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MTI Report No. 80ASE142DR1

Automotive Stirling Engine Mod I Design Review Report

Volume II

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Mechanical Technology Incorporated

August 1982

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN 3-32

for
U.S. DEPARTMENT OF ENERGY
Conservation and Renewable Energy
Office of Vehicle and Engine R&D

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B.8 CONTROL SYSTEM AND AUXILIARIES

8.1 INTRODUCTION

The Basic Stirling Engine, which has previously been subject to a Design Review, forms together with auxiliaries and control system the Stirling Engine System. The auxiliaries and the control system for the ASE MOD I will be described in the following chapters.

These components and sub-systems have the following main functions:

- Provide the required fuel and air flows for a well controlled combustion process, generating heat to the Stirling cycle.

- Provide a driver acceptable method for controlling the power output of the engine.

- Provide adequate lubrication and cooling water circulation.

- Generate the electric energy required for engine and vehicle operation.

- Provide a driver acceptable method for starting, stopping and monitoring the engine.

- Provide a guard system, that protects the engine at component or system malfunction.

In chapter 8.2 the control principles and the way the different components and sub-systems interact is described.

Chapters 8.3 and 8.4 contain detailed descriptions of the different auxiliaries, the air fuel system, and the power control systems.

Chapter 8.5 covers the electronics and in chapter 8.6 the arrangement and location of auxiliaries and other major components are described.

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The control system and auxiliaries of the MOD I engine are to a great extent based on P40 and P75 experience. Within the ASE program work is being performed to improve existing, and develop new, concepts of components and sub-systems. Some components have already been tested successfully, e.g. check valves and speed increasing transmission for the combustion air blower, whereas the function of others, e.g. air fuel system, power control valve actuator, and microprocessor based electronics, remains to be verified in practice. The final testing and adjustment of some of these will be possible only in the engine test cell, or even in the vehicle.

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8.2 ENGINE CONTROL SYSTEMS

8.2.1 The Controlling Systems

In the internal combustion engine power output is primarily controlled by the intake throttle valve position which specifies the cylinder mass air charge.

The Stirling engine cannot become self-sustaining and deliver shaft output until the heater tubes have been heated by the external combustion. Before the vehicle can move off from a cold start, air flow over the heater is established by a battery driven blower. Fuel and ignition are added to start combustion, and the heater tubes are warmed up. After it has been briefly cranked by the starter, the engine becomes self-sustaining. The blower drive is transferred from the electric motor to the engine and the power control system is activated to vary engine power output.

The applied automotive power control system varies the engine hydrogen pressure as required to the desired power output. In order to simulate the accelerating and decelerating response characteristics of an internal combustion engine electronic programming of control valves is used.

The main combustion air control is accomplished by a variable ratio belt drive, giving the combustion air blower a speed proportional to engine load.

An air throttle on the blower inlet is used for fine adjustments of the combustion air flow, in order to keep heater tube temperature within its limits.

Emission limitations are satisfied by appropriate control of the air/fuel ratio.

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Safety systems and guards are provided to protect the engine from malfunctions. Fig 8.2:1 shows the control systems described within this section:

Power Control
Burner Blower Speed Control
Temperature Control
Air/Fuel Control
Start-Stop Sequences
Guards

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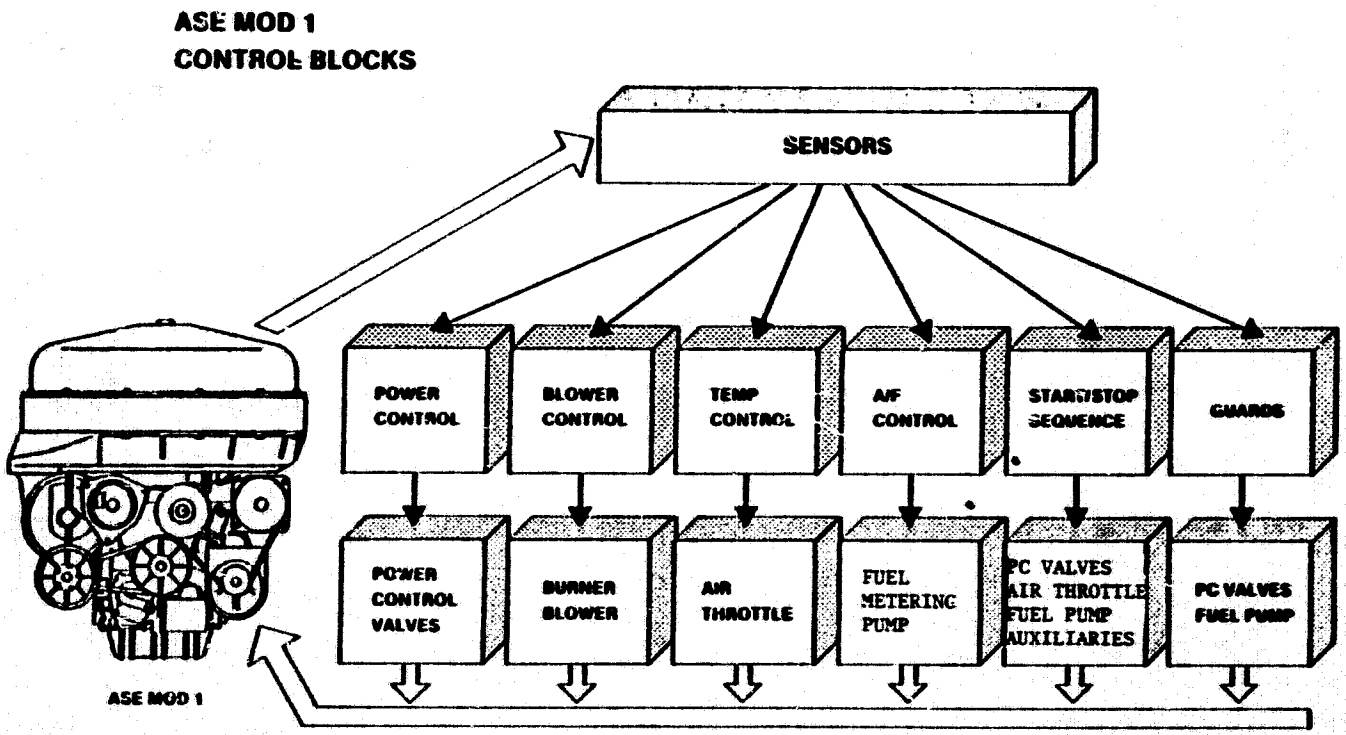


Fig. 8.2:1 ASE Mod I Control Systems

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8.2.2 Control System Interactions

The engine control systems interact according to fig 8.2:2.

Primary System and Secondary, Slave Systems

The primary action is taken by the power control system. The driver sets, via the accelerator, an engine power demand. This is converted by the power control system to a hydrogen pressure within the engine, thus giving a certain engine power output.

Secondary systems are blower speed control, temperature control and air/fuel control. The engine load conditions are sensed by the power control sensors, and converted by the burner blower speed control to a convenient blower speed. The demand of heat to the engine for engine output is sensed by the thermocouples within the heater (constant heater tube temperature). Via the temperature control the demand is converted to a throttle position, giving the right amount of combustion air to the engine.

An air flow transducer measures the combustion air flow and the appropriate amount of fuel (according to excess air demands) is delivered to the engine from the air/fuel system.

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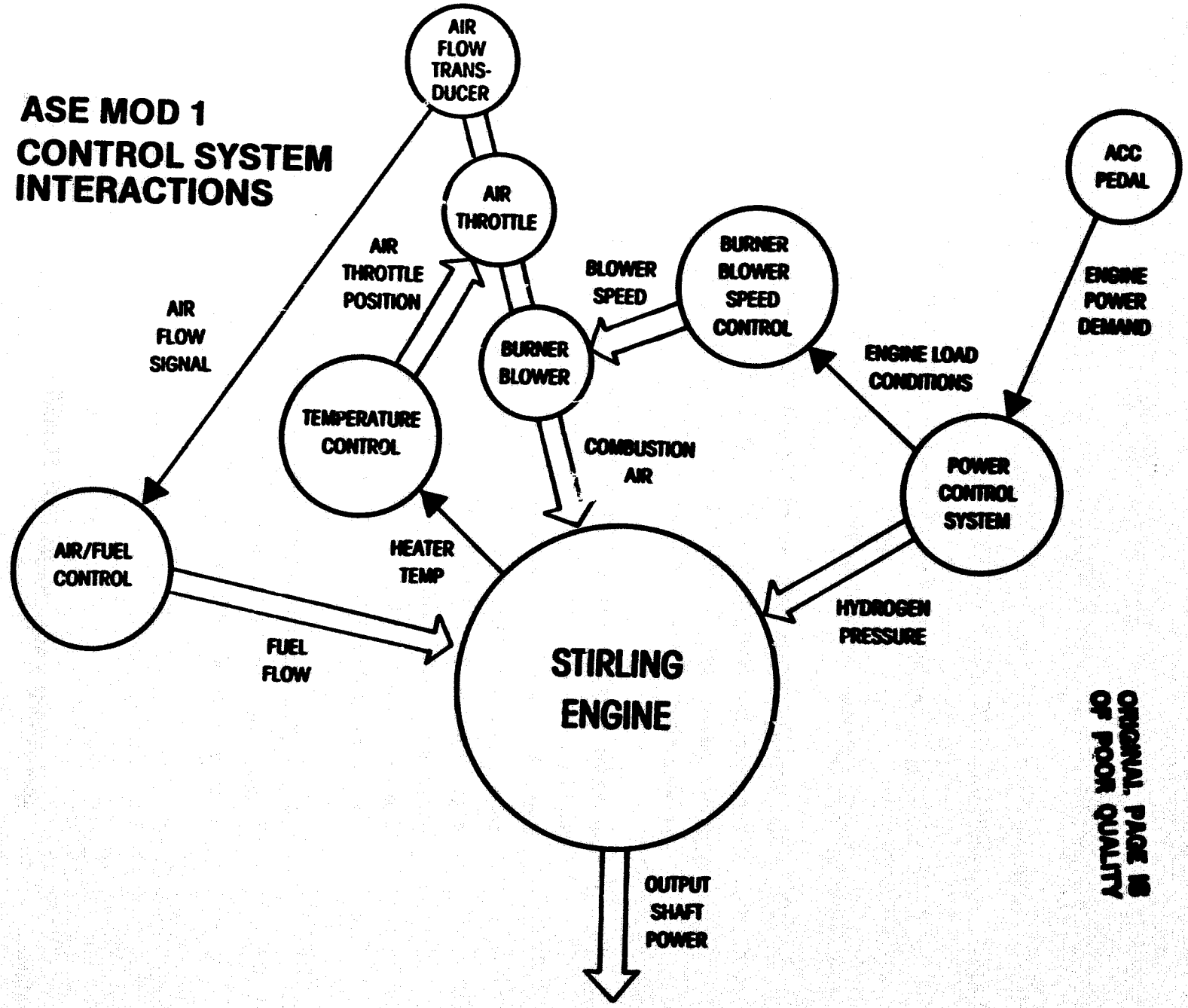


Fig. 8.2:2 ASE Mod I Control System Interactions

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8.2.3 Control Characteristics

8.2.3.1 Power Control Characteristics

The power control system shall via the accelerator (driver) control the engine output in such a way that the driveability of the engine/ /vehicle combination is subjectively acceptable, and so that switching between Stirling powered vehicles and I C powered vehicles would not introduce unacceptable operator adjustments. See power control system specification chapter 8.4.4.

Stirling power control methods compared to those of petrol- and diesel engines are described as follows.

Petrol Engine:

The driver operates the throttle directly. Thus the whole control range can be covered. There is no governor, neither for normal operation nor for idling. The torque curves in the idling area are steep enough to get a stable idling speed.

Diesel Engine:

The pressure in the engine is determined by the position of the fuel pump rack. Idling is not possible without using a governor because the working point is unstable. Often the governor is also used as a maximum speed limiter in view of the soot limit. Under normal driving conditions the governor can be dispensed with. Either the speed or the vacuum in the inlet manifold is fed back.

Stirling engine:

The flat idling torque curves of a Stirling engine are similar to a diesel engine and a torque controlled idling speed is not possible on a Stirling engine. Fuel economy reasons as well as design requirements make pure speed control unacceptable for the selected

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method of power control. The ASE MOD I has power control characteristics similar to a diesel engine at idling. At higher loads and increased speed the control shifts gradually towards torque control.

As mentioned earlier, the way to change power output from the engine is to change the working gas pressure. How this is done to satisfy the control demands is described in chapter 8.4.

Description of the Power Control Loop:

Note: All values, set points, breaking points etc. may be changed during the proceeding development work or the initial engine testing phase.

Fig 8.2:3 shows the power control scheme of ASE MOD I.

(All symbols refer to fig 8.2.3.).
The driver sets the accelerator position which is detected by a potentiometer. That signal (A) together with actual engine speed (RPM) enters a torque rise function (fig 8.2:4).

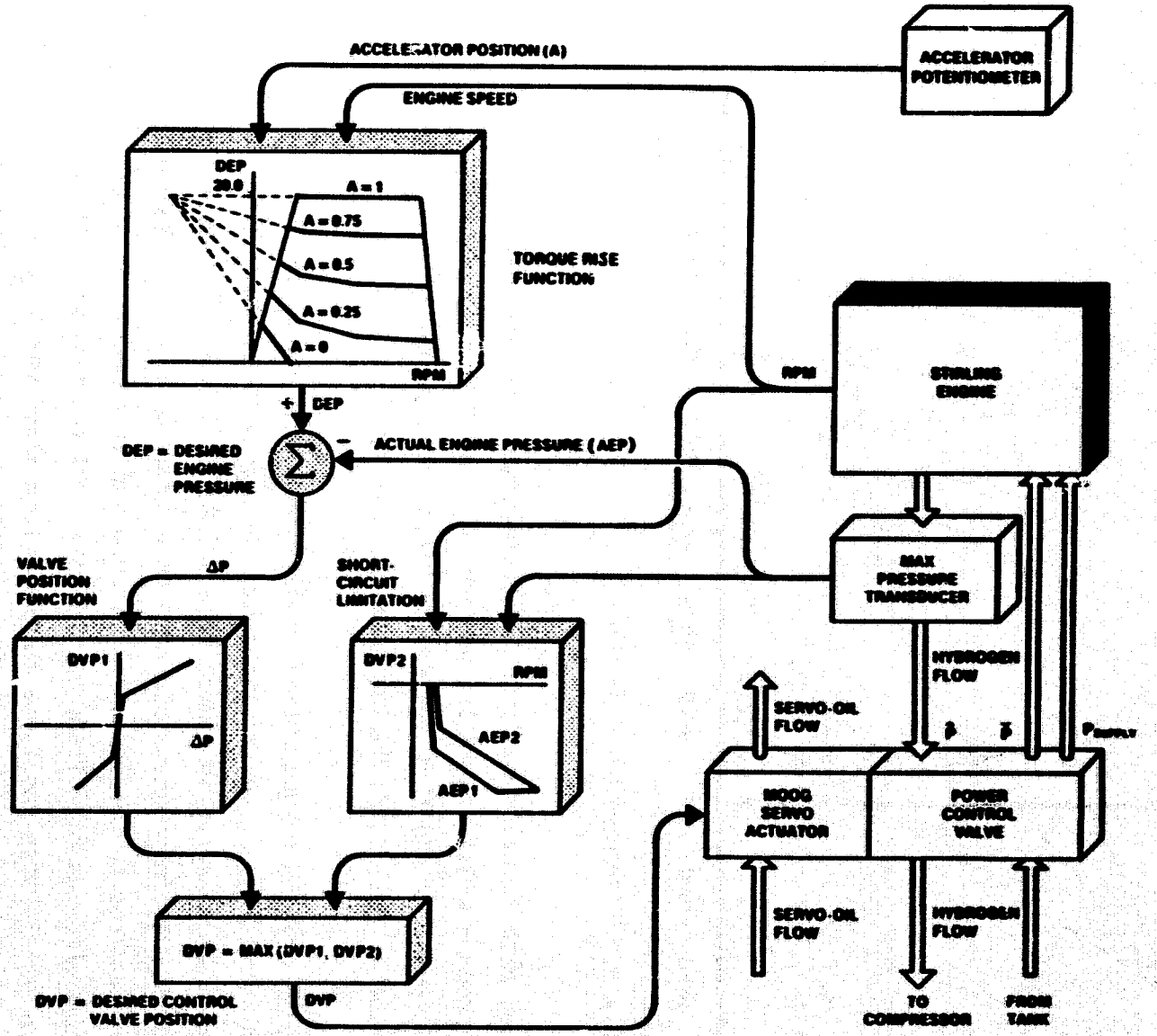


Fig. 8.2:3 Power Control Scheme of ASE Mod I

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ASE MOD 1 DESIRED HYDROGEN PRESSURE (DEP) AS A FUNCTION OF ACCELERATOR LEVEL (A) AND SPEED (RPM)

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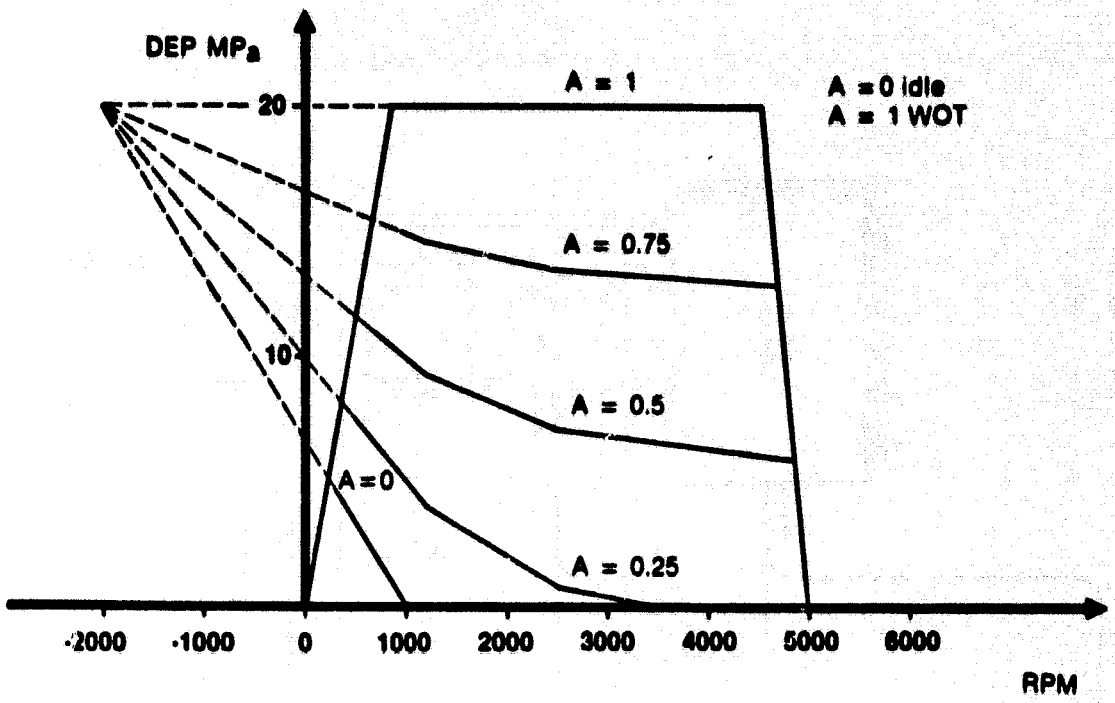


Fig 8.2:4 Torque Rise Function.

The torque rise function converts the accelerator position and engine speed to a desired engine pressure in such a way that stable idling is possible and that the driver "feels" a significant torque rise at part loads during speed decreasing.

Fig 8.2:5 shows the torque curves at different mean pressures. A typical road load curve is also shown. It can be seen that at idle the pressure curve is almost parallel to road load. That means that a stable idling point would have been impossible if accelerator position had been proportional to pressure and independent of speed. The torque rise function reshapes the part load torque curves for different accelerator positions to the dotted lines making the torque curves always steep enough to road load insuring a stable control.

Within the torque rise function there is also a limiting curve preventing high pressure at extreme low speeds and also guarding

ASE MOD 1

TORQUE AS A FUNCTION OF SPEED FOR DIFFERENT MEAN PRESSURES AND FOR DIFFERENT ACCELERATOR LEVELS

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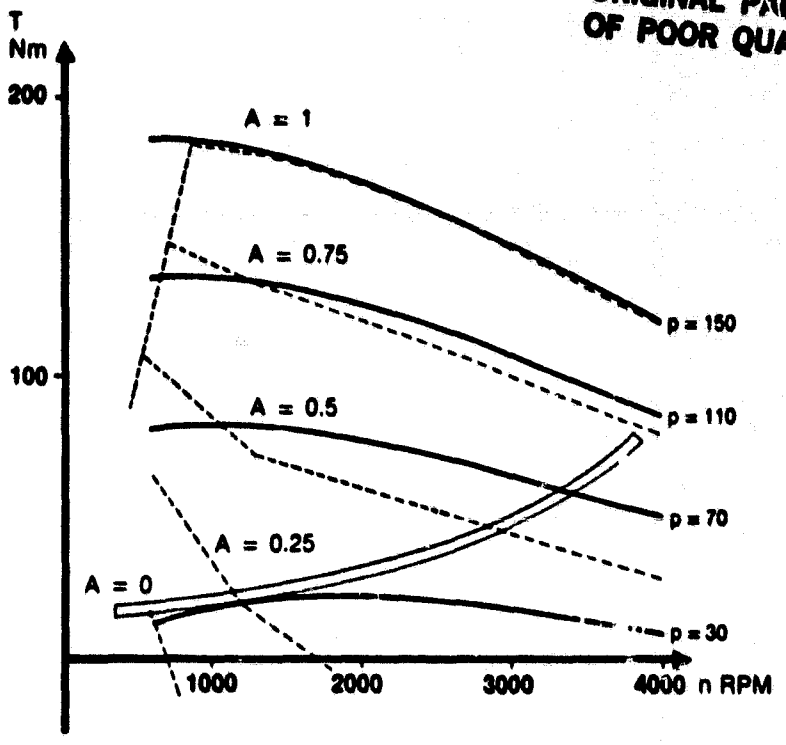


Fig 8.2:5 Torque as a Function of Speed at Different Mean Pressures and at Different Accelerator Levels.

against lack of air during very high load conditions at low speed.

There is also an overspeed limiting curve reducing max desired pressure at speeds exceeding 4 500 RPM.

The desired engine max pressure from the torque rise function is compared to the actual pressure from a max pressure transducer. The difference (Δp) enters a valve position function where a desired control valve position will be determined (fig 8.2:6).

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ASE MOD 1 POWER CONTROL VALVE POSITION (DVP 1) VS DIFFERENTIAL PRESSURE (ΔP)

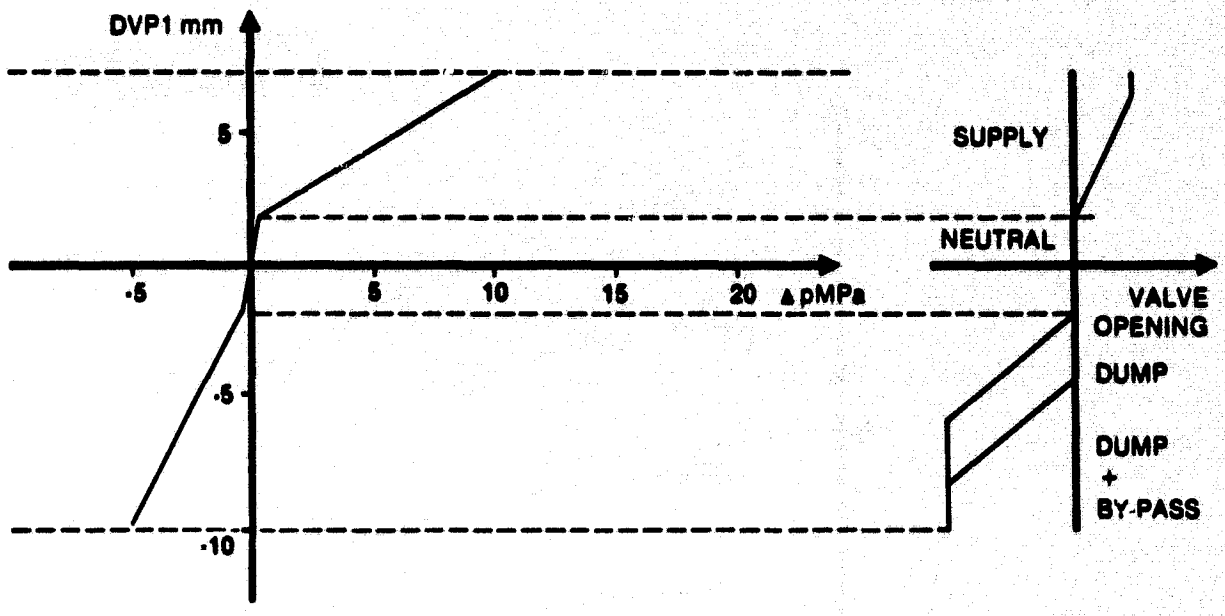


Fig 8.2:6 Control Valve Position Function.

The slope of this function is different depending on whether valve position should be in dump short-circuiting position, the neutral dead band or supply position. The exact shape of the slopes will be determined at the initial testing phase of the engine.

The difference between actual and desired pressure is not sufficient information for power control positioning during all load conditions.

During periods of engine short circuiting, the control valve position must also be determined by actual engine pressure and actual engine speed.

Fig 8.2:7 schematically shows how the negative torque during short-circuiting of a Stirling engine is depending on pressure and speed.

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The desired negative torque due to the driveability of the engine is also shown in fig 8.2:7.

To follow the desired curve, the short circuiting holes within the control valve must be adjusted very carefully to actual engine pressure and actual engine speed.

Fig 8.2:8 shows the short-circuiting limitation function.

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ASE MOD 1

SCHEMATIC CURVES OF SHORT CIRCUITING EFFECTS

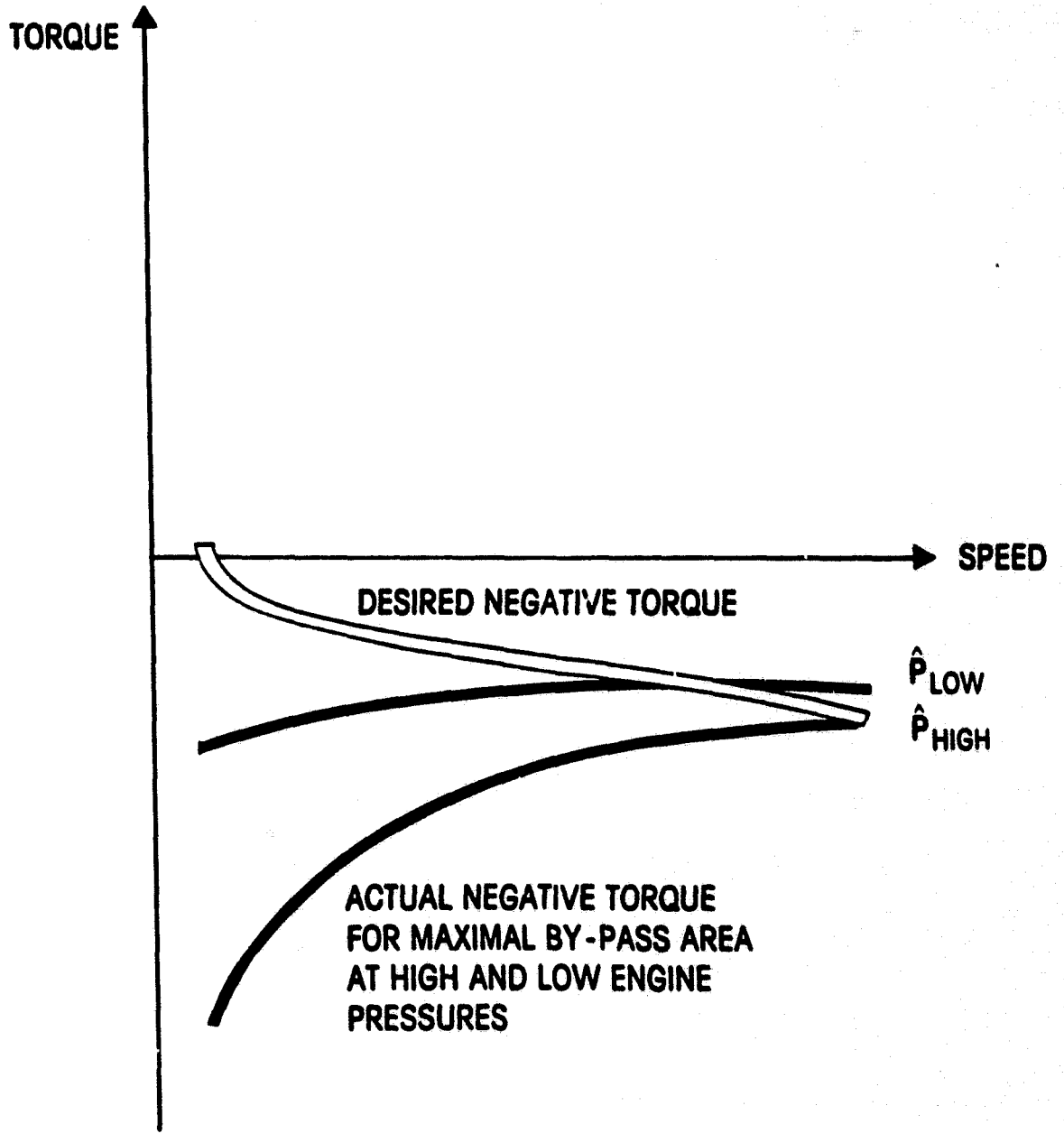


Fig 8.2:7 Schematic Curves of Short-Circuiting Effects.

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**SHORT-CIRCUITING LIMITATION
MAX BY-PASS AREA AS A FUNCTION
OF ACTUAL SPEED AND HYDROGEN
PRESSURE**

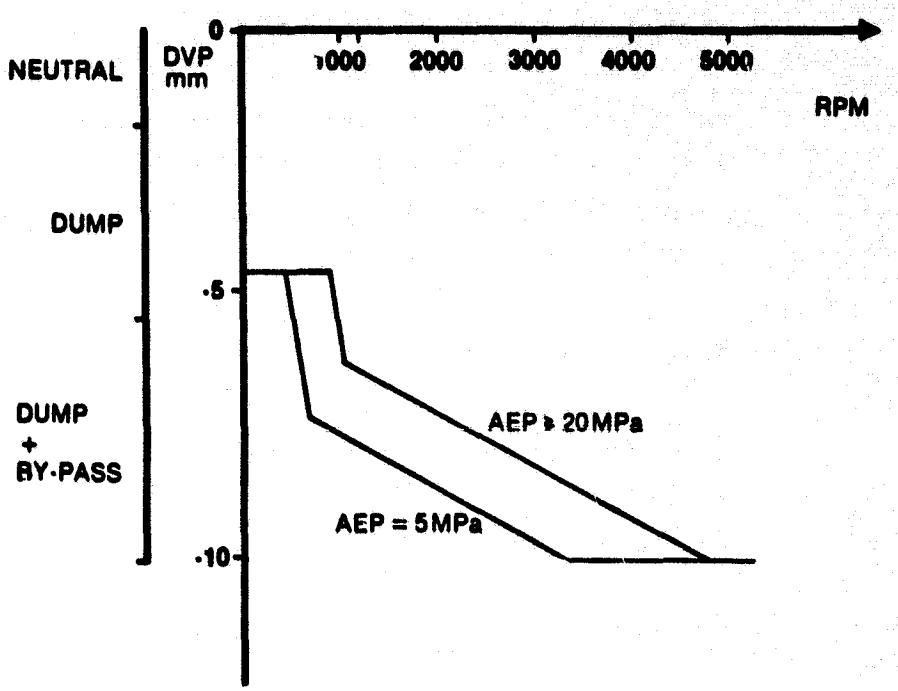


Fig 8.2:8 Short Circuiting Limitation Function.

The short circuiting limitation function limits the maximum available short-circuiting area depending on actual engine pressure and engine speed.

The highest value from the valve position function and the short-circuiting limitation function will be selected as the desired valve position and will be sent to the control valve actuator.

The actuator (see 8.4.8.2) adjusts the control valve position, if necessary, for appropriate action (supply, steady state, dump and short-circuiting) by the power control system.

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8.2.3.2 Burner Blower Speed Control Characteristics

With the new type of blower and air throttle (see section 8.3.6, 8.3.7) used on ASE MOD I, it is essential for blower power consumption and blower efficiency that the blower speed (amount of air) is adjusted in such a way that restriction losses in the air throttle are eliminated as far as possible. Therefore, the ASE MOD I will be equipped with a blower speed control system, which will control the blower variator ratio depending on engine load in a way that most of the air throttle restriction will be eliminated.

Specification:

The burner blower speed control shall control the blower speed in such a way that blower capacity exceeds air demand with 10% during most load conditions.

- The Burner Blower Speed Control Loop:

Note: All values, set points, breaking points etc. may be changed during the proceeding development work or the initial engine testing phase.

Fig 8.2:9 shows the burner blower speed control scheme of ASE MOD I (All symbols refer to fig 8.2:9.).

The burner blower speed loop is a kind of feed forward system for the demand of air.

The correct amount of air is set by the temperature control (section 8.2.3.3). The blower speed control loop adjusts the blower speed so that the available amount of air is slightly higher than the demand.

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Engine max pressure ($\hat{p}$) and engine speed (RPM) from the power control transducers are multiplied and the product enters an air demand map (fig 8.2:10).

This curve is an approximation of the air demand of the engine versus speed and pressure.

This air demand will enter a blower characteristic function where the corresponding blower speed will be calculated (fig 8.2:11).

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ASE MOD 1
BURNER BLOWER SPEED CONTROL SCHEME

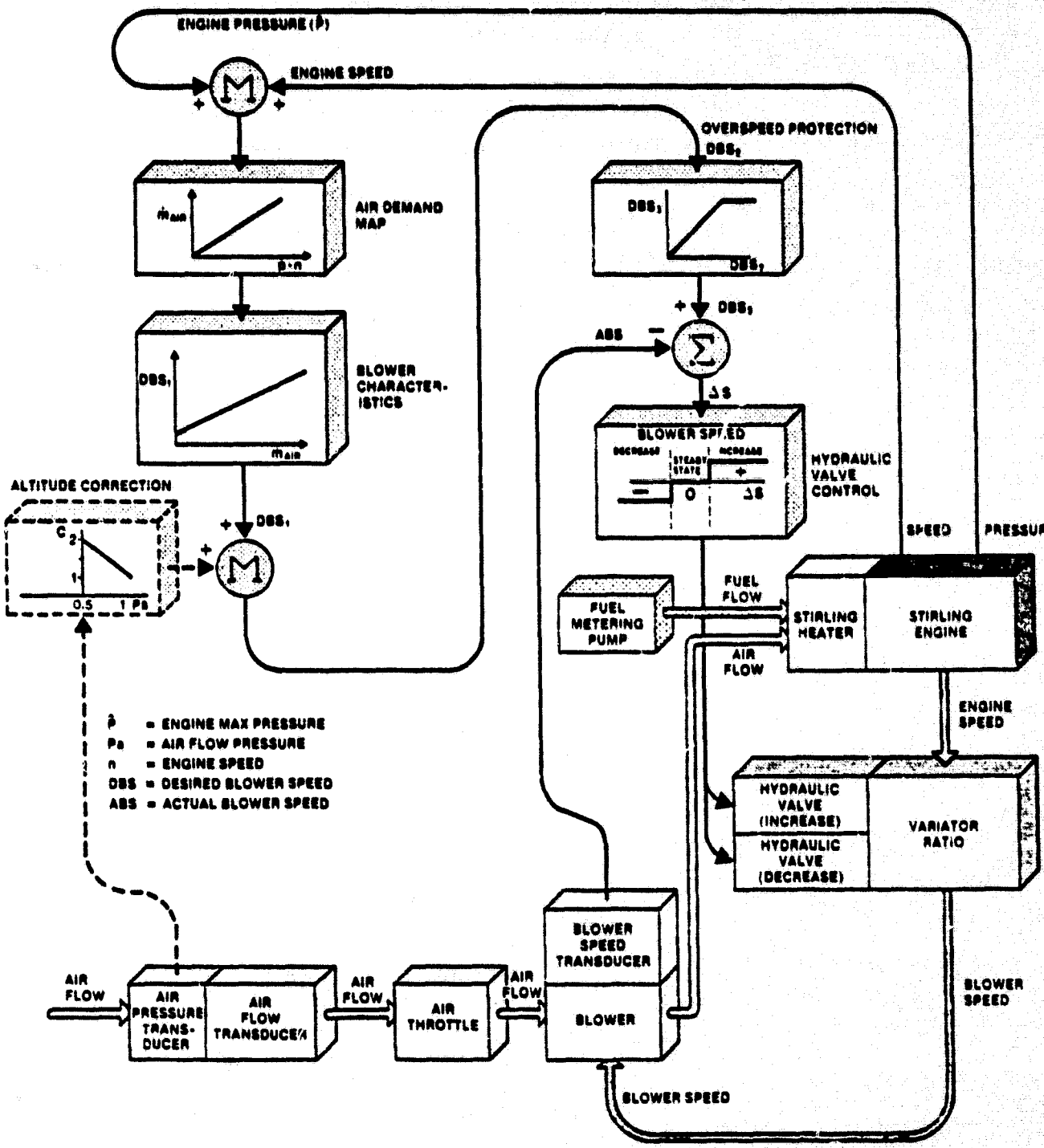


Fig S.2:9 ASE MOD I Burner Blower Speed Control Scheme.

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ASE MOD 1

BURNER BLOWER SPEED CONTROL
AIR DEMAND MAP

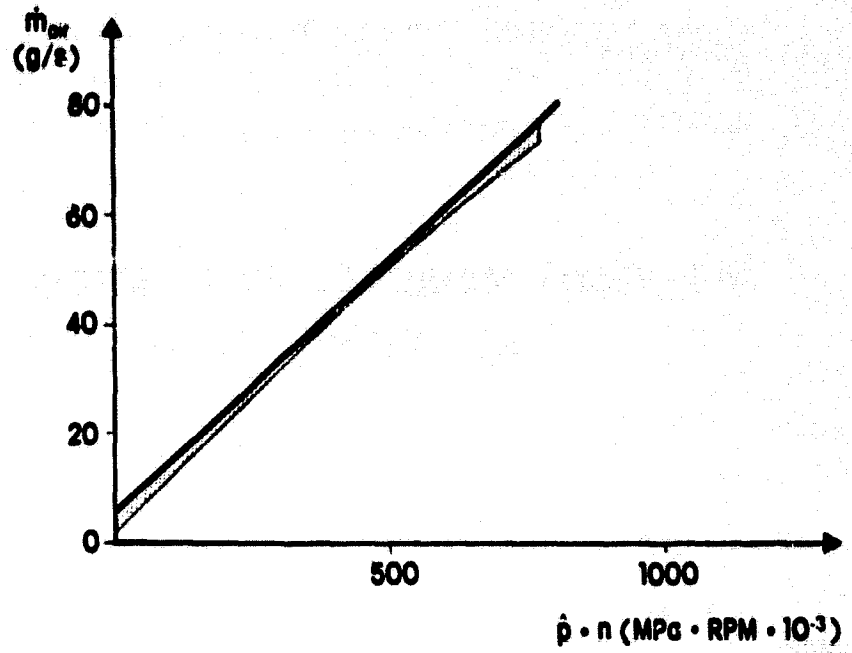


Fig 8.2:10 ASE MOD I Burner Blower Speed Control Air Demand Map.

ASE MOD 1

BURNER BLOWER SPEED CONTROL
BLOWER CHARACTERISTICS

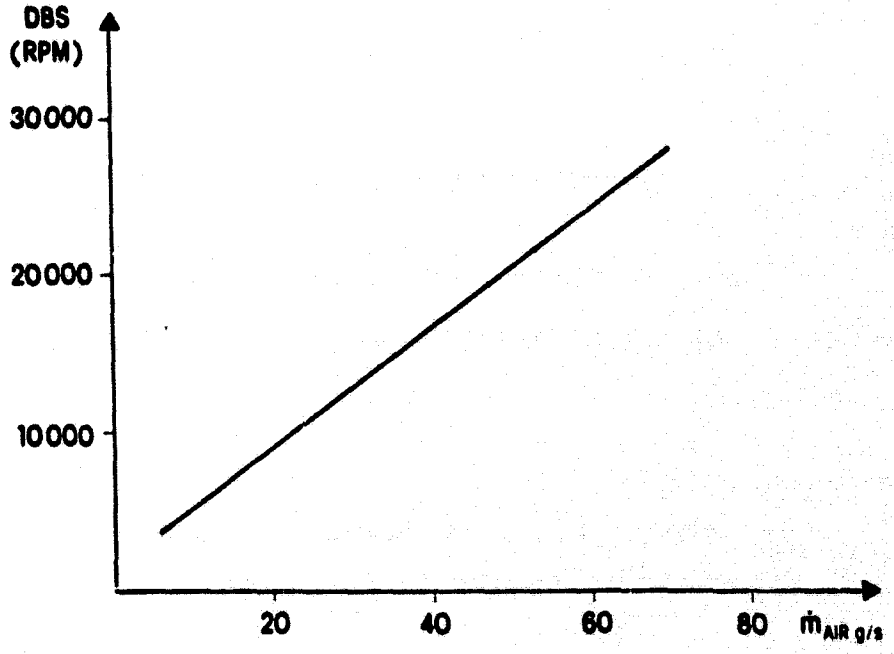


Fig 8.2:11 ASE MOD I Burner Blower Speed Control. Blower Characteristic Function.

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Optionally the blower speed control will contain an altitude correction. The signal from the air pressure transducer (a unit within the air/fuel system) will be converted within the altitude correction function to a factor, at sea level = 1, by which the desired blower speed should be multiplied (fig 8.2:12).

ASE MOD1 BURNER BLOWER SPEED CONTROL ALTITUDE CORRECTION

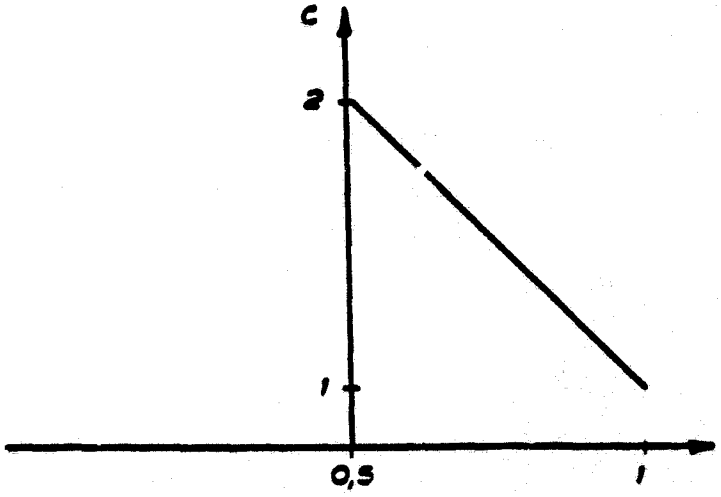


Fig 8.2:12 ASE MOD I Burner Blower Speed Control. Altitude Correction.

There is also an overspeed protection preventing the speed signal to exceed max allowable blower speed (8.2.13).

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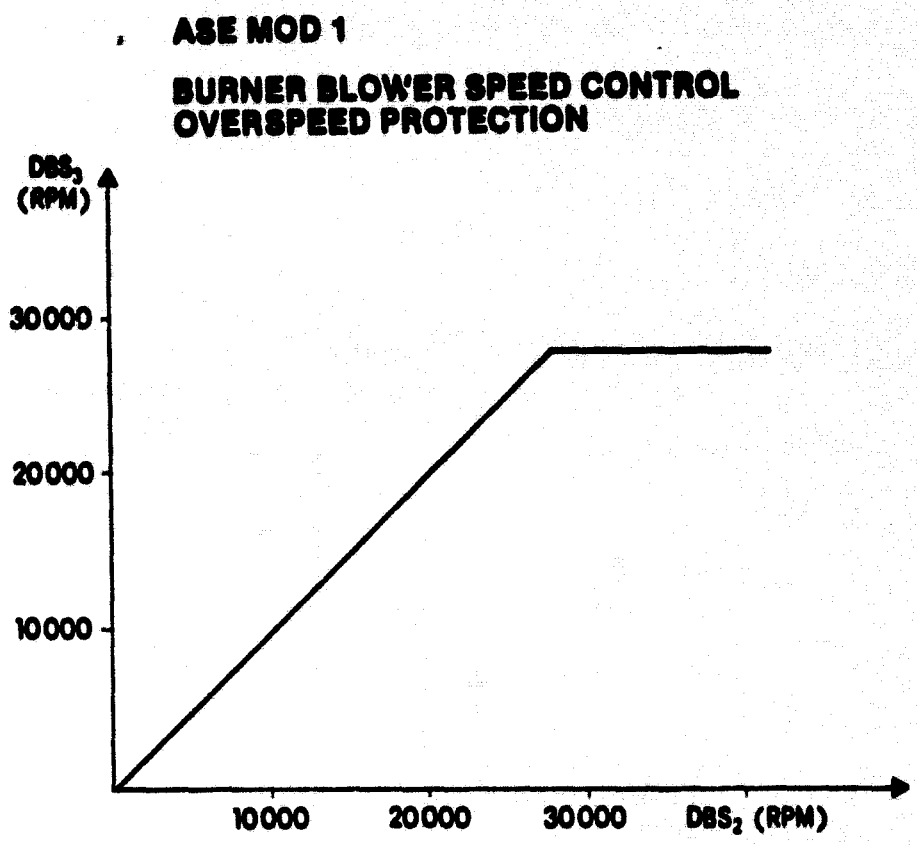


Fig 8.2:13 ASE MOD I Burner Blower Speed Control. Overspeed Protection.

The corrected desired blower speed (DBS_3) is compared to actual blower speed sensed by a speed transducer on the blower input shaft.

The difference enters a hydraulic valve control function where signals to the two hydraulic solenoid valves controlling variator ratio will be defined.

Fig 8.2:14 shows the valve signal logics.

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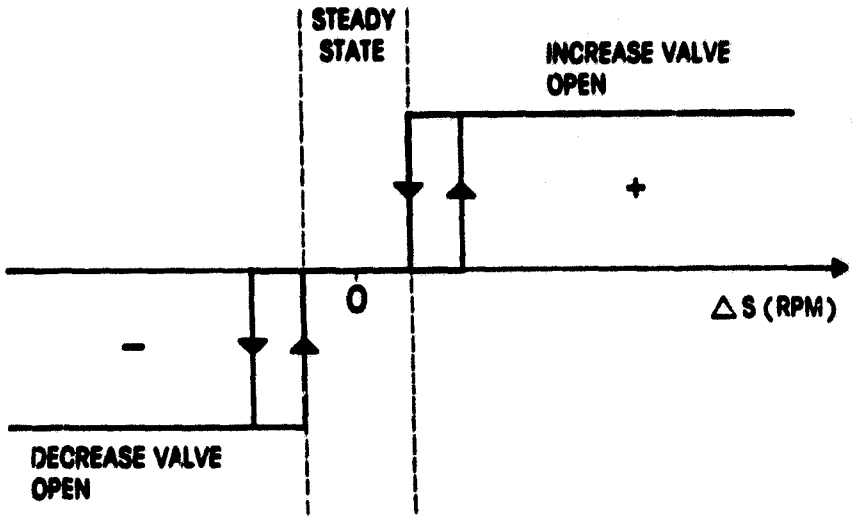


Fig 8.2:14 ASE MOD I Burner Blower Speed Control Hydraulic Valve Control Logics.

The function shows that the increase and the decrease valve will be opened depending on the value of the speed difference (Δs). Small Δs values are within a dead band where the speed control is at steady state.

The signals from the valve control function are thus controlling the two solenoid valves which in turn control the variator ratio (and the blower speed).

High Altitude Power Reduction:

At high altitudes the overspeed protection of the blower may cause lack of combustion air if the engine is running at high speeds and at high load conditions.

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Therefore, a limiter could be implemented to the burner blower speed control, lowering the max desired hydrogen pressure in the engine and thus reducing demand of air.

Fig 8.2:15 shows the control logic. A signal from the intake air pressure transducer is converted to a limiting function within the power control torque rise function. That limiter prohibits high hydrogen pressures at high speed and thus reduces the max amount of air needed for the combustion.

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ASE MOD 1

BURNER BLOWER SPEED CONTROL

HIGH ALTITUDE LIMITER

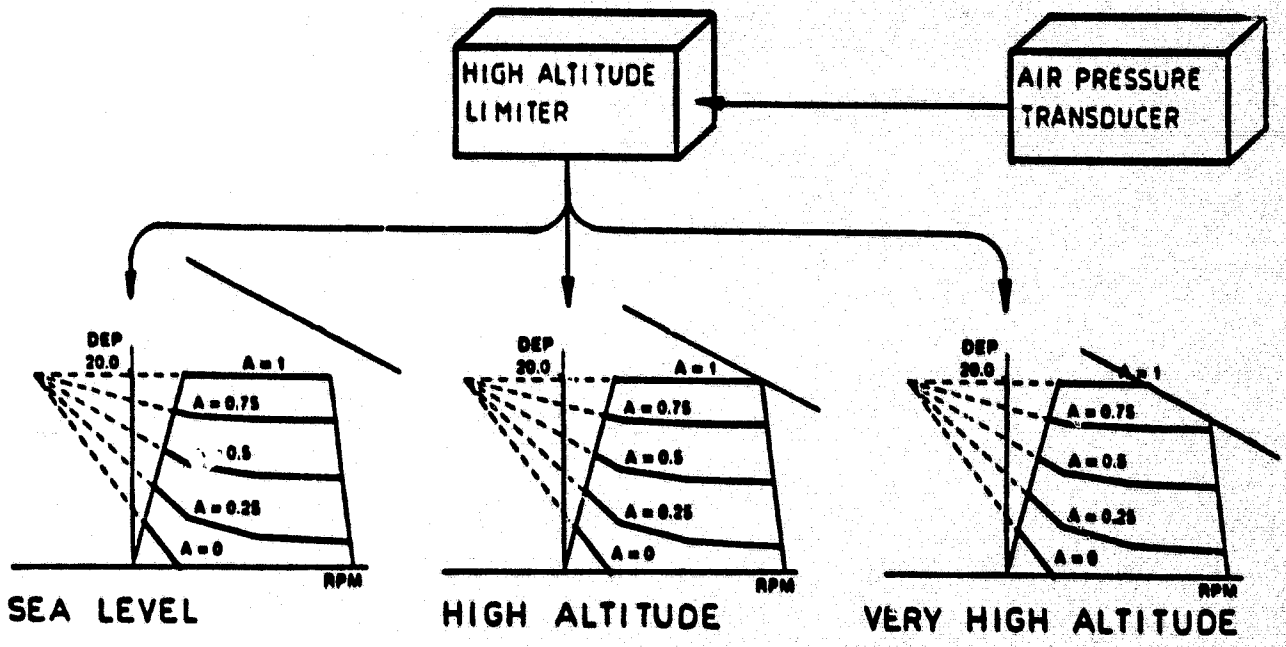


Fig. 8.2:15 High Altitude Limiter

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8.2.3.3 Temperature Control Characteristics

With varying demand of heat from the working cycle, depending on engine power output, the temperature control varies the amount of combustion air in a way that heater tube temperature will be kept constant.

It is very important in terms of engine power and engine efficiency that heater tube temperature is kept close to the max allowable material temperature. Major elements of the temperature control are themocouples, electronic regulator and air throttle.

Sensing elements for temperature are the thermocouples, located in thermocouple sleeves on the rear side of front and rear row of tubes. The thermocouples could also be located inside the heater tubes sensing working gas temperature.

The external thermocouple location causes the set temperature to be set higher than the desired tube temperature.

Temperature Control Specification:

The temperature control system shall maintain the maximum temperature to the set point temperature $\pm$ 7° C within $\pm$ 7° C at steady state.

The temperature control system shall maintain the maximum temperature within +20° C, -80° C during any system transient.

Temperature Control of ASE MOD I:

Thermocouples: ASE MOD I is equipped with 12 thermocouples for heater head temperature measurements.

Four are placed in sleeves on the rear side of the front row of heater tubes.

Four are placed in sleeves on the rear side of the rear row of heater tubes.

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Four are placed inside the rear tubes.

Control will be by the four externally located rear row thermocouples, but there will be a possibility to control the temperature also by the four internal thermocouples.

Type of Thermocouples:

The thermocouples are M08-1CAI16 (K-type) manufactured by Inor, Sweden. They are all insulated in a tube of Inconel 1.6 mm in diameter.

Air Throttle

The air throttle is described in section 8.3.7.

The Temperature Control Loop:

Note: All values, set points, breaking points etc. may be changed during the proceeding development work or the initial engine testing phase.

Fig 8.2:16 shows the temperature control scheme of ASE MOD I. (All symbols refer to fig 8.2:16.)

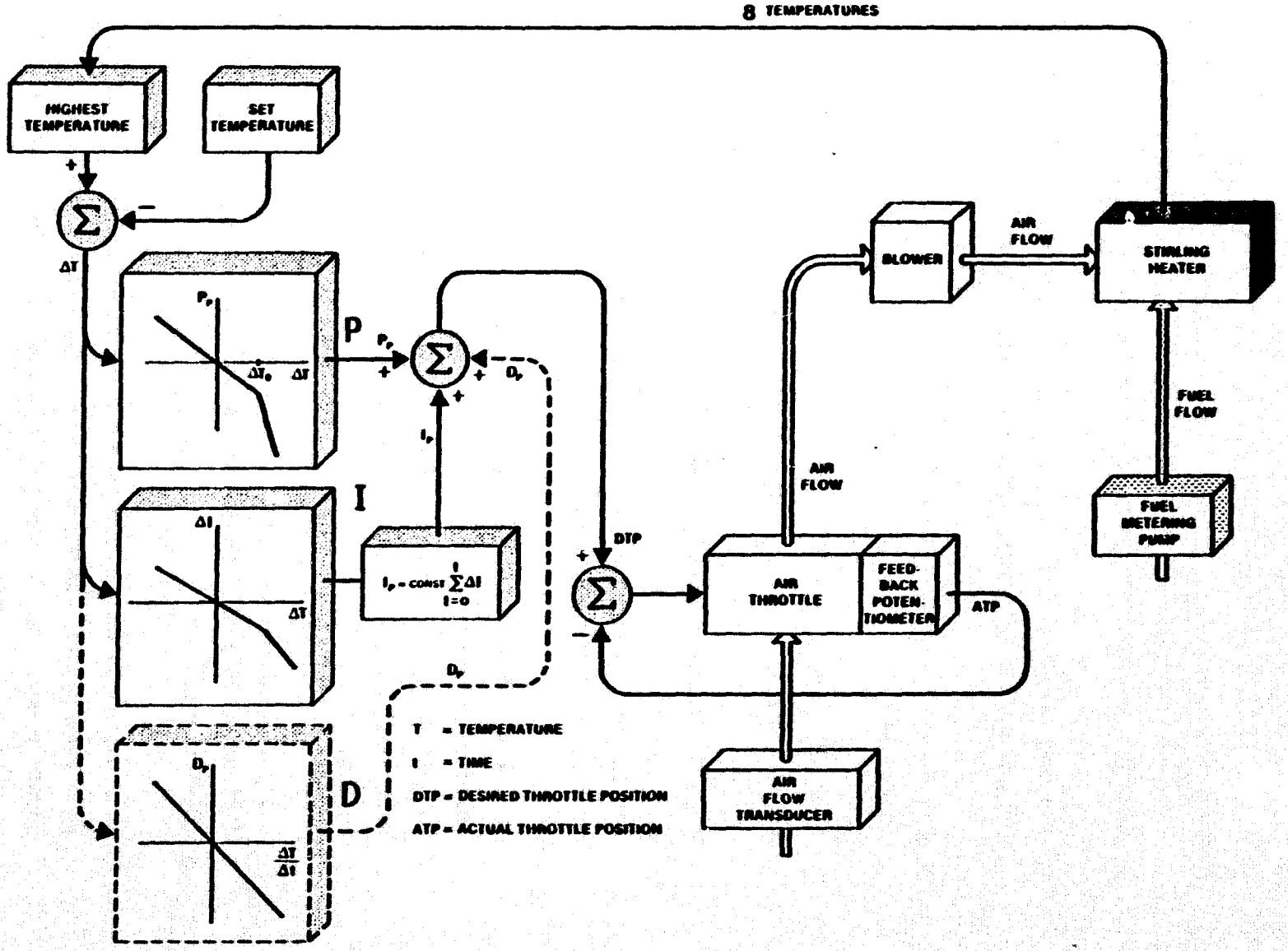


Fig. 8.2:16 ASE Mod I Temperature Control Scheme

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Four temperatures from heater head thermocouples center the electronic regulator. The average of the four back row temperatures will be compared to the set temperature.

The temperature difference ΔT enters a regulator with a proportional, an integral and a derivative part.

Fig 8.2:17 shows the proportional part of the temperature regulator.

The proportional amplification is sharply increased at a certain temperature to give a quick response to threatening overtemperatures.

Fig 8.2:18 shows the integral part of the temperature regulator.

The integral part also increases at a certain ΔT to keep transients in temperatures low.

To further refine the temperature regulator, a derivative part could be implemented to the control logics. This part would react to fast changes in heater temperatures. Fig 8.2:19 shows the derivative part which could be implemented as an option to the electronic control logics.

All parts, the proportional (P_p), the integral (I_p) and the derivative (D_p) will be summed to a desired throttle position (DTP) signal.

This signal is compared to the actual throttle position, from a feed back potentiometer, and if not equality is attained, the throttle moves to the new position and the right amount of air will be fed into the engine.

ASE MOD1 TEMPERATURE CONTROL.
PROPORTIONAL PART OF TEMP. REG.

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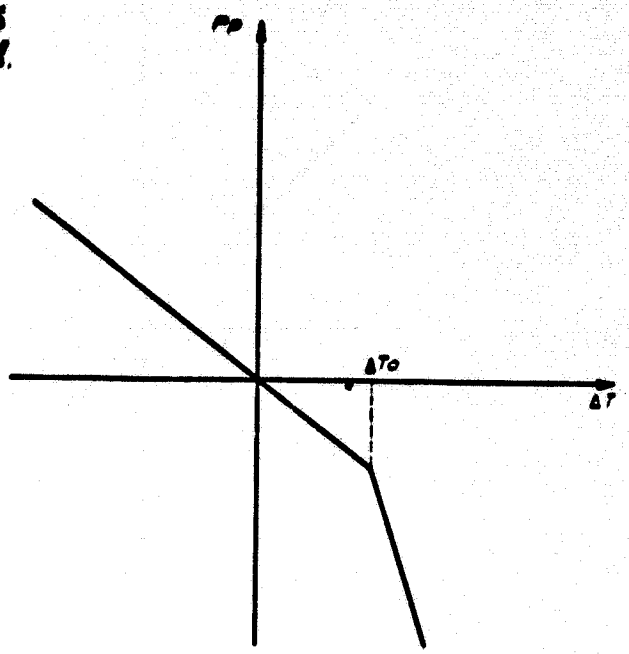
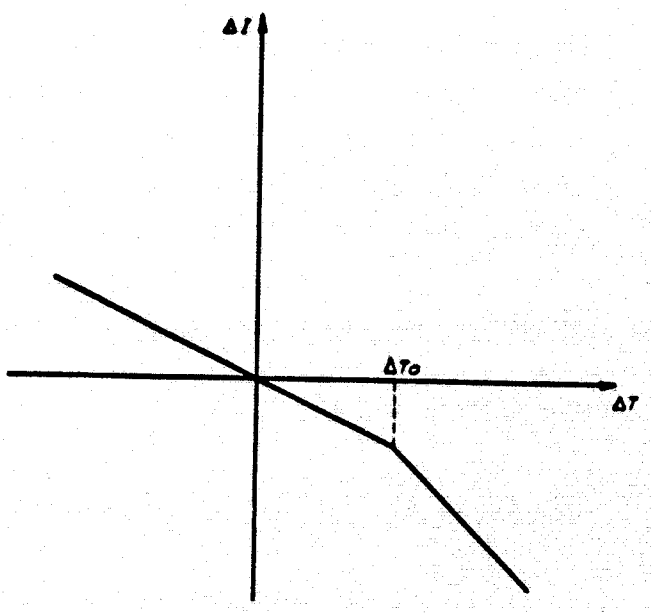


Fig 8.2:17 Proportional Part of the Temperature Regulator.

ASE MOD1 TEMPERATURE CONTROL.
INTEGRAL PART OF TEMP. REG.



$$I_P \cdot \text{CONST.} \sum_{I=0}^{I} \Delta I$$

Fig 8.2:18 Integral Part of the Temperature Regulator.

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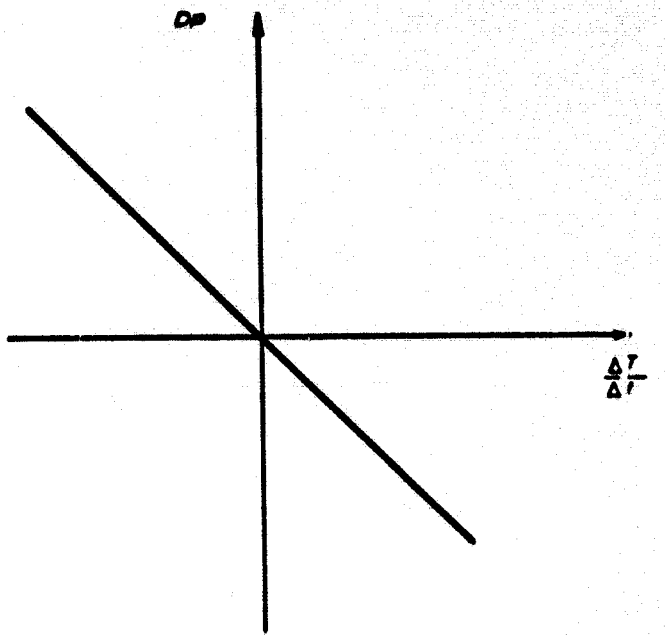


Fig 8.2:19 Derivative Part of the Temperature Regulator.

8.2.3.4 Air/Fuel Control Characteristics

The air/fuel ratio must be accurately controlled to meet the requirements for low exhaust gas emissions. The air/fuel system controls the engine fuel flow specified by the combustion air flow and the air excess (fig 8.3:8). Thus, the air fuel system is a slave system to the temperature control system.

ASE MOD I Air/Fuel Control Specification:

See section 8.3.13.

ASE MOD I Air/Fuel Control System:

ASE MOD I will be equipped with an air fuel system similar to the Autotronic system, described in section 8.3.13. Sensing and controlling elements of this

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system are, air flow transducer, air temperature sensor, air pressure transducer and a fuel metering pump.

Air Fuel Control Loop:

Note: All values, set points, breaking points etc. may be changed during the proceeding development work or the initial engine testing phase.

Fig 8.2:20 shows the air fuel control scheme of ASE MOD I.

(All symbols refer to fig 8.2:20.)

Signals from the air pressure transducer, the air temperature sensor and the flow transducer are entering a flow meter linearisation module, which linearizes and compensates the flow transducer signal for different air temperatures and pressures. Signal out from that module is proportional to air mass flow ($\dot{m}_{AIR}$).

A λ map (fig 8.2:21) converts the air mass flow signal to the desired fuel flow. The shape of this curve depends on the demand of excess air (see fig 8.3:8).

The demand of fuel flow is converted to a desired metering pump speed. Due to the design of the metering pump, this fuel pump characteristic is a linear curve (fig 8.2:22).

The desired pump speed (DPS) enters a fuel metering pump drive, which by sensing the actual pump speed via a speed transducer exactly controls the speed of the metering pump motor.

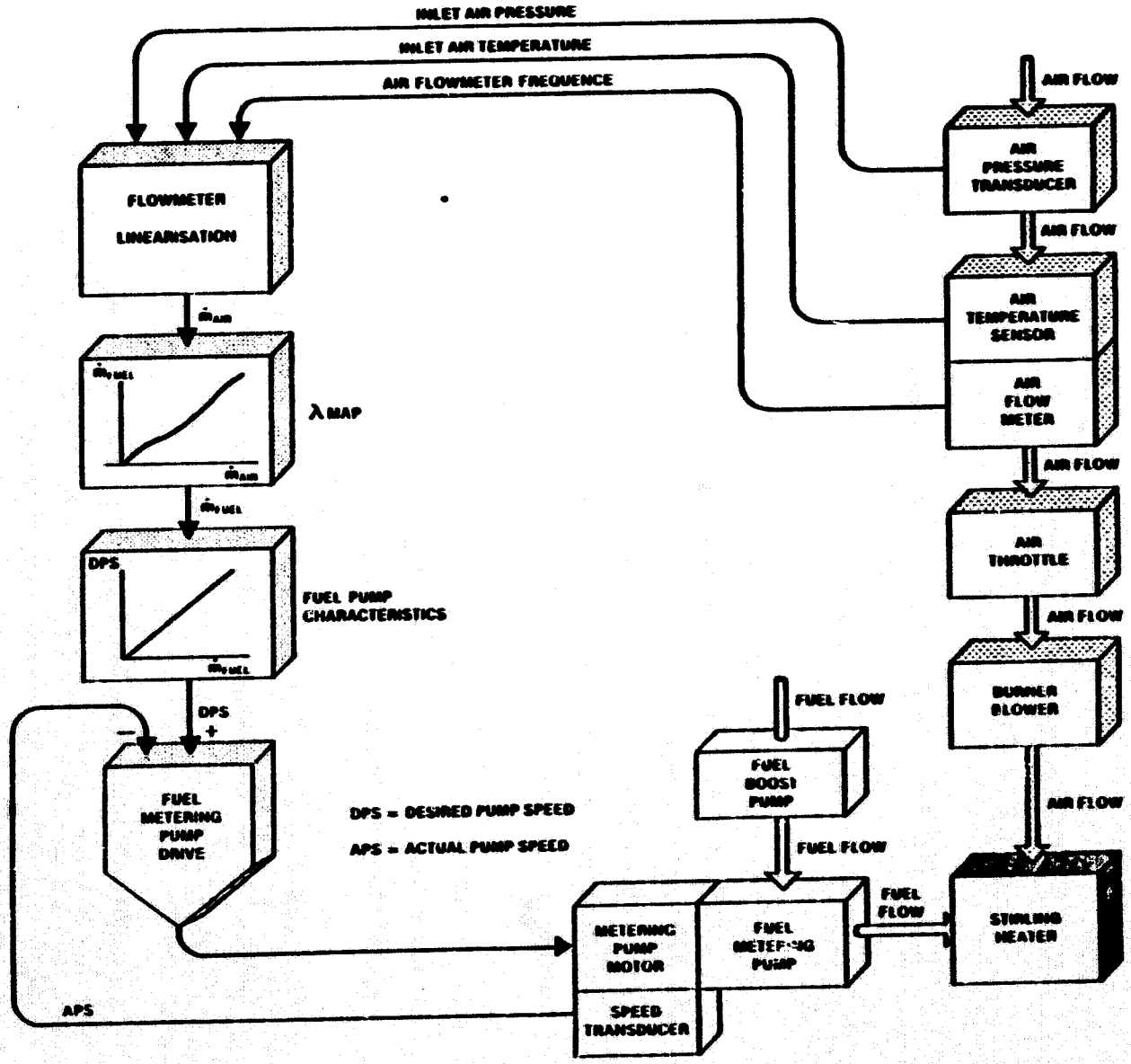


Fig. 8.2:20 ASE Mod I/Fuel Control Scheme

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ASE MOD1 AIR/FUEL CONTROL λ MAP

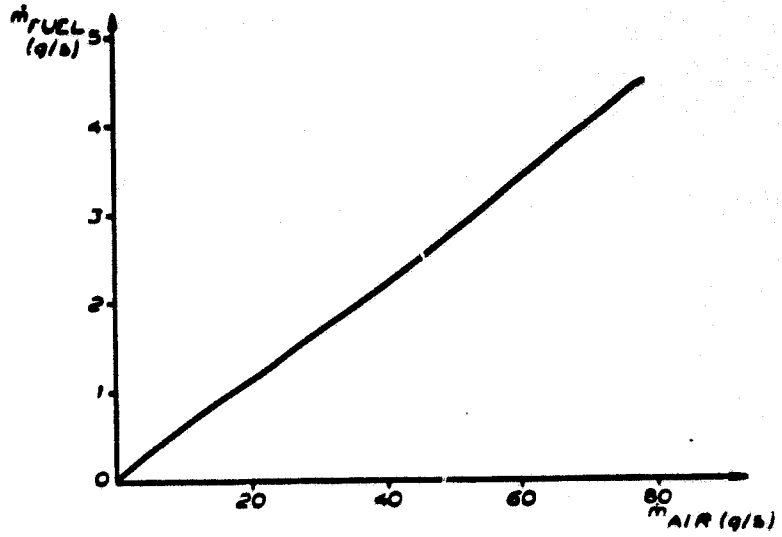


Fig 8.2:21 ASE MOD I Air Fuel Control System. λ Map.

ASE MOD1 AIR/FUEL CONTROL
FUEL PUMP CHARACTERISTICS

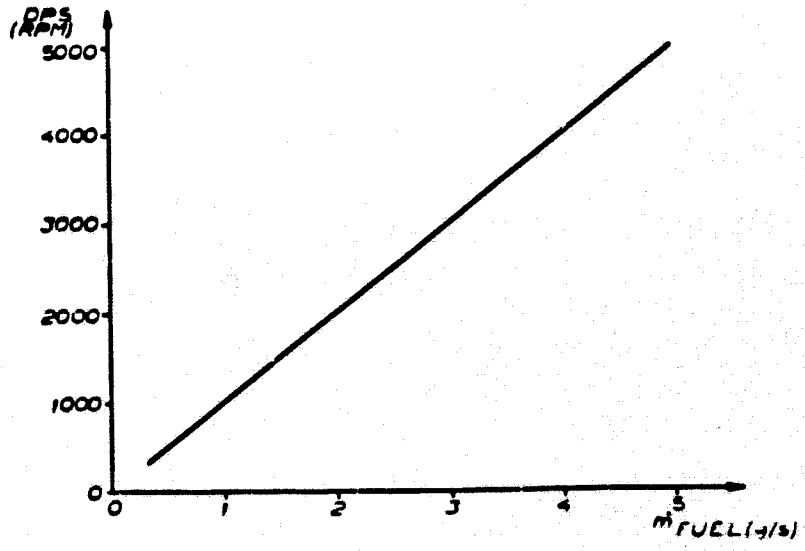


Fig 8.2:22 ASE MOD I Fuel Pump Characteristics.

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8.2.4 Start/Stop Sequence

8.2.4.1 General Description

In contrast to an internal combustion engine a Stirling engine cannot be self-sustained and deliver shaft output until the heater tubes have been heated up by the external combustion. An automatic starting sequence provides all the logical and controlling functions during the start of a Stirling engine.

The external combustion principle and the amount of heat stored within the heater keeps the engine running after fuel flow has been stopped. Therefore, a stop sequence controls the engine parameters and keeps the electronics activated until the engine has stopped and the risk of cooling water overtemperature is eliminated.

8.2.4.2 Starting Sequence

Approximative time schedule

Fig 8.2:23 shows the automatic starting sequence of ASE MOD I.

0 sec

Manual Operations: The driver turns the key to pos 1 and the electronics will be activated. The driver turns the key to pos 2 and the automatic starting sequence will be initiated. At release the key will return to pos 1.

The driver can cut off the starting sequence at any time by turning the key back to pos 0.

Automatic Operations: When the automatic starting sequence is initiated, the electric blower motor is activated. The blower motor drives the burner blower, atomizer compressor and servo oil pump. The blower feeding the engine with combustion air will be running at approximately 19 000 RPM.

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The fuel boost pump and the fuel metering pump starts, but the fuel shut off valve has not opened the fuel flow to the combustor.

The air throttle is moved to a start position being about 60% open.

The power control system will supply the engine with gas until it reaches the starting pressure of 6 MPa and the control valve then is blocked in neutral position (during the starting sequence the driver cannot control the hydrogen pressure by using the accelerator).

3 sec — A 3-second-period of venting the combustion chamber and preheater will follow, preventing accidental combustion of any remaining vaporized fuel or any leaking hydrogen.

After 3 seconds, the ignition starts and the fuel shut off valve opens fuel flow to the combustor and combustion starts.

8 sec — The air throttle continuously opens and is 100% open at a heater temperature of 200° C. At this temperature the risk of flame out is eliminated.

15 sec — At a tube temperature of 600° C the starter is engaged.

When the engine is running the blower variator is set to max speed increasing ratio.

17 sec — The starter will be disengaged at about 500 RPM and the Stirling engine will be self-sustained.

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22 sec
(Possible drive away point)

The engine accelerates with the preset starting pressure and at 1 000 RPM the power control is partly activated. Engine control is possible within the starting pressure limits. The blower motor is disengaged and the blower will be driven by the engine. Idle speed will be in a fast mode (1 000 RPM).

25 sec
(Recommended drive away point)

At a rear-row tube temperature of 600° C the power control is free from any starting sequence restrictions and this is the recommended drive away point.

The blower speed control will be activated when the air throttle leaves the opening degree of 100%. Idling speed continuously decreases from 1 000 RPM to a normal 700 RPM at an exhaust gas temperature of 100° C (700 RPM = idle speed when gearshift is in neutral position).

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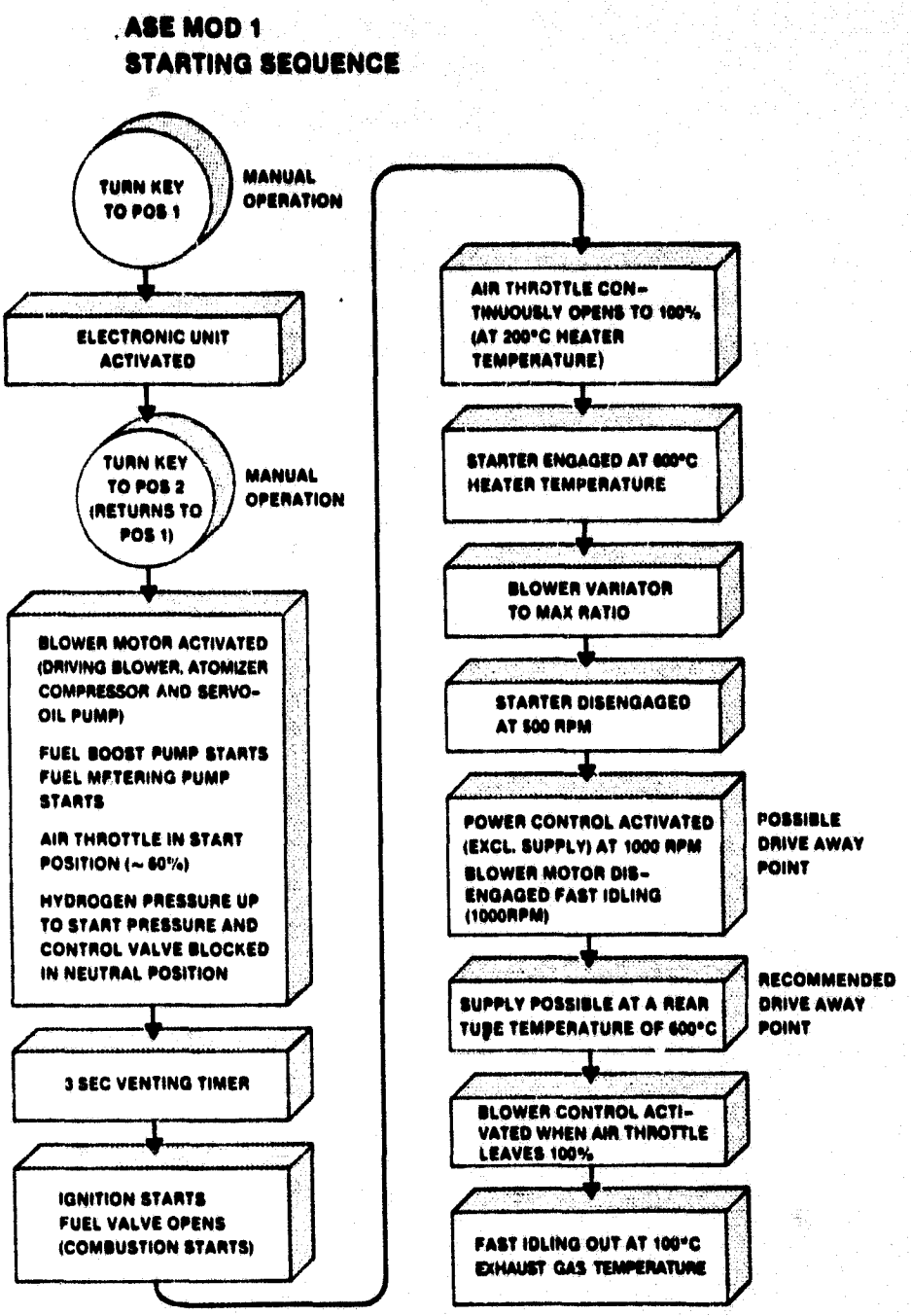


Fig 8.2:23 ASE MOD I Starting Sequence.

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8.2.4.3 Stopping Sequence

Fig 8.2:24 shows the automatic stopping sequence of ASE MOD I.

Manual operation: The driver turns the key to position 0.

Automatic operation: When the automatic stop sequence is initiated the fuel valve shuts off fuel flow to the combustor. The fuel boost pump and fuel metering pump stop. Ignition stops.

The power control system dumps hydrogen to idle pressure independent of accelerator position.

The engine is running on the remaining heat and stops when temperature and speed are too low to sustain the Stirling cycle.

A temperature sensor guards the engine against cooling water over-temperature when the engine has stopped. If the temperature goes up, an after-cooling pump is started. A timer keeps the electronics activated until the after-cooling pump has been out of use for 10 minutes. Then the electronics are switched off.

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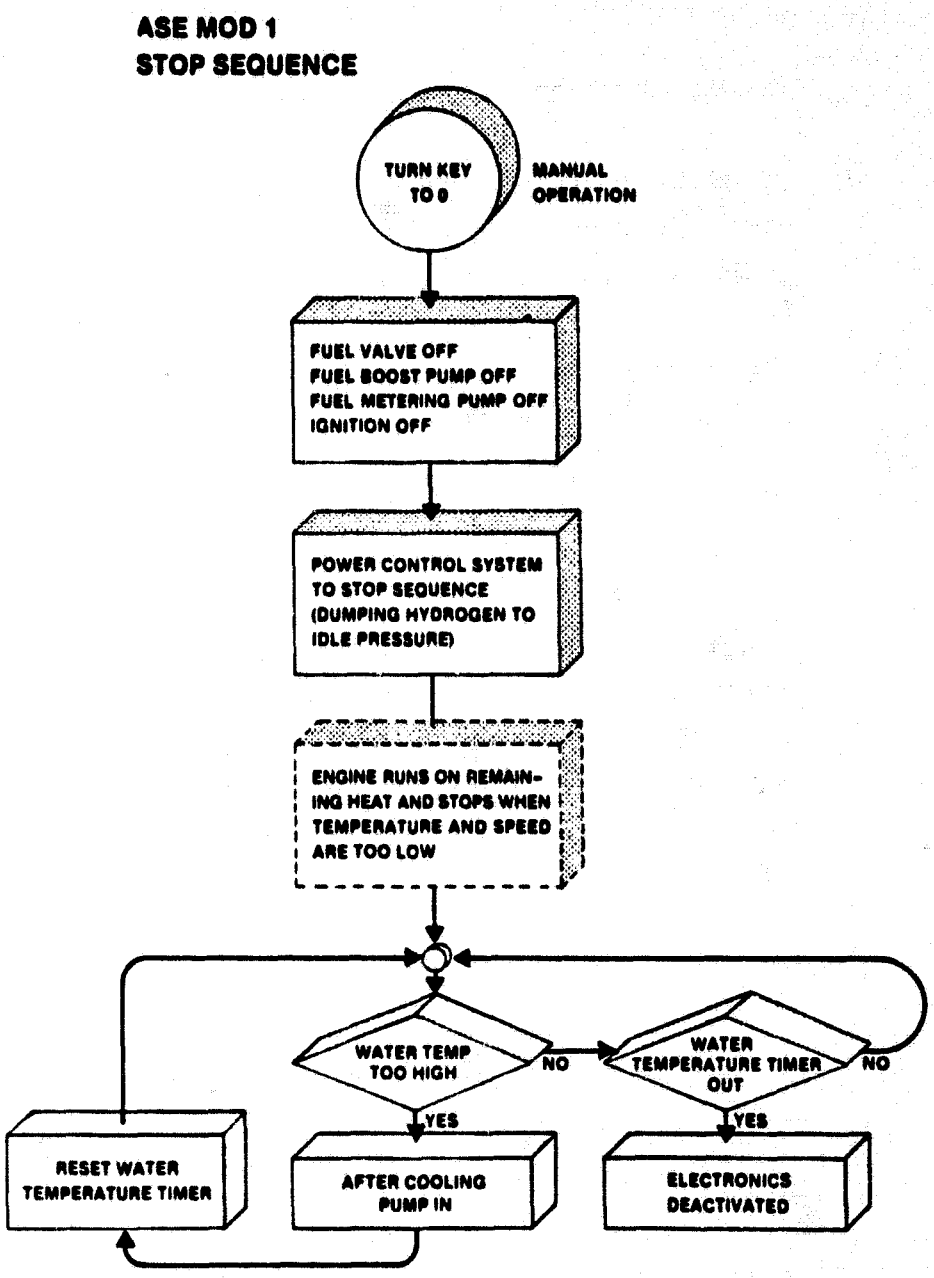


Fig 8.2:24 ASE MOD I Stopping Sequence.

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8.2.5 Safety Systems and Guards

A guard system consisting of electronic logical functions and safety devices guards the engine against malfunctions of the control systems and improper driver operations.

The guards sense vital engine parameters and stop these from exceeding their limits.

Below is a listing of the malfunctions and appropriate guard actions for different control systems.

Guards for Starting Sequence:

Failure	Guard/Action
Hydrogen is leaking into the combustion system or unburned vaporized fuel remains in the preheater.	A 5-second-timer vents the combustion system with air before fuel valve opens and ignition starts.
The engine does not ignite.	After a few seconds a positive temperature derivate shall be obtained and maintained to starter engagement. If not, fuel off.
The starting pressure will be too high caused by an improper function of the control valve (starting torque too high).	The starting sequence will be cancelled.
The starting pressure will be too low during the heating up period due to me or external leakage.	The starting sequence will be cancelled.

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Failure	Guard/Action
Starter will not disengage due to low engine speed.	Starter timer will disconnect starter. Starting sequence cancelled and fuel valve closed.
The engine stops during the starting sequence after the starter is disengaged.	The starter will be engaged again.
The blower motor will not disengage.	Blower motor timer will disengage blower motor.

Guards for Power Control System:

Failure	Guard/Action
Storage tank pressure too high.	Safety valve (22.5 MPa) vents hydrogen to atmosphere.
Storage tank pressure too high with manual shut-off valve closed.	Bursting disc uncharges the storage tank.
Temperatures at storage tank too high.	Fusible temperature plug uncharges the storage tank.
Electrical connection to control valve actuator broken.	Power control valve to dump position (mechanical bias).
Compressor pumping against pressure which is too high.	Safety valve 22.5 MPa vents hydrogen to atmosphere.
Compressor short-circuiting valve not working.	Normal engine control, though at high pressure, possible.

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Failure	Guard/Action
Pressure transducer failure.	Engine shut down to idle.
Pressure within the engine too high.	Safety valve vents hydrogen to atmosphere.
Speed transducer failure.	Engine to lowest pressure. Fuel off.
Accelerator potentiometer failure.	Engine shut down to idle.
Engine speed too high.	Fuel off.
Engine out of operation due to guard failure or other cause.	Manual override of power control actuator.
	External dumping of hydrogen via the external dumping solenoid valve. Valve operation separated from other electronics.
	Fuel shut off by manual operation. Operation separated from other electronics.

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Guards for Temperature Control:

Failure	Guard/Action
Overtemperature in heater.	Fuel off.
Broken thermocouple.	One thermocouple broken: Lower temperature set. More than one: Fuel off.
Thermocouple short-circuited.	One Thermocouple broken: Lower temperature set. More than one: Fuel off.
Overtemperature in preheater.	Fuel off.
Water temperature too high.	Fuel off.
Oil temperature too high.	Fuel off.

Guards for Electronics:

Failure	Guard/Action
Input value incorrect due to noise or transmission failure.	Logical check of reasonable values. If not, engine shut-down.
Microprocessor runs out of control.	Electronic watchdog restarts processor within 50 ms.

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8.3 AUXILIARIES AND AIR-FUEL SYSTEM

8.3.1 GENERAL DESCRIPTION

The auxiliaries on a Stirling engine are in many cases identical to the ones used on an IC-engine. A few are used in the same manner, but are modified or specially developed, and some are unique for the Stirling engine. Listed below are the auxiliaries and some accessories with a note about their individuality.

Starter motor	- Same as IC-engine
Alternator	- Same as IC-engine
Lubrication oil pump	- Same as IC-engine
Water pump	- Same function as on IC-engine, but specially developed with higher capacity.
Combustion air blower	- Unique for Stirling Engines
Air throttle	- Same function as on IC-engine, but specially developed.
Variable ratio belt drive	- Not used on IC-engines today
Atomizer air compressor	- Unique for Stirling engines
Servo oil pump	- Unique for Stirling engines
Electric blower motor	- Unique for Stirling engines
Air-fuel system	- Same function as on IC-engines, but specially developed
Hydrogen compressor	- Unique for Stirling engines
After cooling pump	- Unique for Stirling engines
Air-conditioning compressor	- Same as IC-engines
Power steering pump	- Same as IC-engines

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Fig. 8.3:1 is a front view of the MOD I engine with the auxiliaries and some accessories. Later on these are described in detail.

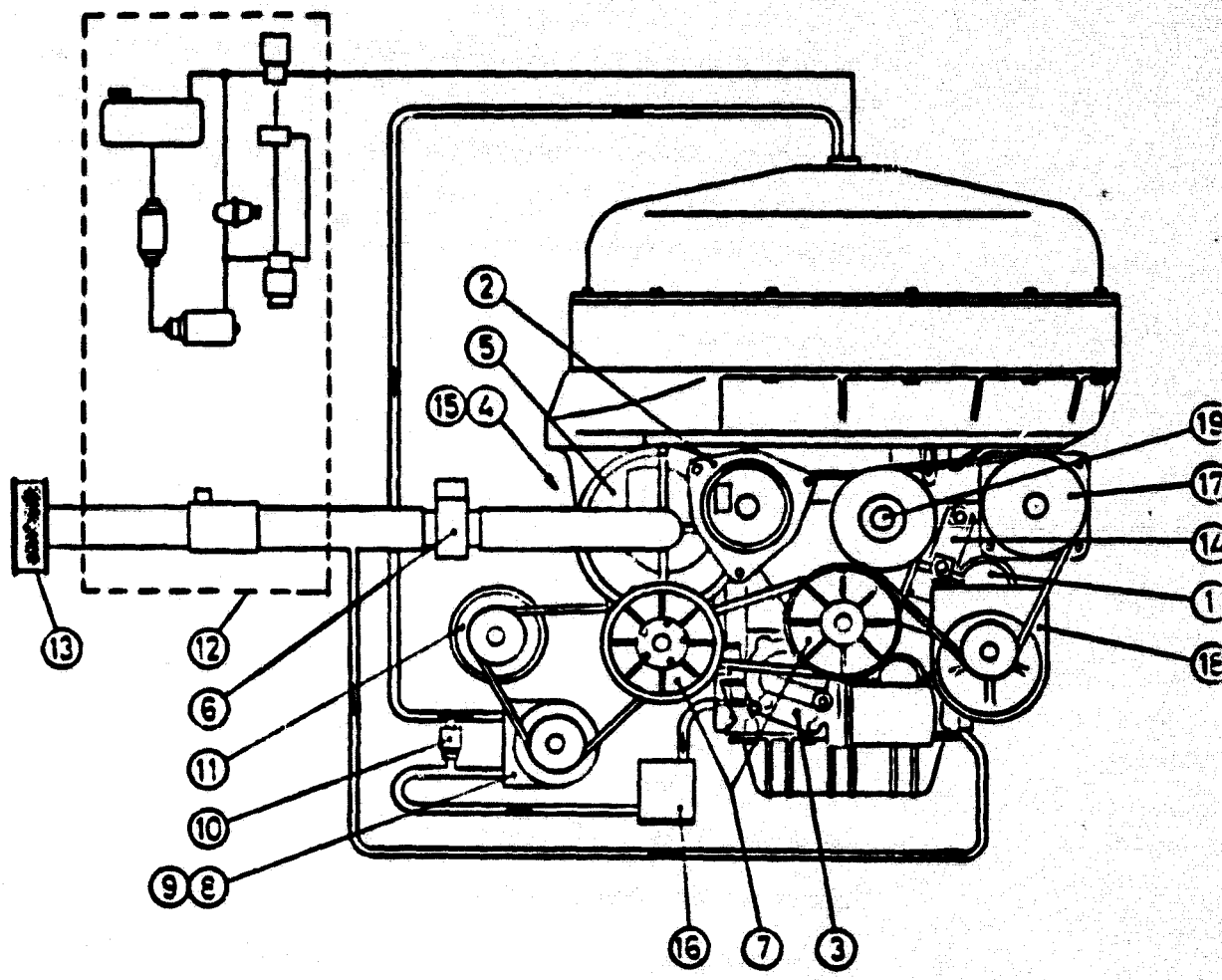


Fig. 8.3:1 Front View of a Mod I Engine

1. Starter motor
2. Alternator
3. Lubrication oil pump
4. Water pump
5. Combustion air blower
6. Air throttle
7. Variable ratio belt drive
8. Atomizer air compressor
9. Servo oil pump
10. Pressure relief valve
11. Electric blower motor
12. Air-fuel system
13. Air filter
14. Hydrogen compressor
15. After cooling pump
16. Crank case ventilation system
17. Air-conditioning compressor
18. Power steering pump
19. Radiator fan shaft

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In order to get a comparison of an improved up-to-date P-40 engine with quiet, efficient and reliable auxiliaries, the following auxiliaries need special attention with separate development programs:

Starter motor

The existing starter motor, which fits to the specified automatic transmission, has too low power output to start the Mod I engine at low temperatures. USS specification for the starter will be found in chapter 8.3.2, and the efforts to develop this motor was taken care of by MTI.

Combustion air blower

Previous blowers have been of the regenerative type, which are heavy, bulky and inefficient. Besides this a special by-pass air throttle was necessary to reduce the blower power consumption at low engine loads. The goal for the ASE MOD I blower development program is to increase the peak efficiency from present 57% to 70%, and reduce the weight to about half of the regenerative blower and maintain the low noise level. The size has to be reduced to facilitate the installation problems. This is accomplished by a high speed centrifugal blower with a flat belt drive as speed increaser. MTI is developing the impeller and volute housing, and USS the flat belt drive.

Variable ratio belt drive

The present blower variator on the P40 engine is only speed governed. If a load governed variator is used, the blower speed can be varied independently of the engine speed. This method will save a lot of blower power especially at low loads and high engine speeds. The developed variator is a hydraulically actuated load governed unit, which is using the same hydraulic system for position control, as the power control valve.

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Atomizer air compressor and servo oil pump for power control

On the P40 engine these two are separate units. The air compressor is a heavy cast iron device, made for industrial purposes, and the oil pump has too high pressure capacity. On the ASE MOD I the oil pump will be integrated with the compressor and driven from the same shaft. The main objectives are to save space and weight and to simplify the auxiliary arrangement.

Air throttle

As mentioned earlier, the regenerative blower has a complicated and bulky throttle arrangement. On a centrifugal blower, the air flow can be controlled by inlet throttling, using an ordinary butterfly valve, Such a throttle, developed by USSw outside the ASE program, will be used on Mod I engine.

Air-fuel control system

Previous engines have used the Bosch K-jetronic mechanical fuel injection system as air-fuel control device. This system is used on fuel injected IC-engines. The disadvantages with this system are: high pressure drop, especially at low air flows, difficulties in changing the air-fuel ratio, difficulties with altitude and pressure corrections, low multi-fuel capability and expected delivery difficulties from Bosch in the future. The alternative is a system with a separate air flow transducer with temperature and pressure corrections, and a fuel metering pump with equalizer valve as fuel distributor. This system will give a lower air pressure drop with lower blower power consumption as a result, and a simple way of changing air-fuel ratio. The multi-fuel capability is also good. This later system is the primary choice for the MOD I engine.

After-cooling pump

Outside the ASE-program a new after-cooling pump has been developed, as the P40 pump was too noisy and had too low capacity. This pump will be used on the MOD I engine.

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Water pump

The waterpump is very similar to the pump used on the USS 4-275 engine with for example the same impeller. The pump is designed by Ricardo Consulting Eng., and no special development program has been established. The capacity and efficiency is supposed to be about the same as for the 4-275 pump.

The rest of the auxiliaries are either normal equipment for IC-engines, or developed earlier outside the ASE-program.

Based on Fig. 8.3:1 the following section is a closer description of auxiliaries with their application, specification and available drawings.

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8.3.2 Starter motor

Application: The starter motor is used during the starting up period to crank the engine when the heater temperature has reached a certain level.

Specification: The necessary power output of the starter motor for a Stirling engine is dependent of several factors, for example: engine and ambient temperature, oil viscosity, condition of seals and piston rings, mean hydrogen pressure, battery capacity and rate of charging.

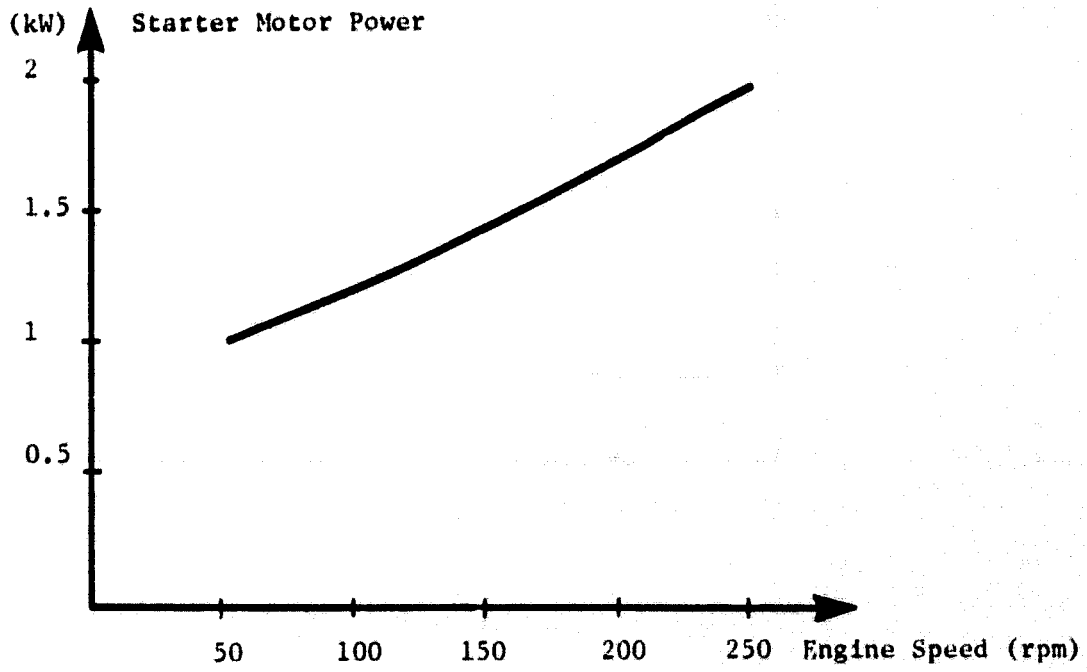


Fig. 8.3:2 Mod I Engine Starting Power Requirement

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A rough estimation of the power requirement for the Mod I engine as a function of engine speed can be seen in Fig 8.3:2. The oil temperature is 20°C and oil viscosity 320 cSt. This estimate is based on comparisons with and simple scaling from measured motoring power requirements of other USSw engine designs. It is further valid for a starting up hydrogren pressure of 5 MPa.

A Stirling engine in good condition will start at an engine speed of about 100 rpm. With the somewhat irregular performance todays piston rings and the necessity of having a margin, it is more advisable to assume a speed of 200 rpm.

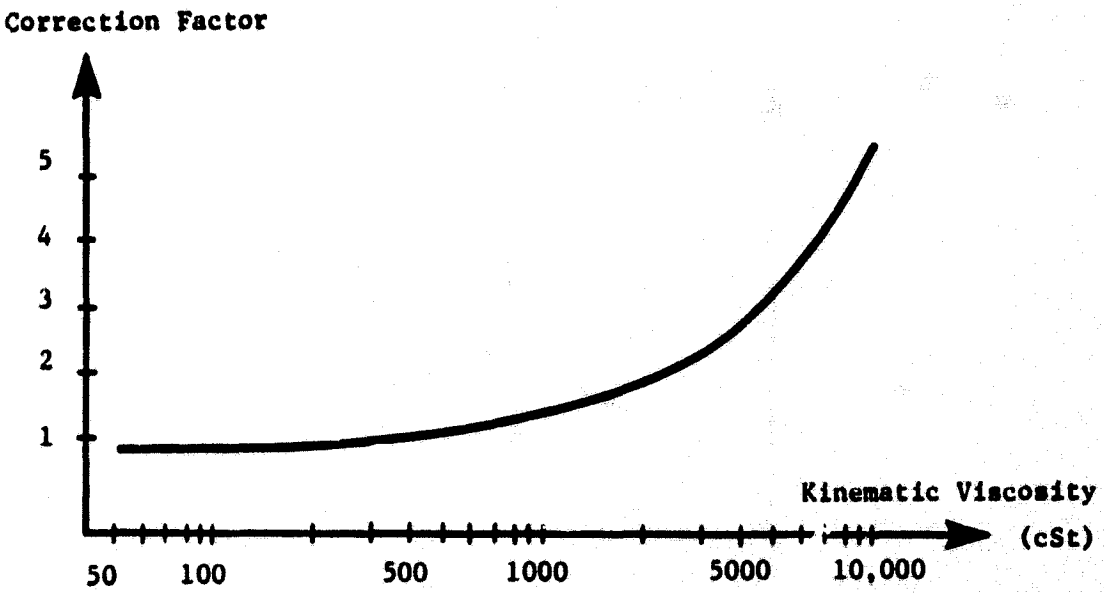


Fig. 8.3:3 Oil Viscosity Influence on Starting Power

The influence of oil viscosity on the power requirement can be seen in Fig. 8.3:3. To get a low value, it might be necessary to change oil at lower temperatures. The correction curve for different oil viscosities is based on tests with Stirling engines mostly in the temperature range 20-60°C, but also at lower temperatures.

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Special attention is necessary when choosing the gear ratio between the starter motor and engine shaft, because the torque must be sufficient to overcome the initial engine friction, and the peak power output from the motor should be utilized when starting the engine at lowest possible temperature.

Description: The starter motor has to fit to a specified automatic transmission case, which requires a starter with integral, offset gearbox and a defined mounting interface. The most powerful motor, which meets these requirements is a 1.8 Hp unit. This motor has to be used on the MCD engine, which will limit the ambient conditions when the engine can be started.

A performance map of the 1.8 hp motor is presented on page II-90.

8.3.3 Alternator

Application: The alternator is used in the same manner as on an IC-engine, although the continuous electric power requirement is somewhat higher.

Specification: The contionuous electric power demands are:

Electronics	24 W
Ignition	15 "
Fuel pumps	9 "
Solenoid valves	30 "
Fuel valve	6 "
Air throttle	6 "
Total	90 W

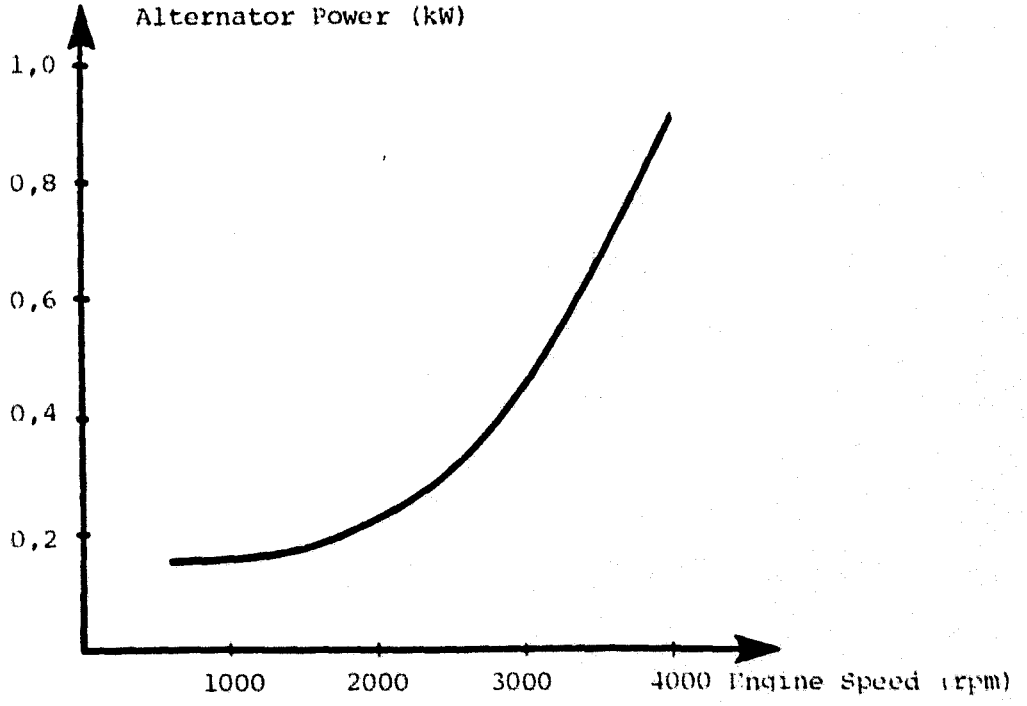
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During the starting period, both the starter motor and electric blower motor needs a lot of electric power for a short time. The energy drain from the battery for an average start will be about 1.4 Ah (Ampere hours).

Installed in a car there will also be an intermittent electric power demands from head lights, windshield wipers, compartment blower, and so on.

Calculations made by ANG, indicates that a 55 Amp. alternator with a ratio of 2.55 between engine shaft and alternator, is sufficient. This calculation is made in the same manner as when the alternator is selected for an IC-engine. With this ratio the alternator speed at 4 000 engine rpm will be 10 200 rpm, which is well within the Bosch limiting speed of 12 000 rpm.

Description: The alternator is a Bosch 14V/55 Amp unit, driven by a Poly V-belt from the main engine shaft via the radiator fan shaft. The continuous alternator power demand will be:



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This is taken from page II-91 showing the power demands at different loads. On page II-92 a USSw drawing of the alternator is illustrated.. Page II-93 contains a graph of current-speed characteristics for the alternator

8.3.4 Lubrication oil pump

Application: The pump is used for pressurizing the lubrication oil for the main shaft and crank shaft bearings. This oil is also fed to the piston rods just below the rod seals to get rid of the generated heat from the seals, and to oil jet nozzles in the syncronizing gear train.
Page II-141 shows a scheme of the lubrication oil system

Specification and description:

Chapter 7.1.7 in the Basic Engine Design Assessment includes a specification and description of both the oil pump and oil filter.

The power demand of the lubrication oil pump is included in the friction losses of the complete engine. The friction losses are based on tests with other Stirling engines. Page II-94 shows an overall view of the oil pump and filter, and a picture of the unit is shown on page II-95.

8.3.5 Water pump

Application: The water pump is used to circulate the cooling water in the main water-circuit. To cool the hydrogen compressor and the compressed hydrogen after the compressor, a parallel connection to the main circuit is used.
Page II-96 shows a scheme of the water circuit.

Specification: The design water flow at full engine speed and load is 4.23 kg/s.

The most suitable pump speed has to be tested out to ensure sufficient cooling, at all engine speeds and loads.

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The maximum possible gear ratio between pump and engine shaft is 0.916:1, which means a water pump speed of 3 664 rpm at 4 000 engine rpm. This is chosen for the first engine builds. As option a lower speed (3 137 rpm) can be obtained.

Fig. 8.3:5 is the calculated pressure drop for the Stirling coolers, cylinder block and pipes and fittings. Together with the water pump flow at 3 664 rpm, the maximum allowable pressure drop in the radiator is 75 KPa, as can be seen in Fig 8.3:5

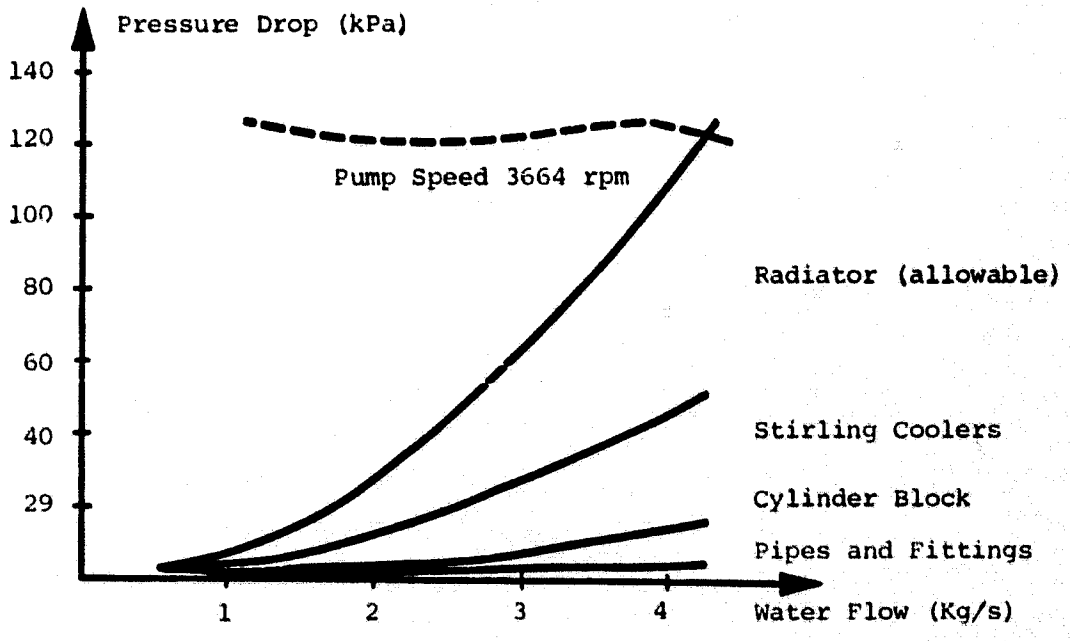


Fig. 8.3:5 Cooling System Pressure Drop

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Description: The pump is a centrifugal pump, gear driven from one of the crank shafts via an idler gear. The impeller is bought as a spare part to a submergeable waterpump (Grundfos SP 25), and is welded together from stamped stainless steel parts. The gear drive, bearings, volute housing and shaft seal arrangement is designed by Ricardo Consulting Eng. The efficiency curves in Fig. 8.3:6 are taken from a test with a 4-275 water pump, which has the same impeller. The differences in efficiency between the pumps are expected to be very small.

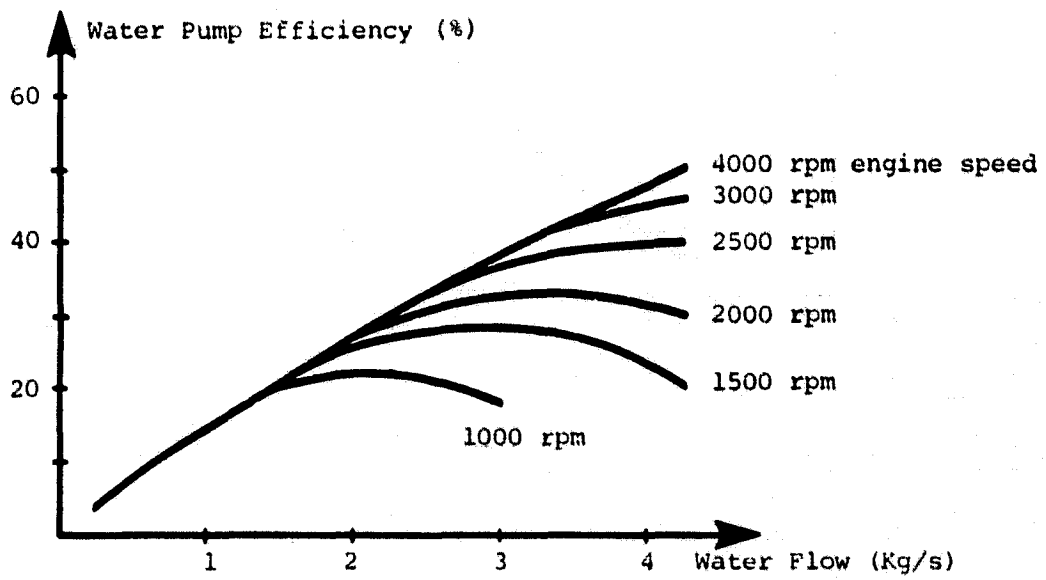


Fig. 8.3:6 Water Pump Efficiency Characteristics

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The water pump power demand can be seen in Figure 8.3:7.

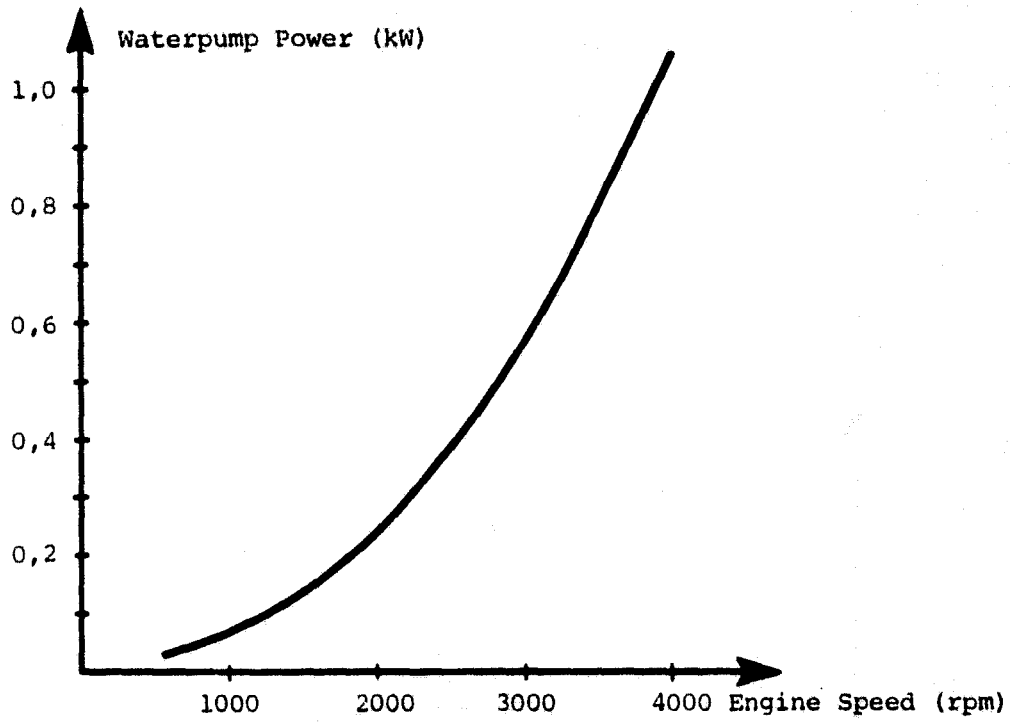


Fig. 8.3:7 Water Pump Power Requirements

A drawing of the water pump is displayed on page II-97.

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8.3.6 Combustion Air Blower

Application: The blower is supplying the combustor with the necessary air for the combustion process. During the first three seconds of the starting up period, it is also used for venting the combustion system.

Specification: The specified fuel flow range is 0.3 - 4.5 g/s. With a specified maximum external leakage of 5% and an air excess curve according to Figure 8.3:8, the blower air flow range is 5.3 - 78.8 g/s.

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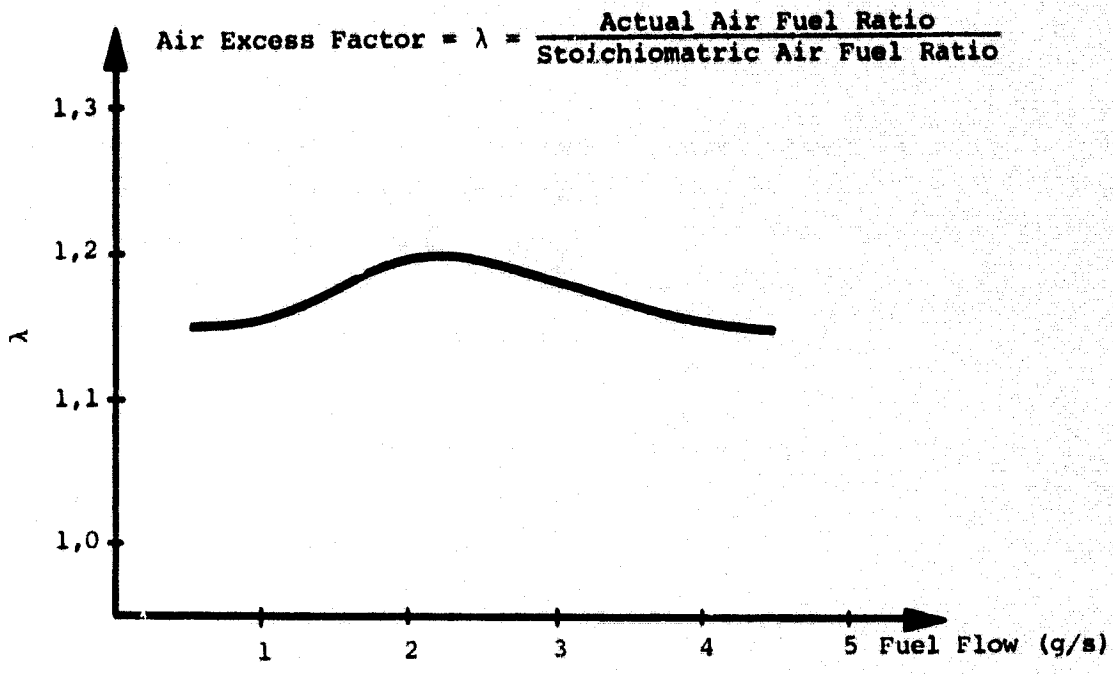


Fig. 8.3:8 Combustor Air/Fuel Characteristics

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The different pressure drops in the system can be seen in Fig. 8.3:9.

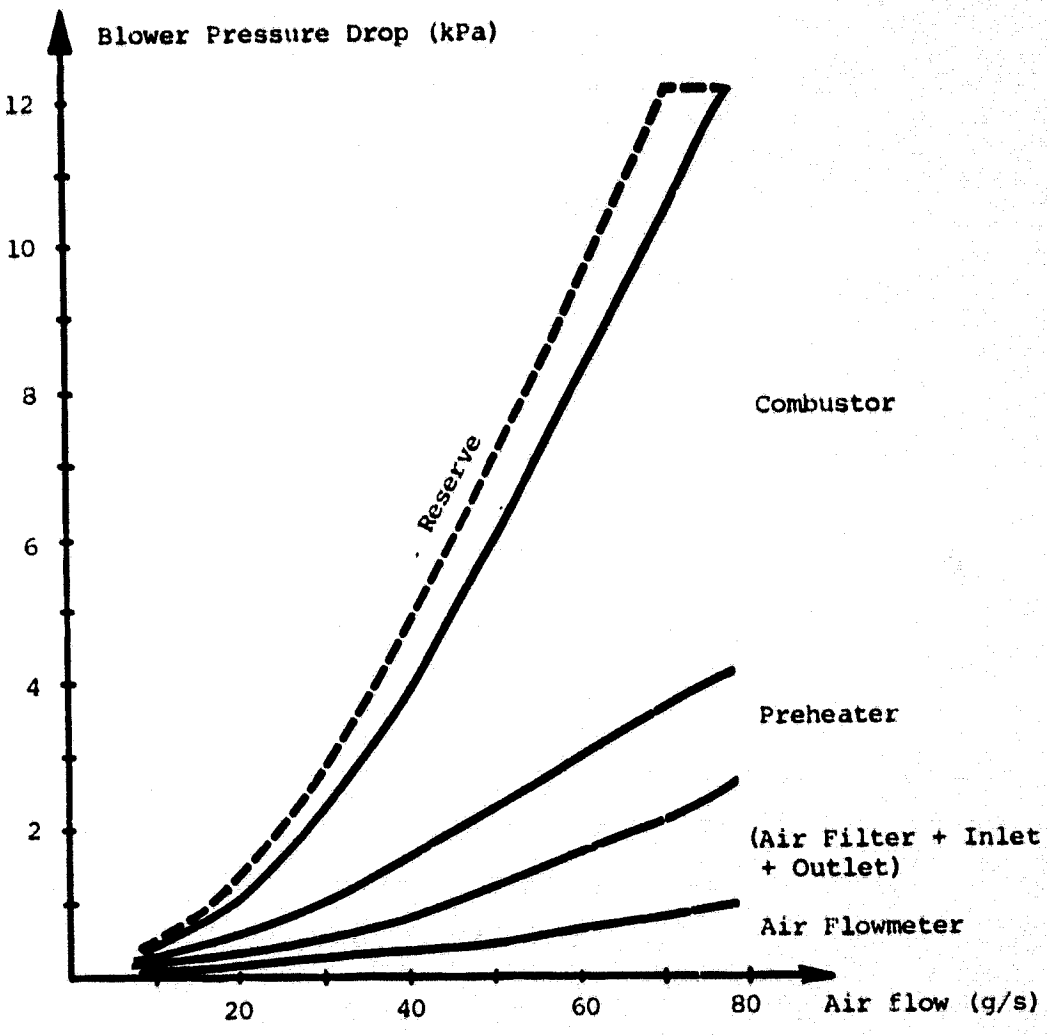


Fig. 8.3:9 Combustion System Pressure Drop Characteristics

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The speed increaser with the flat belt drive is developed by USSw and has the following data:

Maximum impeller speed	28,000 rpm
Gear ratio	1:6.33
Large pulley dia. (crowned)	95 mm
Small pulley dia. (crowned)	15 mm
Belt tensioner pulley dia.	25 mm
Belt type	Siegling UUIOE alt. Fredfors NE21
Belt width	15 mm
Belt length	470 mm
High speed bearings	Barden 201SSTX1
Max. transmitted power	1.5 kW

See page II-118 for a drawing of the blowers assembly.

The belt tension is accomplished by a separate spring loaded pulley. The spring consists of a number of cup springs in series. To avoid oil close to the belt transmission, the bearings have to be shielded and prelubricated with grease.

The blower housing will be made of cast aluminum.

A number of successful component tests with the flat belt drive have been made. For example: noise level tests, accelerated endurance test for 1000 hours, 100,000 cycles, acceleration and deceleration test and cold start tests in temperatures down to -28°C.

One of the pulleys of the variable ratio belt drive is located on the input shaft to the flat belt transmission. This shaft is via an overrunning clutch connected to a concentric shaft, which extends through the transmission case, opposite the input shaft from the variable speed drive. This shaft extension is by means of a toothed belt coupled to the atomizer air compressor and the electric blower motor. The free wheel allows the blower motor to drive only blower and atomizer air compressor unit during the starting up period.

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The blower is inlet throttled, and the throttle position is normally somewhat closed to give an air reserve for quick demands.

So far no efficiency measurements on the blower and variator have been made, and the figures in Fig 8.3:10 are theoretical values.

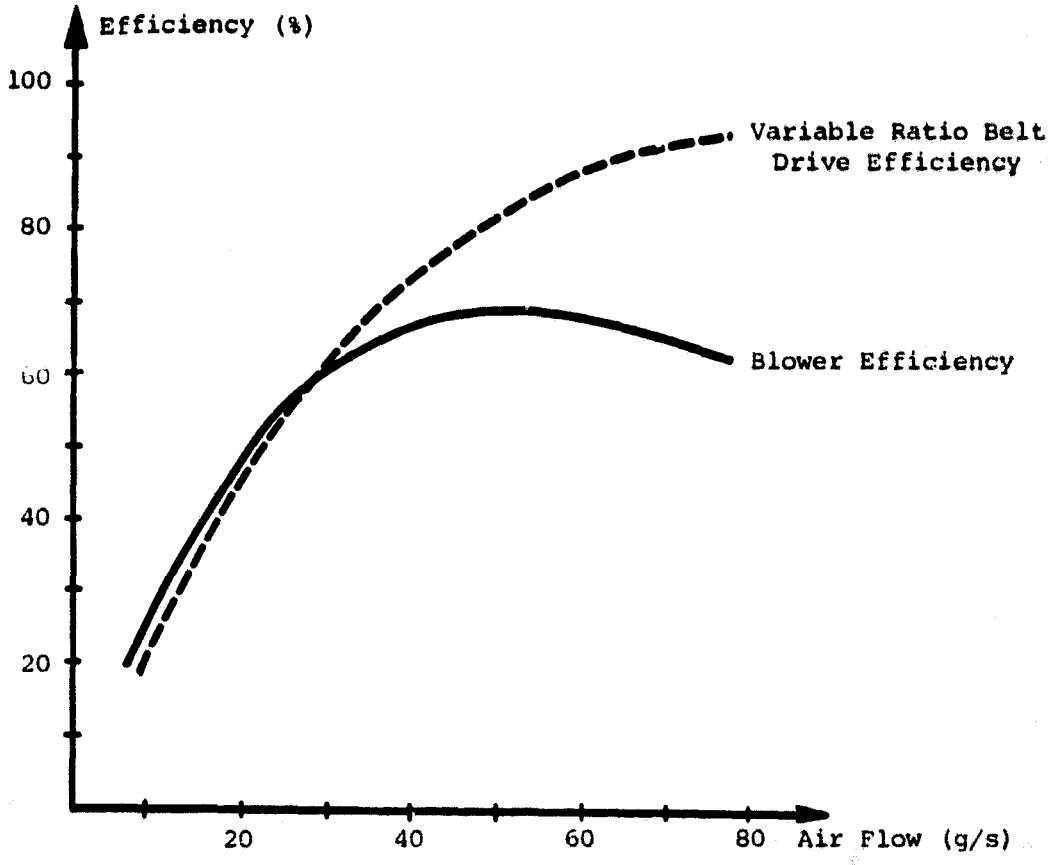


Fig 8.3:10 Variable Ratio Belt Drive Efficiency

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The blower power can be calculated as follows:

$$P = \frac{\Delta P * \dot{m}_f * \lambda * 14,5 * 1,05}{\rho_a * \eta_B * \eta_V}$$

ΔP = Total pressure drop with throttle in somewhat closed position. (Fig. 8.3:9)

$\dot{m}_f$ = Fuel flow

λ = Air excess factor (Fig. 8.3:8)

ρ_a = Air density 1,11 kg/m^3 (+40°C)

η_B = Blower efficiency (Fig. 8.3:10)

η_V = Variable ratio belt drive efficiency (Fig. 8.3:10)

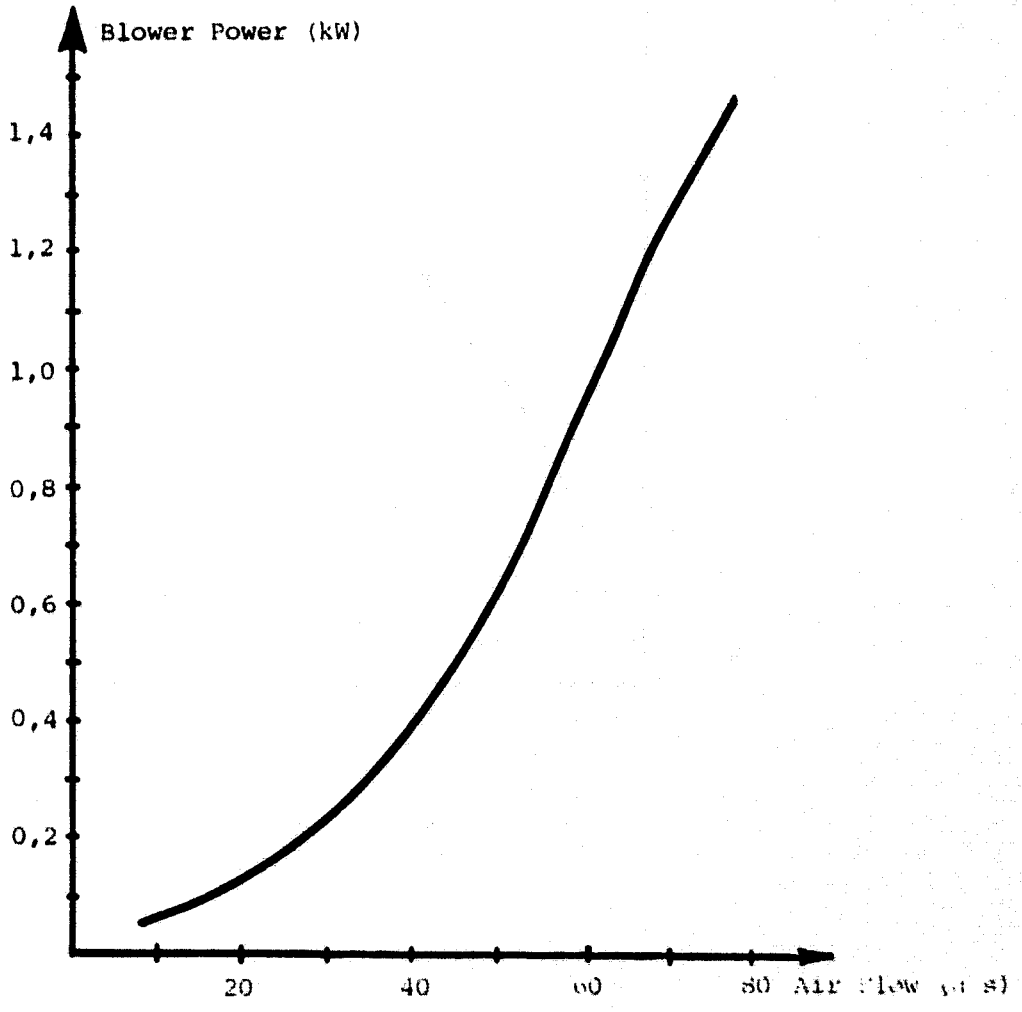


Fig. 8.3:11 Blower Power Characteristics

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8.3.7 Air throttle

Application: The throttle is used for fine adjustment of the air flow to the combustor. It is also activated when a fast air flow shut off or air flow increase is necessary.

Specification: The air flow characteristics shall be as linear as possible with the throttle disc position.

Although, no fast throttle movements are necessary during normal operating conditions, the travel time from one end position to another should be 0.7 sec., when fast response is needed.

Normal position of the throttle is somewhat closed to give an air reserve for fast demands. The main combustion air flow control is accomplished by the variable ratio belt drive.

Tests with a similar throttle indicates a torque requirement of 0.02 Nm to turn the shaft.

Description: The throttle is a butterfly valve with a friction clutch between the plate and the DC servomotor. The clutch is used instead of limit switches to protect the servomotor. The motor has a two step gearbox with a total gear ratio of 354:1. The first step is a spur gear transmission with a ratio of 11.8:1. The second step is a worm gear with a ratio of 30:1. The motor speed is 12 000 rpm and with the total gear ratio it will give the valve shaft a speed of 33.9 rpm.

With this speed the valve travel time between end positions will be 0.44 sec.

The throttle has a feed back potentiometer connected to the valve shaft via a gear transmission.

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A test with five different control methods of a centrifugal blower indicates that the power saving with inlet guide vanes compared to a butterfly valve is not big enough to justify the more complex inlet guide vanes (pg. II-119).

Page II-120 provides a drawing of the butterfly valve.

8.3.8 Variable ratio belt drive

Application: The driving pulley of the variator is located on the main engine shaft, and the driven one on the blower input shaft.

The main purpose of the variator is to make the combustion air blower speed independent of the engine speed. If there was a fixed gear ratio between the blower and engine, there would be a lot of excess air, for example at high engine speeds and low loads, which had to be throttled away with high power consumption as result. Fig. 8.3:12, which shows the blower speed vs engine speed at different loads can illustrate this.

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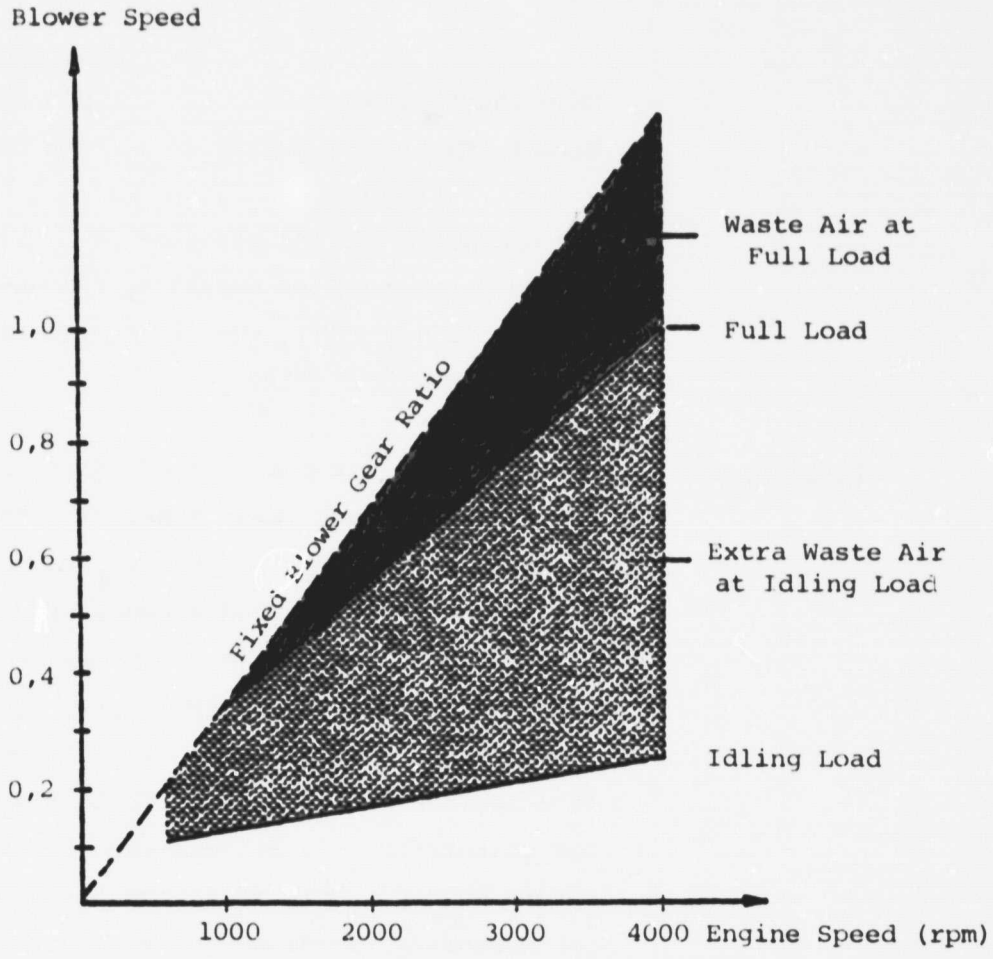


Fig. 8.3:12 Blower Speed/Engine Speed Characteristics

The variator is also driving the atomizer air compressor and servo oil pump for power control from the blower input shaft. This will also give a power gain compared to a direct drive from the engine shaft.

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Specification: The gear ratio for the variator is 1:1,6 - 2,6:1, and the speed ranges for the pulleys are:

	Min. speed	Max. speed
Main shaft pulley	600 rpm	4000 rpm
Blower input shaft pulley	960 rpm	4400 rpm

The hydraulic control pressure is 3 MPa.
To control the pulley position, two on-off solenoid valves are necessary, one to increase and one to decrease the gear ratio.

Description: The variator is a modified RF 150 b unit from C. & W. Berges AG, Germany (page B.8-116). The modification consists of a built-in hydraulic piston on the main engine shaft pulley, and a redesigned spring on the blower input shaft pulley. The oil to the piston is taken from the power control servo system, and transmitted by a rotating union in the front of the pulley. If a failure occurs due to the loss of oil pressure, the gear ratio will be decreased because of the spring loaded pulley. This will give the blower a speed corresponding to idling speed, and the risk for overspeeding the blower is eliminated.
Engine tests have shown that the air demand changes from the combustor are rather slow, which reduces the requirements for a fast response of the variator control system.

The complete control process is described in chapter 8.2.3.2

A drawing of the engine main shaft pulley with the built-in hydraulic piston is illustrated on page II-122.

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8.3.9 Atomizer air compressor

Application: The compressor is used to compress air for atomization of the fuel in the fuel nozzle. It is also used for crankcase ventilation.

Specification: The necessary air flow and pressure for atomization is 1.5 g/s at 70 kPa.

Description: The compressor is an oil less vane type compressor with carbon vanes. It is a modified Gast 0740 compressor, where most of the cast iron parts are replaced by aluminium parts to reduce the weight from 4.2 to 2.5 kg.

The servo oil pump for power control, described below, is integrated with the compressor and driven from the same shaft.

The compressor has to take the air downstream of the air flow meter in the blower inlet, to make sure that the correct air-fuel ratio is maintained. This air flow is passed through the crankcase to provide a ventilation system.

The compressor power demand is as shown in Fig. 8.3:13

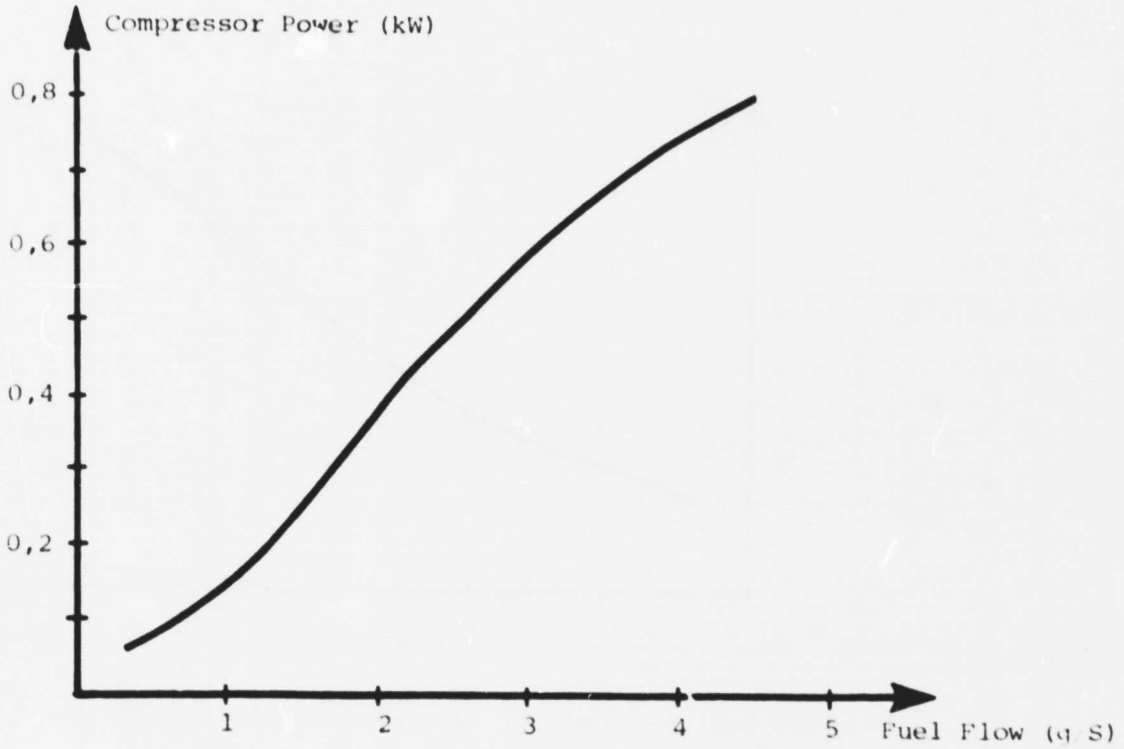


Fig. 8.3:13 Compressor Power Demand

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A drawing of the compressor with the integrated oil pump is presented on page **II-123.**

8.3.10 Servo oil pump

Application: The pump is used for the power control system, which is described in shapter 8.4, and for the control of the variable ratio belt drive.

Specification: Minimum oil flow from the pump is 1 l/min., and pressure 3 MPa. It must be able to handle a normal hydraulic oil like Mobil DTE 24.

Description: The pump is a specially designed gear pump, with gears from W.H. Nichols Company, type 8030 class II. The width of the gears is 6.5 mm. The power demand is as shown in Fig. 8.3:14.

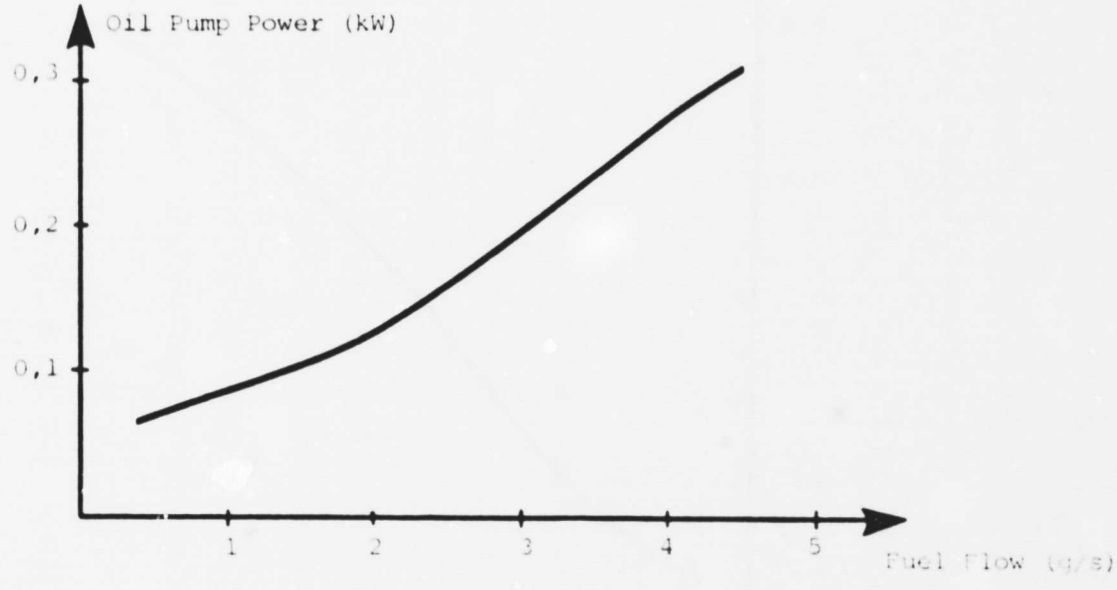


Fig. 8.3:14 Servo Oil Pump Power Requirements

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8.3.11 Pressure relief valve

Application: The valve is used to limit the atomizer air pressure.

Specification: The pressure must not exceed 70 Kpa.

Description: The valve is standard equipment from Gast Mfg. Corp., part No. AA 600. A section view of the valve is presented on page II-124.

8.3.12 Electric blower motor

Application: The blower motor is used during the starting up period to drive the combustion air blower, atomizer air compressor and the servo oil pump for power control. During this period the engine is not running and a system of free wheeling clutches makes it possible to run only these three auxiliaries.

Specification: The motor shall be a 12 V unit with a power output of at least 0.78 kW.

Description: The motor is series-wound and a standard Bosch unit with part No. 0 136 350 003. A free wheeling clutch is build into the hub of the belt pulley. During the starting up period this clutch is engaged and the electric motor is driving the above mentioned auxiliaries. When the engine has started and is driving the auxiliaries the clutch is disengaged and the motor is stationary. The motor is thoroughly tested outside the ASE program with good results. A drawing of the motor is on page II-125.

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8.3.13 Air-fuel system

Application: The system is used to control the air-fuel ratio, and to supply the engine with fuel.

To minimize delays between the air and fuel supplies to the combustor, it is preferable to primarily contr the air supply from the heater tube temperature. The fuel supply is then governed from an air flow signal and the desirable air excess factor.

Specification: The air-fuel ratio shall be kep within the numbers in Fig. 8.3:15 at the fuel nozzle, both during steady state and transients.

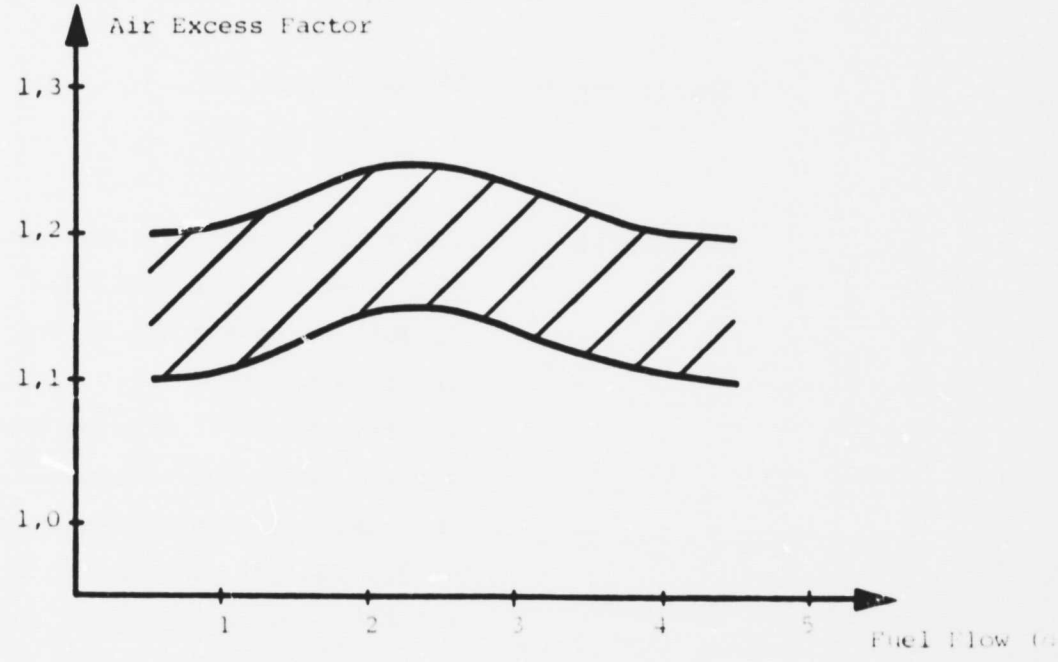


Fig. 8.3:15 Combustor Air/Fuel Characteristics

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The air control response shall respond to within ± 5% of final air flow value for a step from idle flow to full power flow in 5 sec., and shall respond to within ± 5% of final air flow value to a step from from full power air flow to idle air flow in 0.5 sec. The air/fuel control system shall be designed to be insensitive to changes in atmospheric pressure and temperature between -30°C to +45°C.

The air pressure drop in the air flow meter shall be kept as low as possible, and below 1.5 kPa at full air flow.

Description: The air-fuel system consists of the parts included in Fig. 8.3:16.

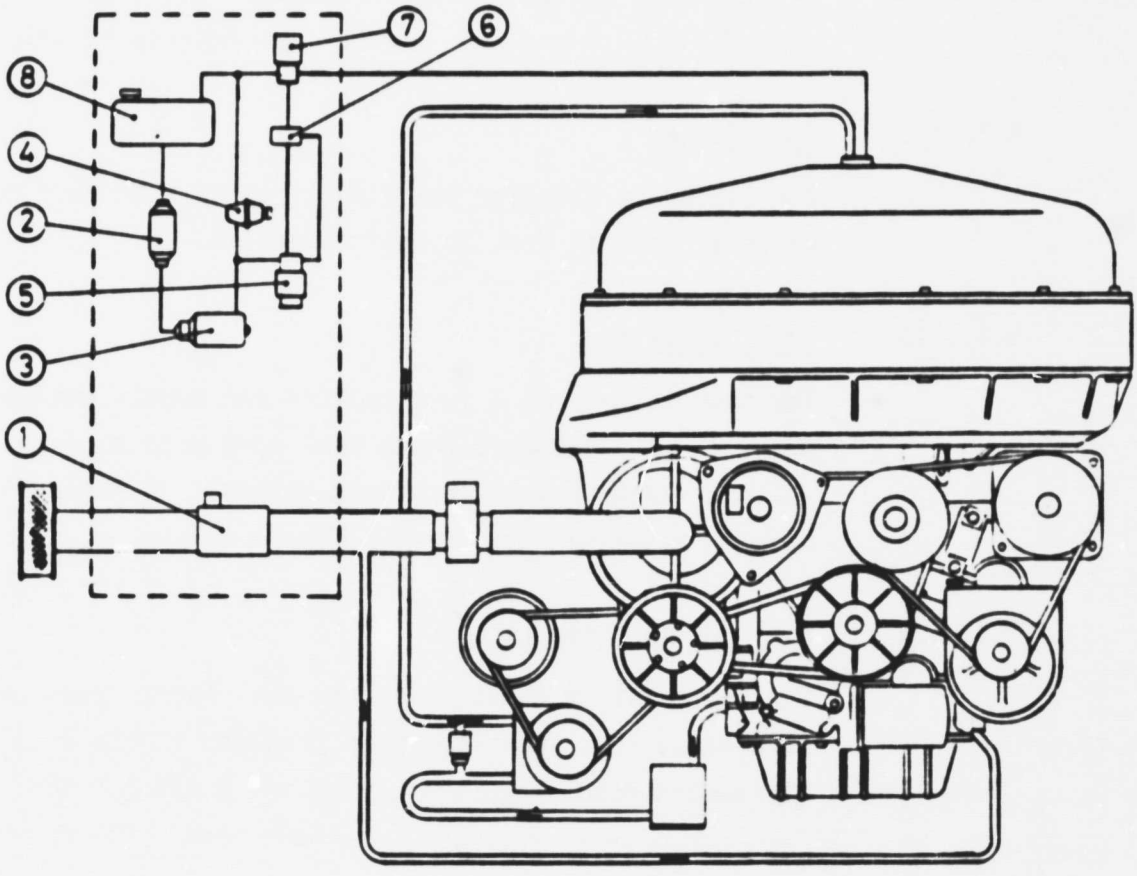


Fig. 8.3:16 Front View of a Mod I Engine

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1. Air flow meter
2. Fuel filter
3. Fuel supply pump
4. Fuel pressure regulator
5. Fuel metering pump
6. Fuel equalizer valve
7. Fuel valve
8. Fuel tank

As an example, an existing air-fuel system from Autotronic Controls Corp. is described below.

8.3.13.1 Air flow meter

The meter is a turbine flow meter with a carbon fiber reinforced plastic impeller. It is a volume flow meter with compensation for air pressure and temperature. The air flow range is 15:1. Page II-127 contains a further description of the flow meter.

8.3.13.2 Fuel filter

The filter is standard Bosch fuel filter with part No. 0450 905 010, which is used in fuel injection systems.

8.3.13.3 Fuel supply pump

The pump is used as a fuel booster and supply pump to the metering pump. It is a standard Bosch fuel pump with part No. 0580 254 996, which is used in fuel injection systems. A drawing of a similar pump is presented on page II-128.

8.3.13.4 Fuel pressure regulator

The regulator is controlling the fuel boost pressure and is set at a value just above the peak pressure in the fuel nozzle. It is a standard Bosch regulator, part No. B 438 180 000.

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8.3.13.5 Fuel metering pump

The pump is delivering the correct amount of fuel to the fuel nozzle in proportion to a signal from the electronic control system, based on air flow and air excess factor. It is a gear pump with multifuel capability, driven by a DC motor with a feedback tachometer. See page II-129 for further description of the pump.

8.3.13.6 Fuel equalizer valve

The valve is used to keep the pressure drop over the metering pump constant and at a low value. The internal leakage in the metering pump will thus be very low, and the output is proportional to the metering pump speed. A further description of the system and the valves is presented on page II-130.

8.3.13.7 Fuel valve

The valve is a three-way solenoid valve used for fuel shut off. When the fuel line to the engine is closed, a by-pass line to the tank makes it possible to run the metering pump.

8.3.14 Air filter

The filter is used to filtrate the combustion air and is an ordinary air filter used on IC engines.

8.3.15 Hydrogen compressor

The compressor is described in detail in chapter 8.4 . 8.7
The power demand is based on the compressor running in a short circuiting mode. See Fig. 8.3:17.

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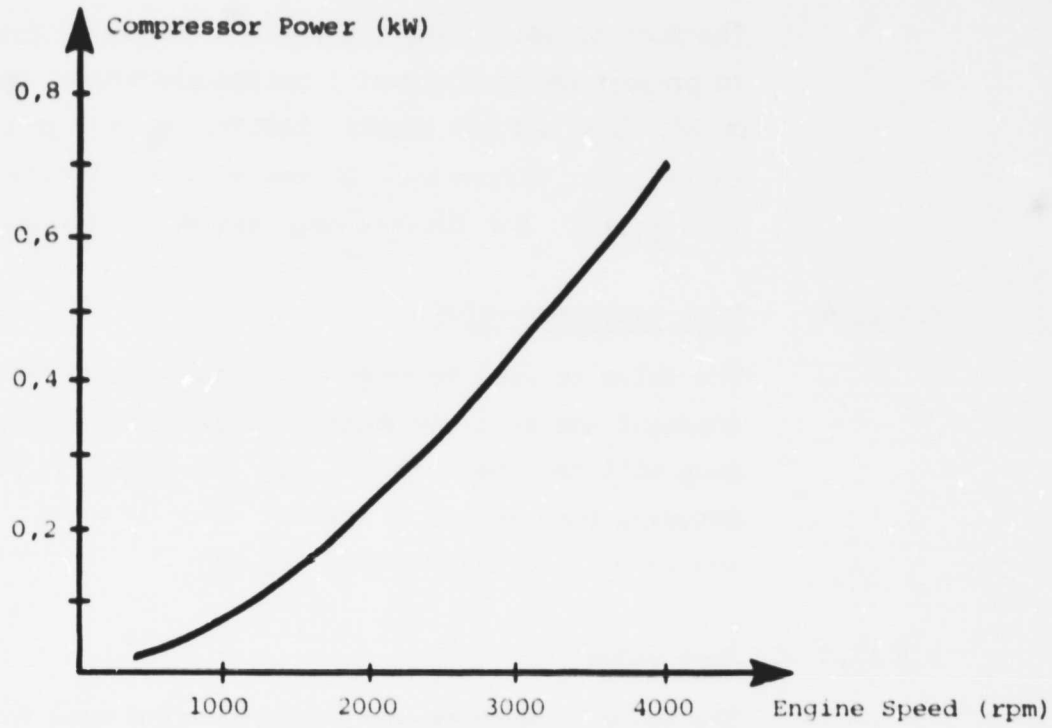


Fig. 8.3:17 Hydrogen Compressor Power Requirements (Short Circuit Mode)

8.3.16 After cooling pump

Application: The pump is used to circulate the cooling water to avoid overheating of seals, after engine has been shut off.

Specification: The pump is running intermittently and is governed by a thermocouple in one of the Stirling coolers. Tests have shown that a waterflow of about 0.5 l/sec. is sufficient.

Description: The pump is an axial flow pump, inserted in a bend in the main water circuit. The impeller is driven by a DC motor via a toothed belt transmission. A drawing of the pump is presented on page II-131.

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8.3.17 Crankcase ventilation

Application: To avoid the risk of an explosion due to hydrogen accumulation in the crankcase, a ventilation system is needed.

Description: The atomizer air compressor suction side is connected to the crankcase via an oil separation filter. The filter is needed to prevent oil from reaching the carbon vanes in the compressor and the fuel nozzle.

8.3.18 Air-conditioning compressor

The compressor is a standard unit for IC engines, manufactured by Sankyo, model No. SD 507. The power demand of the compressor is for example dependent of discharge pressure and evaporating temperature. The power demand at 240 psig and 40°F is shown in Fig. 8.3-18.

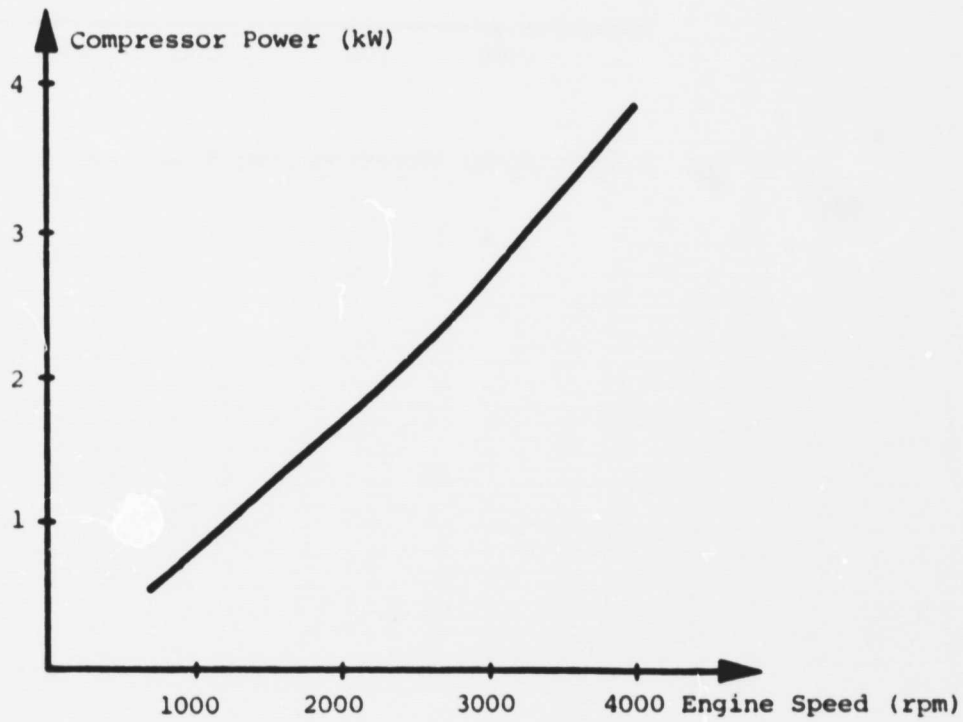


Fig. 8.3:18 Airconditioning Compressor Power Requirements

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8.3.19 Power steering pump

The pump is used both for power steering and power braking, and is a standard unit for IC engines. It is manufactured by Saginaw Steering Gear Div., model No. 783 00 00. The power demand is calculated to be as shown in Fig. 8.3:19.

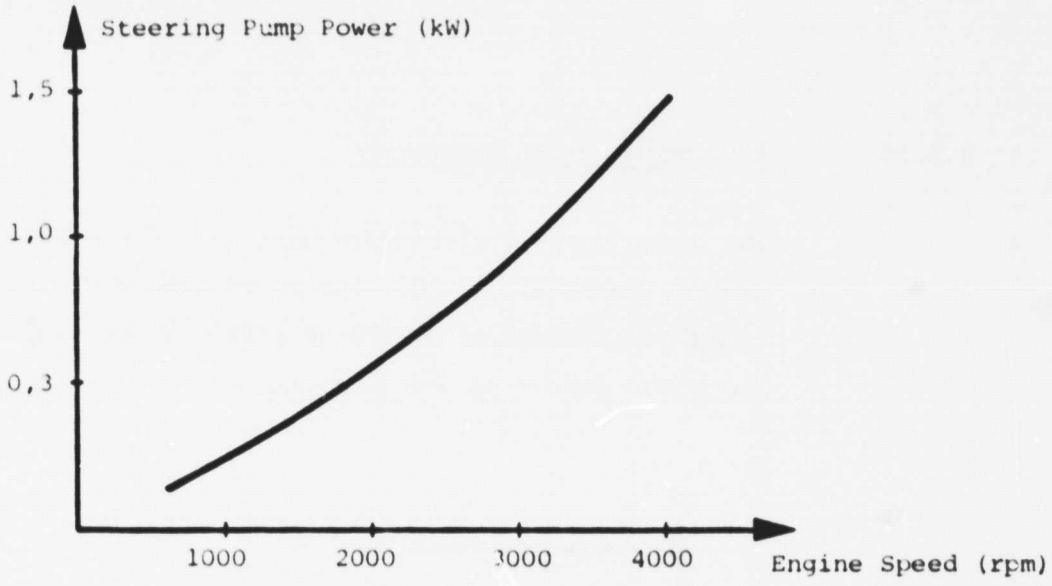


Fig. 8.3:19 Power Steering Pump Power Requirements

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8.3.20 Total auxiliary power demand

The total auxiliary power demand at full load condition can be seen in Fig. 8.3:20. The power demand at no load condition can be seen in Fig. 8.3:21. Excluded are the air-conditioning compressor and power steering pump.

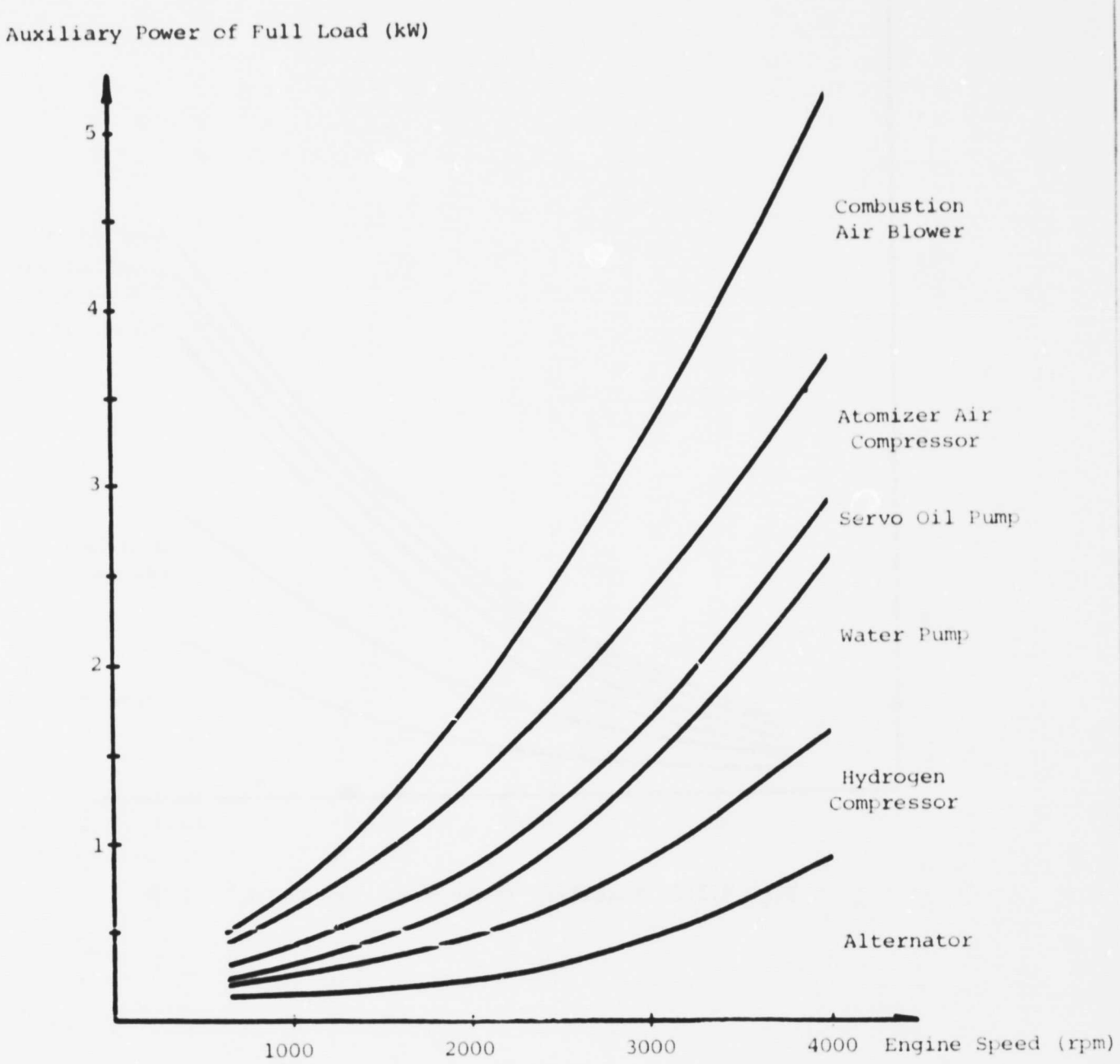


Fig. 8.3:20 Auxiliary Power Requirements at Full Power

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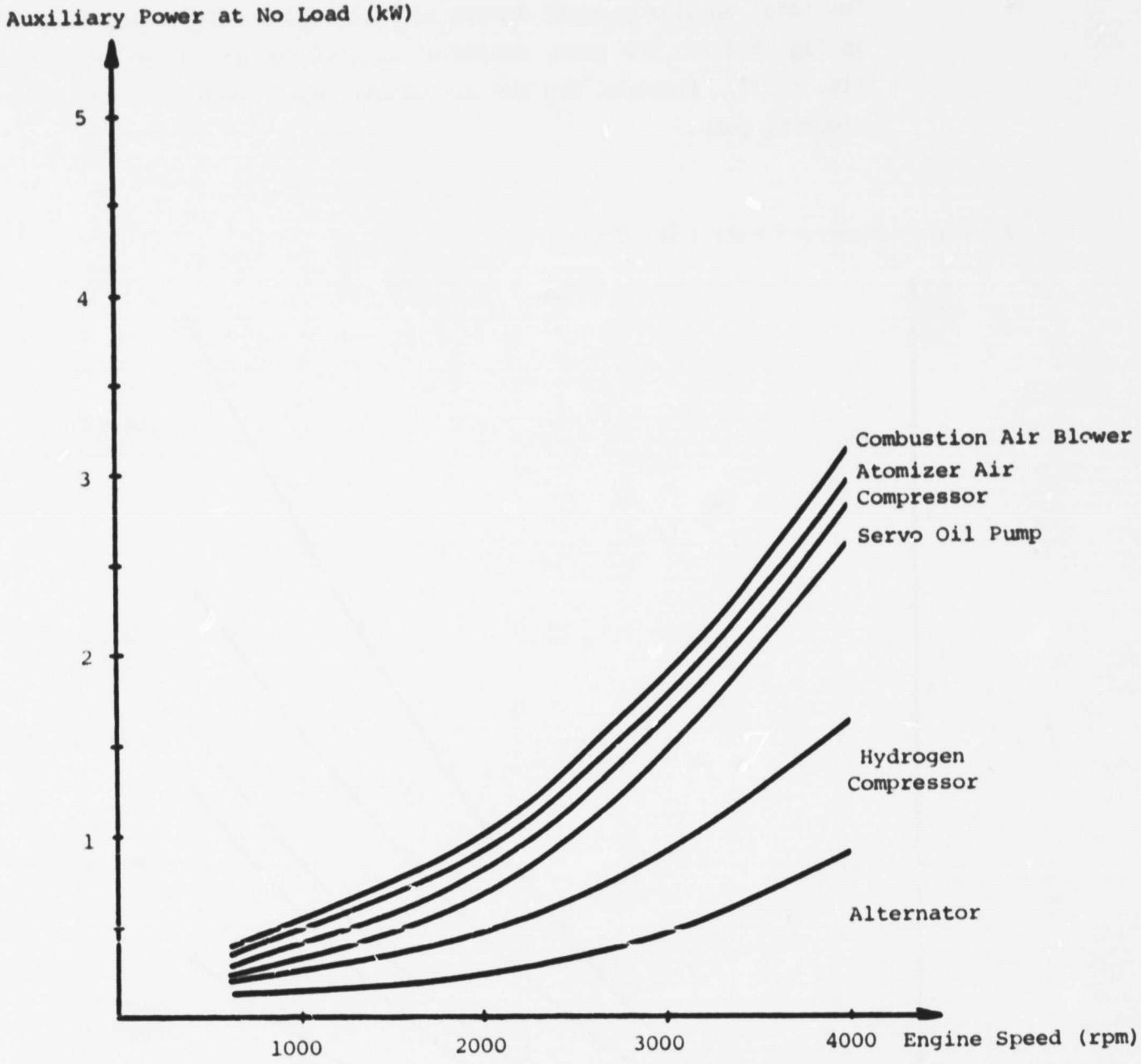


Fig. 8.3:21 Auxiliary Power Requirements at No Load

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APPENDIX

STARTER MOTOR REPORT

Conventional internal combustion engines require a starter to provide torque to overcome static friction and dynamic drag of the engine and to raise engine rpm to a level sufficient to initiate combustion and sustain operation. As illustrated in Figure 1, these same considerations are complicated in the Stirling engine by the additional power generated by the engine once the piston rings have seated. The combined power generated by the starter and the engine accelerate the engine. Piston rings which provide an optimum seal between the hot and cold spaces, allow the engine to significantly assist in the starting sequence. Sufficient power is generated by the engine so that, with termination of the starter motor at a relatively low rpm, the engine is self sustaining. Deteriorated piston rings, however, require continued cranking of the starter motor until the starter motor contribution is almost negligible before termination.

Insufficient knowledge of the engine's contribution requires qualifying the Mod I design by comparing it with P-40 starting experience and by limiting the analysis to a comparison of starter power to engine drag.

Figure 2 illustrates the engine drag starting requirements with ESSO 20W40 oil for summer temperatures, and a low viscosity oil, Mobil SHC 824, for winter (cold day) temperatures. Automotive practice includes temperature to -29°C (-20°F). The engine's drag includes piston ring, cross head and seal friction, bearing and gear losses, and oil pump and water pump power extraction. For the Stirling engine, these requirements demand an engine starter motor power that exceeds the power of motors commercially available that fit the Spirit transmission.

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Several options were investigated to meet the starter requirements. All were considered to be either an excessive compromise to the engine/vehicle system or too expensive and time consuming. The following items were considered:

- The Mod I engine is primarily a laboratory experimental engine with only one engine devoted to vehicle operation.
- The higher power required is principally due to higher piston ring friction and engine drag, items being addressed under component development activity.
- The starter motor requirement is not well understood without considering the Stirling engine's self-assistance characteristic.

The selected 12 volt starter motor system, a Chrysler 1.3 kW (1.8 hp) [P/N 4111680] unit, is compared to F-40 experience in the AMG Spirit in Figure 3. Figure 3 illustrates that for summer temperatures with 20W40 oil, the Mod I engine starting capability is similar to that of the Spirit. Figure 4 shows that by replacing the oil (recommended for temperatures below 10°C) with a low viscosity oil such as Mobil SHC 824, starting is possible to 0°C.

Mod I engine testing should include the acquisition of data to better define the engine's contribution to the starting sequence and a more accurate definition of the RESD starter requirements.

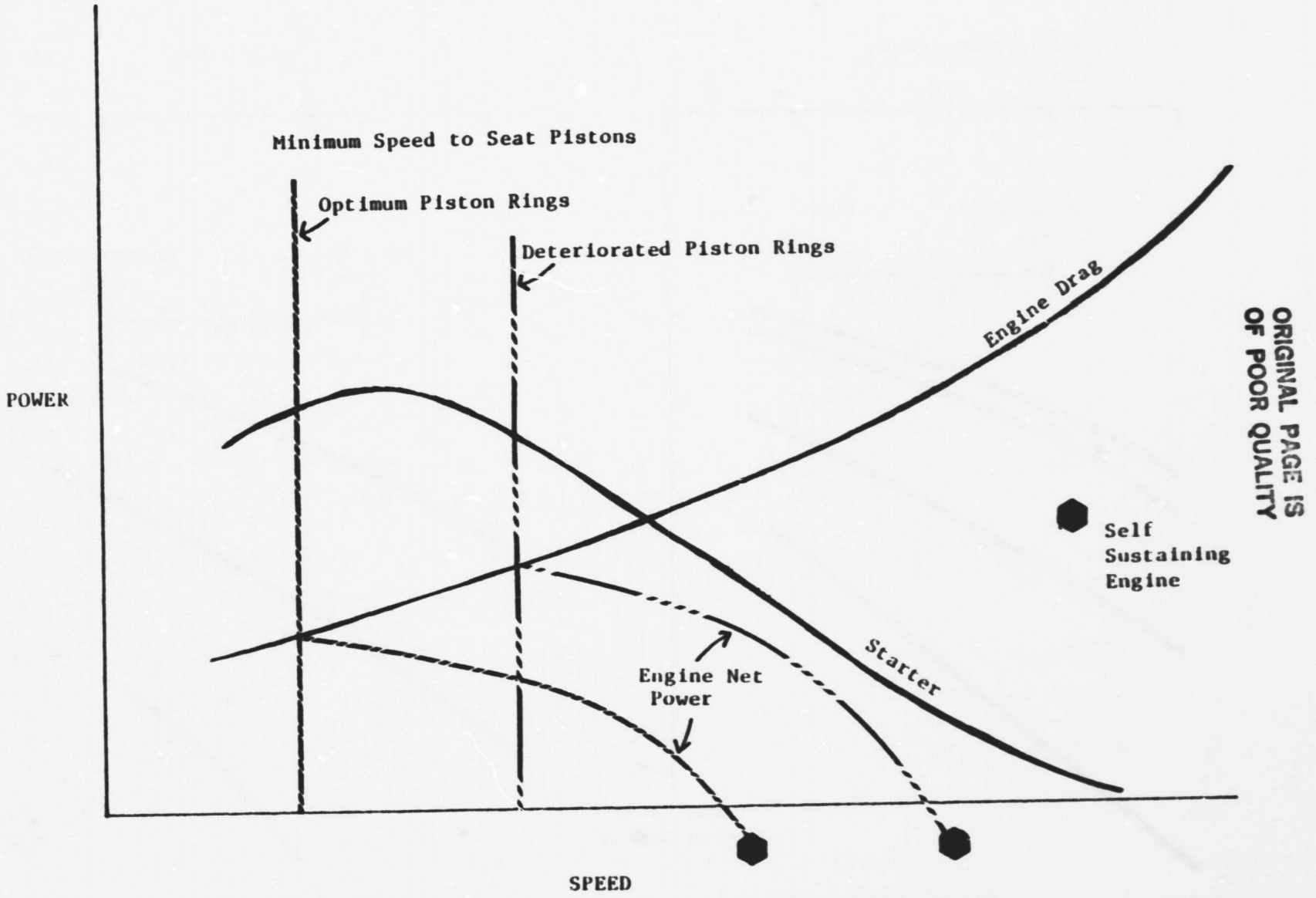
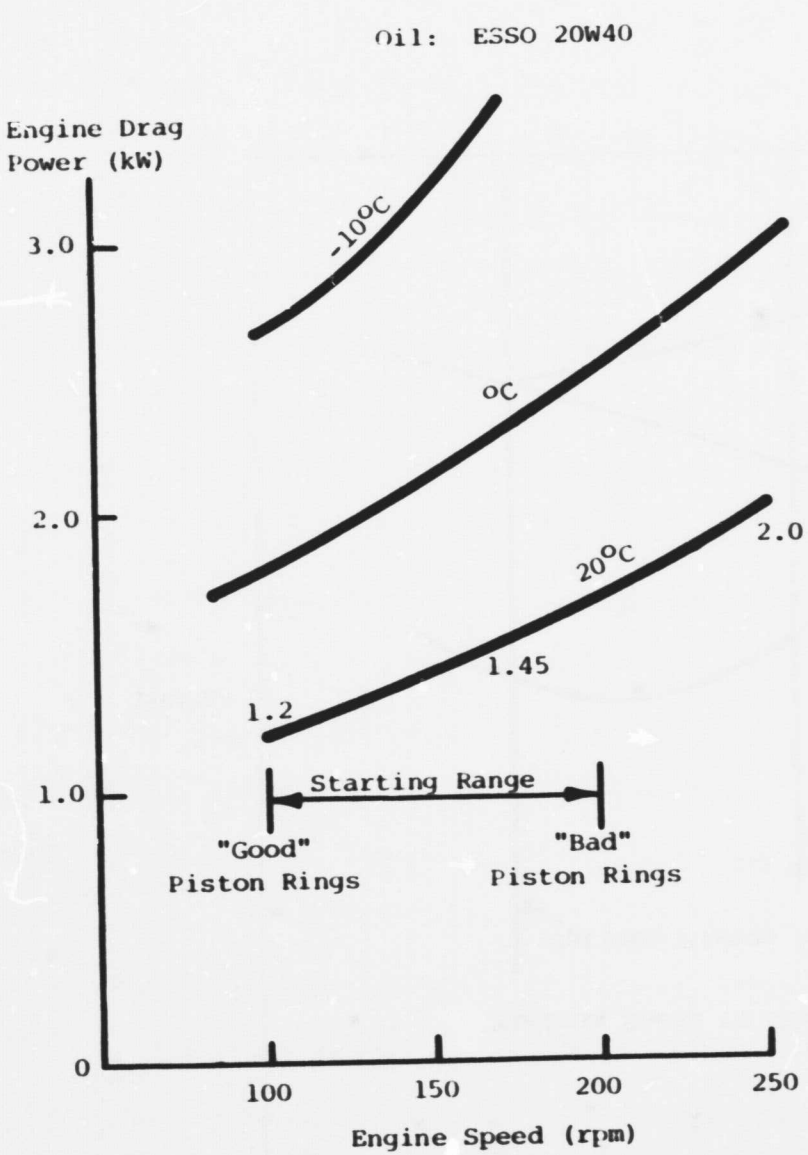


Fig. 1 Starting Power Characteristics

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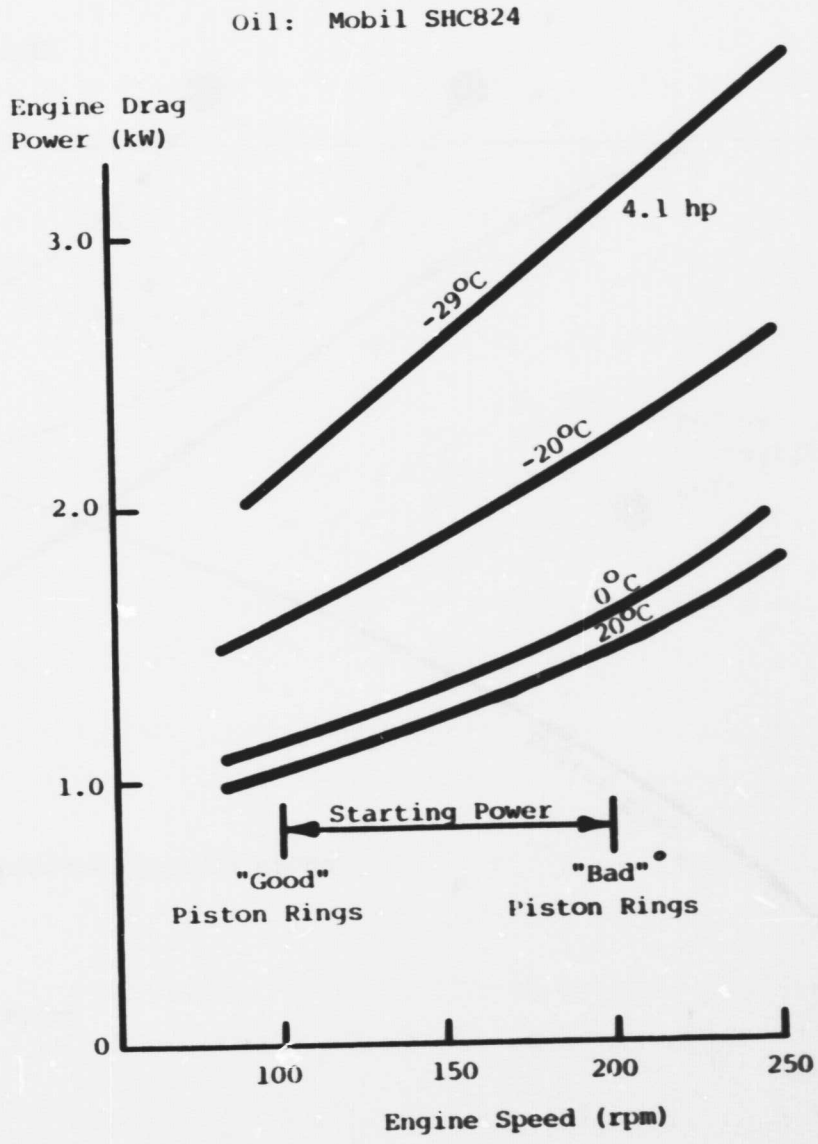


Figure 2 Mod I Starter Requirements
-29°C Starting Temperature, Automotive Standard

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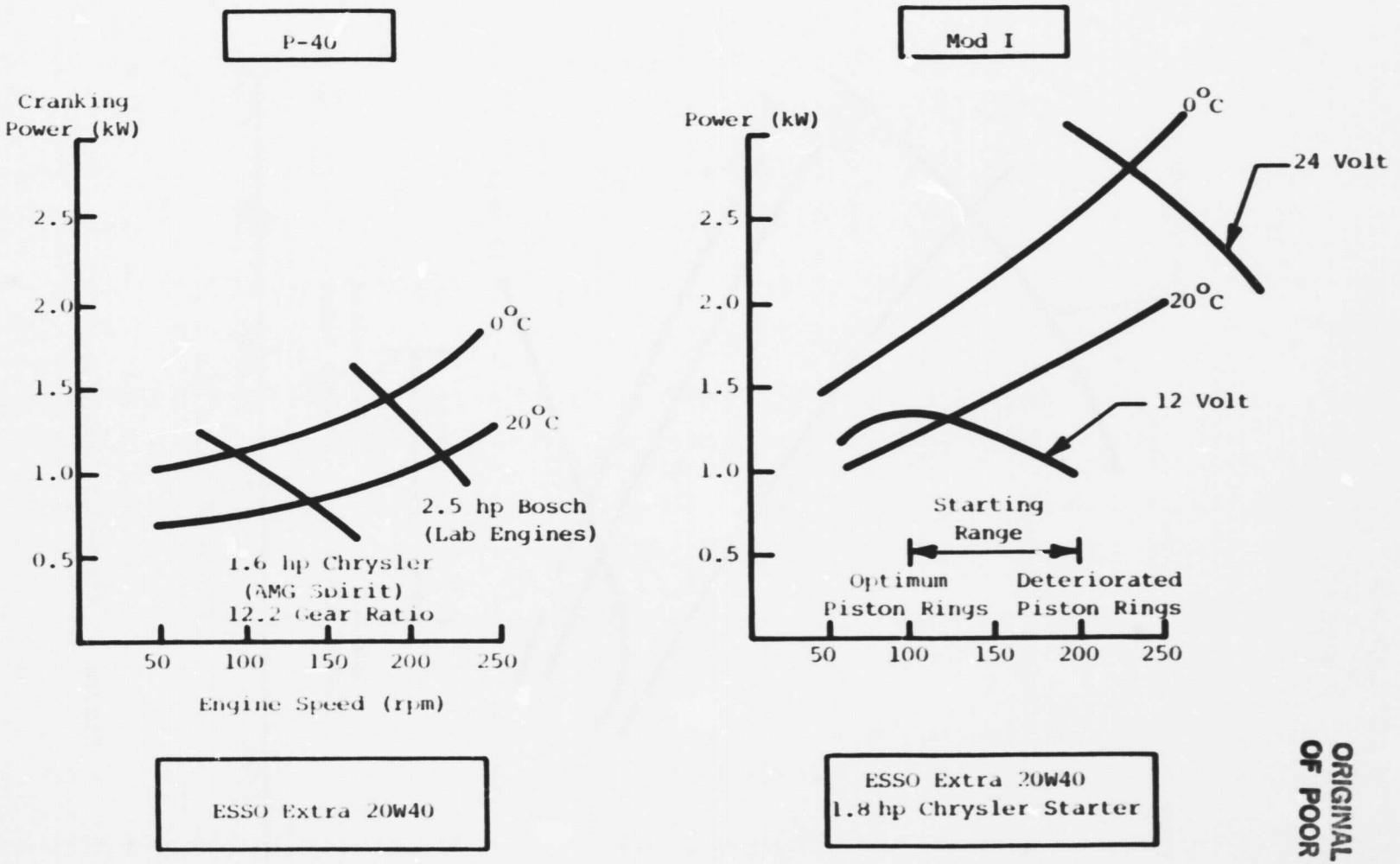


Figure 3 Mod I Starter - Comparison of P-40 to Mod I

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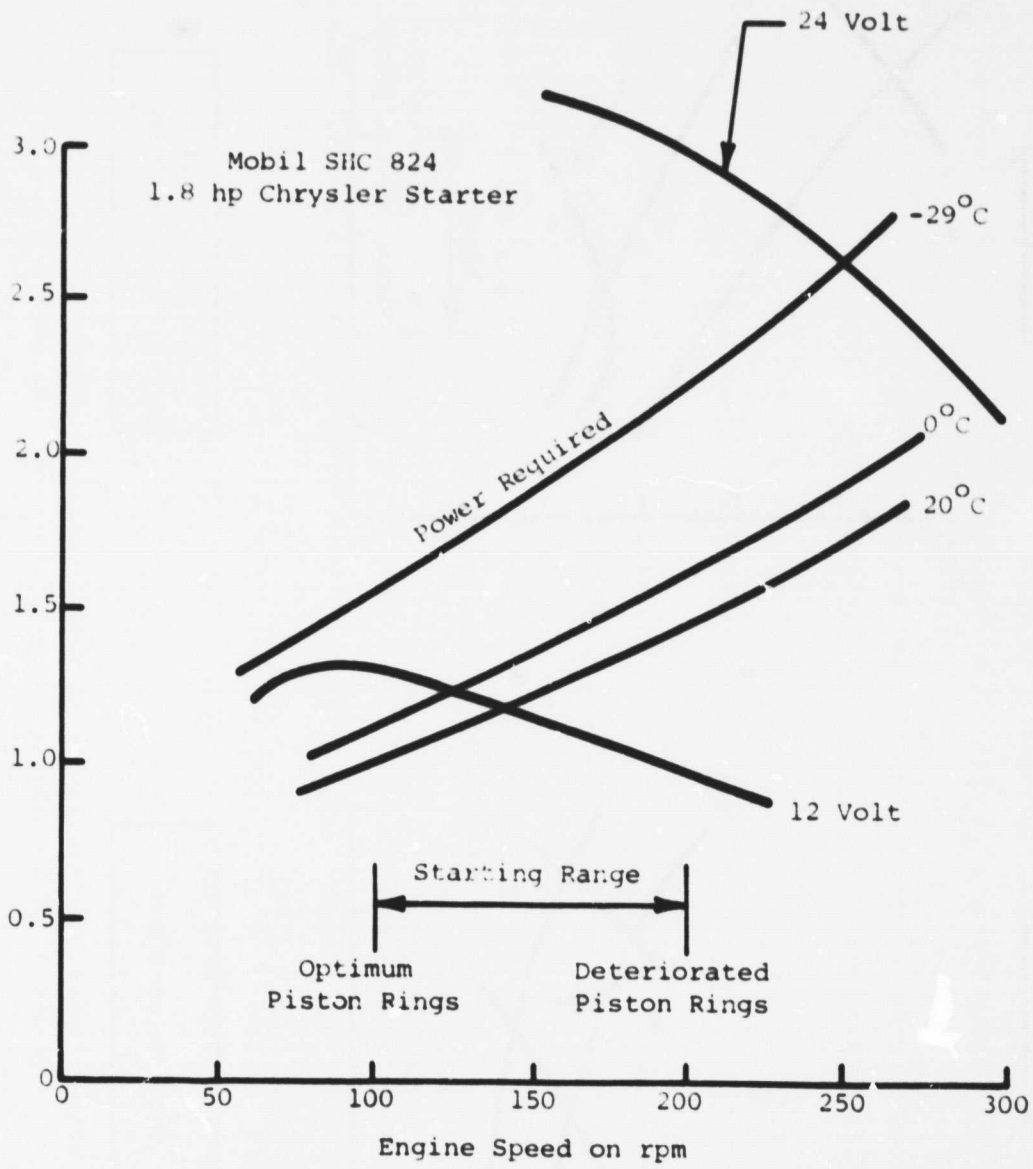


Figure 4 ASE Mod I Starting

TABLE I

MOD I ENGINE STARTER MOTOR

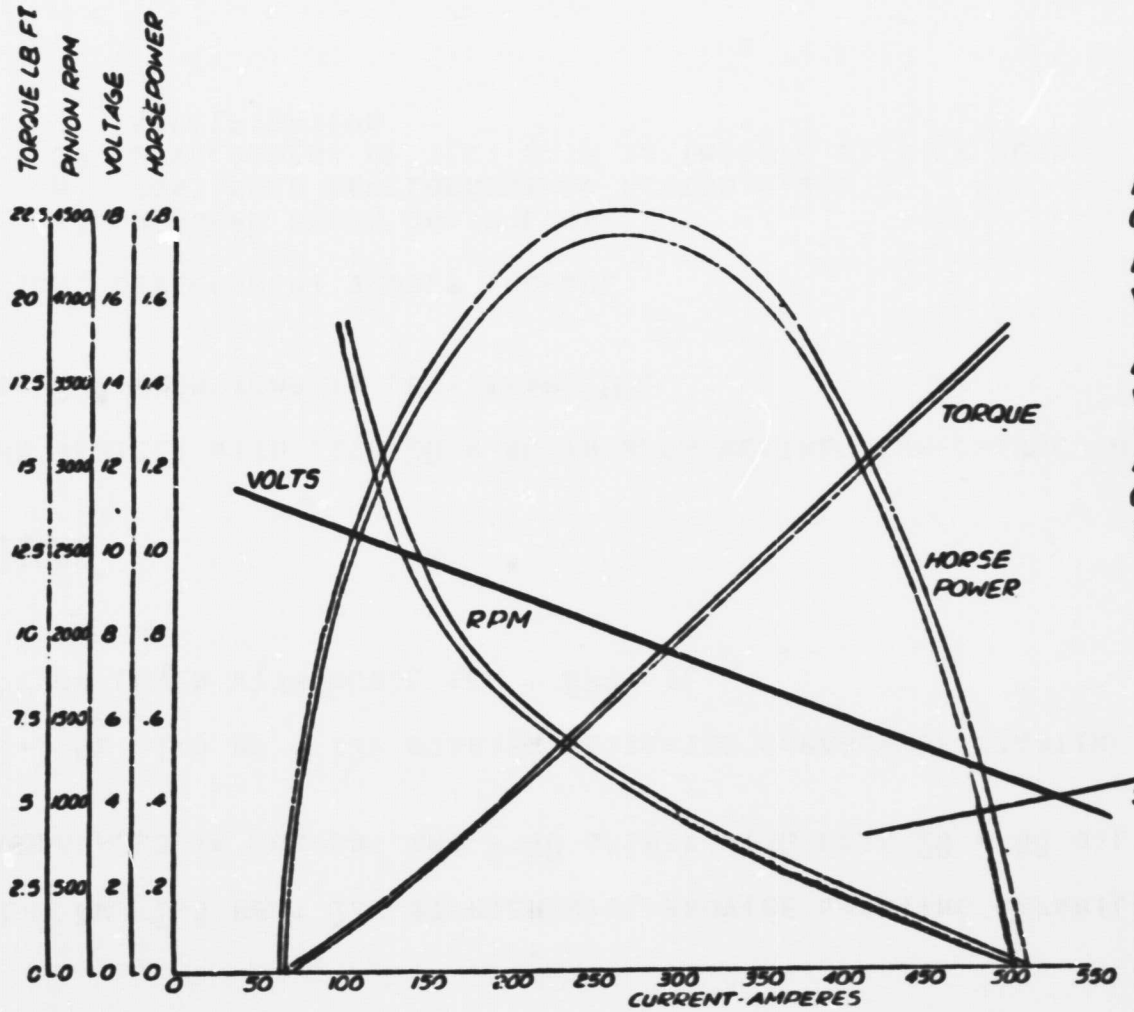
CONCLUSIONS

- (1.3 Kw) 1.8 HP - 12v STARTER WILL PROVIDE STARTING CAPABILITY COMPARABLE TO CURRENT AMG P-40 SPIRIT WITH ESSO 20 W 40 OIL
- (1.3 Kw) 1.8 HP - 12v STARTER ESTIMATED CAPABLE OF STARTING MOD I TO 0°C OR LOWER WITH MOBIL SCH - 824 OIL

RECOMMENDATIONS

- RUN VEHICLE WITH ESSO 20 W 40 IN WARM WEATHER AND CHANGE TO MOBIL SCH 824 WHEN TEMP IS LESS THAN 10°C
- FUTURE DEVELOPMENT SHOULD INCLUDE:
  - A) REDUCED POWER CHAIN LOSSES
  - B) CONTINUED DEVELOPMENT OF PISTON RINGS
  - C) DEVELOPMENT OF TEST DATA TO IMPROVE STARTER MOTOR SPECIFICATION

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PERFORMANCE CHECK
(BRUSHES NOT SEATED)

NO-LOAD TEST:

VOLTS	MAX AMPS	RPM
11.0	90	5700-6300

LOCKED RESISTANCE TEST:

VOLTS	AMPS	RESIST (Ω)
4.0	475-550	.0084-.0073

RUNNING PERFORMACE:
(VOLTAGE-USE VOLTS CURVE)

TORQUE	AMPS	RPM
10	300-315	800-900
5	190-200	1550-1650

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Fig. 5 Cranking Motor Performance Curves Motor Type-Gear Reduction P/N 3755250
Gear Ratio-Arm Shaft to Pinion 2:1

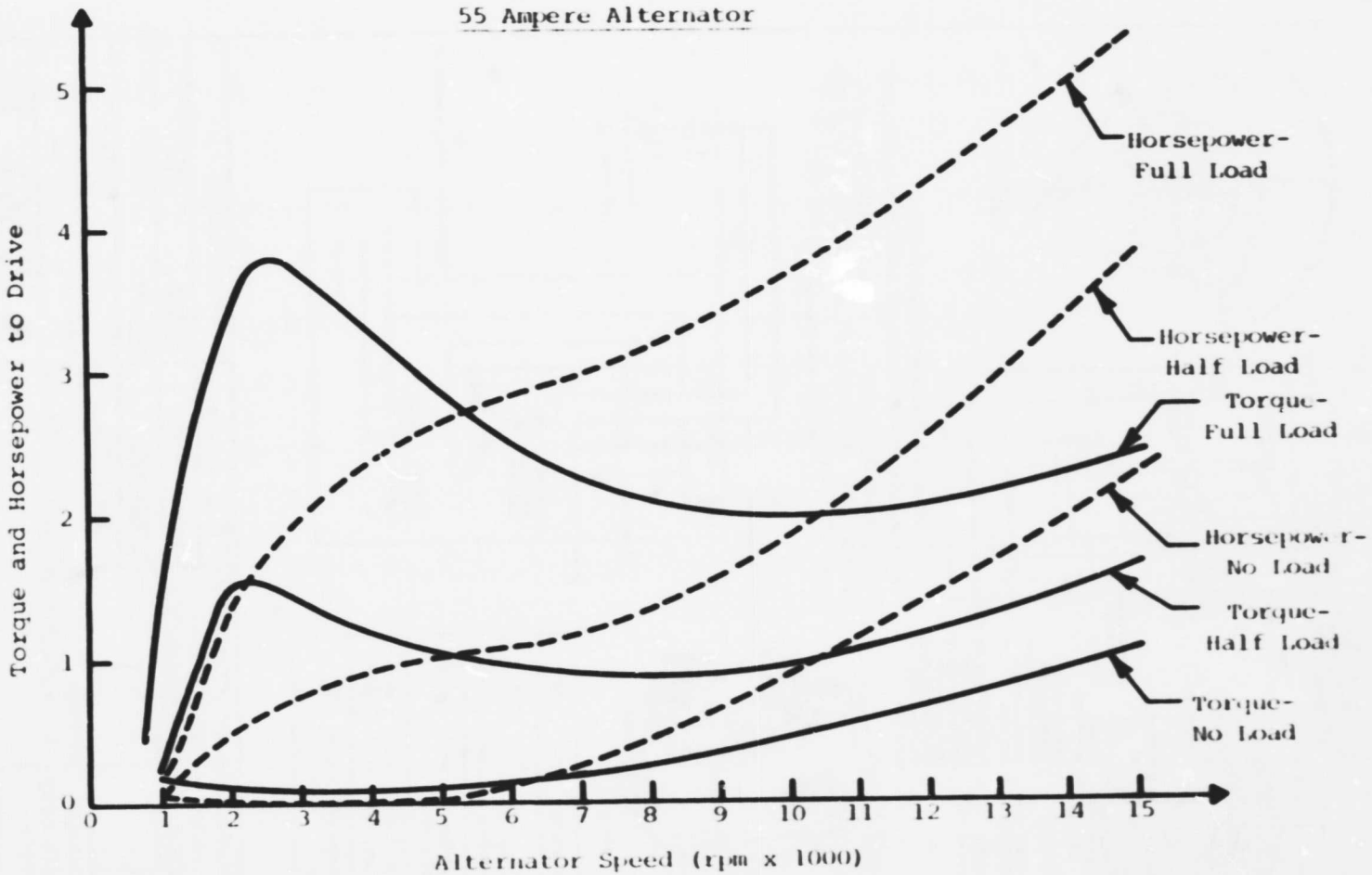


Fig. 6 55 Ampere Alternator Power Demands

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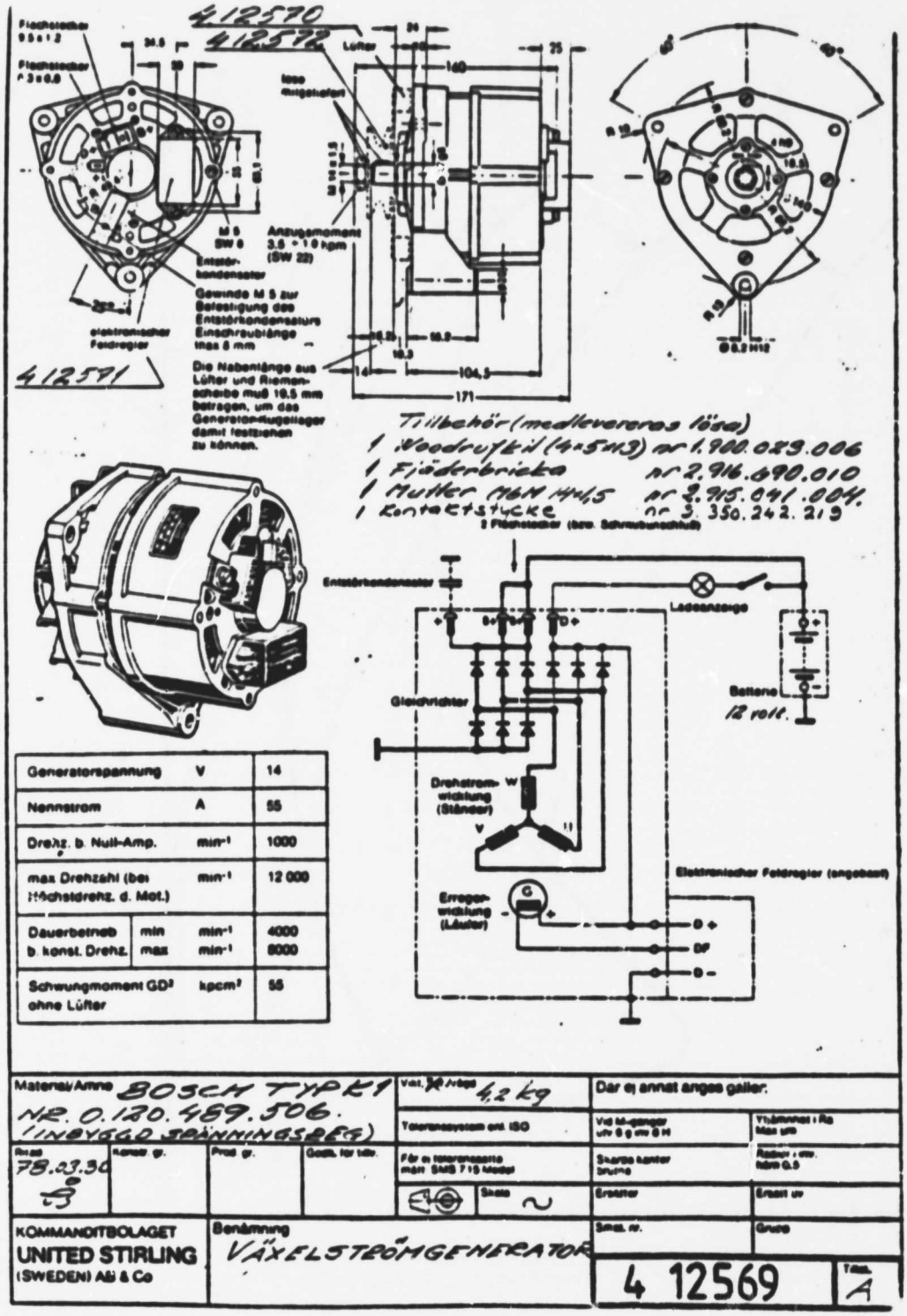
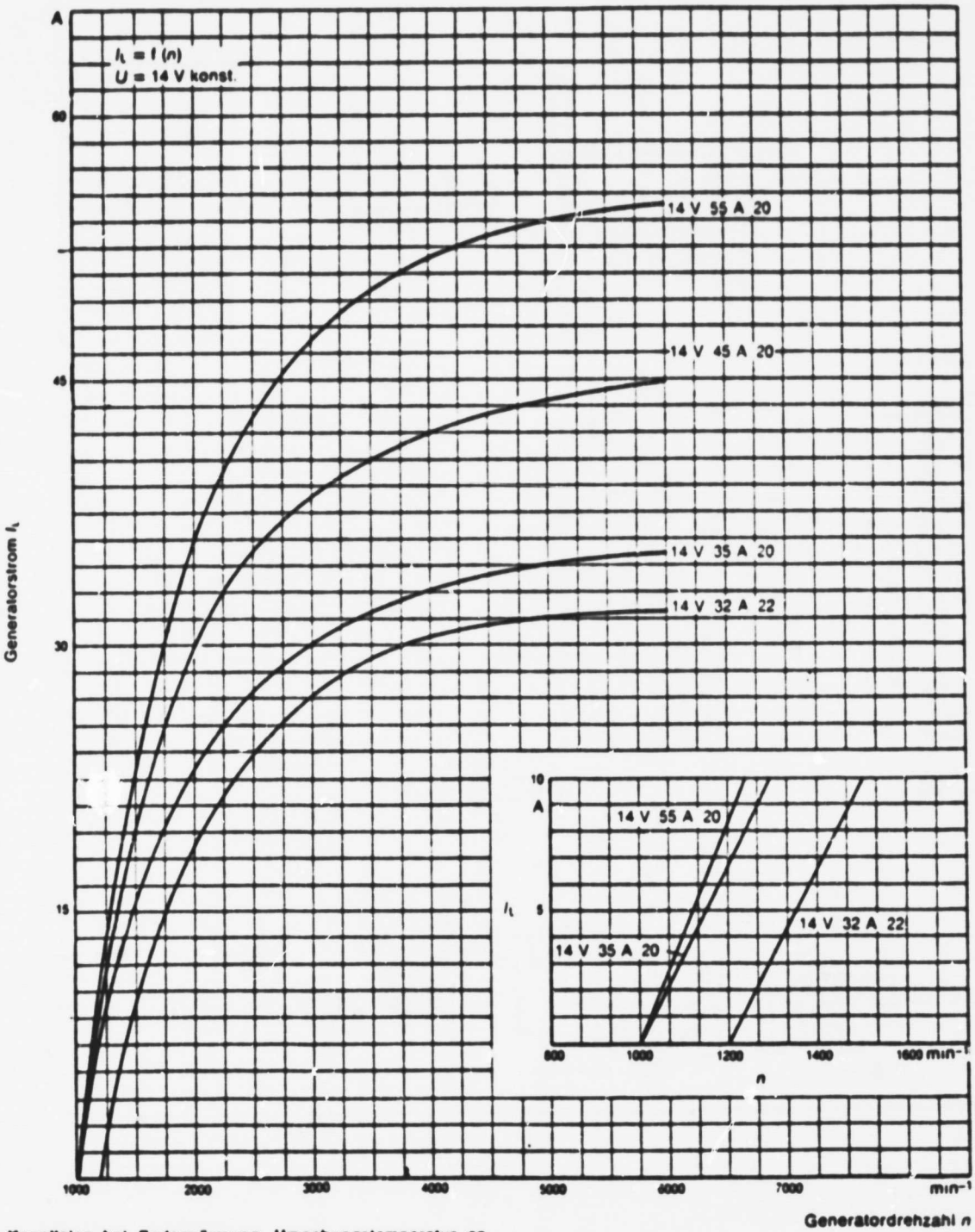


Figure 7

Kennlinien

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Typ K1 14 V 55 A 20
14 V 35 A 20
14 V 32 A 22
14 V 45 A 20



Kennlinien bei Enderwärmung, Umgebungstemperatur ca. 25°C. Regler berücksichtigt.

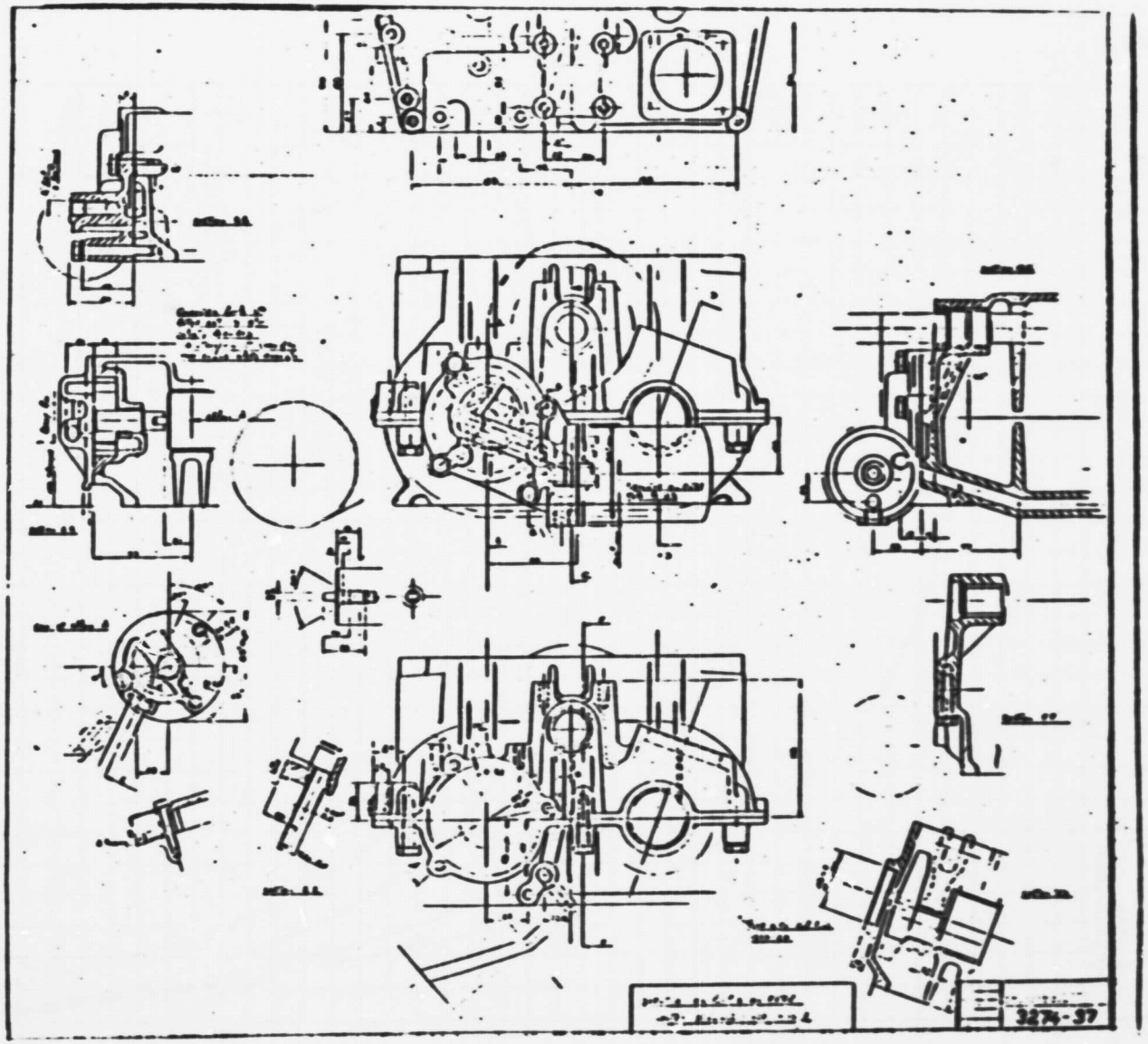


Fig. 9 Overall View of Oil Pump and Filter

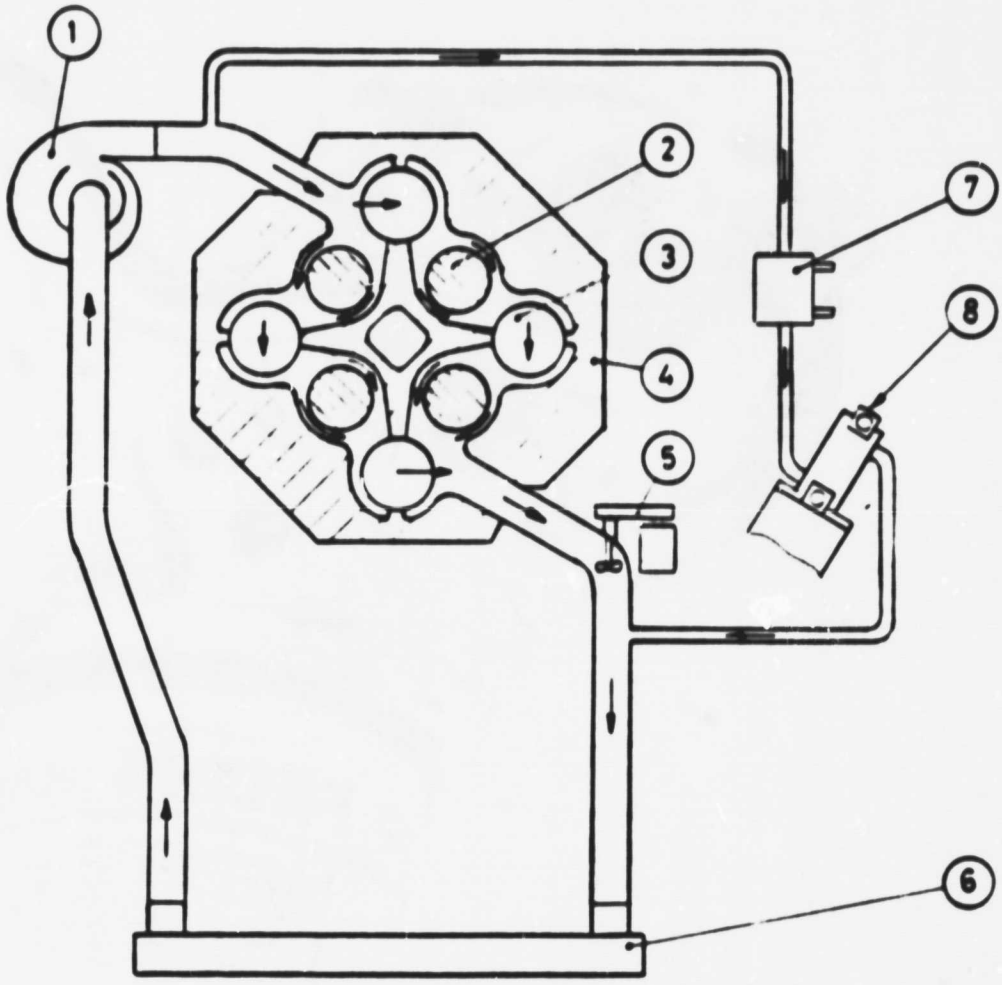
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Fig. 10 Oil Pump (42313K-9K)

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1. Water pump
2. Cylinder
3. Gas cooler
4. Water jacket
5. After cooling pump
6. Radiator
7. Hydrogen cooler
8. Hydrogen compressor

Fig. 11 Water Circuit

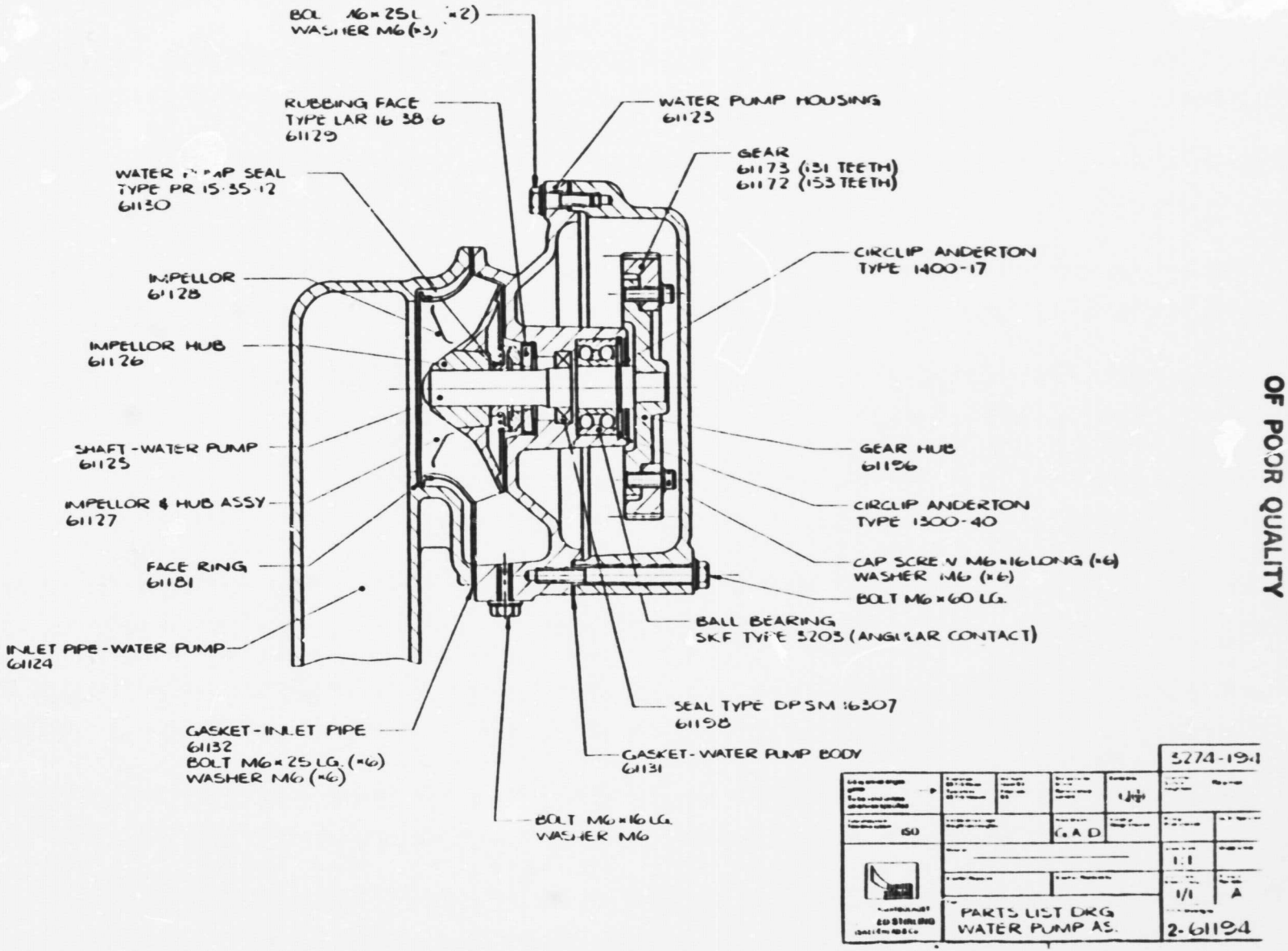


Fig. 12 Water Pump Assembly

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COMBUSTION AIR BLOWER DEVELOPMENT REPORT

As shown in Table 1, the investigation of P-40 auxiliary parasitic losses quickly identifies the combustor air blower as a major contributor. At max power, approximately 40% of the auxiliary losses can be attributed to the blower. A 30% reduction in the total auxiliary power extraction has been realized through the development of an efficient blower which was sized for the automotive application.

The blower specifications, which are defined in Table 2 and Figure 1, not only address improved efficiency but reduced component size, improved durability, and low noise. The geometry of the blower was selected by performing a parametric study. The study had three principal objectives:

- Selection of the type of blower for the Mod I application.
  - Selection of the centrifugal-type blower was accomplished by applying the blower specification values and required mechanical characteristics to the parameters defined on the turbo-compressor performance chart, shown in Figure 2.
- Optimization of the geometry for performance and cost.
- Analysis of bearing life and control speed.

The objectives and parameters investigated are defined in Table 3. For the Mod I flow, pressure rise, and desired diameter, a radial centrifugal blower provides the best opportunity for high efficiency. Table 4 compares the advantages and disadvantages of the various types of blowers.

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The optimum geometry of the centrifugal blower, its advantages and its disadvantages are also shown in Table 4. These options do not provide sufficient performance gains to justify the development and production cost for the Mod I application. Selection of the impeller diameter and designed speed of the blower was determined by evaluating the blower efficiency with a fixed volute housing outer diameter and optimum impeller tip speed shown in Figure 3. The blower geometry study resulted in the following conclusions, also shown in Table 5:

- The centrifugal blower is the best for the Mod I application.
- The least complicated geometry is the most cost effective.
- The optimum blower speed is 26,000 to 28,000 rpm.

After the desired blower speed was defined, a critical speed and bearing life analysis were performed. Figure 4 shows that the critical speed of the blower does not limit the desired speed range. With the estimated bearing stiffness, speeds up to 50,000 rpm are possible before the first critical speed limits the blower operation.

In the bearing life study, bearings were evaluated for fatigue, lubricant starvation and life, and cage failure. The significant conclusion and recommendations of this study are shown in Table 6. The environmental temperature of the bearing, which influences lubricant starvation, and, to a lesser degree, grease life, are the factors that limit the desired speed of the blower to under 30,000 rpm. The Bardon 201 SST x 1 bearing with phenolic and aluminum cages were selected for improved cage life.

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The experimental blower and the currently proposed production blower are shown in Figure 5. The experimental blower is machined and provides the flexibility to optimize the impeller height and volute housing. The proposed production blower has a cast housing and a slight reduction in the impeller diameter to improve the diffuser diameter ratio.

The experimental test rig is shown in Figure 6. Test results from this rig are shown in Figures 7 through 9.

Figure 7 shows the peak efficiency island and the data acquired from 5,000 to 26,000 rpm. All the data shown, from 0 flow to 20% in excess of the required flow, was stable. There was no blower surge experienced in any tests to date.

Figure 8 illustrates that the power demand of the straight vane combustor will require approximately 1.25 kW. This is about 1/3 of the previous power demand requirements of the blower.

Figure 9 shows the sensitivity of the blower performance to the impeller clearance and volute volume. The pressure rise of the blower is significantly altered by the impeller clearance. Decreasing the clearance from .015 to .008 inches improves the pressure rise about 5% (approximately 2-1/2% in efficiency). In other testing to reduce blower size, the volute housing volume was reduced by about 25% with no discernable reduction in performance.

TABLE 1

MOD I COMBUSTION AIR BLOWER

BENEFIT OF DEVELOPMENT

P-40 AUXILIARIES PARASITIC LOSSES

AUXILIARIES	LOSSES (@ 15MPA & 4000 RPM) Kw	%
COMBUSTION AIR BLOWER	3.2	43%
ALTERNATOR	2.2	57%
LUB OIL PUMP	.45	
WATER PUMP	.65	
OIL SERVE PUMP	.27	
ATOMIZING AIR COMPRESSOR	.55	
FUEL PUMP	.15	
TOTAL	7.47	100%

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CONCLUSIONS

- COMBUSTION AIR BLOWER 43% OF AUXILIARIES PARASITIC LOSSES
- SIGNIFICANT REDUCTION IN PARASITIC LOSSES IS AVAILABLE THROUGH DEVELOPMENT OF A SMALL, EFFICIENT BLOWER
- ESTIMATED AVAILABLE POWER REDUCTION - 2Kw @ MAX POWER OR 30% OF PARASITIC LOSSES

TABLE 2

COMBUSTION AIR BLOWER SPECIFICATION

GEOMETRY:	MAXIMUM DIAMETER	- 200MM (7.87 IN.)
	ROTATION	- CLOCKWISE LOOKING INTO BLOWER INLET
	USS/NTI INTERFACE	- BACK SURFACE OF IMPELLER HOUSING
PERFORMANCE:	SPEED	- TO BE DETERMINED BY PARAMETRIC STUDY
	EFFICIENCY	- 65-70% AT LOWEST AIRFLOW POSSIBLE, WITHOUT AN EXCESSIVE COMPROMISE OF MAX FLOW EFFICIENCY REQUIRED FOR ENGINE ACCELERATION
	AIRFLOW CAPACITY	- 5.3 GRAMS/SEC MIN (START/IDLE) 79 GRAMS/SEC MAX, INCLUDES 4 GRAMS/SEC MARGIN FOR LEAKAGE
	PRESSURE DROP	- 2 ALTERNATIVE COMBUSTION SYSTEMS (FOLLOWING FIGURE)
		1) <u>CGR SYSTEM WITH VALVE</u> BLOWER MUST PROVIDE FULL RANGE OF FLOW WITH STABLE OPERATION, AND WITH CGR VALVE FAILED, STABLE OPERATION BUT NOT FULL RANGE OF FLOW.
		2) <u>CGR STRAIGHT VANE SYSTEM</u> BLOWER MUST PROVIDE FULL RANGE OF FLOW WITH STABLE OPERATION.
	NOISE	- LOW NOISE LEVEL (APPROXIMATELY 78 DBA GOAL) WITH INLET MUFFLER INSTALLED.
	ENVIRONMENT	- MAXIMUM INLET TEMP. 40°C (104°F)

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TABLE 2 (CONT.)

BEARING/ROTOR ANALYSIS:

LIFE -3000 HOURS

OPERATION	-%TIME	% MAX SPEED
	5%	MAX
	30%	83%
	50%	67%
	15%	44%

BELT FORCE: $-\left(\frac{32 \text{ KG} + \text{BLOWER FORCE}}{2}\right) \leq 50 \text{ KG MAX}$

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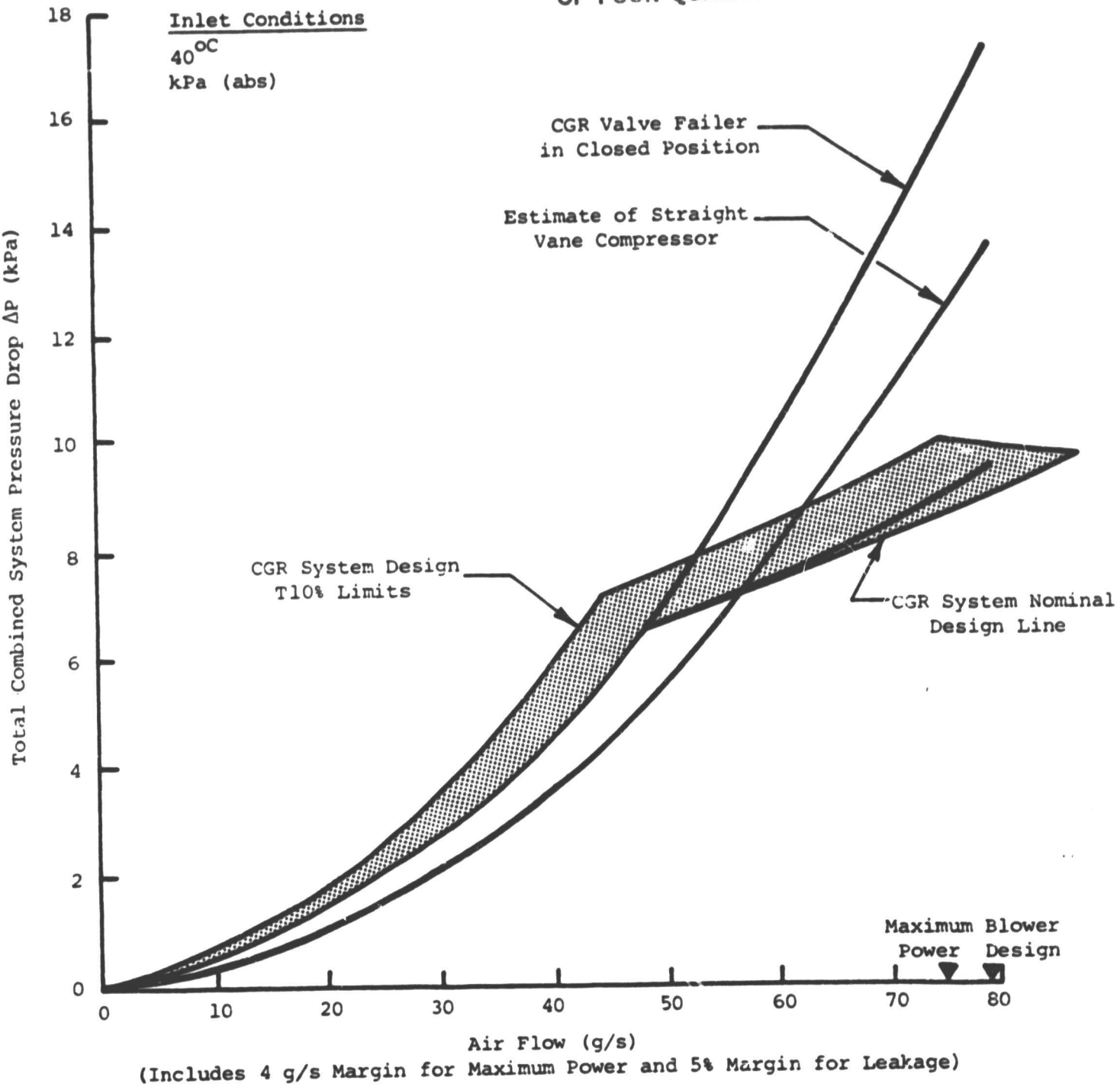


Figure 1 Combustion Air Blower Design Requirement

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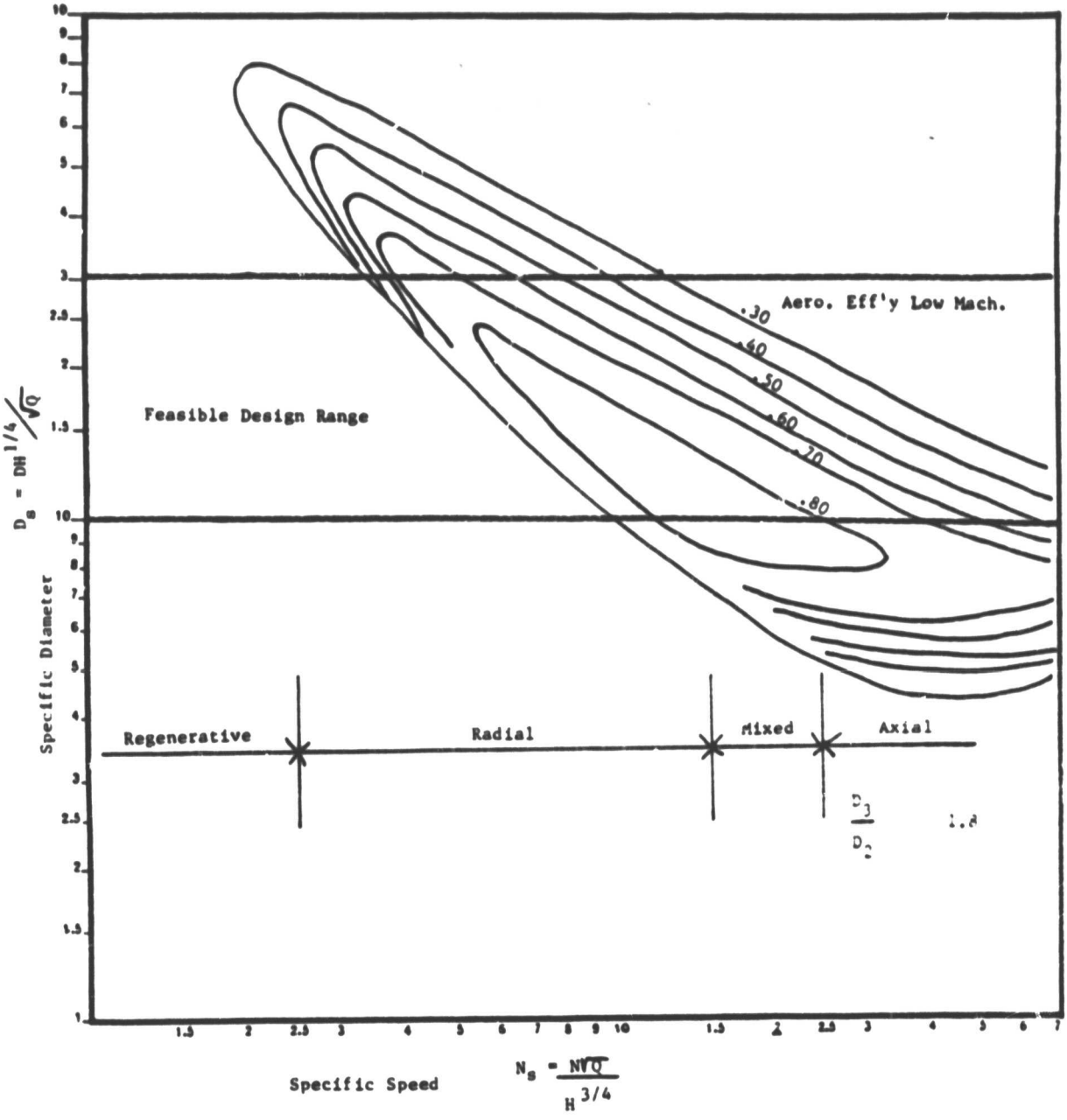


Fig. 2 Mod I Combustion Air Blower (Feasible Design Range)

TABLE 3

MOD I COMBUSTION AIR BLOWER PARAMETRIC STUDY

OBJECTIVE

- DEFINE TYPE OF BLOWER FOR APPLICATION
- DEFINE OPTIMUM BLOWER GEOMETRY
- DEFINE BLOWER SPEED

PARAMETERS INVESTIGATED

- BLOWER GEOMETRY
  - REGENERATIVE
  - CENTRIFUGAL
  - AXIAL OR MIXED
- OPTIMUM BLOWER PERFORMANCE
  - BLOWER SPEED
  - 3 DIMENSIONAL VANES
  - BACKWARD LEANING VANES
  - VANED DIFFUSER
  - IMPELLER DIAMETER
- CRITICAL SPEED
- BEARING LIFE - EVALUATE BEARINGS FOR:
  - FATIGUE
  - LUBRICANT STARVATION & LIFE
  - CAGE FAILURE

TABLE 4

MOD I COMBUSTION AIR BLOWER

PARAMETRIC STUDY

BLOWER GEOMETRY

GEOMETRY	ADVANTAGES	DISADVANTAGES
REGENERATIVE	• HIGH PRESSURE CAPABILITY • LOW SPEED • NO SURGE FLOW RESTRICTION	SIZE WEIGHT INEFFICIENT
AXIAL	• STEEPLY RISING PRESSURE CHARACTERISTICS • SMALL DIAMETER	• POOR SURGE MARGIN • INEFFICIENT • SIZE FOR IMPROVED EFFICIENCY (LONG DIFFUSOR) • ULTRA HIGH SPEED
CENTRIFUGAL	• GOOD SURGE MARGIN • COMPACT SIZE • GOOD EFFICIENCY	• HIGH SPEED
- STRAIGHT BLADES	• 65-75% EFFICIENCY	
- BACKWARD LEANING BLADES	• <1% CHANGE IN EFFICIENCY @ LOW PRESSURE RATIO	• PRODUCTION & DEVELOPMENT COST
- 3 DIMENSIONAL BLADES	• <1% CHANGE IN EFFICIENCY	• PRODUCTION & DEVELOPMENT COST
- DIFFUSER VANES	• 2% CHANGE IN EFFICIENCY	• PRODUCTION & DEVELOPMENT COST • EFFICIENCY IMPROVEMENT GOOD FOR ONLY A NARROW RANGE OF FLOW

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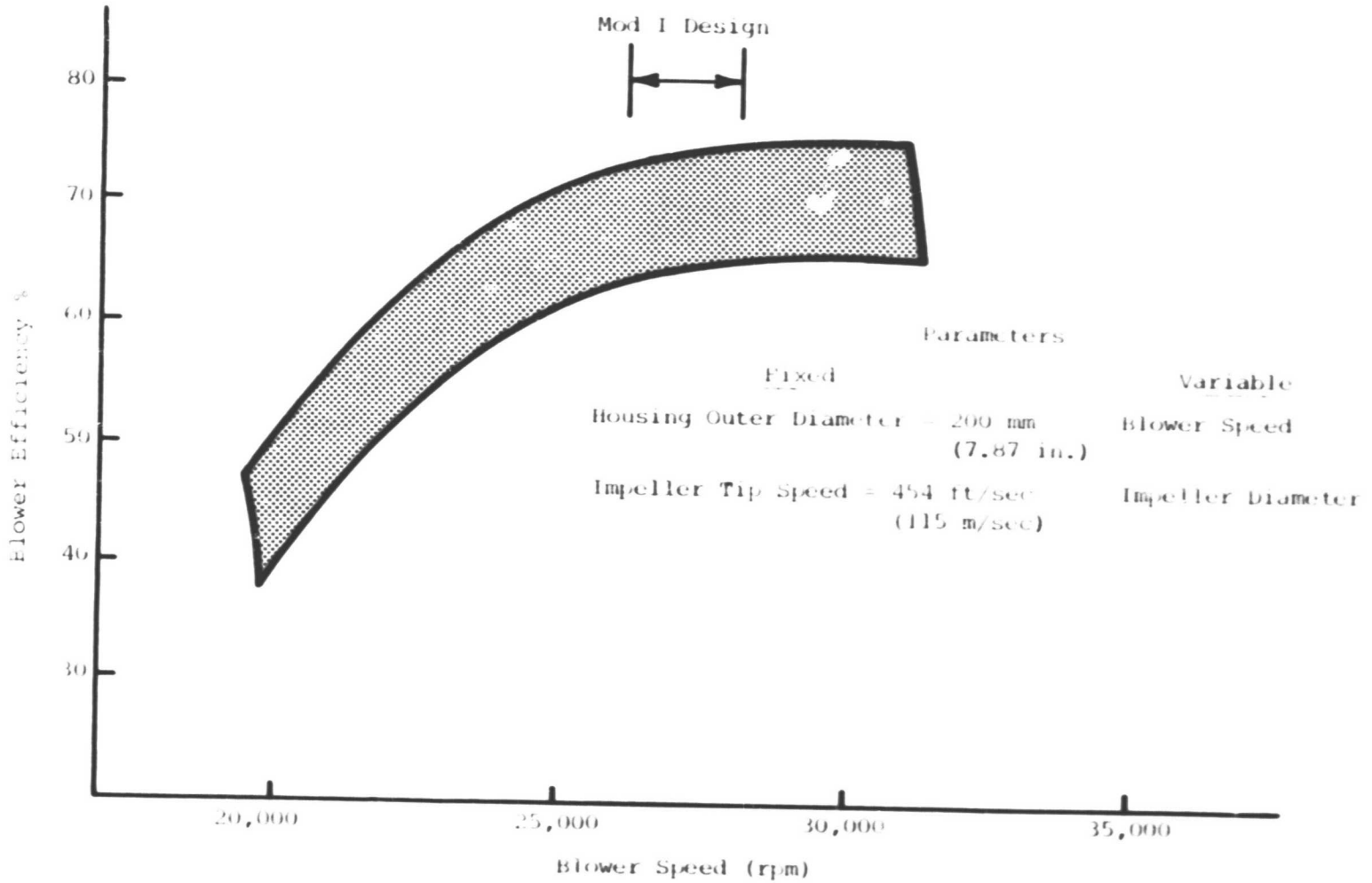


Figure 3 Mod I Combustion Air Blower Parametric Study

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TABLE 5

MOD 1 COMBUSTION AIR BLOWER

PARAMETRIC STUDY

CONCLUSIONS OF BLOWER GEOMETRY STUDY

- CENTRIFUGAL BLOWER BEST FOR AUTOMOTIVE APPLICATION
- LEAST COMPLICATED GEOMETRY MOST COST EFFECTIVE
- OPTIMUM SPEED FOR SIZE 26,000 - 28,000 RPM

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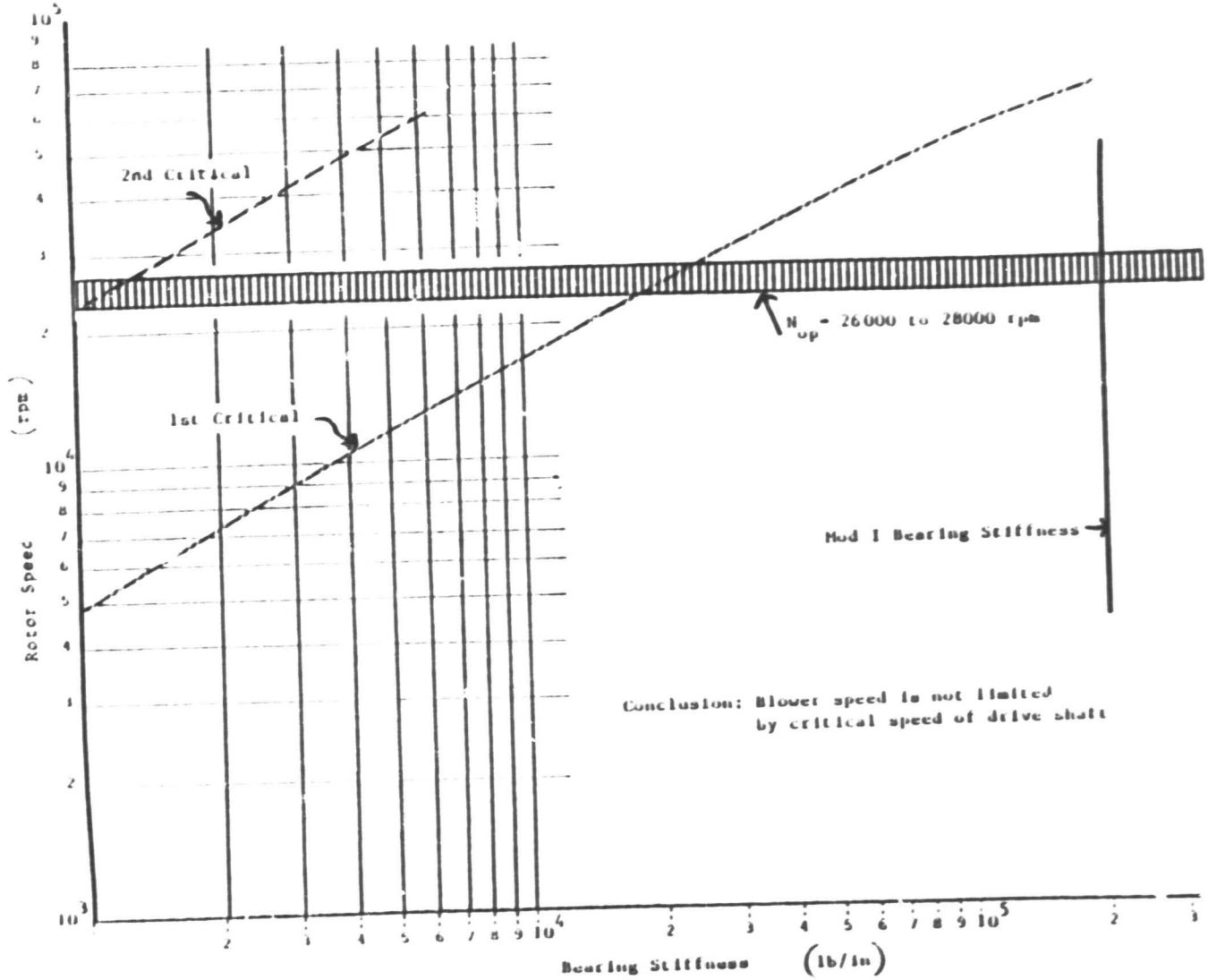


Fig. 4 Mod I Stirling Blower Critical Speed Map

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TABLE 6

MOD I COMBUSTION AIR BLOWER

BEARING LIFE STUDY

CONCLUSIONS

FATIGUE

- BEARINGS HAVE MORE THAN ADEQUATE LIFE WITH NORMAL CONDITIONS OF LUBRICATION AT 28,000 RPM
- BLOWER SPEED IS NOT LIMITED BY BEARING FATIGUE

LUBRICANT STARVATION AND LIFE

- NORMAL LUBRICATION CAN BE MAINTAINED AT BEARING ENVIRONMENTAL TEMPERATURE OF 150°F OR LESS
- 2100 HRS GREASE LIFE REQUIRES 1 REPLACEMENT DURING LIFE OF BLOWER
- BLOWER SPEED LIMITED BY GREASE LIFE

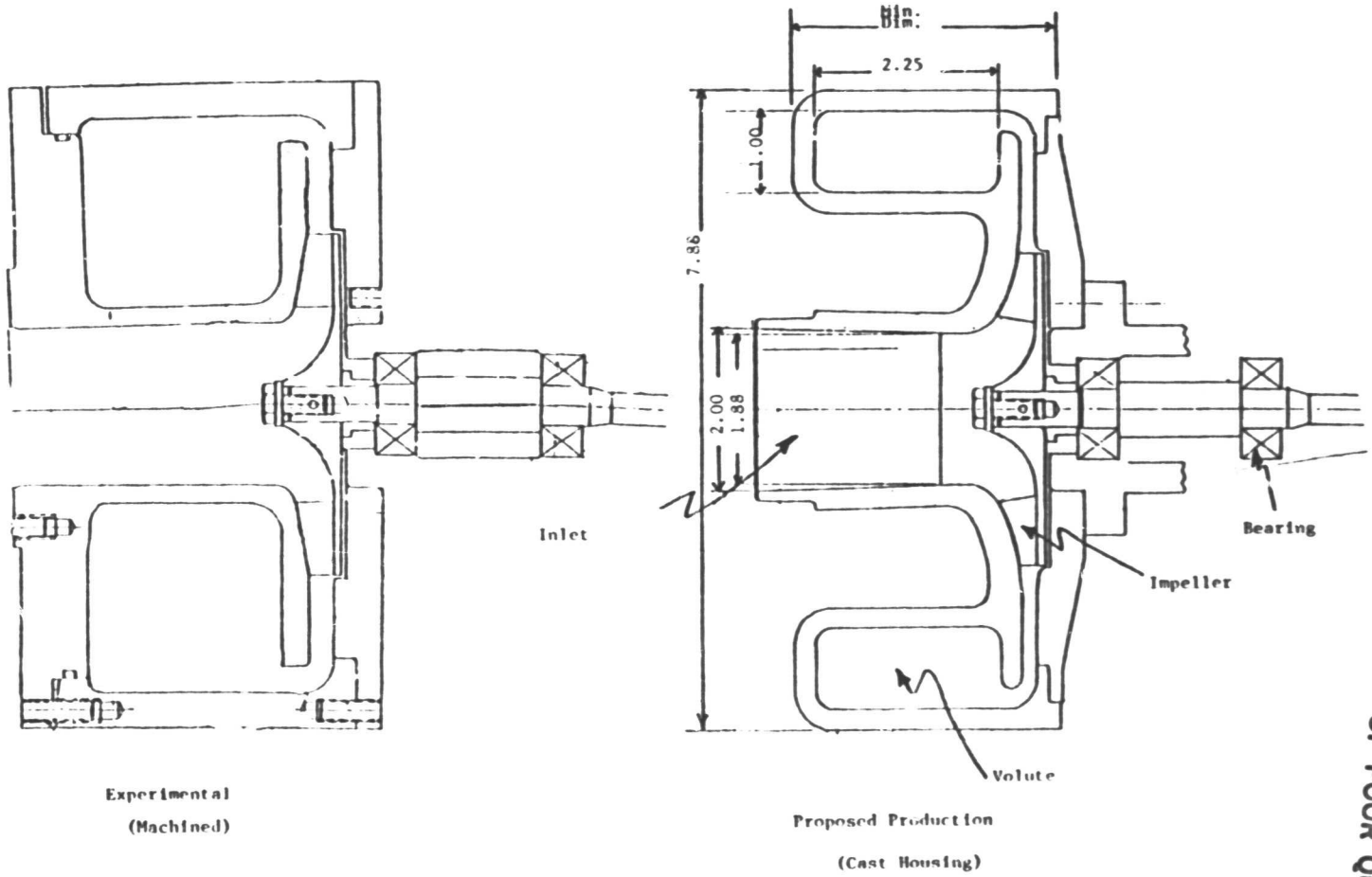
CAGE FAILURE

- SELECTED A BARDON 201 SST x 1 BEARING WITH PHENOLIC AND ALUMINUM CAGES FOR IMPROVED CAGE LIFE

RECOMMENDATIONS

- USE EXXON ANDOK "C" OR TEXACO MULTIFAK - ALL PURPOSE GREASE
- THE ENVIRONMENTAL TEMPERATURE OF THE BEARINGS SHOULD BE MONITORED IN TESTS

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Mod I Combustion Blower Air Blower Designs

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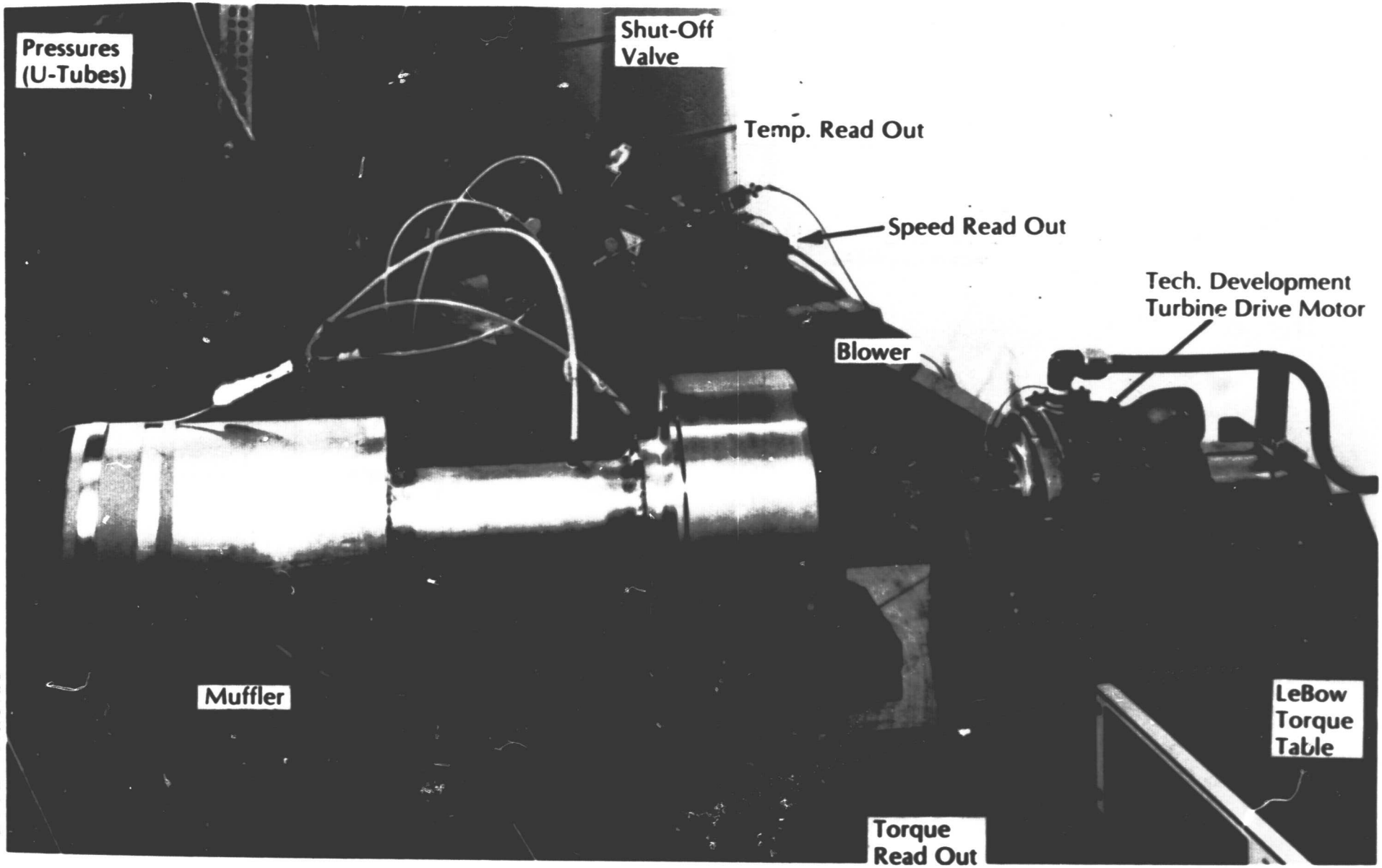


Fig. 6 Mod I Combustion Air Blower Experimental Test Rig

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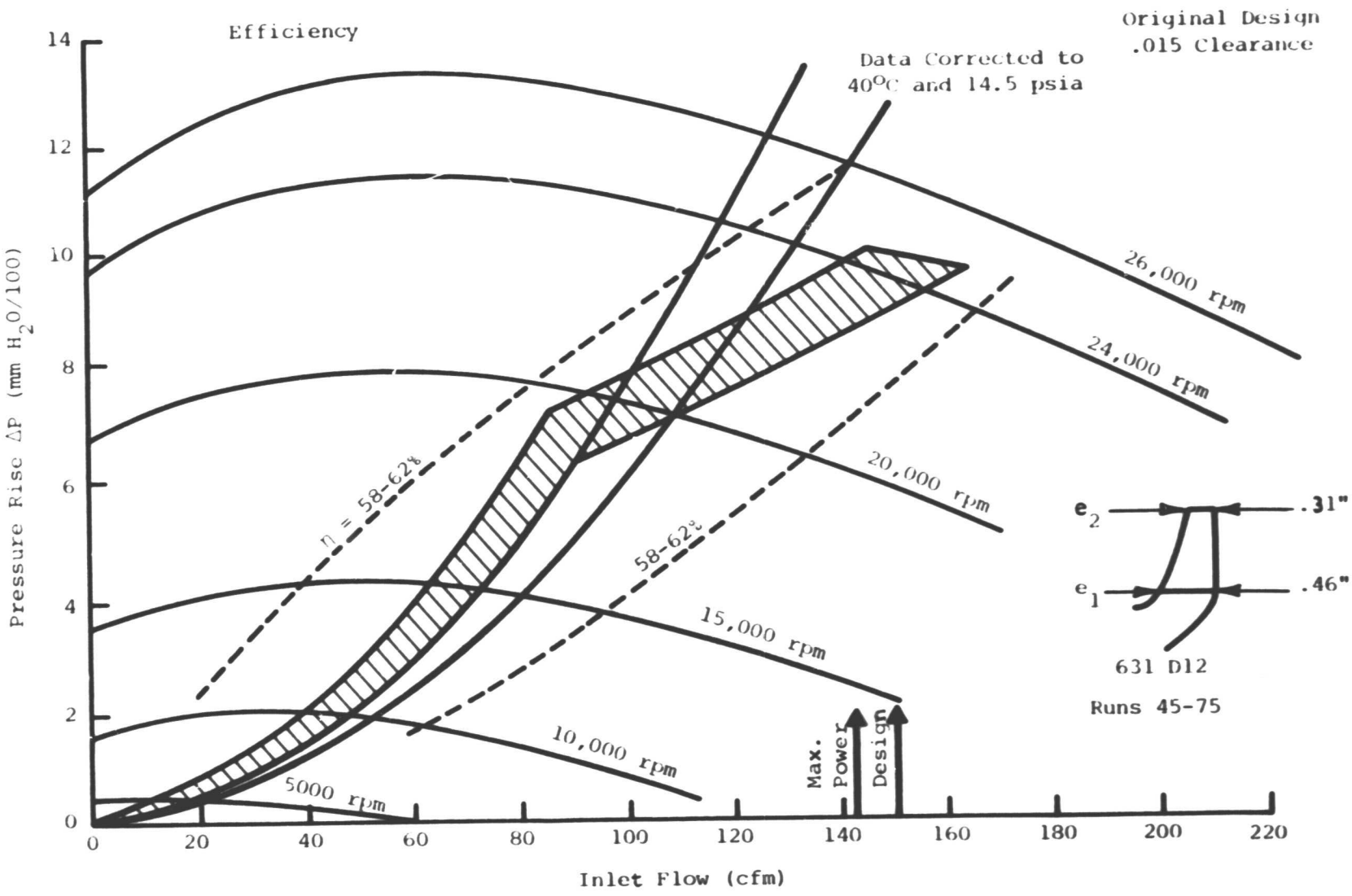


Figure 7 Mod I Combustion Air Blower Test Results

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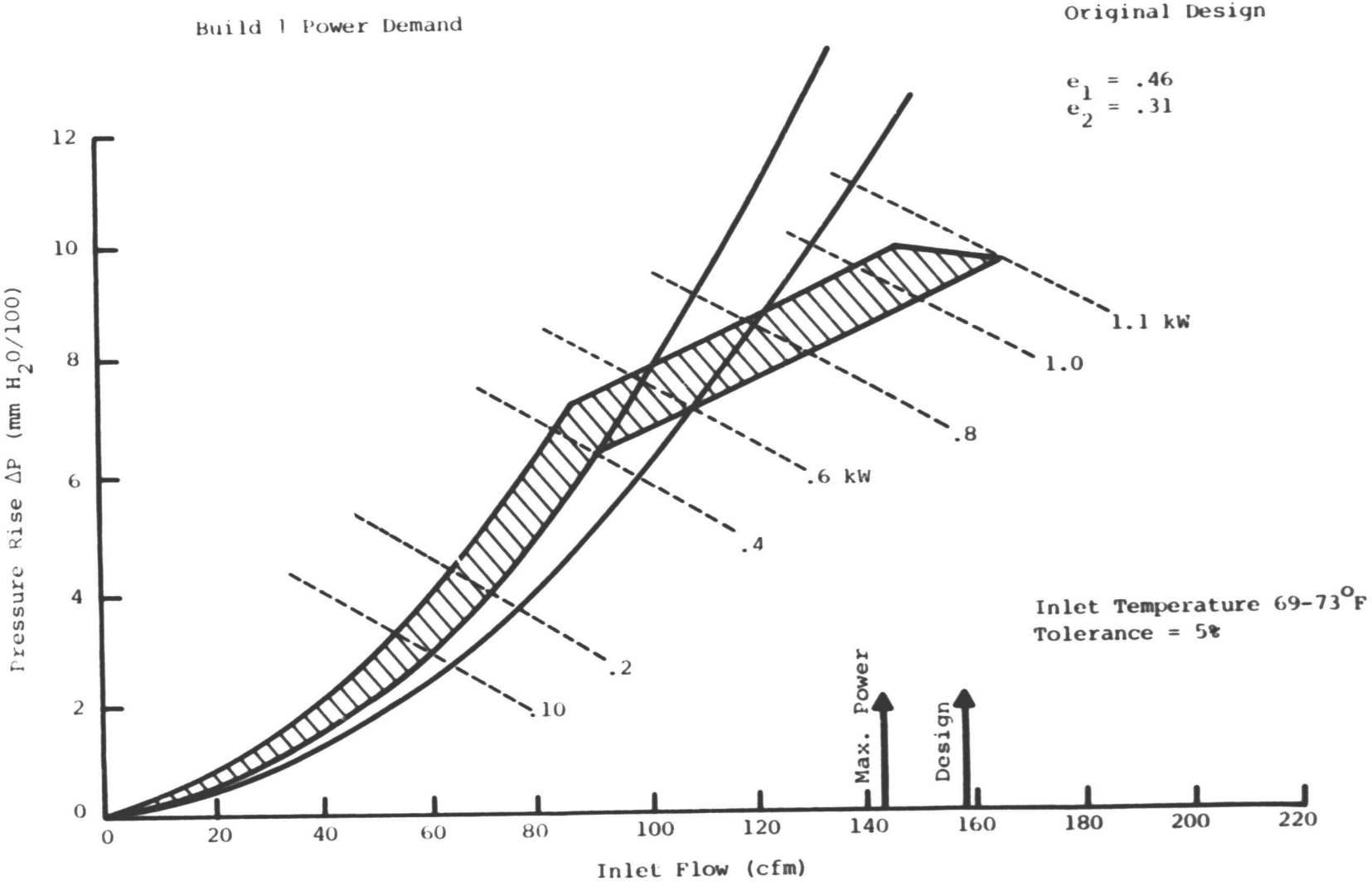
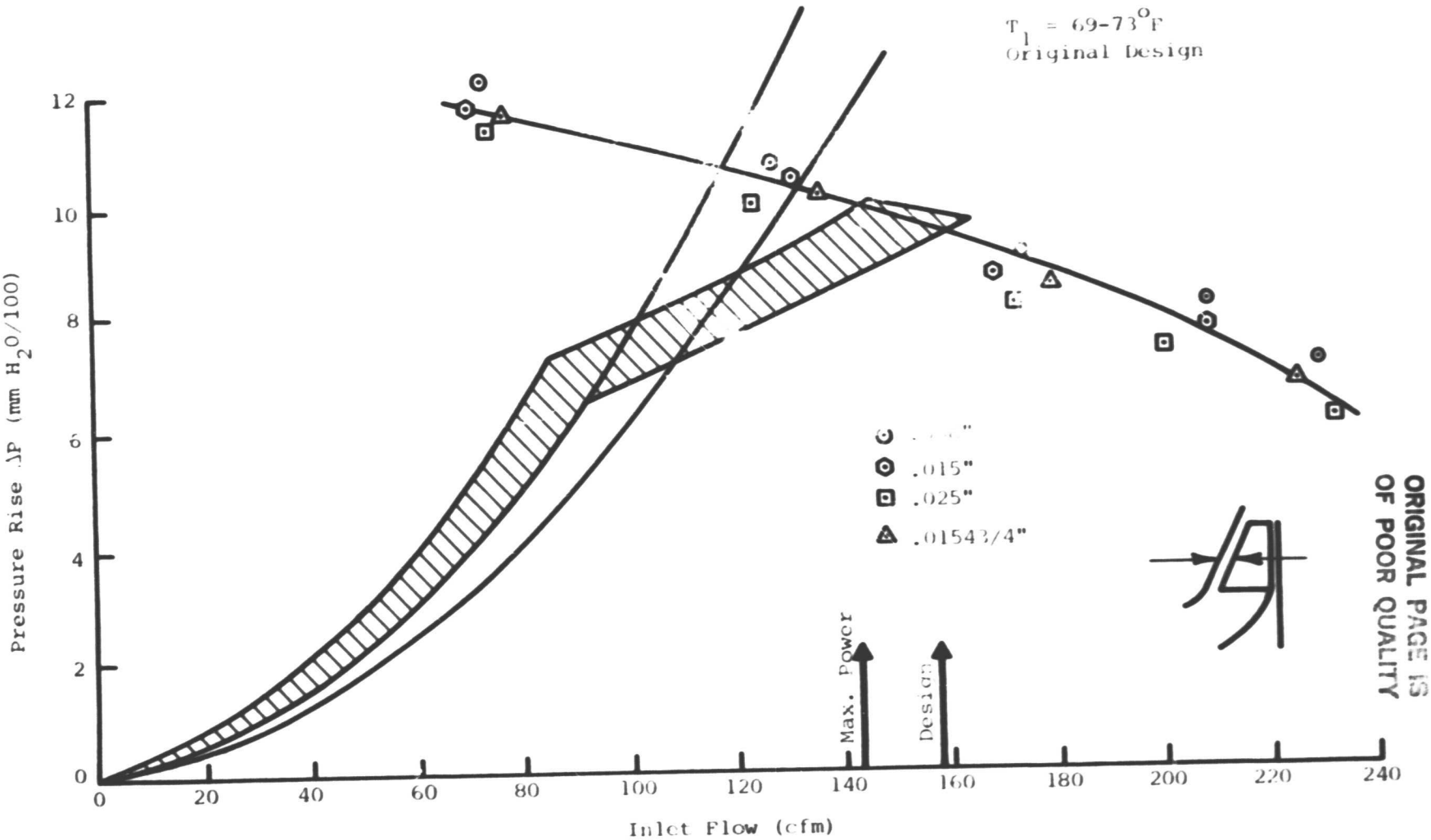


Figure 8 Mod I Combustion Air Blower Test Results

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Figure 9 Mod I Combustion Air Blower Test Results
Effect of Vane Clearance and Volute Volume at 24,000 rpm

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Power consumption versus airflow for different control systems tested with Sulzer 21500 blower

Document No | Page No

Graph No | Appendix No App 9

Author(s) ST | Date 79 02 21

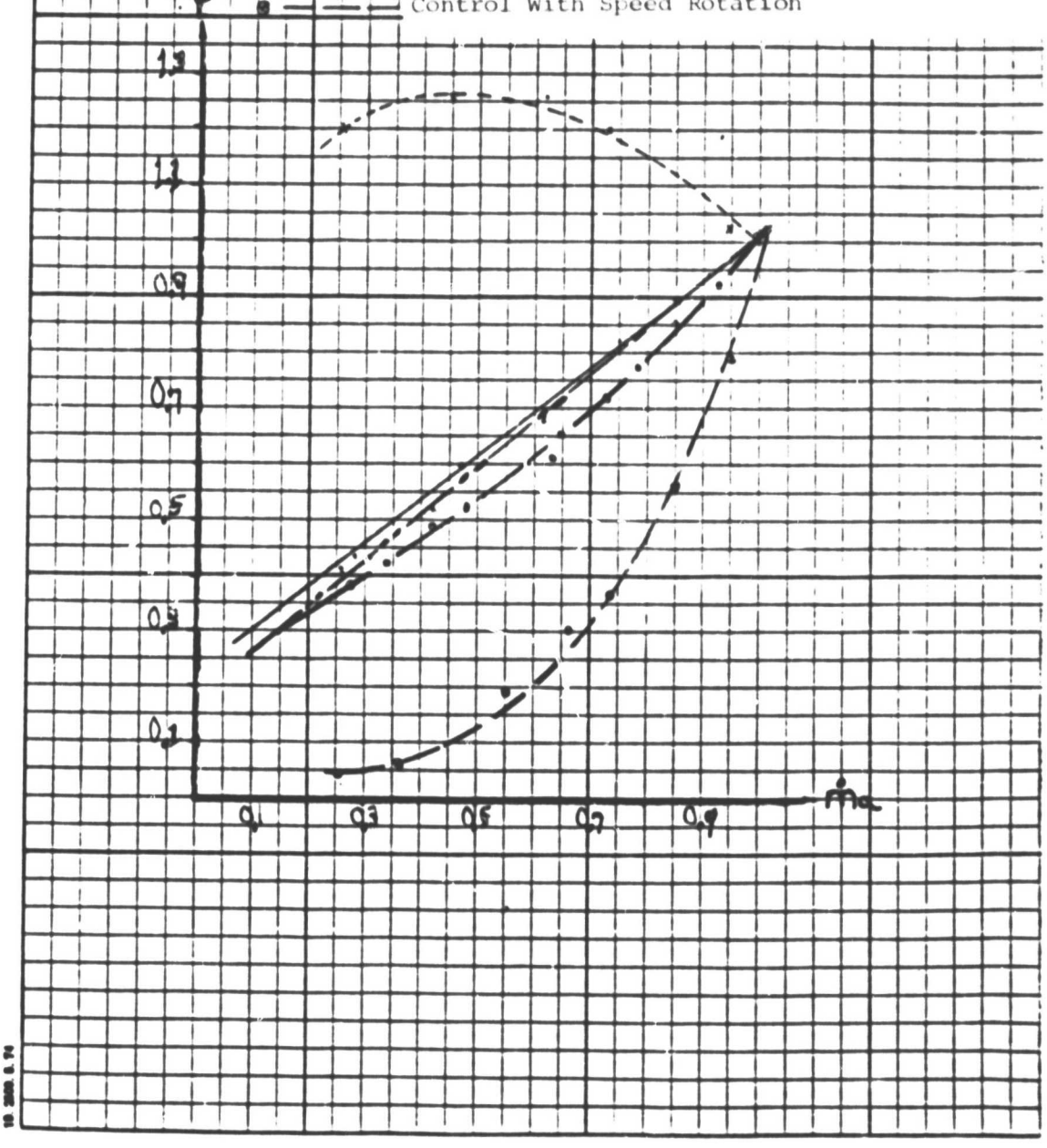
Control Valve on the Outlet Size at 16,000 rpm

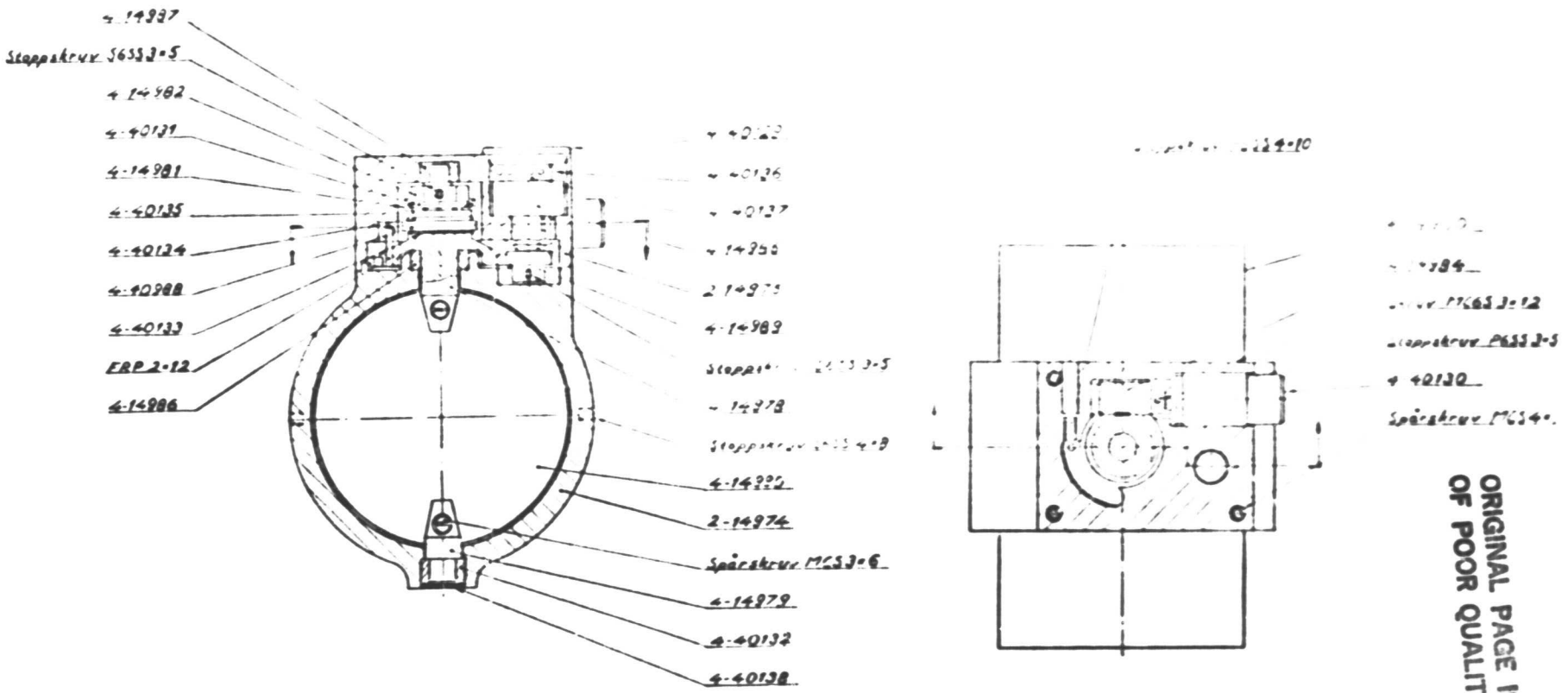
Bypass Air Throttle on the Outlet Pipe at 16,000 rpm

Air Throttle on the Inlet Pipe at 16,000 rpm

Guide Vanes on the Inlet Pipe at 16,000 rpm

Control With Speed Rotation





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Fig. 12 Air Throttle

ISO				
			1:1	
UNITED STIRLING	Air throttle			2-14888

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C. & W. Berges
Maschinenfabrik
5277 Marienheide
☎ 02264/6057 Tx 884116

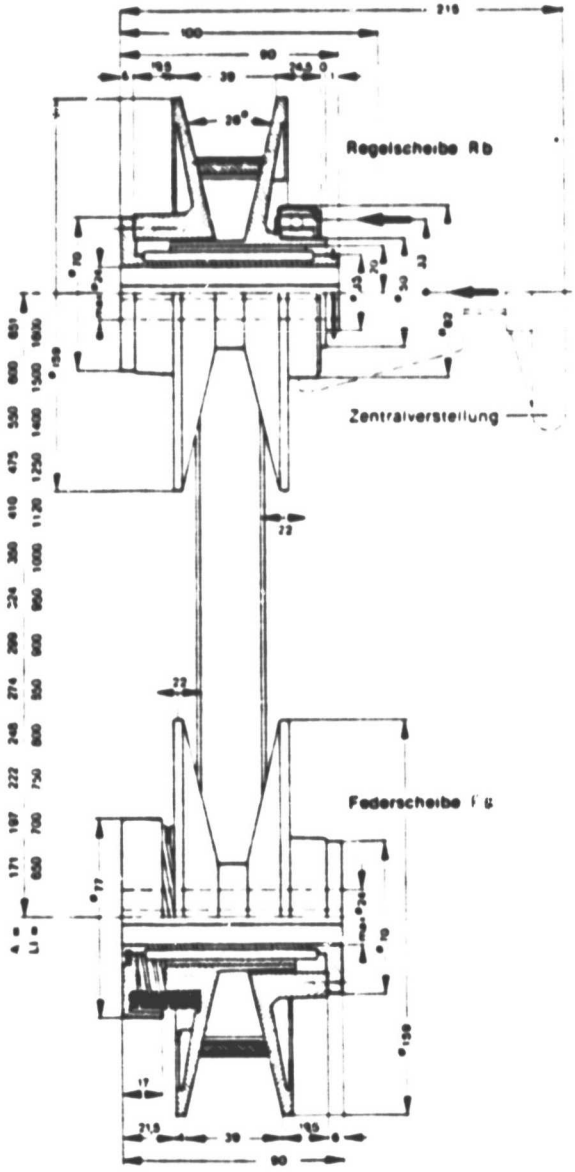
Regelscheibensatz RF 150b

für Riemenprofil CW 28
(Breitkeilriemen)

Regelbereich 1 : 6,5
Motorleistung 2 KW

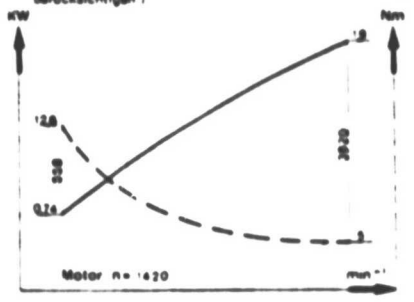
Steuerdruck	= 800 N in ► Richtung
Steuerhub	= 22 mm
Gewicht	= 3,9 kg
Schwungmasse	= A = 0,011 / F = 0,013 kpm2
Keilnuten	= nach DIN 6885 Bl. 1 (3)
Bohrungen	= nach ISO H 7
max. Riemen-Innenlauf- ⌀	= 144
min. Riemen-Innenlauf- ⌀	= 48

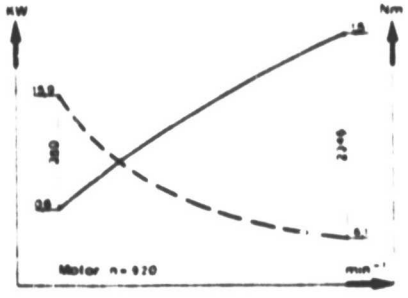
Sofern der Gesamtregelbereich nicht benötigt wird, können abweichend von der Achsabstandtabelle längere Keilriemen verwendet werden, um die kleinsten Laufdurchmesser der Regelscheiben zu vergrößern. Die Leistungsübertragung wird dann gleichzeitig bei n_{min} und n_{max} gesteigert.
Achtung: Es entsteht in diesem Falle ein Versatzmaß zwischen den beiden Scheibenenden. Bitte fragen Sie bei solchen Anwendungsfällen bei uns an. Für Einsatz in explosionsgefährdeten Räumen sind elektrische leitfähige Riemen erforderlich.

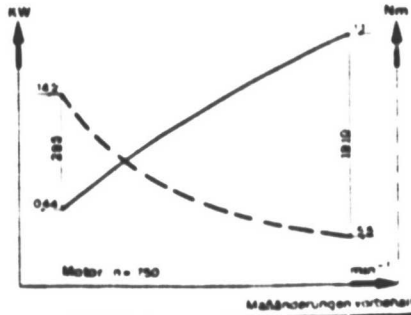


Leistungen - Drehzahlen - Drehmomente

(Bei stoßbelastetem Betrieb die üblichen Sicherheitszuschläge berücksichtigen.)







Maßänderungen vorbehalten!

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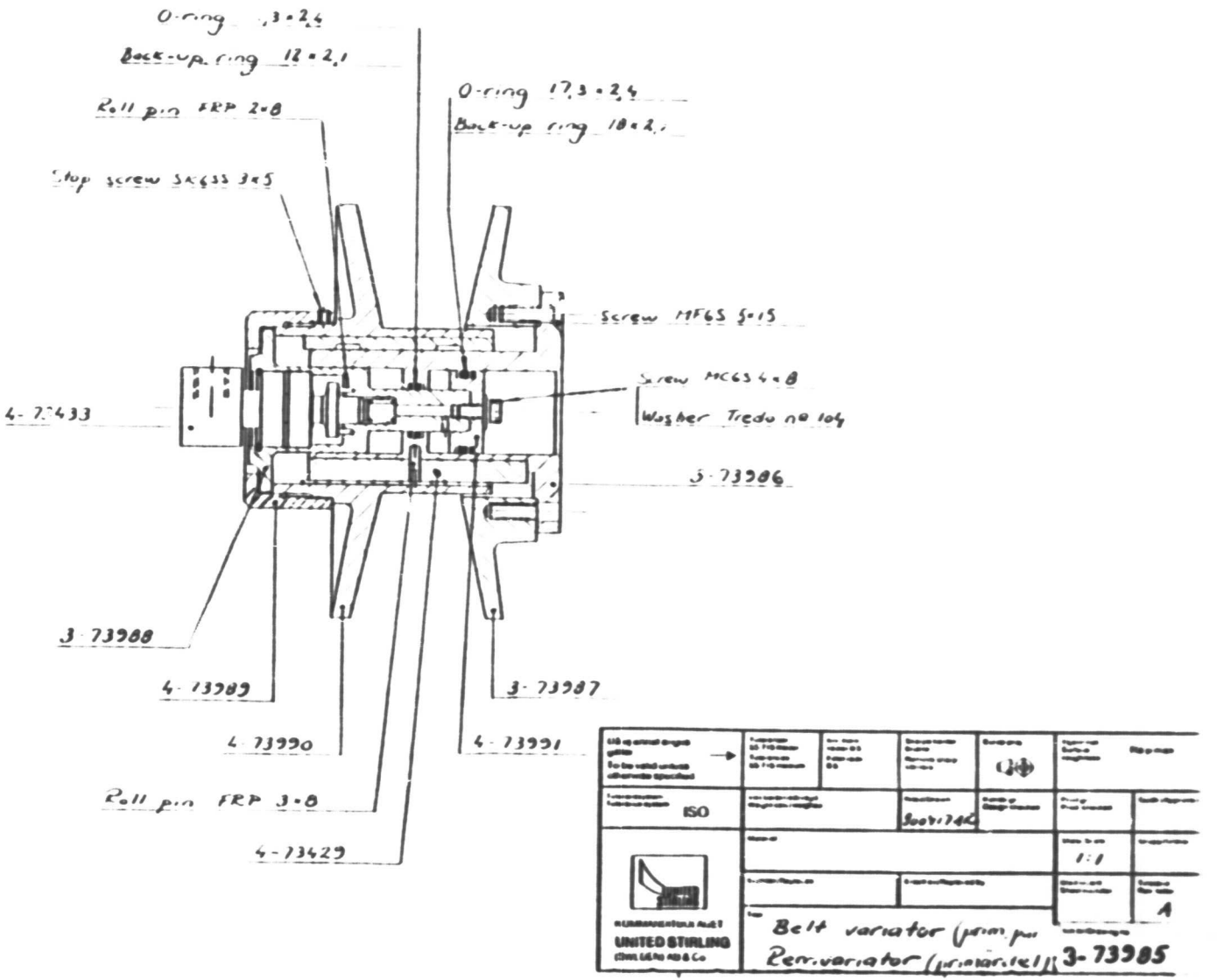
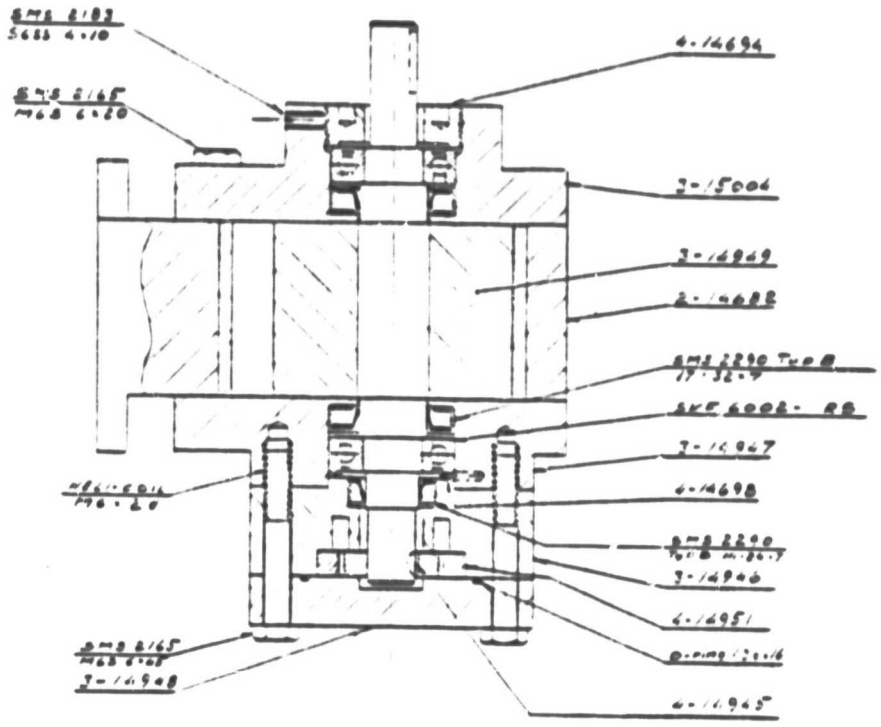
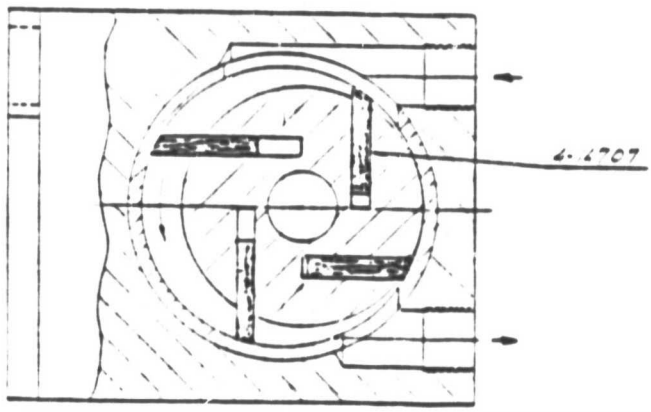


Fig. 14 Belt Variator

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UNITED STIRLING

KOMPRESSOR - OLJEPUMP

3-14950

Fig. 15 Oil Pump Compressor

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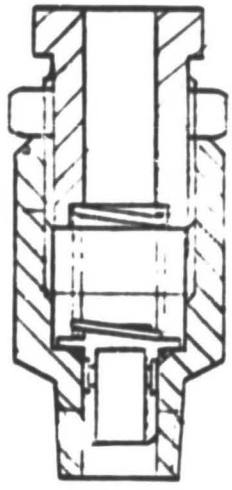
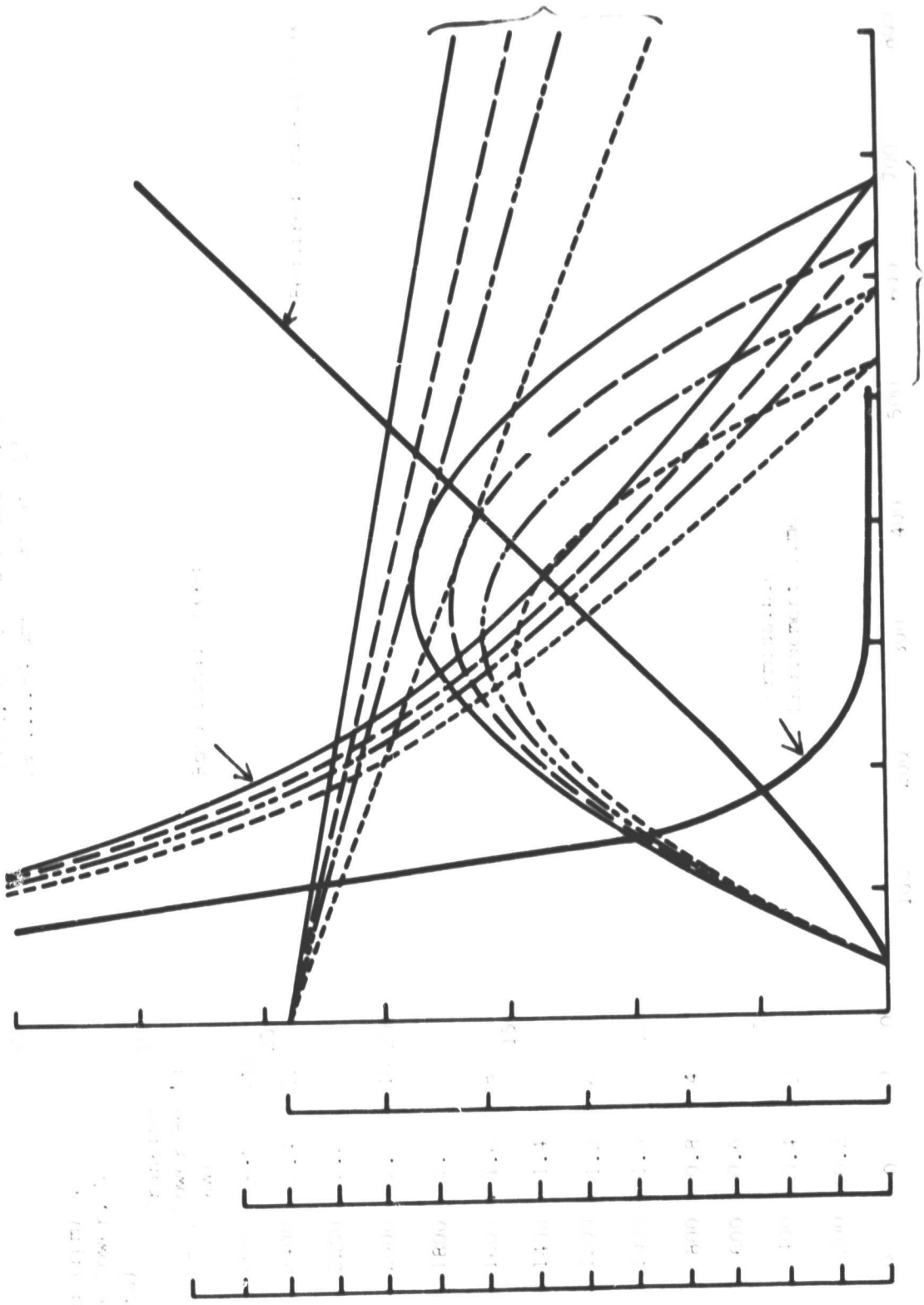


Fig. 16 Check-Valve

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Entwurfs-Zeichnungs Nr	Typ/Aufschrift	Bestell-Nr	Gew ca kg	Kurvenblatt	Bemerkungen
KM74/3B27	JRA-12V 1000W 2800U/min		8.2	EDA/MM-K11	mit Simmerring
KM74/3B4	JRA-12V 1000W 2800U/min	0 136 350 003	8.2	EDA/MM-K11	ohne Simmerring

Gleichstrom-Motor

Zubehör (wird nur mitgeliefert)
1 Federring 2 916 690 010
1 Sechskantmutter 2 915 041 004
1 Scheibenfeder 1 900 023 008

Sonderzubehör (muß besonders bestellt werden)
2 Schutzkappen 1339 999 020

KM74/3B4A

Angebotszeichnung

KM74/3B4A

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FEATURES

- LINEARITY OF SIGNALS VS. AIRFLOW
- REPEATABLE ACCURACY ±0.1%
- WIDE DYNAMIC RANGE ±1% @ 40:1; ±3% @ 50:1
- FAST RESPONSE
  10-35ms DEPENDING UPON STEP SIZE
- PROVEN AUTOMOTIVE DURABILITY
  ½ MILLION MILES
- EXCELLENT ELECTRICAL NOISE IMMUNITY
- SIMPLICITY AND LOW COST $6 in 50,000 PRODUCTION

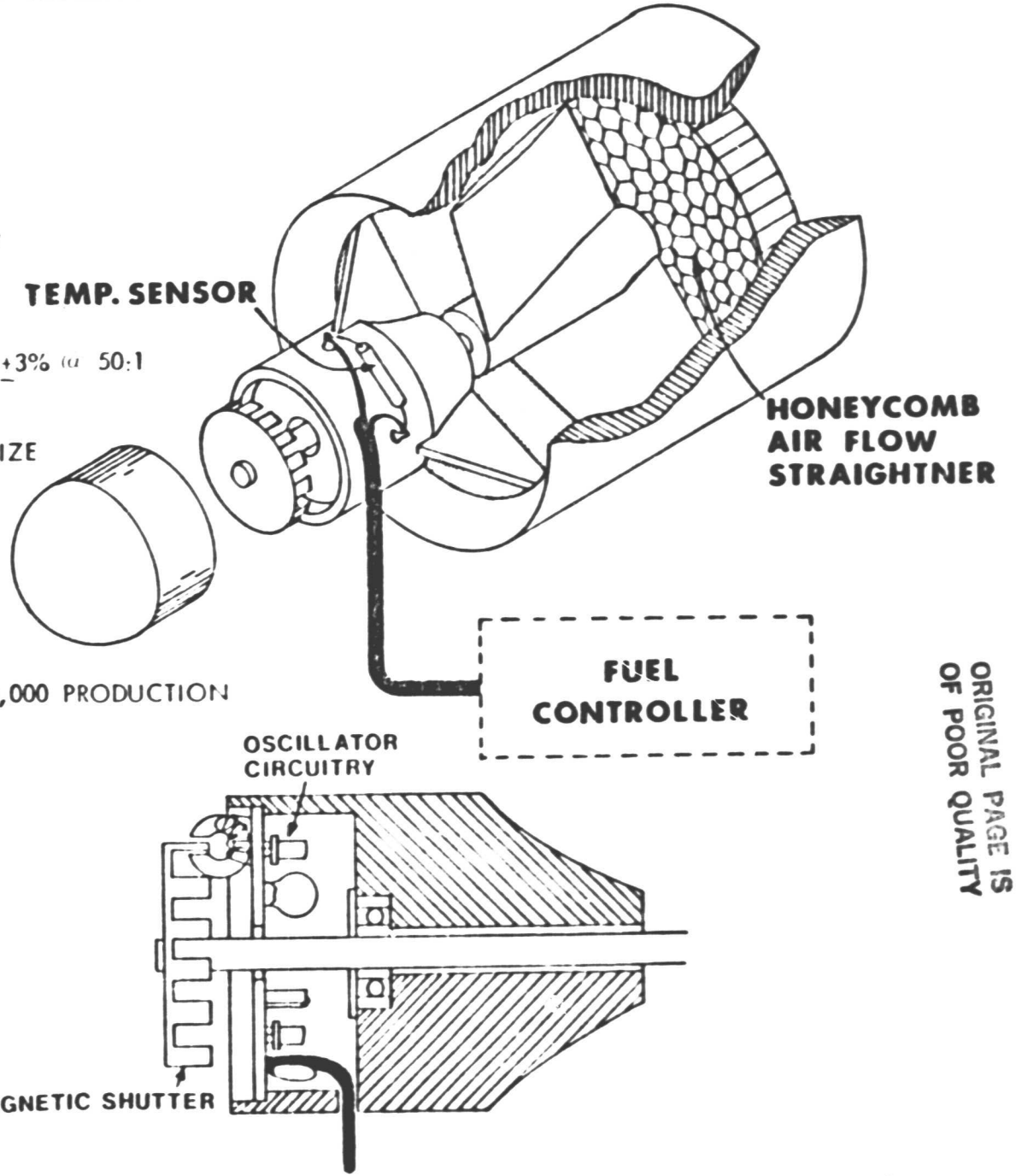
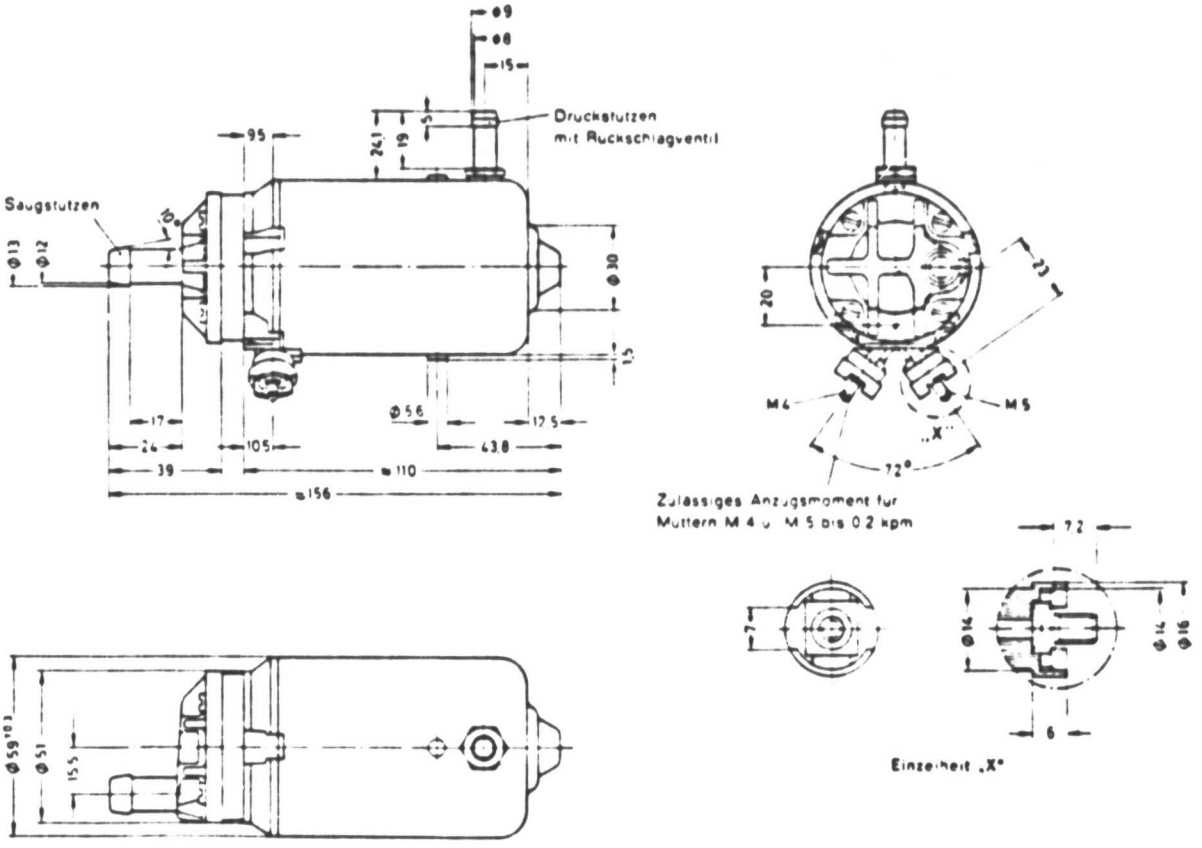


Fig. 19 Airflow Transducer

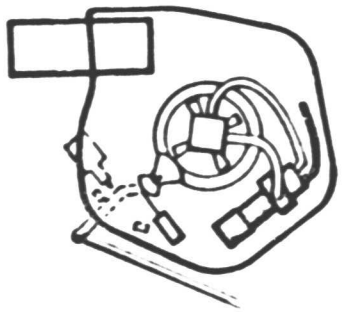
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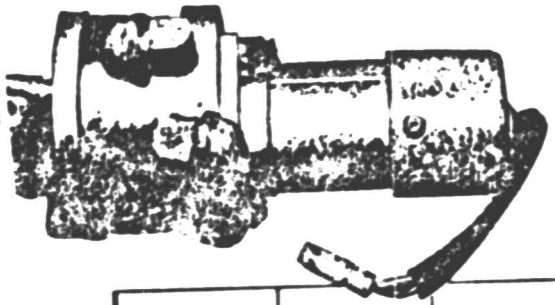
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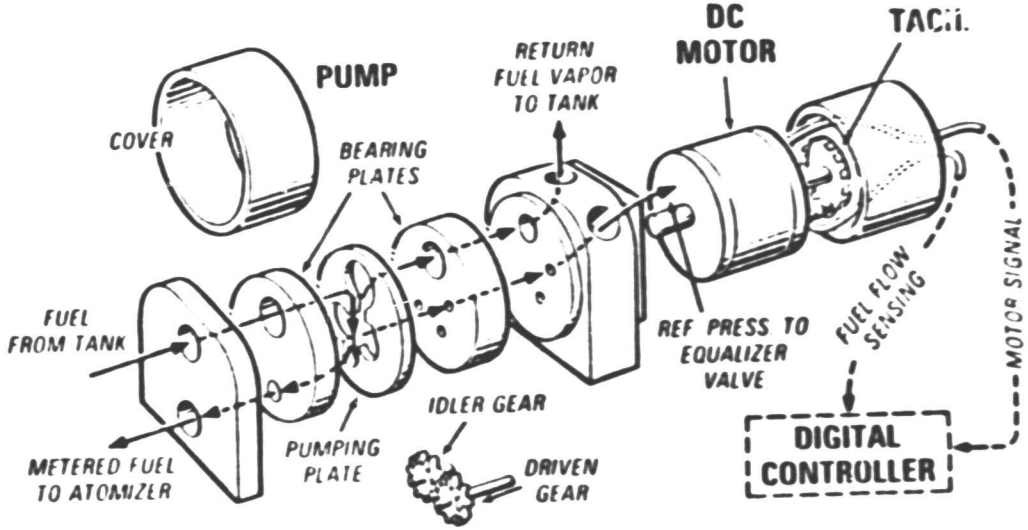
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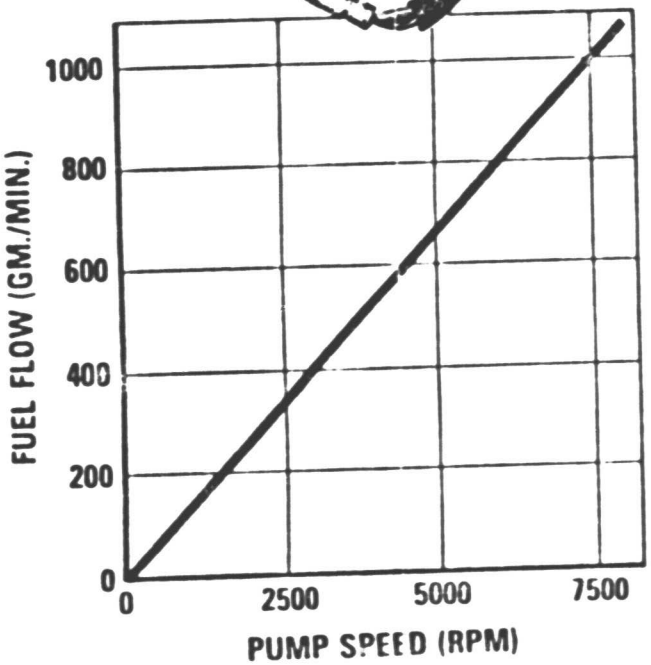


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FEATURES:

- **LINEARITY OF SIGNALS Vs FUEL FLOW** ±1% @ 50:1 range
- **ACCURATE METERING**
- **FAST RESPONSE** 5 millisec.
- **LOW POWER CONSUMPTION** ½ Amp aver. ; 2 Amp max.
- **HIGH DURABILITY** All metallic construction
- **MODULAR CONSTRUCTION**

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Fig. 21 System Component Fuel Metering Pump

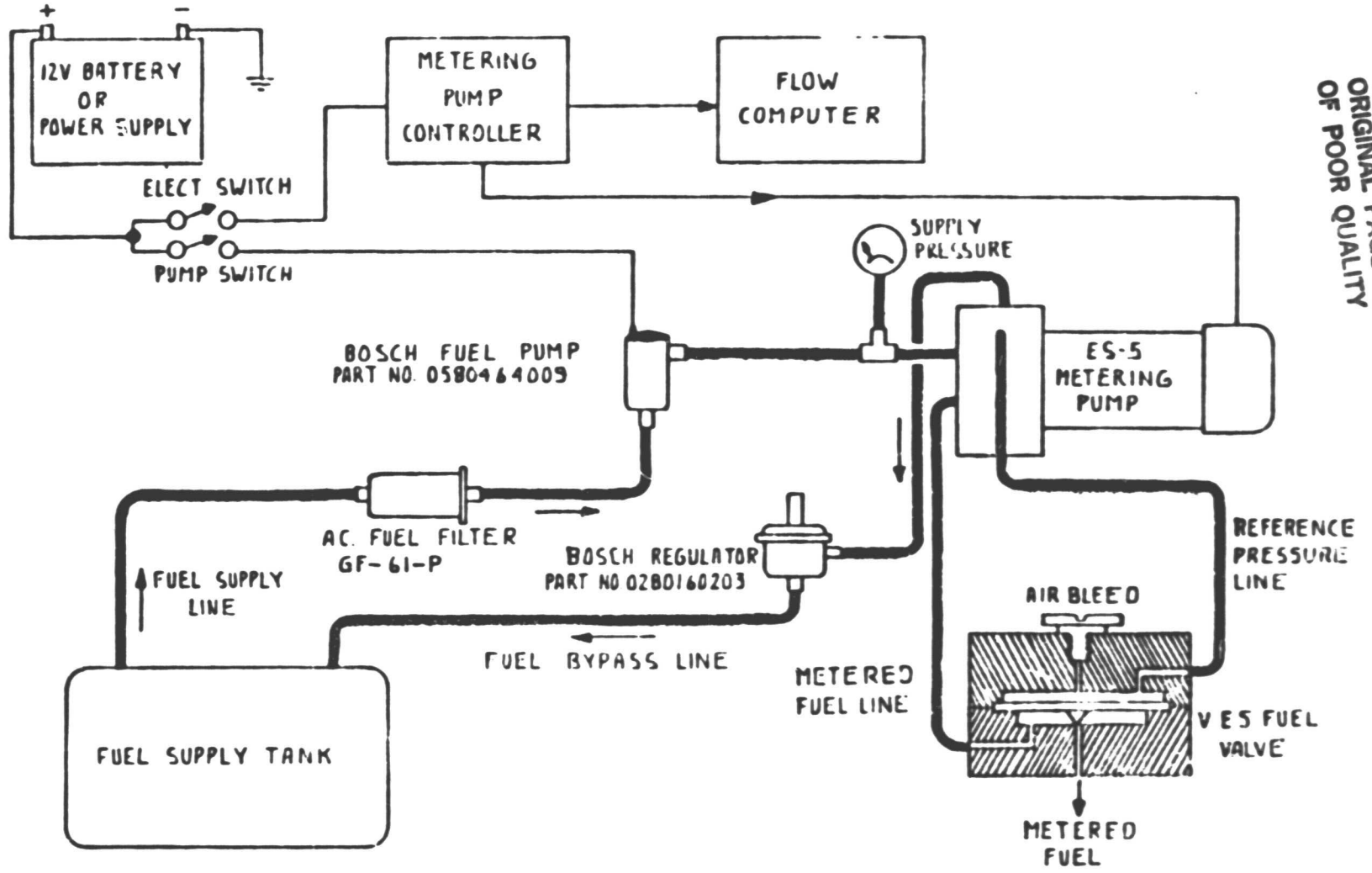


Fig. 22 Electrosonic-5 Fuel Flow Diagram

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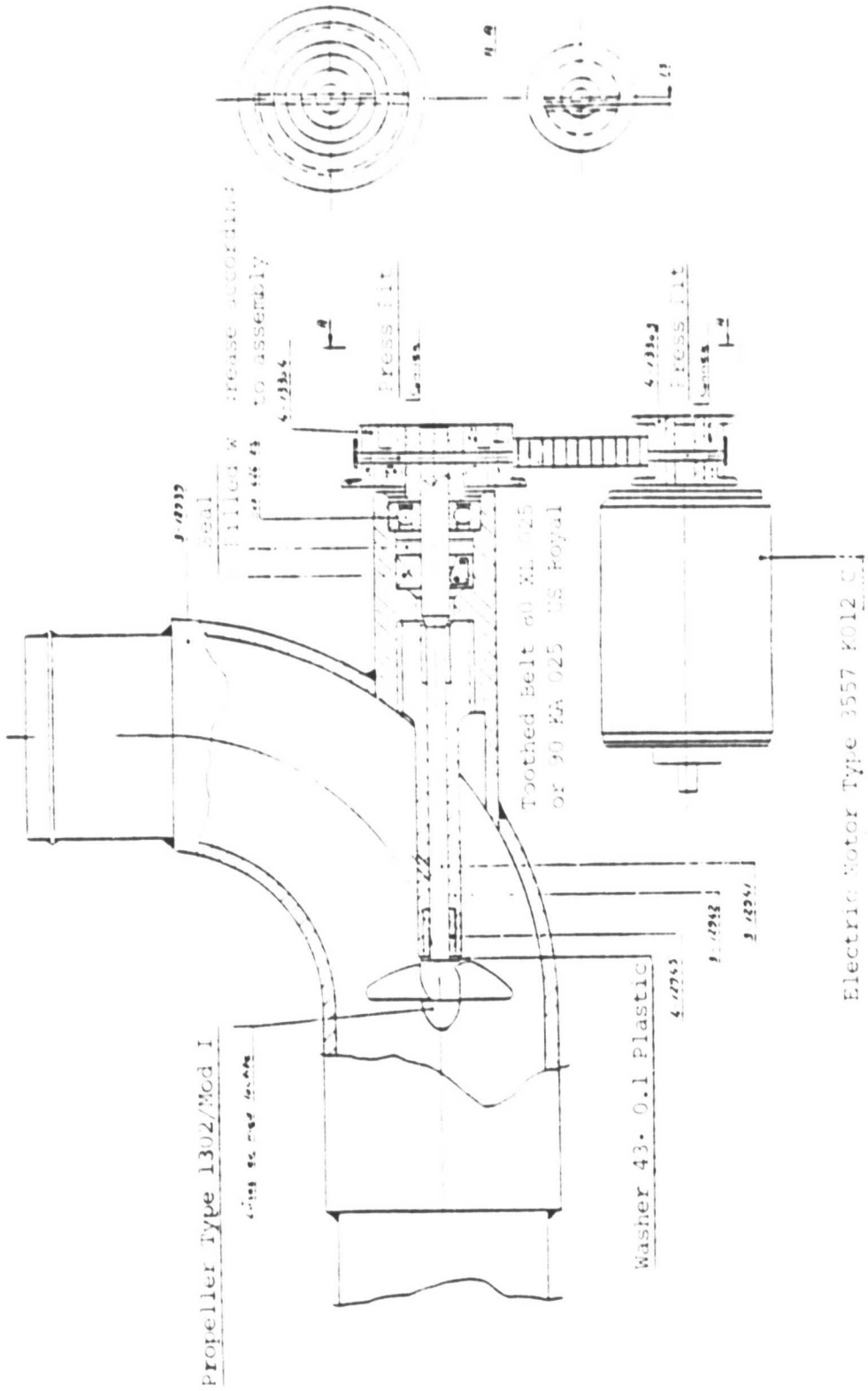


Fig. 23 After Cooling Pump Assembly

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STARTER MOTOR

Conventional internal combustion engines require a starter to provide torque to overcome static friction and dynamic drag of the engine and to raise engine power to a level which is sufficient to initiate combustion and sustain operation. As illustrated in Figure 1, these same considerations are complicated in the Stirling engine by the additional power generated by the engine once the piston rings have seated. The combined power generated by the starter and the engine accelerate the engine. Piston rings, which provide an optimum seal between the hot and cold spaces, allow the engine to significantly assist in the starting sequence. Sufficient power is generated by the engine so that, with termination of the starter motor at a relatively low rpm, the engine is self-sustaining. Deteriorated piston rings, however, require continued cranking of the starter motor until the starter motor contribution is almost negligible before termination.

Insufficient knowledge of the engine's contribution requires qualifying the Mod I design by comparing it with P-40 starting experience and by limiting the analysis to a comparison of starter power to engine drag.

Figure 2 illustrates the engine drag starting requirements with ESSO 20W 40 oil for summer temperatures, and a low viscosity oil, Mobil SHC 824, for winter (cold day, to -29°C) temperatures. The engine's drag includes piston ring, cross head and seal friction, bearing and gear losses, and oil pump and water pump power extraction. For the Stirling engine, these requirements demand an engine starter motor power that exceeds motors which are commercially available and that fit the Spirit transmission.

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Table 1 tabulates the options investigated to meet the starter requirements. All were considered to be either an excessive compromise to the engine/vehicle system or too expensive and time consuming. The following items were considered:

- The Mod I engine is primarily a laboratory experimental engine with only the engine devoted to vehicle operation.
- The higher power required is principally due to higher piston ring friction and engine drag, items being addressed under component development activity.
- The starter motor requirement is not well understood without considering the Stirling engine's self-assistance characteristic.

The selected starter motor system, a Chrysler 1.3 kW (1.8 hp) [P/N 4111860] with 12 volts for the vehicle and with 24 volts for the laboratory engines, is compared to P-40 experience in the AMG Spirit, shown in Figure 3. Figure 3 illustrates that for summer temperatures with 20W 40 oil, the Mod I engine starting capability is similar to that of the Spirit. Figure 4 shows that replacing the oil (recommended for temperatures below 10°C) with a low viscosity oil such as Mobil SHC 824, starting is possible to 0°C. The addition of a supplementary 24 volt system, as shown in Figure 5, will provide the starting capability to the required -29°C. The modified wiring of the starter system for 12 or 24 volts, requires severing a bus bar in the starter solenoid to limit solenoid voltage to 12 volts to prevent damage. The starter motor is capable of accepting 24 volts without damage as long as prolonged cranking is avoided.

Mod I engine testing should include the acquisition of data to better define

the engine's contribution to the starting sequence and to be prepared for a more accurate definition of the Mod II starter requirements. Component development directed at the Mod II should include continued development of reduced engine drag during starting and reduced blow-by piston rings.

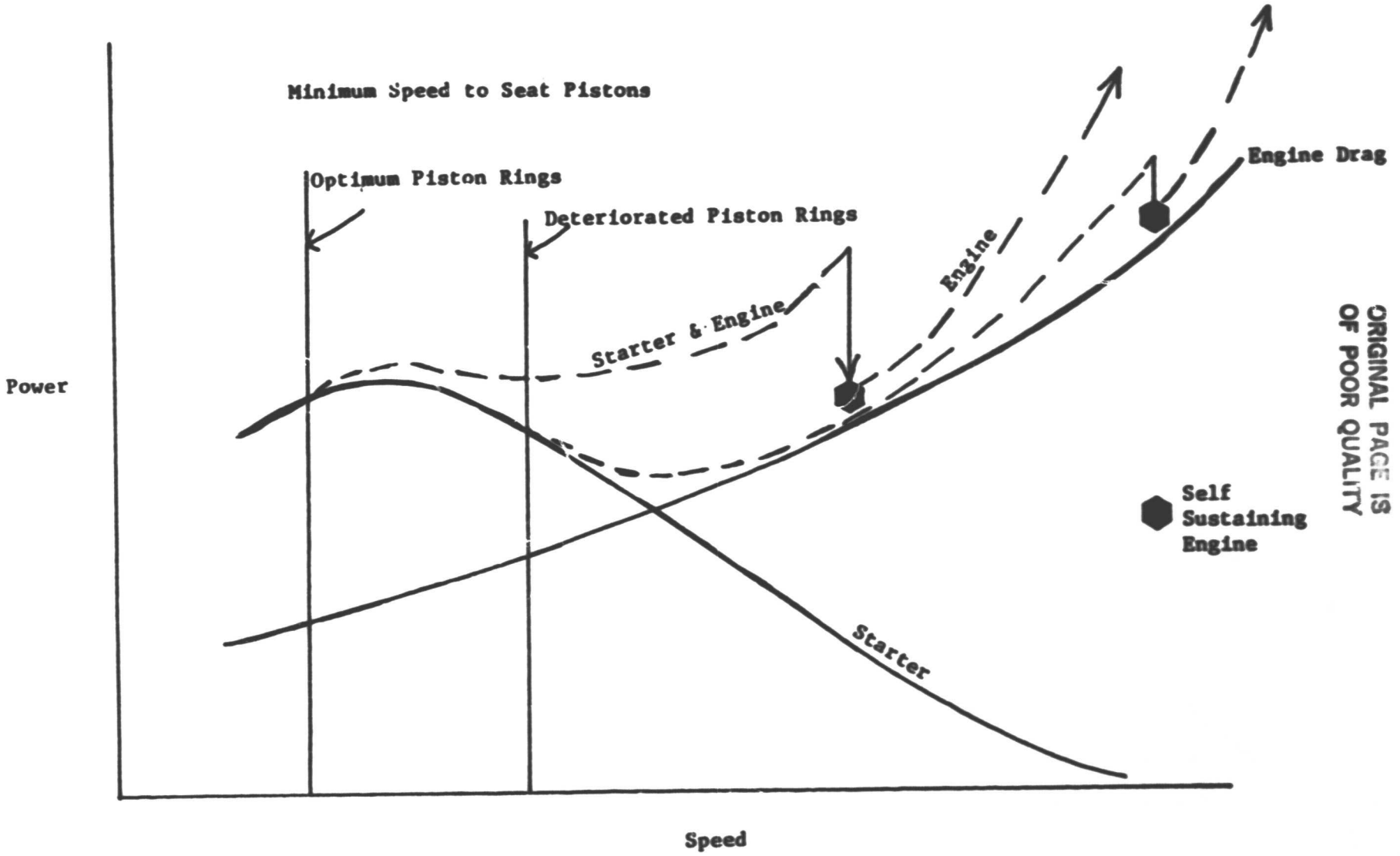
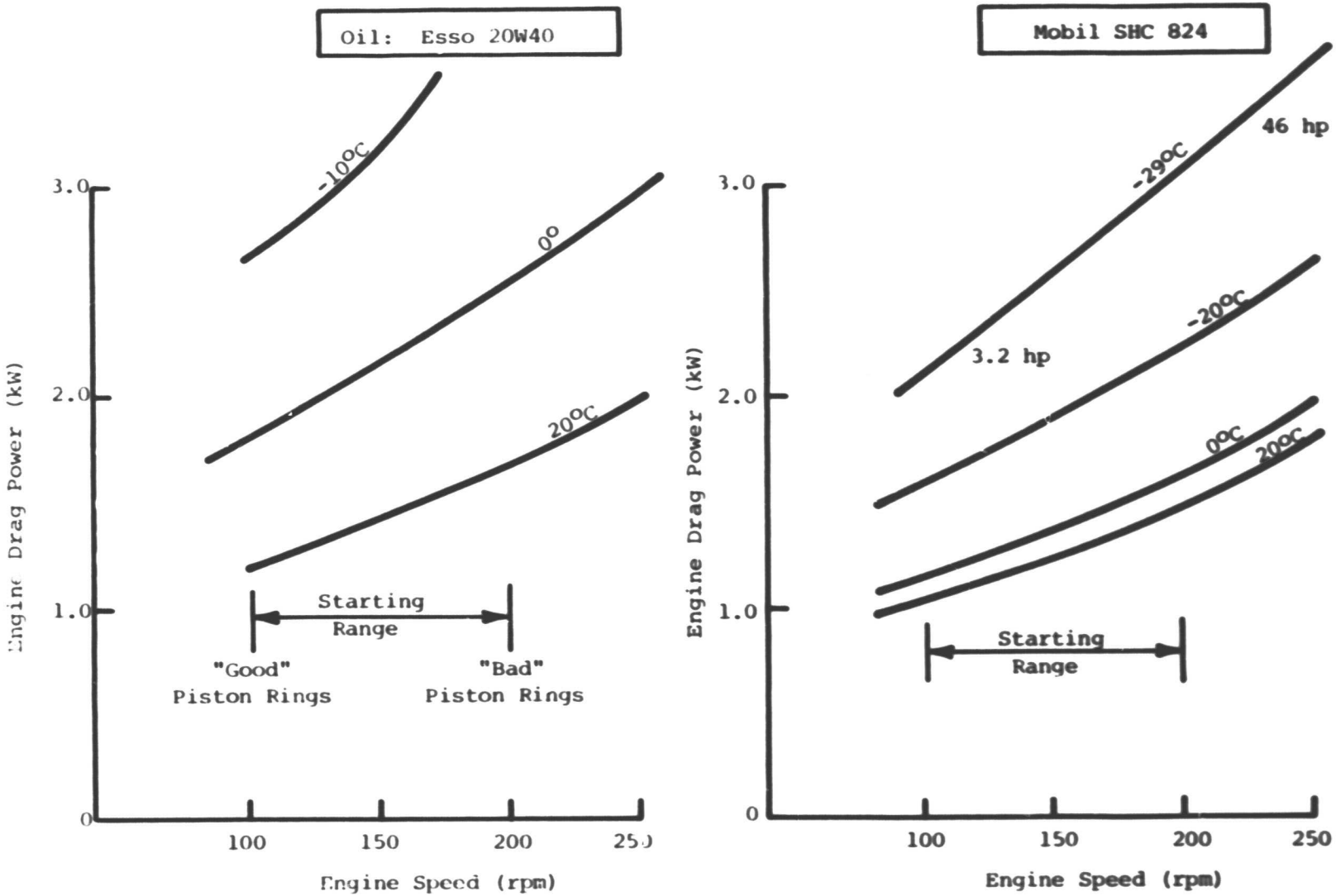


Fig. 1 Starting Power Characteristics

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-29°C Starting Temperature, Automotive Standard



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Figure 2 Mod I Starter Requirements

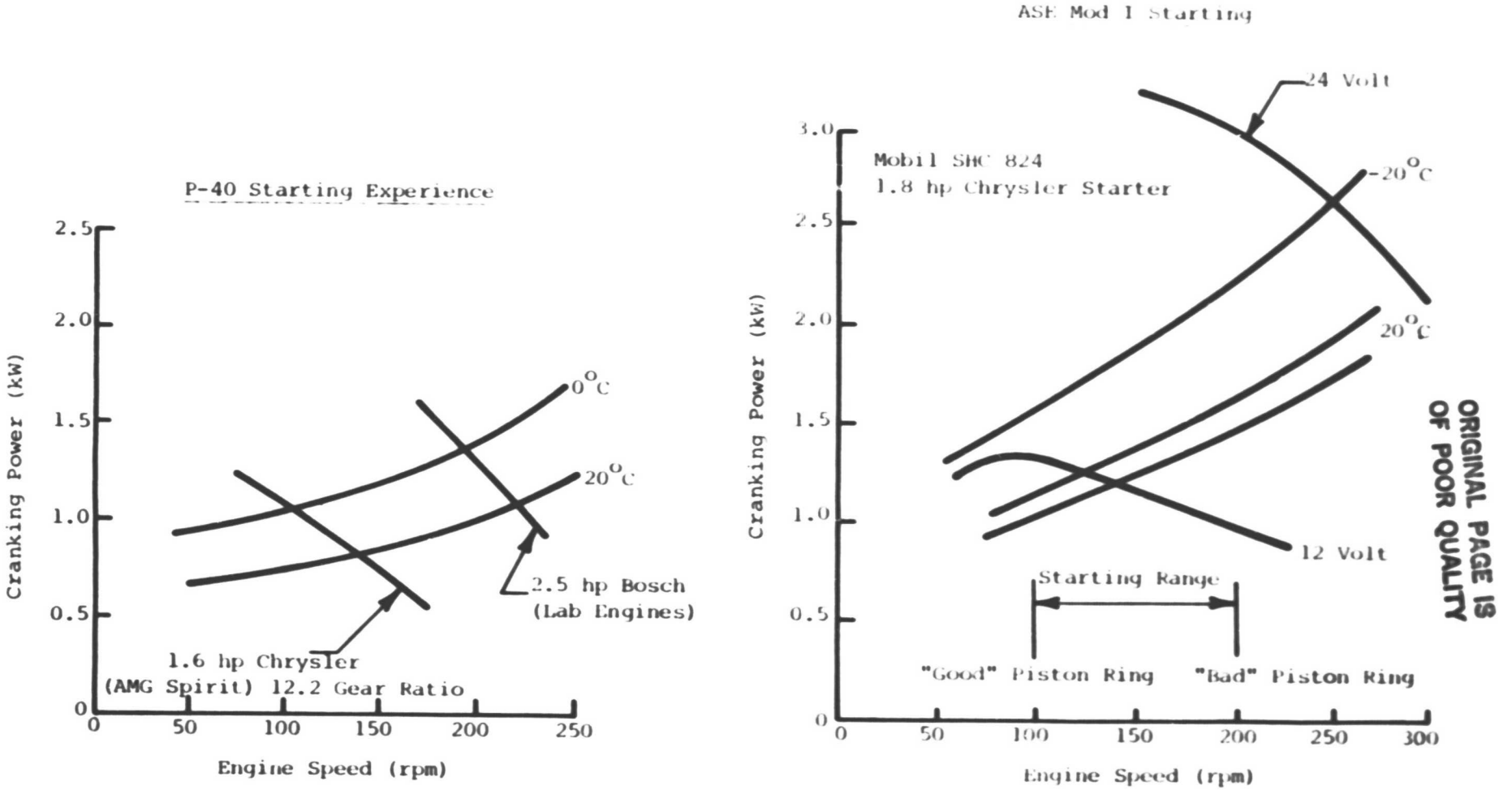


Figure 3 Mod I Starter - Comparison P-40 to Mod I

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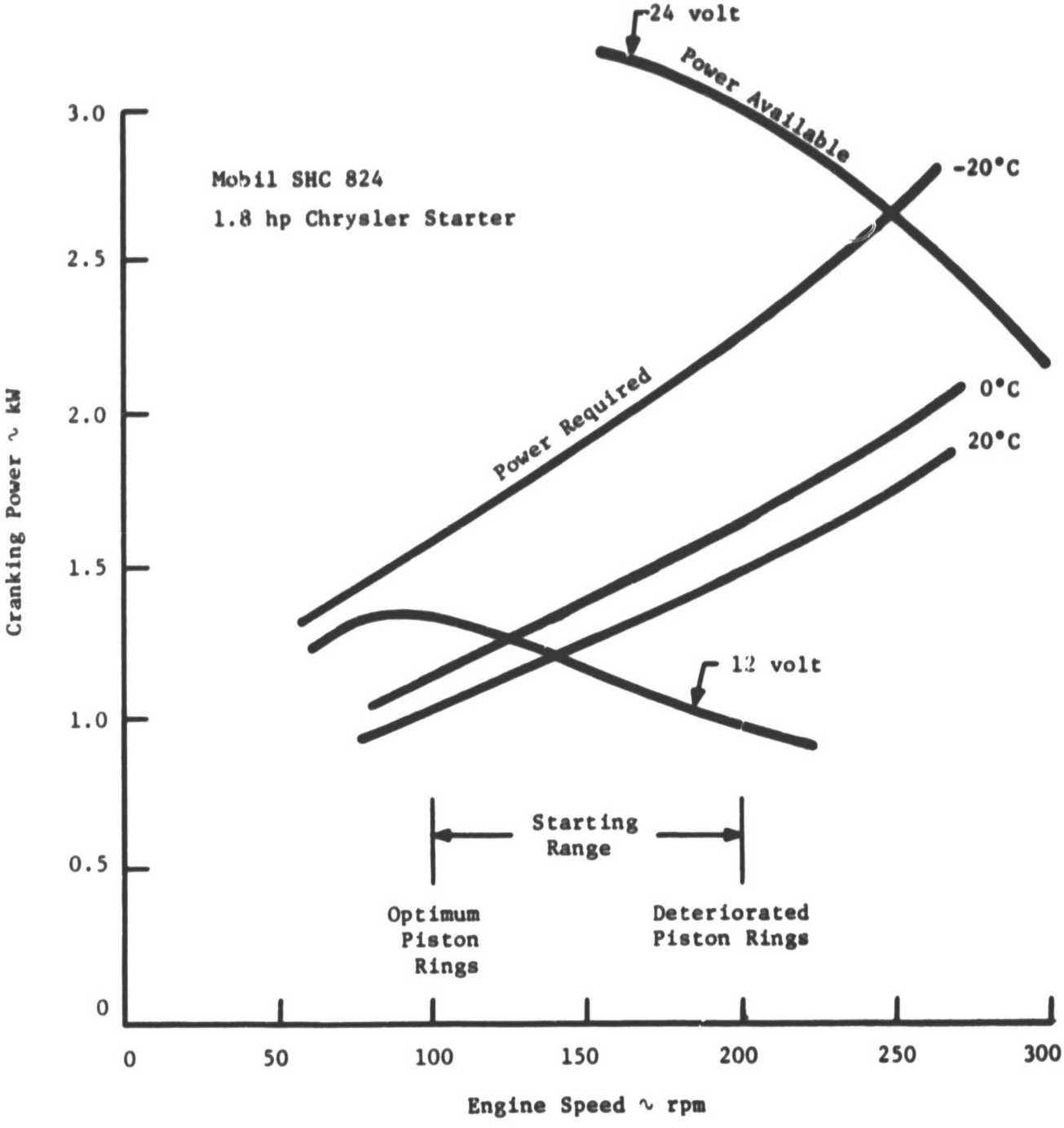


Figure 4 ASE Mod I Starting

TABLE I

MOD I STARTER OPTIONS

OPTIONS	PROBLEMS	COMMENTS
1) 2.2 Kw (3 HP) - 12v STARTER	1) APPROX. 75 MM (3") LONGER THAN 1.3 Kw (1.8 HP) CHRYSLER STARTER - INTERFERENCE WITH STEERING GEAR BOX	VEHICLE MODIFICATIONS UNACCEPTABLE
	2) REQUIRES REBUIDING CHRYSLER STARTER - TIME AND DOLLARS	
	3) REQUIRES REDESIGNED OFF-SET GEAR BOX TO IMPROVE STARTER/ENGINE SPEED MATCH - TIME	
2) 4111860 - 24v STARTER	1) VOLTAGE DROP DISRUPTS ELECTRONIC CONTROL	COMPLICATES CONTROL SYSTEM
A) 2 BATTERIES		
B) 3 BATTERIES	1) LOCATION OF EXTRA BATTERY 2) EXTRA WEIGHT OF VEHICLE	
3) 4111860 - 12v SYSTEM AND 24v EXTERNAL BOOST SYSTEM	1) MODIFIED STARTER AND ELECTRICAL WIRING	LEAST IMPACT ON VEHICLE

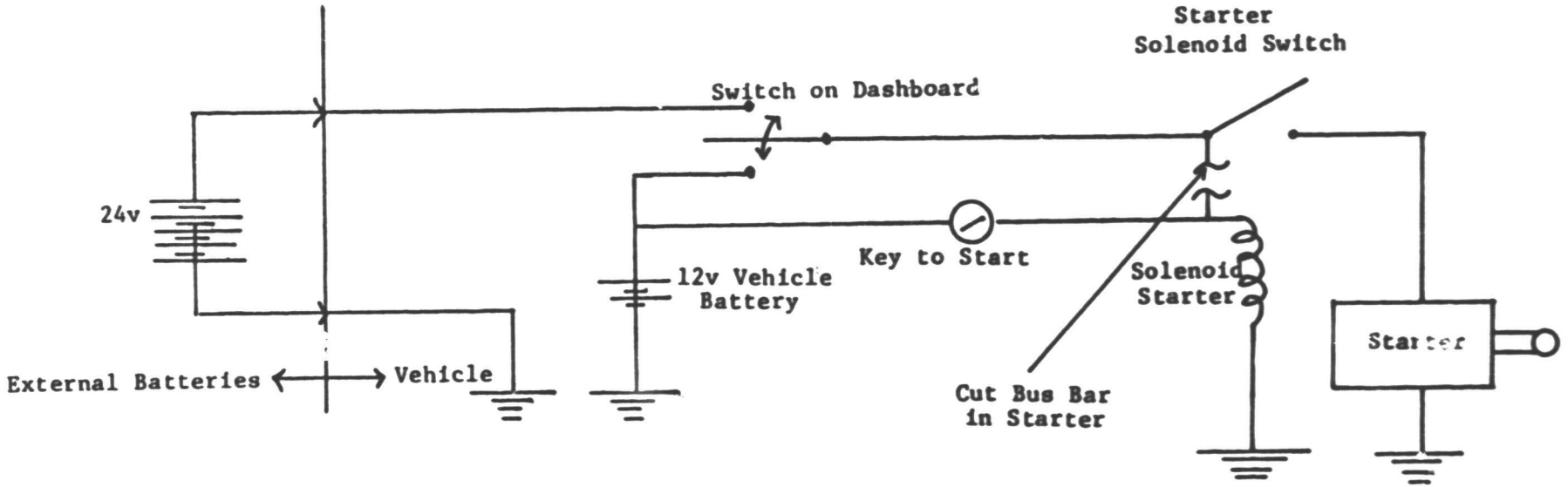
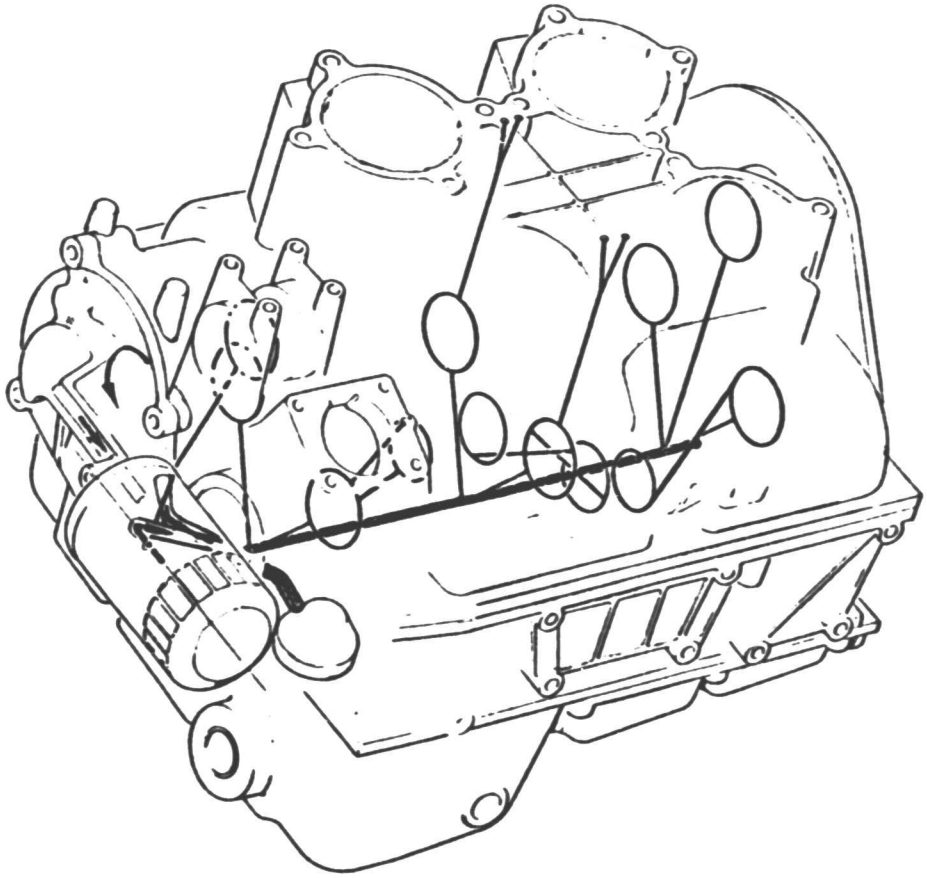
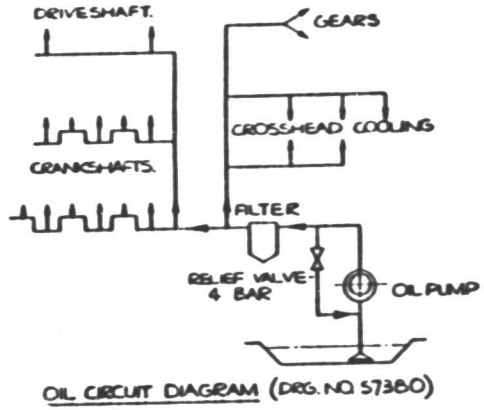


Fig. 5 Mod I Engine Starter (24 v Wiring Diagram)

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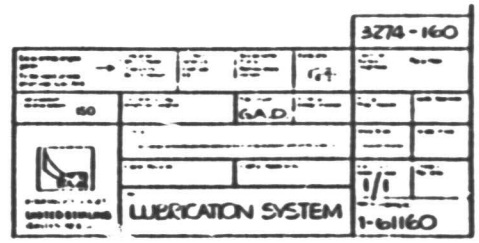




OIL CIRCUIT DIAGRAM (DRG. NO. 57380)

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Fig. 6 Lubrication System



AM General was requested to size the alternator for the Mod I engine with AMC vehicle electrical load. The electrical load for the Mod I was rated at 150 watts or 12.5 amps. Based on this, an investigation was conducted by the AMC electrical group to size the alternator for the engine.

A general outline of the AMC computer program which explains the rating procedure and its derivation is representative of the technique being employed to evaluate the final electrical loads.

1. A rating procedure has been developed which evaluates any given charging system on any given vehicle. A charging system is determined to be adequate by meeting the limits set on three (3) operating conditions and a rating number. The rating number is the result of an empirical equation that weights the three (3) operating conditions with respect to each other and to actual usage. The conditions are:

   a. Amperes discharge at idle.

   b. Minimum MPH required to match typical load.

   c. Minimum MPH required to match maximum load.

   The parameters applied to the procedure to obtain the three (3) operating conditions are:

   1) Alternator output characteristics.
   2) Vehicle electrical loads.
   3) Alternator drive ratio.
   4) Axle ratio.
   5) Tire Size
   6) Idle RPM
   7) Driving schedule electrical loads.

      a) Typical summer or winter

      b) Maximum summer or winter

      c) Idle summer or winter.

   The empirical equation was derived to correlate with the ratings. The AMC equation is:

$$\text{Rating} = \frac{(\text{MPH to Match})^2\ (\text{MPH to Match})^2\ (\text{Idle})Y2}{\text{Typical Load}) + (\text{Maximum Load}) + (\text{Discharge})}{2}$$

2. The limits for rating a charging system should be as follows:

(a) MPH to match typical load - 25 maximum.

(b) MPH to match maximum load - 30 maximum.

(c) Amperes discharge at idle - Summer - 25 maximum.
Winter - 20 maximum.

(d) Rating number - Summer - 22.5 maximum
Winter - 19.5 maximum.

The rating procedure as it finally evolved can be described in three phases: 1) input, 2) output, and 3) limitations. Phase No. 1 may be reviewed by listing each of the parameters put into the computer.

Phase No. 1

(a) Alternator performance curve based on the particular machine being evaluated and whether it is a summer or winter application.

(b) Pulley Ratio based on driving and driven pulley sizes.

(c) Road Factor based on vehicle and worst case axle and tire size options.

(d) Idle RPM based on engine and A/C off or on.

(e) Maximum Electrical Load as derived in parameter description.

(f) Typical Electrical Load as derived in parameter description.

(g) Idle Electrical Load as derived in parameter description.

Phase No. 2 consists of the results of the calculations. They are:

(a) MPH to match typical load.

(b) MPH to match maximum load.

(c) Amperes discharge at idle.

(d) Charging system rating number.

Phase No. 3 consists of applying limits to the results to determine adequacy of the charging system vs. vehicle combination. The evaluations covered by the data included in this report are based on the following limits.

Maximum mph to match typical load was set at 25. This value was chosen based on typical constant speeds in heavy traffic, % of time at various engine speed, and % of time at idle.

The maximum mph to match maximum load was set at 40. This value is based on a typical country driving condition where the maximum load would be applied as a continuous mode.

The maximum discharge at idle was limited to 25 amperes for summer and 20 amperes for winter. These levels are such that the charging system should be able to recover the ampere hours lost while being operated in a typical driving condition. The system rating numbers were given maximums of 22.5 for summer and 19.5 for winter. These maximum rating numbers can be referred to as the "beginning of battery problems". More ideal limits are considered to be 21.0 for summer and 18.0 for winter.

The AMC rating number is not the only indicator of an adequate charging system. Each operating condition limit is also imposed It is obvious that if limit values for each of the three operating conditions were taken, the rating number would exceed the limits of 22.5 and 19.5. This situation displays the interdependency of the operating conditions. It is common for a given system in a given vehicle to have the mph to match typical load at a near limit value and have a low discharge at idle with the combination having a proper rating number. The reverse can also prove to be true. However, each condition must conform to its specific limit.

The AMC computer recommendations are as follows:

PART I

AMG input - 1980 Concord 4-door sedan
- Air Conditioning
- 3.07 Axle
- 12.5 Amp load for Stirling Engine

ALTERNATOR MATCHING PROGRAM

Input Data

1 - Design description AMG

2 - Alternator file name - see below.

3 - Pulley ratio - see below

4 - Road factor 41.3

5 - Idle RPM - 600

6 - Max. Load 45

7 - Typical load 41

8 - Stall load 48

BOSCH 55 AMP ALTERNATOR WITHOUT A/C

3 - Pulley ratio 2.4

Calculated results:

Max. load velocity	24.22
Typical load velocity	20.82
Idle discharge	22.07
System rating	19.41

BOSCH 55 AMP ALTERNATOR WITH A/C

3 - Pulley ratio 2.55

Calculated Results:

Max. load velocity	22.79
Typical load velocity	19.60
Idle discharge	20.22
System rating	18.11

BOSCH 65 AMP ALTERNATOR WITH & WITHOUT A/C

3 - Pulley ratio 2.2

Calculated Results:

Max. load velocity	21.72
Typical load velocity	19.76
Idle discharge	20.53
System rating	17.91

5 - Idle RPM - 600

6 - Max. load 45

7 - Typical load 41

8 - Stall load 48

PART II

AMG Input - Same as Part I except
Vehicle Rear Axle Ratio - 2.3

Input Data

1 - Design description AMG

2 - Alternator file name - WBO-55:

3 - Pulley ratio - 2.55

4 - Road factor 32.0

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BOSCH 55 AMP ALTERNATOR WITH A/C

3 - Pulley ratio 3.0

Calculated Results:

Max. load velocity	29.63
Typical load velocity	25.00
Idle discharge	16.72
System rating	21.11

BOSCH 65 AMP ALTERNATOR WITH A/C

3 - Pulley ratio 2.55

Calculated Results:

Max. Load velocity	27.49
Typical load velocity	24.18
Idle Discharge	19.14
System rating	20.66

ALTERNATOR CURVE FOR BOSCH 55 AMP ALTERNATOR

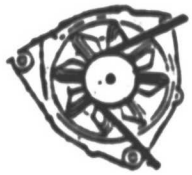
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1600.00	25.68
1800.00	31.28
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2400.00	41.04
2600.00	42.96
2800.00	44.64
3000.00	46.00
3200.00	46.94
3400.00	47.63
3600.00	48.40
3800.00	49.20
4000.00	50.00
4200.00	50.82
4400.00	51.63
4600.00	52.35
4800.00	53.00
5000.00	53.50
5200.00	53.75
5400.00	53.88
5600.00	54.16
5800.00	54.54
6000.00	55.00

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ALTERNATOR SIZING PROGRAM

INPUT

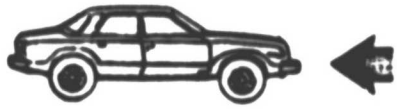
A. ALTERNATOR PERFORMANCE CURVE



B. PULLEY RATIO



C. ROAD FACTOR



D. ENGINE IDLE RPM



E. MAXIMUM VEHICLE ELECTRICAL LOAD - AMPERES

F. TYPICAL VEHICLE ELECTRICAL LOAD - AMPERES



G. IDLE ELECTRICAL LOAD - AMPERES

OUTPUT

A. MPH TO MATCH TYPICAL LOAD



B. MPH TO MATCH MAXIMUM LOAD



C. AMPERES DISCHARGE AT IDLE



D. CHARGING SYSTEM RATING NUMBER

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Bosch K-jetronic air-fuel system

The first ASE MOD I engines have to be equipped with a Bosch K-jetronic air-fuel system, instead of the system described in section 8.3.13. The application and specification are the same, with exception from the air pressure drop in the flow meter, which is higher, and the possibilities of measuring mass air flow.

Description: The Bosch K-jetronic air-fuel system consists of the parts included in fig. 8.3:22.

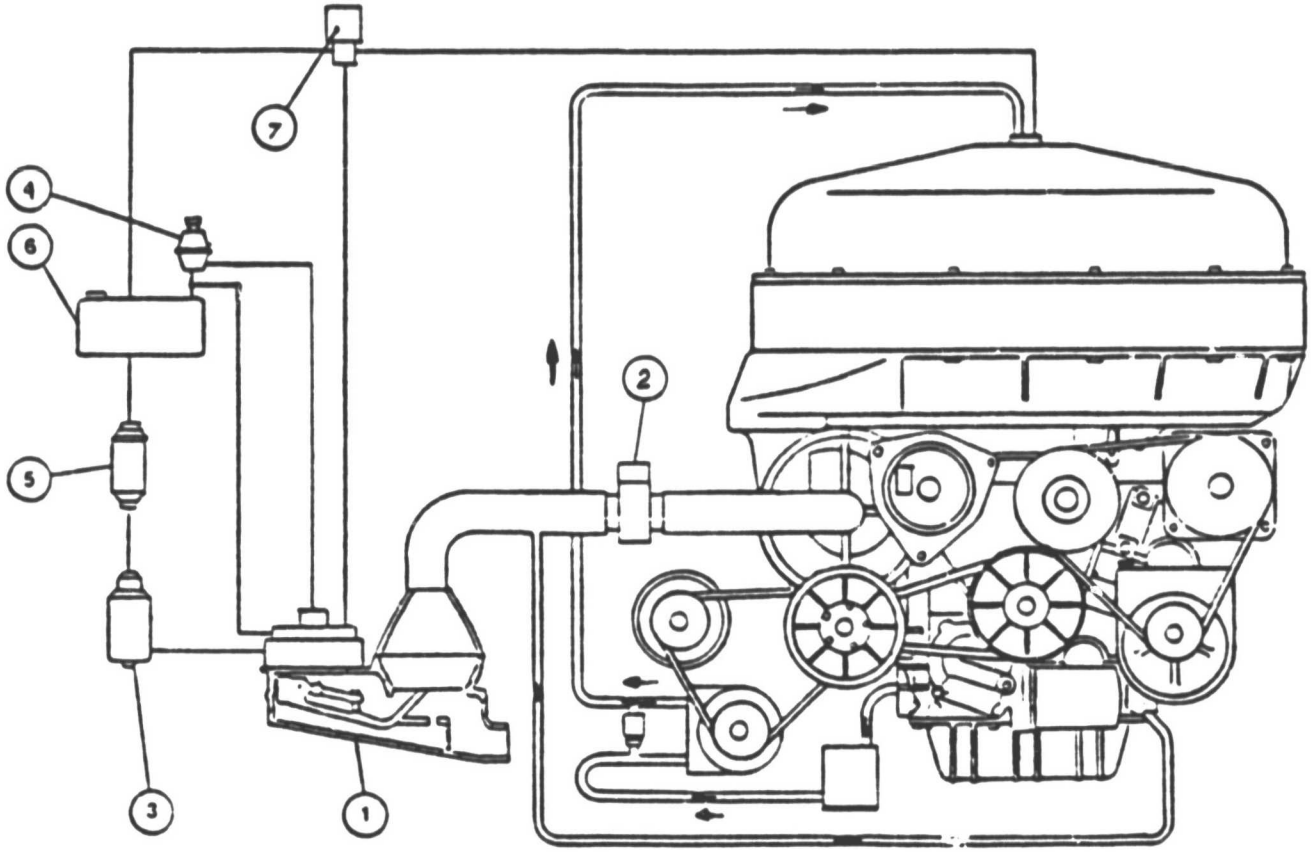


Fig. 8.3:22

1. Bosch K-jetronic
2. Air Throttle
3. Fuel Pump
4. Pressure Relief Valve
5. Fuel Filter
6. Fuel Tank
7. Fuel Valve

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The K-jetronic unit is a mechanical fuel injection system used on fuel injected IC-engines.

The air measuring device is a movable disk in special shaped cone. The more air going through the system, the higher the disk is lifted. A lever, connected to the disk, is controlling the position of a plunger, which acts as the fuel measuring device. From the plunger the fuel flows to the fuel distributors, which are designed to keep the delivered amount of fuel independent of fuel nozzle back pressure.

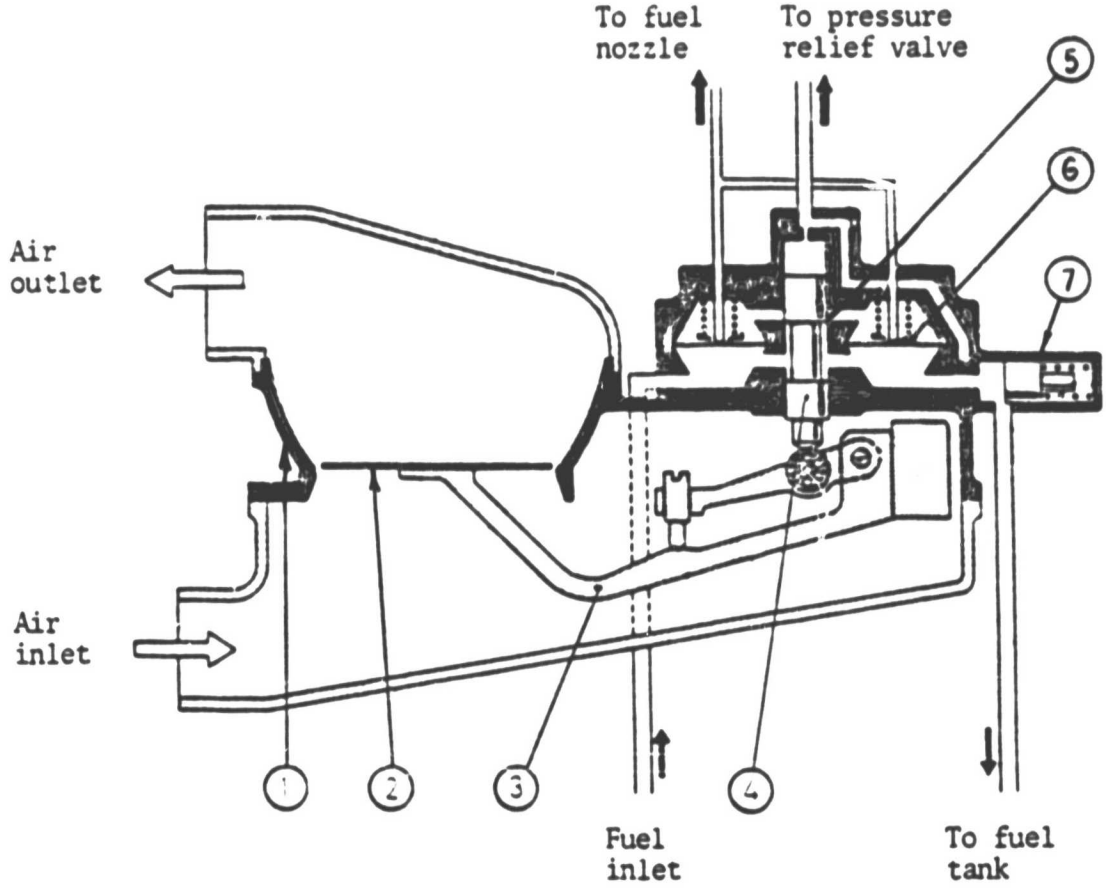


Fig. 8.3:23 K-Jetronic Cross-Section

1. Cone
2. Disk
3. Lever
4. Plunger
5. Fuel Measuring Slot
6. Fuel Distributor
7. System Pressure Relief Valve

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The K-jetronic is modified by USSw to fulfill the special requirements. The main modifications are:

The cone and disk are special designed to obtain the right air excess curve. Three of the four fuel distributors are connected to each other. The fourth is plugged.

This air-fuel system will influence the engine in the following ways:

The pressure drop of the air flow measuring device will increase, as can be seen in fig. 8.3:23

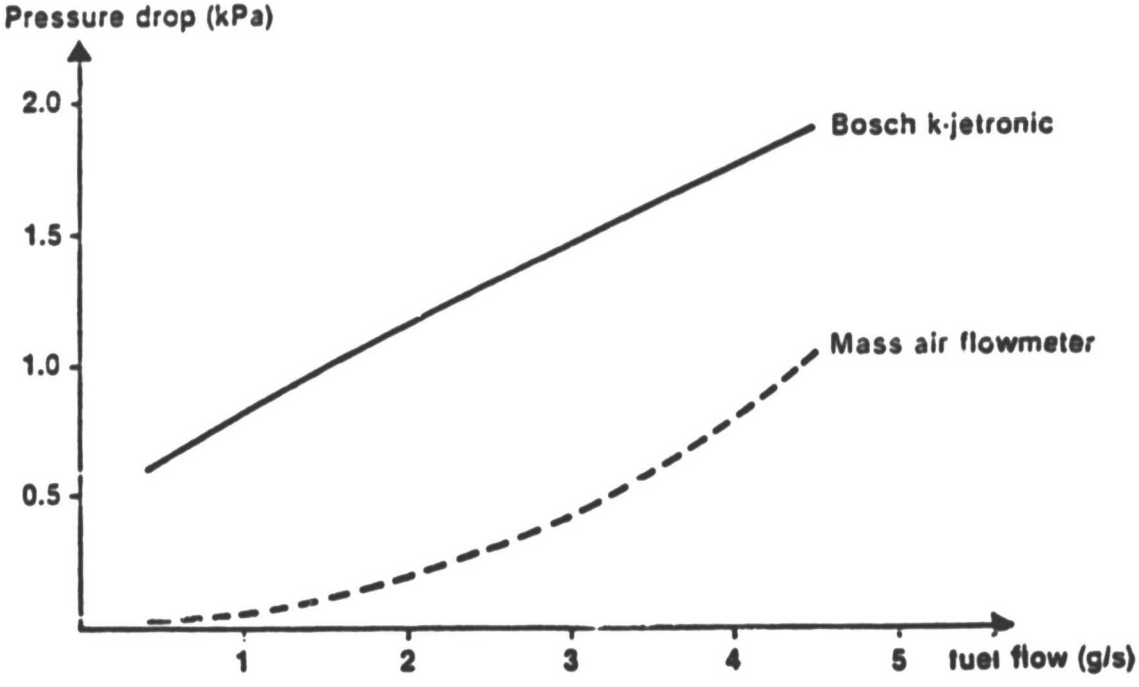


Fig. 8.3:23 Comparison of Pressure Drop Characteristic

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This will influence the combustion air blower power demand according to fig. 8.3.24.

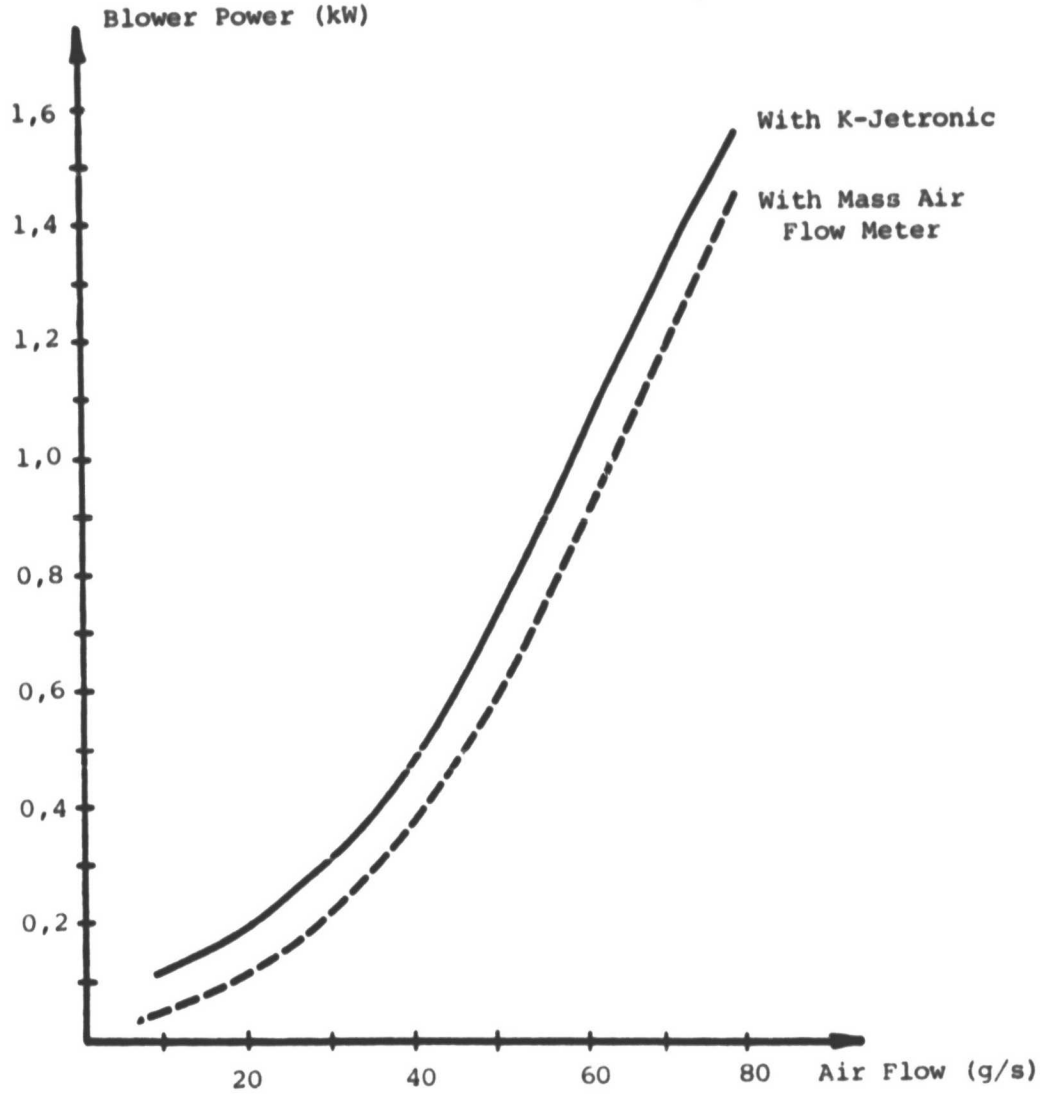


Fig. 8.3:24 Comparison of Blower Power Characteristics

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The fuel system pressure will increase, with a higher fuel pump power demand as result. Fig. 8.3:25 shows the increased alternator power demand from the engine.

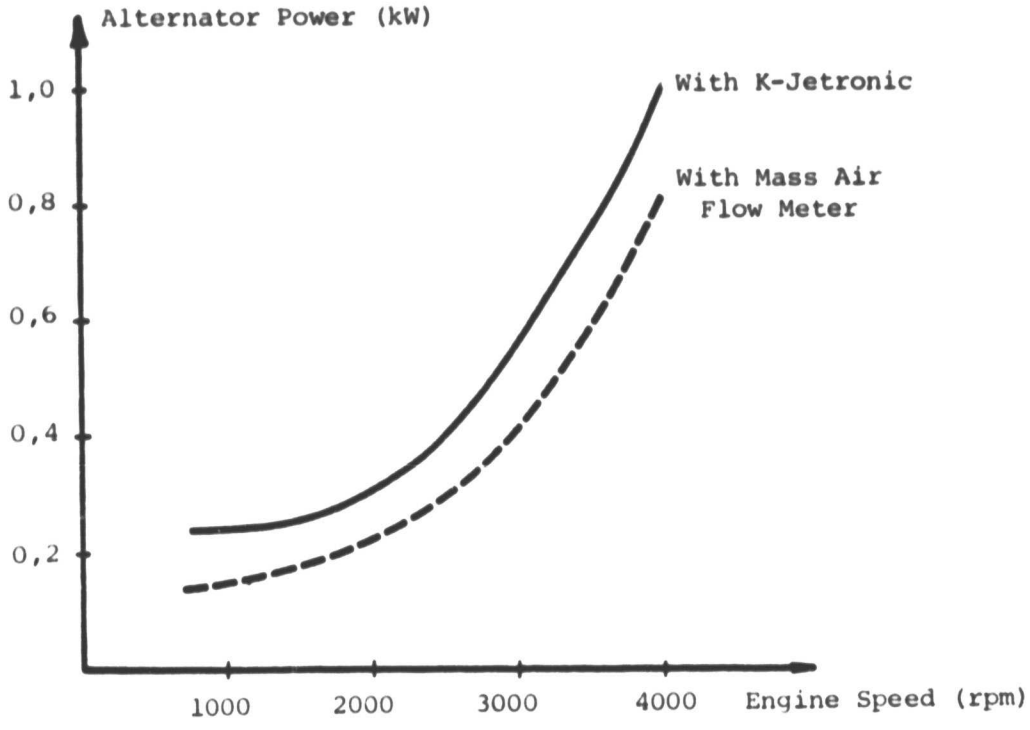


Fig. 8.2:25 Comparison of Alternator Power Characteristics

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General Description

The Poly-Rib belt is gaining wide acceptance in the automotive field due to the advantages it exhibits over conventional V-belts. Small section Poly-Rib belts (H and J sections) have been used in the appliance and business machine industry for many years. Recently, due to the need for smaller pulley diameters and increased performance, Poly-Rib belts have been selected for use on many applications. These applications range from the single multi-rib accessory drive concept (SMAC) for passenger cars to water pump, alternator, and air conditioning drives on heavy trucks. In addition, some fan drives on construction equipment are being placed in production.

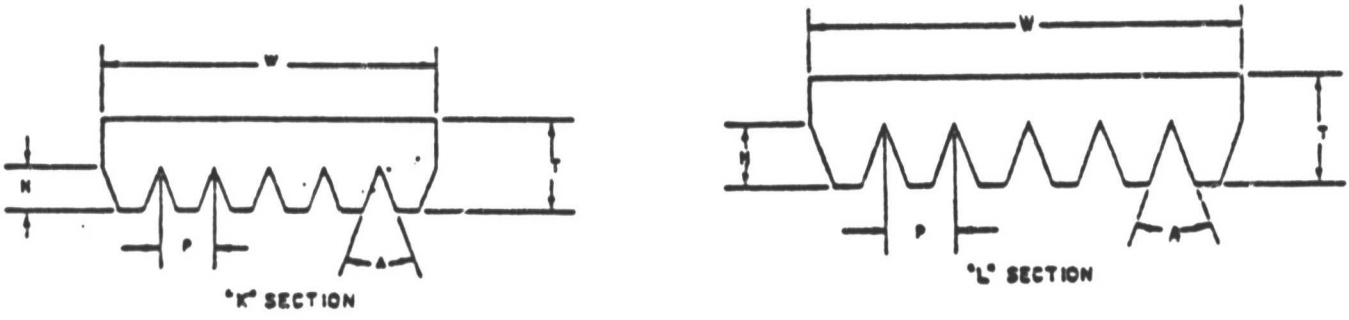


Fig. 1 "K" and "L" Sections

"K" and "L" Sections

The demand that smaller pulley diameters and increased performance has placed upon belts has brought about the utilization of the "K" and "L" section Poly-Rib belts.

The "K" and "L" section Poly-Rib belts have been found to be correctly dimensioned to allow satisfactory application on the majority of automotive accessory drive systems. Poly-Rib belt sections which are designated by the letters "K" and "L" are easily identifiable by measuring the distance between the adjacent ribs within a belt. Dimensions and rib profiles can and will be changed to improve belt performance and to alleviate problems which may occur on specific applications.

Dayco Poly-Rib belts can be manufactured in a variety of lengths, number of ribs and material specifications. The capability of Dayco to manufacture Poly-Rib belts using several different processes enables the rubber compounder and engineer total freedom in the selection of materials to obtain maximum belt performance on each individual application. Belt lengths between 20 in. (500mm) and 120 in. (3050mm) are easily manufactured, while belt lengths outside this range are available as special production orders.

Advantages

The Poly-Rib belt sections are chosen for applications where single or multiple standard V-belts are deficient. Poly-Rib belts offer the following advantages over standard V-belts:

Flex Life

Increased flex life with equal or better load carrying capability.

Strength

The design of the Poly-Rib belt allows placement of the tensile member across the total width of the belt. The tensile member of the V-belt is placed down in the wedge area of the belt. The tensile member width of a V-belt is always less than the overall width of the belt. It is therefore obvious that the Poly-Rib belt will exhibit superior strength to a V-belt of equal width

Tension Loss

The majority of the tension loss associated with a V-Belt is due to belt wear and seating. As a V-belt wears or seats, the belt rides lower in the pulley groove causing a substantial tension loss. Because the Poly-Rib functions more similarly to a flat belt, belt wear is reduced and the wear that does occur does not cause the belt's relative position to the pulley to change a significant amount. The ability of the Poly-Rib belt to hold its ride position will increase the tension holding capability.

Belt Adjustment

Due to the Poly-Rib belt's advantage in strength and ability to hold tension, fewer retentions and less belt adjustment is required.

Backside Bending

The thinness of the Poly-Rib belt cross section allows it to perform up to 50 times better than a conventional V-belt over severe backside bends.

Efficiency

Laboratory testing has shown that the Poly-Rib belt requires less energy to operate than a conventional V-belt.

AM General has reviewed the United Stirling engine accessory drive and has offered an alternate drive arrangements using a Poly-Rib Belt drive.

We offer the proposal because of the advantages such a drive system offers. AMC has experienced mileage improvements in the urban driving cycle of 0.4 of a mile per gallon. Cost savings are also seen at AMG with engines with optional accessories. With the Serpentine Poly-Rib belt system, accessories can be mounted in a fixed position with one accessory being used as the belt adjuster.

AM General has presented three proposals using existing AMC accessory hardware. New pulleys will be required for the output shaft and the fan drive. The accessory mounting package will have to be revised to incorporate the drive system. The service benefits can be appreciated after reviewing the present V-belt drive with the Serpentine drive.

With the present schedule, it is questionable if the output shaft and variator drive pulley could be revised to accept the output drive pulley for a Poly-Rib Serpentine drive system. The proposal shows great promise for future engines in the Mod I program, but at this time we will replace the conventional V-belt accessory drive with Poly-Rib belts. The Serpentine drive will be developed for later Mod I engines.

Dayco Corporation has been contacted to review our proposals and run a belt study using our layouts. The results of the study will be forwarded to the Program Office as soon as they are received.

AMC Recommended Accessory
Drive Ratios

Fan: 1.20

Alternator: 2.90

A/C Compressor: 1.38

Power Steering: 1.25

AMG Serpentine
Drive Ratios

Fan: 1.28	Pulley Size - 114.3mm (4.50)
Alternator: 2.55	Pulley Size - 57.15mm (2.25)
A/C Compressor: 1.21	Pulley Size - 119.88mm (4.72)
Power Steering: 1.10	Pulley Size - 132.08mm (5.20)
Output Shaft:	Pulley Size - 146.05mm (5.75)

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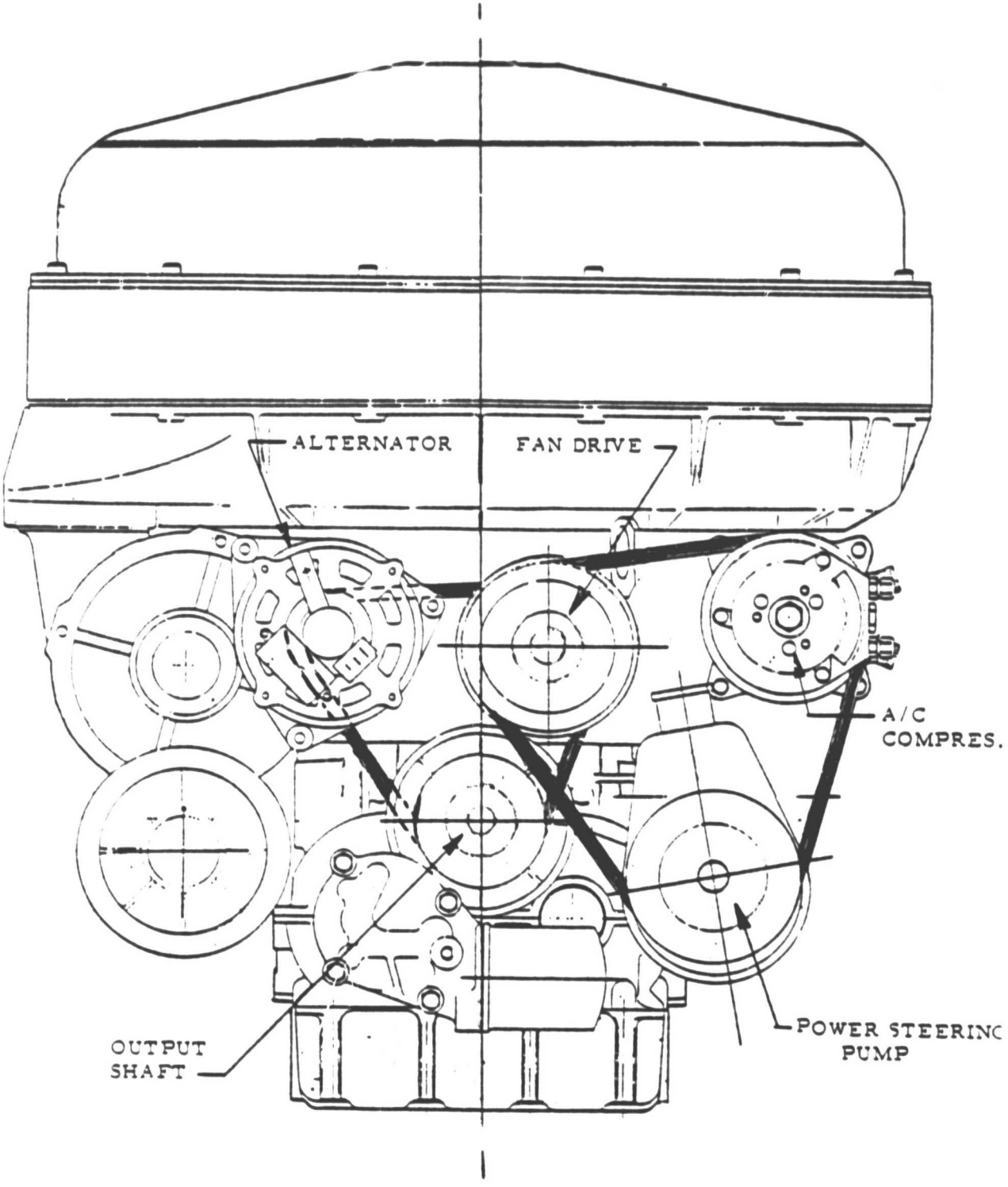


Fig. 2 Conventional V-Belt Accessory Drive

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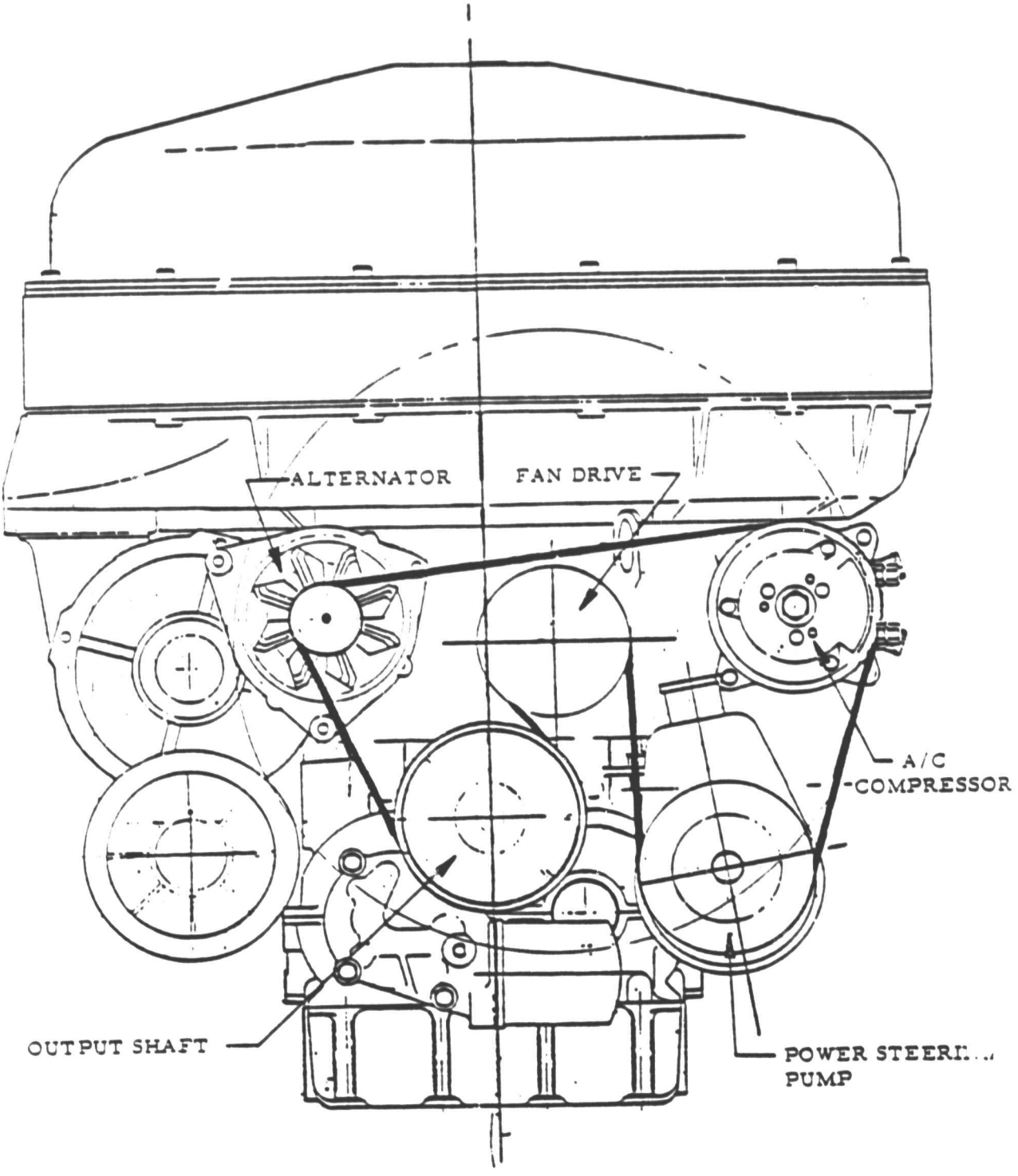


Fig. 3 Serpentine Poly-Rib Belt Accessory Drive Proposal No. 1

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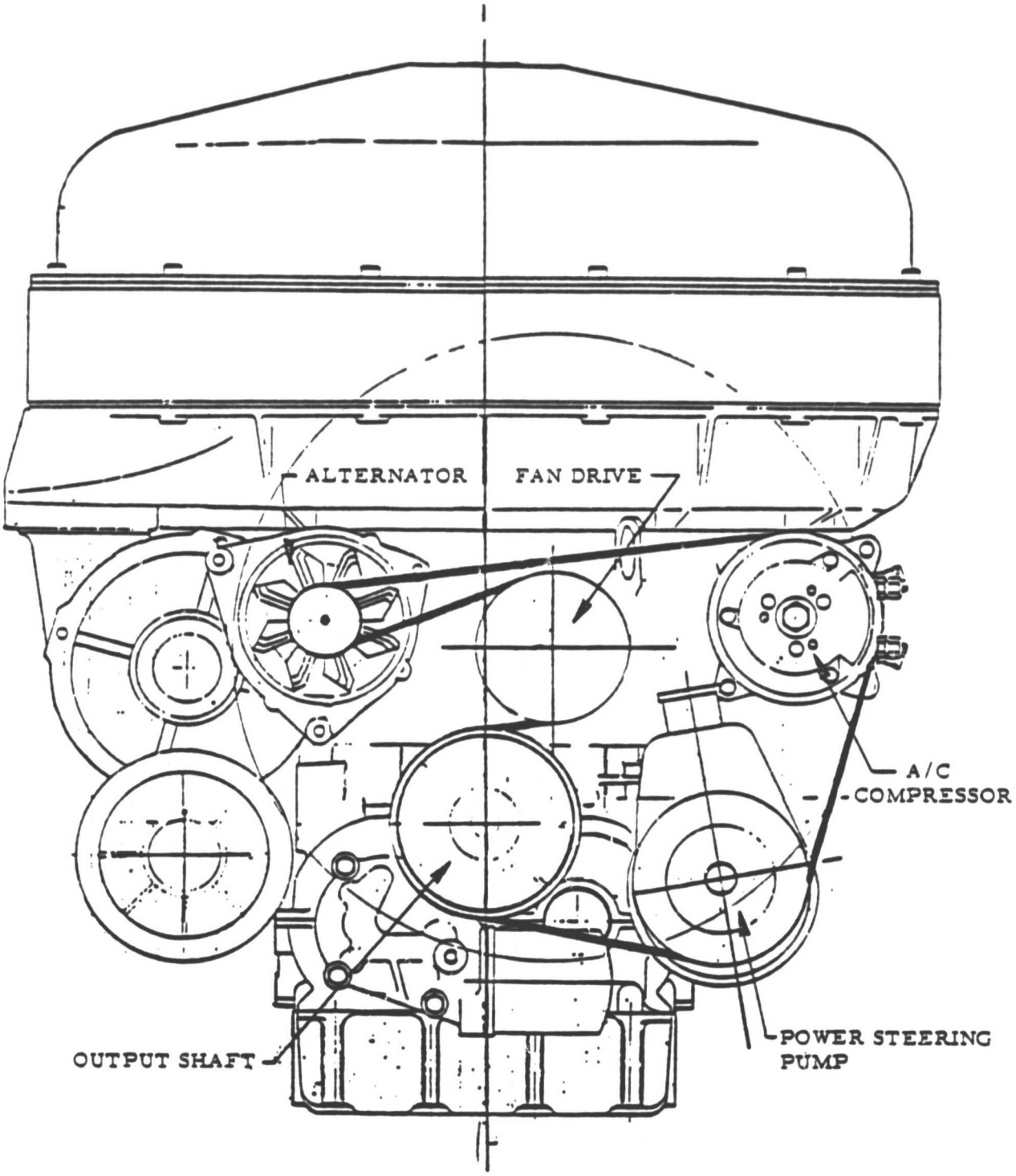


Fig. 4 Serpentine Poly-Rib Belt Accessory Drive Proposal No. 2

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8.4 POWER CONTROL SYSTEM

8.4.1 General Description

To control the power of the Stirling engine, we have the option of varying a number of quantities occurring in the formula for the shaft power. According to the isothermal theory, this formula has the form:

$$P_{ind} = \frac{1}{2} * w * \bar{p}_m * V_{Emax} * (1-\tau) * \frac{\delta}{1 + \sqrt{1 + \delta^2}} * \sin\theta$$

Definitions:

w = angular velocity

$\bar{p}_m$ = mean cycle pressure

V_{Emax} = maximum value of the expansion space volume

τ = temperature ratio $(\frac{Tcompr}{Texp})$

δ = measure for the compression ratio

θ = phase angle between the motion of the expansion piston and pressure

By varying one or more of these quantities, the power of the Stirling engine can be affected. When the engine is used in automotive applications w must be a variable.

To control the power we still have the option of using the following quantities:

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1. V_{Emax} can only be varied with a variable drive concept.

2. $\tau = \frac{TC}{TE}$: By varying this quantity the efficiency is directly influenced. Moreover, it is impossible to effect temperature variations rapidly and they cause large material stresses.

3. $\bar{P}_m$: Variation of the mean cycle pressure
   (Mean Pressure Control, A).

4. δ can be altered by adding one or more spaces to the cycle. This is Pressure Amplitude Control, B.

5. By letting working gas flow from cycles at max pressure to cycles at min pressures θ will change. This is called By-pass or Short circuiting control, C.

Varying the engine output also changes the area of the PV-diagram. Mean Pressure Control, Pressure Amplitude Control and By-pass Control will all change the PV-diagram in different ways:

Fig 8.4:1 shows the influence on the PV-diagram from different power control methods.

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INFLUENCE ON THE PV-DIAGRAM FROM DIFFERENT POWER CONTROL METHODS

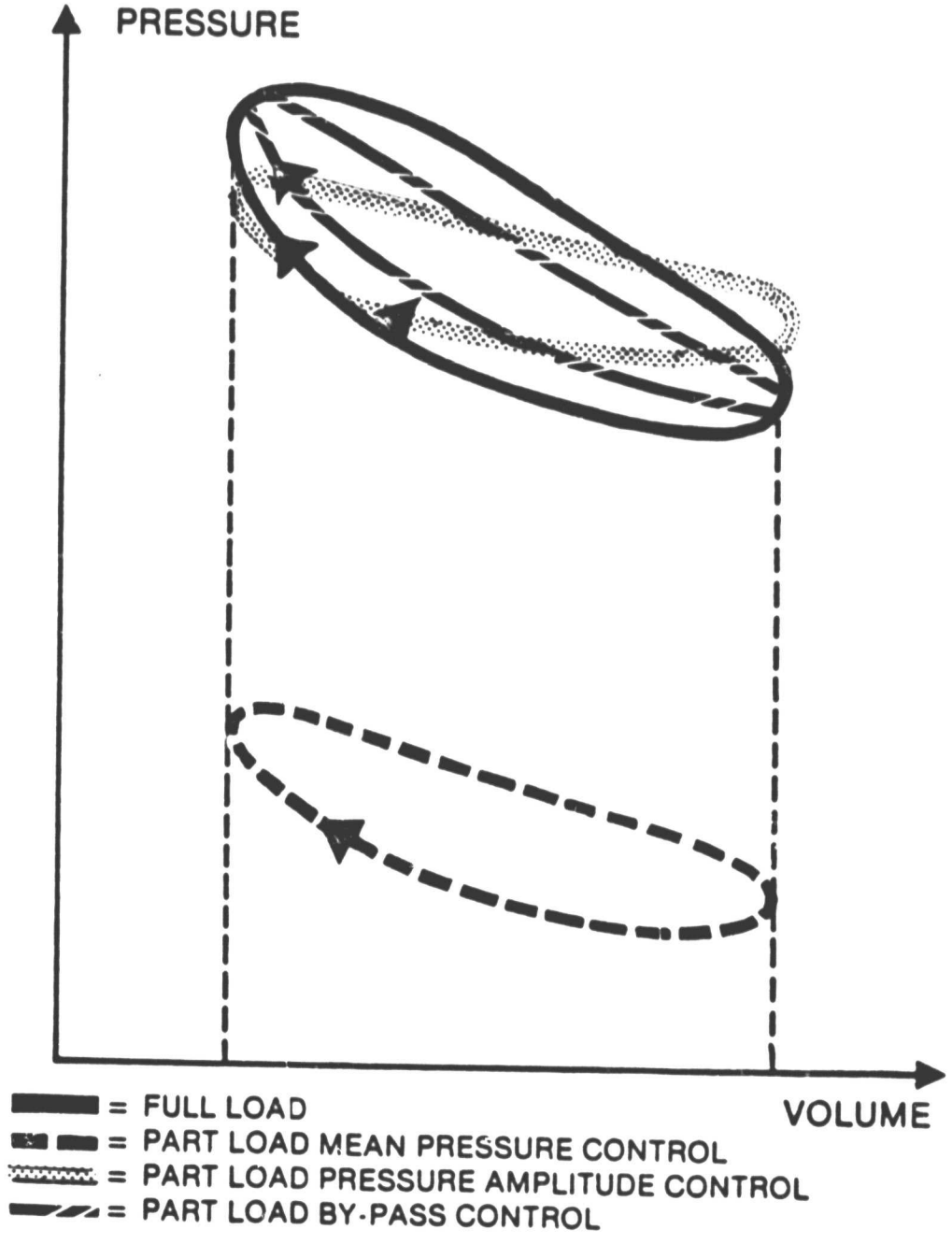


Fig 8.4:1 Influence on the PV-diagram from different power control methods.

Advantages and Disadvantages of Available Power Control Systems:

A. Mean Pressure Control

Advantages: Good part load efficiency
Low fuel consumption at idle
Fast torque increase response
Good control stability

Disadvantages: Complex components
Slow torque decrease response

B. Pressure Amplitude Control

Advantages: Good part load efficiency
Fast torque increase and decrease response
Good control stability

Disadvantages: Discrete power levels
Many internal leaking points
Large externally added engine volumes
Relatively high idle fuel consumption

C. Short-circuiting

Advantages: Very fast torque response
Engine braking capabilities
Few components

Disadvantages: Bad part load efficiency
Very bad idle fuel consumption
Bad control stability

On ASE Mod I as well as for the P-40 engines a combination of mean pressure control and short-circuiting control have been chosen as the power control system. Mean pressure control, because of its good part load efficiency, idle fuel consumption, and for its good controlability in automotive applications. The slow power decrease response and the engine braking are performed by short-circuiting. Short-circuiting is used only for short periods of engine braking and during pressure decrease periods. The fuel penalty due to bad part load efficiency is therefore small.

This combination of mean pressure control and short-circuiting is generally called Mean Pressure Control.

8.4.2 Description of the ASE Mod I Mean Pressure Control System

(All figure references in this section refer to Figure 8.4:24 block scheme of the ASE Mod I Power Control System).

Major parts of this power control system are hydrogen storage tank (18), hydrogen compressor (7), supply, dump and short-circuiting spool valve (1) and the servo actuator (2).

To increase power the spool valve (1) is moved upwards by the actuator (2). Thereby, hydrogen flows from the high pressure storage vessel (18) via the spool valve and the P-supply line to the engine. The hydrogen enters the engine via a timed supply system (P_{max} supply), built into the engine. This timed supply system is described in section 8.4.3. The system mainly supplies hydrogen into the cycles when the cycle pressure is near its maximum valve. If gas flow into the cylinders without the timed supply system, an undesireable torque drop during increase of pressure results.

To decrease power the spool valve (1) is moved downwards. During the first part of the movement, dumping of hydrogen from the engine via the compressor (7) to the storage vessel (18) lowers the power output. During the second part of the valve movement short-circuiting of hydrogen between the cycles occurs, thus giving a quick decrease of power.

Between the accelerator pedal (23) and the spool valve actuator (2) there is an electronic unit (24) with built in control functions, described in sections 8.2 and 8.5, which for different accelerator levels moves the control valve in such a way that an engine pressure corresponding to desired power output will be reached and maintained.

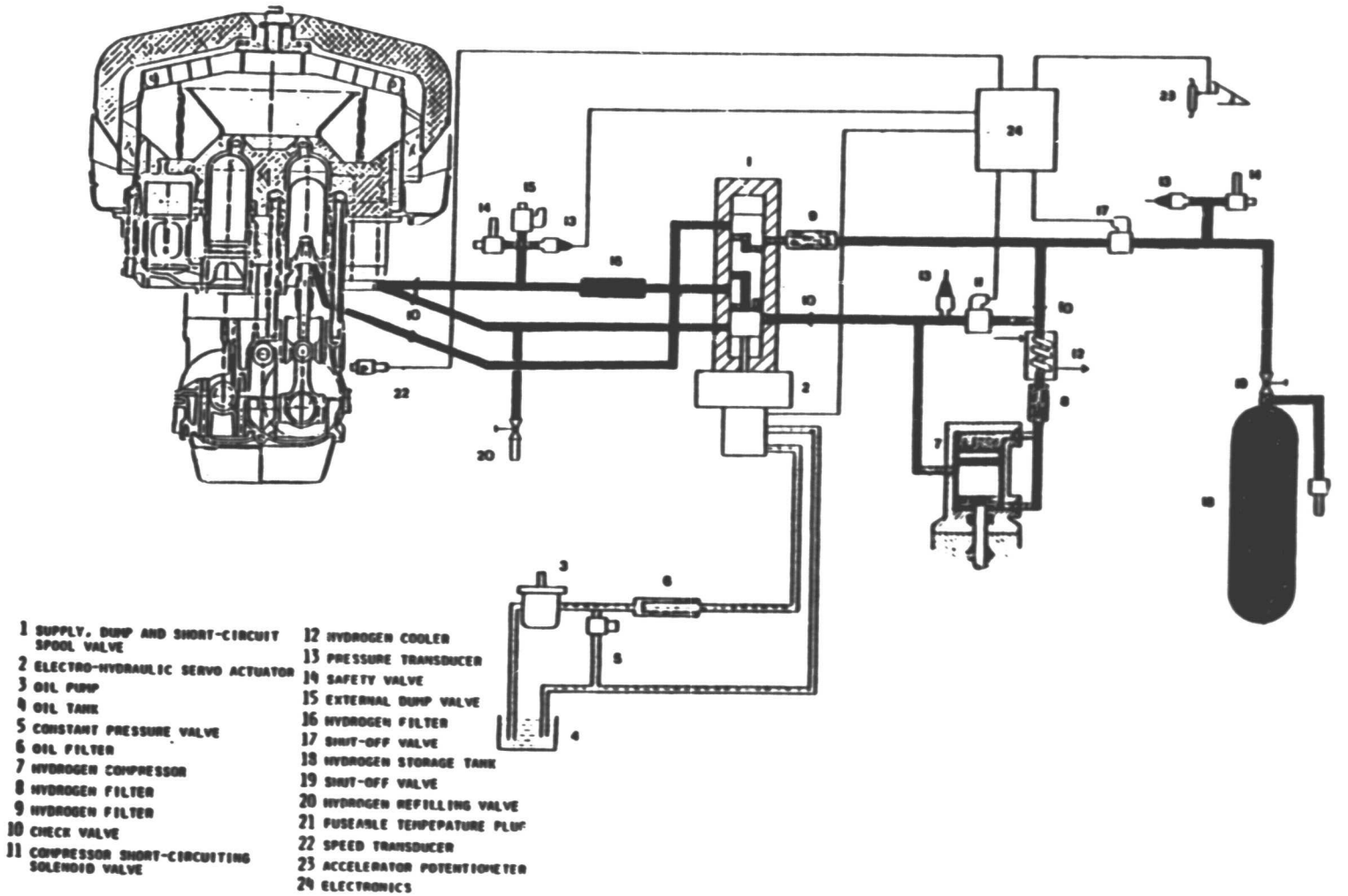


Figure 8.4:24 Block Scheme of the ASE Mod I Power Control System

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To perform the control functions the electronic unit needs signals from transducers: maximum pressure transducer (13) and speed transducer (22).

There are an additional number of components within the power control system. The following are items, included in a complete power control system.

There is a total of 16 check valves (10) within the control system. One max pressure, one min pressure and one P_{supply} check valve for each cylinder. Two compressor exhaust check valves and two check valves before and after the compressor short-circuiting circuit.

The system contains three hydrogen filters (8, 9 and 16) preventing rulon dust from piston rings/or other contaminents from entering the gas system.

The compressor short-circuiting circuit contains besides a filter (8), a hydrogen cooler (12) (reducing compressed hydrogen temperature), a short-circuiting solenoid valve (which unloads the compressor during increase of power and steady-state conditions), and a monitoring pressure transducer (13).

The P_{max} line contains besides the max pressure transducer (13) and filter (16) of an external dumping solenoid valve (15), as emergency unit, and a safety valve (14) preventing too high pressures within the engine.

There is also a safety valve (14) preventing too high pressures after compressor, including too high tank pressures.

Between the tank and the control valve there are also a fusible temperature plug (21), preventing tank overtemperature, a manual shut-off valve (19), a solenoid shut-off valve (17), monitoring

tank pressure transducer (13) and filter (9).

To charge the system from external hydrogen tanks, there is a refilling valve (20) at P_{min} line.

The servo oil system components are: oil pumps (3), oil tank (4), constant pressure valve (5), and oil filter (6).

8.4.3 P_{max} Supply

(Timed Supply)

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To meet the fast response specification during increase of power in automotive applications it is necessary to supply the engine in a special way.

Fig 8.4 : 3 shows the PV-diagram during a supply of hydrogen through the P_{min} line. Most hydrogen enters the cycle during periods of minimum pressure when pressure difference between tank and cycle has max value.

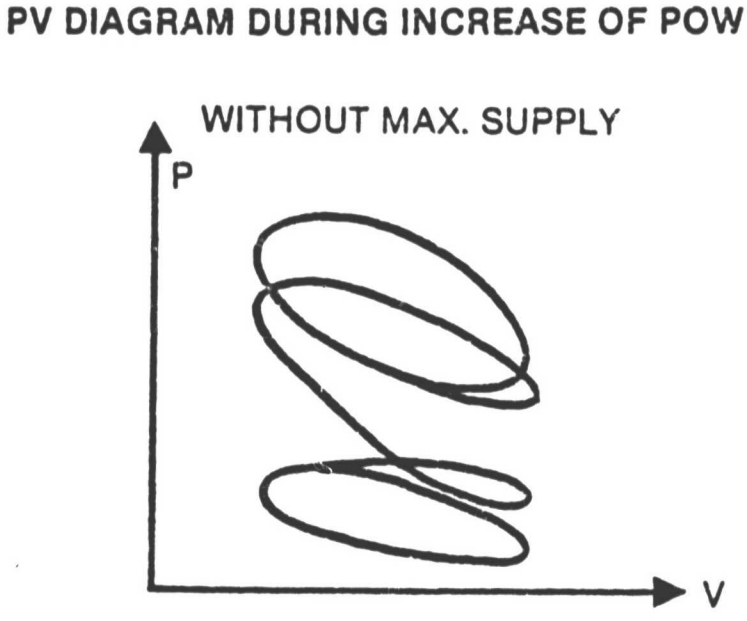


Fig 8.4:3 PV-diagram during increase of power without timed supply system.

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The small PV-diagram areas, shown during the transient phase, indicate a big torque drop during fast acceleration periods. This has also been verified during tests.

If the periods of supply are limited to periods where cycle pressure is almost near its maximum, see fig 8.4.4. The PV-diagram during the transient phase looks different. Fig 8.4.5 shows the PV-diagram during the transient phase of a timed supply. The big areas of the PV-diagram indicate that no torque drop occurs.

USS$_W$ TIMED SUPPLY SYSTEM
TIMING INTERVALS

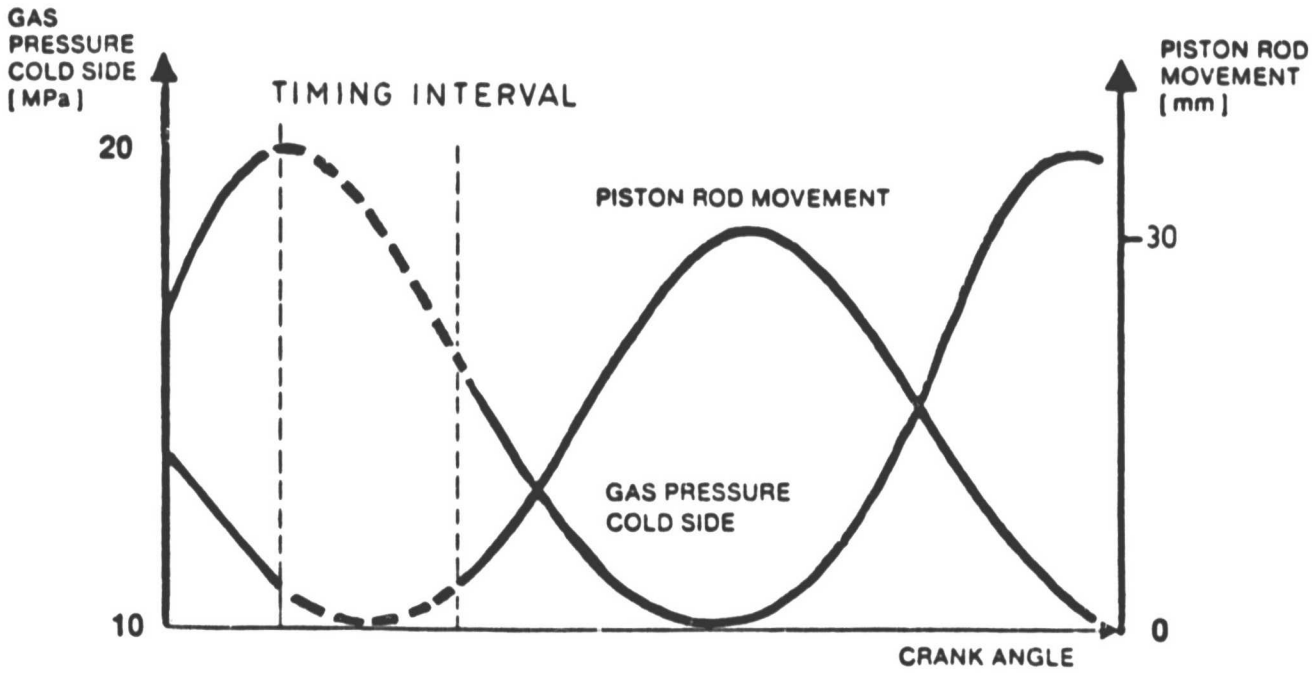


Fig 8.4:4 Timing interval of P_{max} supply.

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PV DIAGRAM DURING INCREASE OF POWER

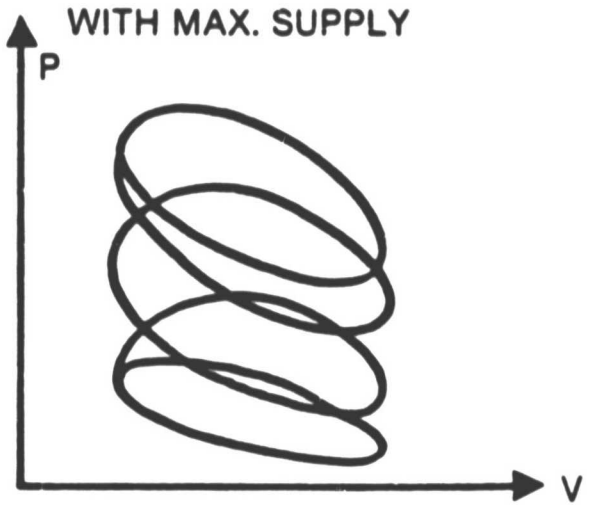


Fig 8.4:5 PV-diagram during increase of power with timed supply system.

It could also be said that without P_{max} supply, the hydrogen entering into the cycles, must first be compressed before it can do useful work during an expansion. With P_{max} supply hydrogen will first do useful work during the expansion before being compressed.

On all USSw mean pressure controlled engines as well as on ASE Mod I the timed supply system is incorporated in the piston rod and seal housing. Figure 8.4.6 shows the principle of the USSw timed supply system.

The timing element is a slot or groove in the piston rod, opening a connection between the supply line and the cycle only during periods when the piston is near its bottom dead center. When piston is leaving the supply interval, some hydrogen may still flow through the clearance between piston rod and seal housing.

This has the advantage that some supply also occur during the rest of the revolution making it possible to supply the engine at high part load conditions when maximum pressure of the cycle is higher than tank pressure. During these high part load transients the torque drops are negligible.

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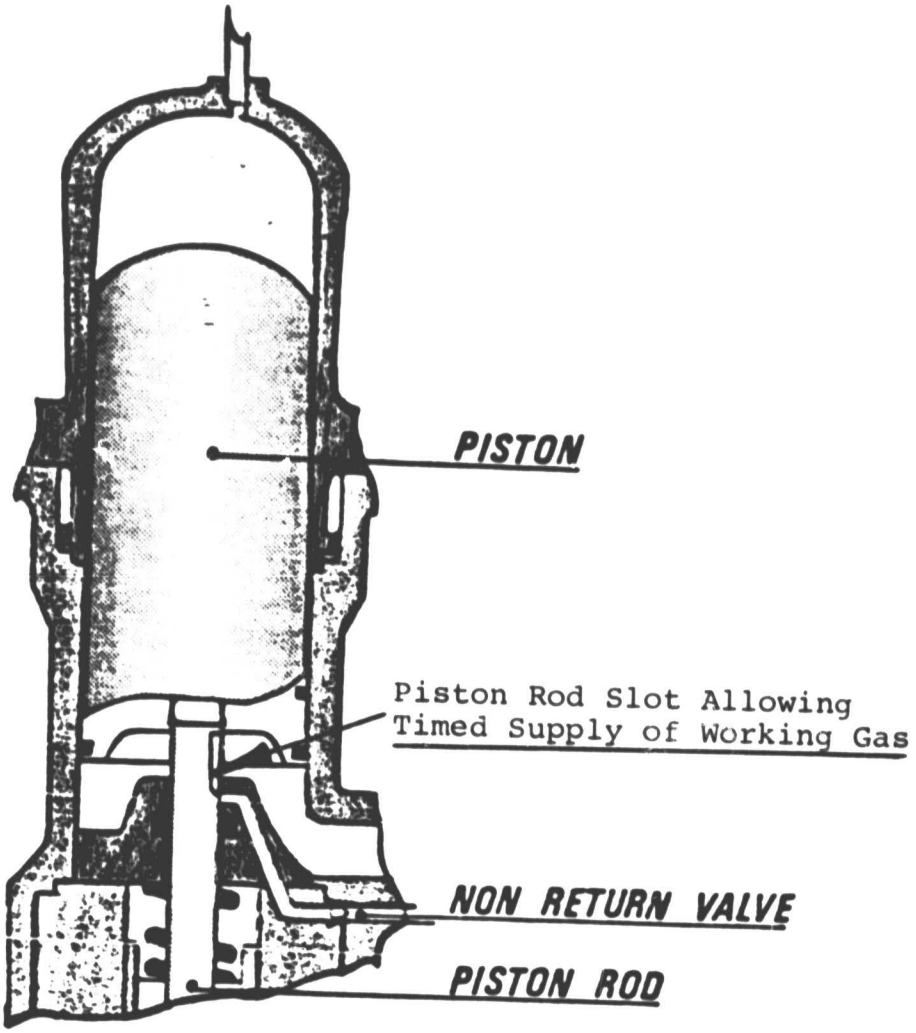


Fig. 8.4:6 Principle of USSw Timed Supply System

8.4.4 Power Control System Specification

The ASE MOD I power control system will follow the general MOD I specification regarding temperatures, pressures etc.

In addition to that, the following more specific goals have been aimed for:

A. The power control shall be designed in such a way that the driveability of the engine/vehicle combination is subjectively acceptable, and the switching between Stirling powered vehicles and IC powered vehicles is judged to introduce no unacceptable operator adjustments.

B. Stable oscillations, noise and/or discontinuous changes in controlled variables which would be noticeable or objectionable to the operator shall not occur.

C. Environmental variation encountered in the automatic vehicle application shall not be the cause of unsatisfactory control performance.

D. Safety provisions shall be incorporated in the power control system to prevent pressure vessel rupture in case of pressure buildup due to fire or malfunctions, to protect the engine from damage due to abnormal operating conditions, and to guard against engine runaway (failsafe).

E. Control system ducts and electrical leads shall be so routed and protected that no damage can be sustained due to accidental impacts during the course of routine operation and maintenance.

F. Response Specifications

At any constant speed the changes from no load power to full load power shall occur with the following

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guidelines:

1. The torque at the drive shaft shall reach 90% of final torque within 0.8 seconds. Fig 8.4:7.

2. A lag time or a torque dip in the up power maneuver shall be terminated in less than 0.3 seconds, and shall not be greater than 10% of final torque magnitude.

3. During power decrease, the torque at the drive shaft shall drop to 20% of the initial valve in less than 0.3 seconds (Fig 8.4:8).

4. During steady state (fixed set point and load), engine torque variation shall not exceed ± 3% of the set point torque value.

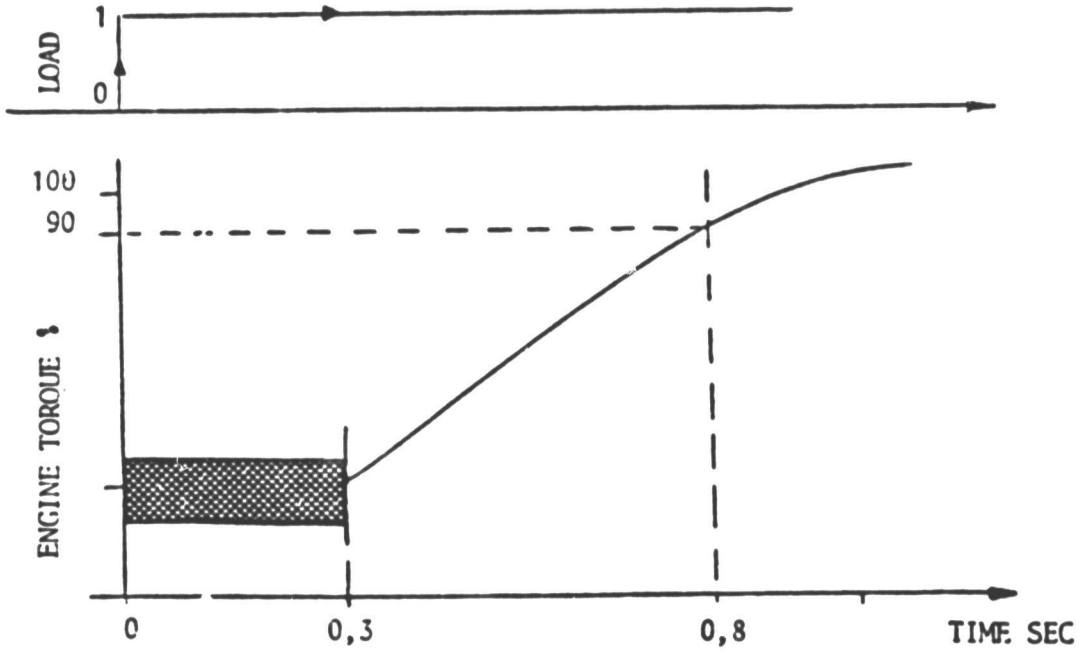


Fig 8.4:7 Torque response specification during torque increase

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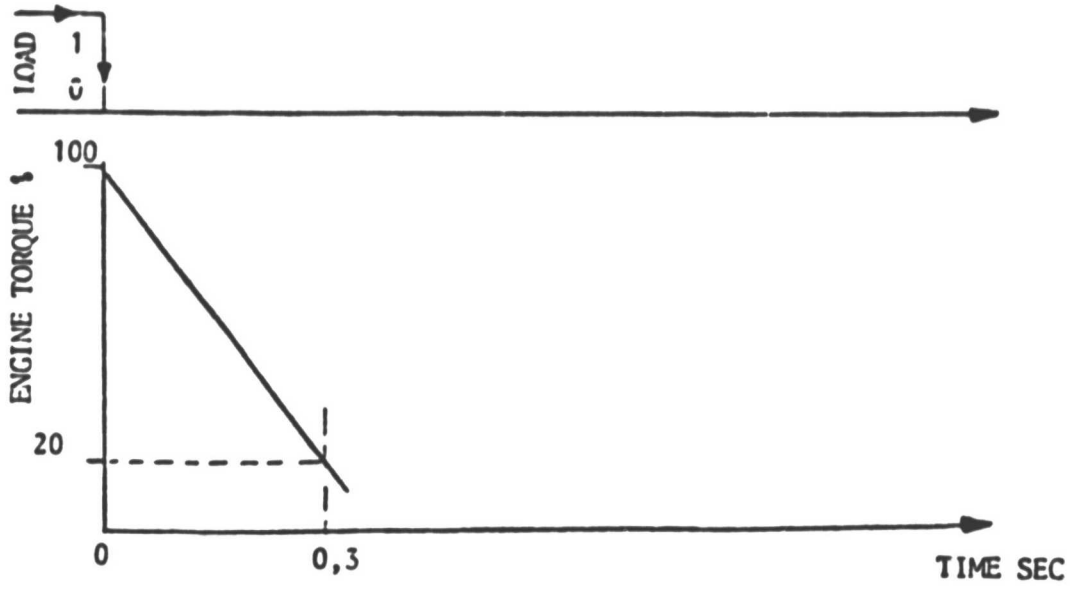


Fig 8.4:8 Torque response specification during torque decrease

8.4.5 Design Criteria and Improvements of P40 Power Control System

Scope of ASE MOD I power control design:

- Reduced number of external seals
- Reduced number and complexity of tubing
- Improved component performance and reliability

Results:

Compared with P40 power control system ASE MOD I power control system has modular block design of components resulting in:

- 48% less external seals
- Total number of external gas tubes reduced by 54%
- Number of tubes outside $\hat{p}$, $\check{p}$ and p_{supply} connection reduced by 74%
- Much simplified tubing layout

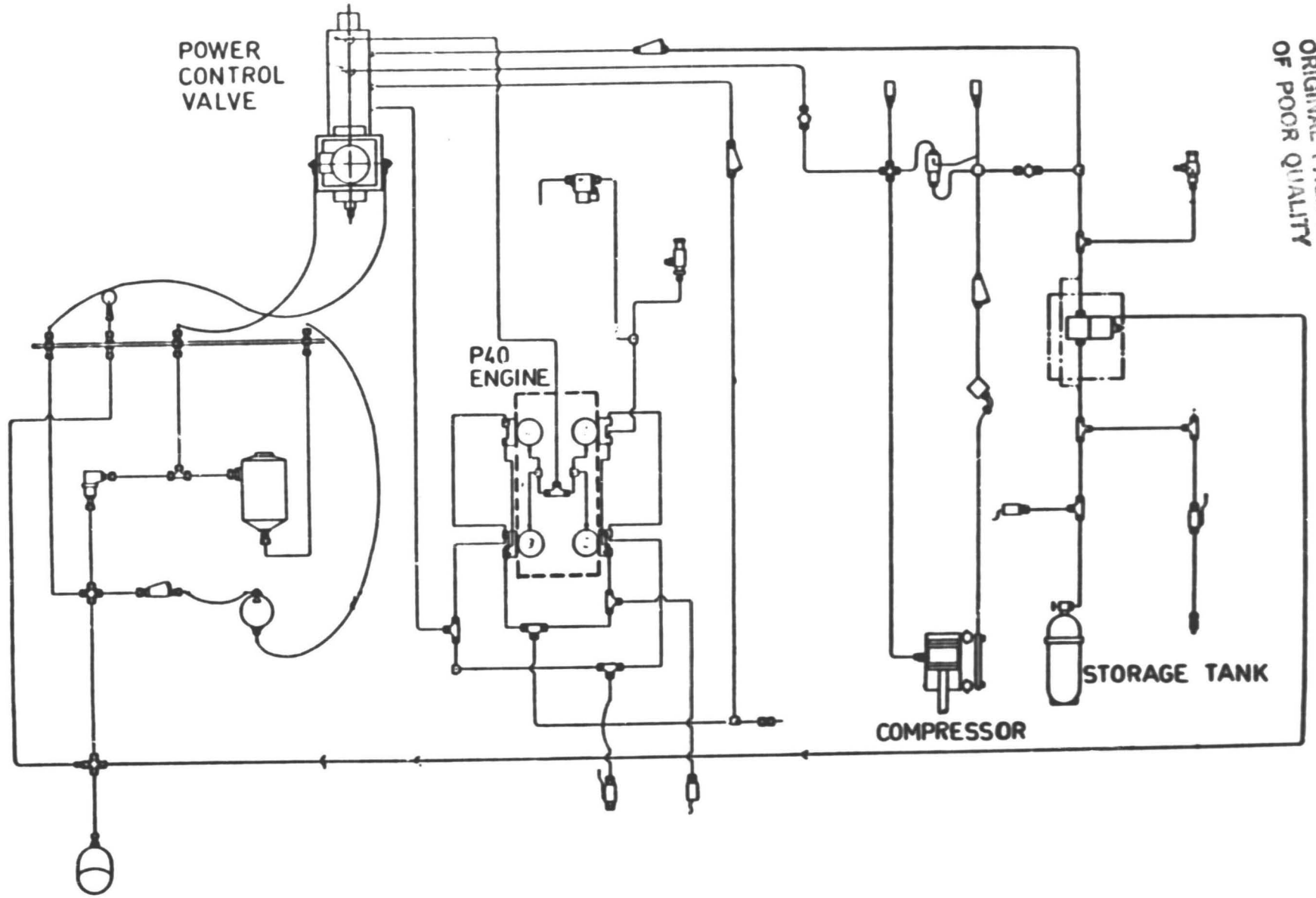
(Fig 8.4:9 and Fig 8.4:10 show the reduction in tubing between P40 and ASE MOD I.)

New check valve design with an accumulated testing time of 600 hours in test rig and 2 000 hours of engine operation.

New spool valve design with lower friction forces and simplified installation capabilities also with 2 000 hours of engine operation.

Special built servo-actuator in a cost-reducing design.

Custom designed plug in solenoid valves built to USSw specification.



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Fig. 8.4:9 Tubing Arrangement of the P-40 Power Control System

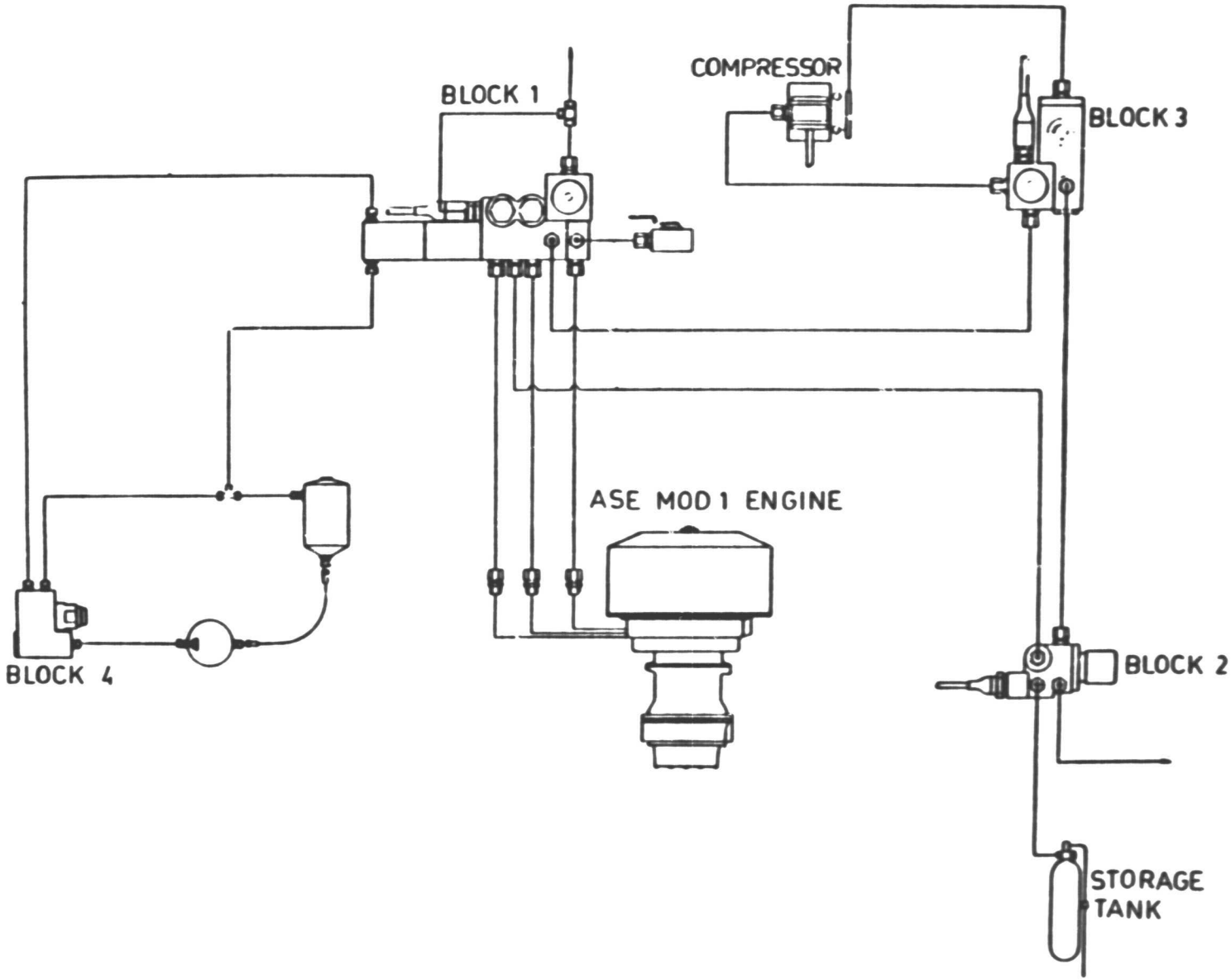


Fig. 8.4:10 Tubing Arrangement of the ASE Mod I Power Control System

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8.4.6 The Modular Blocks

All major components of the ASE MOD I power control system have been built into four modular blocks.

Fig 8.4:11 shows the modular block boundaries of ASE MOD I power control system.

Block 1 incorporates:

- Power control spool valve
- Electrohydraulic servo-actuator
- Safety valve
- External dumping solenoid valve
- Max pressure transducer
- Hydrogen filter
- Check valve

Block 2 incorporates:

- Safety valve
- Hydrogen tank pressure transducer
- Shut-off solenoid valve
- Hydrogen filter

Block 3 incorporates:

- Hydrogen filter
- Hydrogen cooler
- Check valve
- Compressor short circuiting solenoid valve
- Compressor pressrue transducer

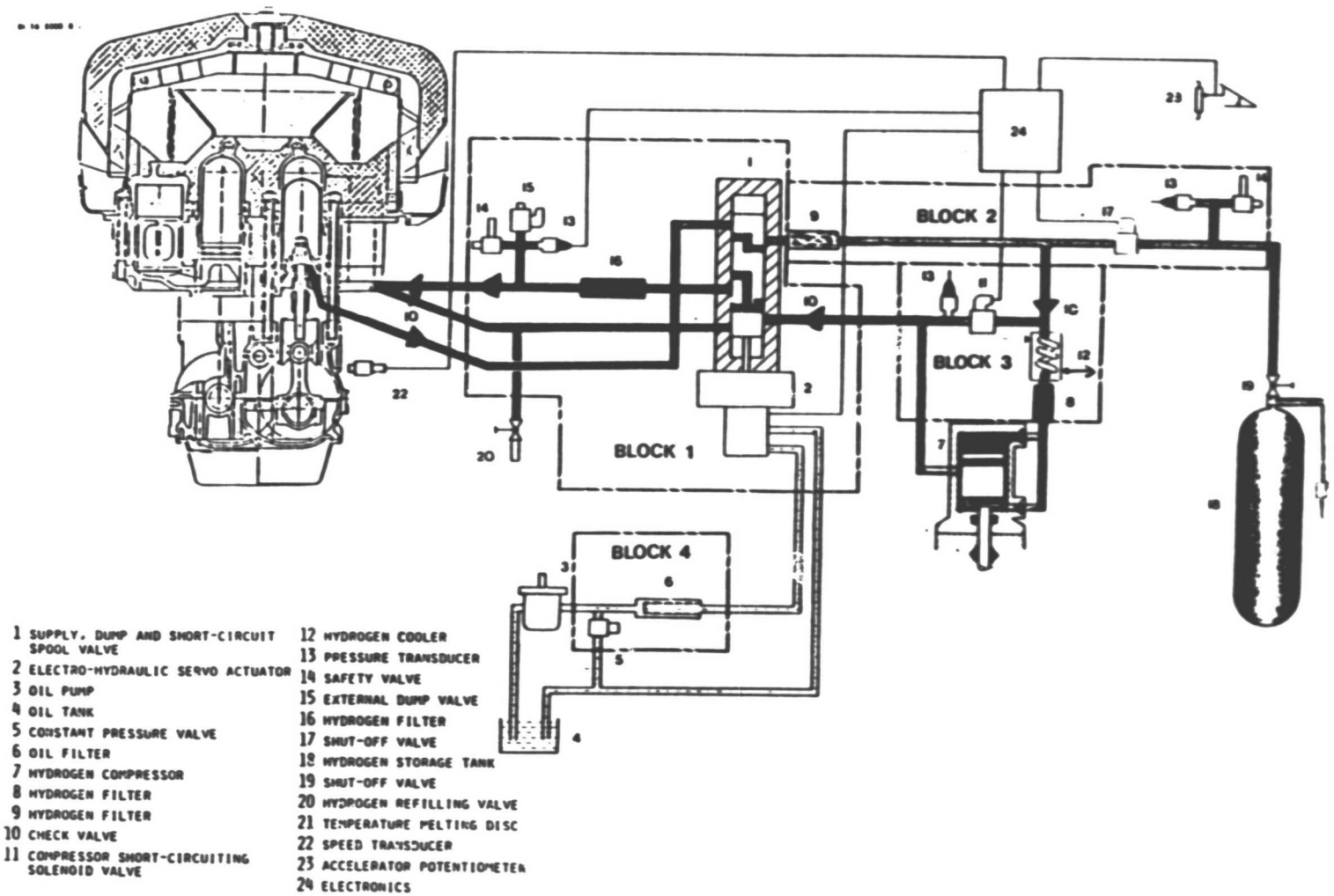


Figure 8.4:11 ASE Mod I Power Control System
Modular Block Boundary Lines

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Block 4 incorporates:

- servo oil constant pressure valve
- servo oil filter

Fig 8.4:12-15 show the design of each block (see also section 8.4.9 Drawings).

Materials and Weights:

	Material	Weight
Block 1	SS 2172	7.1 kg
Block 2	SS 2172	3.2 kg
Block 3	SS 2337	5.3 kg
Block 4	SS 4212	0.8 kg

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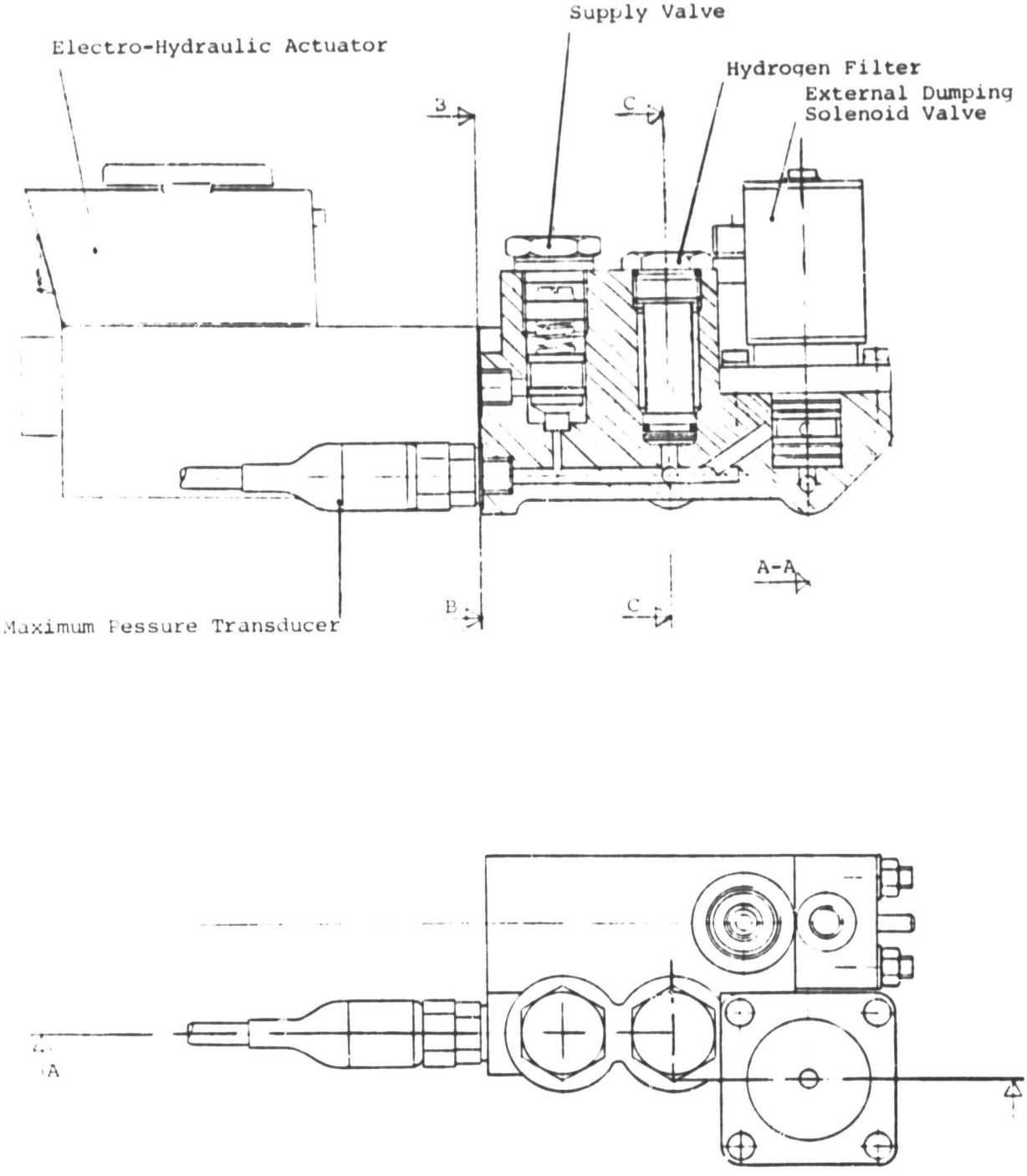


Fig 8.4:12 ASE MOD I. Power Control Block 1.

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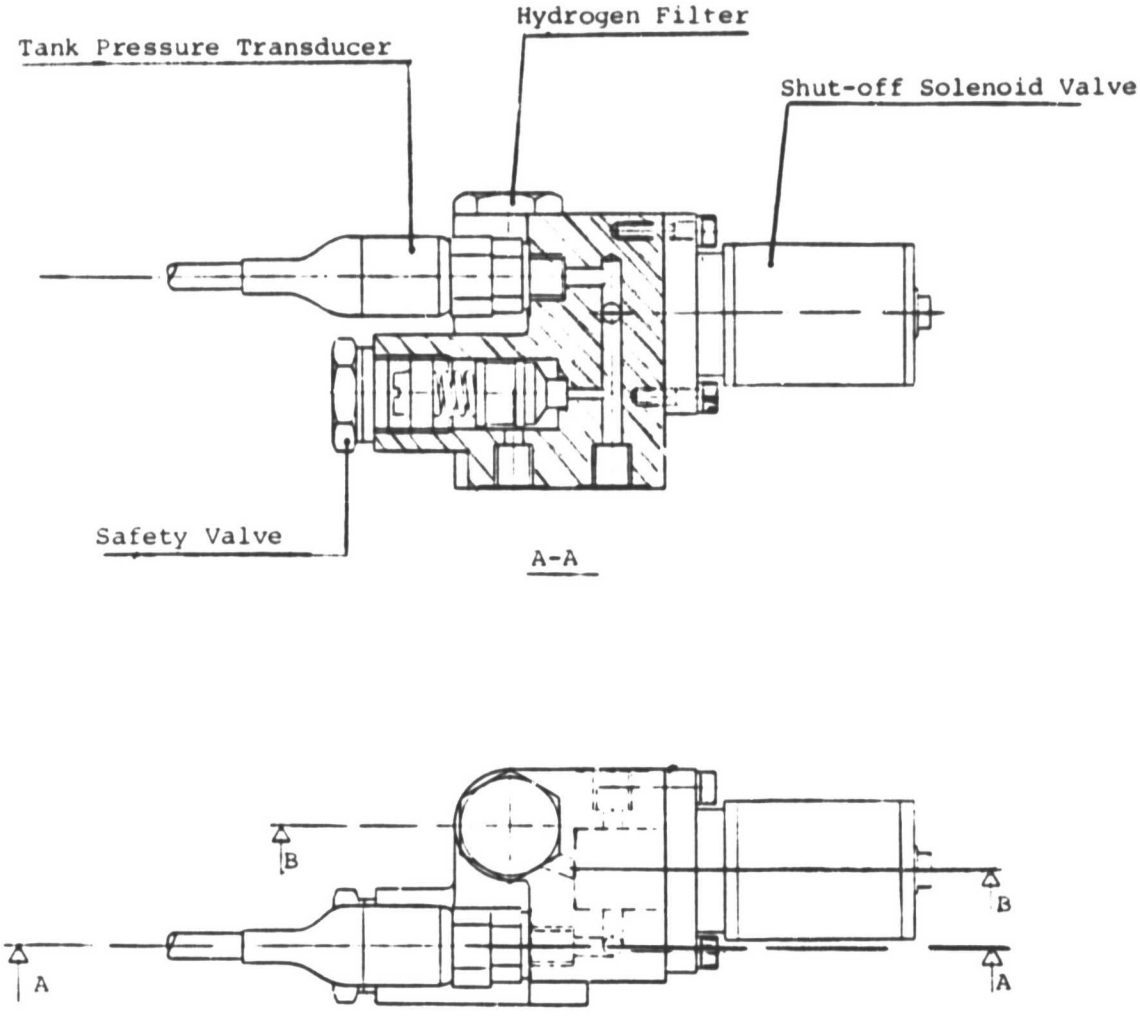


Fig 8.4:13 ASE MOD I Power Control Block 2.

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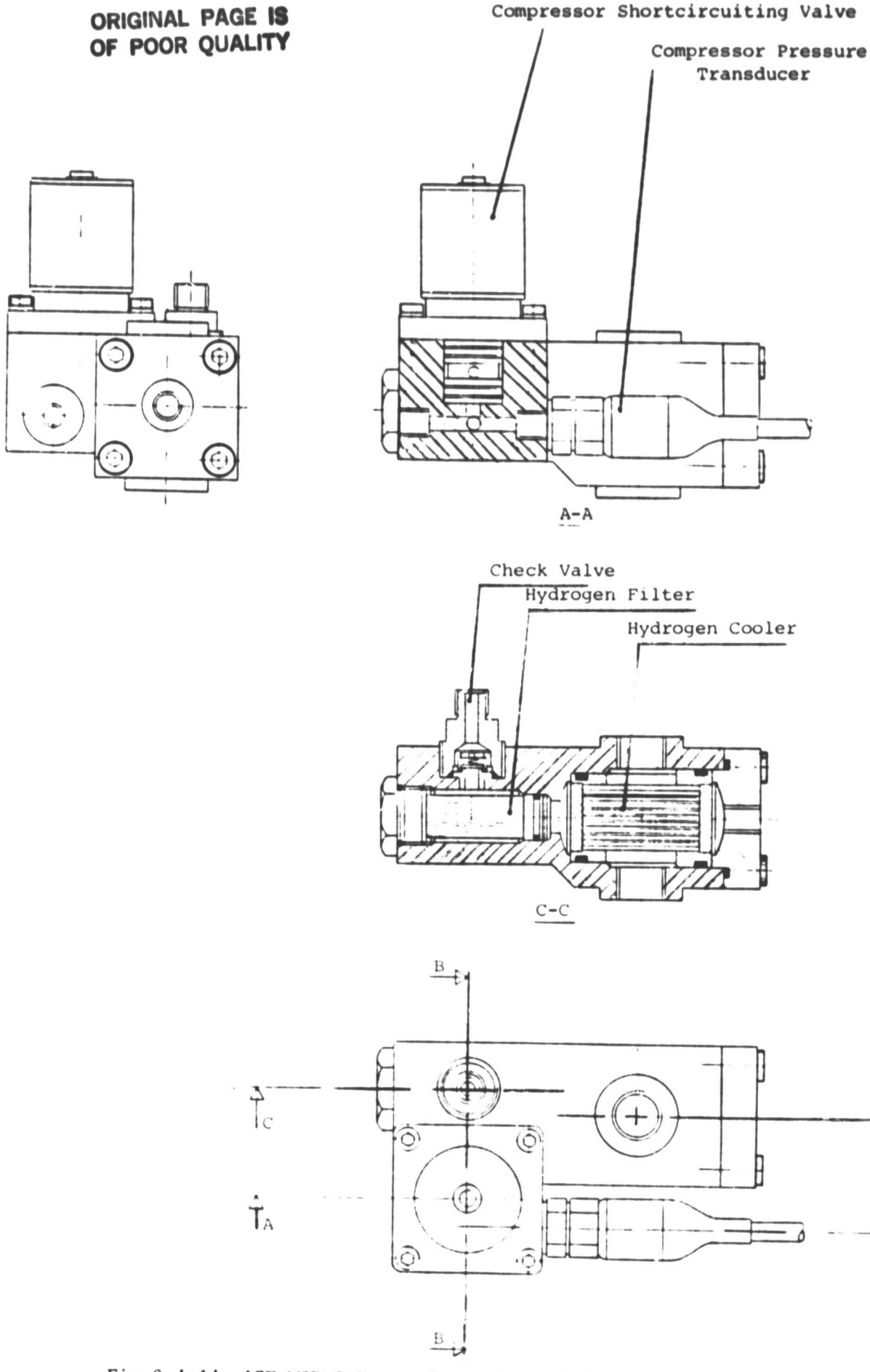


Fig 8.4:14 ASE MOD I Power Control Block 3.

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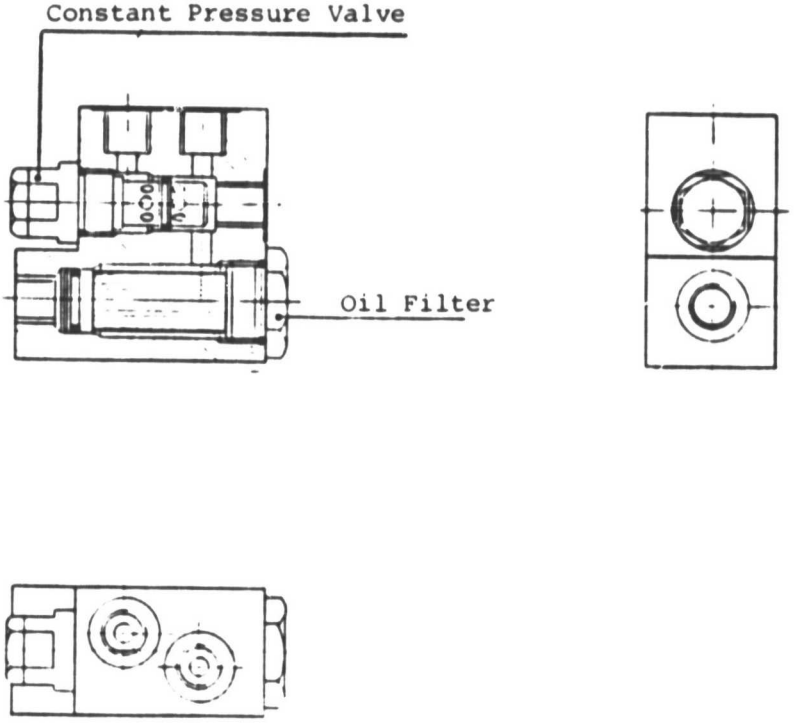


Fig 8.4:15 ASE MOD I Power Control Block 4.

8.4.6.1 Stress Analysis of Power Control Block 1 - 4

A. Power Control Block 1

a. Drawings

The analysis is based on the geometry showed in drawing No 1-S-1470 (800304), see section 8.4.9.

b. Material

Steel SS2172 is assumed in the analysis and the following material property data have been used.

Yield stress: σ_y = 300 MPa
Ultimate tensile strength: σ_u = 490 MPa
Fatigue limit $\sigma_f = \pm$ 210 MPa
$\sigma_{fp} = (185 \pm 185)$ MPa

c. Loads

The maximum temperature is 100° C and different internal parts of the block will be exposed to the following pressure loads:

p = (12 ± 7) MPa
p = (7.8 ± 4.8) MPa
p = (21.5 ± 3.5) MPa

In the static stress analysis the maximum internal pressure (p_{max} = 25 MPa) is used, and in the fatigue analysis the pressure cycle with the maximum amplitude p = (12 ± 7) MPa is used.

The total number of pressure cycles is estimated to 2.5×10^6, but in the analysis the corresponding stress cycles will be compared to the fatigue limit.

d. Stress Analysis

The gas pressure in the channels of the block results in stresses, which will be estimated by considering the material around each channel as an equivalent thick walled cylinder exposed to an internal pressure. In most cases the pressurized thick cylinder has a hole in the cylinder wall, and the stress concentration factor is then taken from R E Petersson: Stress Concentration Factors. The stress analysis is summed up in the table in fig 8.4:17 and the following notations have been used:

R_o = outer radius of the equivalent thick walled cylinder

R_i = inner " " " " " " "

$k = R_o/R_i$

p = internal pressure

σ_n = nominal hoop stress ($\sigma_n = p \cdot \frac{k^2 + 1}{k^2 - 1}$)

K_t = stress concentration factor

Point = point used in the fatigue analysis (showed in figure 8.4:16)

S_f = safety factor against fatigue

The stress cycles in point 1 - 5 have been entered in the Haigh diagram in fig 8.4:17 and the corresponding safety factors can be seen in the table in the same figure.

e. Summary

The minimum safety factor against yielding is 6.4 and the minimum safety factor against fatigue is 3.3.

B. Power Control Block 2

a. Drawings

The analysis is based on the geometry shown on drawing No 2-S-1471(800306), see section 8.4.9.

b. Material

Steel SS2172 is assumed in the analysis, and the following material property data have been used.

Yield stress: σ_y = 300 MPa

Ultimate tensile strength: σ_u = 490 MPa

Fatigue limit: $\sigma_f = \pm 210$ MPa

$\sigma_{fp} = (185 \pm 185)$ MPa

c. Loads

The maximum temperature is 100° C and the internal parts of the block will be exposed to the following maximum pressure load: $p = (21.5 \pm 3.5)$ MPa. The total number of pressure cycles have been estimated to 2.5×10^6, but in the analysis the corresponding stress cycles will be compared to the fatigue limit.

d. Stress Analysis

The analysis has been done in the same way in the case of block 1 and the result is shown in the table in figure 8.4:19. The points, which have been analyzed can be seen in figure 8.4:18, and the stress cycles have been entered in the Haigh diagram.

e. Summary

The minimum safety factors against yielding and fatigue are 4.9 and 2.1 respectively.

C. Power Control Block 3

a. Drawings

The analysis is based on the geometry shown on drawing No 1-S-1480 (800314), see section 8.4.9.

b. Material

Stainless steel SS2337 (AISI 321) is assumed in the analysis, and the following material property data have been used.

Yield stress: σ_y = 174 MPa

Ultimate tensile strength: σ_u = 490 MPa

Fatigue limit: $\sigma_f = \pm 270$ MPa

$\sigma_{fp} = (240 \pm 240)$ MPa

Brinell hardness HB = 190

c. Loads

The maximum temperature is 100° C and the internal parts of the block will be exposed to the following maximum pressure load: p = (17.5 ± 7.5) MPa. The total number of pressure cycles have been estimated to 2.5×10^6, but in the analysis the corresponding stress cycles will be compared to the fatigue limit.

d. Stress Analysis

The analysis has been done in the same way as in the case of block 1, and the result is shown in the table in figure 8.4:21. The points which have been analyzed can be seen in figure 8.4:20 and the corresponding stress cycles have been entered in the Haigh diagram.

The endplate, see fig 8.4:20, is considered as a simple supported circular plate exposed to an uniform pressure.

Using the maximum pressure p = 25 MPa gives it the maximum stress σ_n = 86 MPa with a stress concentration factor of K_t = 2.0, the following stress cycle σ = (120 ± 51) MPa is obtained which has been entered in the Haigh diagram. Thus, the safety factors against yielding and fatigue are 2.0 and 2.8 respectively.

The bolts in the endplate have been designed to fulfill the requirements of VDI 2230 (Verein Deutscher Ingenieure, "Systematische Berechnung hochbeanspruchter Schraubenverbindungen")

The precedure described in VDI 2230 gives the following values of the main parameters used in the analysis.

Maximum gas pressure p_{max} = 25 MPa

Minimum - " - p_{min} = 10 MPa

Number of bolts i = 4

Maximum bolt load due to the gas pressure F_{AO} = 12 300 N

Minimum - " - F_{AU} = 4 900 N

Load relation Φ_{en} = 0.2

where

$$\Phi = \frac{1}{\frac{k_c}{k_b} + 1} \quad \text{and}$$

k_b = spring rate of the bolt

k_c = spring rate of the bolted material

The coefficient of friction used μ = 0.14.

The calculation precedure shows that M10 bolts of quality 12.9 will meet the requirements of VDI 2230.

e. Summary

The minimum safety factors against yielding and fatigue are 2.0 and 2.8 respectively. The endplate bolted with 4 x M10 of quality 12.9 meets the requirements of VDI 2230.

D. Power Control Block 4

a. Drawings

The analysis is based on the geometry shown on drawing 3-S-1452, see section 8.4.9.

b. Material

Aluminum SS4212 is assumed in the analysis and the following material property data has been used.

Yield stress: σ_y = 250 MPa

c. Loads

The maximum temperature is 100° C and the internal parts of the block will be exposed to a constant gas pressure p = 5 MPa, and consequently only a static analysis needs to be done.

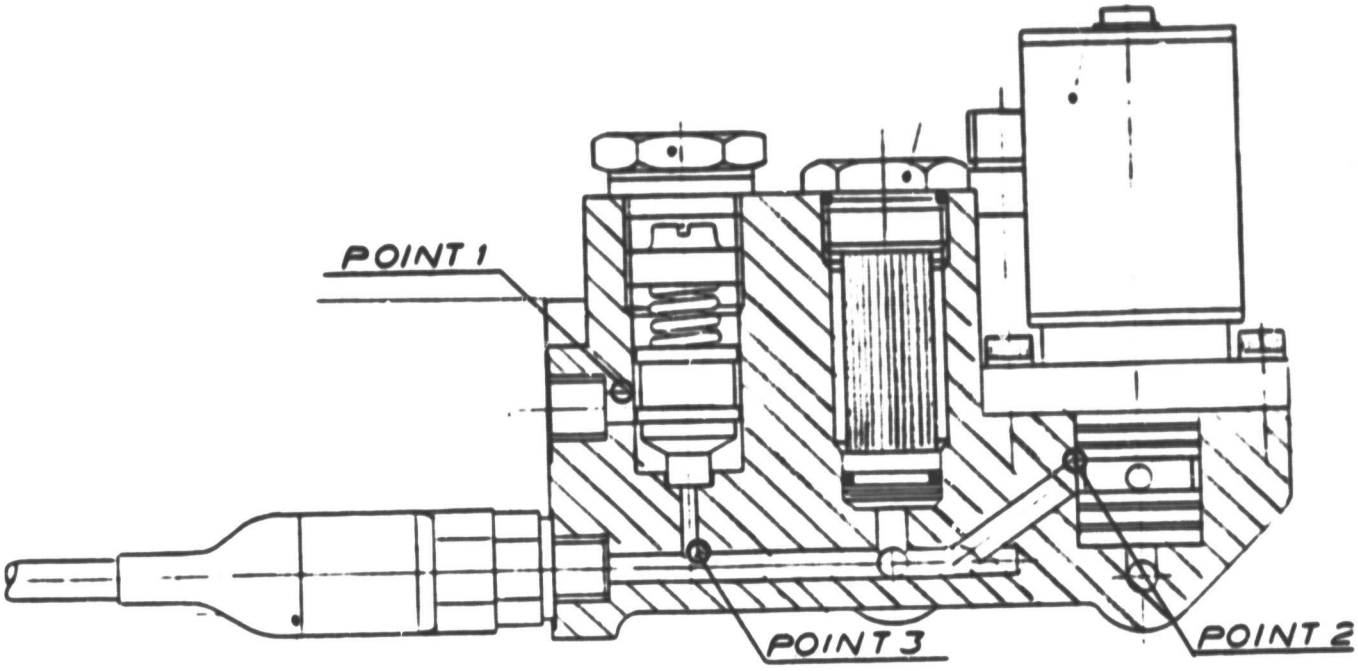
d. The analysis has been done in the same way as earlier and the stress levels in the equivalent thick walled cylinder, see figure 8.4:22, are shown in the table below.

	Cylinder	
	1	2
R_o (mm)	18	20
R_i (mm)	11	9
k	1.64	2.22
σ (p) (MPa)	11	8
S_f	23	31

e. Summary

As can be seen in the table, due to the low pressure, the safety factors against yielding are very high.

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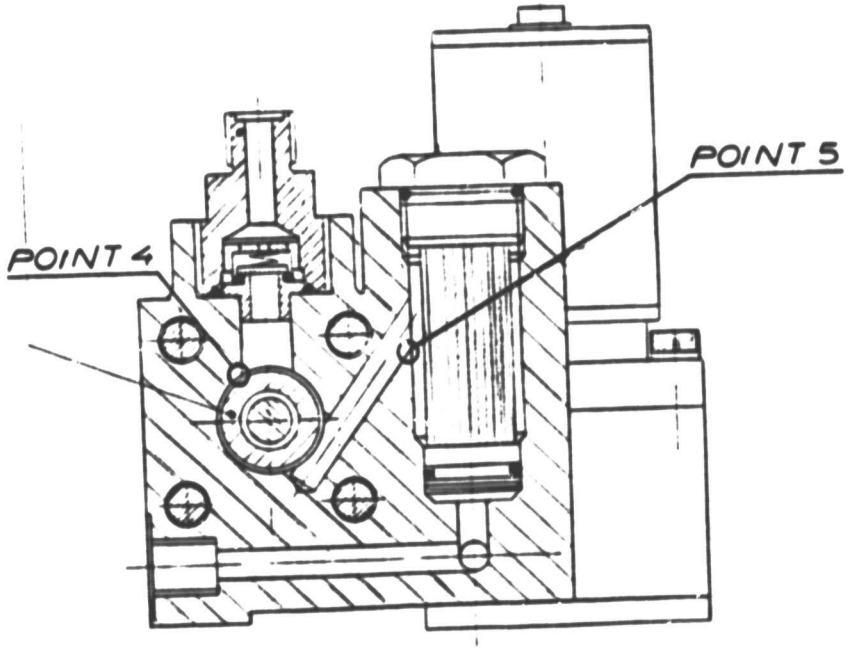


Fig 8.4:16 Points on block 1 used in the stress analysis

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TABLE 1

Point	R_o (mm)	R_i (mm)	K	$\sigma_m(P_{max})$ (MPa)	σ_y/σ_m	K_f	$\sigma(p)$ (MPn)	S_f
1	20	11	1.82	47	6.4	3.34	75 ± 44	3.3
2	31	16	1.94	43	7.0	3.40	71 ± 41	3.5
3	9	3	3.0	31	9.7	2.84	43 ± 25	5.9
4	25	11	2.27	37	8.1	3.44	61 ± 36	4.1
5	20	11	1.82	47	6.4	3.34	75 ± 44	3.3

Haigh Diagram SS2172

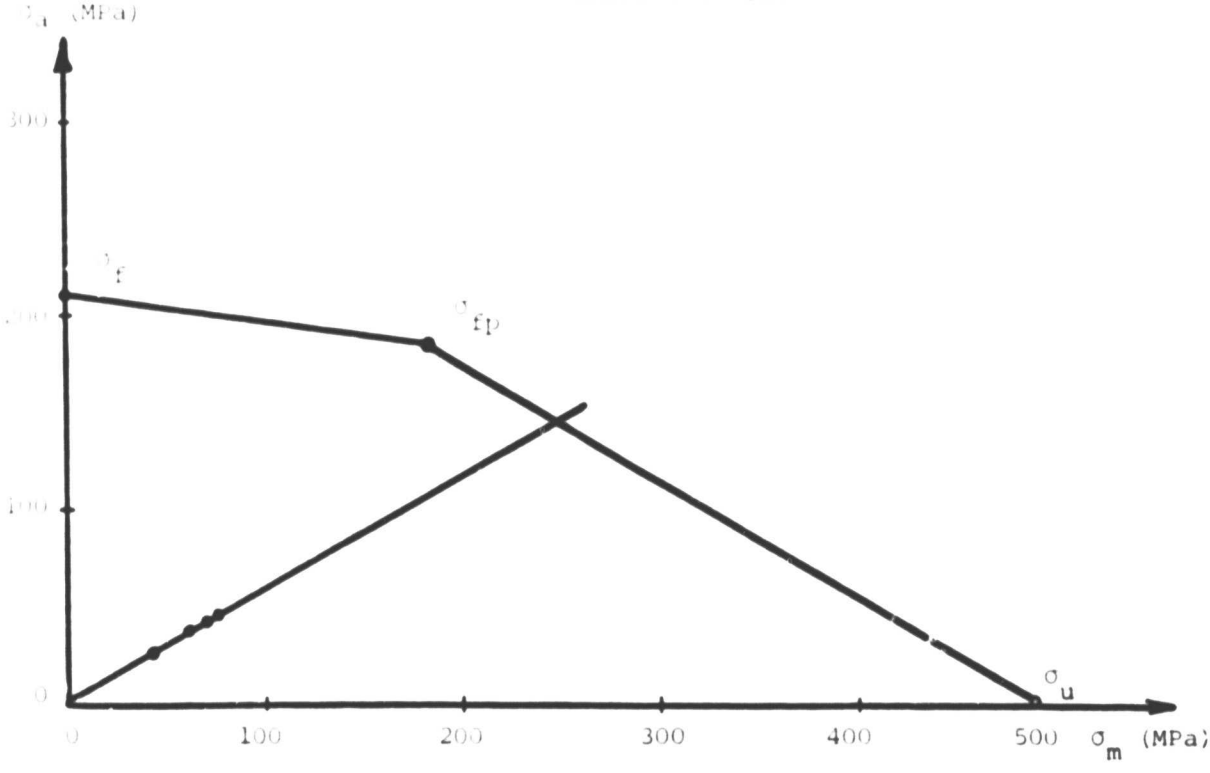


Fig 8.4:17 Results from the stress analysis of block 1.

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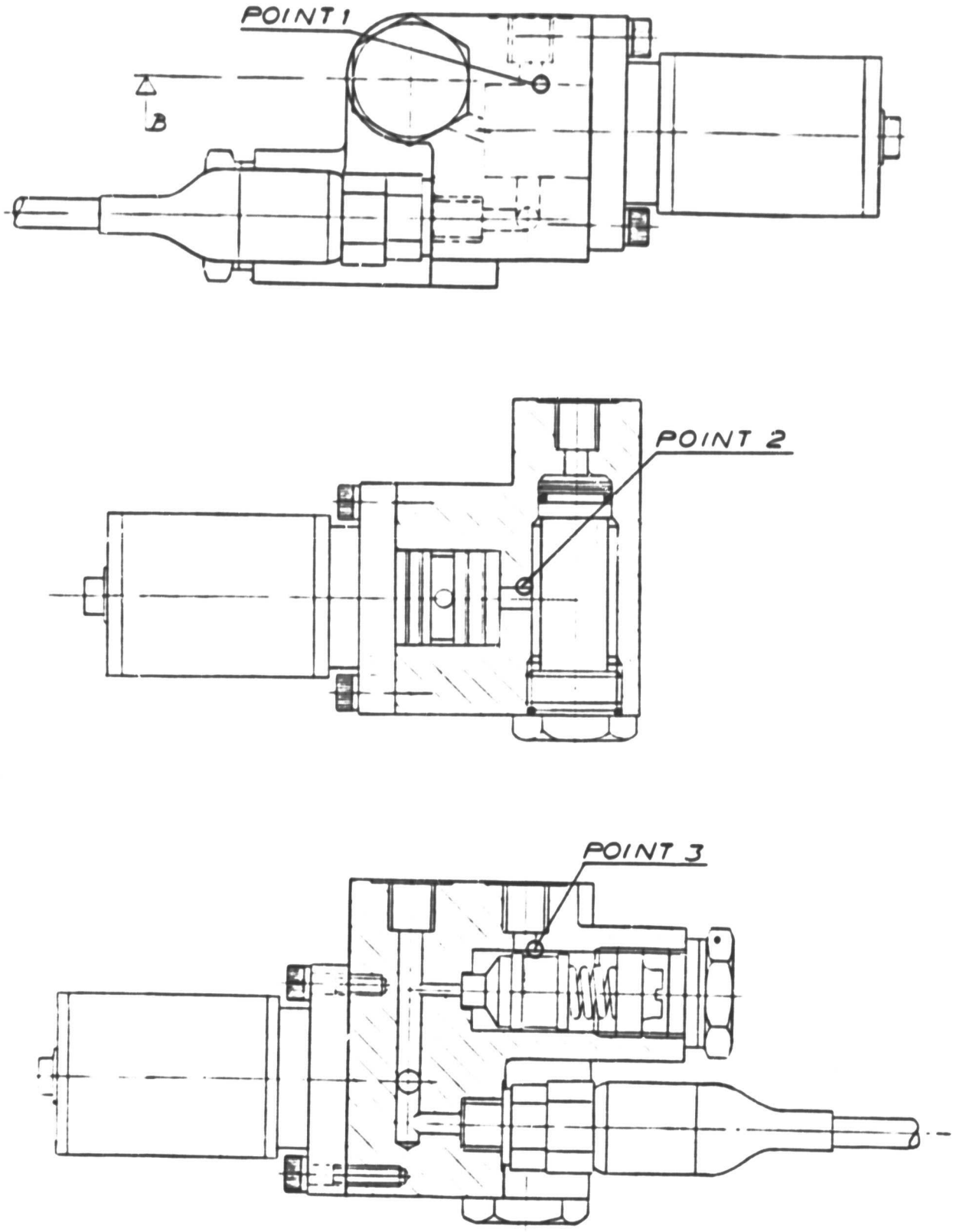


Fig 8.4:18 Points on block 2 used in the stress analysis

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TABLE 2

Point	R_o (mm)	R_i (mm)	K	$\sigma_m(P_{max})$ (MPa)	σ_y/σ_n	K_f	$\sigma(P)$ (MPa)	S_f
1	31	16	1.94	43	7.0	3.44	129 ± 21	3.0
2	17	11	1.55	61	4.9	3.54	185 ± 30	2.1
3	18	11	1.64	55	5.5	3.50	166 ± 27	2.3

Haigh Diagram SS2172

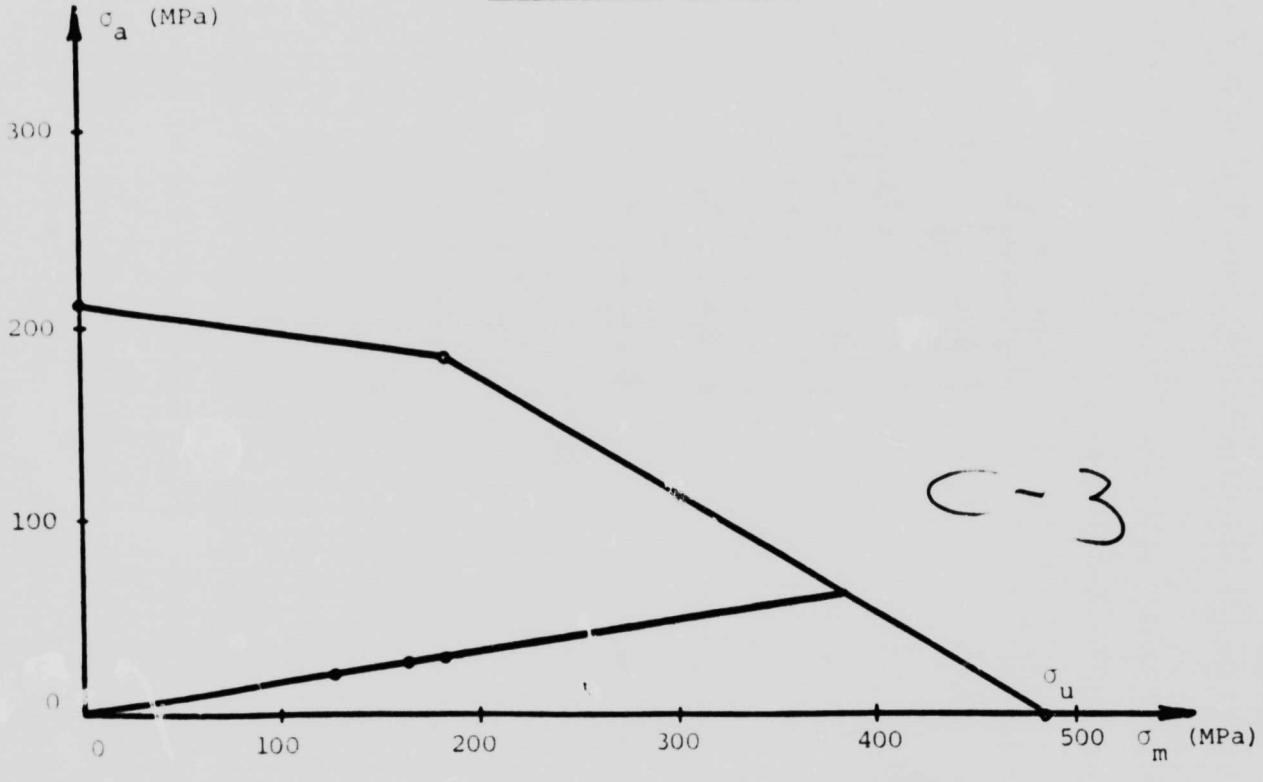


Fig 8.4:19 Results from the stress analysis of block 2.

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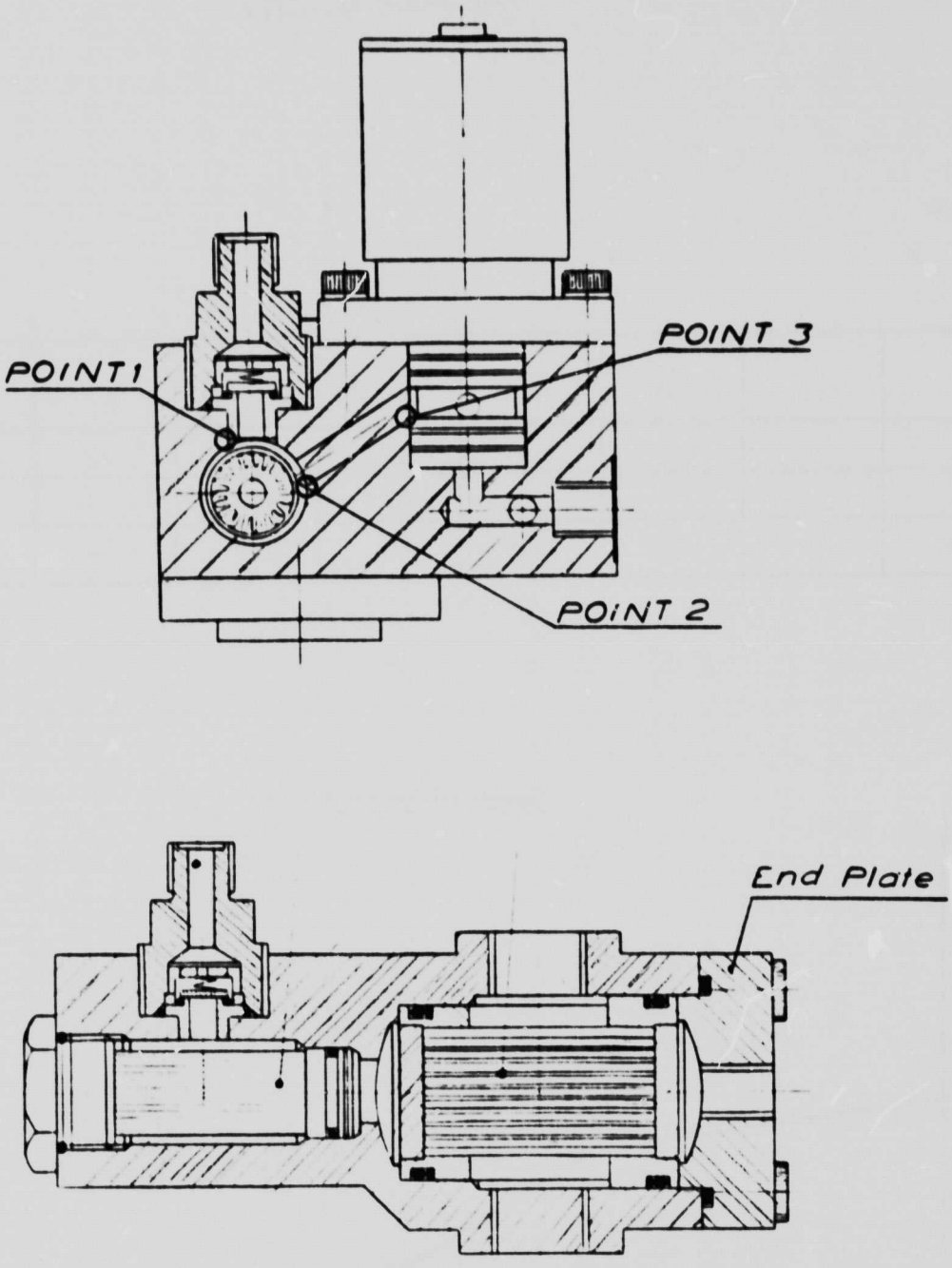


Fig 8.4:20 Points on Block 3 Used in the Stress Analysis.

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TABLE 3

Point	R_o (mm)	R_i (mm)	K	$\sigma_m(P_{max})$ (MPa)	σ_y/σ_n	K_f	$\sigma(P)$ (MPa)	S_f
1	20	11	1.82	47	3.7	3.28	107 ± 46	3.2
2	20	11	1.82	47	3.7	3.34	109 ± 47	3.1
3	31	16	1.94	43	4.0	3.41	116 ± 50	3.0

Haigh Diagram SS2337

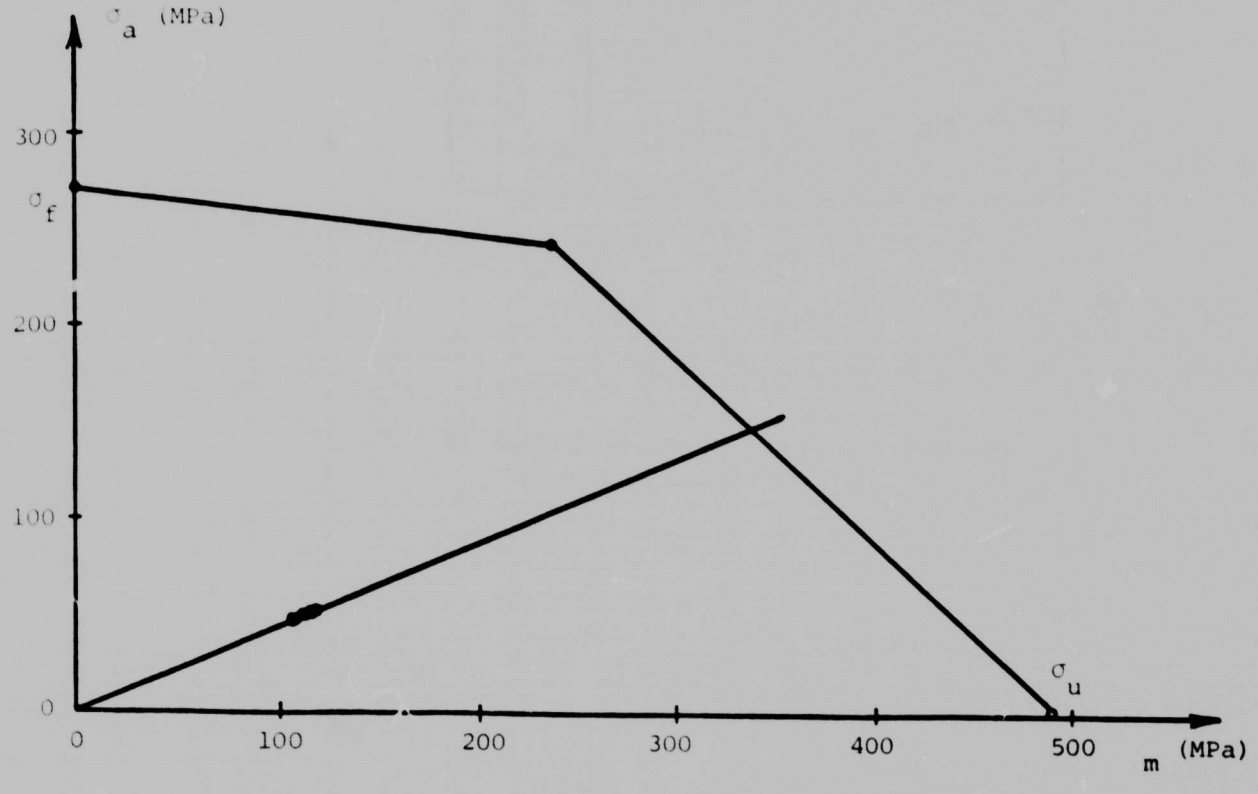


Fig 8.4:21 Results from the Stress Analysis of Block 3.

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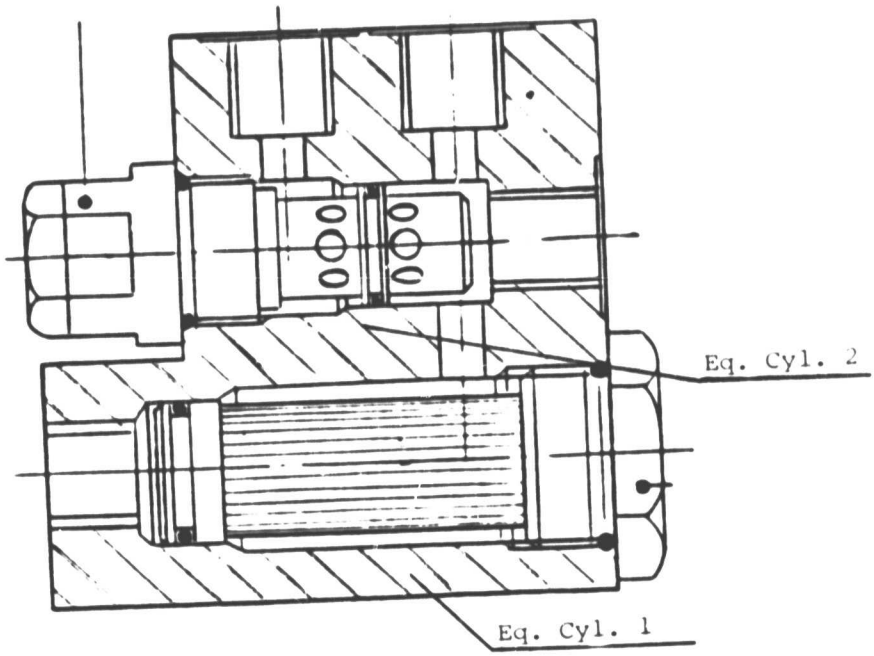


Fig. 8.4:22 Parts of Block 4 Used in the Stress Analysis

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8.4.7 Tubes and Fittings

Most tubes of the power control system will be 8 mm outer diameter and 6 mm inner diameter. Somes tubes will have a 6 mm O.D. and a 4 mm I.D.

All tubes will be made of stainless steel SS 2353.

All tube fittings will be of the same design as in P-40. That means a welded nipple and an O-ring seal (Figure 8.4:23).

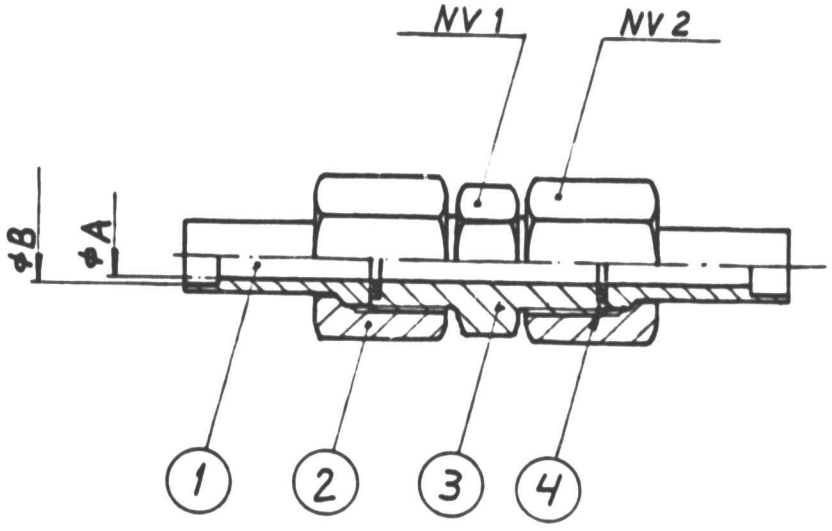


Fig. 8.4:23 USSw Hydrogen Tube Fitting

The fittings will be made of stainless steel SS 2353.

8.4.8 Power Control Components

All component numbers refer to Fig 8.4:24 (Block scheme of the ASE MOD I Power control system).

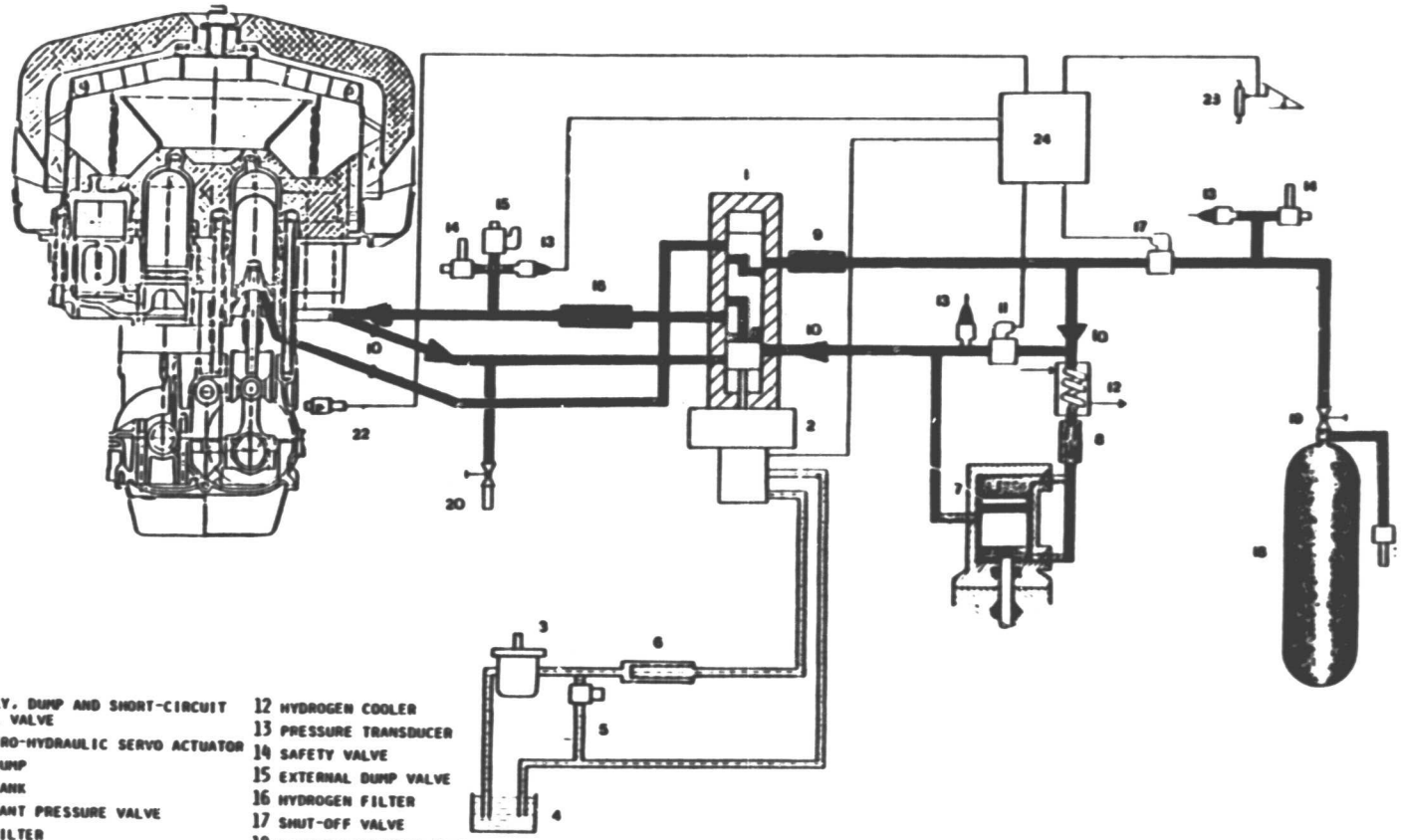


Figure 8.4:24 Block Scheme of the ASE Mod I Power Control System

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8.4.8.1 Supply-Dump and Short Circuiting Spool Valve (1)

To perform the main functions of the mean pressure control as supply, dump and short-circuiting the power control system of ASE MOD I (as well as P40) is equipped with a sliding spool valve. The ASE MOD I valve has the same characteristics as the P40 valve, but is designed differently. To be able to mount the sliding spool valve in a modular block the ASE MOD I valve has the sliding seals on the spool instead in the housing. The design of the spool can be seen in Fig. 8.4:25. A P-40 valve is also shown.

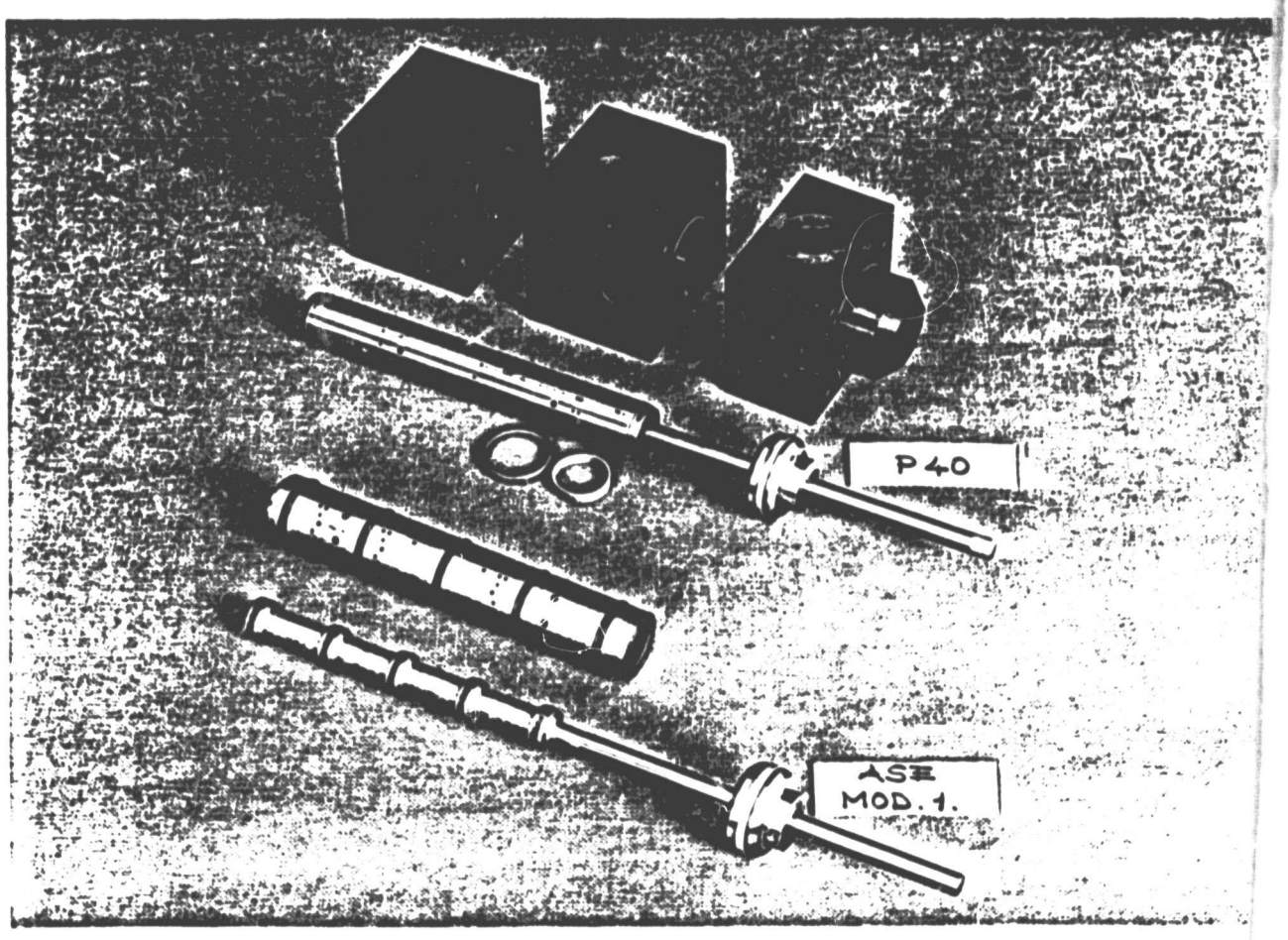


Fig 8.4:25 ASE MOD I control valve spool and P40 control valve spo

This design reduces the spool dimensions and seal sizes. Friction forces with this kind of spool valve will be lower.

This spool valve is mounted in power control block 1, where it will be controlled by the electro-hydraulic actuator.

The opening pattern of the vale is shown in Fig. 8.4:26.

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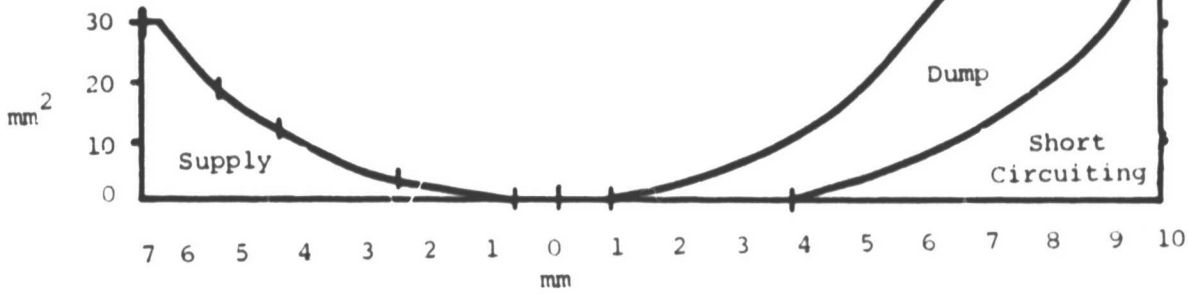


Fig 8.4:26 Opening pattern of the ASE MOD I sliding spool valve.

There is a dead band of 1.4 mm around neutral position. If the valve moves to the left, supply of hydrogen to the engine occurs. Max orifice area is about 30 mm^2. The exact shape of the opening curve is uncertain.

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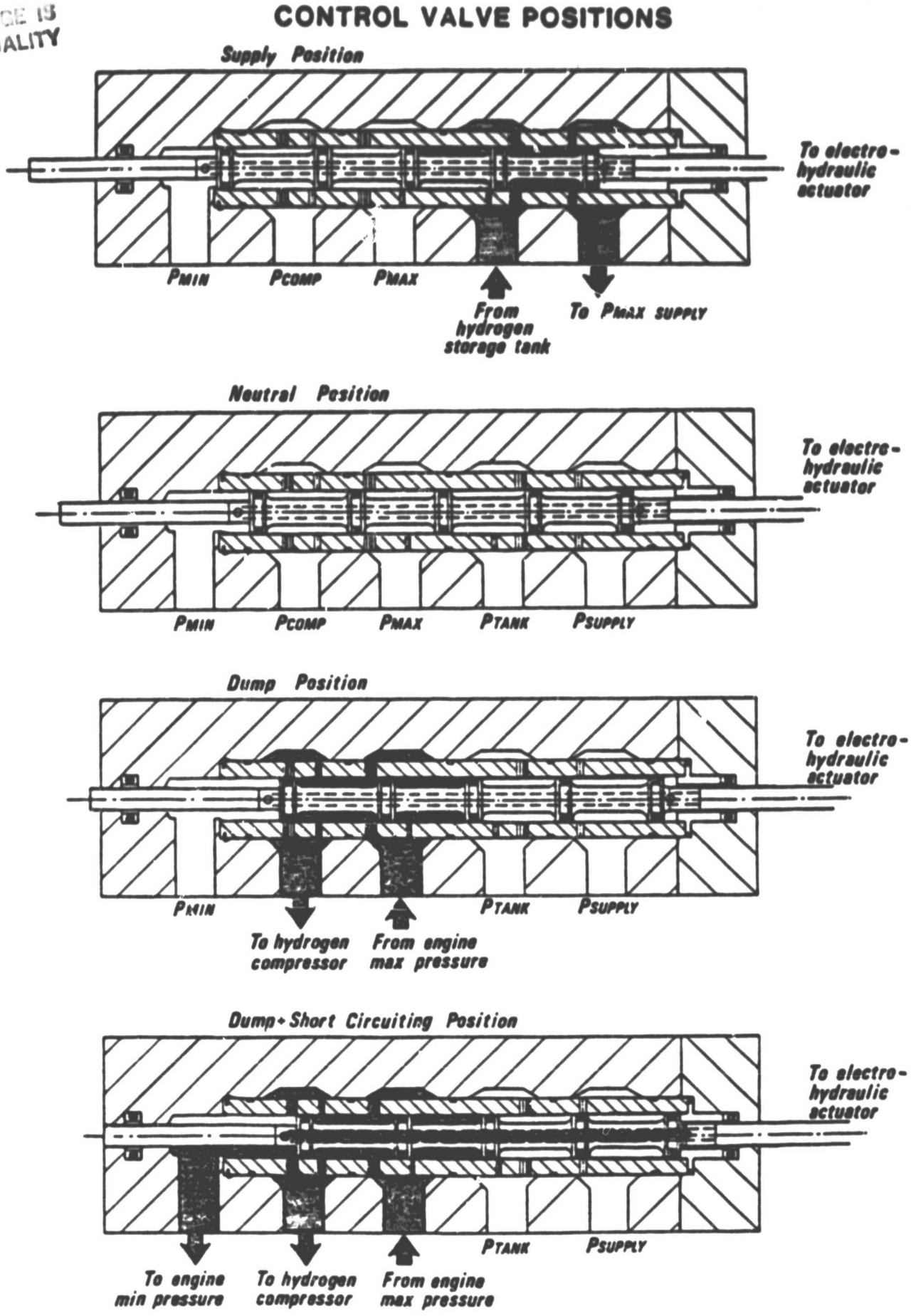


Fig 8.4:27 Flowpattern of the ASE MOD I Sliding Spool Valve.

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If spool moves to the right, dumping of hydrogen occurs. With another 3 mm of movement short-circuiting starts. Maximum orifice area is about 40 mm^2. The exact shape of these opening curves is uncertain.

Flow pattern through the valve during supply, dump and short-circuiting can be seen in Fig 8.4:27.

Sliding Seal Design

Special considerations have been made to avoid seal wear. All ports the seals have to pass during spool motion, have higher pressures than surrounding areas of the seal. Fig 8.4:28 shows what happens when a seal passes a hole.

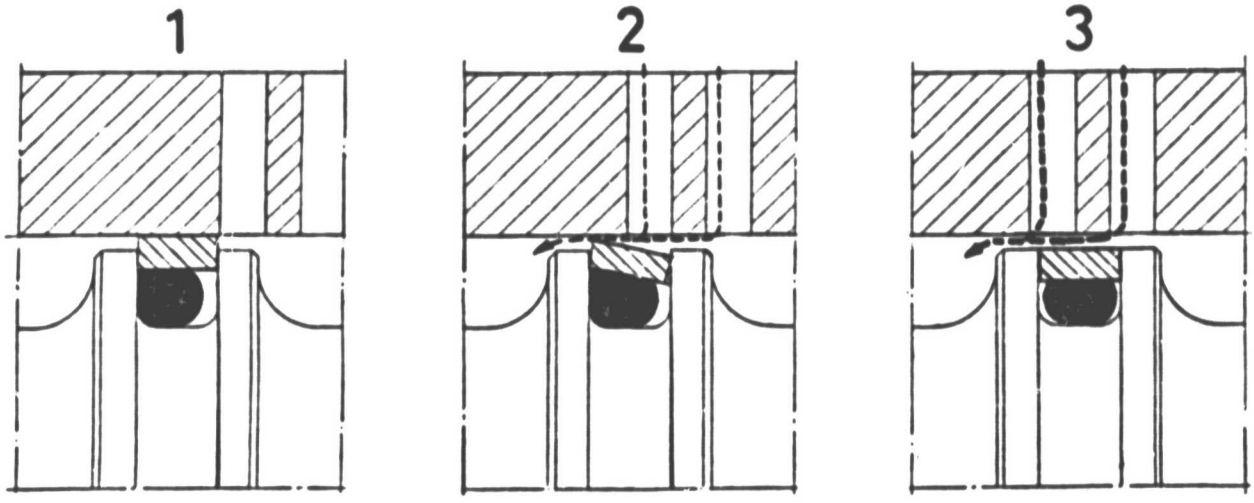


Fig. 8.4:28 Seal Passage of a Hole in Sliding Spool Valve

The exact position when the seal is lifted is uncertain and that is why the opening area curves in Figure 8.4:26 are uncertain.

Material

Slide spool: Brass 5170-04
Spool housing: Steel SS 2140 hardened to 60 HRC
Seals: Shamban S 30059-6-5

Experience

A test unit of the ASE Mod I spool valve has been testing together with P-40-4. The valve has been operating in cyclic tests between 5 and 10 MPa $\bar{p}$. The valve has logged more than 2000 hours and 144,000 cyclic variations with the same seals and housing. After 1900 hours the valve was removed and inspected. No significant wear could be seen.

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8.4.8.2 Electro-Hydraulic Actuator (2)

The actuator controls the motion of the spool valve. In supply and dump position the actuator controls the flow rate in or out from the engine (integrating functions). In short-circuiting mode it precisely controls the opening area of the short-circuiting holes. This is a proportional control where a certain area corresponds to a certain power level. Furthermore, it is an unstable control mode and the actuator speed and position accuracy must be very precise.

On P40 engines the actuator consists of a MOOG electro-hydraulic servo valve controlling servo oil flow to an oil piston. Position accuracy is quaranteed by a feed back potentionmeter.

This MOOG 76 pilot operated servo valve is a precision built unit which can not be built in quantities to much lower price. The performance of this valve is also superior to our demands.

Under ASE MOD I component development program MOOG Ltd (England) is developing a special Stirling engine actuator to our specification and based on the pilot stage of their directional control valve A65. The design goal has been to develop an actuator built to an industrial standard where mass production costs would be low. However, this emphasis makes the new actuator heavier than the P-40 unit and with lower performance.

The principal elements of the actuator are a flapper valve similar to MOOG 62 series and an oil piston as actuating element. Position of the oil piston is fed back mechanically to the flapper valve.

Torque motor and flapper valve work according to fig 8.4:30.

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OPERATION

- input current creates magnetic forces on ends of armature
- armature / flapper assembly rotates about flexure tube support
- flapper closes off one nozzle and diverts flow to one end of spool
- spool moves and opens P_s to one control port; opens other control port to R
- spool pushes ball end of feedback wire, creating restoring torque on armature/flapper
- as feedback torque becomes equal to torque from magnetic forces, armature/flapper moves back to centered position
- spool stops at a position where feedback wire torque equals torque due to input current
- therefore spool position is proportional to input current
- with constant pressures, flow to load is proportional to spool position

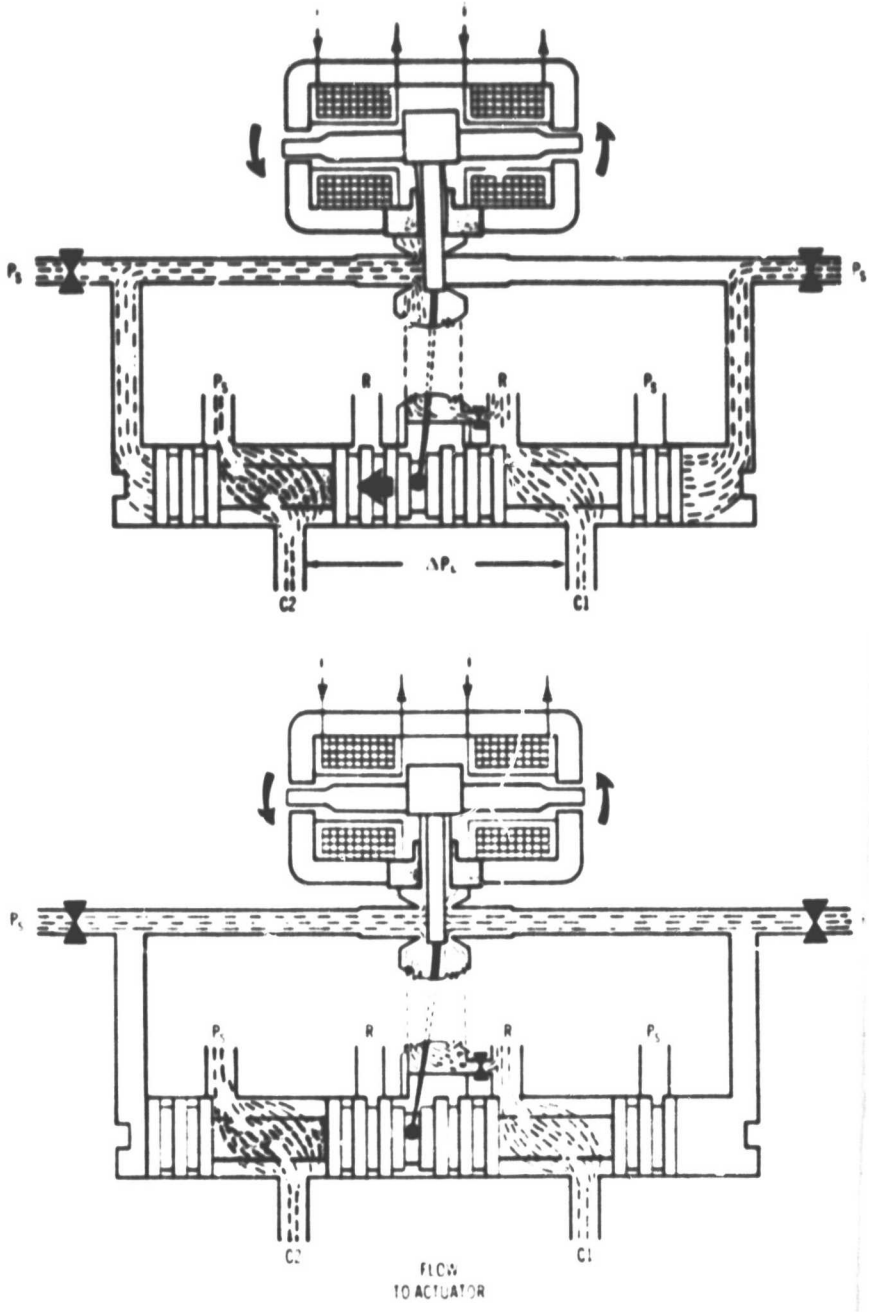


Fig 8.4:30 Working Principle of MOOG Torque Motor and Flapper Valve Used on ASE MOD I Power Control Actuator.

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Fig 8.4:31-32 show the MOOG electro-hydraulic servo actuator.

The oil piston in fig 8.4:30 is on the Stirling actuator replaced by the actuating piston itself (see fig 8.4:31). Feed back of the piston position is done via the feed back springs surrounding the flapper armature.

The torque motor has for safety reasons a manual override making it possible to roughly control the unit in case of electrical breakdow .

For safety reasons the actuator also has a null bias so that zero position in case of servo oil pressure loss would be dump position of the spool valve.

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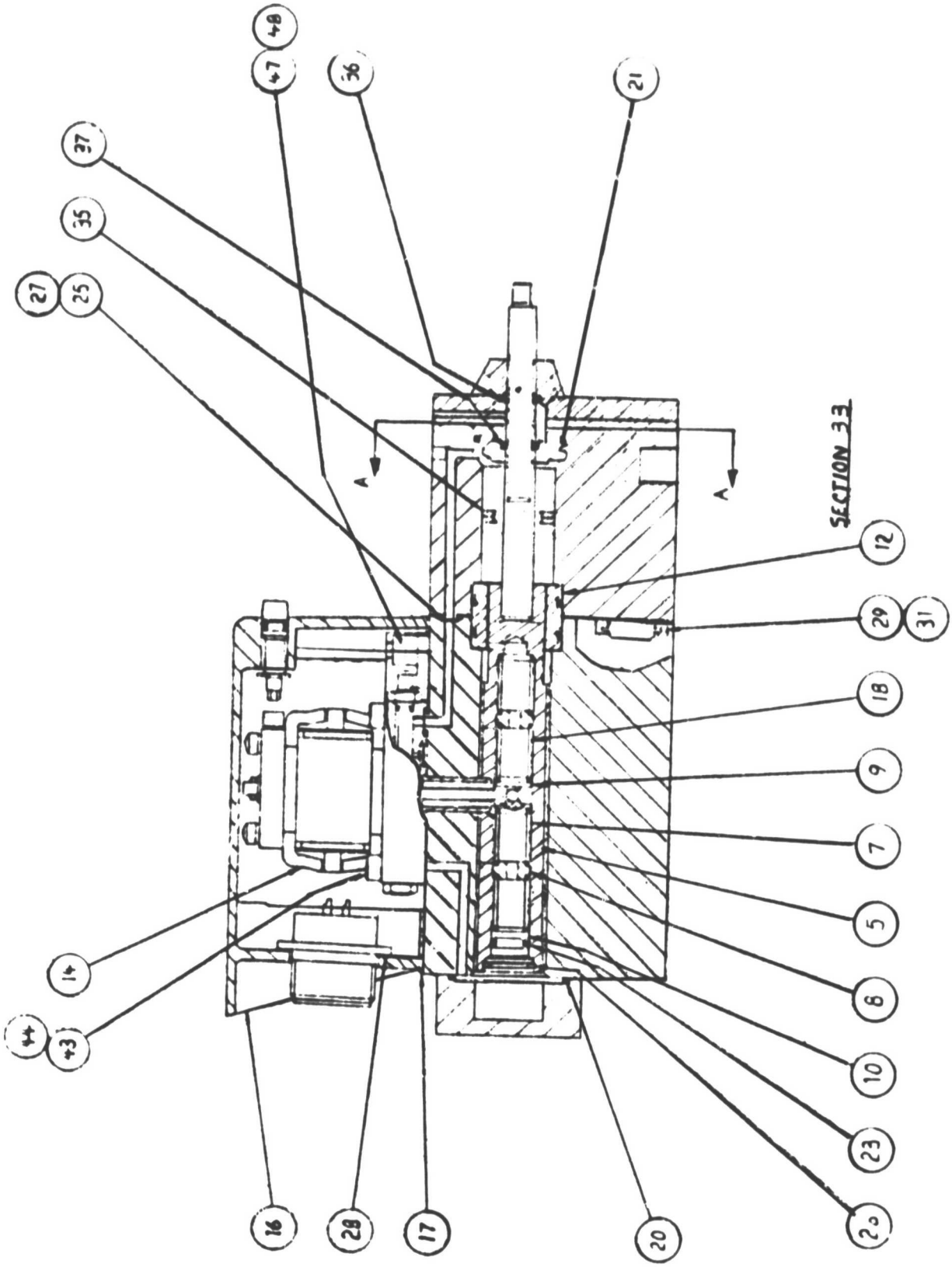
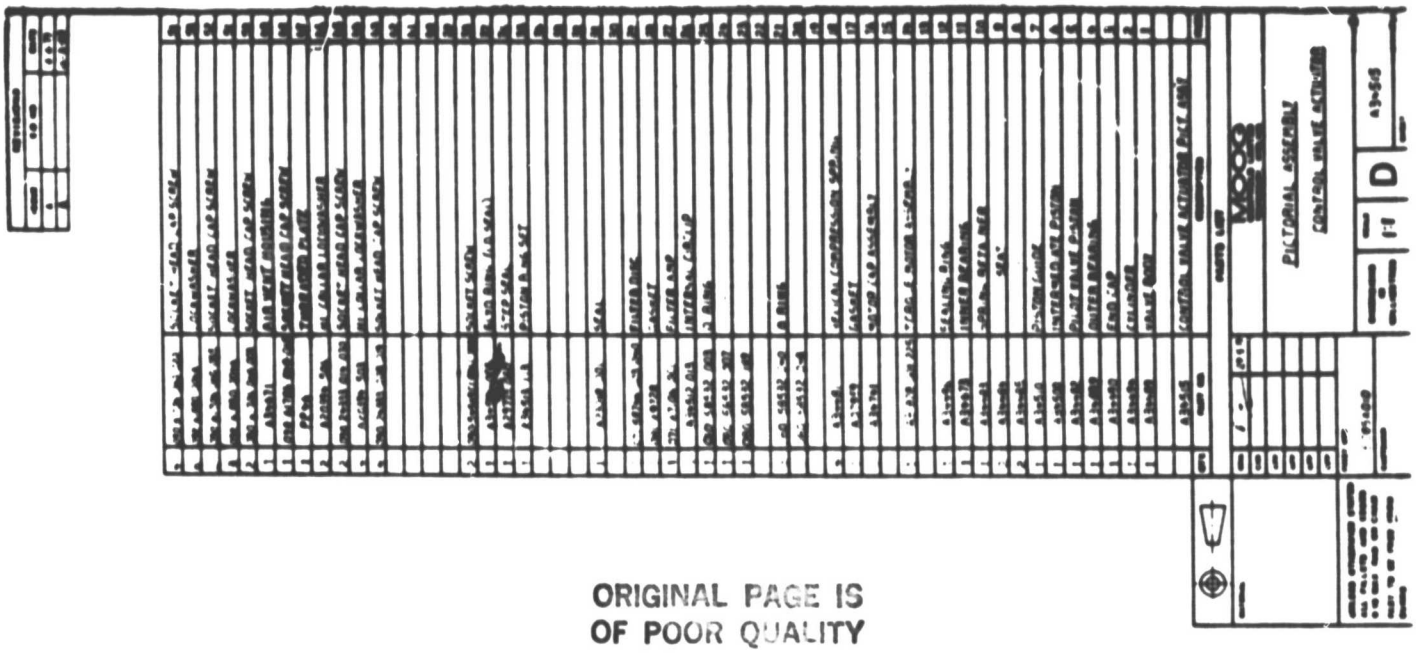


Fig 8.4:31 MOOG Electro Hydraulic Stirling Control Actuator.



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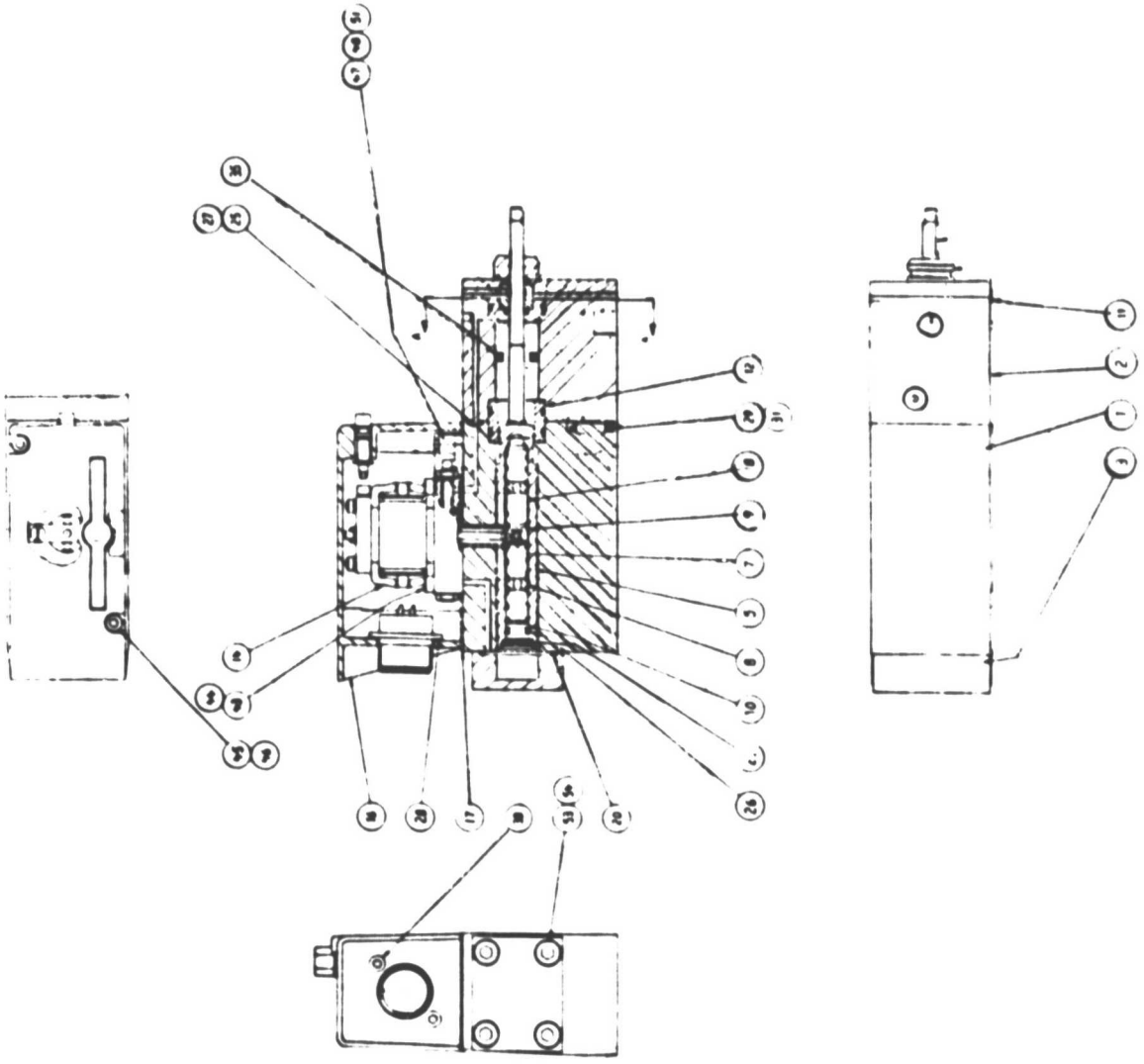


Fig 8.4:32 MOOG Electro Hydraulic Stirling Control Actuator.

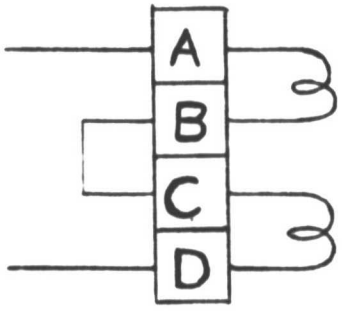
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Specification of the actuator.

Supply pressure	= 3 - 5 MPa
Max pressure	= 15 MPa
Rated thrust at 5MPa	= 970 N
Normal operating stroke	= 10 mm (left), 10 mm(right)
Rated actuator velocity	= 50 mm/sec
Working fluid	= mineral oil
System oil filtration	= 10 μ absolute or better recommended

Electrical connector mates with M53106 R-145-25

Coil connection (series coils)



Flow out port "A"	A+, D-
Rated current	± 0.05 A
Voltage for rated current at 23° C	± 27 VDC
Maximum voltage	24 VDC

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Experience

Present experience of MOOG electro-hydraulic servo valves has been very good. No fault has occurred with the servo valves on present P40 engines and during the four years of operation on the Taunus car. Performance and behaviour of the valve have been much better than demands.

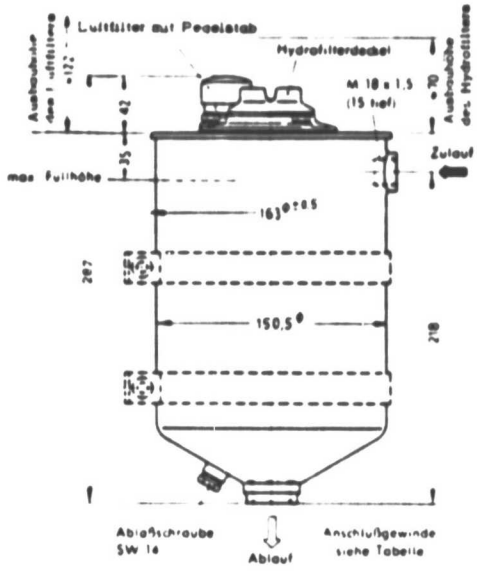
8.4.8.3 Oil Pump (3)

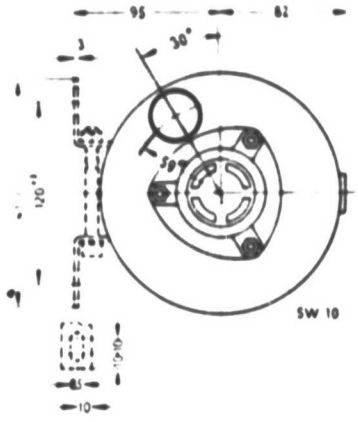
The power control valve of ASE MOD I power control system needs hydraulic oil for its actuation. The servo oil pump in the hydraulic oil system is described in section 8.3.10.

8.4.8.4 Oil Tank (4)

The servo oil system needs a small oil tank of about 1.5 litre in volume. In automotive application ASE MOD I will be equipped with a custom made tank placed in the engine compartment. Test cell engines will be equipped with a Bosch 3 litre standard tank (Bosch 0542901010), see fig 8.4:33.

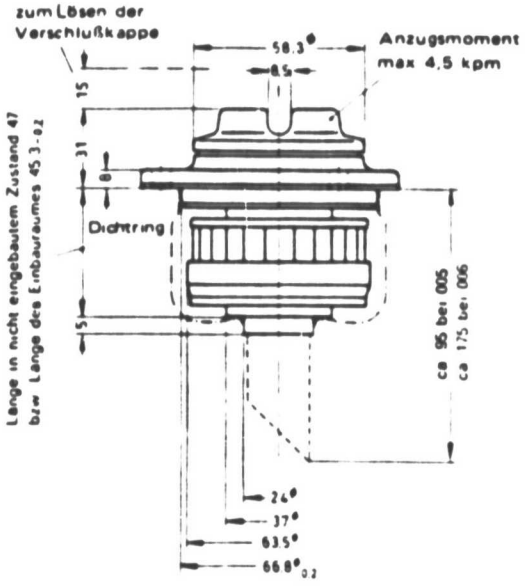
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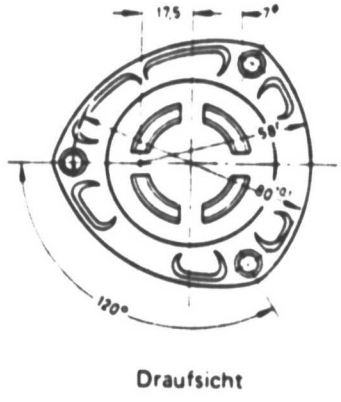




Note: This tank will also contain an oil suction filter. Filteration 60 µ absolute, see Fig. 8.4:34.

Fig. 8.4:33 Servo Oil Tank ASE Mod I





Draufsicht

Fig. 8.4:34 Suction Oil Filter (Power Control System, ASE Mod I)

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8.4.8.5 Constant Pressure Valve (5)

The ASE MOD I servo oil system is equipped with a constant pressure valve to maintain servo oil pressure at a constant pressure.

ASE MOD I will use a valve from Nordhydraulik in Sweden (DDTR-04 AP).

It is a plug in type of valve and will be mounted in block 4. The valve will be preset to a pressure of 3 MPa, see fig 8.4:35.

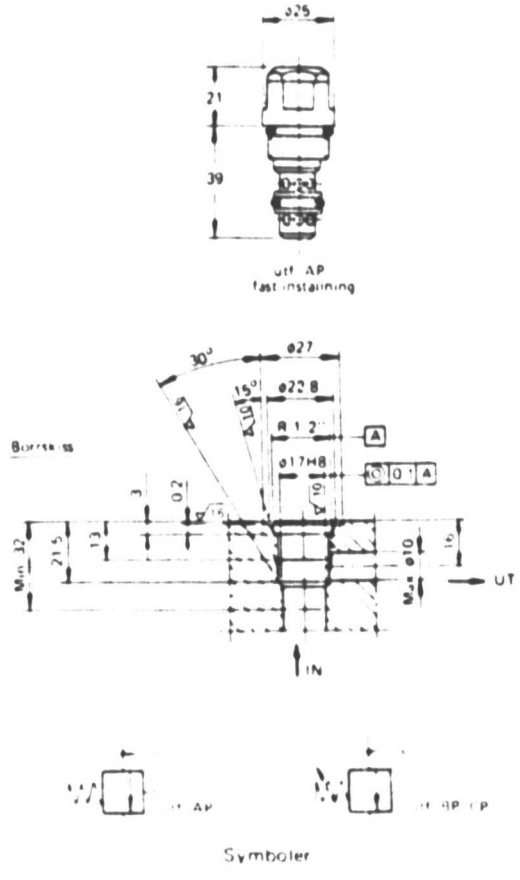


Fig 8.4:35 Constant Pressure Valve Nordhydraulik DDTR-04AP.

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Experience

The P40 constant pressure valve is from the same manufacturer but is not a plug-in version. The active elements are, however, similar. The P40 valves have operated without malfunction on all P-40 engines.

8.4.8.6 Oil Filter (6)

The servo oil system is equipped also with a pressure filter. This filter is the same as the hydrogen filter described in section 8.4.8.8.

A type 1 filter 10 μ nominal will be used. This is the same type of servo oil filter used in all P40 engines.

8.4.8.7 Hydrogen Compressor (7)

To dump hydrogen from the engine to the storage vessel a hydrogen compressor pumping hydrogen to the high pressure tank is needed. The compressor is fixed to the engine and runs with engine speed.

On all USSw mean pressure controlled Stirling engines the compressor has been single stage and double acting. The compressor is running oil free and with piston rings of rulon acting as suction valves. Maximum pressure ratio is about 1:10. To keep temperatures low all compressors have been water cooled.

8.4.8.7.1 Compressor Design

The ASE MOD I compressor has the same general layout as earlier versions of compressors. It is also a single stage double acting compressor. A test unit with a different piston rod attachment has been built within the component development program. It has logged more than 3 000 hours in a separate test rig.

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Stress analysis has shown that this type of piston attachment can not meet the safety factor required for the MOD I operation during engine lifetime and with max expected pressure differences without major modifications. Therefore, a redesign of the compressor has been made and a new piston - piston rod attachment has been designed.

This design enlarges the piston rod to 12 mm in diameter and to keep compressor capacity equal the piston diameter has been enlarged from 20 mm on test unit to 22 mm.

Fig 8.4:36 shows the test unit of the compressor and fig 8.4.37 shows the new design of the compressor.

Principal dimensions of the ASE MOD I compressor will be:

Bore = 22 mm
Stroke = 20 mm
Piston rod diameter = 12 mm
Displacement: 7.6 cm^3 up, 6.5 cm^3 down.

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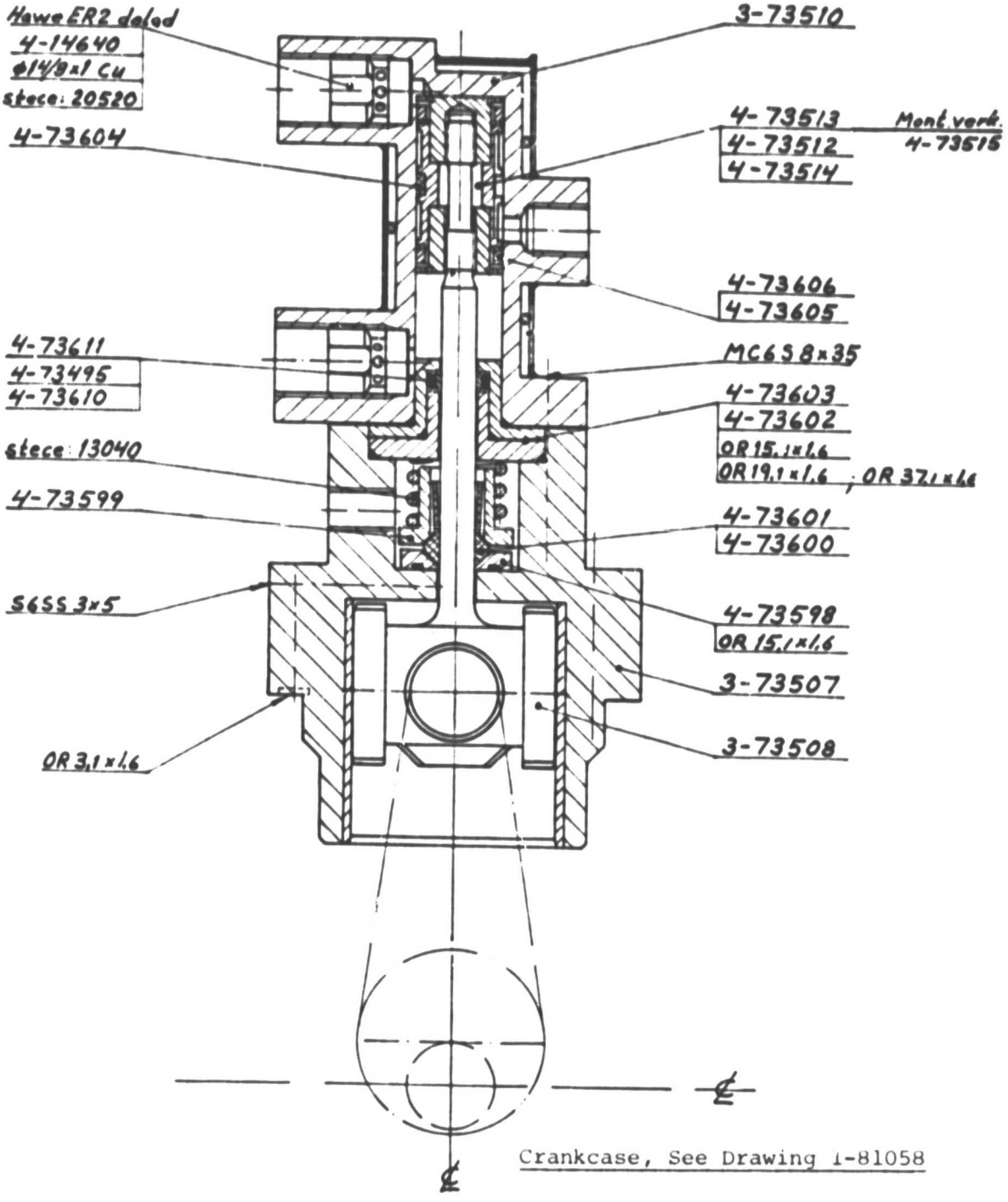


Fig. 8.4:36 Test Unit of ASE Mod I Compressor

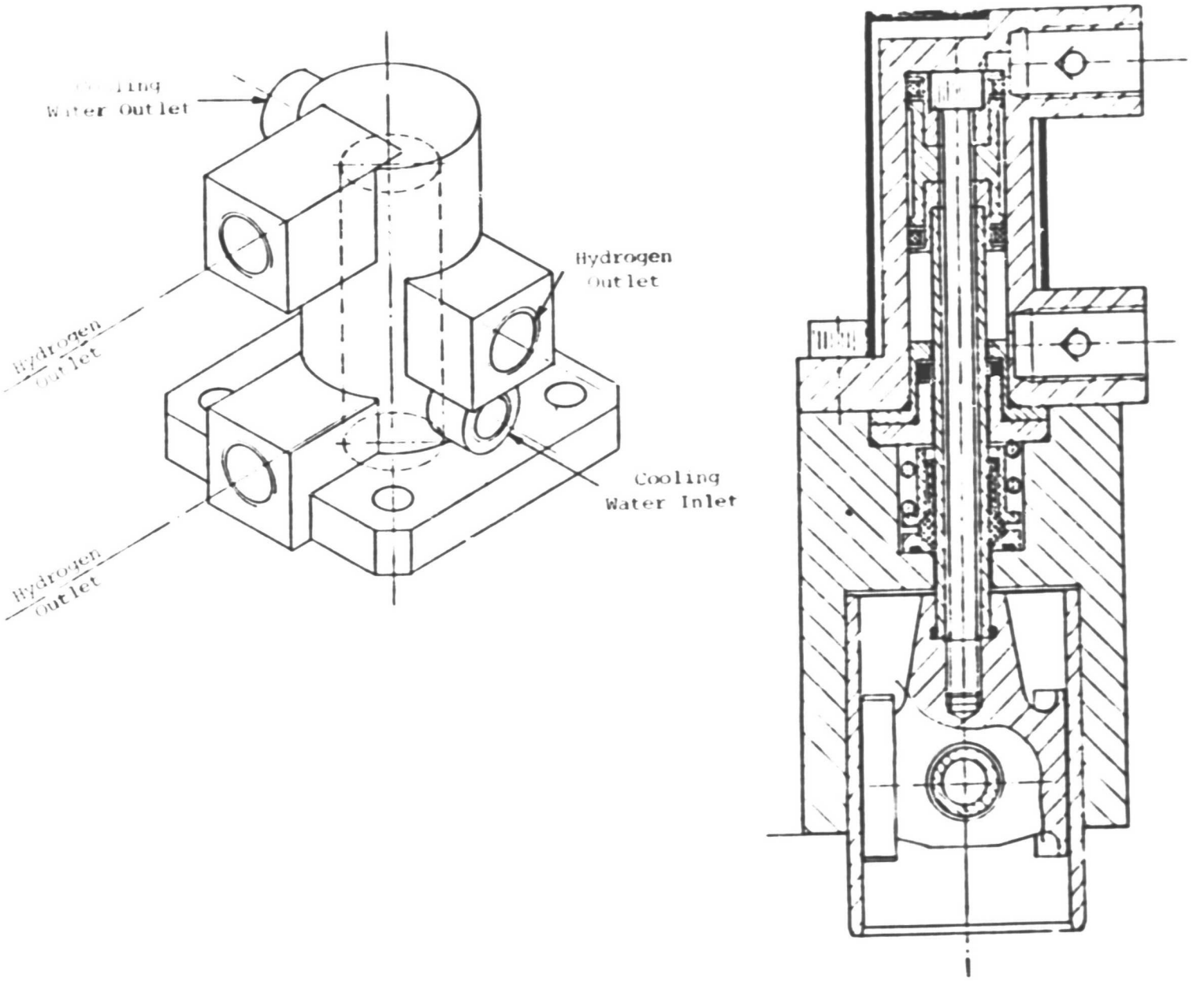


Fig. 8.4:37 Sectional Drawing of ASE Mod I Hydrogen Compressor

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8.4.8.7.2 Compressor Components

COMPRESSOR CYLINDER: In principle the same design as P-40 compressors. Inlet and outlet of hydrogen and cooling water connections can be seen in Figure 8.4:37.

MATERIAL: SIS 2324

COMPRESSOR CHECK VALVES: See Section 8.4.8.9.

COMPRESSOR PISTON: Almost identical to P-40 (Ø 22) compressor piston.

MATERIAL: SIS 2541-03

PISTON RINGS: Identical to P-40 piston rings. Rulon LD with backup rings of vespel SP1.

PISTON ROD: Preloaded between piston and crosshead, Ø 12 mm.

MATERIAL: SIS 294-04 Nitrified to 70 HRC.

SEAL SYSTEM: Similar to normal USSw seal system though with reduced dimensions.

CROSSHEAD: Connected to piston and piston rod with a tie bolt.

MATERIAL: SIS 2225-05.

TIE BOLT: A necked down tie bolt M 8 x 1, Umbrake quality 12.9.

8.4.8.7.3 Stress Analysis

The piston and the piston rod are bolted to the crosshead with a necked down tie bolt. The arrangement is shown in fig 8.4:37.

The following material data are valid:

BOLT

M 8 x 1 Unbrako quality 12.9
Yield stress (0.2%) σ_y = 1 170 MPa
Ultimate tensile strength σ_u = 1 300 MPa
Young's modulus E = 206 000 MPa

PISTON

SS 2541-03
Yield stress (0.2%) σ_y = 700 MPa
Ultimate tensile stress σ_u = 900 MPa
Young's modulus E = 205 000 MPa
Brinell hardness HB = 275

PISTON ROD

SS 2940-04
Yield stress (0.2%) σ_y = 700 MPa
Ultimate tensile strength σ_u = 950 MPa
Young's modulus E = 205 000 MPa
Brinell hardness HB = 290

CROSSHEAD

SS 2225-05
Yield stress (0.2%) σ_y = 700 MPa
Ultimate tensile strength σ_u = 900 MPa
Young's modulus E = 206 000 MPa
Brinell hardness HB = 270

The load is cyclic with a maximum net pressure of 17.5 MPa between the upper and lower side of the piston. This gives a net force between 4 100 N and -7 200 N.

Considering the spring constant of the bolt and its environment, the stress amplitude caused by the applied load is 54 MPa. This is below the maximum allowed value of 64 MPa for rolled M8 bolts. Thus, the bolt design fulfils VDI 2230 (Verein Deutscher Ingeneure, "Systematische Berechnung hochbeanspruchter Schraubenverbindungen").

The amplitude force in the bottom crosshead threads will be 5 650 N since these threads are not preloaded by the bolt torque. This gives a nominal amplitude stress equal to 12 MPa, which is below the maximum value of 30 MPa for cut M8 bolts with the corresponding material strength.

Assuming a friction coefficient of 0.1 the recommended preload is 20 800 N for a M 8 x 1 12.9 tie bolt. This gives a good margin against loosening by the maximum applied tensile force of 4 100 N. In combination with the applied load the preload gives a maximum compressive load equal to 28 000 N. This gives a compressive stress of 446 MPa in the piston rod sleeve, which corresponds to a safety factor against yielding of 1.6. Thus, the surface pressure of the clamped parts are far below allowed levels. The recommended preload corresponds to 23 Nm.

8.4.8.7.4 Compressor Performance

From tests, using the test stand compressor, the pumping (down time) for the ASE Mod I compressor has been calculated.

Fig 8.4:38 shows the pumping down sequence for the MOD I compressor. The diagram shows the engine and storage vessel pressure as a function of number of compressing strokes.

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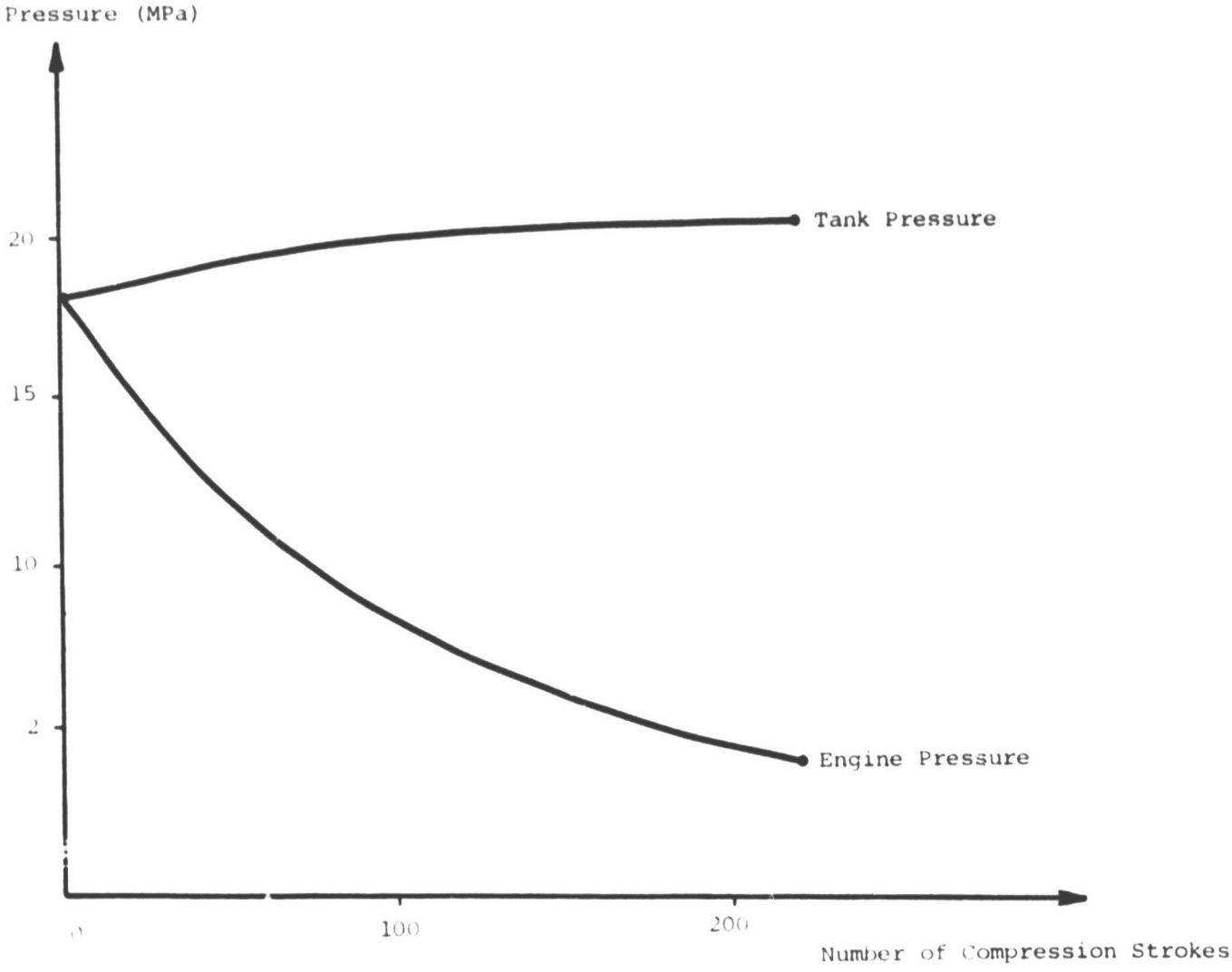


Fig. 8.4:38 Pumping Performance of the ASE Mod I Compressor

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Experience:

This type of external single stage double acting, water cooled hydrogen compressors has been used in all USSw Stirling engines the last four years. Total logged running time is far more than 10,000 hours.

8.4.8.8 Hydrogen Filter

To prevent rulon dust or other particles from wearing or destroying seals and sealings in the ASE Mod I control system hydrogen filters are placed in block 1, block 2 and block 3.

The filters will be of two types with opposite flow direction due to the design of the control blocks, see Figure 8.4:39.

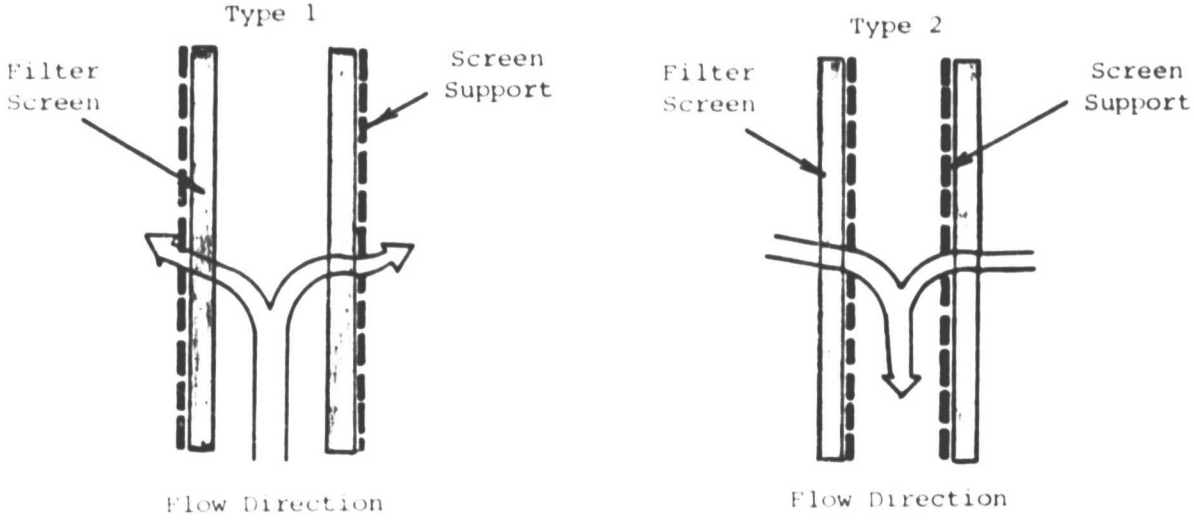


Fig. 8.4:39 Type 1 will be Placed in Block 1 and 3, While Type 2 is Placed in Block 2

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Filtration for all units will be 10 μ nominal and both types have the same mounting dimensions.

The manufacture of these filters is SOFRANCE and their type numbers are:

Type 1 = PC 85 Rev flow

Type 2 = PC 851

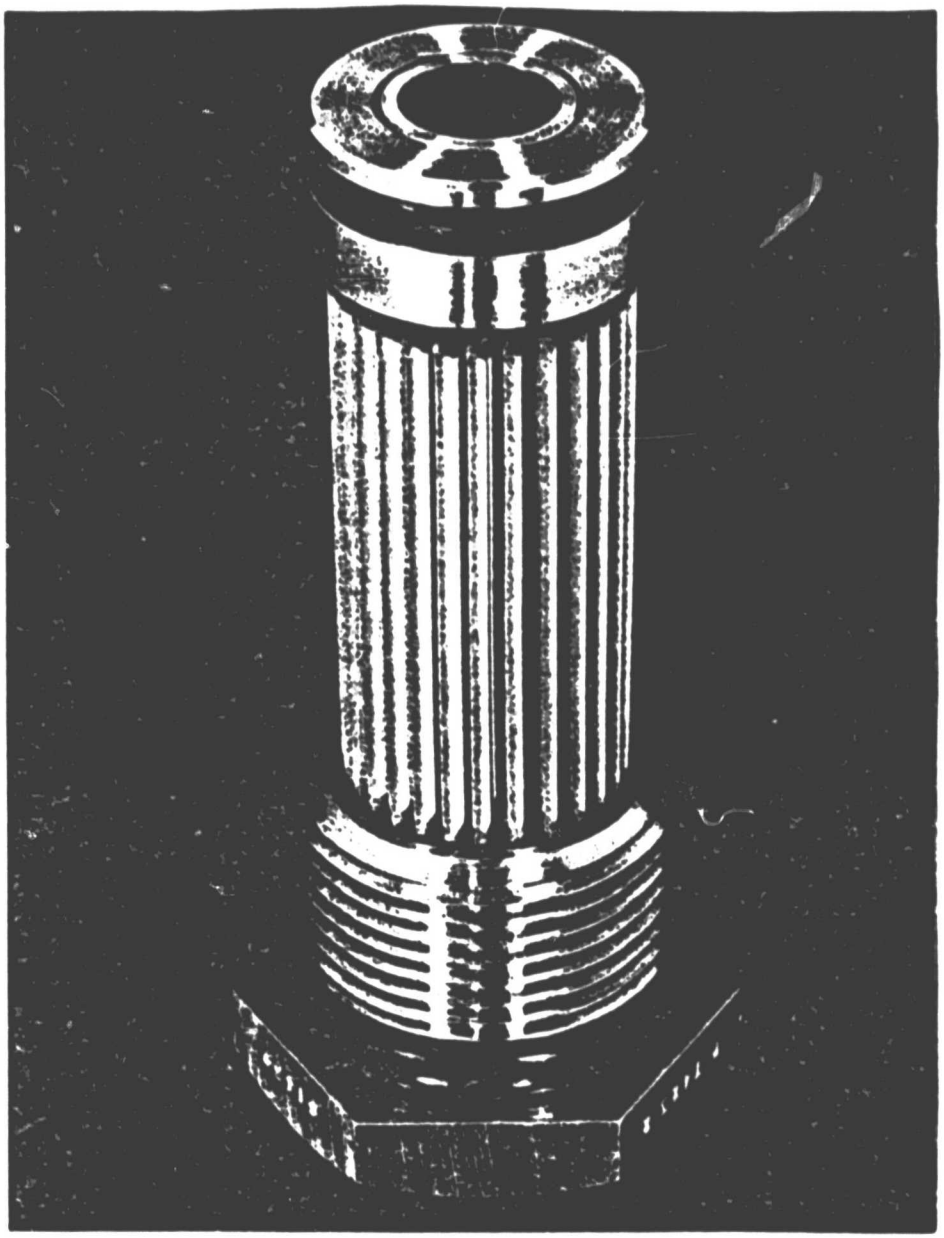


Fig 8.4:40 Hydrogen Filter Type 2 SOFRANCE = PC 851

Experience:

The type 2 filter is currently used on all our P40 engines with very good result. No problems with these filters have occurred during the two years they have been used at USSw.

8.4.8.9 Check Valves

There are within the ASE MOD I power control system a total of 16 check valves.

A. Four check valves allowing working gas to flow out from each cylinder.

B. Four check valves are placed as min pressure check valves allowing working gas to flow into each cylinder.

C. Four check valves are placed as min pressure check valves in the P_{max} supply line allowing gas to enter each cylinder from the storage vessel and preventing cycle pressure variations in the supply line.

D. Two check valves are placed as pressure valves in the hydrogen compressor.

E. Two check valves with O-ring seals are placed before and after the compressor short circuiting circuitry. One check valve is preventing hydrogen at tank pressure from flowing into the short circuited compressor. One check valve prevents the hydrogen in the short-circuitry tubings from flowing into the engine during idling when the spool valve is in the dump and short circuiting position, and the compressor short-circuiting solenoid valve is open.

8.4.8.9.1 Check Valve Specification

Type:	**Check valves for Stirling engines** have a working principle similar to air compressor exhaust valves.
Size:	Internal flow corresponding to at least 6 mm ESEOD (Equivalent Sharp Edge Orifice Diameter).
Working gas:	Hydrogen.
Pressure:	Max total pressure 25.0 MPa Max diff pressure 22.5 MPa
Gas temperature:	Min, max check valves <150° C Compressor check valves 225° C
Leakage:	External none In checked direction <0.001 g/s
Working frequence:	10 - 75 Hz
Material:	Hardened material in seat, plate and in spring holder. Large flows may occur in open direction.
Outer dimension:	As small as possible.
Design:	The check valve should be of a cartridge type and be able to mount in a block. The design should be resistant to particle contamination (Teflon dust) in the working gas.
Endurance:	2 000 H continuous operation.

8.4.8.9.2 Check Valve Selection

An extensive test program has been carried out within the ASE Component Development Program. A screening test of available check valves has been performed and each check valve has been tested 600 hours in continuous operation. See appendix 8.4:1.

From that screening test a check valve was selected and introduced to later versions of P40 engines. Also earlier engines were rebuilt and fitted with the new check valve. One set of these valves have logged more than 2000 hours in cyclic operation on a P-40 engine.

8.4.8.9.3 Check Valve Design

The ASE MOD I check valves are derivates from these P40 valves except the o-ring sealed check valves (E), which are identical except for the outer housings.

P_{max}, P_{min} and Psupply valves

The ASE MOD I P_{max}, P_{min} and P_{max} supply check valves are identical in design.

Seat and plate are elements from the HAWE ER2 check valve, see fig 8.4:43. These parts are identical with the P40 check valves. The spring is also identical.

The spring holder is modified to allow simplified flow passages which is possible because of the location of the ASE MOD I check valves.

Fig 8.4:41 shows the ASE MOD I p_{max}, p_{min} and P_{max} supply check valve and fig 8.4:44 shows the spring holder of that check valve.

Fig 8.4:42 shows the difference in flow passages between the ASE MOD I check valve and the original P40 valve.

The flow of the MOD I valve is only diverted 90^{o} once before leaving the valve thus reducing flow resistance and making the spring holder easier to manufacture.

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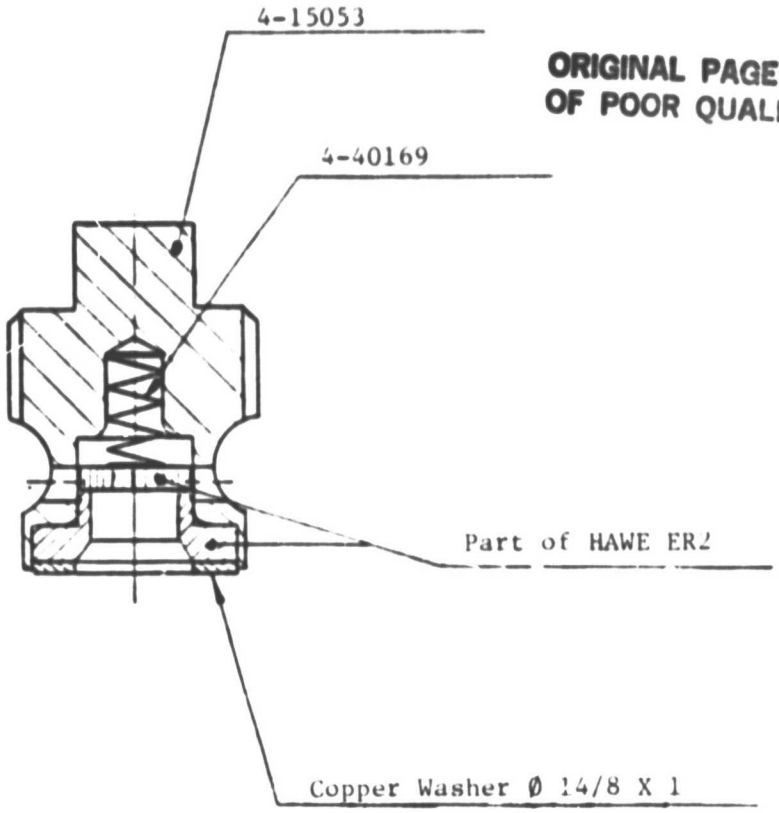
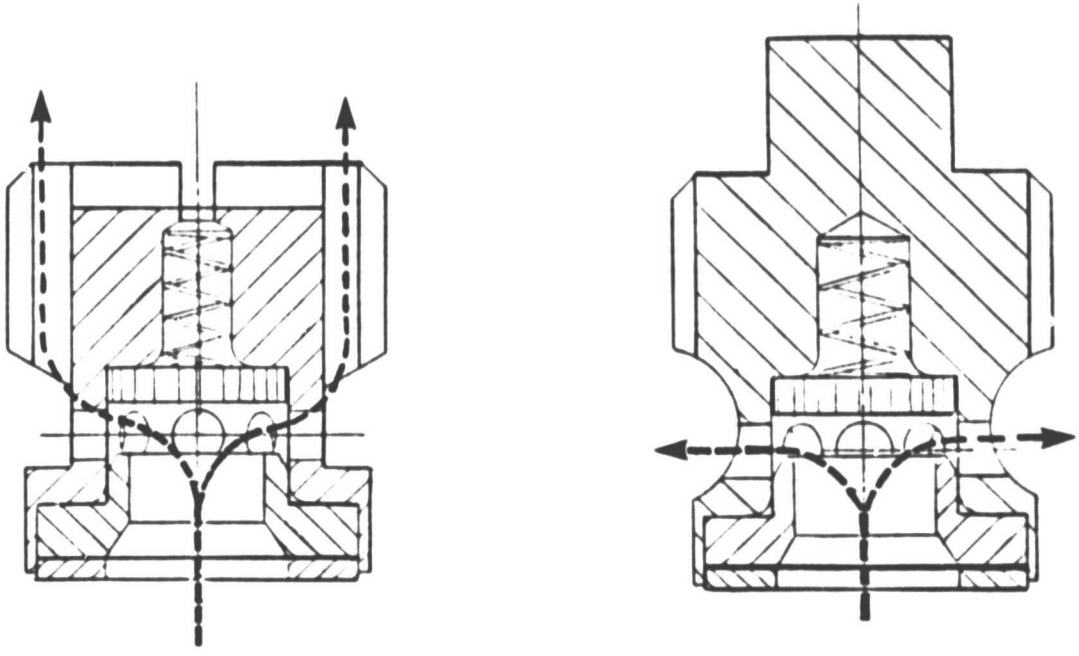


Fig. 8.4:41 The ASE Mod I $\hat{P}$, $\bar{P}$ and $P_{max\ Supply}$ Check Valve



P-40 Check Valve

ASE MOD I Check Valve

Fig. 8.4:42 Difference in Flow Passages Between the ASE Mod I Check Valve and the P-40 Valve

Location

The p_{max}, p_{min} and p_{max} supply check valves are all located in the seal housings and 8.4:45 shows the valve arrangement.

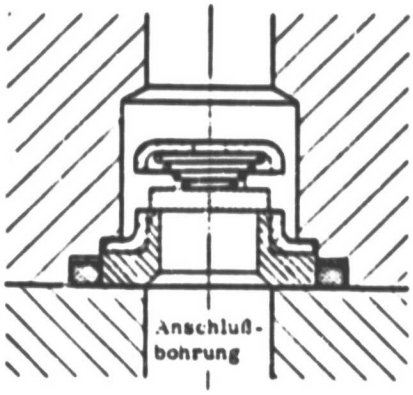
The compressor check valves

These valves are almost identical with the P40 valves and only modified to lower the flow resistance during passage of the spring holder. Fig 8.4:46 shows the spring holder.

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1. Allgemeines

Rückschlagventile geben den Durchflußstrom in einer Richtung frei und sperren ihn in Gegenrichtung. Sie gehören deshalb zur Gruppe der Sperrventile (DIN 24 300).



2. Beschreibung

Der Typ ER ist ein federbelastetes Einsteck-Rückschlagventil und wird vorzugsweise in Hydrogeräte eingebaut, die für Montage auf Grundplatten geeignet sind. Das Ventil wird in eine abgesetzte Bohrung eingeführt und mit einem Bund zwischen Grundplatte und Bohrungsabsatz eingeklemmt, so daß es unverrückbar festsitzt. Die Abdichtung an der Trennfuge übernimmt ein O-Ring.

Der Typ ER ist ein Plattenventil. Ventilsitz und -platte sind gehärtet und geschliffen, der Käfig gehärtet. In gesperrter Durchflußrichtung ist es hermetisch dicht.

3. Kenngrößen

3.1. Allgemein

Sinnbild und Typenbezeichnung	 ER 0, ER 1, ER 2, ER 3
Bauart	Einsteck-Rückschlagventil
Geräteabmessungen	siehe Pos. 4
Einbaulage	beliebig

3.2. Hydraulische Kenngrößen

Betriebsdruck	max. 500 bar
Druckmittel DIN 51 524	Hydrauliköl HL (P) 9, 16, 25, andere Druckmittel auf Rückfrage
zul. Viskositätsbereich	4 ... 150 cSt
Umrechnungsleiter von cSt in E	
Temperaturen	Umgebung: -35 ... +70°C Druckmittel: -40 ... +80°C
zul. Durchfluß	ER 0 = 6 l/min; ER 1 = 12 l/min; ER 2 = 30 l/min; ER 3 = 65 l/min

Δp-Q-Kennlinien

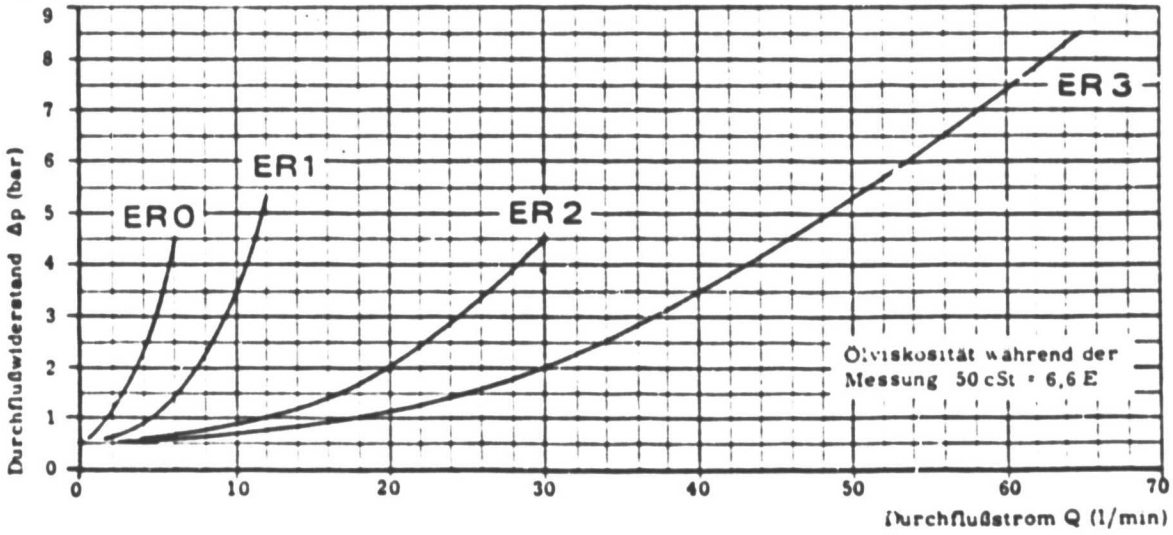


Fig 8.4:45 HAWE ER Check Valve.

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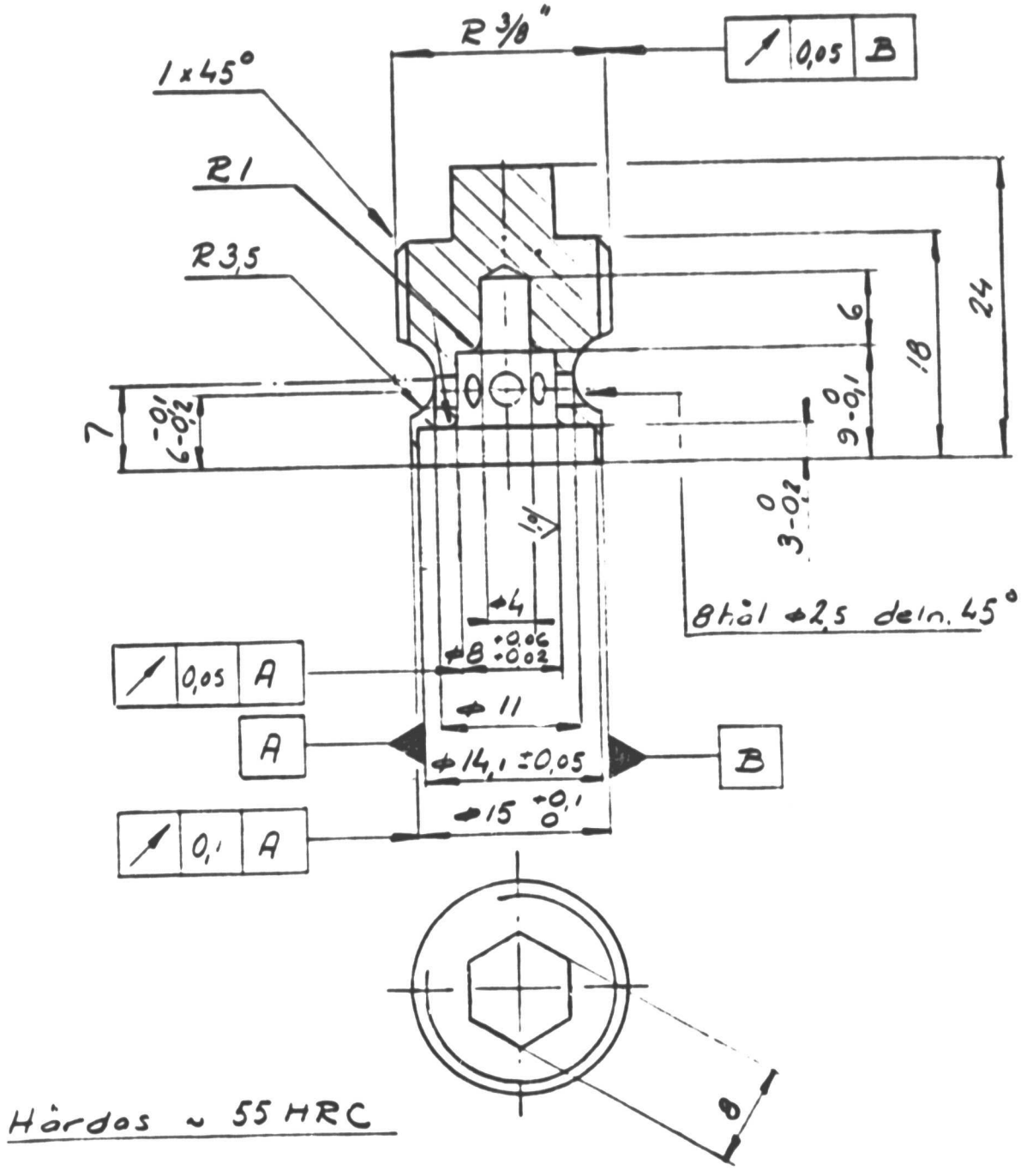


Fig. 8.4:44 Spring Holder of the ASE Mod I $\hat{p}$, $\bar{p}$, $\hat{p}_{supply}$ Check Valve

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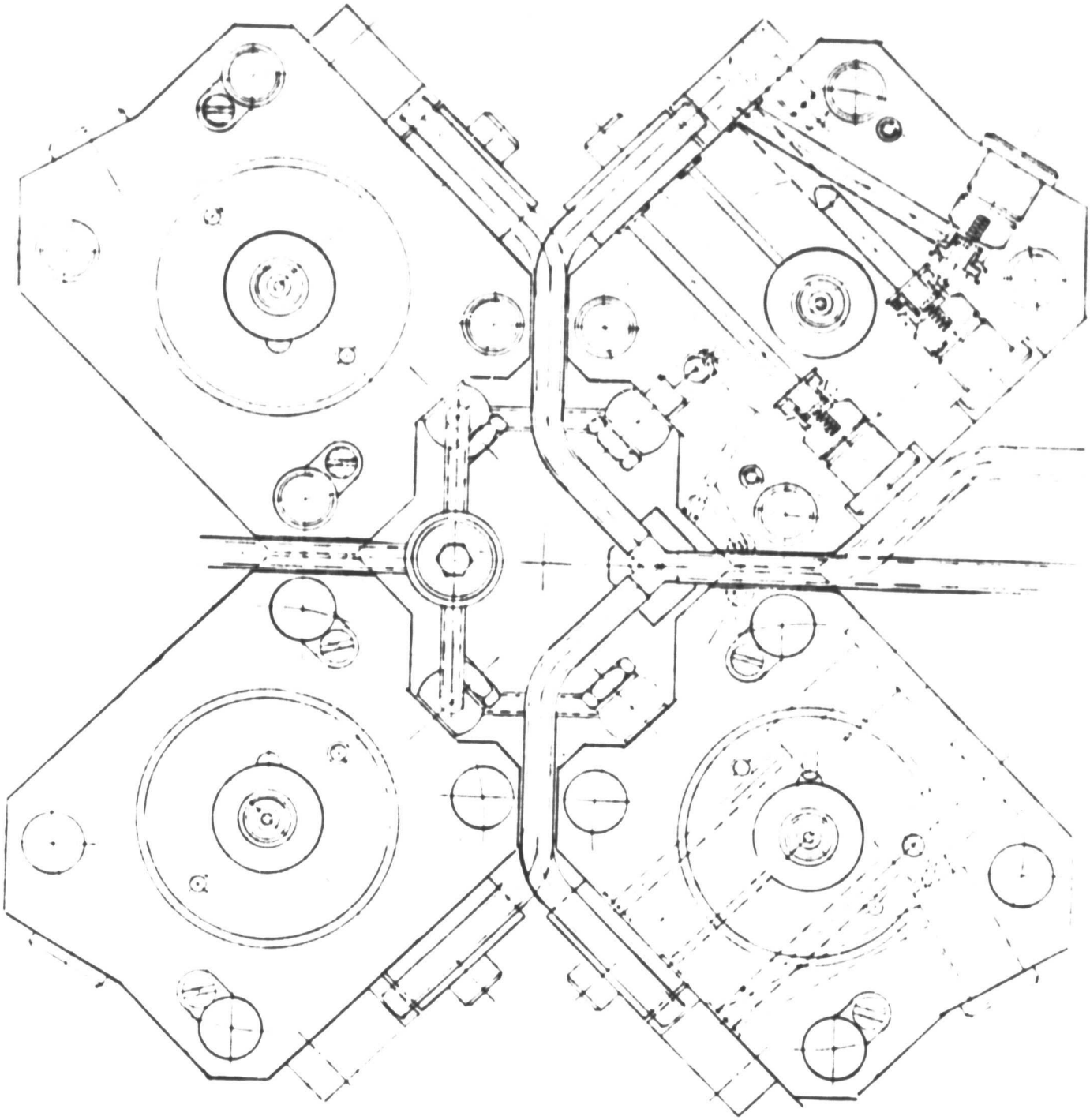


Fig. 8.4:45 Check Valve Arrangement of the ASE Mod I. The Check Valves are Placed in the Seal Housings.

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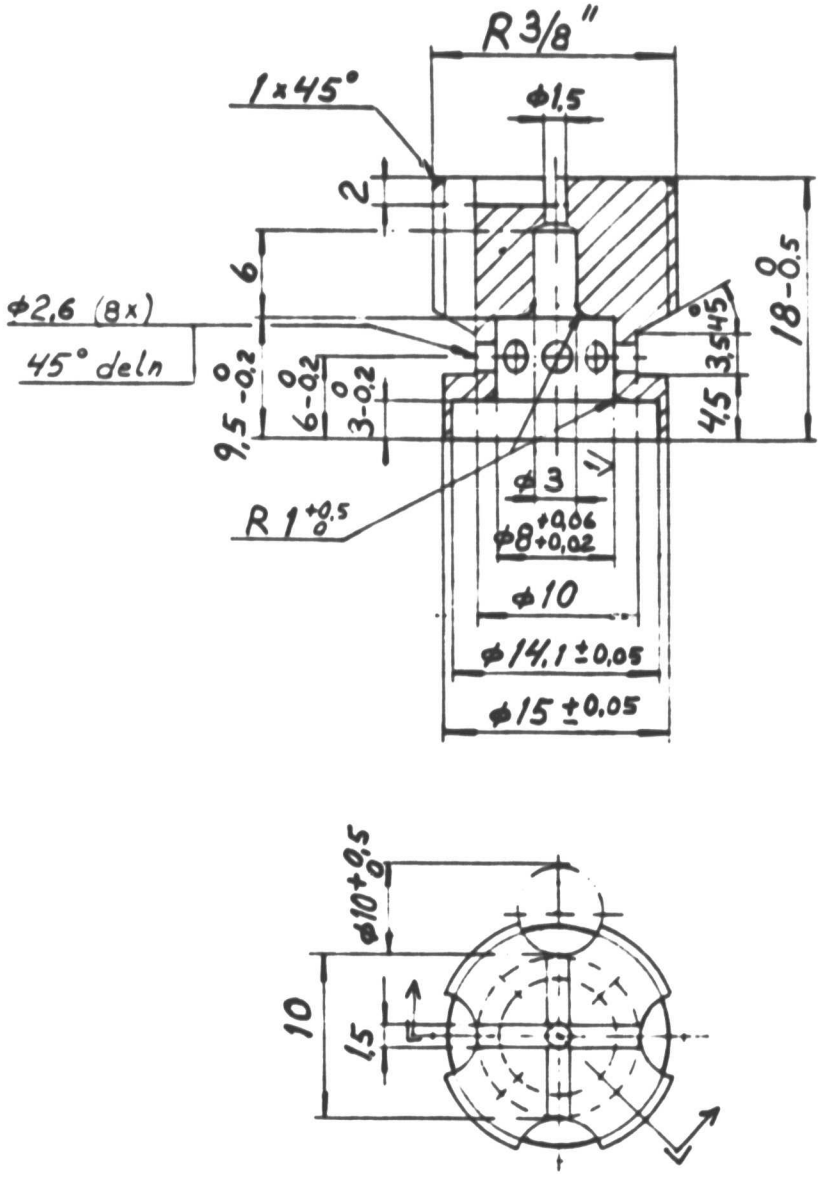


Fig. 8.4:46 Spring Holder of the ASE Mod I Compressor Check Valve

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The O-ring sealed check valves

The active parts are identical to the P40 design. Plate and spring are coming from HAWE ER3 check valves. Contrary to the P40 check valves the ASE MOD I valves are not mounted in a separate housing but mounted in block 1 and block 3.

Fig 8.4:47 shows the design of the o-ring sealed check valve.

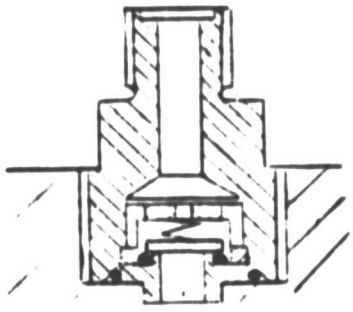


Fig 8.4:47 The O-ring Sealed Check Valve.

8.4.8.9.4 Check Valve Material

P_{max}, P_{min}, P_{max} supply and compressor check valves.

Seat, plate: Hardened material from HAWE.

Spring holder: SIS 2140 hardened to 55 HRC

Spring: Stainless steel SIS 2331-06

O-ring sealed check valve.

Seat: SIS 2140 hardened to 55 HRC

Plate, spring from HAWE

Spring holder: SIS 2140 hardened to 55 HRC

Experience:

Through the screening test and endurance testing of new check valve designs within the ASE MOD I component development program, lifetime and reliability of check valves have increased strongly. New designs have replaced old valves on P40 engines. The ASE MOD I valves are further improved and simplified. Lifetime of P40 check valves are now more than 2 000 hours in cyclic operation with a P40 engine (P40-2 high temperature engine).

8.4.8.10 Solenoid Valves

The power control system of the ASE MOD I needs three solenoid valves. Although almost identical in orifice size, cycling rate and pressure differences, all three differ from each other.

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Solenoid valves for light gases with internal orifice diameter of 4 - 6 mm are not common. These kinds of solenoid valves are mostly used in aerospace and military applications. On the P-40 engine solenoid valves from two different vendors are used (Valcor Inc and Herion GMBH) These valves were not intended for use with hydrogen at our cycling rate. Our present experience with solenoid valves have been somewhat discouraging but the Valcor valve has performed far better than the Herion valve.

To obtain such valves and to extend service life a request for custom built solenoid valves has been passed out to some vendors.

Valcor Engineering Corporation declared themselves willing to develop valves to our specifications according to their technical proposal 792-015.

ASE Program Office has concurred to a development contract for Valcor Inc. according to technical proposal 792-015 based on the ability of Valcor Inc. to design custom build valves and present experience of Valcor valves on P-40 engines.

8.4.8.10.1 Compressor Short Circuiting Valve

This solenoid valve unloads the compressor during steady state conditions or supply periods and is controlled from the electronic unit. This solenoid valve has the highest cycling rate and highest gas temperature in the power control system. The valve will be mounted in modular block ".

Valve Design

The compressor short-circuiting valve will be of a special "balanced spool" design that utilizes a combination of materials that can easily achieve the long cycle life required. Valcor will conduct an endurance cycle test to a Valcor procedure on one prototype of this valve that duplicates in some respects actual pressure differential conditions that exist on the engine.

All tests will be made with nitrogen.

Fig 8.4:48 shows the design of compressor short circuiting valve.

Valve Specification

Operating pressure:	3 000 PSIG Differential
Proof pressure:	4 500 PSIG
Burst pressure:	12 000 PSIG
Ambient temperature:	-25^{o} F to $+212^{o}$ F
Voltage:	10-14 VDC operation
Voltage:	8 VDC cold start zero differential across valve
Duty:	Valve must be capable of being continuously energized at $+212^{o}$ F without failure.
Flow capacity:	Equivalent to 0.240 sharp edge orifice. Cd 0.65.
Hydrogen temperature:	Max $+260^{o}$ F (short duration)
Leakage rate:	300 scc/min at 1 500 PSIG differential pressure
Cycle life:	>2 500 000 cycles at system conditions

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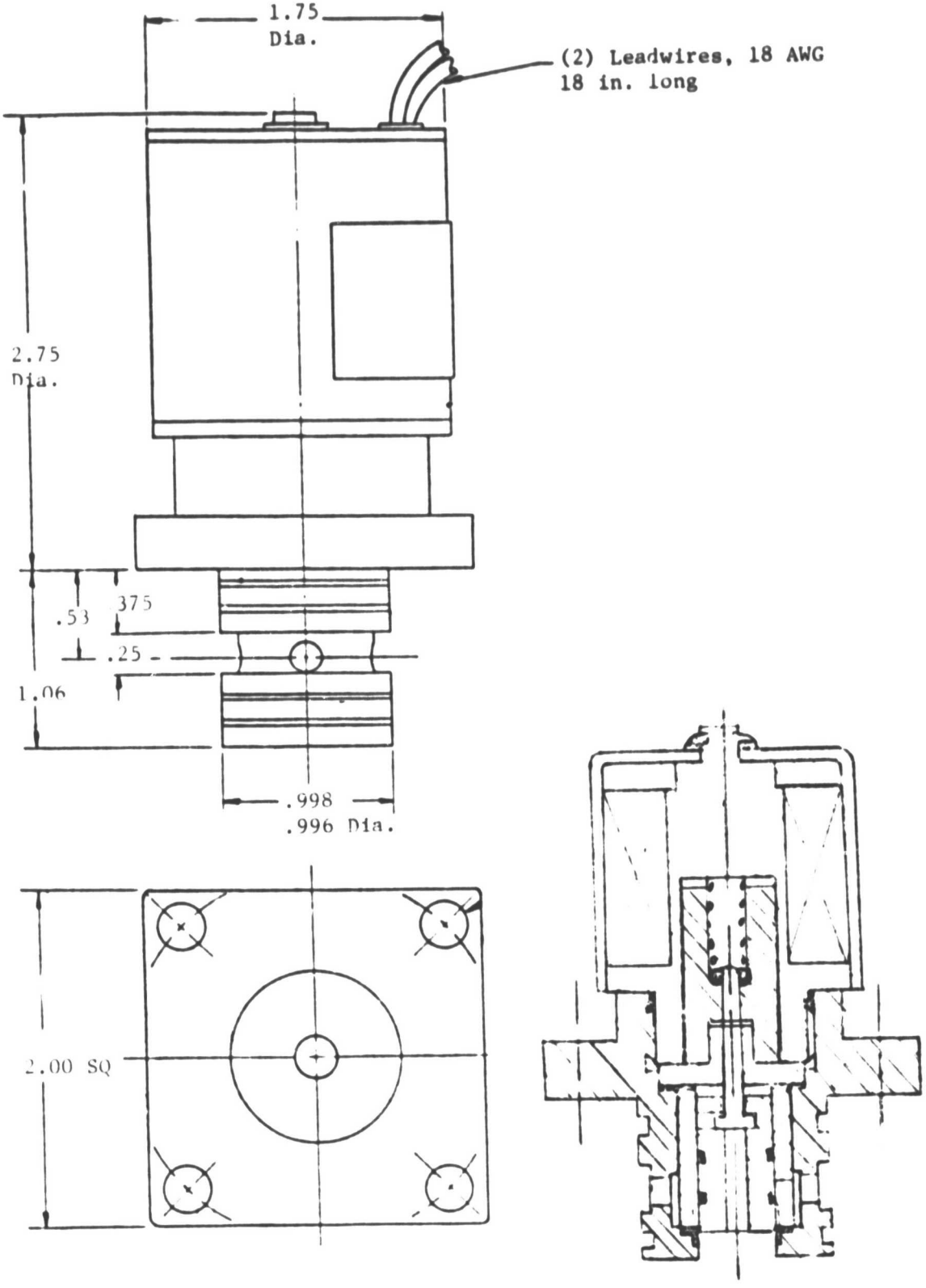


Fig 8.4:48 Compressor Short-circuiting Solenoid Valve.

8.4.8.10.2 **External Dump Valve**

This valve is an emergency valve for dumping hydrogen to atmosphere in case of any emergency.

This valve is <u>not</u> controlled by the engine electronics, but will be operated by the driver from a push-button on the dash board. The valve is energized directly from a battery.

There are extreme specifications for low leakage across this valve (all leakage will be external leakage), but the number of operations is very low.

This valve will be plugged into modular block 1.

Valve Design

The valve is a pilot operated type that utilizers a soft plastic for sealing at both main seat and the pilot. Valcor intend to investigate the use of both E D R and Kel-F in a special proprietary configuration.

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Valve Specification

Operating pressure:	3 000 PSIG differential
Proof pressure:	4 500 PSIG
Burst pressure:	12 000 PSIG
Ambient temperature:	-25° F to +212° F
Voltage:	10-14 VDC operation
Voltage:	8 VDC cold start low differential across valve
Duty:	Valve must be capable of being continuously energized at +212° F without failure
Flow capacity:	Equivalent to 0.160 diameter sharp edge orifice Cd 0.65
Leakage rate:	2 scc/24 hrs at 1 500 PSIG differential pressure
Cycle life:	Replace seat after 2 cycles, if necessary
Hydrogen temperature:	Max +212° F

Fig 8.4:49 Shows the design of External Dumping Solenoid Valve.

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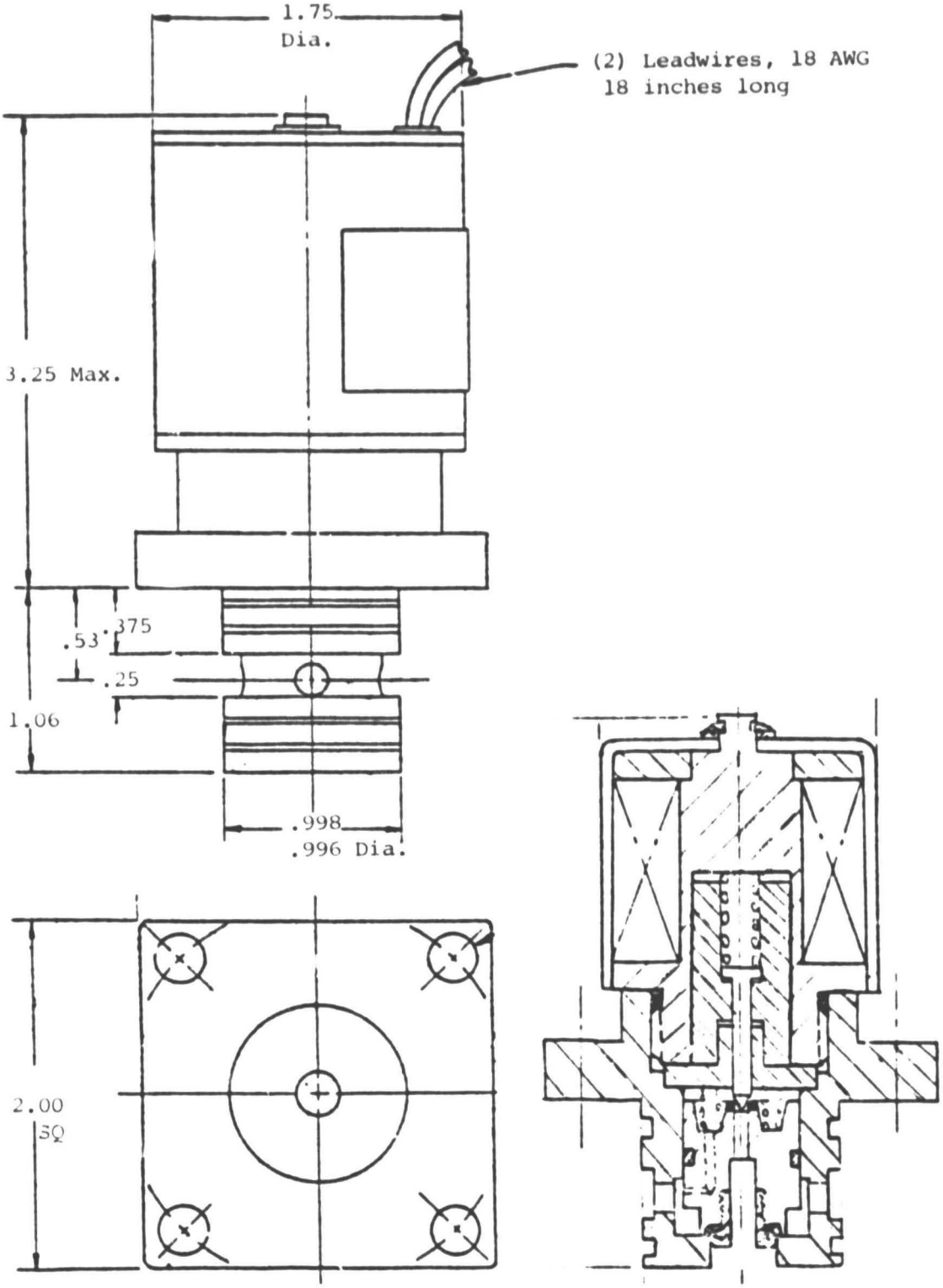


Fig 8.4:49 External Dumping Solenoid Valve.

8.4.8.10.3 Shut-off Valve

This valve is a shut-off valve preventing hydrogen flowing from the storage vessel to the engine during non operation periods. The power control valve with its sliding seals can not be leaktight and thus an extra shut-off valve is needed. The valve will be controlled by the electronics and opened by the starting sequence and closed by the stop sequence.

Also this valve has a rather extreme specification on leakage.

This valve will be mounted into block 2.

Valve Design

The design of this valve is similar to that of the external dumping valve but the increased demand for cyclic operations requires a harder seat material.

Valve specification

Operating pressure:	3 000 PSIG Differential
Proof pressure:	4 500 PSIG
Burst pressure:	12 000 PSIG
Ambient temperature:	- 25° F to + 212° F
Voltage:	10-14 VDC Operation
Voltage:	8 VDC cold start high differential across valve
Duty:	Valve must be capable of being continuously energized at + 212° F without failure

Flow capacity:	Equivalent to 0.160 diameter sharp edge orifice Cd 0.65
Leakage rate:	1 scc/hr at 1 500 PSIG differential pressure
Cycle life:	>54 000 cycles
Hydrogen temperature:	Max + 212° F

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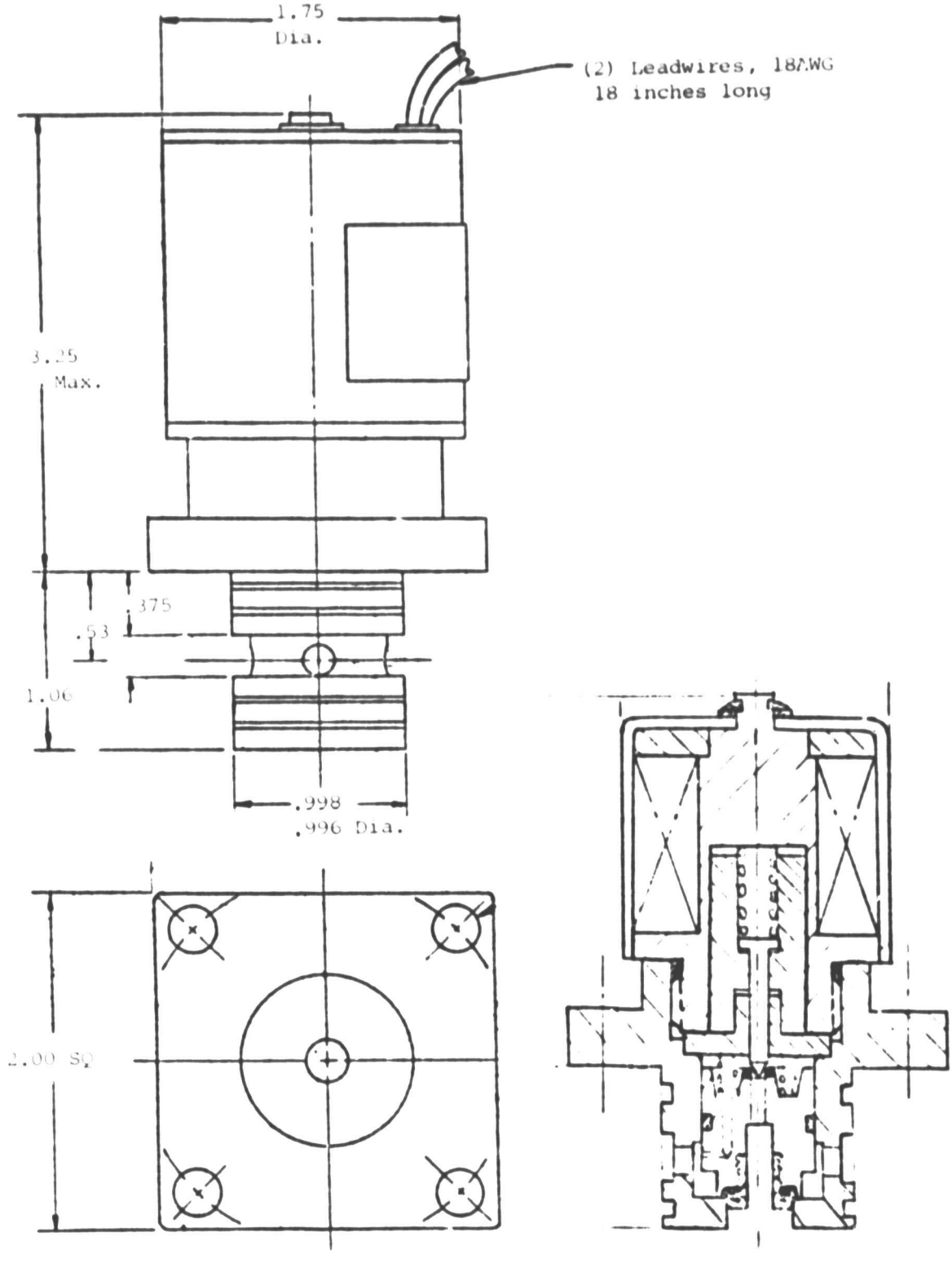


Fig 8.4:50 Shut-off Valve.

Common Features for the Solenoid Valves

The valves are designed to plug into a smooth bore on the power control blocks. They will be mounted in place via four stainless steel socket head bolts. The solenoids will be fabricated from Class H materials and will be fitted with standard 18 gauge lead wires 6 inches long.

Materials of Construction

Valve body:	High strength aluminum or stainless steel
Solenoid core:	430 St/St
Solenoid shell, end pieces:	Cold rolled stee-nickel plated
Springs:	17-7 PH or 302 stainless steel
O-rings:	Ethylene propylene

Leakage Rates

The ASE Mod I leakage requirements for the solenoid valves are very tight. Valcor points out that requirements for spacecraft applications generally range from 1 scc/hr* to 10 scc/hr. On a production basis today they consider the Mod I requirements difficult to achieve. At this time Valcor will accept the leakage specification as a "Target" and firm requirement could be established after completion of the initial prototypes.

Experience

USSw has used Valcor solenoid valves for more than two years with good results. The failures noted with valves have generally been caused by wear from particles (pieces of check valves, dirt) within the hydrogen system due to bad or no filteration of hydrogen gas.

*SCC = STD x cm^3

The Valcor experience with solenoid valves for high pressure gases is extensive from Technical Proposal 792-015:

".....Our series 680, 800, 27211 are currently in use throughout the aero space industry on critical applications for both flight and ground support. We designed, developed and flight qualified a very special high pressure gas valve for NASA on the Skylab space vehicle. This valve controlled, by means of a precise pulse of high pressure gas, the altitude of the space vehicle. Operating pressures varied from 500 - 3 500 PSIG. Ambient temperature ranged from - 150° F to + 180° F. The valve thruster system functioned flawlessly throughout the design mission life.

It also continued to function beyond its designed life so that the pneumatic system was available for use during several emergency situations at the end of the mission.

We have also supplied valves for various classified airforce satellite programs, that required extremely low leakage rate at 3 500 PSIG helium and nitrogen gas.

In addition to our rather exotic work in the aero space industry we manufacture a low cost line of commercial valves that are produced in very large quantities, typically lots of 50,000 to 10,000.

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8.4.8.11 Hydrogen Cooler (12)

To lower the compressed hydrogen temperature a hydrogen cooler is mounted in modular block 3. This cooler prevent overtemperature at the o-ring sealed check valve and the compressor short-circuiting valve.

The design of the cooler is similar to the normal cycle gas cooler used in the P-40 but is a much smaller size. Number of tubes are 80 and their length = 50 mm

Fig 8.4:51 shows the cooler within block No 3.

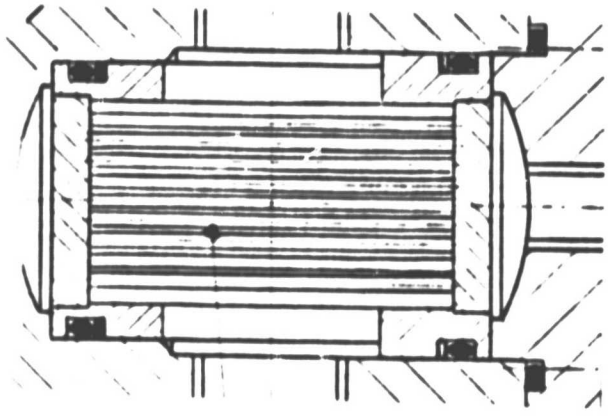


Fig 8.4:51 Hydrogen Cooler in Power Control Block 3.

Material: The material of the cooler will be similar to the cycle coolers.

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8.4.8.12 Pressure Transducers (13)

The ASE MOD I power control system is equipped with three pressure tranducers.

Transducer 1: Measures engine max pressure. This transducer controls the hydrogen pressure and is the only pressure transducer which is needed for control purposes. The transducer is mounted in block 1.

Transducer 2: Measures tank pressure. This transducer is only for monitoring the tank pressure. The transducer is mounted in block 2.

Transducer 3: Measures compressor suction pressure. This transducer is for monitoring the suction pressure of the compressor. The transducer is mounted in block 3.

All transducers will be of the same type (Bell Howell BHL-4050-00-250 Bar).

This is an unbonded strain gauge transducer with stainless steel body and diaphragm. Connecting cable is waterproof.

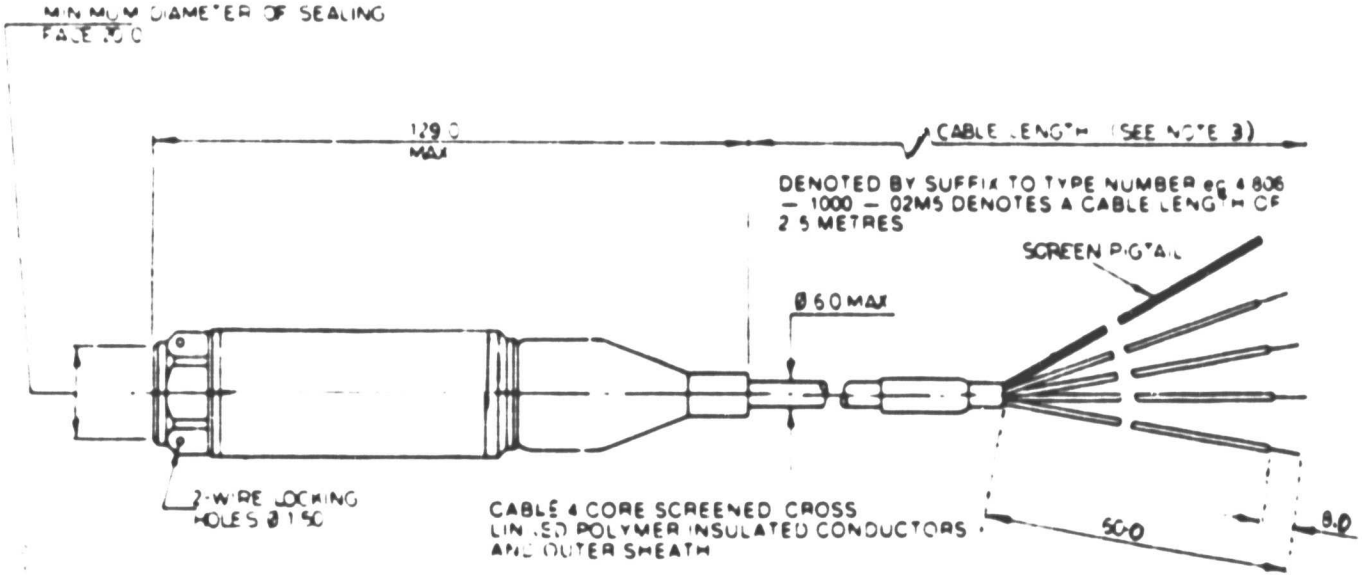


Fig. 8.4:52 Pressure Transducer BHL 4050-00-250 Bar

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This transducer is almost identical to the tank and compressor transducers used on P40 engine. These transducers have been very reliable and so far no failures have been reported.

These transducers have following specification:

Type number:	BHL-4050-00
Input	
Pressure range:	0-25 MPa Sealed Reference
Pressure limits:	Twice rated pressure, when applied for 3 minutes will not cause a zero set exceeding 0.5% FRO.
Rated electrical excitation:	10 V d c
Maximum electrical excitation:	12 V d c
Resistance:	350 Ω nominal 330 Ω maximum at 25 °C
Output	
Full range output (FRO):	Minimum 36 mV, maximum 48 mV into open circuit at rated excitation and +25 °C.
Residual unbalance:	0 mV ± 5% FRO
Natural frequence: (mechanical)	Approx 40 kHz
Resistance:	350 Ω ± 10% at +25 °C.
Resolution:	Continuous
Environmental	
Temperature range:	-10 °C to +120 °C (compensated) -54 °C to +120 °C (operable)
Thermal zero shift: Coefficient	Maximum 0.05% FRO/°C
Thermal sensitivity shift: Coefficient	Maximum 0.03% FRO/°C
Vibration:	At 30 g peak sinusoidal vibratic from 5 Hz-2 kHz the response is less than 0.04% FRO/g

Ambient pressure:	2 MPa absolute, will not damage the transducer.
Humidity:	Meets MIL-E-5272
Mechanical shock:	1 000 g half sine wave pulse for a duration of one millisecond will not damage the transducer.

Physical

Material in contact with pressure media:	17-4 PH stainless steel
Electrical connection:	Integral sealed cable assembly
Pressure connection:	R 1/4" to DIN 259
Weight:	125 g including cable
Insultion:	Cable screen, exitation and output, isolated from case. Insulation resistance 500 MΩ min at 85 V d c.

8.4.8.13 Safety Valve (14)

The ASE MOD I power control system is equipped with two safety valves. One is mounted in the p-line before the control valve. This valve prevents too high pressures inside the engine caused by any malfunction of the power control valve or improperly supply gas to the engine through the refilling valve. This safety valve is on Mod I mounted in block 1.

The other safety valve is placed between compressor and hydrogen tank preventing too high pressures after the compressor in the event the manual shut-off valve (19) should have been closed (by accident). The safety valve also prevents too high tank pressure during engine operation. This safety valve is mounted in block 2.

The safety valve is of own design eliminating the problems which have occurred with the P40 safety valves.

Fig 8.4:53 shows a sectional drawing of the components belonging to the safety valve.

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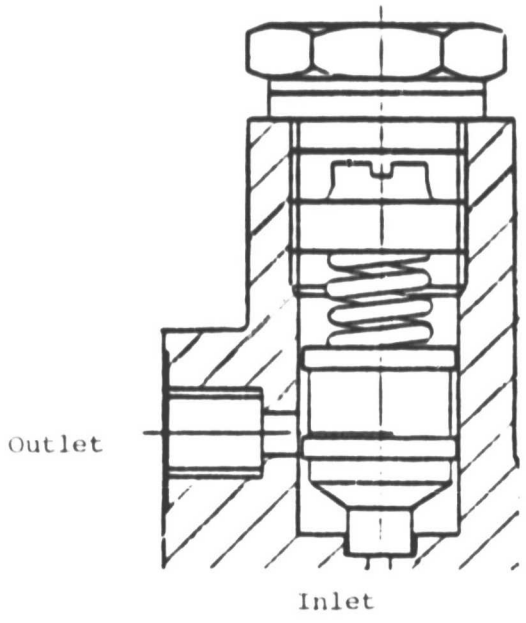


Fig. 8.4:53 ASE Mod I Power Control Safety Valve

To assure a leak tight valve the sealing element is an O-ring. To prevent damage on the O-ring during the opening and closing phase, special attention has been made to design the sealing elements. Fig. 8.4:54 shows the valve in a closed position, during the opening phase and in open position.

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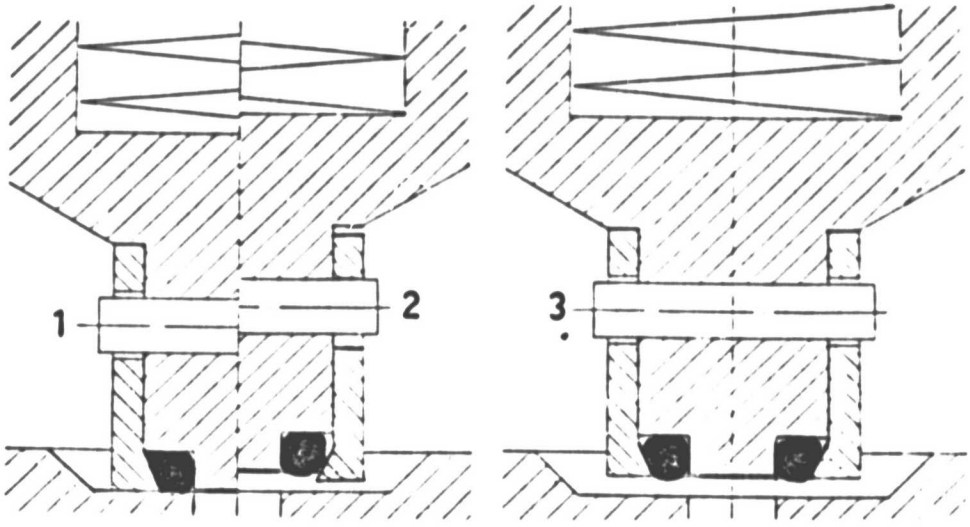


Fig. 8.4.54 Safety Valve Operation

1: shows the safety valve in closed position

2: shows the safety valve during the opening phase (note that the outer O-ring holder prevents the O-ring from blowing away.

3: shows the valve in open position.

Tests have shown that this valve could be opened several times and still seal against atmosphere.

Component specification

Working medium:	Hydrogen
Opening pressure:	22.5 MPa
Closing pressure:	> 17.5 MPa
Entrance orifice:	3 mm ESEOD

8.4.8.14 Hydrogen Storage Vessel (18)

A pressure vessel of 7 litres in volume will be used although partly closed piston domes permit use of a smaller vessel in automotive application. In laboratory test cells connecting tubes from monitoring manometers and measurement transducers adds a volume equal or larger than that of the closed dome volumes.

8.4.8.14.1 Design

The hydrogen storage vessel in MOD I will be identical to that of all P40 engines.

It is a standard off-the-shelf pressure vessel for compressed air used for diving and fire-rescuing purposes. It is designed by the Swedish company AGA (their designation ORC7) and manufactured by Industriwerke Karlsruhe (IWK) in Germany.

The bottle is seamless and cold-drawn. The starting material is a round piece, which by deep drawing in many stages will get the final cylindrical shape. Between the deep drawings the material is annealed and phosphatized.

The bottle neck is thermoformed in rotating tools. The bottle is also tempered.

Fig 8.4:55 shows a drawing of the hydrogen vessel ORC7.

Design Criteria

The bottle is designed for a proof pressure of 45 MPa and a nominal working pressure of 30 MPa (air).

The design, manufacturing, inspection and testing are in accordance with Druckgasverordnung (German Pressure Vessel Code) and Swedish Pressure Vessel Code.

Bottle specification:

Volume:	7.0 - 7.3 litre
Diameter:	140 mm
Length:	610 mm
Weight (empty):	9.2 - 9.7 kg
Wall thickness:	4 + 0.3 mm
Bottom thickness:	4.8 + 0.3 mm
Max charge pressure:	30.0 MPa
Proof pressure:	45.0 MPa

Material specification.

The pressure vessel is made of:

Steel 30 CrNiMo 8, DIN 16580

C	0.26 - 0.33%	S	< 0.035%
Si	0.15 - 0.40%	Cr	1.8 - 2.2%
Mn	0.3 - 0.6%	Mo	0.3 - 0.5%
P	< 0.035%	Ni	1.8 - 2.2%

Yield strength $\sigma_S \geq 100$ kp/mm^2
Ultimate tensile strength $\sigma_B \geq 111$ kp/m^2
Elongation $\geq$ 13%

Manufacturing Inspection

In production all pressure vessels are carefully inspected regarding wall thickness, hardness, inner and outer surfaces, threads etc. All pressure vessels will then be tested to 45 MPa.

8.4.8.14.2 Hydrogen Pressure Variation in the ASE MOD I Storage Vessel

Depending on the engine power variations there will be pressure variations in the storage vessel. The amplitude of these variations depends on: Tank size, gas volume of the Stirling cycles, open or closed domes, volume of the outer gas system and gas volume of monitoring equipment (manometers etc).

Amount of hydrogen in the engine cycles (at 15 MPa $\bar{p}$)		
ASE MOD I off-d 3-00	=	0.0115 kg
Amount of hydrogen in domes ($\check{p}$) (partly open)	=	0.0025 kg
Seal filter 150 cm^3 ($\check{p}$)	=	0.0013 kg
Amount of hydrogen in power control tubes, valves, filters about 80 cm^3 $\hat{p}$	=	0.0012 kg
Total amount of hydrogen in engine at 15 MPa $\bar{p}$	=	0.0165 kg

Of this total amount of hydrogen a great part is transferred to or from the engine during engine operation.

p variation from 19.1 MPa (full load) to 4 MPa (idling) gives an amount of transferred hydrogen of 0.0130 kg.

Lowest practical tank pressure at full load conditions will be set to 15 MPa. Corresponding amount of hydrogen in a 7 litre tank at $70^{o}C$ will be 0.0840 kg.

At engine idling 0.0130 kg of hydrogen will be transferred to the tank giving a total amount of 0.0970 kg.

That corresponds to a tank pressure of 17.3 MPa.

To compensate for external leakage during the refilling intervals an extra amount of hydrogen must be added to the tank. This reserve capacity has been set to 210 standard litres of hydrogen giving an increase of pressure of 3 MPa. As a consequence, during engine operation the tank pressure will vary between 20.3 MPa (engine idling) and 18 MPa (full load conditions).

In test cell installations an extra amount of about 1000 cm^3 of hydrogen (0.0122 kg) must be added to the transferring amount of hydrogen due to monitoring equipment. In that case the tank pressure levels will vary from 22.5 (engine idling) to 18 MPa (full load conditions).

Figure 8.4:56 shows storage vessel pressure and corresponding engine pressure in an automotive and a test cell installation.

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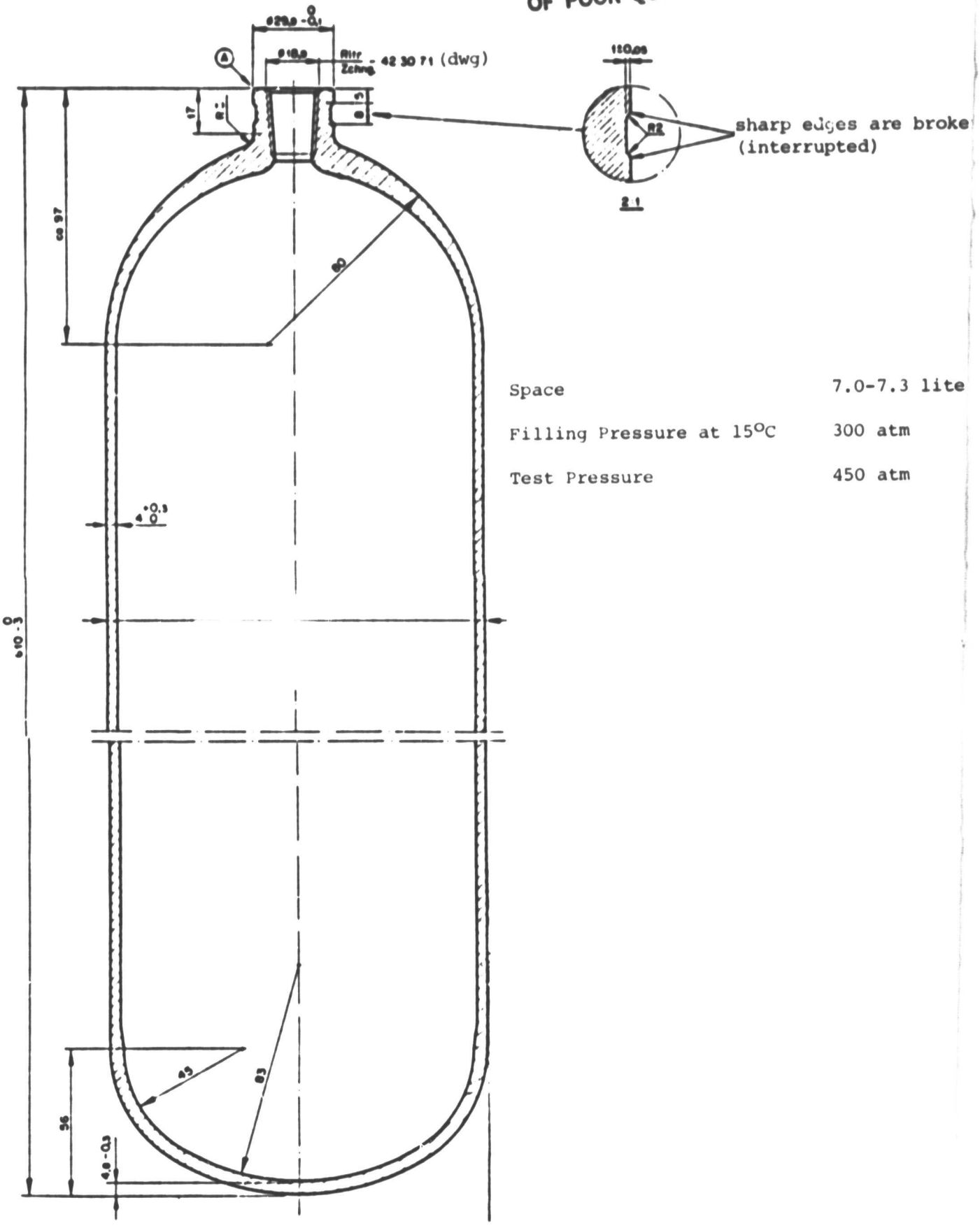


Fig 8.4:55 Hydrogen Vessel AGA ORC7

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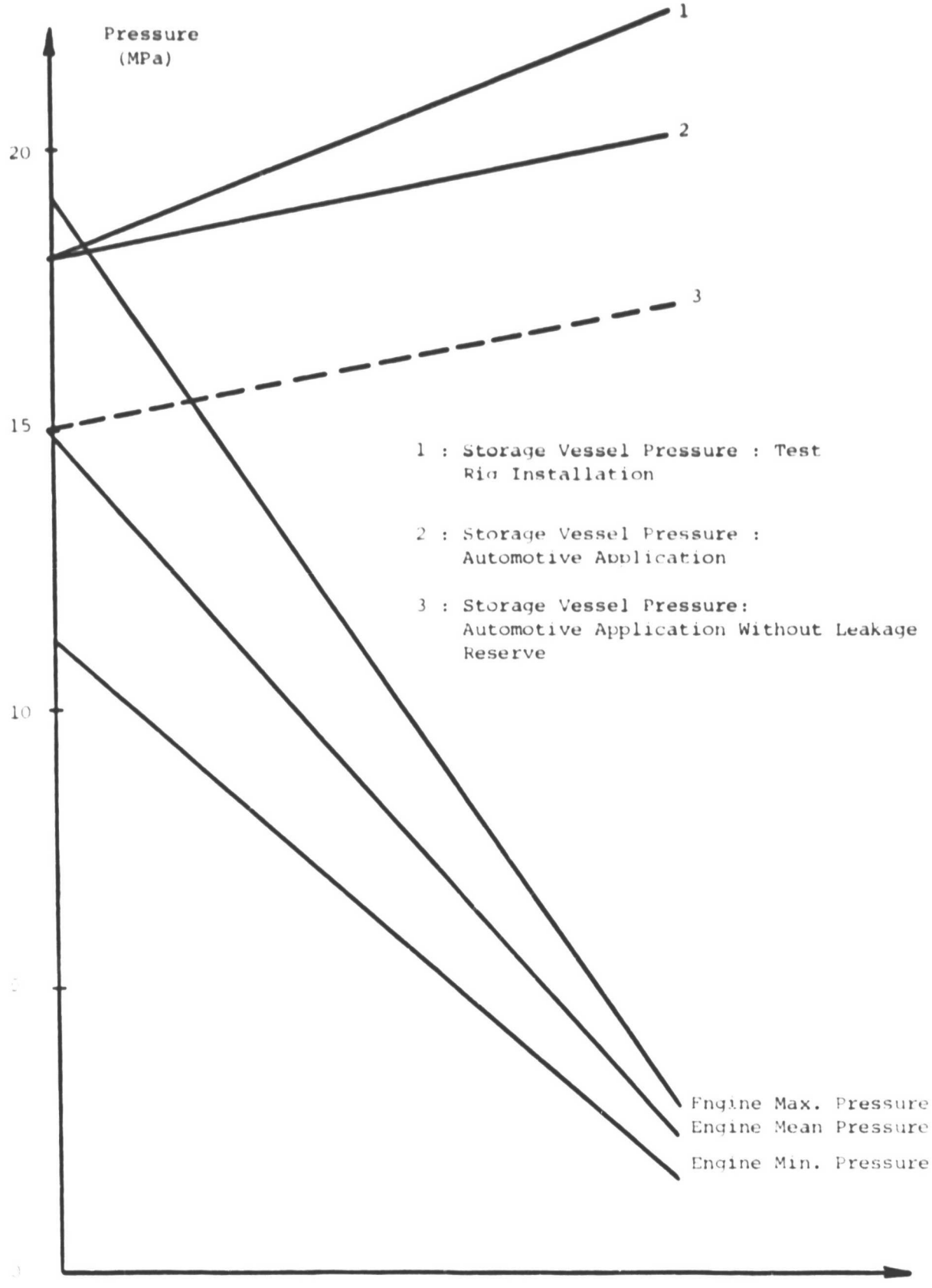


Fig 8.4:56 Hydrogen Storage Vessel Pressure and Corresponding Engine (Mean) Pressure for ASE MOD I.

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8.4.8.14.3 Stress Analysis

The hydrogen storage vessel is made of tempered steel DIN 16580. The following material data are valid:

Yield stress (0.2%) σ_y = 980 MPA

Ultimate tensile strength σ_u = 1 090 MPa

Fatigue limit σ_f = 460 MPa

The design fulfils the Swedish as well as the German gas bottle code (a safety factor of 1.33 against yielding at the proof pressure of 45 MPa).

A burst test with oil has been performed by NASA Lewis Research Center. The yielding started at a pressure of 62 MPa (9000 psi) and the burst occurred at a pressure of 77 MPa (11,200 psi). Using the mean diameter formula with the actual thickness of 4.5 mm these pressures correspond to Hoop stresses of 992 MPa and 1232 MPa, respectively. The Hoop stress at yielding is in good agreement with the specified yield stress. The discrepancy considering the burst stress and the ultimate tensile strength is probably due to the listed value being a minimum guaranteed rather than a nominal one.

The vessel is subjected to a cyclic pressure of 19.15 $\pm$ 1.15 MPa in a vehicle and 20.25 $\pm$ 2.25 MPa in a test cell. The vessel has been fatigue tested with a cyclic pressure of 21.5 $\pm$ 3.5 MPa at the Swedish National Testing Institute. The test was successfully performed with 10 million cycles, see the enclosed certificate, appendix 8.4:2.

Fig 8.4:57 shows a Goodman diagram for the pressure vessel. The fatigue limit is reduced because of the surface conditions. The mean diameter formula gives the following Hoop stresses using the nominal size of the cross section of the pressure vessel:

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1. Test cell = 345 ± 21 MPa
2. Vehicle = 365 ± 41 MPa
3. Fatigue test = 387 ± 63 MPa

The diagram shows that the pressure vessel has a safety factor larger than 2 for the two specified applications.

Four pressure vessels have been delivered to NASA Lewis Research Center for hydrogen embrittlement investigations.

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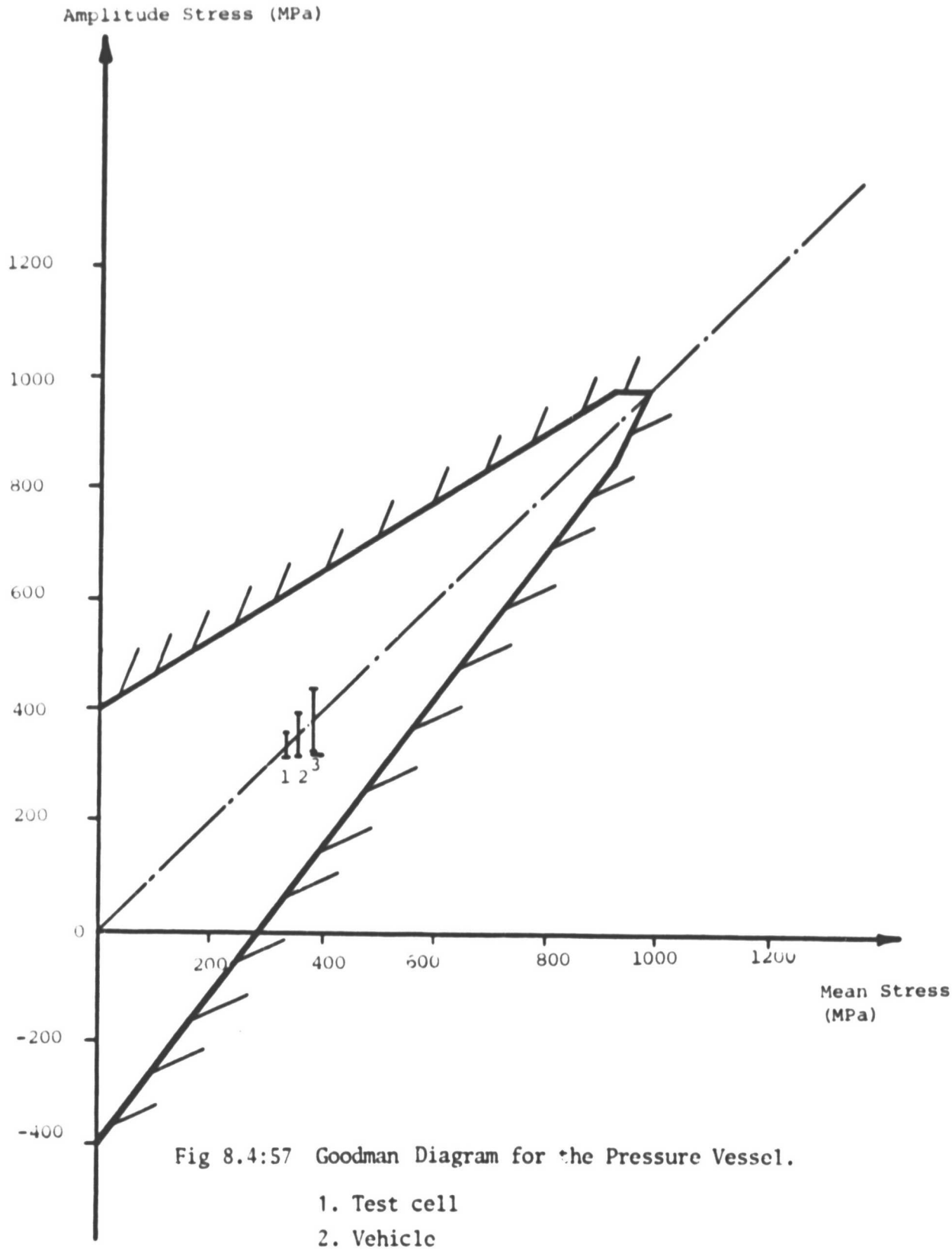


Fig 8.4:57 Goodman Diagram for the Pressure Vessel.

1. Test cell
2. Vehicle
3. Fatigue test

SP STATENS PROVNINGSANSTALT

REPORT
Datum
1980-04-09 8011,048a

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KB United Stirling
(Sweden) AB & Co
Fack
201 10 Malmö

Object Gas bottle

Assignment Fatigue testing, your order no 3912

Test procedure Fatigue testing on gas bottle has been completed to 10^7 cycles without failure. The load has been oscillated between 18 and 25 MPa at a frequency of 12 Hz. The temperature of the bottle was kept below 35°C during the test.

Swedish National Testing Institute
Division of solid mechanics

Christer Ljung

Christer Ljung

SP 4:1

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☐ HUVUDFÖRVALTNING	Box 857	501 15 Borås	Brinelligatan 4	033-10 20 00	362 52 testing S	715-1053
☐ STOCKHOLM	Box 5608	114 86 Stockholm	Drottning Kristinas väg 11	08-24 80 60	103 58 testing S	POSTGIRO
☐ GÖTEBORG	Box 240 36	400 22 Göteborg	Gibraltargatan 35	031-20 08 70	271 61 testing S	1 56 82 8
☐ LUND	Box 712	220 07 Lund	gen 11	046-12 43 10		

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8.4.8.15 Shut-off Valve (19)

The hydrogen storage vessel is equipped with a manual shut-off valve. It will be used during long non operation periods or when it is necessary to remove the tank from the system without emptying the tank. Fig 8.4:58 shows the valve design.

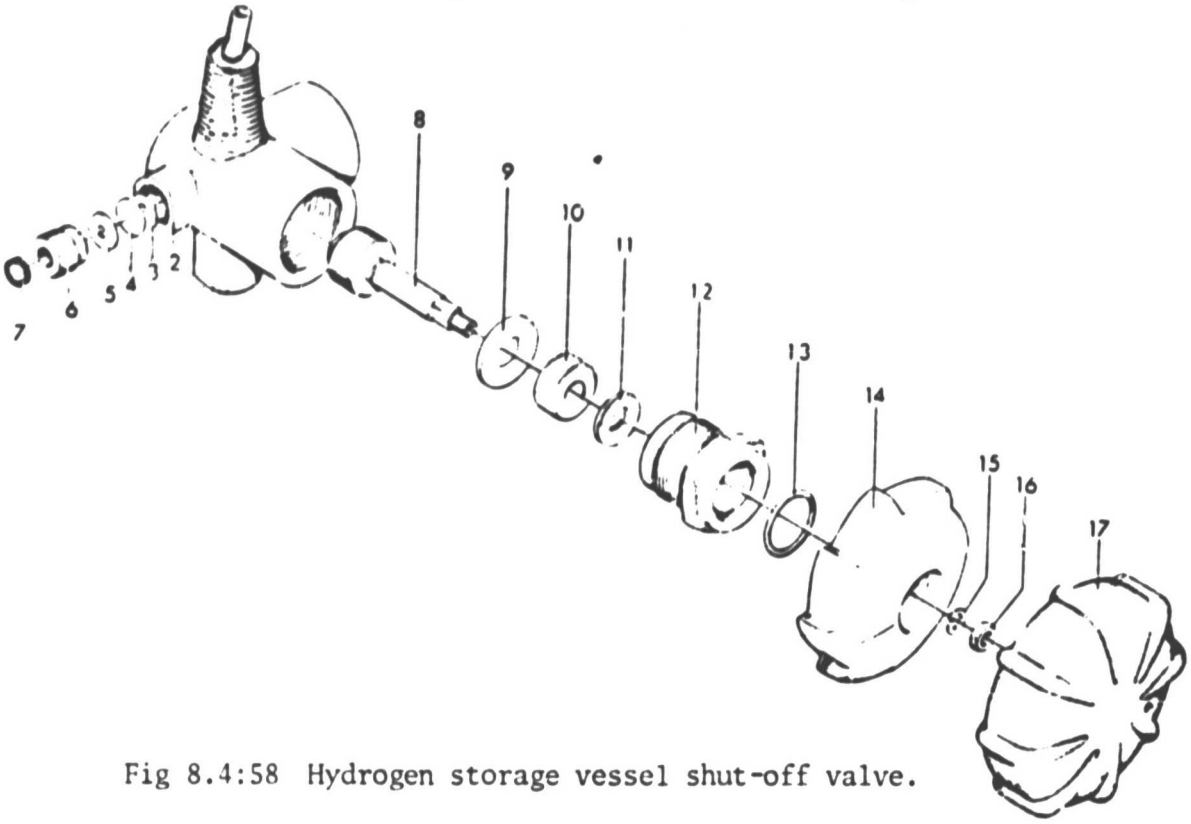


Fig 8.4:58 Hydrogen storage vessel shut-off valve.

The valve is modified according to USS drawing 4-13364 (Fig 8.4:59). A bursting disc preventing overpressure in the storage tank is also built into this valve. Burst pressure of this disc is about 35 MPa. Dumped hydrogen will flow directly to the atmosphere from the valve body as there is no outlet tube.

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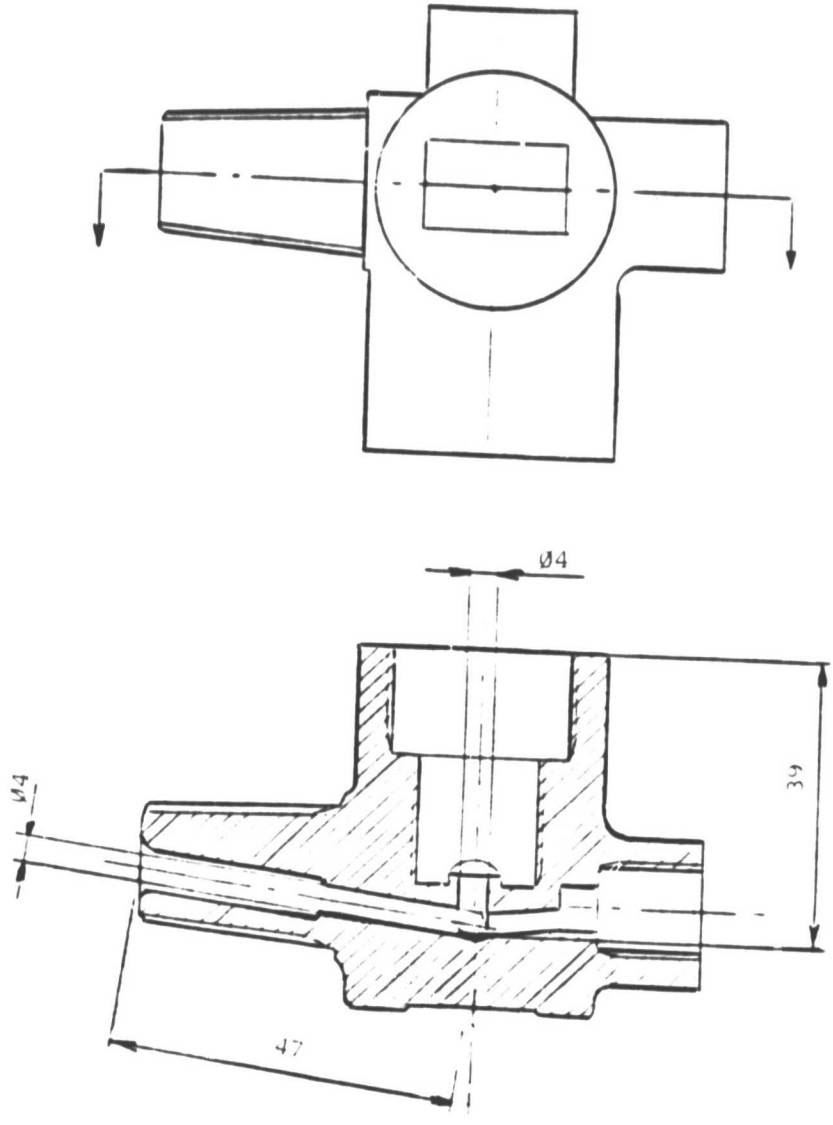


Fig. 8.4:59 Modifications on Shut Off-Valve

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8.4.8.16 Hydrogen Refilling Valve (20)

For refilling the engine from external hydrogen supply the power control system is equipped with a simple manual shut-off valve.

The ASE MOD I will be equipped with the same type of valve as the P40 engines:

Rotelman 1/4" DN 640

8.4.8.17 Fusible Temperature Plug (21)

To avoid overtemperature at the hydrogen storage vessel due to accidental fire and with the risk of vessel bursting a fusible temperature plug will be mounted at the pressure vessel.

The plug consists of a fitting with a melting plug (see fig 8.4:60).

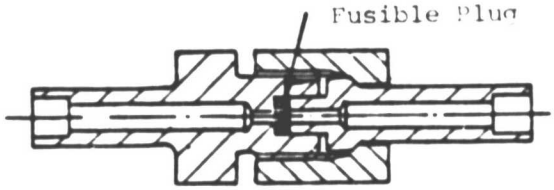


Fig. 8.4:60 Fusible Temperature Plug

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Material of the melting disc: Bi 58% Sn 42% alloy, with a melting point of 138.5° C.

This plug is mounted close to the vessel and is connected directly via the vessel shut-off valve to the inner volume of the tank.

When the disc melts the storage vessel will be emptied through the plug independent of the hydrogen shut-off valve position.

8.4.8.18 Speed Transducer (22)

As a feed back signal of the engine speed to the power control system and the control panel a speed transducer is used.

On ASE MOD I a magnetic pick-up placed in the transmission intercasing will sense the teeth of the synchronizing gears.

The speed transducer will be:

Airpax Standard Magnetic Pick-up 1-0001.

Specification

Output voltage:	30-70 V (P-P)
DC Resistance:	100-130 ohms
Typical Inductance:	32-46 mH
Output Polarity:	Pin "B" positive
Operating Temperature:	-100° to +225° F
Mounting Thread:	5/8 - 18 UNF - 2A

Fig 8.4:61 shows the general layout of the speed transducer.

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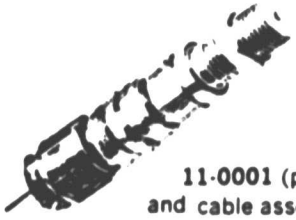
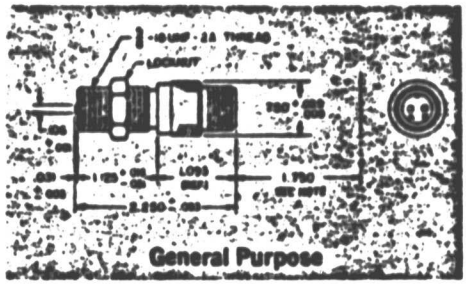
PART NUMBER	SPECIFICATIONS	OUTLINE DRAWING
1-0001 (pickup only)  **11-0001** (pickup and cable assembly)	**Output Voltage:** 30 70V (P P). **DC Resistance:** 100 130 ohms. **Typical Inductance:** 32 46 mH. **Output Polarity:** Pin "B" positive. **Operating Temperature:** -100 to +225°F. **Lead Length:** 10 feet of shielded cable. **Mounting Thread:** ⅝-18 UNF 2A.	

Fig 8.4:61 ASE MOD I Speed Transducer.

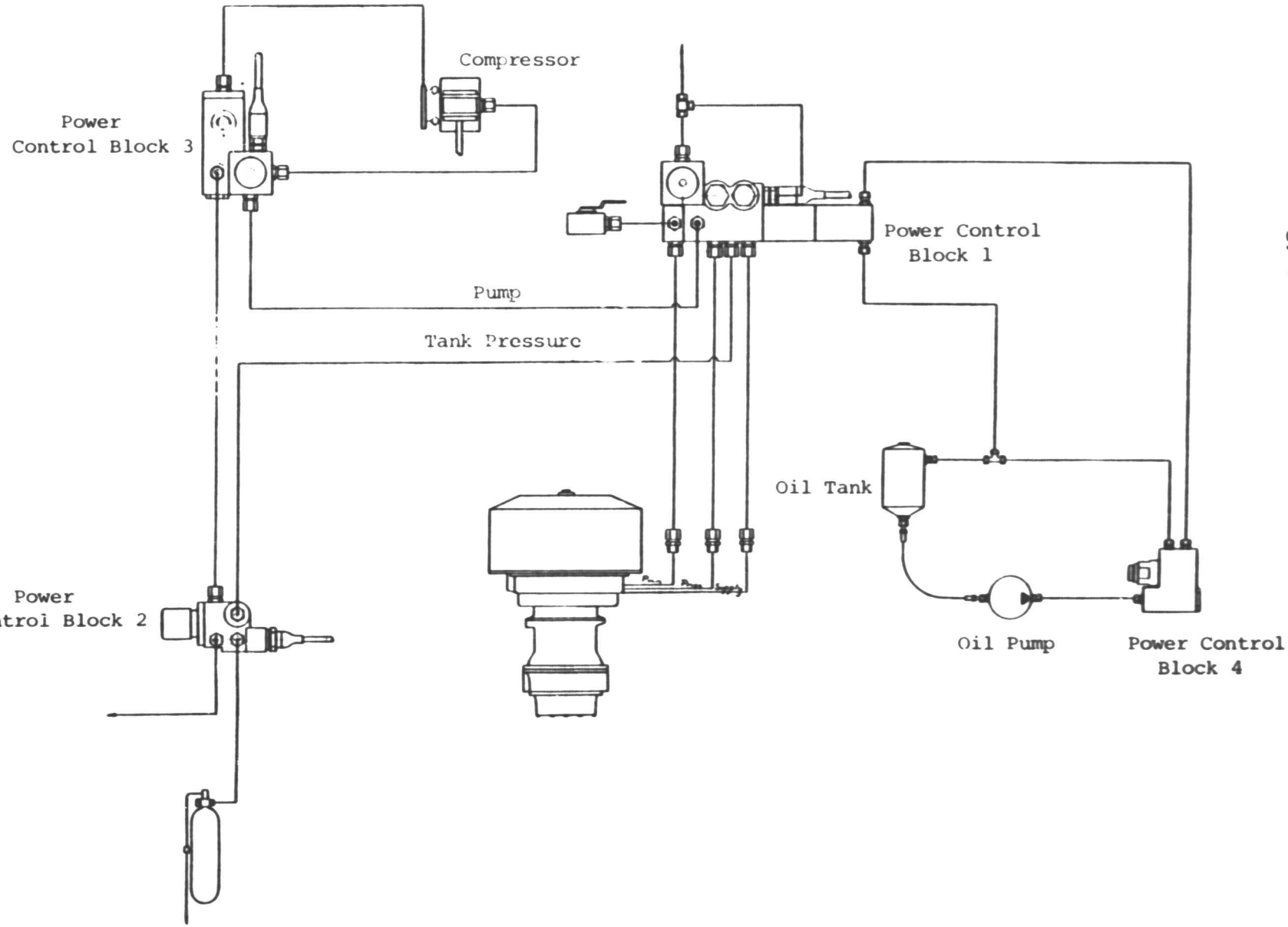
Experience:

The same transducer has been used in all P40 engines.

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8.4.9 Drawings

Figure	Drawing	Description
1	1-S-1489	Power Control System
2	1-S-1470	Power Control Block 1
3	2-S-1471	Power Control Block 2
4	1-S-1488	Power Control Block 3
5	3-S-1452	Power Control Block 4
6	3-73281	Power Control Valve Sleeve
7	4-73282	Power Control Valve Spool
8	A34515	Control Valve Actuator
9	A34898	Gas Valve Actuator
10	3-73506	Test Unit Hydrogen Compressor
11	3-S-1510	ASE Mod I Hydrogen Compressor
12	4-15052	Check Valve
13	4-15053	Check Valve Spring Holder
14	4-14399	Compressor Check Valve Spring Holder
15	152-106-275-B	Hydrogen Bottle AGA ORC7
16	4-13364	Modification of Valve Body for Hydrogen Reservoir



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Fig. 1 Power Control System

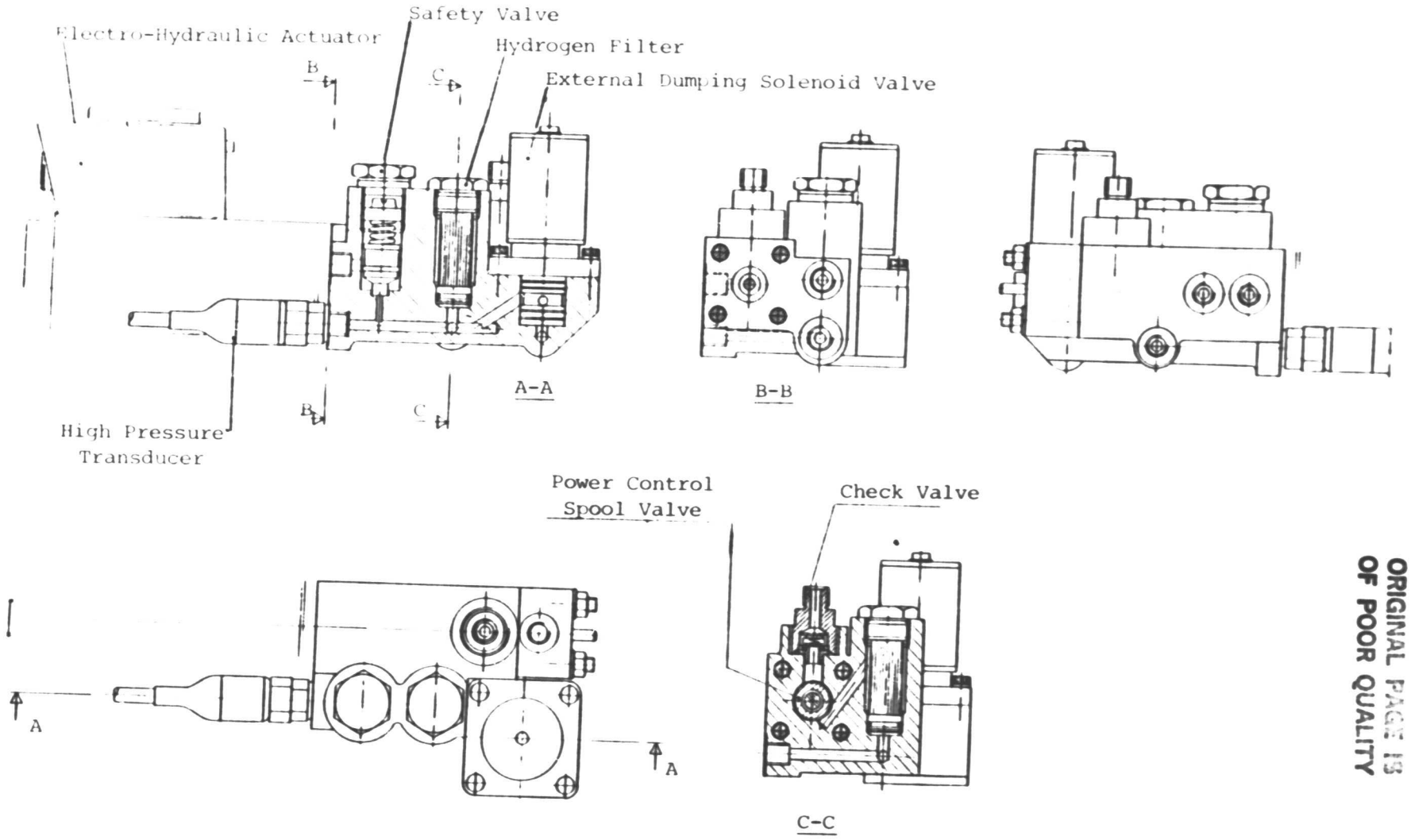


Fig. 2 Power Control Block 1

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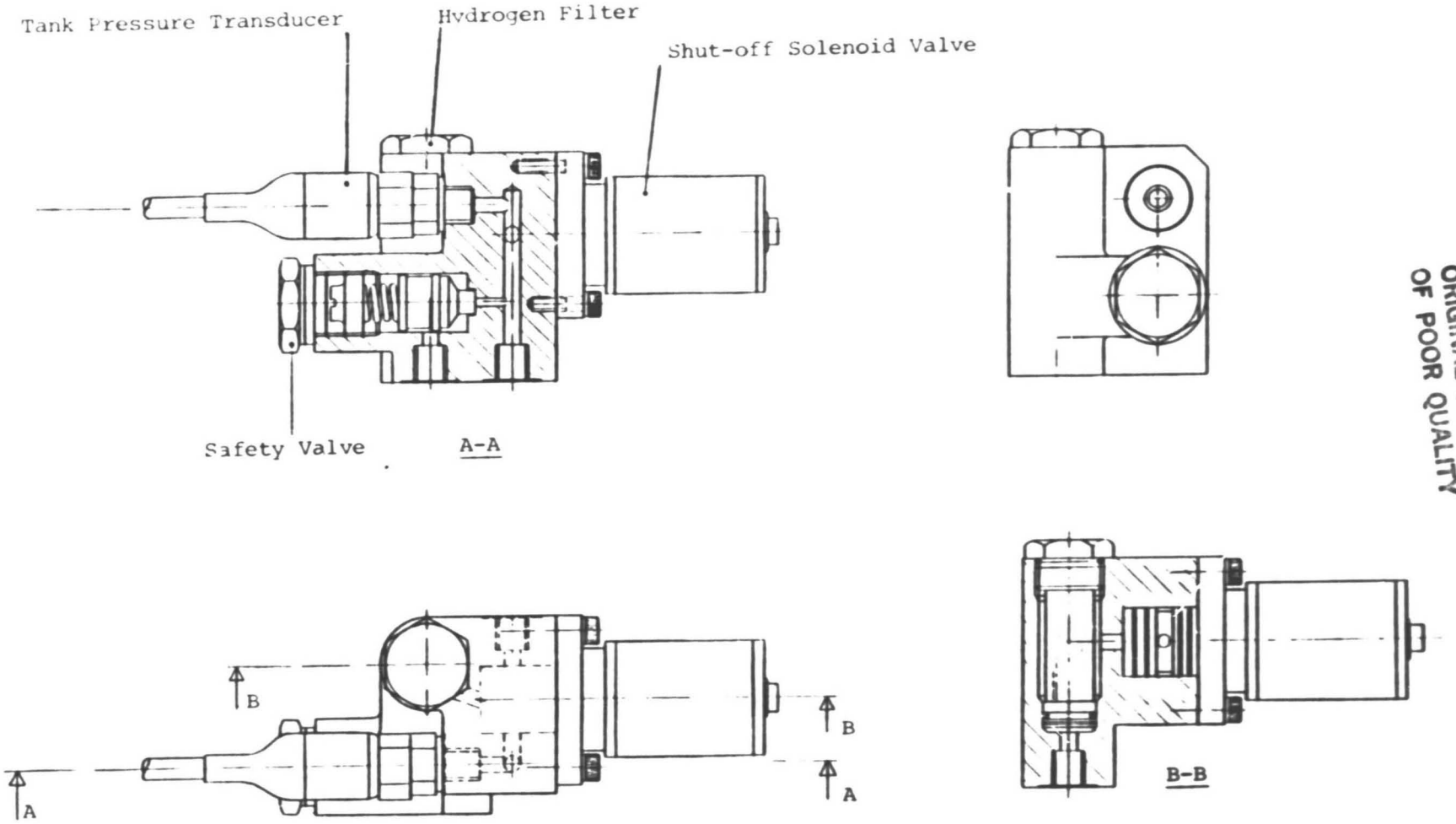


Fig. 3 Power Control Block 2

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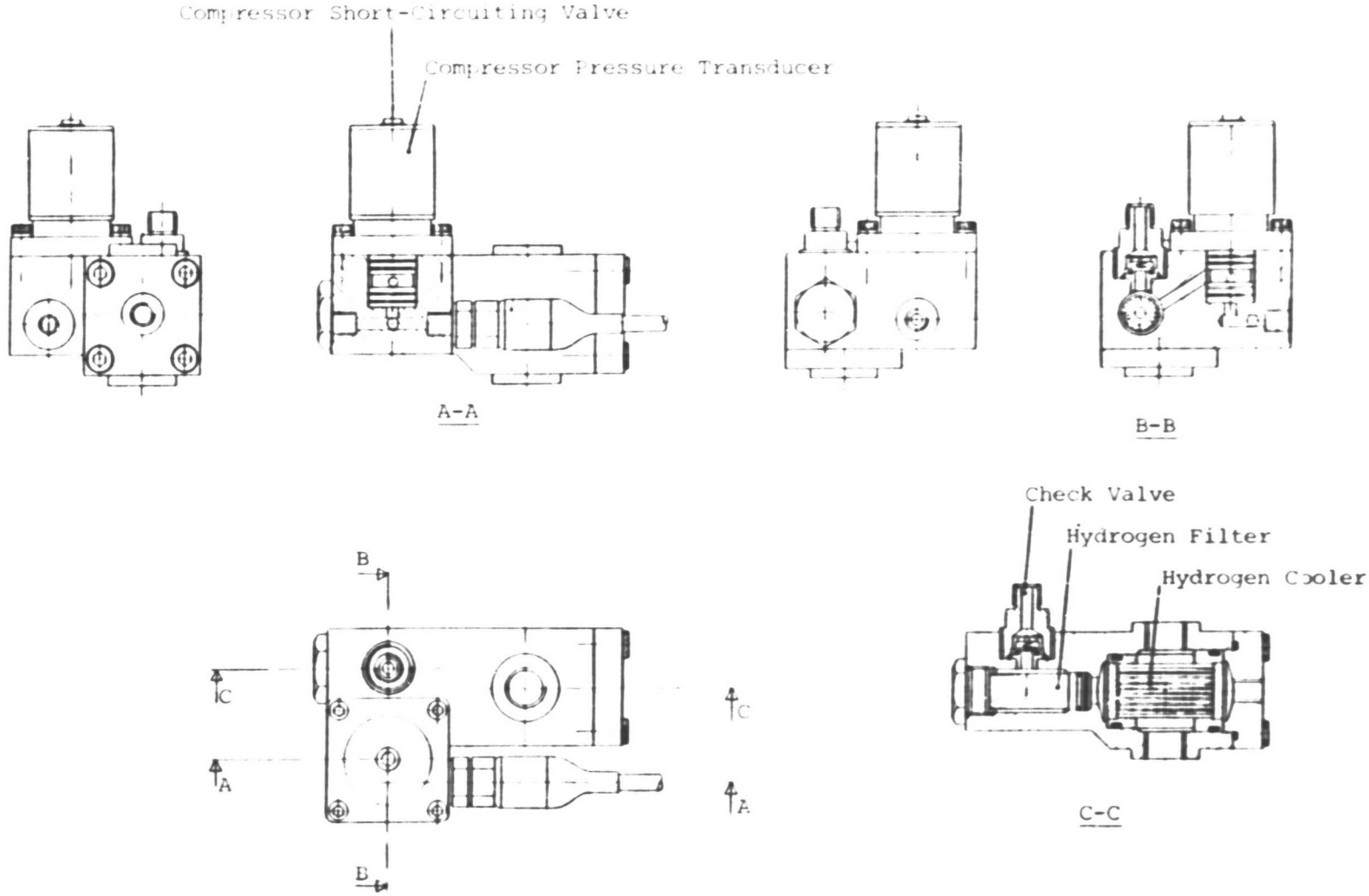


Fig. 4 Power Control Block 3

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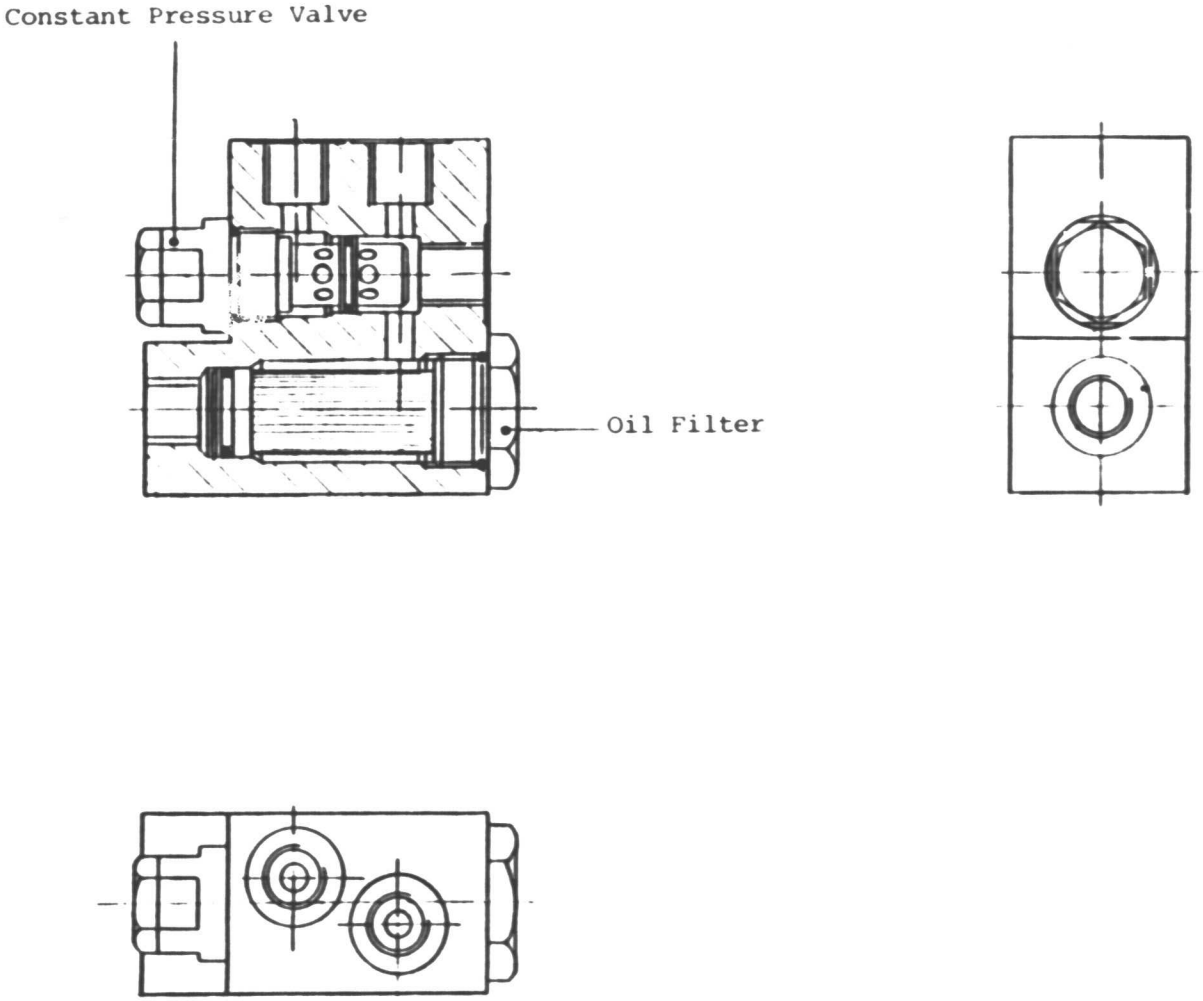
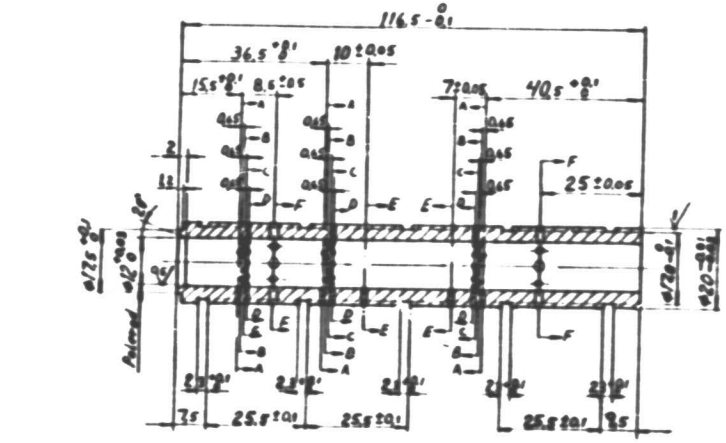
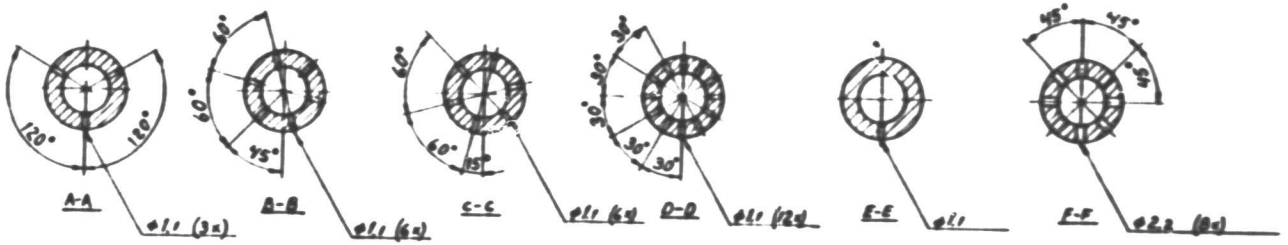


Fig. 5 Power Control Block 4

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Tempered 58-62 ground and polished



Sharp edges are deburred, 0.05-0.10
according to the carbon chemical
liquid hone method

Fig. 6 Power Control Valve Sleeve

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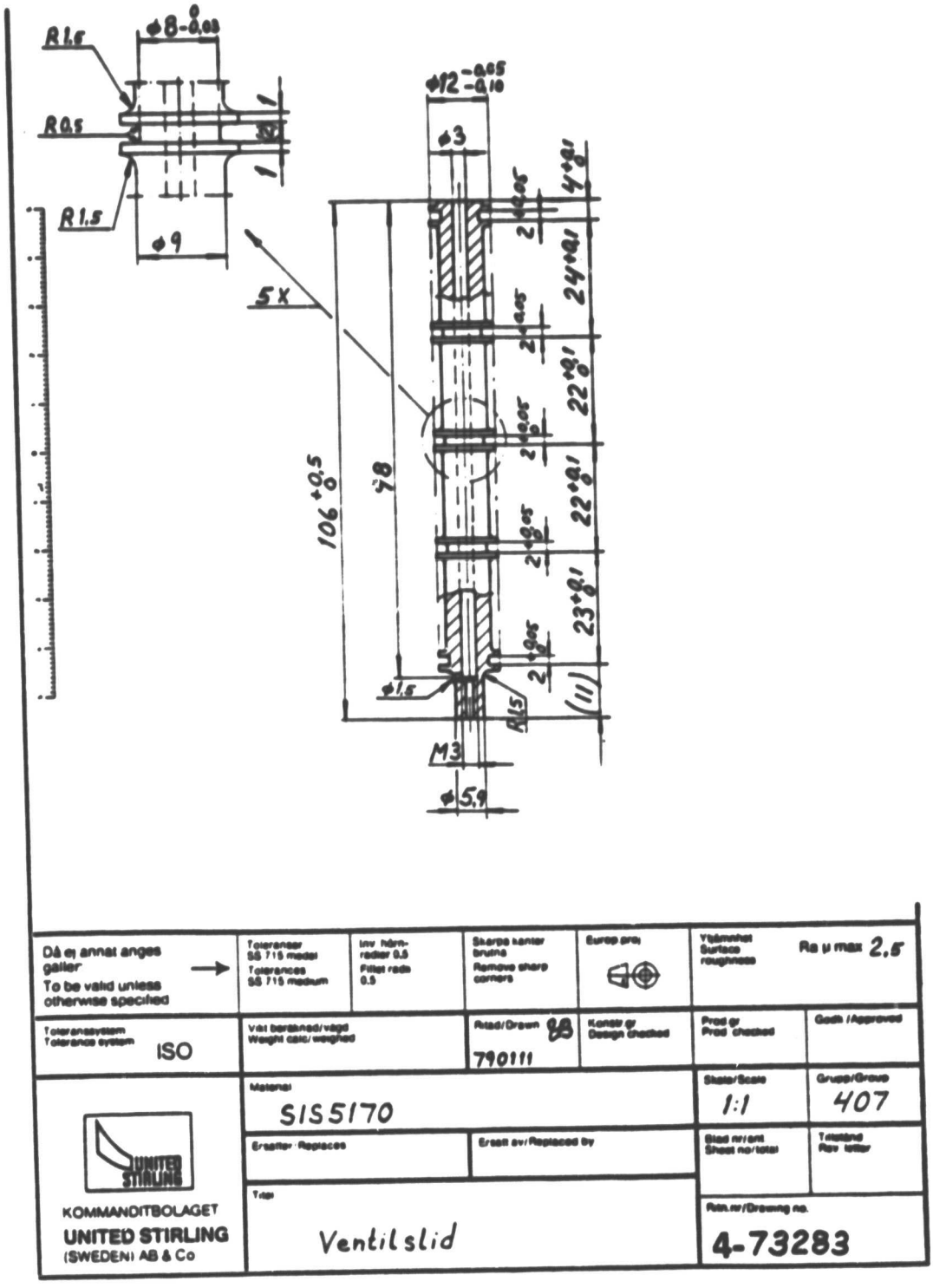


Fig. 7 Power Control Valve Spool

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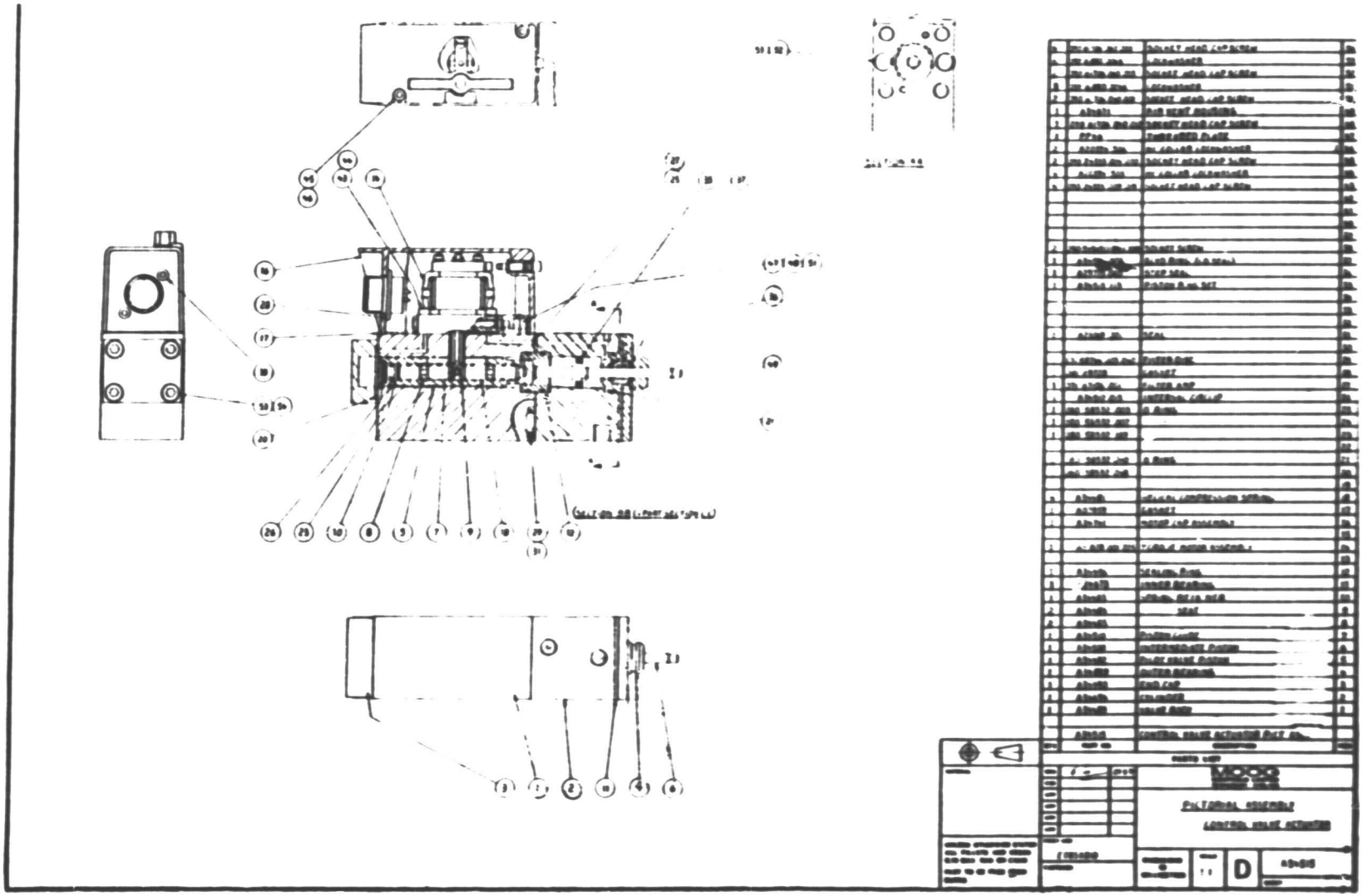


Fig. 8 Pictorial Assembly Control Valve Actuator

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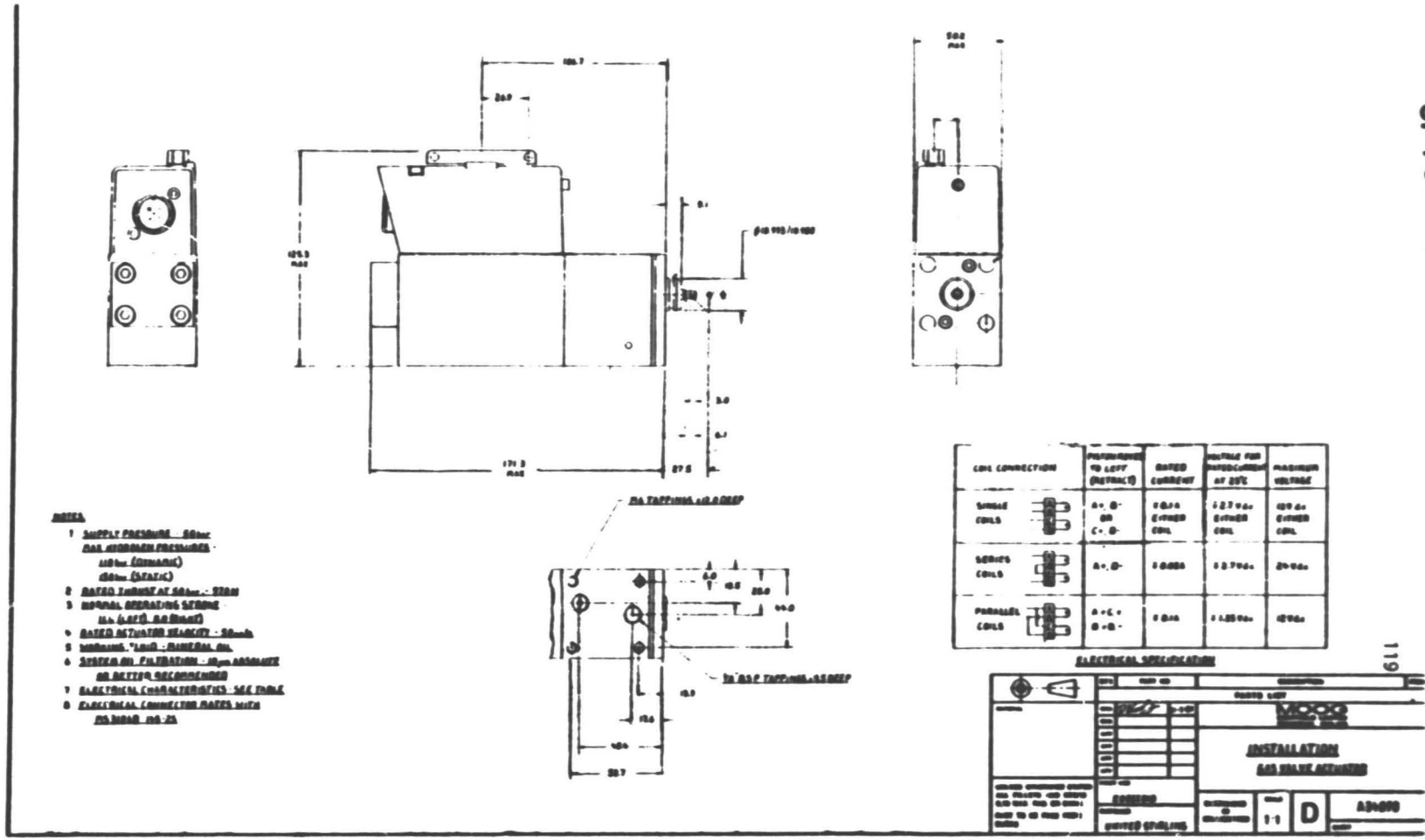
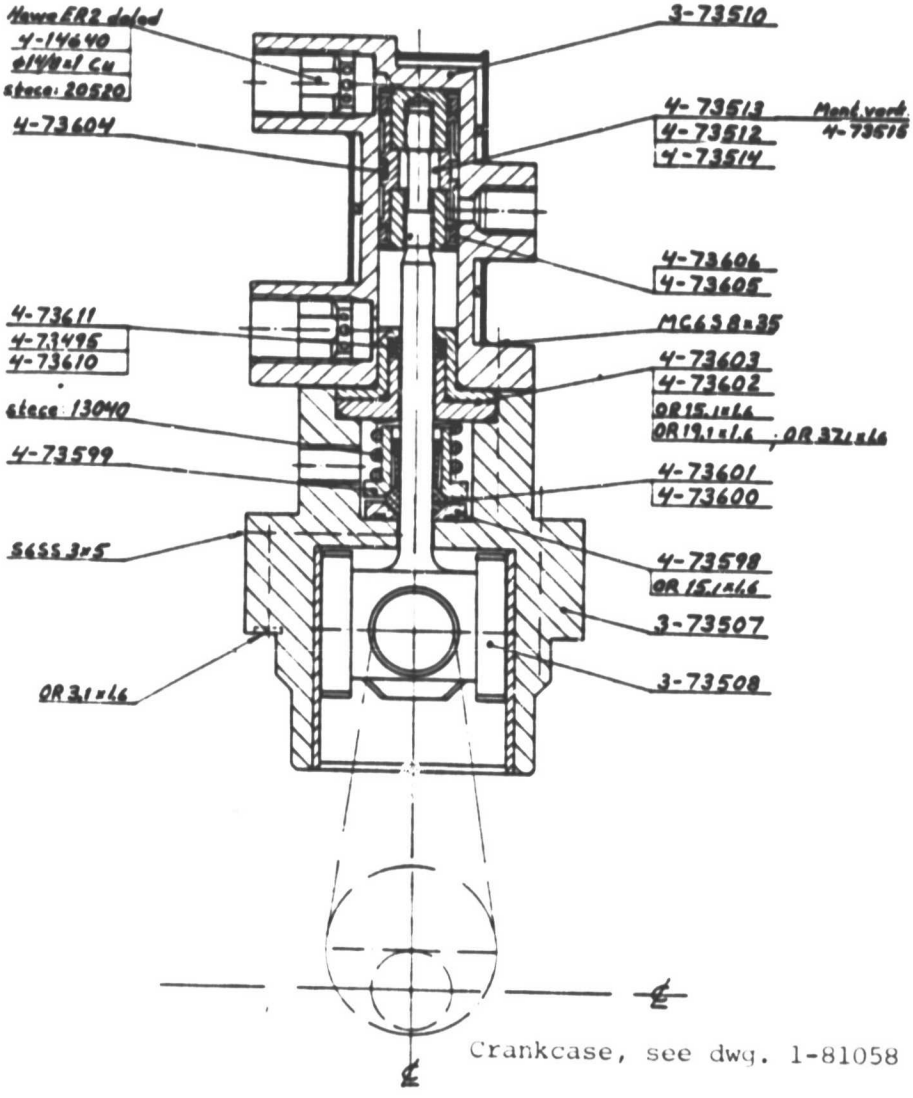


Fig. 9 Installation Gas Valve Actuator

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Prototype for Experiment

Fig. 10 Test Unit Hydrogen Compressor

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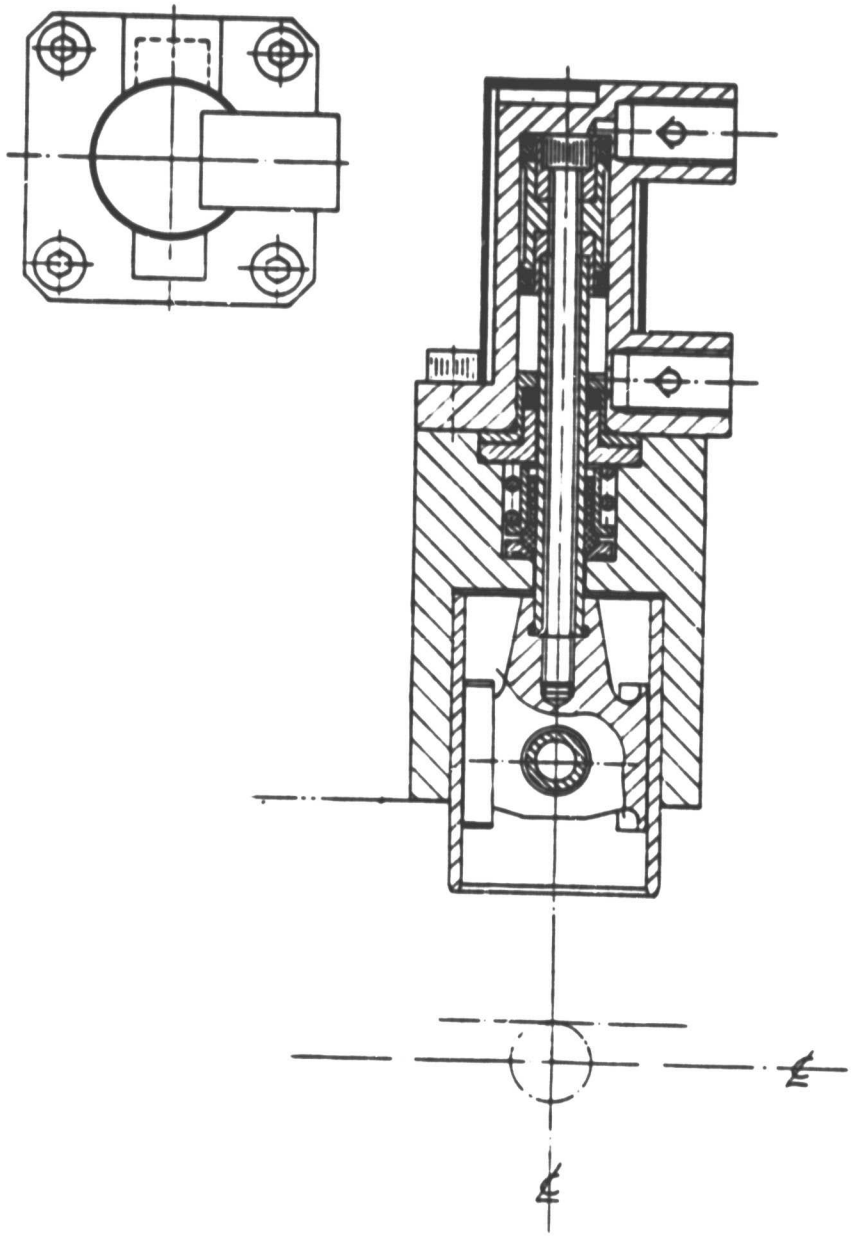


Fig. 11 ASE Mod I Hydrogen Compressor

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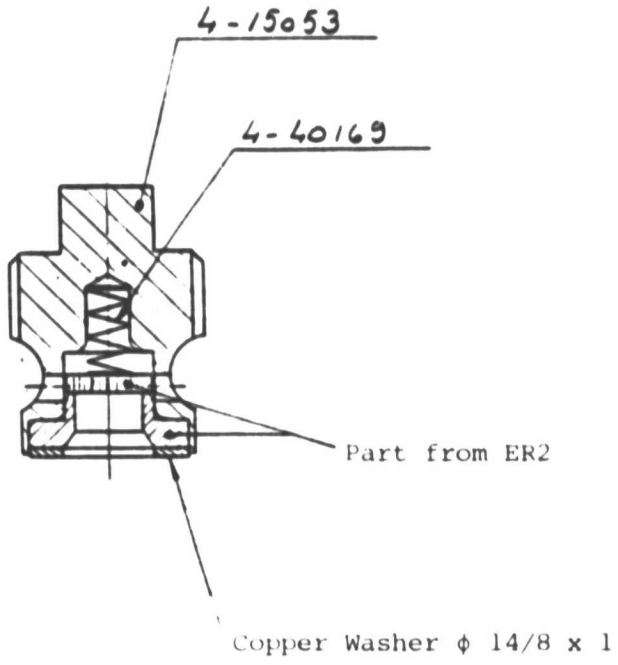


Fig. 12 Check Valve Assembly

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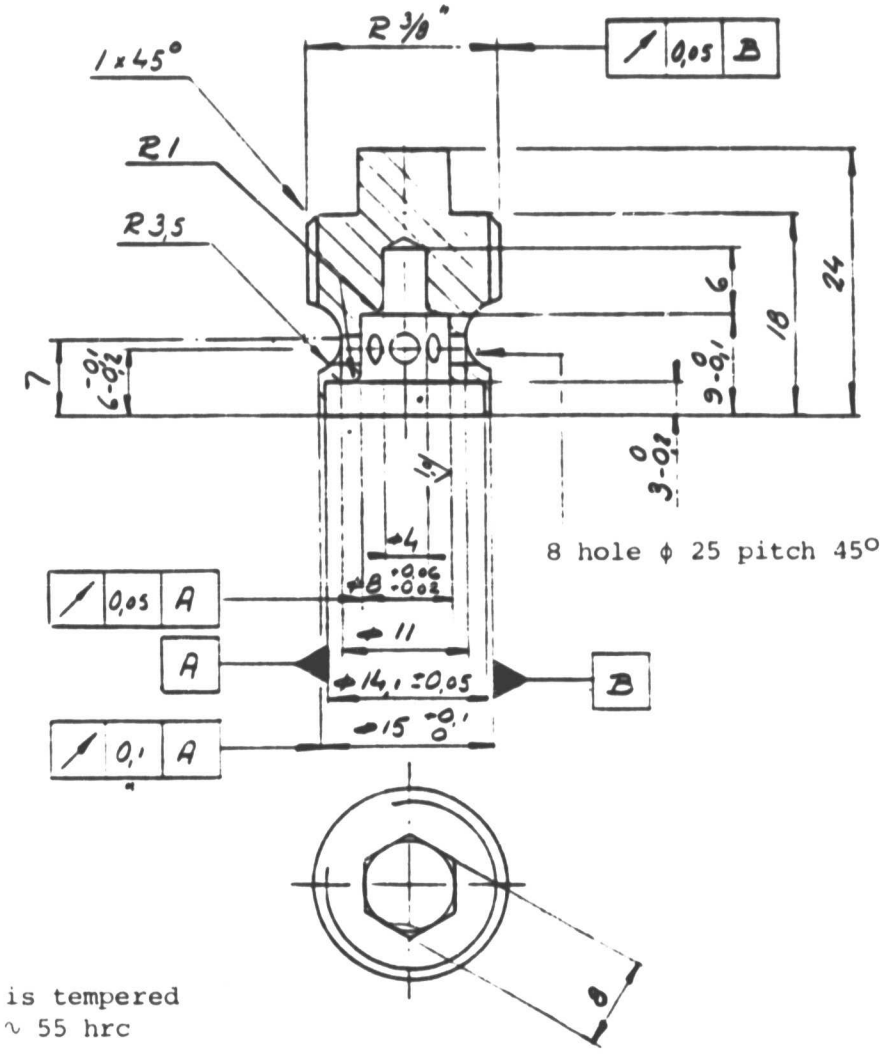


Fig. 13 Check Valve Spring Holder

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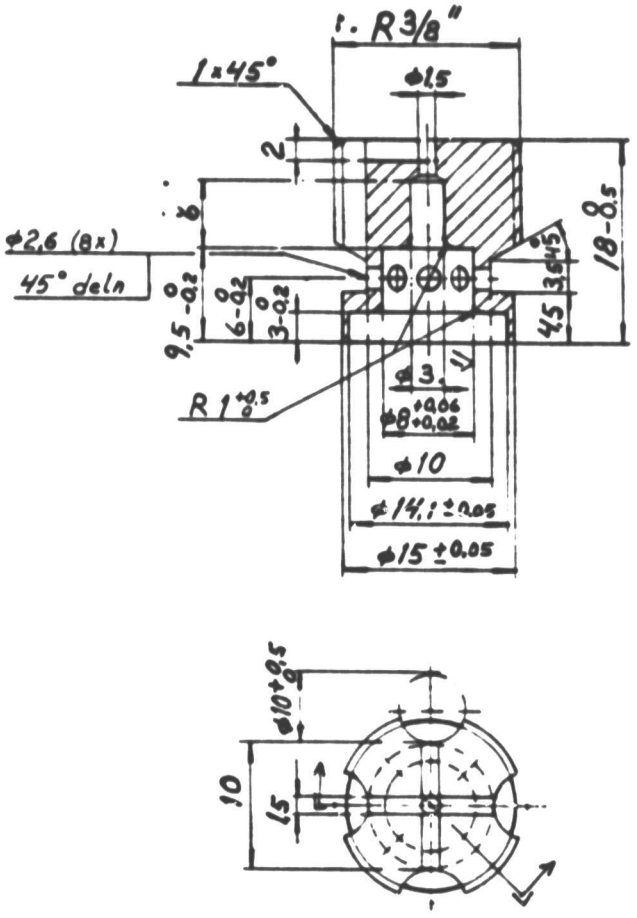


Fig. 14 Compressor Check Valve Spring Holder

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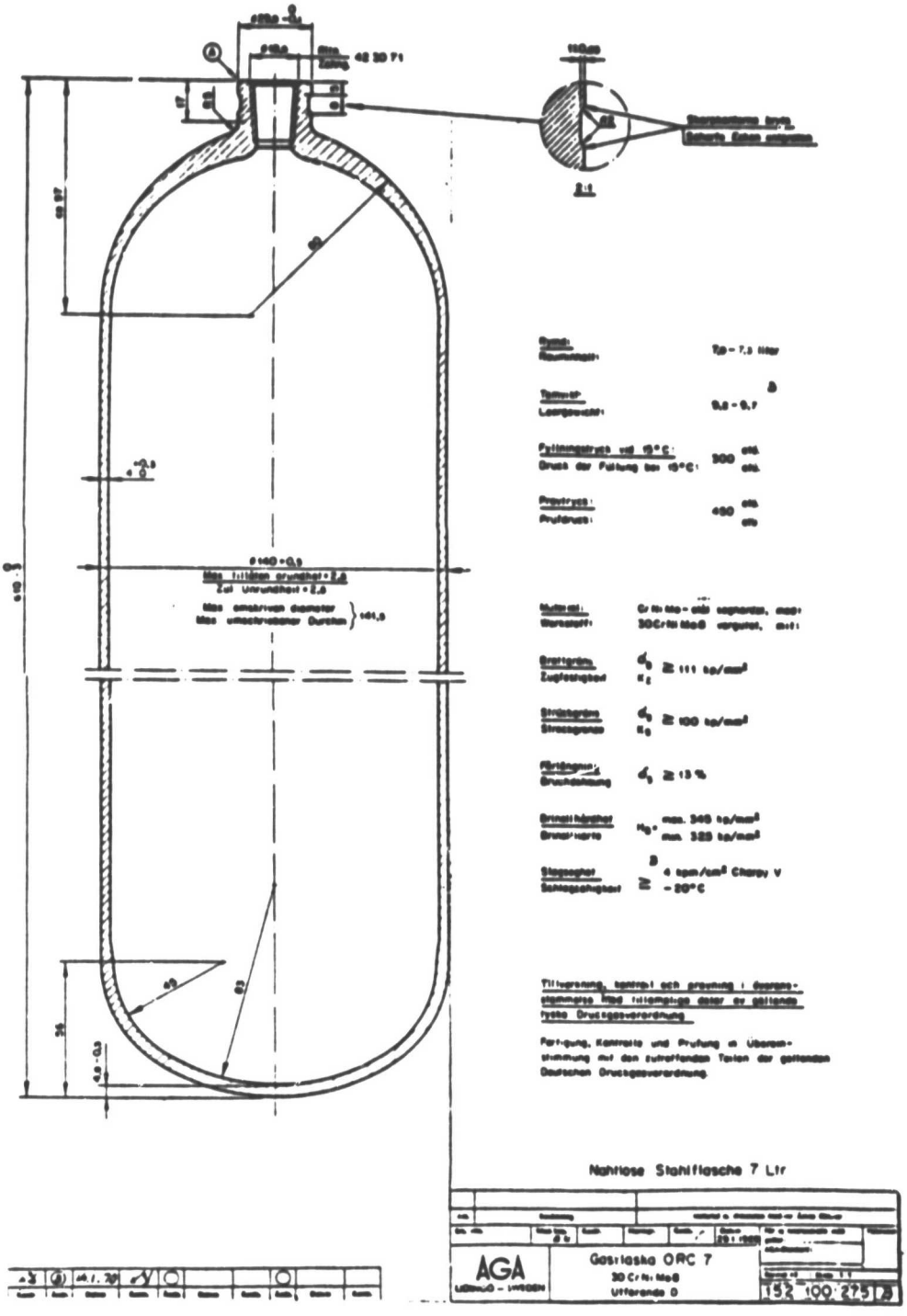
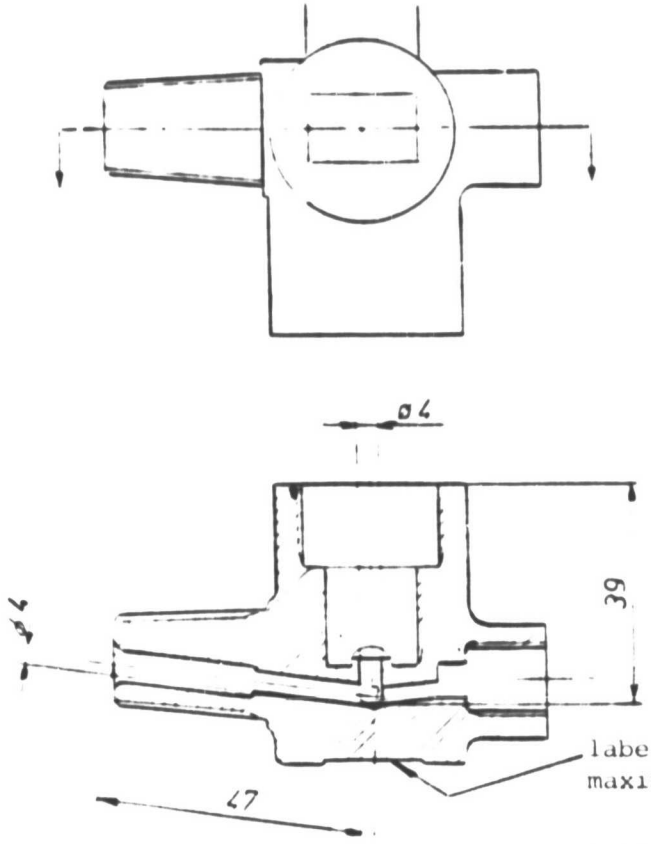


Fig. 15 Hydrogen Bottle AGA ORC

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test pressure and model approval according to maximum work pressure

labeled here with number and maximum work pressure and type

Article Number
Valve Body 331-100-366

Fig. 16 Modification of Valve Body for Hydrogen Reservoir

APPENDIX

POWER CONTROL SYSTEM

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Check Valves for Stirling Engines

Specification for ASE Mod I check valves and a short survey of tested check valves for maximum, minimum pressure and for the hydrogen compressor.

ASE MOD I

CHECK VALVE SPECIFICATION

Type:	Check valves for Stirling engines with working principle similar to air compressor exhaust valves.
Size:	Internal flow corresponding to at least 5 mm ESEOD (Equivalent Sharp Edge Orifice Diameter).
Working gas:	Hydrogen
Pressure:	Max total pressure 25.0 MPa Max diff pressure 22.5 MPa
Cracking pressure:	< 0.1 MPa
Gas temperature:	Min, max check valves < 150 °C Compressor check valves ≈ 225 °C
Leakage:	External none Inchecked direction < 0.001 g/s H_2 at 20 °C and 10.0 MPa ΔP
Working frequency:	10 - 75 Hz
Material:	Hardened material in both seat, plate and spring holder as large flows may occur in open direction.
Outer dimension:	As small as possible.
Design:	The check valve should be of a Cartridge type and be able to mount in a block. The design should be resistant to particle contamination (Teflon dust) of the working gas.
Endurance:	2000 hrs continuous operation.

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Check valves tested by United Stirling

1. HAWE ERI: This check valve was used in the early versions of our V4X engine. The valve has an inner diameter of about 4 mm. The valve was sensitive to rulon dust-oil mixture which caused the valve to leak. Later the valve has been used as a check valve in the timed supply line (V4X, P40) with good result. The valve has not been tested in the check valve test rig.

The size of valve which is too small for the short-circuiting demand of MOD I and the above mentioned experience makes this valve not suitable as a min, max or compressor check valve for MOD I.

2. BAUER 7780 K: [illegible] originally designed as a suction valve in the third stage of an air compressor. The valve has been used by USS as an min and max check valve in the V4X3 engines including the Taunus car. The valve seams to be tolerable to rulon dust and has worked well in the engines. In drive units with very hard running extensive wear in the spring holder and broken valve plates have occurred after about 400 hours. The flow diameter is about 4 mm (similar to HAWE ERI). The valve has not been tested in the check valve test rig and is due to size and wear not suitable to MOD I.

3. BAUER 7780K: The valve is the pressure valve in the same air compressor as Bauer 7780K. The flow passage is similar to 7780 but the valve is more rugged and seems to have stronger and more heat resistant spring. The design of the valve makes it difficult for us to use it as check valve in our engines. The valve has been tested in P40-2. It shows good results and no wear has been detected. The valves have not been tested in the check valve test rig.

Their size make them not suitable for MOD I.

4. LEECHEK 375: Designed as a check valve for hydraulic and fuel system in air crafts. The check valve was the tightest ever tested in our static test rig and the difference between the best and the worst valve was small. The valve has been tested in the P75 4-189 engine. Flow rate equivalent to an appr 7 mm inner diameter hole. The check valve was extremely sensitive to rulon dust and after about 10 hours of engine testing the valve stopped in open position. Due to its size and behavior not suitable for Mod I.

5. LEECHEK 250: A smaller model of LEECHEK 375. The valve has been tested on a V4X3 engine with the same results as with LEECHEK 350.

* Allmänblankett

6. HAWE ER2:

This check valve has been tested together with the P75 4-189 engine including the Volvo van. The P40 and P75 hydrogen compressor is also equipped with ER2. The valve has a flow diameter of 6 mm. The valve has not been tested in the check valve test rig.

In the Volvo truck installation the valve has operated with good results (200 hours).

The check valves placed in the P75 compressors show significant wear in the spring holder after about 200 hours of operating. The gas temperature also was too high for the spring material with broken and damaged springs as a result. A new model with temperature resistance springs has been ordered but not yet tested.

The size and outer dimensions make this valve suitable for MOD I but the wear on the spring holder and the spring problem must be solved before we can choose this valve for MOD I.

7. HAWE RC2:

This valve has about the same flow rate as ER2 and was chosen for the Opel installation in favour of ER2 because its better mounting facilities. RC3 has the same plate and spring as the ER2 but the spring holder was of another design. After about 50 hours of running in the Opel car the spring holder was broken. Later the check valve was tested in the check valve test rig and the same fault occurred after 68 hours. The valve could be suitable for MOD I only after major redesigning.

8. HAWE RC3:

A bigger model of RC2. The valve is used on the 4-275 engine. Flow capacity is larger than RC2. The valve has been tested 600 hours in the check valve test rig with good result and no wear could be seen. The valve could be a choice for MOD I but the size makes other valves better suited as MOD I check valves.

9. BAUER 9312:

The valve looks like Bauer 7780K but with a ø 5 mm entrance hole. The valve has been tested in the test rig (not under MTI contract) 600 hours with good result. The design makes the valve difficult to mount but the concept looks promising for further development and specially built check valves (described later) origin from this valve.

10. BAUER 9313:

This valve is similar to Bauer 7790K but with a 5 mm entrance hole. 9313 and 9312 were originally used as check valves in the second stage on air compressors. The plate of this valve was not strong enough and after about 100 hours in a P40 drive the plate was broken.

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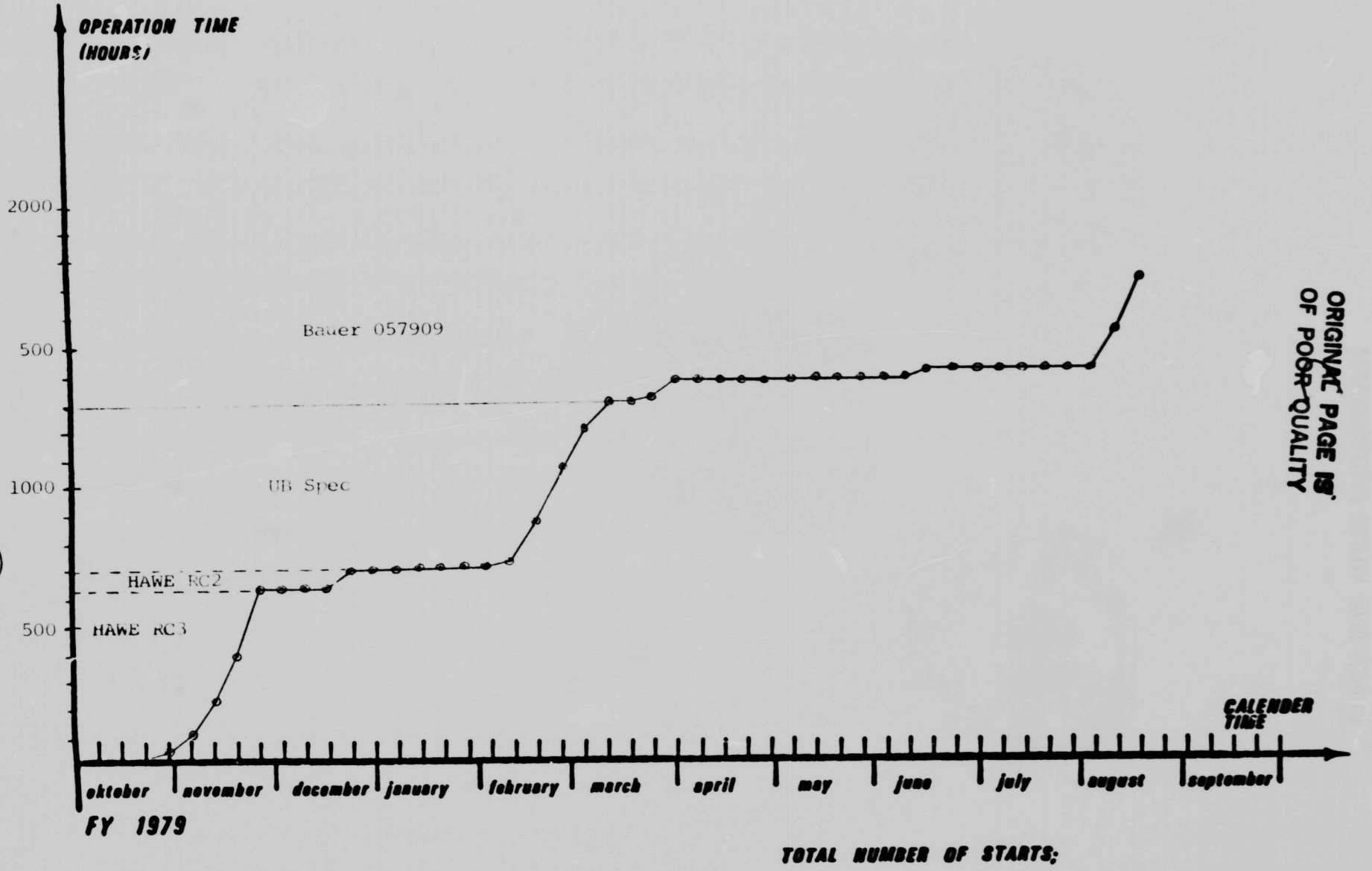
11. UB Special: This valve is a USS modified HAWE ER2 where we use the seat and plate from ER2 and built our own spring holder (hardened) and use a temperature resistant spring.

The valve has been tested in the check valve test rig 600 hours with good results. No significant wear could be seen. In our compressor test stand we have achieved 2 000 hours of proper operation. This valve model is also mounted on our last P40 engines.
This valve is today our choice for MOD I.

12. BAUER KB057909: This valve is a development of Bauer 9312 with its plate, seat and spring but a housing similar to 9313. The valve is now being tested in our test rig and has up till now logged about 300 hours. Nothing can be said about wear but the spring holder is not hardened. The valve has an internal restriction and is probably too small for the MOD I engine.

13. BAUER KB058259: This valve is specially made for us and is a bigger version of KB057909 and with hardened spring holder. We are waiting for the first samples and this valve will be the last valve to be tested in this selection test at the check valve test r
The concept looks promising and the final choice of MOD I check valves will probably be between UB Special and Bauer KB058259.

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Fig. 1 Accumulated Operation Time for Engine/Rig: Check Valve Test Rig (S-H Almstrom)

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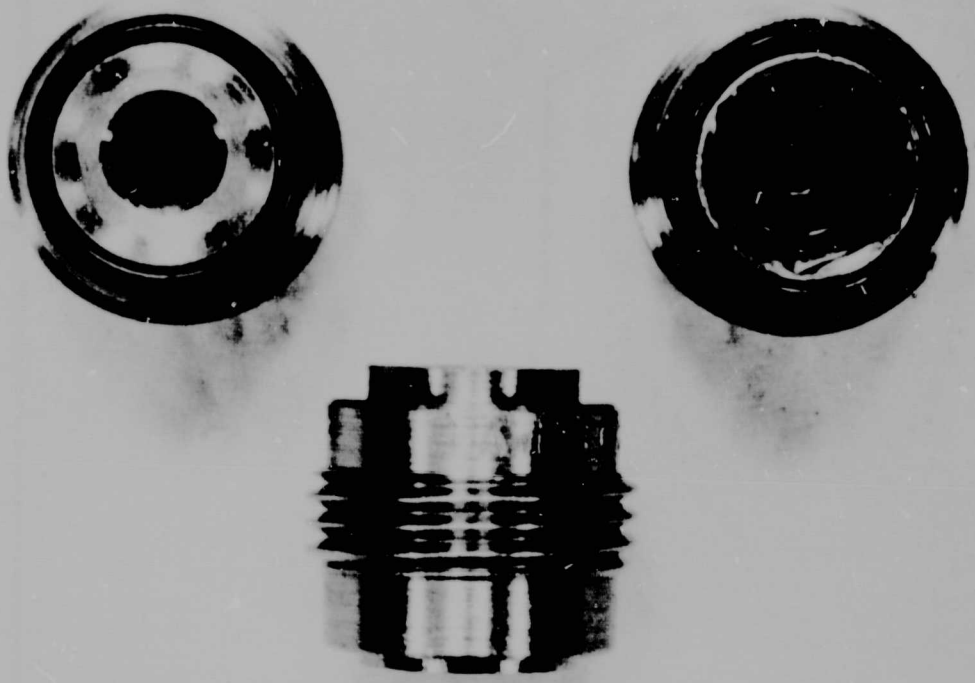
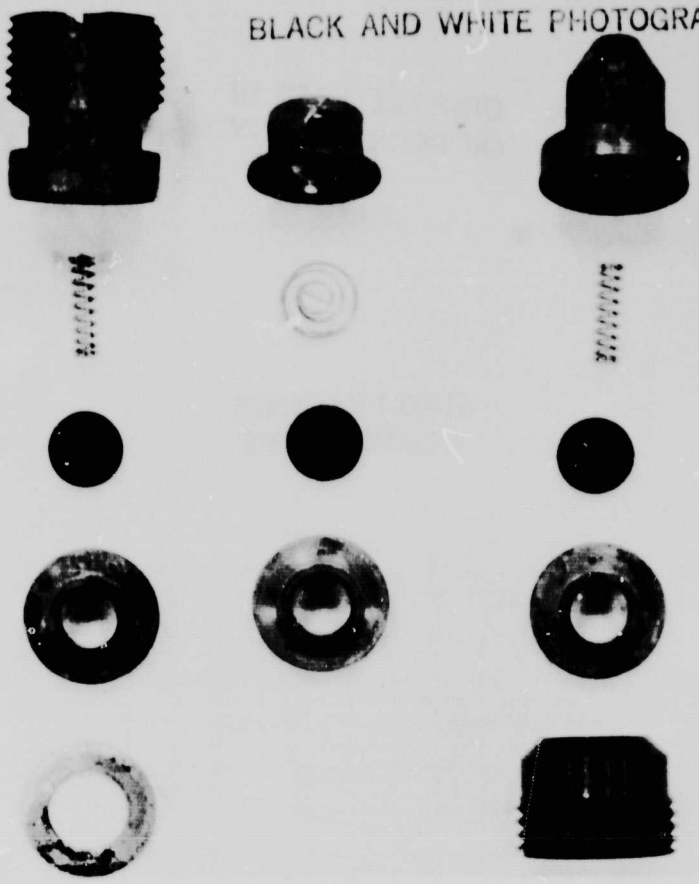


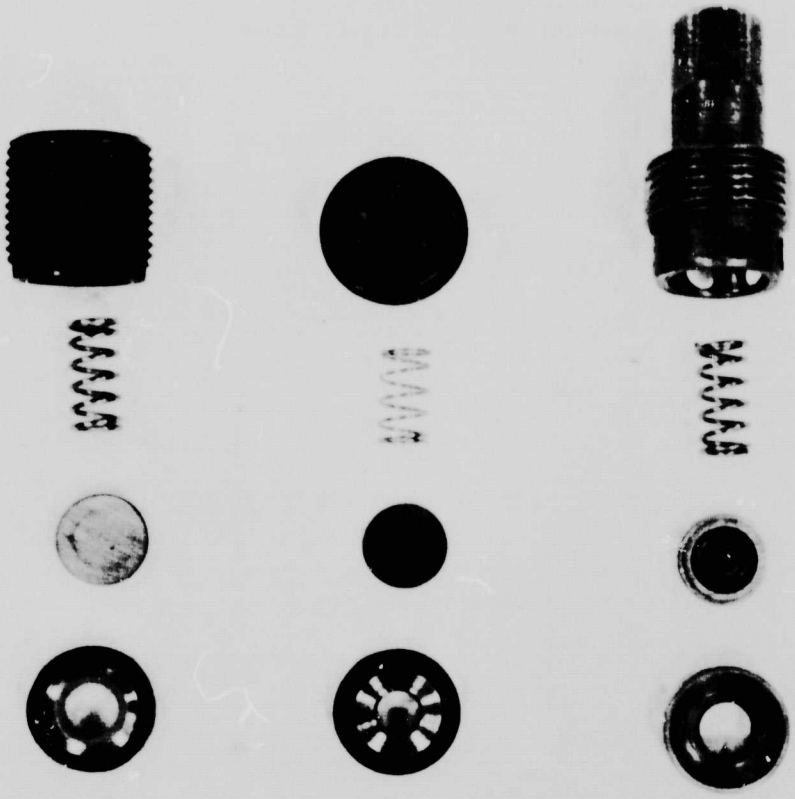
Fig. 2 HAWE Check Valve

Neg. K278

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UB SPEC HAWE ER2 UB SPEC



Bauer 057909 Bauer 7790K Bauer 9312

Neg. K279

Fig. 3 Alternative Check Valves

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250 LEECHEK
FLOW CURVE

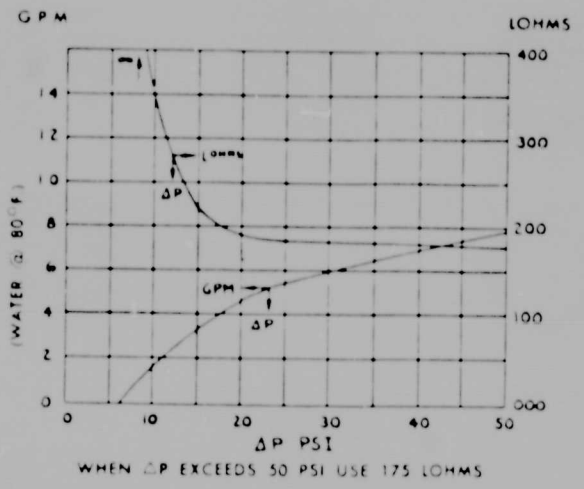


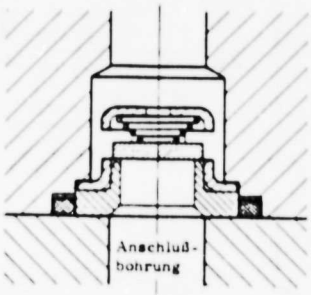
Fig. 4 250 Leechek Check Valve

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Einsteck - Rückschlagventile Typ ER

1. Allgemeines

Rückschlagventile geben den Durchflußstrom in einer Richtung frei und sperren ihn in Gegenrichtung. Sie gehören deshalb zur Gruppe der Sperrventile (DIN 24 300).



2. Beschreibung

Der Typ ER ist ein federbelastetes Einsteck-Rückschlagventil und wird vorzugsweise in Hydrogeräte eingebaut, die für Montage auf Grundplatten geeignet sind. Das Ventil wird in eine abgesetzte Bohrung eingeführt und mit einem Bund zwischen Grundplatte und Bohrungsabsatz eingeklemmt, so daß es unverrückbar festsitzt. Die Abdichtung an der Trennfuge übernimmt ein O-Ring.

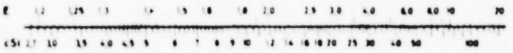
Der Typ ER ist ein Plattenventil. Ventilsitz und -platte sind gehärtet und geschliffen, der Käfig gehärtet. In gesperrter Durchflußrichtung ist es hermetisch dicht.

3. Kenngrößen

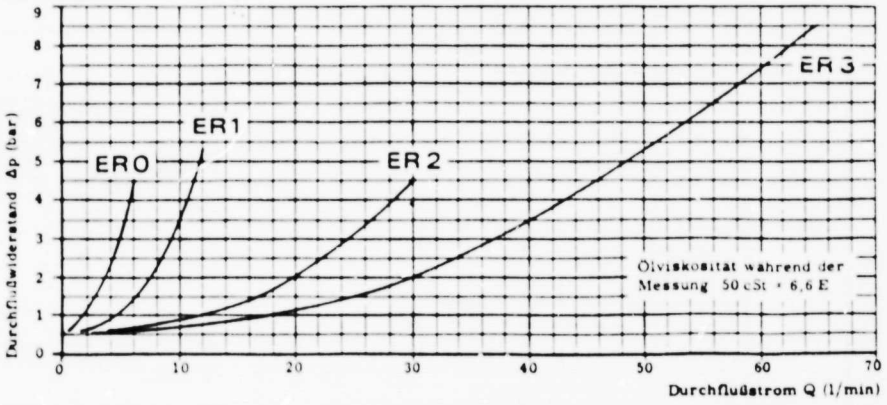
3.1 Allgemein

Sinnbild und Typenbezeichnung	 ER 0, ER 1, ER 2, ER 3
Bauart	Einsteck-Rückschlagventil
Geräteabmessungen	siehe Pos. 4
Einbaulage	beliebig

3.2 Hydraulische Kenngrößen

Betriebsdruck	max. 500 bar
Druckmittel DIN 51 524	Hydrauliköl HL (P) 9, 16, 25, andere Druckmittel auf Rückfrage
zul. Viskositätsbereich	4 ... 150 cSt
Umrechnungsleiter von cSt in E	
Temperaturen	Umgebung: -35 ... +70°C Druckmittel: -40 ... +80°C
zul. Durchfluß	ER 0 = 6 l/min; ER 1 = 12 l/min; ER 2 = 30 l/min; ER 3 = 65 l/min

Δp-Q-Kennlinien

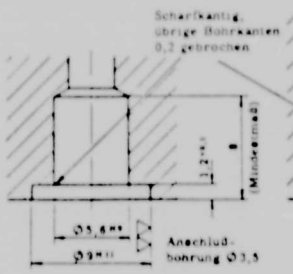


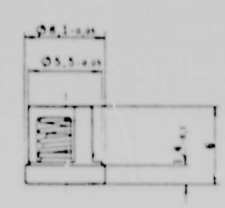
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4. Maßbilder und Aufnahmebohrungen

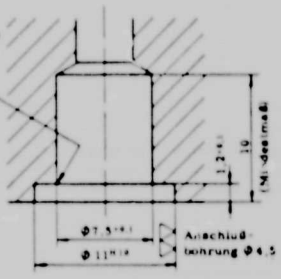
Typ ER 0

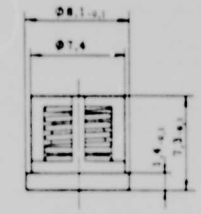




O-Ring 5 x 1,5

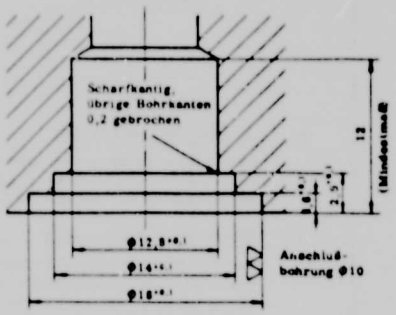
Typ ER 1

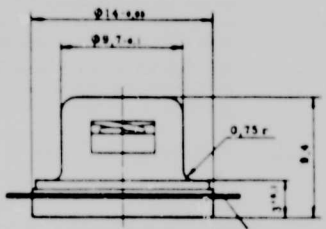




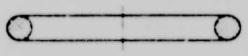
O-Ring 5 x 1,5

Typ ER 2





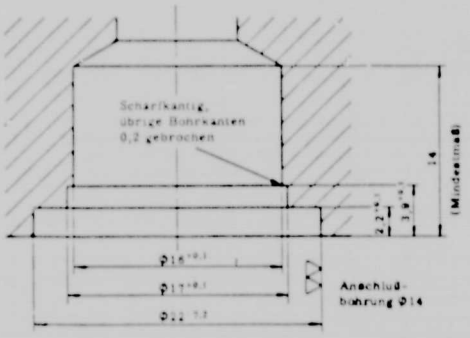
Scheibe
(HAWE-Nr. 4830 006)
Ø18 e9 x Ø14 +0,05 +0,1 x 0,2

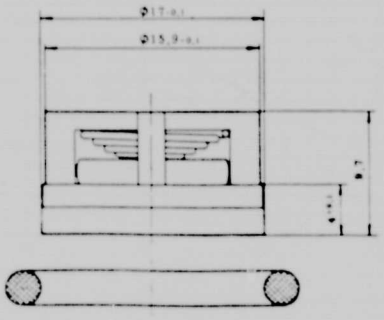


O-Ring 14 x 2

Achtung: O-Ringe sind getrennt zu bestellen!

Typ ER 3

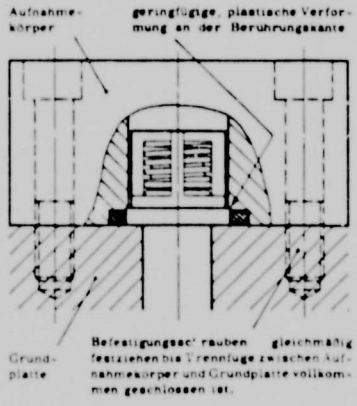




O-Ring 17,12 x 2,62

Einbauhinweise:

Die exakte Fixierung des Rückschlagventiles im Aufnahmekörper erfolgt beim Anziehen der Befestigungsschrauben durch eine bewußt herbeigeführte, geringfügige plastische Verformung an der markierten Berührungskante. Diese Montageforderung setzt ein fließfähiges Material des Aufnahmekörpers voraus. Es können alle im Hydro-Ventilbau üblichen Werkstoffe außer gehärteten oder naturharten Materialien verwendet werden.



Maßbilder sind maßstäblich, alle Maße in mm. Änderungen vorbehalten.

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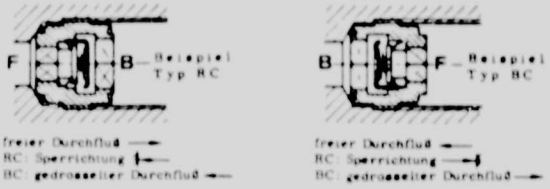
Rückschlagventile Typ RC und BC

zum Einschrauben in Gewindebohrungen

1. Allgemeines

Die Ventile ermöglichen freien Durchfluß in einer Richtung und sperren den Durchfluß in Gegenrichtung (Typ RC) oder drosseln diesen gemäß der Wirkungsweise einer Lochblende (Typ BC). Die Ventilgehäuse sind so ausgebildet, daß sie in normale, im Kernloch abgesetzte Gewindebohrungen sowohl in der einen als auch in der anderen Wirkungsrichtung eingeschraubt werden können.

Schematische Darstellung



freier Durchfluß —→
RC: Sperrichtung —|
BC: gedrosselter Durchfluß —→

freier Durchfluß —→
RC: Sperrichtung —|
BC: gedrosselter Durchfluß —→

2. Kenngrößen nach VDI 3268 / 3269

Benennung und Bauart	Sinnbild	Blende Ø mm	Typenbezeichnung (Bestellbezeichnung) Einschraubelement: große Schlüsselweite Anschluß B, Baugröße 1	Einschraubelement: kleine Schlüsselweite Anschluß F, Baugröße 2	Einschraubelement: kleine Schlüsselweite Anschluß F, Baugröße 3
Rückschlagv. mit federbelasteter Ventilplatte	(F) (B)	0	RC 1	RC 2	RC 3
Blenden-rückschlagventil mit federbelasteter, als Lochblende ausgebildeter Ventilplatte	F B	0,4	BC 1-0,4	BC 2-0,4	BC 3-0,4
		0,6	BC 1-0,6	BC 2-0,6	BC 3-0,6
		0,8	BC 1-0,8	BC 2-0,8	BC 3-0,8
		1,0	BC 1-1,0	BC 2-1,0	BC 3-1,0
		1,2		BC 2-1,2	BC 3-1,2
		1,5		BC 2-1,5	BC 3-1,5
		1,8			BC 3-1,8
		2,0			BC 3-2,0

Gehäuseausführung

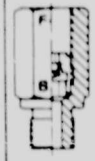
beidseitiger Rohrleitungseinbau	auf einer Seite Einschraubzapfen Form B DIN 3852 Blatt 2	
Ausf. G	Ausf. E	Ausf. F



RC... G und BC... G



RC... E und BC... E



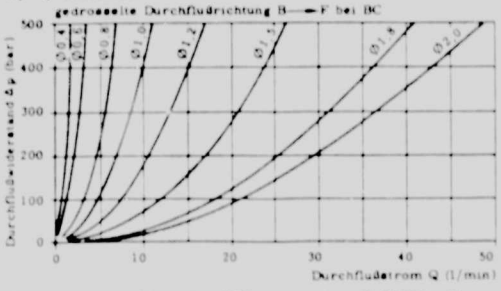
RC... F und BC... F

beide Ausführungen unterscheiden sich nur durch die Einschraubrichtung des Rückschlagventiles

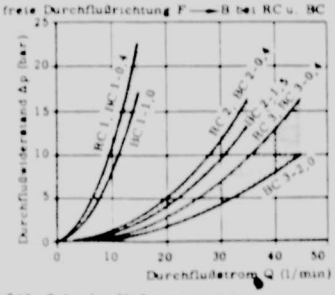
zul. Betriebsdruck (bar)	700		500	entsprechend der Baugröße des Einschraubelementes
zul. Durchflußstrom (l/min)	15	35	45	
Öffnungsdruck F→B (bar)	0,1	0,1	0,15	
Gewindegröße	R 1/4"	R 3/8"	R 1/2"	
Einbaulage	beliebig			
Druckflüssigkeit	Hydrauliköl 9 ... 68 cSt/50°C			
Viskositätsbereich	4 ... 1500 cSt (1 cSt = 1 · 10^{-6} m^2/s)			
Temperaturen	Umgebung und Öl: -40 ... +80°C, auf Viskositätsbereich achten			

Δp-Q-Kennlinien

gedrosselte Durchflußrichtung B→F bei BC



freie Durchflußrichtung F→B bei RC u. BC



Die Messungen erfolgten bei 20°C, verwendetes Öl MOBIL - Flowrex E 7,62 cSt bei bei Meßtemperaturen

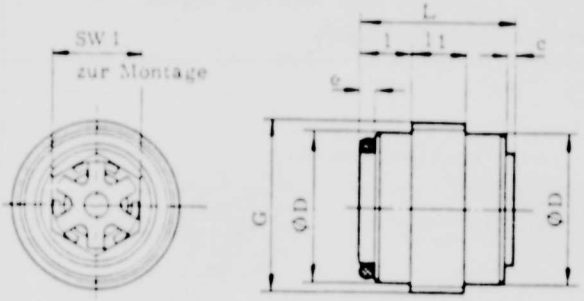
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Juni 1976

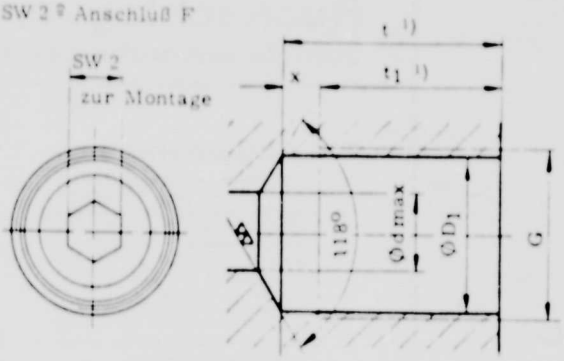
3. Geräteabmessungen (Alle Maße in mm, Änderungen vorbehalten)

3.1. Einschraubelement

Große Schlüsselweite
SW 1 ≙ Anschluß B



Kleine Schlüsselweite
SW 2 ≙ Anschluß F



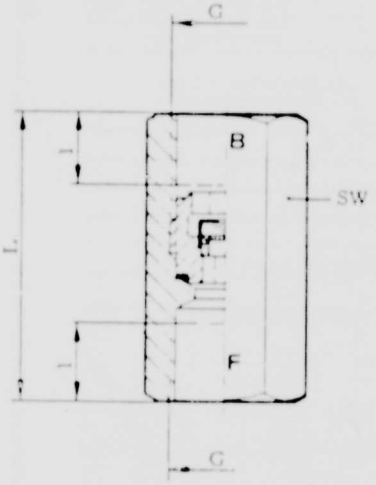
Achtung: Steckschlüssel nicht mit Gewalt einführen, damit der Ventilkäfig nicht beschädigt wird.

Typ	L	l	l1	c	e	D	D1	d	t	t1	x	G	SW	SW 1	O-Ring Abmessungen	Gewicht (g)
RC 1, BC 1	13	4,3	4,7	0,5	1	11,6	11,8	8	25,5	22,5	3	R 1/4"	8	4	9 x 1	6
RC 2, BC 2	15	5	5,3	0,7	1,6	14,8	15,25	9	27	24	3	R 3/8"	9	5	10 x 1,5	13
RC 3, BC 3	18	6,2	6,1	0,8	1,6	18,5	19	12	32,5	28,5	3,5	R 1/2"	12	8	14 x 1,5	24

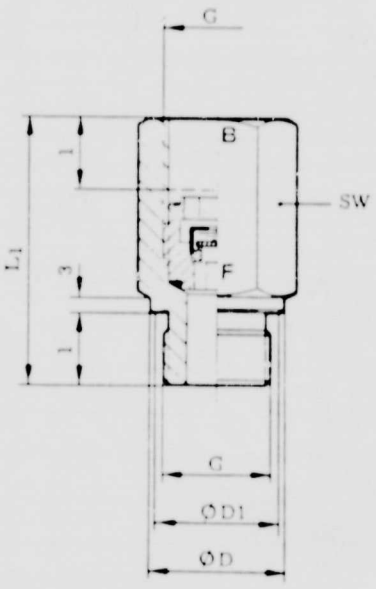
1) Maß t und t_1 sind Mindestwerte. Gewindeauslauf x ist, um eine ordnungsgemäße Montage gewährleisten zu können, unbedingt einzuhalten. Er darf kleiner, aber nicht größer sein.

3.2. Gehäuseausführung

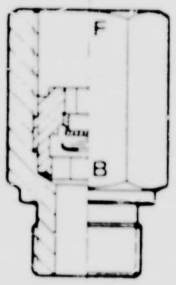
Gehäuseform G



Gehäuseform E



Gehäuseform F

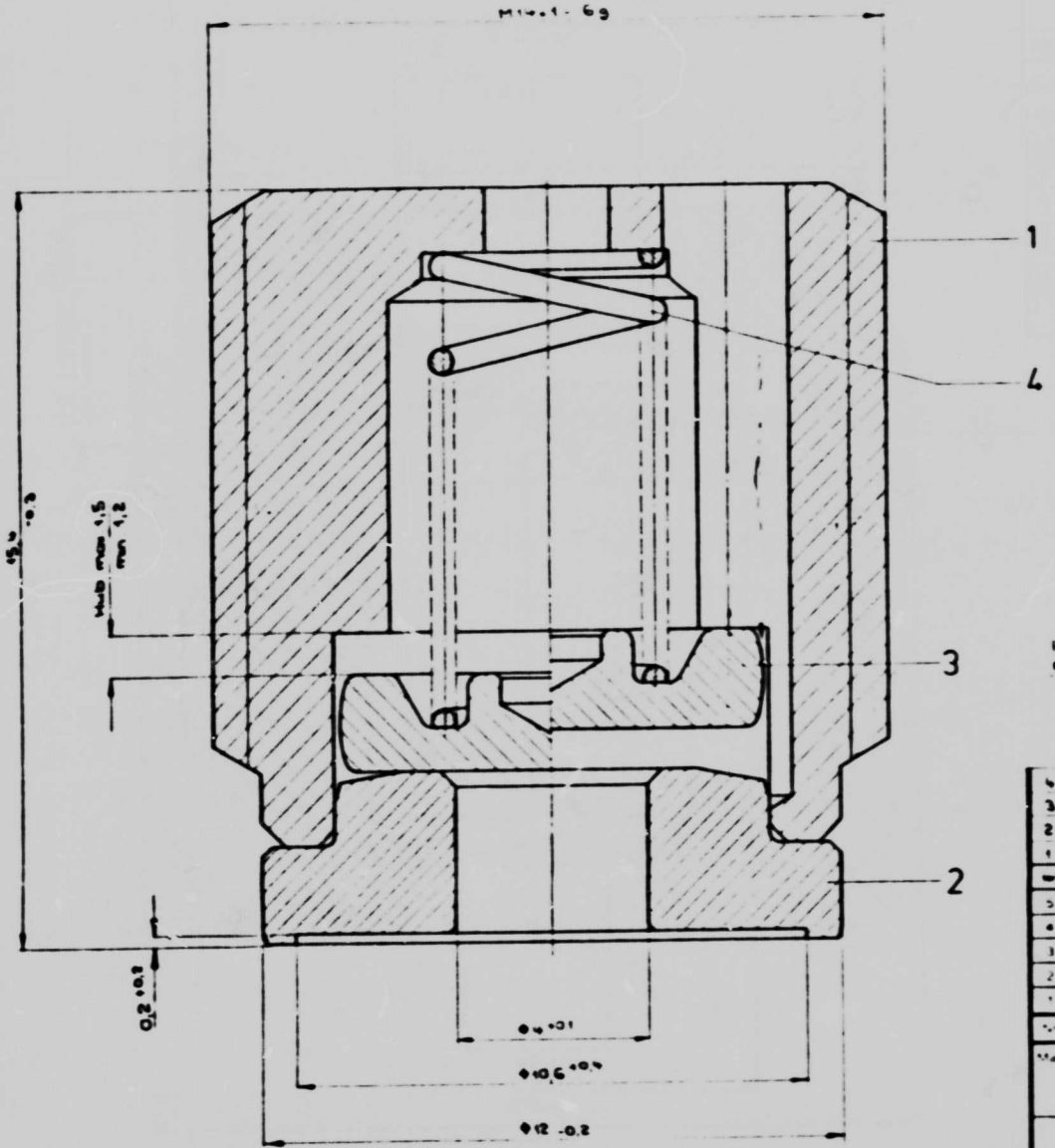


Abmessungen wie Form E

Typ	G	L	L1	l	D	D1	SW	Gewicht komplett (g) Gehäuseform G	Gehäuseform E u. F
R(B)C 1-...	R 1/4"	46	42	12	19	16	19	75	60
R(B)C 2-...	R 3/8"	50	44	12	22	20,5	22	105	85
R(B)C 3-...	R 1/2"	56	52	14	26	24	27	170	145

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BAUER Kompressoren
M28c308

IB-M 31. 10. 78

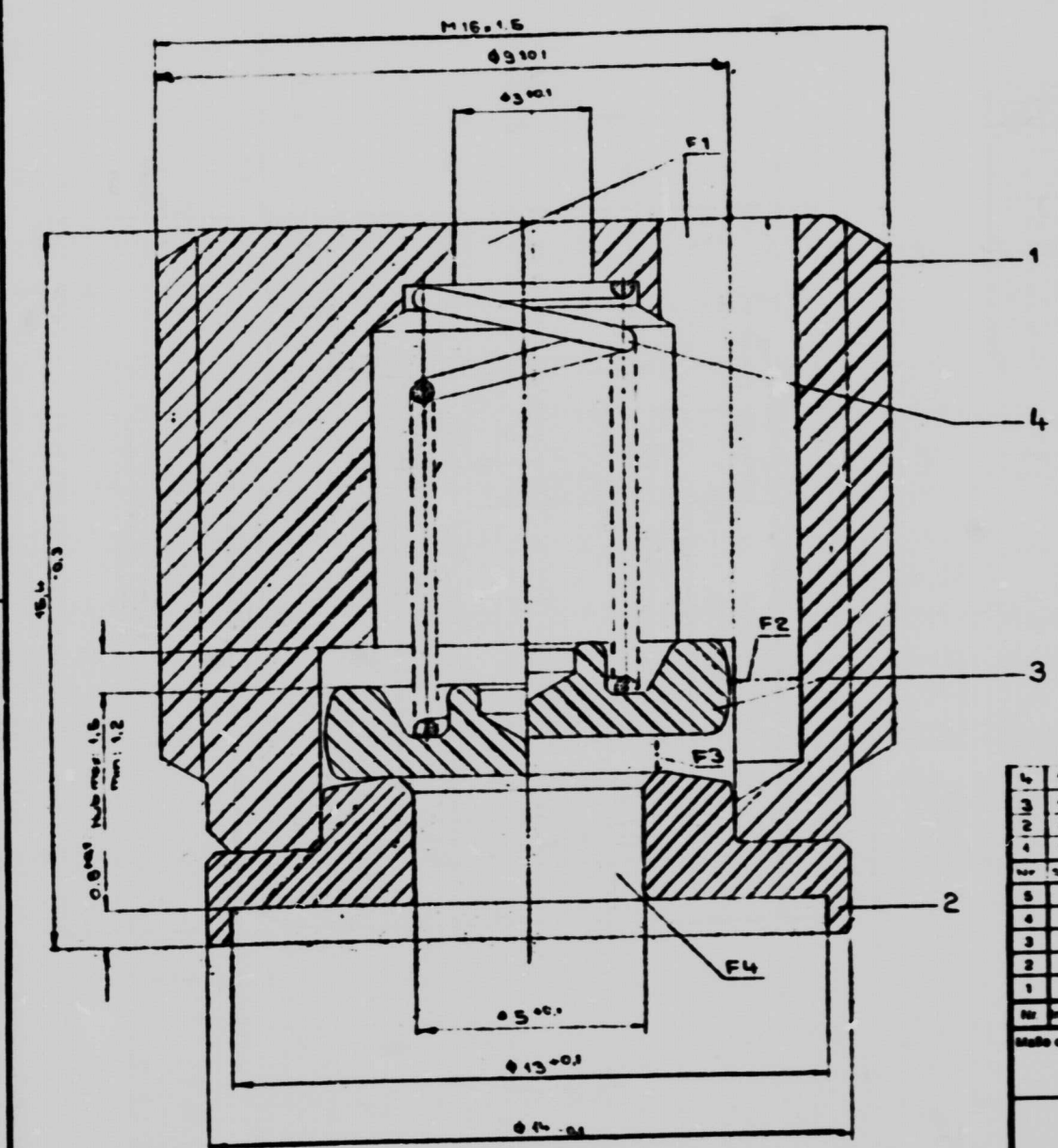
Pos.	Stck.	Benennung	Sach-Nummer	Bemerkung
4	1	Ventilfeder	1878 b 6-0	
3	1	Ventilplatte	12454 080	
2	1	Ventilsitz	12454 080	
1	1	Saugventilverschraubung	57907 080	

Pos.	Stck.	Änderung	Index der Änderung	Datum
5				
4				
3				
2				
1				

Maße ohne Toleranzen gelten	Werkstoff	Type für Saug-Hoch	Maßstab 10 : 1	30.10.78

BAUER KOMPRESSOREN 8 München 71	Saugventil : 2 - 1,5 mm Hub	KB-057909-080
	Ersatz für	Ersatz durch

eingebaut in	Herzu	Heraus	aus	057908-080 ähnlich



Querschnitte

F1 = 49.4 mm² (7.03)

F2 = 268 mm²

F3 = 263 mm² (Hub max. 1.5)
211 mm² (max. 1.2)

F4 = 19.6 mm² (5)

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TB-M02.5.79

Pos.	Stück	Benennung	Sachnummer	Bemerkung
4	1	Ventilfeder	0786.040	
3	1	Ventilplatte	42550.080	
2	1	Ventilsitz	-080	
1	1	Saugventilgehäuse	-080	

Nr.	Index	Art der Änderung	Grund der Änderung	Datum	Name
5					
4					
3					
2					
1					

Maße ohne Toleranzangabe: —	Werkstoff: —	Type: für Stating-Motor	Maßstab: 10:1	30-78
BAUER KOMPRESSOREN, 8 München 70	Saugventil 1.2 - 1.5 mm Hub			KB-056259-080
	Ersatz für:		Ersetzt durch:	

057909-080
057808-080

eingebaut in | Hierzu | Hierzu | Ers.

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8.5 ELECTRONICS

The electronics of the MOD I engine is built up around the Texas 9900-family microcomputer system. For the prototype system a TMS 9900-microprocessor is used.

8.5.1 Engine Control Software

The software for engine control is a modular package written in assembly language for the 9900. Routines are called upon as needed when the processor is interrupted. The interruption interval is 10 milliseconds, but most routines will execute when several interruptions have occurred.

In the prototype version a communication module has been included which will allow data and constants to move between the engine unit and the monitor.

8.5.2 Engine Control Hardware

8.5.2.1 Processor Unit

CPU 16 (bit) TMS 9900-3 Mc clock frequency.

Memory: Static RAM, EPROM.

Other circuits: UART/USRT, 2 timers with 8 interrupt lines.

8.5.2.2 Input System

Analogue inputs: CMOS multiplexer with input protection, filtering differential low level inputs (0-100 mV) and single ended high level inputs (0-10 V).

A/D converter: Successive approximation type. The converting time is 30 s.

Digital inputs: TTL data multiplexer.

RPM pulse input: Binary counter with Schmitt trigger.

8.5.2.3 Output System

Analogue output: 12 bit D/A converter with buffered sample and hold demultiplexer.

Digital outputs: TTL latches and relays.

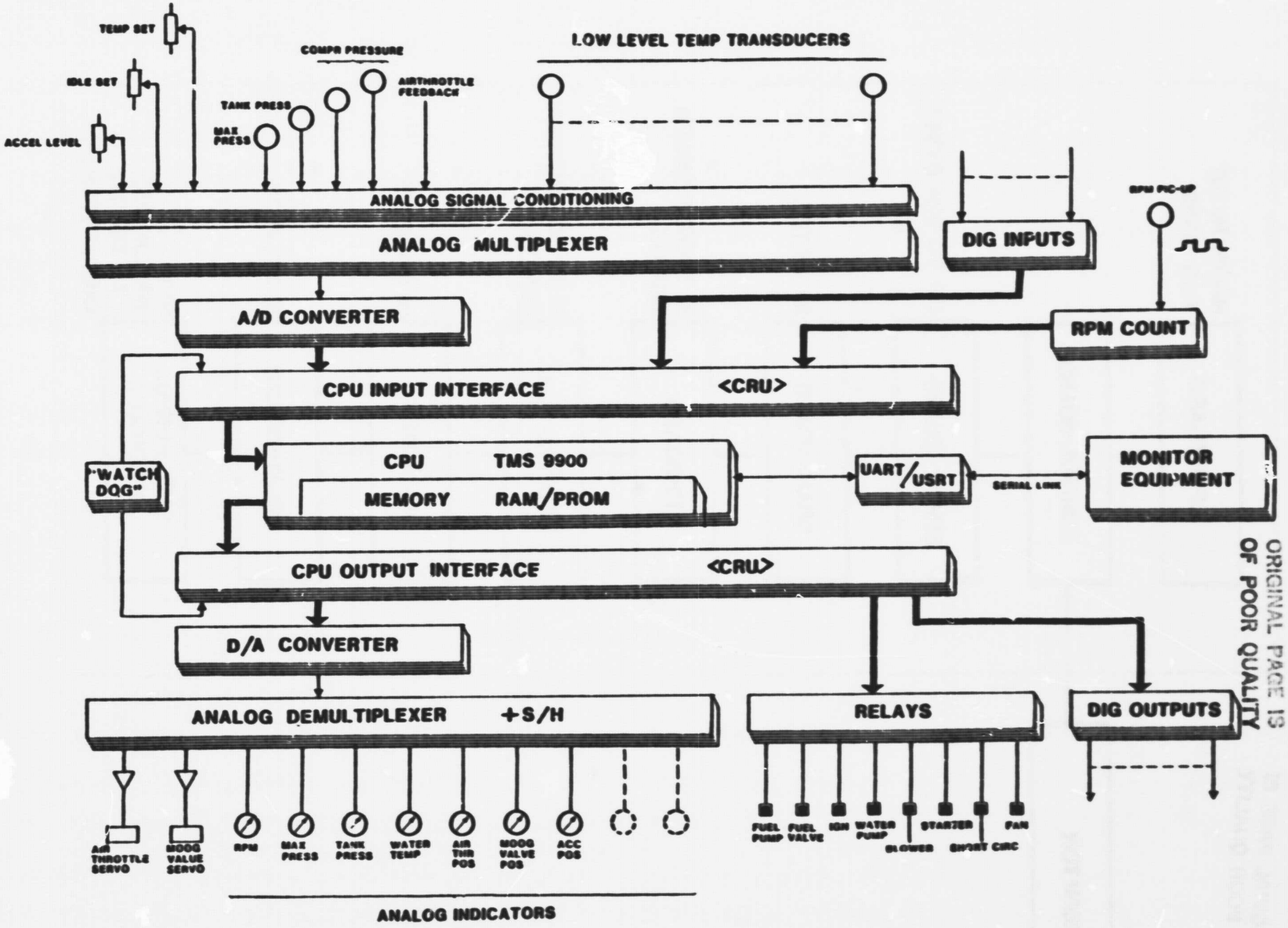


Fig. 8.5:1 Engine Electronics

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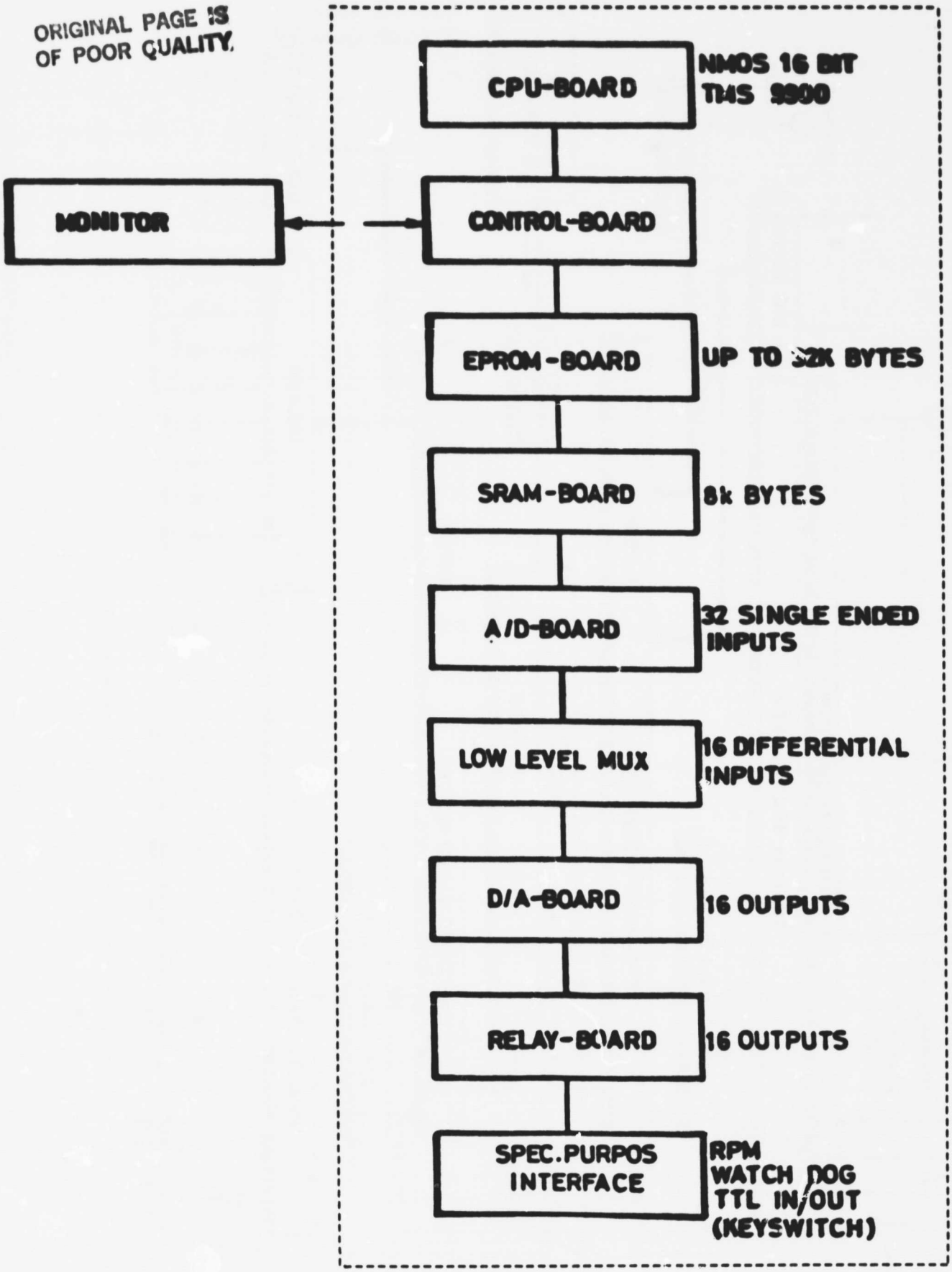


Fig. 8.5:2.1 Engine Electronics (Block Diagram)

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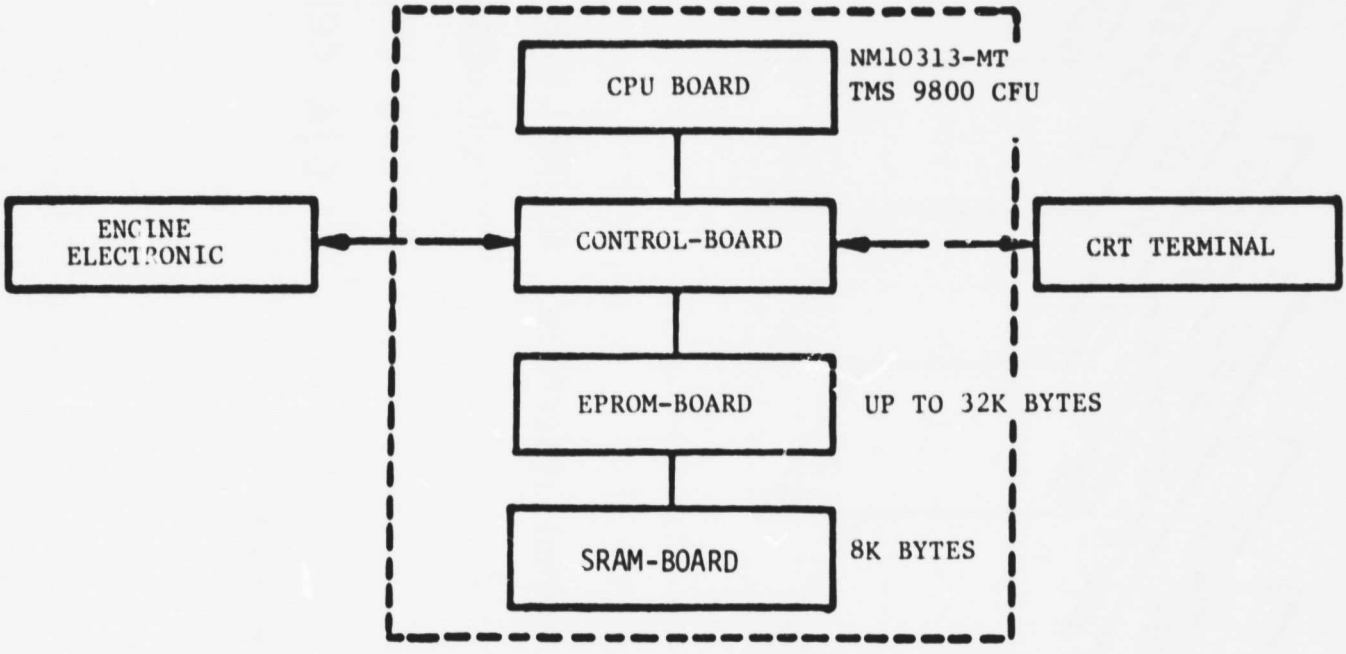
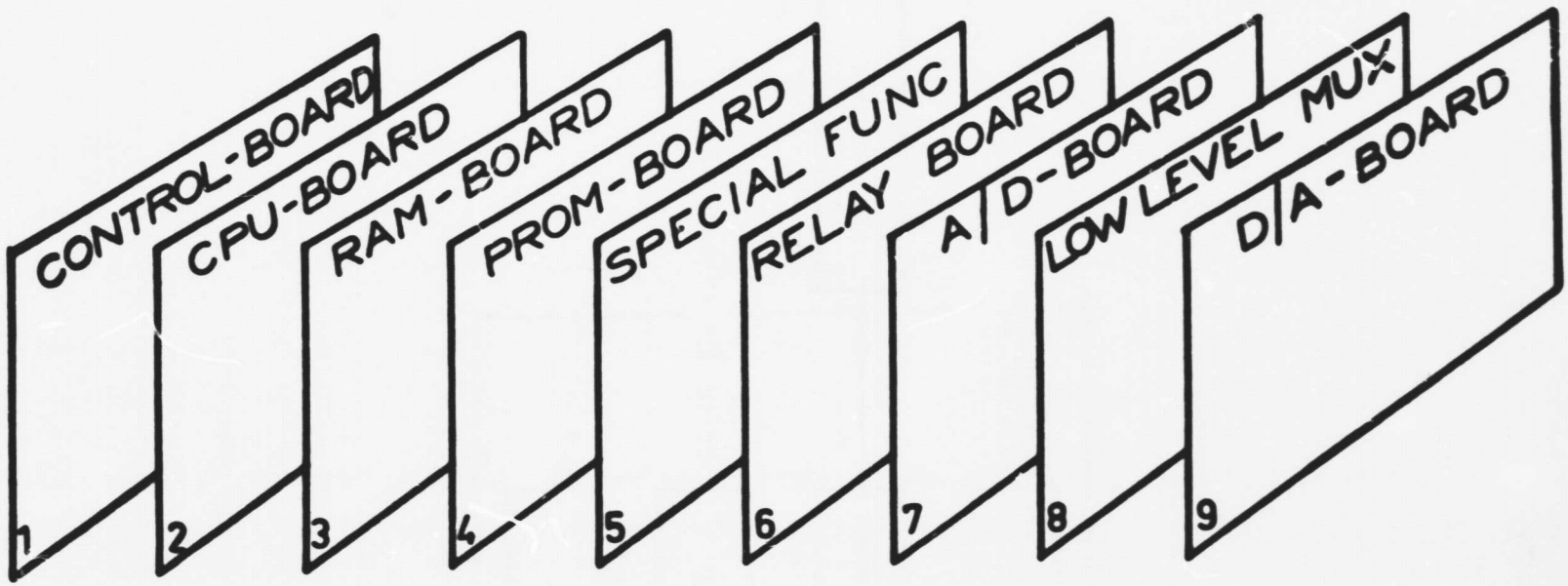


Fig. 8.5:2.2 Engine Electronics (Monitor)



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1: Memory and I/O select, Serial interface

2: CPU

3: RAM

4: PROM

5: DIGITAL I/O, WATCH DOG ETC.

6: RELAYS

7: A/D-CONVERTER + H16H LEVEL MUX

8: LOW LEVEL MUX + FILTERING

9: D/A CONVERTER, DEMULTIPLEXER

Fig. 8.5:3 Prototype Board Configuration

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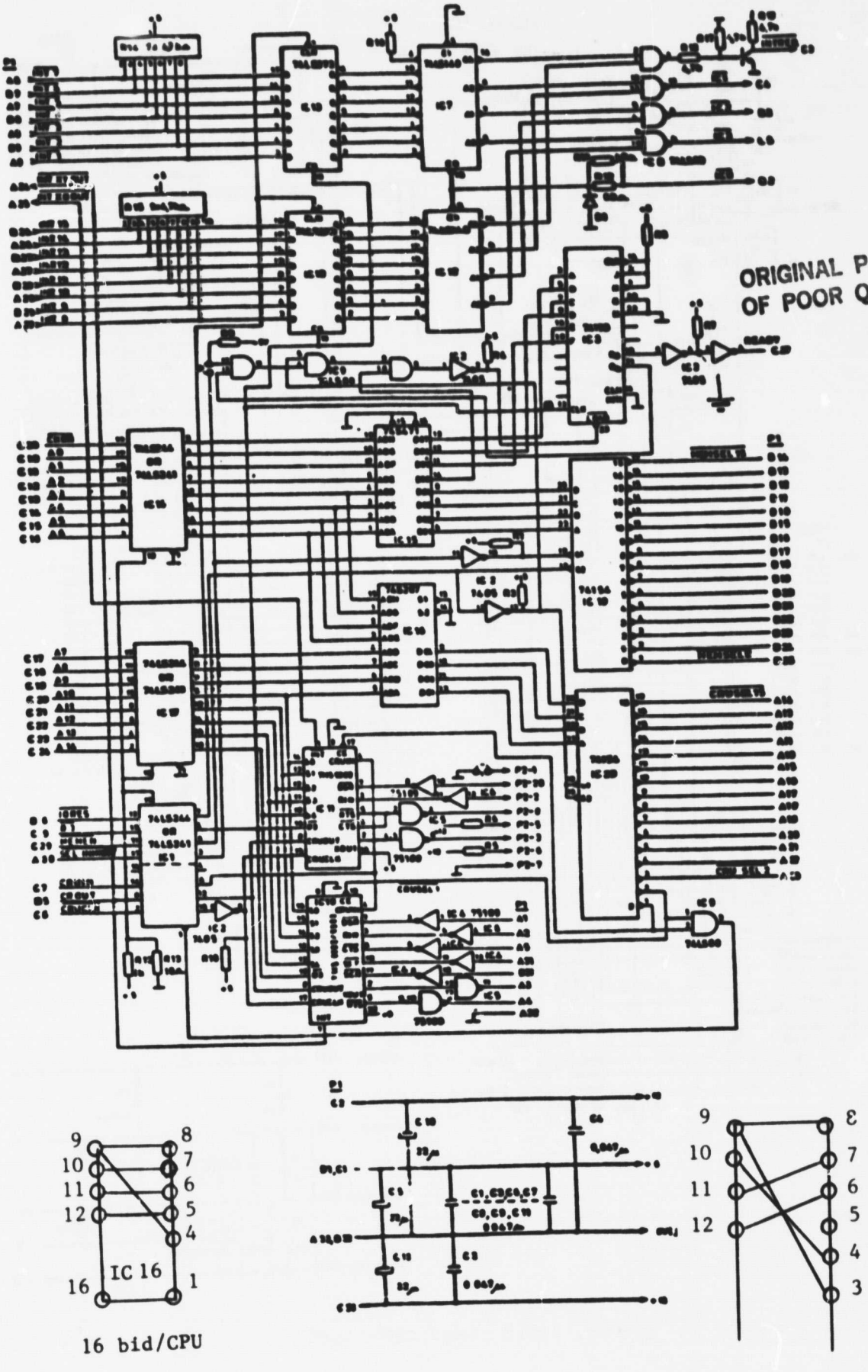


Fig. 8.5:4 S-99 Control Board

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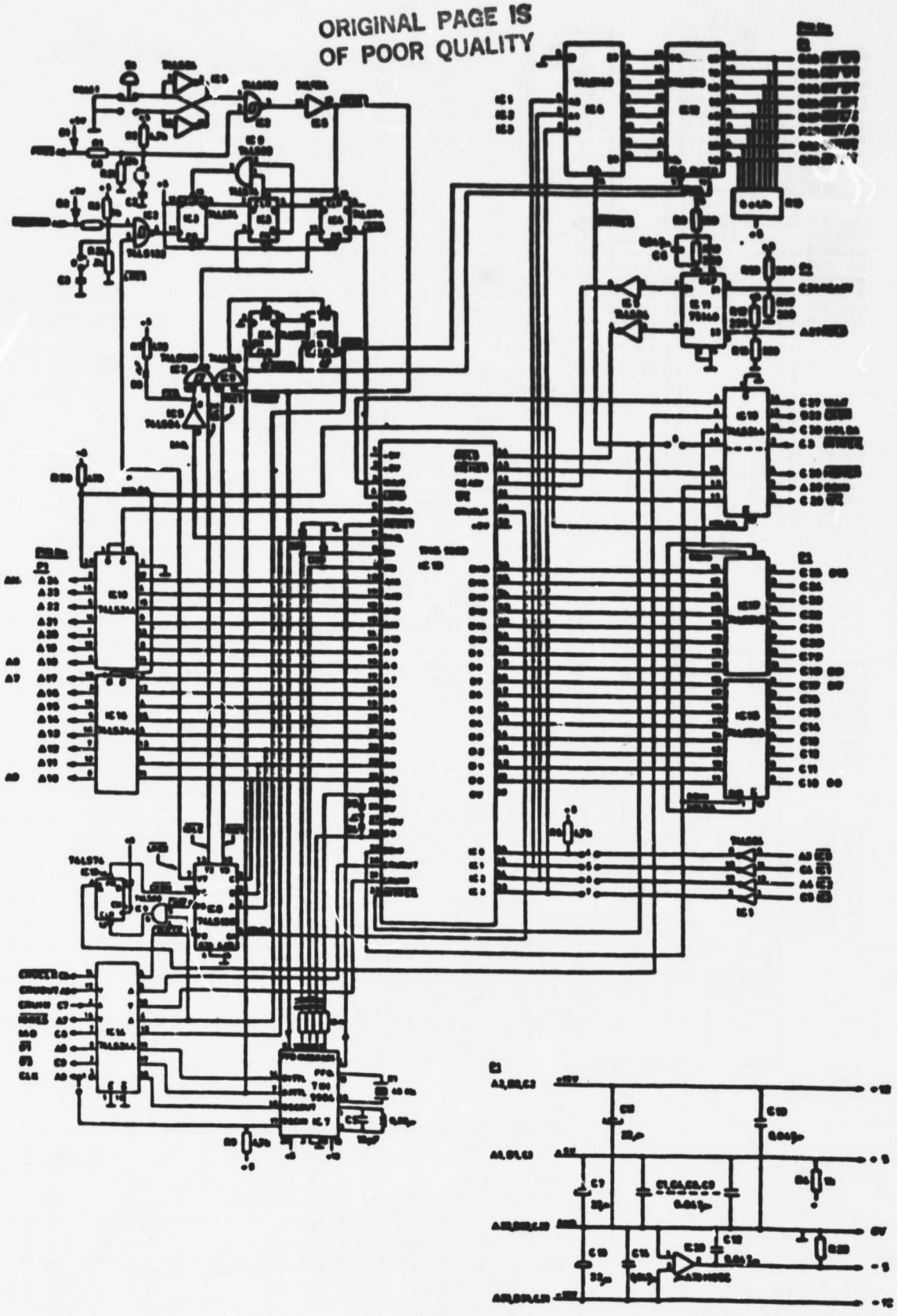
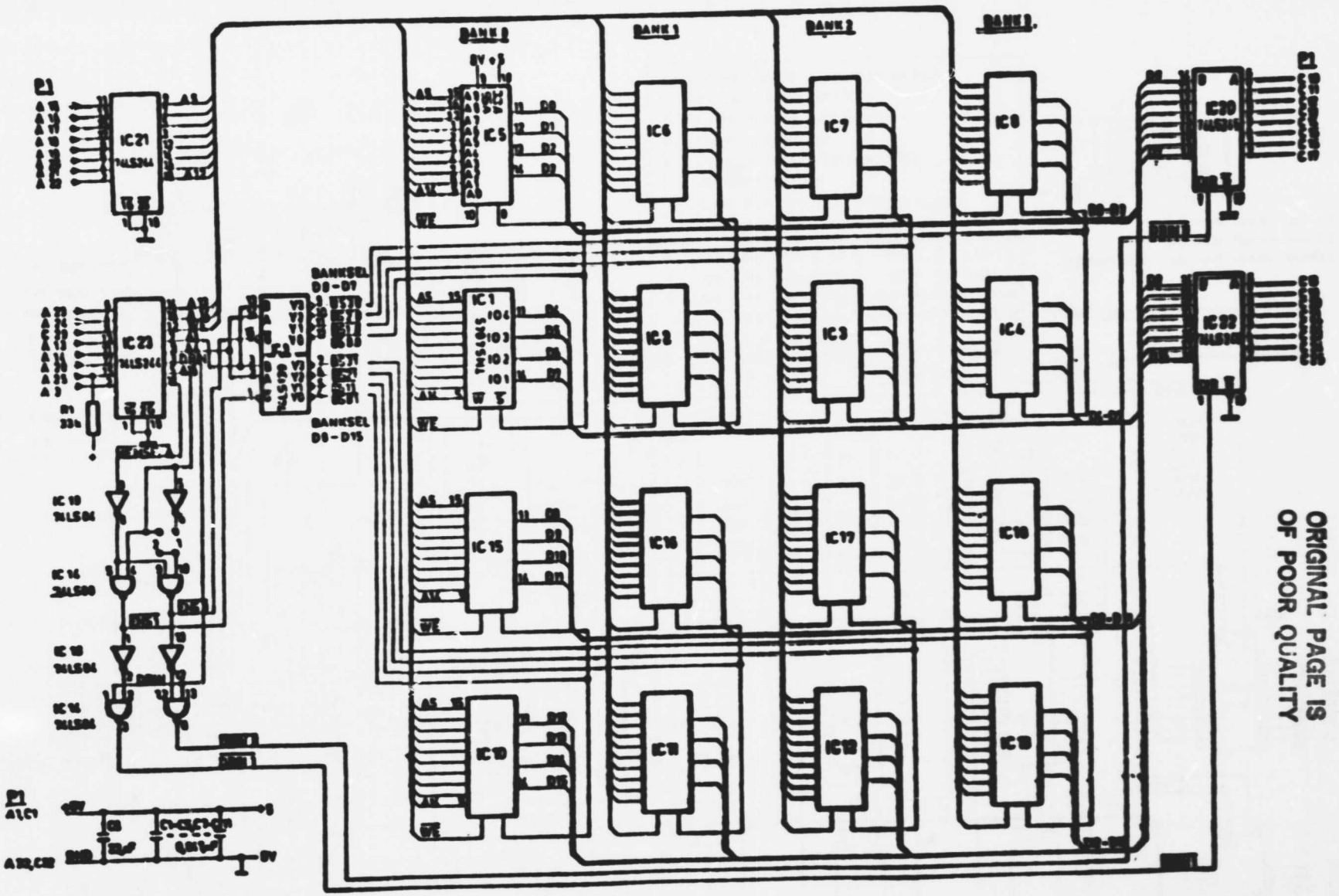


Fig. 8.5:5 S-99 CPU-Board



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Fig. 8.5:6 S-99 8K Sram Board, P/N S99320-01
(Circuit Diagram)

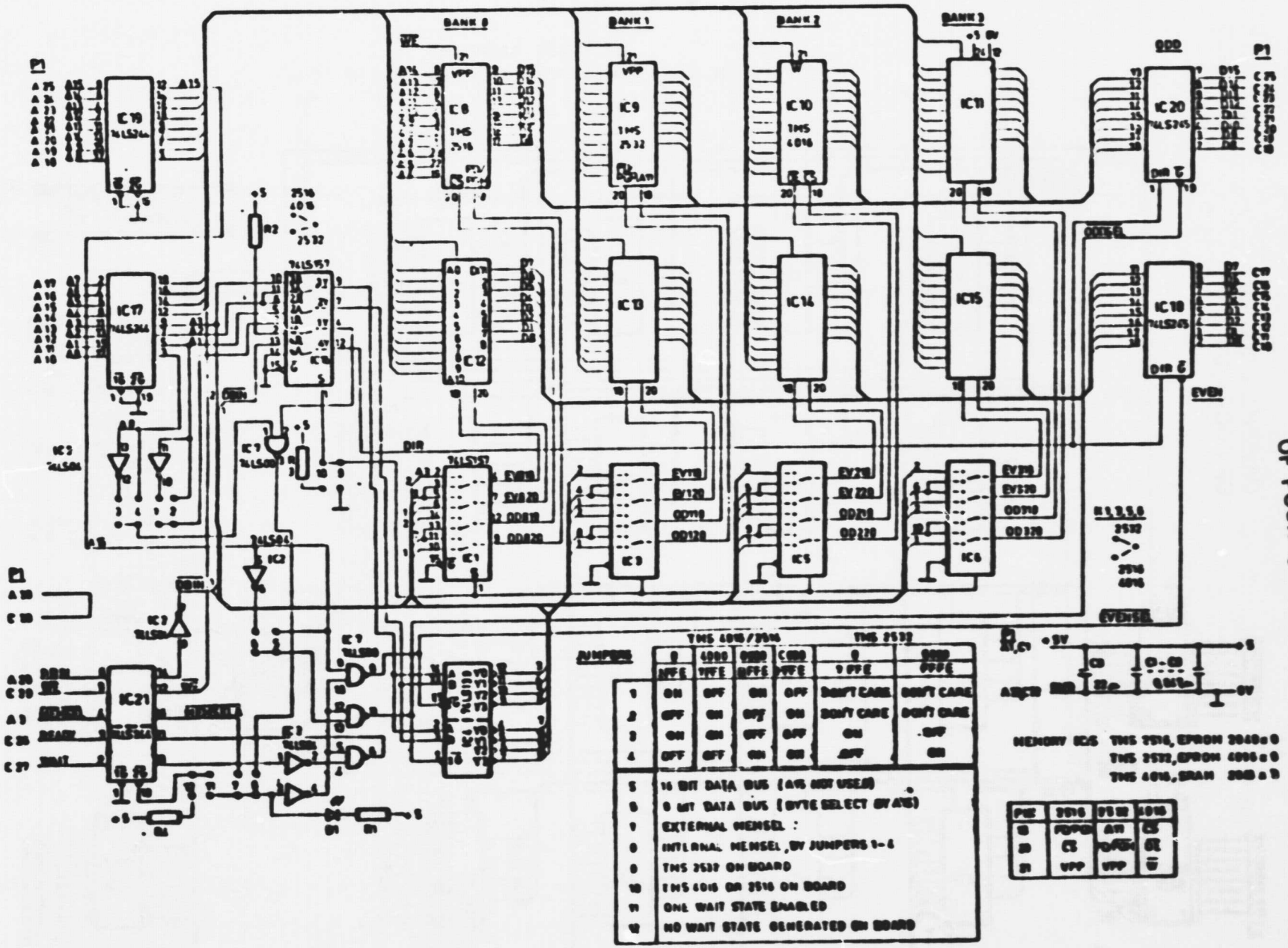


Fig. 8.5:7 S-99 EPROM 16/32 K Board, P/N S-99 420-01

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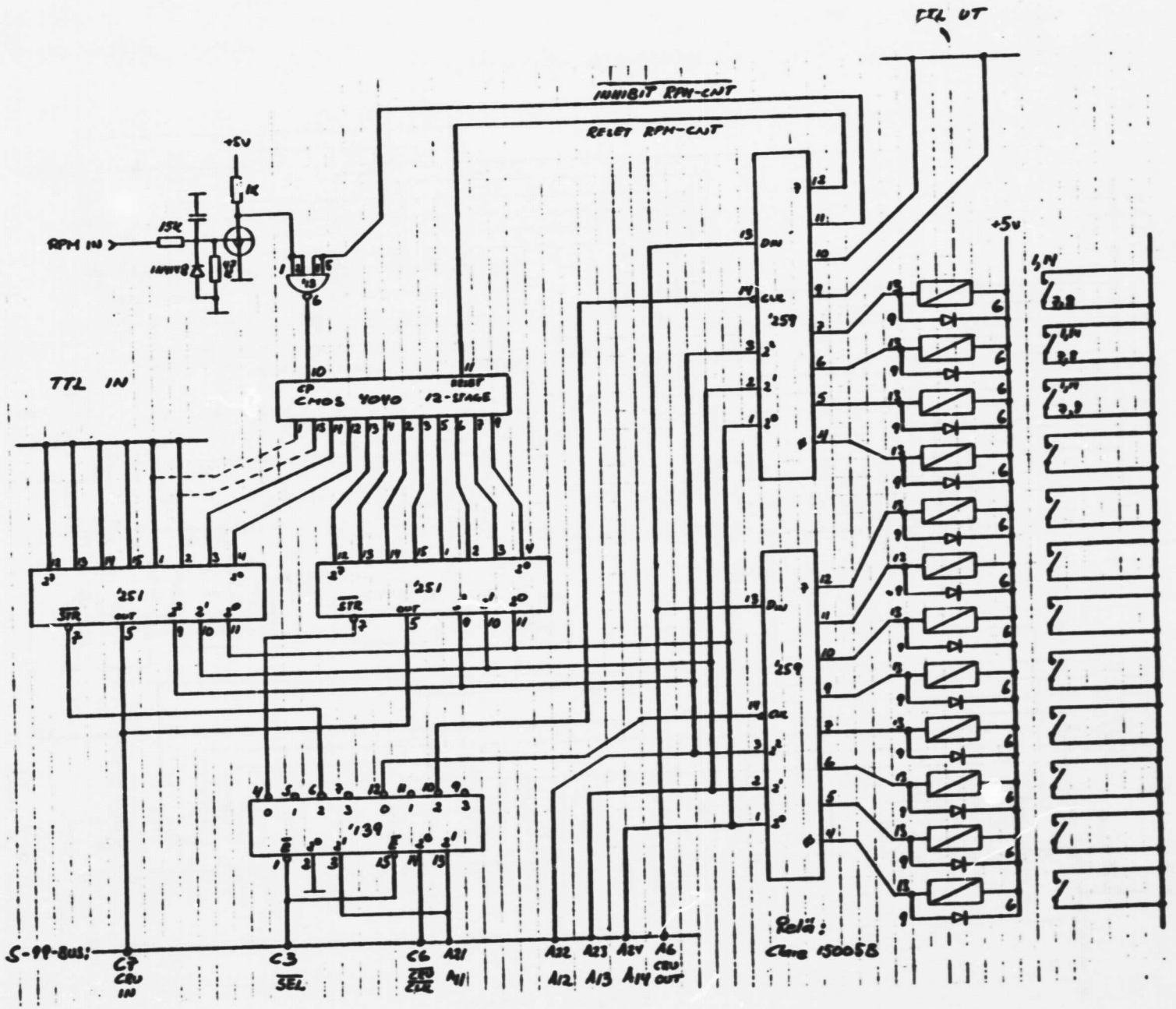


Fig. 8.5:8 I/O Board Relay/TTL In-Out/rpm-Circuit

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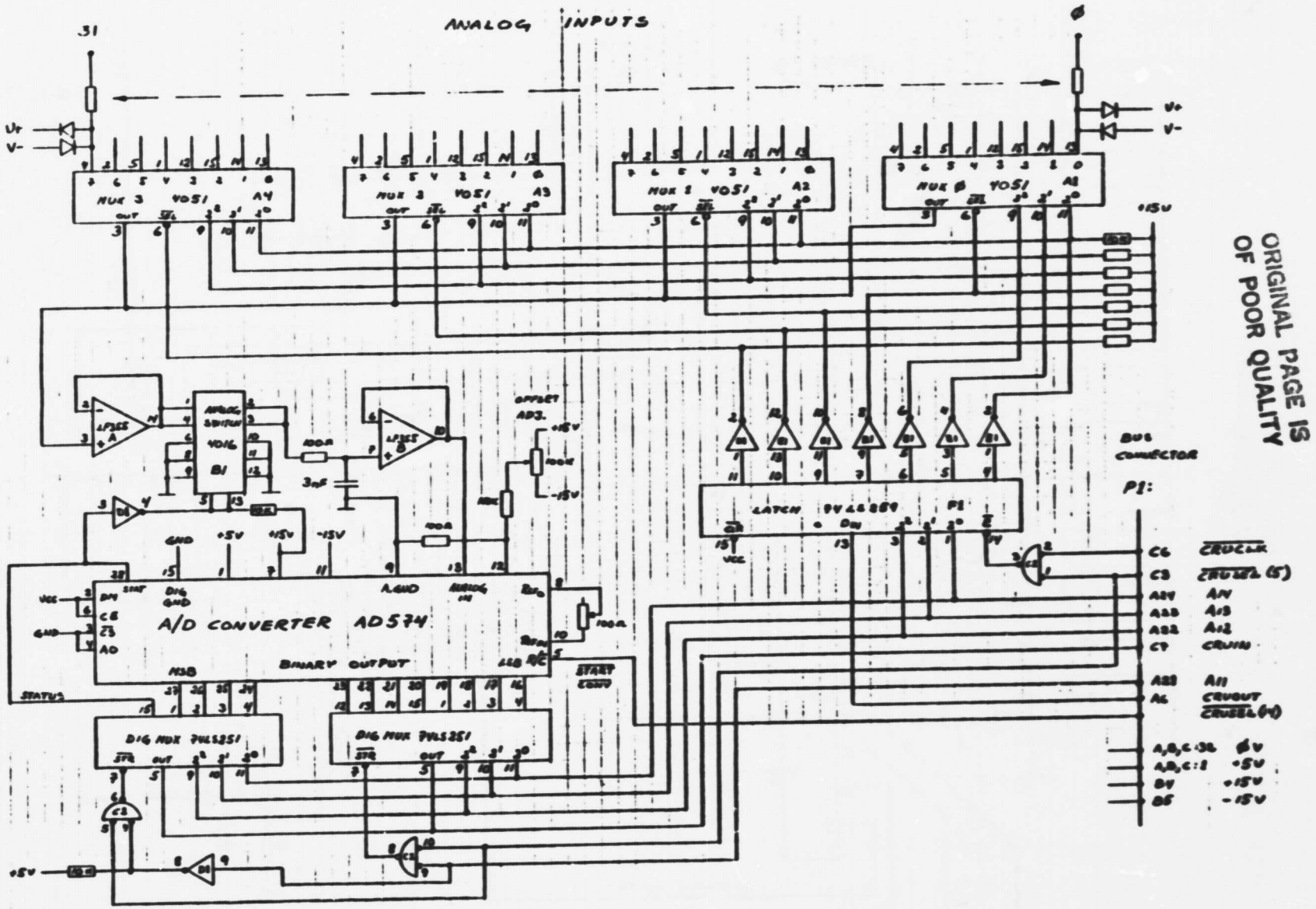


Fig. 8.5:9 A/D Board

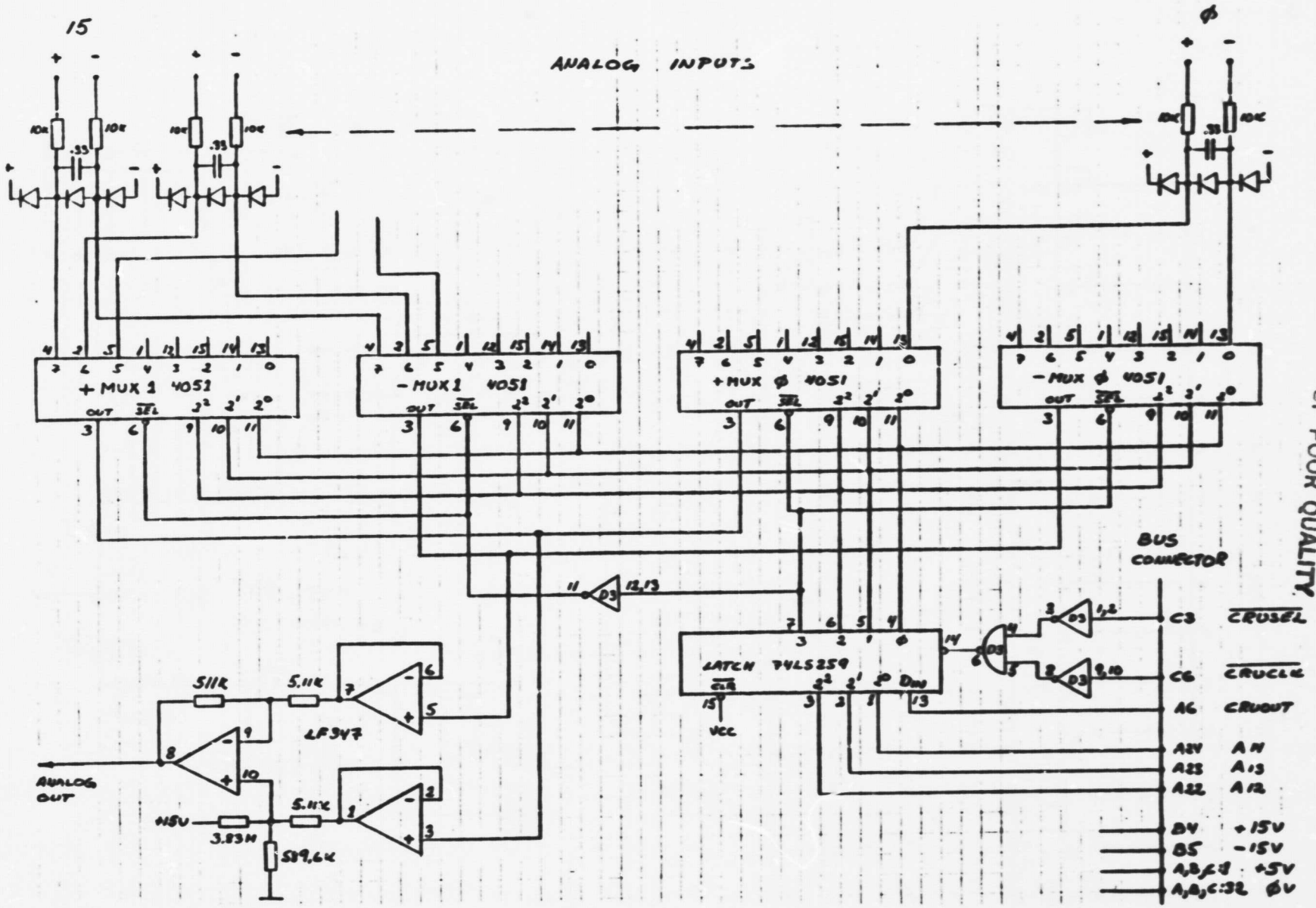


Fig. 8.5:10 Low Level Multiplexer

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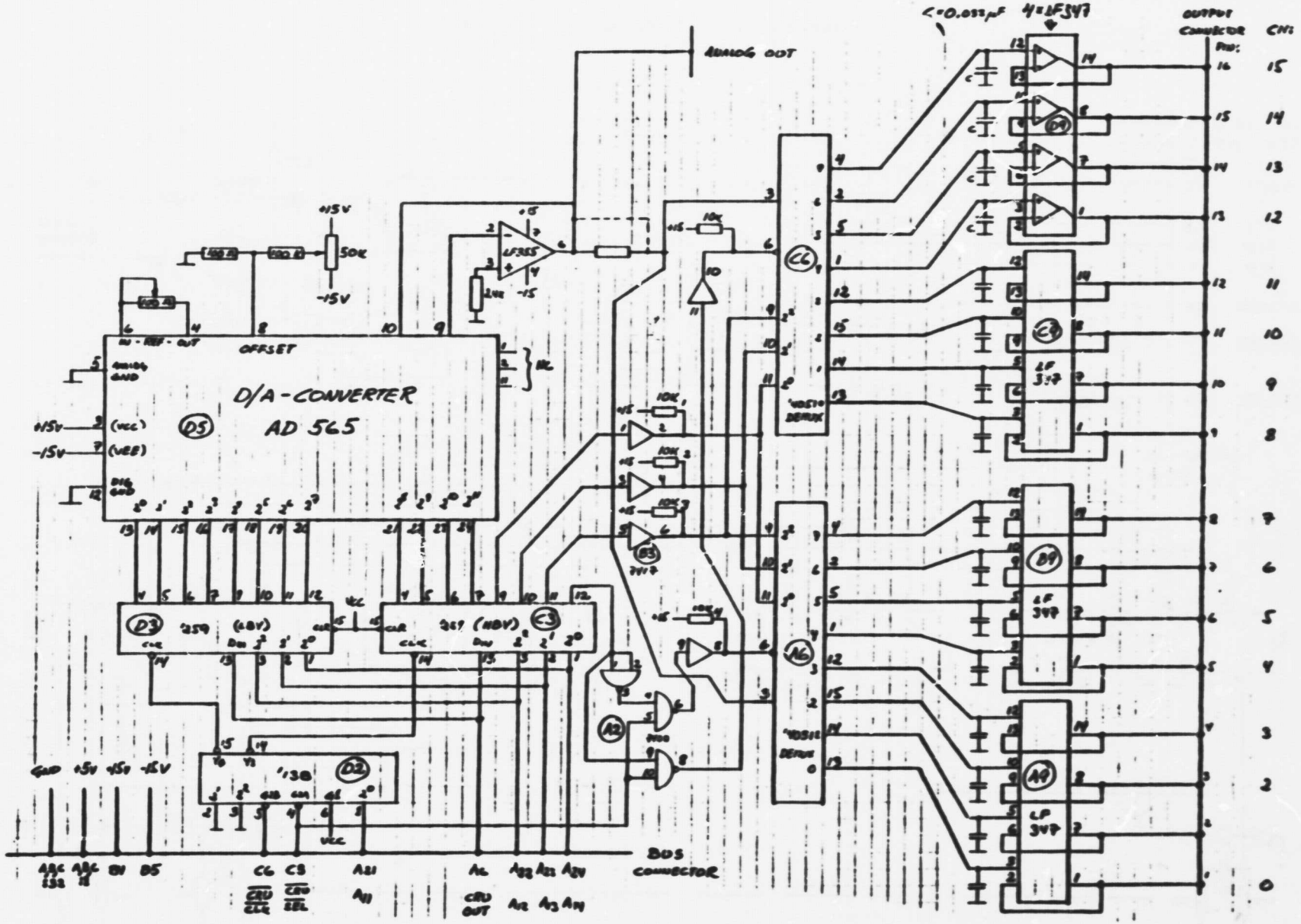


Fig. 8.5:11 D/A Board

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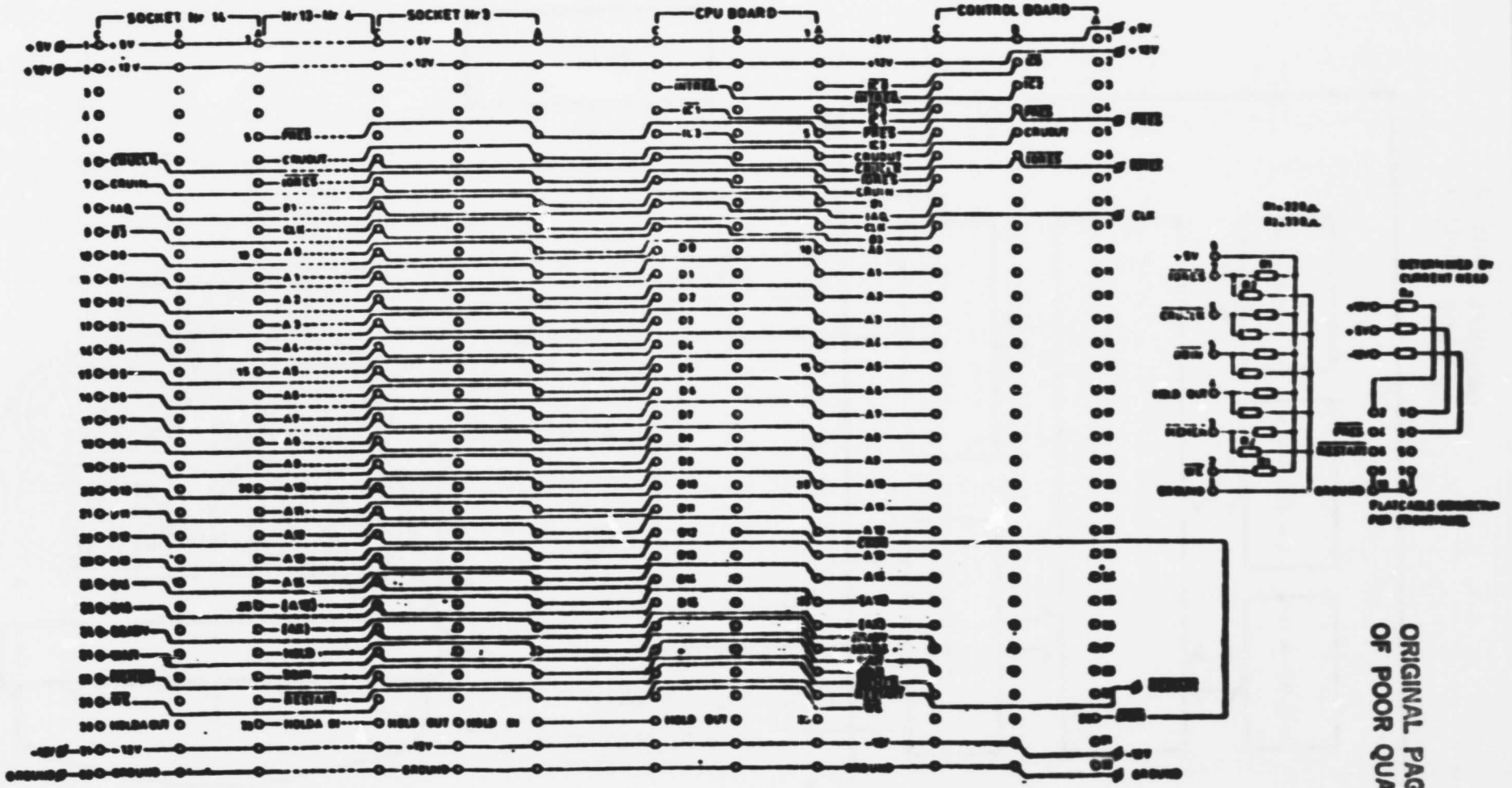


Fig. 8.5:12 Backboard of S-99 System

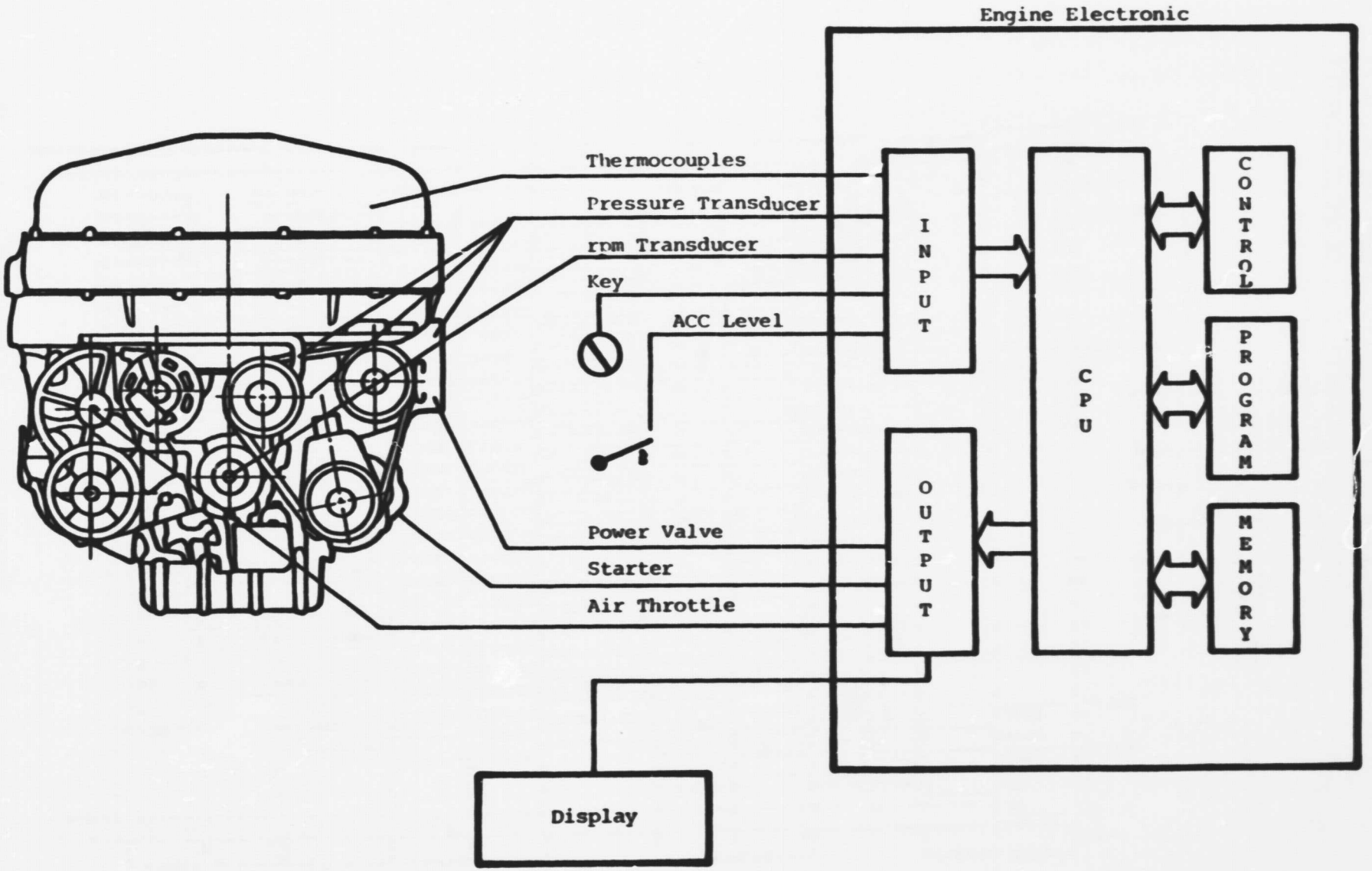


Fig. 8.5:13 Schematic of Digital Control System

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Fig. 8.5:14 Stirling Electronic Development

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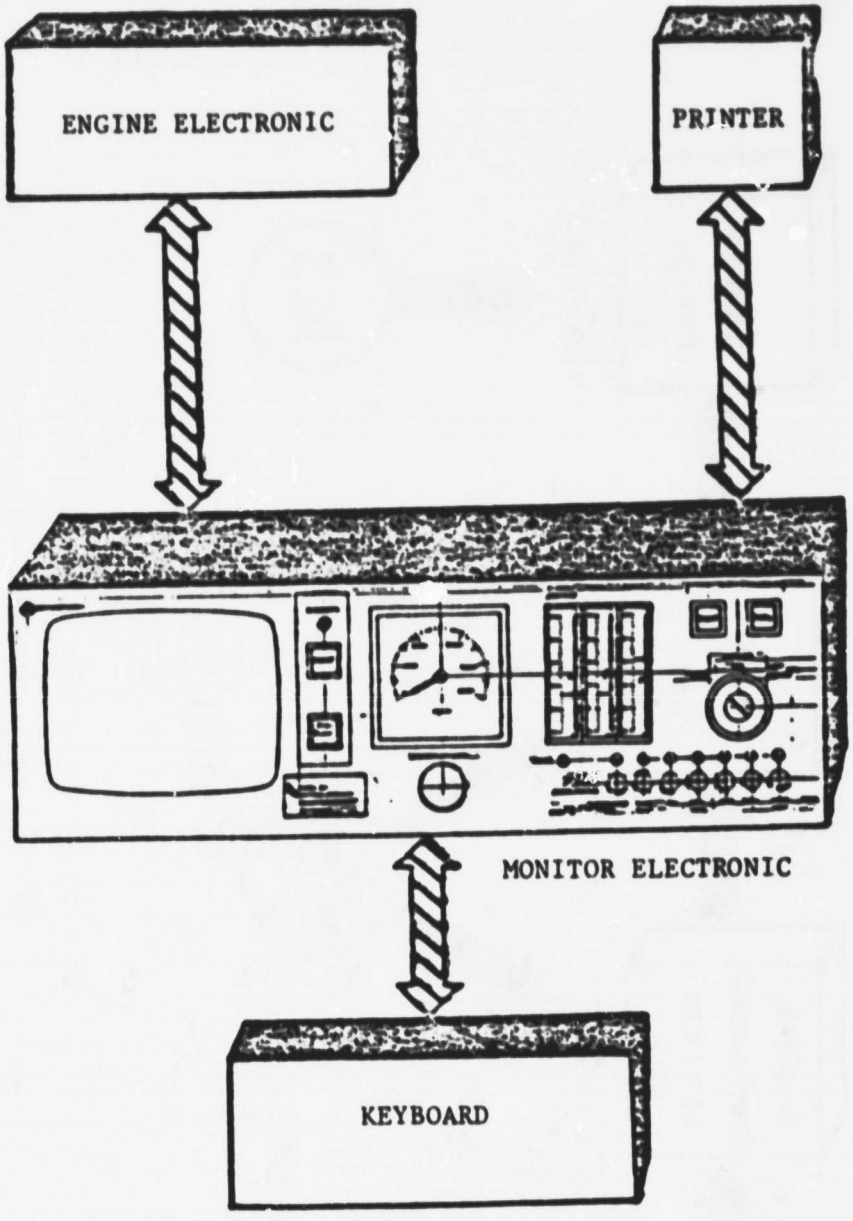


Fig. 8.5:15 Mod I Prototype Electronic Test System

MONITOR

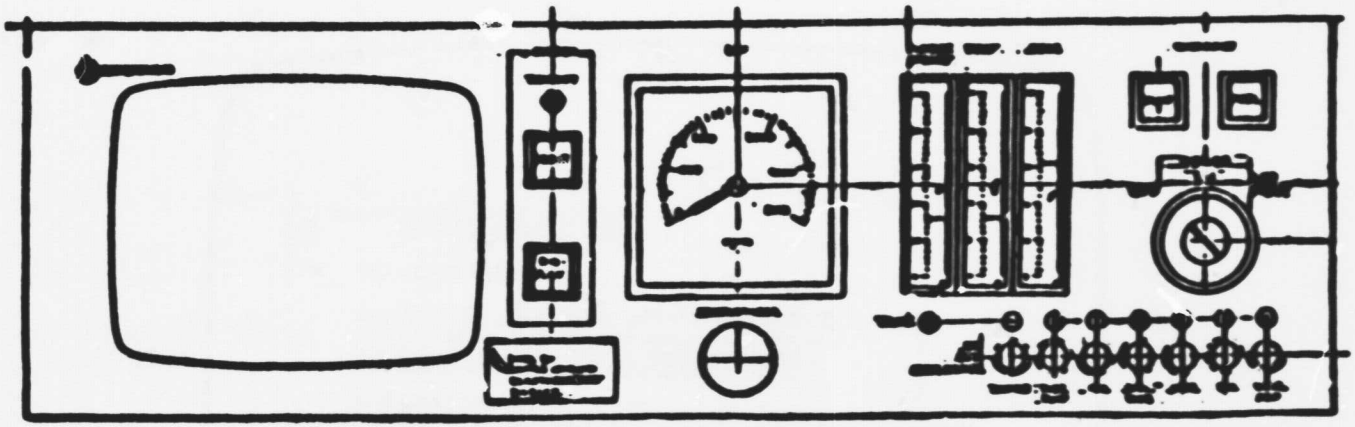


Fig. 8.5:16 Operator's Panel

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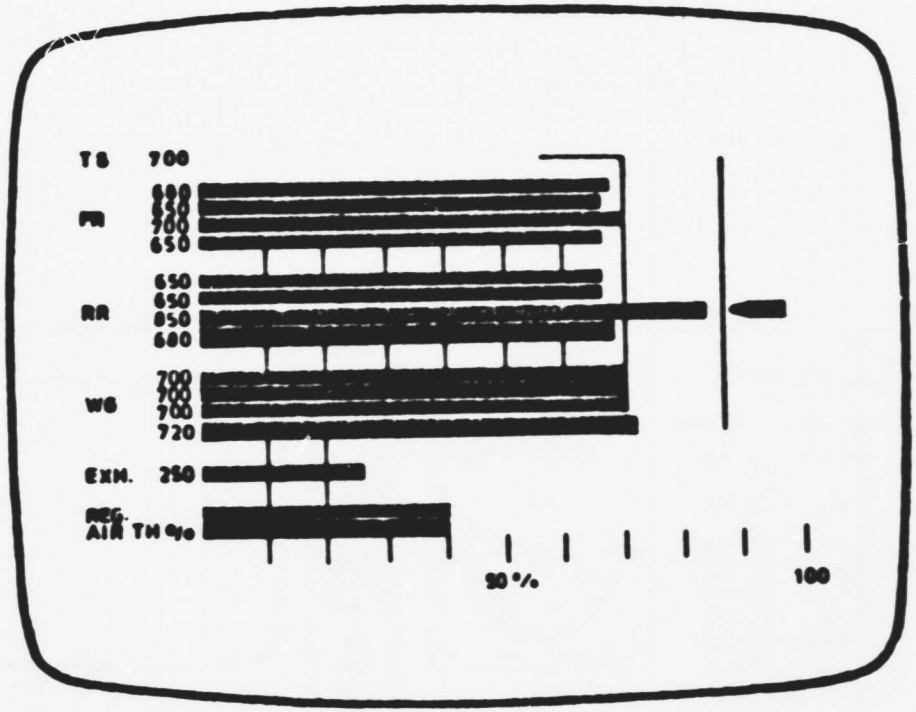


Fig. 8.5:17 Video Screen (Display 1)

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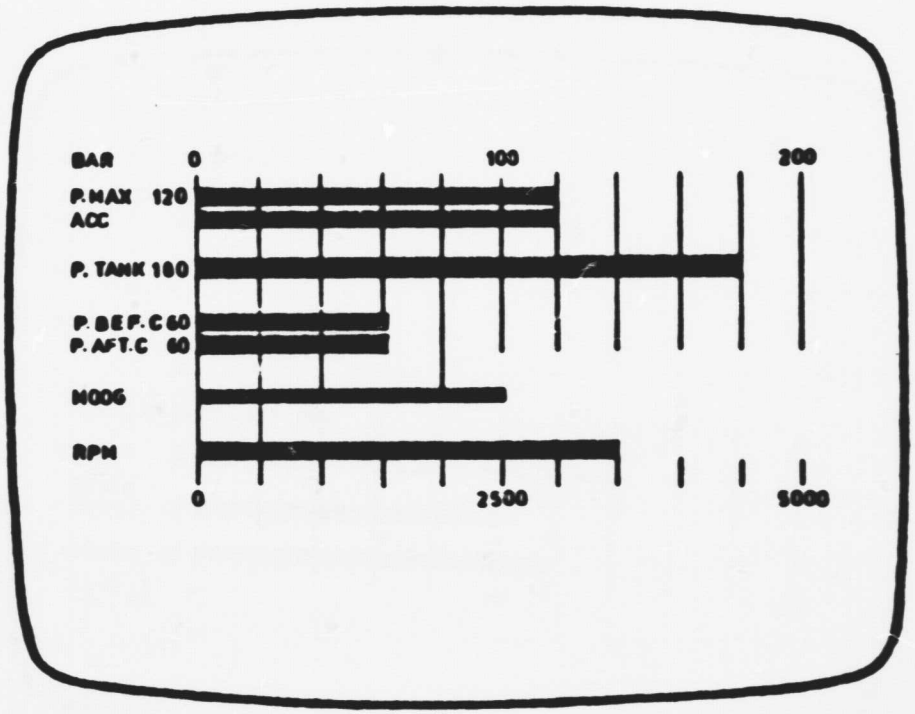
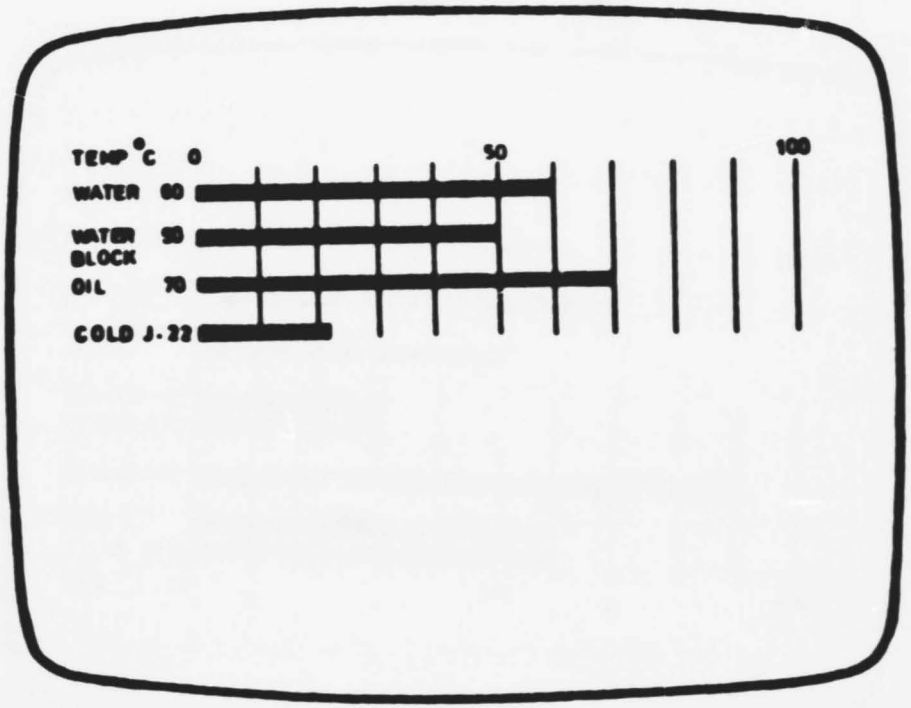


Fig. 8.5:18 Video Screen (Display 2)

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DISPLAY 3

Fig. 8.5:19 Video Screen (Display 3)

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MOD I ELECTRONICS

INPUTS: Others

1. Max Pressure Transducer input level: 2,5-12,5 V e q v 0-300 bar
2. Tank Pressure Transducer- " - - " - - " - - " - - " -
3. Compressor Pressure Transducer - " - - " - - " - - " - -
4. - " - - " - - " - - " - - " - - " - - " - - " -
5. Temp set input level: 0-10 V DC eqv o-10,00^{0}C
6. Feedback from airthrottle servo. Input level: 0-15 V DC eqv 0-90^0 angle.
7. Rpm pick-up. Input level 0-3833.33 Hz, 30-70 V (P-P) DC resistance: 100-130 ohm eqv 0-5000 rpm.
8. Accelerator level. Input level: 0-10 V DC eqv zero-full speed
9. Idle. Input level 0-10 V DC
10. Keyswitch input level: 10 V DC at three different positions (Digital
11. Spare
12. Spare
13. Spare
14. Spare
15. Spare

Fig. 8.5:20 Inputs

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MOD I ELECTRONICS

INPUTS: Temperatures (TC = thermocouple)

1.	Front	row	TC	type	Cr/AL	input	level	appr:	32	mV	e q v	770°C
2.	"	"	"	"	"	"	"	"	"	"	"	"
3.	"	"	"	"	"	"	"	"	"	"	"	"
4.	"	"	"	"	"	"	"	"	"	"	"	"
5.	Rear	row	"	"	"	"	"	"	"	"	"	"
6.	"	"	"	"	"	"	"	"	"	"	"	"
7.	"	"	"	"	"	"	"	"	"	"	"	"
8.	"	"	"	"	"	"	"	"	"	"	"	"
9.	Working	gas	"	"	"	"	"	"	"	"	"	"
10.	"	"	"	"	"	"	"	"	"	"	"	"
11.	"	"	"	"	"	"	"	"	"	"	"	"
12.	"	"	"	"	"	"	"	"	"	"	"	"
13.	Exhaust		"	"	"	"	"	":	10	mV	"	250°C
14.	Watertemp		TC	type	Fe/konst	input	level	appr:	3.73	mV	eqv	70°
15.	Oil temp	variable	resistance	input	level	100-161.7	ohm	e q v	0-100°			
16.	Water temp	block	variable	resistance	input	level	100-161.7	ohm	eqv	0-100°C		
17.	Cold junktion	temperatur	compensation									

Fig. 8.5:21 Signal Levels During Working Conditions

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MOD I ELECTRONICS

1. **Fuel pump relay**
2. **Fuel valve relay**
3. **Ignition relay**
4. **Waterpump relay**
5. **Blower relay**
6. **Starter relay**
7. **Short circ valve relay**
8. **Fan relay**
9. **Rec drive led**
10. **Dump relay. NOTE: Not controlled via computer**
11. Spare
12. Spare
13. Spare
14. Spare
15. Spare

Fig. 8.5:22 Outputs: On/Off

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MOD I ELECTRONICS

1. Air Throttle servomotor: ± 15 V DC. 500 mA
2. Pressure transducer P max: + 15 V DC. 20 mA
3. - " - - " - P tank: + 15 V DC. 20 mA
4. - " - - " - P before compr: + 15 V DC. 20 mA
5. - " - - " - P after compr: + 15 V DC. 20 mA
6. Spare
7. Spare
8. Spare
9. Spare
10. Spare

Fig. 8.5:23 Outputs: Power Supply for Tranducers and Others

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MOD I ELECTRONICS

1. Air throttle servomotor signal: 0-10V DC eqv 0-90^0 angle
2. Air throttle position. Instrumentsignal 0-10V DC 1 mA
3. Rpm. Instrumentsignal 0-10V DC 1 mA eqv 0-5000 rpm
4. Max pressure. Instrumentsignal 0-10V DC 1 mA eqv 0-200 bar
5. Accelerator position. Instrumentsignal 0-10V DC 1 mA eqv 0-200 bar
6. MOOG valve poistion. Instrumentsignal 0-10V DC 1 mA
7. MOOG valve servo signal 0-10V DC
8. Temperature front row TC 1 signal 0-10V 1 mA eqv 0-1000^{0}C
9. - " - -"- -"- TC 2 - " - - " - - " - 0-1000^{0}C
10. - " - -"- -"- TC 3 - " - - " - - " - 0-1000^{0}C
11. - " - -"- -"- TC 4 - " - - " - - " - 0-1000^{0}C
12. Temperature rear row TC 5 - " - - " - - " - 0-1000^{0}C
13. - " - -"- -"- TC 6 - " - - " - - " - 0-1000^{0}C
14. - " - -"- -"- TC 7 - " - - " - - " - 0-1000^{0}C
15. - " - -"- -"- TC 8 - " - - " - - " - 0-1000^{0}C
16. Temperature working gas TC 9 signal 0-10V 1 mA eqv 0-1000^{0}C
17. - " - - " - -"- TC 10 - " - - " - - " - 0-1000^{0}C
18. - " - - " - -"- TC 11 - " - - " - - " - 0-1000^{0}C
19. - " - - " - -"- TC 12 - " - - " - - " - 0-1000^{0}C
20. Water temperature signal 0-10V DC 1 mA eqv 0-100^{0}C
21. Max pressure signal 0-10V DC 1mA eqv 0-200 bar
22. Tank pressure -"- - " - - " - - " - -"-
23. Spare
24. Spare
25. Spare

Fig. 8.5:24 Outputs: Analog Signals

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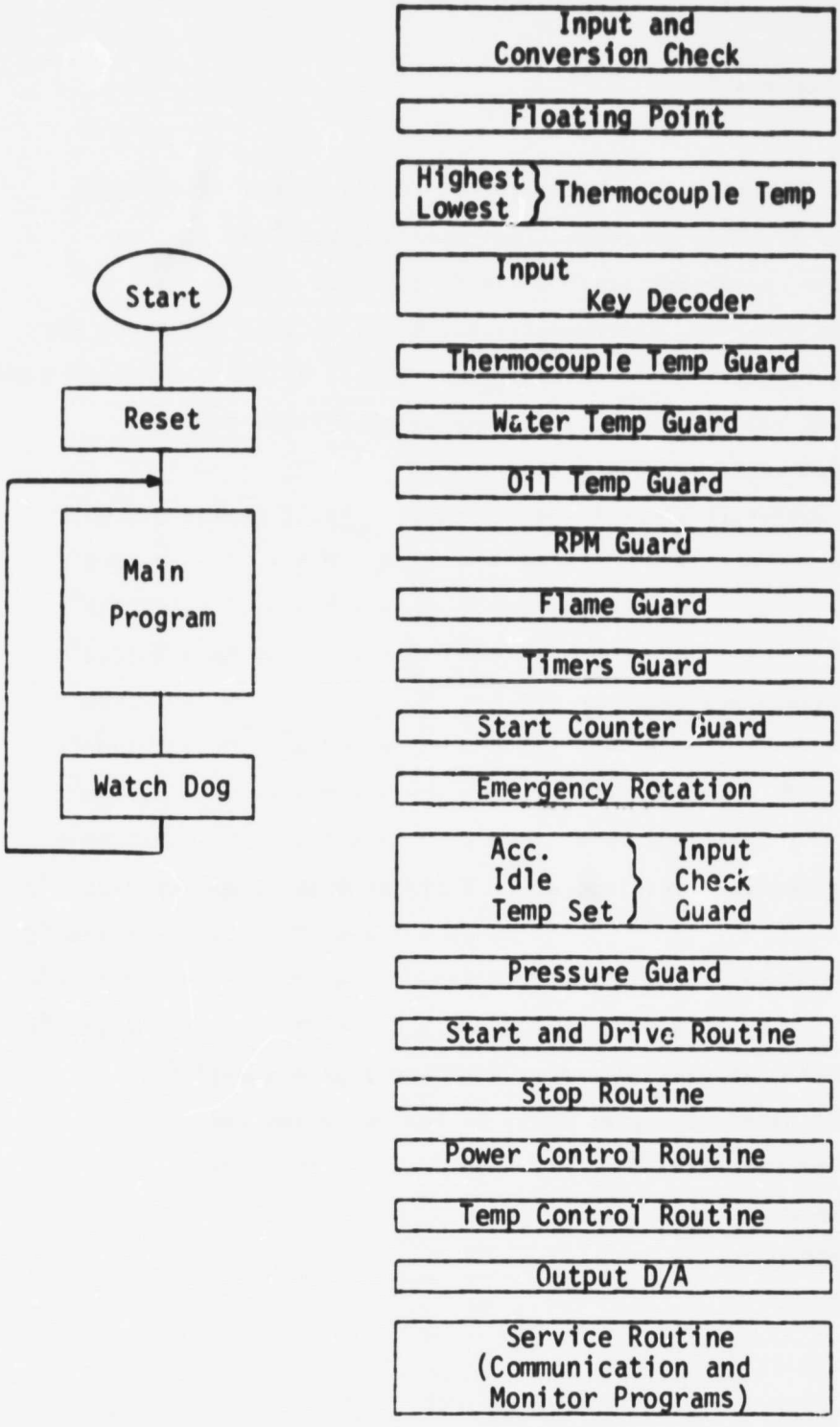


Fig. 8.5:25 Main Program Flow Diagram

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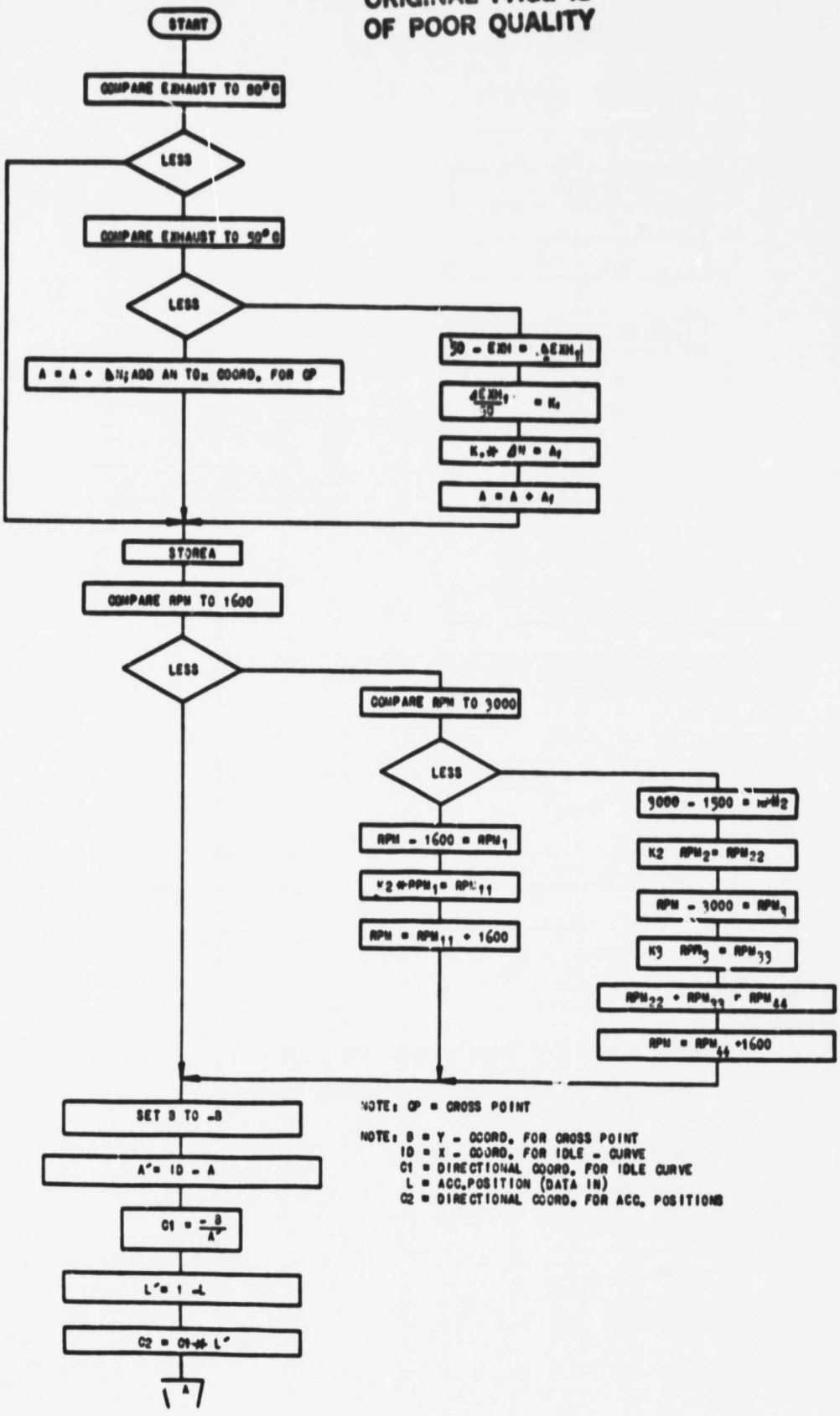


Fig. 8.5:26. Power Control Board

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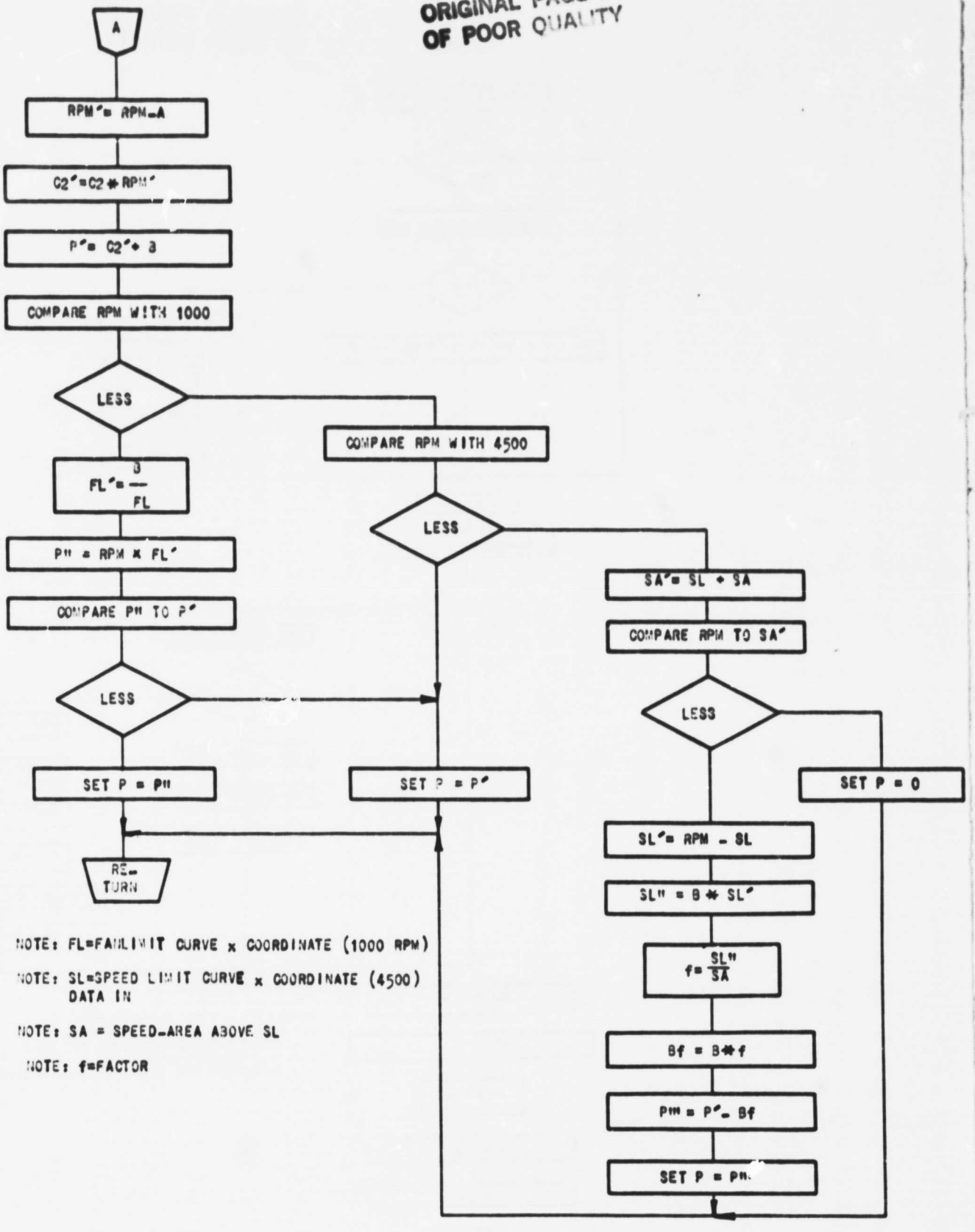


Fig. 8.5:27 Control Program Flow Diagram

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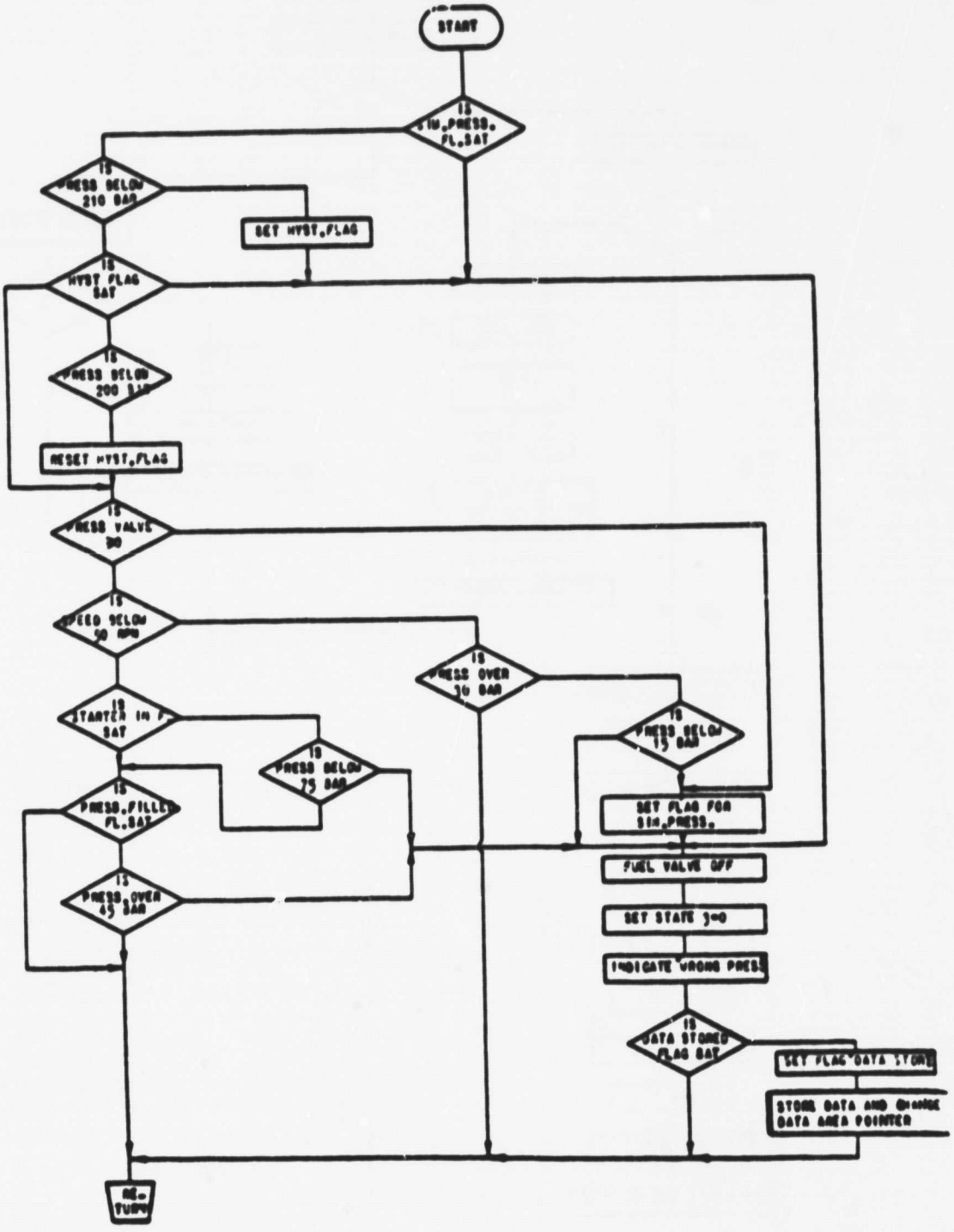


Fig. 8.5:28 Pressure Guard

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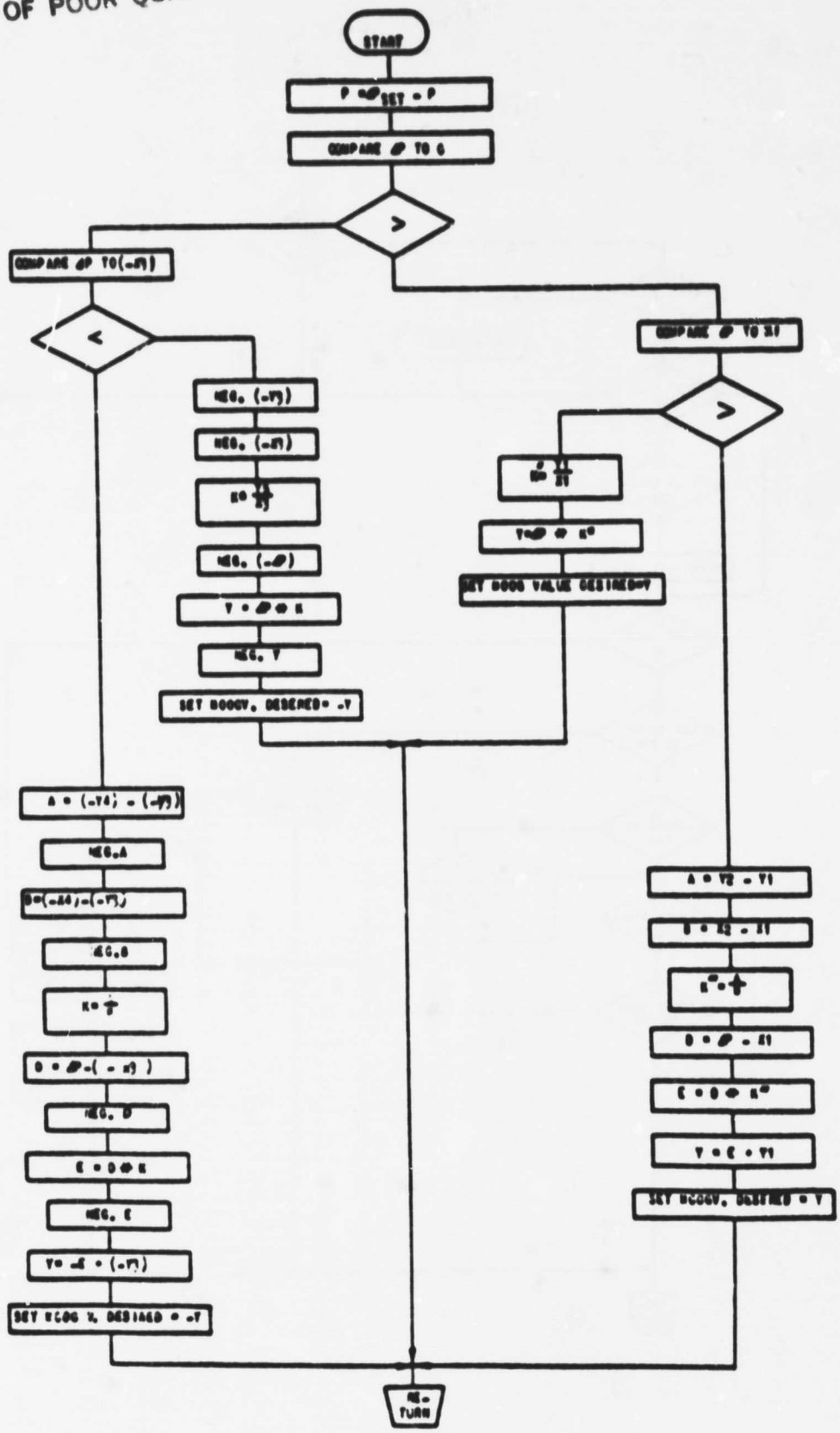
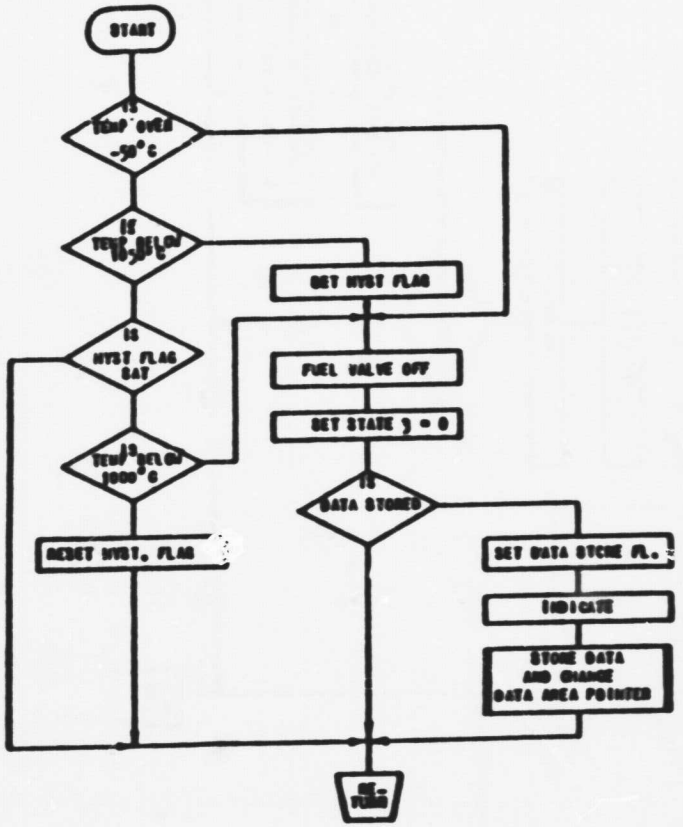


Fig. 8.5:29 Power Control S Curve

EXHAUST GASES

BEFORE PREHEATER



WATERTEMP

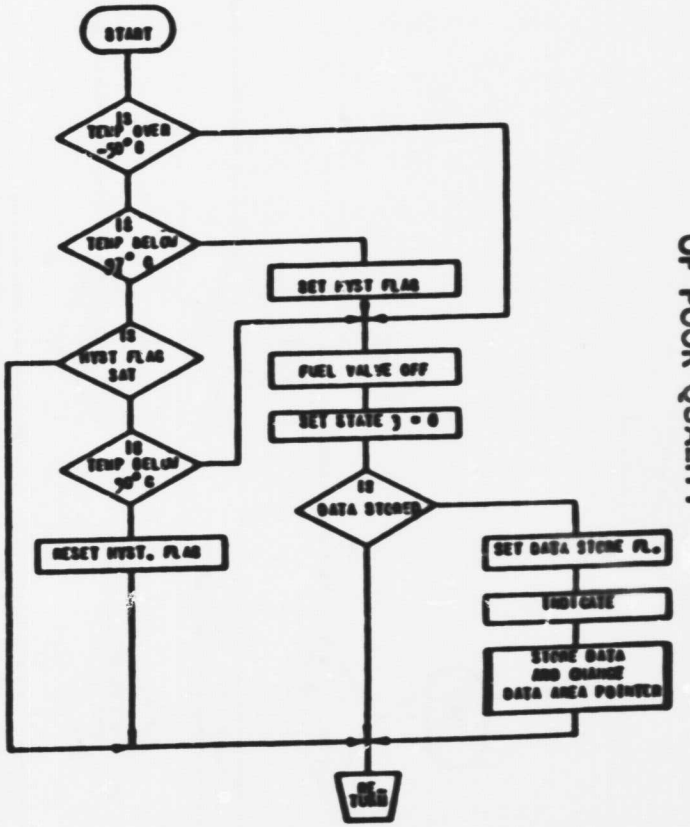


Fig. 8.5:30 Exhaust Gases and Watertemp

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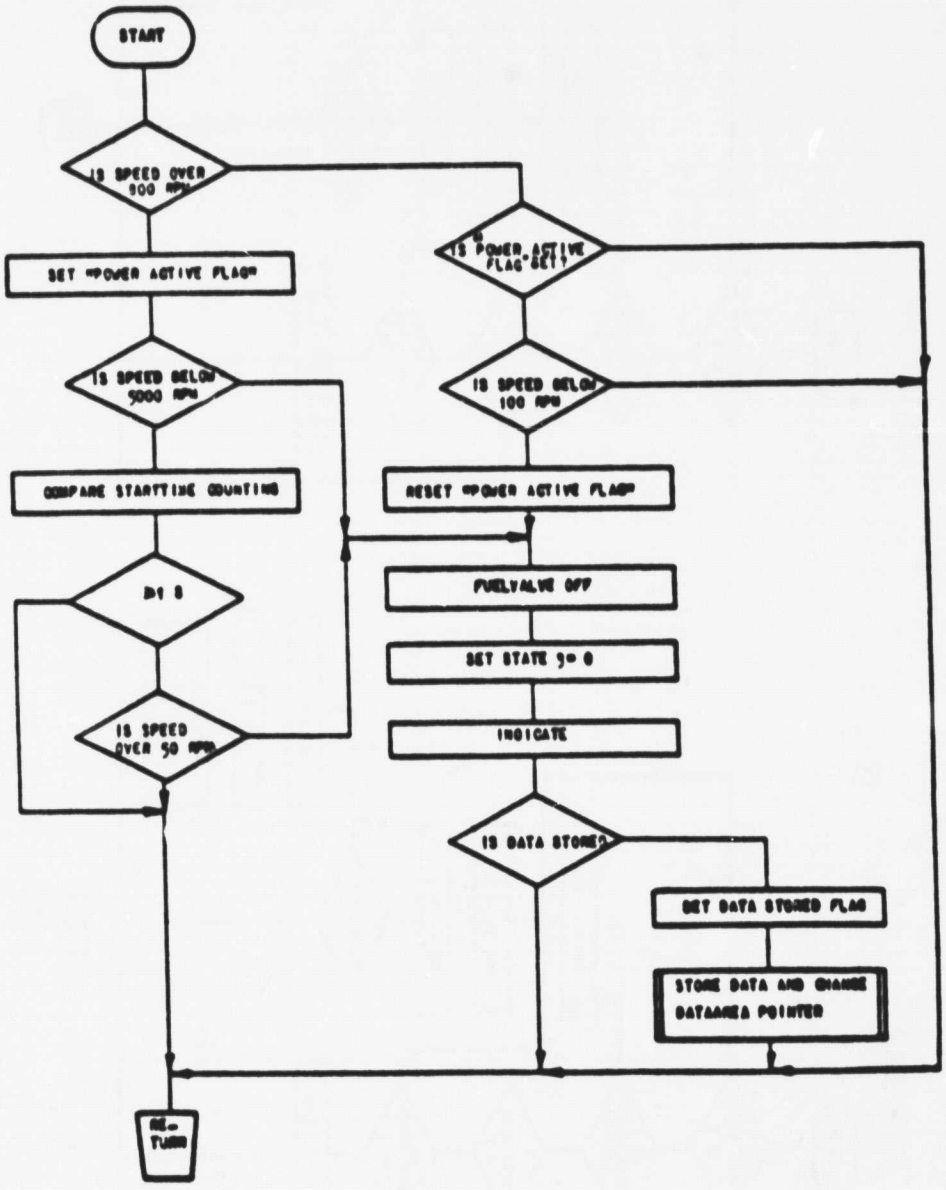


Fig. 8.5:31 Speed (RPM) Guard

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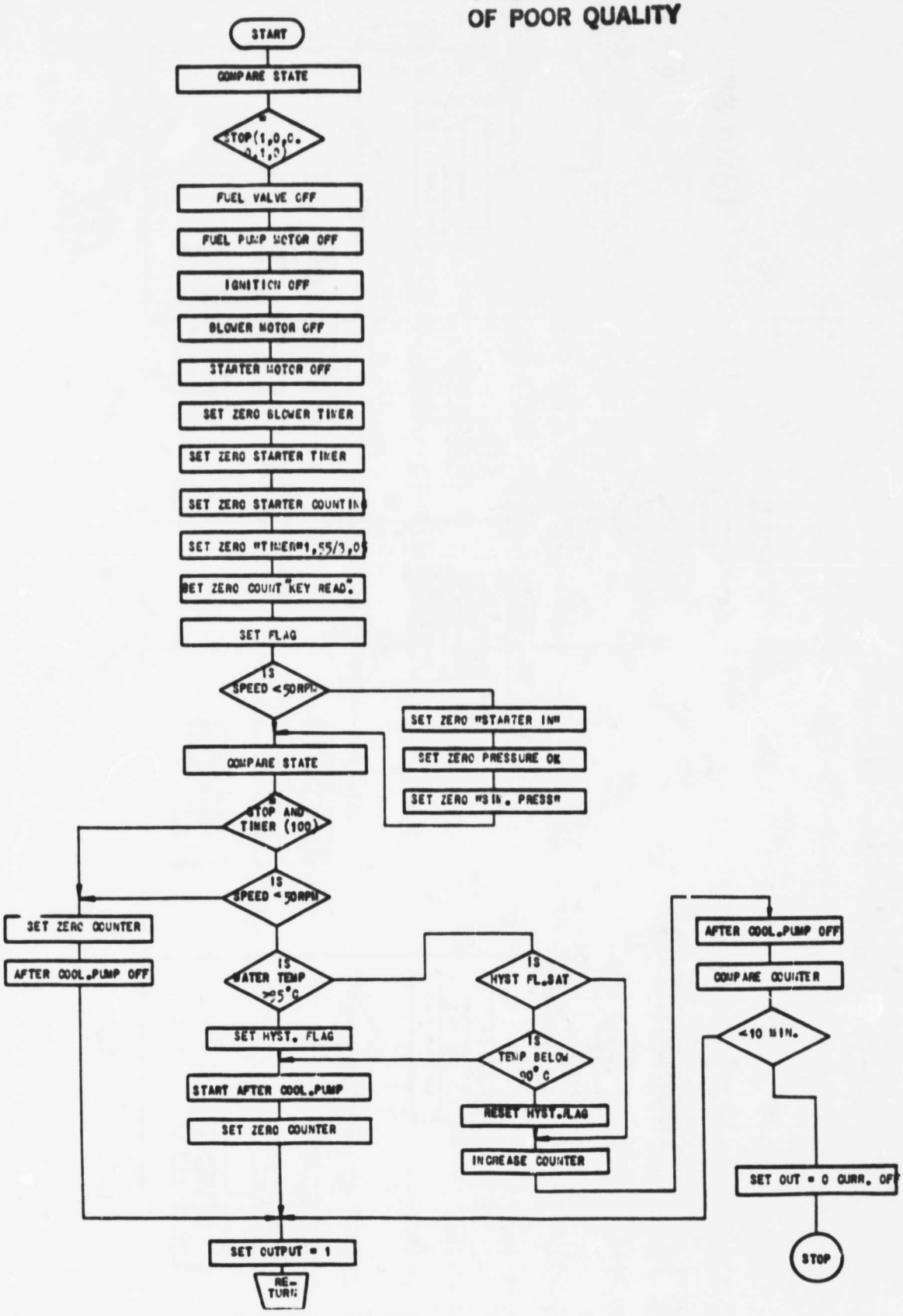


Fig. 8.5:32 Stop Routine

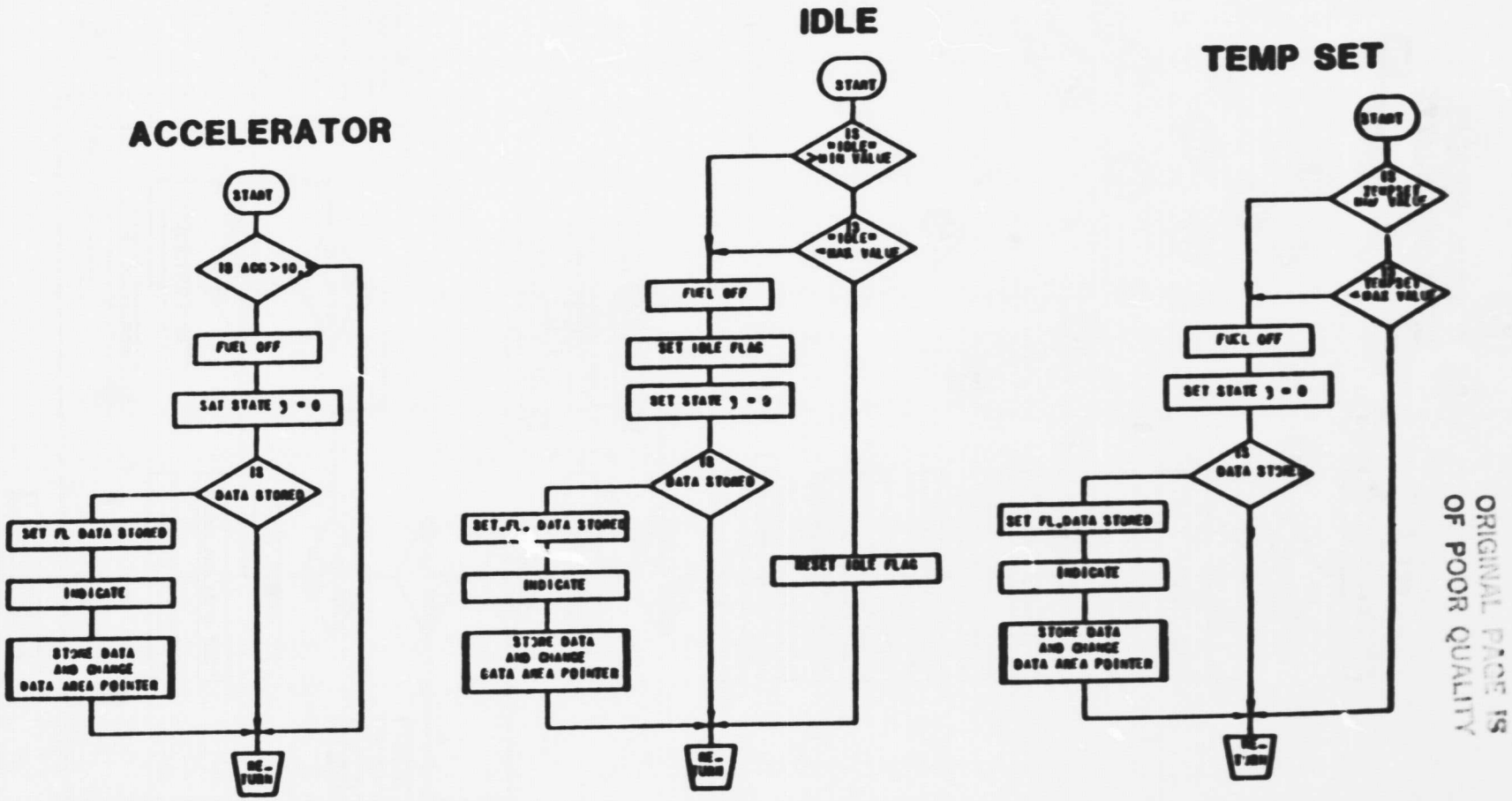
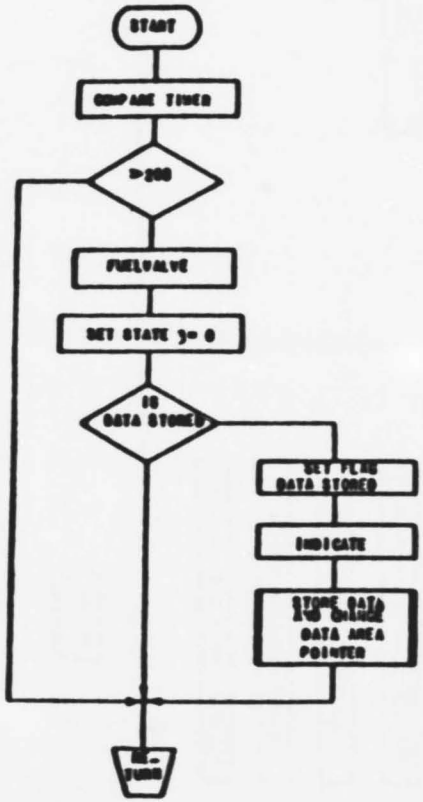


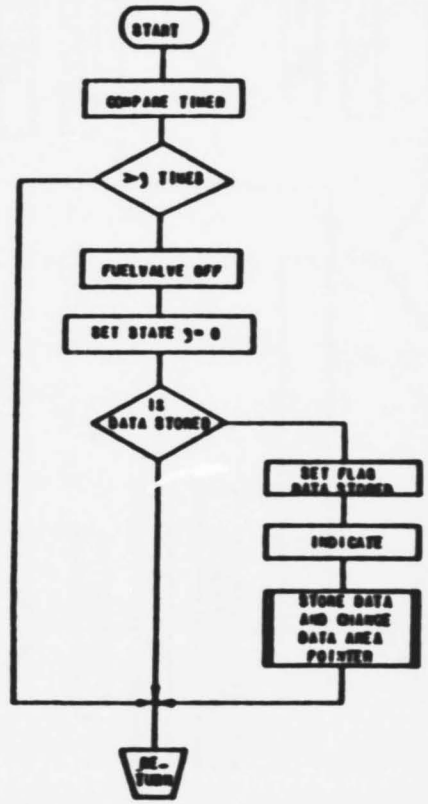
Fig. 8.5:33 Test Data In

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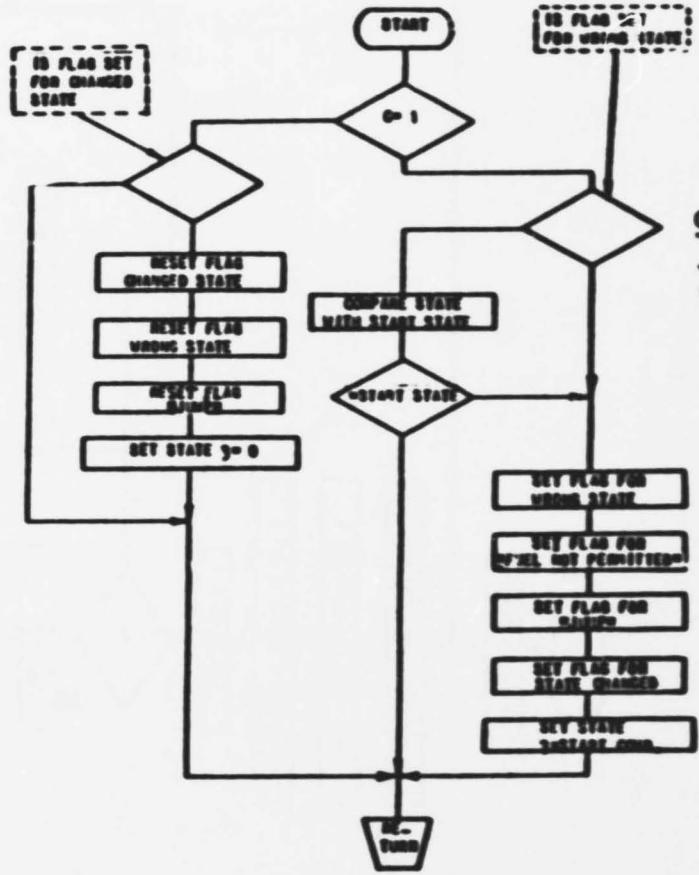
START TIME GUARD



START COUNTER GUARD



EMERGENCY GUARD



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Fig. 8.5:34 Guards

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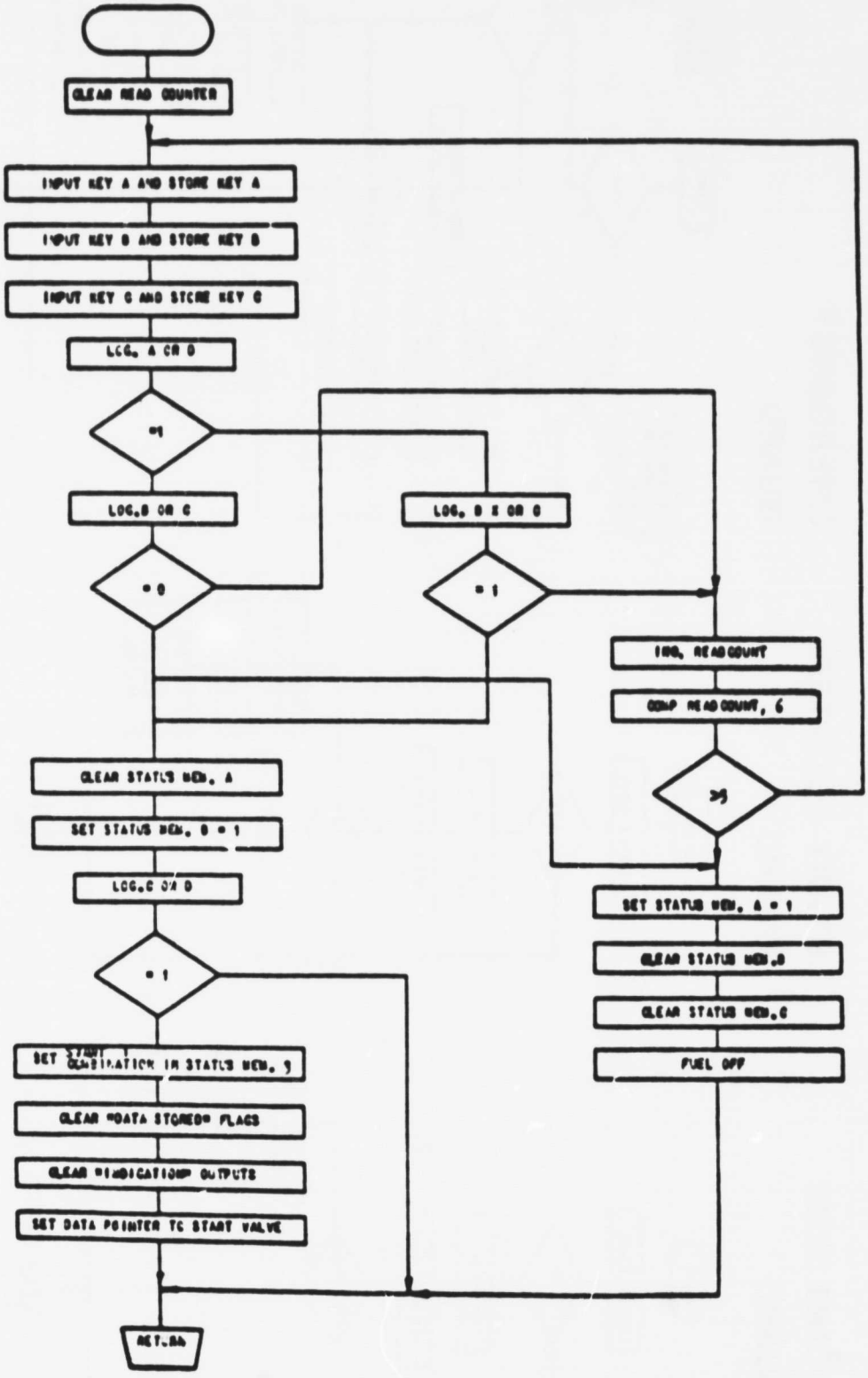


Fig. 8.5:35 Key Decoder

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8.6 ARRANGEMENT

8.6.1 Arrangement of Auxiliaries, Accessories and Driving Belts

8.6.1.1 Front View

Alternator, radiator fan, air conditioning compressor, hydraulic pump (for power steering and brake booster), and the variator belt, are all driven by the front end of the drive shaft. The hydraulically governed variator transmission is used for driving both the burner blower and the combined atomizer air compressor-oil pump.

The alternator and the radiator fan are driven by one belt from the front end of the main shaft, with the alternator mounted on a pivot to act as a belt tensioner. The speed ratio* for the alternator is 2.55:1, which has been checked by AMG to give sufficient charging capacity with the chosen Bosch 55 Amp alternator. The radiator fan drive ratio shown on the drawing is 1.2:1, and space is available for a clutch arrangement between the shaft and the fan hub. The optimal fan speed ratio and clutch arrangement has to be chosen after the planned cooling system tests.

In front of the above mentioned belt is the belt for the variable speed drive for the burner blower, with the driving pulley located also on the main shaft.

A second pulley in front of the driven pulley on the fan shaft is used to drive the power steering pump and the air conditioning compressor, with speed ratios relative to engine speed of 1.2:1 and 1.3:1 respectively. Here the belt tension can be adjusted by means of the power steering pump, which is mounted on a pivot.

The belt for the variator drive is a special type of V-belt, and the

* Speed ratio is defined as speed of driven member divided by speed of driving member.

other two belts are of the Dayco Polyrib type.

For location of these parts, see photos on page II-351.

The hydrogen gas compressor is located in the front of the engine and directly driven by a connecting rod from the left hand crank shaft. Consequently, the speed ratio is 1:1 for this unit.

8.6.1.2 Left Hand Side (Steering Column Side)

On this drawing is shown the 1.8 HP starter motor, which is directly mounted on the transmission case, close to the steering gear.

8.6.1.3 Right Hand Side

The burner air blower is located on this side in the front end of the engine. The air inlet pipe is located in the front of the blower (directed forwards) and the air outlet pipe is directed upwards. The air is guided by a short pipe directly into the air manifold of the preheater housing.

The driven pulley of the variator drive is directly mounted on the low speed shaft of the blower transmission. This shaft goes through the blower transmission case via an overrunning clutch, and is by means of a toothed belt driving the atomizer air compressor-oil pump unit, which is located behind the burner blower. The speed ratio between these two shafts will be approximately 0.8:1.

A toothed belt will be used for this transmission.

The same toothed belt is coupled to the electric motor for driving the burner air blower and the atomizer air compressor during the starting up period, before the engine is running. This electric motor is located behind the blower and above the atomizer air compressor. The speed ratio between the electric motor and the burner air blower is 1:1. The pulley on this electric motor shaft also includes a freewheel to prevent the electric motor from being driven by the engine.

The water pump is located behind the atomizer air compressor unit, and is directly driven by a gear transmission from one of the engine crank shafts. Speed ratio for the water pump is 0.916:1. The inlet pipe for the water pump is located in the front of the pump behind and below the atomizer compressor unit. From the water pump only a short hose is needed to lead the water into the engine.

8.6.2 Location of Engine Mounted Equipment

8.6.2.1 Power Control System

The power control system consists of several components such as safety valves, filters, check valves etc. Most of these components are integrated in blocks. One of these blocks, designated no 1, also includes the power control valve and consequently forms a rather big unit. Another block, designated no 3, includes the gas cooler and the short circuiting valve for the hydrogen compressor and, for this reason, be located close to the compressor. These two units should preferably be mounted on the engine in order to avoid the need of flexible tubing. The most suitable place for block no. 1 is on the front end of the engine close to and above the hydrogen gas compressor.

Block no 3, will be located behind the compressor.

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A third block, designated no 2, can from a functional point of view be mounted either on the engine or on the vehicle. There will be further comments in Section 8.6.3.

8.6.2.2 Air-Fuel System

Components used for the air fuel system are principally mounted on the vehicle, except for some small valves, connections, inlet pipes, etc (not shown). See section 8.6.3.

8.6.2.3 Cooling System

The hydrogen compressor and the hydrogen gas from the compressor outlet have to be cooled. Therefore, the compressor and block no. 3 (containing the gas cooler) have to be connected to the cooling water circuit. The tubes will be only 8-10 mm O.D., and they are not shown on the drawings.

It is not anticipated that an oil cooler will be needed in the vehicle installations.

The water pump has been dealt with in section 8.6.1.3.

8.6.3 Location of Vehicle Mounted Equipment

8.6.3.1 Power Control System

Block no 2 will be mounted on the vehicle, close to block no. 1.

The hydrogen gas storage vessel will be located in the left hand front fender as is shown in the referring photo. This is the same place as for the P40 installation in the Spirit automobile. This is also valid for the servo system oil tank.

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8.6.3.2 Air-Fuel System

The air-fuel system , chosen for the MOD I engine, is based on a separate air flow meter and metering fuel pump.
Since, the shape and arrangement of the air flow meter is not settled, the photos of the mock-up installation instead show the back-up system (Bosch K-Jetronic), which needs more space than the planned system. The air throttle is located in the duct connecting air flow meter and burner blower inlet.

Several minor components belonging to this system will also be mounted on the vehicle, at locations which will be decided later.

8.6.3.3 Cooling System

The after cooling pump will be placed in one of the tube bends in the water return line form the radiator to the water pump.

8.6.3.4 Electronic system

A junction and relay box, about 250 x 150 x 150 mm will be located in the engine compartment, close to the power control box no. 2. If required, the terminal box will be divided into two units.

The electronic unit, containing microprocessor and interface systems, is approximately dimensioned 200 x 150 x 75 mm. The proposed location for this is in the recess in the transmission tunnel.

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8.6.4 Drawings

Figure	Drawing No.	Description
1	0-17150	Front View
2	0-17150	Right Side View
3	0-17150	Left Side View

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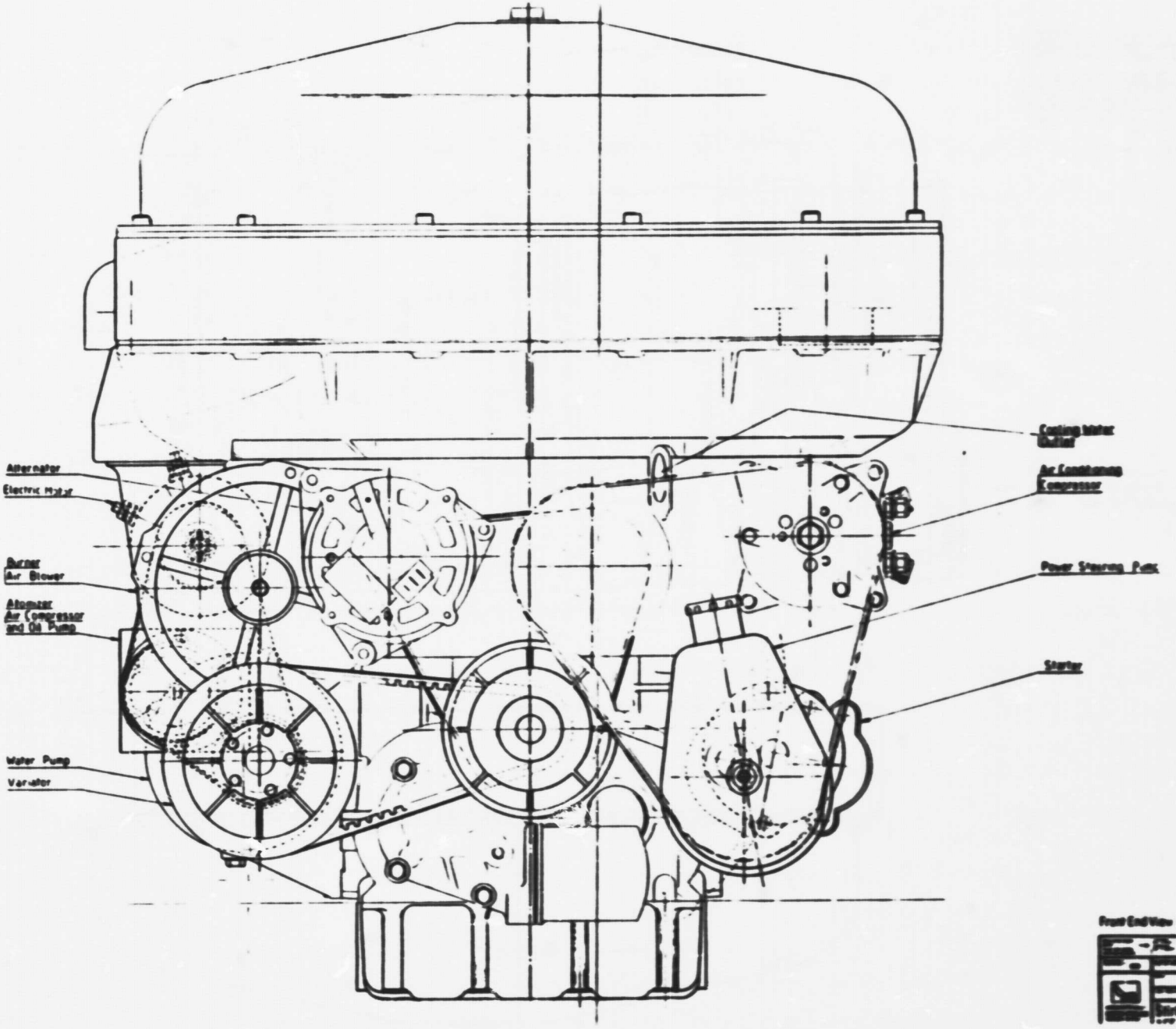


Fig. 1 SES (Front View)

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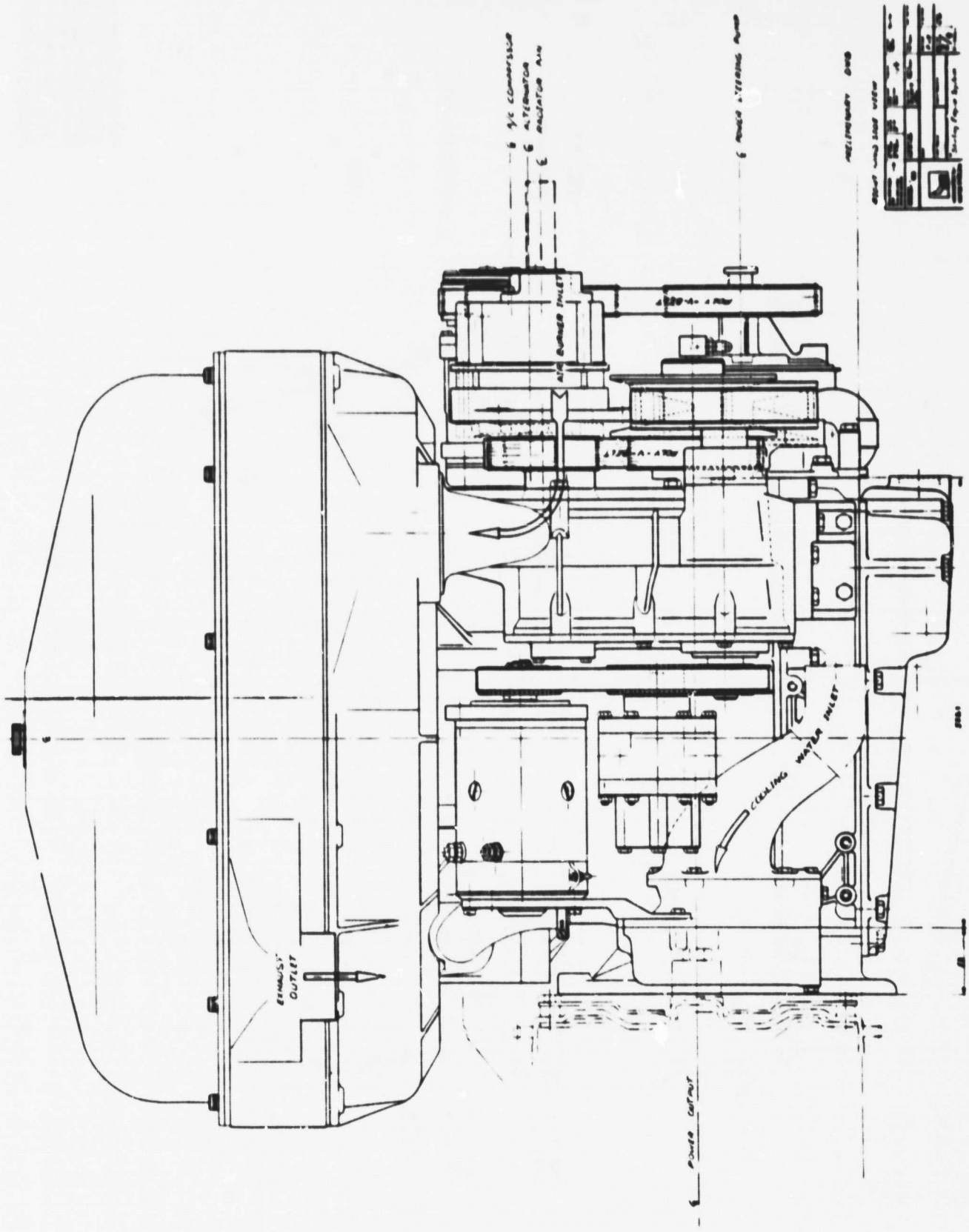
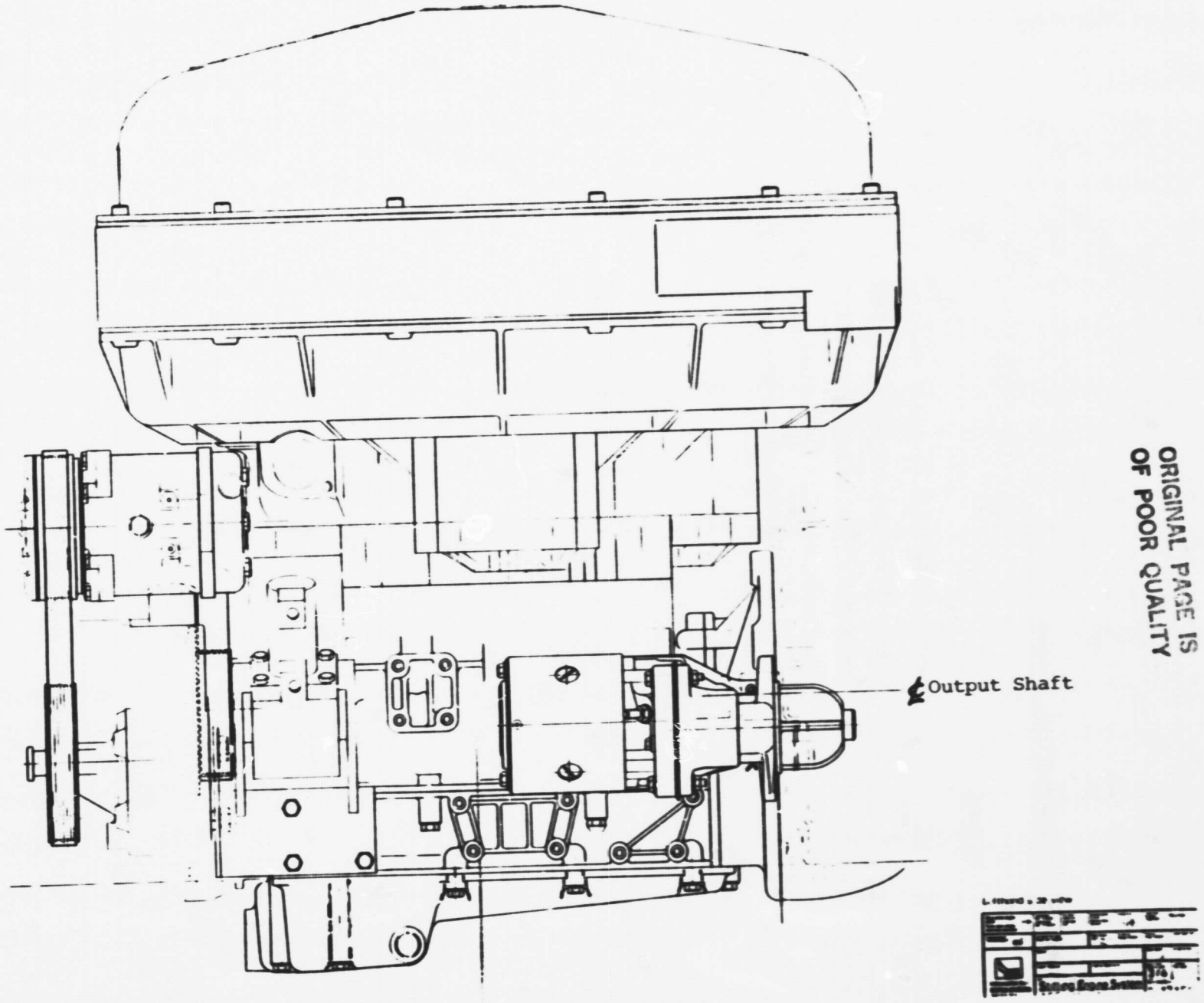


Fig. 2 SES (Right Side View)



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Fig. 3 SES (Left Side View)

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8.6.5 Mod I Mock-Up Photos

Figure

4	Front Side View of Mod I Mock-Up
5	Right Side View of Mod I Mock-Up
6	Left Side View of Mod I Mock-Up
7	Underside View of Installed Mock-Up
8	Front View of Engine Compartment with Mod I Mock-Up Installed

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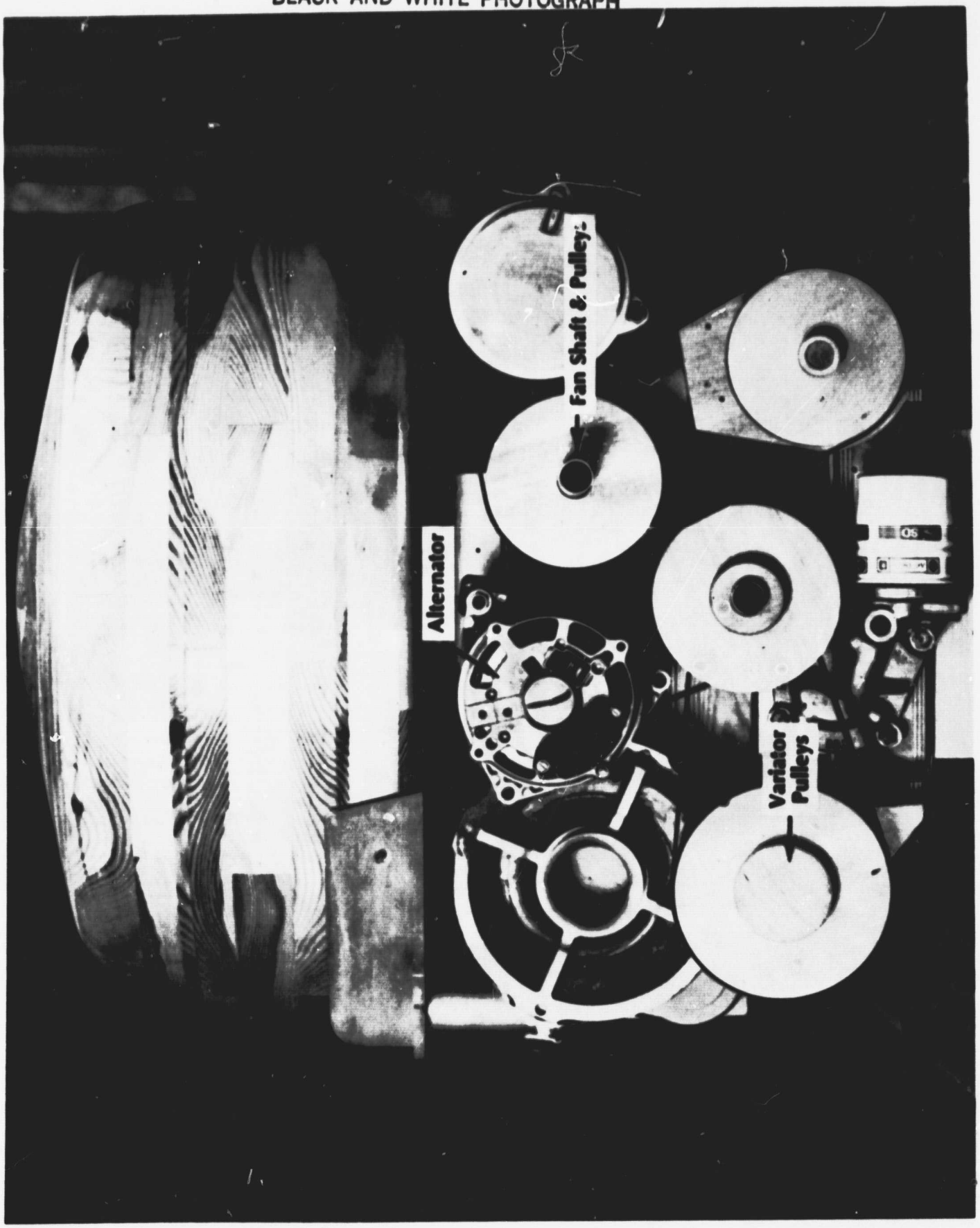


Fig. 4 Front View of Mod I Mock-Up

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Fig. 6 Left Side View of Mod I Mock-Up

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Fig. 7 Rear of Engine Compartment with Vol. Module Installed

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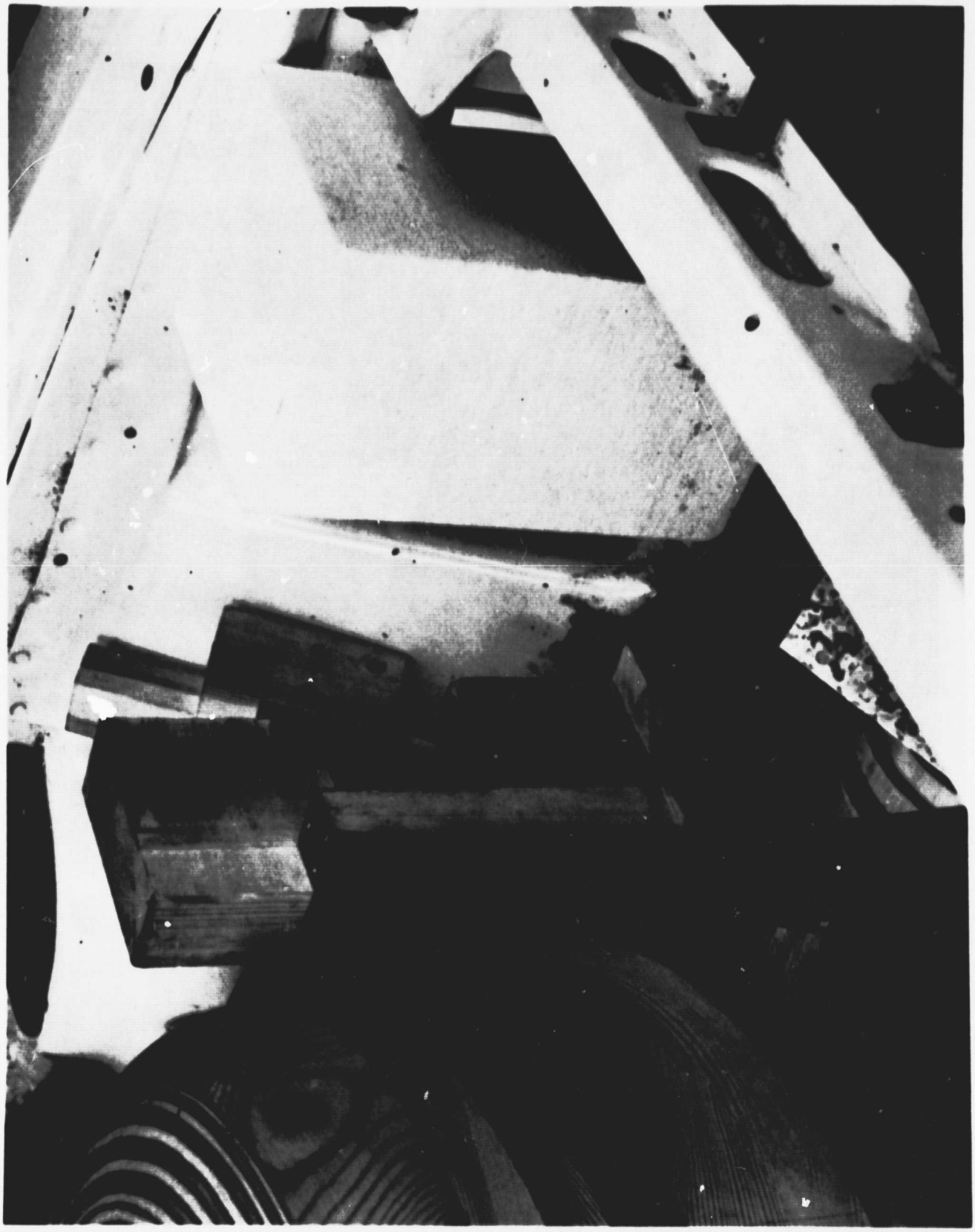


Fig. 8 Front of Engine Component with Mod I Mock-Up Installed

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Fig. 9 Power Control Block No. 3 Mock-Up

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Fig. 10 Left Front View of Engine Compartment with Mod I Mock-Up Installed

8.7 MOD I COOLING SYSTEM

Contents

8.7.1 Scope of Design
8.7.2 Design Approach
8.7.3 Objectives
8.7.4 Radiator Sizing and Positioning
8.7.5 Candidates for Radiator Core and Fan
8.7.6 Prediction of Cooling System Performance
8.7.7 Pressure Drop on Water Side
8.7.8 Influence of Antifreeze Additives to Cooling Water

8.7.1 Scope of Design

The development of a cooling system for the Spirit MOD I installations assumes the use of a fairly traditional automotive cooling system. The fan is assumed to be mechanically driven and the coolant is assumed to be water with glycol or ethanol added when needed for freeze protection.

The FWD MOD I version may be difficult to fit with a mechanically driven fan. Electrically or hydrostatically driven fans are possible alternatives. It is planned to evaluate these three types of drives as part of next RES update. Experimental data for a hydrostatic drive are being generated when testing the Opel, which is presently fitted with such a system.

8.7.2 Design Approach

The procedure when designing an automotive cooling system is still to a large extent based on trial and error. The main reason for this is that it is impossible to calculate the amount of air flow thru the radiator produced by ram air or by a fan with any degree of accuracy. Such data can only be obtained by measurements, which can either be performed on the road or in a wind tunnel. These measurements require theuse of a live engine or - in case a wind tunnel test is chosen - at least a reasonably exact mock-up of the engine installation.

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This fact has influenced the design approach to the cooling system. Basically a multi-step approach has been chosen.

* Define objectives of cooling system
* Based on objectives and rough calculations identify size and position of radiator and allocate sufficient space for a mechanically driven fan.
* Identify candidates for radiator core and fan.
* Manufacture radiator prototypes from core candidates and characterize these prototypes in a uniform way.
* Generate air flow data for the most likely combinations of radiator prototype and fan in a wind tunnel.
* Perform computer simulations of cooling system and vehicle performance and assess to what extent the cooling system meets the objectives.
* If necessary modify cooling system and repeat air flow tests and computer simulations until satisfactory performance has been achieved.

It is obvious that a large part of this effort has to be delayed until the design of the MOD I SES is frozen and a mock-up becomes available for air flow testing. Thus only the results of the first three steps can be reported here.

Because of the influence of cooling water temperature on engine efficiency the cooling system performance has an impact on vehicle performance and mileage. To support predictions of these a simplified computer analysis has been made, which will also be reported.

8.7.3 Objectives

The objective of a standard automotive cooling system is to minimize cost while allowing the vehicle to be subjected to certain duty cycles and ambient air temperatures without overheating the system. Secondary objectives such as minimum weight and fan power consumption are part of system engineering.

Because of the impact of cooling water temperature on engine efficiency the most important thing is to minimize cooling water temperature without excessive use of fan power. From previous experience at USSw it is judged that it should be possible to use ram air only at speeds above 20 mph. It is also judged that during normal driving without excessive vehicle loads a good cooling system should not allow top tank temperature to rise to more than 30°C (86°F) above ambient temperature. In practice city driving will likely give somewhat higher and highway driving somewhat lower temperature. Therefore the following main objective was established.

A. Design a cooling system, which during a combined EPA cycle gives an average top tank temperature not exceeding ambient temperature by more than 30 °C (86 °F), while cutting off the fan at all speeds above 20 mph.

   This objective is compatible with the standard conditions for which MOD I SES maps are given, i.e. ambient temperature 20 °C (68 °F) and cooling water temperature 50 °C (122 °F).

Of course the system must not overheat when subjected to high ambient temperatures and load. Thus a second objective has been established, which is more similar to the objective of the cooling system for an IC engine.

B. Top tank stabilized temperature must never exceed 90 °C (194 °F) while the vehicle is subjected to any of the following duty cycles at 100 °F ambient air temperature

1.	Wide open throttle	0 % grade
2.	60 mph	6 % grade
3.	30 mph	7 % grade
4.	Idle, 750 rpm, in gear, brakes on -- after test	No wind

8.7.4 Radiator Sizing and Positioning

To meet the objective A the radiator must capture as much ram air flow as possible without fan assistance and make the best use of it, i.e. transfer as much heat as possible to each air molecule passing by.

The heat rejection capability of a radiator can be approximated by the formula

$$\text{HEAT REJECTION} = K \cdot A \cdot \sqrt{Q/A}\,(T_w - T_a)$$

where K = Delta Heat Exchanger Coefficient
A = Heat Exchanger (Radiator core) Frontal Area
Q = Air flow
T_w = Coolant Inlet (Top Tank) Temperature
T_a = Ambient Air Temperature

The air flow causes a pressure drop across the radiator, which balances the pressure increase caused by the ram air effect. If - as a first approximation - we assume that the pressure increase caused by the ram air effect is not influenced by the type of radiator, and that all other pressure drops except for the drop across the radiator may be neglected, the best radiator core will be the one which maximizes

$$\text{CORE GOODNESS NUMBER} = K \cdot \sqrt{Q\,(\Delta P_{ref})/A}$$

where $Q\,(\Delta P_{ref})$ is the air flow measured for a given reference pressure drop ΔP_{ref} across the radiator core. It is evident that the best result is achieved if all the following measures are taken simultaneously.

A. Maximize total radiator core area
B. Find a core which maximizes the Core Goodness Number for a suitably chosen Reference Air Pressure Drop ΔP_{ref}
C. Minimize extra pressure drop and reduction of air flow caused by blockage before and after the radiator
D. Increase air pressure in front of the radiator by suitable sheet metal work in the front of the vehicle.

A and C will be treated in detail in the chapter on vehicle installation. The maximum core area has proved to be 835 mm x 575 mm (33" x 23"). Minimum blockage is achieved by pushing the engine as far back in the engine compartment as possible and the radiator as far forward as possible. Presently the distance between the radiator and the engine is approximately 130 mm (4 1/2") as measured to the edge of the closest pulley in the auxiliaries drive system. This is judged reasonably adequate to prevent significant blockage of ram air. As has been stated above C and D must be carried out in wind tunnel tests by trial and error before it is possible to state definitely whether satisfactory results have been achieved.

Unfortunately the distance between engine and radiator is insufficient to avoid a certain degradation of fan efficiency due to the necessity to divert the air flow thru sharp angles immediately behind the fan. It also causes restrictions on the choice of fan and limits the possibilities to use different types of shrouds.

It should be noted that the radiator core cannot be chosen independtly of the fan. To limit fan power during idling the core must still maintain a sufficiently high Core Goodness Number for a low Reference Pressure Drop ΔP_{ref}, chosen to be representative of the pressure drop which can be maintained by a fan without ram air assistance.

8.7.5 Candidates for Radiator Core and Fan

8.7.5.1 Radiator Candidates

Five different radiator core candidates have been identified. They are briefly described in Table 8.7:1. They are all expected to have higher Core Goodness Numbers than other candidates. All core candidates are procured in Sweden and the radiator prototypes made in Sweden, as it was not possible to find US vendors willing to offer suitable alternatives.

Manufacturer	Tube		Fins		
	Row(s)	Size mm	Spacing mm	No per "	Comments
Blackstone "tropik"	2	2 x 16	14.43	18	AEB 1.5 Blackstone Tropic
Gränges Metallverken	2	2 x 16	9	36	Splitter Fins without louvers (Lam-Rad)
" "	2	2 x 16	9	25	Splitter Fins with louvers
" "	1	2 x 16	9	25	Splitter Fins with louvers. Only for testing
Arco	2	2 x 16	8	12	Recessed copper sheets width 40

Table 8.7:1 Some Characteristic Data for Radiator Prototypes

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It should be noted that a large part of the radiator core material used in the automotive industry is supplied from Sweden.

The Splitter Fin with louvers is a new product from Gränges. The difference between this core material and their previous developed Splitter Fin without louvers (or Lam-Rad) is that it gives a significantly better Core Goodness Number during idling conditions. The penalty is a somewhat inferior Core Goodness Number at high vehicle speeds.

The single row Splitter Fin is included in the test program in order to allow calibration of the computer codes for a sufficiently wide range of data.

The "Tropic" is the most efficient radiator available as a standard product from Blackstone (Sweden). ARCO is a small company specialized and highly skilled in radiator production.

8.7.5.2 Fan Candidates

A large number of potentially usable fans exist, all with different requirement for mounting and shrouding. The overall arrangement can be categorized as belonging to one of several groups. Some of these are:

1. Stiff fan mounted on engine with shroud mounted on the radiator covering a rectangular part of the core extending from top to bottom.
   Example: ASE P40 Opel.

2. Flexible fan mounted on engine and without a shroud.
   Example: ASE P40 Spirit (Figure 8.7:1)

3. Orificed rotating ring fan mounted to vehicle body, with or without shroud.
   Example: Electric fan for GM X-car.

4. Stiff fan with close clearance ring-shaped tip shroud both mounted on radiator.
   Example: USSw Ford Taunus installation.

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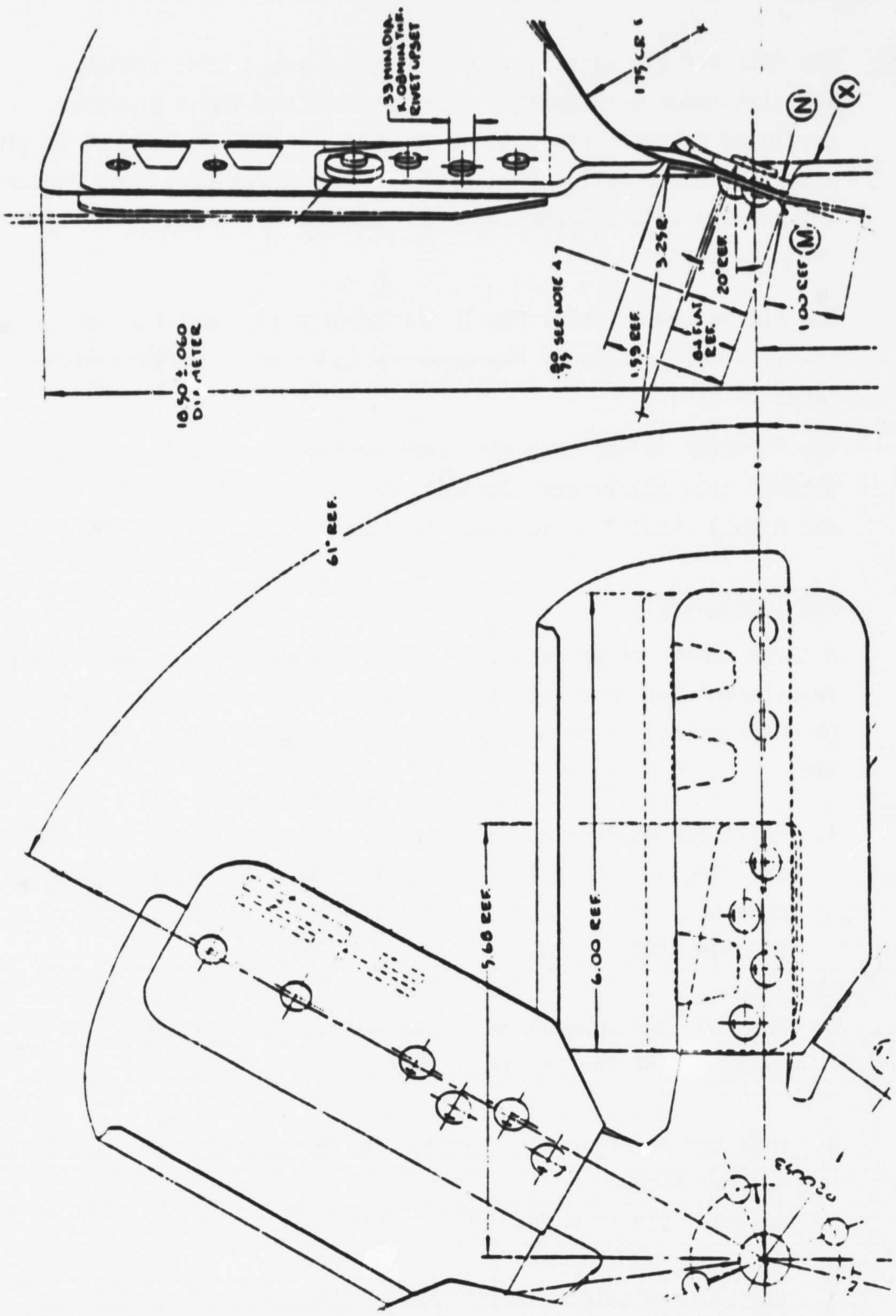


Fig. 8.7:1 Flexible Fan

In the Stirling application it is very important that the fan arrangement does not introdue added blockage to the fan. Large shrouds as in Candidate 1 should therefore be avoided. Candidates 2 and 3 without a shroud, or 4 are therefore preferred solutions. From an efficiency point of Candidate 4 is the best solution. The short distance between the engine and the radiator may in this case preclude its usage for two reasons. The ring shroud may introduce some blockage effect to ram air. The inclusion of two joints in the fan drive shaft requires a minimum length of the shaft. The second problem may possibly be handled by retracting one joint into the fan hub.

8.7.5.3 First Choice of Radiator and Fan Spirit MOD I Installation

The Lam-Rad radiator which is installed in the Opel is identical to the first of the Gränges prototypes except for frontal area. Because this type of radiator has been extensively tested in the Opel cooling tests, it will be used for the performance predictions in next section. However, the Gränges two row radiator having Splitter Fin core with louvers is the preferred choice for the Spirit MOD I installation. As mentioned previously this radiator has a lower pressure drop at low air speeds, which makes it easier to find a fan which does not cause unacceptable ram air blockage.

The preferred fan is the flexible fan shown in figure 8.7:1. It causes minimal blockage of ram air and allows the use of a simple on/off clutch without absorbing unreasonably high power at high engine speeds. Its major drawback is its poor efficiency, but this should have little influence on mileage, as the system is designed to work without fan assistance during a major part of the combined cycle.

The second alternative is a type 4 fan arrangement. This requires the on/off clutch to be exchanged for a variable slip coupling. An electrically controlled coupling, which allows the amount of slip to vary as a function of both top tank temperature and engine speed is recommended.

8.7.5.4 Fan Drive Strategy

The final drive strategy must be elaborated by testing. A tentative strategy is as follows:

fan on, if top tank temperature greater than 80 °C (176 °F)
fan off, if top tank temperature less than 80 °C and
vehicle speed is greater than 20 mph
fan on, if vehicle speed is less than 20 mph and
accelerator not in kick-down position
fan off, if accelerator is in kick-down position and
top tank temperature less than 80 °C

The accelerator control of the fan is intended to improve performance in the low speed region.

8.7.6 Prediction of Cooling System Performance

8.7.6.1 Ram Air Flow without Fan Assistance

In absence of other suitable data the performance of MOD I Spirit cooling system is scaled from the Opel cooling system testing, assuming that the Lam-Rad core is chosen. This means that the Core Goodness Number is identical for both installations. If it is assumed that the pressure drop profiles are comparable at all speeds, this means that the radiator heat rejection is proportional to Radiator Core Frontal Area A only as shown in Table 8.7:2.

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Vehicle	Width mm	Height mm	Area A m^2	Area Relative to Opel	Area Relative to P-40 Spirit
Opel	830	510	.423	1.00	1.31
P-40 Spirit	705	457	.322	.76	1.00
Mod I Spirit	835	575	.480	1.13	1.49

Table 8.7:2 Radiator Core Frontal Areas

In Figure 8.7:5 air mass flow data for the P-40 Spirit (as obtained from flow measurements on Concord with P-40 mock-up) has been compared with Opel data. Mod I Spirit air mass flow has been projected from both data sets using Radiator Core Frontal Area scaling factors from Table 8.7:2. This is in a strict sense not allowed, because the Opel has a different air pressure profile than the Mod I Spirit will have, and the P-40 Spirit has neither the same Core Goodness Number nor the same pressure profile as the Mod I Spirit will have. It is done in the absence of experimental data for the Spirit Mod I installation. The most conservative of the two curves is used in the projections, implying a scaling factor of 1.13.

8.7.6.2 Fan Air Flow with no Ram Air

The projection of fan performance is based on fan characterization data obtained from the manufacturer as shown in figure 8.7:6. In the same diagram is shown a pressure drop curve for the fan working together with the Lam-Rad core. This curve has been obtained by calculating the pressure drop across the radiator core by means of the USSw cooling system code and multiply this by an estimated factor of 1.5. This factor accounts for the sum of pressure drops in front of core between core and fan and after fan. The pressure drop factor is very uncertain, but it should be noted that because of the shape of the fan curves, pressure drop increases do not produce large changes in air flow.

Together with heat rejection data obtained by simulating MOD I SES and Opel cooling system using USSw computer code it is now possible to calculate the numbers given in Table 8.7:3, which projects the cooling system performance during idling conditions. As will be shown later this heat rejection capacity corresponds to a scaling factor of approximately .8 when comparing with the Opel fan installation.

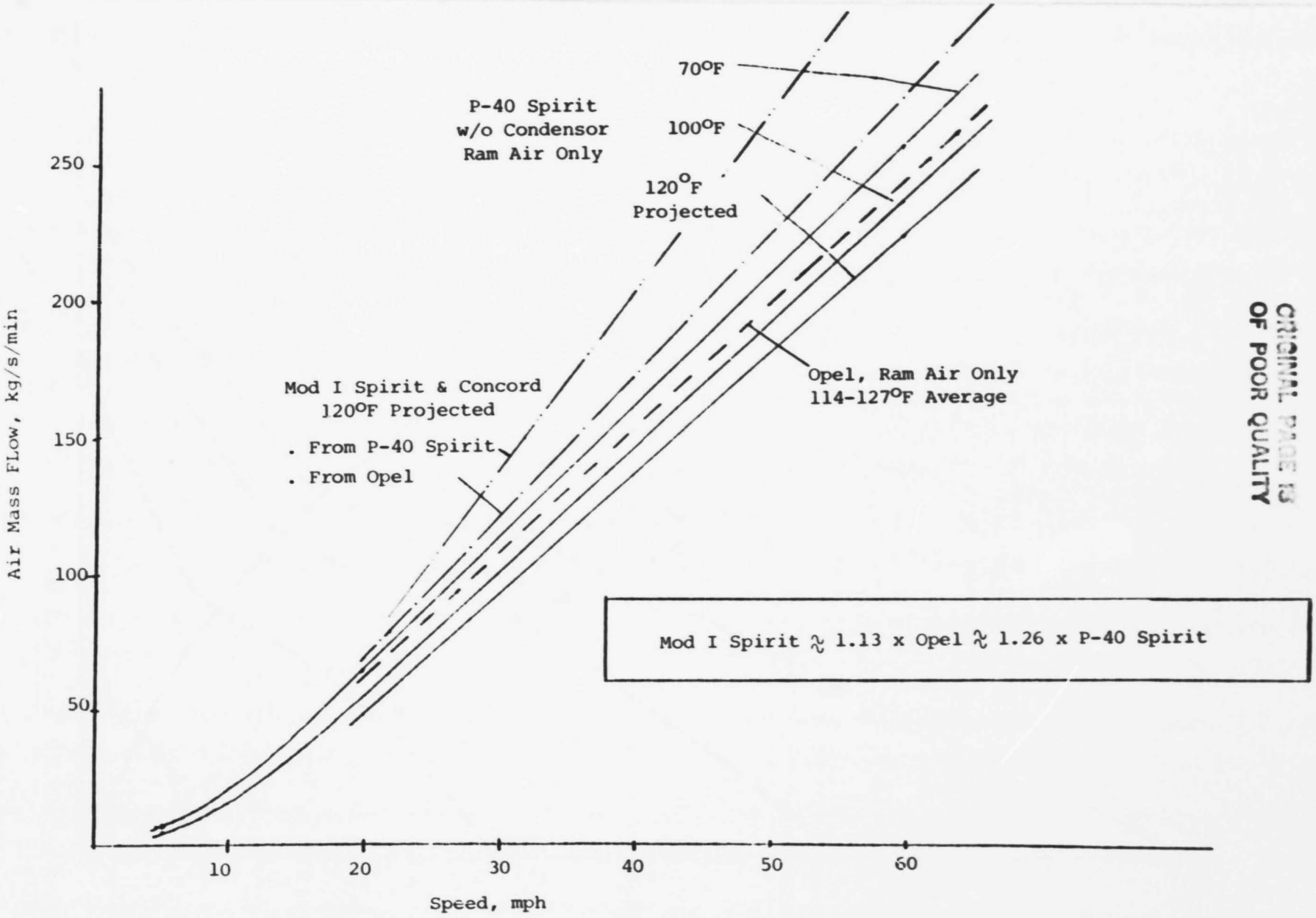


Figure 8.7:5 Comparison of Air Mass Flows with Ram Air Only

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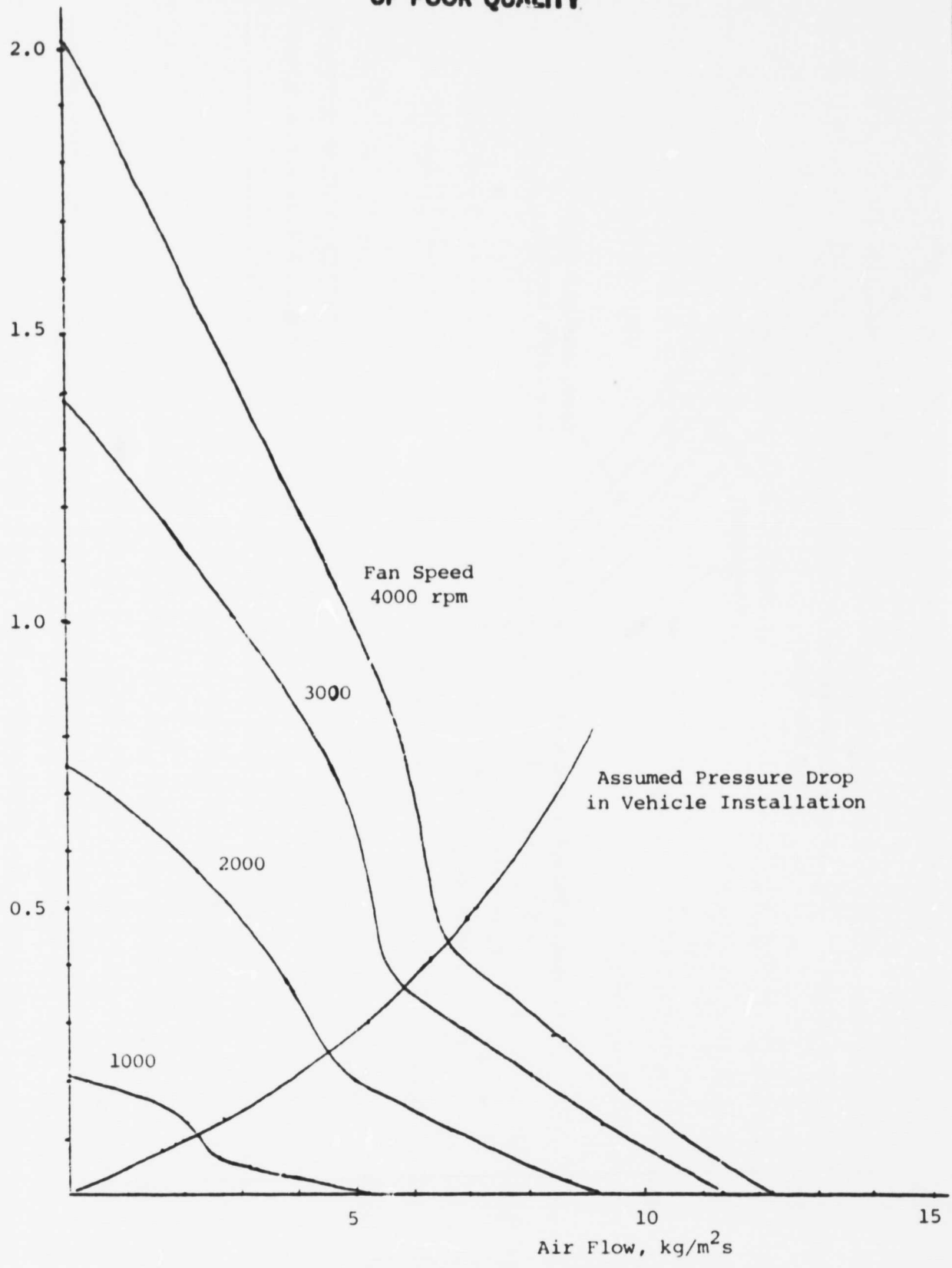


Figure 8.7:6 Pressure Drop Curves for Flexible Fan

Engine speed rpm	Fan speed rpm	Fan power kW	Pressure drop Pa	Air flow $\frac{kg}{m^2 s}$	Air flow $\frac{kg}{s}$	Heat rejection kW	Air temp °C	Water temp °C
600	720	0.12	75	1.6	0.33	7.1	59.2	61
700	840	0.15	90	1.85	0.38	8.2	59.3	61
750	900	0.18	95	2.0	0.42	8.85	58.8	61
1000	1200	0.3	130	2.7	0.56			
2000	2400	1.5	300	5.2	1.08			
3000	3600	2.8	410	6.3	1.31			
4000	4800	3.8	480	7.0	1.46			

Table 8.7:3 MOD I cooling system performance during idling in gear with brakes on and at other engine speeds without ram air. Ambient temperature 100 °F (37.8 °C).

8.7.6.3 Ram Air and Fan Air

Air flow test data for the Opel and the Omega show that the fan produces a constant or with vehicle speed slowly increasing addition of flow to the pure ram air flow.

Typical values are .9 kg/s (120 pounds/minute) for the Opel and 0.35 kg/s (48 pounds/minute) for the Omega fan. After comparing with figure 8.7:6 the assumption is made that the flexible fan produces .45 kg/s (60 pounds/minute) additional air flow on top of the ram air.

The experimental air flow data for the Opel may now be compared with the projections for the MOD I Spirit. Figure 8.7:7 gives a scaling factor of .9.

8.7.6.4 Top Tank Temperature Predictions and Conclusion

Using the scaling factors computed above and applying the rule of three on the Opel test data gives the final results as shown in Table 8.7:4. The Opel cooling system has been modelled into the vehicle simulation in order to produce MOD I heat rejection at approximately correct cooling water temperature levels.

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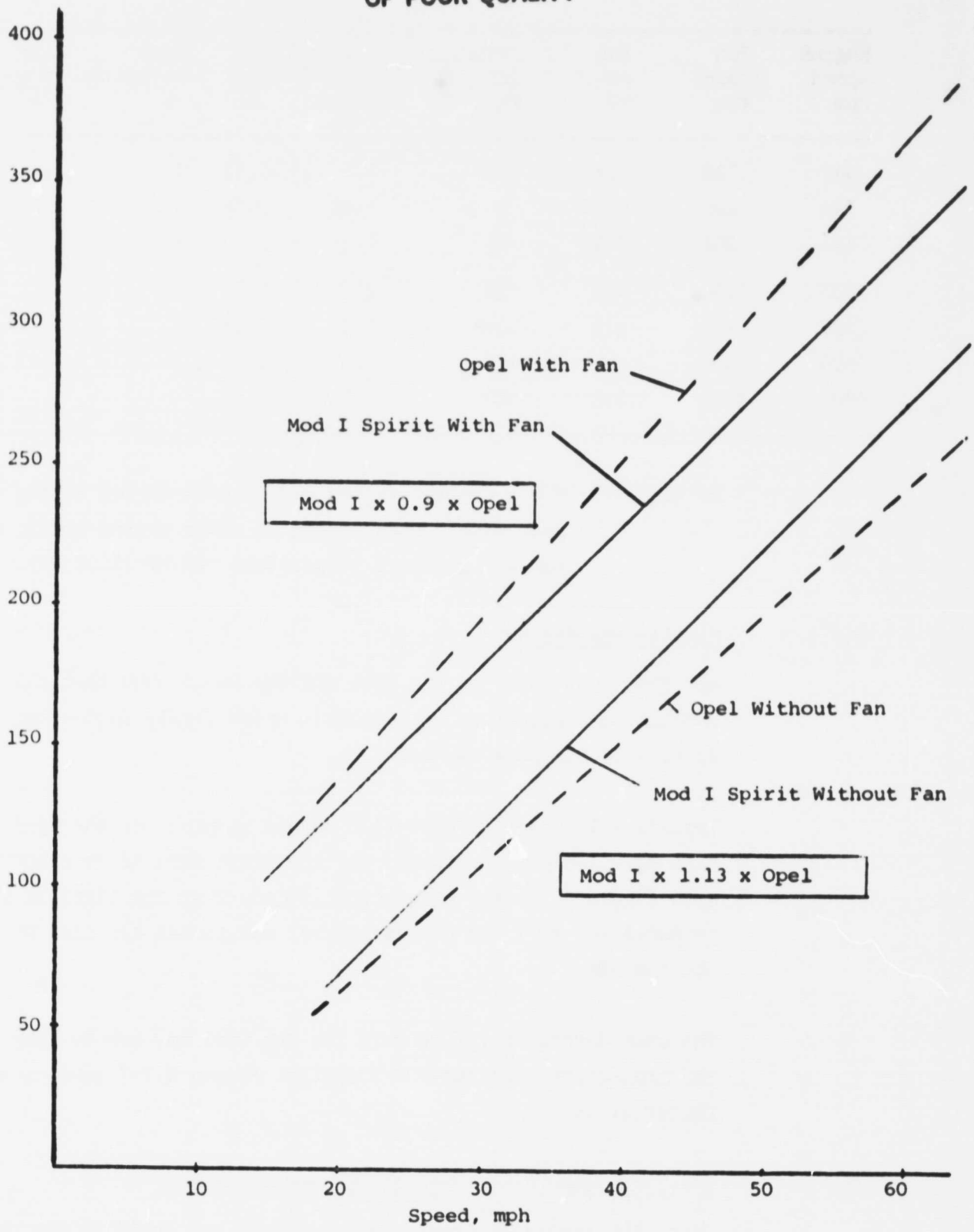


Figure 8.7:7 Comparison of Air Mass Flows with and Without Fan in Operation

Speed mph	Opel Test Data rpm	Btu/min	T_w-T_F, °F	Scaling Factor	Mod I Projected Data: Heat Rejection kW	Btu/min	T_w-T_A °F	°C	T_w at 100°F Ambient °F	°C	Engine rpm
RAM AIR ONLY, ROAD LOAD, THIRD GEAR											
23	1400	857	57.2	1.13	9.9	563	33.1	18.4	133	56	920
33	1720	1286	50.0	1.13	12.9	734	25.2	14.0	125	52	1120
44	2600	1801	53.6	1.13	19.4	1103	29.1	16.1	129	54	1640
55	3000	2990	68.5	1.13	28.7	1632	33.1	18.4	133	56	2050
64	3400	4313	93.2	1.13	40.4	2297	43.9	24.4	144	62	2380
RAM AIR + FAN AIR, MAXIMUM POWER, THIRD GEAR											
33	1900	3544	55.2	.90	111	6312	109.2	60.7	209	99	4000
44	2300	4188	55.7	.90	111	6312	93.3	51.8	193	89	4000
54	2600	5255	53.3	.90	111	6312	71.1	39.5	171	77	4000
65	3300	6011	56.0	.90	111	6312	65.3	36.3	165	74	4000
RAM AIR + FAN AIR, 7% GRADE, SECOND GEAR											
30	1750				39.3	2235	44.5	24.7	145	62	1753
RAM AIR + FAN AIR, 6% GRADE, SECOND GEAR											
50	2420				76.7	4362	55.3	30.7	155	68	2800
IDLE, IN GEAR, BRAKES ON, NO WIND (FAN AIR ONLY)											
0	700	654	63.6	.80	8.2	466	56.7	31.5	157	69	700

Table 8.7:4 Top Tank Temperature Predictions at 100°F Ambient Temperature

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The conclusion is that the proposed system meets all the objectives. The reason that no data are given for the 60 mph, 6% grade case is that the vehicle cannot achieve this with a rear axle gear ratio of 2.7:1, which was used during the calculations.

In the analysis the assumptions about air flows are most critical. After the MOD I SES design has been frozen these assumptions should therefore be checked by separate air flow measurements using a fan powered by an electric motor and a mock-up of the MOD I SES.

8.7.7 Pressure Drop on Water Side

Experimental data for the pressure drop across the radiator exist for the Opel. These data were obtained before the radiator height was diminished by approximately 27 mm from 537 to 510 mm, involving elimination of three layers of water tubes.

For this type of core with horizontal tubes the pressure drop Δp is related to the size of the radiator by the following formula

$$\Delta p_2 = \frac{w_1}{w_2} \cdot \frac{h_2}{h_1} \cdot \Delta p_1$$

when w = width of radiator core or tube length
h = height of core or number of tubes per row multiplied by the tube spacing
1,2 = indices referring to two different radiators

Using this formula and the Opel data the pressure drop for Spirit MOD I installation has been projected according to figure 8.7:8.

Figure 8.3:5 can now be modified as in figure 8.7:9 to include projected pressure accross the radiator instead of allowable pressure drop. At maximum engine speed the water flow will be 4.7 kg/s, giving a certain margin to the 4.23 kg/s, which is the design water flow.

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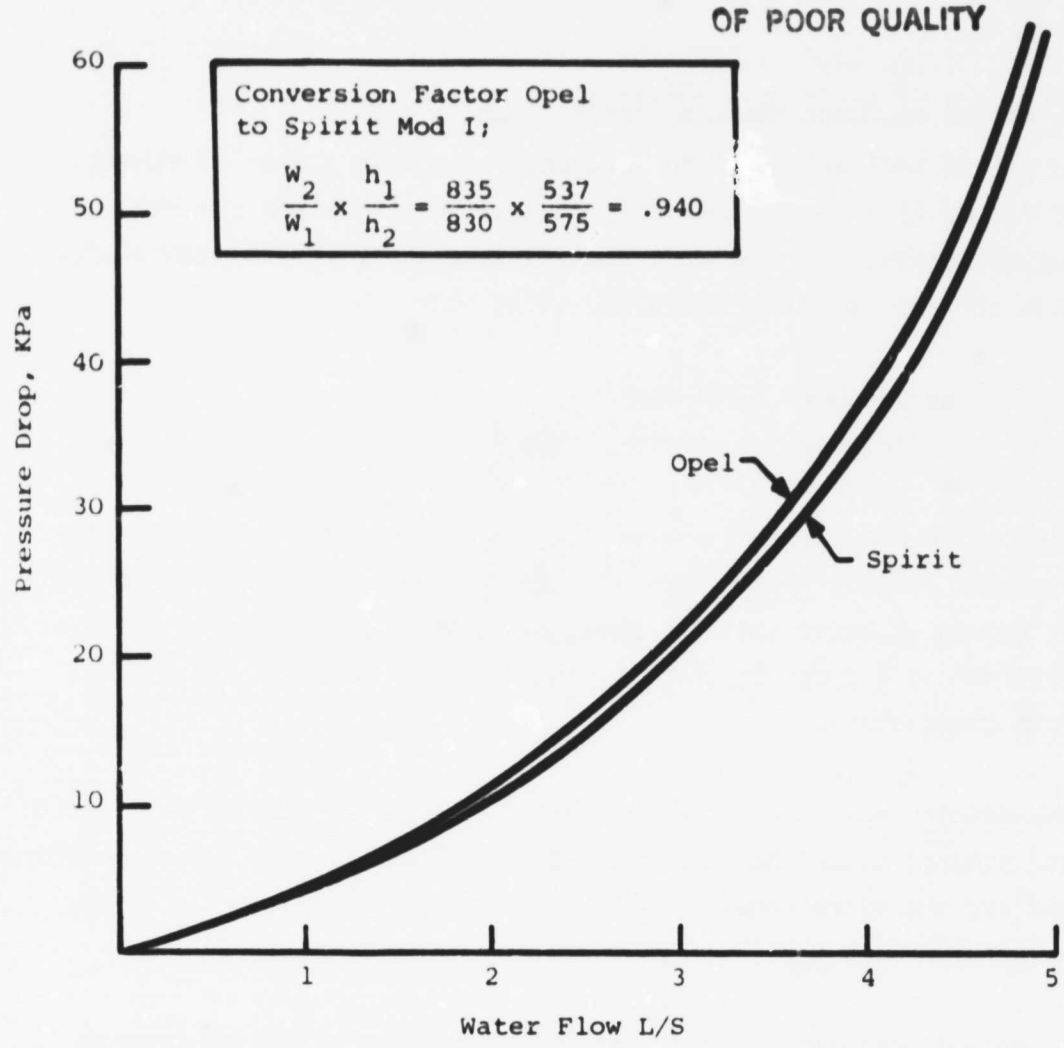


Figure 8.7:8 Pressure Drop on Water Side Across Radiator

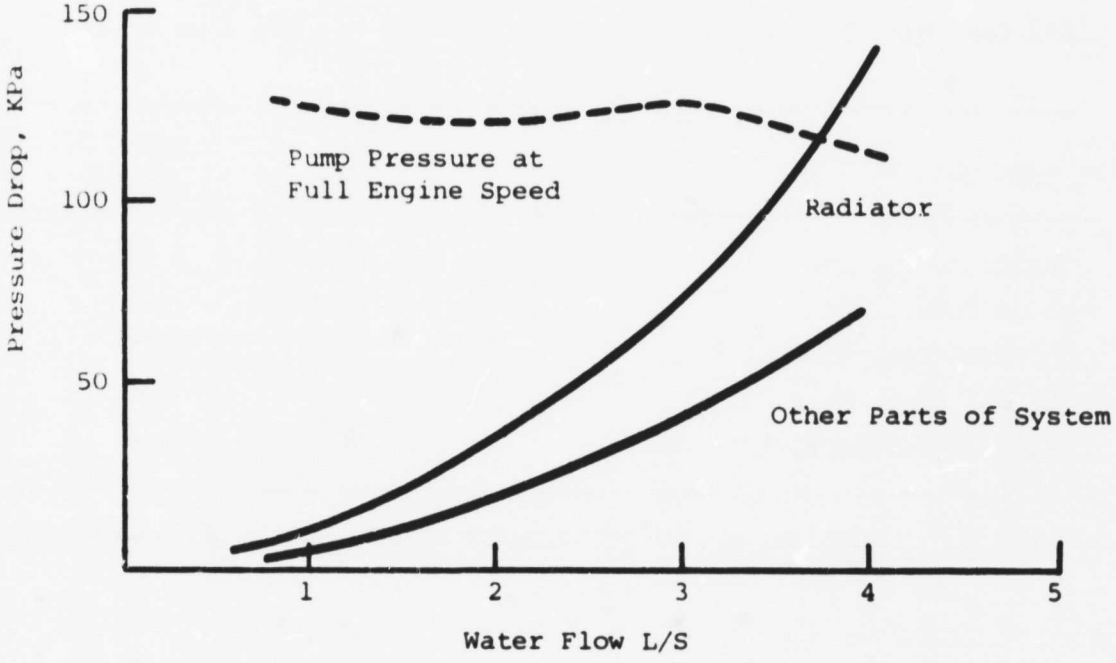


Figure 8.7:9 Pressure Drop and Flow in Water Circuit

8.7.8 Influence of Antifreeze Additives to Cooling Water

To get an indication of the influence of adding glycol or ethanol to the cooling water, some calculations were made for the P40 engine working with the Opel cooling system. The calculations were made for the following operating point only

Engine speed 2 000 rpm
Engine pressure approx 10 MPa

The engine pressure was varied slightly to meet the criterion of constant coolant volume flow and engine shaft power for all coolants. It should be noted that the auxiliary power consumption is not relevant to a specific installation but this does not affect the main conclusions.

The results are summarized in Table 8.7:5. It is evident that glycol and ethanol cause the same degradation of the cooling system performance and are therefore equal from that point of view. Because of other advantages with glycol it is therefore the preferred additive.

A coolant containing 50% glycol raises cooling water temperature about 12 °C at the chosen operating point and reduces efficiency by .7 points or by 2.5%. Although the calculations were made for the P40 they may be regarded as representative of the MOD I as well.

Quantity	Pure water	50% glycol	50% ethanol
Engine speed, rpm	2 000	2 000	2 000
Shaft power, kW	16.94	16.94	16.94
Top tank temperature, °C	56.6	65.4	66.2
Coolant flow, kg/s	1.80	1.90	1.65
Engine efficiency, %	28.53	27.80	27.80

Table 8.7:5 Cooling system performance at constant shaft power output.

B.9 VEHICLE SYSTEM

9.1 INSTALLATION, MOD I ENGINE INTO 1981 SPIRIT

The Mod I engine is located on the center line of the vehicle. The reference dimensions of the engine in the fore-aft and vertical positions are illustrated in Figure 1. The angular relationship of the engine with respect to ground, as shown in Figure 1, is 2.812°. The reference point on the vehicle is the center of a 1.00" master gage hole located directly in both the right-hand and left-hand frame rails. The holes are directly opposite each other and are easily located in the vehicle. The gage hole is shown on the engine installation layout and is located with respect to the layout 5" lines. The reference point on the engine is the intersection point between the rear face of block and the center line of the output shaft. This intersection point is 2.67" (67.82 mm) below the master gage hole and 15.61" (396.49 mm) rearward of the master gage. These dimensions plus their location with respect to the layout inch-lines are all shown in Figure 1.

STEERING SYSTEM

Significant modifications were required to accommodate the Mod I engine. The major changes involved: lowering the steering gear, moving the cross link forward of the oil pan, and shortening the tie-rods. These modifications resulted in changes in the Ackermann curves and toe-in, toe-out characteristics.

ACKERMANN

For comparison purposes the printout on the production Ackermann can be compared to the Mod I vehicle. Program data for both vehicles is as follows:

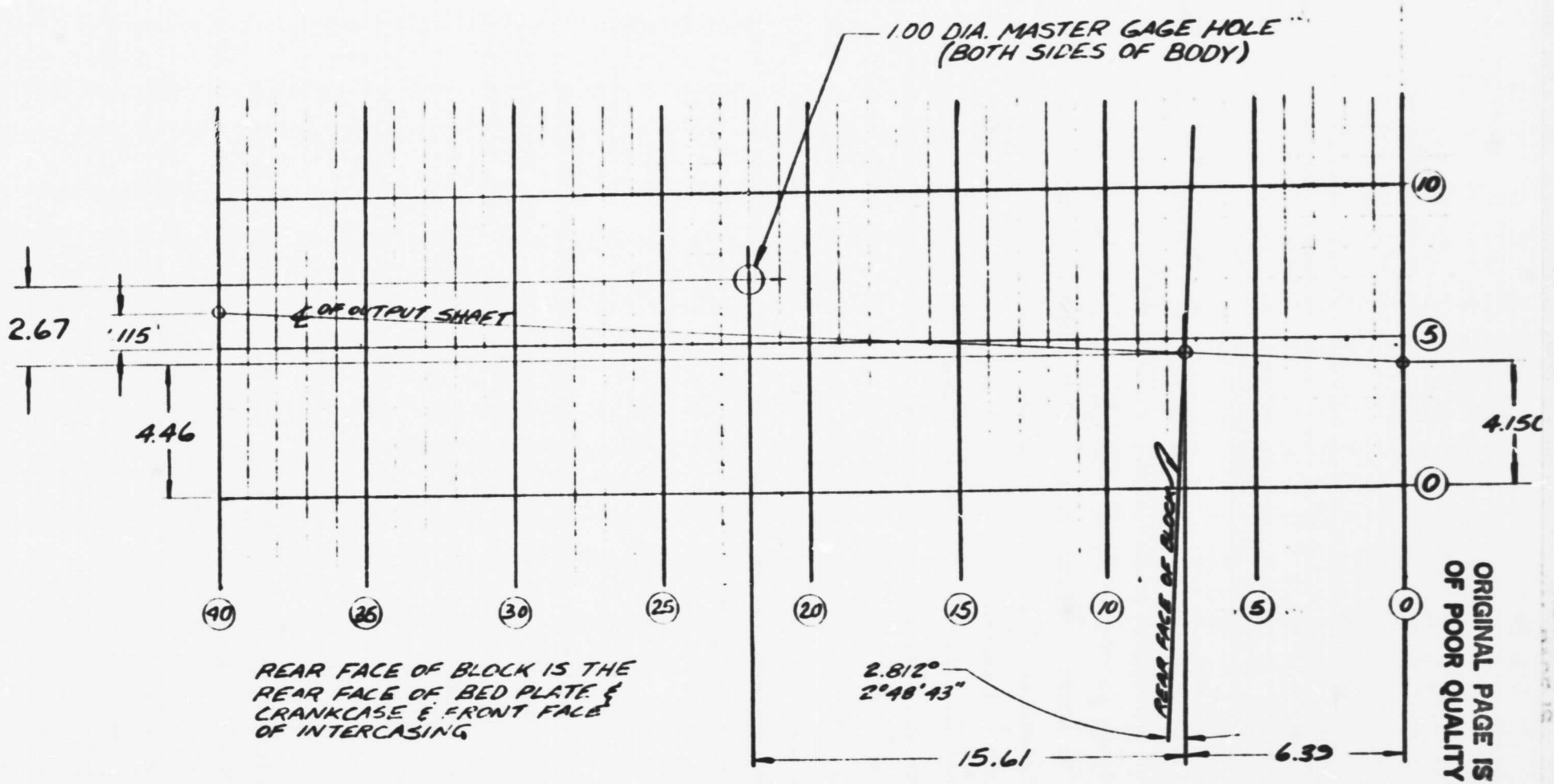


Fig. 1 Mod I Engine Location During 1980-81 Spirit)

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Fig. 2 Front View of Installed Mod I Mock-up

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Fig. 2a Left Front View of Engine Compartment with Mod I Mock-Up Installed

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Fig. 3 Front of Engine Component with Mod I Mock-up Installed

Fig. 4 Front of Engine Components with Weld Repair

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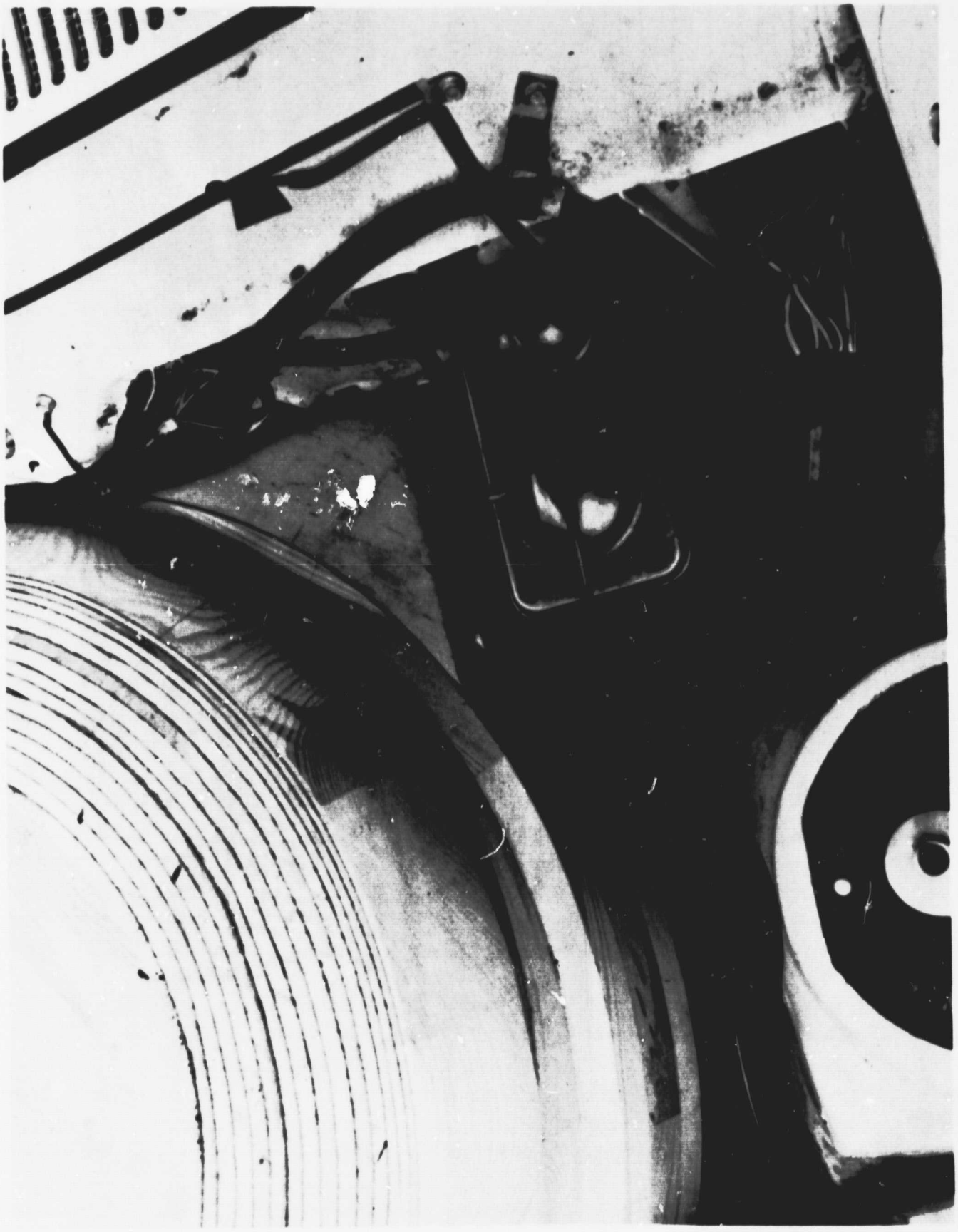


Fig. 5 Rear of Engine Compartment with Mod I Mock-up Installed

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Fig. 5 Rear of Engine Compartment with Mod I Mock-up Installed

Fig. 6 Power Control Block No. 3 Mock-Up

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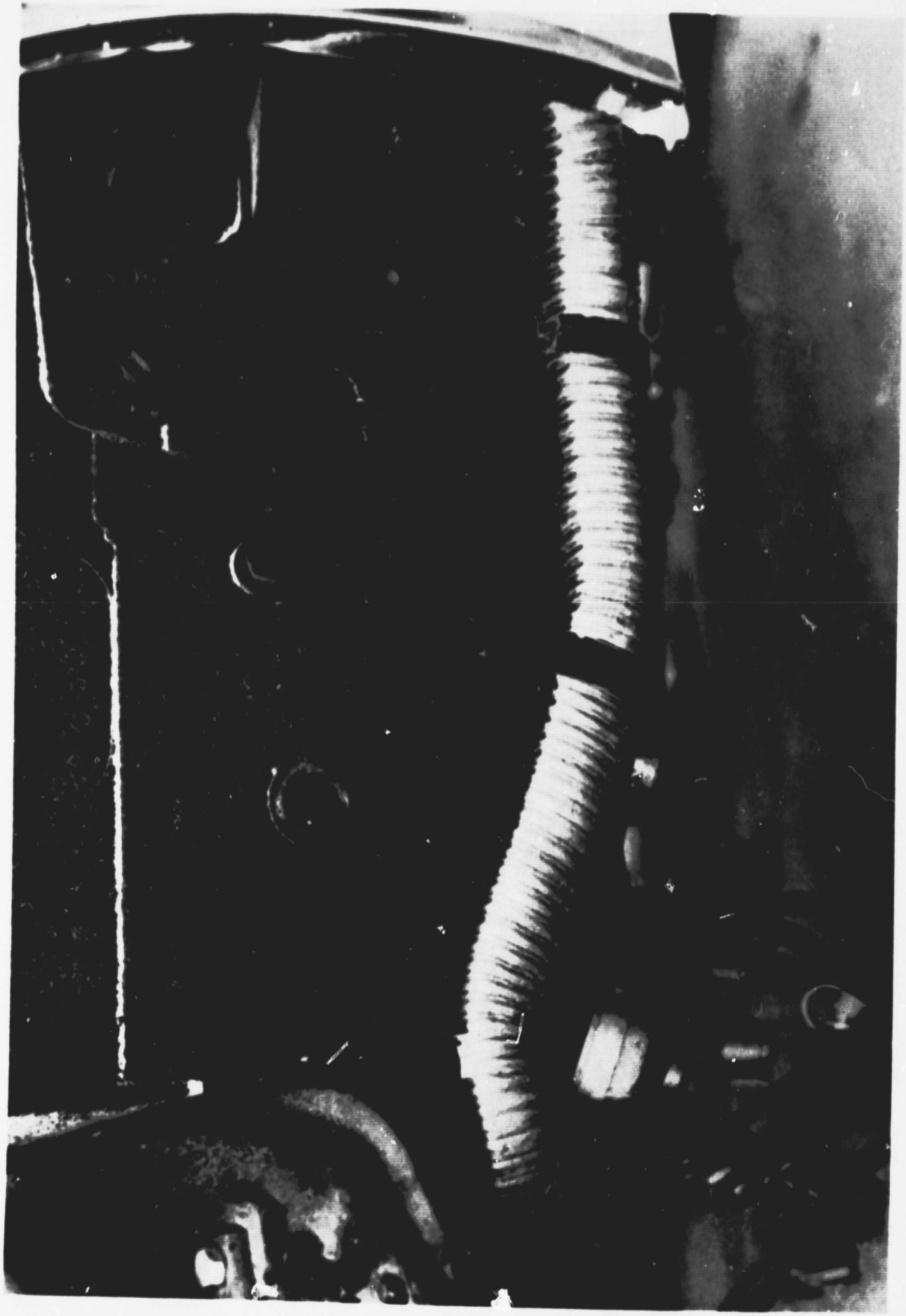


Fig. 7 Vehicle Rebound

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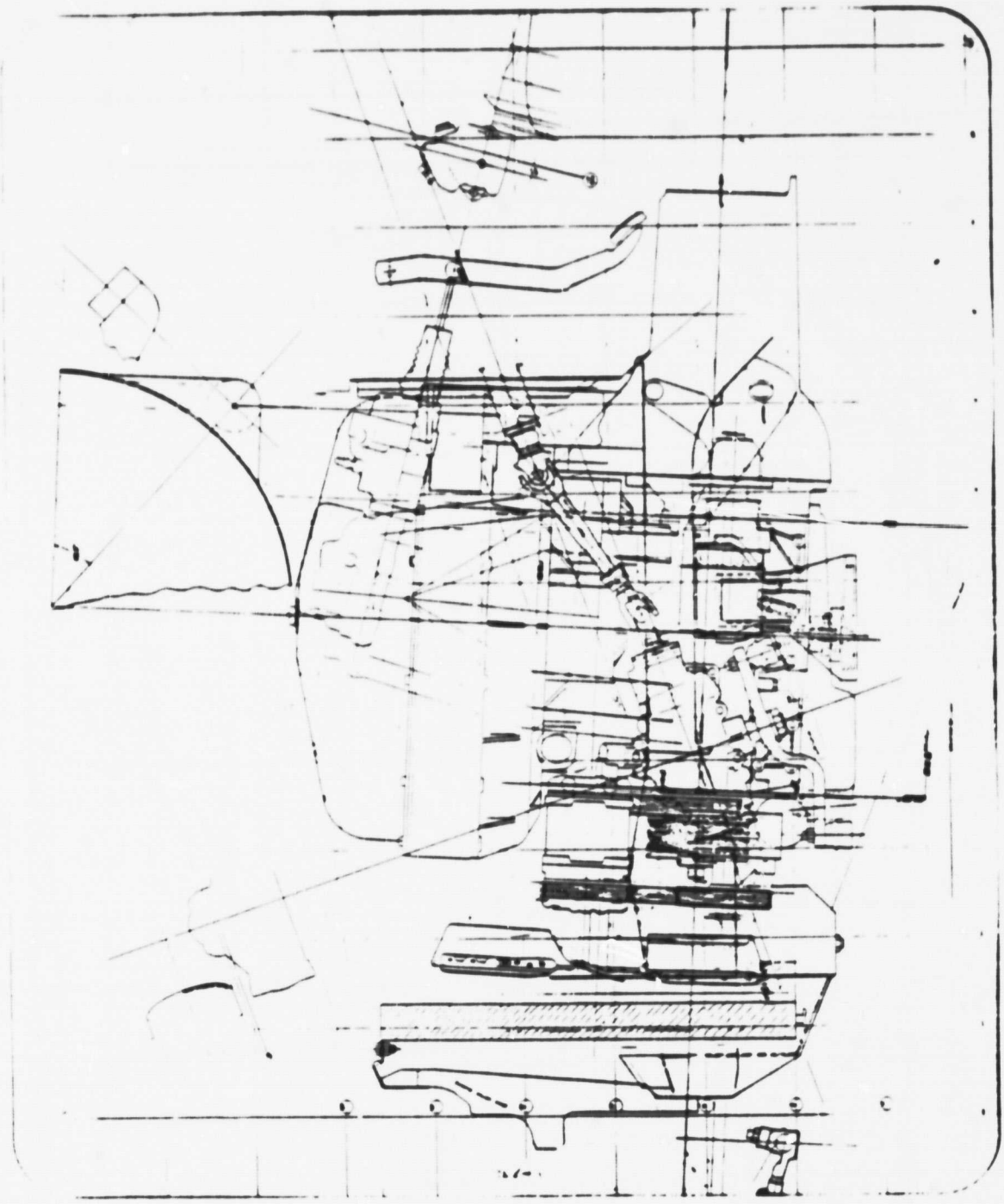


Fig. 8 Left Side View of Engine Installation

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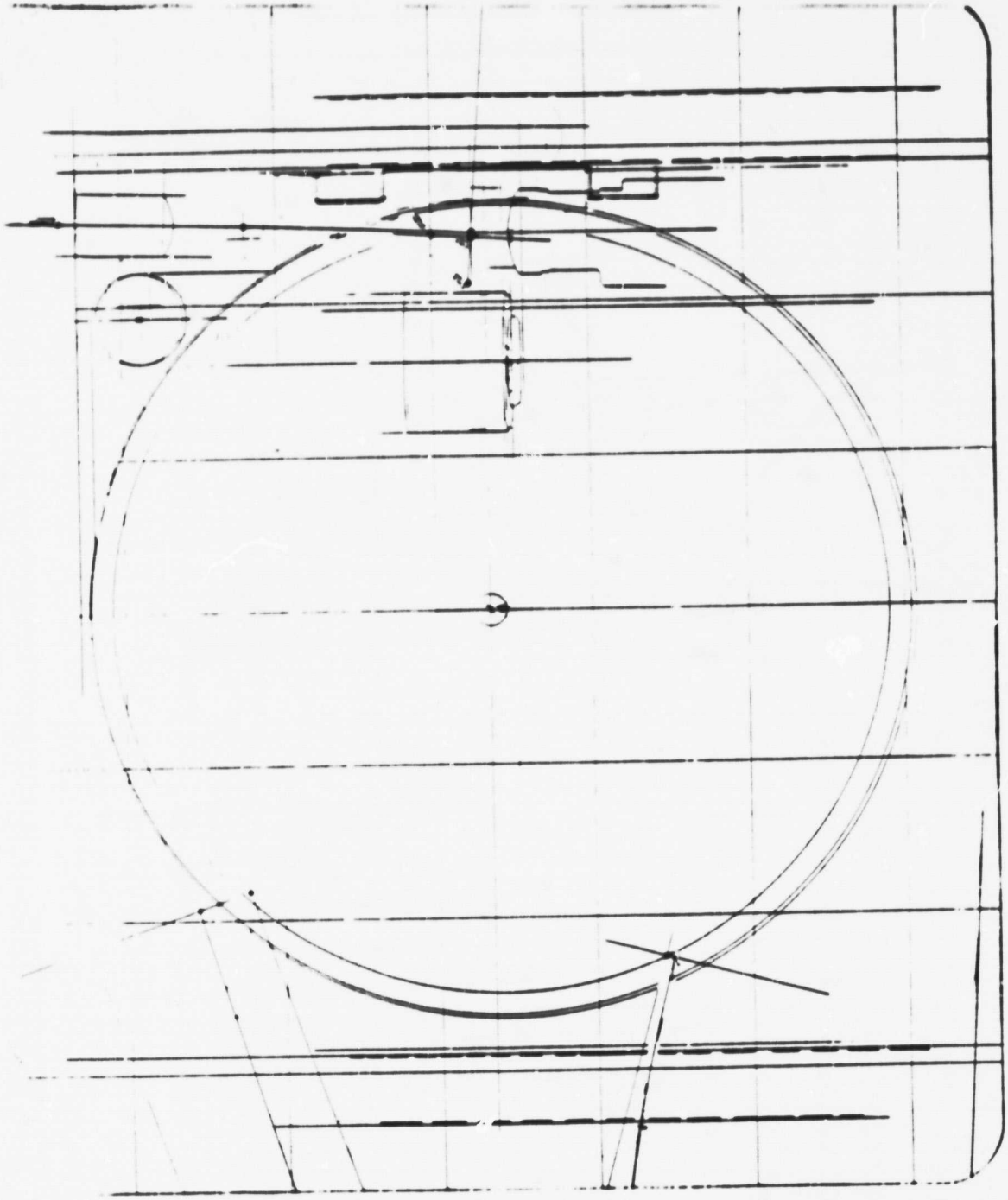


Fig. 9 Plan View of Engine Installation

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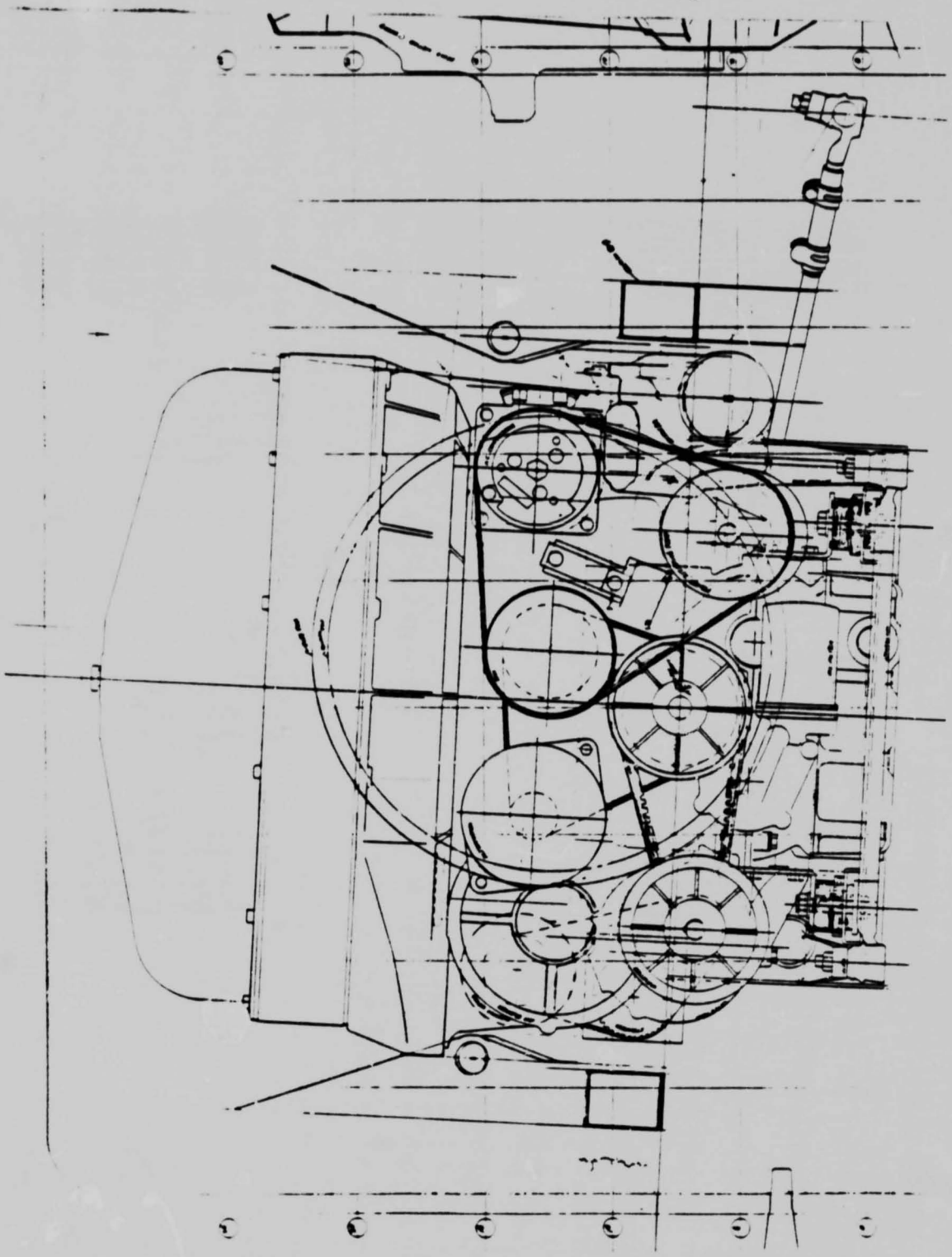


Fig. 10 Front View of Engine Installation

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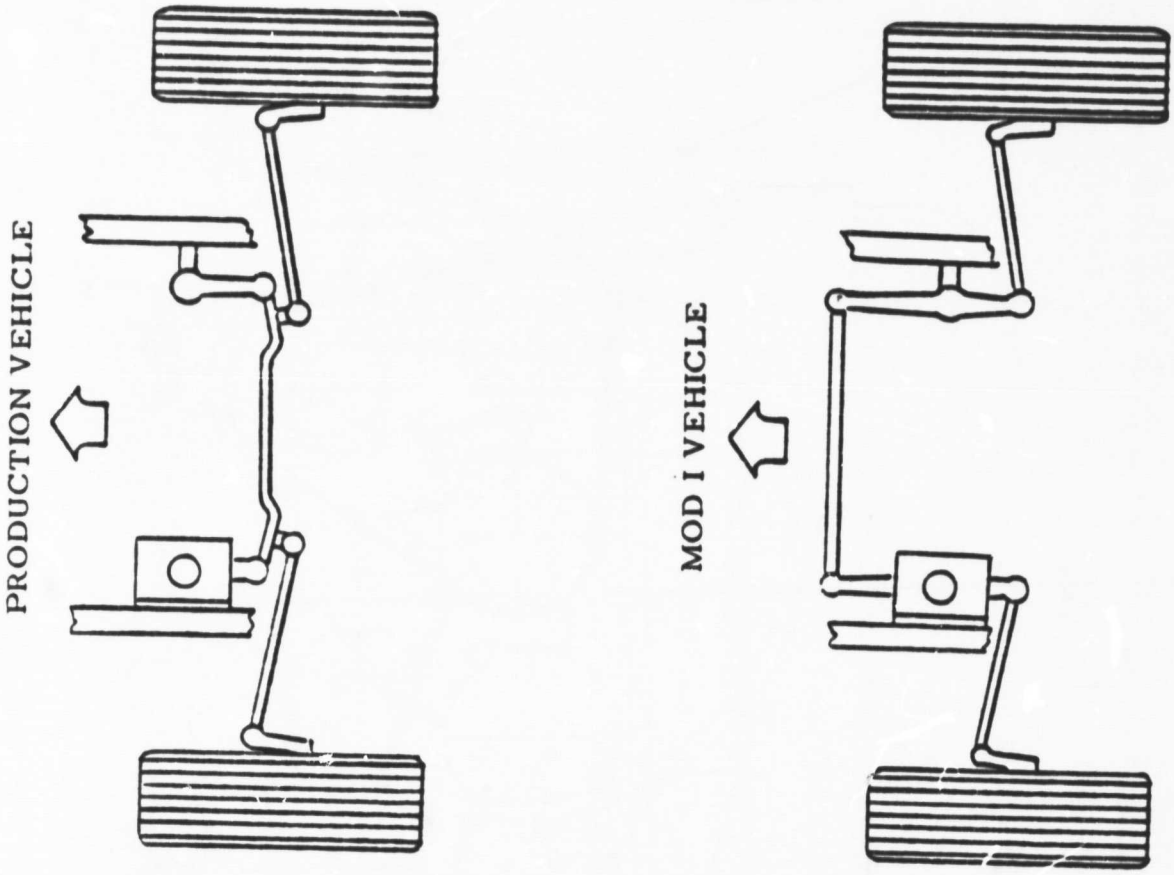


Fig. 11 Mod I Vehicle Steering Linkage Modifications

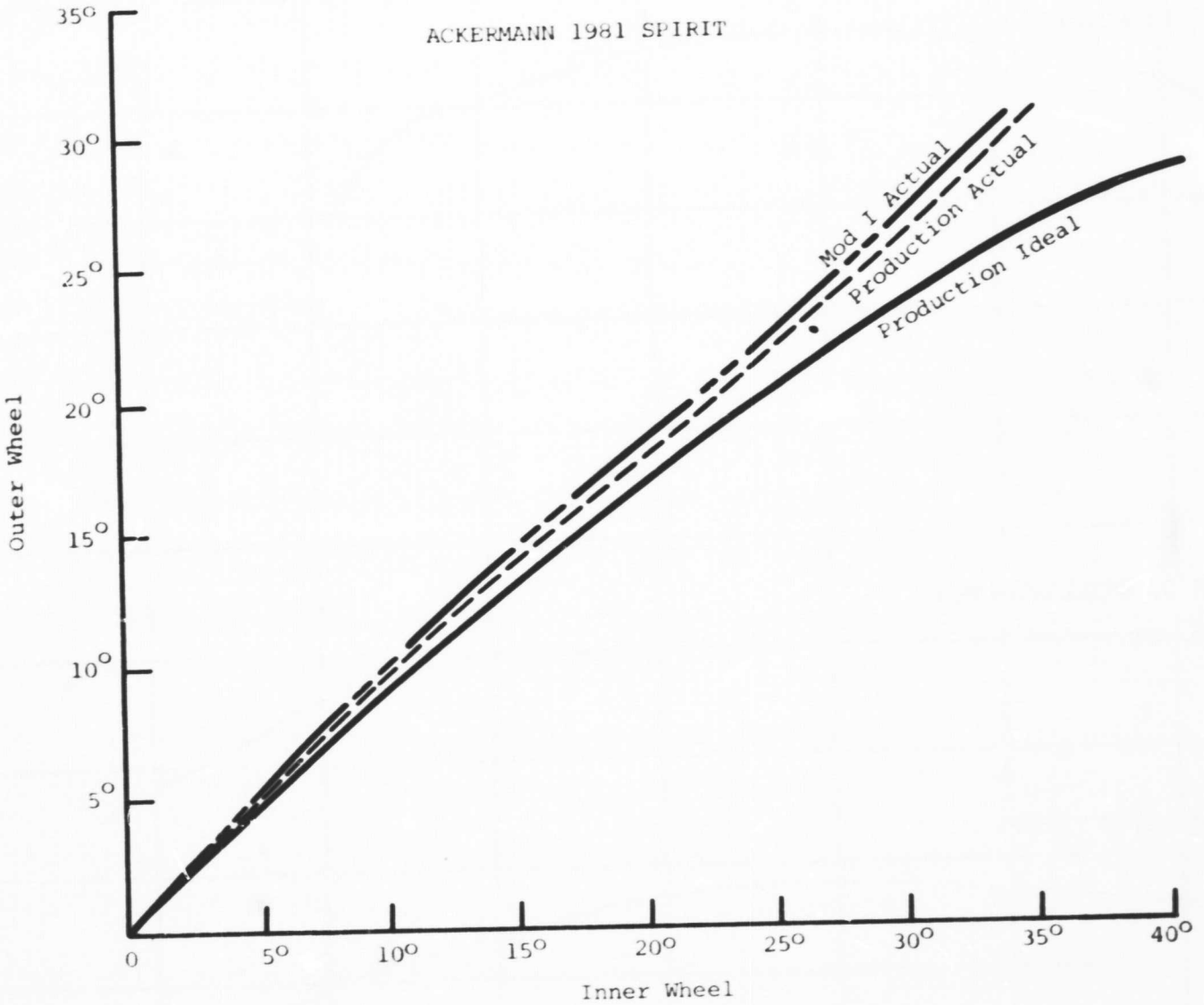
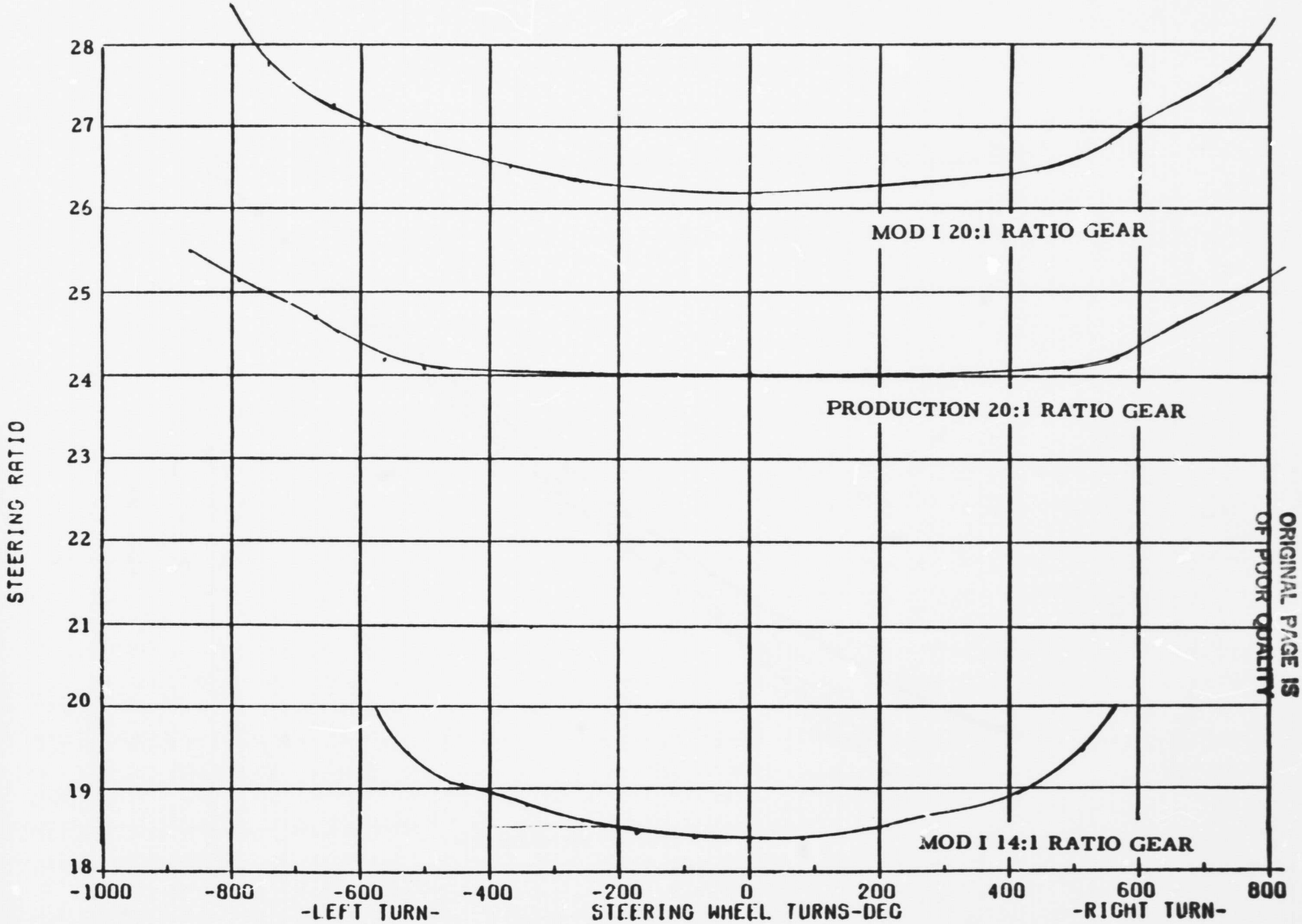


Fig. 12 Comparison of Vehicle (Ackermann) Characteristics

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Fig. 13 Vehicle Steering Wheel Ratio Characteristics

PRODUCTION VEHICLE

Date: 03-18-77 Front Suspension Program C025
Time: 8.065 Hours
7701 Front Suspension For Assembly Drawing
Data File Name 77701FS
Table I: Ackermann Turning Data For Constant Ratio Pitman
Ackermann Tread = 51.542 In. Wheelbase = 96.00 In.
Turning Dia. (At Wheel Center) = 31.314 Ft.

Axle Angle	Req'd. BETA	Actual BETA	Error	Steering Ratio	Steering Wheel Turns (Deg.)
38.000	28.829	33.090	-4.261	25.549	-865.354
35.000	25.971	30.383	-3.412	25.111	-793.117
30.000	23.785	26.136	-2.352	24.700	-678.008
25.000	20.452	22.035	-1.582	24.382	-566.485
20.000	16.934	17.942	-1.008	24.188	-456.193
15.000	13.182	13.761	-0.577	24.082	-345.483
10.000	9.150	9.417	-0.257	24.033	-233.141
5.000	4.777	4.846	-0.071	24.014	-118.216
0.000	0.000	0.002	-0.002	24.005	0.019

Steering Gear Ratio = 20:1
Design Camber = 0.36168 Deg.
Design Spring Length = 14.1114"
Design Castor = 1.07613 Deg.

Note: . . . . . Steering ratio is the incremental angular rotation of steering wheel at any point divided by incremental angular rotation of steered wheels.

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MOD I VEHICLE

Date: 04-22-80 Front Suspension Program C025
Time: 7.394 Hours
Stirling-Front Suspension
Data File Name . . . . . Front-Steering:
Table I: Ackermann Turning Data For Constant Ratio Pitman
Ackermann Tread = 51.542 In. Wheelbase = 96.000 In.
Turning Dia. (At Wheel Center) = 36.514 Ft.

Axle Angle	Req'd. BETA	Actual BETA	Error	Steering Ratio	Steering Wheel Turns (Deg.)
32.000	25.075	27.989	-2.915	28.425	-802.441
30.000	23.785	26.308	-2.523	27.848	-750.653
25.000	20.452	22.175	-1.723	27.277	-624.766
20.000	16.934	18.045	-1.111	26.805	-501.545
15.000	13.184	13.827	-0.643	26.506	-378.850
10.000	9.150	9.450	-0.300	26.328	-255.111
5.000	4.777	4.857	-0.080	26.233	-129.142
0.000	0.000	0.002	-0.002	26.203	0.022

Steering Gear Ratio = 20:1
Design Camber = 0.36172 Deg.
Design Spring Length = 14.1114"
Design Castor = 1.07611 Deg.

Note: . . . . . Steering ratio is the incremental angular rotation of steering wheel at any point divided by incremental angular rotation of steered wheels.

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MOD I VEHICLE

Date: 05-12-80 Front Suspension Program C025
Time: 13.683 HRS
Stirling-Front Suspension
Data File Name..... Frt-Ster;

Table 1: Ackerman Turning Data For Constant Ratio Pitman
Ackerman Tread = 51.542 In. Wheelbase = 96.000 In.
Turning Dia. (at Wheel Center) = 36.514 Feet

Axle Angle	Required Beta	Actual Beta	Error	Steering Ratio	Steering Wheel Turns (Deg.)
32.000	25.075	27.989	-2.915	19.898	-561.709
30.000	23.785	26.308	-2.523	19.494	-525.457
25.000	20.452	22.175	-1.723	19.094	-437.336
20.000	16.934	18.045	-1.111	18.763	-351.081
15.000	13.184	13.827	-0.643	18.554	-265.195
10.000	9.150	9.450	-0.300	18.429	-178.578
5.000	4.777	4.857	-0.080	18.363	-90.399
0.000	0.000	0.002	-0.002	18.342	-0.015

Steering Gear Ratio = 14:1 Des. Spring Length = 14.1114 In.

Design Camber = 0.36172 Deg. Design Caster = 1.07611 Deg.

Note:

Steering Ratio is the incremental angular rotation of steering wheel at any point, divided by incremental angular rotation of steered wheels.

Note: . . . . . The production vehicle has an axle angle of 38.000° whereas the Mod I vehicle has an axle angle of 32.000°. This accounts for the difference of the turning diameter of 31.314 feet for the production vehicle and 36.514 feet for the Mod I vehicle. The restriction was imposed on the Mod I vehicle in order to eliminate an interference condition between the tie-rod inner ball joint and the block in a full turn condition.

DISCUSSION - ACKERMANN

Axle Angle = Road Wheel Angle Of Inside Wheel In A Turn
Required BETA = Theoretically Perfect Angle of Outside Wheel
Actual BETA = Actual Angle Of Outside Wheel
Error = Difference Between BETA Angle & Actual Angle

The degree of error shown is well within the acceptable range for production vehicles on both the production vehicle and the Mod I vehicle. The error can be directly compared in the turning angles from 0° to 30°. Interpolation is required for the 32° angle wheel cut.

STEERING COLUMN & GEAR

The steering gear was relocated by moving it down along the center line of the sector shaft (2.70"). The center line of the input shaft was maintained at an orientation parallel to the steering column. Under these conditions two Cardan "U" joints would produce constant velocity. A slip joint is located between the "U" joints to compensate for vehicle deflections. Critical clearance areas include; steering gear to power steering pump, steering gear to starter, steering column to left-hand exhaust outlet.

CONCLUSION

Steering system installation clearances are marginal but acceptable for experimental purposes. Steering performance would be acceptable from a production standpoint.

SUSPENSION

In the suspension system program, wheel jounce is shown as positive and rebound as negative. Comparison of the production castor to the Mod I vehicle castor shows that the Mod I vehicle is basically equivalent to the production vehicle in the jounce mode and is better than the production vehicle in the rebound mode (the castor change is less). In both cases, the Mod I vehicle castor change characteristics are in the acceptable range.

Production camber change is in the negative mode throughout the jounce and rebound of the wheel. The Mod I vehicle is also in the negative, although the characteristic values are less but well within an acceptable range.

Caster change and camber change values are in degrees.

In the jounce mode the production toe change is better than the Mod I toe change characteristics, however, the Mod I characteristics are acceptable. In the rebound mode the Mod I toe change characteristics are considerably better than the production toe change characteristics.

Toe change characteristics are illustrated in both degrees and inches.

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The suspension data characteristics are tabulated below.

SUSPENSION CHARACTERISTICS

PRODUCTION VEHICLES

JOUNCE	CASTOR CHANGE	CAMBER CHANGE	TOE CHANGE
4.700	0.980	-2.343	-0.376
4.000	0.942	-1.710	-0.220
3.000	0.823	-0.980	-0.115
2.000	0.629	-0.449	-0.053
1.000	0.356	-0.117	-0.021
0.000	0.000	0.000	-0.001
-1.000	-0.448	-0.135	0.036
-2.000	-1.001	-0.596	0.123
-3.000	-1.685	-1.544	0.336
-3.800	-2.368	-2.923	0.725

MOD I VEHICLE

JOUNCE	CASTOR CHANGE	CAMBER CHANGE	TOE CHANGE
4.700	1.014	-2.306	-0.455
4.000	0.911	-1.737	-0.307
3.000	0.734	-0.067	-0.151
2.000	0.524	-0.555	-0.054
1.000	0.280	-0.196	-0.007
0.000	0.000	0.000	-0.001
-1.000	-0.319	-0.011	0.022
-2.000	-0.683	-0.212	0.060
-3.000	-1.103	-0.774	0.086
-3.800	-1.495	-1.630	0.054

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Note: . . . . . Positive toe change means to-in (tire front-to-rear tread change).

Nominal castor is +1°, nominal camber is +1/4° and nominal toe-in is 1/8°.

AUXILIARY DRIVE

The crankshaft power output pulley drives the alternator and fan. The fan pulley in turn drives the power steering pump and air conditioning compressor. The variator which drives the blower is powered from the crank shaft. The upstart motor is located on the right-hand side on the rear side of the blower.

The fan is an 18.5" diameter blade flex fan with a 2.75 pitch width and is driven through an electric clutch. There is approximately 1" clearance between the fan and radiator.

Saginaw Pump - Gear - Remote Reservoir

Bendix Hydro-Boost II System with integral accumulator

Alternator Bosch 55 amp.

POWER STEERING - POWER BRAKES

The power steering system incorporates a Saginaw power steering gear, Saginaw pump with remote reservoir and the plumbing required for the circuitry. In series with the power steering system is the Bendix Hydro-Boost II System incorporating an integral accumulator. The Bendix Hydro-Boost Power Brake is an open center spool valve and hydraulic cylinder assembled into a single unit. With pressure applied by a power steering pump, it actuates the brake system master cylinder. It has a stored energy reserve via a gas charged accumulator for power-off braking.

Fig. 14 Vehicle Suspension Aim Modifications

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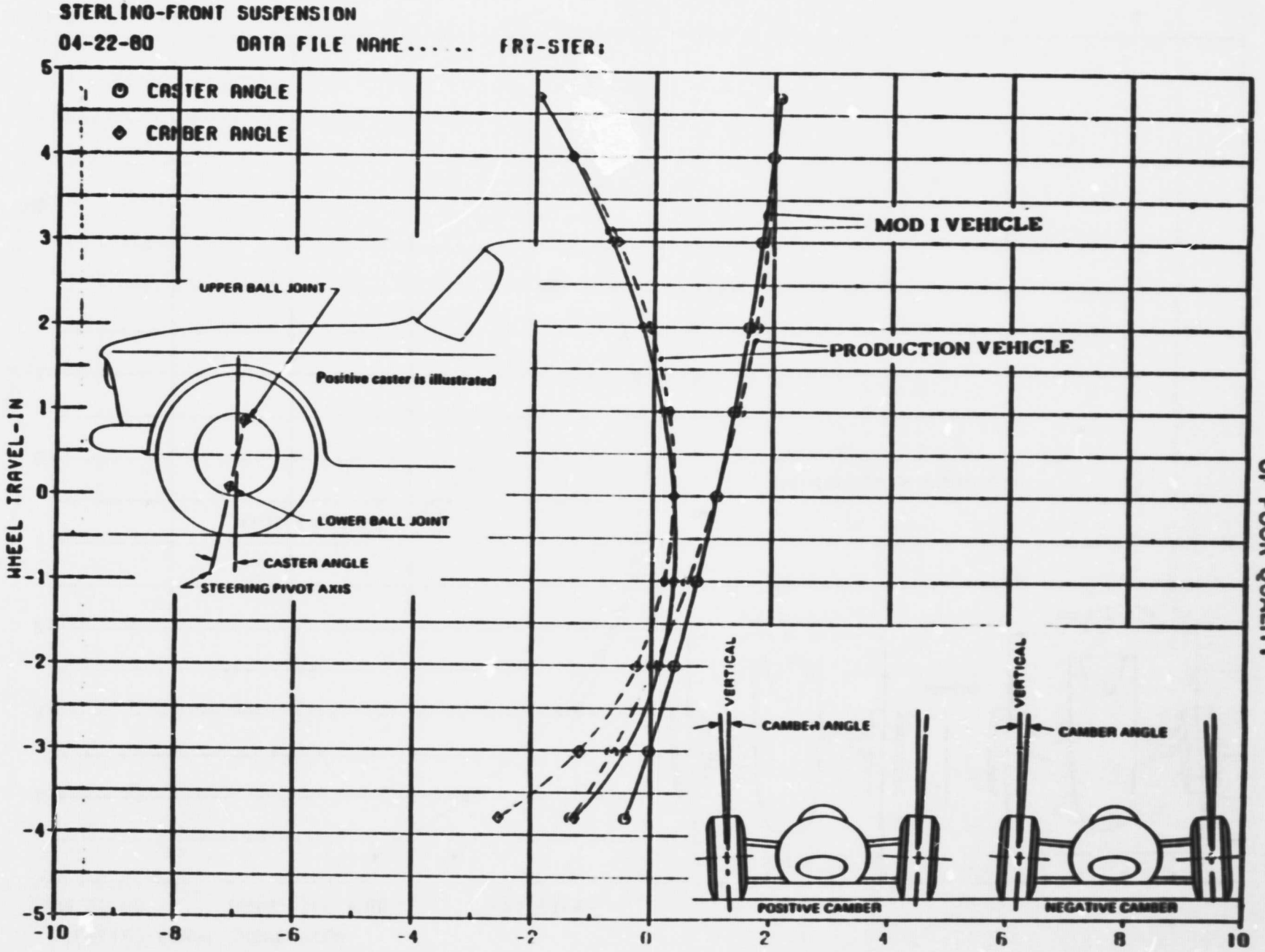


Fig. 15 Castor/Camber Characteristics

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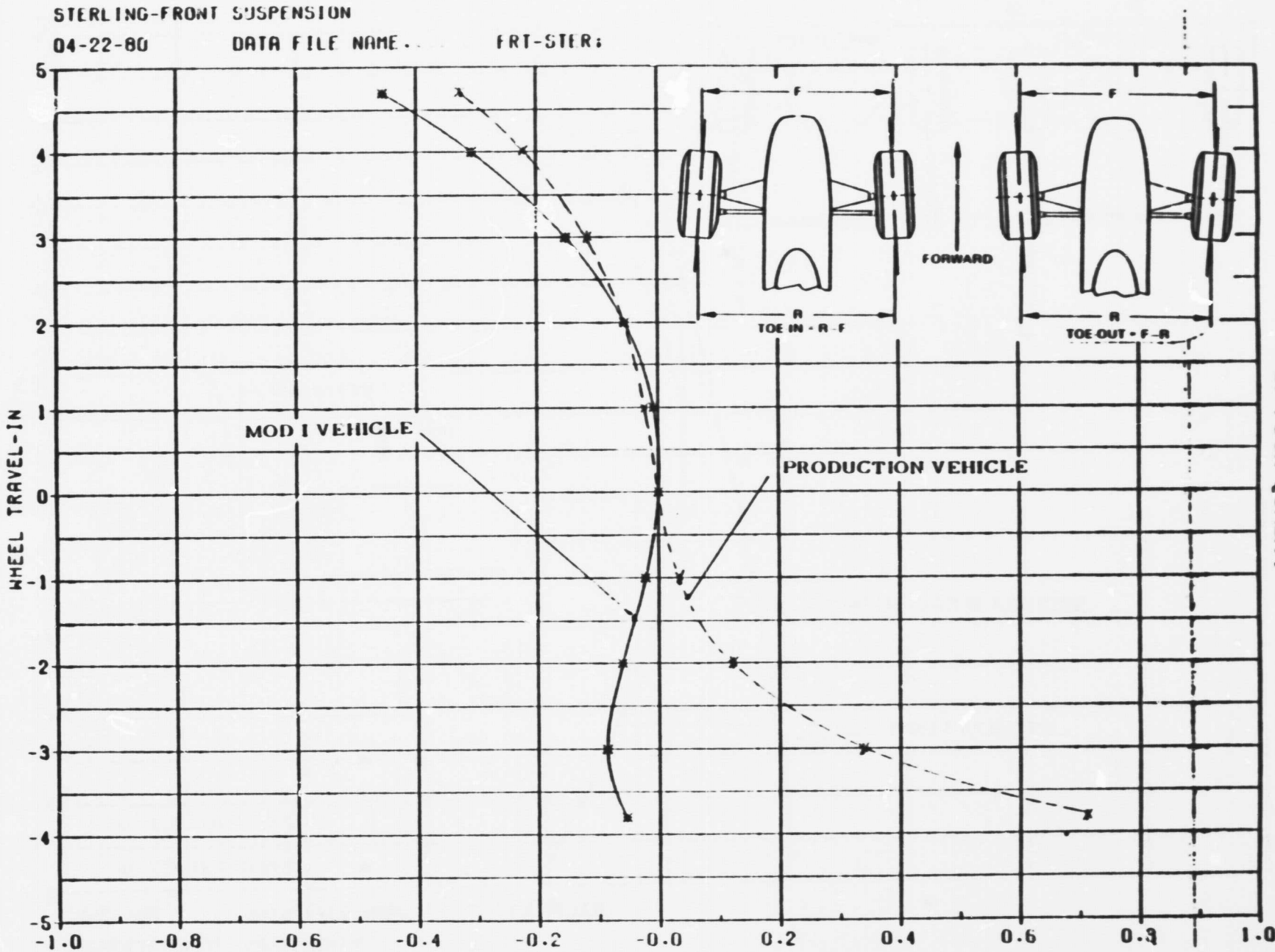


Fig. 16 Toe-in/Toe-out Characteristics (inches)

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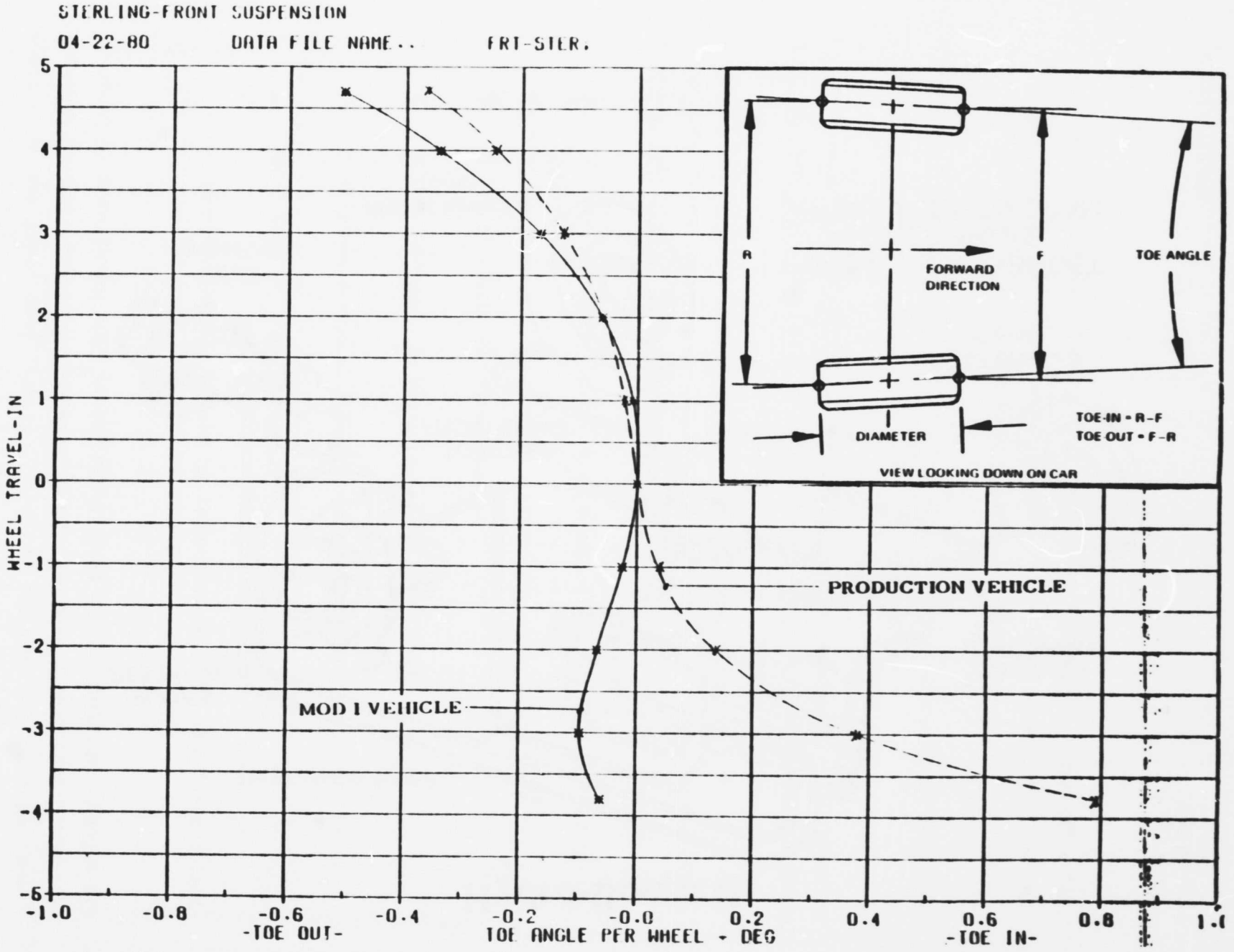


Fig. 17 Toe-in/Toe-out Characteristics (Degrees)

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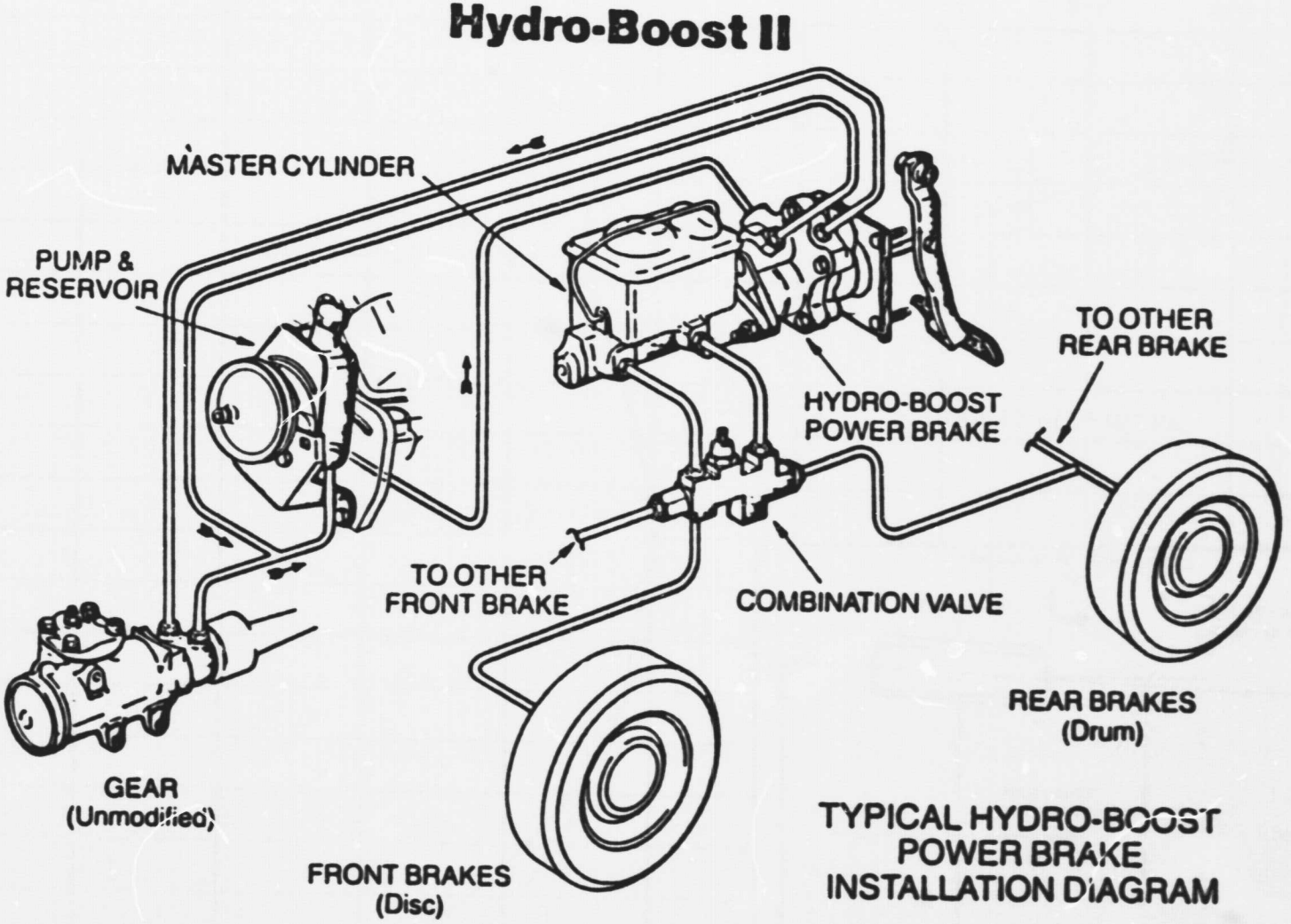


Fig. 18 Hydro-Boost II

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COOLING SYSTEM

Radiator -

The external dimensions of the frontal area of the radiator are 38" x 23". All candiate radiator manufacturers must fit these dimensions and be constructed to accommodate mounting as shown on the layout.

Radiator will be the cross-flow type with a 2" diameter outlet on the lower right corner and the inlet on the upper left hand corner.

Assuming candidate radiator manufacturers use 2" tanks, the core dimension would be 34" x 23" resulting in a core frontal area of approximately 782 sq. in.

At present there are five candidate cores and sample radiators in the process of being fabricated. Core descriptions are as follows:

A. 2 Rows of Tubes, 36 FPI w/o Louvers (this core is similar to the Opel).

B. 1 Row of Tubes, 25 FPI w/Louvers

C. 2 Rows of Tubes, 25 FPI w/Louvers

D. Blackstone 2 Rows of Tubes (Tropic Tradename)

E. Arco Special Rows of Tubes, FPI

All five of these radiators should be interchangeable in the Concord Test car.

In order to accommodate this large frontal area of the radiator, changes are required to the front end of the vehicle. The most significant being: a) removal of the inner head lights, b) cut off the ends of the frame and

reconnect with adequate brackets, c) remove hydraulic shock absorber bumper brackets and replace with fixed brackets (presently in Eagle production) and, d) redesign the grille and turn signals.

All structural components that obstruct the direct flow of ram air to the core of the radiator will be slotted (if possible).

From an installation and structural standpoint, the end objective is to develop maximum unobstructed frontal core area for direct ram air impact.

HEATING - AIR CONDITIONING

Present plans for the Heating - A/C system include the utilization of a console duct package that is in the production car under the dash. The blower and the evaporator core will be retained. The heater core will be removed and water lines from the engine will not be necessary. The assembly is illustrated in Figure 20.

Heating

The maximum requirement for heat delivery is 11,000 kW/hr. The maximum heater airflow is 130 kg/sec in the car. A 10% reduction in these output requirements would not result in objectionable performance.

In a review of possible heat sources of adequate capacity, electrically heated air was deemed unsuitable, because the current draw would require an oversize alternator. The use of supplementary electric heat to the engine coolant would also affect alternator sizing. Therefore, a gasoline heater is the type of heater to be installed.

American Motors Engineers would recommend the use of a gas heater only if installed outside the passenger compartment. The unit selected for this application is an ESPAR make - Model B1L. It is 14.5" long and rectangular in shape (4.33" x 5.71"). It will produce 4,700 BTU/HR output and will burn

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one gallon of gasoline in 18 hours of operation. This heater, used in conjunction with passenger compartment air recirculation, will probably supply adequate heat for average cold weather.

Air Conditioning

The system presently in the production vehicle will remain intact, except for the condenser. A Sankyo compressor will be substituted for the production compressor. A deterioration is expected in the A/C performance, due to the use of this compressor, by 3 degrees in discharge air and subsequently 5 to 6 degrees in interior temperature under ambient conditions of 100°F. The maximum A/C capacity required is 16,500 kW/hr.

The primary condenser location will be in the trunk area or the rear fender wells. It will be an integral unit of condenser and electric fan or fans. If this effort is not successful, the condenser will be located in the conventional position in front of the radiator.

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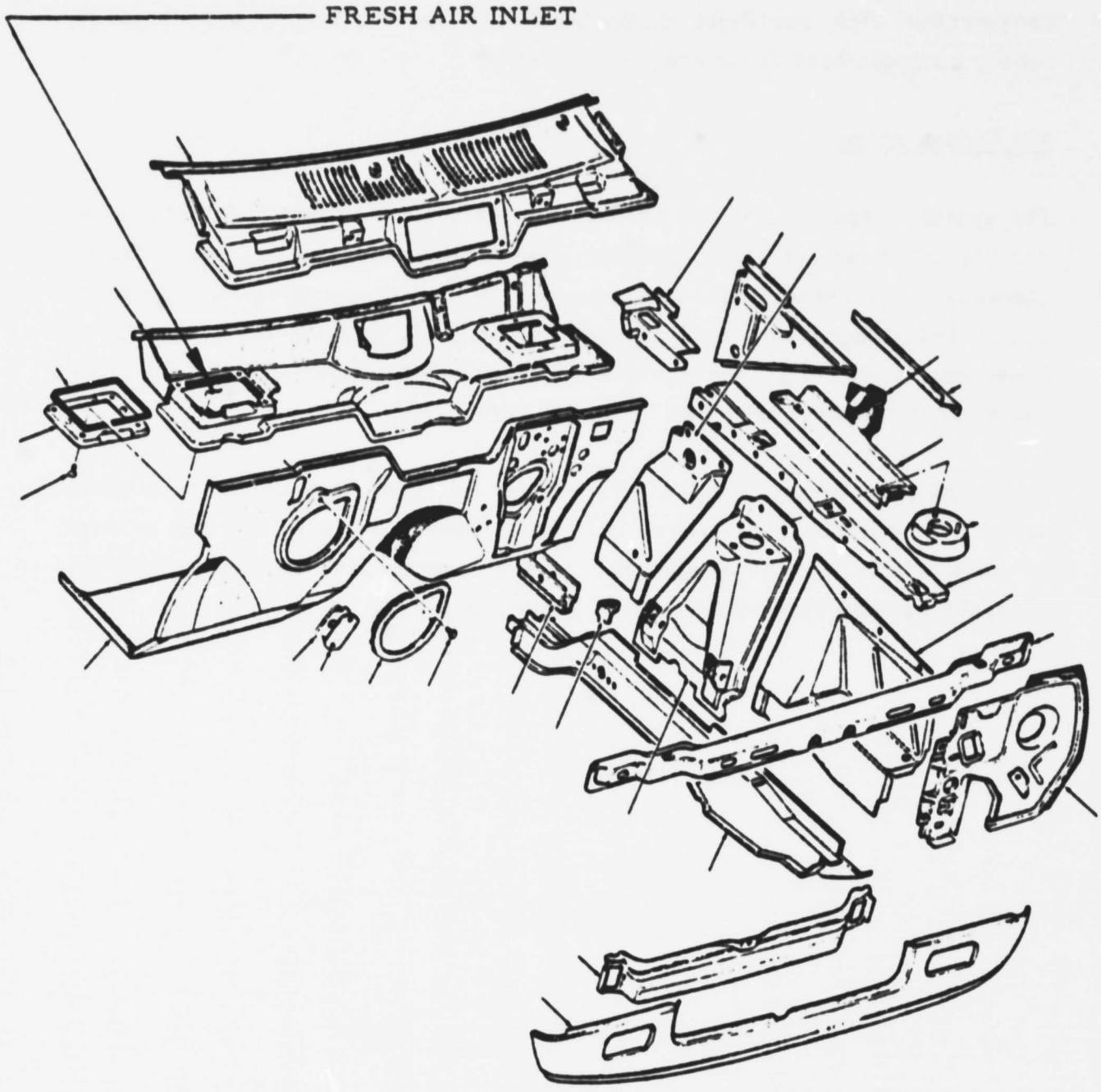


Fig. 19 Dash and Front Wheelhouse Panel

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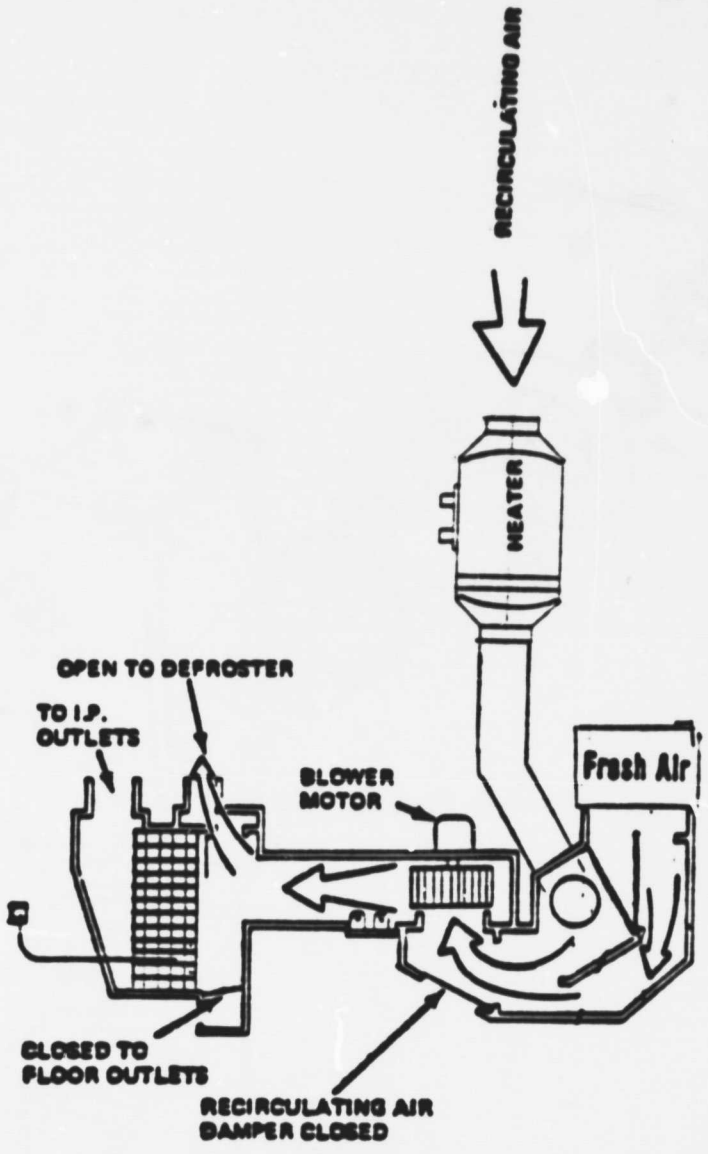


Fig. 20 Heater A/C Fresh Air System

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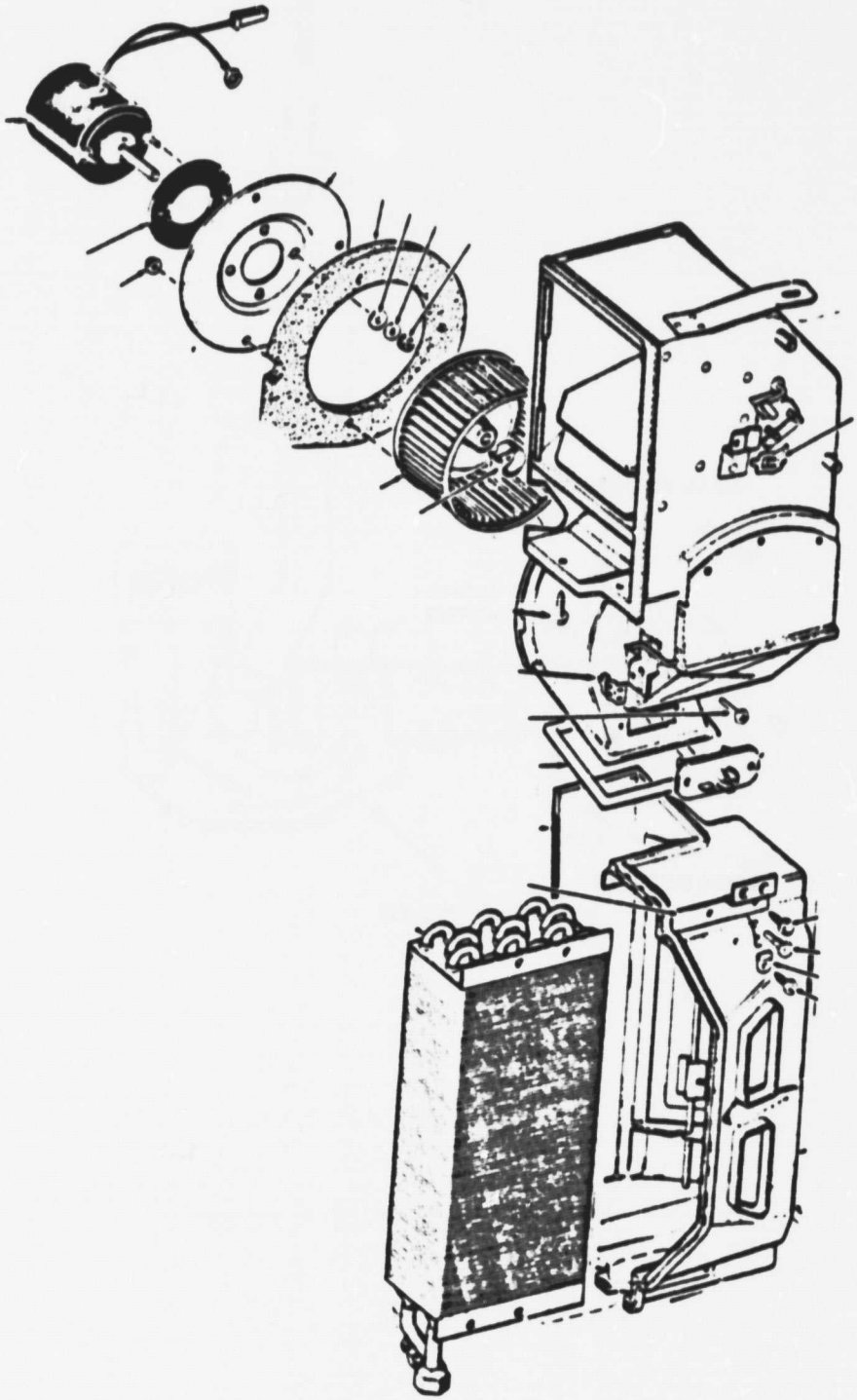


Fig. 21 Heating and Air Conditioning Fresh Air Distribution Chamber

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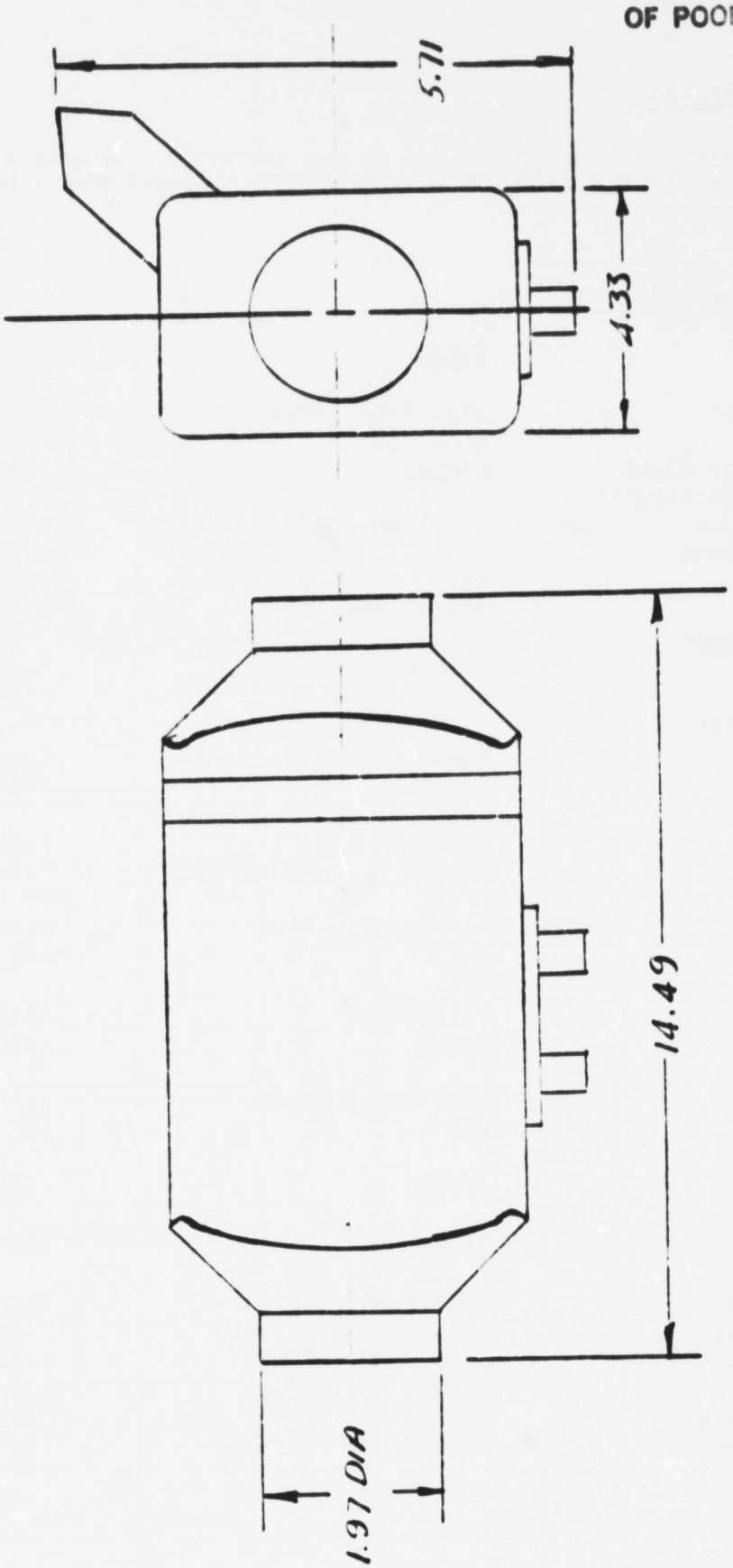


Fig. 22 Espar Heater Model 311

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Mod I Vehicles

Mod I engines will be installed in one vehicle by AM General. The selected vehicles will be a 1981 AMC Spirit 2-door liftback and a 1981 AMC Concord 4-door sedan.

Vehicle Description

	Concord	Spirit
Model Year	1981 4-Dr. Sedan Model 05	1981 2-Dr. Lift Bank Model 43
E.P.A. Car Class	Compact	Subcompact
Maximum Passengers	2 Frt./3 Rear	2 Frt./2 Rear
Engine Type	Stirling U-4	Stirling U-4
Displacement	0.494 (30.1 cu. in.)	0.494 (30.1 cu. in.)
Power	58 KW (77.8 H.P. @ 4000 RPM	58 KW (77.8 H.P.) @ 4000 RPM
Transmission Type	3-Speed Automatic Chrysler 904	3-Speed Automatic Chrysler 904
Accessories Type	Power Steering Saginaw	Power Steering Saginaw
	Power Brakes Bendix Hydro-Boost II	Power Brakes Bendix Hydro-Boost II
	Air-Conditioning AMC	Air-Conditioning AMC
	Alternator Bosch	Alternator Bosch
Axle	Rear Wheel Drive AMC	Rear Wheel Drive AMC
Ratio	(TBD)	(TBD)
Tires	75R14	75R14
Wheelbase	2743.2 MM (108.0)	2438.4 MM (96.0)
Front Tread	1464.055 MM (57.64)	1475.49 MM (58.09)
Rear Tread	1449.3 MM (57.06)	1448.22 MM (57.04)
Overall Length	4699.0 MM (185.0)	4247.89 MM (167.24)
Overall Width	1803.4 MM (71.0)	1825.75 MM (71.88)
Overall Height	1303.17 MM (51.306)	1307.89 MM (51.492)

ASE Mod I

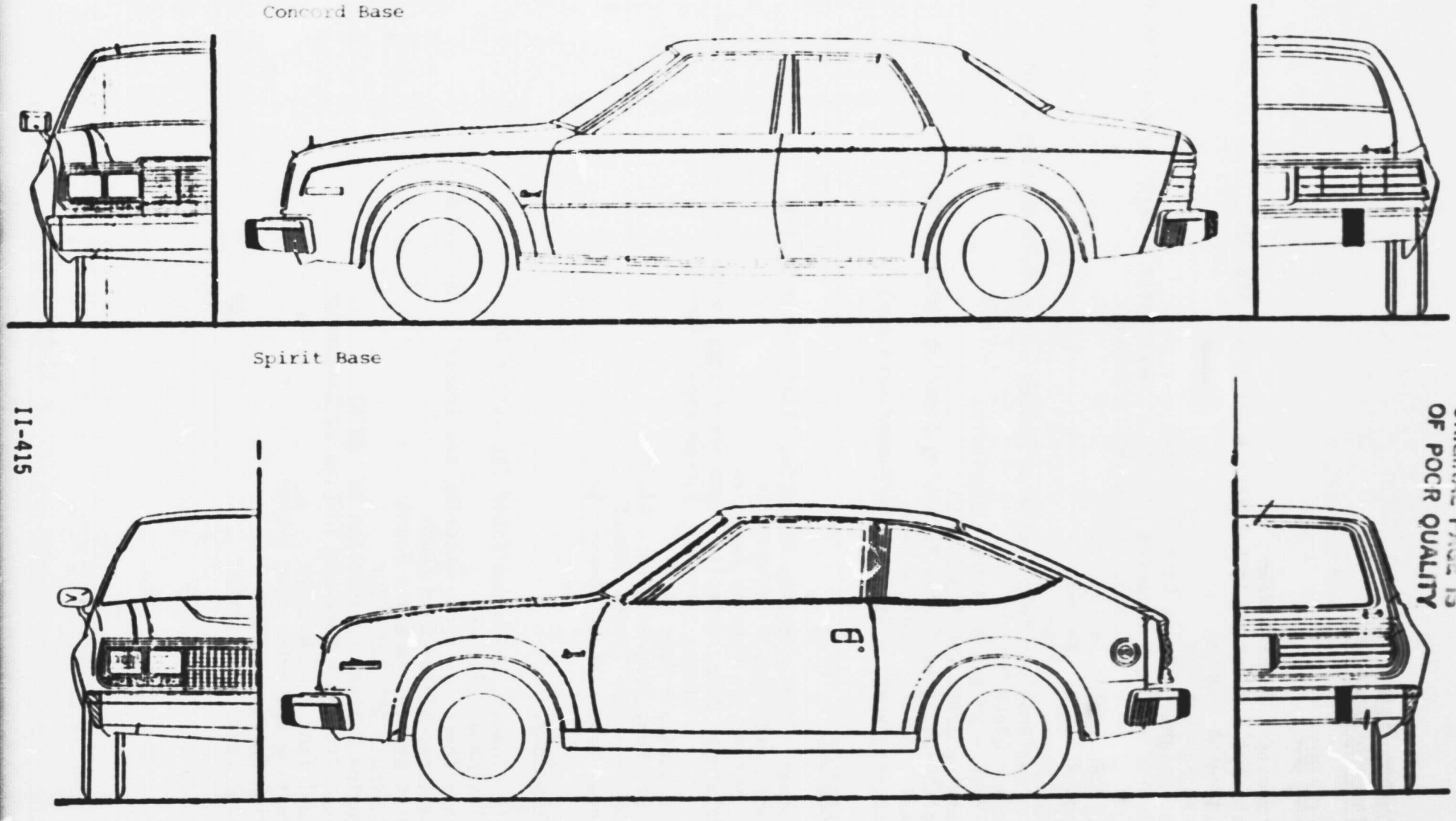


Fig. 23 Stirling Powered Automobile Specification

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CONCORD (BASE)
STANDARD EQUIPMENT

Interior

Armrests - Front & Rear
Ashtrays - Front w/Light & Rear (Rear Right Side Only)
Cigarette Lighter
Coat Hooks - Two
Dome Light (w/Front Door Switches)
Floor Covering - Carpeted Passenger Compartment & Cargo Area (Color-Keyed)
12 oz. (Trunk Mat - Sedan)
Hood Restraints - Two Front
Hood Release - Inside
Key Warning Buzzer
Lap and Shoulder Belt Restraint System w/Seat Belt Warning Light
Lock - Glove Box
Mirror - 8" Inside Mirror (Non-Tilt)
Parking Brake Warning Light
Safety-Designed Energy Absorbing Steering Column & Wheel
Seat - Bench - Front & Rear
Steering Wheel - Standard (w/Padded Horn Bar)

Exterior

Bumper, Front - Energy Absorbing (Static Rear System)
Dual Headlights - Rectangular
Mirror - Left Outside Rearview
Moldings: Drip, Wheellip, Hood Front Edge, Windshield Surround, Rear
Window Surround, Rocker Panel (Narrow)
Pin Stripe - Body Side
Scuff Body Side Moldings (Black)
Two Stop Door Hinges (Front)
Wheel Trim - Wheel Covers (4)

Functional

Air Cleaner, Cellulose-Fiber for Carburetor
Alternator, 42 Amp
Anti-Theft Ignition, Steering and Transmission System
Battery, 55-380 Cold Crank
Disc Brakes, Manual - Front
Engine - 151 CID, Four
Engine Coolant (Protection to -20°F)
Fuel Filters - Tank and in-Line at Carburetor
Fuel Tank - 22 Gallons
Heating and Defrosting System
High Energy Ignition
High Level Ventilation

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CONCORD (BASE)
STANDARD EQUIPMENT (Continued)

Horn - Single
Oil Filter, Full-Flow
Self-Adjusting Brake Linings
Single Unit Body Construction
Spare Tire - High Pressure
Sway Bar, Front
Tires - D78 x 14 Black (Unbelted) 4)
Transmission - 4-Speed Manual Floor Shift (Full-Synchromesh)
Wipers and Washers - 2-Speed Electric

AMC Anti-Pollution Devices

Fuel Vapor Control System
Thermostatically-Controlled Air Cleaner
Transmission - Controlled Spark System
Exhaust Gas Recirculation System
Air Injection "Air Guard" System
Catalytic Converter
Engine designed to run on non-leaded gasoline only.

SPIRIT (BASE)
STANDARD EQUIPMENT

Interior

Armrests - Front
Ashtray - Front w/Light
Cigarette Lighter
Coat Hooks - Two
Floor Covering - Carpeted Passenger Compartment & Cargo Area
(Color-Keyed) 12 oz.
Front Seat Back Lock
Glove Box Lock
Head Restraints - Two Front
Hood Release - Inside
Instrument Panel Insert - Black
Key Warning Buzzer
Lap and Shoulder Belt Restraint System w/Seat Belt Warning Light
Light - Dome (w/Front Door Switches)
Mirror - 8" Inside Mirror (Non-Tilt)
Parking Brake Warning Light
Safety-Designed Energy Absorbing Steering Column & Wheel
Seats - Front Vinyl Buckets & Rear Bench (Fold Down)
Spare Tire Cover - Color-Keyed Carpet
Steering Wheel - Standard (w/Padded Horn Bar)

SPIRIT (BASE)
STANDARD EQUIPMENT (Continued)

Exterior

Bumper, Front - Energy Absorbing (Static Rear System)
Flip-up Rear Hatch
Headlights - Dual (Rectangular)
Mirror - Left Outside Rearview Mirror
Moldings: Wheellip, Drip, Hood Front Edge, Windshield Surround, Rear Window Surround, Narrow Rocker Panels
Pin Stripe - Body Side
Two Stop Door Hinges
Wheel Trim - Wheel Covers (4)

Functional

Air Cleaner, Cellulose - Fiber for Carburetor
Alternator, 42-Amp
Anti-Theft Ignition, Steering and Transmission System
Battery, 55-380 Cold Crank
Disc Brakes, Manual - Front
Engine - 151 CID, Four
Engine Coolant (Protection to -20°F)
Fuel Filters - Tank and In-Line at Carburetor
Fuel Tank - 21 Gallons
Heating and Defrosting System
High Energy Ignition
High Level Ventilation
Horn - Single
Self-Adjusting Brake Linings
Spare Tire - High Pressure
Single Exhaust System
Single Unit Body Construction
Sway Bar - Front
Tires - C78 x 14 White (Unbelted (4)
Transmission - 4-Speed Manual Floor Shift (Full Synchromesh)
Wipers and Washers - 2-Speed Electric

AMC Anti-Pollution Devices

Fuel Vapor Control System
Thermostatically-Controlled Air Cleaner
Transmission-Controlled Spark System
Exhaust Gas Recirculation System
Air Injection "Air Guard" System
Catalytic Converter
Engine designed to run on non-leaded gasoline only.

CONCORD DL

Equipment in addition to or in place of Base.

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CONCORD DL (Continued)

Interior

Individual Reclining Seats in Vinyl
Custom Door Panels
Molded Fiberglass Headliner and Sun Visors in Alpine Cloth
Day/Night Mirror - 10"
Custom Steering Wheel w/Wood Grain Horn Cover
Two Rear Ashtrays
Electric Clock (Vacuum Flourescent Quartz Digital)
Two Courtesy Lights (Under I.P.)
Trunk Carpeting - Floor, Wheelhouse & Spare Tire (05/06)
Wood Grain Instrument Panel Overlay

Functional

D78 x 14 White (Unbelted) Tires (4)
Dual Horns
Extra-Quiet Isulation Package

Exterior

Landau Vinyl Roof (06), Full Vinyl Roof (05)
Opera Quarter Windows (05/06)
Striping - Hood, Upper Body Side, Quarter Window, Quarter Window Argent (06)
DL Crest - "B" Pillar (05/08), "C" Pillar (06)
Bumper Guards - Front & Rear
Styled Wheel Covers - Argent (4)
DL Nameplates: Front Fenders
Moldings - "B" Pillar Overlay (05), Color-Keyed Wide Scuff
(NA w/Two-Tone or 08), Belt, Wide Rocker

Vehicle Function

AMG - Spirit

The AMG vehicle will be a base test bed with function only. Vehicle and engine tests will be conducted to optimize the total package with prime objective that the optimized package will be turned over to the EPA for their tests in the spring of 1983.

The aesthetic changes to the AMG Spirit will be minimal. They will only relate to functional items.

MTI - Concord

The MTI vehicle will be a vehicle with a two-fold function. The prime function will be a vehicle test bed for component development. Its secondary function will be demonstration of the Mod I engine vehicle package.

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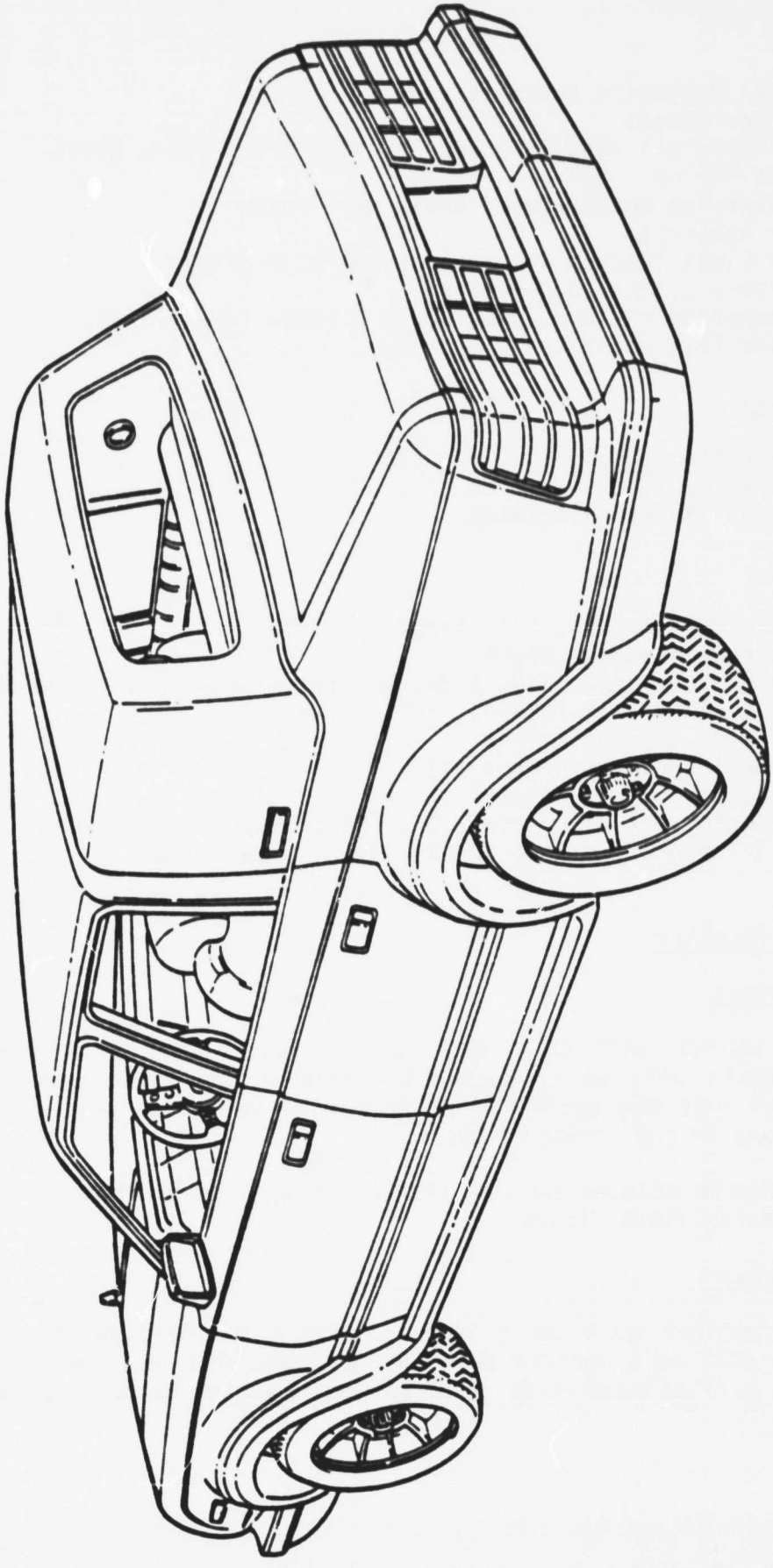


Fig. 24 Mod I Concord

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The aesthetic changes to the MTI Concord will be similar to the P-40 Concord demonstration vehicle. They will incorporate styling modifications to roof, deck lid and front end. With this type of vehicle the exposure demands a package that attracts attention by function as well as aesthetics.

Exterior Modifications

Studies relating to exterior modifications have been completed which are related to the front end frontal area. These studies were tied to the cooling system development group which recommended that the grille opening had to be enlarged to enhance the performance of the cooling system which incorporates a large radiator core.

A review of the present production front end revealed that with the present dual (rectangular) headlights, the grille opening was at its maximum size. Considering costs to the program and time to develop a special front end, we investigated modifying production hardware. The first proposal was to remove the high beam headlight from the dual headlight system. The necessary grille opening was achieved but we questioned the package as being saleable with a low beam headlight system only. Researching AMC headlight systems revealed that we could incorporate a 1978 Concord single high low beam rectangular headlight in the 1981 front end. With the 1978 light package we covered any questions that would arise concerning safety. Another problem came about with 1978 light package. We lost the area needed for turn signal lamps. After an investigation, it was agreed small lamps would be installed in the bumper slots witout affecting our cooling system performance.

The front bumper mounting package was revised to incorporate the larger radiators. The exterior bumper face will remain the same except the cooling slots have been enlarged in width to help cooling performance.

The grille for the Mod I will be a modified AMC-AMX grille shell. We selected this type grille, not only for aesthetic value, but the function of the grille screen can be controlled as to density of the grille screen openings. The grille shell is a simple basic design which can be modified to have a wide opening or split type grille. Aesthetics, with either grille, is pleasing to everybody that has seen the proposals. A final selection has not been made at this date.

The vehicle development task force has made the recommendation that the air-conditioning condenser not be mounted in front of the radiator. The task force recommendation is to mount the condenser in the vehicle trunk. The present state-of-the-art in the automobile industry is to mount the air-conditioning condenser in front of the radiator. So serious study as to the feasibility of a trunk mounted air conditioning had to be given to the adaption. Mechanically there are no major problems in mounting the condenser in different locations as long as a sufficient amount of air can be supplied to extrapulate the heat from the condenser.

Exterior modifications of the vehicle would be needed to incorporate a condenser in the trunk. With the changes in the environment, trunk area and the condenser would need protection from the elements. We have made two

proposals for trunk mounted condensers. Both proposals show that the air inlets would be installed in the rear quarter panels. One proposal shows a standard deck with louvers stamped in it for heat discharge. The other proposal shows the deck lid redesigned to maintain trunk space with exhaust vents to the rear of the vehicle. Both designs would require a plenum to house the condenser and control the air flow and elements from the trunk area.

Additional studies were needed to solve the air-conditioning condenser problems for lift back vehicles (no trunk). Our proposal requires mounting two condensers in the front fenders with electric fans. Air would be ducted from below the front bumper through the condensers and ducted to exhaust outlets mounted in the sides of the front fenders. If the feasibility of the design proves to be satisfactory, the design would eleminate the trunk mount design for other vehicles.

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CONCORD SEDAN

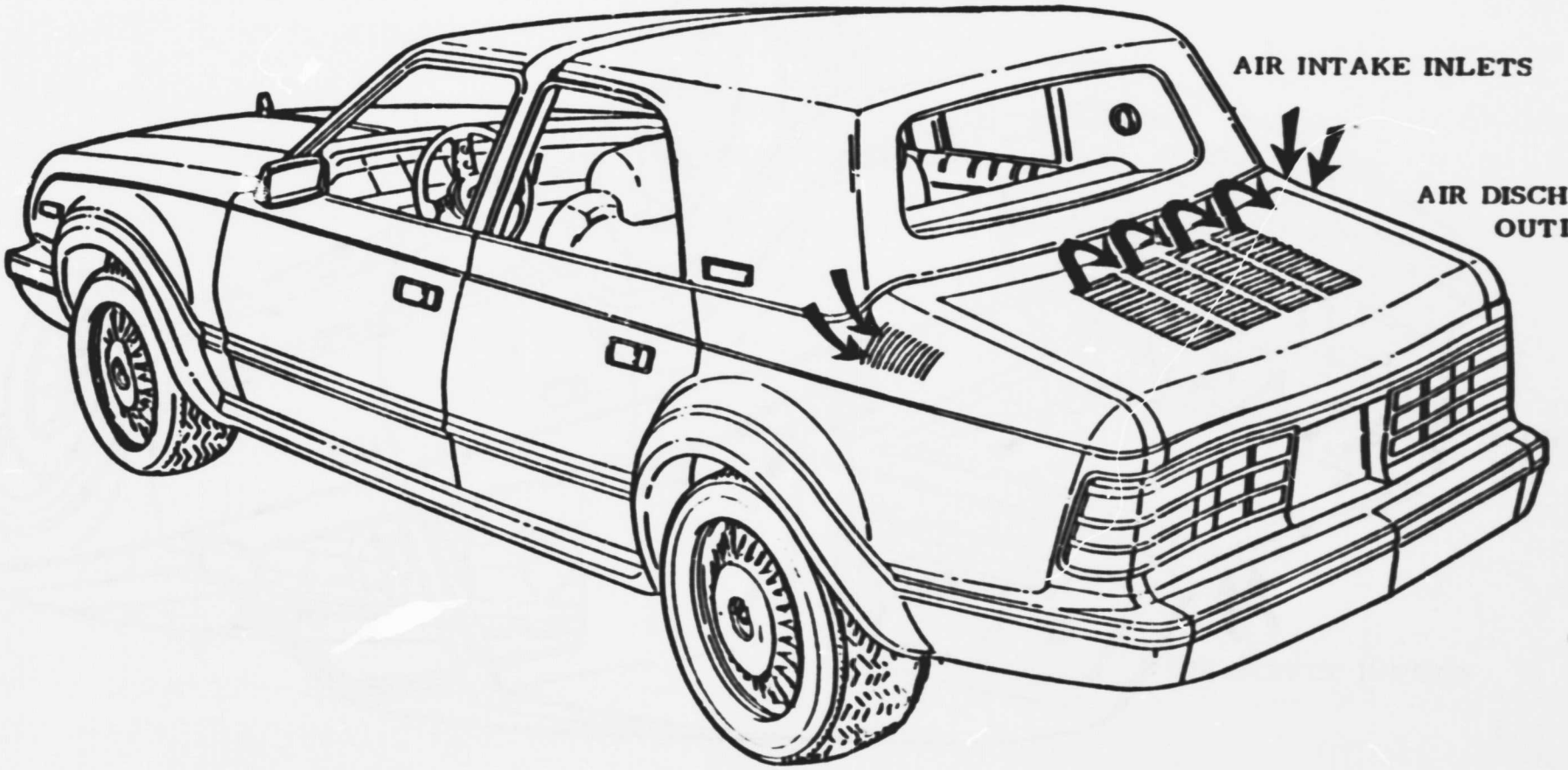


Fig. 25 Rear Trunk Mounted Air Conditioning Condenser Package

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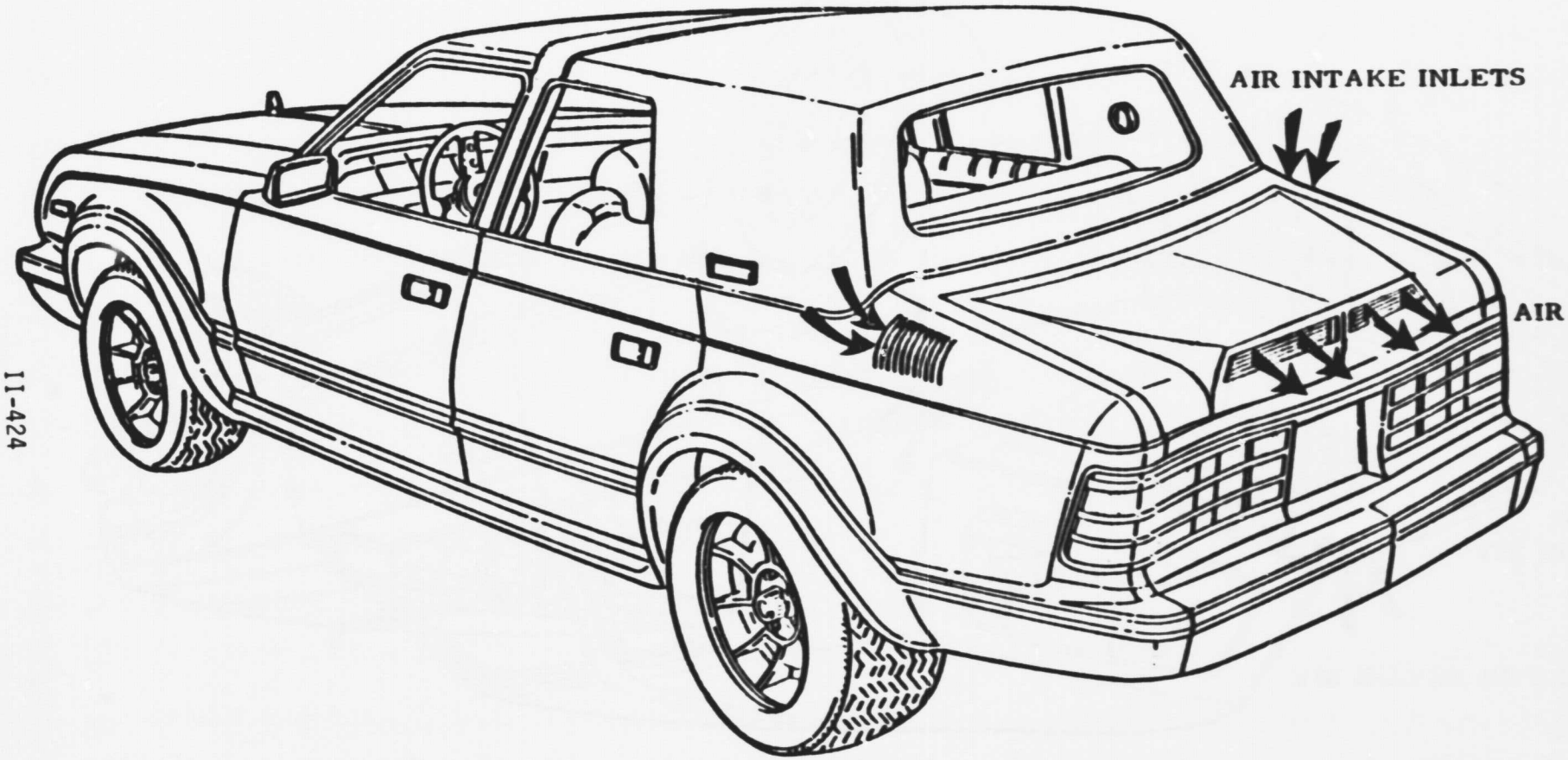


Fig. 26 Dual - Front Mounted Air Conditioning Condenser Package

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Fig. 27 Dual - Front Mounted Air Conditioning Condenser Package

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9.3 MOD I VEHICLE SYSTEM ANALYSIS

This section presents the results of the vehicle simulation and optimization study performed on the MOD I system. The basic tool used in this study was the MTI vehicle simulation program, which is a growth version of the University of Wisconsin program. MTI has tailored this program to the specific needs of the Stirling powered vehicle, while retaining its utility as a general heat engine vehicle simulation.

9.3.1 Validation

The program was validated for spark ignition vehicles by the University of Wisconsin. MTI has performed further validation comparisons on the P-40 Spirit vehicle, and will continue to validate the results of the program as more vehicle data becomes available. Figure 1 shows the baseline P-40 Spirit validation.

9.3.2 Cold Start Penalty

The MOD I cold start penalty is now believed to be somewhat less than the fuel equivalent value of the stored heat in the hot engine system. There are several reasons for this; during cold start up some of the fuel that is spent heating the engine would normally be lost through conduction or radiation. Additionally, the thermal efficiency of the engine is somewhat higher due to the low cooling water temperature.

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P-40 Spirit experience indicates a reduction of the urban mileage by approximately 6%. Using this 6% penalty one calculates a fuel equivalent value of 86½ grams of fuel. Applying a correction factcr from the MOD I to the P-40 Spirit equivalent to the ratio of store heat, yields a value of 116 grams for the MOD I cold start penalty.

9.3.3 MOD I System Characteristics

The Mod I vehicle is a 1982 Lerma. The general vehicle characteristics are shown, Figure 2. Additional plots of detailed simulation input data are shownin Figures 3 through 18.

The road load of 16.5 hp shown in Figure 2 indicates that coast down testing of the vehicle will yield a lower road load value. This is evidenced by calculating the C_D for the vehicle using 10 hp as the equivalent air drag. One obtains a C_D of .62, which is abnormally high.

The fuel used in this analysis is unleaded regular with a heating value of 113,525 BTU/gal as specified by NASA.

The cooling fan has been assumed to be similar to the P-40 Spirit. A flex fan horsepower curve was used with an electric clutch cutout at 20 mph vehicle speed and during wide open throttle.

9.3.4 Optimization

An optimization study was performed on the MOD I system. This was accomplished by optimizing first the shift schedule and then the rear

axle ratio.

P-40 Spirit experience was used in the MOD I shift schedule optimization. The wide open throttle downshift curves were raised to 95% throttle setting. The part throttle curves were shifted to lower drive shaft speeds and the angle of the mid-throttle shift was rotated counterclockwise. Cross plotting the shift curves on to a torque/rpm map of the engine shows that this shift pattern forces the engine to operate in higher pressure (and thus, higher efficiency) regions of the map than the standard shift pattern. The MOD I shift schedule is shown in Figure 11.

The rear axle ratio was varied to establish the relationship between 0 to 60mph acceleration time and the combined mileage value (Note that these combined mileage values do not include cold start penalty). Figure 19 shows this relationship. Due to hardware availability no axles below a numerical value of 2.0 were run. A further mileage improvement would be achieved at a lower axle number. However, due to the slope of the curve, the 0 to 60 time goes up dramatically with small increases in mileage with rear axle ratios below 2.

9.3.5 MOD I Projections

The projections for the various mileages and performance tests are shown in Figure 20. Several notes should be made on various values. First, the five to ten mph acceleration is approximately 10 feet per second squared which is excellent. Secondly, the abnormally high road load value used has a significant impact on the highway mileage and all of the cruise mileages.

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We expect that the acceleration performance and the cruise mileages will be substantially higher, as the actual (not EPA) road load applies.

Some of the more interesting output plots of the MDD I vehicle analysis are shown in Figures 21 thru 32. Figure 25 shows a hp/rpm map with the path of the engine through the urban cycle shown. This in conjunction with a similar plot of the highway mileage will aid in further engine optimization studies. Figure 28 shows the wide open throttle acceleration of the MDD I. One should note however that the velocity profile in the input included five seconds at idle speed after which the vehicle was requested to accelerate to 99mph. The auto driver within the simulation program anticipated the take off time by 5/10ths of a second. Therefore, any acceleration from 0 time should be reduced by 4.5 seconds. However, the shown value of 0 to 60 acceleration also includes a 2/10ths of a second delay time, which might be expected as a result of hydrogen introduction time.

P-40 SPIRIT COMPARISON

Value mpg	Urban*	Highway	Combined
Predicted	13.7	22.2	16.5
Measured	13.3	22.5	16.3
Error	3%	1.3%	1.2%

* With Cold Start Penalty

Fig. 1 Code Validation

1981 SPIRIT

Test Weight	3421 lb
Inertia Weight	3375 lb
Total Road Load (A/C)	16.50 hp
Dyno Power Setting (A/C)	11.10 hp
Fuel	Unleaded Regular (113, 525 Btu/gal)
Drivetrain	A-904 3-Speed Automatic, 10.75 Torque Converter with Lockup, Hypoid Axle
Torque Converter	Production Unit with Loose Tolerances
Gear Box Ratios	2.45, 1.45, 1.00
Front Pump	U of W Test Case Data (Modified)
Shift Schedule	Optimized for Stirling Power
Final Drive Ratio	Optimized for Stirling Power
Efficiencies	All Eff. Calculated using Ricardo Polynomial
Accessories	Power Steering, Brakes, A/C
Cooling Fan	Flex Fan, Cutout at 20 MPH + WOT

Fig. 2 1981 Mod I Vehicle Characteristics

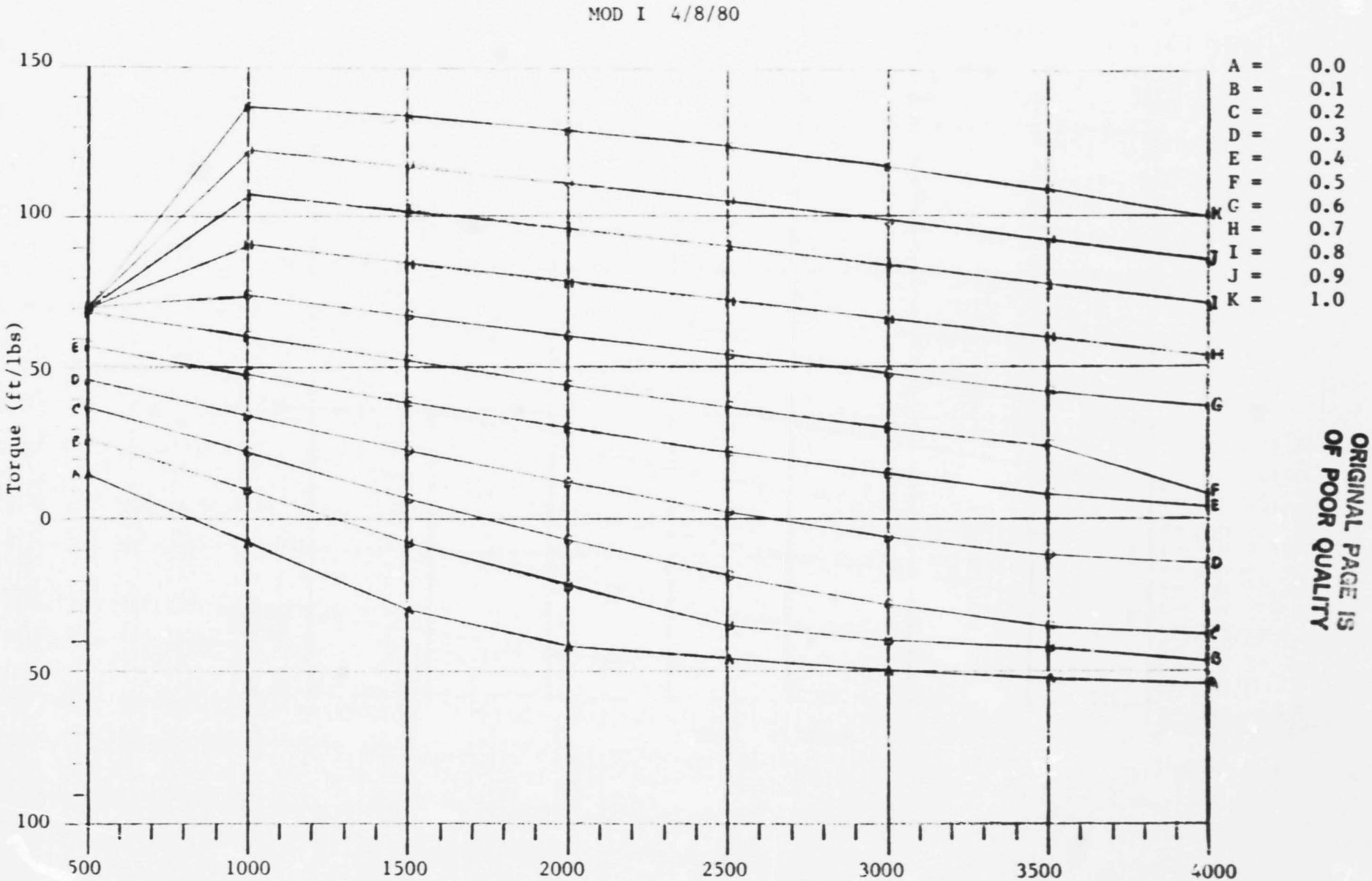


Fig. 3 Mod I Torque Versus rpm (Constant Throttle)

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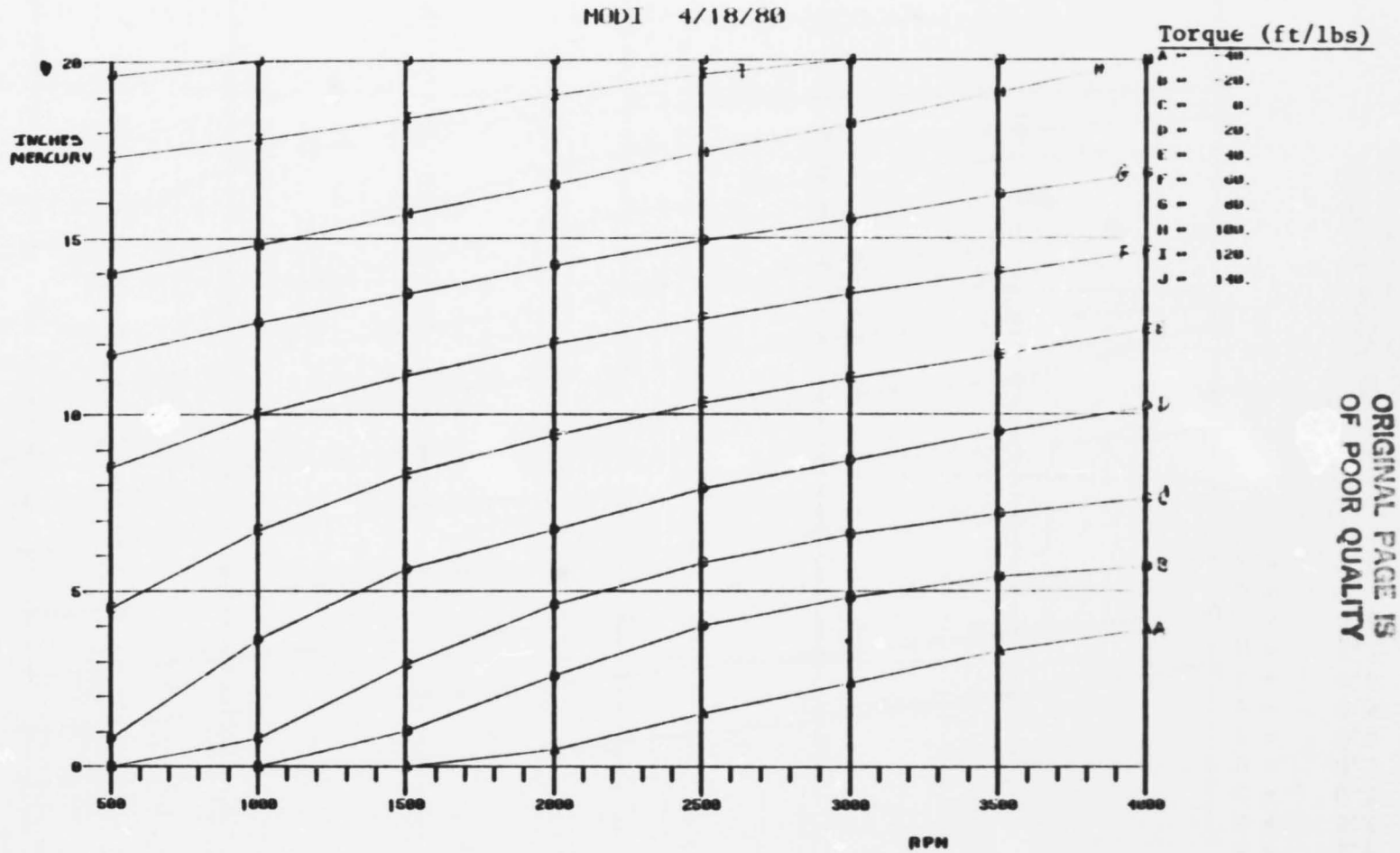


Fig. 4 Mod I Vacuum Versus rpm (Constant Torque)

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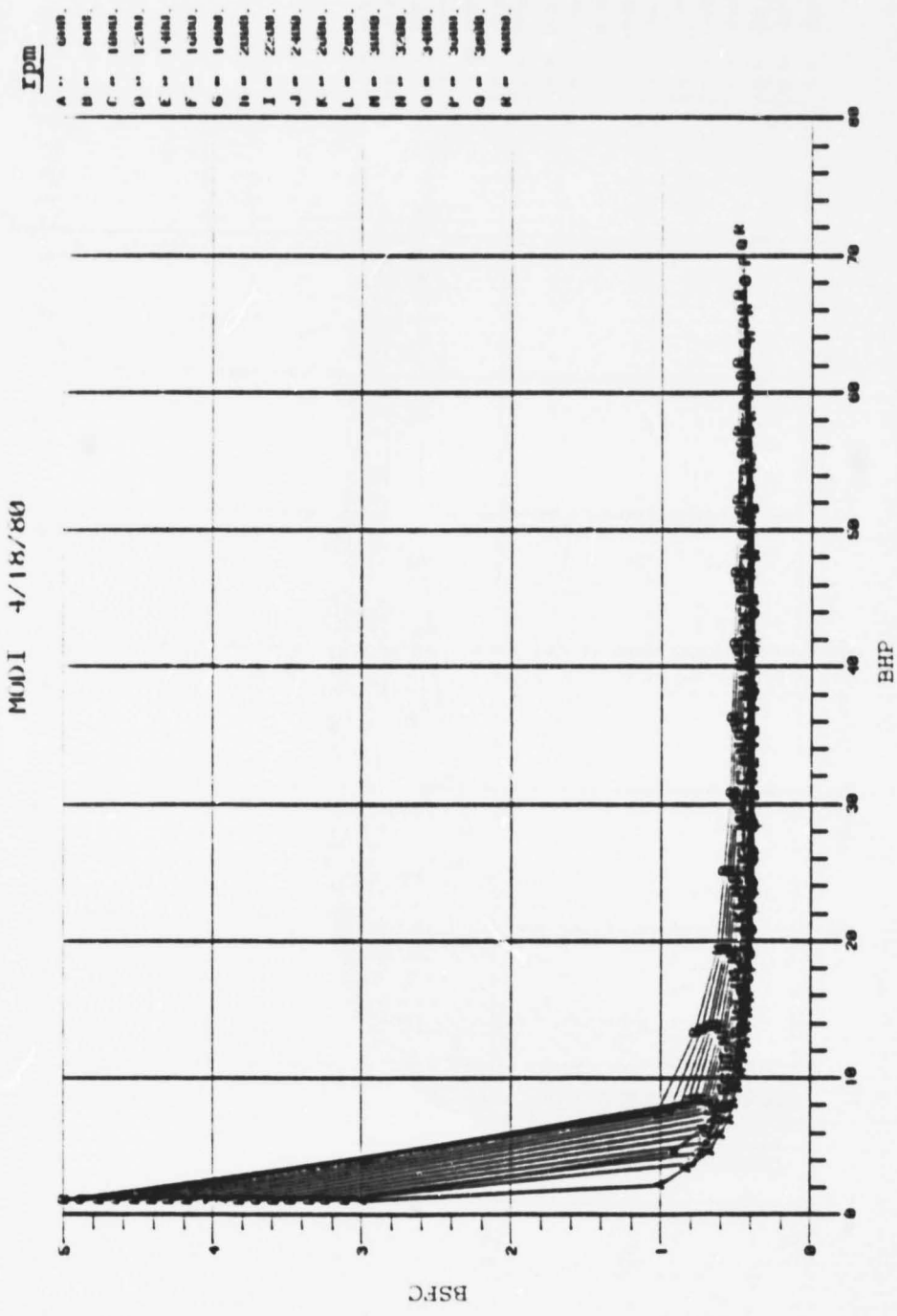


Fig. 5 Mod I BSFC Versus Brake hp (Constant rpm)

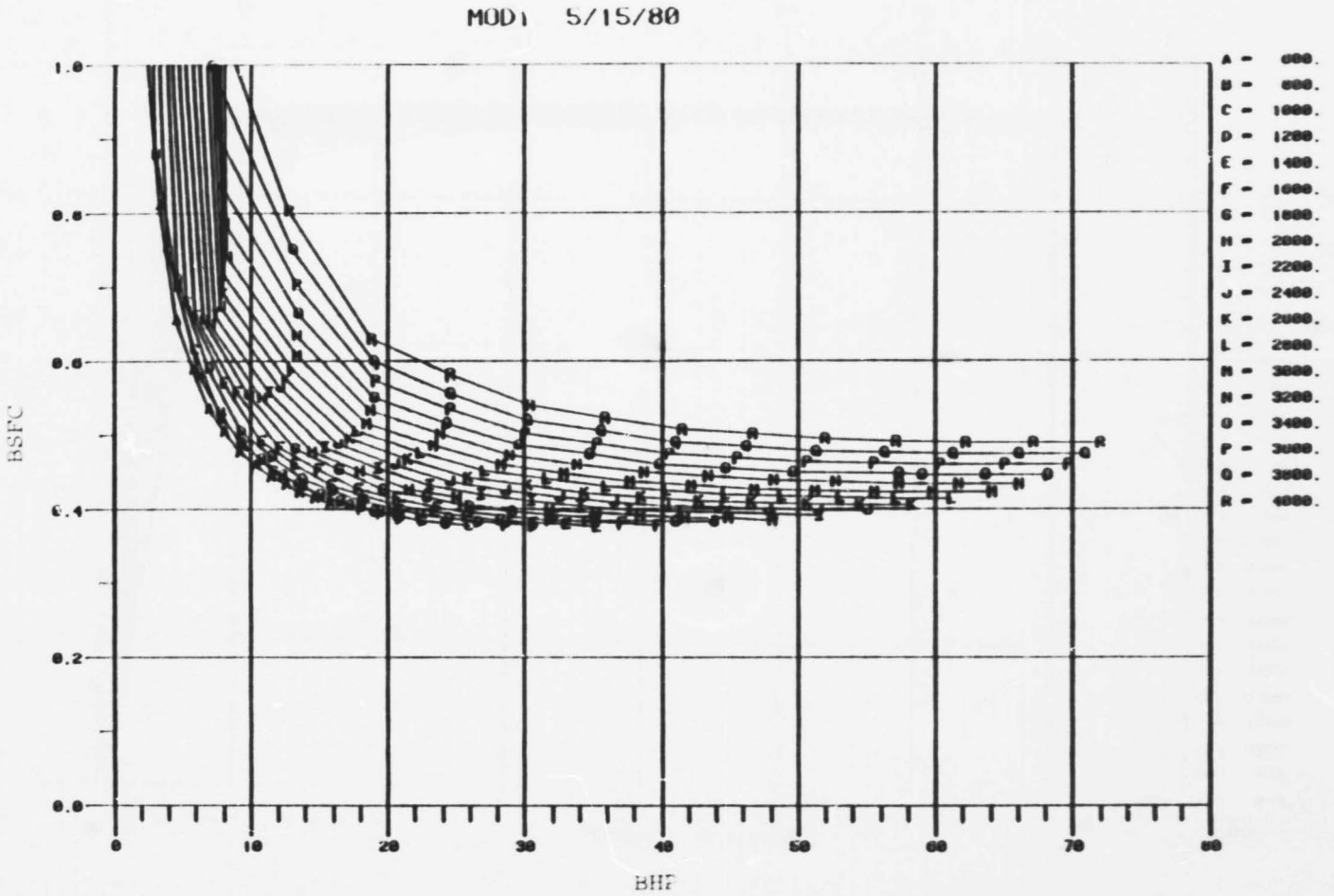


Fig. 6 Mod I BSFC Versus Brake hp (Constant rpm)

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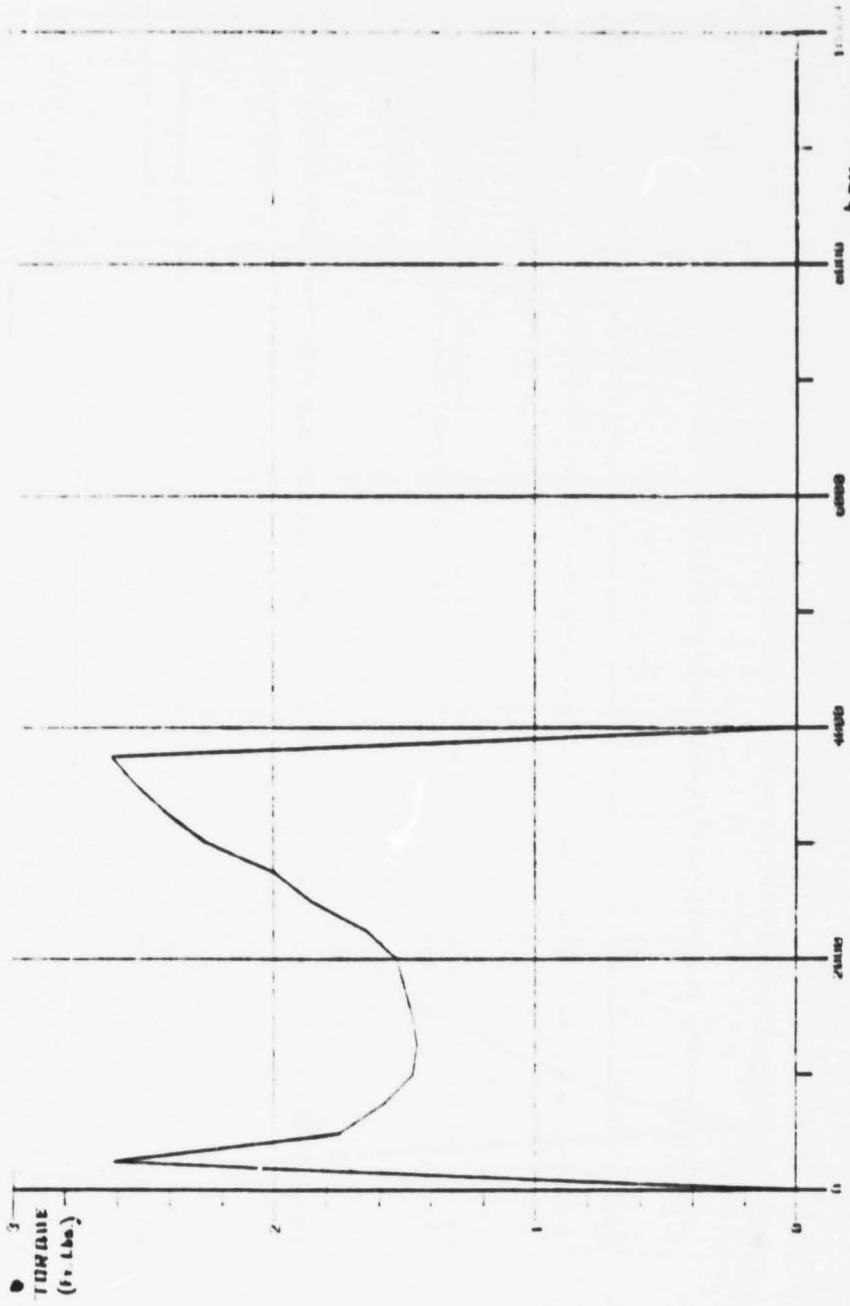


Fig. 7 Mod I Power Steering Load Characteristic

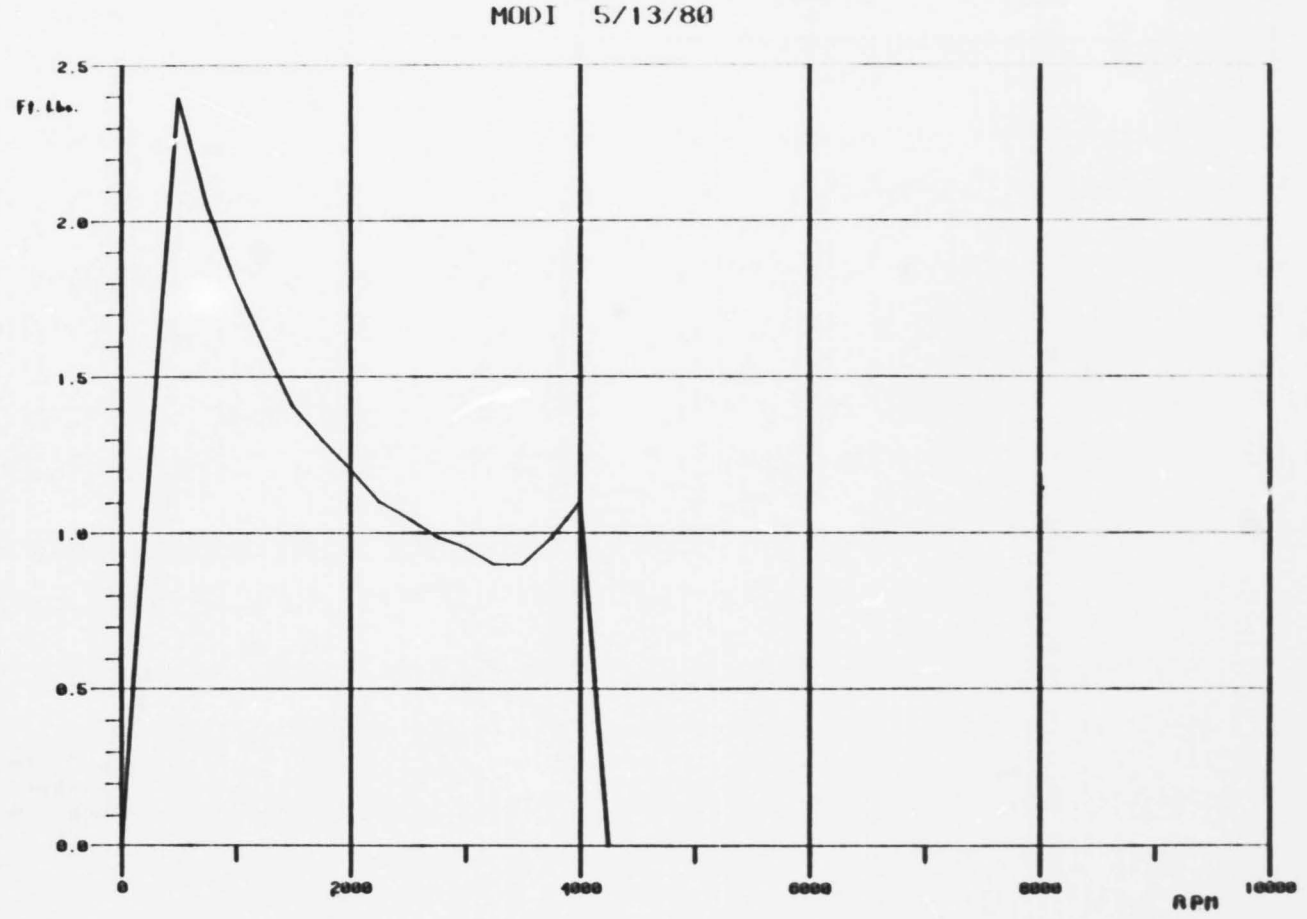


Fig. 8 Mod I Cooling Fan Load Characteristic

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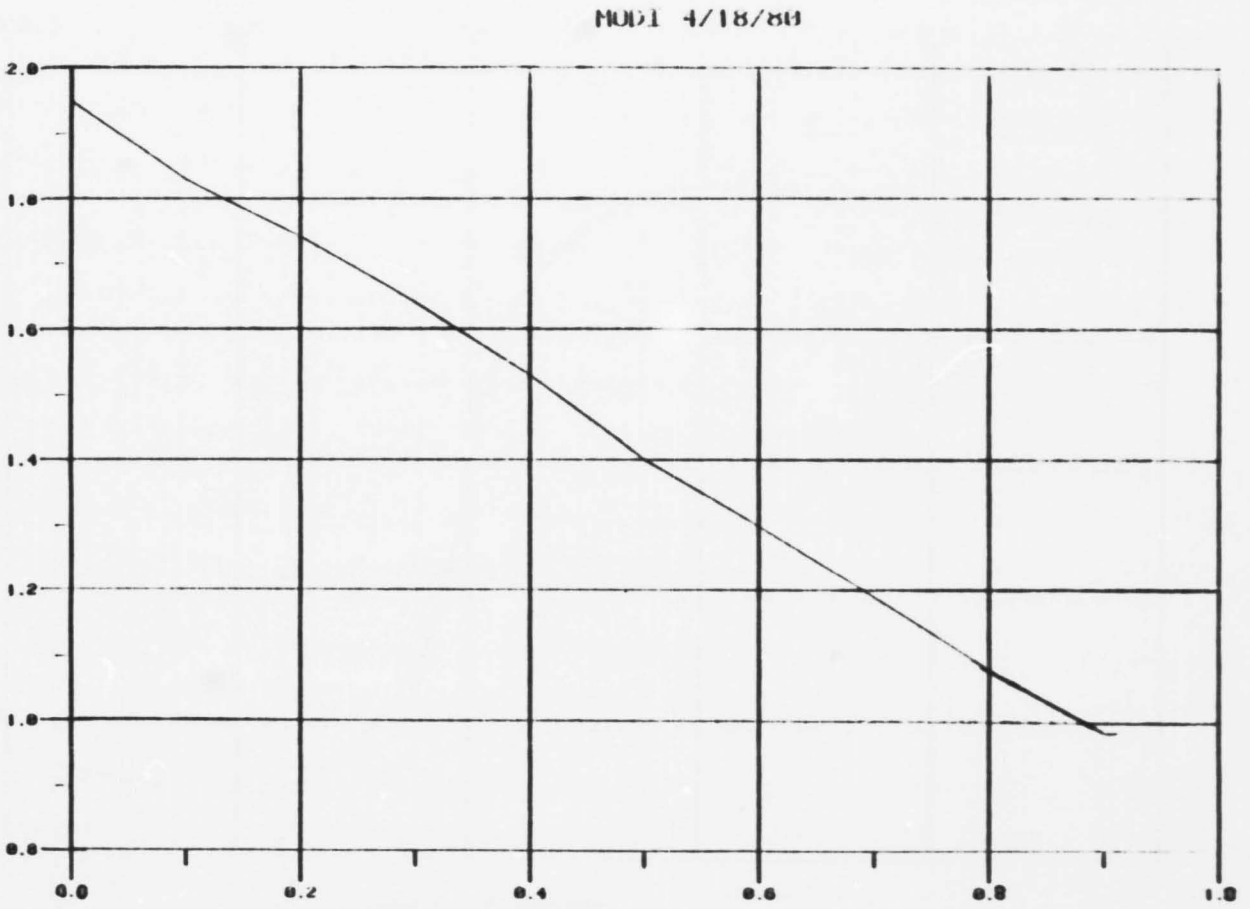


Fig. 9 Mod I Torque Converter Characteristic

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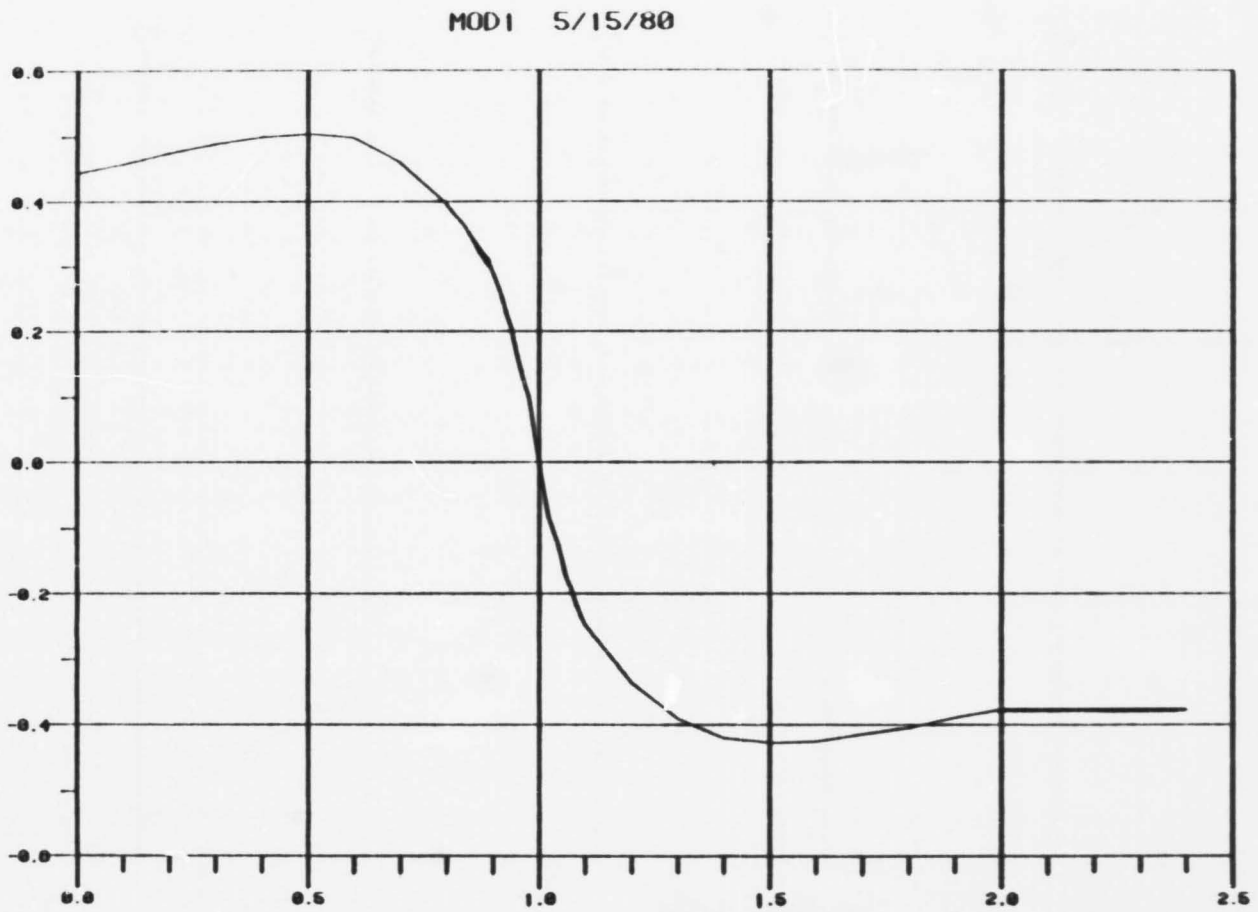


Fig. 10 Mod I Torque Converter (Inverse Size Factor Versus Speed Ratio)

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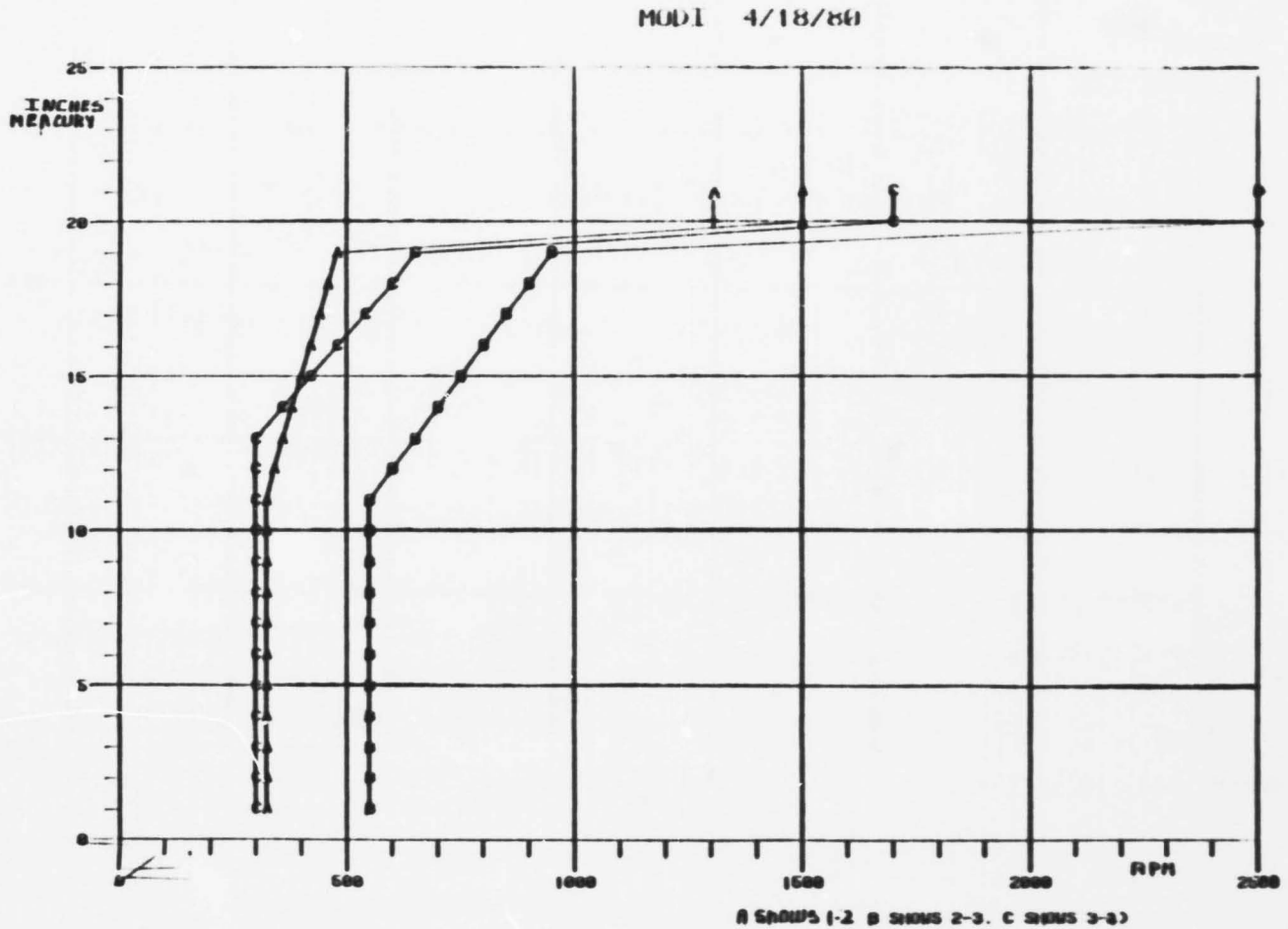


Fig. 11 Mod I Vacuum Versus Drive (Shift Curves)

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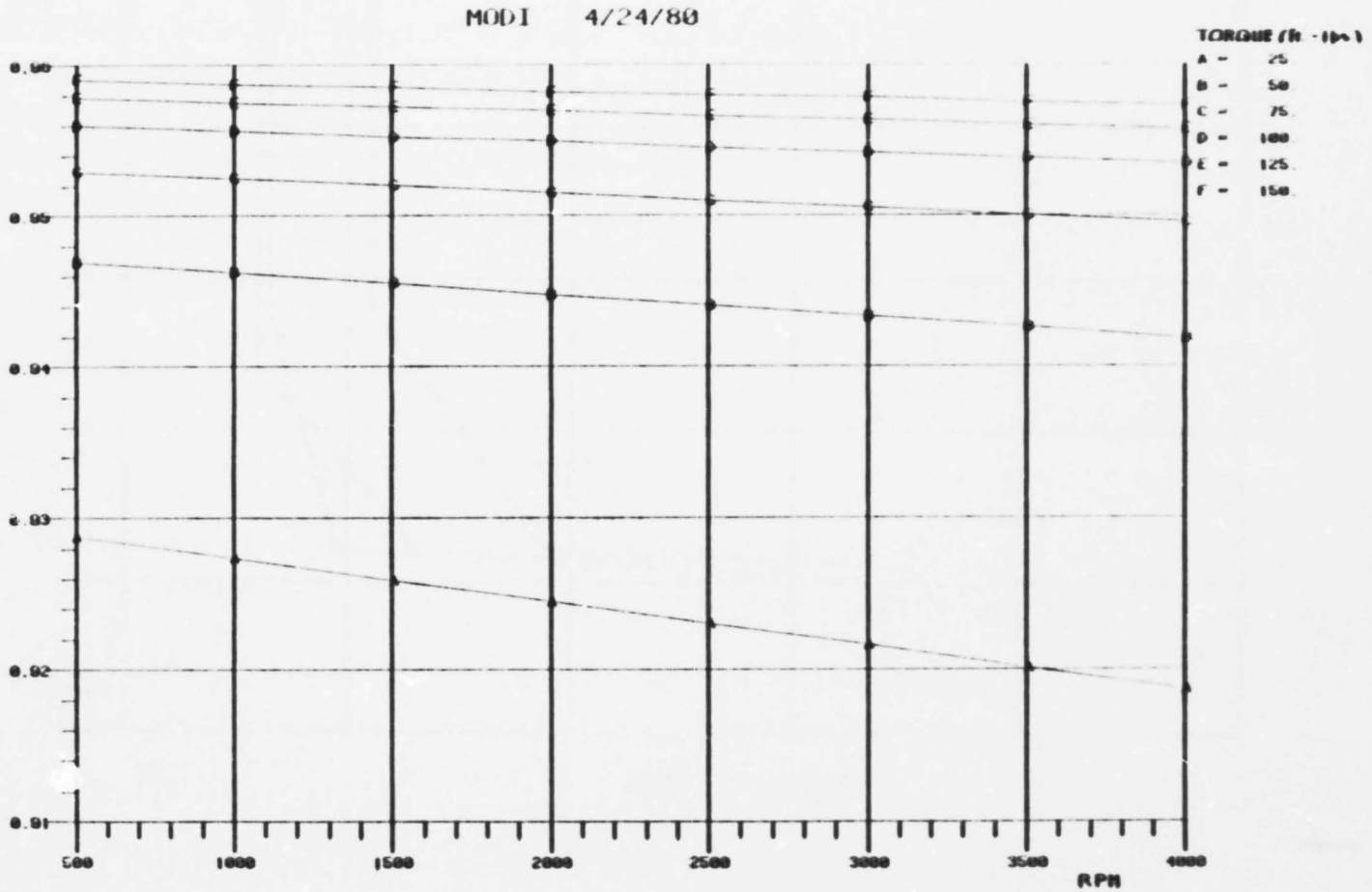


Fig. 12 Mod I Transient Efficiency Versus Driveshaft Speed (Constant Torque) First Gear

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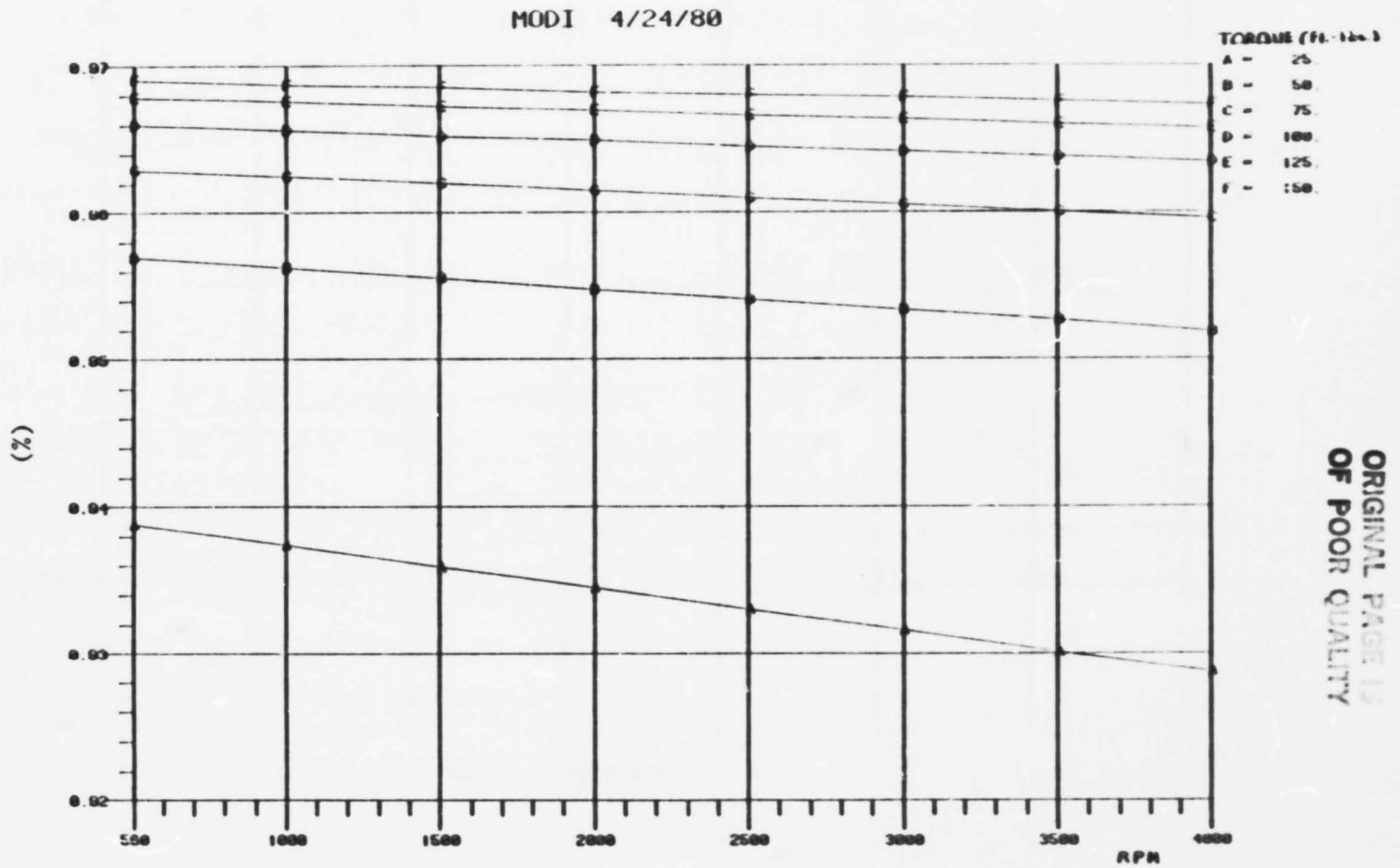


Fig. 13 Mod I Transient Efficiency Versus Driveshaft Speed (Constant Torque) Second Gear

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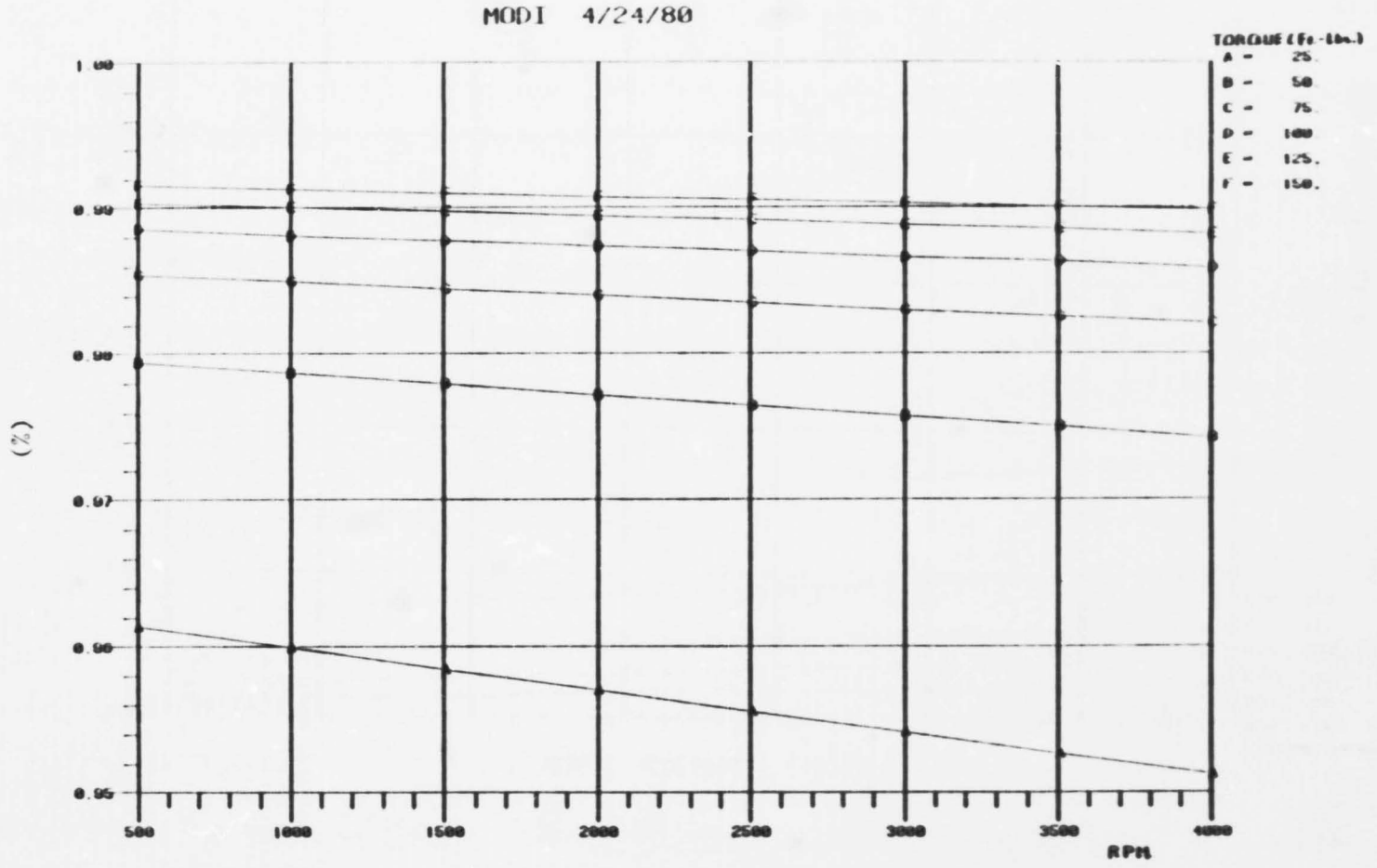


Fig. 14 Mod I Transient Efficiency Versus Driveshaft Speed (Constant Torque) Third Gear

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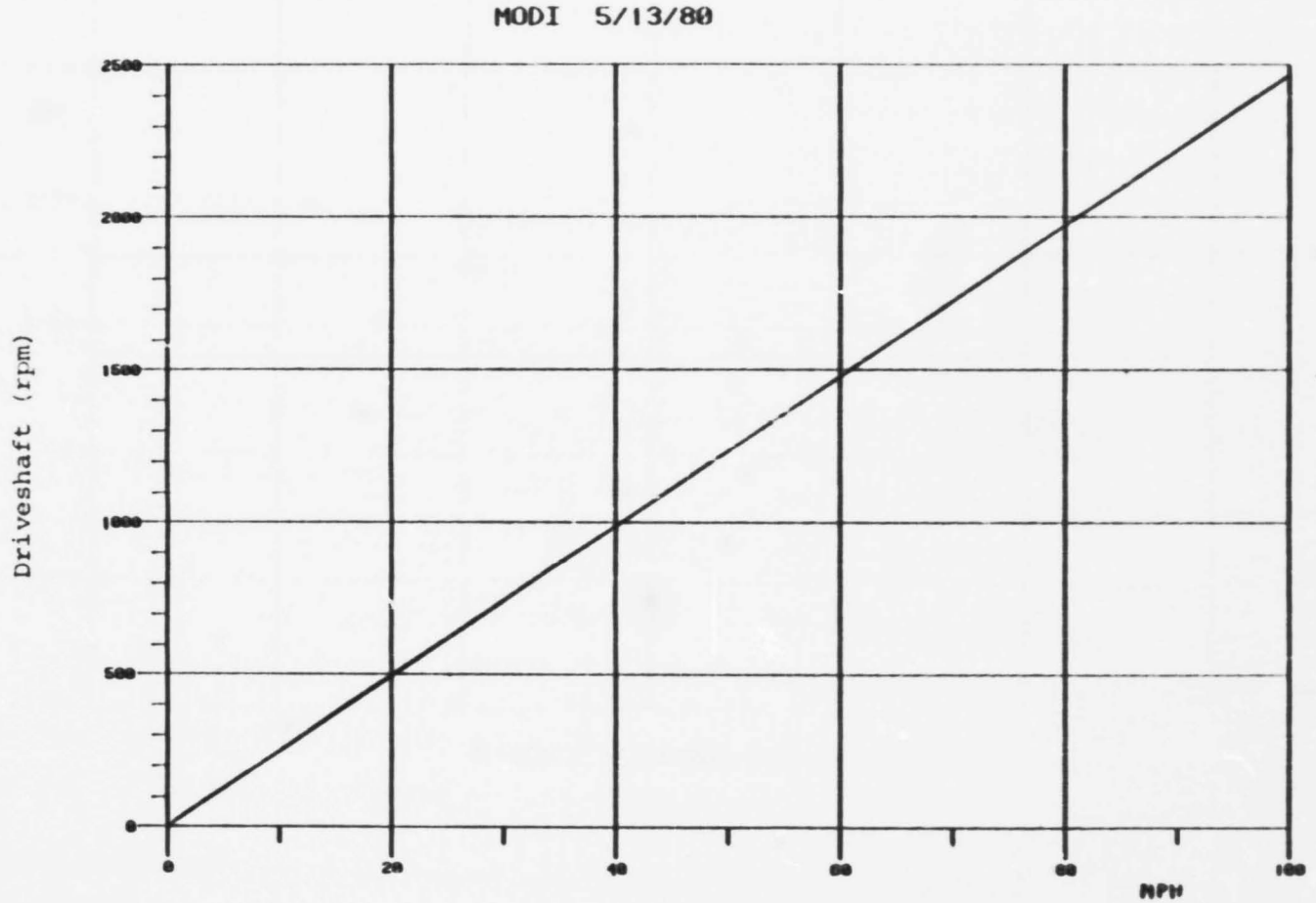


Fig. 15 Mod I Rear Axle Ratio (Vehicle Speed Versus Driveshaft Speed)

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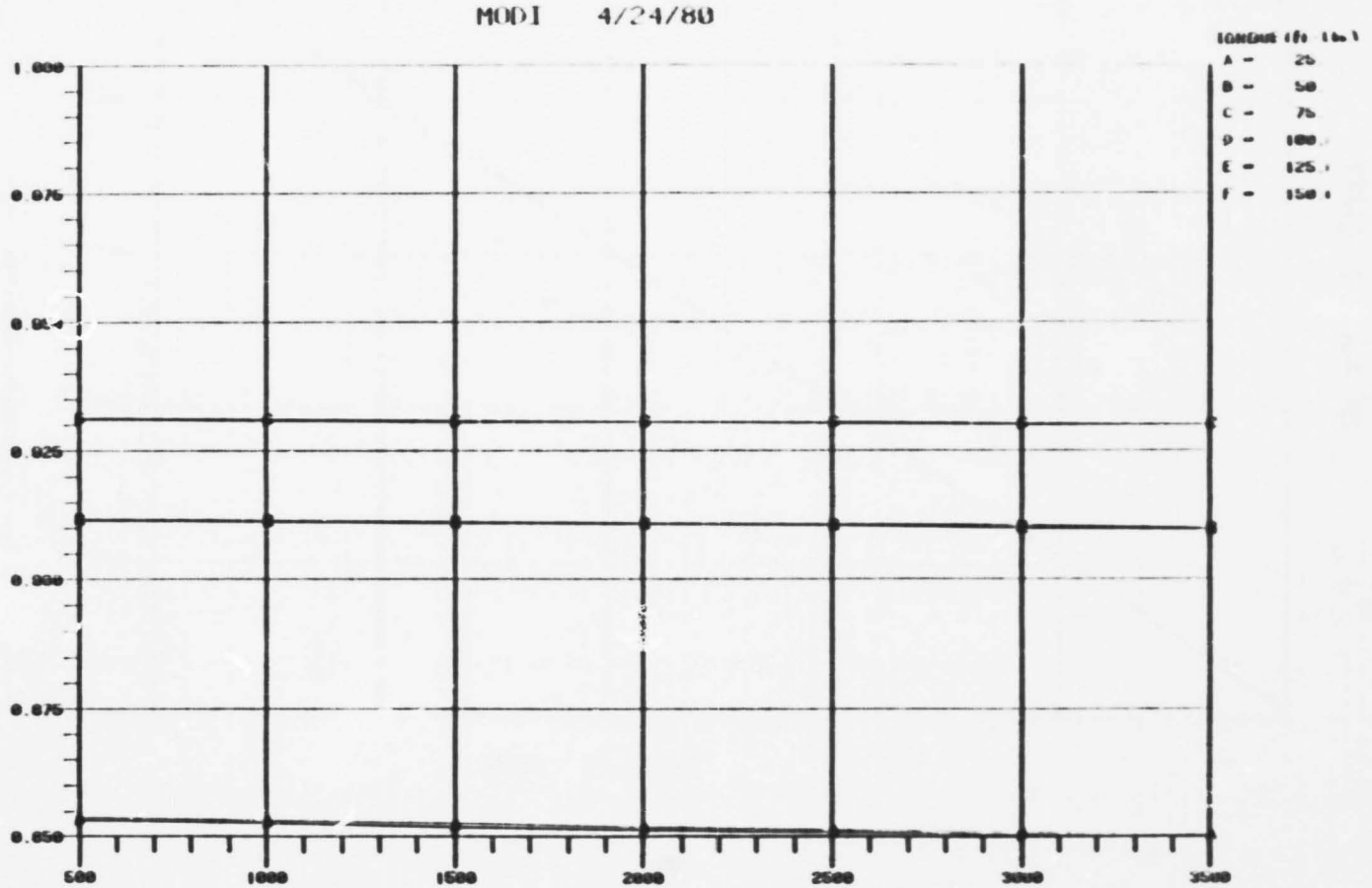


Fig. 16 Mod I Rear Axle Efficiency Versus Driveshaft Speed (Constant Torque)

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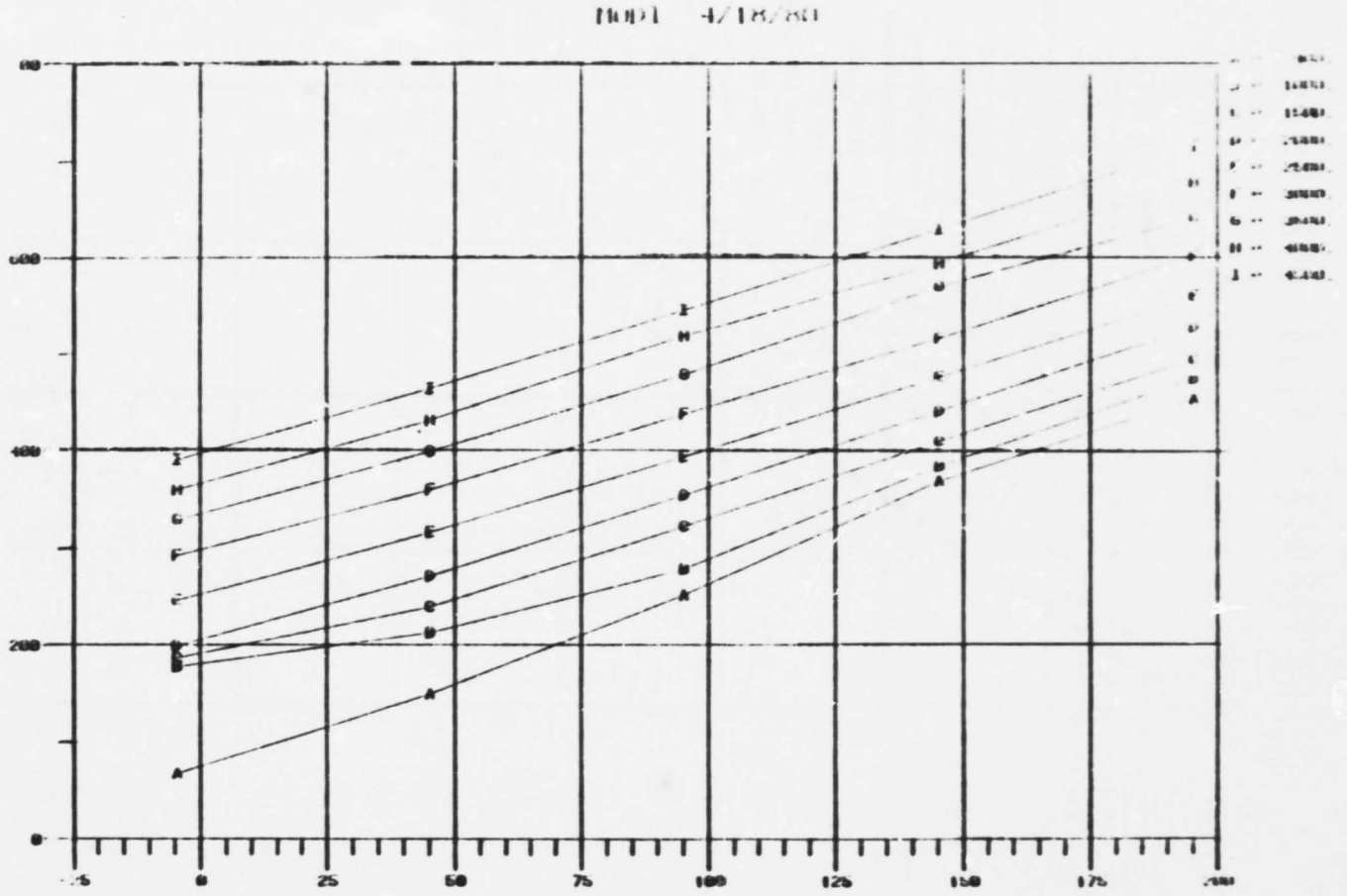


Fig. 17 Mod I Front Pump Torque Versus Line Pressure (Constant Engine Speed)

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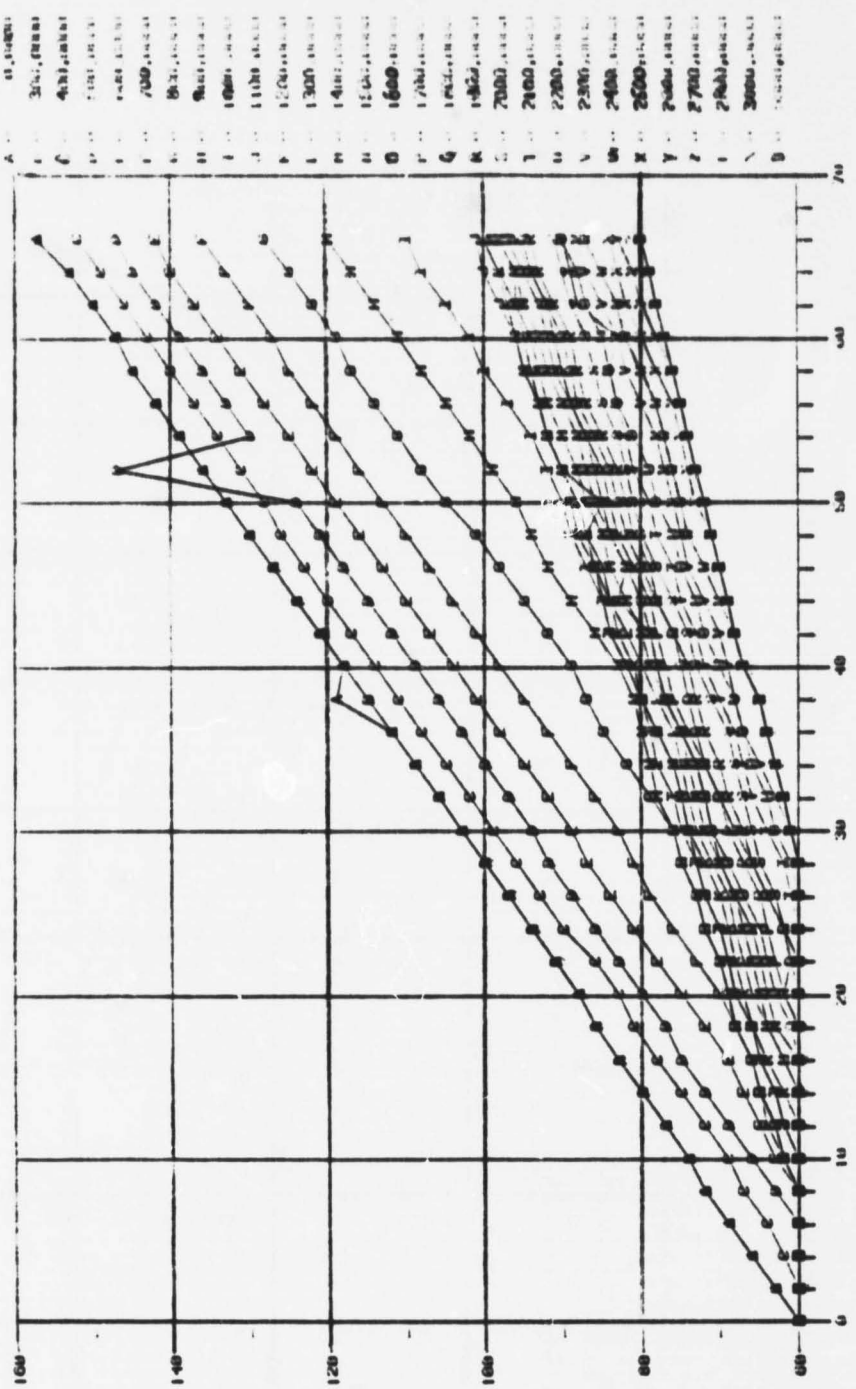


Fig. 18 Mod I Line Pressure Versus Throttle Pressure
(Constant Engine Speed)

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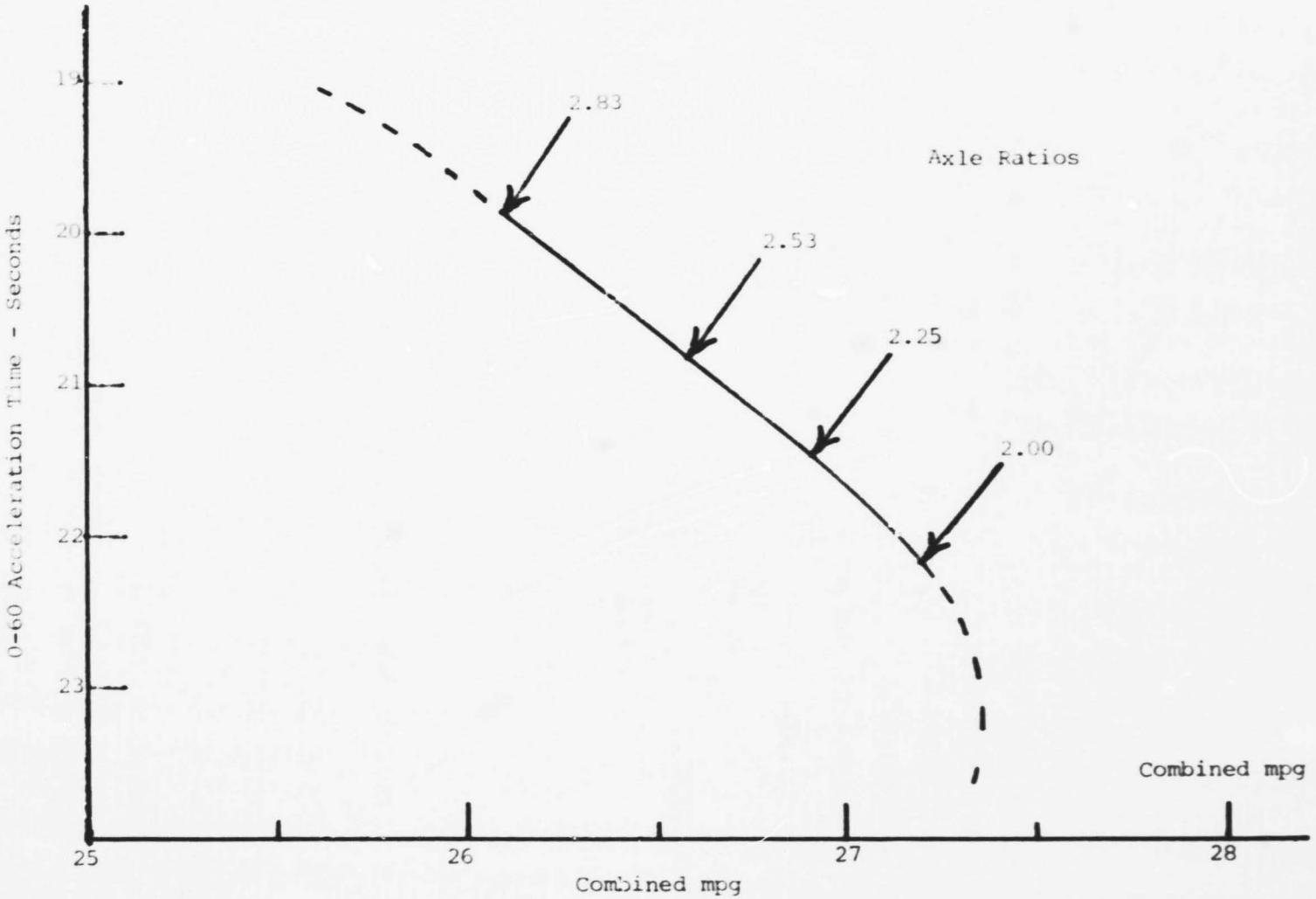
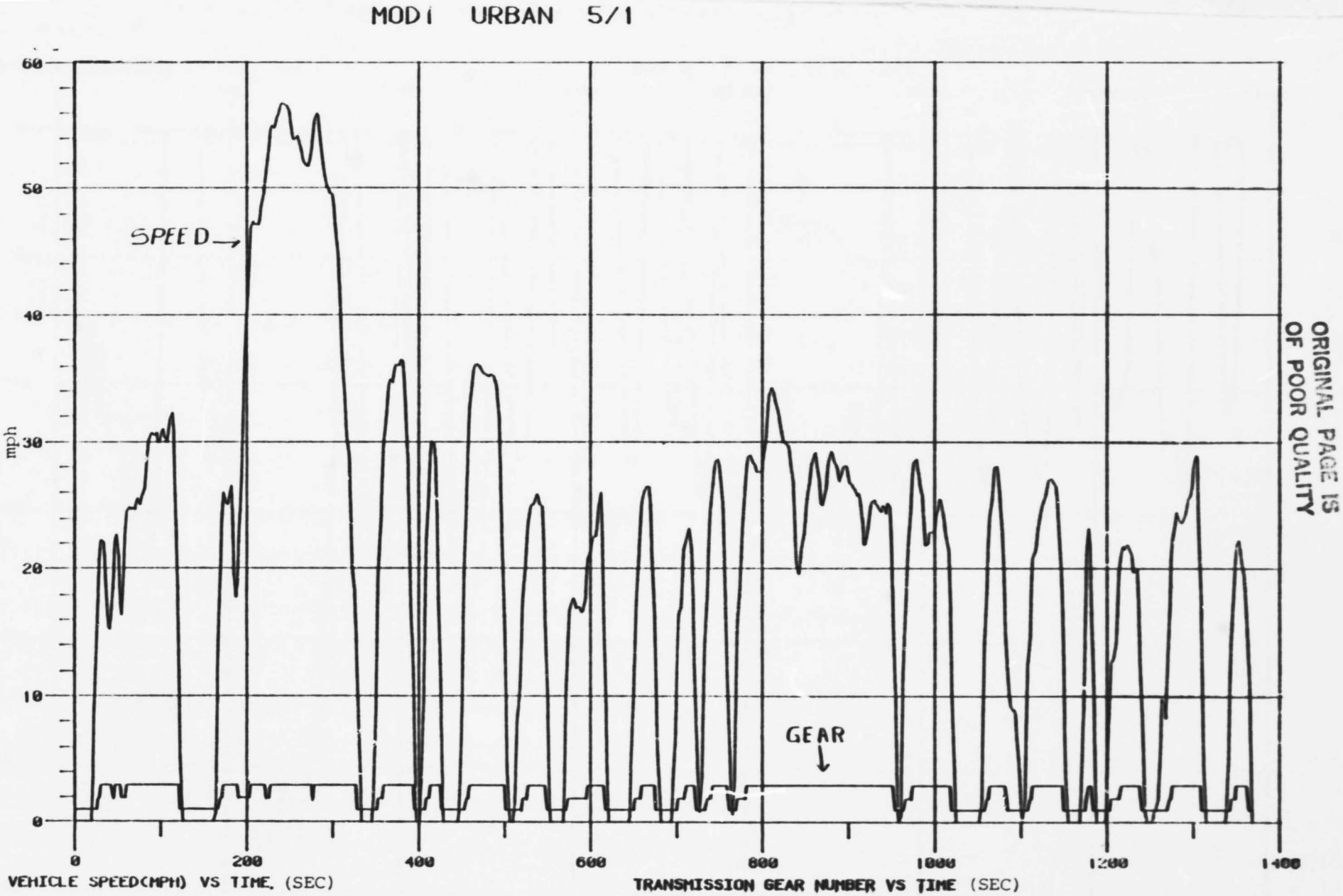


Fig. 19 Mod I Mileage/Performance

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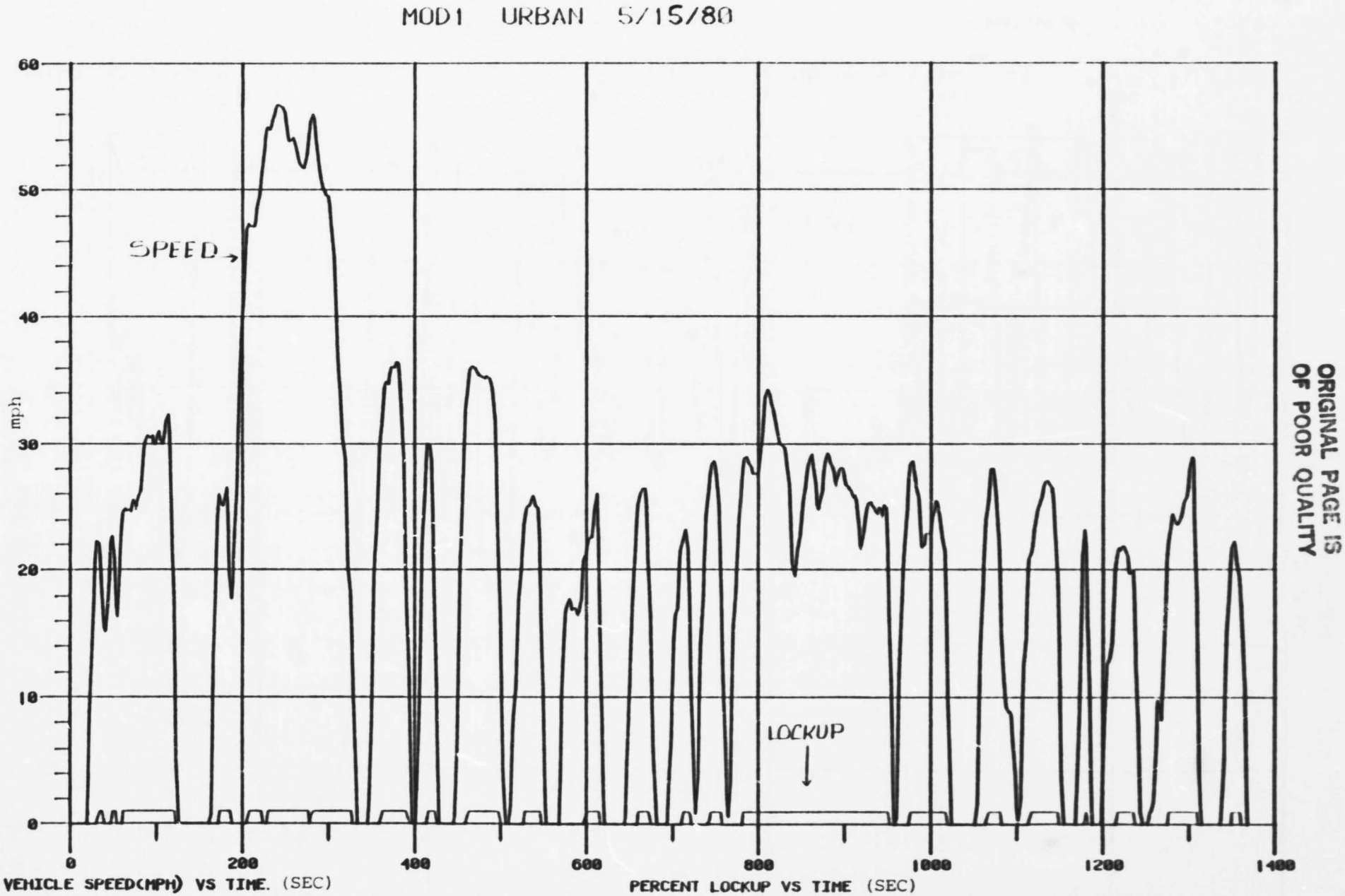
Urban Mileage	25.55
Urban Mileage with CSI	22.42
Highway Mileage	34.95
Combined Mileage	26.73
Combined Mileage (Diesel)	30.74
Cruise Mileage:	
50 mph	37.1 mpg (43.4)
60 mph	31.1 mpg (35.8)
Performance:	
0-60 mph accel	21.2 sec
5-10 mph accel	10 ft/sec^2

Fig. 20 Mod I Projections (2.37 Axle Ratio)



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Fig. 21 Mod I Vehicle Speed and Shift Characteristics for Urban Driving Cycle



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Fig. 22 Mod I Vehicle Speed and Loading Characteristics for Urban Driving Cycle

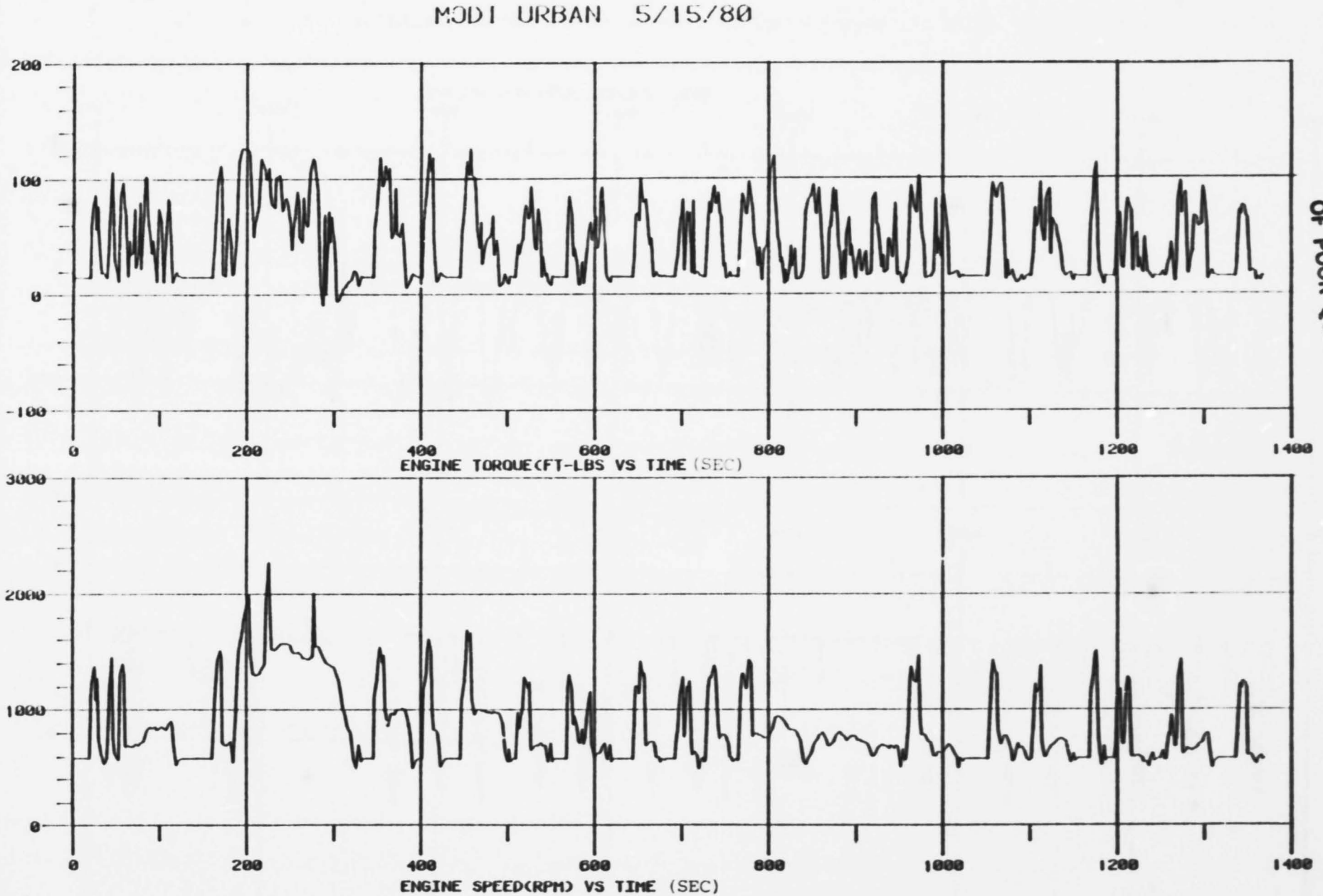
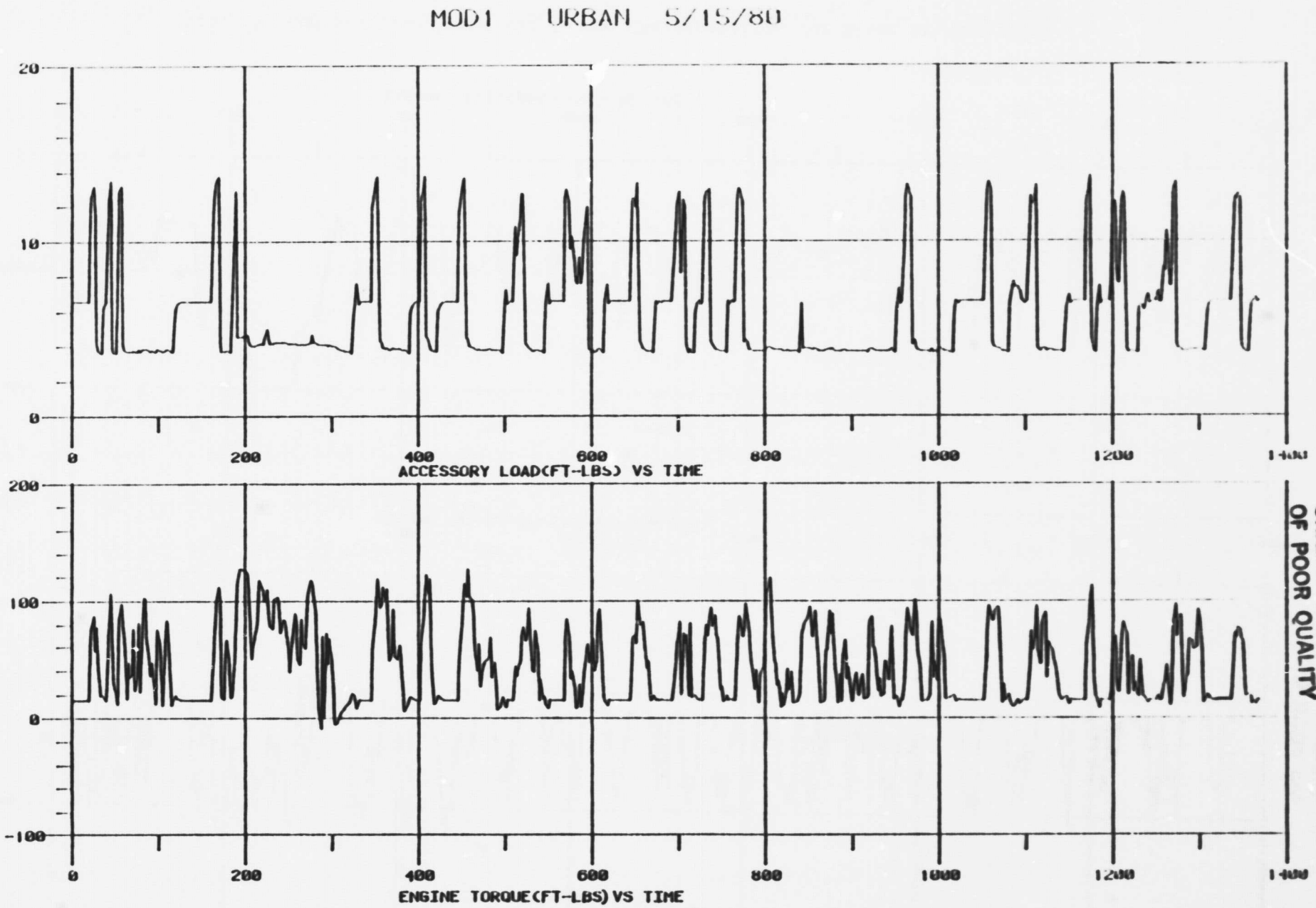


Fig. 23 Mod I Engine Speed and Torque Characteristic for Urban Driving Cycle

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Fig. 24 Mod I Engine Torque and Accessory Load Characteristics for Urban Driving Cycle

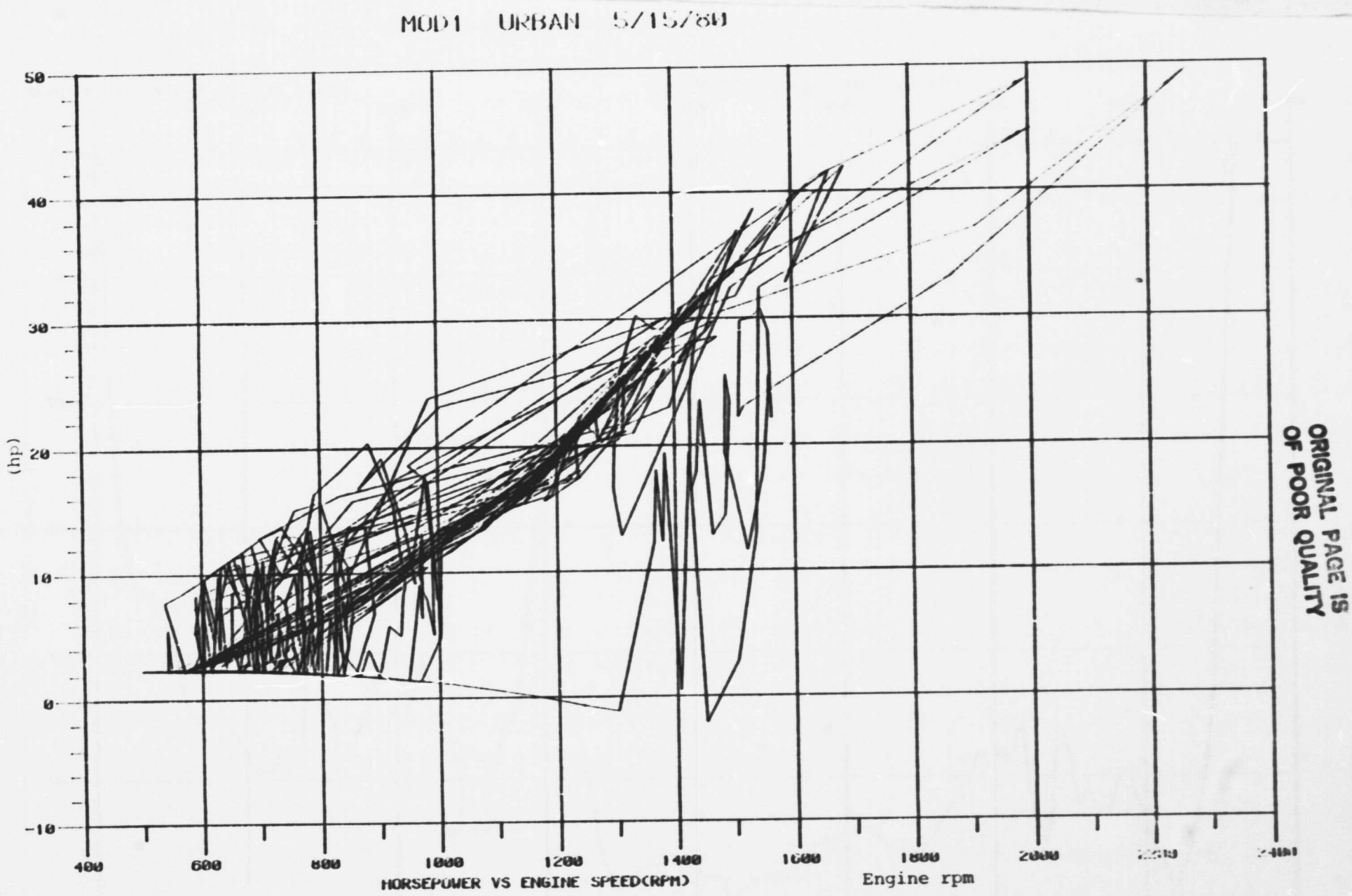


Fig. 25 Mod I Urban Cycle Horsepower/Speed Histogram

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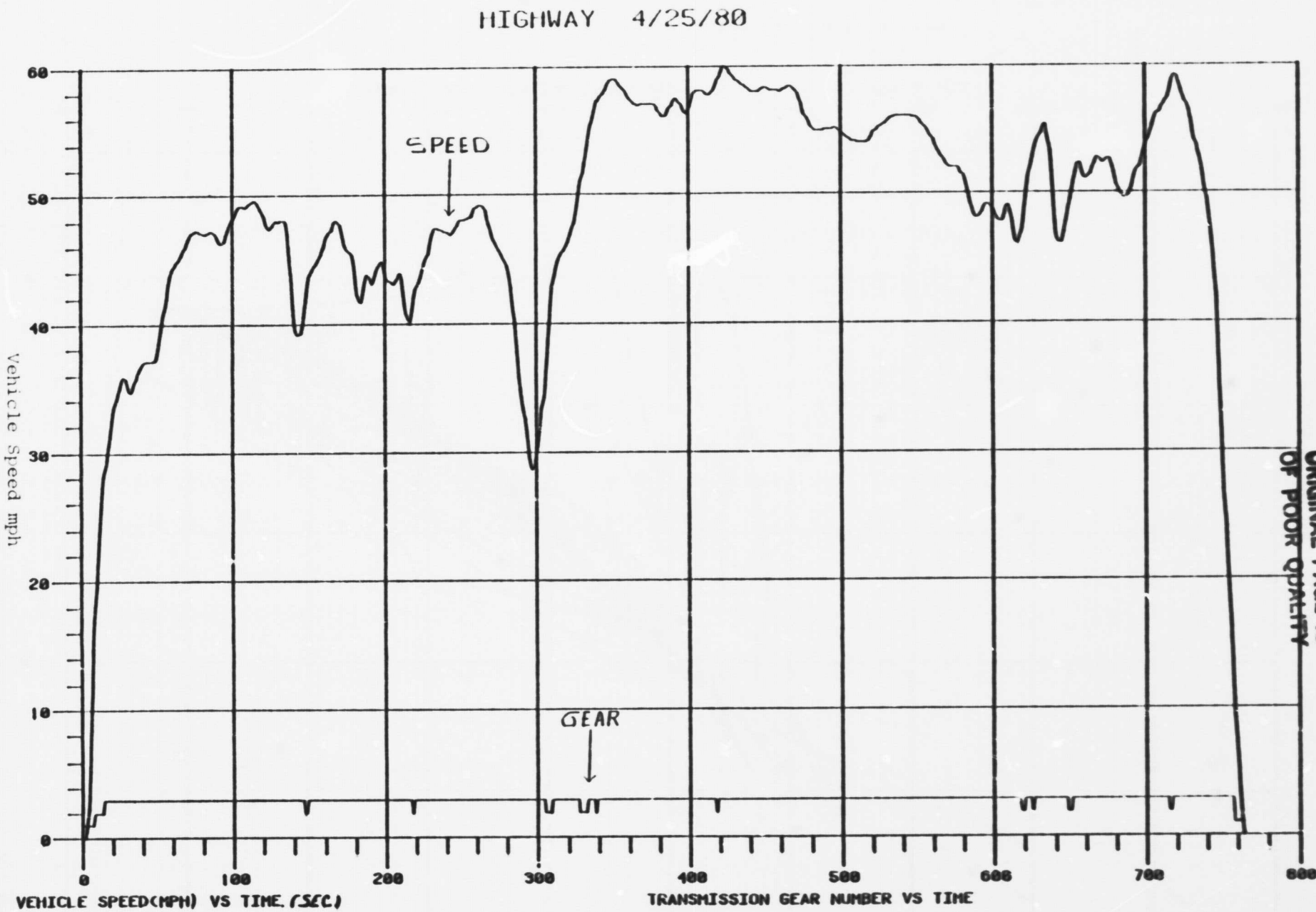
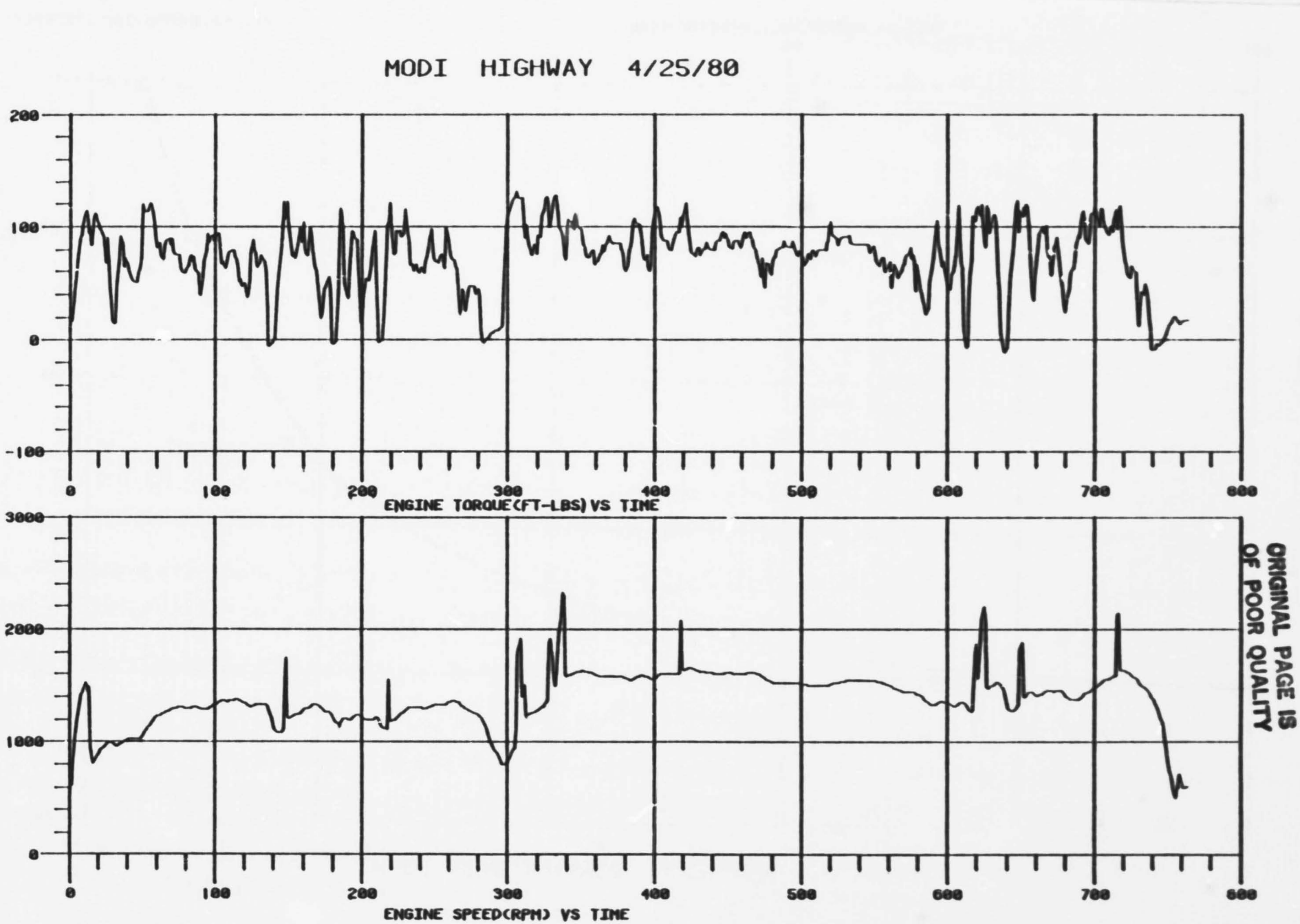
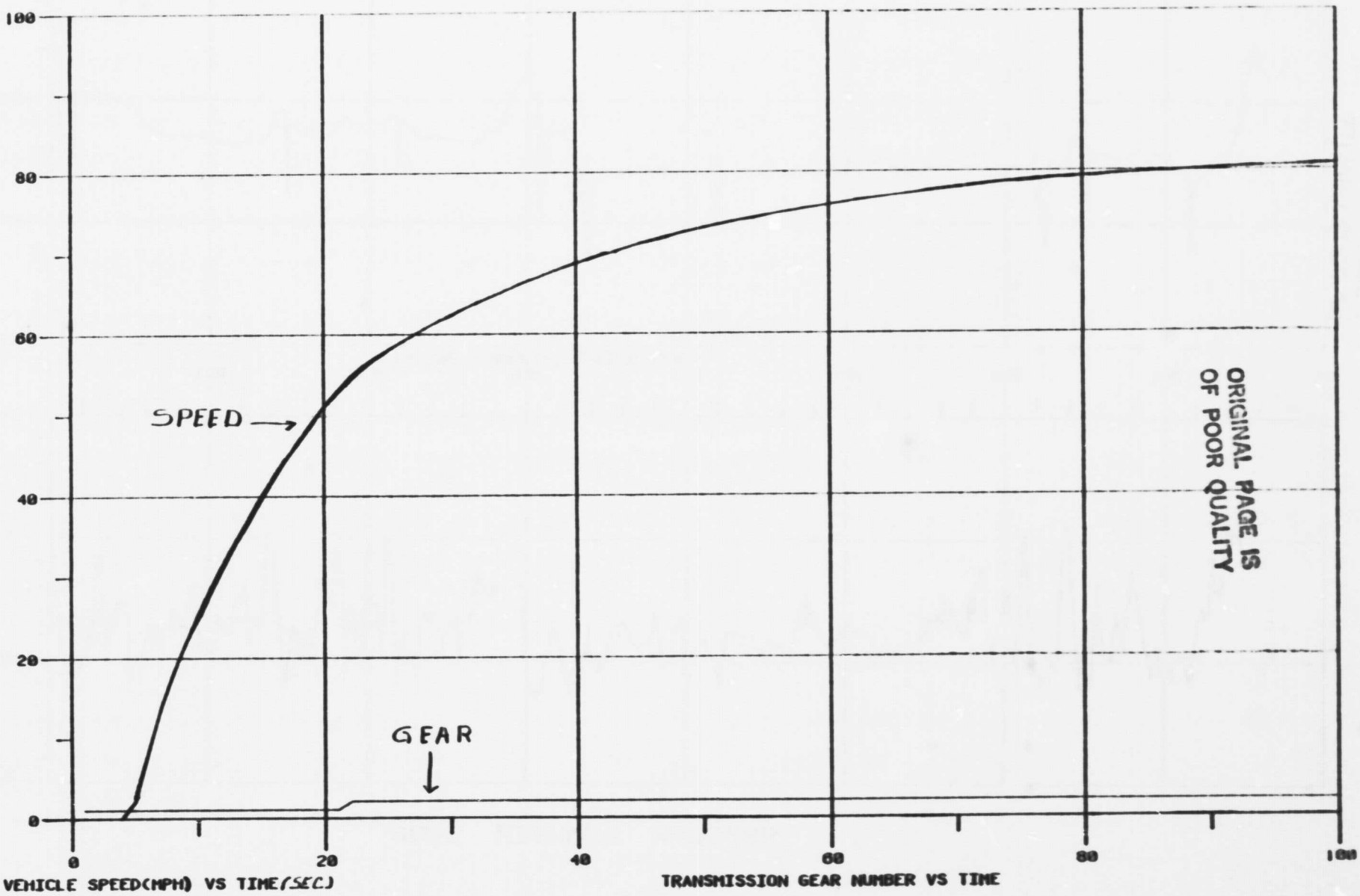


Fig. 26 Mod I Vehicle Speed and Shaft Characteristics for Highway Driving Cycle



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Fig. 27 Mod I Engine Speed and Torque Characteristics for Highway Driving Cycle


MODI PERFORMANCE 4/25/80
100
80
60
40
20
0
SPEED
GEAR
0
20
40
60
80
100
VEHICLE SPEED(MPH) VS TIME(SEC)
TRANSMISSION GEAR NUMBER VS TIME
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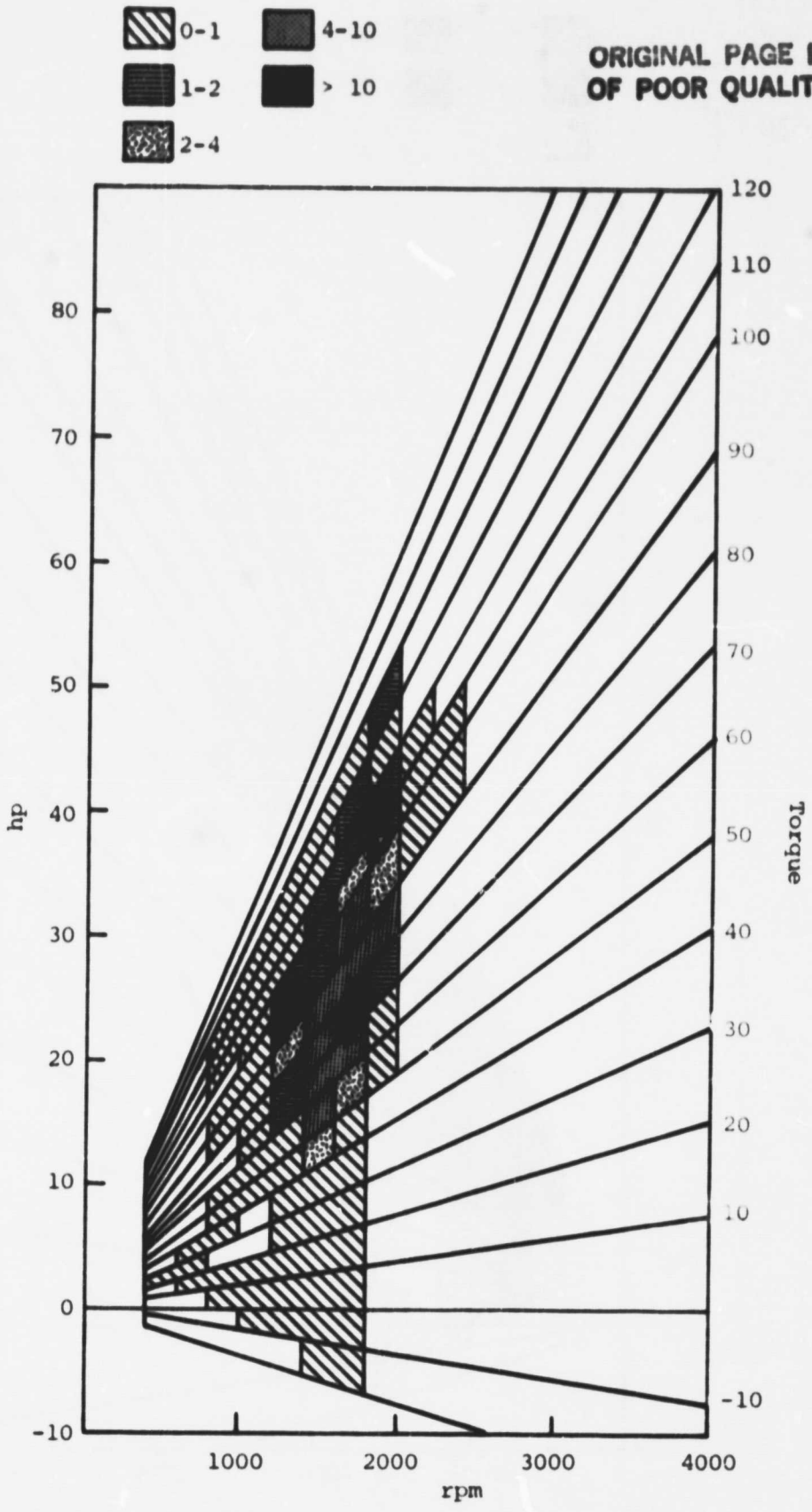


Fig. 29 Highway Driving Cycle/Horsepower/
Torque Characteristics

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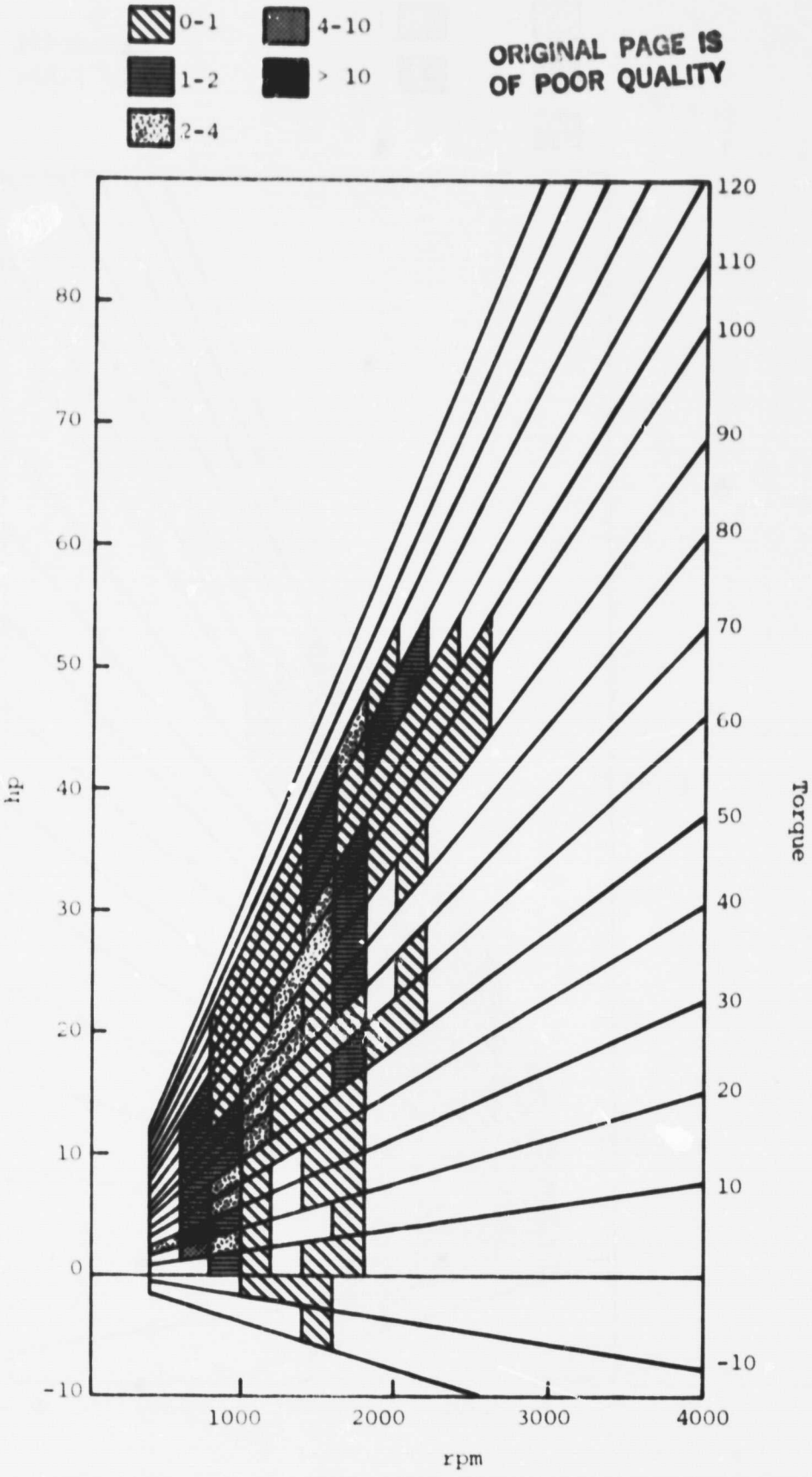


Fig. 30 Urban Driving Cycle/Horsepower/
Torque Characteristics

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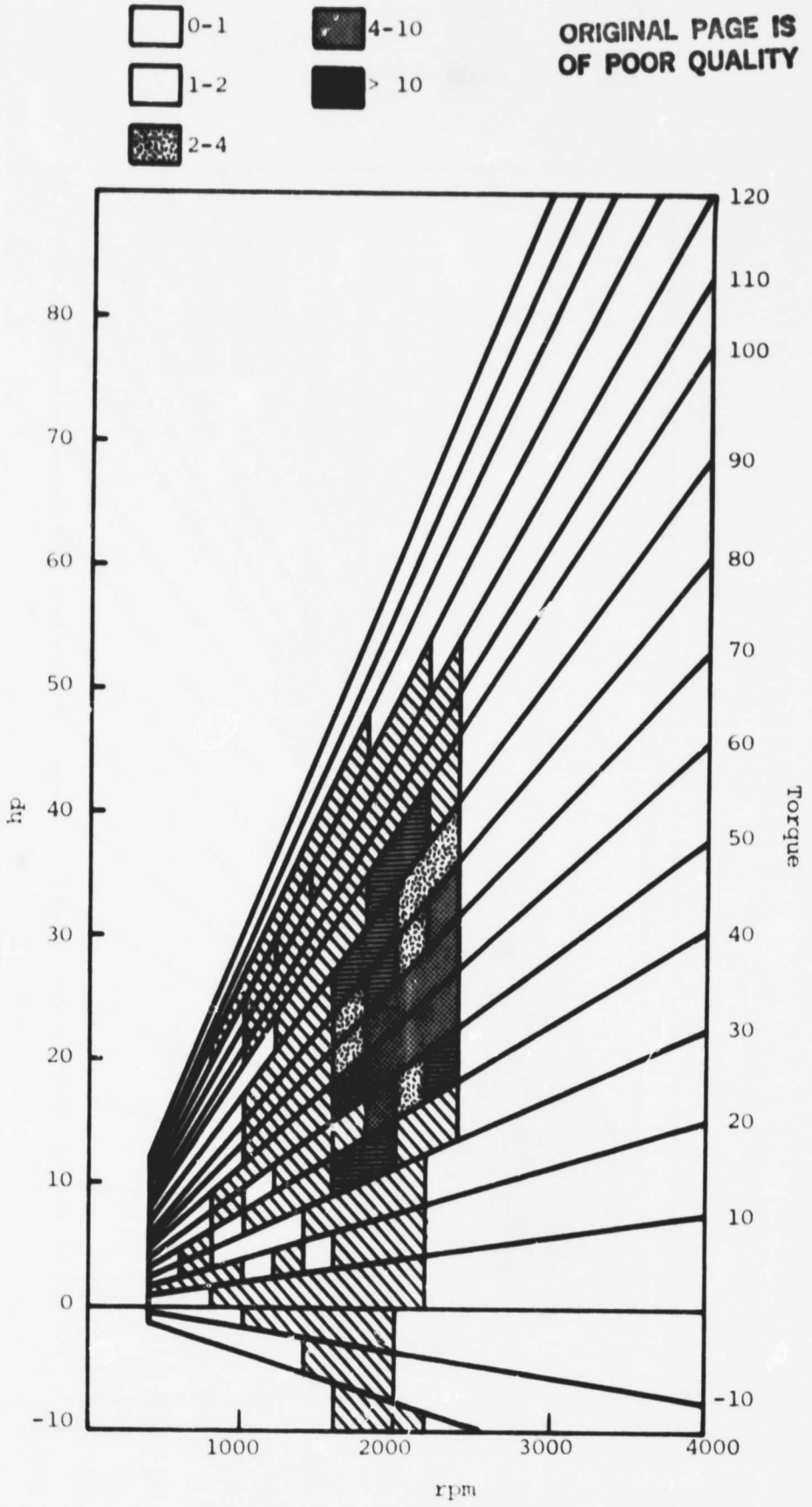


Fig. 31 Highway Driving Cycle/Horsepower/
Torque Characteristics

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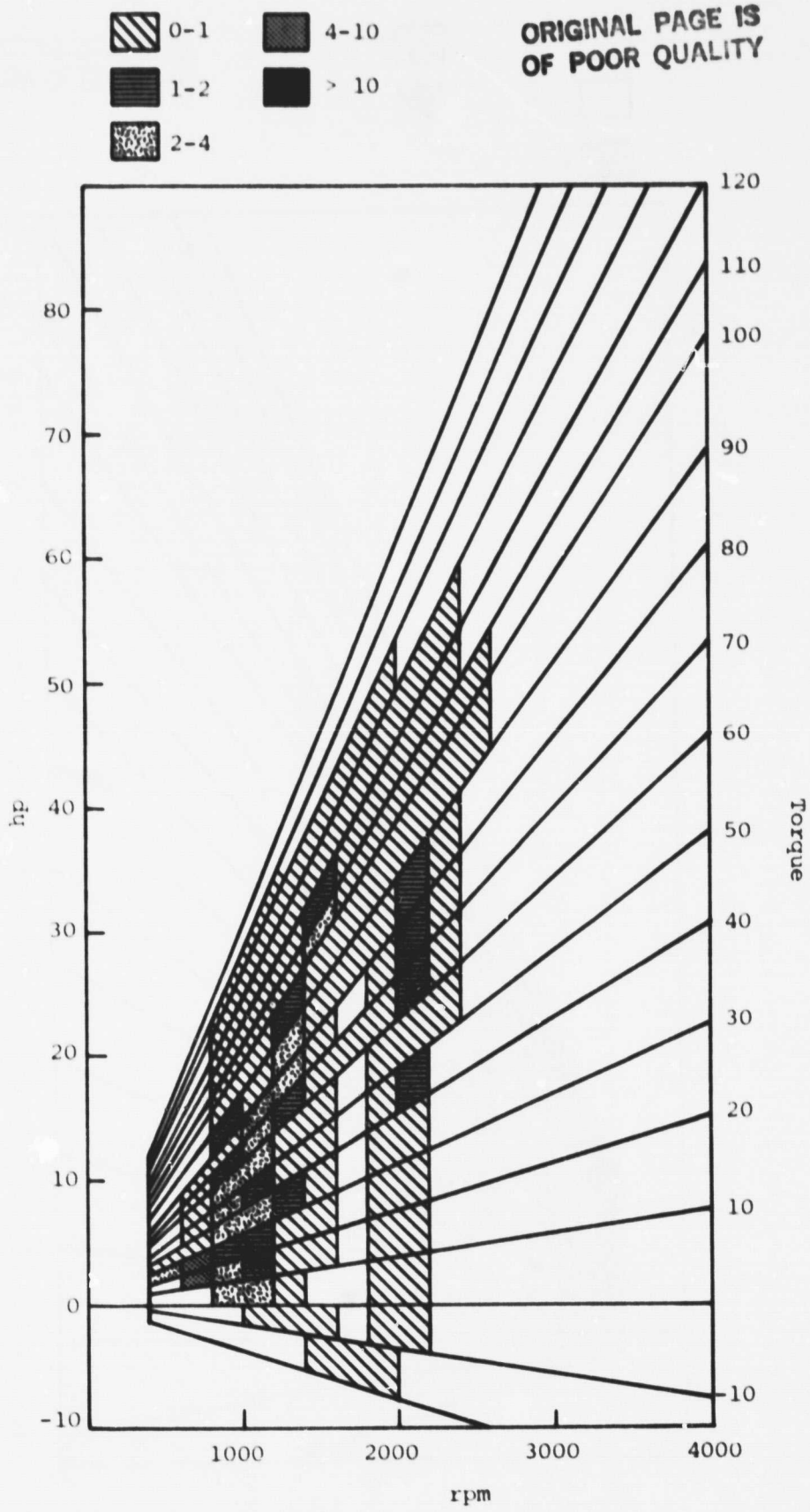


Fig. 32 Urban Driving Cycle/Horsepower/
Torque Characteristics

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10 COST ANALYSIS

10.1 BASE FOR THE ANALYSIS

Layout drawings.

Production program.
200 000 engines/year

Comparison
To be able to make a fast analysis, comparison has been made as follows.

A 2.1 liter Otto engine in high production.

A 2.0 liter Otto engine in high production

Earlier calculation for a 74 kW Stirling engine based on 15 000 units a year.

Earlier calculation for a 40 kW Stirling engine based on 100 000 units a year.

A small Stirling engine based on 10 000 units a year.

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10.2 BASIC DATA FOR PRODUCTION COST

The analysis is based on:
Salary average on Swedish salary and social cost for the first half of 1979.

Interest, based on 10%.

Machinery investments, Western Germany and US prices for 1979.

Tool, Swedish prices for the first half of 1979.

Power, Based on Swedish prices for 1979.

Maintenance Based on statistik figures in Sweden.

10.3 EXTENT OF THE ANALYSIS

The cost analysis include:

- Material
- Salary
- Social cost
- Tool consumption
- Rental of buildings for production
- Power
- Maintenance
- Factory overhead
- Amortization and Interest for machinery

The cost analysis don't include:

- Interest on work in progress (PIP)
- Cost for storing of ready manufactured engines
- Developing cost (design and prototypes)
- Sales cost
- Administration cost
- Profit
- Starting up cost

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10.4 PRODUCTION SYSTEM

All cold parts have been compared to similar parts for Otto or Diesel engines. The machinery for producing cold parts are chosen as they normally will be in any engine factory situated in an industrially high developed country.

For this production the following has been foreseen:

Crankcase	Transfer line
Bedplate	Transfer line
Crankshaft	Line production
Oil pump cover housing	Machine for complete machining
Flywheel	Line production
Gear ring, starter	Group production
Flywheel cover	Line production
Front cover	Line production
Connecting rod	Transfer line
Piston rod guide	Line production
Piston rod	Group production
Piston	Line for the lower part, group for the upper
Cylinder block	Transfer line
Liner	Line production
Guide and seats	Machine for complete machining
Water pump housing	Machine for complete machining
Impeller	Group production
Fan housing	Machine for complete machining
Fan wheel	Line production

For hot parts will known methods be chosed, which means that most of the machinery and equipment are available on the machine tool market. Many of these must be more automatic before they can be used for high production.

The following has been foreseen:

Cas cooler	Automatic machine for welding and cutting. Assembling machine. Vacuum furnace.
Heater tubes	Automatic weling, winding and cutting line.

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General and piston housing	Line production
Assembling, heater	Assembling machine. Vacuum furnace.
Air preheater	Cutting and bending line. Automatic resistence welding
Assembling	To be able to keep a high quality the assembling has been foreseen as station assembly. Each station has to take care of one function of the engine. For instans heater, insulation, burner and so on. Transport between the stations is electrical driven carriers controlled by magnetic slings in the floor and a computor. The carrier transports the engine on a loop from beginning of the assembly through testing reject and will be unloaded in the store for ready manufactu engines.
Testing	Test cells. Computorized test program

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10.5 MATERIAL PRICES

	Sw cr/kg
Cast aluminum	15.00
Cast iron	2.50
Heat resistant steel sheet:	
t = 1.0	10.00
t = 0.15	20.00
t = 0.03	80.00
CRM or Stellit	27.00
Normal steel sheet	2.00
Insulation	10.00

The 87% rule has been applied to purchased parts when no other information was available.

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10.6 SALARY, SOCIAL COSTS ETC

Valid for new factory unit when using 2-shift

	Sw cr/h
Salary	32.00
Social costs 58%	18.50
Tool cost	17.00
Cost for renting building, 200 Sw cr/m^2 a year. 30 m^2/mach.	2.50
Electricity and maintenance	4.00
Various	2.00
Overhead	16.00
Total	92.00
1 man/ 2 machines	67.00

Cost/hour including amortization and interest.
Amortization + interest = 18%

Investment	Sw cr/h	Sw cr/man hour 1 man/ 1 mach.	Sw cr/mach. hou 1 man/ 2 mach.
100 000.- (18 000/ 3000 hrs)	6.00	98	73
200 000.-	12.00	104	79
300 000.-	18.00	110	87
400 000.-	24.00	116	91
500 000.-	30.00	122	97
600 000.-	36.00	128	103
700 000.-	42.00	134	109
800 000.-	48.00	140	115
900 000.-	54.00	146	121
1 000 000.-	60.00	152	127