DETAILED DESIGN

RANKINE-CYCLE POWER SYSTEM WITH ORGANIC-BASED WORKING FLUID AND RECIPROCATING EXPANDER FOR AUTOMOBILE PROPULSION

VOLUME I – TECHNICAL REPORT

Prepared for

Division of Advanced Automotive Power Systems Development

Environmental Protection Agency

Ann Arbor, Michigan

Prepared by

Thermo Electron Corporation

Research and Development Center

101 First Avenue

Waltham, Massachusetts

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DETAILED DESIGN

RANKINE-CYCLE POWER SYSTEM WITH ORGANIC-BASED WORKING FLUID AND RECIPROCATING EXPANDER FOR AUTOMOBILE PROPULSION

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1. ABSTRACT

The Division of Advanced Automotive Power System Development of the Environmental Protection Agency (EPA) is sponsoring part of the development of a low-emission Rankine-cycle propulsion system for automobiles at Thermo Electron Corporation (TECO); the Ford Motor Company is contributing both financially and technically to the development effort.

The system under development at TECO is based on use of an organic-based working fluid with reciprocating expander. The working fluid used is Fluorinol-85, a mixture of 85 mole percent trifluorinol and 15 mole percent water. In this report, a description is presented of the detailed, optimized design of the system including packaging of the complete system in the reference car, the 1972 Ford Galaxie. The results of experimental development in several critical areas are also presented. The system is designed to provide performance approximately equivalent to use of a 351 cubic inch displacement internal combustion engine in the reference car. Performance calculations indicated that a system designed to provide 131 hp net shaft horsepower (feedpump and condenser fan power subtracted) at an expander speed of 1800 rpm (equivalent to 90 mph vehicle speed) provides acceptable performance for the 1972 Ford Galaxie. Some predicted performance characteristics are:

Wide-Open Throttle Performance:

0-60 MPH, 0% Grade	13.4 seconds
Grade for 70 MPH Constant Speed	6.8%
Top Speed, 0% Grade	103 MPH

Fuel Consumption:

Steady Speed - 30 MPH	15.4 MPG
60 MPH	11.8 MPG
Federal Driving Cycle for Emissions	10.8 MPG

Emission measurements were made with a burner designed for a 100 shp Rankine-cycle system. The measurements were made with the burner operated transiently to simulate operation of a Rankine-cycle system over the Federal driving cycle for emission measurements. The Federal specified procedure for the measurements was followed exactly including use of the constant volume sampling unit. The measurements confirmed the low emission potential of the Rankine-cycle system:

	Emission Level, grams/mile	
Pollutant	Measured	Federal 1976 Standard
NO_x	0.29	0.4
CO	0.22	3.4
UHC	0.14	0.41

The next phase of the program involves development of all components and testing of the complete preprototype system in the laboratory.

2. SUMMARY

2.1 INTRODUCTION

The Division of Advanced Automotive Power System Development of the Environmental Protection Agency (EPA) is sponsoring development of a low-emission Rankine-cycle propulsion system for automobiles at Thermo Electron Corporation (TECO). The system under development at TECO is based on use of an organic -based working fluid with reciprocating expander. In this report, a description is presented of the detailed, optimized design of the system including packaging of the complete system in the reference car, the 1972 Ford Galaxie. The results of experimental development in several critical areas are also presented. The system is designed to provide performance approximately equivalent to use of a 351 cubic inch displacement internal combustion engine in the reference car. Experience gained in 1-1/2 years of testing of a complete 5-1/2 hp Rankine-cycle power system at TECO provides a firm technical basis for the design. The same working fluid and similar cycle conditions are used for the automotive system design as in the 5-1/2 hp system.

The Ford Motor Company (FOMOCO) is contributing both financially and technically to the development effort, particularly in the areas of: (1) integration of the system into the 1972 Ford Galaxie, (2) manufacturing considerations, (3) expander design, and (4) transmission design.

2.2 SYSTEM DESCRIPTION AND CHARACTERISTICS

Performance calculations indicated that a system designed to provide 131 hp net shaft power output (feedpump and condenser fan power subtracted) at an expander speed of 1800 rpm (equivalent to

90 mph vehicle speed), and with condenser and condenser fan characteristics based on ram air resulting from 90 rpm vehicle speed, provided acceptable performance for the 1972 Ford Galaxie relative to the EPA specifications.[1] All component sizes are, therefore, based on this design point condition. The working fluid used is Fluorinol-85;* the state point diagram for the system at the design point is presented in Figure 2.1.

The system schematic including all control functions is presented in Figure 2.2. The driver interface to the system is limited to the ignition switch, accelerator pedal, gear shift lever, and brakes as in present automobiles. System startup and operation are completely automatic.

2.2.1 System Integration and Packaging in 1972 Ford Galaxie

In the design of the components described in Section 2.3, an essential input has been packageability of the complete system in the 1972 Ford Galaxie with only minor modifications to the vehicle. The complete system layout is illustrated in Figure 2.3, a side view looking from the driver's side of the engine compartment, and in Figure 2.4, a view from the top of the engine compartment. Sectional views are provided in Figures 2.5, 2.6 and 2.7, as identified in Figure 2.3. As can be seen from these drawings, the expander-feedpump-transmission subassembly is located to the rear of the engine compartment, the condenser is located in the very front of the engine compartment with the condenser fans mounted to the condenser shroud on the rear of the condenser, and the combustion system-boiler subassembly is located between the expander and condenser and is placed as close as possible to the expander, leaving

*Halocarbon Products, Inc., Hackensack, N. J.

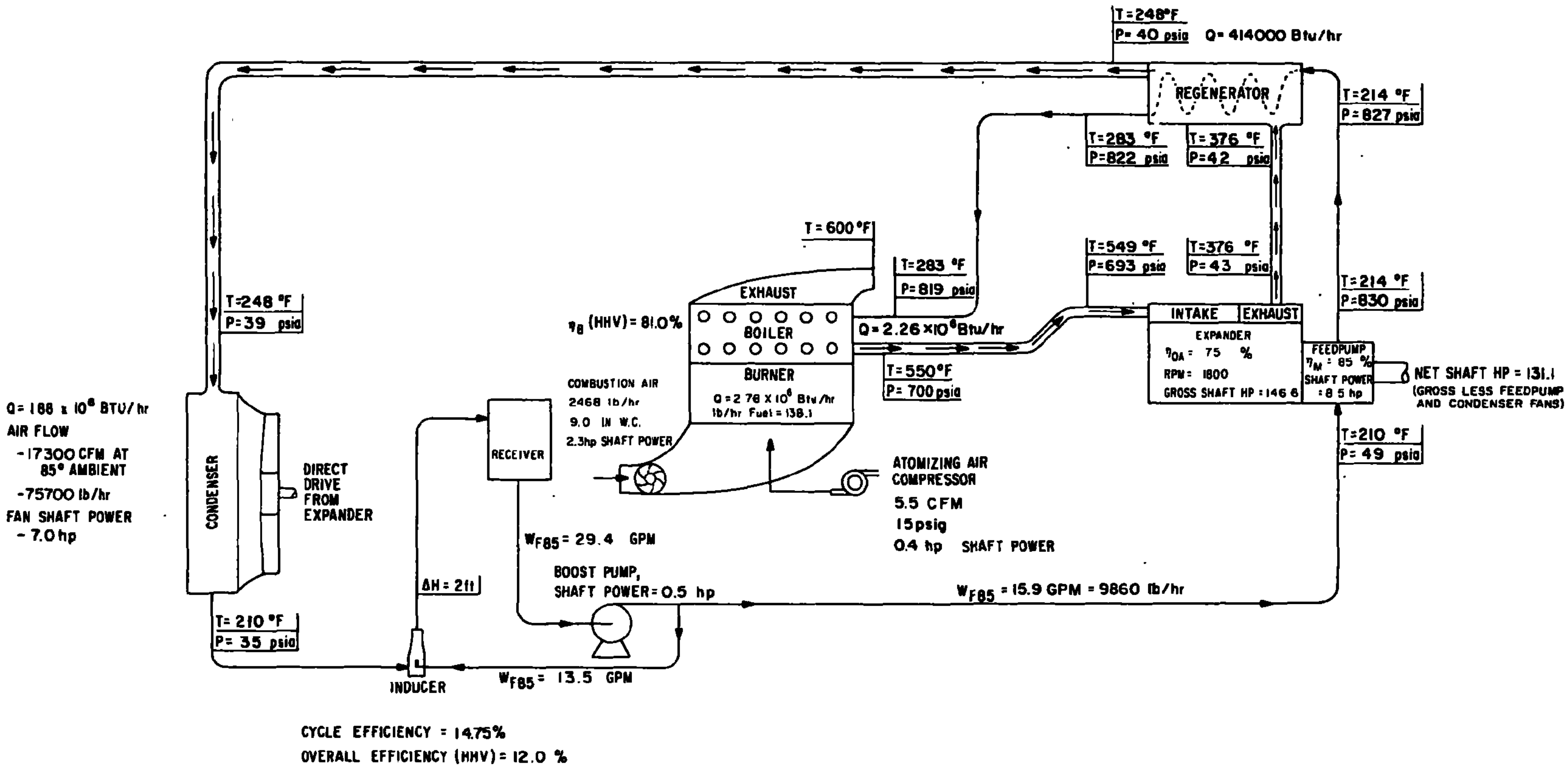
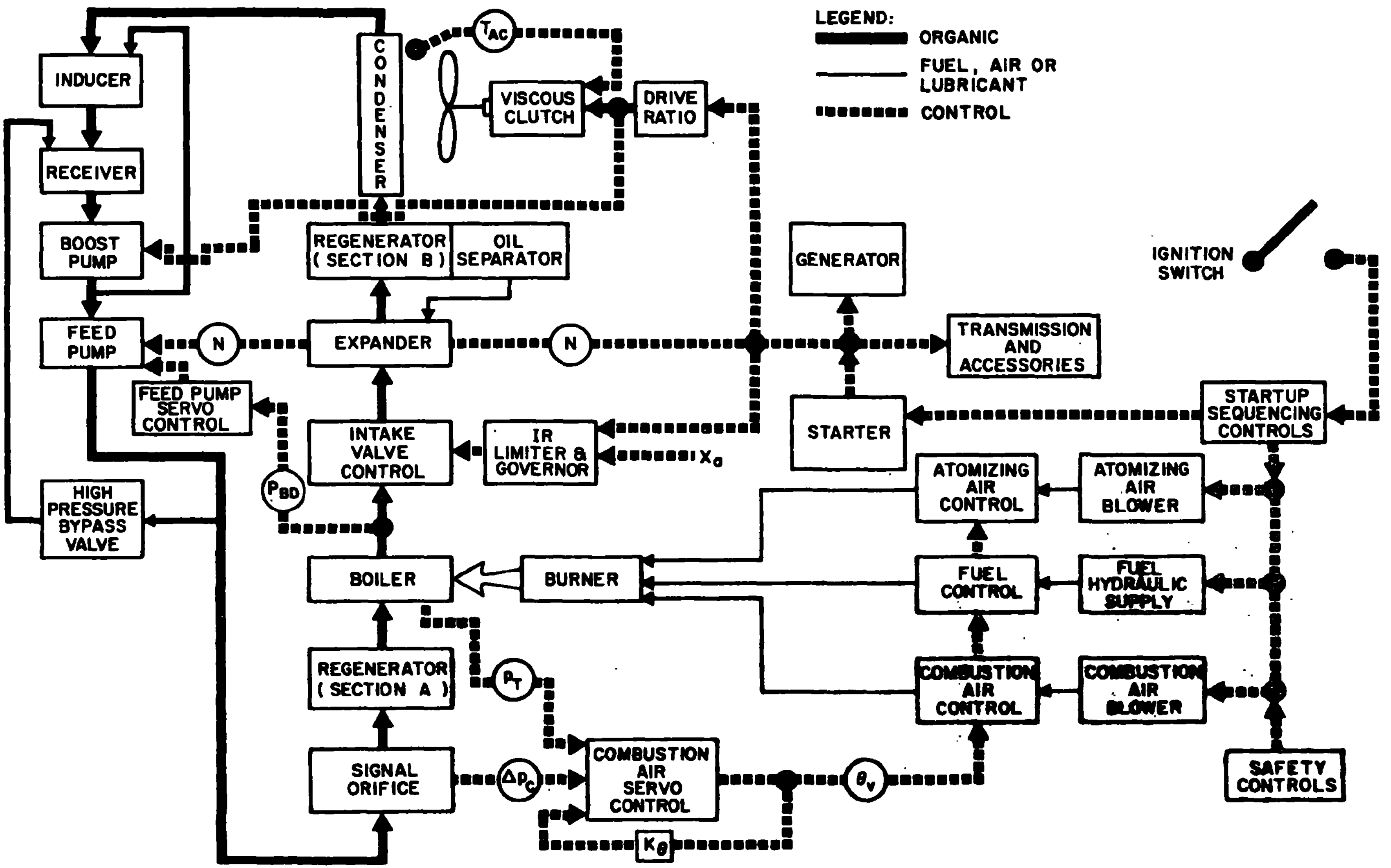


Figure 2.1 State Point Diagram.



<u>SYMBOLS:</u>

p_T - PRESSURE, INDICATING ORGANIC TEMPERATURE
p_{BD} - PRESSURE, BOILER DISCHARGE
Δp_C - PRESSURE DIFFERENTIAL, SIGNAL ORIFICE
X_a - DISPLACEMENT, ACCELERATOR
N - SPEED, EXPANDER CRANKSHAFT
θ_V - DISPLACEMENT, AIR CONTROL VALVE
K_θ - GAIN FUNCTION, FEEDBACK
T_{AC} - TEMPERATURE, CONDENSER DISCHARGE AIR

Figure 2.2 System Schematic.

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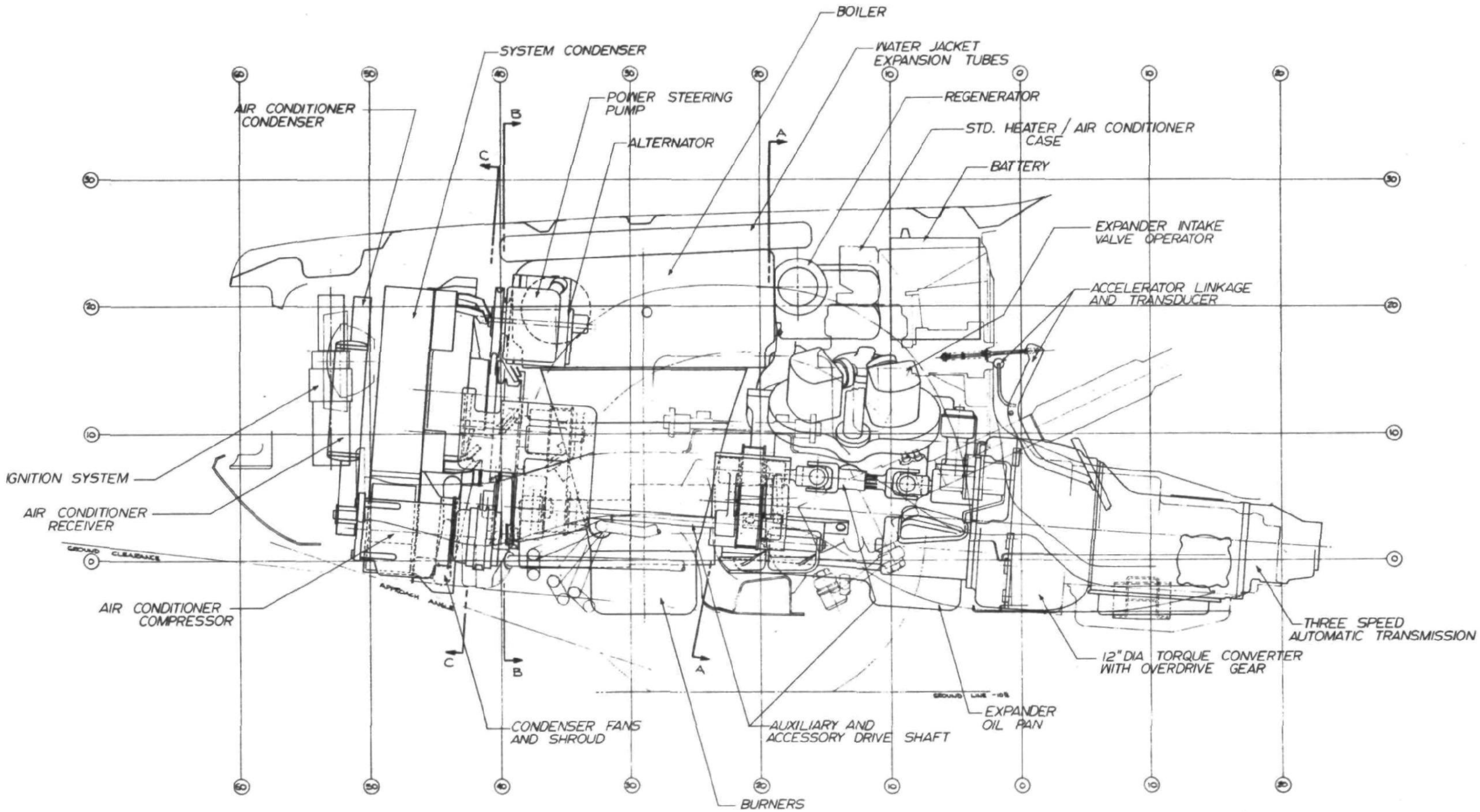

Figure 2.3 Side View, Packaged System.

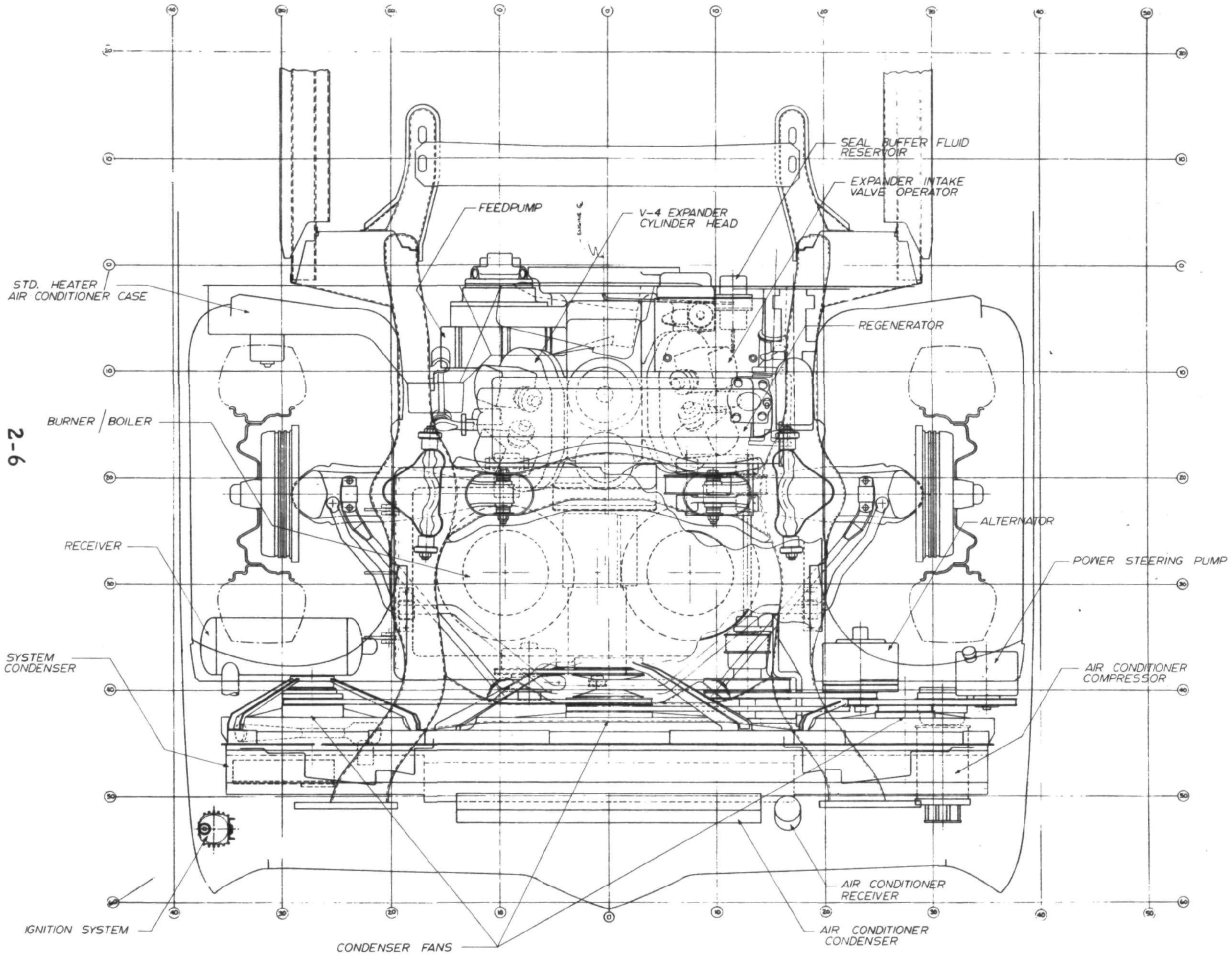

Figure 2.4 Top View, Packaged System.

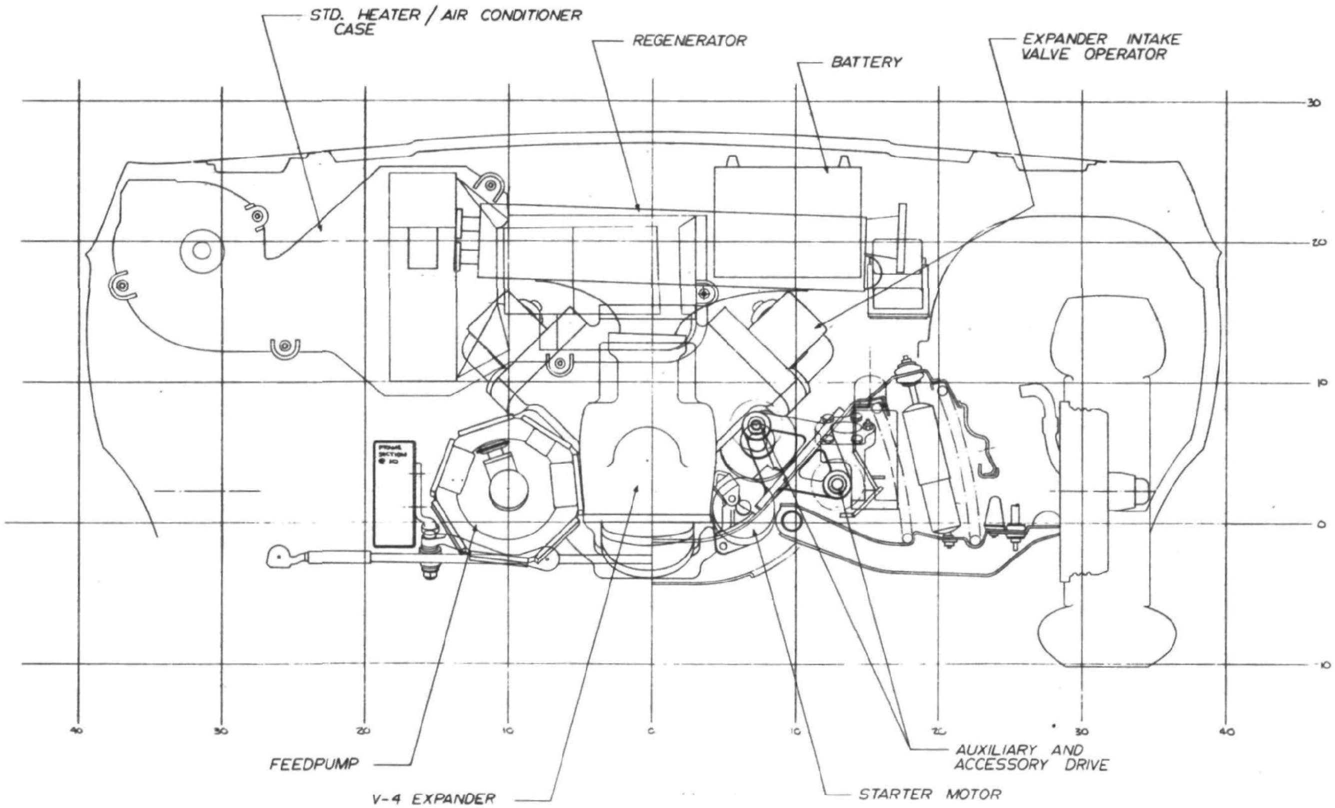

Figure 2.5 Section A-A from Figure 2.3.

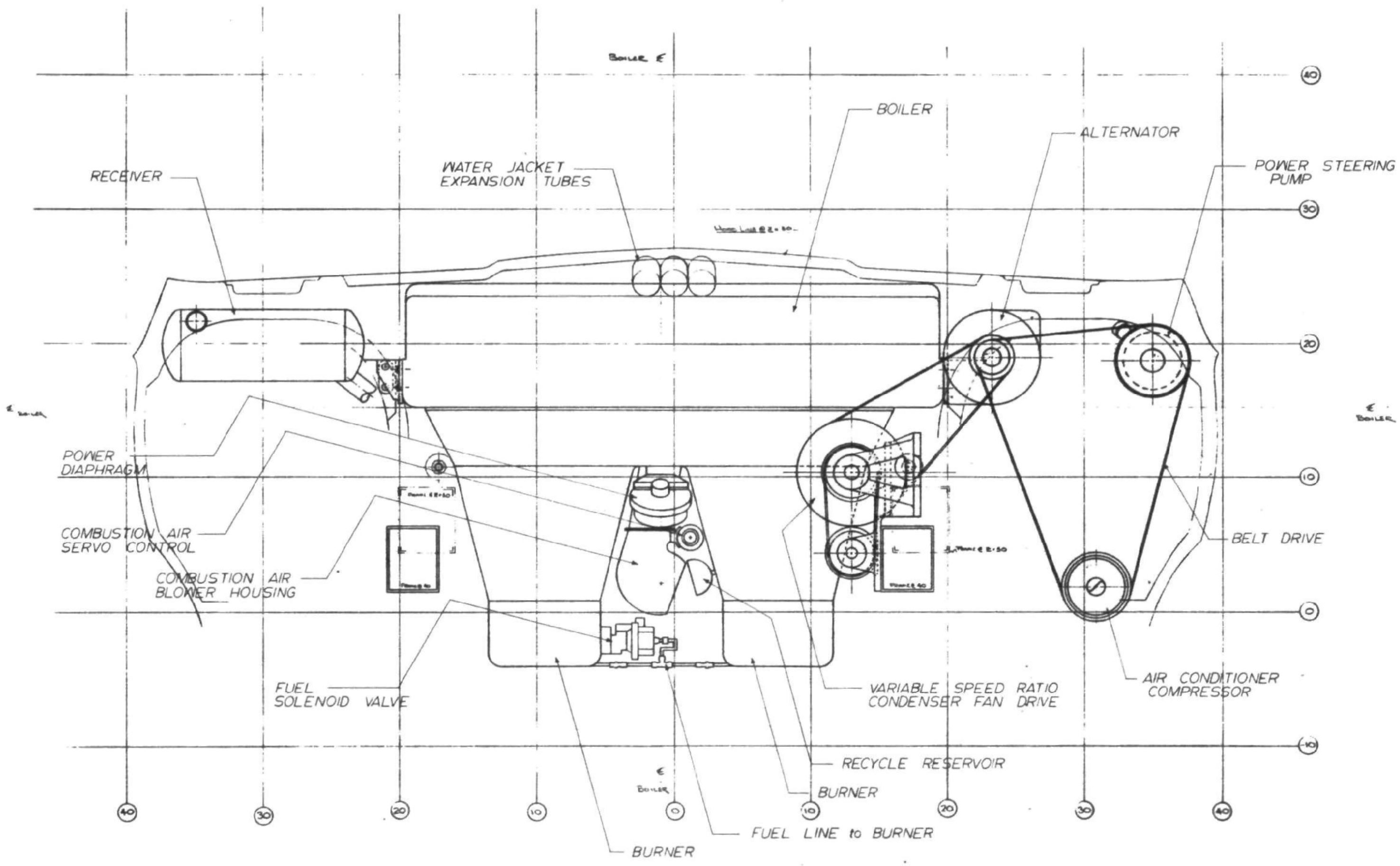

Figure 2. 6 Section B- B from Figure 2. 3.

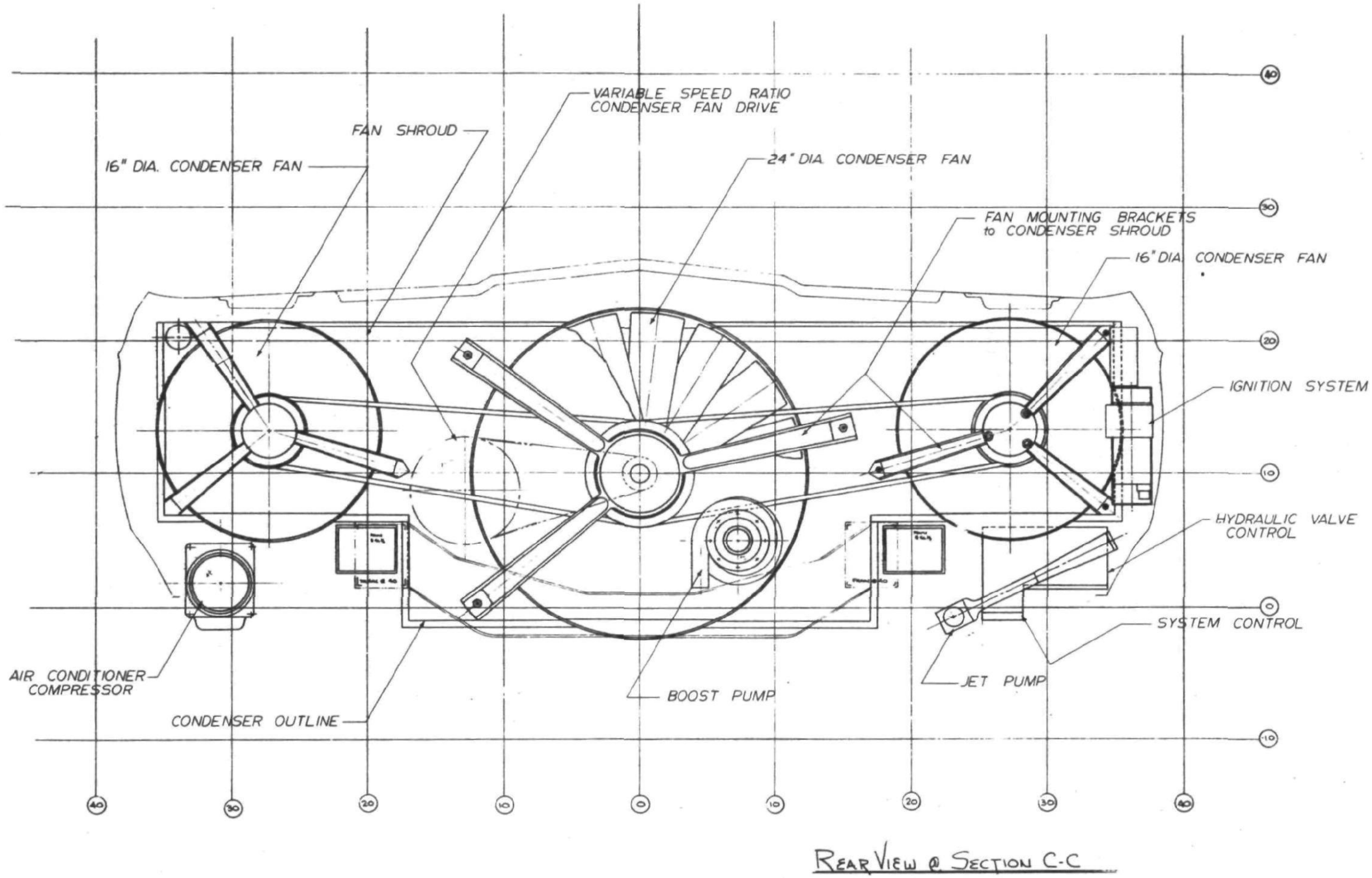

Figure 2.7 Section C-C from Figure 2.3.

sufficient flow area for exhaust of the condenser cooling air. The regenerator is located directly above the expander and is mounted to the expander. All auxiliary components for the Rankine-cycle system as well as accessory components for the normal automotive functions are also packaged in the system.

2.2.2 Acceleration Performance and Fuel Consumption Calculations

Computer models of the Rankine-cycle system and the vehicle have been used in calculating the acceleration performance and fuel consumption of the vehicle over typical driving conditions. In Figures 2.8 and 2.9, performance maps of the Rankine-cycle power system are presented in the form of net shaft horsepower vs. expander rpm with lines of constant efficiency shown. The maps are based on use of a special two-speed transmission designed by the Dana Corporation; the first map applies to "Lo" gear with a high expander rpm relative to vehicle speed, and the second to "Hi" gear with a low expander rpm relative to vehicle speed.

The acceleration performance and gradability of the vehicle are presented in Tables 2.1 and 2.2, respectively. These estimates are based on a vehicle curb weight, fully-fueled, of 4276 lbs, with 300 lbs passenger load for acceleration performance and 1000 lbs passenger load for gradability. The vehicle with the Rankine-cycle system, which weighs 210 lbs more than the same vehicle with the 1972 Ford 351 CID internal combustion engine, meets EPA specifications for acceleration and gradability.

The fuel consumption is presented in Table 2.3 both for steady speed on 0% grade and for three driving cycles: the Federal driving

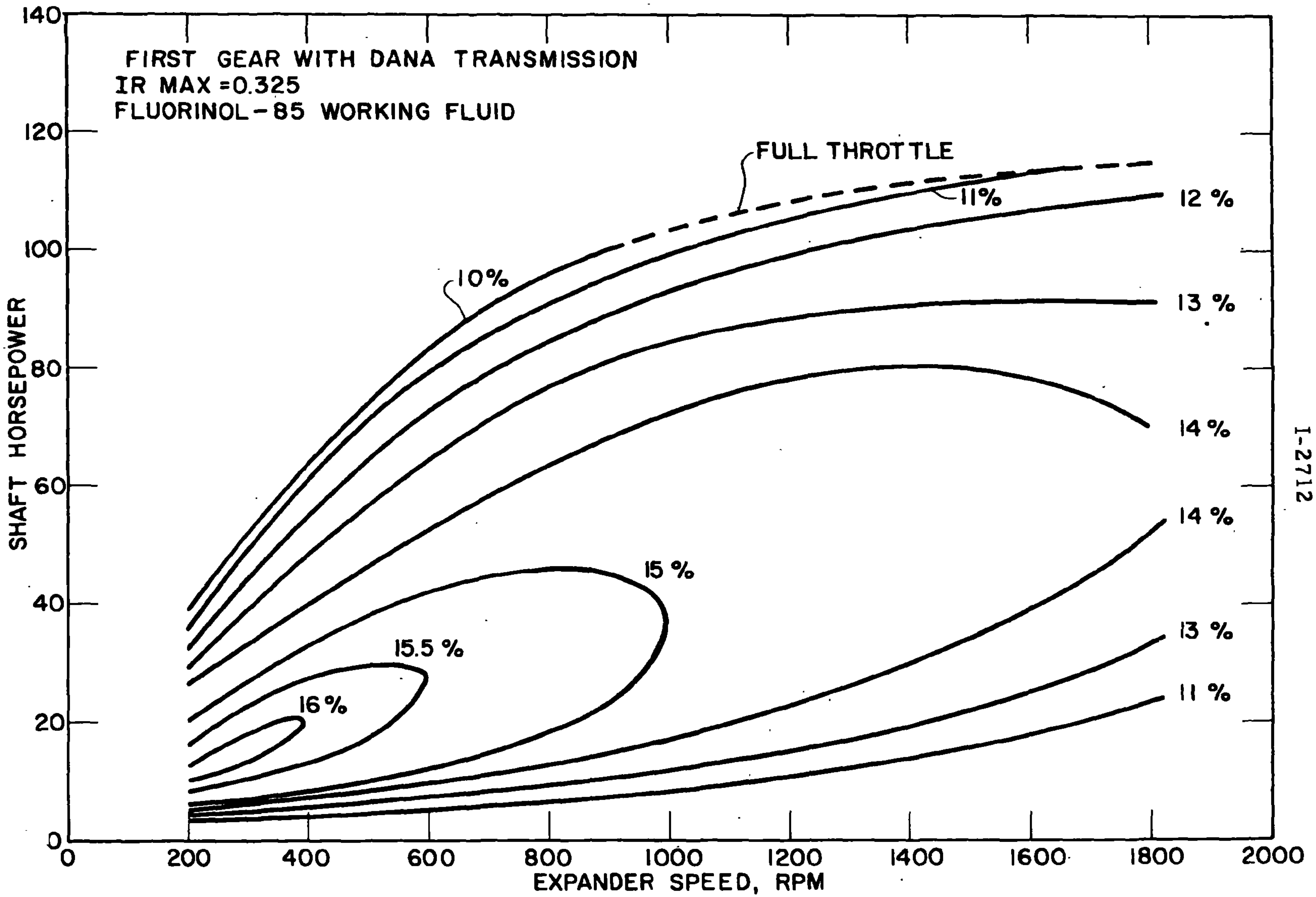

Figure 2.8 Performance Map with Transmission in First Gear (High Expander Rpm Relative to Vehicle Speed).

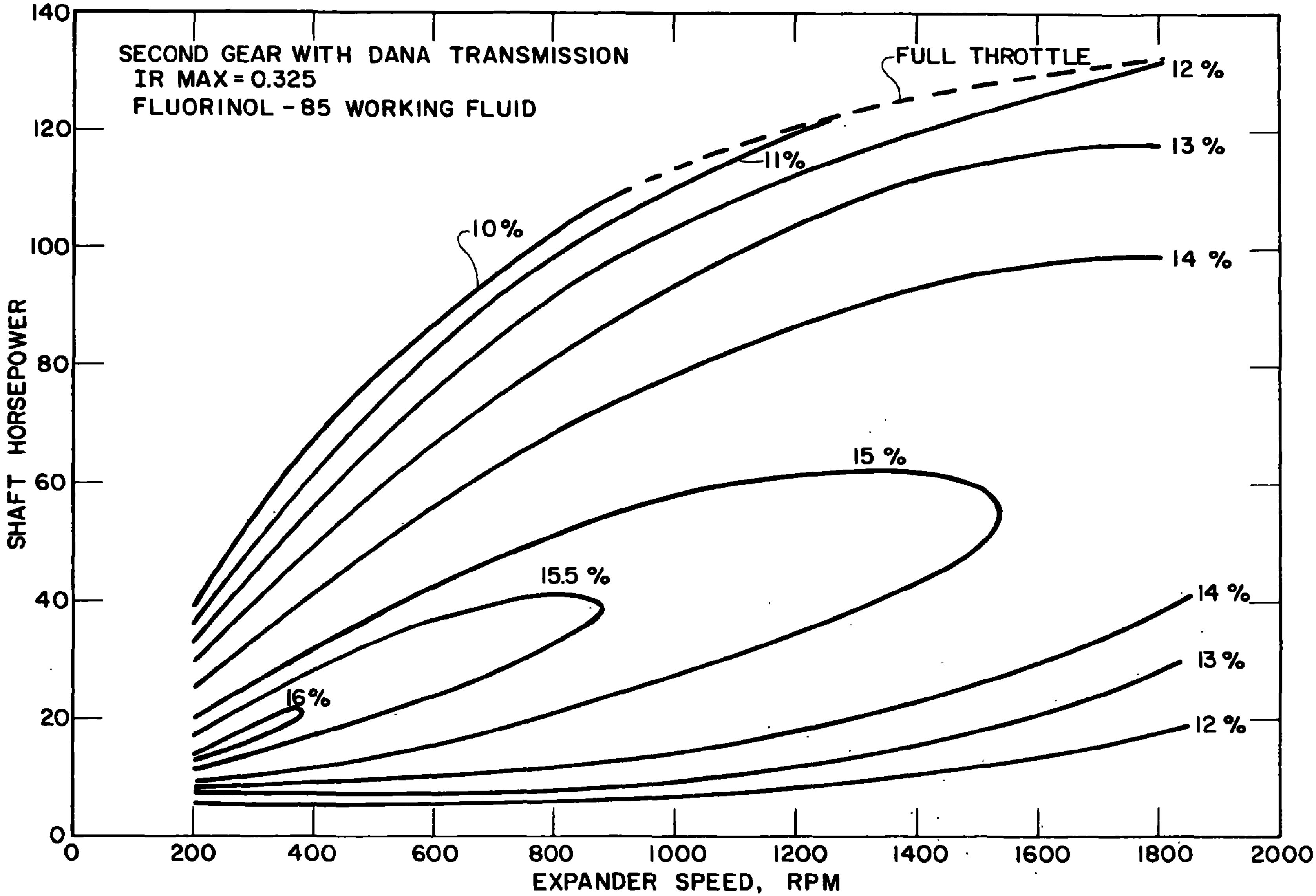

Figure 2.9 Performance Map with Transmission in Second Gear (Low Expander Rpm Relative to Vehicle Speed).

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TABLE 2.1

VEHICLE ACCELERATION PERFORMANCE

Vehicle Test Weight	4576 lbs	
Ambient Temperature	85°F	
Dana Transmission	Two Speed	
	System Performance	EPA Spec
0 - 60 mph	13.36 sec	≦ 13.5 sec
0 - 10 seconds	457.9 ft	≧ 440 ft
25 - 70 mph	15.0 sec	≦ 15.0 sec
Passing, 50 - 80 mph	15.4 sec	≦ 15.0 sec ≧ 1400 ft

TABLE 2.2

GRADABILITY

Vehicle Test Weight	5276 lbs	
Ambient Temperature	85 °F	
Dana Transmission	Two Speed	
	Grade %	
Vehicle Speed, mph	System Performance	EPA Spec.
0	35.3%	≧ 30%
10	34.6%	≧ 30%
15	30.7%	≧ 30%
20	26.8%	-
30	19.8%	-
40	13.8%	-
50	11.3%	-
60	8.97%	-
70	6.84%	Grade for 70 mph < 5%
103	0%	-

TABLE 2.3

FUEL CONSUMPTION

Vehicle Test Weight	4576 lbs
Ambient Temperature	85°F
Dana Transmission	Two Speed
Constant Speed 0% Grade	
MPH	MPG
30	15.38
40	15.15
50	13.52
60	11.76
70	10.33
80	8.54
85	7.84
Constant Speed, 5% Grade	
60	5.68
70	4.99
Driving Cycles	
Federal Driving Cycle for Emissions	10.81 mpg
FOMOCO Suburban Cycle	11.41 mpg
FOMOCO City Cycle	8.61 mpg
FOMOCO Customer Average	10.01 mpg

cycle for emission measurements, and the Ford Motor Company suburban and city driving cycles. The FOMOCO customer average is the arithmetic average of the FOMOCO suburban and city driving cycles.

2.2.3 Emission Levels Measured Over Federal Driving Cycle

To demonstrate the potential of the Rankine-cycle automotive propulsion system for very low emission levels, Thermo Electron Corporation has made emission measurements on a full-size burner for a 100 shp automotive system, operated transiently over burning rates, corresponding to operation of the vehicle over the Federal driving cycle for emission measurements. The fuel/air control used in these transient tests was similar to that to be used in the automotive system. The burner fired into a water-cooled "boiler" with approximately the same configuration as in the system. The procedure for making the emission measurements on the exhaust from the boiler was identical to that of the Federal Register and included use of the three-bag constant volume sampler.

The test results with a ceramic-lined burner are summarized in Table 2.4. The gram/mile emission levels are below the 1976 Federal standards by a factor of 1.4 for NO_x, 15.4 for CO, and 2.9 for UHC. Steady-state measurements indicate that use of exhaust gas recirculation (EGR) will result in even lower NO_x emission rates; thus, EGR will be used on the system.

2.3 COMPONENT DESCRIPTIONS

A description of the component designs and characteristics is presented in this section. These components are identical to those used in the system packaging.

TABLE 2.4

EMISSION LEVELS MEASURED OVER
FEDERAL DRIVING CYCLE

Emissions (grams/mile)	Transient Test Result*	Federal 1976 Standard
NO_x	0.29	0.4
CO	0.22	3.4
UHC	0 14	0.41

*Actual gas mileage used for tests was 12.1 mph. The latest performance calculation predicts 10.8 mpg for the Federal emission test driving cycle and the measured emission levels have been increased by 12% to reflect the change in fuel economy

2.3.1 Expander-Feedpump-Transmission Subassembly

2.3.1.1 Expander Design

The expander is a single-acting V-4 with 4.42 inch bore, 3.00 inch stroke, and total displacement of 184 in^3. The expander speed at a vehicle speed of 90 mph is 1800 rpm. Variable cutoff intake valving is used for power control from the expander to maximize: (1) wide-open-throttle acceleration performance and (2) the part-load efficiency of the system. Power is controlled completely by the expander variable cuttoff intake valving; no throttle valve is used between the boiler and expander. The automatic exhaust valve is similar to that used on Thermo Electron's 5-1/2 hp expanders and permits exhaust over most of the piston return stroke, thereby maximizing the power output per unit of expander displacement.

The front and side layouts of the expander are illustrated in Figures 2.10 and 2.11, respectively. With the exception of the valving, the expander construction is similar to that of internal combustion engines. The block and cylinder head are cast iron, the piston and connecting rod are cast aluminum, and the crankshaft is cast steel. Needle bearings are used throughout to reduce initial development difficulties. Spray lubrication is used. The lubricating oil is thermally stable and compatible with the working fluid at the peak cycle temperature of 550°F, so that oil blowing by the expander can be allowed to pass through the boiler with no deleterious effects on the system.

The variable cutoff intake valves for the preprototype expander, which are hydraulically actuated, are being developed by the American Bosch Corporation. The valve actuator was constructed and bench-testing during the program by American Bosch. With a hydraulic

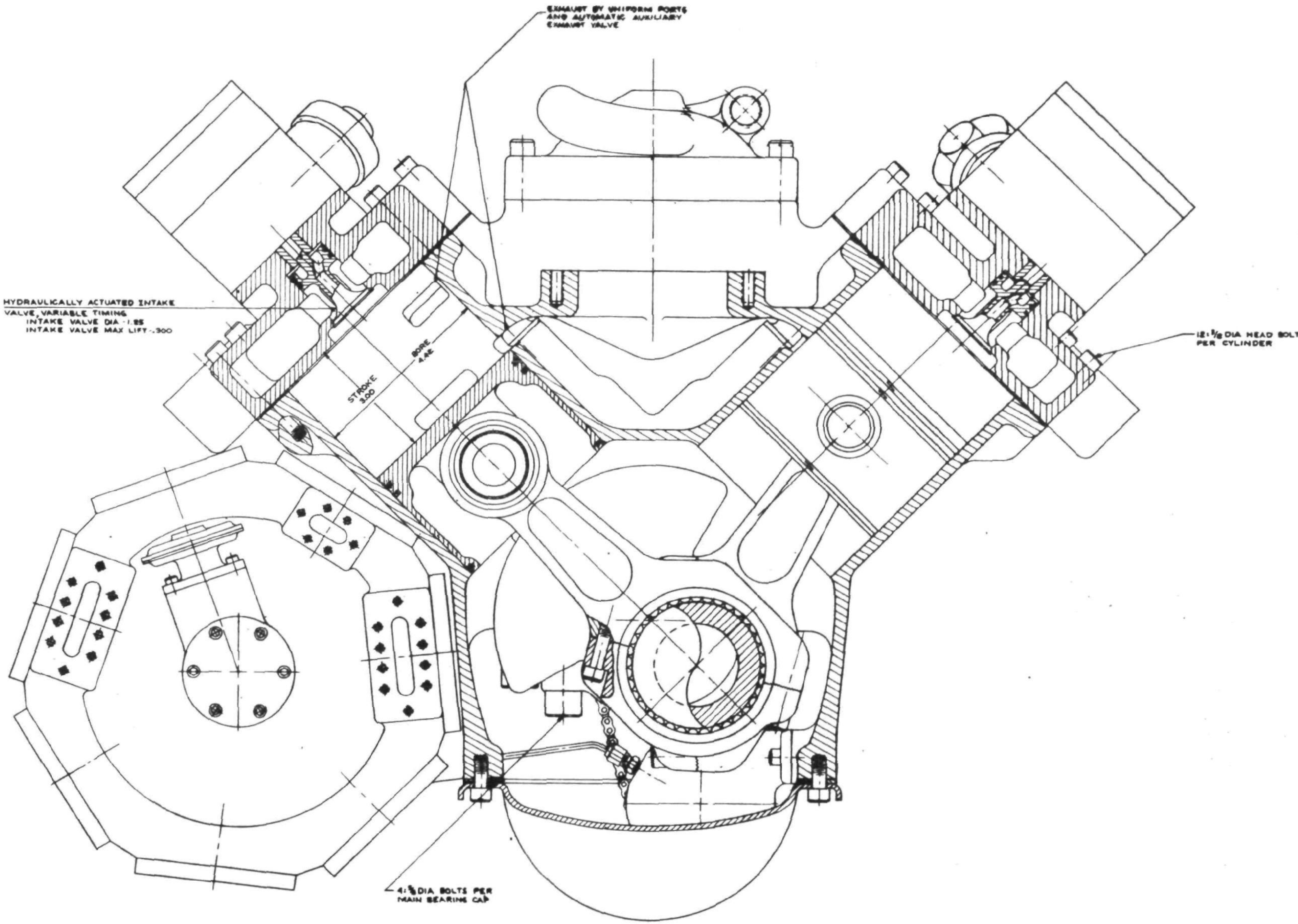

Figure 2.10 Expander Layout with American Bosch Valving - Cross Section Through Front Cylinders.

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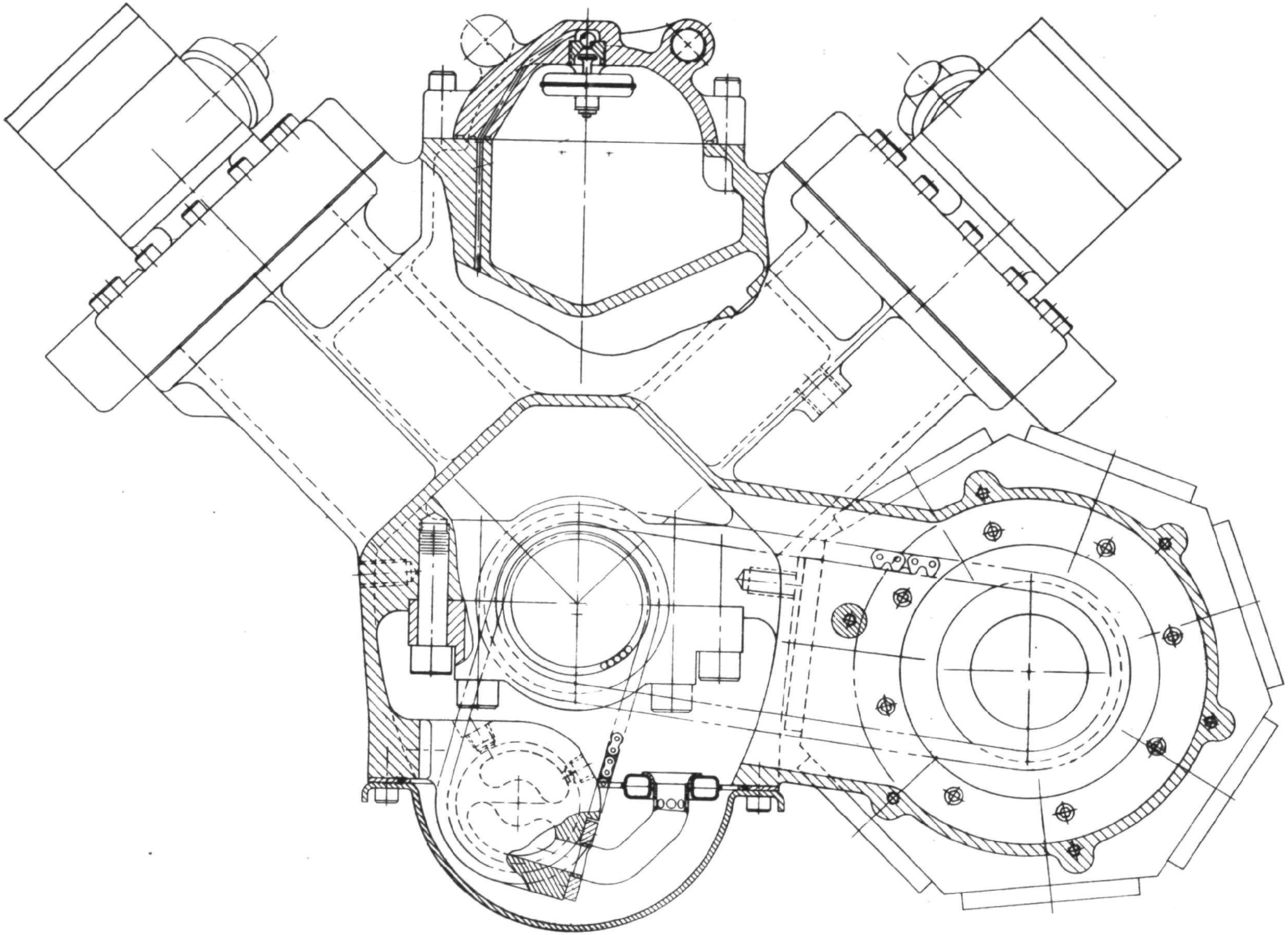

Figure 2.11 Expander Layout with American Bosch Valving - Cross Section Through Rear of Expander Showing Feedpump and Oil Pump Drive.

pressure of 1500 psi, the full-lift opening and closing times are 2.0 and 3.0 milliseconds, respectively. The intake valves are 1.25 inches in diameter with 0.3 inch lift. In addition to this system, a detailed design of a mechanically (cam-driven) actuated valve system was developed and is presented in the report. A second hydraulic system, financed by Thermo Electron Corporation, is under experimental development.

The lubricating oil for the expander is used as the hydraulic fluid for the valving system. The hydraulic pump and a separate oil reservoir for the valve system are integrated in the expander design.

2.3.1.2 Feedpump

The main system feedpump is a radial, 7 cylinder, reciprocating piston pump. The pump is directly driven by the expander and is integrated with the expander, as illustrated in Figure 2.10. Since, at any expander speed, the required system pumping rate can vary from zero to a maximum of about 16 gpm, depending on the intake valving cutoff point (or system power output), the pump is variable displacement to minimize the feedpump power requirement at part-load conditions. An additional development goal was maintaining high pump efficiency over a wide speed and pumping rate range.

The feedpump design is illustrated in the cross section of Figure 2.12. The seven cylinders are located radially around the axis of the rotating shaft. Each cylinder houses seal-ring pistons which can be varied from zero to full stroke by axially moving the angled portion of the shaft through the center eccentric ring. The cylinders receive fluid from a common inlet plenum and discharge into a common outlet plenum. Both the inlet and outlet valves are simple spring-loaded washer types. The piston shoes ride on a septagonal ring, with a circular ring used to return the pistons. The cylinder bore is 1.50 inches, stroke is 0.466 inch, and maximum displacement is 5.76 inches. The pump delivers 15 gpm with full stroke at a speed of 600 rpm. The design discharge pressure is 850 psia. The pump has an aluminum housing with steel pistons and steel drive mechanism.

The pump has been constructed and tested; some of the test results are presented in Figures 2.13 and 2.14. The volumetric efficiency varies from ~98% at low speeds to 71% at the maximum speed of 1800 rpm. The overall pump efficiency (hydraulic power/shaft power) varies from about 82% at low speeds to 68% at high speeds. The efficiency at a given speed is insensitive to discharge pressure and relatively insensitive to the pump displacement. Pressure pulsations are very small and acceptable over the entire range of operating speed and displacement tested.

Boiler outlet pressure is controlled by varying the pumping rate to the boiler through control of the pump displacement. The feedpump design of Figure 2.12 includes a spool control valve operated by a spring-biased diaphragm to which boiler outlet pressure is directly applied. This spool valve controls application of the feedpump discharge fluid to a power piston which is connected to the stroke control shaft of the pump and is used to vary the displacement.

2.3.1.3 Transmission

The system uses a conventional, three-speed automatic transmission with 12 inch diameter torque converter coupling (FOMOCO

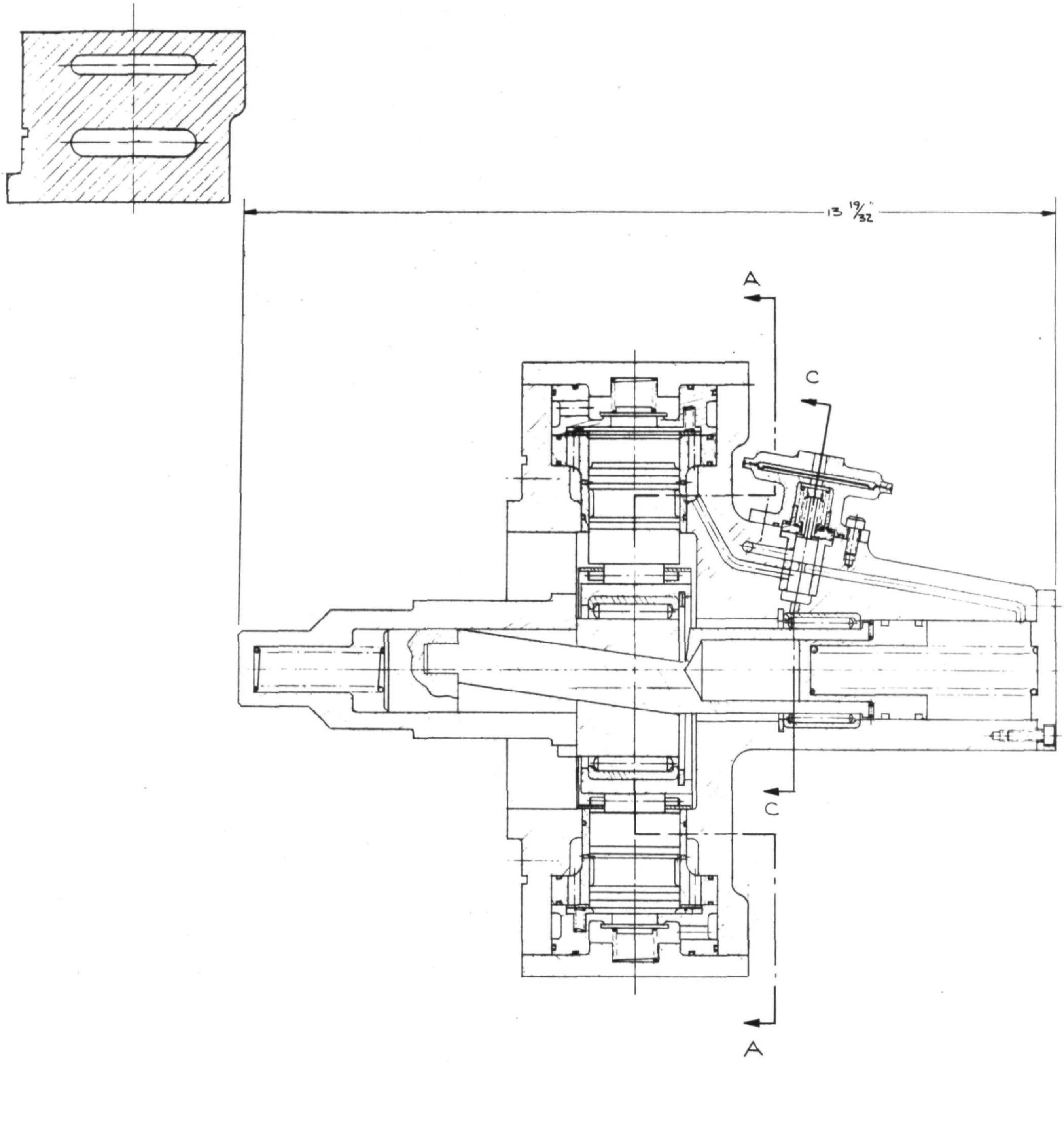

Figure 2.12 Reciprocating Piston Pump with Variable Displacement - Cross Section.

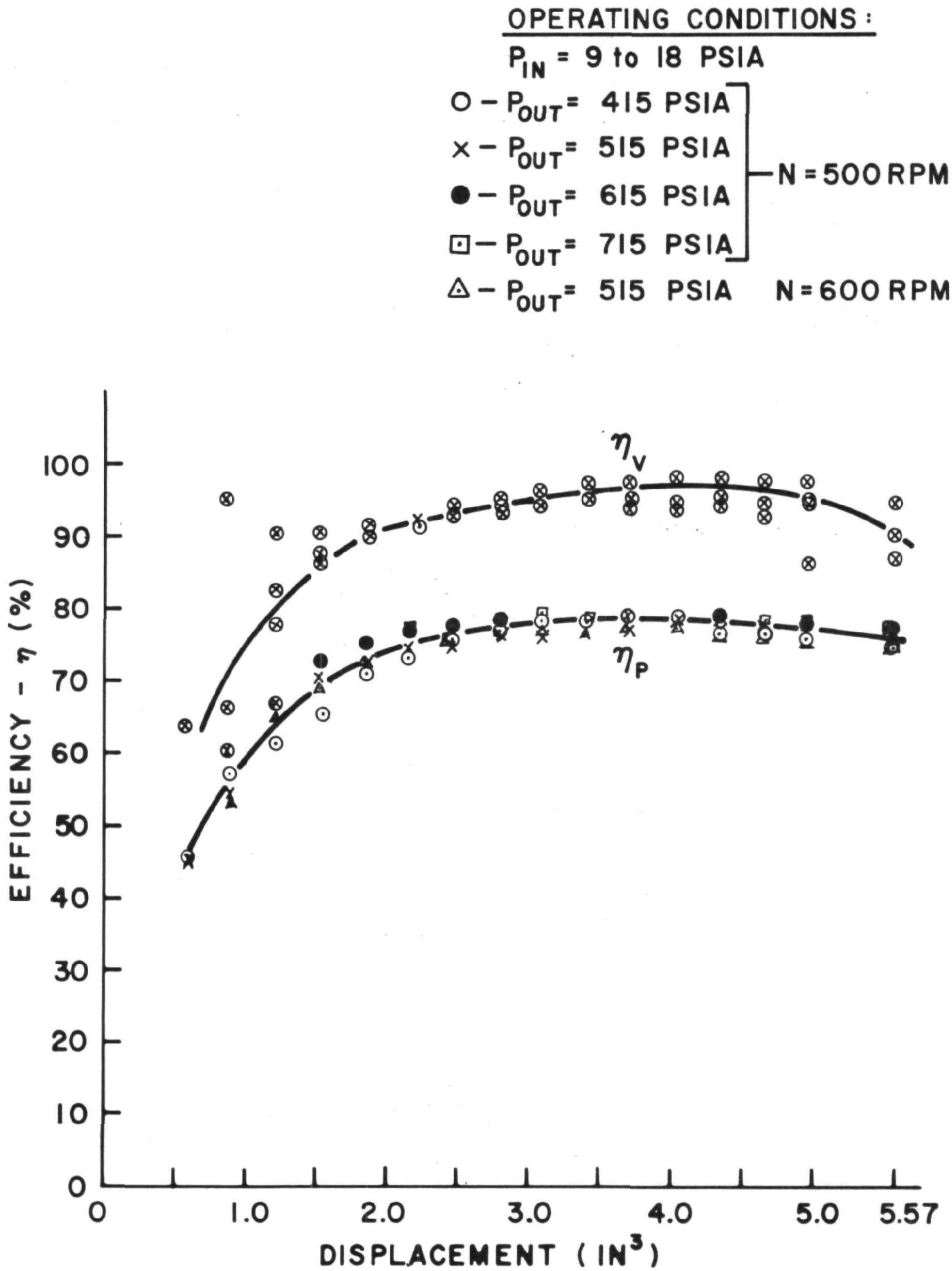

Figure 2.13 Efficiency vs. Displacement for 7-Cylinder Feedpump.

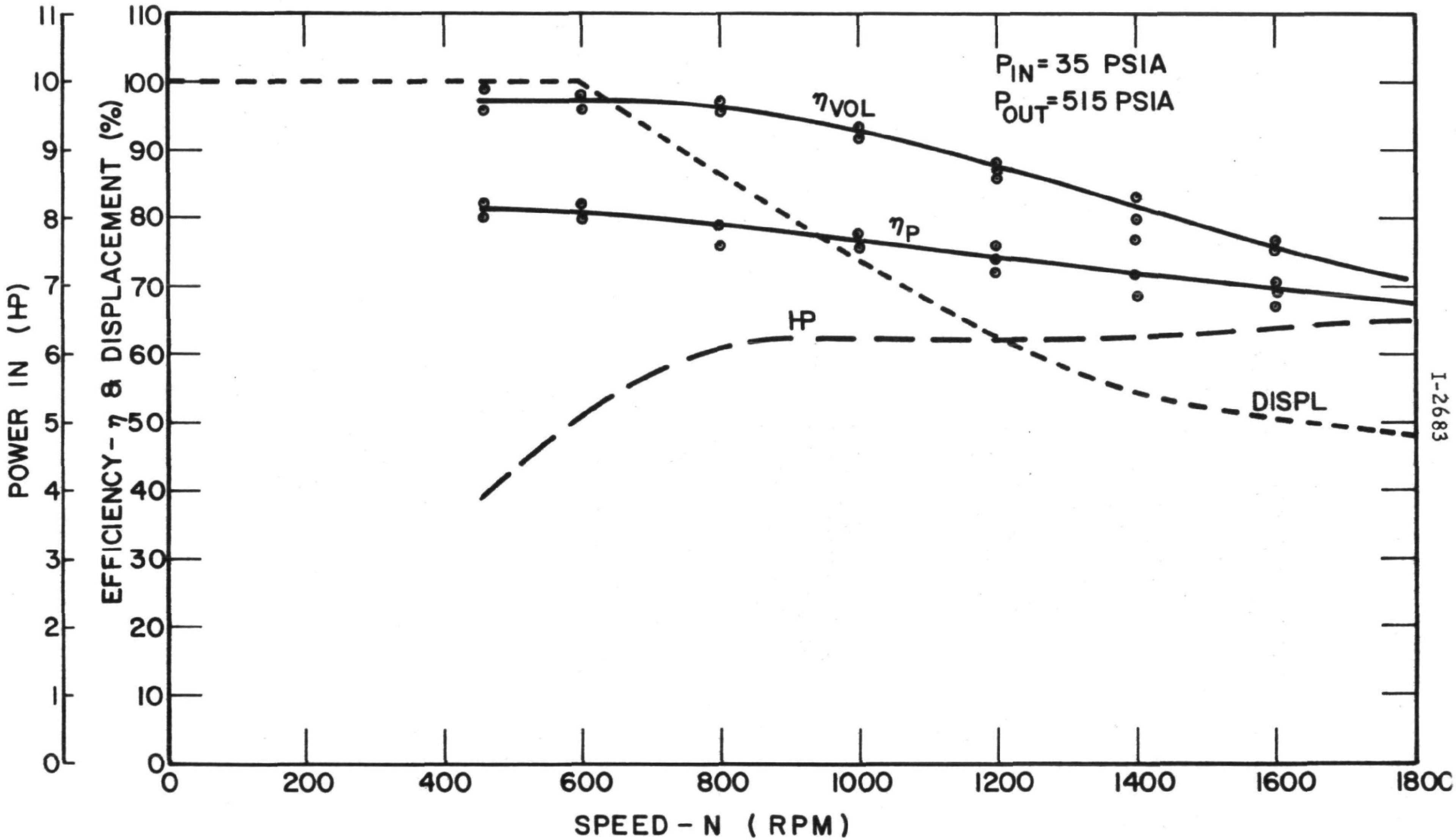

Figure 2.14 Efficiency, Displacement, and Shaft Power Input vs. Speed for 7-Cylinder Feedpump.

2-25

I-2683

C-4 automatic transmission). A planetary speedup gear with 1:2.25 speed ratio is used between the expander and transmission. Gear ratios in the transmission are 2.40 low, 1.47 intermediate, and 1.00 high. The driveline is standard for the 1972 Ford Galaxie with 2.75:1 axle ratio.

2.3.1.4 Rotary Shaft Seal

The system has been designed so that only one dynamic shaft seal, that on the 3 inch diameter expander shaft, is required. To positively prohibit either air leakage into the system or working fluid leakage out of the system, a double shaft seal is used with prepressurized buffer fluid between the two seals. The system lubricating oil is used as the buffer fluid so that any leakage into the system through the seal is the system lubricant. The two seals are of the face-seal type.

Bench-testing of two designs of the full-size seal has been carried out under simulated system conditions with excellent results. Over a test period of 3187 hours in one test, the average leakage rates of the lubricating oil were 0.183 pints/1000 hrs operation into the crankcase, and 0.255 pints/1000 hrs operation through the outboard seal. Leak rates in the shutdown mode are approximately a factor of 10 lower than the operating leakage rates.

2.3.2 Combustion System-Boiler Subassembly

The assembly cross sections of the combustion system-boiler subassembly are presented in Figures 2.15 and 2.16. These drawings include the boiler, burners, combustion air blower and motor drive, atomizing air compressor, and fuel/air controls.

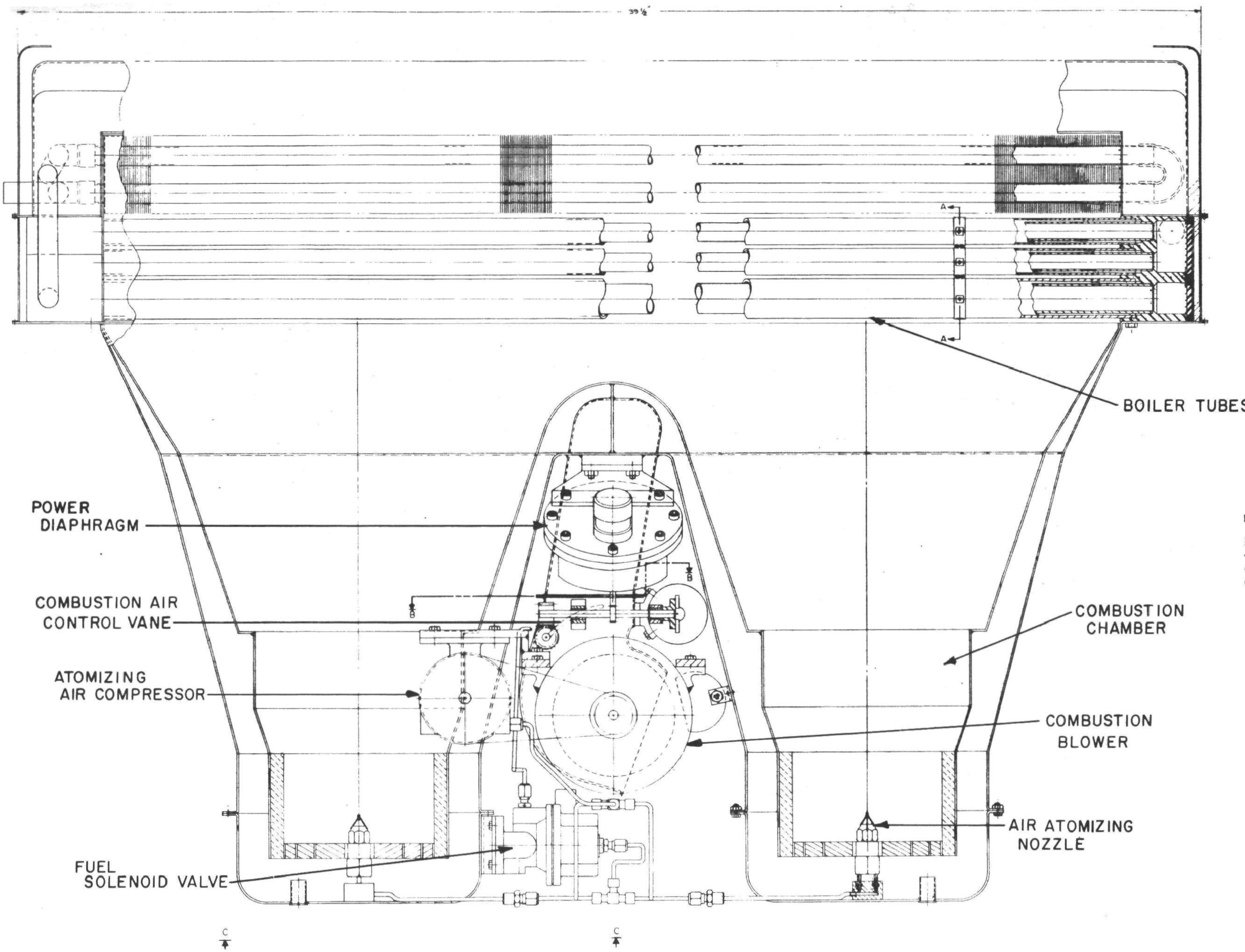

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Figure 2.15 Front View of Combustion System-Boiler Subassembly.

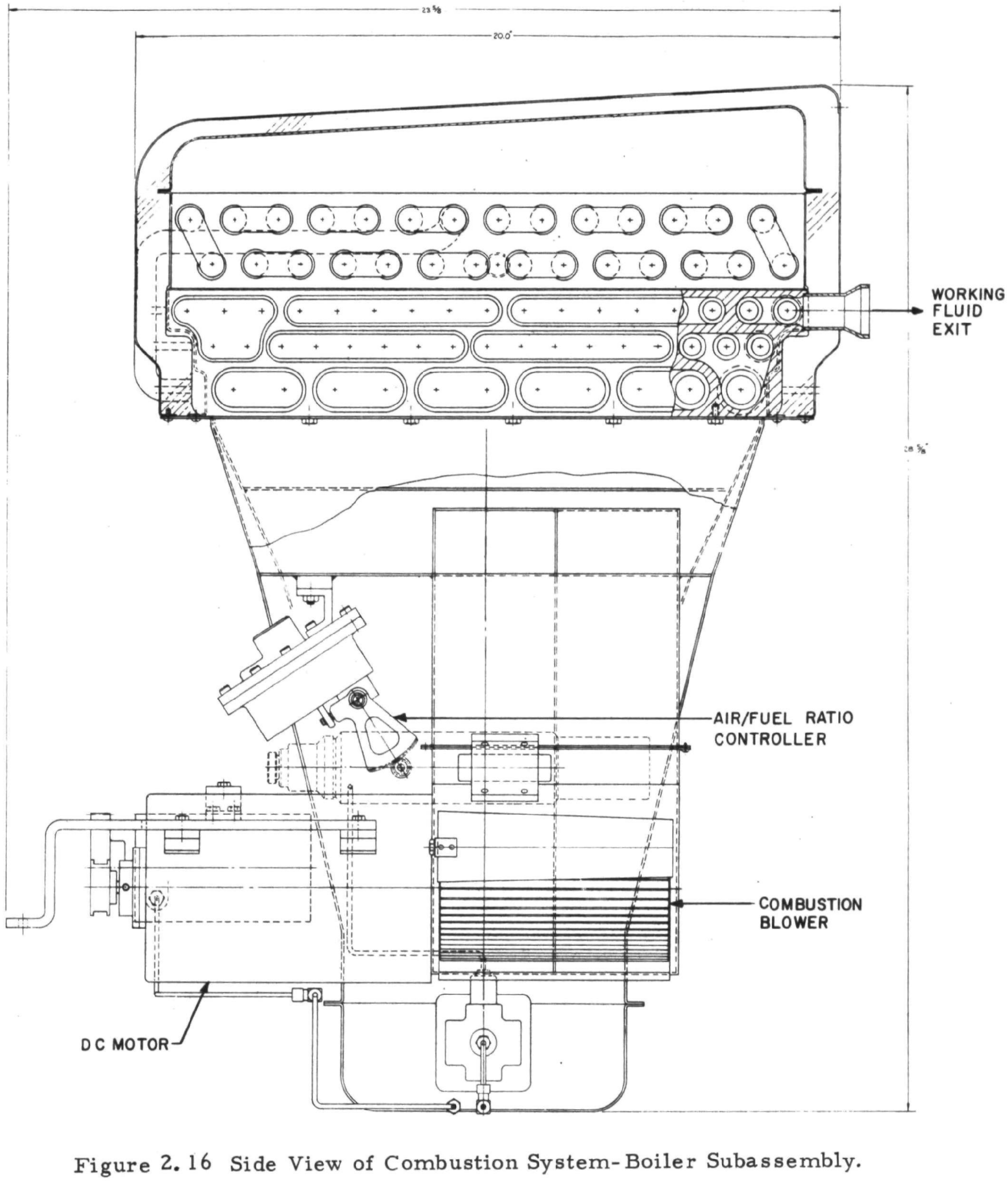

Figure 2.16 Side View of Combustion System-Boiler Subassembly.

2.3.2.1 Boiler Design

The boiler tube bundle is designed for a maximum heat transfer rate of 2.25 x 10^6 Btu/hr with 81% efficiency, based on the fuel higher heating valve (HHV). At 10% load, the boiler efficiency rises to 88% (HHV). The flow path of working fluid and combustion gases through the boiler is illustrated in Figure 2.17. At the design point, combustion gases enter the bottom of the boiler at 2975°F and exit from the top at 600°F. The working fluid enters the preheat state as liquid at 287°F and 826.5 psia and exits from the superheat stage as superheated vapor at 550°F and 700 psia. The maximum tube wall temperature on the working fluid side is 569°F. The design point characteristics of the boiler stages are summarized in Table 2.5.

The boiling and superheat stages of the boiler are bare tube bundles with a water jacket buffer to positively prevent either gross or local overheating of the working fluid. Dual tube construction is used for these stages, as illustrated in Figure 2.15, with organic flowing through the inner tube. The annular space (~60 mil gap) between the tube bundles is sealed and filled with water, with an external thermal expansion tank to permit thermal expansion of the water. Heat transfer from the combustion gases to the organic occurs by boiling the water on the inner surface of the outer tube and condensing the water vapor on the outside of the inner tube. The organic tube wall temperature can therefore not exceed the saturation steam temperature corresponding to the water jacket pressure; this pressure provides a convenient and sensitive means of controlling the maximum temperature to which the working fluid is exposed. The boiling and superheat stages are brazed construction with the tubes brazed into machined steel headers.

The preheat stage is a conventional, finned-tube heat exchanger. A water jacket is not used for this stage, since the combustion gas temperature is relatively low and the tubes are filled with liquid Fluorinol-85.

The fins, tubes, and headers for all stages are constructed of AISI 4130 steel.

2.3.2.2 Burner and Fuel/Air Supply Designs

Two burners firing in parallel are used to provide a burning rate (HHV) of 2.78×10^6 Btu/hr with JP-4 fuel. The design turndown ratio is 20:1. With reference to Figure 2.15, the cylindrical combustion chamber is air-cooled with combustion air entering at the top of the combustion chamber and flowing down the space between the combustion chamber and outer wall of the burner. At the bottom of the burner, the air flow reverses direction and flows through swirl vanes into the combustion chamber. The fuel nozzle is an air-atomizing Sonicore nozzle. Near the nozzle, the combustion chamber is lined with a ceramic insert. The combustion chamber wall is flared outward above the main combustion zone to diffuse the combustion gases and provide a uniform gas velocity over the entire area of the rectangular boiler tube bundle. To insure proper balance of fuel/air flow between the two burners, the air and fuel flow paths from the common fuel control and common air control are symmetrical.

The combustion air blower is a cross-flow or transverse type. The air blower provides a pressure head of 9 in. W. C. at the design air flow rate of 770 CFM and a mean mix temperature of 165°F. The design is based on 20% excess air (2468 lbs/hr at 60°F) and 20% exhaust gas recirculation (521 lbs/hr at 600°F). A gerotor-type fuel

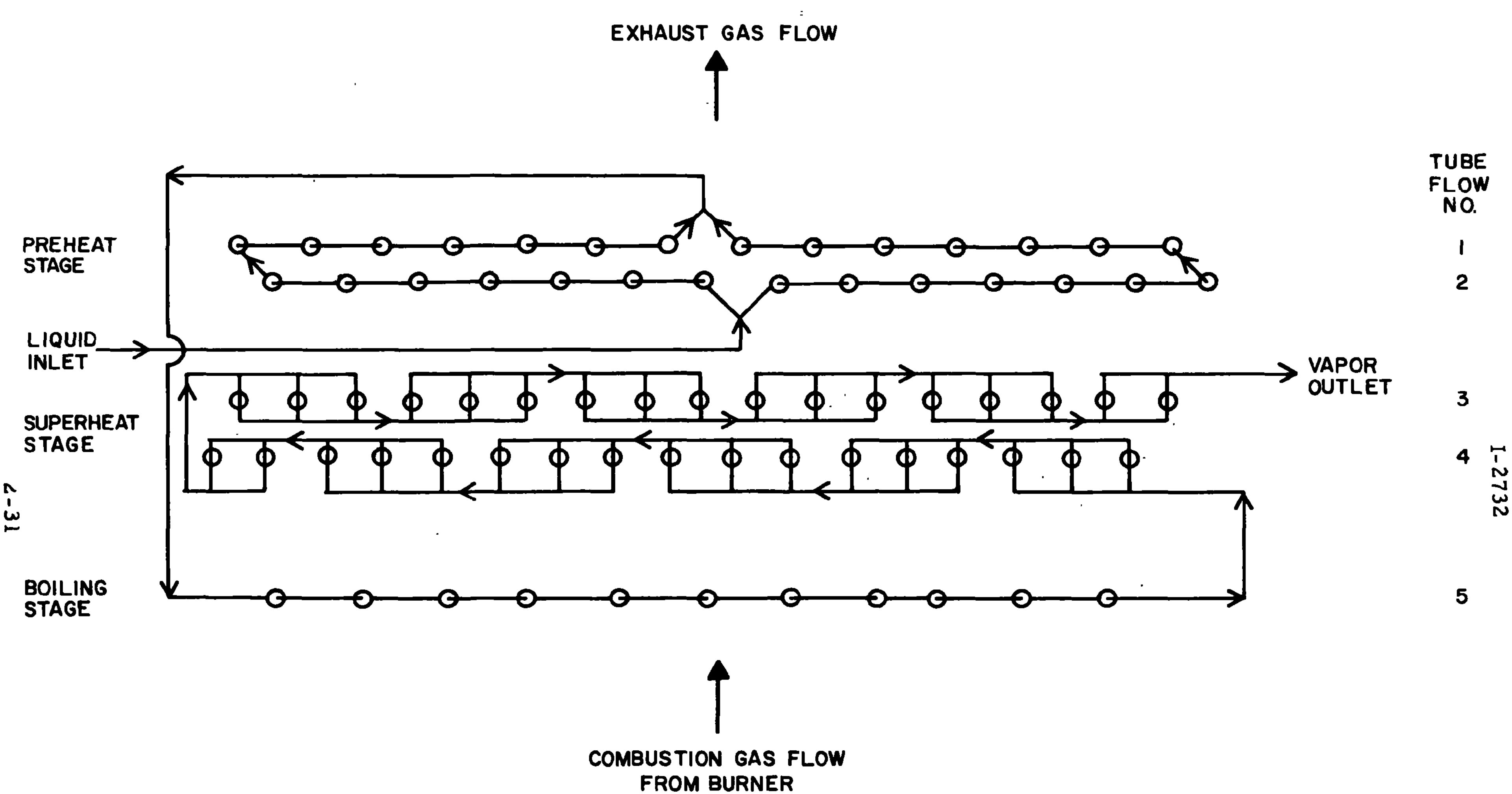

Figure 2.17 Flow Path Schematic of Working Fluid Through Boiler.

TABLE 2.5

BOILER DESIGN POINT CHARACTERISTICS

Stage	Tube Row No.	Heat Transfer Rate Btu/hr	Combustion Gas Temperature		FL-85 Temp.		Pressure Drop		Water Jacket Pressure psia	Mass of Core Without Water lbs.	Mass of Water lbs
			Inlet °F	Outlet °F	In °F	Out °F	Gas Side in w.c.	Organic Side psi			
Boiling	5	625,000	2975	2371	437	445	2.38	40	1470	66	2.7
Superheat-I	4	444,800	2371	1939	445	502	0.74	30	1405	40	2.7
Superheat-II	3	224,400	1939	1709	502	550	0.11	32.5	1431	40	2.7
Preheat	1 and 2	958,000	1709	607	287	437	0.17	24	-	80	-
Total		2.25×10^6					3.40	126.5		226	8.1

TUBE SPECIFICATIONS

<u>Boiling Stage</u>

Inner Tube - 1.00" O.D., 0.083" wall
Outer Tube - 1.313" O.D., 0.093" wall

<u>Superheater I and II Stages</u>

Inner Tube - 5/8" O.D., 0.049" wall
Outer Tube - 7/8" O.D., 0.058" wall

<u>Preheat Stage</u>

5/8" tube expanded (0.577" O.D., 0.035" wall)
18 fins/inlet (rippled)

<u>Tube and Header Material</u> = AISI 4130 Steel

pump is used to provide constant pressure (25 psig) fuel to the burner fuel metering valve. The atomizing air compressor is of the rotary-vane type.

The functional schematic of the burner controls is presented in Figure 2.2 and the location of the control components on the burner-boiler are illustrated in Figures 2.15 and 2.16. Detailed designs of the control elements are presented in Chapter 5 of this report. The burner controls are designed to provide rapid response to large power transients of the system, regulating the burning rate to a level corresponding to the new power level in a time of about 200 milli-seconds. A ΔP created across an orifice in the inlet line to the boiler is applied to the air servo control; this control is diaphragm-actuated, thus providing rapid response to organic flow rate changes to the boiler and an approximate balance between the organic flow rate entering the boiler and the burning rate. For fine tuning of the burning rate, a thermal-expansion temperature sensor responding to the vapor temperature is provided. This sensor generates a pressure signal, proportional to the organic temperature, which is also applied to a diaphragm in the combustion air servo control. The combustion air servo control modulates the air flow to the burner in response to these input signals. For low emissions, reasonably tight control on the fuel/air ratio must be provided. The fuel control is thus directly linked to the combustion air control.

2.3.3 Regenerator

The regenerative heat transfer rate at the design point is 414,000 Btu/hr, with vapor entering and leaving at 375°F and 239°F, respectively, and liquid entering and leaving at 208°F and 287°F. The regenerator design is illustrated in Figure 2.18. The core is brazed aluminum construction with flat tubes carrying the liquid and with fins on both the liquid and vapor sides of the core. The liquid and vapor flow paths are illustrated in Figure 2.19.

The regenerator is also used as an oil separator to collect and return to the crankcase a major portion of the oil droplets in the exhaust vapor from the expander. Provision is made for gravity return to the expander crankcase of the separated oil.

2.3.4 Condenser Subassembly

In the design of the condensing subassembly, a prime goal has been minimizing the parasitic fan power. To meet this goal, (1) the maximum condenser frontal area which can be packaged in the 1972 Ford Galaxie without major modifications has been used; (2) the condenser fan drive includes a control to optimize the fan speed under part-load and low vehicle speed operating conditions, and (3) an inducer is used to maintain the condenser free of condensed liquid so that the entire condenser core is effective for condensation.

The condenser design is illustrated in Figure 2.20. The condenser is T-shaped, as illustrated, in order to provide the maximum condenser frontal area without modifying the frame at the front of the car. The central condenser section sits between the two frame members and the two side condenser sections sit on top of the frame members (see Figure 2.7). Total core frontal area is 8.21 ft^2 and the core thickness is 4.3 inches. At the design point rate of 1.88×10^6 Btu/hr, the air flow rate is 17,300 CFM at 85°F, the air side pressure is 3.5 in. W.C., the ideal air power with the fans on the downstream side is 11.2 hp,

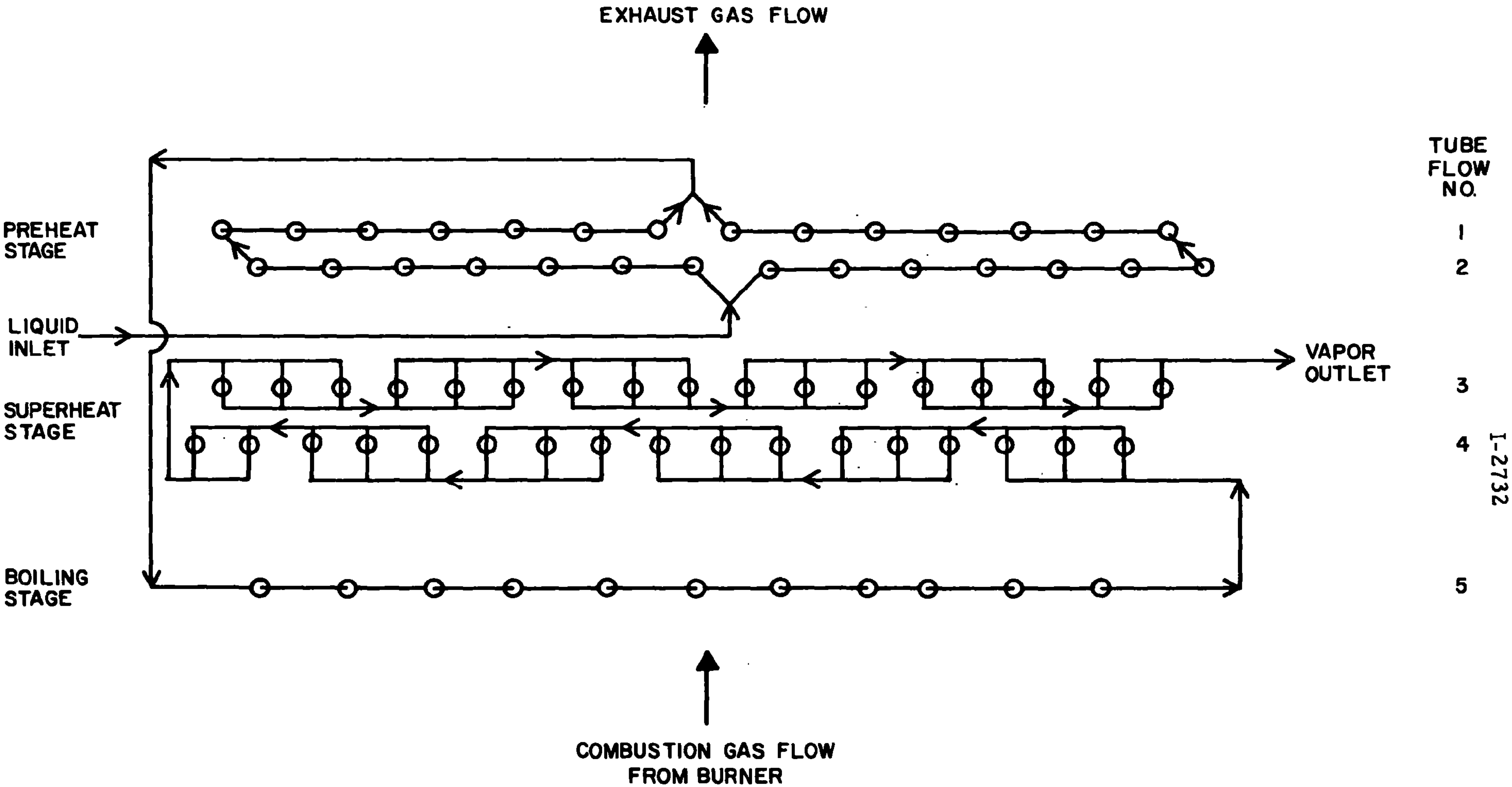

Figure 2.17 Flow Path Schematic of Working Fluid Through Boiler.

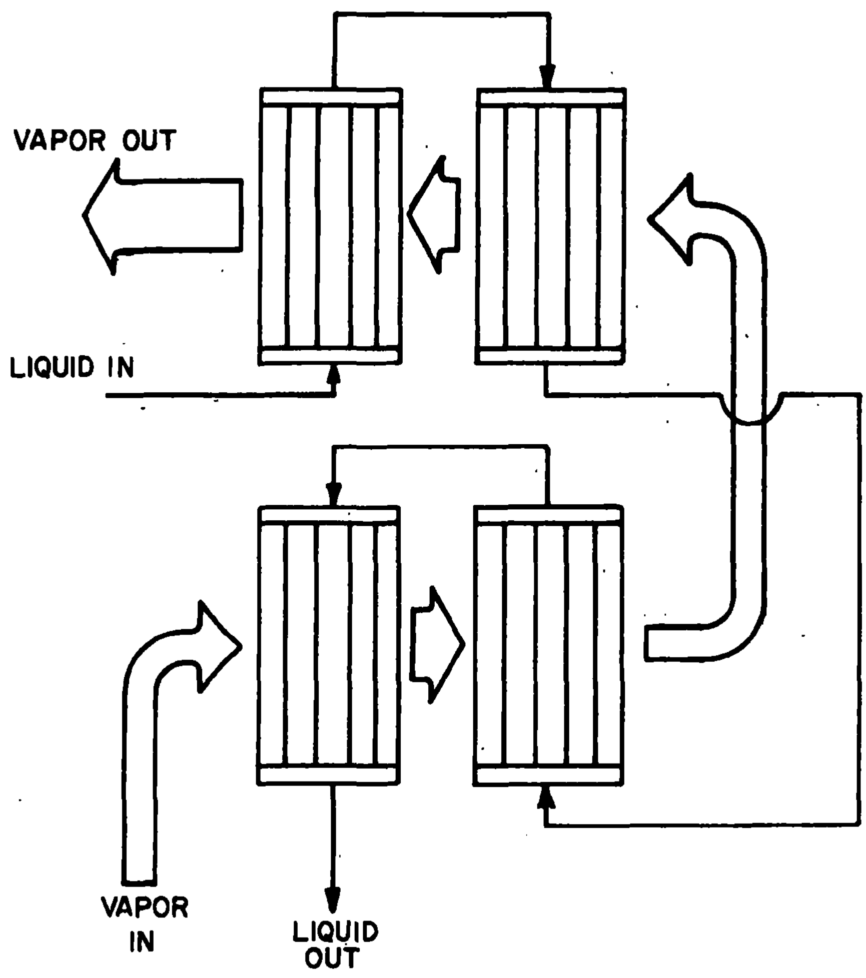

Figure 2.19 Liquid and Vapor Flow Paths in Regenerator.

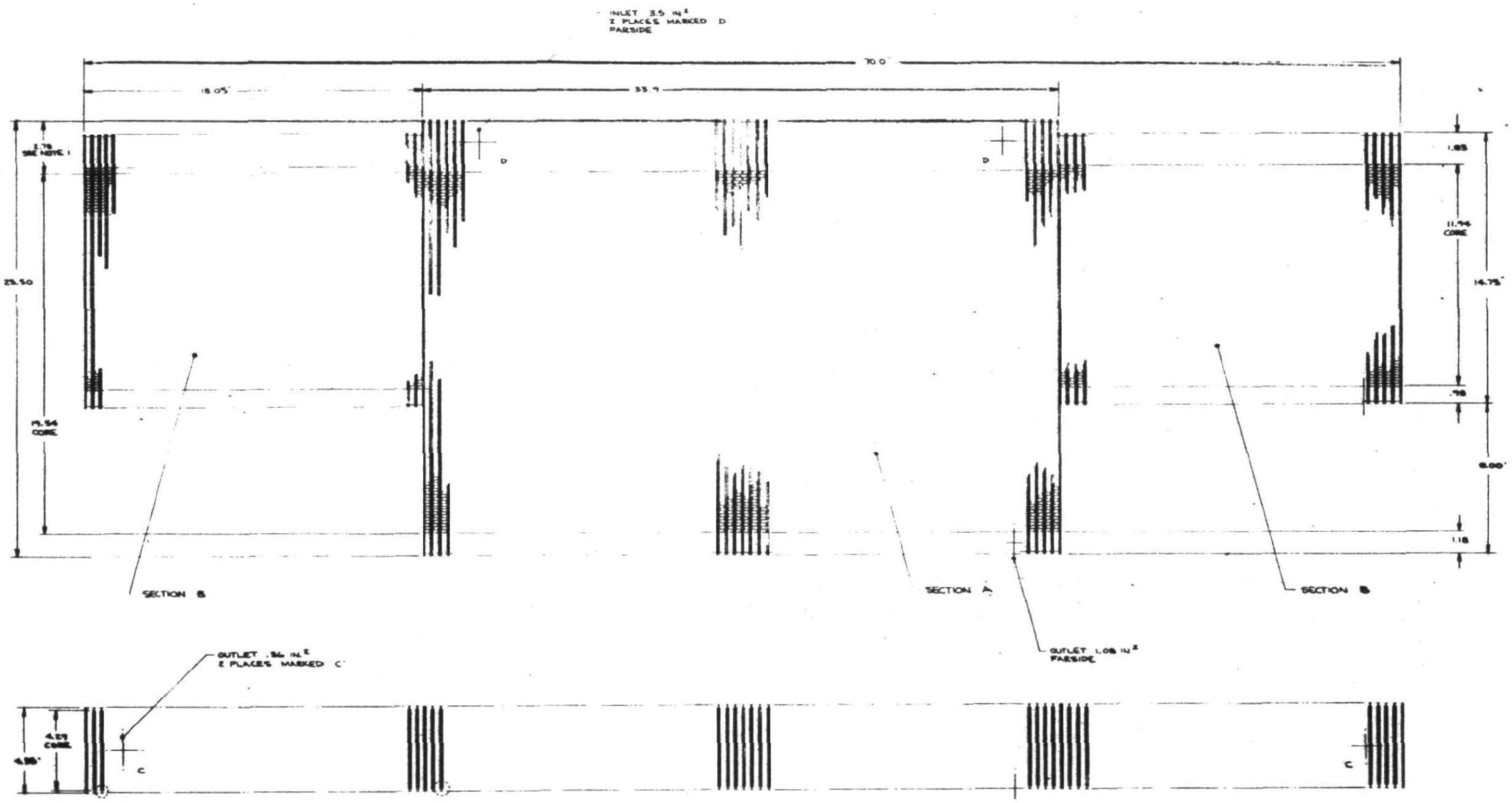

Figure 2.20 Condenser Design Layout.

and the organic side pressure drop is 5 psi. The condenser sections are constructed with top and bottom headers connected by flat tubes; both internal and external finning is used. The condenser construction is brazed aluminum and the design is suitable for one-step brazing for ease in manufacture.

As seen in Figure 2.7, the three condenser fans, one 24 inches in diameter and two 16 inches in diameter, are required to provide uniform condenser air flow and the high cooling air flow rate. The hub thickness of the fans is 2.5 inches and the design point rpm's are 2400 and 4200 for the 24 inch and 16 inch diameter fans respectively. The fans are designed to provide 1.5 inches W. C. head at the design point, with 2.0 inches provided by ram air at the vehicle design point speed of 90 mph.

For optimizing the fan speed, a Morse variable-speed belt drive is used in conjunction with an Eaton Tempatrol viscous clutch. The variable speed drive uses a centrifugally-controlled sheave to vary the expander-fan drive speed ratio. This control provides a constant fan/expander speed ratio of 4.91:1 up to 550 rpm expander speed. Above an expander speed of 550 rpm, the centrifugal sheave maintains a constant fan drive speed of 2700 rpm irrespective of expander speed.

The viscous clutch provides control for part-load conditions as well as ambient temperature variation. This clutch modulates fan speed by sensing the air temperature leaving the condenser and, if the air temperature is low (indicating excess air flow and fan power), the clutch slips, thereby reducing the fan speed. One clutch, constructed integrally with the central 24-inch fan, provides control for all three fans, since the 16-inch fans are belt-driven from the clutch rim.

2.3.5 Startup Sequencing, Safety Controls, and Accelerator Pedal Linkage

Startup sequencing controls are provided for completely automatic system startup, requiring the driver only to turn on the ignition switch. Safety controls are provided to protect the system in case of abnormal operation. These include flame sensing to cut off fuel in case of flameout and overpressure and over-temperature cut-off switches.

The American Bosch hydraulic valving system is electrically controlled. An electrical interface through a LVDT is used between the accelerator pedal and the intake valving control system.

2.3.6 Boost Pump - Inducer - Reservoir Subassembly

The flow schematic of this subassembly is indicated in Figure 2.21. The components of this subassembly provide the following functions: (1) produce the NPSH required by the feedpump during normal operation and during startup; (2) eliminate required condenser subcooling, thereby making more efficient use of the available frontal area for condensing; (3) provide reservoir capacity for working fluid inventory transfer during start condition and transient operation; and (4) prevent separation of lubricant from working fluid in condenser and reservoir.

The centrifugal boost pump, illustrated in Figure 2.22, is designed for a flow rate of 29.4 rpm with head rise of 26.6 feet. The minimum NPSH is 10 inches. The shaft power required is 0.5 hp. The pump is driven by the accessory drive shaft. To eliminate the requirement for a dynamic shaft seal, a permanent magnet drive is used. The pump is constructed primarily of aluminum.

The aluminum reservoir has a capacity of 1.16 gallons and is located at the very top of the engine compartment (see Figure 2.4 and 2.6).

The inducer is of conventional construction; nozzle flow is taken from the centrifugal boost pump.

2.3.7 Accessory and Auxiliary Components

The system includes power steering, power brakes, heating, and air conditioning; these components are identical to those now used in automobiles.

The battery-alternator supplies electrical power to both the system and the normal automotive functions requiring electrical power. The electrical system is 12 Vdc.

An analysis of battery-alternator requirements for both starting and sustained operation at high powers was made to establish the size of these components. For high-power system operation, electrical power is drawn from both the battery and alternator. The battery is an AABM size 24C 84 amp hr battery; the 14 volt alternator is a standard, heavy-duty type and provides 130 amps at 5000 rpm and 90 amps at 2000 rpm (idle condition).

2.4 MAJOR CONCLUSIONS

- A Rankine-cycle automotive propulsion system based on a reciprocating expander and organic based working fluid and competitive in performance to a 1972 351 CID internal combustion engine can be completely packaged in the engine compartment of a 1972 Ford Galaxie with only minor internal sheet metal modifications in the engine compartment required.

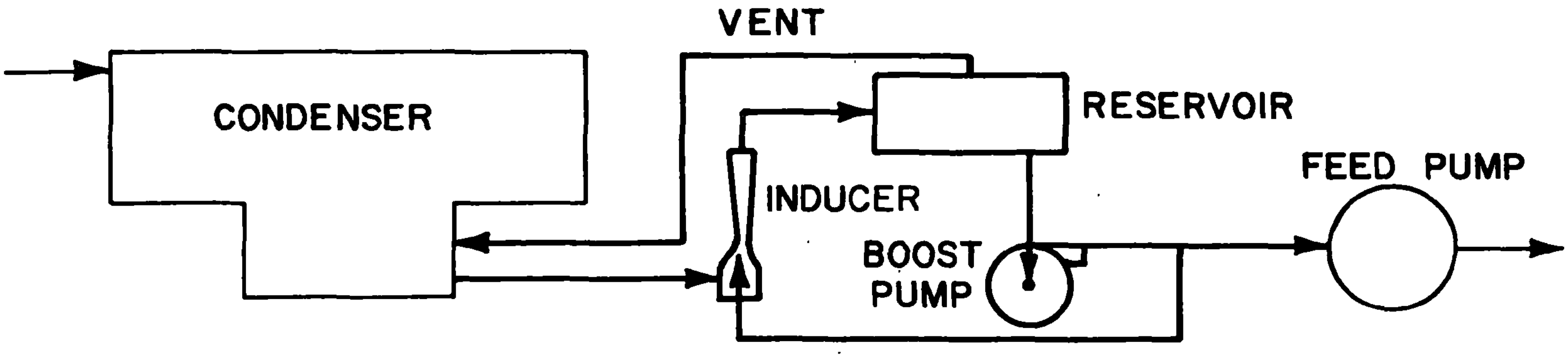

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Figure 2.21 Boost Pump-Inducer-Reservoir Subassembly.

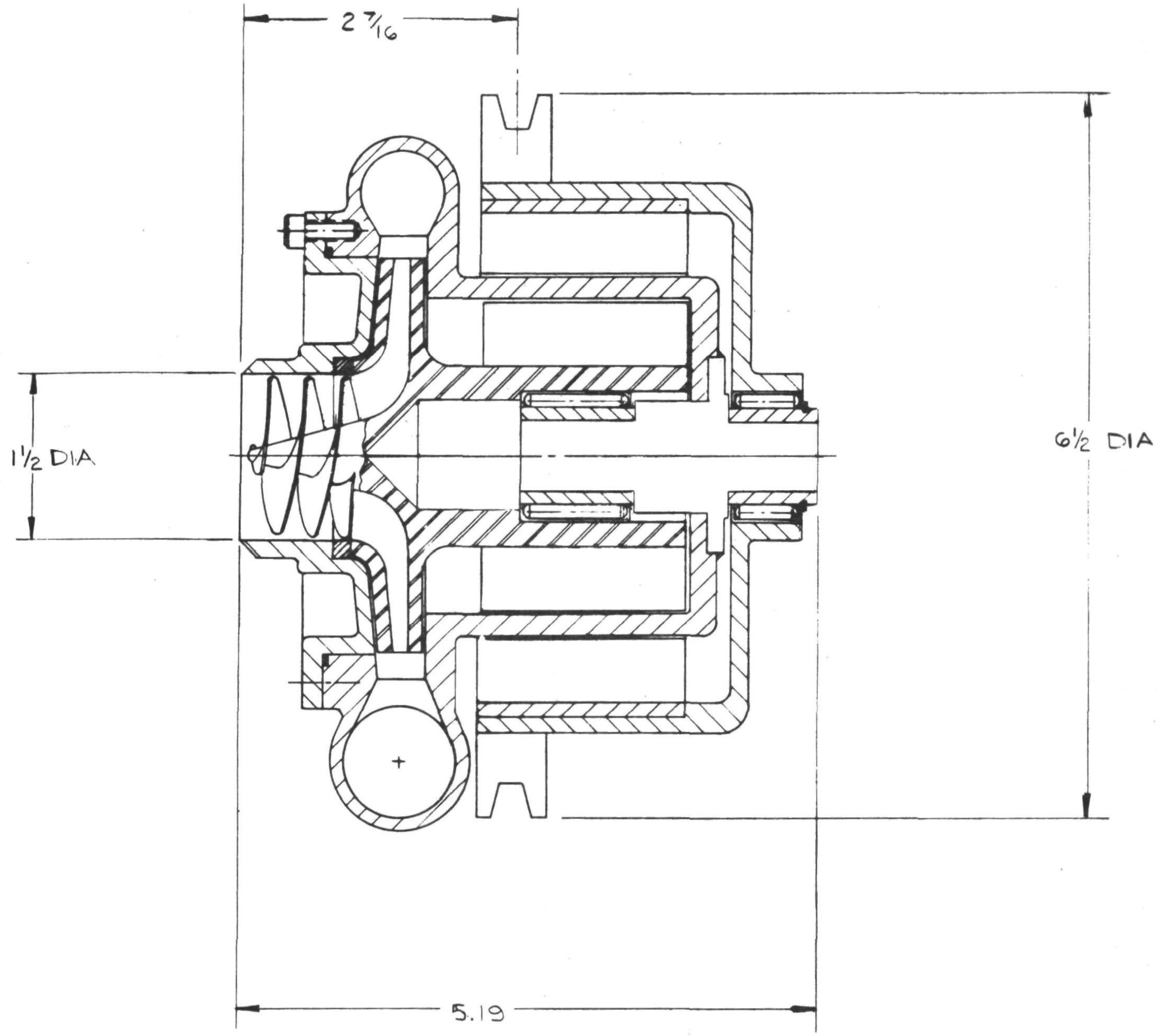

Figure 2. 22 Cross Section of Centrifugal Boost Pump.

The customer-average fuel consumption with the 1972 Ford Galaxie is predicted to be 10 mpg.

- Transient burner tests on an automotive size burner confirm the potential of the Rankine-cycle automotive propulsion system for very low emission levels of NO_x, CO, and unburned hydrocarbons.

- Thermo Electron's system design is based on operation of a complete 5-1/2 hp system with the same boiler outlet temperature as that used for the automotive system. This experience provides a firm technical basis for the system design presented in this report.

- Only low-cost materials are used in the system and the system design is adaptable to high volume production techniques.

3. INTRODUCTION

3.1 PROGRAM GOALS AND HISTORY

The Division of Advanced Automotive Power System Development of the Environmental Protection Agency (EPA) is sponsoring development of low-emission power systems for automotive propulsion systems. The primary goal of these development programs is demonstration of one or more power systems which: (1) produce very low gram/mile emission levels of unburned hydrocarbons, carbon monoxide, nitrogen oxides, and particulates, and (2) fulfill all other requirements for an automotive power system. As part of this program, the development of a Rankine-cycle power system for automobiles is underway at Thermo Electron Corporation. The effort at Thermo Electron is funded partly by EPA and partly by the Ford Motor Company through funds made available under a Ford-TECO business agreement. In addition to its financial support, the Ford Motor Company is providing substantial technical support to the development effort.

The system under development at Thermo Electron Corporation is based on use of an organic-based working fluid with reciprocating expander. Work on this system started in June, 1969; this report presents the results of work performed between June 1970 and November 1971. The program history is summarized below.

3.1.1 <u>June 1969 - June 1970: Conceptual Design Study</u>

The preliminary designs of components for a 100 shp powerplant and package drawings for installation of the powerplant in a 1969 Ford Torino, an intermediate-sized family car, were prepared. A mockup of the powerplant was also fabricated and installed in the engine compartment of a 1969 Ford Torino. Computer models for the components and system were prepared and used in performance and fuel consumption estimates. Thiophene was used as the reference working fluid.

Results of this study are described in the final report issued in June 1970.[1]

3.1.2 June 1970 - November 1971: Detailed Design and Experimental Development in Selected Areas

This phase of the program involved conversion of the conceptual design to a much more detailed, optimized design and experimental development in several of the more crucial areas, specifically:

a. Analysis and bench-testing of full-size, variable cut-off expander intake valving system.

b. Analysis and bench-testing of full-size expander exhaust valve.

c. Simulated testing of full-size rotary shaft seal.

d. Heat transfer and pressure drop measurements on ball matrix fin and evaluation of its use for very compact heat exchangers.

e. Construction and testing of a variable-delivery feedpump based on the conceptual design study.

f. Bearing-lubricant testing with thiophene working fluid.

The detailed design developed in this phase and the experimental results obtained are described in this report. The main body of the report is the final detailed design; the experimental results are described in the appendices, except for the expander test results, which are covered in the main body of the report.

During this phase of the program, two significant changes were made. The reference car size was increased by EPA from the intermediate size (Ford Torino) to the full size (Ford Galaxie) car and new EPA performance specifications[2] were issued. These changes required an increase in the system design point power level from 100 shp to 131 hp. At the beginning of this phase, thiophene was the reference working fluid. In January, 1971, after approximately 4 months of system testing in a 5-1/2 hp system with Fluorinol-85, the decision was made to convert from thiophene to Fluorinol-85 as the system working fluid. This change was made primarily because the safety characteristics of Fluorinol-85 are superior to those of thiophene and the thermodynamic properties are almost as good.

3.1.3 Starting November 5, 1971: Experimental Development Testing of Preprototype System

In this phase of the program, the complete system, as described in this report, is to be experimentally developed and tested as a preprototype system. The schedule calls for initial testing of the complete system in December, 1972.

3.2 TECHNICAL BASIS OF SYSTEM

While its potential for very low emission levels is the main reason for consideration of a Rankine-cycle propulsion system, any new powerplant for automotive application must fulfill many other characteristics, if it is to be seriously considered for large-scale use in automobiles. These other characteristics are:

- Manufacturing cost.
- Reliability and maintenance requirements.
- Driver convenience.
- Acceleration performance
- Fuel economy.
- Packageability and weight.
- Safety.
- Startup and operation over ambient temperature range of -40°F to 125°F.

The selection of the system under development at Thermo Electron was based on attempting to meet all of these characteristics in order to have a competitive and practical system. The most important system considerations which influence meeting these characteristics are working fluid and peak cycle temperature and expander type.

The automotive system design presented in this report is based on experience gained in 1-1/2 years' testing of a 5-1/2 hp system at Thermo Electron Corporation. The working fluid, lubricant, peak cycle temperature, most materials of construction, expander construction, automatic startup sequencing, etc., are based on this test experience. The emphasis behind the 5-1/2 hp system development is commercialization of Rankine-cycle systems in the 5 - 20 hp range for applications such as fork lift trucks, engine generator sets, and other applications where low noise and low pollution are important. Since the prime competitor for these applications is the internal combustion engine, the development goals for the small horsepower Rankine-cycle systems are similar to those for the automotive system where the prime competitor is also the internal combustion engine.

In selecting the working fluid, Fluorinol-85*, for these systems, several requirements were set, based on the anticipated use in commercial systems:

a. The maximum acceptable freezing point is -40°F.

b. With respect to flammability, the fluid must be self-extinguishing. Also, vapor-air mixtures should either be non-explosive or have a very mild reaction.

c. The vapor inhalation and dermal application toxicities should be acceptable for use in the development laboratories at Thermo Electron Corporation with no special requirements other than good ventilation.

d. The maximum working fluid cost when produced in large volumes must be no greater than $1 - $2 per pound.

e. The working fluid must be compatible with low-cost materials of construction.

f. A lubricant which is thermally stable and compatible with the working fluid at peak cycle temperature must be available.

Fluorinol-85 is the best available working fluid which meets all of these requirements in a system with reciprocating expander while

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*Halocarbon Products, Inc., Hackensack, New Jersey.

still providing a useable cycle efficiency. This fluid is used in the 5-1/2 hp system being tested at Thermo Electron Corporation at a boiler outlet temperature of 550°F, and can probably be used at boiler outlet temperatures up to 600°F. It is expected that fluids with higher thermal stability than Fluorinol-85 will become available in the future, permitting higher cycle efficiencies while still retaining all of the required characteristics for the working fluid. If the flammability requirements were relaxed, some currently available fluids would provide a higher cycle efficiency.

The expander used is of the reciprocating type with variable cutoff intake valving. The primary advantages of this type of expander for this application are its low shaft speed without gearing and maintenance of high expander efficiency over a wide speed and power range, thus permitting use of a relatively simple and inexpensive transmission. In addition, production technology for internal combustion engines is directly applicable to the reciprocating expander, and the optimum fluids for a Rankine-cycle system with reciprocating expander require a small regenerator size.

CHAPTER 3

REFERENCES

1. Morgan, D. T. and Raymond, R. J., "Conceptual Design, Rankine-Cycle Power System with Organic Working Fluid and Reciprocating Engine for Passenger Vehicles," Report No. TE4121-133-70, June 1970, Thermo Electron Corporation, Waltham, Massachusetts.

2. "Vehicle Design Goals - Six Passenger Automobile," Revision C, Division of Advanced Automotive Power Systems Development, Environmental Protection Agency, issued May 28, 1971.

4. SYSTEM DESCRIPTION AND CHARACTERISTICS

4.1 INTRODUCTION

The propulsion system design is based on meeting the performance and vehicle specifications established for the Advanced Automotive Power Systems Program by the Division of Advanced Automotive Power Systems Development of the Environmental Protection Agency.[1] The EPA vehicle specifications are for a full-sized American automobile, such as the Ford Galaxie 500 sedan, and the performance specifications are approximately equivalent to those for a 351 CID internal combustion engine with 1970 emission controls and three-speed automatic transmission. A summary of the primary EPA performance specifications is given in Table 4.1.

The vehicle selected as the reference car for system packaging and for performance and fuel economy estimates is the 1972 Ford Galaxie 500 four-door sedan. Performance calculations presented in this chapter indicate that a system designed to provide 131 hp net shaft power output (feedpump and condenser fan power subtracted) at 90 mph vehicle speed provides acceptable performance for the reference car relative to the EPA specifications. All component sizes are therefore based on this design point condition.

The system design has been developed to provide driver convenience equal to that provided by current automobiles. The driver interface is limited to the ignition switch, accelerator pedal and gear selector; system startup and operation are completely automatic. A very important consideration in the system design has been insurance of startup and proper system operation, regardless of environmental

TABLE 4.1

PERFORMANCE SPECIFICATIONS FOR ADVANCED AUTOMOTIVE POWER SYSTEMS PROGRAM

A. GENERAL REQUIREMENTS

Startup: 65% of full power in 45 seconds at 60°F ambient temperature.

Self-sustaining idle within 25 seconds at -20°F ambient temperature.

Idle Fuel Consumption: Not to exceed 14% of fuel consumption rate at maximum power conditions.

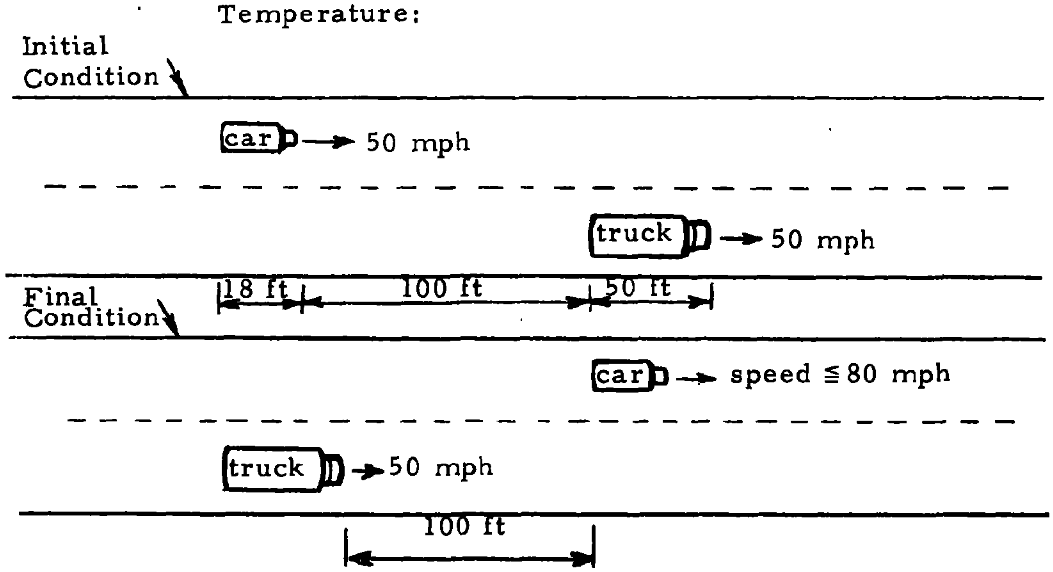
Performance Degradation with Ambient Temperature:
All performance specifications are to be degraded by no more than 5% at ambient temperature of 105°F.

B. ACCELERATION REQUIREMENTS; VEHICLE WEIGHT = FULLY FUELED VEHICLE PLUS 300 LBS; 0% GRADE

Acceleration from Standing Start at 85°F Ambient Temperature:
Distance in 10 seconds ≧ 440 ft
0-60 mph time ≦ 13.5 seconds

Acceleration in Merging Traffic at 85°F Ambient Temperature:
25 to 70 mph time ≦ 15.0 seconds

Acceleration, DOT High Speed Pass Maneuver at 85°F Ambient Temperature:



Time to Accomplish Maneuver ≦ 15 seconds
Distance Traveled by Automobile ≦ 1400 feet

TABLE 4.1 (continued)

C. GRADABILITY AND REQUIRED MAXIMUM SPEED REQUIREMENTS AT 85°F AMBIENT TEMPERATURE; VEHICLE WEIGHT = FULLY FUELED VEHICLE PLUS 1000 LBS

Grade	Vehicle Speed
30%	Start from rest and accelerate to 15 mph
5%	60 mph continuous
	65 mph for at least 180 seconds after acceleration from 60 mph
	70 mph for at least 100 seconds after acceleration from 60 mph
0%	85 mph continuous

conditions or orientation of the vehicle. Consideration has thus been given to conditions such as startup with the car parked uphill or downhill on a steep grade and startup at low ambient temperatures when the internal system pressure is very low ($<$ 0.1 psia).

In the system design and packaging the complete powerplant has been packaged in the existing engine compartment of the 1972 Ford Galaxie with only minor modifications to the base vehicle. In addition, the standard accessory equipment for passenger convenience and comfort has been retained, and packaged with the Rankine-cycle system. These functions include power steering, power brakes, air conditioning, and heating.

In the remainder of this chapter, a description is given of the overall system integration and packaging in the 1972 Ford Galaxie, and of the car performance in fuel economy projections. In Chapter 5, the detailed designs of the system components are described.

4.2 WORKING FLUID-LUBRICANT, SYSTEM SCHEMATIC, AND DESIGN POINT CONDITIONS

4.2.1 Working Fluid-Lubricant

The selected working fluid is Fluorinol-85, a mixture of 85 mole percent trifluoroethanol and 15 mole percent water. The characteristics of the working fluid are summarized in Table 4.2. This working fluid is currently produced in industrial quantity by Halocarbon Products Corporation, Hackensack, New Jersey. Fluorinol-85 has been used at Thermo Electron since September, 1970 in testing of small horsepower (5-1/2 hp) Rankine-cycle power systems with satisfactory results.

A lubricating oil which is thermally stable and compatible with low-cost materials of construction and the working fluid at the peak

TABLE 4.2

WORKING FLUID CHARACTERISTICS

Chemical Composition	85 Mole Percent Trifluoroethanol CF_3CH_2OH, 15 Mole Percent Water (Fluorinol-85)
Average Molecular Weight	87.74
Freezing Point	-82°F
Boiling Point	165°F
Critical Temperature	452°F
Critical Pressure	800 psia
Flammability Characteristics	Non-Supporting, Non-Explosive
Toxicity Characteristics	Not classified as toxic via dermal or inhalation pathways (MCA, 1970)[2,3]
Liquid Density (200°F)	1.25 gms/cm^3
Lubricant	Commercial Refrigeration Oil - Immiscible with Fluorinol-85
Thermal Stability and Material Compatibility	Demonstrated at 550°F (Boiler outlet temperature) by operation of complete power system for 450 hours Capsule tests indicate potential for use at higher boiler outlet temperature

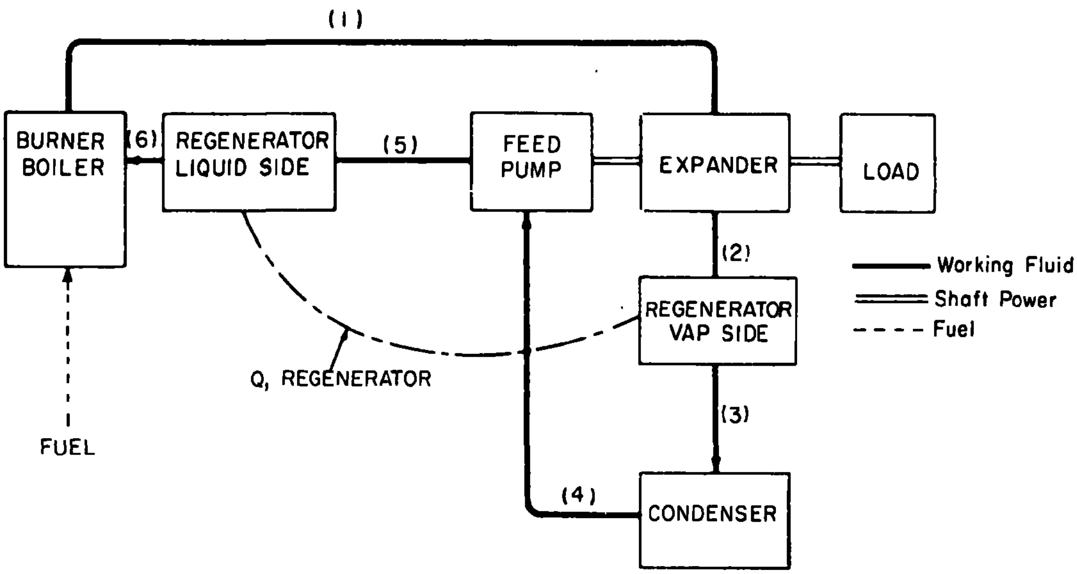
cycle temperature of 550°F is used for lubrication of the expander and feedpump bearings and sliding surfaces. Because of its thermal stability, it is not necessary to absolutely prohibit the oil from passing through the boiler and high-temperature side of the expander; it is expected that a small fraction ($< 1\%$) of the circulating flow through the condenser and boiler is the lubricant. Satisfactory experience has been obtained with use of this oil in complete Rankine-cycle systems operating with the same boiler outlet temperature as the automotive system. The vapor side of the regenerator is used as an oil separator to return oil to the crankcase and maintain the circulating oil fraction at an acceptable level to prevent serious degradation of the regenerator, condenser, and boiler performances.

The lubricant is immiscible with the working fluid and has a density less than that of the working fluid. Consideration in the system design and integration has therefore been given to insure that no lubricant traps occur in the system, resulting in an inadequate lubricant level in the crankcase. The immiscibility of the working fluid-lubricant facilitates adequate and positive lubrication of the expander-feedpump bearings on startup and provides a supply of a working fluid - free oil to the expander hydraulic valving pump.

4.2.2 System Description and Design Point Conditions

4.2.2.1 Basic Cycle

The basic components and cycle conditions comprising the Rankine-cycle powerplant are illustrated in terms of a flow schematic and a T-S diagram in Figure 4.1. The Fluorinol-85 vapor leaves the boiler at a temperature of 550°F and a pressure of 700 psia (State Point 1). This vapor is expanded through the reciprocating expander, producing shaft power applied to the load. The exhaust vapor leaves the expander



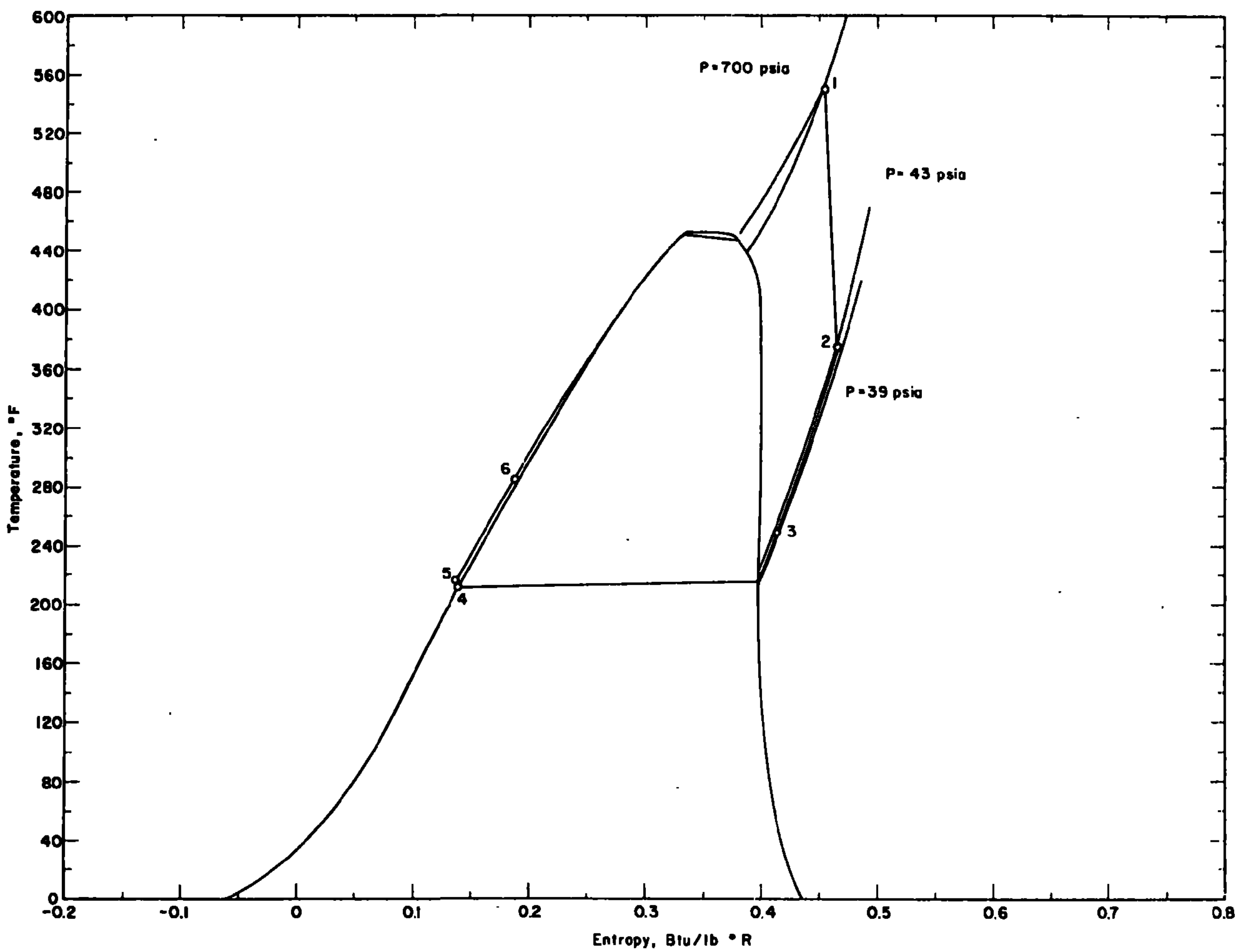


Figure 4.1 Illustration of Basic Components and Cycle for Rankine-cycle Power System with Fluorinol-85 Working Fluid.

at a pressure of 43 psia and temperature of 376°F in a superheated condition (State Point 2). Because of the relatively high temperature of the vapor, the overall cycle efficiency can be improved by using a regenerative heat exchanger in which energy is transferred from the expander exhaust vapor (State Point 2 → State Point 3) to the feed liquid going to the boiler (State Point 5 → State Point 6), thereby reducing the fuel requirement for a given system power output. The vapor leaves the regenerator at a pressure of 39 psia and enters the air-cooled condenser, where the vapor is completely condensed at an average temperature of 215°F (State Point 3 → State Point 4).

The condensed liquid then enters the feedpump, where the fluid pressure is raised from the condenser discharge pressure of 35 psia to 820 psia (State Point 4 → State Point 5). The liquid then flows through the regenerator liquid side (State Point 5 → State Point 6) and enters the boiler at State Point 6. Energy from the combustion of fuel is then transferred to the Fluorinol-85 flowing through the boiler, producing the high pressure-high temperature vapor at State Point 1 and completing the basic cycle.

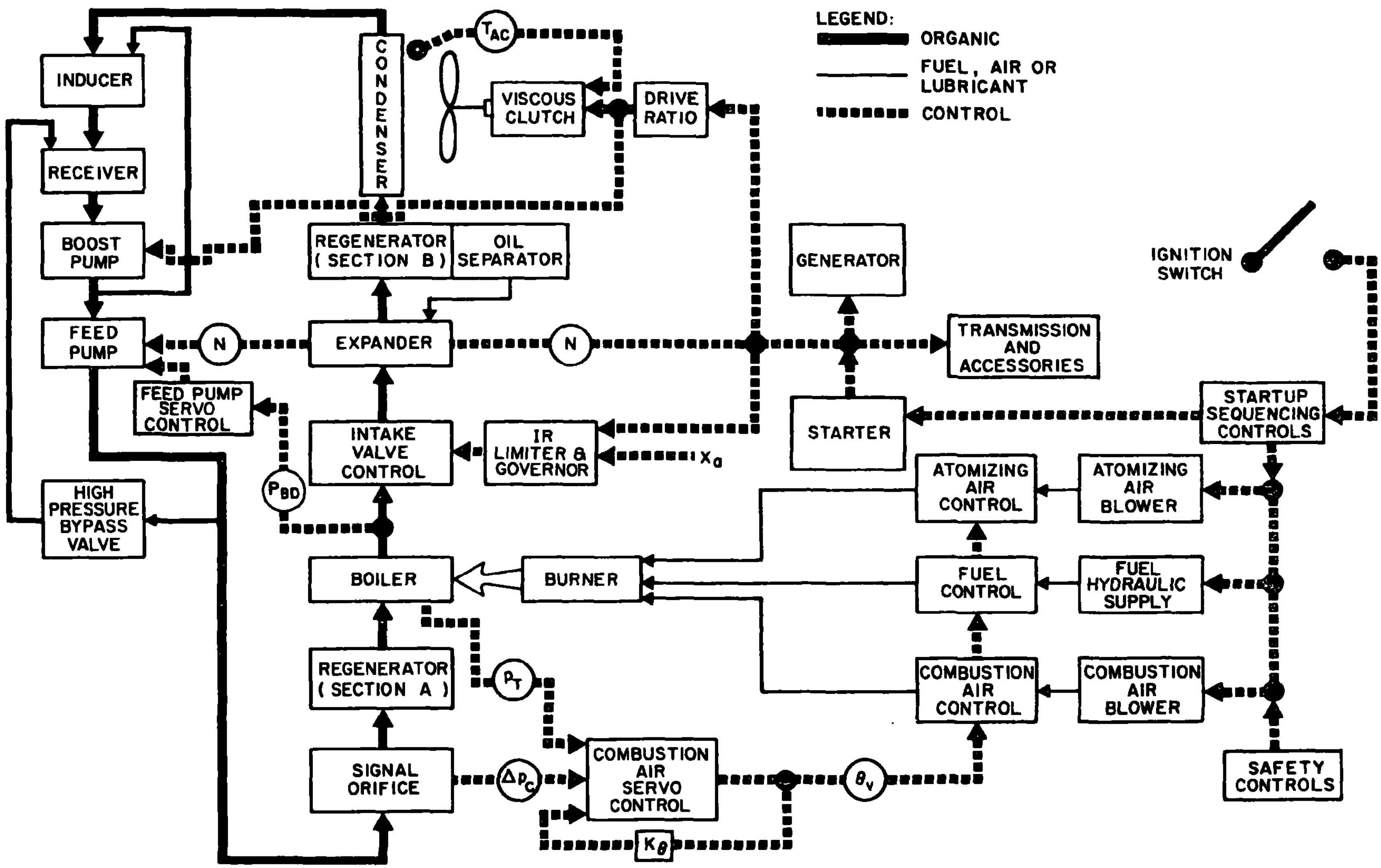
4.2.2.2 System Description

In arriving at a complete system for automotive propulsion, many alternative choices are available on the approach to be followed in synthesis of the system. In this section, a listing is presented of the various components and subassemblies making up the complete system, and the logic underlying the selected approach is briefly outlined. In Chapter 5, a detailed description of the various components comprising the complete system is presented.

The system block diagram is illustrated in Figure 4.2 and includes all components or subassemblies making up the system. For purposes of discussion, here as well as in Chapter 5, the various components are grouped in subassemblies based either on function (as the accessory and auxiliary components) or on preassembled and integrated units to be installed in the system (as the expander-feedpump-transmission subassembly).

a. Expander-Feedpump-Transmission Subassembly: The expander is a reciprocating piston type with hydraulically operated intake valving. The method of controlling power output from the expander is to vary the cut-off point (or intake ratio) of the expander intake valving with vapor at full boiler pressure supplied to the intake valves. Thus, for high power or torque outputs, a large intake ratio is used; for low power outputs, a small intake ratio is used. The intake valve control is directly controlled by the accelerator pedal, as illustrated in Figure 4.2. The hydraulically-operated intake valves used in the system can be controlled down to the point of zero lift with no vapor flow through the expander. Thus, no throttle valve is required between the boiler and expander.

An alternative control procedure for the expander power output is use of fixed cut-off intake valve timing with a throttle valve to reduce the working fluid pressure entering the expander. The reasons for selection of the variable cut-off approach, discussed in the June 1970 report,[4] are: (1) to maximize the wide-open-throttle acceleration performance of the system with given component sizes, and (2) to maximize the system efficiency in the low-power and low-speed range where an automobile operates on the average, thereby providing good fuel economy.



SYMBOLS:

p_T - PRESSURE, INDICATING ORGANIC TEMPERATURE

p_{BD} - PRESSURE, BOILER DISCHARGE

Δp_C - PRESSURE DIFFERENTIAL, SIGNAL ORIFICE

X_a - DISPLACEMENT, ACCELERATOR

N - SPEED, EXPANDER CRANKSHAFT

θ_V - DISPLACEMENT, AIR CONTROL VALVE

K_θ - GAIN FUNCTION, FEEDBACK

T_{AC} - TEMPERATURE, CONDENSER DISCHARGE AIR

Figure 4.2 System Schematic.

Included as part of the expander is an intake ratio limit control and a governor using expander speed as input. At any given expander speed, as the cut-off point of the intake valving is increased, the organic flow rate increases. Since the boiler heat transfer rate required is approximately linear with organic flow rate, assuming constant boiler outlet pressure and temperature, the intake ratio limiter automatically prevents exceeding the intake ratio at which the boiler capacity would be exceeded with a resultant drop in boiler outlet pressure and temperature. Since, as discussed below, the expander is allowed to idle at zero vehicle speed in order to drive the automotive accessories, a governor is required to maintain the expander idle speed under varying power demands by automatic adjustment of the intake ratio.

An automatic transmission is used between the expander and vehicle drive shaft. The transmission serves two functions: it permits the expander to idle at zero vehicle speed, to operate the vehicle accessories, and it supplies either two or three gear ratios to provide improved wide-open-throttle acceleration response from a given system. Its function is therefore identical to the transmission used in I/C engine-powered cars.

The feedpump is directly driven by the expander at expander speed and integrated with the expander. This procedure eliminates any dynamic seal for the feedpump. With the variable intake expander valving, the organic flow at a given expander speed can vary from zero to a maximum corresponding to the boiler capacity. A variable displacement piston pump is used to provide the proper organic flow rate in an efficient manner over the complete power-speed range of the system, thereby minimizing the feedpump power at any operating point. The boiler outlet pressure is used to vary the feedpump displacement through a servo-control and force amplifier to maintain constant boiler outlet pressure.

b. Combustion System-Boiler Subassembly: This subassembly includes the burners, boiler, burner controls, combustion air blower, fuel supply, and atomizing air compressor. The combustion air blower and atomizing air compressor are driven by an electric motor operating from the battery on startup or from the alternator during normal operation. This requirement results from the desire to operate the burners at peak burning rate during startup to minimize the cold startup time. In order to have reasonable motor, battery, and alternator sizes, as well as to reduce the parasitic load on the system, a prime consideration in the design of the burner-boiler unit has been low combustion side pressure drop within the packaging restrictions for the 1972 Ford Galaxie.

The burner fuel/air control has been designed for rapid response to transient power changes of the system and regulation of the burning rate to a level corresponding to the new power level in the fastest practical time. The approach followed to provide rapid response to power transients is use of an orifice in the liquid organic feed line to the boiler; the ΔP created across this orifice by the organic flow is then applied to the air servo control, providing rapid response to organic flow rate changes to the boiler. Thus, for a power increase resulting from depression of the accelerator pedal by the driver, the vapor flow rate through the expander and from the boiler increases. This results in a transient decrease in boiler outlet pressure; the feedpump displacement is then automatically increased by the boiler outlet pressure control, bringing the organic flow rate to a higher level. The increased organic flow rate provides an increased orifice ΔP to the combustion air servo control, which responds in approximately 200 milliseconds to bring the burner rate to the level corresponding approximately to that required for the organic flow rate. Power decreases result in the inverse of the above.

The orifice control input provides an approximate balance between the organic flow rate entering the boiler and the burning rate. For fine tuning of the burning rate to insure constant boiler outlet temperature, a temperature sensor responding to the vapor temperature leaving the boiler is provided. This sensor generates a pressure signal, proportional to the organic temperature, which is also applied to the combustion air servo control.

The combustion air servo control modulates the air flow to the burner in response to the input signals. Feedback for stability is also provided in the servo control. For low emissions, reasonably tight control on the fuel/air ratio must be provided. The fuel control is thus directly linked to the combustion air control.

The burner uses an air atomizing fuel nozzle. For low emissions, it may be desirable to control the atomizing air pressure for different burning rates. Current tests at Thermo Electron indicate that this control may not be required with constant atomizing air pressure used at all burning rates.

c. Regenerator: The regenerator is located directly above the expander and mounted to the expander exhaust flange. The regenerator is also used as the system oil separator, returning most of the oil in the expander exhaust vapor to the crankcase. A screen separator is used in the vapor inlet header to remove a major fraction of the oil before the flow enters the regenerator vapor-side fins in order to minimize the effect of the oil film on the regenerator performance. The vapor side fins of the regenerator act as an additional separator. The separated oil drains by gravity back to the expander crankcase.

The oil leaving the regenerator in the vapor passes through the condenser, pumps, boiler, and expander back to the regenerator.

d. Condenser-Condenser Fans-Condenser Fan Drive and Controls Subassembly: The power required for the condenser fans represents the largest parasitic load to the system; strong consideration has thus been given in the condenser-condenser fan selection to minimizing the fan power for a specific condensing rate. In addition, a fan drive and control system has been devised which approximately optimizes the fan speed for peak system performance under all operating conditions. The drive and controls are made of commercially available parts.

The condenser fans are directly driven by the expander through a centrifugally-controlled variable speed belt drive. This drive provides a high speed ratio (high relative fan speed) at low expander speeds and low speed ratio at high expander speeds (low relative fan speed) for maximum utilization of ram air. A thermostatically-controlled viscous clutch operating on the air temperature leaving the condenser provides additional control. This viscous clutch is identical to those now used on some automotive I/C engines.

e. Startup Sequencing and Safety Controls: Startup sequencing controls are provided for completely automatic system startup requiring the driver only to turn on the ignition switch. The startup sequencing first operates the burner at full firing rate for rapid boiler heatup. When the boiler has been heated to a preset temperature, the starter or cranking motor is switched on, turning over the expander and feedpump until the system becomes self-sustaining. The startup can be safely and effectively accomplished, even if the boiler is initially completely dry or initially filled with working fluid.

Sufficient safety controls are provided to prevent damage to the system and vehicle if system malfunction occurs. These controls include functions such as high pressure and high temperature shutdown of the system and flame sensing to stop fuel flow if flame-out occurs.

f. Inducer-Receiver-Boost Pump Subassembly: These components, while not part of a basic Rankine-cycle system, are extremely important for reliable startup and operation of the system under any possible operating conditions of the vehicle and for proper handling of the lubricant passing through the condenser, that is, for preventing accumulation of oil in the condenser. The use and design of these components is based on startup and operating experience with the completely automatic 5-1/2 hp systems constructed and tested at Thermo Electron. For application in a car, the condensate header at the bottom of the condenser is the lowest part of the system, and, if the vehicle is parked on a steep downhill grade, can be lower than any part of the system.

The feedpump is a piston pump with spring-loaded suction valving and requires an NPSH of several feet for operation of the suction valving and for pump operation without cavitation. On cold startup, the NPSH to the pumping system must be provided solely by liquid head; in addition, the internal pressure in the system on cold startup is normally very low (0.08 psi at 0°F). During operation, the required NPSH can be provided by subcooling of the liquid from the condenser. However, this procedure detracts from the frontal area available for the condenser, requires an additional air-cooled heat exchanger, reduces the system efficiency, and is not applicable for system startup. By utilizing the centrifugal boost pump to pressurize the feedpump suction to a pressure $\geqq$ 5.0 psia, positive feedpump operation under all conditions without cavitation can be guaranteed without subcooling. The centrifugal boost pump is designed for an NPSH of 10 inches, which the receiver provides under all conditions. The boost pump also provides flow for operation of the inducer. The boost pump is driven by the condenser fan drive, as illustrated in Figure 4.2. To eliminate a dynamic seal, a standard magnetic drive is used for the boost pump.

The inducer operates as a sump pump and maintains the condenser free of liquid, maximizing the condenser performances as well as prohibiting accumulation of oil in the condenser. The inducer is operated by flow from the boost pump and pumps directly into the receiver.

The receiver provides allowance for working fluid inventory changes in the system and guarantees liquid supply to the boost pump and feedpump under all conditions. Under normal operation, the receiver would be about three-fourths filled with liquid. Under abnormal conditions or on start-up, when liquid might accumulate in the condenser or some other location in the system, the receiver provides sufficient liquid volume to insure liquid priming of the boost pump. The receiver is located at the top of the engine compartment and provides 10 inches head to the boost pump when completely empty.

g. Accessory and Auxiliary Components: Auxiliaries are defined as other components required for operation of the system, and accessories are components required for the vehicle operation and for passenger convenience and comfort. These components include the alternator, battery, starter or cranking motor, air conditioning and heating of passenger compartment, power steering, and power brakes. In arriving at the alternator size for the system, consideration has been given to the worst operating conditions for the vehicle to insure sufficient alternator capacity for the electrically-operated system components as well as normal vehicle requirements.

4.2.2.3 Design Point Conditions

Component sizes have been based on a net shaft horsepower output of 131.1 hp (gross shaft power less condenser fan power and feedpump power) at a vehicle speed of 90 mph and an expander speed of 1800 rpm, with ambient temperature of 85°F. Performance calculations indicate that this system power output and components sized on this basis satisfy the EPA performance specifications as outlined in Table 4.1 when installed in the 1972 Ford Galaxie.

In Figure 4.3, the state points at the design point are illustrated on the system flow schematic; in Table 4.3, the design point conditions for the major components are summarized. The design point conditions were calculated from the computer model of the complete system.

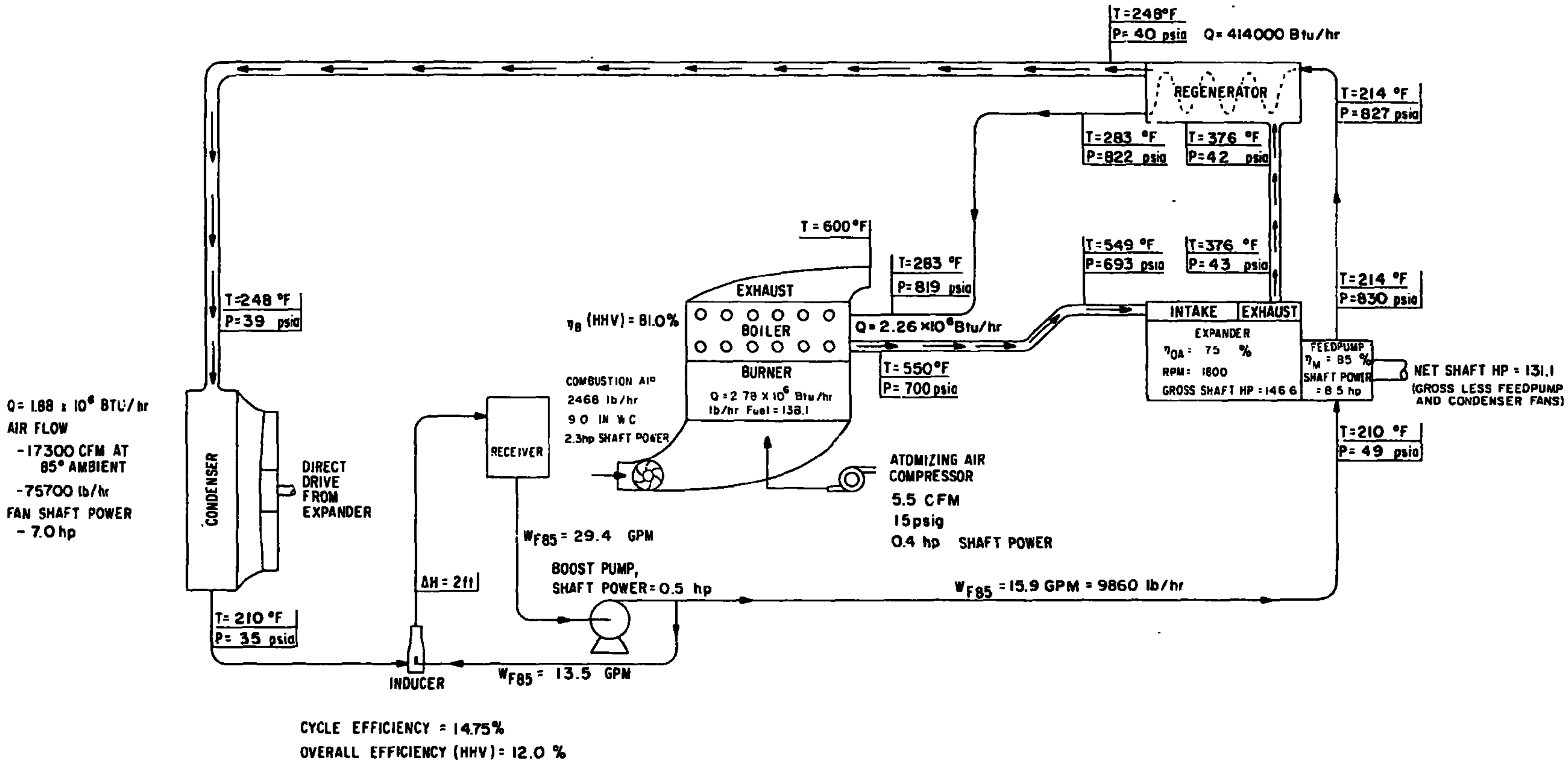


Figure 4.3 State Point Diagram.

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TABLE 4.3

DESIGN POINT CONDITIONS

Vehicle	1972 Ford Galaxie
Boiler	
Outlet Temperature	550°F
Outlet Pressure	700 psia
Heat Transfer Rate	2.26×10^6 Btu/hr
Efficiency (HHV)	81.0%
Burning Rate (HHV)	2.78×10^6 Btu/hr
Expander	
Intake Ratio	0.175
Speed	1800 rpm
Gross Shaft Power	146.6 hp
Displacement	184 in^3
Bore (V-4)	4.41 in
Stroke (V-4)	3.00 in
Overall Efficiency	75%
Regenerator	
Heat Transfer Rate	414,000 Btu/hr
Effectiveness	81%
Vapor Temperature	376°F → 248°F
Liquid Temperature	214°F → 283°F
Condenser	
Heat Transfer Rate	1.88×10^6 Btu/hr
Average Pressure	37 psia
Average Condensing Temperature	214°F
Ambient Air Temperature	85°F
Air Flow Rate	75,700 lb/hr 17,300 CFM Entering
Effectiveness	80%
Fan Power (with Utilization of Ram Air at 90 mph)	7 hp

TABLE 4.3 (continued)

Feedpump	
Organic Flow Rate	9860 lb/hr
	15.9 gpm
Efficiency	85%
Shaft Power	8.5 hp
System	
Net Shaft Power (Less Feedpump) and Condenser Fan)	131.1
Cycle Efficiency	14.8%
Overall Efficiency (HHV)	12.0%

4.3 SYSTEM INTEGRATION AND PACKAGING IN 1972 FORD GALAXIE

In designing the components described in Chapter 5, an essential input has been packageability in the engine compartment of the 1972 Ford Galaxie 500 with only minor modifications. Considerable iteration in the component designs has been required in fitting the system into the car. The system packaging is described in this section, and includes every component of the system as well as power steering, power brakes, and passenger compartment heating and air conditioning. It has been possible to retain the standard air conditioning-heater case assembly on the 1972 Ford Galaxie 500, which occupies considerable volume in the engine compartment.

In packaging the system, the following modifications to the vehicle were required to eliminate local interference points.

- Transmission Tunnel

  A standard three-speed automatic transmission with torque converter has been used in the packaging. The diameter of the torque converter plus the necessity to locate the expander-feedpump-transmission subassembly as far to the rear of the engine compartment as possible without violation of the fire-wall necessitated some increase in the transmission tunnel size near the firewall. The transmission tunnel is thus enlarged locally (adjacent to torque converter housing) by 1-1/2 inches vertically and 1-1/2 inches on the passenger side. The position of the accelerator pedal and its operation are not affected.

- Steering Linkage
  The drag link of the steering linkage must be depressed locally by 3/4 inch to clear the expander oil pan.

- Number Two Cross Member
  The rear upper edge of the number two cross member must be depressed locally (at the center) by 3/4 inch to clear the expander oil pan.

- Sway Bar
  The center of the sway bar must be depressed locally by 2 inches to clear the combustion blower motor.

- Rear Expander Mount
  The rear mount for the expander must be moved 10 inches rearward.

- Head Lamp Sockets
  The headlamps must be moved 1-1/4 inches forward to clear the condenser.

The following modifications are required to improve the flow of air to the condenser and through the engine compartment:

- Redesign of grill.

- Replacement of four headlamps with two headlamps.

- Louver front surface of wheel aprons to increase flow area for exhaust of cooling air.

- Removal of horizontal panels at front of apron assembly.

The complete system layout is illustrated in Figure 4.4, a side view looking from the driver's side of the engine compartment, and Figure 4.5, a view from the top of the engine compartment. Overlays are included with these figures for aid in identifying the various compartments of the system. Sectional views are provided in Figures 4.6, 4.7 and 4.8, as identified in Figure 4.4. Section A-A, given in Figure 4.6, is a front view just in front of the expander looking toward the rear of the vehicle; this view includes the V-4 expander, feedpump, starter motor, regenerator, and battery. Section B-B, given in Figure 4.7, is a front view just in front of the combustion system-boiler, looking toward the rear. Section C-C, given in Figure 4.8, is a rear view of the condenser-condenser fan arrangement.

As can be seen from these drawings, the expander-feedpump-transmission subassembly is located to the rear of the engine compartment. The cranking motor is integrated with this assembly. The regenerator is located directly above the expander and is mounted on the expander. The condenser is located in the very front of the engine compartment, with the condenser fans mounted to the condenser shroud on the rear of the condenser. The combustion system-boiler is located between the expander and condenser and is placed as close as possible to the expander to leave sufficient flow area for exhaust of the condenser cooling air. The combustion blower and burner controls are located between the two burners.

The accessory drive is taken from the rear housing of the expander so that only one dynamic shaft seal is required in the system. A splined shaft with universals is used to bring the accessory drive

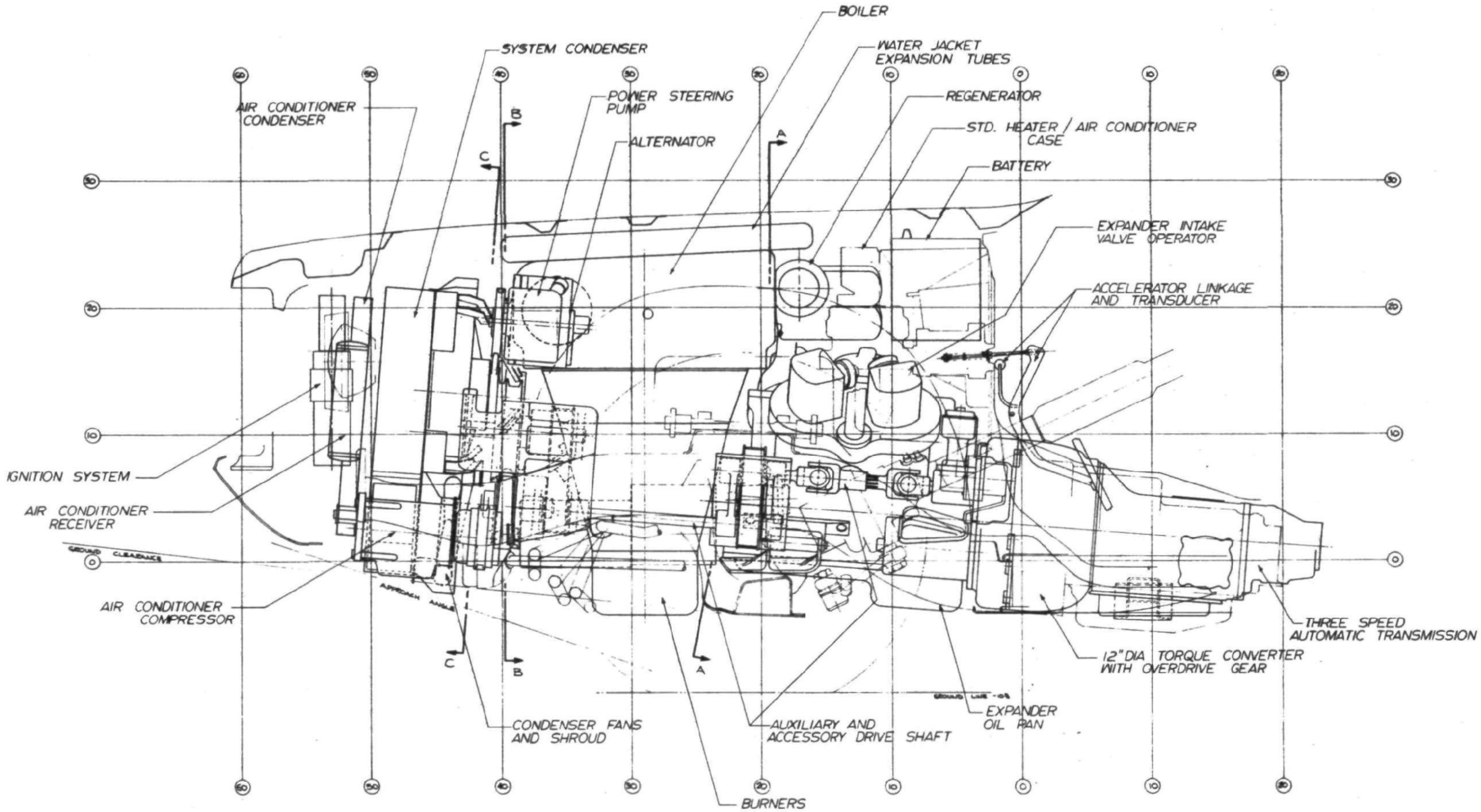

Figure 4.4 Side View, Packaged System.

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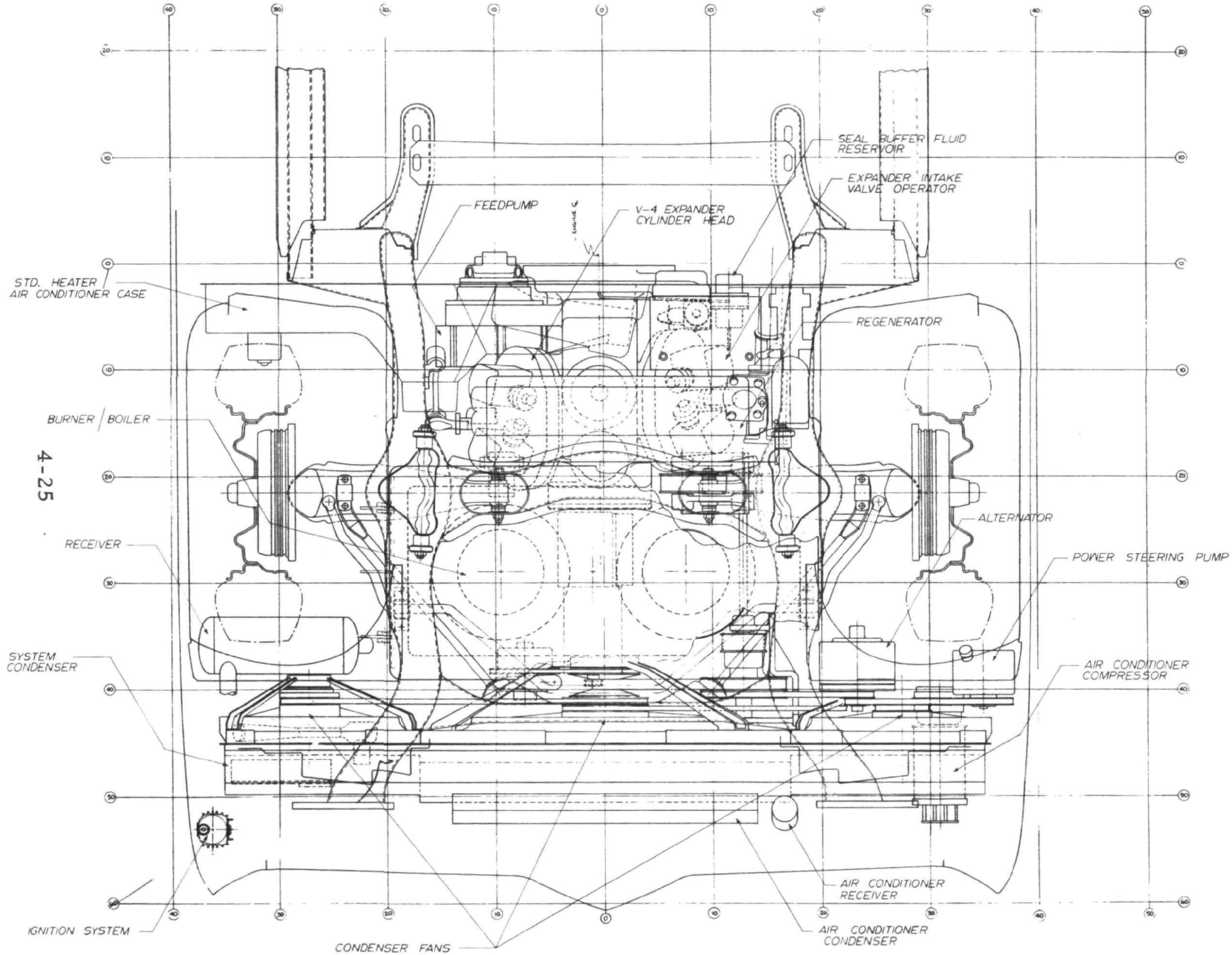

Figure 4.5 Top View, Packaged System.

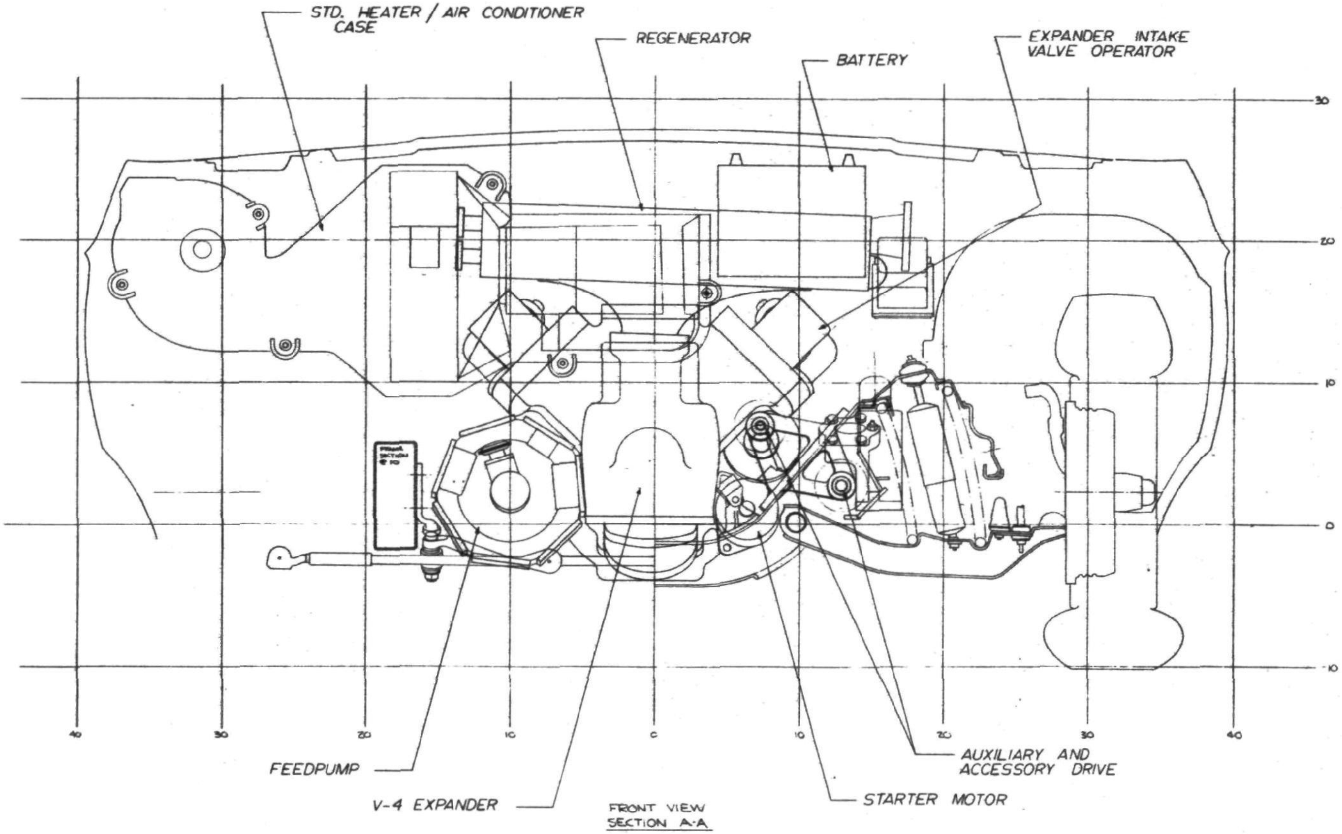

Figure 4.6 Section A-A from Figure 4.4

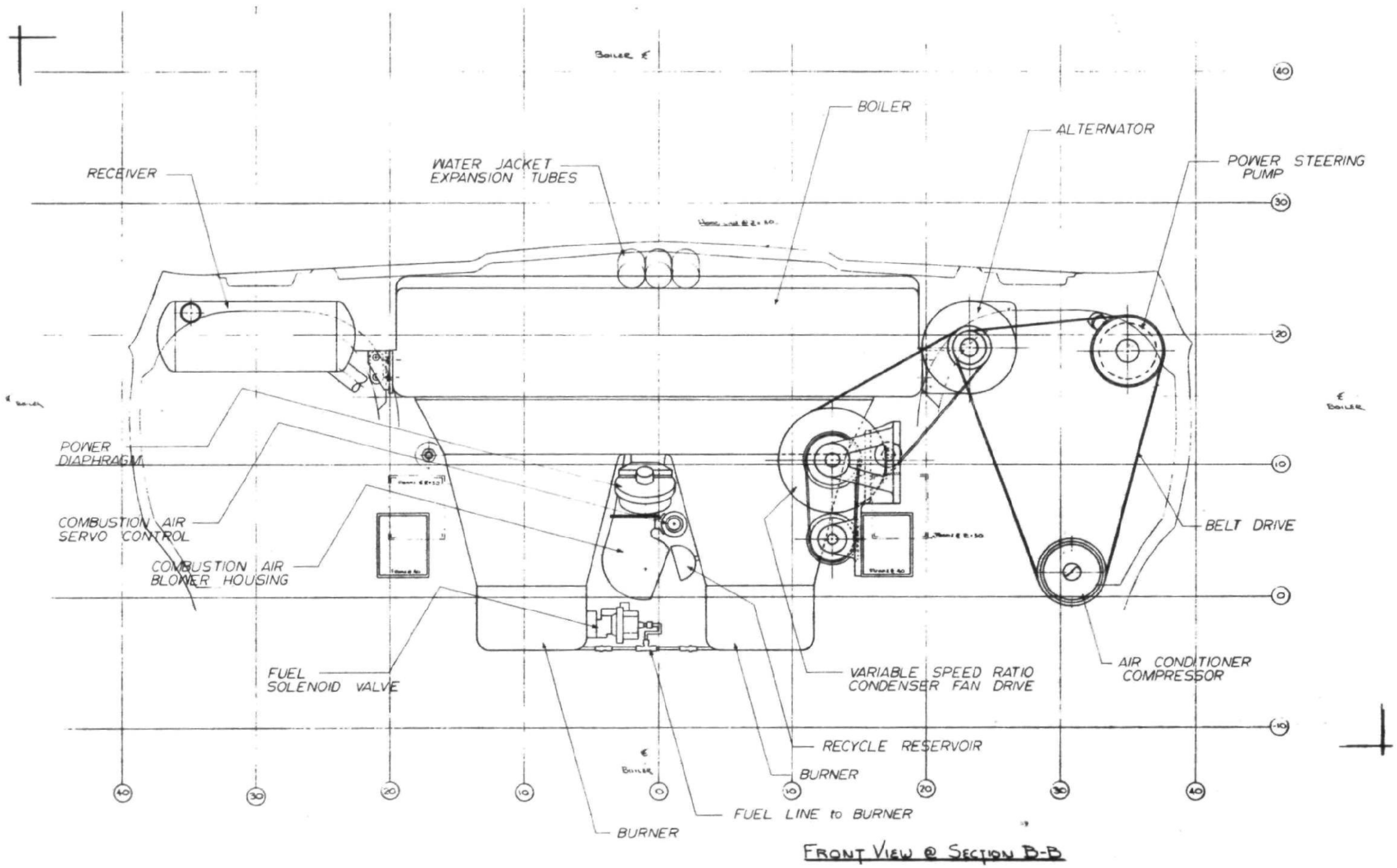

Figure 4.7 Section B-B from Figure 4.4

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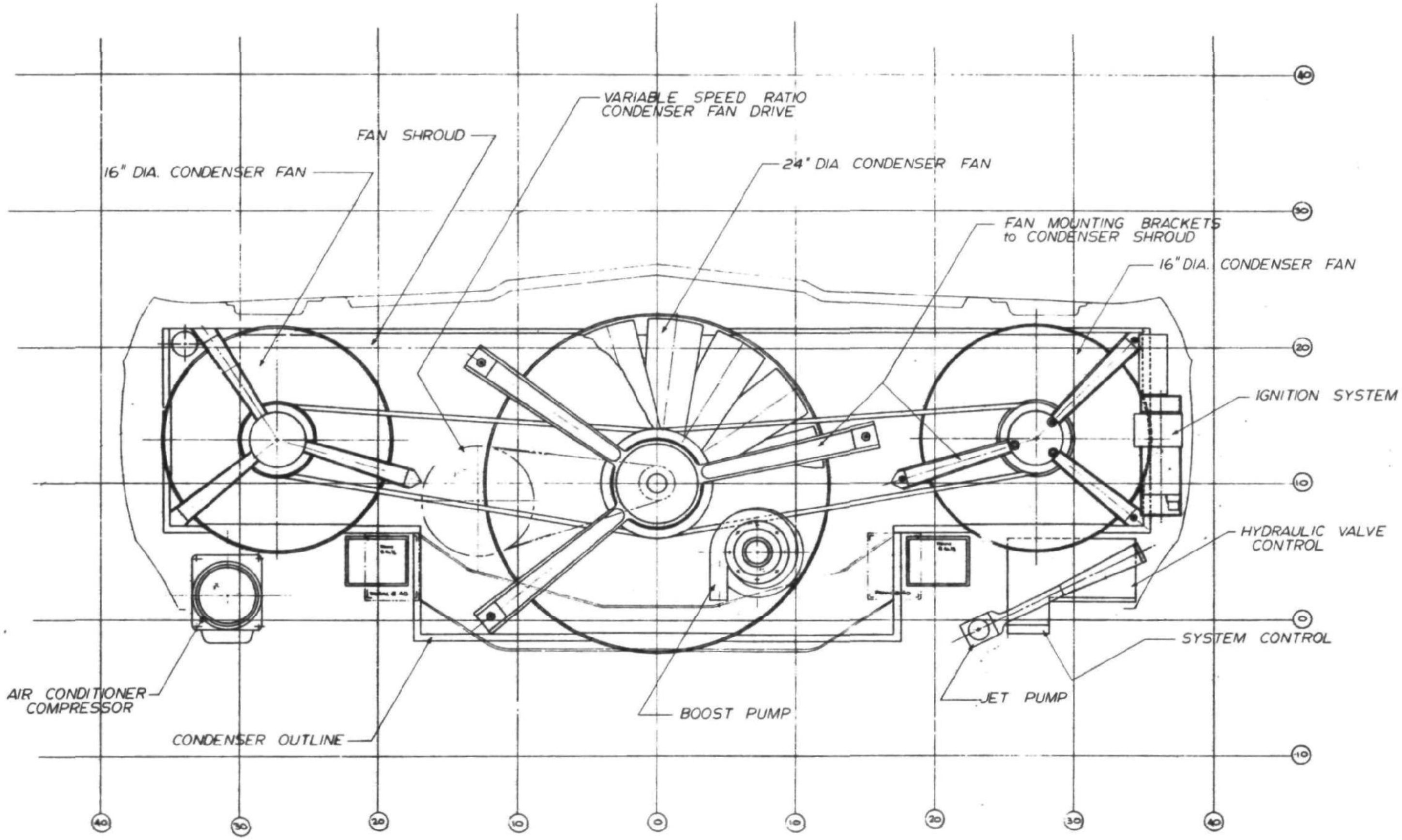

Figure 4.8 Section C-C from Figure 4.4

to the front of the expander. A belt drive is used at this position to a second shaft to bring the accessory drive to the rear of the condenser. Belt drives from this point drive the condenser fans (through the condenser fan variable speed belt drive), boost pump, air conditioning compressor, power steering pump, and alternator. These components are all located just to the rear of the condenser and positioned so as not to restrict condenser cooling air flow significantly through the engine compartment.

The battery is located at the top rear of the engine compartment, above the expander and to the rear of the regenerator. The receiver for the Rankine-cycle system is located beside the boiler tube bundle, at the top of the engine compartment on the passenger side. The inducer and boost pump are located near the bottom of the engine compartment, as required by functional considerations.

Components located in front of the condenser are the ignition system, air conditioning condenser, and air conditioning receiver. The system control assembly is located at the front of the engine compartment to provide a relatively cool environment for the electrical components. The system control assembly includes inlet valve controls, relays, startup sequencing, and safety cut-offs. An electrical transducer in the passenger compartment and attached to the accelerator pedal linkage provides the operator signal to the control assembly.

Because of relative motion between components mounted on the vehicle frame and those mounted on the expander, flexible connections are provided in the lines connecting the boiler outlet and expander, boost pump discharge and feedpump suction, and regenerator outlet and condenser inlet. For clarity, the piping has not been shown on the drawings presented here. Combustion gas exhaust ducts (not shown on these drawings) are used to bring the exhaust gases to the bottom of the engine compartment, and to the rear of the vehicle. These ducts are taken from the top rear of the boiler and pass on either side of the expander near the rear of the engine compartment.

4.4 ACCELERATION PERFORMANCE AND FUEL CONSUMPTION CALCULATIONS

Calculations of acceleration performance and fuel consumption over typical driving cycles have been made using steady-state computer models of both the Rankine-cycle power system and the vehicle. The calculational procedure is outlined in Figure 4.9. Computer models of the Rankine-cycle system components plus the Fluorinol-85 thermodynamic and physical properties are used in the system performance prediction program. System performance characteristics are generated by this program in the form of tables providing hp, burning rate, and system efficiency (as well as any other desired system characteristic) over the range of expander speeds and intake ratios encountered in system operation.

In Figures 4.10 and 4.11, performance maps, cross plotted from these tables, are presented for the 131.1 hp system in the form of hp vs expander rpm, with lines of constant efficiency shown. The maps are based on use of the Dana two-speed transmission, with the first map applying to first gear with a high expander rpm relative to vehicle speed and the second map to second gear with a low expander rpm relative to vehicle speed. Characteristics of the Dana transmission are discussed in Appendix VII.

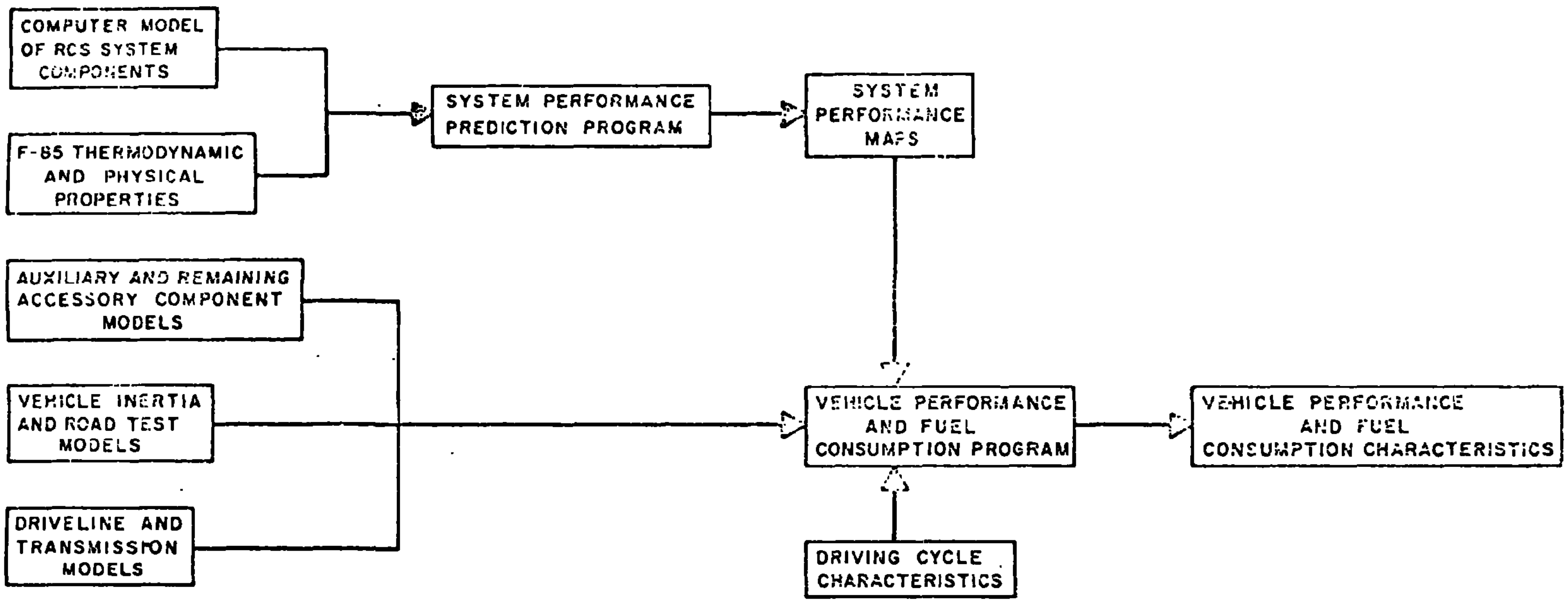

Figure 4.9 Vehicle Performance and Fuel Consumption Calculation.

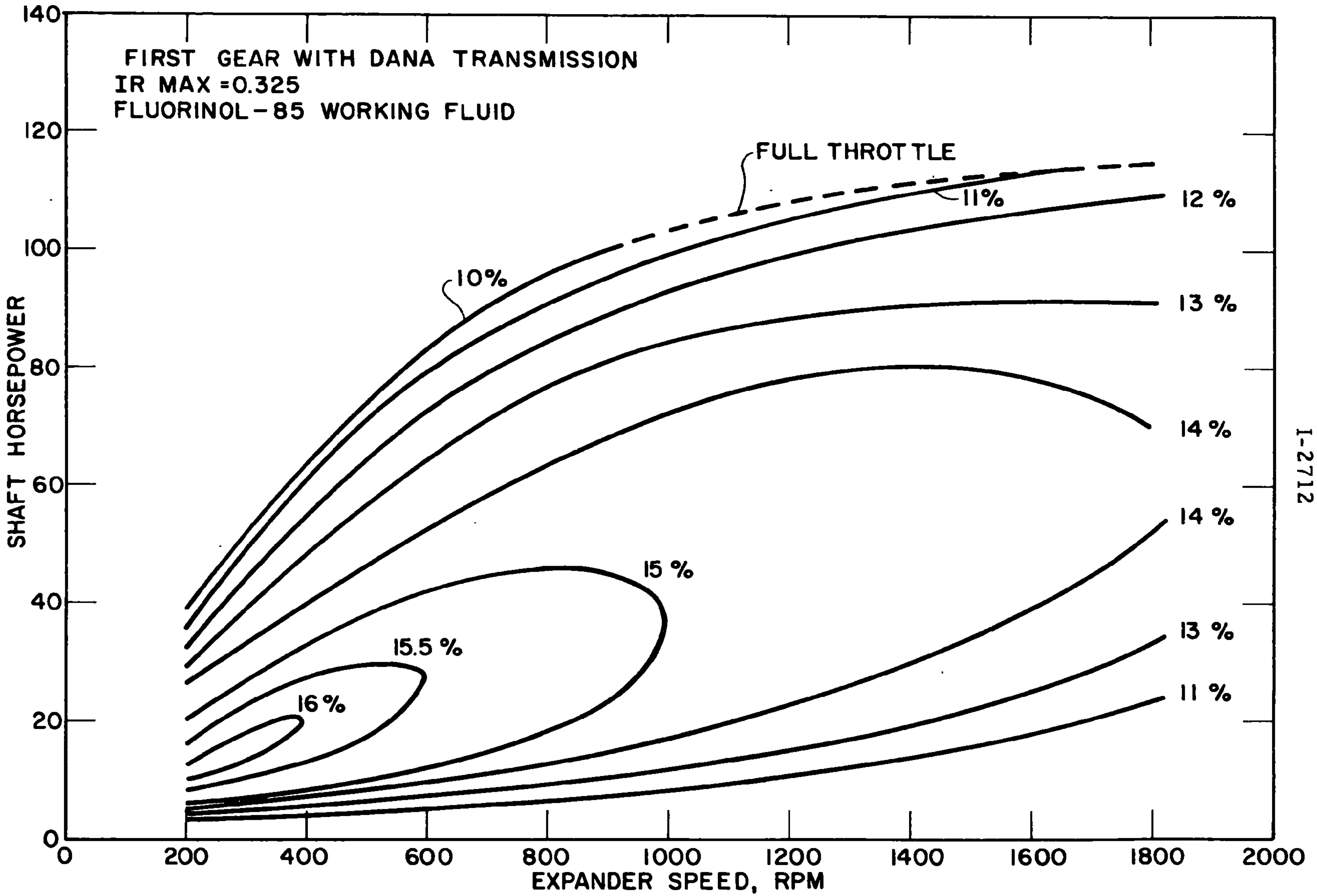

Figure 4.10 Performance Map with Transmission in First Gear (High Expander Rpm Relative to Vehicle Speed).

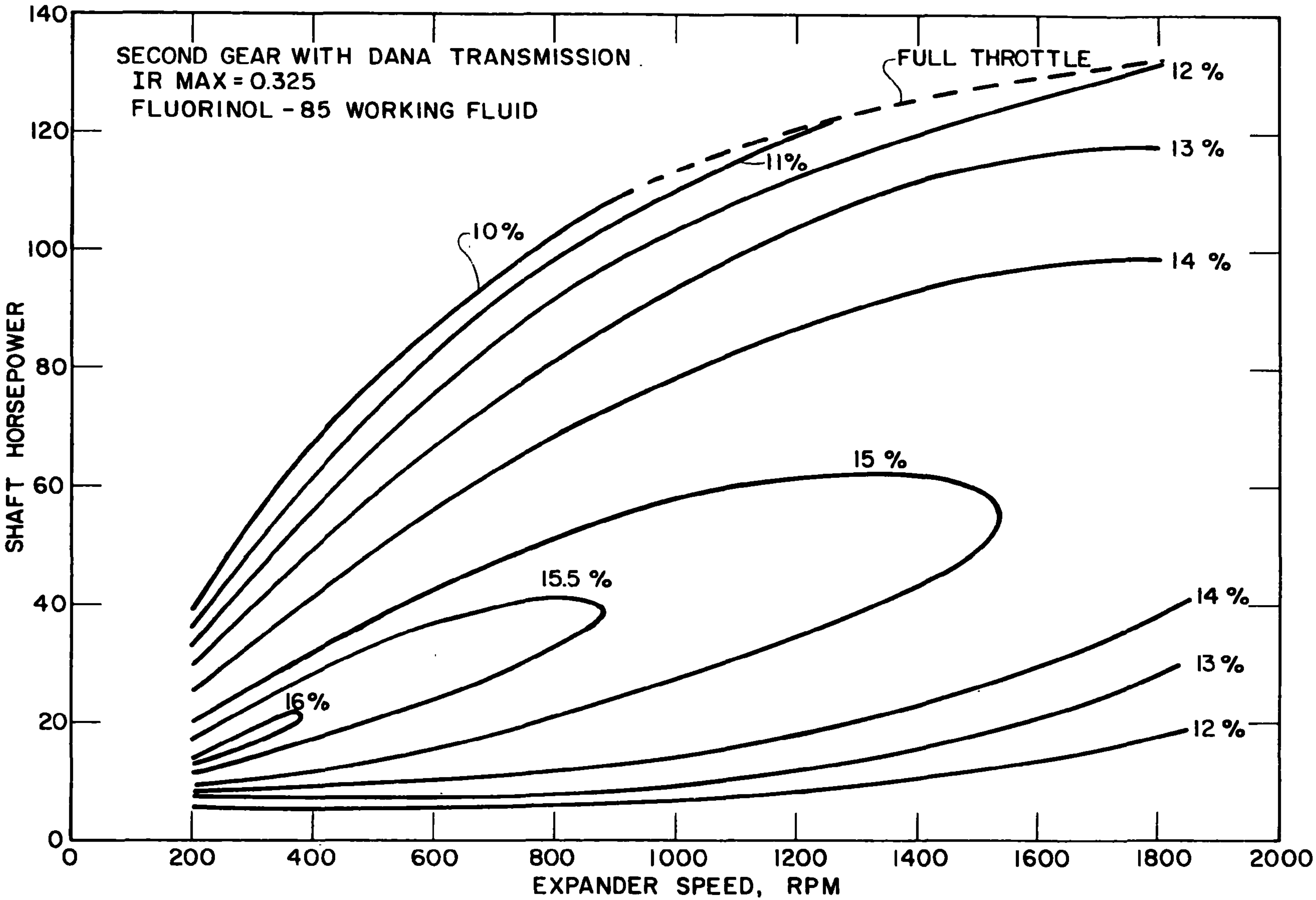

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Figure 4. 11 Performance Map with Transmission in Second Gear (Low Expander Rpm Relative to Vehicle Speed).

These performance maps are used as input to the vehicle performance and fuel consumption program. This program includes all inertia effects, road load, mechanical efficiencies of rear end and transmission, vehicle accessories such as power steering and air conditioning, and any accessory loads not included in the system performance predictions. Any desired driving requirement or condition can be provided as input to the program.

The estimated weight of the 131. 1 hp system is given in Table 4. 4. In Table 4. 5, the vehicle weight breakdown is provided for the 1972 Ford Galaxie with a Rankine-cycle powerplant. The vehicle curb weight with full fuel tank is 4276 lbs; for comparison, the weight of the same car with 351 CID I/C engine and three-speed automatic transmission is 4066 lbs. For wide-open-throttle acceleration and fuel consumption calculations, 300 lbs weight was added to the curb weight to provide a "test" weight of 4576 lbs. For gradability, 1000 lbs weight was added to the curb weight, to provide a gradability "test" weight of 5276 lbs.

The wide-open-throttle acceleration performance of the vehicle is provided in Table 4. 6 and compared with the EPA specifications. The system meets the specifications for 0-60 mph acceleration, distance traveled in 10 seconds from standing start, and 25-70 mph acceleration. The system does not quite meet the passing specification but is very close.

In Table 4. 7, the gradability of the vehicle is presented. The vehicle is able to pull off a 35% grade, and is able to maintain 15 mph on a 30. 7% grade. The car can maintain 70 mph on an 8. 9% grade. The vehicle gradability meets all EPA specifications. The vehicle top speed is approximately 103 mph on a level road.

TABLE 4.4

ESTIMATED SYSTEM WEIGHT BREAKDOWN

Expander	345
Burner-Boiler	330
Combustion Blower Atom. Air Compressor Fuel Pump	9
Combustion Blower Motor	15
Feedpump	42
Transmission	158
Step-up Gear	19
Regenerator	21
Condenser	72
Condenser Shroud	3
Condenser Fans, Pulley, Brackets	22
Variable Speed Fan Drive	21
Boost Pump Inducer	12
Reservoir	2
Starter	12
Alternator	19
Igniter	2
Buffer Fluid Reservoir	2
Exhaust Ducts	12
Electrical Box	4
Plumbing	16
Fluid and Lubricant	30
Battery	45
TOTAL	1213

TABLE 4.5

VEHICLE WEIGHT BREAKDOWN
FOR PERFORMANCE CALCULATIONS
BASED ON 1972 FORD GALAXIE 500

Vehicle Less Propulsion Powerplant	3063 lbs
Complete Rankine-cycle Powerplant with Three-Speed Automatic Transmission	1213 lbs
Vehicle Curb Weight (Fully Fueled)	4276 lbs
Passenger Weight for Performance and Fuel Consumption	300 lbs
Test Weight for Performance and Fuel Consumption	4576 lbs
Passenger Weight for Grade Velocity	1000 lbs
Test Weight for Grade Velocity	5276 lbs

TABLE 4.6

VEHICLE ACCELERATION PERFORMANCE

Vehicle Test Weight 4576 lbs Ambient Temperature 85°F Dana Transmission, Two Speed		
	System Performance	EPA Spec
0-60 mph	13.36 sec.	≦ 13.5 sec.
0-10 seconds	457.9 ft.	≧ 440 ft.
25-70 mph	15.0 sec.	≦ 15.0 sec.
Passing, 50-80 mph	15.4 sec. 1472 ft.	≦ 15.0 sec. ≧ 1400 ft.

TABLE 4.7

GRADABILITY

Vehicle Test Weight 5276 lbs Ambient Temperature 85°F *Dana Transmission* Two Speed		
Vehicle Speed, mph	Grade % System Performance	Grade % EPA Spec.
0	35.3%	Start from Rest on 30% grade
10	34.6%	
15	30.7%	30%
20	26.8%	—
30	19.8%	—
40	13.8%	—
50	11.3%	—
60	8.97%	—
70	6.84%	5%
103	0%	85 mph

The vehicle fuel economy is presented in Table 4.8 for steady speed and for three drive cycles. One driving cycle is that specified in the Federal test procedure for emissions measurement. The other two are driving cycles used by the Ford Motor Company for evaluation of their automobiles, one cycle being typical of suburban driving conditions and the other typical of city driving conditions. The Ford Motor Company customer average is the arithmetic average of these two driving cycles and gives 10.0 mpg for the 1972 Ford Galaxie with Rankine-cycle powerplant.

4.5 EMISSION PROJECTIONS FROM RANKINE-CYCLE SYSTEM

The primary incentive for development of a Rankine-cycle automotive propulsion system is its potential for very low emission levels which not only meet the 1976 Federal objectives, but also are significantly less than the Federal objectives. To demonstrate this potential, Thermo Electron Corporation has made emission measurements on a burner designed for a 100 hp automotive Rankine-cycle system for an intermediate-size American car such as the Ford Torino; the burner was operated transiently over burning rates corresponding to operation of the vehicle over the Federal emission test driving cycle. The fuel/air control used in these transient tests was similar to that to be used in the automotive system. The burner fired into a water-cooled "boiler" with approximately the same configuration as in the system. The procedure for measuring the exhaust emissions from the boiler was identical to that of the Federal Register[5] and included use of the three-bag constant volume sampler. The details of the measurement are described in Appendix VI.

The results of this test are presented in Table 4.9. The measured emission levels in grams/mile are below the 1976 Federal standard by

TABLE 4.8

FUEL CONSUMPTION

Vehicle Test Weight	4576 lbs
Ambient Temperature	85°F
Dana Transmission	Two Speed
Constant Speed, 0% Grade	
MPH	MPG
30	15.38
40	15.15
50	13.52
60	11.76
70	10.33
80	8.54
85	7.84
Constant Speed, 5% Grade	
60	5.68
70	4.99
Driving Cycles	
Federal Driving Cycle for Emissions	10.81 mpg
FOMOCO Suburban Cycle	11.41 mpg
FOMOCO City Cycle	8.61 mpg
FOMOCO Customer Average	10.01 mpg

TABLE 4.9

EMISSION LEVELS MEASURED OVER
FEDERAL DRIVING CYCLE

Emissions (grams/mile)	Transient Test Result*	Federal 1976 Standard
NO_x	0.29	0.4
CO	0.22	3.4
UHC	0.14	0.41

*Actual gas mileage used for tests was 12.1 mph. The latest performance calculation predicts 10.8 mpg for the Federal emission test driving cycle and the measured emission levels have been increased by 12% to reflect the change in fuel economy.

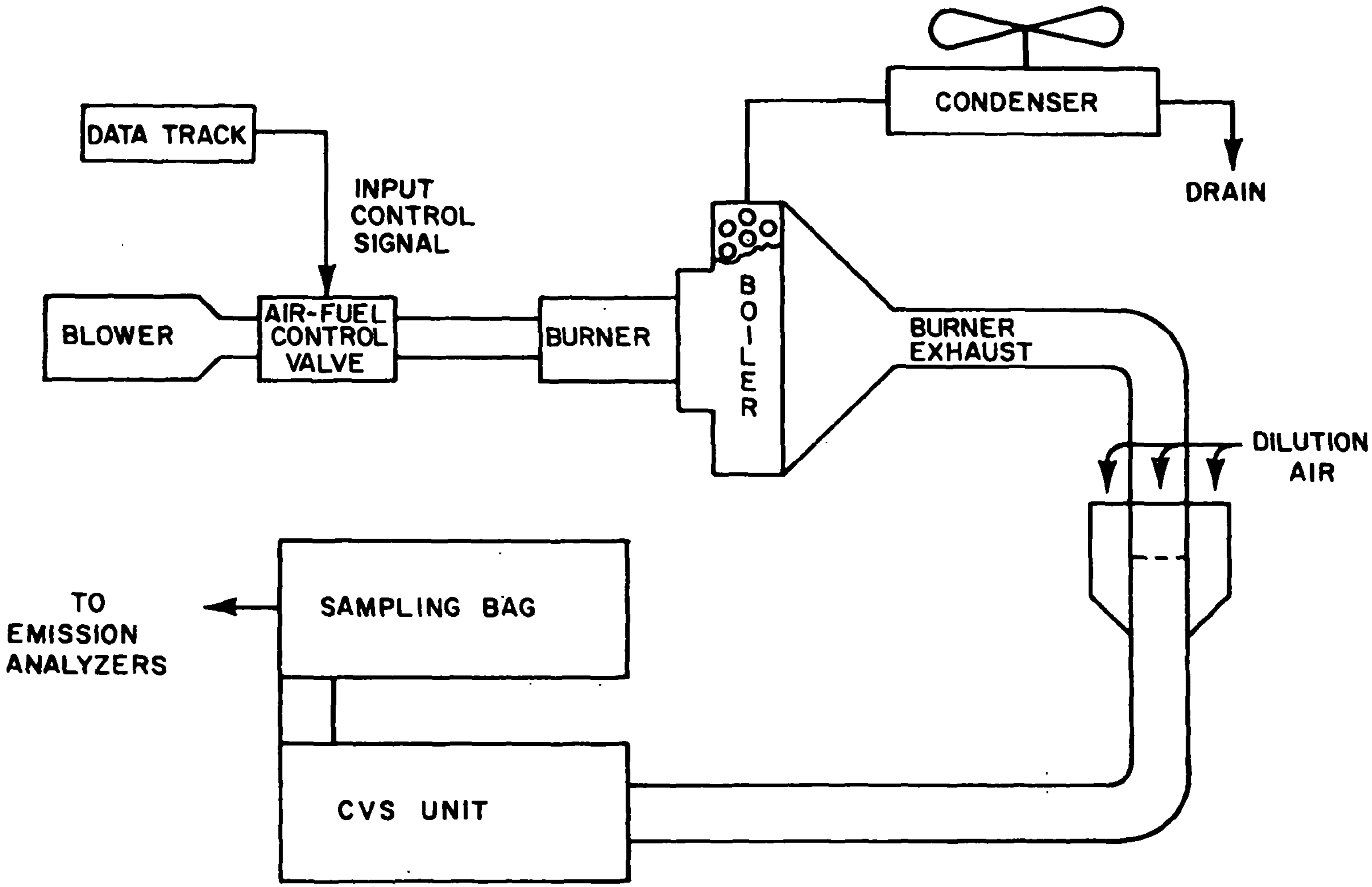

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Figure 4.12 Burner Configuration Used in Emission Tests.

a factor of 1.4 for NO_x, 15.4 for CO, and 2.9 for UHC. These tests conclusively demonstrate the low emission levels attainable with a Rankine cycle power plant for automobiles.

Steady-state measurements indicate that use of exhaust gas recirculation (EGR) would result in even lower NO_x emission rates. Transient tests with EGR have not yet been made, but such tests are planned in the future.

CHAPTER 4
REFERENCES

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2. Manufacturing Chemists Association (1970) Guide to Precautionary Labeling of Hazardous Chemicals, 7th Ed., Manual L-1, Washington, D. C.

3. Blake, D. A., and Brown, D. R., Evaluation of Trifluoroethanol Toxicity and Hazard, April 1971, University of Maryland, Baltimore, Maryland.

4. Morgan, D. T., and Raymond, R. J., "Conceptual Design, Rankine-Cycle Power System with Organic Working Fluid and Reciprocating Engine for Passenger Vehicles, Report No. TE4121-133-70, June 1970, Thermo Electron Corporation, Waltham, Massachusetts.

5. "Exhaust Emission Standards and Test Procedures," Federal Register, Vol. 36, No. 128, Friday, July 2, 1971, Part II. Environmental Protection Agency.

5. COMPONENT DESCRIPTIONS

In this section, a detailed description of the component designs and characteristics is presented. These components are identical to those used in the system packaging described in Chapter 4.

5.1 EXPANDER-FEEDPUMP-TRANSMISSION SUBASSEMBLY

5.1.1 Expander Design

5.1.1.1 Variable Cut-Off Intake Valving Systems

System performance analyses have indicated that load control by variable inlet valve timing, as opposed to throttling, is a very desirable feature for a Rankine-cycle automotive powerplant because of the higher part-load efficiency and higher wide-open-throttle performance which can be achieved. During the initial conceptual design study,[1] both mechanical and hydraulic schemes were derived conceptually to accomplish variable timing. During the second phase of the program reported here, the most promising valve concepts were analyzed in more detail and one approach was selected for experimental bench testing. Work carried out under this program was concentrated on (1) analysis of a mechanical or cam-driven approach with two inlet valves in series, and (2) analysis and bench-testing of a hydraulic approach by the American Bosch Company of Springfield, Massachusetts. In addition, another hydraulic approach is under experimental investigation at the British Internal Combustion Engine Research Institute in England, and is financed entirely by Thermo Electron Corporation.

For the mechanical and hydraulic schemes to be comparable, the valve sizes and motions would have to be such that they both would give the same power and efficiency when installed in a given expander operating at the design point conditions. A computer program was developed to calculate an expander cycle, given operating characteristics such as bore, stroke, speed, inlet pressure and temperature, etc. Instantaneous inlet valve area as a function of expander crank angle is also an input variable. The output of this program includes a pressure-volume diagram, flow rate, power output and expander cycle efficiency. Since the mechanical and hydraulic valves have different actuating mechanisms and, therefore, different opening and closing rates, the size and lift of the two valving approaches must be different to give the same expander performance.

The maximum flow area of the hydraulically actuated valve had initially been estimated at 1.20 in^2, and this size was used to calculate the valve mass and response characteristics in the studies on this system performed by American Bosch under subcontract from Thermo Electron. Once the size of the valve was chosen, the response for various servo pressures was experimentally determined and the power to drive the valve gear calculated. This information was then used to conduct an analytical study of expander performance as a function of inlet valve size, servo pressure, bore, stroke, and speed. The results of this study determined the characteristics of the expander for 147 gross shaft horsepower output as given in Table 5.1.

These characteristics represent a reasonable optimum in the trade-off between expander size (as well as the size of other components of the system) and expander efficiency at the design point condition. Increasing valve size and/or servo pressure improves the expander indicator diagram performance but the improvement is nullified by the additional power required to drive the valves. Attempts to use a higher design point rotational speed in order to reduce the expander size and the overall system size were not successful with the current intake valving systems, since intake valve throttling losses increase above 1800 RPM. A higher intake ratio is then required to maintain a given power level and the expander efficiency is reduced requiring a larger boiler and condenser as well as other components of the system.

TABLE 5.1

EXPANDER CHARACTERISTICS
WITH
HYDRAULICALLY ACTUATED INTAKE VALVES

Number of cylinders	4
Bore	4.42 inches
Stroke	3.00 inches
Total Displacement	184 in^3
Maximum Average Piston Speed	900 ft/min
Corresponding Maximum Speed	1800 rpm
Inlet Valve Size	1.25 inch diameter x 0.3 inch lift
Hydraulic Pressure	1500 psia
Intake Ratio	0.175

The mechanical, two valves-in-series design had to provide the same expander performance as the hydraulic design in order for the comparison to be valid. The principle of the two valves-in-series approach is shown in Figure 5.1. The valve operated by cam No. 1 operates with fixed timing whereas the timing of the valve operated by cam No. 2 is varied. The overlap of the two valve events determines the effective intake ratio of the expander. It is obvious from the valve event diagram shown that each valve of the two valves in series must have an appreciably larger flow area than the single hydraulic valve, since the full lift of the cam is not utilized at the short intake ratio (0. 175) at the design point. The final configuration taken by the two valves in series is discussed in more detail in part b of this section, and the detailed design illustrated in Figures 5. 15, 5. 16 and 5. 17. The size shown in Figures 5. 15 gives the same performance as the hydraulically-actuated valve. Figure 5. 2 shows valve area as a function of crank angle for the two systems at the design point conditions. Figure 5. 3 shows the instantaneous flow rate through the two valves, and Figure 5. 4 is an expander P-V diagram for the two valving schemes at the design point condition. It can be seen from these results that the two schemes give very nearly equal performance at the design point. No off-design runs were made comparing the two systems, but one would expect that the hydraulically-actuated valve would show an advantage, since its lift vs. crank angle diagram approaches a square wave as expander speed is decreased; the two valves-in-series profile remains the same for a given valve timing as speed decreases.

a. American Bosch Hydraulically Actuated Inlet Valve: The valve actuating mechanism is illustrated in Figure 5. 5 and consists of the following basic elements: A double-spool servo valve, a spool check valve, the driving plunger and piston, and a solenoid or magnetic actuator (not shown on Figure 5. 5).

Referring to Figure 5. 5, the right-hand sketch shows the valve in the closed position. High pressure fluid is applied to the top of

(5a)

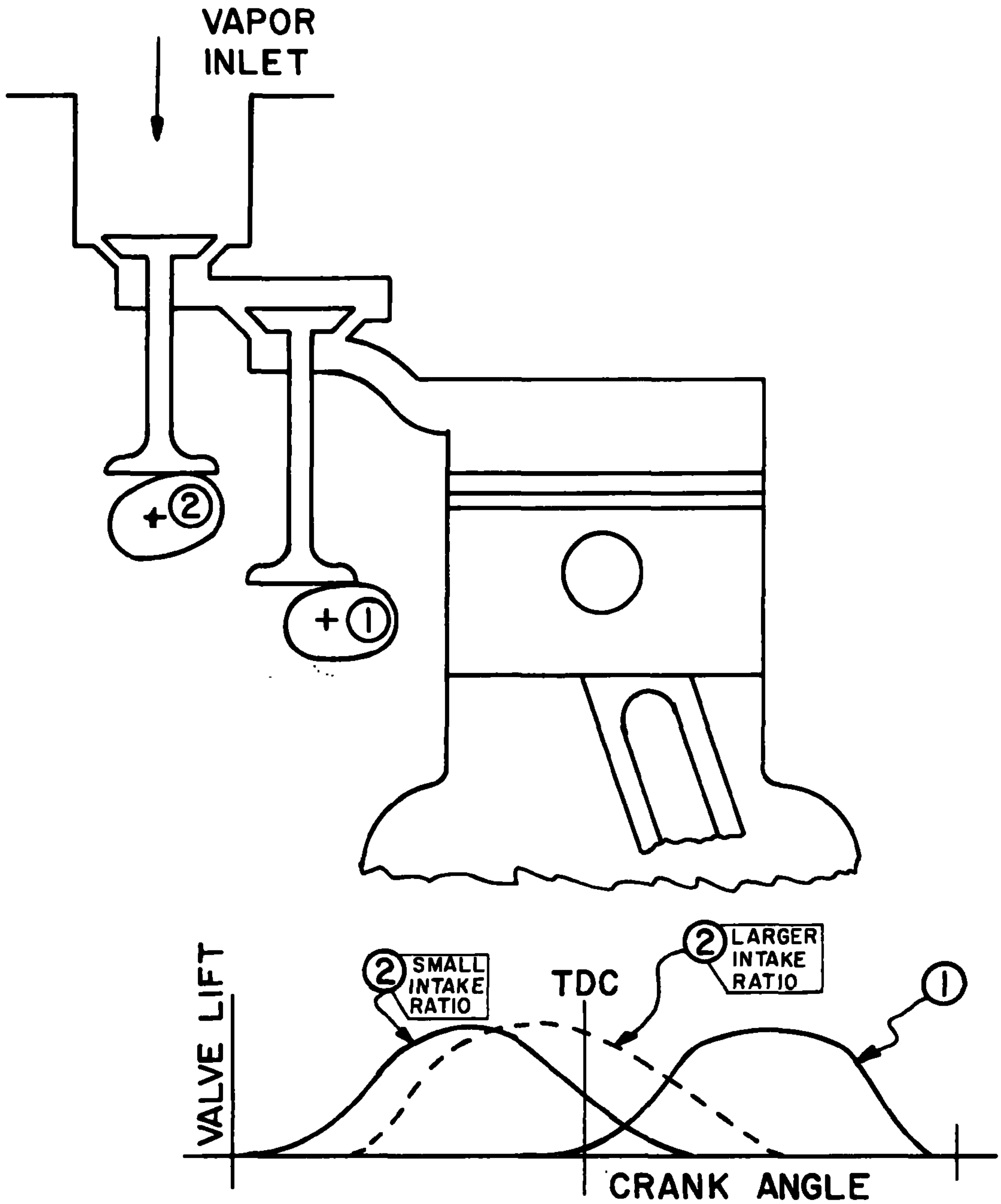

Figure 5. 1 Two Inlet Valves in Series,
Variable Cut-Off Mechanism.

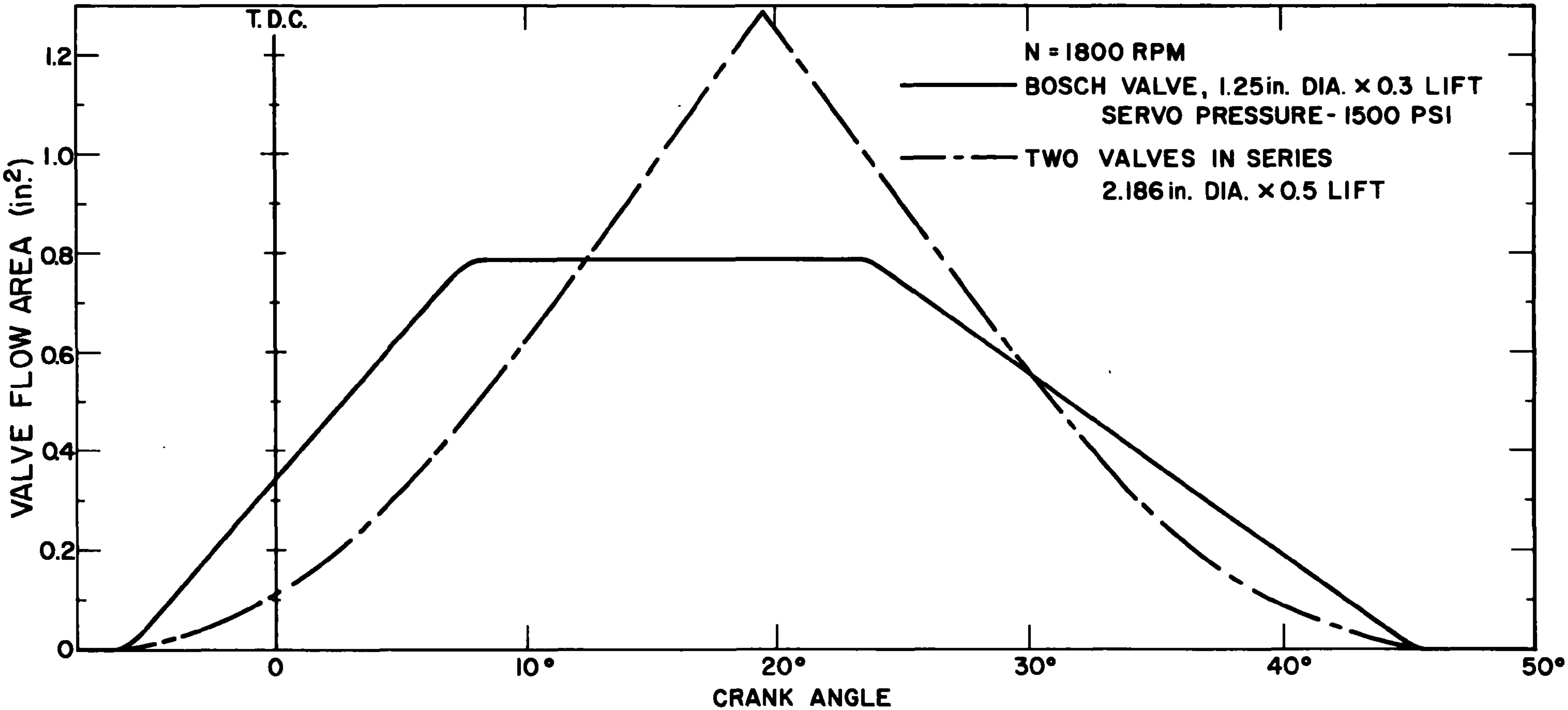

Figure 5.2 Valve Flow Area as a Function of Crank Angle at Design Point Conditions.

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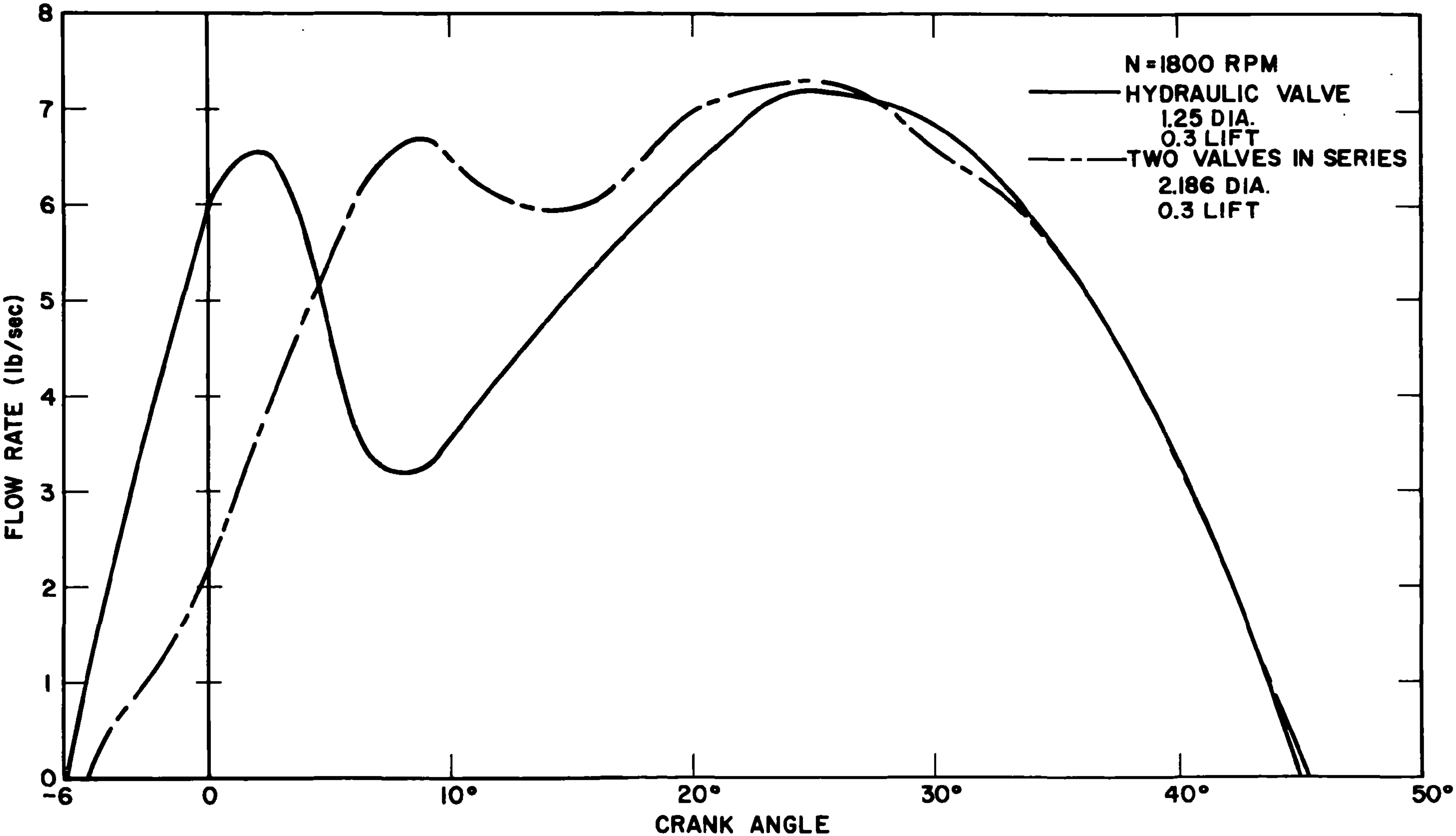

Figure 5.3 Flow Rate Through Intake Valve versus Crank Angle at Design Point Conditions.

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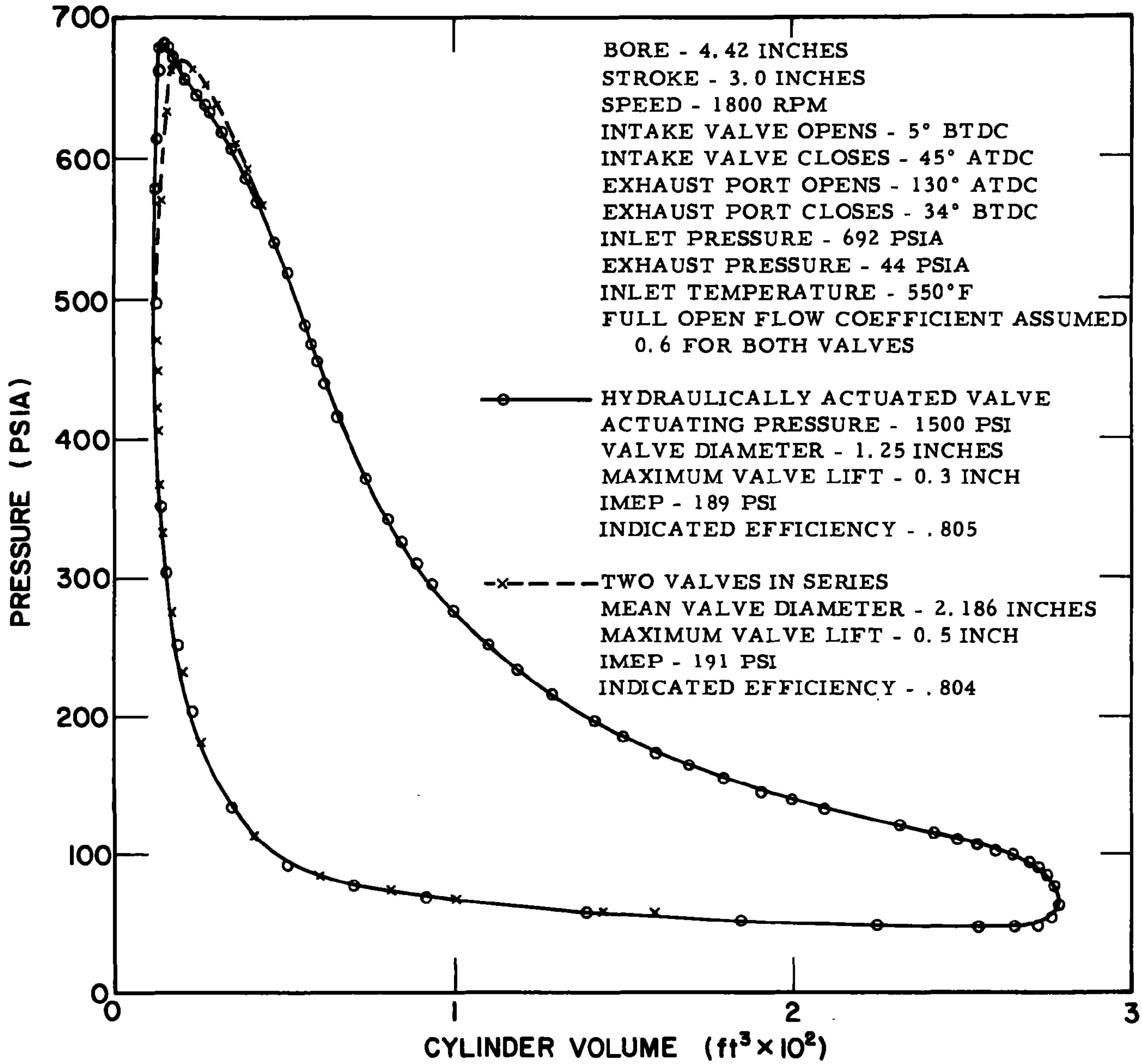

Figure 5.4 Expander P-V Diagram at Design Point Conditions.

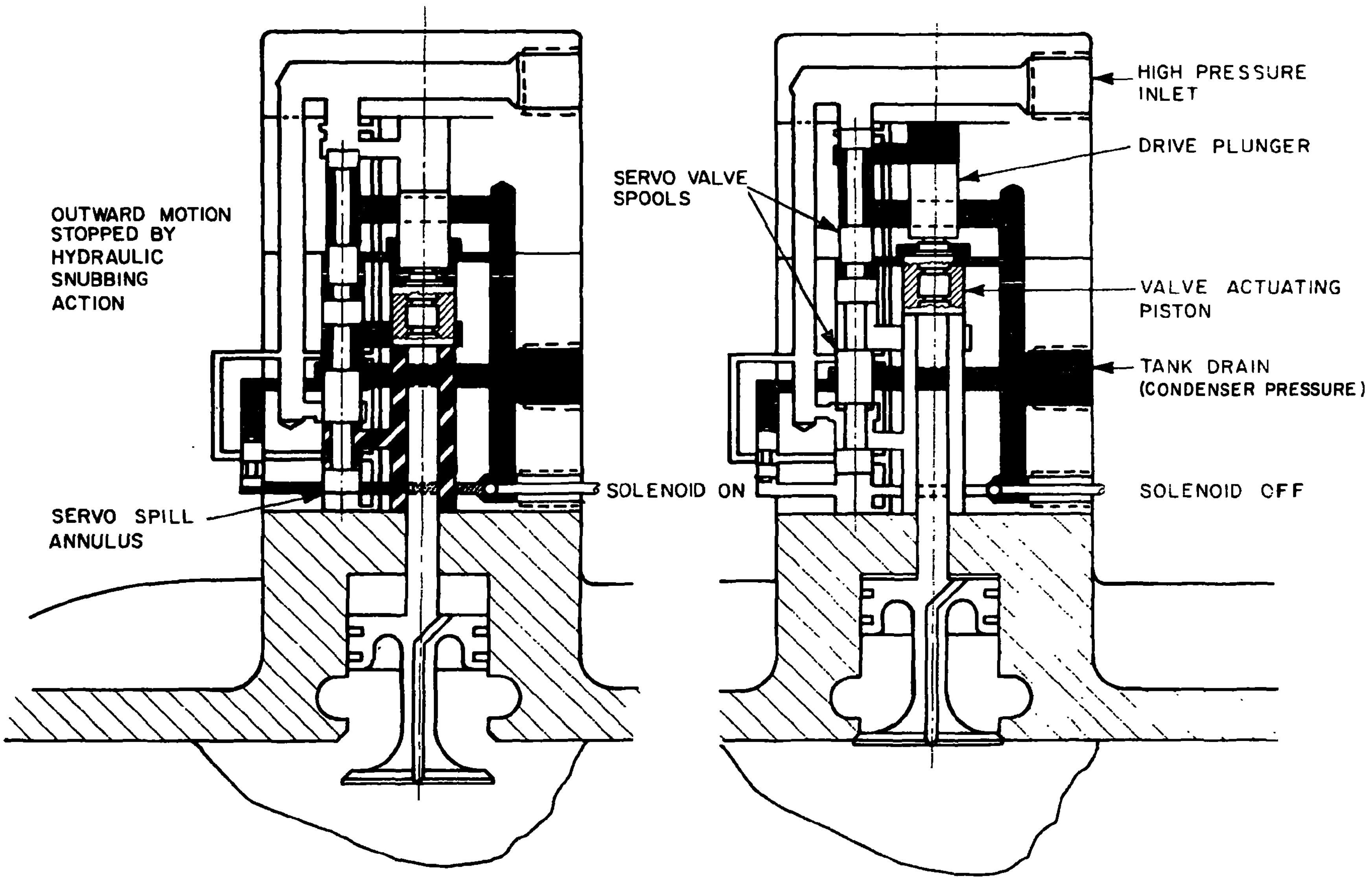

Figure 5. 5 Servo Actuated Inlet Valve (American Bosch).

I-1644

the small spool of the servo valve. A bleed line equipped with an orifice and a spool-type check valve is provided across the two spools of the servo valve. This line produces pressure force equalization across the spools when the solenoid valve is closed. The servo spools are thus maintained in their closed position by the force differential generated by the difference in spool diameters. The high pressure fluid is ducted through the servo valve passages and applied to the underside of the valve actuating piston. This fluid pressure holds the engine valve in its closed position.

Refer now to the left-hand sketch. When the solenoid actuated valve is opened, the fluid pressure under the large spool valve is sharply reduced and a large pressure drop occurs across the two spools which causes them to move downward and the spool check valve to close by spring action. When the servo spools have moved to the point where the spill annulus is closed off (position shown in sketch), the pressure under the large spool increases sufficiently to stop the motion of the spools. Since the bleed line is open, the pressure under the spools will increase, and they will start to move upward. Motion in this direction, however, will cause the spill annulus to be reopened slightly, which re-establishes flow through the solenoid control valve. When this flow exactly matches the flow through the bleed orifice, the spools become stabilized in their "open" position.

With the servo valve spools in their "open" position, the area under the valve actuating pistion is switched from high pressure to "tank" pressure ($\simeq$ condenser pressure) and the area above the drive plunger is switched from "tank" pressure to high pressure. The resultant high pressure drop across the drive plunger and valve actuating piston drives the intake valve open. When the valve actuating piston moves to the point where its spill annulus is closed off (position shown in sketch), the pressure under the piston increases sufficiently to stop the motion of the valve. This "snubbing" pressure is indicated by the dot-dash area. The engine valve will remain open as long as the solenoid valve is energized.

When the solenoid valve is closed, flow through the bleed line causes the pressure under the large servo spool to increase. This forces the spool check valve upward, thus opening a large parallel feed passage under the servo spools for fast response. The servo spools are driven upward to the point where the top end of the small spool closes off the bleed line annulus (position shown in right-hand sketch of Figure 5.5). Further slight upward motion causes the pressure under the servo spools to decrease to a value which will achieve force balance, at which point all motion will cease and the servo spools will be in their "closed" position. If in this condition there is leakage into the bleed line, the servo spools will move slowly upward until they contact a mechanical stop (not shown). During operation the spools will never contact the stop, due to insufficient time for this leakage to occur. When the servo valve is in this closed position, the area above the drive plunger will be switched from high pressure to "tank" pressure, and the area below the valve actuating piston will be switched from "tank" pressure to high pressure. The resultant pressure drop across these members will now drive and hold the engine valve closed.

Control of the valve actuating mechanism described herein is accomplished by a speed-sensitive phototransistor timer which energizes the capacitor discharge circuit that feeds the solenoid at the proper time in the expander cycle. A circuit is also required to turn this signal off at the proper time to control the length of time that the valve is open. The timer is equipped with a speed-sensitive automatic advance mechanism which maintains the valve opening at the optimum point relative to crank angle as a function of speed. Also built into the valve duration control is an electronic speed sensor which is used to limit the maximum intake ratio as a function of speed, so that the expander cannot overdraw the capacity of the boiler. The control schematic is shown in Figure 5.6 along with the other components required for operating the valves. Note that the hydraulic reservoir for the valving is separate from the lube oil sump in the crankcase. The lube oil pump provides only enough oil for make-up of leakage out of the valve system. Leakage occurs only down the valve stem and around the vane pump and is quite small. A separate reservoir is used to insure that the oil used for the valving system is free of Fluorinol-85, which may be present in the crankcase, particularly during start-up.

The vane pump is of the variable displacement type with a speed sensing system to control its output pressure as a function of speed, as shown in Figure 5.6. Use of a variable displacement pump minimizes the power required to operate the intake valves. Fast valve response is required only at higher expander speeds and the servo pressure can therefore be allowed to drop at lower expander speeds from 1500 to 800 psi without sacrificing performance but saving almost half the power.

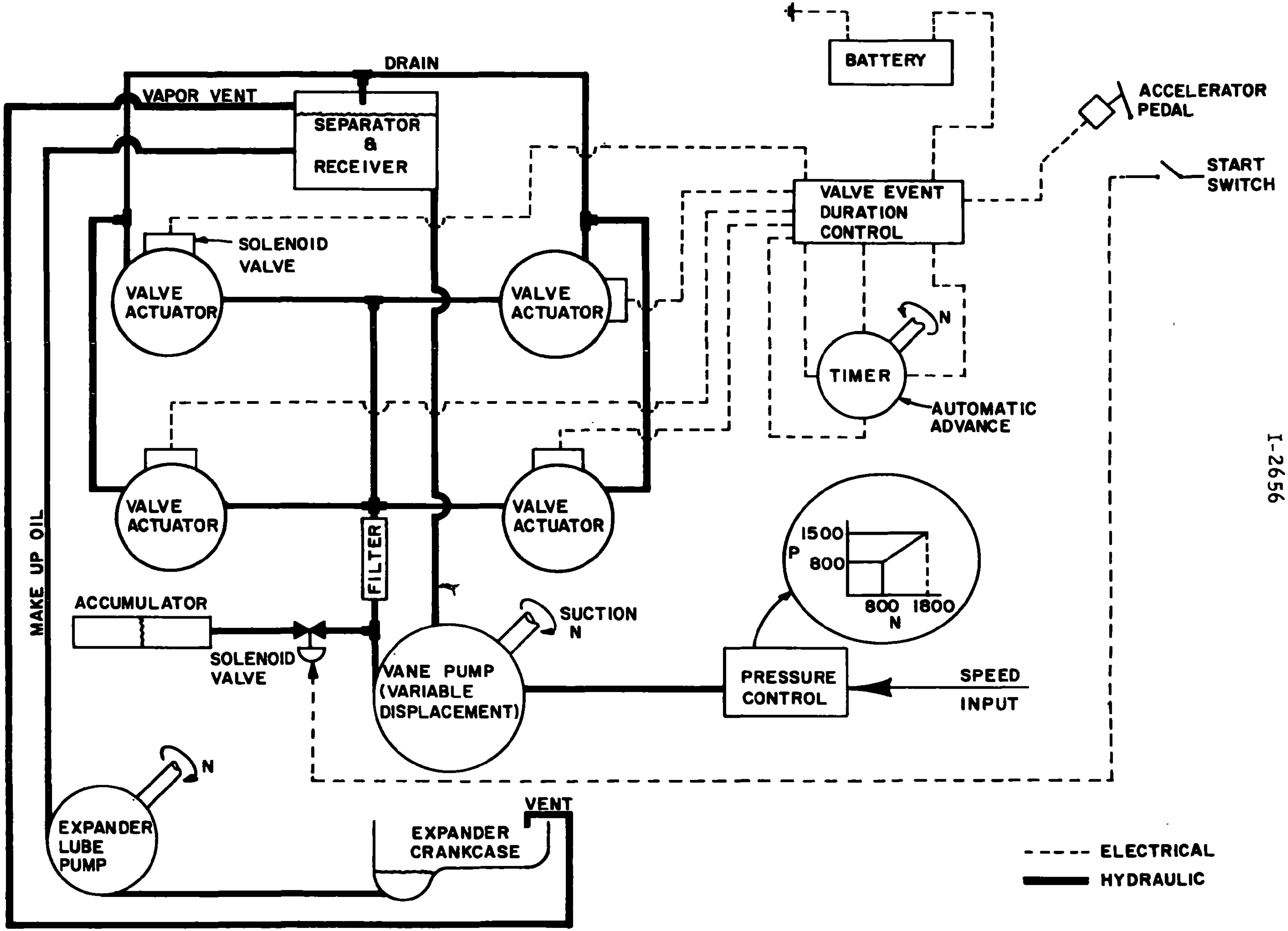

Figure 5.6 Schematic of Control and Hydraulic System for American Bosch Intake Valves.

I-2656

An accumulator is required to provide the high instantaneous flow rates during the stroking of the inlet valve.

The system is equipped with a solenoid valve which holds pressure in the accumulator during shutdown to provide fast system cold startup. During startup, this pressure closes any expander intake valve that might have opened due to leakage during the period the expander is shut down. The closed valves permit faster buildup of boiler pressure and temperature during startup cranking, since no vapor is passed through the expander until the boiler pressure reaches a predetermined level. Operation of a 5 hp system at Thermo Electron has demonstrated that the procedure results in a significant reduction in the cold startup time. During the initial evaluation of the hydraulically-actuated valve, the American Bosch Company performed a design analysis of the actuator illustrated in Figure 5.5. The object of this study was to size the various actuator and servo spools, determine the servo pressure required and calculate the response time of the valve. Table 5.1 summarizes the results of this study. During the second phase of the subcontract with American Bosch, a complete actuator was built and its response was measured in bench-testing. A photograph of the actuator tested is presented in Figure 5.7. The valve head was not built with a seat and balancing piston, but was fitted with a weight to simulate the mass of the real valve (0.3 lbs). The actuator was tested in air; no attempt was made to simulate the pressure loads or thermal situations which might occur in a real expander. The primary object of this work was to see if the actuator could be made to operate the valve with the proper time response.

The opening and closing time response of the experimental unit was measured as a function of servo pressure with the results presented in Table 5.3 and Figure 5.8.

TABLE 5.2

DESIGN CHARACTERISTICS OF
AMERICAN BOSCH HYDRAULIC ACTUATOR VALVE

Supply Pressure	- Up to 2000 psi
Supply Flow	- 1.75 gpm/valve assembly at 2000 rpm
Valve Drive Piston Diameter	- 0.650 inch
Valve Return Piston Diameter	- 0.695 inch
Valve Stroke	- 0.3 inch
Valve Weight	- 0.3 lb
Stroke Time	- 0.001 second
Required Force	- 466 lb
Flow Rate During Stroke	- 5.78×10^{-2} ft^3/sec (26 gpm)
Servo Valve Stroke	- 0.200 inch
Upper Servo (Small) Diameter	- 0.250 inch
Lower Servo (Large) Diameter	- 0.281 inch
Servo Stroke Time	- 0.001 second
Force to Drive Servo	- 30 lbs
Solenoid Valve Diameter	- 0.071 inch
Lower Servo Feed Diameter	- 0.086 inch minimum
Total Valve Event	- 0.003 second

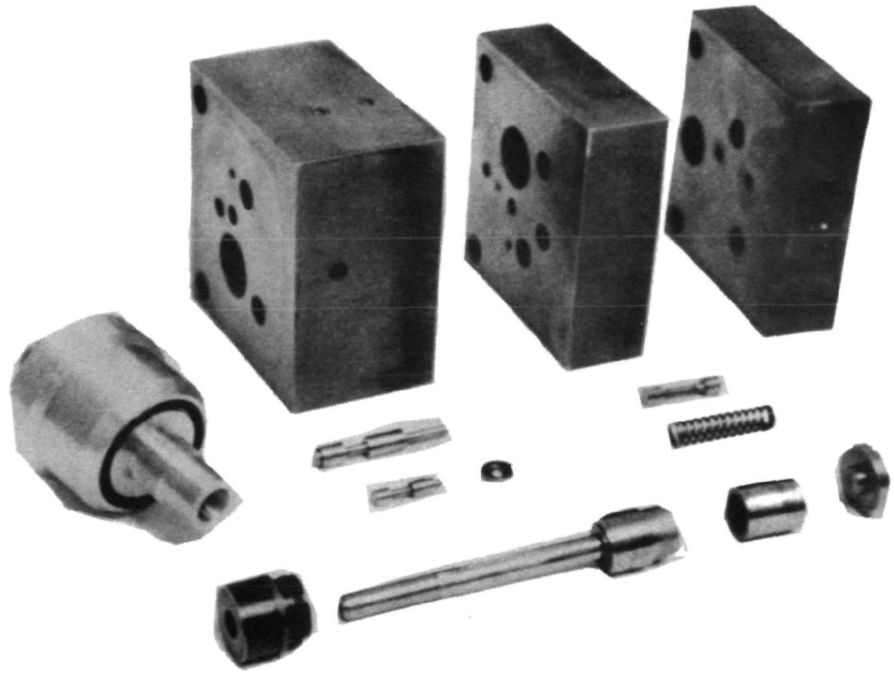

Figure 5.7 American Bosch Hydraulic Valve Actuator Used in Bench Testing.

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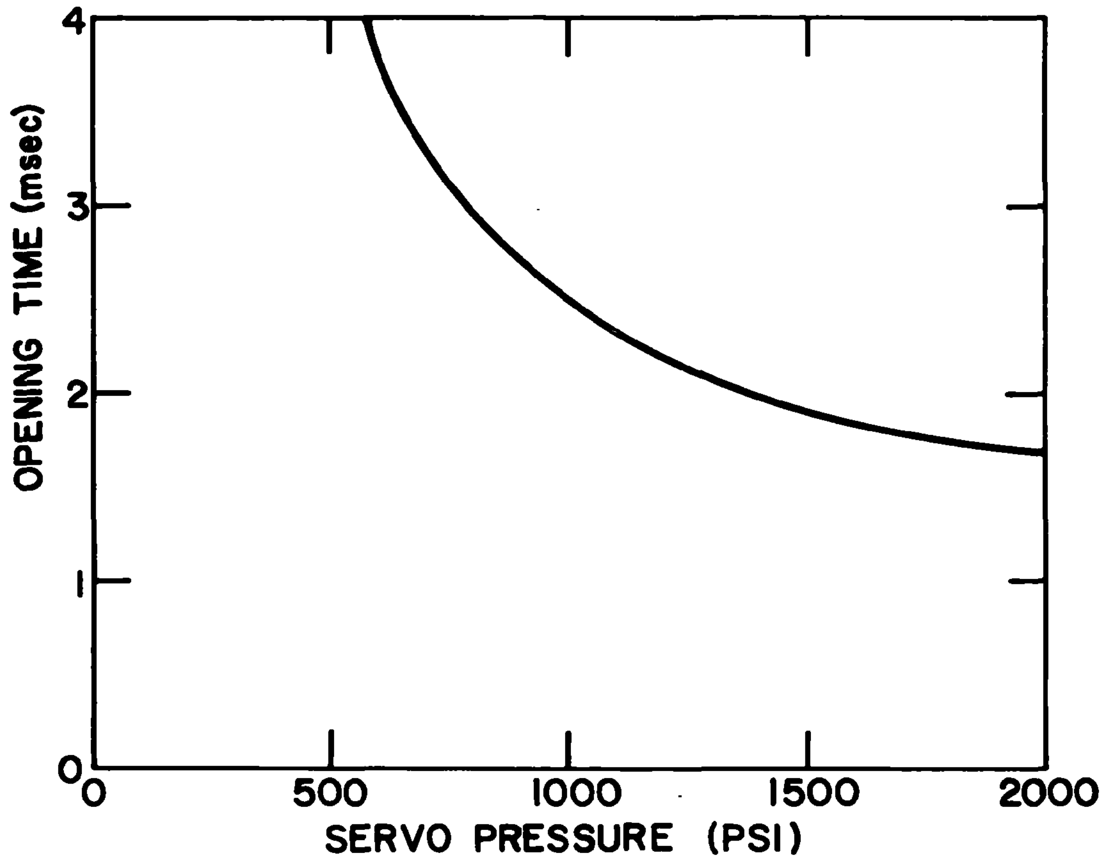


Figure 5. 8 Inlet Valve Opening Time versus Servo Pressure for Servo-Actuated Inlet Valve.

TABLE 5.3

MEASURED OPENING AND CLOSING TIMES
FOR AMERICAN BOSCH HYDRAULIC VALVE ACTUATOR

Servo Supply Pressure (Psi)	Time to Open (Millisecs)	Time to Close (Millisecs)
600	4.0	4.0
800	3.0	4.0
1000	2.3	3.5
1200	2.2	3.5
1500	2.0	3.0
2000	1.7	2.5

These data illustrate that the valve response time is levelling off with increasing supply pressure above about 1500 psi; the increased pressure is being used primarily to accelerate the fluid in and out of the mechanism rather than to move the valve.

Oscilloscope traces of the valve response were made with the different events occurring described in Figure 5.9. The effect of supply pressure on the valve motion is illustrated in Figures 5.10 and 5.11. These traces were used to determine the valve opening and closing times.

The part stroke performance of the valve was also measured with the results presented in Table 5.4. The corresponding valve motion traces are shown in Figure 5.12. The cycle time is 1.5 milliseconds greater than the sum of the opening and closing times because of a dwell at the top of each event, as illustrated in Figure 5.12. This dwell results from the deceleration-acceleration times

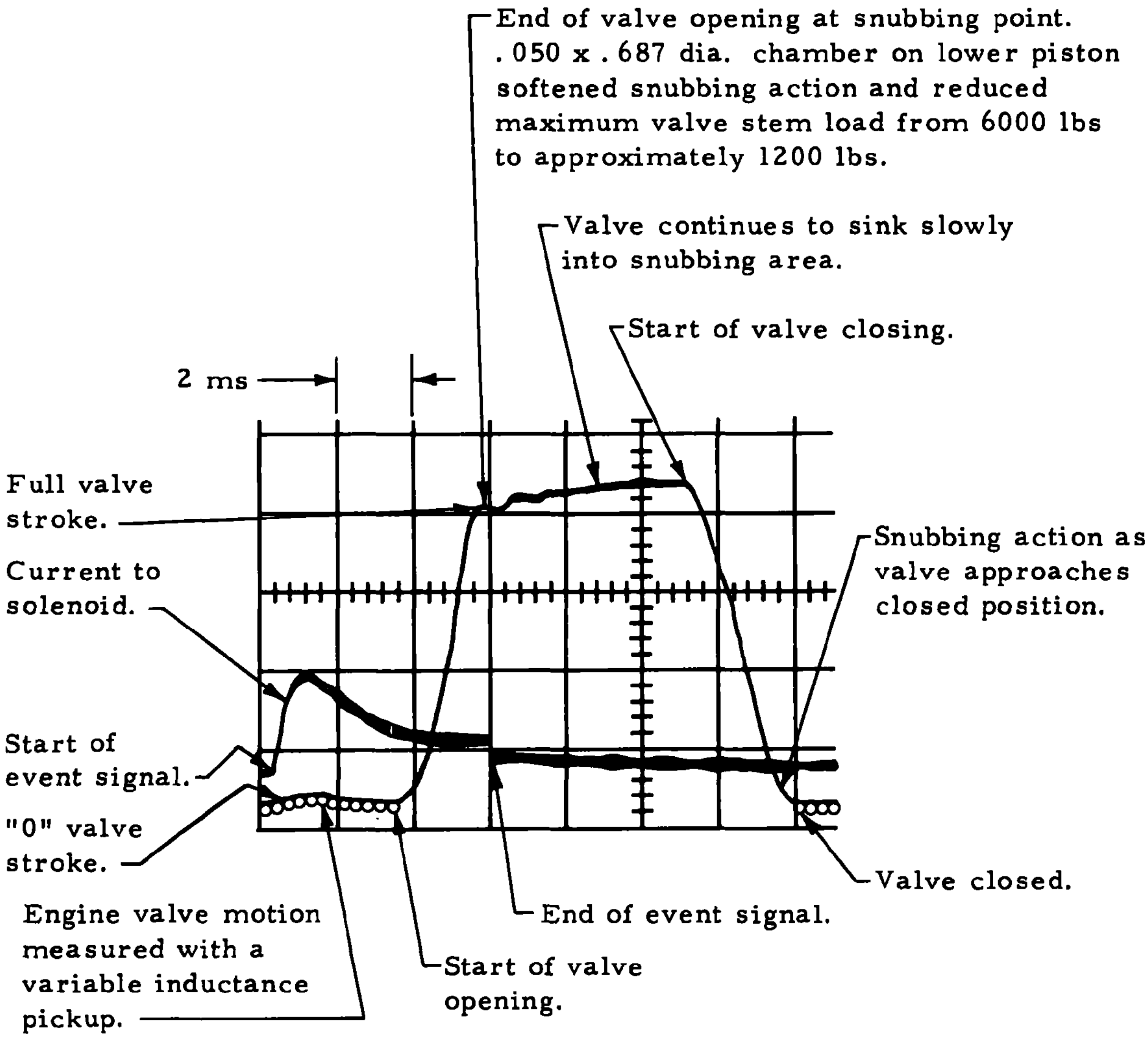


Figure 5. 9 Valve Opening and Closing Motion for Servo-Actuated Inlet Valve.

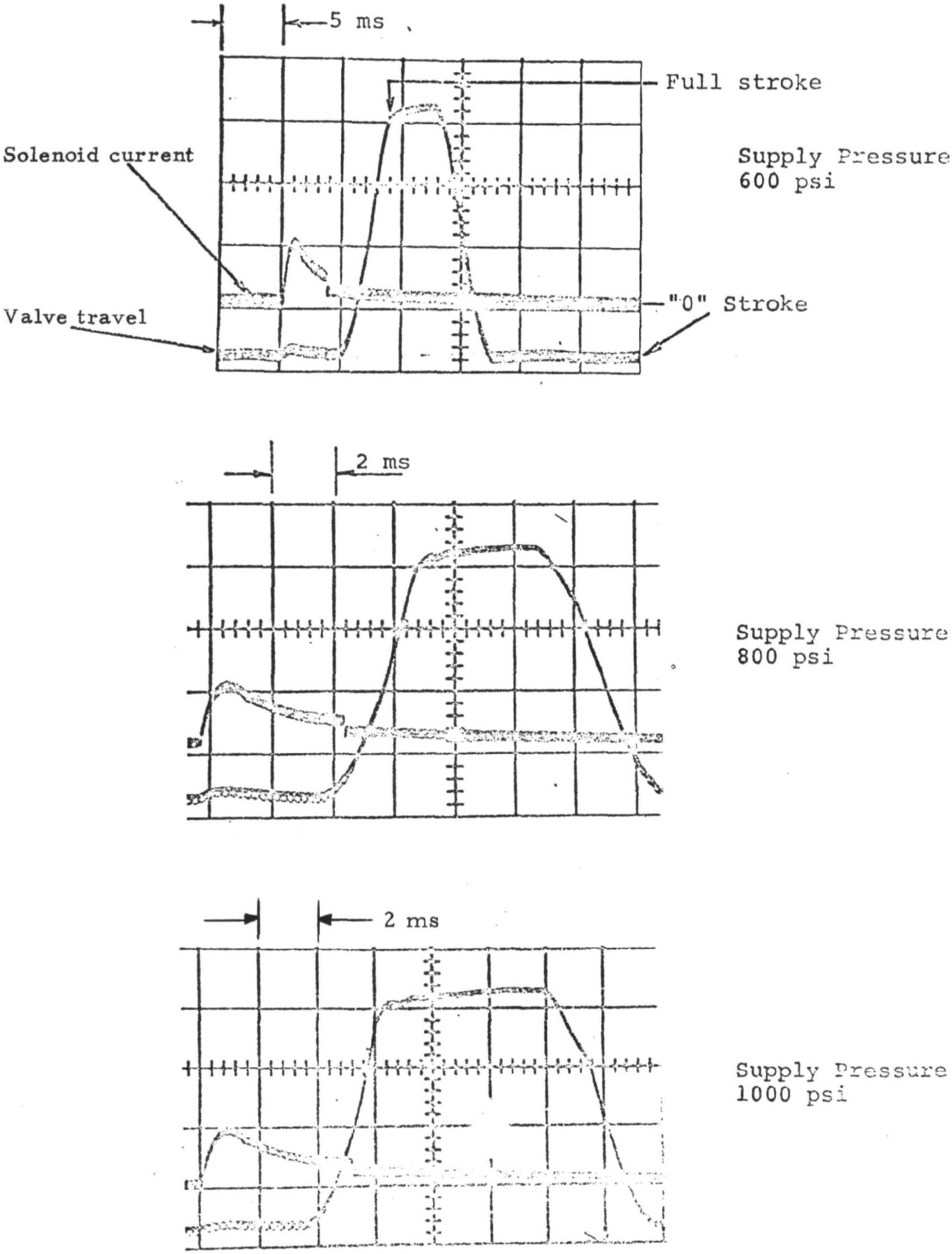


Figure 5.10 Effect of Supply Pressure on Valve Motion.

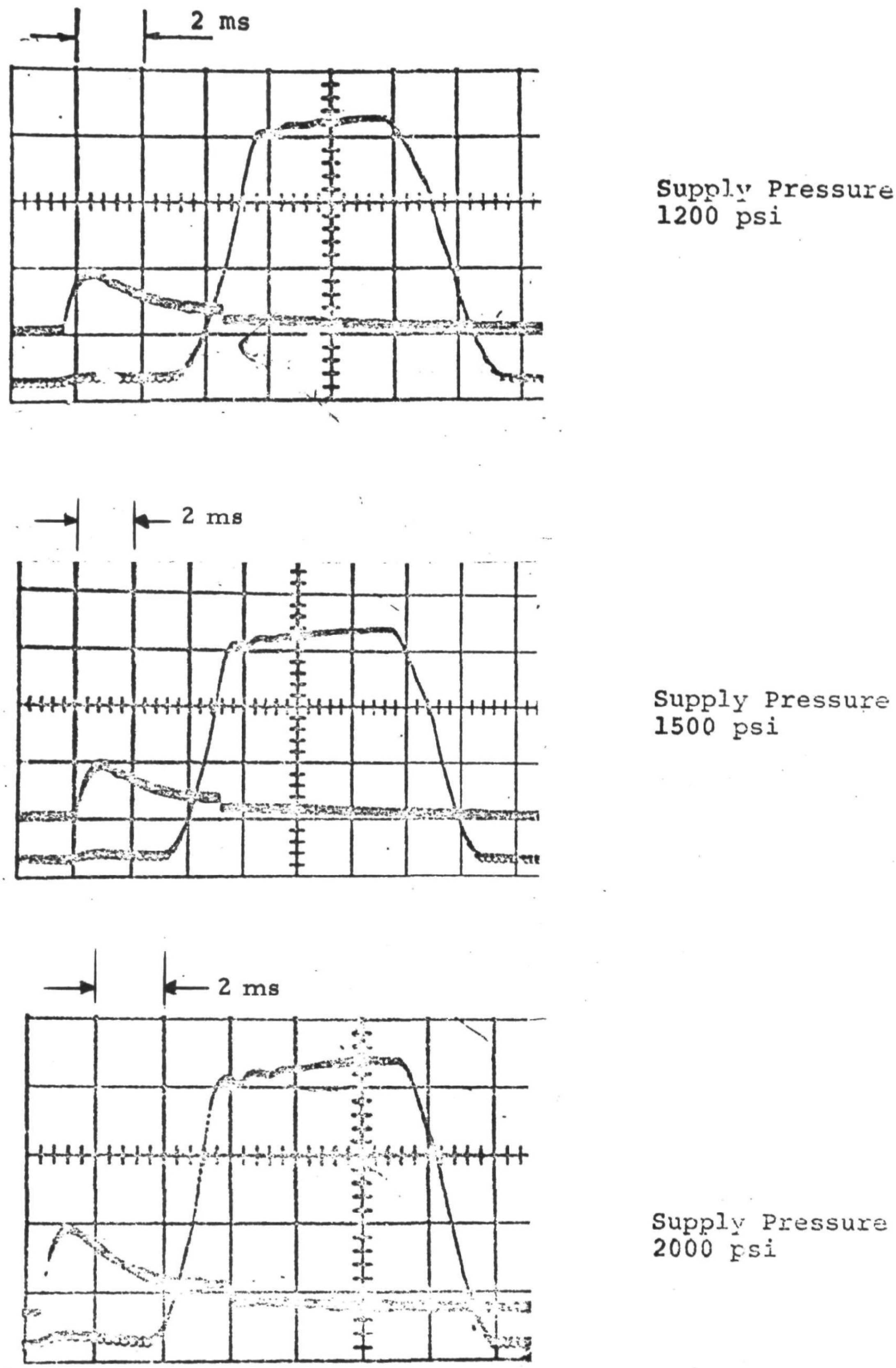
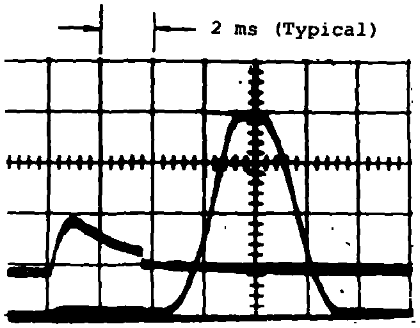


Figure 5. 11 Effect of Supply Pressure on Valve Motion.



Min. time for full stroke cycle.

Fig. 7

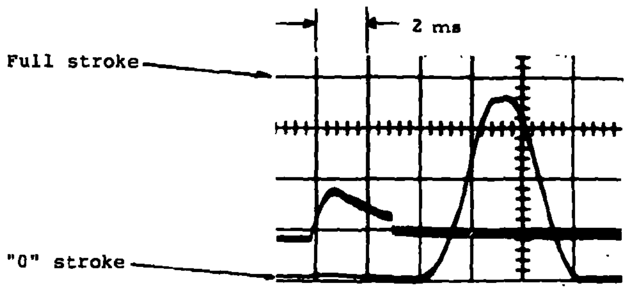


Fig. 8

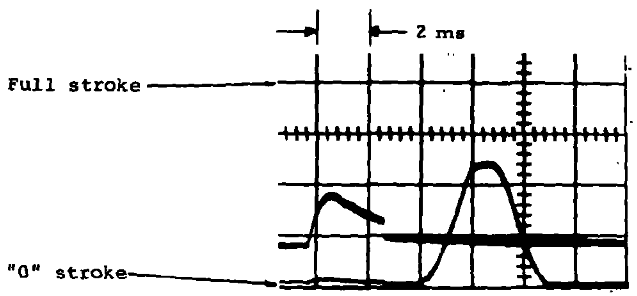


Fig. 9

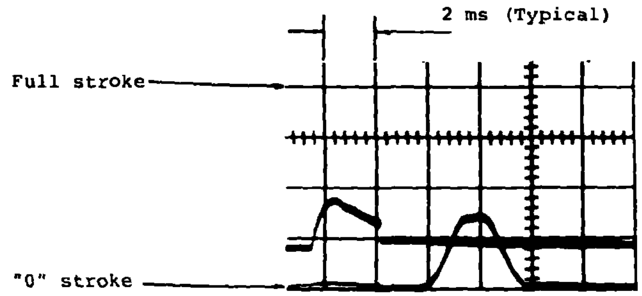


Fig. 10

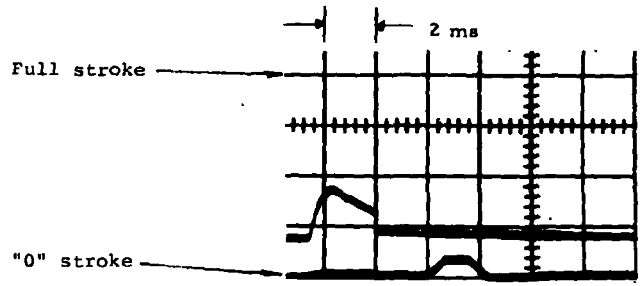
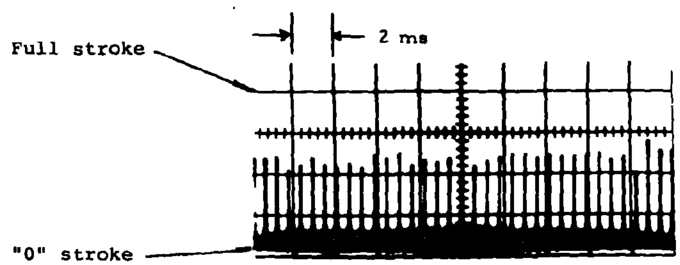


Fig. 11



Repeatability of part stroke.

Figure 5.12 Part-Stroke Performance of Servo-Actuated Valve.
Servo Pressure = 1500 psi.

for the valve. The valve motion can be modulated down to zero lift if desired. Some cycle-to-cycle variation occurs at less than full lift and is shown in the last trace of Figure 5.12. This variation is not sufficient to affect the expander performance. Valve motion measurements were made with and without the bypass piston in the servo feed line and with orifices from .027 inch to .040 inch diameter. The best combination was a .040 inch orifice with the bypass piston included. Figures 5.13 and 5.14 illustrate the results with the .027 inch and the .040 inch bleed orifice, respectively. Note that the delay to initiate closing after the end of the electrical signal is approximately 6 milliseconds with the .027 inch bleed, and is reduced to approximately 4.5 milliseconds with the .040 inch bleed. The opening delay was increased approximately 0.5 millisecond by this change.

TABLE 5.4

PART-STROKE PERFORMANCE
OF SERVO-ACTIVATED VALVE

Supply Pressure = 1500 psi	
Fraction of Full Stroke	Cycle Time (milliseconds)
Full	6.5
.9	5.5
.6	4.5
.35	3.5
.08	2.0

The valve displacement versus time curves for the various conditions shown here were curve fitted and the resulting expressions were used in the expander cycle simulation program. The valve flow area-crank angle curve is shown in Figure 5.2 for the design point.

This experimentally determined curve was used to arrive at the design point condition previously discussed. A peak supply pressure of 1500 psi was found to be adequate for reasonable performance. The power to operate the four (4) valves at the design point is estimated at 3.6 hydraulic horsepower.

The integration of the hydraulic actuator with the expander cylinder head is discussed in Section 5.1.1.3. The timing device and vane pump and controls have been designed and their integration with the expander is also discussed in Section 5.1.1.3.

b. Mechanically Operated Two Valves-in-Series: This concept in its simplest form with poppet valves is shown in Figure 5.1. The valve event is determined by the amount of overlap between cams No. 1 and No. 2. Cam No. 1 determines the beginning of the event and has fixed timing with respect to the expander crankshaft; cam No. 2 determines the end of the valve event and must have variable timing with respect to the crankshaft in order to effect variable cut-off. Since valve No. 2 determines cut-off, all of the volume from this valve to the cylinder is clearance volume. The roles of the two valves cannot be interchanged, because if valve No. 1 were the "cut-off valve," it would open somewhere on the exhaust stroke of the expander, releasing the volume of high pressure vapor between the two valves directly to exhaust. This would constitute an unacceptable efficiency penalty.

The unavoidable clearance volume resulting from use of poppet valves as illustrated in Figure 5.1 results in the undesirable characteristics. If adequate flow areas are used for a reasonable pressure drop, the clearance volume is sufficiently large to seriously degrade

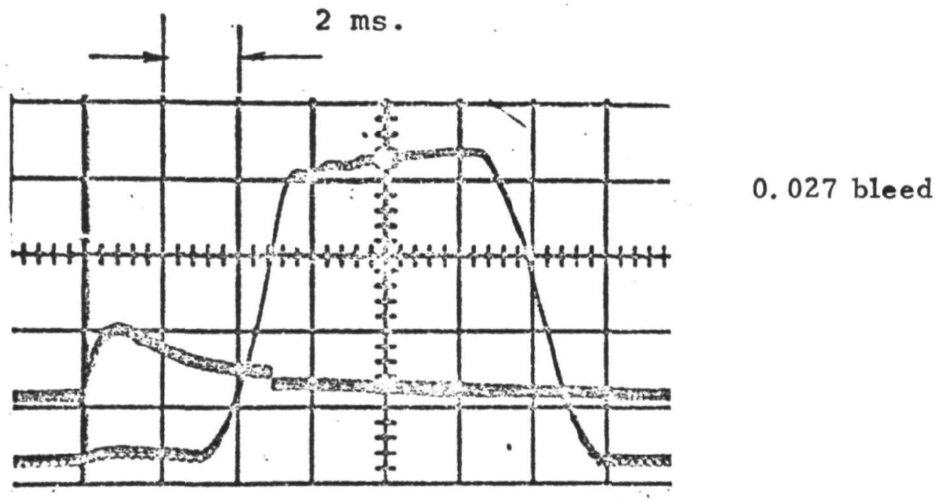


Figure 5. 13 Valve Motion with 0. 027 inch Bleed Orifice and Bypass Piston.

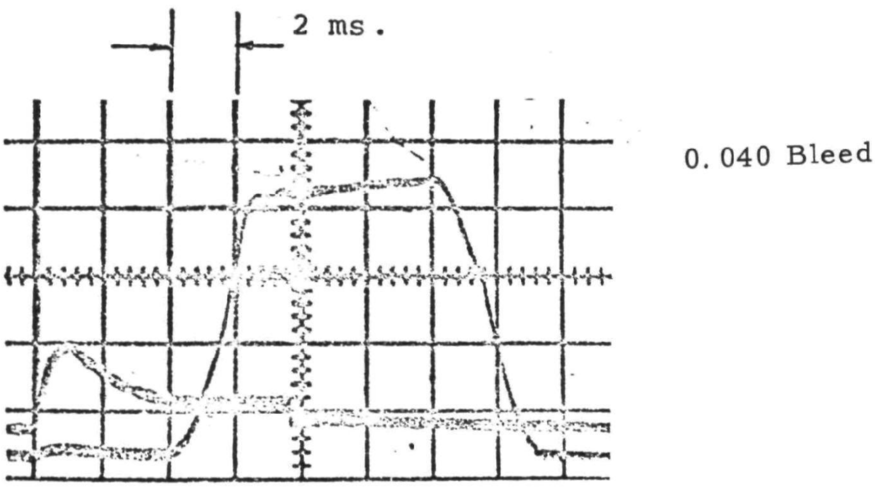


Figure 5. 14 Valve Motion with 0. 040 inch Bleed Orifice and Bypass Piston.

the expander efficiency. In addition, the minimum expander ratio is the volume between the two valves. For adequate flow area, this minimum intake ratio is large relative to that required for idle and light load conditions. A throttle valve would thus be required between the boiler and expander to permit operation of the expander under these light-load conditions.

In addition to these problems, the poppet valves are pressure unbalanced. The pressure forces and resultant cam loading is excessive and renders the use of poppet valves as illustrated in Figure 5.1 unfeasible.

An extensive evaluation of various approaches to the two-valve-in-series concept was carried out in both the initial and current programs with EPA in order to eliminate or alleviate these problems. This evaluation resulted in the use of concentric and annular valves as illustrated in Figures 5.15 and 5.16. The inner valve is operated with fixed timing (cam No. 1) and the outer valve with variable timing (cam No. 2), so that the function of the valves is identical to that illustrated in Figure 5.1. Use of the concentric valves eliminates the volume trapped between the two valves so that the intake ratio can be reduced to zero. The clearance volume is also reduced to a level small enough so that no significant effect on expander efficiency occurs. Pressure imbalance forces are reduced by use of ports in the transition section from the valve stem to the valve sleeve thereby equalizing the pressure across the valve. Pressure forces are thus reduced to those due to the valve stem and due to any pressure differences on the valve sleeve. A detailed analysis of the mechanical valve design presented in Figure 5.15 is given in Appendix I; this analysis demonstrates that all stress levels, including cam stresses, are acceptable. The maximum contact stress between the cam and follower is 198,400 psi necessitating a roller follower. Seal rings are used to reduce leakage of vapor by the valve sleeves.

The differential gear arrangement for changing phase between the cam shafts for the variable intake function is illustrated in Figure 5.17.

(i) Valve Opening Force. Due to the finite size of the valve seat, a pressure force unbalance occurs before the valve lifts, and a relatively high force is required to initiate valve movement. Figure 5.18 illustrates the unbalanced pressure force situation when the valves are both seated and unseated. (See Appendix I for pressure force evaluation.) To reduce this force and the stresses and vibration it produces, two steps were taken in the design of the valve gear. First, to reduce the magnitude of the force required to initiate valve movement, the thickness of the valve seat is made less than the .0930 inch thickness of the valves as illustrated in Figure 5.19. With the valve ends shaped as detailed, the unbalanced pressure force will be greatly reduced. As a second measure, the cam curve is designed so that the ramp will take up the lash in the system, and then initiate the movement of the valve off its seat while there is a low constant velocity in the valve train. This will reduce any jerk amplification of the valve lifting force and initiate the motion of the valve smoothly into the main cam event.

(ii) Cam Curve and Return Springs. Since the event is controlled by the opening of one valve and closing of the other, the sharper the opening and closing curves, the better the overall flow coefficients will be. The maximum positive acceleration is effectively limited by the

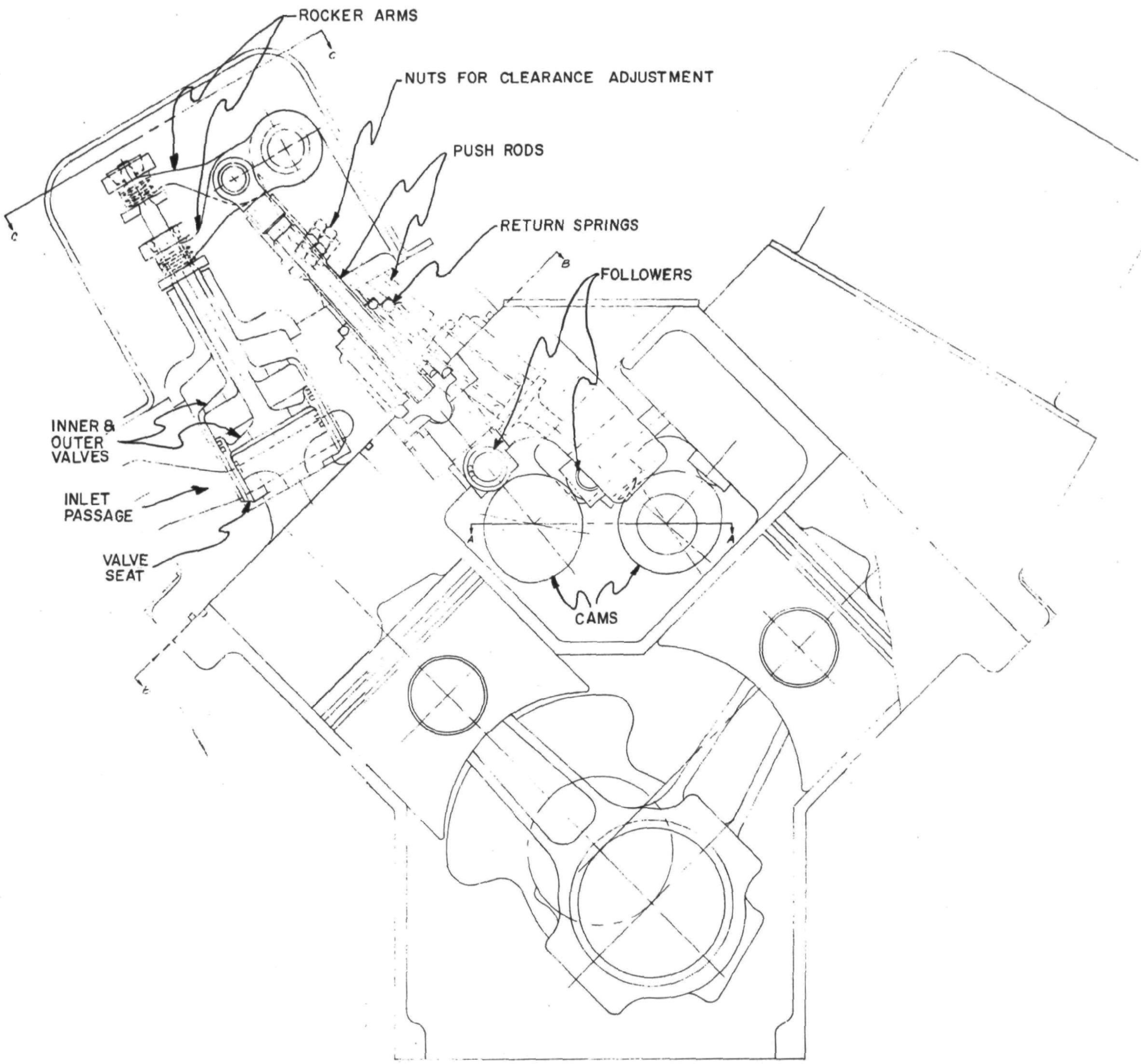


Figure 5.15 Mechanically-Driven Variable Cut-Off Intake Valve.

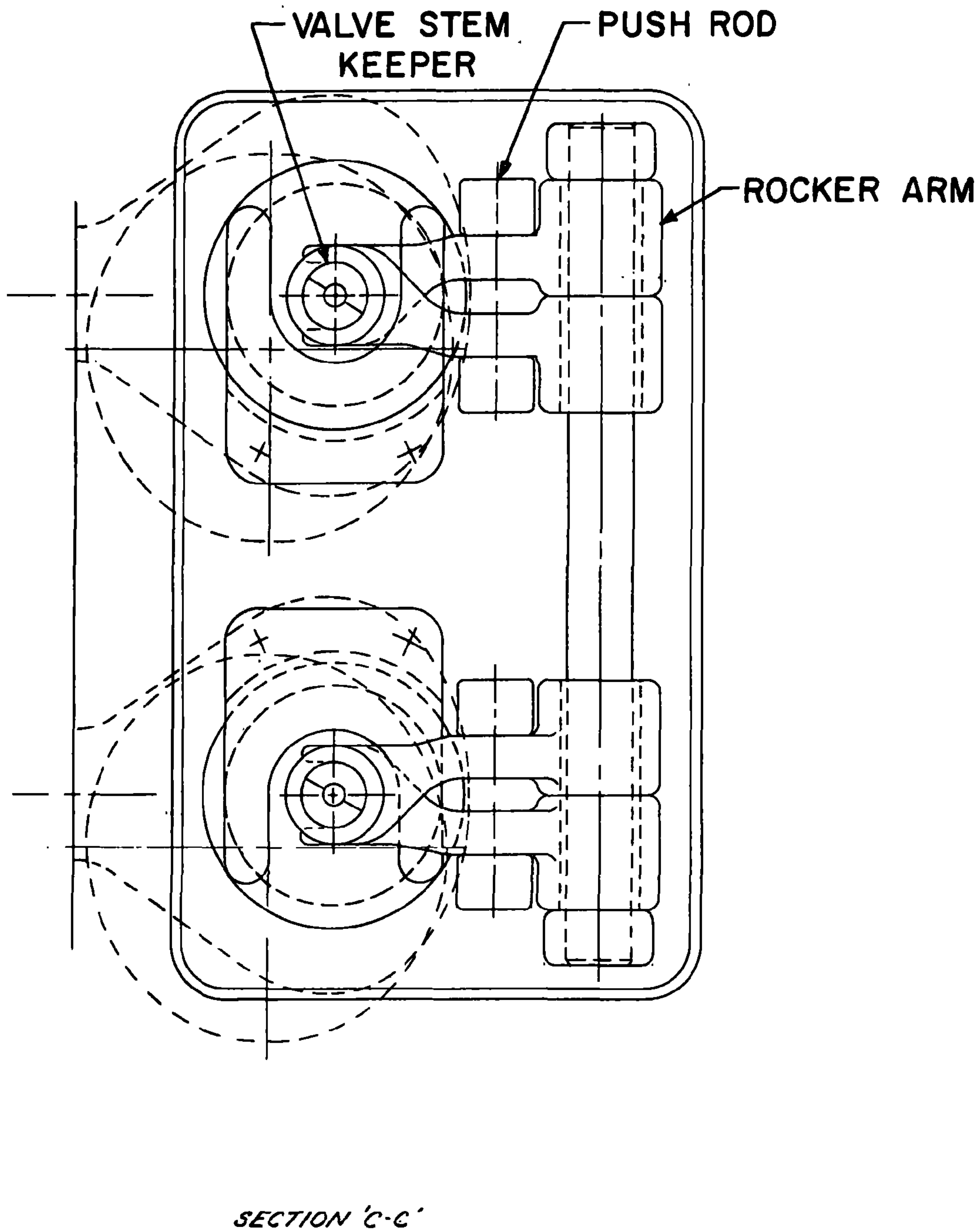


Figure 5. 16 Mechanically-Driven Variable Cut-off Intake Valve.

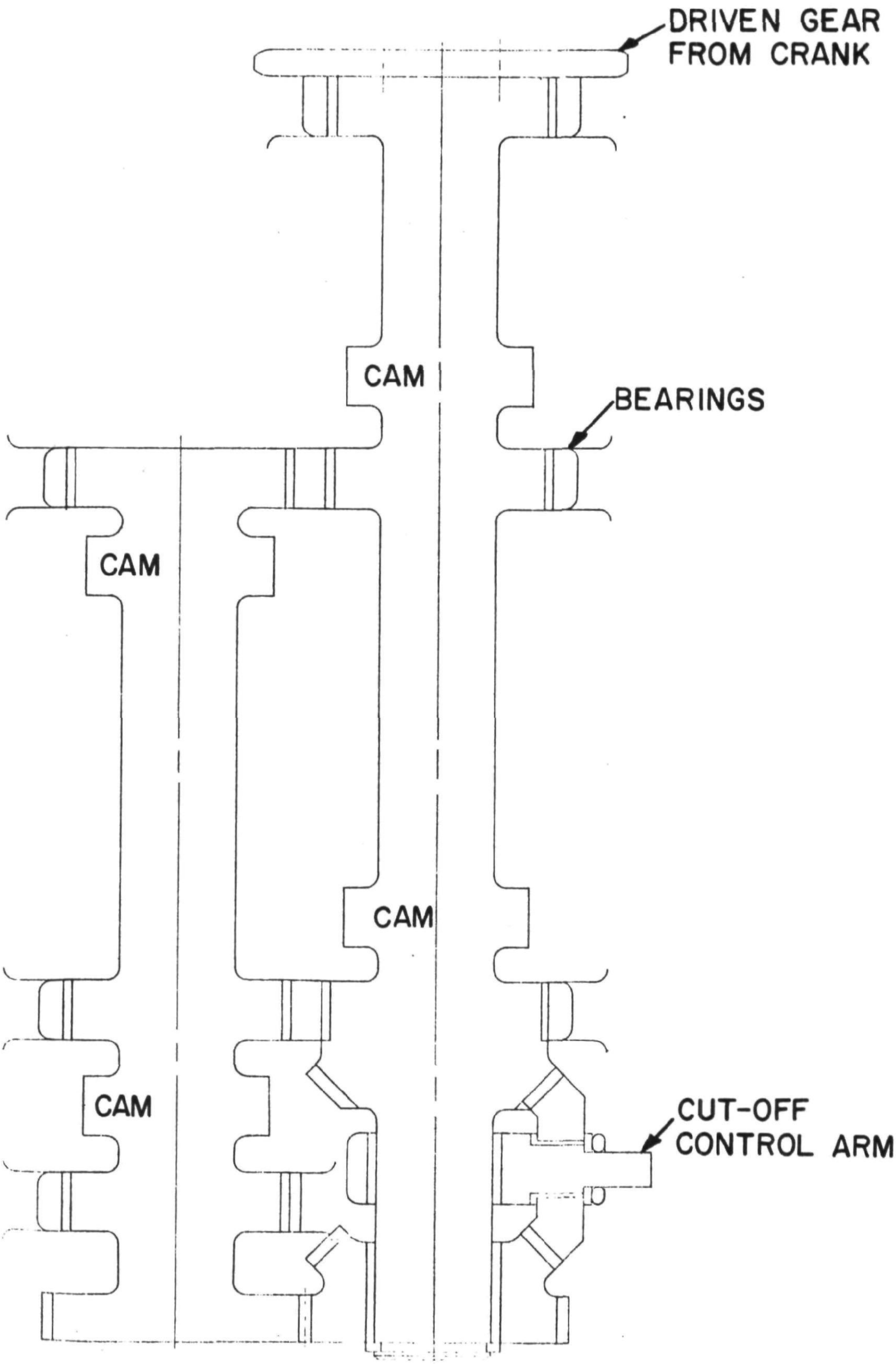


Figure 5.17 Differential Gear Arrangement for Mechanically-Driven Variable Cut-off Intake Valve.

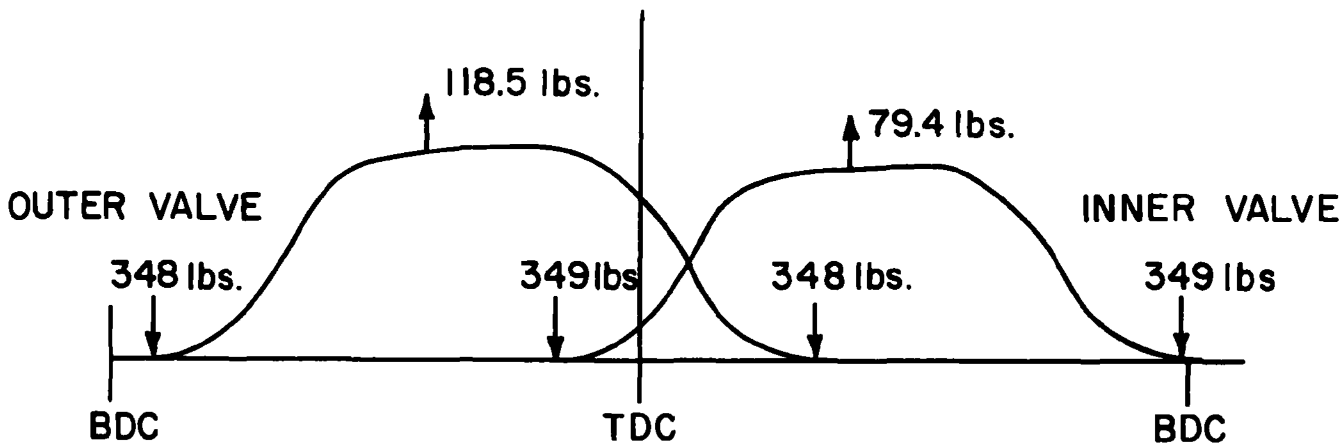


Figure 5.18 Pressure Forces on Sleeve Valves with 0.0930 inch Valve Seat Thickness.

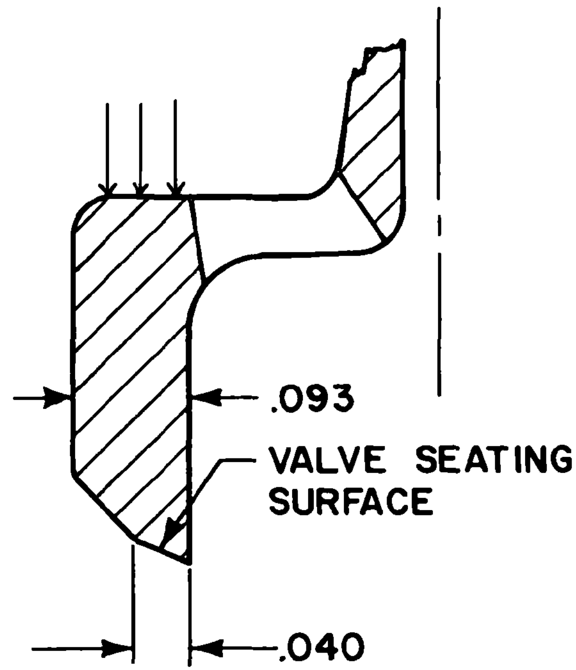


Figure 5.19 Valve Seat Profile to Reduce Opening Pressure Force on Valve.

requirement to keep the cam convex to minimize grinding cost. The minimum acceleration that can be obtained is limited by the magnitude of the spring force required to keep the cam and follower in contact. The resulting cam curve given in Figure 5.20 is a compromise between these factors. The cam velocity and cam acceleration in/degree and in/degree^2, respectively, are also presented in Figure 5.20. The cam characteristics are given in tabular form in Appendix I. Concentric coil springs are used in the design to obtain the required spring force. Consideration was also given to the use of torsion bar return springs mounted in the rocker arm pivots, but they are more difficult to package, although a higher spring force could be obtained by their use.

(iii) Valve Stem Leakage. An analysis was carried out to determine the magnitude of the leakage of working fluid past the valve stems. Laminar flow assumptions were used, and the effect of grooves in the valve stem was examined. Two predictions were made. The first assumed that the stem was concentric in the bore and the second assumed that the stem lay on one side of the bore. The second geometry gave a higher leakage rate.

The analysis showed that if the diametrical clearance is kept below ~ .0003 inches, the leakage rates are acceptable (~ 15 lb/hr for four valves). It also showed that with clearances of this magnitude, the effects of grooves in the valve stem were negligible or even increased the leakage rate. They would also act as stress risers. It was therefore concluded that grooves should not be cut into the valve stems.

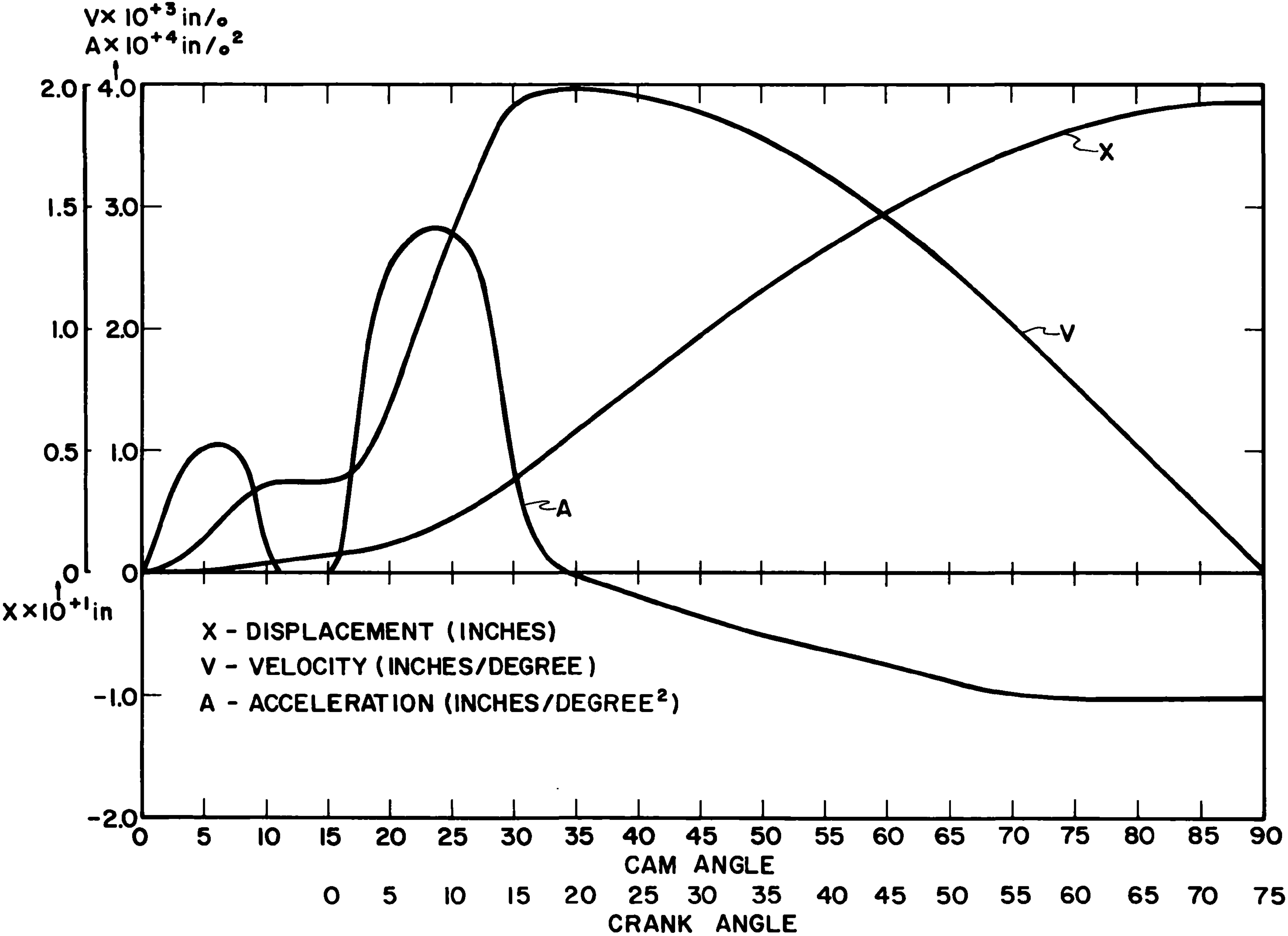


Figure 5.20 Cam Profile, Velocity, and Acceleration for Mechanically-Driven Intake Valve.

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(iv) Assembly. Several schemes for construction and assembly of the valves were considered. These were: (a) thread or bolt the bell-shaped cylindrical valves onto the valve stems; (b) weld the bell shaped cylindrical valves to the stems after assembly; or (c) make the valves and stems in one piece and use the split collet arrangement shown in Figure 5.15 to connect the valve stems to the rocker arms. Scheme (c) was chosen for several reasons. Welding the stems and valve together would mean that the assembly could not be dismantled. Also, serious difficulties might be encountered with distortion of the valves during heating and alignment of the two valves. The fatigue durability of any threaded joint is questionable, especially since it is located at a point of stress concentration between the stem and the valve. Threading the two components together also results in a large increase in the height and weight of the assembly.

Making the valve and stems in one piece eliminates alignment problems. The taper onto which the lower disc is placed, and the upper disc forced onto the taper of the split collets by the Belville washers, add very little either to the weight or height of the assembly. The valve gear is also easily dismantled.

b. Choice of Valving Approach for Preprototype System Testing: The American Bosch system has been selected over the mechanically driven valves as the prime intake valve approach for use in the preprototype system testing. Primary emphasis in this selection was based on the following:

(1) The American Bosch system should provide a higher expander efficiency, particularly at part-load, medium-speed expander conditions.

(2) The American Bosch system is more fully-developed and should require less development time and effort than the mechanically-driven valving approach. Use of the American Bosch system should thus provide satisfactory expander operation at the earliest possible date in accordance with the EPA development schedule.

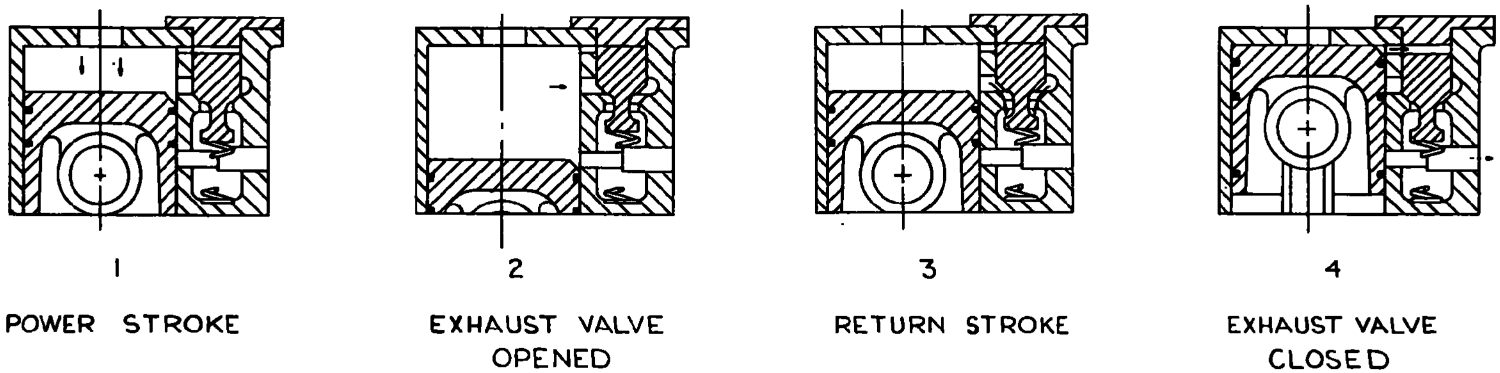
(3) The American Bosch system results in a smaller expander size facilitating packaging of the complete engine in the 1972 Ford Galaxie.

No experimental testing has been carried out on the mechanical valving system. This approach will undoubtedly have vibration problems due to the rapid change in the pressure force on the valves (see Figure 5.18). There are also numerous leakage paths (see Figure 5.15) as well as assembly and machining problems with the mechanical approach. During the next phase of the program, the Ford Motor Company will be working with Thermo Electron Corporation in resolution of these problems and testing of the mechanical valving system. Effort on the BICERI hydraulic valving approach (funded solely by Thermo Electron Corporation) will also be continued. Both of these alternate approaches are simpler than the American Bosch system and will have a lower manufacturing cost, but are in an earlier stage of development.

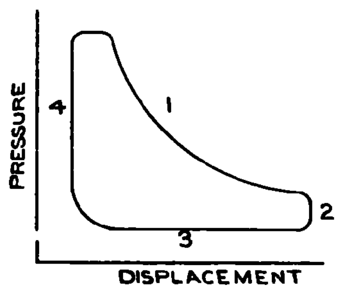
5.1.1.2 Exhaust Valving

The exhaust valve used in the expander is similar in operation to that used in the 5-1/2 hp TECO expanders. The method of operation of the automatic exhaust valve is illustrated in Figure 5.21. The poppet type exhaust valve is connected to the cylinder by three ports; the main exhaust port near bottom dead center which just starts to be uncovered by the piston at 130 degrees ATDC and which is completely uncovered at BDC; an auxiliary exhaust port which is completely covered by the piston at 34 degrees BTDC; and a smaller port near TDC through which cylinder pressure is applied to the top of the valve to insure closure of the valve before the auxiliary exhaust port is uncovered by the piston during the power stroke. During the power stroke (sketch 1 of Figure 5.21) the cylinder pressure is applied to the top of the valve and keeps the exhaust valve closed until the main exhaust port is uncovered and the cylinder pressure vented to the pressure in the expander exhaust line. When the main exhaust port is uncovered, the pressure force across the valve is equalized and the spring force opens the exhaust valve so that the auxiliary port is also vented to the expander exhaust line (sketch 2 of Figure 5.21). During the return stroke (sketch 3 of Figure 5.21), vapor in the cylinder is pushed through the auxiliary exhaust port until this port is closed by the piston. When this port is closed, recompression of the remaining vapor in the cylinder occurs; this pressure is applied to the top of the exhaust valve through the small port near TDC, and the pressure imbalance initiates closing of the exhaust valve as illustrated in sketch 4 of Figure 5.21. This type of exhaust valving gives an ideal P-V diagram as illustrated in Figure 5.21 and maximizes the power per unit displacement of the expander.

In scaling this valve to the automotive-size expander, much larger flow areas are required. As a result, the valve travel and the valve diameter are considerably larger than in the 5-1/2 hp expander. This



EXHAUST VALVE SEQUENCE



INDICATOR DIAGRAM

Figure 5.21 Schematic of Exhaust Valve Function.

greater travel results in a requirement for higher average velocity and the larger valve diameter results in higher pressure forces on the exhaust valve than for the 5-1/2 hp expander. As a result of these differences, bench-testing of the full-size exhaust valve was carried out as part of this phase of the EPA program to establish the operating characteristics of the exhaust valve to be used in the preprototype expander. Since it is difficult to exactly simulate in the bench test unit operating conditions of the expander, additional development of the exhaust valve may be required in testing of the preprototype expander to attain optimum expander performance.

The test fixture used for testing of the exhaust valve is illustrated in Figure 5.22. The test fixture consists of a rotary valve and drive arrangement to provide timed pressure pulses for operating the exhaust valve. Compressed air is fed into one end of the rotary valve which is rotated by a variable speed drive, and a square-wave pressure pulse is then generated and applied to the top side of the exhaust valve, causing it to alternately close and open. As in the expander, the opening force is supplied by a spring. Originally, the exhaust valve was fitted with a velocity transducer, as shown in Figure 5.22, but it proved impossible to keep the magnetic core attached to the valve, due to vibrations caused by the exhaust valve hitting its seat. This method was abandoned in favor of using a stroboscope to plot valve displacement versus time.

The valve development during the testing involved primarily (1) reduction of the seating velocity to an acceptable level while still maintaining the required closing time for the valve, and (2) selection of the appropriate spring to provide the proper valve opening time.

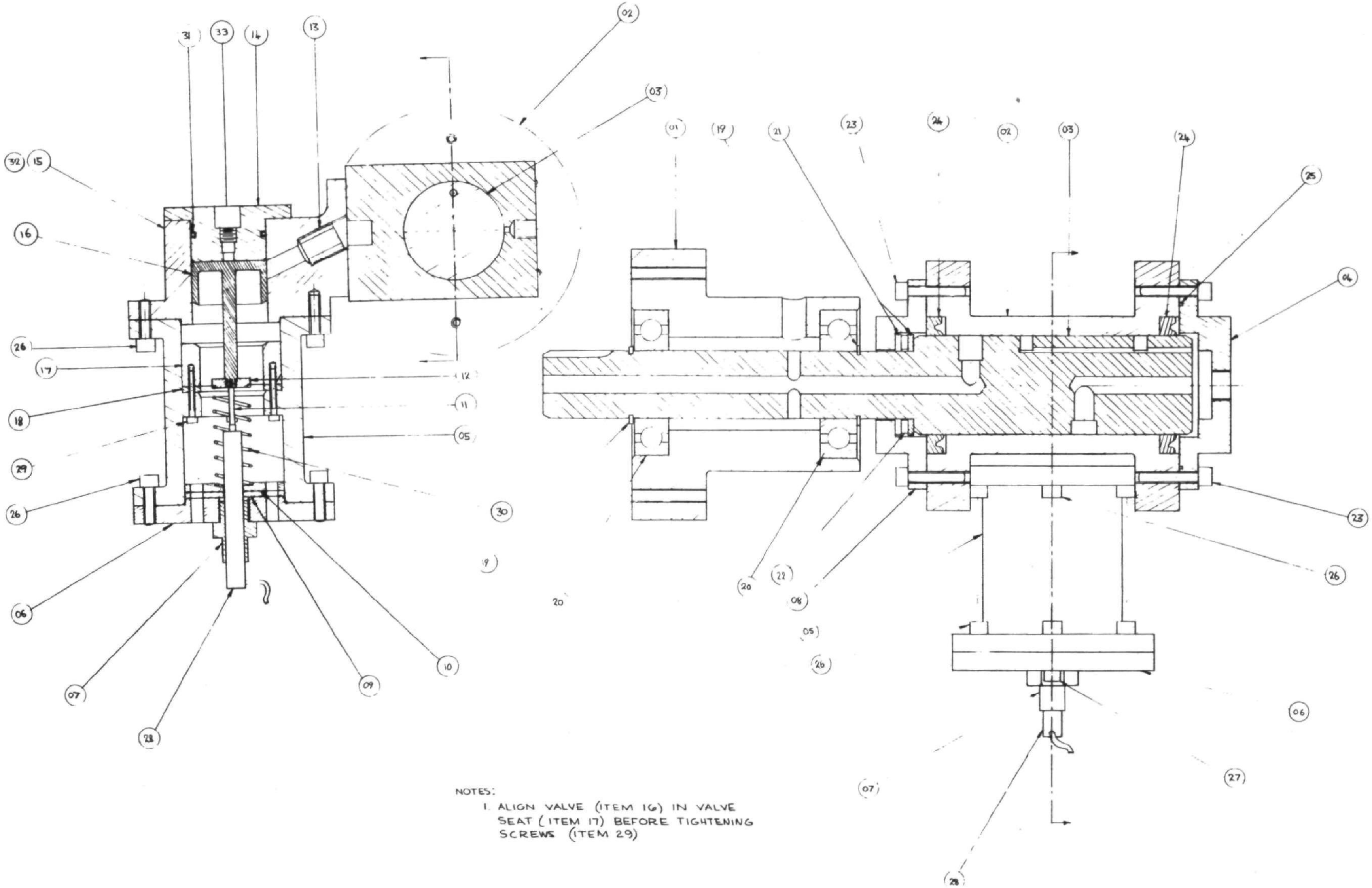


Figure 5.22 Exhaust Valve Test Apparatus.

The required opening and closing times are established by operation of the expander at its maximum speed of 1800 rpm. For closing of the exhaust valve, closing action starts at 34° BTDC and the valve must be closed at 34° ATDC when the auxiliary exhaust port is again uncovered. Leakage by the exhaust valve guide and piston at the top of the valve is small enough to have negligible effect on expander performance.

For the specified travel of 0.3 in., the minimum average valve velocity during closing is:

$$\frac{0.3 \text{ in.}}{68 \text{ degrees}} \times (1800)(360) \frac{\text{degrees}}{\text{min.}} \times \frac{\text{ft}}{12 \text{ in.}} \times \frac{\text{min}}{60 \text{ sec}} = 3.97 \text{ ft/sec.}$$

The maximum closing time is 6.30 milliseconds. During opening of the valve, opening action starts at 130° ATDC and the valve must be completely open at 130°BTDC (the main exhaust port is open during this period). The minimum average velocity during opening is thus 2.70 ft/sec and the maximum opening time is 9.25 milliseconds.

The primary experimental effort was concentrated on development of a damping method to reduce the seating velocity to a level of 10 - 12 ft/sec. Also, since the pressure differential varies across the valve during the recompression, and for different expander operating conditions, it was desirable that the method of damping provide a relatively small variation in the valve velocity for different pressure differentials. A large number of damping methods to accomplish these requirements were evaluated or tested during the program. These can be divided into two categories:

a. Pressure Restriction Dampers. This type of damper is illustrated in Figure 5.23. A check valve with a small opening is placed between the valve and expander cylinder. This valve allows unrestricted flow during exhaust valve opening and limits the rate of pressure buildup during valve closing. A number of variations on this principle were evaluated, but all fell short of the mark for one or both of the following reasons. Either the time to close was excessive for a reasonable seating velocity, occupying more than the allotted 68 crank degrees at 1800 rpm (6.30 milliseconds) and/or a particular design operated properly at a particular operating conditions, but did not operate properly at another operating condition where the shape of the P-V diagram (i.e., the forcing function on the valve) was different. With variable intake valving, the P-V diagram varies greatly, depending on the intake ratio and expander rpm.

b. Hydraulic Dampers. A hydraulic type of damper was developed which operated satisfactorily and this design, illustrated in Figure 5.24, was selected for incorporation in the expander design; the key dimensions of the exhaust valve are summarized in Table 5.5. During exhaust valve opening, the check valve opens permitting a large flow area for the lubricant so that the opening characteristics of the exhaust valve are not affected by the hydraulic valve. During closing of the exhaust valve, the check valve closes so that the lubricant must flow through the orifice down the center of the check valve. Since the flow rate through the orifice (which is directly proportional to the valve velocity) varies as the square of the pressure differential, the valve velocity is relatively insensitive to the pressure applied to the top of the valve, and thus to variations in the P-V diagram of the expander.

TABLE 5.5

EXHAUST VALVE DIMENSIONS

Table Seat Diameter	1.5 inches
Valve Piston Diameter	0.75 inches
Valve Travel	0.30 inch
Damping Plunger Diameter	0.40 inch
Valve Return Spring	71 lbf/in 0.3 inch recompression with valve in open position.
Damper Plunger Return Spring	20 lbf/in 0.4 inch precompression
Orifice Size	1/32 inch diameter

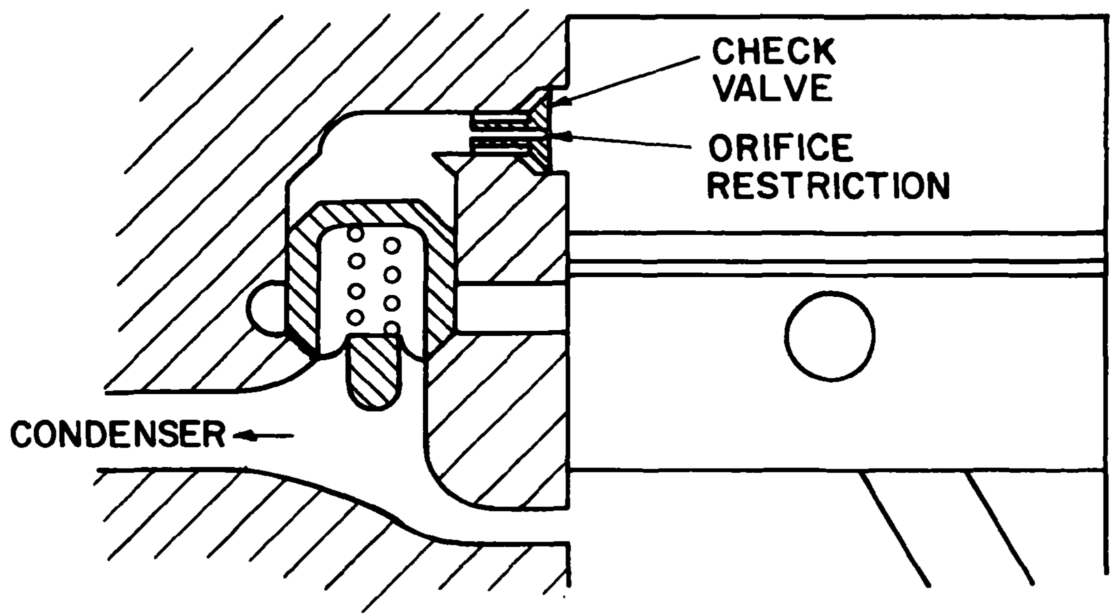


Figure 5.23 Pressure Restriction Damper for Exhaust Valve.

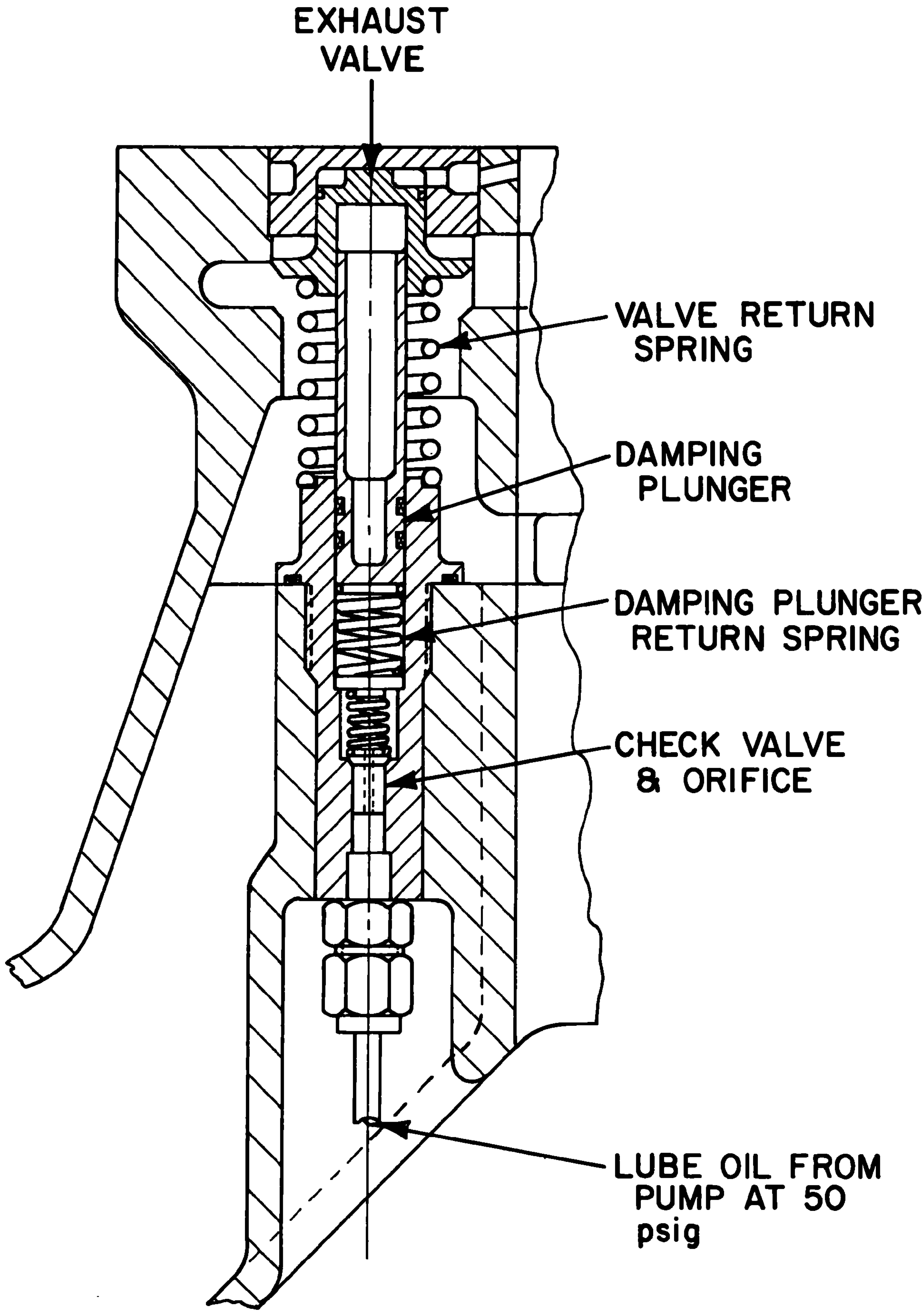


Figure 5.24 Exhaust Valve Design with Hydraulic Damper.

The top of the exhaust valve to which vapor pressure is applied is made smaller than the valve seat diameter to limit the pressure force and the closing velocity of the valve. A seal ring is incorporated in the top valve guide to reduce leakage by the valve guide, since the exhaust valve will be open during the initial part of the power stroke during some operating conditions. This seal ring thus eliminates significant leakage of vapor from the cylinder through the exhaust valve under these conditions.

Typical valve motion during closing is illustrated in Figure 5.25 for a pressure pulse of 60 psig applied to the valve piston. Curves are presented for no damping and with damping with 1/32 inch and 1/64 in. diameter orifices. At the maximum travel of 0.3 inch for the exhaust valve, the valve velocity and total travel time are summarized in Table 5.6. An orifice size of 1/32 inch diameter was selected for the initial design; the optimum orifice size will be experimentally determined during testing of the preprototype expander.

Measured valve motion during opening is illustrated in Figure 5.26. The opening time is about 5 milliseconds for 0.3 inch lift which meets the requirement for an opening time of less than 9.25 milliseconds at 1800 rpm expander speed. The opening time is determined primarily by the spring characteristics used in the valve.

5.1.1.3 Expander Layout

The detailed layout of the expander with American Bosch intake valving and hydraulically damped exhaust valve is shown in Figures 5.27 through 5.33. The main cylinder block and cylinder head are of cast iron, the crankshaft is of cast steel, and the pistons and connecting rods are of cast aluminum. The primary forces are

TABLE 5.6

MEASURED EXHAUST VALVE CLOSING VELOCITY
AND TOTAL TRAVEL TIME
FOR 0.3 INCH LIFT AND 60 PSIG PRESSURE PULSE

Case	Seating Velocity (ft/sec)	Total Travel Time (milliseconds)
No Damping	15.7	3.25
With Damping		
1/32 inch Orifice	12	3.7
1/64 inch orifice	10	3.9

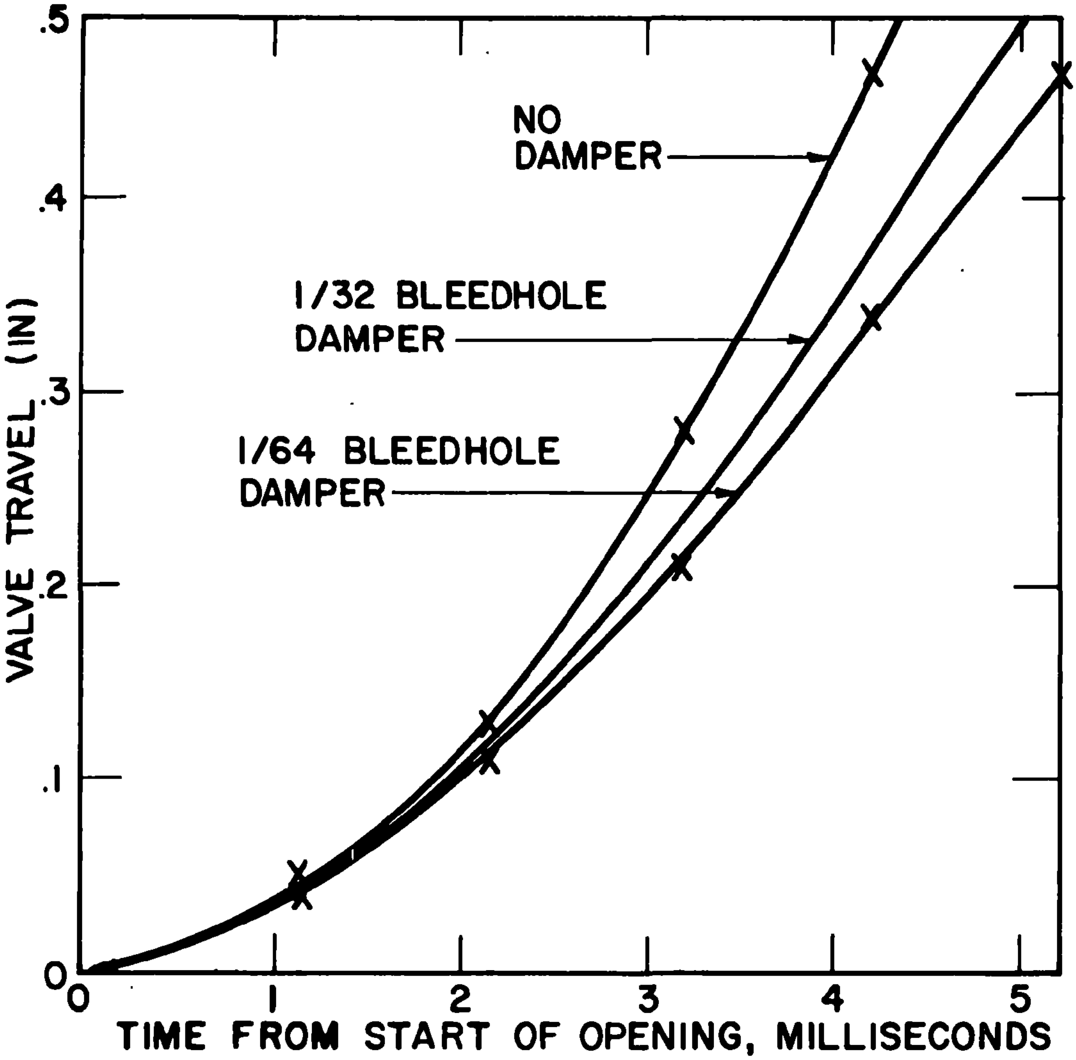


Figure 5.25 Exhaust Valve Closing Travel vs. Time.

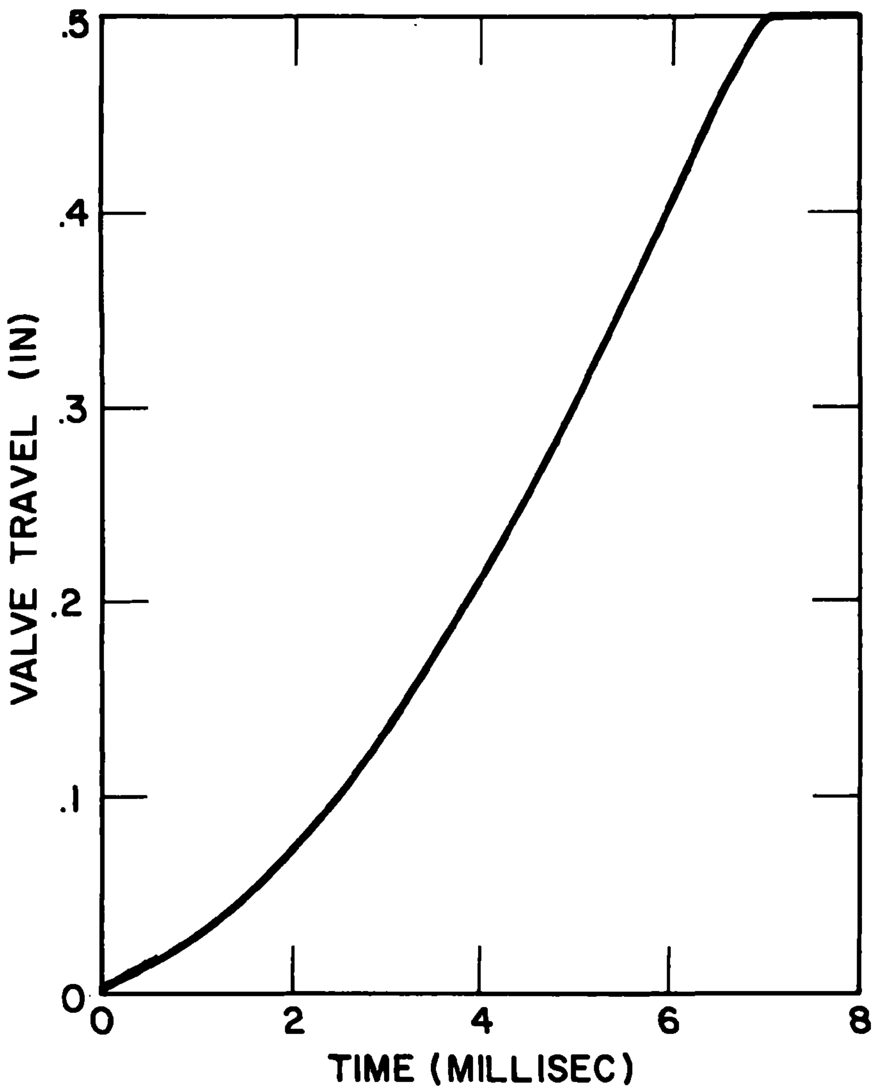


Figure 5.26 Exhaust Valve Opening Travel vs. Time.

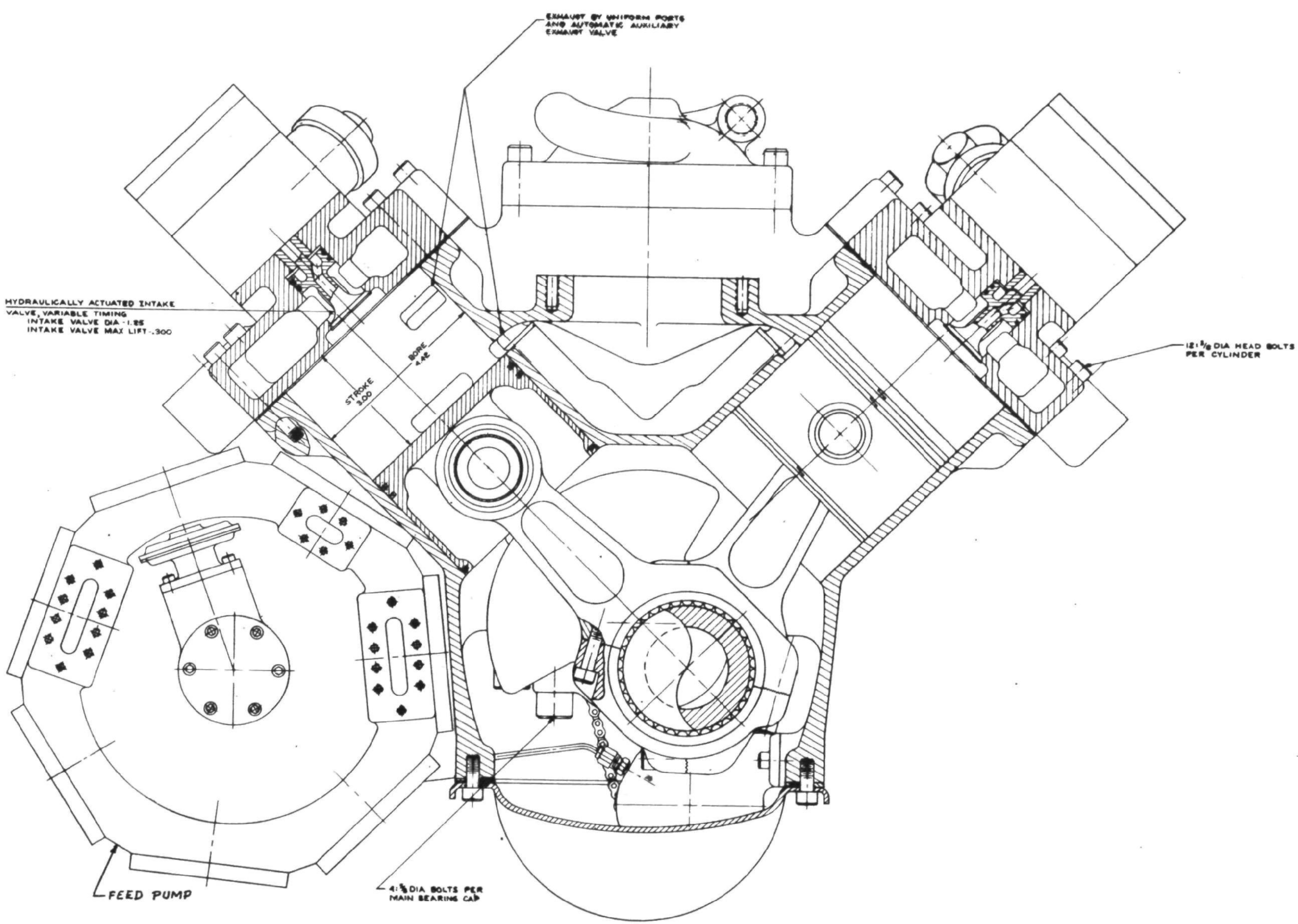


Figure 5.27 Expander Layout with American Bosch Hydraulic Valving - Cross Section Through Front Cylinders.

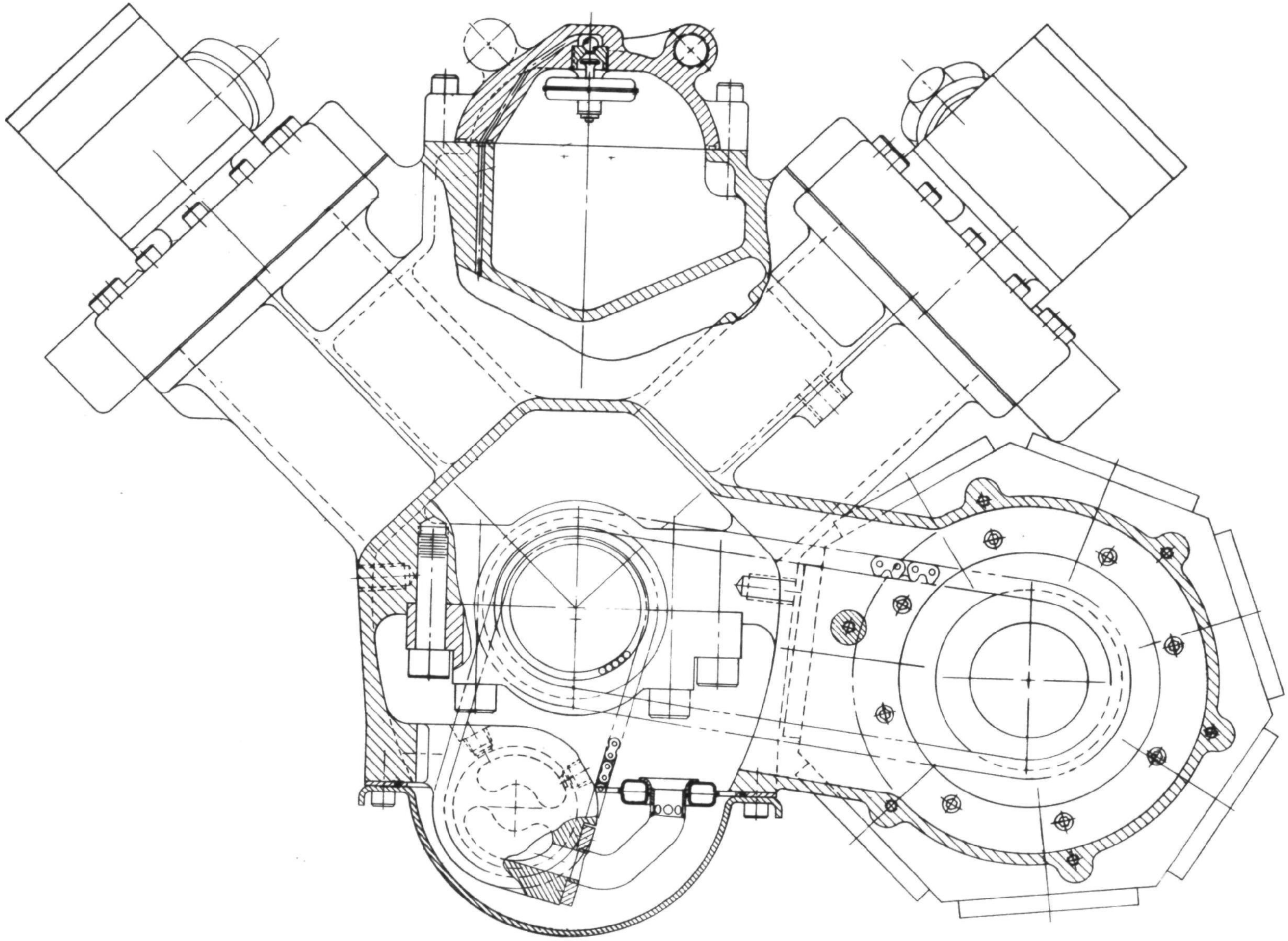


Figure 5.28 Expander Layout with American Bosch Hydraulic Valving - Cross Section Through Rear of Expander Showing Feedpump and Oil Pump Drive.

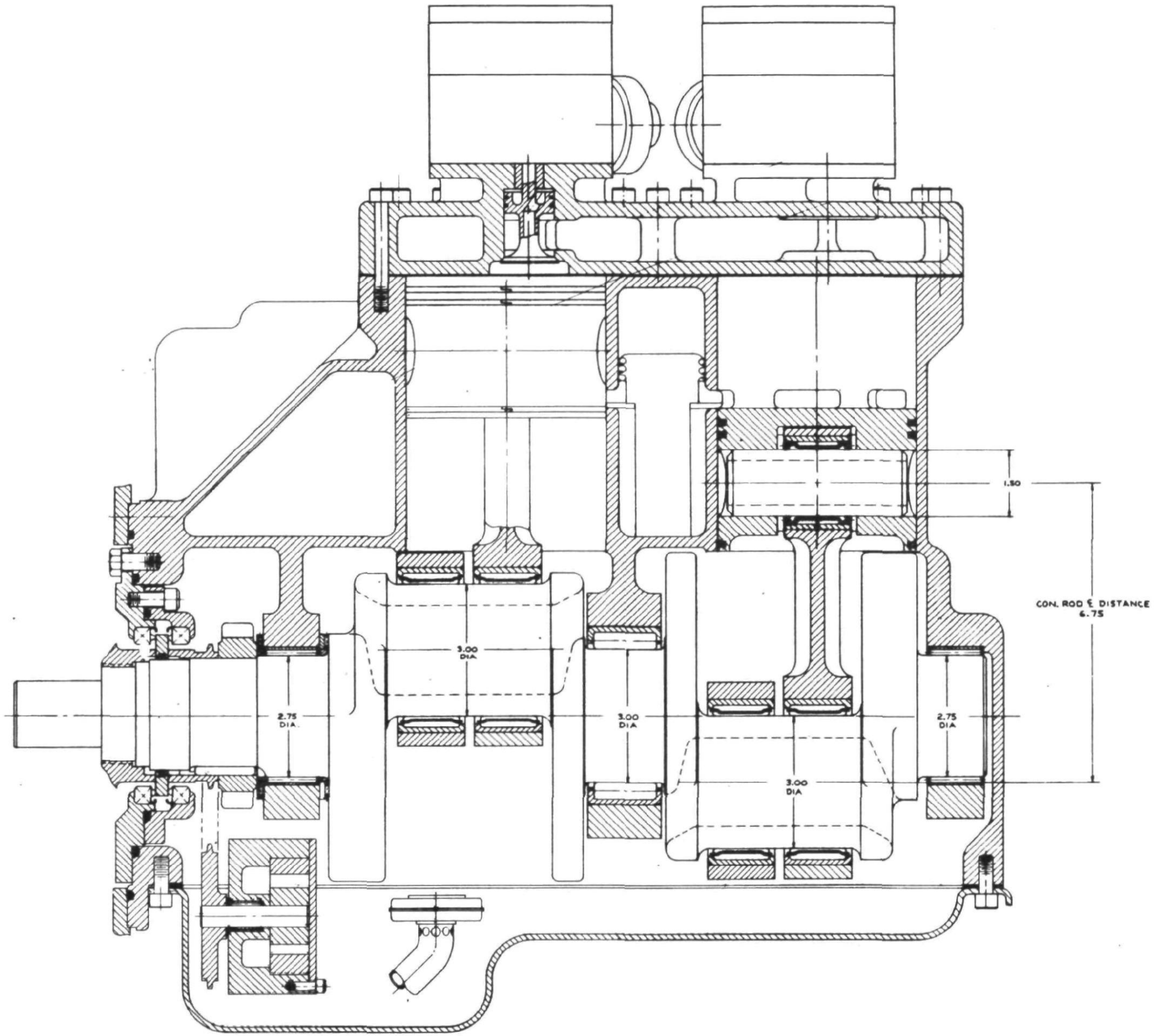


Figure 5.29 Expander Layout with American Bosch Hydraulic Valving - Side Cross Section Through Cylinder Bank and Crankcase.

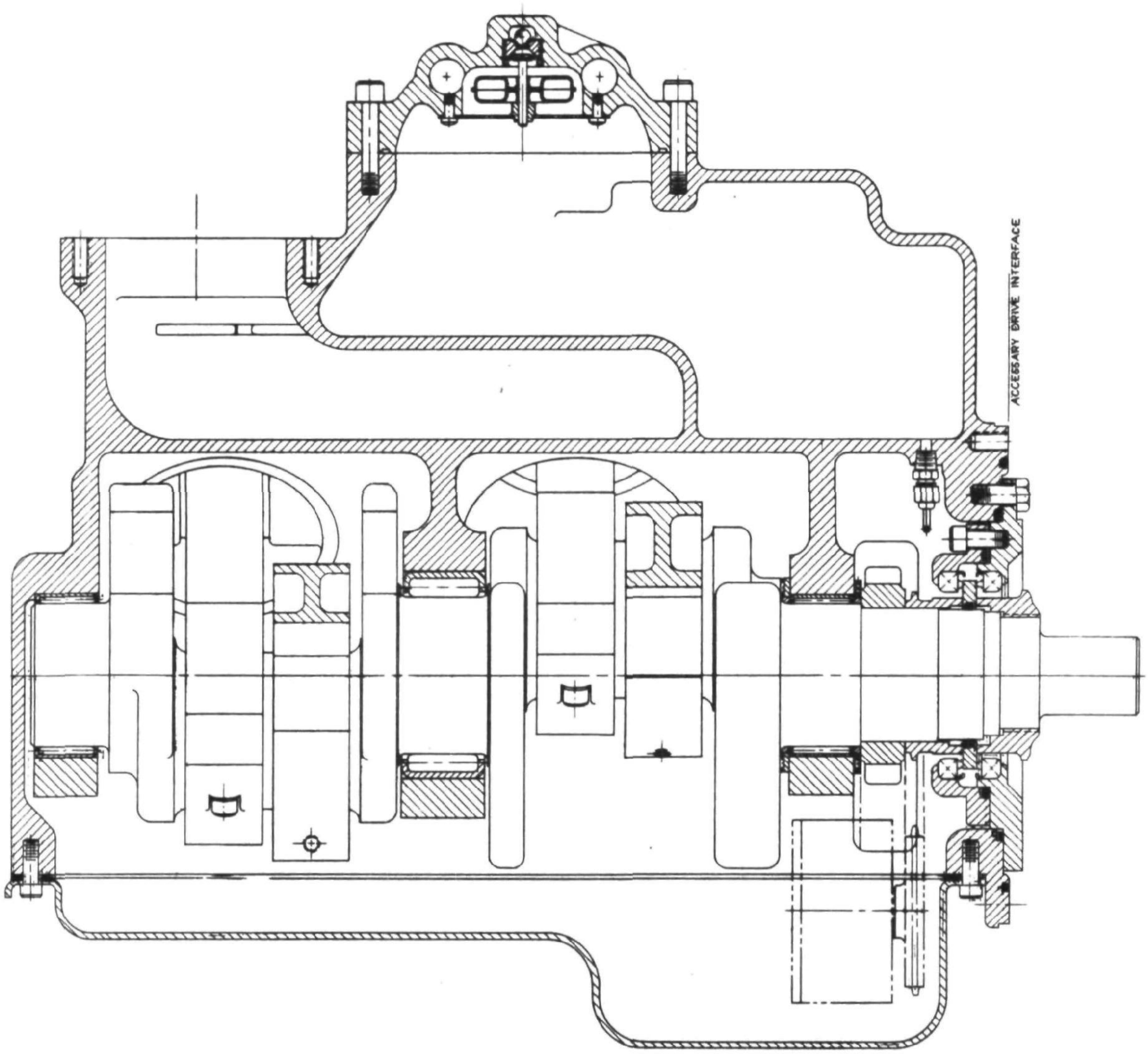


Figure 5.30 Expander Layout with American Bosch Hydraulic Valving - Horizontal Cross Section through Crankcase Along Crankshaft.

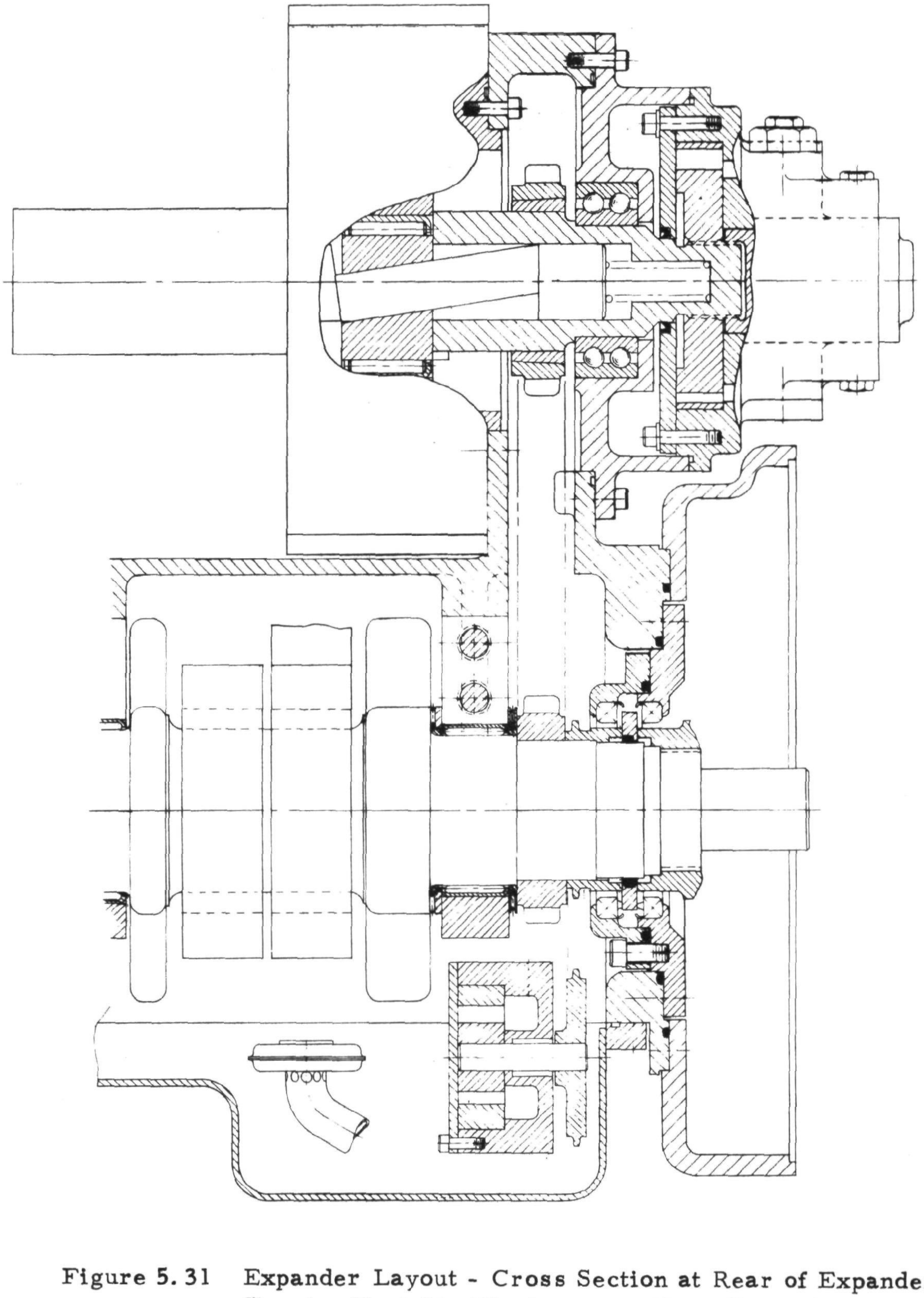


Figure 5.31 Expander Layout - Cross Section at Rear of Expander Showing Variable Displacement Vane Pump for American Bosch Hydraulic Valving System.

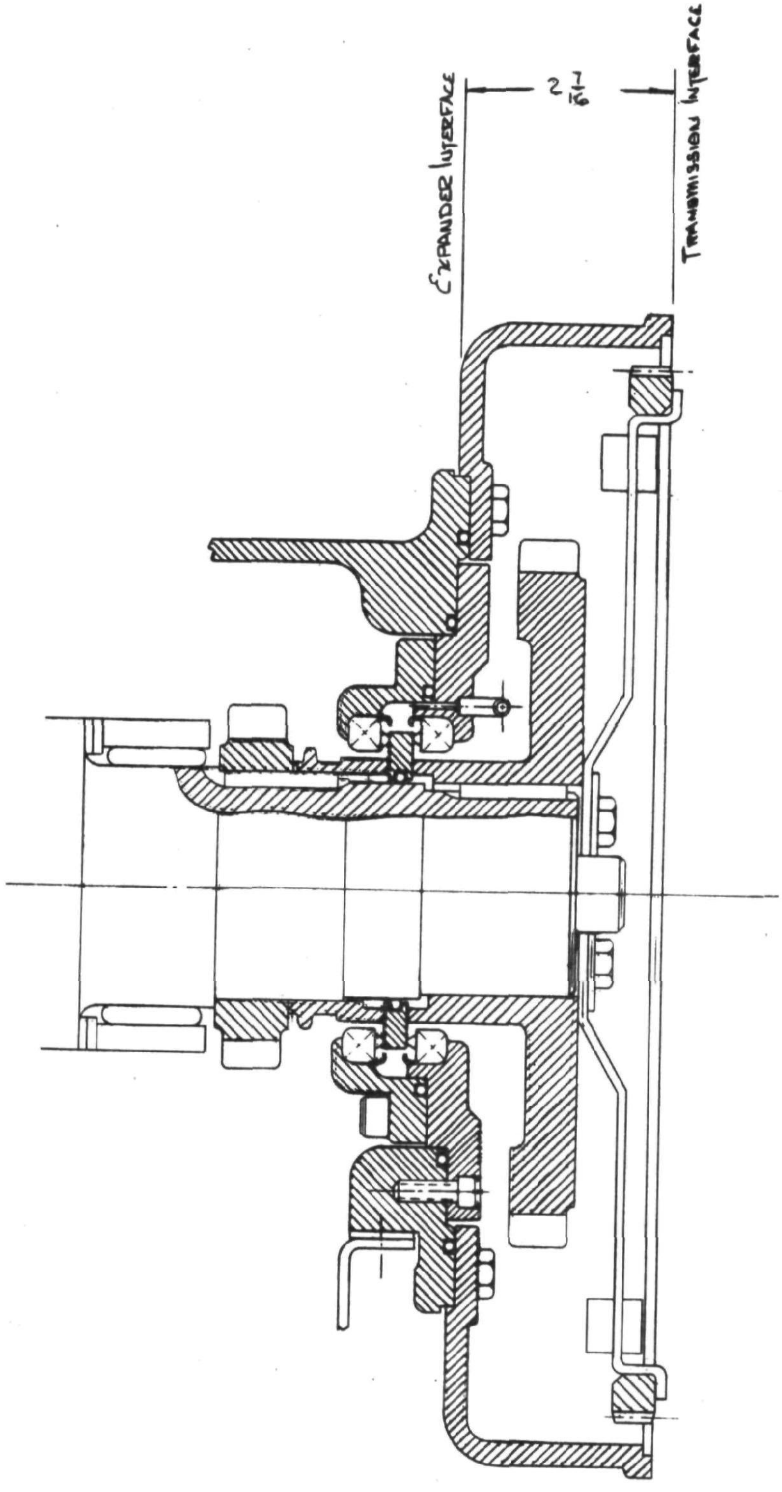


Figure 5.32 Expander Layout - Cross Section at Rear of Expander Showing Accessory Drive Bell Housing and Transmission Interface.

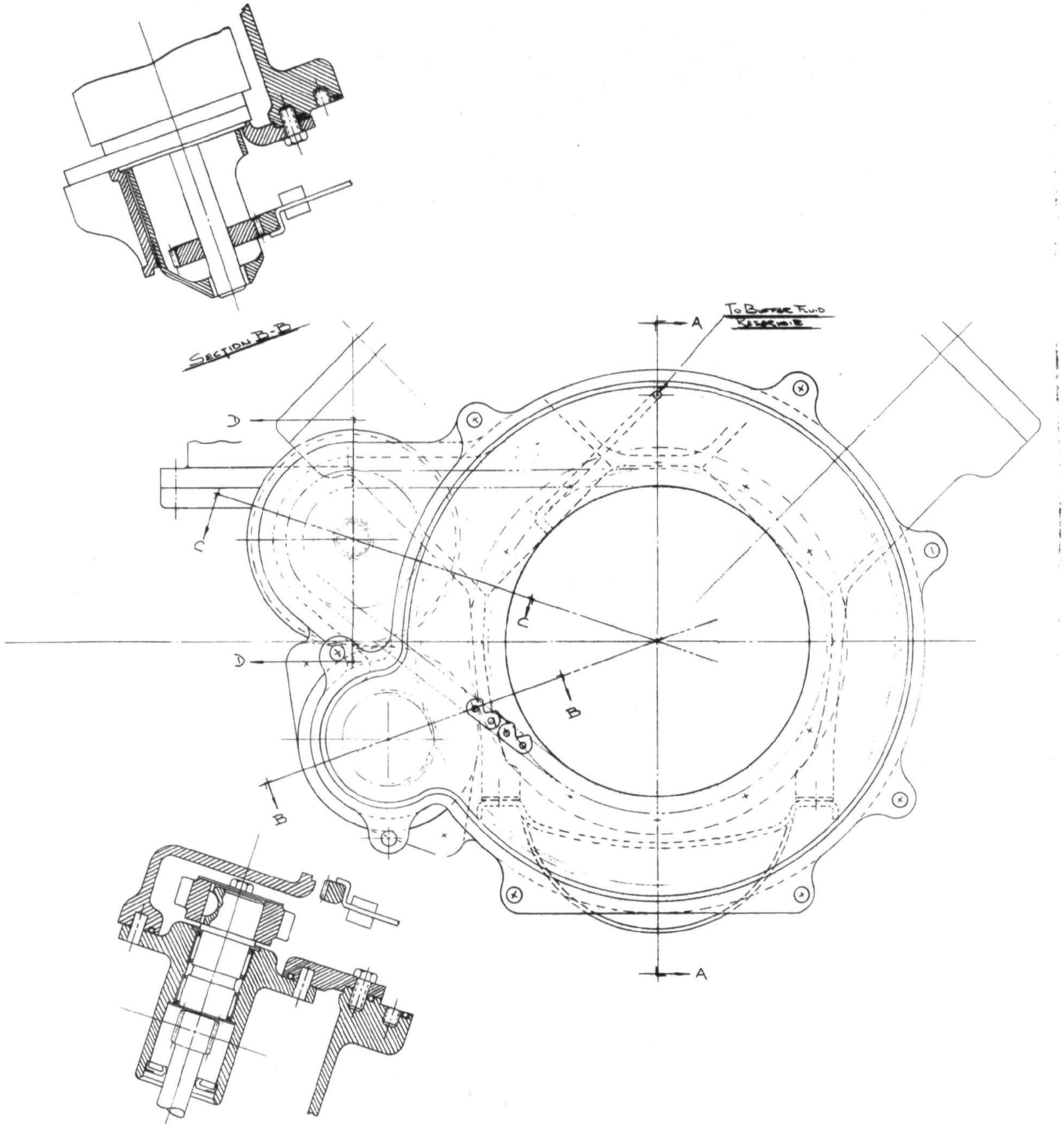


Figure 5.33 Expander Layout - Rear View Showing Accessory Drive.

naturally balanced, while the primary moments are balanced by the counterweights (note that the crankpins are hollow to reduce rotating unbalance). The secondary forces are unbalanced in the horizontal plane and have a peak value of 381 pounds. There is no secondary pitching moment.

Needle or roller type bearings are utilized throughout to minimize difficulty with bearings during initial development of the pre-prototype expander. The bearings are sized for a life of approximately 1500 hours based on expander speed and load equivalent to a vehicle speed of 60 mph.

In Figures 5.27 and 5.31, the location and method of driving the feedpump and lube oil pump are illustrated. The variable displacement vane pump for operating the inlet valve, shown in Figure 5.31, is driven off the same shaft as the feedpump. Figure 5.28 shows the oil separator and reservoir for the valving oil supply.

The accessory drive bell housing is shown in Figures 5.32 and 5.33. The electric starter is shown in Section B-B and the accessory drive shaft in Section C-C. The timer which controls the inlet valve opening is attached to the accessory drive shaft housing. Since the accessory drive is driven at 2.7:1 speed ratio, the timer speed is reduced back to expander speed.

In Figure 5.27, the method of pressure balancing the poppet intake valves is illustrated. A balancing piston is incorporated in the valve design as illustrated to balance pressure forces, thereby greatly reducing the force required to operate the valve. A port through the center of the valve provides cylinder pressure on top of the balancing piston equalizing forces due to the cylinder pressure.

The pressure of the incoming vapor is applied to the bottom of the balancing piston and top of the valve equalizing these forces. Piston type rings are used to reduce leakage around the balancing piston. The balancing hole is 0.25 inch in diameter which is large enough so that flow rates through the valve stem do not cause excessive pressure differentials.

For ease in manufacturing and assembly, the connecting rod is split on a bias so that it can be installed and removed through the bore.

5.1.2 Feedpump Design and Test Results

Many of the features incorporated in the feedpump are dictated by the overall system requirements. The pump is directly driven from the expander, since it represents a significant power requirement. A positive displacement, piston type of feedpump is the most suitable type to provide high overall efficiency at high pressure for a wide range of speeds, with a variable flow rate requirement at any speed. A multiple piston pump is used to minimize pressure pulsations without the use of accumulators.

Two feedpump designs, a 5-cylinder axial and a 7-cylinder radial, were designed, fabricated, and tested in the period covered by this report.* Testing of the axial 5-cylinder pump design, as described in Appendix II, has been terminated in favor of the 7-cylinder radial feedpump. The design features and test results of the radial feedpump, which was selected for the Rankine-cycle power system, are described in the following sections.

*The axial 5-cylinder pump was developed as part of the EPA program. The radial 7-cylinder pump development was financed by Thermo Electron Corporation funding.

5.1.2.1 Performance Requirements

The performance requirements for the feedpump are summarized in Table 5.7.

TABLE 5.7

PERFORMANCE REQUIREMENTS FOR FEEDPUMP

Outlet Pressure	850 psia
Flow Rate	0 to 17 gpm, modulated to any condition under the peak flow curve (Figure 5.40)
Inlet Pressure-Operation	4 - 90 psia
Overall Efficiency	75% at full flow 60% at 30% full flow
Non-operating Temperature Range	-40°F to 150°F ambient
Operating Temperature Range	Fluid inlet temperature from 100 - 250°F; Start-up temperature from -20°F.
Speed Range	300 to 1800 rpm. The pump is directly driven from the expander and uses variable displacement for flow control.

5.1.2.2 Pump Design

The pump is a reciprocating piston type with variable displacement. Views of the pump are shown in Figures 5.34 and 5.35. The pump has seven cylinders located radially about the axis of the rotating shaft. Each cylinder houses seal ring pistons which can be varied from zero to full stroke. The cylinders receive fluid from a common inlet plenum and discharge into a common outlet plenum. Both the inlet and outlet valves are simple spring-loaded washer types. The

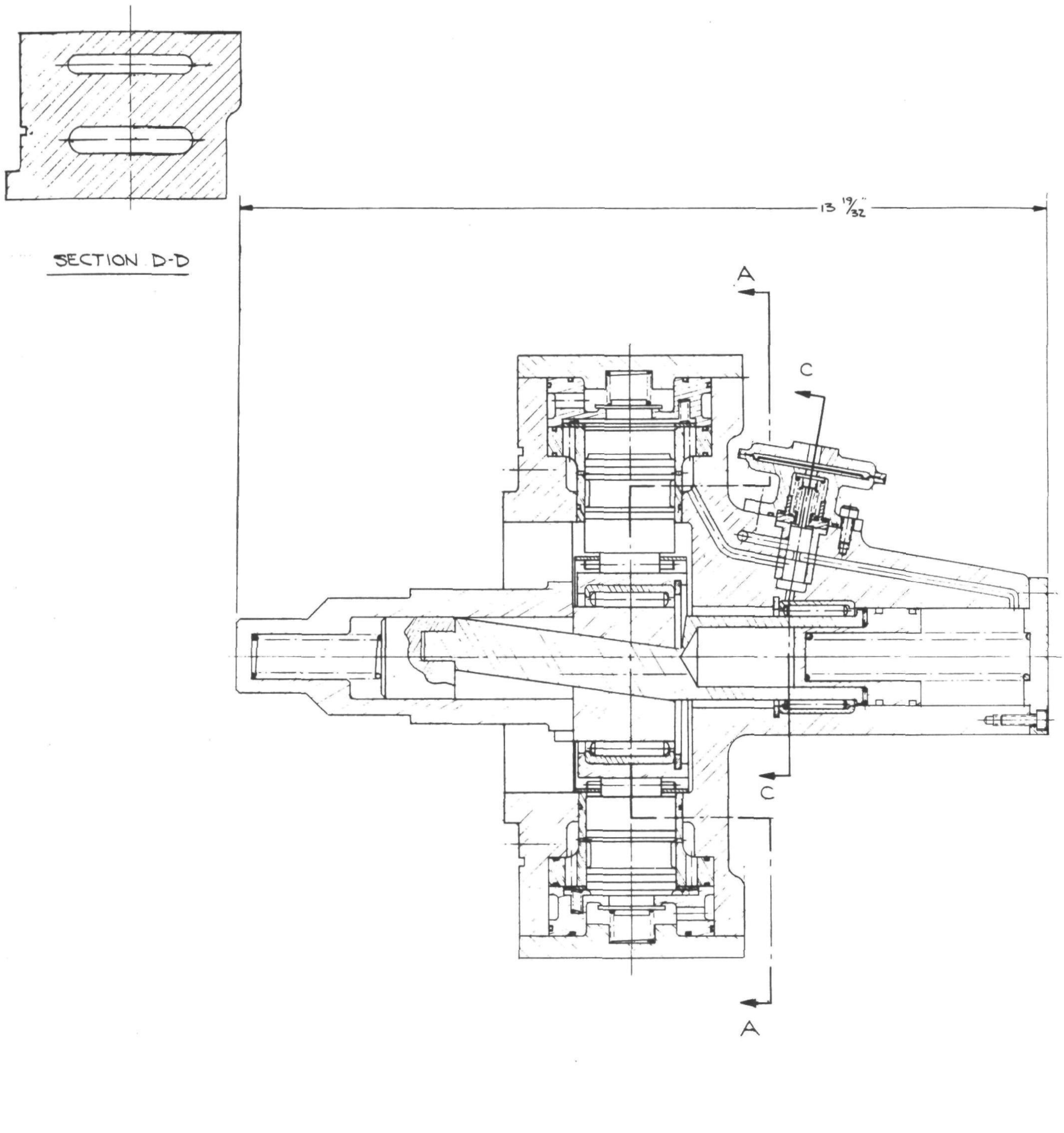
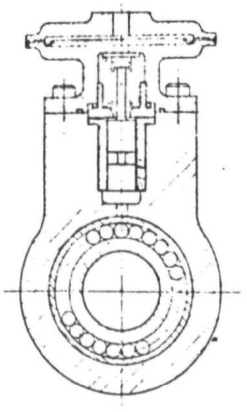


Figure 5. 34 Reciprocating Piston Pump with Variable Displacement - Cross Section.



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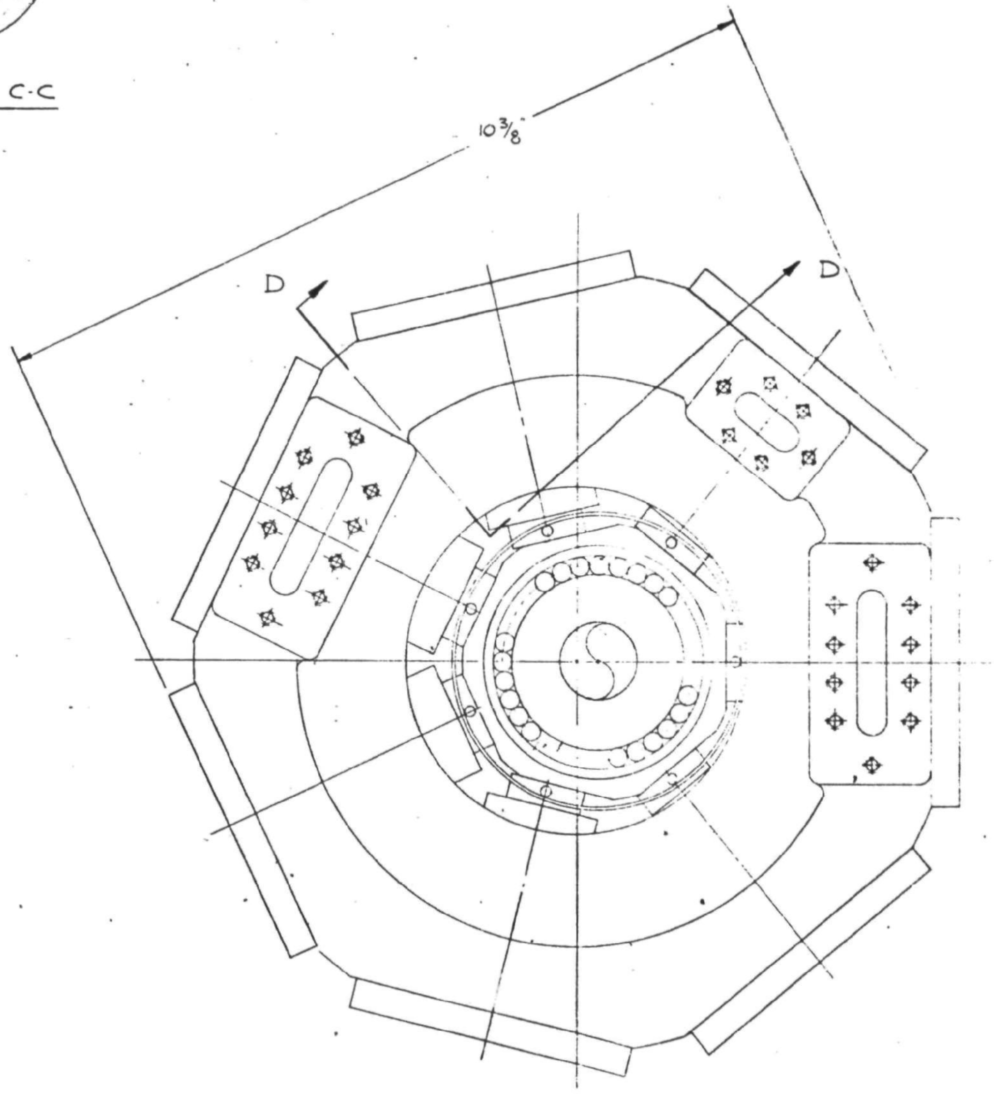

SECTION A-A

Figure 5.35 Reciprocating Piston Pump with Variable Displacement - Section A-A.

piston shoes ride on a septagonal ring with a circular ring used to return the pistons. These rings can be seen in Figure 5.35.

Variable displacement of the pump is achieved by axially moving the angled portion of the shaft through the center eccentric ring. Figure 5.34 shows the pump in its maximum displacement position with the shaft fully engaged. As the shaft is pulled out axially, the stroke of the piston decreases until full extension is achieved at which point the stroke and, therefore, displacement are zero.

The feedpump has an aluminun housing with steel pistons and drive mechanisms. Other important characteristics are summarized in Table 5.8.

5.1.2.3 Feedpump Test Facility

The feedpump test stand is comprised of two systems, the mechanical drive system and the fluid system. The feedpump itself is the connecting link between these systems.

Since the pump is to be tested over a speed range of 300 - 1800 rpm, a variable speed drive (Reeves Model 400) is used to drive the feedpump. The drive unit is supplied with a tachometer for speed indication and a strobotac is used to check the tachometer. A rotating through-shaft torque sensor is used to measure driving

TABLE 5.8

FEEDPUMP CHARACTERISTICS

Bore	- 1.50 inches
Stroke	- 0.466 inches
Maximum Displacement	- 5.76 inches
Speed Range	- 300 - 1800 rpm
Design Point Discharge Pressure	- 850 psia
Maximum Pumping Rate	- 17 gpm
Design Point Pumping Rate	- 15.4 gpm

torque. To protect the torque sensor from overload, a torque limiting clutch is used to couple the drive unit to the torque sensor. The drive unit system is shown in Figure 5.36.

The pump test loop is shown schematically in Figure 5.37. Some of the instrumentation readouts associated with the sensors in the test loop are shown on the extreme right of Figure 5.36. The test loop can be isolated from the pump by closing the ball valve in the discharge line and the shutoff valve in the intake line. The reservoir is heated by strip heaters so that the pump suction pressure can be set by control of the fluid temperature in the reservoir.

A critical aspect of operation of the test loop is to ensure that all air has been removed from the loop, particularly on the intake side of the system. Vents are provided at all possible traps to ensure that any air may be removed from the system.

A turbine flowmeter with an associated digital read-out is used to measure the flow rate output of the pump. The rating of the turbine flowmeter is 1.8 to 18.4 gpm; suitable calibration curves have been provided by the manufacturer, Fischer and Porter Instrument Company.

Variable displacement of the pump was achieved by utilizing an external double-acting piston actuator with a controlling servo valve to maintain the set position of the displacement shaft and, therefore,

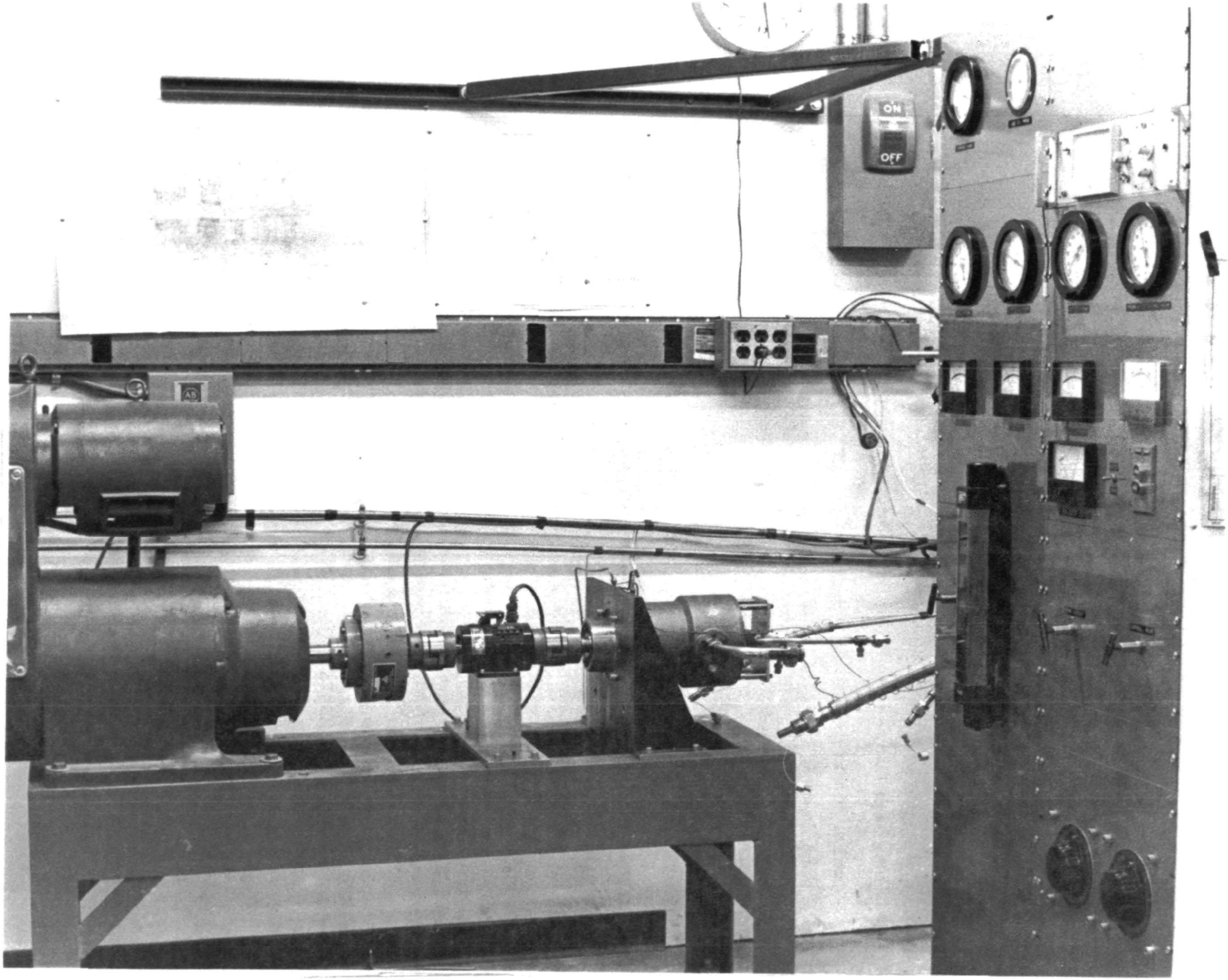

Figure 5.36 Feedpump Test Facility.

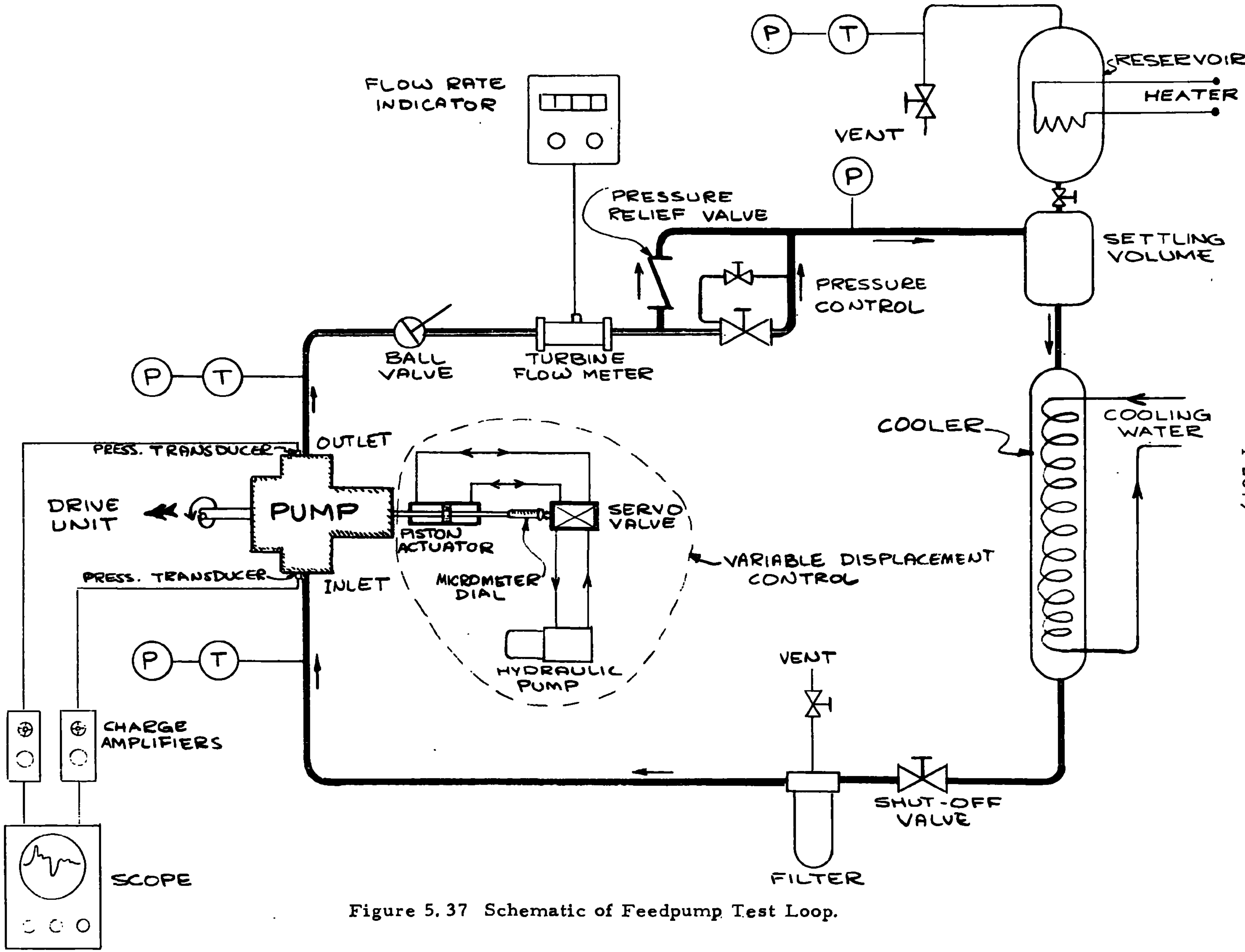

Figure 5.37 Schematic of Feedpump Test Loop.

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the displacement of the pump. A micrometer dial is used to measure the axial setting on the displacement mechanism, which allows for precise indication of actual pistion stroke.

In summary, the test loop provides the following capabilities:

- Variable inlet and discharge pressure.
- Variable inlet temperature.
- Variable pump speed.
- Variable pump displacement.
- Measurement of inlet and discharge static and dynamic pressures.
- Measurement of inlet and discharge temperature.
- Measurement of pump delivery rate.
- Measurement of actual pump displacement.
- Measurement of pump speed and torque to drive the pump.

5.1.2.4 Feedpump Testing

A test version of the pump was fabricated for development testing. It is essentially identical to the system pump shown in Figures 5.34 with a main bearing added to the drive shaft side. In the system where the feedpump is integrated with the expander, this bearing is part of the expander accessory drive housing. The pressure servo control was not incorporated into the test pump.

Initial tests on the radial feedpump were performed at conservative operating conditions with discharge pressures set from 200 psi to 400 psi and speeds not exceeding 700 rpm. The initial testing was performed to accumulate "run-in" time on the pump, to assess the mechanical and basic functional operation of the feedpump, and to accurately correlate the axial movement of the displacement mechanism with the actual stroke of the pistons. The displacement measurement was established to achieve accurate pump performance in subsequent testing, since volumetric efficiency is a direct function of actual pump displacement. After these initial tests, the pump was disassembled and inspected for excessive wear patterns and other mechanical disorders. The feedpump was found to be in excellent condition and was subsequently reassembled and installed back in the loop for performance testing.

In Figure 5.38, the volumetric and overall pump efficiency (hydraulic power/shaft power) is presented as a function of displacement for discharge pressures from 415 psia to 715 psia and pump speeds of 500 rpm and 600 rpm. Suction pressure for these tests varied from 9 to 18 psia. From Figure 5.38, it is evident that the pump volumetric and overall efficiencies are insensitive to pump discharge pressure. Also the efficiencies maintain a high level down to low displacements (or pumping rates). This characteristic is important in maintaining a high system efficiency at part-load operating conditions. The peak overall efficiency of the pump is 78% at these speeds.

The effect of pump speed on efficiency is indicated in Figure 5.39 at a discharge pressure of 515 psia and for various displacements. Both the volumetric efficiency and the overall efficiency decrease as the pump speed is increased up to the maximum speed of 1800 rpm. At a speed of 420 rpm, the volumetric efficiency is very close to 100% and the overall efficiency is 83%. At the maximum speed of 1800 rpm, the volumetric efficiency has decreased to 72% and the overall efficiency to 63%. At all pump speeds, the efficiencies are relatively insensitive to the pump displacement.

In Figure 5.40, the maximum pumping rate required for the system is presented as a function of feedpump (or expander speed). This curve is based on the system performance calculations at the peak power output. The fraction of total pump displacement and required horsepower to achieve these maximum pumping rates are also presented at several pump speeds in Figure 5.40, based on the experimental performance at a discharge pressure of 515 psia. In Figure 5.41, a plot is presented of the horsepower, pump displacement, volumetric efficiency, and overall efficiency as a function of pump speed for the maximum pumping rate, as presented in Figure 5.40, and for a discharge pressure of 515 psia. The peak power requirement of 6.55 hp occurs at the maximum speed of 1800 rpm. The displacement required to achieve the maximum required pumping rate at 1800 rpm is 48% of full stroke. Since the pump efficiency is relatively insensitive to discharge pressure, the pumping power required is closely proportional to the pressure differential across the pump at a given rpm and displacement. At the maximum discharge pressure of 830 psia at 1800 rpm, these results indicate a required pump shaft power of:

$$6.55 \text{ hp } \frac{830 - 49 \text{ psia}}{515-35 \text{ psia}} \simeq 10.6 \text{ hp.}$$

Testing of the pump will be continued to establish performance up to the design discharge pressure of 850 psia and to establish acceptable life for the pump. The tests to date have demonstrated the feasibility of the concept.

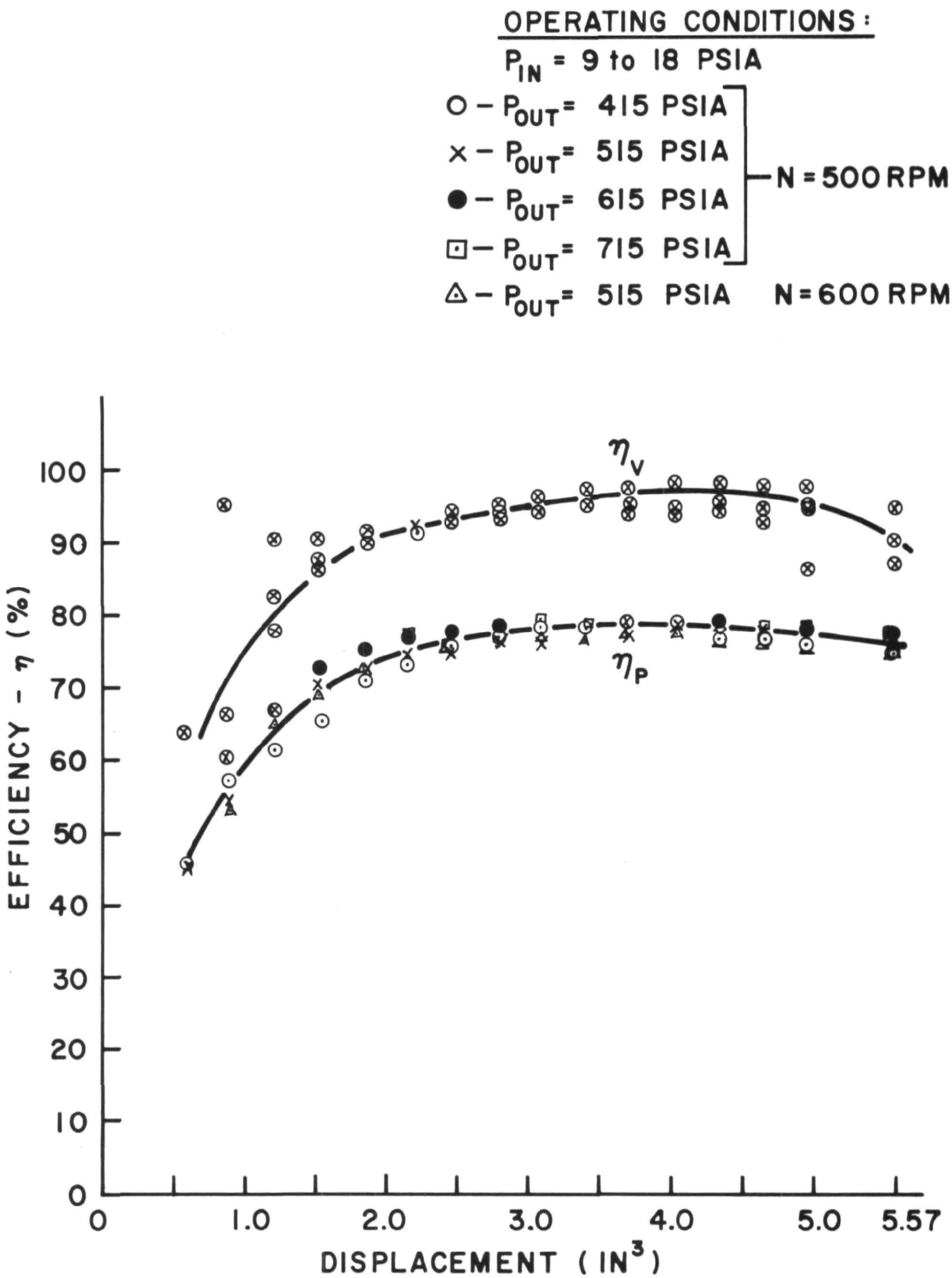

Figure 5.38 Efficiency vs. Displacement for 7-Cylinder Feedpump.

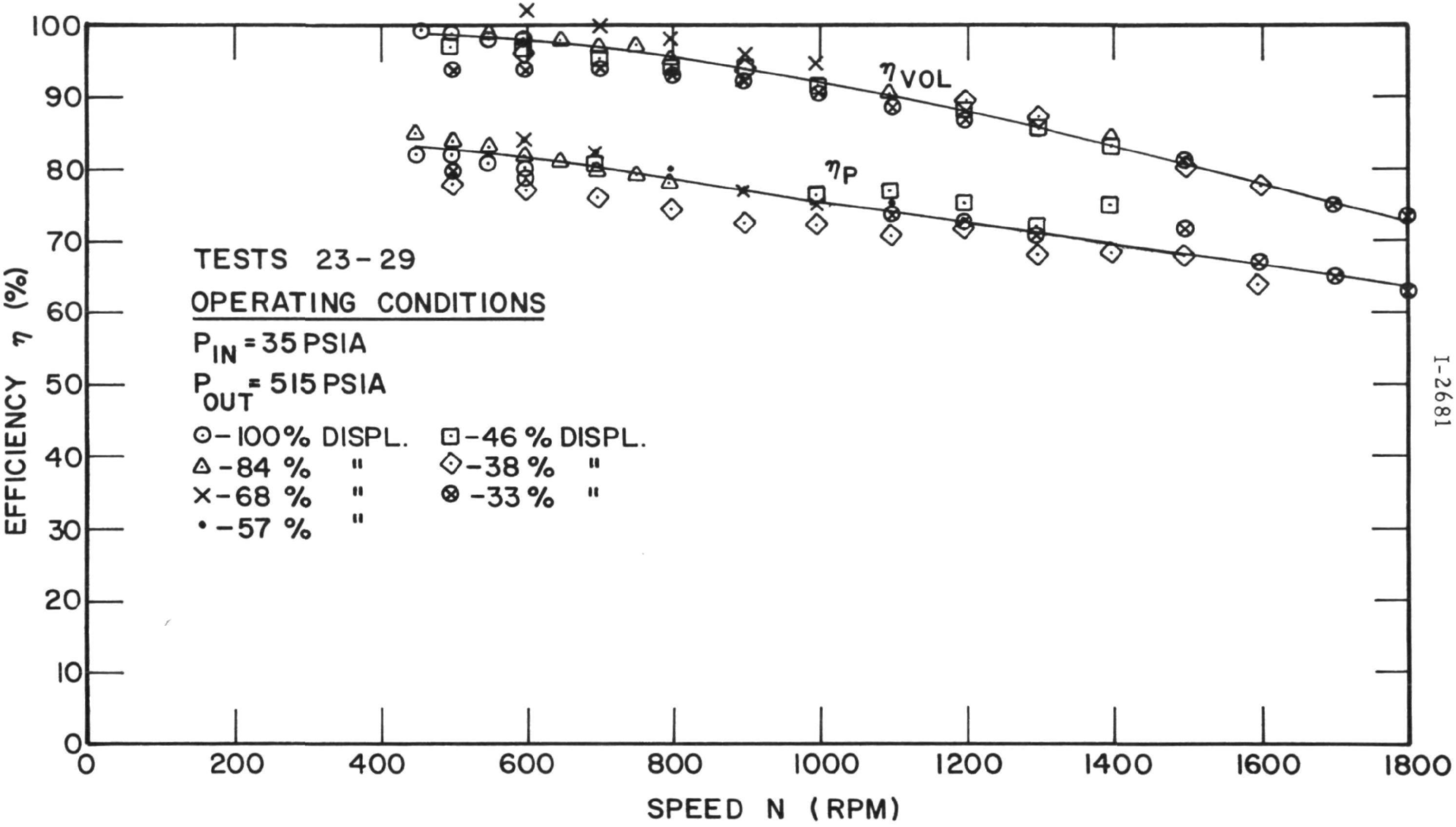

Figure 5. 39 Efficiency vs. Speed for 7-Cylinder Feedpump.

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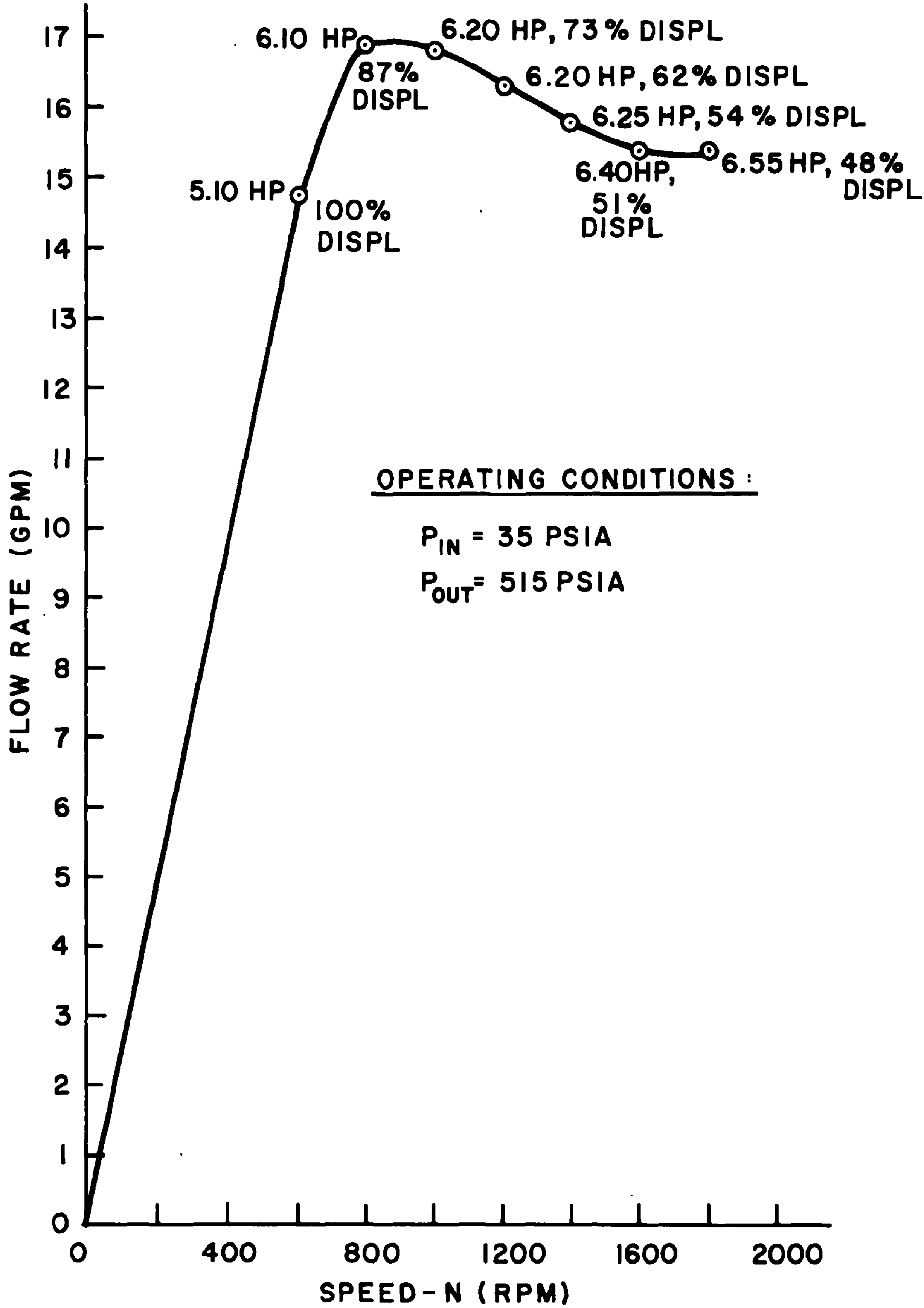

Figure 5.40 Flow Rate vs. RPM for 7-Cylinder Feedpump.

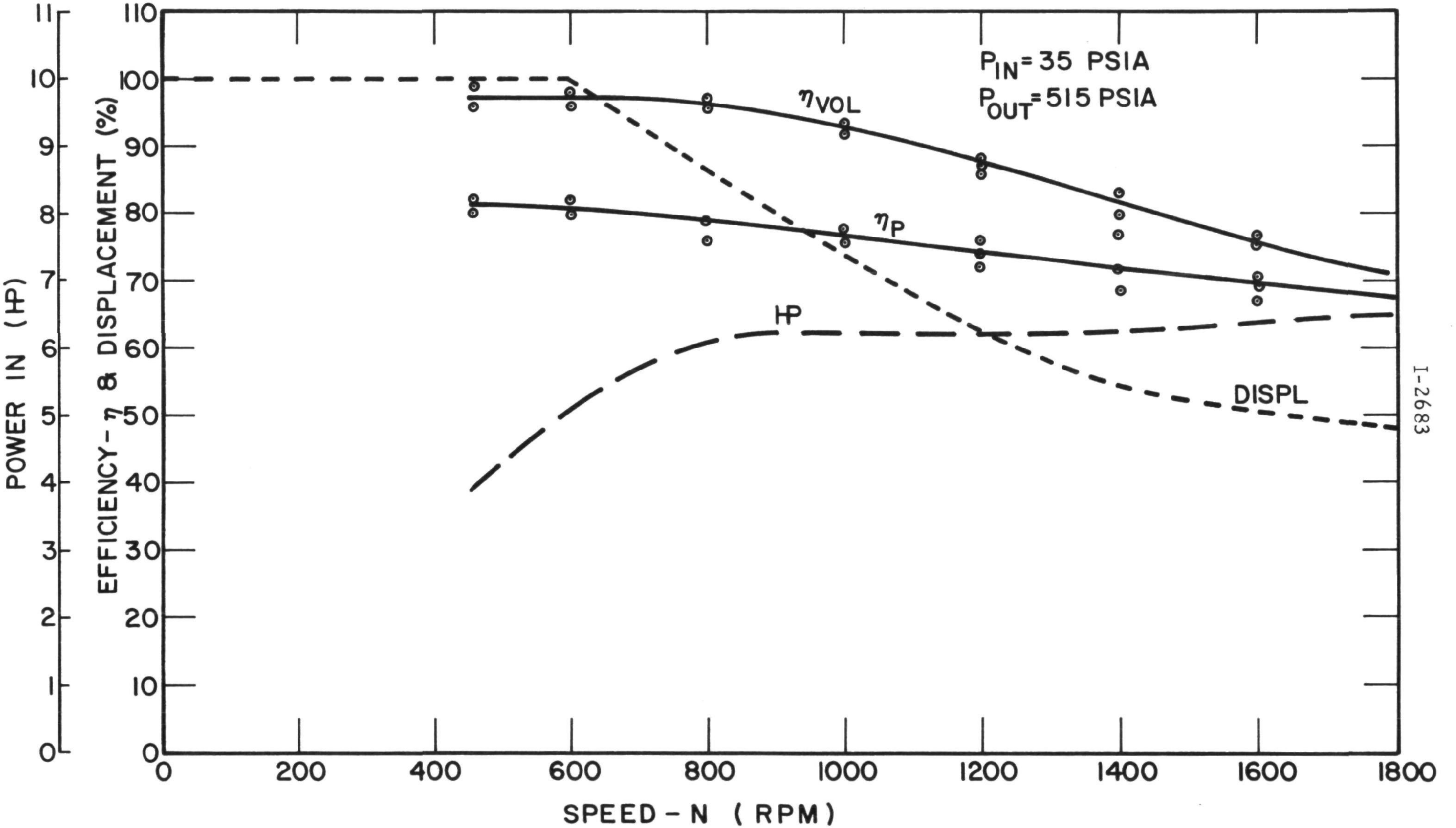

Figure 5.41 Efficiency, Displacement, and Shaft Power Input vs. Speed for 7-Cylinder Feedpump.

The test pump delivers the required range of flow rates at high pressure over the operating range of shaft speeds. The efficiency of the pump remains high over a large variation of displacement. Pressure pulsations are very small and acceptable over the entire range of operating speed and displacement tested.

5.1.2.5 Feedpump Control

The function of the feedpump displacement control is to maintain a constant boiler discharge pressure over the dynamic system operating range from idle to full power by varying the organic flow rate to the boiler in response to the boiler outlet pressure. The feedpump control is illustrated in Figure 5.42 and the position of the control on the feedpump is illustrated in Figure 5.34. The control employs a spool control valve operated by a spring-biased diaphragm to which the boiler outlet pressure is directly applied. This spool valve controls application of the feedpump discharge fluid to a power piston, which is connected to the stroke control shaft of the pump and is used to vary the displacement of the feedpump. The power piston is required because of the large force (~600 lb) required to vary the pump displacement.

Steady-state positioning of the shaft is achieved when a force balance is reached between the pump shaft force and the power piston force. Force from the pump pistons acts on the inclined section of the control shaft to yield an axial force component that is applied to the shaft and power piston assembly. The motion resulting from this force tends to reduce the pump displacement, unless an opposing pressure force is applied on the power piston side of the control shaft. Controlled movement of the shaft and power piston assembly is obtained by varying the working fluid pressure applied to the power piston from the spool control valve (3). In Figure 5.42, the two-land spool valve is shown in the null position, which indicates that the required boiler design pressure has been reached. An increase in boiler pressure will move the diaphragm (4), load spring (5), isolating bellows (6), and spool valve (3) assembly downward to a new equilibrium position. This motion ports the power piston cylinder to the low pressure return port (7) of the spool valve, thus decreasing the power piston pressure and, consequently, the pump displacement. Shaft motion ceases when the spool valve is again nulled at the boiler discharge pressure design point.

A decrease in boiler discharge pressure will cause the power piston pressure to rise as the spool valve assembly moves upward to meter flow from the supply port (8) to the power piston chamber. The resulting control shaft motion will increase the pump displacement until a new equilibrium position is reached at the boiler discharge pressure design point.

5.1.3 Transmission

The transmission requirements are:

- Completely automatic operation.
- Permit expander to idle at zero vehicle speed.
- A minimum of two-speed ratios for improved performance.
- High efficiency under both part load and WOT conditions.
- Small enough to install in existing transmission tunnel with expander next to fire wall.

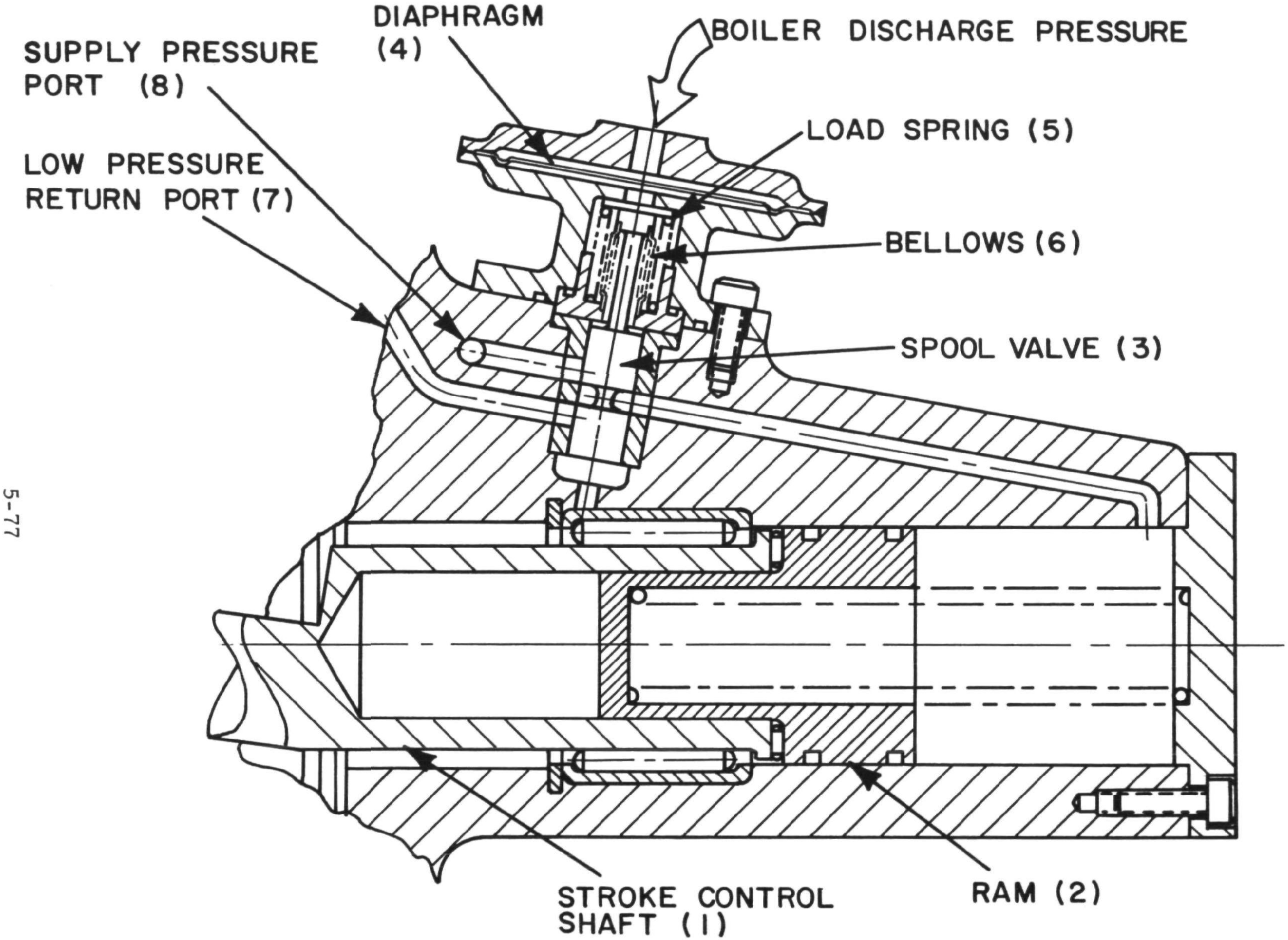

Figure 5.42 Feedpump Control.

- Must be available with minimum modifications required for integration with Rankine-cycle powerplant.

Several transmission choices are available. As part of the previous work, the Dana Corporation has developed, under subcontract, a layout design of a transmission specifically for the Rankine-cycle powerplant. The transmission design is described in Appendix VII. This transmission is a two-speed unit with a hydraulically-operated, slipping wet clutch used to permit both expander idle at zero vehicle speed and low speed operation of the vehicle. Above a speed of 8 mph, the clutch locks up; the transmission then operates as a direct-coupled unit to provide high efficiency and to use the desirable torque characteristics of the Rankine-cycle expander with variable cut-off intake valving. A control system was also designed for the low-speed slipping mode of operation, as well as to provide the desired shifting map between the two speed ratios. Since this is a special transmission and would require considerable development it is not presently being considered for the prototype cars. However, the performance and fuel consumption characteristics presented in Chapter 4 were calculated using the characteristics of this transmission.

Another alternative is use of a three-speed manual transmission. This approach has not been seriously considered due to the strong preference of the American consumer for an automatic transmission.

The requirement for minimum development cost thus restricted the choice to off-the-shelf or conventional transmissions. Two choices exist:

- Conventional three-speed automatic transmission with torque converter coupling.
- Conventional three-speed automatic transmission with fluid coupling.

To use the torque converter coupling with the relatively low-speed Rankine-cycle expander, a speed-up gear is required between the expander and torque converter to bring the speed of the input shaft to the torque converter to a level corresponding approximately to the speed of the I/C engine. The Ford Motor Company has made performance calculations for several different combinations to provide a basis for preliminary selection of the transmission and the results are summarized in Table 5.9. These calculations were based on the characteristics of Ford's C-4 automatic transmission. System No. 4 from Table 5.9 was selected as providing performance equivalent to that obtained with the 351 CID I/C engine without exceeding the maximum torque rating of the C-4 transmission. The performance of the fluid coupling was definitely poorer, at least for the conditions run. In Figure 5.43, the WOT drive shaft torque output of the RCS with System No. 4 is compared with that of the I/C engine. The RCS system produces higher torque output below about 30 mph; above 30 mph, the torque is slightly less than that obtained with the 351 CID I/C engine. The characteristics of the selected transmission are summarized in Table 5.10; a layout drawing of the overdrive-gear torque converter portion of the transmission is given in Figure 5.44.

5.1.4 Rotary Shaft Seal

The system has been designed so that only one dynamic shaft seal, that on the 3" diameter expander shaft at the rear of the expander,

TABLE 5.9

WOT ACCELERATION PERFORMANCE OF 1972
FORD GALAXIE - EFFECT OF TRANSMISSION

CALCULATIONS BY FOMOCO

Performance Weight = 4486 lbs

Transmission Rates = 2.40 Low, 1.47 Intermediate, 1.00 High

Rear Axle Ratio

System No.	Engine	Axle Ratio	Speedup Gear Ratio	Coupling	0-60 MPH Time, Sec.	0-10 sec Distance ft.	25-70 MPH Time, Sec.	50-80 MPH Time, Sec.
1	351 IC	2.75	1.00	12" D Torque Converter	14.6	406	15.5	16.0
2	RCS	2.75	1.75	12" D Torque Converter	13.8	450	16.1	17.0
3	RCS	2.75	2.00	12" D Torque Converter	14.1	443	16.0	17.2
4	RCS	2.75	2.25	12" D Torque Converter	14.4	435	16.3	17.2
5	RCS	1.50	1.00	11" D Fluid Coupling	16.2	389	18.5	20.1

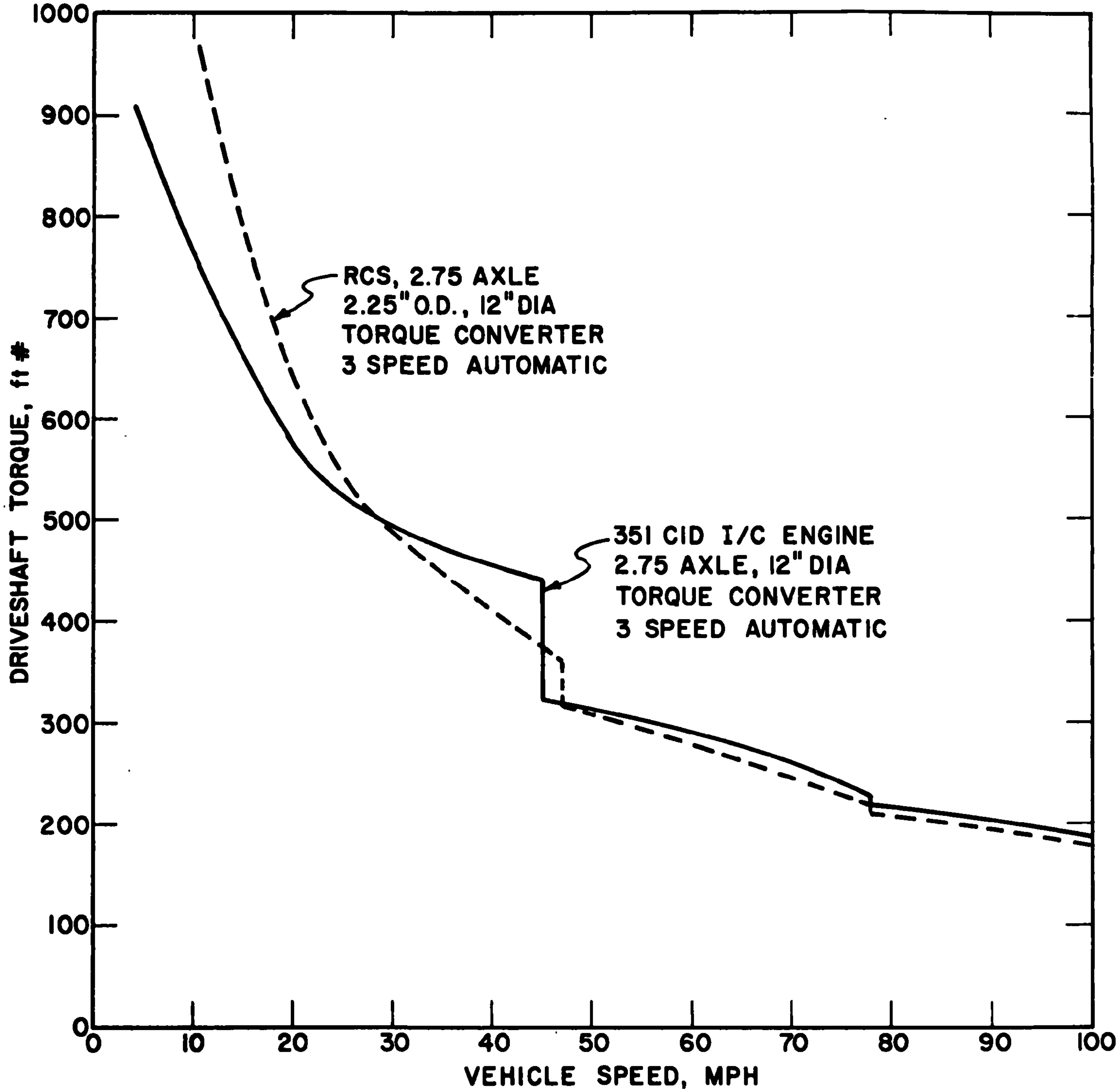

Figure 5.43 Comparison of Wide Open Throttle Torque with Torque Converter Coupling and Three-Speed Automatic Transmission, TECO RCS and 351 CID Engine.

TABLE 5.10

CHARACTERISTICS OF SELECTED TRANSMISSION

Overdrive Gear Between Expander and Torque Converter	
Type	Planetary
Gear Ratio	1:2.25
Ford C-4 Transmission	
12 inch diameter Torque Converter	
Gear Ratios	2.40 Low 1.47 Intermediate 1.00 High
Driveline	
Standard Ford Motor Company Axle Ratio	2.75:1

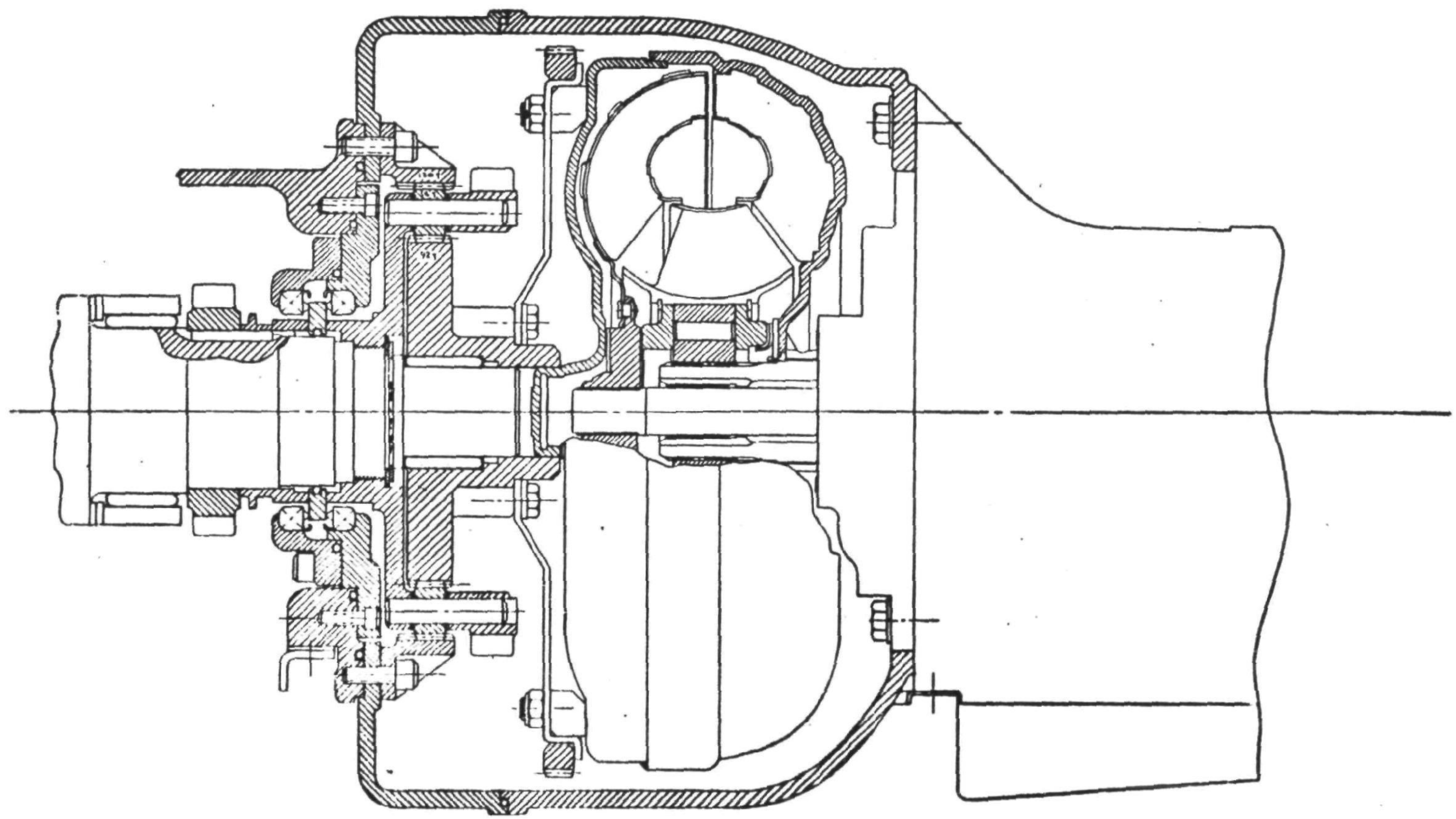

Figure 5.44 Layout of Torque Converter and Overdrive Gear for Transmission (12" diameter torque converter).

is required. During system shutdown, the crankcase pressure is subatmospheric; there is therefore a tendency for air to leak into the system. During operation, the crankcase pressure will vary from subatmospheric to approximately 100 psia, depending on operating conditions and the ambient temperature level. To positively prohibit either air leakage into the system or working fluid leakage out of the system, a double seal is used with a prepressurized buffer fluid between the two seals, as illustrated in Figure 5.45. The pressure of the buffer fluid between the two seals is maintained above the crankcase pressure at all times so that any leakage through the inboard seal is the buffer fluid leaking into the crankcase.

The buffer fluid is the lubricant used in the system. The buffer pressure is always maintained above atmospheric pressure by the springs in the buffer fluid reservoir so that any leakage through the outboard seal is leakage of lubricant to the atmosphere. Leakage through the outboard seal is collected in the power take-off housing on the rear of the expander and can be drained at intervals if required. To reduce the pressure differential across the inboard seal in order to minimize buffer fluid leakage into the system, the buffer fluid pressure is controlled by the crankcase pressure, as illustrated in Figure 5.45. Provision is also made for charging make-up oil to the reservoir when required.

The actual seal construction is illustrated in Figure 5.46. Two face seals are used as illustrated, with the hardened steel mating ring rotating with the shaft and two stationary, spring-loaded carbon rings. This seal is manufactured by the Chicago Rawhide Company and is a modification of a standard seal. Testing on this seal, as

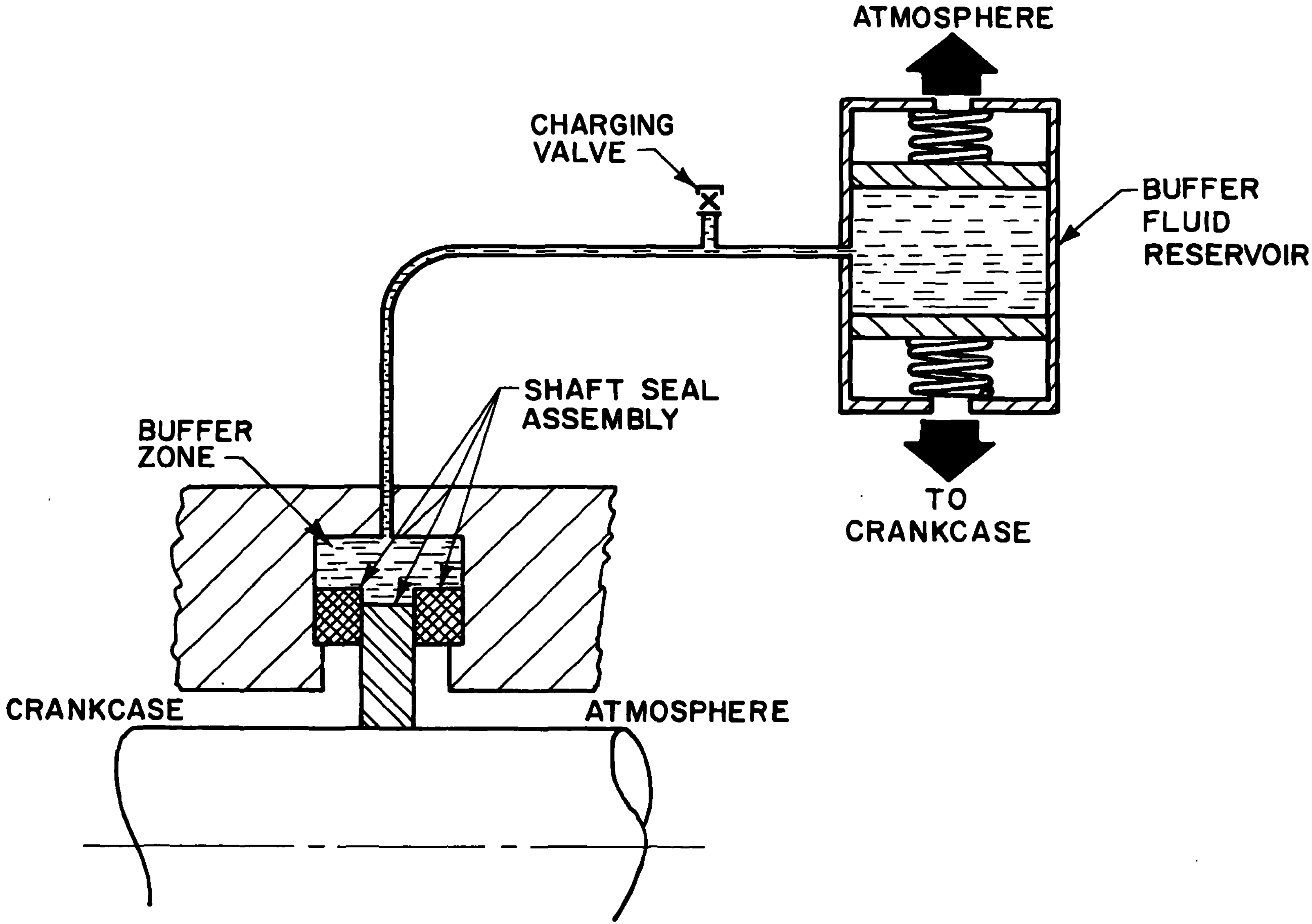

Figure 5.45 Double Seal and Buffer Fluid Concept.

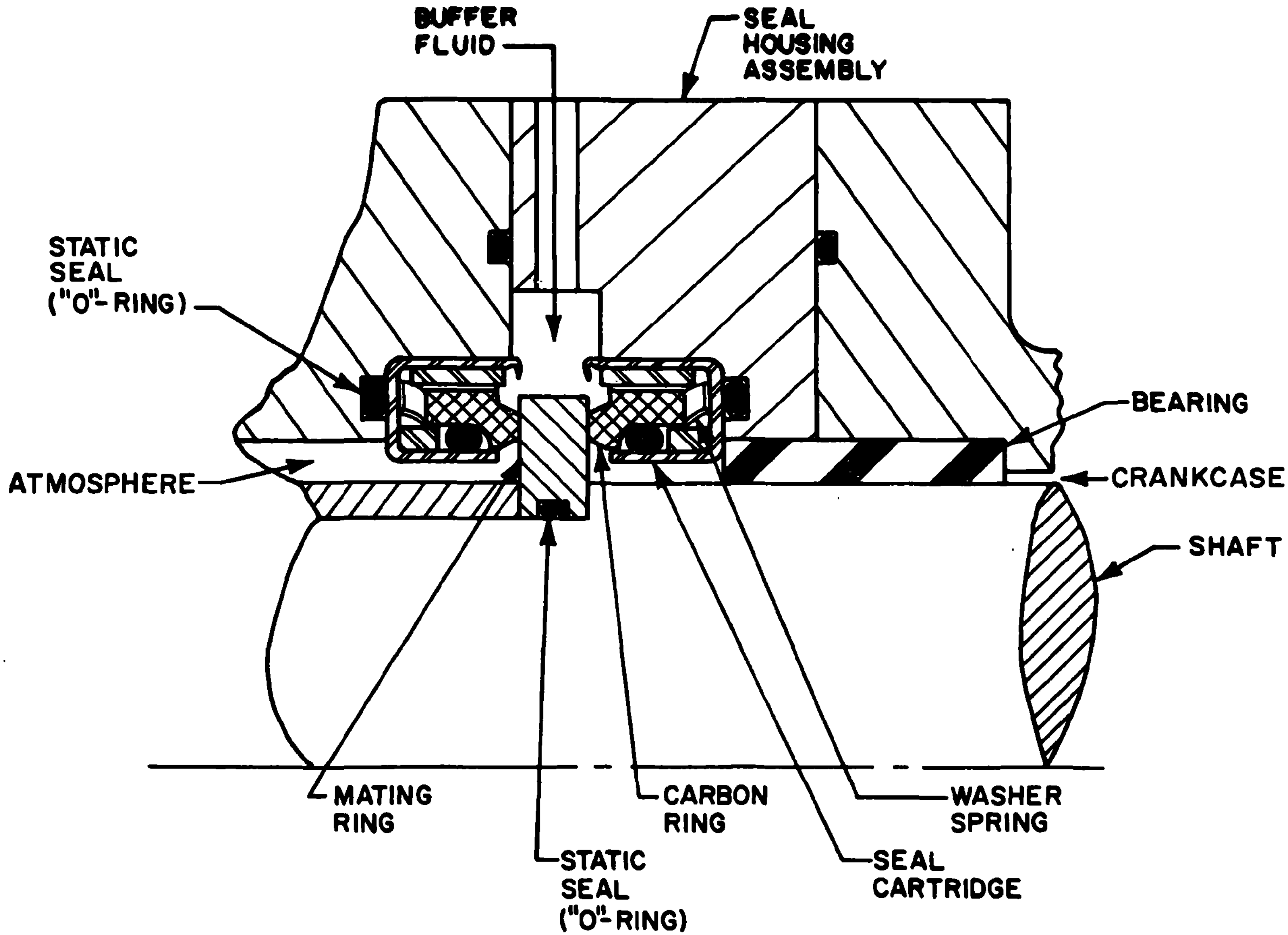

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Figure 5.46 Layout Drawing of Chicago Rawhide Double-Shaft Seal.

well as on an alternative type of seal manufactured by the Crane Company, was carried out at Thermo Electron Corporation under simulated operating conditions; the detailed test program and data are presented in Appendix III. The Chicago Rawhide seal was selected for the preprototype design because of its short length and acceptable leakage rates. In Table 5. 11, measured leakage rates are presented over a 3187-hour test with this seal. The leakage rate when the system is not operating is much lower (by a factor of approximately 10) than when the seal is operating. The measured leakage rates are considerably lower than those that can be tolerated in the system.

The buffer fluid reservoir design is presented in Figure 5. 47. Bellofram seals are used as illustrated to provide a leaktight reservoir and to insure that the buffer lubricant is maintained free of dissolved air. A level indicator is used to indicate the buffer fluid level when in the static condition so that make-up buffer lubricant can be charged to the reservoir when required.

TABLE 5.11

ROTARY SHAFT SEAL TEST

SEAL TYPE	TYPE OF OPERATION	ELAPSED TIME (HOURS)	LEAKAGE RATE (PINTS/1000 HOURS)		
			TOTAL	CRANKCASE	OUTBOARD
CHICAGO RAWHIDE	CONTINUOUS	3187	0.438	0.183	0.255

LEAKAGE RATE GOALS:

TOTAL (BUFFER) LEAKAGE RATE: 2/3 PINT/1000 HOURS

OUTBOARD LEAKAGE RATE: 1/2 PINT/1000 HOURS

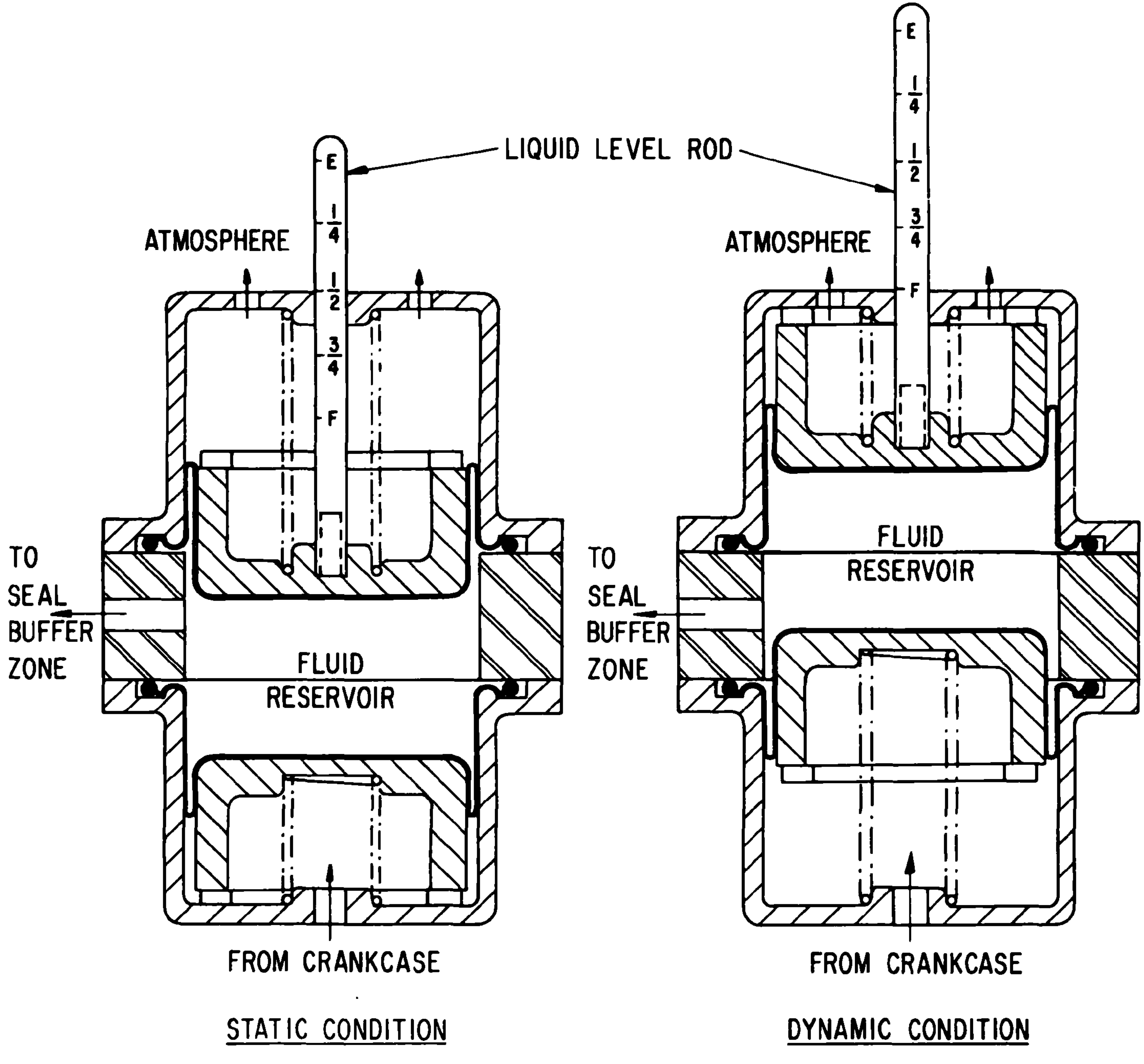


Figure 5.47 Buffer Fluid Reservoir.

5.2 COMBUSTION SYSTEM - BOILER SUBASSEMBLY

In Figures 5.48 and 5.49, the complete assembly cross sections of the combustion system-boiler subassembly are presented. Two partial sectional views are presented in Figure 5.50. These drawings include the boiler, burners, combustion air blower and motor drive, atomizing air compressor, and fuel/air controls. Two burners firing upward are used with control components and the combustion air blower packaged between the two burners. Two burners operating in parallel are used in order to obtain a low burner pressure drop within the packaging constraints for the system and to facilitate obtaining uniform combustion gas flow through the rectangular tube bundle. The boiler tube bundle is positioned at the top of the subassembly because of packaging constraints. Combustion gases flow upward through the boiler tube bundle, leaving at a temperature of 600°F or less, depending on the system power level. The exhaust gases are collected in the outlet chamber and ducted to the bottom of the engine compartment and to the rear of the vehicle if required. Provision is also made (not shown in these figures) for directing part of the exhaust gases to the combustion blower inlet for control of NO_x emissions by exhaust gas recirculation. The combustion chamber walls are air-cooled by the incoming combustion air, as illustrated.

A prime consideration in design of the burner-boiler has been that of minimizing the total power required to operate the unit within the packaging constraints, not only to limit the parasitic load on the system, but also to make practical the use of completely electric drive for the combustion blower and atomizing air compressor. For rapid cold startup, it is necessary to operate the burner at the

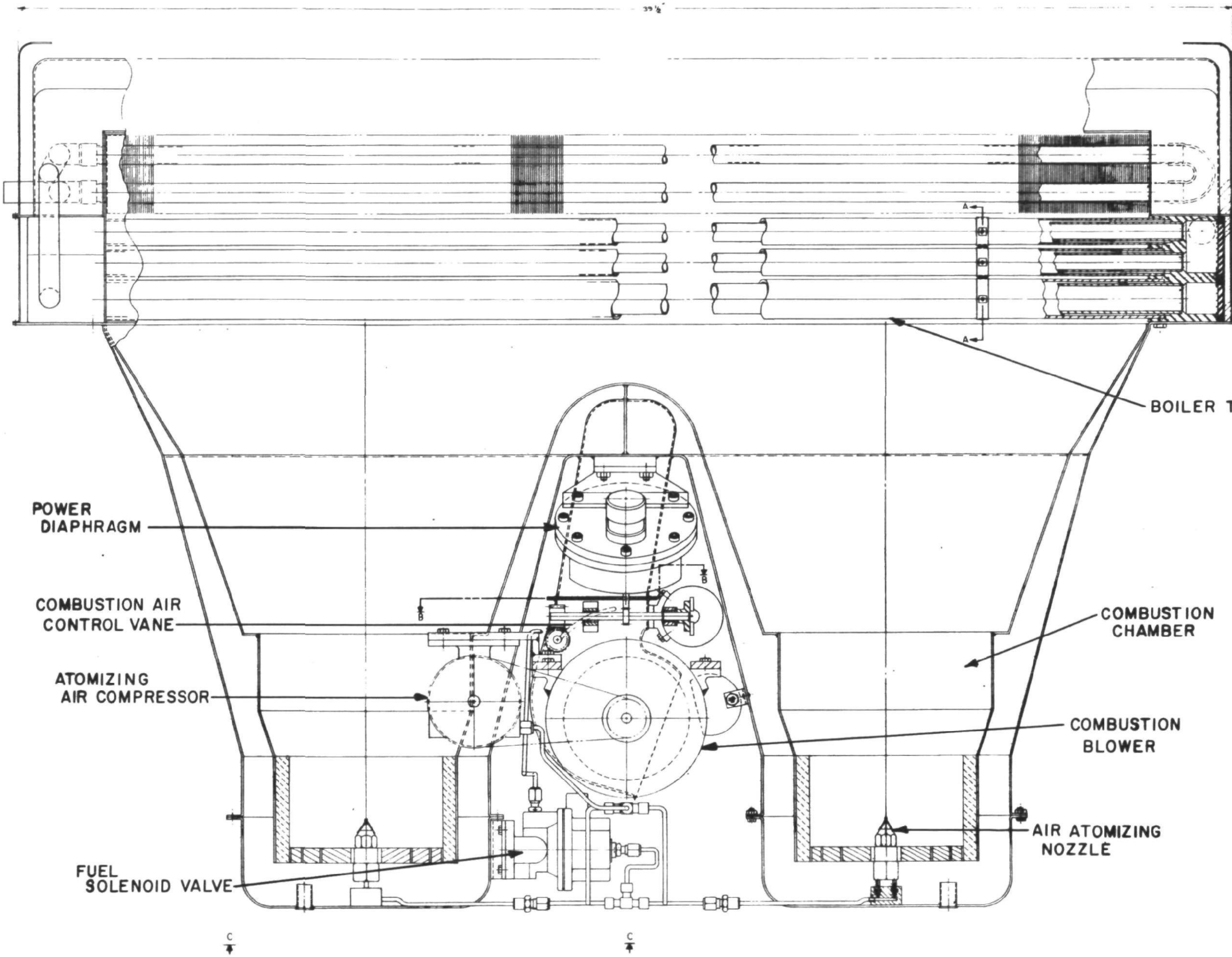


Figure 5.48 Front View of Combustion System-Boiler Subassembly

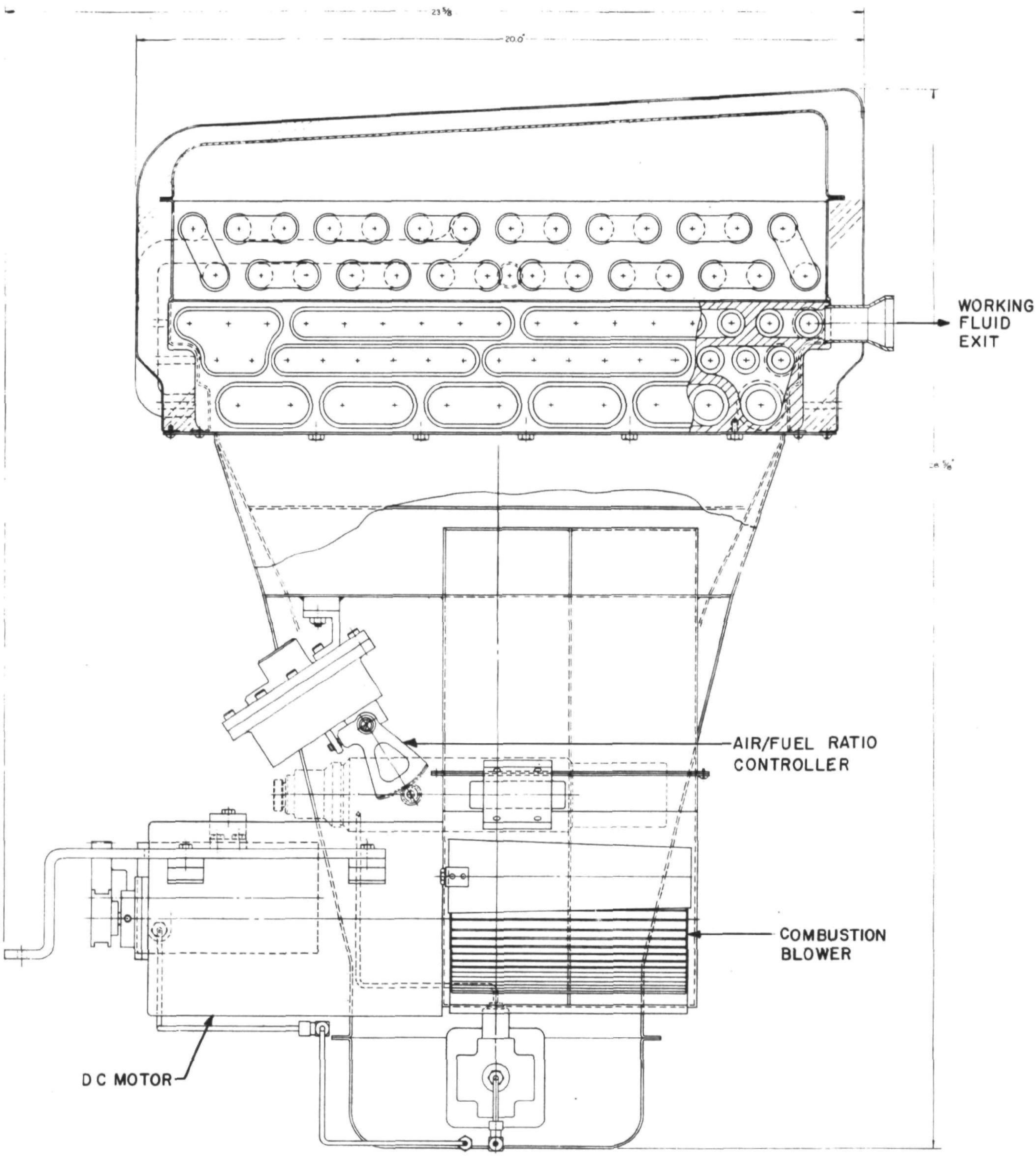


Figure 5.49 Side View of Combustion System-Boiler Subassembly.

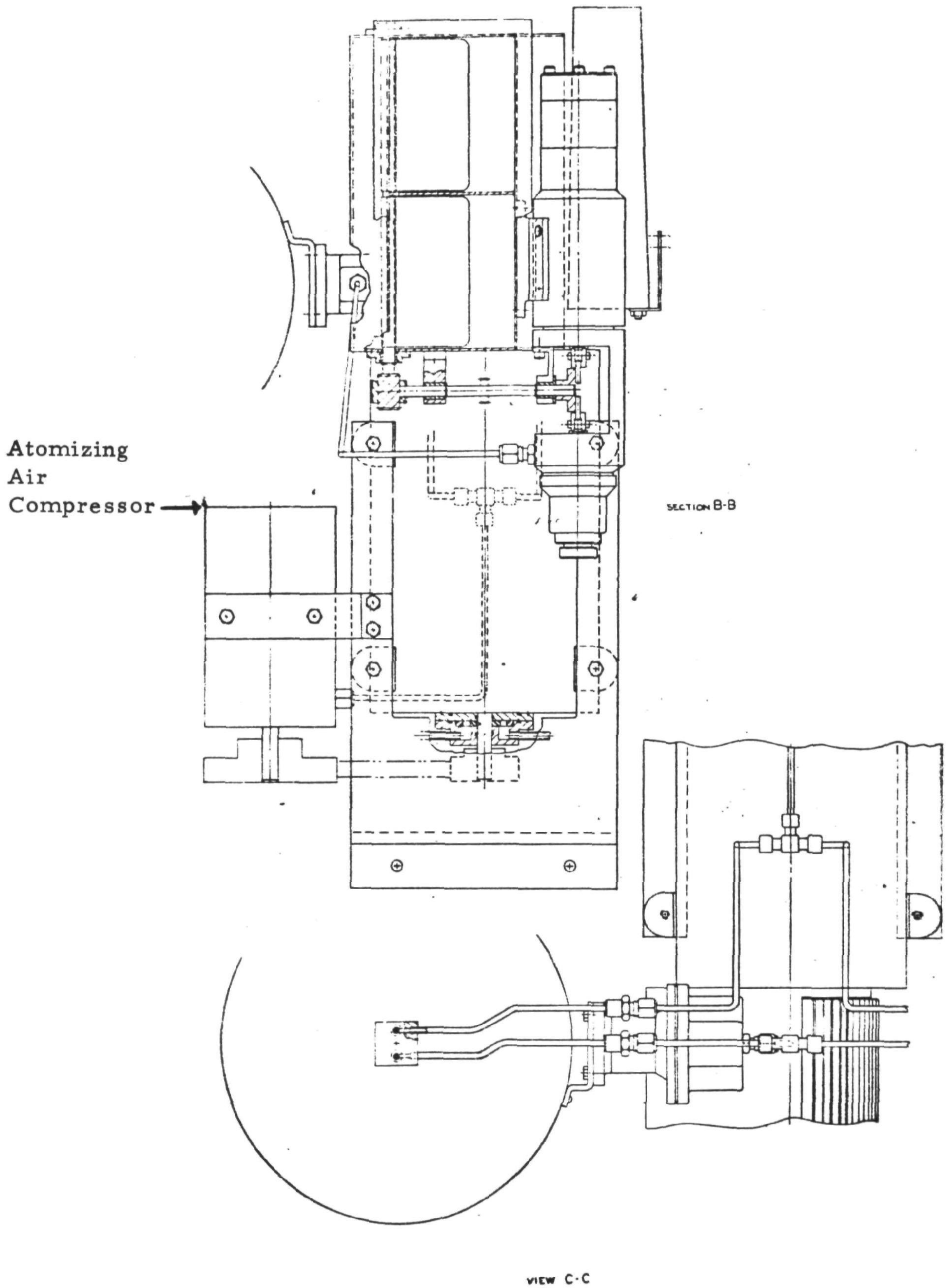


Figure 5.51 Views of Layout Drawing of Figure 5.48 Illustrating Position of Fuel/Air Control.

maximum burning rate on startup. Since the only power available on startup is that stored in the battery, it is necessary that both the atomizing air compressor and combustion blower be driven by electric motors. Another consideration is design of the combustion blower so that the shaft power required decreases as the burner is operated at low firing rates corresponding to part-load operation.

5.2.1 Boiler Design

The boiler design point requirements and characteristics are summarized in Table 5.12. In Figure 5.51, a cross section of the boiler tube bundle across the narrow dimension is presented; Figure 5.52 is Section A-A of Figure 5.51. In Figure 5.53, cross sectional views of the tube bundle across the wide dimension are presented.

The organic flow path through the boiler tube bundle is presented in Figure 5.54. The liquid organic flow enters the preheat stage at the middle of tube row No. 2. The flow is split into two parallel passes through the preheat stage in order to use a smaller tube diameter, thereby reducing the preheat stage size while still maintaining an acceptable organic pressure drop. Flow through the preheat stage is single phase only, so that flow instability with parallel flow is not a problem. The combustion gases exit from the preheat stage, which is the lowest temperature portion of the boiler, so that the boiler efficiency can be maximized without use of air preheat and with a reasonable overall size. The organic flow enters tube row No. 2 rather than No. 1 in order to have the lowest organic temperature where the combustion gases enter the preheat stage, and thus minimize the possibility of overheating the organic.

TABLE 5.12

BOILER DESIGN POINT CHARACTERISTICS

Combustion Rate	2.78 x 10^6 Btu/hr
Heat Transfer Rate to FL-85	2.25 x 10^8 Btu/hr
Fuel	JP-4 (HHV = 20096 Btu/lb)
Fuel Flow Rate	138.1 lbs/hr
Primary Air Flow Rate	2468 lbs/hr
Recirculation Gas Flow Rate	521 lbs/hr
Air:Fuel Ratio	17.85:1
Combustion Gas Temperature at Inlet	2975°F
Combustion Gas Temperature at Outlet	600°F
Efficiency at 100% Load	0.81
Efficiency at 10% Load	0.88
Combustion Gas Pressure Drop	3.40" W. C.
FL-85 Flow Rate	9860 lb/hr
FL-85 Temperature at Inlet	287°F
FL-85 Pressure at Inlet	826.5 psia
FL-85 Temperature at Outlet	558°F
FL-85 Pressure at Outlet	700 psia
Maximum Tube Wall Temperature on FL-85 Side	569°F
Maximum Fin Tip Temperature	877°F
Maximum Water Jacket Pressure at 100% Load	1486 psia
Maximum Water Jacket Pressure at 10% Load	1130 psia
Core Dimensions	34 x 18 x 6.3 inches
Overall Dimensions	40 x 20 x 8.3 inches
Core Weight	226 lbs
Weight of Water	8.15 lbs
Total Weight (Burner, Boiler, Insulation, Headers, Water, Expansion Tanks, etc.)	330 lbs

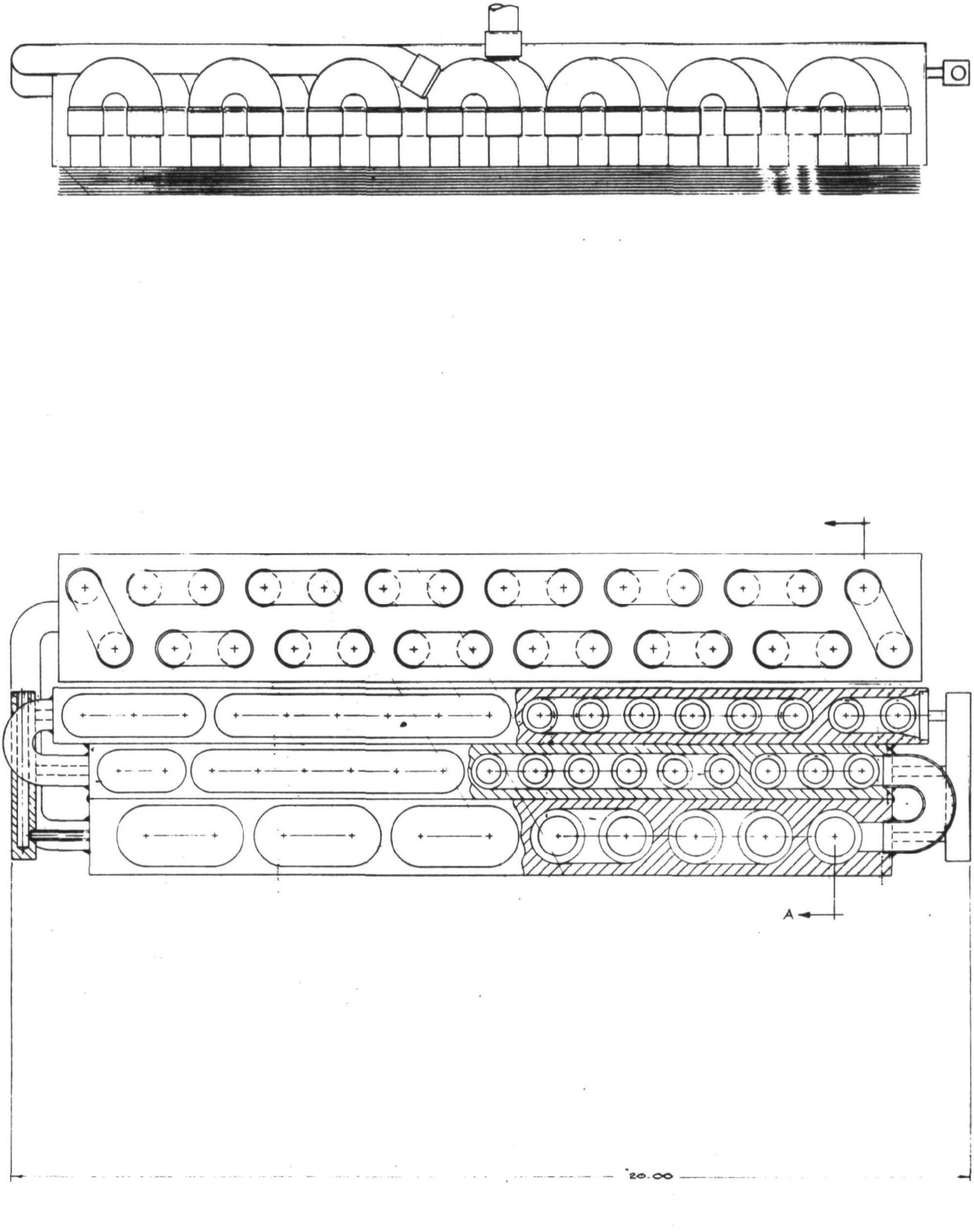


Figure 5.51 Cross Section of Boiler Tube Bundle Across Narrow Dimension.

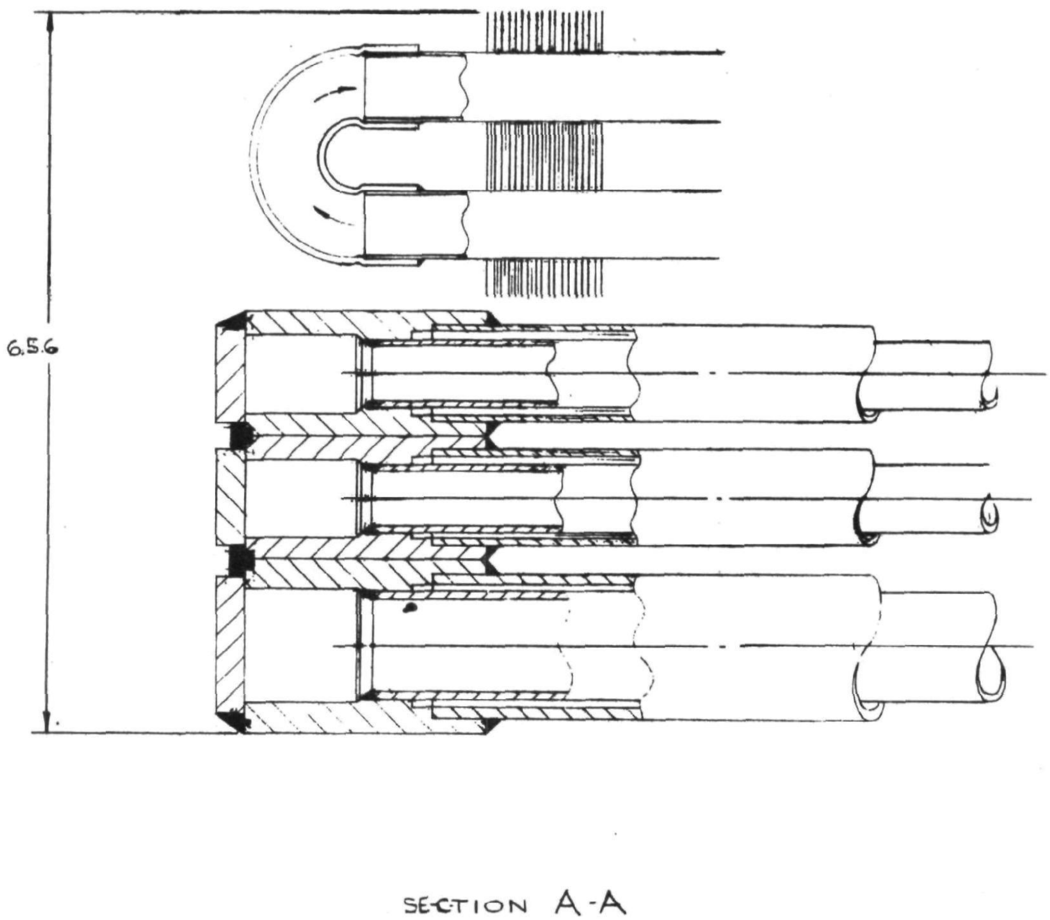
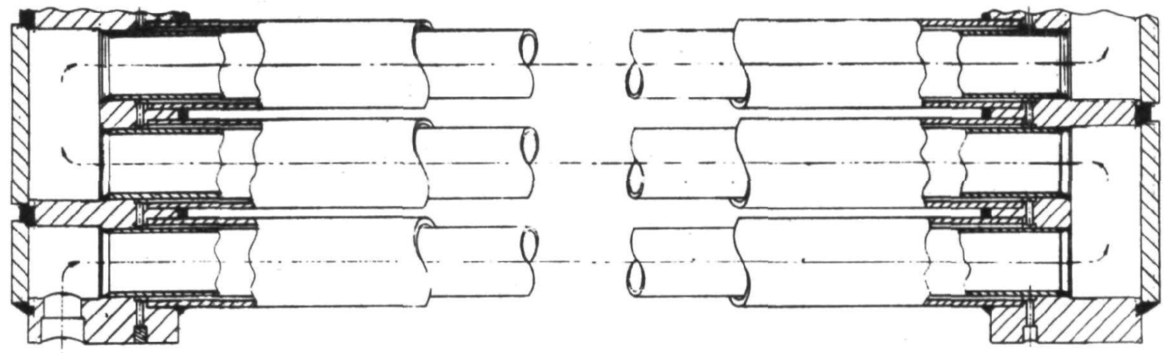


Figure 5. 52 Section A-A of Figure 5. 51 Illustrating Header Construction.



SECTION B-B

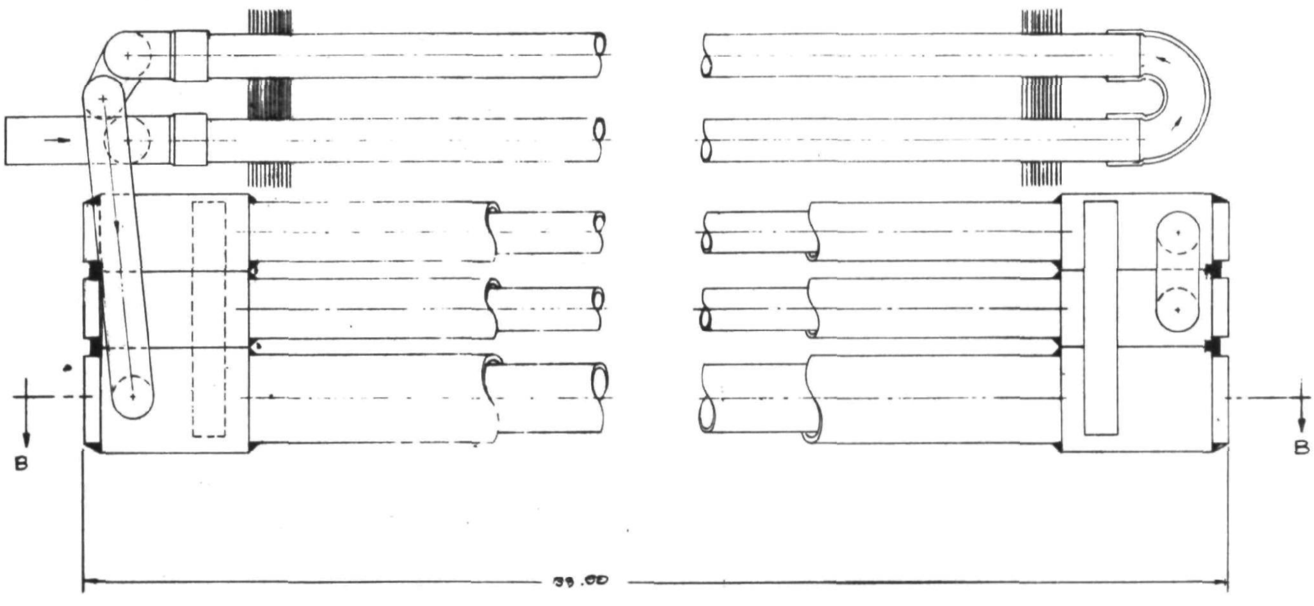


Figure 5. 53 Cross Section of Boiler Tube Bundle Across Wide Dimension.

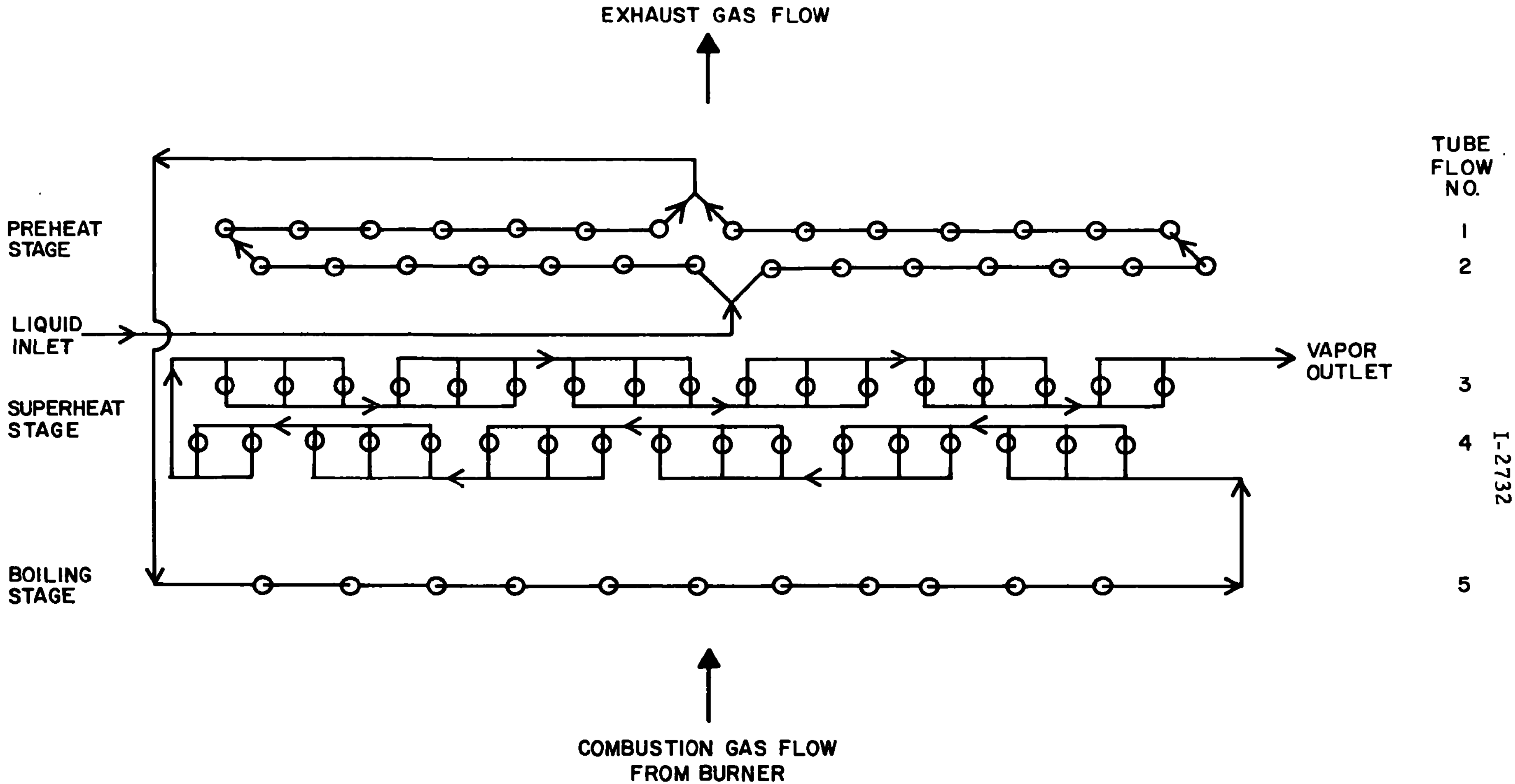


Figure 5.54 Flow Path Schematic of Working Fluid Through Boiler.

The flow leaves the preheat stage close to the boiling temperature and goes to the boiling stage or tube row No. 5. The boiling stage is monotube, with straight-through flow to eliminate any possibility of flow instability. This stage is located where the combustion gases enter, since the organic heat transfer coefficient is highest in the boiling stage and it is thus located where the combustion gas temperature is highest.

The working fluid leaves the boiling stage as high-quality vapor and goes to the first row of the superheat stage (tube row No. 4). Parallel flow is used in the superheat stage, as illustrated in Figure 5.55, since flow is single phase only and no problems with flow instability will be encountered. As in the preheat stage, parallel flow permits use of a small diameter tube with an acceptable pressure drop and minimizes the boiler size. The flow passes through three tubes in parallel except at the exit of each tube row (Nos. 3 and 4), where parallel flow through two tubes occurs. The superheated vapor leaves the boiler from tube row No. 3 and goes directly to the expander.

The stagewise characteristics of the boiler at the design point conditions are presented in Table 5.13. The boiling and superheat stages of the boiler are bare tube bundles with a water jacket buffer to positively prevent either gross or local overheating of the organic working fluid. Dual tube boiler construction is used for these stages, as illustrated in Figure 5.55, with organic flowing through the inner tube. The annular space between the tube bundles is sealed and filled with water, with an external thermal expansion tank to permit thermal expansion of the water. Heat transfer from the hot combustion gases, flowing around the outer tube, to the inner tube carrying the organic

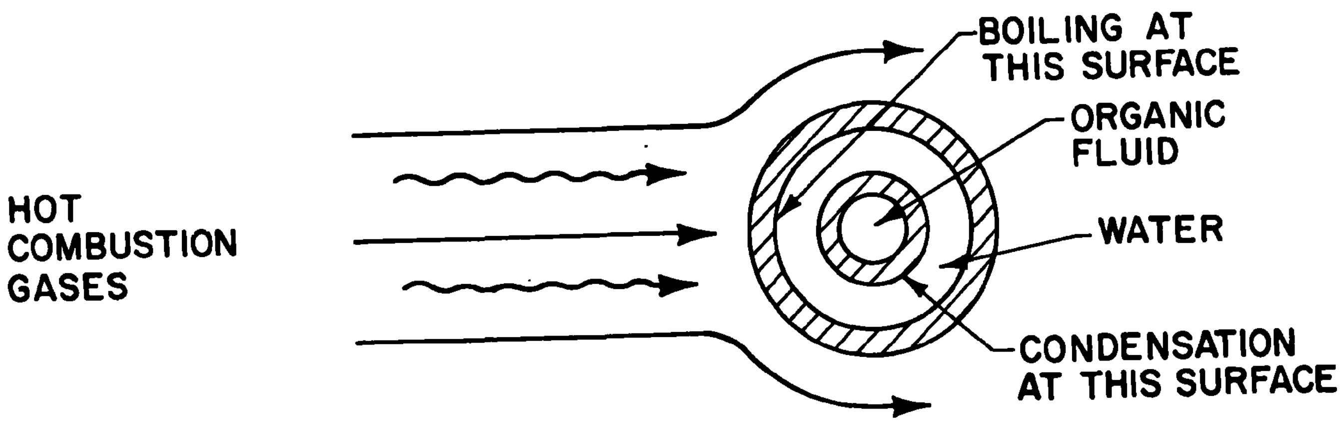


Figure 5.55 Illustration of Water Jacket Operation.

TABLE 5.13

BOILER DESIGN POINT CHARACTERISTICS

Stage	Tube Row No.	Heat Transfer Rate Btu/hr	Combustion Gas Temperature		FL-85 Temp.		Pressure Drop		Water Jacket Pressure psia	Mass of Core Without Water lbs.	Mass of Water lbs
			Inlet °F	Outlet °F	In °F	Out °F	Gas Side in w.c.	Organic Side psi			
Boiling	5	625,000	2975	2371	437	445	2.38	40	1470	66	2.7
Superheat-I	4	444,800	2371	1939	445	502	0.74	30	1405	40	2.7
Superheat-II	3	224,400	1939	1709	502	550	0.11	32.5	1431	40	2.7
Preheat	1 and 2	958,000	1709	607	287	437	0.17	24	–	80	–
Total		2.25×10^6					3.40	126.5		226	8.1

TUBE SPECIFICATIONS

Boiling Stage

Inner Tube - 1.00" O.D., 0.083" wall
Outer Tube - 1.313" O.D., 0.093" wall

Superheater I and II Stages

Inner Tube - 5/8" O.D., 0.049" wall
Outer Tube - 7/8" O.D., 0.058" wall

Preheat Stage

5/8" tube expanded (0.577" O.D., 0.035" wall)
18 fins/inlet (rippled)

Tube and Header Material = AISI 4130 Steel occurs by boiling of the water on the inner surface of the outer tube and condensation of vapor on the outer surface of the inner tube. The annular gap width is approximately 60 mils. The organic tube wall temperature can thus not exceed the saturation steam temperature corresponding to the pressure in the water jacket; the water jacket pressure provides a convenient and sensitive means of controlling the maximum temperature to which the organic is exposed. The water jacket pressure is also used in the startup sequencing to indicate when the boiler has been heated sufficiently to start cranking the expander-feedpump assembly. Use of the water jacket permits safe system startup, even if the boiler is initially dry (empty of working fluid). Freezing of the water buffer is prevented through use of an inorganic salt as an antifreeze agent.

The superheat and boiling stages are brazed construction, with the tubes furnace-brazed into machined steel headers, using a nicro-coat braze alloy. Each row of these stages is constructed separately to facilitate leak-checking and to reduce throwaway cost if any irreparable leak is encountered. The headers are designed, however, so that any leaks can be repaired. The brazed tube rows are finally connected by welding connector tubes between the rows. The steps for fabrication are summarized in Table 5.14. For external corrosion protection in the test boilers, it is planned to coat the entire assembly externally with a brazing alloy.

The preheat stage is a conventional finned tube heat exchanger. A water jacket is not used in this stage, since the combustion gas temperature is relatively low and the tubes are filled with liquid Fluorinol-85; both of these features minimize the possibility of

TABLE 5.14

STEPS FOR BOILER FABRICATION

1. Apply the braze compound and assemble the tubes and headers for each row.
2. Furnace - Braze each row subassembly.
3. Leak test (Mass Spec.) through water jacket charging port. (In the event of a leak, isolate the leak location, rebraze with a lower melting nicrobraze and leak test again.)
4. Weld the end caps.
5. Leak test the inner tube circuit for leaks in the welds.
6. Normalize and stress relieve.
7. Weld row connections.

Materials are:

- Tube and header material: - AISI 4130 Steel (0.9% Cr, 0.27% Mo)
- Tubes coated with nicrocoat-6 for corrosion protection and thermal fatigue characteristics.
- Nicrobraze-200 as brazing alloy (in the event of a leak use nicrobraze-130).

overheating in this stage. The fins and tubes are made of AISI 4130 steel, with brazing used to insure good thermal contact between the fins and tubes as well as to provide external corrosion protection.

5.2.2 Burner and Fuel/Air Supply Designs

The burner design is presented in the layout drawings (Figure 5.48); the burner design point characteristics are summarized in Table 5.15. The burner design is based on the burners tested at Thermo Electron Corporation for emission characteristics, as outlined in Appendix VI. The tested burners were designed for a 100 shp system, so that scale-up to the size for 131.1 hp was required. Some modification to the combustion chamber was required for packaging and for insuring uniform flow into the boiler tube bundle. Steady-state testing has recently been initiated at Thermo Electron Corporation on the full-size burner for 131.1 hp system.

With reference to Figure 5.48, the cylindrical combustion chamber is air-cooled, with the combustion air entering at the top of the burner and flowing down the space between the combustion chamber and outer wall of the burner. The combustion chamber wall operates at approximately 1500°F. At the bottom of the burner, the air flow direction reverses and flows through swirl vanes into the combustion chamber. The fuel nozzle is an air-atomizing Sonicore nozzle, of the same type as used in the burner on which the emission measurements were made. The section of the combustion chamber close to the nozzle is lined with a ceramic insert for improved combustion and to prevent local overheating of the combustion chamber wall in this region at low firing rates. The combustion chamber wall is flared outward, as illustrated in

TABLE 5.15

BURNER DESIGN POINT CHARACTERISTICS

Combustion Rate (HHV)	2.78×10^6 Btu/hr
Number of Burners	2
Fuel	JP-4 (HHV = 20098 Btu/hr)
Fuel Flow Rate	138.1 lbs/hr
Excess Air	20%
Air Flow Rate	2468 lbs/hr
Recirculation Gas at 600°F	20%
Recirculation Gas Flow Rate	521 lbs/hr
Total Combustion Gas Flow Rate	3127.1 lbs/hr
Combustion Gas Adiabatic Temperature	2975°F
Combustion Gas Pressure Drop	4.5" W. C.
Ideal Combustion Gas Fan Power	1.14 hp
Ideal Atomizing Air Pumping Power	0.26 hp
Ideal Fuel Pumping Power	0.005 hp
Combustion Space Rate	1.5×10^6 Btu/hr ft^3
Overall Dimensions	34 x 17 x 19.5 inches

Salient Features

1. Sonicore-air atomizing nozzle for fuel spray.
2. Swirl blades for enhanced air-fuel mixing.
3. Recirculation of exhaust gases to control oxides of nitrogen.

Figure 5.48, above the main combustion zone to diffuse the combustion gases and to provide a uniform gas velocity over the entire area of the rectangular boiler tube bundle.

The burner can also be constructed with ceramic lining the entire combustion chamber wall and without air cooling other than natural convection. Air-cooling was utilized, however, to minimize the weight of the burners. The combustion chamber wall is constructed of Type 316 stainless steel; the outer can operates at low temperature and is constructed of carbon steel.

The experimental testing indicates satisfactory operation of this type of burner over a turndown ratio of 20:1 (70,000 Btu/hr to 1.4×10^5 Btu/hr). This turndown ratio should be adequate for the system. As indicated earlier, packaging constraints and the requirement for minimum burner pressure drop required the use of two burners as illustrated. Two major problem areas must be anticipated in operation of two burners in parallel:

- Unbalance of fuel and/or air flow between the two burners, resulting in non-uniform combustion gas flow through the boiler and off-optimum fuel/air ratio, causing excessive emissions.
- Occurrence of flow instability between the two burners, with pulsing of the burning rate in each burner.

To insure proper balance of fuel/air flow between the two burners, the air and fuel flow paths to the burners have been designed to be symmetrical so that the flow paths from the common fuel control and common air control are identical. In construction of the two burners, the characteristics of the two fuel nozzles and the two combustion chambers will be matched to insure identical fuel/air flow to the two burners. This procedure will provide adequate balance between the two burners. If necessary, trim controls can also be incorporated in the fuel and air supplies to each burner.

To evaluate flow instability, the Northern Research Company of Cambridge, Massachusetts, provided a stability analysis of the system. This study involved a computer stability study using an existing program. The complete system was divided into zones for the purpose of the calculation. A slight air pressure perturbation was then introduced in one burner, and the calculation performed to indicate if the pressure disturbance was damped or continued to oscillate and grow in magnitude. It was assumed that the fuel flow to each burner was not influenced by the pressure oscillations, based on the fuel control design. The conclusions reached were:

- The hot gas side of the burner configuration is stable as designed. While the calculation indicated the configuration is stable, the partition between the two burners should be carried as close to the boiler tube bundle as possible to reduce interplay between the two burners on the hot side and reduce the potential for difficulties.

- The main source of instability results from interplay between the two burners on the cold air side if a common air blower and air supply is used for both burners. The calculations indicated this procedure would result in unstable operation. Use of a separate air supply to each burner eliminates this source of instability.

- A blower whose head-flow characteristic has a negative slope over the burner operating range should be used.

These conclusions were used in the final design iteration, as presented in Figures 5.48 and 5.49 and in selection and design of the combustion air blower.

The combustion air blower requirements and characteristics are summarized in Table 5.16, and the design is illustrated in the layouts of Figures 5.48 and 5.49. The type of blower selected is the transverse blower, which provides uniform air flow across the impeller length. A splitter is constructed as part of the housing to separate and isolate the air flow to each of the burners while still retaining use of a single blower wheel and modulating control. Two separate ducts carry the air to the top of each burner. The blower is motor-driven at constant speed, with air flow controlled by means of a pivoted control vane in the exhaust duct. Pivoting of this vane moves the vortex position in the impeller, thereby modulating the air flow. This type of blower and control results in reduced shaft power at low flow rates with constant blower rpm, thereby reducing the blower parasitic load at low firing rates corresponding to part-load system operation.

The characteristics of the combustion blower motor are summarized in Table 5.17. These characteristics are based on a commercially-available dc motor. The fuel pump and atomizing air compressor are also driven by this motor. The fuel pump, whose characteristics are summarized in Table 5.18, is the gerotor type with standard rotor elements and a special housing. This pump provides

TABLE 5.16

COMBUSTION AIR BLOWER REQUIREMENTS AND CHARACTERISTICS

Design Point Performance Requirements

Mass Flow Rate of Air at 60°F	2468 lbs/hr
Mass Flow Rate of Exhaust Gas at 600°F	521 lbs/hr
Mean Mix Temperature	165°F
Volumetric Flow Rate at 165°F	770 CFM
Pressure Head	9 inches w. c.
Ideal Fan Power	1.14 hp
Blower Efficiency	50%

Requirements for Stability and Part-Load Operation

Negative slope on head flow characteristics required independent air supply to each burner.
Constant speed operation with control for modulating air flow/power characteristic such that reducing air flow reduces shaft power input.

Design and Construction Specifications

Type	Transverse Flow
Impeller Length	7 inches
Impeller Diameter	4 inches
Overall Housing Dimensions	7 x 7 x 6 inches
Impeller Construction	Die Cast Aluminum

Integral flow splitter used to isolate air flow to each burner.
Pivot control vane in exhaust duct to modulate air flow by modifying blower characteristic.
Zero flow shaft power = 20% of design point power.

TABLE 5.17

DC MOTOR FOR COMBUSTION SYSTEM

PERFORMANCE REQUIREMENTS

Rpm	7000
Shaft Horsepower	2.68 hp
Voltage	12 Vdc
Efficiency	70%

SPECIAL REQUIREMENTS AND QUALITY ASSURANCE

Open-air frame motor.

Provide insulation to work in ambient temperature range of -20°F to 200°F

Continuous duty, shunt wound.

DESIGN AND CONSTRUCTION SPECIFICATIONS

Diameter	5-9/16 inches
Length	8-1/8 inches
Weight	18 lbs

TABLE 5.18

FUEL PUMP DESIGN POINT CHARACTERISTICS

PERFORMANCE REQUIREMENTS	
Fuel	JP-4
Flow Rate	138 lb/hr
Delivery Pressure	25 psig
Ideal Pumping Power	0.005 hp
Pump Efficiency	70%
Delivery Pressure Control	Pressure Bypass Valve with Accumulator
MATERIAL AND PROCESSES	
Rotor Elements	Steel
Housing	Cast Iron
MAXIMUM OVERALL DIMENSIONS	
1.5 inches diameter x 0.5 inch thick	

constant pressure (25 psig) fuel to the burner fuel metering valve as well as to the combustion air servo-controller; fuel is used as the actuating medium for these hydraulic-mechanical controllers.

The atomizing air compressor characteristics are summarized in Table 5.19. The compressor is identical in construction to commercially-available vane compressors; however, a special size is required for this application.

5.2.3 Fuel/Air Controller Design

The location of the fuel/air controls are given in Figures 5.48, 5.49 and 5.50, and the functional schematic is presented in Figure 4.2 of Chapter 4. In this section, the detailed layouts of the control components are presented. These component designs are similar to those used in the transient burner emission measurements described in Appendix VI, with modifications made for the different blower characteristics and for packaging in the space between the two burners.

The combustion air servo-control is illustrated in Figure 5.56. This unit controls the fuel pressure applied to the diaphragm actuator illustrated in Figure 5.60. Motion of this actuator positions the blower vane controlling air flow. The fuel pressure to the diaphragm is controlled by the spool valve with constant 25 psig fuel pressure applied at inlet port A. The motion of the spool valve is controlled by a set of diaphragms actuated by pressure signals from the boiler inlet flow rate and boiler outlet temperature and by a cam-operated roller actuated by a cam on the shaft connected to the blower vane. The cam-operated roller provides the feedback in the displacement-controlled servo loop. The orifice ΔP generated by the organic flow rate

TABLE 5.19

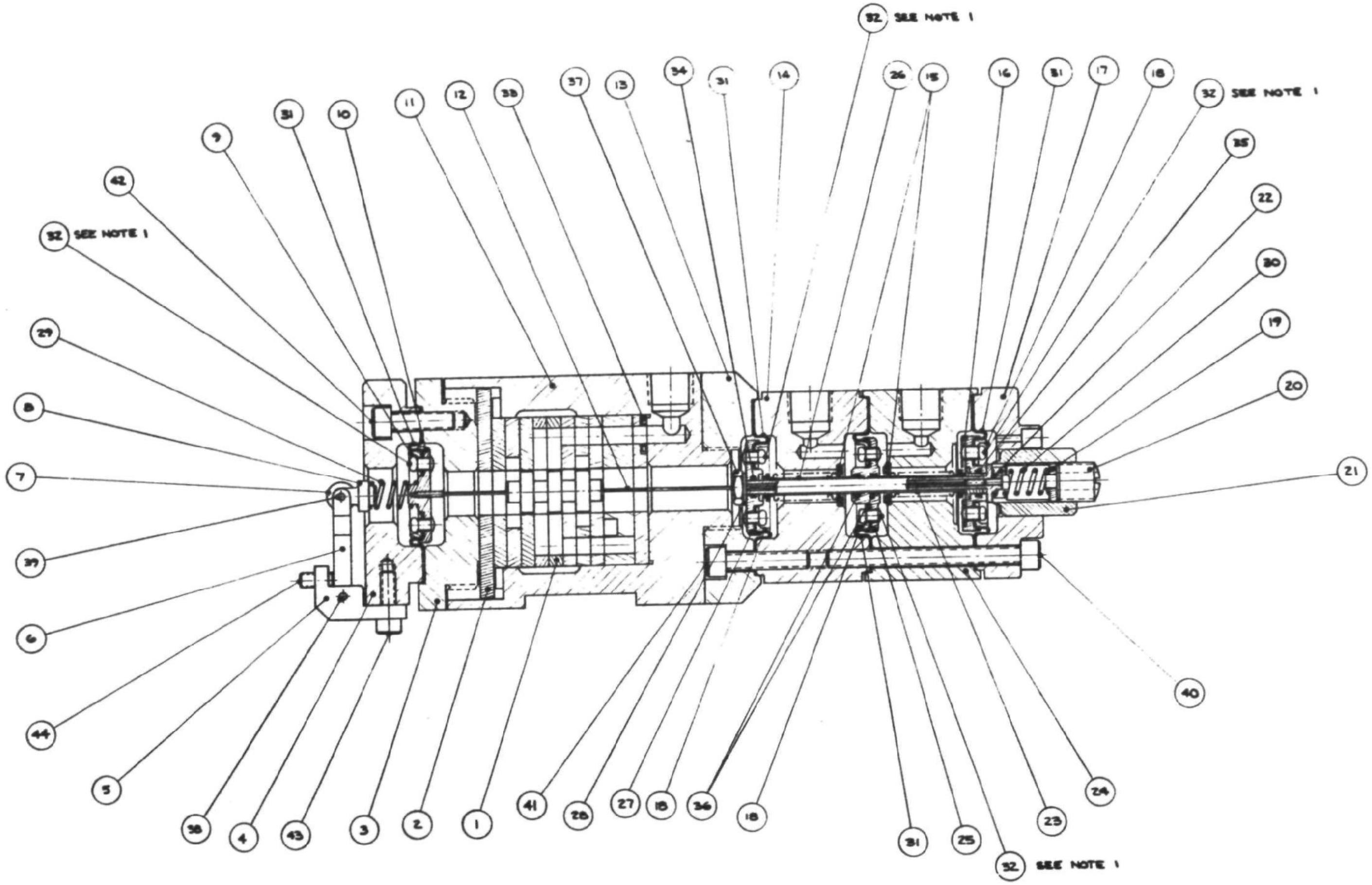
ATOMIZING AIR COMPRESSOR CHARACTERISTICS

PERFORMANCE REQUIREMENTS	
Mass Flow Rate of Air at 60°F	25.3 lbs/hr
Pressure Head	15 psig
Ideal Pumping Power	0.26 hp
Efficiency	65%
SPECIAL REQUIREMENTS	
Continuous constant speed operation	
Oil-free flow output	
DESIGN AND CONSTRUCTION SPECIFICATIONS	
Type	Rotary Vane
External Length	5 inches
External Diameter	3 inches
Speed	3450 rpm
Vane Material	Graphite
Drive	Shaft and Pulley

to the boiler is applied at ports B and C. A buffer fluid is used to transmit the pressure from metal diaphragms located at the orifice to the controller so that high pressure working fluid is not applied directly to the controller. Bellows seals are used in the high pressure part of the controller. The pressure signal proportional to organic outlet temperature is applied through a port (not shown) to the shaft side of the diaphragm on the far right of the layout of Figure 5.56.

The fuel control valve is illustrated in Figure 5.57. Constant pressure (25 psig) fuel is supplied at port A, with port B being the discharge flow to the fuel solenoid valve. The control valve stem is operated by a cam-operated roller, controlled by a cam connected to the blower van controlling the combustion air flow rate. This procedure insures that the fuel flow rate directly follows the combustion air flow rate, providing the desired fuel/air ratio over all transient conditions. If desired, the fuel/air ratio can be changed as a function of burning rate by proper shaping of the cam. This type of control maintained tight tolerance of the fuel/air ratio over all transients encountered in the transient burning tests at Thermo Electron Corporation. The full-swing time response of the controller with a step change in the orifice ΔP is approximately 200 millisec.

The boiler outlet organic temperature sensor is illustrated in Figure 5.58. The prime sensor consists of a stainless steel tube immersed in the organic flow with a central, low thermal expansion inner rod passing through the middle of the tube and fastened at the bottom of the tube. Differential thermal expansion provides movement of the central rod in response to organic temperature changes. Movement of the central rod operates a small valve to control the signal pressure applied to the servo control described earlier. The atomizing air compressor is used to supply constant pressure air for operation of this unit. The constant pressure air enters the supply port (4) and flows through an orifice so that the pressure in the chamber above the valve, or the signal pressure, depends on the air flow rate. The air flow rate is controlled by the valve, with the discharge flow vented to the engine compartment. As the organic temperature increases, the outlet stainless steel tube expands, moving the central rod downward and closing the control valve. This change results in a reduction in the air flow rate through the valve and increases the signal pressure to the servo control unit, thereby reducing the burning rate. A decrease in the organic temperature results in the inverse of this action. The control valve unit is thermally isolated from the thermal expansion unit carrying the organic flow to minimize drift of the controller.



Overall Dimensions:
6-3/4 x 2-1/8" diam. (max)

Materials:
Body - Aluminum Alloy
Metering Elements - Stainless Steel

Figure 5. 56 Layout Drawing of Combustion Air Servo-Control.

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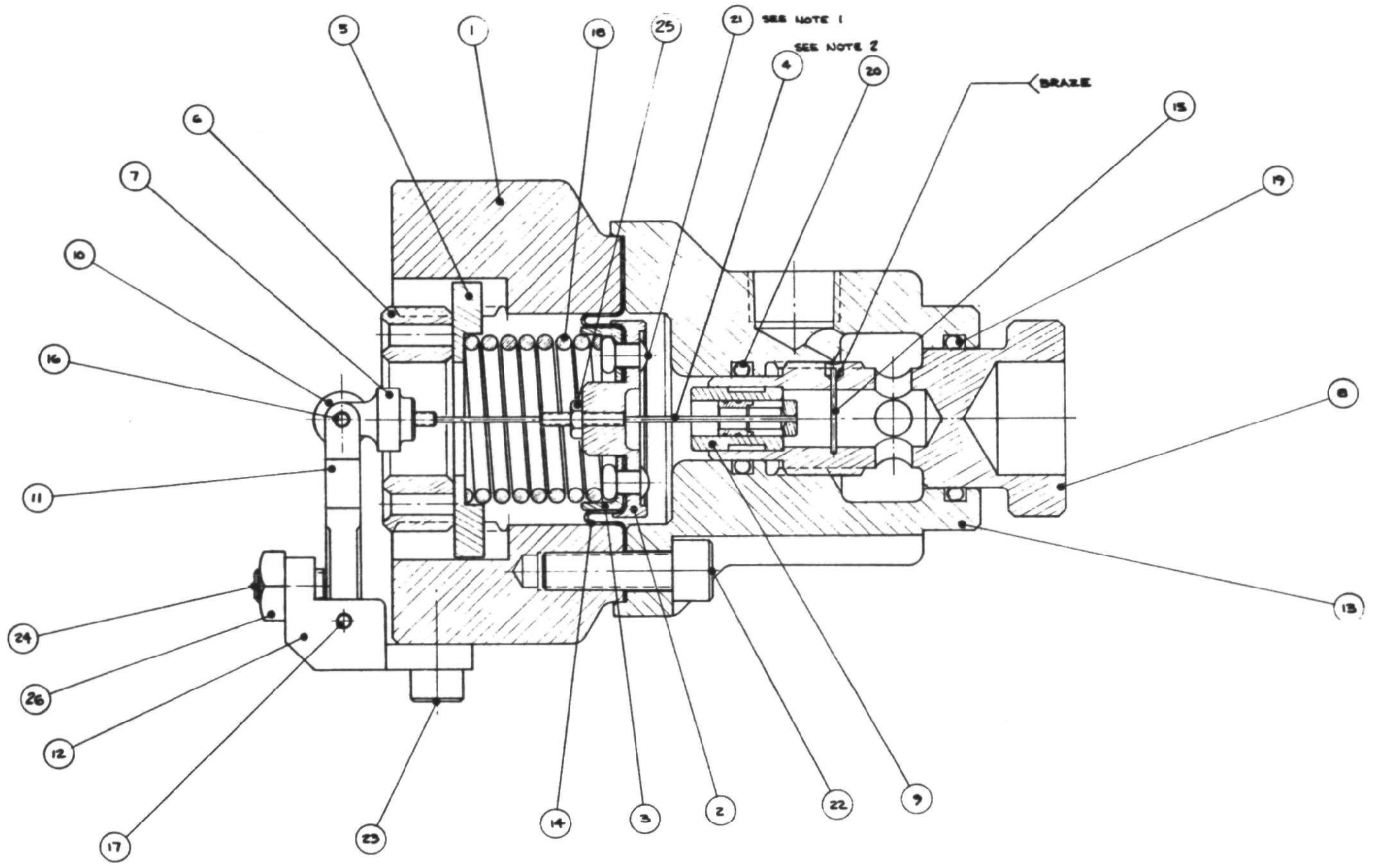


Overall Dimensions:
3-1/2 x 2-1/16" Diam. (max)

Materials:
Body - Aluminum Alloy
Metering Elements -
Stainless Steel

Figure 5.57 Layout Drawing of Fuel Control Valve.

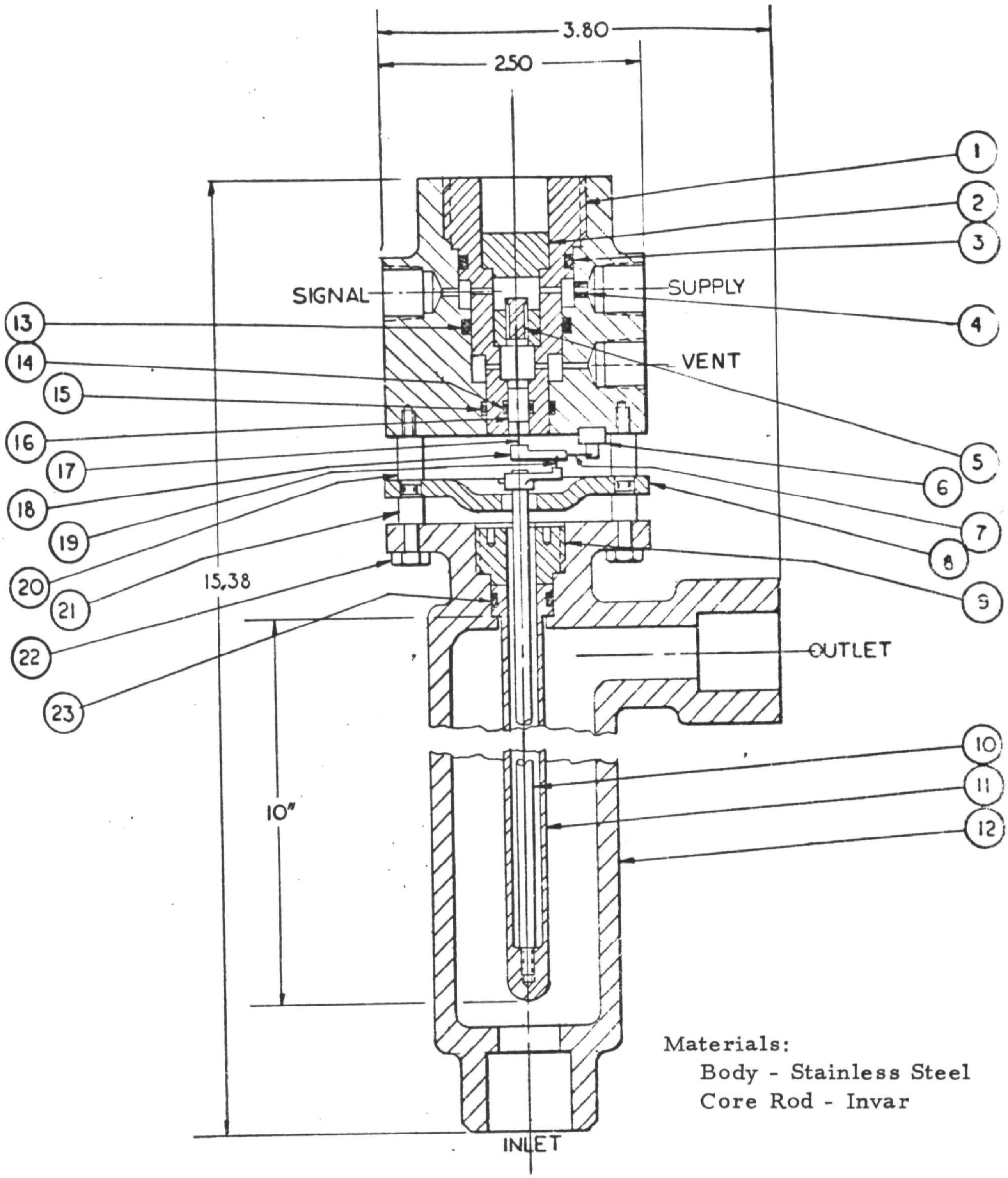


Figure 5.58 Boiler Outlet Organic Temperature Sensor.

5.3 REGENERATOR

The regenerator layout design is illustrated in Figures 5.59 and 5.60, and the regenerator design point requirements and characteristics are summarized in Table 5.20. The liquid and vapor flow paths are illustrated in Figure 5.61. The exchanger can be classed as a multipass, cross-counterflow exchanger with both vapor and liquid mixing between stages, but unmixed within stages. Four stages are used, with the liquid flowing in six parallel circuits in each stage.

The core is brazed aluminum construction with flat tubes carrying the liquid and with fins on both the liquid and vapor sides of the core. The fins on the liquid side serve the dual function of increasing the heat transfer area and acting as stays to permit the flat tube walls to withstand the design pressure of 850 psig on the liquid side with reasonable tube wall thickness. The design is adaptable to one-step brazing for inexpensive high volume production. The fins are based on standard fins produced by the Garrett-AiResearch Corporation, with the basic dimensions summarized in Table 5.20. Experimental measurements of the heat transfer coefficient and friction factor as a function of Reynolds' number are available for the specific fins selected.

The regenerator is also used as an oil separator to collect and return to the crankcase a major portion of the oil droplets in the exhaust vapor from the expander. With reference to Figure 5.60, a mesh screen (item 10) is placed across the vapor inlet to remove an appreciable fraction of the oil droplets before the vapor enters the first stage of the exchanger core. This minimizes fouling of the

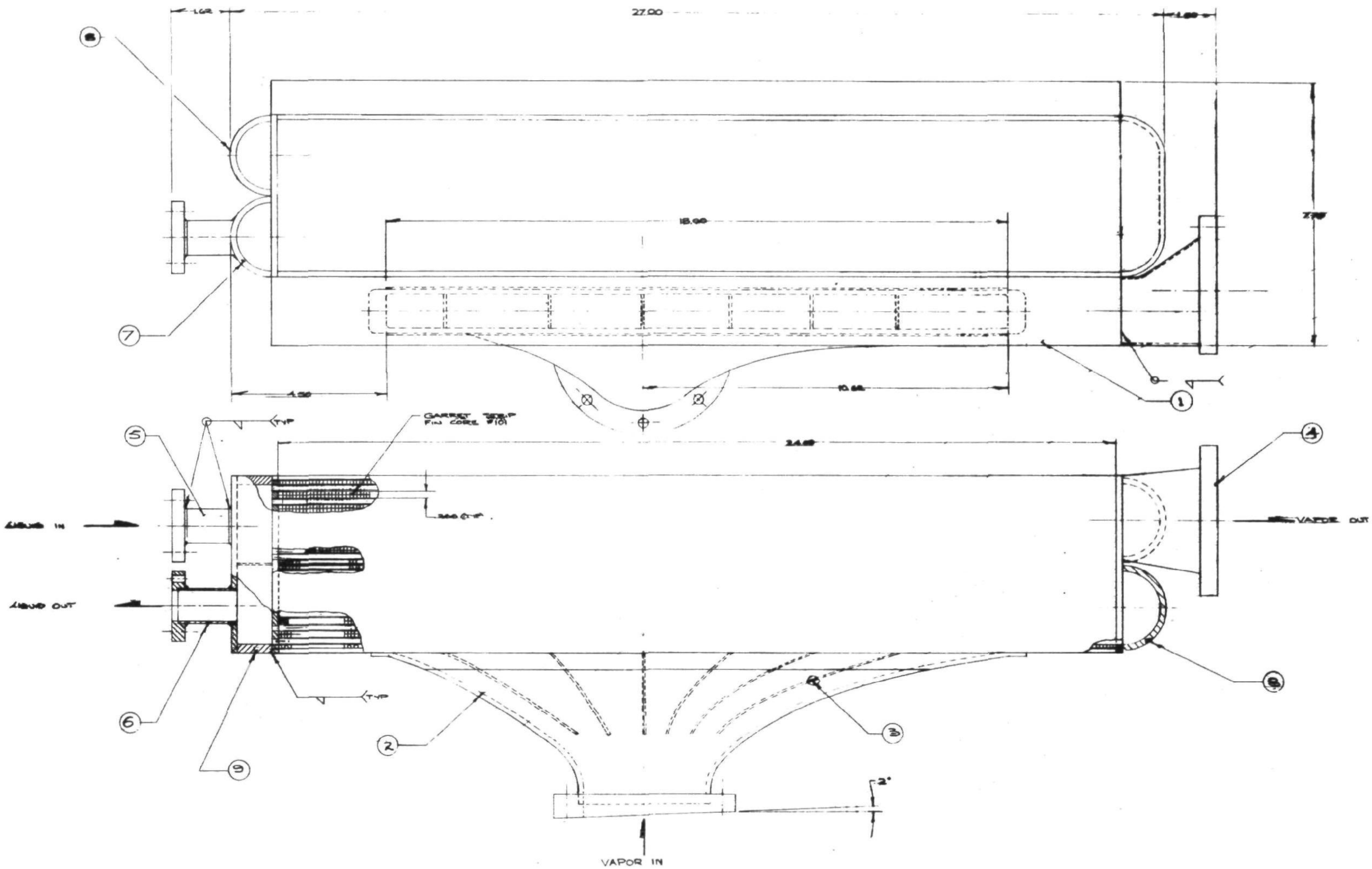


Figure 5.59 Longitudinal Cross Section of Regenerator.

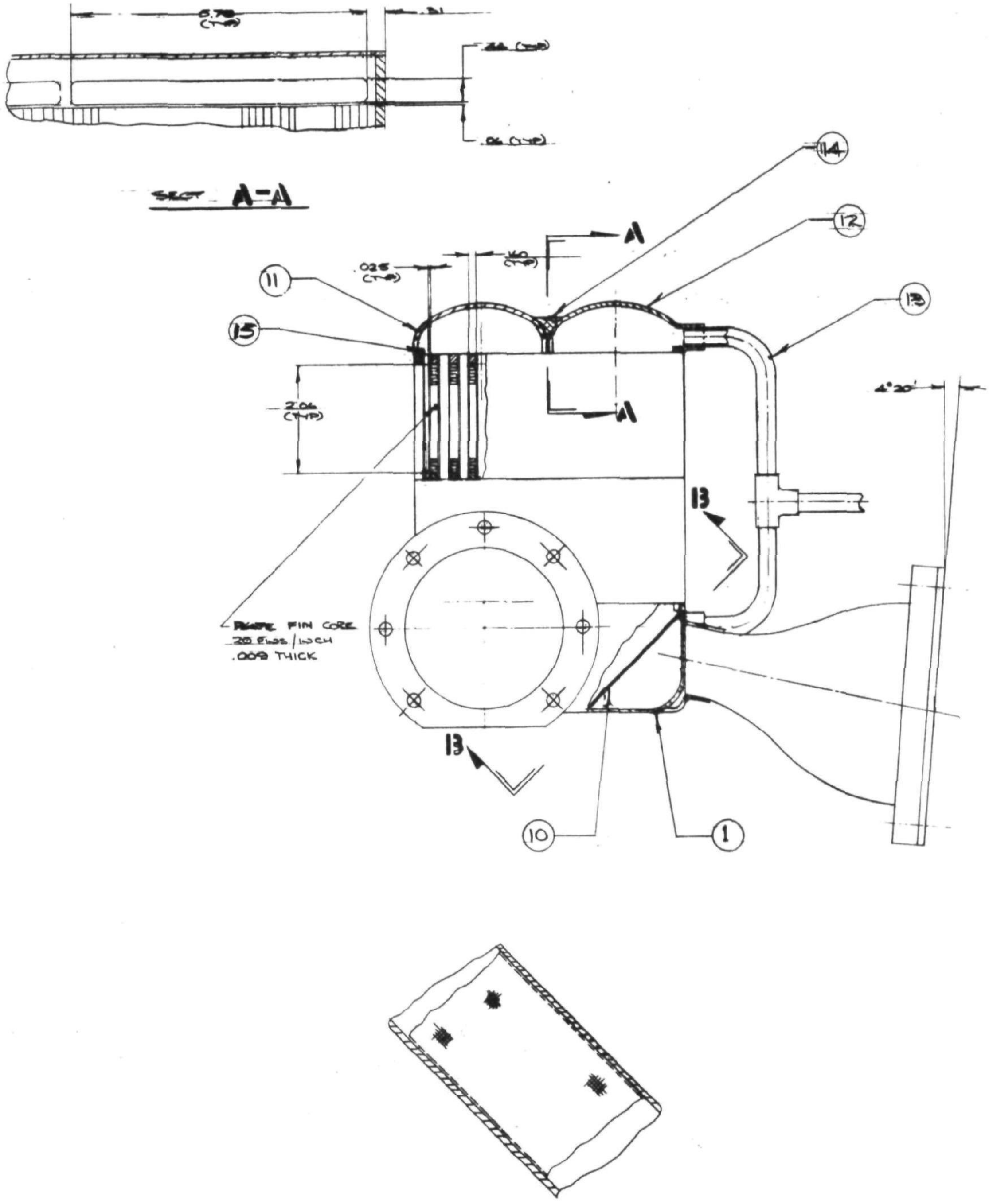


Figure 5.60 Transverse Cross Section of Regenerator.

TABLE 5.20

REGENERATOR DESIGN POINT REQUIREMENTS

Rate of Heat Transfer	414,000 Btu/hr
Effectiveness at 100% Load	81.5%
Vapor Temperatures	
Inlet	375°F
Outlet	239°F
Vapor Pressure	
Inlet	42.5 psia
Pressure Drop	1.328 psia
Liquid Temperatures	
Inlet	208°F
Outlet	287°F
Liquid Pressure	
Inlet	831 psia
Pressure Drop	4.38 psi
Fluorinol-85 Flow Rate	9924 lbs/hr
Number of Stages	4
Number of Parallel Liquid Circuits	6
Overall Core Dimensions	24 x 10.25 x 2.706 inches
Overall Core Weight	16.38 lbs

Liquid Side

Plate Finned Tube-Inside Dimensions 2" x 24" x 0.025" wall x 0.160" Inside Gap

Fins - Plain Plate Fins
20 FPI
Thickness 0.009"
Height 0.160"

Vapor Side

Strip Fin Surface - Strip Fin
25 FPI
Thickness 0.004"
Fin Offset Length (Flow Direction) 1/9"
Height 0.200 inches

Two screens of 5 mil wire 50% open area for oil separation.

All fabrication out of aluminum alloy 6061.

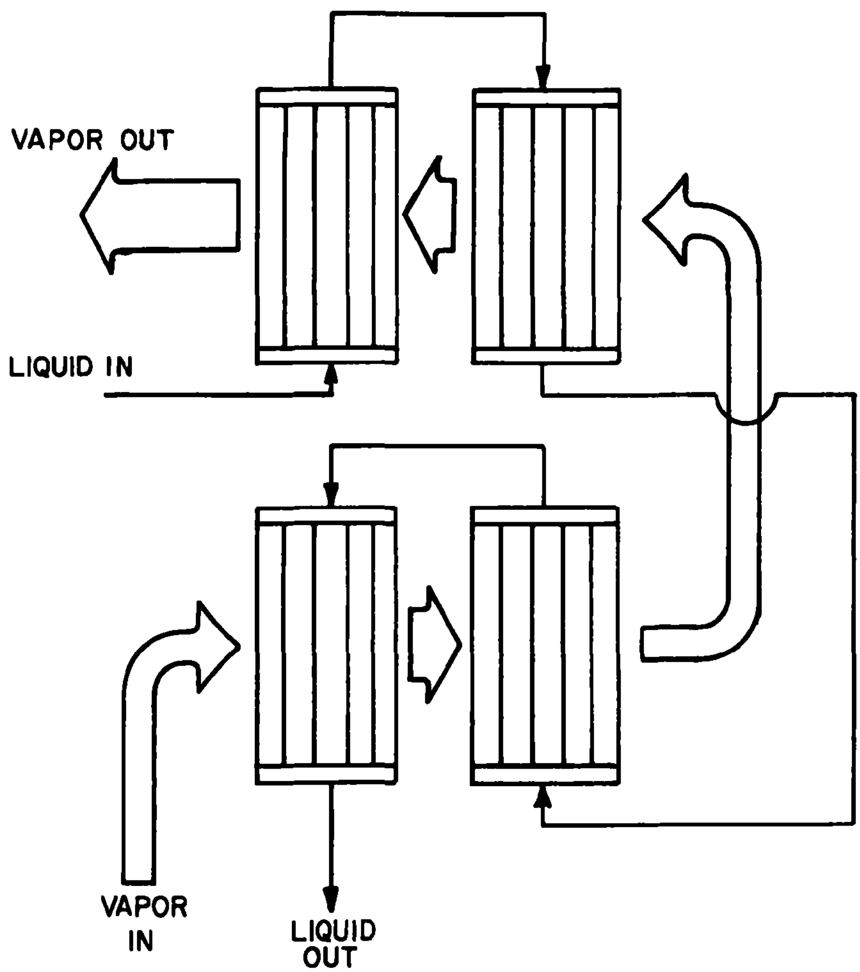


Figure 5.61 Liquid and Vapor Flow Paths in Regenerator.

vapor fins of the first two stages by an oil film. The interception efficiency of the screen is given in Figure 5.62 as a function of oil droplet diameter. The vapor fins of the first two stages are used as an additional separator to remove a fraction of the oil droplets not collected by the screen separator. The vapor plenums between the first two stages are connected by a drain line; oil is collected from both plenums and the oil drains by gravity through a common line to the crankcase. A fraction (~5%) of the vapor is bypassed around the first two stages in the regenerator, through the oil drain line connecting the two plenums, due to the vapor pressure drop across the core of the first two stages. The vapor velocity is low enough so that the oil will drain by gravity from the second plenum against the vapor upflow in this line. Any oil droplets passing the first two stages will pass through the remainder of the regenerator and on through the condenser, pumps, and boiler back to the expander. The flow path has been designed so that no oil traps exist in these components.

The regenerator is directly mounted to the exhaust part of the expander by means of a flanged connector, with a metal "O" ring seal on the vapor inlet duct to the regenerator.

5.4 CONDENSER SUBASSEMBLY

The design of the condenser-condenser fan-condenser fan drive and controls subassembly has a very important influence on the system performance and fuel economy. The condenser fan power represents the largest parasitic power loss in the system. Furthermore, the expander power output and system efficiency are dependent on the condenser pressure; each psia decrease in the condenser pressure represents about 1.5 hp increase in the expander power output. The overall goal, approach followed, and the design point selection for this subassembly, are summarized in Table 5.21. The condenser and fans are designed to meet the design point conditions representing the peak power condition at high vehicle speed. The condenser fan drive and control is then designed to optimize the fan speed for optimum horsepower and efficiency under part-load and low vehicle speed operating conditions. An additional consideration has been use of an inducer to maintain the condenser free of condensed liquid, so that the entire condenser core is effective for condensation. The pump subassembly described in Section 5.6 requires no liquid subcooling for proper operation.

5.4.1 Condenser Design

The condenser design parameters are summarized in Table 5.22 and the condenser design is presented in Figure 5.63. An important consideration has been maximizing the condenser frontal area within the packaging constraints of the 1972 Ford Galaxie engine compartment without any major modifications to the vehicle frame and engine compartment. This large frontal area is particularly important since, for a given core design and condensing heat load

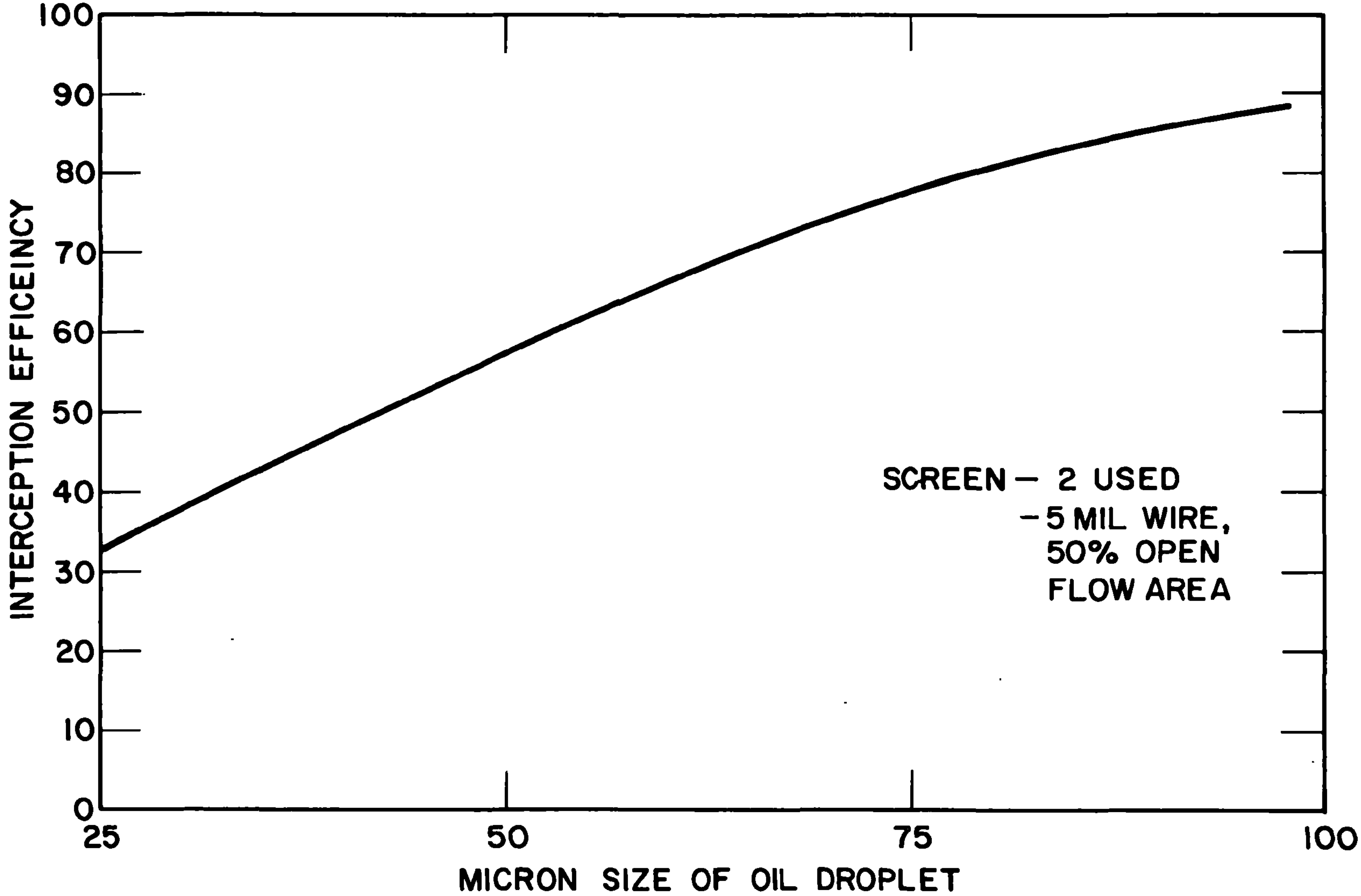


Figure 5.62 Interception Efficiency by Regenerator Screen Separator for Oil Droplets.

TABLE 5.21

DESIGN CRITERIA FOR CONDENSING SUBASSEMBLY

OVERALL GOAL

- Reduce Parasitic Fan Power and Maximize System Performance within Packaging Restraints.

APPROACH

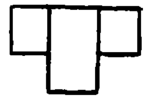
- Maximize Condenser Frontal Area.
- Utilize Condenser Fan Speed Control to Optimize System Part-Load Performance.
- Minimize Condensing Side Heat Transfer Resistance Relative to Air Side Heat Transfer Resistance.
- Use air side fin with optimum characteristics for minimum fan power.

DESIGN POINT FOR SIZING

- Peak System Power Output.
- Ram Air Equivalent to Vehicle Speed of 90 mph.
- Ambient Air Temperature = 85°F.
- Condenser Average Pressure = 40.5 psia.
- Equivalent Condensing Temperature = 217°F.

TABLE 5.22

CONDENSER DESIGN PARAMETERS

HEAT TRANSFER RATE		1.88×10^6 Btu/hr
CONFIGURATION (to maximize A_{fr} within packaging constraints)		
CORE DIMENSIONS -	Central	34.0"x21.0"x4.3"
	Side	18.0"x13.0"x4.3"
	Frontal Area - Central	4.96 ft^2
	Side	1.625 ft^2
	Total	8.21 ft^2
	σ	.757
AIR SIDE	- Fin - Garrett	22R-.326-PERF 9-13) -.004(AL)
	Heat Transfer Area	1450 ft^2
	Air Flow	75,300 lb/hr
		17,300 CFM at 85°F
		20,540 CFM at 188°F
	h	29.8 Btu/in-ft^2-°F
	ΔP	3.47 in W. C.
	Ideal Air Power	9.38 hp at 85°F
		11.17 hp at 189°F
ORGANIC SIDE	Fin - Garrett	20 f/in, 0.004(AL), 1 Strip
	Heat Transfer Area	270 ft^2
	Organic Flow	9860 lb/hr
		5.5 ft^3/sec entering
	h	690 Btu/in-ft^2-°F
	ΔP	5 psi
	P entering	40 psia
OVERALL HEAT TRANSFER PERFORMANCE		
RESISTANCES, % of TOTAL -	Air	73%
	Organic	26.8%
	Wall	0.2%
	$\overline{U}_A$	20.3 Btu/hr-ft^2-°F
	Effectiveness	0.80
	Entering Air Temperature	85°F
	Exit Air Temperature	188.5°F

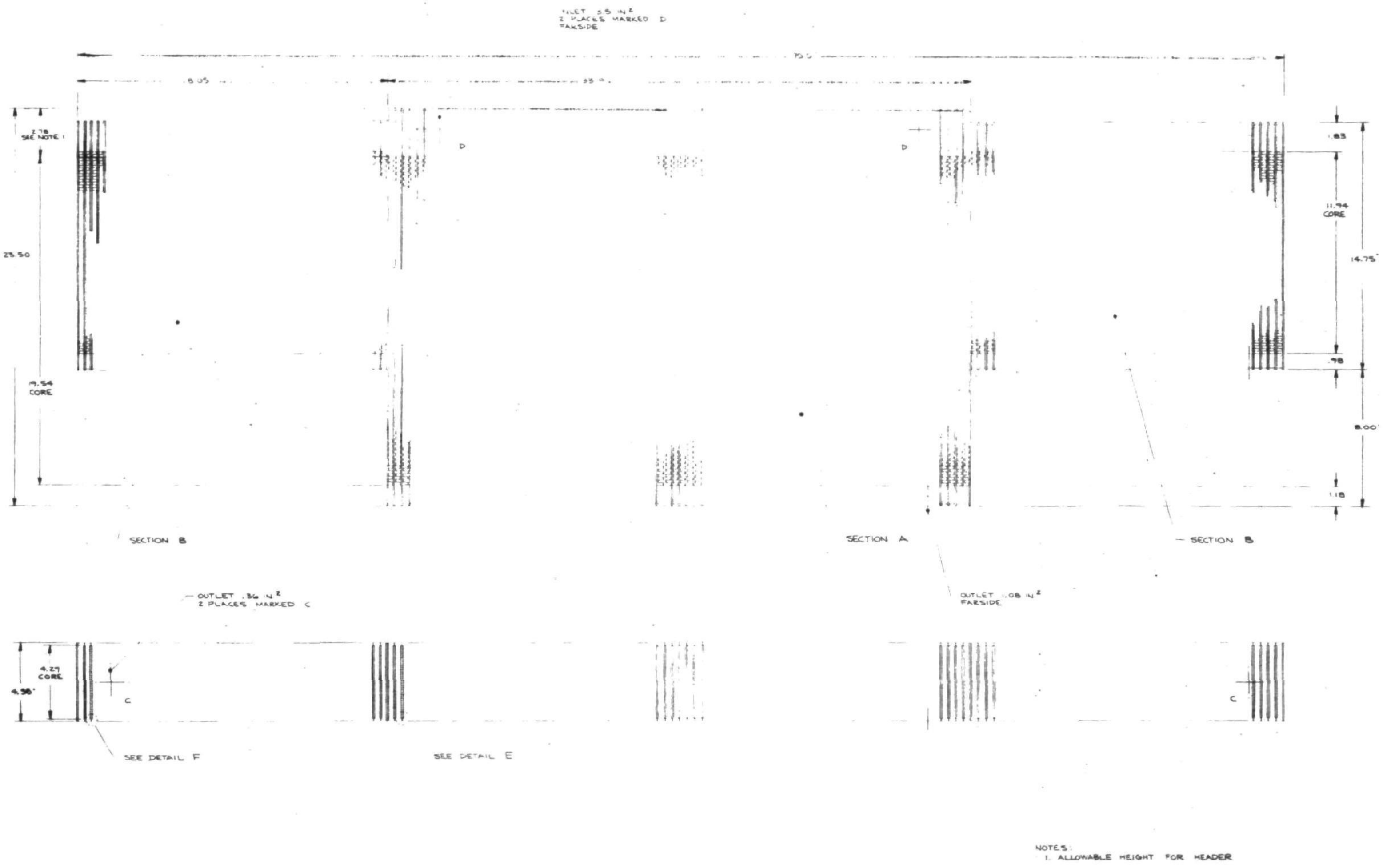


Figure 5. 63 Condenser Design Layout.

and temperature, the fan power required is approximately inversely proportional to the square of the frontal area. A T-shaped configuration has thus been used for the condenser design because of vehicle packaging constraints. With reference to Figure 4.8 of Chapter 4, the top part of the condenser extends across the width of the engine compartment above the frame; the lower part of the condenser sits between the frame members and is brought as low as possible within the limit set by the vehicle driveline illustrated in Figure 4.4 of Chapter 4.

The condenser design is based on the air-side fin and the condenser construction recommended by Garrett-AiResearch; Garrett-AiResearch has concluded an analytical and experimental study to optimize the air-side fins for Rankine-cycle condensers and will provide the condenser, shroud, and condenser fans for the pre-prototype system testing at Thermo Electron. They will also be performing condensing tests with Fluorinol-85. The condenser is brazed aluminum construction and is designed for one-step furnace brazing. The design is thus suitable for high volume production.

The vapor enters an integral top header over the entire top length of the condenser. The header distributes the vapor flow to vertical flat tubes operating in parallel with downflow condensation. The flat tubes have internal fins which serve two purposes: to increase the heat transfer area on the condensing side thereby minimizing the condensing side heat transfer resistance, and to act as stays on the flat tubes to minimize the tube wall thickness for the condenser design pressure of 150 psia. The air side fins are slotted fins and have been optimized by Garrett-AiResearch to minimize the condenser thickness and condenser fan power. Both the internal and air-side fins are adaptable for high volume mass production.

The condensate is collected in bottom headers on the condenser; the condensate headers are directly connected to the inducer. Because of the T construction of the condenser, the tubes have different lengths in the central portion and outer portions of the condenser. The tube internal gap is different in these two portions of the condenser, as indicated in Figure 5.63, for proper balancing of the condensing rate with the same pressure drop across both portions. The condensing side pressure drop at the design point condition is 5 psi.

The ideal fan horsepower with the fans on the rear of the condenser is 11.17 hp. The system is designed for maximum utilization of ram air so that at the design point with a vehicle speed of 90 mph, 7.0 shp is the required fan power input. A model for the grill and engine compartment losses, along with the fan characteristics, has been incorporated in the system performance prediction program and has been used in calculating the condensing system performance at the design point as well as at all other operating conditions.

5.4.2 Condenser Fan Design

The condenser fan requirements and characteristics of the selected fans are summarized in Table 5.23 and the condenser fan arrangement and drive is illustrated in Figure 5.64. The fan design was based on recommendations by a consultant, Flowtron, Inc. of Newton, Massachusetts.

TABLE 5.23

CONDENSER FAN REQUIREMENTS AND CHARACTERISTICS

REQUIREMENTS

- Air Flow Analysis Indicated 1.5" w.c. Required from Fan at 90 mph Design Point - 2.0" w.c. Supplied by Ram Air - Velocity Head Available at 90 mph = 3.73" w.c.
- Air Flow Rate (Hot Side) = 20,540 CFM
- Size and Placement Consistent with Good Air Flow Through Condenser
- Efficiency Maximized Within Packaging Constraints
- Low Noise Level

FAN CHARACTERISTICS

- Tube Axial, Air Foil Blades, Cast Aluminum
- Efficiency at Design Flow, 55%
- Characteristics at Design Point

Number Required	Outer Diameter inches	RPM	CFM	Hub		Number of Blades
				Diameter Inches	Thickness inches	
1	24	2700	10700	5.5	2.4	10
2	16	4050	4920 each-	8.0	2.4	16

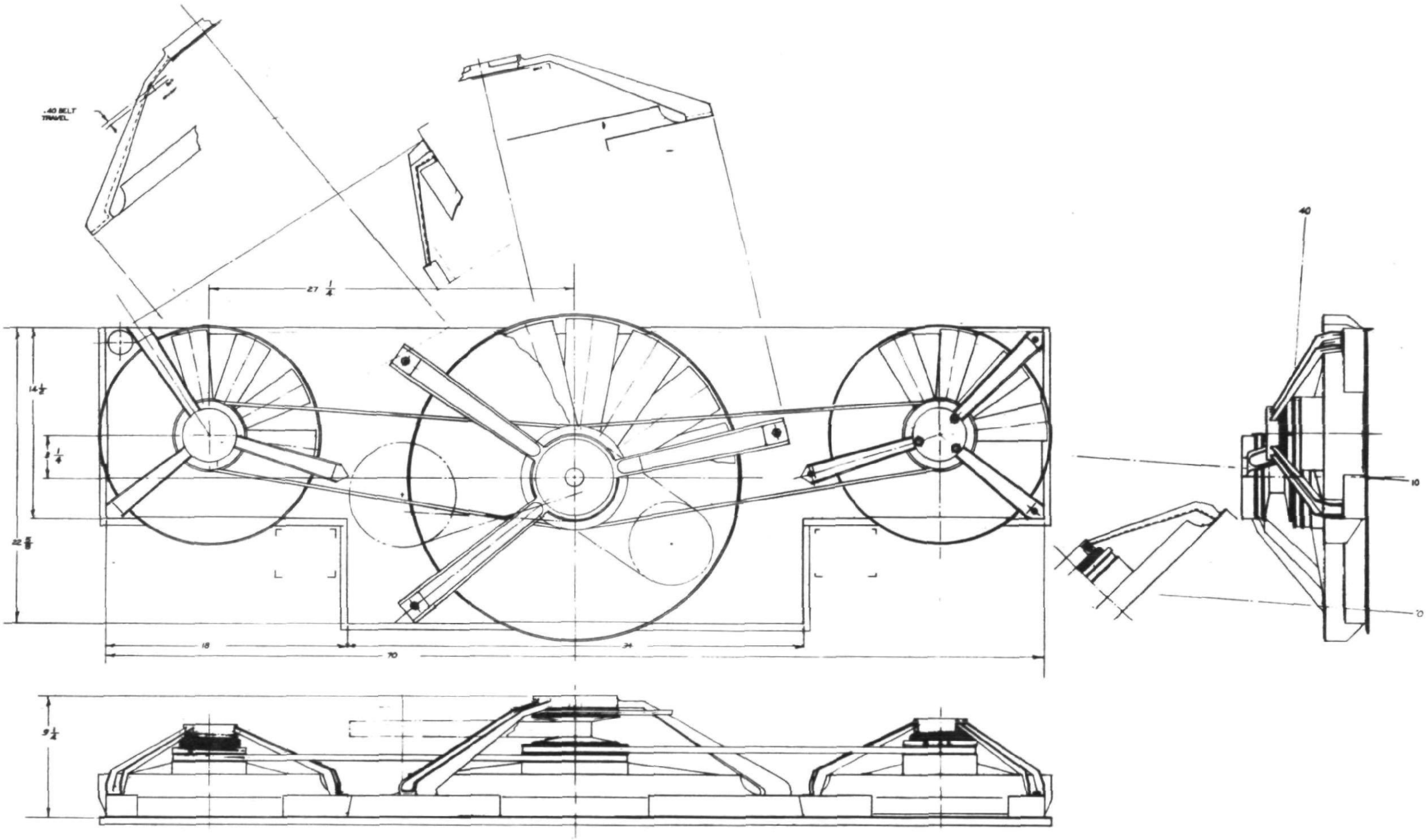


Figure 5.64 Condenser Fan Mounting on Condenser.

At the design point, the condenser pressure loss on the air side is 3.5 inches W. C. The air flow analysis indicated that 2.0 inches W. C. was supplied by ram air at the discharge air flow rate of 20,540 CFM. At 90 mph, the velocity head of the air at 85°F ambient temperature is 3.73 inches W. C., with grill and engine compartment losses of 1.73 inches W. C.

The axial length available for the condenser fans is limited by the available space between the condenser rear face and the burner-boiler subassembly, and by the requirement for sufficient air-flow area for exhaust of the condenser cooling air flow. The selected fans were of the tube-axial with air-foil blades for high efficiency. Three fans are used to provide uniform air flow over the entire condenser area. As indicated in Figure 5.64 and Table 5.23, a 24 inch diameter fan is used in the central portion of the condenser, with two 16-inch diameter fans used on the outer portions of the condenser. The hub thickness of the fans is 2.4 inches. A shroud is used on the back of the condenser and the fans are mounted to the shroud. The largest fans possible were used in the design to provide good coverage of the condenser as well as to minimize the fan speed and fan noise.

5.4.3 Condenser Fan Drive and Speed Control

A condenser fan speed control is required for optimum system performance. At low vehicle speeds, ram air is not effective and the condenser cooling air must be supplied completely by the condenser fans. For wide-open-throttle operation at low vehicle speeds, a large ratio of fan speed to expander speed is required. If this speed ratio is maintained up to high expander speeds, the power consumption by the fans is excessive; it is therefore desirable to reduce the fan/expander speed ratio at high expander speeds to optimize the system performance.

For part-load operation, the condensing load is greatly reduced. Maintaining the condenser fan speed at that required for the wide-open-throttle condition results in a larger than optimum air flow rate; the condenser fan power is again excessive and degrades the system efficiency and fuel economy. It is thus desirable to have an additional control responsive in some way to the condensing rate under part-load conditions. An additional factor is the influence of ambient temperature. At low ambient temperatures, a lower fan speed is required at a given operating condition than at high ambient temperatures for optimum system efficiency.

A preliminary study of the optimum fan speed for various operating conditions was carried out and the results are indicated in Figure 5.65. For the wide-open-throttle condition (maximum intake ratio) the optimum fan speed is approximately constant above an expander speed of 700 rpm, and decreases below 700 rpm down to the idle speed of 300 rpm. For part-load conditions, the optimum fan speed decreases as the power level drops at any expander speed. The lower limit for the speed control is set at the optimum corresponding to an intake ratio of approximately 0.025. The lower limit is not crucial since the shaft power to the fan is much less than the maximum fan power at the low fan speeds.

Many alternative approaches were evaluated for the condenser fan speed control. In Figure 5.66, an illustration is given of the selected control which approximates the desired condenser fan speed

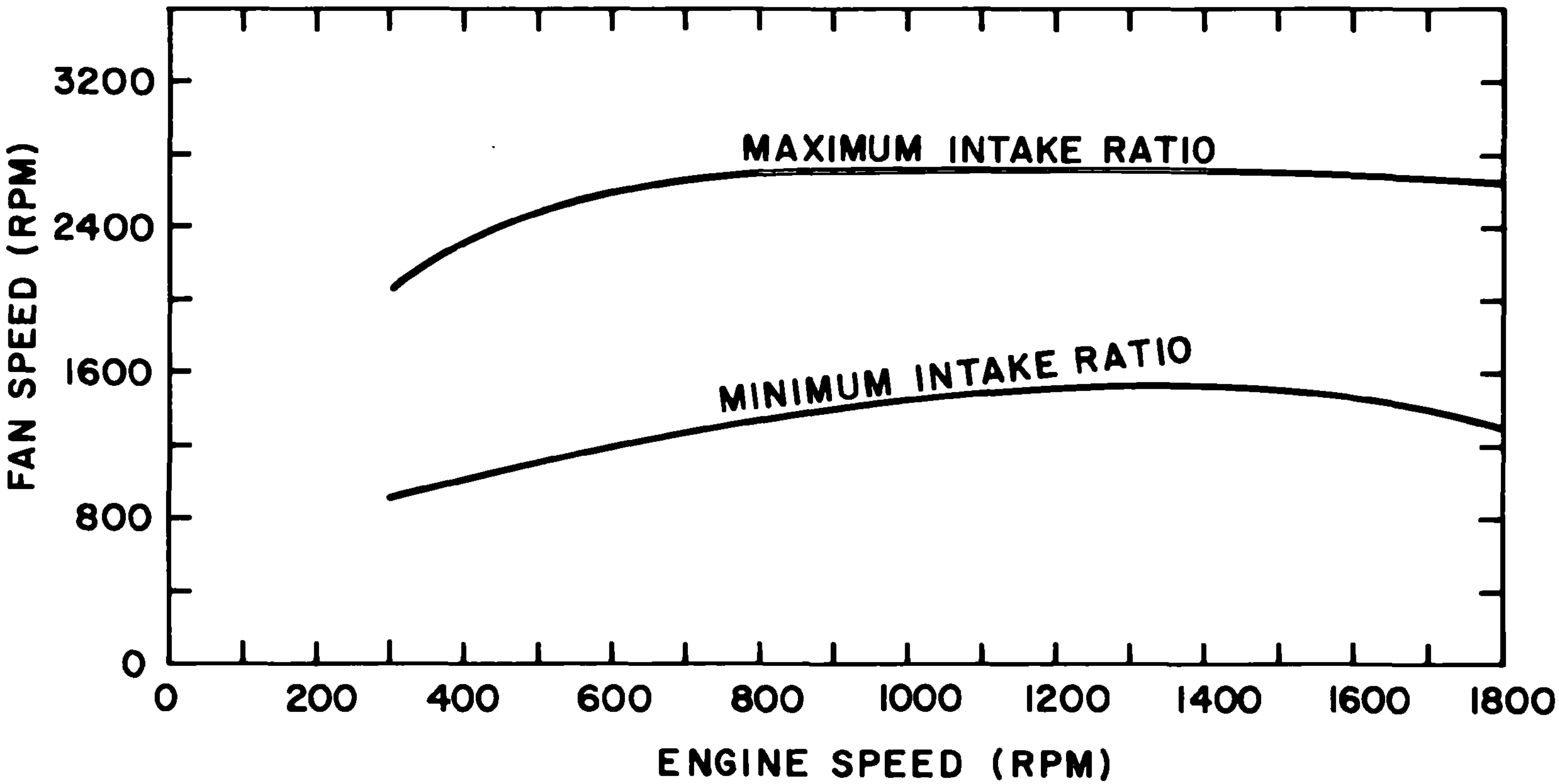


Figure 5.65 Optimum Fan Operating Range.

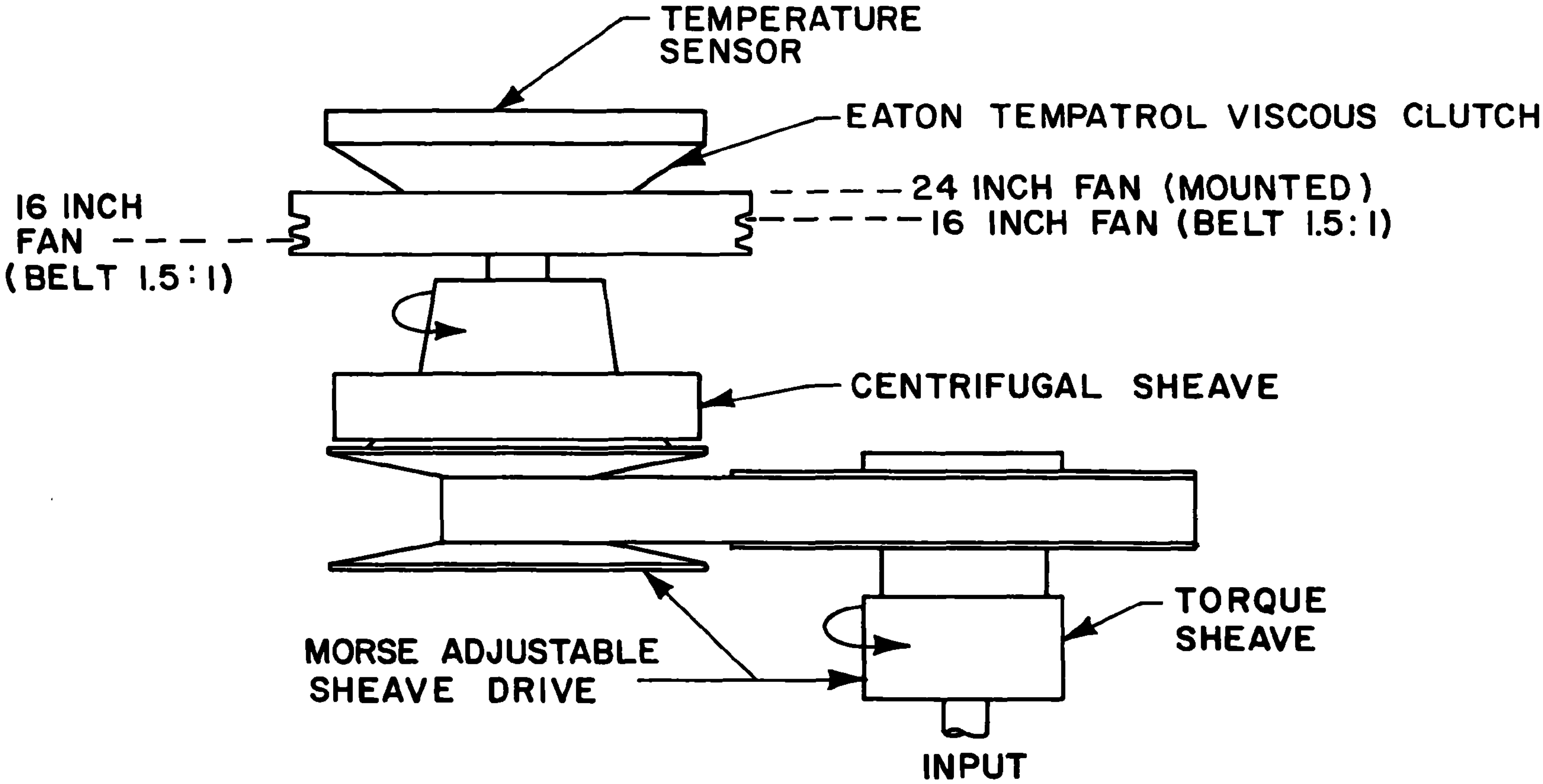


Figure 5.66 Condenser Fan Variable Speed Drive.

variation. Figure 5.67 shows the actual range of operating conditions with the control. The control is made up of off-the-shelf components. To provide the wide-open-throttle speed ratio control, a standard Morse variable-speed belt drive is used with a centrifugally-controlled sheave provided to vary the speed ratio. With an input speed proportional to the expander rpm, this control provides a constant fan shaft/expander speed ratio of 4.91:1 up to 550 rpm expander speed. Above 550 rpm expander speed (or 1490 rpm accessory drive shaft speed), the centrifugal sheave maintains a constant fan shaft speed of 2700 rpm irrespective of expander speed. The maximum fan shaft speed is thus controlled near the optimum except at expander speeds between 300 and 500 rpm, where the fan shaft speed is somewhat below the optimum.

To provide control for part-load conditions as well as ambient temperature variation, a standard Eaton Tempatrol viscous clutch is used; these units are currently used for controlling the fan speed in some I/C engine-powered automobiles. This clutch modulates the fan speed by sensing the air temperature leaving the condenser and, if the air temperature is low, allows the clutch to slip, thereby reducing the fan speed. The amount of slip depends on the air temperature from the condenser. The upper and lower curves of Figure 5.67 illustrate the operating range of the Eaton clutch, which is satisfactory for this application.

The viscous clutch is constructed integral with the central 24-inch fan. The 24-inch fan blades are mounted directly on the clutch rim, and the two 16-inch fans are belt-driven from the clutch rim at a 1.5:1 speed ratio. The belt drive configuration is illustrated in Figure 5.64.

5.5 STARTUP SEQUENCING, SAFETY CONTROLS, AND ACCELERATOR PEDAL LINKAGE

The system is designed for completely automatic startup and operation. The required driver functions are identical to those of current I/C engines with automatic transmission, that is, ignition switch, gear shift lever, accelerator pedal, and brake pedal.

The system startup and safety control logic diagram is illustrated in Figure 5.68 with the nomenclature defined in Table 5.24. The operation of the logic is outlined below:

Startup

When the key is turned on initially, the following events take place:

1. Gate (1) Energizes the atomizing air compressor, fuel pump, and combustion air blower.
2. Gate (2) Energizes the accumulator solenoid valve in the intake valve hydraulic circuit. This applies hydraulic pressure to the intake valves and keeps the expander intake valves closed during startup, permitting a faster buildup of boiler pressure and faster startup.
3. Gate (3) is energized if Gate (4) is not on.
4. Gate (5) is energized when the combustion air and atomizing air pressures are correct.
5. Gate (6) is energized if the fuel pressure is correct and the remaining inputs are true.

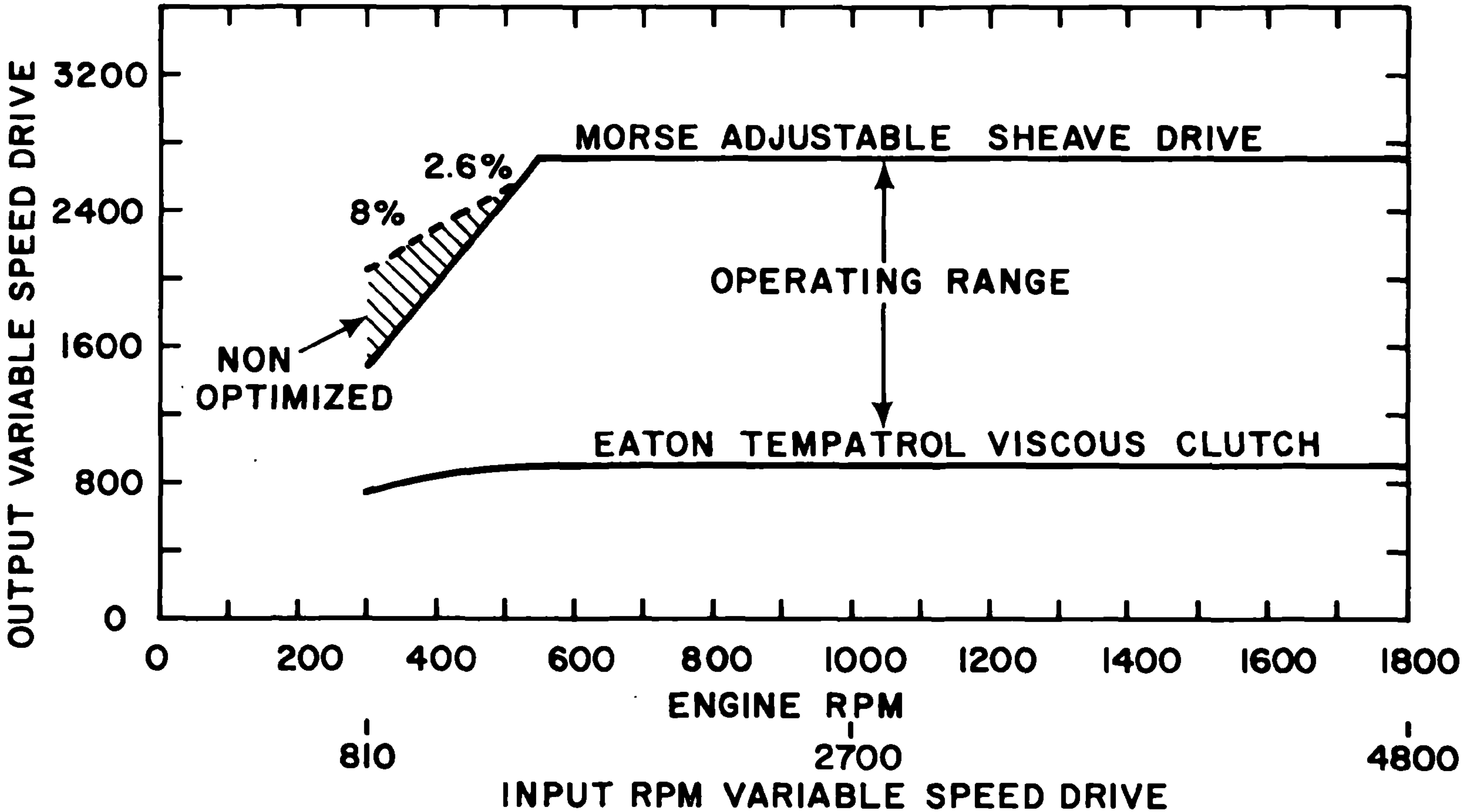


Figure 5.67 Fan Drive Characteristics with Fan Speed Control.

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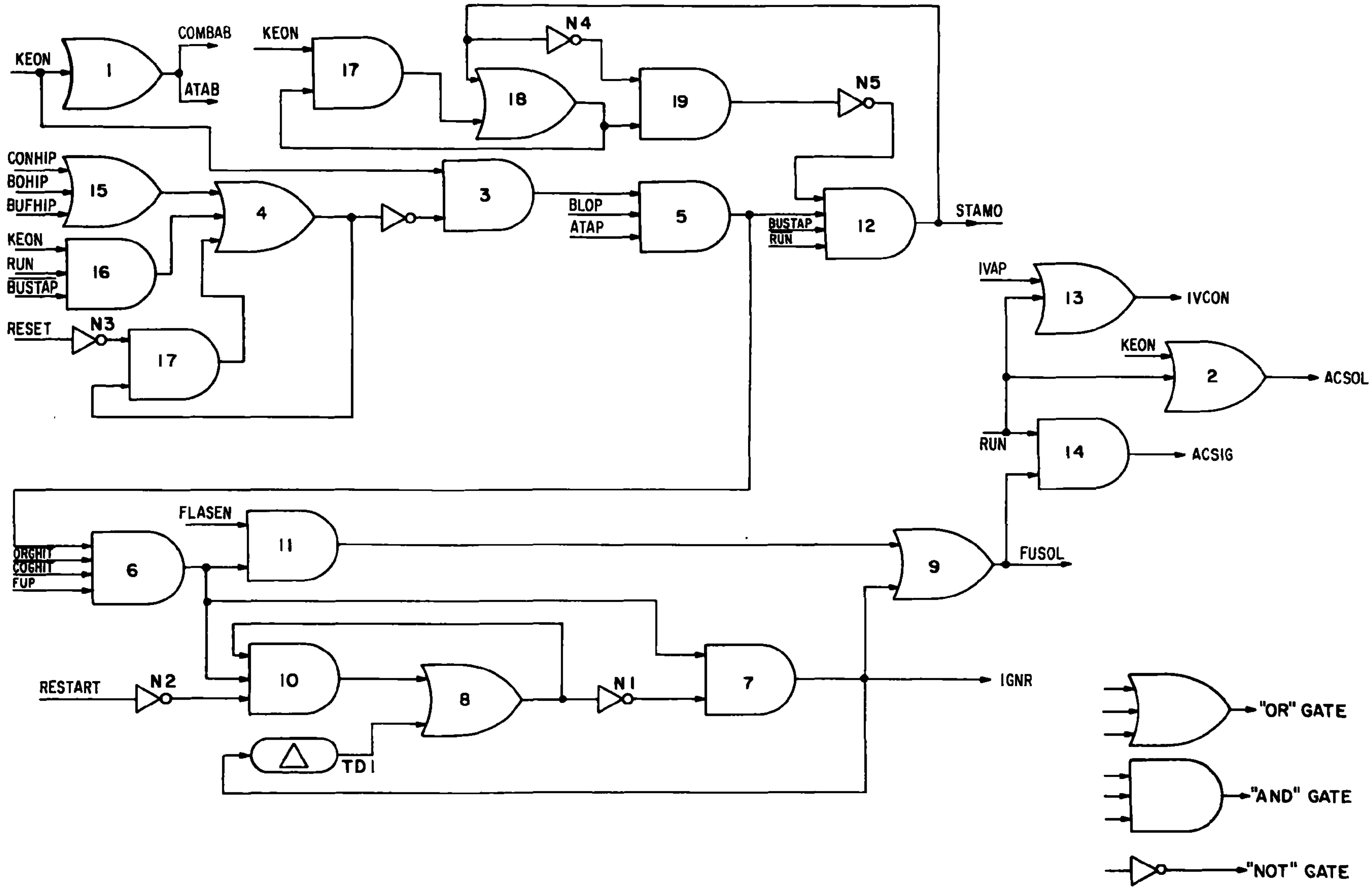

Figure 5.68 System Startup and Safety Control Logic Diagram.

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TABLE 5.24

NOMENCLATURE USED IN SYSTEM LOGIC DIAGRAM

KEON	- KEY ON
COMBAB	- COMBUSTION AIR BLOWER ON
CONHIP	- CONDENSER HIGH PRESSURE
BOHIP	- BOILER HIGH PRESSURE
BUFHIP	- BUFFER FLUID HIGH PRESSURE
RUN	- STARTING RPM
BUSTAP	- BUFFER FLUID STARTING PRESSURE
RESET	- RESET
ORHIT	- ORGANIC FLUID HIGH TEMPERATURE
COGHIT	- COMBUSTION GAS HIGH TEMPERATURE
FUP	- FUEL PRESSURE
FLASEN	- FLAME SENSOR
RESTART	- RESTART
IGNR	- IGNITER
FUSOL	- FUEL SOLENOID
ACSIG	- ACCELERATOR SIGNAL
ACSOL	- ACCUMULATOR SOLENOID
IVCON	- INTAKE VALVE CONTROLLER (ENABLE)
IVAP	- INTAKE VALVE ACTIVATING PRESSURE
STAMO	- STARTER MOTOR
BLOP	- BLOWER PRESSURE
ATAP	- ATOMIZING AIR PRESSURE

6. Gate (7) is energized and the igniter turned on if Gate (8) is de-energized.

7. Gate (9) energizes the fuel solenoid.

8. Gate (11) is energized by the flame sensor. This holds the fuel solenoid on through Gate (9) when the igniter is turned off if ignition has been obtained. A manual restart is provided through NOT Gate N2.

9. Time Delay TD1 energizes Gate (8), which locks up through Gate (10), since the other inputs to Gate (10) are normally true. If a signal from the flame sensor is not received during the time delay period, TD1 opens and the fuel solenoid is closed. Restart can be attempted using the manual restart button.

10. Gate (8) energizes the NOT Gate N1, which de-energizes the igniter through Gate (7).

11. Gate (12) energizes the starter motor until the idle rpm or Run Speed is reached. This input is taken from the intake valving system controller. The starter motor is energized when the buffer fluid pressure, BUSTAP, reaches a preset level.

12. Gate(13) energizes the intake valve controller, which enables the intake valve hydraulic circuit.

13. Gate (14) is energized when the idle rpm is reached. This enables the accelerator signal to the intake valve controller.

Safety

1. Gate (15) de-energizes the fuel solenoid if the maximum pressures are exceeded in the boiler discharge, buffer fluid, or condenser. Lock-up, requiring a manual reset, is provided with Gates (17), (4), and NOT Gate N3.

2. Gate (16) de-energizes the fuel solenoid when the buffer fluid pressure drops below the starting pressure during a running operation. The same reset circuit is used, as above.

3. Gate (6) de-energizes the fuel solenoid when the fuel pressure is low, or when the temperatures of the combustion gas or organic fluid are high. Restart is automatic.

4. Gates (17), (18), (19), N4, and N5 lock out the starter motor circuit after it has been de-energized. The key must be turned off to reset the circuit.

In Figure 5.69, the circuit schematic which provides the startup sequencing and safety shutoff of the system is presented. The control box with this circuit is illustrated in Figure 5.70; the entire control box occupies a space 3-5/8" x 2-1/4" x 3".

An electrical interface through a LVDT is used between the accelerator pedal and the intake valving control system. The schematic of the valving control system is illustrated in Figure 5.71 and the accelerator linkage with the LVDT is illustrated in Figure 5.72. The intake valving control system includes an automatic advance, intake ratio upper limit as a function of expander rpm to prevent exceeding the boiler capacity and a hydraulic pressure control to reduce the hydraulic pressure at low engine speeds when extremely rapid intake valve events are not required, thereby minimizing the hydraulic pump power for the intake valves. Other than the accelerator linkage, these controls will be supplied by American Bosch with the intake valving assembly for the four-cylinder expander.

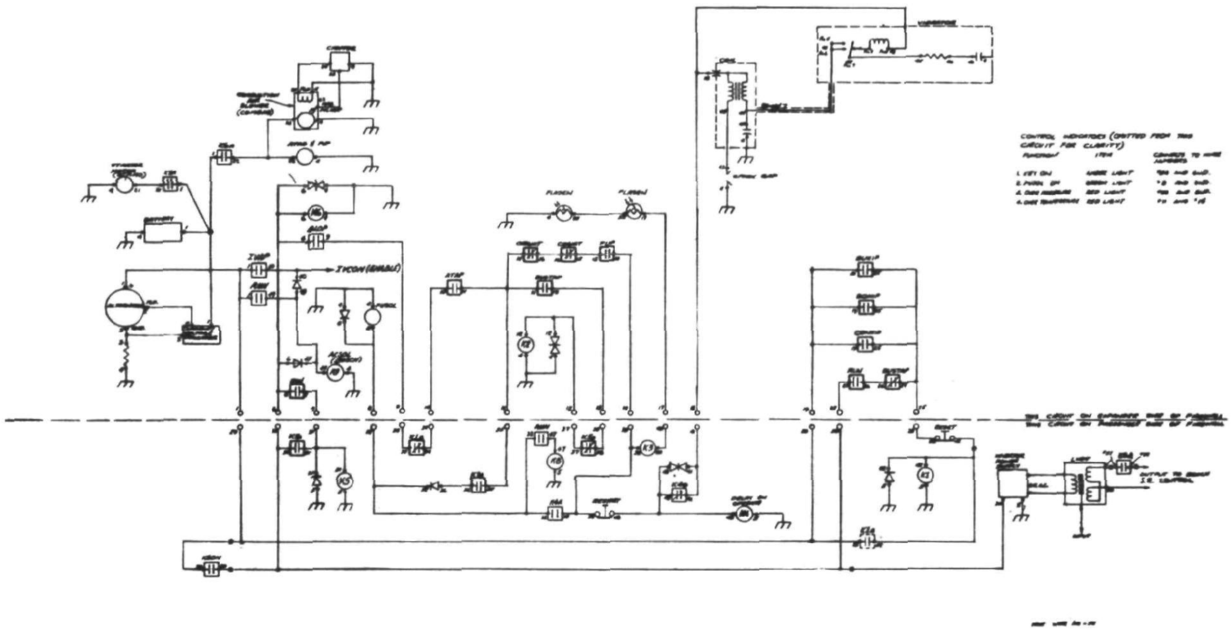

Figure 5.69 Circuit Schematic for Startup Sequencing and Safety Controls.

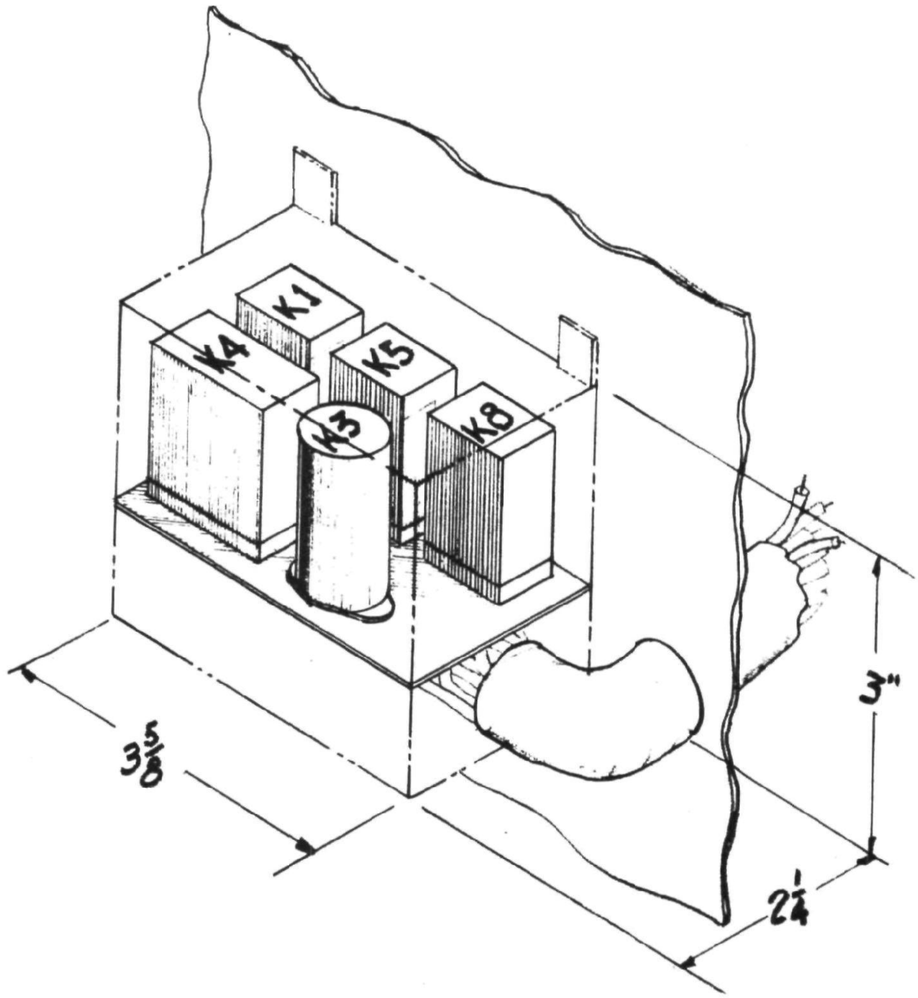

Figure 5. 70 Startup Sequencing and Safety Control Box.

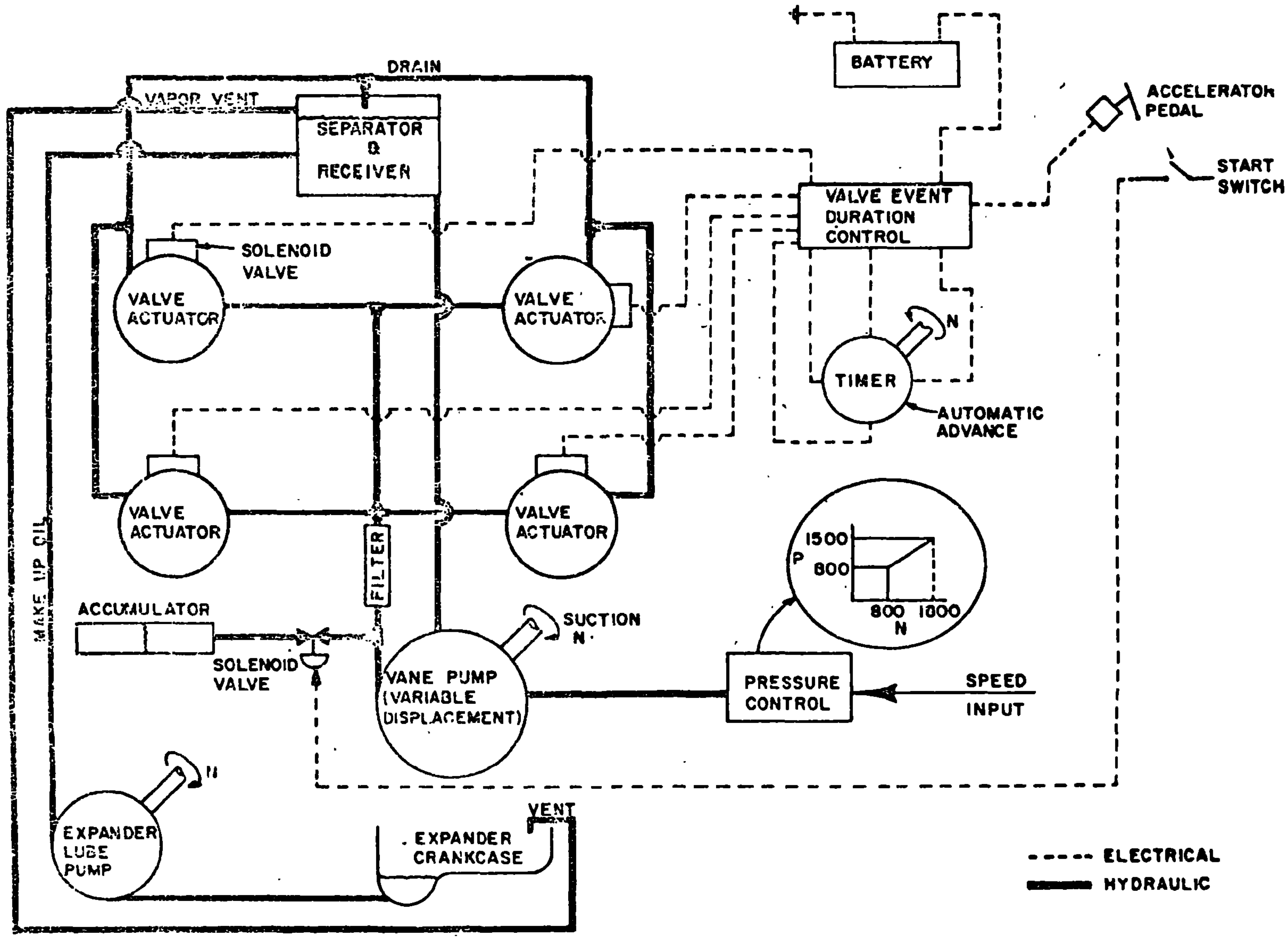

Figure 5.71 American Bosch Valving Schematic.

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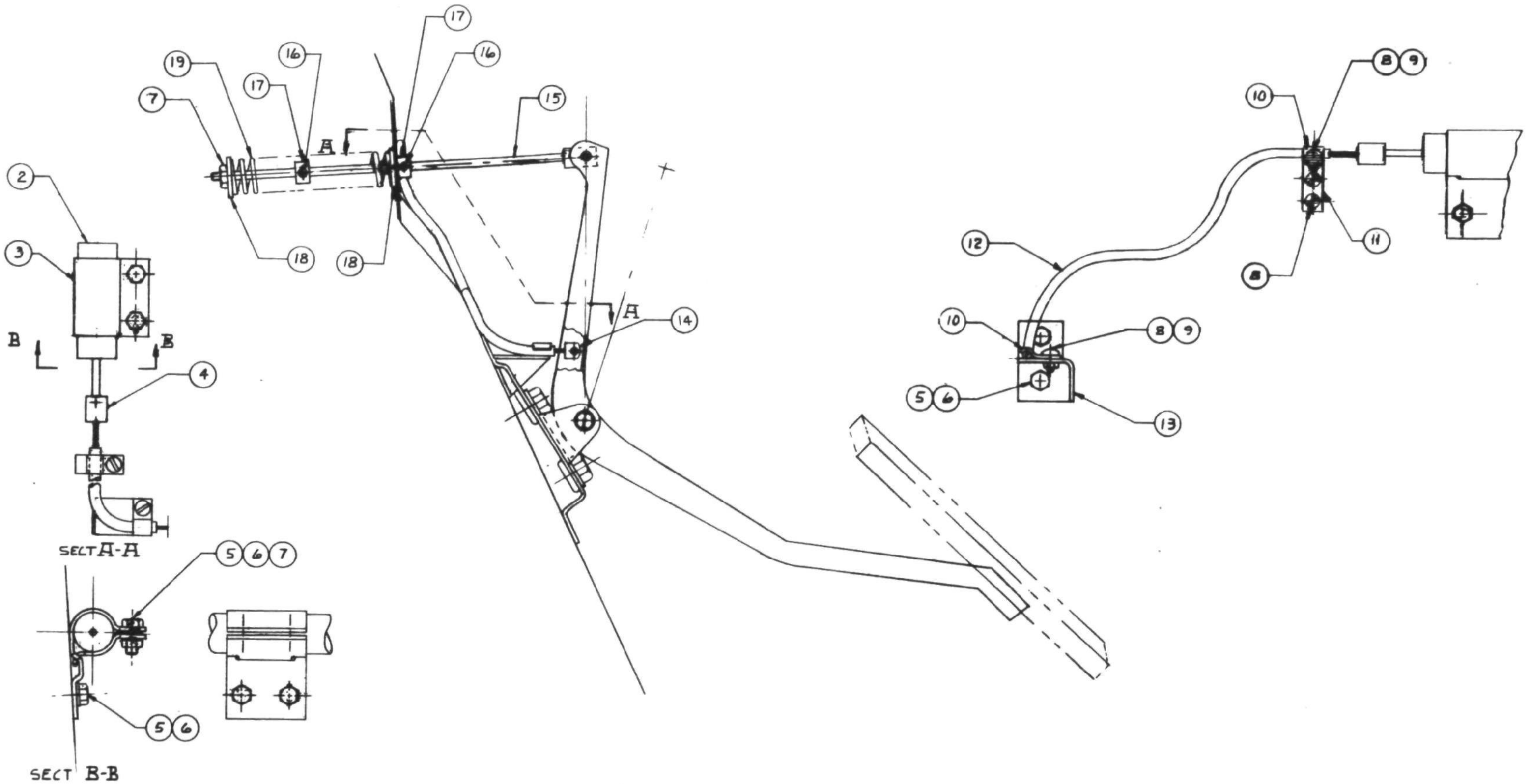

Figure 5.72 Accelerator Linkage to LVDT for Intake Valve Timing Control.

5.6 BOOST PUMP-INDUCER-RESERVOIR SUBASSEMBLY

The components of this subassembly provide the following functions:

- Produce the net positive suction head (NPSH) required by the feedpump for normal operation.
- Produce the NPSH required by the feedpump at the start condition, when condenser pressure is low and all working fluid is at the ambient temperature.
- Eliminate required condenser subcooling, thereby making more efficient use of the available frontal area for condensing.
- Provide reservoir capacity for inventory transfer during start condition and transient operation.
- Prevent separation of lubricant from working fluid in condenser and reservoir.

In Figure 5.73, a flow schematic is presented illustrating the position of these components between the condenser and the feedpump. The inducer is located at the bottom of the engine compartment and serves as the condensate sump pump to maintain the condenser drained of liquid. Part of the flow from the boost pump is used for operation of the inducer. The inducer discharges into the system reservoir located at the top of the engine compartment. This reservoir insures sufficient suction head for proper operation of the centrifugal boost pump during startup and transient operation when working fluid inventory changes in the system may occur. The top of the reservoir is vented to the condensate header at the condenser, so that the reservoir and condensate header pressures are equal. The boost pump provides sufficient pressure to the main feedpump suction line to insure proper inlet valve operation and to insure no cavitation.

This pumping assembly is similar to that used on the 5-1/2 hp systems under test at Thermo Electron, except no inducer is used. In these systems, a horizontal condenser is located on top of the powerplant; the condensate drains directly into the reservoir located under the condenser. In the automotive system, the condensate header is the lowest part in the system, and the inducer was added for draining the condenser and pumping the condensate into the reservoir.

5.6.1 Boost Pump Design

The boost pump requirements are summarized in Table 5.25. The boost pump is driven by the output from the Morse variable speed drive used for the condenser fan speed control (see Section 5.4). The requirements are thus given for two conditions, one representing operation at the expander idle speed (boost pump speed = 1470 rpm) and the other at expander speeds above 550 rpm where the boost pump speed is constant at 2700 rpm. The bottom of the reservoir is 10 inches above the boost pump suction and insures 10 inches head to the boost pump when the reservoir has a low liquid level due to transfer of the working fluid inventory to other locations in the system.

The boost pump layout drawing is illustrated in Figure 5.74. The pump is a centrifugal type with an impeller diameter of 3.5 inches. A screw inducer is used on the pump to meet the NPSH requirement of 10 inches at 2700 rpm. To eliminate the requirement for a dynamic shaft seal, a permanent magnet drive is used. The magnet drive is commercially available and is the type used for the centrifugal boost

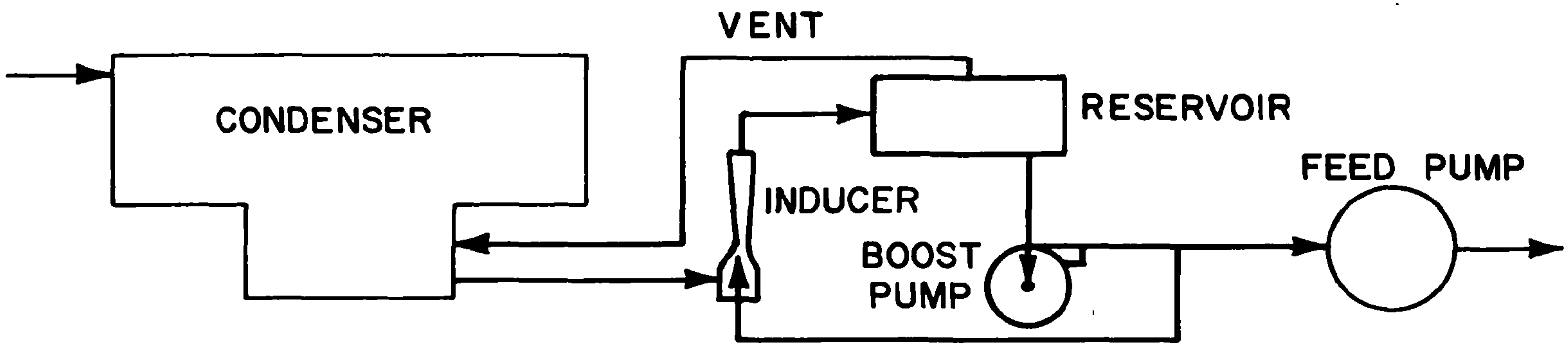

Figure 5.73 Boost Pump-Inducer-Reservoir Subassembly.

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TABLE 5.25

BOOST PUMP PERFORMANCE REQUIREMENTS

• BOOST PUMP AT 1470 RPM (EXPANDER 300 RPM)	
Maximum Flow Rate to Feedpump	7.5 gpm
Maximum Flow Rate to Inducer	8.5 gpm
Maximum Total Boost Pump Flow Rate	16 gpm
Head Rise	7.88 feet
Minimum Available Net Positive Suction Head	5 inches
Shaft Power	0.125 hp
• BOOST PUMP AT 2700 RPM (EXPANDER 550 - 1800 RPM)	
Maximum Flow Rate to Feedpump	17 gpm
Maximum Flow Rate to Inducer	12.4 gpm
Maximum Total Boost Pump Flow Rate	29.4 gpm
Head Rise	26.6 feet
Minimum Available Net Positive Suction Head	10 inches
Shaft Power	0.5 hp

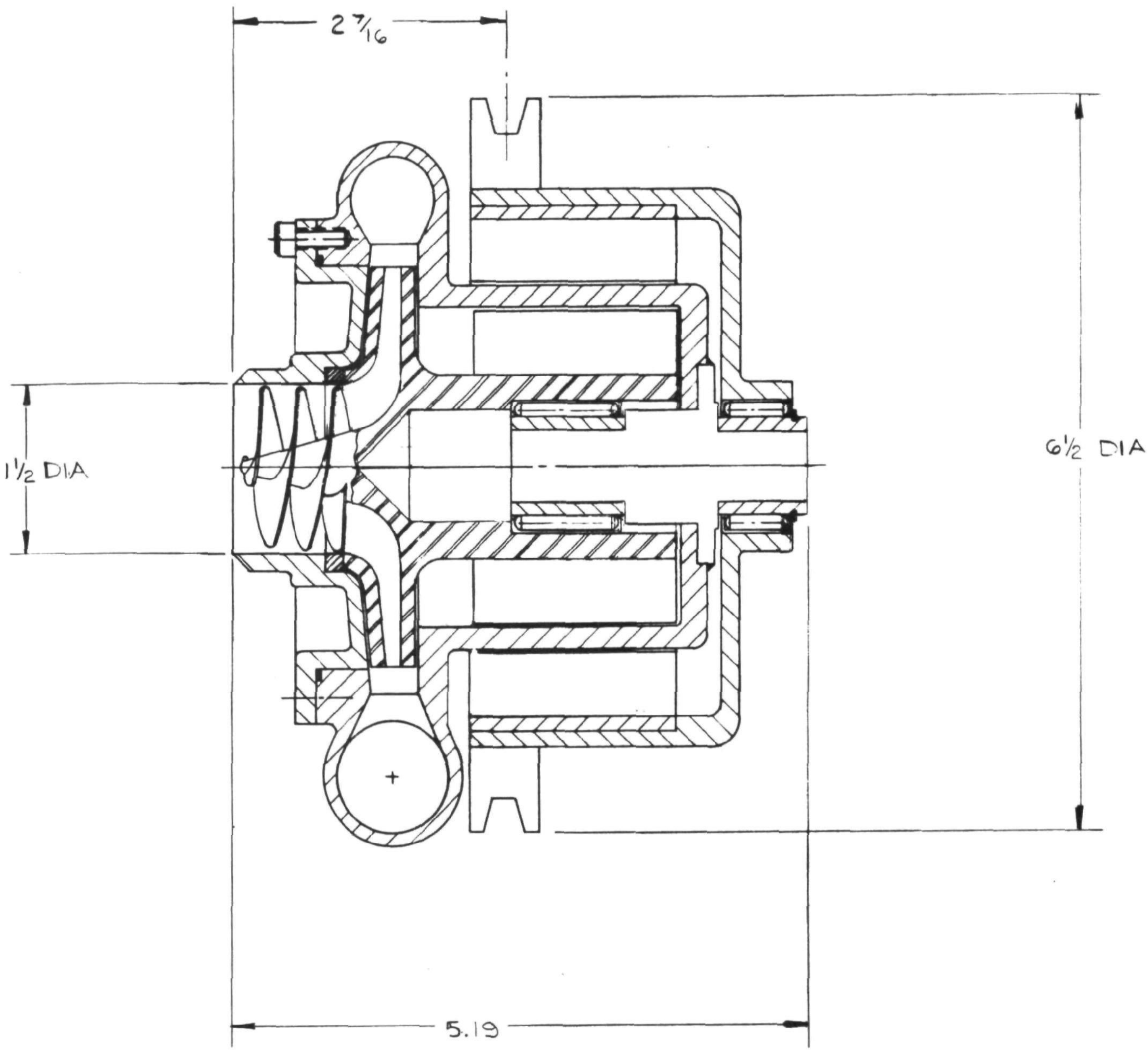

Figure 5.74 Cross Section of Centrifugal Boost Pump.

pump on the 5.5 hp TECO system. In Figure 5.75 the impeller vane and volute construction are illustrated. The impeller uses backward curved vanes as illustrated.

The shaft power required to drive the boost pump is 0.5 hp at 2700 rpm and 0.125 hp at 1470 rpm. The parasitic load from the boost pump is thus very low.

The pump is constructed primarily of aluminum.

5.6.2 Inducer

The inducer performance requirements are summarized in Table 5.26 and the inducer design is presented in Figure 5.76. The required head output is 2.0 feet to pump the liquid from the bottom of the condenser to the reservoir located at the top of the engine compartment. The pump is designed to operate with about 2 inches NPSH. As evident from Figure 5.76, the inducer construction is very simple. Brazed aluminum construction is used and the design is suitable for high volume production. The inducer is brazed to the condenser and reservoir lines.

5.6.3 Reservoir

The reservoir (or receiver) is illustrated in the system packaging drawing of Figure 4.7 of Chapter 4. It is an aluminum tank with a capacity of 1.16 gallons and dimensions 5" diameter by 10" length plus headers. The capacity for the design was based on the total condenser internal volume, so that the system can be started with the condenser completely filled with liquid. The final reservoir size required for the system will be determined experimentally in operation of the complete system to insure adequate capacity for all startup and transient conditions encountered by the system.

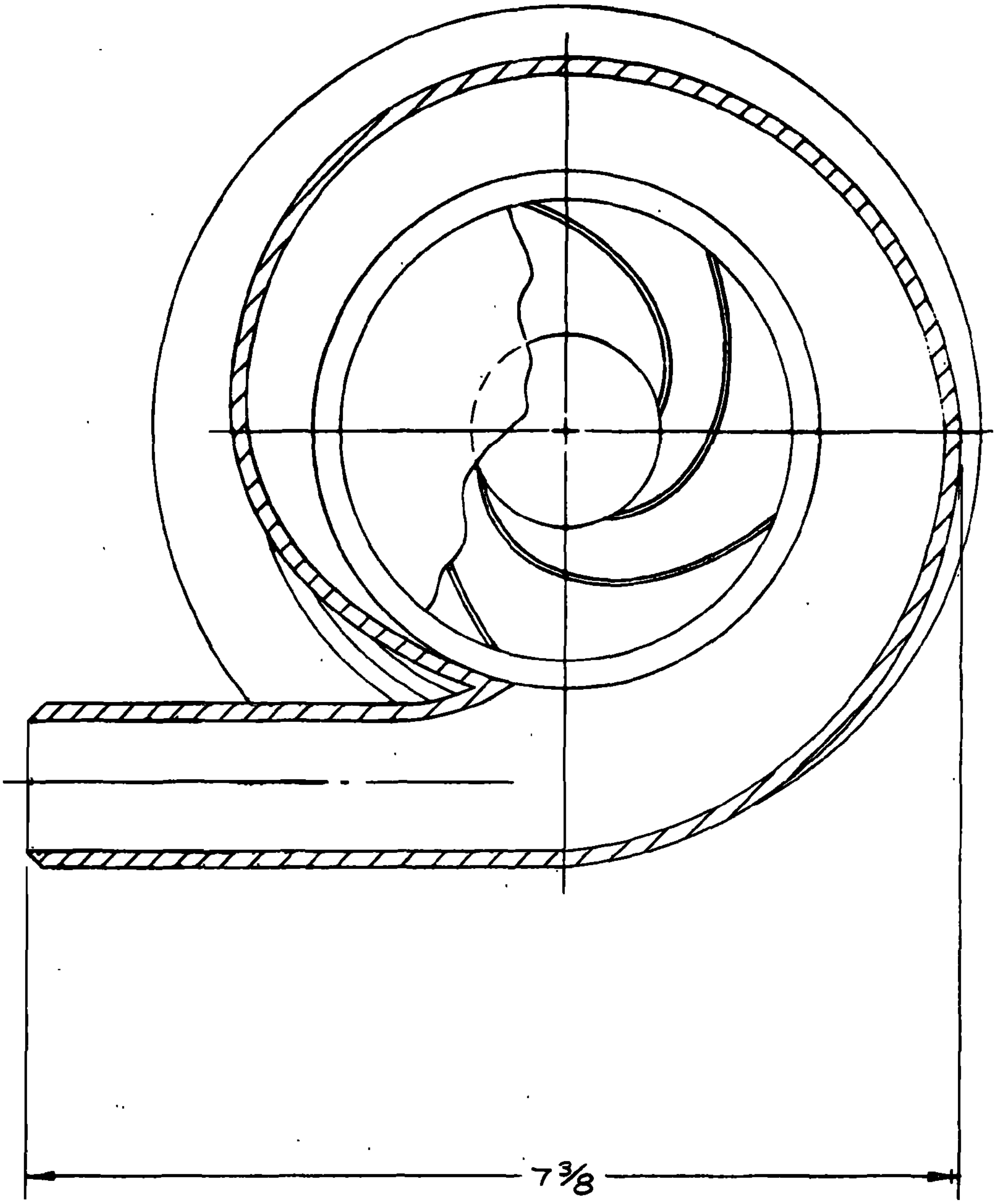

Figure 5.75 Boost Pump Vane and Volute Construction.

TABLE 5-26

INDUCER PERFORMANCE REQUIREMENTS

- BOOST PUMP AT 1470 RPM (EXPANDER 300 RPM)

Primary (Nozzle) Flow Rate	8.5 gpm
Primary Head Available	7.88 feet
Secondary (Condensate) Flow Rate	7.5 gpm
Inducer Head Rise	2.0 feet

- BOOST PUMP AT 2700 RPM (EXPANDER 550 - 1800 RPM)

Primary (Nozzle) Flow Rate	12.4 gpm
Primary Head Required	16.8 feet
Secondary (Condensate) Flow Rate	17 gpm
Inducer Head Rise	2.0 feet

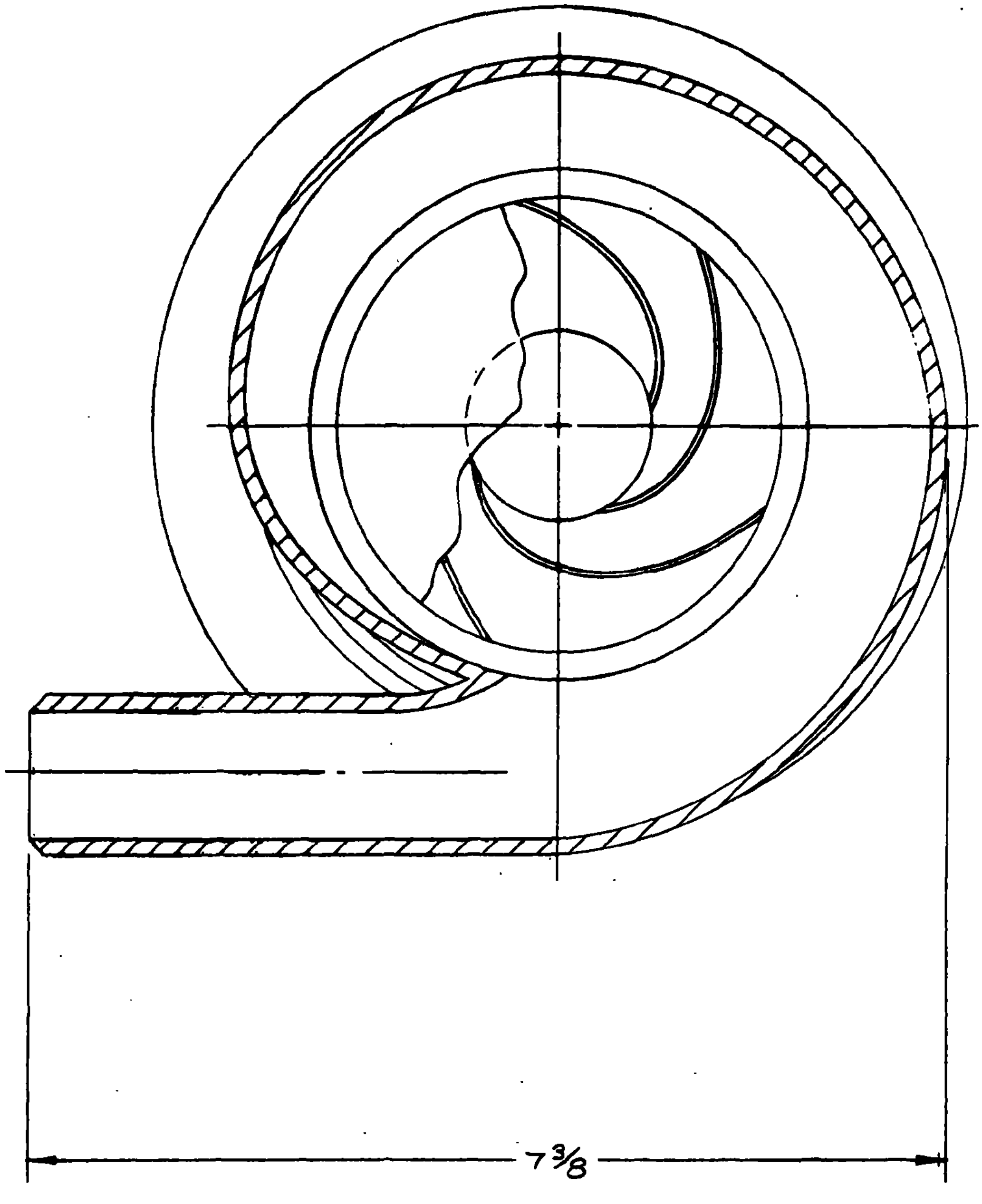

Figure 5. 75 Boost Pump Vane and Volute Construction.

TABLE 5-26

INDUCER PERFORMANCE REQUIREMENTS

- BOOST PUMP AT 1470 RPM (EXPANDER 300 RPM)

Primary (Nozzle) Flow Rate	8. 5 gpm
Primary Head Available	7. 88 feet
Secondary (Condensate) Flow Rate	7. 5 gpm
Inducer Head Rise	2. 0 feet

- BOOST PUMP AT 2700 RPM (EXPANDER 550 - 1800 RPM)

Primary (Nozzle) Flow Rate	12. 4 gpm
Primary Head Required	16. 8 feet
Secondary (Condensate) Flow Rate	17 gpm
Inducer Head Rise	2. 0 feet

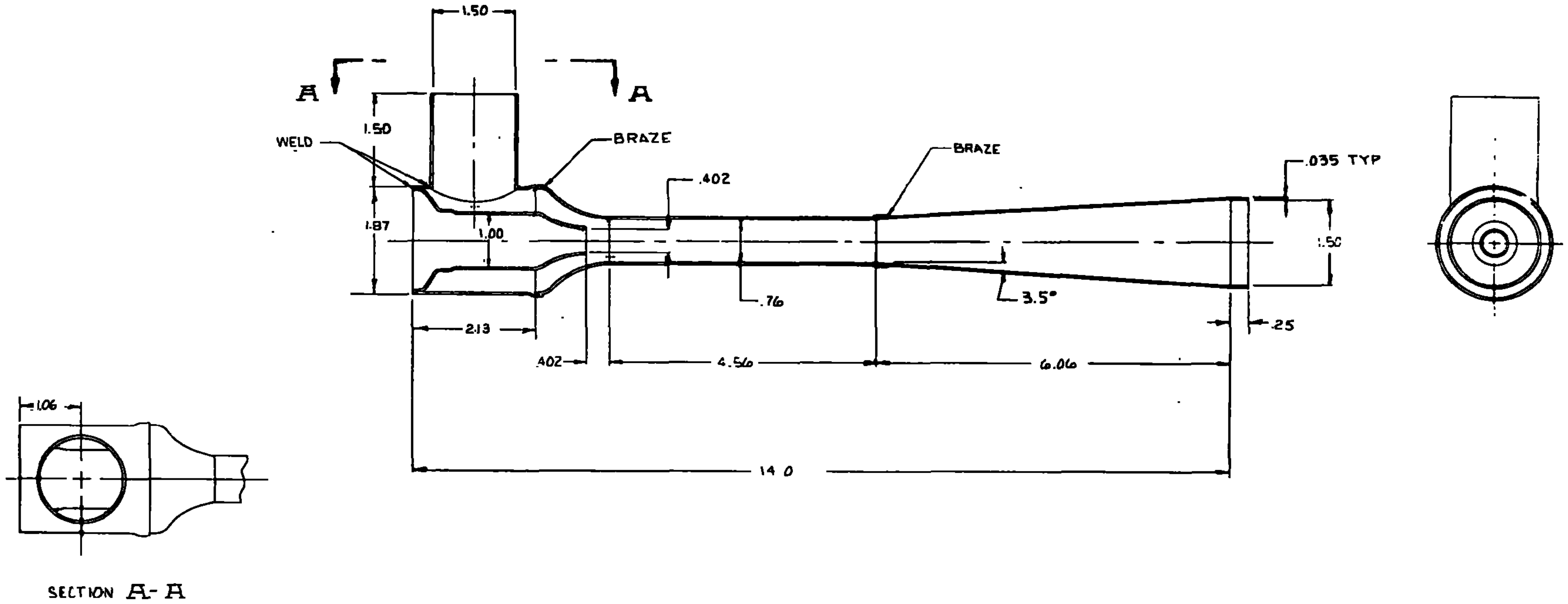

Figure 5.76 Inducer Design.

5.7 ACCESSORY AND AUXILIARY COMPONENTS

The automotive accessories for passenger comfort and convenience selected for the system and incorporated in the packaging described in Chapter 4 are:

- Power Steering - Identical to that on 1972 production Ford Galaxie.
- Power Brakes - Identical to Ford preproduction hydraulic power brake unit.
- Air conditioning Compressor - Identical to that on 1972 production Mach IV Lincoln (swash plate type).
- Heater and Air Conditioning Package - Identical to that on 1972 production Ford Galaxie. To provide hot water to the heating coil, a small heat exchanger will be located in part of the exhaust gas stream from the boiler with circulatory water for transfer of heat.

It was possible to retain the production heating-air conditioning package, even though this package extends into the engine compartment.

The battery-alternator supplies power to both the system and the normal automotive functions requiring electrical power, such as headlamps and the heating - air conditioning blower. A detailed analysis was carried out to insure selection of an adequate alternator and battery capacity for the system. The primary electrical power demands for the Rankine-cycle system are for operation of the combustion system and for startup. The electrical system is 12 V dc as in current automotive practice.

The starting power requirements are summarized in Table 5.27. For the first 25 seconds, the combustion system only is operating with a total electrical power requirement of 2.75 hp. At the end of 25 seconds, the boiler is heated and the starter motor is cranking the expander and valve drive pump, feedpump, boost pump, condenser fans, and automotive accessories. If sufficient boiler pressure exists, as it would if the boiler contained working fluid, the expander will immediately take over. If the boiler is dry, working fluid will be pumped into the boiler, generating the required pressure to operate the system. To handle the latter situation, a maximum of 10 seconds operation of the starter motor is required. The electrical power input for the starter motor is 2.0 hp.

In Table 5.28, the battery supply requirements for the starting sequence are summarized for a nominal 12 V dc system based on these power requirements. The required amps include allowance for voltage drop with the current draw.

The selected battery to meet these starting requirements is a standard AABM size 24C battery with a capacity of 84 amp-hrs and a high rate discharge at 0°C of 500 amps. To insure that this size was adequate for at least two sequential startup attempts, a simulated startup test was made with a 96 amp-hr battery. The results are outlined in Table 5.29. The battery supplied basically the same voltage over both starts, with the terminal voltage dropping to 10.5 volts. in the pre-start operation and to 7.8 volts in the cranking operation.

TABLE 5.27

STARTING POWER REQUIREMENTS

PRE-START (25 sec)	TOTAL POWER: 2.74 HP
• COMBUSTION AIR BLOWER (FULL POWER):	2.28 HP
• ATOMIZING AIR COMPRESSOR:	0.41 HP
• IGNITER, FUEL PUMP AND CONTROLS:	0.05 HP
STARTER MOTOR (10 SEC) (CRANKING SPEED: 300 RPM)	TOTAL POWER: 2.00 HP
• EXPANDER:	0.38 HP
• FEEDPUMP:	0.55 HP
• VALVE DRIVE PUMP:	0.70 HP
• BOOST PUMP:	0.17 HP
• ACCESSORY DRIVE (CONDENSER FANS, ALTERNATOR, POWER STEERING PUMP):	0.20 HP

TABLE 5.28

BATTERY SUPPLY REQUIREMENTS
(FOR NOMINAL 12 V SYSTEM)

PRE-START (25 SEC)

- CURRENT DRAW: 240 AMPS

STARTER MOTOR (10 SEC)

- CURRENT DRAW: 180 AMPS

COMBINATION OF PRE-START AND STARTER MOTOR (10 SEC)

- CURRENT DRAW: 420 AMPS

BATTERY SELECTION

- AABM SIZE: 24C
- CAPACITY: 84 AMP - HR
- HIGH RATE DISCHARGE AT 0°F: 500 AMPS

TABLE 5.29

SIMULATED START-UP TEST
FOR BATTERY

FIRST START - TERMINAL VOLTAGE: 12.5 VOLTS

- PRE-START (25 SEC) - V = 10.5 V: I = 225 AMPS
- CRANKING (10 SEC) - V = 7.8 V: I = 450 AMPS

INTERVAL TIME BETWEEN FIRST AND SECOND START: 20 SEC

SECOND START - TERMINAL VOLTAGE: 12.5 VOLTS

- PRE-START (25 SEC) - V = 10.3 V: I = 220 AMPS
- CRANKING (20 SEC) - V = 7.8 V: I = 450 AMPS

NOTES:

- BATTERY CAPACITY: 96 AMP-HRS
- ALTERNATOR DISCONNECTED DURING TEST SEQUENCE
- TEST PERFORMED ON 1971 FORD LTD

The alternator must be sized to handle the maximum sustained power demand to be encountered in vehicle operation as well as to handle the worst transitory power requirements resulting from system operation on long grades. For these transitory conditions, it is assumed that power will be drawn from both the battery and the alternator; the alternator must have sufficient capacity to prevent total discharge of the battery. In Table 5.30, the alternator specifications and the alternator requirements for sustained driving conditions are presented. The alternator selected is a standard, heavy-duty 14 V dc alternator with 130 amp output at 5000 rpm. This size was based on a continuous current draw of 85 amps for continuous system operation at 70 mph on a 0% grade, plus 35 amps required for normal operation of the vehicle (headlamps, etc.).

The worst transitory condition is operation on long grades requiring high system power output and a resultant high electrical load for operation of the combustion system. For these conditions, power would be taken from both the battery and the alternator. For operating conditions requiring 80% of full system power, the total current requirement is 236 amps; 130 amps are supplied by the alternator and 106 amps are supplied from the battery. Two operating conditions were evaluated to determine the allowable time and total distance traveled with the selected battery-alternator combination, with the results indicated in Table 5.31. At 15 mph on a 30% grade, 80% of full system power is required. The vehicle could travel 3.5 miles continuous at these conditions and the selected alternator-battery is more than adequate. For 70 mph vehicle speed on a 5% grade, 77% of full power is required. The vehicle could travel 17.5 miles at these

TABLE 5.30

SUSTAINED POWER REQUIREMENTS

ALTERNATOR SPECIFICATIONS

- STANDARD HEAVY DUTY 14 VOLT ALTERNATOR
- 130 AMP OUTPUT AT 5000 RPM
- 90 AMP OUTPUT AT 2000 RPM (IDLE)

FEDERAL EMISSION DRIVING CYCLE RATIOED FOR 200 MILES

- AVERAGE BURNING RATE: 12% FULL POWER
- CURRENT DRAW: 56 AMPS

CRUISE AT 70 MPH ON 0% GRADE FOR 200 MILES

- BURNING RATE: 30% FULL POWER
- CURRENT DRAW: 85 AMPS

NORMAL ELECTRICAL LOADS

- ACCESSORIES (AIR CONDITIONING, HEADLIGHTS, ETC.)
- CURRENT DRAW: 35 AMPS

MAXIMUM SUSTAINED POWER REQUIREMENTS

- CRUISE AT 70 MPH PLUS NORMAL LOADS
- CURRENT DRAW: 120 AMPS

TABLE 5.31

TRANSITORY POWER REQUIREMENTS
BATTERY YIELD TAKEN AT 0°F

15 MPH VEHICLE SPEED ON 30% GRADE (80% FULL POWER)	
• ALLOWABLE TIME AT THIS CONDITION:	13.8 MINUTES
• MILEAGE TRAVELED	3.5 MILES
70 MPH VEHICLE SPEED ON 5% GRADE (77% FULL POWER)	
• ALLOWABLE TIME AT THIS CONDITION:	15 MINUTES
• MILEAGE TRAVELED:	17.5 MILES
CONDITION OF 80% FULL POWER BURNING RATE	
• TOTAL CURRENT REQUIREMENT:	236 AMPS
• ALTERNATOR SUPPLY:	130 AMPS
• BATTERY CURRENT DRAW:	106 AMPS

conditions before difficulties were encountered with the battery draw. In Table 5.32, the worst grades in the U.S.A. are summarized, with the most difficult being 11.2 miles with an average grade of 5.1%. In Table 5.33, the recorded grades on a cross country round trip between Chicago, Illinois and Portland, Oregon are presented. For the 3058.5 mile trip, 24.0 miles total had a grade of 5%, 10.5 miles total had a grade of 6% and 3.0 miles had a grade of ≥ 7%.

The selected alternator-battery should thus be adequate for all sustained and transitory driving conditions to be encountered by the vehicle.

TABLE 5.32

HIGHWAY GRADES IN SOUTHWEST UNITED STATES

LOCATION	GRADE LENGTH (MILES)	AVERAGE GRADE (%)
SUPERIOR, ARIZONA	4.0	5.2
DAVIS DAM, ARIZONA	11.2	5.1
GRAPEVINE, CALIFORNIA	13.8	3.4
BAKER, CALIFORNIA	17.0	3.4
JACOB LAKE, ARIZONA	13.3	3.5
FARNEL, ARIZONA	2.1	6.0
MINGUS MOUNTAIN, ARIZONA	2.6	6.0

TABLE 5.33

HIGHWAY GRADES FOR INTERSTATE ROUTES

- PORTLAND TO CHICAGO CROSS-COUNTRY RUN IN 1966
- GRADES RECORDED FOR 3058.5 MILE ROUND TRIP

RECORDED GRADES (+ or - 1/2 %)	MILES	PERCENT
0%	1,399.0	45.7
1%	1,090.0	35.6
2%	371.0	12.1
3%	118.0	3.9
4%	43.0	1.4
5%	24.0	0.8
6%	10.5	0.3
7+%	3.0	0.2