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MASTER

FOURTH QUARTERLY

PROGRESS REPORT

STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

74-310860(12) July 30, 1975

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STUDY ON REDUCTION OF ACCESSORY HORSEPOWER REQUIREMENTS

74-310860(12) July 30, 1975

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ATTACHMENTS: L-3621560 (3 sheets)
Appendix I

REV	BY	APPROVED	DATE	PAGES AND/OR PARAGRAPHS AFFECTED
12	RJH	See Title Page	7-30-75	Fourth Quarterly Progress Report

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PHOENIX, ARIZONA

FOURTH QUARTERLY PROGRESS REPORT STUDY ON REDUCTION OF ACCESSORY HORSEPOWER REQUIREMENTS

1.0 INTRODUCTION

This is the fourth quarterly technical progress report submitted in accordance with requirements outlined in Attachment B, Reports and Reviews, Energy Research and Development Administration (ERDA) E[04-3]-1095. This report covers the period from April 1, 1975 through July 15, 1975.

The program was initiated in July, 1974, under Contract 68-03-2119, for the Environmental Protection Agency, Advanced Automotive Power Systems Division. The contract defines an automotive accessory drive development program consisting of three design phases and an optional fabrication test phase. Phase I, Conceptual Design, and Phase II, Preliminary Design, have been completed. Phase III, Detail Design, has been completed, and Phase IV, Fabrication and Test, is outlined herein. ERDA will direct the program through the Advanced Automotive Power Systems (AAPS) Division (formerly with EPA). The new contract number is E[04-3]-1095.

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 three program phases established concept feasibility, determined potential fuel savings, and selected a drive system design for concept mechanization. Phase IV will carry the program through prototype fabrication, bench tests, engine tests, and vehicle tests. The final program objective will be a detail drive mechanization 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 involves accessory performance, integration, and design improvements associated with limited speed drive.

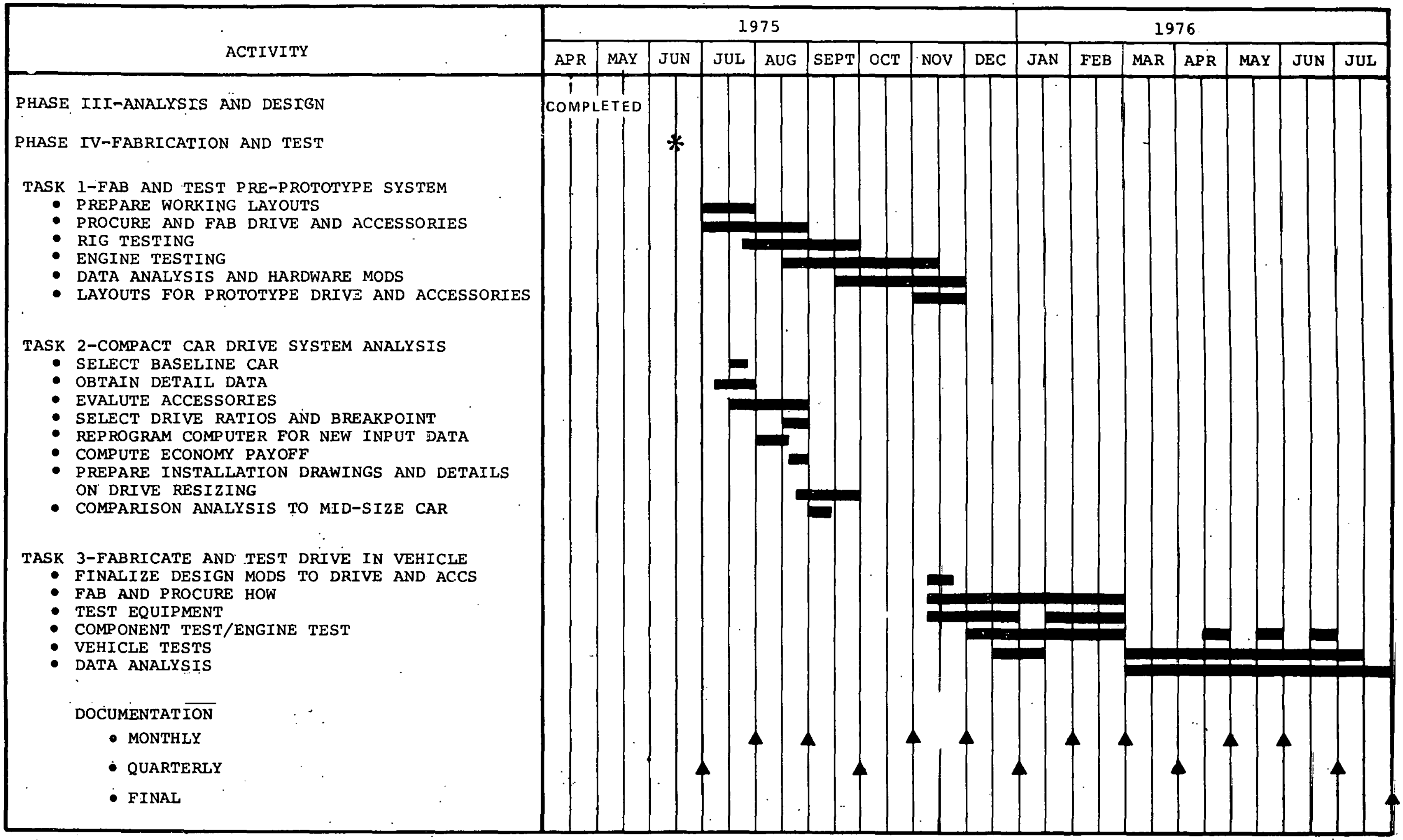
Phase IV consists of three primary development tasks, from the analysis and design definition of Phase III to a functional prototype system evaluated in a baseline vehicle. The three tasks comprising Phase IV are listed below, with scheduling as shown in Figure 1.

Task 1 - Design, fabricate, component test, and laboratory engine test a pre-prototype drive and selected pre-prototype advanced accessories.

Task 2 - Computer modeling and computer drive-cycle analysis of a compact car utilizing an accessory drive system and advanced accessories. Modify drive system design, as required, to adapt to the baseline compact car.

Task 3 - Design, fabricate, component test, engine test, and vehicle system test a prototype drive and prototype advanced accessories. Finalize system analysis, drive design, accessory modification recommendations, and prepare the total development program final report.

Analysis and design modifications for a baseline compact vehicle, in Task 2, will be derived from work completed on the Nova intermediate size baseline vehicle in Phase III. Baseline compact vehicle selection will be mutually determined by ERDA and AiResearch in the initial segment of Task 2.



* Phase IV initiation, June 16, 1975

AUTOMOTIVE ACCESSORY DRIVE SYSTEM DEVELOPMENT

PHASE IV

MASTER PROGRAM SCHEDULE

FIGURE 1

The baseline vehicle utilized in Task 3 will be the compact car selected for Task 2. The vehicle will be equipped with the engine, accessories, and drive train options as mutually determined by ERDA and AiResearch.

1.1 Content

This quarterly report encompasses the following major accomplishments:

- Selection of candidate belt-drive concepts
- Completion of Phase III baseline vehicle drive systems physical and operational envelopes
- Completion of analysis for a mechanically controlled direct-operated belt drive with input programming
- Completion of design layout and analysis for a hydromechanically controlled, servo-operated belt drive with output-speed sensing.

A review of program direction and accomplishments since inception is contained in the technical summary presented as Appendix I of this report. The summary was presented to the ERDA/AAPS Contractors Coordination Meeting on May 6, 1975 at Ann Arbor, Michigan.

2.0 VARIABLE-RATIO BELT CONCEPTS

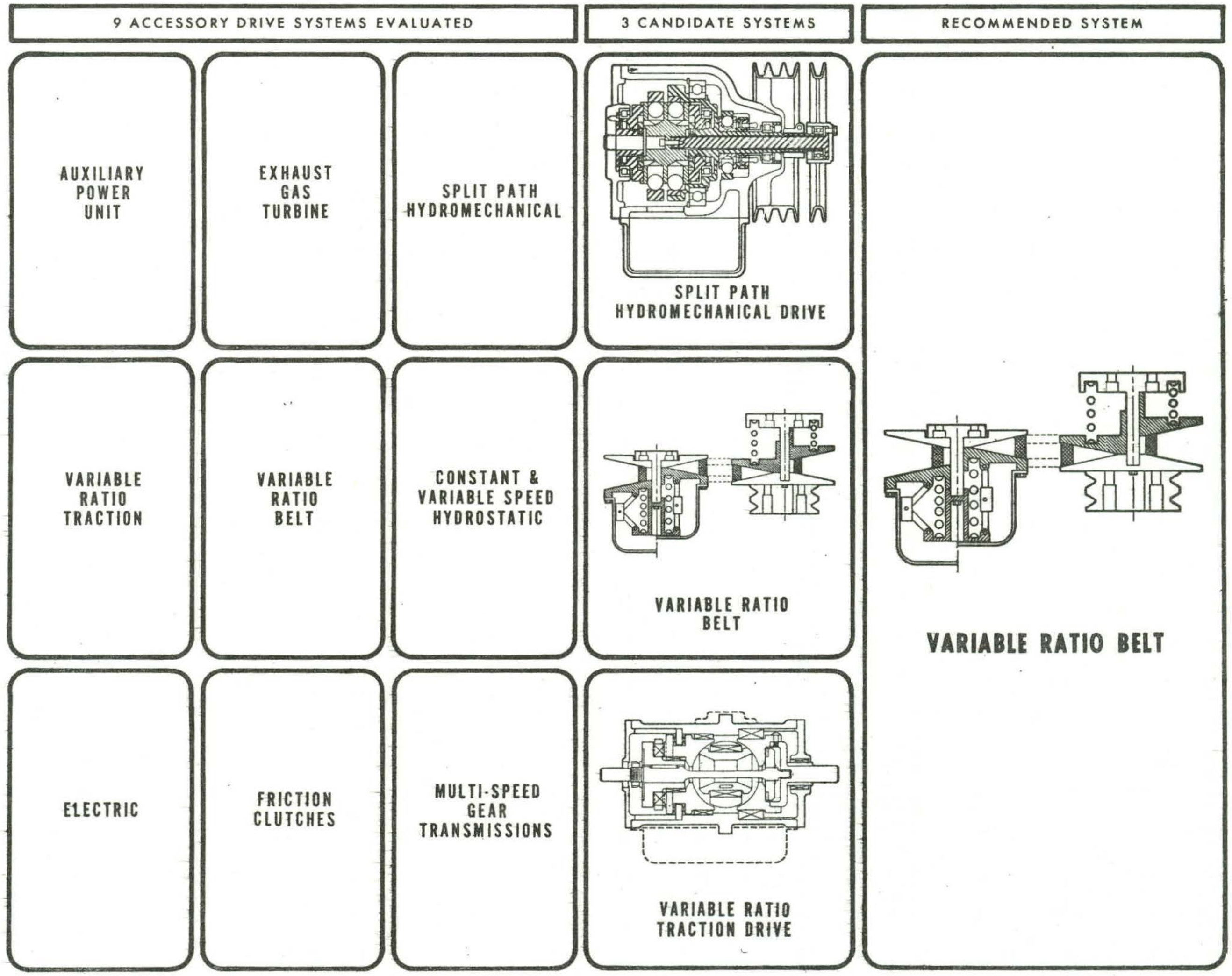
Nine major accessory-drive systems were evaluated. Three systems were selected for additional consideration; split-path hydromechanical, variable-ratio belt, and variable-ratio traction. A trade-off study of the three candidate systems resulted in the variable-ratio belt system being selected as the most practical for near-term automotive applications. The selection process is illustrated in Figure 2.

Primary considerations favoring the variable-ratio belt-drive system are relative simplicity, minimum impact on existing engine compartment layout, and high power-transfer efficiency.

A primary program task was to summarize various methods of mechanizing a variable-ratio belt drive and to analyze each method for practical automotive accessory-drive applications. Several types of variable-speed drives available, within the general classification, are presented in Figure 3. Configurations one through twelve were considered possible drive-problem solutions. A preliminary review of each system was conducted and those with obvious disadvantages were eliminated, as indicated on Figure 3 by cross marks. A summary matrix of selection logic is shown in Figure 4. This process eliminated all but three systems. Option nine, paired split pulley with direct-operated input programming, has been selected as the initial concept. Option eight, servo-operated speed-sensing system, has been selected as a backup design.

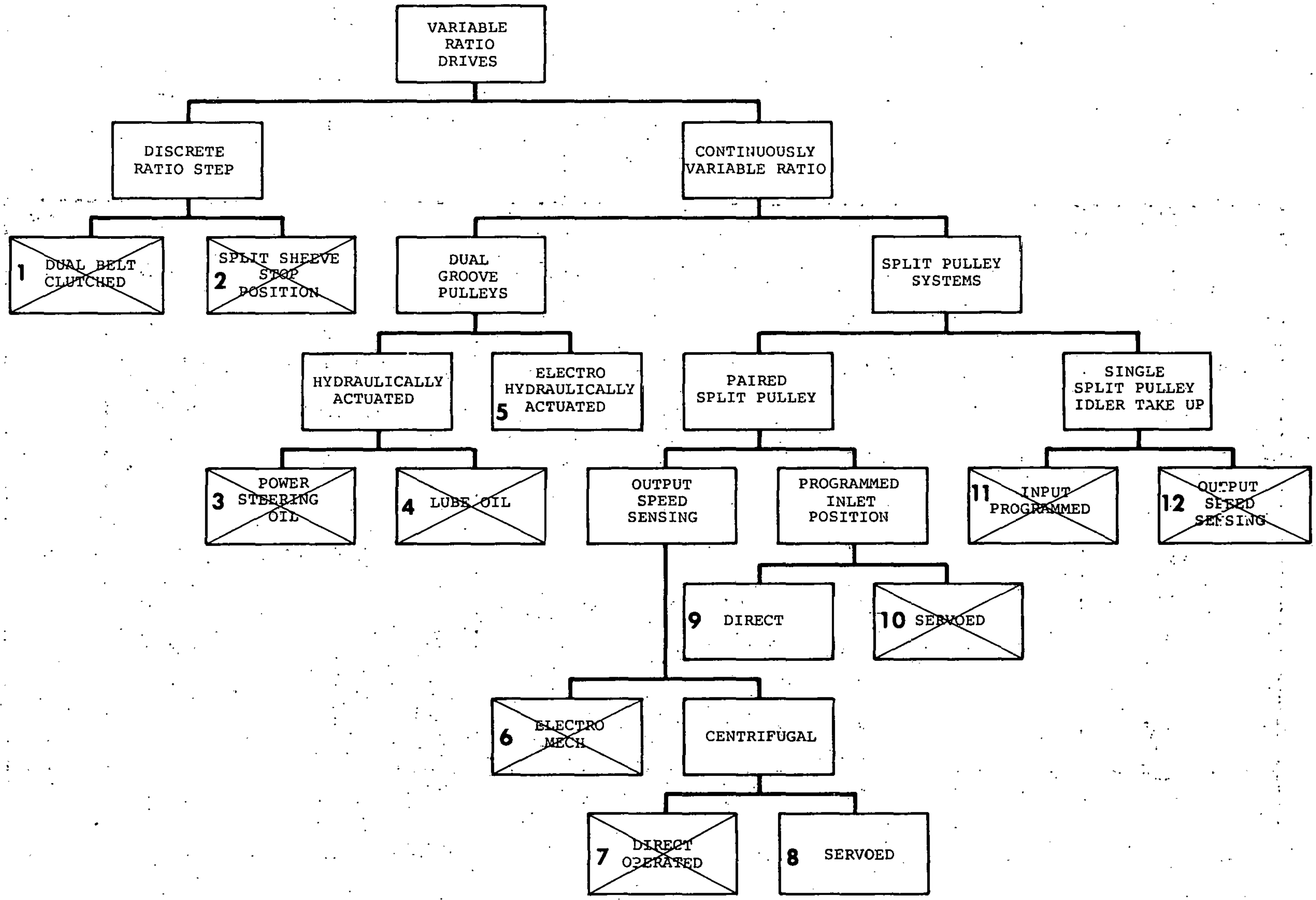
2.1 Drive System Envelope

The basic drive system envelope definition has been completed. The baseline engine (1974 Chevrolet Nova, 350-V8) accessory section is shown on Figures 5 and 6. Numerous variable-ratio, paired, split-sheave drive configurations were considered, and two basic configurations, that represent minimal accessory compartment envelope reconfiguration impact, were selected.



VARIABLE RATIO BELT SYSTEMS

FIGURE 2



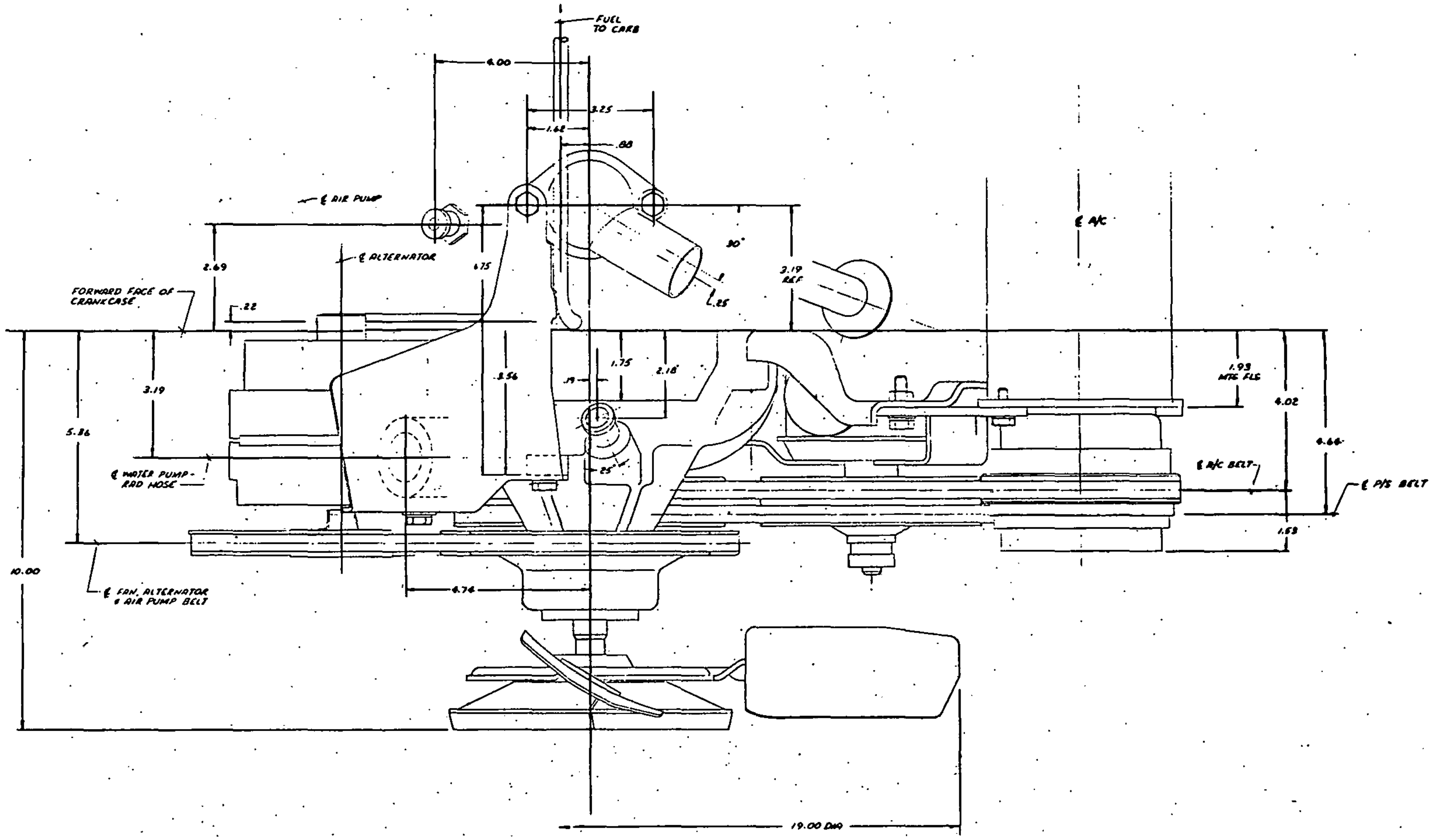
TYPES OF VARIABLE SPEED BELT DRIVES

FIGURE 3

SYSTEM NO.	SIMILAR APPLICATIONS	COMMENTS	RECOMMENDED FOR FURTHER STUDY
1	BUICK STUDY	LOW FUEL ECONOMY PAYOFF	NO
2	OFF ROAD FAN DRIVE	LOW FUEL ECONOMY PAYOFF	NO
3	CONCEPT	STD PULLEYS ON BOTH ENDS REQ P/S MOD	NO
4	CONCEPT	STD PULLEYS ON BOTH ENDS REQ ENG LUBE OIL	NO
5	ADAPTED "GERBING"	STD PULLEYS ON BOTH ENDS	YES
6	"PAM" DRIVE	PARTIALLY DEVELOPED REQ OUTBOARD SUPPORT	NO
7	"SALISBURY" TYPE	DEVELOPED FOR 3600 RPM	NO
8	MODIFIED "SALISBURY" TYPE	ACURATE CONTROL	YES
9	ADAPTED GEN DRIVE	ADEQUATE CONTROL WITH SIMPLICITY	YES (PRIME)
10	CONCEPT	COULD BE USED TO IMPROVE 9 IF REQ	NO
11	CONCEPT	TRADE OFF ONE SPLIT PULLEY FOR IDLER	NO
12	CONCEPT	TRADE OFF ONE SPLIT PULLEY FOR IDLER	NO

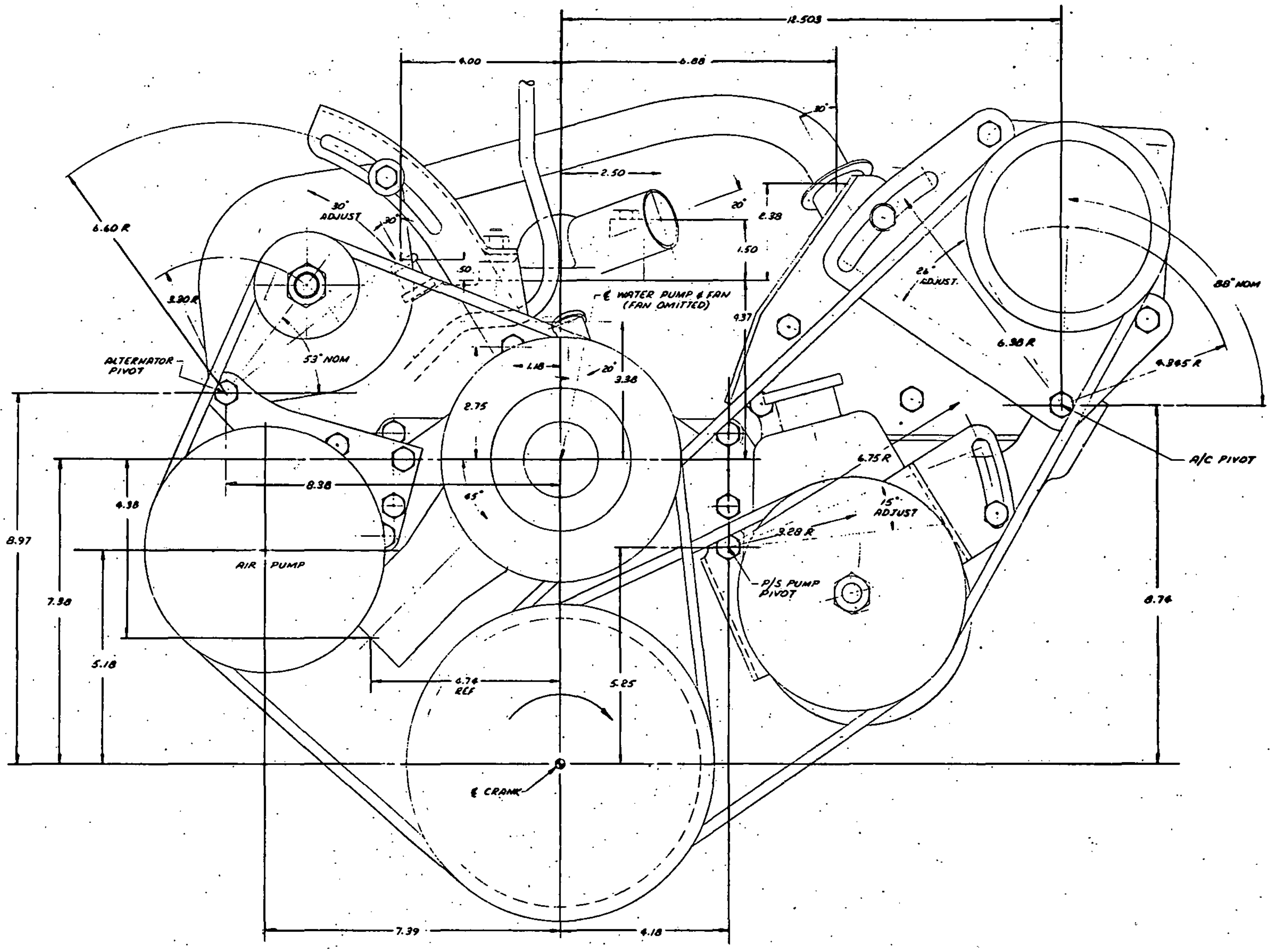
SUMMARY AND PRELIMINARY SELECTION OF BELT DRIVES

FIGURE 4



STOCK ACCESSORY LAYOUT - TOP VIEW
1974 350 V-8 NOVA

FIGURE 5



STOCK ACCESSORY LAYOUT - FRONT VIEW
1974 350 V-8 NOVA

FIGURE 6

(a) Prime Configuration

- o Driving (speed control) sheave on engine crank shaft

- o Driven sheave on water pump/fan shaft

- o Air conditioner compressor and air pump driven with single belt from water pump at fixed speed ratio

- o Power steering pump and alternator driven with single belt from water pump at fixed speed ratio

NOTE:

The air conditioner compressor and alternator exchange positions to balance belt loads.

(b) Secondary Configuration

- o Driving (speed control) sheave on engine crank shaft

- o Driven sheave on power steering pump shaft

- o Air conditioner compressor and water pump/fan driven with single or dual belt from power steering pump at fixed ratio

- o Alternator and air pump driven with single belt from water pump at fixed ratio

Design analyses for the two configurations are similar, with speed and load relationships the major considerations.

A sectional view of the baseline vehicle crank shaft and water pump/fan configuration is presented on Figure 7, which shows the available fore and aft drive-system envelope. First-order design sizing estimates indicate an additional half-inch will be required, which may be obtained by moving the radiator and/or fan forward. A water pump modification may also provide the additional envelope.

2.2 Drive System Load and Speed

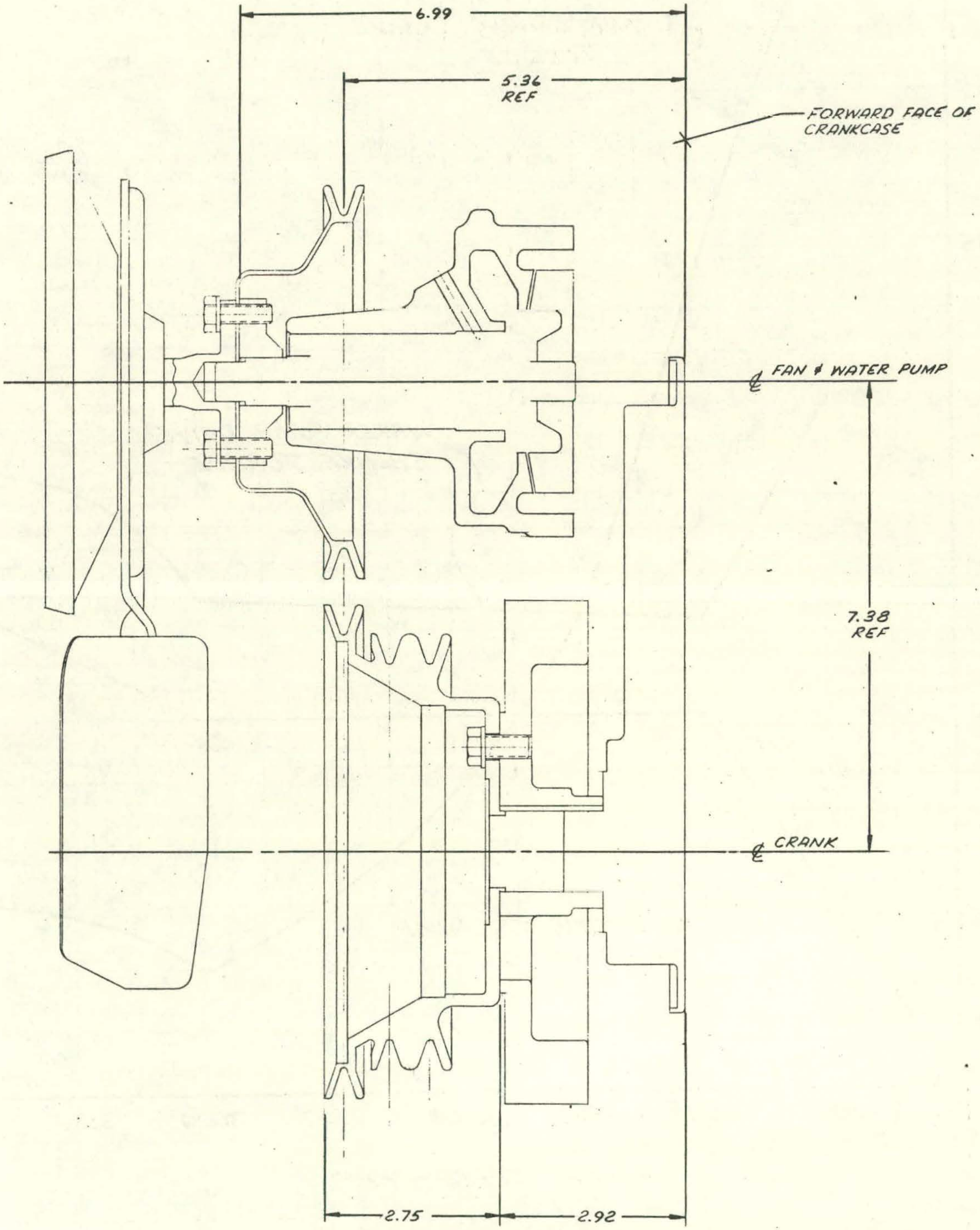
The maximum specified(1) accessory load is shown on Figure 8 as crankshaft driving sheave torque-versus-engine speed. Drive-system speed-ratio and driven-sheave torque curves are also shown. Speed-ratio characteristics have been assumed for initial design analysis and may be considerably different at final design. The peak, low-speed torque requirement results in high drive-belt tension and control system force levels that may cause the selected 1500 rpm constant-speed level to increase. Increasing the water pump and fan constant-speed level above 1250 rpm required existing accessories derating to control input-power requirements.

2.3 Drive System Ratio

Envelope and drive-belt data place certain restrictions on sheave size, and design-speed requirements dictate ratio. The optimum drive system size is shown on Figure 9 for the restrictions listed below.

- o Crank shaft-to-water pump center distance = 7.38 inches
- o Sheave-to-sheave clearance = 0.50 inch

(1) Design specification for the accessory drive system is AiResearch Report No. 75-311125.



STOCK CRANK AND WATER PUMP SECTION VIEW
1974 350 V-8 NOVA

FIGURE 7


ASSUMPTIONS:
1) MAX ACCESSORY LOAD
CRANK DRIVING SHEAVE TORQUE
WATER PUMP DRIVEN SHEAVE TORQUE
CRANK TO WATER PUMP SPEED RATIO
TORQUE ~ LBS-FT
WATER PUMP SPEED ~ RPM
ENGINE SPEED ~ RPM
45
40
35
30
25
20
15
10
3000
2500
2000
1500
1000
500
0
500
1000
1500
2000
2500
3000
3500
4000
CALCULATED BY
TRACED BY
CHECKED BY
APPROVED BY
UNIT NO.
TORQUE AND SPEED CHARACTERISTICS
-CRANK TO WATER PUMP DRIVE SYSTEM-
FIGURE 8
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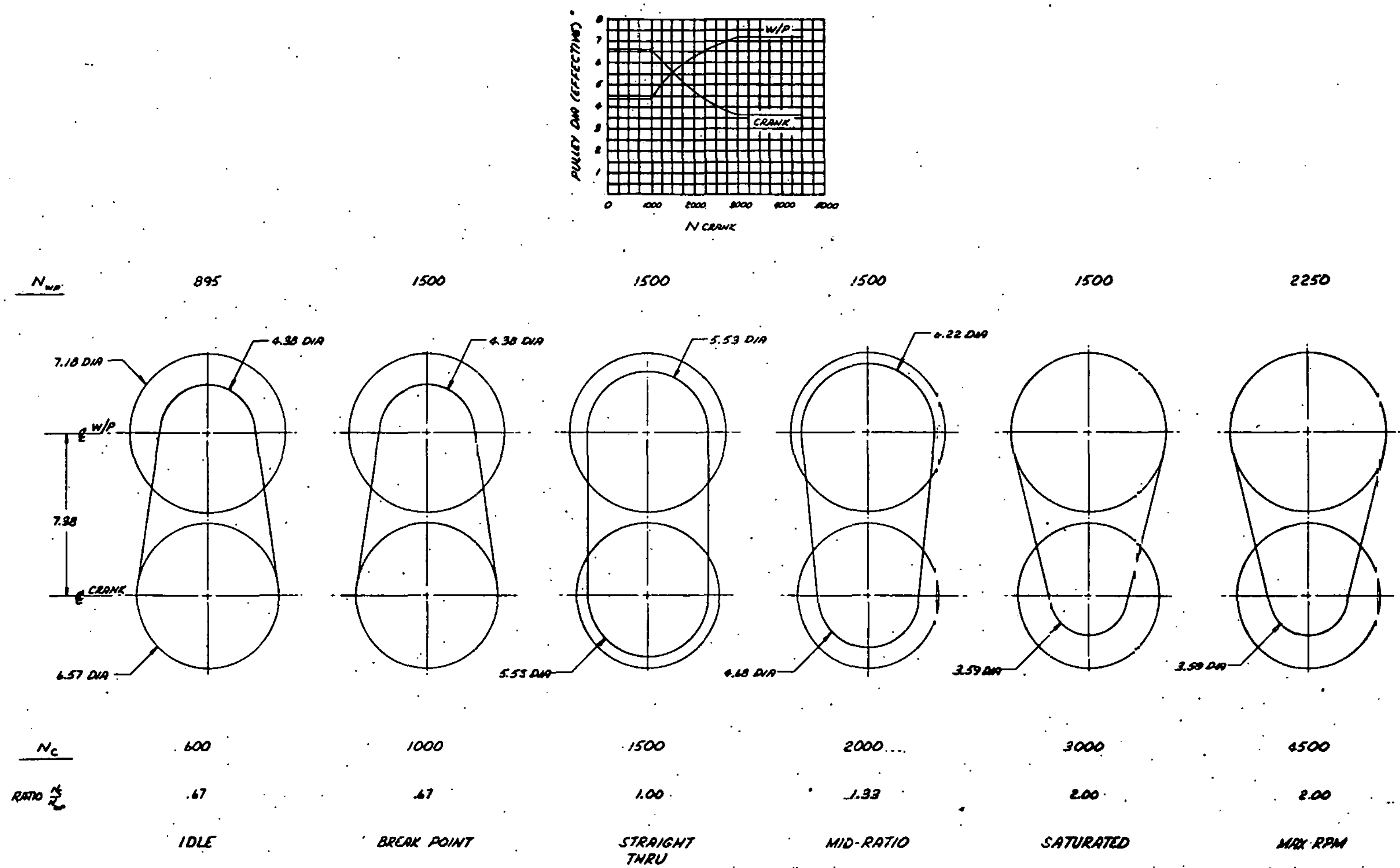


VARIABLE RATIO BELT ACCESSORY DRIVE RATIO SCHEMATIC

FIGURE 9

- o Minimum belt-pitch diameter:

  Water pump, 4.00 inches (envelope restriction)
  Crank shaft, 3.00 inches (belt restriction)

- o Maximum belt width = 1.00 inch

- o Sheave groove angle = 26°

- o Constant output speed = 1500 rpm from 1000 to 3000 engine rpm

3.0 VARIABLE RATIO BELT ANALYSIS

Two paired, split pulley, drive system concepts were selected for analysis and practical automotive application:

(a) Mechanically controlled, directly operated, with input programming

(b) Hydromechanically controlled, servo-operated, with output speed sensing

Both concepts would be mechanized with torque sensing sheaves to achieve proper belt tension over the entire load range.

The mechanical concept preliminary analysis results indicate that this system will not meet speed control requirements over the total required speed ratio and load range. It may be possible to meet system design goals with hybrid mechanical concepts if complexity can be minimized.

3.1 Mechanical Concept Analysis

There are two basic methods of mechanizing the speed controller for a direct-operated-mechanical, variable-ratio belt system. Hybrid combinations of basic methods may also be applied to specific drive-system requirements.

Type I - Utilizes a speed-sensitive regulator at the driven sheave that adjusts pulley width to maintain the driven shaft at an essentially constant speed beyond the predetermined break point

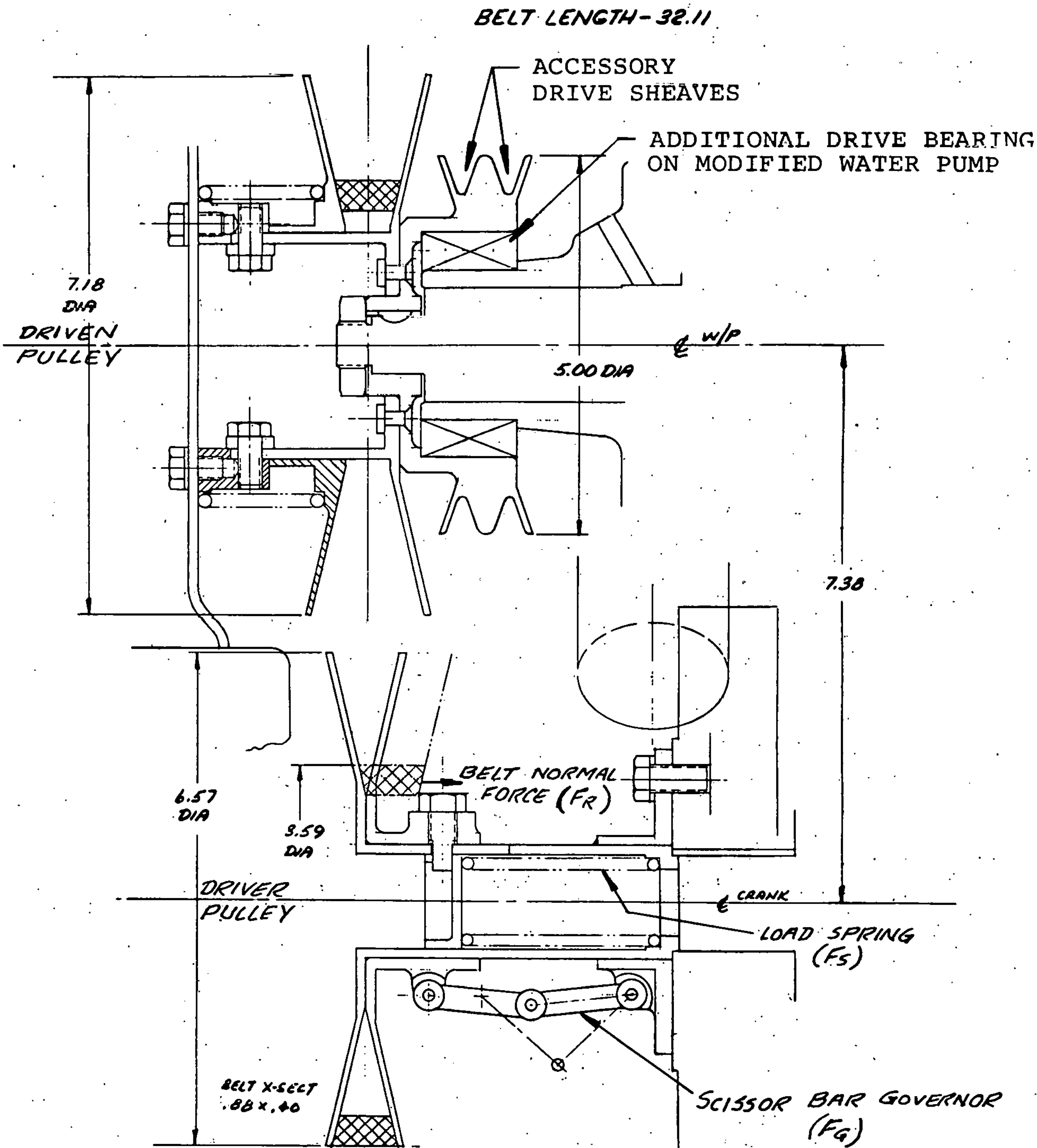
Type II - Utilizes a speed-sensitive regulator at the driver sheave that programs the driver sheave width in accordance with a predetermined schedule selected to achieve the desired constant speed at the driven shaft

From an accuracy standpoint, control at the driven shaft (Type I) is the obvious choice since it is directly sensitive to the controlled variable, (output shaft speed). The difficulty with mechanizing a direct-operated controller at this location is the high gain required. It must have enough sensitivity and power to drive the moveable sheave through required total travel with minimum error signal. The controller must also be stiff enough to maintain position when subjected to widely varying axial forces created by belt torque loading.

Locating the controller at the driver sheave (Type II) simplifies the mechanization task, since the speed range, over which the controller must produce sheave motion, is much wider. This approach basically requires a controller that converts speed into position with enough stiffness to be insensitive to the side force created by the belt.

If the consideration is limited to a direct-operated mechanical control, the Type II system appears to be best for accomplishing a practical set of hardware. The analysis presented in the following sections presents sizing data and performance predictions for an auto-accessory drive using the Type II system.

The selected system is shown in Figure 10, and the analysis is directed to the engine crankshaft mounted pulley (driver pulley). It is assumed that the driver pulley may be mechanized with torque sensing or simple spring loading to transmit all required loads.



FEASIBILITY CONCEPT
VARIABLE RATIO BELT ACCESSORY DRIVE

FIGURE 10

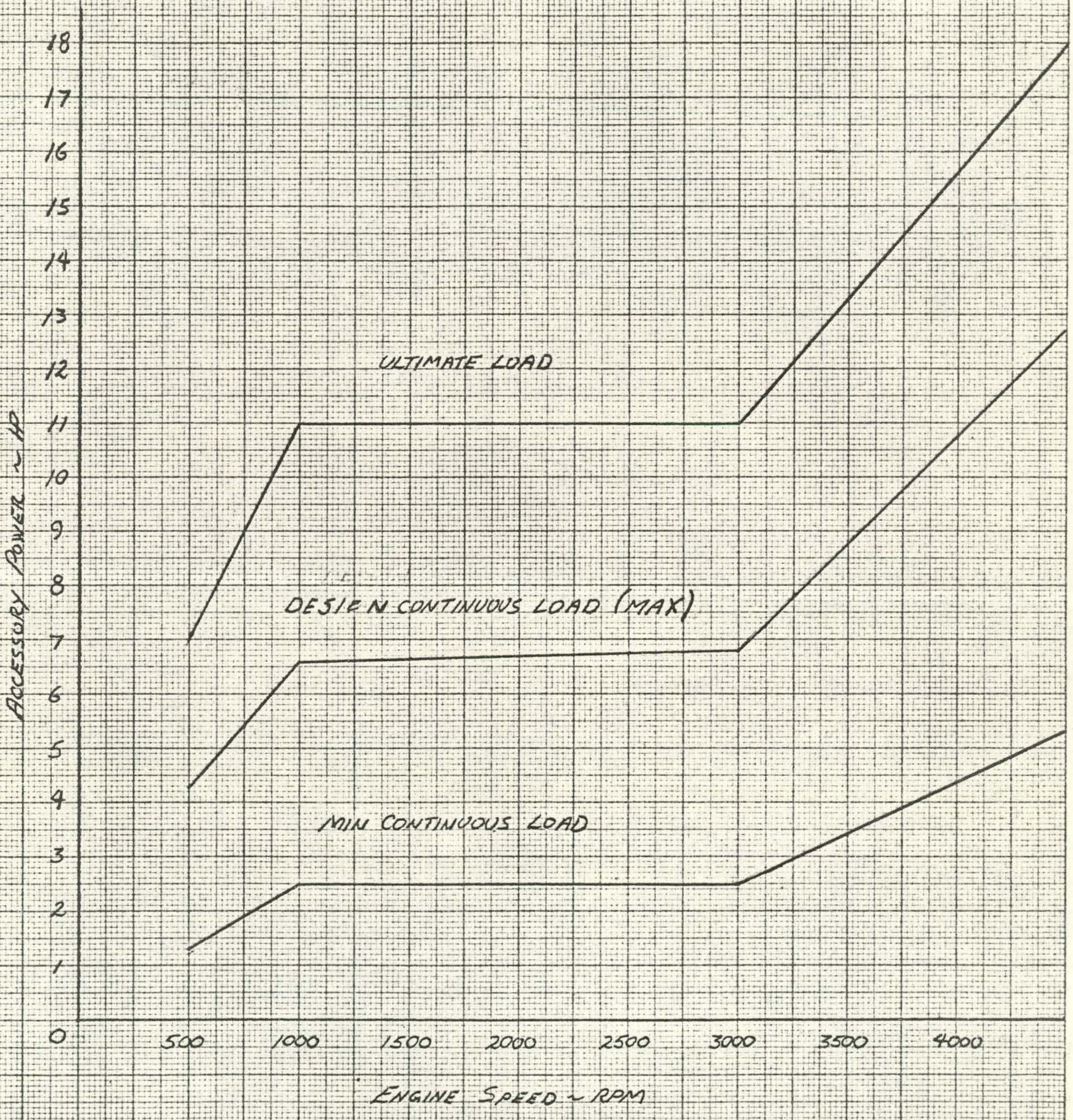
Physical pulley diameter and engine speed relationships, to achieve required accessory speed control for optimum vehicle fuel economy, are as shown in Figure 9. Three load lines that represent the following vehicle operating conditions are shown in Figure 11.

Load	Condition	Percent Time at Load
Minimum continuous	A/C off, 20 amp alt.	50 percent
Design continuous	A/C max., 30 amp alt.	48 percent
Ultimate	A/C and P/S peak	2 percent

The basic, direct-operated, pulley consists of a governor that generates a force proportional to engine speed squared, which is countered by a spring force. The difference in spring and governor force is set to equal the movable pulley-face normal force caused by drive-belt loading at each discrete pulley position and speed. This produces an open-loop control system, with speed-droop characteristics, depending on load variation from the design condition.

Maximum continuous load was selected as the design condition because of close total range mid-load approximation. The belt, spring, and governor-force relationship, required to meet design conditions for mid-load analysis, is presented in Figure 12. Spring and governor characteristics may be shaped by a variety of techniques to satisfy detail design requirements, not considered at this stage of analysis.

A computer program for driver pulley physical and functional relationships was used to determine load capacities at various speed and ratio positions, representing the actual operating range. This data is presented in Figure 13 where the maximum and minimum permissible "traction coefficients" (= 0.85, = 0.2) establish usable load ranges for any speed and ratio position. A possible drive system instability in the 500 to 2000 rpm range is indicated, where loads along any given pulley ratio position from one to six may be transmitted at more than one speed condition.



CALCULATED BY	CNL	6-75	DESIGN LOAD CONDITIONS	
TRACED BY				FIGURE 11
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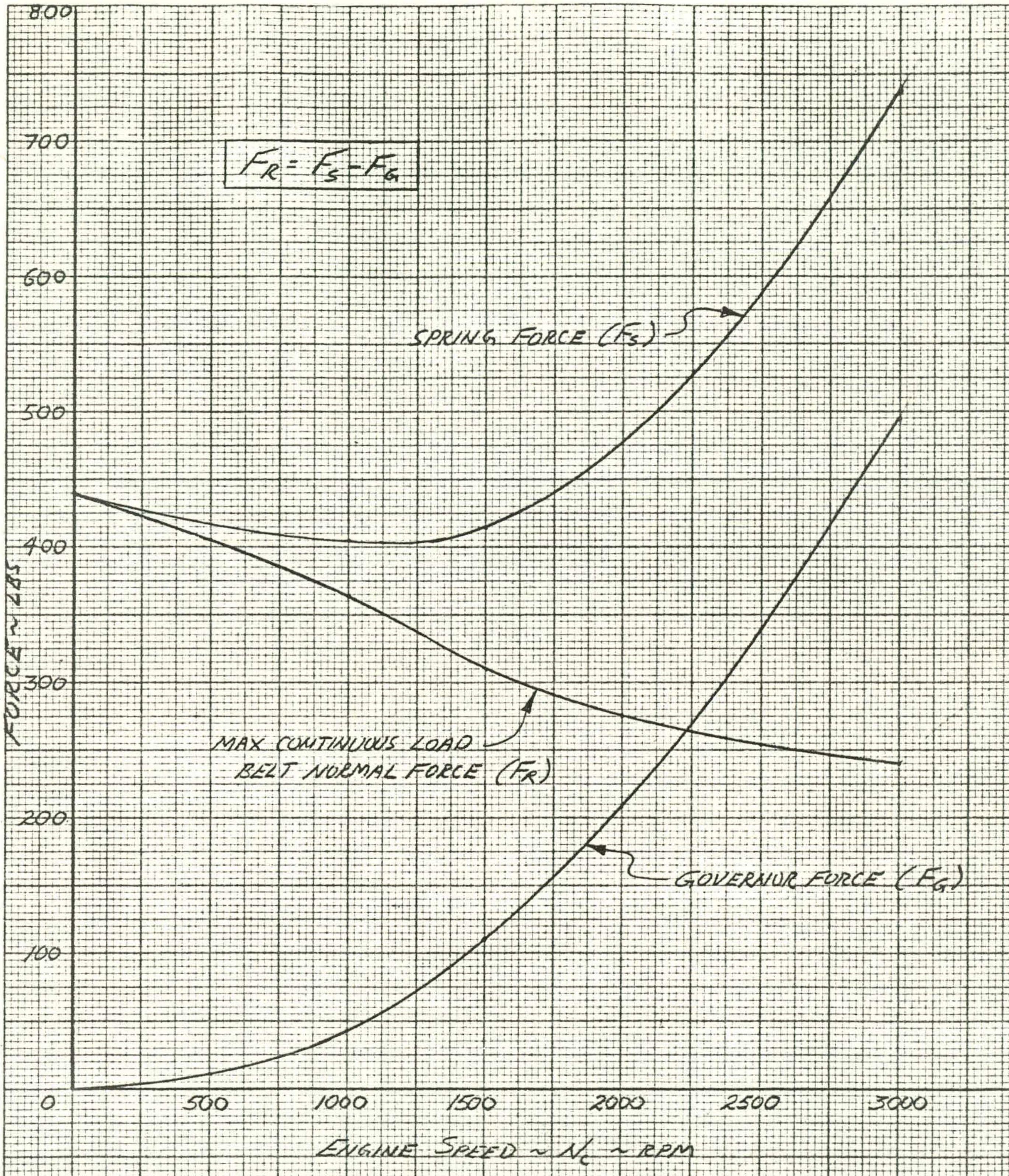

CALCULATED BY	CNC	6-75	DRIVER SHEAVE LOAD SYSTEM	FIGURE 12
TRACED BY			MECH, OPEN-LOOP, CSD BELT DRIVE	
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APPROVED BY			AiResearch Manufacturing Company of Arizona	
UNIT NO.				

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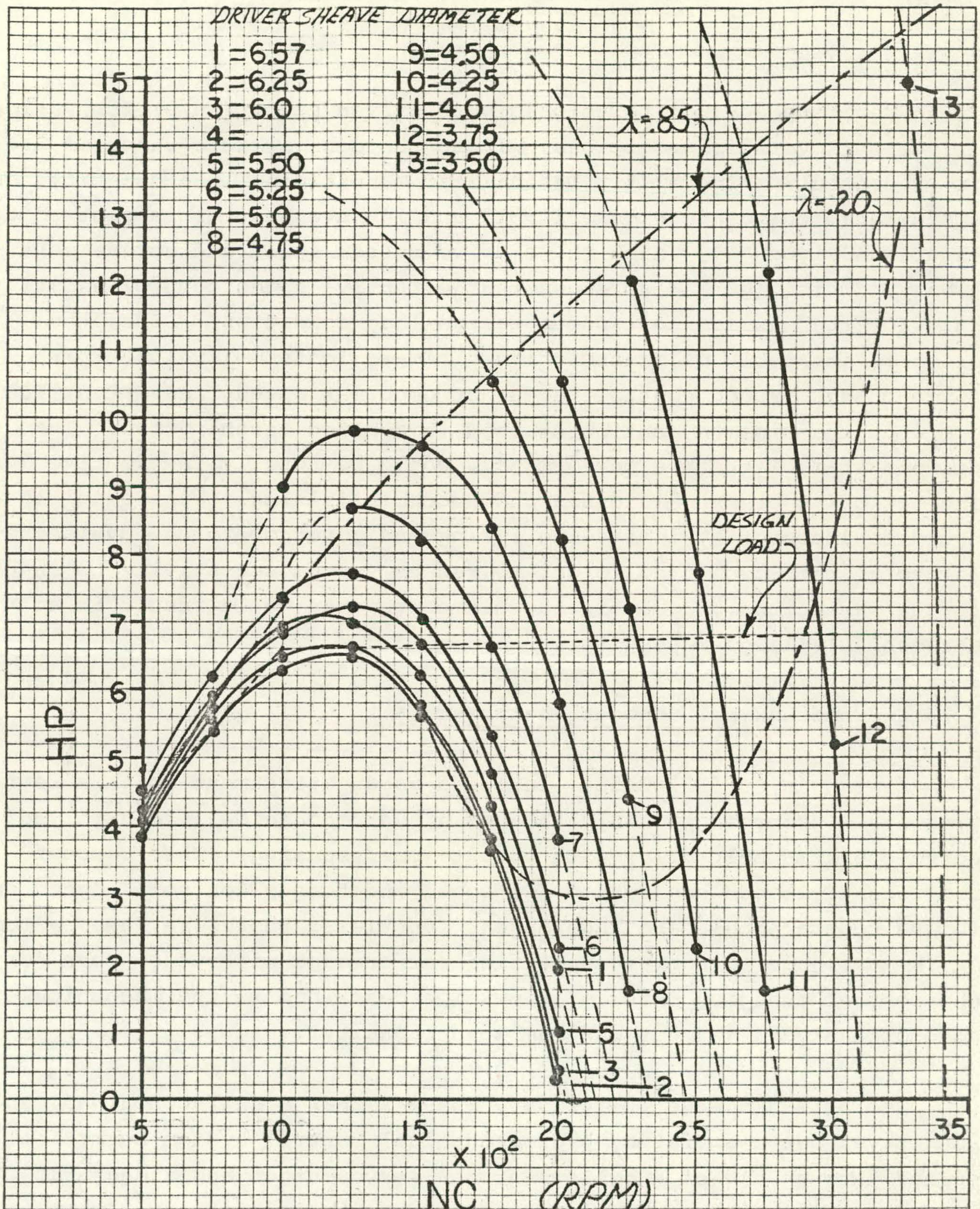

CALCULATED BY	LSD	HORSEPOWER VS NC
TRACED BY		FIGURE 13
CHECKED BY		6-18-75
APPROVED BY		AiResearch Manufacturing Company of Arizona
UNIT NO.	103	

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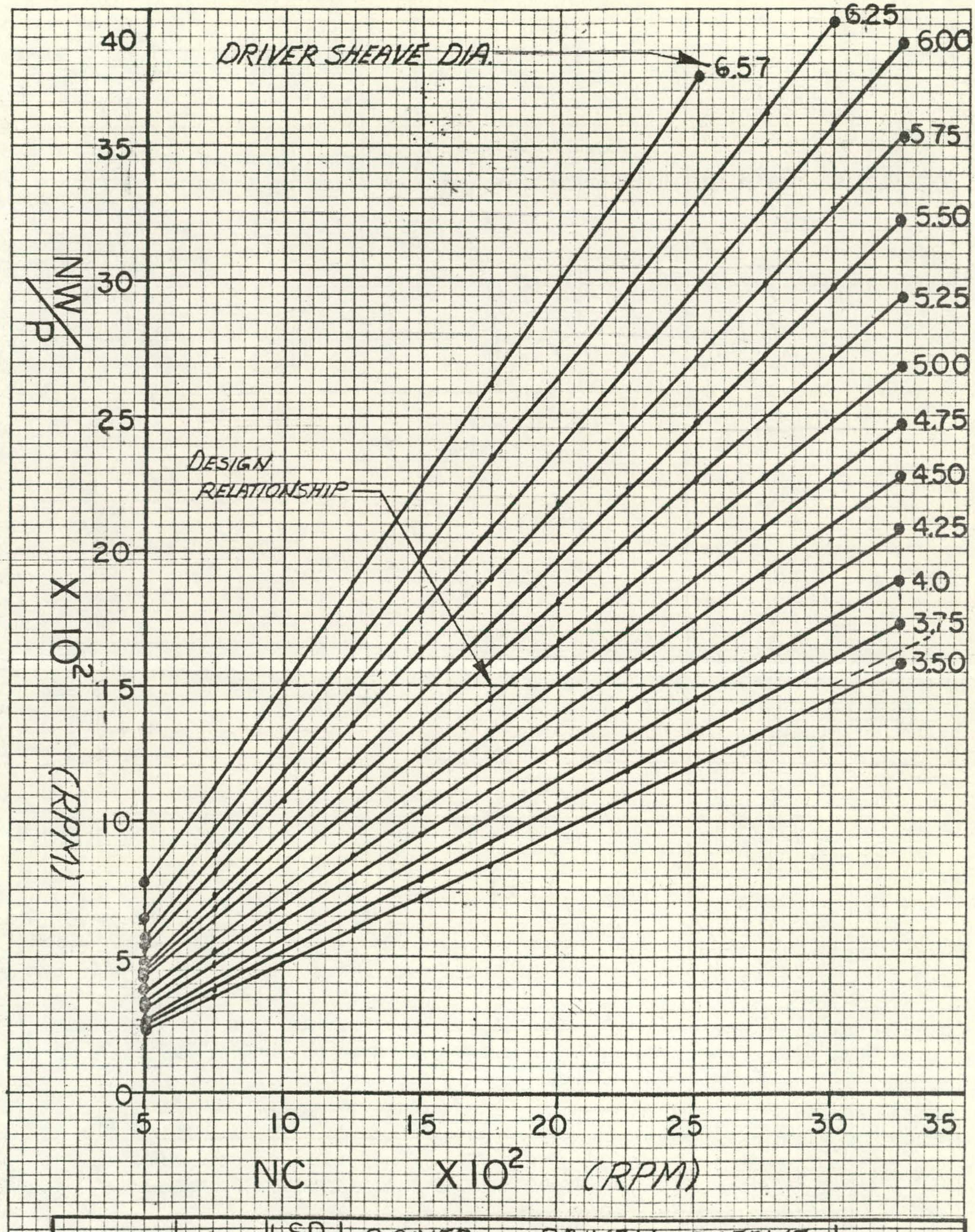
Driver speed (Nc) relationship to driven speed (Nw/p) for various driver-pulley ration positions, within the design ratio range of 3:1, is presented in Figure 14. Design load conditions, load capacity, and speed-ratio curves in Figure 11, 13 and 14 have been cross-plotted to establish Figure 15, which shows drive-system speed and torque sensitivity. This figure also shows that effective speed control exists only for the maximum-continuous design load condition. Also, the drive is incapable of transmitting ultimate loads below 1950 rpm. Ratio change (speed control) does not occur at speeds less than 1750 rpm when the minimum continuous load is applied to the drive. The drive system, with this performance, would be totally ineffective during A/C off operation in urban and suburban driving. This condition could be improved by increasing spring and governor force levels, relative to belt-generated force, to reduce system sensitivity to input torque variations. However, these improvements would result in an unreasonably large mechanism.

3.2 Hydromechanical Concept Analysis

Design layout L-3621560, included as an attachment to this report, presents a proposed mechanization of a hydromechanically controlled, servo-operated belt drive with output-speed sensing. This concept utilizes a closed-loop system for speed and torque control. Speed and torque control logic are independent functions; therefore, the system may be designed to achieve specified performance without concern for the control function interrelationships described in the mechanical concept analysis (except maximum load capacity).

The driven sheave (L-3621560, Sheet 2) has a self-contained, speed sensitive, servo-operated, hydromechanical control system hermetically sealed through use of a bellows at the servo output and a magnetic drive coupling at the pump input. Speed control is achieved with a governor valve located perpendicular to the axis of rotation to use centrifugal force for control sense. Belt pinching forces are applied



CALCULATED BY	LSD	DRIVER vs DRIVEN SHEAVE SPEED	FIGURE 14
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APPROVED BY		AiResearch Manufacturing Company of Arizona	
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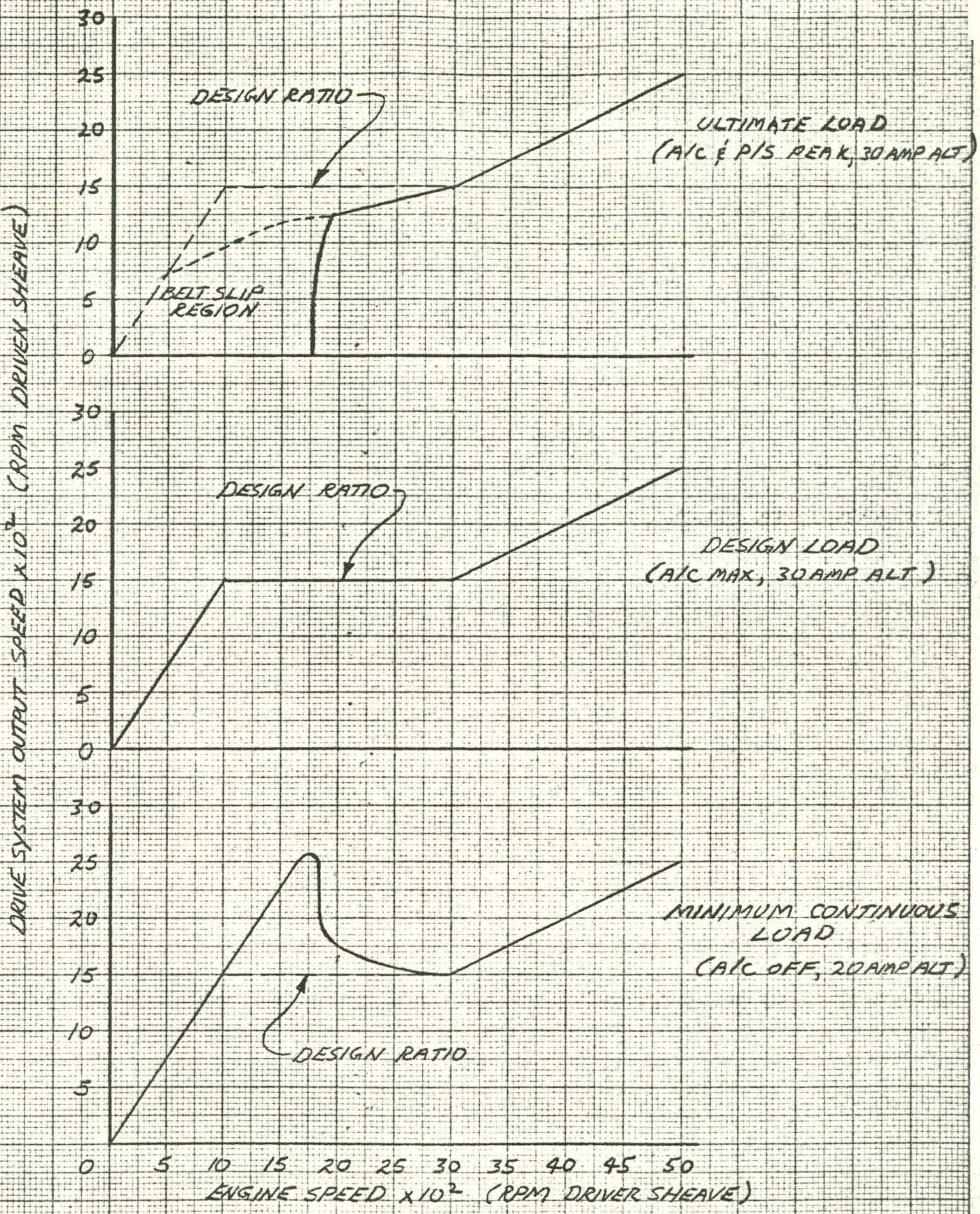

CALCULATED BY	CNL	6-75	DRIVE SYSTEM SPEED CHARACTERISTICS FOR VARIOUS LOAD CONDITIONS	FIGURE 15
TRACED BY				
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APPROVED BY				
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P 5106

to the moveable sheave flange with a leaf type, spider spring that also prohibits angular rotation of the flange. The servo piston balances the spring force over the total accessory input-torque range to maintain sheave speed at the governor set point. The governor is set to maintain a driven-sheave speed of 1500 rpm over an engine-speed range of 1000 to 3000 rpm. The drive system is saturated at engine speeds below 1000 rpm and above 3000 rpm.

Engine - RPM	Speed Ratio (Driven/Engine)
Idle - 1000	1.5:1 (fixed)
1000 - 3000	1.5 to 0.5:1 (variable)
3000 - 5000	0.5:1 (fixed)

Hydraulic pressure is generated by a double element gear pump with sufficient capacity (0.5 gpm) to achieve full ratio change in 0.5 second at a pump speed of 1500 rpm. System response at speeds less than 1500 rpm lags to prevent accessory overspeeds during fast transient engine accelerations. Pump relief-valve pressure is set at 225 psi minimum, which is approximately 50 psi above the maximum required working pressure, to assure stiff control response at all operating conditions. The pump is driven through a magnetic coupling consisting of coaxial ring magnets with ten poles each. The magnet drive is sized to provide 5 lb-in. torque or 150 percent of the torque required for pump operation at the maximum relief-valve setting. The torque margin is required for cold day start-up where hydraulic fluid viscous effects must be overcome.

The driven sheave is shown mounted on the water pump and engine cooling-fan drive-shaft and also carries the fan shroud and grounds the outer magnetic-drive-coupling ring. This configuration allows fan-shroud tip clearances to be greatly reduced, which increases fan efficiency. The shroud is attached to the radiator via a flexible boot.

All hydraulic failure modes are in the direction to reduce accessory speeds to the minimum possible ratio, except one that requires the governor valve to jam in the low-speed (full-pressure) position.

First Order Failure Mode	Resultant Accessory Speed
(1) No pump pressure	Minimum
(2) Relief valve jammed open	Minimum
(3) Relief valve jammed close	Normal operation until magnetic drive slips, then condition (1) exists
(4) Governor valve jammed in bypass mode	Minimum
(5) Governor valve jammed in full pressure mode	Maximum
(6) Loss of hydraulic fluid	Minimum
(7) Magnetic drive slips	Minimum

Note: If magnetic drive coupling slips or disengages, it will automatically re-engage when the engine is shut down.

The probability of overspeeding accessories, due to governor valve jamming in the low-speed position, can be minimized through proper design. The centrifugal, spring, and accessory vibratory input forces acting on the valve, tend to preclude jamming. Internal contamination sources will be held to a minimum and all ferrous particles will be stored on the magnetic drive coupling inner ring. External contamination sources do not exist because the unit is hermetically sealed.

Sheet 3 of L-3621560 shows the driver, or crankshaft-mounted-sheave, representing the torque-sensing drive-system member. The driver sheave utilizes a cam action to translate applied belt-tension forces into belt-pinching forces. This boot-strapping tension-control technique has proven effective on various belt-drive designs currently in service. The design presented in this analysis uses twin cam rollers to share a nominal load of 60 - 100 lbs each. All moving and loaded surfaces are internal and grease-packed to provide low hysteresis and good wear characteristics. The cam ramp-angle has been set to provide belt tension for transmitting 10 to 75 lb-ft torque and the cam spring will supply belt tension for torques of 10 lb-ft or less.

This belt drive design has been heavily biased to maintain the highest possible system efficiency. This has been accomplished by applying optimum design and control techniques to maintain a high system-traction coefficient.

Load Condition	Average Traction Coefficient	Minimum Theoretical Efficiency
Ultimate	0.86	98%
Design continuous	0.74	97%
Minimum continuous	0.45	91%

Speed control is not a function of accessory input loads; therefore, deviations from a specified speed requirement will be determined by governor valve droop characteristics. A special effort must be made during detail design of this valve to eliminate stiction and dead band.

4.0 OTHER CONFIGURATIONS

The hydromechanical variable ratio belt system, discussed in Section 3.2, represents an ideal concept that treats design requirements in the most direct and accurate manner. Mechanization of the basic concept may be achieved with various configurations to find the best possible application within any specific vehicle.

A configuration that physically combines hydraulic controlled driven sheave and power steering pump functions is currently being evaluated in an attempt to:

- o Eliminate repeat of basic components such as pumps, housings and bearings
- o Produce better installation flexibility
- o Reduce cost and weight
- o Increase serviceability
- o Better retrofit capability
- o Reduce sensitivity to belt length tolerance by incorporating an adjustable sheave center distance

APPENDIX I

APPENDIX I
IMPROVING AUTOMOTIVE FUEL ECONOMY WITH ACCESSORY DRIVES

PRESENTED TO ENERGY RESEARCH AND DEVELOPMENT AGENCY,
COMMITTEE ON ALTERNATIVE AUTOMOTIVE POWER SYSTEMS
MAY 6, 1975 AT ANN ARBOR, MICHIGAN

INTRODUCTION (Figure A)

The Garrett Corporation is conducting a study to develop an optimum automotive accessory drive system for the ERDA committee on Alternate Automotive Power Systems. This presentation summarizes the current study progress.

IMPROVING AUTOMOTIVE FUEL ECONOMY WITH ACCESSORY DRIVES

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FIGURE A

(Figure B)

OBJECTIVE

Minimize accessory power consumption and maximize overall vehicle fuel economy.

APPROACH

Study variable speed drives and APUs utilizing system load matching techniques and improved accessory designs.

FACTORS (Prime and Secondary)

- o Prime: Factors other than fuel economy were considered to achieve a viable drive system.
- o Secondary: Initial cost, complexity, maintenance cost, noise, installation flexibility, weight, and durability.

The study considers engine and vehicle accessories that are normally belt driven from the engine crank pulley:

Engine Cooling Fan
Water
Alternator
Air Conditioning Compressor
Power Steering Pump
Emission Control Air Pump

OBJECTIVE

- MINIMIZE ACCESSORY POWER CONSUMPTION AND MAXIMIZE OVERALL VEHICLE FUEL ECONOMY

APPROACH

- STUDY CONTINUOUSLY VARIABLE SPEED DRIVES OR AUXILIARY POWER UNITS IN CONJUNCTION WITH SYSTEM LOAD MATCHING AND IMPROVED AUTOMOTIVE ACCESSORIES

FACTORS CONSIDERED

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

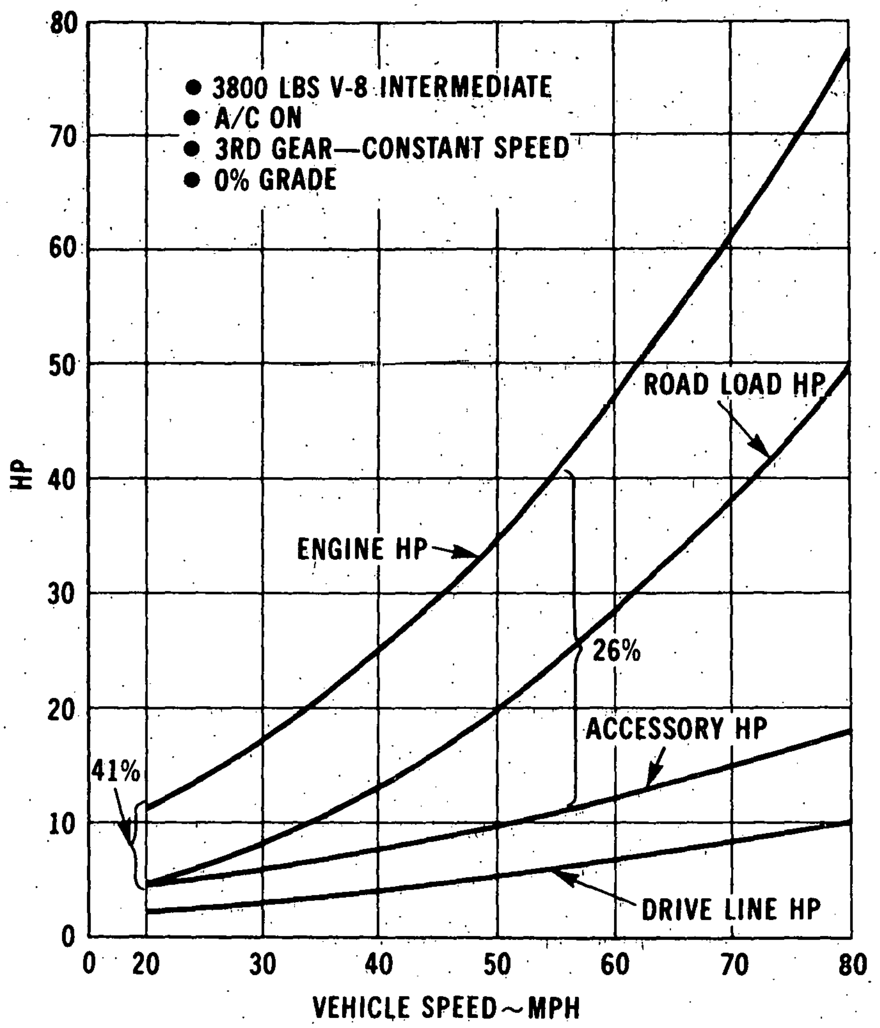
IMPROVING AUTOMOTIVE FUEL ECONOMY
WITH ACCESSORY DRIVES

FIGURE B

(Figure 1)

The significance of the composite accessory load on vehicle fuel economy is apparent when accessory and total engine power requirements are compared. The engine power required to sustain a cruising speed of 55 mph is approximately 40 hp, while the combined accessory power requirement is 11 hp, or 26 percent of the developed power. Unfortunately, the accessory-to-engine ratio of required power worsens as vehicle speed is decreased. This is caused by the accessory imposed loads remaining relatively high while the road loads diminish rapidly.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



WHERE THE POWER GOES

FIGURE 1

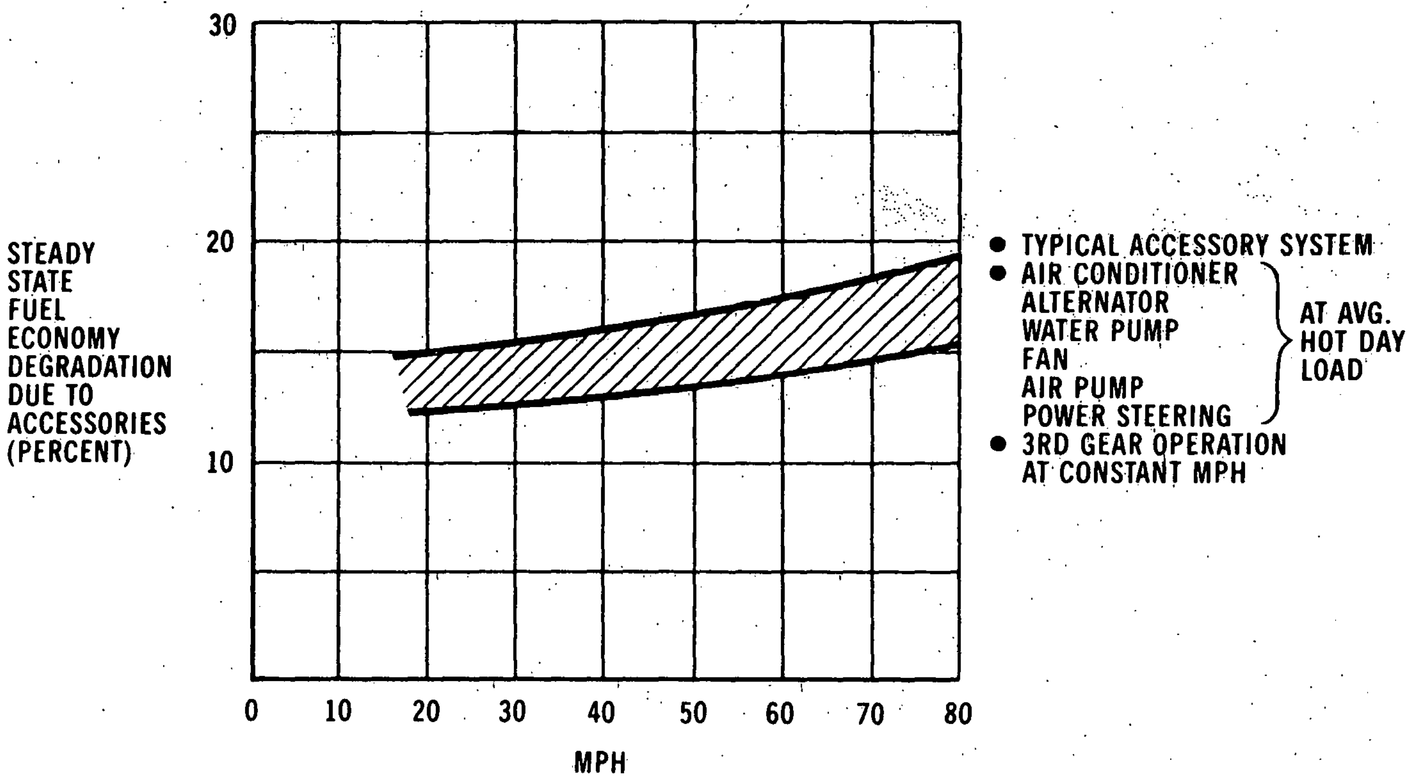
(Figure 2)

The accessory power requirements are reflected in vehicle fuel economy. A degradation in fuel economy of 12 to 18 percent may be attributed to the accessories for steady-state vehicle operation.

Objective

The initial study effort was concentrated on determining how well existing accessories are load matched and describing areas for improvement that would reflect the largest payoff.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



TYPICAL ACCESSORY FUEL CONSUMPTION

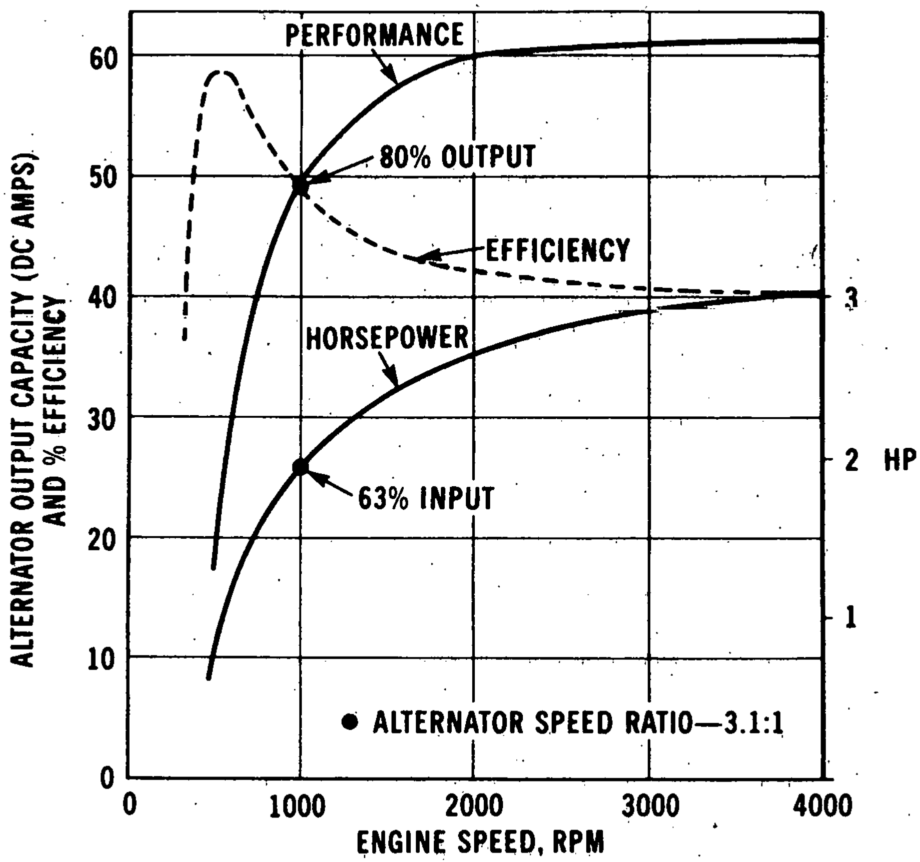
FIGURE 2

(Figure 3)

A typical example of accessory performance characteristics is demonstrated by this alternator map. The 60-amp alternator performance indicates that 80 percent of the maximum output may be obtained at a speed of slightly greater than 3000 RPM where the input power requirements are only 63 percent of the power normally required for full load output.

As each accessory was studied, it became apparent that the designs were heavily influenced by potential load demands at low vehicle and engine speeds. The accessories are belt driven at fixed speed ratios to achieve adequate performance at low vehicle velocity which results in excessive accessory speeds at high vehicle velocities. In effect, the accessories are over-sized for engine speeds greater than 1000 to 1500 RPM.

REPRESENTATIVE ACCESSORY PERFORMANCE

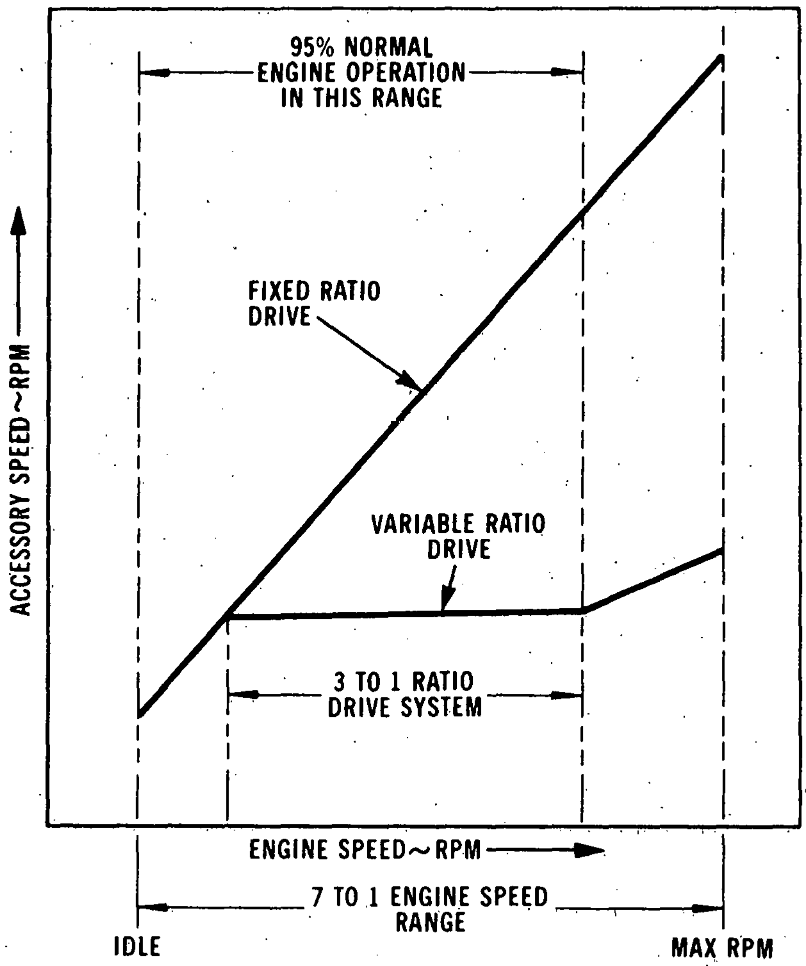


TYPICAL ALTERNATOR LOAD PROFILE

FIGURE 3

(Figure 4)

The normal engine speed ratio is approximately 7 to 1, while acceptable accessory performance can be obtained within a ratio range of 2 or 3 to 1. As shown, a 3 to 1 ratio capability could provide adequate accessory speed at all vehicle velocities, thereby conserving accessory power losses due to excessive speed. This application shows the accessories driven at a fixed ratio from idle to 1000 to 1200 engine rpm, then at constant speed for the remainder of the normal engine operating range. At 3000 engine rpm, the drive becomes saturated and the accessories are again driven at fixed ratio.



TYPICAL SPEED LIMITING EFFECT

FIGURE 4

(Figure 5)

There are decided advantages, in addition to improved fuel economy, to be gained through accessory speed control. Accessory optimization to improve efficiency is possible because near constant speed operation can be obtained. The conservation of excessive accessory power can be reflected in smaller engines with no degradation in vehicle performance. Pollution, noise, accessory life and cost should all reflect improvement from controlled speed operation.

Baseline Vehicle

As an aid to definitizing accessory performance, load requirements and fuel economy, a baseline vehicle was established. A review of 1974 auto market trend data lead to the selection of an intermediate class vehicle. The selection was based on sales volume and accessory options. The leading intermediate at the time of the selection was a 350 V-8 Chevrolet Nova.

- 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

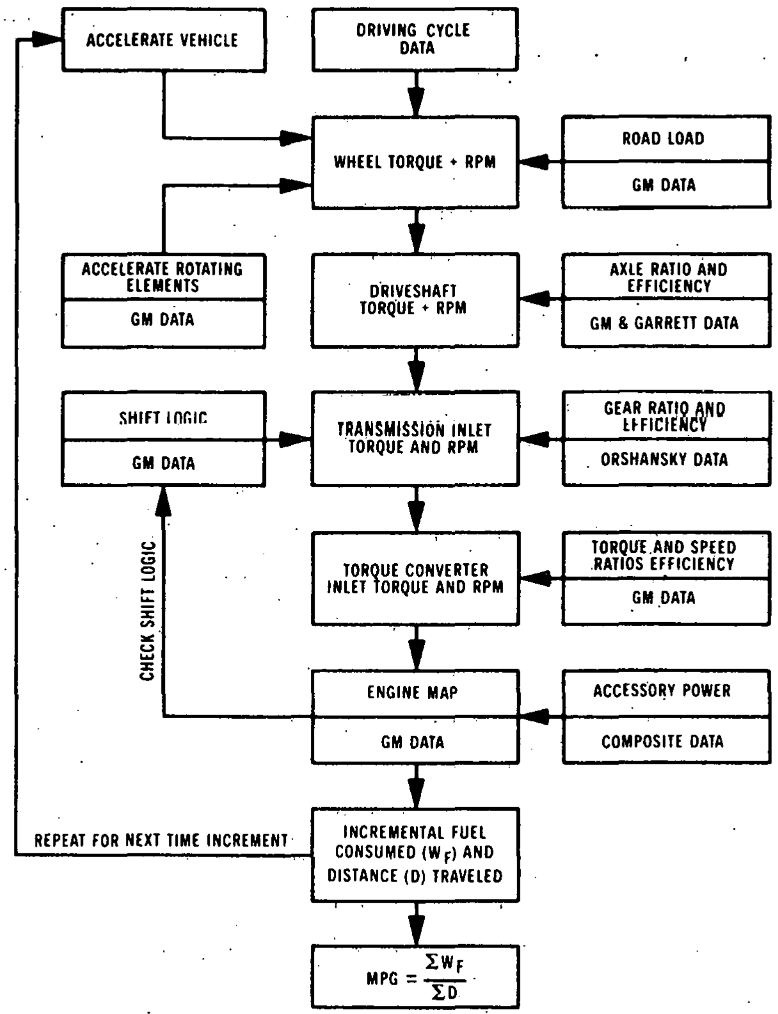
ACCESSORY DRIVE ADVANTAGES

FIGURE 5

(Figure 6)

A baseline vehicle computer model was constructed utilizing GM data for road load, engine, inertia, and drive line characteristics. The model is configured to predict vehicle fuel economy for various accessory performance and driving cycle conditions.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



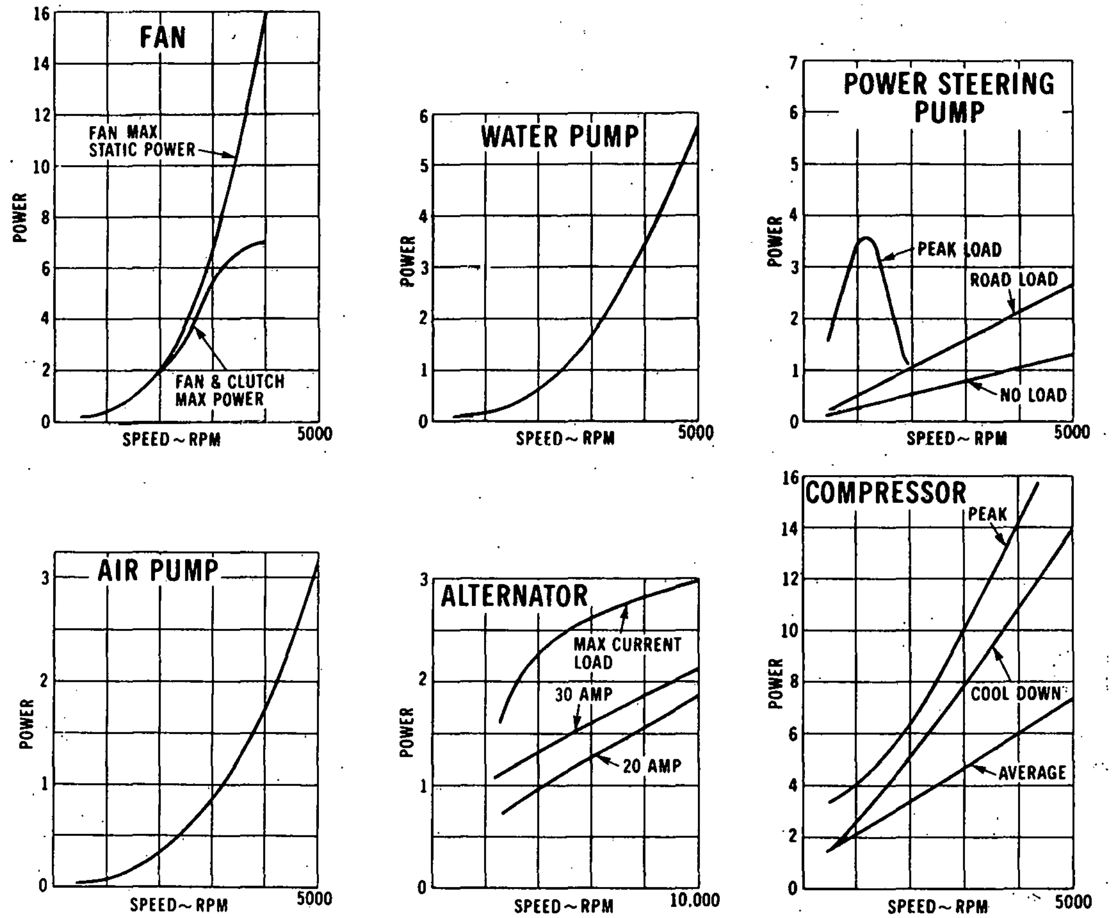
FLOW CHART OF COMPUTER MODEL

FIGURE 6

(Figure 7)

Accessory data was collected from all available sources and reduced to the composite load profiles shown. The composite data compares with Nova accessory data within 15 percent and represents an accurate cross-section of most intermediate vehicle accessory loads.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



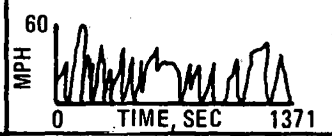
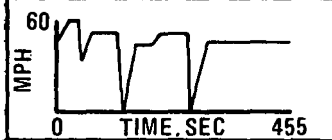
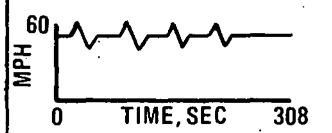
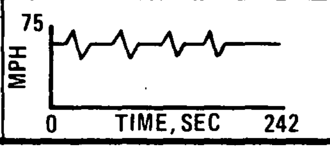
BASELINE ACCESSORY PERFORMANCE

FIGURE 7

(Figure 8)

Seven driving modes or cycles were selected for vehicle simulation. The Federal and EPA cycles represent the driving conditions used to report new car fuel economy; the four SAE cycles are used in conjunction with nationwide "chase car data" to form a composite cycle to predict average, yearly fuel economy. The final cycle consists of a steady-state or constant-speed analysis, useful for performing parametric studies without complications added by accelerations and decelerations in other cycles.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

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

COMPUTER SIMULATED DRIVING MODES

FIGURE 8

(Figure 9)

It was recognized that some accessory loads are influenced by inputs other than speed. The air conditioning system, alternator, power steering system, and cooling fan loads are all modulated to reflect driving cycle and ambient imposed conditions. This figure shows the modulated accessory combinations used for each driving cycle and type of accessory drive. The highest A/C, alternator, and fan loads are used in the slow speed, city driving cycles. These loads are proportionately modified to reflect suburban and highway or A/C-off driving conditions. The speed limited accessory system used a direct-drive fan, which deleted the need for a fan clutch, and also reflected a slightly improved, low-speed air-conditioning system.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

MODEL CODE NO.	TYPE OF ACCESSORY DRIVE SYSTEM	DRIVING CYCLE	A/C MODE	A/C DRIVE RATIO	FAN MODE	ALTERNATOR OUTPUT (AMPS)	TOTAL ACCESSORY POWER CONSUMPTION—HORSEPOWER				
							ENGINE SPEED—RPM				
							IDLE	1000	1500	2000	3000
1	STOCK	ALL	OFF	1.4	40% CLUTCHED	20	1.23	2.13	3.64	5.40	10.77
2	STOCK	FDC URBAN	MAXIMUM PULL DOWN	1.4	75% CLUTCHED	30	3.87	6.25	9.79	14.19	24.87
3	STOCK	HWFET SUBURBAN INTERSTATE	MATCH STEADY STATE LOAD	1.4	60% CLUTCHED	30	3.39	5.25	7.89	10.94	18.47
4	SPEED LIMITED	ALL	OFF	1.6	DIRECT DRIVE	20	1.34	2.40	2.40	2.40	2.40
5	SPEED LIMITED	FDC URBAN	MAXIMUM PULL DOWN	1.6	DIRECT DRIVE	30	4.14	6.75	6.75	6.75	6.75
6	SPEED LIMITED	HWFET SUBURBAN INTERSTATE	MATCH STEADY STATE LOAD	1.6	DIRECT DRIVE	30	3.59	5.60	5.70	5.75	5.80

ACCESSORY SYSTEM SIMULATION CONDITIONS

FIGURE 9

(Figure 9A)

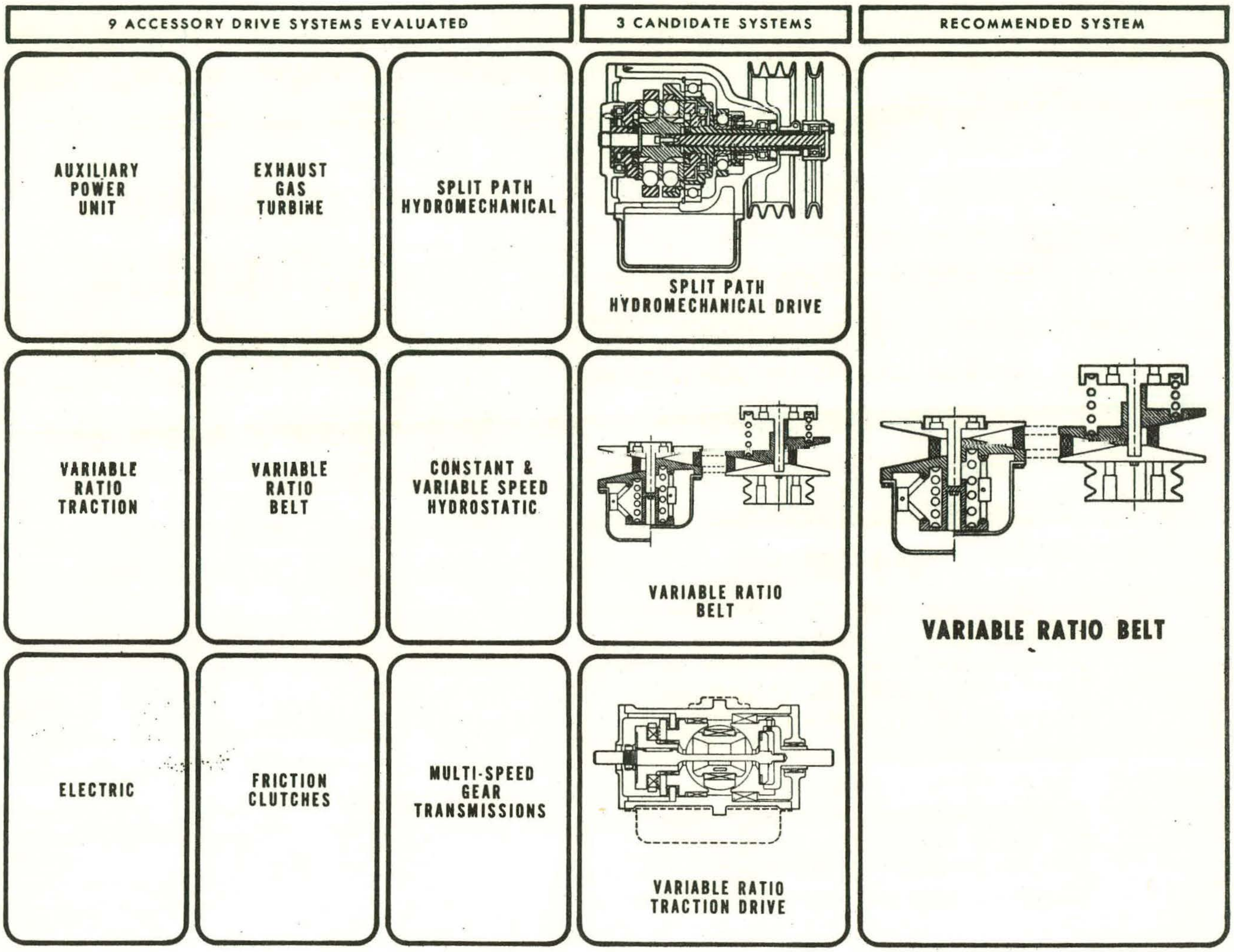
Accessory Drive Systems

Drive systems identified as having sufficient merit to warrant first-order engineering definition were selected in two categories:

Variable Speed Drive (and)
Auxiliary Power Units

Nine specific systems and various combinations were defined:

(1) APUs with Mechanical, Hydraulic, and Electric auxiliary drives
(2) Exhaust Gas Turbines - Both exhaust driven and engine drive augmented
(3) Split-Path - Hydromechanical
(4) Constant and Variable Speed Hydrostatic
(5) Variable Ratio Traction
(6) Variable Ratio Belt
(7) Electric
(8) Slip Clutches - Dry and Wet
(9) Multispeed Gear Transmissions



ACCESSORY DRIVE SYSTEMS

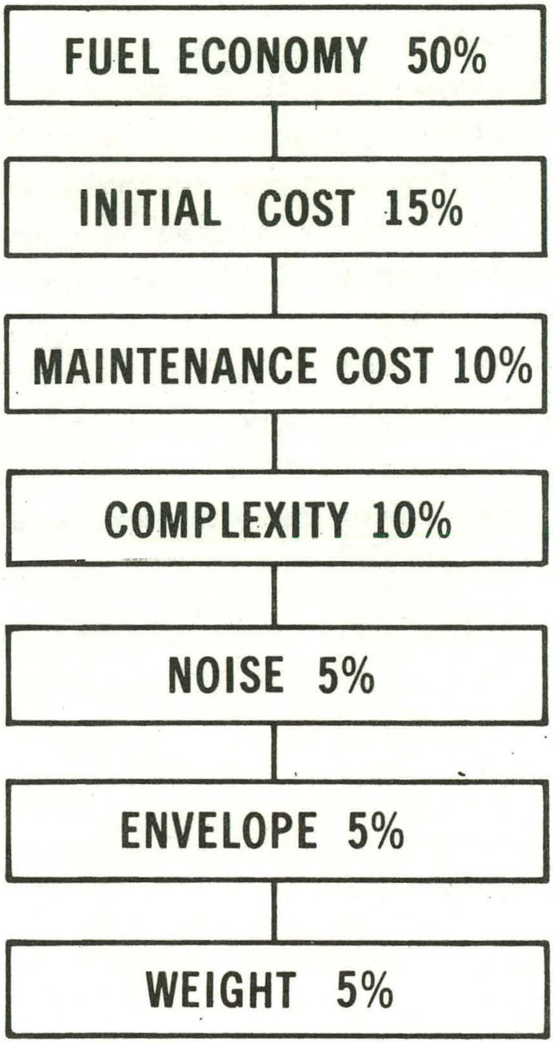
FIGURE 9A

(Figure 10)

Although the study primary goal is to improve fuel economy, this must be accomplished in a manner acceptable to both manufacturer and customer without resorting to extreme hardware sophistication. The drive-system trade-study criteria was developed to definitize study goals and align the candidate systems. A general design specification was also produced to define and establish limits for each criteria category.

The engineering evaluation of all drive systems assumed that a particular device could be mechanized to drive the accessories in their optimum speed range. This assumption permitted a normalized trade study criteria application. Three candidate systems evolved from the preliminary evaluation.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



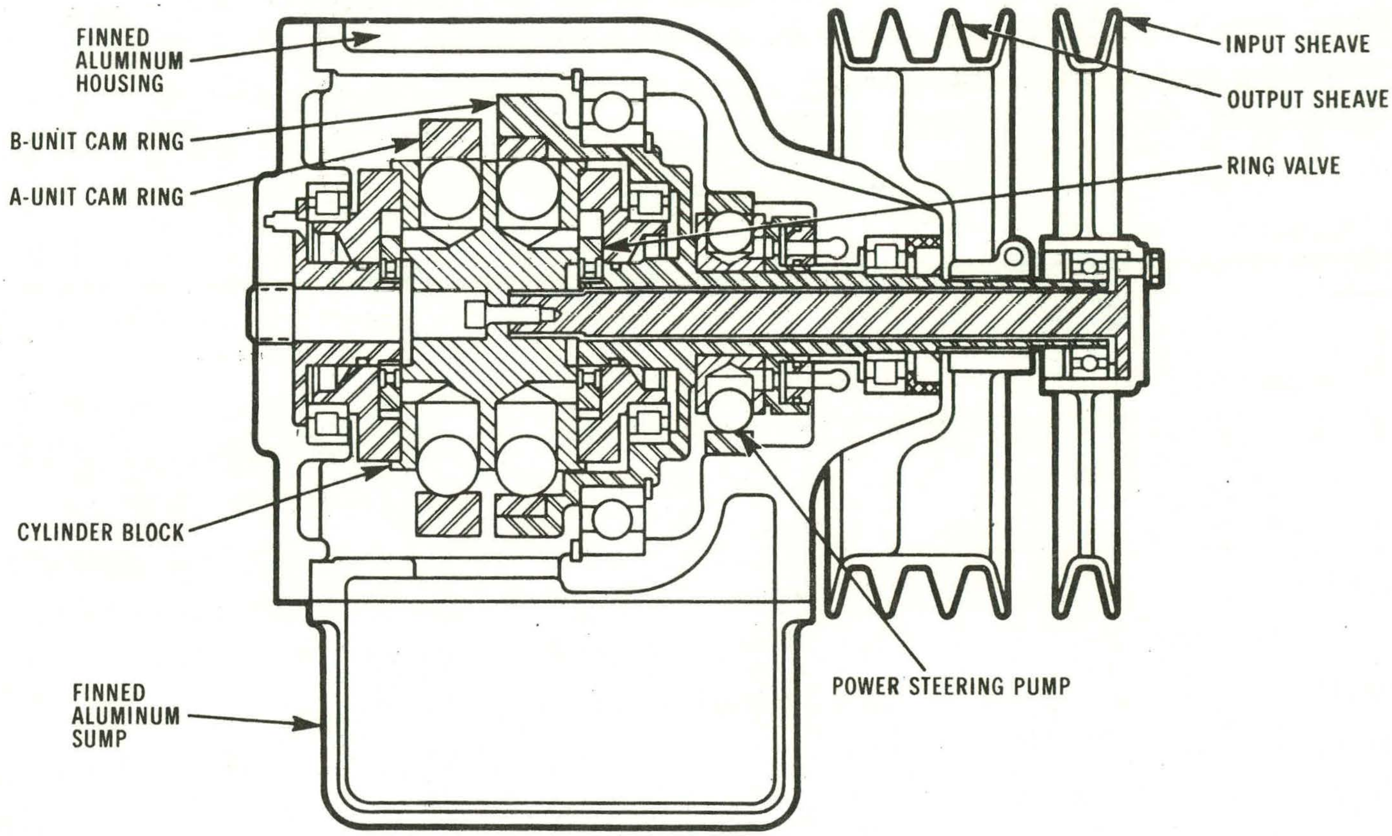
DRIVE SYSTEM TRADE STUDY CRITERIA

FIGURE 10

(Figure 11)

The split-path hydromechanical drive utilizes a compact ball-piston pump and motor with co-axial input and output shafts. The design exhibits good installation flexibility and excellent transient overload capability. The most redeeming feature is the integrated power steering pump that produces the speed control signal and actuation forces. The design is totally self-contained and replaces the existing power-steering pump.

IMPROVED AUTOMOTIVE ACCESSORY

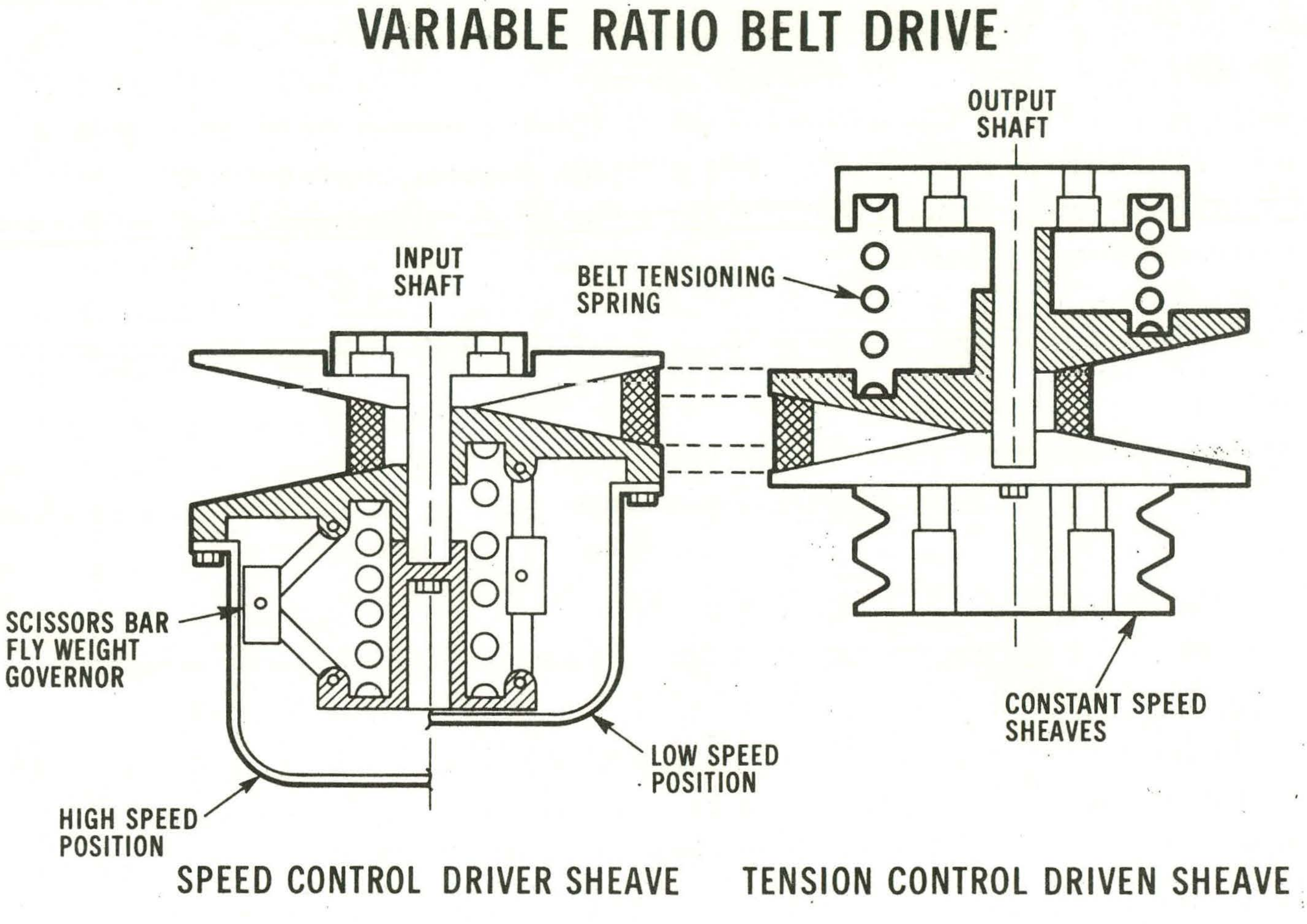


HYDROMECHANICAL SPLIT-PATH DRIVE

FIGURE 11

(Figure 12)

Variable-ratio torque and speed-control belt drives are currently produced in high volume for recreational vehicles and industrial applications. The concept shown here is controlled mechanically, but hydraulic and electrical control schemes may be applied. This concept exhibits low initial cost, conventional maintenance, low noise generation, minimum weight impact, good control characteristics, and excellent installation flexibility. The concept is proven and minimum development is required.



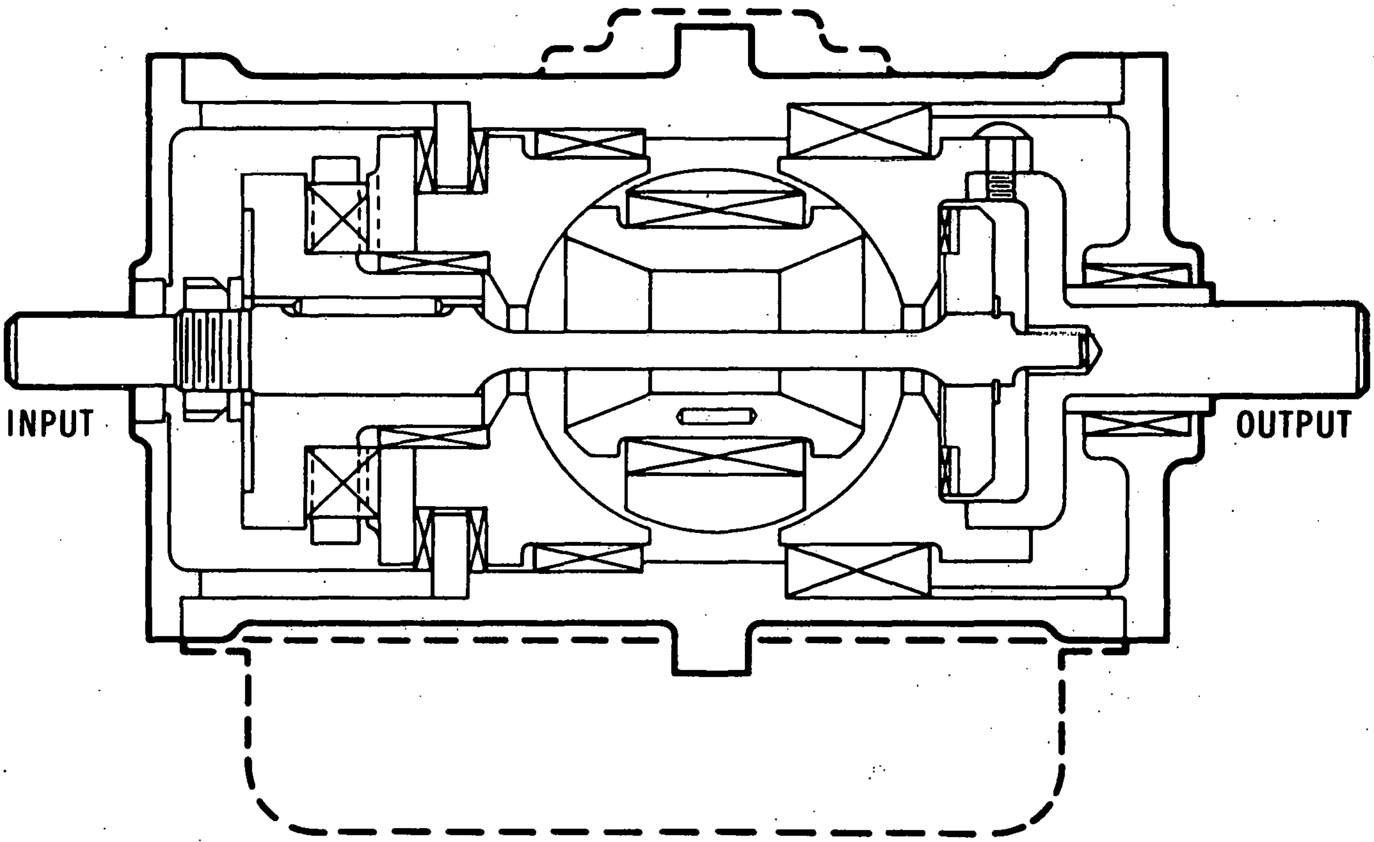
MECHANICAL SPEED CONTROL

FIGURE 12

(Figure 13)

Traction drive designs have been available for many years but only recently have advances in the disciplines of materials and lubricants allowed reasonable applications. The type shown is a variable-ratio spool drive. It is compact, lightweight, and has an excellent speed ratio range of about 6 to 1. Control characteristics and overload capability are good and initial cost would be moderate in high volume production.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



SPOOL TYPE TRACTION DRIVE

FIGURE 13

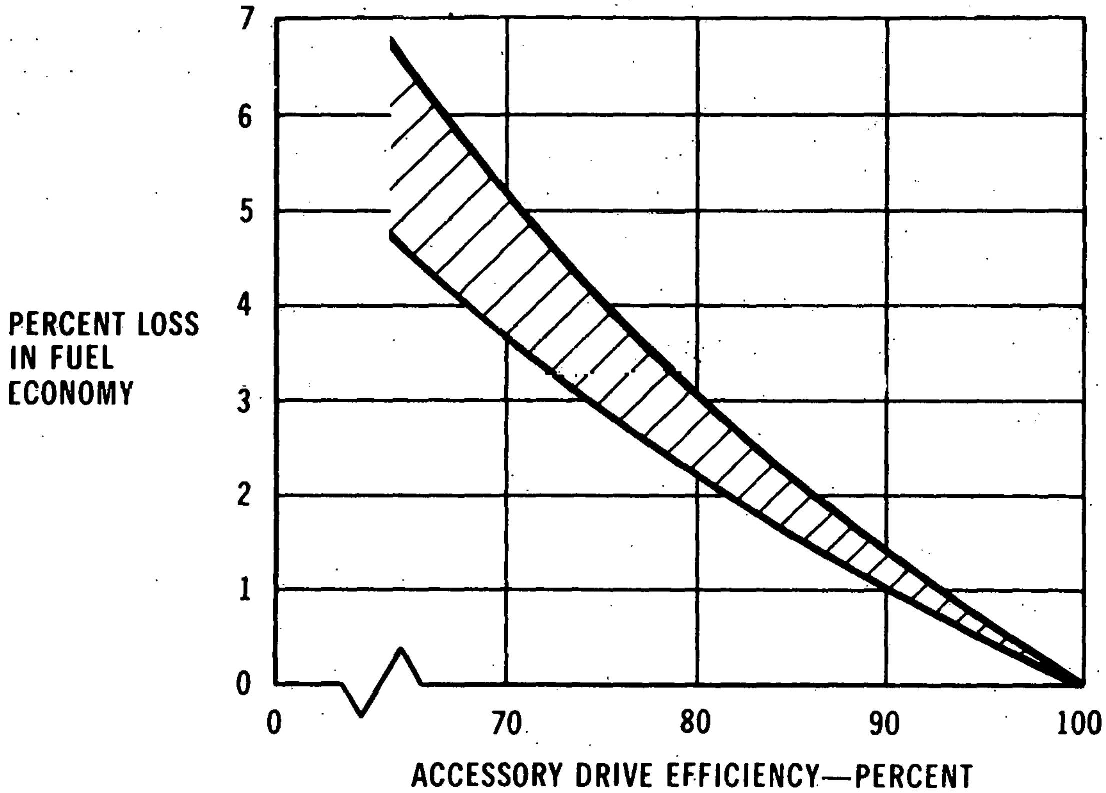
(Figure 14)

The computer model flexibility to perform parametric studies is demonstrated by this figure showing the relationship of accessory drive efficiency to vehicle fuel economy. The curve limits were set for all driving cycles with the maximum sensitivity occurring in the low-speed urban cycles.

The three drive candidates were evaluated for overall system efficiency, and on the bases of major trade-off criteria, fuel economy, the selection was reduced to:

Variable-ratio belt as the prime concept,
Variable-ratio traction as a close contender, followed by the
Split-path hydromechanical

EFFICIENCY
IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



FUEL ECONOMY SENSITIVITY TO DRIVE EFFICIENCY

FIGURE 14

(Figure 15)

Variable-ratio belt drive attributes are being defined in detail as the design phase of the study progresses. Drive efficiency is as high as conventional "V" belts; therefore, if the drive installation replaces one "V" belt, the total system efficiency approaches 100 percent of the standard fixed-ratio system. The critical secondary design considerations of initial cost, maintenance and complexity are inherently lower than the other drive candidates.

<u>Variable-Ratio Belt Drives</u>

Continuously variable ratio belt drives can be mechanized in three design groups:

(1) Dual-groove sheaves
(2) Paired-split sheaves, (and)
(3) Single-split sheave with idler take up

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

EFFICIENCY	90-95%
INITIAL COST	UNDER $40.00
COMPLEXITY	SIMPLE—EXISTING TYPES CURRENTLY IN PRODUCTION
MAINTENANCE COST	$2.00-3.00/YEAR
NOISE	OVERALL REDUCTION
INSTALLATION	BOLT-ON CONCEPT
WEIGHT	ADDITIONAL 8 LBS TO TOTAL ACCESSORY SYSTEM
LIFE—DRIVE BELT	7 YEARS 3 YEARS
DELETIONS	FAN CLUTCH, P/S HEAT EXCHANGER, AND FLOW LIMITERS
POLLUTION	DECREASE

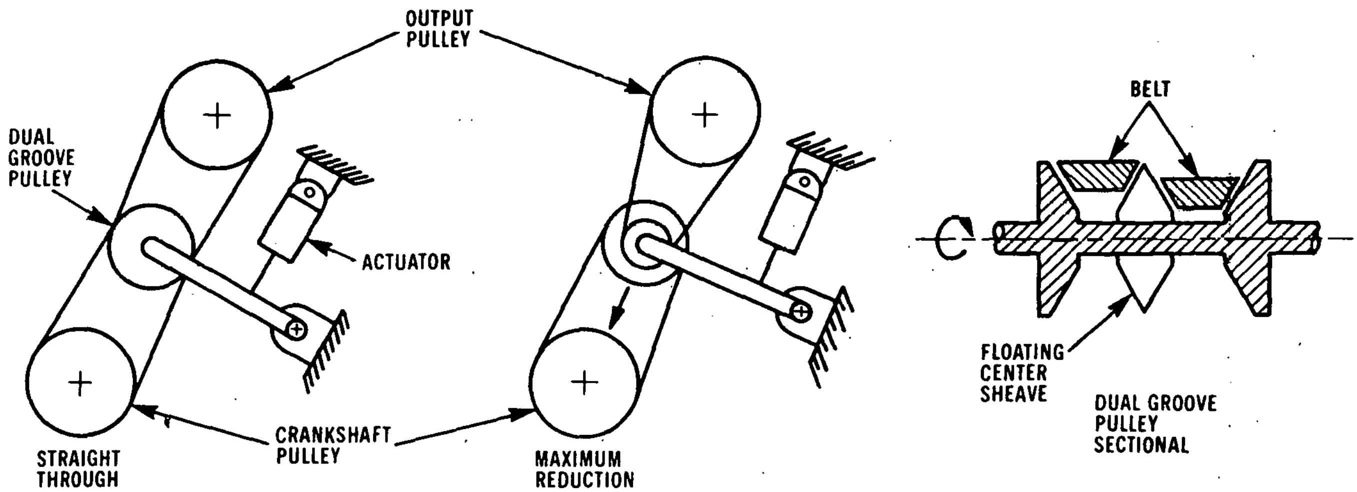
VARIABLE RATIO BELT DRIVE FEATURES

FIGURE 15

(Figure 16)

The dual-groove concept permits large ratio changes with small input movements. The actuation mechanism is external to the sheaves and may be operated mechanically, hydraulically, or electrically.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



"FEATURES"

(1) RATIO CHANGE RESULTS FROM TRANSLATION OF DUAL GROOVE PULLEY MOUNTING SHAFT
(2) INPUT AND OUTPUT PULLEYS ARE STANDARD FIXED GEOMETRY
(3) LARGE RATIO CHANGES POSSIBLE WITH SMALL INPUT MOTIONS
(4) CAN BE INSTALLED WITH MINIMAL CHANGES TO CONVENTIONAL ACCESSORY CONFIGURATIONS
(5) SPEED SENSING AND ACTUATION ALTERNATES
  (a) POWER STEERING PUMP FLOW
  (b) ALTERNATOR FREQUENCY-ELECTROMECHANICAL ACTUATOR
  (c) CENTRIFUGAL

DUAL GROOVE VARIABLE RATIO BELT CONCEPT

FIGURE 16

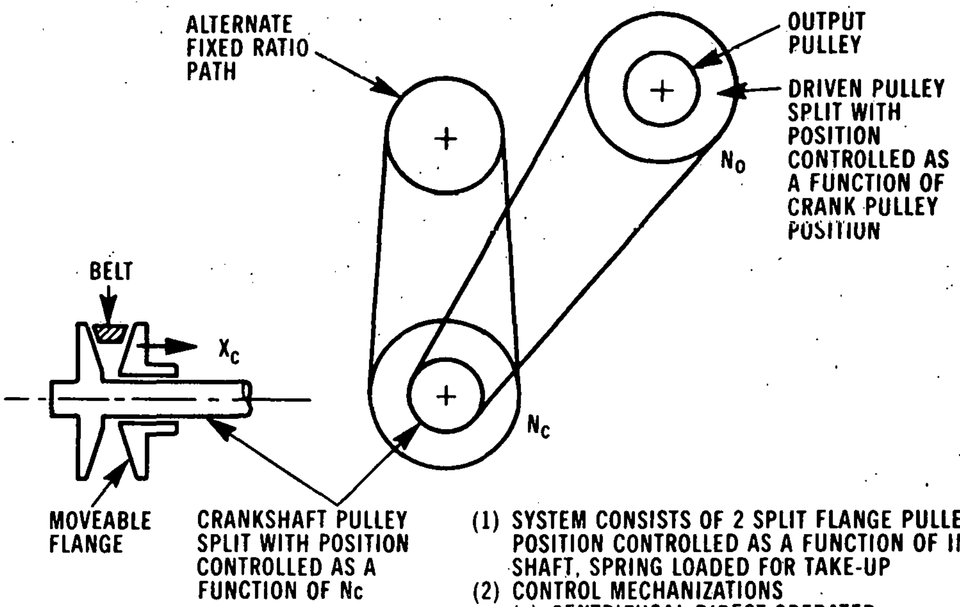
(Figure 17)

The paired-split-sheave design consists of two split-flange pulleys. Both sheaves have integral controls to maintain belt tension while the driver or crank-mounted sheave also produces ratio change through a speed-sensitive device.

The single split-sheave concept uses one split-flange pulley with an externally actuated idler to achieve ratio changes with belt take up.

Each concept is being evaluated for installation on the baseline vehicle. Current emphasis is on the paired-split-sheave design mounted on the crank and water pump or power steering pump shafts.

PROGRAMMED INPUT POSITION CONTROL



(1) SYSTEM CONSISTS OF 2 SPLIT FLANGE PULLEYS, ONE ON-CRANK WITH POSITION CONTROLLED AS A FUNCTION OF INPUT RPM AND ONE ON-OUTPUT SHAFT, SPRING LOADED FOR TAKE-UP
(2) CONTROL MECHANIZATIONS
  (a) CENTRIFUGAL DIRECT OPERATED
  (b) CENTRIFUGAL SERVOED WITH CRANK-FED LUBE OIL
  (c) CENTRIFUGAL SERVOED WITH BALL-SCREW ACTUATOR

PAIRED SPLIT PULLEY

FIGURE 17

(Figure 18)

Fuel Economy Pay-off

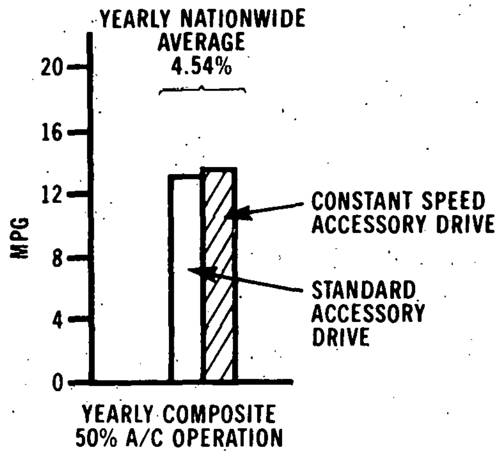
The baseline vehicle fuel economy, with and without the optimum accessory drive system, was computer predicted for all driving cycles and accessory load conditions. The table shows the percent increase in fuel economy due to the accessory drive. The various driving cycles show a range of fuel economy improvement from 2 to 12 percent depending on the average speed, number and magnitude of accelerations and decelerations and idle time for each cycle. The four SAE cycles were combined with percent-miles driven data generated in recent "chase car tests" to predict a yearly nationwide average improvement of 4.5 percent in fuel economy. This improvement represents a 33-gallon fuel savings for a 10,000-mile vehicle year.

Additional Improvements

The full accessory drive potential has not yet been realized. Additional improvements were made possible by near constant-speed accessories and excess accessory power conservation.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

DRIVING CYCLE		PERCENT IMPROVEMENT WITH CSD A/C OFF	PERCENT IMPROVEMENT WITH CSD A/C ON
FEDERAL (FDC)		1.16	3.18
FPA HIGHWAY FUEL ECONOMY TEST (HWFET)		3.67	6.36
SAE URBAN		0.78	1.96
SAE SUBURBAN		3.19	5.14
SAE INTERSTATE (55 MPH)		4.74	8.23
SAE INTERSTATE (70 MPH)		8.16	12.34
CONSTANT VELOCITY SEGMENTS (MPH)	30		0.52
	40		3.40
	50		5.66
	60		8.15
	70		9.66



COMPOSITE DISTRIBUTION
18.2% URBAN
35.5% SUBURBAN
40.7% INTERSTATE 55
5.6% INTERSTATE 70

ACCESSORY DRIVE FUEL ECONOMY ANALYSIS

FIGURE 18

(Figure 19)

Driving the accessories in a narrow speed range allows each accessory to be mechanized and sized to operate at or near maximum efficiency. Each accessory loss source was analyzed for possible obvious areas of improvements. It was concluded that a general overall accessory system efficiency improvement of 15 to 20 percent could be achieved with relatively minor modifications.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

ACCESSORY	LOSS SOURCE
ALTERNATOR	WINDAGE, BEARING, BRUSH & ELECTRICAL
FAN	GEOMETRY, SHROUD CLEARANCE
AIR CONDITIONING COMPRESSOR	PUMPING LOSSES, DUTY CYCLE
POWER STEERING PUMP	FLOW CONTROL ΔP
WATER PUMP	ALL EFFICIENCY IMPROVEMENTS RELATED TO INCREASING LOW SPEED FLOW
AIR PUMP	BEARINGS, PORTING, LEAKAGE

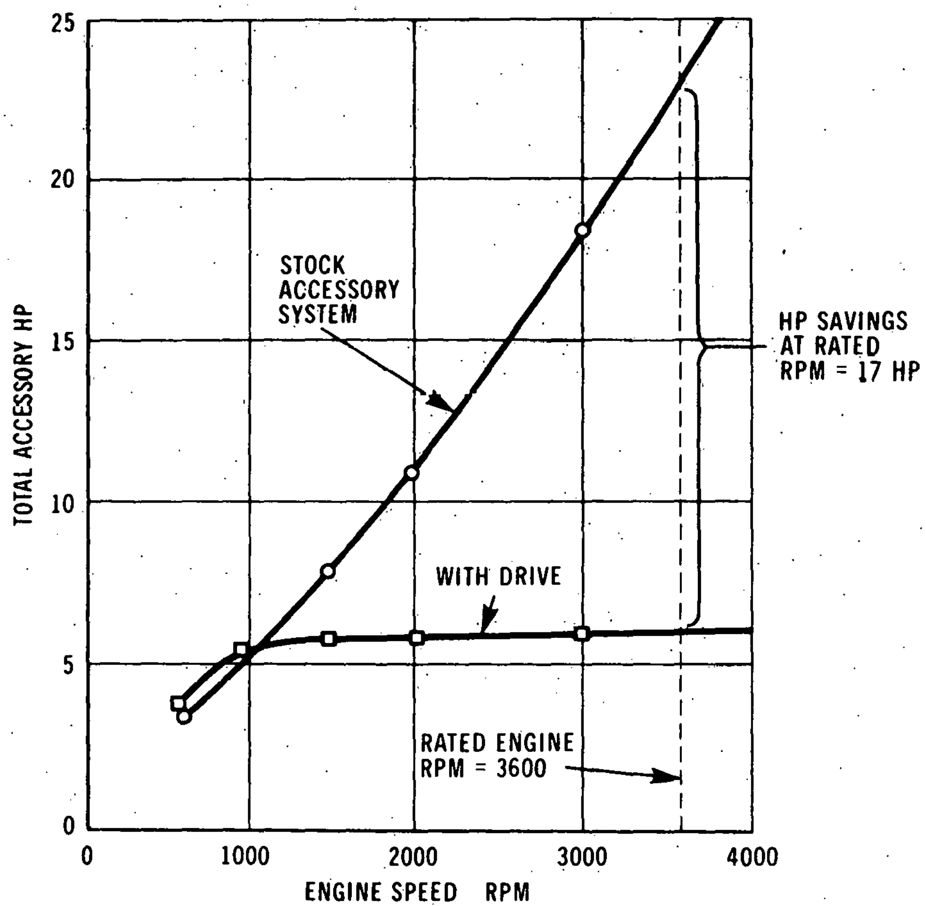
IMPROVING ACCESSORY EFFICIENCY

FIGURE 19

(Figure 20)

The accessory drive system conserves considerable power through most of the engine speed range. At rated engine speed, the power savings is in excess of 17 hp.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM



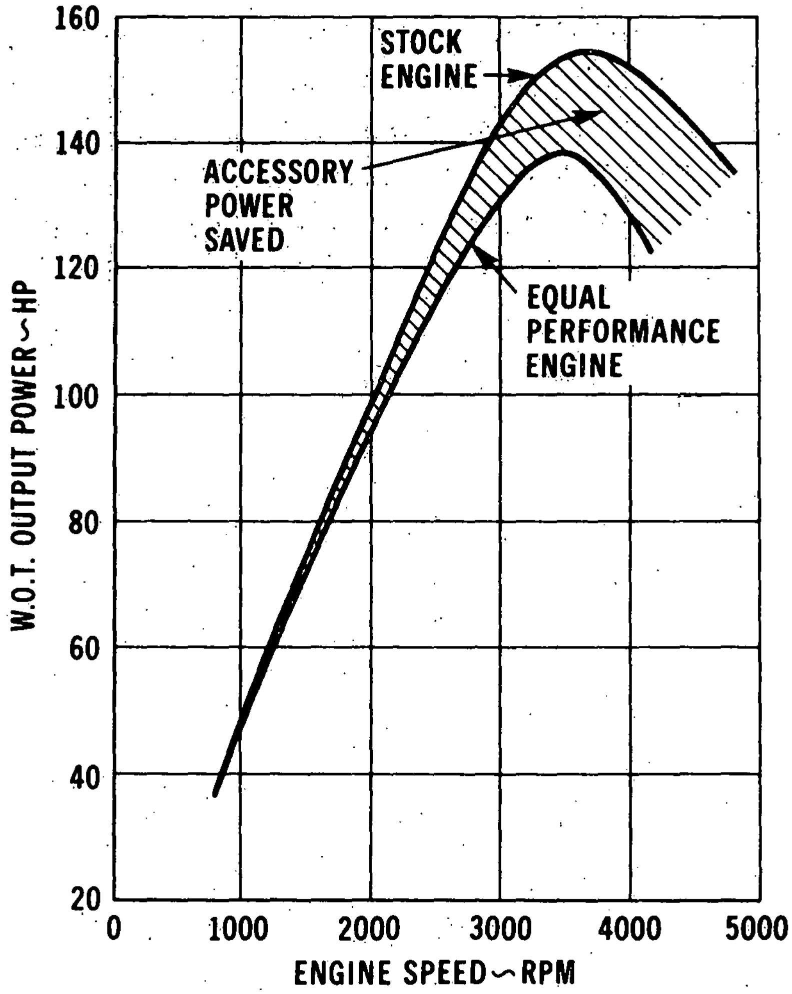
EXCESS HORSEPOWER

FIGURE 20

(Figure 21)

The power saved at any engine speed must now be considered an excess that is available for improving vehicle acceleration margin. To achieve an "equal performance" vehicle, the baseline engine wide-open-throttle power output must be reduced by an amount equal to the power saved by the accessory drive. The difference in W.O.T. power between the baseline and equal performance engines is approximately 10 percent and a like change in engine displacement would produce approximately the desired power reduction.

IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

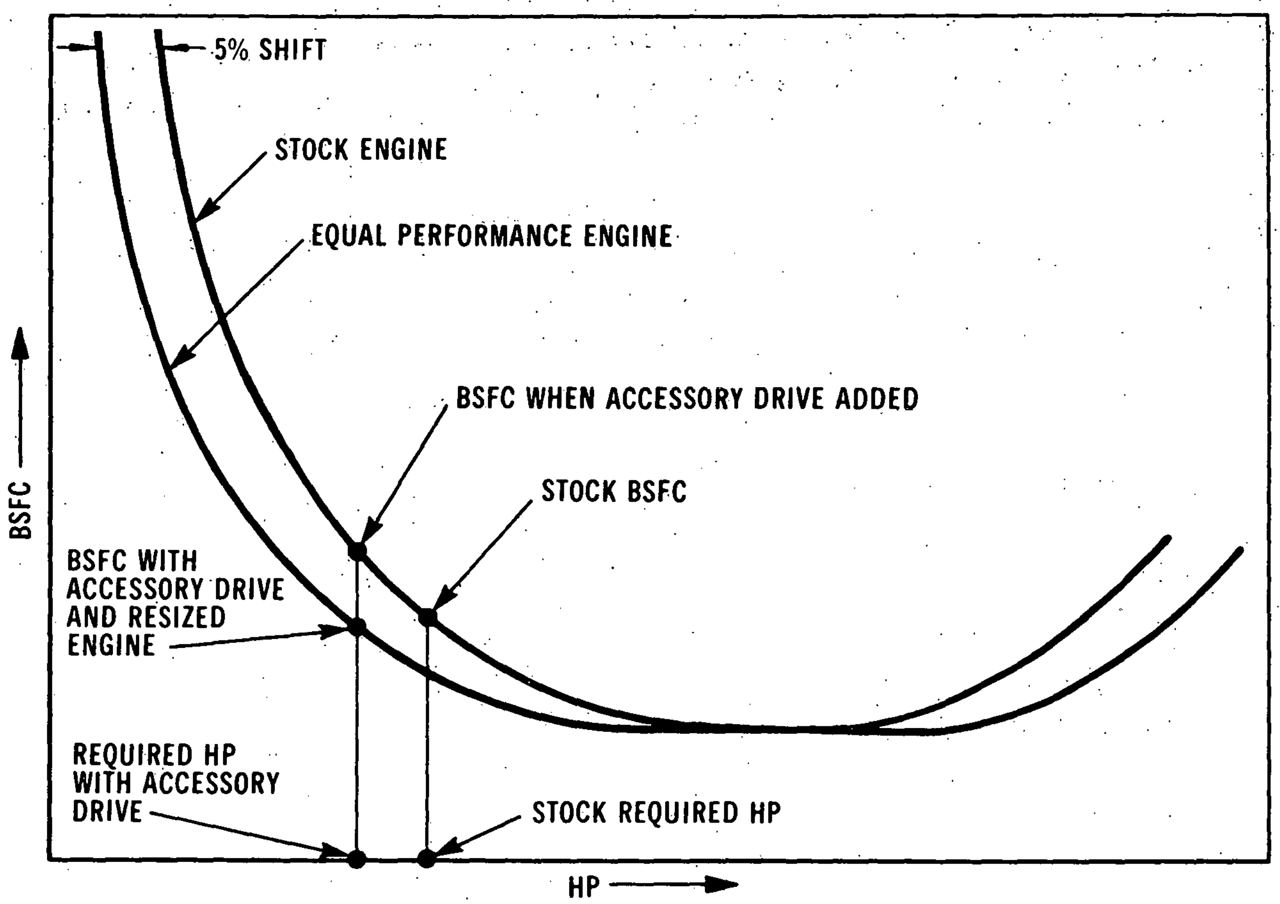


EQUAL PERFORMANCE ENGINE

FIGURE 21

(Figure 22)

The relationships between small changes in engine displacement and part-throttle fuel economy are not on a one-to-one ratio; therefore, the baseline engine BSFC map was modified using only a 5 percent shift in hp. In effect, this produces a smaller engine and regains a major portion of the BSFC lost when the engine was unloaded by the accessory drive.



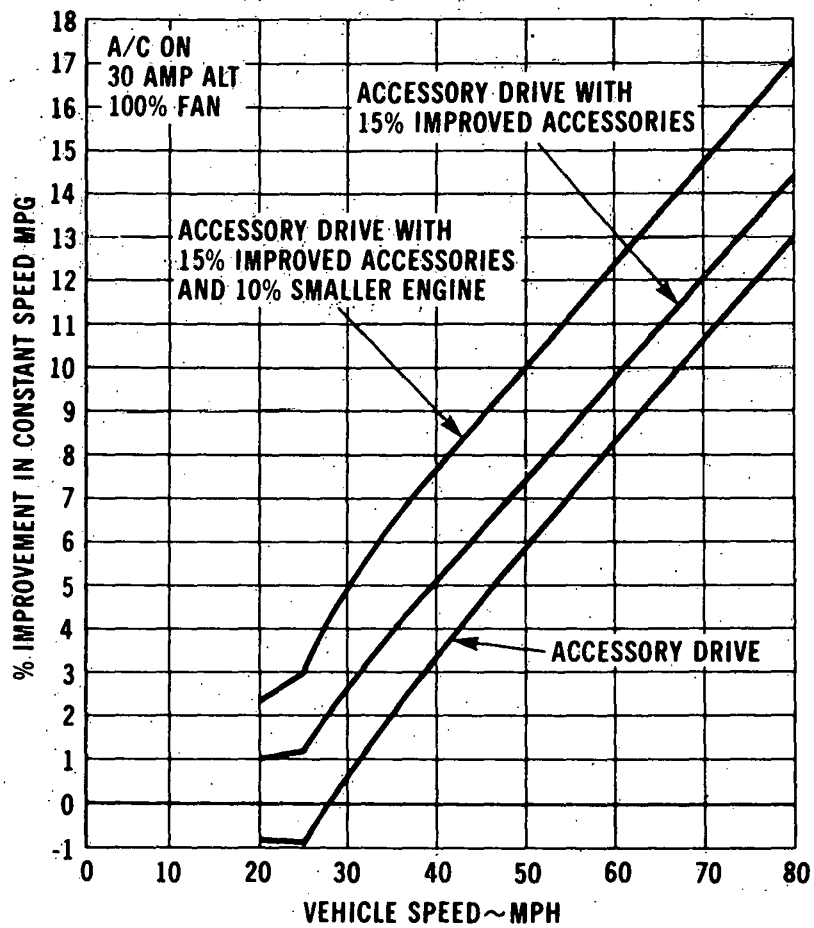
RESIZED ENGINE BSFC

FIGURE 22

(Figure 23)

The computer model was modified to predict fuel economy improvements attributed to the compound effects of an accessory drive, 15 percent improved accessory efficiency and a 10 percent resized engine. The vehicle cruise-speed analysis shown in this figure demonstrates the full potential of accessory drive systems.

DUE TO ACCESSORY DRIVE WITH IMPROVED ACCESSORIES AND RESIZED ENGINE



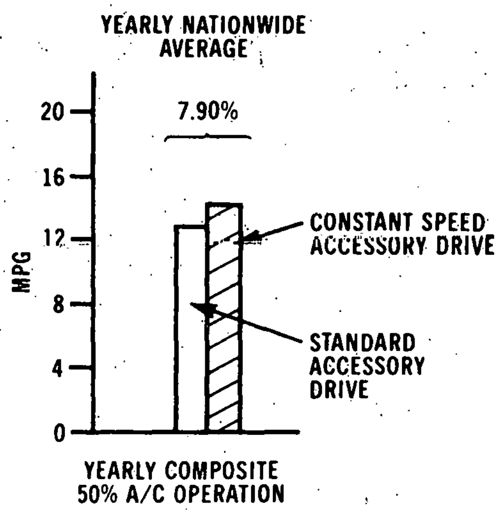
COMBINED FUEL ECONOMY IMPROVEMENT

FIGURE 23

(Figure 24)

Driving-cycle fuel-economy comparisons of the baseline and fully improved vehicle were computed in the same manner as described previously. The various driving-cycle fuel-economy improvements show a considerable change from 2 to 12 percent to 7 to 17 percent. The yearly nationwide average fuel economy was recalculated at 7.9 percent improvement, as compared to 4.5 percent for the accessory-drive-only example. The 10,000-mile yearly fuel savings increased from 33 to 56 gallons.

DRIVING CYCLE		PERCENT IMPROVEMENT	
		A/C OFF	A/C ON
FEDERAL (FDC)		3.66	8.97
EPA HIGHWAY FUEL ECONOMY TEST (HWFET)		6.90	10.77
SAE URBAN		1.91	7.05
SAE SUBURBAN		5.41	8.46
SAE INTERSTATE (55 MPH)		8.08	12.90
SAE INTERSTATE (70 MPH)		12.23	17.67
CONSTANT VELOCITY SEGMENTS (MPH)	30		4.84
	40		7.56
	50		9.71
	60		12.53
	70		13.62



COMPOSITE DISTRIBUTION
18.2% URBAN
35.5% SUBURBAN
40.7% INTERSTATE 55
5.6% INTERSTATE 70

RESIZED ENGINE FUEL ECONOMY ANALYSIS

FIGURE 24

(Figure 25)

Follow-on

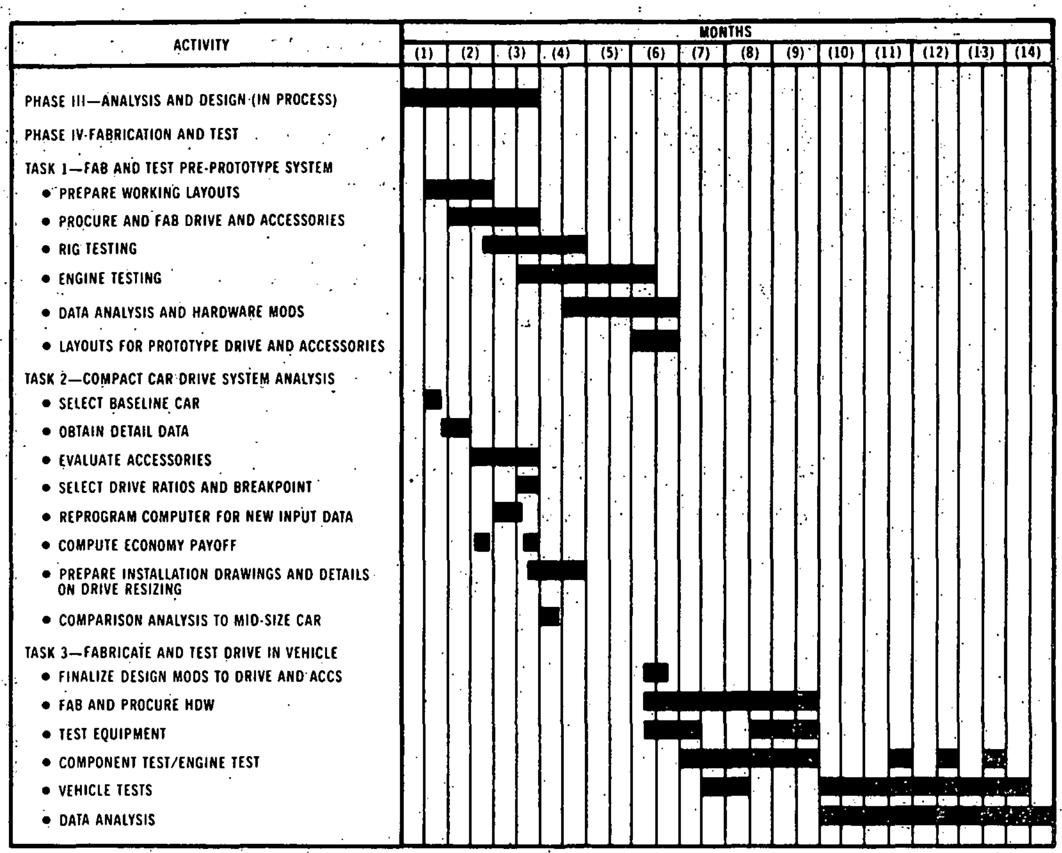
The follow-on program phase consists of 3 major tasks:

(1) Pre-prototype fabrication, bench, and engine testing of an accessory drive system based on current study efforts.

(2) A compact car accessory analysis conducted in the same manner used for the intermediate vehicle with design mods to the pre-prototype system for small car applications.

(3) Vehicle testing with the finalized prototype drive and accessory hardware.

AUTOMOTIVE ACCESSORY DRIVE SYSTEM DEVELOPMENT



PHASE IV MASTER PROGRAM SCHEDULE

FIGURE 25

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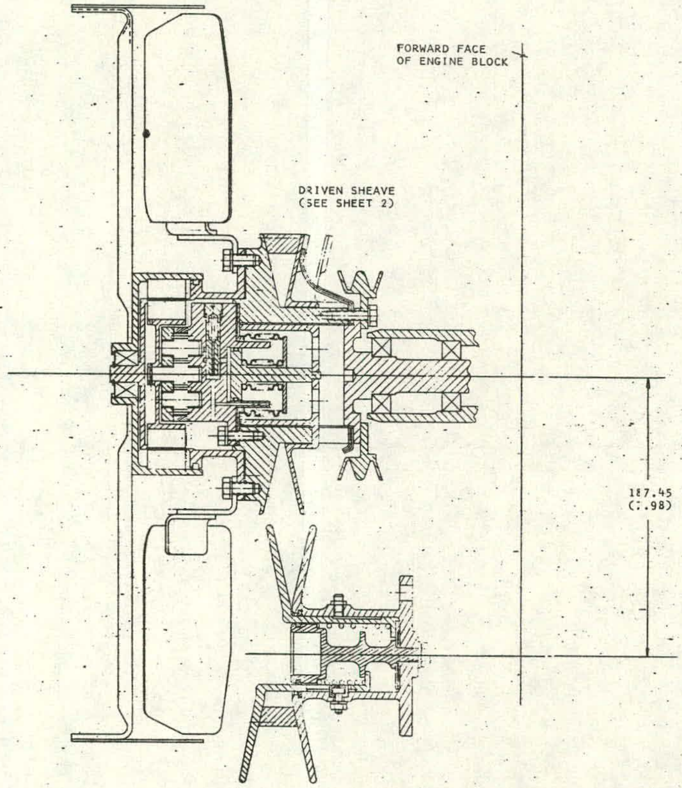
REVISION STATUS OF SHEETS

REV									
SHEET									

REVISIONS

ZONE	LTR	DESCRIPTION	DATE	APPROVED
	A	ADDED W.P. BEARINGS, SEAL, WIPER & LUBE FITTING TO DRIVER	7-23-75	



QUANTITY REQD	FIND NO	PART NO OR IDENTIFYING NO.	SYM	NOMENCLATURE OR DESCRIPTION	CODE IDENT	MATERIAL AND SPECIFICATION	ZONE

← PART NUMBER

PARTS LIST

REQD	NEXT ASSY	USED ON

HEAT TREATMENT — HARDNESS AND SPEC

PROCESS — NAME AND SPEC

SIGNATURES	DATES
DFT [signature]	7-1-75
CHK	
ENG	
MATL	
STRESS	
AERO	
APVD [signature]	7-1-75
APVD	
DESIGN ACTIVITY APVD	
OTHER ACTIVITY APVD	

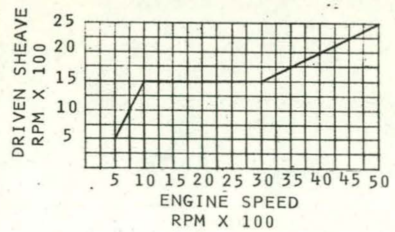
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

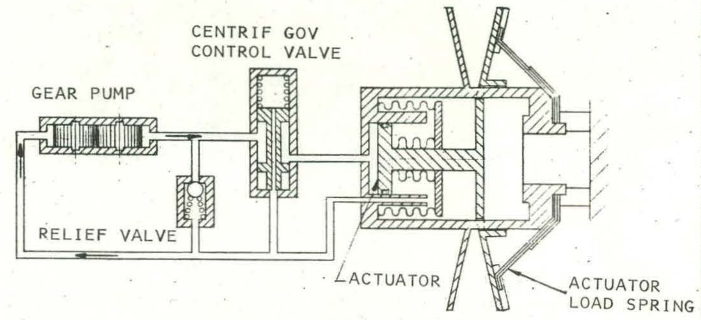
DWG TITLE
LAYOUT
ACCESSORY DRIVE SYSTEM,
VARIABLE RATIO

SIZE E | CODE IDENT NO. 99193 | DWG NO L-3621560

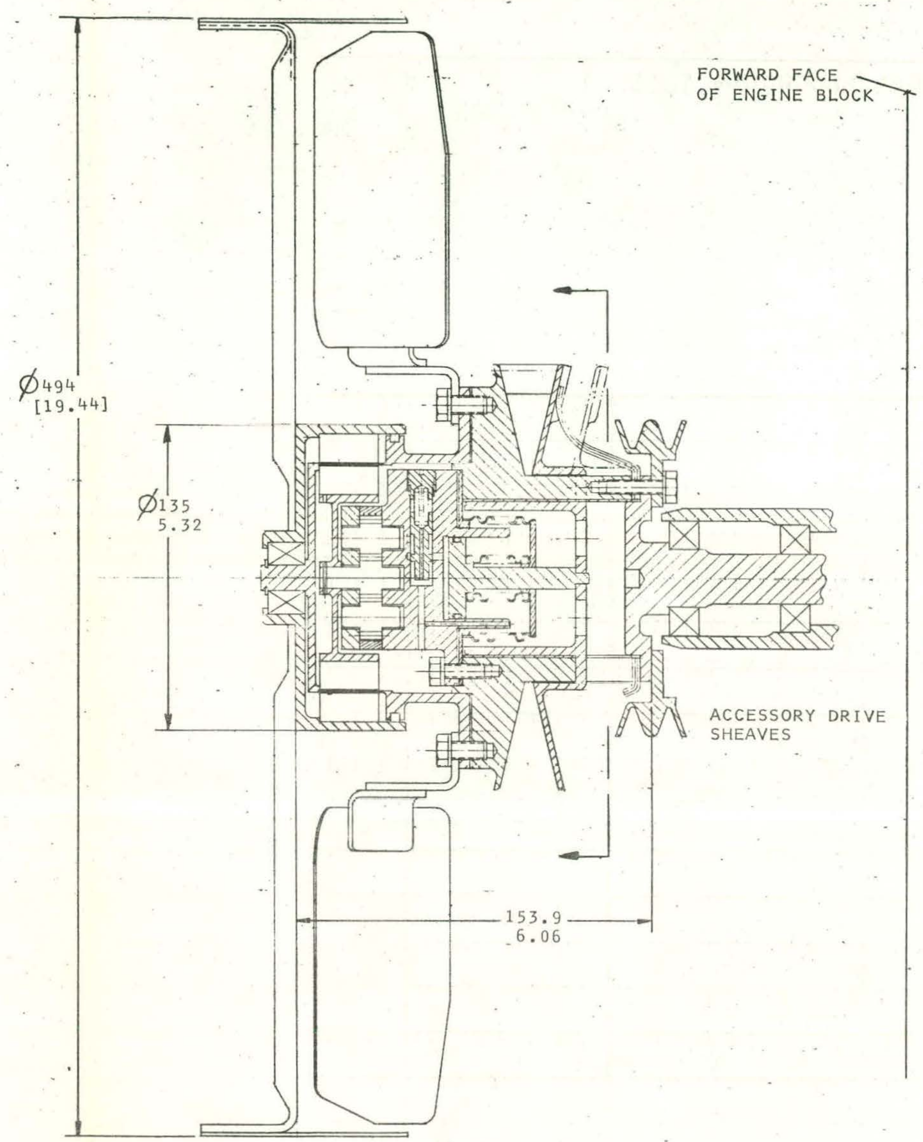
SCALE 1/1 | WT | SHEET 1 OF 3

L-3621560

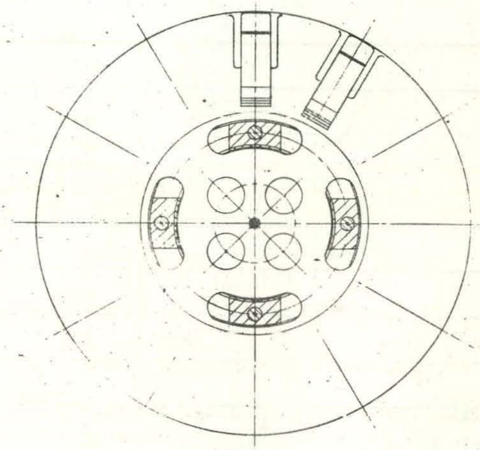




HYDRAULIC SCHEMATIC



DRIVEN SHEAVE
HYDRAULICALLY ACTUATED



PERMANENT MAGNET DRIVE

POLES/RING	10 EACH
MAX TORQUE	0.565 N-M (5 LB-IN.)

FLUID

STD TRANSMISSION FLUID PER CAR MFR RECOMMENDATIONS (HERMETICALLY SEALED).

HYDRAULICALLY DRIVEN SHEAVE

BELT PITCH DIA	111.25 MM (4.38 IN.) MIN 182.37 MM (7.18 IN.) MAX
INCLUDED SHEAVE ANGLE	26°

GOVERNOR CONTROL VALVE

GOVERNOR CONTROL SPEED	1500 RPM
CONTROL SENSE	CENTRIFUGAL FORCE

RELIEF VALVE

CRACKING PRESSURE	1724 ±172 KPA (250 ±25 PSI)

PUMP

VOLUME	1.278 CM^3 (1893 L/MIN) (0.078 $IN.^3$ - 0.5 GPM) AT 1500 RPM
TORQUE	0.384 N-M (3.4 LB-IN.) AT MAXIMUM RELIEF PRESSURE 0.249 N-M (2.2 LB-IN.) AT MAXIMUM WORKING PRESSURE
GEAR WIDTH	8.712 MM (0.343 IN.)
GEAR P.D.	16.942 MM (0.667 IN.) (DOUBLE ELEMENT)

ACTUATOR

PISTON AREA	9.677 CM^2 (1.50 $IN.^2$)
PISTON STROKE	16.459 MM (0.648 IN.)
SPRING LOAD	676-1192 N (152-268 LBS)
SPRING RATE	31.35 N/MM (179 LB/IN.)

SPECIFICATION

REVISION STATUS OF SHEETS

REV									
SHEET									

REVISIONS

ZONE	LTR	DESCRIPTION	DATE	APPROVED

QUANTITY REQD	FIND NO.	PART NO. OR IDENTIFYING NO.	SYM	NOMENCLATURE OR DESCRIPTION	CODE IDENT	MATERIAL AND SPECIFICATION	ZONE

← PART NUMBER

PARTS LIST

REQD	NEXT ASSY	USED ON
HEAT TREATMENT HARDNESS AND SPEC	PROCESS NAME AND SPEC	

SIGNATURES	DATES
DFT H J May	7-1-75
CHK	
MFG	
MATL	
STRESS	
AERO	
APVD [signature]	7-3-75
APVD	
DESIGN ACTIVITY APVD	
OTHER ACTIVITY APVD	

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
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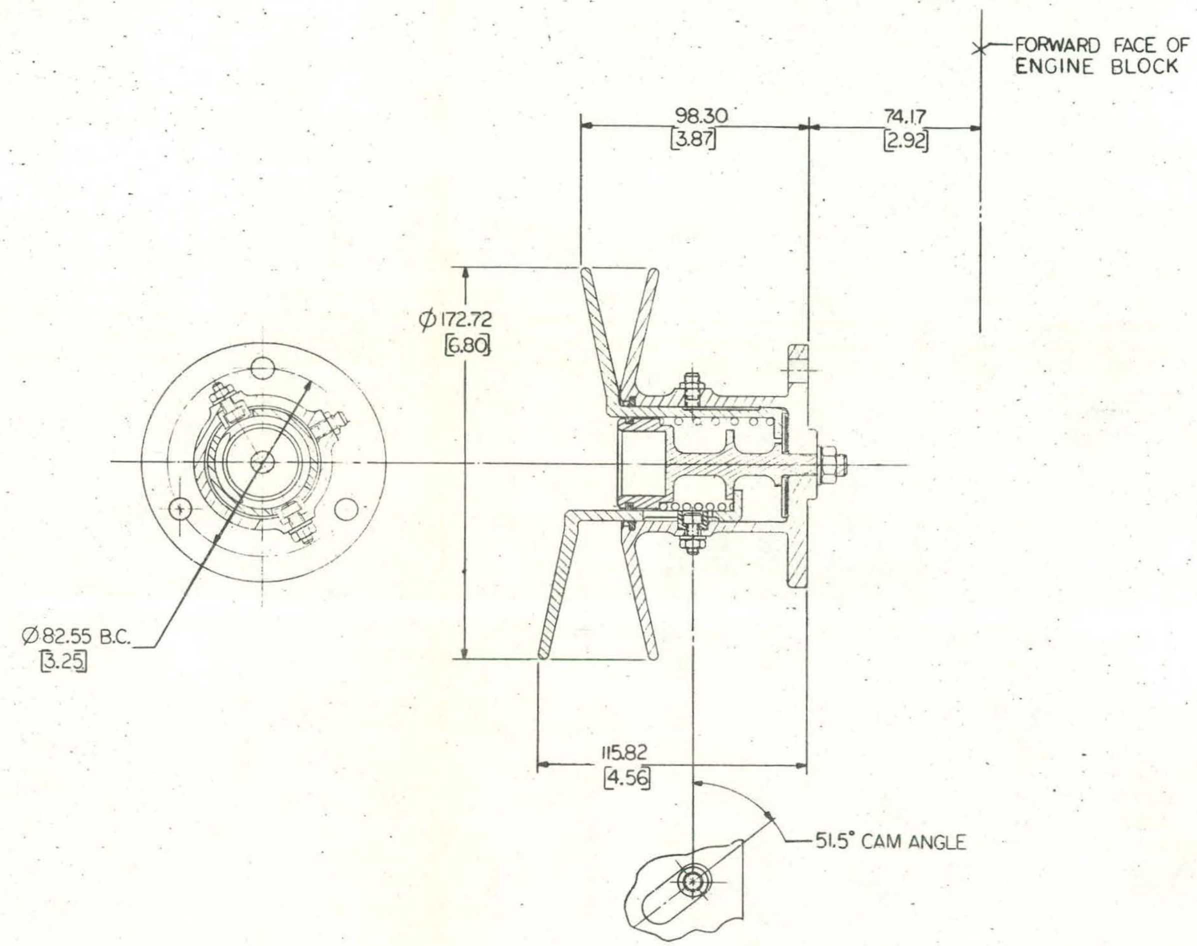
PHOENIX, ARIZONA

DWG TITLE
LAYOUT
ACCESSORY DRIVE SYSTEM,
VARIABLE RATIO

SIZE	CODE IDENT NO.	DWG NO
E	99193	L-3621560

SCALE 1/1 | WT | SHEET 2 OF 3

L-3621560



BELT PITCH DIAMETER:

166.878 MM [6.57] MAXIMUM
91.186 MM [3.59] MINIMUM

OPERATING SPEED:

0-5000 RPM

OPERATING TORQUE:

0-101.685 N-M
[0-75 LB-FT]

CAM SPRING:

LOAD 222.41 253.54N
[50-57 LB]

RATE: 1.75 N/MM
[10 LB/IN.]

SHEAVE TRAVEL:

0-17.526 MM
[0-0.69 IN.]

TORQUE SENSING
DRIVER

SEE SHEET 1 FOR CONTROLLING REV LTR

SIZE	CODE IDENT NO.	DWG NO.
E	99193	L-3621560
SCALE 1/1	REV	SHEET 3 of 3

L-3621560