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ELEVENTH QUARTERLY
PROGRESS REPORT

STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

74-310860(33)

April 30, 1977

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
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ELEVENTH QUARTERLY
PROGRESS REPORT
STUDY ON REDUCTION OF ACCESSORY
HORSEPOWER REQUIREMENTS

1.0 INTRODUCTION

This is the eleventh quarterly technical progress report submitted in accordance with requirements outlined in Attachment B, Reports and Reviews, Energy Research and Development Administration Division of Transportation Energy Conversion (TEC) Office of Highway Vehicles, Vehicular Systems Branch Contract EY-76-C-03-1095. This report covers the period from February 1, 1977 through April 30, 1977.

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 will evaluate hybrid systems utilizing alternate power sources, energy storage techniques and advanced air conditioning concepts. The study intent is to further emphasize fuel economy improvements that may be gained through accessory system optimization with respect to the vehicle and operating cycle. Untapped energy sources, such as engine waste heat and deceleration momentum, will be defined. Concepts to store and transport available power into the accessory drive system will be evaluated. Energy conserving air conditioning concepts will be evaluated to provide alternatives to conventional systems and further reduce accessory power requirements.

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Air conditioning optimization and hybrid accessory drive study results will be applied to the 1975 Mustang II baseline vehicle, used in program Phases I through IV. This allows improvements to be projected from the baseline performance and analytical data already developed.

1.1 Contents

Major technical accomplishments during this reporting period were:

- o All candidate advanced air conditioning concepts were evaluated
- o Advanced air conditioning and hybrid accessory drive component trade-studies were completed
- o Improved alternator, water pump and power steering system concepts were evaluated
- o The vehicle integrated accessory systems trade-study was completed
- o The technical summary report for the Phase V Automotive Accessory Systems Optimization Program was initiated.

2.0 ADVANCED AIR CONDITIONING ANALYSIS

Feasible applications for advanced air conditioning, including preliminary technical analyses, were compiled during this reporting period. The 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.

2.1 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 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.

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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).

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).

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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 1):

- (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.
- (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

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BASIC AIR CONDITIONING SCHEMATIC

FIGURE 1

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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 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 valve in any automotive system is never permitted.

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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.

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

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

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2.2 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.

2.2.1 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."

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

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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.

2.2.1.1 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.

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

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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 I. It can be seen that SO_2 has a 20 and 47 percent system weight advantage over R-114 and R-113 respectively.

2.2.2 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 2. 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 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

TABLE I
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
Total Heat Exchanger Weight, lb (New Designs)	49	59	72

SCHEMATIC OF VAPOR JET AUTOMOTIVE REFRIGERATION

FIGURE 2

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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 pump is the only power consumer. Pump 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 compensated by eliminating the conventional radiator and heater.

Items requiring detailed investigation with this system are:

- o Possible use of a water coolant/Freon boiler system.
- o Sealing of engine block and head with refrigerant compound coolant.
- o Control of engine heat.
- o Strength of block and head with 160 psi operating pressures, and jacket passage geometry for optimum heat rejection.
- o Investigation of the vapor jet operating range with known motive pressure and flow vs. predicted recovery pressures.

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The latter item is critical since 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.

2.2.3 Waste Heat Absorption Air Conditioning

The most common heat driven air conditioning devices are absorption 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 3.

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 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

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AUTOMOTIVE ABSORPTION AIR CONDITIONING

FIGURE 3

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

FIGURE 4

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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.

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.

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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.

2.2.4 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 5. 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 6, 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

COMBINED CYCLE SCHEMATIC DIAGRAM

FIGURE 5

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ROTARY BED DESIGN SCHEMATIC

FIGURE 6

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

2.3 Air Cycle Air Conditioning Systems

2.3.1 Vane Compressor

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

Figure 7 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.

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BASIC ROVAC SYSTEM

FIGURE 7

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 that they can slide inside the rotor slots as the rotor-vane assembly 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 II 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:

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TABLE II
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

Adiabatic Compressor	111 percent
Adiabatic Expander	80 percent
Frictional	60 percent

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, with the ROVAC equipment as shown in Figure 7. 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.

2.3.2 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 III.

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,

TABLE III

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

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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 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 30 to 50 percent 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.

2.4 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.

2.4.1 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.

2.4.2 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 Section 3.

3.0 TRADE STUDIES

The trade studies purpose is 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 is 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 is 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.

3.1 Hybrid Accessory Drive Trade-Study

The performance and physical factors for each candidate hybrid drive are summarized in Table IV. The optimization flow chart, based on Table IV data is presented in Figure 8. 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 IV
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	0.03	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 8

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.

3.1.1 Hybrid Accessory drive Design Models

Two off-peak flywheel, hybrid accessory drive, design concepts are presented in Figures 9 and 10. 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 9 uses a second variable ratio belt transmission to feed and control braking energy into the flywheel at high efficiency, while the Figure 10 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 9

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 10

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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 11. 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 11

3.2 Advanced Air Conditioning Trade-Study

The advanced air conditioning trade-study data summary is presented in Table V and the elimination/selection flow chart is shown in Figure 12. The last line in Table II 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 II 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 V

ADVANCED AIR CONDITIONING
TRADE STUDY SUMMARY CHART

Type	Concept ①	City & Highway Fuel Economy Improvement Potential (%) ②	Overall CCP 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 Hydrids	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 12

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The constant speed "Freon" compressor concept, currently installed in the Mustang II test vehicle was used for the trade-study as a base-line for comparison purposes. Therefore, Table II 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.

3.2.1 Advanced Air Conditioning Design Concepts

A waste heat, Rankine turbo-compressor air conditioning design concept is shown in Figure 13. 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 13

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 14
- o Suction cut-off modulation - Figure 15
- o Helical screw variable delivery - Figure 16

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 14

SUCTION CUT-OFF CAPACITY MODULATION CONCEPT
FIGURE 15

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HELICAL SCREW-VARIABLE DELIVERY CONCEPT

FIGURE 16

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 14 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 of Hartford, Connecticut, 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.

3.2.2 Variable Delivery - Helical Screw Compressor Study

Dunham-Bush, subsidiary of Signal Corporation, in Hartford, Connecticut, has completed the advanced air conditioning study described in Problem Statement ADS030877-008PS which appeared in the February report. The study consisted 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.

The basic variable delivery, helical screw compressor (VHS) discussed in the trade-study, Section 3.2.1 was sized to meet the cooling requirements shown in Figure 17 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 in³/rev). The hot day, steady-state, maximum displacement design performance data is presented in Figure 18. 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 19 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 20, 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 17

VARIABLE DELIVERY - HELICAL SCREW
PERFORMANCE

FIGURE 18

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VARIABLE DELIVERY - HELICAL SCREW
PART LOAD PERFORMANCE

FIGURE 19

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STOCK AND VARIABLE DELIVERY
PART LOAD PERFORMANCE COMPARISON

FIGURE 20

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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 shown on Figure 10. 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 in

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. Figure 21, 22, and 23 show the reverse cycle components and schematic operating modes:

- o Figure 21 - VHS compressor modulating delivery to meet all steady-state cooling demands
- o Figure 22 - VHS compressor at maximum displacement with supercharging to meet pull down capacity requirements
- o Figure 23 - 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 21

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

FIGURE 22

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

FIGURE 23

3.3 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.

3.3.1 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 24, 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 24

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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 per cent. 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.

3.3.2 Water Pump Optimization

The automotive coolant pump is one of the most highly production-ized items on today's 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.

Figure 25 shows a cross section view of a typical automotive water pump with loss areas defined. 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 25

3.3.3 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.

3.4 System Integration Trade-Studies

The advanced accessories component trade-studies produced candidate design concepts based on individual evaluations and comparative analysis. The task in this report section is to integrate the components into a total accessory system concept and evaluate feasibility and system performance. The component trade-studies selected the following hybrid drive and advanced accessory concepts:

Hybrid Accessory Drives (3.1.1)

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

Advanced Air Conditioning Concepts (3.2.1)

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

Optimized Accessories (3.3)

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

Including optimized accessories in any system without regard for power source magnitude or system configuration is permissible; therefore, all systems considered will contain these components and benefit from improved performance and fuel economy.

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

the load is zero. Figure 26 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 27, 28, and 29.

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 VI 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 26

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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 27

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 28

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 29

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TABLE VI

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 ²	Development Potential	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 vari- able 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 tonnage, 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.