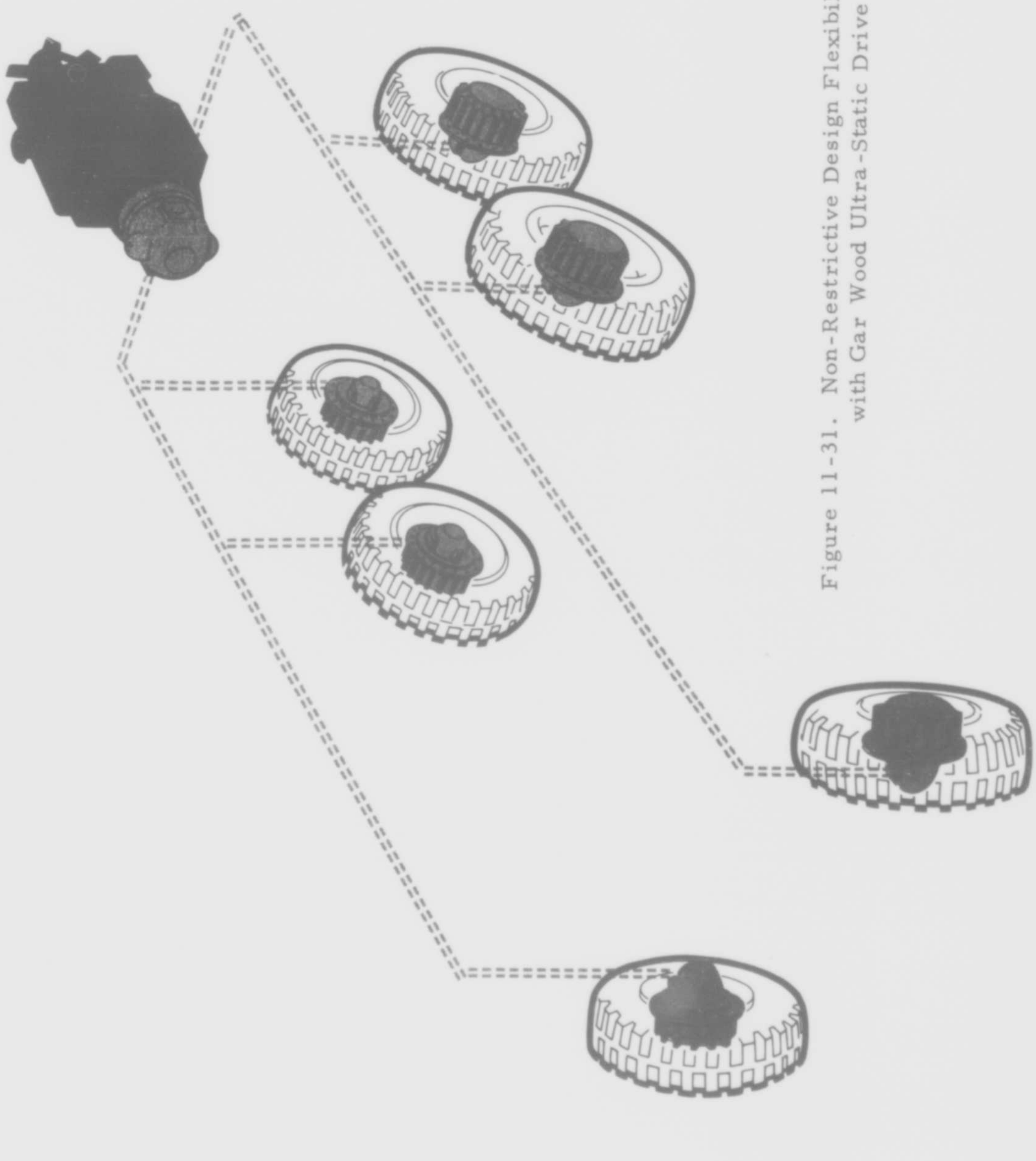


Figure 11-31. Non-Restrictive Design Flexibility Possible with Gar Wood Ultra-Static Drive

12. Creep speeds can be employed without overloading or stalling the engine, even under full load.

13. Higher efficiency is possible (at full power requirements) than with other types of transmissions.

14. Braking is controlled through dynamic braking of the system itself, which eliminates conventional independent braking systems.

15. There is positive drive at all times in any speed range. The slippage inherent in a torque converter is virtually non-existent in the Ultra-Static drive.

16. Steering clutches, required on mechanical drive vehicles, can be omitted.

17. Due to dynamic braking, separate track brakes are not required.

11.9 Vehicle Configurations with Various Hydrostatic Drive Arrangements.

11.9.1 General.

Sundstrand's integral-drive and Gar Wood's split-drive system have been incorporated in various vehicle configurations utilizing both two- and four-sprocket drive arrangements. (A two-sprocket drive has one drive sprocket per track, and a four-sprocket drive has two drive sprockets per track, whether one or two motors are used per drive sprocket.) The distinct advantages and disadvantages of the two- and four-sprocket drive arrangements are outlined below:

11.9.2 Two-Sprocket Drive Versus Four-Sprocket Drive.

The two-sprocket drive incorporates one or two hydraulic motors per sprocket drive unit. They drive the vehicle tracks through reduction gears. Currently available hydraulic motors have torque ratings which generally necessitate the use of two drive motors per sprocket in the two-sprocket drive arrangement for large, high-performance tracked vehicles. Four-sprocket drive units utilize only one drive motor per sprocket drive. In the two-sprocket drive arrangement, the larger torque required per sprocket, in comparison with a four-sprocket arrangement, will require the use of larger drive sprockets in order to accommodate one large or two smaller drive motors within the sprocket hub, the optimum motor location. With maintenance of established vehicle approach and departure angles, the use of larger drive sprockets instead of the smaller ones required by a four-sprocket arrangement will require the use of larger road wheels and a longer track, both of which add unnecessary weight to the vehicle. A further disadvantage in the use of larger drive sprockets and resultant larger road wheels is the fact that the top of the track channel must be raised to provide adequate space for the track. If the vehicle dimensions are fixed, raising the track channel will reduce the usable space within the vehicle. Use of the smaller drive sprocket in the four-sprocket drive arrangement allows the lowest possible track channel location.

The use of a two-sprocket drive limits the possible optimum vehicle trim to something less than could be achieved with a split hydrostatic drive system and a four-sprocket drive arrangement. The reason is that the two-sprocket drive concentrates larger weights at two drive sprocket locations, whereas in the four-sprocket drive arrangement, the weight of the drive sprockets can be located at the four corners of the vehicle. This distributes the weight of the drive motors and sprockets evenly about the center of gravity or buoyancy of the vehicle. The two-sprocket drive arrangement will approach the weight distribution advantages of the four-sprocket arrangement if the sprocket drives are located at one end of the vehicle and the prime mover and power pump are located at the other end. This arrangement has disadvantages both in hydraulic plumbing and in restricting the optimum vehicle prime mover (engine) location. The use of a four-sprocket drive permits the vehicle prime mover to be placed in the optimum location, because only the vehicle trim condition created by the prime mover location need be compensated. This correction can be readily accomplished.

11.9.3 Hydraulic Plumbing.

There will be little, if any, advantage or disadvantage of four-sprocket as compared with two-sprocket drives in regard to hydraulic plumbing.

Plumbing design for a hydrostatic drive system is based on maintaining a given maximum fluid velocity in the system regardless of the arrangement of the drive components. In the two-sprocket drive arrangement, the maximum fluid flow in GPM to the sprocket drives will be double the maximum fluid flow to the individual sprocket drives in a four-sprocket arrangement of equal horsepower and operating pressure. The greater flow of the two-drive system will require larger fluid conduits to maintain the maximum fluid velocity required. The use of larger fluid conduits will require the use of larger fittings and probably more fittings, because larger conduits are harder to fabricate and route than smaller conduits. Even though the double flow requirements of the two-sprocket drive will require larger fluid conduits, they will not necessarily be double the size of those for the four-sprocket drive arrangement. However, experience with tubing and fittings in the size ranges which will be required has shown that, for the same pressure, increasing the conduit size to the next available commercial size will increase tubing weight approximately 50% and fitting weight approximately 80%. This makes an average of 65%, with the difference increasing rapidly as the conduit sizes increase. The two-sprocket drive arrangement, when closely coupled; i.e., prime mover and sprocket drives located at the same end of the vehicle, will probably have a total plumbing weight slightly less than the four-sprocket drive arrangement, but it is doubtful that this weight difference will ever exceed 10% of the heavier system weight.

Reduced flow rates are important also when components such as valves, filters, etc. are to be considered. These components are affected weight-wise by conduit size in approximately the same way that conduit fittings are.

11.9.4 Summary.

The relative advantages and disadvantages of the two-sprocket drive arrangement and the four-sprocket drive arrangement, when compared with each other, may be summarized as follows:

Two-Sprocket Drive

Advantages:

1. There are fewer units to require maintenance.
2. This arrangement has higher efficiency.
3. There are less lines.

Disadvantages:

1. It is impossible to achieve optimum vehicle machinery arrangement, because of requirements for optimum vehicle trim.
2. Arrangement requires heavier individual sprocket drives, which are harder to handle for maintenance.
3. Loss of one sprocket drive due to malfunction means complete loss of use of vehicle.
4. Larger fluid conduits are required, which require more space and are more difficult to route and fabricate.
5. The maximum internal vehicle cubage and usable space are reduced.

Four-Sprocket Drive Arrangement

Advantages:

1. Optimum vehicle machinery arrangement is possible, without detriment to optimum vehicle trim conditions.

2. Lighter individual sprocket drives enable easier handling in maintenance.

3. Smaller fluid conduits are required. These require less space and are easier to route and fabricate.

4. The loss of one sprocket drive will not disable vehicle completely, but merely reduce vehicle mobility.

5. Arrangement permits maximum internal vehicle cubage and maximum usable space.

Disadvantages:

1. There are more units to require maintenance.
2. There are more lines.
3. The arrangement is less efficient.

The vehicle arrangements in these discussions are based on specific sizes given to us by the manufacturer and are feasible for incorporation into this vehicle. Further discussions are presented below:

11.9.5 Test Bed Arrangement - Rear Sprocket Drive, Single Engine, as Shown by Figure 11-32.

This arrangement is presented to show primarily that if a mechanical power train were chosen for the proposed MAC vehicle, a hydrostatic drive system could be readily installed. This would minimize cost and give a reliable comparison for evaluation. This arrangement provides a fair power train arrangement for trim and a fair usable space arrangement.

Gar Wood Industries pumps and motors are shown in this arrangement. Hydrostatic prototype cost of this configuration would be approximately $42,000.

Sundstrand's cost is approximately $75,000.

11.9.6 Actual Design Arrangement - Rear Sprocket Drive, Single Engine, Integral, as Shown by Figure 11-33.

Of all the arrangements, the one shown on Figure 11-33 (Sundstrand Aviation) is the only integral hydrostatic drive system presented. The pumps, motors, valves, and some controls are incorporated into one assembly. This arrangement reduces line losses. It should also increase the drive's efficiency by this reduction of lines. It does not provide the most ideal arrangement of usable space, however, and does not help the water trim condition of the vehicle.

This particular drive was presented for consideration because it approaches the best arrangement of a hydrostatic drive system for efficiency and eliminates the nuisance of line leaks. This drive must still be developed and tested before it can be seriously considered for incorporation, however. Its weight would be approximately 1100 lbs. and its prototype cost approximately $75,000. If it were proved successful, its production cost of $11,000 to $19,000 would be competitive with those of other conventional mechanical drive systems.

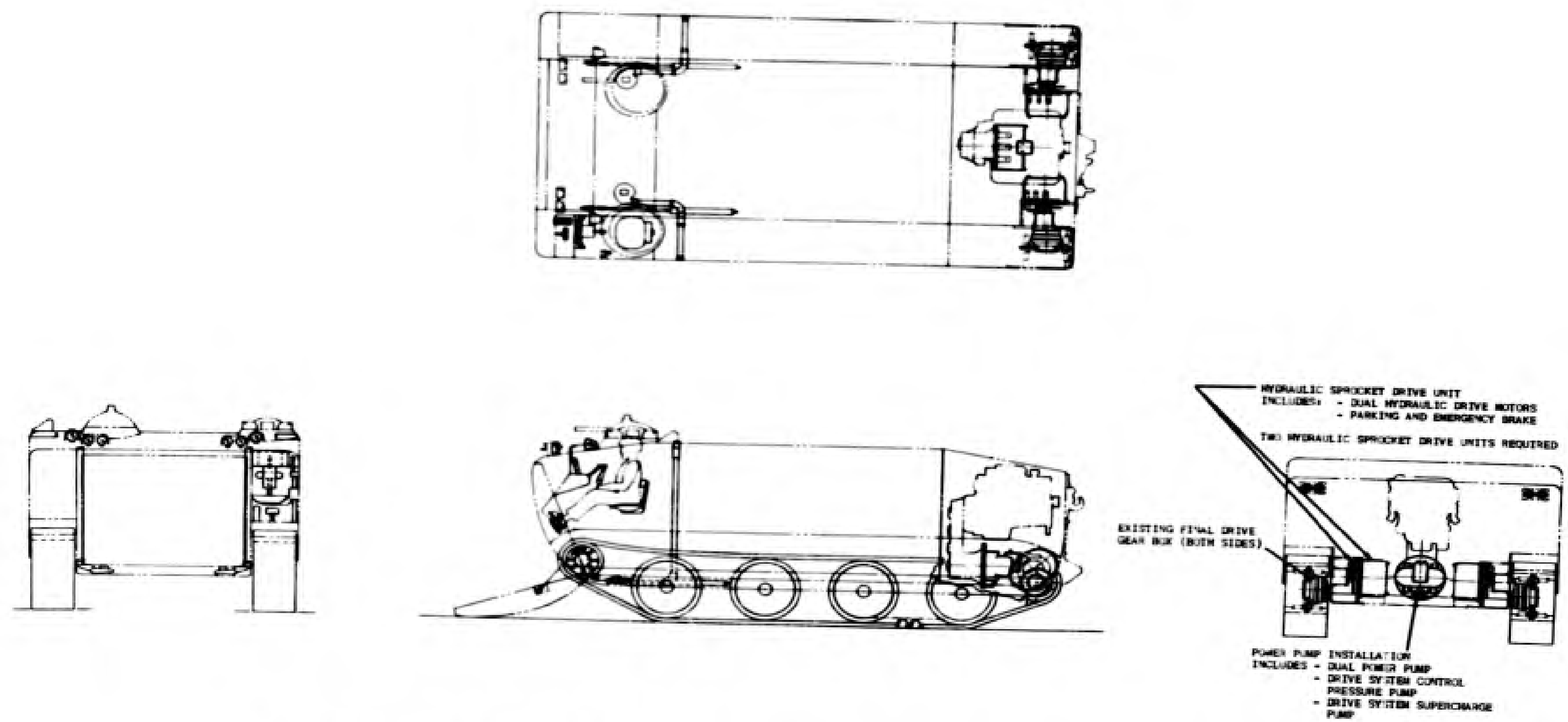


Figure 11-32. Test Bed Arrangement - Rear Sprocket Drive Vehicle, Single Engine

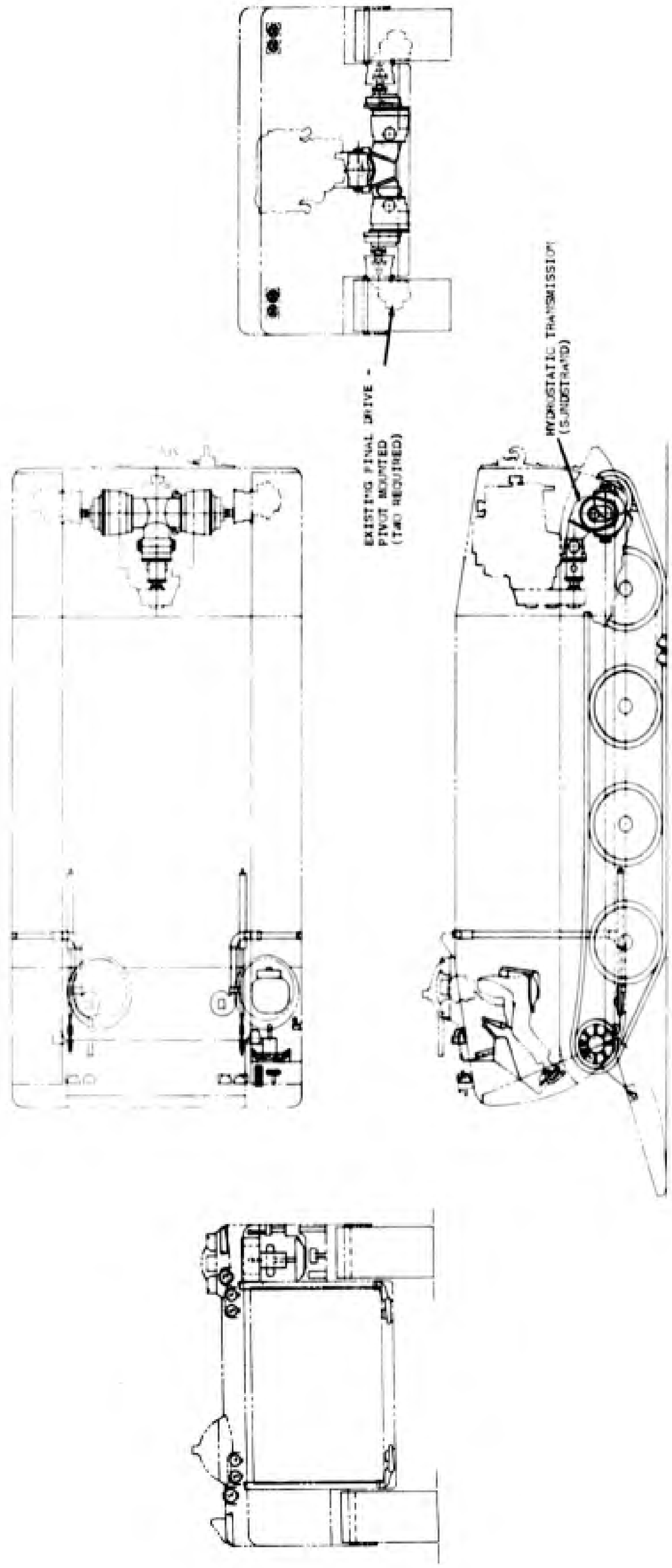


Figure 11-33. Rear Sprocket Drive Vehicle, Single Engine

11.9.7 Actual Design Arrangement - Rear Sprocket Drive Vehicle, Single Engine, as Shown by Figure 11-34

This arrangement indicates the flexibility of a split hydrostatic drive system in that it could be either a front- or rear-sprocket drive. In this single-engine arrangement, it provides additional usable cargo space. It enables close coupling of pumps and motors but would necessitate changing the proposed suspension to gain additional sprocket mounting space for larger hydrostatic drive motors.

Units shown are Gar Wood Industries. The cost of this configuration would be approximately $60,000.

Sundstrand's cost is approximately $75,000.

11.9.8 Actual Design Arrangement - Four-Sprocket Drive Vehicle, Single Engine, as shown by Figure 11-35

This vehicle arrangement provides maximum utilization of space for a single engine. However, it does not provide the usable space as shown for the twin engine configuration. Water trim is good, but not as good as with only front-sprocket drive. This arrangement has longer piping runs than with only rear-sprocket drive. However, the four-sprocket drive will reduce track slap, which in turn increases track reliability and life. It has the advantage of operating with only two sprocket drives under low torque conditions. Parts for the sprocket drives are interchangeable. This configuration portrays a feasible vehicle design and is based on component hardware available today.

Gar Wood Industries units are shown in this arrangement. The prototype cost of this configuration would be approximately $60,000.

Sundstrand's cost is approximately $75,000.

11.9.9 Actual Design Arrangement - Four-Sprocket Drive Vehicle, Two Engines, as Shown by Figure 11-34

Of all the arrangements presented, only this configuration gives a completely open area through the vehicle for maximum utilization of space. As previously indicated, the hydrostatic units are available today. Although this arrangement necessitates the use of two separate pumps and additional controls, it gives maximum reliability by providing two possible power sources for vehicle propulsion. If one compares the extra space available and the reliability for achieving vehicle motivation against the additional control and power components with slight increase of cost and weight, one must favor this vehicle arrangement as the most desirable. Compared to a straight mechanical with hydraulic differential steering, this system could possibly save 300-1300 lbs. in weight.

Gar Wood Industries pumps and motors are shown in this arrangement. The prototype cost of this configuration would be approximately $60,000.

Sundstrand's cost is approximately $75,000.

11.10 <u>Recommendation.</u>

A vehicle specified by the government is basically one in which weight per

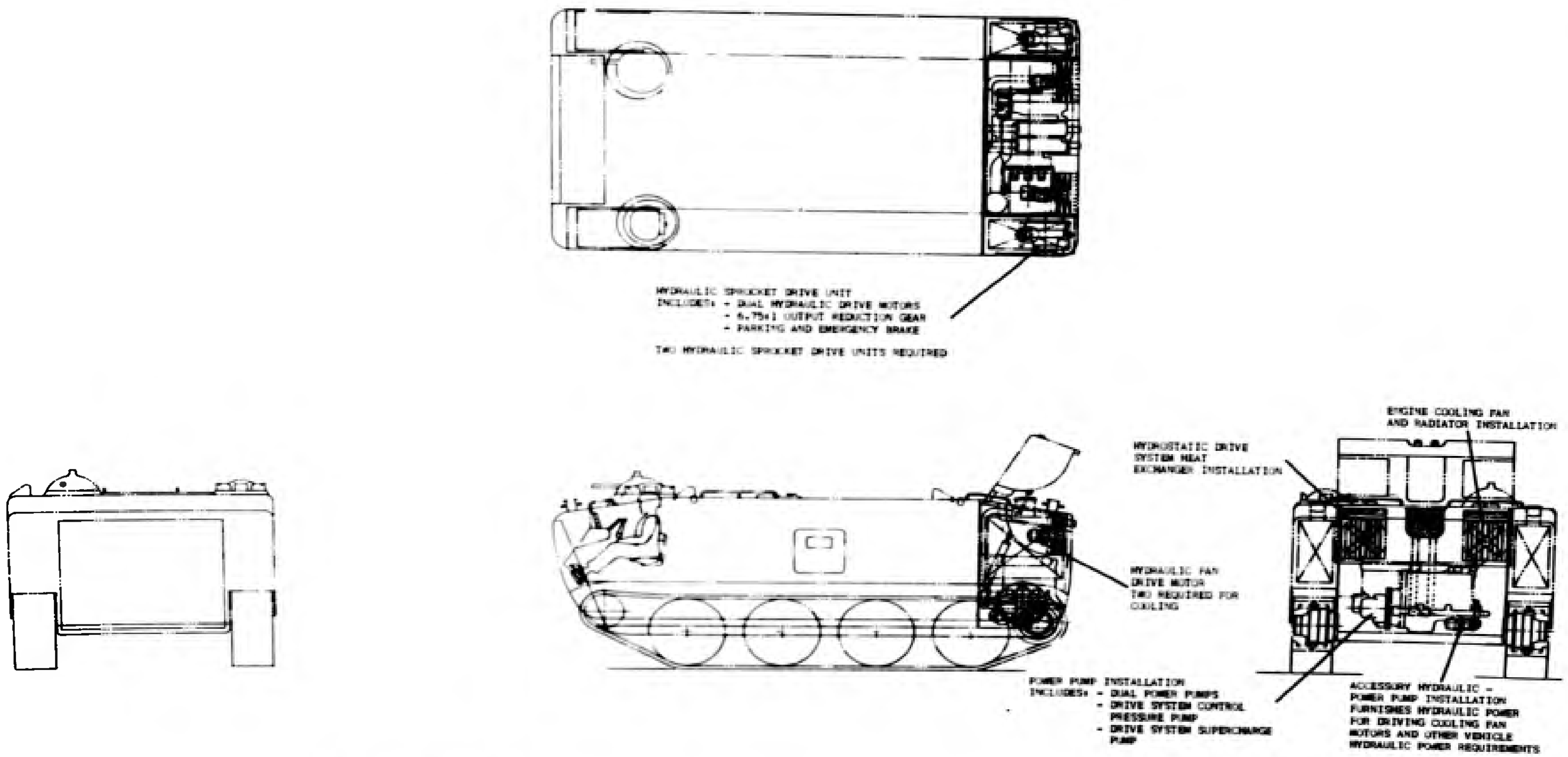


Figure 11-34. Rear Sprocket Drive Vehicle, Single Engine

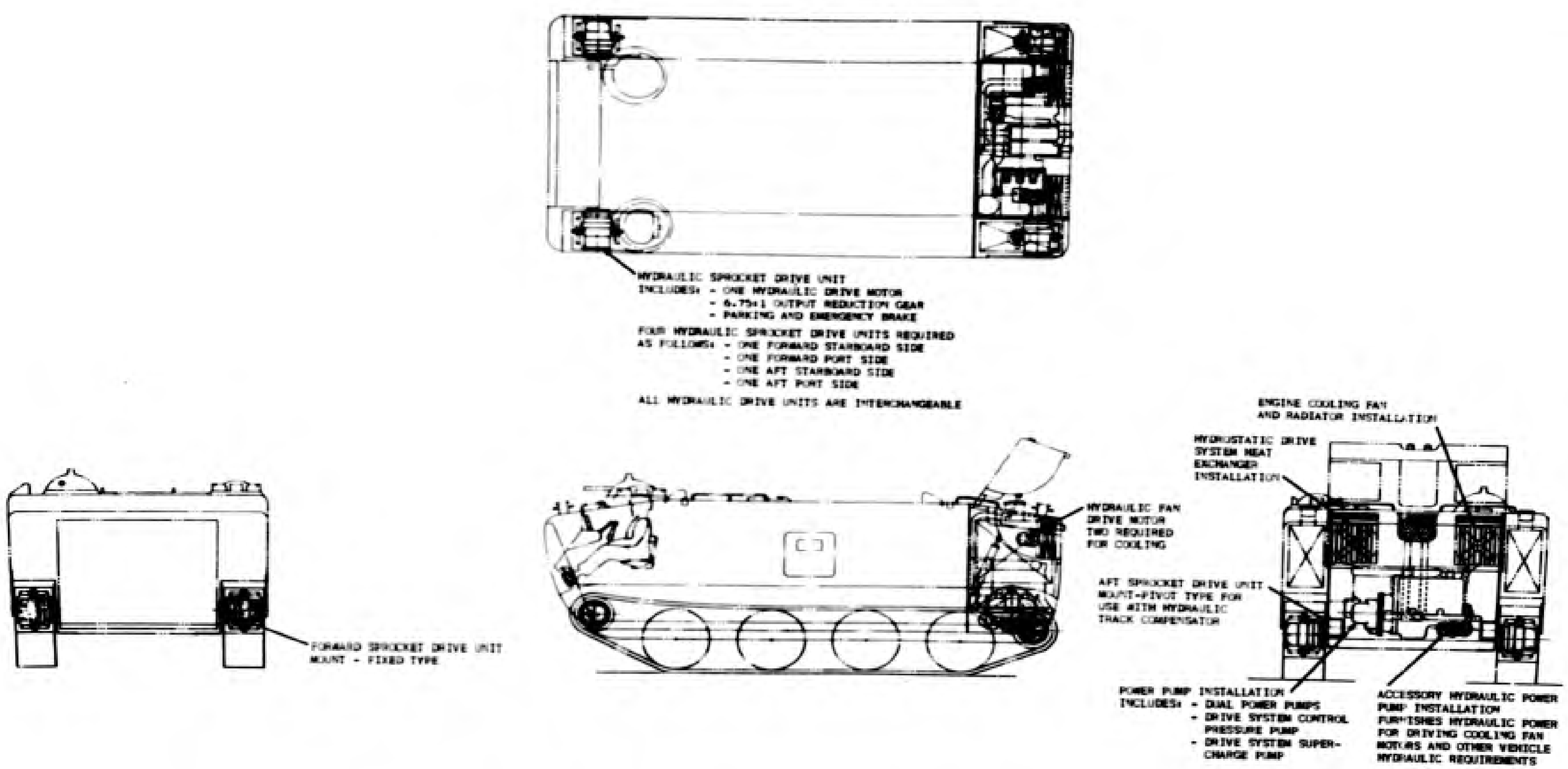


Figure 11-35. Four-Sprocket Drive Vehicle, Single Engine

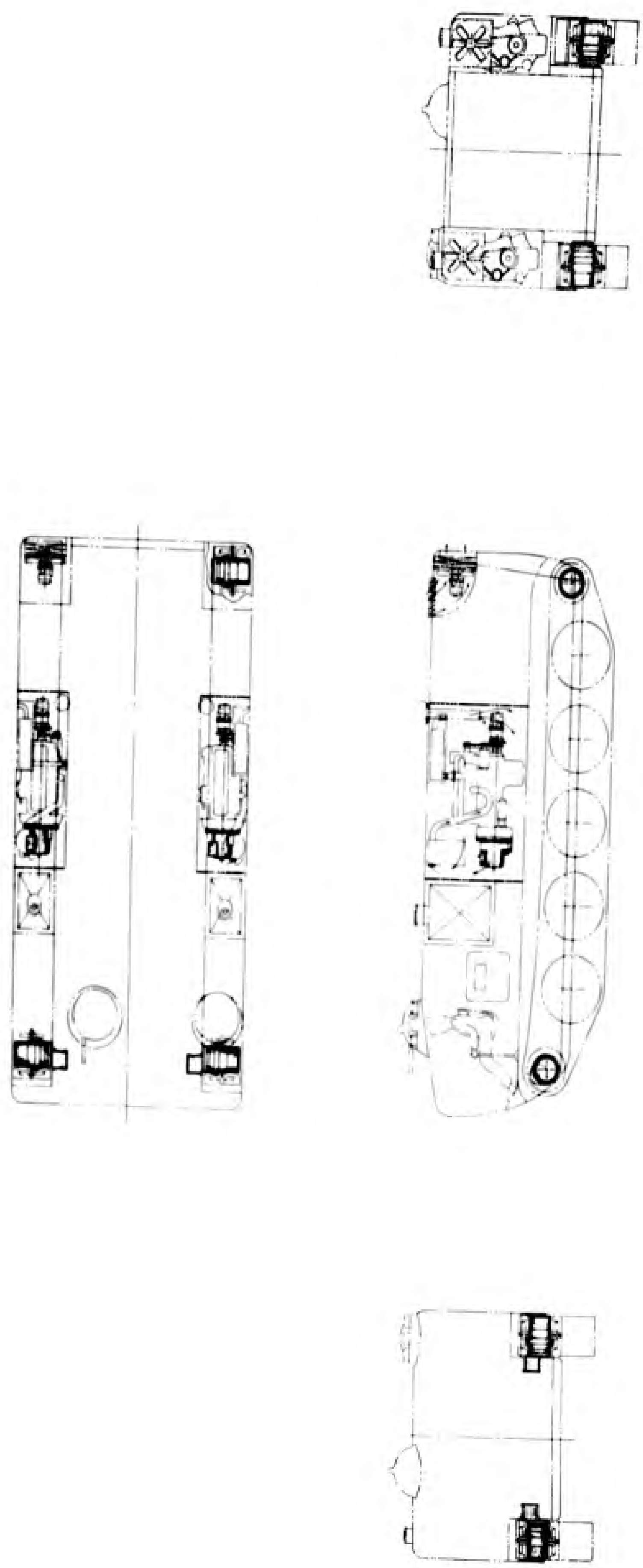


Figure 11-36. Four-Sprocket Drive Vehicle, Two Engines

size and usable space for work are critical. Based on minimum weight per size and maximum usable space, the vehicle configuration shown in Figure 11-36 for twin engines and the configuration shown in Figure 11-35 for a single engine are recommended. The units shown in these drawings are made by Gar Wood Industries. These recommendations are based on these units' being presently available. Our preference, as previously explained, is Sundstrand Aviation. They will have units available within a year's time.

11.10.1 Advantages.

Utilization of a twin-engine vehicle with hydrostatic drive as recommended and shown in Figure 11-36 provides the following advantages:

1. It is the only amphibious vehicle known that is completely open in the cargo area from bow to stern.

2. It has the most usable space area of any of the vehicle configurations.

3. It provides the lowest power train weight for a twin-engine vehicle configuration.

4. It has excellent positioning of components, including modular sprocket drives for maintenance.

5. It has the necessary flexibility to achieve satisfactory water trim.

6. It has the advantage of operating under low torque conditions with only two of the four sprocket drive units.

7. It provides interchangeability of parts among sprocket drives.

8. It provides a secondary power source for all auxiliary hydraulic circuits.

9. It provides positive drive at all times, in any speed range.

Utilization of a single-engine vehicle with hydrostatic drive as recommended and shown in Figure 11-35 provides the following advantages:

1. It provides the lowest weight for the MAC vehicle requirements.

2. It has excellent positioning of components, including modular sprocket drives for maintenance.

3. It has the necessary flexibility to achieve satisfactory water trim.

4. It has the best arrangement in the single-engine group for maximum usable space.

5. It provides interchangeability of parts among sprocket drives.

6. It has the advantage of operating under low torque conditions with only two of the four sprocket drive units.

7. It provides a secondary power source for all auxiliary hydraulic circuits.

8. It saves weight, as compared with a conventional mechanical-drive power train.

9. It provides positive drive at all times, in any speed range.

12.0. A STUDIED TWIN ENGINE LVT

This section contains excerpts of work performed on the design of a steel hull, twin engine concept utilized as a comparison to the MAC in the analysis of the various components and systems ultimately selected. This concept embodies many good features including good overall balance and trim, large cargo and passenger space and greater armor protection and is therefore presented very briefly as comparative data for the reader. Lines are illustrated in Figure 12-1.

12.1 Construction.

The criteria for design loads on the comparison vehicle are:

a) 10 feet of water on the top deck (to withstand surf loading)

b) A 30,000-pound bump force on any bogey wheel on the steel hull. Rigid corner structure indicates that only one-half of this force is absorbed at any one frame.

12.2 Shell Plating.

One-half-inch steel plate is used for the hull cover. With 40-inch frame spacing, the strength is more than adequate, so armor protection is the limiting criterion for this hull. The 1/4-inch-thick cargo deck is sufficient in thickness both to give protection from mines and support the cargo. The mathematical calculations are shown in paragraph 12.7.

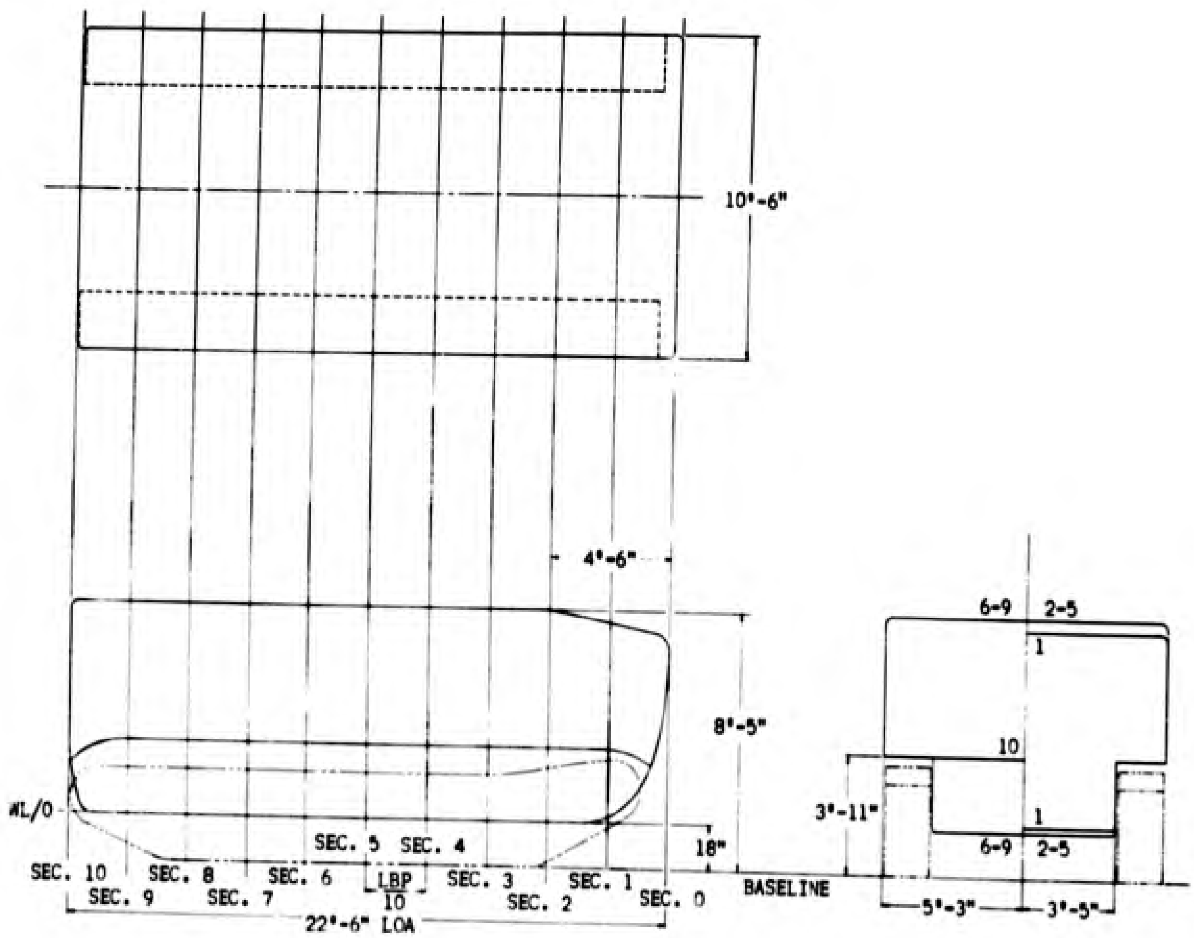


Figure 12-1. Lines

12.3 Armor.

The types of armor that have been considered are steel, aluminum, magnesium-lithium, titanium, Doron II, bonded nylon, unbonded nylon, other plastics, composite armor, and honeycomb, see paragraph 3.15.7. The comparison vehicle has high-strength steel armor plate outside and Doron II inside.

12.4 Protection.

Radiological protection from flash burns, alpha and beta radiation, and fallout and minimum protection from small arms fire and fragmentation are offered by the steel hull. Some additional metal protection may be installed in place of the plastic armor without exceeding 50,000 pounds, but tests show that this would decrease, rather than increase the ballistic protection efficiency of the hull.

12.5 Weight, Trim, and Stability.

Static stability calculations for the comparison vehicle are included on the following pages. For an assumed maximum loading of 10,000 pounds of cargo centered at the geometric center of the cargo department, preliminary estimates show that the vehicle will be stable in the water at any angle of heel up to approximately 120° . Various weight conditions and their relation to trim in various operating conditions and required freeboards have been evaluated and are included with the weight study.

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Hull	18,582	10.74	199,623.6	3.71	68,892.9
2.	Hull Outfittings & Equip.	910	11.89	10,819.9	2.92	2,657.2
3.	Power Train	5,863	11.45	67,039.4	3.39	19,884.8
4.	Suspension	6,874	11.80	81,186.2	.06	425.7
5.	Hydraulics	1,739	9.13	15,878.3	3.02	5,255.0
6.	Electrical	779	11.28	9,389.4	3.90	2,833.6
7.	Total Group Weights	34,747	11.04	383,936.8	2.88	99,949.2

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Total Group Weights	34,747	11.04	383,936.8	2.88	99,949.2
2.	Fuel (--- Full)	680	11.50	7,820	5.40	3,672
3.	Crew	400	3.80	1,520	4.40	1,760
4.	Total Weights	35,827		393,276.8		105,381.2
5.	L.C.G.		10.97			
6.	K.G. Above K.L.				2.94	

7. Mean Draft = 3.30 Ft. (Calc.)
8. L.C.G. = 10.97 Ft. Aft of Sta. 0 (5)
9. L.C.B. = 11.15 Ft. Aft of Sta. 0 (Calc.)
10. Trim Lever = -.18 Ft. (Diff. between 8 & 9)
11. Trim Moment = -6449 Lbs. Ft. (10 x 4)
12. M.T. In. = 2200.0 Lbs. Ft. (Calc.)
13. Trim (In.) = 2.93 (11 ÷ 12)
14. Trim (DEG.) = $0^{o}37'$ (13 ÷ 19 Tan. of trim angle) .01094
15. C.F. = 11.15 Ft. Aft of Sta. 0 (Calc.)
16. Drafts:

    a. Fwd. = 41.05 in.
    b. Aft = 38.15 in.
    c. Mean = 39.6 in.

K.B. = 1.60 Ft. Above K.L.
B.M. = 3.60 Ft.
K.M. = 5.20 Ft. Above K.L.
K.G. = 2.94 Ft. Above K.L.
G.M. = 2.26 Ft.

17. Centroid of projected chine area aft of sta. 0 (Ft.) = -----(Calc.)
18. L.C.B. = ----------- Ft. ----------- of centroid (%L_c) (Calc.)
19. Length W.P. = 267.6 In. (Calc.)

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Total Group Weights	34,747	11.04	383,936.8	2.88	99,949.2
2.	Fuel (---- Full)	680	11.50	7,820	5.40	3,672
3.	Crew	400	3.80	1,520	4.40	1,760
4.	Full Cargo - 27 Troops @ 210#	5,670	13.70	77,679	3.50	19,845
5.	Total Weights	41,497		470,955.8		125,226.2
6.	L.C.G.		11.34			
7.	K.G. Above K.L.				3.02	

8. Mean Draft = 3.65 Ft. (Calc.) K.B. = 1.75 Ft. Above K.L.
9. L.C.G. = 11.34 Ft. Aft of Sta. 0 (6) B.M. = 3.35 Ft.
10. L.C.B. = 11.20 Ft. Aft of Sta. 0 (Calc.) K.M. = 5.10 Ft. Above K.L.
11. Trim Lever = .14 Ft. (Diff. between 9 & 10) K.G. = 3.02 Ft. Above K.L.
12. Trim Moment = 5809.6 Lbs. Ft. (11 x 5) G.M. = 2.08 Ft.
13. M.T. In. = 2200.00 Lbs. Ft. (Calc.)
14. Trim (In.) = 2.64 (12 ÷ 13)
15. Trim (DEG.) $0^{\circ} 34'$ (14 ÷ 20) = Tan. of Trim Angle = .00983
16. C.F. = 11.19 Ft. Aft of Sta. 0 (Calc.)
17. Drafts:
    a. Fwd. 42.49 In.
    b. Aft 45.11 In.
    c. Mean 43.8 In.
18. Centroid of projected chine area aft of sta. 0 (Ft.) = ------ (Calc.)
19. L.C.B. = ---- Ft. -------------------of centroid (% L_C) (Calc.)
20. Length W.P. = 268.5 In. (Calc.)

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity: Station Line: Aft of Sta. 0 (Ft.)	Station Line: Moments (Ft.Lbs.)	Water Line: Above Lowest Point of Keel (Ft.)	Water Line: Moments (Ft. Lbs.)
1.	Total Group Weights	34,747	11.04	383,936.8	2.88	99,949.2
2.	Fuel (----- Full)	680	11.50	7,820	5.40	3,672
3.	Crew	400	3.80	1,520	4.40	1,760
4.	Full Cargo	10,000	13.70	137,000	3.50	35,000
5.	Total Weights	45,827		530.276.8		140,381.2
6.	L.C.G.		11.56			
7.	K.G. Above K.L.				3.06	

8. Mean Draft = 3.90 Ft. (Calc.) K.B. = 1.92 Ft. Above K.L.
9. L.C.G. = 11.56 Ft. Aft of Sta. 0 (6) B.M. = 3.18 Ft.
10. L.C.B. = 11.20 Ft. Aft of Sta. 0 (Calc.) K.M. = 5.10 Ft. Above K.L.
11. Trim Lever = .36 Ft. (Diff. between 9 & 10) K.G. = 3.06 Ft. Above K.L.
12. Trim Moment = 16,497.7 Lbs. Ft. (11 x 5) G.M. = 2.04 Ft.
13. M.T. In. = 2,200.0 Lbs. Ft. (Calc.)
14. Trim (In.) = 7.50 (12 ÷ 13)
15. Trim (DEG.) $1^\circ 35'$ (14 ÷ 20) = Tan. of Trim Angle) .02788
16. C.F. = 11.21 Ft. Aft of Sta. 0 (Calc.)
17. Drafts:
    a. Fwd. 43.08 In.
    b. Aft. 50.52 In.
    c. Mean 46.8 In.
18. Centroid of projected chine area aft of sta. 0 (Ft.) = ---- (Calc.)
19. L.C.B. = ---- Ft. ------------------ of centroid (% L_c) (Calc.)
20. Length W.P. = 269. In. (Calc.)

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Side Plates - 1/2 Plate 291.32 sq ft x 20.4 #/sq ft	5,943	11.50	68,344.5	3.89	23,181.27
2.	Bow Plate - 1/2 Plate 83 sq ft x 20.4 #/sq ft	1,693	.45	761.8	2.53	4,283.29
3.	Stern Plate - 1/4 Plate 72.5 sq ft x 10.2 #/sq ft	740	22.40	16,576.0	3.45	2,553.0
4.	Ramp Plate (Inside) - 3/16 Plate 40.8 sq ft x 7.65 #/sq ft	312	21.90	6,832.8	3.45	1,076.4
5.	Bottom Plate - 1/4 Plate 128.52 sq ft x 10.2 #/sq ft	1,311	12.40	16,256.4	0	0
6.	Deck Plate - 3/16 Plate 134.3 sq ft x 7.65 #/sq ft	1,027	11.90	12,221.3	.42	431.3
7.	Track Channel - 1/4 Plate 79.92 sq ft x 10.2 #/sq ft	815	11.60	9,454.0	2.40	1,956.0
8.	Track Skirts - 1/8 Plate 71.74 sq ft x 5.1 #/sq ft	366	11.50	4,209.0	1.60	585.6
9.	Top Plate - 1/4 Plate 171.85 sq ft x 10.2 #/sq ft	1,753	11.25	19,721.25	6.90	12,095.7

Estimated Group Weight Summary cont'd.

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
10.	Tom Frames - 5 Frames x 10.5 ft x 5.0 #/ft	263	11.62	3,056.0	6.75	1,775.3
11.	Side Frames (Columns) 2 Sides x 5 Frames x 6.33 ft x 5.90 #/ft	373	11.62	4,334.2	3.50	1,305.5
12.	Bottom Frame - 5 Frames x 6.8 ft x 7.2 #/ft	245	11.62	2,846.9	0.25	61.2
13.	Bow Frame at Center-line 7.5 ft x 7.9 #/ft	59	1.50	88.5	2.53	149.3
14.	Bow Frame - Interme-diate - 2 Sides x 6.2 ft x 7.9 #/ft	98	1.50	147.0	2.53	247.9
15.	Ramp Frame - 2 Frames x 6.8 x 6.75 #/ft	92	22.2	2,042.4	3.8	349.6
16.	Cargo Hatch - 1/2 Aluminum Plate	490	14.1	6,909.0	6.9	3,381.0
17.	1/2 Thick Doron (Front Plate) 25.8 sq. sq. ft x 5 #/sq ft	129	.80	103.2	4.6	593.4
18.	1/2 Thick Doron (Foot Plate) 28.3 sq ft x 5 #/sq ft	142	.70	99.4	2.1	298.2

Estimated Group Weight Summary cont'd.

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft.Lbs.)
19.	1/2 Doron (Side Plate) 2 Sides x 13.31 sq ft x 5 #/sq ft	133	2.50	332.5	4.3	571.9
20.	1/2 Doron (Top Plate) 15 sq ft x 5 #/sq ft	75	1.85	138.7	5.8	435.0
21.	1/4 Doron (Cargo Compartment Sides)	552	13.7	7,562.40	3.5	1,932.0
22.	1/4 Doron (Cargo Compt. Top)	420	13.7	5,754.0	6.9	2,898.0
23.	Escape Hatches (2)	240	7.4	1,776.0	4.95	1,188.0
24.	Access Hatches (2)	100	7.3	730.0	7.0	700.0
25.	Mooring Bitts (4)	75	10.6	795.0	6.9	517.5
26.	Towing Eyes (2)	56	11.4	638.4	2.6	145.6
27.	Lifting Eyes (4)	80	10.35	828.0	7.2	576.0
28.	Driver's Cupola	250	3.6	900.0	7.3	1,825.0
29.	Machine Gun Cupola	300	3.6	1,080.0	7.5	2,250.0
30.	Misc. Hinges and Brackets	450	11.3	5,085	3.4	1,530.0
	TOTAL	18,582	10.74	199,623.65	3.71	68,892.96

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Driver's Seat	60				
2.	Troop Seats	190				
3.	Tow Cable	75				
4.	Medical Chest	10				
5.	Flash Lights (2)	2				
6.	Grease Gun	2				
7.	Boat Hooks (2)	8				
8.	Pick	8				
9.	Shovel	4				
10.	Axe	5				
11.	Sledge	8				
12.	Crow Bar	7				
13.	Hand Oiler	1				
14.	Signal Light	12				
15.	Fuel Fill Extension	10				
16.	Gun Kit	10				
17.	Lamp Box w/Spare Lamps	1				
18.	Spare Gun Parts	10				
19.	Bowline and Stern-fast	15				
20.	Track Fixture	35				
21.	1000 Rounds Ammunition	64				
22.	Tow Cable Shackle (Spare)	20				
23.	Track Blocks (Spare) (2)	60				
24.	Pyrotechnic Box	3				
25.	Manuals	20				
26.	Tool Roll	100				
27.	Electrical Cable	50				
28.	Water and Gas Cans (2)	60				
29.	M8A2 Gas Particulate Unit	60				
	TOTAL	910	11.89	11,819.9	2.92	2,657.2

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
	Engine (2)	2,322	18.0	41,796.0	4.4	10,216.8
	Torque Converter (2)	520	15.7	8,164.0	3.7	1,924.0
	Cooling System (2)	250	20.2	5,050.0	5.3	1,325.0
	Exhaust System (2)	130	17.0	2,210.0	6.0	780.0
	Aspiration System (2)	54	14.6	788.4	5.6	302.4
	Angle Drive Shafts (4)	120	9.7	1,164.0	3.3	396.0
	Transmission Drive Shafts (2)	48	3.2	153.6	1.9	91.2
	Final Drive Shafts (2)	66	2.0	132.0	1.4	92.4
	Angle Drive Shaft Brg. (2)	24	9.2	220.8	3.0	72.0
	Angle Drive (2)	600	3.4	2,040.0	2.8	1,680.0
	Transmission	800	2.5	2,000.0	1.6	1,280.0
	Final Drive (2)	574	1.9	1,090.6	1.4	803.6
	Controls	165	2.0	330.0	3.4	561.0
	Machinery Mounts (total)	190	10.0	1,900.0	1.9	361.0
	TOTAL	5,863	11.43	67,039.4	3.39	19,884.8

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Track (Complete)	3,812	11.7	44,600.4	0	0
2.	Road Wheels (10)	2,540	11.29	30,276.8	0	0
3.	Idler Assembly (2)	270	21.5	5,805.0	.55	148.5
4.	Drive Sprocket Assembly (2)	252	2.0	504.0	1.1	277.2
	TOTAL	6,874	11.3	81,186.2	.06	425.7

Estimated Group Weight Summary

No.	Description	Weight (Lbs.)	Moments and Centers of Gravity: Station Line: Aft of Sta. 0 (Ft.)	Station Line: Moments (Ft.Lbs.)	Water Line: Above Lowest Point of Keel (Ft.)	Water Line: Moments (Ft. Lbs.)
1.	Track Comp. Cylinders	200	2.5	500.0	2.0	400.0
2.	Main Hydraulic Pumps	104	1.33	138.3	4.0	416.0
3.	Ramp Cylinders	100	3.5	350.0	2.5	250.0
4.	Bilge Pumps	100	11.0	1,100.0	2.5	250.0
5.	Oil Reservoir	75	16.0	1,200.0	5.0	375.0
6.	Hydraulic Oil	150	11.0	1,650.0	3.0	450.0
7.	Piping	600	11.0	6,600.0	1.75	1,050.0
8.	Fuel System (Complete)	260	11.5	2,990.0	5.4	1,404.0
9.	Main CO_2 System	130	9.0	1,170.0	4.4	572.0
10.	Portable CO_2 System	20	9.0	180.0	4.4	88.0
	TOTAL	1,739	9.13	15,878.3	3.02	5,255.0

Estimated Group Weight Summary

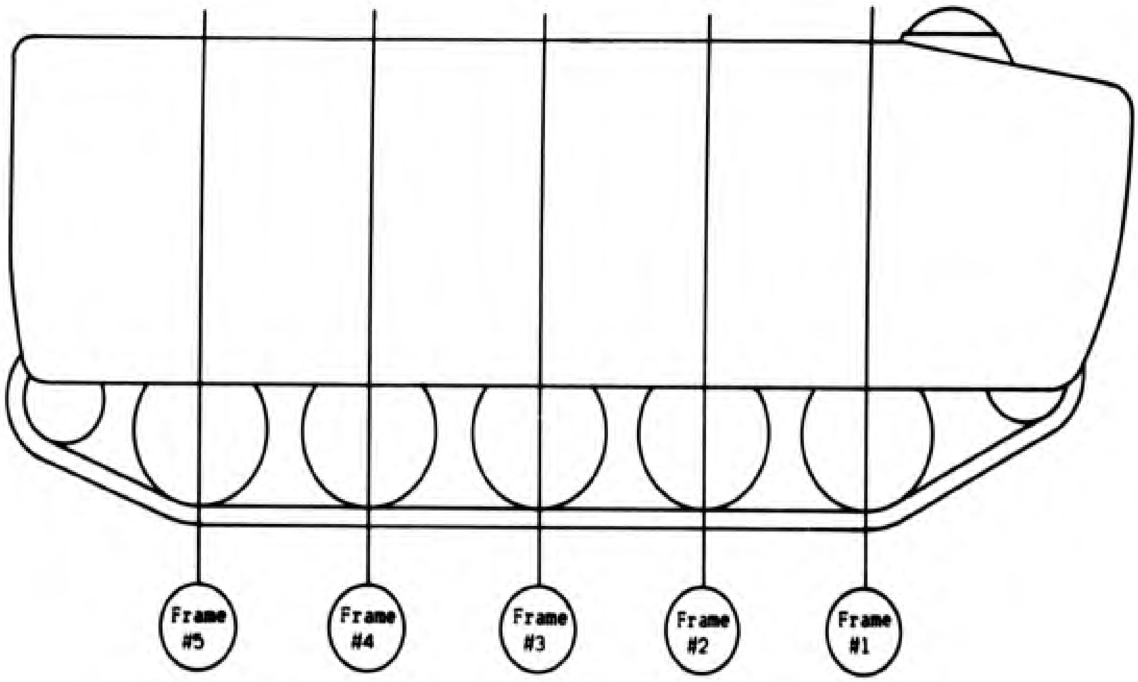
No.	Description	Weight (Lbs.)	Moments and Centers of Gravity			
			Station Line		Water Line	
			Aft of Sta. 0 (Ft.)	Moments (Ft.Lbs.)	Above Lowest Point of Keel (Ft.)	Moments (Ft. Lbs.)
1.	Radio and Antenna	70	1.5	105.0	5.6	392.0
2.	Batteries	144	14.1	2,030.4	4.3	619.2
3.	Instrument Panel	30	2.0	60.0	5.9	177.0
4.	Aux. Bilge Pump	32	15.0	480.0	0.7	22.4
5.	Lights	18	3.0	54.0	6.0	108.0
6.	Distribution System	55	15.0	825.0	4.0	220.0
7.	Wiring	140	11.5	1,610.0	3.0	420.0
8.	Misc. Hardware	125	11.5	1,437.5	2.5	312.5
9.	Generator & Voltage Reg.	110	17.0	1,870.0	3.0	330.0
10.	Crew Ventilation System	30	16.0	480.0	6.5	195.0
11.	Scavenge System	25	17.5	437.5	1.5	37.5
	TOTAL	779	11.28	9,389.4	3.9	2,833.6

12.6 Framing.

In order to achieve the greatest usable interior space and reduce the over-all weight, the same high-strength steel used for the steel hull is used for frame structure. Although this material is considerably more expensive than structural steel, the reduction in weight results in an approximately equal total cost, even on a prototype basis. In this comparison vehicle, frames are spaced to coincide with suspension points, which results in the simplest structure.

Spacing of the transverse hull bottom frames is 40 inches, since this corresponds to the spacing of the suspension system. With this spacing each wheel is supported by a frame, so no additional corner stiffening is necessary. Under these conditions the structural angles provide adequate strength for supporting the top deck. Figure 12-2 shows the typical frame sections. No side frames are employed, since the steel plate installed for armor purposes provides more than sufficient strength without reinforcement.

The columns extend from the hull bottom to the top deck to absorb the torque generated by the suspension system and carry the top deck loads. Maximum surf loading plus simultaneous bump torque were used as design criteria. Square tubing size was calculated for these design loadings but an increased size was selected to insure safety should the tubing be dented or bent by cargo, etc.



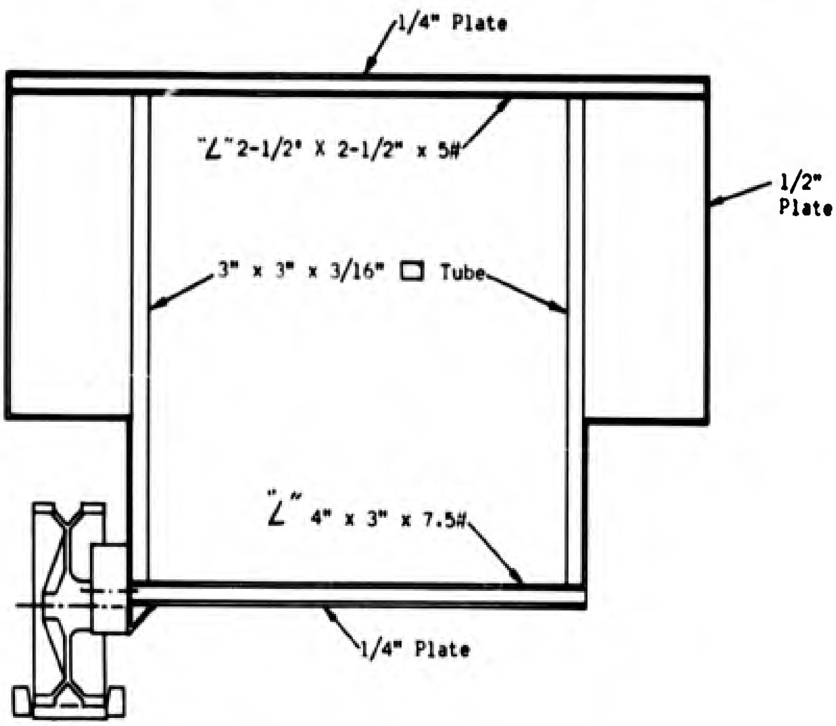


Figure 12-2. Typical Frame Sections

Structural angles were selected as hull bottom transverse frames principally to provide an accessible space beneath the cargo deck and to provide a flat surface for fastening the cargo deck.

12.7 Strength Calculations.

References

Title:	Formula for Stress and Strain
Author:	Raymond J. Roark
Printer:	McGraw-Hill Book Company Third Edition, 1954
Title:	Steel Construction Manual
Author:	American Institute of Steel Construction
Printer:	American Institute of Steel Construction
Title:	Design Manual for High Strength Steels
Author:	H. Malcolm Priest
Printer:	Davis & Warden Inc. for United States Steel Corporation
Title:	USS T-1 Steel
Author:	United States Steel Corporation
Printer:	United States Steel Corporation

HY-125 armor steel plate is used in this design. A factor of safety of 2.5 on the ultimate strength of 160,000 pounds per square inch gives a working stress of 64,000 pounds per square inch. This figure provides a factor of safety of 1.95 on the yield strength, which is 125,000 pounds per square inch.

As outlined in the shell plating paragraph 12.2, the following thicknesses will be used:

1/4 inch for top, bottom, and stern plates;

1/2 inch for sides and bow plates.

A frame spacing of 40 inches will adequately support the suspension system. The top deck plates will be checked for strength under an assumed surf loading of 5 pounds per square inch. Since large deflection, as compared to plate thickness, is allowed in this plate, the design method outlined on page 219, Article 58 of Roark's Formula for Stress and Strain will be used.

w = Unit applied load - pounds per square inch.

a = Larger dimension of panel - inches.

b = Smaller dimension of panel - inches.

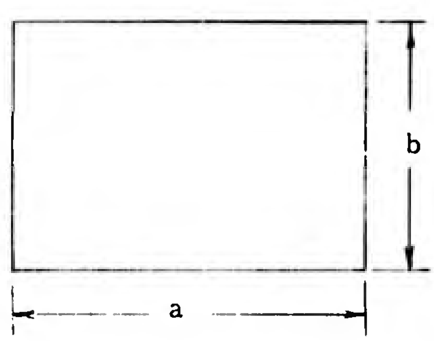
E = Modulus of Elasticity - pounds per square inch.

t = Thickness of plate - inches.

S = Total stress - pounds per square inch.

$$\frac{a}{b} = \frac{126}{40} = 3+$$

Edges held and fixed.



$$\frac{wb^4}{Et^4} = \frac{5 \times 4^4 \times 10^4}{3 \times 10^7 \times (2.5)^4 \times 10^{-4}} = 109$$

$$\frac{Sb^2}{Et^2} = 35 + \left(\frac{41 - 35}{25}\right) 9$$ (Interpolating from table on page 222 - Roark)

$$= 35 + 2.16$$

$$= 37.16$$

$$\therefore S = \frac{37.16 \times 3 \times 10^7 \times (2.5)^2 \times 10^{-2}}{4^2 \times 10^2}$$

= 43,500 pounds per square inch.

This stress is satisfactory, since a working stress of 64,000 pounds per square inch is allowable; therefore, the 40-inch spacing will adequately support the loading.

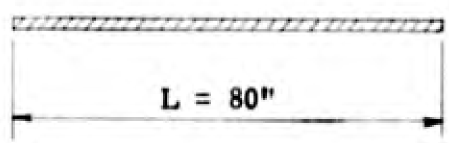
Frames

Frames are assumed to be semifixed. Loads = 5 pounds per square inch.

Top Deck Member

$$\text{Moment} = \frac{WL^2}{10} \qquad W = 5 \text{ lbs. per in.}^2 \times 40 \text{ in.} = 200 \text{ lbs. per inch}$$

$$M = \frac{200 \times 80^2}{10}$$



M = 128,000 inch-lbs.

$$\text{Section Modulus, } Z_{rq'd} = \frac{\text{Moment}}{\text{Allowable Stress}} = \frac{128,000}{64,000} = 2.0 \text{ in.}^3$$

Notations

r = Radius of gyration - inches. $= \sqrt{\frac{I}{A}}$

b = Effective width contributed by continuous plate - inches.

A = Area - square inches.

$\bar{y}$ = Distance from neutral axis of element to axis of reference - inches.

$\bar{Y}$ = Distance from neutral axis of section to axis of reference - inches.

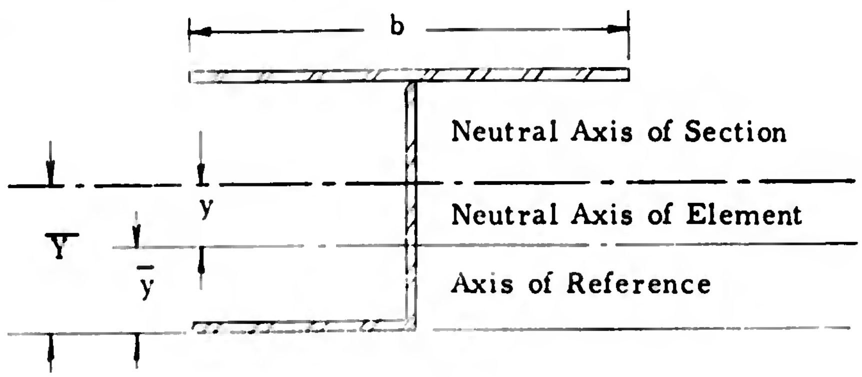
y = Distance from neutral axis of element to neutral axis of section - inches.

I_o = Moment of inertia of element about own neutral axis - inches.

I_t = Total moment of inertia of section about its neutral axis - inches.

Z = Section modulus - inches^3

M = Bending moment - inch-lbs.



Try a 2-1/2 in. x 2-1/2 in. x 1/4 in. angle in addition to the 1/4 inch top plate which provides an effective flange width = 30 x t = 30 x 1/4 inch = 7.5 in. (from page 245, Article 64, Roark).

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	y	y^2	Ay^2	I_o
Plate	7-1/2 x 1/4	1.88	2.63	4.92	0.75	0.56	1.06	----
Angle	2-1/2 x 2-1/2 x 1/4	1.19	0.72	0.86	1.16	1.35	1.60	0.70
		3.07		5.78			2.66	0.70

$$\bar{Y} = \frac{5.78}{3.07}$$

$$= 1.88 \text{ inches}$$

$$I_t = 2.66 + 0.70$$

$$= 3.36 \text{ in.}^4$$

$$Z_{prov.} = \frac{3.36}{1.88} = 1.79 \text{ in.}^3$$

This is not acceptable.

Try a 2-1/2 in. x 2-1/2 in. x 5/16 in. angle.

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	y	y^2	Ay^2	I_o
Plate	7-1/2 x 1/4	1.88	2.63	4.95	0.83	0.69	1.30	----
Angle	2-1/2 x 2-1/2 x 1/16	1.47	0.74	1.09	1.06	1.12	1.65	0.85
		3.35		6.04			2.95	0.85

$$\bar{Y} = \frac{6.04}{3.35}$$

$$= 1.80 \text{ inches}$$

$$I_t = 2.95 + 0.85$$

$$= 3.80 \text{ in.}^4$$

$$Z_{prov.} = \frac{3.80}{1.80} = 2.11 \text{ in.}^3$$

This figure is acceptable, since it is slightly larger than the required 2.0 in.^{3}; therefore, the top member of these frames will consist of an angle 2-1/2 in. x 2-1/2 in. x 5/16 in.

Hull Side Members

Try without structural members.

Load = 5 pounds per square inch.

1/2 inch plate.

$\frac{a}{b} = \frac{51}{40} = 1.28$ Use 1.5, because this value will give a higher stress.

Edges assumed held and fixed.

$$\frac{wb^4}{Et^4} = \frac{5 \times 4^4 \times 10^4}{3 \times 10^7 \times (5)^4 \times 10^{-4}} = 6.8$$

$$\frac{Sb^2}{Et^2} = 0 + \frac{5.75 \times 6.8}{12.5} = 3.12$$ (Interpolating from table on page 222, Roark)

$$S = \frac{3.12 \times 3 \times 10^7 \times 5^2 \times 10^{-2}}{(4)^2 \times 10^2} = 14{,}620 \text{ pounds per square inch.}$$

This value for stress is low, but no reduction in member thickness is recommended, since the 1/2 inch plate is required for armor protection.

Hull Bottom Member

As assumed loading of 5 pounds per square inch is used. Since span and loading are the same as for top member, it is logical to assume that the same requirements will apply for both members; that is, a 2-1/2 in. x 2-1/2 in. x 5/16 in. angle will be adequate for the bottom member. However, since an accessible space is desirable under the cargo deck, 4 in. x 3 in. x 5/16 in. angle will be used.

Columns

Columns will be proportioned primarily to carry a bump torque and bump force which will present the most critical condition. An assumed bump force = 30,000 pounds per road wheel is used. Since there are 10 road wheels, the total force will be 30,000 x 10 = 300,000 pounds. With an estimated total weight of vehicle of 45,000 pounds, this will result in

$$\frac{300{,}000}{45{,}000} = 6.7 \text{ G's.}$$

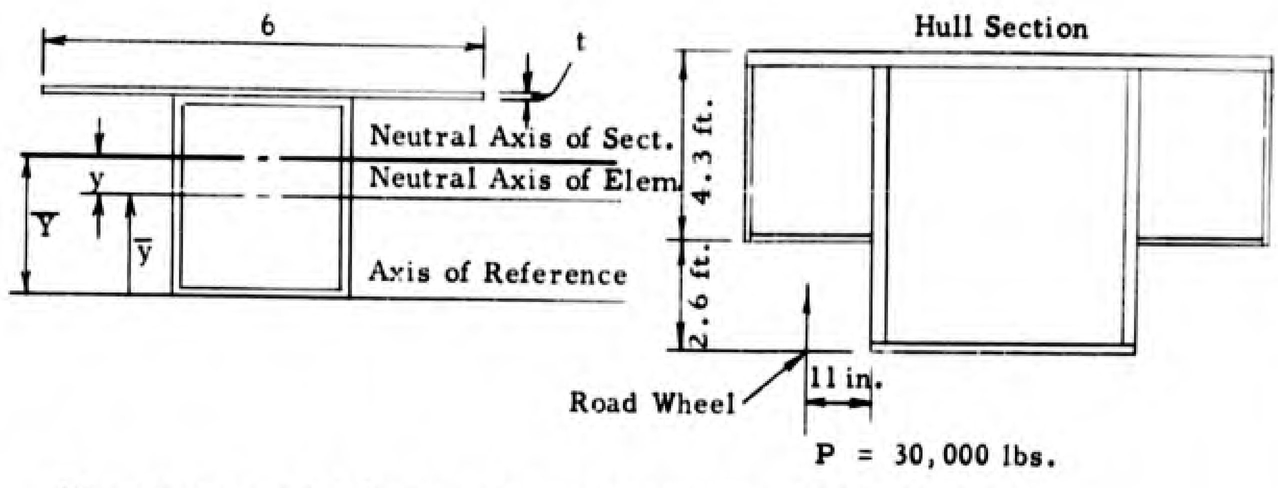
Since each road wheel is cantelivered out at each frame, and the distance from outboard face of column to center line of road wheel is 11 inches, this condition results in a bump torque = 30,000 x 11 = 330,000 in.-lbs. This total torque will be transmitted to the corresponding frame, and it is assumed that the following distribution will occur:

1/3 of torque will be dissipated to adjacent frames;

2/3 of torque will be absorbed by frame in question.

Of the latter amount, 1/2 will be resisted by horizontal bottom member (see paragraph 12.7.5, Roark and Priest) and 1/2 by the vertical member. Therefore, the torque resisted by the column will be = 1/3 x 330,000 in.-lbs. = 110,000 in.-lbs.

$$Z_{rq'd} = \frac{110,000}{64,000} = 1.72 \text{ in.}^3$$



For notations and definitions see pages 12-20 and 12-21.

A square tube is selected because of its greater rigidity.

b = 30 x t = 30 x 1/2 = 15 in.

Try a 3 in. x 3 in. x 3/16 in. tube. $A = 2.0$ in.^{2}; $I_o = 2.55$ in.^{4};

$r = 1.14$ in.

Elem.	Dim.	Area	$\bar{y}$	$A\bar{y}$	y	y^2	Ay^2	I_o
Plate	15 x 1/2	7.5	3.25	24.35	0.37	0.14	1.03	0.16
Tube	3 x 3 x 3/16	2.0	1.50	3.0	1.38	1.91	3.82	2.55
		9.5		27.35			4.85	2.71

$$\bar{Y} = \frac{27.35}{9.5} = 2.88 \text{ inches}$$

$$I_t = 4.85 + 2.71 = 7.56 \text{ in.}^4$$

$$Z_{prov.} = \frac{7.56}{2.88} = 2.64 \text{ in.}^3$$

This column has to carry a combined stress of bending and direct loads, and must be checked by the following formula: $\frac{f_a}{F_a} + \frac{f_b}{F_b} \leq 1$ as stated by the American Institute of Steel Construction Specifications, page 284, Section 12, AISC Handbook.

f_a = Actual axial unit stress - Axiai load divided by area of member pounds per square inch.

F_a = Allowable axial unit stress - pounds per square inch.

f_b = Actual bending unit stress - Bending moment divided by section modulus - pounds per square inch.

F_b = Allowable bending unit stress - pounds per square inch.

$f_b \quad \frac{110{,}000}{2.64} = 41{,}600$ pounds per square inch.

F_b 64,000 pounds per square inch. See paragraph 12.7.

$f_a \quad \dfrac{30,000}{2.0} = 15,000$ pounds per square inch.

KL/r = Slenderness ratio

K 0.75 (Page 20, "Design Manual for High Stength Steels" by H. Malcolm Priest, U. S. S. Corp., Publication No. ADUCO 02215).

L = Length of column - inches.

r = Radius of gyration - inches

$$\frac{KL}{r} = \frac{0.75 \times 2.6 \times 12}{1.14} = 20.55$$

F_a = 123,000 lbs. per in.^{2} (From Fig. 20, page 24, U. S. S. Corp., "T-1 Steel" brochure, Publication No. ADUCO 01060-59).

This stress must be reduced by the same factor of safety as used throughout this design, that is = 1.95 on the yield strength, therefore, allowable F_a =

$\dfrac{123,000}{1.95}$ = 63,000 pounds per square inch.

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} = \frac{15,000}{63,000} + \frac{41,000}{64,000} = 0.24 + 0.64 = 0.88 < 1$$

Therefore, the 3 in. x 3 in. x 3/16 in. tube is adequate.

Bulkheads.

Bulkheads are not integral, but the engine, air intake, and exhaust are enclosed by insulated plates to minimize heat, noise, and fumes.

Deck Support.

The cargo deck is bolted directly to the hull bottom members. Reinforcing members are not required.

Foundations.

Equipment foundations will be designed as required and will be similar to the main hull structure. Shock mounts will be installed where required, as determined by the design. In general, the equipment foundations will be welded to the hull, except where access or maintenance requirements suggest other methods.

Ramp.

Two structural tees, cut to fit the stern shape, form the ramp structure. These are spaced to carry the load of wheeled vehicles. The construction is similar to that used to support the cargo deck.

Bow.

The bow is framed vertically. There are five structural tees: one on the centerline, two which connect with the inside of the track channel, and two intermediate ones. All are formed to the curve of the bow.

Hull Accessories.

Hull accessories were designed both in accordance with marine specifications and in recognition of the shock loading imposed during off-road operations. Safety of personnel, ease of maintenance, and reliability of operation governed the design.

12.8 Power Train.

The increased power requirements of the heavier 50,000 pound class vehicle, when analyzed in conjunction with other vehicle considerations, advanced the concept of a twin engine installation. This design approach evolved into a vehicle general arrangement utilizing the compartments over the vehicle tracks for the engines. The engine is installed facing rearward and is placed at the rear of the cargo compartment with drive directed forward.

The general arrangement presented two distinct and critical problems. It was necessary to evaluate the size and weight of power plants as well as their power output characteristics. Over-all width had to be as small as possible to prevent encroachment into the cargo hatch area. In view of all these factors, it was found that the Continental Motors Corp. Model LDS-465 multifueled, turbocharged compression-ignition engine may be modified for this arrangement.

The selection of Continental Model LDS-465 engine was based on its power characteristics, size, weight, durability, availability, multifuel characteristic and cost, as well as component interchangeability with other Continental military engines. This power plant could be readily replaced by Continental's LDS-427 Ordnance engine which is used in the M-35 truck and M-44 series of vehicles.

The Continental Model LDS-465 is rated at 210 BHP @ 2800 rpm. The displacement is 465 cubic inches and it weighs 1,555 pounds.

Continental LDS-465. The Continental Model LDS-465 compression-ignition engine, is a modified, uprated version of the LDS-427 engine which Ordnance is placing into production for the M-35 truck and M-44 series of vehicles. It is rated at 200 BHP @ 3000 rpm and 445 lb.-ft. torque @ 1800 rpm. This engine is a four cycle, inline 6, of 465 cu. in. displacement. It is turbo-charged and employs the hypercycle combustion system to permit multifuel operation, which gives a 40 percent reduction in fuel consumption over present production gasoline engines.

Its weight with standard accessories is 1,555 pounds. An aluminum version is being developed which will be 394 pounds lighter in weight. One of the most desirable features of the LDS-465 is its availability from a development and production standpoint. Since this will be basically the LDS-427 with modifications, it can be made available after a very short development period and can be placed into production almost immediately with a negligible tooling cost. The modification is primarily that of increasing the cylinder bore size within the block structure by elimination of the liners to allow an increase in displacement from 427 to 465 cu. in. In order to meet the requirements for the track channel space allotted for each engine of a dual installation, the physical location of external components such as turbocharger, aftercooler, and other minor components are tailored to fit within the engine compartment.

This engine is recommended as the first choice of power plants for a vehicle with dual installation because of its advantages of small size, lighter weight and higher engine rpm which gives high horsepower with lower corresponding peak torque to permit lighter weight transmission components. In addition, service is simplified by the large number of interchangeable parts between engines in the military system.

The performance specifications, are the same as those required for the lighter vehicle. However, this vehicle will require considerably greater power to meet these specifications. Due to the increased power requirements, it has been necessary to use an arrangement having a markedly different approach. This 50,000 pound vehicle requires the design and selection of power train components especially tailored to this type of arrangement.

The basic power train arrangement, see Figure 12-3, consists of two Continental, 210 HP, compression-ignition engines with torque converters. The engine-converter is installed facing aft at the aft end of the cargo compartment, directly over the tracks on each side of the vehicle. Power is then transmitted through the cargo compartment for the length of the cargo hatch, by means of drive shafts, to right angle gear units. These right angle gear units, on both sides of the vehicle, direct the power flow to the input shafts of the centrally located steering transmission by means of short drive shafts. The steering transmission is located directly below the operator's cab and is coupled to the hull mounted integral compensator and final drives through drive shafts.

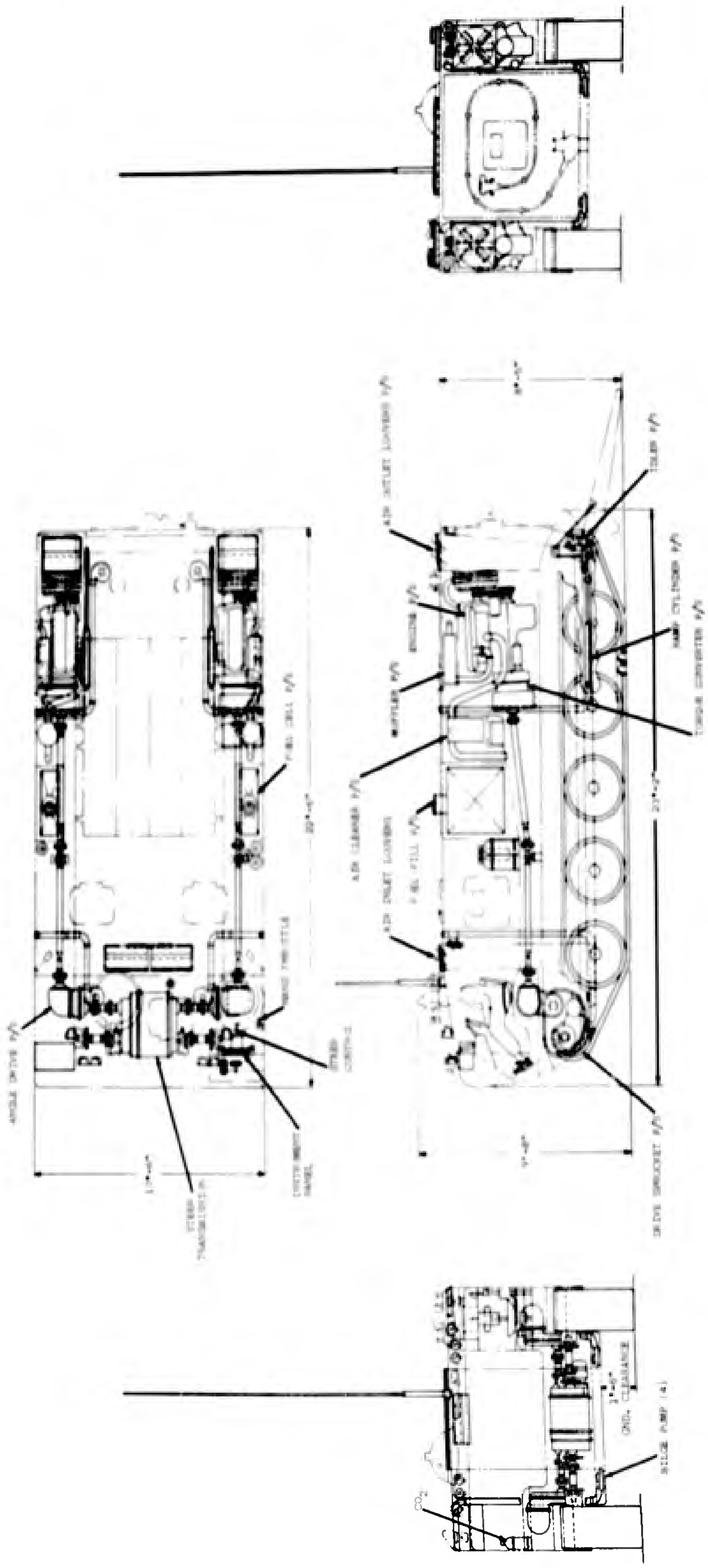


Figure 12-3. General Arrangement - 50,000 Pound Vehicle

Each engine and converter assembly and the steering transmission will have individual cooling systems.

The air for engine aspiration and cooling is drawn into a duct between the operators' cab and the cargo hatch. It then flows beneath the cab, over the steering transmission and out into the engine compartment over the vehicle tracks. The engine air intake is ducted from this compartment to the engines. The air is drawn through the compartment and over the engine by the engine-driven propeller-type fan which forces it through the radiator core into a vertical duct at the extreme rear of the vehicle. This duct exhausts to the atmosphere through a louvered grille on deck. The system of vertical inlet and exhaust ducts provides a means whereby water may enter the grilles, but falls directly into the bilge and cannot be dumped onto the operating mechanisms.

The general arrangement of the vehicle power train requires extensive use of tubular drive shaft assemblies. Each drive shaft assembly is equipped with single cardan type universal joints and one end is provided with a slip yoke. This arrangement will facilitate disassembly, removal of components, and compensate for vehicle dimensional variations and component misalignment.

Due to the drive shaft length required between the torque converter and right angle drive units, it is desirable to use a two-piece drive-shaft assembly.

Since the connection between the drive and driven units are not on the same axis, the two-piece shaft construction permits the total angular misalignment of the axes to be distributed among the three universal joints thereby minimizing wear and vibration. A rubber insulated bearing is utilized to support the center of the two-piece drive shaft assembly. The selection of the two-piece shaft arrangement is predicated, by the offset of the connected units and by limiting critical shaft speeds and the resulting effect on shaft bending or whipping.

Drive shaft power transmission is also utilized from the right angle gear units to the steering transmission and from the steering transmission to the final drives.

The drive shaft assemblies are of seamless welded steel tubing with Mechanics Division of Borg-Warner Corporation universal joints. These universal joints are provided with lube fittings and incorporate spiders supported by needle bearings. All assembly components are of ample capacity to withstand maximum power train loadings.

The final drive assemblies are separate hull mounted drive sprocket supports with integral gear reduction and hydraulcially actuated track slack compensators.

The design of these units is essentially the same as that incorporated in the lighter, 35,000 pound vehicle preliminary design study. The revision to gears and bearings were made to increase the final drive unit capacity as required for a vehicle of the 50,000 pound class.

The modifications will insure fatigue and durability life commensurate to a vehicle of the heavier class.

Aspiration System.

The engine aspiration system is related to the over-all vehicle air system inasmuch as the combustion air intake is located in the combination air duct and drive shaft tunnel. This duct, which is at the bottom of the compartments over the track channel, interconnects the engine and tramission compartments.

The installation has been designed to provide combustion air which is free of water contamination and is cool. The location of the intake, forward of the engine, will insure against these conditions. Principally, the loss of power resulting from excessive induction air temperature which can be present in the engine compartment. The combustion air intake ducts are routed to the air cleaners which are housed in separate compartments. These separate compartments, which are located in both sides of the vehicle are in front of and adjacent to the engine compartment provide additional insulation against engine compartment heat. The air cleaner is then connected, by ductwork, directly to the engine intake manifold.

Exhaust System.

The exhaust system is designed to minimize the heat build-up in the engine compartment which is an important factor in the design of the engine cooling system. The engine cooling system, with air flowing through the engine compartment and then through the radiator cores, requires that the engine compartment temperature be as low as possible to obtain a favorable air to boil temperature differential which is a factor in the radiator core design.

This is accomplish by placing the mufflers outside of the hull in depressions over the engine compartments at the extreme outboard sides of the vehicle. This will permit the mufflers to be contained within the hull silhouette. The location, over the engine compartment, will result in shorter muffler inlet pipes which will be a factor in keeping the engine compartment temperature at an acceptable level.

Cooling System.

The cooling system proposed for the twin engine arrangement is designed to provide maximum air flow without induction of water into the system during surfing operations.

The liquid cooled engines, being installed at the rear of the vehicle and facing aft, will be cooled in the conventional manner. Each engine will have an individual radiator assembly and fan; however, the fan flow direction will be reversed to provide for the induction of air into the vehicle air system.

The vehicle system is arranged with the air intake in the forward section of the vehicle, just aft of the crew compartment. This consists of vertical ducts which have louvered grilles and will provide the means whereby water will be dumped directly into the bilges. The air flow will be directed forward into the transmission compartment under the operators' cab. It will then flow outboard into the combination air duct and drive shaft tunnel which interconnects the transmission and engine compartment and is in the lower section of the compartment over the track channels. The air is then drawn over the engine and forced through the radiator cores. The exhaust arrangement, as is the intake duct, utilizes a vertical duct at the extreme end of the compartment to permit the water entering through the exhaust grille to fall to the bilges.

The exhaust and intake louvered grilles are of the flipper type which will be normally closed. These will be counterbalanced and be opened by the air flow created by the engine cooling fan.

The individual radiator assemblies will consist of two cores, one of which will be used for converter oil cooling. These cores will be of the fin-tube arrangement interconnecting upper and lower tanks.

Controls.

The controls required for a twin engine, rear installation, with front sprocket drive, are essentially the same as those required for a single engine, rear installation, with rear sprocket drive.

The location of the transmission, at the front of the vehicle, simplifies the system as a result of shorter linkage cables which are used. Other than this, the brake, steering, and transmission shift controls are the same.

The accelerator or engine speed control, presents the problem of synchronizing the engine output. This problem is inherent to a twin engine installation and is magnified with the engines being located on opposite sides of the vehicle in separate compartments. As it is quite common for piston engines to vary in output, and for a variety of reasons, it is necessary to provide a means of adjustment. The use of mechanical linkages; that is,flexible cable housed in flexible conduit, is desirable as they are adaptable to various means of adjustment and are easily installed.

A "fail safe" linkage, which additionally will minimize the effects of hysteresis, is proposed for a virtually "fool-proof" system. This will be accomplished by incorporating a throttle control spring of sufficient force to close the throttle in event of failure in the system. The flexible linkage from the accelerator and hand throttle, is designed to operate in tension. This will eliminate the lost motion in the linkage and insure the matching of the engine output throughout the speed range.

A hand throttle lever is located on the port side bulkhead to provide a convenient means of regulating the throttle setting from a standing position, or maintaining a fixed setting for water operation. It is mechanically linked to the throttle linkage and designed to be independent of the accelerator pedal.

The hand control and accelerator pedal are designed and arranged in positions conducive to driver comfort. Accelerator pedal linkage ratios are such that the reaction forces will give the driver positive feel of the controls.

12.9 Suspension System.

The suspension system for the 50,000 pound vehicle is essentially the same as for the MAC except that five pairs of road wheels instead of four are utilized. Accommodation of the extra road wheels also results in extra length of track and additional weight to the vehicle.

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