MASTER

SIXTH QUARTERLY

PROGRESS REPORT

STUDY OF REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

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74-310860(18) January 30, 1976

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AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

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ATTACHMENTS: Appendix I

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N/C	RJH	See Title Page	1-30-76	Initial Issue.

SIXTH QUARTERLY PROGRESS REPORT STUDY ON REDUCTION OF ACCESSORY HORSEPOWER REQUIREMENTS

1.0 INTRODUCTION

This is the sixth quarterly technical progress report submitted in accordance with requirements outlined in Attachment B, Reports and Reviews, Energy Research and Development Administration Division of Transportation Energy Conversion (TEC) Office of Highway Vehicles, Vehicular Systems Branch Contract E[04-3]-1095. This report covers the period from October 16, 1975 through January 15, 1976.

The program objective is to evolve and define an accessory drive system that will minimize system power consumption of driven accessories on an internal combustion engine in a passenger automobile. The initial program phases established concept feasibility, determined potential fuel savings, and selected a drive system design for concept mechanization.[1] The present Phase IV will carry the program through prototype fabrication and bench, engine, and vehicle tests. The final program objective is a detail drive system design and a demonstrated overall vehicle fuel savings potential.

The accessory drive program is being approached as a two-step development involving accessory drive mechanization and improvement of accessories. The drive is intended to be adaptable to 1979 internal combustion engines, utilizing standard automotive accessories, resulting in initial overall vehicle economy improvement with minimum introductory impact. The second step relates to accessory performance, integration, and design improvements associated with limited speed drive verification.

[1] Program Phases I, II, and III were presented in Reports 74-310860(1) through (12).

Although the primary goal is fuel economy improvement, the manner of accomplishment must be acceptable to both manufacturer and customer. Factors such as first cost, maintainability, and impact on existing capital investment were significant considerations in the selection of a variable ratio belt drive as the speed control mechanism.

Phase IV consists of the primary development tasks:

Task 1 - Design fabrication, component, and laboratory engine tests of prototype variable ratio belt drive.

Task 2 - Computer modeling and computer drive cycle analyses of the baseline compact car utilizing the accessory drive system.

Task 3 - Accessory drive integration with the baseline vehicle for system test evaluation. Final drive system development evolution will be an iterative process with the vehicle integration test evaluation.

Tasks 1 and 2 are essentially complete and the program is entering into Task 3.

1.1 Contents

Major technical accomplishments during this reporting period were:

- Hydromechanical and mechanical variable-ratio belt drive fabrications completed.
- Compact vehicle fuel economy analysis completed.
- A formal program status presentation was made at the November 18, 1975 Contractores Conference (included as Appendix I to this report).

- Initial engine check runs for both drive systems were completed.
- Mechanical drive system development and performance mapping was initiated.
- Hydromechanical drive system development tests and modifications were initiated.
- The hydromechanical drive system installation into the test vehicle was completed.
- The engine/dynamometer test rig, including accessory load simulators, was completed.
- The basic test vehicle instrumentation was completed.

2.0 TECHNICAL PROGRESS

2.1 Accessory Drive System Fabrication

Two prototype accessory-drive systems have been fabricated and are shown on Figures 1 and 2. Additional packaging hardware, required to install each drive system in the Mustang II test vehicle, has also been completed.

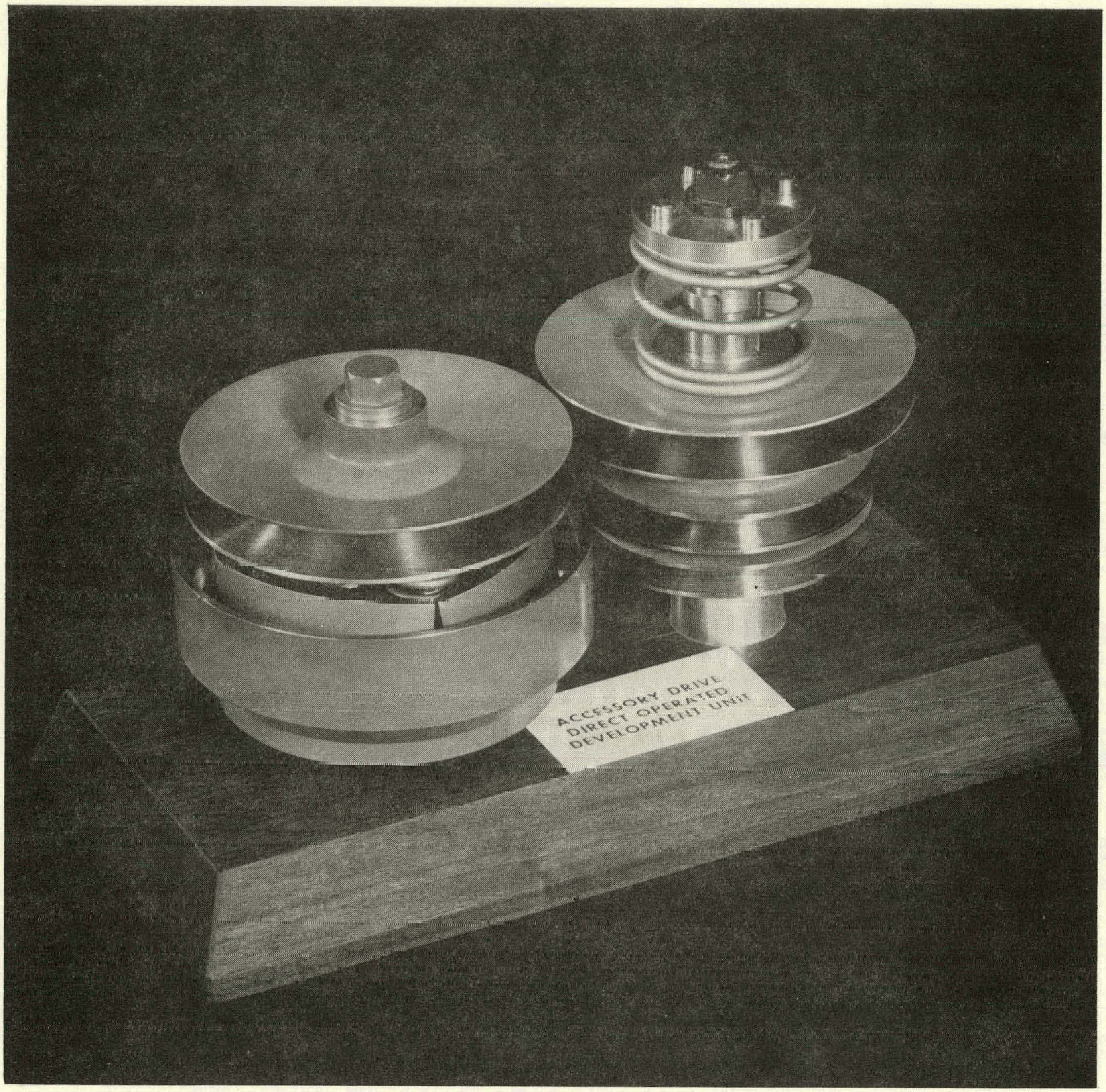
Design verification and development test phases are well under way for both drive systems. The basic test sequence is as follows:

(a) Mechanical System

- Dynamic balance; speed and governor position survey
- Spin-pit test; overspeed capability verification
- Varidrive bench test; green run, adjustments, load and speed profiles, belt evaluation
- Engine test; fit and function under actual engine and accessory operating conditions
- Vehicle test, full vehicle operating profile and final economy test

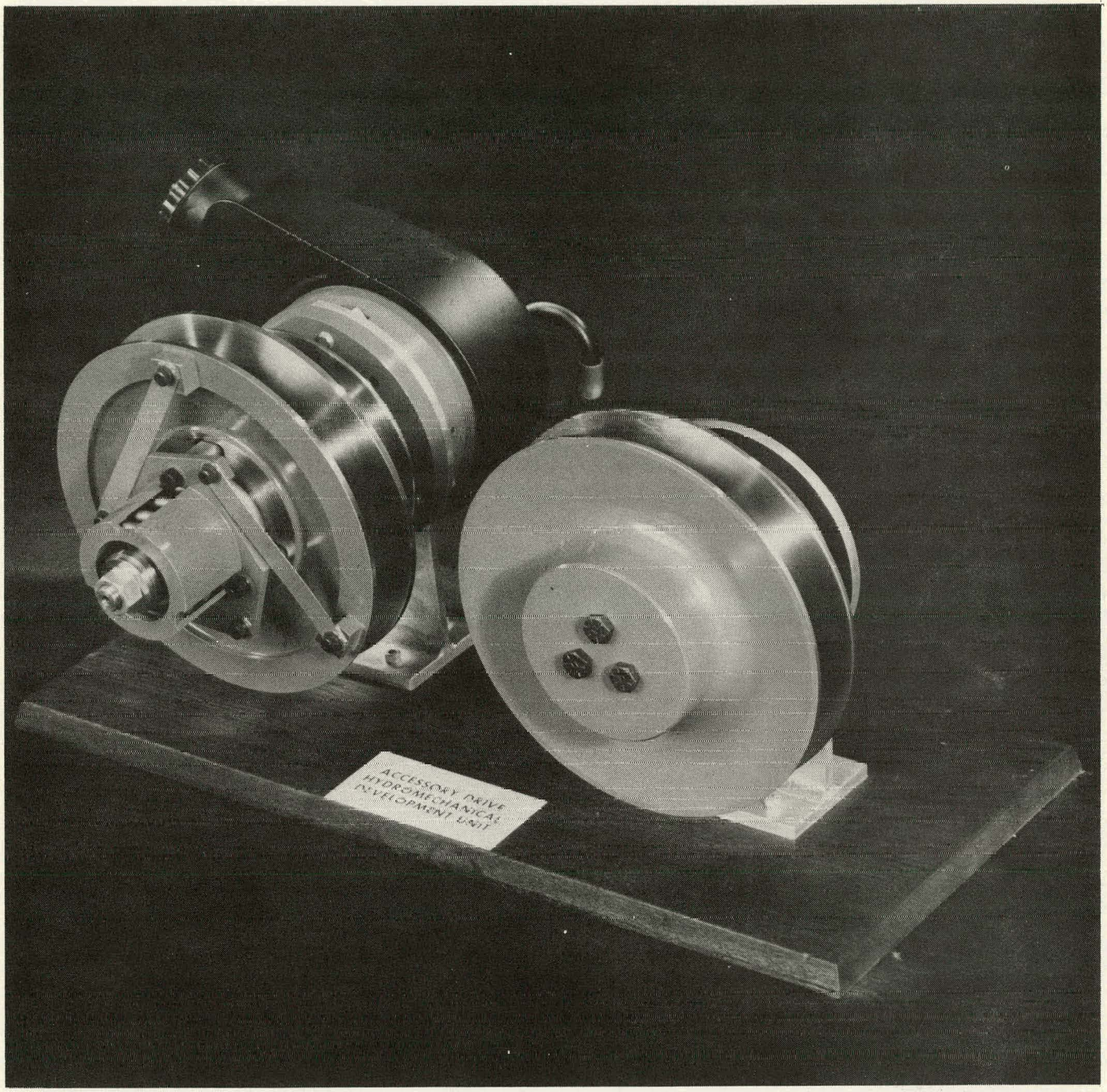
(b) Hydromechanical System

- Pressure test; control and power steering relief valve operational characteristics



DIRECT OPERATED, MECHANICAL
VARIABLE-RATIO-BELT DRIVE
3605140 DIRECT OPERATED, DRIVER SHEAVE
3605141 LOAD CONTROL DRIVEN SHEAVE

FIGURE 1



HYDROMECHANICAL, VARIABLE-RATIO-BELT DRIVE
3605100 HYDRAULIC SPEED CONTROL, DRIVEN SHEAVE
3621590 TORQUE SENSOR, DRIVER SHEAVE

FIGURE 2

MP-52413

- o Static governor adjustment; weight and rate and position measurements for initial governor set point

- o Varidrive bench test; pump run-in, relief valve and governor dynamic adjustments, system response, load and speed control, and efficiency and belt wear evaluation

- o Engine test; fit and function under actual engine and accessory operating conditions. (This test will be conducted concurrently with varidrive bench tests.)

- o Vehicular test; accessory and drive system performance, cycle of operating profile, and fuel economy.

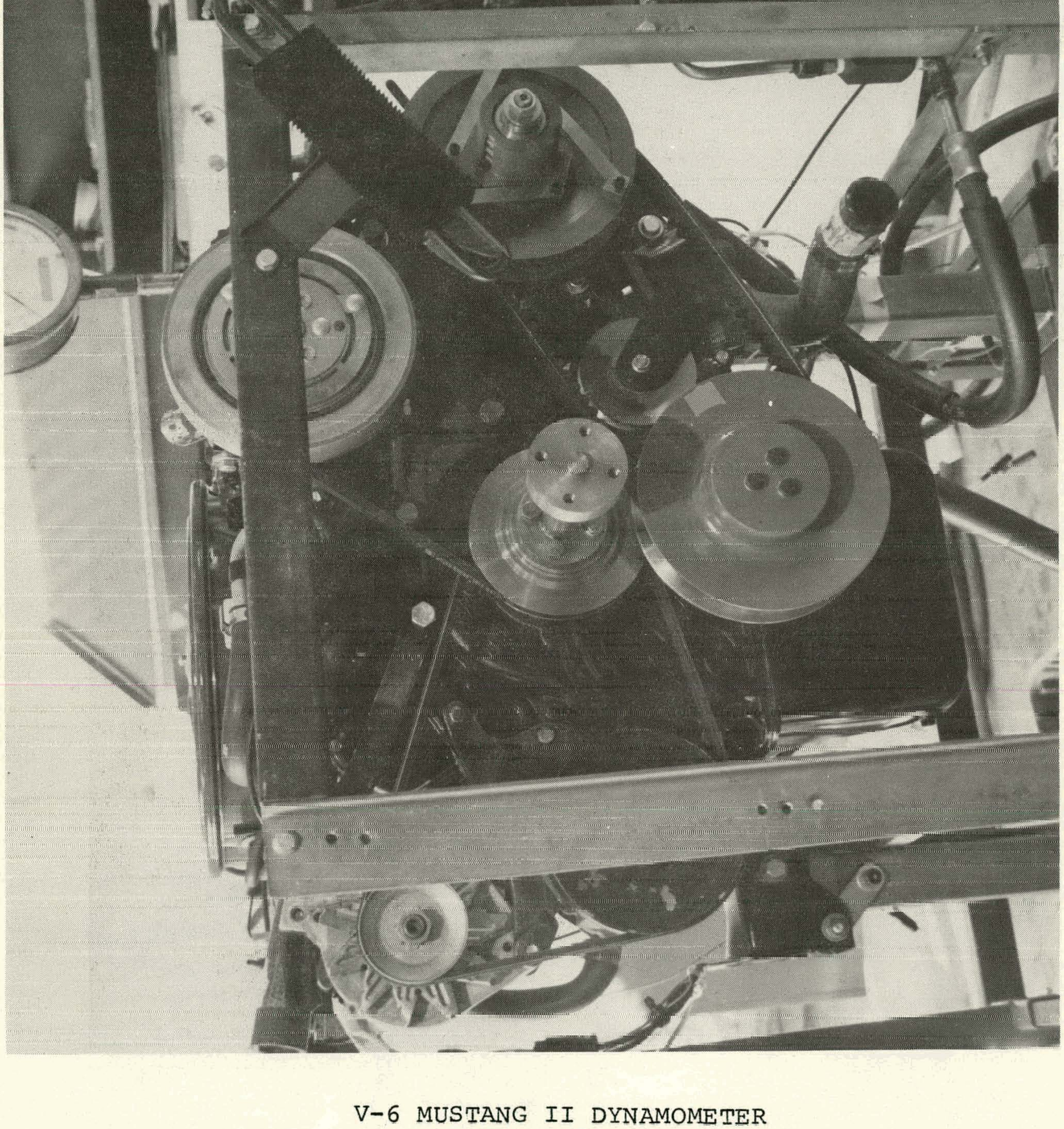
2.2 Hydromechanical Variable Ratio Belt (VRB) Drive Development

The hydromechanical VRB drive system was installed on the V-6 Mustang II engine/dynamometer test stand, as shown in Figures 3, 4, and 5, for initial engine fit and functional check runs. The drive system accumulated four test hours operation throughout the design speed range. This early "green run" was performed to ascertain, qualitative, off-design operating characteristics and to define additional testing within critical operating areas.

The drive system demonstrated full ratio change while driving all accessories. Test anomalies were as follows:

- o The drive system-to-engine compartment noise ratio was unacceptable due to hydraulic pump configuration.

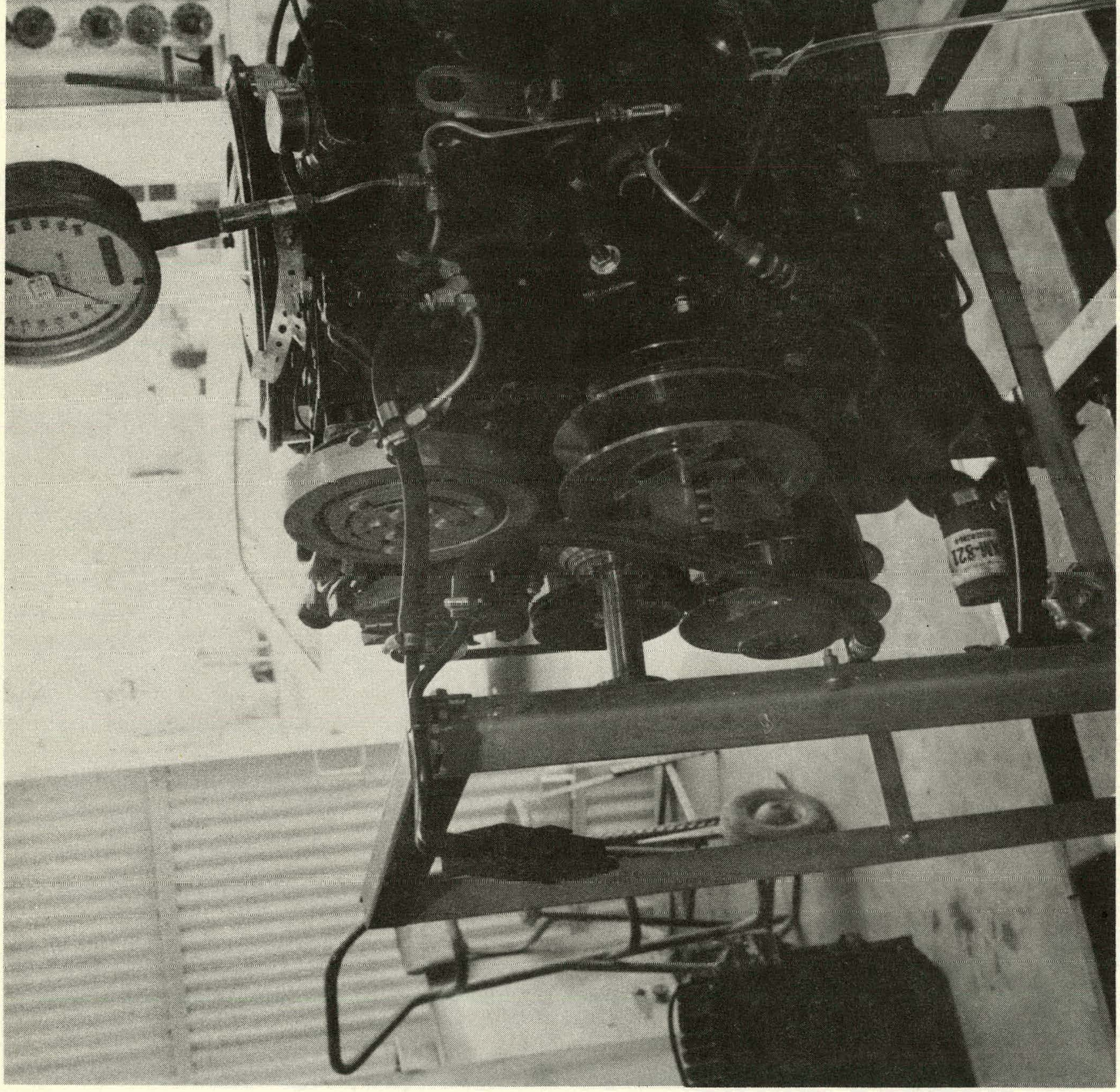
- o The drive ratio control system exhibited minor speed transients.



V-6 MUSTANG II DYNAMOMETER
TEST STAND

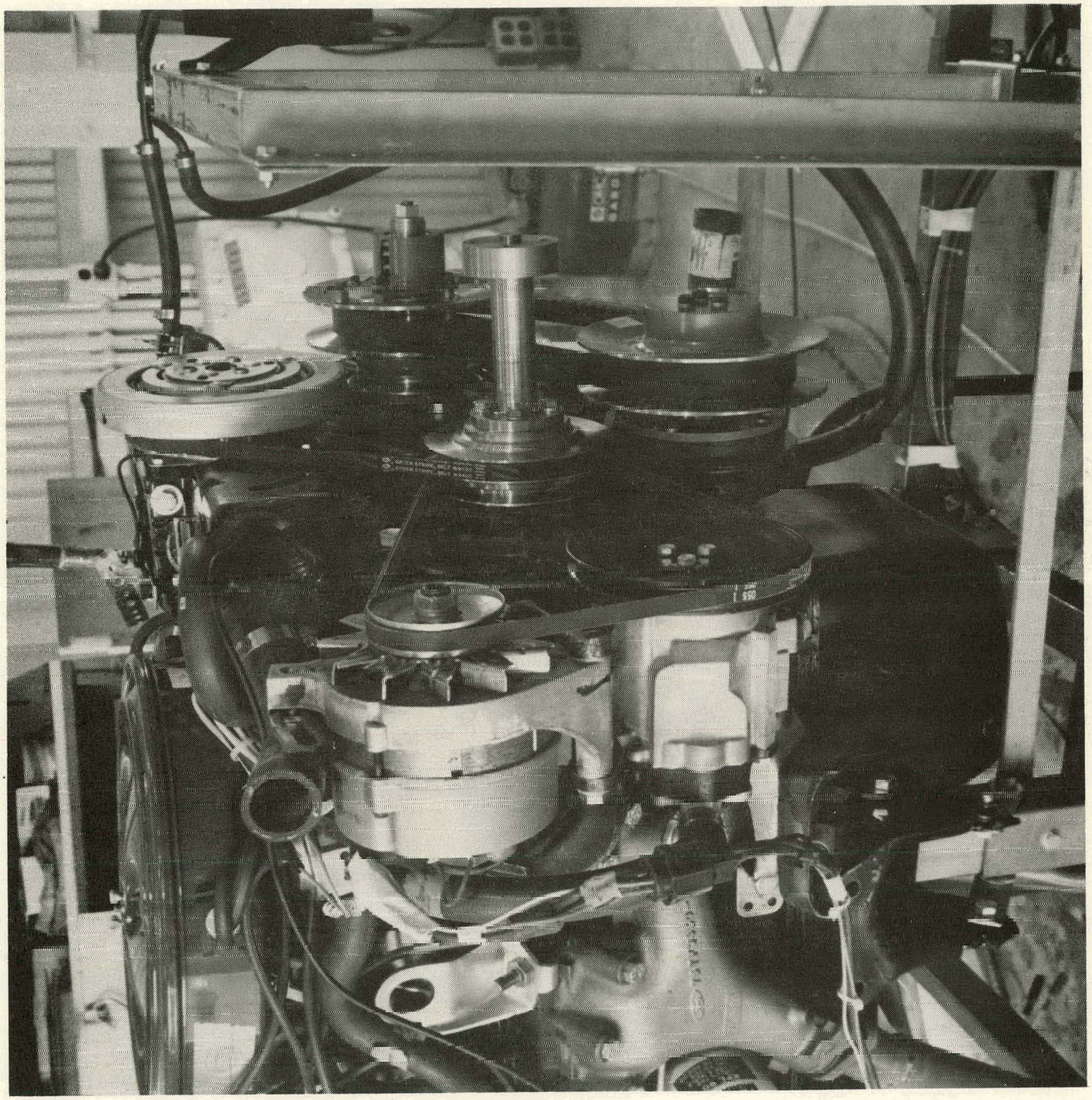
FIGURE 3

MP-52908



V-6 MUSTANG II DYNAMOMETER
TEST STAND

FIGURE 4



V-6 MUSTANG II DYNAMOMETER
TEST STAND

FIGURE 5

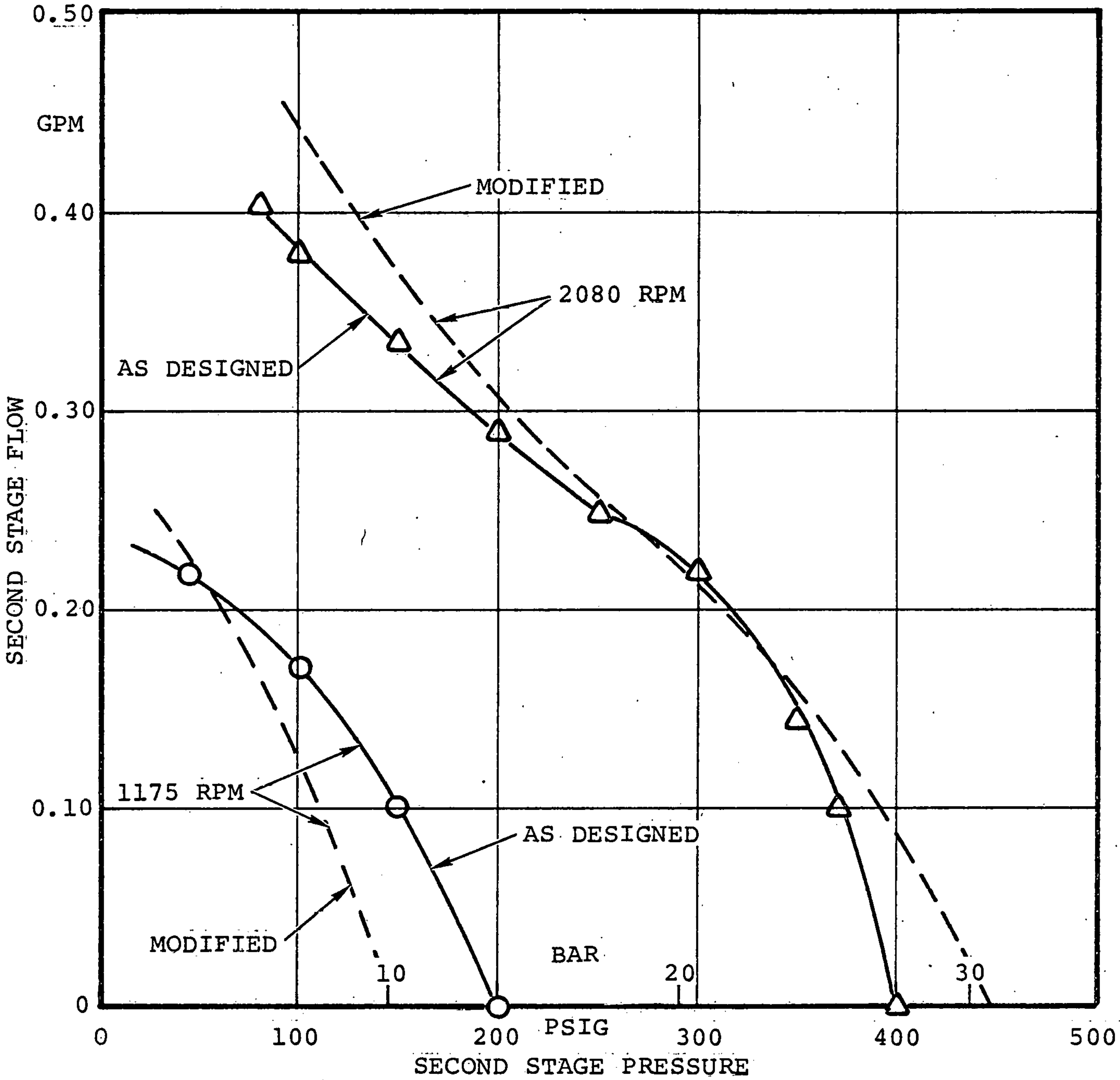
- o The drive system start-up ratio would not progress from maximum underdrive to maximum overdrive as predicted, because minimum control pump output could not be obtained below 625 rpm pump speed.

The drive system was disassembled for hardware inspection and all details were in good condition.

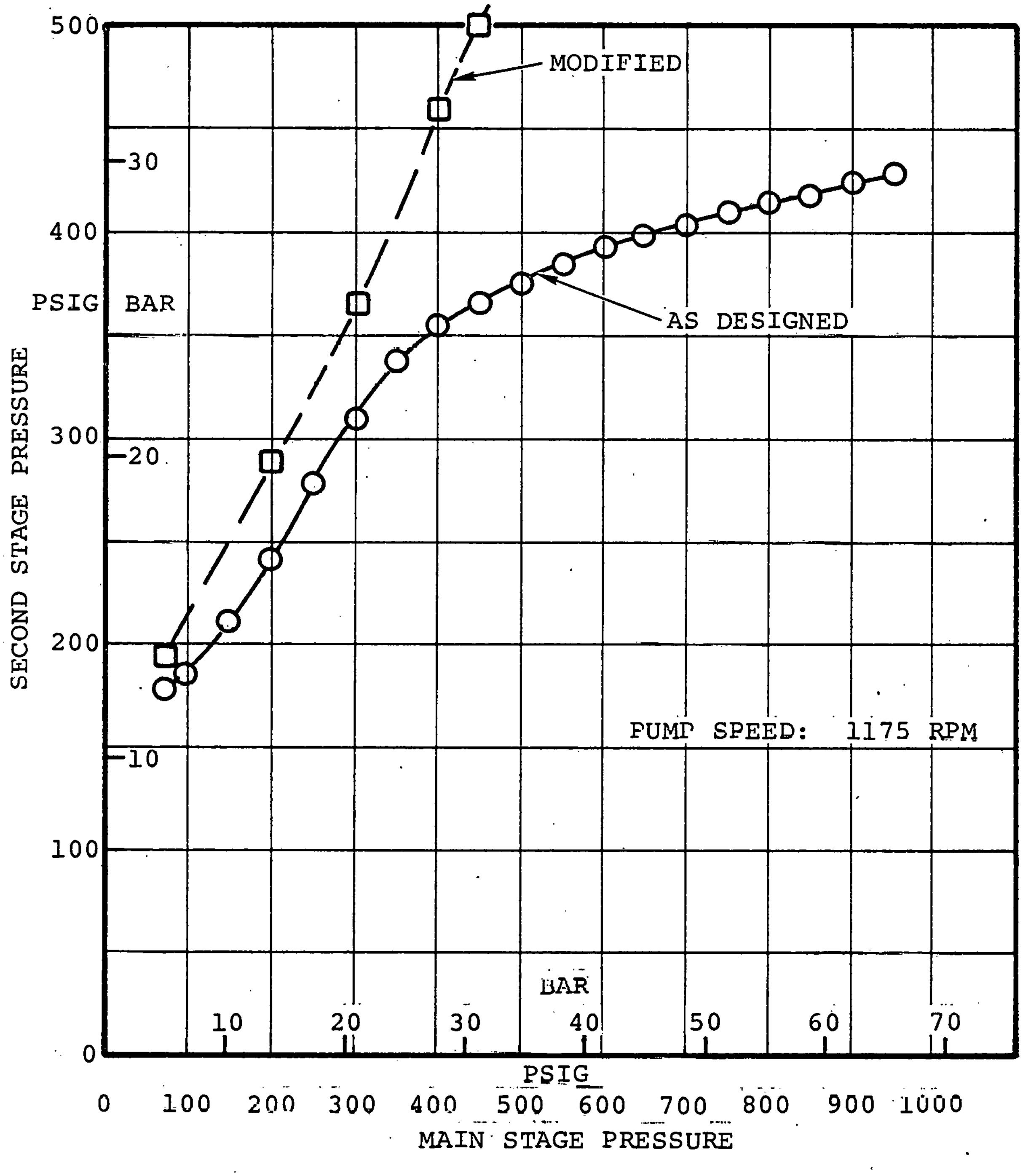
The hardware was modified to improve pump and governor performance. Modifications are as follows:

- o The secondary pump output companion port was machined to increase pump capacity, and decrease noise.
- o The pump inlet was improved by relieving the inlet divider.
- o A double port governor valve was used to improve sensitivity and remove radial unbalance force.
- o The pump running clearance was decreased, to improve performance.

The pump and control hydromechanical VRB driven sheave was assembled, using the above modified hardware, and installed in the bench test rig for evaluation. The dual output power steering pump secondary stage (control pressure) was again mapped to determine the modification effects. The modifications slightly improved the governing speed performance (2080 rpm), but decreased the idle speed performance due to increased cross-port leakage caused by adding the companion port. Refer to Figure 6. This effect becomes visable by comparing main-stage power steering pressure with second stage control pressure. Refer to Figure 7. Governor modifications were not evaluated at this time because of poor pump performance.



FLOW VERSUS PRESSURE - SECOND STAGE
FIGURE 6



SECOND STAGE PRESSURE VERSUS MAIN STAGE PRESSURE
FIGURE 7

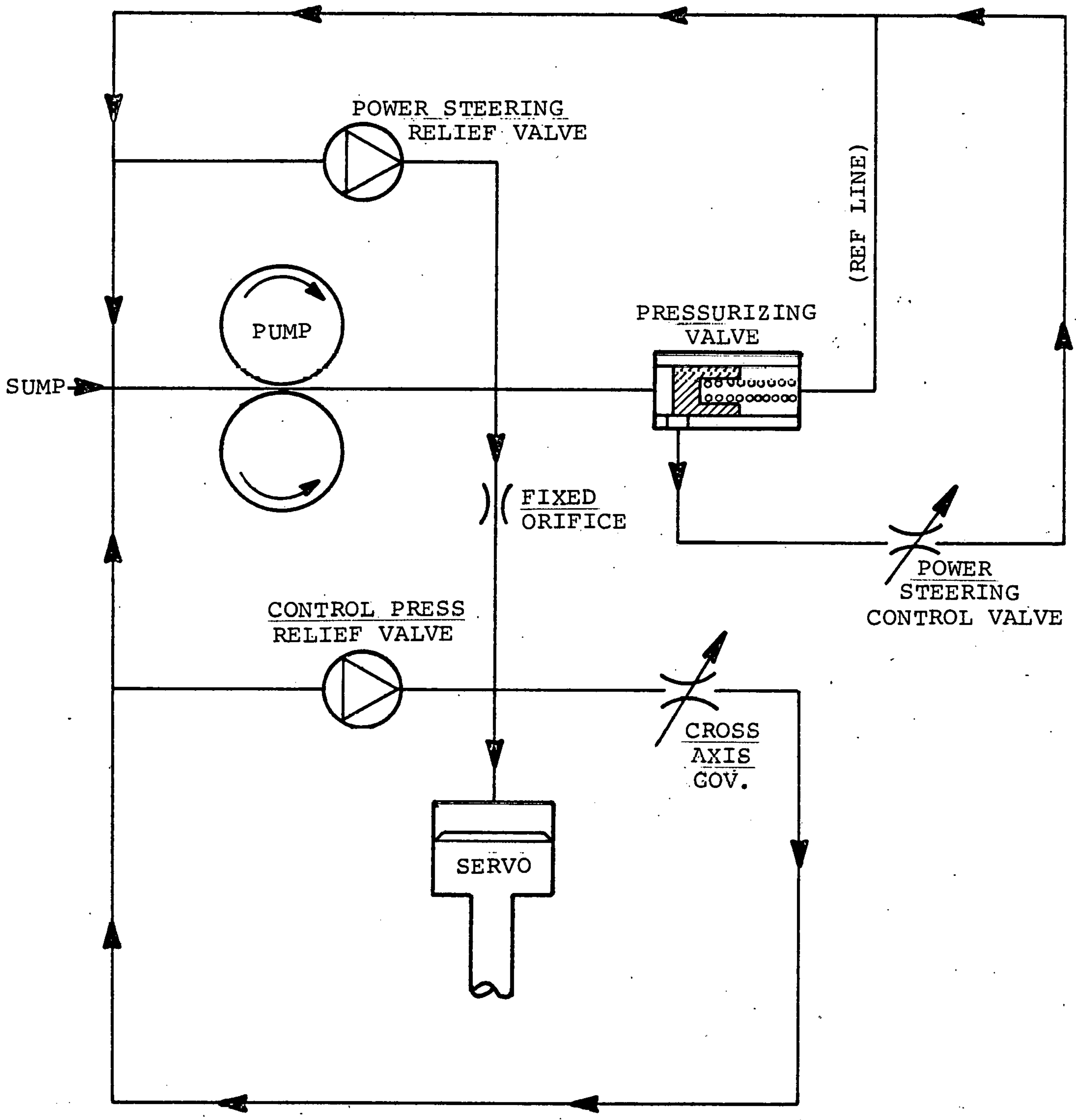
Pump noise remained objectionable, but further changes to improve low speed performance were considered too complex within the program scope. The dual pump output concept would prove viable where operation did not impose rigorous low-speed performance requirements, or cost did not prohibit extremely close tolerance pump designs.

The hydraulic system was then modified to evaluate a single-stage pump concept. Figure 8 presents the schematic diagram of the configuration that was tested. Hardware specifically designed for this concept would not have two separate relief valves. Instead, both functions would be included within the pressurizing valve.

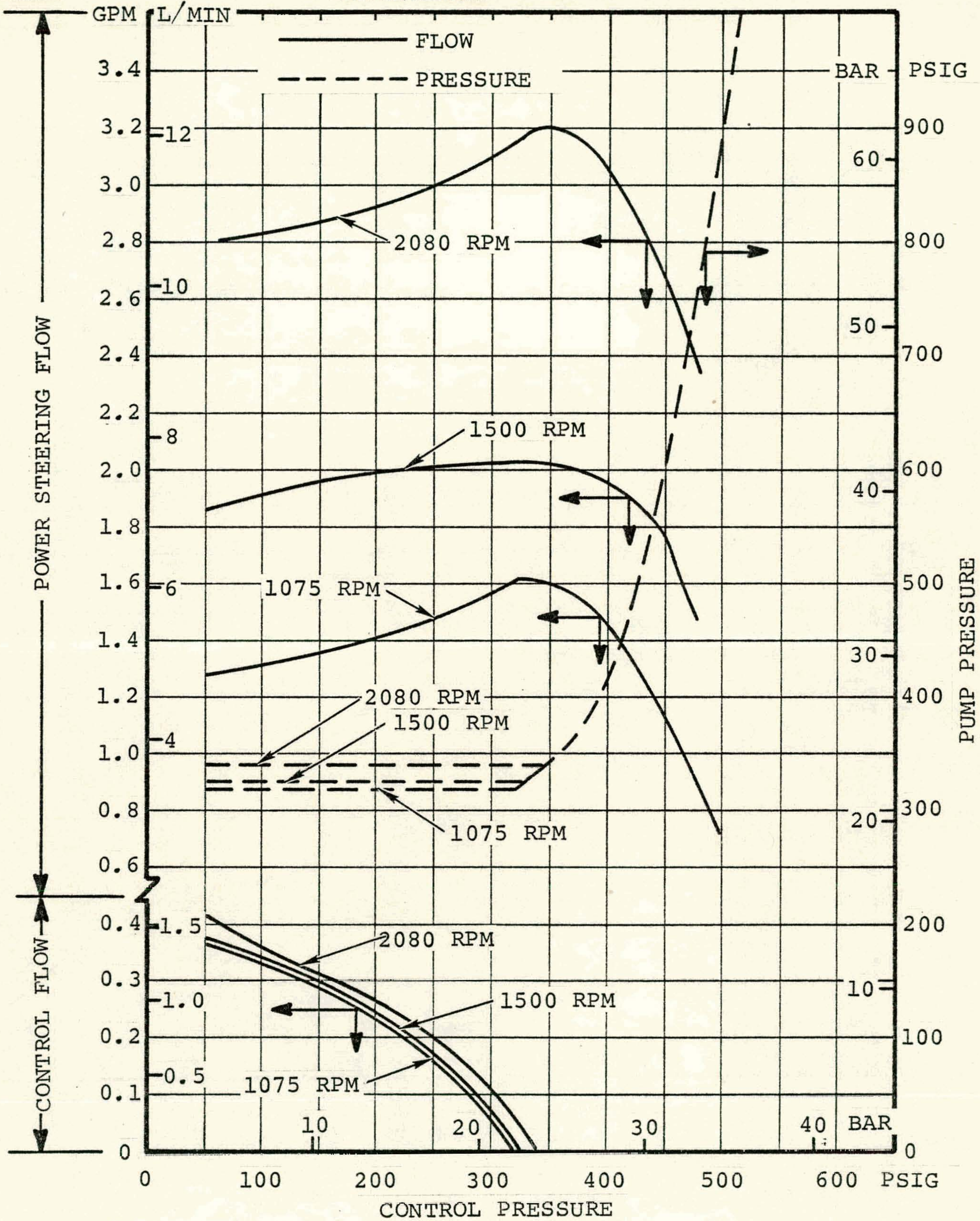
Orifice-controlled flow from a single pump output requires maintaining the upstream orifice pressure at a fixed minimum value to achieve proper servo flow response, and maximum no-flow pressures. This requires the pump load to be raised above that normally expected for no-load, stock power-steering pump operation. An increased pump load results in increased drive power. The power increase is approximately 186 watts (0.75 horsepower) over the stock power-steering pump at 1200 engine rpm. At 1750 engine rpm, the power requirements are equal (55 km/34 mph) due to constant speed drive operation. A net power savings is realized at any speed above 1750 engine rpm.

The single stage pump concept test results are presented in Figure 9. A 0.76 mm (0.030 in.) diameter, orifice was used with the pressurizing valve set at 23 Bar (330 psig). The data indicates that sufficient control and power steering flows can be achieved at all pressure and speed conditions.

Governor stability, hysteresis, and droop tests were then conducted to determine performance characteristics. The data presented in Figure 10 shows that the double-ported governor is stable within a total hysteresis band of approximately 50 rpm, and has a speed droop of 270-280 rpm over the entire governing range.



SINGLE STAGE PUMP SCHEMATIC
FIGURE 8



FLOW AND PRESSURE MAP
FIGURE 9

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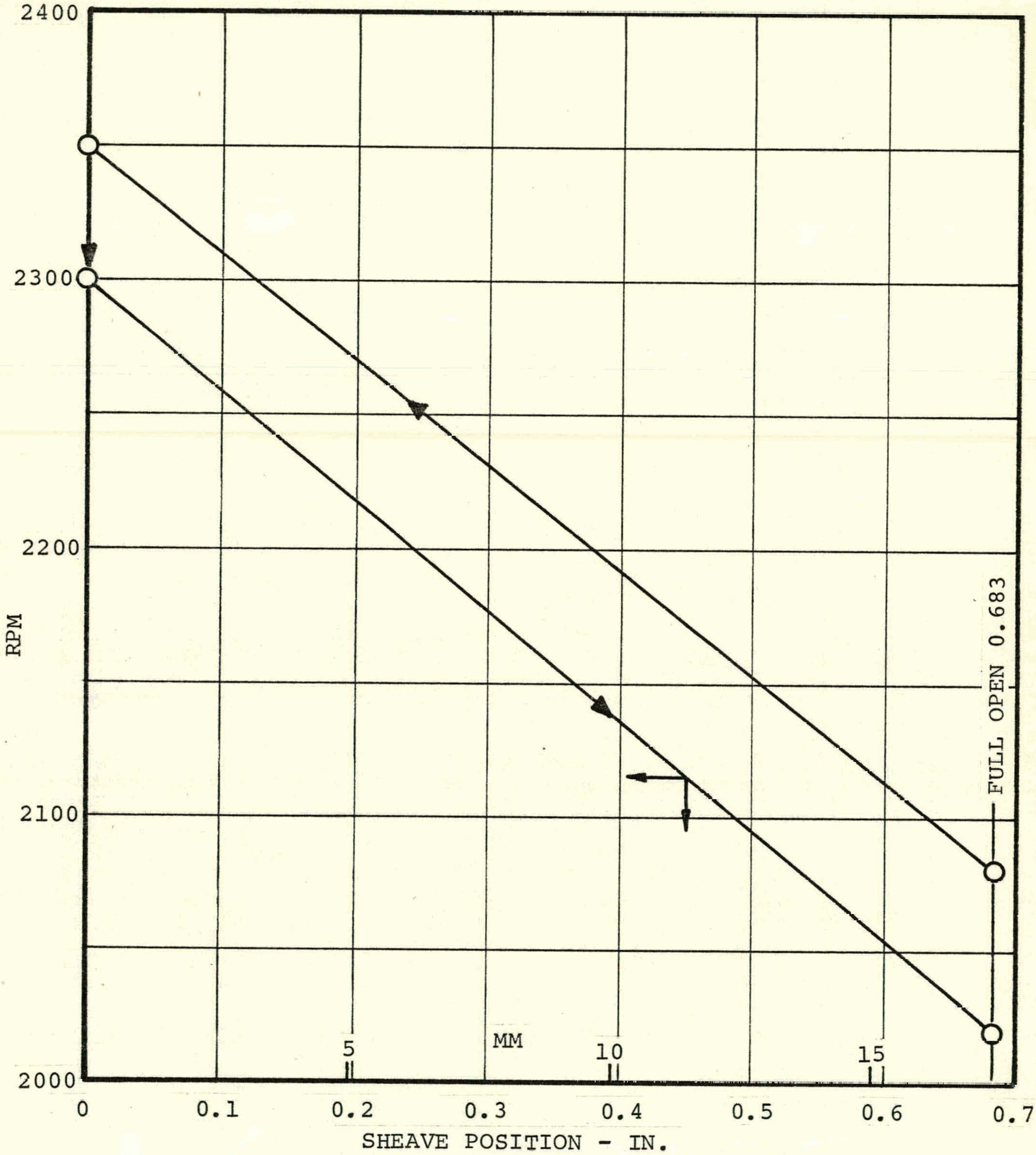

GOVERNOR PERFORMANCE
FIGURE 10

The pump and control data generated by the single-stage concept modified hardware indicates that performance is sufficient to support full scale vehicle testing. This unit will be installed in the Mustang II test vehicle and system performance data will be presented in the next monthly progress report.

2.3 Mechanical Variable Ratio Belt Drive Development

The mechanical drive system was installed on the test engine for qualitative testing and cursory investigation of potential wear areas. Prior to system installation, the driver unit was balanced to an equivalent of 36 mm-g (0.050 in. oz). The unit was then run in the "spin pit", at speeds up to 6,000 rpm, and remained stable throughout the speed range. Dynamic balance was excellent at speeds below, within, and above the operating range.

The drive assembly was installed on the engine and the driven unit connected to all accessories, representing an actual installation as shown in Figures 11, 12, and 13. The first series of tests (four runs of 5 minutes each) revealed that the unit functioned well from a qualitative standpoint. However, the VRB drive belt produced rapid heating during short runs. This was caused by misalignment at assembly. The speed governing system commenced operation at 1400 rpm (designed for 1200). The maximum operating rpm was not reached since a correct length belt was not available and driver maximum stop distance was not set.

Disassembly revealed the driver had not been fully seated on the crank stub, resulting in misalignment and significant fretting corrosion of the driver fixed sheave mount bore.

Continued development testing of the mechanical VRB system was compromised by galling and corrosion fretting associated with the

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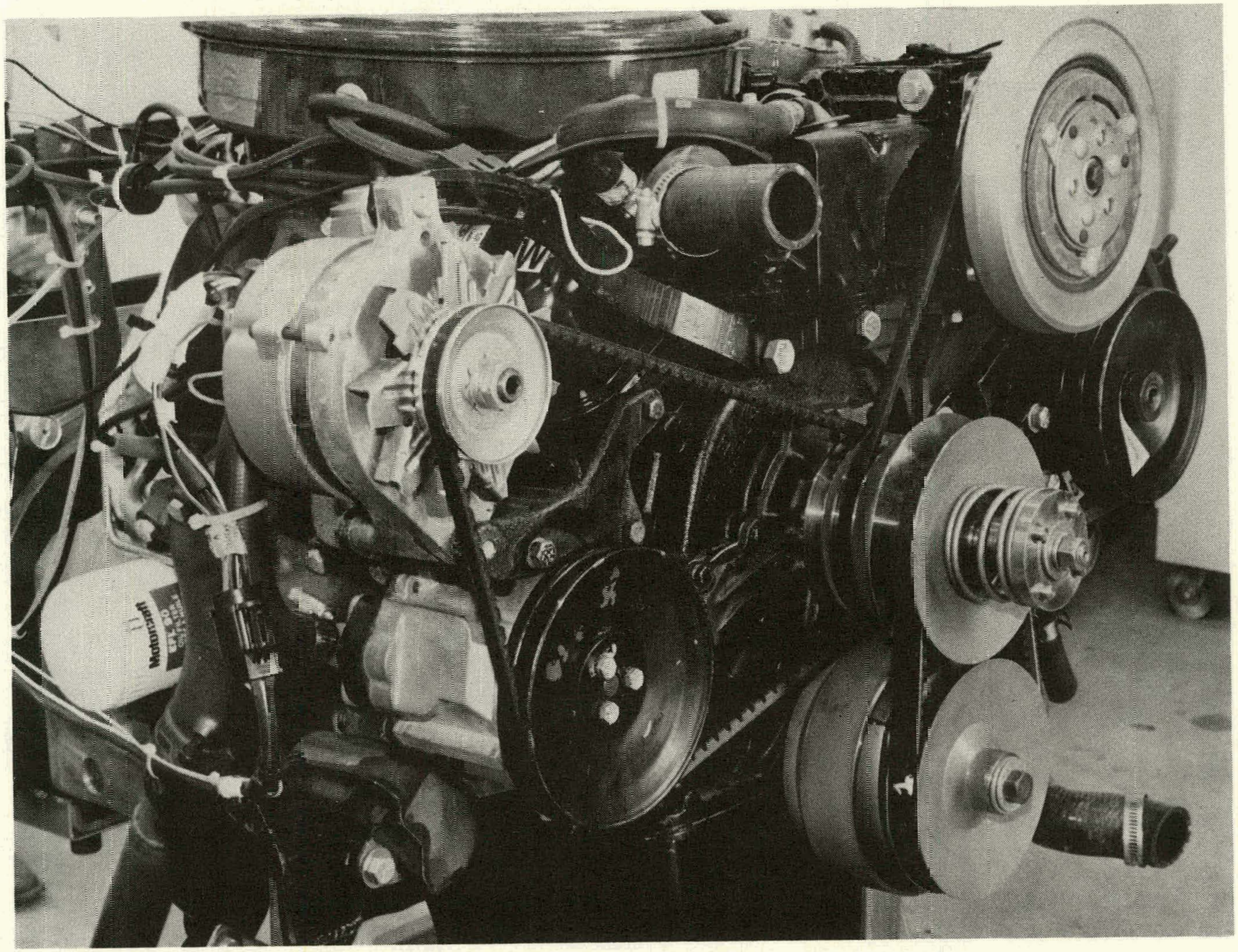
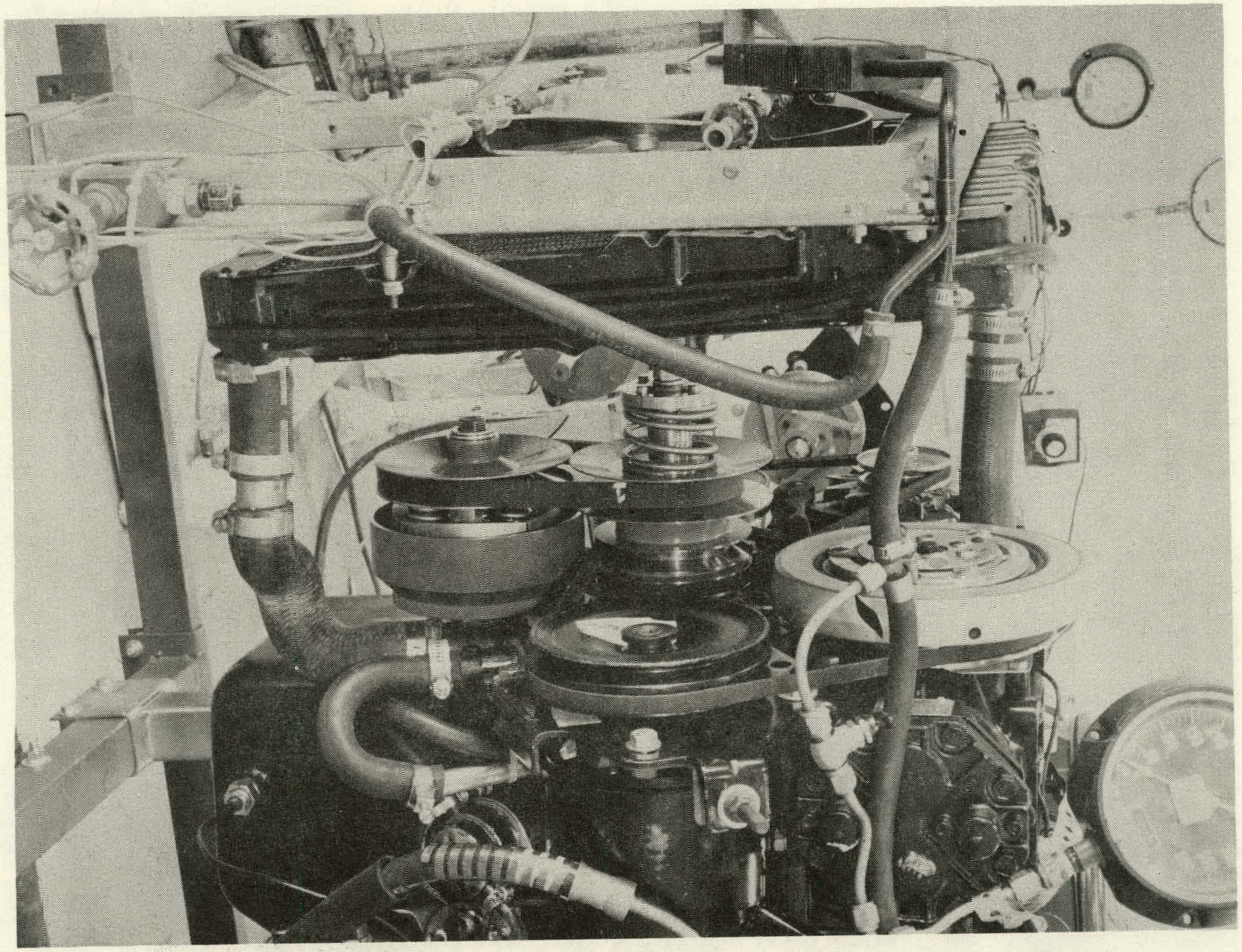


MECHANICAL VARIABLE RATIO BELT DRIVE

FIGURE 11



MECHANICAL VARIABLE RATIO BELT DRIVE

FIGURE 12

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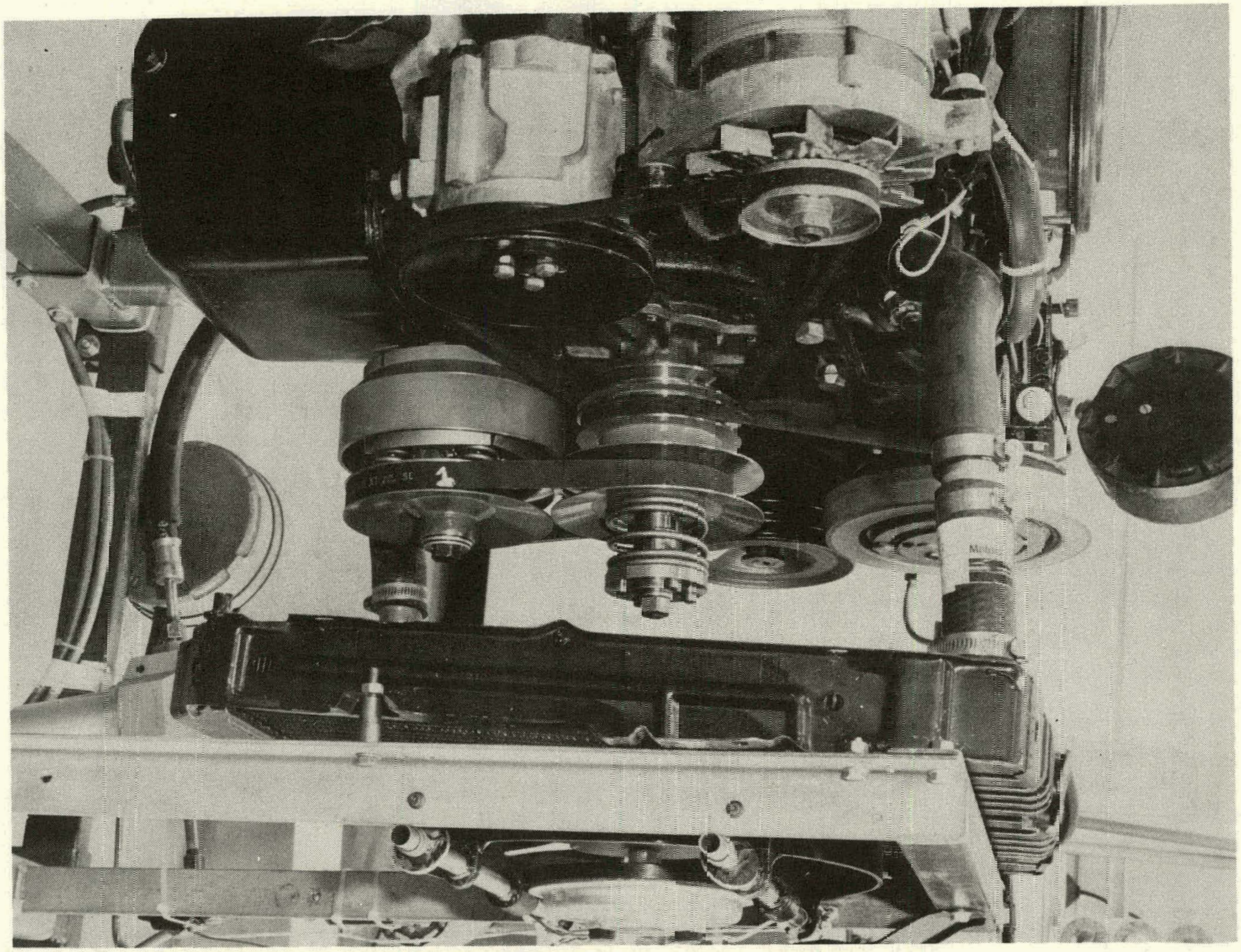


MECHANICAL VARIABLE RATIO BELT DRIVE

FIGURE 13

movable flange in the driven sheave. Several different types of prelubricated materials were used. However, none provided sufficient anti-galling/fretting protection to allow fully instrumented system performance testing. Qualitative (visual) testing revealed excellent speed response; point-to-point hysteresis; full range hysteresis; and minimum load droop effects.

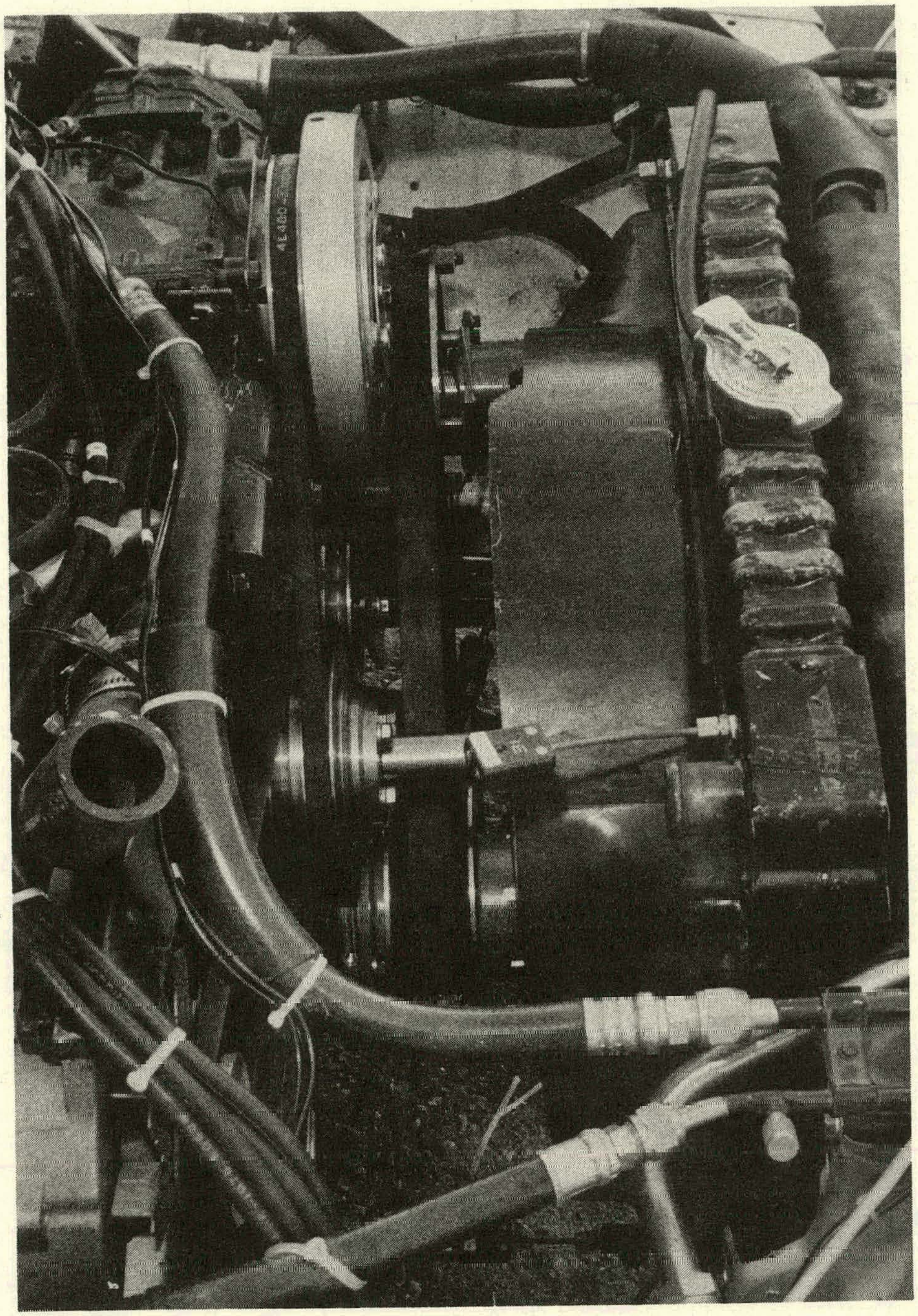
The proper length belts have been determined and are on order. Interim tests were attempted with a "short" belt, however, installation difficulties were prohibitive.

The fretting sheave redesign is complete and parts are on order. The rework consists of chrome plating the sheave mounting shaft, manufacturing a new movable sheave with room to insert a bushing with a long length-to-diameter ratio and a new driving spider with torque arms at a larger diameter. The spider will accommodate slots to provide monopole pulse signals for instrumentation use. These components will arrive and be tested prior to the next reporting period.

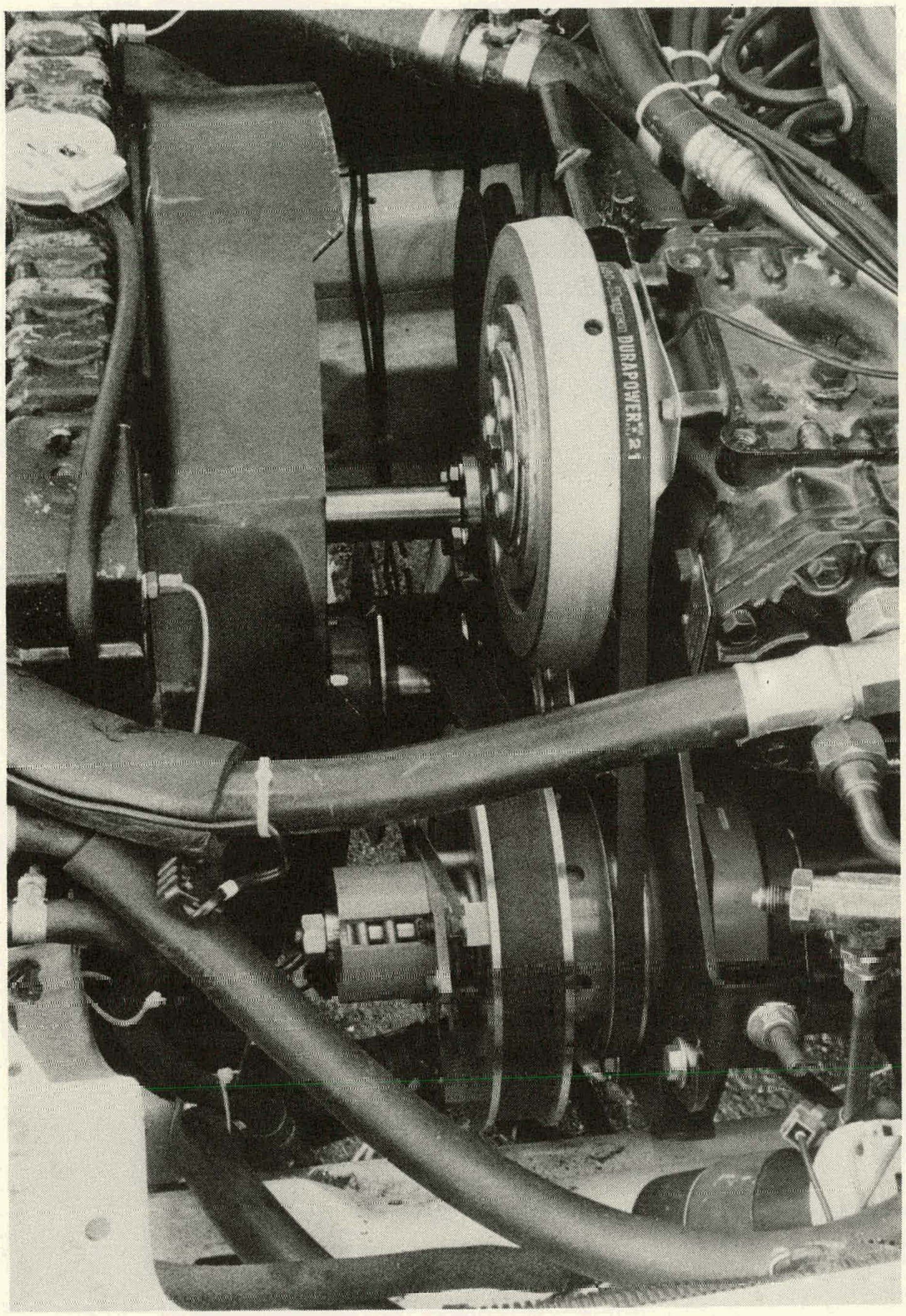
Preliminary test results indicate that an input programmed mechanical system will function as a variable ratio drive device. However, significant test efforts will be required, in conjunction with analysis, to determine the most practical compromises relative to fit, function, materials, and endurance.

2.4 Vehicle Installation

The hydromechanical accessory drive system has been installed in the test vehicle as shown in Figures 14 and 15. The radiator and air conditioning condenser were moved 38 mm (1.5 in.) forward to accommodate the drive system and provide 13 mm (0.5 in.) trailing and 25 mm (1.0 in.) leading edge fan clearance. The radiator and



HYDROMECHANICAL ACCESSORY DRIVE SYSTEM
FIGURE 14



HYDROMECHANICAL ACCESSORY DRIVE SYSTEM

FIGURE 15

MP-53398

condenser have been relocated to the maximum possible forward position for this installation. The minimum possible relocation could be as little as 16 mm (0.625 in.) based on fan clearances found in the 1974, Ford Pinto Wagon.

The air conditioning compressor drive belt has been moved aft by simply reversing the sheave and harmonic damper positions on the clutch assembly. The aft accessory belt, driving the alternator and emissions control air pump, has not been disturbed. The accessory positions and belt tracks showing the dynamometer installation are best illustrated in Figure 3. The accessory drive belts are identified as follows:

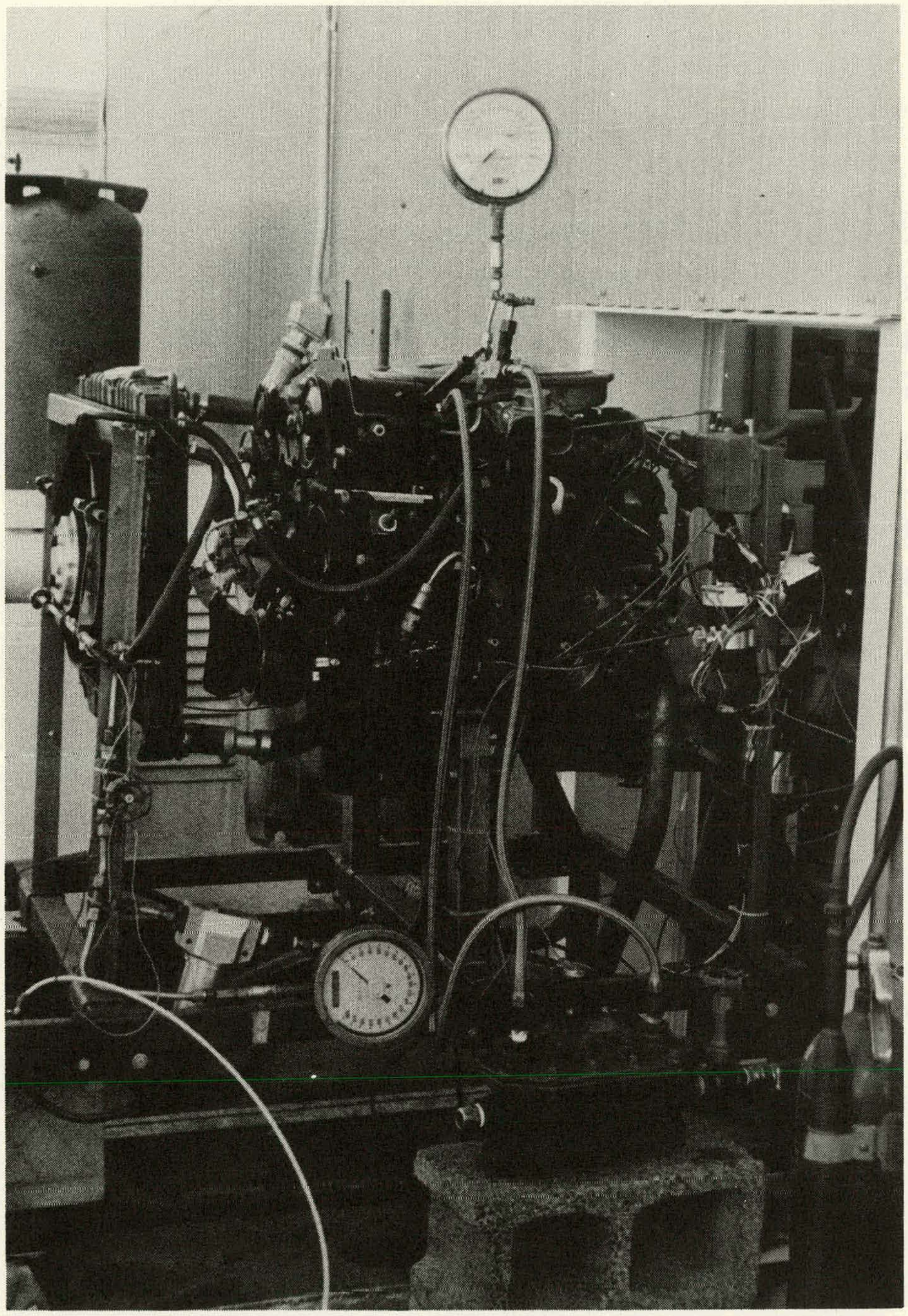
Belt Position	Top Width	Length
Fore-Main Drive (crank to P/S)	25 mm (0.984 in.)	110.5 cm (43.5 in.)
Mid-Power Takeoff (P/S to A/C to W/P)	13 mm (0.5 in.)	122 cm (48 in.)
Aft-Transfer (W/P to Alt to A/P)	11 mm (0.44 in.)	112 cm (44 in.)

The remaining installation hardware consists of a dual belt water-pump sheave, fan extension, longer radiator water hoses, and power takeoff belt idler/adjustment sheave.

The vehicle will also accept the mechanical drive system as shown in Figure 11 by installing the crank-to-water pump shaft drive and stock power steering pump.

2.5 Accessory Load Simulation on Engine Rig

The engine/dynamometer accessory load simulation test setup shown in Figure 16 has been completed. This accessory loading



ENGINE/DYNAMOMETER ACCESSORY LOAD SIMULATION TEST SETUP

FIGURE 16

MP-53400

technique loads each belt driven accessory in a manner very closely simulating actual vehicle operating conditions.

A brief description of the accessory loading test setup follows:

(a) Water Pump - The water pump is loaded as normal in a vehicle, that is, belt driven. The load imposed will be that of the pump circulating water through the cooling system.

(b) Radiator Fan - The fan is loaded normally by pulling air through the radiator.

(c) Air Pump - The air pump is loaded normally by pumping air into the exhaust manifolds.

(d) Alternator - The alternator is loaded by supplying a pure resistive load, applied by ten automobile head lights. This load is applied in increments such that the effects of various alternator loads on the ADS can be evaluated.

(e) Power Steering - The power steering pump is loaded by installing an adjustable needle valve in the discharge line. In this manner, power steering loads can be varied readily from zero to maximum capacity.

(f) Air Conditioning Compressor - Since it is very difficult to accurately determine actual compressor horsepower, due to the number of variables involved, the A/C compressor load is simulated by a second power steering pump. A power steering pump was selected because loading and load measurement can be readily accomplished.

The loading test setup has been tested and accepted. Simulated accessory loads are steady and repeatable. Alternator loads have been varied from zero to 60 amperes in 5-ampere increments. The power steering pump has been loaded up to 8 horsepower. The second power steering pump simulating the air conditioning compressors has also been loaded to 8 horsepower. All other accessories are being loaded in the normal operating manner.

Planned activity will include actual load testing of both the mechanical and hydromechanical VRB systems on the engine dynamometer test rig.

2.6 Test Vehicle Instrumentation

Test instrumentation has been installed in the Mustang II test vehicle. This instrumentation will be used to accurately monitor engine operation (fuel consumption and performance) and accessories operation (output and performance).

The following instrumentation is now installed in the test vehicle:

	Range	
Pressures	Bars	Psig
Manifold pressure	0 to 1	(0 to 15)
Air dump pressure	0 to 1	(0 to 15)
Water pump suction	1 to 0 to 2	(15 to 0 to 30)
Water pump discharge	0 to 2	(0 to 30)
Water pump bypass	0 to 2	(0 to 30)
Spare gauge	0 to 7	(0 to 100)
A/C suction	1 to 0 to 2	(15 to 0 to 30)
A/C suction throttle inlet	0 to 21	(0 to 300)
A/C discharge	0 to 28	(0 to 400)

	Range	
Pressures	Bars	Psig
Spare	0 to 69	(0 to 1000)
P/S discharge	0 to 103	(0 to 1500)
W/P bypass flow (ΔP)	0 to 7	(0 to 100 in. H_2O)
W/P total flow (ΔP)	0 to 14	(0 to 200 in. H_2O)

	Range	
Temperatures	°C	°F
A/C inlet	-18 to 150	(0 to 300)
A/C discharge		
Radiator inlet		
Radiator outlet		
P/C discharge		
Radiator upper tank (left)		
Radiator lower tank (left)		
Radiator lower tank (right)		
Bottom radiator hose		
Top radiator hose		
Engine bypass		

Electrical, dc	
Volts	0 to 15
Amps	0 to 100

In addition to the above described steady-state instrumentation, the following transient data instrumentation will be installed in the test vehicle:

Parameter	Range	Accuracy	Min. Response (-3 dB)
Engine Speed	400 - 6000 rpm	2% F.S.	5 cps
V.R.B. Output Speed	400 - 4000 rpm	2% F.S.	5 cps
Vehicle Speed	10 - 70 mph	3% F.S.	1 cps
Pressure	0 - 1200 psig	2% F.S.	30 cps
Time Pulse	-	-	-

This transient data will be monitored by a portable four channel recorder. This transient data will be used to thoroughly evaluate the "installed" performance of the VRB systems in the test vehicle and their impact on vehicle fuel economy.

2.7 Emissions and Fuel Economy Procedure

In an effort to perform a thorough series of vehicle tests on emissions and fuel economy, Automotive Testing Laboratories (ATL) has been contacted to run the Federal Test Procedure (FTP) and the EPA highway economy test on the 1975 Mustang II test car. These series of tests will be run with and without an Accessory Drive System (ADS) installed. The purpose for performing these tests is as follows:

(a) Verify that the test car's emissions controls are functioning properly.

(b) Determine vehicle fuel economy without ADS installed.

(c) Determine whether ADS effects vehicle emissions.

(d) Determine improvement in vehicle fuel economy with ADS installed.

(e) Compare vehicle computed fuel economy from carbon balance test with measured fuel economy as computed by "Fluidyne" fuel flowmeter.

These series of tests will be completed during the month of February.

2.8 Compact Vehicle Fuel Economy Analysis

The fuel economy analysis for the Mustang II baseline vehicle, with and without the optimum accessory drive system, was completed and the results compiled in Table I. These results represent an in-depth study that considered a variety of total vehicle operating envelopes. The operating envelopes include various apportionments of two major variables:

- o Driving Mode
- o Accessory Loading

The driving modes or cycles are defined in detail in Table II. The selected driving cycles are currently in use throughout the automotive industry and governmental testing agencies. The driving cycles may be divided into three discrete groups that are not necessarily comparable but represent nationwide driving conditions.

- o Federal, city, and EPA highway cycles
- o SAE urban, suburban, and interstate cycles
- o Constant velocity segments

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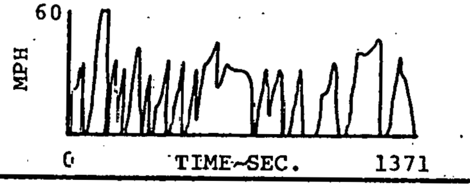
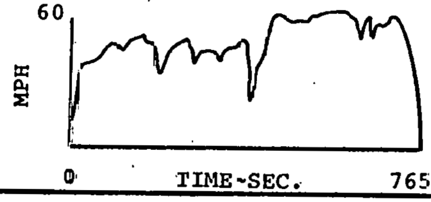
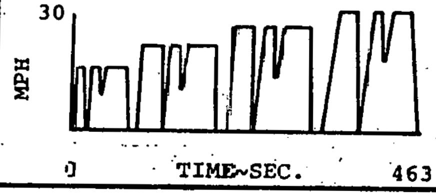
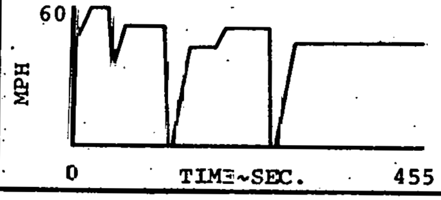
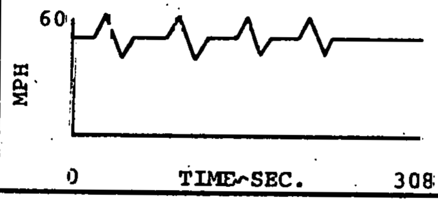
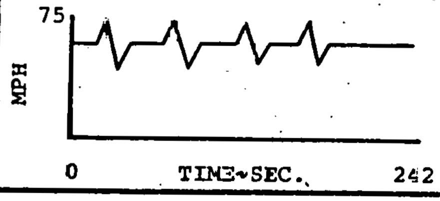
TABLE I

FUEL ECONOMY ANALYSIS

Driving Cycle		Fuel Economy - MPG: Standard Accessory Drive, A/C OFF	Fuel Economy - MPG: Standard Accessory Drive, A/C ON	Fuel Economy - MPG: Constant Speed Accessory Drive, A/C OFF	Fuel Economy - MPG: Constant Speed Accessory Drive, A/C ON	Percent Improvement With CSD, A/C OFF	Percent Improvement With CSD, A/C ON
Federal (FDC)		16.12	14.54	16.45	14.77	2.05	1.58
EPA Highway Fuel Economy Test (HWFET)		21.81	19.72	23.44	21.46	7.37	8.82
SAE Urban		13.14	11.83	13.35	12.08	1.60	2.11
SAE Suburban		19.95	17.85	21.36	19.40	7.07	8.68
SAE Interstate (55 mph)		21.19	19.17	23.06	21.22	8.82	10.69
SAE Interstate (70 mph)		16.91	15.44	19.27	17.94	13.96	16.19
Constant Velocity Segments (mph)	40	25.54	22.65	26.65	23.77	4.35	4.94
	50	22.94	20.57	24.74	22.46	7.85	9.19
	60	19.97	18.15	22.07	20.38	10.52	12.29
	70	17.40	16.07	19.62	18.32	12.76	14.00

TABLE II
DRIVING MODES

DRIVING CYCLE	GRAPHIC DISPLAY	AVERAGE SPEED MPH	CYCLE LENGTH MILE	STOPS PER MILE	PERCENT TIME @ DRIVING CONDITIONS				MAXIMUM ACCEL. RATE (Ft/Sec²)
					IDLE	ACCEL.	DECEL.	CRUSING	
FEDERAL DRIVING CYCLE (FDC LA-4)		19.6	7.45	2.01	14.7	29.9	24.9	30.5	4.84
EPA HIGHWAY FUEL ECONOMY TEST (HWFET)		48.5	10.242	0.098	1.0	23.7	14.5	60.8	4.69
SAE URBAN		15.6	2.01	3.98	13.2	11.7	15.5	59.6	7.00
SAE SUBURBAN		41.1	5.195	0.38	2.6	11.9	10.6	74.9	7.00
SAE 55 MPH INTERSTATE		55.0	4.70	0	0	17.8	17.8	64.4	1.00
SAE 70 MPH INTERSTATE		70.0	4.70	0	0	17.8	17.8	64.4	1.00

Accessory load conditions for each driving cycle, air conditioning mode and accessory drive type are presented in Table III. All driving cycle fuel economy results are obtained for "ON" and "OFF" air conditioning modes of operation. The air conditioning load profile used in the Federal and SAE urban driving cycles represents the power required to achieve heat-soaked vehicle cooling or slow-speed operation in heavy traffic. For the EPA highway, SAE suburban and constant velocity segments, the air conditioning is matching the steady-state cooling requirements imposed by vehicle velocity. The stock engine cooling system fan is modulated via flexible blades. The constant-speed system uses the same fan with de-pitched blades to provide equivalent cooling at higher fan speeds. Air conditioning "ON" operation requires additional alternator loads caused by the air conditioning blower motor and electric clutch operation and other miscellaneous electrical devices. The air conditioning compressor for the constant-speed accessory-drive system must be driven at a higher speed ratio than the stock system (1.43 versus 0.92) to achieve equivalent cooling capacity.

Data presented in Table I shows fuel economy for each particular driving cycle but is not sufficient to determine a gross, yearly average. SAE report 750003 contains current, nationwide, experimental data on driving speeds and conditions and presents a relationship between "miles driven" and driving speeds. The data, reproduced in Figure 17, may be used in conjunction with SAE driving cycles shown in Table II to establish a composite driving cycle that represents all driving conditions for a 10,000 mile vehicle year. Distribution of driving conditions on a yearly basis is:

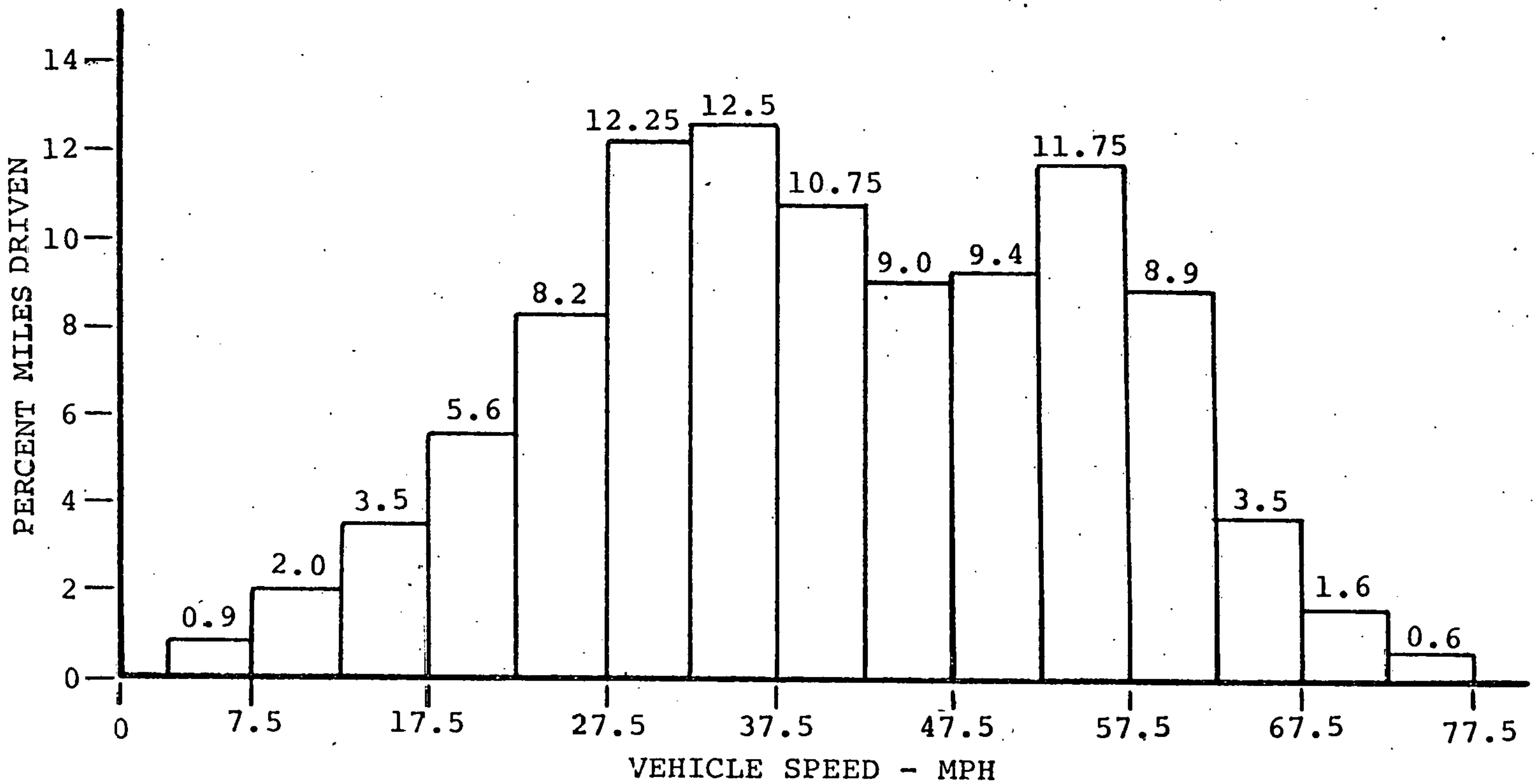
- o 18.2 percent - urban
- o 35.5 percent - suburban
- o 40.7 percent - interstate (55 mph)
- o 5.6 percent - interstate (70 mph)

TABLE III

ACCESSORY SYSTEM SIMULATION CONDITIONS

MODEL CODE NO.	TYPE ACCESSORY DRIVE SYSTEM	DRIVING CYCLE	A/C MODE	A/C DRIVE RATIO	FAN USED	ALTERNATOR OUTPUT (AMPS)
1	STOCK	ALL	OFF	0.92	STOCK	5
2	STOCK	FDC URBAN	MAXIMUM PULL DOWN	0.92	STOCK	30
3	STOCK	HWFET SUBURBAN INTERSTATE	MATCH STEADY STATE LOAD	0.92	STOCK	30
4	SPEED LIMITED	ALL	OFF	1.43	DE-PITCHED STOCK	5
5	SPEED LIMITED	FDC URBAN	MAXIMUM PULL DOWN	1.43	DE-PITCHED STOCK	30
6	SPEED LIMITED	HWFET SUBURBAN INTERSTATE	MATCH STEADY STATE LOAD	1.43	DE-PITCHED STOCK	30

DATA SOURCE: SAE REPORT NO. 750003



PERCENT MILES IN SPEED BANDS-NATIONWIDE

FIGURE 17

Air conditioning usage for the entire year has previously been established at 50 percent in Progress Report 74-310860(7). The yearly fuel savings in gallons is 39 or 6.84 percent, while the increase in fuel economy is 7.34 percent. Additional increases associated with improving accessory operating efficiency and reduced engine size are not represented in this analysis.

2.9 Hydromechanical Drive System Dynamic Analysis

Objectives

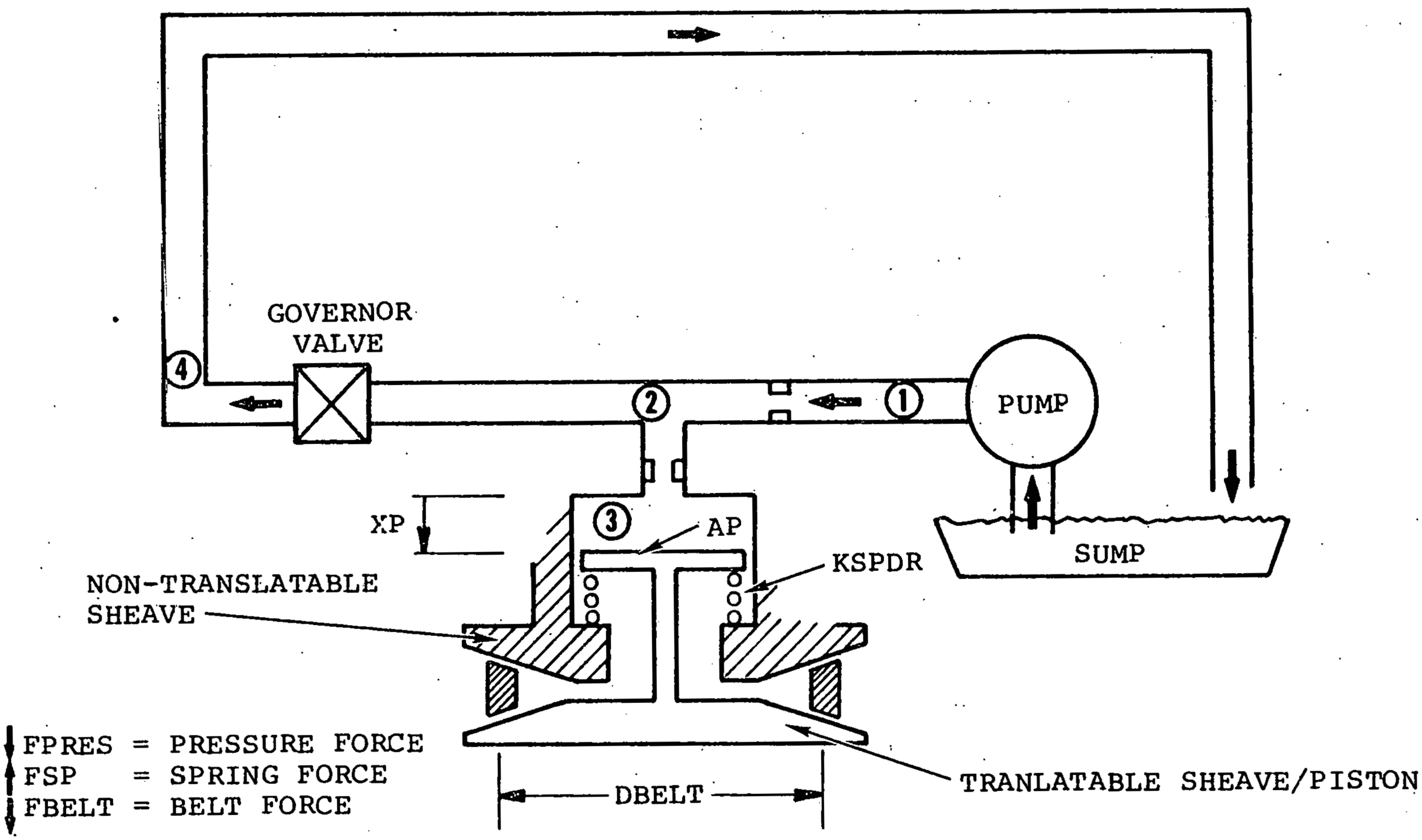
A computer model of the hydromechanical drive was formulated for purposes of:

- Investigating both dynamic and steady-state control characteristics and evaluating proposed design changes.
- Aiding in understanding potential control instabilities and investigating means of improving stability.

Program Description

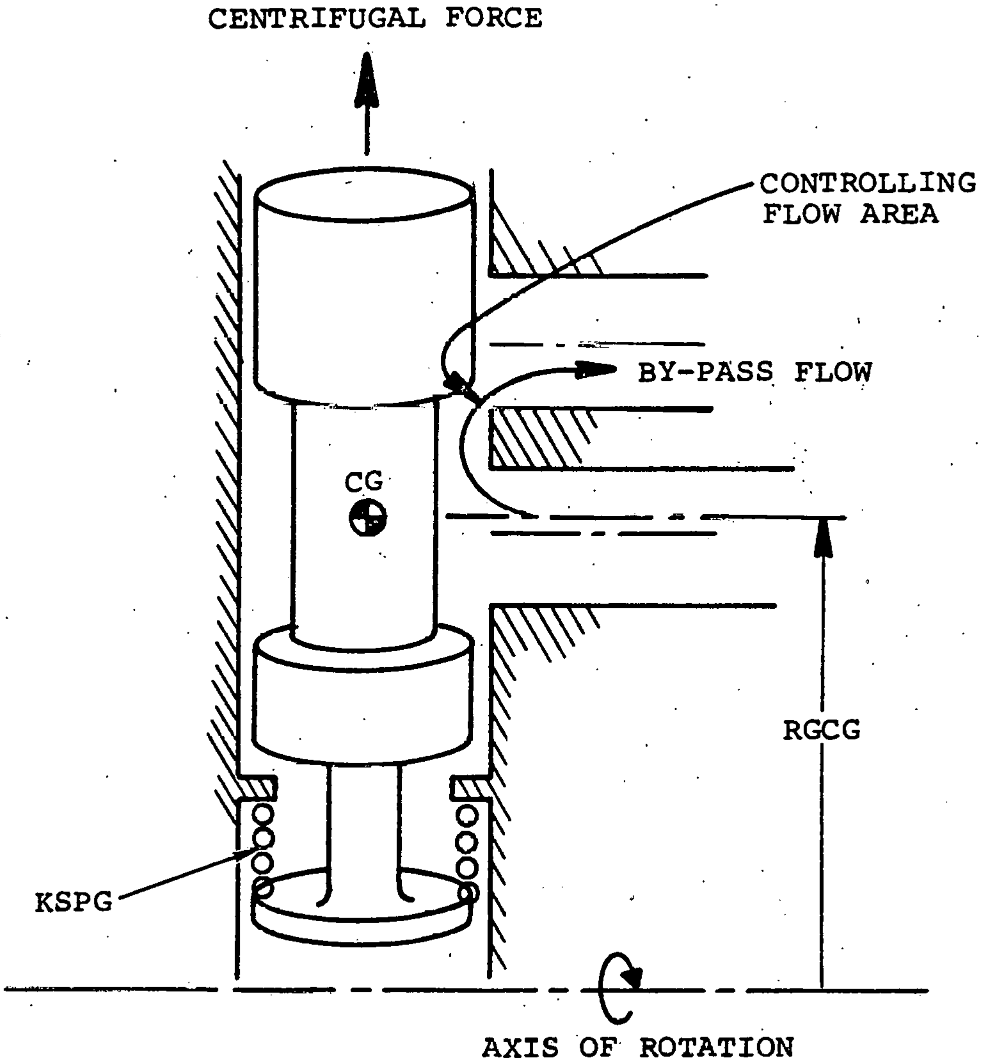
The program is coded in a computer language (JMIMICE) designed specifically to solve simultaneous differential and integral equations such as those encountered in closed loop controls. Input to the program is crank (engine) speed and output is the accessory driving pulley speed.

The system is described in the general schematic diagram of Figure 18. The pump maintains a hydraulic pressure on a piston attached to one of the sheaves. A high pressure tends to spread the sheaves, allowing the belt to run at a small diameter, thus offsetting a low crank speed. The governor valve (Figure 19) modulates as a function of drive speed to control the pressure on the piston.



SCHEMATIC OF HYDRAULIC SYSTEM

FIGURE 18



SCHEMATIC OF GOVERNOR VALVE

FIGURE 19

The model calculates pump flow rate as a function of outlet pressure (P_1). Flow rates and pressures in Regions 1, 2, and 3 (Figure 18) are calculated using the Bernoulli flow equation, assuming representative orifice sizes. Leakage around the piston and fluid compressibility are simulated. The translatable sheave position, and thus the belt diameter, is calculated based on the pressure force on the piston, the belt force, and the spring constant (KSPDR). The belt diameter in the crank pulley is then determined with the known drive pulley diameter and belt length.

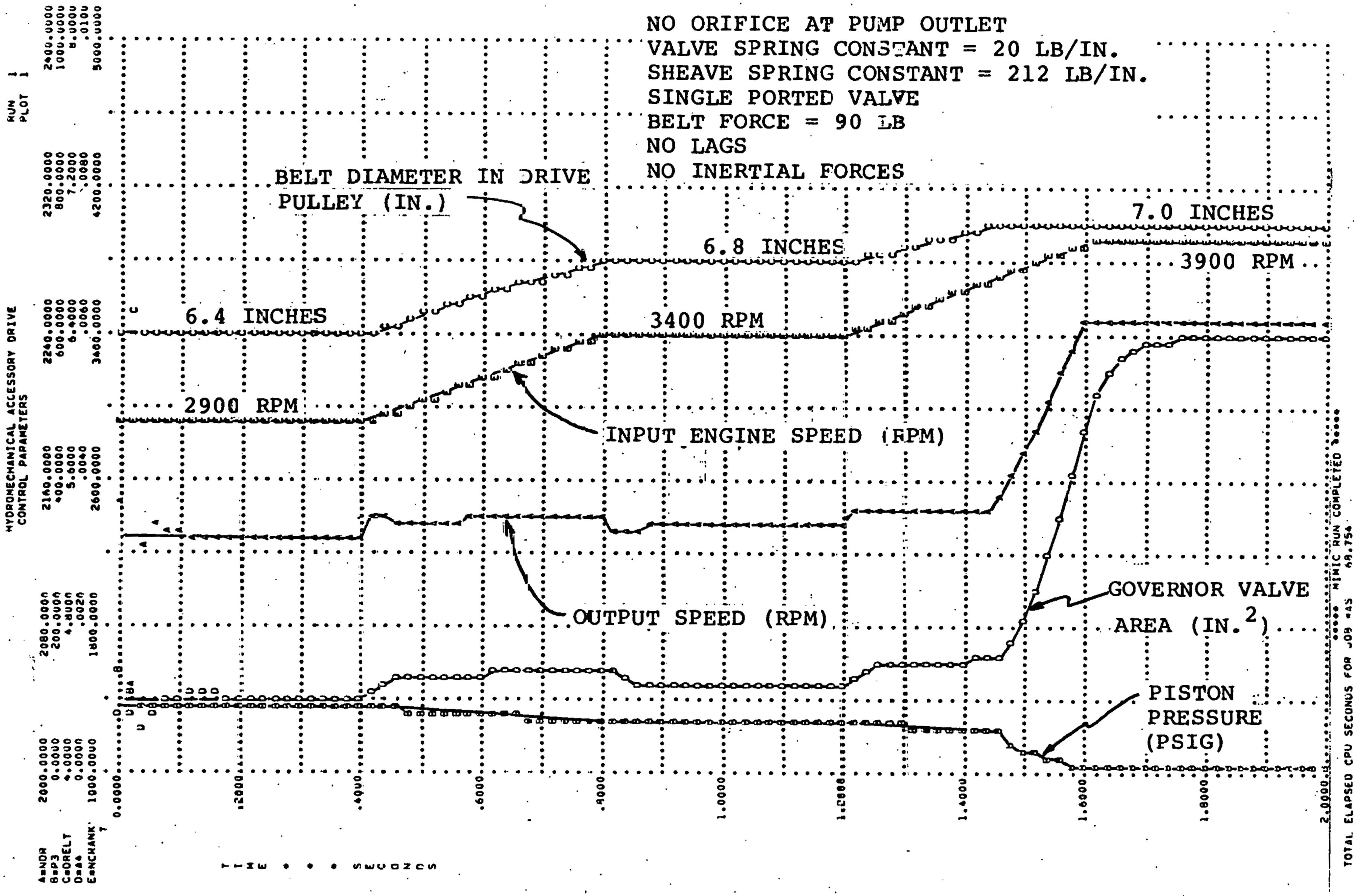
The governor valve open area is determined based on radial position. Centrifugal force on the valve tends to compress the valve spring (Figure 19), opening the valve and relieving the hydraulic pressure acting on the actuator.

Results

A plot for a typical computer run is shown in Figure 20. Input crank pulley speed was varied from 2900 to 3900 rpm as shown. When the crank speed increased from 2900 to 3400 rpm, the output drive speed increases by 6 rpm. The 3900 rpm crank speed exceeded the drive control range, i.e., the drive sheaves were completely closed. Three additional computer runs were made; (1) inserting an orifice between the pump and the piston, (2) double porting the governor valve, and (3) increasing the valve spring constant. The resulting plots were similar to those of Figure 20, with the difference being small and as anticipated.

In addition, computer runs were made using first order lags to simulate belt radial movement in the crank and drive pulleys. No instabilities have thus far been detected using the computer model.

The basic system model is complete and operational. Additional model refinements are planned when experimental data becomes available.



PERFORMANCE OF HYDROMECHANICAL DRIVE

FIGURE 20

APPENDIX I

APPENDIX I

IMPROVING AUTOMOTIVE FUEL ECONOMY WITH ACCESSORY DRIVES

PRESENTED AT ERDA/TEC
CONTRACTORS COORDINATION MEETING
ANN ARBOR, MICHIGAN
NOVEMBER 18, 1975
CONTRACT NO. E(04-3)-1095

This appendix is comprised of visual aid replicas and accompanying narrations presented to ERDA/TEC.

INTRODUCTION (FIGURE 1)

This presentation summarizes AiResearch progress on the Automotive Accessory Drive Program funded by ERDA/TEC.

IMPROVING AUTOMOTIVE FUEL ECONOMY WITH ACCESSORY DRIVES

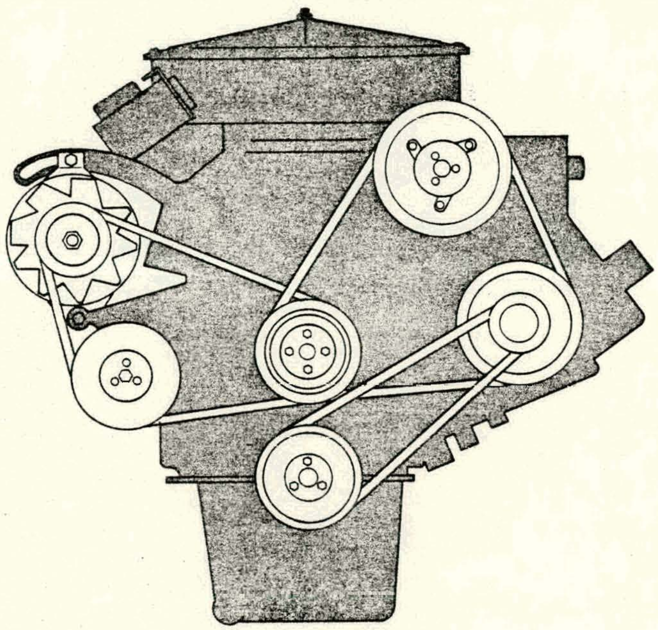


FIGURE 1

(FIGURE 2)

The program objective is to minimize automotive engine-driven accessory power consumption, thereby improving overall vehicle fuel economy.

Although the primary goal is fuel economy improvement, the manner of accomplishment must be acceptable to both manufacturer and customer. Factors such as first cost, maintainability, and impact on existing capital investment are significant considerations in the selection process.

OBJECTIVE

- MINIMIZE ACCESSORY POWER CONSUMPTION AND MAXIMIZE OVERALL VEHICLE FUEL ECONOMY

APPROACH

DEVELOP VARIABLE RATIO BELT DRIVE

- VERIFY FUEL ECONOMY IMPROVEMENT WITH A SIMULATION MODEL OF BASELINE VEHICLE
- DEMONSTRATE THE ACCESSORY DRIVE SYSTEM IN THE BASELINE VEHICLE

FACTORS CONSIDERED

- DRIVE EFFICIENCY/ FUEL ECONOMY
- INITIAL COST
- COMPLEXITY
- MAINTENANCE COST
- NOISE
- INSTALLATION FLEXIBILITY
- WEIGHT
- DURABILITY

IMPROVING AUTOMOTIVE FUEL ECONOMY
WITH ACCESSORY DRIVES

FIGURE 2

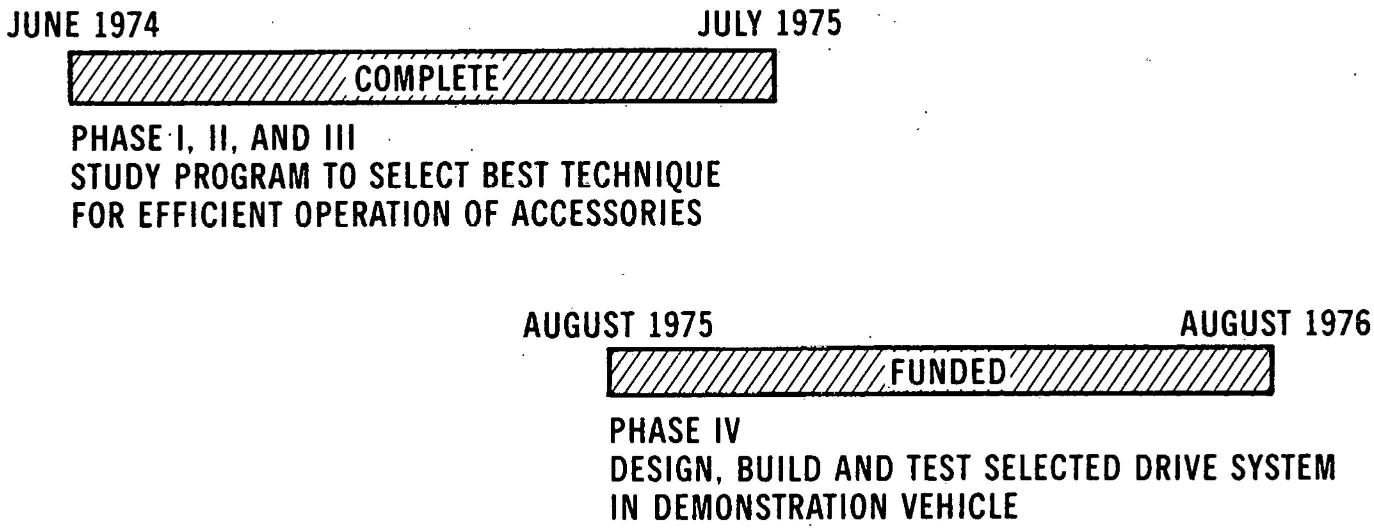
(FIGURE 3)

The accessory drive program was initiated in June, 1974, with a 12-month study directed toward determining the optimum approach to reducing power consumed by engine-driven accessories, and anticipated fuel economy if the selected approach were applied to near-term production automobiles.

Basic study conclusions were:

- Clipping accessory speed at a selected level will reflect worthwhile fuel economy improvements.
- A variable ratio belt is the most appropriate mechanism for clipping or speed limiting.

A follow-up development and demonstration program was initiated in July, 1975 and will extend through July, 1976. This hardware-oriented phase will involve detail design fabrication and variable ratio belt accessory drive development, accessory system installation in the baseline vehicle, and economy improvement verification.



PROGRAM TIMING AND OBJECTIVE

FIGURE 3

(FIGURE 4)

Decided advantages gained through accessory speed control, in addition to improved fuel economy, are listed below:

- Accessory optimization improving efficiency by reducing accessory speed ranges.

- Accessory power reductions permit equal performance with a reduced-size engine.

- Possible elimination of certain components (fan clutches, power steering oil coolers, and flow limiters).

- Allows overdrive of certain components for improved function at low vehicle speeds.

  - Reduced air conditioning cool-down time
  - Increased alternator output
  - Improved engine cooling

- REDUCE ACCESSORY COST
- REDUCE POLLUTION
- 5-10% IMPROVED FUEL ECONOMY
- 5-10% SMALLER ENGINES
- LESS NOISE
- IMPROVED ACCESSORY EFFICIENCY
- LESS WEAR
- LESS MAINTENANCE
- IMPROVED ACCESSORY LIFE

ACCESSORY DRIVE ADVANTAGES

FIGURE 4

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

(FIGURE 5)

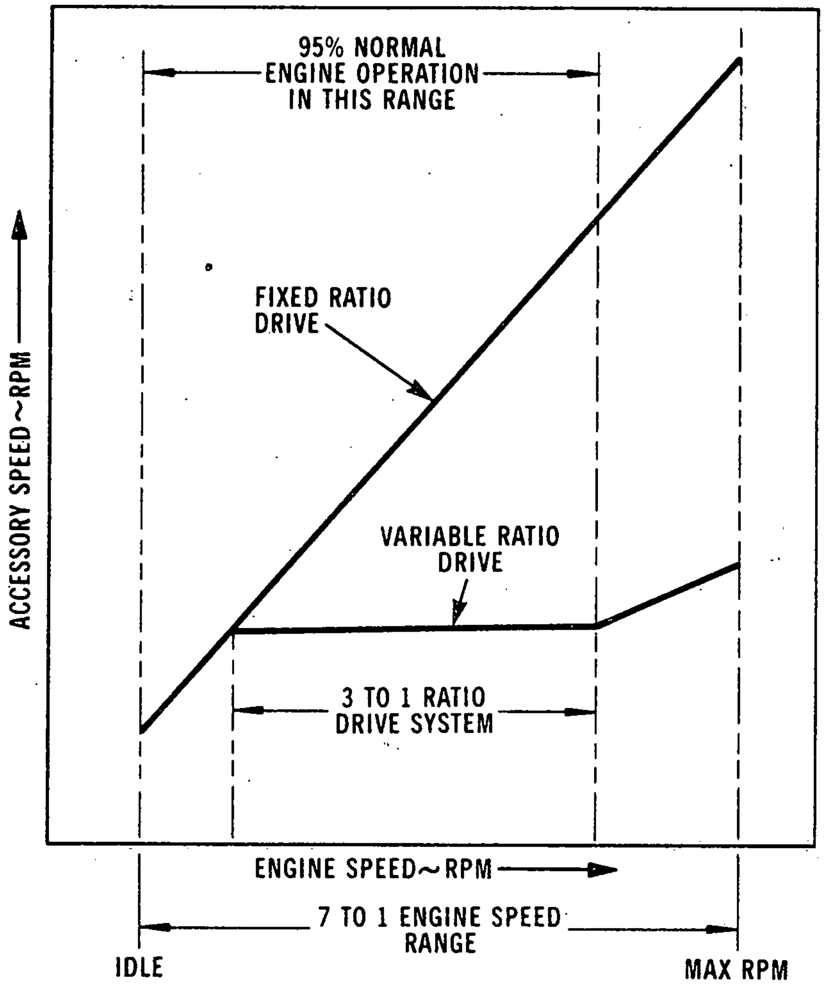
This figure reflects the engine accessory speed limiting concept. Conventional automobiles operate accessories at a fixed ratio-to-engine speed (upper curve). A form of limited speed accessory operation is shown in the lower curve. To achieve the characteristic shown, it is necessary to interpose a drive mechanism, having a ratio change capability of 3 to 1, between the crank shaft and the accessories.

The control or break point onset location, the ratio range and degree of overdrive, if any, prior to break point, are variables that can be preselected to achieve particular objectives.

To achieve maximum fuel economy, the range must be a minimum of 3 to 1, the break point must be low (just sufficient to support the worst case load), and the overdrive minimal.

If the objective were to increase a system output, such as air conditioning, at lower speeds, the overdrive would be increased and a higher break point would be used.

By proper break point, overdrive, and control range selection, a salable combination of economy improvement and accessory system function is achievable.



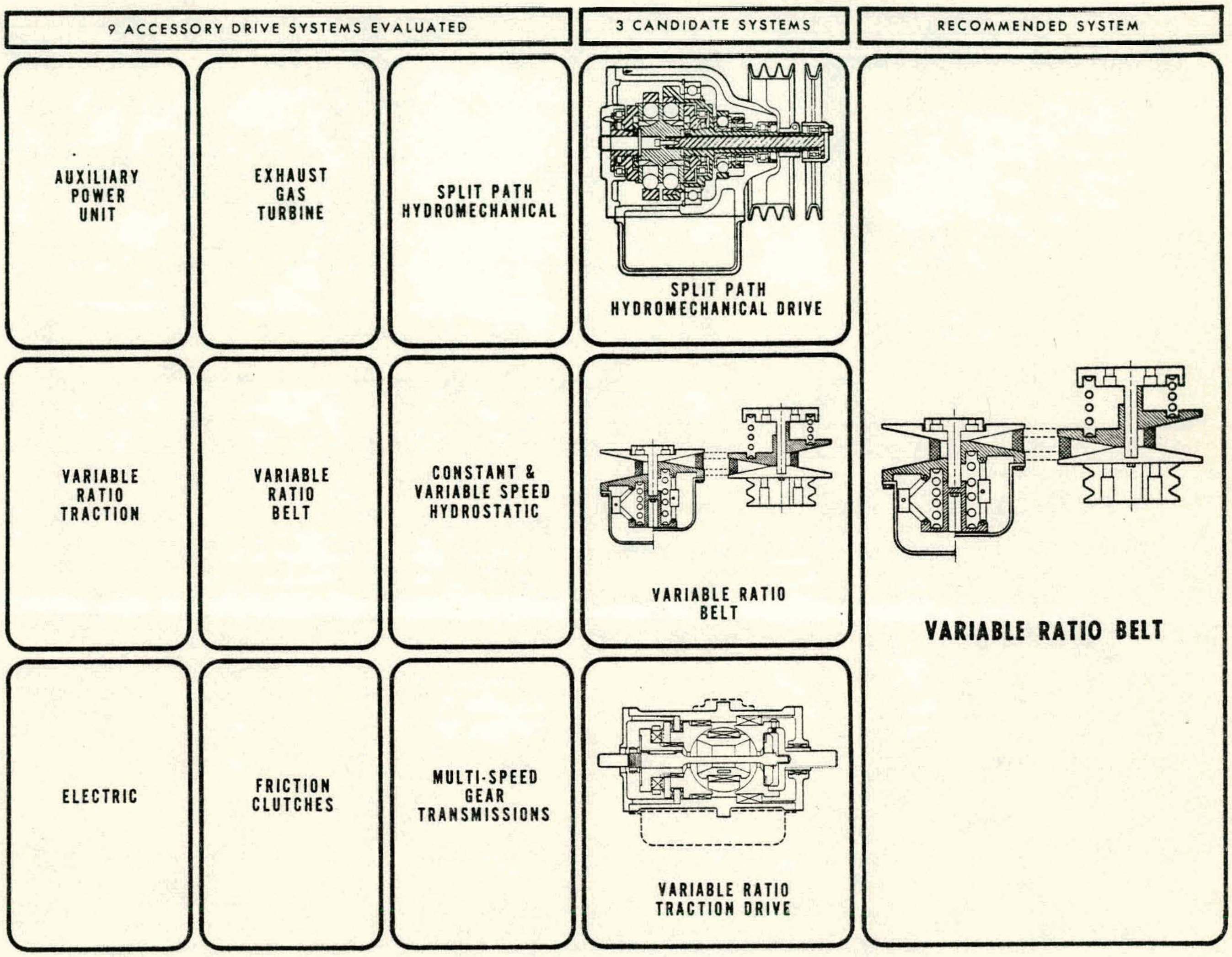
TYPICAL SPEED LIMITING EFFECT

FIGURE 5

(FIGURE 6)

Nine specific systems were evaluated during the study program. The variable ratio belt mechanism was selected as the prime concept for the following reasons:

- Potential for high power transfer efficiency
- Low development risk
- Adequate ratio capability
- Low cost
- Compatible with existing accessories and engine compartment arrangements
- Minimal retraining and equipment for field maintenance required



CANDIDATE SELECTION MATRIX

FIGURE 6

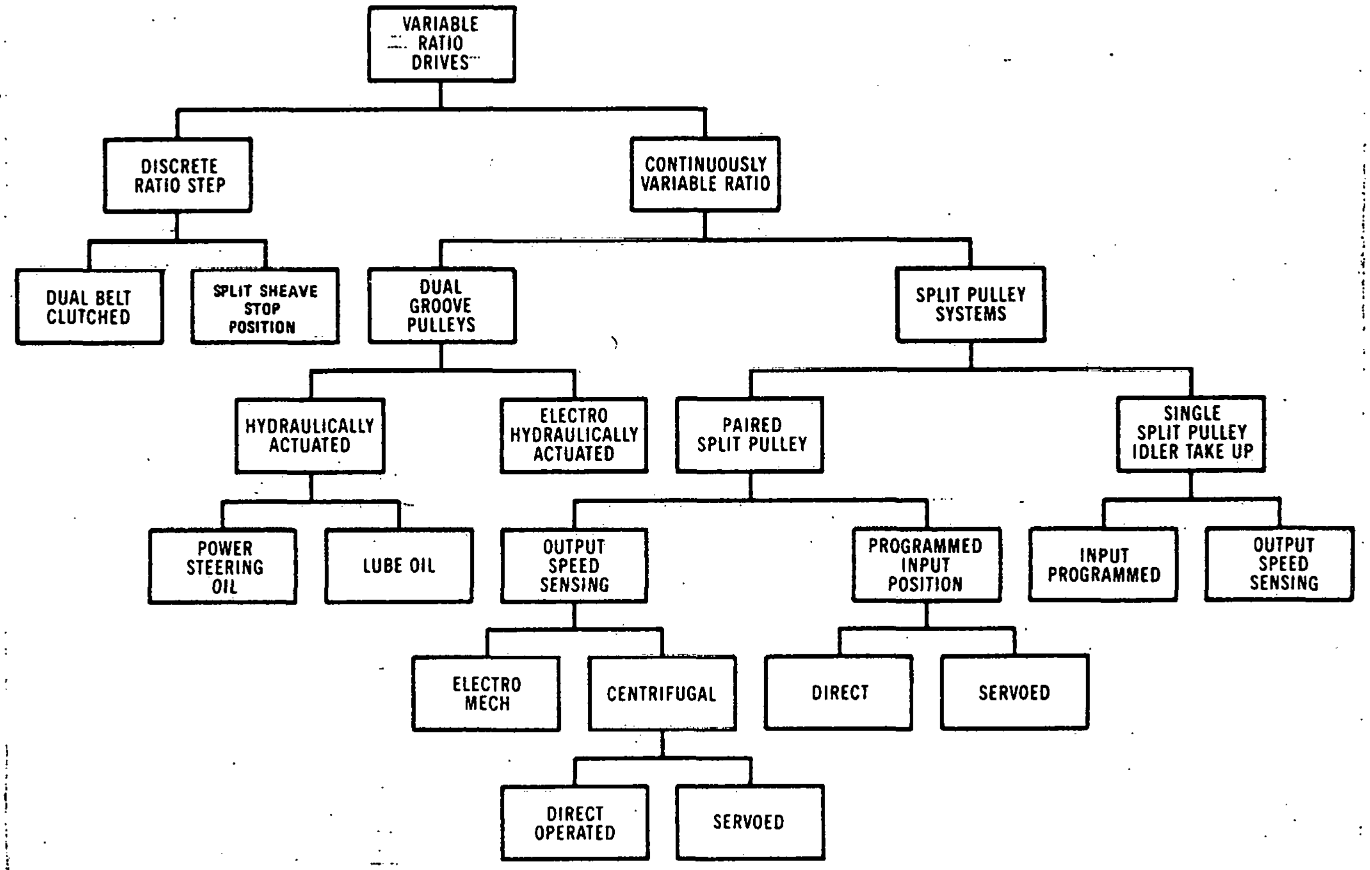
(FIGURE 7)

Selecting the variable ratio belt transmission as the prime concept did not complete the tradeoff study. Figure 7 reflects configuration and control options for this drive class. Configuration types include dual groove or floating center sheaves, paired split sheaves, and single split sheave with idler take-up. Speed control choices range from the apparent simplicity of direct flyweight to electrohydraulic.

Evaluating the range of options, with respect to program objectives, resulted in selecting the following two concepts for hardware development:

- Direct flyweight control
- Hydromechanical servo control

Each system uses a single drive belt operating between paired split sheaves.



TYPES OF VARIABLE SPEED DRIVES

FIGURE 7

(FIGURE 8)

The two drive concepts will be evaluated in the laboratory and on the vehicle during the development program.

Development hardware has been designed and fabricated. Laboratory testing started during the third week in November.

Both systems are configured to fit in the test vehicle with minimum rework of existing vehicle equipment.

Vehicle test program results will be analyzed for accessory drive drive concepts, installation problems, load matching, belt life, fuel economy and relative advantages of each technique.

- TWO CONTROL CONCEPTS
  DIRECT FLYWEIGHT CONTROL
  HYDROMECHANICAL SERVO CONTROL
- DESIGN AND FABRICATE DEVELOPMENT HARDWARE FOR BOTH CONCEPTS
  - DESIGN FOR DEVELOPMENT FLEXIBILITY
  - MUSTANG II INSTALLATION
  - OPERATE STOCK ACCESSORIES
- EVALUATE BY BENCH, ENGINE AND VEHICLE TEST
  - FUEL ECONOMY
  - COMPLEXITY/COST
  - INSTALLATION FLEXIBILITY
  - MAINTENANCE/DURABILITY

DEVELOPMENT PROGRAM

FIGURE 8

(FIGURE 9)

During the study program, it was found desirable to specify a baseline vehicle to calculate definitive performance and economy data. It is more important for the development program that a representative test vehicle be used for proper accessory drive system evaluation.

Primary considerations in selecting the test vehicle were weight, performance, and equipment availability representative of expected near-term trend of contemporary automobiles.

One factor strongly favoring the Mustang II, is availability of an in-line four cylinder, and V6 and V8 cylinder engines. This engine selection allows accessory drive demonstrations over a range of engine configurations in the same basic vehicle.

A Mustang II, 3-door model with a V6 engine was purchased for the initial development work.

CRITERIA

- WEIGHT AND PERFORMANCE SIMILIAR TO EPA COMPACT CAR SPECIFICATION AVAILABLE WITH BROAD RANGE OF ENGINE AND ACCESSORY OPTIONS
- ACCESS TO ENGINEERING DATA
- HIGH PROBABILITY OF CONTINUED PRODUCTION REASONABLE SALES VOLUME

MUSTANG II SELECTED

TEST VEHICLE DESCRIPTION

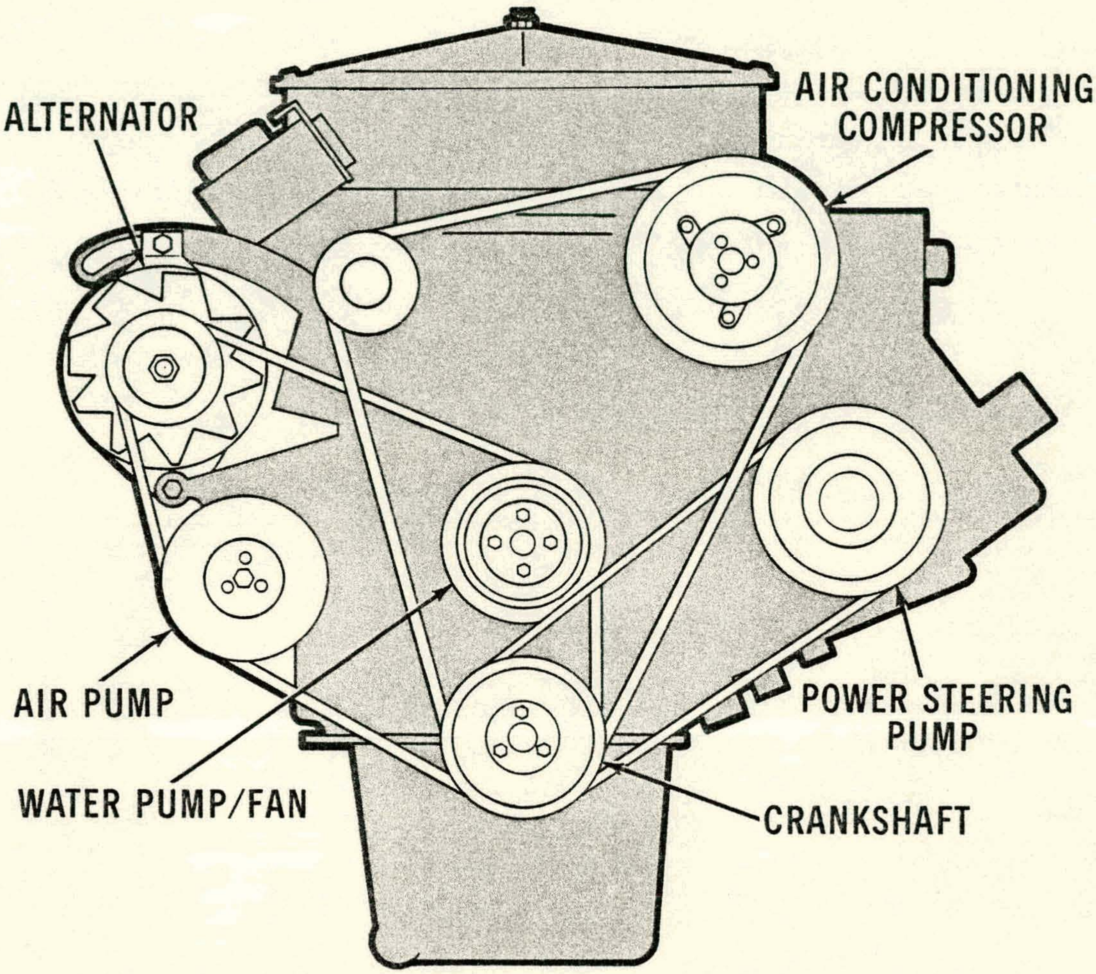
1975 MUSTANG II HATCHBACK	N/V = 50 RPM/MPH
2800 CC V6 ENGINE	0-60 MPH — 15 TO 17 SEC
3 SPEED AUTOMATIC TRANSMISSION	CURB WEIGHT - 3050#
A/C AND P/S	

ACCESSORY DRIVE PROGRAM TEST VEHICLE

FIGURE 9

(FIGURE 10)

The stock accessory locations for the Mustang II, V6 installation are shown. There are three crankshaft-driven belts arranged to drive six accessories.



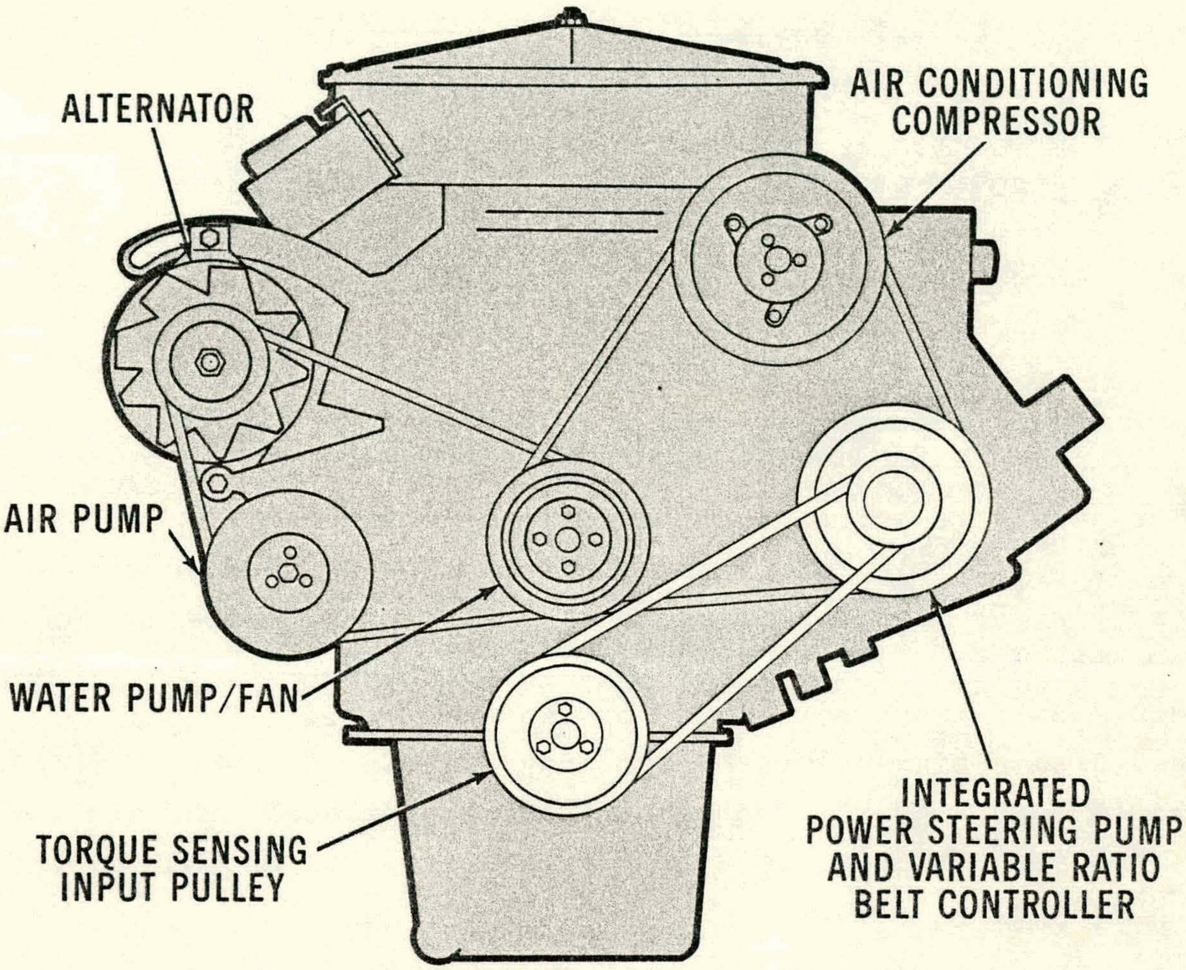
STOCK ACCESSORY INSTALLATION

MUSTANG II

FIGURE 10

(FIGURE 11)

The hydromechanically-controlled variable-ratio-belt installation is shown. The output pulley assembly, which includes the ratio control unit and power steering pump in a single assembly, mounts in place of the stock power steering pump. The input pulley is crank mounted. All other accessories remain in the stock locations, and are belt-driven from a fixed-pitch pulley on the ratio controller.



HYDROMECHANICAL ACCESSORY
DRIVE INSTALLATION

FIGURE 11

(FIGURE 12)

Hydromechanical system development hardware cutaway views are shown. The upper unit includes a moveable and a fixed sheave, a sheave actuator, a cross-axis governor valve, support bearings, a single-element dual-output pump, and a fluid reservoir.

This assembly provides two basic functions:

- Automatic sheave position control required to maintain constant output speed.

- Power steering system flow source.

The sheave position, and therefore output speed, is regulated by modulating a control pressure applied to the single-ended sheave actuator piston. Control pressure is generated by metering pump pressure through a variable area orifice controlled by the cross-axis governor spool. Governor speed-sensing and metering-function are obtained from a single port by locating a spring-referenced spool in a cross-axis bore with the center of gravity displaced from the rotational axis. The mechanization is similar to one type of automatic transmission governor.

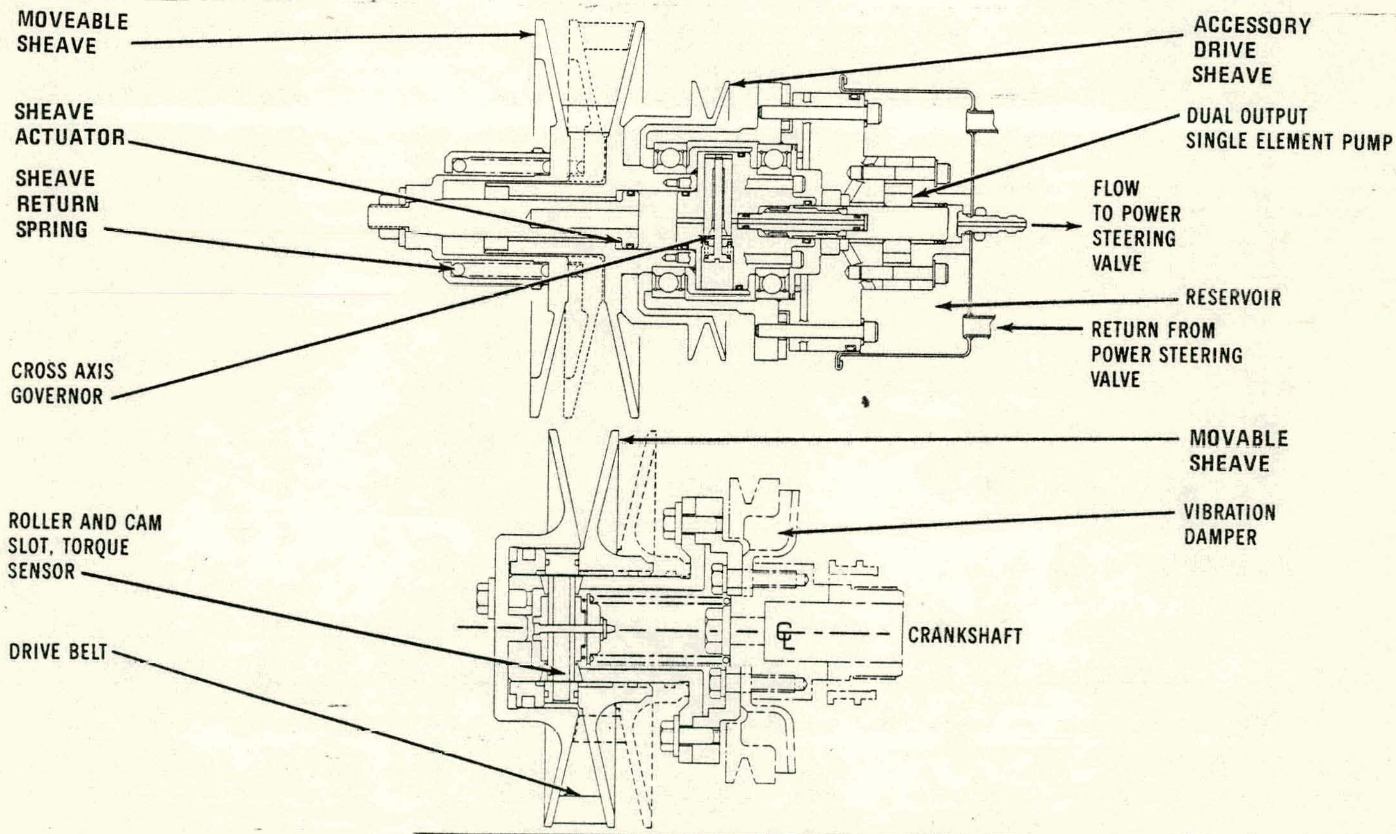
The flow required to operate the speed-control servo is approximately 15 percent of the power steering requirement. This flow is obtained by separately porting out the trapping volume that occurs in the main pump sealing mesh. Normal pump output is used for power steering requirements.

The technique of obtaining dual isolated outputs from a single element pump was developed at Garrett for gas turbine lubrication systems. It is an attractive scheme for the accessory drive system in that it allows generation of two flows that can float independently in pressure level with a single-stage pump.

The package was designed such that external plumbing, reservoir location, and size and fluid maintenance requirements are identical to a conventional power steering system.

The driver pulley, shown in the lower view, is bolted to the stock vibration damper assembly and consists of a fixed and a moveable sheave. The moveable sheave senses differential force in the drive belt tight and slack sides, due to transmitted torque, and transfers a percentage of this torque into squeeze via a helical cam ramp incorporated in the hub. This particular design uses self-centering tapered cam rollers to assure equal load distribution and smooth operation.

The torque sensing mechanism adjusts belt tension in proportion to the load, and results in improved efficiency and life.



HYDROMECHANICAL CROSS SECTION

FIGURE 12

(FIGURE 13)

In summary, the important hydromechanical drive features are:

- Closed-loop hydraulic servo sheave positioning operation results in accurate speed control.
- Inherent speed control system stiffness allows use of a torque-sensing mechanism that automatically adjusts total belt loading as a function of accessory loading. The result is improved belt life and higher transmission efficiency.
- Servo polarity can be selected such that the drive will start in maximum underdrive, resulting in reduced starter load.
- Integrated control unit replaces the power steering pump and installs in the same location.
- Output sheave guide lubrication is inherent in the design.
- Control unit bracket mounting simplifies drive belt installation, and allows center distance adjustment for belt wear and manufacturing tolerance compensation.
- Adaptability to other vehicle installations is good. One basic design can be utilized in a large variety of vehicles, with changes restricted to external parts such as brackets and belt lengths.

- ACCURATE, LOW DROOP, SPEED CONTROL
- TENSION CONTROL DECOUPLED FROM SPEED CONTROL ALLOWING MORE FREEDOM IN MATCHING BELT TENSION TO TORQUE ~ ∴ HIGHER EFFICIENCY AND LONGER BELT LIFE
- START UP IN MAXIMUM UNDERDRIVE — REDUCES STARTER LOAD
- PACKAGES IN SAME SPACE OCCUPIED BY POWER STEERING PUMP
- SHEAVE GUIDE LUBRICATION INHERENT IN DESIGN
- ALLOWS ADJUSTABLE CENTER DISTANCE BETWEEN DRIVER AND DRIVEN — TAKE-UP FOR BELT WEAR AND TOLERANCE ∴ LOWER COST BELT
- READILY ADAPTABLE TO OTHER VEHICLES

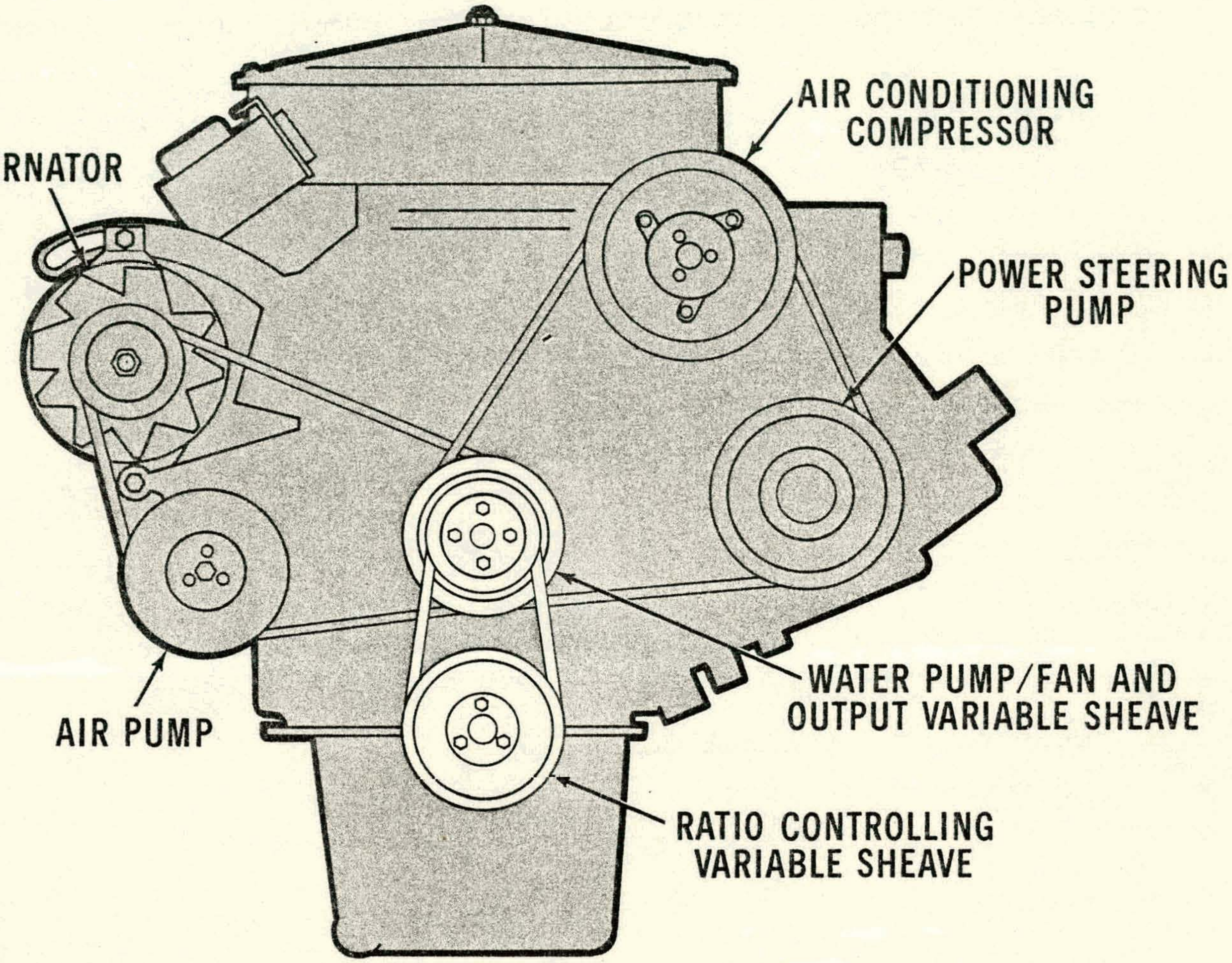
HYDROMECHANICAL DRIVE FEATURES

FIGURE 13

(FIGURE 14)

The direct operated variable ratio belt system is installed on the Mustang II V6 as shown. The flyweight operated control pulley is crankshaft-mounted. The spring loaded output pulley is bearing-mounted to the stock water pump housing.

Center-to-center distance is fixed and established by stock component locations at 5.8 inches. All accessories are located at stock position and are driven by two fixed pulleys at the rear of the output variable pulley.



DIRECT OPERATED ACCESSORY DRIVE INSTALLATION

FIGURE 14

(FIGURE 15)

The input pulley is crankshaft-mounted and contains the speed control mechanism. The output pulley is water pump housing mounted.

The control technique utilized in this drive is input programming. The governor is located at the input pulley and is set to program sheave position as a predetermined function of crankshaft speed. This method of control was selected over an output speed governor because of the more favorable governor operating condition, and because a certain degree of belt tension control is inherent in the design if the take-up sheave is output-shaft mounted.

The control mechanism consists of four flyweights, each connected to a scissors-bar linkage balanced by reference springs. Programmed ratio change, as a function of crank speed, is obtained by balancing the three forces, listed below, acting at the moveable sheave:

- Centrifugal flyweight force
- Reference spring force
- Belt splitting force

The output pulley consists of a fixed sheave and a spring-loaded moveable sheave for belt take-up. The water pump is tang driven from the output sheave, and a fixed pitch fan is mounted at the end of the output shaft.

The drive belt sheave-splitting force enters the control system force balance. This belt-induced force is a direct function of transmitted torque. Acceptable speed regulation requires that the governor output be large, relative to the belt force.

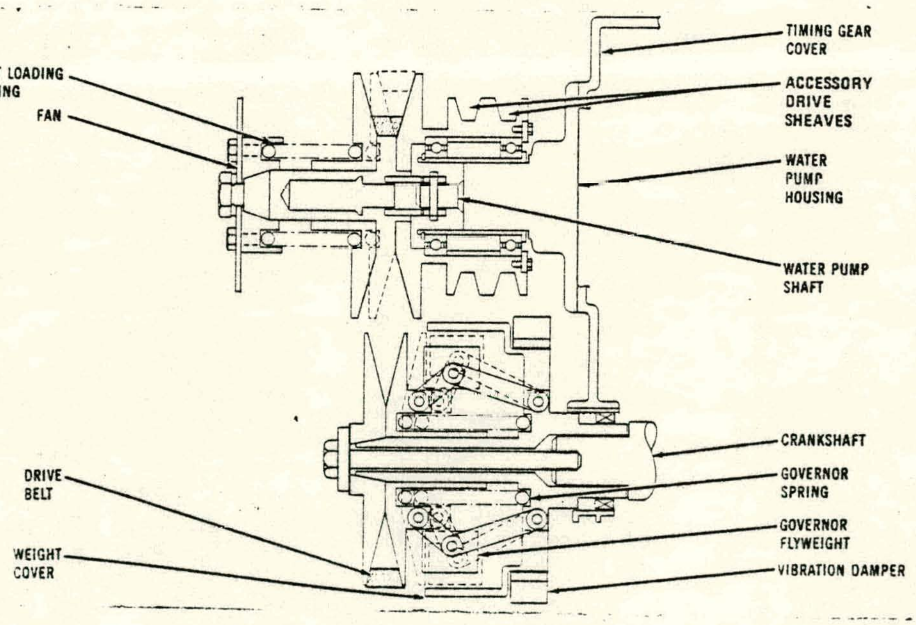
In a direct-operated system, the design must accommodate the variance between maximum and minimum loads with an acceptable speed error.

The device sensitivity to transmitted torque makes it difficult to include a belt tension control feature such as that in the hydromechanical unit.

Although there are some inherent problems with a direct-operated system, potential simplicity presents a strong incentive to pursue development with obtaining an acceptable compromise between belt life and overall accessory system efficiency the objective. Properly mechanized, the mechanical system will reflect some advantage in production cost, but at the sacrifice of some fuel economy gains.

It should be noted that the design shown was configured for development flexibility. Features are included that facilitate rapid investigation of variations in governor weight, linkage ratio, and spring characteristics.

A production version of this concept would include diaphragm springs for improved packaging and torque carry-through from the floating sheave, and dry lubrication or composite plastic sheave guide bearings.



DIRECT OPERATED VARIABLE ACCESSORY DRIVE CROSS SECTION

FIGURE 15

(FIGURE 16)

In summary, the direct-operated variable ratio belt drive features are:

- Direct flyweight control of belt position
- Input programming
- Three-to-one ratio change capability
- Compact crankshaft and water pump locations
- Belt operates continuously at required tension to carry maximum load

DEVELOPMENT HARDWARE

- DIRECT MECHANICAL CONTROL
- FLYWEIGHT GOVERNOR-INPUT PROGRAMMING-TYPE CONTROL
- CONSTANT ACCESSORY SPEED OVER 3 TO 1 ENGINE SPEED RANGE
- FIXED CENTER DISTANCE BETWEEN DRIVER AND DRIVEN SHEAVE
- DRIVE BELT FROM CRANKSHAFT TO WATER PUMP
- DRY LUBE OR GREASEPACK SHEAVE GUIDE DESIGN
- CONSTANT TENSION BELT OPERATION

DIRECT OPERATED VARIABLE RATIO BELT
ACCESSORY DRIVE

FIGURE 16

(FIGURE 17)

The computer program, developed during the study phase, was modified to include Mustang II V6 parameters. This program is sophisticated to the extent that nearly all the drive train characteristics such as transmission shift logic, torque converter characteristics, gear train efficiency, engine map interpolation, and load demand calculation for the significant accessories are included. The program will compute fuel consumption for steady-state speeds and commonly used driving cycles, such as the four SAE cycles and the EPA city and highway cycles.

Data from preliminary check runs were made using the Mustang II input parameters. Correlation with published test data was good.

VEHICLE SIMULATION MODEL

- ROLLING RESISTANCE
- AERODYNAMIC DRAG
- DRIVING CYCLES-CONSTANT SPEED, EPA AND SAE
- A/C COOLING DEMAND LOAD VS VEHICLE SPEED
- INERTIA COMPONENTS MAJOR ROTATING GROUPS
- ENGINE MAP FUEL FLOW, TORQUE, SPEED
- TORQUE CONVERTER EFFICIENCY
- TRANSMISSION SPEED RATIO AND EFFICIENCY
- MANIFOLD PRESSURE SHIFT LOGIC
- REAR AXLE SPEED RATIO AND EFFICIENCY
- SPEED RATIO STOCK AND DRIVE SYSTEM
- STOCK LOAD PROFILES FOR VARYING OUTPUTS
- MODIFIED P/S AND FAN LOAD PROFILES FOR ACCESSORY DRIVE
- A/C DEMAND LOAD MATCHING
- DRIVE SYSTEM EFFICIENCY

CHECK RUN RESULTS

	EPA TEST RESULTS	SIMULATION RESULTS
EPA CITY CYCLE	15.5 MPG	16.1 MPG (+4%)
EPA HWY CYCLE	22.7 MPG	21.8 MPG (—4%)

COMPUTER ANALYSIS OF FUEL ECONOMY

FIGURE 17

(FIGURE 18)

The baseline vehicle computer model was used to predict the fuel economy improvement that can be expected with addition of an accessory drive. The drive parameters used were those associated with the hydromechanical configuration.

- Twenty five-percent overdrive prior to the break point
- Break point at 1200 engine rpm which is approximately 24 mph in third gear
- Three-to-one control range
- An average drive efficiency of 95-percent

The table shows the percent increase for EPA city and highway cycle and constant velocity segments, ranging from 2-percent improvement for the city cycle, to 14 percent at a constant 70 mph.

DRIVING CYCLE		EPA REPORTED MPG	COMPUTER MODEL MPG	PERCENT IMPROVEMENT WITH ACCESSORY DRIVE	
				A/C OFF	A/C ON
FEDERAL (FDC)		15.5	16.1	2.05	1.79
EPA HIGHWAY FUEL ECONOMY TEST (HWFET)		22.7	21.8	7.47	8.82
CONSTANT VELOCITY SEGMENTS (MPH)	40	—	25.54	4.35	4.95
	50	—	22.94	7.85	9.19
	60	—	19.97	10.52	12.04
	70	—	17.50	11.20	14.75

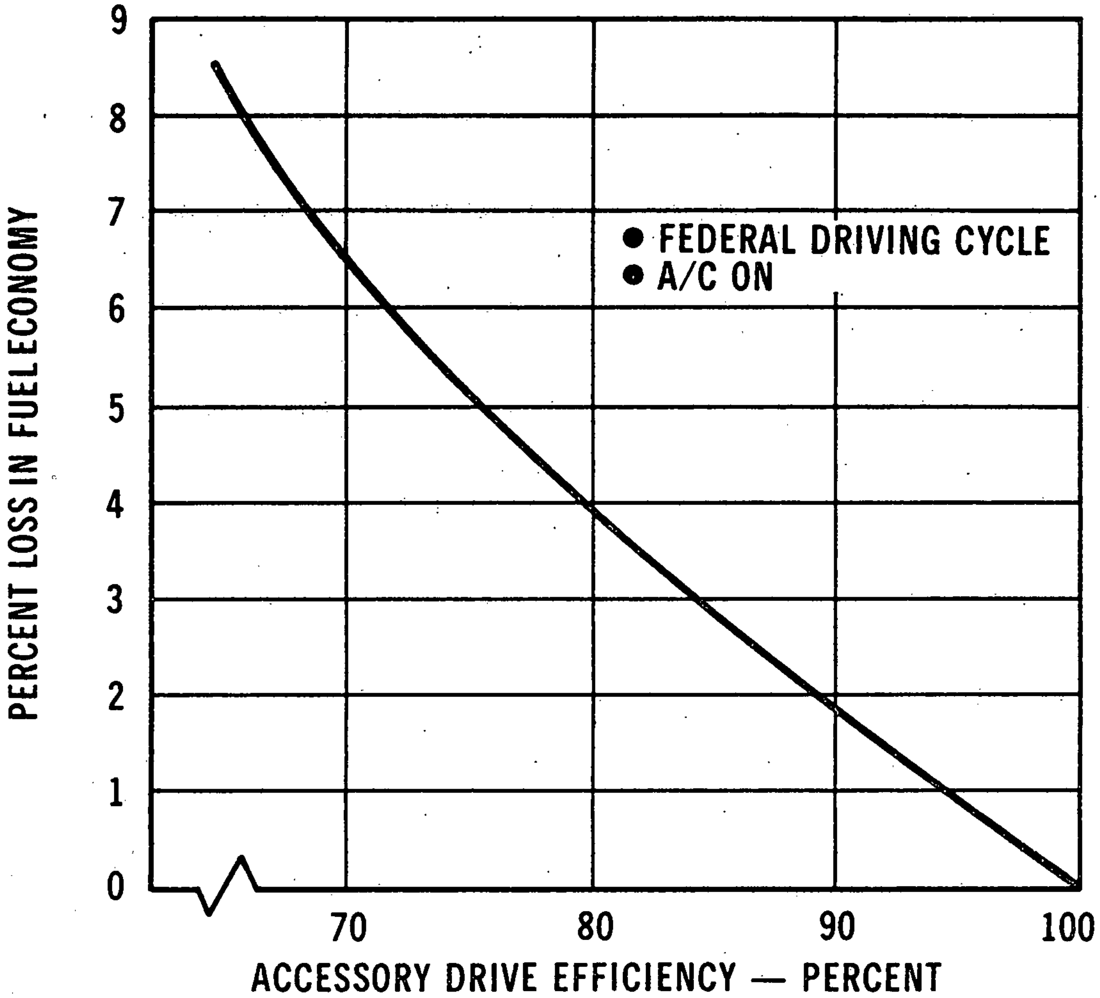
FUEL ECONOMY PREDICTIONS

FIGURE 18

(FIGURE 19)

The computer model prediction of fuel economy to drive efficiency sensitivity for the Federal Driving Cycle is shown. An incremental slope in the expected efficiency range indicates that a 10-percent drop in drive efficiency would result in a 2-percent fuel economy loss.

The importance of high drive efficiency in obtaining maximum economy payoff is clear and emphasizes one of the more critical trade-offs between the two drive mechanizations types.

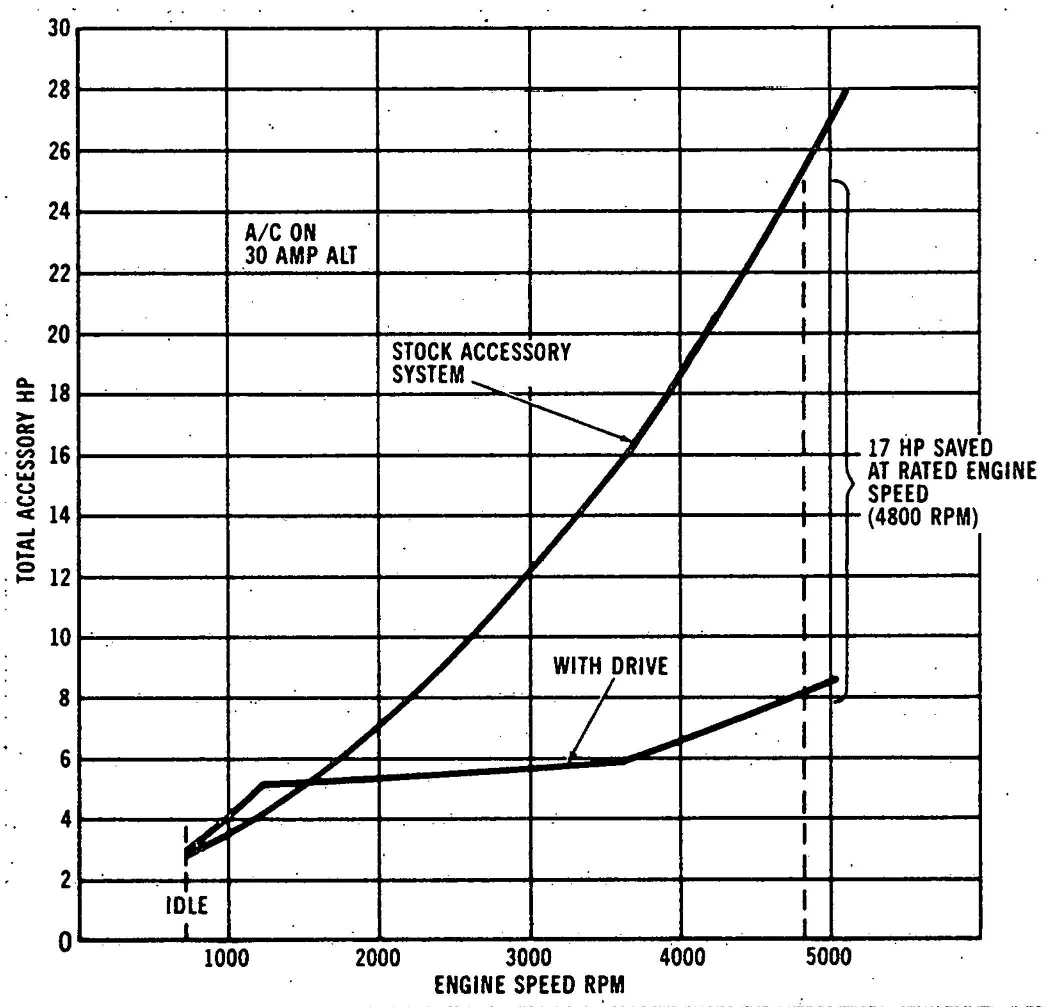


EFFECT OF DRIVE EFFICIENCY
ON FUEL SAVINGS

FIGURE 19

(FIGURE 20)

An accessory drive installation in a vehicle reduces the power required to drive the accessories. If no other change were made, this extra engine power could be used to increase performance. Alternately, for equal performance, the engine could be resized downward for an additional incremental fuel economy increase.



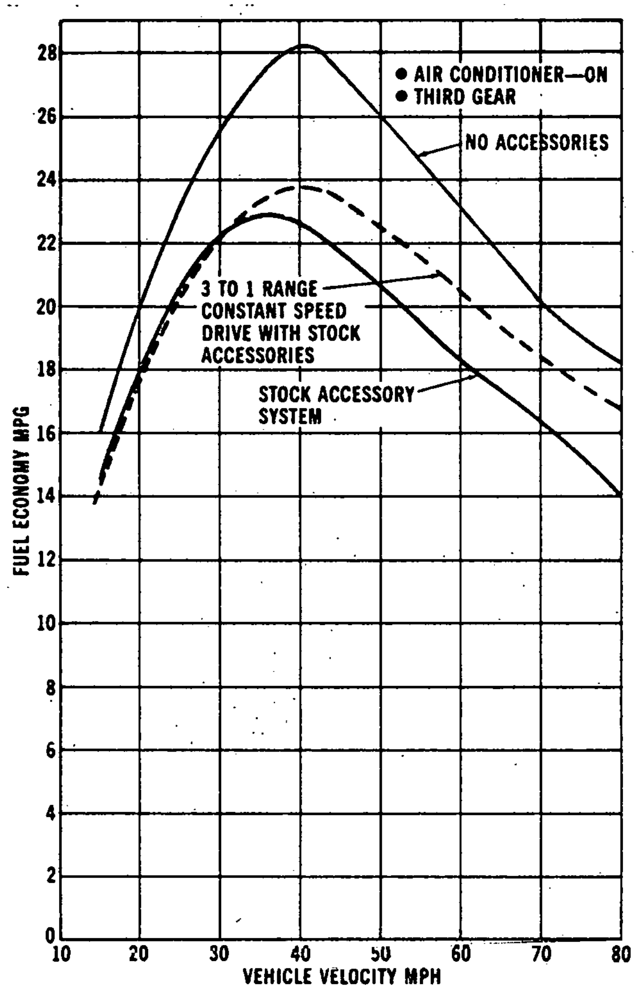
POTENTIAL ENGINE POWER SAVINGS
WITH AN ACCESSORY DRIVE

FIGURE 20

(FIGURE 21)

This figure presents data relative to predicting the maximum fuel economy payoff that can be expected by researching the accessory area.

The obvious maximum is removal of all accessories. As extreme as this case is, it does serve to put an irrefutable upper boundary on the situation. It can be useful in evaluating various accessory schemes. The dotted line indicates the predictions for the 3 to 1 range drive described earlier, operating with stock accessories.



CONSTANT VELOCITY FUEL ECONOMY

FIGURE 21

(FIGURE 22)

Redesign of accessories to take advantage of the limited speed range can result in improved conversion efficiency.

The computer model was modified to predict fuel economy improvements attributed to the compound effects of an accessory drive and a 15-percent reduction in accessory power absorption.

Most low-speed range overdrive losses are recovered and the economy payoff is increased 2 to 3 percent over that obtained from the drive.

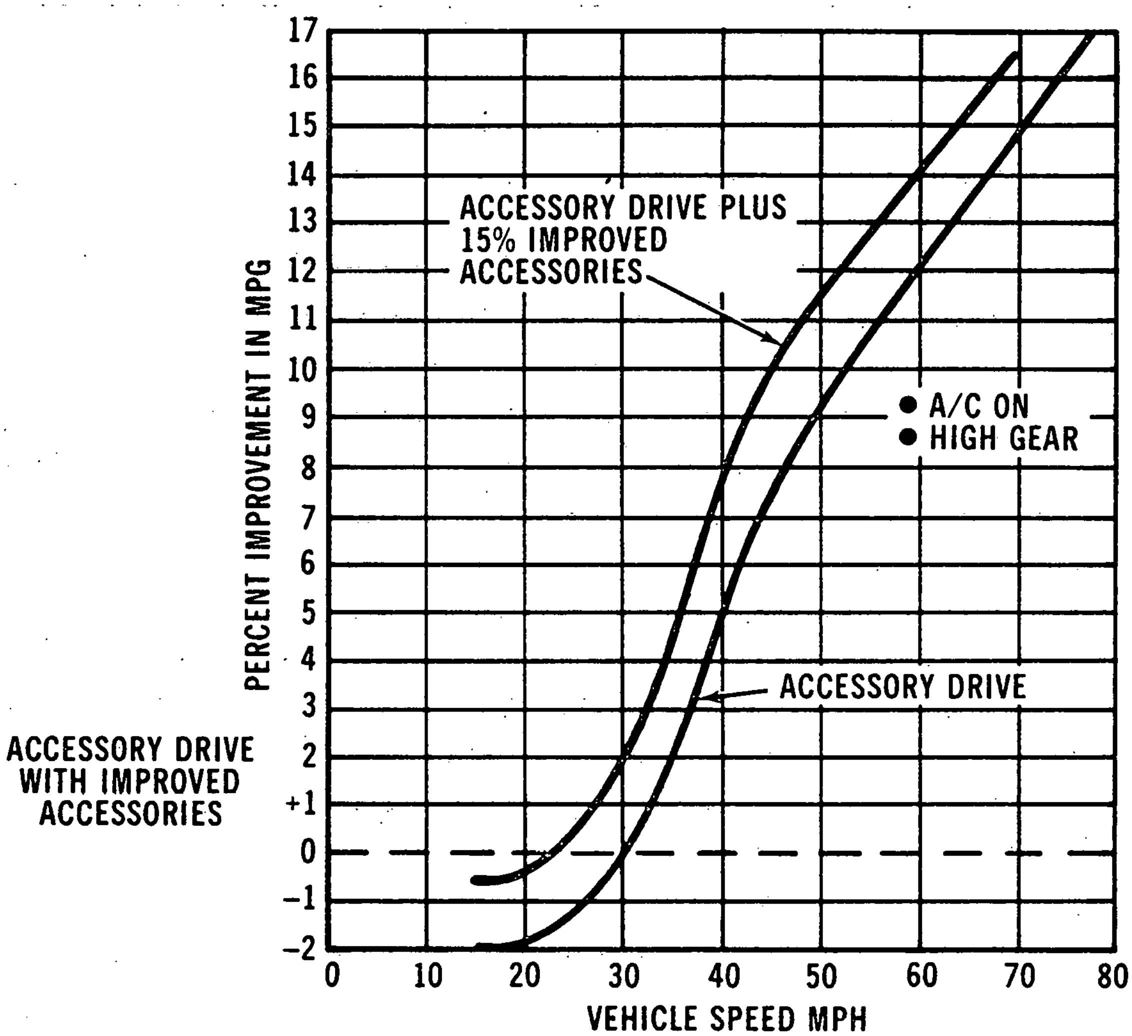

PERCENT IMPROVEMENT IN MPG
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
2
+1
0
-1
-2
ACCESSORY DRIVE WITH IMPROVED ACCESSORIES
ACCESSORY DRIVE PLUS 15% IMPROVED ACCESSORIES
ACCESSORY DRIVE
● A/C ON
● HIGH GEAR
0
10
20
30
40
50
60
70
80
VEHICLE SPEED MPH

FIGURE 22

(FIGURE 23)

Final program status summary:

- The variable ratio belt transmission was selected during the study program as the optimum near-term solution to automotive accessory drives
- Two types of variable ratio belt drives will be investigated during the development program
  - Closed loop hydromechanical control
  - Direct mechanical control
- Development hardware design for both approaches is complete
- Development hardware fabrication is complete
- All components required for drive installation in the test vehicle are complete
- A baseline car has been selected, procurred and instrumented
- Stock vehicle test data required for drive sizing and design has been obtained. This includes:
  - Cooling and air conditioning system performance
  - Power steering and reactor pump load impedance
  - Electric fan cooling capacity
- Accessory test rig complete and in operation. Performance mapping complete on power steering pump and commercial 8-hp constant-speed drive
- Spare V6 engine and accessories procurred. Dyno checkout runs in process

- VARIABLE RATIO BELT SELECTED AS OPTIMUM NEAR TERM SOLUTION TO AUTOMOTIVE ACCESSORY DRIVE.
- BELT CONTROL METHODS REDUCED TO TWO APPROACHES:
  - CLOSED LOOP HYDROMECHANICAL CONTROL
  - DIRECT MECHANICAL GOVERNOR CONTROL
- DESIGN OF HYDROMECHANICAL AND MECHANICAL SYSTEMS COMPLETE
- FABRICATION OF DEVELOPMENT HARDWARE FOR BOTH SYSTEMS COMPLETE
- DESIGN AND FAB OF COMPONENTS REQUIRED FOR INSTALLATION OF SYSTEMS IN BASELINE VEHICLE COMPLETE (BELTS, BRACKETS, PULLEYS)
- BASELINE CAR HAS BEEN SELECTED, PROCURRED AND INSTRUMENTED 1975 MUSTANG II V-6
- STOCK VEHICLE DATA HAS BEEN DOCUMENTED:

COOLING SYSTEM PERFORMANCE	REACTOR PUMP LOAD LINES
POWER STEERING LOAD LINES	ELECTRIC FAN COOLING CAPACITY
	A/C SYSTEM PERFORMANCE

- ACCESSORY TEST RIG COMPLETE AND IN OPERATION, PERFORMANCE MAPPING COMPLETE ON POWER STEERING PUMP AND COMMERCIAL 8 HP CONSTANT SPEED DRIVE
- SPARE V-6 ENGINE AND ACCESSORIES PROCURRED, DYNO CHECKOUT RUNS IN PROCESS

PROGRAM STATUS

FIGURE 23