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PROGRESS REPORT

EIGHTH QUARTERLY
PROGRESS REPORT
STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

AC0376CS51095-T5

74-310860(24)

July 31, 1976

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PROGRESS REPORT
STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

74-310860(24)

July 31, 1976.

Prepared by: Engineering Staff/RJH

Approved by: C. J. Fishap

C.L. Kevan, Supervisor
Documentation and Data Management

A.D. Rottler
A.D. Rottler, Program Project
Engineer

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EIGHTH QUARTERLY
PROGRESS REPORT
STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

1.0 INTRODUCTION

This is the eighth 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 April 16, 1976 through July 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 current 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).

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Although the primary goal is fuel economy improvements, 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 selecting a variable ratio belt drive as the speed control mechanism.

Phase IV consists of the following development tasks:

- Task 1 - Design, fabrication, component, and laboratory engine tests of prototype variable ratio belt drive.
- Task 2 - Baseline compact car computer modeling and computer drive cycle analyses utilizing the accessory drive system.
- Task 3 - Accessory drive integration with the baseline vehicle for system evaluation. Final drive system development will be an iterative process with the vehicle integration tests.

Tasks 1 and 2 are essentially complete.

1.1 Contents

Major technical accomplishments during this reporting period were:

- o The test vehicle was successfully driven to and from Ann Arbor, Michigan from Phoenix with the drive system installed.
- o The program status was formally presented at the ERDA Contractors' Coordination Meeting in Ann Arbor on May 5, 1976.

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- o Accessory drive hardware was calibrated, disassembled and reviewed for wear after round trip from Phoenix to Ann Arbor, Michigan.
- o MADS governor performance was analyzed and parameters were varied to demonstrate flexibility available in modifying accessory drive input/output characteristics.
- o A task list that matched smaller, lower cost accessory components, with the accessory drive equipped vehicle, was prepared. Major emphasis during the remainder of the program will be in this area.

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2.0 TECHNICAL PROGRESS

2.1 ERDA Presentation

The test vehicle and associated test data was presented at the ERDA Contractors' Coordination Conference held in Ann Arbor, Michigan, May 4 through May 6, 1976. This effort involved approximately 800 miles of track and road testing in Arizona for purposes of documenting fuel economy, and performance characteristics of various accessory systems for both stock and modified configurations.

The Mustang II test vehicle was driven uneventfully to and from the conference with the mechanical accessory drive installed. Travel distance was 4264 miles. Fuel economy (tank mileage) was 21.9 mpg which compares to 22.0 obtained during track testing and is an 8.4 percent improvement over stock. Vehicle and component hardware for both accessory drive versions were on display during the three-day conference.

Attendance at this Conference afforded an excellent opportunity to determine industry response to the program and gather data that will be applied to configuring the program for maximum impact and usefulness to the Industry.

The most significant conclusion that could be drawn from the large number of inputs received during the conference is the fact that acceptance of accessory drives depends on more than improved fuel economy demonstrations. A high level of interest was expressed in the possibility that accessory output and life can be improved, certain components can be deleted, and that by proper matching of the drive and accessories, the overall system cost can be less than a stock arrangement.

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The program status and results were presented on May 5, 1976. A copy of the presentation is included as Appendix I to this report and summarizes all recent data obtained on vehicle and accessory drive system performance.

2.2 Accessory Drive Hardware Inspection

The mechanical accessory drive, P/N 3705170, was calibrated, disassembled and inspected after the 4264 mile round trip to Ann Arbor. In addition, approximately 8 hours of operation at curb idle was accumulated at Ann Arbor for functional demonstration.

The accessory drive system configuration was as follows:

- o S/N 43 main drive belt supplied by Goodyear (3/4 inch wide, notched, with Kevlar cords)
- o Grease packed, bronze, sheave guide bushing in control (crankshaft) pulley
- o Nonlubricated, graphite filled Vespel (SP-21) bushing in take-up (waterpump) pulley
- o All accessories and pulleys were stock configuration with the following exceptions:
 - (a) Power steering oil cooler removed
 - (b) Stock five-blade flex fan (16-1/2-inch OD, 2-inch pitch) replaced with four-blade, fixed-pitch fan (16-inch OD, 1-3/8-inch pitch)

Calibration data on drive performance obtained prior to and after the trip is plotted in Figure 1. The drive was removed and disassembled for inspection after calibration. Findings were:

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ACCESSORY DRIVE CALIBRATION
P/N 3605170 MADS S/N 1

FIGURE 1

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- o Drive belt visually acceptable but badly oil soaked due to a crankcase oil leak through the control pulley center. The oil leak was corrected by installing an O-ring seal on the tie bolt.
- o Bronze control pulley bushing was in excellent condition.
- o Take-up pulley dry plastic bushings in essentially new condition with original machining marks still visible in plastic.
- o Pin joints in flyweight assembly were dry but unfretted.

All parts were cleaned and reassembled for further vehicle development. The belt was sent back to the fabricator for documentation of condition and evaluation of expected total life.

2.3 MADS Log Summary

MADS design verification and development testing in the Mustang II vehicle has accumulated 11,897 Km (7394 miles) from February 18 to July 15, 1976. The distribution is:

Ann Arbor Trip	60%
Test Track	10%
Highway Test	15%
Grade Test	3%
Urban Test	12%

During this period the vehicle was also returned to stock configuration and track tested for 483 Km (300 miles) to verify baseline performance.

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The MADS drive belt usage summary is as follows:

<u>Belt S/N</u>	<u>Distance Traveled</u>
41	2394 Km (1488 mi)
43	8307 Km (5163 mi)
44	1196 Km (743 mi)

Belt S/N 41 is currently installed on the MADS. Belt S/N 43 was removed after the Ann Arbor trip due to an oil soaked condition caused by a crankcase leak. Belt S/N 44 broke due to extreme overtensioning and overheating caused by an accidental fan/cranksheave interference. The interference condition was an assembly error and has been corrected.

2.4 Accessory Drive Ratio Modification

The drive ratio characteristic is controlled by the direct operated centrifugal governor in the crankshaft pulley. This control mechanism is basically an open loop programmer positioned according to the balance of three forces; the belt induced splitting force, the reference spring force, and the governor axial component force.

Operation is based on force and speed relationships of both the crankshaft-mounted driver sheave and water pump-mounted driven sheave. Actual hardware dimensions, weights, and measured spring characteristics are used in formulating the following analysis. A cross-section diagram of the MADS is shown in Figure 2.

The governor force applied to the movable sheave (FA) may be calculated for any given sheave position at any rotational speed. This is accomplished by summing the centrifugal force effects of the flyweights, links and pins, and resolving the summation at the movable sheave attach point to obtain FA. The governor system components

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MECHANICAL DRIVE CROSS SECTION

FIGURE 2

geometry, weights, and center-of-gravity (CG) were determined at assembly and a mathematical system description was prepared. Analysis results, where FA has been determined for various sheave positions (τ) over the total engine speed range (N_c) is shown in Figure 3.

Relationships of ratio and sheave travel to engine speed required to obtain the design speed profile are shown in Figure 4. The position/speed dictates in this figures have been cross-plotted on Figure 3 to show the design governor operating line.

The data from Figure 1 presented the actual speed profile attained with the MADS, and was cross-plotted with Figure 4 data to prepare Figure 5 showing the magnitude of off-design ratio control. The difference between actual and design ratio control may be expressed in terms of speed variation at a specific sheave position and used to describe off-design governor operation as shown in Figure 6.

With actual and design governor operation defined, it is possible to describe performance variations in terms of absolute forces required to correct variations. The forces may be modified by spring redesign or changes to the governor mass and linkage system.

The original design target was governor motion initiation (breakpoint) at 1200 rpm crankshaft speed with a predetermined ratio change beyond this speed that would result in a constant output speed up to the mechanism saturation point (3600 rpm crankshaft speed). The hardware, as initially configured, exhibited a breakpoint in the range of 800 rpm and a slight positive slope in the control range; Curve A of Figure 7. The analysis proves (see Figure 6) that the spring preload was insufficient to achieve the design breakpoint and that the spring rate is excessive, causing positive speed droop.

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DESIGN PERFORMANCE MECHANICAL GOVERNOR SHEAVE

FIGURE 3

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DESIGN PERFORMANCE MECHANICAL GOVERNOR SHEAVE

FIGURE 4

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DESIGN VS ACTUAL PERFORMANCE MECHANICAL GOVERNOR SHEAVE

FIGURE 5

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MECHANICAL GOVERNOR SHEAVE DESIGN VS ACTUAL PERFORMANCE

FIGURE 6

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NOTE: ACCESSORY DRIVE OUTPUT SPEED
MEASURED AT A/C COMPRESSOR

ACCESSORY DRIVE CALIBRATION

FIGURE 7

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Experiments were performed to demonstrate that an easily implemented reference spring change can be used to modify these characteristics. Curve B in Figure 2 shows performance achieved by increasing the spring preload by 25 percent and decreasing spring rate by 42 percent. The effect is to raise the breakpoint, minimize rounding at the breakpoint and flatten the characteristics in the control range.

2.5 Phase IV Completion Plan

Phase IV of the accessory drive program includes system optimization by matching conventional accessories to vehicle load and accessory drive characteristics. The following plan outlines this approach and enumerates potential optimization areas.

Objective

Optimize the match between accessories, loads, and drive input/output characteristics to demonstrate improved function and reduced system cost that can be achieved with a properly matched accessory drive system.

Approach

- (a) Obtain background on critical match conditions and areas of highest payoff with respect to cost savings.
- (b) Select items of large impact on manufacturing cost and customer satisfaction for experimental verification.
- (c) Procure/modify components for installation in test vehicle.
- (d) Verify by "on road" test.

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Potential Optimization Areas

(a) Engine Cooling System

- o Reduced fan size (noise and cost)
- o Reduced radiator core size
- o Fan shroud removal
- o Cavitation limiting and effect on impeller configuration and cost
- o Water pump bearing/seal life

(b) Air Conditioning System

- o Cool down time reduction
- o High speed capability
- o Reduced compressor size

(c) Alternator System

- o Substitution of smaller frame size machine
- o Higher alternator speed at near idle

(d) Power Steering System

- o Lower pump speed
- o Reduce pump displacement

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- o Remove flow limiter
- o Effect of oil temperature reduction on life

(e) General

- o Increased accessory belt life
- o Torque sensing on direct operated system
- o Relocation of air conditioning compressor to balance peak loads
- o Relate improved performance to resized engine
- o Document noise profile
- o Integration of torsional damper with drive assembly (reduced cost)

Expected Results

Program completion with a partially optimized vehicle and documented test data supporting the conclusion that a properly implemented accessory drive system not only improves economy and performance, but reduces total vehicle manufacturing costs.

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APPENDIX I

IMPROVING AUTOMOTIVE ECONOMY
WITH
ACCESSORY DRIVES
(58 Pages)

PRESENTED AT ERDA/TEC
CONTRACTORS COORDINATION MEETING
ANN ARBOR, MICHIGAN

MAY 4-6, 1976

CONTRACT NO. E[04-3]-1095

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

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(FIGURE 1)

This presentation reflects the status of the ERDA-funded Automotive Accessory Drive Program being performed by the AiResearch Manufacturing Company of Arizona, a Division of The Garrett Corporation.

Some background information is included in the presentation for those not familiar with the program, but the major portion is devoted to the large amount of development data acquired during the last six months.

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IMPROVING AUTOMOTIVE FUEL ECONOMY WITH ACCESSORY DRIVES

FIGURE 1

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The program objective is to minimize the power consumed by automotive engine-driven accessories, thereby improving overall vehicle fuel economy.

During the study phase, a multitude of possible concepts were reviewed and evaluated for reducing accessory power consumption. Strong consideration was given to selection of a system that could be implemented with minimum impact on existing engine/accessory arrangement and component manufacturing techniques and yet yield a maximum benefit in reduced power consumption and improved fuel economy.

The study conclusion was that the objective could be best achieved with a variable ratio belt transmission located between the crankshaft and the engine driven accessories designed to operate the accessories at essentially a constant speed above road speeds of 20 to 30 mph.

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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

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(FIGURE 3)

The study program was completed July, 1975, and was immediately followed by a 13-month development and demonstration program.

This hardware oriented phase involves detail design, fabrication, development, and installation of an accessory drive system in a test vehicle.

At the end of Phase IV, AiResearch expects to have demonstrated the feasibility of applying an accessory drive system to contemporary automobiles, and that use of such a system will not only yield better fuel economy and performance, but improved accessory system functions as well.

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PROGRAM SCHEDULE

FIGURE 3

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(FIGURE 4)

Program status follows:

- o Both drive system concepts were converted to hardware.
- o Both drive system concepts were installed and operated in the test vehicle.
- o The formal EPA dyno cycles were run with and without the drive installed in the car to determine relative emission performance.
- o Approximately 6000 miles were accumulated on the mechanical drive installed in the car.
- o Fuel economy data was obtained.
- o Preliminary data relative to accessory performance improvements at critical conditions was obtained.

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- **VARIABLE RATIO BELT SELECTED AS OPTIMUM NEAR TERM SOLUTION TO AUTOMOTIVE ACCESSORY DRIVE**
- **TWO BELT CONTROL METHODS SELECTED**
 - **CLOSED LOOP HYDROMECHANICAL CONTROL**
 - **DIRECT MECHANICAL GOVERNOR CONTROL**
- **BOTH SYSTEMS HAVE BEEN INSTALLED AND OPERATED IN THE BASELINE CAR**
- **EPA CITY AND HIGHWAY DYNAMOMETER CYCLES HAVE BEEN PERFORMED WITH ACCESSORY DRIVE INSTALLED**
- **DETERMINATION OF ON-ROAD FUEL ECONOMY PARTIALLY COMPLETED**
- **DETERMINATION OF IMPROVED ACCESSORY PERFORMANCE IN PROCESS**

PROGRAM STATUS

FIGURE 4

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(FIGURE 5)

The program effort for the past six months was focused on accessory drive test and development on bench, dyno engine rigs, and installation, as well as in the vehicle. A large amount of in-vehicle test data was acquired concerning, not only fuel economy, but emission characteristics and individual accessory system function.

A fuel economy improvement of 4 to 9 percent was verified, emission characteristics, measured by the EPA test procedures, were not adversely affected, acceleration performance, which is directly indicative of system power savings, was measurably improved.

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FUEL ECONOMY IMPROVEMENTS HAVE BEEN DEMONSTRATED:

9% @ 55 MPH ON TEST TRACK

4% FOR EPA HIGHWAY FUEL ECONOMY TEST (DYNO/CYCLE)

EMISSIONS HAVE BEEN REDUCED BY:

5% HYDROCARBONS

8% OXIDES OF NITROGEN

8% CARBON MONOXIDE

PERFORMANCE HAS BEEN IMPROVED BY:

12% ON 0-60 MPH MAXIMUM ACCELERATION TIME

VEHICLE DATA SUMMARY

FIGURE 5

(FIGURE 6)

This view shows the mechanical accessory drive system (MADS) installation on the dynamometer mounted V6 test engine. The installation in the car is identical.

MADS features worth noting are:

- o Stock accessories with stock mounting brackets and pulley diameters are used.
- o A/C pulley and torsional damper are reversed on the shaft for proper belt track.
- o All outboard accessories are driven from take-off sheaves located at back of accessory drive output sheave.
- o Three belts total - same as stock.
- o Accessory drive center distance is 5.8 inches.
- o Air conditioning idler pulley is eliminated.

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MECHANICAL ACCESSORY DRIVE INSTALLATION

FIGURE 6

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(FIGURE 7)

This view shows a section view of the mechanical or direct operated accessory drive mechanism. It is basically a pair of split pulleys; one crankshaft mounted, one bearing supported on the stock water pump housing. The crank pulley contains the centrifugal flyweight operating mechanism and the upper pulley is simply a spring-loaded take-up mechanism.

Some design features are:

- o Governor force is transmitted through a scissor bar linkage that allows generation of high axial force levels at low speeds.
- o The drive belt is 3/4 inch wide, 0.4 inch thick molded notch construction with Kevlar cords.
- o All belt and fan induced loads are isolated from the stock water pump bearings.
- o The total ratio change of 3 to 1 is split symmetrically about the straight-through drive position. This minimizes the stroke, belt width, and space required for drive packaging.

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MECHANICAL DRIVE CROSS SECTION

FIGURE 7

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This plot shows the installed drive performance for two load conditions versus the design target. The early breakpoint and positive slope are a result of minor mismatches between the governor force and the reference spring and could be corrected by a spring change. Since the deviations were considered to have a minor effect on overall accessory system performance, the hardware will not be modified until more experience is gained with respect to accessory load match in this particular application.

The ratio to load sensitivity is low--less than 100 rpm droop for full air conditioning load.

Dynamic response to acceleration and deceleration is excellent.

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MECHANICAL DRIVE SYSTEM RATIO CONTROL

FIGURE 8

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(FIGURE 9)

In summary, the direct operated variable ratio belt accessory drive was built, installed in the baseline vehicle, and successfully operated. Good load carrying capability, stability, and tracking of transients has been demonstrated. The installation is straight forward with no relocation or modification of stock accessories or mounting brackets required.

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- **DIRECT MECHANICAL CONTROL — INPUT PROGRAMMING**
- **HIGH LOAD TRANSMISSION CAPABILITY — 7.5 Kw (10 HP)**
- **CONSTANT ACCESSORY SPEED OVER 3 TO 1 ENGINE SPEED RANGE**
- **EASY ACCESSIBILITY FOR MAINTENANCE**
- **LOW SENSITIVITY TO VARYING LOAD**
- **NON-LUBRICATED, NON-METALLIC SHEAVE GUIDE DESIGN**
- **CONSTANT TENSION BELT OPERATION**
- **IMPROVED WATER PUMP BEARING LIFE**

MECHANICAL VARIABLE RATIO BELT DRIVE FEATURES

FIGURE 9

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(FIGURE 10)

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.

Again, the minimum number of belts (3) are used.

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HYDROMECHANICAL ACCESSORY DRIVE INSTALLATION

FIGURE 10

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Appendix I

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(FIGURE 11)

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, dual element pump, and a fluid reservoir.

This assembly proves two basic functions:

- o Automatic sheave position control required to maintain constant output speed above the breakpoint.
- o 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 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.

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HYDROMECHANICAL CROSS SECTION

FIGURE 11

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(FIGURE 12)

Performance data from a dynamometer loaded, bench test on the hydromechanical drive system are shown in this figure. The ratio design target is closely met with this system and the load sensitivity is very low--approximately 6 rpm per transmitted horsepower. There have been some dynamic performance deficiencies with this unit which were primarily a result of insufficient pump output at low speeds. Development tests have established that the most feasible corrective measure would be to generate the servo flow from a pumping stage that is hydraulically independent of the power steering source. The hardware was recently modified to a dual pump configuration as shown in Figure 11, and will be tested in the near future.

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HYDROMECHANICAL DRIVE SYSTEM RATIO CONTROL

FIGURE 12

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In summary, the hydromechanical variable ratio belt accessory demonstrated the predicted attributes of tight speed regulation, low load sensitivity, and variable belt tension control with a ramp type torque sensor. Deficiencies with respect to obtaining sufficient servo output from the single element dual output pump, resulted in a redesign of this area of the assembly. Development testing will resume in the near future.

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- 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

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Vehicle emissions and economy were obtained in accordance with the Federal Test Procedure for new, light-duty automobiles. The tests were conducted by Automotive Test Labs (ATL) of Denver and consisted of back-to-back Mustang II baseline vehicle testing in both the stock configuration and with the mechanical accessory drive installed.

The test series objective was primarily to determine whether installing an accessory drive that affected engine loading and air pump speed, would have an adverse effect on the vehicle emission characteristics. Since the test procedure measures fuel economy by the carbon balance technique, some data was also gathered on the relative fuel economy that an accessory drive could be expected to show on a dynameter loaded driving cycle.

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FUEL ECONOMY AND EMISSIONS EPA FEDERAL TEST PROCEDURE

FIGURE 14

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(FIGURE 15)

The emission test results are shown for the three regulated constituents, HC, CO, NO_x. In all cases, the stock vehicle meets the 1975 EPA requirements and the vehicle with the accessory drive installed demonstrates slightly lower emissions than the stock configuration.

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MADS = MECHANICAL ACCESSORY DRIVE SYSTEM

EMISSIONS CHARACTERISTICS FEDERAL TEST PROCEDURE

FIGURE 15

(FIGURE 16)

Fuel economy for the stock and modified vehicles for both urban and highway cycles are summarized in Figure 16. These data were obtained from the carbon balance technique as defined in the Federal Test Procedure.

Installing the drive system improved fuel economy 1.1 percent for the urban cycle and 4.3 percent for the highway cycle. The change that was predicted by the vehicle simulation computer program is shown for comparison (+0.9 and +4.8 percent).

It should be noted that the fuel economy increments obtained by the Federal Procedure will be low relative to on-road operation due to artificially low accessory loading that results from this test technique.

For comparison, the vehicle simulation program predicts that if the vehicle is street driven to the same speed/time profile with realistic accessory loading, the incremental improvement from the accessory drive system would be +3.5 percent for the urban cycle and +7.8 percent for the highway cycle.

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SIMULATED DRIVING CYCLE	VEHICLE MPG		% IMPROVEMENT	PREDICTED % CHANGE
	STOCK	MODIFIED		
FEDERAL	15.19	15.35	+1.1	+0.9
EPA HIGHWAY	21.97	22.92	+4.3	+4.8

CYCLE FUEL ECONOMY EPA TEST PROCEDURES

FIGURE 16

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(FIGURE 17)

The computer program, developed during the study phase, was modified to include Mustang II V6 parameters. This program is very comprehensive in 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 urban and highway cycles.

The data tabulated in Figure 17 compares the computer prediction to actual test and reported economy for the stock vehicle operated to the urban and highway cycles. On absolute value the computer predicts within 4 percent of the test data. As was shown on Figure 16, the computer simulation has been quite successful in predicting the incremental changes that result from modifications to the vehicle accessory configuration.

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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

SOURCE	MPG	
	CITY	HWY
EPA REPORTED BY MANUFACTURER	15.5	22.7
FEDERAL TEST PROCEDURE BY AIRESEARCH	15.2	22.0
COMPUTER SIMULATION (DEVIATION)	15.8 (4%)	21.7 (1%)

COMPUTER ANALYSIS OF FUEL ECONOMY

FIGURE 17

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(FIGURE 18)

The stock and modified vehicles were track tested to determine constant speed fuel economy. Testing was performed on a closed 2-mile track to minimize or eliminate effects of wind, grade, and passby vehicle influence. The relatively short time required to install the accessory drive system in the vehicle allows comparative tests to be accomplished on successive days which reduces any affects associated with progressive deterioration of vehicle components.

Requirements imposed to ensure valid data are:

- o Totalized fuel flow read from an on-board positive displacement flow meter (Fluidyne Model 1214) at 2-mile increments. Data must repeat within 2 percent to obtain an accepted 10-mile data point.
- o Distance obtained from track survey.
- o Speed regulated by automatic speed control.
- o Consistent track lane used to ensure repeatable power steering load.
- o Accessory outputs monitored, recorded and compared for repeatability.
- o Calibration runs at 55 mph are performed periodically during each track session to detect any tendency for data drift.

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TEST VEHICLE ON TEST TRACK

FIGURE 18

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(FIGURE 19)

The vehicle is extensively instrumented to determine accessory operating conditions and on-road fuel economy. The readout panel is mounted in place of the right front seat and is readable from either the drivers' position or the observers station in the right rear seat.

The instrumentation monitors the following parameters:

- o Fuel Flow
- o Engine and Accessory Speed
- o Twelve Temperature (OAT, Air and Liquid Cooling Temperature, Carburetor Inlet, Power Steering, Fluid Temperature, Air Conditioning Duct Temperature, etc.)
- o Power Steering Pressure
- o Water System ΔP
- o Manifold Pressure
- o Air Pump Output Pressure
- o Alternator Output Current and Voltage
- o Air Conditioning Compressor Suction and Output Pressure, and Evaporator Outlet Pressure

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INSTRUMENT PANEL IN TEST VEHICLE

FIGURE 19

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(FIGURE 20)

Fuel economy data for the stock vehicle at constant speeds between 30 and 70 mph is plotted in Figure 20. These data points were obtained by the previously outlined method on the 2-mile track.

For reference, the computer predicted data is shown as the solid line. The correspondance between predicted and actual in respect to level, slope, and location of peak economy is good.

ROAD TEST CONTROLLED PARAMETERS

- 2 MILE CLOSED TRACK
- MINIMUM WINDAGE EFFECT
- CONSISTENT POWER STEERING LOAD
- POSITIVE DISPLACEMENT FUEL METER
- 10 MILE AVERAGE PER DATA POINT
- 2 MILE INCREMENTAL DATA MUST REPEAT WITHIN 2%
- AUTOMATIC SPEED CONTROL WITH LAP TIME CROSS CHECKS

BASELINE FUEL ECONOMY

ACTUAL VS PREDICTED

FIGURE 20

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(FIGURE 21)

The results from over 700 miles of stock and accessory drive modified vehicle track testing are summarized in Figure 21. The improvement in fuel economy for the accessory drive system, obtained experimentally, is shown as the upper portion of each bar. The predicted change is in parenthesis () to the right of each bar.

Economy improvements range from 4.2 percent at 40 mph to 11.4 percent at 60 mph with maximum accessory load. The difference in accessory loading is obtained by turning the air conditioning and lights on.

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STEADY STATE FUEL ECONOMY COMPARISON

ACTUAL VS PREDICTED

FIGURE 21

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(FIGURE 22)

Integration of a variable ratio accessory drive with a particular vehicle/engine/accessory combination requires evaluation of a number of factors:

- (a) Definition of Objective - Is the objective to achieve absolute maximum fuel economy, obtain improved accessory system performance at some critical condition, allow reduced accessory size and cost, or a compromise involving all of the above.
- (b) Drive Train Characteristics - Manual or automatic transmission, "tight" or "loose" convertor, shift points, rear axle ratio; engine size, idle speed and sensitivity of engine idle performance to load.
- (c) Accessory Characteristics - Output and efficiency versus shaft speed, critical loading conditions, degree of demand matching inherent in the accessory/load system, availability, and economics of small components.

The accessory drive design parameters that can be preselected to meet a certain set of requirements are:

- o Total ratio range
- o Initial starting ratio
- o Engine speed at which the drive starts to shift
- o Slope of the speed relation in the ratio changing range
- o Magnitude of the fixed ratio between the drive take-off sheave and the individual accessories.

Figure 22 illustrates these design parameters graphically.

ACCESSORY DRIVE PARAMETERS

FIGURE 22.

(FIGURE 23)

Data presented in Figures 23 and 24 illustrate the speed relation for the six accessories on the Mustang II test vehicle. In general, the parameters for initial development work were selected for a certain degree of overdrive (accessory running faster than stock) at low car speeds with a crossover in the 20 to 30 mph range. This choice allows investigation of improved accessory function at idle and low speed conditions at some expense in fuel economy improvement in this range.

Figure 23 illustrates the drive path for the air conditioning compressor.

The drive is shown in the low engine speed or "maximum forward ratio" position. For the parameters selected for this installation (3:1 range in the drive, split symmetrically) the forward ratio is $\sqrt{3}$ or 1.7 speed-up. Therefore, the water pump, fan, and accessory take-off sheaves, are turning approximately 70 percent faster than the crankshaft.

There is a constant ratio fixed by pulley diameters between the take-off sheave and the A/C compressor of 0.69. The end result is that the compressor is being driven at 1.7 times 0.69 or 1.2 times faster than the crankshaft. This ratio is maintained up to an engine speed where the drive starts to shift (the breakpoint).

The ratio of 1.2 compared to the stock ratio of 0.92 means that an overdrive of 30 percent has been introduced into this component prior to the drive breakpoint, resulting in reduced cool down time at lower vehicle speeds relative to the stock setup.

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ACCESSORY SPEED SELECTION

FIGURE 23

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(FIGURE 24)

The speed relation for the remaining accessories are shown in Figure 24. All plots were based on actual test data derived from in-vehicle operation.

Some degree of overdrive is present in all components with the existing sheave diameter selections.

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ACCESSORY SPEED SELECTION

FIGURE 24

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The power steering system schematic and test data for one aspect of performance is shown in Figure 25. The system consists of a positive displacement pump with internal flow limiter and relief valve, an open center steering valve and actuator and oil cooler. The pump is sized for approximately 2 gpm at 500 rpm and 12 gpm at 3000 rpm. Pump flow in excess of 2 gpm is recirculated internally through the flow limiter. Although a power steering system is demand-matched (pressure proportional to steering load), the large amount of fluid in circulation at higher speeds results in power losses and associated fluid heating to the extent that many installations require an oil cooler.

One effect of limiting pump speed with an accessory drive is shown in the data plot. Oil temperature is reduced by 20° to 30°F at the critical condition although the cooler has been removed. There has been no affect on power steering performance because the pump is always driven in excess of the flow limiter cut-in point.

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POWER STEERING SYSTEM DATA

FIGURE 25

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(FIGURE 26)

Alternator sizing is generally predicated on output capability at low vehicle speeds. Using an accessory drive, it is possible to increase output at this condition by utilizing overdrive without overspeeding the alternator at high engine speeds.

In Figure 26, the stock (70 amp rated) alternator output is compared for the stock system and the accessory drive installation. At engine idle speed of 700 rpm there is a 39 percent increase in output current.

The accessory drive capability could be utilized to obtain design electrical system performance with a smaller, less costly alternator.

FEATURES

- 39% INCREASE IN ALTERNATOR OUTPUT AT IDLE
- 55% REDUCTION IN MAX. OPERATING SPEED
- INCREASED BELT LIFE
- DECREASED INPUT POWER

	ALTERNATOR SPEED @ ENGINE SPEED OF:	
	700 RPM	6000 RPM
STOCK SYSTEM	1450	12,400
WITH ACCESSORY DRIVE	1800	5,600

ALTERNATOR PERFORMANCE

FIGURE 26
74-310860(24)
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(FIGURE 27)

The emissions air pump is driven slightly faster than stock at low speed and then clipped at higher engine speeds by the accessory drive action. The effect on measured emissions was as previously described--a minor reduction in the three regulated pollutants.

Since emissions control is generally critical at low speeds, there is a possibility of utilizing a lower displacement pump, rematching with the engine system, and achieving reduced power, noise and cost.

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EMISSION TEST DATA (IN GRAMS/MILE)

	STOCK SYSTEM	WITH ACCESSORY DRIVE
HC	1.11	1.05
CO	8.85	8.16
NOX	2.74	2.51

AIR PUMP PERFORMANCE

FIGURE 27

(FIGURE 28)

Since the water pump and fan are directly driven from the accessory drive output pulley, operating speed is 70 percent faster than the engine prior to the breakpoint. Significant improvements in low speed cooling can be realized as a result of increased water and air circulation.

Preliminary data obtained for both configurations indicates that the time-to-boil performance at idle has been improved by 10 minutes, although the accessory drive installation uses a smaller diameter, lower pitch, fixed blade fan in place of the stock 5-bladed flex fan.

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**MUSTANG II V-6 COOLING
SYSTEM SCHEMATIC**

	WATER PUMP & FAN SPEED	
	IDLE	55 MPH
STOCK	740	2950
WITH ACCESSORY DRIVE	1200	2200

WATER PUMP AND FAN PERFORMANCE

FIGURE 28

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(FIGURE 29)

- o Variable ratio accessory drives are adaptable to contemporary automobiles with minimum disturbance to existing accessory equipment and engine compartment configurations.
- o Improved fuel economy for both low speed city driving and cruise conditions has been demonstrated.
- o The computer simulation of the vehicle/engine/accessory system has been validated.
- o Functional performance of accessory systems can be improved by proper selection of accessory drive parameters.
- o The potential exists for reducing overall accessory system cost by elimination or simplification of certain components.

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- **FUEL ECONOMY SAVINGS OF 4 TO 10% SUBSTANTIATED**
- **VARIABLE RATIO BELT DRIVE PROVEN & DEMONSTRATED**
- **SYSTEM MECHANIZATION & COSTS COMPATIBLE WITH INDUSTRY REQUIREMENTS**
- **SYSTEM DEMONSTRATES SIGNIFICANT IMPROVEMENT IN ACCESSORY LOAD MATCHING**
- **DRIVEN ACCESSORY COST REDUCTION IS FEASIBLE**
- **ACCESSORY DRIVE & DRIVEN ACCESSORY IMPROVEMENTS CAN BE INTRODUCED INDEPENDENTLY**

CONCLUSIONS

ACCESSORY DRIVE PROGRAM

FIGURE 29