MASTER

SUMMARY REPORT

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STUDY ON REDUCTION OF ACCESSORY HORSEPOWER REQUIREMENTS

FUEL ECONOMY
OPTIMIZATION
AUGMENTATION
PERFORMANCE

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STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

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STUDY ON REDUCTION OF ACCESSORY
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1.0 EXECUTIVE SUMMARY

The Automotive Accessory System Optimization Program Summary Report has been prepared for the Energy Research and Development Administration Division of Transportation Energy Conversion (TEC), Office of Highway Vehicles, Vehicular Systems Branch. This report presents a summary* of work conducted under ERDA Contract EY-76-C-03-1095, from June 1974 to June 1977, consisting of five program phases concerned with near-term accessory system optimization. The follow-on program effort will evaluate and develop mid-1980 accessory systems for advanced Stirling and turbine automotive and truck propulsion engines.

The program objective was to define, evaluate and develop automotive accessory systems to minimize engine power consumption and significantly improve fuel economy. All tasks have been completed and the program objectives have been accomplished. The basic content of each program phase was:

- o Phase I - Conceptual Design - General automotive accessory systems study include recommendations for component improvement and accessory drive trade study.
- o Phase II - Preliminary Design - Performance and sizing analyses with layouts of mechanical and hydromechanical variable ratio belt accessory drives. Selection of baseline test vehicle.

*All data for this summary report has been compiled from program monthly progress reports numbered 74-310860(1) through (33) submitted in accordance with ERDA/TEC Contract EY-76-C-03-1095 from June 1974 to June 1977. Additional, detail information on any particular subject heading in the summary may be obtained from the monthly reports.

- o Phase III - Detail Design - Development of detail manufacturing, assembly and interface drawings including performance and development specifications.
- o Phase IV - Fabrication and Test - Complete component fabrication, assembly, bench and dynamometer testing. Baseline, Mustang II systems analysis and computer simulation for imperical cycle data extrapolations. Accessory drive/vehicle integration for chassis dynamometer and road test verification and demonstration program.
- o Phase V - Hybrid Drive and Advanced A/C Study - Vehicle/driving cycle energy source evaluation for accessory power augmentation, hybrid drive concept study. Automotive air conditioning concept study for maximum required performance and minimum power consumption. Hybrid drive, improved accessories and advanced A/C integrated trade study evaluating maximum fuel economy improvement potential.

Application of the constant speed concept for the automotive accessories has a number of secondary benefits in addition to the prime objective of improving fuel economy, as shown in Figure 1-1. The variable ratio belt mechanical accessory drive system (MADS) installed in the test vehicle improved the EPA composite driving cycle fuel economy 4.8 percent. The improvement achieved for constant velocity Interstate driving was as high as 10 percent. The MADS also provided improved vehicle and accessory performance, reduced emissions, reduced noise, improved accessory life, and allowed rematching accessory systems to load requirements. The constant speed concept permitted efficient accessory array redesign which eliminated miscellaneous pulleys and brackets, engine cooling system fan and

FIGURE 1-1

and radiator resizing, use of a smaller alternator, and power steering fluid cooler deletion. Cost Analysis, based on a two-year ownership period, demonstrated that the consumer would benefit from an OEM installed MADS.

The hybrid accessory drive and advanced air conditioning study effort expanded the constant speed and accessory load matching concepts to include the feasibility of incorporating engine waste heat and vehicle braking regeneration techniques to further improve fuel economy. These accessory power augmentation techniques, combined with variable delivery air conditioning and efficiency improved, load-matched accessories, produced an estimated EPA composite driving cycle fuel economy improvement of 11.3 percent.

As a result of the Accessory System Optimization Program, it was concluded that the effect of accessory loads on the overall fuel economy of present day internal combustion engine automobiles can be decreased by as much as 50 percent by applying technology well within present manufacturing capability. Further improvements in fuel economy could also be achieved through development of hybrid drive concepts if the market would bear the attendant cost increases.

The current trend to lighter vehicles with smaller displacement, higher speed engines, increases the significance of accessory speed and load control. "Bigger" may no longer be "better" but "comfort" will remain a desirable feature to the consumer and accessory power requirements will continue to consume an increasing percentage of the engine output if no improvements are made.

In conducting the study, the impression was gained that, in the past, the evaluation of accessory systems has progressed in a piecemeal fashion with additions or increased operating requirements imposed by new vehicle designs, Federal regulations or customer desires.

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System engineering, component optimization and load matching of accessory subsystems has apparently been minimal or totally subjugated to manufacturing cost considerations. The "add-on option" marketing concept for items such as air conditioning and power steering is being replaced by fully equipped vehicles marketed with "standard features". The impact of this trend on fleet average accessory loading requirements is significant and requires that accessory/engine integration with improved operating efficiency be considered in new vehicle designs. Manufacturing cost can no longer be accepted as the primary consideration relative to the national goal of fuel conservation, secondary benefits of improved component life, and effective cost of ownership for the consumer.

Nationwide automotive fuel consumption could be significantly improved by application of the accessory control technology described herein. The 1979 intermediate vehicle population, equipped with accessory speed control devices, would conserve 69,000 barrels of fuel everyday. If modified to take full advantage of energy reclamation and advanced accessory designs, the fuel savings could be as high as 150,000 barrels per day. If the entire U.S. automotive population, with full accessory arrays, were equipped, fuel conservation in range of 180,000 to 400,000 barrels per day would be possible.

During Phase VI of the accessory optimization program, advanced accessory components and systems for new vehicle designs incorporating turbine and Stirling cycle engines will be evaluated and demonstrated. The technology developed in program Phases I through V will be applied to a new problem statement specifically written to enhance the effective integration of accessory/vehicle/engine requirements and substantially improve component performance and fuel economy.

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2.0 PHASE I - III TECHNICAL SUMMARY

Automotive accessory system and component baseline studies, drive system concept trade studies and variable ratio belt accessory drive designs were completed during the initial program phases and consisted of the following basic tasks:

- Task 1 - Study existing automotive accessories and systems to provide baseline definition.
- Task 2 - Baseline tests vehicle selection.
- Task 3 - Establish design objectives and criteria for accessory drive trade study.
- Task 4 - Perform accessory drive trade study and prepare preliminary designs of candidate system(s).
- Task 5 - Select prime accessory drive concept for baseline vehicle.

2.1 Stock Accessory Performance Evaluation

Stock accessory performance characteristics evaluation with respect to system performance requirements established the general criteria for accessory drive studies. These studies included the following vehicle and engine accessories:

- o Engine Cooling Fan
- o Engine Water Pump
- o Power Steering System Pump
- o Emissions Control Air Pump
- o Alternator
- o Air Conditioning System Compressor

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A generalized statement of accessory input power versus speed requirements is presented in Figure 2-1. The graphics demonstrate that power requirements are a strong function of speed for all accessories. A detailed evaluation of required performance imposed by vehicle, engine, and ambient loading conditions revealed that the critical accessory operating range occurred at vehicles speed between idle and approximately 48 km/hr (30 mph) in most normal driving modes. It was determined that the accessories were sized and driven at speeds marginally sufficient to meet low vehicle speed load demands while the higher vehicle speed accessory capabilities far exceeded actual requirements. It was recognized that some extreme vehicle operating conditions, such as very high speed driving and steep grade climbing, could impose engine cooling demands that require high speed performance capabilities from the cooling fan and water pump. It was believed that these requirements could best be met through proper component design and load matching techniques rather than relying on uncontrolled speed and the related inefficiencies.

Automotive accessories are classically driven at a fixed ratio with respect to the engine crankshaft. The optimum operating speed must be selected to meet low vehicle speed requirements with the restraint that the accessory will not be destroyed at maximum engine speed. The technique of accessory speed control by incorporating a variable ratio drive device between the engine and accessories is demonstrated in Figure 2-2. The constant speed accessory concept provides the operating envelope flexibility for efficiently matching input and output load requirements; therefore, the excess power consumption associated with fixed drive ratio accessories can be conserved resulting in less engine work and improved fuel economy.

2.2 Baseline Vehicle Selection

A Ford Mustang II was selected as the baseline vehicle for an accessory drive system development. The following criteria and Figure 2-3 were applied in vehicle selection:

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IMPROVED AUTOMOTIVE ACCESSORY DRIVE PROGRAM

FIGURE 2-1

TYPICAL SPEED LIMITING EFFECT

FIGURE 2-2

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CRITERIA

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

**MUSTANG II
SELECTED**

TEST VEHICLE DESCRIPTION

1975 MUSTANG II HATCHBACK
2800 CC V6 ENGINE
3 SPEED AUTOMATIC TRANSMISSION
A/C AND P/S

N/V = 50 RPM/MPH
0-60 MPH — 15 TO 17 SEC
CURB WEIGHT - 3050#

ACCESSORY DRIVE PROGRAM TEST VEHICLE

FIGURE 2-3

- o American manufacture
- o Weight and performance within ERDA compact auto specification scope
- o Accessory options availability
- o Engine options availability
- o Continued production probability
- o Sales volume
- o Engine compartment configuration
- o Manufacturer's engineering data availability

A Mustang II with the following description was procured:

1975 Mustang II, 3-door hatchback
171 V6 Engine with 2-barrel carburetor
3-speed automatic transmission
Factory power steering and air conditioning

Figure 2-4 shows the stock accessory installation of the baseline test vehicle.

2.3 Design Objectives and Trade Study Criteria

The study objective was to minimize accessory power consumption and maximize overall vehicle fuel economy by utilizing the constant speed drive concept.

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STOCK ACCESSORY INSTALLATION

MUSTANG II

FIGURE 2-4

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Although the primary goal was to improve fuel economy, factors such as initial cost, complexity, maintainability, noise, installation, weight, and emissions were included as important considerations to meet consumer and manufacturer acceptability. Figure 2-5 establishes the design philosophy and trade study factors with appropriate considerations.

The magnitude of accessory effects on fuel economy and engine load were determined from baseline vehicle tests and performance analysis. Accessories were found to be responsible for 12 to 23 percent fuel economy degradation and 25 to 60 percent of the total engine load, depending on vehicle speed as shown in Figure 2-6.

Accessory load matching studies dictated an overdrive requirement at engine speeds below 1200 rpm and maintain relatively constant accessory speed from 1200 to 3600 engine rpm. The power conservation obtained with the new accessory operating conditions is shown to be as high as 12.7 kW (17 hp) in Figure 2-7. Accessory drive system efficiency was also determined to be a critical factor in fuel economy which established a major trade study requirement to select devices with maximum efficiency characteristics.

2.4 Accessory Drive System Trade Study

Accessory drive systems for automotive applications were categorized as follows:

- o Conventional Belt or Gear
- o Hydrostatic
- o Hydromechanical
- o Traction
- o Variable Ratio Belt
- o Electrical
- o Auxiliary Power Unit (APU)

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 (50%)
- INITIAL COST (15%)
- COMPLEXITY (10%)
- MAINTENANCE COST (10%)
- NOISE (5%)
- INSTALLATION FLEXIBILITY (5%)
- WEIGHT (5%)
- EMISSIONS

IMPROVING AUTOMOTIVE FUEL ECONOMY

FIGURE 2-5

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STOCK ACCESSORY SYSTEM — 1975 MUSTANG II

- AIR CONDITIONING
- POWER STEERING
- ALTERNATOR
- ENGINE COOLING FAN
- WATER PUMP
- EMISSIONS AIR PUMP

VEHICLE ACCESSORIES - MPG AND LOAD

FIGURE 2-6

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ACCESSORY DRIVE PERFORMANCE TRADE STUDIES

FIGURE 2-7

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- o Friction Clutch
- o Two-Speed Gear Transmission

Preliminary evaluation of the above drive types was performed to determine applicability with respect to power consumption and secondary considerations such as, complexity, cost, packaging, noise, and weight. Preliminary evaluation produced sufficient data to eliminate three types of drives and expand the general category into specific drive systems.

APU and conventional belt or gear systems were eliminated for use as prime drives for the following reasons:

- o APU or totally independent accessory drive and power systems demonstrated poor applicability in all selection criteria categories.
- o Exhaust gas turbine drives were eliminated because the baseline vehicle engine did not have sufficient recoverable exhaust energy to supply accessory power requirements.
- o Conventional belt and gear drives with fixed engine-to-accessory drive ratios require excessive power at engine speeds above 1200 rpm. Secondary consideration factors for this type of drive are attractive and belt or gears would probably be useful for accessory power distribution from a main-drive unit.

The specific drive systems described below were evaluated in detail and this trade study data shown in Table 2-I was developed for relative rating purposes. As a result of this trade study, it was concluded that variable ratio belt and traction transmissions were the most feasible and efficient concepts for development. The variable

TABLE 2-I

ACCESSORY DRIVE SYSTEM TRADE STUDY

ACCESSORY DRIVE SYSTEM	OVERALL DRIVE EFFICIENCY RANGE %	COMPLEXITY	INITIAL COST	INSTALLATION FLEXIBILITY	NOISE	SYSTEM VOLUME	WEIGHT	MAINTENANCE COST	RATING TOTALS	TOTAL RATING PLACEMENT
SPLIT PATH HYDROMECHANICAL	45-80	4	3	2	4	3	2	3	21	3
CONSTANT SPEED HYDROSTATIC	40-75	5	4	5	4	5	4	4	31	5
VARIABLE SPEED HYDROSTATIC	35-75	6	5	4	4	4	5	5	33	6
VARIABLE RATIO TRACTION	85-95	3	2	2	3	2	2	2	16	2
VARIABLE RATIO BELT	90-95	2	1	3	2	2	1	1	12	1
VARIABLE ELECTRIC	35-75	1	6	3	1	4	6	2	23	4
SLIP CLUTCH	15-95	2	1	1	2	1	1	2	10	6
2 SPEED	70-90	5	4	4	3	5	4	2	27	4

- NOTES: (1) Overall efficiency as applied to required engine power.
(2) Ratings are 1 through 6 with 1 indicating most favorable system.

ratio belt concept was selected, based on maximum efficiency and state-of-the-art development potential, resulting from the final trade study process shown in Figure 2-8.

- (a) Split-Path Hydromechanical - A constant-speed drive using a variable-displacement, ball-piston radial pump motor and a similar ball-piston, fixed-displacement pump motor, both sharing a common cylinder block rotation at engine speed.
- (b) Constant-Speed Hydrostatic - This drive uses an engine-driven, variable-displacement pump to drive a fixed-displacement, piston-type, hydraulic motor at constant flow. The pump and motor form an integral package.
- (c) Variable-Speed Hydrostatic - A variable-displacement pump driven at engine speed and controlled to provide constant pressure flow to a variable-displacement hydraulic motor. The pump and motor are packaged individually. This system allows accessories to operate at optimum speed-load conditions.
- (e) Variable-Ratio Belt - This is a V-belt drive where one sheave diameter is controlled as a function of speed and the other sheave diameter is controlled as a function of load. The device may also be used as a constant- or variable-speed drive.
- (f) Variable Electric - This concept is comparable to the variable-speed, hydrostatic drive except direct-current electrical energy is used in place of hydraulic power. An alternator power source would provide electrical energy to drive individual accessory motors.

FIGURE 2-8

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- (g) Friction-Clutch - A speed sensitive clutch capable of drive ratio changes to control accessories at a constant speed over a preselected operating range. Operation requires dissipation of power into heat energy.
- (h) Two-Speed-Gear Transmission - This drive uses two fixed-ratio accessory-gear drives coupled with an engine-speed-sensitive shift mechanism.

2.5 Variable Ratio Belt Drive Design

Variable ratio belt drive transmission speed and load characteristics were studied in depth and two basic concepts were selected for design and development.

The hydròmechanical design consisted of a torque-sensitive sheave on the engine crankshaft to provide belt-tension control and a hydraulic speed-control sheave integrated with the vehicle power steering pump.

The mechanical design incorporated a flyweight governor speed-control sheave mounted on the engine crankshaft, and a spring-loaded belt-tension sheave on the water pump shaft.

2.5.1 Speed Ratio Selection

The prototype drive system operating speed range is shown in Figure 2-9. The drive "break point" selection of 1200 engine rpm is equivalent to a vehicle speed of approximately 39 km/H (in high gear), while the accessories are driven at an equivalent vehicle speed of approximately 48 km/H. The accessories were slightly over-driven up to the drive "break point" to assure output capacity at road speeds above 48 km/H. The drive system total ratio of 3:1 was split evenly between under- and over-drive and becomes saturated at 3600 engine rpm, which is equivalent to 114 km/H.

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MUSTANG II-ACCESSORY DRIVE SYSTEM SPEED PROFILE

FIGURE 2-9

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2.5.2 Accessory Loads

Accessory load input to the drive system, at design ratios, is presented in Figure 2-10. Load values were based on Ford and AiResearch test data.

2.5.3 Hydromechanical Belt Drive Design

The Engineering layout for a hydromechanically controlled, variable ratio, belt drive is shown on Figures 2-11 through 2-14.

2.5.3.1 Torque Sensing Driver Sheave

The driver sheave, shown in Figure 2-11, senses differential force in the tight and slack side belt tensions, due to transmitted torque, and transfers a required portion of torque into belt squeeze via a helical cam ramp. The design incorporates self centering, twin-tapered, cam rollers to assure equal load distribution and smooth operation. All movable and load bearing surfaces are sealed and grease packed, to minimize wear and friction. The cam return spring is mounted on a roller thrust bearing to eliminate wrap-up that would be detrimental to torque sensing control. The sheave senses applied torque over the accessory load range and properly tensions the drive belt to provide maximum efficiency and long life.

2.5.3.2 Hydraulic Speed Control Driven Sheave

The hydraulic speed control driven sheave design incorporates a simplified power steering pump. Belt position, in the driven sheave, is controlled with a servo piston pressurized by the cross-axis governor output. This governor is programmed to maintain the specified accessory speed profile. The governor senses speed deviation from a present position (2080 rpm) due to governor valve spool centrifugal loading, and meters pressure to the servo piston to position the belt and maintain the speed profile. System characteristics are presented in the speed/load diagram shown in Figure 2-15.

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MUSTANG II-ACCESSORY DRIVE SYSTEM LOAD PROFILE (REVISED)

FIGURE 2-10

TORQUE SENSING DRIVER SHEAVE

FIGURE 2-11

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FRONT VIEW

HYDRAULIC SPEED CONTROL DRIVEN SHEAVE

FIGURE 2-12

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HYDRAULIC SPEED CONTROL DRIVEN SHEAVE

FIGURE 2-13

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HYDRAULIC SPEED CONTROL DRIVEN SHEAVE

FIGURE 2-14

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2.5.3.3 Hydromechanical Design Parameters

Envelope restrictions, drive-belt capacities, and design-speed ratios dictate optimum drive system sizing. The following list of design parameters describes the system and the engine installation is shown in Figure 2-16:

- o Driver-to-Driven Sheave Center Distance -33 cm (13.0 inches) (adjustable)
- o Sheave-to-Sheave Clearance -12.7 cm (5 inches) minimum
- o Belt Tensions - Ultimate Load -103 kgs (227 lbs) maximum
Maximum Continuous Load -69 kgs (152 lbs) maximum
Minimum Continuous Load -34 kgs (74 lbs) maximum
- o Theoretical System Efficiency -91-98 percent
- o Belt Pitch Diameters -10.26 cm (4.04 in.) minimum
-17.78 cm (7.00 in.) maximum
-18.14 cm (7.14 in.) over-travel
- o Speed Ratio (Engine to Accessories) -3:1
- o Belt Dimensions -Top width 25 mm (0.984 in.)
-Side angle 26 degrees
-Thickness 11.9 mm (0.47 in.) maximum
-Length 110.5 cm (43.5 in.) minimum
- o Sheave Stroke -17.35 mm (0.683 in.) nominal
-18.16 mm (0.715 in.) with over-travel

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HYDROMECHANICAL ACCESSORY
DRIVE INSTALLATION

FIGURE 2-16

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- o Torque Control
 - Cam diameter 4.445 cm (1.75 in.)
 - Torque angle 38 - 40 degrees
- o System Pressure
 - Power Steering 3.45 - 82.76 bar (50 - 1200 psig)
 - Sheave control 3.10 - 27.59 bar (45 - 400 psig)
- o System Flow at 2080 rpm
 - Power steering 8.33 l/min (2.2 gpm)
 - Sheave control 1.51 l/min (0.4 gpm)

2.5.4 Mechanical Belt Drive Design

The engineering layout for a mechanically-controlled, input-programming, drive system is presented in Figures 2-17 and 2-18.

2.5.4.1 Speed Control Driver Sheave

The driver sheave contains a four-segment flyweight to open the sheave as speed increases, and a return spring to counterbalance governor and belt-produced forces. Sheave force-levels that are generated throughout the total engine speed range are shown in Figure 2-19.

2.5.4.2 Tension Control Driven Sheave

The driven sheave is spring-loaded closed at a force level sufficient to prevent belt slippage under worst case load conditions. The water pump housing and shaft have been modified to accept additional bearings required for drive installation.

2.5.4.3 Mechanical Design Parameters

- o Driver-to-Driven Sheave Center Distance -15cm (5.0 in.) fixed

INPUT PROGRAMMING - MECHANICAL SPEED CONTROL

FIGURE 2-17

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INPUT PROGRAMMING - MECHANICAL OUTPUT SHEAVE

FIGURE 2-18

MUSTANG II MECHANICAL DRIVE SYSTEM DRIVER SHEAVE LOADS

FIGURE 2-19

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- o Belt-Drive Ratio -3:1
- o Belt Sheave Diameters, Both Sheaves
 - Maximum = 14.6 cm (5.75 in.)
 - Minimum = 8.43 cm (3.32 in.)
- o Belt Dimension
 - Width - 19 cm (0.75 in.)
 - Thickness - 10.2 cm (0.400 in.)
 - Pitch Diameter Length - 67 cm (26.4 in.)
 - Side Angle = 26 degrees
- o Theoretical System Efficiency -85 - 95 percent

The mechanical drive engine installation is shown in Figure 2-20.

2.6 Prime Accessory Drive Concept Selection

Feasibility hardware for both hydromechanical (HADS) and mechanical (MADS) accessory drive system concepts were manufactured. An extensive design verification program was conducted at the "bench" and dyno-engine test levels to determine operating characteristics and design modification requirements to advance either or both concepts to vehicle integration and demonstration testing.

The HADS testing effort exposed control pump anomalies associated with the single-stage, dual-output advanced design. Pump control flow output at minimum drive speed and maximum pressure was not sufficient to respond to rapid engine accelerations causing system dynamic instability. The torque sensing crank sheave performed as designed for all steady-state accessory load conditions, but exhibited response instabilities during rapid load transients, such as cycling the air conditioning compressor or power steering pump. Steady-state performance was sufficient to generate the speed/load profile data presented in Figure 2-21. The data shows excellent speed control capability with extremely low load droop characteristics.

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DIRECT OPERATED ACCESSORY DRIVE INSTALLATION

FIGURE 2-20

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HYDROMECHANICAL DRIVE SYSTEM RATIO CONTROL

FIGURE 2-21

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The MADS test series proved excellent speed control characteristics with better than expected load sensitivity. The MADS exhibited fast, stable response to extreme speed and load transients within the design operating range. Minor problems with metal-on-metal load surfaces, inappropriate material selections and surface treatments were easily corrected.

The effort required to develop the HADS concept and eliminate the control stability anomalies was considered out-of-scope and the MADS concept was selected for vehicle integration.

3.0 PHASE IV TECHNICAL SUMMARY

Development and vehicle-accessory integration of an Accessory Drive System for the baseline, 1975 Mustang II test vehicle was completed in program Phase IV and consisted of three basic tasks:

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

Task 2 - Computer model vehicle equipped with MADS to perform cycle analysis.

Task 3 - Integrate MADS into baseline vehicle, evaluate system performance, optimize accessory matching.

3.1 Design

The final developed MADS configuration design and fabrication technique is demonstrated in the exploded hardware pictures presented in Figures 3-1 through 3-3. Figure 3-1 shows all drives system components while Figures 3-2 and 3-3 show the water pump mounting, tension sheave (tension spring and fan bracket removed), and the crankshaft mounted, governor speed programming sheave respectively. Extensive MADS dynamometer engine testing was performed to determine and modify load and speed control capabilities and correct minor hardware load bearing/surface fretting problems. MADS installation on the dynamometer test engine is shown in Figures 3-4 and 3-5.

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DRIVES SYSTEM COMPONENTS
FIGURE 3-1

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WATER PUMP MOUNTING AND TENSION SHEAVE
FIGURE 3-2

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GOVERNOR SPEED PROGRAMMING SHEAVE

FIGURE 3-3

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MADS INSTALLED ON DYNAMOMETER

FIGURE 3-4

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MADS INSTALLED ON DYNAMOMETER

FIGURE 3-5

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3.2 Computer Analysis

A baseline test vehicle computer model was constructed to produce fuel economy for various driving cycles and provide a tool for off-design parametric studies.

The computer Model ⁽¹⁾ simulates a 1975 Ford Mustang II automobile with a 2.8 liter V-6 engine and an automatic transmission. The stock automobile characteristics are defined in Table 3-I and Figures 3-6 through 3-9.

A summary of various conditions which were evaluated is presented in Table 3-II. This included two driving cycles, the Federal Driving Cycle (FDC) and the Highway Fuel Economy Test (HWFET), and constant velocity conditions. Each case was evaluated with air conditioning on, off, and with the stock accessory drive and the MADS. A graphical description of simulator driving cycle capability is shown in Table 3-III.

The last four conditions in Table 3-II simulated the Federal Test Procedure, performed on a dynamometer, rather than a road test.

Accessory loads for the stock vehicle are defined in Figures 3-10 and 3-11. The alternator load was generally assumed to be 15 amps with A/C off and 30 amps with A/C on. However, as indicated in Table 3-II, the alternator load was taken as 55 amps for the constant-velocity A/C-on cases. This condition represented the actual electrical load imposed for the oval-track driving tests conducted at the International Harvester test track. The curve for the 55 amp alternator is defined in Figure 3-12.

(1) All input/output values are presented in the English system of units to satisfy simulator program requirements.

TABLE 3-I

BASELINE VEHICLE COMPUTER MODEL DEFINITION

1975 Ford Mustang II 2-Door Hardtop

Engine (Figure 3-6)

2.8 Liter 171-V6-2V

Inertia - 0.023 ft-lb-s²

Accessories (Figures 3-10 through 3-12)

	<u>Air Cond</u>	<u>Fan</u>	<u>Alt.</u>	<u>PS</u>	<u>WP</u>	<u>AP</u>
Stock Drive Ratio (acc-to-eng)	0.92	1.05	2.06	1.08	1.05	0.95
Inertia (ft-lb-s ²)	0.0341	0.0059	0.0015	0.0014	0.0013	0.0028

Transmission

C-4 Automatic Transmission

	<u>1st Gear</u>	<u>2nd Gear</u>	<u>3rd Gear</u>
Gear Ratios	2.46	1.46	1.00
Inertia (ft-lb-s ²)	0.0225	0.0251	0.0460

Efficiency calculated as a function of HP transmitted using a model of the C-4 transmission supplied by Orshanski.

Torque Converter (Figure 3-7)

Inertias (ft-lb-s²): Impeller = 0.116, Turbine = 0.032

Pump loss calculated using a model, provided by Orshanski, which assumes pump torque proportional to engine torque.

Rear Axle

Gear Ratio = 3.55

Efficiency calculated as a function of speed and torque.

Road Load (including both aerodrag and rolling resistance (Figure 3-9))

Wheel Inertia = 3.01 ft-lb-s² total

Miscellaneous

Vehicle Weight = 3380 lb (including fuel and two passengers)

Fuel Density = 6.173 lb/gal

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ENGINE MAP

FIGURE 3-6

TORQUE CONVERTER

FIGURE 3-7

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UPSHIFT ONLY

BASED ON MEASURED MANIFOLD VACUUM AT SHIFT POINT,
ENGINE DATA, AND CONVERTER CHARACTERISTICS

$$\text{VELOCITY (MPH)} = N_{\text{DRIVESHAFT}}(\text{RPM}) / 50.9$$

TRANSMISSION SHIFT CRITERIA

FIGURE 3-8

VEHICLE ROAD LOAD

FIGURE 3-9

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TABLE 3-II
ANALYSIS

Driving Cycle	Acc. Drive	A/C	P/S	Fan	Alt.	MPG	Δ MPG
FDC on Road	STK VRD	Max	100 psi	STK Flex 4-Blade	30 amps	14.20 14.70	+ 3.5
FDC on Road	STK VRD	Off	100 psi	STK Flex 4-Blade	15 amps	15.73 15.88	+ 1.0
HWFET on Road	STK VRD	SS	100 psi	STK Flex 4-Blade	30 amps	19.57 21.10	+ 7.8
HWFET on Road	STK VRD	Off	100 psi	STK Flex 4-Blade	15 amps	21.50 22.62	+ 5.2
Const. Vel.	STK VRD	SS	100 psi	STK Flex 4-Blade	55 amps	See Figure 14	See Figure 15
Const. Vel.	STK VRD	Off	100 psi	STK Flex 4-Blade	15 amps		
FDC on Chassis Dyno	STK VRD	Off	50 psi	STK Flex 4-Blade	15 amps	15.83 15.97	+ 0.9
HWFET on Chassis Dyno	STK VRD	Off	50 psi	STK Flex 4-Blade	15 amps	21.72 22.77	+ 4.8

SS = Steady State Heat Load
Max = Maximum Heat Load During Initial Cool Down
Wt = 3380 lb.

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TABLE 3-III

DRIVING MODES

DRIVING CYCLE	GRAPHIC DISPLAY	AVERAGE SPEED MPH	CYCLE LENGTH MILE	STOPS PER MILE	PERCENT TIME @ DRIVING CONDITIONS				MAXIMUM ACCEL. RATE (Ft/Sec ²)
					IDLE	ACCEL.	DECEL.	CRUISING	
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

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ACCESSORY POWER CONSUMPTION

FIGURE 3-10

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REFRIGERATION COMPRESSOR PERFORMANCE MATRIX

FIGURE 3-11

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ACCESSORY POWER CONSUMPTION

FIGURE 3-12

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The same accessory loads were also used in evaluating the MADS with exception of the fan. A smaller fan was used with the MADS to provide the same degree of cooling at low engine speeds as provided for the stock case. The fan characteristics are shown in Figure 3-12.

The MADS crankshaft and water pump sheaves are movable, positioned automatically as a function of engine speed, to vary the ratio between water pump and crankshaft speed. This characteristic, determined experimentally for the MADS, is shown in Figure 3-13. Also shown are the various accessory-to-water-pump drive ratios for the MADS vehicle.

The federal test procedure simulation (chassis dynamometer) used the accessory loads described above except the power steering. Since there are no steering loads in dynamometer tests, the power steering null pressure of 50 psi was used. It was assumed to be 100 psi for all other cases. The 50 psi load curve is defined in Figure 3-12.

Constant-velocity analysis results are presented in Figures 3-14 and 3-15. Fuel economy as a function of velocity is shown in Figure 3-14. Fuel economy improvement due to the MADS is shown in Figure 3-15. At 55 mph with A/C on and maximum electrical load, fuel savings, due to the MADS, is 9.2 percent.

Results of the two driving cycles are shown in Table 3-II. The data show fuel savings of 7.8 percent for the highway fuel economy test (HWFET) with A/C on.

The federal test procedure stipulates that fuel economy be evaluated on a dynamometer with the air conditioning off. Since the A/C is a large power consumer, the federal test procedure will reflect lower MADS benefits. There is also a small lower-than-realistic power steering load effect, tending to understate benefits associated with the MADS. For example, the federal test procedure, according to the analysis, would show an increase in fuel economy of 4.8 percent for the HWFET. A more realistic value for the HWEFT would be between 5.2 percent (A/C off) and 7.8 percent (A/C on).

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MADS SPEED CONTROL ACTUAL PERFORMANCE

FIGURE 3-13

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CONSTANT VELOCITY FUEL ECONOMY

FIGURE 3-14

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MADS EFFECT ON CONSTANT SPEED FUEL ECONOMY

FIGURE 3-15

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3.3 System Integration and Tests

The MADS/Mustang II installation is shown in Figures 3-16 and 3-17. The variable ratio belt is in the forward position, driving from the crankshaft to the water pump/fan shaft. Power take-off for the remaining accessories is from the water pump/fan shaft location. The mid position V-belt drives the power steering pump and air conditioning compressor. The compressor drive belt position was moved aft by simply reversing the sheave and harmonic damper positions on the A/C clutch assembly. The aft accessory V-belt, driving the alternator and emissions control air pump remains in the stock position. The stock five-bladed, flex-fan was replaced with a four-bladed, fixed pitch fan to conserve power. Table 3-IV presents physical data for the MADS installation.

Initial MADS testing in the test vehicle proved the design to be totally functional and capable of performing all design goals. Figure 3-18 shows drive system performance for two accessory load extremes over the full speed envelope.

Vehicle emissions and fuel economy were tested in accordance with the Federal Test Procedure for new, light-duty automobiles. The test was conducted by Automotive Testing Laboratories (ATL) of Denver, Colorado. The test consisted of determining vehicle emissions and fuel economy, with and without the MADS, for the Federal Urban Driving Cycle (LA-4) and the EPA Highway Fuel Economy driving Cycle. The purpose of performing these tests was as follows:

- o To verify that test vehicle emissions controls were functioning properly.
- o To determine MADS effects on emissions and fuel economy.

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MECHANICAL ACCESSORY DRIVE SYSTEM INSTALLATION

FIGURE 3-16

31-2612

Page 3-23

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MECHANICAL ACCESSORY DRIVE SYSTEM INSTALLATION

FIGURE 3-17

31-2612

Page 3-24

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TABLE 3-IV

MECHANICAL ACCESSORY DRIVE SYSTEM
INSTALLATION DATA

Accessory	Sheave Pitch Diameter (mm)	Belt Width (mm)	Drive Ratio to Crank Shaft	
			$N_C < 1200$ rpm Maximum Ratio	$N_C > 3600$ rpm Minimum Ratio
Crankshaft	140 to 81	19.0	1.000	1.000
Water Pump and Fan	81 to 140	19.0	1.730	0.577
Air Conditioning	154	11.1	1.200	0.400
Power Steering	133	11.1	1.395	0.465
Alternator	68	9.5	2.804	0.935
Air Pump	150	9.5	1.264	0.422
A/C - P/S Power Takeoff	107	11.1	1.730	0.577
ALT - A/P Power Takeoff	110	9.5	1.730	0.577

Belt Lengths	
Position	Pitch Lengths
Forward (main drive)	66.6 cm
Mid (W/P to P/S to A/C)	122 cm
Aft (W/P to ALT to A/P)	112 cm

AIR CONDITIONING COMPRESSOR SPEED STOCK VS DESIGN VS ACTUAL PERFORMANCE

FIGURE 3-18

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- o To compare emissions fuel economy (carbon balance method) with flowmeter fuel measurements.
- o To compare computer simulation with EPA test data.

A test data summation is shown in Table 3-V and a comparison of results demonstrating overall MADS effects is presented in Figure 3-19. The carbon balance-vs-flow measurement fuel economy comparison is presented in Table 3-VI. The comparisons of computer simulated vs. EPA test fuel economy are presented in Table 3-VII.

The stock and MADS modified vehicle were road tested to determine constant speed fuel economy for both configurations.

The vehicle was fully instrumented for purposes of determining accessory operating parameters and effects on fuel economy. The instrumentation monitors: vehicle, engine and accessory speeds; alternator load; cooling system temperatures and pressures; air conditioning system parameters; emission air pump pressure; power steering pump pressure and temperature; fuel consumption; engine status parameters. The instrumentation panel installed in the vehicle is shown in Figure 3-20.

Constant speed fuel economy testing was performed on a closed oval track to minimize the effects of extraneous conditions that could interfere with the test objective. The controlled comparison of road test and simulated fuel economy is presented in Figure 3-21. Actual constant speed fuel economy data for the stock and MADS vehicle is presented in Figures 3-22 and 3-23 respectively. These figures were all constructed with experimental data points and the mean was subject to change as continued testing resulted in additional data.

TABLE 3-V

EMISSION AND FUEL ECONOMY TEST DATA SUMMARY

Stock Vehicle						
Condition	Emissions, Grams per mile				Fuel Economy, mpg	
	HC	CO	NOX	CO2	Carbon Balance	Fuel Flow
Cold Transient	2.25	27.81	3.97	565.3	14.40	15.15
Cold Stabilized	0.73	3.55	1.99	598.7	14.63	14.87
Hot Transient	0.95	4.68	3.26	507.2	17.15	16.92
1975 Urban Composite	1.11	8.85	2.74	566.9	15.19	15.44
HWFET	0.45	3.26	3.41	397.4	21.97	22.23

MADS Vehicle						
Condition	Emissions, grams per mile				Fuel Economy, mpg	
	HC	CO	NOX	CO2	Carbon Balance	Fuel Flow
Cold Transient	1.88	25.97	3.49	557.2	14.69	15.29
Cold Stabilized	0.77	3.15	2.01	596.4	14.70	14.89
Hot Transient	0.94	4.30	2.73	500.4	17.40	17.21
1975 Urban Composite	1.05	8.16	2.51	562.1	15.35	15.55
HWFET	0.47	2.16	2.78	382.3	22.92	22.94

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VEHICLE PERFORMANCE IMPROVEMENTS

FIGURE 3-19

TABLE 3-VI
FUEL ECONOMY MEASUREMENT ACCURACY

VEHICLE CONFIGURATION	TEST CONDITION	% DEVIATION CARBON BALANCE-vs-FUEL FLOW
STOCK	Cold Transient	+5.2
	Cold Stabilized	+1.6
	Hot Transient	-1.3
	1975 Urban Composite	+1.6
	EPA HWFET	+1.2
MADS	Cold Transient	+4.1
	Cold Stabilized	+1.3
	Hot Transient	-1.1
	1975 Urban Composite	+1.3
	EPA HWFET	0

TABLE 3-VII

COMPUTER SIMULATED VS EPA TEST FUEL ECONOMY

Vehicle Configuration	Test Cycle	Fuel Economy (mps)		% Deviation
		EPA	Simulated	
Stock	Urban	15.19	15.83	+4.2
	HWFET	21.97	21.72	-1.1
MADS	Urban	15.35	15.97	+4.0
	HWFET	22.92	22.77	-0.7

FUEL ECONOMY IMPROVEMENT COMPARISON

Test Cycle	Fuel Economy % Improvement	
	EPA Method	Simulation Method
Urban	1.1	0.9
HWFET	4.3	4.8

INSTRUMENT PANEL IN TEST VEHICLE

INSTRUMENT PANEL IN TEST VEHICLE

FIGURE 3-20

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OVER 10,000 VEHICLE TEST MILES

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 3-21

CONSTANT SPEED FUEL ECONOMY
STOCK VEHICLE

FIGURE 3-22

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CONSTANT SPEED FUEL ECONOMY
MADS VEHICLE

FIGURE 3-23

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The MADS effect on fuel economy is shown in Figure 3-24 as percent-improvement for minimum and maximum accessory load conditions. Minimum accessory load is obtained with the A/C off and electrical loads at minimum. Maximum accessory load occurs with the A/C and headlights on. The approximate difference in accessory load for these two extremes is 2.6 to 3.0 kW (3.5 to 4.0 hp). Figure 3-24 also contains the simulated fuel economy improvement for constant speed operation to demonstrate validity of model predictions.

3.4 Accessory Optimization

Emphasis was on evaluating the cost saving potential of a properly matched accessory drive system during the final portion of Phase IV. The accessory drive developed during this program basically inserts a variable-ratio belt transmission between the engine and accessories. By selecting design parameters, it is possible to achieve a variety of input/output characteristics in this transmission. This flexibility, in conjunction with modified ratios between the drive output and individual accessories, creates a degree of design freedom that can be utilized to reduce driven accessory cost, weight and size.

This concept was pursued for three systems during the Phase IV program final stages; power steering, cooling, and alternator. Sufficient experimental data was obtained on the Mustang II test vehicle to confirm that potential exists in all three systems for a meaningful reduction in cost, weight, and complexity, while retaining the previously demonstrated advantage of increased fuel economy.

3.4.1 Power Steering System Matching

These systems utilize positive displacement pumps of approximately one cubic inch per revolution capacity, with an internal flow limiter that limits output to the steering valve to two gallons per minute (462 in.³/min). Excess pump output is recirculated internally. With

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MADS EFFECT ON CONSTANT SPEED FUEL ECONOMY
ACTUAL VS SIMULATED

FIGURE 3-24

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the conventional accessory setup of approximately 1 to 1 pump speed to engine speed ratio, the pump generates approximately 13 gpm at 60 mph, and recirculates 11 gpm. The associated energy loss generally requires a heat exchanger addition in the power steering fluid circuit.

As demonstrated earlier in the program, pump speed limiting that results from the accessory drive installation reduces energy loss to a level that does not require a cooler (reference Figure 3-25).

3.4.2 Alternator System Matching

Using a speed-limiting accessory drive allows alternator operation at higher shaft speeds near idle without penalty of excessive shaft speed at high engine speeds. The possibility then exists that adequate electrical system performance can be obtained with a smaller machine. For the Mustang II, incentive is particularly strong since the stock 70-amp alternator is a relatively expensive "high output" accessory designed to develop a higher percentage of rated output at low shaft speeds.

A comparison of actual maximum alternator output for the stock accessory system and the MADS system, using a conventional 60-amp alternator with a 2-1/4 inch pitch diameter drive pulley, is shown in Figure 3-26. Maximum output at each engine speed was obtained by loading the vehicle electrical system with an external load bank that was progressively increased to obtain alternator saturation.

Substitution of a smaller alternator proved adequate for all operating conditions (i.e., urban driving with maximum air conditioning, day and night, repeated starting associated with test sequences, and a 2000 mile round trip between Phoenix and San Francisco that encompassed a variety of urban and suburban driving conditions).

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THE POWER STEERING OIL COOLER AND RETURN HOSE MAY BE REMOVED

- 1.5 POUND WEIGHT REDUCTION
- \$18 COST SAVINGS AT RETAIL

POWER STEERING
SYSTEM SCHEMATIC

POWER STEERING SYSTEMS MODIFICATIONS

FIGURE 3-25

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THE 70 AMP STOCK ALTERNATOR IS REPLACED WITH A 60 AMP UNIT

FEATURES

- 12% INCREASE IN ALTERNATOR OUTPUT AT IDLE
- 40% REDUCTION IN MAX. OPERATING SPEED
- INCREASED BELT LIFE
- DECREASED INPUT POWER
- 5 POUND WEIGHT REDUCTION
- \$50 COST SAVINGS AT RETAIL

	ALTERNATOR SPEED @ ENGINE SPEED OF:	
	700 RPM	6000 RPM
STOCK SYSTEM	1450	12,400
WITH ACCESSORY DRIVE	2600	7,500

ALTERNATOR MODIFICATIONS

FIGURE 3-26

3.4.3 Cooling System Matching

The stock three-row radiator was replaced with a two-row radiator from a non-airconditioned Mustang II and the stock five-blade flex fan was replaced with a four-blade fixed-pitch fan. Experimental data was obtained for this configuration and compared to stock configurations for grade and constant speed conditions. Figure 3-27 shows top-tank temperature-versus-vehicle speed for both configurations. These data were obtained on a two-mile track closed to average wind effects on cooling. The air conditioning setting was at maximum cooling. At 70 mph, there is only a 4° to 6°F reduction in air to boil with the modified system, although the radiator size had been reduced considerably. This relatively small cooling capacity reduction, considering the large change in radiator size, is probably a result of reduced water pump cavitation resulting from slower pump operation at high vehicle speeds with the accessory drive system.

Results from comparative tests for hill climbing cooling performance are presented in Figure 3-28. These particular data were acquired on a seven-percent grade northwest of Phoenix (rising approximately 2000 feet in a distance of seven miles). The data is presented as top-tank temperature-versus-elapsed distance for the stock, and modified configurations. The air conditioning was at maximum and the vehicle was fully warmed. Speed was held at 40 mph in third gear for the entire ascent. The modified system was running a top tank temperature 10°F higher than the stock configuration at the end of the climb, which is essentially totally accounted for by the difference in ambient temperature between the two test sequences.

TOP TANK TEMPERATURE VS VEHICLE SPEED

- AC SET AT MAXIMUM
- TWO-MILE OVAL TRACK

STEADY STATE ROAD COOLING SYSTEM PERFORMANCE

FIGURE 3-27

TOP TANK TEMPERATURE VS MILES TRAVELED

- AIR CONDITIONING SET AT MAXIMUM
- SEVEN PERCENT AVERAGE GRADE
- THIRD GEAR AT 40 MPH

GRADE CLIMBING COOLING SYSTEM PERFORMANCE

FIGURE 3-28

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In addition to the described road load and hill climbing data runs, the vehicle was driven over the normal variety of conditions (urban and suburban) with full air conditioning loads through the Phoenix summer, with no indication of cooling system deficiencies.

3.4.4 Cost Summary

A cost summary for the accessory drive, and various options available to the original equipment manufacturer (OEM), are presented in Figure 3-29.

This summary is presented to demonstrate the most cost effective installation technique. Retail, or over-the-counter costs, were used to show consumer benefits. Initial cost and fuel savings costs were combined to demonstrate consumer net costs or savings for a two-year period.

The first column shows a \$28.00 net consumer cost for an after-market accessory drive add-on. The customer's initial stock accessory system cost does not include modification or deletion benefits. The two-year fuel savings do not fully pay for the add-on, but do result in improved vehicle and accessory performance.

The second column illustrates OEM drive installation cost advantages with no consideration given to rematching accessories to required levels. The cost of miscellaneous items deleted by the drive installation are reduced from the drive cost and passed on to the customer at a \$33.00 cost savings over a two-year operating period.

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COST ITEMS	CUSTOMER ADD ON	OEM DRIVE INSTALLATION	OEM SYSTEM INSTALLATION
ACCESSORY DRIVE SYSTEM	\$80.00	\$80.00	\$ 80.00
COOLING SYSTEM MOD	—	—	(55.00)
FAN MOD	—	(15.00)	(15.00)
ALTERNATOR MOD	—	—	(50.00)
POWER STEERING MOD	—	—	(18.00)
MISC. PULLEYS & BRACKETS	—	(46.00)	(46.00)
CUSTOMER INITIAL COST	\$80.00	\$19.00	(\$104.00)
CUSTOMER 2-YEAR COST*	\$28.00	(\$33.00)	(\$156.00)

* 15,000 MILES PER YEAR, \$0.60 PER GALLON, STOCK ECONOMY 18 MPG AVE., 5½% MADS ECONOMY IMPROVEMENT RESULTS IN A TWO YEAR FUEL COST SAVINGS OF \$52.00

() DENOTES SAVINGS

RETAIL COST SUMMARY

FIGURE 3-29

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The final estimate shows a \$156 two-year consumer savings for an OEM drive installation and complete accessory system optimization.

Dollar evaluations in this chart are a means of defining the principle. The absolute values presented are not as important as the fact that vehicle manufacturers have the best control over accessory drive system cost effectiveness, and, if installed by the manufacturer, can result in significant cost reductions.

4.0 PHASE V TECHNICAL SUMMARY

The Automotive Accessory Systems Optimization Program advanced as an incremental process involving conceptual studies, design, fabrication and development of an accessory drive system. Program Phases I through IV show that significant fuel economy and accessory performance improvements resulted from incorporating a variable ratio, constant speed accessory belt drive. Phase V evaluated further accessory system optimization in the areas of air conditioning and advanced hybrid drives utilizing alternate vehicle power sources and incorporating energy storage concepts to provide additional fuel savings.

The air conditioning studies included evaluation of alternative system approaches to conventional automotive systems. For example, the freon cycle air conditioning system was evaluated against alternatives such as the air cycle, and waste heat recovery systems. Trade-off studies were conducted to determine the most suitable air conditioning system for the automobile. The primary goal was to maximize fuel economy, but considerations of cost, complexity, noise, installation flexibility and durability were properly weighed to assure solutions practical for the automotive industry.

The hybrid accessory drives study included a preliminary study of various concepts, detailed analyses of the most promising approaches, final selection, and preliminary design of the system recommended for further development and demonstration.

Air conditioning optimization and hybrid accessory drive study results were applied to the same baseline vehicle as the accessory drive development. This allowed improvements to be projected from the baseline performance and analytical data already developed.

The following program plan outline describes the Phase V efforts:

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A DIVISION OF THE GARRETT CORPORATION
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Vehicle energy envelopes were evaluated relative to the following driving cycles:

- Federal Driving Cycle (LA-4)
- EPA Highway Fuel Economy Test (HWFET)
- Constant Speeds

The existing vehicle computer model was modified, with analytical and actual test data, to determine the nature and magnitude of the following energy sources:

- (a) Engine cooling system waste heat
- (b) Engine exhaust waste heat
- (c) Engine exhaust flow and pressure
- (d) Vehicle driveline deceleration losses
- (e) Vehicle braking energy

Task 2 - Hybrid Drives

Hybrid accessory drive systems and energy storage techniques using Task 1 vehicle energy envelope data were definitized.

The following systems were studied:

- (a) Exhaust gas turbines
- (b) Turbo compounding
- (c) Wheel and driveline braking devices
- (d) Fueled accessory engines
- (e) Waste heat engines (i.e, Rankine Bottoming Cycle)

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The following energy storage concepts were evaluated:

- (a) Flywheel (mechanical)
- (b) Battery (electrical)
- (c) Accumulators (hydraulic)
- (d) Expansion and recovery systems (thermodynamic)

Hybrid systems feasibility was determined on the basis of state-of-the-art-technology evaluation, complexity, adaptability to auto use, efficiency, advanced A/C compatibility, fuel economy potential, envelope, weight, noise, emissions, maintainability, durability, drivability, and vehicle and accessory performance.

The feasible systems were modeled and computer evaluated to determine the optimum system, based on fuel economy and performance.

Task 3 - Advanced A/C

Advanced air conditioning concepts were studied for use with a hybrid drive system. The study included the following concepts:

- (a) Variable displacement pumps - two-phase refrigerants
- (b) Air cycle systems
- (c) Absorption systems
- (d) Waste heat systems
- (e) EGT and turbo-compound systems

Task 4 - Vehicle Data

The Mustang II test vehicle was instrumented, as required, to determine engine, accessory, and vehicle system performance parameters. Test data was collected for 200 miles of city and highway driving.

Task 5 - Trade Study

A complete and systematic trade study, showing the detailed analysis, evaluation and matching of hybrid drives and A/C systems was prepared.

4.1 Vehicle Energy Source Simulation

The hybrid accessory drive system studies are dependent on knowledge of available vehicle energy sources and the energy magnitude and development rate.

The effort to complete this task was organized into four areas:

- o Energy Source Definition
- o Vehicle Testing
- o Source Analysis
- o Energy Envelope Simulator

4.1.1 Energy Source Definition

The vehicle energy envelope is described when all forms of produced or dissipated energy (potential or kinetic) are considered. The general classification of energy sources is as follows:

- o Fuel
- o Road Load (rolling resistances and aerodynamic drag)
- o Inertia
- o Engine (includes work, cooling and exhaust)

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- o Drive Line (includes torque converter, transmission, drive shaft, and differential losses)
- o Accessories (includes water pump, fan, air pump, alternator, power steering and air conditioning)

The objective was to conserve accessory power and decrease engine work by utilizing an available untapped energy source of sufficient magnitude to augment accessory power requirements. Probable sources shown in Figure 4-1 are defined as:

- o Fuel
- o Deceleration (braking and drive line losses)
- o Engine cooling
- o Engine exhaust

Other energy sources are not available due to negative flow or insufficient, recoverable magnitude.

4.1.3 Vehicle Testing

The 1975 Mustang II test vehicle was used to empirically determine deceleration and engine operating parameters to permit energy sources evaluations.

Acceleration coastdown tests were performed to determine drive line losses and engine speed-versus-vehicle speed for computer simulation. The vehicle was allowed to freely decelerate from various velocities with the throttle set in the idle position. Vehicle speed-versus-deceleration time is shown in Figure 4-2 and engine speed-versus-vehicle speed is shown in Figure 4-3. Data presented in Figure 4-3 is used directly in the computer model to determine accessory loads during decelerations. Data presented in Figure 4-2 is used to calculate coast power, which is a function of drive line inertias ($NI\alpha$), and

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CYCLE ANALYSIS

- EPA CITY
- EPA HIGHWAY
- CONSTANT VELOCITY

VEHICLE MODEL

- 1975 FORD MUSTANG II
- 2800 CC V-6 ENGINE
- AUTOMATIC TRANSMISSION/
AIR CONDITIONED

ENERGY SOURCE EVALUATION

FIGURE 4-1

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DECELERATION RATES
1975 MUSTANG II

FIGURE 4-2

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$$N_{\text{ENG DECEL}} = 41.66V + 250.2$$

- INITIAL VELOCITY = 60 MPH
- △ INITIAL VELOCITY = 50 MPH
- INITIAL VELOCITY = 40 MPH
- ◇ INITIAL VELOCITY = 30 MPH

ENGINE SPEED DURING DECELERATION
1975 FORD MUSTANG II
2800 CC ENGINE

FIGURE 4-3

vehicle inertias ($W/S a\bar{V}$). When resultant coast power is compared with known road load power, shown in Figure 4-4, the drive-line drag-loss may be determined.

The engine exhaust system was instrumented to measure exhaust gas temperature and pressure. Exhaust temperature, back pressure, engine speed and intake manifold pressure data was accumulated over the entire engine operating range. Data correlation with the intake manifold pressure/engine torque relationship given in the manufacturers map data, resulted in developing the following expressions for computer model use:

- o $EGT = 980 + 0.136 N_E + 1.9167 \tau$
- o $EMP = 14.014 + 0.0233 (N_E/1000)^{2.5} + 0.00891 \tau$

where:

EGT = Exhaust Gas Temperature ($^{\circ}R$)
EMP = Exhaust Manifold Pressure (psia)
 N_E = Engine Speed (rpm)
 τ = Engine Torque (lb-ft)

Overall emission control air pump performance calibration yielded the following airflow-versus-engine speed relationship, by using manufacturers' air pump map data:

- o $W_{AP} = 0.5175 (N_E/1000)$

where:

W_{AP} = Air pump flow into exhaust manifold (lb/min)

DRAG HORSEPOWER VS VEHICLE VELOCITY
DURING COASTDOWN
1975 FORD MUSTANG II
WITH AUTOMATIC TRANSMISSION

FIGURE 4-4

4.1.3 Source Analysis

The energy source analysis was mainly concerned with simulating engine heat and flow characteristics to be used for calculating exhaust energy.

Manufacturers' engine torque, speed, and fuel flow data were compared with airflow and air/fuel ratio information from other engine sources. Data was cross-plotted and scaled to produce the Mustang II airflow map shown in Figure 4-5. Combustion efficiency, presented on Figure 4-6, is a major function of air/fuel ratio. The lower heating value for gasoline may be expressed as:

$$\circ \quad \text{LHV} = 15,900 + 15,800 \text{ H/C}$$

where:

$$H = 1.008m \text{ [for } H_m]$$

$$C = 12.0/n \text{ [for } C_n]$$

$$\text{Gasoline Hydrocarbon} - C_8 H_{18} \text{ and } H/C = 0.18884$$

$$\circ \quad \text{LHV} = 18,884 \text{ Btu/lb}$$

The exhaust gas constant (R) varies with fuel/air ratio but may be expressed as a constant at stoichiometric conditions with accurate results:

$$\text{Stoichiometric fuel/air ratio (f)} = 0.0667$$

$$R_M = R_A + \frac{382 f}{1 + H/C} (H/C) \frac{1}{1 + f}$$

where:

$$R_A = 53.3 \text{ ft-lb/lb-}^\circ\text{R for air}$$

$$H/C = 0.18884$$

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MUSTANG II
ENGINE AIRFLOW MAP

FIGURE 4-5

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COMBUSTION EFFICIENCY VERSUS
AIR/FUEL RATIO

FIGURE 4-6

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$$\circ \quad R_M = 53.8 \text{ ft-lb/lb-}^\circ\text{R for exhaust gases}$$

The variation in specific heat ratio (α) was determined to be a function of exhaust gas temperature (EGT) and air/fuel ratio:

$$\circ \quad \alpha = 1.4123 - 0.05896 (EGT/1000) + 0.00719 (EGT/1000)^2 \\ + 0.004/1.667 [1/f - 1.50]$$

Total exhaust flow is the sum of: carburetor inlet airflow (from Figure 4-5); engine fuel flow, from existing simulator engine load data; and air pump flow from the expression in the preceding section:

$$\circ \quad W = W_A + W_F + W_{AP}$$

All parameters required to calculate exhaust energy rates were defined:

$$\circ \quad Q_X W_E (\alpha R / \alpha - 1) (EGT - 590) \text{ in Btu/min}$$

From fuel flow rate (W_F), fuel lower heating value (LHV), and combustion efficiency (η_B in Figure 4-6), fuel input energy to the engine may be calculated:

$$\circ \quad Q_F = W_F \cdot \eta_B \cdot \text{LHV in Btu/min}$$

Engine crankshaft work (Q_E) is chiefly a function of driving cycle, road load, vehicle weight, and drive line performance. This energy term existed in the computer model along with the ability to define all accessory loads.

Radiant heat losses are difficult to evaluate. These losses vary considerably with vehicle speed and are estimated to average approximately 700 Btu/min with a range of 400 to 1200 Btu/min.

Engine cooling energy (Q_C) may now be calculated as a result of the engine energy balance:

$$Q_C = Q_F - Q_E - Q_X - 700 \text{ in Btu/min}$$

This expression satisfies the engine cooling load for air conditioning OFF operation. With air conditioning ON, the engine cooling load increases proportionally to air conditioning cooling demands, to reflect the rise in radiator air inlet temperature caused by condenser heat rejection.

4.1.4 Energy Envelope Simulator

The computer program used to simulate the Mustang II baseline vehicle was modified to calculate engine fuel energy input, plus cooling and exhaust energies. Braking energy and drive line loss calculations were added to the program, and an on-line plot routine was incorporated to present these parameters graphically.

Program logic was upgraded to simulate idle and deceleration modes for driving cycles. The major addition, with respect to the deceleration mode, was calculating engine speed as a function of vehicle velocity.

Three computer simulations are included in this report to demonstrate new program features. These included (1) a constant velocity run with velocities from 10 to 80 mph in increments of 5 mph, (2) EPA city driving cycle, and (3) EPA highway driving cycle. In each case, the accessories were driven with stock and variable ratio belt drive, air conditioning on and off. Driving cycle data printouts are extensive, and therefore, could not be included in this report.

Computer plots of various energy rates-versus-velocity are presented in Figures 4-7 through 4-16 for the constant velocity analysis.

[illegible]

FIGURE 4-7

Page 4-18

FIGURE 4-9

COOLING ENERGY (Q_c)

STOCK

VRB

A/C OFF

MPH

31-2612
Page 4-19

31-2612

FIGURE 4-11

ENGINE ENERGY (QE)

FIGURE 4-13

FIGURE 4-14

FIGURE 4-16

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The total vehicle energy distribution is shown, on these figures, in the form of exhaust, fuel, shaft, cooling, and accessory power for A/C "OFF" and "ON" operation.

The data shows that the exhaust energy rate is approximately equal to engine shaft power, with each equal to approximately 25 percent of the total input energy rate. Total cooling system load, including air conditioning condenser heat rejection, is approximately 36 percent of the total input energy rate. This value will decrease to approximately 27 percent for air conditioning OFF operation.

Total cycle energy data for the EPA city and highway driving cycles are presented on Table 4-I. Braking energy and drive line loss-versus-time is presented in addition to the energy sources discussed in the constant velocity analysis. Since only a recoverable drive line loss is of interest, this parameter is set to zero except during deceleration periods.

4.2 Hybrid Accessory Drives

Hybrid accessory drive concepts may be classified relative to energy sources used to power or augment accessory requirements:

<u>Energy Source</u>	<u>Drive Concept</u>
o Engine Crankshaft	o variable ratio belt drive
	o Off-peak flywheel
o Engine Exhaust	o Turboaccessory compounding
	o Rankine bottoming cycle
o Engine Cooling	o Waste heat recovery expander
o Vehicle Deceleration	o Regenerative braking
	o Drive line loss regeneration
o Fuel	o Auxiliary power units (APU)

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TABLE 4-I

ERDA/TEC ACCESSORY STUDY, PHASE 5
1975 MUSTANG II ENERGY ENVELOPES

Vehicle Condition	Accessory Condition	Driving Cycle	mpg	Gallons of Fuel	Total Energy BTU's (1)						
					Fuel	Shaft	Accessory	Exhaust	Cooling	Braking	Driveline
Stock	A/C off	Constant Vel	18.29	.766	80493	19553	3940	22782	21627	0	0
		City	15.72	.474	50576	11281	2222	12029	11306	2029	79
		Highway	21.50	.477	52735	13760	2811	15257	14828	395	168
	A/C on, 30 amp	Constant Vel	16.80	.834	88147	22993	7380	24939	31699	0	0
		City	14.19	.525	56965	14634	5863	13759	20766	1735	239
		Highway	19.58	.524	57949	16035	5154	16790	21339	343	234
VRB Accessory Drive	A/C off	Constant Vel	19.27	.726	76614	18165	2552	21512	20406	0	0
		City	15.88	.469	50090	11197	2138	11880	11054	2029	79
		Highway	22.62	.453	50115	12605	1657	14489	14130	395	168
	A/C on, 30 amp	Constant Vel	18.06	.775	82473	20924	5311	23078	29955	0	0
		City	14.70	.507	54974	14011	5239	13175	19983	1735	239
		Highway	21.10	.486	53757	14232	3351	15544	20196	343	234

NOTES: (1) Radiant heat losses are 700 BTU/min average.

(2) A constant velocity cycle consists of operating the vehicle at constant velocities of 10 to 80 mph in 5 mph increments with constant accelerations between velocities. It does not contain decelerations.

Cycle	Length (miles)	Time (seconds)
(2) Constant Velocity	14.0	1417
City	7.45	1368
Highway	10.26	762

4.2.1 Drive Line Regeneration Concepts

The engine crankshaft concept, where a flywheel and variable ratio belt (VRB) drive have been combined, is shown on Figure 4-17. The flywheel is energized during vehicle deceleration by producing negative drive line torque through the engine and torque converter to provide braking. The flywheel stores braking energy during deceleration then feeds it back into accessories, via the VRB drive, during vehicle accelerations when the engine power margin is critical.

4.2.2 Exhaust Energy Conservation Concepts

Concepts for recovering engine exhaust energy are shown in Figures 4-18 and 4-19. The exhaust energy very nearly equals engine shaft output energy but is not in an easily recoverable form. Engine performance and fuel consumption are sensitive to exhaust back pressure required for the turbo compounding concept. Exhaust flow at engine part throttle operation is very low making the turbo or Rankine recovery concepts marginally feasible.

A detailed analysis explaining performance inadequacies of turbocharging to augment accessory power requirements and improve fuel economy is presented in following sections of this report. The results generally apply to all secondary expansion devices that elevate the primary engine exhaust back pressure.

Hybrid accessory drives involving engine cooling water energy recovery systems are marginally feasible because of heat exchanger and expander size and complexity, and engine temperature control problems. Engine efficiency and emissions production are almost directly proportional to operating temperature. The engine cooling system is thermostatically controlled to maintain engine operating temperature within a narrow range where actual ΔT across the engine is approximately 20°F and the EPA city cycle cooling energy rate is 485 Btu/min (11.4 hp).

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- HIGH MANIFOLD VACUUM (COASTING)
VRB AT MAXIMUM SPEED-UP RATIO DRIVING ACCESSORIES THROUGH OVER-RUNNING CLUTCH AND FLYWHEEL THROUGH ON-OFF CLUTCH
- LOW MANIFOLD VACUUM (ACCELERATING)
FLYWHEEL DRIVING ACCESSORIES THROUGH ON-OFF CLUTCH. FLYWHEEL CAN NOT DRIVE VRB (AT MINIMUM RATIO) THROUGH OVER-RUNNING CLUTCH
- MID MANIFOLD VACUUM (CONSTANT SPEED - IDLE)
VRB DRIVING ACCESSORIES THROUGH OVER-RUNNING CLUTCH. FLYWHEEL ON-OFF CLUTCH DISENGAGED.

CONTROL SENSE

VARIABLE RATIO BELT - ENGINE SPEED

ON-OFF CLUTCH - MANIFOLD VACUUM AND FLYWHEEL/ENGINE SPEED COMPARATOR

OFF-PEAK FLYWHEEL

FIGURE 4-17

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FUNCTION

ENGINE EXHAUST ENERGY CONVERTED TO SHAFT POWER WITH TURBINE. TURBINE POWER ASSIST TO MADS DRIVEN ACCESSORIES VIA REDUCTION GEARS AND OVER-RUNNING CLUTCH

DEVELOPMENT PROBLEM AREAS

- LOW TURBINE POWER YIELD DUE TO LOW EXHAUST FLOW AT PART THROTTLE
- TURBINE EXHAUST BACK PRESSURE REDUCES FUEL ECONOMY
- TURBINE SPEED REDUCTION APPROXIMATELY 45:1
- ENERGY STORAGE (FLYWHEEL) NOT FEASIBLE. PEAK OUTPUT OCCURS AT EXTREME ACCELERATIONS OR W.O.T. OPERATION THAT IS NOT INCLUDED IN NORMAL DRIVING CYCLES.

EXHAUST GAS TURBINE/ACCESSORY DRIVE COMPOUND

FIGURE 4-18

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MUSTANG II - THERMO ELECTRON LONG HUAL
DIESEL COMPARISONS AT 60 MPH

PARAMETER	MUSTANG	DIESEL
ENGINE OUTPUT HP	37	250
ENGINE SPEED-RPM	3000	1800
EXHAUST TEMP.-°F	1040	900
EXHAUST FLOW-LB/MIN	5.8	46
RANKINE CYCLE OUTPUT HP ^a (MAX)	3.3*	26

*DOES NOT CONSIDER FIXED LOSSES
AND ADDITIONAL FAN POWER

RANKINE BOTTOMING CYCLE

FIGURE 4-19

State-of-the-art heat exchanger/expanded processes involving low temperature driving sources convert thermal to mechanical energy in the 5 to 10 percent efficiency range. Assuming critical engine temperature control can be maintained, and packaging of large heat exchanger and expanders (also heavy) can be arranged, a reasonable energy recovery from the cooling system would be 0.5 to 1.0 horsepower. A more feasible use of cooling system energy may be to use the source to drive a chemical or waste heat air conditioning process. These concepts are discussed further in the advance air conditioning study effort.

4.2.3 Wheel Regeneration Concepts

Regenerative braking and accessory/flywheel compounding principles are shown in Figure 4-20. Concepts to achieve regenerative braking are shown schematically in Figures 4-21 through 4-23. Analysis shows that the combined efficiencies of the energy transportation and storage hardware result in poor overall performance for these concepts even when using mechanical storage and dissipation techniques. Electric and hydraulic storage concepts involve further system complexities and inefficiencies. Since highway driving cycle braking energy is 70 percent lower than developed in the city cycle, flywheel accessory augmentation is low.

4.2.4 Auxiliary Power Unit Concepts

Fueled auxiliary power units (APU) can be shown to have an adverse effect on overall fuel economy. As the name implies, an APU matches demand requirements when no other source is available. A small APU could be used to supply accessory power requirements in the Mustang II. Looking at EPA city cycle fuel economy and load data for the MADS equipped vehicle:

- o Fuel economy with full accessory load = 14.2 mpg,
with 0 accessory load = 16.8 mpg

REGENERATIVE BRAKING/ACCESSORY COMPOUNDING

FIGURE 4-20

HYDRAULIC REGENERATIVE BRAKING

FIGURE 4-21

TORQUE CONVERTER REGENERATIVE BRAKING

FIGURE 4-22

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ELECTRICAL REGENERATIVE BRAKING

FIGURE 4-23

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- o Total full accessory load = 5239 Btus
- o City cycle length = 22.8 minutes or 7.45 miles
- o Fuel density = 6.173 lb/gal
- o Assume low BSFC for APU at 0.5 lb/hr-hp

Calculations

- (a) Vehicle fuel consumption with 0 accessory load:

$$1/16.8 \text{ mpg} \times 7.45 \text{ miles} = 0.44345 \text{ gallons}$$

- (b) APU fuel consumption:

$$\frac{(5239 \text{ Btu}) (0.5 \frac{\text{lb}}{\text{Hp-Hr}}) (1 \text{ hr})}{(42.44 \frac{\text{Btu}}{\text{Hp min}}) (60 \text{ min}) (6.173 \frac{\text{lb}}{\text{gal}})} = 0.16665 \text{ gallons}$$

- (c) Total vehicle fuel consumption a+b = 0.6101 gallons

- (d) Fuel economy with APU for city cycle:

$$\frac{7.45 \text{ miles}}{0.6101 \text{ gal}} = 12.21 \text{ mpg}$$

The new city cycle vehicle fuel economy is 14 percent worse than the MADS equipped vehicle. This decreased fuel economy also occurs under ideal APU-accessory load design matching conditions. The actual APU would have to meet peak level requirements two times higher than 0.5 lb/hp-hr. Calculating in reverse, the APU required brake specific fuel consumption (BSFC) to meet a fuel economy of 14.2 mpg would be 0.24 lb/hp-hr, which is not feasible.

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4.2.5 Accessory Turbocompounding

Turbocharging is a well known method of increasing overall efficiency of a reciprocating engine by extending the maximum output capacity. A turbocharger consists of a turbine and a compressor, usually mounted on a common shaft. The turbine is driven by the reciprocating engine exhaust gas stream.

The compressor is mechanically driven by the turbine and functionally takes in and compresses ambient air to some desired level prior to entering the engine air intake. The reciprocating engine volumetric efficiency is greatly improved near W.T.O. operation resulting in higher engine mass flow and increased power. In some cases, turbocharging increases power by as much as 60 percent while slightly improving BSFC.

If the turbine generates more power than is needed to drive the compressor, the excess power can be used to drive accessories. This is referred to as accessory turbocompounding.

Figure 4-24 shows a hypothetical turbocompounded automobile engine system. The power turbine is shown being driven by exhaust gases and, in turn, driving the boost compressor and accessories. The compressor is shown taking in ambient air and "pumping" it into the engine induction system. The compression process heats the air to 300°F or more. To prevent large thermodynamic losses, an intercooler is used to cool the air to about 120°F before it enters the carburetor.

The air conditioning for the vehicle is involved in engine turbocompounding. Compressed and cooled air is bled off via a manually controlled valve and is directed through an expansion turbine, cooling it further before it enters the passenger compartment.

FIGURE 4-24

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The waste gate has a dual purpose. First, it prevents excessive boost pressures at the engine inlet which could cause a mechanical failure. This is accomplished automatically, by sensing intake manifold pressure and opening the waste gate at a predetermined setting, allowing the hot gases to bypass the turbine and exit through the exhaust system. The second waste gate purpose is to avoid excessive exhaust back pressure at conditions where the inlet boost is insufficient to maintain power. The waste gate again opens to allow the gases to bypass the turbine and exit via the exhaust. The latter function must be performed by a separate control circuit which senses both inlet boost and exhaust back pressure. When the waste gate is open, the engine is essentially returned to a naturally aspirated condition, and the accessories are then driven from the engine crankshaft.

Figure 4-25 shows a comparison of naturally aspirated and turbocharged performance for a typical automobile engine at constant engine speed. The region to the left of the intersection of curves is where 90 percent of the driving would occur. Unfortunately, the turbocharged engine fuel economy suffers by comparison in this region. It is apparent that turbocharging both increases power and reduces BSFC at or near wide open throttle.

Using data obtained from the mathematical model of the naturally aspirated 1975 Mustang II baseline vehicle, constant road load condition calculations were performed to predict overall performance with a hypothetical turbocharger installed.

Table 4-II presents three road-load conditions that were investigated. Baseline data is shown for 40 mph (Case A), 60 mph (Case B), and 80 mph (Case C). For each road-load condition analyzed, it is necessary to hold engine output power and speed constant in order to maintain the prescribed road-load conditions.

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COMPARISON OF TURBOCHARGED
AND NATURALLY ASPIRATED PERFORMANCE
AT CONSTANT ENGINE SPEED
FOR A TYPICAL RECIPROCATING ENGINE

FIGURE 4-25

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TABLE 4-II

- o Road Load Conditions
- o Naturally Aspirated, Stock Vehicle
- o Air Conditioning Off

	Case A	Case B	Case C
Altitude	Sea Lvl	Sea Lvl	Sea Lvl
Vehicle velocity, mph	40	60	80
Engine speed, rpm	2035	3008	3935
Total brake horsepower	16.1	35.6	65.6
Fuel flow, lb/hr	9.67	18.5	32.2
Exhaust gas temperature, °R	1337	1508	1682
Exhaust gas pressure, In. Hg. Abs.	30.8	31.6	32.8
Intake manifold pressure, In. Hg. Abs	16.2	18.0	23.8
Carburetor airflow, lb/hr	160.0	309.0	560.0
Total accessory horsepower	3.3	6.9	12.2
Air pump airflow lb/hr	63.2	93.4	122.2
Brake specific fuel consumption, lb/Hp-Hr	0.600	0.520	0.491

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Table 4-III presents the results for all three cases of constant road-load. Back pressure is assumed to be 10 inches Hg (gauge) and the intercooler pressure drop, 2 inches Hg, in all cases. Calculation details for Cases B and C are omitted since they are identical to those in Case A.

The calculations do not reflect equilibrium conditions in cases where accessory turbocompounding takes place. As the turbine begins to carry a portion of the total accessory load, the engine "unloads" slightly and since vehicle velocity is constant, a slightly lower throttle setting would be required, resulting in a slight fuel flow decrease. The study magnitude precludes considering the painstaking process of precisely matching the turbocharger to the engine. The fuel economy for turbocharged performance in Cases A and B would be improved if performance were refined through proper matching. An overall improvement of 25 percent would regain the stock 55 mph fuel economy but an improvement of this magnitude is considered radical.

Since fuel economy improvement is the prime objective, it is imperative that the devices used be effective at vehicle speeds below 40 miles per hour. The average EPA City Cycle Speed is 19.6 mph and the average EPA High Cycle Speed is 48.5 mph. A composite of the two cycles is formed with a 45 and 55 percent split, respectively. In addition, the maximum cycle speed is 60 mph.

Results for the cases considered show that turbocompounding would not be worthwhile on the Mustang II baseline vehicle. Turbocompounding would not be possible below 40 miles per hour because of insufficient exhaust system energy caused by part throttle operation. At higher speeds, some turbocompounding is possible but not enough to improve overall fuel economy.

Figure 4-26 shows that improvements in fuel economy would occur at vehicle speeds above the maximum speeds in the EPA driving cycles.

TABLE 4-III
CASES OF CONSTANT ROAD LEAD

	Case A	Case B	Case C
Vehicle speed, mph	40	60	80
Engine speed, rpm	2036	3008	3935
Total engine horsepower	16.1	35.6	65.6
Naturally aspirated fuel flow, lb/hr	9.67	18.5	32.2
Turbocharged fuel flow, lb/hr	13.54	24.5	38.1
Total required accessory horsepower(1)	3.3	6.9	12.2
Turbine power available for accessories, hp	0.27	1.37	3.1
Percent of total accessory power supplied via turbo-compounding	7.9	19.9	25.4
Naturally aspirated BSFC, lb/HP-HR	0.600	0.520	0.491
Turbocharged BSFC, lb/HP-HR	0.828	0.663	0.554
Percent increase in BSFC due to turbocharger	38	27.5	12.8
Fuel economy (miles/gal) naturally aspirated	24.7	19.4	14.9
Fuel economy (miles/gal) turbocharged(2)	17.7	14.7	12.6
Percent decrease in fuel economy due to turbocharger	28.6	24.5	15.5

NOTE: (1) Reflects air conditioning OFF condition.
(2) Does not reflect effect of engine unloading due to accessory turbocompounding.

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COMPARISON OF FUEL ECONOMY
TURBOCHARGED VERSUS NATURALLY ASPIRATED
1975 MUSTANG II

FIGURE 4-26

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From the results obtained in this analysis, it is obvious why turbo-compounding is used profitably on large trucks where a great deal of operating time is spent under high engine loads at or near wide open throttle. This regime of operation could also apply to passenger vehicles in the Mustang II class if the 2800cc displacement engine were rescaled into the 1400 to 1600cc range. A 3000 lb vehicle with a small turbocharged engine would operate frequently at wide open throttle to meet cycle acceleration requirements and provide good part throttle fuel economy because of reduced engine size. The concept of turbocharging engines in vehicles with low, naturally aspirated, power-to-weight ratios should be considered a prime, cost effective, method for meeting near term legislated fuel economy improvement.

4.3 Advanced Air Conditioning Analysis

Feasible applications for advanced air conditioning concepts were compiled and preliminary analyses were sufficient to critique the systems for physical and performance limits in qualitative terms.

The large variation in system types and operation presented a trade study challenge. During initial efforts, it was determined that quantitative comparisons would not reflect the relative merits of each system. Many of the systems evaluated herein, particularly those in the waste heat and air cycle categories, have been proposed to industry and evaluated by the respective inventors. Performance claims for these systems were somewhat overstated and the advantage emphasis did not address increasing coefficient-of-performance (COP) or decreasing input power requirements.

4.3.1 Existing Automotive Air Conditioning Systems

Existing automotive air conditioning systems are classified in accordance with the number of functions performed and the degree of control automation. In luxury cars having a complete environmental

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system with fully automatic controls, the driver needs only to select a temperature setting and actuate the system. This is an all-weather system providing heating, cooling, fresh air mixing, and humidity control. Defroster operation is the only manually selected function.

The factory-installed system most commonly used provides all the above functions through manual selection of heating, cooling, ventilation, and defrosting. These systems usually have a control to add engine heat during the cooling mode to obtain the desired temperature balance, or to provide cooling and reheat during the heating mode to control humidity.

The least expensive factory installed system provides heating, cooling, ventilating, and defrosting, all as separate, manually controlled functions. Heating and cooling modes are independently temperature controlled; humidity control is not possible during the heating mode.

Add-on or after-market air conditioners provide the single function of recirculating cooled air. This system does not operate in conjunction with the factory installed heating and ventilating system.

The automobile without air conditioning has only one main power consuming device in the cab environmental system, the air recirculation fan. This fan need only provide 20 to 40 m³/min for heating and defrosting, which consumes approximately 0.12 kW (0.16 hp).

When an air conditioning system is added, power consumption increases drastically. The recirculation fan must now provide 87 m³/min with a power penalty of 0.192 kW (0.26 hp). The air conditioning compressor has a magnetic clutch that consumes approximately 0.05 kW (0.064 hp), while the compressor consumes engine power at a rate proportional to cooling load and engine speed. At road speeds of 80 km/hr (50 mph), the compressor will consume about 3.73 kW (5 hp).

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For the example stated above, the power penalty for air conditioning is 3.85 kW (5.2 hp), which is approximately 20 percent of the power required to maintain vehicle speed at zero grade.

The air cooling requirement is seasonal and air conditioner operation is only required during 30 to 40 percent of the year. However, when the air conditioner is linked with the heating system to provide humidity and temperature control, the operational requirements are 60 percent of the year. Utilizing a 24,000 km (15,000 mile) driving year at an average speed of 64 km/hr (40 mph), as an example, the total air conditioner power consumption for the year can be 746 kW-hr (1000 hp-hr). This is approximately 18 percent of the average power required to drive the vehicle. The above estimate does not consider factors other than gross, average performance variables, but results do indicate that air conditioning is a major contributor to fuel usage.

Regardless of how complex an air conditioning system may become, it still contains the following basic components (see Figure 4-27):

- (a) Compressor: The compressor pumps low pressure refrigerant (R-12), received from the evaporator, to the condenser inlet at high pressure. The compression process raises the heat content of the gas. The compression cycle is very close to an adiabatic process.
- (b) Condenser: This refrigerant-to-ambient-air heat exchanger converts the high pressure, hot compressor discharge gas to a high pressure, hot liquid by removing heat. The heat removed is the heat of condensation required for a refrigerant phase change. Condensers usually have additional capacity to further cool the liquid, which is termed "subcooling". The condensing cycle is a near constant pressure process.

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BASIC AIR CONDITIONING SCHEMATIC

FIGURE 4-27

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- (c) Receiver: The receiver stores surplus refrigerant during light load operation. It also contains a filter and dessicant to remove foreign material and moisture from the refrigerant.
- (d) Expansion Valve: This valve controls the flow of liquid refrigerant to the evaporator. The valve senses evaporator outlet temperature and modulates flow accordingly. Low temperatures decrease flow; high temperatures increase flow. The pressure and flow relationship across the expansion valve is maintained to assure complete refrigerant vaporization within the evaporator. By necessity, the valve relies on a small amount of superheat, at the evaporator outlet, to effectively modulate orifice size and guarantee complete vaporization. Superheat is the quantity of heat in addition to that required for complete vaporization. The vaporization cycle is a near constant pressure process. The expansion valve flow may be compensated with respect to evaporator pressure drop by biasing the temperature signal with the change in pressure across the evaporator.
- (e) Evaporator: The evaporator is an air-to-refrigerant heat exchanger. Vehicle compartment air is forced over the evaporator where heat is extracted to vaporize the refrigerant. The air cooling process causes water condensation on the evaporator air side surface; therefore, compartment air is dehumidified as well as cooled.

The basic system described above may be further modified by adding controls to negate the problem of evaporator freeze-up. The basic system components do not maintain the evaporator temperature above 0°C (32°F); therefore, under light load and humid conditions, the evaporator condensate will freeze. This makes the evaporator ineffective as a heat exchanger.

The thermal expansion valve (TXV) is the basic control device for all automotive air conditioning systems. The valve has a preset operating range, which is a strong function of evaporator capacity. Proper valve operation is critical because any liquid refrigerant allowed to leave the evaporator may slug and seriously damage the compressor. On the other hand, if the valve restricts too much refrigerant flow, system cooling capacity system will be degraded. Manual valve control in any automotive system is never permitted.

The most common method used in the past for controlling vehicle environment is compressor cycling. This method uses a temperature sensor on the evaporator surface evaporator that signals the compressor clutch to disengage when the selected temperature level has been reached. The sensor has a built-in lower limit that prevents evaporator freeze up. In recent years, this method has not been used because cycling the compressor at constant road speeds is undesirable, inasmuch as engine loading and unloading affects drivability comfort. When the compressor is cycled off, condensate that has been collected on the evaporator is re-evaporated into the compartment, thus raising the humidity, which is an undesirable effect. Some factory-installed systems still use compressor cycling to prevent evaporator freeze-up, but most systems now use a "suction throttling regulator" for this purpose.

The suction throttling regulator, which may be referred to by different manufacturers as a Robotrol-, EPR-, STV-, or POA-valve, prevents evaporator freeze-up by holding evaporator vaporization pressure above the saturated vapor freezing point. For refrigerant R-12, this minimum pressure setpoint is 303 kilopascals (44 psia). The regulator, which does not function until evaporator outlet pressure falls to this point, extends the system lower operating region by preventing evaporator freeze-up at low cooling load demands. At these low load conditions, the compressor will operate at a suction pressure

that raises the compression ratio and drives volumetric efficiency as low as 50 percent. The overall effect of this type of control is to decrease the system coefficient of performance (COP)*, in return for an extended operating range that eliminates all undesirable effects of compressor cycling.

4.3.2 Waste Heat Air Conditioning Systems

Waste heat systems have been categorized into Rankine cycle, absorption cycle, metal hydride, and molecular sieve systems.

Waste heat powered refrigeration systems offer the optimum economy increase due to minimum power input requirements. The compartment cooled air circulation fan is required in virtually all air conditioning systems and so will not be discussed in power input requirements.

Of the waste heat system discussed, the turbo-compression and vapor jet Rankine concepts appear to warrant additional evaluation. Proper refrigerant fluid selection or development for each system is critical for satisfactory performance and endurance. The cool-down capability of any waste heat system is, at best, marginal because the waste heat source has a low magnitude at engine idle and a long response time during the start and warm-up cycle.

4.3.2.1 Rankine Cycle

A biphasic fluid/gas is used to convert heat energy into mechanical power used for refrigerant gas compression during Rankine cycle operation. The systems under consideration use the same fluid in the boiler and refrigeration circuits, and is referred to as a "unitary Rankine cycle."

*COP is the general expression of system efficiency relative to cooling capacity and power consumption.

$$\text{COP} = \frac{\text{Cooling Capacity}}{\text{Compressor Power}}$$

In the power loop, fluid is boiled and superheated by engine exhaust gas. Super-heated vapor is expanded in the expander, which drives the compressor. After vapor leaves the expander, it passes through the recuperator and is cooled to a liquid in the condenser. The fluid is then passed through the liquid pump and back to the recuperator and boiler. The cooling loop is essentially the same as an ordinary refrigeration loop. A portion of the liquid flow from the condenser is directed to the expansion valve, where the pressure is dropped to the evaporator pressure. The compartment heat load is absorbed by fluid in the evaporator. The vapor is then recompressed in the compressor to a pressure equal to expander discharge pressure. The compressor flow combines with expander flow and is directed back to the condenser.

The basic Rankine Cycle has been selected as the prime waste heat air conditioning system and is discussed in greater detail in the trade-study portion of this report.

4.3.2.1.1 Fluid Selection

The working fluid selection is a trade-off study of system operating characteristics, fluid medium characteristics, and fluid safety considerations.

Fluids under consideration are Sulphurdioxide (SO_2), R-11, R-113, and R-114. Sulphur dioxide has desirable characteristics and has been utilized as a refrigerant for many years. It is a prime candidate for a waste heat system because of a high decomposition temperature, capable of operation up to 850°F. A major drawback is toxicity, requiring more exotic hermetic sealing and extensive crash protection.

Fluids R-11, R-113, and R-114 are low in toxicity, have reasonable condensing pressures, and low cost. However, those fluorinated hydrocarbons are performance limited due to low decomposition temperatures that requires large heat exchanger equipment.

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Heat transfer fluids, such as the family of fluorinals, offer boiler operating advantages. However, the refrigerant qualities are not sufficient for a unitary system.

A refrigerant fluid comparison chart of heat exchanger performance size and weight for one ton (12,000 Btu/Hr) of waste heat air conditioning is presented on Table 4-IV. It can be seen that SO_2 has a 20 and 47 percent system weight advantage over R-114 and R-113 respectively.

4.3.2.2 Rankine Cycle Vapor Jet Refrigeration

One of the waste heat powered air conditioning systems is an integrated cooling system and vapor driven jet pump refrigeration device. The system provides heating and cooling with a substantial decrease in power consumption, with attendant fuel economy improvement. An integrated engine cooling-cab environmental system must dissipate 40,000 to 70,000 Btu/Hr making large heat exchangers necessary and very high system flow rate. The only feasible expander for handling high gas flow rates is the vapor jet pump. Other mechanical expansion devices could be used but would be large and heavy.

A simplified system schematic diagram is presented on Figure 4-28. Several bypass valves for quick warm up, compartment heating, etc. have been omitted.

The operation is analogous to steam jet refrigeration, using high pressure gas through an ejector, creating a low gas pressure on an evaporator partially filled with the refrigerant (liquid phase of the motive gas). The low pressure on the refrigerant causes low temperature boiling, the latent heat of vaporization providing the refrigeration effect.

The automotive circuit uses the fluid, Refrigerant 11, as the engine block coolant. Pump flow and system sizing is such that saturated vapor leaves the coolant jacket, the exhaust manifold providing

TABLE 4-1Y

COMPARISON OF HEAT EXCHANGER PERFORMANCE, SIZE AND WEIGHT

Fluid	SO ₂	R-114	R-113
<u>Evaporator</u>			
Heat Rejection, Btu/min	200	200	200
Inlet Temperature, °F	40	40	40
Inlet Pressure, psia	27	15	2.7
Flow, lb/min	1.5	4.6	3.7
Size, in.	12 x 4 x 18	12 x 8 x 12	17 x 4 x 12
Weight, lb	23.1	23.4	42.9
<u>Condenser</u>			
Heat Rejection, Btu/min	480	646	601
Inlet Temperature, °F	130	130	130
Inlet Pressure, psia	135	74	18.5
Flow, lb/min	3.4	12.6	8.9
Size, in.	14 x 7 x 24	28 x 5.2 x 24	25 x 5.2 x 24
Weight, lb	45.1	60.8	54.3
<u>Boiler</u>			
Heat Load, Btu/min	297	491	400
Outlet Temperature, °F	850	348	353
Inlet Pressure, psia	700	700	240
Flow, lb/min	1.9	8.0	5.2
Size, in.	2.6 x 8.6 x 6	2.6 x 16 x 6	2.6 x 23 x 6
Weight, lb	12.6	14.0	19.9
<u>Recuperator</u>			
Size, in.	1 x 10 x 4	2 x 19 x 6	2 x 22 x 6
Weight, lb	3.5	5.2	6.0
Total Heat Exchanger Weight, lb (Existing Designs)	84.3	103.4	123.1

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SCHEMATIC OF VAPOR JET AUTOMOTIVE REFRIGERATION

FIGURE 4-28

superheat for the motive force, R11 gas vapor. Total ejector output (motive plus induced evaporator gas) flows through the condensor and receiver. The flow splits with the major portion going through the coolant pump to the engine cooling jacket, the remainder going through the expansion valve, evaporator, and into the suction chamber of the ejector. A high side recuperator heats coolant flow and cools total ejector flow prior to the condensor. A low side recuperator uses evaporator-out flow to cool inflow to the expansion valve. The recuperators provide significant system performance improvement and may be required in spite of increased costs. The coolant feed pump is the only power consumer. Feed pump motive flow is 421 pounds of coolant/hour with a 120-140 psi increase, about 0.18 hp at idle. Pump speed or flow modulation will be required to handle the range of coolant flow required from idle to maximum power.

Condensor size estimates are about three times that of existing units. However, this is partially compensated by eliminating the conventional radiator and heater.

Steam jet refrigeration uses $P_{\text{MOTIVE}}/P_{\text{RECOVERY}}$ ratios 40 to 100:1, whereas the proposed vapor jet system has a P_m/P_r of 5.0. A commercial supply catalog indicates successful pumping with P_m/P_r of 4.0, using steam as motive and pumping air, but the recovery process efficiency at low pressure ratios is considered poor.

4.3.2.3 Waste Heat Absorbtion Air Conditioning

The most common heat driven air conditioning devices are absorb-tion units. The only power input to these devices is a circulating pump. The lack of compression machinery makes absorption units a desirable option. A simplified schematic diagram of an absorption unit is presented on Figure 4-29.

AUTOMOTIVE ABSORPTION AIR CONDITIONING

FIGURE 4-29

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The absorbant must be a material that is easily regenerated at temperatures available as waste heat from the automobile. The most common materials are molecular sieves and silica gel, used as the absorbant, and water vapor, used as the refrigerant. This combination will produce refrigeration, but is prohibitively large for an automotive application. Another combination would be to use a metal hydride, such as iron or magnesium titanium, with hydrogen.

Absorption and desorption information has been reported that indicates the metal hydride system shown in Figure 4-30 would produce refrigeration. However, due to the complexity and experimental nature of this combination, coupled with the marginal benefits to be derived should it be developed, this concept was not further evaluated.

A study of absorption refrigeration systems coupled with the quantity and quality of automotive waste heat reveals that the heat exchange equipment becomes large. A normal design point for current automotive freon systems to reject heat is about 135° - 150°F from the condensor, allowing a relatively small condensor. Equivalent absorption systems need to reject heat at 120°F to be practical, thus the heat exchanger equipment becomes very large. Also, heat is rejected from the absorber, condensor and rectifier (if required) and added to the generator, resulting in at least three (3) pieces of heat transfer equipment, all generally large for automotive use.

An absorption system rejects 2.5 - 3.5 times as much heat as a liquid-vapor compression system, coupled with lower approach temperatures for rejection, resulting in roughly 10 times the bulk required for heat exchanger equipment in absorption systems compared to liquid-vapor systems.

Several system fluids used for analysis were a mix of ammonia/water, refrigerant 22/DME-TEG, and water/lithium bromide.

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HYDROGEN GAS/METAL HYDRIDE
BED COOLING WATER POWERED A/C SYSTEM

FIGURE 4-30

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The water/lithium bromide system must operate at approximately 0.120 psia in the evaporator and thus is highly sensitive to ambient leakage, requiring purge devices. Also absorber temperatures must be limited to an unrealistic 105°F - 110°F, otherwise the lithium bromide becomes crystallized. Ambient conditions below 32°F can freeze the water refrigerant during non-operating periods. These items, coupled with the large heat exchanger components required, eliminate this system from consideration for automotive application.

The ammonia/water system has been in use as a refrigeration system for many years in industrial applications. Ammonia/water systems require operating pressure near 300 psi and heat source operation near 300°F, eliminating the cooling system as a generator heat source. Ammonia is a toxic substance, also explosive in certain concentrations. Providing system integrity in the event of a vehicle accident alone is a monumental task. The toxicity problem and potential explosiveness removes ammonia systems from automobile use considerations. Predictions indicate coefficient of performance of 0.23.

Another combination of materials capable of providing absorption refrigeration is a combination of R-22 and DME-TEG. Refrigerant 22 is the refrigerant and dimethyl ether-tetraethylene glycol is the absorbent (solvent). The solution operating regimes are suitable for automotive use. There is limited operational data available on this solution in refrigeration systems so any sizing or performance predictions would require experimental data beyond the scope of Phase V. Using extrapolated data, a coefficient of performance was determined to be 0.26 with condensing temperatures of 123°F, again indicating very large heat exchange equipment.

4.3.2.4 Waste Heat Molecular Sieve Combined Refrigeration Cycle

During studies conducted on Rankine and absorption cycles, it became apparent that a combined cycle would have some merit for this application. Results of preliminary calculations indicate that a refrigeration cycle using chemical absorption and desorption to provide the vapor compression function is a viable candidate. A schematic of such a system is shown in Figure 4-31. The evaporator and condensor perform identical functions as in any Rankine cycle. Compression, however, is accomplished by absorbing the refrigerant in a molecular sieve or silica gel bed. The bed is then heated, which causes the refrigerant to be desorbed. This desorption process will occur at a pressure determined by the condenser internal temperature. The absorption process is an exothermic reaction; therefore, the bed must be cooled during this process. The amount of cooling required is the heat of vaporization of the refrigerant, plus the heat of reaction.

This concept is adaptable to a rotary bed design. This arrangement, shown in Figure 4-32, uses the refrigerant as the heat transport medium. The system can be completely hermetically sealed by enclosing the drive motors for the rotary bed and the two fans. The primary rotary concept advantage over the fixed bed method is that no extended heat transfer surfaces are required in the sorbent; hence, less bed weight is required to be heated and cooled. Because it is easier to maintain a minimum void volume during cyclic changes, higher efficiencies can be obtained. Various fluids and sorbent combinations will work; for example, water and molecular sieve type 4A. Water could also be used with many other molecular sieves and silica gel. Other refrigerants can be used that will operate at higher pressures, thus reducing components sizes. A candidate refrigerant is ammonia, which can be absorbed by molecular sieve types 5A or 13X. Sufficient data

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COMBINED CYCLE SCHEMATIC DIAGRAM

FIGURE 4-31

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ROTARY BED DESIGN SCHEMATIC

FIGURE 4-32

to perform a complete analysis was not available. Therefore, feasibility testing would be required prior to introducing this system into the trade-study.

4.3.3 Air Cycle Air Conditioning Systems

4.3.3.1 Vane Compressor

A new system, employing a positive displacement, rotary vane compressor and expander, has been under development. It has been described as The ROVAC Air Conditioning System in a paper presented at the Automotive Engineering Congress and Exposition in Detroit, Michigan, held February 18-24, 1975. The description and test data presented herein is taken from that paper.

Figure 4-33 illustrates, in a generalized fashion, the substance of the ROVAC system. In the open cycle form, the system consists of three basic components: an input-output duct, a rotary vane combination compression-expander-circulator, and a heat exchanger.

The inlet-outlet duct serves to provide a path for both the incoming flow of warm air and outgoing cold air. This ducting system is equipped with various processing media such as filters, sound control, or moisture separation means, depending upon the actual application of the particular system.

The circulator unit, which can be considered the heart of the system, consists mainly of two items: the stationary portion, termed the stator, and the rotating rotor-vane assembly. The stator housing is also equipped with two endplates which hold the rotor in the axial position and aid in containing the air that circulates through the machine. The rotor-vane assembly consists of a cylindrical rotor equipped with a series of sliding vanes. These vanes are arranged so

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BASIC ROVAC SYSTEM

FIGURE 4-33

that they can slide inside the rotor slots as the rotor-vane assembly is rotated within the stator, and thereby maintain continuous near-contact with the stator wall.

The heat exchanger, also termed the intercooler, serves to dissipate the heat generated during the operation of the system. This heat exchanger is principally a simple externally finned tube. For some applications, internal finning can also be used to increase the rate of heat transfer.

Table 4-V which follows, presents performance test data for a ROVAC unit. The test was performed in a facility employed to evaluate various prototype ROVAC air conditioning systems performance. Ambient operating conditions were: temperature = 110°F, pressure = 14.69 psia, and relative humidity = 57 percent.

It is evident from the information presented, that the test conditions imposed represent ambient conditions that are out of the normal conditions used to determine cooling capacity. To predict ROVAC performance under nominal design conditions selected for this study, the test data was reviewed to determine operating efficiencies of the device.

The apparent efficiencies, as calculated from the above noted test data, are:

Adiabatic Compressor	111 percent
Adiabatic Expander	80 percent
Frictional	60 percent

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TABLE 4-V

ROVAC MODEL 30B/45/9.5-4 PERFORMANCE

Parameter	Computer Predicted Performance	Actual Measured Performance
<u>Compressor Outlet</u>		
Temperature, °F	316	315
Pressure, psia	47.75	48.2
<u>Expander Inlet</u>		
Temperature, °F	105.5	108
Pressure, psia	47.10	47.5
<u>Expander Outlet</u>		
Temperature, °F	9	34
Pressure, psia	14.65	14.65
Rotor Speed, rpm	1510	1510
HP Drive	3.875	5.75
Air Mass Flow Rate, lb/hr	451.0	453
Cooling Capacity, Btu/hr	29,979	23,284
COP	3.04	1.59

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The compressor efficiency, greater than 100 percent adiabatic, indicates that heat transfer is taking place either during the compression process or prior to locating the compressor outlet temperature measurement. It would be expected that some heat transfer would occur between the compressor and expander due to the device construction.

Applying the above efficiencies to the normalized design point used for the turbomachinery equipment presented earlier, yields a higher than stock power requirement for the desired one ton of refrigeration. Even if the device should be developed and some improvement be obtained in efficiencies, AiResearch does not consider it energy competitive with other methods. Therefore, an accurate size determination of this concept was not attempted.

4.3.3.2 Brayton Cycle

The working gas compression and expansion process in a reverse Brayton cycle refrigeration system can occur in any compression expansion device. Of the cycles studied, the one employing a vacuum cycle with 100 percent fresh air required the least power to operate. The selected system characteristics and schematic are presented in Table 4-VI.

System operation is as follows. Fresh air is drawn in through a heat exchanger where it is cooled. Any condensed moisture is collected and then introduced into the cold side and re-evaporated for increased cooling. The air thus cooled is directed through the compartment, where it picks up heat. The compartment exhaust air is directed through an expansion turbine. There the gas is cooled to provide a portion of the power required to drive the compressor, which increases the air pressure to ambient so that it may be exhausted. The

TABLE 4-VI

ESTIMATED PERFORMANCE OF
VACUUM AIR CYCLE AIR CONDITIONER WITH
100 PERCENT FRESH AIR

<u>COMPARTMENT</u>		
Stabilized Temperature, °F	75	
Inlet Airflow, cfm	281	
Inlet Temperature, °F	40	
Inlet Humidity, gr/lb	33.7	
Outlet Temperature, °F	75	
Outlet Humidity, gr/lb	36.8	
Cooling Capacity, Btu/min	200.0	
<u>TURBOCOMPRESSOR</u>	<u>Turbine</u>	<u>Compressor</u>
Airflow, lb/min	22.35	22.35
Inlet Temperature, °F	75	86
Inlet Humidity, gr/lb	36.8	98.0
Inlet Pressure, psia	14.68	10.24
Pressure Ratio	1.414	1.440
Outlet Temperature, °F	34	160
Power, hp	5.23	9.59
<u>HEAT EXCHANGER</u>	<u>Hot Side</u>	<u>Cold Side</u>
Airflow, lb/min	22.35	22.35
Inlet Temperature, °F	90	34
Inlet Humidity, gr/lb	95.0	36.8
Outlet Temperature, °F	38	86
Outlet Humidity, gr/lb	33.7	98.0
Heat Load, Btu/min	496.6	496.6
<u>DRIVE UNIT</u>		
Input Power, hp	4.59	
Horsepower per Ton of Compartment	4.59	
Cooling Capacity

VACUUM AIR CYCLE

turbocompressor would operate at approximately 60,000 rpm, and therefore, would be difficult to drive directly from the automobile engine. Gearing losses would be high.

The drive that appears most promising would be incorporation of an integral electric motor to the turbocompressor shaft. To provide power levels required for this approach, an alternator design similar to the motor would be required so that size and power losses would be at acceptable levels.

Motor power can be reduced by injecting water into the heat exchanger low pressure cold side. Only a small quantity of water would be required. While it is understood that carrying water as an expendable is undesirable, the amount of water carried can be minimized by injecting it only during periods of high demand, such as compartment temperature pulldown.

The vacuum cycle does not achieve the desired low power with resultant fuel economy improvement, and it represents the minimum power air cycle system studied. Therefore, air cycle air conditioning does not qualify as a candidate in advanced air conditioning systems.

4.3.4 Vapor Cycle Refrigeration Concepts

Compressor sizing is a compromise between minimum horsepower, cooldown capacity, and steady state loads, for city and highway applications. Compressor discharge pressure is a function of ambient temperature, so the only control variable for various loads is flow modulation. Two techniques for flow modulation are varying compressor speed or compressor displacement.

4.3.4.1 Variable Speed Compressors

The variable speed hydraulic air conditioning drive can provide idealized compressor flow-versus-load. However, the cost and complexity of a variable displacement pump/motor would be better utilized on a variable displacement compressor. Combined pump/motor efficiency would probably negate the gains achieved by the compressor. Variable speed electric drives are becoming more feasible with the advent of high efficiency solid state motors and high speed permanent magnet alternators. Application of these devices for accessory power and control on mid 1980s, automotive propulsion engines deserve prime consideration.

4.3.4.2 Variable Delivery Compressor Concepts

Variable displacement, suction cutoff, and variable delivery helical screw compressor concepts have been studied in detail to provide sufficient trade-study data. These systems are presented in 4.4.2.1.

4.4 Trade Studies

The trade studies purpose was qualitative and quantitative evaluation of all hybrid accessory drive and advance air conditioning system concepts to determine the most feasible concept(s) for detailed design and analysis. The trade study was used as an engineering tool to derive concept conclusions on a comparative basis with respect to the following matrix of performance and physical factors:

Performance

- o Fuel Economy
- o Efficiency
- o Emissions
- o Durability
- o Drivability
- o Weight
- o Noise

Physical Factors

- o Initial Cost
- o Complexity
- o Weight
- o Maintainability
- o Adaptability
- o Envelope
- o Comfort

The major study effort was directed toward performance, specifically, fuel economy, while other factors will be controlled to preclude unacceptable performance. The "initial cost" function is a strong physical factor that must be controlled at the lowest possible level for consumer acceptance.

The complexities of the trade-study can best be understood by explaining that hybrid accessory drives and advance air conditioning concepts may require common energy sources. The two most suitable sources, vehicle kinetics and engine waste heat, are of insufficient magnitude to provide power for both functions and, depending on driving cycle conditions, very marginal for single function operation. Therefore, the trade-study must be performed at two discrete levels:

- o Component trade-study
- o System intergration trade-studies

The component trade-studies consisted of evaluation and analysis of all candidate hybrid drive and advanced air conditioning concepts to determine the optimum concept in each energy source category. The system integration trade-study combines drive and air conditioning concepts to maximize performance and energy source utilization resulting in the optimum fuel economy improving configuration.

4.4.1 Hybrid Accessory Drive Trade-Study

The performance and physical factors for each candidate hybrid drive are summarized in Table 4-VII. The optimization flow chart, based on Table 4-VII data is presented in Figure 4-34. Four of the seven basic concepts did not enhance fuel economy sufficiently and were eliminated in the initial study effort. Two of the concepts, accessory turbo compounding and accessory power units (APU) were found to actually decrease fuel economy. All concepts include a variable ratio, constant speed, accessory drive as a baseline; therefore, this concept results in zero fuel economy improvement. The probable recovery and utilization of lost driveline energy was discredited when the source magnitude was evaluated and found to have a maximum accessory augmentation potential of only 185 watts (1/4 horsepower).

The off-peak flywheel, Rankine bottoming, and regenerative wheel braking concepts were modeled and evaluated over the EPA city and highway driving cycles in the Mustang II computer simulator. Efficiency losses contributed by reclamation, storage, transportation and delivery hardware for the regenerative wheel braking concept consumed nearly all the available braking energy resulting in very minor fuel economy gains. The three types of Rankine bottoming

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TABLE 4-VII

HYBRID ACCESSORY DRIVES
TRADE STUDY SUMMARY CHART

Concept	City & Highway Fuel Economy Improvement Potential (%)	Maximum Energy Recovery Ratio	Maximum Power to Weight Ratio (hp/lb) ₂	Total System Volume (ft ³)	Relative Factors					
					Initial Cost	Complexity	Maintenance	Adaptability	Noise	Emissions
Turbo Accessory Compounding	3.5 - 7.8	0.019	0.003	1.0	Low	Moderate	Low	Fair	Low	Higher
Exhaust Rankine Bottoming Cycle	6.2 - 13.2	0.088	0.016	2.0	High	Moderate	Moderate	Fair	Moderate	Lower
Engine Cooling Rankine Cycle	4.2 - 9.5	0.047	0.005	5.0	High	High	High	Poor	Moderate	Unknown
Combined Waste Heat Ranking Cycle	6.5 - 15.0	0.07	0.012	6.0	High	High	High	Poor	Moderate	Unknown
"Off Peak" Flywheel	6.0 - 8.5	0.50	0.08	1.0	Moderate	Moderate	Moderate	Fair	Low	Lower
Regenerative Braking	4.5 - 7.8	0.21	0.03	3.0	High	High	Moderate	Poor	Moderate	Lower
Accessory Engines (APU)	0	0.27	0.12	3.5	Moderate	Low	High	Fair	High	Higher

1 The baseline for all accessory systems includes a variable ratio accessory belt drive developed in program Phases I-IV. The fuel economy predictions are for A/C on operation.

2 Power to weight ratio considers maximum accessory power augmentation and system replacement weight.

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HYBRID ACCESSORY DRIVE TRADE STUDY

FIGURE 4-34

systems evaluated, exhaust, engine cooling and combined cooling systems, showed the combined system had a slight fuel economy edge over the exhaust powered system. However, the combined system size, complexity and adaptability resulted in a poor overall rating. Therefore, the exhaust Rankine hybrid accessory drive was selected as the prime waste heat system.

The off-peak flywheel concept out performed all other candidates in the efficient use of braking energy and was selected as the prime regenerative braking concept.

4.4.1.1 Hybrid Accessory drive Design Models

Two off-peak flywheel, hybrid accessory drive, design concepts are presented in Figures 4-35 and 4-36. Both systems are driven from the engine crankshaft and mount in the engine compartment as an accessory. The major difference in the concepts is the method used to charge the flywheels. The constant speed concept shown in Figure 4-35 uses a second variable ratio belt transmission to feed and control braking energy into the flywheel at high efficiency, while the Figure 4-36 concept dissipates some braking energy in the flywheel input clutch in favor of better control and drivability. The operating cycle for both concepts is essentially the same:

- (a) Vehicle deceleration through a braking command engages the flywheel. The flywheel absorbs vehicle kinetic energy as an accessory, and stores it for later use.
- (b) During the idle or constant speed period following a deceleration, one-way clutches, magnetic couplings, and ratio control isolate the flywheel from the accessory array.

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DESIGN DATA

- SPEED RATIOS:**
- CRANK TO ACCESSORY SHAFT 2:1 TO 0.5:1 VARIABLE (4:1 OVERALL)
 - ACCESSORY SHAFT TO FLYWHEEL INPUT SHEAVE 1.73:1 TO 0.577:1 (3:1 OVERALL)
 - FLYWHEEL GEARBOX 7:1
- SPEEDS:**
- CRANKSHAFT - 750 TO 4500 RPM
 - ACCESSORY SHAFT - 1750 RPM CONSTANT FROM 875 TO 3500 ENGINE RPM
 - FLYWHEEL - NOMINAL 21,200 RPM
MAXIMUM 27,300 RPM
MINIMUM 9,000 RPM AT CLUTCH DISENGAGEMENT
- FLYWHEEL:**
- NOMINAL CAPACITY - 120 BTU (93,400 FT/LB)
 - SIZE - 9.0 INCH DIA X 0.96 INCH THICK
 - WEIGHT - 17.4 LB (STEEL)
 - AVAILABLE ACCESSORY ENERGY - 98 BTU MAX
 - MAXIMUM DELIVERY POWER FOR A 16 SEC VEHICLE ACCELERATION - 7.1 HP INCLUDING TRANSMISSION EFFICIENCY

OFF-PEAK FLYWHEEL - CONSTANT SPEED CONCEPT

FIGURE 4-35

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DESIGN DATA

- SPEED RATIOS:**
- CRANK TO ACCESSORY SHAFT 1.73:1 TO 0.577:1 VARIABLE (3:1 OVERALL)
 - ACCESSORY SHAFT TO FLYWHEEL - 10:1 (FLYWHEEL CHARGING MODE)
 - FLYWHEEL TO CRANK - 5.5:1 (FLYWHEEL DISCHARGE MODE)
- SPEEDS:**
- CRANKSHAFT - 750 TO 4500 RPM
1600 TO 4700 RPM FLYWHEEL DRIVEN
 - ACCESSORY SHAFT - 2080 RPM CONSTANT FROM 1200 TO 3600 ENGINE RPM
 - FLYWHEEL - NOMINAL 20,800 RPM
MAXIMUM 26,000 RPM
MINIMUM 8800 RPM AT CLUTCH DISENGAGEMENT
- FLYWHEEL:**
- NOMINAL CAPACITY - 120 BTU (93,400 FT/LB)
 - SIZE - 9.0 INCH DIA X 1.0 INCH THICK
 - WEIGHT - 18.0 LB (STEEL)
 - AVAILABLE ACCESSORY ENERGY - 98 BTU MAX
 - MAXIMUM DELIVERY POWER FOR A 16 SEC VEHICLE ACCELERATION - 6.8 HP INCLUDING TRANSMISSION EFFICIENCY

OFF-PEAK FLYWHEEL CLUTCH CONCEPT

FIGURE 4-36

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- (c) An acceleration signal supplied by manifold vacuum or speed integration devices couples the flywheel to the accessories and effectively unloads the engine reducing peak power demands, conserving fuel.

The flywheels are sized based on an overall collection and storage efficiency of approximately 60 percent with 20 and 50 percent accessory augmentation capability for the EPA city cycle, A/C on and off respectively. Braking energy required during the EPA highway cycle is reduced by a factor of four; therefore, flywheel effectiveness is greatly reduced.

A Rankine bottoming cycle accessory augmentation concept is shown in Figure 4-37. This design assumes a high speed (100,000 rpm) turbine expander feeds power to the accessories through a novel epicyclic gear train that is capable of efficient torque transmission and speed reduction of 50:1 and greater. A helical screw or vane expander running at 10,000 to 20,000 rpm could also be used which would simplify the transmission. A helical screw expander with variable capacity control would also obviate the need for a variable displacement boiler feed pump in the Rankine cycle system.

The Rankine bottoming system will produce accessory augmentation power in the 700 to 2600 watts (1 to 3 1/2 horsepower) range, but unlike the intermittent flywheel operation concepts, this power is continuous and augments accessory requirements by 20 to 75 percent, depending on driving cycle and accessory loading. This type system enhances vehicle performance and fuel economy and will not produce undesired driveability effects. The simulation model was sized based on an overall exhaust energy conversion efficiency of 8 1/2 percent. This efficiency is somewhat lower than could be expected from close tolerance, well designed systems, but is representative of mass produced hardware. A development, prototype Rankine bottoming system could be designed with an overall efficiency in the range of 10 to 14 percent.

RANKINE BOTTOMING CYCLE
ACCESSORY AUGMENTATION CONCEPT

FIGURE 4-37

4.4.2 Advanced Air Conditioning Trade-Study

The advanced air conditioning trade-study data summary is presented in Table 4-VIII and the elimination/selection flow chart is shown in Figure 4-38. The last line in Table 4-VIII describes the weighting factors used to comparatively evaluate each system. Fuel economy improvement potential is of prime importance with cooling performance a strong second, and initial cost third. Weighting factor significance in this configuration, is that they could only be found in a fuel economy study. Manufacturers of automotive air conditioning systems have classically been chiefly concerned with manufacturing cost and performance. Some auto manufacturers and suppliers are currently evaluating and/or developing advanced concepts with little or no intent of improving power consumption beyond the 2 1/2 horsepower/ton rating of yesterdays designs.

The cooling performance ratings assigned in Table 4-VIII require some explanation. All candidate systems could be designed to provide adequate cooling comfort for most steady-state conditions but rapid cool down performance is also critically important to the consumer. Meeting cool down requirements and demands imposed during hot day, city, stop and go driving forces the manufacturer to oversize heat exchangers, circulation fans and compressors, resulting in a heavy, over-capacity system for steady state operation. The throttling techniques currently employed to control capacity are inherently inefficient processes that directly effect cooling without controlling power requirements. This mixed bag of performance and cost mandates dictate the design of inefficient systems. By adding a new, critical mandate to improve the fuel economy while maintaining customer satisfaction in the performance category, the cost must increase, but be controlled to achieve reasonable pay-back through fuel conservation.

TABLE 4-VIII

ADVANCED AIR CONDITIONING
TRADE STUDY SUMMARY CHART

Type	Concept ①	City & Highway Fuel Economy Improvement Potential (%) ②	Overall COF Range (Steady State)	Relative Factors							
				Cooling Performance	Initial Cost ④	Adaptability	Complexity	Emissions ③	Weight	Volume	Development Potential
Engine Driven	Constant Speed Compressor	0	1.5 - 2.2	Good	Low	Good	Low	Freon	Moderate	Low	Developed
	Variable Delivery Compressor	1.5 - 2.5	2.5 - 3.5	Excellent	Moderate	Good	Moderate	Freon	Moderate	Moderate	Excellent
Air Cycle	Vacuum Cycle	Loss	1.0 - 1.8	Good	High	Moderate	Moderate	None	Moderate	Moderate	Good
	Regenerative Bootstrap	Loss	0.8 - 1.6	Good	High	Good	Moderate	None	Moderate	Moderate	Good
	"ROVAC"	Loss	0.5 - 1.5	Excellent	Low	Good	Low	None	Moderate	Low	Excellent
Waste Heat Rankine	Vapor Jet	6.5 - 5.5	0.2 - 0.3	Poor	Moderate	Poor	Moderate	Freon	Moderate	Moderate	Fair
	Compressor/Expander	7.0 - 6.3	0.4 - 0.5	Fair	High	Good	High	Freon	Moderate	Moderate	Good
Rate Heat Absorp- tion	Metal Hydrides	5.5 - 4.5	0.1 - 0.2	Poor	High	Poor	High	Unknown	High	High	Fair
	Chemical	4.5 - 4.0	0.2 - 0.3	Poor	High	Poor	Moderate	Toxic	High	High	Poor
	Molecular Sieve	3.5 - 3.5	0.1 - 0.2	Poor	High	Poor	High	Toxic	High	High	Poor
	Trade Study Weighting	30%	-	20%	15%	5%	5%	5%	5%	5%	5%

NOTES: 1 Vehicle configuration for all systems assumes a variable ratio belt accessory drive is installed.

2 Refers to potential available from Phase IV, MADS equipped Mustang II test vehicle with constant speed air conditioning compressor.

3 Assumes system leaks or fails.

4 Assumes OEM installation of developed system.

ADVANCED AIR CONDITIONING SYSTEMS TRADE STUDY

FIGURE 4-38

The constant speed "Freon" compressor concept, currently installed in the Mustang II test vehicle was used for the trade-study as a baseline for comparison purposes. Therefore, Table VIII shows a 0 percent fuel economy improvement potential for this concept. All air cycle concepts evaluated required engine power in excess of that required to drive Freon compressors to meet the same cooling demands (lower COPs); therefore, fuel economy would not be improved. The engine waste heat Rankine and absorption systems all have the potential to improve fuel economy but most perform poorly. The absorption systems have the lowest overall rating and were eliminated as poor development risks with no near-term market potential. A new evaluation of these systems should be entertained when technology advances. Both Rankine systems, the vapor jet and compressor/expander concepts, were considered strong candidates though the cool-down performance rating is marginal due to engine warm-up lag. The vapor jet concept is principally simple but requires critical engine cooling system redesign and development, while the compressor/expander is an add-on concept that relies only on exhaust heat for a driving source. Also the operating characteristics of a vapor jet using a phase change refrigerant, such as "Freon", are still experimental. The compressor/expander concept has good development potential because the technology exists and has been successfully applied to other waste heat systems.

The engine driven, variable delivery, "Freon" compressor ranked good to excellent in all trade-study categories. The fuel economy improvement potential is lower than the waste heat systems because the engine must still supply the driving power, but the performance rating is the highest overall. The technology required to develop this concept is currently available and consumer acceptance should be good due to the performance-cost factors.

4.4.2.1 Advanced Air Conditioning Design Concepts

A waste heat, Rankine turbo-compressor air conditioning design concept is shown in Figure 4-39. The turbo-compressor unit for the concept is small and light weight with 38 mm diameter rotors designed to operate at 100,000 rpm. The vapor boiler system includes an exhaust bypass control valve to regulate turbine inlet conditions and eliminate the requirements for a variable delivery feed pump. The boiler and heat storage device are located downstream of a well insulated converter to take advantages of additional heat from the catalytic process. The heat storage capability influences idle performance in stop and go city driving. The estimated weight of the boiler, valve and storage module is 14 kg (30 lbs). The condenser and evaporator are standard construction with a combined weight of approximately 44 kg (97 lbs). The boiler feed pump operates at approximately 285 psig and pumps 6 lb/min liquid Freon (R-113) to consume approximately 0.17 horsepower. The pump may be designed as a centrifugal unit and directly-driven from the turbo-compressor shaft. The recuperator is a small efficiency boosting heat exchanger that weighs only 2.7 kg (6 lbs). The total system weight, excluding fans, is 68 kg (150 lbs) which is comparable to existing air conditioning systems.

The design is a unitary fluid system that develops the motive force and cooling capacity from the same fluid. The cooling capacity design point is 18,000 BTU//HR at 40°F, evaporator temperature and 130°F condensing temperature. The required boiler heat load to meet the 1 1/2 ton cooling requirement is 600 BTU/min. The EPA city cycle average exhaust heat energy is 550 BTU/min; therefore, a steady-state delivery of one ton refrigeration is possible and should be adequate. The EPA highway cycle average exhaust heat energy is over 1100 BTU/min which is more than adequate for developing maximum cooling capacity. The idle exhaust heat flow is only 200 BTU/min which would only support about 3000 BTU/hr of cooling at an evaporator temperature of 40°F. Obviously, the idle cooling capacity will not support cool down

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WASTE HEAT, RANKINE TURBO-COMPRESSOR AIR CONDITIONING SYSTEM

FIGURE 4-39

requirements; therefore, the vehicle would have to be driven to achieve reasonable cool down performance. Further detail design concept refinement could possibly result in selecting or developing a more ideal refrigerant that could increase performance and decrease size.

Three variable delivery engine driven "Freon" air conditioning compressor concepts were evaluated for performance and efficiency characteristics:

- o Variable displacement piston - Figure 4-40
- o Suction cut-off modulation - Figure 4-41
- o Helical screw variable delivery - Figure 4-42

All concepts are capable of effective demand modulation to minimize or eliminate the throttling losses associated with fixed geometry compressors. The three concepts are also driven at or near constant speed to produce the best possible operational environment.

The concepts design point operating efficiencies are nearly equal but the helical screw machine performs at an optimum speed that is twice the piston unit speed. The high speed capability results in a smaller and lighter compressor.

Suction cut-off modulation is the most efficient compressor power control technique but the poorest method of meeting system load requirements. Effective system load matching, with the suction cut-off method, would require a multi-cylinder compressor with at least three displacement ranges. Balancing the compressor cranking mechanism might require unloading the cylinders in pairs, again increasing the total number of cylinders.

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VARIABLE DISPLACEMENT COMPRESSOR CONCEPT

FIGURE 4-40

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SUCTION CUT-OFF CAPACITY MODULATION CONCEPT

FIGURE 4-41

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HELICAL SCREW-VARIABLE DELIVERY CONCEPT

FIGURE 4-42

The variable displacement piston compressor loses volumetric efficiency when operating at part load because the percent effective clearance volume increases as displacement decreases. As pressure ratio changes, the degree of modulation changes for any fixed delivery requirement resulting in a control shift. The control system shown in Figure 4-40 should preclude float or oscillations in the low to mid-pressure ratio range but may require special development attention in the higher pressure ratio ranges. Lubrication for the axial piston swash-plate design is extremely critical. The hydrostatic valve plate and piston slipper shoes operate under adverse load conditions and require hard well lubricated bearing surfaces. Preliminary design of this concept was completed to assist in determining detail performance and physical factors. The design incorporates a rotating cylinder block with seven, 26 mm diameter pistons with a maximum stroke of 36 mm for a maximum displacement of 130 cc/rev ($7.94 \text{ in}^3/\text{rev}$). The stroking member is a swash plate with the hinge point selected to maintain minimum clearance volume. The design displacement range of 50 to 130 cc/rev (3.05 to $7.94 \text{ in}^3/\text{rev}$) will provide full capacity modulation over a pressure ratio of 3.5 to 8 representing 70 to 100°F day operation. Volumetric efficiency approaches zero at minimum displacement and pressure ratios above 8 but this condition is not expected to occur because cooling demand on very hot days will always dictate high displacement operation due to vehicle cab heating associated with insulation properties and glass area. The average COP for this machine rated at 48 km/hr (30 mph) steady-state suburban driving on a 90 to 95°F day is approximately 2.45 representing a 35 percent improvement over the MADS and stock configurations and a 634 watt (0.85 horsepower) savings. Energy conservation over the idle to 89 km/hr (55 mph) range is improved by 24 to 68 percent. Operating power requirements continue to improve as the cooling load decreases because the compressor concept is capable of efficiently matching demands while the stock configuration relies on throttling the output of an oversize compressor.

All improved operating characteristics attributed to the variable displacement piston compressor also apply to the variable delivery helical screw machine. The helical screw compressor is smaller and lighter requiring only 51 cc/rev ($3.11 \text{ in}^3/\text{rev}$) maximum displacement to meet all load demands. The variable inlet slide valve concept does not effect volumetric efficiency or require special materials and lubrication. Concept load control should be simple, direct and stable. The concept has been developed by Dunham-Bush and is used in commercial applications where performance and reliability have proved to be unquestionably high. The helical screw compressor concept offers additional, unique, performance capabilities that cannot be obtained with a piston compressor. Compressor "slugging," or liquid ingestion, is a serious problem for piston and vane compressors and has caused massive failures. The helical screw compressor is far less sensitive to slugging and can be operated with less superheat resulting in improved performance. The most advantageous aspect of the helical screw concept is that mid-range pressure, evaporator outlet gas may be injected into the screw anywhere along the length (visualized as stroke), thus supercharging the compressor and increasing the mass flow during high load demand periods, such as vehicle cool-down. This factor allows design of a smaller, compact unit with additional capacity only when needed.

4.4.3 Variable Delivery - Helical Screw Compressor Study

Dunham-Bush, a subsidiary of Signal Corporation, completed the advanced air conditioning study consisting of multiple tasks to determine applicability of various compressor concepts and system configurations to meet automotive requirements and improve fuel economy. The study provided the indepth technical analysis required to complete the trade-study task.

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The basic variable delivery, helical screw compressor (VHS) discussed in the trade-study, was sized to meet the cooling requirements shown in Figure 4-43 while operating over the speed profile dictated by the Phase IV developed variable ratio belt (VRB) accessory drive. The compressor is driven from the output of the VRB at a fixed ratio of 1.54 increasing the speed range envelope and resulting in a maximum displacement requirement of 51 cc/rev ($3.11 \text{ in}^3/\text{rev}$). The hot day, steady-state, maximum displacement design performance data is presented in Figure 4-44. The speed, hp/ton and volumetric efficiency represents an optimized compressor that is capable of providing more than adequate cooling with significantly improved efficiency. Figure 4-45 shows the VHS part load or modulation characteristic covering a 70 to 100°F day operating range at nearly one-to-one power vs capacity. A single point analysis representing capacity modulation capability at a vehicle speed of 64 km/hr (40 mph) is presented in Figure 4-46, also showing stock A/C performance for comparison. If the data for the stock and VHS compressor is extrapolated to meet a 2-ton refrigeration requirement, both units will perform at approximately two hp/ton. This particular single point analysis was selected to show performance at the optimum design speed for the stock compressor, thereby eliminating VRB effects and demonstrate only variable delivery advantages. The VHS compressor performs 29 to 86 percent (COP) better than the stock machine over the required cooling demand range, resulting in a power conservation of 22 to 46 percent. As the stock, fixed ratio and displacement compressor speed increases, excess capacity is generated at a rate of 700 BTU/hr for every 100 rpm at a power penalty of 186 watts (0.25 hp) per increment.

The engine idle, VHS cool down capacity also exceeds the stock compressor capability by approximately 30 percent at 130°F condensing and 35°F evaporator temperature. In this operating condition the VHS compressor also consumes more power than the stock unit. If output capacities are equated, the increased VHS power is 373 watts (0.5 hp). Power added to the accessory load at engine idle has negligible effects

COOLING DEMAND VS VEHICLE SPEED

FIGURE 4-43

VARIABLE DELIVERY - HELICAL SCREW
PERFORMANCE

FIGURE 4-44

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VARIABLE DELIVERY - HELICAL SCREW
PART LOAD PERFORMANCE

FIGURE 4-45

STOCK AND VARIABLE DELIVERY
PART LOAD PERFORMANCE COMPARISON

FIGURE 4-46

on overall fuel economy. Since idle fuel flow is a strong function of engine speed, the fuel flow will not be effected if the engine can support a slight load increase without modification to the idle set point.

The VHS sizing analysis required a 35°F evaporating temperature while matching the maximum load demand. A more realistic and still adequate evaporating temperature of 45°F at maximum steady-state load would result in a smaller displacement unit with the same performance and capacity except for cool-down conditions. Improved cool-down performance for the smaller design could be achieved with the supercharging technique discussed earlier.

The final air conditioning system study effort evaluated the possibilities of incorporating reverse cycle performance to use the VHS as a heat pump. A heat pump would eliminate long warm-up and defrost idle periods in cold winter climates and provide more customer satisfaction. The electric driven reverse cycle air conditioning system is also the only answer to all electric vehicle environmental control. Figures 4-47, 4-48, and 4-49 show the reverse cycle components and schematic operating modes:

- o Figure 4-47 - VHS compressor modulating delivery to meet all steady-state cooling demands
- o Figure 4-48 - VHS compressor at maximum displacement with supercharging to meet pull down capacity requirements
- o Figure 4-49 - VHS compressor with evaporating and condensing cycles reversed to provide cab heating and defrosting

VARIABLE DELIVERY - A/C AND HEAT PUMP SYSTEM -
STEADY STATE COOLING

FIGURE 4-47

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VARIABLE DELIVERY - A/C AND HEAT PUMP SYSTEM -
PULL DOWN COOLING

FIGURE 4-48

VARIABLE DELIVERY - A/C AND HEAT PUMP SYSTEM -
HEATING CYCLE

FIGURE 4-49

4.4.4 Additional Accessory Optimizations

The constant speed accessory drive concept applied to the Mustang II accessory system has demonstrated improved performance, but the potential of improved accessory efficiency has yet to be realized. All power consuming and producing devices operate at a peak efficiency for a particular load and speed range condition. The efficiency benefits that can be achieved from load and speed control have been discussed for variable delivery air conditioning compressors. The purpose of this section is to evaluate efficiency improvement potential that may be obtained by redesigning the alternator, water pump and power steering system for constant speed operation.

4.4.4.1 Alternator Optimization

A typical automotive type alternator consists of three armature windings wound on a laminated stator core and a field bobbin wound around the rotor. Current is furnished, via brushes, to the rotating field coil from the voltage regulator. This current in the field coil creates a flux field that flows axially through the center of the field coil, radially outward through the rotor north pole end plate, across the air gaps of the six north poles, past the armature windings, circumferentially around the armature core to adjacent south poles, and radially inward through the rotor. This results in a time varying field in the armature which develops a voltage in the armature winding. The magnitude of the voltage is proportional to the speed of the rotor and the current in the field.

As shown on Figure 4-50, the alternator output is full wave rectified by six diodes and the output voltage is regulated by the voltage regulator that controls the field current. Output voltage is nominally controlled within the range of 13.4 to 14.6 volts over a temperature range of -18° to $+16^{\circ}\text{C}$ (0° to 140°F). Alternators are typically rated at 35 to 65 amps with full field excitation and at the

DC SYSTEM
FIGURE 4-50

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mid-speed range. Additional load results in reduced voltage which, in turn, causes battery drain.

Developing power over a wide speed range results in an oversized machine at high speeds. Although some improvement in efficiency can be obtained by increasing the core size, adding copper in the field and armature, and optimizing the air gap, friction and windage losses would be higher than for a machine designed for a fixed speed.

Most automotive alternators are designed with an efficiency peak of approximately 60 percent that occurs within the engine idle speed range where electrical output is critical to prevent excess battery drain. The efficiency rapidly declines to an average value of 45 percent as the alternator speed is increased. The above efficiencies also represent rated conditions, therefore, part load efficiencies may be lower.

Incorporation of a constant or near constant speed accessory drive, as used in the Mustang II test vehicle, eliminates the low speed output problem by increasing the idle speed by 80 percent more than is possible with the stock system while limiting the maximum speed by 40 percent. Applying the accessory drive in the Mustang II permitted the use of a smaller alternator with improved performance at idle and adequate performance at highway speeds. The automotive alternator can be redesigned for maximum efficiency of 75 to 80 percent at constant speed with an overall operating efficiency of 60 percent representing a nominal input power reduction of 25 percent. The fuel economy improvement potential for the redesigned alternator is as high as 1.2 percent for the EPA highway driving cycle with air conditioning on.

4.4.4.2 Water Pump Optimization

The automotive coolant pump is one of the most highly production-ized items on todays I.C. engines. The construction technique utilizes a stamped sheet metal or cast iron centrifugal impeller positioned in a rough sand cast housing by a set of double row bearings with spring loaded seals. Engineering of these pumps has been almost totally concerned with life and cost. Engine cooling system performance is approximately one-half as dependent on pump fluid flow as on radiator airflow; therefore, pump performance demands are not critical. If the pump provides sufficient engine idle flow and does not begin to cavitate prior to an equivalent vehicle speed of approximately 105 km/hr (65 mph) it is considered satisfactory. Any effort to increase the water pump efficiency will result in increased cost, but the increase may be recoverable based on fuel savings and high production amortization.

A cross section view of a typical automotive water pump with loss areas defined is shown in Figure 4-51. Taking advantage of the constant speed concept, the pump may be redesigned to include a speed-flow matched, log spiral impeller, reduced housing clearances and optimum area ratio diffusion chambers.

Efficiencies common to the stock water pump are approximately 10 to 20 percent overall. The ideal, peak efficiency for a speed, flow and pressure matched design is 65 to 70 percent. This study proposes the attainable goal of doubling the stock pump efficiency thus reducing the input power requirement by 50 percent. Relating these improvements to the Mustang II test vehicle with constant speed accessory drive, the resulting improvement in fuel economy is 0.5 to 0.7 percent.

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AUTOMOTIVE WATER PUMP LOSSES

FIGURE 4-51

4.4.4.3 Power Steering System Optimization

Power steering pumps are required to deliver approximately 7.6 min (2 gpm) at 43 to 57 bar (600 to 800 psig) near engine idle speed to assist steering for parking maneuvers. If idle performance requirements are met, the pump is capable of producing up to four or five times as much flow at 86 km/hr (55 mph) where the actual requirement is less than at idle. The system contains a flow modulating bypass valve that prevents steering assist rate increases that may exceed the drivers control capability. The bypass valve allows the total pump capacity to be raised to the demand pressure level then dissipates the excess to meet steering system requirements. The net effect is a power input that is four or five times above the load requirement with fluid heating necessitating a small heat exchanger.

A power steering pump driven at near constant speed could be efficiency optimized over a narrow flow range to conserve power, eliminate the heat exchanger and very possibly eliminate the flow control valve. The net result is a lower cost power steering system with 1.2 to 1.4 percent improvement in fuel economy.

4.4.5 System Integration Trade-Studies

The hybrid drive, advanced air-conditioning and accessory optimization trade-studies performed at the component level produced candidate concepts for system integration trade-studies. The task in this report section is to integrate the components into a total accessory system configuration and evaluate feasibility and performance. The component trade-studies selected the following concepts:

Hybrid Accessory Drives

- o Off-peak regenerative braking flywheel drive
- o Exhaust heat, Rankine bottoming accessory drive

Advanced Air Conditioning Concepts

- o Exhaust waste heat, turbo-compressor air conditioning system
- o Variable delivery, helical screw air conditioning compressor

Optimized Accessories

- o Alternator
- o Water pump
- o Power Steering system

All advanced system configurations and analyses include the optimized accessories. Accessory optimization techniques are dependent on consistent speed operation and all advanced systems include this concept.

The two hybrid drive and advanced airconditioning concepts yield a possible four way combination, but use of waste heat (exhaust) for more than one concept at a time is not feasible because the magnitude of the energy source is not sufficient to support the combined load. The feasible combinations are reduced to three:

- o Off-peak flywheel with variable delivery A/C
- o Rankine bottoming drive with variable delivery A/C
- o Off-peak flywheel with waste heat A/C

A multiple concept incorporating off-peak flywheel, Rankine bottoming drive and variable delivery air conditioning is possible, but was not considered due to excessive cost, weight and complexity.

A demonstration of goals and potentials should be made prior to detailed evaluation of the three candidate accessory systems. The maximum, theoretical fuel economy improvement that could be obtained through decreasing the accessory loads on the engine is realized when

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the load is zero. Figure 4-52 shows that the EPA composite driving cycle (45 percent city, 55 percent highway, 50 percent A/C) fuel economy improvement potential for the zero accessory load condition is 15.2 percent. The Phase IV, constant speed, Mechanical Accessory Drive System (MADS) proved the importance of speed control and yielded a 4.8 percent composite fuel economy improvement, representing a 30 percent achievement of the maximum theoretical goal. A first order approximation of attainable improvement resulting from incorporating the candidate systems is 7 to 11 percent representing a 46 to 72 percent achievement. In terms of miles-per-gallon, the stock vehicle composite fuel economy is 18.03 mpg and an 11 percent increase would yield 20.01 mpg

Each of the three hybrid drive, advanced air conditioning and accessory system concepts were fuel economy and performance evaluated using the parametric capabilities of the Mustang II simulator. The simulation detail included assumptions for energy conversion and component efficiencies, accessory and engine speed and load matching characteristics, and approximate accessory system weight replacement penalties. System schematics, descriptions, and fuel economy improvement potential for each concept are shown in Figures 4-53, 4-54, and 4-55.

The off-peak flywheel with variable delivery air conditioning concept demonstrated the lowest fuel economy improvement (7.3 percent composite), but maintains credibility by being the lightest, least complex, and least expensive system. The systems involving exhaust heat regeneration for either air conditioning or accessory drive assistance demonstrated composite fuel economy improvements of slightly over 11 percent. Table 4-IX summarizes the accessory system integration performance and relative factors for each concept.

Fuel economy improvement, performance and development potential dictate the selection of the Rankine bottoming accessory assist drive with a helical, twin screw, variable delivery airconditioning system

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	FUEL ECONOMY IMPROVEMENT OVER STOCK (PERCENT)			
	CITY CYCLE		HIGHWAY CYCLE	
	A/C ON	A/C OFF	A/C ON	A/C OFF
MAXIMUM THEORETICAL IMPROVEMENT ASSUMES VEHICLE WITH NO ACCESSORIES	18.4	6.9	23.6	12.6
	COMPOSITE 15.2			
M.A.D.S. IMPROVEMENT ASSUMES PHASE IV VEHICLE WITH CONSTANT SPEED ACCESSORY DRIVE	3.6	1.0	7.8	5.2
	COMPOSITE 4.8			
MAXIMUM ATTAINABLE IMPROVEMENT ASSUMES HYBRID DRIVE, ADVANCED A/C AND IMPROVED ACCESSORIES	6-15	4-5	10-18	7-12
	COMPOSITE 7-11			

FUEL ECONOMY IMPROVEMENT POTENTIAL

FIGURE 4-52

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FUEL ECONOMY IMPROVEMENT POTENTIAL		
	CITY CYCLE	HIGHWAY CYCLE
A/C ON	5.9%	10.1%
A/C OFF	4.4%	7.2%
EPA COMPOSITE (50% A/C)	7.3%	

SYSTEM DESCRIPTION

- OFF-PEAK REGENERATIVE FLYWHEEL WITH 93,400 FT-LB STORAGE CAPACITY AND OVERALL EFFICIENCY OF 50% DRIVING THROUGH A VRB.
- VARIABLE DELIVERY SCREW COMPRESSOR A/C SYSTEM
- EFFICIENCY IMPROVED WATER PUMP AND ALTERNATOR WITH FLOW/SPEED MATCHED POWER STEERING PUMP.

OFF-PEAK FLYWHEEL AND VARIABLE DELIVERY
AIR CONDITIONING

FIGURE 4-53

SYSTEM DESCRIPTION

- RANKINE BOTTOMING CYCLE TURBINE AUGMENTING ACCESSORY POWER REQUIREMENTS THROUGH SPEED ADD PLANETARY GEARCASE. INCLUDES THERMAL STORAGE.
- VARIABLE DELIVERY SCREW COMPRESSOR A/C SYSTEM
- EFFICIENCY IMPROVED WATER PUMP AND ALTERNATOR WITH FLOW SPEED MATCHED POWER STEERING PUMP.

FIGURE 4-54

RANKINE BOTTOMING CYCLE AND VARIABLE DELIVERY AIR CONDITIONING

FUEL ECONOMY IMPROVEMENT POTENTIAL		
	CITY CYCLE	HIGHWAY CYCLE
A/C ON	15.4%	17.4%
A/C OFF	4.4%	7.2%
EPA COMPOSITE (50% A/C)	11.1%	

SYSTEM DESCRIPTION

- OFF-PEAK REGENERATIVE FLYWHEEL WITH 93,400 FT-LB STORAGE CAPACITY AND OVERALL EFFICIENCY OF 50% DRIVING THROUGH A VRB.
- TURBO-COMPRESSOR WASTE HEAT A/C SYSTEM
- EFFICIENCY IMPROVED WATER PUMP AND ALTERNATOR WITH FLOW/SPEED MATCHED POWER STEERING PUMP.

OFF-PEAK FLYWHEEL AND WASTE HEAT
AIR CONDITIONING

FIGURE 4-55

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TABLE 4-IX

ACCESSORY SYSTEM INTERGRATION
TRADE-STUDY SUMMARY CHART

System Concept ¹	EPA Composite Fuel Economy Improvement (%)	Weight Replacement Penalty (lbs)	Approx Added Volume (ft ³)	Relative Factors					
				Initial Cost	Complexity	Maintenance	Overall Performance	Development Potential ²	Development Improvement Potential ³
Off-peak flywheel with variable delivery A/C	7.3	75	1.0	Mod	Mod	Mod	Good	Fair	Good
Rankine bottoming drive with variable delivery A/C	11.3	150	2.0	High	Mod	Low	Excellent	Good	Excellent
Off-peak flywheel with waste heat A/C	11.1	140	1.5	High	High	Mod	Fair	Fair	Poor

NOTES:

- ¹ All systems include constant speed principle.
- ² Performance includes A/C cooling comfort factor.
- ³ Reflects trade-study assumptions and continuing state-of-the-art advances.

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as the prime system concept. The decision to utilize exhaust waste heat energy source in a constant duty accessory drive realizes the maximum source potential. Accessory drive assistance will always be available regardless of accessory array loading. Vehicles without air conditioning and power steering will also benefit from the incorporation of a Rankine bottoming system.

The waste heat powered air conditioning concept did not have good performance ratings because the 2800 cc displacement, Mustang II engine, exhaust mass flow is not sufficient at low speed driving conditions. A re-evaluation of this concept for turbine powered vehicles would show improvement in concept feasibility.

Regenerative braking studies have revealed that the recoverable magnitude of available energy was lower than expected over the EPA driving cycle. Incorporation of this concept in vehicles such as; city taxis, buses, commuter trains, etc., where the accel/decel schedules are more rigorous and the potential energy is increased due to higher vehicle weight should be further evaluated. Ultimate use of regenerative braking energy for vehicle, prime motive force may prove to be a more suitable application for improving fuel economy if the stored energy can be used at a very high rate to provide initial vehicle acceleration requirements; thus reducing engine peak power requirements in an inefficient operating range.

Developing the helical, twin screw, variable delivery air conditioning compressor that can efficiently operate at high speed with cool-down gas injection and reverse cycle capability represents a near ultimate configuration incorporating state-of-the-art techniques for maximizing performance and efficiency. This concept is certainly not restricted to automotive use. Large tonage, commercial application of this concept utilizing solar powered gas injection augmentation has already proven highly reliable and efficient. Home and automotive applications can readily be developed to take advantage of the energy conservation and performance flexibility offered by this concept.