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CAE REPORT NO. 976
SUPPLEMENT

DESIGN REPORT
CONTINENTAL TS120 TURBOSHAFT ENGINE

POWER CONTROL SYSTEM

DECEMBER 1965

DA-44-009-AMC-760(T)

U.S. ARMY ENGINEER RESEARCH AND DEVELOPMENT LABORATORIES
FORT BELVOIR, VIRGINIA

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FOREWORD

This report, prepared by Continental Aviation and Engineering Corporation, presents the design of the power control system for a family of small turboshaft engines derived from a single basic design, and supplements Section XI (Power Control System) of CAE Report No. 976 which was submitted in fulfillment of Phase I of ERDL contract DA-44-009-AMC-760(T). A change from the initial approach to the control system was made during the Design Study Phase of the program which prevented Section XI from being finalized at the time of the basic engine design study.

ABSTRACT

This report presents the design of the power control system for a basic single-spool turboshaft engine and the variations possible with this basic design. These variations cover engines with outputs of 60, 90, and 120 horsepower, recuperated and nonrecuperated versions, along with direct drive and geared output speeds.

Included herein are schematic diagrams, description of the fuel control scheduling and governing methods, estimated engine fuel schedules, and a brief discussion of those changes which have been made to the power control system since the submission of the main engine design report.

SUMMARY

The basic engine design is a single-spool, direct-drive, 120-horsepower simple-cycle engine. The design variations of the 120-horsepower engine covers engines of 60 and 90 horsepower, recuperated versions, and geared output speeds. The design of the control system for the direct-drive, simple-cycle 120 horsepower engine is presented in detail along with the modifications necessary for the alternate versions.

Details of the engine design are presented in CAE Report No. 976

INTRODUCTION

The TS-120 engine is a single-spool turboshaft engine, flat-rated at 120 shaft horsepower to 8000 feet on a 90-degree day.

The engine accessories are located at the front of the engine immediately behind the power take-off flange, and are mounted at a right angle to the engine axis. A single side entry air inlet is located behind the accessories and can be rotated 90° to either side of the vertical centerline. A single can combustor is located behind the air inlet approximately tangent to the engine and can be rotated to alternate positions. The engine exhaust is discharged axially on the engine centerline. An alternate radial exhaust adapter can be attached to the existing axial exhaust.

The power control system used on the TS-120 engine is a scheduling system consisting of a fuel pump, relief valve, acceleration scheduling system, automatic fuel density compensation, exhaust gas temperature limiting switch, governor, and a fuel cut-off valve. Compensation of the governor set speed for engine optimization as a function of inlet air temperature is provided, along with an electrically operated override for the turbine temperature limiter and a remotely resettable governor speed setting. The system is planned for

maximum versatility with a minimum of modification and it includes provisions for adaptation to isochronous governing with load sharing and paralleling features required in the geared drive system. Provisions will be made for incorporating recuperator temperature compensation and a bypass system for overspeed protection in the recuperated engine design.

The history of the control system design and the applicable design data were outlined in Section XI of CAE Report No. 976. This Supplement describes the system as it is now planned.

Continental has subcontracted with Woodward Governor Company, Rockford, Illinois for the major part of the fuel scheduling, pumping, and governing portions of the system which is presented as an Appendix to this Supplement.

POWER CONTROL SYSTEM

The power control system consists of those elements required to filter, pump, and meter fuel flow to the engine for acceleration scheduling and governing and those elements which are interconnected with the accessories and the sequencing and protective circuits of the engine to ensure satisfactory operation in accordance with ERDL Contract DA-44-009-AMC-760(T).

A schematic diagram illustrating the fuel control system is shown in Fig. 1 of the Appendix. The assembly is an integral fuel pump acceleration scheduling and governing unit of the hydromechanical type but with an electrical speed-setting motor, electrically operated fuel cut-off valve, and an enrichment solenoid. The control system contains all the elements necessary for automatic starting, acceleration, governing, blowout protection, and overspeed protection. This part of the system is described in detail in the Appendix.

TURBINE OVERTEMPERATURE PROTECTION

Two basic alternatives were studied with regard to overtemperature protection of the turbine:

1. A temperature sensor and limiter which will modulate fuel flow to prevent excessive turbine exhaust gas temperatures.

2. A temperature switch which will shut the engine down if the turbine exhaust temperature exceeds a preset value. An explanation of the engine and control consideration is in order to illustrate this choice.

During the start and initial acceleration, the engine airflow and pressure ratio are low. Engine light-off normally occurs at about 6 to 10 percent speed and because the pressure ratio is low, fuel flow is very nearly a constant value from light-off to about 35 percent speed (see Fig. 1). The low pressure ratio results in a correspondingly low temperature difference from turbine inlet to exhaust at the same time that maximum turbine inlet temperature can be expected from the scheduled starting fuel-air ratio. This results in turbine exhaust temperature being much higher (without exceeding desired turbine inlet temperature levels) during starting than corresponding values at operating speeds. As the engine reaches rated speed and when load is applied the turbine temperature differential is considerably higher because of the compressor and output shaft work requirements. Thus a sensor measuring turbine exit temperature can only be related to the turbine inlet temperature at one condition, which is usually the loaded turbine at rated speed.

Use of a temperature switch which will shut the engine down by actuating the fuel cut-off solenoid at rated speed and an overload causing temperature to exceed the limit thus is incompatible with the temperature excursions which will exist during starts. Therefore, the switch must be deactivated during the start cycle. The sequencing system will accomplish this function and then automatically arm the switch circuitry at the fuel enrichment point at 87 percent speed. An override circuit is provided which can be actuated by a manually

operated switch connected to the sequence box for emergency situations where the engine must continue to run even in the event of overload.

The second alternative to temperature limiting studied was a temperature sensor capable of regulating fuel flow by modulation of the pneumatic signal to the control in proportion to the temperature from maximum flow to minimum flow. This unit would allow the engine to continue operation at the temperature limit without shutdown and would also have been operable during starts to help limit starting temperatures. Since no control of this type was found "on the shelf," a development program would be required to perfect a limiter of this type. The advantages to be gained from such a program did not warrant the costs at this time. This type control could be added in the future if these considerations should indicate its desirability.

STARTING

The starting process has been modified from the method outlined in the design report. The starting sequence was to have used an oil pressure switch to energize the fuel shut-off solenoid and to initiate ignition after the engine rotation was sufficient to build up oil pressure. Since then, the oil pressure switch has been replaced by use of a pressurizing valve in the fuel control discharge line. By setting this pressurizing valve to open at 25 to 30 psi the fuel pump (and engine) must be rotating sufficiently to displace fuel to open the valve. The shut-off valve and ignition will be energized with the starting switch but light-off will not occur until the pressurizing valve admits fuel to the engine. This eliminates a pressure switch and relay, simplifying the circuitry.

FUEL NOZZLE PURGE VALVE

The possibility of "coking" of the fuel nozzle on shut-down and the accompanying hot soak may require fuel nozzle purging. This will be determined early in the test program and if necessary, a purge valve will be added to the system.

FILTRATION

The engine will be furnished with two fuel filters, mounted in series, which filter fuel supplied to the engine prior to the fuel pump inlet. The first of these filters is a 10-micron full-flow filter with replaceable paper element, and sized to provide adequate contaminant holding capacity for 200 hours of operation with the flows and contaminant levels specified in the engine model specification. The second filter has the same micron rating and also has the full engine flow passed through it, but has been sized only to provide acceptable flow-pressure drop characteristics. Its purpose is to prevent contamination from reaching the pump inlet while the first filter is being serviced. Although it is also of the replaceable paper element design similar to the first filter, little or no servicing should be required since almost all contaminant will have been removed by the first filter. No bypass is provided for either filter, in order to ensure proper servicing. Both filters are currently being procured from Bendix Filter Division.

The remainder of the system is essentially as outlined in CAE Report No. 976. Figures 1 through 4 show the estimated acceleration characteristics of the system.

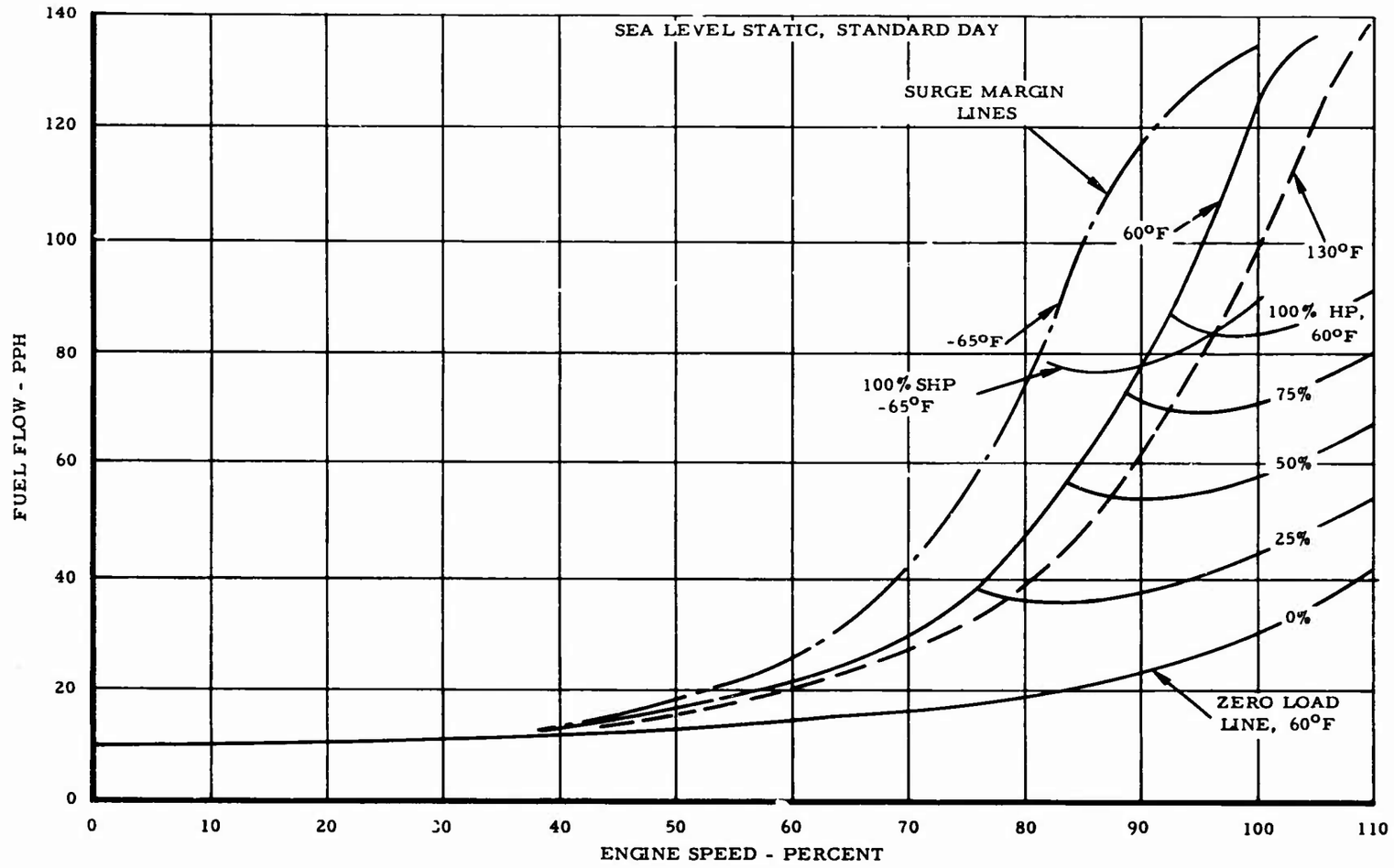

Fig. 1. Model TS-120 Estimated Performance and Acceleration Characteristics.

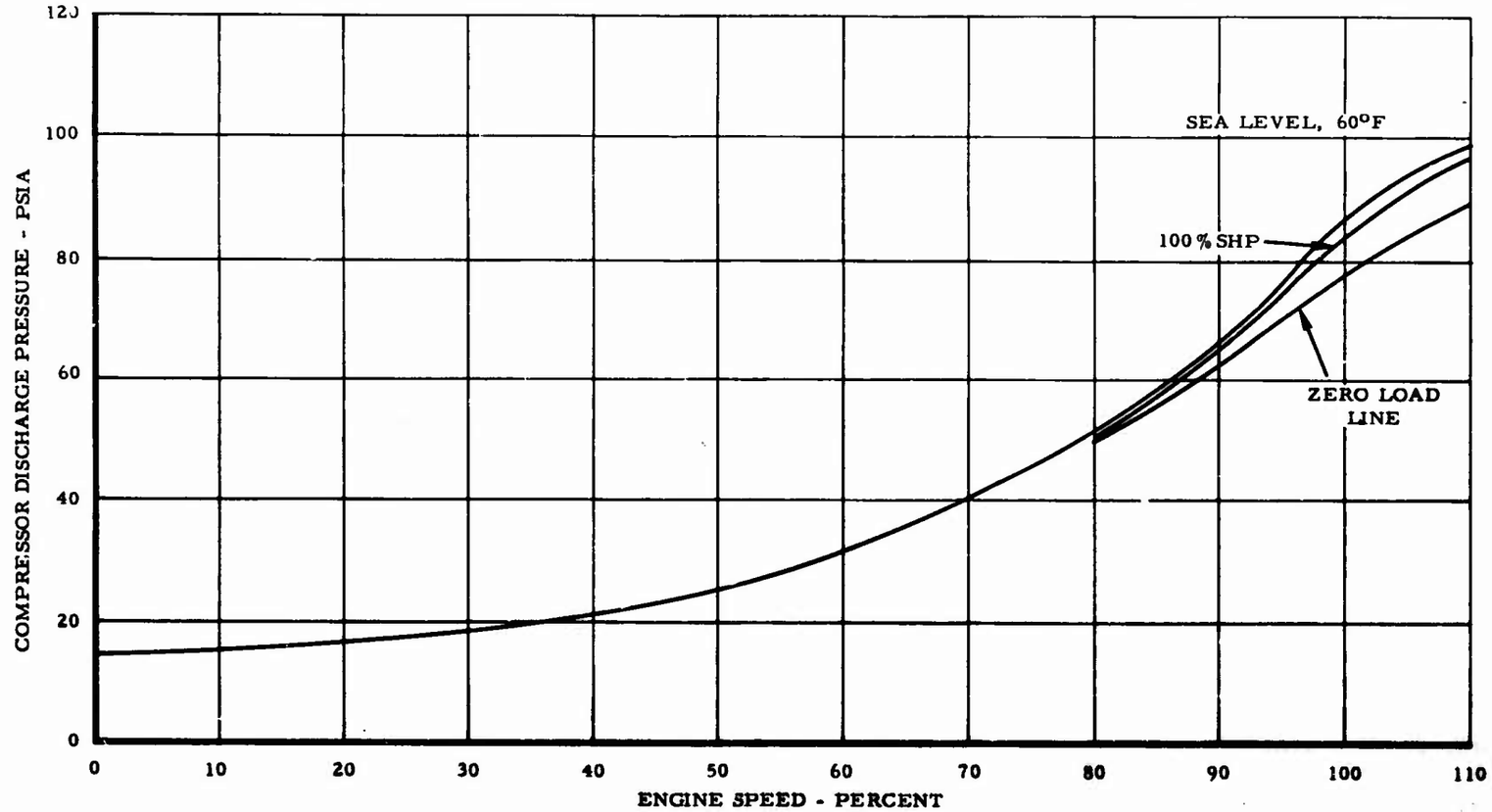

Fig. 2. Model TS-120 Estimated Compressor Pressure Rise Versus Speed Characteristics.

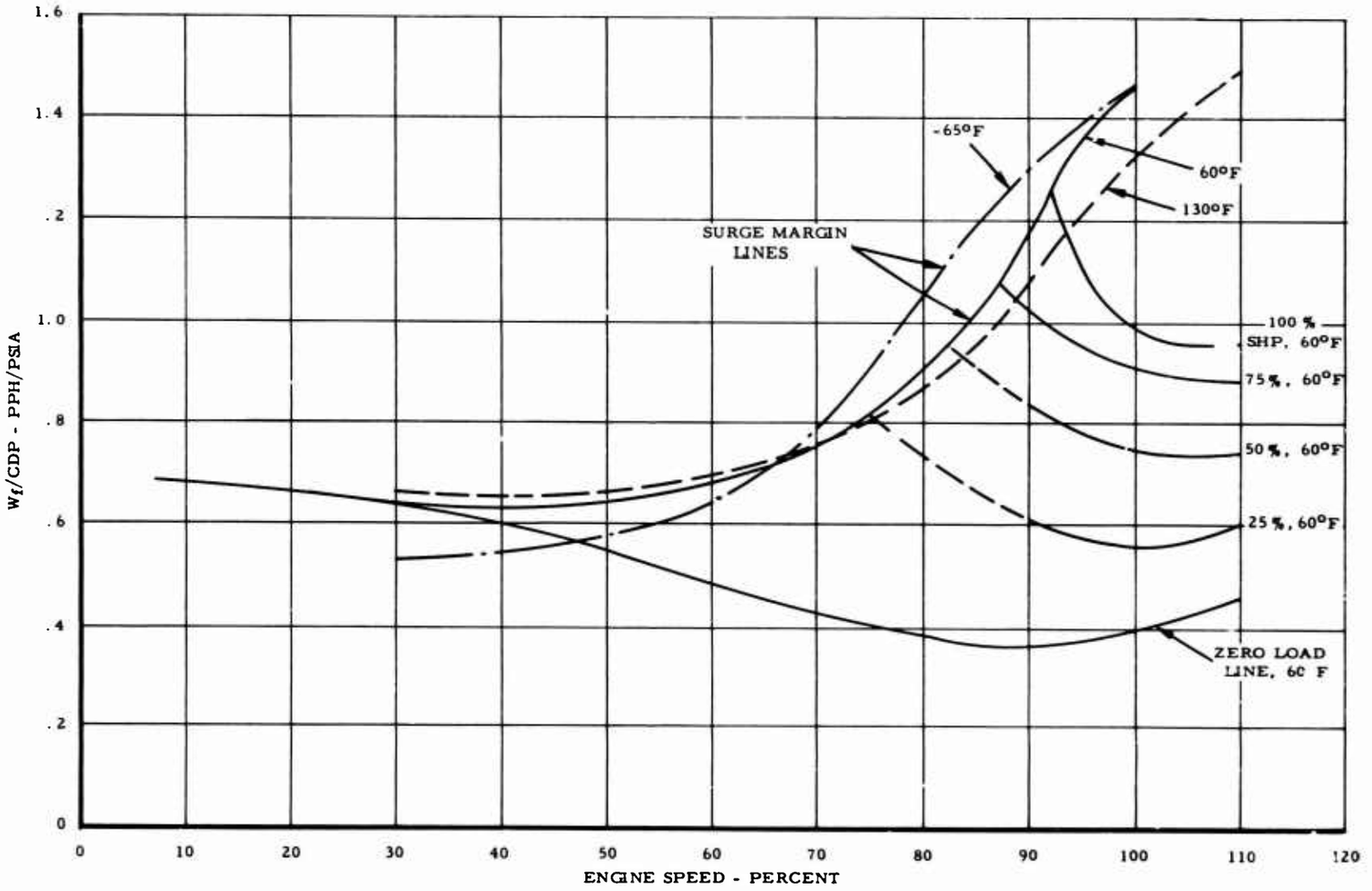

Fig. 3. Model TS-120 Estimated Performance and Acceleration Scheduling Parametric Comparisons.

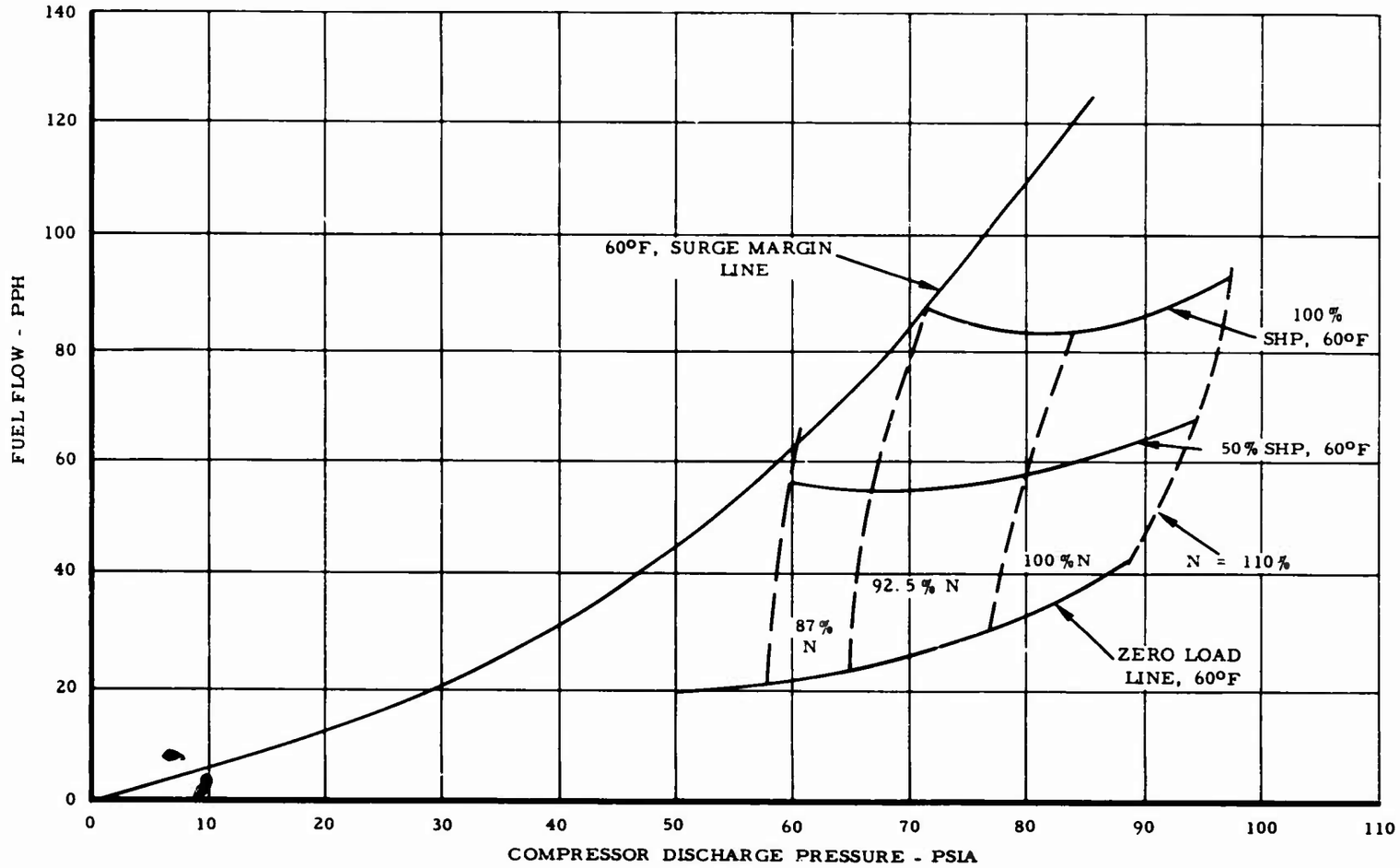

Fig. 4. Model TS-120 Estimated Performance and Acceleration Scheduling Parametric Characteristics.

APPENDIX

DESCRIPTION OF OPERATION AND DESIGN REPORT

X2292 Fuel Control
Woodward Governor Company

WOODWARD GOVERNOR COMPANY

X2292 Fuel Control For

Continental TS120-1 Turboshaft

Engine -- Simple Cycle-- Direct Drive

DESCRIPTION OF OPERATION AND DESIGN REPORT

See Schematic X2292-021, Figure 1,for

Reference to Control System Components

A. General Operation

The X2292 governor is a variable reference speed governor with acceleration and deceleration fuel scheduling control and with 2-1/2% speed regulation from no load to full load. Acceleration fuel is scheduled as a function primarily of compressor discharge pressure (CDP) and is biased by compressor inlet air temperature (CIT). The ratio and level of acceleration fuel relative to CDP is adjustable. The ratio and level of acceleration fuel bias relative to CIT is also adjustable. The acceleration fuel schedule may be further biased by bleeding CDP air from the sensor and at a preset speed the full CDP will be admitted to the sensor in order to establish normal acceleration fuel flow.

An alternate to the above bleed system for controlling acceleration fuel flow is contouring of the fuel metering valve porting.

The governor reference speed setting will be adjustable through a range of ±10% of rated speed either manually or by means of a set of remote electrical contacts.

The governor speed regulation, or change of speed with load, is determined by the combination of the fuel valve porting and the speeder spring and will vary with air density due to changes in temperature and altitude at a given speed setting. The regulation will also vary as a result of changes in speed setting, but will not exceed 2-1/2 per cent.

The governor package will weigh approximately 6 pounds in addition to the fuel pump. The power to drive the X2292 will be negligible.

B. Detailed Description of Governor Components and Operation.

1. Fuel Supply.

The fuel pump (2) is a Pesco Products gear type pump which supplies fuel to the control inlet. Fuel is supplied to the pump through a 10 micron filter (1) to assure long pump life. The fuel will be supplied to the pump at 5-15 psig. Pesco states in their Engineering Report 4882 that the major components of this fuel gear pump are fully developed and are in use on many similar applications. The Pesco pressure loaded fuel gear pumps have been proven by hundreds of thousands of hours' service in the field on fuels as listed in Continental Specification 8-46. The pump seal is of standard Pesco design. The fuel pump has a weight of 1.75 pounds and a power requirement of 0.628 horsepower at 400 psig outlet pressure. There is a 74 micron screen (3) between the fuel pump outlet and the governor inlet.

B. 2. High Pressure Relief Valve

The high pressure relief valve (4) limits the pressure to which the fuel system may be subjected. This valve is incorporated in the control body and is set to bypass at 100 psi above normal maximum system operating pressure. Fuel is bypassed back to boost.

3. Pressure Regulation.

The pressure regulator is a spring loaded, constant differential pressure, bypass valve (5). Excess fuel is bypassed back to boost. Fuel control inlet pressure P_1 and control outlet pressure P_2 are applied to opposite ends of the bypass valve plunger so that the force resulting from (P_1-P_2) is opposed by a reference spring, giving an essentially constant ΔP across the metering port (6). This results in fuel flow being a function of the position of the fuel valve plunger (7) for any given fuel specific gravity.

4. Fuel Specific Gravity Compensation.

The variety of fuels required to operate at from -65°F to 135°F have a range of specific gravity of approximately 0.7 to 0.84,but are not necessarily limited to this. Specific gravity is a measure of the heating value of the fuel by volume. As the specific gravity changes, the ΔP across the metering port must change in order to increase or decrease the volume flow of fuel to the engine for a given metering port opening.

The specific gravity compensation system (9) automatically adjusts the ΔP, and therefore the fuel flow, as a function of fuel specific gravity. The operation of the specific gravity compensation is as follows. The pressure regulator (8) gives a constant pressure supply to the viscosity measuring device (9). This eliminates possible errors due to the changing P_1 pressure with turbine speed and load. The pressure drop across sharp-edged orifice (10) is held constant by spring (12), which assures a constant flow through the orifice regardless of viscosity. Pressure P_6 is determined by the length of laminar flow orifice (13) which passes the constant flow from sharp-edged orifice (10). The pressure drop across a laminar flow orifice is dependent on viscosity. A pressure regulator plunger (11) maintains a pressure upstream from the sharp-edged orifice equal to P_6 plus the spring load. The load on piston (14) increases with P_6, which in turn increases with increasing viscosity. An increase of P_6 therefore unloads the bypass valve (5) proportionally and decreases regulated pressure P_1 and ΔP across metering port (6), which decreases fuel flow for a given position of plunger (7). The slope of ΔP to fuel specific gravity is adjustable by changing the length of laminar flow orifice (13) which is accomplished with adjusting screw (15). The level of initial ΔP is adjustable with screw (16).

B. 5. Shutdown Solenoid Valve.

This is a normally closed valve (17) which requires a voltage supply to remain open and pass fuel to the engine nozzles. Any interruption of voltage to the solenoid coil will allow the valve to close and shut off the fuel flow to the engine nozzles. Any number of protective devices may be utilized to break the circuit to the solenoid coil to stop the engine.

6. Pressurizing Valve.

This is a spring loaded valve (18) in series in the fuel line just before the nozzles. It is set to open at 25-30 psi which is above maximum boost pressure. This assures that the engine and fuel pump are rotating before fuel is admitted to the nozzles and the combustor.

7. Acceleration Fuel Scheduling.

Acceleration fuel control and therefore engine acceleration control is accomplished by proper positioning of the fuel metering plunger (7). Starting fuel flow is set by adjustment (19) and will always be equal to or greater than minimum flow. This adjustment assures sufficient fuel for initial light-off. Acceleration fuel is then controlled and limited by compressor discharge pressure (CDP) acting on the CDP bellows (20).

The metering port opening (6) increases through linkage (21), pivoting at pivot point (22), as the engine accelerates and can use more fuel. Acceleration fuel is scheduled so that the fuel flow is sufficient to accelerate the engine and low enough so as not to exceed the surge limits of the compressor or temperature limits during acceleration.

Acceleration fuel scheduling is biased in level by adjusting nut (23) and in slope relative to CDP by moving pivot point (22). Pivot (22) is moved as a function of compressor inlet temperature (CIT) sensed by CIT sensor (24) acting through bellows (25). The bellows is connected to point (22) through linkage and the pivot point is moved in position relative to CIT and in slope relative to CIT change by adjusting nuts (26). The linkage is arranged so that an increase in CIT increases the slope of accelerating fuel relative to CDP.

The CDP control is through the action of an evacuated bellows (22). The evacuated bellows thus has an absolute pressure reference and gives a constant fuel flow relative to CDP (psia) regardless of altitude changes. This is the same system as used on aircraft propulsion turbines. With atmospheric pressure refence, fuel flow vs. CDP (psia) varies with altitude. See Figure 2. Compensation on this system would give the curves shown in Figure 4. The absolute pressure reference gives a constant fuel flow vs. CDP (psia) regardless of altitude as shown in Figure 3.

B. 7.cont.

There are two methods of obtaining the required acceleration fuel scheduling over the speed range. One method, and the simpler, is to contour the metering valve porting (6) to match engine requirements. This works very well if the engine requirements are established and known, and if there are not excessive variations in acceleration fuel requirements between engine units. Example curves are shown in Figures 5, 6 and 7. The surge margin curves are paper curves developed by CAE. The limiter curve on Figure 7 represents the curve attainable with contoured porting. An alternative method is to use a bleed system whereby the governor does not see the actual CDP signal from the engine compressor. This is accomplished by needle valve settings (28) and (29) and after the CDP chamber in the governor. This system incorporates a speed switch actuated solenoid valve (27) so that at any preset speed, full CDP may be admitted to the governor CDP acceleration control. Full CDP is not admitted instantaneously but at a controlled rate depending on the opening of the needle valve (28) between the engine and the governor and the acceleration rate of the engine. The curves in Figures 5, 6 and 7 indicate the operation of this system as compared to the first described system. The same limiter curve applies.

Either of the above methods gives a low ratio of fuel flow to CDP at low speed, increasing relative to speed. Control of acceleration fuel is necessary to assure acceleration and yet to avoid excessively high temperatures during acceleration and to prevent compressor surge. At and near rated speed fuel flow must be as near to surge as practical in order to give minimum underspeeds on large rapid load applications.

The second system of CDP control has the advantage of a large range of adjustability in order to determine actual engine require ments. When these requirements are once established, the first system with the contoured fuel metering port has the advantage of simplicity.

8. Control on the Governor.

(A) Transfer from acceleration to governor control.

After a start, when the governor reaches the set speed,the flyweights (30), opposed by the speeder spring (31), move out, raising the fuel valve plunger (7), thereby adjusting the metering port opening. The fuel valve plunger is separate from the acceleration control rod (32) at point (33) while the governor is controlling. The CDP still limits the opening of the metering port through rod (32). The maximum fuel adjustment (34) limits the absolute opening of the metering port regardless of CDP or load.

B. 8.(B) Governor Speed Setting and CIT Bias.

Governor speed is set by operating the speed setting motor (35) with a remote switch or by operating the manual speed setting knob (36) at the governor. These are connected through a friction clutch (37) which also prevents motor stall at the high and low speed stops. The high and low speed setting stops (38) set directly on the lever which operates on the speeder spring (31). Regardless of temperature the speed setting may be adjusted from the low stop, 90%, to the high stop, 110%.

The compressor inlet temperature bias changes speed setting as a function of CIT through linkage. An increase in CIT will increase the speed setting. This bias is effective regardless of where the speed is set. The ratio of speed bias to change in CIT is adjustable at (40).

The governor will have a maximum regulation of 2-1/2% from no load to full load. The regulation, or speed change with load, is a function of fuel flow. As load is increased, fuel flow is increased which is effected by moving the metering valve plunger (7) down while the flyweights move in,relaxing the load on the speeder spring and reducing the speed setting. The regulation will vary from a maximum of 2-1/2% at 8000 feet, 90°F, to a minimum of approximately 1-1/2% at sea level -65°F. This is caused by the engine requiring less fuel flow change from no load to full load at low altitudes and low temperatures because of changes in air density.

(C) Governor Transient Action.

Temporary speed errors may arise in the engine from three principle causes:

(1) A change in governor reference speed set point.

(2) A sudden change in engine environment or air flow.

(3) A change in load on the engine.

The amount of speed error depends on the level of the disturbance and there will be a short but finite time interval,while the control is making the appropriate correction,before the engine returns to steady state speed. In any of the causes mentioned above, the action of the control is similar. As an example, if the speed were to decrease below the reference speed, the flyweights would move in,lowering the fuel valve plunger (7) and thereby increasing the metering port opening (6) allowing a higher fuel flow to the engine. The maximum opening is limited by CDP or the maximum fuel stop,whichever is calling for the lesser opening. The limit prevents the

B. 8. (C) compressor from going beyond the surge limit and also prevents excessive temperatures. Conversely, if the speed were to increase above the reference speed, the flyweights would move out, decreasing fuel flow to the engine. The minimum fuel flow stop (41) prevents complete cutoff of the fuel due to control action and maintains fuel flow above engine lean blowout limits. This minimum fuel flow may be biased by CDP by repositioning the stop at adjustment (42) so it is moved relative to CDP by link (43). This decreases the minimum fuel flow as the engine speed decreases. See curve in figure 8.

(D) Governor Response and Control.

For any given set of stabilized environmental conditions, the governing system will maintain speed within the following limits:

(1) Regulation - the engine speed at full load will not be more than 2-1/2% less than the engine speed at no load at any environmental condition.

(2) Steady State Speedband - At any steady state load between no load and full load the governor will maintain engine speed within ±1 per cent of mean with no speed hunt.

(3) Transient Performance - The governor will maintain engine speed within 8 per cent of the speed at which the engine is operating during sudden application or removal of any shaft load to and including 100 per cent rated load. The speed will stabilize within the ±1 per cent steady state speedband within 3 seconds with no more than one undershoot or overshoot outside this steady state speedband.

(4) System Analysis - A system transient analog analysis made by Continental Aviation & Engineering Corporation indicates that this governor type will meet all requirements. The analysis also indicates there will be no undershoot or overshoot outside the ± 1 per cent steady state speed band with a 100 per cent rated load step application or removal and 2-1/2 per cent regulation. With governor gain increased to give 1 per cent regulation, there is still no evidence of excessive undershoot or overshoot.

B. 9. Stress

Pesco Products has made a stress analysis of the pump load carrying capacity at 10g shock load and horizontal mounting. The safety factor is 7 to 1. Thus the strength of the pump is sufficient.

10. Maintenance and Adjustments.

All adjustments are accessible with the removal of a protective cover. Adjustments are covered for protection and to prevent tampering. There is no maintenance except for the possible replacement of the 74 micron screen filter in the base of the governor. In the remote event that it may become clogged the fuel pump, itself, has already been damaged and will probably require replacement.

C. Future Application.

The X2292 droop governor as described above may be used on a 60,90, and 120 hp turbine with minor alterations.

The isochronous version of the X2292 governor system will incorporate an X2301 electric governor which includes an electric-hydraulic actuator and associated electric components. Schematic X2292-026,Figure 9 describes the system. The electric governor will allow equal load sharin during parallel operation of similar units. In this system the flyweights will be removed from the X2292 governor, and this assembly will be used as a fuel valve which is operated by the X2301 governor. The actuator may be mounted on the fuel valve and will operate the fuel valve through linkage. The actuator will require up to 1 pint/minute of oil from the turbine lubricating pump at 40-50 psi. The X2292 will still control acceleration in the same manner as the original version, and will have CIT control of acceleration fuel flow. Specific gravity compensation will be effective during acceleration, but the isochronous characteristics will nullify the compensation during governor control.

The engine version incorporating a regenerator will use a bypass valve to dump regenerator air flow on a load rejection in order to limit overspeeds. An analog study must be made to determine the best methods of modulating the bypass valve operation.

D. Interchangeability

The X2292 droop governor for the 60 and 90 hp version of the single-shaft simple cycle turbine will be identical to the one described except for different metering valve porting and speeder spring to maintain required droop. Since droop is dependent on fuel flow changes from no load to full load, it will be less on the 60 and 90 hp turbines than on the 120 hp turbine if no port change is made. If the reduced droop gives a stable system, there is the possibility the same assembly may be used on all three turbines.

D. Cont.
The isochronous and droop governor assemblies may be interchangeably mounted on the engine. The isochroncus governor will need in addition, the electric components shown in Figure 9. The internal parts of the X2292 assembly will remain the same except for the removal of the flyweights in the isochronous version. Speed bias with CIT will not be required so an entirely different case and cover will be needed, omitting the external speed bias linkage. The CIT bias on acceleration will be required on both versions, as will the specific gravity compensation.

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Figure 1 Schematic - Droop Fuel Control.

Figure 2 W_f vs. CDP - No absolute reference,no compensation.

Figure 3 W_f vs. CDP - Absolute reference.

Figure 4 W_f vs. CDP - No absolute reference with compensation.

Figure 5 CDP vs. N - Turbine.

Figure 6 W_f vs. N - Requirements.

Figure 7 W_f vs. CDP - Requirements.

Figure 8 W_f vs. CDP - Sample deceleration.

Figure 9 Schematic - Isochronous fuel control.

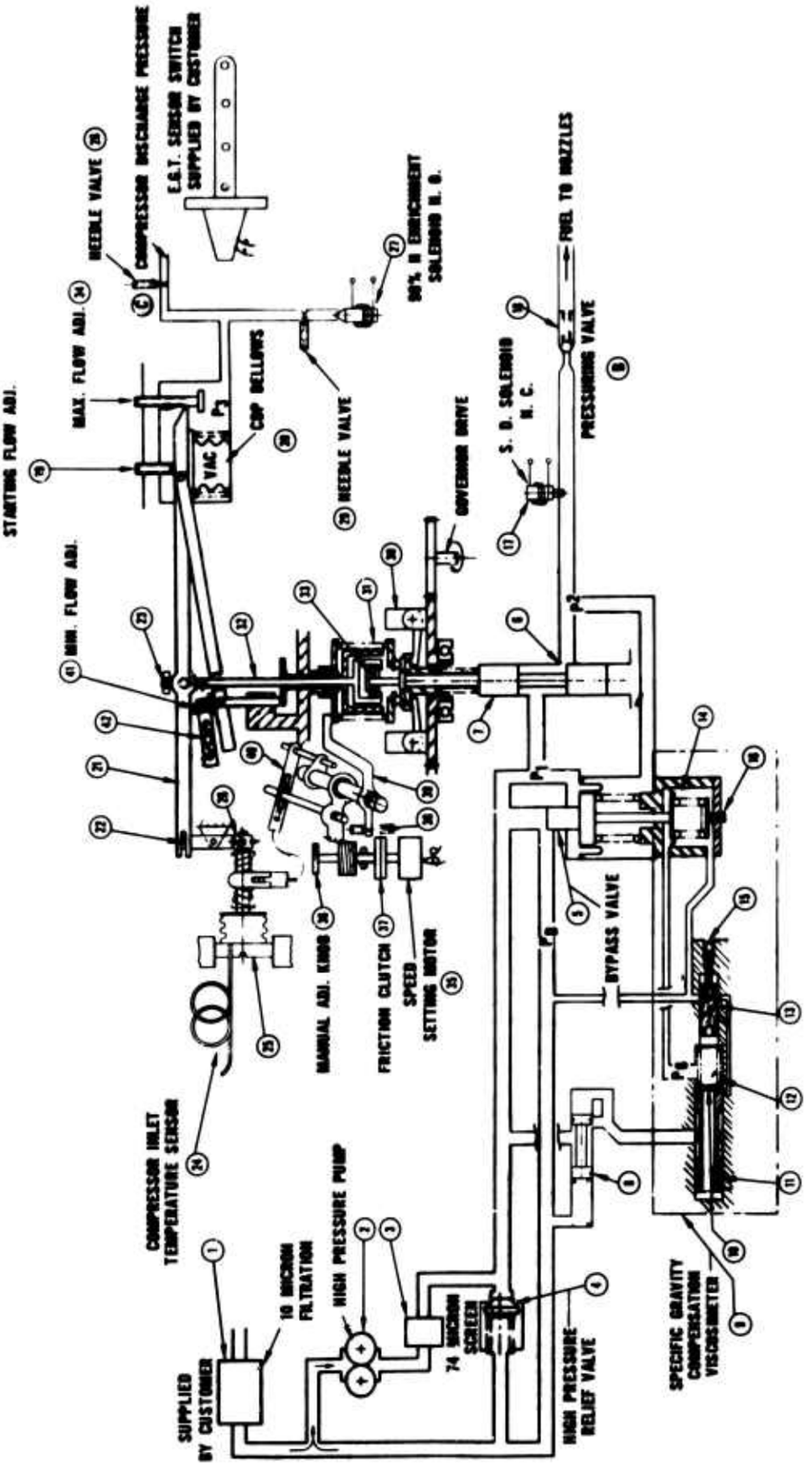

Fig. 1. Schematic - Droop Fuel Control

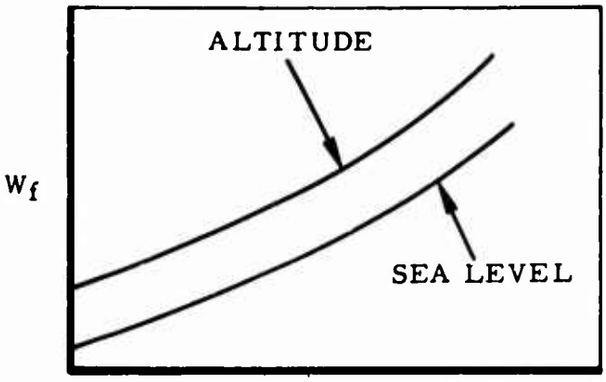

Fig. 2

CDP (PSIA)
NO ABS. REF., NO COMP.

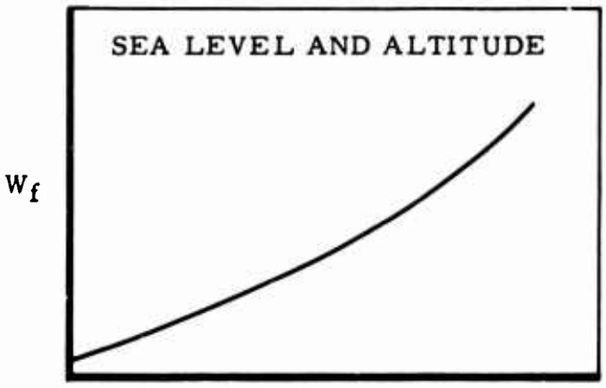

Fig. 3

CDP (PSIA)
ABS. REF.

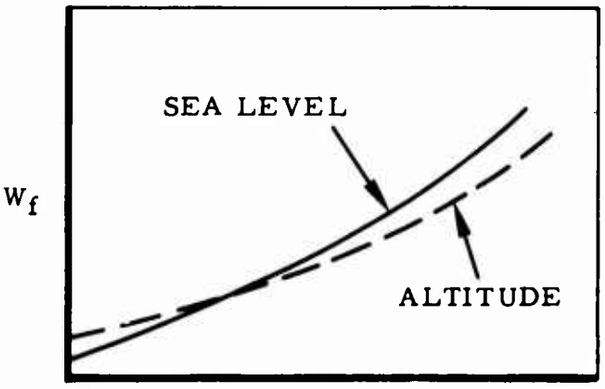

Fig. 4

CDP (PSIA)
NO ABS. REF., WITH COMP.

Figs. 2, 3, and 4. Sample Limiter Curves

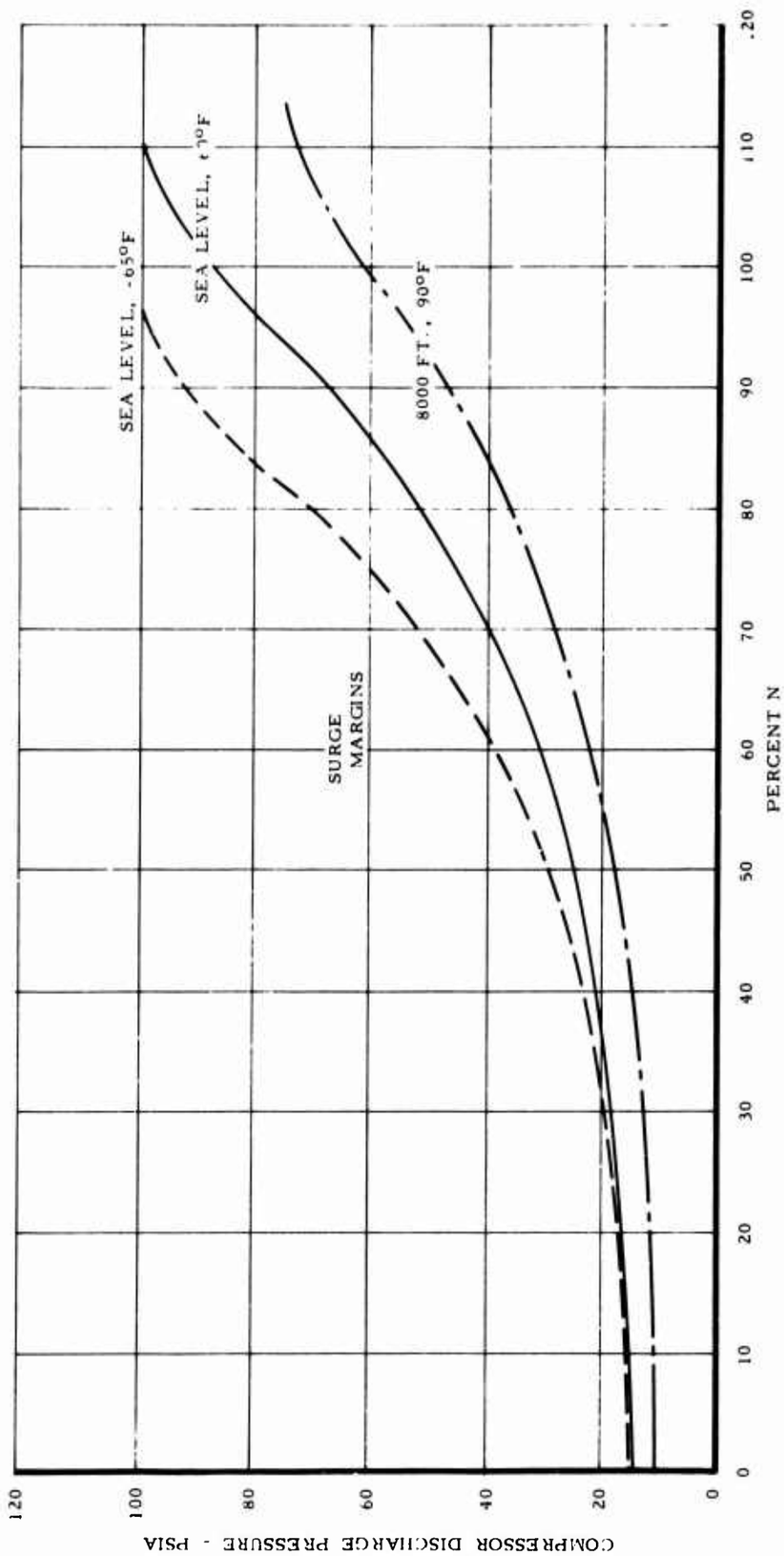

Fig. 5. CDP Versus N - Turbine

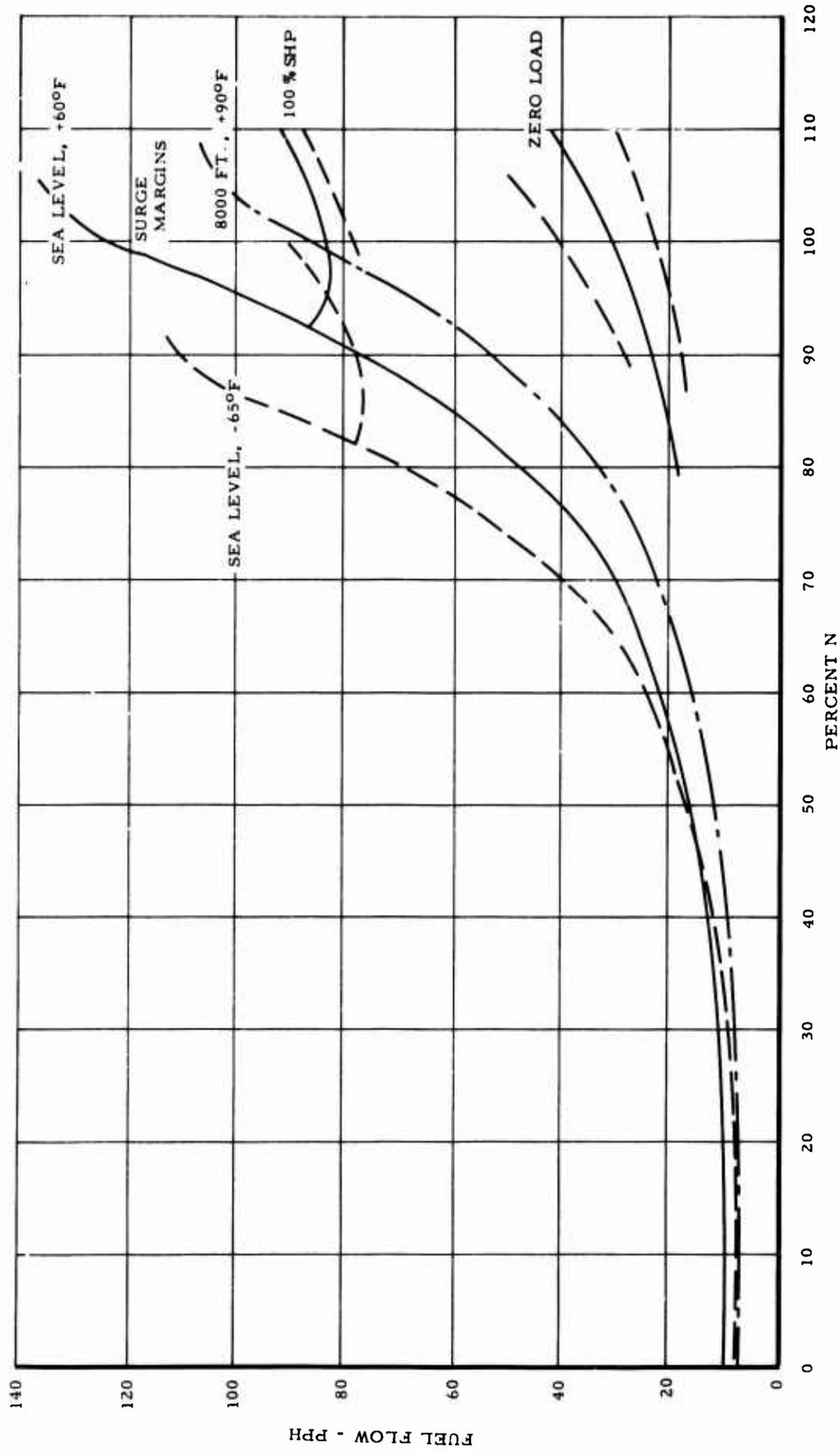

Fig. 6. W_f Versus N - Requirements

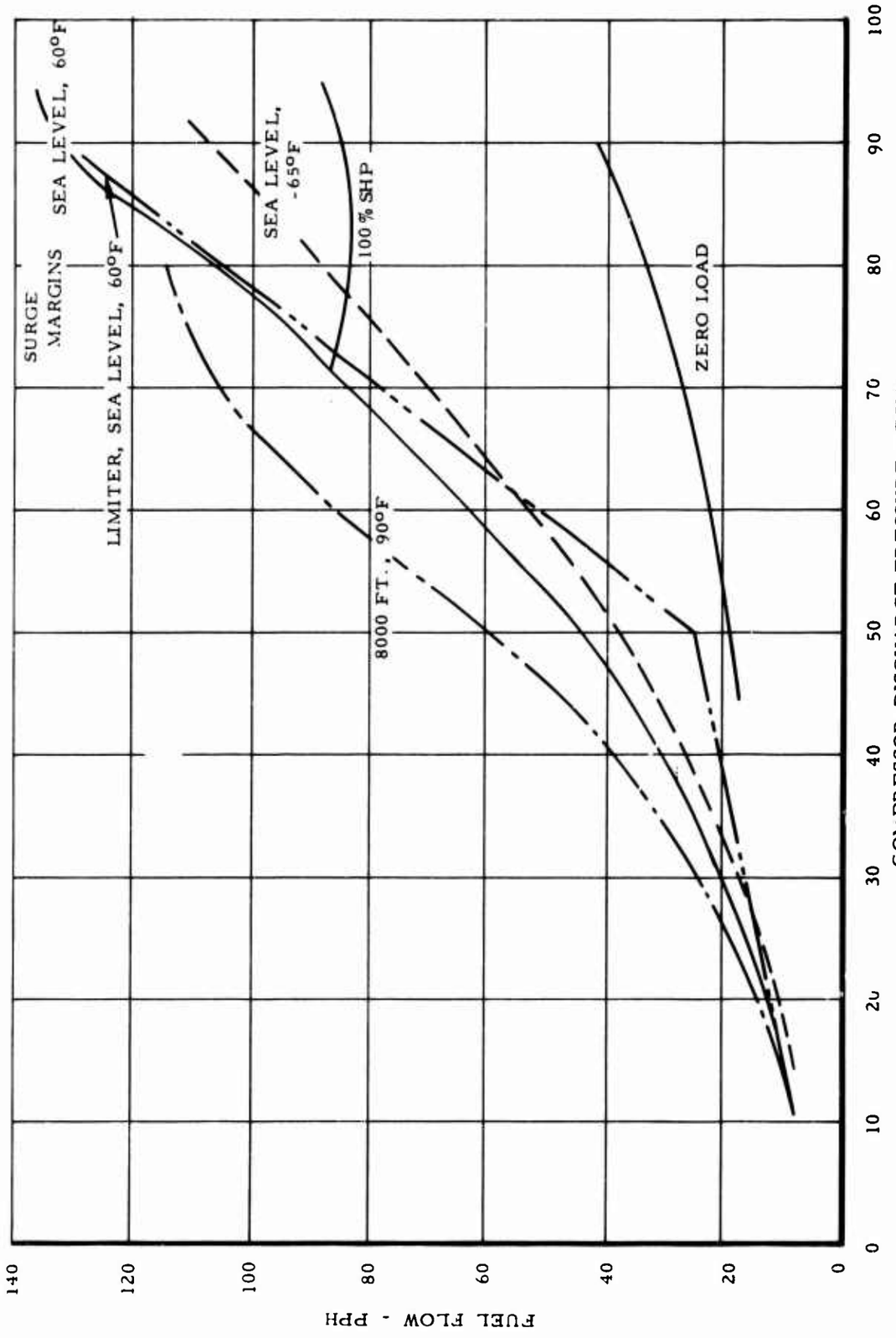

Fig. 7. W_f Versus CDP - Requirements

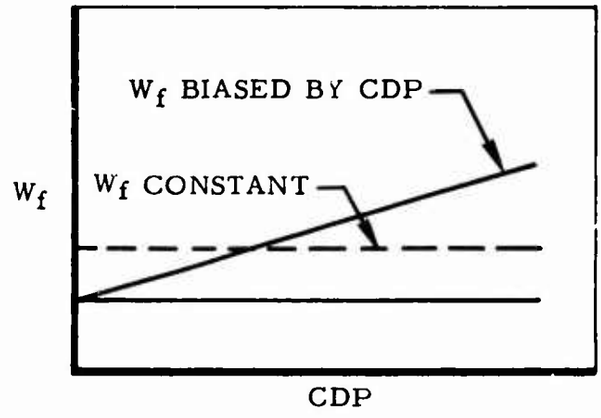

Fig. 8. W_f Versus CDP - Sample Deceleration

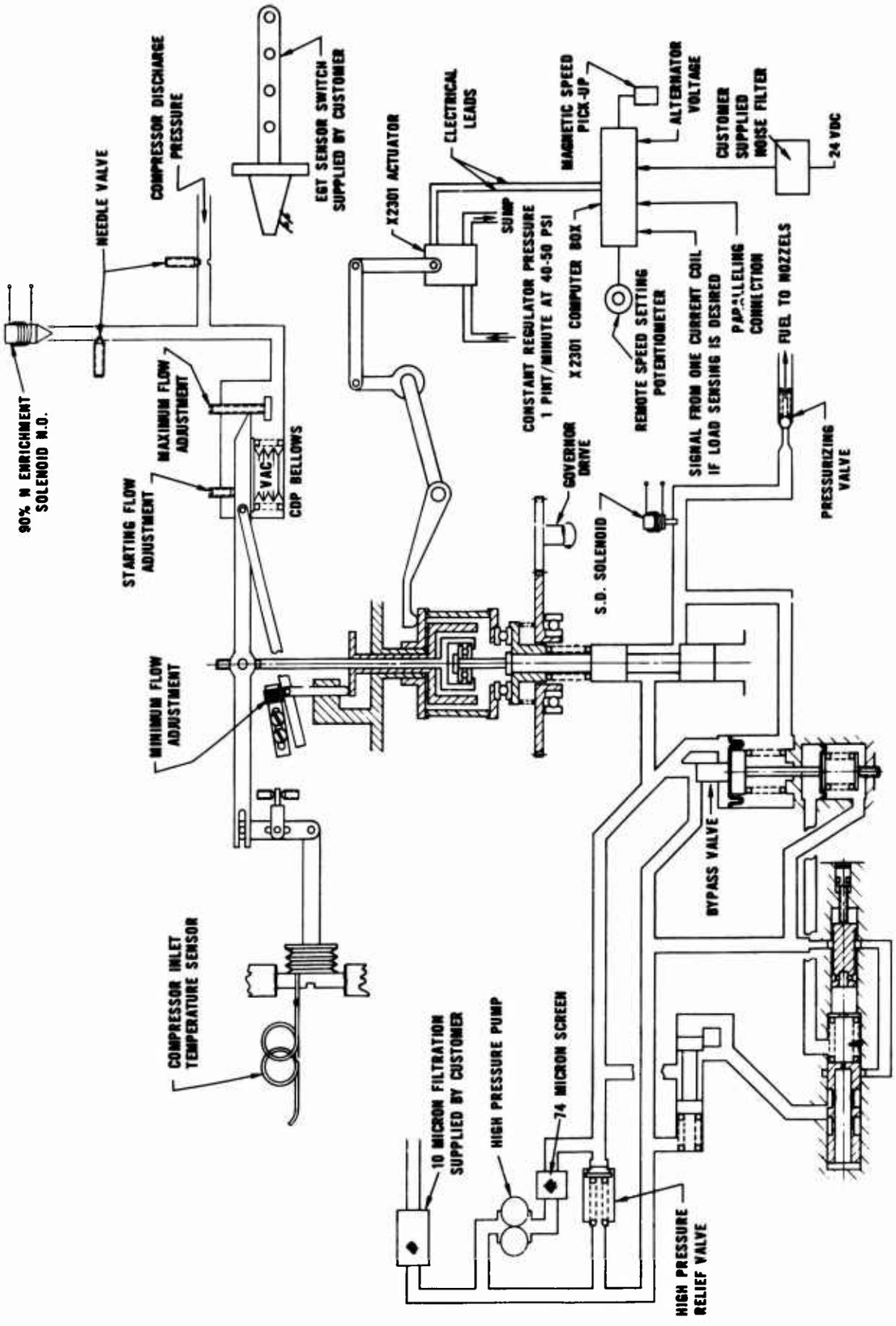

Fig. 9. Schematic - Isochronous Fuel Control