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MEP 354 DRIVE SYSTEM.(U)

AUG 79 R F O'DONNELL, R E BARLOW

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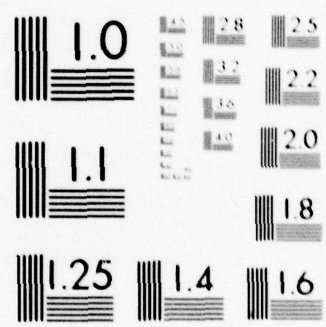

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MEP 354 DRIVE SYSTEM

Handling & Servicing/Armament Division
Ground Support Equipment Department
Naval Air Engineering Center
Lakehurst, New Jersey 08733

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MRP 356 DRIVE SYSTEM

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AIR MOTOR
DRIVE SYSTEM
WHEEL MOTOR

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Presents data accumulated through a study to determine the type of drive system to be included in the new DOD Mobile Electric Power Plant (MEP 354). All suitable drive system types are described and compared for possible inclusion in the power plant. A systematic approach is used to indicate the drive which best meets all the specified requirements.

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SUMMARY

A. GENERAL. Two of the mobile electric power plants currently used in the fleet, the NC-8A and the NC-2A, are to be replaced by a new power plant, MEP 354. This power plant will conform to the latest standard for military aircraft power, MIL-STD-704C. Problems encountered with the drive system of the NC-8A have led to the need for a more reliable and dependable drive system to be incorporated in MEP 354. This report documents the results of a study to ascertain which type of drive system would be most advantageous to include in the new mobile power plant.

B. PROCEDURES AND RESULTS. An attempt was made to analyze all types of possible drive systems, both developed and undeveloped, and determine which exhibits the characteristics most suitable for MEP 354. Figure 1 illustrates the major drive systems considered and the key parameters used to evaluate their relative merits. The end result of this investigation is that a hydrostatic system is the system that displays the most favorable characteristics. It is recommended that such a system be included in MEP 354 as the drive system. A secondary recommendation is that further analysis of an a-c motor controlled by a microcomputer be initiated. It is a novel approach to controlling the speed of a very dependable a-c induction motor. The necessary development time to build and perfect such a system precluded its choice for MEP 354.

PREFACE

The NC-8A and NC-2A mobile electric power plants were designed over a decade ago. Problems encountered with the quality of power have led to the development of a revised standard for aircraft electrical power, MIL-STD-704C, "Aircraft Electric Power Characteristics". Ground power units must conform to this new standard. Drive system problems have also been encountered in the fleet, particularly on the NC-8A. These factors have resulted in a development program to replace the existing Mobile Electric Power Plants (MEPP). This report pertains to the drive system to be incorporated in the new MEP 354.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS AND TABLES

LIST OF ILLUSTRATIONS AND TABLES (CONTINUED)

I. INTRODUCTION

Several of the mobile electric power plants presently used in the fleet were designed many years ago and are no longer capable of efficiently performing their mission. Problems exist in several areas, particularly in the electrical and propulsion systems. An effort is under way to replace the NC-8A and NC-2A power plants with a new power plant, MEP 354.

The goal of this report is to determine what type of drive system would be most advantageous to incorporate in MEP 354. The power plant will be used on carrier decks and shorebased flight lines. Therefore, mobility and ability to function in severe environments are critical. Currently available systems as well as presently undeveloped state of the art systems are being evaluated. This report documents the investigation which results in the final drive system selection.

II. PROCEDURE

A. DESCRIPTION OF ANALYSIS OF COMPETING ALTERNATIVES FOR MEP 354 DRIVE SYSTEM.

In order to determine the most desirable alternative for the drive system, many approaches could be utilized. The approach employed in this study was to select and combine various analytical techniques in the form of a model. The model is adapted to the problem at hand and the output is oriented to the selected evaluation criteria. The model, in itself, is not the decision maker, but is a tool which provides the data necessary in support of the decision making process.

The model utilized should incorporate the following features:

1. The model should represent the dynamics of the system being evaluated in a way that is simple enough to understand and manipulate, and yet close enough to the operating reality to be meaningful.

2. The model should highlight those factors that are most relevant to the problem at hand, and suppress those that are not as important.

3. The model should be comprehensive by including all relevant factors.

4. Model design should be simple enough to allow for timely implementation in problem solving. Unless the tool can be utilized in a timely and efficient manner by the analyst, it is of little value. If the model is large and highly complex, it may be appropriate to develop a series of models where the output of one can be tied to the input of another.

5. Model design should incorporate provisions for easy modification and/or expansion to permit the evaluation of additional factors as required.

An attempt was made to incorporate all of the above features. The model which was selected is shown in Figure 1. The evolution of the complete evaluation process follows.

B. INITIAL ASSUMPTIONS.

MEP 354 is to supersede two other mobile electric power plants currently in the fleet. These two power plants are the shorebased NC-8A and the ship based NC-2A. The following specification manuals, publications, and military standards are used as a guide to determine the specific characteristics to be included in the new MEP 354 to the extent specified herein.

NAVAIR 19-45-10	Technical Manual, Organizational, Intermediate, and Depot Maintenance with Illustrated Parts Breakdown, Mobile Electric Power Plant Model NC-2A.

EVALUATION PARAMETER	WEIGHTING FACTOR	HYDROSTATIC WITH DIFFERENTIAL		HYDROSTATIC WITH WHEEL MOTORS		HYDRODYNAMIC (TORQUE CONVERTER)		HYDRODYNAMIC (AUTOMATIC TRANSMISSION)		A-C MOTOR WITH MICRO COMPUTER CONTROL		A-C MOTOR WITH RECTIFIER INVERTER CONTROL		D-C MOTOR WITH FIXED RECTIFIER CONTROL		D-C MOTOR WITH VARIABLE RECTIFIER CONTROL	
		BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE
COST																	
SIZE																	
WEIGHT																	
RELIABILITY																	
MAINTAINABILITY																	
SUBTOTAL																	
RISK (DERATING FACTOR)																	
GRAND TOTAL																	

FIGURE 1. PROPULSION UNIT MATRIX - MEP 354

NAVAIR 19-45-1	Index and Application Tables for Mobile Electric Power Plants
NAVAIR 19-45-9	Technical Manual, Operation and Maintenance Instructions with Illustrated Parts Breakdown, Organizational, Intermediate and Depot Maintenance, Power Plant, Mobile Electric Model No. NC-8A
MIL-W-8005E	Wheels and Hubs, For Industrial Pneumatic Tires
XAS-1359	Specification for NC-8A Mobile Electrical Power Plant
XFWGS-164B	Specification for NC-2A Mobile Electrical Power Plant
MIL-M-8090C	Mobility Requirements, Ground Support Equipment, General Specification For
MS 35389	Tire, Pneumatic, Industrial Straight Side, Tractor Rear and Tractor Front
MIL-D-23003	Military Specification Deck Covering Compound, Nonslip, Lightweight

The following paragraphs define the properties used in determining the drive system for the mobile electric power plant.

1. SIZE. NC-8A frame envelope as defined in NAVAIR 19-45-1.

Length	112 inches
Width	58 inches
Height	45 inches
Wheelbase	78 inches
Ground Clearance	7 inches

2. WEIGHT. The weight of a fully loaded NC-8A is 5880 pounds (NAVAIR 19-45-1). This weight is used as the base weight. MEP 354 will have a different generator and drive system than the NC-8A. It was assumed that the new propulsion system's weight would be approximately the same as the NC-8A. Therefore, the generator weight was subtracted from the above 5880 pounds as below.

NC-8A fully loaded weight	5880 pounds
A-C/D-C generator	-1000 pounds
Remaining weight	4880 pounds

The weight of the MEP 354 generator, which was obtained from the manufacturer, is added to this remaining weight to obtain the following gross weight.

Remaining weight	4880 pounds
MEP 354 generator	+ 775 pounds
Gross Weight	5655 pounds

To allow for growth and possible inclusion of a transformer rectifier assembly, 6500 pounds was used as the weight to determine the drive system required to meet the mobility requirements.

3. TIRE SIZE. The NC-8A has 7.50 x 10 pneumatic tires (NAVAIR 19-45-9). The diameter of the tires is 25.9 inches with a rolling radius of 11.4 inches (MS 35389) and a 10-ply rating (MIL-W-8005E).

4. COEFFICIENT OF FRICTION. The coefficient of friction chosen for deck surfaces is 1.00 (MIL-D-23003).

5. MAIN PROPULSION UNIT. A Detroit Diesel 4-53 engine will be used as the main propulsion unit on MEP 354. This engine produces a brake horsepower of approximately 100 horsepower at a rated speed of 2000 RPM. The actual horsepower depends on the fuel injectors chosen, which was undetermined at the time of this report. Nevertheless, 100 horsepower, as an approximation of available horsepower, is within a few horsepower of the actual values.

6. AIR RESISTANCE. Based upon frontal area, a coefficient of .002 was used for the necessary calculations (Kent's Mechanical Engineering Handbook - 11th edition).

7. WEIGHT DISTRIBUTION. The weight of the fully loaded unit, with driver, shall be distributed 50 ±10 percent on the front and rear wheels (XAS-1359).

8. ROLLING RESISTANCE. Based on estimated vehicle mechanical friction, a value of 20 pounds per 1000 pounds of vehicle weight was used in the calculations (Kent's Mechanical Engineering Handbook, 11th edition).

9. TURNING RADIUS. MEP 354 will have a maximum turning radius of 130 inches. This vehicle will be used on carrier decks as well as on shorebased flight lines; therefore, the NC-2A turning radius of 130 inches obtained from XFWGS-164B was used as a guideline for maximum permissable outside turning radius.

10. Belts, chains, clutches, and similar devices shall not be included in the drive system (XAS-1359).

The calculations used to size a drive system relative to horsepower and torque, based on the preceding assumptions, appear in Appendix A.

C. MOBILITY REQUIREMENTS.

The following mobility requirements along with the initial assumptions are used to determine the horsepower, speed, and torque required in the drive system.

1. SPEED. A maximum speed of 19 ± 2 MPH (XAS-1359) was initially considered. It was determined by a committee involved with the development of MEP 354 that 15 MPH would be a safer maximum speed because the vehicle will be used on a carrier deck.

2. ACCELERATION. The mobile electric power plant should travel 100 feet in 9.1 seconds from a standing start.

3. The mobile power plant should climb and pass over, from a standing start, the following obstacles (XAS-1359):

a. 3-inch curb against both front wheels.

b. 3-inch curb against both rear wheels.

c. 3-inch-deep, 20-inch-diameter hole possessing straight sides; condition--one front wheel in the hole.

d. 3-inch-deep, 20-inch-diameter hole possessing straight sides; condition--one rear wheel in hole.

4. The mobile electric power plant should be capable of both forward and reverse direction of travel.

D. IDENTIFICATION OF FEASIBLE ALTERNATIVES

The next step is to identify possible alternative solutions to the problem. All possible alternatives must be initially considered. However, the more alternatives that are considered, the more complex the analysis process becomes. It is desirable to list all possible solutions to ensure against inadvertent omissions and then eliminate those alternatives which are clearly unattractive, leaving only a few for evaluation. Those few alternatives are then analyzed with the intent of selecting a preferred approach. A list of all possible solutions and elimination of those which are unattractive follows in Section III.

E. SELECTION OF EVALUATION CRITERIA.

The parameters selected as evaluation criteria should relate directly to the problem statement. In order to minimize the introduction of risk and uncertainty in the analysis, the evaluation criteria must be selected judiciously to capture as many of the relevant system characteristics as possible. Those features that cannot be quantitatively expressed must be identified and all assumptions must be defined.

In the event that a number of parameters are involved, each parameter should be reviewed from the standpoint of relevancy or degree of importance. The degree of importance may be realized by applying parameter weighting factors (the most important items receiving the heaviest weighting). There are a number of known techniques currently in use for assigning weights. The Delphi Technique, which will be utilized for this analysis, is one common method. Expert opinion (from a group of personnel intimately familiar with the problem at hand) is systematically solicited, compiled, and reviewed with quantified value criteria being established. Through numerous iterations constituting additional opinion solicitations, the criteria becomes more and more refined and bias is reduced.

The approach, which will be followed for this study, will be to assign weights to all evaluation factors, then determine the degree of compatibility of the system configuration being evaluated with each of the various evaluation factors. Each system's evaluation factor will be assigned a base rate which is multiplied by the appropriate weighting factor to obtain a score. The scores for every factor are added to obtain a subtotal. In addition, there is a derating factor based on the risk involved in developing a particular option in the allotted development time. Discussions with the section that handles 6.4 engineering development work, and will be responsible for putting the entire MEP 354 package together, have indicated that a development time of one year is necessary for the program's success. Risk factors are included for the proposed one-year development time as well as for two years to indicate the positive effect of time on risk. The grand total for each system is obtained by multiplying each risk factor by the appropriate subtotal to obtain the system grand totals; the lowest grand total represents the most desirable system.

The evaluation factors were chosen following the model features prescribed and by mutual agreement of a committee involved in the total development of MEP 354. Weighting factors were assigned by the Delphi Technique. The evaluation factors chosen include the initial procurement cost (for 1000 units), size, weight, reliability, and maintainability of each system. Reliability and maintainability help to indicate the long-term efficiency of any particular system. Therefore, some aspects of a life cycle cost analysis are included in the evaluation method; however, a full-scale life cycle cost analysis of all the systems was not attempted. The actual evaluation matrix is incorporated in the report in Section IV.

F. DATA GENERATION AND APPLICATION.

The next step in the analysis process is to assemble the appropriate input data. In this analysis, this involves rating all criteria of each alternative on a scale of 1 to 10. The sources of data for the assignment of values include data banks which provide field data on existing systems similar in configuration and function to the item being developed. The accuracy and completeness of input data depend not only on the sources of data available but on the personal experience of the analyst. The ratings and an explanation of each rating follow in Section IV. Also, the completed matrix is shown in Section IV.

G. ANALYSIS OF RESULTS.

The purpose of this study is to select the most desirable drive system for incorporation into a new 30KW Mobile Electric Power Plant. The complete problem was defined previously. Section V gives details on what the evaluation matrix reveals.

III. DRIVE SYSTEMS

A. Many types of drive systems have been investigated for possible inclusion in MEP 354. The four methods of turning a vehicle's wheels to accomplish motion are mechanically, hydraulically, pneumatically, and electrically. Often these four methods are used in conjunction with each other to produce the desired drive system. Each of these methods was investigated thoroughly and those systems which obviously could not meet the performance requirements were eliminated at the outset. Investigation of current vehicle drive systems and contacts made with numerous system manufacturers has narrowed the gamut of possible drive systems down to a selected few which will receive further consideration in this report. There are five major types described in this Section, each having its own subdivisions as indicated below.

1. HYDROSTATIC DRIVE.

   a. With Differential

   b. With Wheel Motors

2. HYDRODYNAMIC DRIVE.

   a. Torque Converter

   b. Automatic Transmission

3. A-C MOTOR DRIVE.

   a. Microcomputer Control

   b. Rectifier-Inverter Control

4. D-C MOTOR DRIVE.

   a. Fixed Rectifier Control

   b. Variable Rectifier Control

5. AIR MOTOR DRIVE.

Characteristics of each system will be discussed, and the advantages and disadvantages inherent in these schemes will be revealed. This Section describes the aforementioned drive systems.

B. DRIVE SYSTEM DESCRIPTIONS.

1. HYDROSTATIC DRIVE.

a. General.

(1) Hydrostatic drives use a pressurized fluid to transmit force (pressure head), whereas hydrodynamic drives transmit force by means of a fluid which moves at high velocity (velocity head). Except for the

use of a fluid as a power transmitting medium, there is practically no similarity between these two drive systems. The hydrostatic drive consists of a fixed or variable displacement hydraulic pump and one or more fixed or variable displacement hydraulic motors. Also the drive system includes a filter, a reservoir, an oil cooler, valves, etc.

(2) The hydrostatic drive pump is a positive displacement type. There are various types of positive displacement pumps available, such as gear pumps, vane pumps, and piston pumps. There are many variations of each of these basic types. The hydrostatic drive normally employs the axial piston pump. The same three basic categories also apply to the motor, with the axial piston motor being generally used for hydrostatic applications. Positive displacement in a pump guarantees that there will be no change in output flow as pressure at the outlet port changes; and for a motor, that the speed will not change as load torque requirement varies.

(3) The hydrostatic drive system is adaptable to various types of input power sources or drives.

b. Description.

(1) The hydraulic pump is driven by a prime mover and pumps hydraulic fluid from a reservoir, through a filter, to one or more hydraulic motors. The motor drives the load. For a vehicle, the motor can drive directly into a wheel (one motor for each driving wheel), or one motor can drive indirectly through a gear train into a differential. Maximum hydraulic operating pressure is approximately 5,000 psi. The drive is controlled by moving one lever, which controls a variable displacement pump and changes the amount of fluid pumped and the direction of flow. Normally a fixed displacement motor is employed; however, when a variable displacement motor or a two-speed motor is necessary, an additional control is required.

(2) There are four types of hydraulic circuits employed:

(a) Open circuits - Pump inlet and motor exhaust use the same reservoir. Capable of one direction of rotation only.

(b) Closed circuit - Used for unidirectional rotation. Motor exhaust is returned to pump inlet. A piston-type pump is generally used, with a charge pump to improve pump inlet conditions.

(c) Split circuit - The various system components are separate and are connected by pipes and hoses. Application flexibility is possible since the hydraulic motor can be located remotely from the pump. Actually all drive system components are individual units. This flexibility increases the chances of fitting into the existing envelope.

(d) Integral circuit - All components are contained in one housing, which is a packaged drive. These drives are the simplest to apply and generally lower in cost.

(3) A general schematic arrangement of a closed, split circuit hydrostatic system is shown in Figure 2.

(4) The axial piston pump converts rotary input motion to axial reciprocating motion of the pistons. The swashplate rotates which moves the pistons axially. The pistons take in fluid while moving toward the thin end of the plate and expel it, under high pressure, while approaching the thick end. Many pumps incorporate check valves to regulate flow into and out of the piston cavity. One check valve allows fluid to flow into the low-pressure cylinder, while the other check valve is held closed by the high-pressure fluid so the fluid cannot flow back into the reservoir. Pumps with a fixed angle swashplate have a fixed delivery rate. Variable delivery rate is accomplished by varying the angle of the swashplate, which is accomplished by a hanger that is trunnion-mounted on the pump body. Reversible flow is accomplished by moving the swashplate over center. There are also bent axis axial piston pumps and radial piston pumps available.

(5) The charge pump is frequently attached to the main pump. The charge pump provides a flow of fluid through the transmission for cooling purposes, supplies pressurized fluid to the low-pressure side of the pump and motor, and provides fluid for control purposes and for internal leakage makeup.

(6) The axial piston motor functions in the same manner as the axial piston pump. The motor may have either a fixed displacement or a variable displacement. The variable displacement is accomplished by means of an adjustable swashplate. There are also bent axis axial piston motors and radial piston motors available.

(7) The manifold valve assembly contains two pilot operated high-pressure relief valves which prevent sustained abnormal pressure surges in either of the main hydraulic lines by dumping high-pressure fluid into the low-pressure line during suddenly applied loading, hard braking and rapid acceleration. The assembly also contains a shuttle valve and a charge pressure relief valve. These two valves release charge pressure fluid into the case pressure system as a means to control the charge pressure level and also to remove the excess cooling fluid added to the circuit by the charge pump.

(8) Clean hydraulic fluid is necessary for successful system operation; therefore, a filter is present in the pump intake. The fluid should be filtered by a 10-micron filter of good quality. To obtain high reliability, the filter system shall not include a bypass. It is better to reduce the transmission performance with a clogged filter than to allow debris to enter the hydraulic system.

(9) A reservoir is required to provide a source of hydraulic fluid. A suggested minimum reservoir volume (gallons) is .625 of the total charge pump flow (GPM) with a minimum fluid volume equal to .5 of charge pump flow. The reservoir outlet should be above the bottom of the reservoir to permit gravity separation and prevent any large foreign particles from

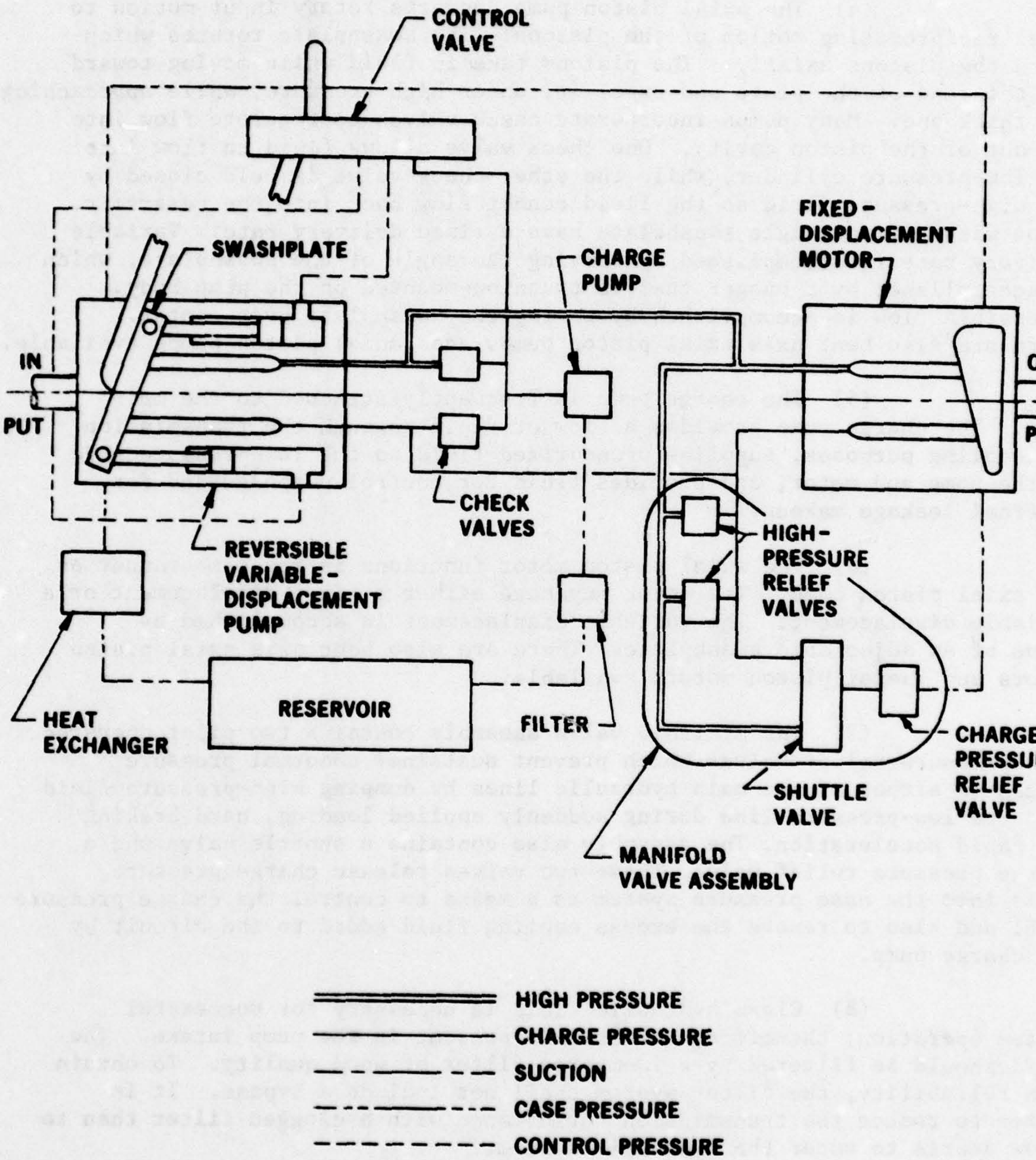

FIGURE 2. CLOSED, SPLIT CIRCUIT HYDROSTATIC SYSTEM SCHEMATIC

entering the outlet line. A 100-mesh screen should be placed over the outlet port. The oil level in the reservoir should be above the charge pump inlet. The reservoir inlet should allow the flow to be below the normal fluid level to provide maximum dwell and efficient deaeration of the fluid.

(10) A heat exchanger or oil cooler is frequently required to dissipate the heat generated. The maximum continuous operating temperature shall not exceed 180°F, however intermittent temperatures of 200°F are acceptable for short periods of time. In the cooling circuit, excess fluid enters the motor housing from the manifold charge pressure relief valve, passes through the motor housing, and then flows through the pump housing. The fluid is cooled in the oil cooler or heat exchanger and returns to the reservoir.

(11) With a variable displacement unit in the hydrostatic system, control of flow, speed, torque, and power can be accomplished. There are many types of controls including handwheel, cylinder, lever or stem servo, pressure compensator, and remote servo. The control device shown in the schematic consists of a control valve which pressurizes one of the servo control cylinders in the pump, which in turn tilts the swashplate and thus varies pump displacement. The swashplate is spring loaded to its neutral position. To reverse direction, the opposite servo control cylinder is pressurized. If circuit pressure tends to overcome the swashplate preset position, the feedback linkage from the swashplate to the control valve will actuate the control valve to compensate for the circuit pressure and maintain the swashplate preset position.

c. Performance Characteristics.

(1) Advantages of a hydrostatic transmission are as follows:

(a) Easy and accurate stepless adjustment of speed, torque, and power over a wide range of speeds.

(b) Smooth and controllable acceleration and deceleration (or dynamic braking).

(c) Maximum torque starting.

(d) Smooth and rapid starting, stopping, reversing, or changing from one speed to another for short, accurate cycling, jogging, or threading.

(e) Ability to be stalled for short periods without damage.

(f) Good speed regulation and low speed drift which permits holding a set speed within reasonable accuracy against variable driven or braking loads.

(g) Speed can be set at zero (neutral) or in either direction of rotation (reversible).

(h) Hydraulic cushioning of shock loads.

(2) The disadvantages of a hydrostatic unit are:

(a) Hydraulic system pressure can be high when developing maximum torque (up to 5,000 or 6,000 psi) depending on the system chosen.

(b) Need for a separate fluid reservoir and heat exchanger.

(c) Need for a manual control to adjust flow and direction.

(3) Efficiency of a hydrostatic transmission is the product of the pump efficiency and the motor efficiency. The efficiency of either the pump or the motor is the product of the volumetric and mechanical efficiencies. Volumetric efficiency is the ratio of actual to theoretical delivery. Mechanical efficiency is determined by bearing friction and surface rubbing. With all inefficiency considered, the overall efficiency of these transmissions may still be higher than 90%.

(4) Typical performance curves for a positive-displacement pump are shown below.

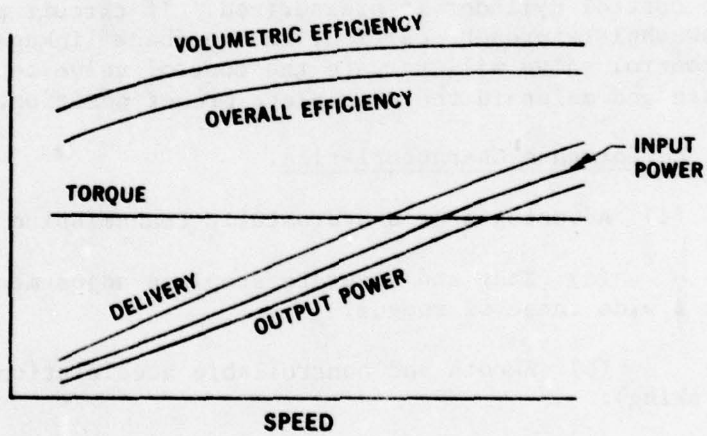

FIGURE 3. PERFORMANCE CURVES FOR A POSITIVE-DISPLACEMENT PUMP

Typical performance curves for a positive displacement motor is shown below:

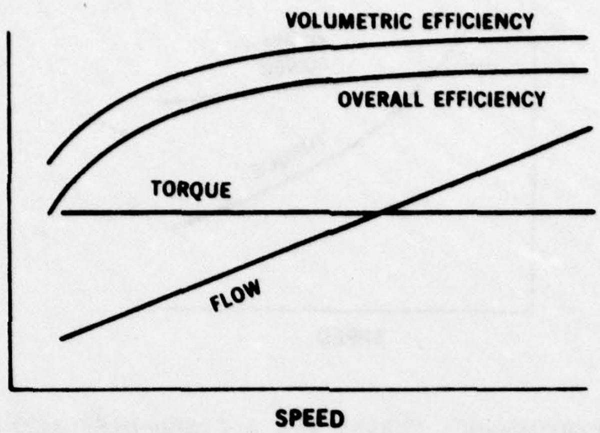

FIGURE 4. PERFORMANCE CURVES FOR A POSITIVE-DISPLACEMENT MOTOR

It is evident from the above performance curves that positive-displacement pumps and motors have very similar characteristics. This is understandable since both units are of similar construction. Motor characteristics do not include input and output power.

(5) Torque and horsepower variations with changes in speed for the three basic hydrostatic circuits are as follows:

(a) Variable-displacement pump with a fixed-displacement motor (as shown in the schematic) is the most common type of circuit used in packaged drives. Simple controls and a wide speed range are the advantages of this system. Speed is controlled by varying pump delivery.

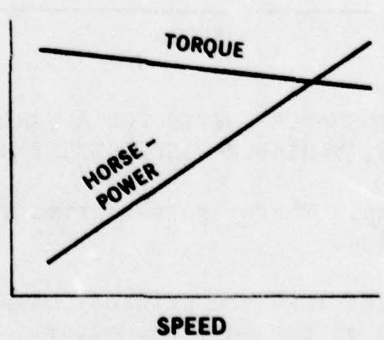

FIGURE 5. PERFORMANCE CURVES FOR A VARIABLE-DISPLACEMENT PUMP, FIXED-DISPLACEMENT MOTOR SYSTEM

(b) Fixed-displacement pump with a variable-displacement motor circuit is used in special applications such as constant tension devices. A narrow speed range is a characteristic of this type of system.

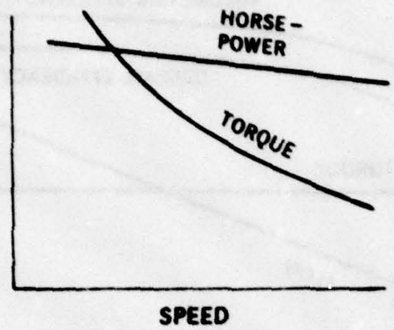

FIGURE 6. PERFORMANCE CURVES FOR A FIXED-DISPLACEMENT PUMP, VARIABLE-DISPLACEMENT MOTOR SYSTEM

(c) Variable-displacement pump with a variable-displacement motor is used where constant torque and constant power is required.

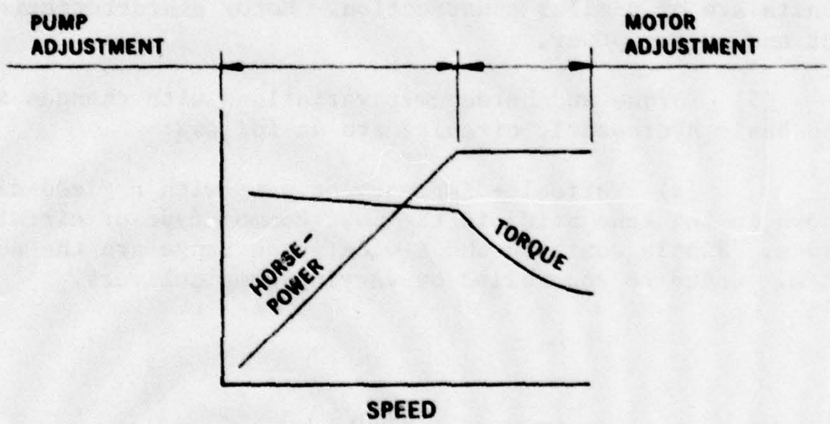

FIGURE 7. PERFORMANCE CURVES FOR A VARIABLE-DISPLACEMENT PUMP, VARIABLE-DISPLACEMENT MOTOR SYSTEM

d. Reliability. There are two primary factors that govern the operating life of the drive.

(1) The bearings are critical components, and their ratings dictate the expected life of the pump and motor. When side loads are applied to the pump or motor shaft, the bearing life will be reduced proportionately. The system is self-lubricating, since the system fluid lubricates the bearings.

(2) The sliding surfaces within the pump and motor are subject to wear and are therefore very critical. The amount of wear on these surfaces is dependent on the cleanliness of the hydraulic fluid. Good fluid filtration will eliminate most sliding surface wear.

e. Maintainability. Routine maintenance includes periodic changing of the hydraulic fluid and the filter. The fluid level must be checked regularly and any water removed from the system. Frequent visual inspection of the hydraulic lines and fittings is also necessary. Any required repair or maintenance should be accomplished as required. The heat exchanger should be regularly inspected and externally cleaned.

f. Conclusions.

(1) This drive appears to be capable of propelling the MEPP in accordance with the specified requirements.

(2) The pump may be driven off the engine front power takeoff (crankshaft forward end). Full engine power is available at this location.

(3) One motor may drive through a rear axle differential to the rear wheels, or individual motors may drive each rear wheel directly. If wheel motors are used which do not produce the exact torque and speed required, a torque hub may be coupled with the motors at each wheel. Torque hubs are planetary gear trains which support and drive the wheels.

(4) If the two torque requirements, one for acceleration and the other for curb climb, cannot be handled by a simple fixed-displacement motor, a special variable-displacement motor or a two-speed motor may be required. If this is the case, a secondary control will be necessary.

(5) The hydrostatic transmission, with wheel motors and with a motor driving through a rear axle, will both be included in the decision matrix.

2. HYDRODYNAMIC DRIVE.

a. General.

(1) Hydrodynamic drives convert the kinetic energy of hydraulic fluid in motion into output torque and rotation. There are two basic types of hydrodynamic drives:

(a) Fluid Coupling. Transmits power without the ability to change torque.

(b) Torque Converter. Transmits power with ability to change torque. The converter torque output is infinitely variable in response to the applied load, within the limits of the converter.

(2) These are both individual units with an input and output shaft. They may be combined with a power shift transmission to provide a greater range of torque and speed. The torque converter with a power shift transmission is the "automatic transmission" in the average passenger car.

(3) The torque converter without automatic gearing and the fluid coupling do not provide reverse motion without an additional reversing transmission. The automatic transmission incorporates reverse gearing. Most converters are equipped with a heat exchanger, and many require a reserve tank with a charging pump and a filter. Hydrodynamic transmission units are contained in a single package with the exception of the heat exchanger. These transmissions necessitate a continuous, rigid drive train from engine to rear axle. Speed control is through the engine accelerator and direction control is by means of a separate selector.

b. Description - Torque Converter.

(1) A typical rotating housing torque converter consists of three parts: impeller or pump, turbine, and stator or reactor (see Figure 8). The impeller is driven by the engine, and directs fluid to the turbine, which imparts motion to the output member. The fluid leaving the turbine enters the stator or reactor where it is redirected to the proper re-entry angle and directed into the impeller for recirculation.

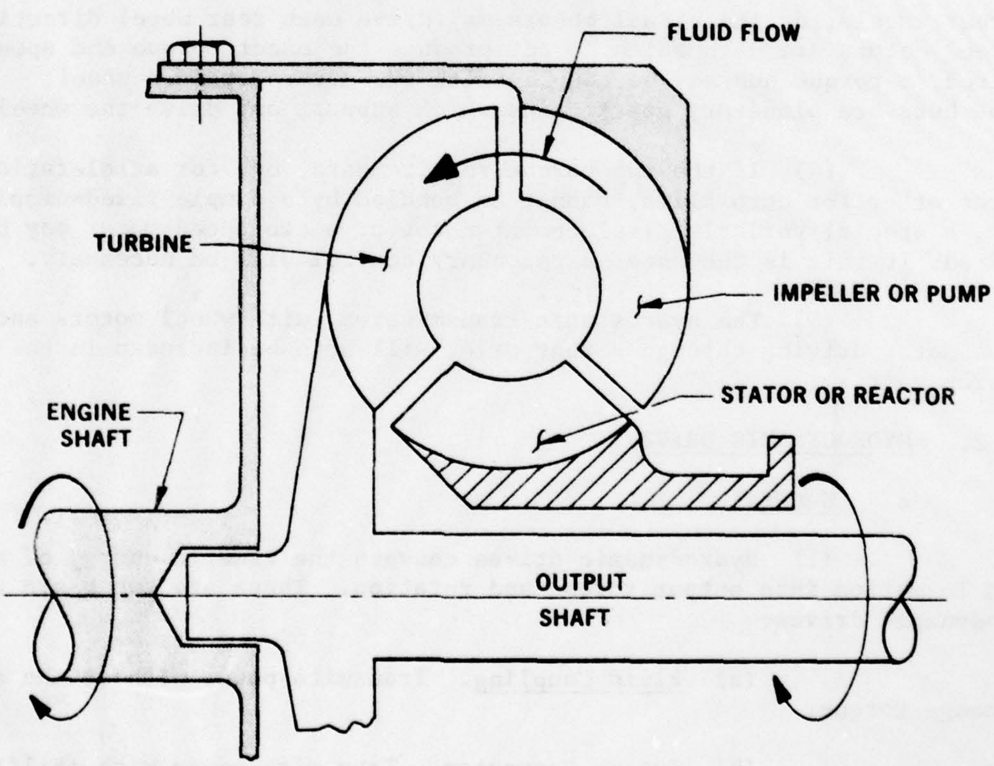

FIGURE 8. TYPICAL ROTATING HOUSING TORQUE CONVERTER

(2) When the loading on the output shaft is light, the fluid passes easily through the converter, since there is little resistance offered by the turbine. Under this condition the fluid strikes the turbine and stator blades at slight angles. When the loading on the output shaft is heavy, the turbine offers more resistance which causes the fluid to strike the turbine and stator blades at sharper angles, which in turn increases the output torque.

(3) There are two basic groups of converters, high torque ratio and low torque ratio. High torque ratio converters have a torque ratio at stall of approximately 4.5:1 to 6:1. They have a stationary housing and are available in the following configurations:

(a) Single stage. The single-stage converter has a single row of blades in the turbine wheel. It is capable of considerable hydrodynamic braking. In many applications this converter can be used without the aid of a transmission, since it has a relatively broad speed range. It is used in cranes, hoists, shovels, etc.

(b) Two Stage. The two-stage converter has a double row of blades in the turbine wheel. The maximum torque ratio attainable is approximately 4:1. These converters are well suited to use on bus applications.

(c) Three stage. The three-stage converter has three rows of blades in the turbine wheel. It is the most commonly used in this country for both industrial and automotive applications.

(4) Low torque ratio converters have a torque ratio at stall of approximately 2:1 to 3.5:1. They are normally single-stage, rotating housing type units, and are used for many industrial applications. Because of their low torque multiplication they are usually used with a three- or four-speed gear transmission.

(5) Torque converter fluid must be cooled to maintain proper operating temperatures. Therefore a heat exchanger is normally required. The heat exchanger takes the cooling medium from the engine radiator, when an engine radiator is available.

(6) Converters that have a reserve tank with a charging pump, vent the converter fluid system to the tank to allow for fluid expansion. At the same time, entrapped air in the fluid is vented to atmosphere. The charging pump keeps the system full and maintains a system pressure of approximately 50 psi. Cavitation can occur if system pressure is too low. A filter in the pump line keeps the fluid clear.

(7) Adjustable torque converters are also available. Adjustment is accomplished by varying the direction or pitch of guide vanes which replace the stator in the fixed converter. When the guide vanes are closed, the flow of power is interrupted. As the guide vanes are opened, power transmission is proportionately increased. This adjustment can be remotely controlled by a control lever. The actual controller may be mechanical, pneumatic with operating cylinder, electric with servo-motor, or electrohydraulic with control valve.

(8) The automatic transmission consists of a torque converter and a gear train, where the output of the converter is modified by various gear combinations to produce the best speed and torque characteristics for various transmission loads. Normal gear ratio changes are affected automatically without manual assist. The clutches are normally hydraulically actuated multiple disc and the gearing is constant mesh planetary type.

(9) Automatic transmissions may have almost any desired range depending on the number of gear combinations and the characteristics of the gears. Generally these transmissions have from one to five forward speeds and one or two reverse speeds. Gear ratios and direction are selected with a single control.

c. Description - Fluid Coupling. Fluid couplings are similar to torque converters with the exception of the stator. They consist of an impeller and a turbine. Figure 8 would apply to a fluid coupling if the stator was removed and the turbine and impeller filled the entire space. A fluid coupling is shown in Figure 9. The impeller is driven by the engine which imparts motion to the turbine.

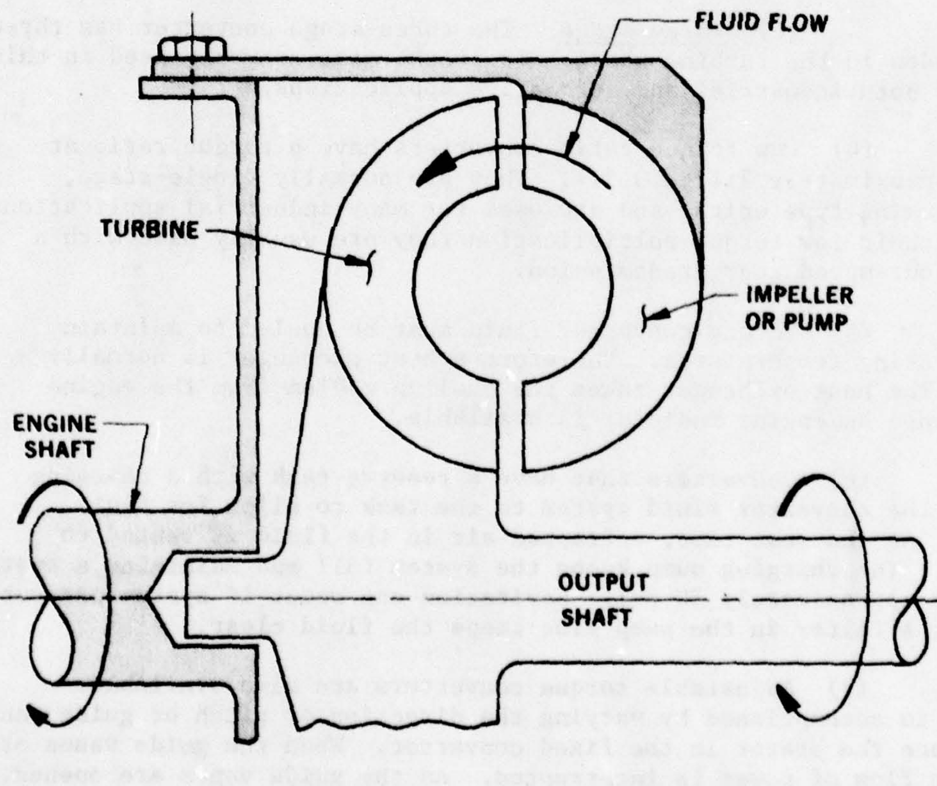

FIGURE 9. FLUID COUPLING

There are two types of fluid couplings: the constant-fill type and the variable-fill type.

(1) Constant-Fill. This coupling contains a constant amount of fluid. It is the normal or basic type of coupling and does not provide any speed regulation. Its main function is to smoothly accelerate a load up to full speed. Normally a heat dissipation device is not required.

(2) Variable-Fill. In this unit the amount of fluid in the coupling may be varied to obtain speed control. The coupling is located within an outer housing which contains the hydraulic fluid. Fluid is supplied to the coupling by a pump which provides a constant volume. The actual amount of fluid between the impeller and the turbine is controlled by an adjustable scoop tube. These units provide all the advantages of fluid couplings with speed control. The variable-fill coupling operates like a constant-fill coupling when the coupling is completely filled.

d. Performance Characteristics - Torque Converter.

(1) Load shocks, vibrations, and torsionals are damped out because torque is transmitted through a fluid mass. An infinite variety of ratios are available, within the converter power limits, an exact proportion to load demands. Power absorption characteristics are the same for all converters and are as follows:

(a) Power varies with cube of speed.

(b) Power varies with fifth power of circuit diameter.

(c) Power varies with number and shape of impeller, turbine, and stator blades.

(d) Torque varies with square of speed.

(2) Output performance characteristics of torque converters can be obtained by three different methods:

(a) K-Factor. The K-Factor is the given speed divided by the square root of the measured torque. This factor is plotted against speed.

(b) Primary-Torque. The Primary-Torque is the torque absorbed by the converter when the input speed is set at 1700 RPM. The various torque measurements are taken over the entire operating range. With the torque-speed relationship, torques at other speeds can be obtained.

(c) Torque-Absorption. At various speed ratios torque is plotted against speed for the entire operating range. This provides a family of curves, and is the most common and simple way to present performance characteristics.

(3) General engine-converter characteristic curves for the various types of converters are as follows:

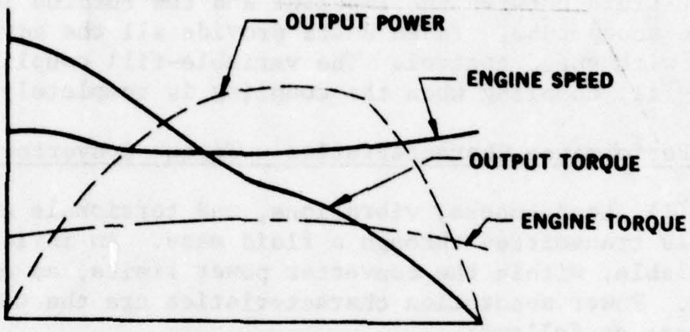

FIGURE 10. CHARACTERISTIC CURVES FOR A HIGH TORQUE RATIO, SINGLE-STAGE TORQUE CONVERTER

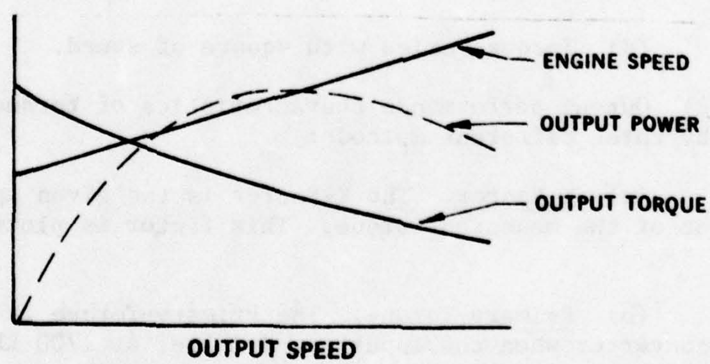

FIGURE 11. CHARACTERISTIC CURVES FOR A HIGH TORQUE RATIO, TWO-STAGE TORQUE CONVERTER

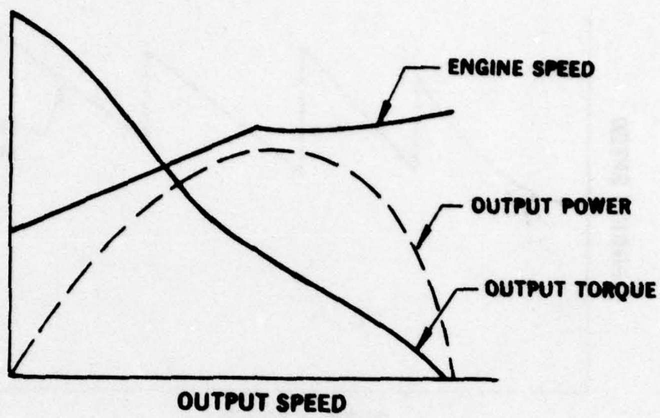

FIGURE 12. CHARACTERISTIC CURVES FOR A HIGH TORQUE RATIO, THREE-STAGE TORQUE CONVERTER

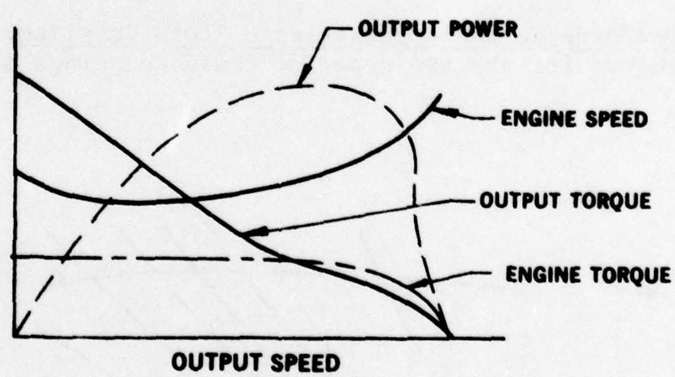

FIGURE 13. CHARACTERISTIC CURVES FOR A LOW TORQUE RATIO, SINGLE-STAGE TORQUE CONVERTER

(4) The performance characteristics of the torque converter will be modified by the output gearing in an automatic transmission. The engine speed curve will have steps equivalent to the number of gear ratios employed. For a typical four-speed transmission, the output speed curve would be as follows:

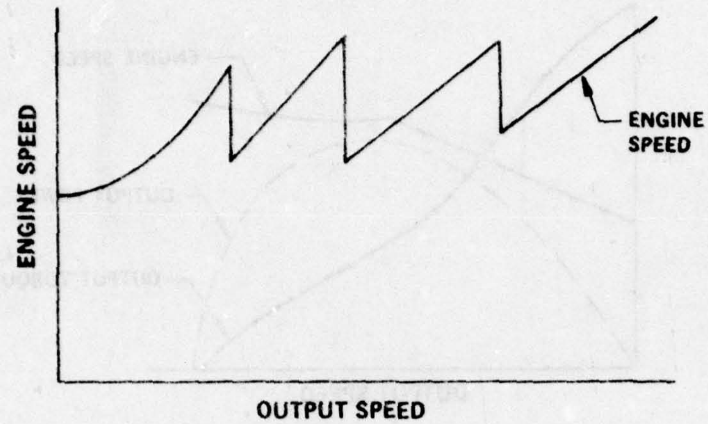

FIGURE 14. SPEED CURVE FOR A TYPICAL FOUR-SPEED AUTOMATIC TRANSMISSION

Within each speed range the output torque and efficiency characteristics as shown for a torque converter without gearing, would be duplicated.

e. Performance Characteristics - Fluid Coupling. General characteristic curves for the two types of fluid couplings are as follows:

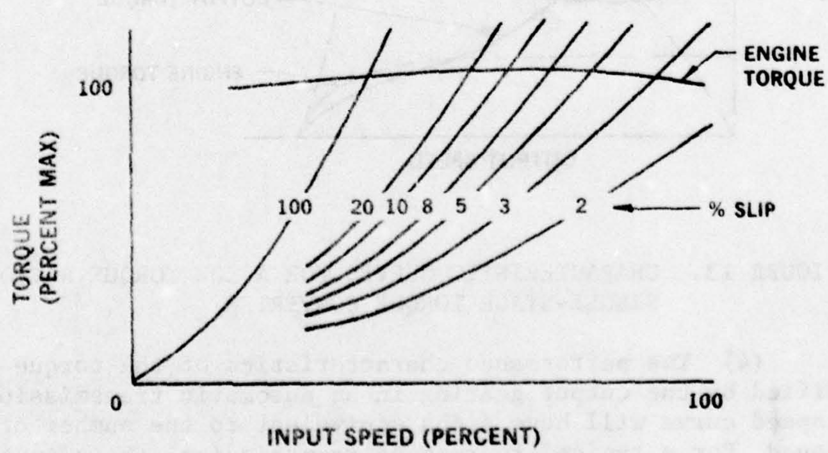

FIGURE 15. CHARACTERISTIC CURVES FOR A CONSTANT-FILL FLUID COUPLING DRIVEN BY AN INTERNAL COMBUSTION ENGINE

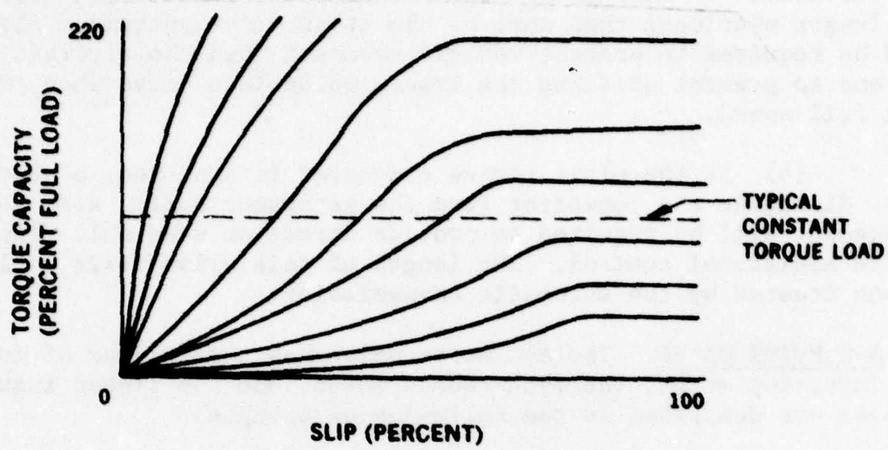

FIGURE 16. CHARACTERISTIC CURVES FOR A VARIABLE-FILL FLUID COUPLING USED AS A VARIABLE SPEED DRIVE

f. Reliability. The expected life of the torque converter and fluid coupling is dependent on the life of the bearings, which are the critical components. The torque transmitting elements are not sensitive to a small amount of debris in the fluid, and therefore high levels of fluid filtration are not required to extend the life of these elements. Since the bearings are lubricated with the converter fluid, reasonably clean fluid is necessary for this purpose. The automatic transmission gear assembly life is dependent on the bearing life, gear tooth wear, clutch wear and control system clearances. Reliability of these transmissions is good based on general knowledge of typical automotive transmissions in general use today. Reasonably clean fluid is also required for the gear assembly since all of the critical wear areas are lubricated by the same fluid.

g. Maintainability. Routine maintenance includes periodic changing of the hydraulic fluid and filters. Most torque converters and automatic transmissions contain replaceable filters. Frequent visual inspection of the converter or fluid coupling and its associated equipment is also necessary. Any required repair or maintenance should be accomplished as required.

h. Conclusions.

(1) Either a plain torque converter or an automatic transmission appears to be capable of propelling the MEPP in accordance with the specified requirements.

(2) The fluid coupling is not capable of propelling the MEPP in accordance with the specified requirements since it can not multiply torque in response to the applied load. Therefore, the fluid coupling will not be included in the decision matrix.

(3) The automatic transmission may be installed behind the generator and driven off the generator shaft. It may be possible to modify the generator by adding a flywheel housing type adaptation and extending the shaft. The transmission could then be attached to the generator in the same manner as it would be attached to an engine. This continuous, rigid drive train will require a longer wheelbase than some of the other drive systems. Also an interlock would be required to prevent vehicle movement when the aircraft is being serviced, and to prevent shifting the transmission into drive when the engine is running at full speed.

(4) If the plain torque converter is employed, a clutch will be required to disengage the converter from the generator shaft, also an additional gear arrangement will be required to provide direction reversal. These components will require additional control. The length of this drive train would be longer than the one created by the automatic transmission.

3. A-C MOTOR DRIVE. The A-C motor drive has three types of motors: the polyphase induction motor, the synchronous motor, and the linear induction motor. These are described in the following paragraphs.

a. A-C Motor Drive - Polyphase Induction Motor.

(1) General. The induction motor is the most common type of alternating-current machine. It is relatively cheap and simple and can be manufactured with particular characteristics to suit one's needs. It consists of two main parts: a rotor and a stator. The rotor may be wound in two ways: a squirrel-cage winding, or a wound rotor winding. Normally, it is a singly excited machine with the field winding receiving the excitation potential. It may be installed in an austere environment and perform well despite the presence of moisture or dust in the atmosphere. Simplicity and ruggedness are two of its major assets.

(2) Description.

(a) The stator, or stationary element in the motor, has wire coils wound around it for connection to the power supply. The rotor is a laminated structure with slots in it into which copper or aluminum bars are placed. The bars are all connected at the ends.

(b) When a polyphase alternating current is applied to the stator winding, a resultant magnetic field rotates around the stator at a speed dependent on the supply voltage's frequency. The stator is like the primary winding in a transformer, whereas the squirrel-cage rotor acts as the secondary winding. A voltage is induced into the rotor bars and a current will flow in these bars. The current's magnitude is limited by the rotor's impedance. As the stator field turns, the rotor field will be compelled to follow it and start revolving. There is a maximum theoretical speed, called synchronous speed, where the rotor would rotate at the same speed as the stator field. This speed actually can never be attained because the rotor bars wouldn't cut the stator field flux, no voltage would be induced and no current would be generated. Since the rotor wouldn't have current flowing in it, no torque could be developed. There must be a difference in speed between the rotor and the stator field. This is a phenomenon called slip. A finite amount of torque must be developed just to overcome friction core losses and windage in the rotor.

(c) The motor torque is determined by the particular design, especially the resistance of the rotor conductors. A standard motor has a fairly low resistance rotor, low starting torque, and low running slip. Variations of these characteristics can be attained at the manufacturing stage, such as a high starting torque motor. The following equation describes the average torque developed by an induction motor, either running or at standstill:

$$T = K_T \phi I_R \cos \theta_R$$

WHERE: T = total torque
K_T = a constant of the motor
ϕ = effective value of stator flux
I_R = effective rotor current
θ_R = power-factor angle of rotor

Additional analysis of this torque equation for running conditions yields:

$$\cos \theta_R = \frac{R_R}{\sqrt{R_R^2 + (SX_R)^2}}$$

$$I_R = \frac{SE_R}{\sqrt{R_R^2 + (SX_R)^2}}$$

WHERE: R_R = effective combined rotor resistance

$X_R = 2\pi F_{ROTOR} L_R$ = combined rotor reactance

E_R = effective value of rotor induced electromotive force (emf)

L_R = effective combined rotor inductance

S = slip

$F_{ROTOR} = Sf_{LINE}$

THEREFORE:

$$T_R = K_T \phi \left(\frac{SE_R}{\sqrt{R_R^2 + (SX_R)^2}} \right) \left(\frac{R_R}{\sqrt{R_R^2 + (SX_R)^2}} \right)$$

AND $$T_R = \frac{K_T \phi S E_R R_R}{R_R^2 + X_R^2}$$ = total running torque

(d) Once the motor is built and the rotor is inaccessible, the starting torque of the motor depends on the stator line voltage. Typical starting torques vary from 125% of full-load torque to 300% of rated torque, depending on the values of R_R and X_R and the line voltage selected.

(e) The rotor speed of an induction motor is defined through the following equation.

$$N_R = \frac{120f}{p}(1-S)$$

WHERE: N_R = rotor speed (RPM)
f = line frequency
p = number of poles
S = percent slip

Therefore frequency, slip, and the number of poles are the variables controlling speed in an induction motor. Varying the input voltage can also affect the motor's speed but only over a limited speed range (usually 3:1). Frequency control is the most popular technique used to control the speed of induction motors and this is due in part to the reliability of the solid state frequency changers available today.

(f) The thyristor is a principal component in many of today's frequency changers. Its switching time is relatively quick, enabling it to produce adjustable frequency quite readily. Several avenues of approach are employed using thyristors or silicon controlled rectifiers (SCR). One is the cycloconverter. It is composed of several switches connected between the load and power source. When properly gated on and off, these switches will present the load with an approximated sine wave. Thirty-six thyristors are needed for a full three-phase output. There is minimum heating involved and full reversing capabilities. The efficiency is high; however, the low frequency output inhibits its use with commercial a-c motors.

(g) Another approach is called an inverter with d-c link. This method takes a d-c source and controls the thyristors conducting periods to produce square wave pulses. Modulating pulse width while controlling frequency lends this technique to be called pulse-width modulation (PWM).

(h) It is important to control the voltage as well as the frequency in these frequency converters. Chopper-type rectifiers produce an adjustable-voltage direct current, which is fed to an inverter for frequency control. There are adjustable-voltage constant-frequency drives for motors as well as slip-power controlled drives used on wound rotor motors. Transistors are now employed in inverter drives; however, the use of transistors is presently limited to about 15 KVA applications due to heat dissipation problems.

(i) With the advent of the microcomputer, it is quite possible to control the input voltage and frequency of the motor through a dedicated a-c synchronous generator. Software could be used to control the voltage and frequency of the generator driving the motor. Therefore, variable speed could be attained. A switching mechanism would be required to disconnect the motor's supply and prohibit movement while servicing an aircraft, and for reversing purposes.

(j) The poles on the stator of an induction motor are created by the arrangement of the conductors. A switching arrangement could be produced to interchange the coil's connections and therefore change the number of poles. Different speeds could be attained by switching the number of poles in the motor. However, the change in speed would be abrupt rather than a continuous smooth transition. Normally, pole changing is performed on the squirrel-cage motor due to simplicity.

(k) Inserting resistance in the rotor to change the slip is one method of controlling speed. This variable resistance rotor has two forms: a double squirrel-cage motor and a wound rotor motor. First, the former rotor will be discussed.

1. It is often advantageous to have an induction motor with a high starting torque and low starting current, but a low running slip. Low starting current is desirable so that the supply line won't be adversely affected by a sudden large current draw. Low running slip is appealing so that there isn't too much heat lost in the rotor and for better speed regulation under load. A double-cage rotor has two squirrel-cage windings in the same rotor core. The first winding has low resistance and a high reactance. The second winding has a high resistance and a low reactance. At standstill, when the rotor current frequency equals line frequency, most of the current flows through the low reactance winding. As the speed of the motor increases, the rotor's frequency decreases and the reactance of the first winding becomes less effective, and more current flows through the higher reactance winding. Therefore, the rotor's characteristics have changed to a lower slip, higher current rotor while running.

2. Second, the wound rotor motor will be examined. This motor has a wire winding in the rotor rather than conducting bars unlike the squirrel-cage motor. The wires are not connected to each other at the ends. They are attached to slip rings; brushes situated on these rings present a method of connecting to them. This method allows for the insertion and removal of an external resistor into the rotor circuit to achieve high starting torque and low starting current as well as low running slip which are all desirable. Varying this external resistance presents an approach whereby speed regulation may be attained through control of the motor's slip characteristics. However, control of speed is limited to about a 2:1 range.

(3) Performance Characteristics.

(a) The following graph depicts the characteristics of a normal squirrel-cage induction motor and a motor with a high resistance rotor. From this graph, we can see that the normal motor has a starting torque of about 145% of full load. The maximum torque available, or pullout torque, is about 280% of full load. The high resistance rotor motor exhibits a much higher starting torque of close to 300% rated full-load torque. One can readily see how available starting torque increases with rotor resistance. Also, the normal motor has starting currents of from six to ten times full load running current when starting on full voltage from a power supply. The higher starting torque motor, however, has a starting current of only about 400 to 500% of a normal full-load value. Therefore, its starting characteristics would be advantageous for our application, high torque and low current. Nevertheless, under running

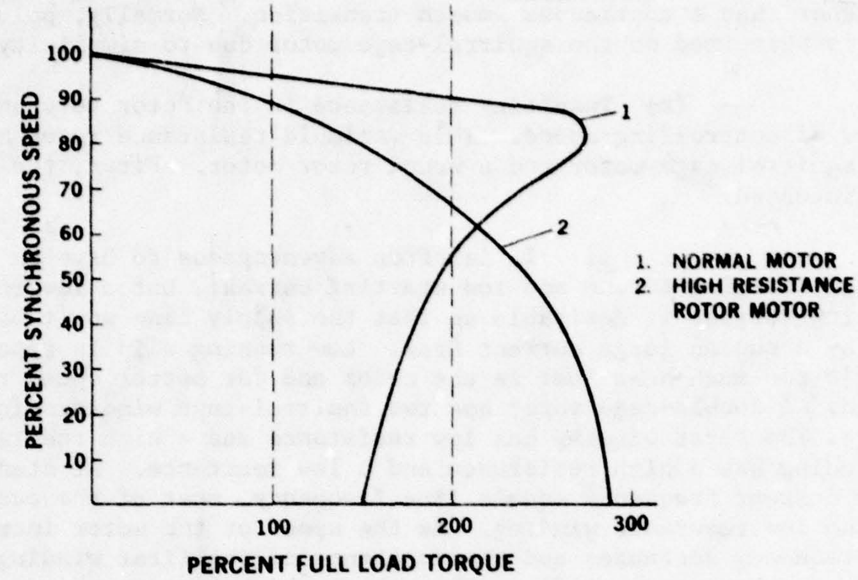

FIGURE 17. SPEED-TORQUE CHARACTERISTICS OF SQUIRREL-CAGE INDUCTION MOTORS

conditions the higher slip inherent in such a motor causes higher heat losses in the rotor and greater speed variations with applied load. Therefore, the running characteristics would have to be altered. A double squirrel-cage motor has the advantages of both types of motors. It will start off with a high resistance rotor with high torque and slip and low current. As the speed increases, the slip and torque will decrease and the current will increase, which is desirable.

(b) The wound rotor motor presents a method of being able to vary the rotor resistance through a slip ring and brush arrangement. As depicted below, the starting torque increases with increasing resistance and the current decreases.

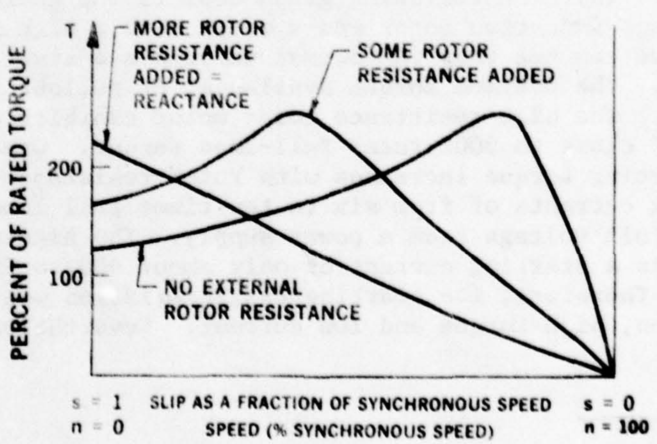

FIGURE 18. EFFECT OF INCREASED ROTOR RESISTANCE ON TORQUE AND SPEED

(c) This method is an effective way of controlling the speed of the motor; however, there are slip rings and brushes and a wound rotor which are less maintenance-free than the squirrel-cage motor.

(d) The magnitude of the voltage applied to the stator does have an effect on the speed and torque of an induction motor. The torque of an induction motor varies as the square of the voltage. Therefore, increasing the line voltage when starting could quickly develop enough torque to overcome internal losses and initiate the motor's starting. The following graph better illustrates the effect of a change in voltage over the entire speed range.

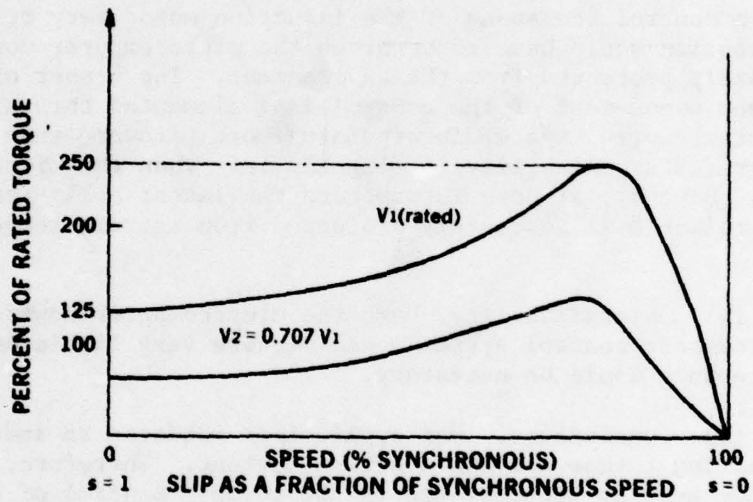

FIGURE 19. EFFECT OF REDUCED VOLTAGE ON TORQUE AND SPEED

(e) The speed adjustment using this method isn't very great when compared to the price paid. When the motor operates at reduced voltage, its reserve capacity for available torque is greatly reduced as is the speed and efficiency. Therefore, reducing the voltage has the disadvantage of reducing the available torque for a given speed.

(f) Another control scheme is the altering of the resistance to reactance ratio of the rotor. Adding resistance decreases the current and torque, therefore, the speed decreases. Speed control can be obtained down to about 50 percent of synchronous speed. However, as the speed decreases, the slip increases and the power delivered to the load decreases. The power is dissipated as heat in the external resistors and is not recoverable. This method allows about a 2:1 speed ratio as compared to about 3:1 for voltage control.

(g) The reactance of the rotor can be varied through frequency control. For a given rotor resistance, increasing the applied voltage's frequency will decrease the available torque for a given slip and increase the current. Reducing the frequency will increase the core losses and exciting current. This will precipitate a decreasing efficiency. Therefore, if one is to control the speed through frequency, it is also necessary to control the voltage simultaneously to maintain operable current levels.

(4) Reliability. The a-c squirrel-cage induction motor is the most reliable motor available. It can be installed in places where dirt or moisture are present and perform satisfactorily with very little attention. It can also withstand high inrush currents without sustaining damage. A microcomputer is a very reliable piece of digital electronics. When suitably protected from adverse environmental conditions such as extremes in temperature and moisture, it can control the speed of the induction motor very reliably. Again very little attention would have to be given the microcomputer control circuitry if it is adequately protected from the environment. The number of components is kept to a minimum since most of the control is implemented through software. A rectifier-inverter control system incorporates more hardware than the microcomputer system; therefore, its reliability is slightly less than the microcomputer control system. However, it does incorporate the latest solid-state components which are very reliable if adequately protected from extreme temperature, current, and humidity.

(5) Maintainability. Both the microcomputer control system and the rectifier-inverter control system would require very little attention and almost no maintenance would be necessary.

(6) Conclusions. Our application mandates an induction motor with a high starting torque and low starting current. Therefore, a larger than normal resistance must be incorporated in the rotor. Control of speed and torque following starting could be implemented in two ways. First, a conventional rectifier-inverter method could be employed necessitating purchase of a commercial unit. Second, the voltage and frequency of the synchronous generator available on the MEPP could be controlled through the use of a microcomputer. Since a microcomputer will be utilized to control voltage and frequency while the MEPP is servicing planes, it could be dedicated to driving a motor while in the propulsion mode. Both alternatives will be incorporated in the matrix decision process later in this report.

b. A-C Motor Drive - Synchronous Motor

(1) General. Synchronous motors are inherently constant-speed motors, where the speed is usually synchronized with the line frequency and independent of load. They are divided into two major types: nonexcited and direct-current excited. The nonexcited motors such as the reluctance, hysteresis, split-phase, capacitor-start, repulsion-start, and shaded-pole designs are built in the fractional horsepower range. Since they are too small for our application, no further discussion on them will follow. Synchronous motors are generally

of the salient-pole type and divided into two groups: motors rated above 514 RPM, and motors rated below 514 RPM. This rotating machine has no intrinsic starting torque; therefore, special starting provisions are required to get it up to its rated synchronous speed.

(2) Description.

(a) The construction of a polyphase synchronous motor is essentially the same as that of a synchronous generator. The rotor often has a squirrel-cage winding on it. This winding which aids starting is often called the damper winding, cage winding, amortisseur winding, or starting winding. If the rotor and stator were to be excited simultaneously from rest, no starting torque would be developed. Therefore, when starting, the rotor isn't energized initially and a polyphase voltage is supplied to the stator. The rotor is energized when the motor nears synchronous speed. This moment of excitation is critical. If it is applied at too low speed, or at the incorrect phase angle, high pulsating current and vibration may result. This excitation must also be removed at rotor pull-out torque (maximum torque developed by the motor at synchronous speed for one minute with rated frequency and normal excitation).

(b) Synchronous speed is dependent on the supply frequency and the number of poles for which it is wound as in an induction motor. The governing equation again is:

$$N_S = \frac{120f}{p}$$

where: N_S = synchronous speed
f = line frequency
p = number of poles

(c) When a load is placed on the motor's shaft, initially the rotor slows down. The field of the stator continues to rotate at synchronous speed, and the current drawn from the supply line increases. This increase in current develops sufficient torque to carry the load. The motor will therefore continue to rotate at synchronous speed, but the angle between the rotor field and stator field will have changed. The power and torque increase under load until the motor reaches the pull-out torque. This is when the developed torque is less than the load torque, and the motor can no longer rotate at synchronous speed. Pull-out torque varies from 150 to 300 percent of the full-load torque. It varies with field excitation for a constant applied voltage.

(d) Changing the field current not only changes the armature current, but also the power factor at which the motor operates. This is a major characteristic of a synchronous motor; to be able to vary its power factor while maintaining synchronous speed. Underexcitation produces a lagging power factor -- the armature current lags behind the terminal voltage. Increasing the excitation until the voltage and current are in phase produces a unity power factor and normal excitation. Further increase in excitation produces a leading power factor and overexcitation. As the field excitation increases, the generated voltage increases. As a result, the power developed and power factor increase. This is a major advantage of synchronous motors. In factories or plants, most equipment is inductive and produces a lagging power factor. This situation results in an increased current, and in greater I^2R losses in lines attached to a load, transformers, and in the power producing alternators. It is advantageous to have a power factor near unity for good efficiency and voltage regulation. Entering an overexcited synchronous motor in the system can improve the system's power factor. Such a motor, designed to carry no mechanical load, is referred to as a synchronous condenser.

(3) Performance Characteristics.

(a) The following graph depicts the speed-torque relationship during the starting period for synchronous motors started through induction methods.

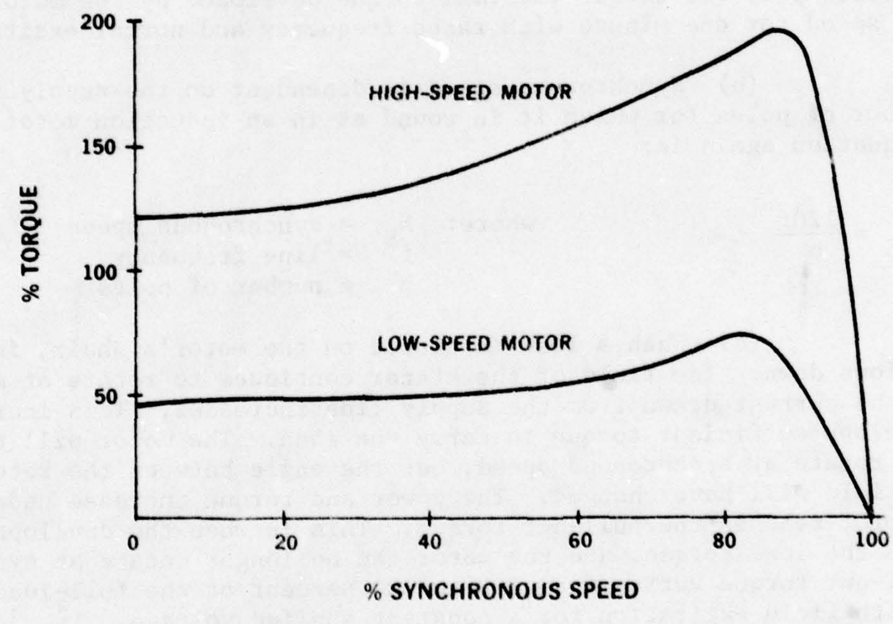

FIGURE 20. SPEED-TORQUE CHARACTERISTICS OF AN INDUCTION START SYNCHRONOUS MOTOR

One can readily see how these starting curves resemble those of the induction motor. Synchronous motors can be designed for high-starting torque by using a double squirrel-cage winding or by increasing the resistance of the damper winding. However, the motor won't approach synchronous speed so closely.

(b) As described previously, the power factor of a synchronous motor may be changed by varying the strength of the d-c field. The excitation required to produce any given power factor varies with load. An increase in load with a given power factor requires more d-c excitation to maintain that power factor. The following "V" curve depicts the excitation needed to maintain unity power factor at various loads.

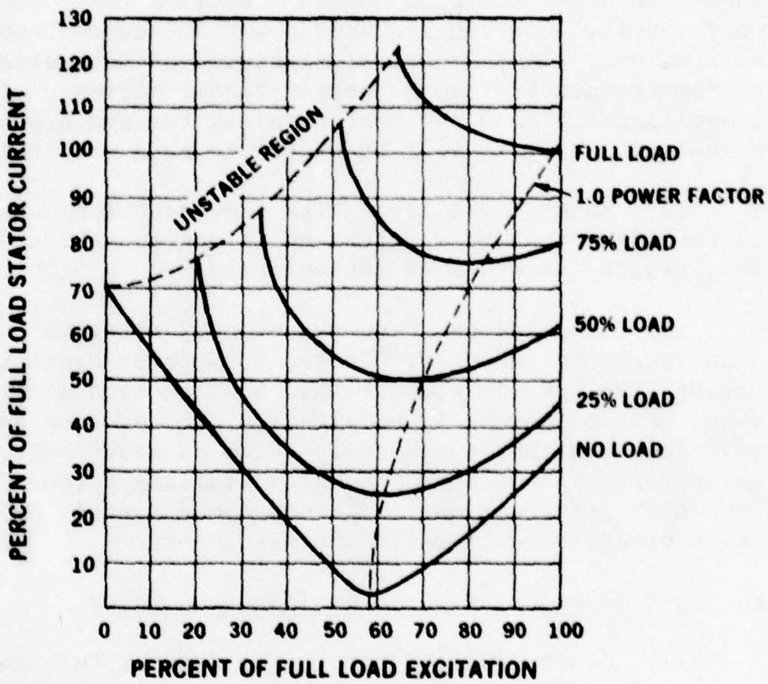

FIGURE 21. "V" CURVES OF A TYPICAL GENERAL PURPOSE 100 PERCENT POWER FACTOR SYNCHRONOUS MOTOR

(c) The synchronous motor does have some advantageous characteristics. It is sturdy and may be wound for high operating voltages. The possibility of running at unity or a leading power factor is often desirable especially when running in an area where induction motors are prevalent. Its efficiency is high, usually 1 to 2% higher than an induction motor. It is very useful in constant-speed applications. The initial cost of this motor is less than an induction motor for ratings below 500 RPM and in horsepower ranges greater than 500 horsepower. Nevertheless, the synchronous motor has distinct disadvantages also. In smaller sizes, it is considerably more expensive than an induction motor, and therefore is rarely used below 50 HP. It needs a separate source of d-c excitation. Starting and control devices are usually expensive. The inertia of the load affects the "pull-in" torque (torque developed by the motor at the speed from which it will attain synchronous speed). Low starting torque is another disadvantage of the standard synchronous motor. The motor is sensitive to system disturbances, and may fall out of synchronous speed due to these disturbances.

(4) Reliability. As a member of the a-c motor family, a synchronous motor is very reliable if given the proper operating conditions. The motor should be protected from suddenly applied loads because oscillations or hunting may result. Hunting produces power and current pulsations which may cause system problems. If the natural frequency of mechanical oscillation approximates the frequency of an important torque harmonic of a load's cycle, intolerable oscillations result. Therefore, if the synchronous motor is protected from such transient loads, it will be proven to be a very reliable motor.

(5) Maintainability. Like the induction motor, the synchronous motor should require very little attention if not abused. The shaft bearings might possibly require maintenance periodically.

(6) Conclusions. The synchronous motor is not a good choice for the propulsion system on the MEPP. In the horsepower range required for the MEPP application, the cost is greater than an induction motor. It does not lend itself to speed control as easily as either a d-c motor or an induction motor. Inherently, it is a constant speed device with no starting torque. Our requirement calls for variable speed and high starting torque. These characteristics along with the high price warrant the synchronous motor's omission from the matrix decision process incorporated in this report.

c. A-C Motor Drive - Linear Induction Motor

(1) General Description. The name of this machine aptly describes its physical characteristics. It is a squirrel-cage induction motor which has been unrolled into a flat arrangement. This motor can be arranged in two forms. First, a single-sided motor as illustrated below has one set of stator windings separated by an air gap from the armature.

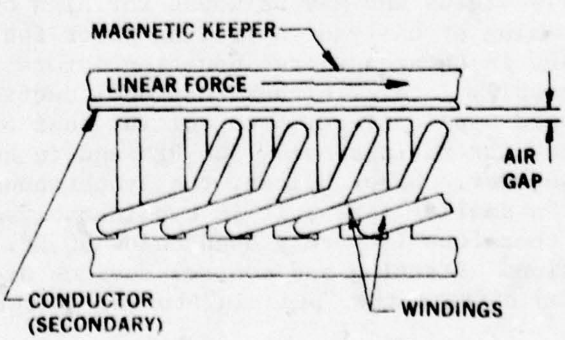

FIGURE 22. SINGLE-SIDED LINEAR INDUCTION MOTOR

Second, a double-sided arrangement has stator windings on both sides of an armature. There is an air gap between the armature and each stator.

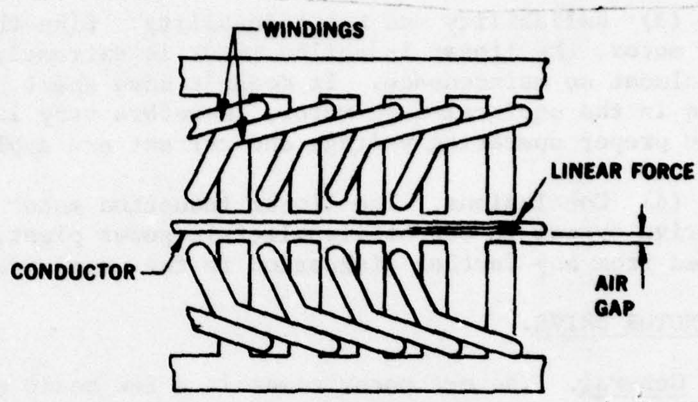

FIGURE 23. DOUBLE-SIDED LINEAR INDUCTION MOTOR

Applying a-c power to the primary winding of a linear induction motor induces a linear sweeping field in the armature. The primary and secondary windings repel each other. The nonstationary element is propelled in a linear fashion.

(2) Performance Characteristics. The motors are very quick in responding to flux changes in the stator. They accelerate rapidly initially and accelerate less rapidly as the speed increases. The speed of these induction machines can be controlled as in a rotary induction motor. Typically, the voltage and frequency are simultaneously varied to change speed without having severe current and heat dissipation problems. Linear induction motors produce a straight line motion in either direction. It would be impractical to use such a device to attain rotary motion as in our application. It is also necessary to maintain a certain air gap between the primary and secondary elements. On a carrier deck, if the deck acted as a stationary armature and a MEPP had the appropriate field windings, linear motion could be attained. An extremely strong attraction force exists between the armature and the primary. This attraction force present in the air gap would help keep the MEPP anchored to the deck. However, there would be problems encountered. A typical air gap is 1/4 inch. If the power plant, were to pass over an obstacle on the deck, this gap would change and the motion of the vehicle would be adversely affected. Speed control would again be complicated and the impracticality of using a linear machine for rotary motion prevails. On dry land the problem is even greater because a stationary member of the motor is needed. Mounting the field

windings on a rotating axle and the fixed armature parts near the wheels would require too many large stationary elements for our application. In addition to these problems, there is a noise problem due to an a-c hum at the air gap.

(3) Reliability and Maintainability. Like the squirrel-cage induction motor, the linear induction motor is extremely reliable and would require almost no maintenance. It doesn't have shaft bearings to maintain periodically as in the squirrel-cage motor, therefore very little attention is required if the proper operating voltage and current are applied.

(4) Conclusions. The linear induction motor is not a good choice for a drive system in our mobile electric power plant. Therefore, it will be excluded from any further discussion in the matrix decision process.

4. D-C MOTOR DRIVE.

a. General. The d-c motor comes in a few basic types. These types exhibit their own characteristics as far as torque and speed are concerned. D-C motors are classified according to the type of field winding. They may be series wound, shunt wound, or compound wound depending on the relationship between the field and the armature windings. A compound wound motor has a shunt and a series field winding, and therefore its characteristics are dependent on the appropriate proportion of each in the motor. Also, the field windings may receive d-c excitation from a separate external source or internally from self-excitation. The method of excitation influences both the steady-state and dynamic behavior of the machine. Therefore, the d-c motor is a versatile machine. It can operate over a wide speed range and change direction quickly. High starting torque is another noteworthy quality possibly attained in a d-c motor. Nevertheless, the machine does have its drawbacks. The need for brushes and a commutator to convert the a-c electromotive force induced in each armature coil to a unidirectional voltage makes it expensive, and less rugged than an a-c motor.

b. Description.

(1) Shunt Motor - A shunt motor has the field winding in parallel with the armature as illustrated below:

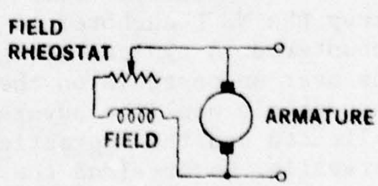

FIGURE 24. D-C SHUNT MOTOR

The field current is constant with a given line voltage, but adjustable, through the use of the field rheostat. Motor speed is also relatively constant with varying loads. The shunt motor's field winding is stationary while the armature is a rotating device. A commutator is positioned on the armature shaft. Its function is to maintain a unidirectional torque in the machine and it acts as a rectifier. Carbon brushes mounted on the commutator provide a means of carrying direct current to and from the armature. The resistance of the shunt field winding is usually fairly high, since it is directly connected to the supply voltage. This resistance is dependent on the motor's application requirements and initial design.

(2) Series Motor - Series motors have the field winding connected in series with the armature. The load current passes through the series field; therefore, the strength of the field will vary with the load. Increasing load will cause a decreasing speed in the motor. The following illustration depicts the arrangement of the field and armature in a d-c series motor.

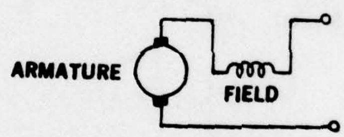

FIGURE 25. D-C SERIES MOTOR

The field windings are usually constructed of heavy wire and connected to a rather long armature of small diameter. The commutator and brush assembly is mounted as in the shunt motor. The series motor is capable of producing large starting torques and operating over a wide range of speed. These characteristics readily illustrate how it has been traditionally used as a traction motor in mobile applications.

(3) Compound Motor - A compound motor has both a shunt and a series winding as shown below.

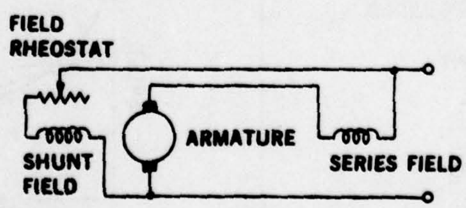

FIGURE 26. D-C COMPOUND MOTOR

Usually a compound motor is wound to have the series field's characteristics aid the shunt field, especially where higher starting torque is desired. This arrangement is called a cumulatively compounded motor. The characteristics of a cumulatively compounded motor are obtained from both windings. Its behavior is intermediate between the two windings, but can approach the features of either depending on the relative strength of the two windings.

c. Performance Characteristics.

(1) The performance characteristics of the three types of d-c motors will be discussed coincidentally. It is easier to compare the advantages of each and the methods of speed control used. The discussion will start with a graphical illustration of the contrasting characteristics of the d-c motors. The speed torque relationship is as shown below:

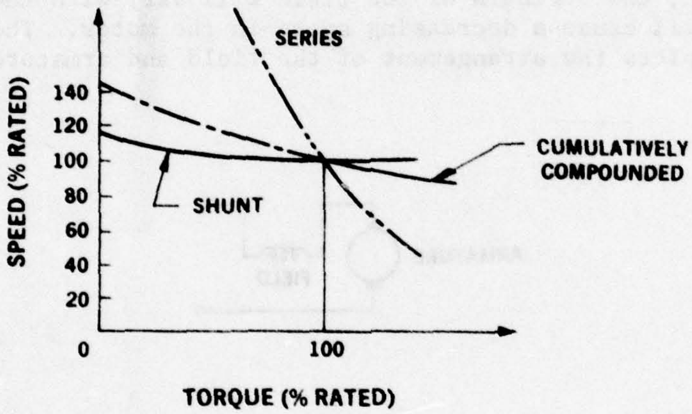

FIGURE 27. SPEED-TORQUE RELATIONSHIP FOR D-C MOTORS

It is evident that the shunt wound motor maintains a fairly constant speed regardless of load. The series motor's speed decreases with load and the compound motor's characteristics lie midway between the two.

(2) An important parameter in a d-c machine is the armature current. Figures 28 and 29 project the general relationships between torque, speed, and armature current.

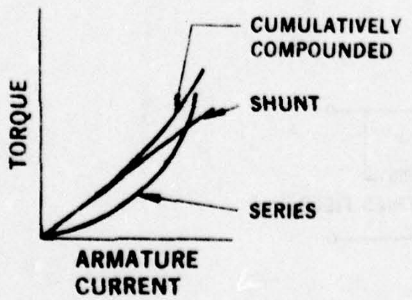

FIGURE 28. CURRENT - TORQUE CHARACTERISTICS OF A D-C MOTOR

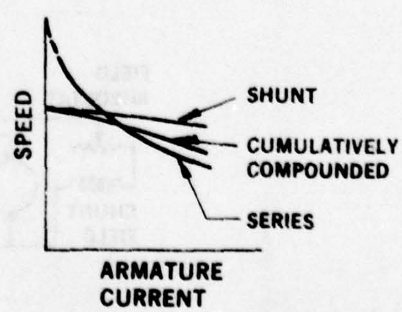

FIGURE 29. CURRENT - SPEED CHARACTERISTICS OF A D-C MOTOR

(3) There are a few equations which help to clarify the operation of a d-c motor. For any d-c motor:

$$T = K\phi_A\phi_F$$

where:
T = torque of the motor
K = proportionality constant
ϕ_A = flux produced by the armature
ϕ_F = flux produced by the field

The armature flux is proportional to armature current; therefore, the equation may be reduced to

$$T = K_T\phi I_A \qquad (1)$$

where:
K_T = total proportionality constant
ϕ = field flux
I_A = armature current

The armature current may be further defined as

$$I_A = \frac{V_T - E_a}{R_a + R_s}$$

where:
V_T = terminal voltage
E_a = counter emf generated in armature
R_a = armature resistance
R_s = series field resistance

The general equation relating to speed of a d-c motor is

$$S = K\frac{V_T - I_a R_a}{\phi} \qquad (2)$$

where: S = speed
K = proportionality constant
V_T = applied voltage
I_a = armature current
R_a = armature circuit resistance
ϕ_a = motor field flux.

(4) Equations (1) and (2) together with the method of excitation and magnetization curve help to determine the torque-speed characteristics of a d-c motor. It is readily apparent that four quantities can be controlled to vary the speed. These quantities are: applied voltage, armature current, armature resistance, and field flux.

(5) The three most common methods of controlling a d-c motor's speed are adjustment of the armature's resistance, adjustment of the flux (usually through a shunt field rheostat), and adjustment of the armature terminal voltage.

(a) Armature circuit resistance control may be used with all three types of d-c motors, series, shunt, and compound. A series resistance is inserted before the armature circuit. For a given value of series resistance, the motor speed will vary widely with the applied load. Therefore, the speed regulation and operating efficiency is adversely affected with this control scheme. There is a certain power dissipated in this series resistor which can be large, particularly at low speeds. Costs of operation may be high if a motor runs at low speeds for a large percent of its running life. However, the initial cost of such a system is low and it may be an attractive method of speed control for short-time or intermittent operation at reduced speeds. This method offers a constant-torque drive unlike the next method of control, shunt-field rheostat control.

(b) A commonly used method of speed control in shunt and compound motors is shunt-field rheostat control. Zero resistance or full field strength corresponds to the lowest speed attainable. Armature reaction limits the highest speed that can be obtained when weak field conditions exist. Stabilizing or compensating windings help to increase the speed range to a maximum of about 8 to 1. This drive control scheme is commonly called a constant-horsepower drive where the torque varies directly with the field flux. Maximum torque occurs at the lowest speed.

(c) Armature-terminal voltage control, which is often referred to as the Ward Leonard system, is often used when control of speed over a wide range in both directions is needed. The conventional method of speed control, as illustrated, is very similar to a method which would be used for our application.

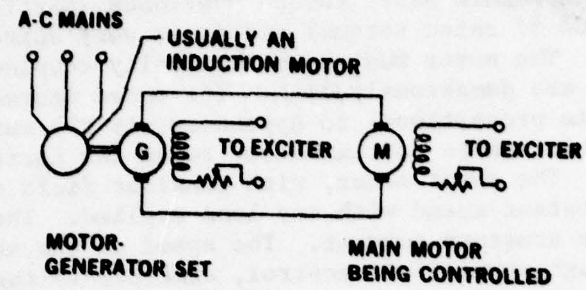

FIGURE 30. WARD LEONARD CONTROL SYSTEM

(6) The generator's output voltage varies with the strength of its field flux, which is controlled by the field rheostat. The varying voltage is applied to the motor's armature terminals for a smooth control of speed. A wide range of speed is attainable using generator voltage control alone. Control of the motor's field flux through the field rheostat, when combined with the voltage control, permits an even wider possible speed range. Voltage control could be used to achieve speeds below the base speed of the motor. At these speeds, constant torque is developed because the flux and armature current are fairly constant. Field rheostat control would permit speeds above the base speed in a constant horsepower mode. Typical torque and horsepower curves are presented below.

FIGURE 31. EFFECTS OF ARMATURE VOLTAGE AND FIELD RHEOSTAT CONTROL ON SPEED, TORQUE, AND POWER

Maximum speeds are normally limited to four times the base speed.

(7) D-C motors are flexible, easy to control, and versatile. The series motor can handle heavy torque overloads, particularly while starting (up to 500% of rated torque) and it is very suitable for traction-type applications. The motor must be mechanically coupled to a load because the no-load speeds are dangerously high. Its speed decreases with increasing load. The torque is proportional to approximately the current squared at low saturation levels and to approximately twice the current at higher saturation levels. The shunt motor, with constant field current, operates at a relatively constant speed with any load applied. The torque is almost proportional to the armature current. The speed of the shunt motor usually is controlled through shunt-field control, armature-voltage control, or a combination of the two. The cumulatively compounded motor, having both a series and a shunt field winding, has characteristics which are intermediate between the series and shunt motor. It can exhibit the characteristics of either type depending on the relative field strengths. The shunt field winding helps to limit the no-load speeds to safe values. A noteworthy attribute of the shunt motor is its adaptability to adjustable speed service. Armature-resistance control or armature voltage control may be applied for speeds above the full-field speed. Field-rheostat control may be applied for speeds above the full-field speed, to further increase the possible speed range. A d-c motor may be reversed by reversing either the armature or the field but not both. The armature is normally reversed because the field's high inductance makes it hard to open and slow in response; however, higher currents in the armature require large reversing contactors.

(8) Since the mobile electric power plant has an a-c generator on it, a rectifier would be required to transform the a-c voltage to a workable d-c voltage for the motor. A diode rectifier is a device that can allow current flow in one direction with a given voltage polarity and inhibit current flow when the polarity is reversed. Figure 32 delineates the characteristics of a diode.

(a) When six diodes are combined correctly, a three-phase bridge rectifier is formed as depicted in Figure 33.

(b) Each diode conducts for a period of each a-c cycle. This results in 120 electrical degrees conduction in each rectifier and provides the maximum d-c power output that can be obtained from a three-phase a-c line with a three-phase rectifier. On the MEPP, the three-phase a-c voltage could be supplied by the 400 HZ synchronous generator. Control of the generator's field excitation could permit a variable a-c voltage output. With a fixed three-phase rectifier as shown, a 115V d-c motor could be controlled. The variable d-c output voltage would be applied to the motor's armature terminals. The a-c ripple on such a system would be relatively small due to the high frequency involved. If a larger speed range is required, control of the motor's field flux could be attained through a shunt-field resistor or a variable source of separate excitation. The two methods of speed control when used together could create a fairly simple system with a wide range in speed.

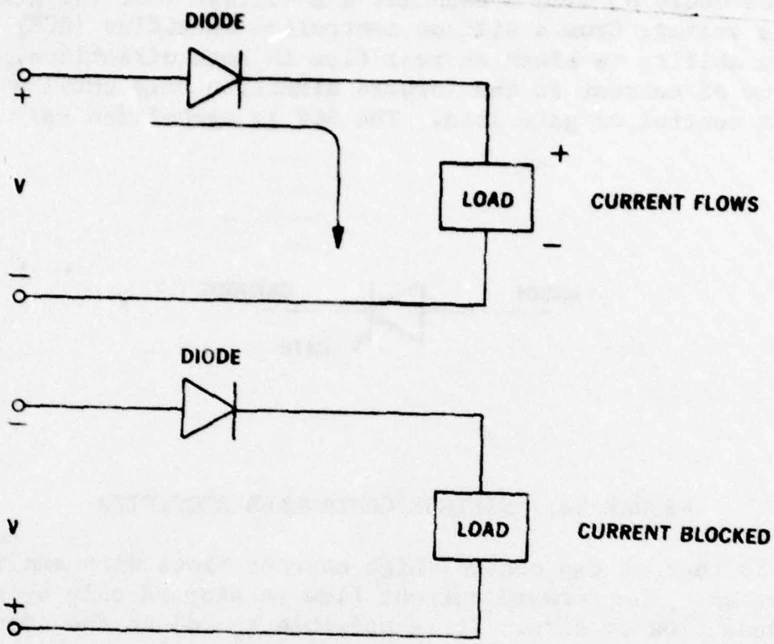

FIGURE 32. DIODE CHARACTERISTICS

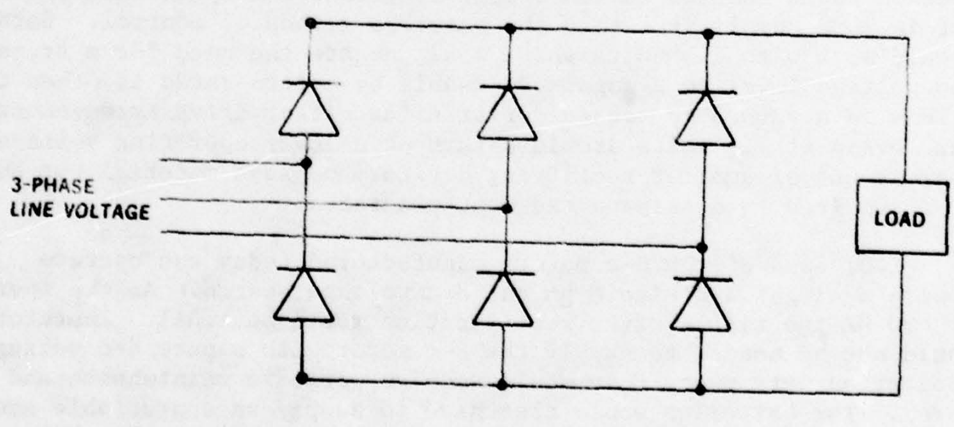

FIGURE 33. THREE-PHASE RECTIFIER

(9) Another possible way of controlling the d-c motor in our application would be with a constant a-c voltage from the generator and a variable d-c voltage from a silicon controlled rectifier (SCR) arrangement. An SCR has the ability to block current flow in both directions. This SCR blocks the flow of current in the forward direction only until a signal is applied to its control or gate lead. The SCR is symbolized as:

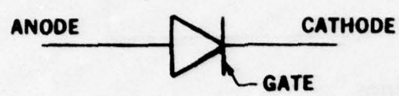

FIGURE 34. SILICON CONTROLLED RECTIFIER

It is useful in that it can control high current flows with small gate signals of short duration. The forward current flow is stopped only by reducing the anode-to-cathode flow to zero. It is possible to adjust the average value of direct current applied to a load. When these SCRs are used in a bridge rectifier arrangement, the output d-c voltage can be controlled by delaying the point in the a-c cycle where each rectifier conducts. This segment of an a-c cycle during which the rectifier passes current, measured in electrical degrees, is called the conduction angle. Controlled rectifiers would only have to be placed in three of the six legs of a three-phase bridge to obtain full range control of output voltage. This method of speed control requires control circuitry to ensure proper turn-on and turn-off of the SCRs. They must function in the correct sequence and also operate quickly enough for the applied frequency. Since this method would require an SCR bridge rectifier and applicable control circuitry, it is more complicated than the previous method of control. Both approaches could work with 115 volts which would negate the need for a transformer to change the voltage level to a magnitude usable by motors rated at other than 115 volts. This is a favorable characteristic for either drive arrangement. Also, current levels at 115 volts are less than at a lower operating voltage. This would permit use of smaller rectifying devices and less material for heat sinks would be required to dissipate the heat generated.

(10) Some of the d-c motors manufactured today can operate effectively with a slight a-c ripple on the d-c voltage source. At the operating frequency of 400 HZ the ripple after rectification would be small. Therefore, batteries would not be needed to supply the d-c motor with a pure d-c voltage level. If batteries were used, they would require periodic maintenance and a charging system. The batteries would also have to supply an appreciable amount of current to the motor, particularly while starting. The size and number of batteries could be prohibitive. Since they are not necessary for the operation of the d-c motor, they will receive no further consideration.

d. Reliability. The d-c motor, unlike a squirrel-cage induction motor, requires brushes and a commutator to function. These items will wear out, at a rate dependent upon the duty cycle of the motor and the current levels encountered. If used intermittently, it is a reliable motor. The rectifier bridge arrangement is very reliable if operated within acceptable current and voltage levels.

e. Maintainability. Aside from the periodic maintenance required on the brushes and commutator, there are shaft bearings which may need servicing on the motor. The three-phase bridge circuitry should require no attention when operated within rated conditions.

f. Conclusions. A drive system utilizing a d-c motor as the main component could propel the MEPP satisfactorily. High starting torque could be attained as well as a wide range of speed. It will be included in the matrix decision process later in this report for further evaluation.

5. AIR MOTOR DRIVE (Reference (a))

a. General. Interest in air motors for industrial applications has been renewed due to the presence of air motors with integral gearing. Inherently air motors were high-speed, low-torque devices. Integral gear reduction now enables an air motor to operate at lower speeds and higher torques. For a given hp, an air motor is much lighter than a comparable electric motor. An air motor requires a muffler to keep the noise within acceptable limits.

b. Description. Typically air motors are classified into three basic categories: rotary-vane, axial-piston, and radial-piston. There are two other types of motors which are used primarily in the medical or dental field rather than in the industrial area. Turbine motors, which operate at extremely high speeds and low torques are used primarily in dental drills. Diaphragm motors, employed in the medical field, exhibit characteristics between those of a rotary-vane and a geared rotary-vane motor.

(1) Rotary-Vane Motors.

(a) These motors use vanes which slide radially in rotor slots. Normally, four vanes are utilized; however, more vanes are used in special applications. They are available in sizes up to 10 hp with free speeds of 3,000 to 15,000 rpm, and in lubricated and unlubricated models as well as governed or ungoverned models.

(b) Geared vane motors are rotary-vane motors with a worm-gear reduction mechanism internal to the motor. Speed ratios range from 5:1 to 60:1, with the higher ratios usually available only in the lower power ratings. Maximum speeds are from 150 to 300 rpm and maximum power ratings are usually 1.5 to 5 hp.

(2) Radial-Piston Motors. These air motors employ four to six pistons positioned radially around a central drive shaft. Maximum power

Ref: (a) Brooks, E. "Air Motors Challenge Electrics", Machine Design, pp. 130-133, 21 September 1978.

is developed at low speeds usually around 1,000 rpm. These motors are nearly vibrationless due to overlapping power strokes and accurate balancing. Radial-piston air motors require lubrication during operation. Free speed is usually 3,000 rpm or less.

c. Performance Characteristics. If an adequate air supply is available, a properly geared rotary-vane motor may be applied where an electric motor has normally been used in the low hp range. Speed control is attained through a flow-control valve, and reversibility is obtained with a four-way valve in a few degrees of rotation. Rated speed is the point at which maximum power is reached, generally about 50% of no-load speed. Rated power and torque are usually the maximum values produced at 90-psi inlet pressure. Motor efficiency drops about 1.23% for every 1% drop in inlet pressure. The following operating curves better describe the general characteristics of air motors:

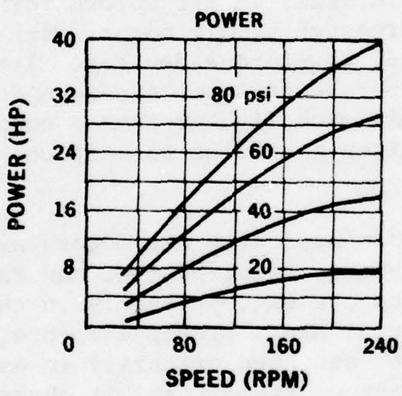

FIGURE 35. POWER - SPEED CURVE FOR A TYPICAL AIR MOTOR

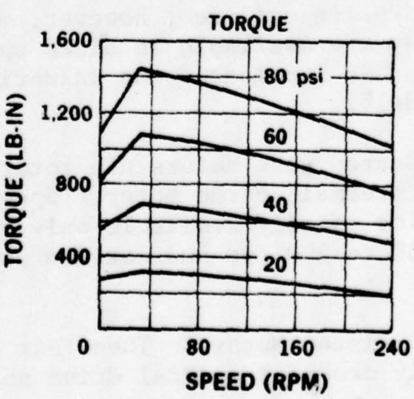

FIGURE 36. TORQUE - SPEED CURVE FOR A TYPICAL AIR MOTOR

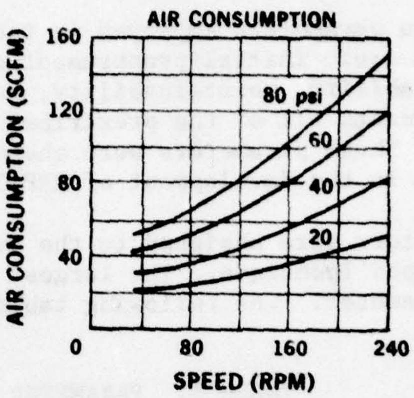

FIGURE 37. AIR CONSUMPTION - SPEED CURVE FOR A TYPICAL AIR MOTOR

d. Reliability. Air motors are reliable machines. They may operate in environments with temperatures up to 250 degrees F. Overloading and stalling doesn't damage the motors. The motors slow down with increasing load until pressure is increased or the load decreased. These motors are inherently explosion proof because they require no electricity for operation and no sparks can be generated. Therefore, no special explosion-proof enclosure is required as in electric motors. Inlet air inherently has moisture in it and this could possibly cause condensation problems.

e. Maintainability. The motor may require lubrication to achieve maximum bearing life. Aside from the possible replacement of bearings and seals, little maintenance is required.

f. Conclusions. The air motor will not be included in our matrix decision process. It is presently not available in power ranges of the magnitude we require. Also, a supply of air is necessary for operation, which is not available on the MEPP.

IV. EVALUATION OF SYSTEMS

The evaluation parameters employed in the matrix to evaluate the various propulsion systems are: initial procurement cost (based on 1,000 units), size, weight, reliability, maintainability, and the risk of developing a drive system that meets all of the prescribed requirements in the allotted development time. These parameters were chosen by mutual agreement of a committee involved in the development of MEP 354.

Weighting factors were assigned to the evaluation parameters using the aforementioned Delphi Technique. The largest weighting factor relates to the most important parameter. The following table indicates the assigned mean weighting factors.

TABLE 1. PARAMETER WEIGHTING FACTOR

Parameter	Mean Weighting Factor
Cost	.274
Size	.071
Weight	.077
Reliability	.425
Maintainability	.152

Scores for each system were obtained as explained below. For a given parameter, determine which system exhibits the best characteristics. Set the score for this system to the appropriate parameter weighting factor and give it a base rate of 1. The base rate for the other systems equals the ratio of their system value to the best system's value. The score for each system is then obtained by multiplying each individual base rate by the main parameter weighting factor. The scores for all the parameters are added for each system to obtain a subtotal.

The risk derating factor, obtained through the Delphi Technique, is multiplied by these subtotals to obtain the individual grand totals. The largest risk corresponds to the largest derating factor, and the lowest grand total corresponds to the most advantageous system.

The following example better clarifies this matrix evaluation process.

	System 1	System 2
Cost	\$100	\$200

System 1 has the lower cost, so it is the best system. It is assigned a score equal to the weighting factor for cost of .274 and a base rate of 1. The base rate for system 2 is computed as follows.

$$\text{System 2} = \frac{\$200}{\$100} = 2$$

The score for system 2 equals the base rate x parameter weighting value or 2 x .274 = .548.

Information on cost, size, and weight was obtained from the system manufacturers. Reliability and maintainability data was obtained mainly from a government industry data exchange program, reference (b). The following table relates each system to its predicted mean time between failure (MTBF) and mean time to repair (MTTR).

TABLE 2. SYSTEM PREDICTED MTBF AND MTTR

Drive System	MTBF (Hours)	MTTR (Hours)
Hydrostatic		
Differential	2,710	1.04
Wheel Motors	2,445	1.13
Hydrodynamic		
Torque Converter	1,111	1.68
Automatic Transmission	5,000	4.0
A-C Motor		
Microcomputer Control	6,753	1.68
Rectifier-Inverter Control	1,229	1.47
D-C Motor		
Fixed Rectifier Control	1,947	2.56
Variable Rectifier Control	881	2.29

The completed matrix follows in Figure 38.

Ref: (b) "Summaries of Failure Rate Data", Government Industry Data Exchange Program, Volume 1, August 1975.

EVALUATION PARAMETER	WEIGHTING FACTOR	HYDROSTATIC WITH DIFFERENTIAL		HYDROSTATIC WITH WHEEL MOTORS		HYDRODYNAMIC (TORQUE CONVERTER)		HYDRODYNAMIC (AUTOMATIC TRANSMISSION)		A-C MOTOR WITH MICRO COMPUTER CONTROL		A-C MOTOR WITH RECTIFIER INVERTER CONTROL		D-C MOTOR WITH FIXED RECTIFIER CONTROL		D-C MOTOR WITH VARIABLE RECTIFIER CONTROL	
		BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE	BASE RATE	SCORE
COST	.274	2.36	.647	2.31	.633	2.61	.715	2.07	.567	1	.274	2.49	.682	4.38	1.200	4.73	1.296
SIZE	.072	2.52	.181	2.31	.166	1.83	.132	1.76	.127	1	.072	2.93	.211	2.52	.181	4.69	.338
WEIGHT	.077	2.07	.159	2.20	.169	2.84	.219	2.33	.179	1	.077	1.86	.143	1.98	.152	2.02	.156
RELIABILITY	.425	2.49	1.058	2.76	1.173	6.08	2.584	1.35	.574	1	.425	5.49	2.333	3.47	1.475	7.67	3.260
MAINTAINABILITY	.152	1	.152	1.09	.166	1.62	.246	3.85	.585	1.62	.246	1.41	.214	2.46	.374	2.20	.334
SUBTOTAL	1.000	2.197		2.307		3.896		2.032		1.094		3.583		3.382		5.384	
RISK (DERATING FACTOR)	1 YR	.10		.13		.90		.85		.70		.75		.32		.55	
	2 YR	.07		.10		.85		.80		.55		.60		.20		.40	
GRAND TOTAL	1 YR	.220		.300		3.506		[illegible]		.766		2.687		1.082		2.961	
	2 YR	.154		.231		3.312		1.626		.602		2.150		.676		2.154	

FIGURE 38. COMPLETED EVALUATION MATRIX - MEP 354

V. DISCUSSION OF RESULTS

As explained earlier a matrix evaluation process has been implemented to indicate the relative merits of several possible drive systems. Several parameters deemed important constitute the backbone of the matrix. These parameters are initial procurement cost (based on 1,000 units), size, weight, reliability, and maintainability. It has been determined that a derating factor is necessary in the matrix. This factor relates to the probability of developing a drive system that meets all of the prescribed requirements in the allotted development time. The numbers in the matrix reflect the risk for development times of one and two years. The lowest number in the grand total section of the matrix relates to the most advantageous drive system overall.

From the resulting matrix values, it is obvious that the hydrostatic system utilizing a rear axle with an integral differential is the most desirable system overall. It can provide infinite control of speed within a given range in both forward and reverse. Starting, stopping, changing direction, and accelerating are all performed in smooth movements without unnecessary cycling or roughness. Maximum torque is available at starting which is desirable for the MEPP. The system is very flexible. The motor can be mounted remotely from the pump but connected through two hydraulic hoses. The numbers in the evaluation matrix indicate it has favorable characteristics. Also, it is presently commercially available and is a proven entity. This is graphically displayed in the favorable risk factors. Figure B-1 illustrates this system's layout.

The hydrostatic system employing wheel motors exhibits characteristics very similar to the previously described hydrostatic system. It is dissimilar from the former system in that a hydrostatic wheel motor would be placed on each driving wheel rather than one motor driving a differential and rear axle as shown in Figures B-2 and B-3. Low-speed, high-torque wheel motors aren't as readily available as a rear axle and this is reflected in the slightly higher risk factor. Nevertheless, these systems are very close as far as overall desirability is concerned. With either system it might be possible to shrink the wheelbase slightly, which is advantageous to the turning radius. More detailed layouts of these systems are under way.

The hydrodynamic (torque converter) system has many drawbacks. A clutch is necessary to engage and disengage the torque converter from the generator to inhibit motion while in the generating mode. It is undesirable to use a clutch-type transmission in MEP 354. Reliability and maintainability suffer severely when a clutch is considered. A reversing gear mechanism is mandatory to enable reverse motion. When all the components are linked together, a long drive train results. The combination of clutch, torque converter, reversing transmission, drive shaft, and rear axle extend the vehicle's wheelbase beyond reasonable limits as shown in Figure B-4. Since this unit will be used on carrier decks, turning radius is an important parameter to keep in mind, and the wheelbase must be kept as short as possible to achieve a satisfactory turning radius.

Any extension of the wheelbase adversely affects the turning radius with a given steering arrangement. Figure B-12 better illustrates the relationship between wheelbase, turning radius, and steering arrangements. If the wheelbase is extended, it is necessary to turn the wheels at a sharper angle to maintain the desired turning radius. There is a limitation on the maximum attainable wheel turning angle. Steering experts like to keep the inside wheel's turning angle at 40° to 45°. Extending the wheelbase would necessitate achieving a very large angle which is impractical and would require changes in the steering mechanism and linkages. Also, increased wheel angles are limited by the wheel well, engine, etc., which would interfere with the wheel as depicted in Figure B-13. Therefore, any extension of wheelbase as in the torque converter scheme is undesirable.

The automatic transmission arrangement could meet the power and torque requirements nicely; however, it would be necessary to extend the wheelbase to utilize it. The transmission could be placed behind the generator, directly behind the diesel, or on a power takeoff from the diesel in parallel with the generator. Figures B-5 through B-7 show how in all cases the wheelbase would have to be extended to implement any of these alternatives.

Of notable interest is the a-c motor system controlled by a microprocessor. This system displayed very favorable characteristics as exemplified in the completed matrix. The a-c motor control scheme is very reliable and would require almost no maintenance. A microcomputer is presently being used in a parallel program at the Naval Air Engineering Center to control the MEP 354 generator's voltage and frequency characteristics. It is very possible that it could be used to vary the generator's frequency and voltage which would control the a-c induction motor's speed and torque. Such a system is currently not developed commercially and would require substantial development time to be fabricated in-house. This is reflected in its high derating factor. Figure B-8 displays the layout for this system.

The a-c motor with rectifier inverter control is presently available commercially. However, its cost, size, and high risk preclude its use on this mobile electric power plant. Figure B-9 illustrates this system's possible layout.

Both d-c motor systems, fixed and variable rectifier, exhibited high procurement costs and relatively poor reliability. This is largely due to the motor itself. It requires periodic maintenance on the commutator and brushes which is reflected in the high reliability and maintainability figures. Figures B-10 and B-11 display the layouts for these two drive systems.

VI. CONCLUSIONS AND RECOMMENDATIONS

The results of this drive system study indicate that a hydrostatic system would be the most advantageous system to be incorporated in MEP 354. An analysis is under way to lay out the two types of hydrostatic drives, using a differential and using wheel motors, to see which best conforms to the given frame envelope. Wheelbase and its effect on turning radius are the major items of concern in this analysis. Pending completion of the layout analysis and when all vendor proposals are received, a unique hydrostatic system will be recommended for inclusion in MEP 354.

As noted earlier the a-c motor with microcomputer control was the leading drive system candidate prior to the risk factor. It is a strategy that would endeavor to marry the most reliable electric motor to a very reliable and efficient control scheme. It is recommended that further consideration should be given to this control strategy. A study of ground support equipment would probably reveal several areas where such a system could be incorporated. An example would be a generator test stand which currently utilizes a d-c motor to drive a generator. It could prove very beneficial to retrofit this test stand with a much more reliable a-c induction motor. New designs of generator test stands or similar equipment could possibly benefit by using such an a-c motor control scheme.

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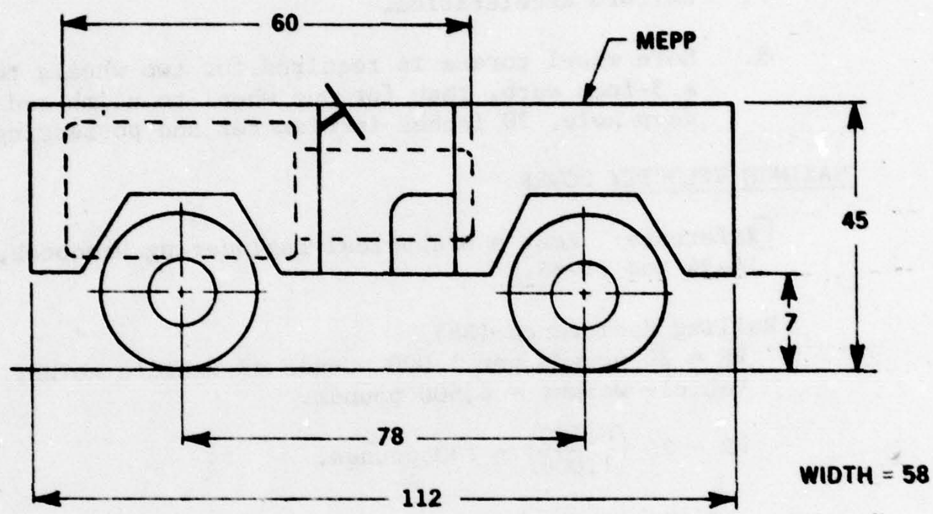
APPENDICES

APPENDIX A - MOBILITY REQUIREMENT ANALYSIS

PROPULSION UNIT

MOBILE ELECTRIC POWER PLANT 354



NOT TO SCALE

PERFORMANCE REQUIREMENTS

1. Maximum velocity - 15 mph on straight and level concrete road.

2. Maximum acceleration - 100 feet in 9.1 seconds from a standing start.

3. Capable of forward and reverse direction of travel.

4. Climb and pass over, from a standing start:

   a. 3-inch curb against front wheels.

   b. 3-inch curb against rear wheels.

   c. 3-inch-deep, 20-inch-diameter hole possessing straight sides; condition--any one wheel in the hole.

ASSUMPTIONS

1. Total vehicle gross weight - 6,500 pounds.

2. Pneumatic tires with diameter of 25.9 inches and a rolling radius of 11.4 inches.

3. Weight distribution - 50 ±10% on rear and front wheels.

4. Rolling resistance - 20 pounds per 1,000 pounds of vehicle weight.

5. Air resistance coefficient - 0.002.

6. Coefficient of friction on a wet, moderately worn deck surface - 1.0.

7. Uniform acceleration.

8. More wheel torque is required for two wheels to climb and pass over a 3-inch curb, than for one wheel to climb and pass over a 3-inch deep hole, 20 inches in diameter and possessing straight sides.

MAXIMUM VELOCITY POWER

[Reference: Kent's Mechanical Engineering Handbook, 11th Edition, Pages 14-54 and 14-55.]

Rolling Resistance (RR)

RR = 20 pounds per 1,000 pounds of vehicle weight.
Vehicle weight = 6,500 pounds.

$$RR = 20 \left(\frac{6,500}{1,000}\right) = 130 \text{ pounds.}$$

Air Resistance (AR)

$$AR = KAV^2$$

K = Body shape coefficient = 0.002.
A = Body frontal area (square feet).
V = Velocity (mph) = 15 mph.

$$AR = 0.002 \left[\frac{(45-7)\ (58)}{144}\right](15)^2 = 7 \text{ pounds.}$$

Total Resistance (R_T)

$$R_T = RR + AR$$
$$= 130 + 7 = 137 \text{ pounds.}$$

Resistance to grade not included.

Power Required

$$HP = R_T\left(\frac{V}{375}\right)$$
$$= 137\left(\frac{15}{375}\right) = 5.5 \text{ HP.}$$

MAXIMUM ACCELERATION POWER

$S = \frac{V_t^2 - V_o}{2a}$ for uniform acceleration.

S = Distance (feet) = 100 feet.

V_t = Terminal velocity (feet/sec) = 15 mph (1.467) = 22.0 feet/sec.

V_o = Initial velocity (feet/sec) = 0 feet/sec.

a = Acceleration (feet/sec^2).

$a = \frac{V_t^2}{2S} = \frac{(22.0)^2}{2(100)} = 2.42$ feet/sec^2.

$t = \frac{V}{a}$.

t = Time (seconds).

$t = \frac{V}{a} = \frac{22.0}{2.42} = 9.1$ seconds.

$F = Ma$.

F = Force (pounds).

M = Mass (slug) = $\frac{\text{Weight}}{32.2} = \frac{6,500}{32.2}$.

$F = Ma = \frac{6,500}{32.2}(2.42) = 488.5$ pounds.

Power required for acceleration:

$HP = \frac{FV60}{33,000} = \frac{488.5(22.0)(60)}{33,000} = 19.5$ HP.

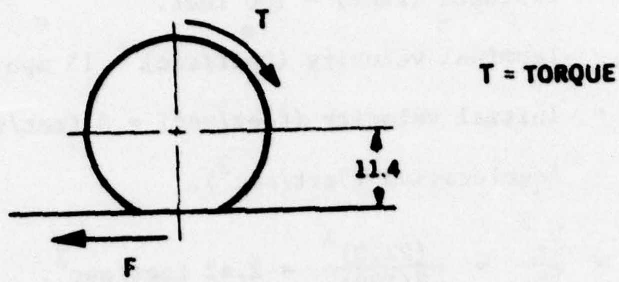
TOTAL POWER AT PEAK ACCELERATION

This is the power required for acceleration plus the power required to overcome rolling resistance and air resistance at 15 mph.

Total power = acceleration power + velocity power.

Total power = 19.5 + 5.5 = 25 HP.

MAXIMUM ACCELERATION TORQUE



F_T = Force required for acceleration plus air resistance and rolling resistance.

$F_T = F + R_T = 488.5 + 137 = 625.5$ pounds.

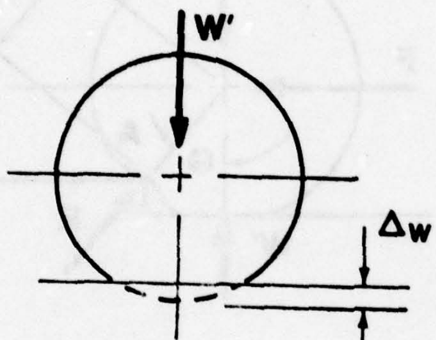
$T = 11.4F_T = 11.4\ (625.5) = 7130$ lb in.

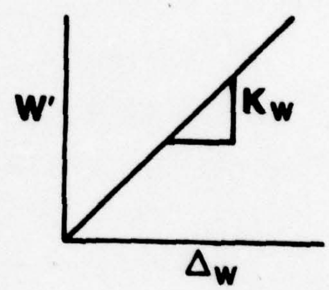
or 594 lb ft.

At 221 rpm (see wheel speed calculations).

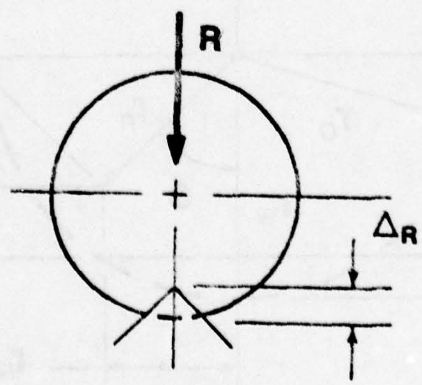
CURB CLIMB POWER

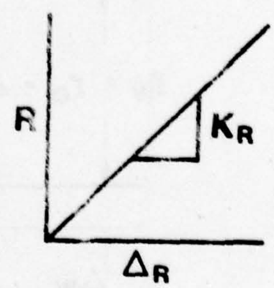
This is a statically indeterminate problem, which can be solved if relationship between tire load and tire deformation is known. This relationship is illustrated as follows:





W' = Load on road surface.
Δ_w = Deflection due to load W'.
K_w = Spring constant for tire.
$W' = K_w \Delta_w$.

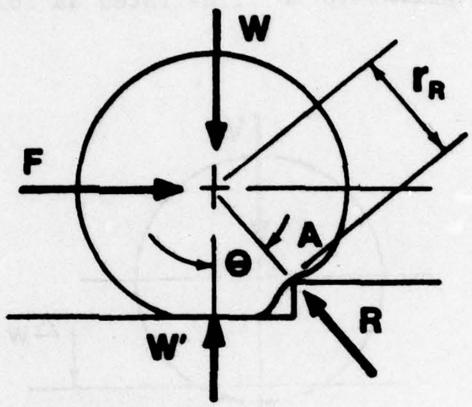




R = Radial load on curb.
Δ_R = Deflection due to load R.
K_R = Spring constant for tire.
$R = K_R \Delta_R$.

<u>Non-driving wheels against curb.</u>

Statics:

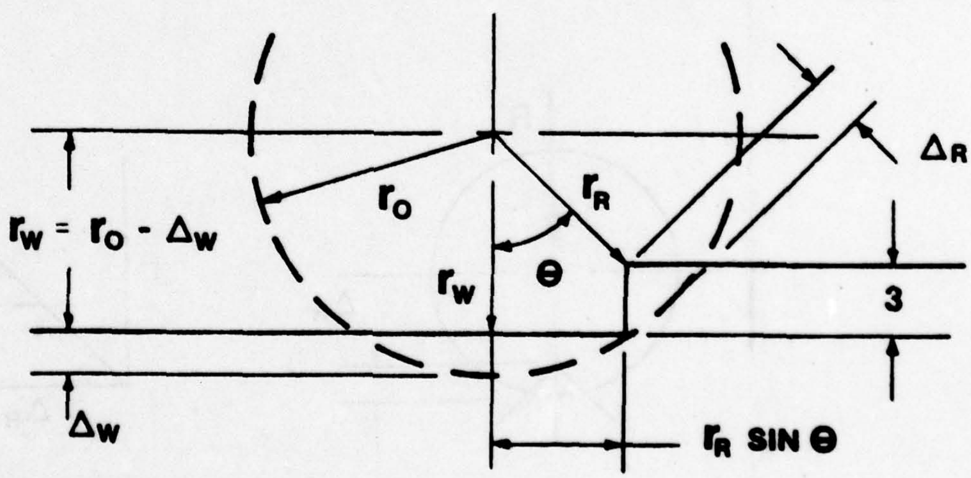


$$\Sigma F_V = W' + R\cos\theta - W = 0.$$

$$\Sigma F_H = F - R\sin\theta = 0.$$

$$\Sigma M_A = W\, r_R \sin\theta - F\, r_R \cos\theta - W'\, r_R \sin\theta = 0.$$

Geometry:



$$r_w = r_o - \Delta_w = r_R \cos\theta + 3.$$

$$r_o - \Delta_w = r_o \cos\theta - \Delta_R \cos\theta + 3.$$

$$\Delta_w = r_o (1 - \cos\theta) + \Delta_R \cos\theta - 3.$$

Wheel Position:

$\Delta_w = r_o\ (1-\cos\theta) + \Delta_R \cos\theta - 3.$

$r_o = 12.9$ inches.

When $\Delta_R = 0$ (wheels just touching curb).

$r_w = 11.4$ in (rolling radius).

$\Delta_w = r_o - r_w = 12.9 - 11.4 = 1.5$ in.

$1.5 = 12.9\ (1 - \cos\theta) + 0-3.$

$\cos\theta = .651.$

$\theta = 49.4°.$

When $\Delta_w = 0$ (wheels just lifting off ground)

Assume $R = W$ and $K_R = K_w$.

Then $r_R = r_w = 11.4$ in. and $\Delta_R = 1.5$ in.

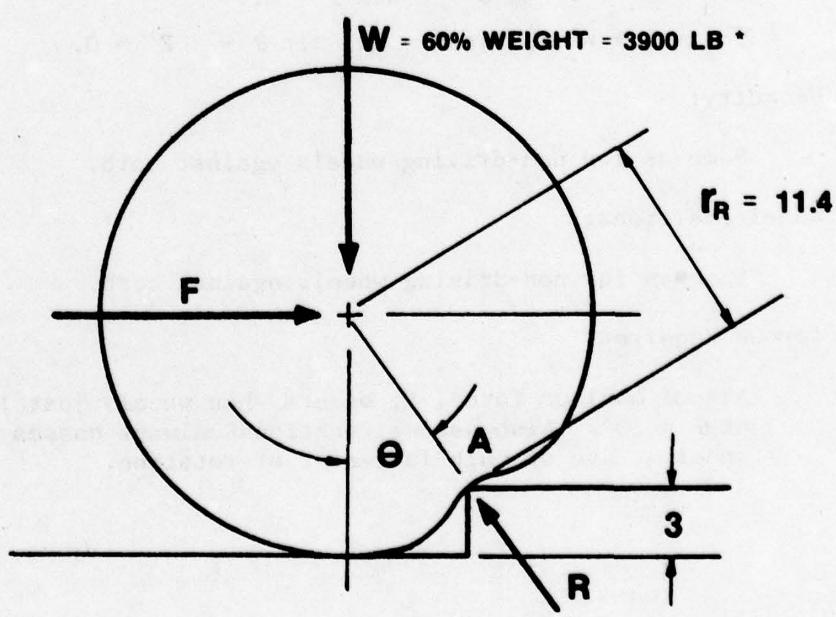
$0 = 12.9\ (1-\cos\theta) + 1.5 \cos\theta - 3$

$\mathrm{Cos}\ \theta = .868.$

$\theta = 29.7°.$

Torque Required:

Assume maximum force, F, occurs when wheels just lift off ground, at $\theta = 30°$. Also assume reaction R always passes through center of wheel.

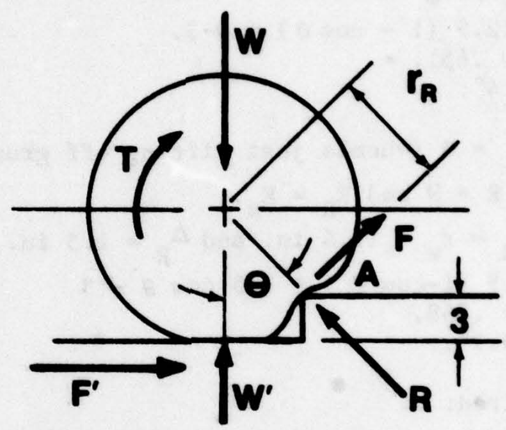


* Spec XAS 1359, Para 3.6.3.

$\Sigma M_A = Wr_R \sin \theta - Fr_R \cos \theta = 0.$
$3{,}900\ (11.4) \sin 30 = F\ (11.4) \cos 30.$
$F = 2{,}252$ lb.
$T = 2{,}252\ (11.4) = 25{,}673$ lb in.

Driving wheels against curb

Statics:



$$\Sigma F_V = W' + R \cos \theta + F \sin \theta - W = 0.$$
$$\Sigma F_H = F' + F \cos \theta - R \sin \theta = 0.$$
$$\Sigma M_A = T + W' r_R \sin \theta - Wr_R \sin \theta - 3F' = 0.$$

Geometry:

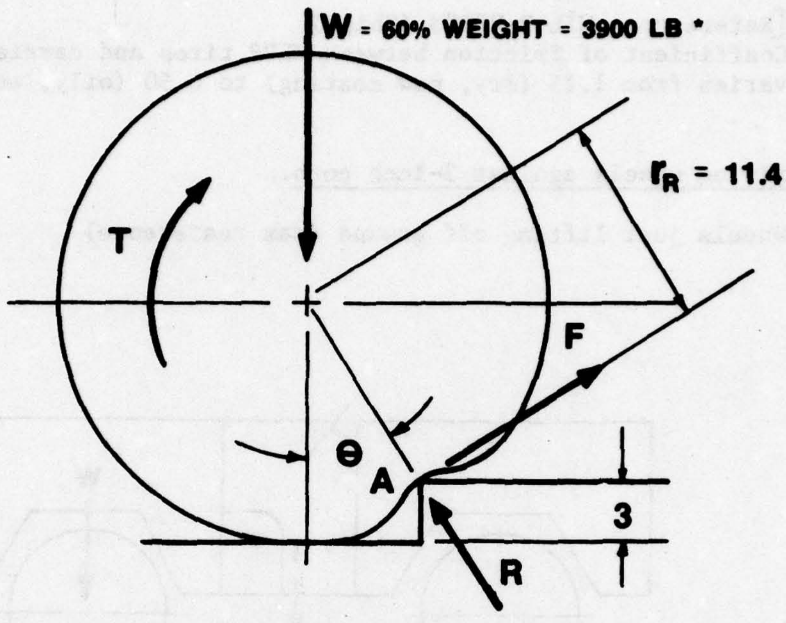
Same as for non-driving wheels against curb.

Wheel Positions:

Same as for non-driving wheels against curb.

Torque Required:

Assume maximum force, F, occurs when wheels just lift off ground, at $\theta = 30°$. Also assume reaction R always passes through center of wheel. Edge of curb is center of rotation.



* Spec XAS 1359, Para 3.6.3.

$$\Sigma M_A = T + W' r_R \sin\theta - W r_R \sin\theta - 3 F' = 0.$$

$$W' = 0,\ F' = 0.$$

$$T = 3{,}900\ (11.4) \sin 30° = 22{,}230 \text{ lb in.}$$

The maximum curb climb torque determined from the previous calculations is 25,673 lb in, and is required when the non-driving wheels are against the curb. This is the torque required to statically balance the forces. The actual torque required to push the wheels over the curb will be slightly higher than this value.

The calculations are based on 60% of the vehicle weight supported by the axle being raised over the curb. This creates the maximum torque, since force is proportional to weight.

The minimum torque required to climb the 3-inch curb is:

$$T' = 25{,}673 \left(\frac{.4}{.6}\right) = 17{,}115 \text{ lb in for the non-driving wheels against curb.}$$

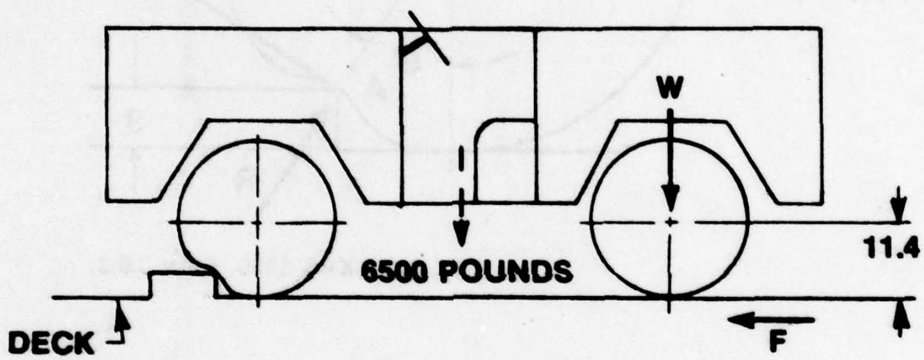
$$T' = 22{,}230 \left(\frac{.4}{.6}\right) = 14{,}820 \text{ lb in for the driving wheels against curb.}$$

TRACTION

[Reference: MIL-D-23003 (Ships)]
Coefficient of friction between MEPP tires and carrier deck coating varies from 1.15 (dry, new coating) to 0.50 (oily, worn).

Non-driving wheels against 3-inch curb.

Wheels just lifting off ground (max resistance)



W = Weight on driving axle
= 50$\pm$ 10% total weight
W = 6500 (.6) = 3,900 pounds
W' = 6500 (.4) = 2,600 pounds.
F = μ N
F = Frictional force developed.
μ = Coefficient of friction = 1.0
(wet, moderately worn surface).
N = normal force = W

Minimum weight on non-driving wheels and maximum weight on driving wheels:

F = 1.0 (3,900) = 3,900 pounds
T = 11.4 F = 11.4 (3,900) = 44,460 lb in.

Torque required = 17,115 lb in. (from curb climb power calculations).

Curb climb torque can be developed.

Maximum weight on non-driving wheels and minimum weight on driving wheels:

$F' = 1.0\ (2600) = 2{,}600$ pounds.

$T' = 11.4\ F' = 11.4\ (2{,}600) = 29{,}640$ lb in.

Torque required = 25,673 lb in. (from curb climb power calculations).

Curb climb can be developed.

Driving wheels against 3-inch curb.

Wheels just lifting off ground (max resistance).

Maximum weight on driving wheels.

$F = 1.0\ (3{,}900) = 3{,}900$ pounds.
$T = 11.4\ F = 11.4\ (3{,}900) = 44{,}460$ lb in.

Torque required: 22,230 lb in. (from curb climb power calculations).

Curb climb torque can be developed.

Minimum weight on driving wheels.

$F' = 1.0\ (2600) = 2{,}600$ pounds.

$T' = 11.4F' = 11.4\ (2{,}600) = 29{,}640$ lb in.

Torque required = 14,820 lb in. (from curb climb power calculations).

Curb climb torque can be developed.

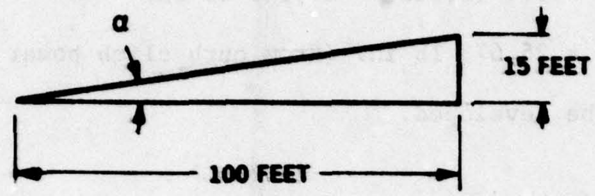
Note:

The coefficient of friction of rubber tires on concrete is less than rubber tires on the deck coating. Therefore, the torque that can be developed on a concrete surface is less than can be developed on a carrier deck, resulting in the possibility that the MEPP may have more difficulty negotiating a 3-inch curb on a concrete surface than on a carrier deck. However, it is always possible to increase the frictional force between the tires and the driving surface by placing additional external weight over the driving wheels.

ABILITY TO NEGOTIATE GRADE

[Reference: Mark's Handbook, Fourth Edition, Page 1451]

15% Grade



$$\text{Tan } \alpha = \frac{15}{100} = .15.$$

$$\alpha = 8.5°.$$

Grade Resistance (GR)

$$GR = \frac{nW}{100}.$$

n = Percent Grade = 15%
W = Weight = 6,500 pounds.

$$GR = \frac{15\ (6,500)}{100} = 975 \text{ pounds.}$$

Rolling Resistance (RR)

$$RR = 20\ \frac{W}{1000} = 20\left(\frac{6,500}{1,000}\right) = 130 \text{ pounds.}$$

Air Resistance (AR)

$$AR = KAV^2 = .002\left[\frac{(45-7)\ 58}{144}\right] V^2 = .031\ V^2.$$

Total Resistance (R_T).

$$R_T = GR + RR + AR.$$

$$R_T = 975 + 130 + .031\ V^2 = 1105 + .031\ V^2.$$

Theoretical Velocity

Consider 25 horse power available (from previous calculations).

$$HP = R_T\left(\frac{V}{375}\right).$$

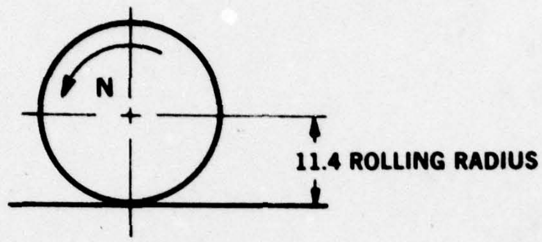
$$25 = (1105 + .031\ V^2)\ \frac{V}{375}\ .$$

$$25 = 2.947V + .0000827\ V^3.$$

$$V \approx 8.5 \text{ mph.}$$

There is sufficient power (25 HP) to propel the vehicle up a 15% grade at 8.5 mph, considering no acceleration.

WHEEL SPEED



Speed at maximum velocity of 15 mph.

$$N = \frac{12\ (88)V}{\pi\ D}$$

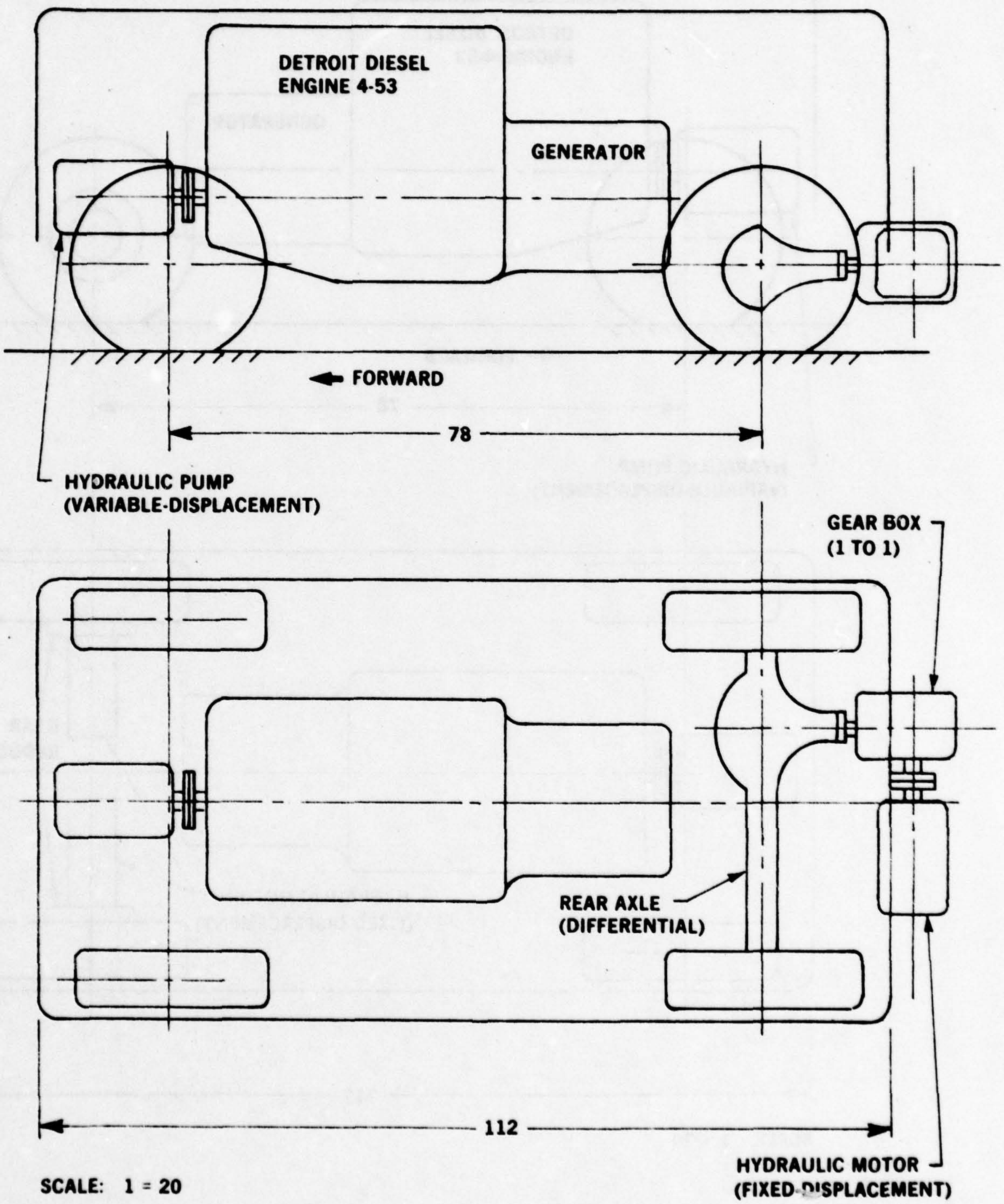
N = Wheel speed (rpm).
V = Velocity (mph) = 15 mph.
D = Rolling diameter = 11.4 (2) = 22.8 inches.

$$N = \frac{12\ (88)15}{\pi\ (22.8)} = 221 \text{ rpm.}$$

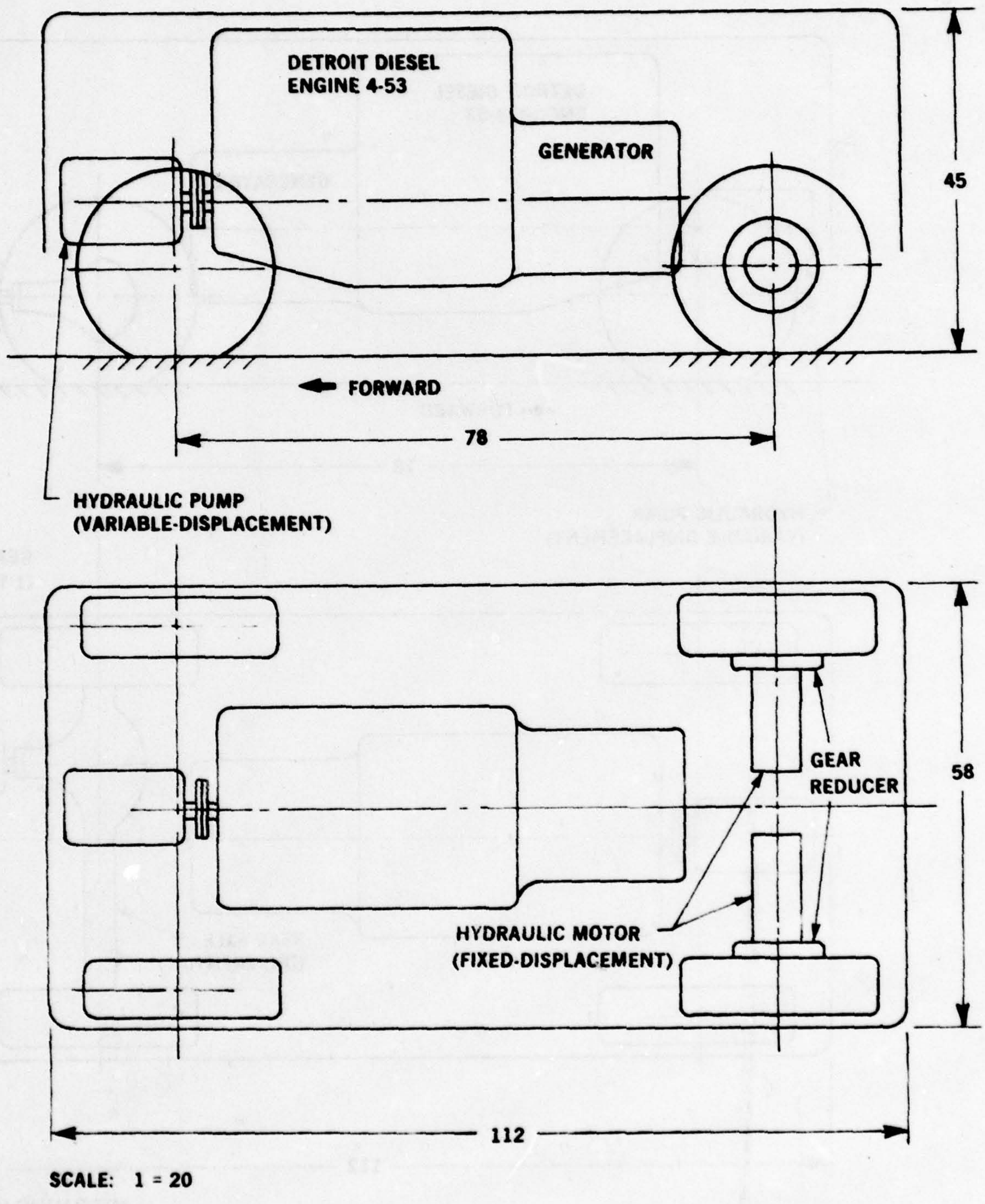
Speed during curb climb. No specific velocity required; therefore, speed must only exceed 0 mph.

$N > 0$ rpm

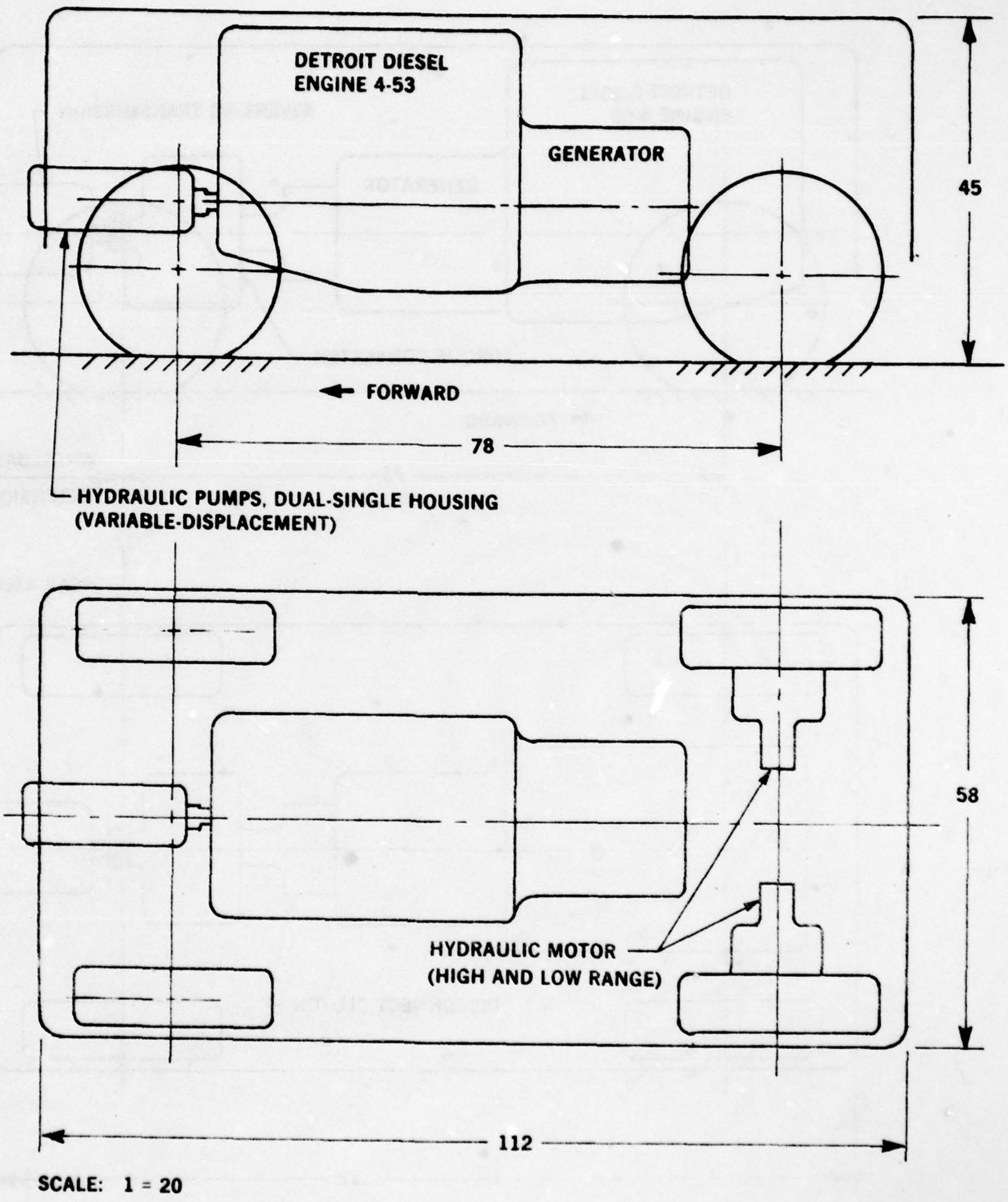
APPENDIX B - SYSTEM LAYOUTS



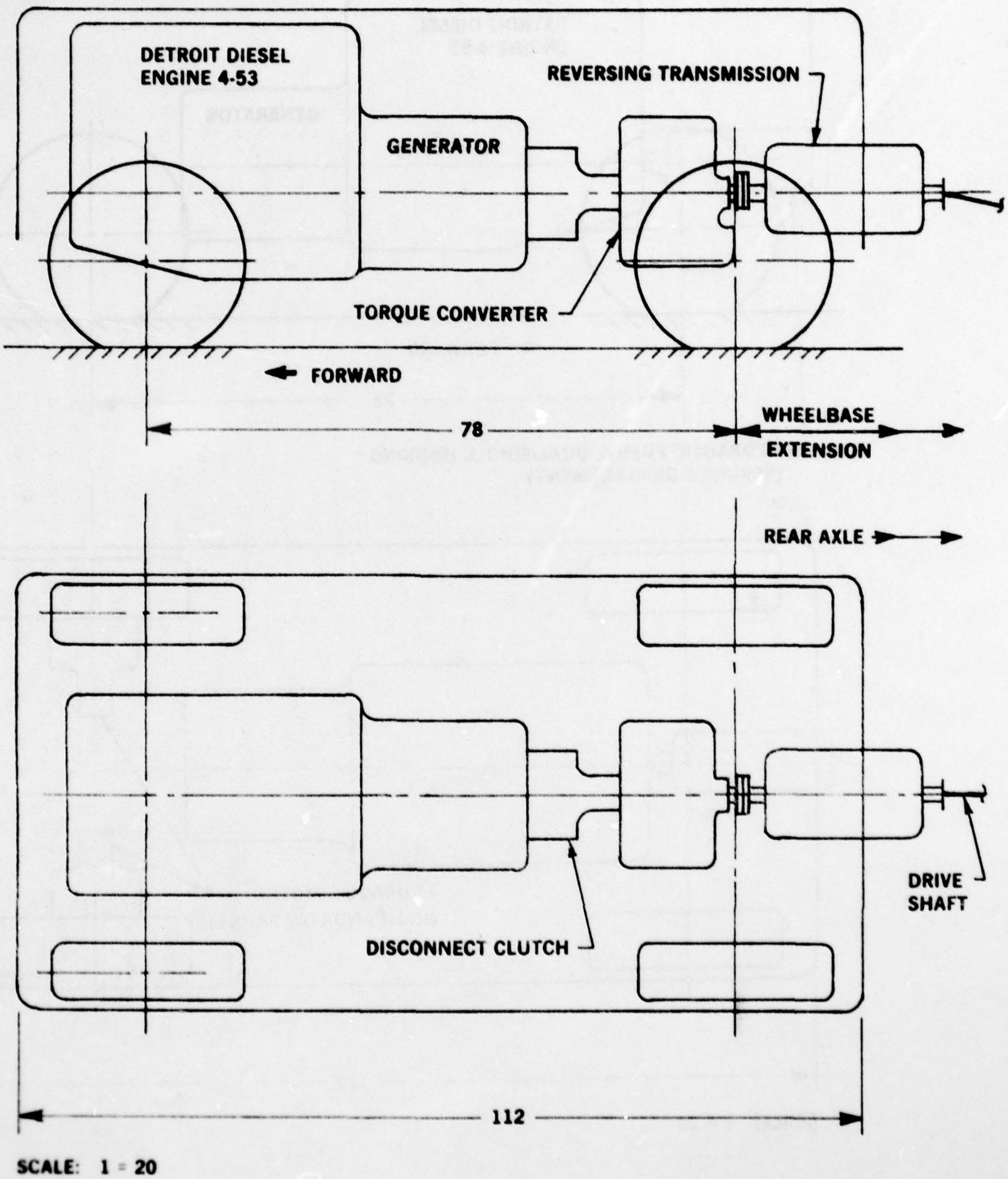
B-1. HYDROSTATIC WITH DIFFERENTIAL



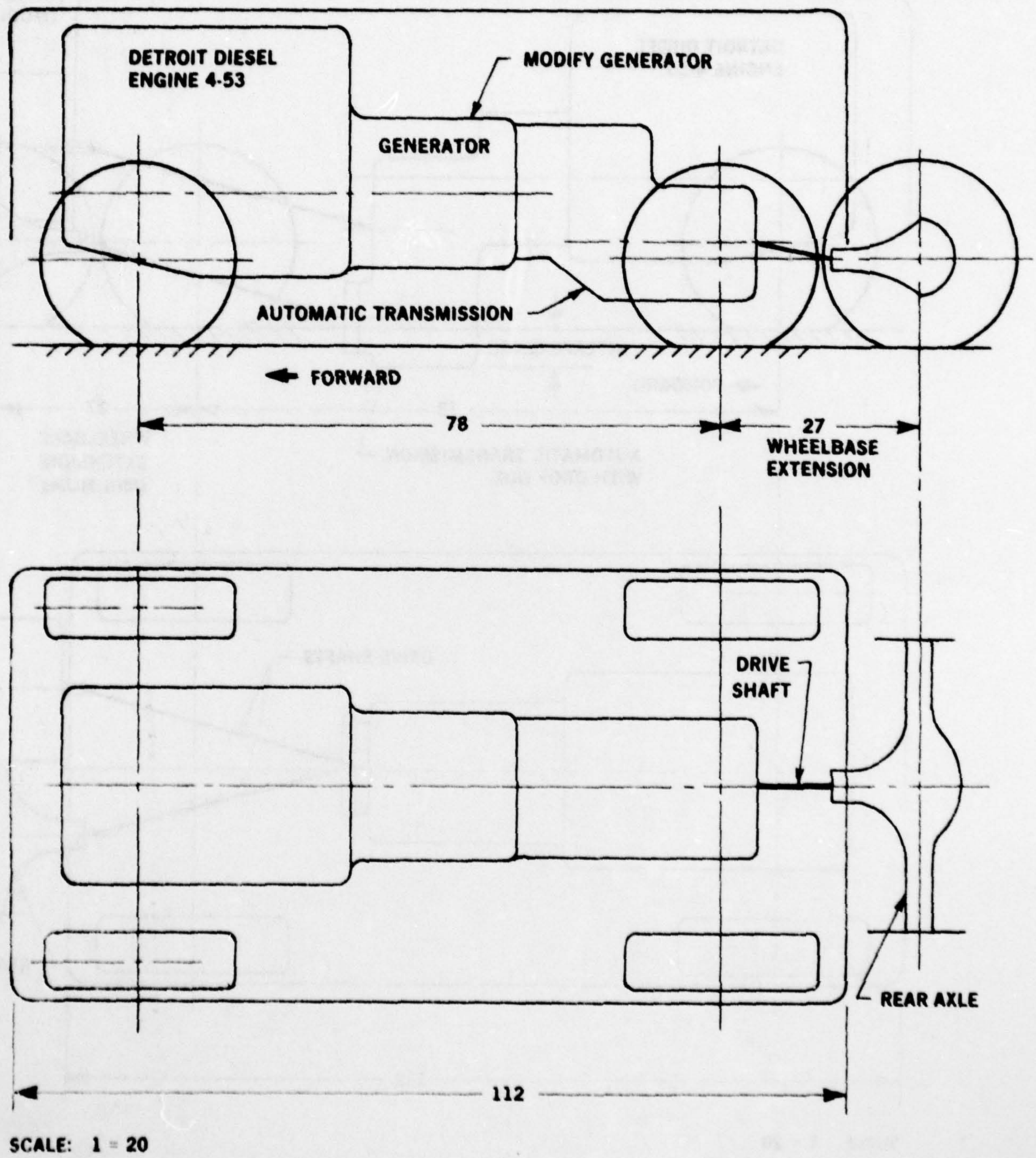
B-2. HYDROSTATIC WITH WHEEL MOTORS WITH GEAR REDUCERS



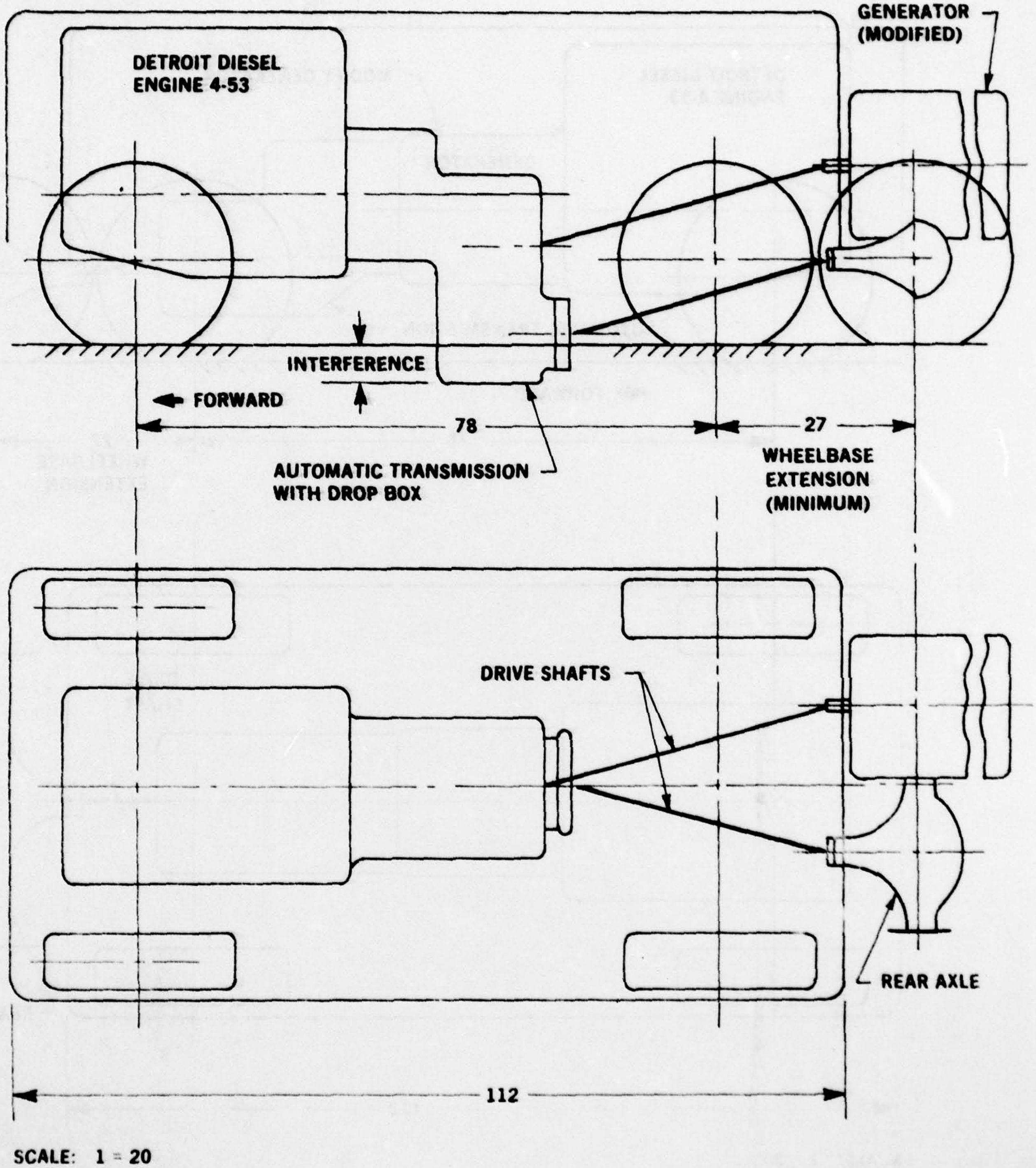
B-3. HYDROSTATIC WITH WHEEL MOTORS.



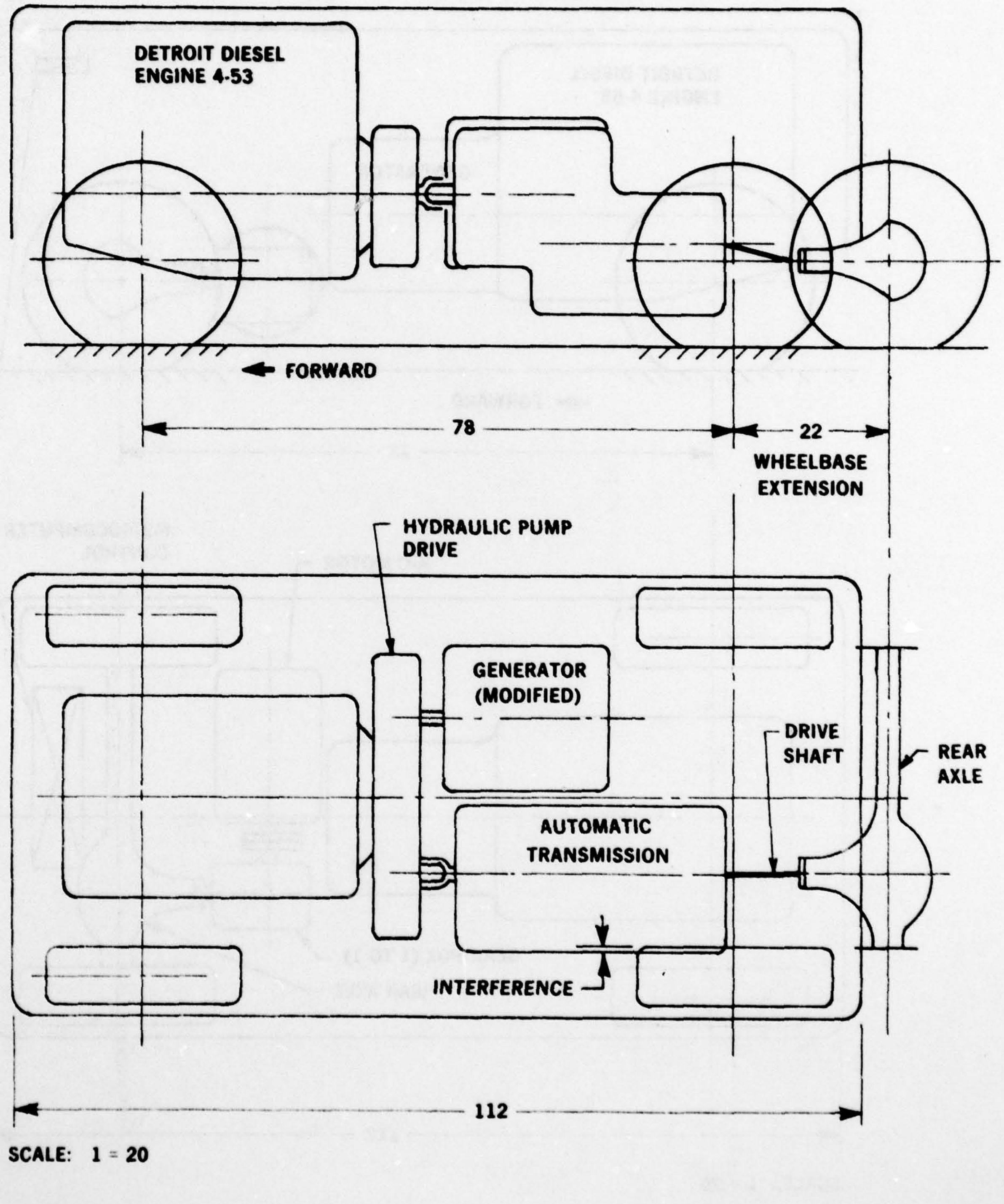
B-4. HYDRODYNAMIC (TORQUE CONVERTER)



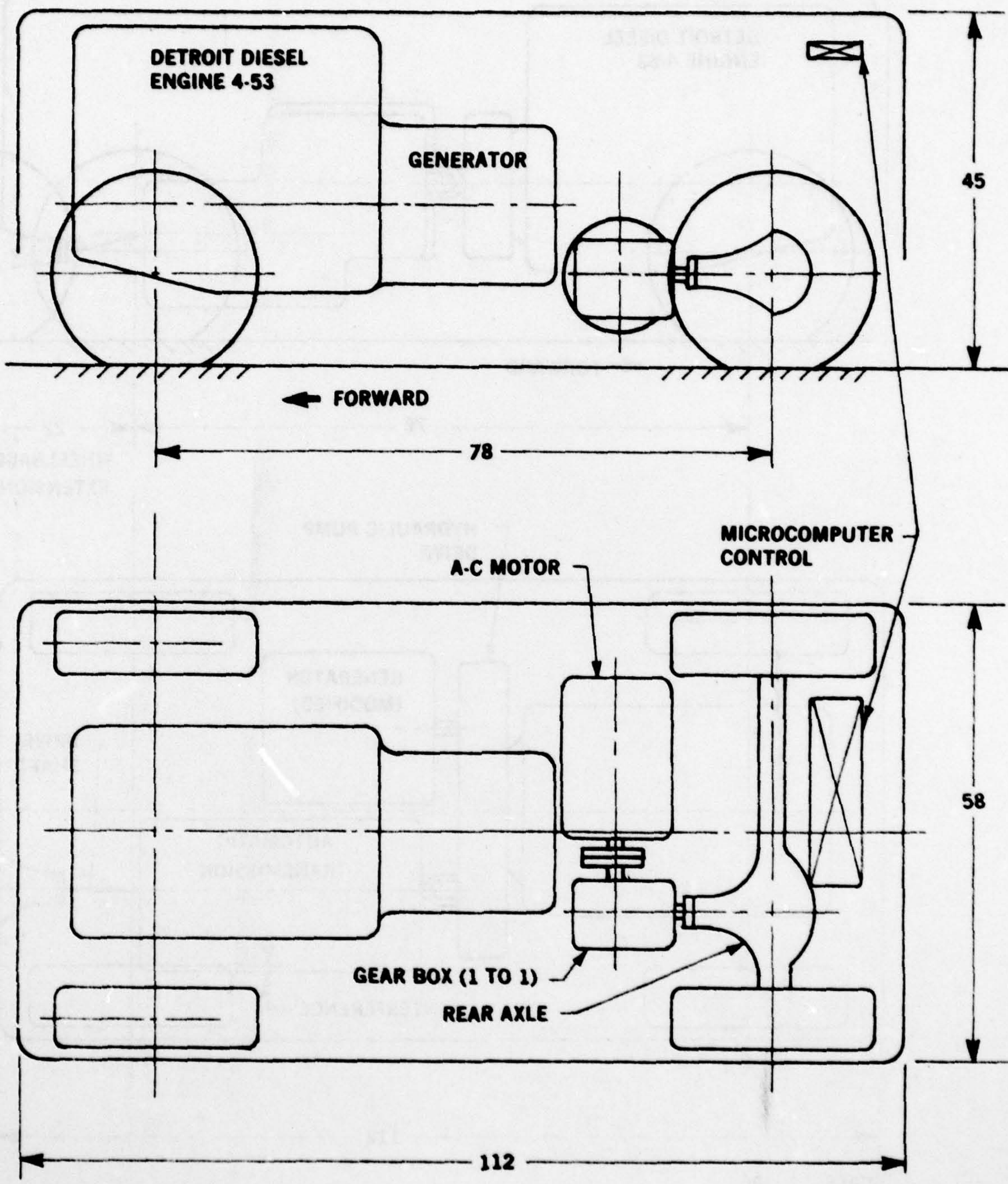
B-5. HYDRODYNAMIC (AUTOMATIC TRANSMISSION) IN-LINE ARRANGEMENT



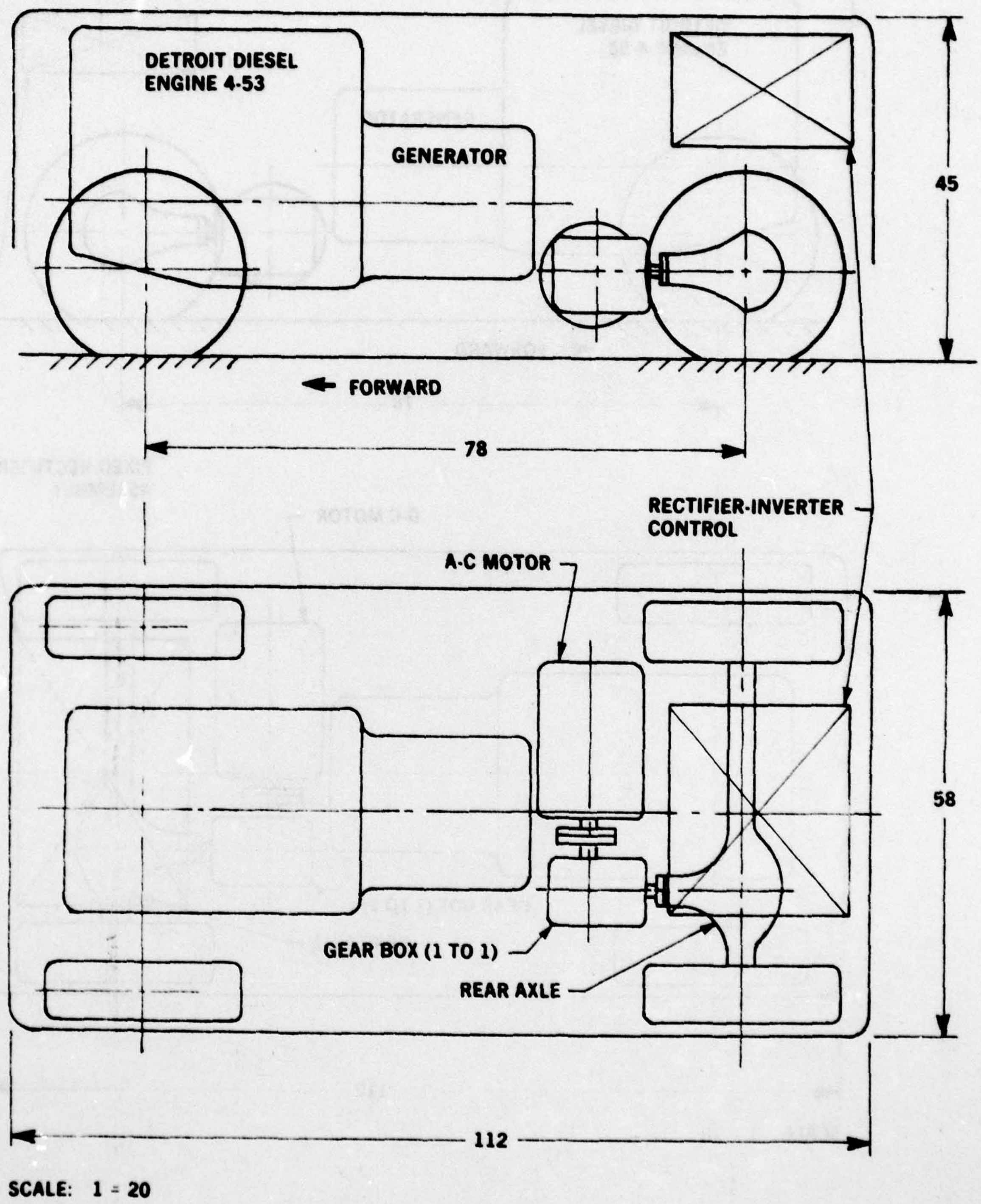
B-6. HYDRODYNAMIC (AUTOMATIC TRANSMISSION) WITH DROP BOX



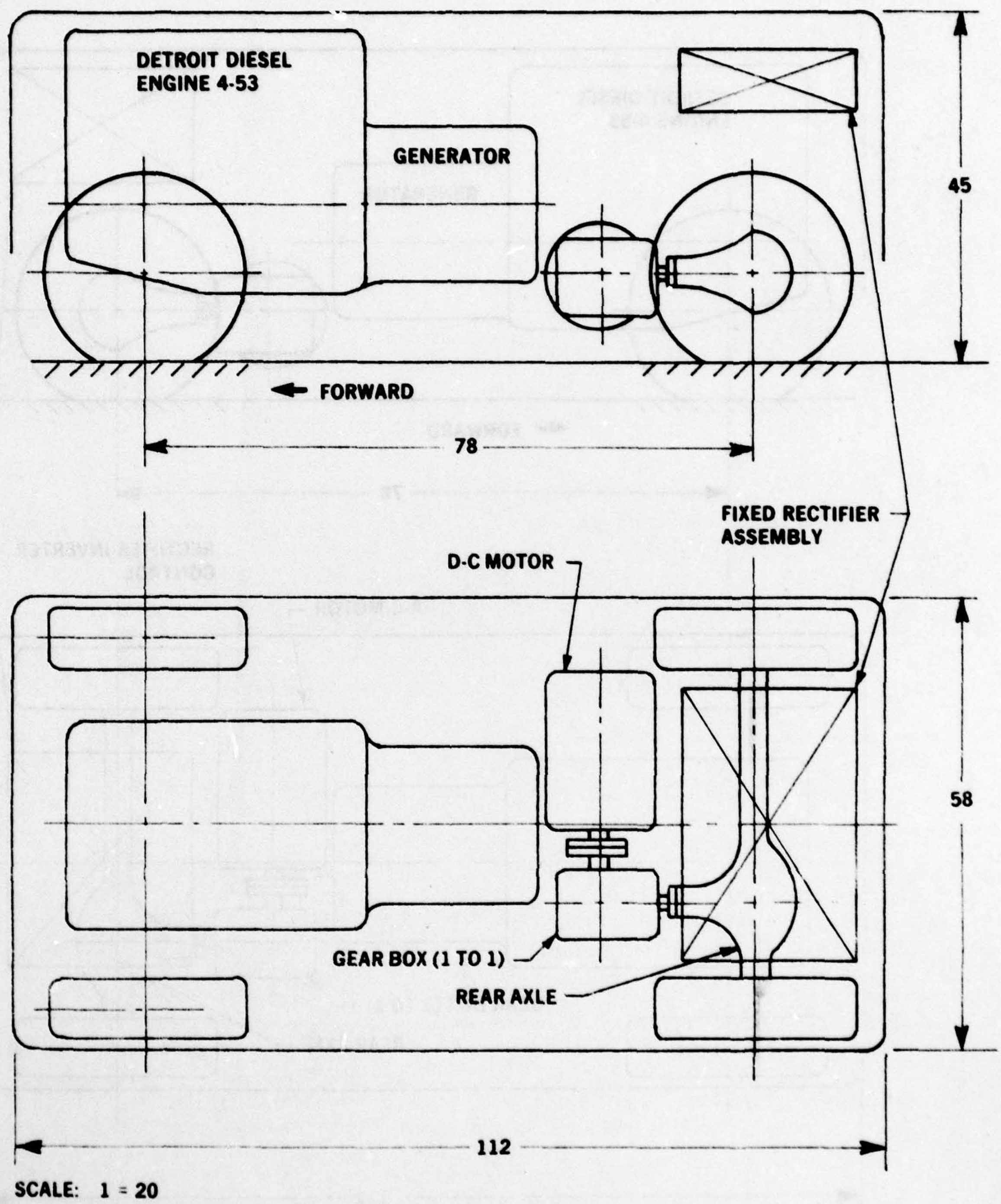
B-7. HYDRODYNAMIC (AUTOMATIC TRANSMISSION) WITH HYDRAULIC PUMP DRIVE



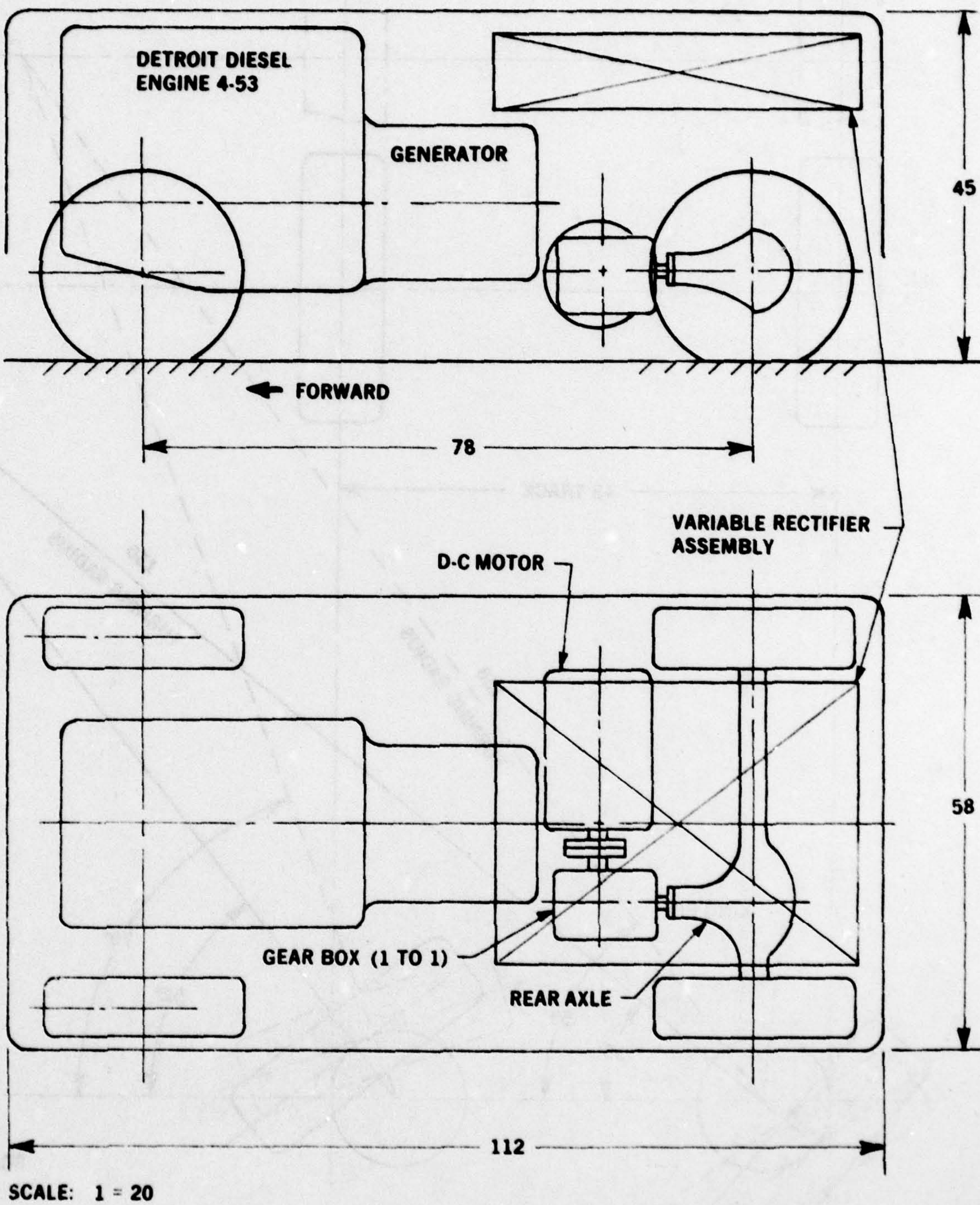
B-8. A-C MOTOR WITH MICROCOMPUTER CONTROL



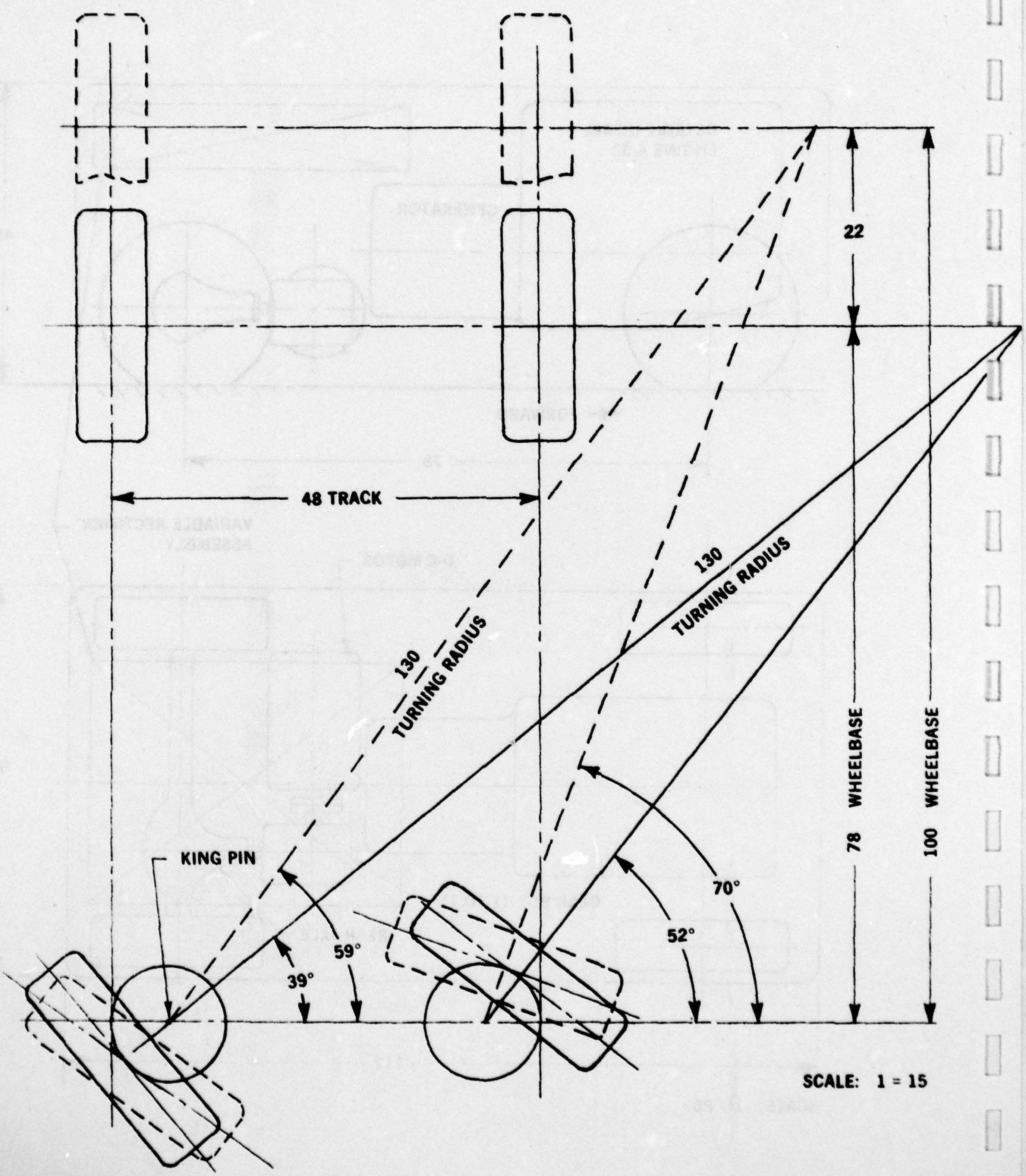
B-9. A-C MOTOR WITH RECTIFIER INVERTER CONTROL



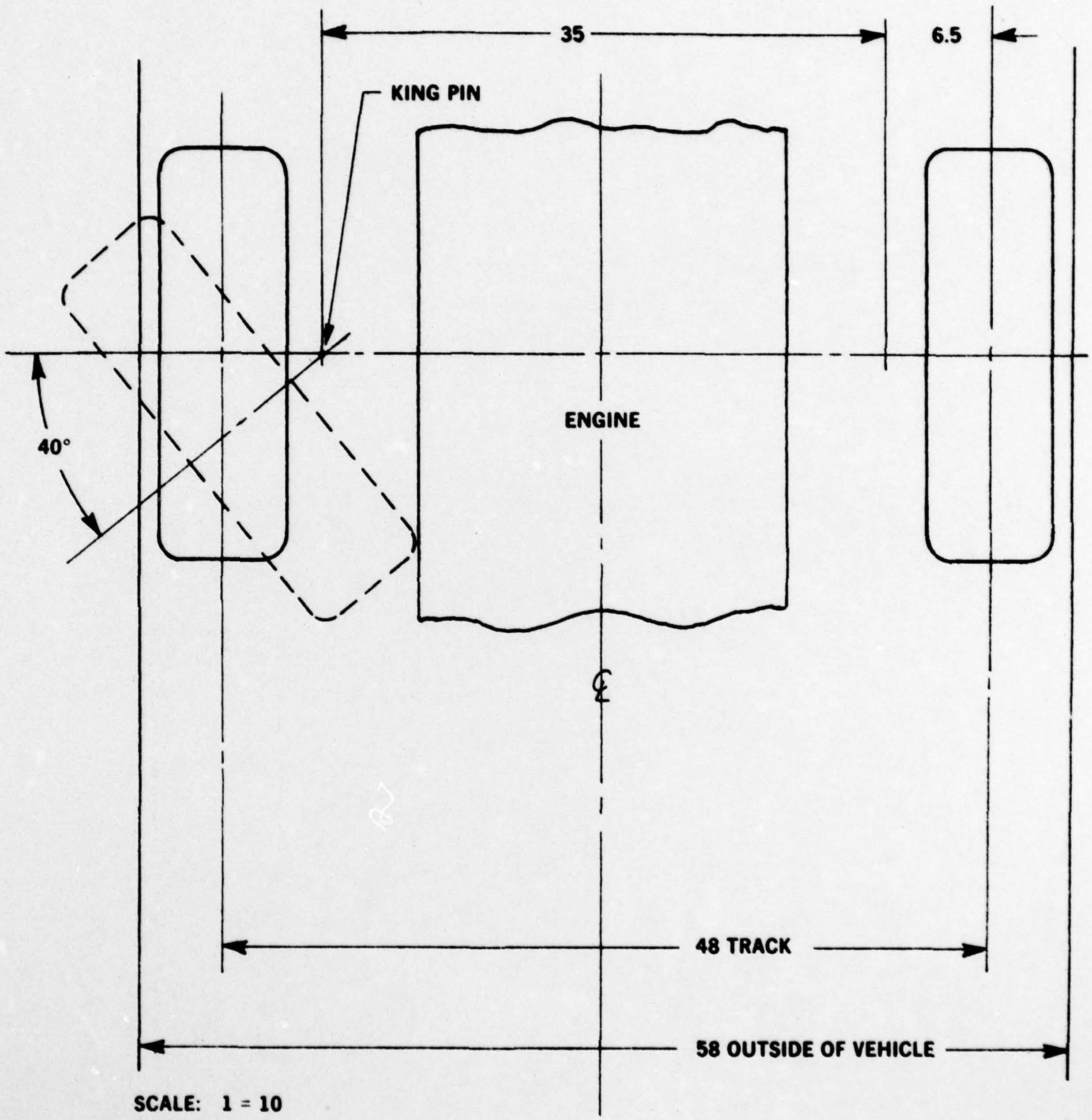
B-10. D-C MOTOR WITH FIXED RECTIFIER CONTROL



B-11. D-C MOTOR WITH VARIABLE RECTIFIER CONTROL



B-12. WHEEL BASE EFFECT ON STEERING ANGLES



B-13. STEERING RESTRICTION DUE TO INTERFERENCE WITH ENGINE

APPENDIX C - LIST OF TECHNICAL CONTACTS

1. Abacus Controls, Inc., Sommerville, NJ
2. Abex Corp-Denison Division, Columbus, OH
3. Advanced Devices Lab, Pompton Lakes, NJ
4. AEG-Telefunken Corp., Englewood Cliffs, NJ
5. Aeroflex Laboratories, Inc., Motor Products Div., Plainview, L.I., NY
6. Airline Hydraulics, Cornwells Heights, PA
7. Airoyal Company, Maplewood, NJ
8. Alan Leland Co., Vandalia, OH
9. Alfa Industries, Inc., Caldwell, NJ
10. Allis-Chalmers Corp., Milwaukee, WI
11. Louis Allis Co., Milwaukee, WI
12. Louis Allis Co., Springfield, PA
13. Ambi-Tech Industries, Inc., Hillsdale, NJ
14. American Standard Inc., Industrial Products Div., Dearborn, MI
15. Applied Motors, Inc., Rockford, IL
16. Archer Engineering, Inc., Philadelphia, PA
17. Atlas Motor Generator Mfg. Co., El Monte, CA
18. Baldor, Collingswood, NJ
19. Baldor, Fort Smith, AR
20. B&B Motor & Control Corp., New York, NY
21. Belyea Co. Inc., Jersey City, NJ
22. Bendix, Eatontown, NJ
23. Bendix, Brake & Steering Div., South Bend, IN
24. Bendix, Electric & Fluid Power Div., Utica, NY

C. LIST OF TECHNICAL CONTACTS (Continued)

25. Allen Bennett Ltd., Orgreave Drive, Sheffield S13 9NR England
26. Bergen Research Engineering Corp., Maywood, NJ
27. Berger Lahr Corp., Jaffrey, NH
28. Beta Marine Corp., Reading, PA
29. Bird Johnson Co.-Fluid Power Div., Walpole, MA
30. Bison Gear & Eng. Corp., Subsidiary of Diebel Mfg. Co., Downers Grove, IL
31. Black Clawson Co., Electro-Flyte Div., Fulton, NY
32. Bogue Electric Mfg. Co., Paterson, NJ
33. Borg Warner Corp., Warner Gear Div., Muncie, IN
34. Borg Warner Corp., Rockford Clutch Div., Rockford, IL
35. Borg Warner Corp., Hydraulics Div., Wooster, OH
36. Boston Gear, Incom International Inc., Quincy, MA
37. Brown Boveri Corp., North Brunswick, NJ
38. David Brown Gear Industries, Inc., Ontario, Canada
39. Brundage Associates Inc., Rochester, NY
40. Burton Industries, Inc., Pawtucket, RI
41. J. I. Case, Racine, WI
42. Caterpillar Tractor Co., Peoria, IL
43. Central Power Co., Troy, MI
44. Century Motor and Compressor Co., Inc., Newark, NJ
45. Chrysler Corp., New Process Gear Div., East Syracuse, NY
46. Chrysler Corporation, Troy, MI
47. Clark Equipment Co., Corporate Labs, Buchanan, MI
48. Clark Equipment Co., Transmission Div., Jackson, MI
49. Cleveland Machine Controls, Inc., Cleveland, OH

C. LIST OF TECHNICAL CONTACTS (Continued)

50. Commercial Shearing, Inc., Youngstown, OH

51. Consolidated Diesel Electric Co., Old Greenwich, CT

52. Control Products Corp., Chicago, IL

53. Control Systems Research, Pittsburgh, PA

54. Cotta Transmissions, Rockford, IL

55. Dana Corp., Industrial Power Transmission Div., Warren, MI

56. Dana Corp. Industrial Power Transmission Div., West Chester, PA

57. Dart Controls, Inc., Zionsville, IN

58. Dayton Electric Mfg. Co., Chicago, IL

59. Delco Products Div. of GMS, Dayton, OH

60. Delta Electronic Control Corp., Irvine, CA

61. Demag Material Handling Corp. Drives Div., Cleveland, OH

62. Detroit Diesel, Allison Div., Detroit, MI

63. Detroit Diesel, Allison Div., Indianapolis, IN

64. Device Engineering Co., Media, PA

65. Doerr Electric Corp., Cedarburg, WI

66. Double A Products Co., Marchester, MI

67. Dresser Industries, Inc., Power Transmission Div., East Orange, NJ

68. Drive All Manufacturing Co., Detroit, MI

69. Eaton Corp., Industrial Drives Div., Cleveland, OH

70. Eaton Corp., Fluid Power Div., Eden Prarie, MN

71. Eaton Corp., Industrial Drives, Fairfield, NJ

72. Eaton Corp., Industrial Drives Div. Dynamatic Plant, Kenosha, WI

73. Eaton Corp., Fluid Power Div., Marshall, MI

74. Eaton Corp., Philadelphia, PA

C. LIST OF TECHNICAL CONTACTS (Continued)

75. Eaton Corp., Richmond, IN

76. Eaton Corp., Spencer Div., Spencer, IA

77. Electra Motors Operations, Dresser Power Trans Div., Anaheim, CA

78. Electric Machinery, Minneapolis, MN

79. Electro-Devices Inc., Servospeed Div., Paterson, NJ

80. Elinco, Norwalk, CT

81. Eltra Corp., Prestolite Electrical Div., Toledo, OH

82. Emerson Electric Co., U.S. Electrical Motors Div., Milford, CT

83. Exide-UPS, Raleigh, NC

84. Fairchild, Industrial Products Div., Winston-Salem, NC

85. Fairfield Mfg. Co., Lafayette, IN

86. Fasco Industries, Inc., Ozark, MO

87. Fidelity Electric Co. Inc., Lancaster, PA

88. Fincor, York, PA

89. Firing Circuits, Inc., Norwalk, CT

90. W. J. Flannelly Co., Hillside, NJ

91. Fluid Power Inc., Blue Bell, PA

92. FMC Corp., Drive Div., Philadelphia, PA

93. Foote-Jones Operations, Dresser Power Transmissions Div., Downers Grove, IL

94. Ford Motor Co., Transmission Div., Livonia, MI

95. Ford Tractor Operations, Industrial Equip. Eng., Troy, MI

96. Franklin Institute Research Labs, Philadelphia, PA

97. Funk Mfg. Co., Div. of Gardner-Denver, Coffeyville, KS

98. Garrett Corp., El Segundo, CA

99. Garrett Corporation, Washington, DC

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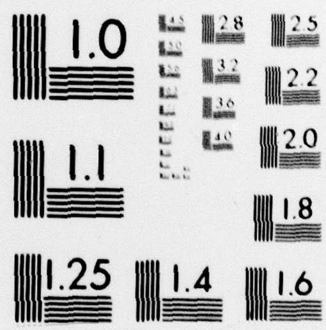

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

C. LIST OF TECHNICAL CONTACTS (Continued)

100. Gear Motions, Inc., Shrewsbury, MA

101. Gear Specialties, Inc., Chicago, IL

102. General Electric, Binghamton, NY

103. General Electric D.C. Motor & Generator Products Dept., Erie, PA

104. General Electric General Purpose Motor Div., Wilmington, MA

105. General Electric, Philadelphia, PA

106. General Electric Marine & Defense Facility Sales Organization, Schenectady, NY

107. General Electric Aerospace Div., Wilmington, MA

108. General Motors Corp., Detroit, MI

109. Gerbing/Formsprag, Subsidiary of Dana Corp., Elgin, IL

110. Gould Imperial-Eastman, Folcroft, PA

111. Gould Inc., Electric Motor Div., Saddle Brook, NJ

112. Gould Inc., Century Electric Div., St. Louis, MO

113. Graham Transmissions Co., Div. of Stowell Industries, Inc., Drive Systems Dept., Menomonee Falls, WI

114. W. W. Grainger Co., Chicago, IL

115. Grant Gear, Inc., South Boston, MA

116. Lewis H. Hein Co., West Conshohocken, PA

117. G. K. Heller Corp., Floral Park, NY

118. Hilliard Corp., Elmira, NY

119. Hobart Brothers Co., Troy, OH

120. John R. Hollingsworth Co., Phoenixville, PA

121. Howell Electric Motors, Plainfield, NJ

122. Hurst Mfg. Co., Princeton, IN

123. Hydreco-A Unit of General Signal, Kalamazoo, MI

C. LIST OF TECHNICAL CONTACTS (Continued)

124. Hydro-Air Inc., Fairfield, NJ

125. Hyper-Loop Inc., Bridgeview, IL

126. Ideal Electric Mfg. Co., Mansfield, OH

127. Indiana Gear Works Div. of The Buehler Corp., Indianapolis, IN

128. International Harvester, Hinsdale, IL

129. International Rectifier, El Segundo, CA

130. J. H. Jones Co., Houston, TX

131. Johnson & Towers, Inc. Mount Laurel, NJ

132. Kalglo Electronics Co. Inc., Industrial Div., Bethlehem, PA

133. Kato Engineering Co., Mankato, MN

134. Kennett Corp., Newton, MA

135. Kershner Assoc., Devon, PA

136. Phil Knox, Lutherville, MD

137. Koppers Co., Inc., Baltimore, MD

138. Thomas F. Krick Co., Lancaster, PA

139. Lear Siegler Inc., Englewood, CO

140. Lincoln Electric Co., Cleveland, OH

141. Liquid Drive Corp., Holly, MI

142. Litton Industrial Products Inc., Louis Allis Div., Milwaukee, WI

143. Lovejoy, Inc., Downers Grove, IL

144. Machine Components Corp., Plainview, L.I., NY

145. Machine Design, Cleveland, OH

146. Frank MacPherson Control Products, Westfield, NJ

147. Marathon Electric Mfg. Corp., Wausaw, WI

148. Marine Electrical Airway Industrial Products, Edison, NJ

C. LIST OF TECHNICAL CONTACTS (Continued)

149. John McClelland and Assoc. Inc., Flourtown, PA

150. Meridian Laboratory Inc., Madison, WI

151. Moog, Inc., Industrial Division, East Aurora, NY

152. National Hydraulics Inc., Elmhurst, IL

153. Nordco Products, Los Angeles, CA

154. Nova Electronics, Nutley, NJ

155. Ohio Gear Inc., Liberty, SC

156. Oilgear Co., Doylestown, PA

157. Oilgear Co., Milwaukee, WI

158. Oklahoma State Univ., Fluid Power Research Center, Stillwater, OK

159. Optimation Engineering Associates, Inc., Medford, NJ

160. Panasonic Motors, Matsushita Electric Corp. of America, Franklin Park, IL

161. Panasonic Co., Precision Motor Dept., Secaucus, NJ

162. Parametrics, Orange, CT

163. T. H. Paris Co., Trenton, NJ

164. Ernest H. Pauli, Warren, NJ

165. Pearse Fluid Power, Rahway, NJ

166. Penske G. M. Power Inc., Philadelphia, PA

167. Plessey Dynamics, Hillside, NJ

168. Plessey Airborne Corp., New York, NY

169. PMI Motors, Syosset, NY

170. Polyspede Electronics, Dallas, TX

171. Polyspede Electronics, Technical Industrial Sales, Wayne, PA

172. H. K. Porter Co. Inc., Pittsburgh, PA

C. LIST OF TECHNICAL CONTACTS (Continued)

173. Precision Controls Inc., Hawthorne, NJ

174. PTI Controls, Fullerton, CA

175. Rae Corp., McHenry, IL

176. Rae Corp., Milford, CT

177. Ramsey Controls, Mahwah, NJ

178. Randtronics, Menlo Park, CA

179. Reliance Electric Co., Cleveland, OH

180. Reliance Electric Co., Master-Reeves Div., Columbus, IN

181. Reliance Electric Co., King of Prussia, PA

182. Renold, Inc., Westfield, NY

183. Rexroth Corp., Bethlehem, PA

184. Robbins & Meyers, Inc., Power Transmission Div., Southington, CT

185. Robbins & Meyers, Inc., Springfield, OH

186. Roberts Electric Co., Chicago, IL

187. Robicon Corp., Pittsburgh, PA

188. Rockford Clutch, Div. of Borg Warner, Rockford, IL

189. Ross Gear Div. of TRW, Lafayette, IN

190. Sabina Electric & Eng. Co., Anaheim, CA

191. Safeguard Automotive Corp., Hub City Div., Aberdeen, SD

192. Saginaw Steering Gear Div. of General Motors Corp., Saginaw, MI

193. Sawyer Industries, Inc., Arcadia, CA

194. Schulz Controls Inc., New Haven, CT

195. SECO Electronics Corp., Lancaster, SC

196. Servo-Optic Systems, Subsidiary of Polyspede Electronics Corp., Dallas, TX

C. LIST OF TECHNICAL CONTACTS (Continued)

197. Servo-Tek Products Co., Hawthorne, NJ

198. Sevcon, Burlington, MA

199. Shingle & Gibb Co., Trenton, NJ

200. Siemens-Allis Inc., Union, NJ

201. Silicon General Inc., Garden Grove, CA

202. Singer Co., Kerfott Div., Little Falls, NJ

203. Skurka Engineering Co., Los Angeles, CA

204. Snow-Nabstedt Gear Div. of Granite State Machine Co. Inc., Manchester, NJ

205. Society of Automotive Engineers, Inc., Warrendale, PA

206. Soleq Corp., Chicago, IL

207. Sperry Rand Corp., Sperry Vickers Div., Troy, MI

208. Sperry Vickers, Marine & Ordnance Div., Jackson, MS

209. Sperry Vickers, Springfield, NJ

210. Spicer Axle, Div. of Dana Corp., Fort Wayne, IN

211. H. P. Stallings & Assoc., Philadelphia, PA

212. Sterling Power Systems, Inc., Irvine, CA

213. Stewart & Stevenson Services, Houston, TX

214. Sundstrand Hydro-Transmission, Unit of Sundstrand Corp., Ames, IA

215. Superior Electric, Bristol, CT

216. Tech Systems Corp., Precise Power Systems Div., Thomaston, CT

217. Topaz Electronics, San Diego, CA

218. Torque Systems Inc., Waltham, MA

219. Transmission Engineering Co. Inc., Ft. Washington, PA

220. Trilectron Ind. Inc., Rectifier Systems Div., Newark, NJ

C. LIST OF TECHNICAL CONTACTS (Continued)

221. TRW - Main Office, Cleveland, OH

222. TRW Globe Motors, Dayton, OH

223. TRW - LSI Div., Lawndale, CA

224. TRW Power Semiconductors, Lawndale, CA

225. Turner Uni-Drive Co., Kansas City, MO

226. Twin Disc, Inc., Racine, WI

227. Unitron, Garland, TX

228. Van Air & Hydraulics, Maple Shade, NJ

229. Vee-Arc Corp., Westboro, MA

230. Voith Transmissions, Inc., Appleton, WI

231. Walker Industrial Products, Roseland, NJ

232. Warner Gear, Div. of Borg Warner, Muncie, IN

233. Web Controls Corp., West Englewood, NJ

234. Welco Industries, Inc., Cincinnati, OH

235. WER Industrial, Div. of Emerson Electric Co., Grand Island, NY

236. Western Gear Corp., Electro Products Div., Lynwood, CA

237. Western Manufacturing Co., Detroit, MI

238. Westinghouse Corp., Buffalo, NY

239. Westinghouse Electric Corp., Hillside, NJ

240. Westinghouse Electric Corp., Aerospace Electrical Div., Lima, OH

241. Westinghouse Electric Corp., Transportation Div., West Mifflin, PA

242. Westinghouse Electrical Supply, Trenton, NJ

243. Windworks Inc., Mukwonago, WI

244. Winsmith Speed Reducers, Div. of UMC Industries, Inc., Springville, NY

C. LIST OF TECHNICAL CONTACTS (Continued)

245. Wollock and Lott Co., Kenilworth, NJ

246. T. B. Wood's Sons Company, Chambersburg, PA

247. Young Radiator, Racine, WI

248. Zero-Max Industries, Inc., Minneapolis, MN

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