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CONCEPT STUDY

FOR

TRACKED AMPHIBIAN PERSONNEL AND CARGO CARRIER (LVTPX11)

CONTRACT NO. NObs-4464

VOLUME IV

APPENDIX [illegible] ALTERNATE DRIVE TRAINS

FINAL REPORT
Of a
CONCEPT STUDY
For a
TRACKED AMPHIBIAN PERSONNEL AND
CARGO CARRIER
(LVTPX11)

VOLUME IV

Prepared for

BUREAU OF SHIPS
DEPARTMENT OF THE NAVY
Washington 25, D. C.

By

Ordnance Division
FMC CORPORATION
San Jose, California

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ELECTRIC PROPULSION SYSTEM FOR LVTPX11

During the past four years, FMC has been studying and evaluating all types of AC and DC electric drive systems for tracked and wheeled vehicles. The electric drive has many recognized advantages, as shown in Figure J-1, but until recently the large size and weight of electrical components to a great extent ruled out the electric drive for practical use in military tactical vehicles.

Now, however, recent advances in the state-of-the-art have provided new types of power control components which permit the use of small, lightweight, and simple rotating equipment. Using this new technology, the Ordnance Division of FMC has been conducting intensive feasibility studies which show that electric drive systems are both feasible and desirable.

In this study, it became apparent that, while the DC system has many desirable features, the large physical size and weight of the motors make them unsuitable for tactical vehicle propulsion where space and weight are at a premium. Also, wound rotating parts and commutators limit the speeds at which DC motors can be run, thereby limiting any reduction in size which might be possible with higher speeds. In addition, wound rotors and commutators do not adapt well to modern production techniques. For the above reasons, only the new, variable-speed AC electric drive system was considered for the LVTPX11 vehicle.

The AC electric drive system proposed herein for development has been under careful study and analysis by FMC engineers for over a year. During this time, FMC has thoroughly analyzed each of the systems proposed by many electrical manufacturers and has discussed the systems and their major components at length with engineers from more than 25 companies specializing in this area.

This system stems from the recent development of the silicon-controlled rectifier (SCR), which provides a completely solid-state control system to afford flexibility of control similar to that of a DC system.

Characteristic	Electrical System	Hydraulic System	Mechanical System
RELIABILITY COMPARISON			
Durability	Few moving parts. Control uses solid-state devices with no moving parts. Extreme life, when used within their capabilities.	Valves and moving parts subject to deterioration with wear and contamination.	Complex gearing and controls subject to wear.
Reliability	High, with few wearing parts. Reduced gear wear with least shock loading.	Good, but subject to wear from exposure and contamination. Low shock loading on gears.	Shock loading maximum on gears and power plant.
Ease of Maintenance	Components are unitized for easy replacement. Cables easily repairable or replacable.	Field maintenance difficult, due to contamination. A broken line loses all power to system.	Mechanical failure difficult to repair in field.
Spare Parts	Minimum in system. Cleanliness not significant problem.	Minimum parts required. Difficult to keep clean.	Larger number of parts.
DESIGN ADVANTAGES			
Space Utilization	Electrical components placed to give maximum space to payload.	Same as electrical.	Components must be placed to align with power train system and hull.
Selection of Power Plant	Any type energy source adaptable with minimum modifications --		
Turbines	Direct drive	Same as electrical.	Reduction gear drive.
Reciprocating Engine	Speed increases	Same as electrical.	Transmission
Fuel Cell	Inverter in place of alternator.	Motor required	Use electric drive
Thermo-electric Generator	Inverter in place of alternator.	Motor required	Use electric drive
Controls	Some complex feedback control mechanisms.	Same	Simplest in design.
Weight Distribution	Components can be positioned for best weight distribution. Drive shafts or high-pressure lines replaced by flexible, easily routed cables.	Same as electrical High-pressure lines must be protected	Components must align with power train. Drive shafts must be enclosed for safety.
PERFORMANCE COMPARISON			
Fuel Economy	Power plant automatically controlled to operate on optimum fuel economy curve (power plant speed not directly related to vehicle speed.	Same as electrical.	Power plant must operate at speed set by vehicle and gearing
Tractive Effort	Higher at low speeds, utilizing up to full power, for limited time. Smooth and easily controlled. Overload capabilities available.	Same as electrical but no overload capability.	Linked to power plant through converter, with higher efficiency losses. No overload available and more difficult for smooth control.
Steering Versatility	Infinitely variable steering ratio, with capability of opposite rotation of tracks.	Same as electrical but opposite track rotation difficult in some systems.	Turns by braking one track for short periods, causing greater speed loss cannot turn about vehicle center.
Braking	Electrical braking. No fading - no wear Infinite control. Heat easily removed by air cooling.	Some systems regenerative. Some use porting, giving added cooling problems. Must design for higher pressures.	Friction brakes with resultant fading on long use
Controls	Steering wheel, foot brake, and accelerator pedal. Good human engineering. Smooth operation.	Similar to electrical with some systems.	Steering levers and accelerator pedal. Rough operation.
Water Speed	Motor-driven propeller gives optimum speed and power for water operation.	Same as electrical.	Mechanical linkage unwieldy. More difficult to stow. Requires fixed physical relation for power shafts.
UNIVERSAL POWER SOURCE			
DC - 60 cycle, 400 cycle, 800 cycle	Auxiliary power while mobile. Precise frequency and DC available up to approximately 40 KW. Power plant for field station capability, to full power of system.	Requires hydraulic motor, control, and alternator for auxiliary.	Requires additional alternator and drive assembly to get auxiliary power.

FIGURE J-1

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

This will give maximum land performance and the capability of efficiently using full engine power for amphibious operation.

The primary system consists essentially of a high-frequency alternator, a frequency converter and control, AC traction motors, and a propeller motor for water operation. A possible arrangement of this system within a vehicle is shown in Figure J-3. Parameters for the electric drive system design were established from the specifications of the LVTPX11 and from FMC's previous experience with LVT vehicles. From these parameters, specific electrical design criteria were established, taking into account the electrical, mechanical, and cooling characteristics for over-all optimum performance.

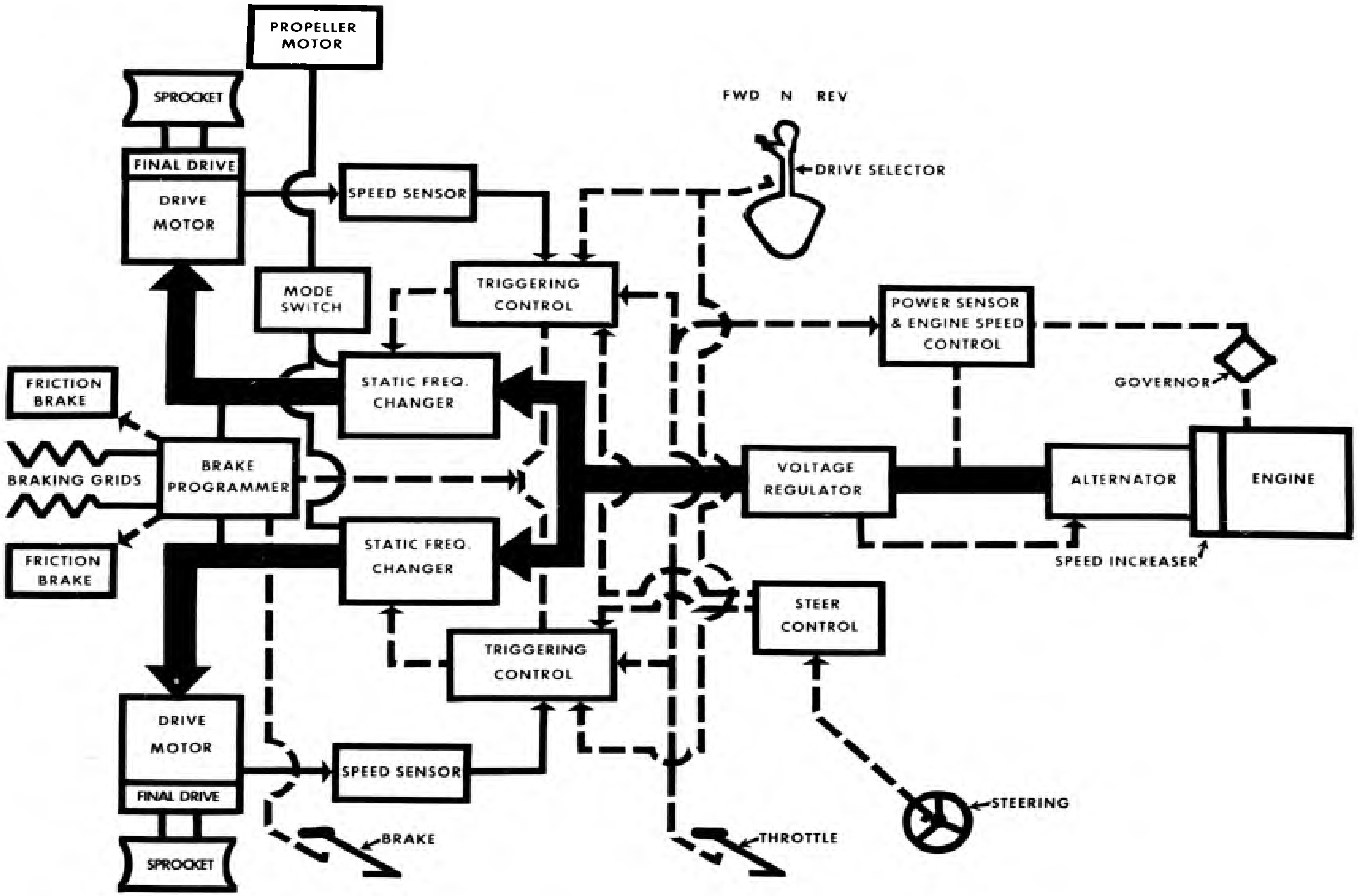
Several electrical companies active in the field of variable-speed drives were asked to propose on the electrical components for this specific AC system. From the data submitted, components for an AC variable-speed drive were selected and compared. Initial investigation indicates that the values listed are realistic, and the equipment for the system is based on present development areas that are felt to be sure of attainment by 1964 or 1965, with production in 1966. More exotic development areas now being studied (such as cryogenic motor cooling and new motor operational theory) may eventually produce marked additional improvements.

In addition, the alternator may be used as a mobile ground power source supplying up to 240 KW of power (400 cycle, 60-cycle, DC, etc.) for such uses as radar, missile and support equipment, command posts, command vehicles, communications centers, and fire direction vehicles.

	Alternator	Traction Motors	Converter	Prop. Motor
Type of Unit	Inductor	Induction	Frequency	Induction
Power Rating	300 KVA	120 HP		225 HP
Speed (rpm)	20,000	1,200-24,000		12,000
Voltage	277/480	277/480		277/480
Current (rated)	370	185	400	370
Frequency	2400-3200	40-800		40-800
Phases	3	3		3
Cooling Method	Oil	Oil	Heat Sink	Submersed
No. of SCR's			72	
Volume (cu ft)			2-1/2	
Length (in.)	22	13		20
Diameter (in.)	16	18		12
Weight (lb)	400	500	200	350
Efficiency	91%	92%	97%	92%
System Efficiencies				
Land Mode*	74.5			
Water Mode*	76.5			

*Includes 2% miscellaneous control power and line losses.

FIGURE J-2



ELECTRIC DRIVE - BLOCK DIAGRAM

FIGURE J-3

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

Power Plant

As can be seen from the data below, this electric drive system can be easily adapted to any type of power plant.

Of the two types available today (turbine and reciprocating engine), the former complements the physical and operating advantages of the electric system to the greater degree. The alternator can be run directly coupled to the turbine shaft (with a resultant decrease in weight and increase in efficiency), due to elimination of the gear increaser required with the reciprocating engine.

Due to the problems of delivering sufficient air to the turbine, the major portion of this electric drive description is based on using the same type of reciprocating engine as the recommended mechanical system. This decreases the over-all efficiency in the land mode, as shown in Figure J-2, from 74.5% to 72%.

This system, with only minor modifications, can be used with any of the following types of power sources:

Turbine

- Directly coupled to high-speed alternator, eliminating the need for a speed increaser.
- The electrical system complements the turbine, since it allows use of either fixed- or free-shaft turbines.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

- A slight modification in the converter fuel control will give the same speed-governing capabilities as for the reciprocating engine.

Reciprocating Engine

- Similar to turbine system, in that output power is transferred through a speed increaser to a very-high-speed alternator (about 30, 000 rpm).

- Converter control governs engine speed, to obtain minimum fuel consumption and to extend engine life.

Fuel Cell

- Converts fuel to DC power, which is fed through an inverter for conversion to AC.

- The inverter can easily supplant the alternator in the previous systems.

- The fuel cell complements the electric drive by eliminating much of the noise, lubrication problems, and power train losses.

Thermonuclear Energy Source

- Converts heat to DC power, which is fed into an inverter, as used with the fuel cell.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

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ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

- Requires only a slight modification of the converter fuel-consumption control used with the fuel cell.

- Complements electrical system by practically eliminating noise of operation, refueling requirements, and fuel storage.

It can be seen that the electric drive system has unlimited compatability with all proposed power sources.

The requirements and parameters for the electrical system were based on the operational requirements of the LVTPX11.

Where the mechanical system horsepower curve has peaks and valleys, the electrical system curve is smooth, providing an over-all performance capability equal to or better than the mechanical system.

Data from the LVTP6, which FMC designed and tested at this facility, was also used to determine the requirements for low-speed performance and braking and steering of the vehicle.

The following design criteria were established:

40 mph on level ground with 100 lbs/ton rolling resistance

15 mph up a 10% slope with 100 lbs/ton rolling resistance

6 mph up a 30% slope with 140 lbs/ton rolling resistance

2-1/2 mph on a 70% slope with 140 lbs/ton rolling resistance (this requires less than full engine power)

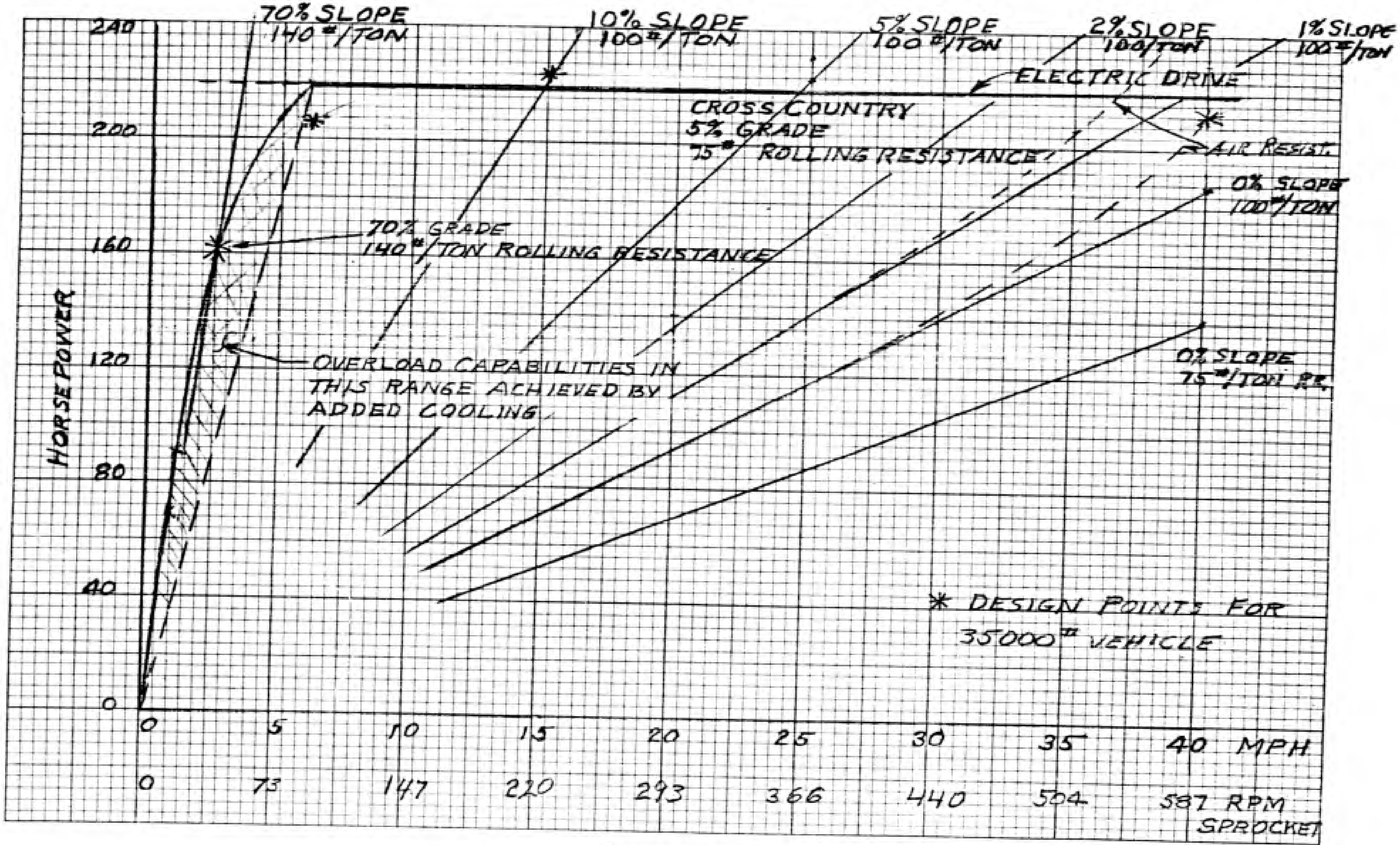

FIGURE J-4

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ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

- Steering capable of a pivot turn from standstill and of continuous turning on 20-foot and 60-foot radii, with up to 100% of full engine power fed to the outside track

- Water operation most efficiently utilizing the full power of the engine in landing operations

Components will be located for maximum utilization of space and for balanced distribution of weight, using high-speed lightweight alternator and motors.

The performance to be expected from the proposed system shown in Figure J-3 meets the established electrical design parameters. This predicted performance was obtained by realistic evaluation, estimation, and calculation of each component in the system, with regard to the efficiency, power, and thermal capacity of each and requirements of the necessary auxiliary systems.

The system is based on utilizing the best logical combination of components and gearing. Figure J-4 indicates its performance closely matches the established requirements. Of particular significance are the high torque capability in the lower speed area and a smooth horsepower-versus-speed curve.

Infinite steering ratio and the ability to run the tracks in opposite directions to obtain a true pivot turn are inherent. Up to 100% of engine power can be regenerated to the outside track.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

Controlled braking is provided entirely by electrical regenerative and dynamic braking means. The motors and braking grids have sufficient capacity to stop the vehicle from 40 mph on downgrades with no additional help.

Mechanical brakes are used for parking and emergency, only.

Vehicle Speed Control

The vehicle speed control is commanded by a standard foot throttle. The throttle actuates a rheostat-type element which varies the frequencies of both control oscillators simultaneously. The resultant increasing converter output frequency is fed to the drive motors and, due to the increasing slip, provides drive motor acceleration. The maximum allowable acceleration, determined by maximum slip, is programmed into the speed control system as a limit, thus ensuring maximum acceleration. The drive motors accelerate toward synchronous speed until the desired vehicle speed is reached, at which time the operator reduces the throttle to maintain this speed. From the driver's standpoint, this system is operated in the same manner as conventional vehicles. Stated simply, the speed control is a power control which controls the motor slip. When the driver removes his foot from the throttle, the oscillators reduce and maintain synchronism with the motors.

The drive motor's speed/torque curve does not coincide with the power plant's curve, so a programmed limit is incorporated to prevent the drive motors from overloading the engine to a stall condition.

ELECTRICAL PROPULSION SYSTEM FOR LVTPX11 (Continued)

Steering

Under constant-speed driving conditions, the two reference oscillators generate identical output voltages and frequencies (which command each of the SCR converters to supply corresponding identical output voltages and frequencies) but, when turning, one reference oscillator frequency is lowered by actuating the steering wheel, which causes a corresponding frequency change in the pertinent converter. The drive motor supplied by this converter reacts by slowing and braking the overrunning load. This regenerated power is passed from the braking motor back through its converter, through the other converter, and is added to the outside motor to help maintain speed.

The independent control of each converter provides the capability of operating the two tracks in opposite directions, which will provide a true pivot turn about the vehicle's center.

Various steering situations have been analyzed and power calculations have been made. Based on these results, this system is designed to apply full engine horsepower to the outside track. Steering conditions which impose larger demands than this will require current limitations for the converter, and the balance of the regenerated energy will be routed to the braking grids of the inside motor. Steering power calculations indicate that these maximum powers are comparable to maximum braking power requirements.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

There is a very definite trade-off in this area between economy and extended-performance capability, in order to incorporate converters which would provide full regenerated power to the opposite track. Since this is the only condition where the full capability of the converter is utilized, the added cost and size of the larger capacity hardly appears justified.

Braking

Braking is accomplished by operating the motors as generators and dissipating the energy in an air-cooled resistive load bank, with some energy fed back through the SCR converters and the alternator to be dissipated in the engine. This action is commanded, through a standard-appearing operator's brake pedal, by lowering the frequency of the SCR converter reference oscillators.

With an electric drive system, the vehicle engine is no longer directly linked to the drive unit. For this reason, the normal closed-throttle compression drag is missing, and a freewheeling effect might be expected. FMC investigated methods whereby, in the closed-throttle position, a small amount of braking would be programmed to correspond to engine braking in a standard vehicle. However, the rolling resistance and track losses provide sufficient drag or deceleration, eliminating the need for this form of braking.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

For greater deceleration, the brake pedal is actuated, causing braking grid contactors to close, dissipating the energy in the resistive load grids. There is a maximum allowable amount of regenerative braking provided by the drive motors. This selected point on the reverse speed/torque curve (see Figure J-5) is programmed into the system. Beyond this slip point, the generated energy falls off to a low value. By this programming and sensing method, the maximum braking point will not be exceeded.

For unusual braking demands, failure of the dynamic braking system, and/or parking, an auxiliary friction braking system is provided. The two braking systems are interconnected in such a manner that the operator normally actuates the dynamic braking for the majority of deceleration requirements but can engage the friction system by further depressing the brake pedal. The parking brake is actuated in a conventional manner.

The braking capabilities of this system, when summing the final-drive losses, track losses, rolling resistances, and motor braking, total 1400 hp. This braking capability will decelerate the vehicle from 40 mph to 0 in 4.84 seconds, corresponding to a deceleration rate of 12.1 ft/sec^2 (see Figure J-6). Calculations show also that there is adequate capability to stop the vehicle from 40 mph on a 30% downgrade in approximately 600 ft.

Additional braking conditions encountered when turning are further discussed under "Steering".

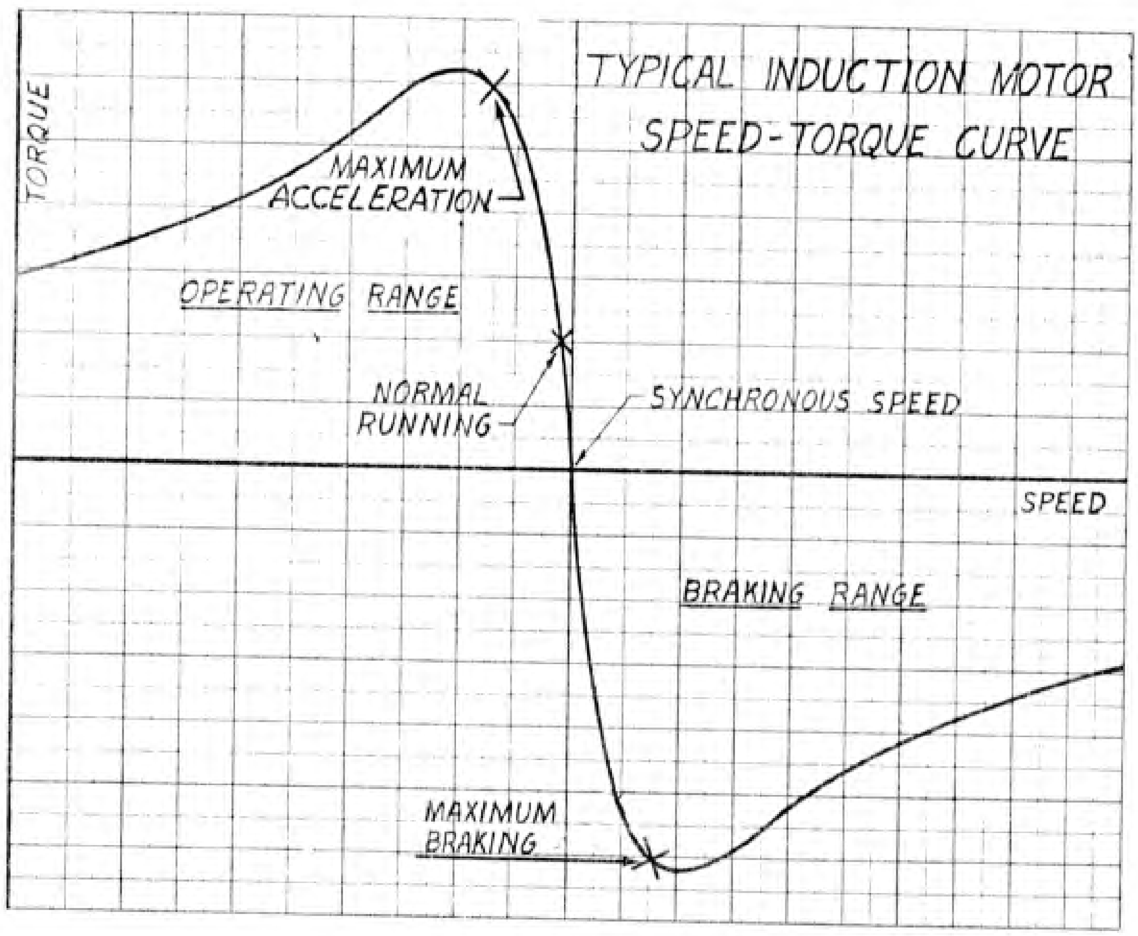

TYPICAL INDUCTION MOTOR
SPEED-TORQUE CURVE
TORQUE
MAXIMUM
ACCELERATION
OPERATING RANGE
NORMAL
RUNNING
SYNCHRONOUS SPEED
SPEED
BRAKING RANGE
MAXIMUM
BRAKING

FIGURE J-5

ORD ENG 108

FOOD MACHINERY AND CHEMICAL CORPORATION

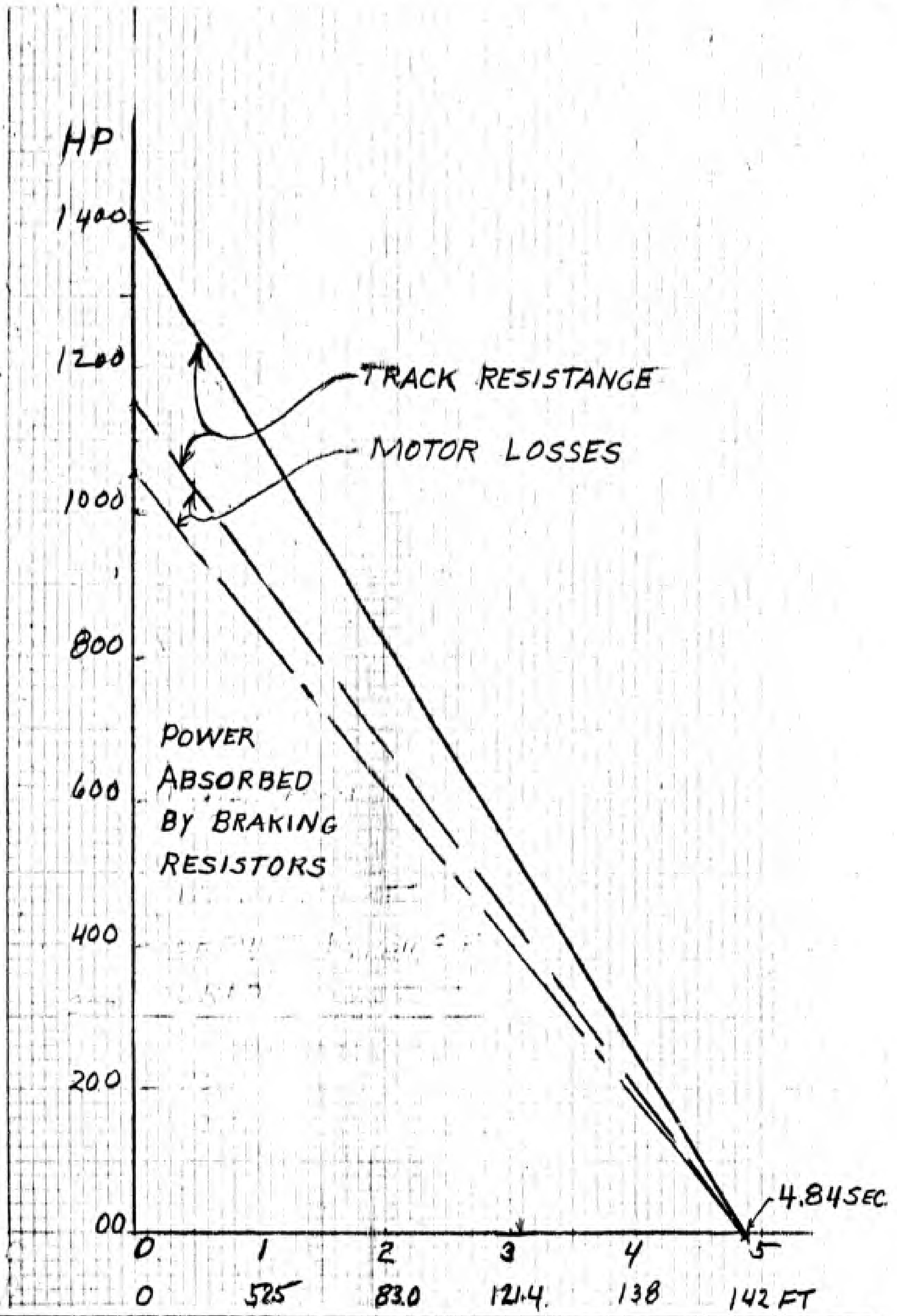
DISCLOSURE SHEET

PAGE ____ OF ____

TITLE BRAKING REQUIREMENT FOR LVT VEHICLE

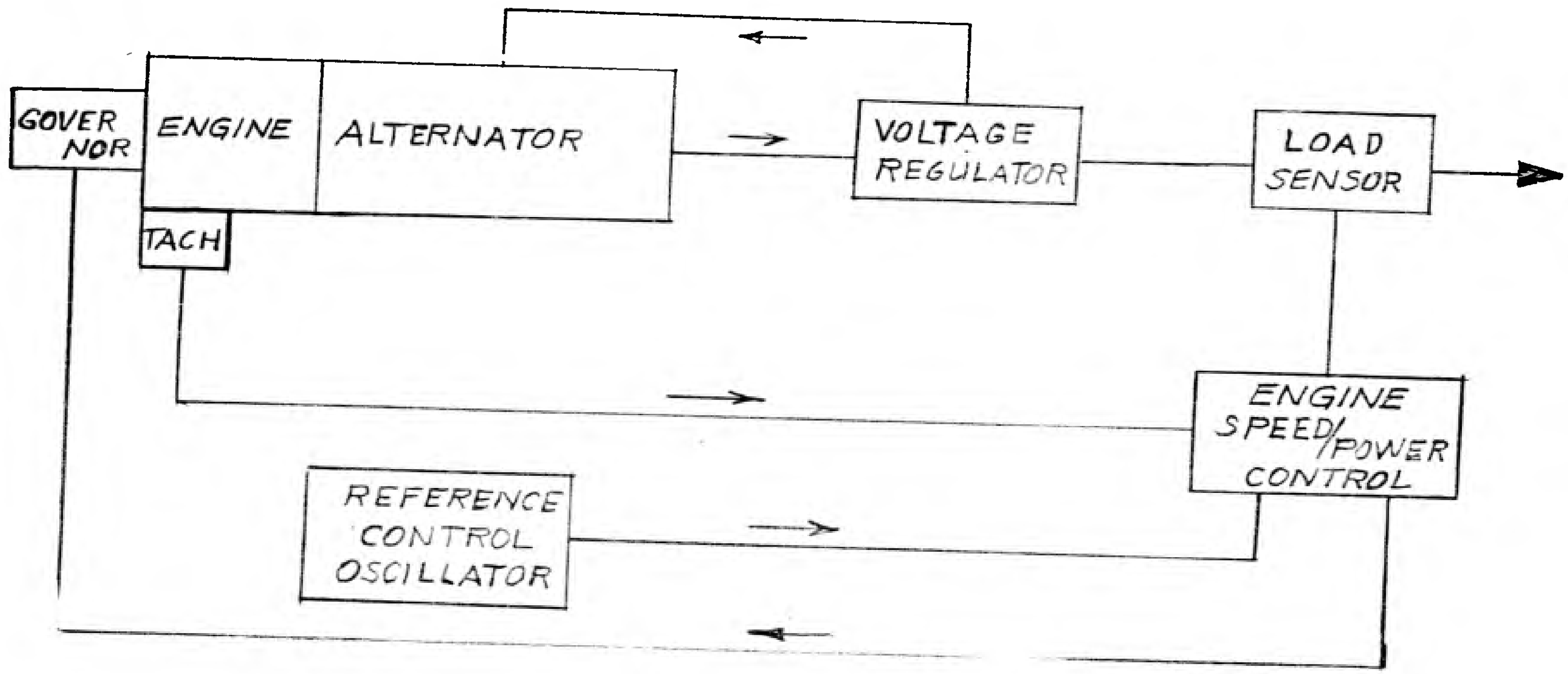
PROJECT 934

MAKE SKETCH. GIVE BRIEF DESCRIPTION, DATE AND SIGN. HAVE WITNESSES DATE AND SIGN.



WITNESS	WITNESS	BY
DATE	DATE	DATE

FIGURE J-6



ENGINE SPEED/POWER CONTROL

FIGURE J-7

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

Engine Speed Control

Because there is no direct link between the engine and the operator, the engine speed control provides automatic throttling of the engine to realize increased economy of operation. In addition, by preventing the engine from overspeeding, when downshifting or while compression-braking downgrade, engine life will be extended. This is accomplished by running the engine at a speed consistent with horsepower required to meet prevailing conditions and maximum fuel economy (see Figure J-7).

In operation, a load sensor constantly senses the hp demanded from the engine. This signal and the engine tachometer signal are compared with a programmed speed/load curve, and the resultant difference and signal are fed to the engine governor. As the vehicle encounters varying terrain, the engine throttle is adjusted to match the load requirements. By monitoring the engine tachometer signal, the engine is prevented from overspeeding.

The same overspeed protection is provided when braking by sensing the maximum allowable engine braking capability, determined by speed sensing, and switching in the braking resistors.

To provide fast response to a speed-change command, another control link from the reference control oscillator provides an anticipator signal.

These engine control features do not require operator participation. Therefore, economical operation as well as longer engine life may be realized regardless of driver experience.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

Arrangement and Component Installation

The SCR power units are placed in the engine compartment on the heavy hull structure, which serves as the heat sink they require. The triggering and control units are also located in the engine compartment.

The driver's controls, located in the driver's compartment, include steering wheel, accelerator pedal, brake pedal (actuating both electrical service brakes and mechanical emergency and parking brakes), and a standard-size instrument panel.

This system can use either a turbine or a reciprocating engine for the power source. With a reciprocating engine, the engine speed is substantially below alternator speed, and a step-up gearbox is required. The amount of this increase influences the size of the alternator, and, through careful design, the smallest, lightest, and most efficient gear/alternator package can be obtained. Through this gearbox, flexibility and positioning of the alternator is obtained. For example, the alternator may be end-mounted to the engine or it may be mounted beside the engine. With the turbine, the speed increaser is eliminated, and the alternator is direct-connected to the turbine shaft.

The drive units include a drive motor, reduction gearing, and mechanical brakes for emergency and for parking, assembled for compactness and simplicity in mounting. The drive sprockets can be located either in the front of the vehicle or in the rear. If a rear drive is deemed advantageous,

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

the drive motor will be located over the sponson, and the power will be transmitted to the final drive by gearing. This flexibility in drive motor positioning insures better distribution of weight.

The cooling fan is electrically driven, and the cooling circuit uses air-to-oil coolers.

Through the ability to either dense-package the drive components or to place them in otherwise unused areas, the electrical drive nets greater usable space in the vehicle and a better distribution of weight.

The eventual use of a fuel cell or a thermonuclear device would eliminate the alternator and add a small inverter package to the frequency converter.

Driver's Controls

The vehicle control system is designed to minimize the demands on the human operator. The logic circuitry (elementary computer techniques) automatically adjusts various controls and other devices to provide maximum performance or maximum economy under varying operating conditions, with only minor decision-making by the operator. This accomplishes two worthwhile objectives by providing better performance and economy and by reducing driver training requirements.

The operator control mechanisms are incidental to the system, and may take the form, for example, of the familiar steering wheel and accelerator

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

and brake pedals. Inasmuch as the control mechanisms operate rheostat-type devices, very little manipulation energy is required, therefore these mechanisms are stiffened to provide the operator with a conventional "feel".

The directional control mechanism consists of a multiposition switch providing FORWARD, NEUTRAL, and REVERSE positions, and the instrument panel appears conventional to the operator.

From a human engineering standpoint, no limitation or difficulties are expected.

Alternator

The alternator is a high-speed unit with a top speed of 20,000 to 30,000 rpm. The final design speed will be determined by the exact power plant speed, in the case of a turbine. The high-speed alternator provides sufficient weight reduction to warrant its use, even with the speed increaser required with a reciprocating engine.

Some of the problems to be resolved for the high-speed alternator (many of which are well on their way to being solved) are as follows:

- Mechanically strengthening the rotor, to withstand high rotational stress.
- Minimizing heat generation by rotor windage.
- Developing high-speed bearings. Turbine manufacturers are advancing in this field.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

- Cooling without weakening structures at critical points.
- Metals which can withstand higher temperatures.
- Electrical insulation which will also conduct heat.

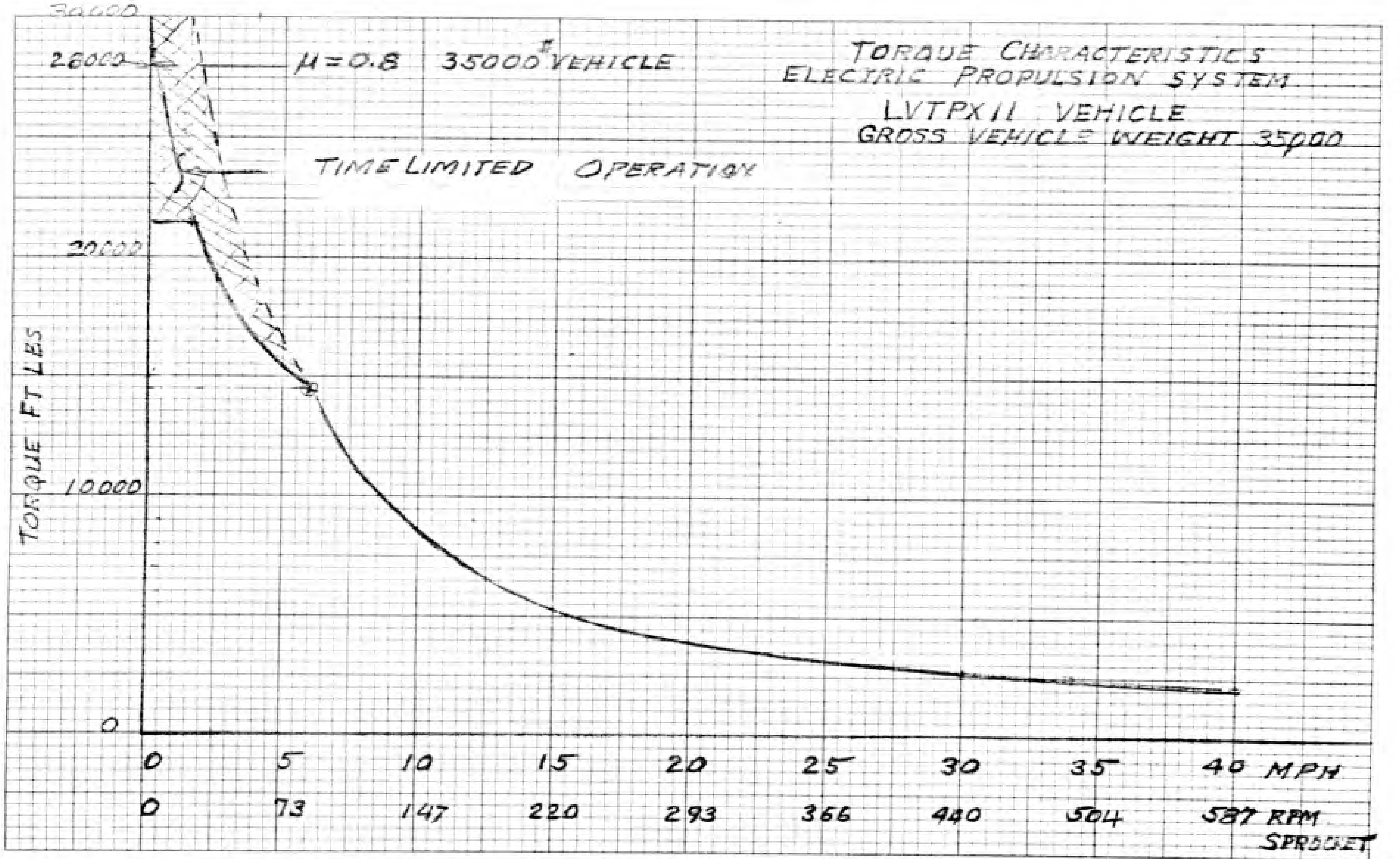
New improvements that can be expected to aid in this development are as follows:

- Better high-strength magnetic materials.
- High-temperature insulation and insulation with greater thermal conductivity.
- Experience in high-speed and high-temperature operation.

Motors

The system motors must operate over a 20-to-1 speed range. Full-power output of 120 hp per motor through the reducer to the sprocket must be supplied over a 6.5 : 1 speed range (6-40 mph).

In order to obtain the best performance in the speed range of the major portion of vehicle operation during the life of the vehicle, the motor will be designed to be most efficient in the range from 6 mph on a 30% slope to 40 mph on the level. Below this speed range, where the full power of



TORQUE CHARACTERISTICS
ELECTRIC PROPULSION SYSTEM
LVTPX11 VEHICLE
GROSS VEHICLE WEIGHT 35000
μ=0.8 35000# VEHICLE
TIME LIMITED OPERATION
TORQUE FT LBS
30000
28000
20000
10000
0
0 5 10 15 20 25 30 35 40 MPH
0 73 147 220 293 366 440 504 587 RPM SPROCKET
9-29-61

FIGURE J-8

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

the engine would not be used, the motor is run at a reduced voltage and overcurrent condition, with added cooling. This cooling is available since the engine load is at reduced horsepower, allowing for greater cooling fan and coolant pump power. In this mode of operation, higher than normal load torques can be obtained for short-time operation (see Figures J-4 and J-8).

For braking and steering, the motors are used as induction generators and must have the capability to transform all of the power from the vehicle into electrical energy to be dissipated in the resistors and, through the alternator, by engine friction drag, for limited periods of time. This is approximately 512 hp peak for each motor (average of 256 hp) for 4.9 seconds, for a complete stop on level terrain (test condition) (see Figure J-6). With ample oil cooling, this braking rate can be continued for longer periods, when braking or stopping on down grades.

The motors are cooled by a flow of oil through passages in the stator cores from a circulating pump, as discussed in the paragraph on Cooling.

The drive motor's maximum speed would be in the range of 20,000 to 30,000 rpm, with speed controlled by frequency variations from the converter.

Currents and voltages will be determined by the most economical design of the converter package.

The use of new magnetic materials will aid in keeping the efficiency high.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

In each case, the optimum operating speeds must be determined by the weight trade-off and efficiency trade-off of the motor and reducer system.

Future improvements in motor efficiency, size, and weight can be expected through the use of better structural and magnetic materials, improved cooling methods (including cryogenics), and additional experience with high-horsepower motors operating over a wide, constant-horsepower, variable-frequency range.

There is little documented experience in this area. One manufacturer provided preliminary information on a new design breakthrough in this area for a lower-weight variable-speed motor operating on a 5 : 1 ratio with constant frequency. The indications are that, if used in combination with a frequency converter, a 20 : 1 torque ratio should be readily achieved.

Gearing

In this system, 3 main gear boxes are required as follows:

Between engine and alternator.

Drive unit reduction

Cooling fan gearbox

The gearing between the engine and alternator consists of a spur-geared step-up transfer case, if the two units are place side-by-side, or possibly

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

planetary gearing if they are end-mounted. With an alternator speed of 20,000 to 30,000 rpm, a ratio up to about 7:1 may be required. For a turbine-powered unit, no gearing is required, since the alternator would be direct-connected to the turbine shaft.

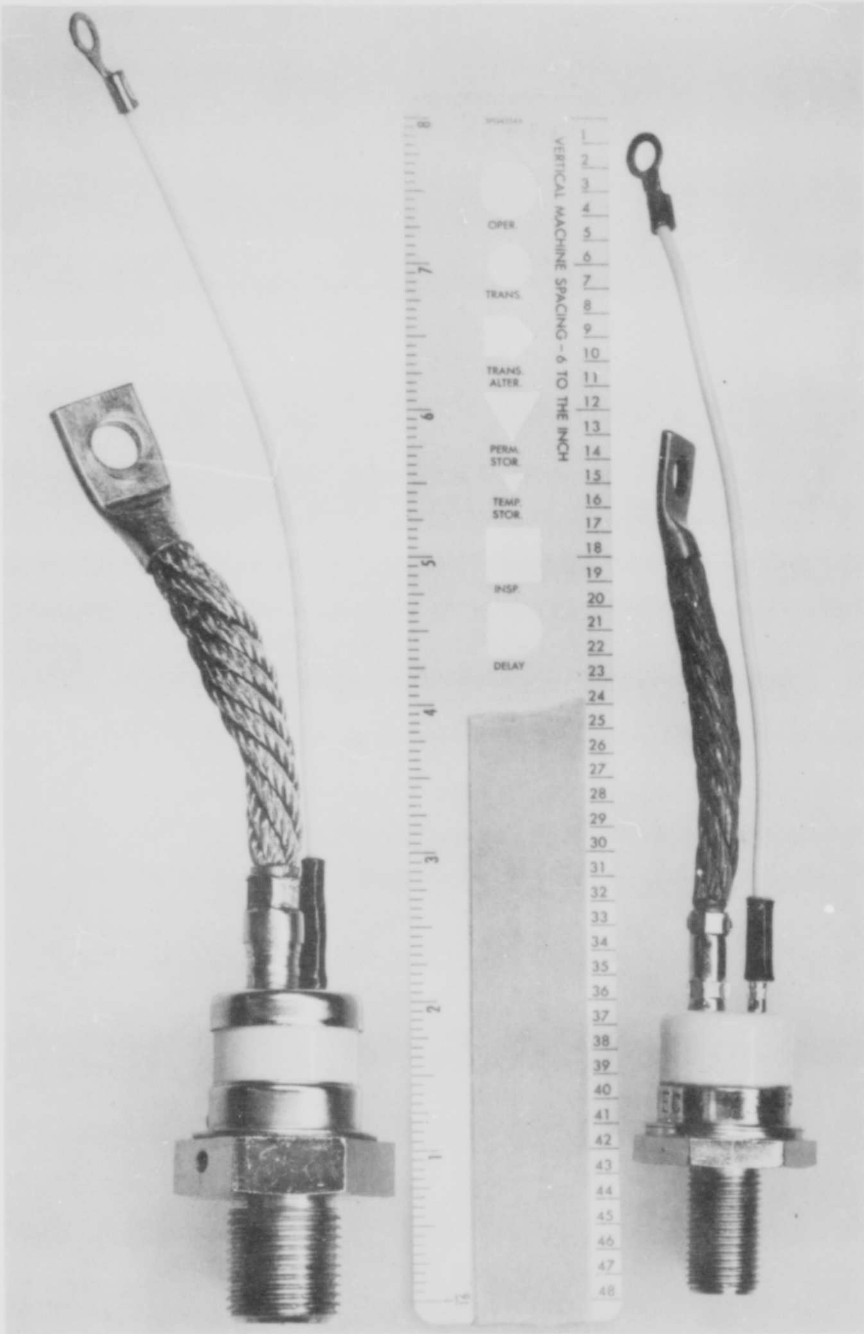
The drive motor-to-sprocket reduction gearing ratio will be in the range of 50 : 1. Special attention must be given to lubrication, and separate stages of lubrication may be required throughout the reduction. The design of the gear-drive motor unit should be done in a single effort, to provide the lightest, most efficient, and most compact system. It may also be possible to incorporate the drive motor and reduction gearing into the sprocket carrier, netting a further saving of usable space. This assembly would also contain disc-type mechanical brakes for parking and emergency service.

In all gearing, consideration will be given to the elimination of fatigue stressing, the assurance of surface durability by reducing scoring and pitting, and protection against road shock or track-engagement vibration.

The cooling fan drive will incorporate an integral fan, gear, and motor assembly, providing an easily located and controlled cooling unit.

Static Frequency Converter

With the recent rapid development of the SCR and its capability of providing efficient, compact, frequency conversion, several electrical companies moved quickly into this field (see Figure J-9).



100 AMP 50 AMP

SILICON-CONTROLLED RECTIFIERS FIGURE J-9

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

The first applications were in electrical generating systems to obtain a constant output frequency with a varying input frequency (VSCF), rather than for variable-frequency output systems required for vehicles and other variable-speed drives; however, the basic converter theory used in VSCF systems is very similar to that required for variable-frequency output applications.

The static frequency converter is the device which only recently has allowed a practical approach to a variable-speed electric drive system. Although the design idea is not new, previous components to accomplish this were large and inefficient. With the advent of the silicon-controlled rectifier (SCR), new vistas of power control were opened.

Although the SCR is a comparatively new device, it is already developing a record of successful applications and a truly amazing reliability, when used within its ratings. Its life expectancy can only be estimated, but it is already apparent that it should outlast devices which it is replacing, by many magnitudes. The electrical companies who have assisted FMC in this proposal have successfully used these rectifiers in similar applications.

The converter consists of SCR's operated in a manner similar to the "cyclo-converter", which required large gas thyratrons, used in Europe some thirty years ago.

TYPICAL SCR SWITCHING CIRCUIT

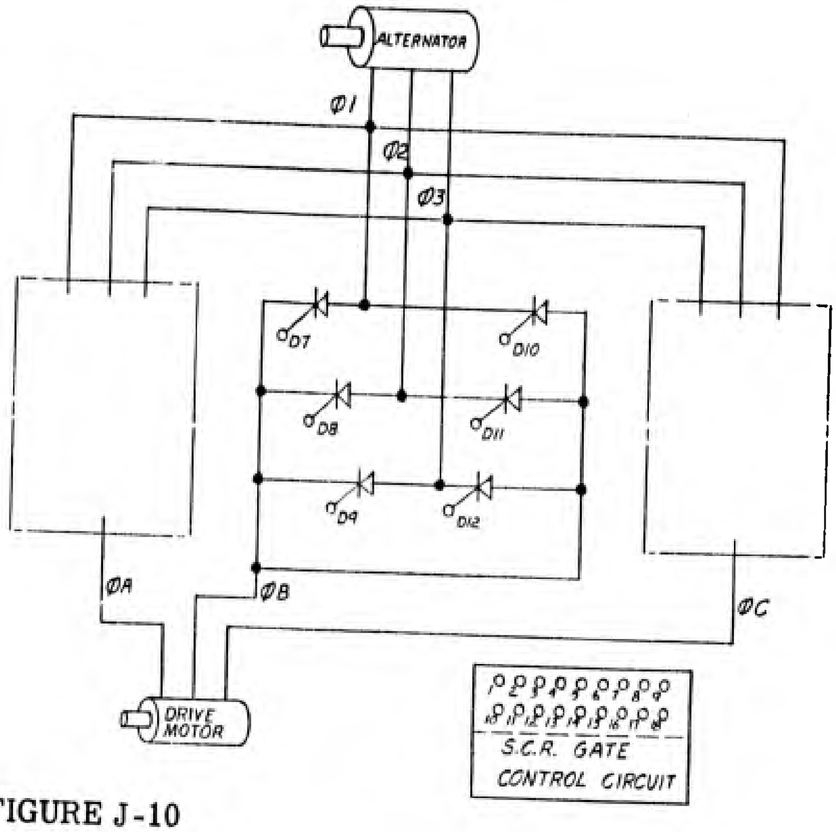


FIGURE J-10

FREQUENCY CONVERTER WAVE FORMS

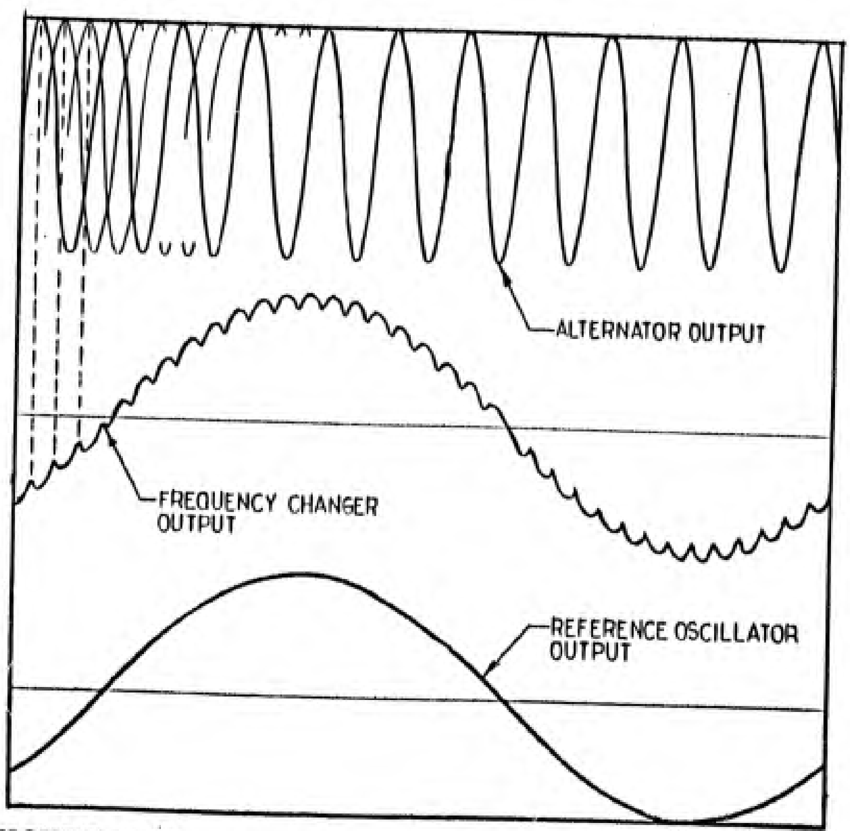


FIGURE J-11

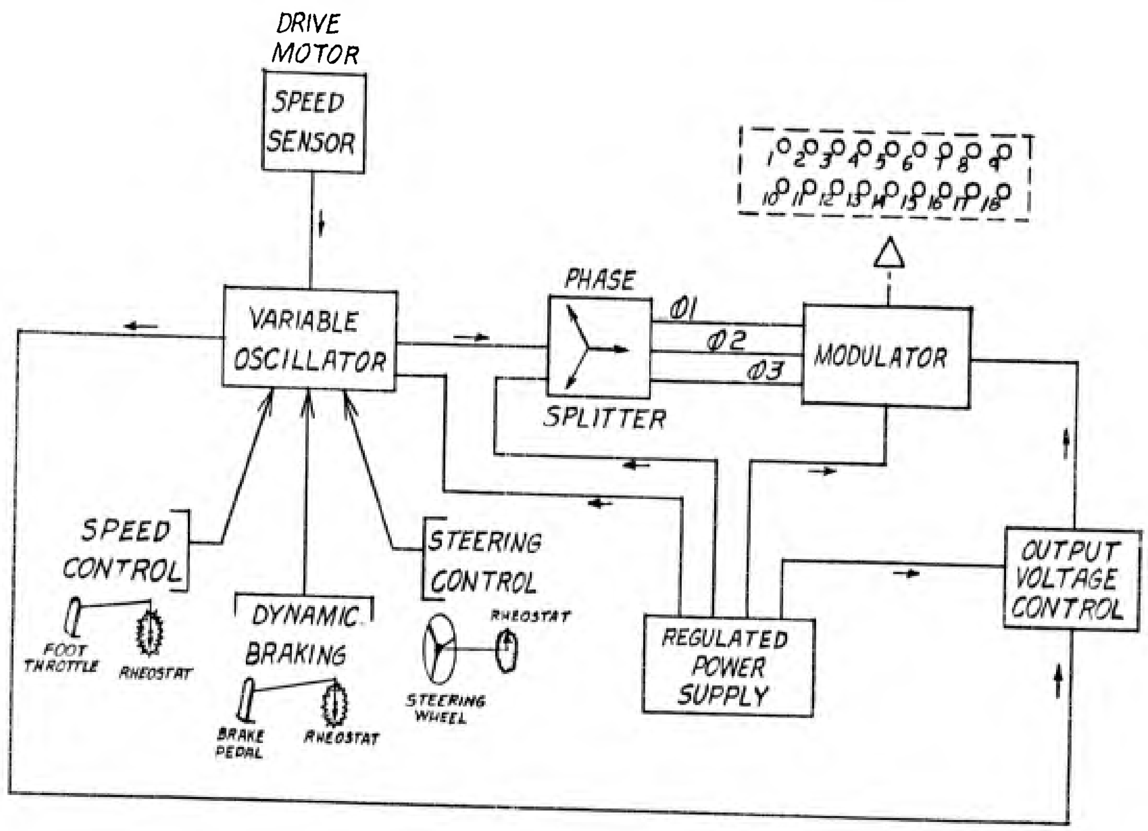
ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

In operation, two banks of SCR's are utilized in such a fashion as to provide a positive and negative sequence of pulses (see Figure J-11). By proper firing or gating control, these pulses are fashioned into a near-sine wave, as in Figure J-11. The output load in this system will accept this waveform with no ill effects; therefore, waveform filtering is not required. The drive motor, having considerable inductance, provides its own filtering. The rate at which the positive and negative banks of SCR's alternate is determined by a control or reference oscillator signal. Varying the frequency of this reference oscillator will vary the output frequency of the converter. The output frequency is theoretically synchronized with, or identical to, the reference oscillator. This theoretical similarity is maintained, independent of the main input-power frequency, with the following practical limitations:

- There is a practical input-to-output frequency division ratio for efficient power transfer. This limit appears to be approximately 3 : 1.

- The SCR's have a maximum upper frequency limit, where switching time limits power transfer. This is caused by the time required to sweep out the junction carriers, and is much higher than the requirement outlined here.

The output voltage of the converter may be varied by advancing or retarding the firing angles of the SCR's. This allows independent voltage and frequency control, to take advantage of the optimum combination of drive motor characteristics.



SCR GATE CONTROL CIRCUIT

FIGURE J-12

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

SCR's have a maximum current rating that is independent of voltage (i.e., an SCR can handle its maximum current at any voltage within its maximum voltage rating). Thus, since power is equal to the product of current and voltage, it becomes obvious that an effort should be made to keep the voltage as high as possible throughout the full constant-horsepower/variable-frequency range, for maximum power-handling capabilities.

The best selection of voltage and current characteristics for the drive motors imposes difficult operating conditions on presently available SCR's, when used singly. To overcome this, the SCR's are operated in a series-parallel arrangement: series for peak voltage protection, and parallel for increased current capabilities. This stacking-up of SCR's adds some circuitry, for series equalization and triggering synchronization, and paralleling requires matching, for equal load division.

The vehicle drive system will require two separate converters and associated circuitry. The converters, by employing separate reference oscillators, can operate simultaneously, for constant-speed conditions, or individually, to allow braking and steering (see Figure J-12).

For the most efficient transfer, the power flow through the converter is from the high-frequency source to a low-frequency output load. However, this flow direction can be inverted if required, where load is reactive and/or capable of energy storage. This inherent characteristic permits bilateral flow of power at any power factor.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

By utilizing all three phases from the alternator output to fabricate each of the three output phases (see Figure J-12). any system load unbalance will be distributed over all the phases of the high-frequency alternator output.

Also under consideration is a six-phase alternator which would allow a better output waveform to be fabricated, by allowing the SCR's to build the waveform with twice as many pulses. This would also effectively parallel SCR's without the need for matching SCR's. Some penalty would be paid in increased control circuitry; however, the gain in reliability and increased ripple frequency offset the small size and weight increase.

The converter is small, lightweight, and efficient. All associated control and logic circuits are transistorized, modular, plug-in types. This type of construction lends itself to faster field service as well as a lower technical qualification requirement. By utilizing this approach, where many circuits are identical (this includes both converters), replacement and spare parts problems are greatly reduced.

It should be pointed out, however, that reliability will be greatly enhanced by the use of solid-state design.

The control and logic circuitry designed to operate these SCR's is sufficient to drive larger rectifiers when they become available. Thus, to update a converter for use with larger SCR's as they become available will require minimum modification of the converter.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

These rectifiers, by using silicon junctions, operate efficiently at a much higher ambient temperature than their cousins of the germanium family. This feature, coupled with the excellent heat-sink capability of the vehicle's aluminum hull, eliminates any special cooling equipment.

The power conversion efficiencies of the static frequency converter range from 97 to 98%, with the losses limited to the small power requirements of the control and logic element and the forward conduction losses of the SCR's. These forward conduction losses are due to the inherent voltage drop of the SCR junction, which is a function of the energy required to raise the current carriers from the low-energy state to the conduction band and is unaffected by external influences.

The basic design criterion for converter power sizing is a function of the power required for regeneration to the outside track, in executing a high-speed shart turn.

Future Outlook

Inasmuch as the price, volume, weight, and output power are functions of the SCR's capabilities, a look at the state-of-the art in the SCR field is in order.

The silicon-controlled rectifier art is in its infancy, but the strides in this industry should result in improved, more economical components. As the rated capacities of the SCR's are increased, the number of rectifiers required for a given task is reduced. This simplifies the static frequency converter circuitry and reduces the size and weight of this component.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

At present, 150-ampere units, with PRV ratings to 500 volts, are available, and 300-ampere 600-volt units should become hardware realities before the end of 1962.

Converter Problem Areas

The primary problem areas are thermal design, overload protection, transient-voltage protection, radio noise, maximum current ratings associated with acceleration and deceleration, and packaging.

- Thermal Design - The effectiveness with which the SCR elements can be cooled determines the size and weight of the package. Liquid cooling would be the most efficient manner, although FMC is quite confident that hull-mounting will provide an effective means of removing the heat from the rectifier. The maximum temperature of the SCR's must not exceed 100°C.

- Overload Protection - Since the SCR elements contain only a tiny thermal mass, relative to their thermal time constant, they are not capable of withstanding large overloads for longer periods than a few seconds. Longer overloads must be designed into the ratings of the converter, and steps must be taken to assure proper operation within the ratings.

- Transient-Voltage Protection - When SCR's are switching inductive loads, they are subjected to high voltage transients. Protection devices must be designed into the converter to protect the rectifiers from this destructive spike.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

- Radio Noise - The switching phenomenon which is inherent in SCR firing produces noise in the frequency range of 5 kilocycles and up. This requires appropriate shielding and radio-suppression filter design in the converter package.

- Acceleration and Deceleration - It is most efficient to accelerate in a constant-torque mode without high inrush current. This means of motor starting should be considered in view of converter SCR sizes and motor loss reductions.

FMC feels that the selected system reduces these problems to that of design, and no major developmental areas are anticipated.

Another problem area, not part of the converter but closely associated, is that of the many control and feedback devices and circuits. Although neither this control system nor its components exceed the state-of-the-art, considerable design work will be required to intregrate the over-all system.

Evaluation of the Various Design Proposals

In judging the relative merits of the converter systems proposed by the electric companies, FMC has reached the following conclusions:

- There are many successful approaches to the solid-state converter design employing varying degrees of complexity

- All result in the same basic operation

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

The preferred system was chosen for the following reasons:

- Considerably higher conversion efficiency, resulting from more efficient logic and gating control
- Increased reliability and stability by incorporating digital pulse techniques. This also allows more precise control of frequency, when providing universal power, as well as easing synchronization of the two control oscillators.
- In considering the converter and drive motor as a welded combination, their frequency-to-voltage relationship takes advantage of the maximum current-to-voltage relationship of the SCR.

These advantages tend to make this system closest to an optimum approach.

Cooling

The maximum calculated heat rejection of the electrical system, including gearing, is 3,190 btu/min. The heat is removed from components by oil-to-air heat exchangers, by airflow from a cooling fan over the housings of the components, and by heat dissipation to the hull.

- The alternator is cooled by oil forced through passages in the stator at the rate of about 45 gpm.

ELECTRIC PROPULSION SYSTEM FOR LVTPX11 (Continued)

- The motors and gear reducers are cooled by oil circulation through the stators and gear cases. Cooling under overload conditions such as slow-speed operation is obtained by increasing the oil flow.

- The SCR's are cooled by mounting on the aluminum hull, using this for a heat sink.

- The heat from the final drive is dissipated into the hull of the vehicle.

- The braking resistors are cooled by the air blown over the grids.

The cooling fan for the radiator heat exchanger is electric-motor driven, with speed control operating at maximum airflow only when the heat rises sufficiently to require full cooling. The cooling-oil flow is also controlled by the heat-rejection demand. This provides additional horsepower to the sprockets under cool ambient conditions.

HYDRO-MECHANICAL
STEERING & POWER TRAIN SYSTEM
FOR LVT PX 11 VEHICLE

F M C
CORPORATION

OCTOBER 27, 1961

ACCESSORY POWER
GENERAL ELECTRIC
AIRCRAFT ACCESSORY TURBINE DEPARTMENT
LYNN, MASSACHUSETTS

A AT 432 C

TABLE OF CONTENTS

INTRODUCTION

Tactical and logistical requirements for tracked vehicles are calling for increased vehicle speeds, torques, and efficiencies as well as lighter weights and lower silhouettes.

The differential hydro-mechanical transmission offers the improvements to meet these requirements as well as offering a smooth steering control and an infinitely variable speed-torque ratio.

The General Electric Company is pleased to offer the Model 7T-ALH10-A01 differential hydro-mechanical transmission to Food Machinery Corporation for use on an improved universal LVT vehicle.

The General Electric differential hydro-mechanical transmission offers:

High efficiencies over a wide speed range

High component efficiencies coupled with an optimum design arrangement give efficiencies impossible with other simple design arrangements.

Smooth steering control at all speeds, with built-in operator "feel"

Inherent characteristics provide this without the use of brakes or clutches.

Regenerative steering

Required vehicle speed and flexibility is maintained during steering operations with reduced unit size and heat rejection.

Control of engine for minimum fuel consumption.

Control of the engine is integrated into the transmission control to insure optimum vehicle efficiency under various engine and vehicle conditions.

Long life and high reliability.

The mechanical simplicity and conservatism of the ball piston design means inherent high reliability and long life. One million hours of operation have confirmed this.

Compactness

The features of the ball piston design - high speed, short stroke, no connecting rods, crankshafts or bearings and radial porting - result in a small diameter, short length transmission.

Lightweight design.

The same ball piston unit characteristics which make for compactness result in an extremely lightweight design.

Versatility

The ball piston unit has the capability of operating with any hydraulic fluid as well as being applied to a wide range of loads -- 1 HP to 500 HP -- and speeds -- several hundred RPM to 25,000 RPM.

DESIGN

General Electric Specification Data Sheet
for
Hydro-mechanical Differential Steering Transmission
Model 7T-ALH10-A01

Speed Range: Input - 2400 RPM max. (or as specified)

Output - 10 MPH reverse to 40 MPH forward

Infinitely variable stepless ratio over complete range from maximum reverse speed to maximum forward speed.

Overall Efficiency:

Vehicle Speed MPH	(4) Engine Power HP	(4) Engine Speed RPM	(3) HP @ Sprocket HP	Trans-mission Heat Rejection Total, HP	(1) Trans-mission Heat Rejection to Oil, HP	(2) Final Drive Heat Rejection HP
2	263	2400	138	120.7	96.6	4.3
2-1/2	263	2400	153	105	84	4.7
15	263	1000	224	32.1	25.7	6.9
40	263	2400	210	46.5	37.1	6.5

(1) It is assumed that the heat rejection to the oil will be approximately 80% of the total heat rejection. The difference is dissipated by convection, radiation and conduction.

(2) Final drive efficiency assumed to be 97%.

(3) Sprocket speed is 640 RPM at 40 MPH vehicle speed.

(4) Max. engine power at 2400 RPM is 263 HP.

Outline Drawing:	1076511-195
Weight:	Dry - 779 lbs. Wet - 834 lbs. (no final drive)
Steering:	By single lever or wheel. Smooth control without use of mechanical brakes or clutches. Regenerative steering inherent with capabilities of 1200 HP applied to a single sprocket.
Braking:	Dynamic within capability of engine to absorb retarding torque. Service brakes for each sprocket - built into transmission.
Controls:	Single mechanical-hydraulic speed control for integrating the transmission and overall vehicle system. Selector lever for forward-neutral-reverse. Emergency braking by single pedal. Parking brake by single lever. Selector lever for high speed towing mode.
Ambient Temperature:	$125^{o}F$ maximum vehicle ambient temperature. $160^{o}F$ maximum engine compartment temperature.
Oil Temperature Range:	$-65^{o}F$ to $325^{o}F$ or within limits of oils - MIL-F-2104 and MIL-O-10295.
Final Drive:	4.25:1 ratio drive to be furnished by vehicle manufacturer.
Oil System:	Reservoir, make-up pumps, filter, valves, lines, integral with transmission. (Cooler to be furnished by vehicle manufacturer.)
Life:	In excess of 1000 hours.
Attitude:	Will operate normally with 70% forward or backward slope and 60% side slope.

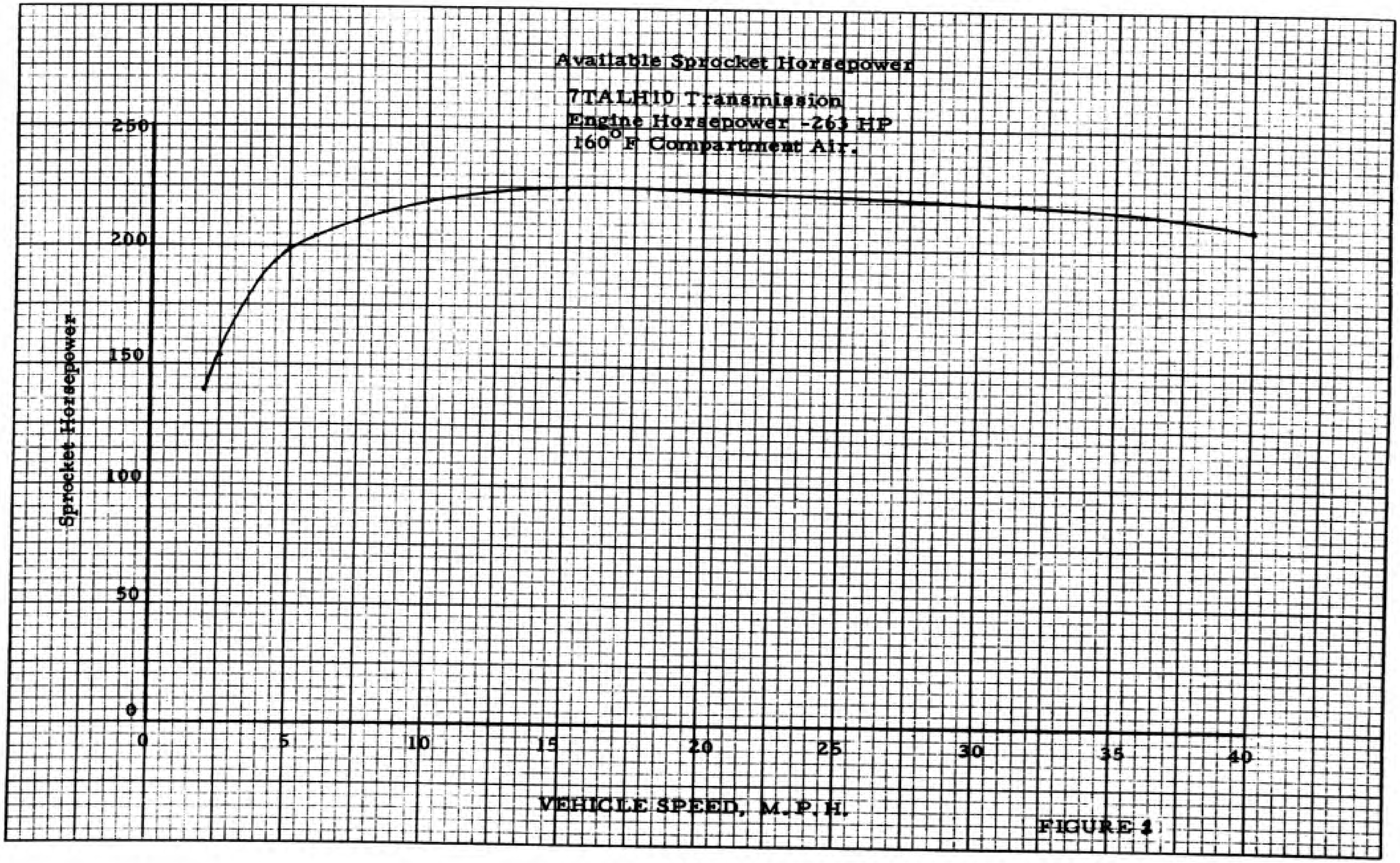
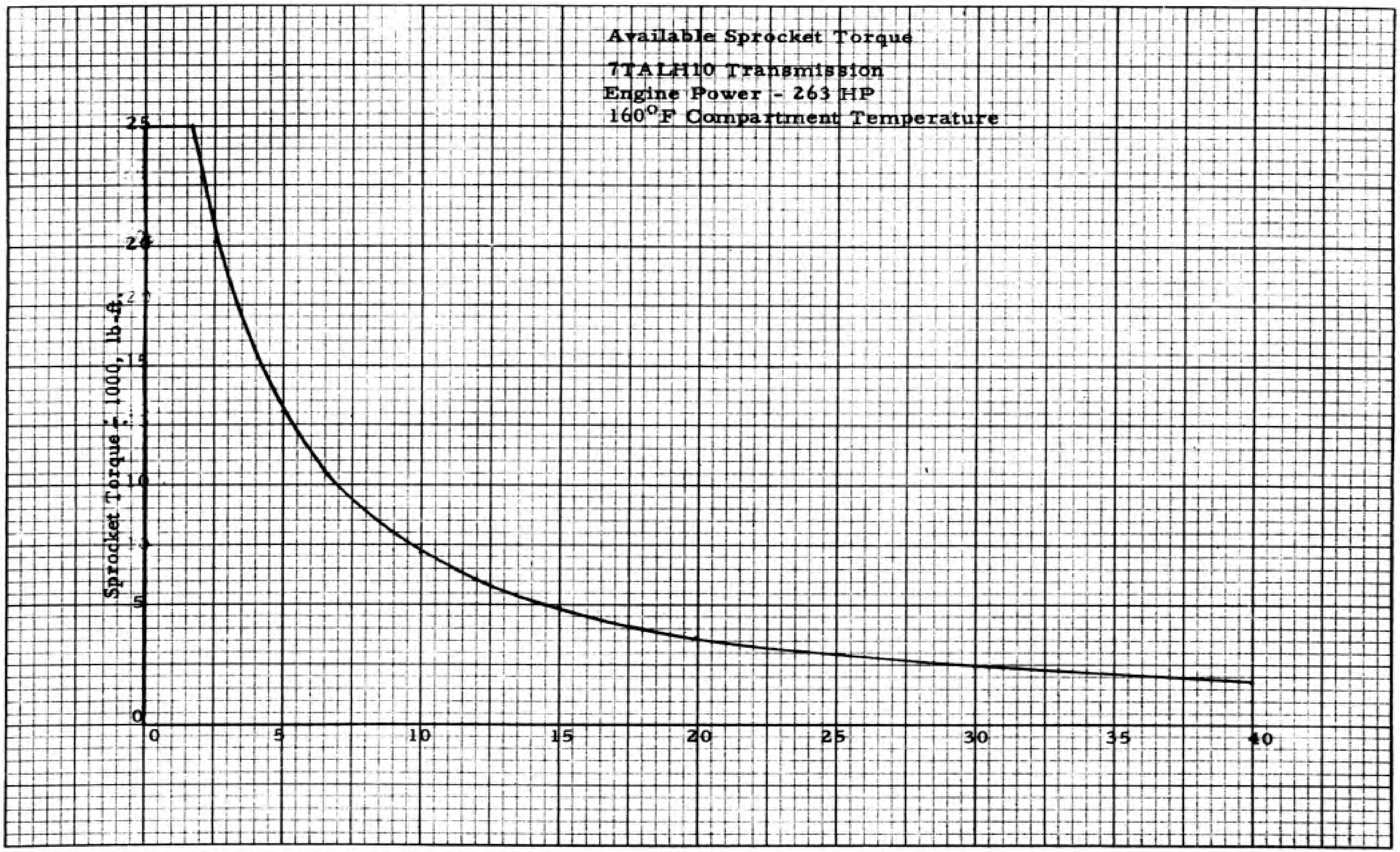
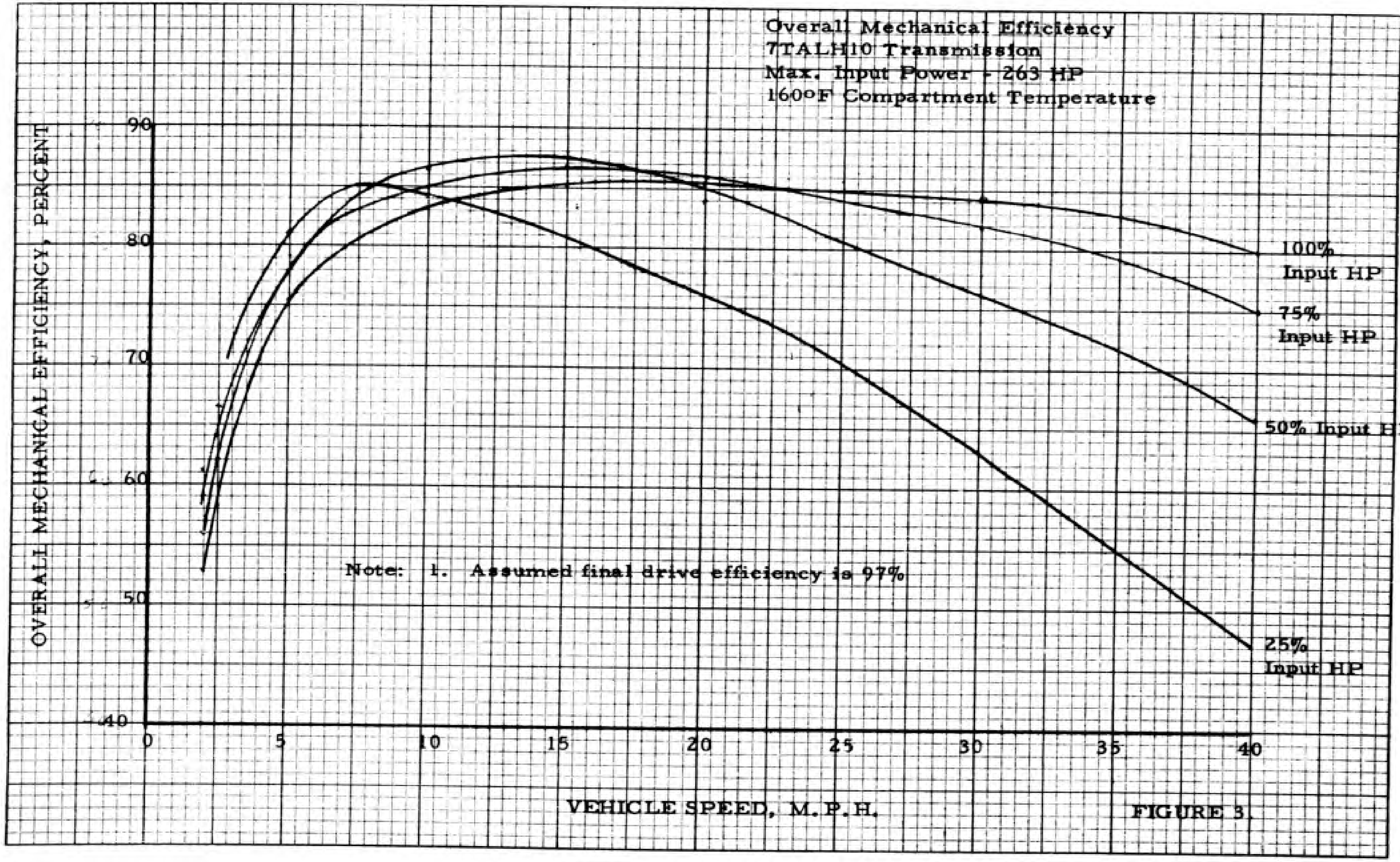

Available Sprocket Horsepower
7TALH10 Transmission
Engine Horsepower -263 HP
160° F Compartment Air.
Sprocket Horsepower
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
VEHICLE SPEED, M.P.H.

FIGURE 1



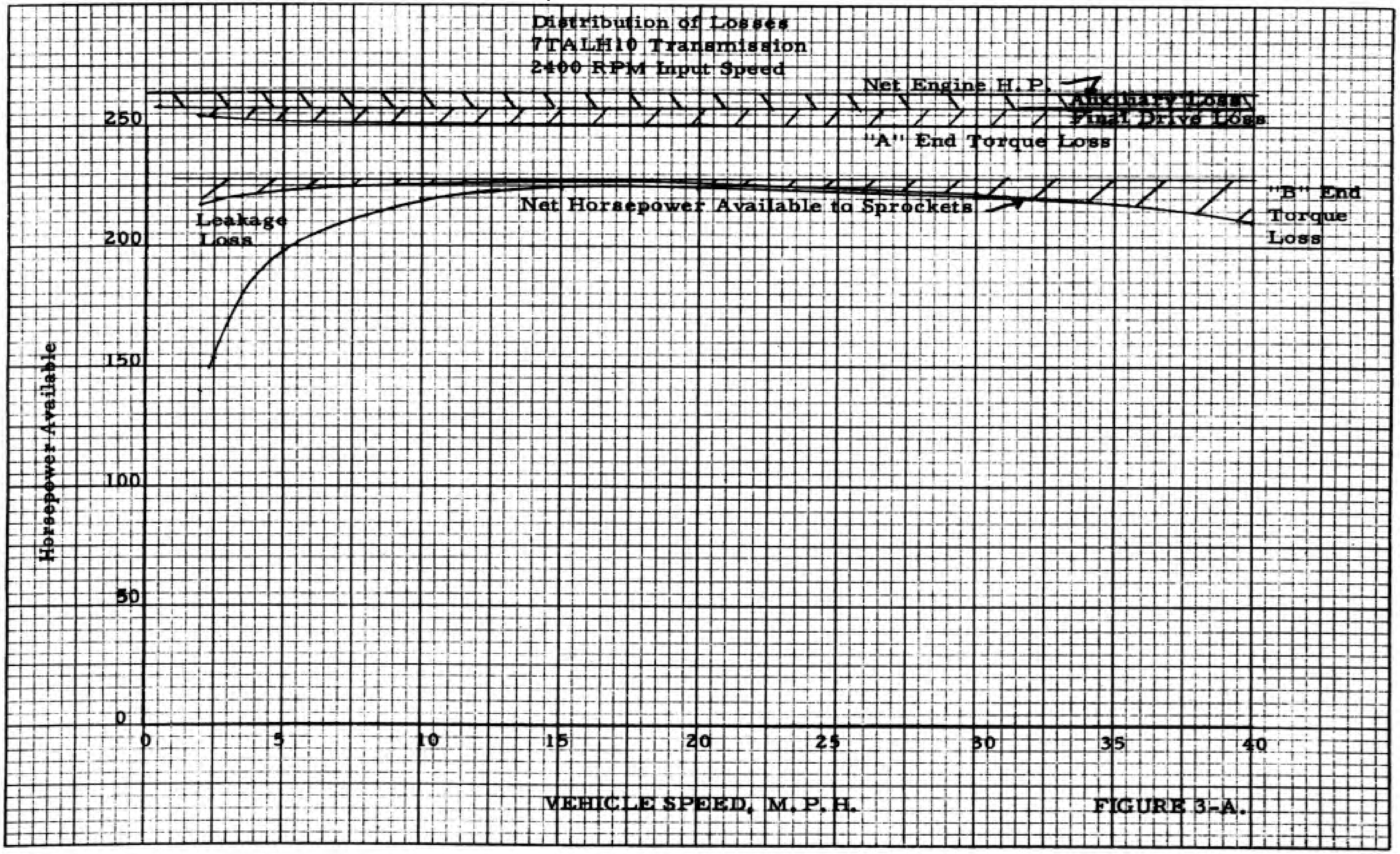
VEHICLE SPEED, M. P. H.

FIGURE 2.



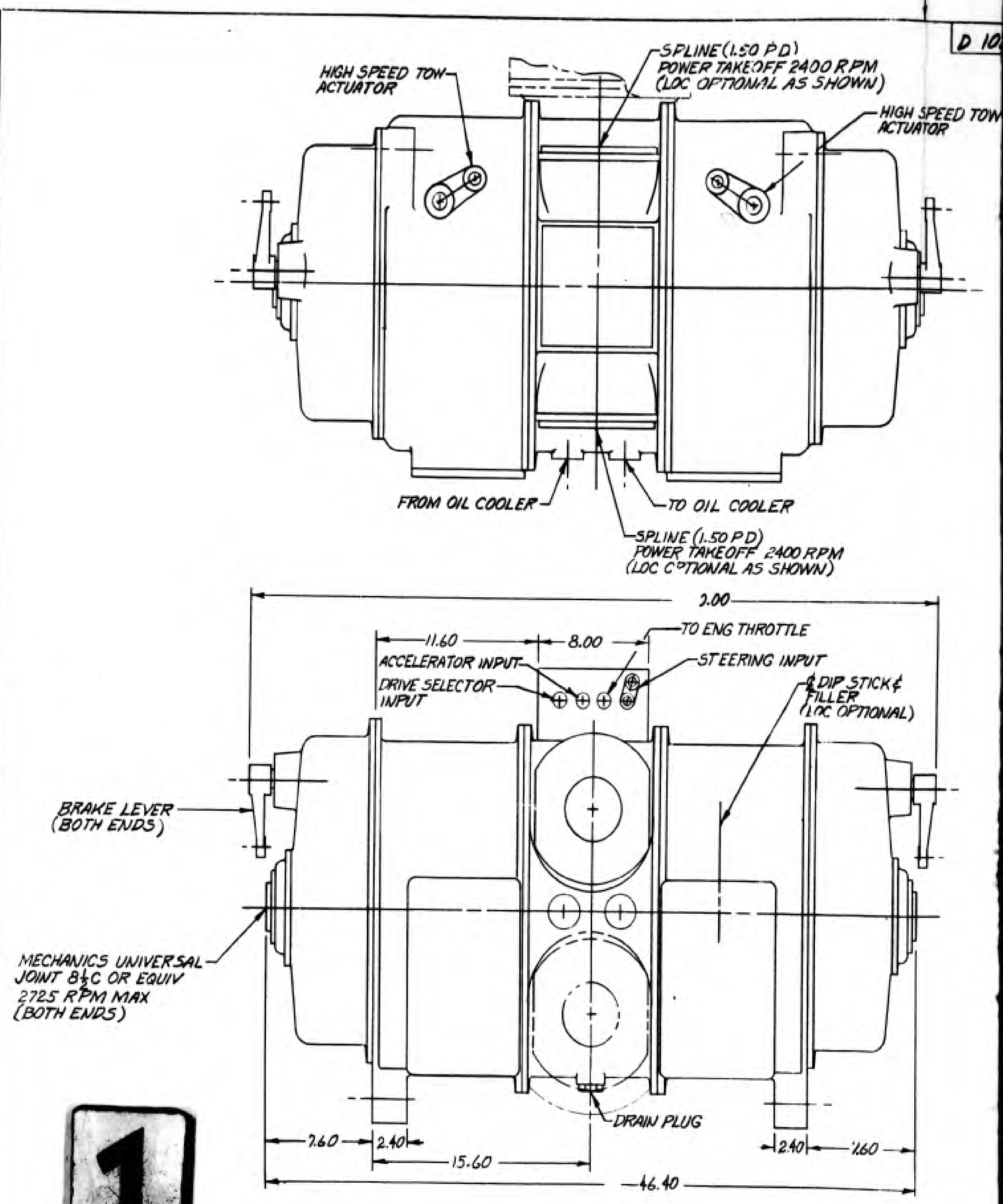
Overall Mechanical Efficiency
7TALH10 Transmission
Max. Input Power - 263 HP
160°F Compartment Temperature
OVERALL MECHANICAL EFFICIENCY, PERCENT
90
80
70
60
50
40
0
5
10
15
20
25
30
35
40
100% Input HP
75% Input HP
50% Input HP
25% Input HP
Note: 1. Assumed final drive efficiency is 97%
VEHICLE SPEED, M.P.H.

FIGURE 3



Distribution of Losses
7TALH10 Transmission
2400 RPM Input Speed
Net Engine H.P.
Auxiliary Loss
Final Drive Loss
"A" End Torque Loss
Net Horsepower Available to Sprockets
"B" End Torque Loss
Leakage Loss
Horsepower Available
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
VEHICLE SPEED, M.P.H.

FIGURE 3-A.



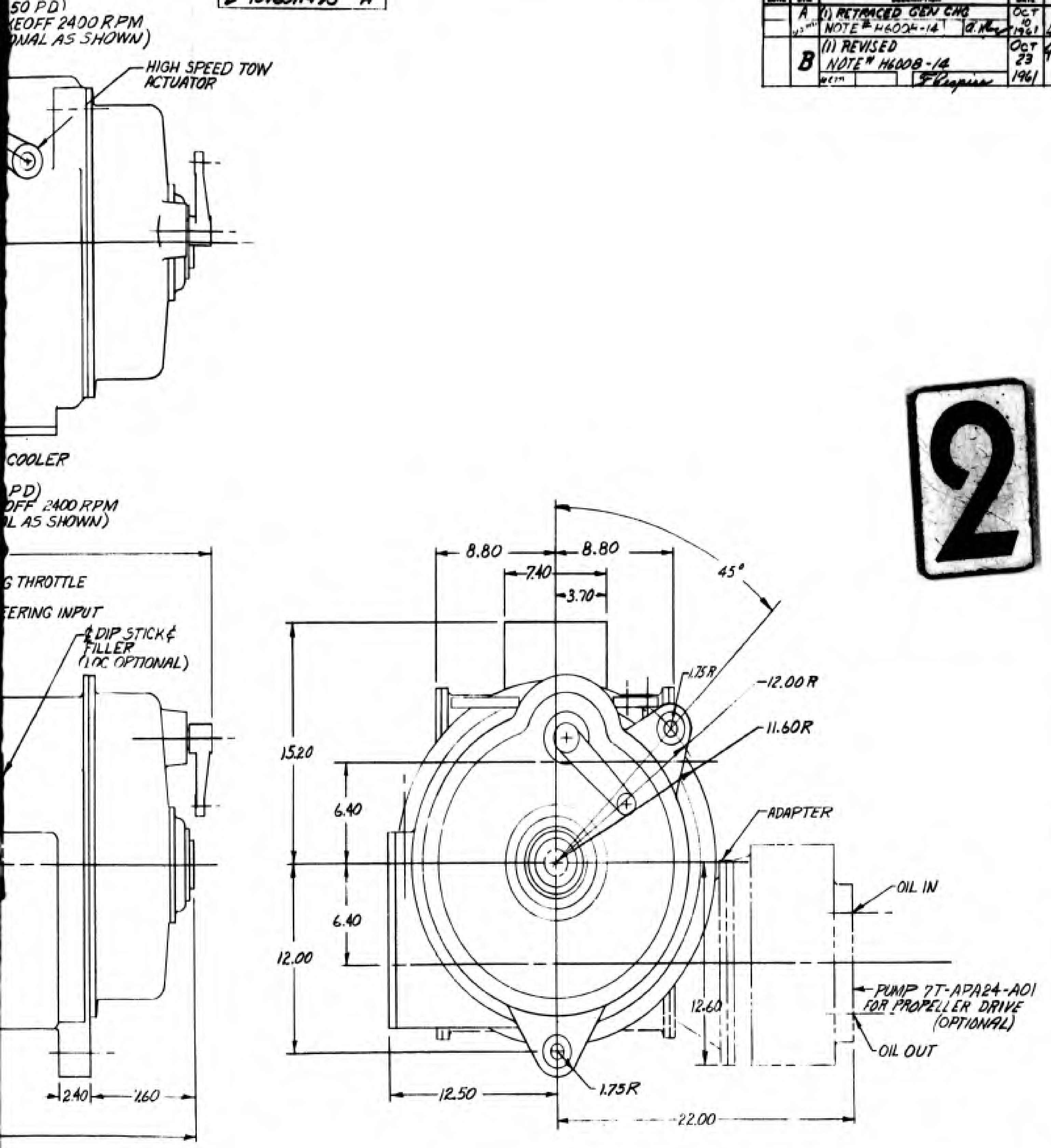
D 10
HIGH SPEED TOW-ACTUATOR
SPLINE (1.50 P D)
POWER TAKEOFF 2400 RPM
(LOC OPTIONAL AS SHOWN)
HIGH SPEED TOW ACTUATOR
FROM OIL COOLER
TO OIL COOLER
SPLINE (1.50 P D)
POWER TAKEOFF 2400 RPM
(LOC OPTIONAL AS SHOWN)
11.60
8.00
TO ENG THROTTLE
ACCELERATOR INPUT
STEERING INPUT
DRIVE SELECTOR INPUT
DIP STICK & FILLER
(LOC OPTIONAL)
BRAKE LEVER
(BOTH ENDS)
MECHANICS UNIVERSAL JOINT 8½C OR EQUIV
2725 RPM MAX
(BOTH ENDS)
DRAIN PLUG
7.60
2.40
15.60
2.40
7.60
46.40

1

D 1076511-195 A

ZONE	SYM	DESCRIPTION	DATE	APPROVAL
	A	(1) RETRACED GEN CHG NOTE # H6004-14	OCT 10 1961	
	B	(1) REVISED NOTE # H6008-14	OCT 23 1961	

2



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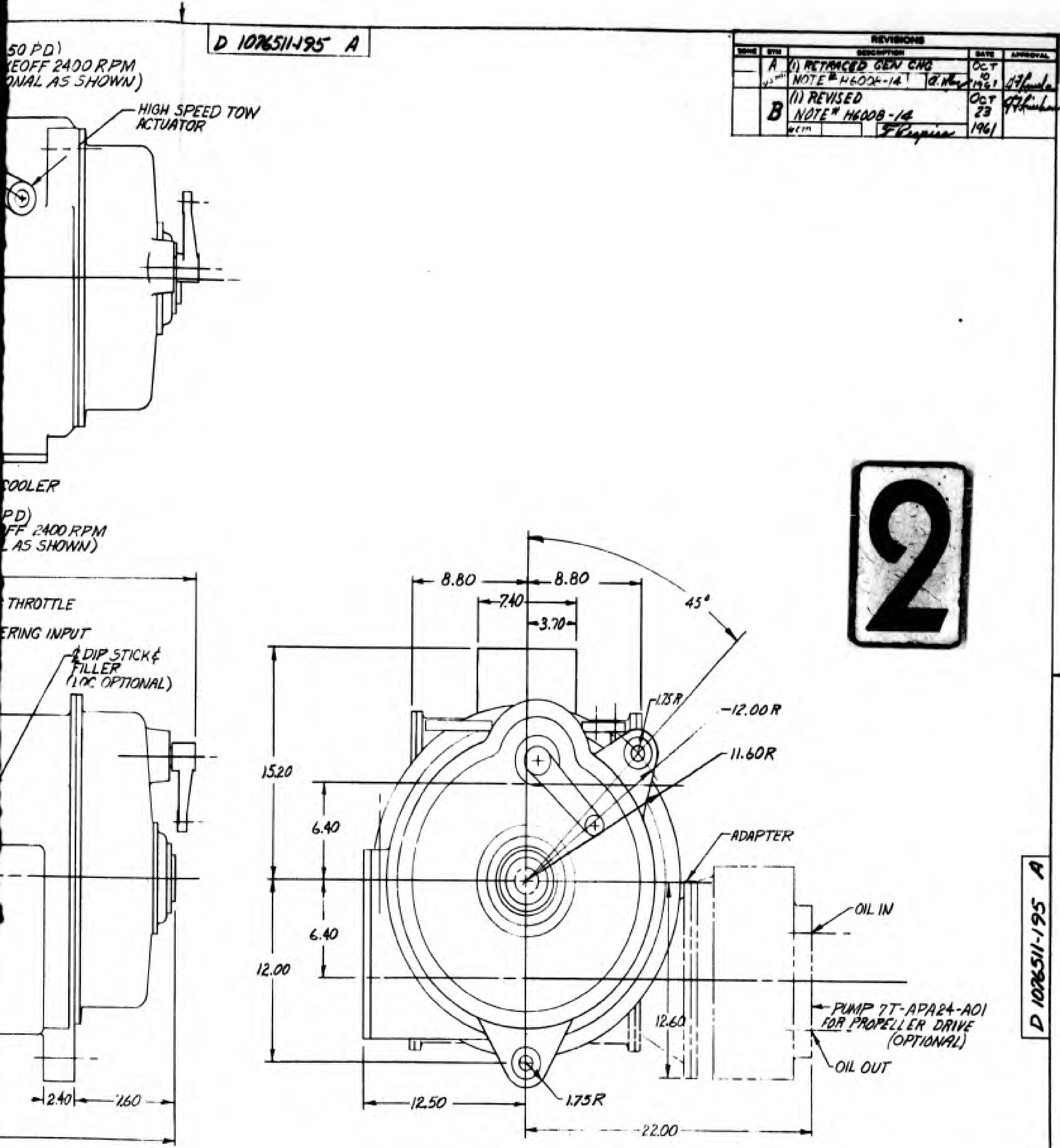
859

DRAWN P. FRASSICA DATE SEPT 21 61 APPD ISSUED SEPT 1 1961 ENGR E.E.L. 9-21-61 MFG MATL	OUTLINE - 300 HP HYDRO-MECHANICAL STEERING TRANSMISSION H6008-14 SCALE / WT	GENERAL ELECTRIC AGT LYNN RIVER D 1076511-195 CONT ON SHEET / SH NO.

CODE IDENT → 99611



REVISIONS

ZONE	SYM	DESCRIPTION	DATE	APPROVAL
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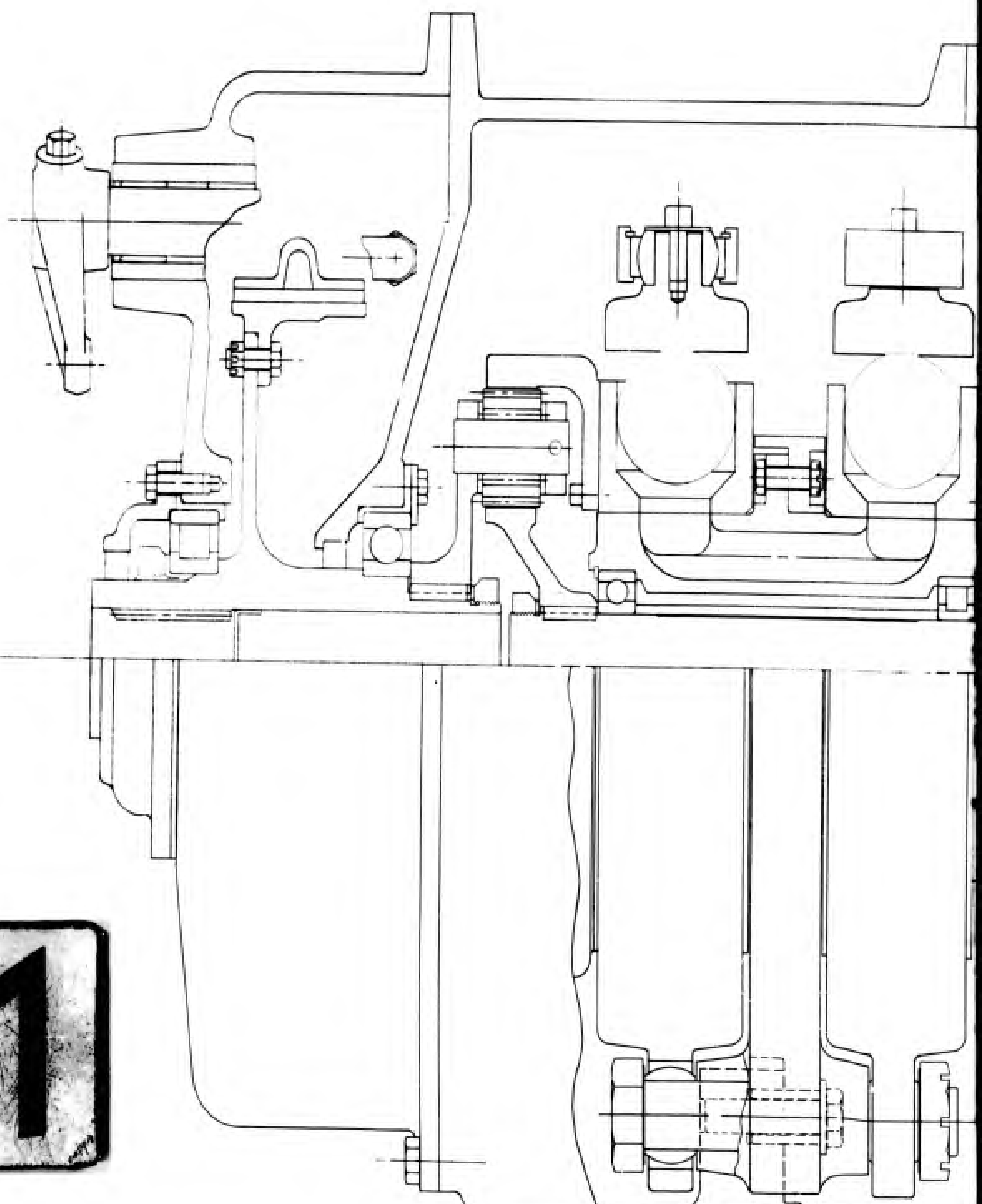
	B	(1) REVISED NOTE # H6008-14	OCT 23 1961	

40V 859

DRAWN P. FRABASCA DATE SEPT. 21 61 APPD ISSUED SEPT 21 1961 ENGR E.E.L. 9-21-61 MFG MATL	OUTLINE - 300 HP HYDRO-MECHANICAL STEERING TRANSMISSION H6008-14 SCALE WT	GENERAL ELECTRIC A.A.T. (DIVISION) LYNN RIVER (LOCATION) SIZE D 1076511-195 CONT ON SHEET SH NO.

CODE IDENT → 99611



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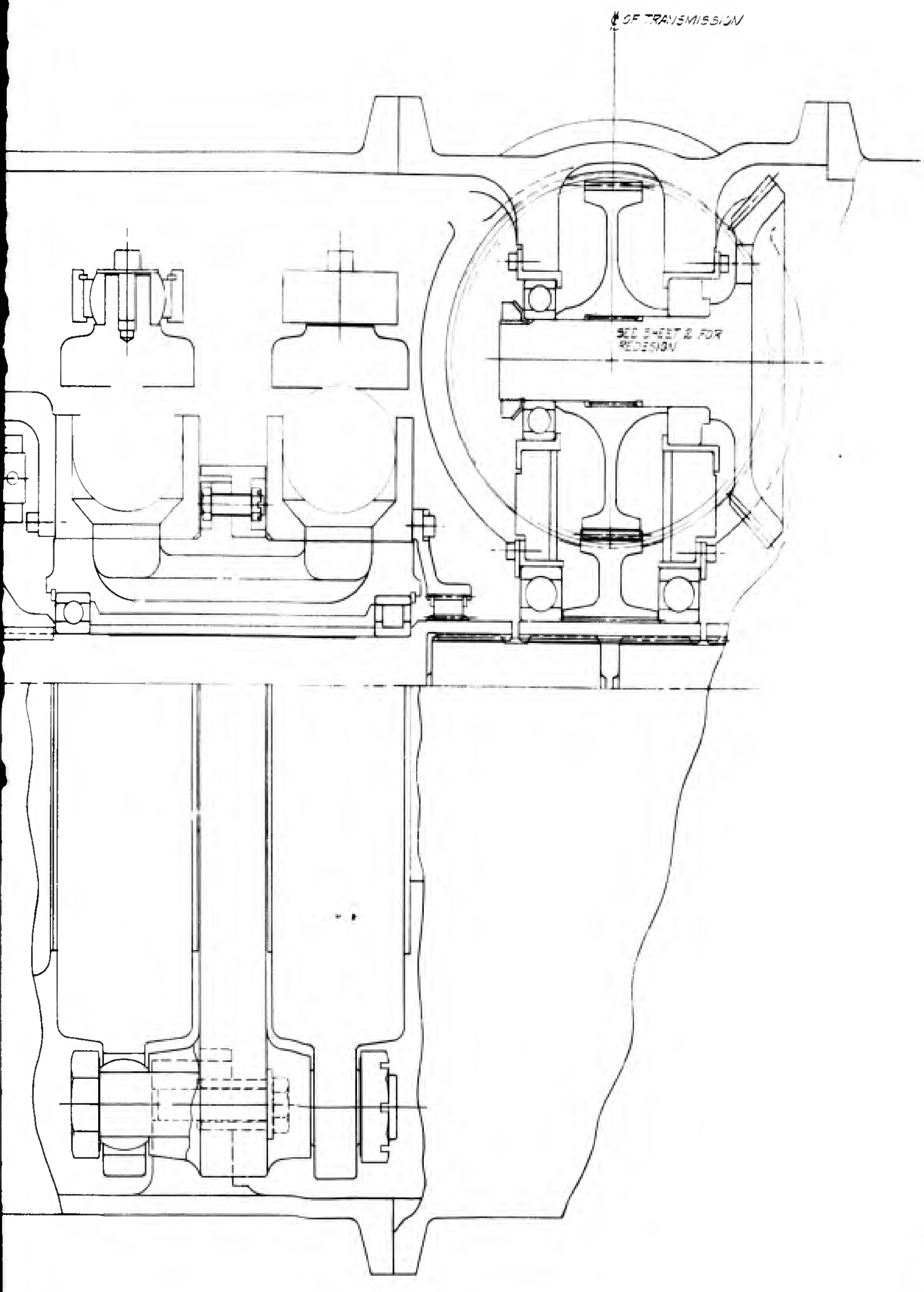
REVISIONS

ZONE	SYM	DESCRIPTION	DATE	APPROVAL



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E1076511-188

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LAYOUT
HYDRO-MECHANICAL
STEERING-TRANSMISSION

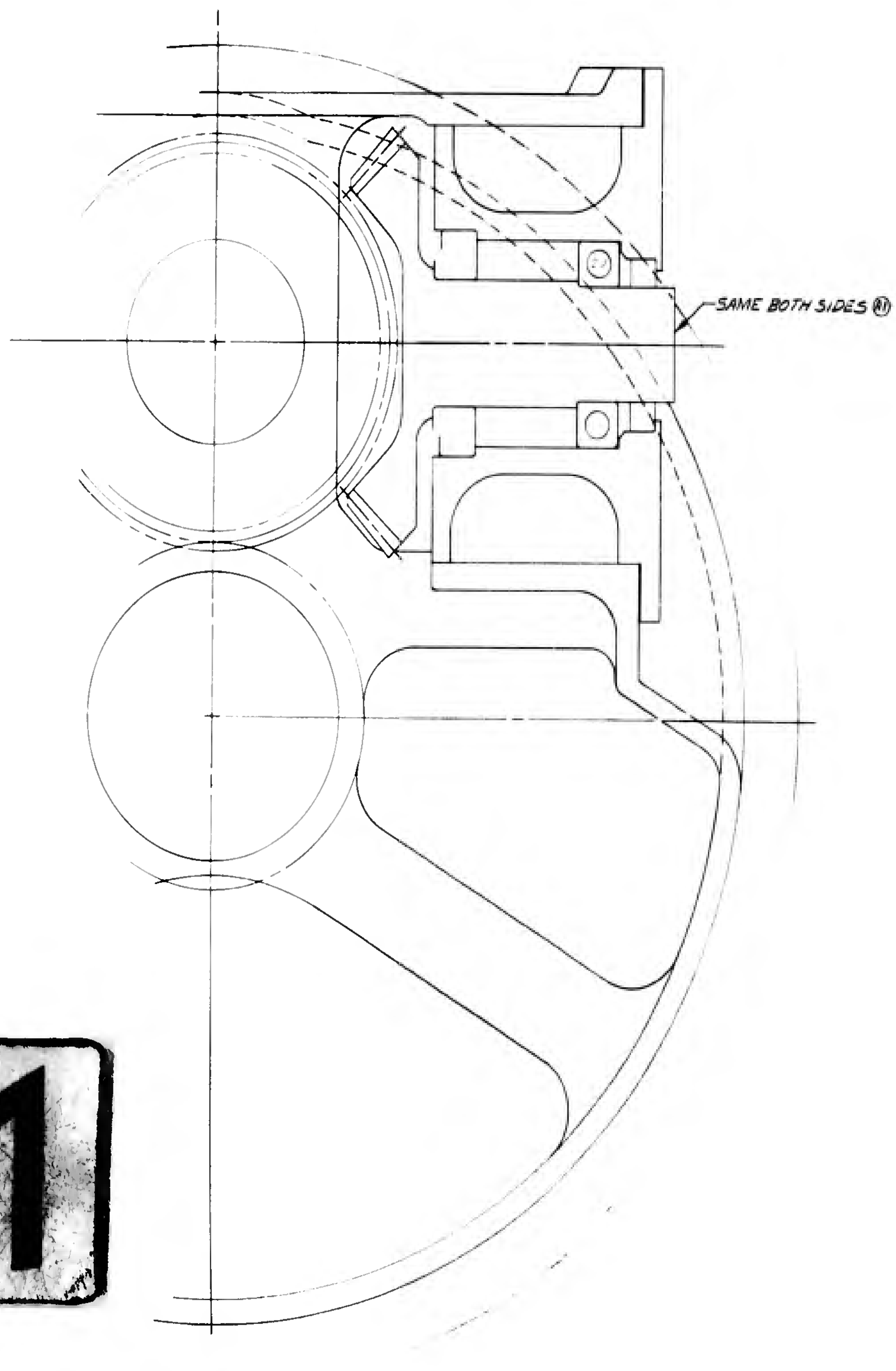
GENERAL ELECTRIC
LYNN RIVER
LOCATION

H6808-14

SIZE E 1076511-188

CONT ON SHEET 2 SH NO 1

CODE IDENT NO. 99611



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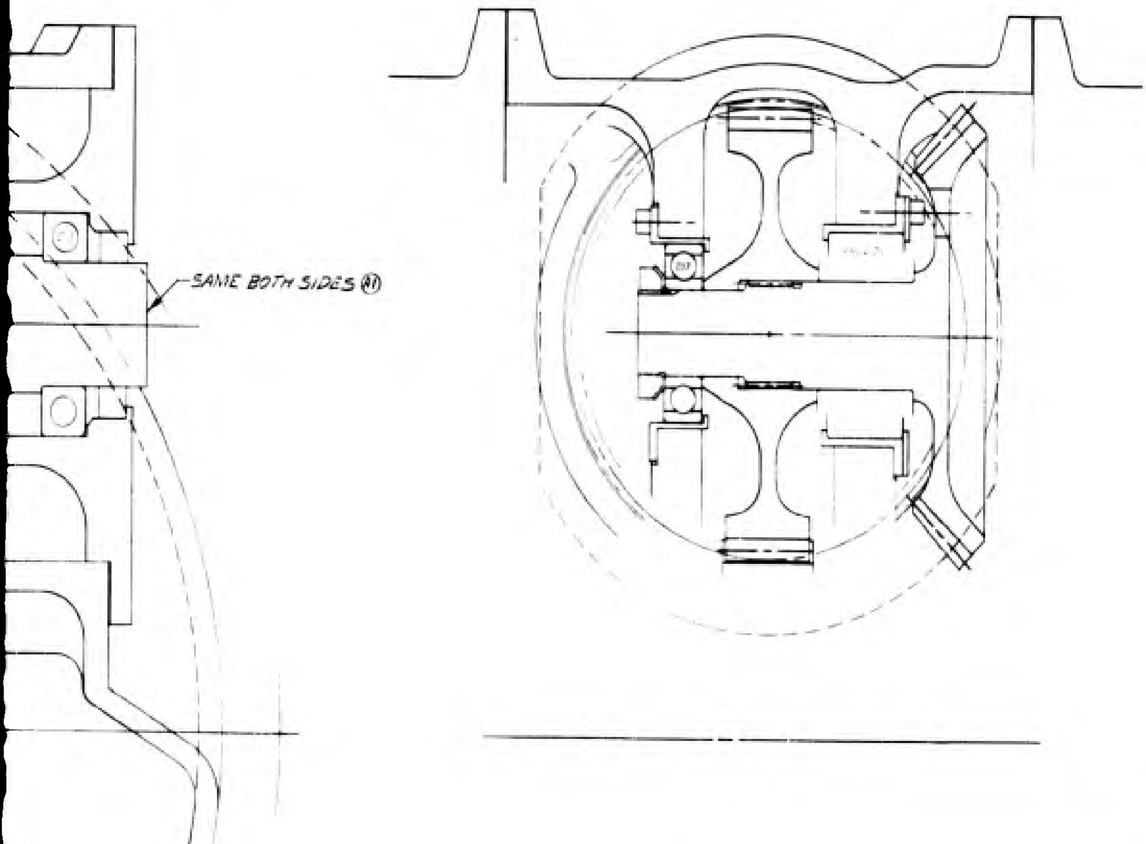
REVISIONS				
ZONE	SYM	DESCRIPTION	DATE	APPROVAL
	2	[illegible]	OCT 10 1961	[illegible]
		NOTE # 6008-14 [illegible]		



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E1076511-188A

E1076511-188

859
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LAYOUT
HYDRO-MECHANICAL
STEERING-TRANSMISSION

H6008-14

SCALE 1/2 | WT

GENERAL ELECTRIC
LYNN RIVER
LOCATION

SIZE E 1076511-188

CONT ON SHEET | SH NO 2

CODE IDENT NO → 99611

WEIGHT ANALYSIS (DRY)

(Without Final Drive)

Items	Material	
Pintles	Steel	88
Races	Steel	180
Balls	Steel	100
Blocks	Steel	105
Pumps	-	20
Brakes	-	50
Controls	-	20
Housing & Misc.	Aluminum	100
Input Gearing	Steel	30
Differential Gearing	Steel	20
Output Bearings	-	6
Stroking Piston	-	30
Center Shaft	Steel	10
Oil System	-	20
	Total	779 lbs.

Estimated oil weight in system - 55 lbs.

DESCRIPTION

Transmission

The proposed transmission is a hydro-mechanical geared differential steering transmission and drive system consisting of:

1) Two drives, each consisting of two hydraulic pump-motor elements mechanically connected through a differential planetary gear as well as hydraulically connected.

2) Input gear train.

3) Mechanical control for integrating sprocket, transmission and engine requirements.

4) Two make-up pumps for replenishing leakage flow in the hydraulic elements and lubricating gears.

5) Necessary regulating, relief, and check valves for proper system operation.

6) Emergency brakes.

The unit is completely self-contained, incorporating the system reservoir, pumps, filters and all the necessary valving for the oil system. The controls system is packaged as a component which can be incorporated in the transmission proper or mounted separately. Servicing and inspection can be accomplished by removal of the end bolts.

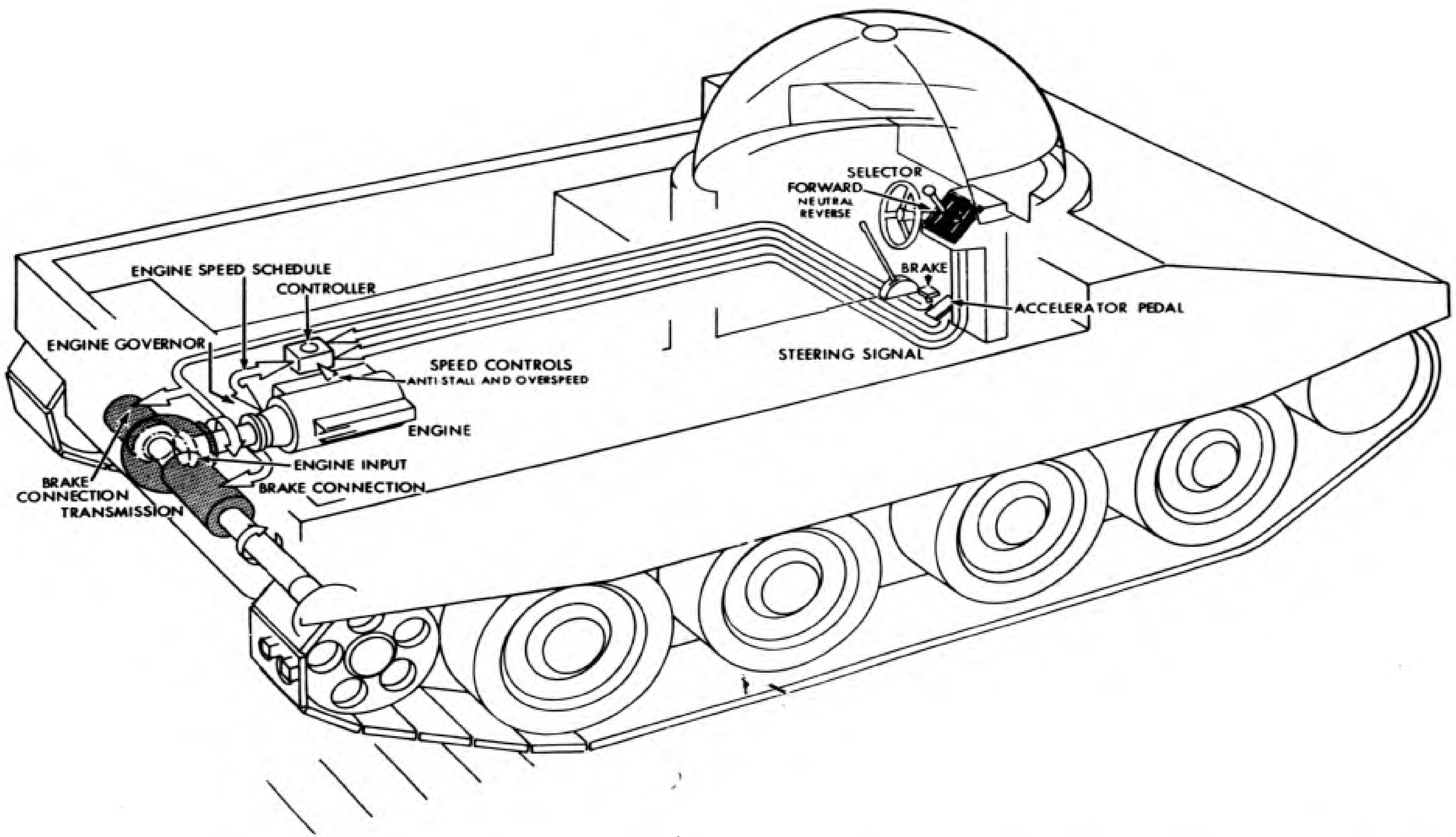
All highly stressed parts will be made of steel while the main housing will be made of aluminum.

The vehicle engine is connected by bevel gearing to both of the drives. The output of each drive unit is then connected through an appropriate gear reduction (external to the transmission) to the sprockets which move the tracks of the vehicle. A basic representation of the mechanical arrangement of the transmission in a vehicle is shown in Figure 4, and is shown in block form in Figure 5.

Each drive has a fixed displacement hydraulic unit connected to the ring gear of the output differential planetary gear, the sun gear connected to the input shaft, the planet connected to the output shaft, and a variable displacement hydraulic unit connected to the input shaft.

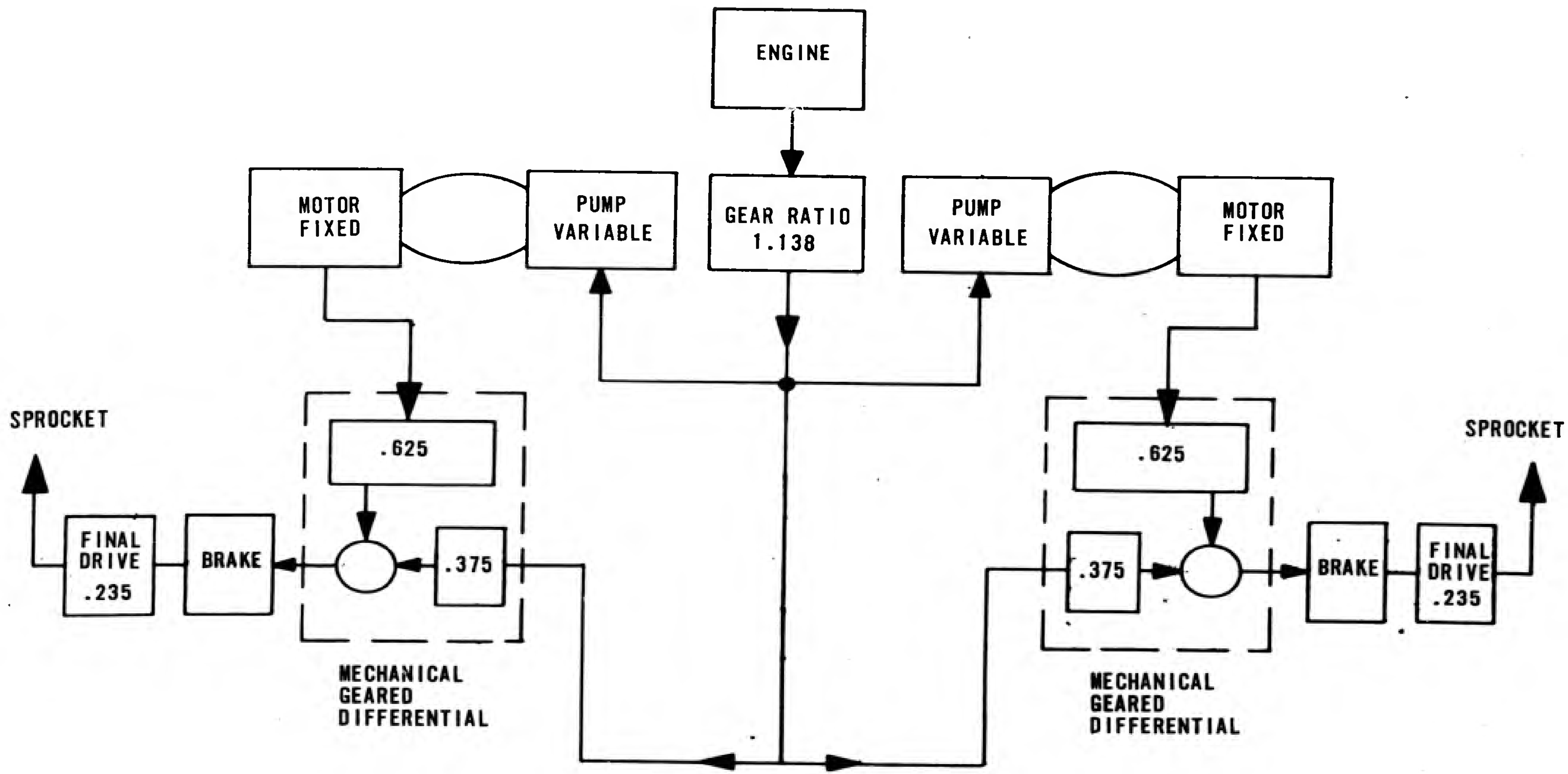
The variable displacement, "A" end, unit is hydraulically connected to the fixed displacement, "B" end, unit. By adjusting the stroke or displacement of the A-end unit oil flow is pumped to the B-end unit causing it to rotate in either one direction or the other depending on the direction of the A-end stroke (Figure 6). The rotation of the B-end unit and hence the ring gear of the differential may then be controlled to either have the same speed and direction as the sun gear and provide a variable forward sprocket speed, or may operate in reverse rotation to give zero sprocket speed and/or reversed sprocket speed.

It is then the direction and speed of rotation of the B-end or fixed displacement hydraulic unit that controls the speed ratio of the differential



HYDROSTATIC TRANSMISSION FOR TRACKED VEHICLES

FIG. 4



POWER BLOCK DIAGRAM

FIGURE 5

HOW THE HYDROSTATIC GEARED DIFFERENTIAL DRIVE OPERATES
(Ratic shown below is for planetary only)

Output Speed = Input Speed - (Input Speed - "B" end speed) .625

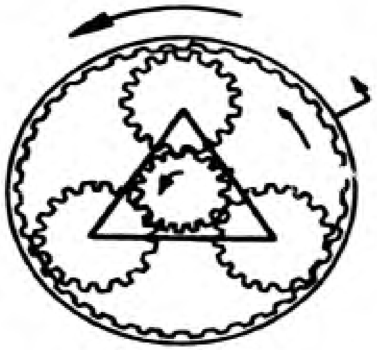
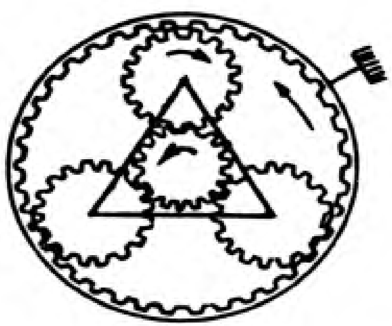
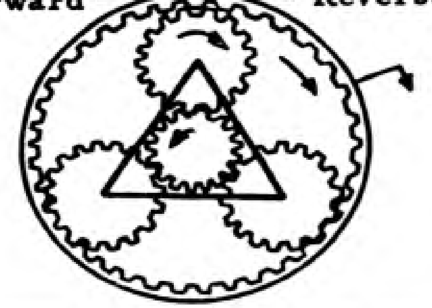
Planetary	Operating Mode and Ratio	"B" end	Flow	"A" end
Forward 	Max. Output Speed Ratio = 1 Output Speed = Input Speed	100% Speed Motor	100%	+100% Stroke Pump .
Forward 	Intermediate Speed Max. Efficiency Ratio = .375 Ring Gear Speed = 0	0 Speed	0	0 Stroke
Forward Reverse 	Min. Speed, Max. Torque Ratio→Inf. 0 Speed, Ratio = Inf. Reverse, Ratio = - .25	-100% Speed Pump	-100%	-100% Stroke Motor

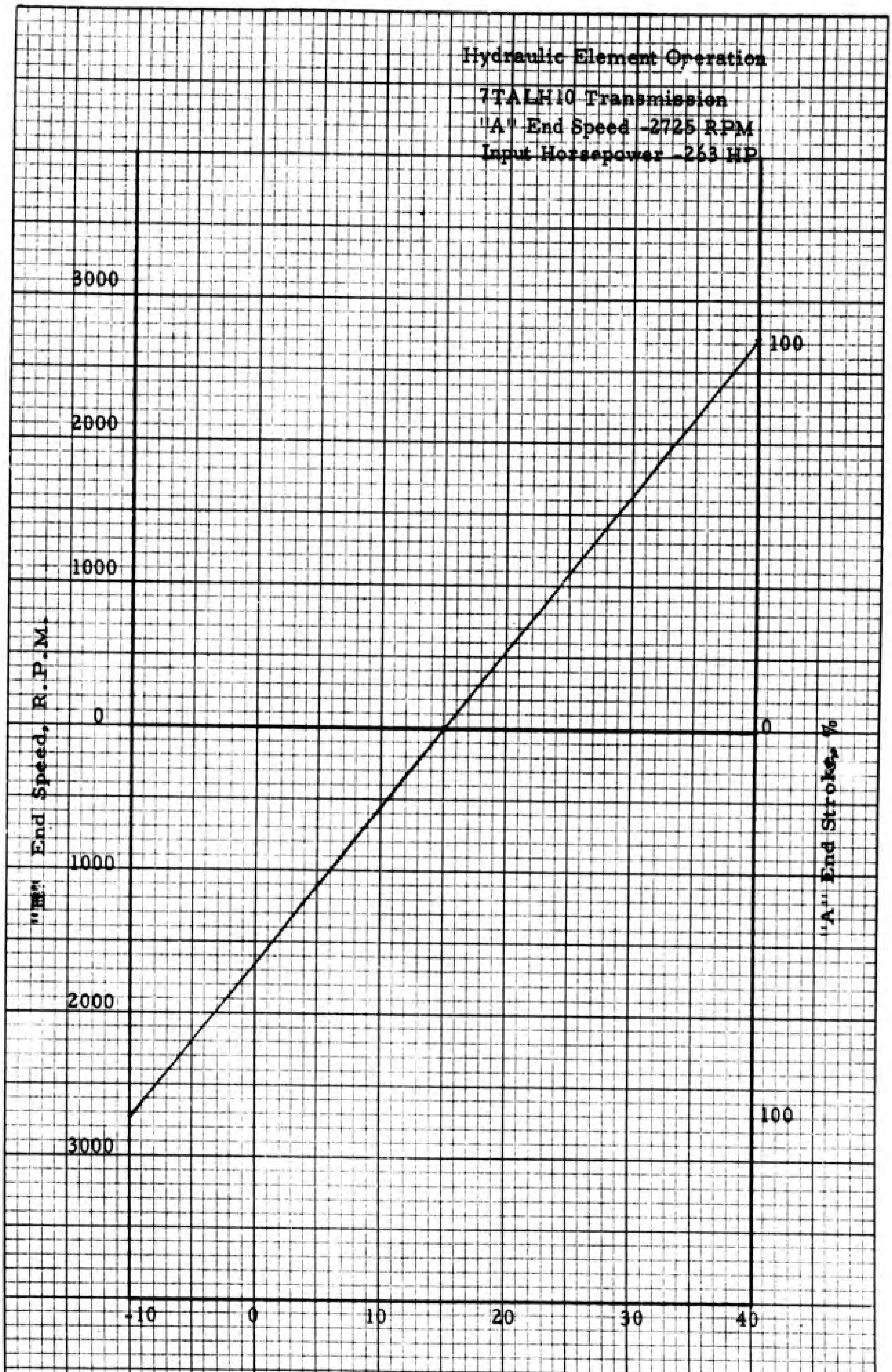
FIG. 6

planetary. By selecting the proper gear ratio of this planetary an infinitely variable speed ratio can be provided for speed ratios from full speed forward, to zero speed, to 25% full forward speed in reverse.

As noted previously the transmission ratio is controlled by the displacement or stroke of the variable displacement hydraulic unit and in turn the "B" end speed. From Figure 7 it can be seen that 100% stroke and maximum "B" end speed provides a maximum forward speed of 40 MPH. Zero stroke and zero "B" end speed provides a forward speed of 15 MPH. Minus 100% stroke and maximum reversed "B" end speed provides a reverse speed (maximum) of 10 MPH.

Since the output or "B" end is fixed displacement, the working hydraulic pressure will be a direct function of load torque from the sprocket regardless of vehicle speed. The drive when sized to provide the required maximum low speed torque is then capable of providing the same torque at any transmission speed ratio. This capability can provide for the large steering torque requirements during high speed turns as well as accept the reverse torques during the transfer of power from one track to the other during a turn.

The division of total power transmitted by mechanical or hydraulic varies as a function of speed ratio. The hydraulic portion of the power is a function of the "A" end displacement and hence "B" end speed. When the "A" end displacement is zero ("B" end at zero speed) the hydraulic power transmitted


Hydraulic Element Operation
7TALH10 Transmission
"A" End Speed -2725 RPM
Input Horsepower -263 HP
3000
2000
1000
0
1000
2000
3000
"B" End Speed, R.P.M.
100
0
100
"A" End Stroke, %
-10
0
10
20
30
40
VEHICLE SPEED, M.P.H.
FIGURE 7.

will be zero (Figure 8). Therefore, all of the power is transmitted mechanically. This occurs at a vehicle speed of 15 MPH. At the two extremes of vehicle speeds, maximum forward and maximum reverse speed, the hydraulic power transmitted is 62-1/2 per cent of the total power.

By using two differential hydro-mechanical drives, one for each sprocket, the two major requirements of a transmission for track laying vehicles are accomplished. First by controlling the strokes or displacement of both differential drive units together, the features of an infinitely variable speed ratio transmission are provided. By changing the two drive speed ratios differently, one higher, the other lower, the vehicle steering is provided with the transfer of torque and/or power between the one track to the other as required to turn the vehicle.

Although the "B" end unit is referred to as a fixed displacement unit, it actually will be capable of assuming two stroke positions - full stroke and zero stroke. Under all conditions of normal vehicle operation the stroke will be fixed at the full or 100% stroke point. Only under the condition of high speed towing will the stroke be moved to zero stroke to insure minimum flow loss as the vehicle is towed. That is, by stroking the unit to zero stroke no oil will be pumped as the units are rotated by the wheel sprockets.

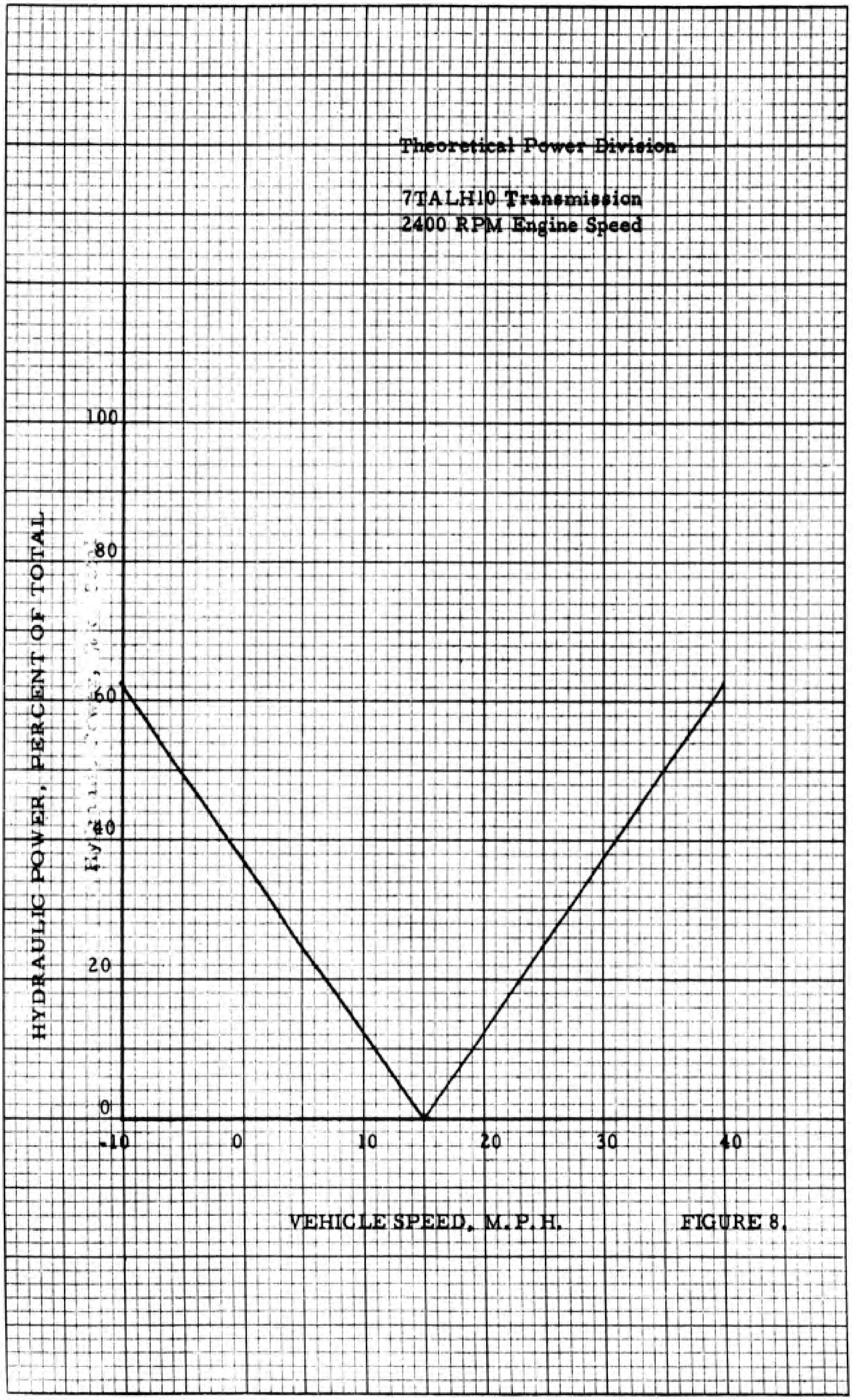
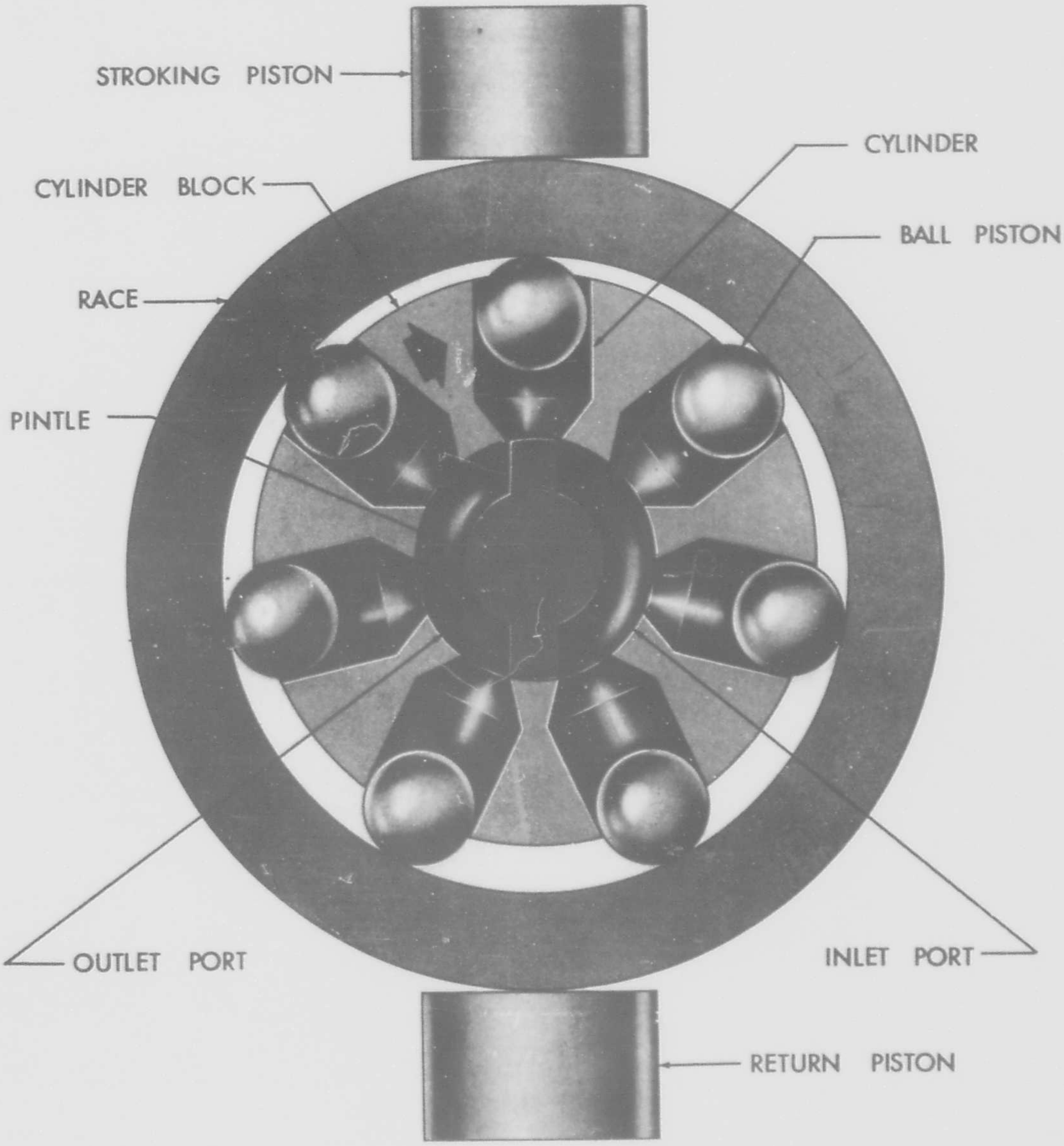

Theoretical Power Division
7TALH10 Transmission
2400 RPM Engine Speed
HYDRAULIC POWER, PERCENT OF TOTAL
100
80
60
40
20
0
-10
0
10
20
30
40
VEHICLE SPEED, M.P.H.

FIGURE 8.

BALL PISTON UNIT



Ball Piston Unit.

Hydraulic Elements

The hydraulic elements are radial piston, positive displacement hydraulic units capable of operating as a pump or motor. The units are unique in that the conventional cylindrical pistons have been replaced by precision steel balls.

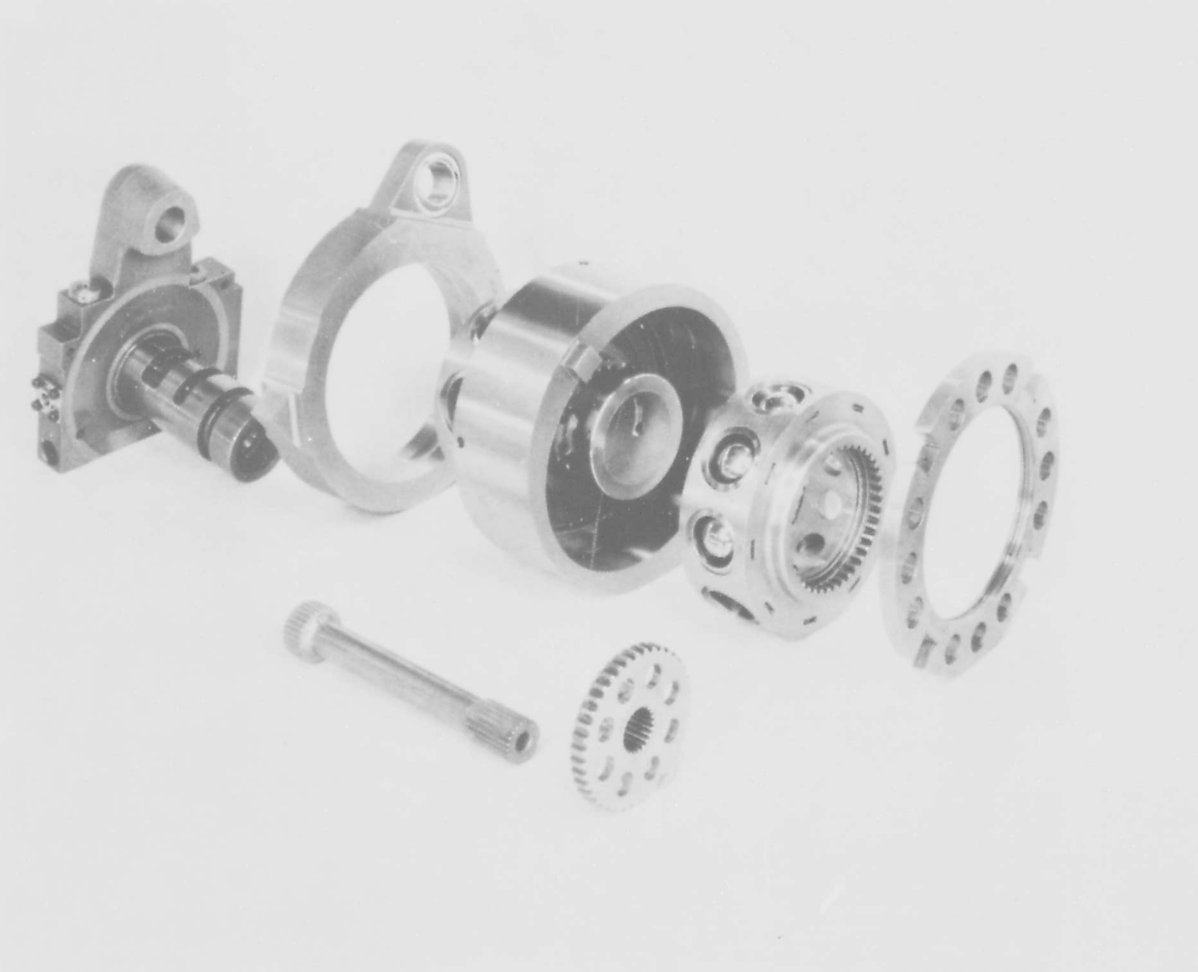
A diagram of the basic ball piston unit is shown on the preceding page. It consists of a cylinder block which contains the ball pistons, a shaft or pintle upon which the cylinder block rotates and which contains the oil passages and ports, a race in which the balls rotate and a stroking mechanism to control the amount of eccentricity of the race relative to the block and in turn displacement.

When the cylinder block is rotated by an external force (engine shaft) in a clockwise direction, the ball in the top cylinder moves radially outward, due to centrifugal force, and continues to do so for approximately 180 degrees. During this time the bottom of the cylinder is opened to the right-hand (inlet) port and oil is drawn into the cylinder. As the ball completes the first half of the cycle, it passes over the land between the ports at the bottom, and the cylinder is then opened to the left-hand (outlet) port. During the remainder of the revolution the ball is forced radially inward in the cylinder by the race, thus forcing the oil from the cylinder into the high-pressure or left-hand outlet.

To change the delivery from the unit all that is necessary is to change the eccentricity of the race by shifting the race with respect to the pintle. For example, if the race is concentric with the pintle there will be no pumping action. In a variable displacement unit, the degree of eccentricity of the race is varied by means of the stroking pistons. If the eccentricity of the race is opposite to that shown in the illustration, and the same rotation of the cylinder block is maintained, the direction of the oil flow will be reversed.

If oil is supplied under pressure, the unit will act as a motor. Oil pressure supplied to the right-hand port will force the balls in the right-hand cylinders outward against the race and part of this outward force will be vectorially resolved into a clockwise rotating force on the cylinder block. After the oil-loaded cylinders pass the pintle land at the end of the first half of the revolution, their charge of oil is returned to the outlet port by the inward movement of the ball as it runs on the race. Similar to the pump above, if the eccentricity of the race is opposite to that shown, the motor will reverse direction of rotation.

The design will be standardized so that the hydraulic elements will be interchangeable to a maximum degree. The four cylinder blocks, pintles and gears will be all interchangeable as well as the strokeable and fixed races.



Transmission Component Parts

TRANSMISSION CONTROL

There are three basic vehicle operator controls required for operating the engine and transmission over the complete sprocket torque-speed range of the vehicle.

Selector Lever for forward, neutral, reverse.

Accelerator Pedal to control vehicle speed.

Steering mechanism for controlled steering.

The control for dynamic braking can be tied in with the selector lever or provided with a separate lever.

A mechanical-hydraulic control box mounted on the transmission receives input from the operator's controls and engine speed to provide the mode of vehicle operation desired at the lowest possible fuel consumption rate. Specifically the control performs the following basic functions:

- Sets the transmission elements for forward, reverse and neutral operation.
- Sets the transmission ratio
- Controls engine speed.
- Selects a combination of engine speed and transmission ratio to provide the vehicle speed and tractive effort called for by the operator at best fuel economy.
- Prevents engine stall due to overloading.

- Minimizes engine overspeed during transients and dynamic braking.
- Provides smooth steering including automatic regenerative steering.

The control is the basic link or "brain" which integrates the transmission with the rest of the vehicle and automatically compensates for excessive transients imposed upon the vehicle by the terrain or the operator. The control senses an accelerator pedal position calling for a given vehicle speed. It adjusts the engine speed in relation to the accelerator position. The transmission ratio is controlled at the same time to keep the engine loaded at its maximum output horsepower for a given speed. This means, then, under practically all vehicle operating conditions the engine will be operating at or close to its minimum fuel consumption point for that speed.

As a result of computer studies, a control has been developed which utilizes proportional signals. This eliminates the necessity for several pilot valves and associated pistons. Furthermore, a new concept for accurate speed sensing has been adapted which eliminates the necessity for using a flyweight governor. The control components are depicted on the block diagram of Figure 11.

The accelerator linkage input motion is used to generate two functions representing the desired engine stroke and speed. The signal

TRANSMISSION AND ENGINE CONTROL BLOCK DIAGRAM

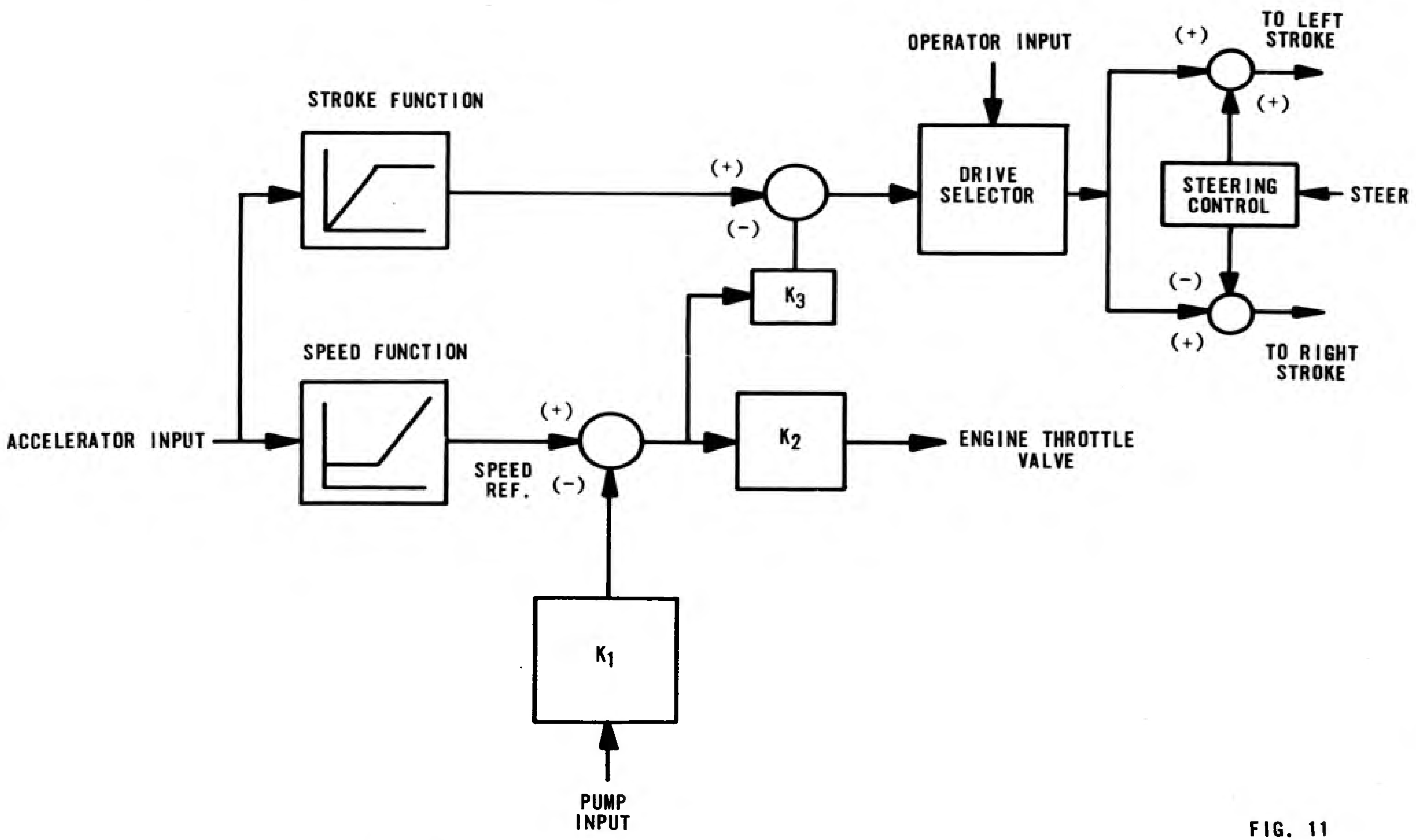


FIG. 11

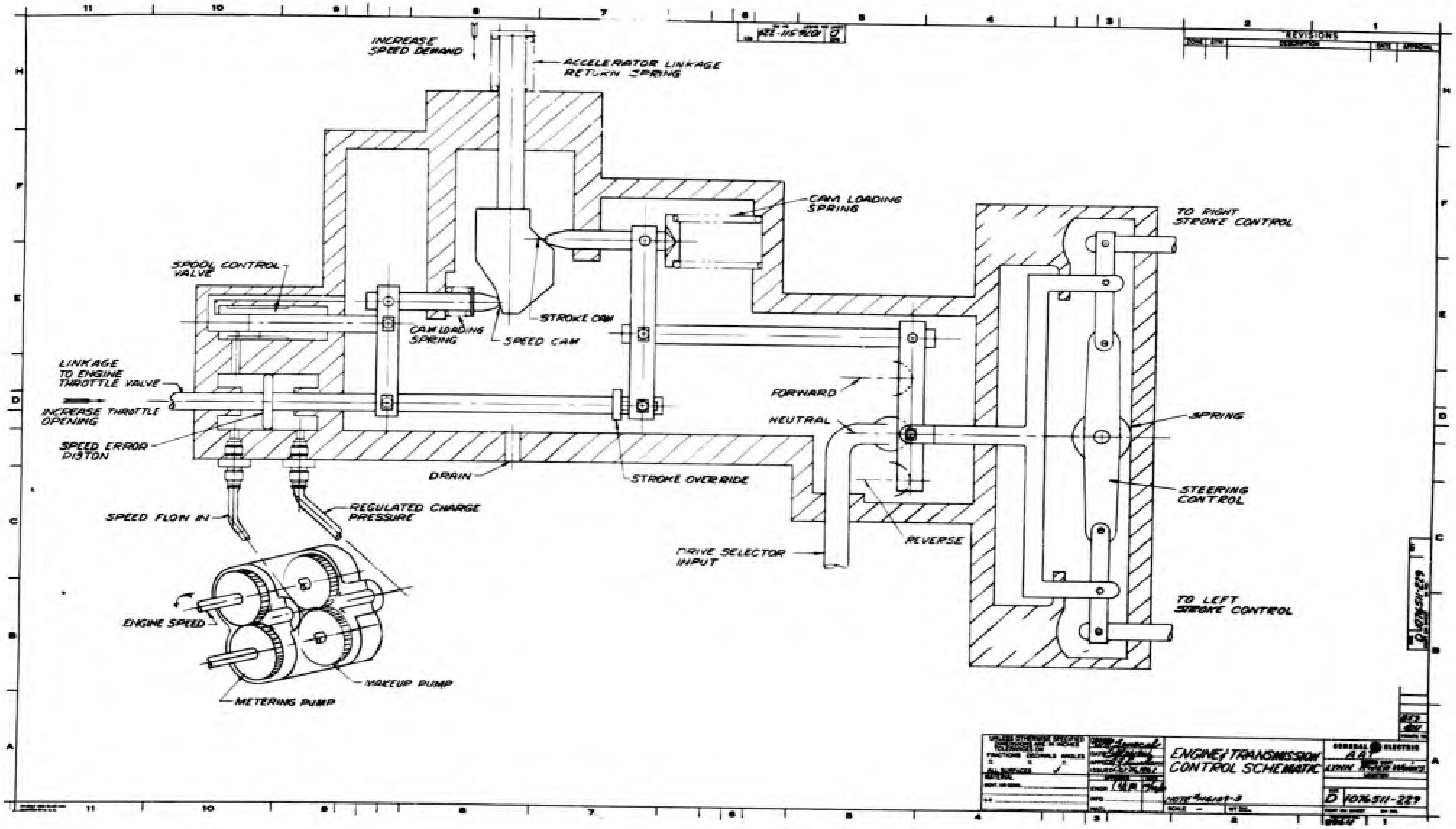
representing the desired engine speed is compared to one representing the actual engine speed and the difference between the two operates the engine throttle valve. The desired stroke (which actually sets the transmission ratio) is obtained by passing the input signal through the drive selector, the output of which positions the steering linkage and subsequently to the stroke valves.

A stroke override feature is also provided which allows engine speed error signals to override the normal stroke setting. The stroke override loop is shown on the block diagram sensing engine speed error and is used to modulate the transmission stroke so that regardless of the road load, the engine can operate at the scheduled speed. Assume a condition where the vehicle is running on a level road, at maximum speed, and suddenly begins to climb a steep grade. Because of the increased load, the engine would start to slow down, thus developing a large speed error. (Assuming the accelerator input is still unmoved.) The called-for stroke by the accelerator position would then be overridden and the transmission stroked back until the load reflected into the engine was such that the engine could run at the scheduled speed. This would, of course, slow the vehicle down, but would allow the engine to run at maximum speed and thus at maximum horsepower.

Inspection of the functional schematic, shown in drawing 1076511-229 will further clarify the control operation. The purpose of the schematic is to indicate functionally how the control will operate with as much simplicity as possible. It is not the purpose of this schematic to indicate the detailed mechanization of the various control components.

The accelerator linkage input is equipped with a return spring which also gives the operator the necessary feel. This input directly positions a double-edge cam whose output sets the engine speed and stroke function. A cam is required so as to obtain a linear output speed versus accelerator position (see Fig. 17). The engine speed and stroke are related relative to the accelerator input, but their relationship is not linear.

The cam follower representing the desired stroke is connected through linkage to a travel reversing component which is controlled by the vehicle operator. Depending on the position of the drive selector (forward, neutral, reverse), the signal to the stroke valves is passed directly reversed or cancelled. The device is shown schematically as a linkage, the pivot point of which is changed by the drive selector input. The override which operates on the same link as the stroke cam follower is actuated by output of the speed error piston. It is desirable for the stroke to change from zero ratio at zero accelerator input to full ratio at an accelerator position corresponding to about 25 percent of its full


INCREASE SPEED DEMAND
ACCELERATOR LINKAGE RETURN SPRING
CAM LOADING SPRING
TO RIGHT STROKE CONTROL
SPOOL CONTROL VALVE
STROKE CAM
CAM LOADING SPRING
SPEED CAM
LINKAGE TO ENGINE THROTTLE VALVE
INCREASE THROTTLE OPENING
SPEED ERROR PISTON
FORWARD
NEUTRAL
SPRING
DRAIN
STROKE OVERRIDE
STEERING CONTROL
SPEED FLOW IN
REGULATED CHARGE PRESSURE
DRIVE SELECTOR INPUT
REVERSE
TO LEFT STROKE CONTROL
ENGINE SPEED
MAKEUP PUMP
METERING PUMP
REVISIONS
ENGINE/TRANSMISSION CONTROL SCHEMATIC
GENERAL ELECTRIC
D 1076511-229

travel. Beyond this point, the transmission will be at full ratio unless affected by the speed error actuated override.

The output of the drive selector is applied simultaneously to two summing links and in turn to the left and right stroking pistons. The steering function is applied to these summing links as well. When the transmission is stroked as a result of accelerator pedal input motion, both the left and right units are stroked the same amount and in the same direction. However, if there is steering, one side is stroked in one direction and the other in the opposite direction. This has the effect of slowing one side down and speeding the other up. Thus an infinite turning ratio is obtained.

A speed sensor is employed which uses the output flow from a small positive displacement pump driven by the engine to give a measurement of its speed. The inlet of this pump is supplied by the low pressure pintle charging pump. This same pressure is applied to one side of an equal area piston. The discharge from the speed measuring pump is ported to the other side of the piston. Therefore, for a piston with negligible loading, the pressure on both sides of the piston must be equal in order to maintain force balance. This means there is zero pressure differential across the speed measuring pump and the internal leakage will be negligible. The steady-state flow represents pump speed to a very good accuracy. The speed indicating flow is ported from the piston cavity through a variable orifice comprising a spool valve and a sleeve with a rectangular port. The output motion of the piston is "fed back" through a linkage to position the spool valve. The motion of the speed function cam follower is also applied to this linkage with the result that the output motion of the piston represents the speed error of the system. It is this motion which positions through appropriate linkages the engine throttle valve and also provides for stroke override.

Engine Speed Scheduling

Since one of the basic functions of the control is to select combinations of engine speed and transmission ratio required to meet vehicle requirements at best fuel economy, this feature should be further explained.

For best fuel economy at a given vehicle speed the ideal situation is to have both engine and transmission operating at their highest efficiency for that power requirement. For a gasoline or diesel engine the lowest fuel consumption rate is almost always at the lowest engine speed that will produce the required horsepower. For the G-E hydromechanical transmission the most efficient operation occurs at zero per cent stroke when the power is transmitted 100% mechanically and the input to output ratio is 0.375. It is not always possible to meet required vehicle speeds and torque with both engine and transmission at highest efficiency. In these cases our studies show that the lowest fuel rate is obtained by optimizing engine speed at the expense of operating the transmission off its peak efficiency point. In other words, the effect of engine efficiency is greater than transmission efficiency on overall vehicle fuel economy.

The stroke and speed functions shown in the block diagram, Figure 11, have been designed by shaping a double-edged cam in the control to rapidly increase the transmission ratio to 1:1 during acceleration while slowly increasing engine speed. In terms of a gear shift transmission this is analagous to shifting into full ratio at the lowest possible vehicle speed to keep engine speed as low as possible. Figure 12 shows the relation of engine speed to stroke (ratio) and output speed as a function of accelerator pedal position, for low output pressures. It can be seen that output speed (vehicle speed) is directly proportional to accelerator position. Further, it is noted that by increasing transmission stroke (directly proportional to ratio) rapidly the output speed increases with very small increase in engine speed up to full ratio. After that, further acceleration is directly proportional to engine speed. This mode of operation designed to conserve fuel is not believed common to other types of conventional transmissions.

Examination of Figure 13 will explain somewhat the scheduling technique. This figure presents sprocket torque versus output speed for lines of constant power and engine speed.

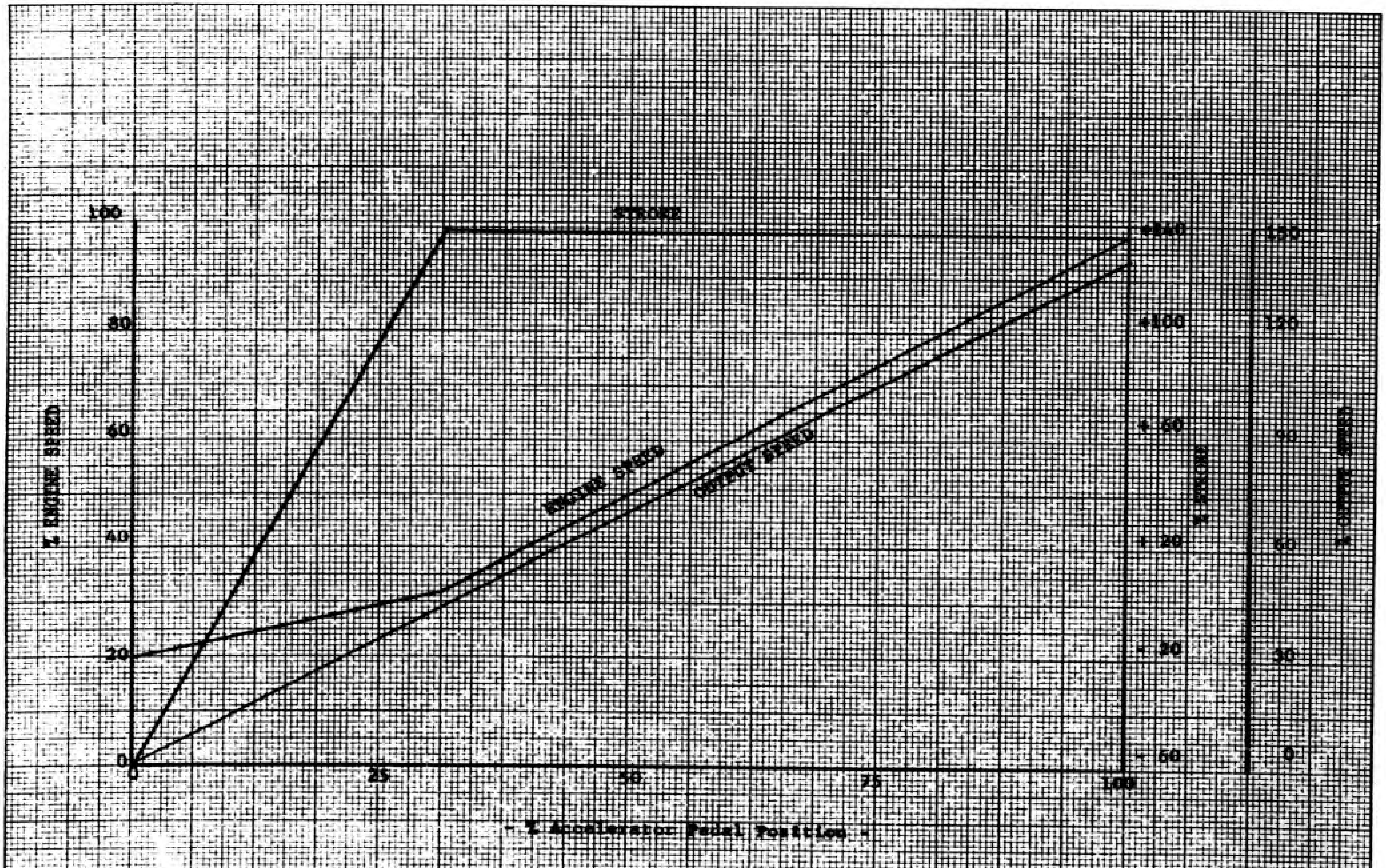

STROKE
ENGINE SPEED
OUTPUT SPEED
% ENGINE SPEED
% STROKE
% OUTPUT SPEED
- % Accelerator Pedal Position -
0
20
40
60
80
100
25
50
75
100
+140
+100
+60
+20
-20
-60
0
30
60
90
120
150

FIGURE 12

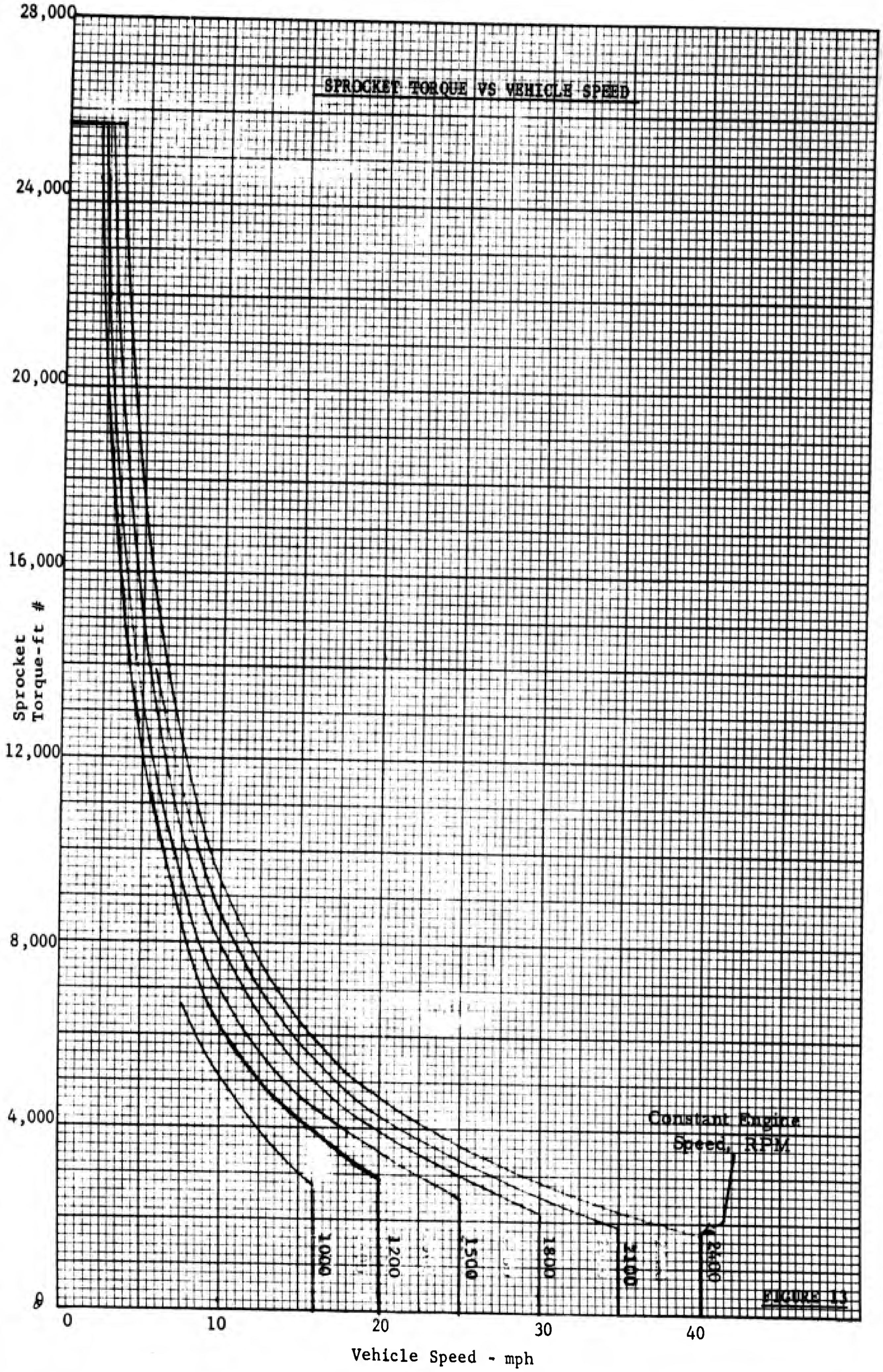

SPROCKET TORQUE VS VEHICLE SPEED
Sprocket Torque-ft #
28,000
24,000
20,000
16,000
12,000
8,000
4,000
0
0
10
20
30
40
Vehicle Speed - mph
Constant Engine Speed, RPM
1000
1200
1500
1800
2100
2400

FIGURE 13

Consider the 1800 RPM line. As the torque required at the output is increased from zero to 2200 ft. lbs. the output and engine speed remain unchanged. At an output load demand of 2200 ft. lbs. the engine is fully loaded for that particular engine speed (1800 RPM). The transmission is in full ratio. As the load is increased above this point and the accelerator position is maintained constant then the stroke override will actuate and reduce ratio and therefore output speed, however, the engine speed will remain unchanged.

Having done the basic analytical work on the control, refinements based on the availability of detailed vehicle and engine characteristics can be easily and quickly accomplished.

Computer Simulation of Hydro-mechanical Transmission Operation in a Gasoline Engine Powered Vehicle

The control concept as just described has evolved from a system study performed utilizing an analog computer. A basic transmission engine and vehicle were simulated. Various control schemes were developed and evaluated. The one described previously represents our best effort to date to achieve a truly integrated engine and transmission system.

Further development in this area rests on the availability of better engine performance data. Something comparable to the performance maps currently available for jet engine compressors and turbines. Given this type of information a control can be developed which will optimize the matching of the engine and transmission in terms of overall efficiency and specific fuel consumption. Furthermore, the promise exists of even further increasing the range and capabilities of this type of transmission over those presently used in military vehicles.

Sample traces which are a result of the computer study are shown in Figure 14. The simulation is one for a 12-ton truck equipped with a 135 HP engine. The transmission is only one-half that proposed here and would take the place of a torque converter. The traces illustrate the transients which occur when the vehicle is started from rest with the engine idling and when the accelerator is returned to zero position from full speed. The only braking applied is that supplied by rolling resistance and the engine.

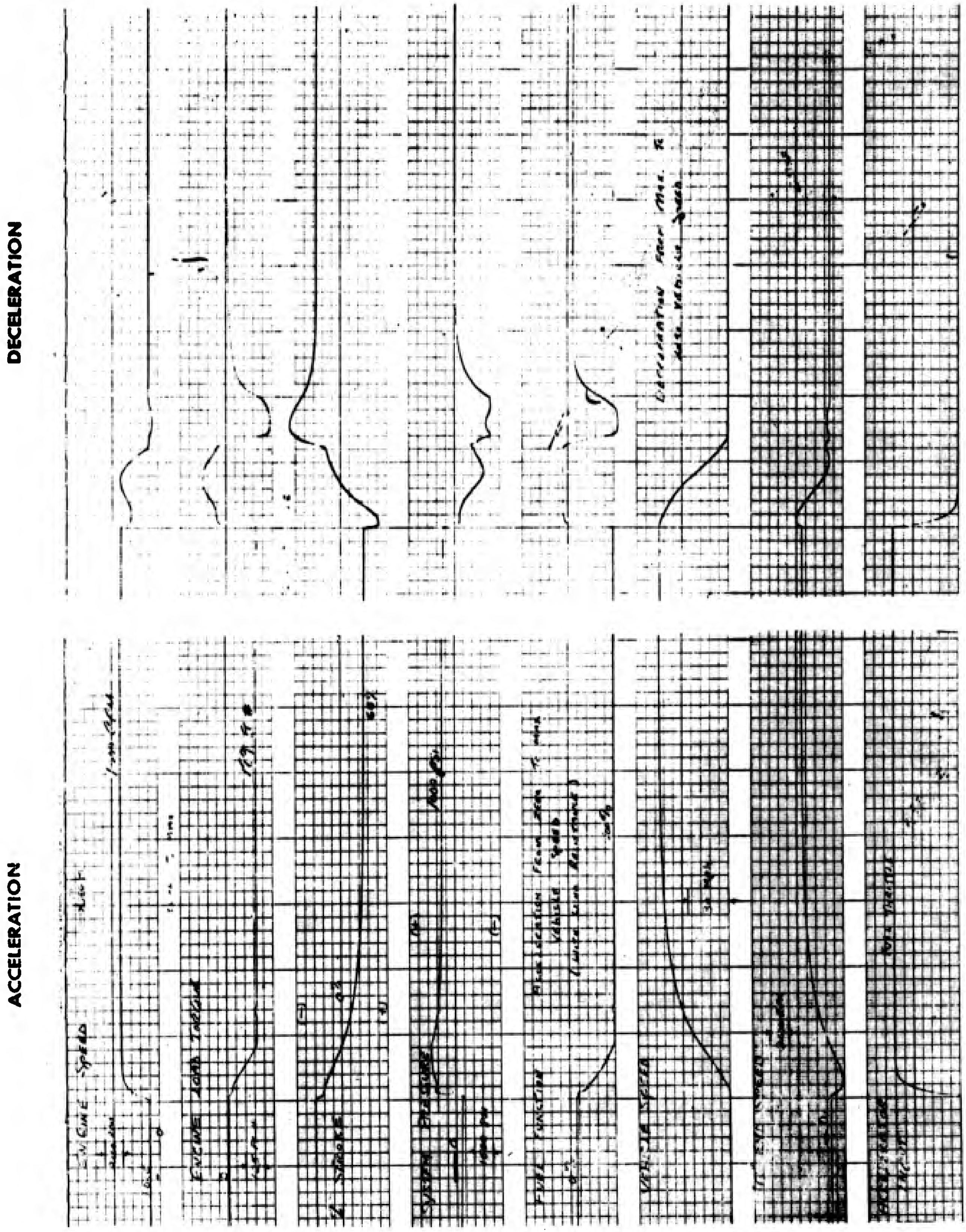


FIG. 14

Towing

The proposed transmission provides for long periods of high speed towing. Suitable operation under this condition is accomplished by (1) provision of a transmission output driven make-up pump, and (2) zero stroke position of the "B" end or output hydraulic unit.

The make-up pump provides adequate lubrication to the output hydraulic units and to the gear train as the units are rotated by the towing.

Stroking of the "B" end to zero results in no flow being pumped by that unit and in turn no excessive transmission drag torque being seen by the vehicle.

Adjustment of the transmission to allow towing is accomplished by a simple lever which has two positions - normal and tow. As noted elsewhere, this lever may be tied in with the selector lever actuating rods to be moved to the tow position when selector is moved to neutral.

Dynamic Braking

The control through the transmission performs a secondary function, dynamic braking. The principle of dynamic braking can best be illustrated by analogy to an automobile with a standard shift. If, when the accelerator pedal is released on such a vehicle, the car is shifted from third to second gear, an immediate increase in deceleration rate is produced. However, with a hydro-mechanical transmission, the gear ratio is infinitely variable, reduce wear and heat generated in the vehicle's service brakes or to minimize wheel skid. With the infinitely variable transmission the ratio can be controlled to gradually change during deceleration such that maximum engine retarding torque is constantly applied throughout the deceleration period by keeping the engine driven at a maximum safe speed. The kinetic energy of the vehicle is then converted to heat in the engine by friction and air pumping.

In the proposed transmission this capability for braking is inherent in the design, but will require a control signal from the operator before it will take effect. Without the control signal, the transmission will remain in the same ratio when the operator reduces the accelerator pedal setting. The control signal can come from an additional position on the forward-neutral-reverse lever or from a separate control lever.

It should be noted that the transmission itself is not providing braking as in true dynamic braking, but is dependent upon the engine for dissipation of the vehicle power. True dynamic braking allows vehicle power to be dissipated hydraulically by throttling, at a high pressure, the fluid flowing between the "B" end hydraulic unit and the "A" end hydraulic unit. In a hydromechanical transmission, the necessary valving and its tie-in to the transmission oil system becomes difficult.

It is therefore recommended that if braking in excess of that obtainable from the engine is required, a simple commercially available hydraulic retarder unit be added between the engine and the transmission. Details of such a device would be defined at a later date.

Tow Starting

The vehicle may be started by towing by simply placing the operation selector in the forward position. (The starting should be done by towing the vehicle in the forward direction.) Basically tow starting is a special case of dynamic braking, wherein the sprocket motion is transmitted to the planet carrier of the planetary gearing of the mechanical-hydraulic differential drive units. At first with transmission input speed and the sun gear at zero speed, the ring gear and B-end hydraulic will start to rotate. As the B-end unit gains speed it will try to pump air and oil and in addition its torque drag or impedance will cause torque to be transmitted to the sun gear causing it to start to rotate. At some very slow speed of the sun gear and input speed of the transmission, the make-up pump will start furnishing sufficient fluid to the drive units so that the input speed and vehicle engine will be immediately increased to starting speed.

Gearing

There are three distinct functions provided by the power gears: right angle input common to both transmissions; differential between each hydraulic drive output and the central shaft which is fixed at 59% of engine speed; and final speed reduction. The arrangements shown give the required compact length and are based on prior successful gear design in details of mounting and unit load.

The gear sizes were established by duty cycle analysis correlated into accumulated experience curves of allowable loads. For this application, all gears can be hardened, either by carburizing or induction hardening, and therefore higher limits for surface compressive stress, scoring, and for tooth bending stress can be permitted.

Suitable mounting is vital to these gears. It has been provided by anti-friction bearings, stiff casings and, for the planetary elements, self-centering planet carriers. By these means, full face contact and equitable load distribution are assured. Rolling-element bearings have the best resistance to momentary overloads and marginal bearing lubrication at sustained low speeds. In addition, they have low losses and are easily lubricated.

Bearing sizes were calculated in critical spots by the same limiting loads and lives as for the gears, using B10 life cycles of twice the required service, and per industry standards of capacity set by AFBMA.

The differential and final-reduction planetary gears are lubricated by dip and splash of the sump oil, usually considered most effective, particularly for starting and low speeds. The input gearing will be lubricated and cooled by leakage flow from the hydraulic drive, augmented by oil jets from the make-up pump or control pump.

Oil System

The proposed transmission has a self-contained oil system requiring only a periodic check of the oil level and scheduled cleaning or replacement of the oil filter element. The working fluid for the hydrostatic elements is also used to lubricate the gears and bearings. A block schematic of the oil system is shown on the following page.

The transmission case serves as an oil reservoir and sump. Oil in the case surrounding the rotating parts will be scavenged to reduce heat generated by viscous drag. Two fixed-displacement make-up pumps, one driven from the input gearing and one driven from the output gearing, will pump oil from the sump to supply the internal leakage of the hydrostatic units and lubricate the gears and bearings. The output driven pump is a small unit whose main purpose is to furnish lubricating flow during periods of high speed towing of the vehicle. The oil passes through a 40 micron filter, through the cooler to the lubrication jets and the hydrostatic units. Check valves are placed in the lines to the hydrostatic units to insure make-up flow regardless of which direction of rotation of the units and in turn pressure levels of the internal passages. High pressure relief valves will be used to prevent working pressures exceeding 3500 psi. The make-up pump inlet and sump are designed to allow operation at angular attitudes as specified without uncovering the pump inlet.

HYDROSTATIC TRANSMISSION OIL SYSTEM BLOCK DIAGRAM

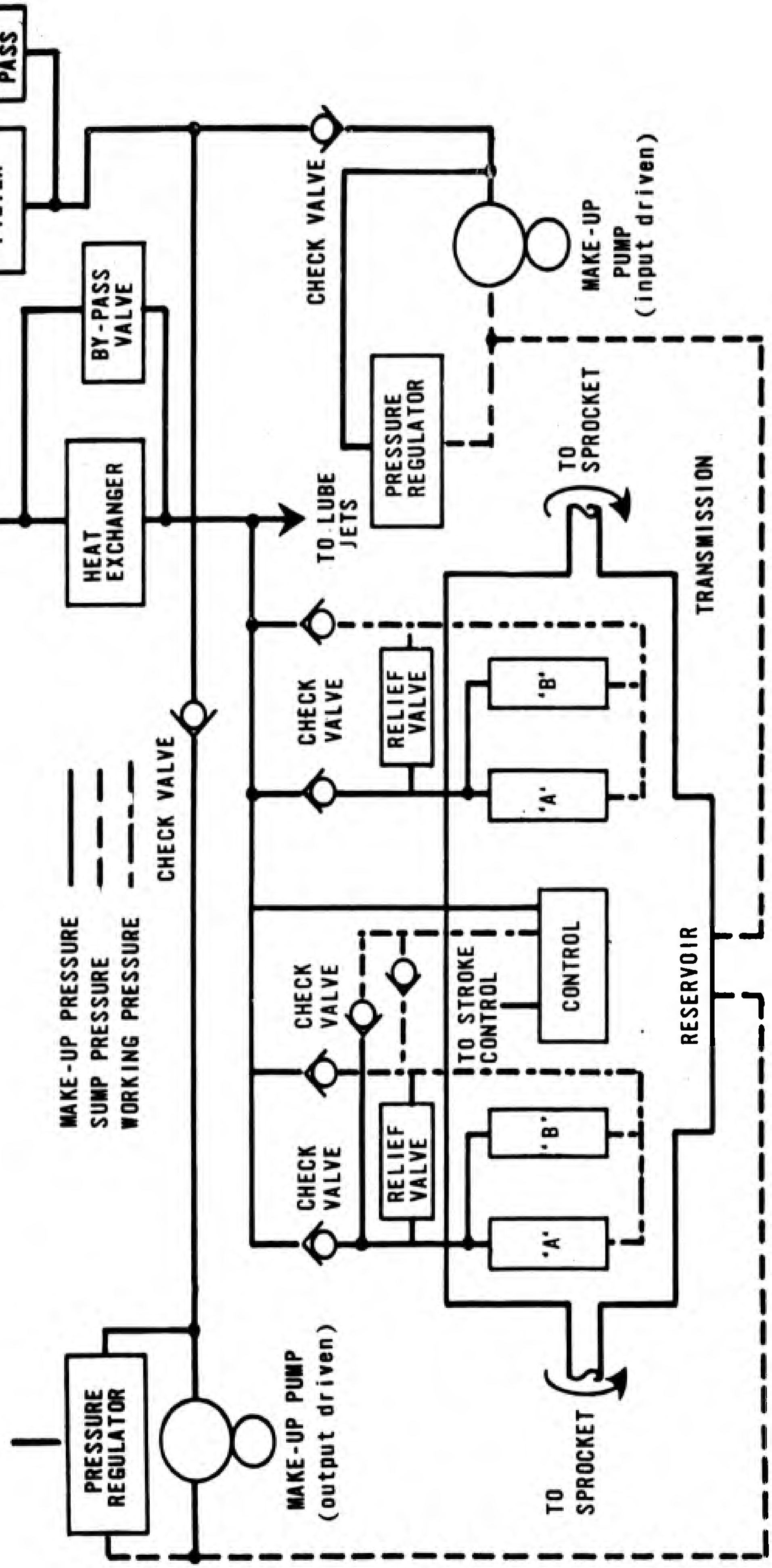


FIG. 15

The full flow of the make-up pump will be supplied to the cooler supply port in the case so that an external cooler may be connected. Thermal and pressure bypass valves are provided to allow the cooler circuit to be automatically bypassed at very low oil temperatures or when cooler is clogged.

Actual cooler rating will be determined after all the vehicle requirements are known. Although the total heat rejection from the transmission is large in terms of total horsepower, only a portion of this need be taken care of by the cooler because normal radiation, convection, and conduction will remove a large portion of this heat.

Service Braking System

In addition to the dynamic braking or engine braking system there will be a secondary braking system, service brakes.

The service braking system consists of two oil cooled hydraulic brakes, one on each output shaft. This system will be sized to provide a braking effort sufficient to meet the specification without use of the dynamic braking system. By doing this and by placing the brake directly on the output shaft, a safe controlled stop is assured even if any of the engine-transmission power train has failed. The brakes are like those presently being manufactured by F. M. C. Corporation.

The brakes are packaged within the transmission but isolated from the rest of the transmission. They will have their own cooling system separate from the transmission.

PERFORMANCE

PERFORMANCE

The performance of the transmission in a vehicle is very basic to the performance of the vehicle. For this reason it is felt that a short discussion of the performance of the 7T-ALH10 differential transmission as it is affected by

1. Basic design
2. Vehicle parameters
3. Environment, and
4. Life

is in order.

Although the vehicle reflects a number of parameters to the transmission - input or engine speed, sprocket speed, sprocket load, ambient temperature, transmission fluid cooling capacity, steering, input power - they all resolve themselves into four basic variables which the transmission sees. These are (1) "A" end and "B" end speed, (2) operating pressure, (3) "A" end stroke, and (4) fluid temperature.

For each vehicle condition there is a corresponding value for each of the transmission parameters.

The design of the transmission including the sizing of the hydraulic elements, gear ratios, and sizing of the auxiliary oil system is based on the worst combination of these parameters.

For the hydromechanical transmission proposed, there are four basic equations which define the operation of the transmission in terms of its parameters.

1. $HP_I = HP_o + HP_{og} + HP_{Ig} + HP_A + HP_{BT} + HP_{AT} + HP_{BL} + HP_{AL}$

2. $T_o = \frac{PD_B}{K_1 75.5} \pm \frac{T_B}{K_1}$ (last term plus if "A" end speed > "B" end speed and minus if "A" end speed < "B" end speed)

3. $N_I K_3 D_A S = D_B N_B + Q_{AL} + Q_{BL}$

4. $N_o = N_B K_1 + N_I K_3 K_2$

Equations 3 and 4 can be combined, however, into the following equation:

5. $N_I \left\{ K_3 D_A S + \frac{D_B K_3 K_2}{K_1} \right\} = \frac{D_B N_o}{K_1} + Q_{AL} + Q_{BL}$

The factors in these equations are defined as follows:

HP_I	= horsepower into transmission
HP_o	= horsepower from transmission
HP_{og}	= horsepower loss of transmission output gears
HP_{Ig}	= horsepower loss of transmission input gears
HP_A	= Horsepower loss of auxiliaries
HP_{BT}	= horsepower loss attributable to "B" end torque loss
HP_{AT}	= horsepower loss attributable to "A" end torque loss
HP_{BL}	= horsepower loss attributable to "B" end leakage loss
HP_{AL}	= horsepower loss attributable to "A" end leakage loss
T_o	= output torque from transmission, ft. lbs.
P	= transmission operating pressure, psi
D_B	= displacement of "B" end, in.^{3}/rev.
T_B	= torque loss of "B" end, ft. lbs.
K_1	= planetary gear ratio - "B" end to output
K_2	= planetary gear ratio - input to output
K_3	= input gear ratio
N_I	= input speed of transmission, RPM
N_o	= output speed of transmission to final drive, RPM
D_A	= displacement of "A" end, in.^{3}/rev.
N_B	= "B" end speed, RPM
N_A	= "A" end speed, RPM
S	= "A" stroke in % divided by 100. (S is positive when $N_I > N_o$ negative when $N_I < N_o$)
Q_{AL}	= "A" end leakage, in.^{3}/rev.
Q_{BL}	= "B" end leakage, in.^{3}/rev.

From these basic equations - 1, 2 and 5 - the basic sizing of the transmission can be made. Obviously, once the basic parameters are determined, then any transmission condition can be determined knowing one or two of the vehicle parameters.

Sizing of the unit is basically a "hunt and try" process because of the large number of variables. Normally the pressure level and the hydraulic units' speed is assumed and in turn the sizes or displacements of the hydraulic units and gear ratios can be initially sized.

However, to finalize the sizing and determine basic efficiency, it is necessary to accurately determine the leakage and torque losses within the unit. Here the basic experience resulting from 15 years in the hydraulic pump/motor/transmission business plays a very important part. This experience backed by many thousands of hours of testing has given the basic parameters to determine: how leakage varies with pressure for a given ball piston size, how torque losses vary with speeds and sizes of the hydraulic units, how temperature of the hydraulic fluid (oil viscosity) affects leakage and torque losses, how pressure affects torque losses and life, losses associated with supply pumps and various types of controls.

Although the majority of the experience has been with hydraulic unit sizes smaller than those to be used in the proposed transmission, the basic relationships between the various parameters have held true.

This has been proven by the testing of the 21.2 cu. in. hydraulic pump/motor, described in another section.

Since efficiency is one of the most important performance criteria of the proposed hydromechanical transmission, it is felt that further elaboration is in order. Figures 16 through 24 are actual test data taken from the 21.2 cu. in./rev. unit and are indicative of the relationships described in the preceding paragraphs.

Figure 16 shows leakage as a function of the operating pressure at constant values of operating speed. The increase in flow due to speed is attributable to the "spin effect" of the ball piston or the effect of the spinning ball wiping oil out of the cylinder. This is only true of a unit acting as a pump. A hydraulic unit acting as a motor tends to act in an opposite fashion, that is, it wipes oil back into the cylinder reducing the apparent leakage.

Figure 17 shows the torque loss of the hydraulic unit as a function of pressure and speed. From this curve it can be seen that pressure does not have a great effect on these losses. The losses result primarily from shearing of oil both in the ball piston bores of the cylinder block and in the journal bearing.

This data, of course, was based on one stroke and one oil temperature. How do the losses vary with stroke of the "A" end unit and with operating oil temperatures or viscosity? Figures 18 through 20 show the effect of leakage loss and torque losses as a function of stroke, pressure and speeds. It can be seen that the leakage losses increase approximately the same amount as stroke increases, regardless of pressure. The torque loss curve, Figure 18, indicates that the torque loss is affected very little by pressure regardless of the stroke.

The effect of oil viscosity changes on leakage and torque losses is again very predictable. It has been determined that the leakage will increase with a decrease in oil viscosity by a factor of $\left(\frac{V_1}{V_2}\right)^{.36}$. and torque losses will decrease with a decrease in oil viscosity by a factor of $\left(\frac{V_2}{V_1}\right)^{.23}$. Figures 21 and 22 give very close correlation with these factors.

It is interesting to note, therefore, that the torque loss decrease and the leakage loss increase tend to balance each other as the oil temperature is increased (viscosity is decreased). In fact if we examine the 40 MPH vehicle speed point on the proposed transmission, the efficiency tends to increase as oil temperature increases. This happens because the operating pressure is very low at this point causing a corresponding low leakage loss. The major portion of the total transmission losses at this point are torque losses. An increase in leakage due to higher oil temperature does not materially increase this leakage loss but does change the torque loss more drastically. This effect is shown in Figure 23 which shows this increase in efficiency.

In like manner, any effect from stroke is minimized at the critical vehicle design points, 2, 2-1/2, and 40 MPH, because of course the "B" end always runs at full stroke and the "A" end stroke is 100% at 40 MPH and above 50% stroke at the lower vehicle speeds. Overall mechanical efficiency of a pump unit as a function of stroke is shown in Figure 24.

In summary, it is felt that the indicated efficiency of the proposed transmission is accurate. It is based on factual, conservative, and proven values based on years of past experience with all types of hydraulic elements and confirmed by test data on a hydraulic element close to the size of the proposed transmission's hydraulic elements, with even more importantly equivalent speeds, pressures, and temperatures.

The only other factor involving performance not discussed in the preceding paragraphs is life. This is discussed in the following pages.

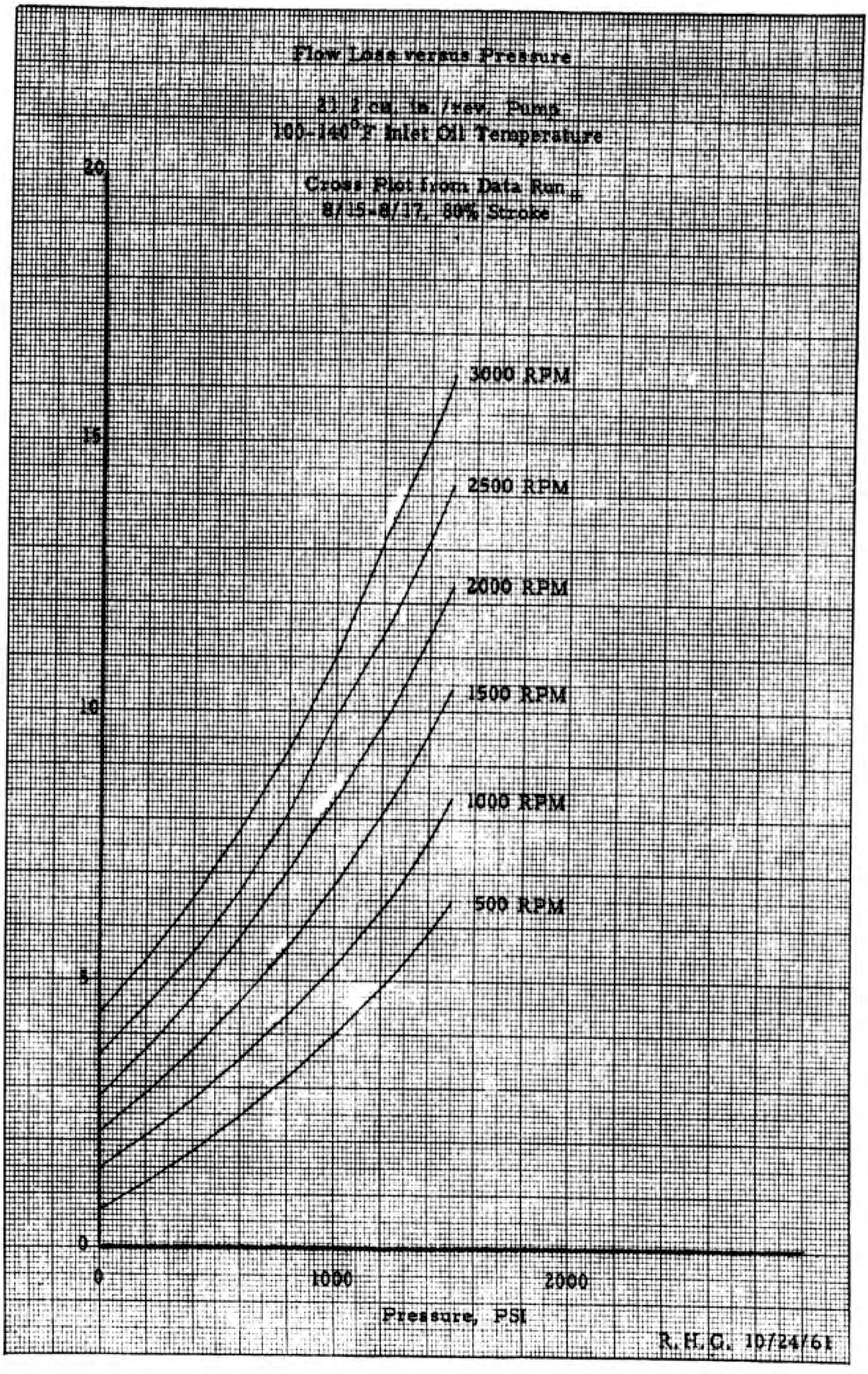
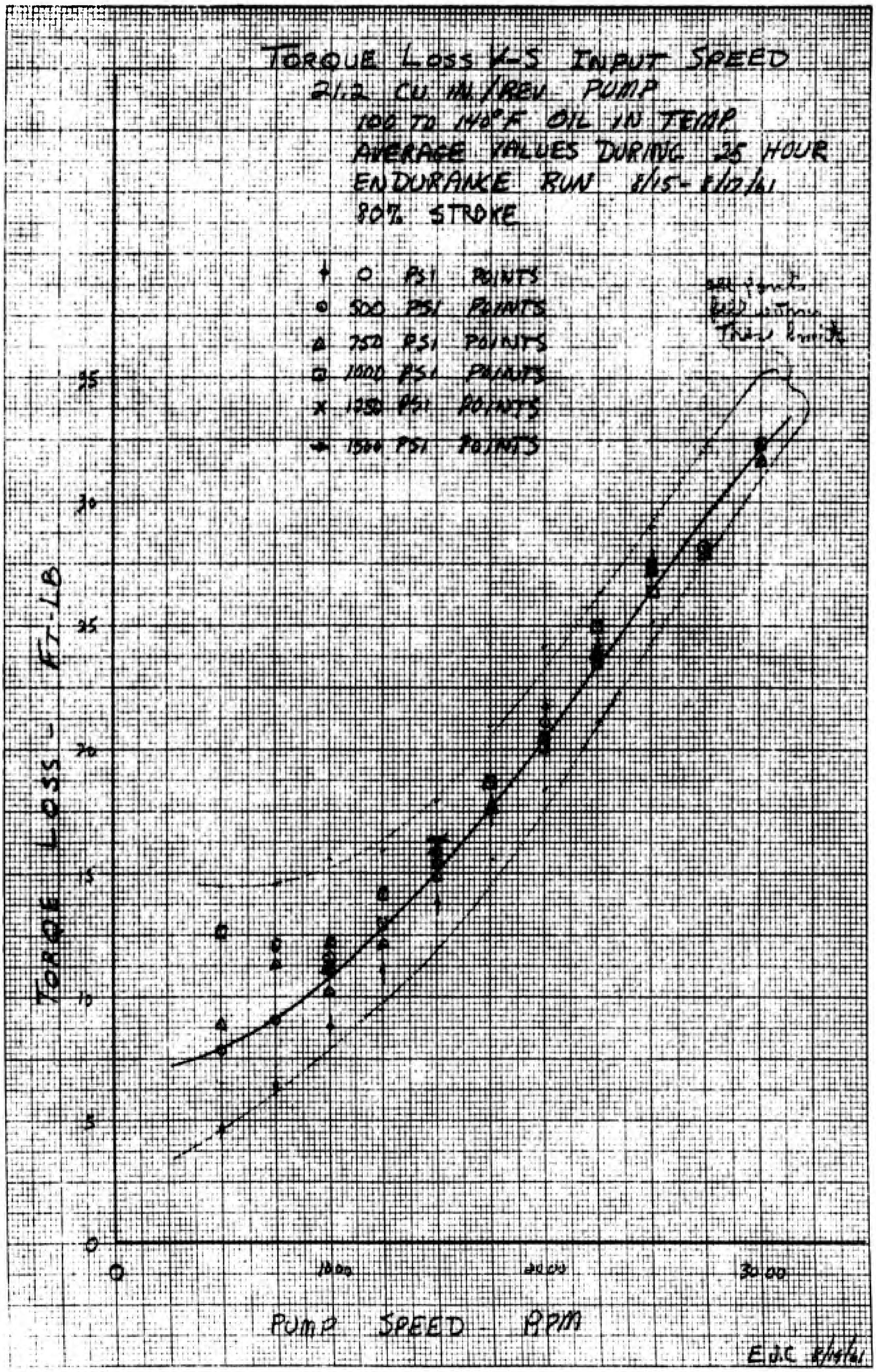

Flow Loss versus Pressure
100-140°F Inlet Oil Temperature
Cross Plot from Data Run
8/15-8/17, 80% Stroke
3000 RPM
2500 RPM
2000 RPM
1500 RPM
1000 RPM
500 RPM
20
15
10
5
0
0
1000
2000
Pressure, PSI
R. H. G. 10/24/61

Figure 16



TORQUE LOSS V-S INPUT SPEED
21.2 CU IN/REV PUMP
100 TO 140°F OIL IN TEMP
AVERAGE VALUES DURING 25 HOUR
ENDURANCE RUN
80% STROKE
0 PSI POINTS
500 PSI POINTS
750 PSI POINTS
1000 PSI POINTS
1250 PSI POINTS
1500 PSI POINTS
TORQUE LOSS - FT-LB
PUMP SPEED - RPM
0
1000
2000
3000
0
5
10
15
20
25
30
35

Figure 17

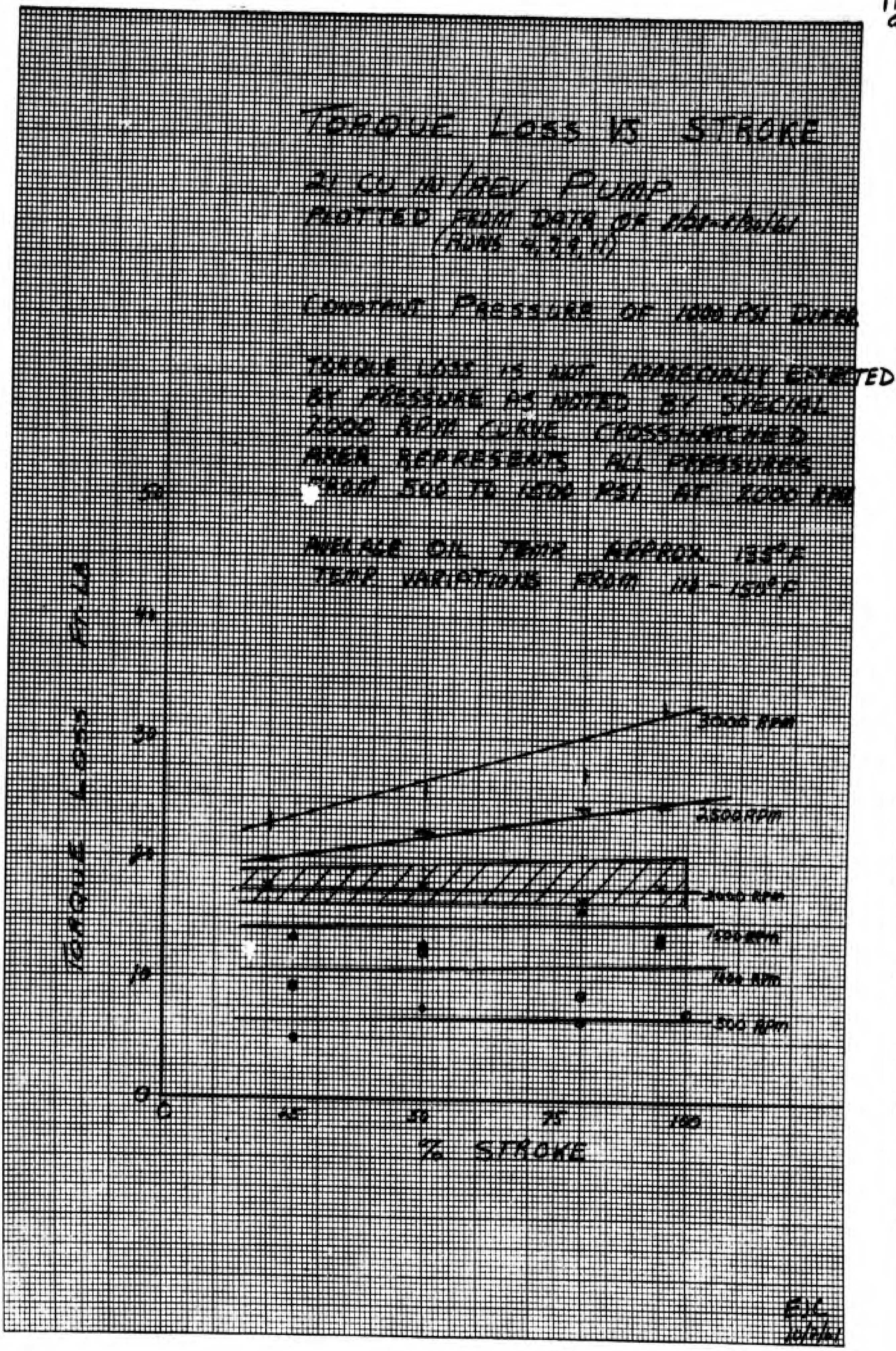

TORQUE LOSS VS STROKE
21 CU IN/REV PUMP
PLOTTED FROM DATA OF
(RUNS 4,7,9,11)
CONSTANT PRESSURE OF 1000 PSI DIFF.
TORQUE LOSS IS NOT APPRECIABLY EFFECTED
BY PRESSURE AS NOTED BY SPECIAL
2000 RPM CURVE. CROSSHATCHED
AREA REPRESENTS ALL PRESSURES
FROM 500 TO 1500 PSI AT 2000 RPM
AVERAGE OIL TEMP APPROX 135°F
TEMP VARIATIONS FROM 114-150°F
TORQUE LOSS FT-LB
% STROKE
3000 RPM
2500 RPM
2000 RPM
1500 RPM
1000 RPM
500 RPM
0
10
20
30
40
50
25
50
75
100

Figure 18

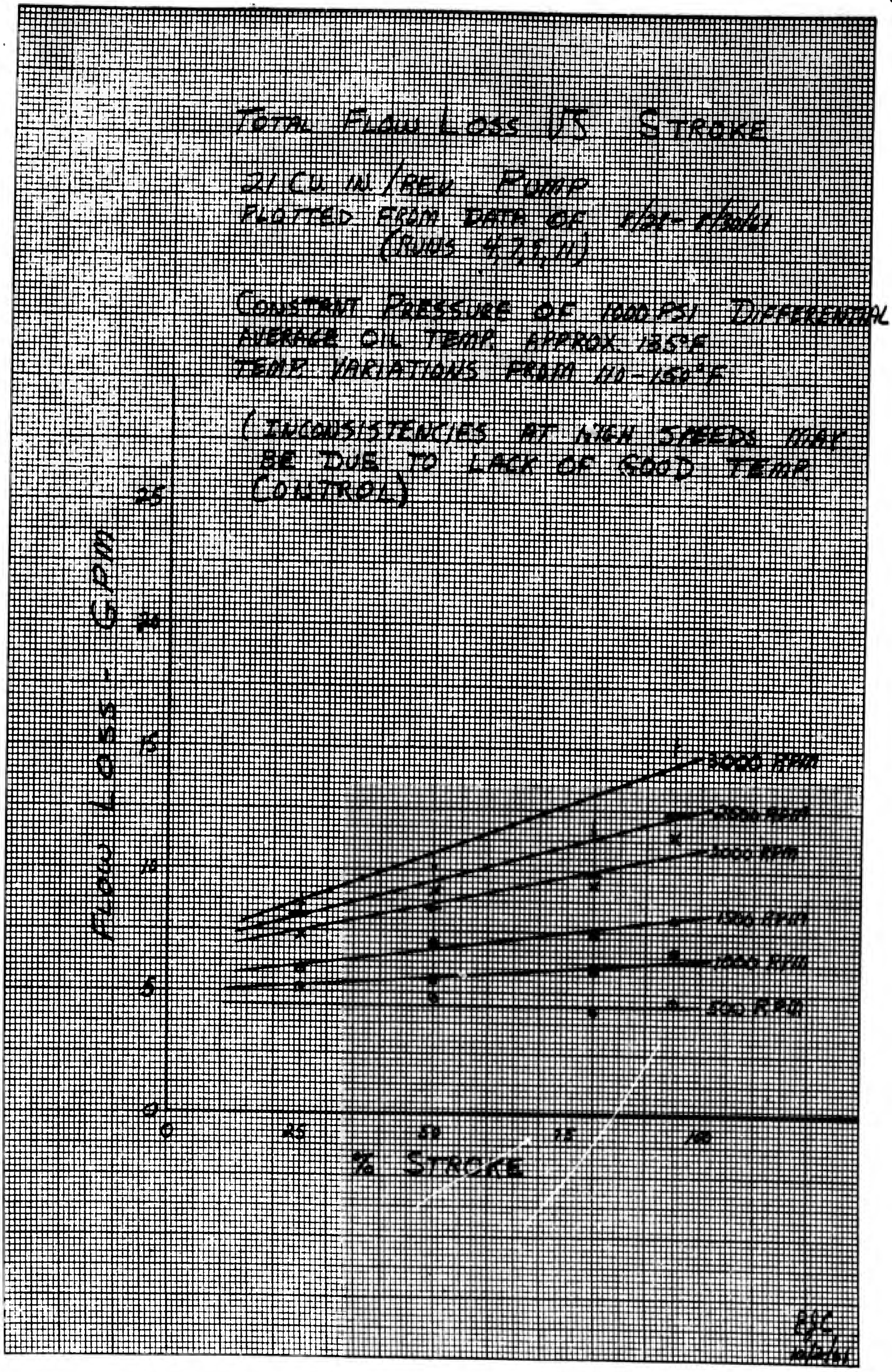
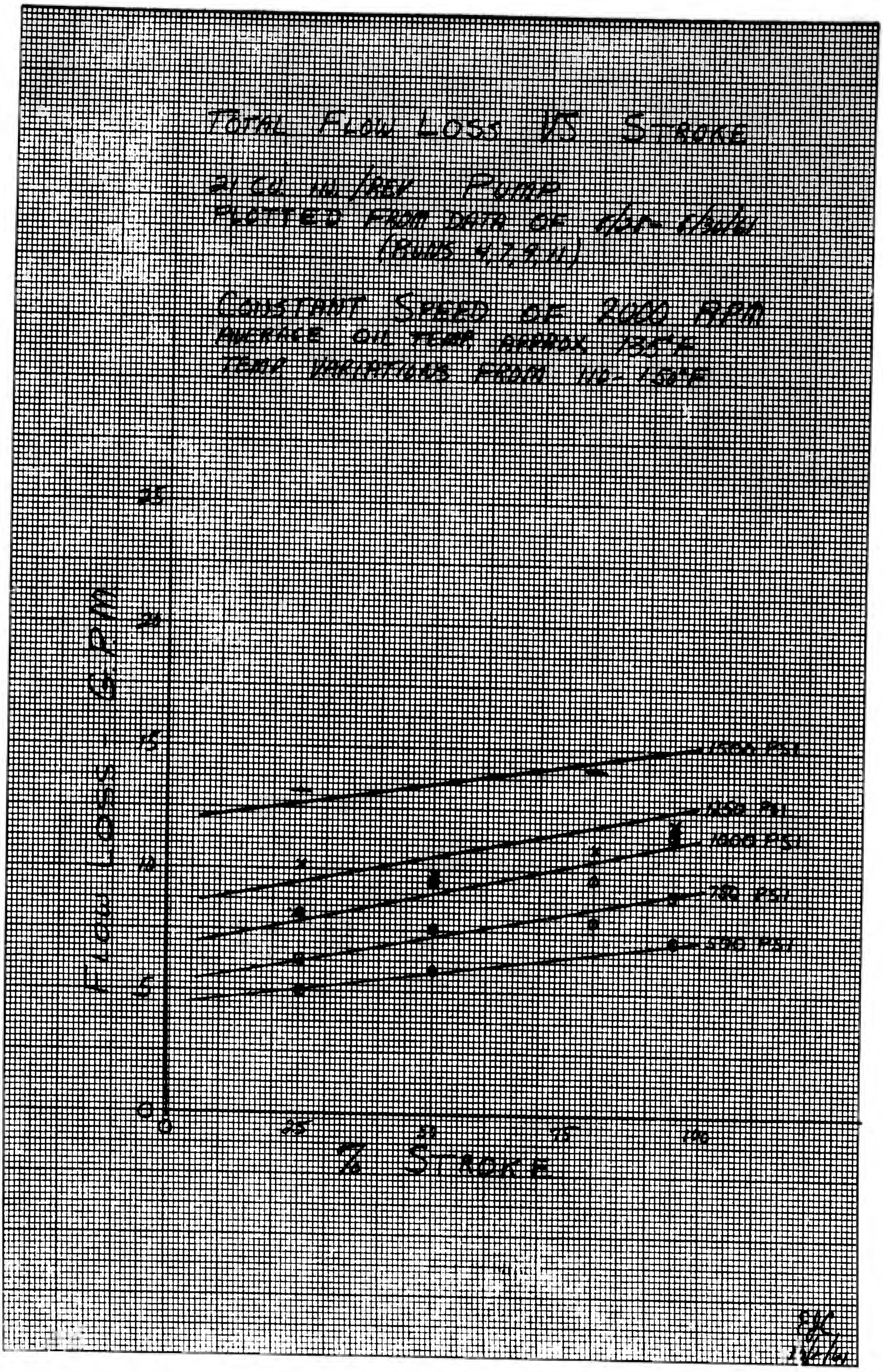

TOTAL FLOW LOSS VS STROKE
21 CU IN/REV PUMP
PLOTTED FROM DATA OF
(RUNS 4,7,9,11)
CONSTANT PRESSURE OF 1000 PSI DIFFERENTIAL
AVERAGE OIL TEMP. APPROX. 135°F
TEMP VARIATIONS FROM 110-150°F
(INCONSISTENCIES AT HIGH SPEEDS MAY BE DUE TO LACK OF GOOD TEMP. CONTROL)
FLOW LOSS - GPM
25
20
15
10
5
0
0
25
50
75
100
% STROKE
3000 RPM
2000 RPM
1500 RPM
1000 RPM
500 RPM

Figure 19



TOTAL FLOW LOSS VS STROKE
21 CU. IN./REV PUMP
PLOTTED FROM DATA OF
(RUNS 4,7,9,11)
CONSTANT SPEED OF 2000 RPM
AVERAGE OIL TEMP. APPROX 135°F
TEMP. VARIATIONS FROM 110-150°F
FLOW LOSS - G.P.M.
25
20
15
10
5
0
1500 PSI
1250 PSI
1000 PSI
750 PSI
500 PSI
25
50
75
100
% STROKE

Figure 20

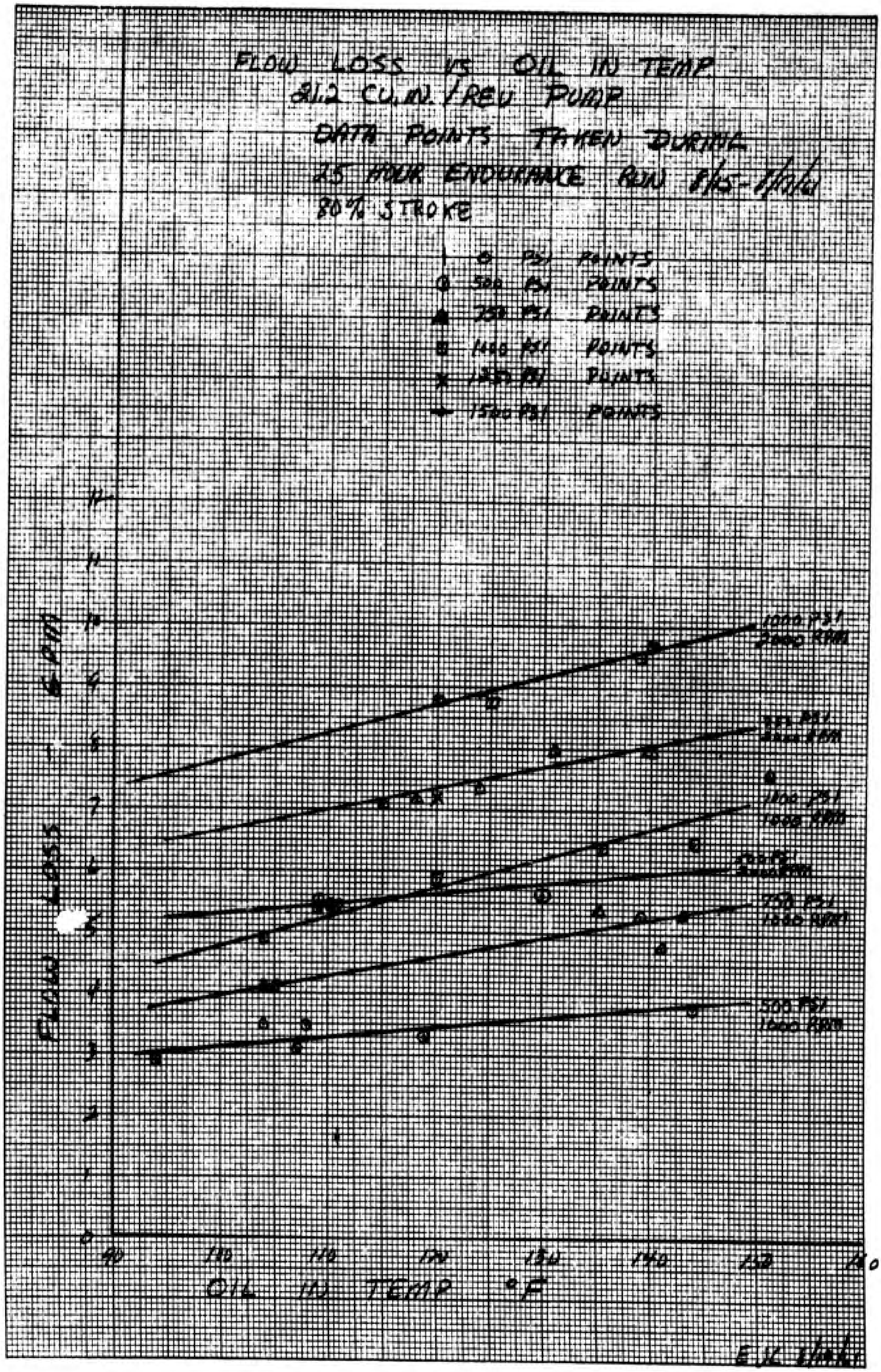

FLOW LOSS VS OIL IN TEMP
21.2 CU. IN. / REV PUMP
DATA POINTS TAKEN DURING
25 HOUR ENDURANCE RUN
80% STROKE
0 PSI POINTS
500 PSI POINTS
750 PSI POINTS
1000 PSI POINTS
1250 PSI POINTS
1500 PSI POINTS
1000 PSI 2000 RPM
1000 PSI 1000 RPM
750 PSI 1000 RPM
500 PSI 1000 RPM
FLOW LOSS GPM
OIL IN TEMP °F

Figure 21

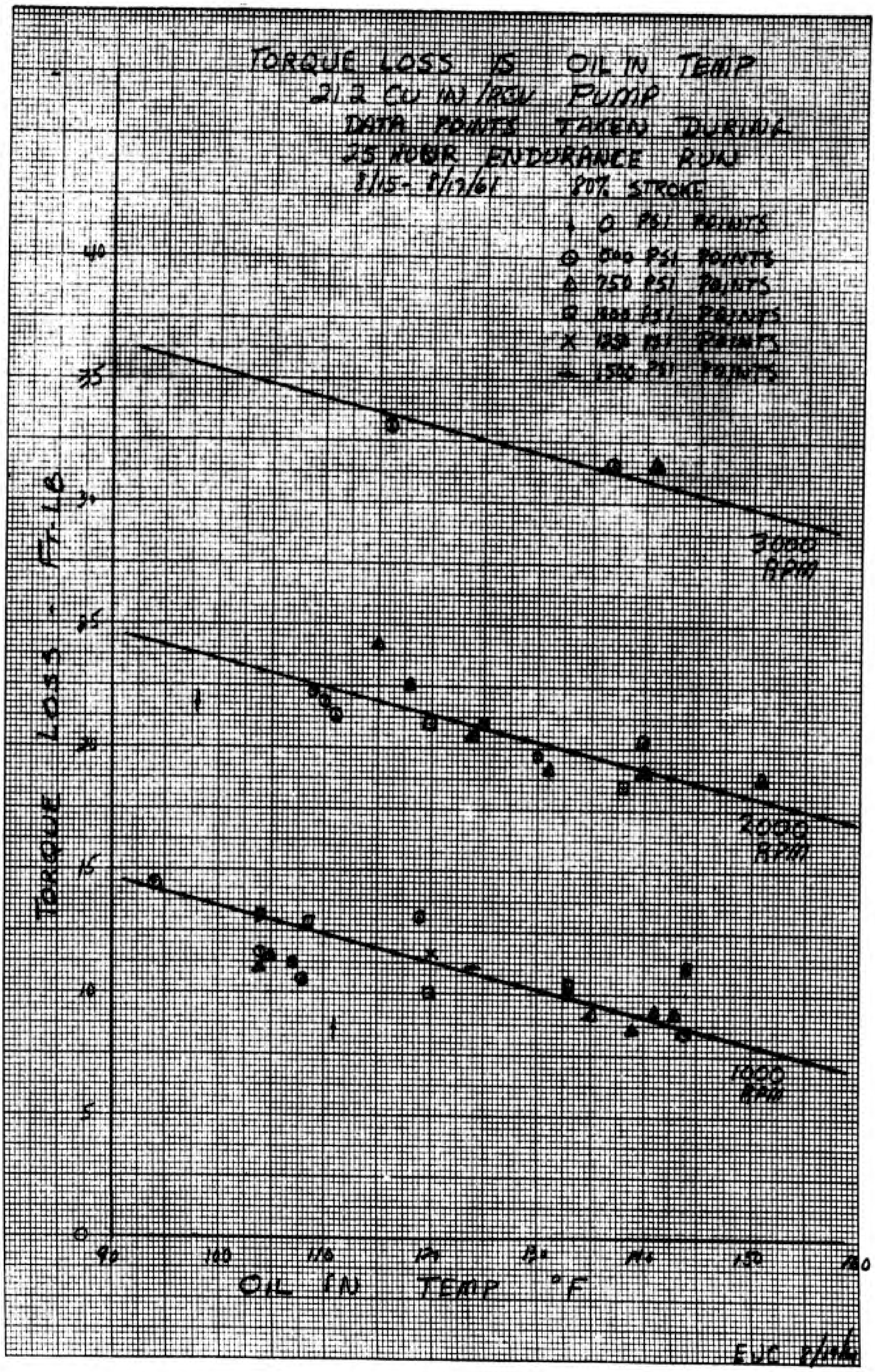

TORQUE LOSS VS OIL IN TEMP
21.2 CU IN/REV PUMP
DATA POINTS TAKEN DURING
25 HOUR ENDURANCE RUN
8/15 - 8/17/61
80% STROKE
0 PSI POINTS
500 PSI POINTS
750 PSI POINTS
1000 PSI POINTS
1250 PSI POINTS
1500 PSI POINTS
3000 RPM
2000 RPM
1000 RPM
TORQUE LOSS - FT-LB
OIL IN TEMP °F
0
5
10
15
20
25
30
35
40
90
100
110
120
130
140
150
160

Figure 22

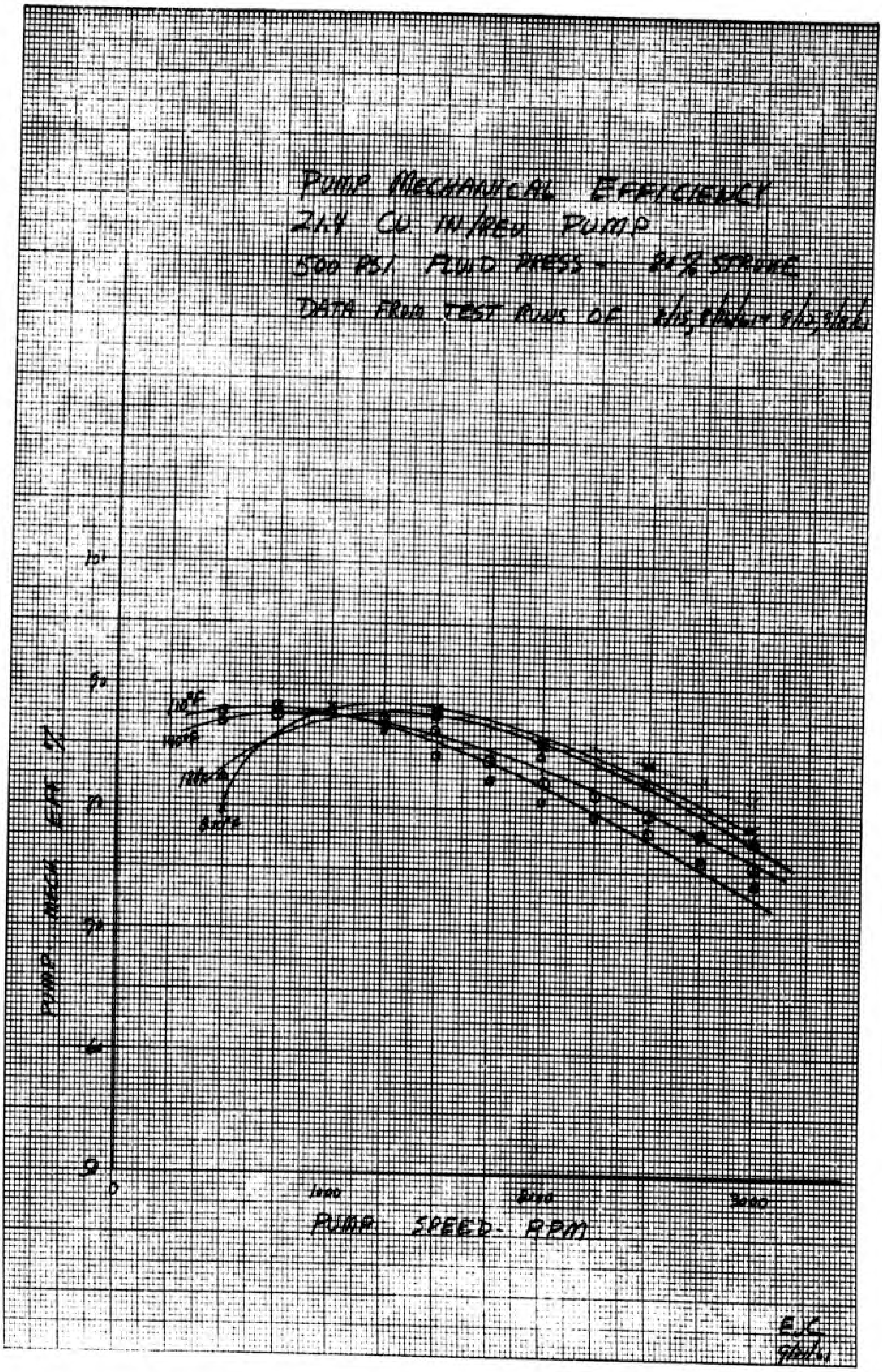

PUMP MECHANICAL EFFICIENCY
2.4 CU IN/REV PUMP
500 PSI FLUID PRESS - 24% STROKE
PUMP MECH EFF %
PUMP SPEED - RPM
0
1000
2000
3000
E.J.C.

Figure 23

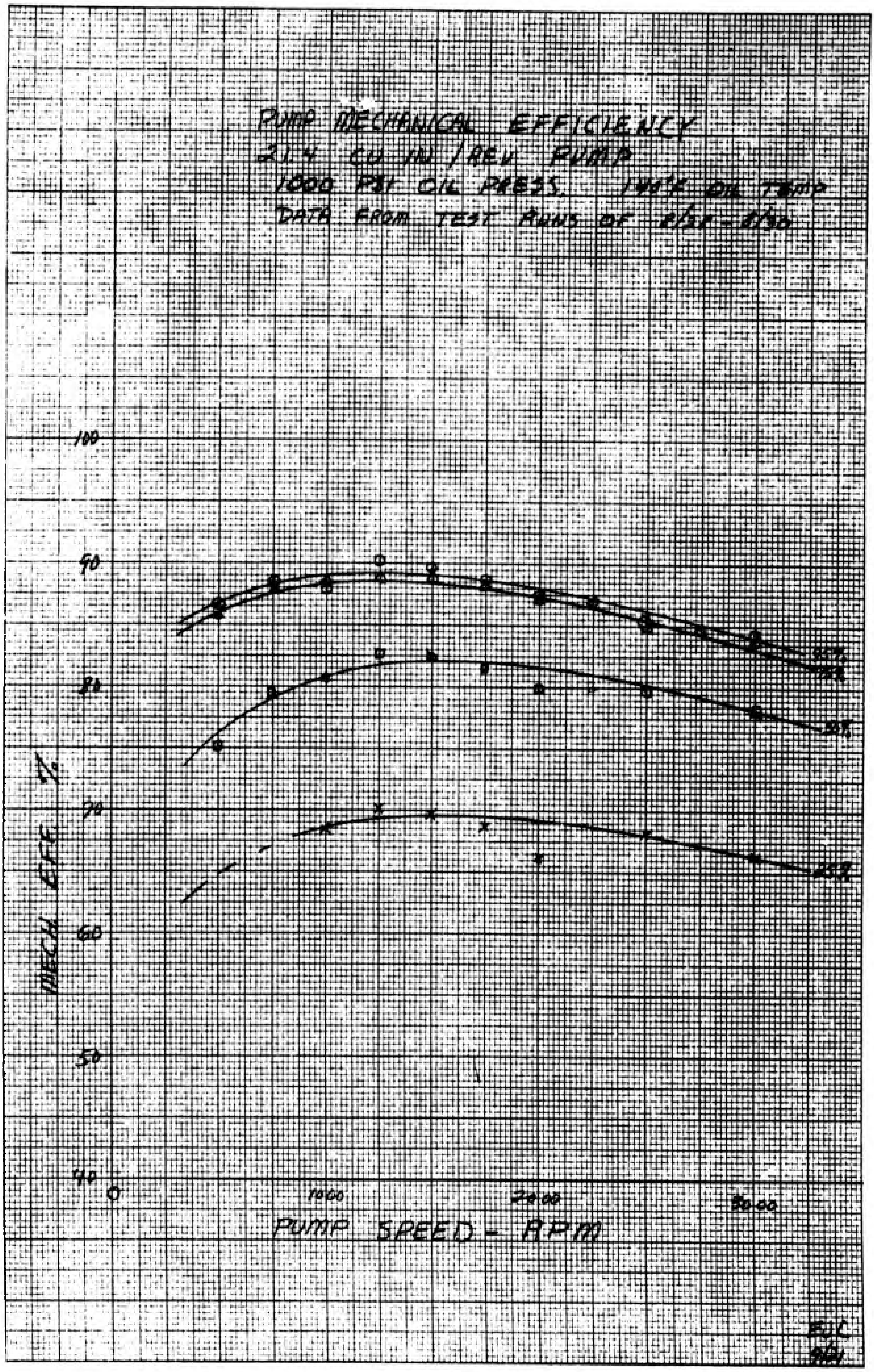

PUMP MECHANICAL EFFICIENCY
2.14 CU IN / REV PUMP
1000 PSI OIL PRESS. 140°F OIL TEMP
DATA FROM TEST RUNS OF
100
90
80
70
60
50
40
0
1000
2000
3000
PUMP SPEED - RPM
MECH EFF %
25%

Figure 24

LIFE EFFECT ON PERFORMANCE

Over one million flight hours have been accumulated on General Electric Constant Speed Transmissions. These hours have proven that there is little or no performance (particularly efficiency) deterioration over the overhaul periods established for these drives. The overhaul periods vary from 720 hours for the 9 KVA drive to a present T.B.O. for the 40 KVA drive of 1500 hours.

A 500 hour qualification life test was run on the 40 KVA drive to an accelerated load, speed, and temperature schedule. Inspection of the unit after the test indicated the performance was well within the allowable limits. Actual measurements of the clearances of the "A" end indicated an increase of 0 to 0.000004 inches and "B" end clearance increase of 0.000007 to 0.000013 inches. A good portion of these changes were within measurement accuracies.

Also on the 40 KVA drive over 100 units have been overhauled at the General Electric Los Angeles Service Shop. None of these drives had any indications of excessive wear or performance deterioration.

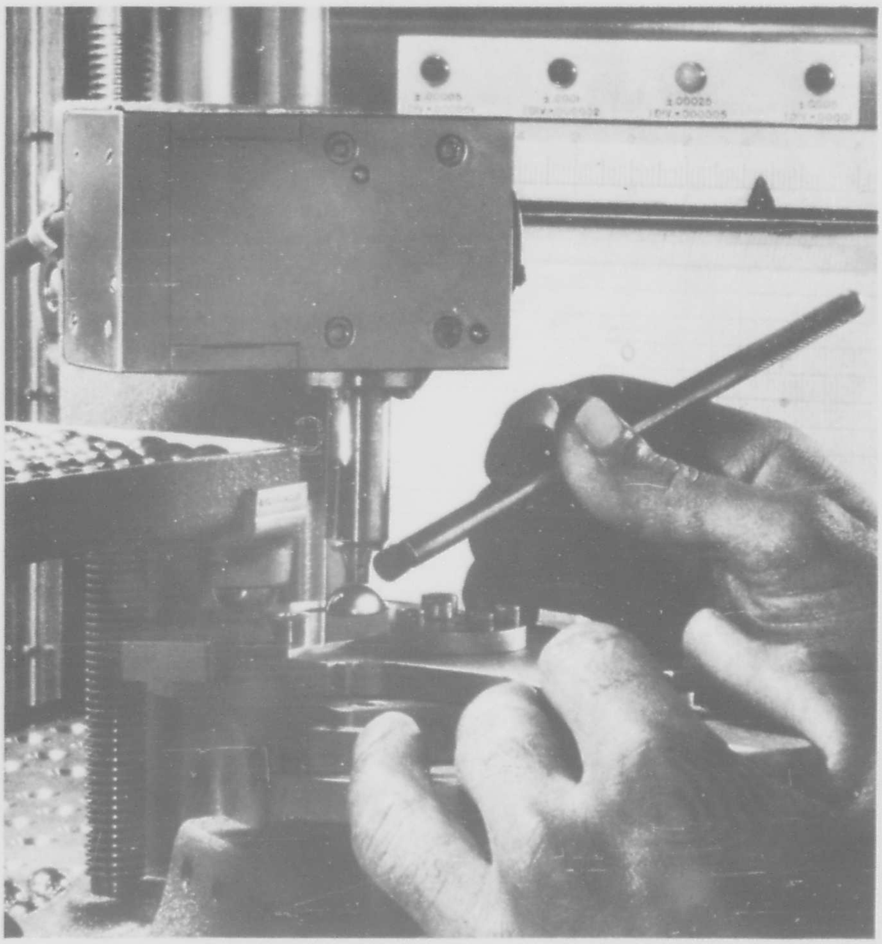
It is interesting to note that based on present experience it is expected that only 4.5% of the ball pistons in the 40 KVA drive would be replaced at the end of 2500 hours of operation.

Another significant test, run on the 40 KVA drive during the development phase, indicated the relatively little wear of the hydraulic elements. This test consisted of 2200 hours of endurance testing to a normal airline schedule. After completion of the 2200 hours the wear on the balls or cylinders were insignificant.

In all of these applications including the larger transmissions as proposed herein, the ball piston spin speed and "in-and-out" velocity are approximately the same. Oil temperatures and pressures in the aircraft applications are equivalent if not higher than those required for the proposed vehicle transmission. It is therefore felt that changes in efficiency of the proposed transmission over 1000 hours of operation would be negligible.

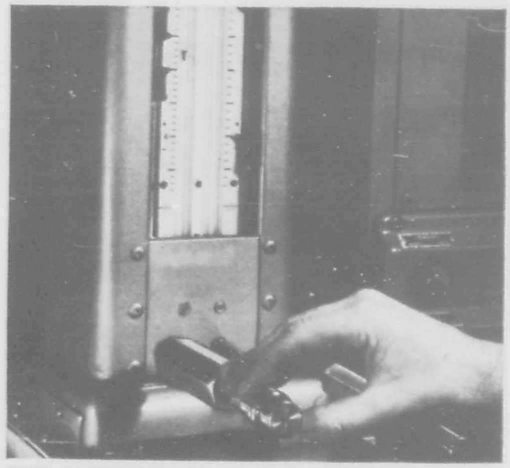
20 KVA CONSTANT SPEED TRANSMISSION LIFE TEST

On one aircraft application for the 20 KVA transmission, the 500 hour qualification life test indicated no change in efficiency during the 500 hours. The test was operated over a range of speeds and loads (greatest percentage of time at full load) and at

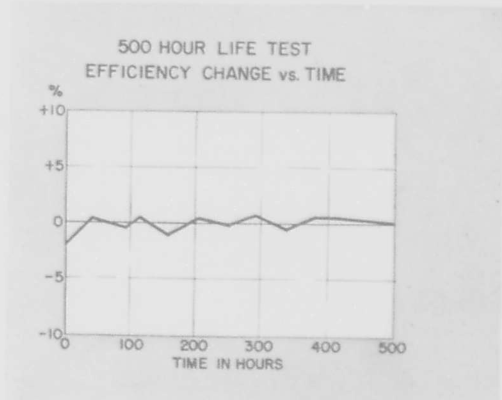


BALL PISTONS used in General Electric Hydraulic Constant Speed Drive show less than .000020 of an inch of wear after 500 hours of continuous operation. Wear is so minute a highly sensitive electronic device is needed for detection.

oil temperatures of from $250^{\circ}F$ - $315^{\circ}F$. The clearance between the ball pistons and the cylinders showed no significant changes. Based on our findings it was felt that the drive could have been operated another 500 hours with little or no loss in efficiency.



INNER DIMENSIONS of ball piston sleeve are well within tolerances after 500-hour test as shown by precision air gauge.



TEST DATA shows no measurable decrease in efficiency of G-E hydraulic constant speed drive after 500 hours of operation.

GENERAL ELECTRIC HYDROSTATIC TRANSMISSION DEVELOPMENT PROGRAM

To support General Electric's entry into the hydrostatic transmission market the Aircraft Accessory Turbine Department is investing this year approximately a quarter of a million dollars of available development funds. This program is comprised of four major engineering activities as follows:

System Analysis and Preliminary Design

Fundamental system studies utilizing analog computer techniques to thoroughly explore hydraulic unit performance and controls characteristics peculiar to vehicle power transmission systems. Configuration studies to optimize mechanical designs.

Cost Reduction

Hardware development program to endurance and performance test a number of promising new materials and design innovations designed to reduce the production cost of basic ball piston elements.

High Pressure Element Development

Hardware development program to perfect a simple hydrostatic element capable of higher operating pressures than the ball unit.

Large Displacement Ball Piston Element Development

Hardware development program in direct support of hydrostatic individual wheel drive proposal activity. Presently, a 21.2 cu. in. /rev.

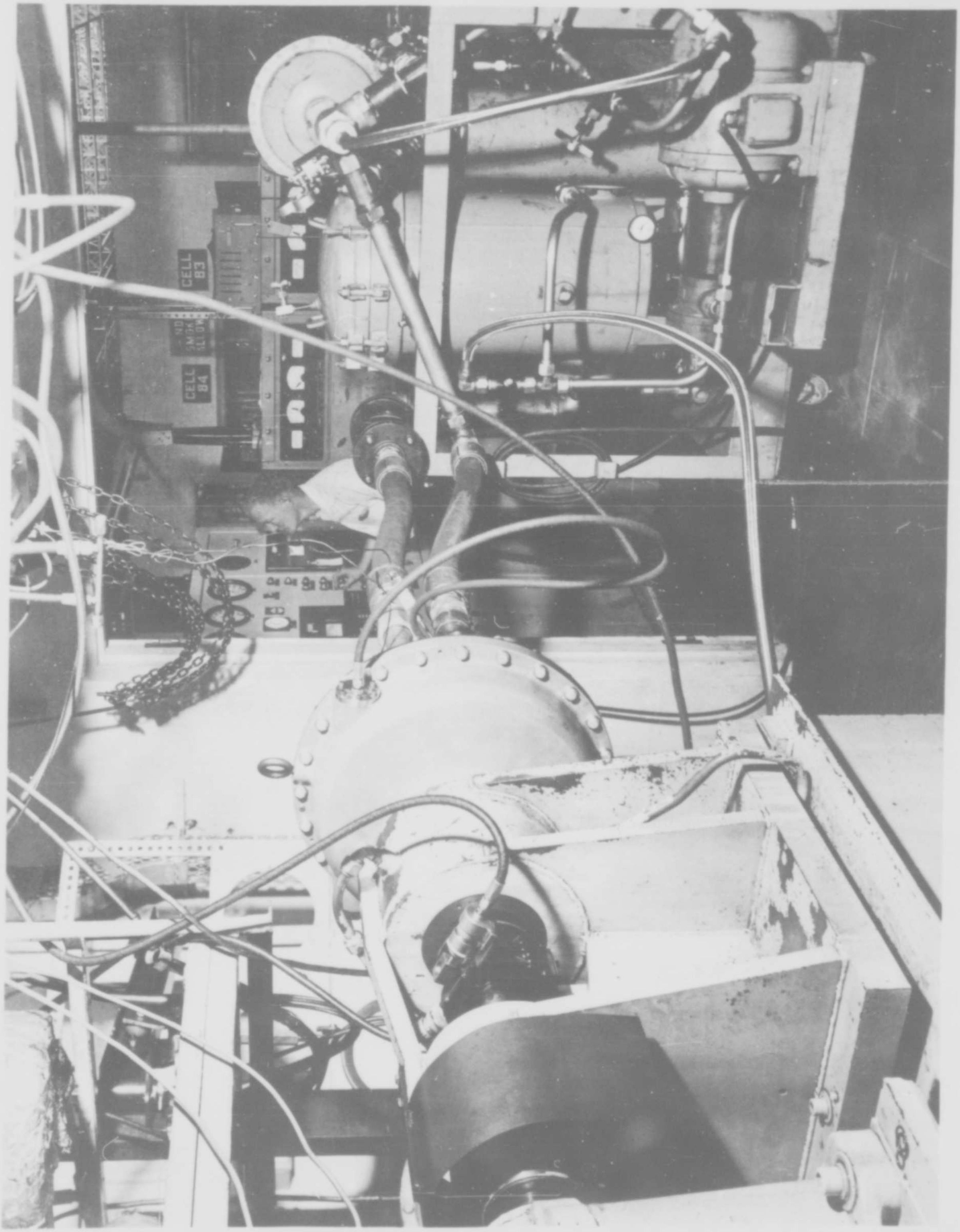
pump-motor unit is undergoing extensive performance and endurance testing. Also promising new cost reduction features developed through test on smaller test vehicles are being tested on this full scale unit. Figure 25 shows this unit prior to assembly and Figure 26 shows it on test in one of our test cells.

At present approximately one hundred hours have been accumulated on this unit with excellent results. The data confirms the predicted performance for the pump within about plus or minus one percent. Performance curves for the unit for speeds up to 3250 RPM and 1500 psi are shown on Figures 27 through 32. Additional testing at higher pressures and oil temperatures and at lower strokes is in process. Motor tests are also to be run in the near future.

A smaller 11.05 cu. in./rev. pump-motor is being built now for similar testing at the conclusion of the present series of tests on the 21.2 cu. in. unit.

AAT 9007

FIG. 25



AAT 9110

FIG. 26

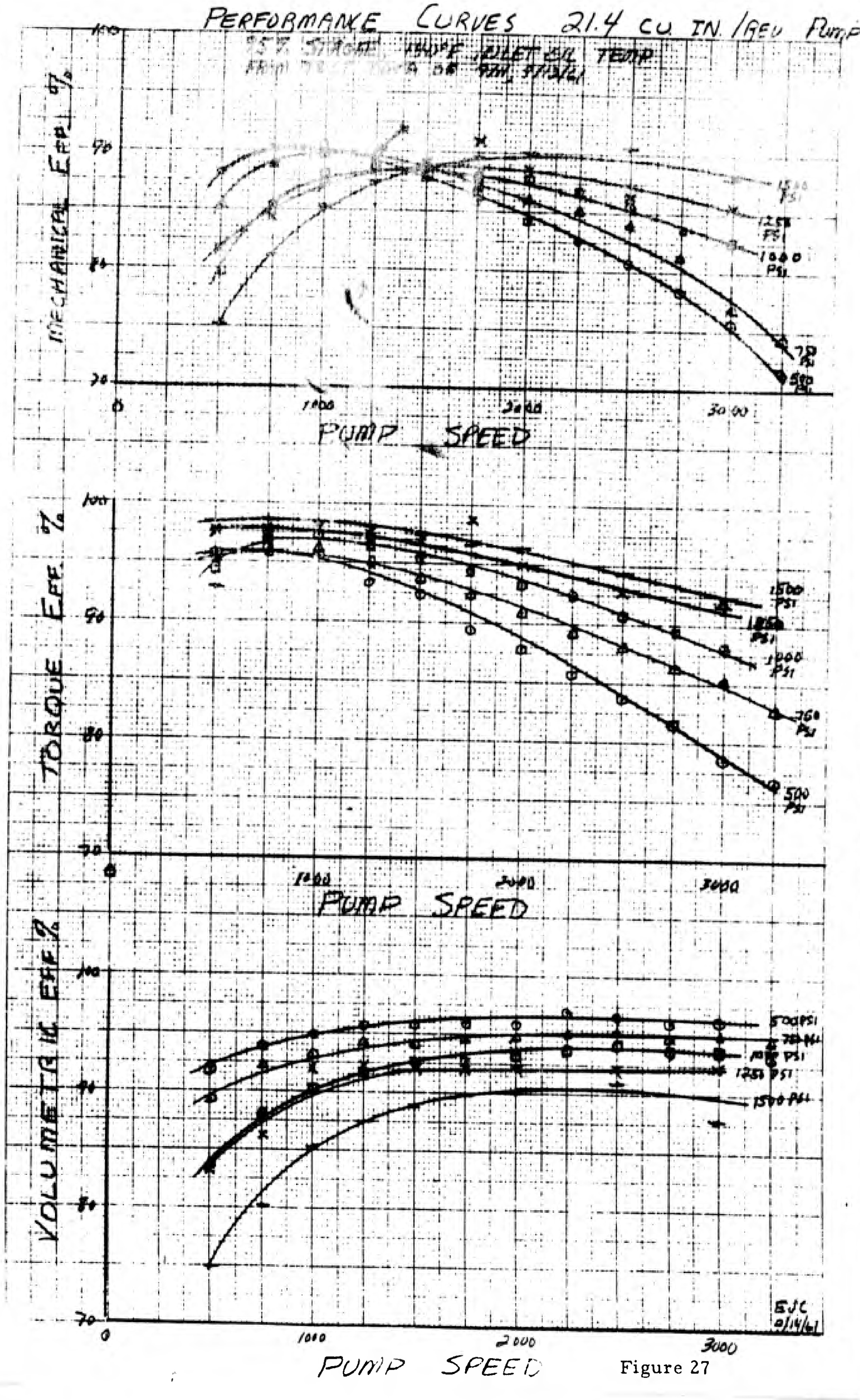

PERFORMANCE CURVES 21.4 CU. IN./REV PUMP
MECHANICAL EFF. %
PUMP SPEED
TORQUE EFF. %
PUMP SPEED
VOLUMETRIC EFF. %
EJC
PUMP SPEED

Figure 27

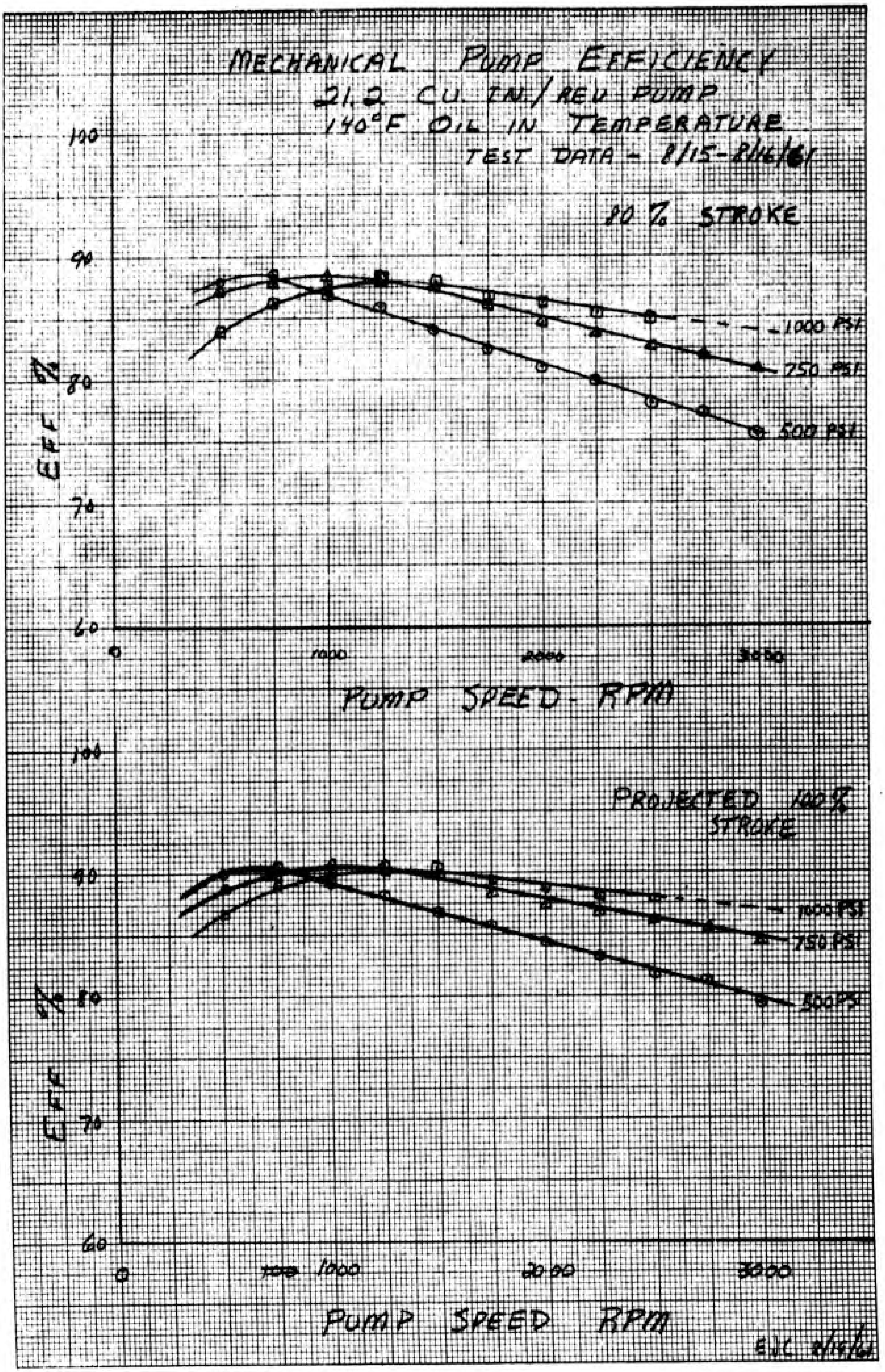
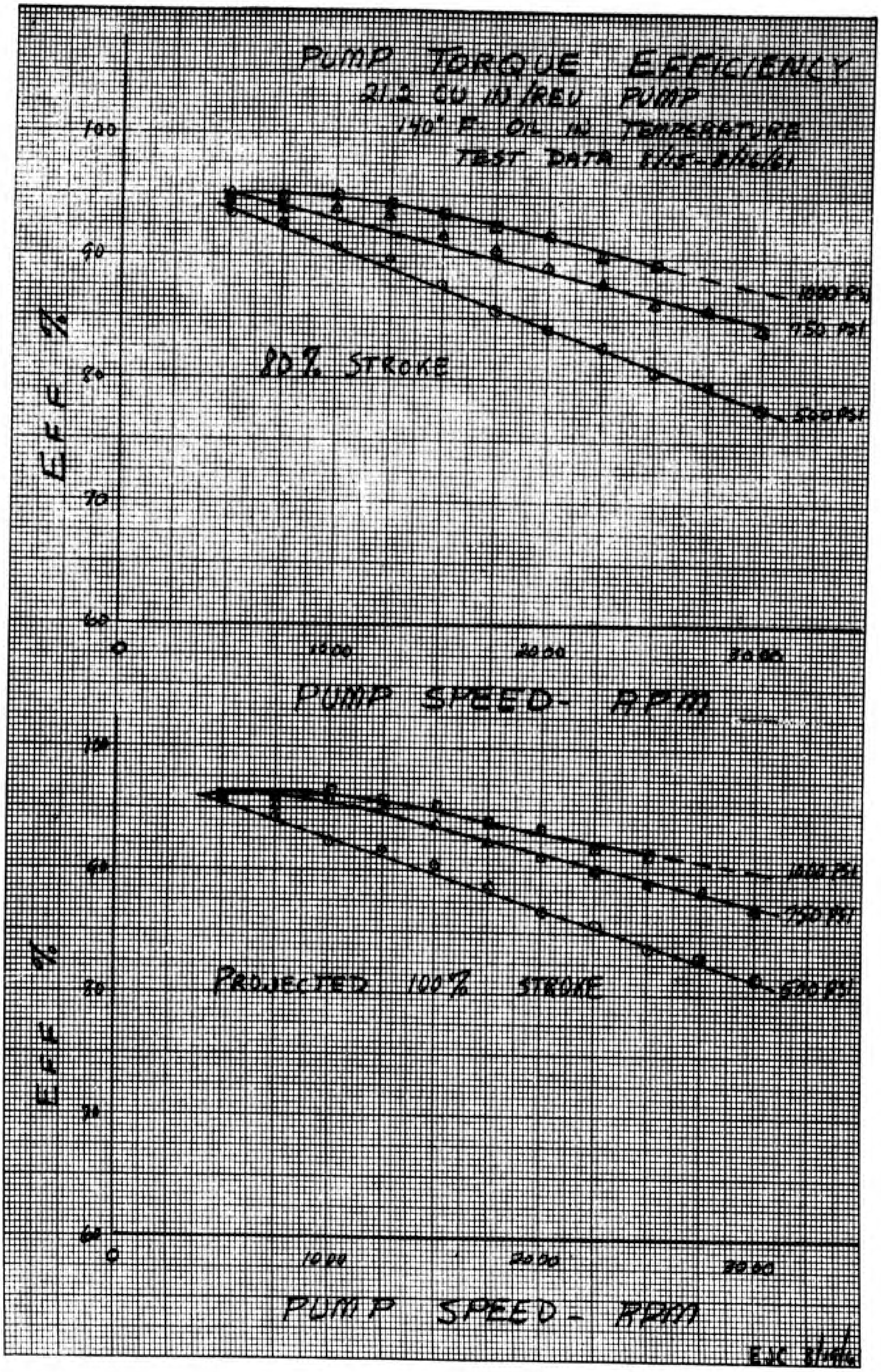

MECHANICAL PUMP EFFICIENCY
21.2 CU IN/REV PUMP
140°F OIL IN TEMPERATURE
TEST DATA - 8/15-8/16/61
80 % STROKE
1000 PSI
750 PSI
500 PSI
EFF %
100
90
80
70
60
0
1000
2000
3000
PUMP SPEED - RPM
PROJECTED 100% STROKE
1000 PSI
750 PSI
500 PSI
EFF %
100
90
80
70
60
0
1000
2000
3000
PUMP SPEED RPM
EJC 8/16/61

Figure 28



PUMP TORQUE EFFICIENCY
21.2 CU IN/REV PUMP
140° F OIL IN TEMPERATURE
TEST DATA
EFF %
100
90
80
70
60
0
1000
2000
3000
80% STROKE
1000 PSI
750 PSI
500 PSI
PUMP SPEED - RPM
PROJECTED 100% STROKE
EJC

Figure 29

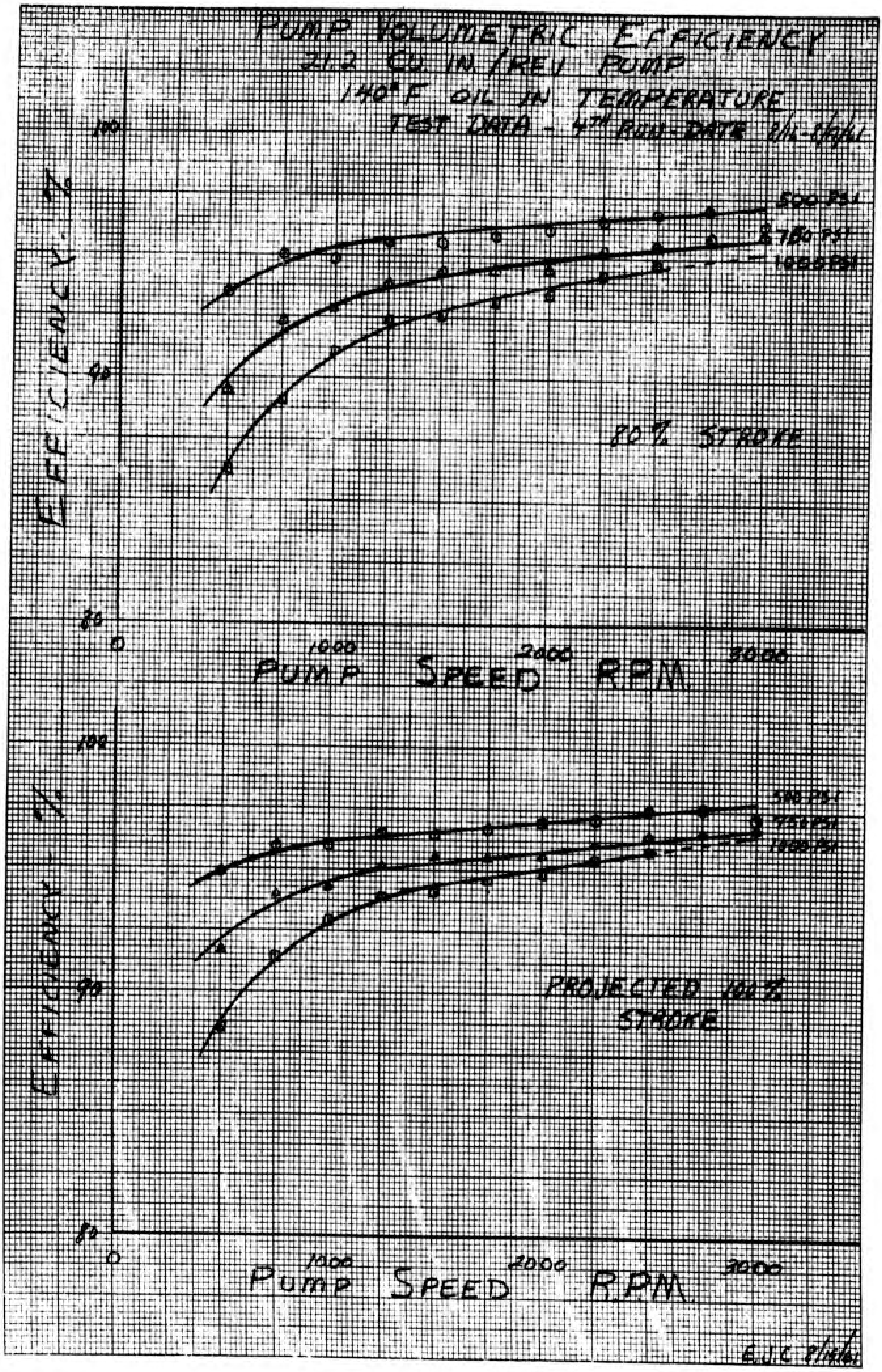

PUMP VOLUMETRIC EFFICIENCY
21.2 CU. IN./REV PUMP
140°F OIL IN TEMPERATURE
TEST DATA - 4TH RUN - DATE
EFFICIENCY %
500 PSI
750 PSI
1000 PSI
80% STROKE
100
90
80
0
1000
2000
3000
PUMP SPEED RPM
EFFICIENCY %
500 PSI
750 PSI
1000 PSI
PROJECTED 100% STROKE
PUMP SPEED RPM
E.J.C.

Figure 30

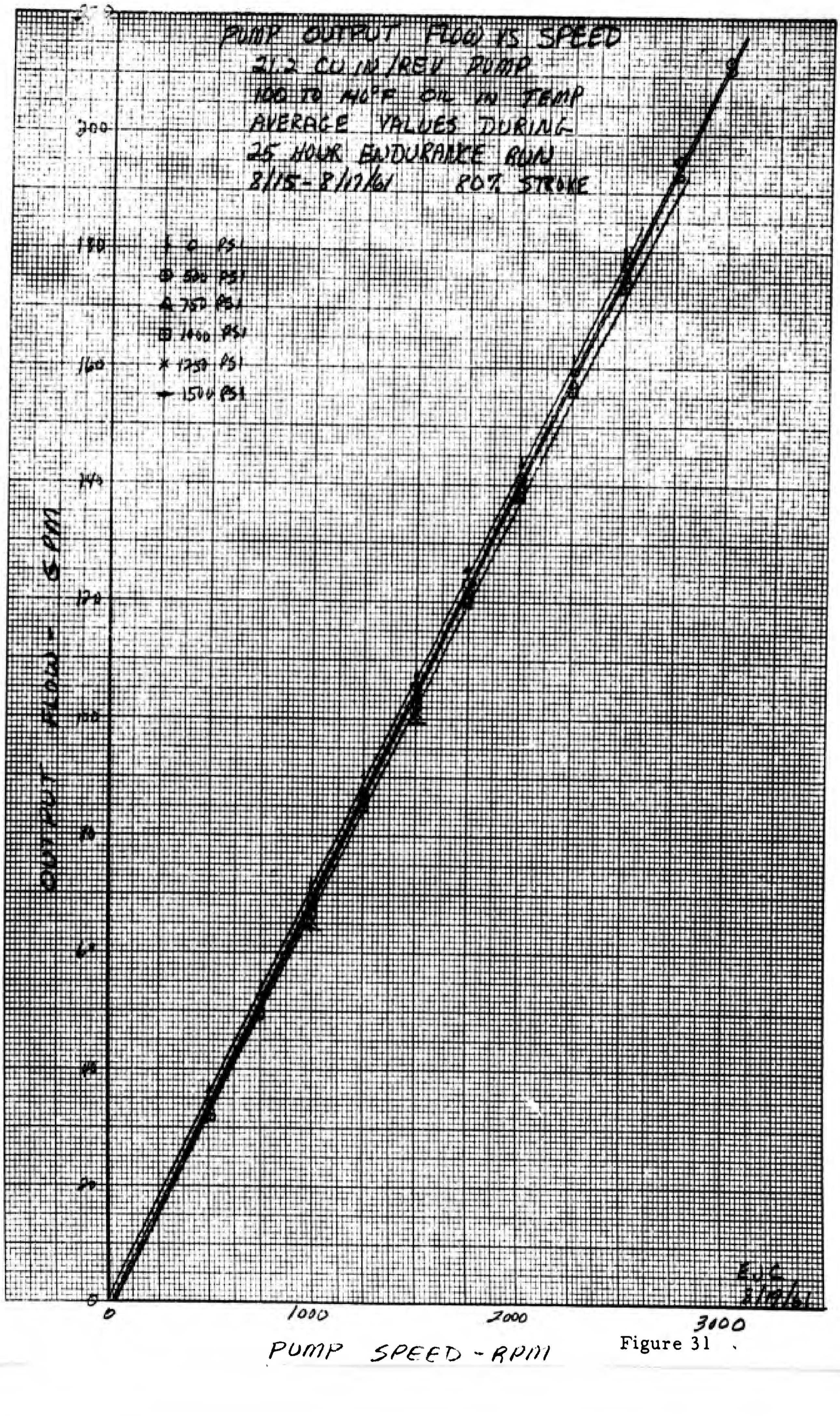

PUMP OUTPUT FLOW VS SPEED
21.2 CU IN/REV PUMP
100 TO 140°F OIL IN TEMP
AVERAGE VALUES DURING
25 HOUR ENDURANCE RUN
8/15 - 8/17/61 80% STROKE
0 PSI
500 PSI
750 PSI
1000 PSI
1250 PSI
1500 PSI
OUTPUT FLOW - GPM
PUMP SPEED - RPM
0
1000
2000
3000
200
180
160
140
120
100
80
60
40
20
E.J.C.
8/19/61

Figure 31

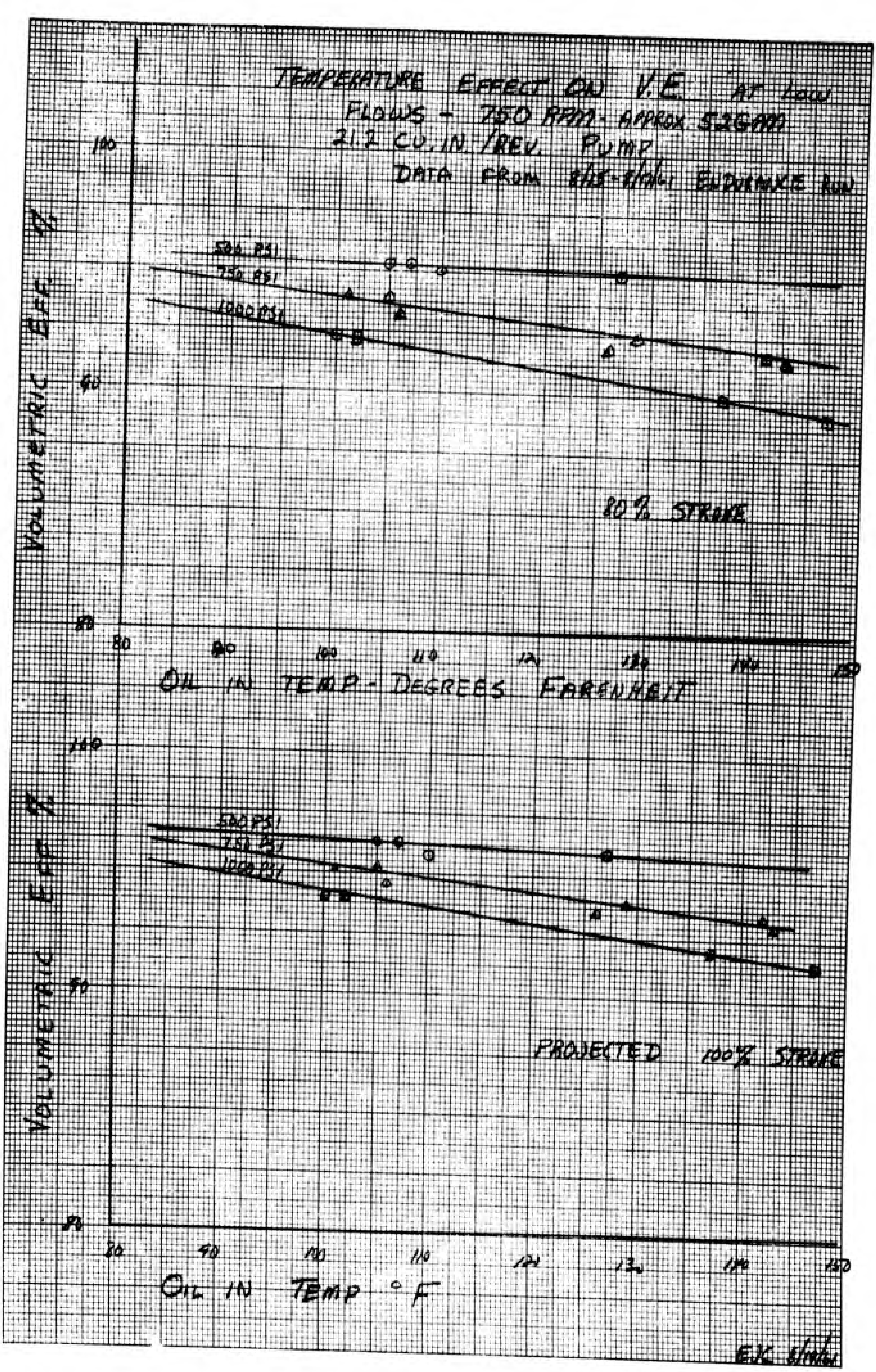

TEMPERATURE EFFECT ON V.E. AT LOW FLOWS - 750 RPM - APPROX. 5.2 GPM
21.2 CU. IN./REV. PUMP
DATA FROM 8/15-8/16/61 ENDURANCE RUN
VOLUMETRIC EFF. %
500 PSI
750 PSI
1000 PSI
80% STROKE
OIL IN TEMP - DEGREES FARENHEIT
VOLUMETRIC EFF %
PROJECTED 100% STROKE
OIL IN TEMP °F

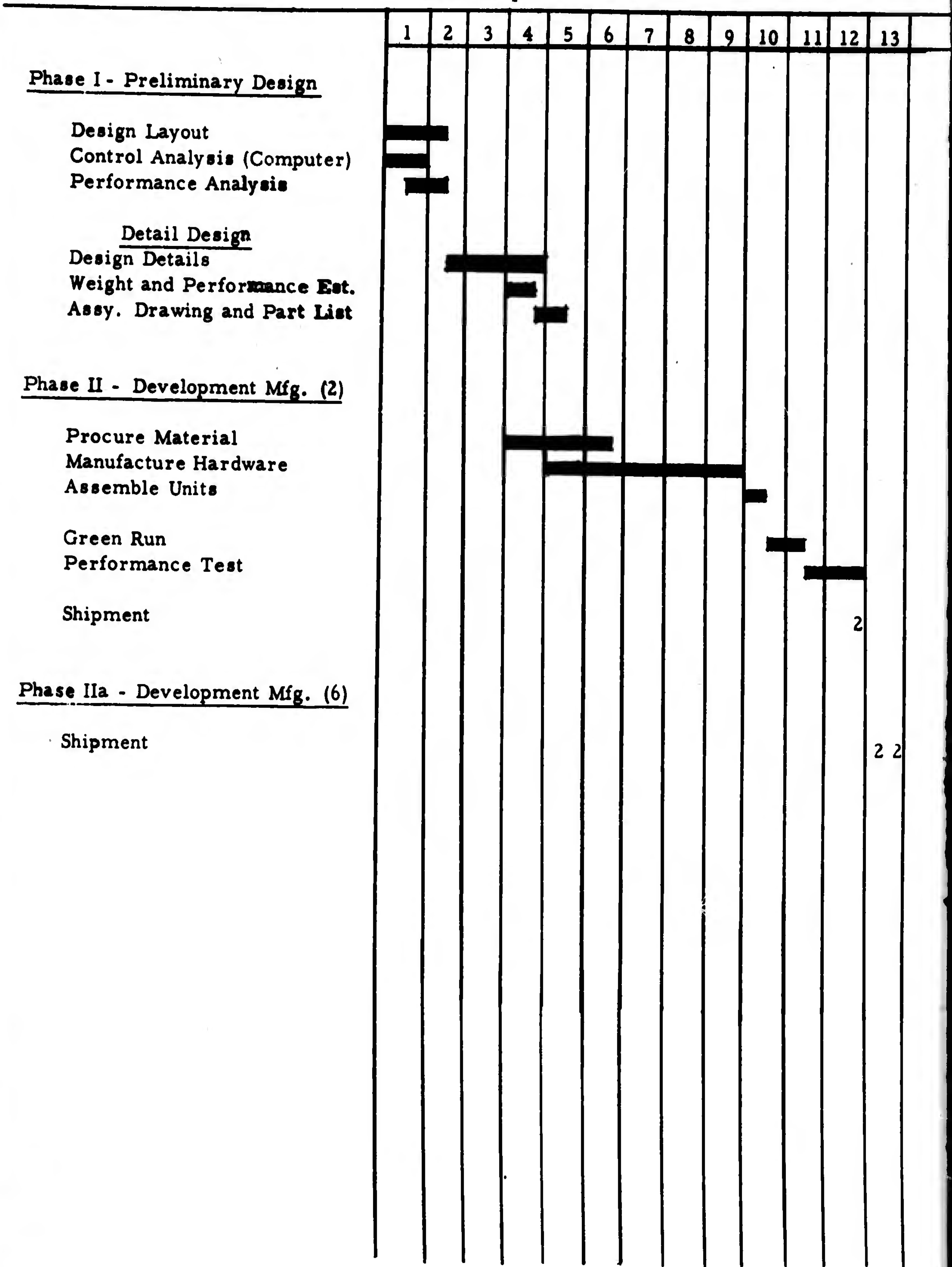
Figure 32

ENGINEERING PROGRAM

The attached milestone chart indicates the major elements of the engineering program required for the design of the hydrostatic transmission and its control. Phase I of the transmission design and the control design will require engineering effort to analyze the requirements and establish component configuration and sizing. Computer studies will be performed simulating each element of the system to establish system responses and characteristics. Layouts will be made to optimize arrangements, locations, and interconnections of transmission components and accessories. From these layouts detail hardware drawings, material lists, assembly drawings and parts list will be prepared.

Phase II consists of manufacturing the development units and testing and shipping. Testing will be done to insure performance, stability and overall application compliance. Details of the required testing will be negotiated.

DEVELOPMENT PROGRAM
300 HP Hydrostatic Steering Transmission
FMC Corp.



PROPELLER DRIVE SYSTEM

INTRODUCTION

At the request of F. M. C. Corporation a preliminary study has been made to determine the physical and operating characteristics of a hydrostatic propeller drive system to be used in conjunction with the proposed General Electric 300 HP Hydromechanical Steering and Power Train system described in the preceding sections. The basic requirements were as follows:

System to be driven from transmission PTO.

Maximum HP in - 250.

Maximum propeller speed - 750 RPM.

Propeller drive motor - 11" O. D. maximum
Length - short as possible

Propeller power requirements were assumed to vary as the square of propeller speed.

The results of this study are included within this section.

DESCRIPTION

Components required:

1 - variable displacement ball piston pump
21.2 cu. in./rev. Dwg. 1076511-37 Weight = 120 lbs.

1 - fixed displacement ball piston motor
66 cu.in./rev. Dwg. 1076511-228 Weight = 157 lbs.

1 - selector lever for water operation for forward-neutral-reverse propeller control

Installation hardware will include: mechanical linkage or flexible cable for controlling pump stroke from selector lever, two 1/2" dia. 150 psi lines to connect transmission oil system to main pump, two 1-1/2" I.D. 1800 psi lines for main supply and return oil between pump and motor, one 1/2" 50 psi line to carry motor leakage oil back to transmission reservoir, flanged adapter for mounting pump to PTO, and means for mounting propeller drive motor on vehicle.

No separate oil system is required. The oil system built into the hydromechanical transmission is adequate to furnish reservoir capacity, filter, make-up pump capacity, etc. to handle the oil system requirements of the propeller drive system.

The variable displacement ball piston pump would be mounted on the power take-off pad of the steering transmission, Fig. 33, and driven at engine speed to supply oil to drive the fixed displacement propeller drive motor. The pump displacement would be controlled by a mechanical

General Electric
Hydrostatic Propeller Drive Schematic
Integrated with
G. E. 300 HP Hydromechanical Transmission

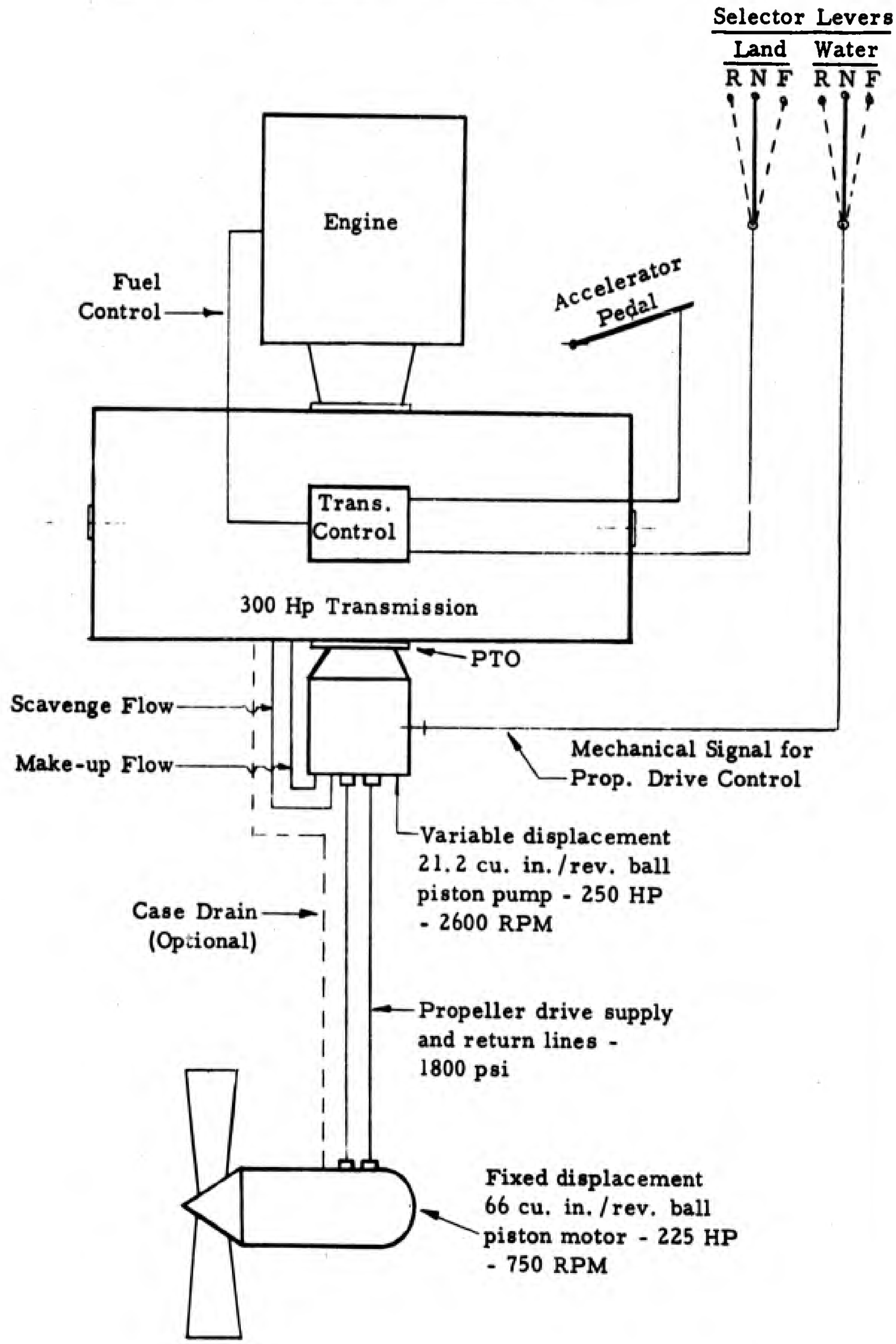
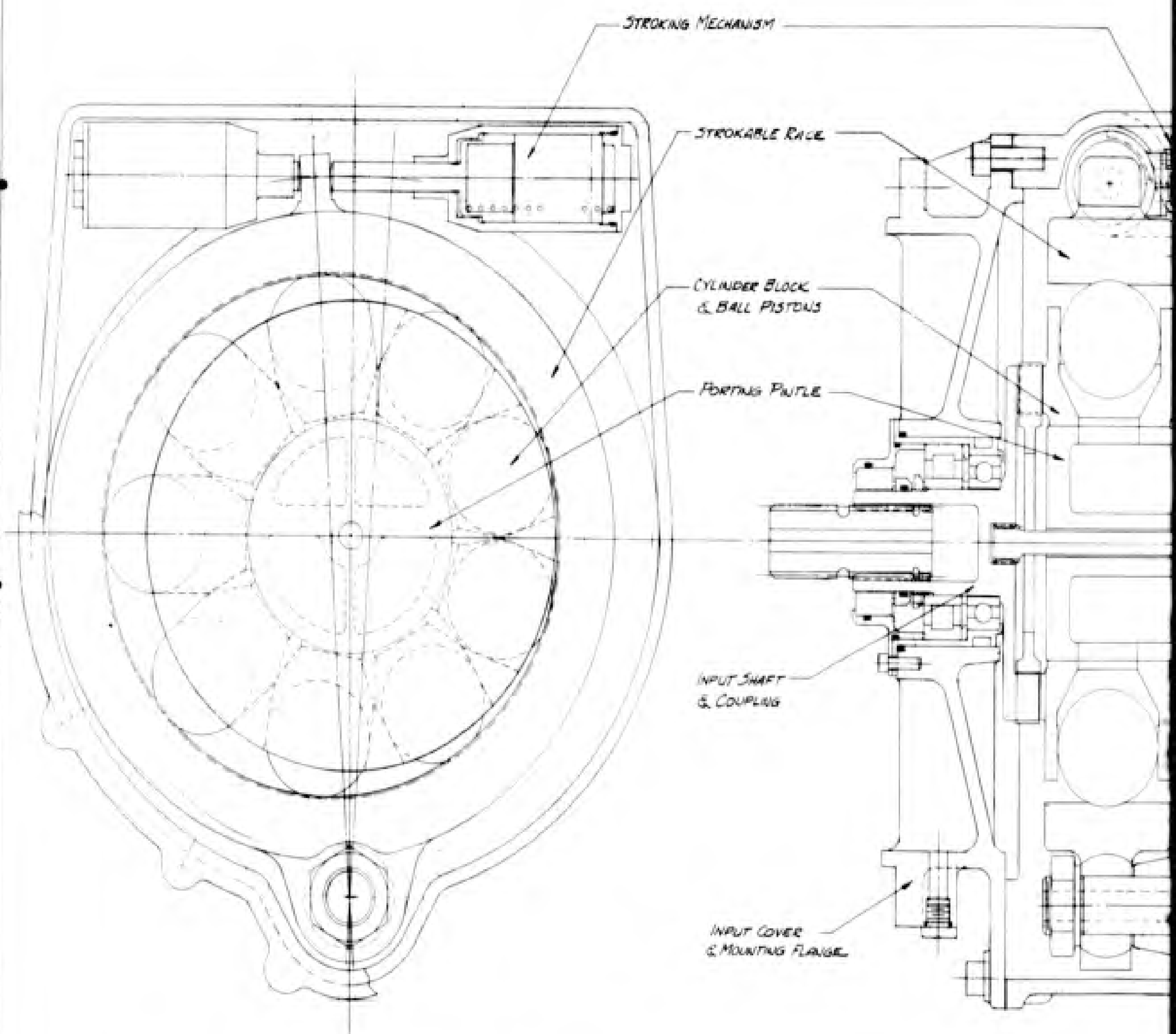


FIG. 33

push-pull signal from a lever in the operator's compartment. By this lever the speed and direction of rotation of the propeller is controlled. Normally, the pump would be set for 100% stroke in either the forward or reverse direction and propeller speed governed by engine speed control.

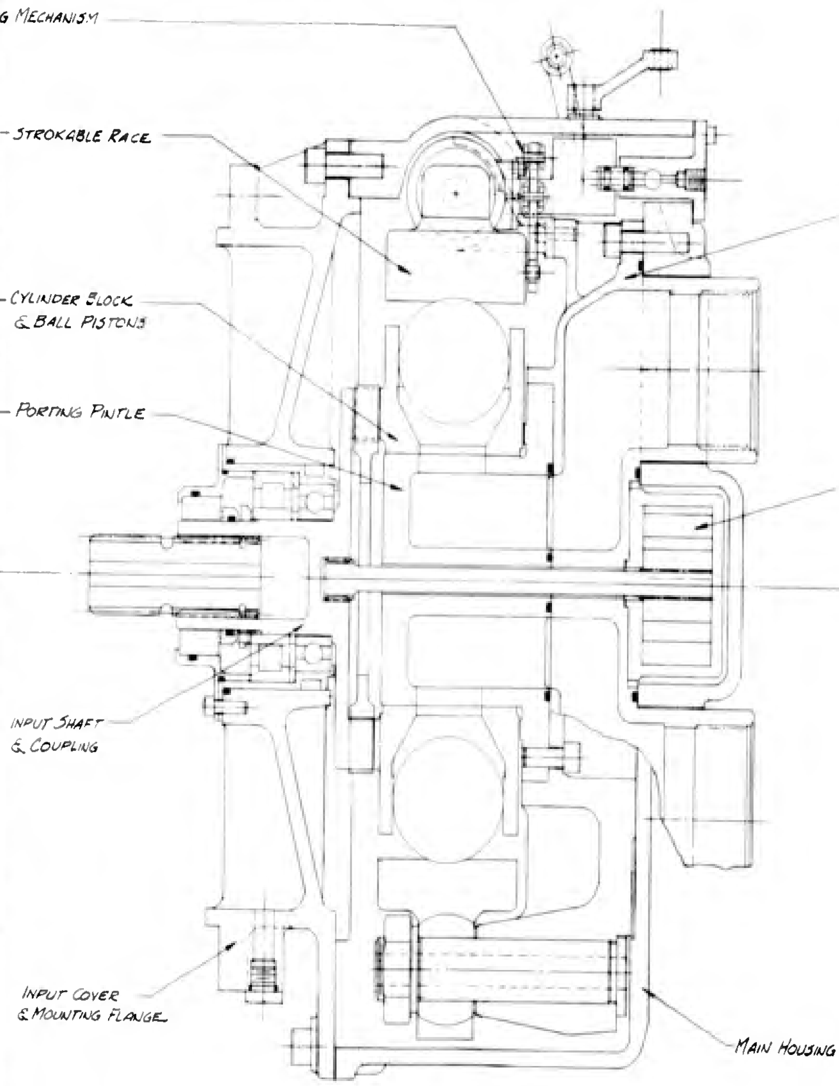
The fixed displacement motor provides for direct drive of the propeller without gearing. A suitable mount would be provided by the vehicle manufacturer to provide steering and retraction functions.

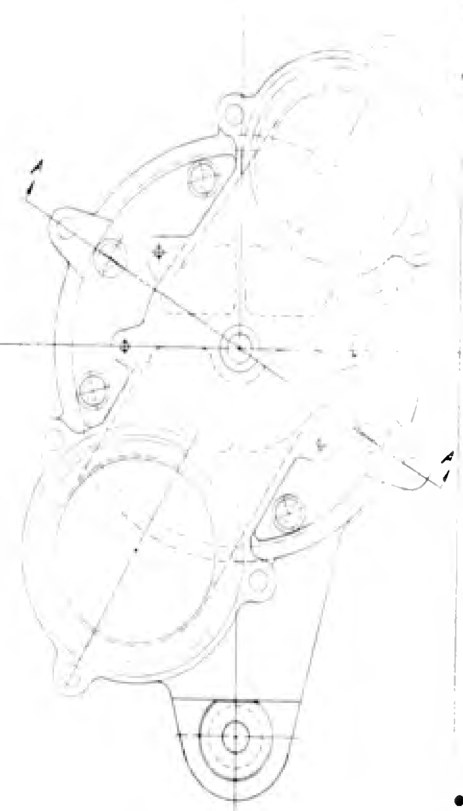
This system can be operated by itself with no output from the main transmission to the sprockets, or it can be operated while the sprocket power train is driving the sprockets and tracks in either forward or reverse direction. A study of the system characteristics indicates that lowest power loss for water operation is achieved if the hydromechanical transmission is set at that ratio giving 100% mechanical power transmission and zero "B" end speed. This would then allow the tracks during water operation to run at approximately the same velocity as the vehicle is travelling in the water. It is assumed that this mode of operation would create minimum drag for the vehicle travelling in the water. Reference to Figure 7 shows that zero "B" end speed at full engine speed provides 15 MPH track speed. By providing a control adjustment, possibly on the selector lever, the steering transmission could be set for a ratio to give 10 to 15 MPH



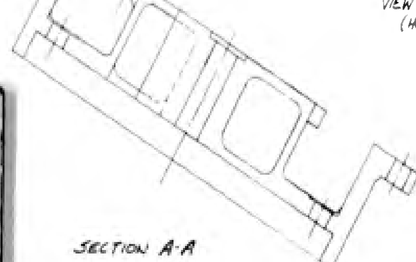
VIEW OF CYLINDER BLOCK & STROKING MECHANISM

1





VIEW OF PINTLE SUPPORT (HOUSING REMOVED)


SECTION A-A

2

E 1076511-37

400
859
DIST

LAYOUT
ENGINE DRIVEN VARIABLE DISPLACEMENT PUMP UNIT

GENERAL ELECTRIC
RIVER WORKS

SCALE 1/1

E 1076511-37

99611

track speed for water operation. This ratio, of course, can be set at the best ratio indicated by actual vehicle testing.

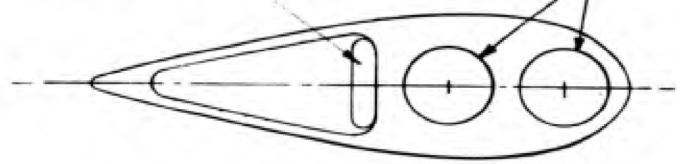
With this mode of water operation the speed of the tracks and propeller will be a direct function of engine speed and accelerator position, and the tracks will always be powered. Normally the tracks will be absorbing no appreciable power operating in a close to no-slip condition with respect to the water. However, should the vehicle run aground the tracks will immediately provide tractive effort to the vehicle with the available power being shared by propeller and tracks.

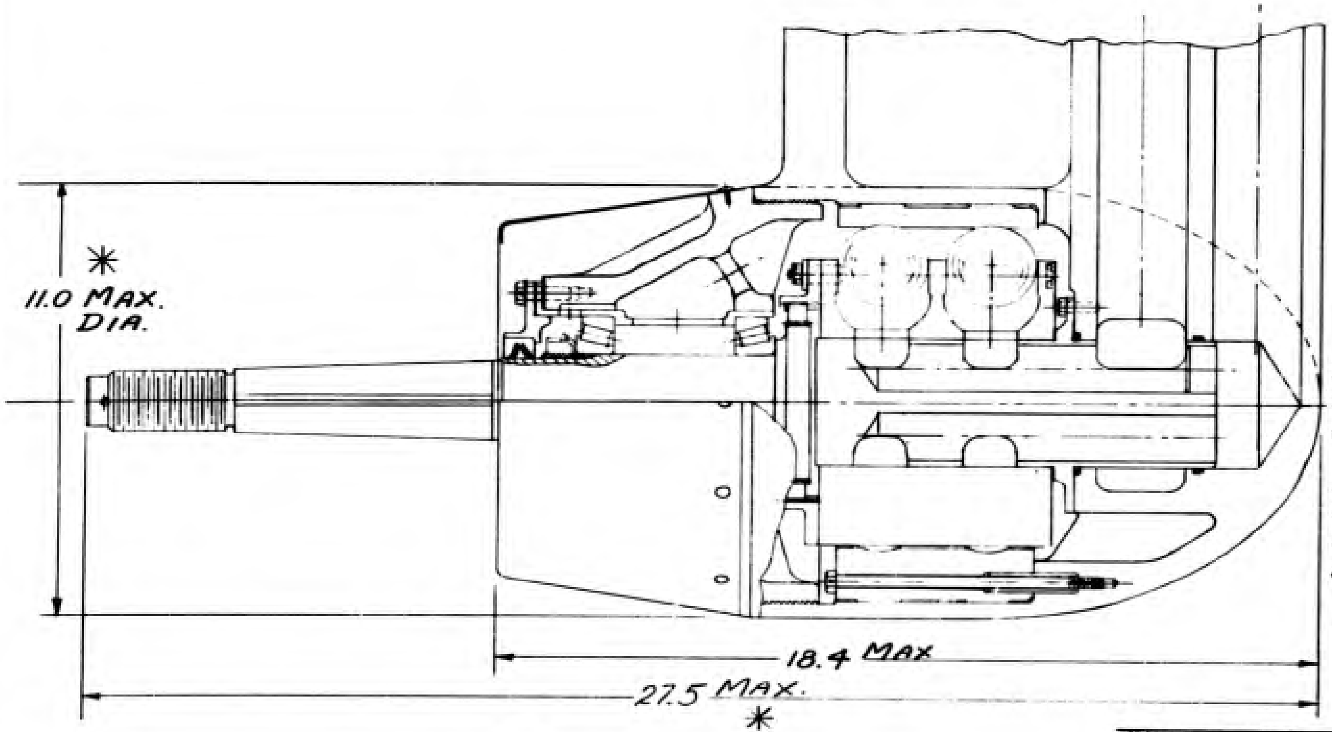
Propeller drive pump

This unit, drawing 1076511-37, is a nine cylinder, 2" piston diameter, 21.2 cu. in./rev. ball piston unit identical in internal construction to the development unit on test in our laboratory. This test unit has been run for over 125 hours with excellent performance. Endurance and performance tests are continuing through the rest of this year.

This pump will be controlled by a mechanically actuated pilot valve controlling a servo piston to set race position or stroke. Control oil for this simple servo is supplied from the working fluid in the pump pintle. An integral scavenge pump will maintain a low oil level in the pump housing to minimize parasitic drag on the internal rotating parts. The scavenged oil would be pumped into the main transmission

C 1076511-228





8 225 HP @ 750 RPM
7 PRESS - 1800 PSIG
6 7 CYL - 2.00 DIA BALL - DOUBLE LOBE - DOUBLE BANK
5 FIXED DISPL 66 CU IN/REV
4 2 PIECE AL HOUSING
3 EXTERNAL CONTOUR OPTIONAL
2 2.00 DIA (SAE STD) MARINE PROPELLER SHAFT
1 WEIGHT CALC ON PORTION MARKED WITH *

C 1076511-228

40U 859

225 H.P. PROPELLER DRIVE UNIT STUDY #2

HG008-14

GENERAL ELECTRIC
AAT
LYNN RIVER WORKS

C 1076511-228

WT 137 LBS
CODE IDENT → 99611

reservoir. Make-up or charge pressure fur the propeller drive would be piped from the make-up pump in the main transmi ssion. The pump as shown has a cast aluminum housing. The pump unit, rated for maximum speed of 3500 RPM and maximum pressure of 3000 psi, will be operating below its peak output capability in this application and hence should be a very reliable unit with time between overhaul over 1000 hours.

No clutch or decoupling device is shown for the pump for land operation because of the low power loss when operated at zero stroke as shown on the enclosed curve.

Propeller Drive Motor

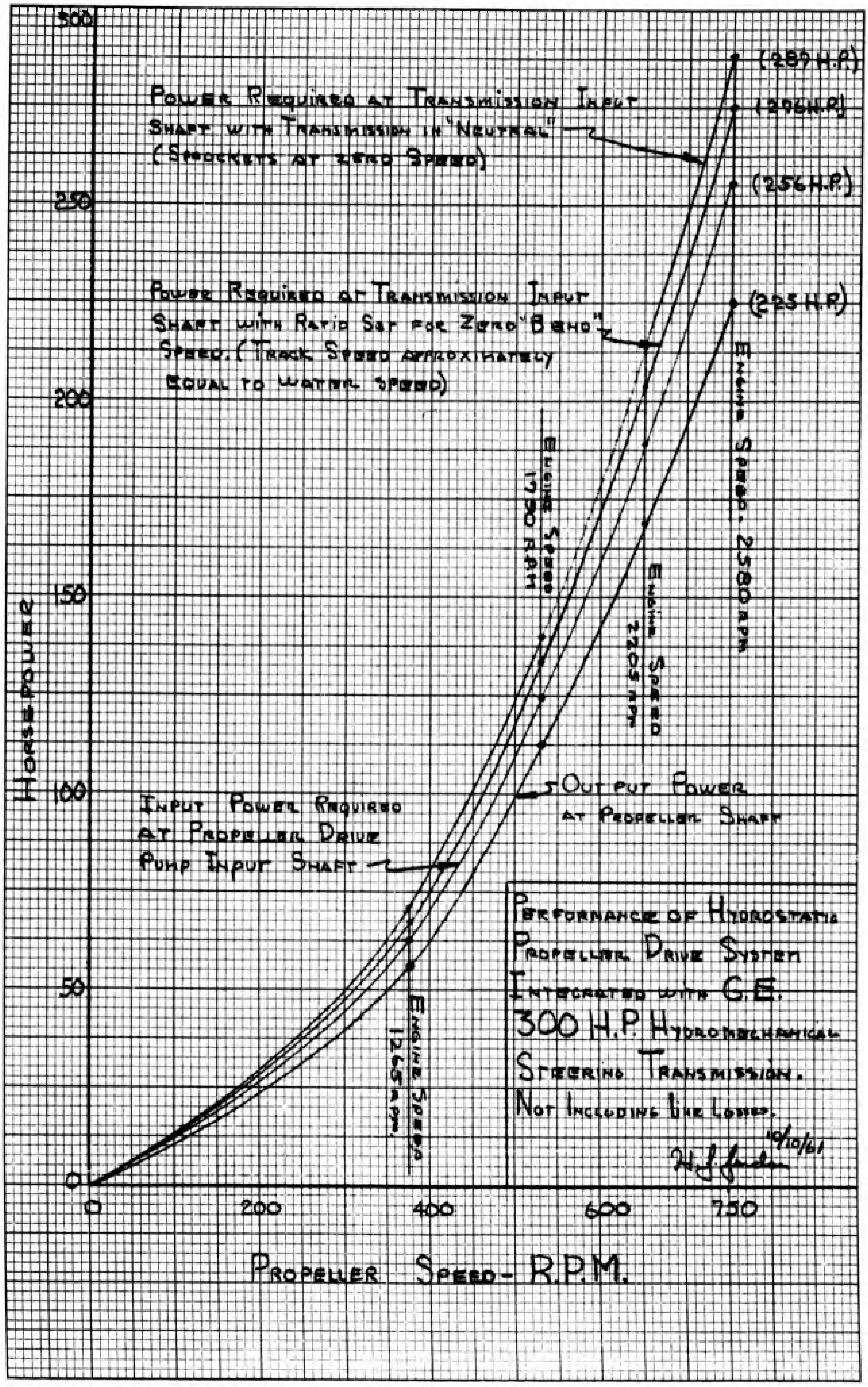
This unit, drawing 1076511-228, is a power package consisting of two, 7 cylinder, 2" piston diameter fixed displacement ball piston motors with elliptical races (double-lobe) providing two power strokes per revolution. The propeller shaft, bearings and seal are provided in the package. An Oldham flexible coupling is provided between the ball units and propeller shaft. A two-piece cast aluminum housing with integral mounting strut and oil lines would be provided. The unit would run "wet case", that is, with housing full of oil at all times and under a small positive pressure determined by the line loss in the "Case Drain" line which returns leakage oil to the main transmission reservoir.

System Performance

A performance curve showing system input and output horsepower as a function of propeller speed is shown in Figure 34 The performance was calculated on the basis of 225 HP output to the propeller at 750 RPM and output power varying as the square of the propeller speed. System characteristics were then calculated for 25%, 50%, 75% and 100% output power with the supply pump at 100% stroke. Hydrostatic system overall mechanical efficiency was then estimated for the four output power points. Pump-to-motor line losses were not included in the efficiency estimates for lack of knowledge of the piping arrangement. However, since lines should be short in this application, and no valves are required, the actual line losses should be negligible (less than 1%). The efficiency of the transmission excluding line losses is shown in Figure 35. System efficiency varies from 88 to 91%.

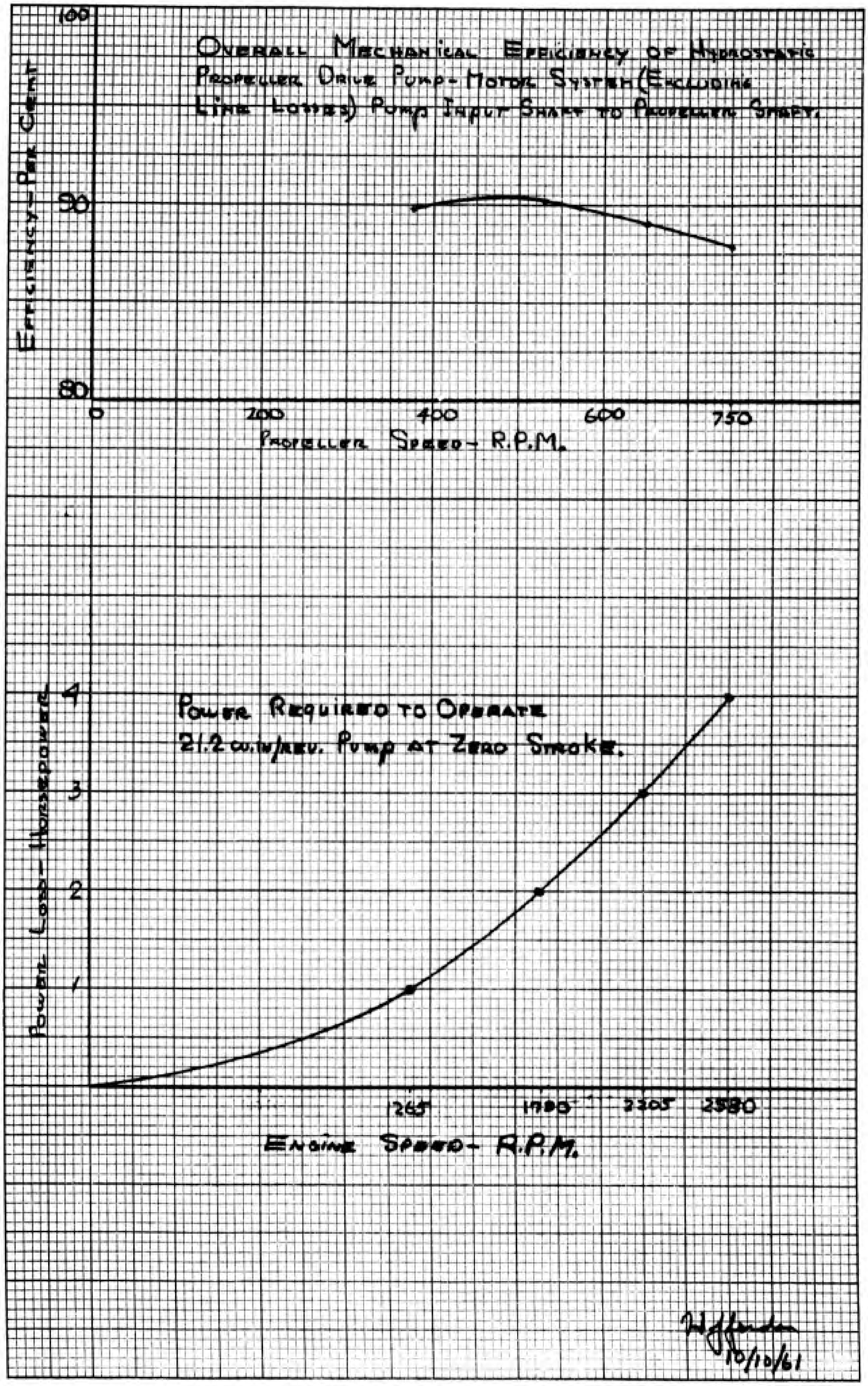
Under normal land operation the pump will require a normal power input even though no power is being supplied to the propeller. This drag loss is also shown in Figure 35. This loss could be eliminated by the use of a disconnect at the hydromechanical transmission's PTO of some type.

In like manner, operation of the propeller transmission during water operation results in losses within the hydromechanical transmission because of the rotation of its gearing and "A" end cylinder



Power Required at Transmission Input Shaft with Transmission in "Neutral" (Sprockets at zero Speed)
Power Required at Transmission Input Shaft with Ratio Set for Zero "Bend" Speed. (Track Speed approximately equal to water speed)
Input Power Required at Propeller Drive Pump Input Shaft
Output Power at Propeller Shaft
(289 H.P.)
(276 H.P.)
(256 H.P.)
(225 H.P.)
Engine Speed 1265 RPM
Engine Speed 1730 RPM
Engine Speed 2205 RPM
Engine Speed 2580 RPM
Performance of Hydrostatic Propeller Drive System Integrated with G.E. 300 H.P. Hydromechanical Steering Transmission. Not Including Line Losses.
10/10/61
Horsepower
Propeller Speed - R.P.M.
0
50
100
150
200
250
300
0
200
400
600
750

FIG. 34



OVERALL MECHANICAL EFFICIENCY OF HYDROSTATIC PROPELLER DRIVE PUMP-MOTOR SYSTEM (EXCLUDING LINE LOSSES) PUMP INPUT SHAFT TO PROPELLER SHAFT.
EFFICIENCY - PER CENT
100
90
80
0
200
400
600
750
PROPELLER SPEED - R.P.M.
POWER REQUIRED TO OPERATE 21.2 cu.in/rev. PUMP AT ZERO STROKE.
POWER LOSS - HORSEPOWER
4
3
2
1
1265
1790
2205
2580
ENGINE SPEED - R.P.M.
10/10/61

FIG. 35

block. If the transmission ratio is set so that it is in neutral or zero output speed condition, the "B" end cylinder block and output ring and sun gears will also be rotating and generating losses. At maximum engine speed these losses can be as high as 32 HP. If however, the transmission ratio is set giving little or no "B" end speed, the losses are considerably less, 20 HP maximum.

SUNDSTRAND ENGINEERING PROPOSAL
NO. 1950A-P1 (SECOND REVISION)

HYDROSTATIC TRANSMISSION FOR
LVT VEHICLE

FMC CORP., ORDNANCE DIVISION
SAN JOSE, CALIF.

Leader In Secondary Power

SUNDSTRAND

SUNDSTRAND AVIATION

SUNDSTRAND ENGINEERING PROPOSAL
NO. 1950A-P1 (SECOND REVISION)

HYDROSTATIC TRANSMISSION FOR
LVT VEHICLE

FMC CORP., ORDNANCE DIVISION
SAN JOSE, CALIF.

Note: Revisions are indicated by asterisks
in Table of Contents.

October 26, 1961

STATEMENT OF PROPRIETARY RIGHTS

II. DESIGN REQUIREMENTS OF THE TRANSMISSION

The proposed Hydrostatic Transmission System is designed to meet all c he required specifications.

A. VEHICLE DA...

1. Type

a. Nomenclature: Universal Landing Vehicle (Amphibious)

b. Traction: Track laying

Weight: 35,0( ) lbs. gross

2. Land Performance

a. Velocity: 40 MPH maximum improved roads
20 MPH maximum cross-country

b. Grade: Negotiate 70% slope at 2.5 MPH minimum forward and reverse
Negotiate 60% side slope

3. Water Performance

a. Forward Velocity - Highest consistent with power required for specified land performance.

b. Reverse Velocity - Highest consistent with simplicity of power train and required maneuverability in the water.

4. Life

   a. Land: 80%

   b. Afloat: 20%

5. Power Requirement of Vehicle

   a. Level at 40 MPH: 240 horsepower

   b. 70% Slope at 2.5 MPH: 177 horsepower

B. ENGINE DATA (In a gross power developing range of approximately 375 horsepower)

1. Spark Ignition

2. Compression Ignition

3. Gas Turbine

C. TRANSMISSION PARAMETERS

1. Angular Sprocket Velocity at 40 MPH: 587 RPM

2. Steering:

   a. Infinite turning radius to pivot turns (desired)

   b. Regenerative steering

3. Power Train: Full engine power into each track

4. Envelope Dimensions: Minimum with respect to required parameters.

5. Weight: Minimum with respect to required parameters.

6. Fuel Programming Control: Desirable

7. Braking:

   a. Secure 35, 000 lbs. gross weight vehicle on 70% slope.

   b. Brake 35, 000 lbs. gross weight vehicle 25 times from 40 MPH at 2 minute intervals at an average deceleration rate of 12 ft./ sec.^{2}

8. Push to Start Capability: Required

9. Towing Capability: Required

10. Cooler: To be furnished by vehicle manufacturer.

III. SYSTEM EQUIPMENT DESCRIPTION

A. GENERAL CONFIGURATION

The Hydrostatic Transmission System is schematically represented by Figure 1 of the Appendix. An Outline Drawing of the transmission is shown by Drawing Number 1950A-E6 and the transmission is referred to as Part Number 4000 268.

The transmission has the general physical characteristics of a "T" configuration i.e. the power input provision is perpendicular to each of the two oppositely directed power output provisions. The proposed hydrostatic transmission is composed of a hydraulic pump that couples to the power output pad of the engine or engine mounted gearbox. On the other end of the pump and through a mutual endcap manifold are connected two hydraulic motors "back to back". In an energy transmitting mode of operation, power is mechanically transmitted from the engine to the pump. The pump then converts mechanical power to hydraulic power and conveys it to the motors. The motors transmit the hydraulic power back to mechanical power and transmits it to a final drive gear train in the respective output provisions of the transmission. In an energy absorbing mode of operation, the direction of power flow reverses for each of the above described components.

Each of the two power output provisions of the transmission has a mechanically actuated disk brake that will be used for parking and emergency conditions. The purpose of the endcap manifold which is common to the three hydraulic units is to internally port hydraulic fluid between the hydraulic units and to house various control valves. Externally located on the basic transmission are the fluid reservoir, fluid filter and bypass valve and certain transmission control valving and plumbing. Wherever practical, it is proposed to use aluminum for housing material.

Transmission metal bearing surfaces will be lubricated by lube jets, as required. A scavenge pump will return the lubricating fluid from the sump to the transmission reservoir.

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

1. Steering Device
2. Accelerator Device
3. Operating Mode Selector Device
4. Parking Brake Lever
5. Braking Device

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

B. SYSTEM COMPONENTS

1. Variable Displacement Hydraulic Pump

The proposed hydraulic pump unit is of an axial piston, crosshead design and has a speed range of 0 to 3700 rpm. The unit is designed to vary its displacement from 0 to 23.0 cu. in. per rev. and has a maximum pressure capability of 6000 psi. The displacement is varied by a hydraulic controlled force multiplying piston which regulates the angular displacement of a swash plate. The pump unit is operated with a scavenged case and will obtain a large degree of lubrication from normal fluid leakage. The housing of the pump unit will be of cast aluminum material.

2. Variable Displacement Hydraulic Motor

The proposed hydraulic motor units are of axial piston, crosshead design and have a speed range of 0 to 2800 rpm.

Each unit is designed to vary its displacement from 0 to 54.0 cu. in. per rev. and has a maximum pressure capability of 6000 psi. Like the pump unit, the displacement is varied by a hydraulic controlled force multiplying piston which regulates the angular displacement of swash plate. The motor units are also operated with scavenged cases and will obtain lubrication from normal fluid leakage. The housings of the motor units will be of cast aluminum material.

3. End Cap Housed Control Valving

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

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

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

d. Flow Reversing Valves - This device is located between the fluid ports leading to each motor so as to direct either high or low pressure flow into the two respective porting provisions of each hydraulic motor.

Reversing the high and low pressure flow to the motors is manually controlled during certain vehicle steering operations through the flow reversing pilots.

e. Flow Reversing Pilot - This hydraulic valve is located between each flow reversing valve and the operator mode selector valve. The purpose of these devices is to activate the flow reversing valves when signaled by the steering mechanism.

4. <u>Transmission Accessory Equipment</u>

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

b. Input Driven Scavenge Pump - Like the charge pump, the device is also a variable displacement pump that will provide the means of scavenging the inner cavities of the transmission housing of fluid derived from component leakage. The displacement of the pump will be controlled by system pressure and engine speed.

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

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

e. Filter and By-Pass Valve - The filtering device will be located in the cooler and reservoir hydraulic circuit to remove contamination from the hydraulic fluid passing through the elements. The by-pass valve will be an integral part of the element housing. The by-pass valve activates at a specified pressure difference across the filter assembly to allow the fluid to pass-by the element in the event the 33 micron element becomes over contaminated.

f. Reservoir - The reservoir tank will be attached on the exterior of the basic transmission located in an accessible position. The purpose of the reservoir is to store excess fluid during transmission shut down and to provide a temporary storage delay for fluid during deaeration while the transmission is operating.

5. Final Drive Gearing

Final Drive gearing will be provided by the vehicle manufacturer as requested.

6. Engine Speed Control Equipment

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

b. Engine Driven Governor - This rotating mechanical governor is a device that receives signals from the vehicle operator through the engine speed control cam and linkage mechanism for properly operating the engine for optimum fuel economy. The governor will be located on the vehicle engine, and will either be specified by or furnished by Sundstrand.

7. Vehicle Control Equipment

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

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

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

d. Parking Brakes - The disk brake that is provided on each output shaft is used for vehicle parking and emergency stopping. The brakes are located in the respective integral final drive gear trains housings.

C. OPERATOR CONTROLS

1. Cab-Mounted Operator Controls

The proposed type and number of cab-mounted controls have been determined on the basis of mobility and safety. By the incorporation of multiple purpose controls, the number in the cab is established at five as listed below. The controls are schematically illustrated in Figure 1 of the Appendix.

a. Steering Wheel
b. Accelerator Pedal
c. Operating Mode Lever
d. Parking Brake Lever
e. Service Brake Pedal

2. Steering Wheel

Steering of the vehicle will be accomplished by turning the wheel in either one direction or the other as in steering a conventional automobile. A maximum turn to either the left or right will cause the vehicle to negotiate a spin turn. The hydrostatic transmission lends itself well for steering with either a steering wheel or a joy stick. However, the steering wheel is more applicable for providing support to the operator during vehicle operation over rough terrain. The power steering system is of a modulated torque type to facilitate smooth steering control of the vehicle with minimum operator effort.

3. Accelerator Pedal

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

4. Operating Mode Lever

The operating mode lever is used to change the direction of vehicle motion from forward to reverse. While the control system may be easily designed to allow continuous change of speed from full-speed forward to zero and full-speed reverse without any lever actuation, it is felt that as a safety precaution some distinct effort should be made by the driver to get the vehicle to go backwards. This lever also includes a neutral position which would be used mainly during engine starting, vehicle towing, and parking.

5. Parking Brake Lever

Actuating the parking brakes will be accomplished mechanically by the lever located in the operator compartment. The brakes can also be used for stopping the vehicle in the event of an emergency.

6. Service Brake Pedal

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

7. Instrument Panel

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

IV. OPERATION AND CONTROL FEATURES

A. VEHICLE OPERATION

1. Velocity Control

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

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

Reversing vehicle direction of travel is accomplished by making the prope[illegible] selection with the operating mode lever. The operating mode lever is mechanically connected to the three way operating mode selector valve. The three way valve controls the direction of working pressure flow to the hydraulic motors. T[illegible]mainder of the t[illegible]mission operation remains the same when changing the direction of vehicle travel. Braking the vehicle is accomplished as before by the power output hydraulic units reverting to pumps and feeding power back to the engine gearbox mounted hydraulic unit.

2. Steering Control

The m[illegible]thod of steering most desirable for this transmission from a standpoint of simplicity is the modulated torque type. Regenerative steering is accomplished by being able to impose infinitely variable torque from full forward torque to full reverse torque on each track independently. This is done by adjusting the motor displacements relative to one another. Since the pressure in the system is determined by the accelerator, variations of the motor displacements will have a direct affect on the torque applied to the tracks. By being able to modulate the track torques, smooth and uniform regenerative vehicle turns are possible.

The motors in this transmission have been designed so that they are unable to achieve negative displacement. This was done with the idea of reducing their size as much as possible. In order to obtain the negative torques that are needed for live track and high speed steering, flow reversing valves are incorporated. These valves reverse the flow to the motor each time the motor swashplate reaches the zero stroke position. When the flow reverser is actuated, the motor swashplate then continues back into stroke to give the required negative torque. Since these motor swashplates are directly connected through linkages to the steering wheel, the recycling of the flow reverser will be automatic as the steering wheel is returned to the center position.

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

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

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

3. Braking

The hydraulic units employed in this propulsion system are equally capable of being either pumps or motors. To reverse their role, which is necessary in braking, the two fluid ports in the end cap manifold for each motor must reverse in high and low pressure exposure. Whenever the flow through the output units exceeds the flow from the engine driven unit, the track unit becomes a pump and the engine driven unit a motor. The pressure level of the system is determined by the relative speed and respective displacements between the track units and the engine mounted unit.

In vehicles with standard gearbox transmission, down-shifting may be employed to help brake on steep hills or slow down gradually. Kinetic energy from the vehicle can be transmitted through the transmission to the engine where it is dissipated in the form of frictional drag horsepower. The hydrostatic transmission is an infinitely variable speed transmission in the driving direction as well as the braking direction. Consequently, there are infinite down-shifting capabilities inherent in the transmission.

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

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

4. Vehicle Towing

The vehicle can be towed either in the forward or reverse direction without the engine operating. This facilitates return of the vehicle after an engine or engine-pump failure or certain control malfunctions. For vehicle towing, the operating mode lever is placed in the neutral position thereby placing a hydraulic short circuit in the system porting. As the vehicle is pulled the output hydraulic units revert to pumps and attempt to motor the engine. The short circuit is accomplished by resetting the working pressure relief valve to a very low pressure setting thereby preventing a build-up pressure and hence, minimizing retarding or towing force. Each of the output units has a charge pump which serves as a replenishing pump as well as a cooling circuit pump. The vehicle can be towed at reasonable speeds for prolonged periods of time.

5. Push to Start Capability

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

B. FUEL PROGRAMMING CONTROL

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

The essential components of the engine speed control system are the accelerator foot pedal, the engine driven governor, the engine speed control program cam linkage mechanism, and the accelerator and braking valve. The foot pedal and its linkage are connected to the engine governor and the accelerator and braking valve through the engine speed cam. Thus, as the foot pedal is depressed, the engine governor is reset to a higher speed. The governor reset linkage is connected to a dash pot to yield smooth operation. Simultaneously, the engine speed cam will advance to its appropriate position to serve as an index of engine speed.

When the speed governor is signaled by an increase of output torque, the linkage will produce a movement to increase the engine speed. The throttle linkage will move until it reaches its full opening position indicating the engine is now fully loaded for the particular speed setting. The control will always insure that the engine will be running at full throttle for any particular vehicle power requirement.

C. OPTIONAL TYPES OF STEERING CONTROL

1. Difference Speed Steering

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

The difference speed steering control system would receive signals from the steering wheel when turned by the operator of the vehicle. As the steering wheel is turned in the desired direction by the operator, the difference speed control system would signal the control pistons of the hydraulic motors to change their respective displacements to achieve a specified turn. The difference speed steering system constantly monitors the output speeds of the two tracks of the vehicle so as to seek and maintain the proper signal to the motor displacement pistons throughout the full speed range of the vehicle for any given vehicle turn. The displacement of the respective motors will, in effect, increase the torque and resulting velocity of one track while decreasing the torque and resulting velocity of the other track.

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

The displacement control pistons of the hydraulic motors receive their signals from two distinct sources, the output driven charge pump and the differential steering control system. The two signals are synchronized with each other by virtue of the fact that difference steering will set the precedence if the vehicle is negotiating a turn and the output driven charge pump will set the precedence when the vehicle is moving at an infinite radius. The purpose of the signals from the differential steering control system is to regulate the relative torques and velocity differences between the two tracks of the vehicle when negotiating turns. The purpose of the signals from the output driven charge pumps is to reduce the stroke of the motors during vehicle acceleration and high velocity operation.

2. Ratio Speed Steering

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

D. SYSTEM PROTECTION FEATURES

1. System Overpressure Protection

The proposed Hydrostatic Transmission System is protected against fluid overpressure by the charge pressure relief valve. As system pressure reaches the maximum limit of the pressure relief valve because of heavy vehicle loading conditions, the valve will "crack" and emit fluid from the high to low pressure port. As long as the heavy loading condition exists, the relief valve will modulate to limit the system pressure at the relief valve setting.

2. Vehicle Overspeed Protection

The transmission and engine are protected from vehicle overspeed control. Vehicle speed is sensed by the output driven charge pumps of the transmission. The displacements of the motor units are controlled by the signals of vehicle speed received by the motor driven charge pumps. As vehicle speed increases, the displacement of the motor units decreases. As vehicle speed increases beyond the maximum design speed of the vehicle, the charge pumps signal this overspeed condition and cause the motor displacements to increase. Thus, the vehicle will decrease in speed by virtue of the hydrostatic transmission functioning as a brake until the vehicle decreases to maximum design speed.

3. Engine Overspeed Protection

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

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

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

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

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

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

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

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

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

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

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

VI. TECHNICAL DATA

A. INSTALLATION AND SYSTEM

1. Transmission Mounting (Ref. Drawing No. 1950A- E6)

   a. Power Input - Shaft coupling

   b. Power Output - Flange mounting

   c. Transmission - Three-point suspension

2. Transmission Weight

   a. Dry - 970 lbs.

   b. Fluid - 48.5 lbs.

3. Transmission Envelope (Ref. Drawing No. 1950A- E6)

4. Transmission Fluid Pressure

   a. Maximum Working Pressure - 6000 psi

   b. Rated Working Pressure - 3000 psi

   c. Back or Return Pressure - 250 psi

   d. Maximum Δ Pressure - 5750 psi

5. Temperature Data

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

   b. Fluid Temperature Range: -65° to 200° F

6. Heat Exchanger Requirements (Based Upon Braking Requirements)

   a. Dissipate a maximum of 6500 BTU/min.

b. Cooling Circuit Flow 40 gpm nominal
60 gpm maximum

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

d. Aerated oil - 20 gpm out of 60 gpm

e. Cooling Circuit Pressure - 50 psi nominal

7. Fluid Reservoir Assembly

a. Capacity - 8 gals.

b. Vented to atmosphere

8. Fluid Type - Type A anti foam transmission fluid

9. Working Pressure Relief Valve Setting - 6000 psi maximum

10. Return Pressure Relief Valve Setting - 250 psi minimum

11. Output Driven Charge Pump

a. Rated Flow - 2 gpm each

b. Rated Discharge Pressure - 250 psi

12. Input Driven Charge Pump

a. Rated Flow - 40 gpm

b. Rated Discharge Pressure - 250 psi

13. Input Driven Scavenge Pump

a. Rated Flow - 60 gpm

b. Rated Discharge Pressure - [illegible] psi

14. Parking B-ake Rated Capacity - Adequate to hold vehicle on 70% slope plus emergency stopping capability.

15. Heat Rejection Data (Ref. Figure 3)

16. Transmission Efficiency (Ref. Figure 2)

B. TRANSMISSION COMPONENTS

1. Variable Displacement Pump

   a. Manufacturer: Sundstrand Aviation Division

   b. Displacement: Variable from 23.0 to 0 cu. in. per rev.

   c. Outlet Pressure: 6000 psi maximum

   d. Inlet Pressure: 250 psi minimum

   e. Housing Material: Aluminum

   f. Speed Range: 0 to 3700 rpm

   g. Rotation: Single direction once selected

   h. Life: 1000 hours

   i. Rated Flow Delivery: 180 gpm

2. Variable Displacement Motor

   a. Manufacturer: Sundstrand Aviation Division

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

   c. Inlet Pressure: 6000 psi maximum

d. Outlet Pressure: 250 psi minimum

e. Housing Material: Aluminum

f. Speed Range: 0 to 2800 rpm

g. Rotation: Reversible

h. Life: 1000 hours

i. Rated Torque Output: 48,000 in. lbs.

C. TRANSMISSION DURABILITY AND DESIGN LIFE

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

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

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

4. Transmission capability of one thousand hours of operation between overhaul of which approximately 400 hours will be at idle.

D. VEHICLE ENGINE SELECTION

For purposes of proposing and cost estimating, Sundstrand assumed that the engine to be selected for use with the proposed hydrostatic transmission will be either spark ignition or compression ignition type. When considering the selection of an engine to be used with the hydrostatic transmission, two important factors should be considered; the engine output speed range and the fuel programming control system. If a gas turbine engine is to be seriously considered for the first prototype vehicle, a redefinition of the fuel programming control will have to be made and very likely an engine mounted gearbox will have to be provided between the hydrostatic transmission and engine output shaft.

E. SUSTAINED DOWNGRADE BRAKING

The suggested cooling capacity of the hydrostatic transmission is based upon repetitive braking as specified in the Proposal. However, the heat exchanger will not be adequate for sustained downgrade braking of the vehicle at high speeds. With the 6500 BTU/min. heat exchanger, it will be possible to brake the vehicle without overheating the transmission fluid up to a speed of 5 mph down a 70% grade or 10 mph down a 35% grade. If braking at higher velocities is required, the heat exchanger capacity must be increased.

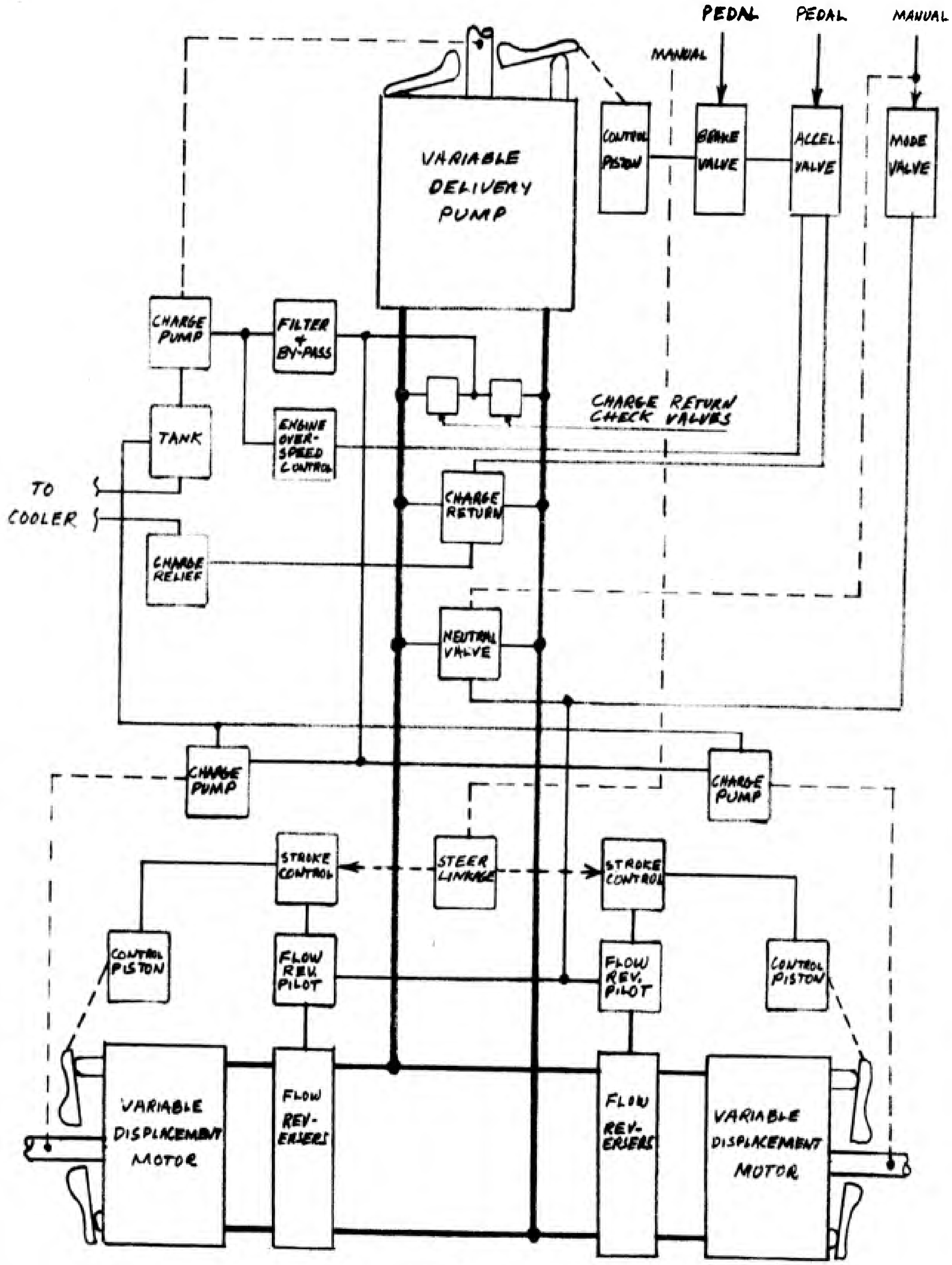


FIGURE 1
CONTROL BLOCK DIAGRAM ~ HST SYSTEM

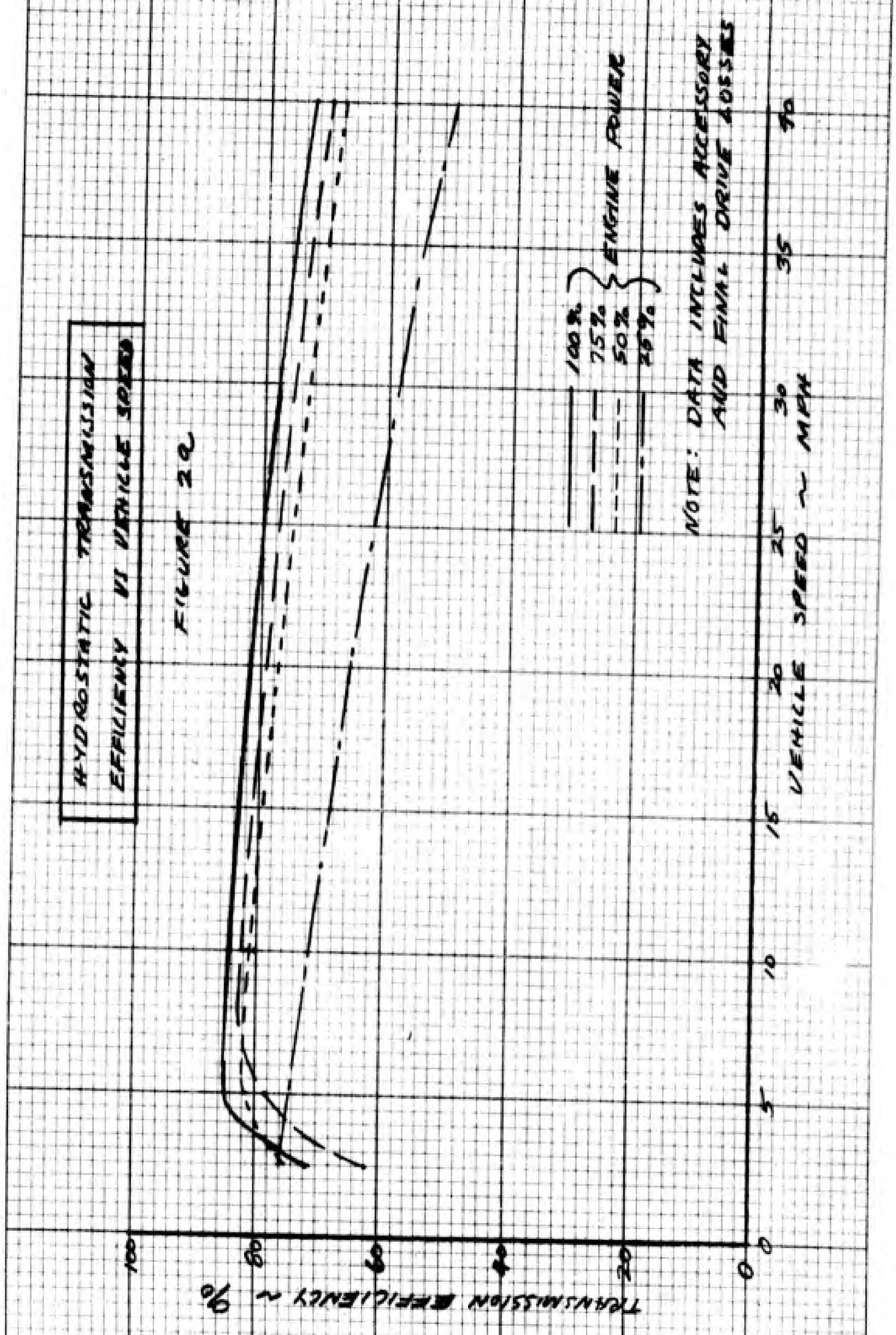

HYDROSTATIC TRANSMISSION
EFFICIENCY VS VEHICLE SPEED
FIGURE 2a
100%
75%
50%
25%
ENGINE POWER
NOTE: DATA INCLUDES ACCESSORY
AND FINAL DRIVE LOSSES
0
5
10
15
20
25
30
35
40
VEHICLE SPEED ~ MPH
0
20
40
60
80
100
TRANSMISSION EFFICIENCY ~ %

FIGURE 2

HYDROSTATIC TRANSMISSION
HEAT REJECTION
VS
VEHICLE SPEED

NOTE: DATA INCLUDES ACCESSORY AND FINAL DRIVE LOSSES

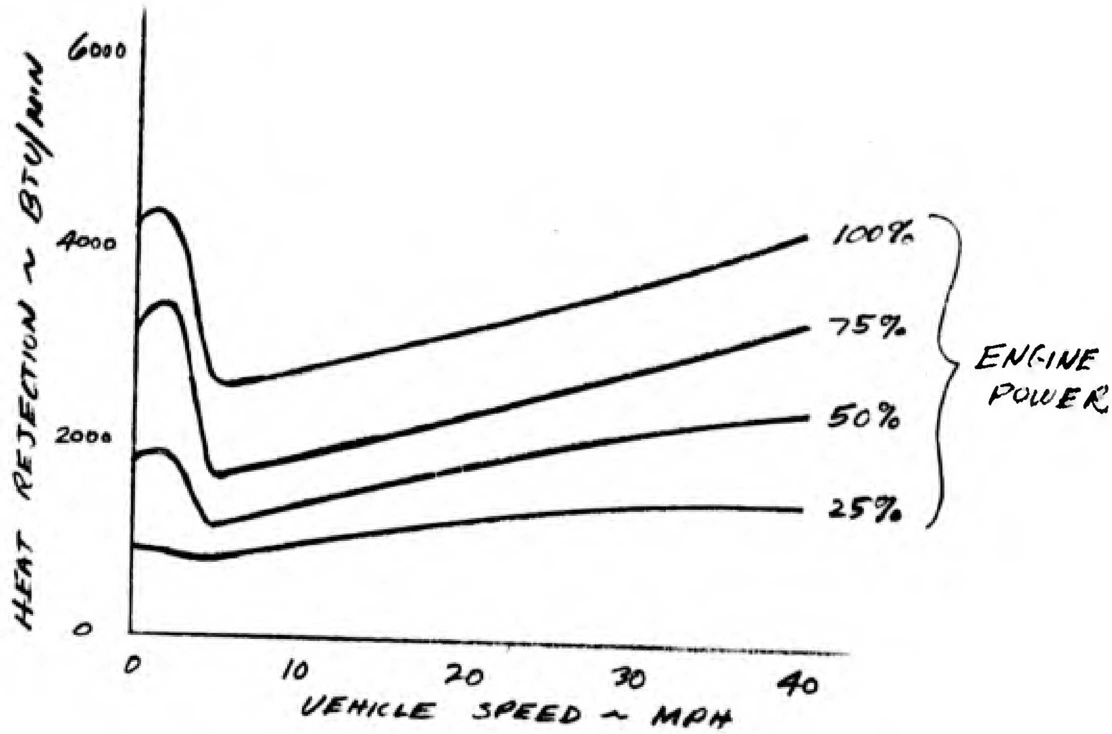


FIGURE 3

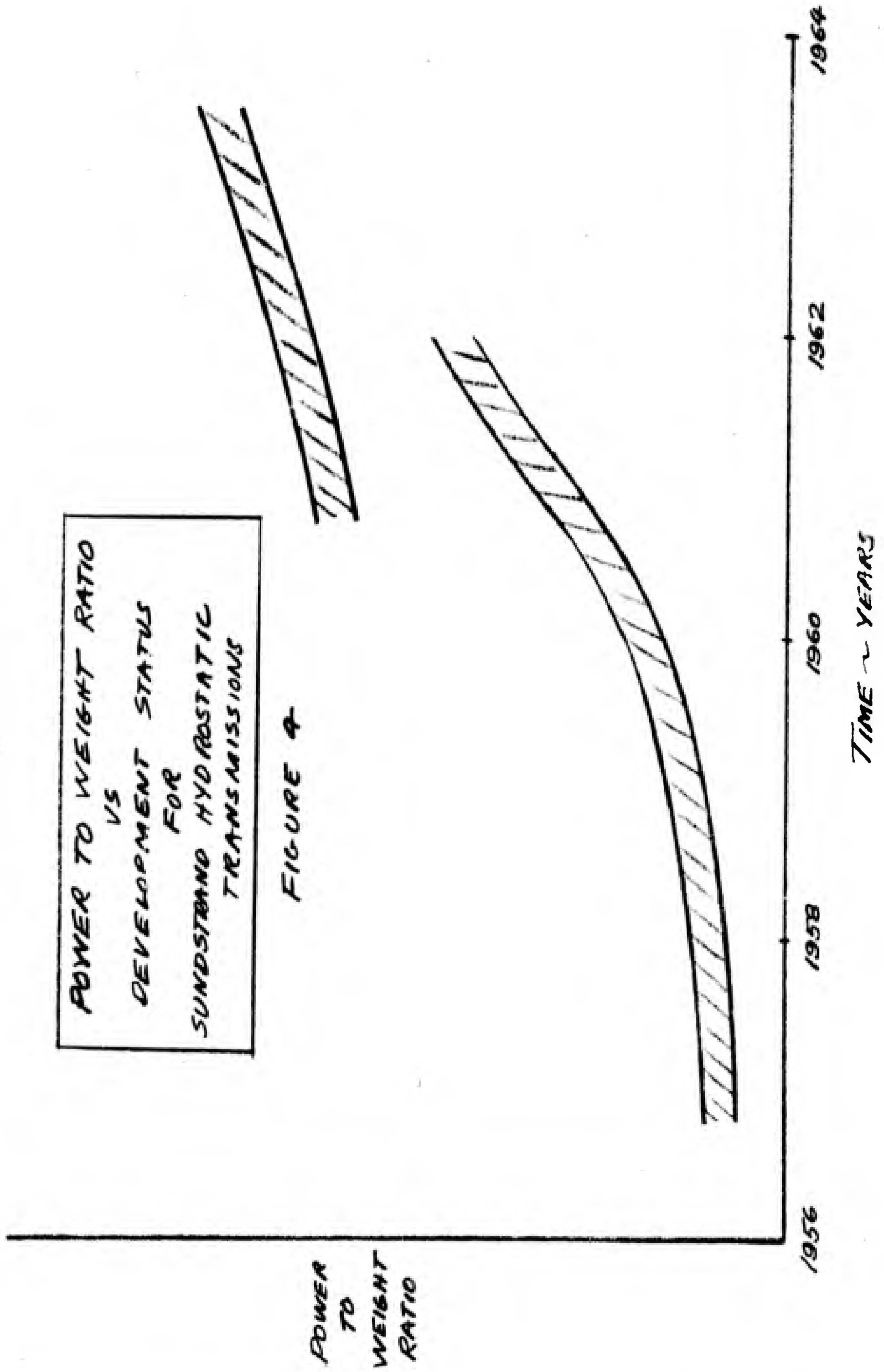
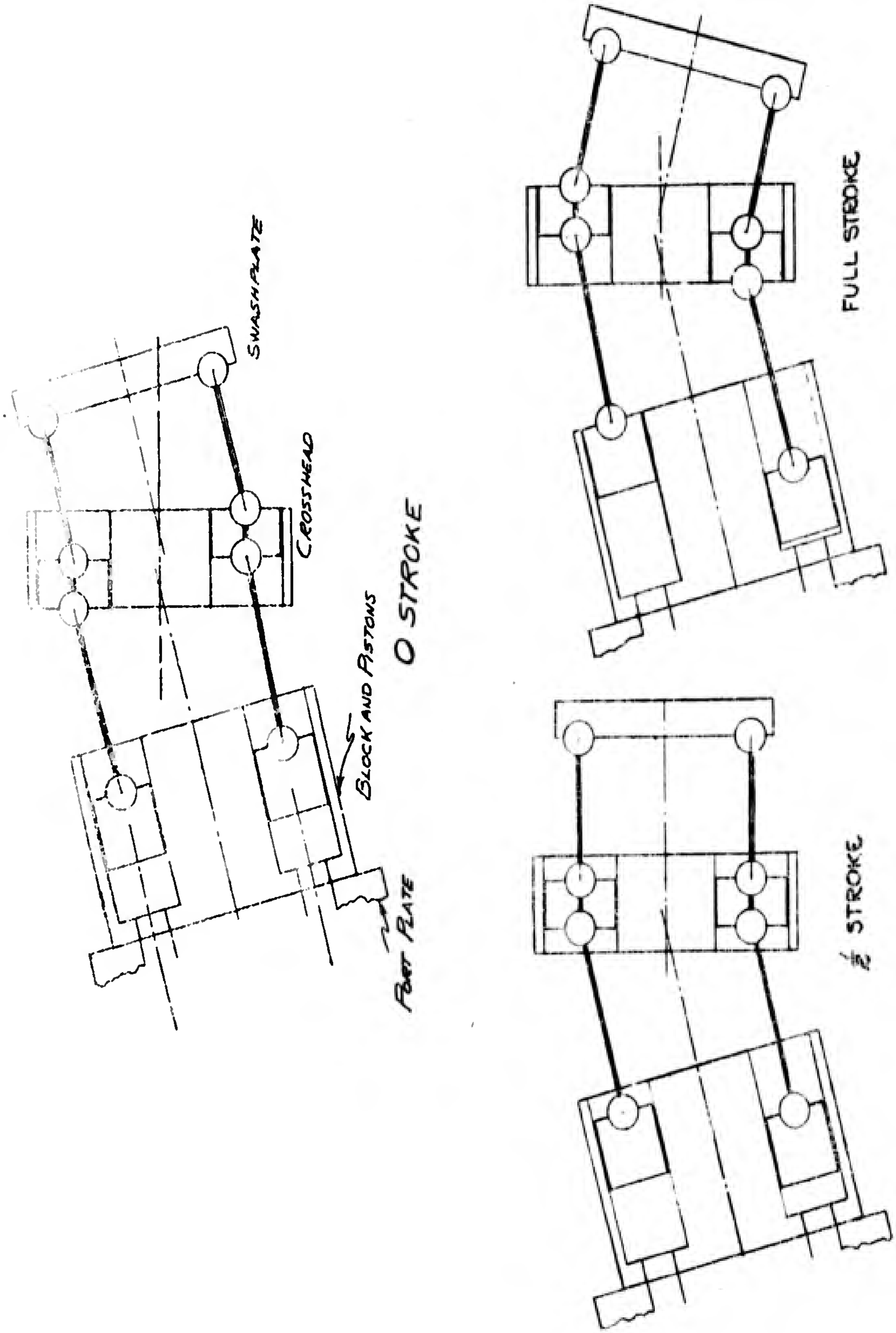

POWER TO WEIGHT RATIO
VS
DEVELOPMENT STATUS
FOR
SUNDSTRAND HYDROSTATIC
TRANSMISSIONS
FIGURE 4
1956
1958
1960
1962
1964
TIME ~ YEARS
POWER
TO
WEIGHT
RATIO

FIGURE 4



CROSSHEAD STROKES
FIGURE 5

FIGURE 5

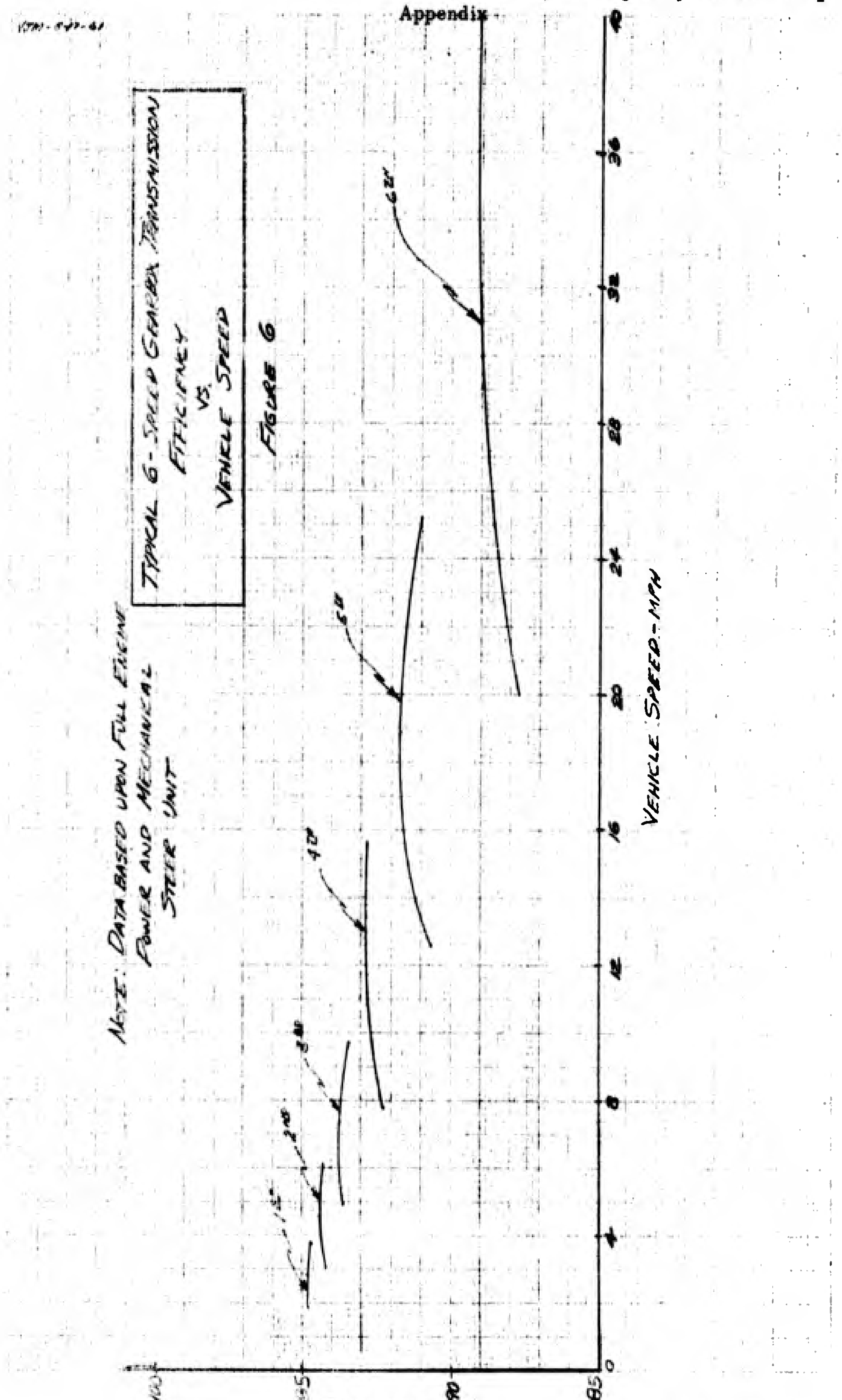

NOTE: DATA BASED UPON FULL ENGINE POWER AND MECHANICAL STEER UNIT
TYPICAL 6-SPEED GEARBOX TRANSMISSION
EFFICIENCY
VS.
VEHICLE SPEED
FIGURE 6
1st
2nd
3rd
4th
5th
6th
EFFICIENCY %
100
95
90
85
VEHICLE SPEED - MPH
0
4
8
12
16
20
24
28
32
36
40

FIGURE 6

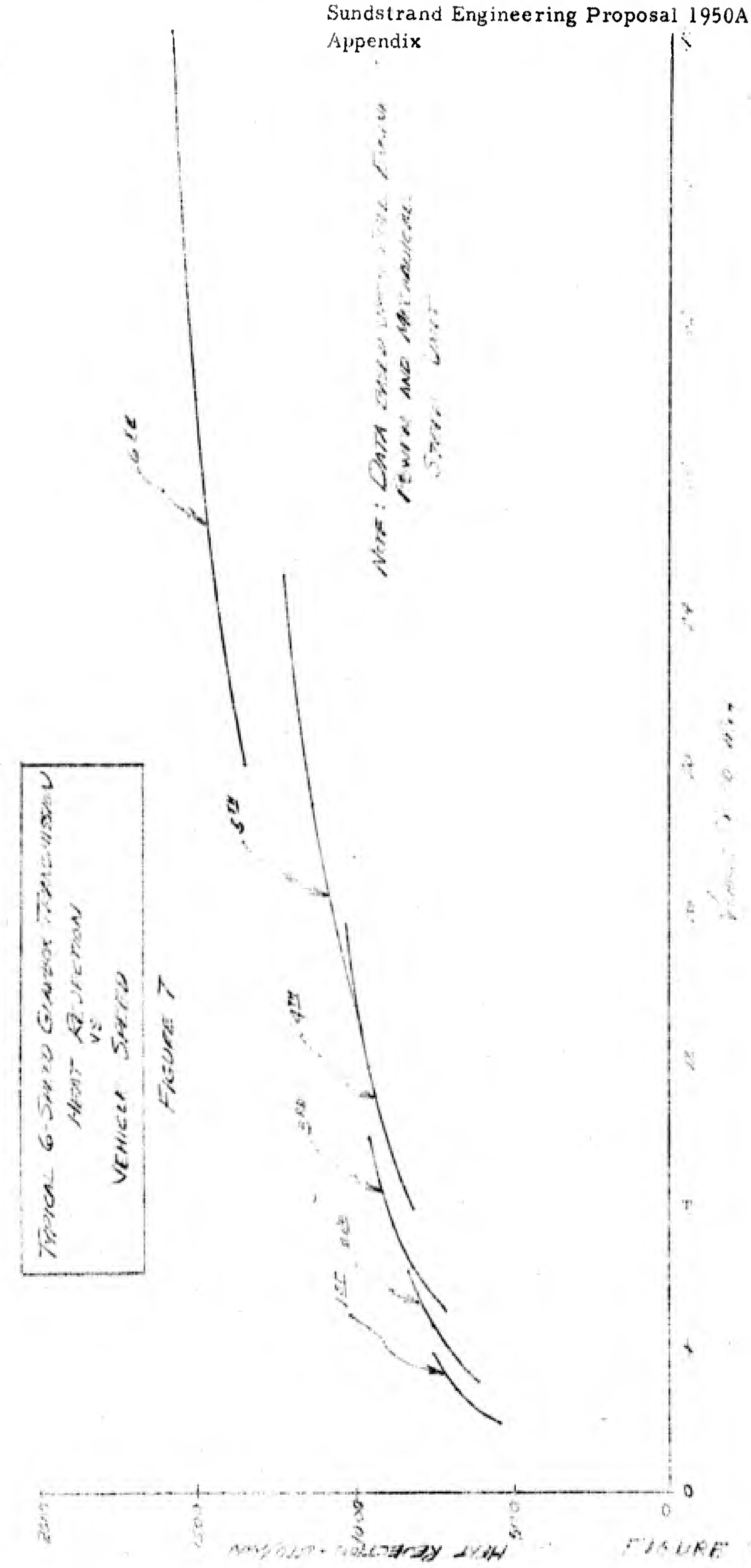

Typical 6-Speed Gearbox Transmission
Heat Rejection
vs
Vehicle Speed
Figure 7
Note: Data Taken With Full Fluid
Power and Mechanical
Steer Load
Heat Rejection - BTU/Min

FIGURE 7

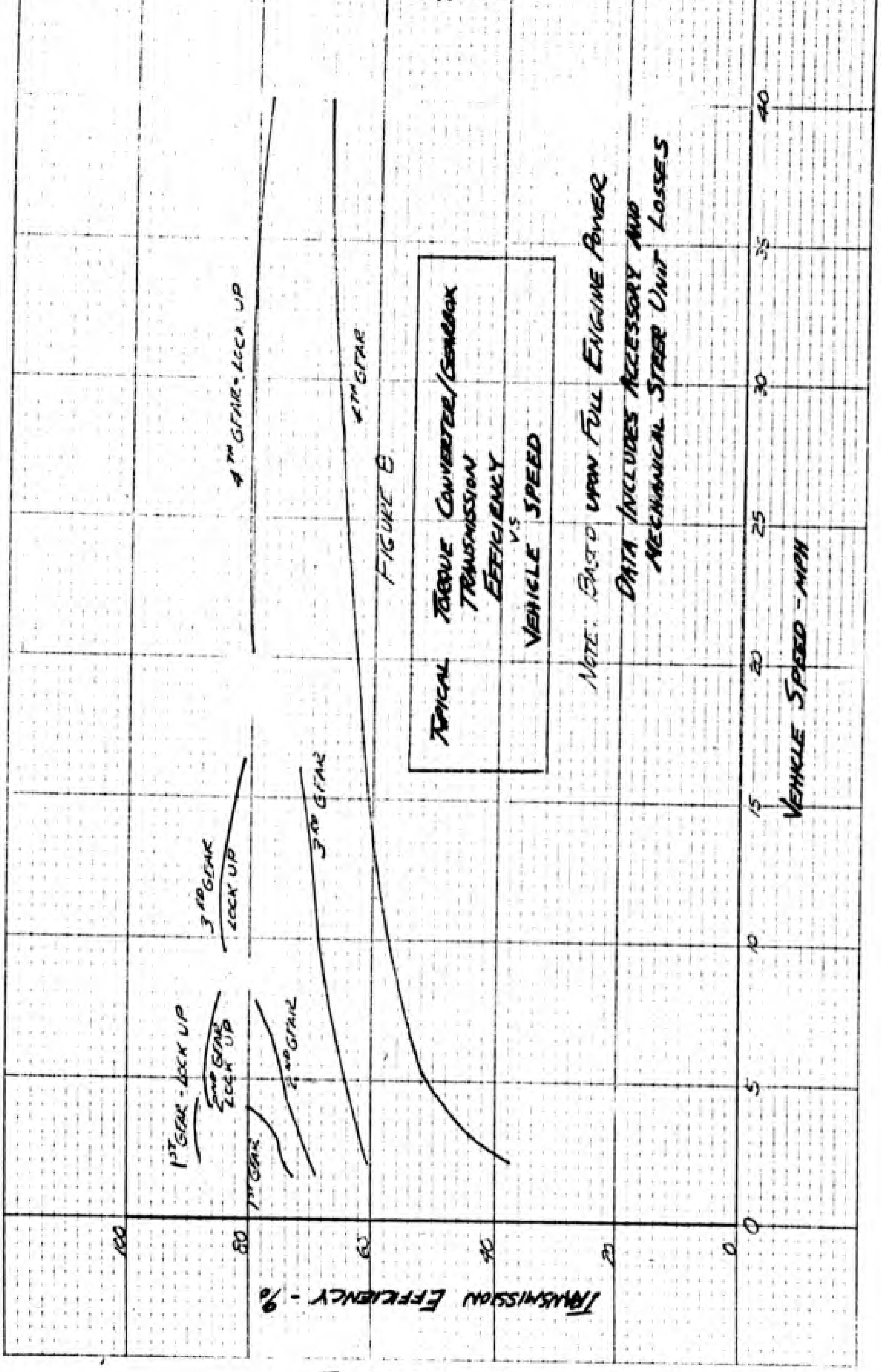

Typical Torque Converter/Gearbox Transmission Efficiency vs Vehicle Speed
Figure 8
Note: Based upon Full Engine Power
Data Includes Accessory and Mechanical Steer Unit Losses
Vehicle Speed - MPH
Transmission Efficiency - %
1st Gear - Lock Up
2nd Gear Lock Up
3rd Gear Lock Up
4th Gear - Lock Up
1st Gear
2nd Gear
3rd Gear
4th Gear
0
5
10
15
20
25
30
35
40
0
20
40
60
80
100

FIGURE 8

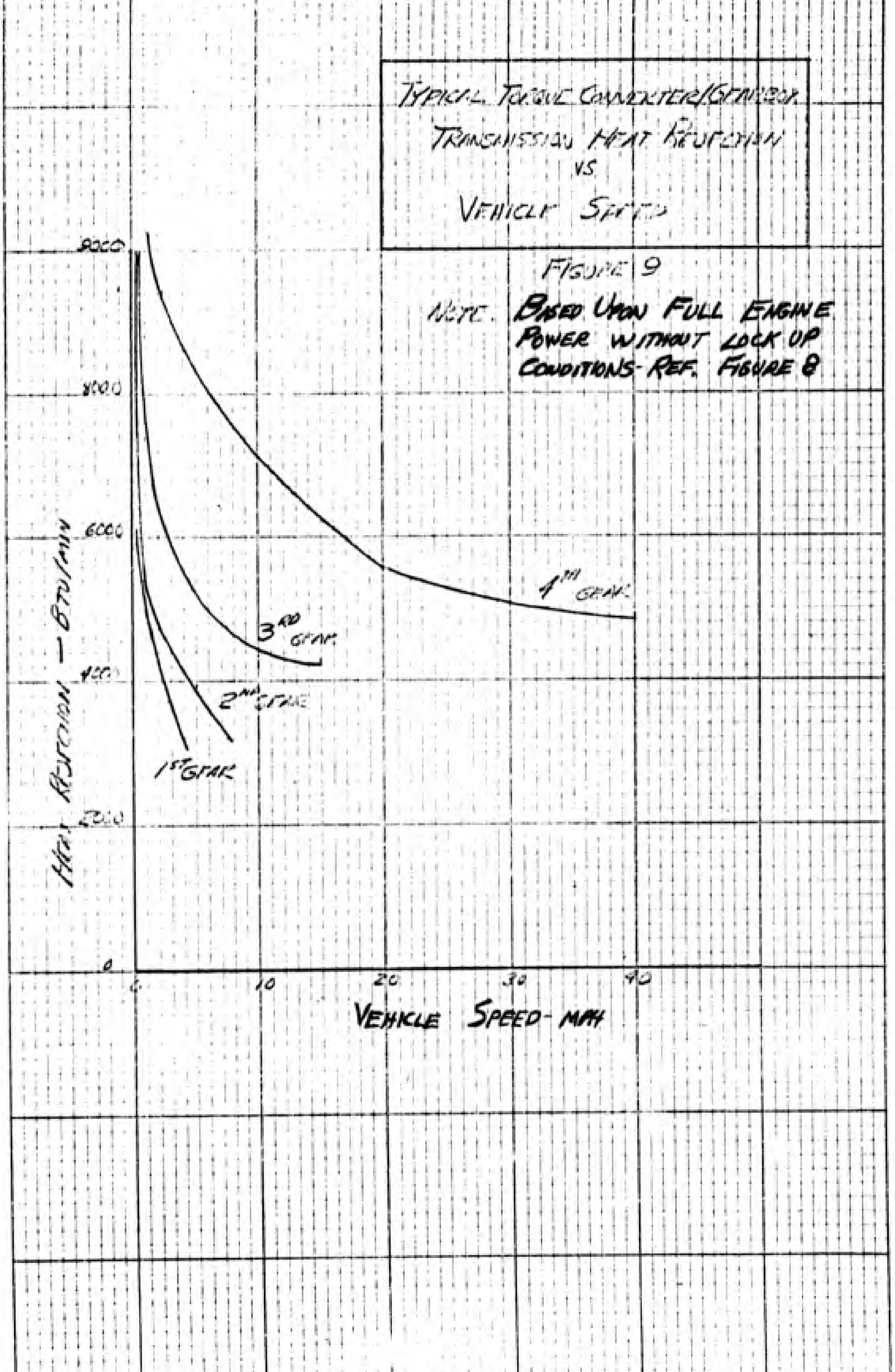

Typical Torque Converter/Gearbox
Transmission Heat Rejection
vs
Vehicle Speed
Figure 9
Note: Based Upon Full Engine Power without Lock Up Conditions- Ref. Figure 8
Heat Rejection — BTU/MIN
8000
6000
4000
2000
0
1st Gear
2nd Gear
3rd Gear
4th Gear
0
10
20
30
40
Vehicle Speed- MPH

FIGURE 9

Sundstrand Engineering Proposal 1950A- P1
Appendix

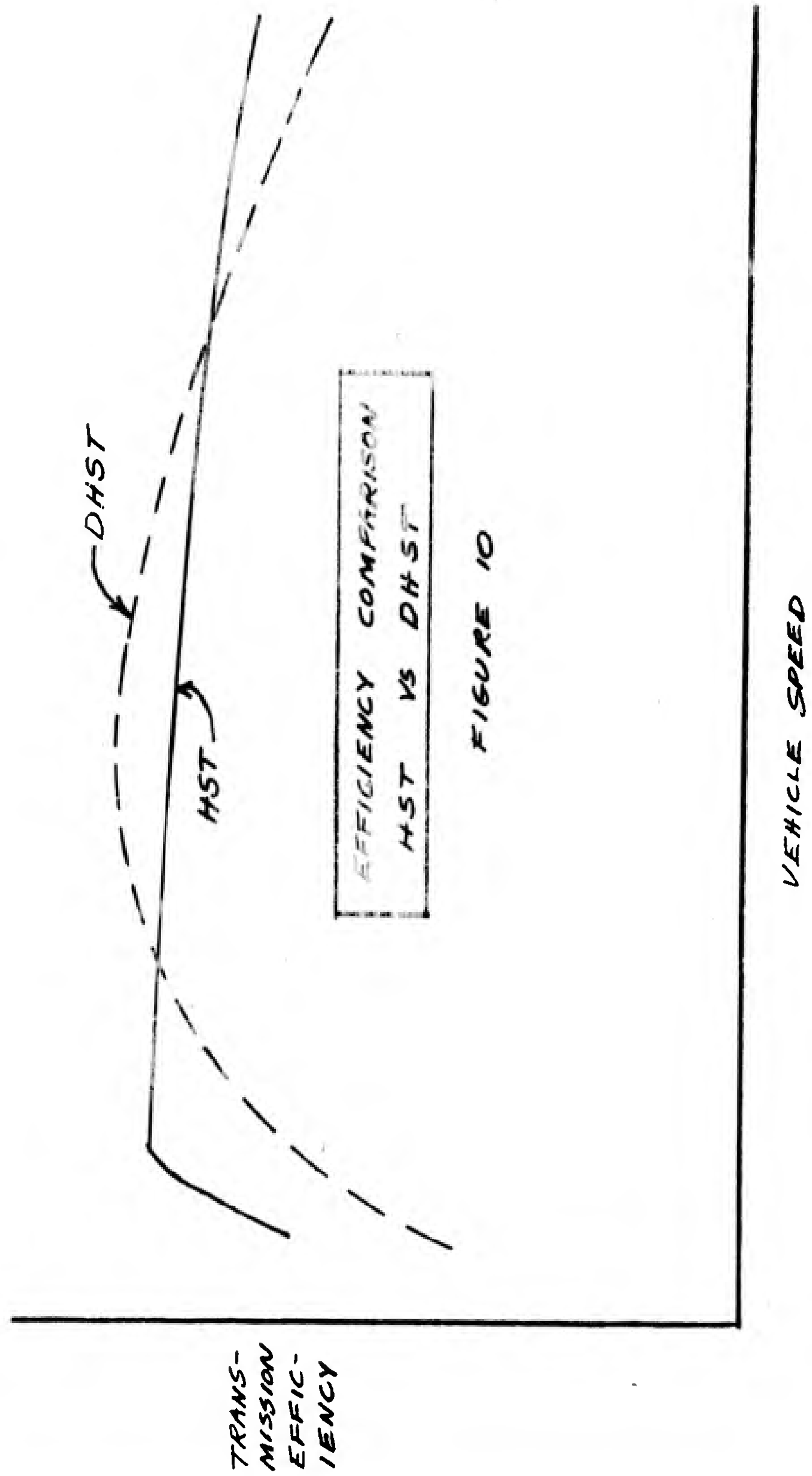
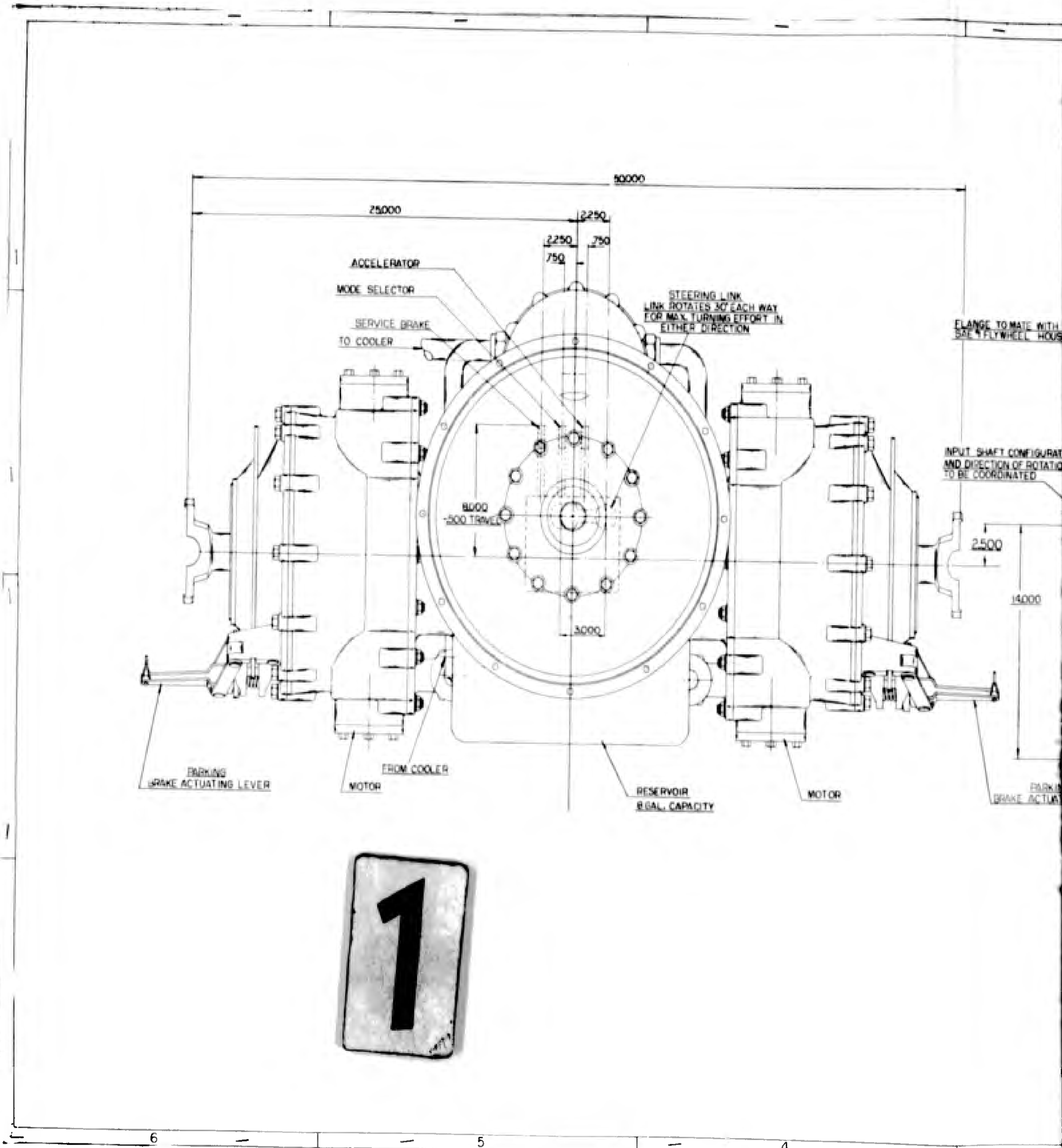
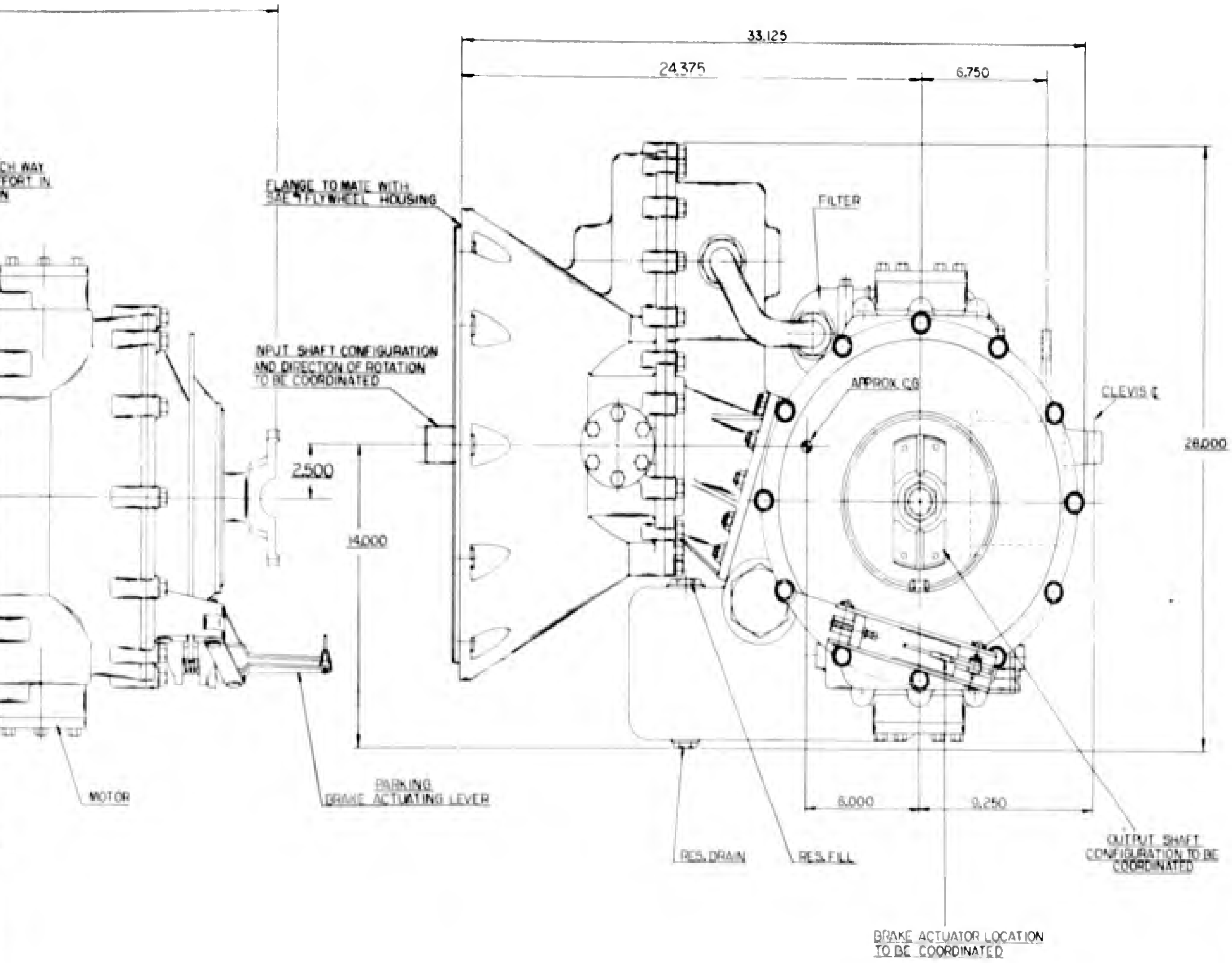


FIGURE 10


50000
25000
2250
2250
750
750
ACCELERATOR
MODE SELECTOR
SERVICE BRAKE
TO COOLER
STEERING LINK
LINK ROTATES 30° EACH WAY
FOR MAX. TURNING EFFORT IN
EITHER DIRECTION
FLANGE TO MATE WITH
INPUT SHAFT
AND DIRECTION OF
TO BE COORDINATED
8.000
-.500 TRAVEL
2.500
14.000
3.000
PARKING
BRAKE ACTUATING LEVER
MOTOR
FROM COOLER
RESERVOIR
8 GAL. CAPACITY
MOTOR
6
5
4

1



MOUNTING PROVISIONS TO BE COORDINATED

PRELIMINARY

INPUT H.P. - 350 H.P.
OUTPUT TORQUE - 48,000 IN.LBS. (EACH SHAFT)
MAX. OUTPUT SHAFT SPEED 2800 RPM

TRANSMISSION OUTLINE VEHICLE

SUNDSTRAND AVIATION

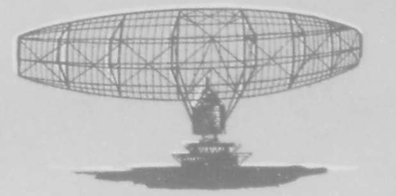
400026B

SCALE HALF

1950A-P1

L.V.T.

1950A-E6



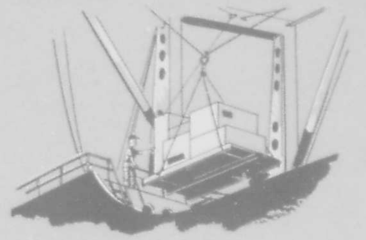
Machinery Hydraulics Division
Marine and Ordnance Department

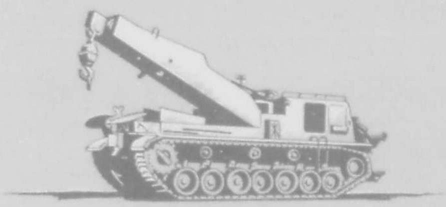
VICKERS

PROPOSAL NO. 2352A

TRANSMISSION FOR

TRACKED LANDING VEHICLE LVT PX11





VICKERS INCORPORATED
DIVISION OF SPERRY RAND CORPORATION
Marine and Ordnance Department
Waterbury 20, Connecticut

DISTRICT OFFICES: Detroit, Michigan • El Segundo, California • Los Altos, California • Seattle, Washington • Washington, D.C. • Waterbury, Connecticut

PROPOSAL NO. 2352A

TRANSMISSION FOR TRACKED LANDING VEHICLE (LVT) PX11)

Prepared by: H. J. Sculthorpe

H. J. Sculthorpe

Approved by: R W Jeffery

R. W. Jeffery

R L Ruth

R. L. Ruth
Chief Engineer

October 6, 1961
Rev. October 27, 1961

VICKERS INCORPORATED
DIVISION OF SPERRY RAND CORPORATION
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PREFACE

This proposal covers a hydro-mechanical transmission for a tracked landing craft weighing 35,000 lbs with a maximum of 245 hp at the sprockets.

INTRODUCTION

The system proposed is a completely integrated self-contained double split drive. Vickers Incorporated has selected the proposed system as the optimum consistent with the performance and size requirements.

Other systems which were considered, included a purely hydrostatic drive, a conventional single split drive and a purely mechanical drive. The first was rejected on the basis of efficiency, the second on the basis of size and the third on the basis of power availability and flexibility.

Specific areas in which the proposed system excels are:

1. The hydraulic components are modified, standard commercially developed units well within their design range. For example, maximum system pressure is 3000 psi, whereas the hydraulic units are capable of 5000 psi operation; a reserve capacity of over 50%.

2. Since a minimum of 70% of total power is transmitted through the mechanical power train, its high efficiency predominates. Use of hydraulics for the remaining 30% (max.) allows smooth, stepless speed variation. Thus, the best features of mechanical and hydrostatic transmissions are provided while supressing the effects of their shortcomings. Moreover, sizing the hydrostatic power train on the basis of 30% of full power yeilds minimum space and weight.

3. Continuous ratio control is provided, allowing engine operation at minimum fuel consumption, and allowing transmission of full power regardless of speed.

4. The hydraulic units are directly coupled through a common valve plate. This eliminates all high pressure piping and connections.

5. Speed modulated, positive steering is provided for all terrestrial and amphibious operations. Steering control sensitivity increases at lower vehicle speeds and stability at high speeds is assured.

6. Excess steering capacity is provided for amphibious maneuvers.

7. Capability of pivot center steering is provided.

8. Capability of 100% steering power regeneration is provided. Power regeneration is through a central cross shaft and therefore does not affect hydraulic component sizing. This permits in excess of 125% of full power to be transmitted to one track.

9. Speed range changes are accomplished with no power on the clutches, assuring smooth transition and optimum clutch life and reliability.

10. This program can be executed utilizing existing personnel and procedures at Vickers. In addition, follow-through capability exists in the areas of testing, development and production. Hardware proposed is within the existing state of the art at Vickers.

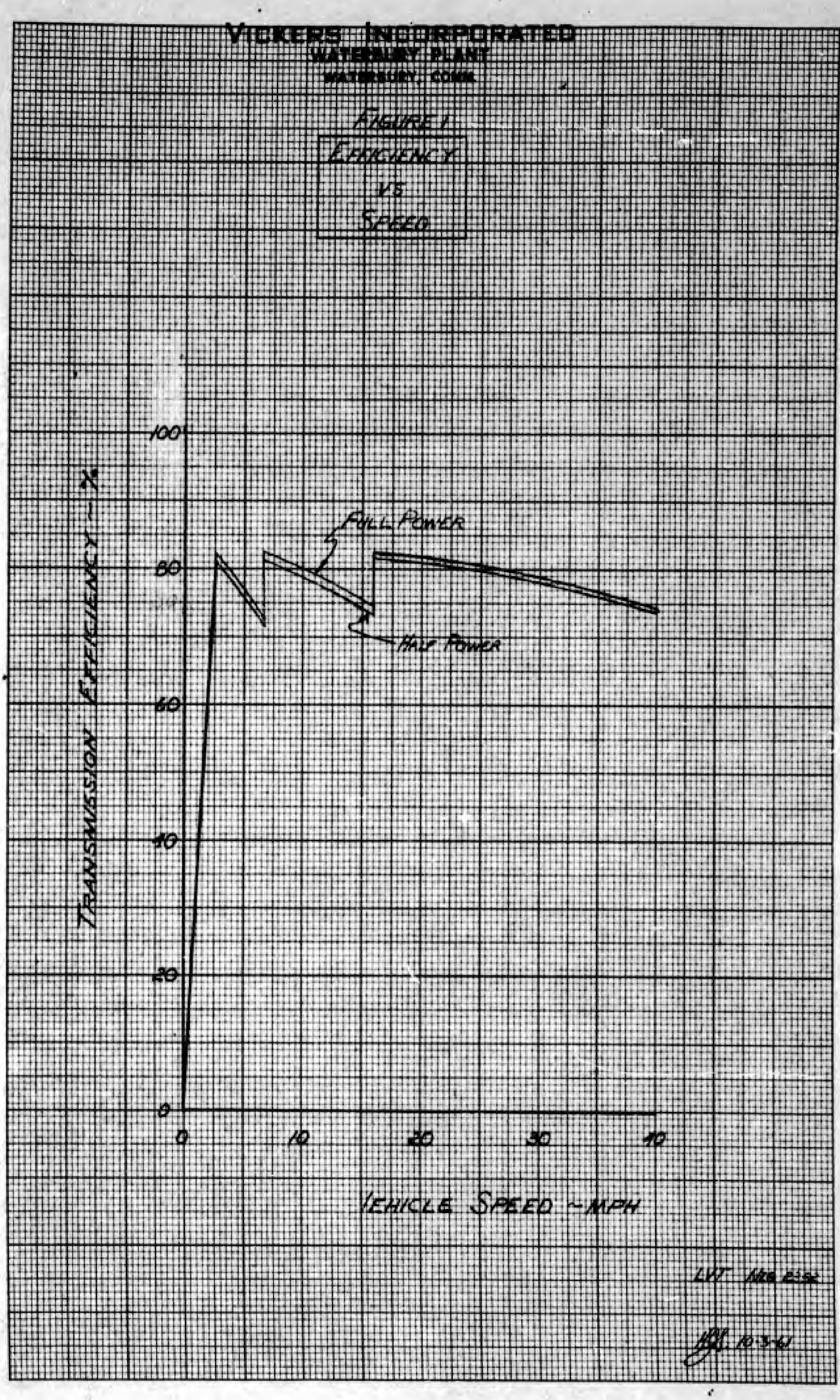
DISCUSSION

System Performance

Estimated overall transmission efficiency is presented in Figure 1. This curve is based on a mechanical power train efficiency of 92% which is somewhat conservative. It is also based on hydraulic unit efficiencies derived from test data.

Salient features of the steering system include excess steering capacity, turning radius modulation, capability of 100% steering power regeneration and pivot center steering.

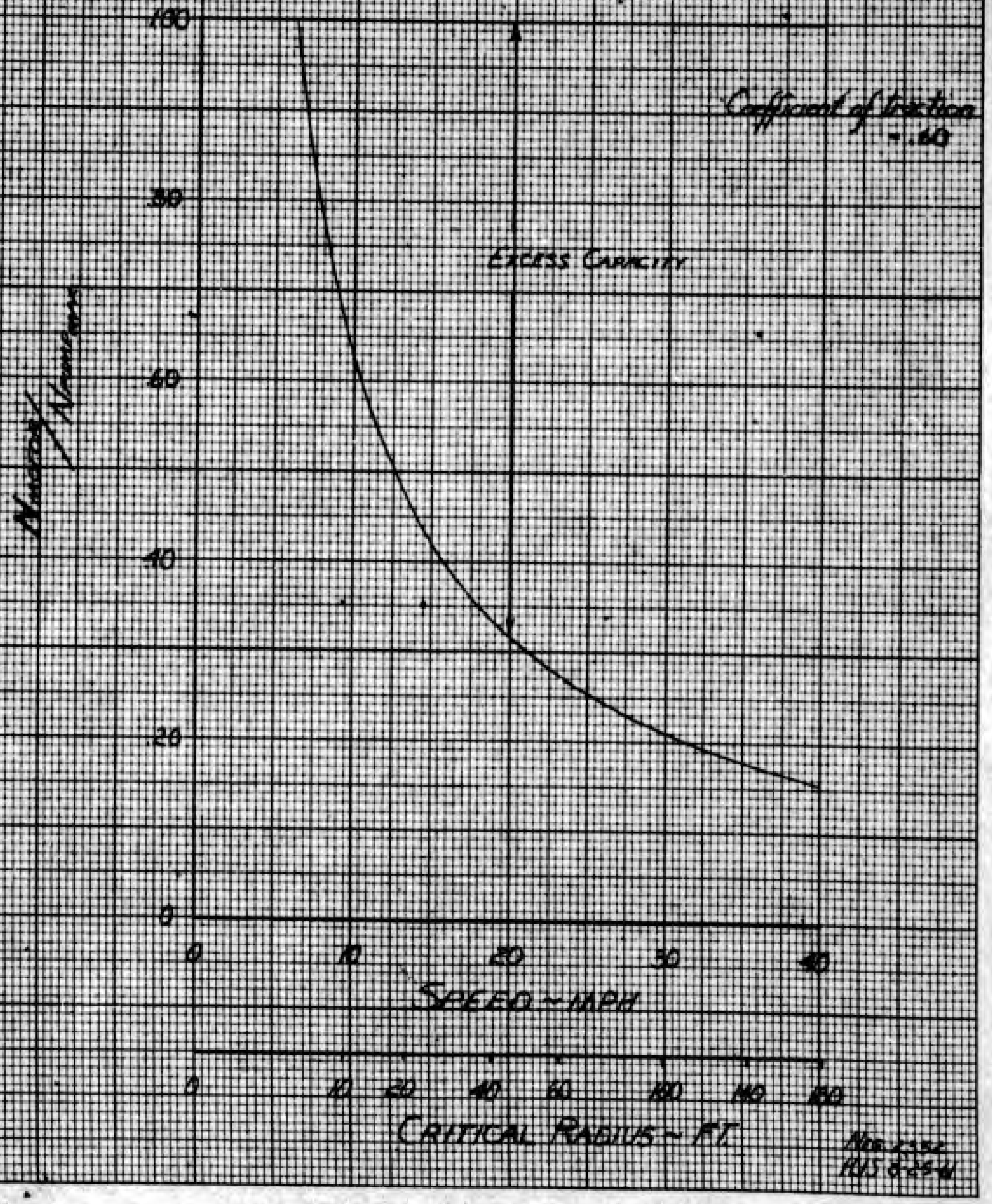
Figure 2, presents the steering motor speed ratio for steering control. The lower curve represents the maximum motor speed that will occur at the point when the vehicle will skid laterally in a turn with a lateral traction coefficient of 0.6. This therefore, presents maximum motor speed for terrestrial operation. The upper line represents maximum pump speed. Since the pump and motor are of the same maximum displacement, the difference represents the reserve steering capability for the lower engine speed and also for amphibious operation where it is desirable to have a much wider track ratio range than required for land operation.


VICKERS INCORPORATED
WATERBURY PLANT
WATERBURY, CONN.
FIGURE 1
EFFICIENCY
VS
SPEED
TRANSMISSION EFFICIENCY - %
100
80
60
40
20
0
FULL POWER
HALF POWER
0
10
20
30
40
VEHICLE SPEED ~ MPH
LVT
10-3-61

VICKERS INCORPORATED
WATERBURY PLANT
WATERBURY, CONN.

FIGURE 2

STEERING PERFORMANCE
AT SIDESLIP



The effect of speed on vehicle turn radius is presented in Figure 3. This illustrates the desirable feature of decreasing sensitivity with increasing speed at a given steer control position. Thus, stability is assured at higher speed. Critical radius is also plotted on this curve, showing that the vehicle can be "spun out" above 6 MPH with full steer and above 12.5 MPH with 1/2 Steer.

Since regenerated power is transferred directly from one output planetary to the other by the central cross shaft, it is not necessary to oversize the ratio system to transmit more than engine power. The availability of 100% steering power re-generation to one track should prove useful for critical maneuvers, such as turning on a side slope or transition from amphibious to terrestrial operation.

Capability of pivot center steering should enhance vehicle maneuverability at low speeds. This feature is possible be-cause the steering system is designed to function independently from the ratio system.

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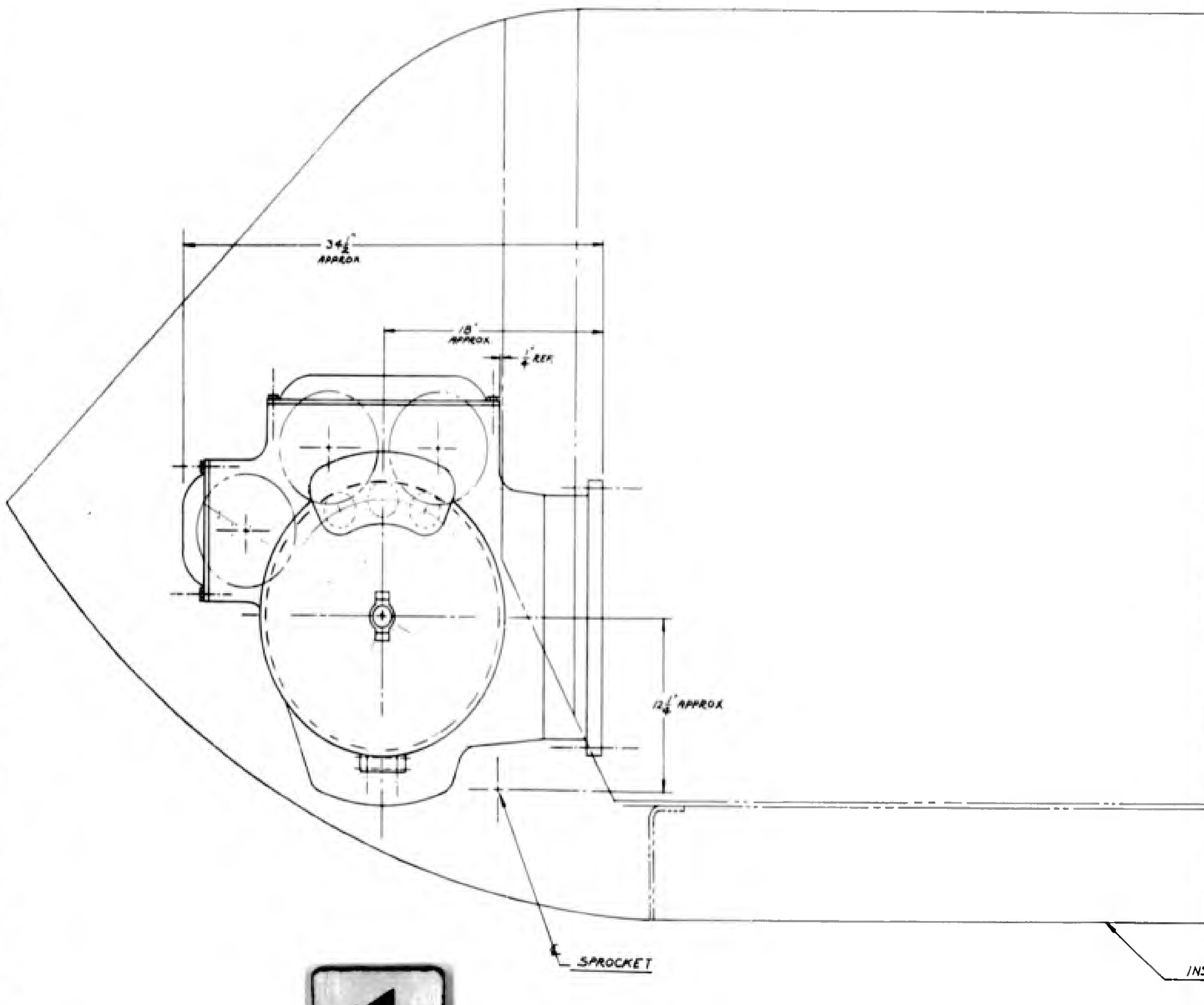
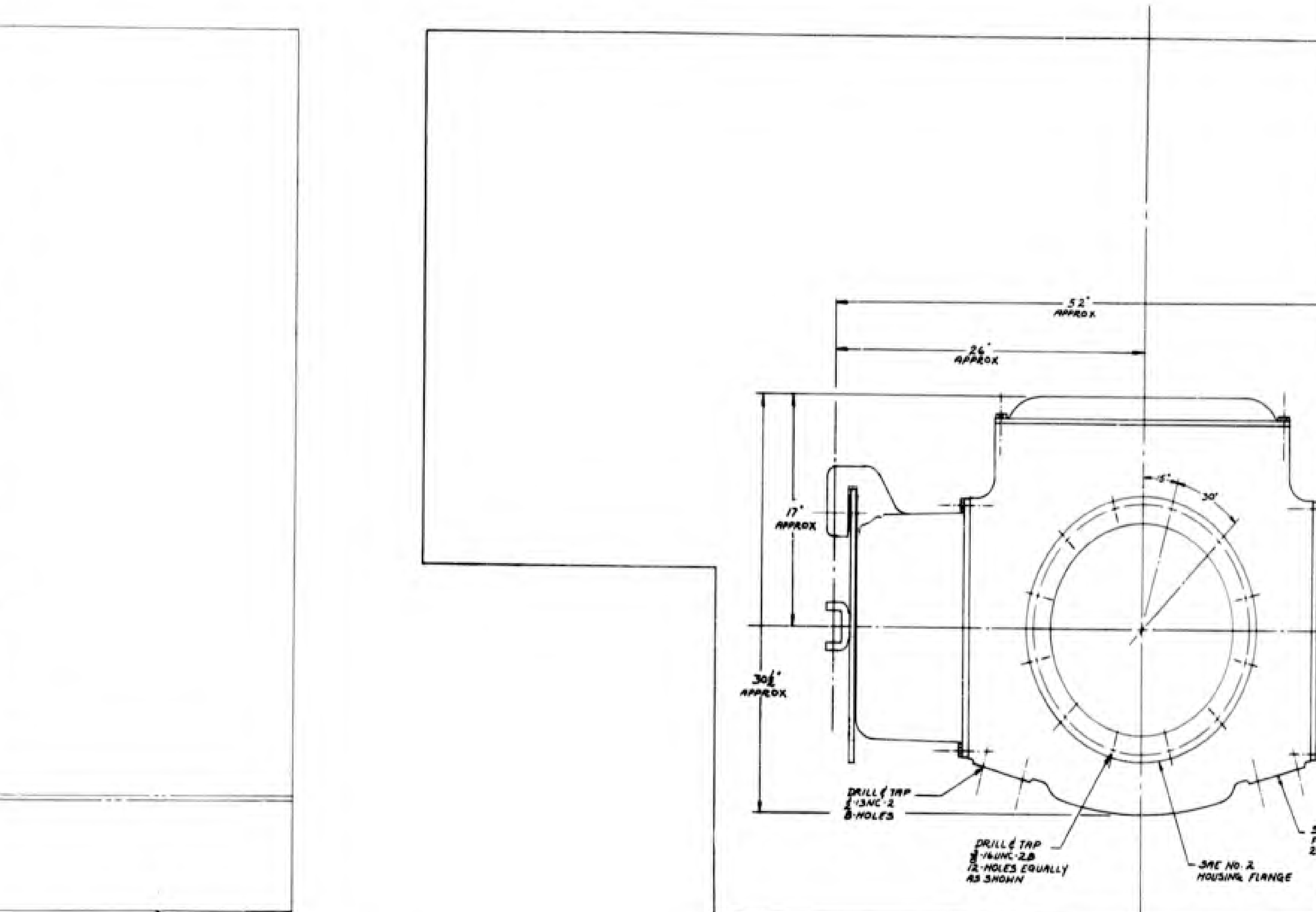
WATERBURY PLANT
WATERBURY, CONN.

FIGURE 3

SPEED MODULATION
OF
TURN RADIUS

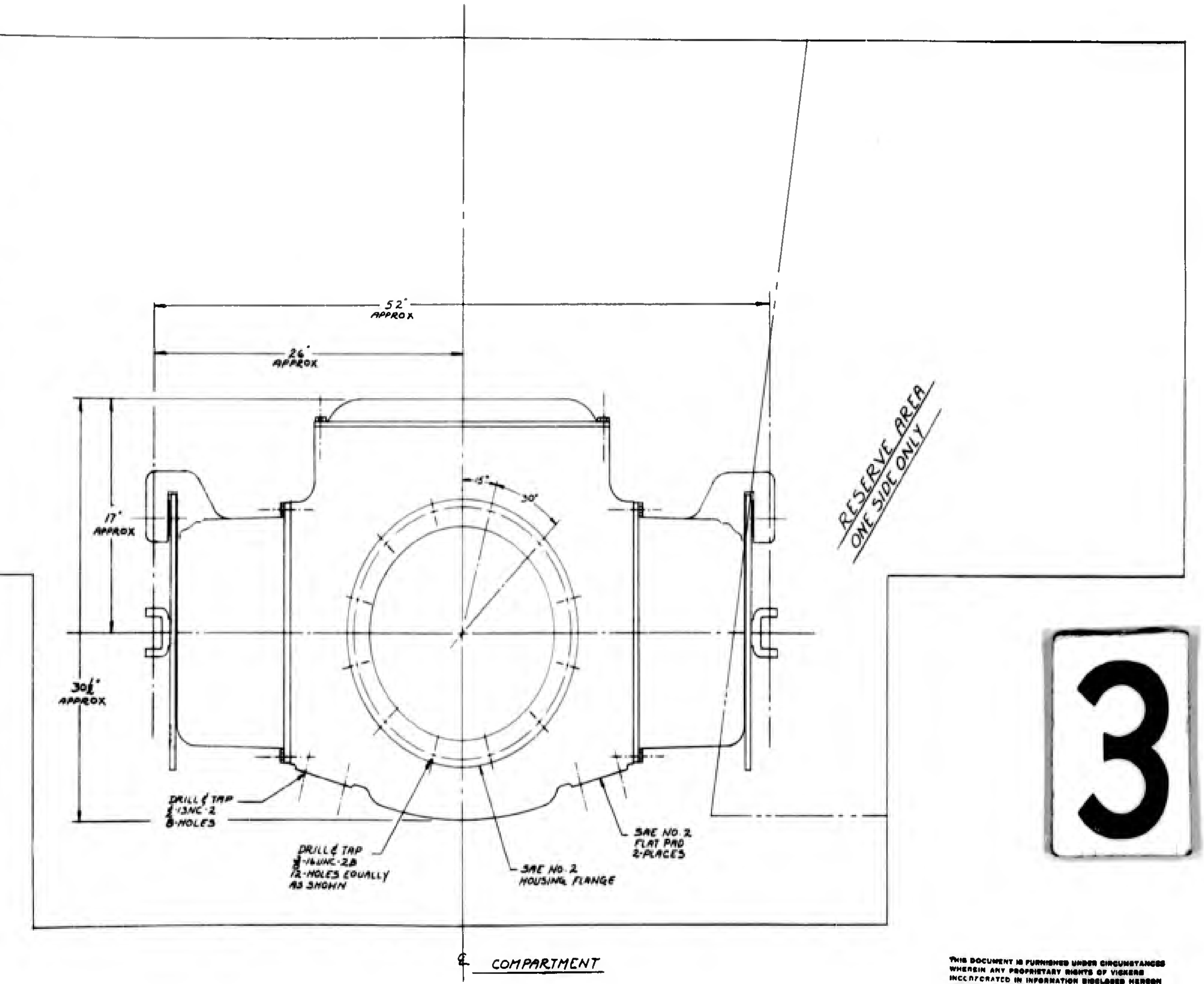


1


52"
APPROX
26"
APPROX
17"
APPROX
30½"
APPROX
15°
30°
DRILL & TAP
½-13NC-2
8-HOLES
DRILL & TAP
⅜-16UNC-2B
12-HOLES EQUALLY
AS SHOWN
SAE NO. 2
HOUSING FLANGE
INSIDE HULL OUTLINE
℄ COMPARTMENT

2

ISSUE
(1) ADDED BRAKE ASSY
(2) Added weight note 10/11/61



3

ESTIMATED WEIGHT 1150 lb.

H. THOMPSON 8/17/61

40,000 LVT
TRANSMISSION
1/4 = 1"

Neg. 2352

[illegible] 8/18/61

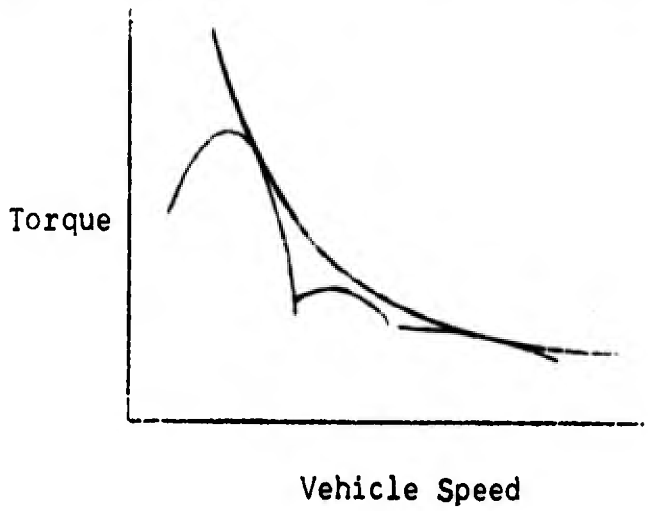
SK-3344 2

Design Approach and Considerations

It was assumed that performance and size considerably advanced beyond that found in existing equipment is needed; therefore, it was apparent that a unique approach was essential.

First examine the choices available. Besides the proposed system there are schemes such as a straight mechanical system, a single split transmission and a purely hydrostatic transmission. A purely mechanical system, while light and efficient by itself, lacks flexibility. At a given vehicle speed, engine speed is fixed at a point which is not the optimum speed for fuel consumption. Continuously variable ratio and steering capabilities are impossible without considerable loss in efficiency due to hydrokinetic elements or slipping clutches. Torque available to the tracks is, at certain points, considerably less than in a continuously variable system. This is illustrated by the figure following. The upper curve illustrates torque available with a continuously variable transmission. The other curves represent torque available from a purely mechanical system. Since variation of ratio occurs in steps, it is not possible to operate the engine at full power except at a number of points corresponding to the number of gear changes.

At the expense of weight and size, the number of gear changes can be increased to more closely approximate the proposed system and other continuously variable transmissions.



Addition of a torque converter adds flexibility but at the expense of efficiency and weight. Onput torques of the proposed system and a six speed transmission with torque converter are compared in Figure 4. Output power of the two systems are compared in Figure 5. Although torque converter action is shown in first and third gear, it is expected that realistic operation would include torque converter action at other speeds as well, reducing output power and efficiency.

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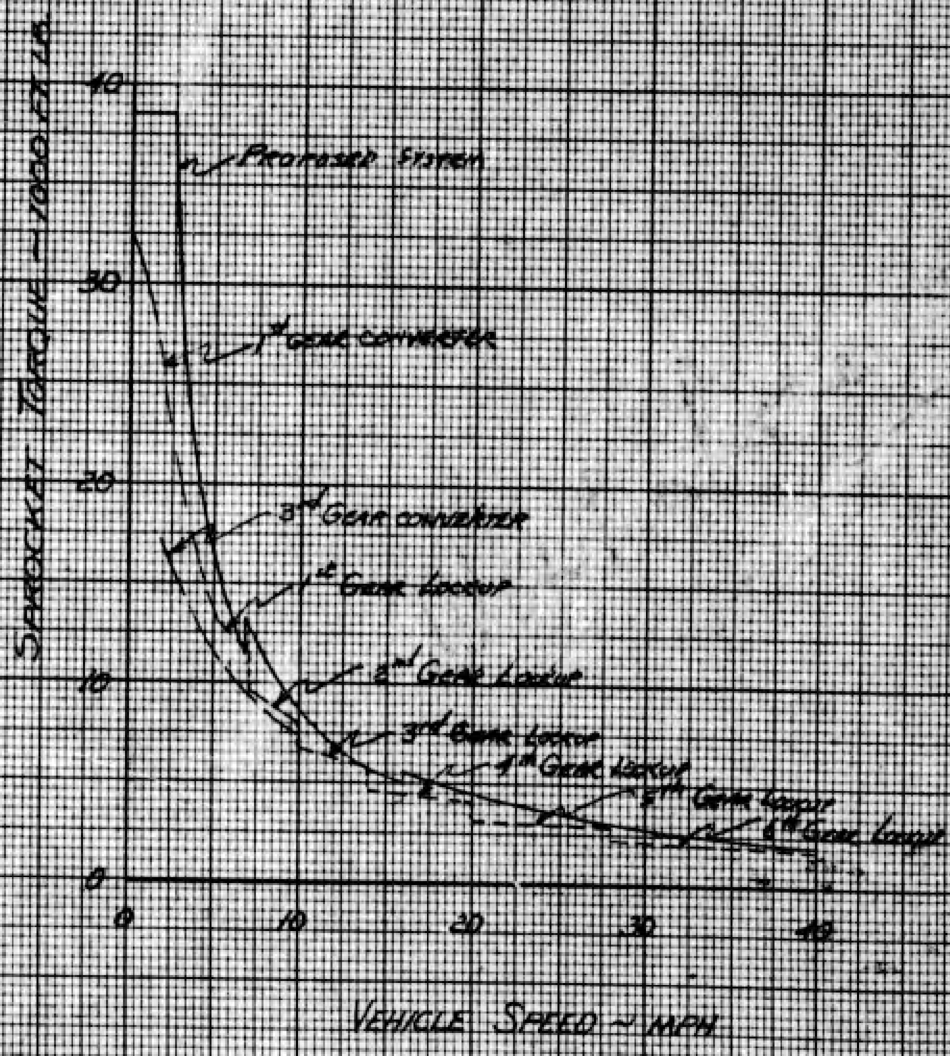
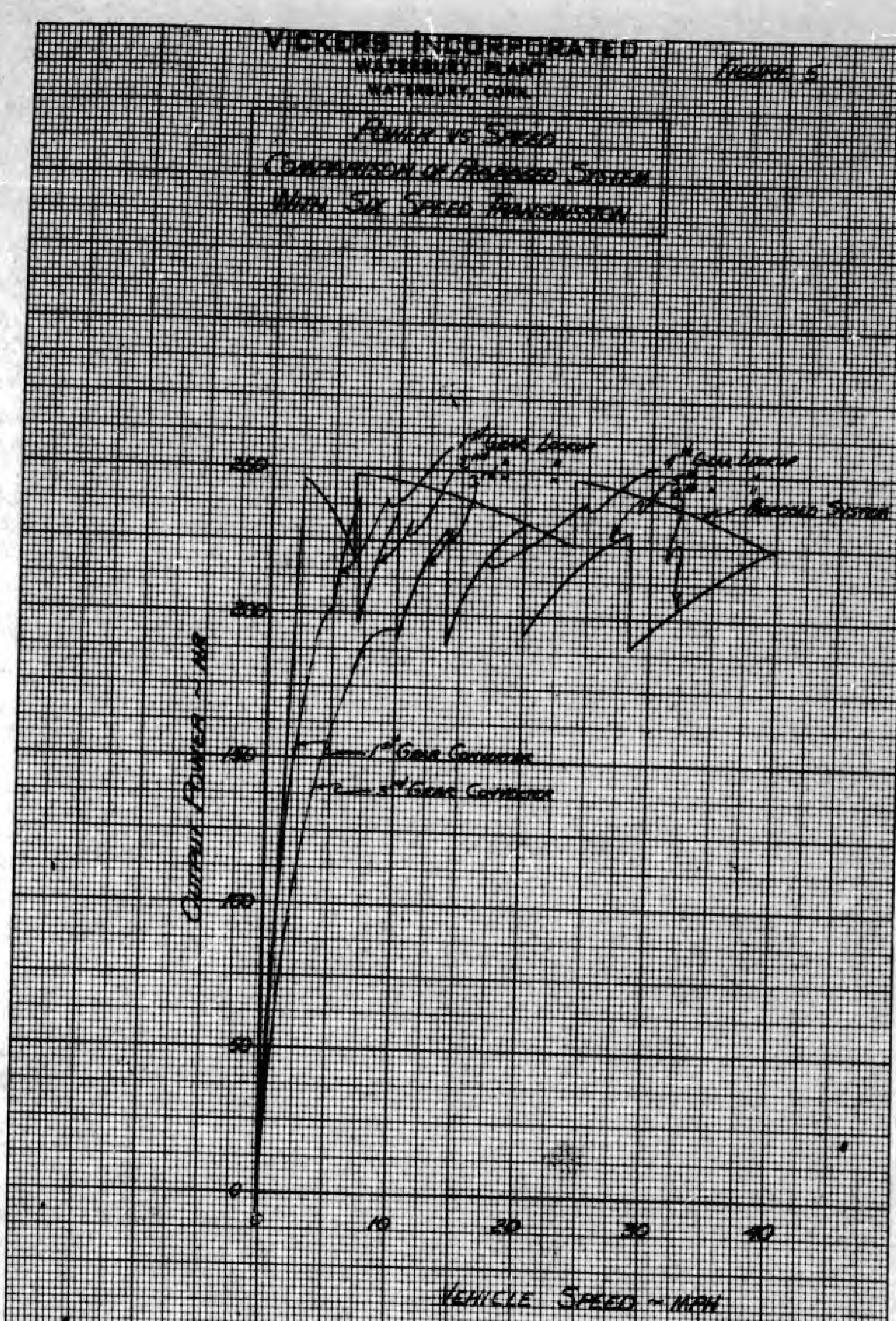
WATERBURY, CONN.

FIGURE 4

TORQUE VS SPEED
COMPARISON OF PROPOSED SYSTEM
WITH SIX SPEED TRANSMISSION





VICKERS INCORPORATED
WATERBURY PLANT
WATERBURY, CONN.
FIGURE 5
POWER VS SPEED
COMPARISON OF PROPOSED SYSTEM
WITH SIX SPEED TRANSMISSION
GEAR LOCKUP
4th GEAR LOCKUP
PROPOSED SYSTEM
1st GEAR CONVERTER
3rd GEAR CONVERTER
OUTPUT POWER ~ HP
250
200
150
100
50
0
10
20
30
40
VEHICLE SPEED ~ MPH

Single split transmissions are generally arranged so that the hydrostatic power train regulates a reaction member of the primary planetary gear train to vary overall gear ratio. Two considerations here affect the size of the hydrostatic portion of the power train. First, the amount of power split through the hydrostatic portion increases with speed range at full power. Secondly, this arrangement demands hydrostatic power train regeneration for reverse operation.

These effects cause size and weight of hydraulic components to increase and overall efficiency to decrease considerably. It should be noted that in a single split transmission full power goes through the mechanical power train at only one speed whereas in the proposed system, this condition cccurs four times.

The greater availability of smaller hydrostatic components was another consideration. It should be noted that, in the proposed system, the hydrostatic components in the steering and ratio circuits are identical.

A purely hydrostatic transmission is an excellent choice from the standpoint of continuous ratio variability. However, it is not the optimum for this system from the standpoint of efficiency. Moreover, hydraulic components in the purely hydrostatic system would be roughly twice the size of those in the proposed system.

The problem therefore, was to select a transmission which combines the high efficiency and light weight of a planetary drive with the continuous ratio variability of a hydrostatic drive.

This we have done. Several secondary advantages accrue as a result of this selection, such as greater availability of smaller units, use of identical hydrostatic rotating groups on both the steering and ratio control circuits, use of higher speed hydrostatic elements (simplifying the planetary and interconnecting gear sets) and improved operation of the hydrostatic elements by reducing piston speed and maintaining constant system pressure. Moreover, the clutches shift under no load conditions, improving life and reliability considerably.

100% steering power regeneration is available in the proposed system with a corresponding increase in vehicle capability. Consideration of some critical maneuvers warrant this built in reserve steering capacity. An example would be landing one track on the bank of a stream during the transition from amphibious to terrestrial operation. Certain undesirable characteristics of other steering transmissions have been eliminated as well. An example would be downhill operation. Many steering transmissions have a tendency to reverse steer in this condition. The proposed system does not.

In summary, after careful consideration of the requirements and choices available, Vickers offers the proposed system as the optimum combination of hydrostatic and mechanical transmissions for this vehicle.

Vickers In-Line Units

The Vickers In-Line Units, which are proposed herein as the hydrostatic portion of the transmission, are of the positive displacement, axial-piston type. They are designed to operate either as fixed or variable displacement pumps or motors.

Nine pistons reciprocate in a cylinder block that is splined to a drive shaft as shown in the illustration following. Shoes swaged to ball-ended pistons are held against an inclined swash plate by a spring-loaded shoe plate.

Another spring is used to load a spherical washer placed in contact with the shoe-plate spherical slot. This spring load is also used to hold the cylinder block against the integral valve plate-cover assembly.

In variable displacement units, the swash-plate angle is adjustable in order to vary the piston stroke. The swash plate is mounted in a movable yoke which can be controlled manually by a lever or handwheel attached to the yoke point.

In pressure-compensated units, the yoke movement is controlled by a compensator valve mounted on the valve plate. The compensator strokes the yoke against the return spring to provide variable delivery through a predetermined pressure range.

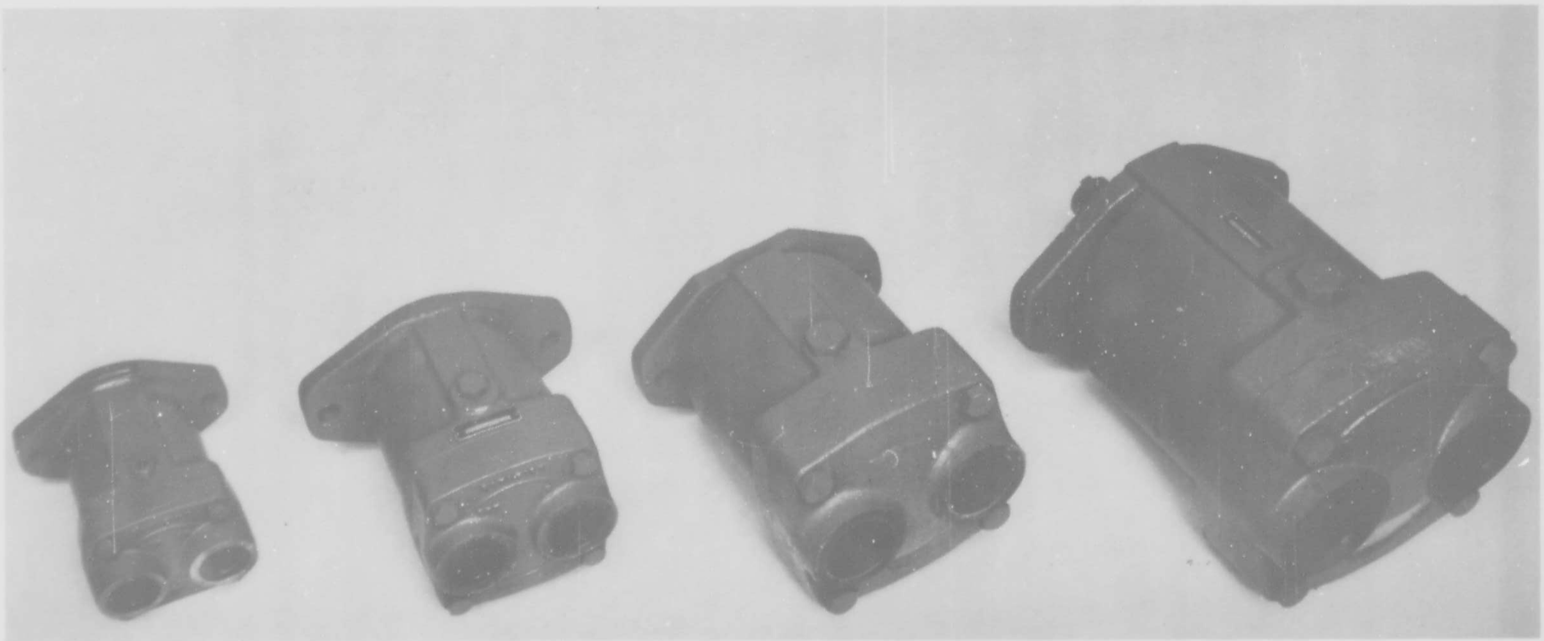


FIGURE T - 5, 10, 20, 40 GPM FIXED INLINE PUMP/MOTORS

VARIABLE DISPLACEMENT PISTON PUMP/MOTOR

VICKERS INCORPORATED
DIVISION OF SPERRY RAND CORPORATION
Marine and Ordnance Department
Waterbury 20, Connecticut

DISTRICT OFFICES: Detroit, Michigan • El Segundo, California • Los Altos, California • Seattle, Washington • Washington, D.C. • Waterbury, Connecticut

PROPOSAL NO. 2425

PROPELLER DRIVE

FOR

TRACKED LANDING VEHICLE

LVT PX11)

Prepared by R R Hurley
R. R. Hurley

Approved by R W Jeffery
R. W. Jeffery

Date: October 26, 1961

SYSTEM DESCRIPTION

The system consists of two servo controlled variable displacement pumps that provide pressure to two fixed displacement motors. The motors drive a common shaft at 2000 RPM max. The shaft speed is reduced by a multiple contact gear train to the required 750 RPM.

The pumps are driven by an engine at 2600 RPM and provide 2717 PSI oil to the motors. The oil is transported to the motors by flexible lines thru the supporting strut. The motors are mounted in the propeller pod and the pod housing is the motor housing, as shown on SK 3353.

COMPONENT DESCRIPTION

Pumps:

The pumps are Vickers Model PVB-80 in-line variable delivery servo pumps. The valve plate will contain the supercharge and replenishing pump, the cross line relief valves, and the replenishing check valves.

The pump is controlled by a servo system. This can be either electro-hydraulic or hydro-mechanical, and the servo valve can be mounted directly on the pumps.

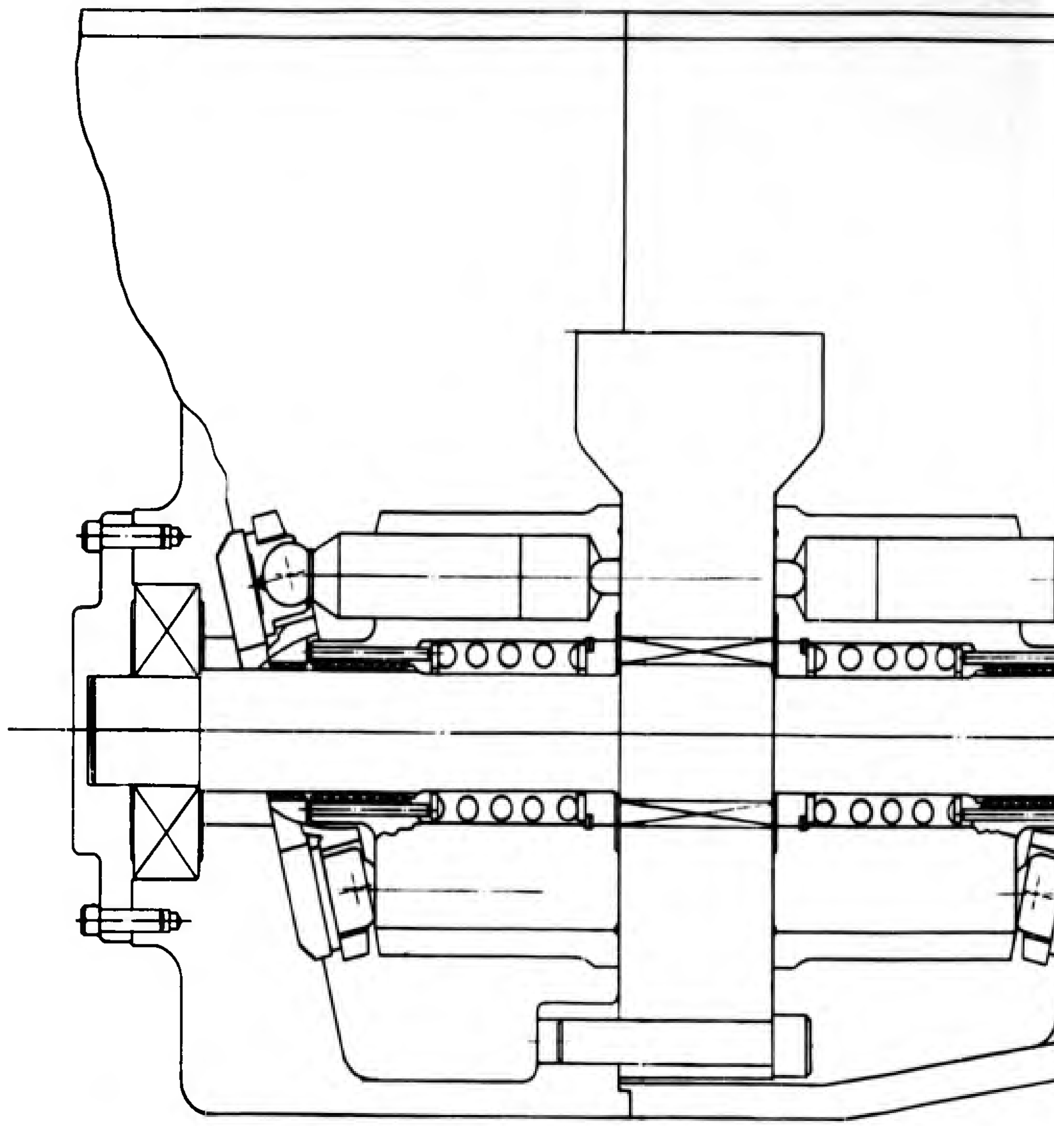
Motors:

The motors consist of two rotating groups and a common shaft. The motor housing is the pod housing, as shown on SK 3353. The motors have a common valve plate mounted between the rotating groups. The shaft is mounted on bearings at either end. Where the shaft passes thru the valve plate, a self-aligning journal bearing is used to limit shaft deflections. This packaging concept achieves the minimum spacial requirements desired for the system.

The motors are Vickers Model MFB-80 fixed displacement in-line motors. Speed is controlled by varying the flow to the motors.

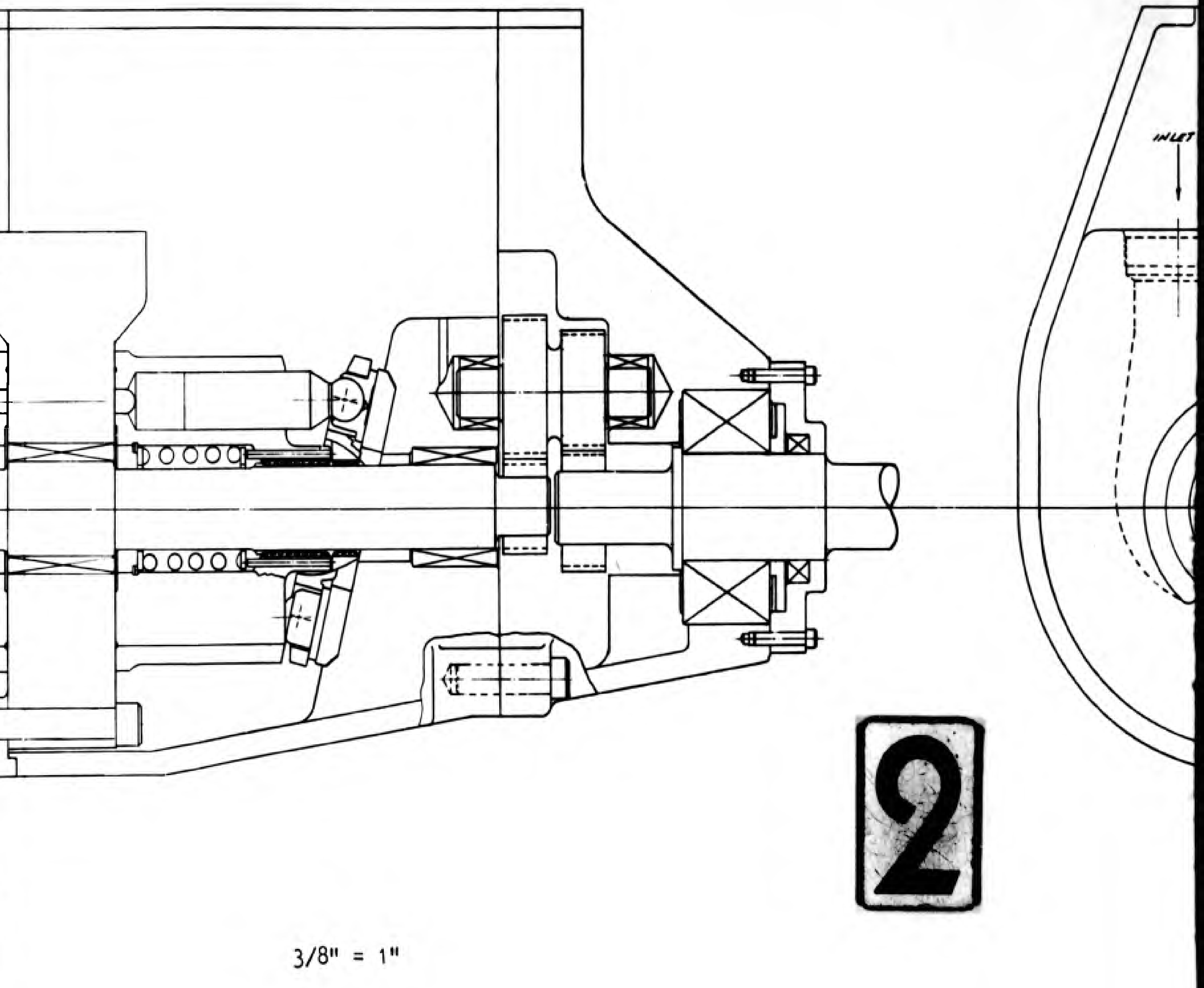
Gears:

The gear train is a multiple contact system. The multiple contact allows the load to be distributed over three gears, yielding a smaller package. The gear ratio is 2.68 to 1.00.



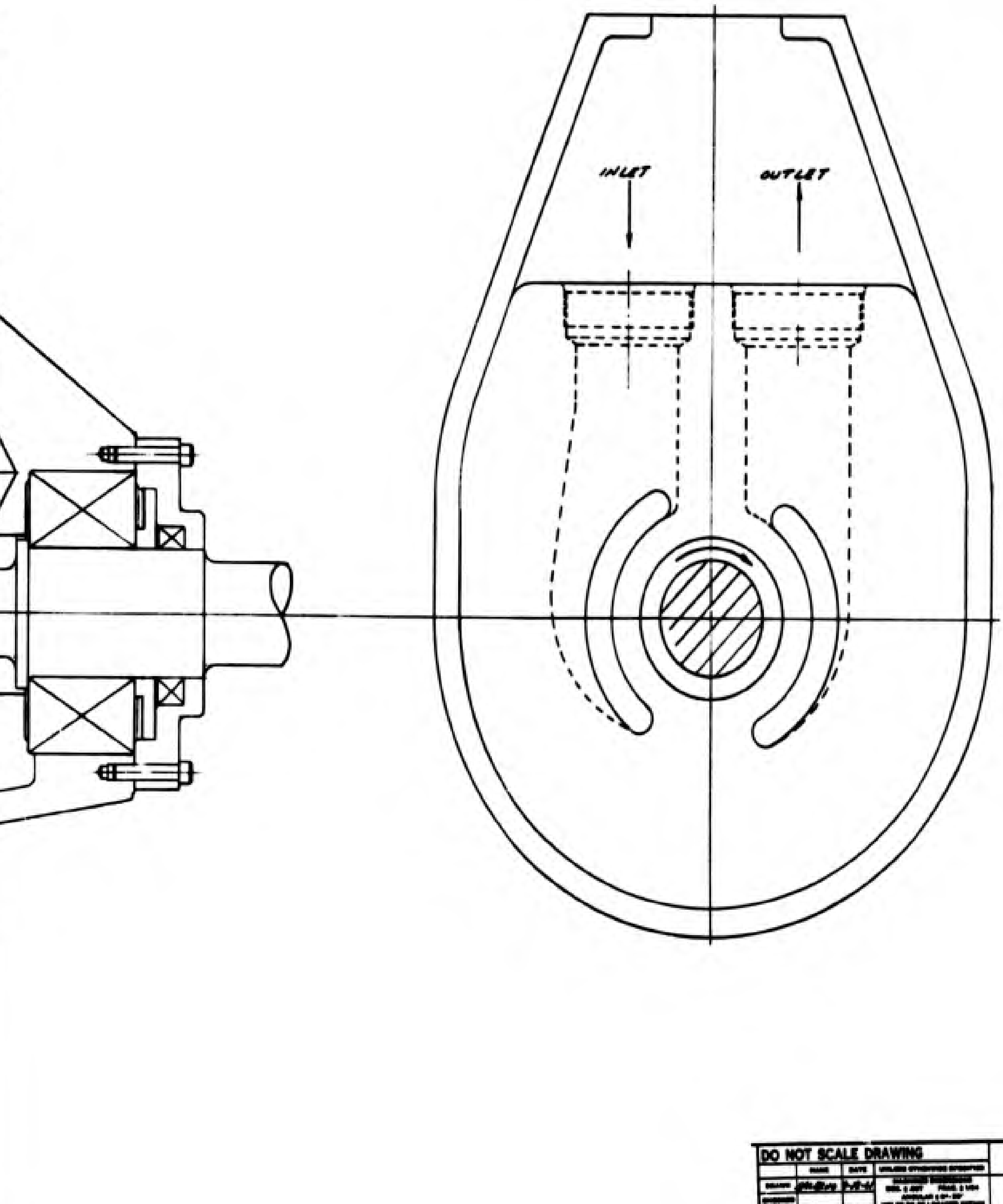
1

3/



2

3/8" = 1"



3

DO NOT SCALE DRAWING

VICKERS INCORPORATED

TITLE: DRIVE-PROPELLER HYDRAULIC

SCALE: FULL SIZE

R SK-3353

PROPOSAL FOR AN INFINITELY-VARIABLE HYDROMECHANICAL STEERING TRANSMISSION FOR THE LVTPX 11 TRACKED VEHICLE

STRATOS DIVISION

FAIRCHILD STRATOS CORPORATION

Bay Shore, Long Island, New York

PROPOSAL
FOR
AN INFINITELY-VARIABLE
HYDROMECHANICAL
STEERING TRANSMISSION
FOR
THE LVTPX 11 TRACKED VEHICLE

Prepared by: J. F. Sheehan
J. F. Sheehan,
Senior Engineer
Advance Design

Approved by: I. D. Miller
I. D. Miller,
Chief
Advance Design

E.P. 9053
October 25, 1961

By

STRATOS

A DIVISION OF
FAIRCHILD ENGINE & AIRPLANE CORPORATION
Bay Shore, Long Island, New York

INDEX

INDEX (Continued)

LIST OF ILLUSTRATIONS

SECTION 1

INTRODUCTION

This proposal is submitted to the Ordnance Division of FMC Corporation by STRATOS, a Division of Fairchild Stratos Corporation. The objective of the proposal is to design, fabricate, functionally bench-test and deliver to FMC the steering transmission system for a tracked combat vehicle of 35,000 lbs. gross vehicle weight, and to further provide the engineering services and support during the performance and development testing of the integrated gas turbine-power train package as may be required. This proposal is responsive to the applicable requirements of FMC Document "Specification for New Universal LVT", dated 20 July, 1961 (Rev. 27 July, 1961). The proposed power train utilizes a novel, hydromechanical transmission. This proposed transmission will provide sustained operation of the gas turbine portion of the power package at pre-determined ideal performance levels by fully automatic selection of the correct torque and speed ratios for any load condition, resulting in optimum overall fuel economy.

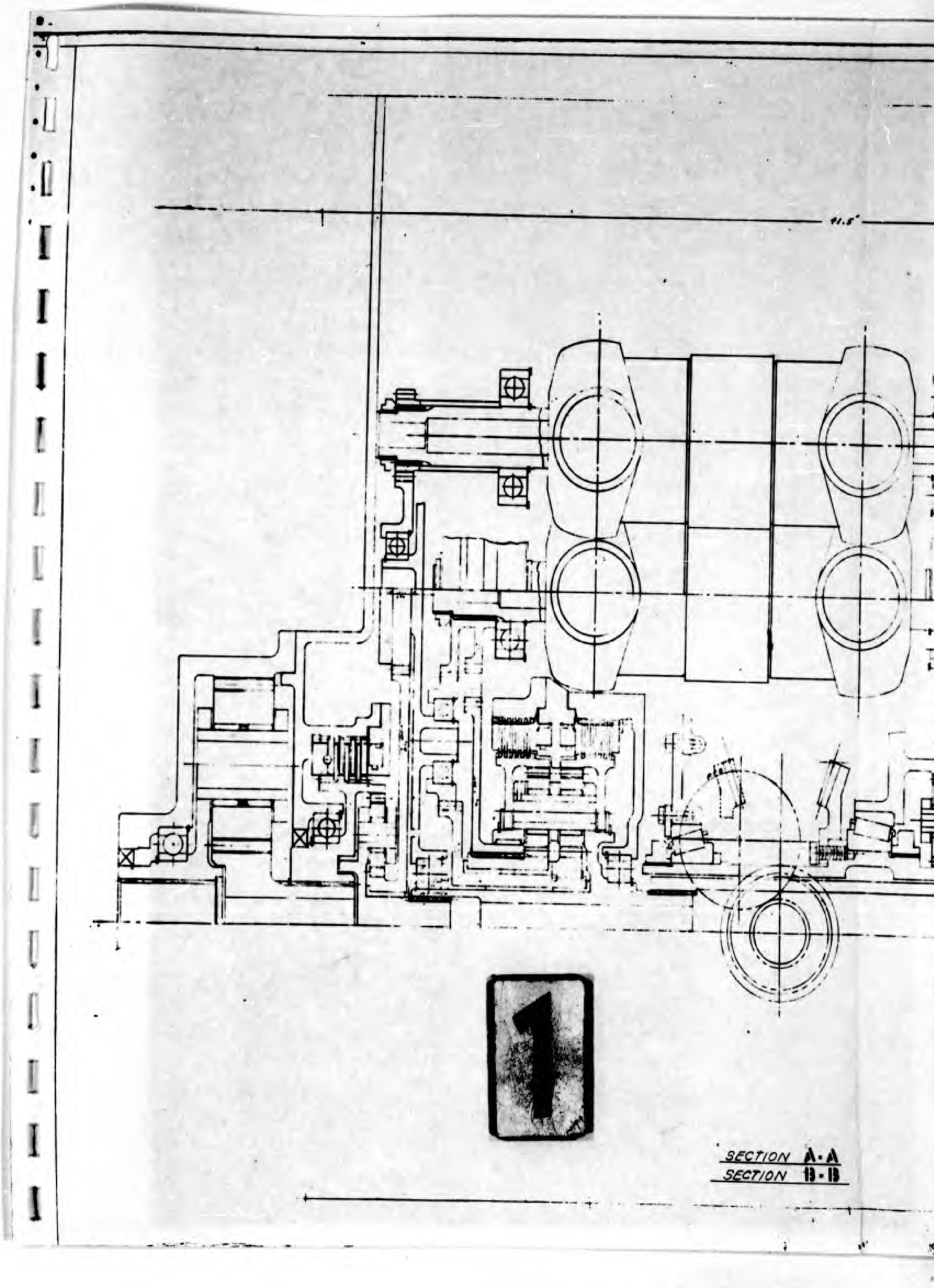
Ease of control will be a corresponding feature of this power train with driver fatigue and the need for driver's skill reduced to the minimum by provision of a simple control system. The control system consists of a power pedal, a steering lever, and a brake pedal, with driver-controlled override of the engine operating speed range available at all times for any desired operational condition. Similar transmissions are in advanced stages of design by STRATOS under the sponsorship of OTAC. The individual hydraulic components of this transmission have been bench and road tested in other transmission assemblies, and predicted system efficiency is thus based on actual test data.

SECTION II

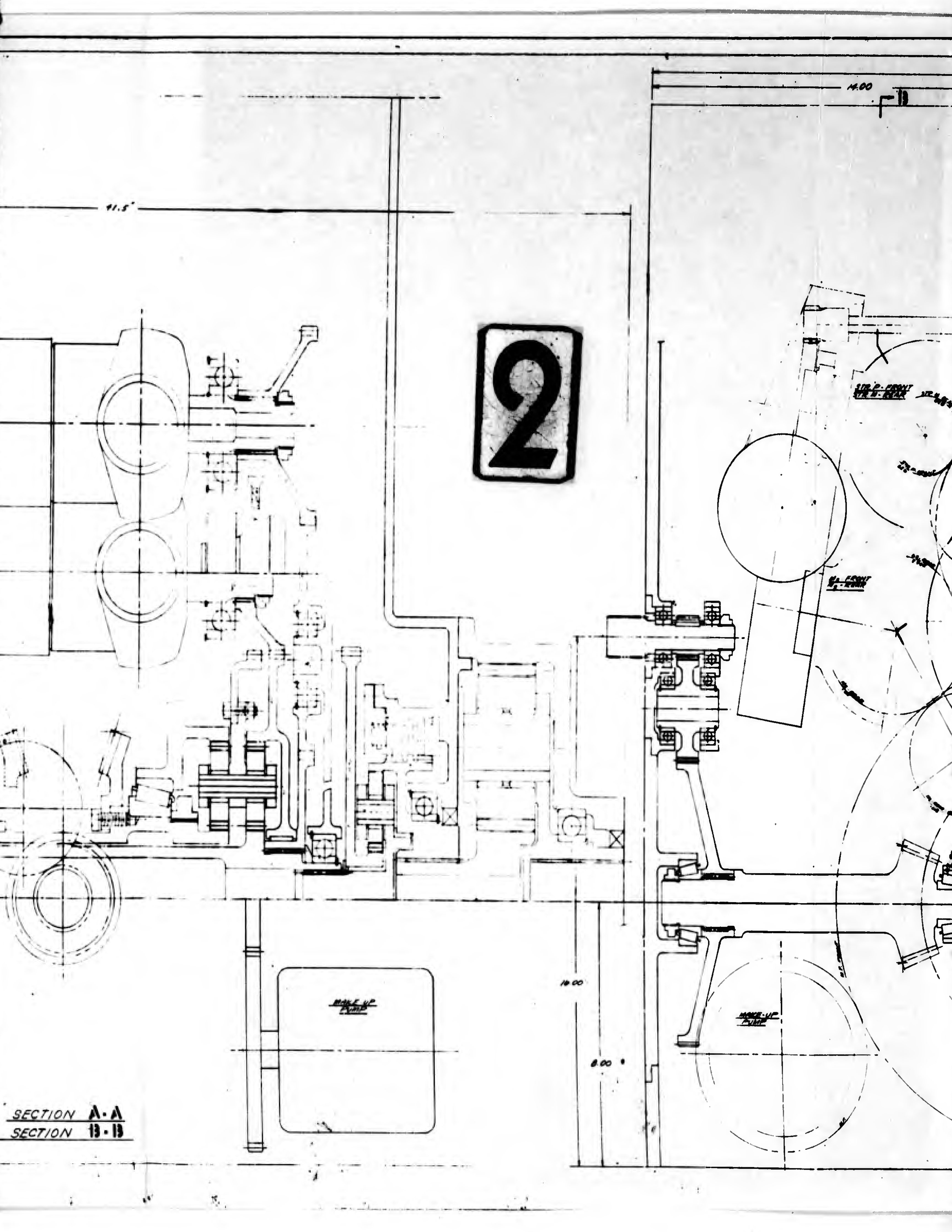
DESCRIPTION OF POWER TRAIN

Type and Definition - The proposed Stratos Hydromechanical power train, see Figure SL 10135, is a self-contained stepless power and steering transmission providing infinitely variable operation with a ratio range of 10:1. Forward and reverse driving modes are provided, with fully automatic control throughout. Vehicle braking capability is also provided. The arrangement of the elements is shown by the two schematics, Figures SK 10322 and SK 10325. This transmission is infinitely variable within its torque ratio limits and automatically controls the prime mover to operate at its best specific fuel consumption without driver effort and thus driver fatigue is minimized.

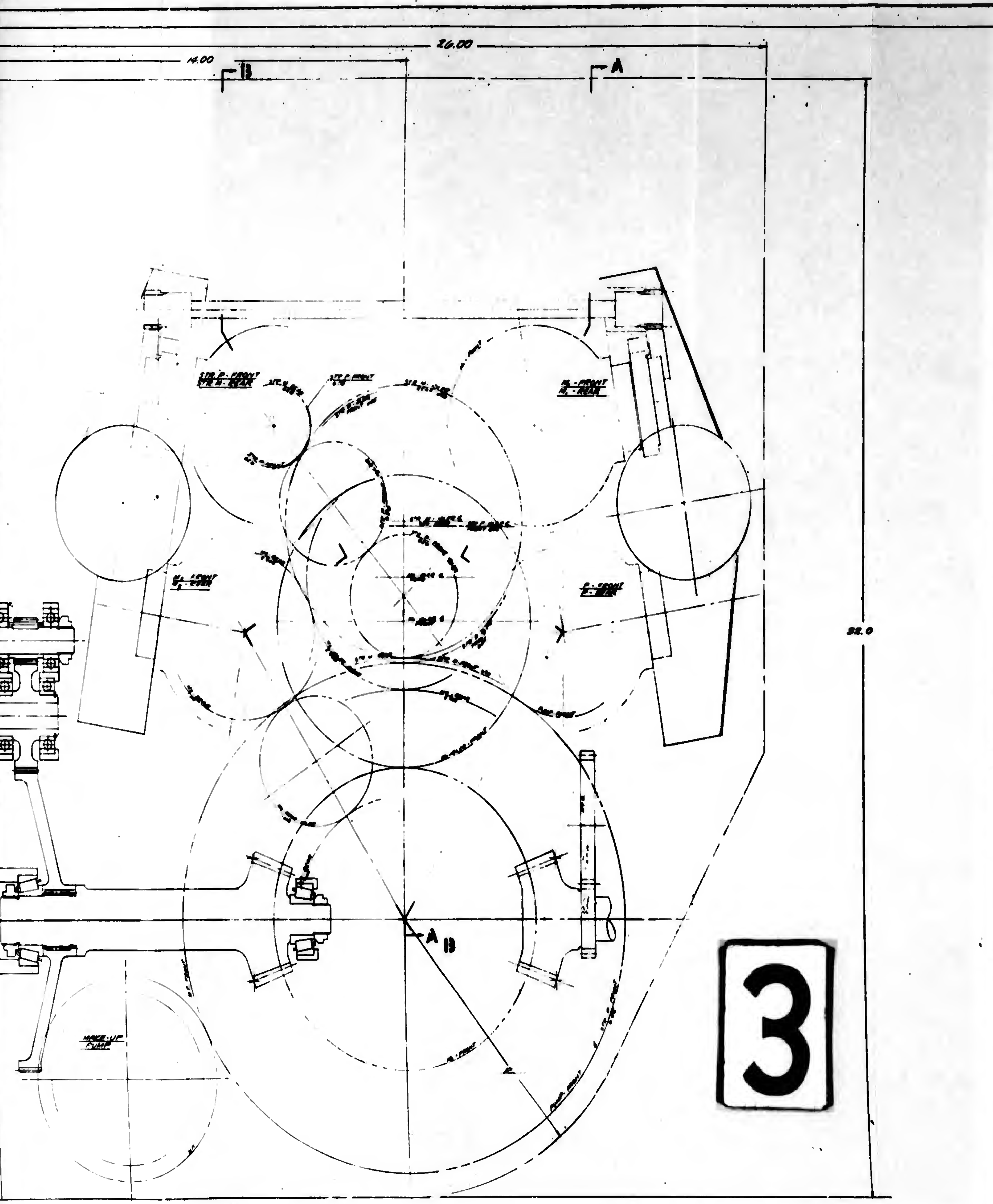
Configuration - Basically the unit consists of: an engine driven planetary differential gear with a hydraulic lock-out in the 1:1 ratio position; six identical variable displacement hydrostatic units with two functioning as pumps driven by the input gear

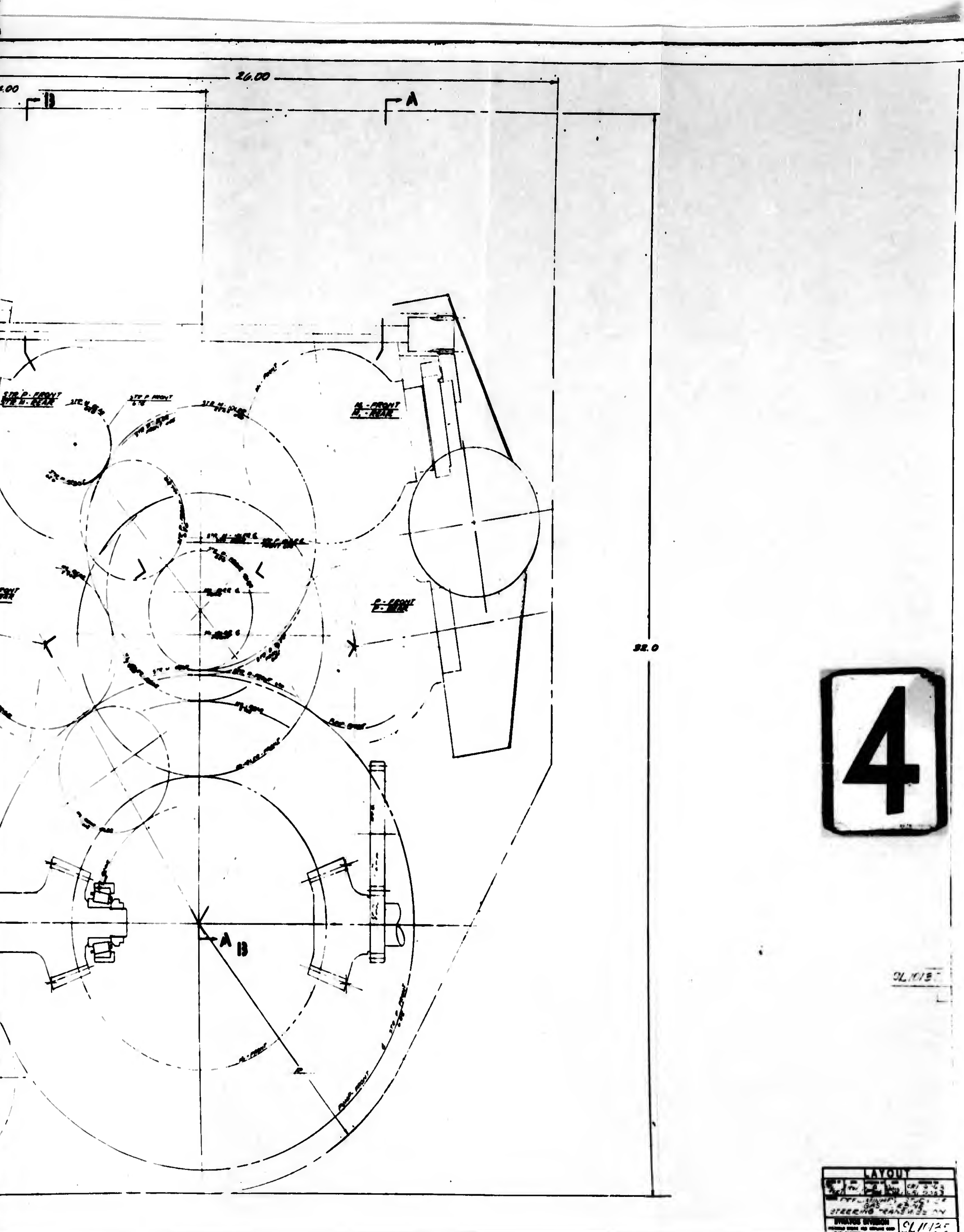


11.5"
1
SECTION A-A
SECTION B-B

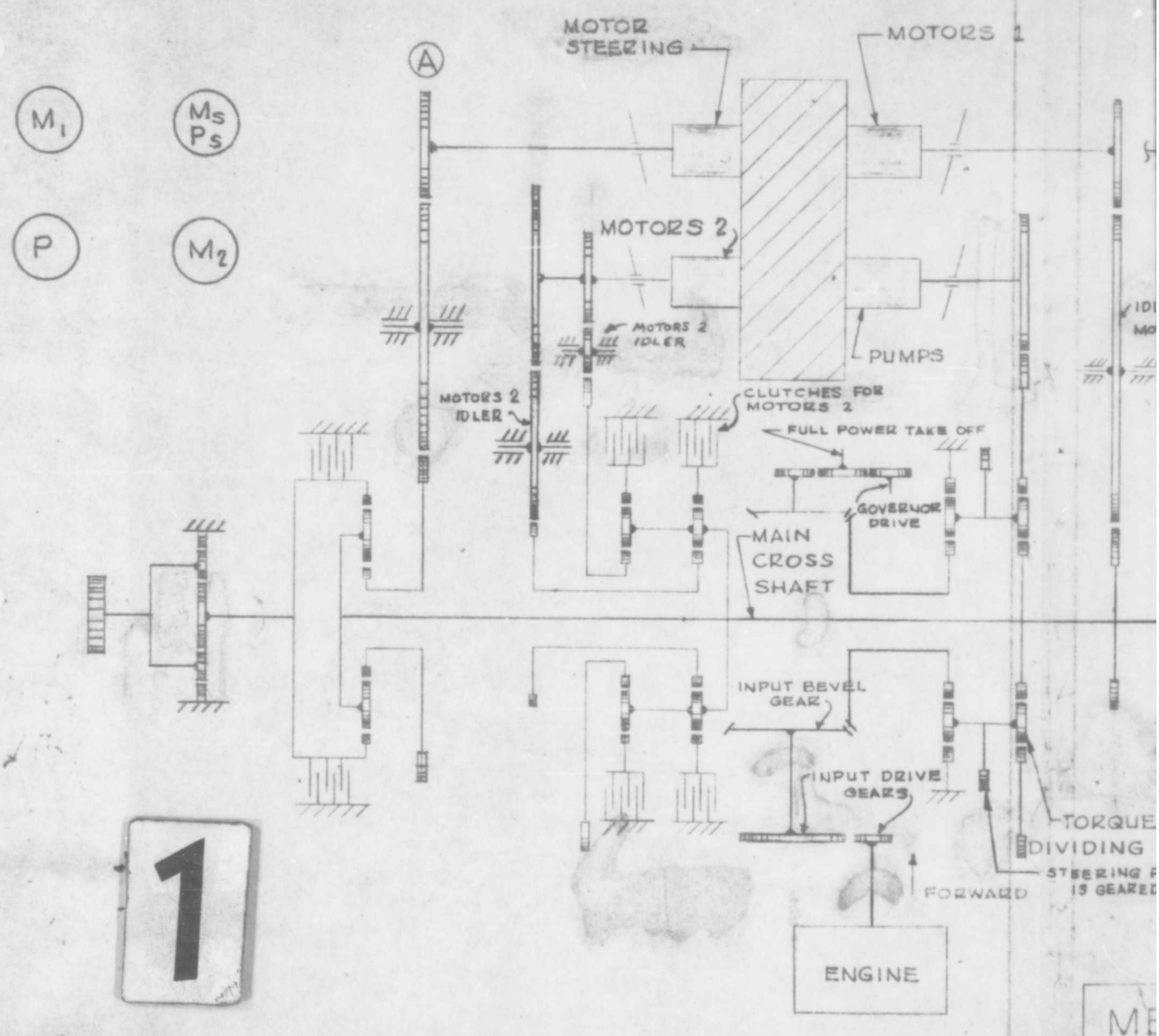


14.00
41.5"
2
MAKE-UP PUMP
MAKE-UP PUMP
16.00
8.00
SECTION A-A
SECTION B-B


26.00
14.00
B
A
A B
32.0
MAKE-UP PUMP
3



26.00
A
B
M.-FRONT
M.-REAR
P.-FRONT
P.-REAR
32.0
4
LAYOUT

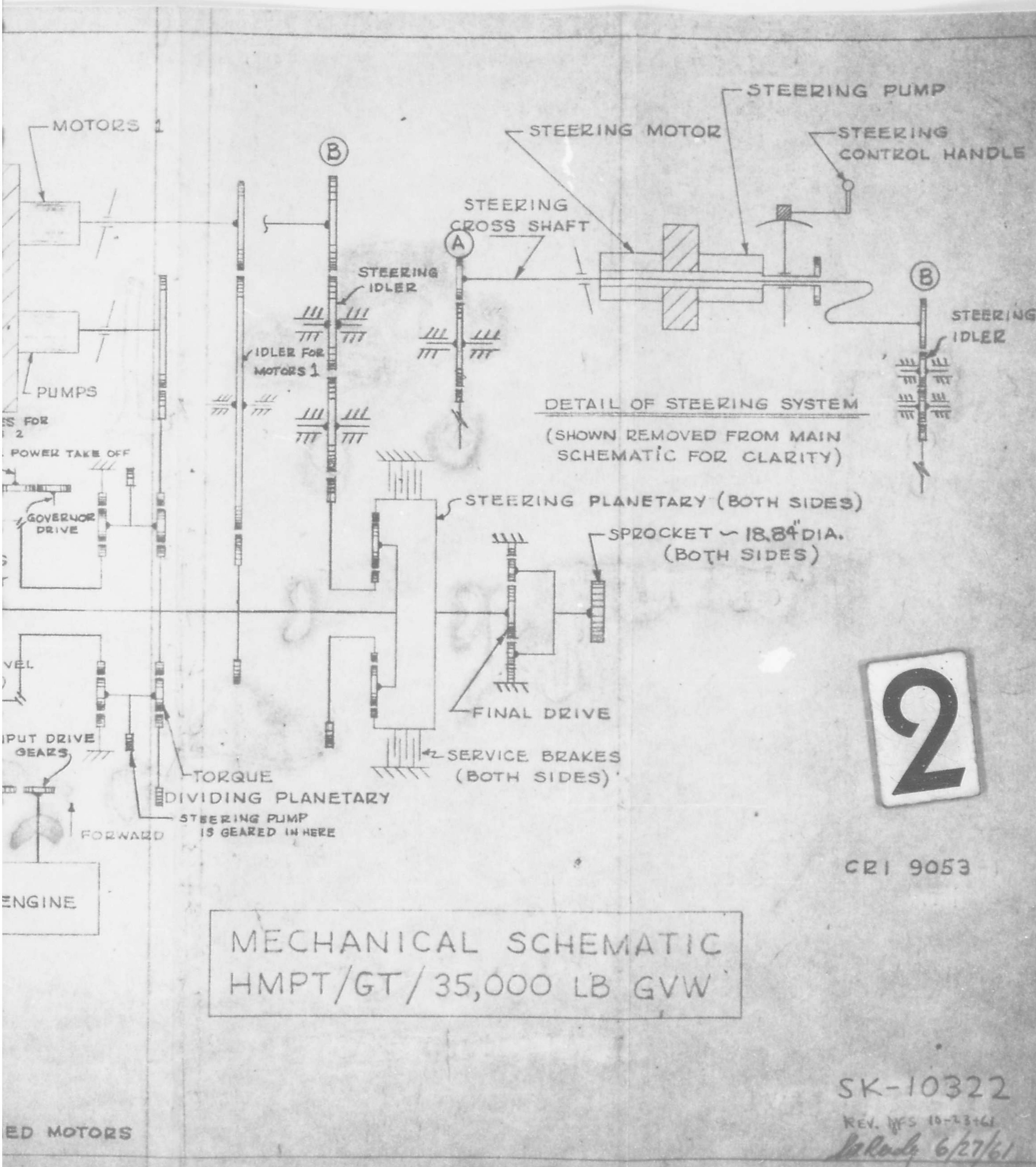


STEERING UNITS

26 MM (1.0236 IN.) PISTON DIA.
ONE (1) PUMP
ONE (1) MOTOR

HYDRAULIC UNITS

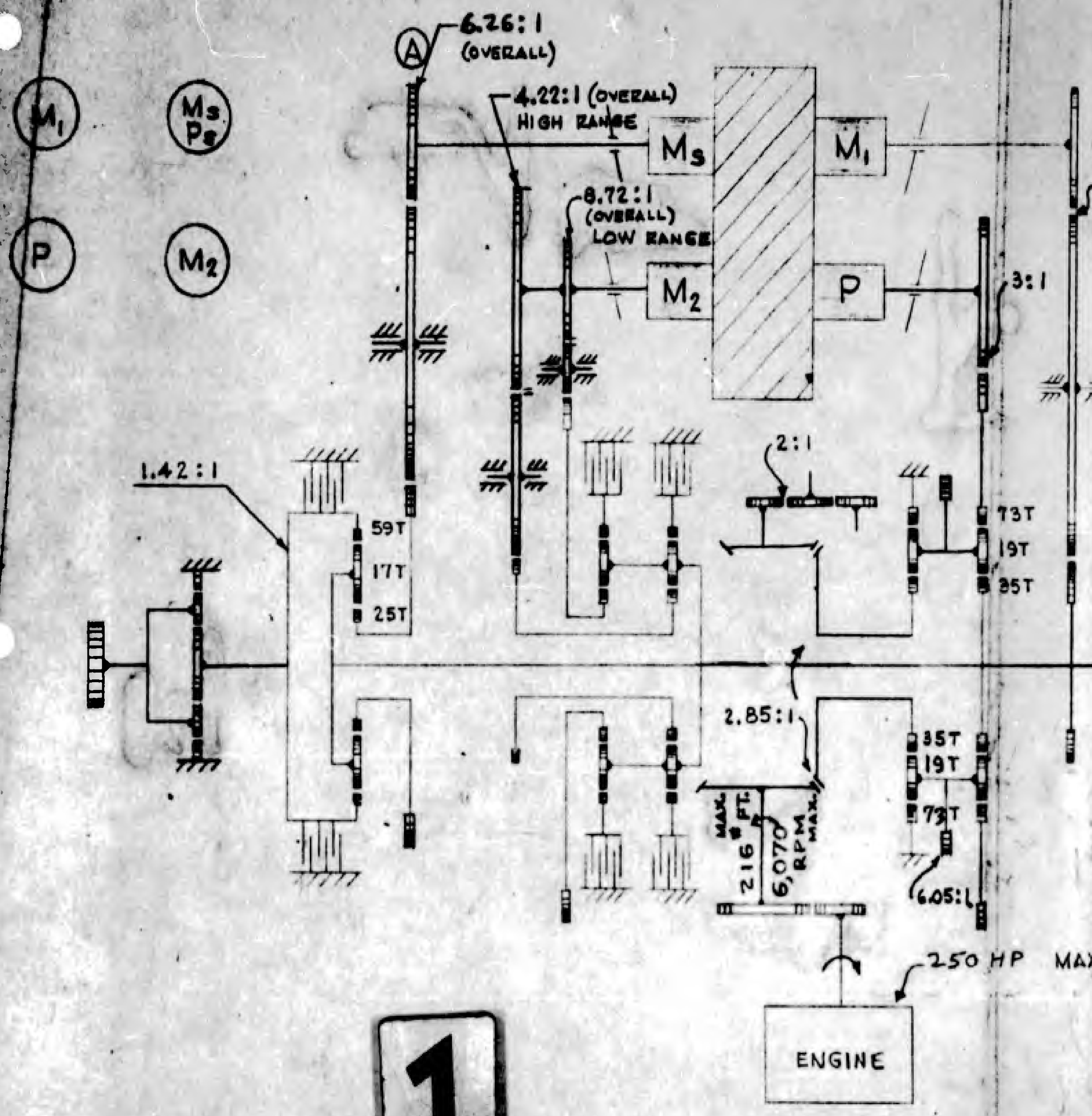
26 MM (1.0236 IN.) PISTON DIA.
TWO (2) PUMPS
TWO (2) GEARED MOTORS
TWO (2) CLUTCHED MOTORS
2 RANGE GEARING ON CLUTCHED MOTORS



CRI 9053

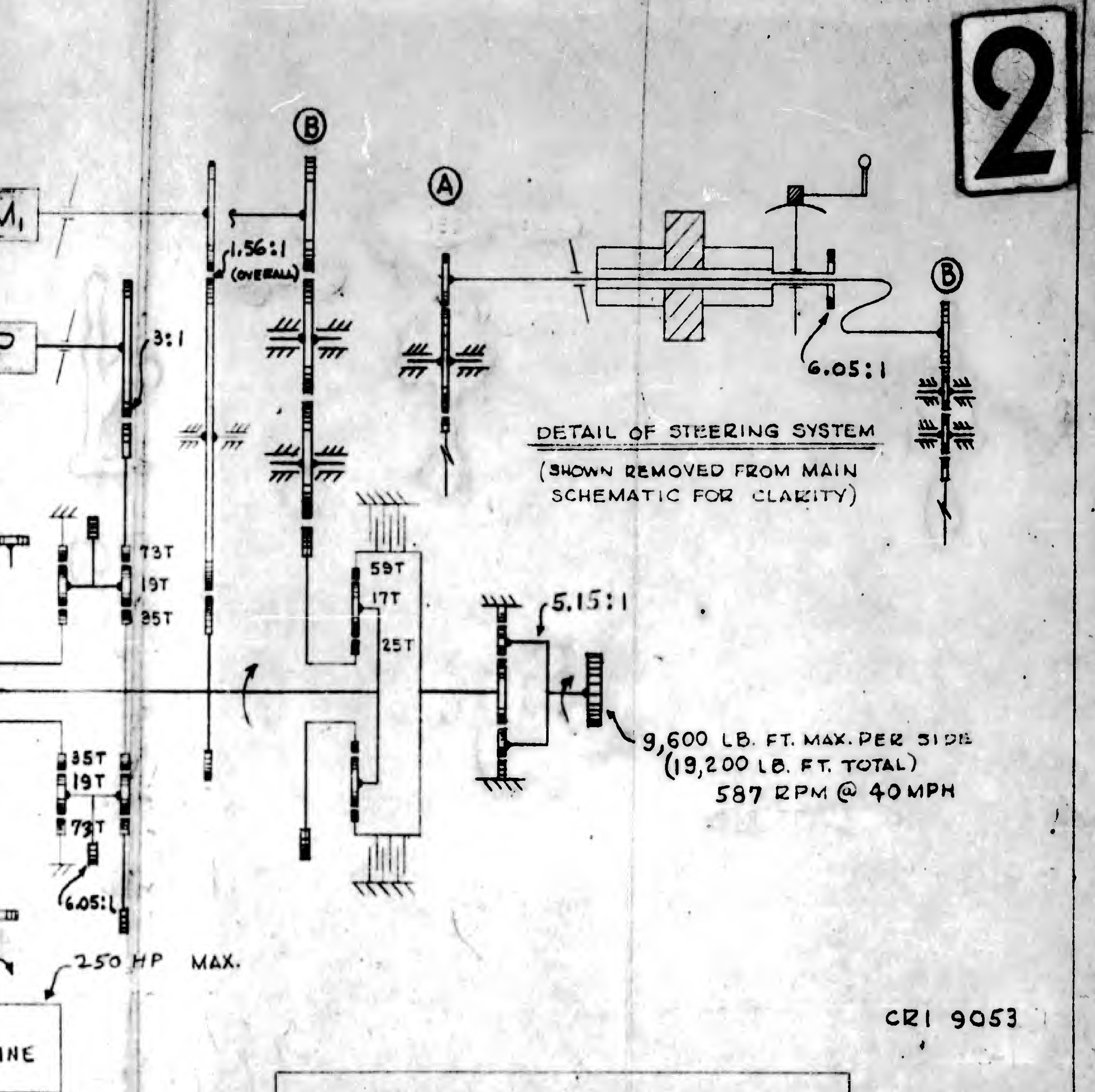
MECHANICAL SCHEMATIC
HMPT/GT/35,000 LB GVW

SK-10322
REV. WFS 10-23-61
6/27/61



6.26:1
(OVERALL)
A
M_1
Ms
Ps
P
M_2
4.22:1 (OVERALL)
HIGH RANGE
8.72:1
(OVERALL)
LOW RANGE
Ms
M_1
M_2
P
3:1
1.42:1
59T
17T
25T
2:1
73T
19T
35T
2.85:1
35T
19T
73T
MAX. 216 # FT.
6,070 RPM MAX.
6.05:1
250 HP MAX
ENGINE

1



CRI 9053

GEAR TRAIN SCHEMATIC
HMPT/GT / 35,000 LB GVW

SK-10325
REV. MFS 10-23-61

and the remaining four as motors geared (two units are clutched) to the output shaft; two steering planetary gear sets operated by a separately driven hydrostatic pump and motor system; and a control system. The hydrostatic elements are axial piston type units with variable displacement controlled by the angular movement of swash plates.

Specification - These data correspond to the mechanical design shown in the accompanying drawings and schematics. Modifications to fit a particular engine of different horsepower can be made easily by changing gear ratios and the torque multiplication of the transmission.

a. The transmission has a "T" configuration, i.e., two output shafts at right angles to the input.

b. The design input torque capacity is 216 lb.ft. at the bevel pinion gear. The transmission torque multiplication capabilities have been designed to give the tractive effort described in sub-paragraph i of this specification utilizing the engine torque occurring between 70% and

100% power.

c. Design input power is 250 HP.

d. Design input speed at design input torque is 6070 rpm.

e. Maximum length, flywheel mounting flange to rear of the housing is 24 inches.

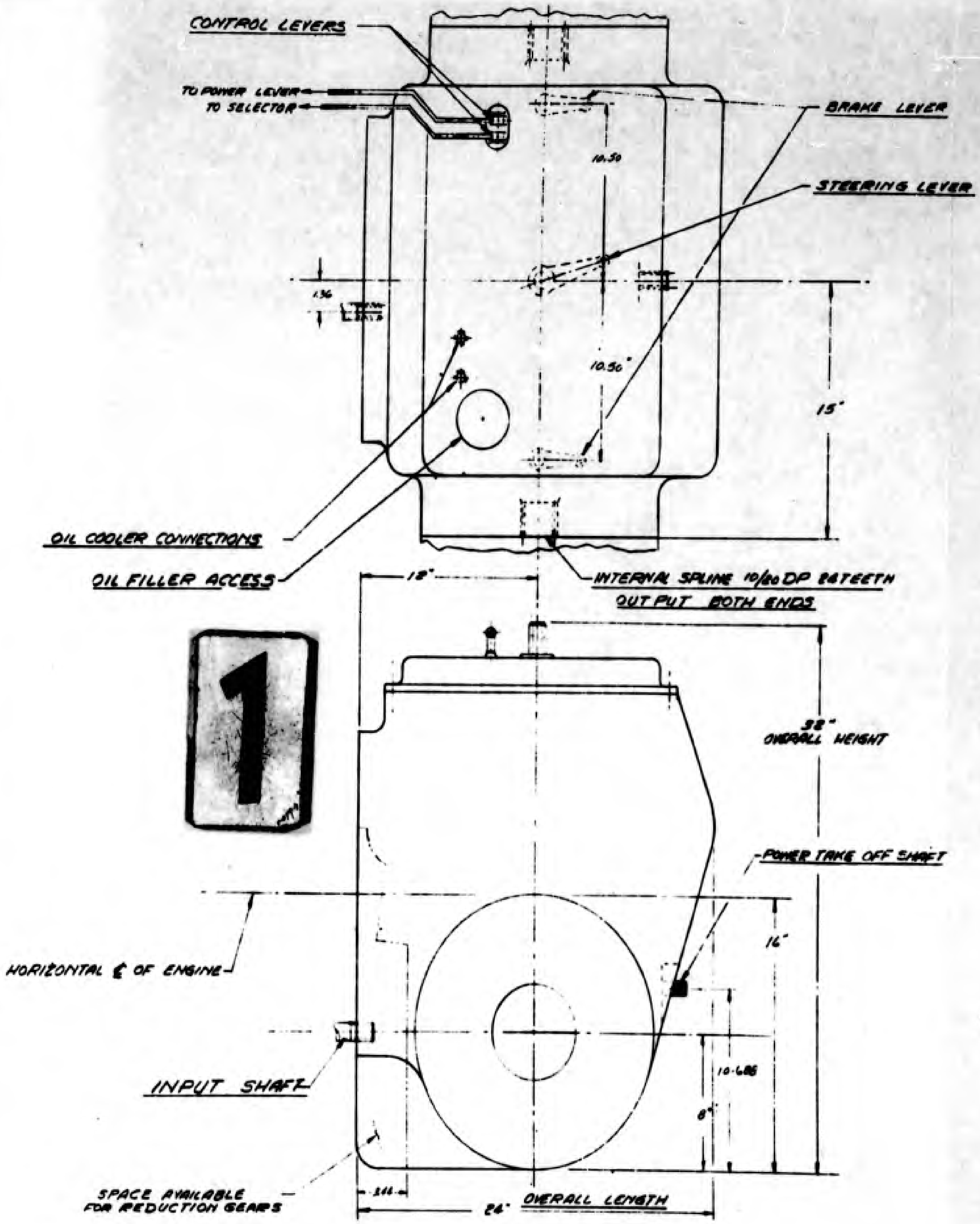
f. Maximum overall width between output flanges is 30 inches. Included within this dimension are the transmission steer system and braking system. Maximum width over final drives is 41.5 inches.

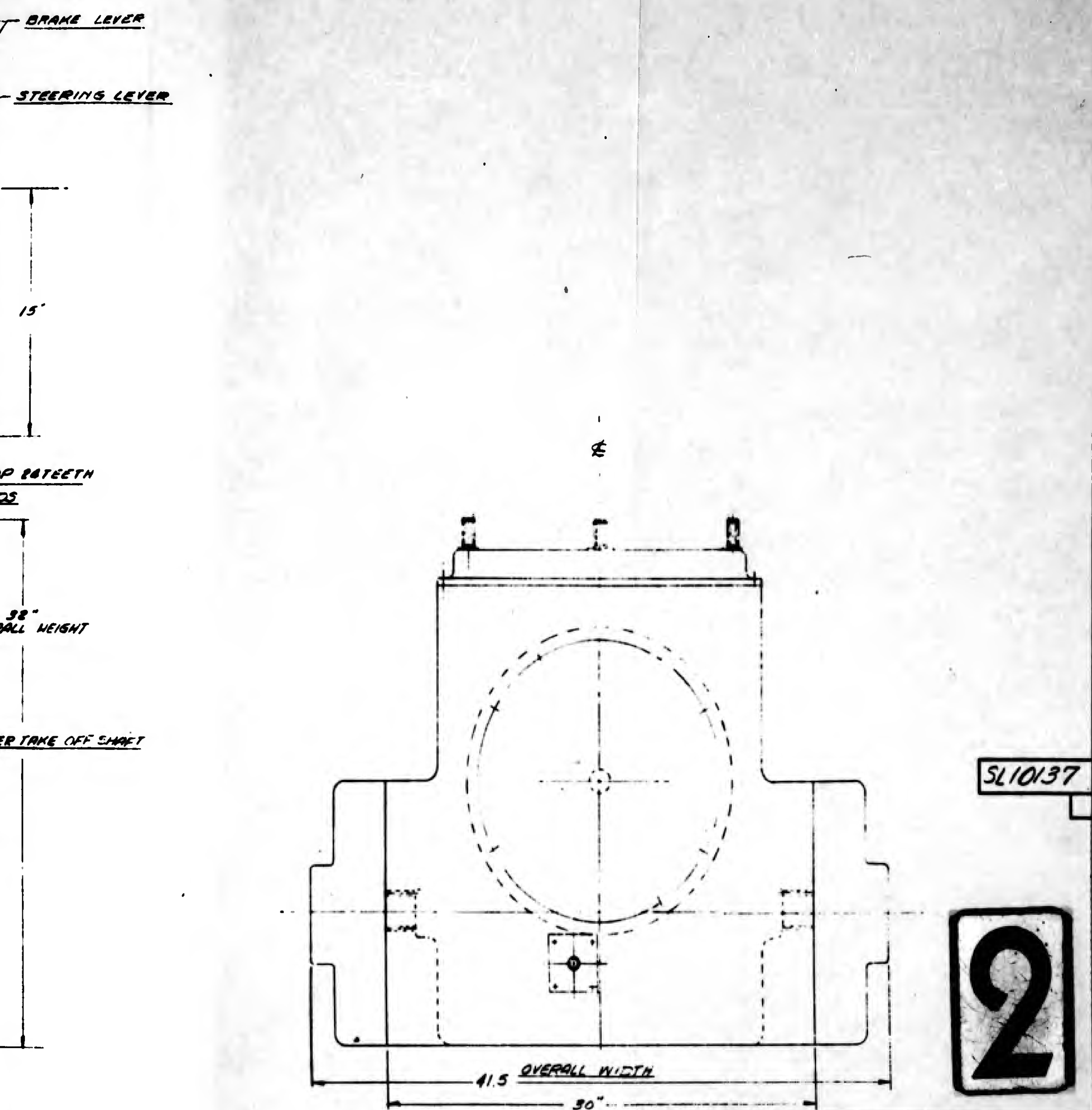
g. Maximum height is 32 inches.

h. Maximum radius of bottom rear of housing from the output shaft is 8 inches.

NOTE: All dimensions are shown on the Power Train Package Drawing SL 10137.

i. The transmission is capable of driving the vehicle 40 mph with 6070 rpm input and provides 22,000 lb. tractive effort up to about 3 mph.


CONTROL LEVERS
TO POWER LEVER
TO SELECTOR
BRAKE LEVER
10.50
STEERING LEVER
1.36
10.50"
15"
OIL COOLER CONNECTIONS
OIL FILLER ACCESS
12"
INTERNAL SPLINE 10/20 DP 24 TEETH
OUT PUT BOTH ENDS
1
32"
OVERALL HEIGHT
POWER TAKE OFF SHAFT
16"
HORIZONTAL ℄ OF ENGINE
INPUT SHAFT
10.686
8"
SPACE AVAILABLE
FOR REDUCTION GEARS
24"
OVERALL LENGTH



REV LTR	REVISION	AUTH DRAFT	CHK DATE

APPROVALS		
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CHECKED		
APPROVED	[illegible]	[illegible]
APPROVED	[illegible]	[illegible]
APPROVED		
APPROVED		
FIRST USE REFERENCE		

MATERIAL

STRATOS
A DIVISION OF FAIRCHILD ENGINE AND AIRPLANE CORP
BAY SHORE NEW YORK U.S.A

HMPT TRANSMISSION ENVELOPE

SCALE [illegible] | APPROX WT

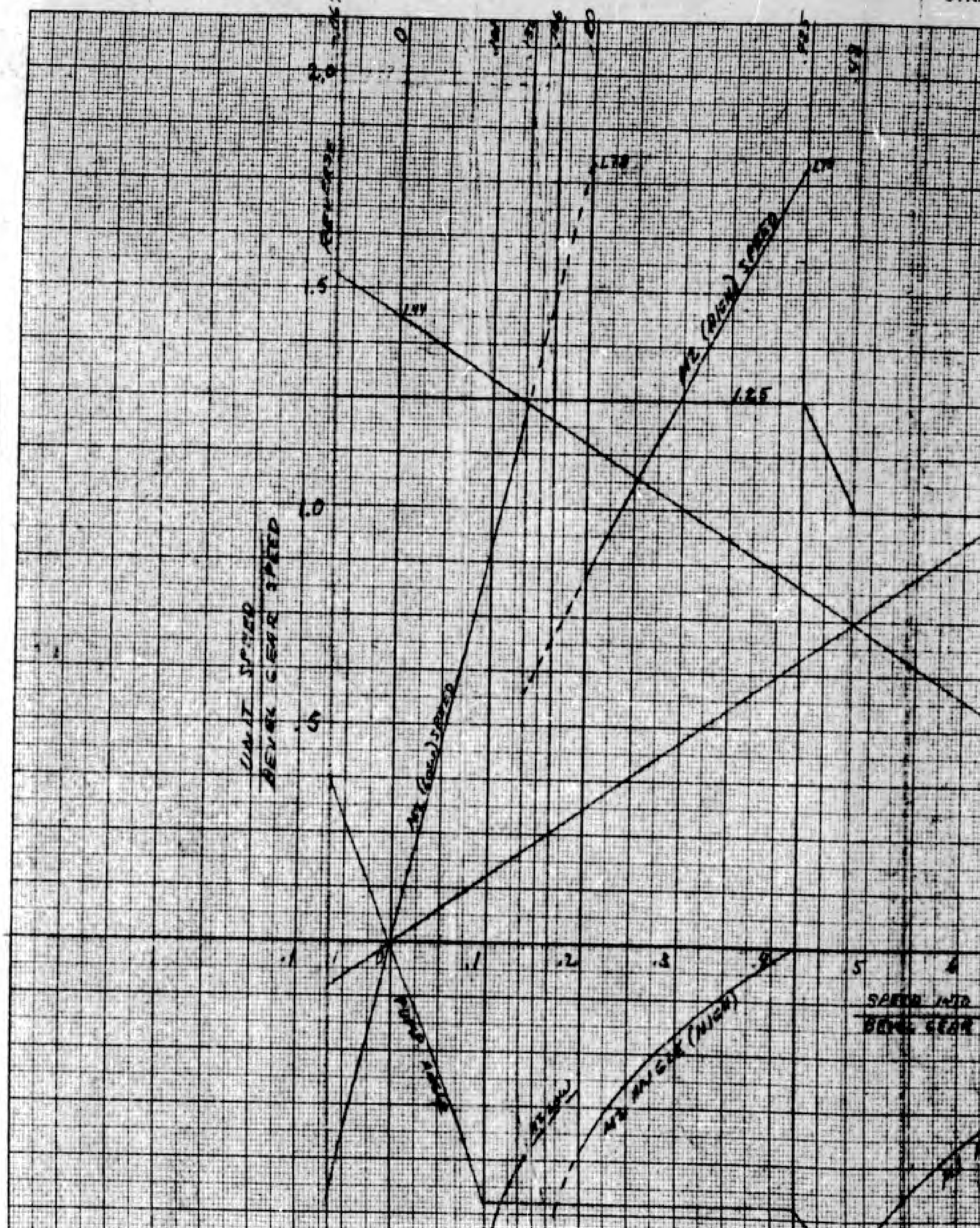
DRAWING NO. SL 10137

SHEET 1 OF 1 | SIZE B

DO NOT SCALE DRAWING — WORK TO DIMENSIONS SHOWN

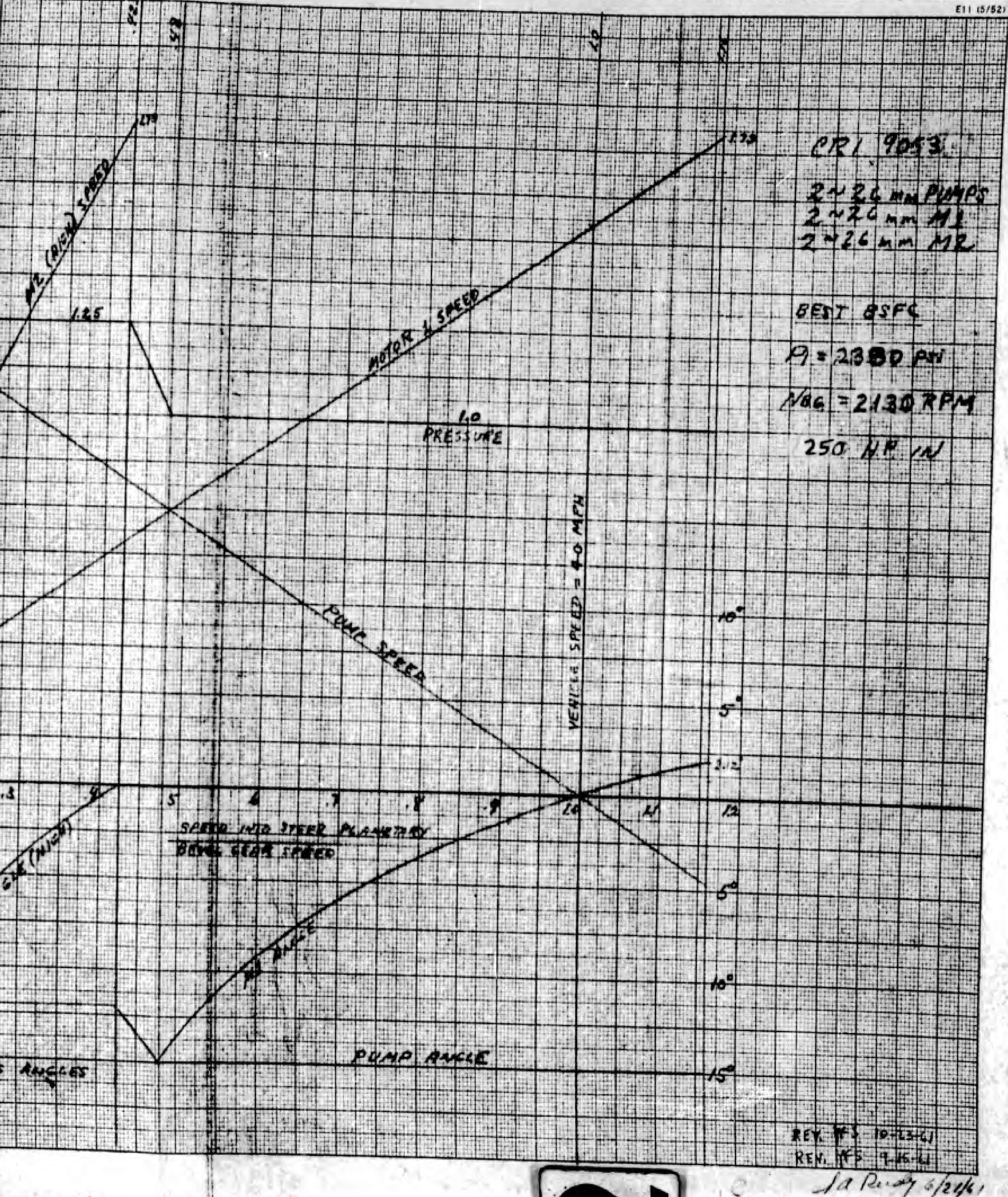
CRI 90434-1 E2
CSI 9953

The curves of pump and motor swash plate angles in SC 9017, Plot of Torque, Pressure, etc., represent theoretical values. The effect of inefficiencies will be accommodated as follows: The swash plate on motor 1 will be opened to a slight negative angle. This will provide 40 mph but at a torque less than the theoretical maximum torque available at the engine. At the other end of the ratio scale the pump angle will be reduced below the theoretical minimum angle. This will insure sufficient pressure to achieve the 22,000 lb. tractive effort, but it will occur at a speed less than the theoretical value of 3 mph. The degree of overdrive and underdrive is shown on SK 10381, Engine Operating Range. The efficiency at the maximum torque position (.101 speed ratio) at full power is 80% as shown in the Efficiency v/s Speed Ratio curve, SC 9009. This results in a speed of 3 mph,


REVERSE
1.78
1.44
1.25
M2 (HIGH) SPEED
M1 (LOW) SPEED
1.0
.5
UNIT SPEED
BEVEL GEAR SPEED
.1
.2
.3
.4
.5
SPEED INTO
BEVEL GEAR
M2 ANGLE (HIGH)
MOTOR ANGLES

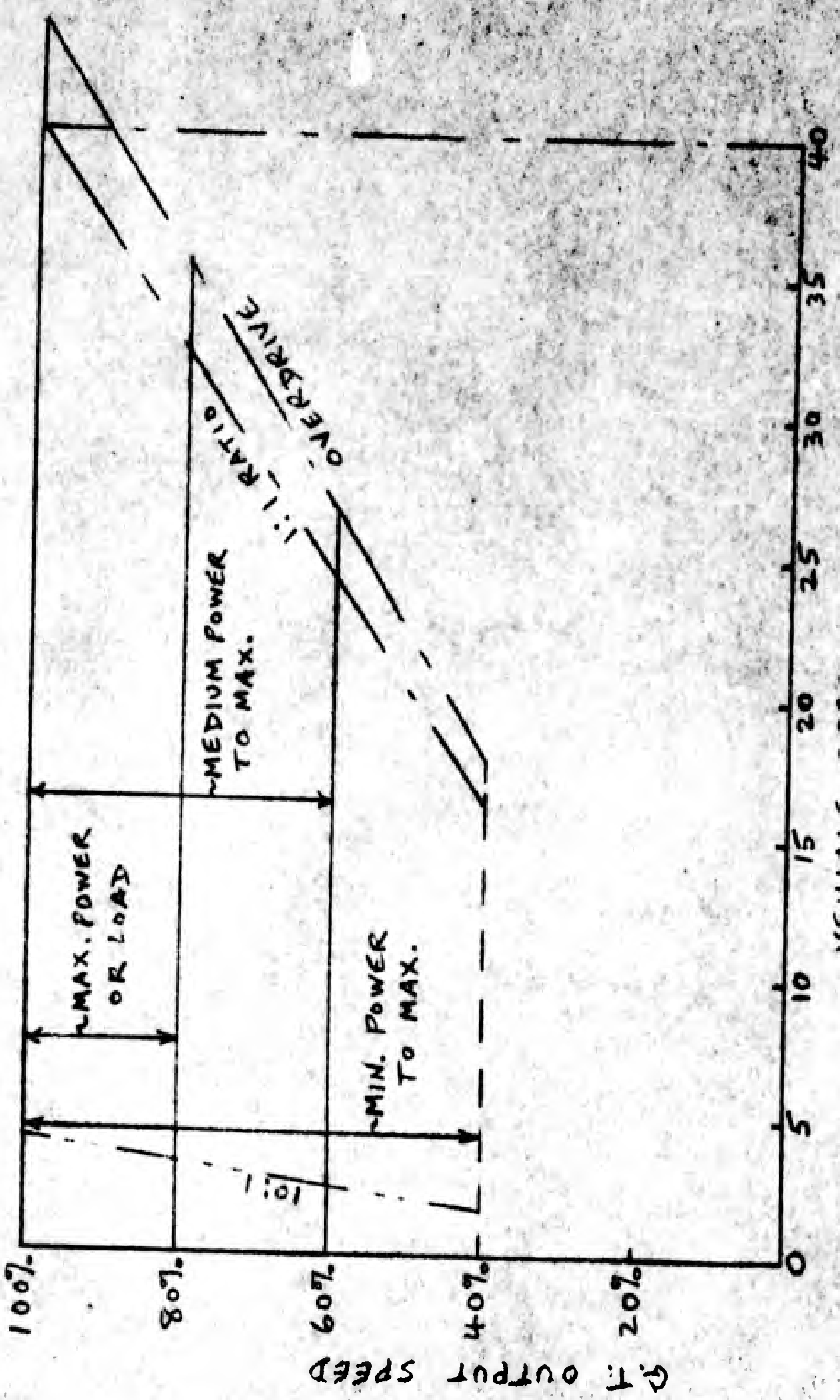
STRATOS DIVISION - FAIRCHILD ENGINE & AIRPLANE CORPORATION BAY SHORE, N. Y. CURVE NO. SC 9017

E11 (5/52)



2

SK-10331



STANDARD DESIGN SHEET - FORM # 162 (7/59)

SUBJECT: GAS TURBINE OPERATION RANGE

PROJECT: HMPT FOR LVTPX 11

ENG. ORD. SUPPL. ________ PROJECT TASK NO. ________

REFERENCE SL. ________

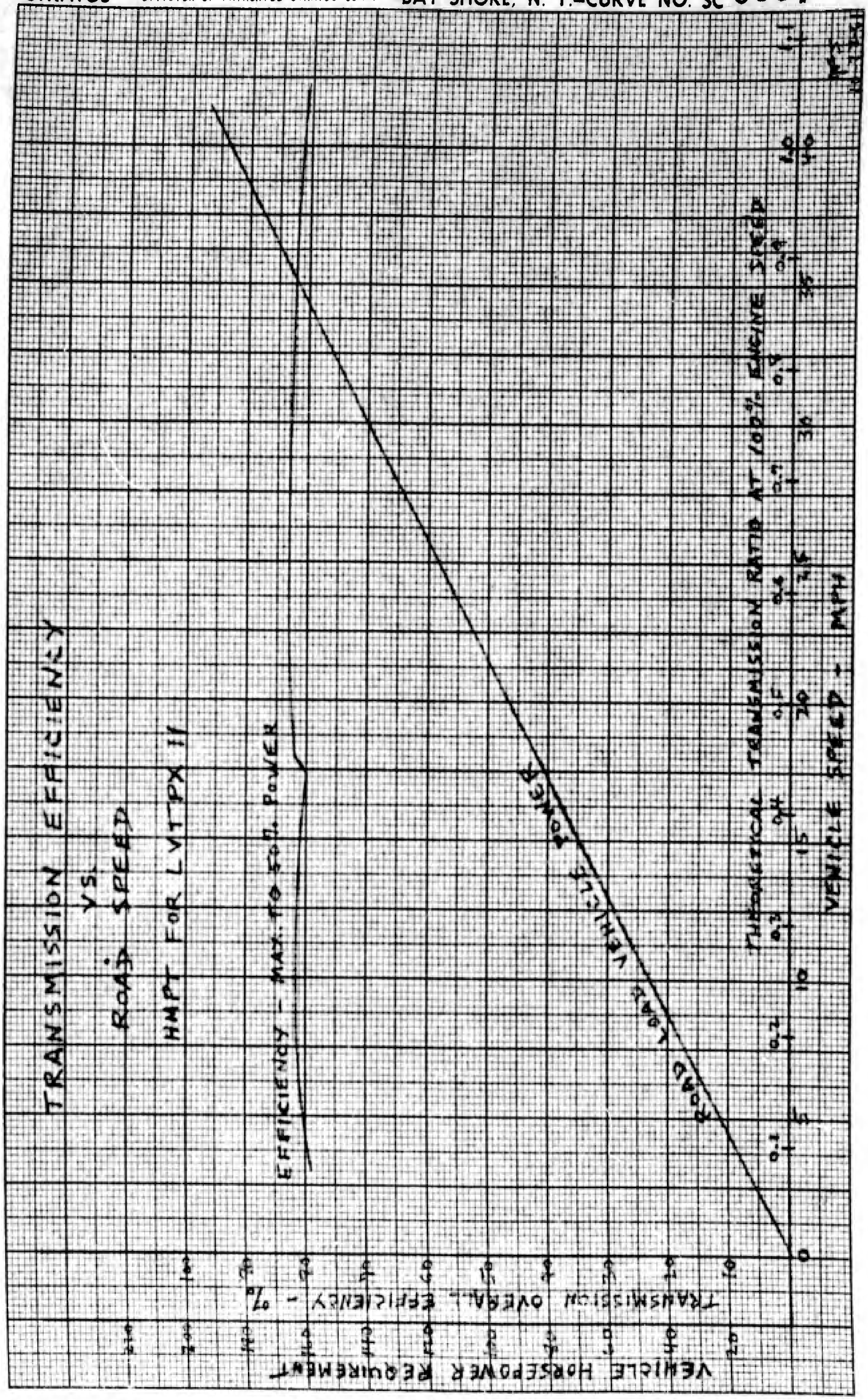
FOR: ________

BY: JFS

DATE: 10-23-61

PAGE ____ OF ____ PAGES

STRATOS A DIVISION OF FAIRCHILD STRATOS CORP. -BAY SHORE, N. Y.-CURVE NO. SC 9009



K&E 10 X 10 TO THE 1/2 INCH 359-11G KEUFFEL & ESSER CO.

when producing 22,000 lb. tractive effort.

As shown in the schematic diagrams SK 10322 and SK 10325, the sprocket diameter is specified at 18.84 inches diameter and the final drive as 5.15:1. The torque required, on the 18.84 inch sprocket, to give 22,000 lb. tractive effort is 9,600 lb. ft. per side, for a total of 19,200 lb. ft. The transmission is also capable of exerting 19,200 lb. ft. through a single sprocket.

j. The transmission brakes are capable of holding a 35,000 lb. vehicle on a 60% slope and are capable, without adjustment, of stopping a 35,000 lb. vehicle 25 times from 40 mph, at 2 minute intervals at an average deceleration rate of 12 ft/sec.^{2}. Since the brakes are self-adjusting, additional stops can be made without mechanical adjustment.

k. Transmission Weight (Net, dry):

809 lbs. (less final drives)

1,015 lbs. (including final drives)

Operating Principles - The Stratos HMPT transmission differs from other purely hydrostatic transmissions by the provision of a drive system which divides the transmitted power between mechanical and hydraulic paths. The power unit provides the input to a planet carrier of a gear set called the ratio planetary. The sun gear of the ratio planetary forms the primary transmission output shaft which connects to the vehicle driving sprockets via a steering planetary gear set. The ring gear of the ratio planetary drives the hydrostatic pump elements and by control of the pump and motor speed relationship, the ratio planetary set is made to operate as a differential gear. The primary output shaft is also driven directly through gearing by the hydrostatic motors of the system. (See SK 10322)

Forward Mode - In automatic drive mode at zero forward speed of the vehicle the ratio planetary set drives the hydrostatic pumps with the transmission primary output shaft held stationary by the vehicle. At low engine speeds when insufficient power is available to overcome load resistance, the transmission down shifts until the pump swash plate is at zero angle. This is the normal idle position. When increased engine power is demanded by the operator, the actuator shift valve responds to a rise in the input speed signal and upshifts the actuator and cam plates and the pump output pressure rises. The hydrostatic motors, which are coupled to the output shaft, then supply additional torque to overcome vehicle load resistance and the vehicle commences acceleration.

As the vehicle gathers speed the sprocket torque available is reduced for any constant horsepower input, and the control system acts automatically to vary swash plate angles to maintain engine loading for optimum torque and economy as the speed ratio changes. As the clutching motors reach their designed maximum speed they are shifted automatically from low to high range and as the speed further increases they are clutched out of the drive system by the automatic control inherent in the basic design. The declutching occurs under no-load conditions of the motors and is synchronized to occur at zero displacement. At a certain vehicle speed dependent on percentage of maximum engine speed attained, the remaining drive motor swash plate angles approach zero with the pump swash plates at maximum angle, thereby preventing circulation of the working fluid and hydraulically locking the ratio planetary ring gear. At this condition a 1:1 drive speed ratio exists within the transmission and the drive to the output sprockets is purely mechanical. A small degree of overdrive is achieved by giving negative angles to the swash plates of the output shaft motors, which therefore act as pumps to drive the main pump (which now runs as a motor) and overdrive the ring gear, enabling maximum vehicle speed to be reached.

<u>Reverse Operation</u> - Reverse operation is provided by moving the driving pump swash plates to negative angles. This causes

reversal of the high and low working pressures within the system which, in turn, causes the motors to reverse direction thereby driving the vehicle in reverse. Automatic engagement of the reverse drive is achieved in the same manner as forward drive, with the exception that only a fixed low ratio of 14.9:1 is provided. Other alternative reverse ratios can be provided if requested, with only minor changes to the transmission control mechanism.

Steering System - A steering system is provided in this transmission as follows: an additional planetary gear train is installed on each primary output shaft between the input planetary and the final drive gear train. This is shown on the schematic drawings SK 10322 and SK 10325 and is referred to as a Steering Planetary. The input to each steering planetary is the planet carrier which is driven by the primary output shaft. The output shaft of each steering planetary is attached to the ring gear. The sun gear is driven directly by the steering cross shaft in one planetary and through an idler gear in the opposite side.

A hydrostatic motor is geared to this steering cross shaft. Its swash plate is fixed at a full angle of 15 degrees. A pump geared to the planet carrier of the ratio planetary is coupled hydraulically to the steering motor.

In the straight running condition the equal but opposite reaction torques on the steering planetary sun gears are taken by the steering cross shaft, and the sun gears are stationary. The steering pump has its movable swash plate set at zero angle. The pump, although it is constantly rotating, is not pumping or building pressure, so the steering motor remains stationary.

In order to steer, the driver's steering control is operated in the desired direction, moving the pump swash plate to either a positive or negative angle by mechanical means. This causes pressure to build up and the steering motor then rotates. The motor turns the steering cross shaft which, in turn, drives the sun gears of the two steering planetary gear trains in opposite directions. The steering planetary gear trains now have an additional input, i.e. the sun gears which are driven at equal speeds in opposite directions. This causes one track to run faster and the other slower.

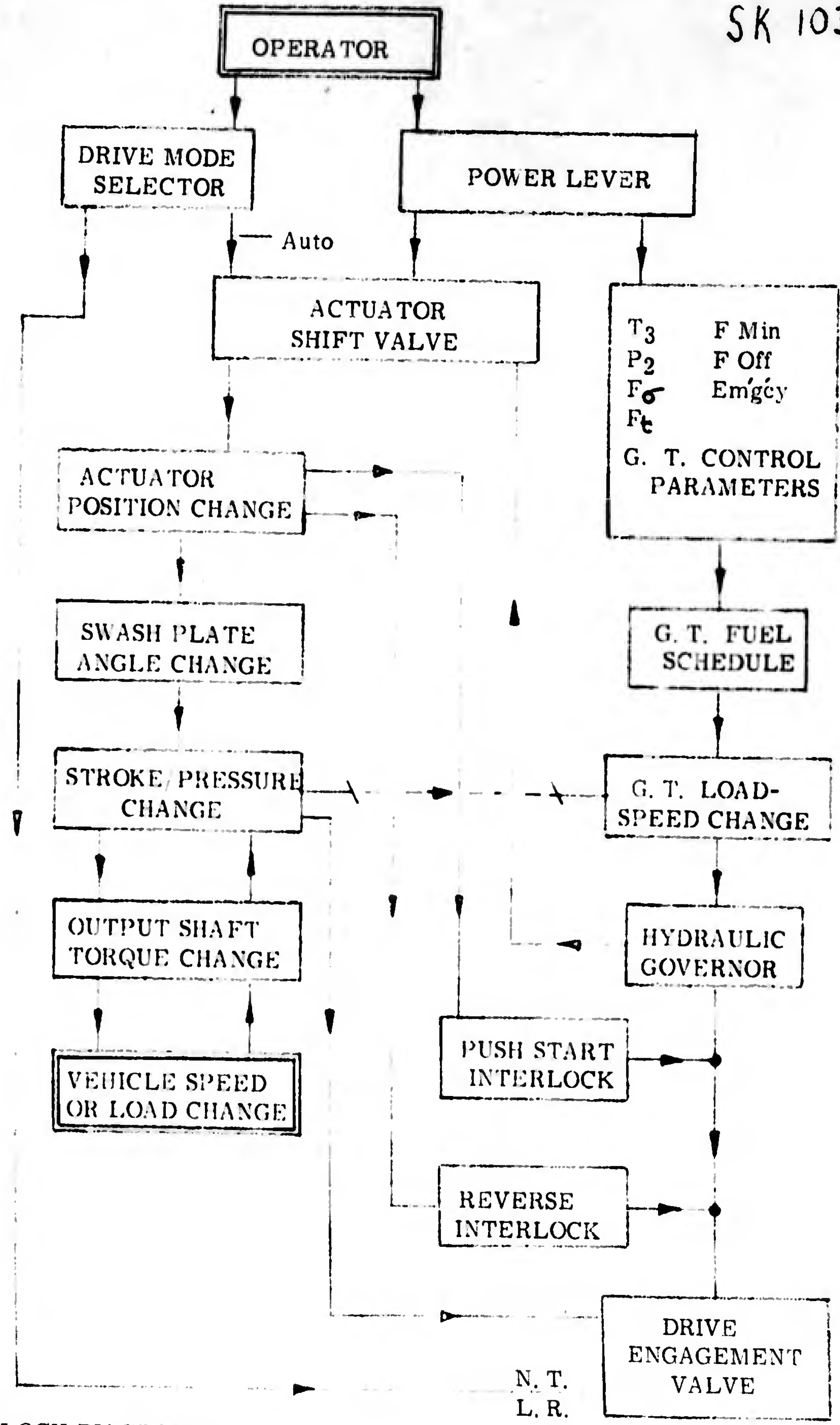
The differential speed setting is positive, regenerative, and infinitely variable within the limits of the design. In a pivot turn, track force requirements dictate a positive and negative torque value to each of the output ring gears. These values being equal and opposite would impose reaction torques on the planet carriers of equal values but opposite direction. The carriers are attached by a continuous cross shaft and therefore will provide this reaction. The sun gear torque value requirement is also equal for each side and opposite in direction. The sun gear torque is provided by the steering motor which is splined to the steering cross shaft. The steering cross shaft is geared to the sun gears directly on one side and through an idler on the other. Thus the sun gear torque for each side is equal but opposite in direction.

Speed and Ratio Control - (See Control System Block Diagram SK 10327 and Schematic SK 10332). The control system provides automatic, infinitely variable speed and torque ratio changing for optimum utilization of the available engine power in the matching of vehicle load resistance. The control system contains the following elements: an input speed governor, an actuator flow director valve, and a hydraulic actuator linked to the hydrostatic pump and motor swash plates. Other manual and servo-operated spool valves are used for drive mode selection and interlocking functions. A pressure signal provided from the drive input speed governor of the control system displaces an actuator shaft control valve to permit upshifting or downshifting of transmission ratios with increasing and decreasing engine speed, respectively. A connection is made between the gas turbine fuel control lever and the actuator shift valve biasing spring, to relate ratio-shifting to the required power setting.

Since for every engine speed and power setting there is a fuel

STRATOS DIVISION
BAY SHORE, L. I., N. Y.

SK 10327



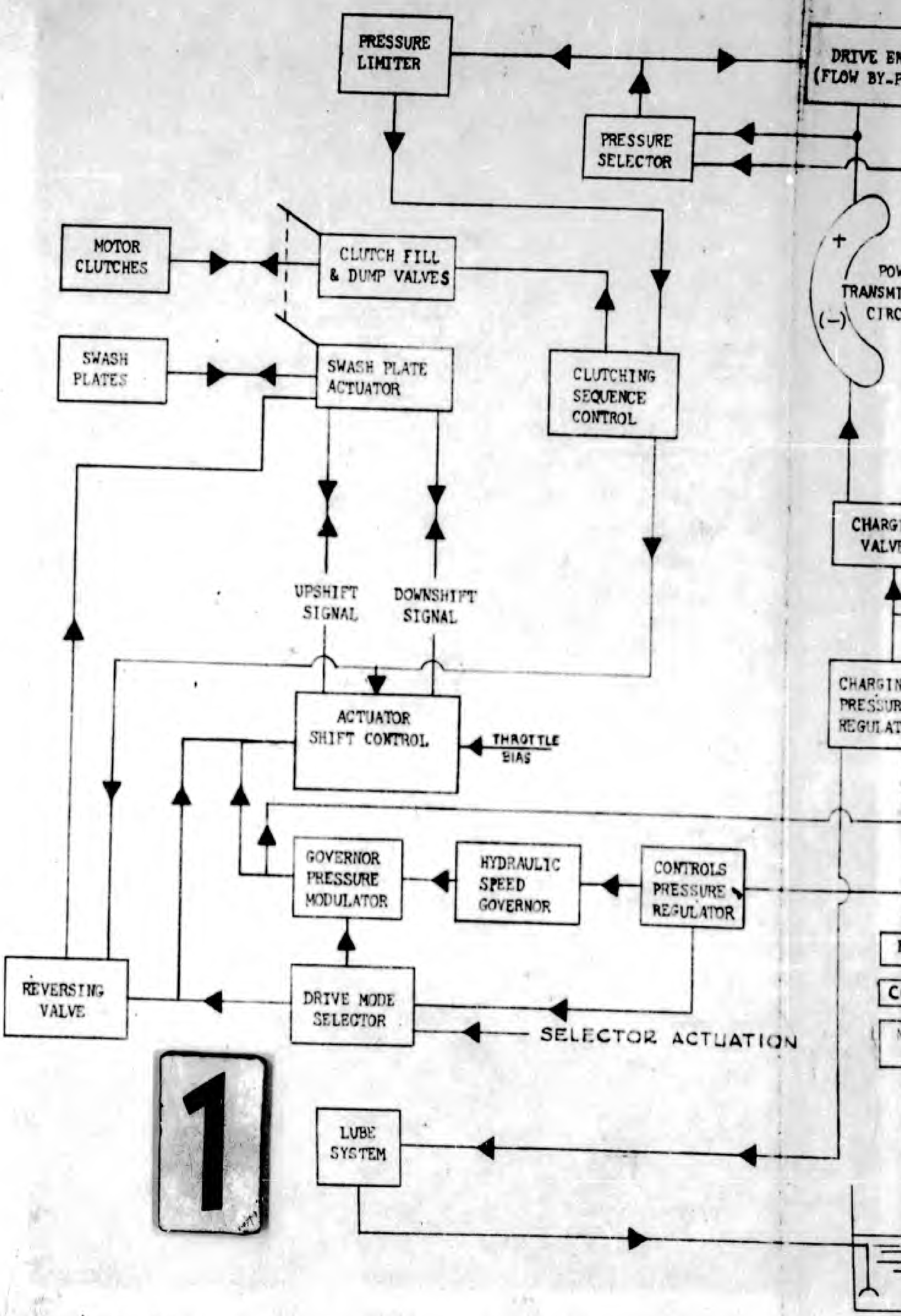
BLOCK DIAGRAM
HYDROMECHANICAL
POWER TRAIN CONTROLS

GAS TURBINE WITH HMPT

K.P

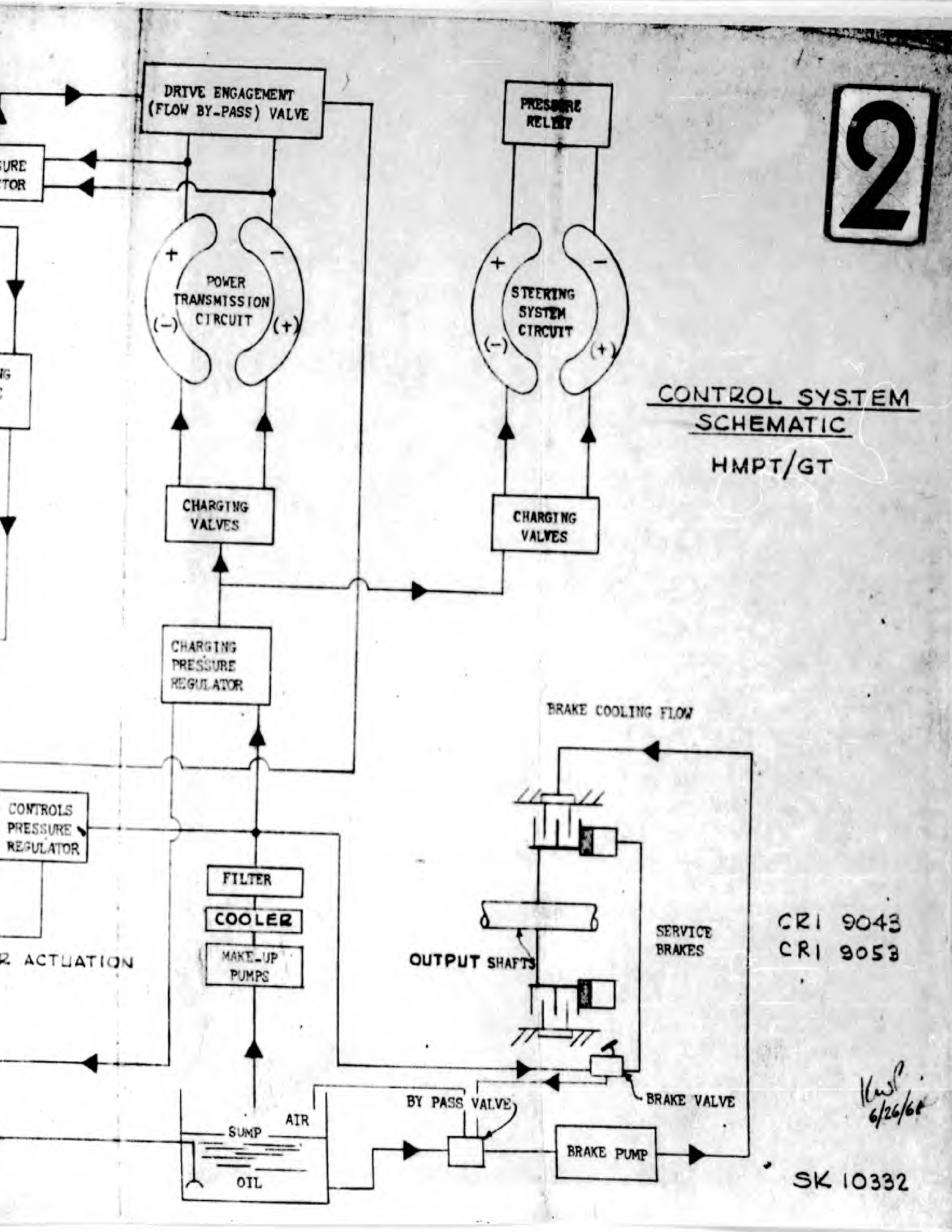
CR1 9053

6/26/61 E-4 (1-53)



PRESSURE LIMITER
DRIVE EN
(FLOW BY-P
PRESSURE SELECTOR
MOTOR CLUTCHES
CLUTCH FILL & DUMP VALVES
SWASH PLATES
SWASH PLATE ACTUATOR
CLUTCHING SEQUENCE CONTROL
+
(−)
POW
TRANSMI
CIRC
CHARG
VALVE
UPSHIFT SIGNAL
DOWNSHIFT SIGNAL
CHARGIN
PRESSUR
REGULAT
ACTUATOR SHIFT CONTROL
THROTTLE BIAS
GOVERNOR PRESSURE MODULATOR
HYDRAULIC SPEED GOVERNOR
CONTROLS PRESSURE REGULATOR
REVERSING VALVE
DRIVE MODE SELECTOR
SELECTOR ACTUATION
LUBE SYSTEM

1

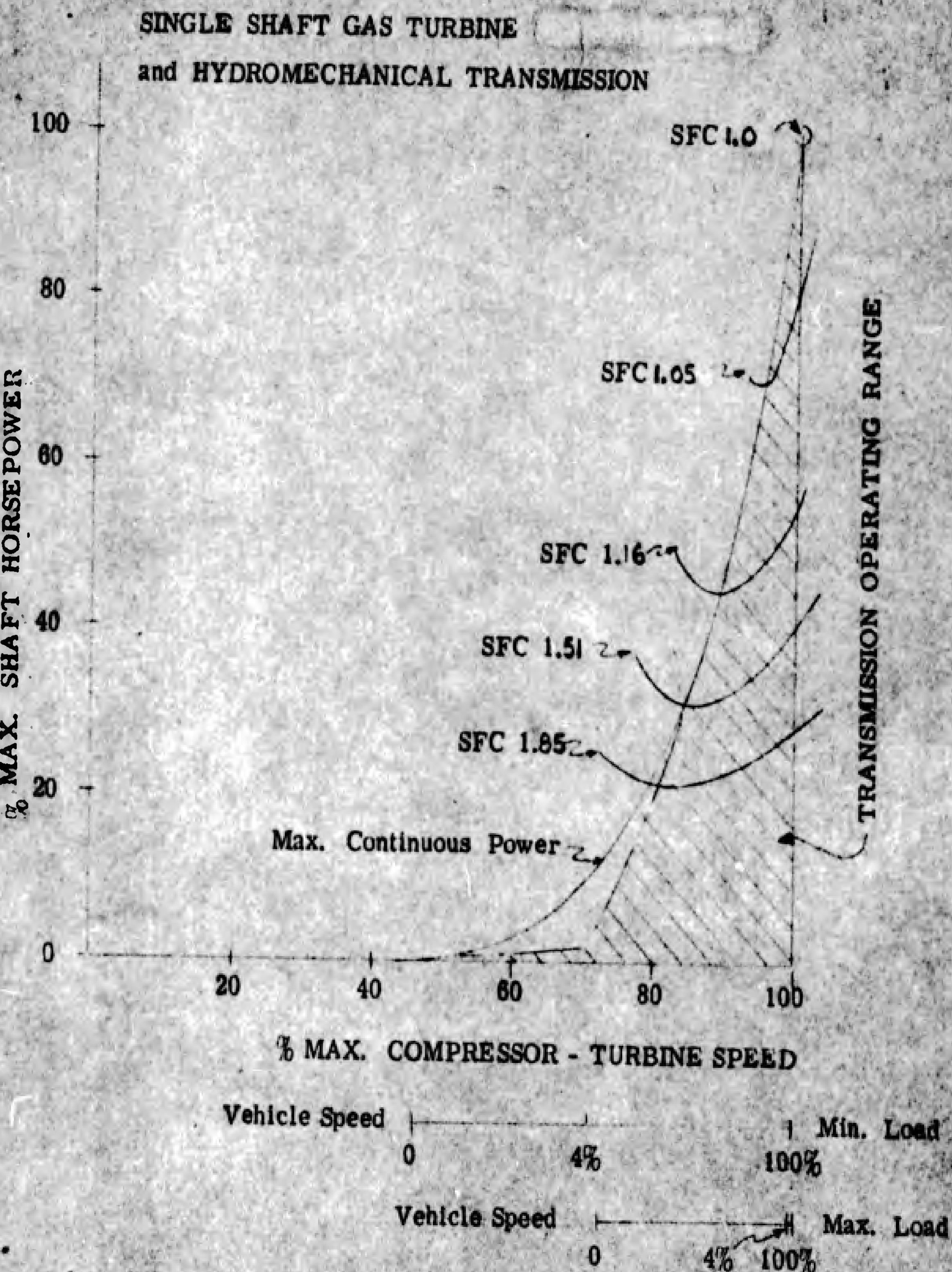

DRIVE ENGAGEMENT (FLOW BY-PASS) VALVE
PRESSURE RELIEF
2
SURE
TOR
POWER TRANSMISSION CIRCUIT
STEERING SYSTEM CIRCUIT
CONTROL SYSTEM SCHEMATIC
HMPT/GT
CHARGING VALVES
CHARGING VALVES
CHARGING PRESSURE REGULATOR
BRAKE COOLING FLOW
CONTROLS PRESSURE REGULATOR
FILTER
COOLER
MAKE-UP PUMPS
R ACTUATION
OUTPUT SHAFTS
SERVICE BRAKES
CRI 9043
CRI 9053
BRAKE VALVE
BY PASS VALVE
AIR
SUMP
OIL
BRAKE PUMP
6/26/68
SK 10332

pump control lever position corresponding to fuel scheduling for best economy, controlling the transmission governor as a function of fuel control lever position permits the transmission to select the precise ratios to maintain the desired engine power and/or speed level over a wide range of vehicle speeds. (See SC 9018).

The transmission is matched to the engine by the design of the shift valve biasing cam, which matches the desired characteristics. Ratio shifting will commence immediately to increase or decrease engine loading (see Figure SK 10331). The transmission will initially downshift to relieve engine loading and allow increased input speed; thereafter the transmission will continue to change ratio or output speed until the newly established equilibrium point is reached. Overspeed limit is obtained by control of ratio with respect to vehicle speed.

Unloading of the drive is automatically achieved by the partial opening of the engaging valve with consequent bleed-off of

SC-9018



SHAFT HORSEPOWER AND RELATIVE FUEL CONSUMPTION
versus
COMPRESSOR - TURBINE SPEED

CRI 9043
A-1
CRI 905

pressure when the torque requirement is greater than the transmission design capacity.

Considerable operating experience, which has borne out the correctness of the concept, has been amassed in field testing with transmissions based on the above principles of construction and control.

Braking - The service and parking brakes are combined in a single brake system consisting of a multiple disc assembly mounted on each of the two output shafts, as shown in the Schematics SK 10322 and SK 10325. The disc packs can be compressed, to set the brakes, by two means. For service braking during normal running they will be compressed simultaneously by a piston actuated by pressurized oil from the make-up pump. For parking, and for emergency braking in case of pump failure, they will be compressed mechanically. This will be accomplished by rotating rings simultaneously, on each brake, on the sides of the disc packs opposite the hydraulic pistons. These rings have inclined-plane shaped slots containing steel balls cut into them. Adjacent to each inclined plane on the rings is a similar type of inclined plane on the housing. Opposite rotation of the rings will cause the inclined planes to ride up on the balls, separating the rings from the housing and compressing the disc packs. The link between the two disc packs, provided for mechanical operation, is outside the transmission housing. This permits the vehicle designer to include a linkage which allows the brakes to be individually set instead of simultaneously set. This type of linkage allows the vehicle, with a dead power plant, to be steered while under tow by cable. The brakes have been sized to meet the most extreme torque condition, the requirement of holding the vehicle on a 60% slope. The design of the discs will include oil grooves for lubrication during dynamic braking. This flow will be sufficient to maintain acceptable temperatures on the rubbing surfaces, and in the core of each disc. The brake oil cooling circuit is shown on the Controls Schematic SK 10332. The degree of engine braking which can be achieved by overriding the throttle-to-governor speed relationship is set by the power absorption capacity of the gas turbine impellers at maximum motored speed. A braking capability is also provided within the transmission. This consists of multi-disc, hydraulically operated service brakes furnished on each steering planetary output shaft, energized by a driver's brake pedal. These brakes are designed for capacity capable of twenty-five brake stops from 40 MPH at 2 minute intervals at an average deceleration rate of 12 ft/sec^2, with no brake adjustment. The self-adjusting nature of the brakes provided in the Stratos transmission, together with a forced flow oil cooling system, will provide additional brake stops, but without the need for any further adjustment. Emergency manual braking by override, and parking brake function are also included in this braking system.

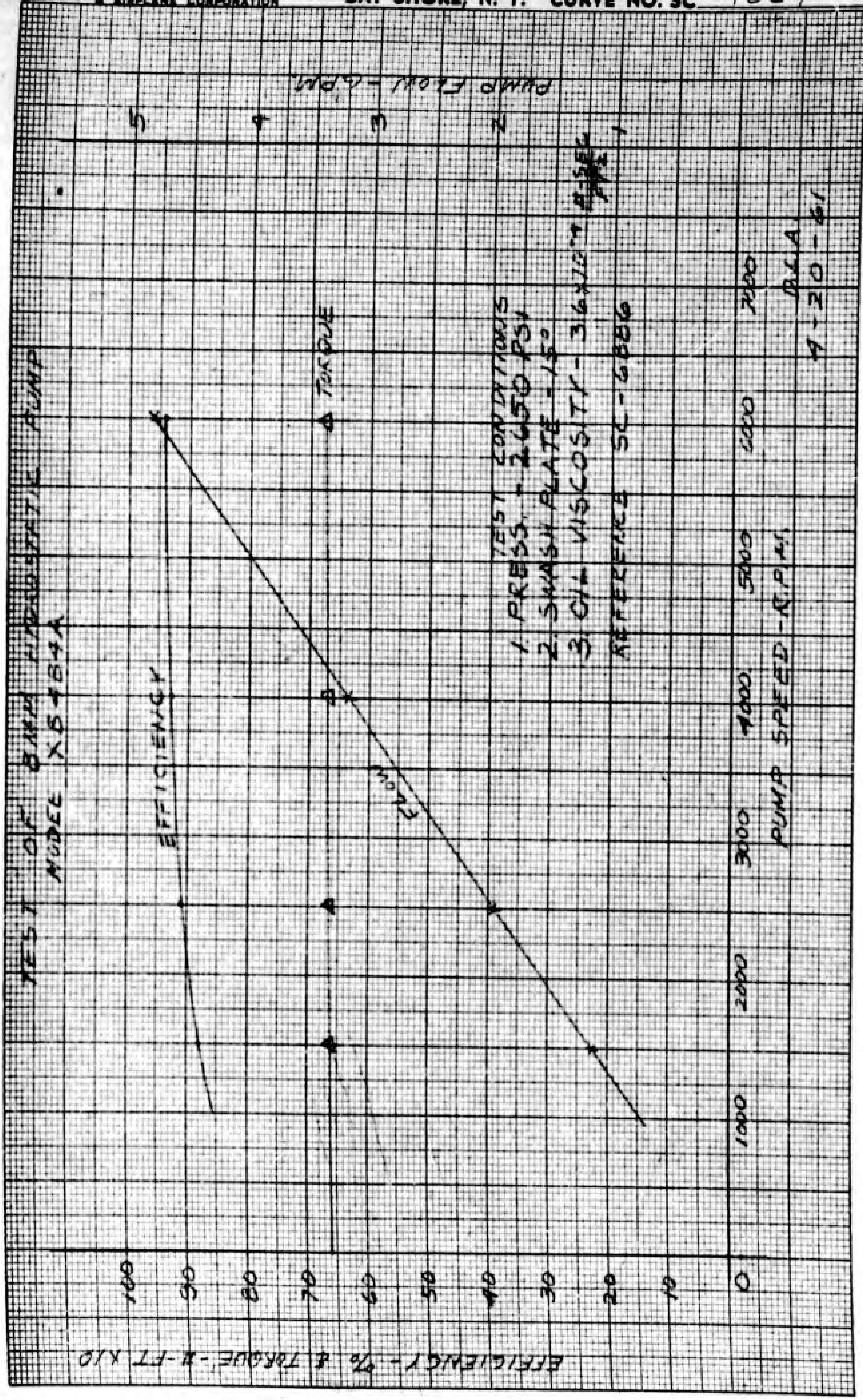
<u>Push Start</u> - The vehicle power plant is capable of being push-started. When the operator selects push-start mode of operation the transmission will start to upshift to the overdrive position, as the vehicle speed builds up. The vehicle may be towed at speeds up to 25 mph in neutral.

SECTION III

REQUIREMENTS

Performance - The system performance analysis of this transmission was made using standard system analysis techniques. The results of actual pump tests were used to evaluate the slip, drag, and friction coefficients of the basic hydraulic element. The performance and test data appear on curve SC 9007. The basic assumption is that the HMPT hydraulic system will operate in a manner similar to the test pump, i.e., the conditions of geometric, dynamic, kinematic, and flow similarity will be satisfied over most of the operating range. Corrections for off-design performance are made, based on actual test results of a similar transmission operating under similar off-design conditions. For design purposes, slip, drag, and friction coefficients were taken as constant throughout the range of operation. Although in an actual transmission this is not entirely true since the slip coefficient at constant load varies as the 1.5 power of the

STRATOS DIVISION · FAIRCHILD ENGINE & AIRPLANE CORPORATION BAY SHORE, N. Y. CURVE NO. SC 9007



speed, the use of a relatively high constant over the entire operating range makes all performance calculations conservative.

The dynamic flow losses were assumed to be negligible. This is a valid assumption because actual test data proves that the efficiency curve rises with increased speed at a constant torque.

The system equations, relating efficiency to theoretical ratio, were derived entirely from actual test data for hydraulic and mechanical efficiencies of the pump and the HMT 400, (a currently operating truck transmission), respectively. The derived system equations give efficiency as a function of ratio. These equations relate pressure, speed, load, and performance to the overall ratio and hence to the efficiency of the system, and are used to calculate maximum performance and economy operation when matched to a specific vehicle and engine. These calculations take account of actual torque and speed ratios when these differ from the theoretical ratios, and are programmed on a digital computer because of the complexity of the problem and because of the non-linearity of the performance equation.

The overall efficiency calculated for the proposed steering transmission (hereafter referred to as the HMPT) was compared with test efficiencies of the HMT 400 transmission which was tested under Contract DA-30-069-ORD-2340. The HMT 400 is a similar transmission (hydromechanical, power dividing). The actual dynamometer test curves for the HMT 400 shown in the accompanying graph, SC 9010, were used to establish the straight through mechanical efficiency. This efficiency was used to calculate the mechanical efficiency of the HMPT transmission for a tracked vehicle, which has a lower overall mechanical efficiency. The mechanical efficiency for the HMT 400, taken from the test, is 92%. The mechanical efficiency for the HMPT, taking into account the differences already mentioned, was calculated as 90.5%.

The parasitic power losses, or power losses that are independent of input power and operating conditions, were calculated

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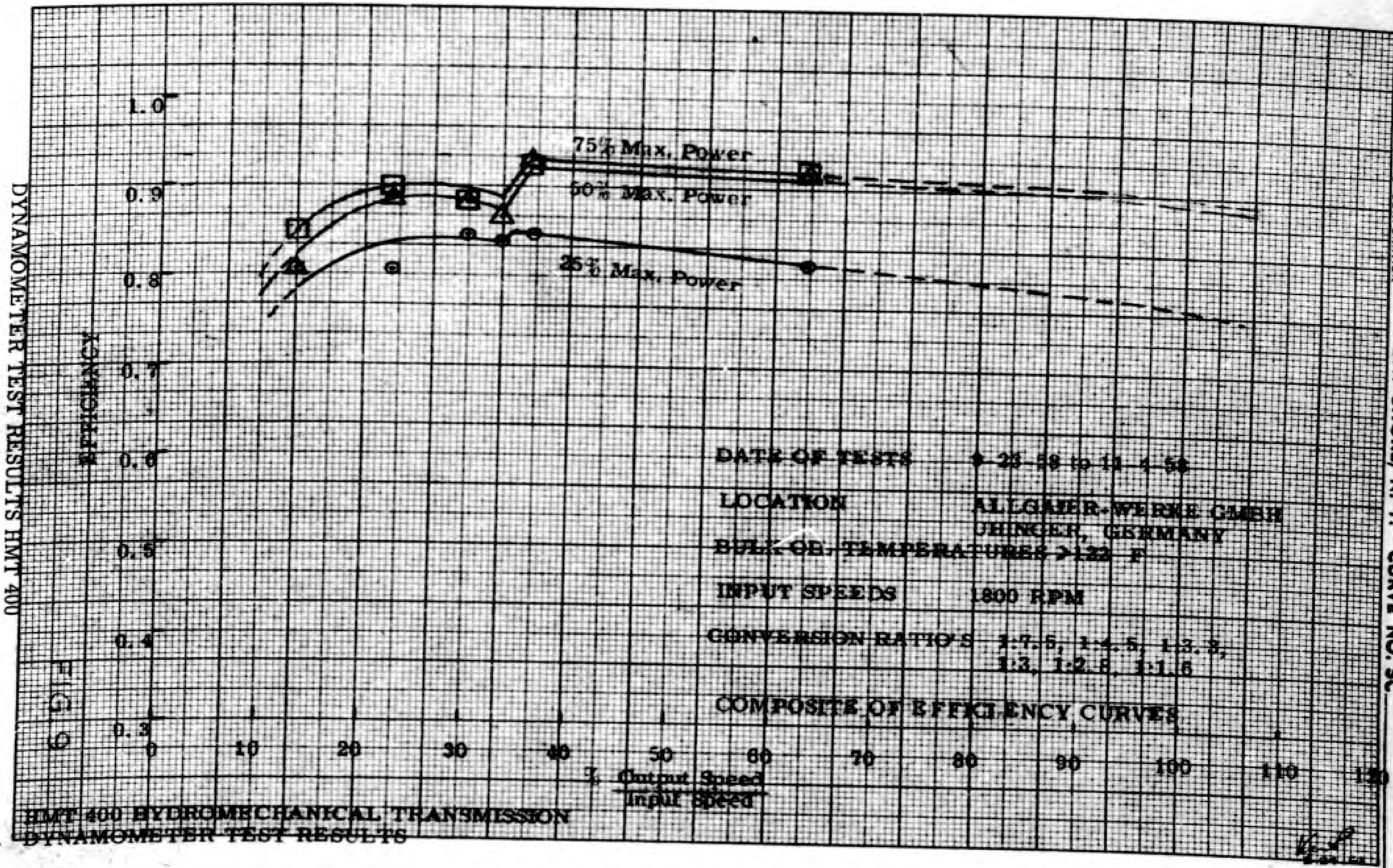

FIG. 9

DYNAMOMETER TEST RESULTS HMT 400

from the HMT 400 tests comparing the 100% power efficiency and the 25% power efficiency. This power loss was taken as a function of the maximum input power and used in calculating the constant power loss of the HMPT transmission. The calculations for the mechanical efficiency and parasitic power losses are valid if the test system and the HMPT system are similar, as is the case.

Comparing the overall efficiencies as a function of ratio, it will be seen that the two curves have the same basic shape. The maximum efficiency of the HMPT and the HMT 400 occurs near maximum ratio. The efficiency of the HMPT is lower throughout the range because of the added steering planetaries and the potential for increased ratio coverage which requires more hydrostatic elements. When all configuration differences are taken into account, it may be seen that comparable performance is achieved.

Vehicle Speed - On a level, hard surface road, with a vehicle rolling resistance of 100 lb/ton and 18.84 inch pitch diameter sprocket, the proposed power package will provide the vehicle with a maximum speed of 40 mph.

Hill Climbing - At an assumed vehicle rolling resistance of 100 lb/ton, the proposed power package will propel the vehicle continuously at 2.5 mph up a 70 per cent slope, drawing 183 HP from the prime mover, with the heat losses from the transmission satisfactorily discarded to the atmosphere by the heat exchanger of the air/oil cooler, and with attainment of an acceptable maximum sump temperature in the order of 180°F. Going downgrade on 70% slopes at 5 mph can also be maintained indefinitely within the transmission and brakes design temperature limits. Similarly, the power package will function continuously and satisfactorily on 60% side slopes.

Durability - The proposed transmission is designed to provide maximum durability. A duty cycle was assumed for a basic requirement of 5000 miles consisting of 3,300 miles of cross-country operation at 5-7 mph with equal amounts of running at 50% and 100% of rated engine load, and 1,700 miles of secondary road operation at 15-18 mph with equal amounts of 55% and 80% of rated load. This is equivalent to approximately 650 hours of operation at about three-quarter power. Although the hydro-mechanical transmissions with regenerative steering are in the advanced stages of design, field experience is lacking. Sufficient testing has been conducted upon similar components and complete truck transmissions, however, to justify the conclusion that the proposed assembly will operate satisfactorily for a period of time well in excess of 650 hours under the loading conditions specified above.

<u>1000 Hours Between Overhauls</u> - As pointed out in the discussion of the requirement for 5000 miles of durability, no steering transmissions of this sort have been operated for 1000 hours. Components have demonstrated equivalent service life, however, in the HMT 400 which has been tested for over 15,000 miles of Ordnance vehicle operation. It is concluded that a developed transmission of the proposed arrangement would furnish more than 1000 hours of trouble-free operation.

Hydraulic Fluid - A wide range of lubricating oils of various viscosity grades, produced in accordance with Specification MIL-L-2104A has been demonstrated to be satisfactory hydraulic fluids for hydromechanical transmissions. Lubricating oil to MIL-L-10295A having similar lubricating qualities, but other viscosity temperature characteristics, is expected to give comparable performance.

Environmental Temperature Range - The proposed transmission will operate satisfactorily in extreme environmental temperatures of -65° to 125°F and may be stored in temperatures of -65°F to 155°F in accordance with military specifications, provided hydraulic oil of appropriate viscosity is used. In addition, the proposed power train is water, corrosion and fungus proof in compliance with government specifications.

Space Requirement - The volume of the steering transmission is approximately 14.5 cubic feet. The overall dimensions are: length 24 inches; height 32 inches; and width 30 inches (width 41.5 inches over final drives).

Weight - Based on analysis of the preliminary layout the estimated dry weight is 809 lbs, exclusive of final drives. The total weight (dry) is estimated at 1015 lbs, including final drives.

Cooling - Cooling requirements of the transmission are provided for through the use of an oil cooler. The heat rejection capacity provided by the cooling system is adequate for continuous operation of the power package throughout the speed and load range and for the dissipation of heat generated during vehicle braking.

Power Take-Off (Optional) - Full power take-off is provided by a shaft extension at the outboard face of the transmission.

Maximum Tractive Effort - The maximum tractive effort is set by the requirement of 2.5 mph up a 70% slope. The proposed transmission meets this requirement. Maximum sprocket torque is 19,200 lb-ft, developed from 0 to approximately 3 mph. With a sprocket pitch diameter of 18.84 inches, The corresponding tractive effort is [illegible],000 lbs.

Drawings - Facility drawings of the proposed power train would be furnished in accordance with the requirements of the OTAC Standard Selection of Components and Drawing Clauses for Inclusion in Development - Type Engineering Contracts, Revision dated 15 November, 195[illegible].

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