DESIGN AND DEVELOPMENT OF AN AUTOMOTIVE
PROPULSION SYSTEM UTILIZING A RANKINE CYCLE
ENGINE (WATER BASED FLUID)

FINAL REPORT

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1.0 INTRODUCTION

In fulfillment of the objectives of Reference 1, Scientific Energy Systems Corporation has devoted significant efforts to the design, fabrication and test of a low emissions Rankine cycle automotive steam engine. The program initially funded by private sources, was sponsored by a Government contract since 1971. The program was originally funded by the Environmental Protection Agency (EPA), Division of Motor Vehicle Research and Development, for its low emissions potential. In January 1975, the program sponsorship was shifted to the new Energy Research and Development Administration (ERDA).

Under EPA and ERDA sponsorship, SES successfully designed, fabricated and tested the first federally sponsored steam powered automobile. The automobile - referred to as the "simulator" - is a 1975 Dodge Monaco standard size passenger car with the SES preprototype Rankine cycle automotive propulsion system mounted in the engine compartment.

In the latter half of 1975, the simulator successfully underwent test operations at the facilities of SES in Watertown, Massachusetts and demonstrated emission levels below those of the stringent federally established automotive requirements originally set for implementation by 1976. The demonstration was accomplished during testing over the Federal Driving Cycle on a Clayton chassis dynamometer.

During the period of performance covered by this report, significant achievements were made by SES in the design, fabrication and test of subsystems such as the vapor generator, feedwater pump, controls, expander, lubrication and systems integration. While each detail of program performance is not included herein, the report reflects the detailed level of effort, and the evolution of design concepts during the pursuit of contract objectives.

Due to reductions in program funding and changes in program emphasis, one original program objective, that of providing design specifications for a prototype Rankine cycle automotive propulsion system, was not concluded. The prototype system specifications would have included a light weight expander, vapor generator, condenser and controls, all optimized based upon subsystem and system analysis and test results, and packaged to operate in the engine compartment of a production automobile. In addition, shifts in program emphasis reduced or terminated work in lubrication testing, particularly at temperatures in excess of 811°K (1000°F), friction and wear testing, and freeze protection.

Throughout the period of performance SES successfully sought, as program activities required, the specialized expertise of individuals and industry as consultants or subcontractors. Among the contributors were Bendix (preliminary controls specifications), American Motors Corporation (preliminary powerplant/Vehicle integration), AiResearch (preprototype condenser design and fabrication), Energy Technology Incorporated (feasibility study of a radial outflow turbine/gearbox alternate expander). The major subcontractors were:

Chrysler Corporation	-	Powerplant/Vehicle Integration, Vehicle Performance Analysis, Drive Train Selection, Cost Analysis and Vehicle Modification
Exxon Research and Engineering	-	Expander Lubricant Development
Ricardo Consulting Engineers, Ltd.	-	Expander Design and Manufacturing

The work reported herein was funded through the Division of Transportation Energy Conservation, Highway Vehicle System of the Office of the Assistant Administrator for Conservation of the Energy Research and Development Administration under their Contract E(11-1)2701. Mr. James Buchar was the Contracting Officer. Mr. Patrick Sutton was the Project Officer, and Mr. Roger L. Demler of Scientific Energy Systems Corporation was the Program Manager.

The material presented in this report is a condensation of information prepared by the engineering staff at SES for regular progress reports previously submitted to ERDA.

2.0 SUMMARY

Scientific Energy Systems Corporation has been engaged in the development of advanced engine systems since its incorporation in 1968. The initial purpose behind the Company's founding was to develop and manufacture steam engines for motive power and stationary applications. Early development work was carried out on Company funds, with emphasis on low emissions combustors, compact steam generators, and high temperature steam expanders. During this period, an advanced homogeneous combustion system and compact steam generator was conceived and developed, and in 1970, a complete 112 kW (150 HP) V-4 engine system was operated at 6.9 MPa and 811°K (1000 psia and 1000°F).

In March 1971 the Company received a major development contract from the U. S. Environmental Protection Agency (EPA) to continue the work. The program was one effort to demonstrate a low emissions, competitive alternative powerplant to the conventional spark ignition engine. The contract has led to the development of a modern high performance steam powerplant installed in a 1975 Dodge Monaco passenger automobile. A photograph showing the automobile in the SES test facility is shown in Figure 2.1.

The vehicle design goals and specifications set forth for this program by the EPA are summarized in Table II.1. Salient features of this specification are:

1. The use of a six-passenger auto having a maximum curb weight of 1950 kg (4300 lb).

TABLE 2.1

EPA VEHICLE DESIGN SPECIFICATIONS
FULL-SIZE SIX PASSENGER FAMILY CAR

Weight:	
Vehicle weight without propulsion system:	2700 lb (1225 kg)
Maximum propulsion system weight:	1600 lb (725 kg)
Maximum curb weight:	4300 lb (1950 kg)
Test weight:	4600 lb (2087 kg)
Gross vehicle weight, maximum:	5300 lb (2404 kg)
Volume (maximum):	
(Entire propulsion system must fit within engine compartment)	35 ft ³ (1 m ³)
Emissions - g/mile over federal driving cycle	
NO _x	0.2 (max)
CO _x	1.7 (max)
HC	0.2 (max)
Start-up time:	
From key-on to 65% power	45 sec
Acceleration performance:	
(level grade, 29°C ambient)	
0-60 mph	13.5 sec (max)
25-70 mph:	15.0 sec (max)
0-10 sec:	440 ft (min)
DOT high speed pass, 50-80 mph:	15.0 sec, 1400 ft (max)
Gradability:	
30% grade:	15 mph (24 km/hr)
5% grade (continuous):	60 mph (97 km/hr)
Level grade (continuous):	85 mph (137 km/hr)
Minimum range:	200 miles (322 km)

FIGURE 2.1 - SES Steam Engine in 1975 Dodge Monaco

2. Acceleration performance equal to the majority of cars of this class sold in 1969 (for example, 0-96.6 kmpm (0-60 mph) in 13.5 s).

3. The entire Rankine cycle powerplant must fit into the existing engine compartment volume of $1. \text{ m}^3$ (35 ft^3). Total propulsion system weight not to exceed 725 kg (1600 lb) (approximately 136 kg greater than the current gasoline engine powerplant).

4. Exhaust emission goals equal to one-half of the original 1976 federal standards.

Vehicle performance analysis has shown that a steam engine capable of producing a maximum of about 112 kW (150 HP) gross expander power will deliver the required acceleration performance and will be more than adequate to meet gradability requirements.

2.1 Steam Powerplant Characteristics

The steam powerplant developed by SES incorporates the following general features:

1. Pure water as the working fluid. Although water freezes within the normal operating temperature range, its superior thermodynamic, thermal stability, and heat transfer properties make it an attractive choice over other presently known fluids. The superior thermodynamic and heat transfer properties of water are responsible for the achievement of higher thermal efficiency and more compact system hardware

than has been attained using other working fluids. The freezing problem could be adequately handled by properly designing for drainage into an insulated sump; and the use of a small, automatic heater which prevents freezing and the attendant inconvenience.

2. High pressure, high temperature steam conditions for compactness and efficiency.

3. A 4-cylinder, single-stage reciprocating uniflow expander employing conventional automotive "trunk-type" pistons with the steam cylinder, engine crank bearings, and cam surfaces all lubricated by a common crankcase oil. Output power is controlled by regulating the period of inlet steam flow during the expansion stroke (variable cut-off).

4. Expander-driven auxiliaries and accessories such as the feedpump, alternator, air-conditioning (A/C) compressor, power steering, and power brakes. The expander idles to drive the accessories at zero road speed; a three-speed automatic transmission provides for suitable acceleration and part-load economy characteristics.

5. A highly compact burner/vapor generator package incorporates features which reduce the total length of heat transfer tubing, provide efficient utilization of space, and provide excellent control over emissions, especially NO_x , by means of a unique, premixed homogeneous combustor.

6. An air-cooled condenser and cooling fan system capable of condensing all steam passed through the expander. A variable-speed drive on the fans and a floating condenser pressure permit a fully condensing system with reasonable condenser size and peak fan power, while allowing for reduced fan power and improved fuel economy during cruise conditions.

A schematic diagram of the powerplant is shown in Figure 2.2. Major system components are the expander, burner/vapor generator, condenser, and feedpump, which are arranged in a closed loop as illustrated. The expander drives the vehicle through a conventional three-speed automatic transmission and differential axle. Accessories such as the alternator, A/C compressor, and hydraulic pump for steering and braking assist are all belt-driven from the expander, which idles when the vehicle is at rest. Brief descriptions of the major components and systems are given below. Detailed specifications, operating characteristics and subsystem test results are contained in Sections 3.0 through 7.0.

Expander

The main features of SES steam expander are as follows:

- Brake power of 112 kW (150 HP)
- Speed range of 6.7-42 Hz (400-2500 RPM)
- 4-cylinder in-line, trunk piston, uniflow exhaust

FIGURE 2.2 STEAM POWERPLANT SCHEMATIC DIAGRAM

- Total displacement of 2.2 liters (135 in³);
8.89 cm bore x 8.89 cm stroke (3.5" x 3.5")
- Inlet steam conditions of 6.9 MPa, 811^oK
(1000^oF, 1000 psia)
- Exhaust pressure: 2.2-5.1 kPa (15-35 psia)
- Cruciform crankshaft, 1-3-4-2 admission order
- Series poppet admission valving with variable cut-off
- Shell bearings, pressure lubricated
- Counter-rotating balance shaft giving complete balance of primary and secondary forces

Longitudinal and transverse cross sections of the prototype expander are shown in Figures 2.3 and 2.4.

The expander block casting is made in two pieces in order to simplify prototype manufacture and to provide flexibility for changes to meet vehicle installation requirements. The top cylinder block casting is designed with its bottom face at the crankshaft centerline, and the main bearing caps are integral in a second box casting which also carries the oil pumps and the balancer shaft.

The cast-iron cylinder block carries pressed-in, Porous chrome-plated iron liners, these can be easily removed for modification to exhaust ports or for changes in liner material or surface finish. Exhaust steam passes through the discharge ports in the liner to a common gallery in the cylinder block and leaves through an elongated port connecting to a bolt-on manifold.

FIGURE 2-3 - PROTOTYPE L-4 EXPANDER CROSS SECTION

FIGURE 2-4 - PROTOTYPE L-4 EXPANDER TRANSVERSE CROSS SECTION

Individual cylinder heads of Meehanite G40 cast iron are used to minimize differential longitudinal expansion between the hot cylinder heads and the cooler cylinder block. Each head is retained in the block by six bolts spaced around the cylinder and four additional bolts in the valve/tappet chest area.

The series poppet valve system is shown diagrammatically in Figure 2.5. High pressure steam enters the cylinder through a series pair of poppet valves where the admission valve has fixed timing with respect to the crankshaft and controls the start of admission, while the other (cut-off) valve has variable timing and regulates the end of the admission period. Steam admission to the cylinder occurs only during the overlap of the two valve events; variable overlap is obtained by shifting the phase of the cut-off cam with respect to the crankshaft, and thus, controls the steam flow and power output.

Mechanical design features of the series valve arrangement are shown in Figures 2.3 and 2.4. Two camshafts are employed, one for the cut-off valves and one for the admission valves. They are driven from the crankshaft through a double roller chain. The phase relation of the cut-off camshaft to the crankshaft is varied by means of a phase change mechanism attached to the front end of that camshaft.

SERIES INLET VALVE ARRANGEMENT

FIGURE 2.5

Steam Generator

The SES automotive steam generator is a very compact unit, capable of supplying superheated steam at 6.9 MPa and 811°K (1000°F and 1000 psia). A cross section of the unit is shown in Figure 2.6.

Combustion air enters through the center of the combustion chamber. Uncooled combustion products are drawn through the recirculation ports (E) and mix with the incoming air to produce a final mixture temperature in the range 394-450°K (250-350°F). The flow of air is modulated by the air throttle valve (A) located downstream of the recirculation ports. The fuel and atomizing air are brought to the fuel nozzle (B) where the fuel is sprayed against the direction of air-flow to provide good spatial distribution of the fuel aerosol prior to entering the blower. The air velocity at the air throttle valve is maintained high enough so that the fuel spray does not penetrate upstream of the throttle valve.

The fuel-air mixture passes through the blower (C), through a vaneless diffuser (D), and then flows radially inward through a vaned diffuser (F) which removes most of the swirl velocity component. Final flow straightening is accomplished in a short honeycomb (G). By the time the mixture has reached the flameholder (H), the fuel is vaporized and has been quite well mixed going through the blower and diffusers. The flameholder is a perforated, sheet-metal, Meker-type of burner, the perforations being 0.84 mm (0.033 in) round holes representing approximately an

MODEL 8 AUTOMOTIVE STEAM GENERATOR

FIGURE 2.6

18% open area. The combustion products flow through the combustion chamber to the heat exchanger (I) and then to the exhaust duct (J).

An important feature of the combustor is the positioning of a bare coil approximately midway through the combustion zone. The coil partially cools the combustion gases to a temperature of 1673-1873°K (2550-2910°F), effectively quenching the NO_x reaction, but permitting continued oxidation of CO and HC.

The burner/vapor generator operates in a fully modulated control mode to supply steam at a rate which matches the demand. The steam generator operates over a steam flow range of 27-550 kg/hr (60-1200 lb/hr). The fuel flow varies from 1.8-45 kg/hr (4-100 lb/hr), as the blower speed changes from 33 Hz to 100 Hz (2000-6000 RPM). The clean emissions characteristics of the steam generator are illustrated in Figure 2.7.

Feedwater Pump

SES began development of a feedwater pump when a commercial pump which would satisfy all requirements of the steam powerplant could not be found. To satisfy system requirements, the pump must be compact, have a delivery pressure of 8.3 MPa (1200 psia), supply 9.5 l/min (2.5 gpm) at 20.8 Hz (1250 RPM) expander speed, be able to pump pure water at a temperature of 405°K (270°F) and operate with a low value of net positive suction head (NPSH) to avoid pump cavitation. The result of this development effort, a positive displacement, efficient high pressure pump is shown in Figure 2.8.

FIGURE 2.7 - MEASURED MODEL 5 VAPOR GENERATOR EMISSION VERSUS LOAD
AT OPTIMUM AIR/FUEL RATIO, FULL LOAD COMPRISES 100 LB/HR
FUEL FLOW AND 1200 LB/HR STEAM FLOW AT 1000°F/1000 PSI

SES FEEDWATER PUMP

FIGURE 2.8

The feedwater pump is driven directly off the expander. Its control is accomplished by unloading solenoids that keep the intake valves open. Each of the pump cylinders is separately controlled by modulating the fraction of time (pulse width modulation) during which its intake valve is unloaded, resulting in an efficient and smooth water flow modulation technique.

Heat Rejection System

The heat rejection system of the SES steam engine carries out a variety of functions as follows:

- Condensation of exhaust steam to maintain the exhaust pressure between .10 MPa and .24 MPa (15 and 35 psia)
- Subcooling of feedwater to avoid feedpump cavitation
- Condensation of steam that leaks into the crankcase
- Engine oil cooling
- Transmission oil cooling
- Hydraulic oil cooling
- Air conditioning condenser

The condenser is constructed of fluxless brazed and welded aluminum of the plate-fin type, using perforated air-side fins which were developed for automotive Rankine cycle condensers by Garrett AiResearch under an EPA contract. The cooler package is constructed of Wolverine finned tubing. The layout of condenser and coolers is shown in Figure 2.9.

SIDE VIEW OF COOLERS PACKAGE

FIGURE 2.9

Condenser air-flow is provided by twin 19 in. cast aluminum axial fans driven by the expander through a variable ratio hydrostatic drive. This permits modulated disengagement of the fans under cruise conditions when ram air is sufficient to cool the condenser. Under high-speed cruise conditions, during decelerations, or at very low ambient temperatures, additional modulation to air-flow is provided by shutters in the grille, which are controlled by condenser pressure. Modulation of the pump displacement in the fan hydrostatic drive and the shutters is done by a Bellows type actuator which uses the condenser pressure such that the actuation force is proportional to the condenser pressure.

Auxiliary-Accessory System

Figure 2.10 shows a schematic diagram of the steam engine accessory drive system. The driven components are:

1. Feedwater pump - direct drive.
2. Burner blower - variable ratio hydraulic transmission enables independence between engine speed and burner operation.
3. Condenser fans - variable ratio hydraulic transmission enables condenser pressure modulation. This drive can be disengaged, reducing the starting power requirement and taking advantage of ram air supply when the vehicle is cruising.

CURRENT SYSTEM AUXILIARY DRIVES SCHEMATIC
FIGURE 2.10

4. A fuel boost pump and a metering pump are driven electrically off the vehicle battery.

5. The air pump, used to atomize the fuel before vaporization - electrical drive.

6. Subcooler pump, circulates water through the subcooler - electrical drive.

7. Actuating oil pump, supplies oil to the hydraulic actuators - electrical drive.

There are two electrical starters in the system. The first one is used to start the steam generator burner and water pump. The second one is used to start the expander after the steam pressure is sufficiently high. The expander originally was self starting with steam pressure, but a reversal of the camshaft functions prevented inclusion of a self starting feature. Self starting of the expander could be incorporated in a new block design.

Control System

The basic steam engine controls philosophy is shown in the system schematics of Figure 2.5. There are six major logic functions in the system.

1. Power control package: takes into account the driver command (foot pedal position) and expander parameters (speed, intake pressure and exhaust pressure) and controls the cut-off angle to provide smooth engine response and gear changes.

2. Burner control package: Controls the required amount of fuel and air by modulating the blower hydraulic pump stroke, the air valve position servo motor and the fuel metering pump.

3. Steam generator control package: Controls the steam outlet conditions and boiler tubing temperatures by operating the feedpump unloading solenoids and injecting water into the superheater.

4. Condenser control package: This package consists of a Bellows type actuator that actuates the condenser shutter and hydraulic pump stroke to maintain the condenser pressure between .10 MPa and 0.24 MPa (15 and 35 psia).

5. The system is equipped with automatic startup and shutdown sequencing controls. On startup, the feedwater pump and blower are driven by a DC motor. After ignition, the expander is cranked by a conventional start motor. When the expander is running, the system is switched to the normal control mode. On a shutdown, all water in the system is blown to the condenser sump, which would be insulated and heated in a final version to provide freeze protection.

6. An emergency subsystem continuously monitors important engine parameters: critical temperatures, speed and pressure, and triggers a shutdown procedure upon abnormal circumstances.

2.2 Test Result Summary

Major systems and components of the powerplant were tested individually during their development. After the completion of individual component testing, the system was assembled in a preprototype configuration. All components were connected, but were not in their prototype (engine compartment) positions. Back-up systems, as for example condensing, feedwater and steam generation, were available to substitute for any inoperative component to minimize interruptions in testing. A photograph of the preprototype (PPT) engine configuration is shown in Figure 2.11. During the course of component and PPT testing, the steam generator was operated for over 1000 hours, and the expander for over 800 hours.

In the final preprototype configuration, the expander was coupled to a 300 kW (400 hp) dynamometer through a three-speed automatic transmission, with a flywheel simulating vehicle inertia placed between the transmission and the dynamometer. This enabled simulation of transient and steady-state EPA performance criteria, with the exception of the 30% grade capability due to the steady-state torque limitation of the dynamometer.

In the limited time available for the transient PPT testing much was learned about the system. Significant results were noted in the following areas:

1. Clean, smooth start-up in 20 to 25 seconds.
2. The system is responsive and drivable.

FIGURE 2.11 SES PREPROTOTYPE SYSTEM ASSEMBLED ON TEST STAND

3. Emissions are good although an improvement of air to fuel ratio control during rapid load changes is needed.

4. Smooth and clean gear shifts.

5. Good steam generator pressure and temperature control, however, manual corrections were still required by operator.

A summary of the PPT test results is shown in Figures 2.12 and 2.13. As can be seen, the steam engine has sufficient power, good emission characteristics and encouraging fuel economy for a first generation engine.

Subsequent to PPT testing, the prototype, power-plant system was placed in the engine compartment of a Chrysler Corporation modified 1975 Dodge Monaco vehicle, for actual demonstration in an automotive environment and proof-of design testing. Many development problems and pains were experienced; however, the final 'simulator' system performed very well, accumulating over 2,100 km (1300 miles) of chassis dynamometer operation and met the Federal emissions standards of 1976.

Table 2.II presents the test results of the urban and highway Federal Driving Cycles (FDC). A comparison between the above test results and the steady-state fuel consumption and emissions reveals that there is still room for improving the dynamic response of the system. The urban driving cycle in particular, being a highly power varying cycle where fast transients are required, suffers a penalty in both fuel economy and emissions. Nevertheless, the original 1976 emissions standards were met and extremely valuable experience was gained.

EXPANDER PERFORMANCE

NET HORSEPOWER vs SPEED, WITH
LINES OF CONSTANT FUEL FLOW
BASED ON PROTOTYPE TEST RESULTS

FIGURE 2.12

STEADY-STATE ROAD LOAD FUEL ECONOMY

4600 POUND VEHICLE

FIGURE 2.13

TABLE 2.II
SUMMARY OF SIMULATOR PERFORMANCE RESULTS

URBAN FDC RESULTS (1975 FTP)

	MPG	NO _x (GMS/MILE)	UHC (GMS/MILE)	CO (GMS/MILE)
ORIGINAL 76 STANDARDS	-	0.40	0.41	3.4
29 OCT 76	6.8	0.34	0.47	4.4
30 OCT 75	7.1	0.36	1.10	6.9
4 NOV 75	7.5	0.26	0.58	3.7
6 NOV 75	7.5	0.25	0.37	3.4

HIGHWAY CYCLE

AUG 75	10.2 MPG	(105°F AMBIENT)
5 NOV 75	11.9 MPG	
6 NOV 75	12.2 MPG	

COLD START EMISSIONS DEVELOPMENT

COLD TRANSIENT - BAG ONE CARBON BALANCE

TEST DATE	FUEL (GAL)	NO _x (GMS)	UHC (GMS)	CO (GMS)
29 OCT 75	0.67	1.53	4.42	47.1
30 OCT 75	0.61	1.77	12.0	69.1
4 NOV 75	0.57	1.40	1.40	46.8
6 NOV 75	0.58	1.27	2.13	41.5

FUEL ECONOMY - ROAD LOAD
 - 15.5 MPG @ 39 MPH
 - 14.9 MPG @ 52 MPH

MAX SPEED - 45 MPH (1st GEAR)
 TESTED - 60 MPH (2nd GEAR)
 (NOT FULL - 70 MPH (3rd GEAR)
 POWER)

MAX ACCELERATION RATE - 3.5 MPH/SEC
 NO OF SIMULATOR TEST MILES - 1321
 SHORTEST CRANKING TIME - 13 SECONDS - COLD START
 - 5 SECONDS - HOT START

Currently, SES is working on the adaptation of its steam engine into a coal haulage underground vehicle under U.S. Bureau of Mines Contract Number HO 155121. Knowledge accumulated throughout the simulator testing was used to further analyze the boiler dynamic behavior and apply hardware changes resulting in great improvements in both the dynamic and steady-state characteristics, together with a change over to heating oil instead of gasoline. Backed by these new data, SES has no doubt that the original emissions goals are feasible and even conservative.

The development of high temperature lubrication know-how has recently led SES to believe that a high temperature, high efficiency two-stage reheat steam engine is also a feasible, near-term goal. Such an engine could have a fuel economy exceeding that of a Diesel engined automobile of equal acceleration performance. SES has carried out a preliminary design for a reheat steam engine and is currently engaged in the development of a reheat boiler.

3.0 PREPROTOTYPE EXPANDER

This chapter will summarize the design philosophy, basic configuration and drive system of the SES-Ricardo L-4 uniflow steam expander for the steam engine powerplant.

A major portion of this contract effort was devoted to expander design and development. In all, three different expanders were ultimately tested, two strictly as research devices (V-4 and Single Cylinder) and one as the "preprototype" (L-4) intended for vehicle testing with a complete preprototype engine system. This section discusses this work with little distinction being made between the three expanders due to the similarity in basic components particularly between the single cylinder and the L-4 (inline four cylinder) and the fact that the V-4 was phased out early in the program.

In chronological order, the first expander was a four cylinder V type was built and under test at SES before award of the contract and was a simple modification of a production air cooled diesel manufactured by Bernard in France. The second expander to run was an extensively modified single cylinder Bernard diesel (Figure 3-1). All critical components of the cylinder/piston/valve train were nearly interchangeable with like parts of the final expander, the inline 4 (L-4) illustrated in Figures 3.2 and 3.3. The single cylinder version accumulated over 2400 hours of development testing including several continuous runs of 150 hours each and one continuous run of 300 hours. Two inline 4 cylinder expanders were built, the first L-4 accumulated 325 hours on dynamometer testing and the second testing plus an additional 150 hours in the car testing phase.

FIGURE 3-1

The "Basic Expander Configuration" section 3.1 provides the rationale for selecting an inline four cylinder expander for the project and some of the detail features such as variable cut off control and trunk piston design.

The Expander Drives Section 3.2 describes the trade offs in design of the vehicle drive train and accessory drive.

Piston/Ring lubrication discussed in section 3.3 covers the most intensive durability development effort of the contract. In summary, suitable oils and oil additives were developed by Exxon with a natural hydrocarbon base stock oil and somewhat better oil by Mobil with their synthetic hydrocarbon oil. Several ring coatings proved to be satisfactory but a conventional cylinder bore coating (hard chrome) could not tolerate the high liner temperature of 370°C (700°F) at the top of the ring travel due to softening of the chrome. Copious lubrication could prevent wear of the chrome but the high oil consumption was intolerable. Excellent results were ultimately obtained with Tribaloy (DuPont proprietary) plasma sprayed rings and liner with no measureable wear or visible softening in 150 hours of testing. The Tribaloy tests were conducted with the Exxon lubricant.

The Bearings and Tappets section 3.4 discusses the success achieved in development of the crankshaft, connecting rod and sliding valve tappets for the severe environment of a steam engine. Long duration loads and unidirectional loads of a "two stroke" type (a single acting steam engine required relatively large bearings to survive). The valve

tappets and cams ultimately survived the very high loads, imposed by an unbalanced poppet valve being pushed open against 6.9 MPa (1000 psi) steam.

Piston structural development discussed in Section 3.5 points on the severe thermal gradients and distortion problems and how they were overcome.

Valve guide wear and sealing is discussed in Section 3.6.

Oil and water separation is discussed in Section 3.7.

Expander performance development and analysis is presented in Section 3.8. A peak expander efficiency of 70 percent was achieved but this was only realized at full power. Better vehicle fuel economy, perhaps as much as 20 per cent better could have been possible had a much smaller expander been developed. This smaller expander could have had the same peak power output, but the peak efficiency would have occurred at lower powers where the bulk of the fuel is consumed in normal driving.

3.1 Basic Expander Configuration

Early in the program the task of choosing the expander configuration was completed. Dual expansion, single expansion crosshead and single expansion trunk piston expanders were evaluated for the automotive application. Detail computer thermodynamic studies by Ricardo (the SES subcontractor for the initial expander hardware) indicated that a single expansion trunk piston expander was the most attractive expander.

A simple uniflow expander, when operated with highly superheated steam will not have any appreciable condensation losses, thus the incentive for compounding was eliminated.

Design studies of crosshead expanders revealed several points which made such an expander undesirable for automotive use. A crosshead expander is much more bulky and more costly to produce. The packaging of the expander with the automobile powerplant cavity becomes acute. Moreover, cylinder lubrication becomes more difficult at the higher wall temperatures in the crosshead design.

Three expander configurations, all of the single expansion trunk-piston type, were studied from the point of view of suitability for packaging within the automobile, expander balance, and simplicity of design without compromising performance. A horizontally opposed 4, a V-4 and an in-line 4-cylinder expander were studied in detail by Ricardo and SES.

The horizontally opposed expander was rejected on the basis that it provided no packaging advantages; it required an additional camshaft when used with a series poppet valve system and had poor drainability.

Both the in-line 4 and V-4 expanders were studied in great detail to determine the best configuration for the automotive application. From the packaging point of view the in-line 4 allows more freedom in locating the steam generator and auxiliaries and, what is more important, the design provides for less condenser fan flow restriction.

Complete balancing for both primary and secondary forces on the V-4 expander can only be accomplished by adding two balance shafts (for secondaries) whereas a completely balanced in-line 4 expander can be accomplished with the addition of only one balance shaft (for primary balance). It is possible to accomplish the primary force balance on the in-line 4 engine by adding weights to one of the camshafts thus doing away with the balancer shaft; however, this design results in a somewhat more bulky expander.

The specifications for the preprototype expander design agreed upon by SES and Ricardo were as follows:

- 4 cylinders in-line, single acting trunk piston design, 81.9 cm (3 1/2") bore and 8.9 cm (3 1/2") stroke nominally.
- 1-3-4-2 crankshaft and counter-rotating balancer shaft.
- Series-admission unbalanced poppet-valves, with cut-off to be accomplished by a camshaft phase changing mechanism.
- Sliding tappet valve train with room for roller tappet if found necessary.
- Uniflow exhaust porting with recompression relief valves provided between cylinder and steam manifold.

- Palm end wrist-pin bearing design, with a double-eccentrically machined pin design as a back-up system should it be required.
- Wet sump design with freedom to alter system to a dry sump should that be preferred later.
- Auxiliaries driven from the front end of the expander.

An analytical study was undertaken to determine the importance of various admission valve parameters on indicated engine performance.

1. Single poppet valve with fixed cam period, and a timing and/or throttling control.
2. Ideal "square-wave" function (instantaneous opening and closing) with cut-off control.
3. Rotary valve (trapezoidal function) with cut-off control.
4. Two poppet valves in series with cut-off control achieved by phasing the valves.

The effect of the following parameters was analyzed:

1. Cut-off control versus timing and/or throttling control.

2. Maximum valve opening flow area.
3. Speed of opening or closing.
4. Time at which valve opening is initiated (lead angle).

A number of generalized conclusions were reached. Cut-off control improves engine efficiency over other techniques such as timing control, inlet pressure throttling control, or a combination of the two. This is particularly true at reduced power levels corresponding to vehicle cruise conditions. Increased valve flow area increases efficiency, although practical limits can be determined where the rate of improvement diminishes. The rate of opening or closing is relatively unimportant, such that a small increase in maximum flow area can offset the effect of a large decrease in opening rate. An optimum lead angle of about $.09 - .17$ rad ($5-10^\circ$) exists for nominal valve functions. This optimum is a function of many parameters, but is very shallow. The series poppet valve can give performance only slightly lower than the "ideal" valve function, and does not have the many inherent problems that other systems have. Figures 3.2, 3.3 and 3.4 show cross sections and the valving arrangement of the expander.

The admission period control is achieved by varying the timing of the cut-off valve. Thus, steam admission to the cylinder occurs nominally only during the overlap of the two valve events; the variable overlap obtained by shifting the phase of the cut-off cam controls the steam flow and power output.

Mechanical design features of the series valve arrangement are shown in Figures 3.2 and 3.3. Two camshafts are employed, one for the cut-off valves and one for the admission valves. They are driven from the crankshaft through a double roller chain. The phase relation of the cut-off camshaft to the crankshaft is varied by means of a phase change mechanism attached to the front end of that camshaft.

The mechanism, shown in Figure 3.4, is a rotary hydraulic actuator, comprising an outer casing driven by the timing sprocket and containing a two-vaned rotor which is splined to the cut-off camshaft. Expander lubricant under pressure for operating the mechanism is supplied from the adjacent camshaft bearing and is discharged through the center of the camshaft into the camshaft tunnel.

Phasing of the rotor relative to the outer casing is set by axial movement of a spool valve with two pairs of helical grooves; the spool is coaxially located within the rotor shaft, which provides entry and exit of oil for each of the four segmental chambers. In the equilibrium position all ports are very nearly closed, so that the system is virtually solid. Any deviation from the set point is resisted by increased inlet port area and reduced exit port area in one pair of chambers, and the converse in the other pair of chambers, resulting in a torsionally stiff system. The mechanism provides a phase change of 11 rad (70°).

L-4 PREPROTOTYPE EXPANDER RIGHT SIDE SECTIONS

Figure 3.2

L-4 PREPROTOTYPE EXPANDER FRONT CROSS SECTION

Figure 3.3

DEVELOPED VIEW OF VALVE & PORTING

FIGURE 3.4

3.2 EXPANDER DRIVES

An installation in which the expander was mounted coaxially with the automotive driveshaft was chosen for the preferred packaging arrangement. Off-the-shelf transmission components were chosen for use with this expander installation. American Motors Corporation, Warner Gear and Borg and Beck all assisted in this task.

In order to use a standard automotive transmission it was necessary to provide a 2:1 speed increase in expander output, resulting in a similar reduction in torque input to the transmission. A Warner marine reduction planetary gear set was found which provided a 2.1:1 speed-up of expander speed when used as a speed increaser. It had sufficient capacity to handle in excess of 949 Nm (700 ft-lb) of torque input. In order to limit the expander length, both SES and Ricardo undertook to integrate the gear set with the flywheel to the extent that this was feasible. Several schemes were studied for providing proper lubrication of the gears. One method was to provide an integral oil pump and sump. Another method was to use a wet-sump/splash system. The final choice was to provide positive oil pressure from the expander rear-main-bearing to the planetary gear set. The oil from the gear set housing drained back into the expander oil pan.

Design studies included front-end drives of auxiliary and accessory components from the expander. An attempt was made to integrate components to the extent that it

was possible to do this (i.e., mount several components on a common driveshaft). This approach would allow for pad mounting of some of these components on the expander crankcase and provide for a more compact and cleaner looking package. It was also considered feasible to drive some components from a camshaft.

Two failures of the expander step-up gear input shaft occurred over a period of 600 engine hours. A flexible drive was designed and built incorporating helical springs for flexibility similar in principle to the conventional clutch disc drive. (See Reference 16 for a detailed discussion of the flexible drive.) Subsequent automatic start-up and transient testing revealed that the installation of the expander flexible drive resulted in marked improvement in smoothness of the system operation. It was found that the 13.0 Hz (800 RPM) wide span of the "noisy" region encountered without the flexible drive decreased to two bands, one around 4.17 Hz (250 RPM) which was avoidable being sufficiently below the proposed idle speed, and the second around 9.17 Hz (550 RPM) which was potentially dangerous. In the latter region the noise became pronounced at higher loads only.

The ultimate solution comprised a means of preventing the engine speed from dropping into the critical level at any but zero or light load. This was done by using a frictional clutch (modulation of the rear clutch in the Chrysler 727 automatic transmission) to get the car in motion. The torque converter was eliminated. The operation

of the steam car with this modification was by automatic control of a pressure modulating valve that would engage the rear clutch in the transmission in such a way that the expander speed would not drop below a predetermined value while in gear. In addition, the clutch would automatically disengage when the car is stopped, saving fuel normally lost in the torque converter drag. Subsequent testing demonstrated that the expander and drive system were capable of preprototype operation in the simulator vehicle.

3.3 PISTON/RING LUBRICATION

It was stated in early reporting that contact wear in cylinders occurs primarily through lubricant failure. Consequently, the progress in lubrication was toward improved high-temperature viscosity, decreased volatility and maximum thermal stability. In the Rankine cycle expander, low volatility is essential. Low volatility assures that a viscous film will survive the evaporation which can occur on cylinder liners at 633.2°K (680°F). Sufficient oil must remain to make a hydro-dynamic film, or contact wear will result. Figure 3-5 illustrates the cylinder liner temperature environment as a function of expander power.

Exxon Research and Engineering Company analyzed samples of used oils as well as samples of filtered out material. Analyses revealed high Fe_3O_4 deposits in all samples, and this result led to the successful switch from the Rokon-Parebar 448 additive to the ENJ-1031 anti-rust agent. The results of the laboratory test of a sample of the lubricant 2415-122-1 after 280 hours of running-in the L-4 indicated that only negligible degradation of the oil occurred in this service. Used oil inspection results are presented in Table 3-I and the new oil properties of all the candidate lubricants are presented in Table 3-II for purposes of comparison.

Synthetic oils possess lower volatility characteristics than mineral base hydrocarbon oils. SES, therefore, initiated a synthetic oil investigation. As excellent demulsibility of the oil is a prerequisite for SES's lubricating oils, demulsibility tests were designed to screen oil candidates prior to any expander evaluation. Exxon first

MAXIMUM CYLINDER TEMPERATURE AT TOP
RING POSITION

FIGURE 3-5

TABLE 3.I

PROPERTIES OF LUBRICANT 2415-122-1 AFTER 280 HOURS OF OPERATION	
<u>Properties</u>	
Vis & 311°K(100°F)(SUS)	774
Vis & 372°K(210°F)(SUS)	79.6
VI	101
% Vis Increase at 311°K(100°F)	2.1
Pentane Insolubles (Wt. %)	.04
Water By Distillation (Vol. %)	1.6
Phosphorous (Wt. %)	.08
<u>D-664 Neut No.</u>	
PH	6.0
TAN	.64
TBN	.06
<u>Metals Levels (ppm)</u>	
Aluminum	3
Chromium	12
Copper	32
Iron	163
Lead	7
Silicon	19
Zinc	3
<u>4-Ball Wear Test Results</u>	
Max Contact Stress MPa (PSI $\times 10^3$)	3850
Seize load kg	100
Weld Load kg	158
Wear Index	33.5
Avg. Wear Rate (cm ³ cm ⁻¹ kg ⁻¹)	5.53 $\times 10^{-12}$

TABLE 3.II

SUMMARY OF THE COMPOSITION AND PROPERTIES OF CANDIDATE LUBRICANTS

SHOWN IN CHRONOLOGICAL ORDER OF DEVELOPMENT

Blend Number	1 50-1	2 113-1	3 122-1	4 138-1	5 151-1
Additives (Wt. %)					
Ethyl-702 (oxidation inhibitor)	1.0	1.0	1.0	1.0	1.0
Tricresyl Phosphate (boundary lubrication additive)	1.0	1.0	1.0	1.0	1.0
ENJ-1031 (Rust inhibitor)	0.3	-	-	0.5	0.5
Parabar-448 (Rust inhibitor)	-	.25	.25	-	-
Rokon (Rust inhibitor)	-	.25	.25	-	-
PC-1244 (Monsanto's additive)	-	-	300 ppm	-	-
DC-200	30 ppm	30 ppm	60 ppm	30 ppm	30 ppm
ECA-4543 (Polymer additive)	-	-	-	-	plus 20%
<u>Properties</u>					
Vis & 311°K (100°F) (SUS)	663	754	758	780	1819
Vis & 366°K (210°F) (SUS)	74.0	77.9	79.2	80.1	167
VI	102	101	102	101	141
Phosphorous (Wt. %)	.07		.08	.08	0.09
Flash Point (COC) (°F)	(1)	588°K	560°K	566°K	-
<u>D-664 Nuet No.</u>					
pH	7.3	6.7	6.0	6.3	8.1
TAN	.57	.97	1.14	.86	0.92
TBN	.12	.08	.11	.17	1.38

considered and then screened ester-based fluids used for gas turbines - specifically ETO-25, ETO-5251, and ETO-2380 -then the more recently developed synthesized hydrocarbons and the combination of synthesized hydrocarbon and ester fluids. (The synthesized hydrocarbon fluids recently demonstrated excellent performance in both Wankel and reciprocating I.C. engines.) The oils passing the demulsibility requirements were then evaluated for: distillation, rust protection, oxidation stability and compatibility with elastometric seal materials used in automotive engines. The initial Exxon investigation was limited to the evaluation of currently available synthesized oils.

The Lubrizol additive package formulated specifically for chrome carbide rings and chrome liners was evaluated with poor results. The Lubrizol oil consisted of the Exxon mineral base stock (without Exxon's polymer additive ECA-4543) combined with Lubrizol's "Anti-wear" additive package. The higher liner wear rate confirmed that low volatility was the best guide to the quality of an oil to prevent cylinder contact wear in the SES Rankine cycle expander.

The metallurgical program included the evaluation of the Alumina-Titania-Calcium Fluoride (A/T/Cf) ring, ceramic liners and friction and wear rig testing.

The A/T/Cf ring was evaluated on hard chrome vapor blast liner with Exxon lubricant 151-1. This combination gave a higher wear rate than the chrome carbide ring on a

chrome liner and rings. To reiterate, the contact wear in cylinders occurs primarily through lubricant failure.

Koppers demonstrated that if a lubricant failure occurs with chromium, interfacial temperatures of the chrome are at least as high as 922°K (1200°F) and possibly 1033°K (1400°F). The chromium under these conditions undergoes irreversible softening, is plastically deformed and transferred to its mating surface. Thus, chromium liners were, therefore, replaced by porous plasma sprayed ceramic liners. The materials selected for evaluation were A/T/Cf, Tribaloy 400 and Molybdenum (Metco #505).

The piston ring and cylinder liner wear problems were resolved, and pertinent lubricating oil volatility data was obtained. The following is a review of the developmental approach leading to the wear problem solution, and the identification of lubricating oil deficiencies and oil improvements.

The predominant factor in obtaining zero wear was material compatibility. Both the ring and cylinder liner that were plasma sprayed with Tribaloy 400 exhibited zero wear after 150 hours of endurance testing. The plasma spray process was selected over casting because it could apply a dense and porous coating which was inherently valuable in the oil's retention to the cylinder liner and the oil's transportation by the ring. Tribaloy grade 400 is structurally a hard intermetallic phase in a softer matrix. Both the phase and the matrix resist plastic flow and cold welding to a mating surface.

Prior to the successful Tribaloy evaluation, two other plasma sprayed ceramic liner materials and electro-chemically deposited chrome liners were evaluated and found to be without the required degree of durability. The hard chrome liners were initially selected for this project based on their previous use in high performance engines. Hard chromium, however, was unsuitable for Rankine cycle application because, without sufficient lubricating film, chromium tends to weld microscopically to its mating surface (adhesive wear). The chromium under these conditions undergoes irreversible softening and plastically deforms and wears. Of the five plasma sprayed rings evaluated on hard chrome, the chrome carbide exhibited the best compatibility, the tungsten carbide scuffed, while the boron-nitride-nichrome, moly-nickel, and the alumina-titania-calcium-fluoride rings exhibited no signs of incipient scuffing. The ceramic liners, which are thermally stable, illustrated how surface combinations can be made to apportion wear between the ring and the liner under marginal lubrication conditions if excellent compatibility between the material does not exist. The alumina-titania-calcium-fluoride ceramic exhibited zero wear, while the chrome carbide ring wore excessively. Both ring and liner wore excessively in the case of the CaF₂S₂B liner and the chrome carbide ring. Only the Tribaloy ring and liner combination was totally successful.

The cylinder liner temperatures in Rankine cycle engines are 422°K (300°F) higher than the current I.C. engines, but in a steam environment free oxygen is absent which allows research on cylinder liner lubrication phenomenon to be conducted independent of oxidation limitations.

The cylinder component temperatures are limited, however, by the oil's thermal stability - cracking temperature. Both mineral and synthetic oils crack at 658-672°K (725-750°F), with subsequent viscosity loss and smoking. Volatility is the most important characteristic of a lubricating oil as it assures that a viscous film will survive the evaporation to make a hydrodynamic film, or contact wear will result.

The Exxon mineral based oil, developed for SES, had its volatility reduced by the addition of 10% polymer with a corresponding reduction in piston ring and cylinder liner wear. This oil, 151-1, was selected for all the ring/liner material evaluation test, including use in the simulator. To assess volatility characteristics, ASTM1160 distillation curves were obtained on both synthetic and mineral based oils. It was apparent from these curves that the Mobil hydrocarbon synthetic, 1301'H', polymerized to 2000 SUS 311°K (100°F), exhibits the lowest volatility, with only a 13% distillation loss at 644°K (700°F). The Exxon polymer oil has a 95% distillation loss at the same temperature. The 1301'H' oil demonstrated satisfactory cam protection with only ½% TCP, rust protection and excellent demulsibility characteristics during the intial engine screening tests.

Excellent demulsibility is obviously a prerequisite to any water-based Rankine cycle engine relying on splash lubrication. The aviation ester-based fluids, specifically ETO-25, ETO-5251, and ETO-2380, failed all the demulsibility tests, while Exxon mineral oil showed satisfactory performance. The Mobil hydrocarbon demonstrated the best demulsibility characteristics of all the oils screened.

Dry friction and scuffing tests were conducted by operating a pin in contact with a rotating disc (Figure 3.6). Disc RPM was adjusted to duplicate various mean piston speeds. As a known normal force was applied to the pin the friction force between pin and disc was measured by means of a calibrated strain gauge ring. Wear rates were expressed as a function of the weight changes of the pin and disc, and the wear scar due to contact. The test specimens were mounted within a kiln capable of producing temperatures to 811°K (1000°F). By introducing steam from an external generator, the material samples were subjected to high temperature steam. Thus friction and wear data were determined in an environment similar to that existing in a steam expander.

Preliminary tests of rig operation were carried out using a cast iron pin on a graphite disc under ambient temperature and pressure conditions at a velocity of 228.6 cm/sec (90 in/sec) and a normal force of 2 kg (4.4 lb). Friction coefficients from .18 to .12 were measured. The latter (.12) after the surfaces were "broken in". Strain gauge output of friction forces was recorded on a Sanborn strip chart recorder and found to be reliable. Unlubricated test results are presented in Table 3-III.

The conclusions of the Materials and Lubricating activities are:

That the durability of the Rankine cycle piston rings and cylinder liners is equivalent to automotive components, and the Rankine cycle piston rings and cylinder liners operate at peak liner temperatures of 422°K (300°F) higher than current I.C. engine liner temperatures.

SES FRICTION & WEAR RIG

FIGURE 3.6

TABLE 3-III

FRICTION AND WEAR DATA

PIN SAMPLE	K1020	K1000C	K28	K28	K28	K28	K1020	K28
DISC SAMPLE	HI-TI-LUBE	HARD CHROME	AL ₂ O HOT PRESSED	AL ₃ O SINTERED	NASA LUBE PS101	SILICON NITRIDE	HARD CHROME	HARD CHROME
TEMP °K	FRICTION COEFF.	FRICTION COEFF.	FRICTION COEFF.	FRICTION COEFF.	FRICTION COEFF.	FRICTION COEFF.	FRICTION COEFF.	FRICTION COEFF.
297°K (75°F)	.14	.37	.49	.62	.49	.59	.70	.47
394°K (250°F)	.29	.39	.45	.53	.44	.59	.67	.43
533°K (500°F)	.31	.38	.34	.32	.39	.45	.65	.36
672°K (750°F)	.29	.38	.31	.32	.42	.34	.51	.37
811°K (1000°F)	.21	.38	.31	.31	.39	.27	.42	.30

That 20% polymeric additive in the Exxon mineral oil reduced the oil's volatility and component wear. This oil, 151-1, was selected and used during the ring-liner material evaluations, and is used in the simulator.

That the Mobil hydrocarbon synthetic lubricating oil 1301'H' is potentially the best crankcase lubricant currently available for high temperature Rankine cycle engines because of its low volatility and excellent demulsibility characteristics.

3.4 BEARINGS AND TAPPETS

Unbalanced poppet valves were selected to minimize expander complexity. Since the unbalanced valves operate against 6.9 MPa (1000 psi) very high cam and tappet stresses result, particularly when very short valve events are desired. Initial short tests resulted in considerable wear, but over 1000 hours was ultimately accumulated on one valve train with no measureable wear and excellent surface appearances.

The successful design incorporated:

1. 90 MPa (130,000 psi) design Hertz stress.
2. Chilled cast iron tappets hardened to 53 Rc, having 50 inch spherical radius faces.
3. Cam nose radii of 0.3 inches, 0.75 inch width and a 5 minute transverse taper to promote rotation of the tappet.
4. Ferrox treatment of both the tappets and cams.
5. Copious oil supply with 10 microns filtration.

Lubrication of the rotating plain bearings was relatively straight forward. Minor problems were encountered in the single cylinder connecting rod big end bearing primarily because the small original diesel, journal dimensions were retained. Consequently, the bearing pressures were approximately 50 per cent higher than the L-4 rod bearing pressures. Finer oil filtration corrected abrasive

wear problems in the single cylinder rod bearing. Both the single and L-4 expanders suffered bearing wear problems in early tests of several short runs and cold starts were attempted. Both problems were corrected by providing a large clearance between the sump and oil pump inlet so that more condensed water could be accommodated before reaching the level of the pump inlet. Five to ten minutes of running was sufficient to boil off any water in the crankcase so that long term water accumulation in the expander oil sump was not a problem.

The plain bearing between the piston and connecting rod wrist pin required some development before satisfactory durability was achieved. In a single acting steam expander there is never any reversal of forces on the piston so that the wrist pin bearing is always loaded in the same direction thereby limiting the opportunity for oil to flow into the pressure zone. A similar problem occurs in two stroke cycle internal combustion engines but the steam expander is aggravated by a longer period of high pressure in the cylinder made possible by variable cut off up to very large steam admission periods.

A "palm end" bearing was selected for the wrist pin bearing. In this design the wrist pin is bolted to a platform or cradle at the top of the connecting rod so that the full length of the pin surface facing the direction of loading is available as a bearing surface. A bearing shell is put in the piston across the full width of the piston. Figure 3.7 in the next section is a photograph of the complete connecting rod, bearing and piston assemble of the crosshead type. This design technique resulted in a bearing

pressure of 17 MPa (2500 psi) based on projected area of the bearing shell. While this is typical of diesel engine practice, additional care was required to make this bearing survive the longer load duration of the steam expander.

The original piston incorporated a bronze bushing for the wrist pin bushing. A drilling up the connecting rod led from a high oil pressure zone in the connecting rod to crankshaft journal through the wrist pin and into a distribution arc in the center of the bushing. Grooves extended away from the distribution groove axially along the pin to the ends. The oil flow from the crankshaft and out the shell provides both lubrication and cooling of the wrist pin bearing. Distortion of the pin, limited oil flow and subsequent overheating of the wrist pin bearing weren't completely eliminated until the two piece piston was adopted. A major contribution to solving this problem was the critical piston broaching and production bearing shells made available by the General Motors Detroit Diesel Allison Division as they had already solved the same set of problems for their two cycle diesels.

3.5 PISTONS

The following descriptions of piston developments are broken up into three parts, arranged chronologically with developments.

Modified Diesel Piston - The original diesel piston had a combustion cavity at the top and oil draining holes in the bottom of the upper oil control ring grove (above the wrist pin). The drain holes, as well as the combustion cavity were filled with threaded aluminum plugs. These were secured in position by high temperature epoxy resin applied to the threads. Material was removed from the top of each piston to maintain inertia balance and the desired clearance volume. The weakest point in the pistons was the inadequate wrist pin bearing load capacity when used in a "two cycle" steam expander. To improve the bearing load capacity, the pistons were bushed with grooved bronze inserts. Even with this modification the wear rate of the bearings was unacceptable, although at no time did any of the bearings fail structurally. The only other problem was overheating and erosion of the upper land of the pistons, which occurred after testing at the full power output (26 bhp/cylinder). The development of a suitable steam piston began after the modified diesel piston work pointed out the limitations of the design. This led to the development of a bolted crown/palm bearing piston described below. This design was conducted jointly by Ricardo, Wellworthy and SES.

The Bolted Crown Piston - In a single-acting uniflow expander, the loading on the wrist pin is unidirectional, similar to a two-stroke I.C. engine, except that the maximum cylinder pressure is sustained through a larger crank angle during late cutoff operation. Several designs were considered to alleviate the small end load, the most attractive scheme being a palm bearing with the wrist pin bolted to the connecting rod. This allows a much larger bearing as there is no bearing area removed, for the normal connecting rod eye; thus the bearing extends the full length between the skirt walls. Bolting the wrist pin to the connecting rod imparts a positive rocking action to the wrist pin which allows the oil film to be replenished during each swing of the rod.

An aluminum piston was preferred to cast iron in order to minimize the reciprocating mass, and a steel crown was provided for strength, erosion protection, and to act as a heat barrier between the steam and the piston body. The prime function of the crown was to act as an insulator, thus the piston was designed to have a large air gap between the steel crown (SAE HNV6) and the aluminum body, (Wellworthy-W52). The crown was bolted to the aluminum body by four alloy steel bolts (AISI 4037) with an asbestos gasket between the crown and the aluminum to seal the air gap and to reduce heat transfer to the piston body.

The lead bronze palm bearing, designed for a maximum unit pressure of 2500 psi, was a press fit in an aluminum cross member, as were the end caps which prevented oil from leaking through the exhaust ports. Oil passing out of the end of the palm bearing drained down to the crankcase via a drilled pathway in the cross member.

There were a number of problems associated with the original piston assembly. The following paragraphs describe the design modifications which were incorporated to remedy those problems.

During the initial stages of development testing, failure of the crown gasket and bolts occurred. The crown asbestos gasket was an unsatisfactory choice, as it suffered from thickness reduction with steam load, and was unprotected against steam erosion. In order to correct this problem an Inconel 'C' ring was used to seal the cavity's periphery with the bolt head also acting as a seal. This arrangement worked well, sealing and reducing the heat flow. The heat transfer from crown to skirt remained the same as with the asbestos gasket, because the groove for the ring increased the total air gap between the crown and the piston's body. Although the seal modification worked well, the static loading on the crown bolts was excessive under certain conditions. The components of forces acting on the bolt are listed below:

1. Installation load to compress the seal.
2. Tensile load, due to differential expansion between the bolt and the aluminum skirt, which was particularly high at engine shut down heat soak.

3. Bending load, due to differential radial expansion between the alloy steel crown and the aluminum, which was also highest at a heat soak.

The above forces could be best accommodated by long flexible bolts from beneath the crown, but the palm bearing configuration eliminated the possibility of this arrangement as there was insufficient space for a bolting pattern. The bolting arrangement through the crown was, therefore, retained, but the following changes were made:

1. The alloy steel bolts (thermal expansion coefficient = 6.5×10^{-6} in/in/ $^{\circ}$ F) were replaced by stainless steel bolts (cold worked AISI-304) with a high coefficient of thermal expansion (= 10.4×10^{-6} in/in/ $^{\circ}$ F).
2. The bolt was necked and increased in length, to provide a greater strain length and to reduce bending stresses. Spherical seats below the bolt head were considered, but the corrosive environment (steam), together with size limitations, made this approach unworkable. A "location diameter" below the head was used to minimize the installation bending loads.
3. The number of bolts was increased from four to eight to provide a more uniform distribution of load on the seal and to avoid separation of the sealing surfaces between the bolts.

With the bolt modifications completed, a number of changes were made to the piston to resolve a scuffing problem and to improve the palm bearing design. The modifications are reviewed in the following paragraphs and a discussion of the scuffing problem is examined in detail in the Ring and Liner section.

To alleviate the problem of rings and skirt scuffing experienced initially with this piston, the top three compression rings were reduced to two for improved top ring lubrication. The skirt surface finish was changed from 60 to 200/250 rms and graphite coated to eliminate potential scuffing during "running-in". The skirt profile was determined by measuring the piston and liner temperatures at several operating conditions, and was subsequently machined to normal engine practice of tapering and barreling. These changes helped to alleviate the scuffing problem. To reduce the piston's cost and weight the ring at the bottom of the skirt, which seals exhaust steam from crankcase, was incorporated into the oil scaper ring.

With the scuffing problem resolved, tests at high bmep loads established the need to increase oil flow through the bearing and to reduce cross member flexing. This was achieved by extending the oil grooves to the end of the bearing and ribbing the cross member to the crown.

Testing of the modified piston successfully demonstrated 270 bmep (two stroke operation). Although the piston as described has proven to be a successful steam piston, due to its complexity it is not cost competitive with automotive pistons.

Crosshead Piston - To avoid the problems and cost associated with the bolted crown concept, an entirely new piston was designed, utilizing the "crosshead" principle, employed at the Detroit Diesel Allison Division of General Motors in their two cycle engines. The design retained the palm bearing scheme of the previous piston, but was otherwise fundamentally different. It consisted of two parts: a cast iron crown, which carries the compression rings and the palm bearing, and a separate skirt. Both parts are held in place by the wrist pin. A photograph of the piston assembly is shown in Figure 3.7 and a cross section in Figure 3.8. The advantages of the crosshead design in comparison to the previous one can be summarized as follows:

1. The two parts of the piston, the crown and the skirt, are held together by the wrist pin - a reliable and inexpensive arrangement. (Hence the problem of crown bolt failure is eliminated.)
2. The skirt is free from distortion caused by pressure and heat loads imposed on the crown. The skirt runs cooler. These factors contribute to easier control of the running clearances, and to good bedding of the skirt.
3. Ring and ring groove wear is reduced as a result of less piston crown motion.

TWO PIECE PISTON
TEMPERATURE ($^{\circ}$ F) AT 21 HORSEPOWER
PER CYLINDER
THRUST SIDE

FIGURE 3.8

The piston crown which carries the compression rings is in contact with the skirt through the wrist pin, and the skirt oil sealing ring only. There is no direct mechanical contact between the two parts under any operating conditions. Thus, each can react independently to normal stresses developed during engine operation. The crown transmits the vertical force, resulting from steam pressure, directly to the palm bearing bushing. The piston skirt absorbs all the side to side load from the connecting rod, and because it is free from steam thermal or mechanical loads, distortion is virtually eliminated. Because the skirt is not fixed rigidly to the crown, it is possible for the crown to remain stabilized in one position relative to the linear and, therefore, provide a stable platform for the compression rings, reducing motion and thus minimizing ring groove wear. The small skirt-to-linear clearance made possible by the relatively cool and distortion free skirt, minimizes the horizontal motion of the wrist pin center and therefore, the change in geometry of the ring to liner contact is minimal. The crown ring belt is uniformly supported, and configured for even distribution of heat and mechanical loads. The result is potentially long ring life and small steam blow-by regardless of the engine operating temperature.

The piston design allows for cooling the ring belt should it become necessary, with anticipated increase in steam temperature and pressure. In that case, all the oil supplied to the wrist pin bearing through the connecting

rod would flow along the palm bearing to the enclosed area under the crown, which is sealed from the outside by the sealing ring, and by the close crown-to-skirt fit in the wrist pin section. The oil would drain back to the crankcase through holes at certain height above the bottom, thus maintaining an optimum oil level in the cavity for effective "cocktail shaker" cooling. Passing the oil through the bearing enables it to cool both the bearing and the ring belt, and as the oil would be delivered to the outside area of the crown first, improved cooling of the rings can be achieved without extracting heat from the center of the crown. This is an important consideration since all of the heat lost through the piston represents a direct loss in efficiency.

A piston for a uniflow steam engine should have one important feature with respect to heat dissipation. Namely, in addition to heat insulation, the surface in contact with the steam should have "zero heat capacity" ideally. There are two periods during a cycle when the heat transfer coefficient and the temperature differential between the piston and the steam is high; one is the admission period, the other the blow-down period. During the admission period the top of the piston heats up, the absorbed Btus being lost for useful work. During the exhaust process the cooled steam flushes the surface of the piston, removing some of the stored Btus which have to be dissipated by the condenser unit. Calculations have shown that some one-third of the piston heat losses could be eliminated by applying a thermally protective coating to the top of the piston.

though none of the coatings investigated fulfilled the zero heat capacity requirement, zirconium oxide was chosen as the most promising candidate. The coating was applied in one case not only to the top but also to the side of the crown down to the first ring. A limited amount of testing did not reveal any bonding problems between the 0.035 in thick zirconium oxide coating and the cast iron and, as expected with a relatively thin coating, only an insignificant improvement in brake efficiency was detected.

In addition to the vertical load on the crown of the crosshead piston there is a lateral one resulting from friction in the wrist pin bearing. This forms a couple that tends to swing the crown from one side of the liner to the other around the wrist pin axis. In the large diameter highly-loaded bearing of a steam engine, the resulting force can be quite high and a back-up wavy spring behind the second compression ring was originally proposed, to center the crown in the liner. The tests done without it showed that the crown stabilizes and runs tilted to the nonthrust side. This occurs in correspondence with the relative magnitude of forces generated by the friction between the wrist pin and its bushing in relation to the friction between the rings and grooves. At top dead center, the steam pressure on the rings holds the crown in place but at BDC, when the steam pressure is low, the friction in the bearing overcomes the ring groove friction and tilts the crown, touching the liner. Because the crown remains in this position, the complication of a back-up spring was not necessary. It was

observed, however, that the wear of liners at TDC due to ring load was clearly larger on the thrust side. Should it become a problem a "half way" back-up spring which pushes the crown permanently toward the thrust side could be adopted. Thus the force resulting from the bearing friction in TDC will be carried to the liner directly by the well-lubricated and relatively cool lowest land of the crown instead of through the semilubricated piston rings.

Calculations of the mechanical and thermal stresses revealed that to comply with the minimal thickness of the castings as required by the foundry, most sections were stronger than necessary. The only critical stress concentration was found in the root of the first ring groove. It was corrected by an enlarged radius inside the crown and by closer tolerances of the groove depth. Detailed stress analysis of the improved version using the method of finite elements confirmed the soundness of the design.

Two different assemblies of the piston were designed and tested. One utilized a nodular cast iron (80-60-03) crown and skirt and the other had an identical crown but aluminum alloy (SAE 321) skirt. The weight of the cast iron piston (3 lb, 11 oz) was higher than that of the bolted crown piston (2 lb, 15 oz), while the light alloy skirt version was lighter (2 lb, 12 oz). Both assemblies were tested and developed.

The running profile of the aluminum skirt was not affected by operating temperatures so that a simple straight cylindrical grinding of that skirt was possible. The cast iron skirt, on the other hand, needed an 0.001 in barreling

of the upper section to obtain satisfactory bedding-in. The reason for the difference in the required skirt profile lies in the different thermal conductivities of the respective materials. To insure easier running-in, the light alloy skirt was graphite coated and the cast iron one was tin plated (for rust protection also). The initial 0.0007 in to 0.0015 in thickness tolerance of the tin coat proved unsuitable. The coating peeled off on several occasions and was very sensitive to overheating. The cure was to limit the thickness to less than 0.0005 in.

Both the aluminum as well as the cast iron versions of the pistons ran satisfactorily for hundreds of hours.

3.6 VALVE GUIDES AND SEALS

Valve guide wears on the single cylinder was excessive due to a slight valve to pushrod misalignment. This misalignment was corrected on subsequent builds and the guide wear was reduced significantly. Main guides are of Mechanite and inlet valves are made by machining down spark ignition engine exhaust valves.

In order to minimize steam leakage down the valve guides, seals were fitted to the valve stems. Stainless steel compression rings were used on a local tungsten carbide guide. They are recommended for use only up to 755°K (900°F). Above this temperature the material oxidizes. The seals were, therefore, located at the coolest location, at the bottom of the guide housing. During an engine start, condensing steam was normally visible escaping from the guide area until the engine was warm. With the seals, however, steam leakage was eliminated at start-up.

On disassembly of the valve, the seals and the tungsten guides were in perfect shape. Further tests were successfully conducted to evaluate the rings and guides at the top of the guide housing in order to keep the guide cooler, and to eliminate any possibility of the valve sticking due to foreign particles being trapped in the guide by the seals. The final valve guide and seal design is shown in Figure 3.8a.

VALVE STEM SEAL ARRANGEMENT

FIGURE 3.8A

3.7 OIL SEPARATOR

A small amount of lubricating oil will always be present in the exhaust steam as long as fluid lubrication is used in the expander. The lubricant likely to mix with the steam will come from three sources as follows:

1. The oil which passes by the piston rings.
2. The oil scraped off the piston skirt and ring area as the piston moves by the exhaust ports.
3. The oil washed off the cylinder walls by blowby steam.

The first two sources are, within the limit of observation, completely independent of the load on the expander and are dependent of the speed of the expander. Measurements made by others on internal combustion engines have shown that oil consumption is very nearly directly proportional to the total number of engine revolutions.

The third source will vary according to the sealing of piston skirt and rings. Initially the blow-by with a new expander may be high. However, once the rings are properly worn-in the amount of oil which is washed from the walls by blow-by will be reduced.

The amount of lubricant which will get into the exhaust in this way will only be slightly dependent on the load of the engine, in any case, the amount should be small compared with the other sources.

The cylinders of industrial reciprocating steam engines are lubricated with 50 to 200 ppm of oil which is purposely sprayed into the incoming steam. Only a small portion of this oil reaches the cylinder walls, suggesting that the amount of oil sufficient for adequate lubrication of the cylinder walls is much smaller than the total quantity added.

The performance of the condenser is not significantly lowered when quantities of oil as high as 150 ppm are contained in the steam. This indicates that, as far as condenser capacity is concerned, a fully loaded engine in good mechanical condition could run without oil-separation.

As pointed out earlier, the amount of oil within the exhaust steam depends substantially on engine speed, once the engine is "run-in". Any amount of oil greater than the quantity which is dissolvable will tend to form a layer on the condenser hotwell and ultimately will get to the boiler if not separated.

The steam passing the piston rings which are located below the exhaust ports, will get into the crankcase in the form of small droplets or wet steam. During the start-up period the blow-by into the crankcase is substantial, such that water collects on the bottom of the crankcase sump. The amount of water increases until the crankcase oil gets hot enough to evaporate the water, unless the water is drained. The expander will be operated at a crankcase oil temperature of approximately 394°K (250°F) which provides sufficient protection against continued accumu-

lation of water within the crankcase. For the case of frequent start-up concurrent with relatively cold, low-loaded engine runs, separate oil-water separation is required.

Studies of condensate water and possible boiler effects were conducted during the early phases of expander testing.

Lube oil water separation considerations included the following.

1. Removal of "blow-by" steam which passes by piston rings into the crankcase and elimination of accumulated condensed crankcase "blow-by" during repeated cold engine start-ups.
2. Removal of lube oil entrained in expander exhaust steam to keep oil concentration in the condenser and boiler low.
3. Prevention of the accumulation of oil skim in the condenser sump to ensure proper functioning of feed-pump and boiler.

The L-4 test expander was used to analyze the exhaust steam and the crankcase blow-by. These analyses were made over the range of expander speeds and cut-off angles equivalent to vehicular cruising speeds. The results are given in Table 3-IV.

TABLE 3.IV

MEASUREMENTS OF OIL ENTRAINMENT IN STEAM AND
CRANKCASE BLOW-BY OF L-4 EXPANDER

	EXHAUST STEAM	CRANKCASE BLOW-BY
Water { 1. Average	5.0×10^{-2} kg/sec(1)	1.0×10^{-3} kg/sec
2. Range	0 - 0.15 kg/sec	$0 - 2.5 \times 10^{-3}$ kg/sec
3. Idling	6.3×10^{-3} kg/sec	----
4. Cold engine start up	----	0.95 kg(2)
Oil	4.7×10^{-5} kg/sec	To be Determined
Temperature	$366^{\circ} - 422^{\circ}$ K	$366^{\circ} - 411^{\circ}$ K
Pressure	0 - 274 kPa	0 - 108 kPa

NOTES: (1) For 90% of vehicular engine operational power, steam flow is equal to or less than this rate. (2) During initial 5 to 10 minutes of cold engine start-up

A centrifugal liquid-gas separator for the exhaust steam of the single cylinder expander was studied. The device, 0.127 m DIA x 0.432 m long, was the smallest, most economical commercially available separator. Its efficiency was determined to be less than 50% during average operating conditions of the single cylinder expander.

A complete oil separation scheme was never fully developed by a basic scheme was selected and various parts were tested. The separation scheme is shown in Figure 3.9, and was selected based on the following criteria:

1. Size and weight of equipment
2. Energy requirement
3. Control
4. Maintenance

The required separations are provided by the following three methods:

- A. Separation of Condensed Blow-By and Lube Oil in Crankcase.

It was observed that condensed blow-by in the crankcase amounted to as much as 0.95 kg when the engine was cold. Repeated cold engine start-ups were expected to cause sufficient accumulation of condensed blow-by in the crankcase to be detrimental to engine lubrication. However, after 5 to 10 minutes of engine operation, the crankcase becomes hot enough to evaporate the crankcase blow-by. Therefore, only during the initial 5 to 10 minutes of engine operation, separation of condensed blow-by and lube oil is necessary.

It was proposed to accelerate the boiler off the condensed crankcase blow-by during the initial minutes of engine operation. The most practical source was boiler steam.

B. Separation of Oil Spray from Blow-by Steam
in Crankcase Vent Discharge

As the crankcase temperature reaches the boiling point of water, the condensed blow-by is boiled off through the crankcase vent, carrying with it some oil droplets. A phase change mechanism (PCM) separator was designed and tested for recovering this oil spray. The PCM separator is an engine-driven device that removes oil droplets from steam by way of spinning the crankcase vent discharge when it passes through the narrow passage between the PCM drum and its cover. The measured efficiency of the device was virtually 100%.

C. Separation of Entrained Lube Oil From Expander
Exhaust Steam

Evaluations revealed that all commercially available liquid-gas separators have one or more of the following shortcomings:

- A. Low efficiencies
- B. Sizes too bulky to fit in an automobile
- C. Maintenance problems: Knitted wire mesh oil mist eliminators were prone to the formation of a semi-solid material resembling mayonnaise.

PROPOSED LUBE OIL - WATER SEPARATION SYSTEM

FIGURE 3.9

D. Any liquid-gas separator has the further problem of an accumulation of a layer of lube oil in the condenser sump. The build-up of oil skim formed by gravity separation must be eliminated, and the cost of commercial skimming devices was found to be prohibitive.

Since the maximum oil entrainment is 0.17 kg/hr, (0.375 lb/hr) and the average steam rate is 181.4 kg/hr, (400 lb/hr), the entrained oil concentration is 940 ppm during most of the engine operating time. The condenser functioned effectively under the presence of this oil concentration.

A liquid-liquid separator, however, was required in order to reduce oil concentration in the condenser sump discharge for proper boiler tube functioning. Commercially available liquid-liquid separators come in two basic types: centrifugal and coalescing. They are slightly bulky to fit in an automobile. Redesigning of existing commercial liquid-liquid separators was investigated.

A parallel program of oil-water coalescing separation was undertaken by Exxon to investigate the effectiveness of a separator unit charged with different oil coalescing agents such as Polystyrenes, Polypropylenes, Polymethacrylates, Polyamides and Polyfluorocarbons. The influence of mass flow rates, pressure drops across the coalescer and direction of flow upon the effectiveness of separation was determined and found to be quite bulky in application.

Subsequent to this program an exhaust steam separator consisting of a combination centrifuge for high power and gauze coalescer for low flows was constructed and tested under a U.S. Bureau of Mines contract and appeared to have better than 90 per cent effectiveness.

3.8 EXPANDER PERFORMANCE

Computer analysis was used to determine the effect on efficiency and power output of various options in expander design. The alternative considered included: normal valve arrangement (cut-off valve downstream of admission valve) with a short duration cut-off cam, throttling, reversed valve (cut-off valve upstream of admission valve) arrangement, and exhaust port size and placement. The studies indicated that a reversed valve expander with revised valve timing offers the best potential for improvement of the efficiency, particularly at low power levels, but that the minimum power without throttling might be too high for idle. The original expander builds incorporated the normal valve arrangement but was later abandoned because of the poor light load efficiency.

In addition, the exhaust ports were decreased in size, giving improvement in ring and cylinder wall lubrication, with no loss of power or efficiency. A computer program was developed, which accounted for intake and exhaust flow processes, cam and valve opening geometries, effect of inter-valve volume, provision for evaluating reversed intake valves, variable size and placement of exhaust ports, and throttling inlet steam to regulate power output. The following studies constituted the bulk of the analysis work.

3.8.1 EFFECT OF IVV AND CUT-OFF CAM TIMING

The first improvements were tried on the normal valve arrangement and involved finding a valve timing scheme to improve the cycle efficiency, without

reducing the maximum power output. For a constant value of Indicated Horsepower (IHP) efficiency improves with decreasing Inter-valve Volume (IVV) size. IVV steam is dumped into the cylinder when the cut-off valve opens, which occurs when the piston is on the upstroke. Work is done to recompress this additional steam, thus reducing the net work of the cycle and the thermal efficiency. Decreasing the IVV size minimizes this effect and improves efficiency. Similarly, by delaying the blowdown of steam from the IVV, recompression work is reduced. This can be accomplished by using a shorter duration cut-off cam since, for a given cut-off angle, it will cause the cut-off valve to open later on the upstroke. A 1.1 rad (60°) cut-off cam provided the shortest practical duration, because .96 rad (55°) cut-off was necessary to obtain maximum power, and .09 rad (5°) of admission advance was required.

3.8.2 THROTTLING WITH NORMAL SERIES VALVES

The effect of IVV blowdown was largest at small cut-off angles, where IVV steam is admitted to the cylinder early on the upstroke, resulting in large recompression work. This accounted for part of the trend of decreasing efficiency at low IHP (small cut-off angles).

Since early cut-off was found to have a detrimental effect on efficiency with the normal series valves, the possibility of obtaining low power by throttling the inlet steam, rather than reducing the cut-off angle, was studied. This involved holding the cut-off angle to

.44 rad (25°) and reducing the manifold pressure of the inlet steam. Due to the delay of IVV blowdown, it was more efficient to obtain low power outputs at .44 rad (25°) cut-off and a reduced manifold pressure, than at 6.9 MPa (1000 psia) manifold pressure and smaller cut-off angles. The IHP's were approximately the same, but the steam flow was much lower when throttled to 5.5 MPa (800 psia) at .44 rad (25°) cut-off angle than at 6.9 MPa (1000 psia) at .25 rad (20°) cut-off, and consequently the efficiency was better with throttling. These results were confirmed experimentally in both L-4 and Single Cylinder Expander tests.

3.8.3 REVERSED VALVES

The largest improvement in expander efficiency was made by reversing the positions of the admission and cut-off valves, that is by placing the admission valve nearest the cylinder and the cut-off valve nearest the manifold. This valve arrangement eliminates blowdown of steam from the IVV to the cylinder, thus minimizing the recompression work. In order to maximize efficiency with reversed valves, different cam timing must be used, although the available range of cut-off angles 0 - .96 rad ($0-55^{\circ}$) and admission lead angle .09 rad (5°) remain the same. It was most desirable to close the admission valve as soon as possible after the cut-off valve, to obtain maximum expansion ratio of the steam within the cylinder, and the best efficiency. For this reason, a 1.1 rad (60°) admission cam was chosen to provide .96 rad (55°) cut-off and .09 rad (5°) Lead. Cut-off cam duration was not critical, and was taken to be 2.1 rad (120°). When the cut-off valve opens, steam is admitted only to the IVV, and no additional recom-

pression work is done because there is no IVV blowdown. The reversed valve arrangement produced considerably better efficiency at low horsepower levels than did the normal series valve arrangement, or the normal arrangement with throttling. A comparison of normal and reversed valve indicator diagrams is given in Figure 3.10 showing the additional recompression work due to IVV blowdown with the normal valve arrangement. Finally, computer cases were run with throttling applied to a reversed valve expander. This resulted in reduced efficiency, which led to the conclusion that throttling is an inefficient means of obtaining low power output. It improved the normal valve arrangement expander only because with that valve set-up, throttling imposes a smaller penalty than does the blowdown loss associated with small cut-off angles.

3.8.4 EXHAUST PORT SIZING AND PLACEMENT

A reduction of the exhaust port radius from .61 cm to .48 cm (.24" to .19") was made to improve ring and cylinder wall lubrication. Accordingly, computer analysis was done to determine the effect of smaller exhaust ports on efficiency and maximum power output. In addition, it was felt that the efficiency might be improved slightly by moving the exhaust ports downwards from the 2.36 rad (135°) position, which would have the effect of increasing the work done during expansion. The smaller port actually improved the maximum power slightly by restricting the exhaust blowdown, more expansion and hence more power. Investigation showed that as the ports were moved downward, the exhaust could not expand down to .14 MPa (20 psia);

FIGURE 3.10

COMPARISON OF NORMAL AND REVERSED VALVE ARRANGEMENTS AT EQUAL IHP

1500 RPM IVV = .80 IN³

consequently, this option was not considered. A reduction of the exhaust port radius to .48 cm (.9") was made for L-4 cylinder liners. This change provided improved ring and cylinder wall lubrication and slightly improved efficiency.

Computer predictions indicated that the reversed valve modification would increase efficiency at low cut-off angles, by eliminating blowdown of steam from the IVV into the cylinder. With this valve arrangement a short admission cam period provided the best efficiency, while the cut-off cam period had little or no effect. For a particular brake horsepower level and expander speed, the reversed valve arrangement required a smaller amount of steam flow, and hence was more efficient.

Considerable attention was given to making the reversed valve expander self-starting; however, no simple method was found. Valve timing for best efficiency will allow self-starting only at maximum cut-off or with a complex system of starting valves. Consequently, a starter was required.

A reversed valve expander with 1.3 rad (75°) admission cam and 2.1 rad (120°) cut-off cam was studied with the aid of computer techniques. This configuration could readily be set up on the single cylinder test engine with existing camshafts and cylinder head. Efficiencies were not as good as with a 1.1 rad (60°) admission cam; however, this test was run, and the measured results verified the computer predictions for reversed valves.

Analysis of the heat flows in the expander gave preliminary results which are discussed here. Plotting the measured single cylinder expander liner temperatures as a function of distance from the top of the liner indicates a temperature gradient of $616/\text{cm}$ ($650^{\circ}\text{F}/\text{in}$) at the top of the liner, which corresponds to a heat flow of 2344 W (8000 BTU/hr) into the top of each liner, or a steam flow of 41.8 kg/hr (92 lb/hr) for 4 cylinders. These solutions were integrated to obtain heat flows into and out of the cylinder liner. They indicate that 2184 W (7450 BTU/hr) flow into the top of the liner, 2125 W (7250 BTU/hr) are conducted out into the air through the side of the liner, while 59 W (200 BTU/hr) are conducted away through the bottom of the liner. A similar analysis of measured piston temperatures indicates the 806 W (2750 BTU/hr) per piston are conducted away from each cylinder, corresponding to about 14.1 kg/hr (31 lb/hr) steam flow for 4 cylinders. Thus, heat losses through the liners and pistons may account for as much as 54.4 kg/hr (120 lb/hr) of the difference between measured and predicted steam flows, according to these preliminary results. A different and more rigorous approach to heat transfer loss evaluation was therefore undertaken.

In order to provide a basis for advanced expander design, the effect on the expander efficiency of such variables as IVV, clearance volume, condenser pressure, secondary exhaust, and recompression work was studied. One conclusion was that a large clearance volume and recompression work account for lower expander efficiency at early cut-offs. Thus secondary exhaust seems a desirable feature if an appropriate expander can be designed, that is, one in which the exhaust steam does not cool the head and upper

cylinder walls. Reduced clearance volume is also beneficial as long as the cylinder volume at late cut-off permits enough steam to enter so that the necessary power output is obtained and as long as the cylinder pressure is low enough to remain below the cylinder exhaust pressure at early cut-off. Obviously, it is much easier to reduce clearance volume in an expander with secondary exhaust, where the recompression pressures can be controlled.

3.8.5 EXPANDER THERMAL ANALYSIS

The objective of the thermal analysis effort was to improve expander efficiency by minimizing heat losses in the expander via the following mechanisms:

1. The to and fro heat flow from liner to rings and rings to liner.
2. The to and fro heat flow from steam to crown and crown to steam.
3. The heat flow through the crown to the piston body.
4. The heat flow across the head gasket.

Briefly, the procedure followed in this effort included:

1. Searching the literature to establish the heat transfer coefficient from the steam to piston crown and to determine coefficient variation during a cycle.

2. Searching the literature to establish the thermal conductance between the compression ring and liner.

3. Application of the above to lumped parameter network models of the piston crown and compression rings.

4. Computation of temperatures and heat flows over one cycle, using forward differencing numerical techniques.

5. Comparison of computed results with previously reported experimental Templug data.

The analysis provided an estimate of the heat losses as well as a tool which can be used to predict heat loss reductions due to material changes, coatings and design changes.

Below is a summary of work done on the compression rings and the piston crown.

3.8.6 HEAT TRANSFER VIA COMPRESSION RINGS

The compression ring was modeled as a network of eleven lumped thermal masses connected to a boundary node representing the liner. The twelve nodes were series connected by thermal conductors. The thermal mass (product of the mass and the specific heat) of each of the nine interior nodes is equal to 10% of the total volume of the ring. The thermal mass of each of the two exterior nodes is half that

of an interior node. The ten internal conductances (the product of the thermal conductivity and the cross-sectional area normal to the heat flow path, divided by the distance between nodes) all share the same numerical value, neglecting the variation in ring diameter.

The thermal conductance between the ring and the liner was obtained from:

1. Conductance values reported in the literature.
2. Conductance data on highly loaded (thermal grease filled) static interfaces.
3. Calculations based on oil film thickness between compression ring and liner.

The liner temperature distribution observed by the ring from its top dead center position to its bottom dead center position was taken from single cylinder liner thermocouple data. The ring position was plotted versus crankshaft angle giving the temperature versus time function used in the calculations.

Using the ring model described, a 45°K (350°F) sawtooth driving function at the boundary, and an assumed expander speed of 25 Hz (1500 RPM), the ring temperatures and heat flows were calculated over one cycle. The result was a 300°K (80°F) peak-to-peak cycling of the ring outside surface temperature, and a heat flow from liner to ring and back of 615.5 W (2100 BTU/hr). Including the contribution of the second compression ring yields a total heat flow via the two compression rings of 1143 W (3900 BTU/hr)/ Cylinder.

Additional efforts in this area examined effects of coatings and configuration changes on the heat flow via this mechanism.

3.8.7 HEAT TRANSFER VIA PISTON CROWN

A thirty-degree segment of the piston crown was modeled as a lumped parameter network of 44 nodes inter-connected by a conductance network. The four nodes modeling the top surface of the crown segment were connected to a boundary node representing the steam. These conductances were put into the computer model so that they, along with the steam temperature, could be continually varied during the cycle. One crown node on the bottom side was connected to a boundary node representing the piston. This node was set at 566°K (560°F), a temperature obtained from Templug data.

The variation in heat transfer coefficient versus crankshaft angle was computed using Annand's formula shown below:

$$h_c = a \frac{k}{D} (Re)^{.7}$$

where

k = Thermal conductivity of steam

D = Piston diameter

a = Proportionally constant equal to:
0.38 for non-turbulent conditions,
0.80 for turbulent conditions

R_e = Reynolds No.

The method of calculations differed during four discrete intervals within the cycle:

1. Steam Admission - during the admission period, half the inlet port steam velocity was used to compute the Reynolds number. The value of "a" was taken as 0.8 to account for the high turbulence and high impingement velocity of the admission steam.

2. Expansion - during the expansion stroke, the mean piston velocity was used to compute the Reynolds number. The high turbulence assumption was retained, thus "a" remained equal to 0.8.

3. Exhaust - during the exhaust period $286^{\circ}\text{K} - 355^{\circ}\text{K}$ ($55 - 180^{\circ}\text{F}$), the high turbulence assumption was retained and half the downward steam velocity in the cylinder was used in computing the Reynolds number.

4. Compression - finally, during the compression period, low turbulence was assumed, thus "a" = 0.38 and the mean piston velocity was used to compute the Reynolds number.

Figure 3.11 shows the variation of the heat transfer coefficient (h_c) versus time for the .87 rad (50°) cut-off case.

CONVECTIVE HEAT TRANSFER COEFFICIENT VERSUS CRANK ANGLE

FIGURE 3.11

An average value of h_c and steam temperature was used to compute the steady-state crown temperature and heat flows. These temperatures compared closely with the Templug data from the single cylinder expander and were used as the starting temperatures for the periodic solution. Solutions showed heat flows of the following magnitudes:

1. Heat flow from steam to crown =
257.8 W/Piston (3000 BTU/Hour)
2. Heat flow from crown to exhausting steam =
239.1 W/Piston (1000 BTU/Hour)
3. Heat flow through the crown to the
piston body =
586.2 W/Piston (2000 BTU/Hour)

Expander thermodynamics for a 1.3 rad (75°) cut-off cam/reversed valve were studied to determine the suitability of a short duration cut-off cam for reversed valve operation. (Such a cam had already been built for use with normal valve arrangement.) The objective in using this cam was to save the time and expense of fabricating 1.66 rad (95°) or 2.1 rad (120°) cut-off cam, as had been previously proposed. The primary concern with this cam was that the admission valve might be forced off its seat during the compression stroke, due to later cut-off valve opening angles than with the 1.66 rad (95°) or 2.1 rad (120°) cams. This could occur if cylinder pressure were to exceed IVV pressure before cut-off valve opening on the compression stroke. Computer results for .14 MPa (20 psia) condenser pressure indicated that this condition would never occur and that even for condenser pressures to .25 MPa (36 psia), the shortened duration cut-off cam was satisfactory from the standpoint of admission valve lifting.

Computer runs were made to study the effect of IVV size with reversed valves. These were used to substantiate single cylinder test results in which measured efficiency improvements were obtained due to reversed valves, reduced IVV, and reduced valve guide leakage.

3.8.8 FARNBORO INDICATOR DIAGRAM ANALYSIS

Indicator diagrams were obtained from the single cylinder expander for the range of speeds and cut-off angles from 25 Hz (1500 RPM) and .35 rad (20°) cut-off, to 37.5 Hz (2250 RPM) and .785 rad (45°) cut-off. From the analysis, the following conclusions were reached:

1. Friction power, as determined from the Farnboro test data, is reasonably well predicted by the computer friction power model.

2. For a given speed, friction appears to be a function of cut-off angle.

3. Examination of discontinuities on the P- θ indicator diagrams indicates steam blow-by occurs when the piston is near to top dead center (TDC) position.

Data from thirty-three Farnboro indicator diagrams was reduced and used to establish the friction power (FP) versus speed relation. Coordinate grids were plotted on each of the indicator diagrams, and the pressure versus crankshaft rotation angles were converted to expansion volume, and the data replotted in a P-V format. Each of these P-V diagrams was then integrated to determine the indicated power (IP). The difference between this and the brake power (BP) at each operating condition is the friction power (FP). The graph of friction power versus speed is shown in Figure 3.12 along with the friction power model used in the cycle analysis program.

Also shown in Figure 3.12 are the maximum and minimum friction power data points for each speed. There are two sources of this scatter:

1. Accumulated error in data acquisition
2. A functional relationship between the friction power, FP, and cut-off angle, or BMEP.

First, errors associated with the friction power data can become significant since friction power is the difference between the two larger numbers (BP and IP). Errors associated with either the IP and BP are increased by a factor of the ratios of the IP to the FP.

Secondly, there appears to be a functional relation between the friction power and cut-off (or BMEP). This is shown in Figure 3.13. At cut-offs of .52

FRICTION POWER vs SPEED
 - AVERAGED DATA -
 SINGLE CYLINDER TEST DATA
 CUTOFF RANGE: 20° TO 45°

FIGURE 3.12

rad (30°) and .785 rad (45°), FP increased smoothly as a function of speed. For other cut-offs, only random data points fell exactly where they might be expected. Determination of the analytical relation between friction power and cut-off was thus beyond the accuracy of the test data.

Examination of the indicator diagrams revealed steam blow-by during the first .52 rad (30°) of shaft rotation after TDC. An example of a typical P- θ diagram is shown in Figure 3.14. The discontinuity in the P- θ diagram, Figure 3.14 at $\theta = .38$ rad (22°), can be easily discerned on 85% of the diagrams taken. The discontinuity was always located between .38 rad (22°) and .51 rad (29°) after TDC. At this angle of shaft rotation, the top piston ring was on the .71 cm (.28 in) width band on the liner where the greatest wear was found. The eccentricity of the liner in this location together with the minimal oil film between the ring and the liner would both contribute to poor top ring sealing. Additional steam would be admitted to make up for that loss. Thus, the power would not be affected. This mechanism combined with heat transfer loss accounted for low predictions of BSFC that were given by the cycle program. The program was subsequently updated.

Further expander analytical work focused on some of the many Farnboro (P- θ) diagrams generated during single cylinder testing. These were used to improve computer predictions of expander performance. A heat loss computation

FRICTION POWER vs SPEED AND CUTOFF ANGLE

FIGURE 3.13.

TYPICAL FARNBORO INDICATOR DIAGRAM SHOWING PRESSURE vs θ
DISCONTINUITY DURING EXPANSION PROCESS

ROTATION DIRECTION $\rightarrow$
FIGURE 3.14

was included in the expander program, which calculates cylinder pressures and temperatures as a function of crank angle, in the presence of a heat loss which must be specified as a function of crank angle.

In the past, computed efficiencies and steam consumption differed considerably from both single cylinder and L-4 measurements. This poor water flow agreement had always been particularly true at low power conditions where heat losses apparently amounted to a larger fraction of the heat delivered to the expander. Consequently, a heat loss model was incorporated into the SES expander program and compared to Farnboro diagram.

The measured single cylinder performance at 20.2 kW (27.2 BHP), .73 rad (42°) cut-off and 25.3 Hz (1515 RPM) produced a water flow about 132.5 kg/hr (292 lb/hr), for a specific steam consumption of 653 kg/kW-hr (10.74 lb/hp-hr). Computed values with a 4083 W (17,000 BTU/hr) heat loss produced a water flow of 112.0 kg/hr (247 lb/hr), corresponding to a specific steam consumption of 5.52 kg/kW-hr (9.09 lb/hp-hr). This compared with values of 100.7 kg/hr (222 lb/hr) and 4.97 kg/kW-hr (8.18 lb/hp-hr) that were previous estimates at this condition, assuming no heat losses. The computer model of the heat loss was updated based upon continued Farnboro work.

Analyses of the Farnboro diagrams revealed that the single cylinder expander exhibited lower recompression pressures (at TDC) than were predicted theoretically. This was attributed to:

1. Condenser pressure measurement error.
2. Difference in effective clearance volume.
3. Actual admission valve opening on the single cylinder expander later than expected due to mechanical reasons.
4. Heat loss during the admission process beginning at .17 to .26 rad (10° to 15° BTDC) - possibly due to the hot supply steam being cooled as it comes in contact with the cooler piston and liner.
5. Ring leakage.

The shape of the P- θ diagram could be made to agree fairly well with Farnboro data at .785 rad (45°) cut-off, 25 Hz (1500 RPM) (high water flow condition). This was accomplished by slightly increasing the clearance volume and admission valve lead used in the computer analysis. The shape of the computed P- θ diagram was found to be quite sensitive to admission valve lead angle, with this parameter having significant effect on the peak cylinder pressure and indicated horsepower. Measured P- θ diagram at .785 rad (45°) cut-off, 25 Hz (1500 RPM) was indicative of significantly later admission valve opening and larger clearance volume than expected. Admission valve lead appeared to be about .04 to .06 rad (2.5° to 3.5° BTDC), rather than the nominal .09 rad (5°). Initially, the clearance volume of the single cylinder was taken as 26.2 cm³ (1.6 in³) but additional volume due to the relief valve and Farnboro transducer brought this to about 31.1 cm³ (1.9 in³). Best computer agreement was obtained with CV = 36.6 cm³ (2.05 in³).

The largest effect of heat losses was noticeable at a condition of short cut-off, low power and speed, and low water flow. A comparison of computed P- θ diagrams at .26 rad (15°) cut-off, 8.3 Hz (500 RPM), with and without heat loss, is given in Figure 3.15, based on the revised heat loss model. The heat loss corrected the shape of the diagram to match the characteristics of the Farnboro data, where low recompression pressure and an inflection point near TDC were usually evident. For the computed cases shown in Figure 3.15, these characteristics were obtained by introducing a heat loss which begins as the admission valve is lifted from its seat at .17 rad (10° BTDC). By introducing the heat loss, power was reduced from 3.78 kW/CYL (5.07 BHP/CYL) to 3.45 kW/CYL (4.62 BHP/CYL), steam flow increases from 17.28 kg/hr/CYL (38.1 lb/hr/CYL) to 22.63 kg/hr/CYL (49.9 lb/hr/CYL), and specific steam consumption rose from 4.57 kg/kW-hr (7.521 lb/hp-hr) to 6.56 kg/kW-hr (10.794 lb/hp-hr). The current L-4 expander data showed a steam flow of about 24.95 kg/hr/CYL (55 lb/hr/CYL) at 3.45 kW/CYL (4.62 hp/CYL) and 8.3 Hz (500 RPM). Consequently, measured and predicted efficiencies were then within 10% agreement at this early cut-off condition. This neglected any leakage which when applied to the computer calculation with heat loss would tend to make the agreement even closer.

The final expander performance results for the inline 4 cylinder L-4 is shown in Figure 3.16.

COMPUTER CALCULATED PRESSURE VS. CRANK ANGLE
(WITH AND WITHOUT HEAT LOSS)

FIGURE 3.15

PPT PHASE II EXPANDER DATA

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FIGURE 3.16

4.0 STEAM GENERATOR

The preprototype steam generator went through several generations of design and evolution, starting with a gas fired Model No. 4 unit and ending with Model No. 8 that was used in the vehicle simulator. A summary of the steam generator specifications is given in Table 4.I; a photograph of the unit and its cross section are shown in Figures 4.1 and 4.2.

Combustion air enters through the center of the combustion chamber. Uncooled combustion products are drawn through the recirculation ports and mix with the incoming air to produce a final mixture temperature in the range $394 - 450^{\circ}\text{K}$ ($250-350^{\circ}\text{F}$). The flow of air is modulated by the air throttle valve. The fuel is sprayed against the direction of air-flow to provide good spatial distribution of the fuel aerosol prior to entering the blower.

The fuel-air mixture passes through the blower and then flows radially inward through a vaned diffuser. Final flow straightening is accomplished in a short honeycomb. By the time the mixture has reached the flameholder, the fuel is vaporized and has been quite well mixed going through the blower and diffuser. The flameholder is a perforated, sheet-metal, Meker-type of burner, the perforations being 0.84 mm (0.033 in) round holes representing approximately 18% open area. The combustion products flow through the heat exchanger into the exhaust duct.

MODEL NO. 8 VAPOR GENERATOR ON TEST STAND

FIGURE 4.1

CROSS-SECTION OF MODEL NO. 8 VAPOR GENERATOR

FIGURE 4-2

TABLE 4.1
MODEL 8 VAPOR GENERATOR SPECIFICATIONS

A. Design Point (Full Load)	<u>Model No. 8</u>
1. Fuel Flow, kg/hr	49
2. Steam Flow, kg/hr	635
3. Air Flow, kg/hr	835
4. Stack Temperature, °K	550
5. LHV Efficiency, %	88
B. Physical Characteristics	
1. Outside Diameter, cm	40.6
2. Heat Exchanger Weight, kg	18.6
3. Package Weight, kg	40.8
C. Heat Exchanger	
1. Number of Passes	7
2. Total Gas-Side Surface, m ²	7.0
3. Total Tubing Length, m	36.6
4. Design Point Effectiveness, %	90

An important feature of the combustor is the positioning of a bare coil approximately midway through the combustion chamber. The coil partially cools the combustion gases to a temperature in the range of 1700-1900°K, (2600-2900°F) effectively quenching the NO_x reaction, but permitting continued oxidation of CO and HC.

The burner/vapor generator operates in a fully modulated control mode to supply steam at a rate which matches the demand. The steam generator operates over a range of steam flows 27-550 kg/hr (60-1200 lb/hr). The fuel flow ranges from 1.8-45 kg/hr (4-100 lb/hr) as the blower speed changes from 33-100 Hz (2000-6000 RPM). The desired air-fuel ratio is maintained by positioning the air throttle valve as a nonlinear function of blower speed.

4.1 Liquid Fueled Burner Development

In order to expedite early design and development of the vapor generator, a gaseous-fueled burner was employed which was capable of high combustion intensity, with low draft loss and low emissions. As part of the automotive design program, SES was required to develop a liquid-fueled burner capable of similar performance. The burner requirements were as follows:

1. A firing rate of $.64 \times 10^6$ W (2.2×10^6 BTU/hr) (HHV), which is sufficient to produce the required $.5 \times 10^6$ W (1.7×10^6 BTU/hr) steam generator output at a boiler efficiency of 77%.

2. A turndown capability of 20:1, $.64$ to $.032 \times 10^6$ W (2.2×10^6 to $.11 \times 10^6$ BTU/hr), which matched the turndown capability of the steam generator.

3. A combustion intensity of 41×10^6 W/m³ (4×10^6 BTU/hr-ft³) in order to fit within the combustion chamber provided in the vapor generator, and in order to limit NO formation.

4. Emission of hydrocarbons, carbon monoxide, oxides of nitrogen and particulates below the EPA goals measured over the Federal Driving Cycle.

4.1.1 Fuel Preparation

During proof-of-principle tests it was demonstrated that vaporized fuel or sufficiently finely atomized fuel, when premixed with combustion air, could be used in SES's proprietary gas burner. The concept was to develop a means of clean and reliable fuel preparation and to incorporate those means in a compact burner, meeting the above requirements and suitable for interfacing with the steam generator.

The basic fuel preparation concept consisted of utilizing an air atomizing nozzle to form a fuel aerosol consisting of droplets of 10 to 50 microns in diameter. The aerosol was vaporized by mixing with combustion air which was heated to $394-450^\circ\text{K}$ ($250-350^\circ\text{F}$) by the addition of recirculated combustion products.

Careful attention was given to the design of the metering air valve. It was observed that with a simple butterfly valve, the emission characteristics were not satisfactory due to non-uniformities in the fuel distribution. This problem is illustrated in Figure 4.3.A.

Consequently, the butterfly valve was changed to an axisymmetric four-piece air valve actuated by a servo motor. A photograph of the valve inside the inlet duct is shown in Figure 4.4. The valve was operated successfully over the load range of the burner and provided satisfactory precision in achieving the desired air/fuel ratio control.

Figure 4.3.B shows the interaction between the fuel jet and the air jet generated by the axisymmetric four-piece air valve utilized in Model 5 S/N #2 burner. The flow is nominally symmetrical, but the impinging jet configuration is not necessarily stable enough, so that the air/fuel ratio is maintained circumferentially uniform. If there is slight misalignment of the nozzle and air jet, one side of the mixture flowing into the blower will be rich and the other will be lean.

The solution to the problem is shown in Figure 4.3.C. A perforated plate baffle is inserted between the fuel jet and air valve. The baffle stabilizes the flow geometry by eliminating the direct impingement of the two jets and washing out any slight misalignment or jet asymmetry. The perforations are included to minimize the wetting of the baffle and generation of large droplets. The configuration was tested and proven insensitive to air valve position and atomizing pressure.

FIGURE 4.3

SCHEMATIC OF THE INTERACTION BETWEEN THE FUEL JET AND AIR VALVE IN THE MIXING ZONE IMMEDIATELY UPSTREAM OF THE BLOWER

AXISYMMETRIC AIR THROTTLE VALVE IN TEST RIG
FIGURE 4-4

4.1.2. Combustion Air Blower

Early development of the liquid fuel burner was performed with a commercially available fan of the radial flow, backward curved type manufactured by Volkswagen for engine cooling. Although satisfactory for burner development, the VW blower was not well matched to the steam generator requirement in the vehicle. The head/flow characteristics were not correct, its peak efficiency was only about 65%, and it had a strong tendency to surge at low flow, highly throttled conditions. Moreover, its mass was unacceptably high (1.4 kg), limiting the rate of change of speed with the available drive torque.

Accordingly, a new blower was designed to match the requirements of the steam generator and overcome the deficiencies observed in the old unit. The diffuser was also redesigned to match the new contours of the blower and its discharge aerodynamics.

Table 4.II gives the blower specifications as manufactured. Figure 4.5 is a photograph of the new blower impeller. Figure 4.6 schematically shows the differences in air flow path between the original design and final design.

The new blower system was tested in a steam generator and its head/flow characteristics measured. Figure 4.7 shows the blower performance map. The peak total-to-total efficiency for the impeller was measured at 84%, and occurred at a slightly lower flow than design point. Additionally, the blower was observed to be surge free, operating at tight shut-off with no measurable oscillations in discharge pressure.

TABLE 4.II
BLOWER SPECIFICATIONS

1.	Design Point	- 18.4 m ³ /min (650 cfm) @ 422°K (300°F), 15 cm. (6 In) of Water Total Pressure Rise at 100 Hz (600 RPM)
2.	Flow Range	- 0.7 to 18.4 m ³ /min (26-650 cfm)
3.	Speed Range	- 25 to 130 Hz (1500-8000 RPM)
4.	Stall Margin	- 5 cm (2 In) Water Pressure Rise at 26 cfm Without Surge
5.	Inlet Diameter	- 12.7 cm (5 In)
6.	Discharge Diameter	- 22.9 cm (9 In)
7.	Discharge Blade Height	- 2.3 cm (0.90 In)
8.	Inlet Angle	- 61.5° to Radius
9.	Discharge Angle	- 45 ° to Radius
10.	Blade Number	- 18
11.	Construction	- Aluminum 6061-T6 Sheetmetal, 1.6 mm (.062 In) Thick, Furnace Brazed, Balanced, Hard Anodized
12.	Weight	- 0.45 kg (1 lb.)

FIGURE 4.5 - SES COMBUSTION AIR BLOWER

BURNER AERODYNAMICS

OLD DESIGN

NEW DESIGN

FIGURE 4.6

SES BLOWER MAP

MEASURED TOTAL HEAD RISE
VERSUS AIR FLOW AT VARIOUS
SPEEDS AND VARIOUS THROTTLE POSITIONS
FOR SES COMBUSTION AIR BLOWER, AND AMBIENT
AIR TEMPERATURE AND PRESSURE

FIGURE 4.7

The blower power consumption was measured under simulated operating conditions at rated air temperature in a steam generator. The results are compared to those of the old combustion air blower in Figure 4.8. As evident, the new blower system reduced the shaft power consumption substantially - 40 to 50% over most of the load range of the steam generator.

The aluminum blower has undergone many hundreds of hours of operation in SES steam generator Models 5, 7, and 8. The design has proven excellent in performance and durability.

MEASURED INPUT POWER TO OLD COMBUSTION
AIR BLOWER (INCLUDING V-BELT DRIVE)
COMPARED TO BENCH TEST DATA FOR
NEW SES BLOWER (INCLUDING COG BELT DRIVE)

FIGURE 4.8

4.1.3 Emissions

Emission testing of the Model 5 S/N #2 steam generator was performed while it was mounted in the engine compartment and was generating 811[°]K, 6.9 MPa (1000[°]F 1000 psi) steam.

Figures 4.9, 4.10 and 4.11 show the measured emissions levels as a function of firing rate and excess air.

Effort was made to model NO_x and CO formation analytically in the Model 5 burner in order to understand the effects of various features and for use as a design tool for possible further improvement of performance. The model is based on work by Sarofim, Williams, and Padia in modeling NO emissions in premixed gas fired burners. The present design of the burner/steam generator was aided by earlier work by the same group.

It was found that the positioning of a bare, low heat duty, coil approximately midway through the combustion chamber is very important. The coil partially cools the combustion gases to a temperature of 1673-1873[°]K (2552-2912[°]F), effectively quenching the NO_x reaction, but permitting continued oxidation of CO and HC.

4.1.4 Gasoline Igniter

Early SES ignition systems had utilized propane as the fuel source for a small torch which ignited the main flame. For automotive applications, propane was only an interim solution to the ignition problem. Subse-

FIGURE 4.9

FIGURE 4.10

FIGURE 4.11

quently, a program was undertaken to eliminate the propane requirement, resulting in the development of a gasoline fired ignition torch with very low emissions.

Figure 4.12 is a photograph of the gasoline igniter including its fuel solenoid. The design of the unit is based on an air atomizing type nozzle providing a stream of combustible mixture of fine fuel droplets and air. The mixture flows through a swirl-type stabilizer, generating an intense high velocity flame with good penetration into the combustion chamber. Development testing of the igniter included reliability testing over a variety of air pressure conditions and both hot and cold torch conditions to insure reliable starts under all circumstances. Specifications of the ignition torch are listed in Table 4.III. The requirements of small size, low air consumption, low fuel consumption and low emissions were all met. The igniter was easily interfaced with existing burner geometry and gave reliable and clean burner ignition.

4.2 Heat Exchanger Development

The SES steam generator heat exchanger is of a once through montoube design, utilizing several passes of finned tube and one or more passes of bare tube. One of the bare tube passes is located close to the flameholder to provide quenching of combustion reactions as discussed in the previous section. The passes are wound into 60° cones as shown in Figure 4.13. Figure 4.14 shows the various passes of the heat exchanger and their nestling together into a compact package. The relative location of the heat exchanger coil in the steam generator can be seen in Figure 4.2.

GASOLINE IGNITER INCLUDING THE FUEL SOLENOID

FIGURE 4.12

TABLE 4.III
GASOLINE IGNITER SPECIFICATIONS

<u>NO.</u>	<u>ITEM</u>	<u>DESCRIPTION</u>
1.	Dimensions	1.3 cm dia. x 13.3 cm long (1/2" x 5 1/4")
2.	Fuel Requirement	0.1 kg/hr at approximately 70 kPa (10 psi)
3.	Air Requirement	1.0 kg/hr at approximately 25 kPa (3 psi)
4.	Spark Requirement	Approximately 5 watts power, 10,000 Volts
5.	Operating Time	2 seconds per start
6.	Flame Characteristics	Pencil-like, 5-10 cm long
7.	Air cooled when not in use	
8.	Emission Generated	44 milligrams UHC and 1 milligram CO mean (measured at the Vapor Generator stack)

4.3 Further Steam Generator Development

During further research on the U.S. Bureau of Mines Ramcar project, Contract No. H0 155121, it was found that the primary reason for the deviation from the optimal emission curves during simulator tests was that the system is designed to operate optimally with intake and exhaust ducts having exactly the same barometric pressure. In the car, the intake is under the hood pressure while the exhaust duct leads to the rear of the vehicle. Ram air and condenser fan operations would disturb the pressure balance on the boiler, causing an upset in the air-to-fuel ratio and higher than expected emissions.

Model 9 steam generator for the Ramcar is designed with an internal pressure balancing system. The system works so well that an air metering valve in the inlet duct is no longer necessary. At the time of this writing, SES has accumulated 90 hours of running experience on this boiler and disturbances such as test cell door slamming and exhaust fan on/off have been applied many times. It was found that regardless of external disturbances or ambient conditions, the Model 9 steam generator operates very close to its optimal emission characteristics at both steady-state and transient operations. The unit has been developed for operation with No. 2 fuel oil and has proven even cleaner and more responsive than the Model 8 gasoline-fired steam generators. Remaining development, as of this writing, involves refinement of start-up/shut-down control, a more difficult task with No. 2 fuel oil than with gasoline.

CONICAL BOILER PASSES
FIGURE 4.13

VIEW FROM QUENCH AND SUPERHEATER PASS END

FIGURE 4.14

The heat exchanger of the Model 5 boiler was designed with four heavily finned passes, and one bare superheater pass all connected in series. The gas flow path followed a 5-3-4-2-1 sequence relative to the steam side flow (Pass No. 5 is the superheater and Pass No. 1 is the first economizer) as illustrated in Figure 4.15A. The fins are of stainless steel material, brazed on stainless steel tube. Very quickly it was learned that with such an arrangement the average metal temperature at high load is much higher than the low load temperature. Therefore, a large amount of energy must be pumped in (or out) of the heat exchanger metal before a transient response can be realized. The first attempted transient operation is shown in Figure 4.16. The response lag was judged totally unacceptable for a vehicle application. Accordingly a development program was undertaken to solve the problem.

As mentioned, the problem lay in the characteristic change in heat content of the heat exchanger metal from one load to another load. Computer studies showed that a revised pass order and lower weight materials would result in much improved transient performance. Model 7 was designed with the pass arrangement shown in Figure 4.15B, with extremely lightweight economizers of stainless steel tubing with aluminum fins.

The characteristic transient response for Model 7 is shown as measured in Figure 4.17. Second order effects in heat and mass transient distribution cause some oscillation in steam flow, but the net change in heat content from load to load is virtually zero. More sophisticated control of water flow enabled the elimination of the steam flow oscillation, resulting in smooth rapid response to command for power from the steam generator.

A
Model #5
S/N 1

B
Model #7

C
Model #8

FIGURE 4.15
COIL CONFIGURATIONS

STEP IN LOAD DEMAND: 17% - 70%

MODEL NO. 5 BOILER

FIGURE 4.16

STEP IN LOAD DEMAND: 29% - 78.5%

MODEL NO. 7 BOILER

FIGURE 4.17

With Model 7 demonstrated, the transient response solution had been proven, and the final design of a steam generator for the vehicle was executed. Model 8 was designed with all of the key features of Model 7, with the additional goals of smallest package, lightest weight, and highest efficiency which could be achieved within the program constraints.

Figure 4.15C illustrates the passa arrangement for Model 8. It is the same as that for Model 7 with the addition of a third economizer pass and increased fin density. Figure 4.18 compares the steady state stack temperatures of Model 8 to Model 7 over the load range, showing the improvement in effectiveness. Figure 4.19 compares the efficiencies (net) of the two units, showing the considerable improvement in operating performance of the Model 8.

The detailed pass specifications for the Model 8 heat exchanger are shown in Table 4.IV. The combustion products flow in order from left to right. This combination appeared to be the optimum for the given constraints of maximum efficiency, low weight, low volume, fast response, combustion interface, and reasonable cost for the vehicle application. The finished Model 8 steam generator is shown in Figure 4.20 as assembled and ready to be installed in the engine compartment of the vehicle.

Table 4.V summarizes the progression in design from Model 5 to Model 7 to Model 8, and some of the critical parameters. The package weight was reduced by 45% and the volume by one third. The warm-up time was reduced by one third, the water inventory by one half, and the energy level variation (heat content) almost completely eliminated. The characteristic steam flow response time was reduced from 7 seconds to under one second.

MEASURED MODEL NO. 8 STACK TEMPERATURE vs LOAD
COMPARED TO COMPUTER MODEL PREDICTIONS

FIGURE 4.18

MEASURED MODEL NO. 8 EFFICIENCY vs LOAD COMPARED TO
COMPUTER MODEL PREDICTIONS

FIGURE 4.19

TABLE 4.IV

MODEL #8 HEAT EXCHANGER PASSES SPECIFICATIONS

Pass Ident.	Quench	Superheat	Evaporator	Evaporator	Economizer	Economizer	Economizer
Tube OD (cm)	1.270	1.588	1.588	1.588	0.953	0.953	0.953
Tube ID (cm)	0.940	1.257	1.339	1.339	0.810	0.810	0.810
Fins/CM	(Bare)	(Bare)	2.36	3.94	3.94	3.94	3.94
Fin D (cm)	--	--	2.540	2.540	2.223	2.223	2.223
Fin Thk (cm)	--	--	0.064	0.064	0.025	0.025	0.025
Length (m)	4.62	5.23	5.08	5.32	7.01	5.54	3.78
Weight (kg)	2.07	3.02	4.12	5.59	1.88	1.49	1.02
Tube Mat.	SS	SS	SS	SS	SS	SS	SS
Fin Mat.	--	--	SS	SS	Alum.	Alum.	Alum.

MODEL 8 STEAM GENERATOR

FIGURE 4.20

VAPOR GENERATOR DESIGN SUMMARY

	<u>MODEL 5</u>	<u>MODEL 7</u>	<u>MODEL 8</u>
1. HEAT EXCHANGER COIL WEIGHT - kg (lb)	44.5 (98)	27 (60)	21 (47)
2. PACKAGE WEIGHT -kg (lb)	75 (165)	58 (127)	41 (90)
3. OVERALL DIMENSIONS LENGTH X DIAMETER - m (in)	0.47 X 0.50 (18.5 X 19.5)	0.47 X 0.50 (18.5 X 19.5)	0.47 X 0.41 (18.5 X 16)
4. COLD START WARM UP			
a. JOULE (Btu)	5.4×10^6 (5100)	4.5×10^6 (4300)	3.9×10^6 (3650)
b. TIME (sec)	13.5	11	9
5. WATER INVENTORY			
a. MAXIMUM LEVEL - kg (lb)	1.6 (3.5)	1.3 (2.9)	.68 (1.5)
b. VARIATION kg (lb)	.45 (1)	.18 (0.4)	NEG
6. ENERGY LEVEL CHANGE			
JOULE (Btu)	1.5×10^6 (1400)	1.1×10^5 (100)	NEG
7. STEAM FLOW RESPONSE TIME ESTIMATE (sec)	7	1	<1

5.0 FEEDWATER PUMP

During the early development work on the steam engine feedwater pump, SES had tried to adopt commercially available pumps having the potential for satisfying the following pump requirements:

1. The ability to pump pure water (deionized - no lubricants).
2. Maximum delivery pressure of 8.3 MPa (1200 psia).
3. Maximum water temperature of 394°K (250°F).
4. Maximum flow of .01 m³/min (3 gpm).
5. Low NPSH capability.
6. Size suitable for packaging in an automotive system.
7. Self priming capability.
8. Drainability for freeze protection.

Two commercial pumps were selected for preliminary tests. One, a Hypro four cylinder, radial piston pump, had been used in earlier SES steam engine development work. The other was a Delavan three cylinder, axial piston pump. At the conclusion of the preliminary testing, it was decided that neither pump was suitable as delivered. Each had developed rusting conditions during closed loop testing,

which had to be corrected by treating internal passages in the cast iron pump housings. Cavitation problems were also present to some extent in both pumps at the higher speeds and/or low NPSH operating ranges.

The Delavan water pump was selected as the more promising starting point for further developmental work. The original Delavan pump specifications are given in Table 5.I. The following sections will describe how modifications, including piston bearings and seals for high pressure operation, through valves and finally the block, were made; until the original product became unrecognizable, evolving into the SES water pump whose specifications are given in Table 5.II.

5.1 THE DELAVAN PUMP

The Delavan 3-piston axial pump has a capacity of $15 \text{ m}^3/\text{min}$ (4 gpm) with a maximum output pressure rating of 6.8 MPa (1000 psi) (as suggested by the manufacturer) and a speed range to 30 Hz (1800 RPM). This was a relatively new pump on the market and was chosen for evaluation in this program because of its suitability for packaging with the system (i.e., equivalent in shape and size to an automotive generator).

The pump was tested over a complete range of speeds to 30 Hz (1800 RPM) and to pump outlet pressures of 7.6 MPa (1100 psia) and water temperatures to 339°K (150°F). Under closed cycle testing, an excessive rust problem developed. This was due to rusting of some passages within the pump casting. This problem was quickly remedied by chrome plating the interior passages.

TABLE 5.1

DELAVAN PUMP SPECIFICATIONS

- o Axial Piston Type-Fixed Displacement, 3 Cylinders
- o Bi-Directional Rotation - Belt or Direct Drive
- o $.015 \text{ m}^3/\text{min}$ (4 gpm @ 1750 RPM) $.525 \text{ Cu.In./Rev}$
(Theoretical) Displacement
- o Theoretical Displacement: 8.60 ml/Rev ($0.525 \text{ in}^3/\text{Rev}$) [15 l/min @ 29 Hz (4 gpm @ 1750 RPM)]
- o Working Pressure: 4.8 MPa (700 psi)
- o Maximum Operating Temperature: 333°K (140°F)
- o Size: 25 cm (9 3/4") Long (Excluding Shaft Projection) x Approximately 14 cm (5 1/2") Diameter
- o Approximate Weight: 18 Kg (40 lbs)

TABLE 5. II

SES PROTOTYPE FEEDWATER PUMP

- o Configuration
 - Fixed Displacement, 3 Cylinder in Line
 - Flow Control by Modulation of Inlet Valve Unloading Solenoids
 - Weight: 6.4 kg (14 lb)
 - Bi-Directional Drive Capacity
 - Automatic Drain on Shutdown for Freeze Protection
- o Performance
 - At Maximum Speed of 42 Hz (2500 RPM)
19 l/min (5 gpm) at 8.3 MPa (1200 psi)
90% Volumetric Efficiency
75% Overall Efficiency
394°K (250°F) Inlet Water Temperature
 - Self Priming: 20 cm (8 in) Hg Dry Lift

An inspection of the pump after some of the initial testing recalled a high friction condition on the piston actuating shafts and their guide bushings; noticeable fretting had developed on the actuating shafts at the bearing points.

The first series of tests resulted in changes in the piston actuating shaft bearing arrangement, and the main thrust bearing to provide reliable operation and adequate life at pressure of 8.6 MPa (1250 psi). Specifically, bushings were added to the piston ends of the actuating shafts to alleviate the drive mechanism side forces from acting on the pistons and piston water seals. A larger diameter and capacity thrust bearing was installed without increasing the overall pump size. These changes eliminated the occasional problem of a piston shaft sticking with pressures over 5.5 MPa (800 psi), reduced the friction and power loss, and extended piston seal-ring life.

With the pressure capability of the pump increased, a second series of developmental tests followed for the purpose of raising the working temperature to 405°K (270°F). Consequently, the following series of modifications were made:

- All static seals were changed from Buna N to Viton.
- The back-up rings were added to the cylinder liners and intake valve cages.
- The discharge valves were changed from glass-filled nylon to #303 stainless steel.

- The intake valve cage material was changed from Delrin to bronze.
- The low pressure piston seal rings were changed from a square cross section, graphite composition ring to a tee cross section, compression type seal ring made from ethylene propylene.
- The drive case lubricating oil was changed from SAE #20 to SAE #30.

After making these changes it was found during testing that the steel valves caused an appreciable increase in noise level, particularly with speeds in excess of 25 Hz (1500 RPM). The steel valves were then replaced with valves made from a fiber reinforced Teflon. This change was effective in quieting the pump and providing efficient high pressure sealing, without the necessity for lap fitting of the valves to the seats as is normally required with metal valves. Inspection of these valves after 96 hours of operation revealed no signs of deterioration.

Subsequent endurance testing at maximum pressure, temperature, and speed disclosed a faulty condition in the high pressure piston rings. These original rings consisted of two butt cut piston rings per piston made from: 15% glass, 5% molydisulphide, filled Teflon. Their running performance had been satisfactory at temperatures up to 366°K (220°F); however, after the pump stopped and cooled off, these rings had a tendency to lose their tension and relax sufficiently to cause a 20% to 30% drop in volumetric efficiency.

Another condition that became apparent during the course of endurance testing was the relatively short life of the low pressure seal rings. Although the replacement, tee cross section type seal rings exhibited an improvement over the original seals, it was found that the wear rate in a thirty-hour period was enough to permit external leakage of water.

The seal that was finally found suitable is a spring loaded U cup type seal made from a Teflon graphite composition material. One cup was used as an excellent solution at the time, due to lack of space on the piston. The cup was applied with the pressure actuated side to the water, with reliance on spring tension for air sealing. Ideally, it would be preferable to use two cups, back to back, to utilize pressure actuation of the seal lips with spring tension to seal both mediums.

Generally, the lip type seal can provide longer life than the compression type because the lip deflection principle permits more follow up for wear without introducing as much friction.

Some consideration was given to the possibility of operating the Delavan pumps at speeds higher than the upper limit of 30 Hz (1800 RPM), specified by the manufacturer. Interest in operating at higher speeds resulted from further experience with the component drive and packaging development work, which indicated that speeds to 42 Hz (250 RPM) might be of some advantage. One of the known

speed limitations was a volumetric efficiency drop commencing at approximately 29 Hz (1750 RPM), with fairly high losses and flow instability at 33 Hz (2000 RPM). The problem was found to be associated with the discharge valve lift and back flow conditions. Installation of limit stops on the discharge valves, extended the speed range to 38 Hz (2300 RPM) with practically no change in volumetric efficiency.

The total of 12 modifications made to the Delavan pump are summarized as follows:

1. Bearings - Main shaft bearing and wobble plate thrust bearing replaced with larger capacity, bearing cap and shaft revised.
2. Push Rod Bushing - Bushing added to push rod end to keep push rod from tilting and hanging-up.
3. Pistons - Diameter increased to reduce clearance and prevent piston seal ring extrusion. Length of piston head increased to accomodate additional seal rings.
4. Piston Seal-Ring - Teflon carbon blend, angle cut seal-rings incorporated to maintain positive contact.
5. Cylinder - Redesigned to match longer piston.
6. Static Seals - Changed to Viton or silicon throughout.

7. Low Pressure Water Seals - Spring loaded, moulded U-cup seals, made from Teflon graphite compound, incorporated in place of replacement tee ring seals.

8. Air Seals - added to low pressure side of pistons Spring loaded, moulded U-cup seals, made from virgin Teflon.

9. Back-up Rings - Stainless steel back-up rings added to cylinder liner O-rings at push rod end.

10. Intake Valve Cages - Original moulded Delrin cages replaced with type 303 stainless steel, and stops added.

11. Discharge Valves - Original nylon valves replaced first by stainless steel and then by fiber reinforced Teflon; bronze stops added to limit valve lift and improve volumetric efficiency.

12. Oil Case Seal - Original Buna-N oil seals replaced with spring loaded Teflon graphite type seals; oil changed from SAE #20 to SAE #40.

Figure 5.1 shows the final stage of development with all modifications incorporated into the Delavan pump, resulting in a reliable, high temperature, high pressure, and low NPSH water pump. The problems that were still to be addressed at that time were drainability and primability of the pump. It was decided that a further modification of the pump to provide these features must involve a major re-configuration of the pump. The development of this "new" pump is discussed in the following sections.

PARTIAL CROSS SECTION OF DELAVAN PUMP

FIGURE 5.1

5.2 PREPROTOTYPE SES FEEDPUMP

The SES feedwater pump is a fixed positive-displacement three-piston in-line pump, with eccentric-operated pistons and positive piston return (no springs), shown in Figure 5.2. The pump weighs only 6.1 kg (13.5 lb) with overall dimensions of 8.9 x 18.4 x 17.8 cm (3-1/2 x 7-1/4 x 7 in). Precision heavy-duty ball and roller bearings are used on the driveshaft and eccentrics, and the entire piston actuating mechanism is oil immersed in a sealed case located above the pumping section. Good drainability on shutdown and minimum leakage of water into the oil crankcase are two important features which were provided by locating the crankcase above the pumping section. The inlet valves and passages to the pump were carefully designed to result in a very low NPSH capability, approximately 0.6 m (2 ft) water head at maximum required water flow. Over 2300 hours of pump operation have been accumulated while pumping 383°K (230°F) water at a discharge pressure of 8.3 MPa (1200 psig).

The feedwater pump is driven directly off the expander. Flow control is accomplished by unloading solenoids that keep the intake valves open. Each of the pump cylinders is separately controlled by modulating the fraction of time (pulse width modulation) during which its intake valve is unloaded, resulting in an efficient and smooth water flow modulation technique.

SES FEEDWATER PUMP WITH FLOWCONTROL SOLENOIDS

FIGURE 5.2

Throughout the period of performance many hours of experience were accumulated. The pump was continually improved until it was evolved into a unique product. The major areas of improvement were as follows:

1. The original bronze, hydrodynamic bearings on the eccentric were replaced by anti-friction roller bearings, in order to improve low speed operation.

2. The piston seals were redesigned into an interchangeable three piece unit made up of one "U" cup with an integral preload spring, a metal back-up ring, and one Teflon composition guide bushing. The interchangeable unit gave excellent results under all operating test temperatures, speeds and cool-off conditions. This three-piece unit was tested for 200 hours with negligible wear or leakage.

3. A third modification to the pump was a result of a search for a more durable, lower cost discharge valve. Previously, the discharge valves were of the disc type, made from duroid and lap fitted to stainless steel seats. These were a carry-over from the Delavan modification work. Endurance tests revealed that their life was between 350 to 400 hours. These valves were replaced with stainless steel ball valves operating on a reamed stainless steel seat. The ball is self seating and eliminates the necessity for a lap fit. The ball is more durable and economical than the composition material disc.

4. The intake valve assemblies were changed from spring loaded poppet valves to ball type assemblies, in order to provide good drainability of the pump. The ball is positioned by three-point contact guides, thus minimizing water collection between the ball and guides.

5. Much development effort and testing time was dedicated to the operation of an intake valve unloading mechanism without a sacrifice in NPSH and priming capability. The final arrangement involves spring loaded solenoids and push rods that do not interfere with the valve operation when the solenoid is activated. The mechanism is shown in Figure 5.3.

SES had spent more effort on the development of the feedwater pump than originally intended, however, the results are worthwhile. Figures 5.4 and 5.5 demonstrate the performance of this excellent reliable trouble-free unit. Figure 5.4 shows that the volumetric efficiency has only deteriorated to 80% after 2300 hours of operation. Figure 5.5 shows that flow modulation of the direct driven pump is achieved with only a minor efficiency penalty.

SIMULATOR FEED WATER PUMP—END VIEW
VARIABLE FLOW UNLOADING MECHANISM

FIGURE 5.3

SES FEEDWATER PUMP - SERIAL NO. 6

WATER DELIVERY DURING 2300 HOUR ENDURANCE TEST (MAY - JULY, 1974)

FIGURE 5.4

S.E.S. FEEDWATER PUMP (SERIAL #7)
WATER FLOW PERFORMANCE WITH ELECTRONIC
CONTROLLED UNLOADING CONTROLLER

FIGURE 5.5

6.0 PREPROTOTYPE CONDENSER

The design of the condenser was quite straightforward, and the final design differs little from the proposal version. The optimization criteria were minimum fan power and minimum core weight (which should represent roughly minimum cost), with a secondary criterion of minimum core volume. Following the optimization analysis given in the SES proposal, the design effectiveness of 85% resulted in minimum fan power for a given frontal area. However, decreasing the design effectiveness to 75% increased the power requirement by only about 7% while effecting a 25% reduction in core volume. The selection of this design condition assured that close to the minimum fan power would be required at full load, resulting in the lowest fan design speed and lowest noise, while efficient condensation would be accomplished at part load.

An alternative design approach was to under-design the condenser for full load, and to design for minimum fan power at part load. This would result in improved part load efficiency and a smaller condenser, but would require either increased full load fan power or increased full load condenser pressure. The latter choice would actually improve overall system efficiency at full load, but creates more severe condenser overpressure problems under WOT, low speed high ambient conditions. For the preliminary design, therefore, the more conservative approach was taken.

The problem of condenser freezing under low ambient conditions was studied. The air flow at which the steam side wall temperature drops to 273°K (32°F) at the condenser air inlet face was computed as a function of condenser heat load. The results indicated that in all cases, the air flow required to cause freezing is greater than that required under WOT, normal ambient conditions. This means, that if a shutter system is employed to prevent over-cooling under more moderate ambient temperatures, it would more than suffice to prevent freezing.

Auxiliary components such as the feedpump, accessory drives, piping and ducting were explored and analyzed to the extent required to proceed with overall system analysis and the design of the major system components.

Condenser fouling due to oil carryover was studied using a model which supposes that all of the oil entering the condenser finds its way to the wall and flows down the wall in a laminar film under the action of gravity and condensate shear. The results of the study demonstrated that as long as no non-Newtonian emulsions are formed, an oil content as high as 100 ppm by weight ($M_{\text{oil}}/M_{\text{steam}} = 10^{-4}$) can be tolerated while reducing the ratio of overall steam conductance (U_{st}) to condensing conductance alone (h_{cond}) to 0.84, which has the effect of a loss in effectiveness of only about 5%. As the peak steam flow is 585 kg/hr (1290 lb/hr), the allowable oil flow to the condenser would be .06 kg/hr (0.129 lb/hr). Thus, assuming the oil

pick-up to be 5.85 kg/hr (12.9 lb/hr), (1%), the steam/oil separator would have to be 99% effective. The oil content would be further reduced in the hotwell of the condenser by a skimmer so that the oil content of the feedwater would be of the order of a few parts per million.

The design pressure of the condenser was .14 MPa (20 psia). However, under low speed WOT operation or under high ambient conditions, condenser pressure would be allowed to rise to 0.21 MPa (30 psia), and would act as a safety valve at .34 MPa (50 psia) even with no non-condensables present.

6.1 CONDENSER FANS

The condenser fans are needed to pull ambient air through the condenser core in sufficient volume to condense all the steam. The air flow required by the preliminary condenser design was determined for various steam flow and ambient temperature conditions. At wide open throttle (WOT) above an engine speed of 25 Hz (1500 RPM), the required cooling air flow is a constant $6.75 \text{ m}^3/\text{sec}$ (14,300 CFM) since the steam flow to the expander is limited to 9.75 kg/min (21.5 lb/min) during WOT conditions. Below 25 Hz (1500 RPM), the expander cut-off is constant and the air flow decreases as steam flow decreases. At lower steam flows or lower ambient temperatures, the air flow will, of course, be lower, but the same general characteristics apply.

The fan must be capable of producing enough pressure rise to overcome the condenser core pressure drop and all other entering and leaving losses (all lumped as "ducting loss"). The ducting loss was assumed to be equal to 3 velocity heads at the condenser outlet. This value was optimistic compared to present radiator design standards of 4 - 8 leaving velocity heads; however, it was necessary in order to keep fan power as low as possible, and can be achieved by careful design of the air ducting.

The total pressure loss which the fan must overcome is reduced by the ram effect of air entering the condenser while the vehicle is in motion. Since this effect increases as velocity to the second power, it can be considerable at high vehicle speeds. The fans were designed to make full use of the ram pressure available.

6.1.1 FAN DESIGN POINT

The fan was initially designed to maintain a 394°K (250°F) condenser temperature at design ambient temperature 303°K (85°F) at full power, low gear. At any condition where the fans would not be able to provide the required air flow, then the condenser temperature first would be allowed to rise, then under very severe conditions, the maximum steam flow would be limited.

Even at the stated ambient temperature, there was more than one design point which could be used even at WOT. First, the design point could be WOT at 8.3 Hz (500 RPM) in low gear. Then, as engine speed increased

from 8.3 - 25 Hz (500 to 1500 RPM), the fans could provide slightly more flow and pressure than required with a constant drive ratio.

Alternatively, the design point could be WOT at 25 Hz (1500 RPM) in low gear. At speeds above 25 Hz (1500 RPM), there would be no problem to provide adequate flow, since the required flow is constant and required fan pressure is less and the engine speed is higher. However, below 25 Hz (1500 RPM), the flow and pressure which the fan would provide at its reduced speed was less than required, and as a result, the condenser temperature would increase. The condenser temperature at 8.3 Hz (500 RPM), WOT condition in low gear would increase to 394^oK/21 MPa (250^oF/30 psia) if the fan speed is 1/3 the 25 Hz (1500 RPM) design speed. It was seen, however, that for a thermostatically controlled slip-clutch drive system, the total engine power to the fans was 3.9 kW (5 hp) less for the 25 Hz (1500 RPM) design point. Furthermore, a power savings could be achieved over the entire operating range, by choosing the design point at 25 Hz (1500 RPM), while sacrificing only increased condenser temperature and pressure at less frequently encountered operating conditions.

It was felt that some type of variable speed drive must be used for the fans. Two types of drives were studied in some detail. First was a hydraulic drive, with a single variable displacement hydraulic pump driving a constant displacement motor. Second was a thermostatically controlled viscous slip-clutch drive, with a clutch at each

fan and a mechanical step-up drive from the engine. A preliminary choice of the viscous clutch was made based on initial cost and drive efficiency. The hydraulic drive would have a design point overall efficiency of approximately 50% with an output power of 8.9 kW (12 hp), 4.5 kW (6 hp per motor), and cost \$75.00 in production quantities. The viscous slip-clutch cannot be compared on a direct efficiency basis, but to be able to transmit 8.9 kW (12 hp) at the design point will require maximum input power of only about 11.2 kW (15 hp). The cost of two clutches would be in the range of \$12.00 - \$24.00.

The performance of the viscous clutch system during various operating conditions was predicted. The assumption was made that the slip ratio of the clutch could be varied by thermostatic control (as is done on present automotive systems) or by condenser pressure control, so that the fan speed is exactly that needed by the fan to produce the required air flow and pressure rise (except that the slip ratio cannot be negative, i.e., output power greater than input power). The fan horsepower and torque can be determined as discussed below. Since there is no torque difference from the clutch input to output, the horsepower dissipated by the clutch is:

$$HP_{cl} = T_F (w_E - w_F)$$

or since $T_F = HP_{F/F}$ and $w_F = 2\pi N_F$

The clutch horsepower is: $HP_{cl} = \frac{HP_F}{N_F} (N_E - N_F)$

where N_F = Fan speed, RPM

N_E = Drive speed into clutch (a constant multiple of engine speed)^s, RPM

HP_F = Horsepower to drive fan, hp

The fan horsepower was determined from "ideal" fan characteristics presented by Eckert, and from variable speed fan law relations. These relations are given for vane-axial fans at constant speed by Eckert in the form:

$$\Delta P / \Delta P_o = f_1 (Q / Q_o)$$

$$\eta / \eta_o = f_2 (Q / Q_o)$$

$$HP / HP_o = f_3 (Q / Q_o)$$

The fan law variations with speed take the form:

$$Q_o / Q_D = N / N_D$$

$$\Delta P_o / \Delta P_D = (N / N_D)^2 (\rho / \rho_D)$$

where

ΔP = Required fan pressure rise, in H_2O

Q = Required fan air flow, CFM

ΔP_D = Design point pressure rise

Q_D = Design point air flow

η_D = Design point fan efficiency

HP_D = Design point power
 N_D = Design point speed

By combining "curve fit" forms of Eckert's characteristics with the fan law equations, the fan speed, fan power and clutch power can be determined for any steam flow rate and engine speed.

Analyses using the above equations showed that the power decreased rapidly when either steam flow or ambient temperature was decreased. They indicated that no fan power was required during many conditions (including most cruise conditions at 303°K (85°F) or less) when there is sufficient ram pressure available to overcome all losses including losses in the fan itself in the system. Although this zero-power condition is theoretically possible, it cannot be achieved in the real system unless a separate disconnect is put in the drive system.

6.1.2 PRELIMINARY FAN DESIGN CONSIDERATIONS

The design point flow and pressure rise for the condenser fan design were the result of the optimization study described above. The rest of the conditions for the fan design were defined by available space, operational flexibility, and by production considerations. The following list is a brief summary of the conditions selected for the preliminary fan design by other than fan design considerations.

1. Fan would be tube-axial type. The absence of stator vanes makes the tube-axial fan more compact axially by 5.1 - 7.6 cm (2 - 3"). Tube-axial fans are simpler and less expensive. However, the total-to-static efficiency (η_{TS}) was expected to be 10 - 15 points lower than for a comparable vane-axial fan, and the fan speed had to be 10 -20% higher for tube-axial fans to meet the same pressure requirements.

2. The fan would be located downstream of the condenser. The available space and the fan drive arrangements dictate the fan location. It is known that the fan power is proportional to the absolute air temperature for the same condenser performance (not including leaving losses).

The fan power would be 18% higher than with the fan located upstream of the condenser.

3. The design point flow rate should be equal or higher than the best efficiency point flow rate. The fan optimization study showed that a large percentage of the fan operation would be at reduced pressure and at flow rates lower than the best efficiency point at the given speed. This condition was defined for minimum overall power consumption by the fan.

4. Tip clearance between the fan shroud and blade tips is .25 cm (0.10"). The mounting and the rigidity of the shroud require a large tip clearance, but

the fan efficiency decreases rapidly with increasing tip clearance.

5. Ram air effect is utilized. The air inlet would be at or near the front stagnation point of the air stream lines around the car, therefore, the ram effect can be utilized.

6. Twisted air foil blades would be used. The production method would permit the use of twisted fan blades which result in more efficient and quieter fan design.

7. Axial space available for the fan between the discharge face of the condenser core and the nearest component (air flow obstruction) would be 27.9 cm (11"). This was the maximum value available for the fan because of the space limitations. Good efficiency and low noise generation require proper axial spacing free of stationary components upstream and downstream of the fan rotor.

Some of the hardware design decisions were made simultaneously with the analytical phase of the condenser fan design. They were:

1. It is desirable to locate the two fans in the center of the two condenser core-halves.

2. The fan shroud for each fan connects a transition piece from the rectangular shape of the core-half to the circular shape of the fan shrouds. The directional change where the transition piece meets the fan shroud would be achieved with a generous radius of curvature.

3. Separate the two condenser halves and the two fans completely with the transition pieces. The possible instabilities due to parallel operation of fans were eliminated and the addition of an air-conditioner condenser in series with one of the condenser halves becomes possible without upsetting the operation of the two fans. With A.C. one of the fans would have to operate at an increased pressure rise.

4. A nose piece for the fan hub should be used which blends into the hub diameter.

5. Optimum axial spacing between the core and the blade inlet edge.

6. The fan drive and the speed control can be the same or separate for the two fans.

The design point overall efficiency of the fan was assumed to 60% for the above system study. Therefore, for the preliminary system design the fan efficiency of 60% at design point was used for system performance determinations.

Analytical studies were made of the effects of condenser shape and fan placement on flow maldistribution, and the resulting reductions in performance. Additional studies were directed to the optimization of design point condenser pressure for maximum BHP output under wide open throttle conditions.

6.1.3 FLOW DISTRIBUTION

The condenser core was separated into four sections and estimates were made of the aerodynamic resistances due to grill restriction, core losses entering, leaving, and turning losses. Following this, the average flow through the condenser sections were calculated, from which the heat transfer effectiveness, the required total air flow, and the resulting pressure drop (or fan pressure rise) were computed. The results show that a .41 m x 1.47 m (16 x 58") core would require 25% - 42% greater fan power than an ideal (perfectly distributed) condenser, while the penalty for the .48 m x 1.24 m (19 x 49") core is only about 11%. In addition, the sharper turning angle into the fans with the wider core would undoubtedly make prevention of flow separation a more serious problem. Significant flow separation at the fan inlet would reduce fan efficiency and create a serious noise (and perhaps vibration) problem. For these reasons, the .48 m x 1.24 m (19 x 49") core was definitely preferred over the wider one. (Note: This analysis used an AMC Ambassador, and was subsequently updated for the 1975 Dodge Monaco.)

6.1.4 OPTIMUM DESIGN

The final analytical design focused on the optimal design for minimum condenser weight (and cost), and minimum pressure drop, and the selection of condenser operating conditions for maximum system performance. The latter objective was directed toward the selection of a

design which would provide maximum system BHP at constant steam flow for a given sized condenser. In this case the variables are condensing temperature and air flow rate. The study showed that peak performance with the $.6 \text{ m}^2$ (6.5 ft^2) condenser occurs with a condensing pressure of $.17 \text{ MPa}$ (25 psia). Also studied were optimum fin spacings, fin thicknesses, and plate thicknesses.

The preliminary design of the condenser and the analysis of its performance were all based on a design steam flow rate of 918 kg/min (21.5 lb/min). Subsequent system performance analysis indicated that vehicle performance goals could be met with maximum steam flows of the order of $8.2 - 9.1 \text{ kg/min}$ ($18 - 20 \text{ lb/min}$) and correspondingly lower heat rejection rates. The condenser design for the lower heat rejection rate had the same frontal area as the preliminary design, but had a smaller depth, and was consequently designed for a lower maximum air flow rate.

6.1.5 HARDWARE

Twin 48.5 cm ($19''$) tube axial fans were mounted on Eaton "Tempatrol" viscous fan clutches through an integrally cast hub. The clutches were mounted in extended (twin element) automotive water pump type bearings which were retained in the hub of a spider fastened to the fan shrouds which were designed for mounting on the vehicle frame. The clutches were each driven through a vee-belt pulley through a central pulley driven off the expander through a splined propeller shaft with universal joints.

Cooling for the viscous clutches was provided by drawing air through the hub. Included was the bellmouth entry to the fan which occurred only at the sides; the fan diameter being equal to the core height.

The twin fans were located behind the condenser and placed beside each other in either side of the condenser centerline to provide maximum clearance for the front wheel wells. Placement of the fans in front of the condenser was considered as this would reduce the fan power requirement. However, the requirement to diffuse the flow out of the fans if placed before the condenser and the added complexity in the fan drive, forced placement behind the condenser.

The final analytical and hardware design of the condenser incorporated changes in the design point for the condenser and in its configuration. The new design point was different in two respects: The condenser heat load was reduced from the original $.38 \times 10^6$ W (1.3×10^6 BTU/hr) to $.35 \times 10^6$ W (1.21×10^6 BTU/hr) as a result of the lower maximum steam flow rate of 9.1 kg/min (20 lb/min), compared to the original 9.8 kg/min (21.5 lb/min). Secondly, the design point pressure was increased from .14 MPa (20 psia) to .23 MPa (24 psia) at full load. However, the control point, that is the pressure at which the control system would attempt to maintain the condenser, remained at .14 MPa (20 psia). This change had the effect of reducing the maximum air flow requirement as well as the maximum fan head rise requirement, thereby easing the fan power requirement significantly.

The final condenser design utilized the AiResearch* perforated plate fin as its air-side heat transfer surface. The design point change had little effect on the core geometry, since it was designed for a slightly higher air-side effectiveness than the preliminary design, thereby compensating for the lower heat flow and higher temperature difference.

* (NOTE: AiResearch under separate contract to the EPA designed and constructed the condenser core and fan assemblies to SES specifications. The condenser and fans were supplied to SES as Government Furnished Equipment.)

6.1.6 FINAL FAN-DRIVE DESIGN

The fan design analysis was begun with two specific drive train systems. The first choice was a 2-speed standard transmission because of better fuel economy, while the second choice was a 3-speed automatic transmission. Fan designs were determined for each transmission. Each design allowed vehicle performance goals to be met, while keeping maximum drive power and dissipated clutch power low, and minimizing fan power during cruise. The best design for the two drive train choices proved to be considerably different. The 2-speed system required a fan drive capable of transmitting up to 13.8 kW (18.5 hp) with a 3:1 speed increase between expander and fan, while dissipating over 7.5 kW (10 hp) in the clutches. The 3-speed system had to transmit only 8.6 kW (11.5 hp) in the clutches. In

addition, the cruise fan power for the 2-speed was 2 to 3 times higher than for the 3-speed. This last effect reduced fuel economy by as much as 13% (versus less than 4% for 3-speed) in high gear cruise and negated the main reason for choosing that drive train.

As an alternative to the 2-speed manual transmission, a 3-speed manual was investigated. This drive train had gear ratios and rear axle similar to the automatic. The fan and drive design for this system was not too different from the automatic. The maximum power which must be transmitted was up to 10.4 kW (14 hp) with a speed ratio of 2.2:1, and the cruise fan power increased by approximately 25%.

For future convenience, it was decided to use a single fan/fan drive design for either drive train. The design was for the 3-speed manual transmission, being oversized for the automatic. The predicted power requirements were analyzed for total fan power needed as a function of expander speed and steam flow at an ambient temperature of 303°K (85°F) for transmission gear ratios. At other ambient temperatures, the engaged and disengaged powers would be similar. However, the speed at which the clutch disengages for a given steam flow will shift (increase at higher ambient temperature and vice versa). The peak power, at 41.6 Hz (2500 RPM) expander speed, is just over 10.4 kW (14 hp), under the worst ambient conditions.

Previously, the design comprised twin tube-axial fans driven by a layshaft from the expander through thermostatically (or steam pressure) controlled viscous clutches. Subsequent drive design studies and packaging studies indicated that the linkages in the layshaft that would be required to isolate the expander motions from the relatively fixed condenser system would have to be capable of significant angular motion and displacement. Furthermore; discussions with Eaton, the manufacturers of the Tempatrol thermostatically controlled viscous clutch, indicated that modifications to the automotive clutch to achieve the required controlled response would require significant development effort. For these reasons, an alternative drive mode was ultimately selected that consisted of a hydrostatic drive comprising a variable displacement expander-driven hydraulic pump, driving a single fixed displacement hydraulic motor.

Modulation of fan speed in the hydrostatic drive system was accomplished through the use of a condenser pressure-operated pump displacement control.

6.2 CONDENSER PERFORMANCE TESTS

The condenser test stand was incorporated into the preprototype engine test stand in relatively the same location as in the vehicle. A schematic of the system is shown in Figure 6.1. A photograph of the test stand is given in Figure 6.2. The lower header of the condenser was operated

FIGURE 6-1

SCHEMATIC OF CONDENSER SYSTEM

REAR VIEW OF CONDENSER AND FANS INSTALLED IN ENGINE TEST STAND

FIGURE 6.2

as a dry sump, with the condensate draining by gravity into a separate sump. No steam trip was used, so there was always a steam/water interface in the external sump. Deaeration was accomplished in a small, air-cooled re-flux, i.e., flowing loop condenser which drew steam from above the steam/water interface, condensed it and returned the condensate to the sump. The non-condensable gases were concentrated at the far end of the deaerator and were purged through a conventional steam-radiator air purge valve.

In an attempt to take advantage of the low net positive suction head requirements of the feedpump, the feedwater boost pump was eliminated from the preprototype test set-up. However, in order to provide for operation of the system in case the available NPSH was insufficient, a small water-cooled feedwater subcooler was provided. The degree of additional subcooling could be varied from 255°K (0°F) to over 289°K (60°F). For the purpose of priming the feedpump, which was located above the level of the sump, a check valve was located at the outlet of the sump and priming was accomplished by pumping make-up water into the line downstream of the check valve.

The air-handling system was designed to provide a close simulation of vehicle aerodynamics without entailing an excessive amount of ductwork. The major problem in the design was to provide a good simulation of ram-air boost

without using a condenser inlet plenum or requiring complicated flow straighteners. By using an induced-draft blower at the condenser exhaust instead of a forced-draft blower upstream of the condenser, uniform flow into the condenser was obtained, and the condenser inlet was completely free of any ductwork. This allowed easy access to the condenser and the engine compartment. Ram-air boost was simulated by maintaining the exhaust plenum pressure below atmospheric pressure by the same amount that the ram-air stagnation pressure would be above atmospheric pressure.

The air from the condenser was exhausted from the engine compartment through orifices which simulated the opening between the vehicle sheet metal and the ground, into an exhaust plenum which surrounded the sides and rear of the compartment. The air flowed through an exhaust duct, past pressure control dampers and into the induced draft blower. Another set of dampers at the blower output was controlled to recirculate sufficient hot exhaust air back to the test cell to maintain the cell ambient temperature between 288°K and 303°K (58° and 86°F).

The S/N #2 steam generator was operated on a separate test stand and piped directly to the air-cooled condenser. The steam outlet conditions were controlled to produce slightly superheated steam so that the condenser inlet enthalpy could be determined accurately. The condenser and fans were located in their prototypic location in the engine compartment, and the fan hydraulic drive

was operated from an electrically driven hydraulic pump. In order to overcome the exhaust loss in the air ducting system, the ram-air (suction) blower was operated to produce zero static pressure in the air exhaust manifold surrounding the engine compartment. With the test cell pressure at the ambient reference pressure, this procedure simulated the effect of zero vehicle velocity. Since no attempt was made to obstruct the condenser inlet in the manner of bumper and grille restriction the only "installation" loss was that due to the engine components within the engine compartment and due to the restriction between the engine compartment sheet metal and the "ground line".

Test data were taken at .08, .18, .26 and .35 x 10^6 W (300,000, 600,000, 900,00 and 1,200,000 BTU/hr) heat rejection with .12 - .16 MPa (17-23 psia) steam and inlet air from 287°K to 300°K, (57° to 80°).

The data showed the measured air-side heat exchange effectiveness as a function of air flow. Comparisons were made at the design point value of 76% at 17,237 kg/hr (38,000 lb/hr) predicted by AiResearch, and SES' extrapolation to lower air flows. The results at full load agreed quite well with AiResearch's prediction. At lower load, however, the effectiveness was lower than expected. It was quite possible that the lower than predicted performance was due to temperature measurement error 2.8°K (5°F). (Temperature measurement error would account for almost all of the discrepancy). It was equally plausible, however,

that the observed effect was real. In any event, the measured 80% effectiveness at low and moderate loads imposed no significant system performance penalty.

The total-to-static air-side core pressure drop measurements were recorded. The core outlet pressure tap was located in the wall of the transition duct between the core and the fans. All of the measured values were lower than predicted, and probably evidence the fact that the pressure distribution in the transition duct was highly non-uniform. The fan power measurements suggested that the actual average core pressure drop was higher than indicated.

The indicated fan power at the design point was seen to be slightly higher than predicted despite the fact that there was no bumper or grille loss (this effect was counteracted substantially by there being no ram-air boost). In any event, the power requirements were not far from predictions, and the system performed well in subsequent test activities.

7.0 FREEZE PROTECTION

Initial evaluation of system requirements for freeze protection considered designs of chemical and thermomechanical systems. Feasible candidates for chemical protection were narrowed down to low molecular weight volatile additives (e.g., methanol) which could be added to the feedwater on shut-down and distilled off prior to start-up. There was a distinct trade-off with these systems between start-up time and equipment size. It appeared that only if the system were rarely activated so that a start-up time of the order of several minutes was tolerable, then equipment size could be reduced to reasonable limits. Work on thermomechanical systems concentrated on the system design and critical component design for the freezable sump system. Preliminary analytical designs of insulation, sump penetrations, sump, and thermal check valve were completed. A preliminary system design was performed to determine start-up and shut-down mode.

7.1 CHEMICAL SYSTEMS

Chemical solutions for the prevention of freezing had the advantage that they positively prevented damage by freezing. They can be used in two ways: either left in the water so that they become a part of the working fluid; or separated from the water before start-up. The first would be preferred from the point of view of start-up simplicity.

However, when one examined the list of possible candidates, several problems arose. Any additive which was left in the water had to have high thermal stability to survive the high temperatures in the steam generator. This ruled out organic fluids. Most inorganic fluids which are sufficiently stable are corrosive (e.g., lithium bromide, sodium chloride), toxic or otherwise highly noxious (e.g., ammonia). Many have too high a molecular weight to afford enough freezing point depression without requiring excessively high concentrations. (Others which have a low molecular weight are not sufficiently stable, e.g., ammonia.)

By separating the additive from the feedwater prior to operation more possibilities existed and organics were considered. Two methods of separation were possible: use an additive with low volatility and distill off the water for start-up; or use one with high volatility and distill it out of the water. The former type was not practical since low volatility fluids have high molecular weights so that high concentrations would be required. Also, the heat required to distill out the water was at least seven times as great as that required to thaw the water if it were allowed to freeze. There were, however, numerous possibilities if the latter method were chosen. High volatility was compatible with a low molecular weight requirement. A prime candidate fluid, then would be methanol. It has a low molecular weight (32) so that only 25% by weight would be required for protection to 253°K or -4°F, any methanol remaining in the water would decompose into gaseous components, and only three or four rectification

stages would be required for separation of 92% of the methanol. Moreover, it is a low cost fluid and is relatively innocuous.

Three methanol systems were considered:

(1) A system which would be automatically activated at every winter shut-down, and would therefore not be allowed to delay start-up appreciably. The additive would have to be separated within about 30 seconds.

(2) A system which may be activated only rarely (e.g., twice a month); as for example, if a sump temperature drops to freezing. Two minutes would be allowed for separation.

(3) A system which would activate only as a back-up system if the primary system failed. Ten minutes would be allowed for separation.

Representative equipment sizes and heat loads for the three methanol systems are summarized below:

<u>System</u>	<u>1</u>	<u>2</u>	<u>3</u>
Start-up Time - Min	0.5	2	10
Heat Load - BTU	4500	4500	4500
Condenser Heat Rate - BTU/hr	540000	135000	26000
Column Diameter - cm	43.2	21.6	10.2
Column Height - cm	10.2	10.2	10.2

More complete rectification freezing or suppression below 253°K (-4°F) would, of course, require larger components and higher heat loads.

The conclusions reached from that study were that for all but system 3, equipment sizes (including boiler and condenser) were too large to be practical and that the heat input and the time required for system 3 were both greater than would be required if the water in the sump were allowed to freeze and were thawed out prior to start-up.

7.2 THERMO-MECHANICAL SYSTEM

The design of the thermo-mechanical freeze protection system had to satisfy five main objectives: prevent damage to the system from freezing, permit a normal start-up without any noticeable delay, be fail-safe in the sense that failure to operate properly would result in no permanent damage and the vehicle could be started in a short time (e.g., as in the case of a dead battery), have a high reliability and be inexpensive in first cost and in operating cost. The thermo-mechanical system selected for prime consideration appeared to meet all of these objectives.

The system, shown in schematic in Figure 7-1, consisted of an insulated sump into which substantially all of the working inventory of water was drained upon shut-down. The sump was designed so that it would withstand

FIGURE 7-1
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FREEZE PROTECTION SYSTEM

without damage, as for example, with sloping walls or with a collapsible internal bladder. However, it was insulated with one to two inches of polyurethane foam so that it would not freeze over normal overnight or weekend subzero shut-downs. In the event of extended shut-down, if the water temperature dropped below a present level, a small wick-type burner would be activated to prevent freezing. To cope with the problem of transferring heat to the sump without allowing heat losses through the same heat path, a thermal check valve was used which is described below. Water in the make-up tank or any water remaining in the condenser hot well was allowed to freeze and was thawed out during system operation.

Several different sump shapes were considered: both conical and prismoidal sumps were designed. However, they suffered from the disadvantages that in order to provide sufficient usage at various vehicle angles, a large volume was required, and, having a less than ideal surface to volume ratio, heat losses were greater than with more compact shapes. On the other hand, a spherical sump having a central collapsible bladder to prevent damage from freezing had intrinsically high wall strength, a minimal surface to volume ratio, and was insensitive to angular position. With 5 cm (2") of polyurethane insulation, it would withstand freezing for 38.5 hours at 244°K (-20°F), would have less than 10 watts heat loss at 244°K (-20°F), and could be kept from freezing on less than one gallon of fuel per month under continuous 244°K (-20°F) temperatures.

The sump was to be heated with a wick-type burner similar to a kerosene stove. On demand, fuel would be admitted to the burner reservoir, sized to accept only 10 to 20 minutes worth of fuel. Heat was transferred to the interior of the sump through a wickless heat pipe, shown in Figure 7.2 which acted as a thermal check valve. Heat was transferred as in an ordinary heat pipe by boiling water in the lower bulb, and condensing it inside the sloping tube which was inside the sump. The condensate drained back to the bulb by gravity. When not in use, the water in the bulb freezes, the pressure and density of the vapor fall below their triple point values, and heat is transferred only by conduction along the thin tube walls and through the vapor, and by the slight amount of natural convection to present at the low steam density and adverse density gradient.

The sump is located at the low point in the vehicle. Upon shut-down the expander is stopped, which also stops the expander driven feedpump and condenser fans. The condenser hot-well drains to the sump, and a line by-passing the feedpump is opened, allowing the steam generator to blow down into the sump. The feedpump drain lines open, and it drains back to the sump. After miscellaneous lines are similarly drained, isolation valves close off the sump to prevent migration of water to colder parts of the system.

Upon start-up, the water is displaced out of the sump, either by expanding the bladder inside the sump or through a self-priming start-up pump. The normal start-up

GROSS HEAT INPUT = 4000 BTU / HR.

NET HEAT INPUT = 3000 BTU / HR.

DIAGRAMATIC SKETCH OF HEAT PIPE

FIGURE 7-2

then proceeds. It would not be necessary to pre-warm any of the piping or components prior to start-up since, unless the passage is extremely small or the wall section extremely heavy, the high heat of fusion of water allows only a thin coating of ice to form on the wall. It was expected that the main problems would be in designing valves and other moving parts so that they would not be susceptible to seizure by small pockets of ice. Similarly, the feedpump was designed to be fully drainable.

A preliminary design of the freezable sump system was completed, and work was begun on an elevational schematic of the entire system to study in detail the drainage of every component.

7.3 PRELIMINARY DESIGN

The strongest influences on the design of the system were the geometric requirements of the installation. The only practical space available for the insulated water sump was below and to the rear of the condenser bottom header, as shown in Figures 7.3, 7.4 and 7.5. Fortunately, this location was quite compatible with the physical requirement of gravity feed of condensate to the sump and did not obstruct the flow area of the condenser fan discharge. Both normal system operation and start/stop sequences required the sump to be at the lowest point so that it can take advantage of gravity drainage. It was decided, therefore, to select this location for the base-line design. A more remote location of the sump would be more complicated and

FIGURE 7-3

FIGURE 7-4
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FIGURE 7-5

require either non-return valves and liquid/vapor traps (with the condenser operating at a higher pressure than the sump), or a scavenging pump in the bottom header along with its own freeze protection system (i.e., drain, insulation or rapid thaw device, etc).

The sump is a long slender vessel placed parallel to the bottom header. The feedpump is located almost centrally close to the sump. The nominal height of the suction head provided by the pump remained about 20.3 cm (8 inches) at most attitudes of the automobile.

The size of the sump depended on the size and length of piping, valves, fittings, feedpump capacity, nominal operating capacity of the boiler (which was by far the strongest influence) and the needs of the start-up and shut-down sequences.

Some design work was done on sensors which were used to detect the onset of ice formation; but after review, it seemed that a simpler temperature switch system would be more reliable, and, for the initial design, more expeditious. An ice formation sensor allowed ice to form in the sump, in effect allowing a greater heat loss before the sump would have to be reheated. One problem was that during start-up, the flow of ice cold water to a sub-freezing pump might freeze it. The temperature controlled sump would, at the expense of a higher heat loss rate, hold the sump at a higher temperature, and would warm a feedpump that might otherwise freeze incoming water.

The test plan provided for testing of the following components in a cold chamber:

(1) The feedpump proper with its drain valves installed: its drainability; ability to start when frosted.

(2) A section of the air-cooled condenser: its capability for blowdown and drying.

(3) The sump: the effectiveness of its insulation: the proper operation of its isolation valves; the degree and consequences of vapor migration; the effectiveness of temperature control by the heater (and its relation to cold starts of the feed pump); the consequences of allowing the sump to freeze.

The first testing was done with an electric heater in the sump controlled by a temperature switch. Subsequent testing considered the following:

(1) The blowdown process of the steam generator. It was projected that most of the heat of the water/steam in the boiler would remain in the boiler tubing with the rest rejected by the condenser.

(2) The freezability of, or system blockage by, residual water in low spots or piping. Water remaining in small amounts in the steam generator caused no damage and would be quickly thawed when the burner was fired.

(3) The effects of oil contamination. It was projected that oil helped rather than hindered the freeze protection system since a layer of oil on the water surface in the sump reduced the evaporation from the sump to colder regions. In addition, an oil coating on, for example, the condenser surface allowed water to run off to the sump to a greater degree.

(4) Alternative cross-sectional shapes and materials for the sump. It had to be able to survive freezing cycles in case of heater failure.

The subsystem studied consisted of a specially modified feedpump, a newly designed heated condenser sump, a section of the present air-cooled condenser, and secondary components (piping, valves, etc.). These components were studied as a subsystem and also as individual items.

Figure 7.6 is a schematic of the subsystem, with instrumentation, which was subjected to cold testing. The prototype feedpump was modified to provide drainage of water. The condenser section measured the full height, but reduced width of the condenser. The components were arranged at their prospective vehicle elevations as a closed-loop system, to be placed in the cold chamber.

At the beginning of the test, water was pumped around through the test loop by the pump which was then shut off. Water was allowed to drain by gravity through appropriately placed valves from the pump and condenser to the sump, which was then isolated by valving. The system was

FREEZE PROTECTION TEST LAYOUT

FIG. 7-6

allowed to cool; the temperatures of the feedpump and condenser were allowed to go below freezing, but the sump was kept warm by the immersion heater. An electric heater was used for the freeze test as an expedient rather than a fuel-burning heater, to show the ability to maintain a reservoir of warm water for start-up. The results were the same, and extra complications were avoided in the test equipment. In one set of tests however, the sump was allowed to freeze in order to demonstrate the capability of the design to eliminate damage due to ice expansion.

Figure 7.7 shows a sectional view of the feedpump, and illustrates the operation of the drain mechanism. Upon shut down, the actuator arm was moved to raise the inlet valve, allowing water within the cylinder to drain into the inlet passage and then out. Water on the down-stream side of the delivery valve drained into the delivery passage and then to the sump via the blowdown valve. Initial tests showed that water drained reliably with this system.

Figure 7.8 shows a cross-section of the test sump. A bladder tube ran the length of the sump. Its position in the cross-section was unimportant, since its only function was to prevent complete filling of the sump so as to provide expansion room when ice formed. The bladder resilience allowed the sump to be made of thin-section non-resilient metal which easily handled the normal operating temperatures and pressures.

SECTION THROUGH FEEDPUMP

FIGURE 7.7

CROSS SECTION OF TEST SUMP

FIG. 7.8

Figure 7.9 shows the test fixture that was used to demonstrate the failure of an unprotected tube sample and the successful prevention of damage to a tube protected by the bladder system. An unprotected tube, in one freezing cycle, showed severe formation before rupture (the tube was partly supported laterally by the fixture bolts), subsequently suffering a split for most of its length. A protected tube sample showed no deformation during or after a test under identical conditions.

The cold test chamber, a commercial freezer designed with the low temperature requirements necessary, was in-house and was checked for correct operation. The chamber was able to maintain a temperature near 247°K (-15°F). Slowly varying freezing cycles were obtained by controlling the set point temperature. For rapid freezing cycles, the cooling capability of the chamber was increased cryogenically by the addition of liquid nitrogen.

7.4 SYSTEM TESTING

The insulation selected for testing and eventual use in the vehicle was a commercial grade of expanded polyurethane foam: Dow "Thurane." It has one of the lowest K-factors of the readily available insulations, being better than polystyrene foam, and in addition having a higher rated operating temperature, up to 408°K (275°F). In the area of design, one sump design was selected out of several alternatives and its fabrication completed. The test sump design represented a shortened version (having the same cross-

FREEZE PROTECTION TEST FIXTURE

FIGURE 7.9

section but about 1/3 the capacity) of the actual sump to be used in the vehicle. Differences were in the end fittings and heating method. The framework for the test stand was finished as shown Figure 7.10.

Testing involved examination of the condenser core drainability. The completeness of drainage was measured, and it was found that under normal conditions it would be likely that on the order of 2.5 cm (1") of height of water would remain in the core. Preliminary testing was run involving freezing of the core section, held horizontally and completely filled with water; the result showed that some plastic deformation occurred in each freezing cycle. The test system comprising the sump, feedpump, blowdown valve, condenser section and drive motor was supported by facility power supplies, timers and counters, etc.

The Test Procedure was as follows:

(a) The sump was frozen with its protective bladder installed to ascertain if this was an effective method for the prevention of ice damage. After thawing-out, no measureable damage was observed.

(b) The sump and its attaching pipes and connectors were subjected to ambients of 250° to 247°K (-10° to -15°F) to determine the best transfer rate for electrical heater sizing. The rate was found to be approximately (75 BTU/hr) for a full-size which was an improvement over previous estimates. The ball-type isolation valves remained

FREEZE TEST STAND LAYOUT

FIGURE 7.10

free at the above temperatures, and there was no sign of vapor migration beyond the valves, as the result of overnight cold soaking.

The feedpump itself underwent many "pump-drain-freeze" cycles. In every case, the drain mechanism and sequences worked as intended to rid the pump of the water upon "shut-down". In no case did ice block, or prevent the pump from turning over. The primary problem that was encountered, was one of general stiffness in the pump bearings and seals. Several different facility drive motor arrangements were tried, but none had sufficient starting torque to operate the pump when cold. When this drive motor problem was solved, the cold-priming capability of the pump was determined. Part of the cold stiffness was due to the increased oil viscosity, at the freeze testing temperatures used. Hydraulic transmission oil was tried, and resulted in easier operation than lube oil; however, it did not satisfy the other operational requirements of the feedpump. The original solenoids were of insufficient strength to operate the inlet valve mechanism at low temperatures and were replaced with new direct acting units with satisfactory results.

Tests on the two inch condenser core sections continued. Seventy-eight freeze/thaw cycles were run on the horizontal test section, and thirty-seven cycles were run on the vertical section. The horizontal section, using city water, representing the "worst case" of water hang-up and

and showed a cumulative increase in thickness of about .022 cm (.009") from an initial thickness of .259 cm (.102"), with the largest deformation being at the center. The vertical section, using actual "oily" boiler feedwater, represented the condition of free natural drainage, increased slightly less than .005 cm (.002") along the lower edge. The center position showed no deformation. Testing continued to find the extent of this expansion. This information was used to specify the required drainage surfaces for the design of the prototype condenser.

7.5 FINAL SYSTEM CONCEPT

The basic scheme for protecting the steam system from freezing was to remove the working fluid from those components that would be damaged by ice expansion. The more problematical drainage areas were in the low pressure section of the system. Previous testing confirmed that the various modifications incorporated in the feedpump provided the required drainage and cold-start capability. Adequate draining of the condenser core and bottom header was the goal of the design effort. Four basic condenser problems and their proposed solutions are set forth below:

Condenser core drainage and good heat transfer performance required opposite characteristics. Drainage required a passage of large hydraulic diameter, D_h , to minimize the effect of surface capillary action which retained water within the core; and in the case of the AiResearch condenser, testing revealed that plastic deformation of the core occurred due to ice formation from residual

water. For this case, the passage thickness was 1.2 mm (.046 in.), as was D_h , and it was observed that water remained in the core to a height of about 3.8 cm (1.5 in.) from the bottom edge. Because of the multitude of fins crossing the passage, it was difficult for any water/ice mixture to extrude from the core, thus causing more deformation. Better heat transfer performance was achieved by reducing D_h ; thus, a compromise was required. Preliminary analysis showed that enlarged passages in the shape of flattened tubes and the elimination of fins could reduce the capillary height to about 10.2 mm (.4 in.). Also the absence of fins allowed the freezing slurry to extrude.

Other methods of reducing the capillary height were evaluated, one being the use of wicking to draw the water from the core. Geometric changes were considered to prevent plastic deformation irrespective of capillarity. For example, the passage side might be extended into the bottom header the fins removed, allowing elastic deformation when ice forms.

A related problem was to allow start-up of the system even when water droplets had frozen in the bottom edge of the core, thereby plugging the condenser. The proposed solution was to install a by-pass tube, a passive element, from the condenser inlet pipe to the bottom header. This directed exhaust steam to the underside of the frozen region at start-up, providing both quick thawing and ready

elimination of air and/or condensate from the condenser through one of the steam traps connected to the bottom header.

The significant problem in relation to packaging the condensing system in the vehicle was the drainage of the bottom header when the vehicle was tilted. Since the steam system required the maximum amount of condenser frontal area, practically the entire width and height of the front region of the engine compartment had to be used. The result was that the bottom header became long, and any transverse tilting caused water to gather at one end and back up into the core. Any attempt to provide a deeper or triangular header resulted in a loss of heat exchanger area. The proposed solution also satisfied the fourth problem, that of operational drainage. It was required that all condensate be delivered from the condenser, even from low corners, and that start-up not be prevented due to blocked and frozen outlet tubes. The freeze protection concept, therefore, consisted of what was called the three-trap system.

Instead of using gravity drain tubes at each end of the header, which would need check valves to prevent back-flow into the header (and low point drainage is not ensured), three drain tubes in conjunction with a steam trap of the thermostatic type on each tube were connected to the bottom header. The important point was that the tube inlets were located in different planes. Specifically, one inlet was placed at each end of the header, and the third at the

FREEZE PROTECTION SYSTEM SCHEMATIC

FIGURE 7.11

center of the header in an elevated position with respect to the others. Figure 7.11 shows the schematic of the system, and Table 7.I details its start-up and shut-down sequences.

Figure 7.11 also includes two new proposals: the relocation of the hotwell, and the addition of a stop valve at the boiler outlet to aid in the blowdown sequence. Because the condenser was no longer drained by gravity but by steam traps which use condenser pressure to transfer condensate to where it was desired, the previous restricted location requirement no longer existed. Thus, the hotwell was elevated, which aided packaging and enabled a heating system to have more freedom in its design.

The stop valve between the boiler and the engine was required not only for reasons of safety, but also to allow a time period between system shut-down and the moment of blow-down. Not only was system hot-start time reduced, but a reduction of fuel consumption was possible. In the former concept, when the burner was extinguished the engine would have continued to run until blowdown pressure was reached. In the current scheme, steam remained in the boiler until blowdown pressure was reached by natural cooling, this process taking considerably more time than the one above. Because a stop valve was employed, it was possible to blow out the boiler and feedpump in the event of an aborted start.

TABLE 7-I

PROPOSED LOW PRESSURE SYSTEM OPERATION

START-UP

Isolation valves open
Blow-down valves shut
Feed pump begins operating,
check valve allows priming.
Water flows to steam generator.
Subcooler circulation pump may start.
Exhaust steam enters condenser
By-pass tube allows thawing of blockage,
if any, of bottom core.
Steam traps allow air and condensate to be
blown to hotwell using condenser pressure.
Sub-cooling water mixes with condensate to
maintain hotwell at desired temperature.
The three steam traps ensure condensate
delivery at any vehicle attitude.
Cooling fins on hotwell vent line recover
water from venting vapors.
Atmospheric pressure hotwell eliminates
feed pump inlet pressure variations, en-
gine crankcase pressure build-up.

SHUTDOWN

Burner stops, throttle (stop) valve shuts.
When boiler pressure drops to, say 700 kPa
(100 psi), feed pump valves open, water is
blown from boiler and feed pump back to hot-
well.
Hotwell outlet (lower) isolation valve shuts,
steam or supplemental air continues to clear
feed pump, subcooler circulation pump, and
subcooler. Air leaves via vent, water goes
to hotwell.
Residual water at and below steam traps drains
back into condenser bottom header.
Hotwell inlet (upper) isolation valve shuts.
The three non-coplanar traps allow start-up -
one is always open. Exhaust steam, entering
the bottom header via the by-pass line on
start-up, thaws any ice in this region.

A tank of greater volume and of a more nearly spherical shape has a proportionately low cooling rate due to the reduced surface to volume ratio. Utilizing this shape and a high-performance insulating method, namely superinsulation, (which uses multiple layers of aluminized Mylar between the double-walls of a Dewar type vessel) resulted in cool-down times on the order of fourteen days, at an ambient temperature of 255°K (0°F). In addition, the heat loss rate was low enough, about 1W at the above ambient, that electric heating from the vehicle's battery was feasible, in which case the complications of a combustion heater were eliminated. If the more conventional insulations are used, the elevated hotwell would have made insertion of a heat-pipe freeze prevention system significantly easier. For proper performance, the heat pipe had its source below its delivery point, virtually impossible in the former system.

Several alternative designs were made for the sump container. Since vacuum insulation (together with aluminized mylar superinsulation) has significant advantages over urethane foam insulation, its use allowed more freedom in locating the sump. Previously, a sump insulated with urethane foam had been designed and a sample section tested; it was intended to be placed just below and to the rear of the condenser. Although this location had the advantage of gravity drainage, the long, slender tank was far from optimal from an insulation point of view. In addition, the inclusion of a sump heating system became difficult because, for example, a heat pipe requires components that would have to be located below the sump creating a ground clearance problem.

However, if a high performance insulation (vacuum plus mylar) is used, the tank's shape is no longer important since the connections now become the dominant factor for heat loss. Also, since this insulation system is so effective, electric heating becomes feasible, eliminating the packaging problem and, more importantly, eliminating the need for a fuel-burning heating system altogether.

An elevated sump located behind one fan hub was the preferred alternative to the sump location described above. The fan hub location was selected since;

- (1) A tank of greater capacity was possible.
- (2) Gravity priming of the feedpump was possible.
- (3) A fuel burning heater was allowable should urethane insulation be used.
- (4) The hub location was more protected from road obstacles.

Gravity drainage to the sump was no longer necessary since, during operation, a positive pressure condenser and an atmospheric pressure sump was used, and a steam or air-supplemented blowdown was required after shut-down.

In the system shown in Figure 7.11, steam (or air admitted to the system at the boiler outlet) forces water out of the steam generator back through the exhaust desuperheater and the feedpump to the sump. The line between the sump and the pump is of larger diameter, about 25 mm (1

in.), so that air can change places with water and the line will drain. After the outlet isolation valve is shut, the remaining water is then blown through the subcooler to the sump inlet. Finally, the inlet isolation valve is shut and the freeze protection system is in operation.

In conjunction with the positive blowdown, the feedpump was redesigned to allow direct reverse-flow blowdown, thus eliminating the need for the separate blowdown valve that was part of the old design. The system uses a solenoid and linkage to lift the three outlet valve balls. Along with the existing inlet valve lifters, provision was thus made for reverse flow through the pump.

One of the first studies of vacuum-insulated sumps considered a "wide-mouth" design. Its advantages were simplicity and accessibility, but it could not reduce the conduction heat losses to the low level required if electric heating was to be used. This requirement was met by the present design which consisted of a complete double vessel with "pant leg" inlet and outlet ports. Solenoid valves were designed for these ports with the valve element located inside the tank and the solenoid coil outside. The significant feature was the ability to disassemble the valve and withdraw it from the tank in case of trouble. The heat loss rate was recalculated on the basis of the actual design, and it came to about .2 watts (.6 BTU/hr) for a T of 22°K (40°F).

To make up the loss for freeze prevention, an electric resistive element and a thermal switch were selected for mounting within the tank. Originally a thermister was to be used with its control logic (on the exterior and adjustable) to operate the heater, but by using a third "pant leg" a much simpler system evolved which allowed access to an adjustable mechanical temperature switch. The heater was a nominal 500 watts at 120 volts giving rapid thawing when plugged into house current, and it gave 5 watts when connected to the vehicle's battery for regular freeze prevention. The cycle time for the latter was approximately 6 hours "on" followed by 140 hours "off".

The cold chamber for freeze protection testing (a Bally Walk-In Freezer) was delivered and has interior test space of 1.6 m by 2.2 m with 2.4 m height (5'2" x 7'1" x 7'10"). Check out of its operation showed fast cool-down. The cooling unit within the chamber had a strong fan which decreased the time required to chill test components.

Completion of fabrication and testing of the final system design was precluded by termination of the prototype phase.

8.0 PREPROTOTYPE CONTROL

Some of the important factors influencing the market acceptance of new engines for passenger automobiles are responsiveness, ease of operation and smooth handling. All these features together are sometimes called driveability or controllability of the engine. Driveability in general does not lend itself to a mathematical description, its design and performance evaluation are, therefore, very subjective and difficult to assess in engineering terms. In order to ensure reasonable driveability, SES has attempted to follow automotive practices wherever possible. The resultant powerplant has good driveability, in the sense that it can follow the transitory Urban Federal Driving Cycle. The steam car driver "feeling" however is different than an I.C. engine driver. In the steam engine a fixed pedal position calls for a constant power whereas in an I.C. engine a pedal position calls for a certain torque.

The second goal of the steam engine control system was to maintain the high engine steady state performance at all times. In particular, with the measured steady state emission characteristics, the steam engine has the potential of emitting less than half of the original 1976 emissions standards for CO, UHC and NO_x. As can be seen in Table 2.II this goal was not fully met. However, SES is very encouraged by the fact that its engine is the first engine in the higher vehicle weight class to meet the 1976 emission standards, and is confident that with further controls development the original goal can be accomplished. In a further development of the SES steam engine for a coal

haulage vehicle inside deep mines, SES has utilized know-how accumulated throughout the simulator test to modify its burner control and is presently achieving much better steady state and transient emissions characteristics with its Mod #9 steam generator and Mod #3 control system. It should be noted that the original steam car contract called for the development of Mod #3 control system on the simulator vehicle. Unfortunately, however, when the contract was modified, the scope was narrowed down and the Mod #3 control system was eliminated.

The initial development of Mod #0 control system was subcontracted to Bendix Research Laboratories of Southfield, Michigan. The specific items covered by their work were:

1. Fuel Control Unit for Burner. This was a standard Bendix RSA fuel injector (pressure carburetor) used on aircraft engines. It provided metering of fuel flow in proportion to the air flow passing through the venturi. The metered fuel flow would be delivered to the fuel nozzle of the SES burner. Bendix made the necessary modifications to this unit to provide the desired turndown range and schedule of fuel/air ratio versus output.

2. Air Throttle Valve for Burner. The combustion air blower would be throttled to control air flow to the burner. Bendix provided a servo-motor actuated throttle valve and a control circuit that operated the valve with anticipatory control sensing steam flow, and with a feedback trim control sensing superheater steam discharge temperature.

3. Feedwater By-pass Control Valve. The feedpump would always run at a capacity in excess of that required by the expander, and the automatic by-pass valve would divert excess flow back to the pump inlet to maintain fixed steam outlet pressure. This valve was designed and built by Bendix.

In addition to these hardware items, Bendix also performed control system design analysis using the hybrid computer system simulation. Several design details of the control hardware were developed by means of the system simulation model.

Testing of the Bendix control system had revealed many inadequacies. The steady state, as well as dynamic accuracy was very poor; the system was unstable at some loads and uncontrollable at others. Nevertheless, in the process of testing and analyzing the system errors, much was learned about the dynamic behavior of the system and it served as a starting point in the design of Mod #1 control system. It was also learned that a good steady state accuracy and controllability must be achieved before attempting to develop dynamic accuracy and responsiveness.

The Mod #1 control system was designed to enable steady state operation and testing. At that stage of the engine development, the Mod #5 boiler was still employed and it was too premature to devise a highly responsive control system with such a slow component in it. The responsiveness of Mod #5 boiler is illustrated by Figure 4.16. The primary features of the SES Mod #1 control system are:

1. Combustion air metering by a four flapper metering valve whose position is scheduled as a function of the blower speed. A detailed discussion of this valve is given in section 4.1.1.

2. An electrically driven variable speed fuel metering pump. The fuel flow is scheduled as a linear function of the blower speed.

3. Feedwater flow control by unloading solenoids that unload each of the pump cylinders individually. Flow control is achieved by cycling these solenoids once a second while varying the ratio between the loading time and the unloading time.

4. A "passive" normalizer. At that time it was already understood that a fractional evaporator by-pass helps the boiler controllability and responsiveness. This fraction was set by a fixed orifice.

5. Heat Balance Controller. Figure 8.1 shows schematically how the accelerator pedal and the steam pressure dually control the cut-off angle. Depression of the accelerator pedal would cause the cut-off angle to increase. If the steam flow was such that the pressure deviates from 6.9 MPa (1000 psi), the bellows would cause the cut-off angle to change to reduce the deviation.

6. A Start-Up Sequencing System. This subsystem automatically controls the start-up of the engine from burner ignition to expander cranking and is aimed at getting the engine to be self sustained in the minimum amount of time.

FIGURE 8.1

Testing of Mod #1 control system was carried out with a varying degree of success. The burner control by a scheduled air metering valve and a variable speed fuel pump was found to have sufficient steady state accuracy. The feed-water control by unloading pump cylinders had not resulted in any damage to the pump. The start-up sequencing controller could start the engine in 30 seconds from room temperature and 5 seconds when the boiler is hot. It turned out that the heat balance controller tends to compete against the driver, preventing him from idling the expander when the boiler pressure is high. Consequently, this function was taken off and during the preprototype tests the heat and power balance were performed manually. It was also found that a passive normalizer cannot provide sufficient accuracy in controlling the steam outlet temperature. Above all, it was reconfirmed that a much faster boiler is needed in order to provide a responsiveness that would be comparable to a modern passenger automobile.

Figure 8.2 presents a functional diagram of the engine system and the Mod #2 broad control strategy. The power control package takes into account the driver command (foot pedal position) and expander parameters: speed, intake pressure and exhaust pressure, and controls the cut-off angle to provide smooth engine response and gear shifts. At the same time, the power control package sends a command to the burner and boiler such that the energy balance, steam generation in the boiler must equal steam consumption in the expander, is satisfied.

FIGURE 8.2

STEAM POWERPLANT SCHEMATIC DIAGRAM

The burner control package controls combustion blower speed, combustion air valve position and a fuel metering pump and tries to maintain the desired firing rate while keeping the air to fuel ratio within the tolerance limits for emissions, shown in Figure 8.6. The steam generator control package has the task of maintaining heat and mass balance inside the boiler. The coil temperature is sensed in several locations; the feedwater flow and normalizer injection are controlled to provide the desired temperature distribution.

The condenser controller is an independent mechanical-hydraulic package that controls the condenser shutters and fans speed to maintain the condenser pressure between 15 and 35 psia while making full possible use of available ram air.

In addition to the functions described in Figure 8.2, SES has also developed automatic start-up and shut-down sequencing controls. On start-up, the boiler is operated by an external power source "starter" and steam is brought up to a high temperature and pressure before it is admitted into the expander. On shut-down, all water in the system is blown to the condenser sump which would be insulated and heated to provide freeze protection.

8.1 POWER CONTROLS

The following requirements have been taken into account in the design of the power control subsystem:

1. The boiler must be kept within safe and operable temperature and pressure limits. Since the boiler is the slowest component of the steam engine, this requirement dictates continuous monitoring of the load and processing the information immediately in the boiler control package to avoid large unexpected steam flow changes.

2. For the same reason, the steam consumption (and power level) must be insensitive to engine speed changes. Otherwise the power output behavior will confuse the operator.

3. The system must respond immediately to power demand changes.

4. The system must be capable of idling at a very low steam consumption in order to achieve economical operation over the Federal Urban Driving Cycle.

Figure 8.3 shows how these four requirements were incorporated into the control system. The right hand side of the figure is a block diagram representation of the engine dynamics. The expander power is determined by cut-off, speed, steam pressure and condenser pressure. The controller takes into account a first order approximation of the power level and steam consumption as follows:

$$SF \propto \text{Power} \propto SP \times N \times (\text{Cut-off} - k \times CP) \quad (\text{Equation 8.1})$$

Where SF is the steam consumption rate, SP is the steam pressure, N is the expander speed, CP is the condenser pressure which is also equal to the exhaust pressure, and k is a factor representing the volume of steam trapped inside the expander cylinder when the recompression begins. Since the first requirement insists on knowing the load (power and/or steam flow) at all times to avoid unexpected mismatch of boiler-expander heat balance, the cut-off must be controlled by the relationship given in Equation 8.2 in order to satisfy Equation 8.1:

$$\text{Cut-off} = \frac{k_1}{SP} \frac{SF}{N} + k_2 \times CP \quad (\text{Equation 8.2})$$

Referring again to Figure 8.3, it can be seen that the power demand is transferred directly to the burner control logic box which will regulate the firing rate in direct proportion to the pedal position. Thus, the steam generation rate is primarily controlled by the driver's pedal position. On the other hand, for the purpose of cut-off control, the power demand is conditioned as required by Equation 8.2; it is divided by the engine speed and added to the condenser pressure with a scale factor.

The division by steam pressure, however, is eliminated in order to enable inherent corrections of small errors in heat balance: i.e., when the steam pressure is low, the expander consumes less than scheduled steam flow and vice versa. Moreover, further corrections of heat balance errors are done by a trim controller that senses the steam pressure and changes the firing rate command to eliminate the error.

Section 3.0 describes the cut-off mechanism in detail. Basically, the mechanism is a rotary actuator that maintains the cut-off cam at a certain angle relative to the admission cam. This angle is controlled by a linear displacement of a lever, located at the front end of the expander. A servo motor is used to displace the cut-off actuator lever.

This basic control scheme has been operated in the preprototype configuration and later in the simulator. It has demonstrated good controllability and smooth gear shift operation.

Upon a gear shift, the expander speed change automatically changes the cut-off via the demand signal conditioning of Equation 8.2 such that the expander steam consumption, boiler steam generation and power level before and after the shift remain the same.

Although the system "feels" different than an I.C. engine with power output that is proportional to the pedal position, the driver is not inconvenienced. His feeling is one of a highly responsive powerful system at moderate and high power levels. At very low power levels, however, the cut-off mechanism can no longer control the steam admission period and the power must be controlled by a steam throttle valve in the same way as it is done on fixed cut-off machines.

After an extensive search for suitable hardware, SES had decided to develop its own lightweight and compact steam valve, that uses high pressure feedwater for actuation and cooling of the steam as shown in Figure 8.4.

The steam valve has been tested in operation on the simulator with moderate success. Some galling developed between the stem and valve body. As explained in the previous section the controls development effort had been stopped at that point and, therefore, no friction reducing coatings and/or other modifications were tried. Nevertheless, the steam throttle valve had served its purpose on the simulator, enabling idling of the steam engine and very low power level operation.

In Figure 8.3 it can be seen how the throttle valve control is incorporated into the power control logic. At very low and "foot-off" power demands the steam pressure is throttled to 400 psi. At higher demands the throttling action is gradually reduced until at a pedal position of 15% (approximately 20 hp) the throttle valve is held wide open and the cut-off control effectively takes over. The addition of this "foot-off" throttling feature enabled a very smooth engine operation over wide changes in operating conditions varying from 0-120 shaft hp and 7-100 lbs/hr of fuel flow.

With a decision to use the transmission without the torque convertor, the power logic was further modified to prevent the expander from stalling upon clutch engagement. This so-called idle speed controller is shown on the upper

STEAM THROTTLING VALVE

- SCHEMATIC -

FIGURE 8.4

left corner of Figure 8.3. When the expander speed drops below 10 Hz (600 RPM), the power level is commanded to increase sharply. At any higher speed the idle speed controller remains inactive, allowing the normal control scheme to regulate the speed and power without interference.

From a controllability point of view the power control subsystem has demonstrated a satisfactory performance on the simulator, regulating and balancing the burner-boiler-expander system over the Federal Driving Cycles without stalling the engine and without boiler overpressurization.

8.2 BURNER CONTROLS

The functional operation of the SES burner is shown in Figure 4.2. Fresh combustion air enters through the center of the combustion chamber and mixes with uncooled combustion products to produce a final mixture temperature of 390-450°K (250-350°F). The hot air is sucked through an air metering valve into the mixing chamber in which the finely sprayed fuel is evaporated. The mixture is then blown by the combustion air blower through the flameholder into the combustion chamber.

As can be seen, the burner system has the possibility of three control inputs: blower speed, inlet air throttling and fuel flow while only two parameters are controlled: firing rate and air/fuel ratio. In that sense the system is redundant. In the initial development stages it was intended to use a fixed drive ratio between the blower and expander, and regulate air flow by varying the throttle valve only.

However, SES was unable to devise a control system that would be accurate and fast enough to respond to the fast speed changes during engine gear shifts. Consequently, it was decided to utilize a variable ratio hydrostatic drive for the blower. With the variable ratio drive system, there is no longer an incentive to use a complex relationship for positioning the air valve to control the air-to-fuel ratio. Nevertheless, the air valve has proved very useful, contributing stiffness to the burner system at light load. Note that with the air intake being under the hood, each time the condenser fan changes speed a pressure disturbance is introduced and an error in the air-to-fuel ratio occurs. Thus, only a stiff air system can eliminate emission spikes during a highly transitory driving cycle.

The basic burner control logic is given schematically in Figure 8.5. Upon receiving the required firing rate signal from the power control package, the signal is passed through a rate limiting device that produces input-output relationships as follows:

$$\frac{dy}{dt} = \begin{matrix} \alpha y & @ & y < x \\ 0 & @ & y = x \\ -\alpha y & @ & y > x \end{matrix} \quad (\text{Equation 8.3})$$

Where x is the firing rate demand, y is the blower speed command, α is a fixed coefficient and t is the time in seconds. From the controllability point of view, it is desirable to choose α as large as possible in order to transfer the power demand immediately and avoid a heat balance discrepancy. However, due to our inability to completely eliminate emissions spikes during fast transients, α has to be a finite number. On the simulator vehicle, α was selected to be 3 sec^{-1} as a reasonable compromise between burner

FIGURE 8.5
BURNER CONTROL LOGIC

emissions and system responsiveness. A τ of 3 sec^{-1} permits the burner to receive a command step from idle to maximum firing rate in one second.

The blower speed controller is a fast electro-hydraulic controller that varies the drive ratio between the expander and the blower such that the blower speed varies linearly with the command between 33 and 100 Hz (2000 - 6000 RPM) regardless of the expander speed. A further description of this controller will be given below.

The actual blower speed is sensed by a magnetic pick-up type sensor. This signal is used to control the fuel metering pump such that the fuel flow varies linearly between 2.3 kg/hr (5 lbs/hr) at 33 Hz (2000 RPM) and 45 kg/hr (100 lbs/hr) at 100 Hz (6000 RPM). The fuel pump is a very accurate metering unit.

The air valve positioning schedule is a diode break function generator (DBFG) whose profile can be tuned to yield a desired air-to-fuel ratio. On the simulator, the optimum schedule, shown in Figure 8.6 was used. Note that at low firing rate, the air valve is now nearly shut, making the air metering stiff enough to permit idle operation of the vehicle in spite of the varying intake pressure. The valve is actuated to its scheduled position by a servo motor.

A photograph of the air valve inside the intake duct is given in Figure 4.4.

FIGURE 8.6

MIXTURE RATIO CONTROL WINDOW REQUIRED TO MAINTAIN
EMISSIONS BELOW 1976 STANDARDS AT ALL FIRING RATES
(ASSUMING CONSTANT 1.0 MPG FUEL ECONOMY)

Figure 8.7 shows a schematic diagram of the blower drive. An expander driven, variable stroke hydraulic pump is used in conjunction with a fixed displacement hydraulic motor. The drive ratio is controllable by a hydraulic actuator that varies the pump stroke. Specifications of the hydraulic components of the system are listed in Table 8.4.

The electronic controller of the blower drive performs the following function:

$$V = k_1 (D - N_b) - k_2 P_h - k_3 \frac{d P_h}{dt} \quad (\text{Equation 8.4})$$

Where D is the speed demand, P_h is the hydraulic pressure, dP_h/dt is the rate of change of the hydraulic pressure, N_b is the blower speed, k_1 , k_2 , k_3 are constant coefficients and V is the servo-valve command. The first term in Equation 8.4: $k_1 (D - N_b)$ is aimed at satisfying the driver's command; the pump's stroke actuator keeps actuating the stroke until the actual blower speed approaches the demand. The second and third terms are designed to shield the blower from expander speed changes and eliminate emission spikes during gear shifts and fast accelerations. On a down-shift, the pressure tends to rise due to the increased expander speed. These changes are picked up by the proportional-derivative terms and a correction of the pump stroke is made before a significant blower speed change occurs. The blower speed controller has demonstrated excellent performance and reliability on the steam engine. Figure 8.8 is a recorder chart that shows how the blower speed follows the demand very accurately while being almost totally insensitive to the drive speed.

BLOWER DRIVE SCHEMATIC

FIGURE 8.7

TABLE 8.1

BLOWER DRIVE SYSTEM CONSTANTS

Blower: J - Inertia $\hat{=}$.00359 Newton Meters sec²
 - Torque Gradient = 0.025 Newton Meters/Hz

Pump: Lucas Variable Displacement
 Dp - Maximum Displacement = 0.725 cm³/Radian

Stroke Control Piston: Bore = 1.905 cm
 Rod = 0.098 cm
 Travel = 0.843 cm (with stop)
 A - Area = 2.85 cm²

Servo Valve: Supply Pressure = 2.067 MPa (300 psi)
 Maximum Flow = 62.94 cm³/sec
 Maximum Current = 8 MA
 Maximum Voltage = 10 Volts

Pressure Transducer: Maximum Pressure = 13.78 MPa (2000 psi)
 Supply Voltage = 5 VDC
 Output Voltage = 0.1 Volts/13.78 MPa

Hydraulic Motor: Hyrevz
 D_m - Displacement = 0.2426 cm³/Radian

Control Constants:

L - Leakage Gradient = 5.495 cm³/sec/MPa
 B - Bulk Modulus = 344.5 MPa
 V - Entrained Volume = 86.87 cm³
 K₁ - Gain = 0.34 Volt/Volt
 K₂ - Gain = 6.0 Volt/Volt
 K₃ - Gain = 0.939 Volts/MPa
 T - Lead Time Constant = 0.0825 Sec

Blower Drive Tests 12/10/74
Cold Test, Air Valve on schedule
Drive Motor Speed changes; Blower Speeds = 66.7 & 83.3 Hz

Error Signal

Chart Speed 5mm/sec. →

80%

Firing Rate Demand Signal

0%

BRUSH ACCUCHART

13.1 MPa (1900 PSI)

Hydraulic Pressure

0

41.7 Hz (2500 RPM)

Drive Motor Speed

0

Figure 8.8

100 Hz (6000 RPM)

Blower Speed

16.7 Hz (1000 RPM)

8.3 BOILER DYNAMICS

The development of the fast, responsive and controllable Mod #8 SES boiler has been one of the most crucial milestones of this contract. Extensive dynamic studies were carried out in order to understand the parameters that influence the response of the boiler. Basically the initial study that was done on the Mod #5 boiler had revealed a common phenomenon: at light load most of the heat transfer is done in the last three passes. As a result, the two economizer passes and the first evaporator pass are relatively cool at light load. On the other hand, at a high load, the heat load is more uniformly spread out among the passes, and the economizer and evaporator are significantly hotter. Thus, a step demand in the steam flow is not realizable before the heat exchanger mean temperature is elevated.

A computer program was developed in order to calculate the amount of energy that needs to be pumped into the heat exchanger coil before a large step in the steam flow can be materialized. The program predicted a difference of 3000 BTU between the heat content of the coil at maximum and minimum firing rate having a maximum firing rate of 500 BTU/sec, we conclude that a dead time of six seconds is the best realizable dynamic performance of Mod #5 steam generator. Moreover, if an attempt is made to increase the steam flow before six seconds have elapsed, the delay will be extended even further. Figure 4.16 shows how well the

STEP IN LOAD DEMAND: 17% - 70%
MODEL NO. 5 BOILER

FIGURE 4.16

computer results agree with actual test data. Note that in that Figure, the difference between the line of fuel flow (energy input) and the line of steam flow (energy output) is the rate of energy flow into the coil. An integration of that area reveals that it amounts to eight seconds worth of dead time, somewhat longer than the ideal predicted delay. Further, the computer program agrees very well with steam and air side temperature measurements along the coil.

After the response problem was fully assessed and understood, the second phase of the study explored a variety of changes in the coil configuration in order to minimize the tubing energy changes between high and low load. It was found that a combination of the following steam generator coil changes can theoretically reduce the coil energy changes between low and high load to zero, thus yielding a fast and responsive boiler:

1. Lightweight economizer. The Mod #8 economizer was built from MODINE tubing which is made of aluminum fins brazed on a stainless steel tube. In addition to the obvious energy reduction due to the light weight of aluminum, the thermal conductivity is improved, the fins are colder at high load, and the result is a very significant energy reduction.
2. Quench pass. At light load the economizer exit water temperature is in the order of $175-205^{\circ}\text{C}$ ($350-400^{\circ}\text{F}$). If we let the water flow directly to the evaporator it will keep the first, heavily

finned, evaporator pass cool, and a delay will be caused when the load is stepped up. The solution that was incorporated on Mod #7 and #8 boilers is a quench pass - the closest pass to the flame. The quench pass is a bare tube pass that enjoys a very high relative heat load at low firing rate and vice versa. The water flows from the economizer directly into the quench pass and then into the evaporator. Thus the first evaporator always receives boiling, 285°C (545°F), water and its temperature difference between light load and high load is small.

3. Evaporator by-pass injection (EBI). By-passing sections of the heat exchanger at low load causes the by-passed sections to stay hotter. The Mod #8 boiler EBI uses a variable by-pass of between 10% at light load and 2% at high load. The section mostly affected by this operation is the second, so-called "evaporator" which becomes superheated at light load. This important feature actually stores usable energy at light load. When a step in the demand occurs, saturated steam water mixture is pushed into the second evaporator and, using the stored energy for evaporation, produces large amounts of steam to meet the demand.

Figure 8.10 shows a schematic diagram of the Mod #8 coil that incorporates the above three features. SES has measured excellent response times on this boiler. In fact, with the improved boiler response, the delay function

of the rate limiter, that limits the rate of firing rate increases as discussed in Section 8.2, became very noticeable. On the simulator, the boiler response was limited only by the burner response and not by its own energy difference. Figure 8.9 illustrates the Mod #8 steam generator response to a demand step. For all practical purposes the steam flow response is instantaneous.

FIGURE 8.9

MEASURED STEAM FLOW RESPONSE TO
A STEP IN FUEL FLOW AND AN OVER/
UNDER SHOOT IN WATER FLOW

8.4 BOILER CONTROLS

The responsive boiler in itself does not guarantee constant steam temperatures. In fact, since the boiler is so fast, any mismatch of energy balance can cause a temperature error that develops fast. Thus, some control mechanism that measures the quality of energy balance and corrects the water flow and/or firing rate must be incorporated to avoid such errors. The boiler steam outlet temperature is of particular interest because a high temperature error at that point can cause extensive damage to the expander.

In the Mod #1 control system the energy balance mismatch was measured by sensing the outlet temperature and the temperature was controlled by changing the feedwater flow. This control method was extremely sluggish and could not handle large load steps without high temperature errors. It was later understood that pouring water into the boiler does not cool the superheater unless steam is generated at the same time. Thus, a long delay between an increase of water flow into the economizer and a consequent reduction of steam temperature is unavoidable and the usage of such a control function to regulate the steam outlet temperature cannot yield satisfactory dynamic accuracy.

With the decision to change the coil configuration and use a normalizer, the Mod #2 control scheme uses a strategy of injecting normalizer water to control the superheater outlet temperature. The injected water evaporates immediately, reducing the superheater inlet temperature to whatever is required to hold the outlet temperature constant.

The injection flow is controlled by a servo valve on the EBI normalizer. This control strategy has been tried manually during the PPT testing and was found very successful. The operator could correct expander inlet temperatures almost instantly regardless of the evaporator outlet temperature. A further improvement of the outlet temperature regulation was achieved by sensing and controlling the superheater inlet temperature to a constant 620°K (650°F). Since the temperature sensor is located close to the injection point, there is no delay whatsoever in the control loop and a very tight control can be exercised. This scheme allows the superheater outlet temperature to "float", being 870°K (1100°F) at light load and 755°K (900°F) at high load. Such temperature behavior is not undesirable because it increases the part load efficiency. The elevated light load temperature would not damage the expander whose cylinder walls are relatively cool at light load; on the other hand, it helps to reduce the cylinder wall temperatures at high load.

Figure 8.10 shows how the normalizer and superheater inlet temperature control are incorporated into the boiler control logic. It can be seen that regardless of the temperature at the outlet of the last finned pass (so-called second evaporator), the superheater inlet temperature is kept constant by the EBI operation. Thus, the problem of steam temperature at the coil outlet, that was previously related to the interface between the boiler and expander, is now shifted in location to the evaporator outlet and becomes an internal problem of the boiler. There are several advantages associated with transferring this sensitive degree of freedom upstream into the boiler:

STEAM GENERATOR TEMPERATURE CONTROL SCHEMATIC

FIGURE 8.10

1. The boiler does not have any high temperature moving parts and thus, it can tolerate higher temperature errors than the expander.
2. The desired evaporator outlet temperature varies between 755°K (900°F) at low load and 645°K (700°F) at high load for good responsiveness. These temperature levels are lower than the boiler outlet temperature.
3. The response time of the evaporator outlet temperature is shorter than that of the superheater outlet temperature because the coil is shorter.

The control logic of Figure 8.10 uses the evaporator-outlet temperature to influence the boiler feedwater flow, trying to maintain the temperature schedule of 755°K (900°F) at light load and 645°K (700°F) at high load. This control logic was tested in the preprototype configuration and found very satisfactory at steady states and small power steps. At large power steps however, there is still the problem of a relatively long time delay between a change in water flow and a resultant change in evaporator outlet temperature, and a faster control action is needed to maintain the temperature errors within 385°K (200°F). Hence, it was recognized that a measurement of the evaporator outlet temperature can only be used for long term accuracy but not as an instantaneous measure of the quality of heat balance between firing rate and water flow.

Having the tool to correct slow heat balance errors, the problem is now further reduced to a measurement whose only important feature is a quick response to changes in the heat balance, without a requirement for absolute accuracy. It should be noted here that these types of dual measurements: one for long term accuracy and another for quick response are common practice in the control field. A simple example of such a device is the proportional integral controller that uses the integral (reset) action to correct slow droops and the proportional action for responsiveness.

As shown in Figure 8.10, the Mod #2 boiler control system uses the economizer outlet temperature as a quick (if inaccurate) measure of heat balance. Being filled with incompressible fluid, the economizer responds very rapidly to changes in the water flow rate and thus it enables a correction in the water flow to be made long before this error reaches the evaporator outlet temperature sensor, resulting in smaller temperature errors and a very smooth operation over the tested driving cycles.

A further improvement in the feedwater control logic was achieved with the incorporation of a lead-lag function. Upon a demand step, the water flow increases by 50% more than the proportional amount necessary for heat balance. The flow stays high for about five seconds and then drops to a deficient flow and stays at that low flow also for about five seconds. After this lead-lag operation, the flow is allowed to go to its steady state level to maintain the boiler heat balance. The excess water flow

helps push saturated steam inside the boiler onto the superheated evaporator surfaces, instantaneously generating enough excess steam to meet the power demand. Such a lead must be followed by an undershoot in the water flow to correct the heat balance error generated by the lead. The influence of a feedwater lead-lag function on the responsiveness of the boiler is shown in Figure 8.9. For all practical purposes, the boiler with its Mod #2 control scheme has an instantaneous response.

8.5 CONDENSER PRESSURE CONTROLLER

In a totally condensing steam engine system, the importance of the condenser in maintaining the heat balance of the steam engine train (pump, boiler/burner, expander and condenser) should not be overlooked. A mismatch in the condenser heat balance would cause the condenser pressure to change drastically, upsetting the expander operation to the point where it cannot deliver its rated power. As the engine goes through power changes, the condenser load changes almost instantly. It was calculated that upon a large power step, the condenser pressure increases at a rate of 48 KPa (7 psi/sec). Hence, the condenser control system must be capable of changing the cooling air flow through the condenser between idle to full load cooling rate in one to two seconds to avoid a large expander disturbance that would confuse the driver.

In the simulator vehicle there are two mechanisms to drive the cooling air through the condenser. First, the condenser fans are driven by a variable ratio hydrostatic drive, comprised of a variable displacement, expander driven pump and a fixed displacement motor. When the pump outlet pressure is 21 MPa (3000 psi), the fan can be accelerated to its full load of 37.5 Hz (2250 RPM) in 1.4 seconds. Second, at highway speeds and low power levels, the ram air provides enough cooling air flow. Under these conditions the fan drive pump is declutched and the cooling air flow is regulated by variable pitch shutters in front of the condenser.

The steady state control schedule of the shutters and hydraulic drive operation is shown in Figure 8.11. As can be seen, the shutters are scheduled to be completely

CONTROL SCHEDULE

FIGURE 8.11

closed at 103 KPa (15 psia), condenser pressure. This is done in order to prevent the condenser from operating under vacuum and admitting non-condensables into the water system. At 138 KPa (20 psia), the shutters are fully open, making full use of the available ram air. In order to avoid competition between the shutters control action and the fan, a dead band between 138 and 172 KPa (20 and 25 psia) is designed into the system; however, when the condenser pressure exceeds 165 KPa the pump clutch is engaged to prepare the fan drive control which starts operating at 172 KPa. The fan drive pump stroke is operated linearly between zero at 172 KPa (25 psia) and 100% at 207 KPa (30 psia) at which pressure the fan is brought up to full speed as rapidly as possible.

Figure 8.12 is a schematic diagram giving the method of mechanization of the condenser pressure dual action controller. The condenser pressure is sensed by two bellows mechanisms. The left hand side bellows response is directly proportional to the condenser pressure. Upon a change in pressure the proportional bellows changes the spool valve position, admits hydraulic oil into the dual actuator which changes the shutters and/or pump stroke until the proportional portion of the controller is satisfied. The bellows train on the right hand side of Figure 8.12 is sensitive to the condenser pressure rate of change. This pressure derivative sensor provides some anticipatory action which starts, upon a large power change, accelerating the fans before a high pressure error is sensed, and prepares the condenser system for its new load level.

SES had very good results with the condenser pressure dual controller. During the simulator tests, the pressure stayed within the design bounds of 103-207 KPa (15-30 psia), without using the relief valve or going to vacuum even once throughout the duration of the tests.

CONDENSER PRESSURE CONTROLLER SCHEMATIC

FIGURE 8.12

8.6 START SYSTEM

The start-up of the SES steam engine differs from a commercial vehicle start-up in that for the former it is necessary to heat up the steam generator coil before hot, high pressure steam can be provided to the expander, whereas in the latter, the existence of combustion is enough to guarantee a successful start.

The start-up control system executes a sequence of operations from the time the key is turned on until the engine is self-sustained. It is equipped with indicator lights to inform the operator of the state of the engine throughout the start sequence which proceeds as follows:

1. A key to "start" position activates the boiler starter, enabling the operation of the combustion air blower and the water pump.
2. The blower is brought to 33 Hz (2000 RPM) (idle speed) and the air valve is opened for two seconds. This is done in order to purge traces of residual fuel from previous operations.
3. The air valve is brought back to its scheduled position. The igniter fuel solenoid valve and spark generator are activated for two seconds.
4. When ignitor flame is sensed, the main burner fuel solenoid is opened and the burner is ignited. A safety controller shuts down the system if no flame is sensed after two seconds.

5. The burner is brought to 70% of its load to rapidly heat up the steam generator and shorten the start-up duration.
6. The feedwater pump is actuated to provide a high water flow into the boiler for five seconds. This duration of flow is sufficient to fill up about half of the coil volume and insures an initial generation of steam at temperatures of 533-700° 500-800°F.
7. When the steam pressure reaches 1.38 MPa (200 psi), the directional expander starter is activated and the expander is cranked. At the same time, the system controls command is transferred from the start-up mode to the normal operating mode.
8. When the expander becomes self sustained, both starters are turned off and the start-up is completed.

In general, the steam engine start-up controller has performed very successfully. During the testing period, the system has been started hundreds of times. The steam pressure and temperature are maintained within safe limits throughout the start sequence and the expander neither overspeeds nor stalls. The start-up times vary between ten seconds on a warm engine and thirty seconds on a cold engine. During the start period, the engine consumes 0.90 to 0.23 kg (0.2 - 0.5 lbs) of fuel, depending on the start-up duration.

8.7 MOD #3 CONTROL SYSTEM

Although the task of developing the Mod #3 control system on the "Controls Development Simulator" was deleted from this contract, SES has had the opportunity of applying the steam engine controls know-how that has been accumulated throughout the simulator tests, to a steam powered coal haulage vehicle that is currently being developed for the U.S. Bureau of Mines. This section will highlight the Mod #3 controls developments that have led to a very significant improvement in the emissions characteristics and in the simplicity and functionality of the engine.

1. By changing the inlet and exhaust ducts of the burner, SES has been able to eliminate the possibility of outside pressure disturbances interfering with the burner air to fuel ratio. Thus, the combustion air valve stiffening function is no longer necessary. In the Mod #9 burner, the air valve was removed, resulting in a much simpler burner whose emissions characteristics are even better than the ones of the Mod #8 boiler used in the simulator.
2. The fuel pump was replaced by a fast responding, very accurate fuel metering valve designed by SES.
3. The blower drive controller is now an integral part of the hydraulic pump. It is simpler and responds smoothly to demand changes.

4. The combination of the three changes discussed above, has enabled a much better transient air to fuel ratio control, resulting in a much faster and cleaner burner.
5. The boiler temperature control logic has been changed to a combined firing rate and water flow controller. Fewer sensors are now needed to achieve a more accurate temperature control.
6. The expander is controlled by a mechanical speed governor. The driver's feeling is similar now to a diesel engine driver's.
7. The condenser pressure controller is an integral part of the hydraulic drive pump. It is simpler and operates the fan more efficiently.

SES is very pleased with the performance so far demonstrated on its coal haulage vehicle and it is confident that with the above changes, the simulator vehicle would be able to meet the original emissions goals without difficulties.

9.0 PREPROTOTYPE RANKINE CYCLE SYSTEM DESIGN DEVELOPMENT AND PERFORMANCE

9.1 THE PPT RANKINE CYCLE SYSTEM

9.1.1 TEST FACILITY

By the fifth quarter, work began on the planning and design of the preprototype system test facility. The preprototype configuration was established. The testing of the preprototype system was separated into two phases. After the completion of individual component testing, the system was assembled in a breadboard configuration. All components were connected, but were not in their prototype position. Back-up systems, as for example condensing, feed-water and steam generation, were available to substitute for any inoperative component to minimize interruptions in testing.

The preprototype test cell was constructed and the engine bed and 298 kW (400 hp) eddy current dynamometer were installed. The dynamometer controls were modified from their constant-speed control mode which simulated road load. For EPA performance runs the expander was coupled to the dynamometer through a three-speed automatic transmission, and a flywheel simulating vehicle inertia was placed between the transmission and the dynamometer. This enabled simulation of EPA performance runs with the exception of the 30% grade capability due to the steady-state torque limitation of the dynamometer.

9.1.2 TEST SYSTEM

The preprototype test (PPT) system comprised a power plant consisting of the major components in a form substantially identical to that planned for vehicle installation; but with certain auxiliary components and control hardware in a form providing for greater flexibility in test cell operation and not suitable for packaging in a vehicle. The expander was identical except that the main bearing beam and oil pan were not suitable for vehicle installation due to steering linkage interferences. It was fitted with a 2.1:1 step-up gear and a Chrysler 3-speed automatic transmission. The burner/vapor generator package was in prototype configuration with the exception that an externally mounted air throttle valve was employed. While the external air throttle valve added to the system bulk, it also provided greater flexibility for early preprototype system tests and was easily modified if required. The condenser supplied by AiResearch was a flat unit having the same frontal dimensions and core thickness as planned for the prototype. Condenser fans were in prototype configuration. Major components were packaged on the PPT test stand in the same orientation in which they were mounted in the vehicle installation.

Auxiliary component drives were handled by means of a flexible, variable ratio hydraulic drive system: Two variable displacement hydraulic pumps were belt driven from the expander crankshaft. One pump supplied a

fixed displacement motor driving the condenser fans through v-belt drives while the other pump supplied a fixed displacement motor driving the atomizing air pump and, through fixed ratio belt drives, the feedpump and combustion air blower. In addition, two other matching variable displacement hydraulic pumps were driven by electric motor to provide a means of starting the system. The circuits were provided with suitable check valves so that either the expander driven pumps or the electric motor driven pumps could supply system activation. This auxiliary drive system provided maximum power plant testing flexibility throughout the preprototype test and development phase. Various drive ratios and control modes were examined very simply by employing different schedules of pump displacement.

No provisions were made for vehicle accessories such as alternator, power steering and air conditioner in the preprototype (PPT) testing. The PPT system is shown in Figure 9.1 as seen from the left or driver's side of an equivalent car.

In preparation for system testing, eight bolt pistons, reversed valve arrangement, reduced port area liners, 1.1 rad (60°) admission camshaft and 1.3 rad (75°) cut-off camshaft were installed and run-in. Phase I testing covered the work necessary to determine expander performance improvements. Phase II testing studied overall system performance.

PREPROTOTYPE TEST SYSTEM ON ENGINE DYNAMOMETER
FIGURE 9.1

Phase I testing was completed except for speeds at which torsional vibration would not permit operation. This vibration was associated with the mismatch of inertias between the expander, the output shaft, drive shaft and dynamometer rotor. An attempt was made to correct the problem by adding a flywheel between the output shaft and the drive shaft. Unfortunately, the end result of this modification was a shift in the natural frequency of the system to a frequency that was also within the speed range required for testing.

After Phase I testing, portions of the system requiring modification for Phase II testing were disassembled and modified. In addition new equipment was installed to allow all of the auxiliaries to be driven directly from the expander.

The feedwater pump was driven through an over-running clutch by a shaft coupled directly to the front end of the expander. The shaft drove the blower hydrostatic transmission through a timing belt. The condenser fans were driven by twin v-belts from a hydraulic motor mounted on the left side fan shaft. Power for the fan hydraulic motor was provided by a hydraulic pump driven by a timing belt off the front end of the crankshaft. The fans however were not used during road load testing because ram air was sufficient.

The outside power source to the system was the hydraulic starter motor which drives the hydrostatic transmission shaft through an over-running clutch. The combination of the two over-running clutches allowed the steam generator to operate without the expander and the expander to operate without driving the hydraulic starter motor.

The servo controller for the hydrostatic transmission and the atomizing air pump were the only system components which remained outside of the simulated engine compartment. These components represented test type hardware and their absence from the interior of the compartment was of no significance.

9.2 THE RANKINE CYCLE SIMULATOR VEHICLE

During the contract period of performance, three different automobiles were selected in turn to be used as the simulator vehicle (a 1969 AMC Ambassador, a 1974 Plymouth Fury and a 1975 Dodge Monaco). The Simulator Vehicle was the 1975 Dodge Monaco "C" Body Vehicle. While each automobile was equally satisfactory for installation and testing of the preprototype Rankine Cycle propulsion system, the final selection was based upon contract modification or redirections of contract intent. Photographs of the final installation are given in Figures 9.2, 9.3, 9.4 and 9.5.

Preliminary analyses were completed by SES to determine values of design parameters that would be required to meet vehicle acceleration and gradability goals. The

RANKINE CYCLE SIMULATOR ON CHASSIS DYNAMOMETER

FIGURE 9.2

RANKINE CYCLE SIMULATOR LOOKING OVER CONDENSER
FIGURE 9.3

RANKINE CYCLE SIMULATOR VIEW OVER LEFT FRONT FENDER
VEHICLE AUXILIARIES SIDE

FIGURE 9.4

RANKINE CYCLE SIMULATOR ON CHASSISS DYNAMOMETER
RAM AIR BLOWER ABOVE EXHAUST SAMPLING AT SIDE:
EXHAUST HEAT EXCHANGER, TURBINE FLOW METER, AND
SAMPLE COLLECTION PUMP AND BAG SYSTEM AT RIGHT

FIGURE 9.5

initial recommendation was that present torque converter automatic transmissions were unsuited for our application because of the high torque/low speed capability in the steam engine. Thus, a standard transmission or a fluid coupling automatic, with two speed forward plus reverse was recommended. It was also agreed that the expander would have a 1:2 step-up gearing into the transmission to make speed and torque more compatible with available drive train components.

The analyses included 6 industrial type transmissions (available from Borg-Warner), all with 1:1 high gear ratios, and low gear ratios ranging from 1.558 to 1.832; and 5 rear axles (available for the AMC Ambassador) with ratios ranging between 2.87 and 4.10. The goals which controlled drive train choice were the ability to cover 134 M (440 ft) in 10 seconds from a standing start, and the 0 to 96.56 km/hr (60 mph) acceleration in 13.5 seconds. The 40.24 - 112.65 km/hr (25 - 70 mph) and high-speed pass goals could easily be met. The drive train selection was then made using the lowest rear axle ratio (3.54:1) and the necessary transmission (1.832:1 low gear) which met all acceleration goals. The use of the lowest possible rear axle ratio would improve fuel economy and emissions per mile.

The gradability of the vehicle with the selected drive components was determined. The 5% grade goal could easily be met. With a gross weight of 2404 kg (5300 lbs), the vehicle had a top speed on the 5% grade of 136.79 km/hr (85 mph) compared to the 112.65 km/hr (70 mph) goal. The 30% grade goal could not be met with all of the initial

assumptions. For the selected drive train, and a weight of 2404 kg (5300 lbs), the expander would require a valve opening of about .91 rad (52°) (instead of the .87 rad (50°) being used) in order to accelerate to 18.05 km/hr (15 mph) on a 30% grade specified by the goal. Alternately, with the propulsion system and drive train as specified, the vehicle could accelerate up a 28% grade with a 2404 kg (5300 lb) gross weight or accelerate up a 30% grade with a 2268 kg (5000 lb) gross weight (1814 kg (4000 lb) actual curb weight plus 454 kg (1000 lb) for passengers and baggage). Alternately, a higher, less efficient, rear end ratio could be used to meet the goal.

The propulsion system and drive train as specified above adequately met the performance goals for the vehicle. The over all gear ratios with the selected components were 3.24:1 in low, and 1.77:1 in high which are only slightly higher than those assumed in earlier analyses. The new values would be used in subsequent analyses for final component designs.

9.2.1 DESIGN OF VEHICLE INSTALLATION

Subsequent to SES/AMC efforts, contract changes included the selection of Chrysler as the sub-contractor to furnish the SES preprototype Rankine Cycle simulator vehicle. Initial designs were modified based upon use of a 1974 Plymouth Fury as the simulator vehicle.

9.2.1.1 EXPANDER

The 4-cylinder in-line expander was located at the rear of the engine compartment with the cylinder axis inclined approximately $.79$ rad (45°) from the vertical toward the left side of the vehicle. The longitudinal position was maintained 7.6 cm (3 ") rearward from the "interference-free" position originally suggested by Chrysler. A detailed study of vehicle interference points was made for this expander location. Minor sheetmetal changes were required in the transmission tunnel and possibly a slight relocation of the accelerator pedal. These changes were readily accomplished. The rear transmission mount located conveniently with this expander position. Several changes were required to the expander main bearing beam and to the oil pan as a result of steering gear linkage interferences; however, it appeared that these changes could be readily accomplished with this proposed expander location. (Ricardo studied the required changes to determine feasibility for the prototype expander.) This type of change was anticipated, and indeed was the principal reason for incorporating the separate main bearing beam in the expander design.

9.2.1.2 STEAM GENERATOR

The steam generator (with integral burner and blower) was mounted alongside the expander on the right hand side of the vehicle with its axis parallel to the expander with the 49.5 cm (19.5 ") diameter portion nested

between front and rear expander flanges. This resulted in minimum interferences with the engine compartment hood and suspension components (upper control arm pivots). These interferences could be handled by localized indentions in the outer steam generator cover. The steam generator was mounted rigidly to the expander which was mounted to the vehicle frame through suitable rubber-suspended engine mounts. Details of the engine mount design were studied by Chrysler.

9.2.1.3 COOLING AIR INLET

The air conditioning condenser, transmission and engine oil coolers were mounted in front of the steam condenser. Only the exposed face area above the bumper was covered by these heat exchangers. Besides saving valuable axial space, this helped alleviate the poor air flow distribution caused by bumper flow blockage. Another means for reducing the bumper flow blocking was the provision of two air inlet port openings in the bumper. The primary source of condenser air flow, however, was through the combination grille/shutters located directly above the bumper and extending the full width of the condenser.

9.2.1.4 FEEDWATER PUMP

The feedwater pump used for these installation studies was the SES design based on the feedpump development work accomplished. The electric starter drives the combustion blower, atomizing air pump and feedpump

independent of the expander for system start-up. Over-running clutches were incorporated in the starter drive pulley and the atomizing air pump drive gear permitting the expander to pick up the drive function as it comes up to speed. An electric starter motor was mounted at the expander transmission interface to insure proper rotational direction of the expander during start-up.

Expander-driven vehicle accessories are mounted above the expander. The alternator and power steering pump were coupled together, and the air conditioning compressor was mounted separately.

9.2.1.5 EXHAUST SYSTEM

The preliminary design and installation of a combustion exhaust system was examined in a layout drawing. A large cut out of the rear cross member similar to that of the 1974 "C" body preprototype chassis was utilized. The use of this cut-out required the reduction of the exhaust duct area to about 50.8 cm (20") through the rear cross member. The 60.9 cm² (24 in²) area was maintained throughout the remainder of the exhaust pipe.

9.2.1.6 CONDENSER AND FAN ASSEMBLY

The vehicle installation of the AiResearch preprototype steam condenser and fan assembly was examined. It was concluded that since the preprototype condenser is flat there would be insufficient clearance for

the shock absorbing bumper. Thus, either the vehicle would have to be lengthened or the shock absorbing bumper would have to be eliminated. It was decided that the use of a rigid bumper system would be preferable on a preprototype installation. In addition, the AiResearch layout of the preprototype condenser assembly indicated that the condenser support bracket would be rigidly attached to both the radiator yoke and the frame. This concept was unacceptable.

The sheet metal structure to which a production car radiator was mounted (the yoke) is the structural member responsible for rigidly supporting the front body sheet metal. The frame members deflect relative to the body sheet metal (and the yoke). Therefore, to avoid overstressing, the condenser had to be mounted only to the yoke as is a standard radiator. The yoke must therefore be redesigned to support the condenser and fan as well as provide its body structure function. The fans must be isolated from the yoke to prevent the transmission of fan vibrations to the body sheet metal.

The problems encountered in the installation of the preprototype condenser and fan assembly were reviewed between SES and Chrysler. Sufficient design work was performed to insure that the preprototype condenser system could be installed in the vehicle.

Some details not included for the preprototype condenser installation were considered for the installation of a prototype condenser. The preprototype

mounting brackets shown by AiResearch appeared to neglect the effects of thermal expansion of the condenser. Also, a method for supporting the auxiliary heat exchangers (A/C condenser, transmission oil cooler, engine oil cooler) and the air shutter control mechanisms had to be developed for a prototype installation. A location for a hood latch assembly also had to be provided. The feasibility of spreading the frame rails so that they would straddle the prototype condenser was examined. This required both frame rails forward of the front cross member to be offset 19.1 cm (7.5") outboard from their production positions. The additional offset would simplify the bumper mounting and allow a larger yoke cross section between the fan and frame rail. Preliminary calculations indicated that the 19.1 cm (7.5") offset would require the frame rail metal to be nearly .6 m ($\frac{1}{4}$ ") thick to maintain stress limits during a 8.05 km/hr (5 mph) impact.

9.2.1.7 ACCESSORY DRIVE

The accessory drive required the use of a matched set of three 1.3 cm ($\frac{1}{2}$ ") v-belts with the alternator and power steering pump on a common shaft. Since belt tensioning problems were likely with this arrangement, it was decided that the power steering pump could be driven with one belt which could be tensioned independently from the two belts on the alternator. This would be accomplished by moving the power steering pump forward of the alternator

pulley and reversing its direction of rotation. All three belts would drive the air conditioning compressor. The air conditioning compressor pulley and clutch mechanism were designed to accomodate a maximum of two 1.3 cm ($\frac{1}{2}$ ") v-belts. Thus, either an adapter to the pulley or a redesigned clutch assembly would be required.

An alternative to the above arrangement was developed to reduce the required number of belts. This was accomplished by increasing the belt speed for a given accessory load. A 63% increase in belt speed at idle was required for the use of two 1.3 cm ($\frac{1}{2}$ ") v-belts. The size of a pulley on the crankshaft was limited to 20.3 cm (8") because of interference with the frame cross member. Since the size of the crank pulley could not be increased it was decided that the speed of the accessory drive pulley had to be increased. This was done by driving the drive pulley from an intermediate step-up gear driven from the expander crankshaft. The diameter of the accessory drive pulley was reduced to about 16.3 cm (6.4") for clearance with the cut-off phase change mechanism. This required a 2:1 speed ratio between the drive pulley and the expander crankshaft.

9.2.1.8 TIRES

The tire size for the vehicle was increased from HR78-15 go JR78-15, for increased life on the dynamometer rolls.

9.2.1.9 BODY MODIFICATION

The body modifications were completed by Modern Engineering Service Company, and the vehicle was then returned to Chrysler for the reassembly of the interior, the electrical system, and final assembly of the front end parts (headlights, grille, etc.).

9.2.1.10 SUBCOOLER AND AUXILIARY COOLERS

In the interim between final assembly of the vehicle by Chrysler and the installation of the pre-prototype propulsion system at SES detailed analyses of preprototype (PPT) test data provided for definition of subcooler and auxiliary coolers.

A method for providing subcooled water to the feedwater pump was determined. It consisted of extracting water from the condenser sump, circulating it through an air-cooled heat exchanger and returning it to the sump to be mixed with saturated condensate providing a subcooled mixture. The crankcase blowby was condensed by heat exchange with water coming from the subcooler.

9.2.1.11 SUBCOOLER CIRCUIT

The technique for providing subcooled water involved a modified sump design, a circulating pump and an air-cooled subcooler. The subcooler circuit was designed for a flow rate for which the selected pump has

excess capacity at the expected subcooler pressure drop. The blowby condenser was incorporated into the subcooler circuit.

The sump remained a long cylinder lying below the condenser and receiving condensate by gravity drainage. It was divided into an upper and lower section separated by a baffle so that the upper section received and stored saturated condensate which flowed as needed to the lower section which also received subcooled water and acted as a mixing chamber for the subcooled and saturated water, resulting in a subcooled mixture. The feedpump and subcooler circulating pump both draw from the lower sump section. In addition, a float valve was incorporated to allow make-up water to be transferred to the sump as needed by gravity drainage from a pressure-balanced make-up tank. The make-up rate was small so that random water splashing would not cause the sump to overfill. A non-condensable purge valve with reflux condenser was accessible to the upper sump section.

9.2.1.12 AUXILIARY COOLER DESIGNS

The required auxiliary coolers were: air conditioning condenser, hydraulic oil cooler, transmission oil cooler, engine oil cooler, subcooler, and crankcase blowby condenser. For the air conditioning condenser, a Chrysler core was used consisting of 9.5 mm (3/8") tubes on

25 mm (1") centers in a wavy fin air side matrix with 5 fins/cm (13 fins/in), two tube rows deep, with a 3.2 cm (1.25") fin depth. The other coolers used Wolverine Trufin^(R) integral fin aluminum tubing with 9.5 mm (3/8") I.D., 11.3 mm (.445") O.D., tubes on 25 mm (1") centers, 25 mm (1") fin diameter and 3.5 fins/cm (9 fins/in).

The available flow area was the frontal area of the main condenser which was 5574 cm^2 (6 ft^2). The flow through the various air-cooled heat exchangers were assumed to be in direct proportion to that fraction of the frontal area covered by the cooler.

a. Hydraulic Oil Cooler

The design point corresponded to maximum pump flow at 21 Hz (1250 RPM) of the engine. Of the roughly 7.5 kW (10 hp) maximum delivered to the fan drive, about 1.86 kW (2.5 hp) was lost as heat. Of the .75 kW (1 hp) delivered to the blower, .37 kW (0.5 hp) was lost as heat. The total heat rejection was 2.24 kW (3 hp). At higher engine rpm, a higher flow rate would be available but implied ram air assistance would reduce the torque, i.e., power requirement. The oil flow rate at design point was 1860 kg/hr (380000 lb/hr) of which an amount proportional to frontal area was assumed to go to the hydraulic oil cooler.

The cooler has a frontal area of 269 cm^2 (0.29 ft^2). The configuration is two parallel passes, two rows deep, resulting in a maximum oil side pressure drop of

68.9 kPa (10 psi). The design point leaving air temperature of 356°K (181°F) with an inlet oil temperature of 353°K (176°F) with an inlet air temperature of 303°K (85°F).

b. Transmission Oil Cooler

The design point for the transmission oil cooler was provided by Chrysler and determined by SES to correspond to the most difficult design point expected by the steam engine/automatic transmission combination. This was 408 kg/hr (900 lb/hr) oil flow, 8.8 kW (30,000 BTU/hr) heat rejection, 422°K (300°F) oil inlet temperature, 303°K (85°F) air temperature, 2462 kg/hr (5428 lb/hr) condenser air flow corresponding to 48 kg/hr (30 mph) vehicle velocity. A pressure drop limitation of 138 kPa (20 psi) was specified.

The cooler has a frontal area of 1356 cm² (1.46 ft²). The configuration is two passes in series, two rows deep resulting in a design point pressure drop of 103 kPa (15 psi). The leaving air temperature is 356°K (181°F) and the leaving oil temperature is 385°K (233°F).

c. Engine Oil Cooler

Based upon PPT test data, the design point for the engine oil cooler was heat rejection of 4.4 kW (15,000 BTU/hr) oil flow of 1360 kg/hr (3000 lb/hr) corresponding to 16.7 Hz (1000 RPM), a condenser air flow of 3447 kg/hr (7600 lb/hr), an inlet oil temperature of 396°K (253°F) and an air temperature of 303°K (85°F).

The cooler has a frontal area of 585 cm^2 (0.63 ft^2). The configuration is two passes in parallel, two rows deep resulting in a design point pressure drop of 62 kPa (9 psi). A 138 kPa (20 psi) by-pass relief valve was incorporated resulting in maximum cooler oil flow of 2007 kg/hr (4424 lb/hr). A maximum air flow (a heat rejection of 9.4 kW (32,000 BTU/hr) results) which was more than adequate in spite of the oil flow limitation. Design point leaving air temperature was 347°K (165°F) while design point leaving oil temperature was 390°K (242°F).

d. Air Conditioning Condenser

The design point for the air conditioning condenser was supplied by Chrysler. It specified a heat rejection of 8.3 kW (28,000 BTU/hr), a condensing temperature of 345°K (110°F) and an air flow corresponding to 48 km/hr (30 mph). The air flow was determined to be 2462 kg/hr (5430 lb/hr).

The condenser, using a Chrysler core, had a frontal area of 2731 cm^2 (2.94 ft^2). The design point leaving air temperature was 341°K (154°F).

e. Subcooler

The design point for the subcooler was taken to be 14.1 kW (48,000 BTU/hr) which would provide steady-state subcooling of 278°K (40°F) at full steam flow of 544 kg/hr (1200 lb/hr) and full condenser air flow of 17,237 kg/hr (38,000 lb/hr). Thus at the elevated condenser pres-

sure of 241 kPa (35 psia), the sump would be subcooled to below saturation temperature at 138 kPa (20 psia) of 382°K (228°F), and no flashing would occur with condenser pressure transients. A by-passed cooler flow of 544 kg/hr (1200 lb/hr) was selected. The design point water inlet temperature was taken to be 378°K (220°F), and the air temperature to be 303°K (85°F).

The subcooler had a frontal area of 660 cm² (0.71 ft²). The configuration is two passes in series, two rows deep. The water outlet temperature is 356°K (130°F). The pressure drop is 34 kPa (5 psi) and is constant since a constant by-pass flow is used.

Because under all but design point conditions, the cooler is considerably oversized, the subcooled water was used to condense steam blowby which enters the crankcase via the lower piston ring. The subcooler size was kept the same, but the net performance was different as will be discussed below.

f. Crankcase Blowby Condenser

Since the crankcase blowby was to be condensed by a fixed flow rate of subcooled water, it was shown that the effectiveness is constant. Moreover, beyond condensing all the available vapor, no additional heat transfer would result. The requirement over the normal operation range was for a constant flow of 4.5 kg/hr (10 lb/hr) to be condensed at 373°K (212°F). If a 50% effectiveness is provided

then a circuit steady-state heat balance including sub-cooler, condenser and sump results in 2.93 kW (10,000 BTU/hr) to be available at 20% of maximum condenser air flow with excess capacity above this air flow. This was enough to condense the blowby over most of the range with a deficit below 20%.

Moreover, the heat balance showed that with a sump pressure of 138 kPa (20 psia), the normal system condensing condition, the sump would remain at $368^{\circ}\text{K} \pm 3^{\circ}\text{K}$ ($230^{\circ}\text{F} - 5^{\circ}\text{F}$). A difficult condition would arise during high power transients if the system condenser pressure reached 241 kPa (35 psia) since the blowby is proportional to the difference between expander exhaust pressure and the fixed crankcase pressure of 101 kPa (14.7 psia). Thus, at this high power condition a rare occurrence, a blowby of 18 kg/hr (40 lb/hr) could occur. With the system, the sump would rise to a steady-state of 382°K (228°F) which would be inadequately subcooled. Moreover, the steady-state condensing capability of 7.26 kg/hr (16 lb/hr) would result in venting some steam to atmosphere. However, with a 4.5 kg/hr (10 lb) inventory of sump water, a time constant of 18 seconds was computed so that initially a condensing capability of 12.2 kg/hr (27 lb/hr) would be available for short duration, WOT condition. Little vapor would be lost.

Thus, this design point was taken to be adequate: Design effectiveness of 50% with water flow of 547 kg/hr (1200 lb/hr) on one side and condensation on the other.

9.2.2 FINAL CONFIGURATION

With minor exceptions, the design details developed for the 1974 Plymouth Fury and summarized above were revised and incorporated into the 1975 Dodge Monaco "C" body vehicle. The Dodge Monaco was used as the vehicle incorporating the preprototype Rankine Cycle automotive propulsion system during dynamometer tests in fulfillment of contract goals.

The shutter-condenser-auxiliary heat exchanger design was rearranged to locate the auxiliary heat exchangers in front of the condenser face. Cooling air enters the condenser from the nose cap grille and also from the cut-outs in the bumper. The original design, which used the shutters for a grille, would have allowed an excessive amount of air flow through the bumper openings and the condenser when the shutters were closed thus permitting the condenser to sub-cool and create sub-atmospheric condenser pressures if the vehicle were to be operated at low ambient temperature. The final arrangement placed all the auxiliary heat exchangers between the nose cap grille and the shutters and provided auxiliary cooling at all times. In addition to the shutter-condenser-auxiliary heat exchanger design modifications, the 1975 Dodge bumper was replaced by the 1974 model year bumper so that there would be an increase in the air flow frontal area. The 1975 bumper has only two cut-outs with a total inlet area of 0.15 m^2 (1.13 ft^2). The 1974 bumper had three cut-outs with a total inlet area of 0.120 m^2 (1.29 ft^2).

Therefore, the use of the 1974 bumper increased the air flow frontal area of the bumper by 14%. The grille air flow frontal area was also increased after the removal of the shutters from the front end cap.

9.2.3 POWER PLANT INSTALLATION AND CHECK-OUT

The simulator vehicle power plant installation was completed early in May 1975. Subsequently a test was conducted to check out the system. The test consisted of an acceleration from rest to a speed of approximately 100 km/hr (62 mph). Transmission engagement and gear shifts during "drive-off" were smooth, and system control was stable. The dynamometer load was increased from zero to 13 kW (17hp), and 55 kilometers (34 miles) were accumulated on the vehicle odometer prior to shut-down. All critical parameters were monitored during the run to provide early engine shake-down data prior to completion of the remaining instrumentation installation and calibration.

9.2.4 SIMULATOR DRIVE TRAIN MODIFICATION

Several means were considered for suppression of a serious step-up gear noise problem. The solution that was chosen as the most feasible (from the point of view of time constraint) consisted of a modification to the engine control that would prevent the expander from running at the system's resonant speeds. This modification

required that the expander not be loaded at certain low speed conditions; thus, the use of a frictional clutch that would disengage at certain speeds was required. A friction clutch on the other hand enabled elimination of a hydrodynamic clutch (or the torque converter), which would result in improvement of fuel economy on any driving cycle consisting of large idling portions.

The above solution was realized in the following manner. A pressure modulating valve, externally controlled, was incorporated into the automatic transmission hydraulic circuitry in such a way that the valve controlled the oil pressure feed to the rear clutch. Although the heat capacity of any clutch in the transmission was not sufficient to sustain slipping conditions for an extended length of time, it was postulated that the clutch would probably not overheat in the closely controlled laboratory condition of simulator testing. The use of the transmission's frictional clutch would allow driving transmission with a short shaft directly from the expander flywheel. But since the envelope of the torque converter carried the starter ring and the front pump drive which were still needed, parts of a 27.3 cm (10 3/4") torque converter were modified to provide the direct drive to the transmission input. The modification required practically no new parts. Five cases were investigated:

1. No load on the drive train (transmission in "N")
2. Light load in first gear
3. High load in first gear

4. Light load in direct drive
5. High load in direct drive

Although the analysis was based on several estimated input data, the predicted resonant frequencies were representative of the actual system within acceptable limits. The practical impact of the results was:

The engine can idle at any speed above 5 Hz (300 RPM), providing the inlet and exhaust steam pressures are within the design limits. The expander can accept full throttle from any speed above 12 Hz (720 RPM). The expander cannot be loaded at lower speeds and must be brought to the "loadable" speed quickly, crossing the critical speeds in fractions of a second. The engine can be decelerated to its idle speed under light load conditions due to the non-linear characteristics of the flexible coupling.

The modification was confirmed during several tests performed with the simulator. Control of the pressure modulating valve was incorporated into the overall control system. After simulator system shake-down was completed, remedial action was instituted on those problem items which could readily be corrected within the scope of the contract. After shake-down and remedial action, the simulator vehicle was operated on the chassis dynamometer over the Highway Fuel Economy Cycle and the Urban Federal Driving Cycle.

There were three recurring problems which required development throughout the shake-down period. None were of a serious nature, but all of them required a significant amount of testing to determine the cause of their behaviour.

The first problem involved the condenser pressure controller which failed many times because of control cable failure. All of the commercially available cables which were used could not tolerate the heat radiated to them from the steam accumulator. Each failure resulted from melting of the inner liner of the cable. The appearance of the outer sheath of the cables, however, gave no indication of heat related seizure. The final solution to the problem was accomplished by designing and fabricating a cable which had no plastic or fiberglass parts that were susceptible to malfunctioning in a high temperature environment.

The second problem was one of improper fuel scheduling. The original fuel schedule for the simulator was the same as the fuel schedule that was used with success in the Burner in the Preprototype Test Cells. However, after the system was run in the simulator, it became apparent that the fuel/air mixture was too rich at loads above 20%. A series of tests was conducted for the purpose of rescheduling the burner. The results of those tests indicated the existence of a reduction in air flow from the blower when it was packaged in the simulator engine compartment. The mixture was corrected by increasing the blower speed at high firing rates and reducing fuel flow at low loads. A drastic reduction in emissions resulted from the burner rescheduling.

The third and least significant problem was related to the ignition system. As originally designed, the igniter fuel line was dead-ended at the igniter fuel solenoid, contiguous to the steam generator. With this type of system, after long periods of continuous testing, the fuel at the igniter fuel solenoid became vaporized and prohibited igniter fuel flow on subsequent hot restarts.

Three failures occurred after the start of simulator testing. The rear clutch on the automatic transmission burned out because of excessive slipping at idle. The transmission was replaced. The pump housing on the new transmission was modified so that complete disengagement of the clutch could be effected, and future failure of that nature could be precluded.

The step-up planetary gears failed during a UFDC. This was attributed, in part, to the inability of the original automatic transmission to completely disengage at idle. Incomplete disengagement reduced expander speed to a critical speed, and deleterious torsional oscillations ensued. The 2.1 to 1 ratio step-up gear was removed from the system, and the rear axle ratio was changed from 3.21:1 to 2.7:1, resulting in a 77% increase in expander speed versus vehicle speed in each gear. With this drive train configuration driveability was significantly improved, and there was no indication of increased fuel consumption.

The expander oil cooler was not maintaining the desired oil temperature. It was replaced by a water-cooled oil cooler so that critical heat load data could be obtained. There was insufficient data to determine the cause of the deficiency in the original oil cooler. The testing schedule did not permit testing for the sole purpose of accumulating that data.

Prior to termination of simulator testing, steady-state and driving cycle tests were extensive and impressive. The system performed well, exceeding the emission goals specified in the contract. System Performance is outlined in the following section.

9.3 THE RANKINE CYCLE PERFORMANCE

9.3.1 PERFORMANCE STUDIES

1. Acceleration Studies

A computer program to simulate the SES system vehicle under wide open throttle accelerating conditions was written. A flow diagram for the program is given in Figure 9.6. The data used included the EPA specification for rolling resistance, drag, drive train, tires and expander speeds. The expander map used is shown in Section 3.

The program was run in conjunction with a boiler dynamic response study to determine the effect of response time on performance. The parametric study consid-

FLOW CHART FOR ACCELERATION STUDIES

FIGURE 9.6

ered boiler time constants ranging from 0.1 sec to 7.0 seconds. From the early results it was concluded that a time constant of 1.0 second would satisfy EPA acceleration specifications.

2. Federal Driving Cycle Fuel Economy and Emissions

Fuel Economy results obtained by computer simulation of several variations of the Rankine Cycle steam configuration while driving over the Federal Driving Cycle (FDC) were considered.

The powerplant variations examined to determine fuel economy sensitivity to changes were:

1. Accessory load (with and without air conditioning).
2. Total power plant auxiliary load (atomizing air pump (AAP), feedwater pump, combustion air blower (CAB), and condenser fans).
3. Expander idle speed.
4. Torque converter characteristics.

The analysis was for the standard size car and drive train.

The first three conditions indicate the effect of power plant auxiliary load changes from no load to full load, excluding condenser fan load, and then full load with condenser fans. The following observations were made:

- a. The effect of auxiliaries on fuel economy was minimal and were about 2 to 2.5% in all cases.
- b. The fourth condition showed the effect of fuel economy if the accessory load could be reduced from that currently specified by EPA.
- c. When air conditioning was not operating, the load was assumed to decrease from the 1.1 to 3.7 kW (2 to 5 hp) range to a .7 to 3.7 kW (1 to 5 hp) range. This produced a fuel economy improvement of 5%.
- d. With the air conditioning operating, the load was assumed to decrease from the 3 to 11.2 kW (4 to 15 hp) range to a 1.7 to 7.5 kW (2.25 to 10 hp) range. The effect on fuel economy was an 11% improvement.

With a higher idle speed, reducing the accessory load without air conditioning gives a fuel economy improvement of 3½%, while with air conditioning, the load reduction gives a 9% improvement in fuel economy.

The effect of a different torque converter in the vehicle transmission showed that the larger torque converter gave a small improvement (1 to 2%) under all accessory load conditions when the idle speed was 4.7 Hz (250 RPM). However, if the idle speed is raised to 8.33 Hz (500 RPM), the larger torque converter gives poorer fuel economy (11 to 12%) depending on accessory load.

3. Effect of Idle Speed and Torque Converter Performance

The effects of idle speed and torque converter performance are related because of the interaction of the torque converter with the expander during idle conditions. This relationship determines the minimum system steam flow. Idle speed means the speed at which the expander operates when the vehicle is stopped, with the automatic transmission in gear and the brakes applied to keep the vehicle stationary. Since the torque converter is rotating, the expander must produce a given amount of power to overcome hydraulic resistance. The amount of input torque required by the torque converter is a function of its size; and since torque increases with the square of the input speed, the power required by the torque converter is a function of its size; and since torque increases with the square of the input speed, the power required increases as the cube of speed. In addition to torque converter demand there is also power required by all accessories and auxiliaries during the idle condition. With a given expander and steam generator performance the steam flow and fuel flow required to maintain an idle speed is fixed by the total power and speed required.

The above assists in explaining the effect of expander idle speed on fuel economy. Referring to the performance characteristics of the smaller torque converter 27.3 cm (10.75" dia) the torque needed to rotate the torque converter at given input speeds (expander speed x step-up gear ratio) with a speed ratio of zero (vehicle stationary) is 17 Nm at 8.75 Hz (12.5 lb ft at 525 RPM) 4.17 Hz expander idle (250 RPM) and 66.4 Nm at 17.5 Hz (49 lb ft at 1050 RPM) 8.33 Hz expander idle (500 RPM) and powers of .93 and 7.3 kW (1.25 hp and 9.8 hp) respectively. When this power is added to that for accessories (normal value without air conditioning) and auxiliaries, and idle steam flow for the two cases is 16.5 kg/hr and 66.2 hg/hr (80 lb/hr and 146 lb/hr) respectively. Using the latter values during the driving cycle would obviously give poorer fuel economy.

Using the performance characteristics for the larger torque converter 29.8 cm (11.75" dia) the values obtained under the same conditions are:

- a. For 4.17 Hz (250 RPM) idle speed. 25.8 Nm (19 lb ft) torque, 1.4 kW (1.9 hp) and 38.6 hg/hr (85 lb/hr) minimum steam flow.
- b. For 8.33 Hz (500 RPM) idle speed. 103.1 Nm (76 lb ft) torque, 11.5 kW (15.2 hp) and 88.5 kg/hr (195 lb/hr) minimum steam flow.

These values again show why there is poor fuel economy during the driving cycle with the higher expander idle speed. Both they did not compare the performance of the two torque converters for a vehicle in motion, at any given instant of time during the driving cycle, where velocity and acceleration were specified. For a 2087 kg (4600 lb) vehicle with a specific drive train and torque converter the torque required at the wheels was determined by the EPA driving cycle. The larger torque converter could produce this output condition more efficiently (generally with a higher speed ratio) than the smaller one and thus required a smaller input power. Consequently the larger torque converter gave two distinct results:

1. It required larger power input to produce a given expander idle speed (and any other condition when the system steam flow might be less than at idle) but required smaller power under other conditions.

2. The fuel economy for the cycle depended on the extent one effect offsets the other. With the conditions presented previously, it can be seen that with a high idle speed, the negative effect of increased idle steam flow greatly outweighs the positive effect the larger more efficient torque converter has during running conditions.

4. Analysis of Fuel Economy Studies

Studies of fuel economy obtained by computer simulation of the Federal Driving Cycle (FDC) and steady speed conditions are presented for:

- 1) Updated Steam Generator (SG) and Combustion Air Blower (CAB) performance, with a parametric study of steam generator performance.
- 2) Effect of simplified drive system for the CAB.
- 3) Parametric study of condenser-fan drive input power when the fans are not required.
- 4) Parametric variation in accessory load for the above three conditions.

In addition, the program used to simulate the driving cycle was modified to accept as input, any arbitrary driving cycles which can be defined by vehicle velocity versus time in one second intervals. This modified program was used to simulate the EPA Highway Fuel Economy Test (HWFET) driving cycle. All the results presented used the larger A731 torque converter 29.8 cm (11.75" dia) and lower expander idle speed of 4.17 Hz (250 RPM) which were found to give the better fuel economy in an earlier study.

Some general conclusions were made from this analysis:

- 1) The updated (increased) combustion air blower power requirement had a minor effect on fuel economy. The maximum change was only 0.6% at 32.19 km (20 mph) and decreased to 0.3% at higher steady speed.

2) Steam generator efficiency had a significant effect on fuel economy. There was a one-to-one relationship between efficiency and fuel economy.

3) The four cases considered had a 6 to 7% change in fuel economy on the FDC, up to 10% change at 32.19 km (20 mph), and a 5% change at 80.47 km (50 mph).

4) The heat loss from the exterior of the steam generator accounted for up to 2/3 of the total changes.

The second part of the systems fuel economy work was an investigation of an alternative drive method for the combustion air blower. The CAB drive used an Eaton hydrostatic transmission (integral variable displacement pump; fixed displacement motor). This unit was reasonably efficient; however, due to size and speed limitations the overall drive (transmission, sheaves, belts, and support structure) was heavier than desired. With the new SES blower, it was possible that a simpler, lighter weight drive could be used. The final drive system used a fixed displacement hydraulic pump, a fixed displacement hydraulic motor mounted directly on the blower shaft and a by-pass flow valve for blower speed control. Sufficient data was not available to be able to predict the overall efficiency of this drive under various conditions, but it is believed to range from 30 to 70%, improving with increasing blower speed and power.

Fuel economy simulations were run assuming a constant 30% drive efficiency, as a conservative prediction. The fuel economy penalty using the simpler drive was small being .09 km/l (0.23 mpg) (1.5%) under the worst conditions of 32 km/hr (20 mph), and .06 km/l (0.14 mph - 1.4%) during the FDC for the worst case.

A third portion of the study was an investigation of the effect of condenser fan drive power on Fuel economy. Earlier analysis had shown that the condenser fans were not required during most steady speed conditions, since ram velocity pressure could supply sufficient air flow through the condenser core to give full condensation of the steam. Computer simulation of the condenser fan requirements during the driving cycle indicated the fans were needed during only 30 to 40% of the cycle time (driving acceleration at moderate rates). In these analyses, it was assumed that the fans could be completely disengaged, thus using no power. In the preprototype system, the input shaft of the fan drive hydraulic pump would always be coupled to the expander, and thus use some power even at zero pump displacement. Tests conducted showed that this minimum pump power could be considerable at high speed.

Computer simulations were run to determine what effect the minimum fan drive pump power had on fuel economy. Three cases were compared: first, a drive system where the fans were fully disconnected by means of a clutch, when not needed; second, a system where the pump

remained engaged, but had only mechanical friction losses; and third, the existing preprototype system where the pump had mechanical losses plus "charge pump" losses (associated with internal pump by-pass and hydraulic fluid cooling requirements). Having the fan drive pump coupled to the expander at all times imposes a significant penalty on fuel economy. Conclusions reached were:

- 1) During the Federal driving cycle, fuel economy was reduced by 2 to 2.5%, or up to .11 km/l (.25 mpg).
- 2) At all steady speeds the penalty was worse: over 2% at low speeds and over 3.5% at high speeds, and up to .21 km/l (0.5 mpg).
- 3) Mechanical friction in the pump accounted for over 75% of the total losses during FDC and low speed cruise.
- 4) Eliminating the "charge pump" loss alone, without uncoupling the drive pump, provides only a small advantage.

The final portion of the fuel economy study was concerned with the EPA Highway Fuel Economy Test (HWFET) driving cycle. The computer program used to simulate the driving cycle was one that was modified to accept as input data any driving cycle defined by vehicle velocity at one second intervals. The program determined acceleration rates required to follow the specified cycle, and the total elapsed time and distance of the cycle. The simulation for fuel economy on this cycle was run for many of the conditions discussed earlier.

The FDC was designed for vehicle emission testing, which was to be determined as a weighted average of both cold start and hot start tests. The highway cycle is used to determine fuel economy, with a fully warmed-up running engine.

It would be expected, therefore, that the fuel economies determined using these two cycles would be considerably different. Comparing the cycles starting with the same power plant condition, either warmed-up or not, the highway cycle gave about 40% better fuel economy. Comparing the highway cycle, starting with a fully warmed-up vehicle, to the FDC which includes some warm-up, there was a difference of about 50%.

9.3.2 PPT PERFORMANCE RESULTS

A series of tests was conducted with the expander driving all auxiliaries except the atomizing air pump. Previous maps were presented in terms of "gross power" (gross hp), which was the power before auxiliaries, for ease of comparison with data. For this testing, the expander maps were updated in terms of "net expander power" (net expander hp), using all of relevant previous data (without expander driven auxiliaries) corrected to the same basis.

It had been demonstrated that the condenser fans were not required during most steady speed cruise conditions; thus, the preprototype system was equipped with a clutch which completely disengaged the hydraulic fan drive pump when the fans were not needed.

1. Total Heat Balance of Engine System

Testing was completed on the L-4 expander preprototype engine to determine heat and power flow through the engine compartment at various simulated vehicle speeds from 48 km/hr to 91 km/hr (30 mph to 60 mph). Measurements of various temperatures and flow rates through the engine shroud as the control volume were used to calculate the distribution of energy in the system. The steam conditions were $811^{\circ}\text{K}/6.89\text{ MPa}$ ($1000^{\circ}\text{F}/1000\text{ psi}$), and an external water-cooled condenser was used. Figure 9.7 is a schematic of the energy flow.

The heat input was calculated from the known chemical energy of the fuel and air mixture. The net expander output was measured by a dynamometer. Calculating the condenser heat loss was complicated by the fact that an accurate measurement of the steam flow was not possible to within $\pm 10\%$ at the low 68 - 159 kg/hr (150 - 350 lb/hr) steam flow of the test. Therefore, an analytical method of calculating steam flow rate based on a heat balance around the cooler was used. The steam temperature and pressure leaving the engine compartment, the water temperature and pressure entering the compartment, and the flow rate sufficed to give the condenser heat loss. This heat loss is accurate to within $\pm 5\%$. The steam generator stack loss was calculated from the emissions data and is accurate to within $\pm 9\%$. The valve stem and crankcase blow-by heat flows were found by measurement of actual water flows and temperatures. The heat flow to the air inside the shroud was calculated indirectly, since an actual measurement of the air dynamic

ENERGY FLOW IN SYSTEM

FIGURE 9.7

pressure from which the air velocity and mass flow rate could be found failed due to instrumentation problems. The air mass flow through the front of the engine was found graphically from the pressure drop through the air cooled condenser, which formed the air intake portion of the control volume. The shroud exhaust air was then found from the inlet air flow and the steam generator air flow. The inside shroud exhaust heat was then calculated from air mass flow and temperature changes from inlet to outlet. This result is probably accurate to $\pm 20\%$ due to leakage and measurement variances. The heat flow to the air outside the shroud was found by using a natural convection model with an average shroud temperature of 344°K (160°F).

The testing showed that the reversed valve configuration reduced the valve leakage from 10,260 Watts (35,000 BTU/hr) to 3,080 Watts (10,500 BTU/hr). The disagreement between the heat input and heat output of 4% at 48 kg/hr (30 mph), 3% at 64 km/h (40 mph), 6% at 80 km/h (40 mph) and 1.5% at 97 km/h (60 mph) was due to experimental uncertainties. Lowering the condenser heat loss by 5% would be enough to make the heat input and output agree.

2. Net System Horsepower Map

A net system horsepower map was completed utilizing L-4 test data. The map incorporated the following configuration changes:

1. Valve stem seals
2. Desuperheater
3. Direct steam line from boiler to expander
4. Heated compartment air
5. Low manifold pressure for low horsepower operation.

Figure 9.8 presents the results of adding the above to the map. This map represents the net system horsepower into the transmission (i.e., the net after all engine auxiliaries, but with no allowance for vehicle accessories such as alternator, power steering, air conditioning).

3. Light Load Performance Results

Tests were conducted on the L-4 system to ascertain the influence of boiler pressure and automatic transmission on fuel flow and efficiency at expander conditions near idle.

Test points were obtained at boiler pressures of 4.8 MPa (700 psi) and 6.9 MPa (1000 psi), cut-off angles from .09 rad to -.07 rad (5° to -4°), expander speeds of 6.6 Hz (400 RPM) and 8.33 Hz (500 RPM) with and without a transmission and 29.8 cm (11.75") diameter torque converter in the system. The Model No. 5 boiler/burner was used.

NET SYSTEM POWER MAP

L-4 EXPANDER 88.9mm x 88.9mm (3.5" x 3.5") BORE & STROKE
REVERSED VALVES

60° ADMISSION CAM-75° CUTOFF CAM

MAX RPM 2500 (41.6Hz) MAX PISTON SPEED 7.4m/sec (1458 ft/min)

(RAW DATA PLUS CORRECTIONS)

FIGURE 9.8

The condition simulated in this case was that of idle creep, i.e., a car with its transmission in third gear, selector in "Drive" and no braking. For an expander speed of 8.33 Hz (500 RPM) and .09 rad (5°) cut-off, the simulated vehicle speed was about 40.2 km/hr (25 mph). The torque converter speed ration was between .9 and .96 during the testing.

Figure 9.9 shows the fuel flow in the system as a function of the cut-off angle and boiler pressure. For the expander speed of 8.33 Hz (500 RPM) one can see that the fuel flow of 7.47 kg/hr (16.5 lb/hr) at conditions of 6.89 MPa (1000 psi), .09 rad (5°), can be reduced to 5.4 kg/hr (12 lb/hr) either by staying at .09 rad (5°) and lowering the pressure to 4.8 MPa (700 psi), or by lowering the cut-off to -.02 rad (-1°) and staying at 6.89 MPa (1000 psi).

For a given cut-off angle, the idle data showed that the fuel consumption was lower by about 20 - 25% when the boiler pressure was lowered from 6.89 MPa (1000 psi) to 4.8 MPa (700 psi). The relationship between fuel flow and cut-off at constant expander speed and boiler pressure was linear at cut-off angles between .09 rad (5°) and .52 rad (30°), so the relationship was assumed to be linear at the zero and negative cut-off range.

Figure 9.10 shows the fuel flow in the system as a function of net system power, steam pressure, and the presence of the automatic transmission. When the automatic transmission is added, with steam pressure at 6.89 MPa (1000 psi), the net system power drops by 3.3 kW (4.5

FUEL FLOW vs CUT OFF @ EXPANDER SPEED = 8.3 Hz (500 RPM)

FIGURE 9.9

hp) at the same fuel flow. With 4.8 MPa (700 psi) steam pressure, for a constant fuel flow, the automatic transmission loss is therefore about 2.2 kW (3 hp) at the test conditions.

In the other direction, the graph shows that the fuel flow must increase by about 2.7 kg/hr (6 lb/hr) to maintain the same brake power at 6.89 MPa (1000 psi) steam pressure and about 2.2 kg/hr (5 lb/hr) at 4.8 MPa (700 psi) steam pressure when the automatic transmission is installed. For the expander conditions tested, the test data imply transmission and torque converter efficiencies between 45% and 75%. In normal operation at vehicle speeds from 40 to 128 km/hr (25 to 80 mph) the automatic transmission should be about 80% efficient. The low efficiencies observed are due to the near constant converter spin loss and front pump work as these losses dominate the efficiency at light load.

4. Condenser Fan Requirements for Cruise Conditions

An analysis of condenser fan requirements for the L-4 system with a 27.3 cm (10.75") torque converter was done to show more exactly the regions on the net system map which required the use of condenser fans.

The analysis was similar to previous work where knowledge of the heat rejection rate to condenser, condenser temperature effectiveness, core pressure drops, installation loss and the drive train combined to

APPARENT TRANSMISSION LOSS AT LIGHT LOADS

EXPANDER SPEED 8.3Hz (500 RPM)

FIGURE 9.10

give the maximum power which the system can deliver at a given vehicle velocity without requiring condenser fans. However, the previous analysis assumed a constant torque converter speed ratio of .9. The results given in Figure 9.11 were calculated from actual speed ratios which are a function of output speed and torque of a 27.3 cm (10.75") torque converter and a transmission in third gear. Any region below the threshold does not need the fans to produce condensation since the ram air flow alone is sufficient. Note that the fans are not required for cruise conditions.

5. Condenser Fan Power Requirements

Test data on the preprototype L-4 system was analyzed to provide information on the condenser fan performance and power requirements. Figure 9.12 presents the fan shaft input power as a function of vehicle speed and steam flow. The fan shaft input power is the shaft power to the fans only and does not include the hydraulic fan drive input power. The map presented is general and will apply to a range of fan drive systems.

6. Automatic Transmission Modeling and Testing

Performance of the 29.8 cm (11.75") diameter torque converter and transmission at speed ratios near unity and expander speeds near 8.33 Hz (500 RPM) are discussed in the section on idle testing. Testing was done on the L-4 system with the automatic transmission at simu-

CONDENSER FAN REQUIREMENTS AS A FUNCTION
NET SYSTEM POWER AND EXPANDER SPEED

FIGURE 9.11

FAN SHAFT INPUT POWER vs VEHICLE SPEED AND STEAM FLOW

(COND. PRESS. $\sim 0.145 P_0$ (21.0 PSIA))

FIGURE 9.12

lated steady-state speeds from 16 to 128 km/hr (10 to 80 mph) and variable boiler pressure. Three data points were obtained for each test point in order to increase measurement reliability.

Figure 9.13 shows a graph of torque converter input power versus expander speed for the 29.8 cm (10.75") diameter torque converter at zero speed ratio, which was developed from information furnished by Chrysler. Because of the fluid nature of the torque converter, the input torque is proportional to the square of the input speed. Therefore, the input power is proportional to the cube of the input speed.

For the 29.8 cm (11.75") torque converter, one can see that reducing the idle speed from 8.33 Hz (500 RPM) to 6.7 Hz (400 RPM) reduces the required power from 11.3 kW (15.2 hp) to 5.8 kW (7.8 hp). Reducing the idle speed to 5 Hz (300 RPM) reduced the required power to 2.5 kW (3.4 hp). The 27.3 cm (10.75") torque converter behaved in an analogous fashion.

7. PPT Transient Tests

The PPT test set-up simulated the total SES steam system as a vehicle with the basic components: Steam generator, expander, transmission, feedpump, condenser, and auxiliaries. The dynamometer was controlled to

CALCULATED TORQUE CONVERTER IDLE CHARACTERISTICS
(AT ZERO SPEED RATIO)

FIGURE 9.13

simulate the road load, and the flywheel was used to simulate the vehicle dynamics. In addition, the controls were integrated for completely automatic control operation.

The flywheel was designed to simulate a 2089 kg (4600 lb) vehicle, but the design was based on the then-planned 3.37 rear axle ratio. Subsequent to that, the rear axle was changed to 3.21 (Chrysler production axle ratio closest to the 3.37). Consequently, the flywheel simulated only a 1900 kg (4200 lb) vehicle with a road load torque function set on the dynamometer representing a 2087 kg (4600 lb) vehicle.

The tests were divided into steady-state tests, acceleration tests and hot and cold start-ups for determining self-sustaining capabilities.

Driveability tests were performed by an externally controlled function generator which changed the pedal position. The objective of the driveability tests was to adjust all the control functions and gains to attain a balance among stability, dynamic performance and fuel economy.

The biggest progress in the SES steam engine was noted in the following areas:

1. The excellent and smooth start
2. The controllability had improved by an order of magnitude

3. The system was responsive and driveable
4. Emissions were good although more attention to dynamics was required
5. Good gear shifts

Vibration Problem: During system start-up, there was a noticeable increase in the vibration or noise level. The noise started at idle and remained up to about 20 Hz (1200 RPM). In essence, the noise originated from repeated disengagement and engagement of the planetary gear mesh caused by crankshaft torque fluctuations and torque reversal. The inertia of the torque converter was approximately 3.5 times the inertia of the expander (including flywheel). As a consequence, the planetary step-up gear serving as the transferring member between the expander flywheel and the torque converter did not provide sufficient rigidity to transfer the fluctuating torque completely and not enough flexibility to absorb the excessive energy, resulting in a torque discontinuity transfer that gave noise and vibration.

The solution was a set of helical springs placed between the flywheel and the outer ring gear of the step-up transmission. The simulator, being a softer system, did not suffer the drive train problems experienced on the PPT.

The step-up gear loss model was determined from test data supplied by Borg Warner. Analysis of the data indicated that the step-up gear loses 10% of the

transmitted power at low loads and 5.5% at high loads. This was viewed as approximate since the Borg Warner tests were run with an electric motor on a gear box operating as a step-down gear. It was decided to operate the simulator vehicle without the torque converter in view of high torque converter losses.

8. Effect of Steam Pressure on BSFC at Road Load

A question was raised as to whether variable steam pressure could reduce BSFC. Tests were run in May 1973 on the PPT system which measured gross horsepower (normal valve arrangement) and in May 1974 which measured net power without transmission. The results of the tests are that a slight tendency to lower BSFC's at 48, 64, 80 km/hr (30, 40, 50 mph) occurred at steam pressures around 5.2 MPa (750 psia). However, the scatter in the points was about equal to the estimated reduction (5 to 8%) in BSFC. Consequently, additional tests were run at varied steam pressure:

Data were collected for two PPT systems with a transmission installed. One with the Model No. 7 and the other with Model No. 8 steam generator and water-cooled condenser with desuperheater. Road Load power levels and speeds were calculated on the basis of a 3.21 rear axle and an effective wheel radius of 35 cm (1.15 ft) at all vehicle speeds.

Only at 48 km/hr (30 mph) could one see any consistent tendency for lower BSFC at lower steam pressures. At 64 and 80 km/hr (40 and 50 mph), the data points showed essentially the same BSFC with tendencies of the order of 3% lower BSFC at lower pressures. At 96 km/hr (60 mph) there was no improvement going to lower pressures.

9. Performance Test Analysis

Acceleration tests were conducted for the following cases:

1. 0 to 96.5 km/hr (0 to 60 mph)
2. 40 km/hr to 112.6 km/hr (25 to 70 mph)
3. 80.5 km/hr to 128.7 km/hr (50 to 80 mph)

In general, the tests were carried out smoothly. The system exhibited good stability and gear shifts without operational problems.

The acceleration times were higher than the ERDA specification requirements. For example, the 0 to 96.5 km/hr (60 mph) case called for 13.5 seconds to reach 96.5 km/hr (60 mph), and the test results showed 20.8 seconds. Similarly, the 40.2 km/hr (25 mph) to 112.6 km/hr (70 mph), and the test results show 25 seconds. The key factors that caused these penalties were identified as follows:

1. The low boiler pressure at idle operation hindered the power response. The solution is the introduction of a steam throttle valve.

2. Due to a slight mismatch in the boiler/expander operation, the steam pressure excursions of 2.1 MPa (300 psi) reduced the engine power.

3. Another effect of the mismatch limited the maximum cut-off angle to .70 rad (40°), further reducing the engine power.

4. The high drive train losses were mentioned earlier. The penalty here was about 20% above the predicted loss.

5. Clutch engagement was done manually with conservative loading rates.

9. Start-up Test Analysis

The engine started automatically by turning the start key. The start was smooth with the steam generator firing first, and on start of pressure build up, the expander turned and came up to a self-sustaining power level. The start-up control logic designed into the control system is presented in the Controls Section.

In general, start-ups were very successful, and the system was self-sustaining in 20 to 30 seconds. Table 9.1 presents a summary of the results for various criteria with 2 seconds included in the results for the time from the key start to ignition. Both hot and cold start results are presented. A hot start is defined as one where the shut-down period prior to the start had not exceeded 30 minutes.

Table 9.I also shows the average time to reach the self-sustained condition. The fuel used is about 0.23 kg (0.5 lbs). The steam temperature after 30 seconds is approximately 588.7°K (600°F).

In addition to the presented data, it should be noted that the system was started every day for approximately two months, and various start-up modes were tried.

9.3.3 SIMULATOR VEHICLE RESULTS

Simulator Data Reduction and Analysis Program

A data reduction and analysis program was developed based on PPT experience for the Chassis Dynamometer Simulator tests. It was used to analyze system performance for all steady-state performance tests on the Chassis Dynamometer and component performance analysis. The simulator had less instrumentation compared to the PPT engine dynamometer test facility.

The data acquisition program would accept, interpret and calculate data for the vehicle simulator and also print out calibrated and calculated values including heat balance, expander efficiency, steam generator efficiency, emissions and vehicle and power train efficiencies.

A program of updating L-4 test data normalized the data to a common base. The base approximated the simulator engine and included:

TABLE 9.I

TIME IN SECONDS

<u>TYPE START</u>	<u>TIME TO 2.07 MPa (300 PSIA)</u>	<u>TIME TO 5 Hz (300 RPM)</u>	<u>TIME TO 11.7 Hz (700 RPM)</u>	<u>OPERATOR TURNS OFF STARTER</u>	<u>STEAM TEMP @ OPERATOR TURN OFF</u>	<u>PRESSURE</u>		<u>FUEL USED KG (LBS)</u>
						<u>5 Hz</u>	<u>11.7 Hz</u>	
						<u>@(300 RPM)</u>	<u>@(700 RPM)</u>	
						<u>MPa (PSIA)</u>		
HOT	22	22.4	23.6	28	460.9°K (370°F)	2.18 (316) 2.62 (380)		0.31 (0.68)
HOT	21.8	22.4	25	28	533.2°K (500°F)	3.26 (473) 3.26 (473)		0.22 (0.48)
COLD	25	30	33.6	32	644.3°K (700°F)	4.81 (697) 4.83 (700)		0.23 (0.51)

TIME INCLUDES 2 SECOND BLOWER PURGE

1. Expander valve stem seals
2. Desuperheater
3. Drive train step-up gear
4. Model No. 8 steam generator
5. Direct steam line from boiler to expander
6. Heated compartment air

Test data from components not conforming to the normalized base was corrected:

The effect of the desuperheater on fuel flow and steam generator performance was found from test data and the steam generator computer program. Heating the feedwater to the steam generator decreases the amount of fuel needed to heat the water to boiling, but it also can lower the steam generator efficiency since the exhaust stack gases leave at a higher temperature and consequently higher exhaust energy. The steam generator computer program indicated how these two trade-offs affected the fuel flow.

The test data showed that the desuperheater heated the feedwater to about 263°K (13°F) below the saturation temperature of the expander exhaust steam. Consequently, for a given test point the maximum feedwater temperature rise was calculated knowing the feedwater outlet temperature and the exhaust steam pressure. Then the change in fuel flow resulting from the calculated feedwater temperature rise for a given steam generator was used to determine a corrected fuel flow.

The expander valve steam leakage correction was found from the measured leakages in prior testing, assuming a 75% seal effectiveness. A linear function of cut-off angle was chosen to fit the data and the steam flow and fuel flow correction resulting from the seals was found accordingly.

The drive train step-up gear power loss was a function of the gear efficiency and the power being transmitted through the gear. A 96% efficiency was used.

Simulator System Results

Steady-state, driveability, HWFDC and UFDC data were analyzed for the drive train configuration without the torque converter.

A detailed study was conducted for steady-state simulator data in the 56 to 80 km/hr (35 to 50 mph) range. For the 63 km/hr (39 mph) second gear case, the raw data showed a fuel consumption of 7.3 kg/hr (16 lb/hr) resulting in a fuel economy of 6.6 km/liter (15.5 mpg). The fans were operating at 313°K (104°F) ambient temperature and using 1.4 kW (1.9 hp). These results correlate well with earlier test data. The gross expander power is 13.7 kW (18.4 hp), and the steam flow is 99.3 kg/hr (219 lb/hr). If a correction is made for the 1.4 kW (.19 hp) fan power, the fuel economy for the 63 km/hr (39 mph) point would be 7.3 km/liter (17.3 mpg).

Driveability

Numerous gear shifts and start-ups were performed in the simulator vehicle with no significant problems. The time to shift was comparable to the PPT test results: 0.3 seconds for the first-to-second gear shift and approximately 0.5 seconds for the second-to-first down shift.

Highway Federal Driving Cycle

Figure 9.14 presents a typical trace of the SES system operating on the HWFDC compared to the ERDA baseline for the HWFDC. The SES steam system followed the trace except for the start and finish. This was not a system problem. The start was not lined up because the two points were initially not timed. The SES steam system had an initial acceleration close to the desired ERDA curve as can be seen in the figure.

Also the finish does not line up because the vehicle deceleration during the braking phase was less than expected by the operator. Future tests improved on this by having the operator simply brake harder at the finish.

The above test resulted in a fuel economy of 4.43 km/liter 10.25 mpg). A clutch slippage study was made during the HWFDC which revealed no slippage, except at the 11.7 Hz (700 RPM) point where the clutch is engaged and re-engaged.

SES SYSTEM RESULTS FOR HIGHWAY
FEDERAL DRIVING CYCLE COMPARED
TO BASELINE HIGHWAY FEDERAL
DRIVING CYCLE

FIGURE 9.14

FIGURE 9.15

FIGURE 9.16

URBAN AND HIGHWAY CYCLE RESULTS

4500 POUND INERTIA WEIGHT

<u>URBAN FDC RESULTS (1975 FTP)</u>				
	MPG	NO _x (GMS/MILE)	UHC (GMS/MILE)	CO (GMS/MILE)
ORIGINAL 76 STANDARDS	-	0.40	0.41	3.4
29 OCT 75	6.8	0.34	0.47	4.4
30 OCT 75	7.1	0.36	1.10	6.9
4 NOV 75	7.5	0.26	0.58	3.7
6 NOV 75	7.5	0.25	0.37	3.4
14 NOV 75	7.0	0.28	0.36	2.1

<u>HIGHWAY CYCLE</u>	
AUG 75	10.2 MPG (105°F AMBIENT)
5 NOV 75	11.9 MPG
6 NOV 75	12.2 MPG
13 NOV 75	12.4 MPG

SUPPLEMENT TO: CONTRACTORS' COORDINATION MEETING
 DIVISION OF TRANSPORTATION, ENERGY CONSERVATION
 ERDA CONTRACT E(11-1)-2701
 SCIENTIFIC ENERGY SYSTEMS CORPORATION
 NOVEMBER 1975

TABLE 9.II

Urban Federal Driving Cycle

Results of the UFDC were encouraging with respect to emissions and fuel economy. The throttle valve was installed for the UFDC testing. A typical trace is shown in Figure 9.15, for bag 1 to 340 seconds. Figure 9.16 shows eight pertinent analog traces associated with Figure 11.16.

A summary of simulator performance results is presented in Table 9.II.

The vehicle inertia weight used in these tests was 4,500 pounds. This was based on the estimated vehicle weights presented in Table 9.III. The preprototype weight represents the estimated system weight as tested except that test instrumentation was deducted and prototype component weights were assumed for those components not included in the test. The prototype weight column is the result of a brief study to estimate the weight reduction possible in a short term redesign of the system without changing the steam cycle or expander displacement. The prototype was estimated to weigh only 289 pounds more than an equivalent performance production car.

TABLE 9.III

FULL SIZE 1972 CAR WEIGHTS WITH STEAM ENGINES

	<u>Preprototype</u>	<u>Prototype</u>
Powerplant		
Expander	570	420
Condenser	113	80
Condenser Fans	66	20
Vapor Generator	96	90
Feed Pump	16	16
Auxiliary Drives	100	65
Working Fluid, 5 Gal.	42	42
Lubricant	12	8
Freeze Protection System	25*	25
Miscellaneous	<u>118*</u>	<u>118</u>
Total Powerplant Weight	1158	884
Equivalent 318 CID V-8		
Engine and radiator	<u>595</u>	<u>595</u>
Additional System Weight	563	289
Other		
Heavy-duty transmission increase	32	32
Fuel, 20 Gal	<u>126</u>	<u>126</u>
Total other	158	158
Vehicle Curb Weight**	4639	4365

* Prototype components assumed

** 1972 Plymouth Fury dry weight, V-8, air conditioning, and automatic transmission: 3918 lbs.

10.0 PROTOTYPE ENGINE STUDIES

The subject contract included scope for design and long lead procurement of a second generation or prototype engine for a compact car. The contract was ultimately terminated during the prototype design study phase and thus a definitive design and performance specification was never completed. The following discussion therefore touches on early studies and is only representative of the spectrum of possible designs.

Two engine designs will be discussed to illustrate the range of designs contemplated and some of the potential of future automotive steam engines. The first engine, designated Mark I, represents a current cycle with rematched components to maximize the fuel economy with the current component state-of-the-art. The Mark II engine is designed to maximize materials utilization in a high efficiency moderate cost engine. The resultant engine performance estimates are then used to calculate the cruise fuel economies in a compact car for comparison with other candidate engines.

The Mark I Engine - Current Cycle - The current SES/ERDA steam engine cycle is illustrated in Figure 10.1 and the energy balance, component and system net efficiencies are presented in Table 10.I.

A second generation engine operating on the current cycle has been studied as a baseline for the best current technology. Assuming no improvement in any component peak efficiencies the Mark I design incorporates only

MARK I	CRUISE CYCLES	MARK II
1000 PSIA, 1000 ⁰ F NONE	BOILER REHEAT	1500 PSIA, 1250 ⁰ F 200 PSIA, 1500 ⁰ F
20 PSIA, 270 ⁰ F NONE	EXPANDER EXHAUST FIRST STAGE SECOND STAGE	200 PSIA, 810 ⁰ F 8 PSIA, 830 ⁰ F
NONE	RECUPERATOR FLUID EXIT VAPOR EXIT	450 ⁰ F 255 ⁰ F
20 PSIA, 230 ⁰ F	CONDENSER	8 PSIA, 180 ⁰ F

FIGURE 10.1
CURRENT CYCLE AND REHEAT CYCLE
STATE POINTS AND TEMPERATURE
VERSUS ENTROPY

better component matching. The expander displacement to power match provides the major improvement. The low speed cruise fuel economy can be increased by approximately 15% by doubling the expander design point mean effective pressure with increased maximum cut-off. The peak engine efficiency will not change but the peak efficiency island will move down from 50% power to 25% power and the peak efficiency island will also move across the speed range towards the road load curve. At this higher specific power, a smaller, higher rpm expander results without increasing piston speed. The smaller expander will retain the high torque capability with the higher cut-off. The high torque at low speed will eliminate the need for a torque converter for low speed acceleration.

Improvements to this simple cycle approach are possible without resorting to superalloy components. Stainless steel boilers can be designed to deliver steam at 1500°F at over 1000 psi. A simple cycle engine using 1250°F, 1000 psi steam would increase the net engine efficiency an additional 14% and reduce specific steam flow 21%. The higher temperature simple cycle would be directly competitive in fuel economy with current spark ignition engines. The higher specific power would further reduce engine size, weight and cost.

The Mark II Engine - Reheat Cycle - An advanced reheat cycle steam engine could operate on the cycle illustrated in Figure 10.1. This cycle incorporates a single reheat pass mid-way in the expansion process, higher initial pressure, higher temperatures, recuperation and lower condenser pressure.

Assuming no improvement in component peak efficiencies in the heat balance and secondary losses, Table 10.I illustrates that the Mark II Engine peak efficiency could be 63% higher (39% lower fuel consumption) than the efficiency of the current Mark I SES engine. The specific power (work per pound of steam) would be doubled.

The reheat cycle results in a large reduction in heat exchanger size. Relative to the Mark I cycle at the same power level, the Mark II heat exchangers handle 50% less steam flow, 39% lower heat input and 15% less heat rejection. The boiler temperature/pressure schedule requires stainless steel only in the first stage superheater and in the reheater. The reheater can operate at a higher temperature than the first stage superheater as the steam pressure is lower.

Expander design studies have concluded that the two reciprocating stages' total displacement can be one cubic inch per horsepower. A maximum volumetric effectiveness of 30% in each stage would result in a peak efficiency in each stage of 70% at 50% power.

Lubrication of the expander would be the major applied research task required in the development of this engine. A counter flow expander design provides the most control of the piston ring environment. Steam would be exhausted through an exhaust valve in the head rather than through liner exhaust ports at the bottom of the stroke. This provides maximum separation of steam and oil. The piston rings can now be positioned well below the top of the piston as they no longer control the exhaust timing. The ring temperatures can then be selected by positioning the rings on the piston to operate in the cooler portion of the liner temperature gradient.

MARK I		BASE ENERGY BALANCE	MARK II	
+ 1246	{ + 1246 -	Boiler Heat Input (BTU/LB) Q_{IN1} Q_{IN2} Recuperator Q_R ($\epsilon = 0.9$) Condenser Q_c	+ 1204 + 372 275 - 1022	+ 1576
+ 5	+ 5	Pump W_p ($\eta = 0.6$) Expander W_1 ($\eta = 0.7$) W_2 ($\eta = 0.7$)	+ 8	+ 8
- 282	{ - 282 -		- 208 - 353	- 561
- 277		NET BASE WORK		- 553
(277/1246)	22%	NET BASE EFFICIENCY	35%	(553/1576)
(No Recuperator) (3%) 4% 1% 10% 15%		SECONDARY LOSSES Boiler LHV Loss @ 25% Load Burner Blower & Air Pump Expander Heat Rejection % Boiler Heat Input Step-Up Gear/Other Spin Loss Total Secondary Loss	4% 1% 5% 2% (No Step-Up Gear) 12%	
	19%	NET BRAKE EFFICIENCY	31%	
	10.8	SPECIFIC STEAM CONSUMPTION (LB.-Steam/HP-HR)	5.2	

TABLE 10.1
CURRENT CYCLE AND REHEAT CYCLE ENERGY BALANCES

The net engine performance envelope of the Mark II design would be essentially identical to the rematched Mark I engine in torque versus speed. The location and shape of the efficiency contours would be similar to, but proportionately more efficient, than the Mark I engine.

Total Mark II powerplant weight is estimated to be about $4 \frac{1}{2}$ pounds per horsepower. The component weight distribution would be 20% heat exchangers, 50% expander, and 40% auxiliaries and fluids.

A preliminary design of such an engine is illustrated in Figure 10.2. Two high pressure cylinders in the rear of a V-6 would feed six low pressure cylinders in front after a return to the top mounted boiler for reheating.

Vehicle Cruise Fuel Economy - Figure 10.3 results from a simple analysis of cruise fuel economy for several alternative engines. The ground rules were that: all engines and resultant cars weigh the same, the drive train efficiency is constant after the torque converter, and all engines are scaled to the same power at 50 mph. The gas turbine has slightly more power at 50 mph to compensate for the relatively flat power curve in the higher speed range. Published data was scaled, without changing the efficiency, and adjusted to a common vehicle accessory load where gross engine performance was available. The engine data sources and scaling requirements follow:

FIGURE 10.3

CRUISE FUEL ECONOMY
ALTERNATIVE ENGINES

Mark I Current Cycle Steam Engine - The estimated performance map derived for the rematched current cycle is given in Figure 10.4. Condenser fan power is not included. During steady cruise, vehicle ram air is sufficient for cooling up to an ambient temperature of 85°F.

Mark II Reheat Steam Engine - The estimated performance map of the reheat cycle engine is given in Figure 10.5. Condenser fan power is not required for steady cruise up to an ambient temperature of 85°F.

Spark Ignition Engine - This is a 1972 model year production Otto cycle engine (2*). The performance map for this engine is given in Figure 10.6.

Current Chrysler/ERDA Gas Turbine - The cruise fuel economy for this engine is a ratio of the Reference (3*) test car fuel economy by the inverse of the vehicle weights. The reference test vehicle includes a torque converter.

Chrysler/ERDA Gas Turbine Upgraded - This engine is a scale down of the Chrysler/ERDA 123 horsepower gas turbine (4*).

Diesel - The diesel projection is based on the Daimler-Benz 4-cylinder Diesel (5*). Two cylinders were added and then scaled up to the desired power.

Stirling - The Stirling engine is the projected performance for the Ford/Stirling 4-215, 170 horsepower engine (6*) scaled down to the desired power.

FIGURE 10.4
MARK I CURRENT CYCLE STEAM ENGINE

POWER STEERING AND BRAKES
NO CONDENSER FAN POWER

FIGURE 10.5

MARK II REHEAT STEAM ENGINE

HIGH GEAR PERFORMANCE

POWER STEERING & BRAKES
NO CONDENSER FAN POWER

FIGURE 10.6

SPARK IGNITION OTTO CYCLE ENGINE

POWER STEERING AND BRAKES

The steam engine's superior match to the duty cycle is evident in the relative cruise economies. The Mark I steam engine fuel economy at 50 mph is only 10% below that of the spark ignition engine economy even though the spark ignition engine has a peak efficiency 63% higher than that of the Mark I steam engine. The Mark II reheat steam engine has a better cruise fuel economy than the Stirling with identical peak engine efficiency. Some of this steam engine advantage would be lost in higher power duty cycles but the Mark II steam engine would continue to have superior fuel economy by virtue of its high peak efficiency.

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