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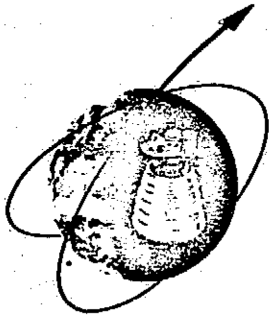

Space Shuttle Main Engine Definition (Phase B)

ENGINE DESIGN DEFINITION REPORT

VOLUME V

VALVES AND INTERCONNECTS

MSFC-DRL-163, ITEM 30
DRD SE-275

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George C. Marshall Space Flight Center
Marshall Space Flight Center, Alabama 35812

Pratt & Whitney Aircraft DIVISION OF UNITED AIRCRAFT CORPORATION
FLORIDA RESEARCH AND DEVELOPMENT CENTER
BOX 2691, WEST PALM BEACH, FLORIDA 33402

.d in the United States of America

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INTRODUCTION

The steady-state thermodynamic cycle balance of the single preburner staged combustion engine, coupled with dynamic transient analyses, dictated in detail the location and requirements for each valve defined in this volume. Valve configuration selections were influenced by overall engine and vehicle system weight and failure mode determinations.

Where possible, the valves have been designed to perform several functions to minimize weight and avoid unnecessary synchronizing problems. The modulating elements within the valves have been kept as simple as possible to avoid maintenance problems that require separating the high pressure flanges. In keeping with this, the modulating valve actuators are external to the valve and are line replaceable. Development and satisfactory demonstration of a high pressure dynamic shaft seal has made this configuration practical.

Pneumatic motor driven actuators, to be supplied by the Bendix Corporation, that use engine pumped hydrogen gas as the working fluid are used. Although the piece-part cost for hydraulic actuators would seem to make them an attractive alternative, our experience with high response close coupled hydraulic systems has been unsatisfactory, and the total cost to develop an adequately stable hydraulic system for engine power control is unpredictable. The pneumatic motor actuator is stable, and is least dependent upon the vehicle system for safe operation.

The helium control system is proposed as a module containing a cluster of solenoid actuated valves. The valve configuration was chosen after a competitive design review in which contamination insensitivity, demonstrated ability to seal, and design approach to fail operative were heavily favored. The selected design, submitted by the Parker-Hannifin Corporation also includes a weight saving approach for incorporation of the required bifilar solenoid windings.

The separable couplings and flanges are designed to assure minimum leakage with minimum coupling weight. The deflection of the seal surface in the flange is defined by finite element analysis that has been confirmed with test data. The seal design proposed has passed preliminary pressure cycling and thermal cycling tests and has performed consistently well during the recent staged combustion rig test series at P&WA.

SECTION I
CONTROL VALVES

A. PREBURNER OXIDIZER VALVE

1. Introduction

The translating sleeve valve, which divides and schedules oxidizer flow to the preburner, is based upon a P&WA design that has been tested throughout 137,000 operating cycles under cryogenic high pressure conditions. In addition, our 250K lightweight design has successfully supported a series of hot preburner rig and staged combustion rig firings, up to 100% power level conditions. The 550K flight weight design is a scale-up of the substantiated model. A single 25-lb valve, close-coupled to the preburner injector, will meet all CEI specification requirements and the functional requirements of the SSME cycle.

Tests have verified the sleeve valve effective area versus stroke schedules and flow dividing characteristics. We have also substantiated shut-off seal leakage rates under 10 scc/sec after 10,000-cycle cryogenic endurance tests of the same design that is used in the 550K design. This valve shuts off all preburner oxidizer flow and is mounted inside of the lightweight spherical preburner housing. This close-coupled arangement with the injector ensures rapid oxidizer filling for starting and stable flow rates for smooth engine acceleration. The low oxidizer residuals also provide the basis for consistent rapid engine shutdown. The translating sleeve has proved to be the most effective and least complicated design to provide 276:1 area turndown ratio, flow splitting, and shutoff sealing in a single unit. A commerical nonflight-type valve actuator has been used to control our 250K model to date. The proposed flight pneumatic actuator concept is currently being tested in the P&WA F100 jet engine program, for actuation of the exhaust nozzle.

2. Design Description

The preburner oxidizer valve, shown in figure I-1, consists of a translating sleeve moving over a valve guide that incorporates two sets of metering ports to divide total oxidizer flow before it enters the injector. A single shutoff seal is located upstream of these ports, and the sleeve seats against this seal at the end of its travel. Progressive opening movement initiates primary and then secondary oxidizer flow into the dual-orificed injector. The valve assembly is mounted inside of the spherical preburner housing dome proximate to the injector to reduce oxidizer cavity volume and to obtain the lightest weight structure possible. Direct linear drive by the dome-mounted actuator provides minimum valve weight and complexity.

The drive for the sleeve valve is optimized for direct linear actuation because of its inherent simplicity. Several angular drives were considered and the most practical scheme is shown in figure I-2. Excessive complexity is clearly evident compared to the chosen design in figure I-1 shown previously. Since the linear drive meets the envelope requirements, the angle drive was not selected as the optimum configuration.

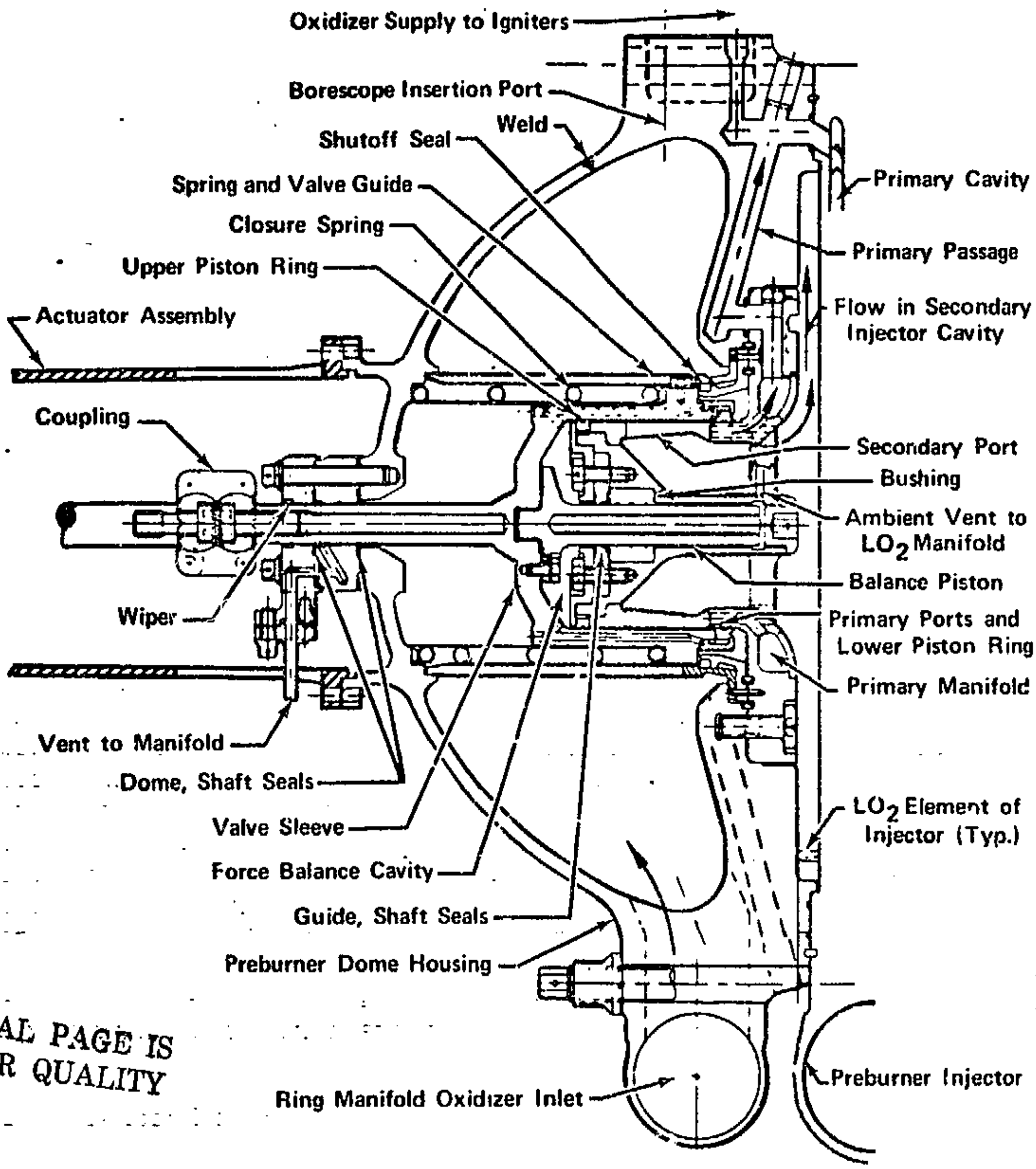


Figure I-1. The Translating Sleeve Shuts Off, Divides, and Schedules Oxidizer Flow to Our Single Preburner With a Substantiated Design FD 46056

The proposed design has evolved from an original Phase I 250K concept demonstration design that is shown in figure I-3. The original was heavy and complicated but the flow dividing and scheduling capability with single shutoff was substantiated by 117,000-cycle endurance and calibration tests. This initial sleeve valve was selected from nine candidate schemes which were considered during a systematic design evaluation. Five of these alternative concepts are illustrated in Valves and Interconnects Trade Studies, PWA FR-4442, together with the rationale for rejecting all but the translating sleeve.

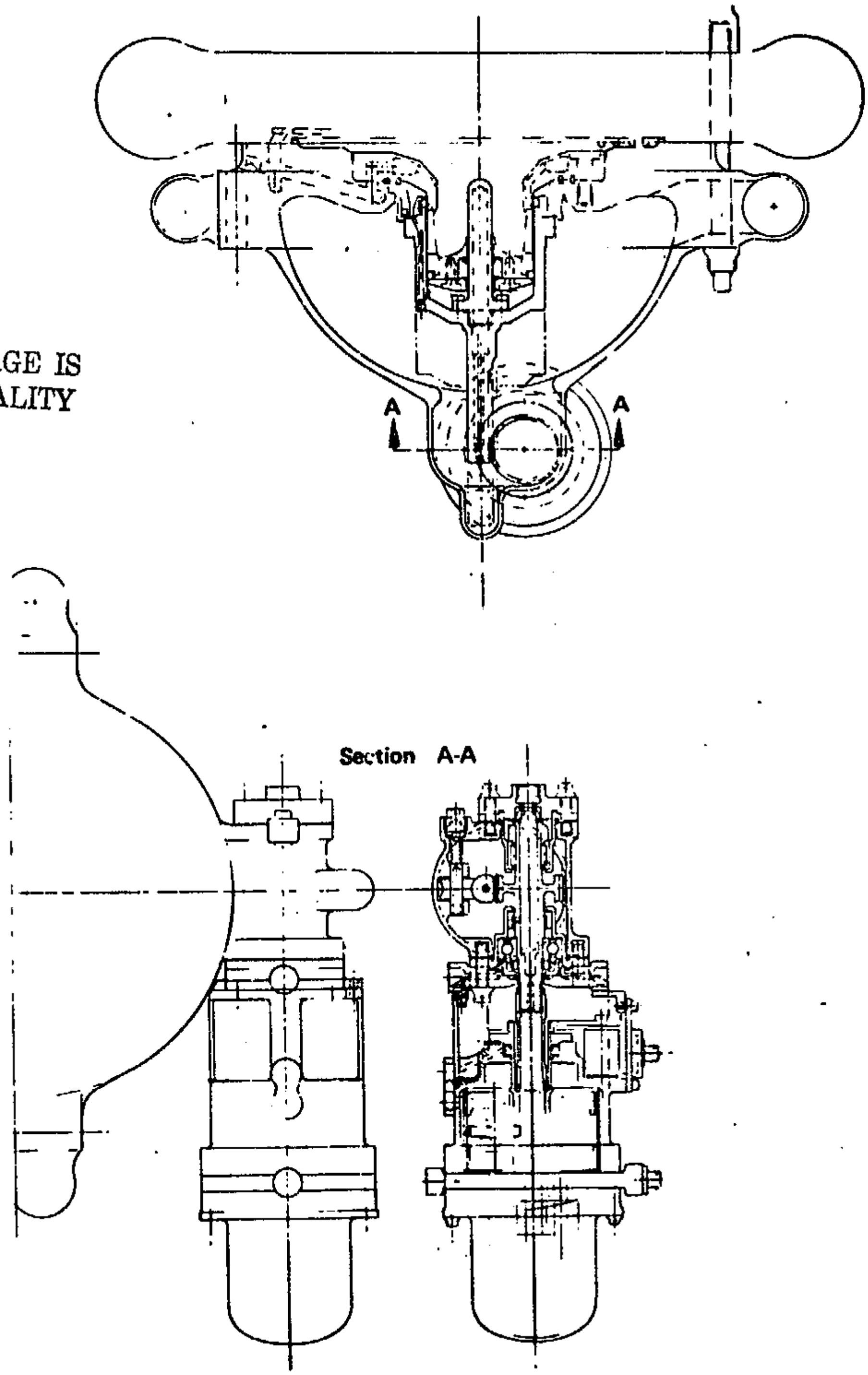


Figure I-2. Preburner Oxidizer Valve-Angle Drive Is 9 in. Shorter, but Complexity Is Excessive FD 46058

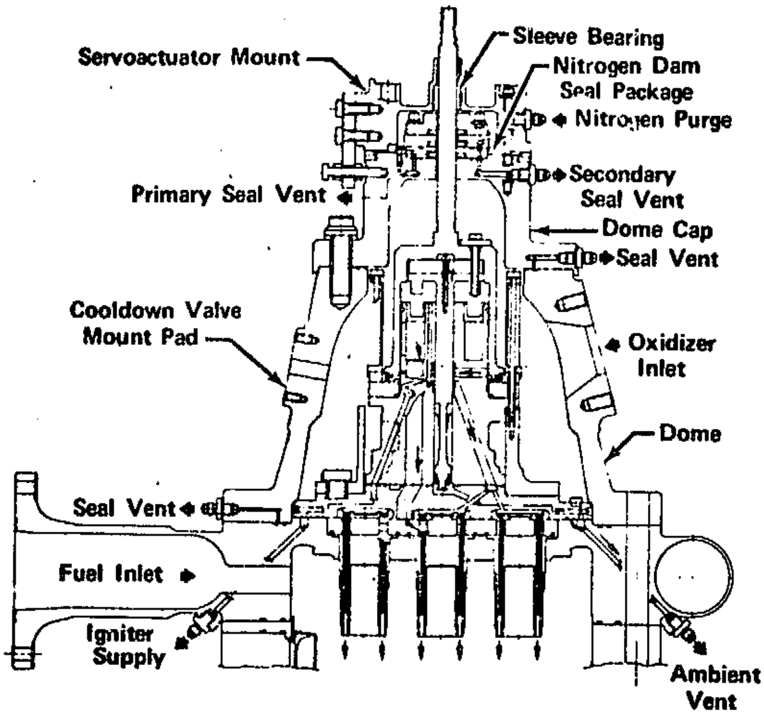


Figure I-3. Oxidizer Flow Divider Valve, XLR129, Phase I (250K) FD 46390

The dome housing has a 180-deg ring manifold with centrally located oxidizer inlet and a boss at one end for mounting a recirculation valve. A baffle plate inside the manifold directs incoming oxidizer through the ring manifold and into the dome. Flow enters through 14 ports and vents out of four ports through the recirculation valve. This prechill flowpath has proven entirely effective during tests of the similar 250K design. The dome periphery incorporates five fluid bosses for 0.250-in. tube connections to the engine system. Oxidizer supply to the igniters is obtained by internal tap-off from one of the 10 radial primary passages in the dome. Three helium inlets are provided for preburner purging at engine shutdown. Two holes communicate with the secondary injector cavity and a single hole intersects another primary passage. A shaft seal vent to the balance piston is incorporated, consisting of holes through the dome and valve guide, with a jumper tube that bridges the primary manifold for access to the balance cavity. A sixth boss is added for borescope insertion into the dome housing cavity for maintainability inspection of the engine mounted valve.

The dome has an integral bottom plate that partially covers the injector and supports the valve. The advantage of this design is the singular flange joint to restrict overboard oxygen leakage. It also minimizes preburner weight, complexity, overall length, and flow path volume. Alternative designs considered a separable injector top plate to provide an externally mounted valve concept for ease of service. Trade study results and one alternative design are presented in Valves and Interconnects Trade Studies, PWA FR-4442.

The valve guide and dome contains 10 radially spaced passages and 24 contoured ports for metering and transmitting primary flow to the injector. This flow was controlled by nine fixed orifices in the 250K design, but the SSME inlet pressure variation of 20 to 225 is so broad that a 12.8:1 area turndown ratio during the engine start is necessary to provide uniform liquid oxygen flow to the igniters and preburner for consistent engine starting. The solution chosen is a scaled-down version of the secondary port scheme to avoid new problems with an untested approach. Figure I-4 illustrates the geometry and relative locations of both sets of metering orifices.

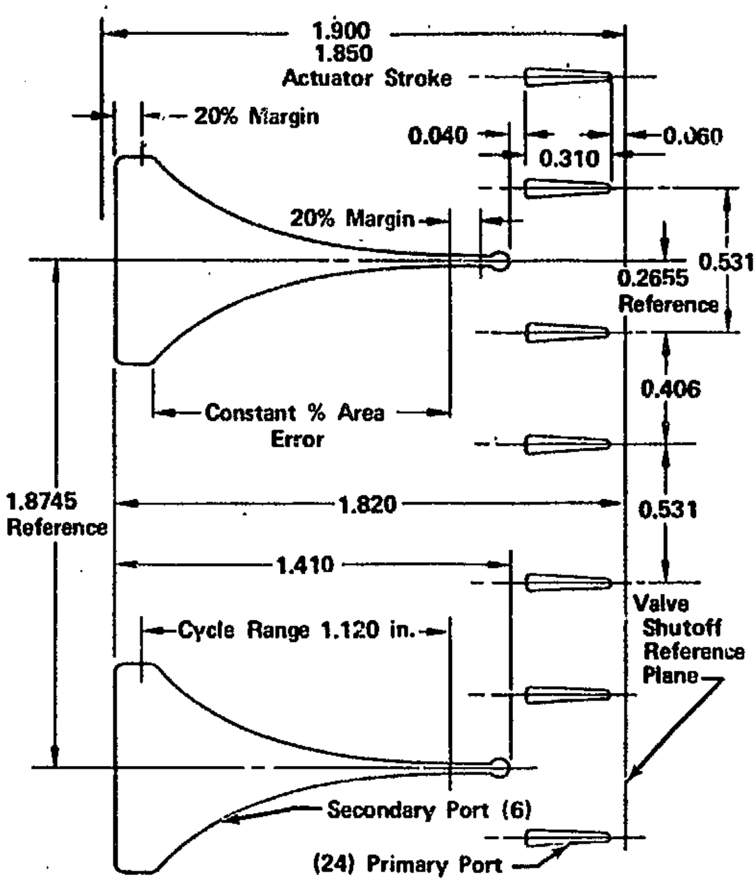


Figure I-4. Contoured Ports for Precision Area Scheduling FD 46064

There are six secondary ports that are contoured to produce consistent flow rate accuracy over 95% of the cycle operating range from 20% to EPL. These ports incorporate 20% excess area and reflect flow coefficient data obtained from 250K valve calibration tests, thereby ensuring correct port sizing. The valve flowpath schematic shown in figure I-5 presents results of computed losses and liquid oxygen volumes for the primary and secondary flowpaths at design point; i.e., 100% thrust and 6.5 mixture ratio.

Valve actuation stroke of 1.850 in. is the result of limiting the secondary discharge velocity into the injector cavity to 200 ft/sec (which defines the valve cylinder diameter), maintaining a minimum shutoff seal diameter and short valve stroke. The valve motion schedule versus effective area is shown in figure I-6. The valve opening force is 2900 lb when starting under 148 psia inlet pressure, and diminishes to 530 lb total force to open against the 20 psia minimum. Operational loads are 2160 lb maximum at 100% and r = 5.5, consisting of drag forces from the shaft seals and piston rings, fluid pressure forces and 120 lb of spring load. At 20% thrust, the total actuation load is 600 lb in the deceleration mode and only 350 lb when opening during engine acceleration.

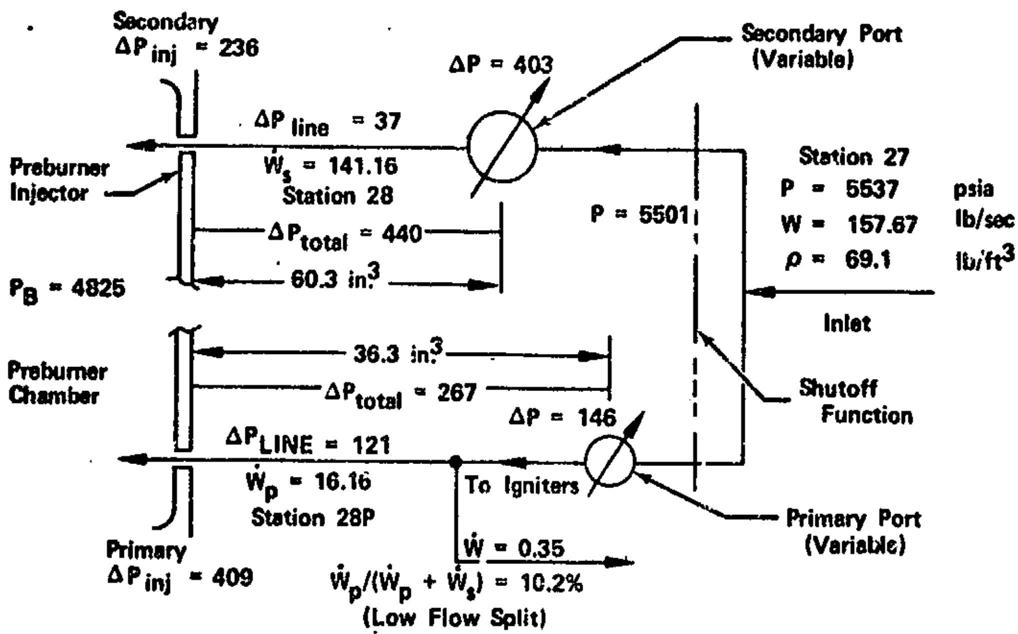


Figure I-5. Low Oxidizer Volumes Typify the Close-Couple Valve-to-Injector Design Concept FD 46065

Piston ring drag reaches a total of 500 lb at 1365 psi maximum pressure drop across the secondary ports. Three and a half years of testing 250K-sized rings has produced a predictable, low drag, pressure-balanced design with acceptable leakage and durability. They are fabricated from Berylco 25 and rub against chrome coated eight micro-finished, Inconel 718. An alternative molybdenum-chromium plating offered better wear properties but was rejected because coating results were too inconsistent for acceptance. Details are discussed in Valve and Interconnects Trade Studies, PWA FR-4442. Balance grooves in the lower ring are carefully arranged to avoid interference with any metering ports to prevent any effective area disturbance. Dowel pins are used to maintain angular alignment between this ring and the metering ports in the valve guide.

The shaft seals are PWA-developed laminated plastic lip seals. The Teflon and Kapton materials have consistently demonstrated acceptable wear rates, drag loads, and leakage control at SSME cycle conditions. Maximum friction force from both lip seal packages in the valve design is 120 lb at the 6841 psi maximum cycle pressure inside of the preburner dome. Teflon omniseals and other commercial types were tested and rejected because of poor durability at high pressures. Description and test data are provided in Valves and Interconnects Trade Studies, PWA FR-4442. The bearings that guide the translating valve are Berylco 25 bushings that mate with the chrome coated Inconel 718 valve shaft, using lapped surface finishes. Optimum alignment between the two supporting bushings is achieved by positioning the dome bearing/seal package laterally to center freely on the valve shaft. A compact double-ball universal coupling design is used to join the actuator and valve shafts. An eccentricity allowance of 0.040 in. is provided which tilts the coupling a maximum of 3 deg. Bearing radial loads are approximately 110 lb. Maximum bearing stress in the coupling is 11,100 psi, occurring under the 2900-lb valve opening condition. The bearing stresses drop off to the 2000 to 8500 psi level over the cycle range of 20% to EPL operation. Stress margins are therefore conservative in relation to our 30,000 psi allowable.

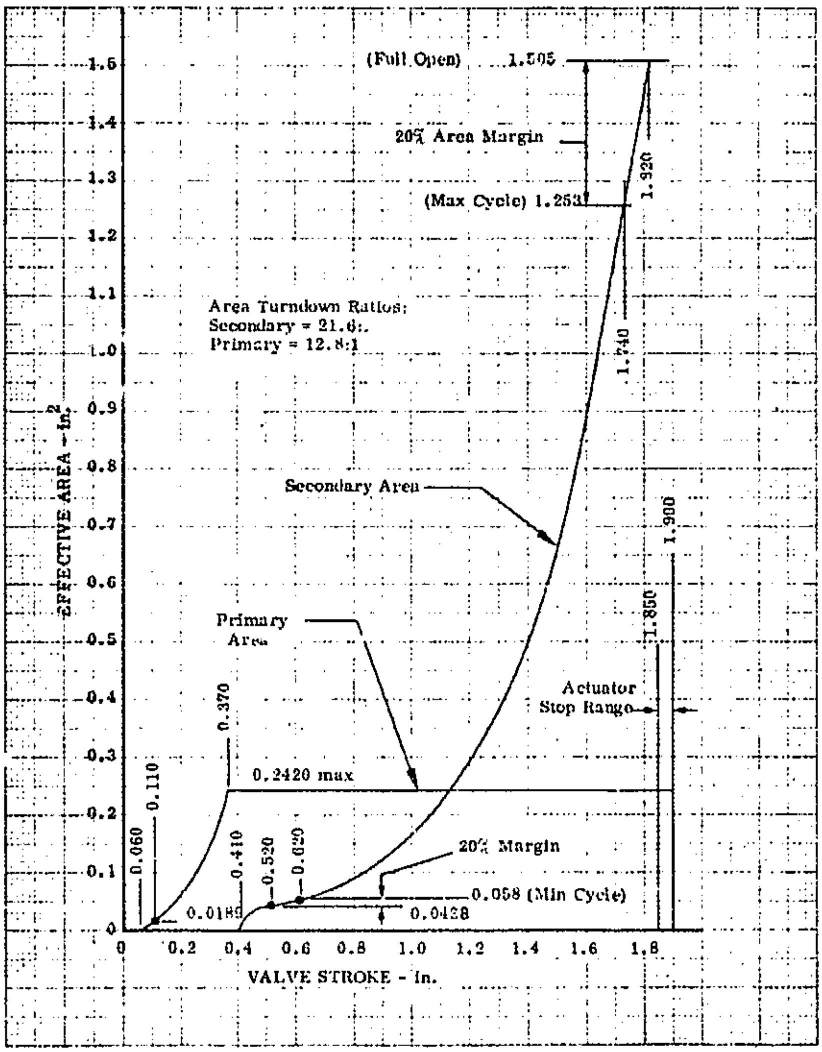
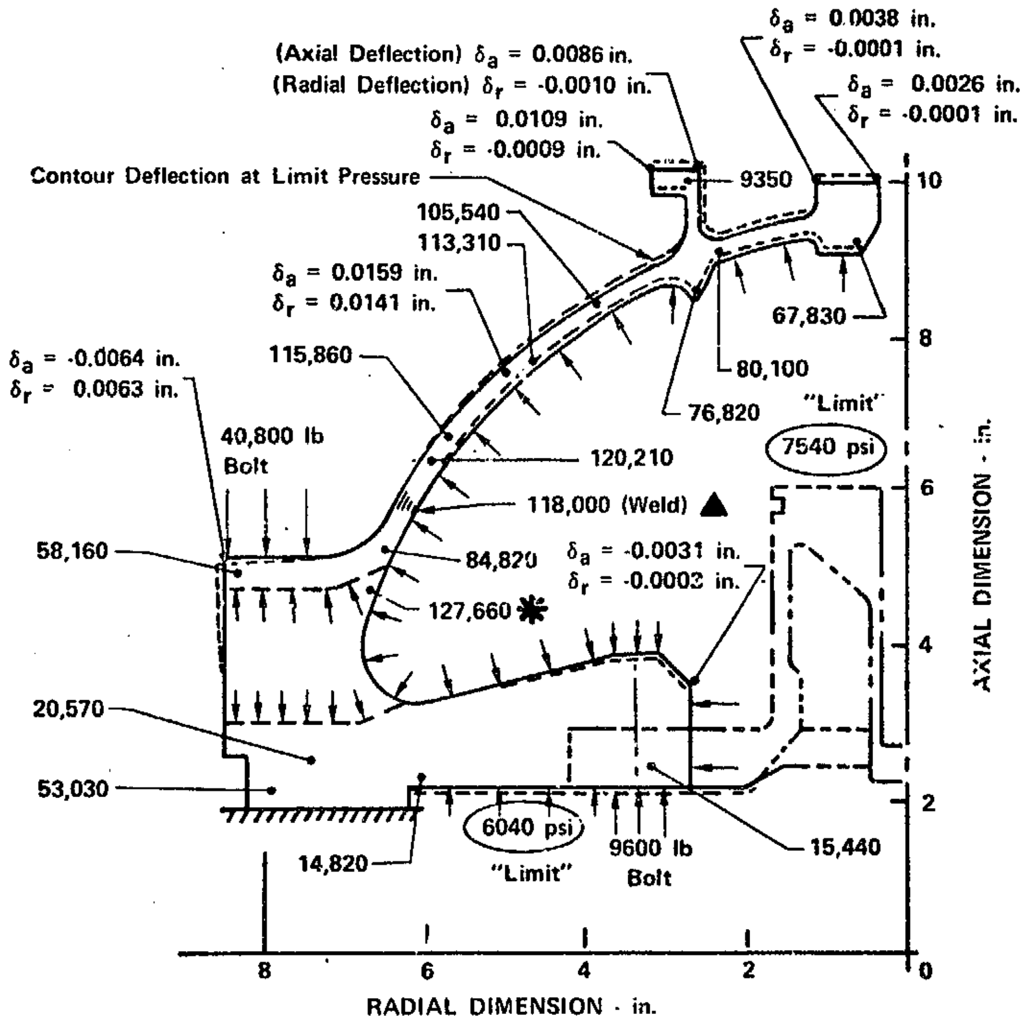


Figure 1-6. Metering Area vs Stroke Shows Flow Dividing Preburner Oxidizer Scheduling DF 84750

The preburner housing is an Inconel 718 hemispherical structure that is fusion welded into a single part from several forged components. The relation of the sphere to the major flange preserves the line of action concept which effectively prevents flange rotation under high pressure.

This condition avoids any loss of static seal performance at the injector joint. A finite element structural analysis of the housing is summarized in figure I-7. The margin of safety at 7540 psi limit pressure is 0.062 based on 203,000 psi "A" basis ultimate, at 220°R operating temperature. The inner flange is low stressed because thickness is set by inlet and primary flow passages. Housing axial deflection between the inner flange and actuator mounting flange is only 0.014 in. This relative motion between the actuator and the valve secondary ports is acceptable with our closed-loop control system. The oxidizer inlet and recirculation valve flanges are structurally sized to comply with Plumbing Design Criteria PWA FR-4455 for bolt selection and preload requirements, and to set the flange thickness for a total axial deflection at the seal contact point of 0.002 in.



Dome Housing - Maximum Effective Stresses and External Deflections

Computer Deck No. 6174

Material: Inco 718 at -240°F

Margins of Safety Are:

✱ 0.070 at Proof and 0.062 at Burst Pressure

▲ 0.042 at Proof and 0.032 at Burst Pressure

Figure I-7. Lightweight Spherical Housing Encloses Preburner Oxidizer Valve FD 46426

The housing studs are designed to meet the 1.4 factor of safety on the ultimate strength of the MP35N stud material. The allowable stress of 181,500 psi at room temperature occurs at the 40,000-lb maximum specified assembly preload. The minimum preload requirement of 32,000 lb accounts for blowoff load at 6200 psi limit pressure, seal preload at 28,300 lb, and external moments. These are caused by the wet weight of the valve and actuator assemblies under engine acceleration loadings and the oxidizer inlet pipe design moment. A 5200-lb loss of preload is also included for a 100° thermal transient between the stud, housing, and injector.

3. Design Requirements

The proposed design is in compliance with all requirements of CEI Specification CP2291. Specific specification requirements that influenced the concept selection and design elements are as listed:

1. Engine performance precision in accordance with paragraphs 3.1.1.3 and 3.1.2.4.

   Compliance - The valve design requires the use of "constant percent area error" metering ports, since preburner oxidizer flow control is influenced by valve position repeatability.

2. Thermal conditioning within 30 minutes in accordance with paragraph 3.2.6.4.

   Compliance - The fully-immersed oxidizer valve, mounted proximate to the injector, provides an optimum flowpath arrangement for oxidizer recirculation.

3. Preburner chamber pressure oscillations must not exceed ± 5% of steady state values in accordance with paragraph 3.2.9.1.

   Compliance - The close-coupled oxidizer valve holds the volume upstream of the injector to 60 cubic in. to ensure flow stability margin at all thrust levels.

4. Simultaneous H_2 and O_2 dumping is not permitted in accordance with paragraph 3.2.12.

   Compliance - Separate seal drains are provided at shaft seals to vent oxygen leakage to engine manifolds for safe disposal.

5. Components must withstand exposure to sand, dust, salt spray, and 100% humidity in accordance with paragraph 3.4.

   Compliance - The valve is enclosed and the shaft high pressure seals are protected with a shaft wiper element where the shaft is exposed to ambient.

6. Oxidizer inlet pressures shall vary from 20 to 225 psia in accordance with paragraph 3.5.2.1.1.

   Compliance - A 12.8:1 primary area turndown ratio is provided to schedule a consistent oxidizer flow to the igniters and preburner for reliable engine starting.

7. 7.5 hours time-between-overhaul service with 100 engine starts is a requirement in accordance with paragraph 3.6.1.

   Compliance - Life-limited bearings and dynamic seals are designed to pass a 10,000 cycle endurance test.

8. Engine shutdown due to component malfunction to occur without damage to adjacent systems, in accordance with paragraph 3.6.3.

   Compliance - This valve is designed to be self-closing in the event of actuator or coupling malfunctions, and to withstand 7540 psi limit pressure across the closed valve without the risk of structural failure.

Additional design criteria derived from the P&WA performance cycle or pertinent XLR129 development experience are:

1. Design point for metering port sizing is 100% thrust and 6.5 mixture ratio. Cycle area, flow rate, and pressure values are:

   a. 1.253 $in.^2$, secondary eff area

   b. 158 lb/sec secondary and 16.1 lb/sec primary flow

   c. Flow split: Primary = 10.2% of total flow

   d. Overall secondary ΔP = 440 psi

   e. Overall primary ΔP = 267 psi

2. Structural design limits occur at 100% thrust and 5.5 mixture ratio. Cycle pressure values are 6880 psi maximum internal pressure and 1365 psi pressure drop across the valve secondary.

3. Smallest secondary port area occurs at 20% thrust and 5.5 mixture ratio. Value is 0.058 $in.^2$ which establishes a 21.6:1 turndown ratio.

4. Oxidizer exit velocity must be 200 ft/sec maximum into the injector for good flow distribution to obtain a flat mixture ratio profile across the preburner combustion chamber.

5. Valve actuation loads must be minimized to avoid excessive actuator unit weight, or hydrogen consumption by venting the valve inner shaft section to ambient pressure and limiting shutoff seal diameter.

6. Primary flow tap-off provided for the liquid oxygen source to both igniters, and three helium inlets for purging the primary and secondary injector cavities at engine shutdown.

7. Provide 20% area margin in the secondary ports over cycle requirement for injector rematching if necessary.

8. Provide borescope access port for inspection of:

   a. Shutoff seal

   b. Shutoff lip on valve

   c. Lower piston ring

   d. Ring retainer

   e. Spring

   f. Wear surface and ports in guide

4. Design Capability

Oxidizer flow rates 20% greater than the present 550K cycle are possible with no ΔP increase because extra secondary port area is incorporated. This margin can be increased to 30% with minor changes in port contour, and with no significant increase in actuator stroke requirement. The initial 20% excess is to provide flexibility for possible injector change or to compensate for any valve inlet pressure deficiency.

5. Design Substantiation

Performance data that are incorporated into the design are identified below in relation to the two sleeve valve models that have been tested. Test results are presented in the Preburner Oxidizer Valve Design Calculations PWA FR-4419.

1. 250K, Phase I design:

Tested Feature	Number Cycles	Data Sheet (Refer to FR-4419)
Actuation Forces	41,000	DF 56862
Shutoff Seal Leakage	11,000	DF 58030
Piston Ring Leakage	65,000	DF 58031 and DF 59537

2. 250K, staged combustion rig, Phase I preburner oxidizer valve and injector:

34 hot firings with 9 min, 33 sec run time

3. 250K, Phase II design (similar to figure I-1 previously shown, except for the use of fixed primary orifices):

The data listed are typical of that obtained for several calibration and endurance runs to qualify the preburner oxidizer valve and its actuator for combustion rig hot firings.

Tested Feature	Number Cycles	Data Sheet (Refer to FR-4419)
Piston ring drag force	10,000	DF 77781
Piston ring leakage	10,000	DF 74996
Shaft seal drag force	10,000	DF 77782
Shaft seal leakage	10,000	DF 77786 and DF 74995
Shutoff seal leakage	10,000	DF 77790
Balance piston seal leakage	10,000	DF 77787 DF 77788 DF 77789
Secondary effective area versus stroke	Calibration	DF 74997
Secondary port C_d versus stroke	Calibration	DF 74998

4. 250K, preburner combustion rig tests, with Phase II preburner oxidizer valve:

22 hot firings with 6 min, 14.9 sec run time, August 1969 through February 1970

5. 250K, staged combustion rig tests, with same Phase II preburner oxidizer valve:

14 hot firings with 251 sec run time through March 1971

B. MAIN OXIDIZER VALVE

1. Introduction

The main oxidizer valve is a dual-purpose valve located in the main flow path between the oxidizer turbopump and main chamber injector. The valve regulates oxidizer flow to the main chamber injector and provides positive oxidizer shutoff.

The P&WA butterfly valve design is based on testing that included endurance cycles for seal leakage and wear, flow calibration, operating torque, and structural integrity. In addition, the 250K butterfly design operated in a series of staged combustion rig hot firings, up to 100% power level conditions.

2. Design Description

The butterfly valve design selection for the main oxidizer valve shown in figure I-8 resulted from a seven-valve selection study of various valve types. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.) This study included sleeve valves and pintle valves and showed that a butterfly valve offered the simplest, most compact design for this application. Actuation power requirements are low and the range of operation is within a band that yields good flow metering characteristics.

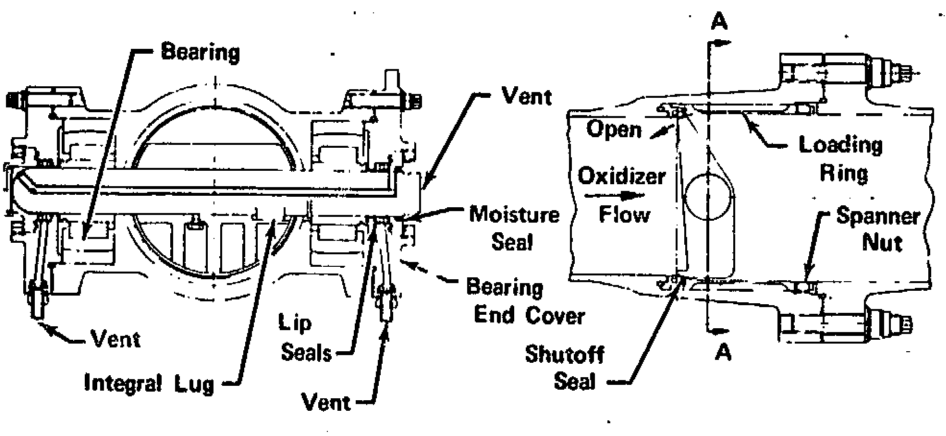


Figure I-8. Main Oxidizer Valve - Lightweight, Dependable, Dual Purpose FD 52317

The main oxidizer valve is positioned by a servo actuator as a function of the engine thrust level and the scheduled mixture ratio. The position is controlled by an error signal from the control system, which compares the actual position as a function of oxidizer flow with the desired requirement.

The main oxidizer valve has a regulating range of 1.856 to 11.1 in.^{2} effective area. This represents a 30% margin over the required maximum area to compensate for undetermined engine component variations. The effective area schedule as a function of disk angle is shown in figure I-9. Because of the shape of the butterfly disk, the valve has an approximate constant ratio of percentage of area change per degree of shaft rotation over the operating range.

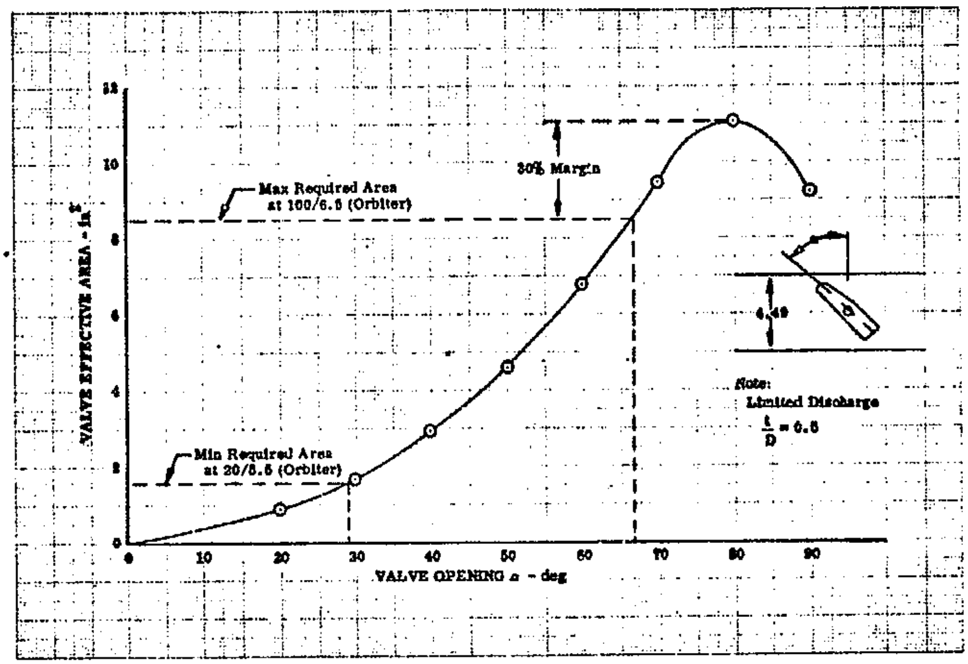


Figure I-9. Predicted Effective Area from Proven Experience DF 85704

Positive oxidizer shutoff is provided when the spherical surface of the valve disk is seated within a pressure energized, hoop-type shutoff seal as shown in figure I-10. The hoop-type shutoff seal was selected as the best of four shutoff seal designs built and tested after a study of 21 design concepts. Tests included shutoff cycle endurance (10,100 valve shutoff cycles) at cryogenic and ambient temperatures, and durability at high flow and high pressure conditions. The silver plated hoop seal was selected as it provided the most consistent endurance test results, met all of the design requirements for shutoff cycle endurance and sealing capability, and was still serviceable after all tests were completed. The tests and the four shutoff seal candidates are described in Valves and Interconnects Trade Studies, PWA FR-4442, together with the rationale for selecting the hoop-type shutoff seal.

The main oxidizer valve is driven by a rotary actuator bolted to the housing. Since the housing configuration does not permit an integral disk and shaft, a torque-drive feature is required to transmit shaft torque to the disk. Spline, tapered, flat, full-length key, and integral lug configurations were considered for driving the disk, but because of desirable features such as minimum backlash, good tolerance control, and no interference with the adjacent shutoff seal, the integral lug as shown in figure I-8 is the best solution.

Actuation power requirements are reduced by partially force-balancing the butterfly disk. This is done by cutting a spoiler notch in the front face of the disk. This results in more symmetrical pressure distribution across the disk and reduces the dynamic flow torque.

Total torque in the operating range is the summation of dynamic torque, bearing friction torque, lip seal friction torque, and the shutoff seal friction torque. Shutoff seal torques in the operating range and dynamic disk torque coefficient as a function of valve position are illustrated in Main Oxidizer Valve Design Calculations, PWA FR-4417. The total main oxidizer valve torque plotted against the valve rotational speed is shown in figure I-11.

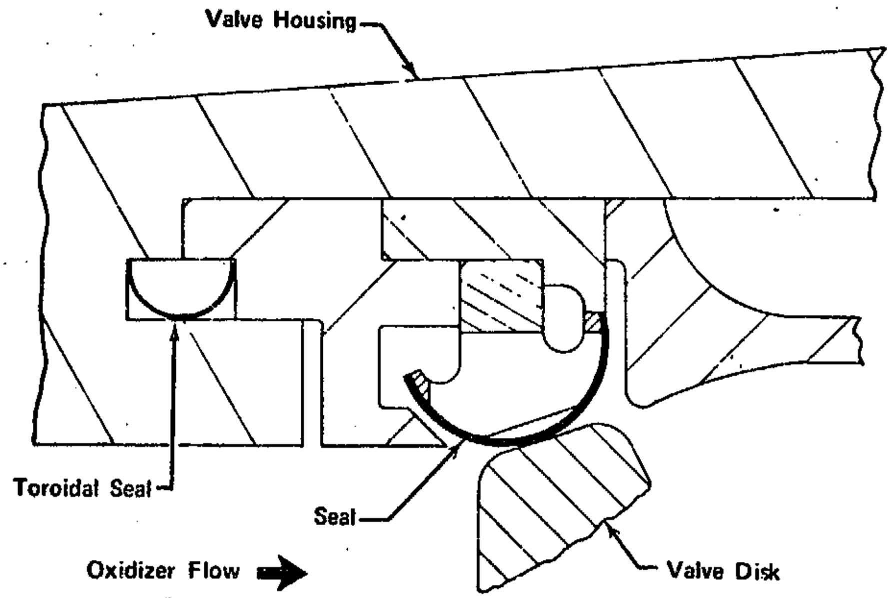


Figure I-10. Hoop Seal - Positive, Reliable Shutoff FD 52318

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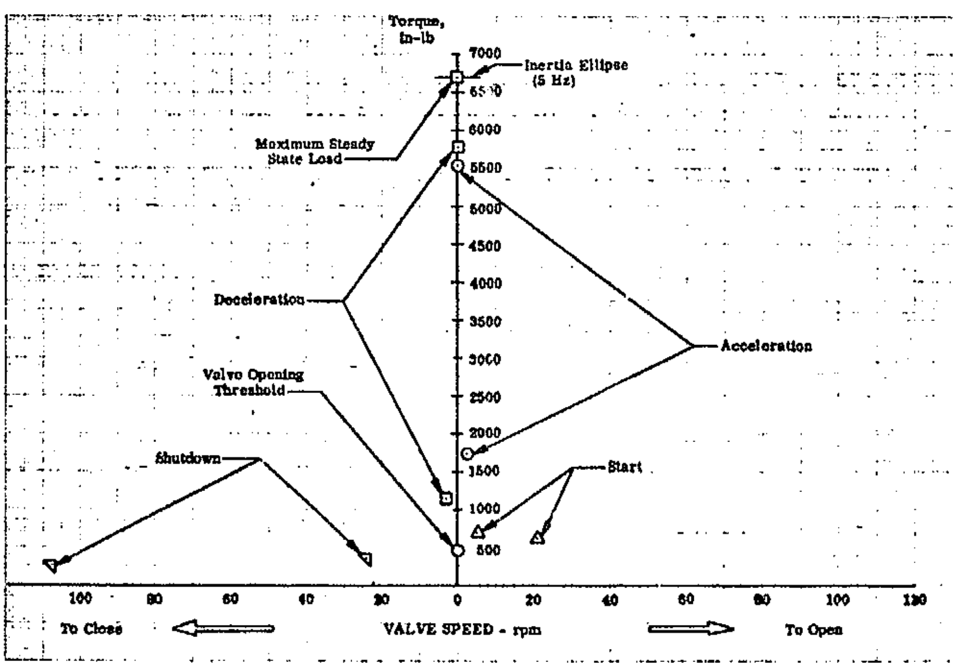

Figure I-11. Predicted Load - Speed from Proven Experience DF 85705

Shaft bending stresses, bearing loads, slopes, and deflections were computed on the basis of maximum pressure load across the valve disk (28,000 lb). This load is supported by two roller bearings which incorporate spherical outer races to allow for 0.00535 in./in. slope due to shaft deflection. These are full complement bearings made of 440C stainless steel (AMS 5630). The bearings are designed to withstand Hz stress levels of 260,000 psi. Larger Hz stress levels have been run on turbojet engines; however, cryogenic experience has indicated that higher Hz stress levels are a risk because of race fracturing at cryogenic temperatures.

Possible misalignment of 0.012 in. at the shaft spline is accommodated by a quill shaft that couples the valve shaft to the actuator shaft. Shaft end play and alignment required to locate the butterfly disk in the shutoff seal is controlled by a fitted spacer.

The shutoff seal element is a 0.005 in.-thick hydroformed Inconel X-750 (AMS 5598) hoop welded to an Inconel X support member. After heat treatment, the seal element is silver plated to enhance sealing and durability. A static seal is installed in the housing to prevent leakage around the shutoff seal as shown in figure I-10. A loading ring is used between the shutoff seal and spanner-type retaining nut. The purpose of the loading ring is to be part of the flow liner and provide uniform circumferential loading of the seal.

Double shaft lip seals of Kapton/FEP Teflon laminate are located outside of the roller bearings as shown in figure I-12. Each lip seal is constructed of three layers of Kapton sandwiched between two layers of Teflon. Actually, only one layer of Teflon is needed, but two layers foolproofs assembly by ensuring that Teflon, which has better rotating shaft seal wear characteristics, is always next to the shaft. A loading ring backed by a spanner type retaining nut loads the double lip seals to their required compression. The amount of lip seal compression is based on test data of similarly constructed lip seals. (Refer to Main Oxidizer Valve Design Calculations, PWA FR-4417.) As illustrated in figure I-12, leakage vents are located between the lip seals. Identical seal packages are used at both ends of the shaft. The valve bearing covers contain the double shaft lip seal packages. Test experience indicates very low leakage for lip-type shaft seals. Tests showed maximum primary lip seal axial and radial leakages of only 0.37 sccs after 10,000 shaft cycles, with pressures up to 6000 psig at liquid nitrogen temperature. Tests and configurations are illustrated in Valves and Interconnects Trade Studies, PWA FR-4442. This seal effectiveness allows a reduction in the total number of seals required by eliminating separate static seals, since it seals radially as well as axially. By venting the shaft at the "blind" end, shaft axial thrust load is prevented and a thrust bearing is not required.

A reversed Bal-Seal is used at the drive end of the shaft to prevent external dirt and moisture from entering the valve. The high pressure toroidal segment static seals are Inconel X-750 (AMS 5598) metal rings. To provide constant sealing for the static seals, the bearing covers are deflection limited (0.002 in. maximum). Low pressure static seals are fluorocarbon gaskets of TFE Teflon film.

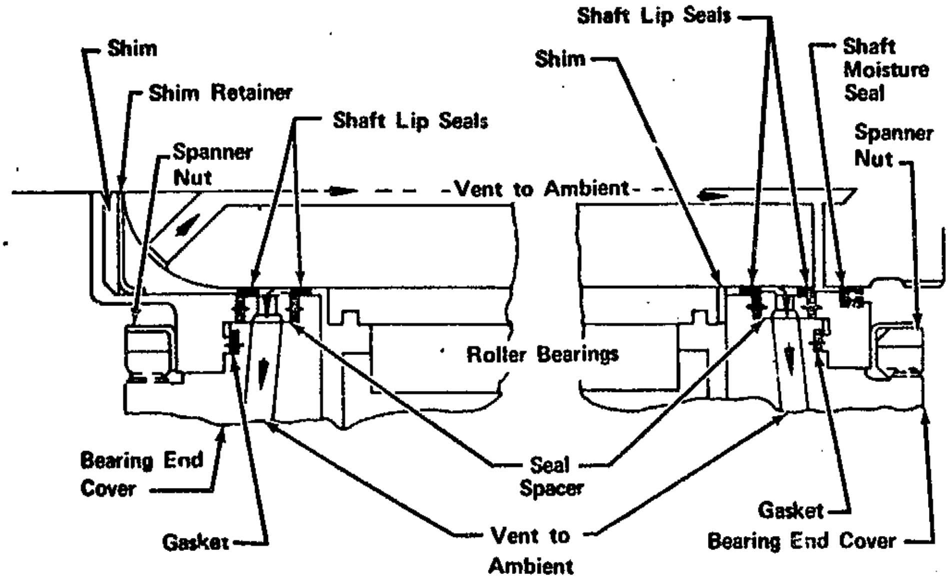

Figure I-12. Compact, Low Risk Seal Package FD 52327

The housing, valve disk, shaft, and bearing covers are machined Inconel 718 nickel alloy. This alloy offers high strength and good ductility at cryogenic temperature. Flange and bearing cover bolts are made of MP35N multiphase high strength alloy to help reduce weight. Devices that require material deformation for locking are soft stainless steel (300 series) because of its ductility. Silver plating is used on rubbing surfaces (thrust faces and bolts) to prevent galling.

The main oxidizer valve is designed to be integral with the main oxidizer plumbing line which is bolted to the main chamber injector. Integrating the main oxidizer valve with the plumbing provides the lowest weight and does not impair maintainability. Service will require removal of the line in any case, and valve service or element replacement can be easily accomplished in the service area. A main oxidizer valve trade study shows that this arrangement is the lightest, most compact design. It also eliminates two high pressure static external seals and reduces the effects of thermal lag in the bolts.

3. Design Requirements

The main oxidizer valve design meets all the requirements of CEI Specification No. CP2291, however, the following is a list of pertinent requirements:

a. CEI Specification No. CP2291 Requirements

1. Engine reliability and service life of at least 100 starts for a total of 7.5 hr in accordance with paragraph 3.6.1. Compliance. The valve is designed for a minimum of 10,000 valve cycles, 100 starts for a total of 7.5 hr; bearings 5 Hz or 135,000 cycles.

2. The engine shall be capable of continuously variable thrust and mixture ratio in accordance with paragraphs 3.1.1.1 and 3.1.2.

Compliance. The valve regulates oxidizer flow by varying effective area. (Reference figure I-9; effective area versus angular turn.)

3. Internal leakage of engine propellants or fluids shall not occur in such a manner as to impair or endanger proper function of engine or vehicle, in accordance with paragraph 3.7.12.

Compliance. The main oxidizer valve provides positive oxidizer shutoff with an energized hoop shutoff seal.

4. Structural criteria using factors of safety, "A Basis" material strengths and verification pressure in accordance with paragraphs 3.7.7.1, 3.7.7.1.1, and 3.7.7.1.2.

Compliance. Valve designed to SSME design criteria.

5. Flange joint design shall be in accordance with paragraph 3.7.13.1.

Compliance. Valve flanges are designed to PWA FR-4455, which includes all requirements specified in paragraph 3.7.13.1, including minimum bolt circle diameter and weight.

b. Pratt & Whitney Aircraft Imposed Requirements:

1. Provide equal percentage area/stroke design.

Compliance. Our design features a butterfly valve design which has good equal percentage area modulation characteristics.

2. Provide 30% effective area margin over maximum cycle requirements to compensate for engine component variations and provide growth capability.

Compliance. Maximum cycle requirement is 8.365 in.^{2}. Maximum valve area is 11.1 in.^{2} which gives a 30% margin.

3. Satisfy all operating conditions of design cycle 128D in accordance with paragraphs 3.1 through 3.1.3.

Compliance. Valve designed to all operating conditions of cycle 128D, including the following maximum parameters:

a.	Pressure	5710.4 psi
b.	Pressure drop	1747.7 psid

c. Flow 1095.96 lb/sec

d. Temperature 211.8°R

e. Effective area 8.365 $in.^2$

4. Design Capability

a. The main oxidizer valve has 30% effective area margin. Area margin is required to compensate for engine component variations and provides growth capability. Tests will verify how much of the 30% margin is actually needed.

b. The main oxidizer valve maximum flow rate can be increased by utilizing area margin.

c. Growth program can be met with redesign of valve housing and high pressure flanges. Other valve elements will require no change.

5. Design Substantiation

Butterfly type main oxidizer valve tests included proof pressure testing, environmental cycling, endurance cycles for seal leakage and wear, flow calibration, operating torque and structural integrity. The 250K staged combustion rig, Phases I and II main oxidizer butterfly valves operated in 32 hot firings for a total of 548.19 sec run time, and 14 firings for a total of 204.51 sec respectively. Refer to Main Oxidizer Valve Design Calculations, PWA FR-4417, for test results.

a. Butterfly Valve Shutoff Seal Configuration

Four shutoff seal designs were built and tested: silver-plated hoop seal, cam actuated seal, strap actuated seal, and loose leaf Kapton F-FEP Teflon seal. Testing included installing the seals in representative butterfly valve rigs and subjecting each of them to 10,100 valve shutoff cycles at cryogenic and ambient temperatures and pressures up to 6000 psi. The silver-plated hoop seal was selected as it met all of the design requirements for shutoff cycle endurance and sealing capability and was still serviceable after all tests were completed. Two of the other three schemes were unsatisfactory in shutoff durability, the other failed during subsequent flow tests.

b. Shaft Lip Seal Configurations

A Teflon-Kapton laminated lip seal package was selected because of its superior sealing, durability, and packaging characteristics relative to five other shaft seal configurations tested. Tests showed the primary lip seal was capable of maximum leakage of only 0.37 sccs after 10,000 shaft cycles. Teflon omniseals and other commercial-type seals were tested and rejected because of poor durability under high pressure.

c. Flange Static Seal Configuration

A toroidal segment design was selected for the valve static seal configuration on the basis of consistent minimum leakage performance during rig testing over a pressure range of 50 to 7000 psi for 500 cycles. A lead-plated pivot ring seal was the closest alternative of the other seven basic configurations evaluated.

C. PREBURNER FUEL VALVE

1. Introduction

The preburner fuel valve is patterned after the 250K preburner fuel valve, which was flow tested and found to have excellent flow control characteristics. The valve function is to modulate flow only and since the fuel shutoff valve is upstream, no shutoff seal is required. The butterfly valve was chosen instead of a sleeve, pintle, or ball-type valve because it provides the best combination of lightweight configuration and flow-modulating characteristics necessary for this location (refer to Valves and Interconnects Trade Studies, PWA FR-4442).

2. Design Description

The valve disk is on an offset shaft that is supported by two roller bearings. The shaft as shown in figure I-13 is sealed by pairs of lip seals on each end, with a wiper seal on the open end of the shaft. All high pressure static seals are toroidal segment seals.

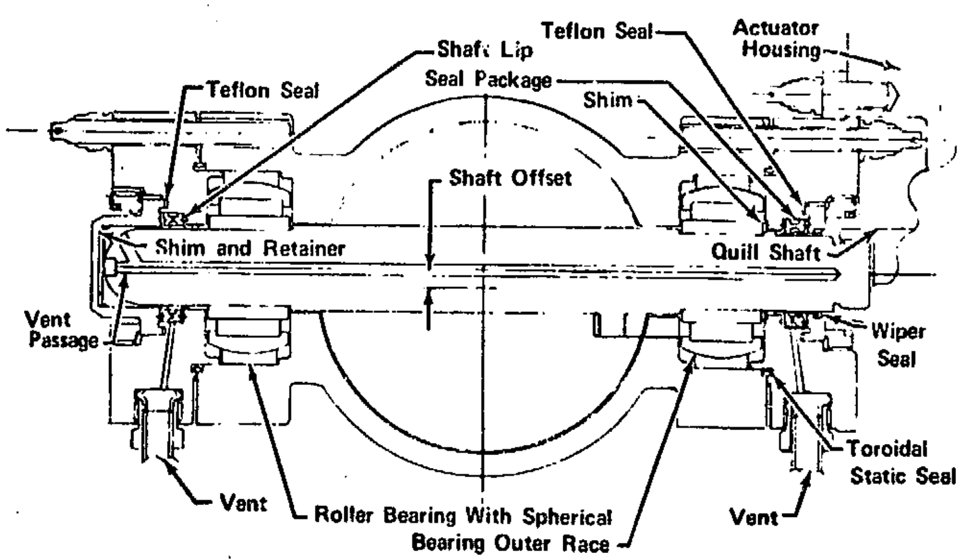

Figure I-13. Valve Shaft Cross Section FD 52314

The valve housing is welded to the preburner fuel plumbing line. A trade study that compared an integral valve housing line to a separate valve housing sandwiched between the preburner injector and fuel line flange showed significant advantages for the integral configuration. This eliminates one high pressure external seal, tie bolt weight, and reduces bolt thermal lag problems without hindering maintainability. (Refer to Engine System Trade Studies, PWA FR-4438.)

The butterfly disk has good aerodynamic contour with the upper half of the disk symmetrically contoured to reduce flow losses as shown in figure 1-14. Actuator torque is reduced by pressure balancing cut-outs on the lower half of the disk. The shaft is offset from the center of the disk so that fuel flow tends to open the disk providing a safe failure mode.

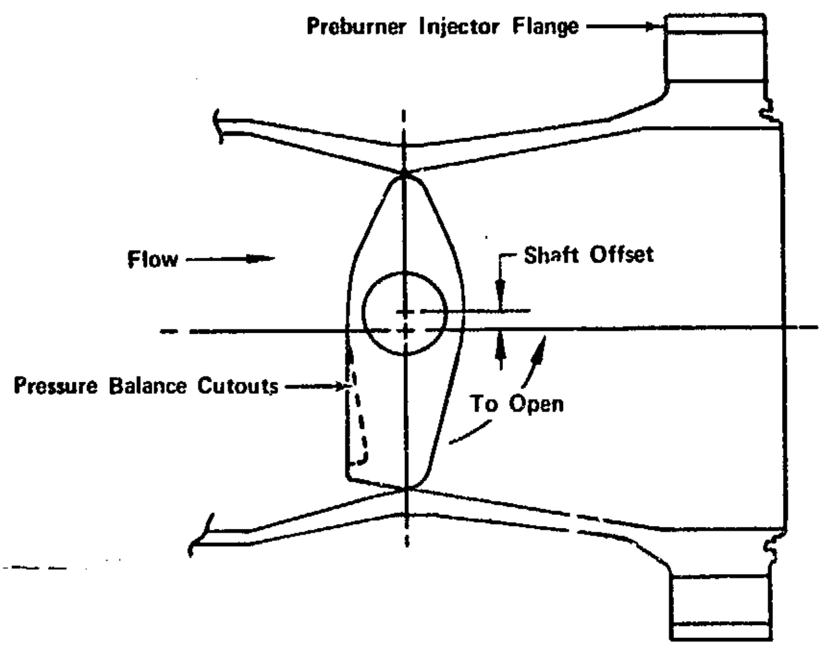

Figure I-14. Flowpath Cross Section FD 52316

The shaft lip seals are made of laminated Kapton and FEP Teflon. As illustrated in figure I-15, the lip seals are used in pairs, separated by a vented seal plate.

High pressure fuel that leaks past the primary lip seal flows through the vent passage to the fuel drain manifold for safe disposal. The secondary lip seal is used to seal the vent passage. Each lip seal has five laminations; an 0.005-in. thick layer of Teflon is on each outside surface to ensure good surface contact with the shaft, while three 0.005-in. thick layers of Type F Kapton are used in the middle. The Kapton layers have a high spring rate and will press firmly against the shaft. The lip seals extrude into the grooves on the seal back-up surfaces when compressed. This serves to hold the seals in place and controls radial leakage to the vent. A spring-loaded Bal-seal is used on the open end of the shaft to prevent contamination and moisture from reaching the lip seals. A hole is drilled through the center of the shaft to vent the end cap. This vent prevents excessive end load on the shaft and eliminates the need for a thrust bearing.

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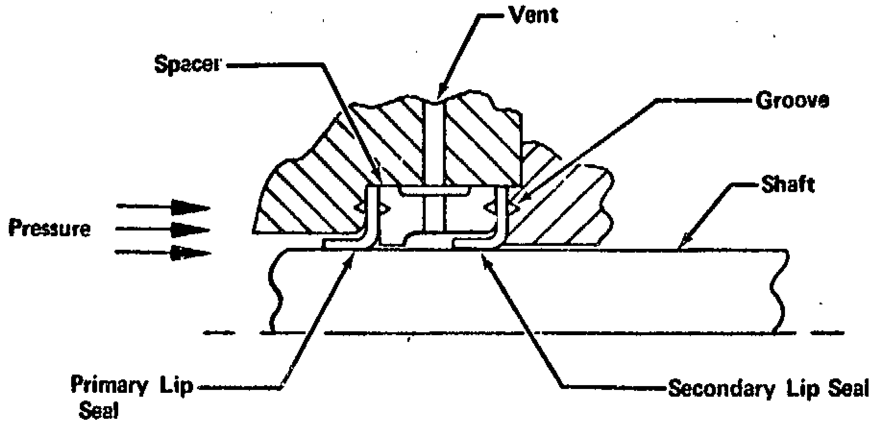

Figure I-15. Lip Seal Configuration FD 52315

The high pressure static seals are toroidal segment seals of lead plated Inconel X-750 (AMS 5542). The high pressure end cap flanges are designed for a maximum of 0.002 in. total deflection at the seal grooves. The larger high pressure flange at the preburner injector uses a wider toroidal seal and is deflection limited to 0.003 in. total at the seal groove. Trapped seals made of TFE Teflon film are used to prevent drain manifold leakage past the spanner nut threads.

The two full complement bearings that support the shaft are subjected to a maximum load of 3400 lb each. Each bearing has a spherical bearing type outer race which allows the inner race to align with the slope in the shaft. Perfect alignment of the shaft in the bearings under all load conditions prevents uneven loading on the rollers. By preventing uneven loading, we can use fewer roller elements and have a smaller bearing than would be required without a self-aligning outer race. To prevent race fracture at cryogenic temperatures, the bearings are designed for a maximum Hz stress of 260,000 psi. The outer race is slotted in order to assemble the spherical race.

The valve housing, shaft, disk, and end plates are all made of Inconel 718 (AMS 5663). This nickel alloy was chosen for its high strength and good ductility at cryogenic temperatures. All of the bolts are made of MP-35N, a high tensile strength alloy.

The disk has a 3.69-in. diameter which gives a maximum effective flow area of 8.55 in.^{2}, a 50% area margin over the EPL requirements. This is provided to allow for engine component variations, and degradation between overhauls. The valve is 64 deg open at EPL and is 31 deg open at 20% thrust as shown in figure I-16. Throughout the operating range, the valve maintains a nearly constant percentage of area change per degree of shaft rotation.

The valve butterfly is positioned by a rotary actuator bolted to the valve housing. The actuator can be removed without disassembling the valve, and is calibrated as a separate unit apart from the valve. A quill shaft is used to connect the valve shaft to the actuator shaft. The end of the valve shaft deflects 0.007 in. when under maximum load, and the quill shaft is necessary to relieve this misalignment. The lip seals are located close to the bearings so they can accept the shaft deflection.

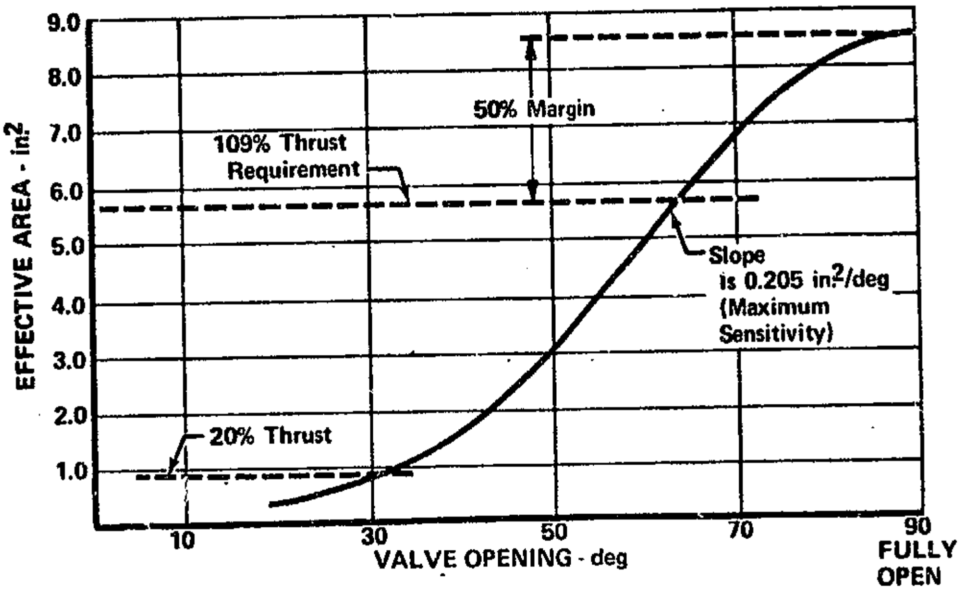

Figure I-16. Valve Capacity FD 52313

3. Design Requirements

a. CEI Specification No. CP2291 Requirements

The valve design meets all the applicable requirements of CEI Specification No. CP2291. The following is a list of the most important specification requirements:

1. The valve must be capable of satisfying all operating conditions in accordance with paragraphs 3.1.0 and 3.1.3.

   Compliance. The valve is designed for operation at the following worst case conditions:

   a. 6579.5 psia fuel pressure

   b. 8.5% pressure overshoot

   c. 635 psi drop across the valve

   d. 163.5 lb/sec flow

   e. 8.55 in.^{2} effective area (50% margin over the 5.7 in.^{2} required for EPL)

   f. 37° to 600°R temperature range

2. The valve must be capable of withstanding 100 engine starts and have 7.5 hr service life in accordance with paragraph 3.6.1.

   Compliance. The valve is designed for a durability of 10,000 cycles. The bearings are designed for 7.5 hr at 5 Hz or 135,000 cycles.

3. Valve components must be designed to meet the structural criteria listed in paragraph 3.7.7.1 and 3.7.7.2.

   Compliance. All components are with a 1.1 safety factor on the yield strength and a 1.4 safety factor on the ultimate. All pressure vessels are designed for a burst pressure of 1.5 times the limit pressure.

4. The valve must limit leakage to meet paragraphs 3.7.12 and 3.7.12.1.

   Compliance. External leakage is prevented by toroidal segment seals. Primary shaft lip seal leakage is vented to drain mainifold.

b. Pratt & Whitney Aircraft Requirements

1. The valve must have 50% margin in effective area to provide for engine component variations and degradation during the operating life.

   Compliance. The valve has 8.55 in.^{2} effective area when 90 deg open. This is a 50% margin over the EPL requirement.

2. The valve must shut off 95% of the fuel flow when closed. This is necessary to prevent cryogenic fuel from flowing through the valve and lowering the temperature of the warm fuel injected into the preburner during start transients.

   Compliance. The valve can shut off a calculated 97% of the flow when closed.

4. Design Capabilities

By increasing the flow velocity, the flow capacity can be increased without significantly raising the pressure losses.

Growth programs would necessitate increasing the valve housing and flange thicknesses but would not affect the internal components.

5. Design Substantiation

This valve design is based upon tested and reliable concepts proved during the 250K program. The 250K preburner fuel valve is a thick disk butterfly valve with a shutoff seal. Both the 250K and the 550K valves use the same types of static seals and shaft seals. The 250K valve was proof tested to 7300 psia with no resulting detrimental seal leakage.

Water flow tests were performed to determine the effective area characteristics of the 250K valve. These tests were also used to determine torque loads on the disk and friction factors of the lip seals. From this data, we calculated the emperical flow coefficients, lip seal friction coefficients, and the disk pressure load factors which were used to accurately predict all of the performance parameters of the 550K valve. The 250K butterfly valve has a shutoff seal, and therefore, the disk has a different contour from that of the 550K disk. It was necessary to modify certain emperical flow coefficients to allow for these differences in disk contours.

The shaft lip seals were tested during the 250K program and were proven to be superior to either Bal-Seals or omniseals. The Teflon-Kapton laminated lip seals used on a rotating shaft were tested for 10,000 shaft cycles and 500 pressure cycles at 6400 psig, all at cryogenic temperatures. The average leakage was 1.4 sccs of nitrogen through the primary seal, which is well below the P&WA standards of 10 sccs maximum leakage through a shaft seal.

D. TRANSPIRATION COOLANT VALVE

1. Introduction

The transpiration coolant valve is a cam-shaped butterfly valve which modulates the main chamber hydrogen coolant flow.

This modulation provides maximum specific impulse by closely matching cooling flow with chamber requirements during steady-state operations and improves main chamber durability by providing excess coolant flow during engine transients.

2. Description

The cam-shaped butterfly valve as shown in figure I-17 was selected over five other concepts because it was lighter, and because of undesirable features associated with other concepts. (Refer to PWA FR-4442, Valves and Interconnects Trade Studies.)

The rotary cam profile is a low gain design which provides small modulations in flow areas with large angles of rotation

A rectangular flow area cross section was selected as the best cam surface to meet the area requirements for standard steady state operation. Conventional fabrication methods such as broaching or slotting will be used to cut the rectangular passage through the valve housing. Percent of booster engine thrust and oxidizer-to-fuel ratio compared to required valve effective flow area and valve actuator rotation is shown in table I-1.

The valve shaft is supported by two roller bearings, each loaded to a maximum of 920 lb. The maximum design stress limit is 260,000 psi Hz stress, which is an acceptable level substantiated by cryogenic testing in hydrogen with bearings made of 440-C alloy. Dynamic test runs under this condition and stress level indicated no race or roller fracture.

The bearings are full complement bearings which permit the load distribution to be spread over the maximum number of rollers for a minimum Hz stress.

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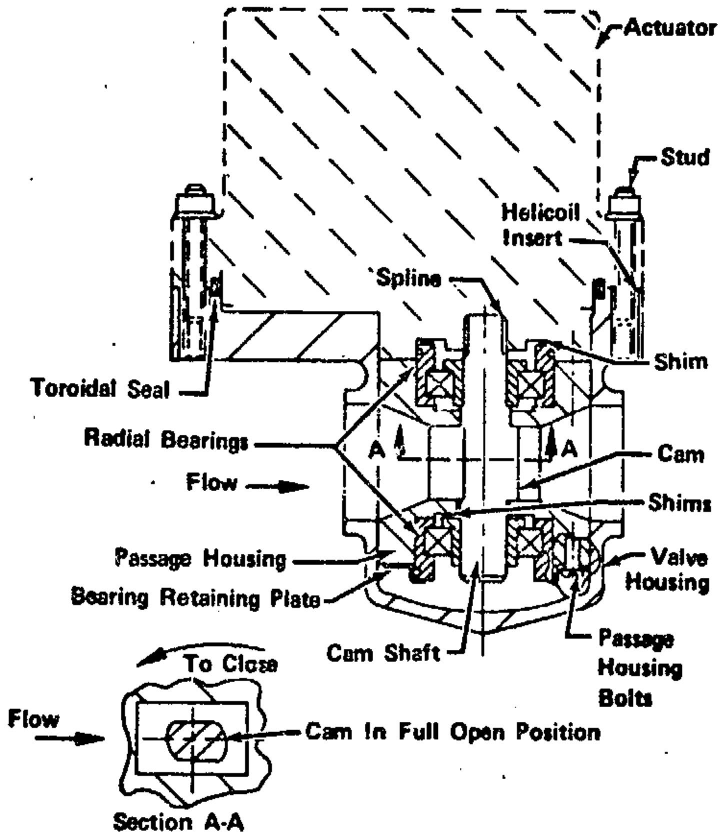

Figure I-17. Cam and Flow Passage Cross Section "Cam Butterfly" Transpiration Coolant Valve   FD 52236

The bearing service in this valve is mainly static and vibratory. The risk of bearing fretting is minimized by maintaining close manufacturing tolerances in the bearing assembly. P&WA computer deck A120 has been developed to ensure proper bearing installation fits to avoid overstressing bearings during assembly and operation.

The actuator and valve interface shown in figure I-17 is accomplished with a splined shaft connection and a high pressure toroidal section seal loaded and retained in a cantilevered flange housing configuration. Studs driven into the valve housing join the actuator flange and a dowel pin assures correct relative position between the housing and flange. The shaft spline has one tooth blanked to index the correct cam position.

The valve and actuator are exposed to the same pressure within the housing package. As can be seen in figure I-17 this pressure equilibrium eliminates the need for a shaft thrust bearing and no pressure differential exists across the flow passage housing wall. No shaft lip seals are required.

The toroidal section seal shown in figure I-17 is made of Inconel X-750 and has been substantiated by test under high pressure at cryogenic temperature. (Refer XLR129 Design Handbook, PWA FR-4461) on PWA static seal test rig 35120-39 and 40. Pressure cycles of 0 to 7000 psig were applied and leakage rates at 50 to 7000 psig were consistently low.

Table I-1. Percent of Booster Engine Thrust and Oxidizer-to-Fuel Ratio Compared to Required Valve Effective Flow Area and Valve Actuator Rotation

% Engine Booster Thrust	Mixture Ratio	Valve Effective Flow Area in.^{2}	Actuator Rotation From Start Position, Degrees
Start	5.5	0.286	0
20	5.5	0.161	85
50	5.5	0.143	100
75	5.5	0.129	108
100	5.5	0.114	125
20	6.0	0.177	65
50	6.0	0.159	85
75	6.0	0.145	97
100	6.0	0.130	108
109	6.0	0.126	110
20	6.5	0.191	45
50	6.5	0.174	74
75	6.5	0.159	85
100	6.5	0.145	97

Minimum weight is achieved by welding the valve housing into the plumbing line without hindering maintainability. The valve and actuator can be removed from the line for maintenance without disconnecting the plumbing. This arrangement eliminates a high pressure seal in one location and the probable requirement for tie bolts which could present a thermal lag problem. The weight reduction advantages for this concept are described in the trade study titled "Integral Butterfly Valve and Line," PWA FR-4442.

3. Design Requirements

The valve meets all of the requirements in CEI Specification No. CP2291 as follows:

1. The valve shall be capable of modulating coolant flow in accordance with chamber cooling requirements as a function of engine thrust levels and oxidizer-to-fuel ratios in accordance with paragraph 3.1.

Compliance - The valve effective flow area is variable from 0.191 in^2 to 0.114 in^2 to satisfy chamber cooling requirements over the oxidizer to fuel ratios range of 5.5 to 6.5 and thrust range of 2% to 109%.

2. Engine reliability and service life shall be in accordance with paragraph 3.6.1 required 7.5 hr service without overhaul and a capacity of 100 starts at normal power level.

Compliance - Valve housing is designed to withstand 400 pressurization cycles from ambient to maximum limit pressure of 6600 psi (includes a transient overshoot of 2.8%).

The valve bearings are designed to meet a 7.5 hr life requirement at 5 cps or 135,000 cycles at maximum limit radial loads of 920 lb.

3. The structural criteria, material properties and design allowables shall be as defined in paragraphs 3.7.7.1, 3.7.7.1.2, and 3.7.7.1.2.1.

Compliance - Safety factor requirements are 1.1 for minimum yield and 1.4 for minimum ultimate.

For pressure vessel design the proof pressure is 1.2 times the limit pressure at design temperature and the burst pressure is 1.5 times the limit pressure at design temperature.

All structural criteria in compliance with the above sections are based on Space Shuttle Main Engine Structural Design Criteria, PWA FR-4449, Plumbing Design Criteria, PWA FR-4455 and for Material Properties PWA Materials Manual "A Basis" strength values.

4. Design Capability

The valve is capable of opening to an effective flow area of 0.286 in^2 at engine start. This area is 50% greater than the maximum required effective flow area of 0.191 in^2 which occurs at 20% thrust and mixture fuel ratio of 6.5.

The valve flow area range can be altered by installing a cam shape with a revised profile. This may be accomplished with no rework to the valve housing unless the cam profile requires additional clearance at the installation slot.

The Phase I growth potential will not alter the effective area modulation range of the valve, but an increase in pressure difference across the cam butterfly may require the installation of larger bearings. The higher inlet and exit pressures of the growth program operating parameters may also require heavier housing walls and a thicker flange design.

5. Design Substantiation

The service environment for the transpiration coolant valve is applicable to the conditions set forth in the testing program of a PWA designed butterfly valve for the XLR129 high pressure engine program. The main chamber oxidizer valve testing is described in SSME Related Data (XLR129 Final Report), PWA FR-4460.

The valve housing was hydrostatically proof pressure tested to 6000 psig for 15 min and water flow calibrated at specified valve pressure differentials. 2500 shutoff cycles at 50 psid nitrogen pressure were then performed with the valve submerged in liquid argon and then 7500 cycles at ambient temperature. Cycling was done at one cycle per sec with 50 psid across the valve disk seal. The shutoff seal was also water tested to 1300 psid at ambient temperature with no evident leakage. The Phase I valve withstood 32 hot firings of staged combustion rig for a total of 548.19 sec duration. Also 14 firings or 204.51 sec on the Phase II valve was accomplished.

E. FUEL SHUTOFF VALVE

1. Introduction

The fuel shutoff valve is located in the main fuel line directly downstream of the high pressure fuel pump. The purpose of this valve is to provide low pressure drop and positive fuel shutoff. By welding the valve into the line, a substantial weight savings has been achieved (Refer to Trade Study, Fluid Components; Integral Butterfly Valve and Line, PWA FR-4442). The valve is a two-position ball valve having a maximum pressure loss of 30 psi at 109% thrust. The integral actuator is operated by helium pressure to open the valve and a downstream fuel pressure actuated detent is used to hold it open during engine operation if helium pressure is lost. The valve is closed by a spring when helium and fuel pressures are removed. The ball was the best choice of three valve types studied because it is lighter and more compact than the poppet and blade valves considered. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

2. Design Description

The fuel shutoff valve shown in figure I-18 is a two-position valve actuated by 700 psi minimum helium pressure. The valve is located in a 4.240 inside diameter line.

The valve ball is driven by a splined shaft through a rack and pinion. The ball is supported by two full complement roller bearings. These bearings have a maximum radial load per bearing of 1010 lb. A ball type bearing is used at the end of the pinion to accept the axial thrust of 1610 lb. These bearings are made of 440C stainless steel (AMS 5630) and are designed to a 260,000 psi maximum Hz stress level.

The ball valve uses a deflecting plate metal-to-metal type seal which gives the valve positive shutoff. The seal is supported by a thin metal cylinder that allows the silver plated lip to center on the chrome coated spherical surface of the ball and provides support when the lip is deflected by contact with the ball surface. The upstream pressure provides added sealing capability by causing the seal to act with greater force against the ball. This seal is held in place by a spanner nut and the flow of the fluid is guided through the ball by an inlet guide which serves to reduce pressure drop and prevent erosion of the seal. This entire seal package is made of Inconel 718 (Refer to Test Data on Ball Seal, Recirculation Valve; Propellant Recirculation Valve, PWA FR-4425.

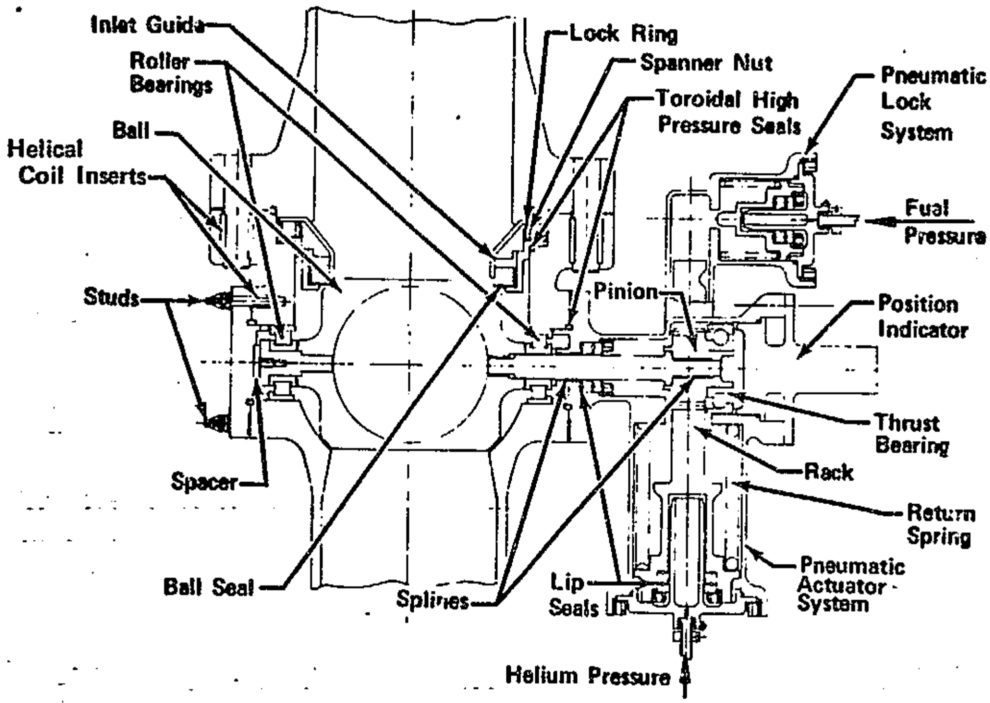

Figure I-18. Fuel Shutoff Valve FD 51910

Toroidal segment static seals are used to seal against external leakage (Refer to Valves and Interconnects Trade Studies, PWA FR-4442). Two Kapton/FEP Teflon laminated shaft seals are used to prevent high-pressure fuel leakage into the actuator; in addition, there are two seals on the actuator cylinder to prevent helium leakage. The actuator cavity will be vented to the fuel drain manifold. Low pressure flanges on the actuator make use of trapped TFE Teflon seals in double piloted grooves to prevent Teflon extrusion.

The helium actuator drives the normally closed ball through a rack and pinion, which is splined to the valve shaft as shown in figure I-19. A roller bearing supports the rack which is an integral part of the actuator piston. The material chosen for these parts is A-286 stainless steel (AMS 5737), which has good tensile strength and excellent impact strength at cryogenic temperatures. MoS_2 dry film lubricant is used on the contacting surfaces of these parts.

Provisions for a position indicator are incorporated, which will be bolted to the actuator housing and keyed to ensure proper orientation.

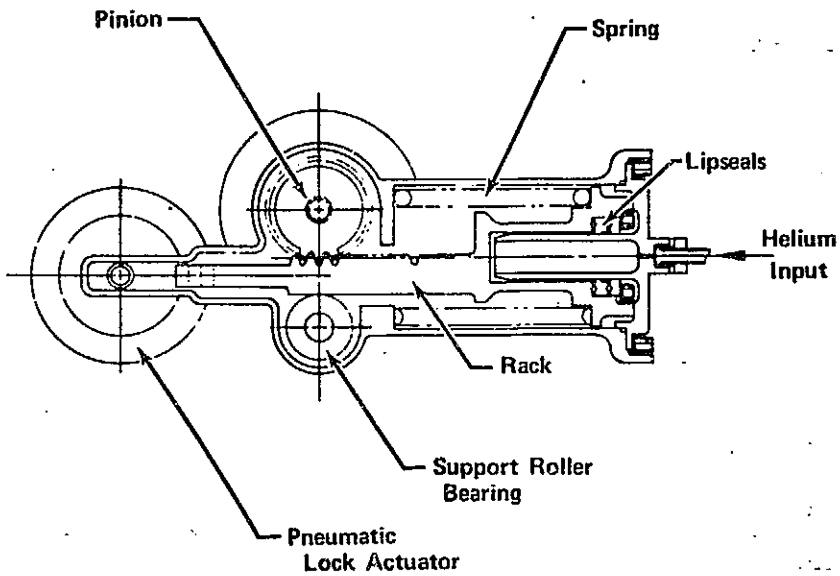

Figure I-19. Actuation System FD 46427

3. Design Requirements

a. The requirements of the applicable paragraphs of CEI Specification No. CP2291 have been met as follows:

1. The fuel shutoff valve shall meet the functional and/or performance requirements in paragraphs 3.1 through 3.1.3.

   Compliance -

   a. Maximum pressure = 6675.5 psi

   b. Maximum Δ P = 30.0 psi

   c. Maximum flow = 216.0 lb/sec

   d. Minimum temperature = 37°R

2. The fuel shutoff valve shall meet the structural criteria as established by paragraphs 3.7.7.1, 3.7.7.1.2, and 3.7.7.1.2.1.

b. Further requirements placed on design.

1. The valve shall have positive sealing when closed and a maximum pressure drop of 30 psi when open. It must have a durability of 150 operational cycles and 1500 check-out cycles.

Compliance - These requirements are fulfilled by the use of the deflecting plate seal which maintains positive sealing in the closed position. A 30 psi pressure drop through the valve is a direct result of proper flow path configuration and sizing of the inner diameter of the ball.

2. The valve shall open with an upstream pressure of 50 psi and close against an upstream pressure of 200 psi maximum.

   Compliance - This requirement is met by use of the valve's integral actuator, which when supplied with a helium pressure of 700 psi minimum, will provide sufficient force to open the valve. When pressure is removed, a spring generates enough force to close the valve.

3. The valve limit pressure shall be 7265.5 psi.

   Compliance - This requirement is met by sizing valve inlet flange, side plates, bolt arrangements and seal leakage paths to loads imposed by that pressure; all have appropriate safety factors.

4. Design Capability

The conditions at which the valve will open and close can be controlled by varying the spring characteristics while maintaining the same size and shape actuator package. Helium pressure, however is correspondingly related to the spring force, in that as the spring rate goes up, minimum required helium pressure increases to open the valve.

The growth potential of this valve is as follows: The valve will be capable of being used in the Growth Criteira; however, the main housing will have to be modified and flanges resized to meet the higher pressure limits. The internal components will remain the same.

5. Design Substantiation

Over 9000 RL10 engine firings including 90 firings of 76 flight engines with ball-type shutoff valves, conducted in a space environment, has supplied the substantiation for this valve design. Ambient temperature tests of all seals, and actuation pressure tests on similar smaller valves such as the recirculation valve have also been performed. Tests after extended storage have been conducted with no appreciable deterioration in the valves. (Refer to Propellant Recirculation Valve Design Calculations, PWA FR-4425.)

This design for the fuel shutoff valve is a feasible low-risk concept, and best fulfills our requirements for a helium actuated two-position valve with fuel pressure interlock.

F. FUEL BYPASS VALVE

1. Introduction

The fuel bypass valve allows gaseous fuel to bypass the preburner fuel valve for stable engine starts. The valve is normally open, and is closed by downstream fuel pressure during the start transient.

The valve is located in the fuel line from the intermediate heat exchanger discharge to the preburner injector inlet downstream of the preburner fuel valve.

2. Design Description

A ball valve as shown in figure I-20 is used in this application as a result of a trade study (refer to Fluid Components Trade Studies, PWA FR-4442 that compared poppet, blade, and ball valves. The study showed that a ball valve has significant advantages such as positive sealing, low pressure loss, light weight, and compactness. The basic valve dimensions were determined by design optimization of the flowpath through the ball element.

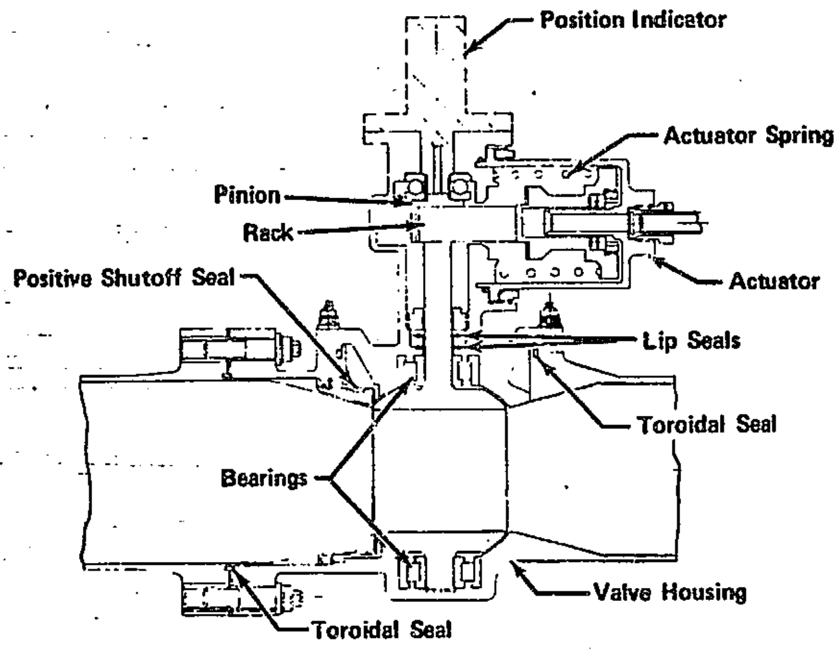

Figure I-20. Fuel Bypass Valve Cross Section FD 52336

The fuel bypass valve shown previously in figure I-20 is composed of an integral valve-actuator package. The ball and shaft are integral and a downstream metal-to-metal shutoff seal prevents internal leakage. The valve is held in the normally open position by spring force and is forced to the closed position by downstream fuel pressure.

A dual lip seal piston actuator drives the ball through a rack and pinion operator. The valve shaft and actuator lip seal leakage is vented to the fuel drain manifold through a No. 4 Dynatube connection on the actuator housing. The actuator is designed to initiate closure at 400 psia actuator inlet pressure.

The ball is supported by two full complement roller bearings with a maximum design load of 760 lb per bearing. The maximum thrust load of 1260 lb on the shaft is supported by a ball thrust bearing. All bearings are made of 440c (AMS 5630) material. These bearings were computer designed and incorporate state-of-the-art bearing technology based upon years of turbojet experience and 250K engine bearings operating at 6000 psi and cryogenic temperature. A maximum Hz stress level of 260,000 psi is used to prevent race fracturing at cryogenic temperatures.

Materials selected for the valve components include: AMS 5673, spring; AMS 5737, rack and pinion; Inconcel 718, housing; MP35N, bolts; AMS 5735, spanner nut; and AMS 5510, cup washer.

Seals include a deflecting plate shutoff seal, toroidal segment high pressure static seal, trapped Teflon low pressure static seal and Kapton/Teflon lip seals (for both translatory and rotary applications). The toroidal segment seal housing is designed to allow 0.002 in. total deflection at the sealing diameter. The Kapton Teflon lip seal lamination depends on the application. Rotary lip seals contain-layers of Teflon in contact with sealing surfaces and three layers of Kapton between (TKKKT). Linear lip seal applications utilize Kapton in contact with the sealing surface and three layers of Teflon between (KTTTK). Previous 250K tests have verified leakage rates of less than 10 sccs for these seals at operating pressures up to 6000 psi. (Refer to Valves and Interconnects, PWA FR-4442.) The shutoff seal incorporates a deflecting seal plate supported by a thin cylinder. An initial seal plate deflection of 0.005 assures positive sealing at all valve positions. The seal contact surface is silver plated in accordance with AMS 2410 to provide a good wear surface.

A rotary potentiometer position indicator is driven by the ball shaft. A combination of limited shaft rotation, rectangular drive, and offset bolts ensures proper installation.

3. Design Requirements

1. All CEI Specification CP2291 requirements have been met with particular attention given to the following:

   a. The engine shall be capable of 100 starts before overhaul in accordance with paragraph 3.6.1. Therefore, the valve shall be capable of 100 off-on cycles.

      Compliance - The valve is designed for a minimum of 150 operational cycles and 1500 checkout cycles.

   b. All engine components shall be capable of withstanding vibration, shock, and aerodynamic loads in accordance with paragraph 3.4.5.1.

      Compliance - The valve will be tested to withstand a two-g ground handling load in each orthogonal direction and a four-g longitudinal and six-g lateral acceleration load. The natural frequency of all valve components is above the 500 Hz maximum engine induced vibration.

c. All materials used in the design must be in agreement with paragraph 3.7.1.

Compliance - Inconel 718 is the basic valve material. Seal materials are Kapton/Teflon (lip seals) and Inconel X-750 (toroidal seals).

d. Structural design criteria shall meet the requirements of paragraph 3.7.7.

Compliance - An allowable yield stress safety factor of 1.1 and an ultimate stress safety factor of 1.4 are used. The design margin provides for a proof test capability of 1.2 times limit pressure without detrimental yield and a burst test of 1.5 times limit pressure without failure. Welded joints are limited to 90% of the parent material strength. Structural design values are per MIL-HDBK-5 "A" values.

2. Other design requirements include:

a. The valve must be single acting, two-position, and normally open.

b. The valve is closed by downstream fuel pressure.

c. Closed valve internal leakage is predictable and is minimized by a shutoff seal.

d. The valve begins to close at 400 psi increasing actuator inlet pressure.

e. The valve must be completely open with 200 psi decreasing actuator inlet pressure.

4. Design Capability

Increased flow requirements can be met by the fuel bypass valve without major redesign by increasing the allowable pressure loss. The valve can be made to operate at different transient pressures by changing the actuator spring. If different operating characteristics are required while the system pressures remain the same, the actuator piston area may be changed. The net result of these features is added growth potential.

5. Design Substantiation

Test results from the 250K engine demonstrate that a ball valve with positive shutoff seal will give excellent low risk performance in the fuel bypass valve application. The 250K recirculation valve (which is operationally similar to the 550K fuel bypass valve) demonstrated acceptable leakage rates (as required by CEI Specification No. CP2291, paragraph 3.7.12). This valve has also demonstrated a storage life of over four months. In addition, ball valve performance has been demonstrated by the RL10 fuel and oxidizer inlet valves.

G. PROPELLANT RECIRCULATION VALVE

1. Introduction

The recirculation valve is a two-position, normally closed, ball-type, shutoff valve. Three valves are required to recirculate gaseous and liquid propellants during tanking and hold. We selected a ball valve because it has the best combination of light weight and low actuation requirements of the three candidate valve types considered. The recirculation valves are common because performance requirements at the one fuel and two oxidizer valve locations are similar to the extent that the complexity of separate valve designs is not justified.

2. Design Description

We chose a spring-loaded ball valve based upon the results of a design selection study. (Refer to Valves and Instruments Trade Studies, PWA FR-4442.) This study shows a ball valve has a better combination of low weight, low actuator volume, and small actuator leakage circumference than blade or poppet vlaves, the two other candidate valves for the recirculation valves. A ball valve also provides the best package, smallest high pressure seal diameter, and the best contamination resistance due to seal wiping action. This design, shown in figure I-21, incorporates an integral ball and shaft that is actuated open by helium pressure and closed by spring force. The valve uses a metal-to-metal shutoff seal in the inlet.

We found that a common valve could be used at the three valve locations: on the preburner injector oxidizer supply manifold, upstream of the fuel shutoff valve, and upstream of the main oxidizer valve. This is because a SSME cooldown study indicates recirculation requirements are sufficiently similar at the three valve locations that the additional design fabrication, test effort and inventory for separate valve designs is not warranted. (Refer to SSME Childown Studies, PWA FR-4454.) The recirculation valves are sized to provide 0.90 $in.^2$ effective area, based upon the trade study to establish minimum recirculation systems weights.

The recirculation valve uses a shutoff seal similar in design to that used for the fuel shutoff valve and the fuel bypass valve. The shutoff seal shown in figure I-22, uses a deflecting seal plate supported by a thin metal cylinder that allows the silver plated lip to center on the spherical surface of the ball and provides support when the lip is deflected by contact with the ball surface. The seal carrier radially supports the thin cylinder at high valve inlet pressures. Propellant leakage is minimized by limiting the valve assembly high pressure sealing requirements to the shutoff seal. All shutoff seal material is Inconel 718 (AMS 5663).

The valve internal static seals are compressed, trapped TFE Teflon. These seals are exposed to static pressures of less than 275 psi. The seal recesses are double-piloted with a maximum loose fit of 0.0015 in. to prevent Teflon extrusion.

The valve inlet and discharge static seals are toroidal segment seals. These seals were selected for their consistent minimum leakage performance over a pressure range of 50 to 7000 psi for 500 cycles. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.) Valve flanges have been designed to limit the total deflection of the seal to 0.002 in. to maintain effective seal point contact. Toroidal segment seal material is Inconel X-750.

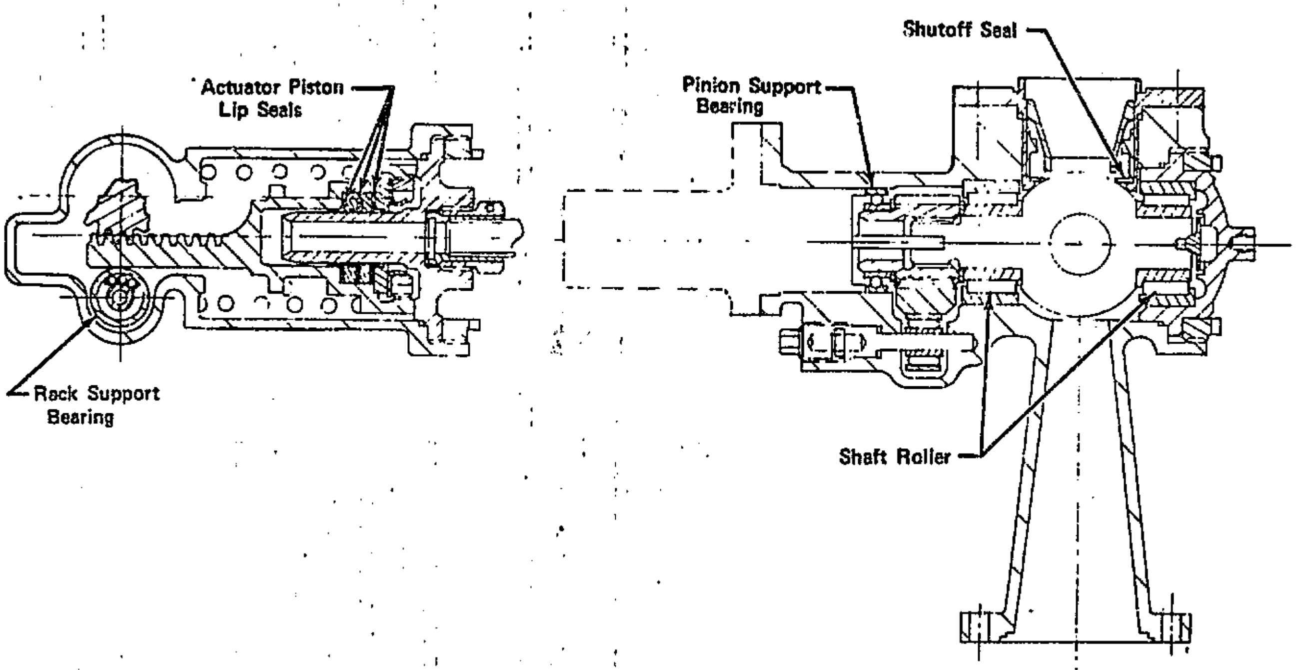

Figure I-21. Propellant Recirculation Has 0.9 in.^{2} Effective Area in Minimum Weight Package FD 52285

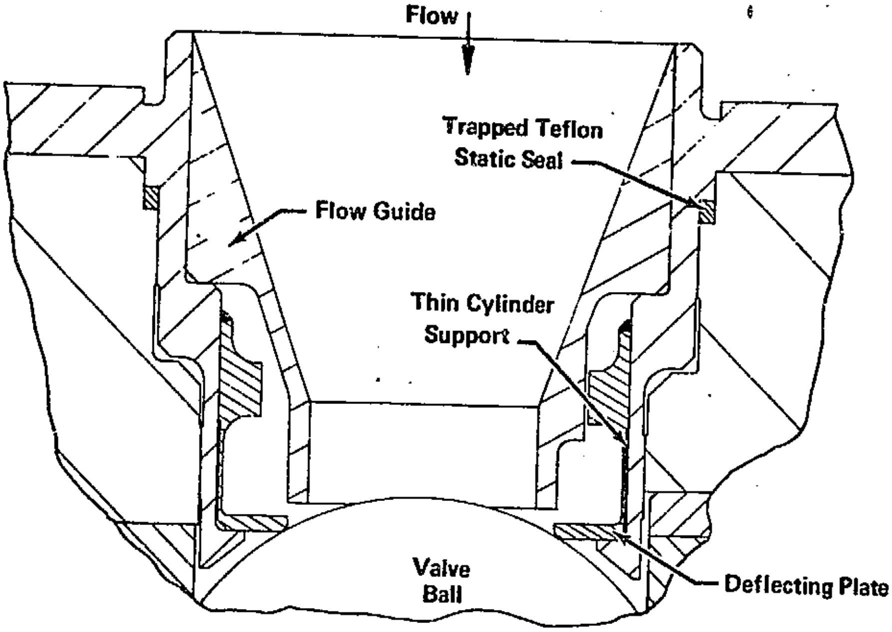

Figure I-22. Deflecting Plate Shutoff Seal Provides Positive Shutoff Metal-to-Metal Seal FD 52319

Two needle roller bearings support the ball shaft. The maximum static load per bearing is 2650 lb and the maximum dynamic load is 525 lb per bearing. These bearings are designed to a Hz stress limit of 260,000 psi to prevent race fracturing under cryogenic conditions. (Refer to Propellant Recirculation Valve Design Calculations, PWA FR-4425.) A full complement bearing supports the pinion. This bearing is lightly loaded and serves also to axially position the pinion and ball shaft. The rollers, balls, and races of these bearings are made of AISI 440c stainless steel (AMS 5630).

A rack and pinion converts actuator translation to ball shaft rotation. The rack is pneumatically translated and drives the pinion, which is splined to the valve shaft, to actuate the valve to the open position. The rack is supported by a lightly-loaded roller bearing. The maximum contact stress between the rack and pinion involute surfaces is 100,000 psi. We selected A-286 stainless steel for the rack and pinion because of its good tensile strength and excellent impact strength at cryogenic temperatures. The involute surfaces of the rack and pinion are coated with MOS_2 dry film lubricant to minimize wear. We selected a combination of 17 gear teeth and 18 splines for the pinion to facilitate alignment of the valve ball.

A double lip seal prevents helium leakage into the recirculation line. A single lip seal prevents propellant leakage into the helium line. These seals are laminated Kapton-F and FEP Teflon. We chose this seal type based upon its superior sealing durability and packaging characteristics over five other shaft and face seal configurations. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

A 17-7 PH stainless steel (AMS 5673) spring returns the valve to its normally closed position when actuator pressure is vented. We selected this spring material because of its high strength. The spring has a minimum fatigue life of 10,000 cycles.

The recirculation valve design provides for installation of a valve position indicator to monitor the valve position. The position indicator is a dual element rotary type potentiometer keyed to the valve body and the valve ball shaft.

3. Design Requirements

The recirculation valves are designed to meet the following requirements of CEI Specification No. CP2291 and Pratt & Whitney imposed requirements:

1. The recirculation valves shall allow recirculation of gaseous and liquid propellants during tanking and hold in such a manner as to prevent engine initiated geysering as described in paragraph 3.5.2.

   Compliance - The recirculation valves have a maximum effective area of 0.90 in.^{2} which allows flow rates necessary to prevent geysering in the vehicle inlet line.

2. The recirculation valves shall allow engine conditioning within the 30-minute time requirement as specified in paragraph 3.2.6.4.

   Compliance - The 0.90 in.^{2} valve effective area allows engine conditioning within the allowed time period.

3. The recirculation valves shall have a durability goal of 10,000 valve cycles in order to ensure compliance with the engine reliability and service life of paragraph 3.6.1.

   Compliance - All valve components subject to fatigue are designed to minimum of 10,000 cycles life.

4. The recirculation valve shall incorporate a positive shutoff seal to meet the fluid leakage requirements of paragraph 3.7.12.

   Compliance - The recirculation valve uses a deflecting plate type shutoff seal to minimize propellant leakage.

5. The valve shall be capable of actuation closed to open with 150 psi maximum upstream static pressure.

   Compliance - The recirculation valve has an integral actuator sized to actuate the valve closed to open with 150 psi inlet pressure.

6. The valve shall be capable of opening and remaining open with 150 psi maximum downstream pressure.

   Compliance - The recirculation valve integral actuator exerts sufficient force for the valve to open and remain open with 150 psi acting against the actuator piston.

7. The valve shall be capable of withstanding a maximum closed valve static pressure at the inlet of 7455 psi.

   Compliance - The valve inlet flange, seal package, ball shaft, and bearings are designed to the loads imposed by 7455 psi inlet limit pressure with appropriate safety factors applied.

8. The recirculation valve shall be designed in accordance with all applicable requirements of CEI Specification No. CP2291.

   Compliance - A list of all applicable specification requirements has been compiled in the DVS and the recirculation valve has been designed to comply with this list.

4. Design Capability

The recirculation valve will open against 600 psi higher inlet pressure, a 400% increase, by providing 800 psi higher helium actuation pressure. To meet Phase I and Phase II growth potential requirements, the valve inlet flange, seal package, and bearings must be redesigned to the loads imposed by the increased limit pressures. The valve integral actuator will meet the growth potential requirements as presently designed.

5. Design Substantiation

The selection of the ball valve concept for the propellant recirculation valve is based upon the successful performance of the RL10 propellant inlet shutoff valve and on the successful initial testing of a propellant vent valve (designed and fabricated for the XLR129-P-1), both of which use the ball valve concept.

The RL10 propellant ball-type inlet valves have been used in space in 76 engines for 90 firings with no failures. Over 9000 total engine firings for 1,142,200 sec have been accumulated on the RL10 engine.

Five XLR129 propellant vent valves were built and tested. Limited testing consisted of subjecting these valves to ambient temperature actuation and pressure tests of all seals. The valves were then stored for 4 months and retested to determine whether the seals had deteriorated. (Refer to Propellant Recirculation Valve Design Calculations, PWA FR-4425.) This testing has shown that all data acquired from the testing was within acceptable limits and the 4-month storage period caused no appreciable deterioration of the valves.

Because the propellant recirculation valve is very similar in design concept to both of the above valves, this ball valve concept is a practical, low risk, design which will best fulfill the requirements of the propellant recirculation valves.

H. FUEL CHECK VALVE

1. Introduction

A check valve is provided as a failsafe feature in the low pressure turbopump turbine drive return line. It is incorporated to prevent hot gas backflow from the main case in the event of a loss of pressure in the turbine drive line.

A swing check valve was chosen over ball, cone, poppet, and butterfly types because it provides the best combination of low pressure drop and low weight.

2. Design Description

The fuel check valve is a swing check valve as shown in figure I-23, spring-loaded against a floating stop, which holds the flapper open when the engine is not operating. During an emergency when the valve is subjected to a reverse pressure differential, the flapper depresses the spring-loaded stop and moves to the fully closed position. Although the cracking pressure for a fully closed flapper is low, about 0.5 psi, the flapper is held slightly open to ensure proper engine start, since the pressure available in the line at the time is low, nominally 5 psi. The maximum flow rate required to hold the valve open to the full 83-deg opening is approximately 1.5 lb/sec. The backflow pressure differential required to fully close the valve is less than 1 psid.

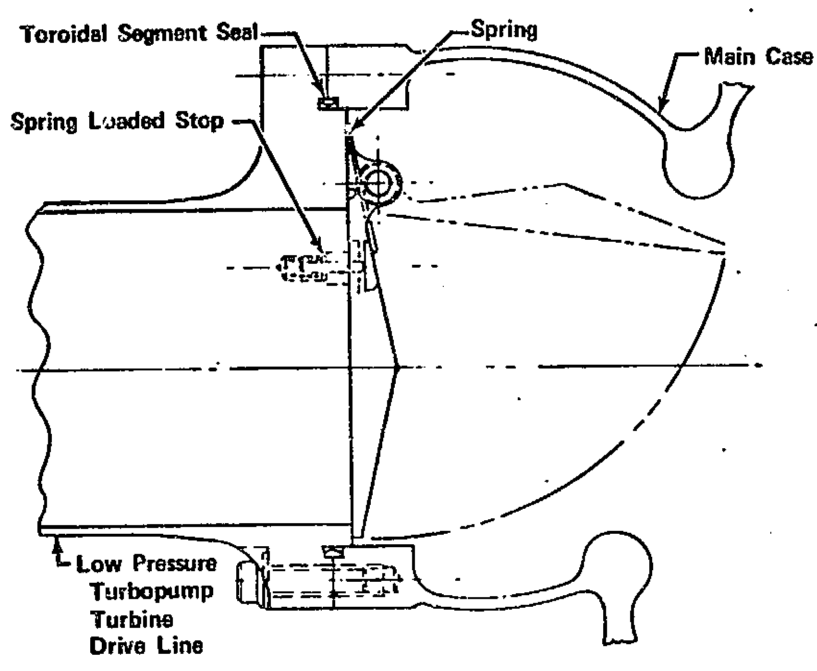

Figure I-23. Fuel Check Valve Prevents Backflow of Main Chamber Hot Gases FD 52284

The flapper is fabricated from PWA 1052 (modified A-286 stainless steel) to provide high strength and resistance to hydrogen degradation. It is designed to hold full main case pressure (approximately 3600 psi maximum) when closed. A metal-to-metal seal is used to maintain simplicity. A resilient seal is not needed because a relatively large leakage (0.1 lb/sec) is acceptable when the valve is fully closed.

The torsion spring's primary function is to initiate closure of the valve when the pressure differential across the valve drops to 5 psi. The spring is fabricated from AMS 5688 (302 stainless) which is resistant to hydrogen degradation.

The hinge pin is fabricated from AMS 5646 (347 stainless); it is only lightly loaded during normal engine operation and does not carry any of the reverse pressure load when the valve is closed. No rolling-element bearings are used; the pin is coated with PWA 586 for its anti-galling and low friction properties. The pin is locked in place by a collar of AMS 5646 and a rivet of AMS 7229 (347 stainless).

The floating stop assembly consists of a plunger fabricated from AMS 5646 (347 stainless) coated with PWA 586 dry film lubricant, a compression spring of AMS 5688, and a bolt of AMS 5735 (A-286). A tab lockwasher of AMS 5510 (321 stainless) is used to lock the bolt in place.

3. Design Requirements

The check valve is designed to meet the following requirements of CEI Specification No. CP2291 and Pratt & Whitney Aircraft imposed requirements:

1. The valve shall not impede engine start by preventing fuel flow through the low pressure turbopump turbine line, as required by paragraph 3.2.1.1.

   Compliance - At engine ignition, the valve is open a minimum of 5 deg.

2. The valve shall open and remain open during normal engine operation, including periods when the engine is experiencing acceleration loads of 3 g's maximum, as specified in paragraphs 3.4.3.5 and 3.4.4.4.

   Compliance - The valve will remain open at any fuel flow rate above 1.5 lb/sec in the turbine drive line.

3. The valve shall meet the requirements of paragraph 3.6.3 regarding failsafe design.

   Compliance - The valve closes within 5 deg of its seat when the pressure differential across the valve, in the normal direction of flow, drops to 5 psi; it closes fully when the negative pressure differential backflow mode becomes 1 psi.

4. The valve shall be designed in accordance with all applicable paragraphs of CEI Specification No. CP2291.

   Compliance - A list of all applicable specification requirements has been compiled in the DVS and the valve is designed to comply with that list.

4. Design Capability

The valve is capable of accommodating increased flow in the normal operating mode, limited only by the allowable increase in pressure drop across the valve, which would be approximately 0.5 psi for each percent increase in the flow rate at constant temperature.

Structurally, the valve is designed to withstand reverse pressure (presently about 3600 psi, including overpressure). It is capable of withstanding higher pressures within the extent of the 1.4 safety factor used in computing allowable stresses.

5. Design Substantiation

Low pressure drop was judged to be of paramount importance for this valve, with minimum weight also an important consideration. Several types of check valves were investigated. A ball check valve exhibits a high pressure drop, together with the disadvantages that the weight of the ball increases with the cube of the line size, while in a flapper type valve, the weight of the closure is more nearly proportional to the square of the line size. A poppet valve requires a close fit between the sliding element and the valve body thereby being subject to contamination problems, and the mass of the moving parts is comparatively large. A butterfly valve has poor sealing properties at low loads, high shaft loads, and unstable torque reversals throughout its travel. A split flapper valve is bulkier than the single flapper type valve, has a greater seal perimeter, and also creates a larger pressure drop due to the supporting structure in the middle of the line.

Our extensive experience with springs of all types and with bushing-type bearings, such as those used at the compressor bleed doors of the J58 turbojet, should preclude any problems with those components.

The swing check valve offers practically unrestricted passage to flow with little pressure drop. It is a feasible, low-risk, and simple design which will best fulfill the requirements for this component.

SECTION II
CONTROL VALVE ACTUATORS

A. VALVE ACTUATORS

1. Introduction

Pneumatic motor driven actuators provide stable, precise position control for the SSME control valves. Linear response and deadband characteristics necessary to meet the engine transient and steady-state performance requirements are provided by these actuators.

The actuators integrate with the engine cycle naturally. Hydrogen gas from the engine cycle is utilized as the primary energy source for the actuators, then is returned to the engine cycle. By using the gas from the engine cycle, problems associated with engine-to-vehicle interfaces, piping, and ancillary fluid conditioning have been eliminated.

The electrical power requirement for the actuator position sensors and the electromechanical servos is 32 watts maximum. Torque or force outputs are produced with relatively light weight actuators through the use of small, high speed, pneumatic gear motors driving through efficient high ratio transmissions. The gear motors operate with gas differential pressures up to 1500 psia. Flat-armature torque motors, directly positioning jet pipe servo valves or flapper type servo valves, control the gas flow to the gear motors.

Design concepts proven by testing under similar operating conditions have been incorporated into these actuators. In those cases where background testing is not available, design margins sufficient to ensure structural, performance, and life integrity of the actuators have been applied. The resulting actuators will meet requirements to control the SSME valves reliably and with a high power to weight ratio.

2. Design Description

The preburner fuel valve actuator (PFVA), the main oxidizer valve actuator (MOVA), and the transpiration coolant valve actuator (TCVA) rotate the respective valves as commanded. The preburner oxidizer valve sleeve translation is provided by the preburner oxidizer valve actuator (POVA).

Schematic representations of the POVA, the PFVA, the MOVA, and the TCVA are shown in figures II-1, II-2, II-3, and II-4, respectively. The actuators are part of a closed loop, valve position control system shown in block diagram form in figure II-5.

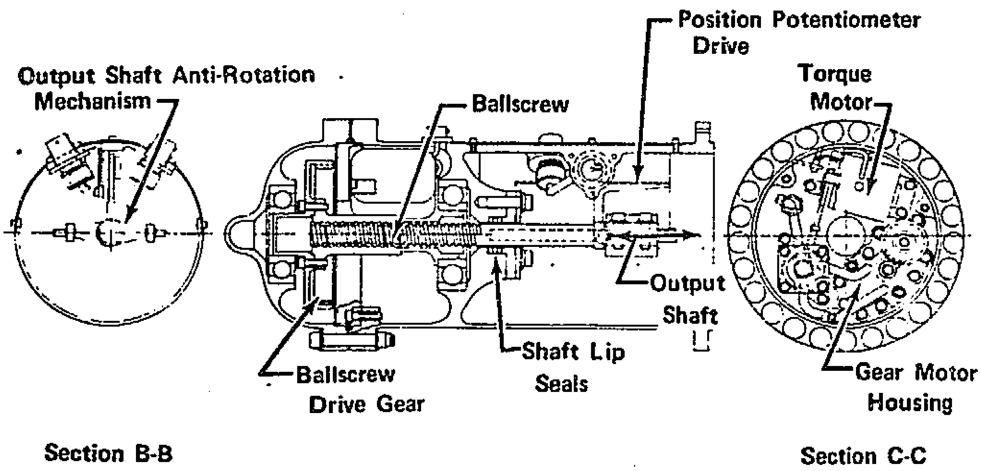

Figure II-1. Precise Control, and Excellent Dynamic Performance Provided by Preburner Oxidizer Valve Actuator FD 52300

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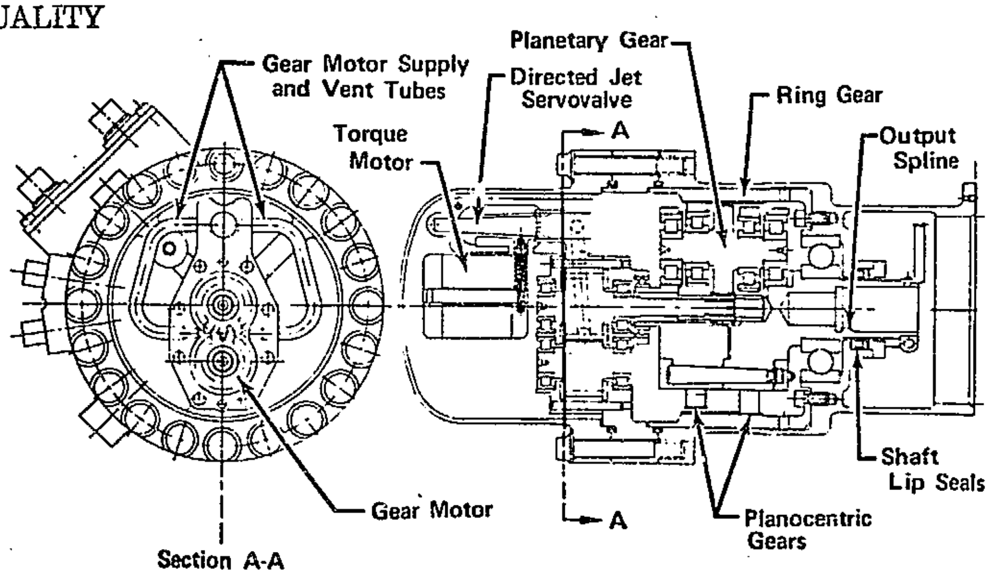


Figure I-2. Preburner Fuel Valve Actuator Provides Rotational Position Control and Peak Output Torque at 503 Thrust FD 52556

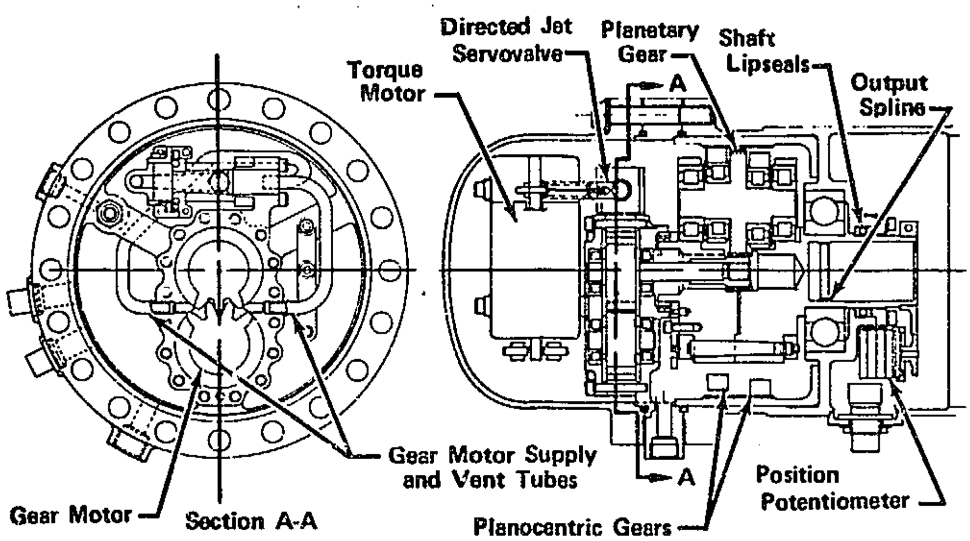


Figure II-3. High Slew Velocity and Large Stall Torque Provided by Main Oxidizer Valve Actuator FD 52555

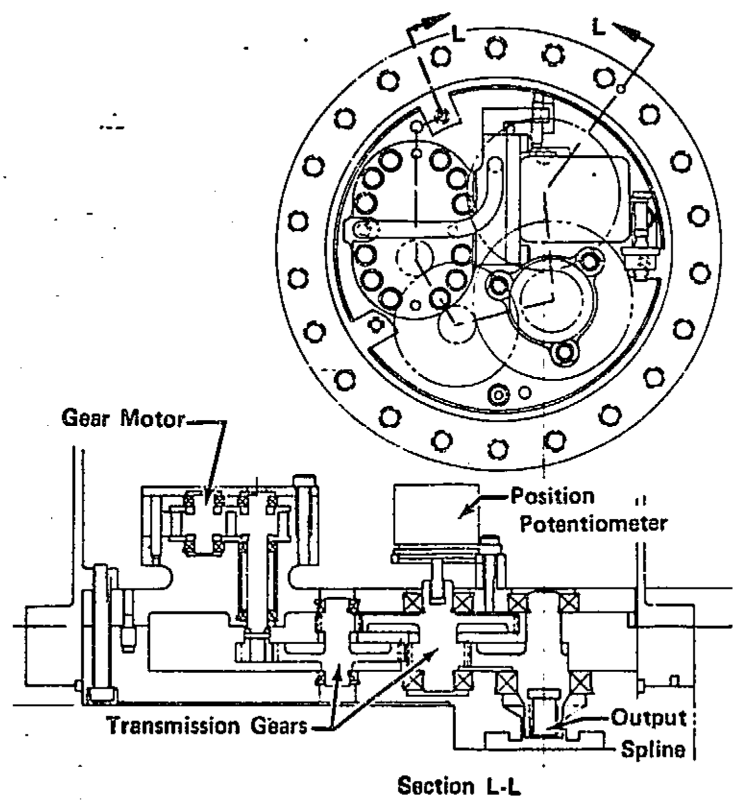


Figure II-4. Transpiration Coolant Valve Actuator Is Powered by Hydrogen Gas Flowing in Parallel With the Valve FD 52549

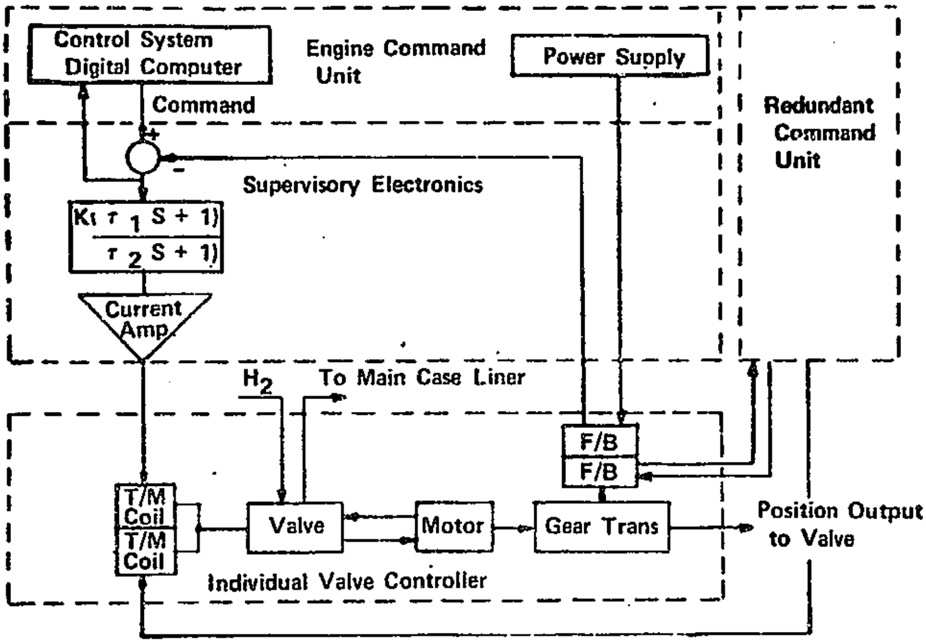


Figure II-5. Precision Position Control With Dynamic Compensation Provided for Excellent Transient Response Characteristics FD 52548

The actuators each contain the following elements:

1. A flat-armature torque motor with two independently wound coils
2. A gas servo valve
3. A two gear gas motor
4. A transmission (rotary or translational output)
5. Two rotary film type potentiometers
6. A pair of vented shaft seals (not required on the TCVA).

All of the actuators function alike. An electrical signal, that is a function of the difference between a commanded actuator position and the measured actuator position, is generated in the engine command unit (ECU). This signal is applied to the torque motor inside the actuator housing. The torque motor positions a servo valve in proportion to the applied electrical signal. Gaseous hydrogen is directed to a two-gear type gas motor by the servo valve. The resultant motor speed is a function of the displacement of the servo valve from its null position, the regulated gas supply differential pressure, the gas supply temperature and the load applied to the motor. The two gear gas motor output shaft drives a transmission to provide a rotary output drive for the PFVA, the MOVA, and the TCVA. A translating output drive is provided through a ball screw for the POVA.

A position potentiometer measures the actual position of the actuator output shaft and converts this position into an electrical signal for feedback transmission to the ECU for the difference comparison previously noted.

The electrical portion of each actuator control loop is dual-redundant. In addition to the independently wound torque motor coils and the dual potentiometers, a separate electrical connector for the leads for each torque motor coil and each potentiometer is provided for the POVA, the PFVA, and the MOVA. The leads from one torque motor coil and one potentiometer are attached to each of the two connectors for the TCVA.

The dynamic response and accuracy that can be achieved through the life of the POVA will exceed that of a control system defined by a quadratic transfer function with a natural frequency of 10 Hz and a damping ratio of 1 at an input amplitude of ±2.5% of maximum displacement and with a deadband due to friction no greater than 3 percent of maximum stroke. The MOVA, PFVA, and the TCVA performance is likewise defined except that the natural frequency of the mathematical model is 5 Hz. The electrical portion of the actuator control loop provides a flexible means of adjusting the loop gain and of varying the dynamic compensation within the loop.

Engine protection is achieved in the event that electrical signal input is lost by mechanically biasing the servo valves to drive each actuator toward safe shutdown position. The POVA and the MOVA will move to the closed valve position. The PFVA and the TCVA will move to the open valve position. The servo valve bias with no electrical power to the torque motor will limit the slew velocity of the actuators to safe values. Mechanical feedback linkage within the PFVA, the MOVA, and the TCVA provide closed loop travel limits at both the open and closed valve positions. The POVA travel is limited in the closed valve position when the valve seats. Mechanical feedback linkage within the POVA provides a closed loop travel limit at the open valve position. This same feedback linkage will limit the velocity of the POVA as it approaches the closed position.

Except during starting and shutdown, the actuators are powered by hydrogen gas obtained from the engine cycle and exhausted back into the engine cycle. During the start and shutdown modes, helium gas powers the POVA, MOVA, and PFVA. The helium also serves to purge the actuators and is exhausted into the engine cycle. Check valves in the valve actuator pressure regulator (VAPR) and in the helium control system prevent helium backflow through the VAPR and hydrogen from flowing into the helium system when the hydrogen supply pressure to the actuators exceeds the helium supply pressure.

The TCVA is powered by hydrogen gas extracted from upstream of the TCV and exhausted downstream of the TCV. The flow through the actuator is accounted for in sizing of the TCV. The TCV area is controlled to maintain a predetermined pressure differential across the transpiration liner.

To avoid the problems associated with independently sealing the servo valve and gas motor and reduce the possibility of contamination of moving parts, all actuator drive components are enclosed in a sealed housing. Gas pressure within these housings is equal to the actuator discharge pressure. The maximum pressure in the POVA, PFVA and MOVA actuator housings will be 3497 psi and the pressure in the TCVA housing will be 3810 psi at EPL.

Toroidal segment static seals conforming to Pratt & Whitney Aircraft Drawing L-220467, located in PWA FR-4377, and flange design procedures described in Plumbing Design Criteria, PWA FR-4455 are being used. Laminated Kapton and Teflon shaft seals are being used.

Flat armature torque motors power the servo valves that control the gas power to the gear motors. A schematic representation of a torque motor is shown in figure II-6. These electromechanical force transducers provide relatively large forces and small displacements. The armatures are suspended by torsion springs to eliminate pivot deadband.

Flat Armature Torque Motor Schematic

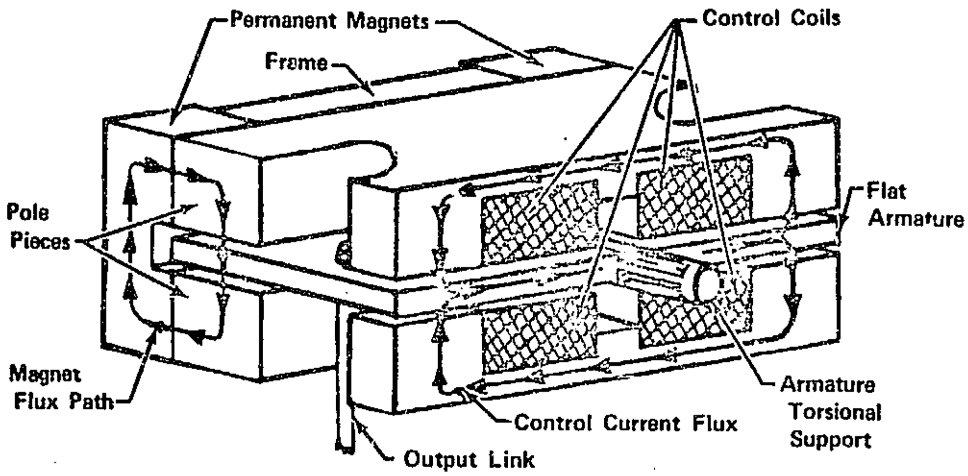


Figure II-6. Torque Motor Uses Redundant Coils FD 51547

During normal operation, with both torque motor coils energized by their respective amplifers, rated output force is produced when 50% rated current is supplied to each coil. Stops are provided to limit the armature deflection and maintain the displacement versus the current approximately linear. This stop position corresponds to the servo valve saturated area position.

The torque motors have been sized to provide the required displacement under maximum loads imposed by the servo valve at rated current. The loads imposed by the servo valve are due to the servo valve pivot spring, Bernoulli or pressure forces, and stiction forces. The torque motor armature also incorporates counter balances to reduce response to externally imimposed mechanical motions.

In the event that power is removed or lost from one torque motor coil, it will have no effect on the remaining coil operation. If a short occurs in one coil, the force produced by the torque motor would be reduced for the same current input. A back-emf in the shorted coil section will be produced if oscillatory input signals are applied, the effect being to increase the damping coefficient of the torque motor and reduce the transient response band pass. When a torque motor with one coil shorted was tested at an input frequency of 40 Hz, the output attenuation increased from 1.6 to 14.0 db, and the phase lag increased from 28 deg to 65 deg. This effect was reduced by using fewer coils (higher current) in the torque motor.

A directed jet servo valve is used to control the gas power to the gas motor in the PFVA, the POVA, and the MOVA as shown in figure II-7. The directed jet servo valve concept was selected, rather than a more conventional spool valve or flapper type servo valve, for the following reasons:

1. The directed jet servo valve has a low null quiescent flow, which is desirable during start and shutdown.

2. The directed jet servo valve has proven to be much more tolerant to contaminants than a spool type servo valve.

3. High gas flow rates are possible without requirement for a two-stage pneumatic servo valve in which stability is difficult to attain because of the low damping coefficient available.

A more detailed discussion of the trade studies that led to the selection of the directed jet servo valve are presented in Valves and Interconnects Trade Studies, PWA FR-4442.

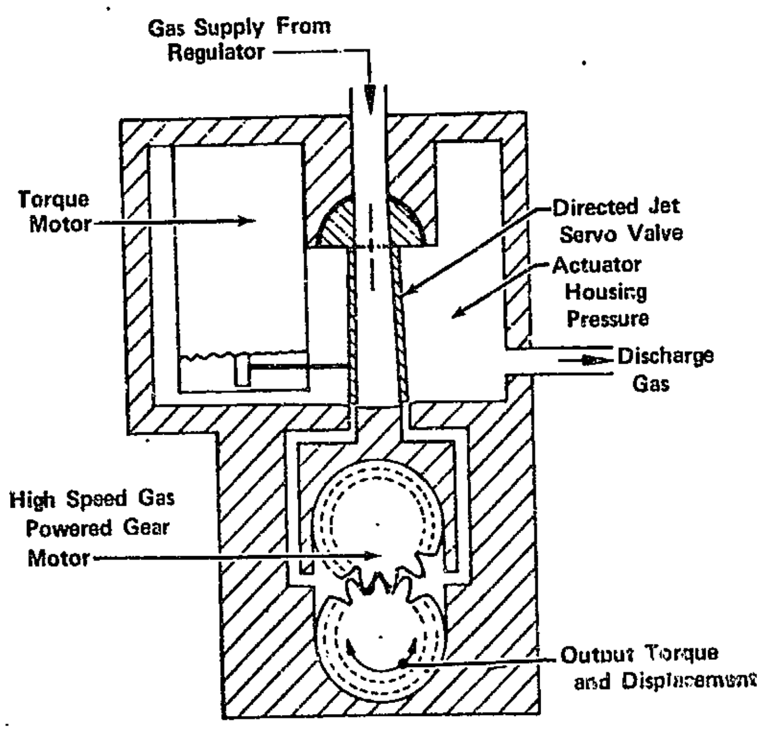


Figure II-7. Pneumatic Motor Actuator Insensitive to Contamination

FD 52546

The directed jet servo valve design is an extension of the jet pipe concept that was originally developed for hydraulic servo valves. It consists of a fixed supply nozzle, a movable jet pipe, and a gas flow receiver. From the inlet section to the jet pipe exit, an expanding cross-sectional area is provided. Wall thickness of the jet pipe is dictated by stress analysis utilizing the maximum operating differential pressure and the CEI specification safety factor. The jet pipe flexural pivot used is similar to a flexural pivot used in Bendix temperature sensors used on the P&WA J-58, F100-PW-100, and F401-PW-400 engines. The spring rate of the pivot is approximately 50 in.-lb per radian. The two receiver ports are formed by concentric cylinders to match the cylindrical geometry of the jet pipe. At null, the two receiver ports are closed by the jet pipe. When the jet pipe is deflected by the torque motor, one receiver port is opened to the gas supply pressure and the other port is opened to the exhaust pressure.

Gap clearances between moving and fixed parts have been set at 0.0010 ± 0.0005 in. to allow for manufacturing tolerances, thermal differential expansion between the valve and body, and deflection due to loads. The motor performance calculations were based on maximum gap clearances.

Mathematical models of the directed jet servo valve, gear type gas motor and flat armature torque motors were developed. A computer program was written for the solution of the equations defining the model. Gas flow reaction forces on the valve at various operating conditions were obtained. These force values were used in the sizing of the torque motor. No test data is available to date to corroborate the force values used or to indicate what stiction forces may occur due to contaminants in the hydrogen and helium. The servo valve calculations for the particular valve actuators may be referred to in the following calculation books: Preburner Oxidizer Valve Actuator Design Calculations, PWA FR-4421; Main Oxidizer Valve Actuator Design Calculations, PWA FR-4424; and Preburner Fuel Valve Actuator Design Calculations, PWA FR-4426.

In the event the reaction forces on the servo valve exceed those expected, other approaches may be taken to drive the directed jet servo valve. One concept utilizes a stepper motor driving through a small ball screw to position the servo valve. This mechanism can provide 10 pounds of force to the servo valve at a response of 10 Hz. It was not selected because there are more operating elements and the electrical power requirements are higher than for a torque motor.

Another concept utilizes a small gear motor operating through a ball screw and levers to drive the directed jet servo valve. The small gear motor is controlled by a torque motor with a flapper-type servo valve. This device will produce a very large force margin for the directed jet servo valve. It is complex and would be more expensive than the torque motor approach; therefore, it was not selected. A discussion of these approaches is provided in Valves and Interconnects Trade Studies, PWA FR-4442.

A flapper-type servo valve is used to control the gas flow to the TCVA. This control method is shown schematically in figure II-8. The flapper-type servo was selected for the TCVA for the following reasons:

1. Efficient use of the hydrogen gas used to power the actuator is not an important criteria. The pressure drop across the TCV flows through the TCVA in parallel to the TCV.

2. The flapper-type servo valve is simple and relatively inexpensive compared to either a spool or directed jet servo valve.

3. Since no close clearance parts are displacing relative to each other, hangups due to contaminant wedging in the clearance spaces are precluded.

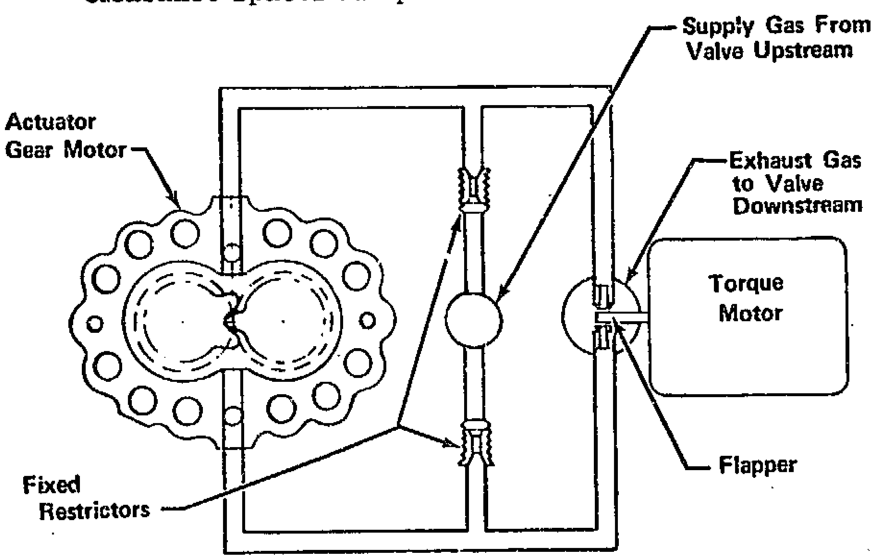


Figure II-8. A Flapper Type Servo Valve is Used Control Gas Flow to TCVA FD 52545

All actuators have gas powered gear motors that utilize the same basic design approach. The motors are high-speed, two-gear types. Roller or ball bearings are used to support the gears, and carbon end plates are used for sealing as well as for the limited thrust loads that may exist. The motor displacement is designed to meet the specific power requirements for each actuator.

Torque or force versus speed characteristics for the actuators show that adequate performance can be attained. The curves are shown in the following calculation books: Preburner Oxidizer Valve Actuator Design Calculations, PWA FR-4421; Main Oxidizer Valve Actuator Design Calculations, PWA FR-4424; Preburner Fuel Valve Actuator Design Calculations, PWA FR-4426; and Transpiration Coolant Valve Actuator Design Calculations, PWA FR-4427.

Because both gears share the torque load, the contact stresses on spur gear motors is less than 105,000 psi. The number of teeth has been set at

12 minimum to ensure good rolling contact. The gears, motor housing and end plates will be machined from 17-4 PH condition 1025 steel.

The bearings will be corrosion resistant 440C steel. The contact stresses at the maximum motor stall torque is less than 350,000 psi. Bearing separator material has not been selected because conclusive test data in the hydrogen gas environment is not available. Bendix will test several candidate materials, among these are Vespel and Salox. Vespel has been used by Bendix successfully in Nerva turbine power control actuators, which operated with hydrogen gas.

Motor end plate clearance will not exceed 0.001 in. total. The radial clearance between the gear OD and the housing ID will not exceed 0.001 in. P5N carbon end plates will be used to prevent metal-to-metal contact between the gears and the end plate.

Hi-T-Lube or an equivalent molybdenum disulfide film will be used on the motor gears, the motor shaft, the bearing bores, the bearing OD's, the bearing races, and the end plate bearing bores. The bearing bores and bearing OD's will be coated with molybdenum disulfide to preclude failure of an actuator due to a frozen motor bearing. Using a coefficient of friction of 0.025, which was obtained from data on page 276 of "Advanced Bearing Technology", NASA SP-38 for friction of 440C on 440C with a phenolic-epoxy bonded molybdenum disulfide coating, the added friction as a percent of available stall torque for one frozen motor bearing sliding between the bearing OD and the end plate bore would be 5.3%, 3.0%, 13.5%, 8.5% for the PFVA, the POVA, the MOVA, and the TCVA, respectively. These torques will not preclude the actuators from functioning even though the response and accuracy will be degraded.

The transmissions used in the PFVA and the MOVA were selected after several types had been evaluated. (Refer to the Valves and Interconnects Trade Studies, PWA FR-4442.) The transmission that provided the best gain-to-weight ratio was the planocentric design shown in figure II-9. This transmission has a ratio of 129:1 and a normal efficiency of 85 percent. Over the life of the transmission, the efficiency will not drop below 60 percent.

In operation, the high speed gear motor drives the input shaft directly. The input shaft drives three planetary gears that cause oscillating planocentric pinions to precess around the internal housing gear. The output shaft is a part of the pinion support, and rotates as the pinions precess.

Several variations on the use of the spur gear and ball screw transmission for the POVA were evaluated. The design shown in figure II-1 represents the best approach studied from a consideration of shaft seal design, maintainability, and weight.

The all spur gear transmission for the TCVA was selected because of the ease with which the gears could be packaged relative to the gear motor, potentiometer, and valve shaft. The torque requirement of this actuator is small (103 lb in. maximum); therefore, the loads on the gears are small. Small face width spur gears can be used without producing excessive gear stresses. This transmission is shown in figure II-4.

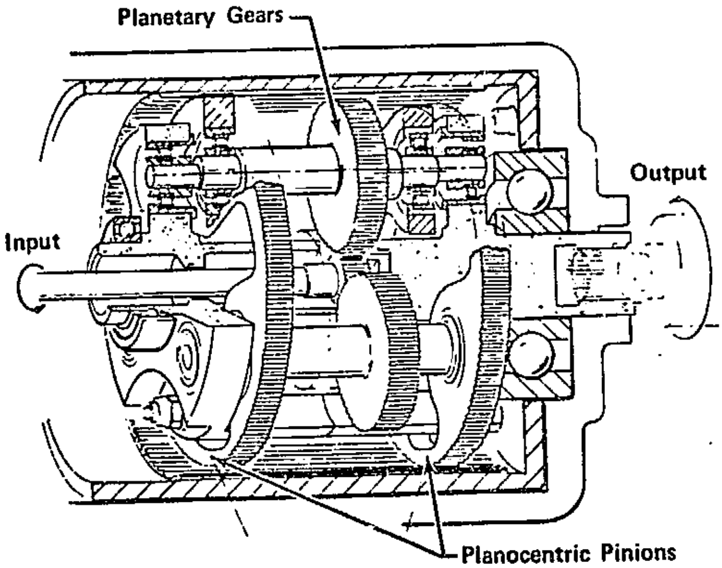


Figure II-9. Planocentric Reducer Provides Large Mechanical Advantage in Small Package FD 52544

Hi-T-Lube or an equivalent molybdenum disulfide film will be used to lubricate the gears, bearings, and ball screw. Lubrication will also be provided between the bearing outer races and the housings and between the bearing inner races and the shafts to ensure operation in the unlikely event that a bearing freezes.

The gears will be construced of 17-4 PH condition 1025 steel. The bearings will be 440C steel heat treated to Rc60. A bearing separator material will be selected based on the previously referenced tests to be performed by Bendix. A limiting value of static bearing contact stress of 350,000 psi will be used.

Dual-rotary, conductive plastic precision potentiometers mounted on a common shaft provide electrical signals to the engine control unit that are proportional to the actuator position. The selection of potentiometers was made rather than differential transformers because of the following reasons:

1. Comparable accuracy can be achieved.
2. The associated electronic circuit requirements for a potentiometer are significantly less than those required for differential transformers.
3. The number of interconnecting wires between an actuator and the engine control unit for a potentiometer is 3 for each element compared to 6 for a differential transformer. The selection of potentiometers results in less weight for the electrical connectors and harness.
4. Adequate life, proven through test experience on gas turbine engine controls, can be achieved with potentiometers.

A valve actuator pressure regulator (VAPR) is used to limit the pressure differential across the actuators. The VAPR shown in figure II-10 incorporates a combination throttling and bypass valve that maintains the total equivalent restriction of the regulator and the actuator constant as shown in figure II-11. The regulator valve, in conjunction with the valve sleeve, forms two metering ports. These ports are arranged so that the area of one port increases as the other decreases and vice versa as the valve is translated. As a result of this arrangement, the effective area of the two restrictors in series is approximately constant. This is done to maintain the coolant flow to the main case liner to the specified values.

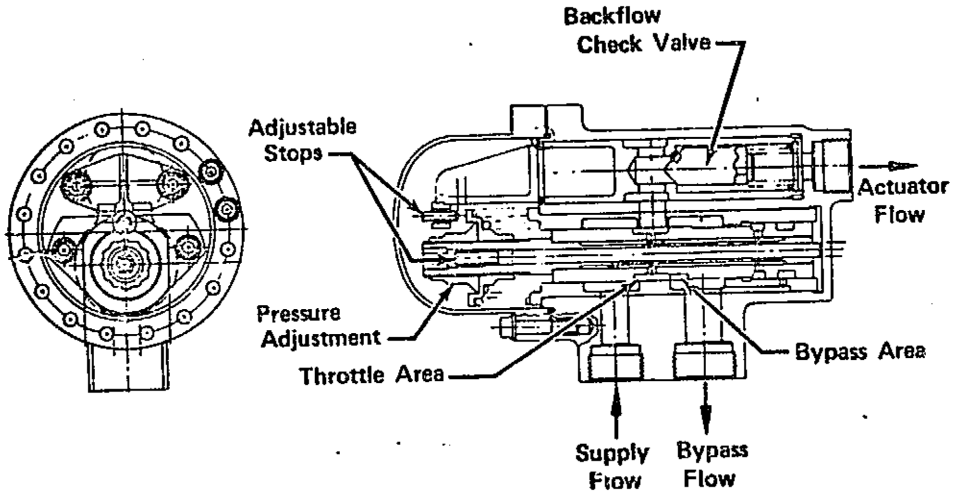


Figure II-10. Actuator Growth Potential and Design Flexibility Provided by Valve Actuator Pressure Regulator FD 52558

The dual regulator reference springs and the check valve spring will be made of A-286 steel with a maximum compression stress of 50,000 psi. During shutdown, the stress on the springs will be approximately 25,000 psi. If a regulator spring should break, the VAPR will continue to function and the POVA and MOVA will function normally. The PFVA response at the 50-percent thrust level will be degraded, but the actuator will function.

The regulator slide valve, sleeve, and helium check valve, will be made of 440C steel at RC 55-62 hardness. Diametral clearances at each end of the regulator valve and the sleeve will be held to 0.0004-0.0008 in. The diametral clearance of the center land will be held to 0.0028-0.0032 in. The OD of the regulator and check valve and the ID of the respective sleeves will be coated with Hi-T-Lube or an equivalent molybdenum disulfide surface lubricant.

Combined viscous and turbulent flow damping of the regulator is accomplished by restricting flow between the large end of the valve and bypass flow pressure. The restriction is provided as a close clearance path between the valve sleeve and body.

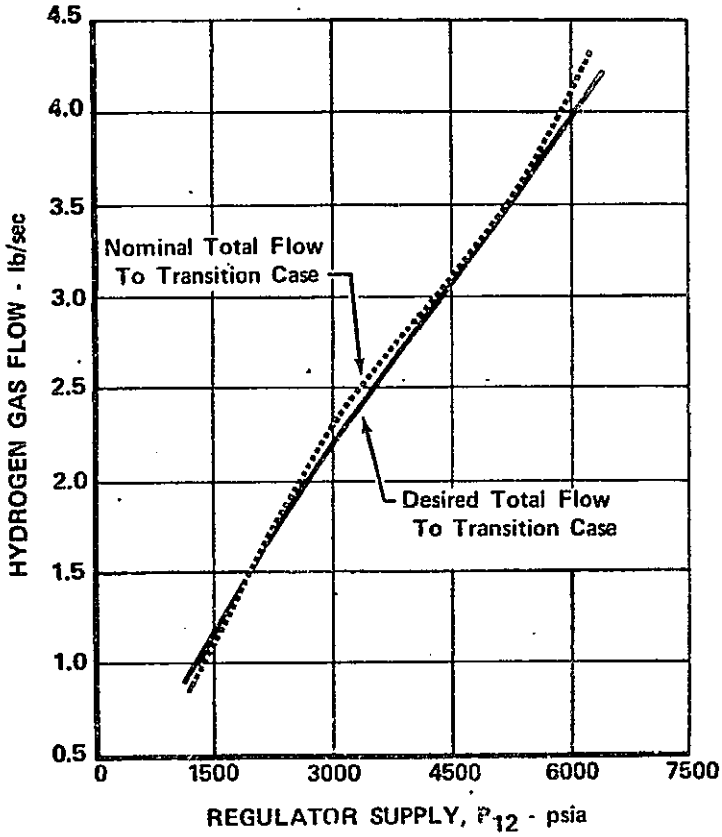


Figure II-11. Transition Case Flow Maintained by VAPR FD 52541

In the trade studies conducted on the gas power source characteristics for the actuators, the performance of each actuator was evaluated for a regulation system consisting of two restrictors in series such as that shown schematically in figure II-12. Adequate performance could be achieved for the POVA and the MOVA; however, the operation of the PFVA was not compatible with this mode of regulation. Maximum stall torque of the PFVA occurs at 50% thrust at a mixture ratio of 5.5. The series restrictor type regulator provides only about 550 psi pressure differential at this operating point. The resulting gas motor, servo valve, and torque motor sizes were too large. The pressure regulator shown in figure II-10 provides approximately 1100 psi differential pressure for the actuators at 50% thrust. As a result of the higher pressure available, the motor displacement was reduced approximately 50 percent. The VAPR selected also provides growth potential and flexibility for possible changes in the failsafe positions in the actuators.

The corroborative analysis for the VAPR is provided in Valve Actuator Pressure Regulator Design Calculations, PWA FR-4423.

3. Design Requirements

All applicable requirements of CEI Specification No. CP 2291 will be met. The pertinent specific requirements are as follows:

1. The actuators shall operate in conjunction with the engine control unit to provide the functional, performance, and

operational requirements in accordance with paragraphs 3.1 and 3.2.

Compliance - The actuators have been sized to provide the transient response, accuracy, speed, and torque necessary to power the propellant control values as required.

2. The actuators shall function as required when subjected to the thermal environment in accordance with paragraph 3.4 as modified by the engine heatshield.

Compliance - The nonoperating temperature limits of the torque motors and potentiometers are -360°F to +400°F. The operating temperature limits of the torque motors and potentiometers are -360°F to +250°F. These limits are not exceeded within the local environment. The housing bolts are designed for maximum loading (during bolt up) at +135°F.

3. Helium shall be supplied to the actuators during engine start and shutdown at temperatures between 490°R and 660°R in accordance with paragraph 3.5.2.4.

Compliance - The actuators have been sized to provide the required performance during the start and shutdown modes using helium at the specified conditions. Maximum total helium flow will be 0.181 lb/sec.

4. Electrical power to the actuator torque motors and the feedback potentiometers shall be derived from vehicle supplied electrical power at 28 ±4 vdc, ripple voltage ±1 volt peak as specified in paragraph 3.5.3.

Compliance - The current to the torque motors will be modulated by the engine control unit. Maximum voltage at +200°F will be 10 vdc. The total power requirement for the four torque motors will be 20 watts. The maximum resistance of the potentiometers will be 1000 ohms at +135°F. At this temperature, each potentiometer uses approximately 1.5 watts.

5. The actuators and the VAPR shall function within specification for a 7.5-hr service life as specified by paragraph 3.6.1.

Compliance - Bearings in the actuator are designed for a maximum contract stress of 350,000 psi when the actuator is productng 120% of the maximum output load. In the worst case the rotational life capability of the POVA motor bearings is 82 times the actuator rotational life requirement. Gears are designed for no greater than 264,000 psi contact stress when the actuator is producing 120% of the maximum output load.

6. The actuators shall be designed so that a single failure in a functional component permits the engine to be shut down safely as specified in paragraphs 3.6.3 and 3.7.11.

   Compliance - Dual redundant electrical control circuits are provided in the actuators and the engine control unit. In the event complete electrical power fails, the POVA and the MOVA will move to the closed position and the PFVA and the TCVA will move to the open position.

7. The design criteria and recommended practices for the guidance of design shall be set forth in the contractors manual in accordance with paragraph 3.7.

   Compliance - Design effort is being conducted per the SSME Structural Design Criteria, PWA FR-4449, using only high grade materials suitable for the purpose. A-286 corrosion-resistant steel will be used for the housings and other structures.

8. Leakage past internal and external static seals shall be minimized and where feasible, disposed at drains provided for the purpose in accordance with paragraph 3.7.12.1.

   Compliance - Dual seals with vents are used for actuator shaft seals. Static seals and the associated flanges comply with Plumbing Design Criteria, PWA FR-4455.

9. The actuators must comply with the weight, stroke, envelope, and interface requirements specified in the following Pratt & Whitney Aircraft Company drawings which can be found in the Drawing List, PWA FR-4377.

   - L-222175 Envelope Drawing - Preburner Oxidizer Valve Actuator
   - L-222176 Envelope Drawing - Preburner Fuel Valve Actuator
   - L-222177 Envelope Drawing - Main Oxidizer Valve Actuator
   - L-222178 Envelope Drawing - Transpiration Coolant Valve Actuator

   Compliance - The actuators have been designed to meet these requirements. The weights of the actuators will be as follows:

POVA	32 lb
PFVA	31 lb

MOVA 53 lb

TCVA 10 lb

10. The valve actuator pressure regulator must comply with the weight, envelope, and interface requirements specified.

Compliance - the VAPR meets the envelope and interface requirements. The weight of the VAPR will be 7 lb.

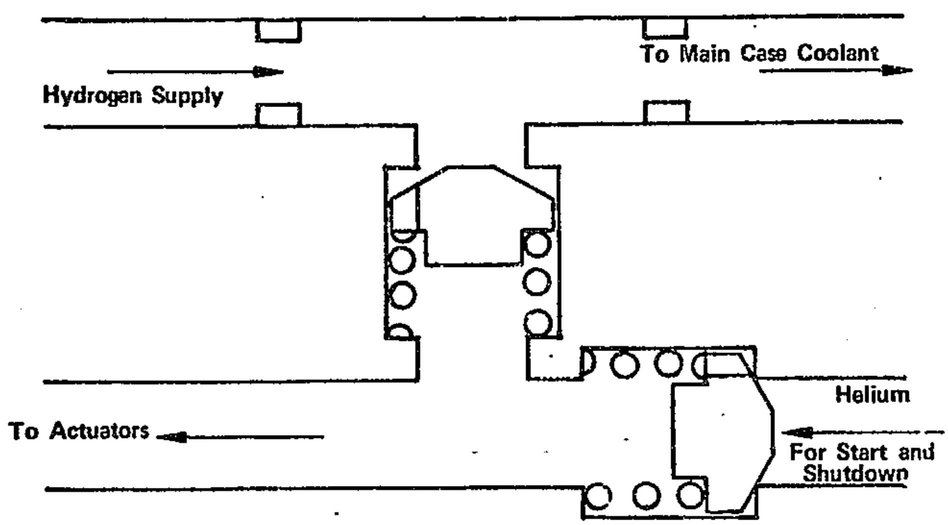


Figure II-12. Schematic Series Restrictor Type Pressure Regulator FD 52550

4. Design Substantiation

The overall concept of the actuators has been proven through experience. A turbine power control valve actuator for the Nerva engine was developed by Bendix which was similar in design concept to the PFVA and the MOVA. The Nerva actuator contained a flat-armature torque motor, a spool-type servo valve, a gear-type gas motor, a nutator transmission, a potentiometer, and a shaft seal. These actuator components were contained in a housing. The actuator was powered with hydrogen gas at temperatures from -350°F to room temperature. The actuators produced approximately 850 lb in. of stall torque and could rotate at 86 rpm unloaded. A total of 760 test hr were recorded on 23 actuators built by Bendix. Bendix Nerva Program Phase I Final Report No. 2014, Phase Extension Final Report No. 2201, and Summary Report BPAD-863-4-15122R covered the developement work on the Nerva actuators.

Additional background experience was obtained from the Bendix Model TO-D2 Exhaust Nozzle Actuator that is used on the Pratt & Whitney Aircraft F100-PW-100 and F401-PW-400 gas turbine engines. This system contains a spiral two-gear pneumatically operated servo motor, gear transmission, ball screws, ball bearings, and roller thrust bearings. The system operates with dry film lubricants on the motor gears and has operated with dry film lubricants on the ball screw and gear transmissions. Over 275 hr engine operation has been accumulated on the nozzle actuators system. As of March 31, 1971, the air motors for this system had accumulated over 12,000,000 revolutions. Additional developement work on the thrust bearings for the system has accounted for over 700,000 revolutions on three sets of bearings under the simulated loads and temperatures encountered on these engines.

Although no test hours are available, additional experience on bearings, seals, gears, and the transmission was obtained on two model MV-B1 pneumatic actuators for the J2 rocket engine. These units were designed for 75 hr of operation using gaseous hydrogen for the power medium. These test data are reported in Bendix Report BPAD-864-15479 on the Electropneumatic Linear Gimbal Actuation System.

a. Torque Motor

The torque motor loads at 20%, 50%, 75%, 100%, and 109% thrust-level operating conditions were examined to determine the maximum load requirements. The data in table II-1 were obtained from the torque speed maps in the Design Calculations Actuators. It shows the minimum force margin that is available for striction forces with rated current and the condition at which this force occurs.

Table II-1. Torque Margins Are Provided To Assure Reliable Operation Of Torque Motor-Servo Valves

Actuator	Torque Motor Deflection % Max	Thrust Level %	Speed rpm or in./sec	Load lb or lb/in.	Servo Valve Required lb	Force Margin lb
POVA	27	20	3.8	498.0	2.13	12.84
FFVA	40	50	11.8	3388.0	2.76	6.96
MOVA	49	20	108.5	250.0	6.16	10.37
TCVA	84	100	143.0	106.5	1.18	1.76

b. Directed Jet Servo Valve

The directed jet servo valve concept was selected based on the test and analysis of a similar type servo valve used in a bang-bang reaction controller for the C-3 Poseidon A valve. This valve controlled the flow of a perchlorate gas at 2200°F and 740 psia. Results of the tests and analysis are covered in a report prepared by The Bendix Corporation Research Laboratories under subcontract 18-16038 from Lockheed Missiles and Space Company.

The test results indicated that the aerodynamic torque required to operate a similarly sized valve would be in the order of 6 lb in. The test also indicated that stiction due to contaminant with the extremely "dirty" perchlorate fuel combustion products was not present. On the basis of this test, it was concluded that a directed jet servo valve operating in relatively clean hydrogen and helium gas should likewise not experience stiction due to contaminant particles. Refer to Valves and Interconnects Design Trade Studies, PWA FR-4442.

The data from a detailed analysis of the forces acting on the directed jet servo valve indicated that the aerodynamic or Bernoulli forces acting on the valves was on the order of those measured in the previously referenced report. For example, the maximum torque calculated in the operating regime of the MOVA servo occurred at 20% thrust, at a servo valve displacement of 0.0105 radians, a gas motor speed of 13,950 rpm, and a motor output torque of 1.26 lb in. This servo valve torque was 2.19 lb in. Refer to Preburner Oxidizer Valve Actuator Design Calculation, PWA FR-4421, Main Oxidizer Valve Actuator Design Calculations, PWA FR-4424, and Preburner Fuel Valve Actuator Design Calculations, PWA FR-4426.

c. Design Calculation Results for the Gas Motors, Transmissions, and Spline Couplings

Life calculations were made for the motor gears and bearings, the transmission gears and bearings, and for the output shaft splines for the rotary output actuators. The life calculations were based on the maximum valve loads with the load being cycled at ±2.5% of the total actuator stroke at 1 Hz. Normal transmission efficiencies were assumed for these calculations. Under these load conditions, the minimum ratio of bearing life capability in terms of B-10 life, to life required for 7.5 hr under the same cyclic conditions is 82. The minimum ratio of life capability for gears compared to life required for 7.5 hr is 11.8. Adequate life margins are available to ensure that the actuator bearings and gears will meet the required life.

The life margins for the motor bearings for each actuator are shown in table II-2. The transmission bearings and gears for each actuator are shown schematically in figure II-13. Corroborative analyses for the data presented is contained in the design calculations referenced above.

To ensure that the bearings and gears would meet all stress requirements, both bending and contact stress were evaluated. The motor gears and bearings and the input gears to the transmissions were evaluated for the maximum motor stall torque condition. The transmissions were evaluated at the loads that would be produced if the actuators were delivering 120% of the maximum valve loads. The criteria used to determine the transmission bearing and gear loads is realistic; however, the criteria used to determine the motor loads is considered to be too conservative. The highest contact stresses occur in the POVA. These occur when 120% of the POVA breakaway force is required with the fully degraded efficiency or when the actuator must be driven at a much higher frequency and amplitude than is required. Even under these unrealistic conditions, the gear and bearing compressive yield stress limits will not be exceeded.

Table II-2. Transmission Bearing and Gear Stresses Are Low Enough to Assure Reliable Operation

Actuator and Key Letter		Maximum Contact Stress-psi	Maximum Load, lb		Life Rev, in.*	
			120% Load or Stall	Operating	at 7.5 hr	at Max Load
PFVA	A	214,000	90	43.8	280,000	6.6×10^6
	B	214,000	90	43.8	31,200	6.6×10^6
	C	60,500	96.3	80.3	31,200	10^8
	D	60,500	96.3	80.3	31,200	10^8
	E	333,000	562	468	31,200	6.85×10^6
	F	329,000	1243	1036	31,200	20.9×10^6
	G	312,000	1243	1036	31,200	27.5×10^6
	H	318,000	3739	3696	31,200	5.15×10^6
MOVA	A	225,000	133.3	65.5	280,000	3.3×10^6
	B	225,000	133.3	65.5	31,200	3.3×10^6
	C	57,500	144.0	120	31,200	10^8
	D	57,500	144	120	31,200	10^8
	E	331,000	785	655	31,200	3.15×10^6
	F	323,000	1875	1560	31,200	21.2×10^6
	G	333,000	1875	1560	31,200	17.1×10^6
	H	338,000	7445	7385	31,200	2.57×10^6
POVA	A	224,000	123.3	60	302,000	4.4×10^6
	B	224,000	123.3	60	151,000	4.4×10^6
	C	264,000	246.6	120.0	151,000	0.54×10^6
	D	264,000	246.6	120.0	23,200	0.54×10^6
	E	364,000	3840	3200	23,200	0.283×10^6
	F	364,000	3840	3200	23,200	0.283×10^6
	G	347,000	4856	4.72	23,200	0.64×10^{12}
	H	325,000	5205	3208	23,200	16.8×10^6
	I	311,500	389	189	151,000	415×10^6
	J	242,100	235	114	151,000	1290×10^6

*Note: Bearing life based on B-10 life. Gear life based on contact stress.

Table II-2. Transmission Bearing and Gear Stresses Are Low Enough to Assure Reliable Operation (Continued)

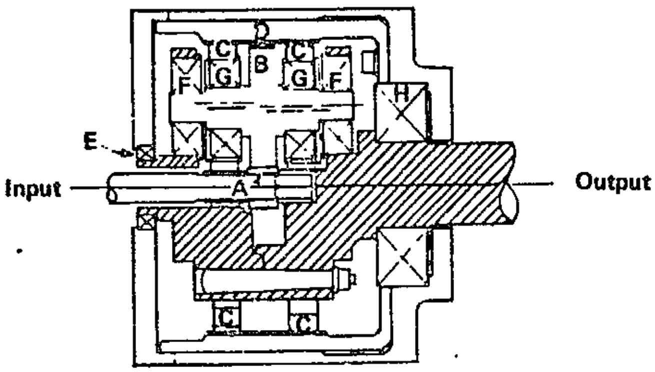
Actuator and Key Letter		Maximum Contact Stress-psi	Maximum Load, lb		Life Rev, in.*	
			120% Load or Stall	Operating	at 7.5 hr	at Max Load
TCVA	A	154,000	19.35	12.55	35,000	50×10^6
	B	154,000	19.35	12.55	10,500	50×10^6
	C	184,000	64.50	41.90	10,500	0.8×10^6
	D	184,000	64.50	41.90	2,460	8×10^6
	E	230,000	172.00	112.00	2,460	0.5×10^6
	F	230,000	172.00	112.00	967	0.5×10^6
	G	278,000	28.6	18.6	35,000	770×10^6
	H	284,000	59	38.4	10,500	120×10^6
	I	315,000	28	18.2	10,500	120×10^6
	J	202,000	74.8	48.5	2,460	2475×10^6
	K	233,000	113	73.5	2,460	712×10^6
	L	193,000	64	41.6	967	3940×10^6
	M	236,000	119	77.5	967	607×10^6

*Note: Bearing life based on B-10 life. Gear life based on contact stress.

A limiting value of static bearing contact stress of 350,000 psi was used based on the bearing manufacturer's recommendation for the bearings used. This value seems reasonable based upon tests performed on the previously referenced TO-D2 actuator. In this actuator, a roller thrust bearing was subjected to contact stresses in excess of this magnitude without any detrimental local yielding. The Navigation and Controls Division of The Bendix Corporation built an actuator for the Control-Moment-Gyroscope on the Apollo Test Module. Bearings in this actuator were designed for Hertz stresses up to 653,000 psi. In a design review report of this actuator performed by Battelle Memorial Institute, Columbus Laboratories, it was reported that the bearings could be used at this Hertz stress level; however, they recommended that in future applications the Hertz stress levels be kept below 500,000 psi. The ball screws in the TO-D2 actuator were designed for contact stresses in excess of 500,000 psi. None of the SSME actuator bearings are subjected to contact stresses in excess of 347,000 psi.

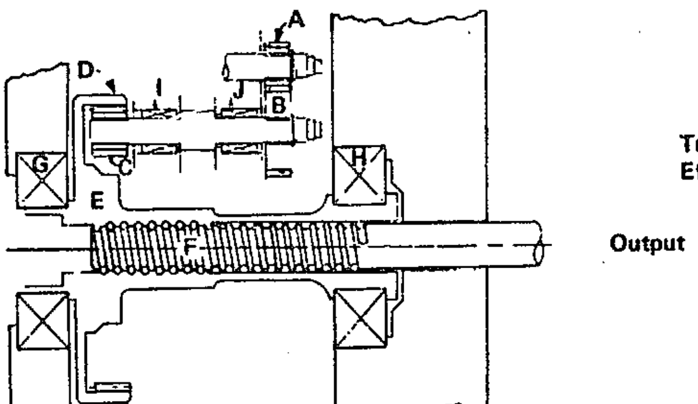
The maximum gear contact stress was calculated to be 264,000 psi. With this stress imposed on 17-4 PH Condition 1025, some local yielding may occur; however, it should not impair the performance of the gear.

The maximum bending stress that was calculated for the stalled motor condition was 80,000 psi; consequently, no detrimental yielding of the gears should occur.



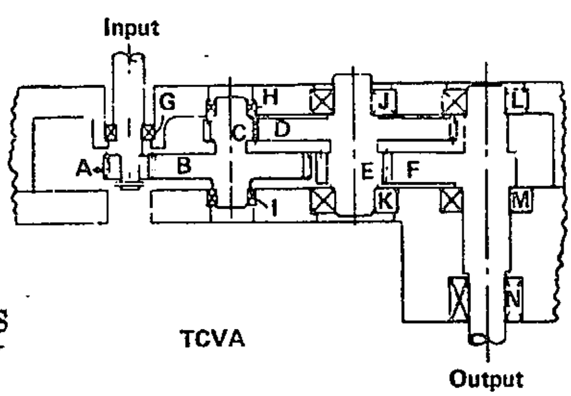
MOVA and PFVA

Transmission Ratio - 129:1
Efficiency
New - 0.85
End of Life - 0.60



POVA

Transmission Ratio - 52 rev/in.
Efficiency
New - 0.735
End of Life - 0.469



TCVA

Transmission Ratio - 36.2
Efficiency
New - 0.927
End of Lıfe 0.857

Figure II-13. Actuator Transmissions Provide High Efficiency Mechanical Advantage FD 52540

The maximum ball screw contact stress for the POVA was calculated to be 364,000 psi. For the 440C material at a hardness of RC58, the limiting contact stress is 500,000 psi. The ratio of life capability in terms of revolutions of the ball screw nut at the maximum operating load, compared to the revolutions required to produce 7.5 hr life unde the cyclic conditions cited for the gear and bearing life is 12. An adequate contact stress margin and life margin as shown in table II-3 at the maximum load exits to assure that excellent performance can be achieved with the POVA ball screw.

Table II-3. Contact Stresses and Life Capability of Gear Motor Gears and Bearings Assure That Life Requirement Can Be Met

Actuator	Maximum Contact Stress, psi Gear	Bearing	Maximum Bearing Loads, lb Stall	Operating	Bearing Life - Rev At 7.5 hr	B10 At Max Load
PFVA	102,000	298,000	645	313	93,500	242 x 10^{6}
POVA	92,000	340,000	465	226	302,000	132 x 10^{6}
MOVA	104,500	321,000	720	352	93,500	162 x 10^{6}
TCVA	67,000	333,000	49.3	32	35,000	108 x 10^{6}

The shear stresses in the spline couplings is shown in table II-4 for 1.2 times a maximum shall torque for the PFVA, the MOVA, and the TCVA. The stress limits for the materials used are also shown. These splines will be lubricated with Hi-T-Lube or an equivalent molybdenum disulfide lubricant.

Table II-4. Spline Couplings Provide a Safe Means of Transmitting Actuator Output Power

Actuator	Maximum Torque 1.2 x Stall lb in.	Shear Stress Actual psi	Limit psi
PFVA	4000	46,000	72,000
MOVA	8100	32,250	72,000
TCVA	107	3,580	86,500

d. High Pressure Sealed Electrical Connector

Four electrical connector vendors stated that they could provide weld mount, shell size 8, 3 pin, bayonet coupling electrical connectors that could seal hydrogen gas at 3500 psi operational pressure and with leakage rates less than 1 x 10^{6} scc/sec of helium. One vendor, Gulton Industries Inc., Newport Beach, California, provided test data on a weld mount, shell size 8, 5 pin, bayonet coupling electrical connector. This connector was leak tested as part of an acceptance test at 5460 psi with helium for 2 minutes. The leakage rate was less than 1 x 10^{-8} scc/sec of helium. On the basis of this test, we are confident that the high pressure electrical connectors will provide the required life expectancy.

5. Design Capability

Power margins have been built into the actuators. The actuators motors were designed to provide a torque capability that will assure that fractional errors in the output will not exceed 3% of the output stroke when all moving elements have degraded to end of life efficiencies. It is not probable that these efficiencies will ever be experienced because of the design margins used in the bearings and gears. It is even less probable that all components will simultaneously degrade; therefore, greater power outputs may be provided with some possible reduction in the life margin.

The transmissions were designed to transmit 120% of the specified maximum loads. It was previously shown that generous life margins exist at these loads. Larger loads may therefore be specified without increasing the transmission sizes.

Without increasing the transmission ratios, the MOVA and POVA maximum slew rates could be increased 20%. The PFVA maximum slew rate could be doubled. At conditions other than the 100% thrust level valve position and load, the TCVA slew rates can be increased. The limiting unloaded speed is approximately 2.5 times the specified maximum slew rate.

The actuators are capable of operating with changed cycle conditions provided that at some condition the combination of source gas power, output loads, output response, output accuracy, and output speeds required do not exceed the capability of the actuator. The control system is capable of compensating for the loop gain changes that will occur at the various engine thrust levels.

Changes in the failsafe valve positions can be accommodated by adjusting the servo valve bias position relative to the null position when no power is applied to the torque motor. Mechanical feedback linkage provides travel limits to each extreme actuator stroke position.

SECTION III
HELIUM SYSTEM

A. INTRODUCTION

A compact, easily maintained modular arrangement has been designed which ensures reliable helium management for the SSME. Requirements for this system are identified in P&WA purchase specifications T-10 and T-11. A contract was awarded to the Parker-Hannifin Corporation Systems Division after a competitive proposal effort, based upon the most reliable design and the most directly applicable past experience. The Parker-Hannifin Corporation Systems Division proposed an advanced design concept proved to be practical and effective in qualified hardware used in current NASA and Air Force Space programs. The application of this related experience permits these design concepts to be used with high confidence. The actual valve elements are similar in detail to hardware used in the Apollo, Lunar Excursion Module, and Saturn programs, together with Air Force space vehicles.

Modular construction, component accessibility, hardware commonality, and integral position indication for fault detection permit rapid maintenance with a minimum of logistics. Although preventive maintenance of the helium module is not required during the life of the SSME, the design includes the flexibility needed for obsolescence protection by allowing new or modified components to be easily incorporated to accommodate different or expanded performance requirements, as well as permitting component replacement in the event of failure or inadvertent damage.

B. DESIGN DESCRIPTION

Twenty-four components are packaged together to form an integrated helium module having minimum weight and envelope. Figure III-1 shows the module that provides helium control for 15 SSME functions in an envelope that weighs 55 lb. Qualified flexures guide the moving parts to preclude valve jamming due to contamination. These are discussed in Valves and Interconnects Trade Studies, PWA FR-4442. Mechanically retained Teflon seats offer positive, low leakage helium shutoff with maximum environmental and contamination resistance. Ultra clean assembly and a Zeiss interferometer inspection provide initial quality while the ability of the Teflon to "flow" around contaminant particles ensures continuing low leakage operation. Multi-ply diaphragms replace dynamic sliding seals to increase reliability and valve life. These diaphragms, designed for 20,000 cycles, eliminate the problems of high leakage and high friction which are characteristic of the sliding seals used on ordinary pilot-operated solenoids.

In addition to providing positive no-sliding-fit armature guidance for both solenoids and check valves, the flexure guides generate the spring force needed to return solenoids to their normal position when electrically deenergized. The solenoid valve flexure guide as a Belleville-type disk spring refined by computer design to match its spring characteristics with the solenoid force characteristics. The Belleville spring (short-stroke, high-force) is ideally suited for use with the bipole solenoid which is optimized for short stroke and high force. It also provides low mass, and variable spring rate, and edge damping in the flexure guide, which result in superior vibration tolerance. The flexure can be easily analyzed and designed by computer techniques to ensure operating stresses for long life and proper force characteristics.

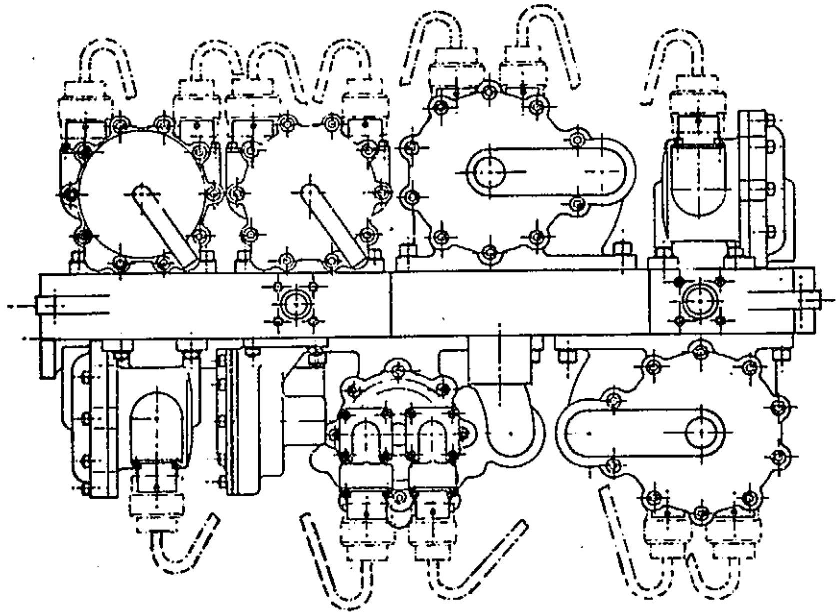


Figure III-1. Modular System Simplifies Maintenance FD 52418

Mechanically retained Teflon seats offer good contamination resistance and excellent sealing characteristics while providing the thermal environment tolerances needed for this application. The combination of good contamination resistance and sealing ability cannot be matched by the other common seat concepts and meet the 100 scch leakage limit required in this application. Both the poppet and seat are flat to facilitate precision manufacture and inspection, and the solenoid valve poppet is allowed to overtravel and align itself on the Teflon seat for uniform sealing stress distribution. Mechanical Teflon retention is positive and completely inspectable for reliable fabrication. Narrow metal bumpers, recessed approximately 0.001 to 0.002 in. below the inside and outside of the Teflon insert, provide the structural backup needed for high pressure operation, and precisely limit the amount of Teflon compression to ensure proper sealing stress and negligible cold-flow during the life of the SSME.

Bipole dc solenoids are used. Six direct and five pilot operated valves are used in the helium module. The pilot solenoid operated main valves use multi-ply diaphragm actuated, flexure guided, main shutoff valves. All solenoids employ first-failure-operational electrical redundancy, and are designed to assume second-electrical-failure-safe positions by conservatively designed mechanical springs. Holding coils are used to reduce continuous operation electrical power, and weight optimized bifilar windings suppress deenergization back-emf.

Bipole dc solenoids were selected after a trade study of possible state-of-the-art solenoid actuator concepts (refer to Valves and Interconnects Trade Studies, PWA FR-4442). As noted in this study, the bipole solenoid design approach is inherently suited for flexure guide usage, and the increased reliability advantages are gained without any weight or power penalties. This design approach minimizes the possibility of valve jamming which is the prevalent failure mode of traditional side-loaded plunger-type solenoids. This type of failure is especially common in long-life, reusable gas systems where no lubricity is provided by the operating fluid, and the introduction of contamination during maintenance is a realistic possibility. This possibility must be considered in application for the SSME.

As shown in figure III-2 the helium module has six direct operating solenoid valves and five pilot solenoid operated valves. In designing the module, the basic simplicity of direct operation was maintained for the maximum number of valves permitted by the total helium module weight allowance of 55 lb.

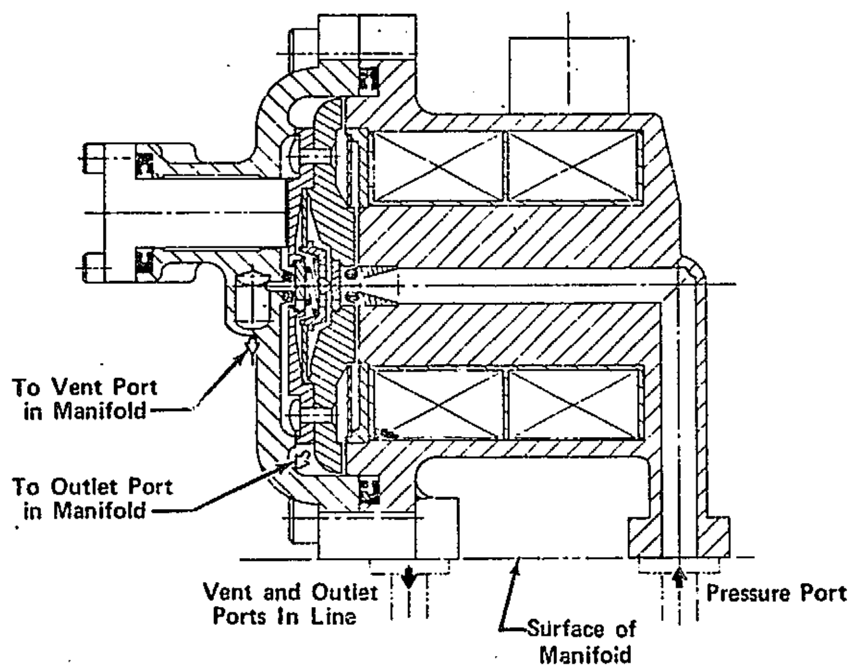


Figure III-2. Three Way, Two-Position Normally Open Valve FD 52417

The direct operated valves shown in figure III-3 position the valve poppet directly using solenoid generated magnetic forces when electrically energized. Spring forces return the valve poppet to its normal position after solenoid deenergization.

Design commonality has been achieved where possible in the six direct operated valves. The solenoid actuators are identical in all six valves and valve elements are identical where possible. This is true for normally closed valves 1-S, 2-S, and 10-S and normally open valves 3-S and 4-S.

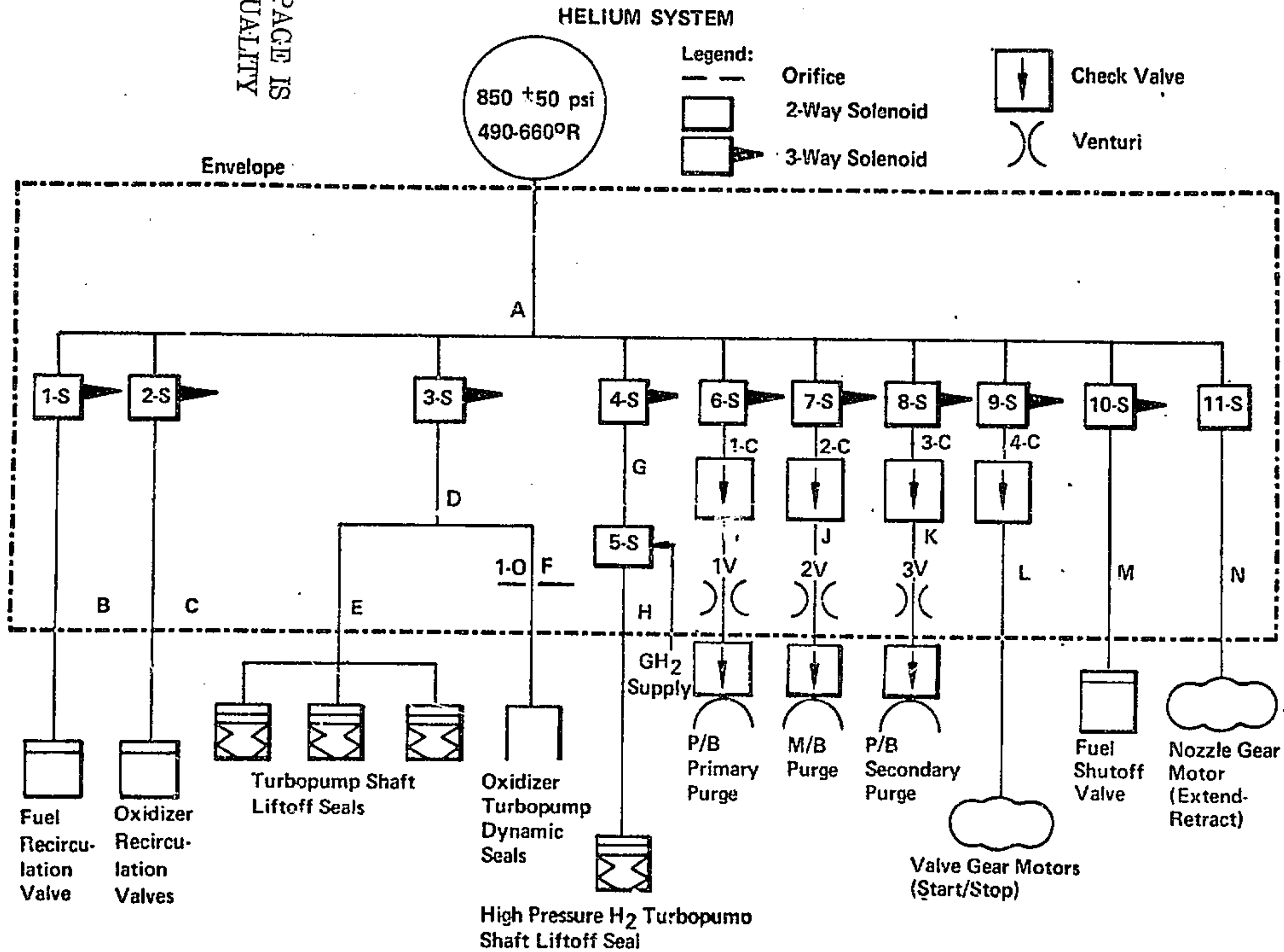


Figure III-3. Reliable Helium Management

FD 47205C

The pilot operated solenoid valve shown in figure III-4 utilizes normally closed 3-way solenoid valves to pressurize multi-ply diaphragm controlled main valve poppets. The main valve poppet actuates when the pilot valve solenoid is energized to establish a pressure differential across the diaphragm. Deenergizing the pilot valve solenoid reduces the pressure differential across the diaphragm by venting the helium, and the main valve returns to its normal position by mechanical spring forces. As determined in trade studies, computer optimization of the required actuation force and weight resulted in the selection of pilot-operated valves over direct operated valves.

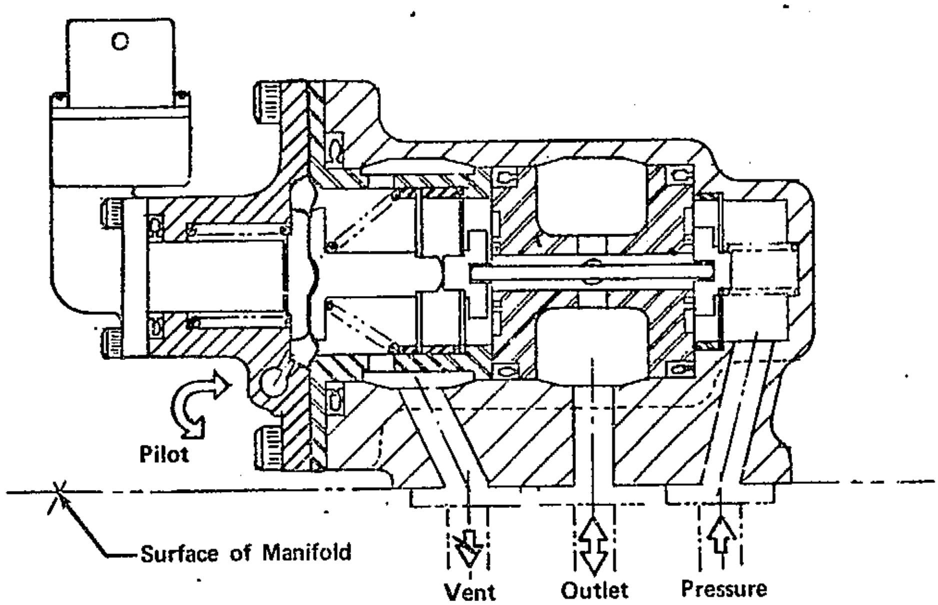


Figure III-4. Three Way, Two-Position Normally Closed Valve FD 52416

Design commonality is also achieved in the pilot-operated valves. Normally open valves 7-S, 8-S, and 9-S are identical and a normally closed 3-way solenoid valve, identical to the solenoid valve configuration used for 1-S, 2-S, and 10-S is used for each of the pilot operated valves.

The electrically parallel coil arrangement which was optimized by computer design analysis, results in a bipole solenoid valve that is at least 16% lighter than other configurations studied. Three coils, essentially trifilar wound, are utilized in each of the redundant coil assemblies used in each solenoid valve. This trifilar wound coil represents a pull-in coil, a holding coil, and a transient suppression coil. Transistor switching is utilized to energize both the holding and pull-in coils during valve operation. After approximately 100 milliseconds, power to the pull-in coil is removed and the valve is held open by the holding coil. Power consumption is reduced by a factor of five by this feature, which is possible because of the greatly reduced solenoid ampere turns required to maintain the valve in its energized position. When the holding coil is deenergized, the valve returns to its deenergized position by mechanical spring force. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

A bifilar coil is provided for each solenoid to limit flux decay rates to acceptable valves. As the magnetic flux decays during the deenergization of the holding and pull-in coils, the collapsing field generates a back-emf across the bifilar coil. This back-emf generates a current in the bifilar coil that tends to maintain the original flux, thereby reducing flux decay rate. Dynamic computer simulation was used to determine the bifilar coil wire size, and therefore coil resistance, to limit the back-emf to 42 vdc.

The dynamic computer simulation also revealed that the bifilar coil wire size could be reduced due to the surge-suppression action of the momentary current increase in the energized holding coil when the opening coil is deenergized. (Although back-emf surge suppression must be provided when the holding coil and opening coil are deenergized, more suppression is required to suppress the deenergization of the larger wire size/lower resistance opening coil.) As noted in the trade study, only a parallel coil configuration offers this reduction in the bifilar coil size by taking advantage of the inherent surge suppression gained by the energized holding coil.

Multi-ply Kapton single-convolution diaphragms were selected instead of conventional dynamic sliding seals to pressure actuate the main poppets of the pilot solenoid-operated valves. (Refer to Fluid Components Trade Studies, PWA FR-4442.) Conventional pilot-solenoid operated valves use one or more pressure actuated pistons, and at 900 psi pressure, it is necessary to use close diametral clearance to avoid piston seal extrusion. The sliding piston seal is subject to wear-out, leakage, and exhibits variable friction.

There is substantial experience that demonstrates the practicality, performance capability, and reliability of this design approach. For example, pilot-solenoid operated valves (Parker P/N's S62A1600 and S62A1630) in the hydrogen and oxygen fuel control system supplied for the Air Force Dynasoar vehicle demonstrated operation at 2300 psi and up to 250° F. Because these diaphragms easily exceeded the 10,000-cycle life test for the higher temperature and pressure Dynasoar vehicle application, and in other tests in which they have passed life tests in excess of 10^7 cycles, no difficulty is expected in meeting the 20,000-cycle life requirement of the helium module.

Passive, inductance type position indicators are used on selected solenoid valves to monitor valve position for checkout, fault detection, and fault isolation. A trade study shows this advanced all-electronic position transducer is more reliable than conventional mechanical miniature snap-action switches, as it completely avoids the fragile snap action mechanisms and the making/breaking of delicate mechanical electrical contacts. In addition, this is an off-the-shelf item, modified only slightly for this installation and requires no new moving parts in the solenoid valves. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

This type position indicator uses a variable transformer action caused by varying the proximity of the solenoid valve ferromagnetic armature with respect to the two coils of the position indicator. A 10-volt, 5000 Hz voltage excites one coil, inducing a voltage in the second coil; the magnitude of the induced voltage is determined by the transformer coupling effected by the proximity of the valve armature, hence, when the armature is close to the sensor, flux coupling is increased and the induced voltage is higher than when the armature moves away as the result of valve actuation. It has been established in the Air Force Minuteman ICBM Roll Control System solenoid valve application that, when properly designed, the flux generated in the solenoid armature has second order effects on indicator operation.

In addition to the solenoids valves, there are four check valves included in the module. These incorporate flexure-guided, flat-lapped poppet sealing against a Teflon seat. Reliability and low contamination sensitivity were overriding considerations in choosing this scheme.

Three replaceable venturis, designed to be choked when flowing helium and which are not interchangeable, limit engine purge flows. Conventional venturis, which provide an 80% pressure recovery, were chosen for these applications to allow a constant purge flow rate over a wide range of back pressures. Because of a low back pressure, flow to the high pressure oxidizer turbopump dynamic seals is controlled by a simple, replaceable, sharp-edged orifice. All of these flow control devices are "cartridge" type units offering complete and independent accessibility. This feature is important if minor engine changes create the need for adjusted flow rates or pressures.

Modular construction allows complete maintenance at the component level without disturbing hard system line connections. A forged aluminum alloy manifold using steel inserts at the plumbing interfaces and helical coil inserts at all screw threads provides a low weight, high strength, serviceable design. All solenoid valves are flange-mounted to the manifold, and all check valves, venturis, and the orifice are cartridge units inserted into the manifold. Independent component accessibility with standard tools expedites and simplifies maintenance. Noninterchangeable components use dissimilar mounting bolt patterns and different cartridge diameters to avoid improper assembly. Mounting flanges are nonsymmetrical so that components can be installed only in the proper orientation and redundant electrical receptacles on solenoid valves utilize different shell sizes.

C. DESIGN REQUIREMENTS

The helium system conforms to all applicable requirements of the CEI Specification CP2291. The following is a list of pertinent specification requirements and P&WA requirements and the manner in which the helium system complies to these requirements.

1. CEI Specification No. CP2291 Requirements

The helium system meets all the requirements of CEI Specification No. 2291 as follows:

1. Helium system functional performance requirements shall be met by paragraphs 3.1 through 3.1.3.

   Compliance - The helium system supplies all helium to the engine as it is required during the engine cycle. The maximum helium flow rate is 0.621 lb per second.

2. Failsafe operation requirements shall be in accordance with paragraph 3.6.4.

Compliance - Fail op/failsafe operation is provided by dual solenoid coils and selected solenoid failure modes. After the first solenoid coil failure, the system is fully operational. After the second solenoid coil failure in the same valve, the engine remains operational and can be safely shut down.

3. Structural criteria requirements shall be met in accordance with paragraphs 3.7.7.1 and 3.7.22.

   Compliance - All system components were designed in accordance with SSME Structural Design Criteria, PWA FR-4449.

4. Helium maintainability requirements shall be met in accordance with paragraph 3.7.7.3.

   Compliance - Modular design permits field replacement for easy maintainability.

5. Electrical power requirements shall be in accordance with paragraph 3.5.3.1.1.

   Compliance - After studying the possibility of using 110 vdc the helium system is designed to function using the vehicle supply power of 28 ±4 vdc and to operate with minimum power demand. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

6. The electromagnetic compatibility requirements shall be in accordance with paragraph 3.7.11.4.

   Compliance - All system solenoids have bifilar windings to provide surge suppression.

2. Pratt & Whitney Aircraft Requirements

   1. System solenoids are designed for continuous duty over the operating temperature range without external environment conditioning. The valve components are designed for 20,000 cycles without exceeding calibration limits. All diaphragms are qualified for 20,000 cycles and all flexures and springs will be designed for at least a 10^7 cycle life.

   2. Solenoid valve response from either extreme position will not exceed 100 milliseconds.

   3. Each solenoid is equipped with two coils to minimize peak and steady-state electrical power. A holding coil drawing 0.25 ampere operates during steady-stage operation and an energizing coil drawing 1.0 ampere cycles the solenoid.

D. DESIGN CAPABILITY

Failure modes derived from the failure mode and effect analysis of the helium subsystem provide for a safe engine shutdown despite the complete loss of electrical power. Helium is supplied to all purges and valves required to protect the engine from damage during shutdown after a power failure.

Replaceable flow control valves and metering orifices/venturis facilitate minor flow adjustments. Possible uprating or downrating of the engine in the future is simplified by these "cartridge" type units.

No helium system components require active temperature control during the engine cycle. Individual components have been designed to withstand the predicted environments.

E. DESIGN SUBSTANTIATION

This design approach, although representing a significant advancement in valve technology, has been demonstrated entirely practical and highly effective for valves currently in production. The LEM propellant storage system valve (Parker P/N 5640014) and the thrust chamber valve (Parker P/N 5690023) and the hydrazine propulsion module valve (Parker P/N 5677001) used on Classified Air Force Satellites are typical applications of the bipole, flexure guided, Teflon seat solenoid concept. Modular construction has demonstrated benefits of low weight, minimum envelope, minimum external leak points, structural integrity, and maintainability on such projects as the LEM oxygen modules (Parker P/N 5661000), Dynasoar oxygen and hydrogen modules (Parker P/N 562A0017M4), and the XB-70 nitrogen inerting module (Parker P/N 560-E0010).

Producibility of the bipole solenoid valve is firmly established. Over 900 of the LEM propellant storage system valves have been manufactured. New records for high performance have also been demonstrated; for example, the LEM bipole flexure guided, Teflon seat solenoid valve (thrust chamber valve) has been qualified for 10^6 cycles of operation at temperatures up to 250° F, while maintaining a seat leakage of less than one scc gaseous nitrogen per hr.

All parts of the current module will undergo a comprehensive test and qualification program as described in DVS No. 14.

SECTION IV
ARTICULATING MAIN PROPELLANT DUCTS

A. MAIN PROPELLANT DUCTS

1. Introduction

The P&WA main articulating propellant ducts have been routed to allow a reduction in the booster engine spacing from 109 in. to 105 in. The duct diameters have been reduced to 7.0 in., tight bends (R/d < 1) with turning vanes have been used where required, and the ducts have been routed slightly out of the gimbal plane. This design approach was necessary in order not to violate the 104-in. diameter maximum static envelope. P&WA is aware of General Dynamic Corporation's desire for a 105-in. engine-to-engine booster spacing and both ducts were routed accordingly.

The P&WA duct routings have enabled gimbal joints to be used that are within the current state-of-the-art. Preliminary designs for the bellows gimbal joints have been received from three suppliers. These designs permitted us to make a trade study that showed that an externally pinned gimbal was needed for the liquid oxygen duct, while the lighter internal ball strut design could be used for the fuel duct. Flow induced bellows fatigue problems have also been eliminated by specifying that flow liners be positioned in each gimbal joint. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

2. Design Description

Both stage mounted low pressure turbopumps transfer propellants to the engine high pressure turbopumps through 7.0-in. articulating ducts. These articulating ducts, shown in figures IV-1 and IV-2, allow the relative motion between the two turbopumps resulting from engine gimbaling, tolerances, deflections, and thermals. The fuel duct is insulated to meet both chill down and surface temperature requirements. The main flowmeters required for engine control are also a part of the fuel and liquid oxygen duct assemblies.

Approximately 20 lb of weight has been saved by routing the fuel duct with a 7-in. inner diameter rather than a 7.5-in. inside diameter. Some convergence is required at the inlet of the high pressure fuel turbopump, but routing a 7.5-in. diameter line would have meant a 20-lb weight penalty. Therefore, the line is routed with a 7-in. diameter and diffused up to 7.5 in. prior to entering into a convergent area flow section at the pump inlet. The additional pressure drop to accommodate this design was less than 10 psi. The total duct weight savings was 30 lb but the low pressure fuel turbopump weight was penalized 10 lb to regain the lost 10 psi. A similar weight optimization was performed for the liquid oxygen duct. As a result, the liquid oxygen duct is routed with a 6.9-in. inside diameter and diffused to a 7.35-in. diameter, approximately 5 diameters upstream of the high pressure turbopump inlet.

Because of the overall 104-in. diameter engine envelope, it was necessary to route the fuel duct with less than a one diameter bend at the high pressure pump inlet. With this type of bend, the pressure losses were too high and the flow distribution would have degraded pump performance. The inlet, shown in figure IV-3, was therefore designed which provides a uniform flow distribution for a minimum pressure loss. The inlet also provides a uniform flow distribution for a minimum pressure loss. The inlet also provides transition from a 7.5-in. diameter to an annulus 10 in. in outside diameter and 6.72 in. in inside diameter.

Inconel 625 was used for fabrication of both the main articulating propellant lines, most of the ducting, flanges, and various details, while the gimbal joint bellows and fuel duct at the low pressure turbopump discharge are formed from higher strength Inconel 718 sheet.

High torsion loads resulted from dynamic loading of the duct sections between the gimbals. A constant force spring hanger between the vehicle structure and the duct will damp most of these loads. We recommend that during Phase C/D, such hangers be coordinated as additional vehicle/engine interfaces.

Three internally pinned ball strut gimbal joints permit the fuel duct to articulate. The joints shown in figure IV-4 are vacuum jacketed and have flow liners. A trade study indicated that this configuration would be the lightest of three possible designs. The externally pinned and ball race gimbal designs required larger envelopes and were heavier. Each gimbal joint is designed to deflect ±15 deg from nominal. Externally pinned gimbal joints as shown in figure IV-5 are required for the liquid oxygen duct because of the higher torsion loading that the heavier liquid oxygen fluid creates under dynamic loading. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

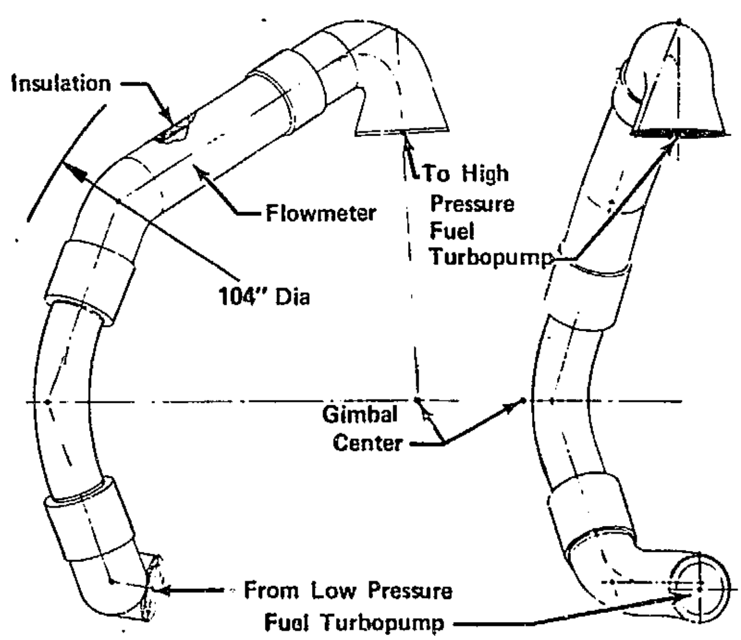


Figure IV-1. Fuel Duct Routed Inside 104 in. Diameter Powerhead Envelope FD 46378

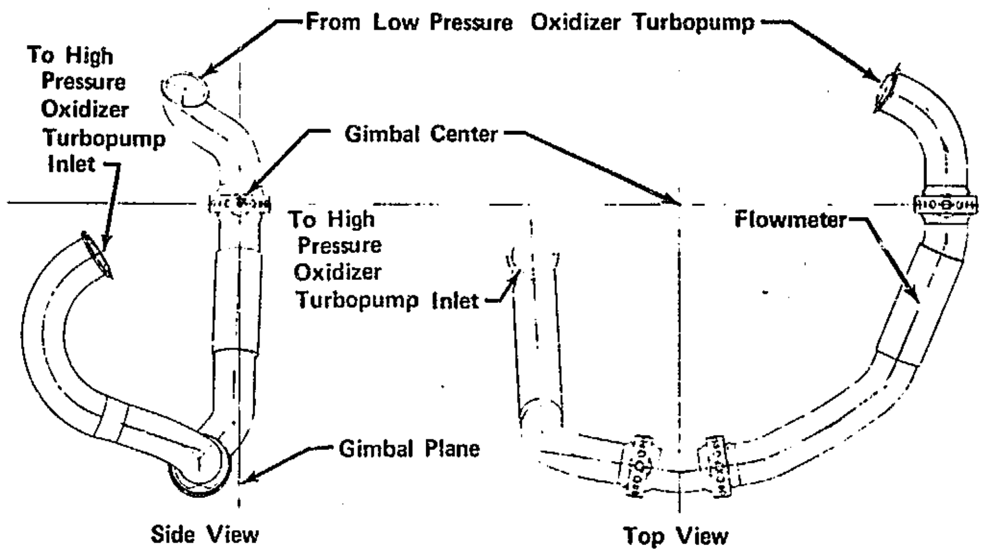


Figure IV-2. Oxidizer Duct Integral Flow Meter Saves Weight FD 46379

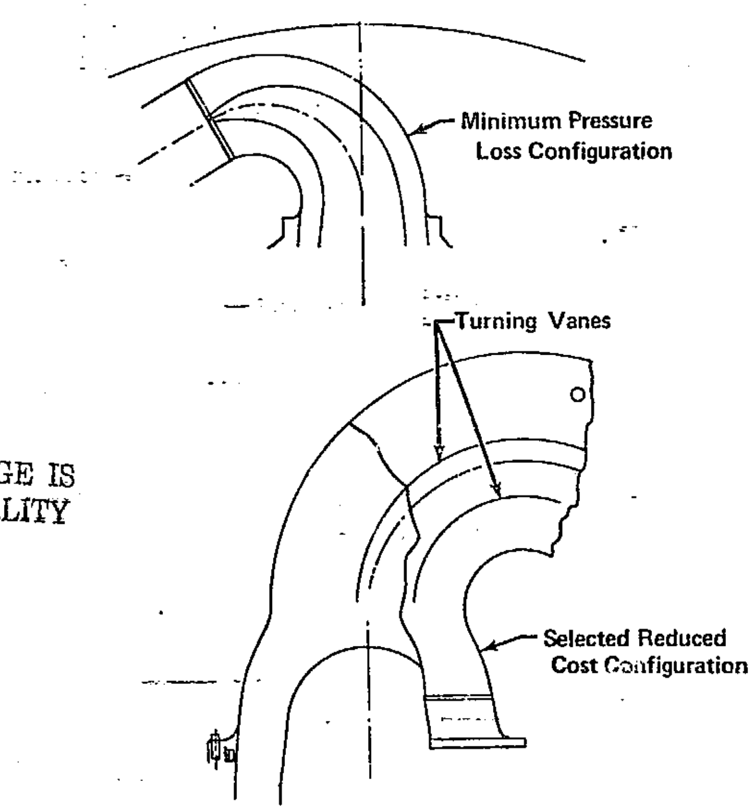


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Figure IV-3. Alternate Designs Considered for Reducing Pressure Drop at Fuel High Pressure Turbopump Inlet FD 52355

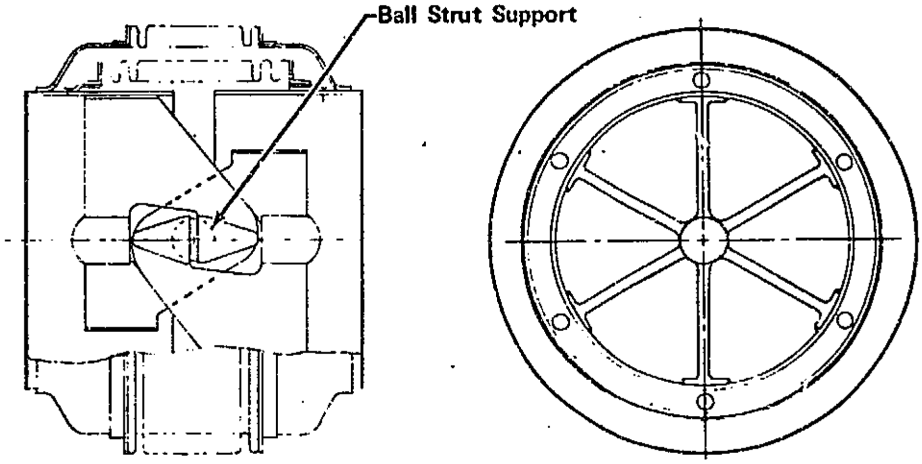

Figure IV-4. Vacuum Jacket Design Prevents Liquid Air Formation FD 46380

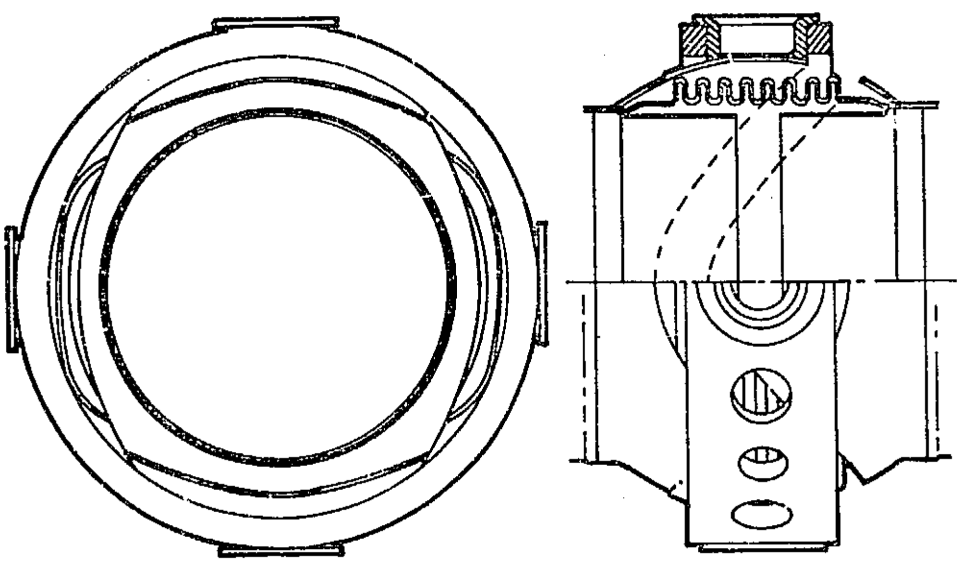

Figure IV-5. Torsion Loads Require Externally Pinned Gimbal Design FD 52354

A hybrid insulation system has been selected for the fuel duct. The gimbal joints are individually jacketed and depend upon cryopumping, in which an interstitial gas freezes onto the cold surface, to create a vacuum. A 1-in. layer of foam insulates the remainder of the duct. A higher percentage of the duct is insulated using foam, than is possible with pure vacuum jacketed systems. Foam samples are currently being tested from various suppliers. The system of foam and individual gimbal jackets that have been selected will allow a reduction in the duct assembly weight with a corresponding decrease in cost. Should this system prove inadequate under development testing, conversion to a total vacuum jacketed system can readily be affected.

A low loss flowmeter is welded in as an integral part of both ducts. Calibration of the flowmeter will be performed in a fixture which will simulate the actual flow conditions. After the duct assembly is completed, the flowmeter will be recalibrated to verify accuracy. The flowmeter is of a venturi type and is welded into the duct to eliminate the weight of two 7-in. diameter flanges per assembly.

3. Design Requirements

The proposed design is in compliance with all of CEI Specification CP2291. Specific requirements that influence the concept selection are listed below:

1. The engine gimbaling capability shall be ±7.5 deg for the orbiter and ±10.5 deg for the booster. This includes 0.5 deg for snubbing and overtravel. In addition, 0.5 deg shall be allowed for engine misalignment, (paragraphs 3.1.4 and 3.3.3 of the CEI Specification and paragraph 4.4.2 of the ICD).

   Compliance - Each articulating duct contains three bellow type gimbal joints. The ducts are routed near the gimbal plane with the joints close to the Y and Z axes. This routing allows for all of the relative motion between the stage-mounted low pressure propellant pumps and the engine high pressure turbopumps that results from gimbaling ±11 deg, tolerances, and thermals.

2. The engine shall prevent the formation of liquid air that might come into contact with the vehicle structure, (paragraph 3.2.11 of the CEI Specification CP2291).

   Compliance - One-in. thick foam on the duct sections and cryopumped vacuum jacketed gimbal joints prevent the formation of liquid air on the fuel duct.

3. The engine shall be designed to withstand the operational and nonoperational gimbal duty cycles imposed for the life of the engine, (paragraph 3.5.1.6 of CEI Specification CP2291).

   Compliance - The fatigue criteria of SSME Structural Design Criteria, PWA FR-4449, (Miners rule) was used to verify cycle life. All gimbal joint bellows have as a minimum twice the required life.

4. Flexible metal ducts of 2-in. inside diameter or less shall be of multiple-ply construction, (paragraph 3.7.7.2 of CEI Specification CP2291).

   Compliance - Although the main articulating ducts have 7-in. inner diameters, the high gimbal angles causes gimbal bellows to be multi-ply.

5. The average velocity of the fluid at any station in the duct shall be less than that corresponding to 0.3 Mach number.

   Compliance - For the fuel duct, the maximum average velocity is 161 ft/sec, which corresponds to a Mach number of 0.046. The main liquid oxygen duct has a 72.9 ft/sec maximum average velocity with a Mach number of 0.025.

6. The design of flexible metal ducts shall be such that the mechanical resonant frequency of the bellows system in its operating environment is not coincident with the vortex shedding frequency range and shall be capable of meeting the test requirements of 20M02360 unless otherwise approved.

   Compliance - Flow liners in each gimbal joint shield the bellows. Vortex shedding does not occur.

4. Design Capability

In both ducts, the wall thickness is limited to 0.030 minimum because of handling considerations. The maximum effective stress in the fuel duct is 53,295 psi with the yield stress of Inconel 625 at -300°F being 77,000 psi. The Inconel 718 portion of this duct has a maximum operating stress of 126,200 lb with the limit stress being 134,000 lb. To meet the leakage criteria, the flanges for both ducts have been designed in accordance with SSME Plumbing Design Criteria, PWA FR-4455, with deflection at the seal contact points being limited to 0.0015 in. per flange. Therefore, the flanges are not presently stress-limited and have the capability of withstanding higher reaction, shock, and vibratory loads than are currently predicted.

If needed, higher flow rates can be accommodated in the ducts with the pressure drop being the limiting criteria. This is possible because flow liners are provided in all of the flexible gimbal joints, which eliminates any possible flow induced fatigue stresses in the bellows sections. The bellow stresses will be independent of flow rate.

5. Design Substantiation

Forces, moments, and stresses due to thermal changes, accelerations, and external static loads were calculated. Those due to fluid momentum and pressure effects are calculated using High Pressure Plumbing Design Procedure. The two sets of data are combined on the "Tubstres" Program in the FRDC Minicomp System (a system of IBM 2471 remote computer terminals) to give the effective stress levels. The loads are tabulated in Engine Plumbing and Miscellaneous Hardware Design Calculations, PWA FR-4430.

The gimbaling motions of the ducts were checked using a half-size mockup as shown in figure IV-6, and verified by layout drawings. This procedure led to the conclusion that a gimbal joint with a maximum angulation capability of ±15 deg would be sufficient at all locations. Preliminary gimbal joint designs were requested from three different vendors, all of which were suppliers of flexible ducting for the Saturn Program. Their designs were reviewed and the best picked for a trade study.

The trade study determined that three designs; ball race, externally pinned, and internal ball strut gimbals, were all possible design selections. The internally pinned ball strut design is lighter and has the smallest envelope. Its main weakness is that torsion loading will be transmitted through the bellows section. The externally pinned design is slightly heavier but does not put torsion into the bellows. The ball race design was heavier, limited in angulation, and transmitted torsion through the bellows. (Refer to Valves and Interconnects Trade Studies, PWA FR-4442.)

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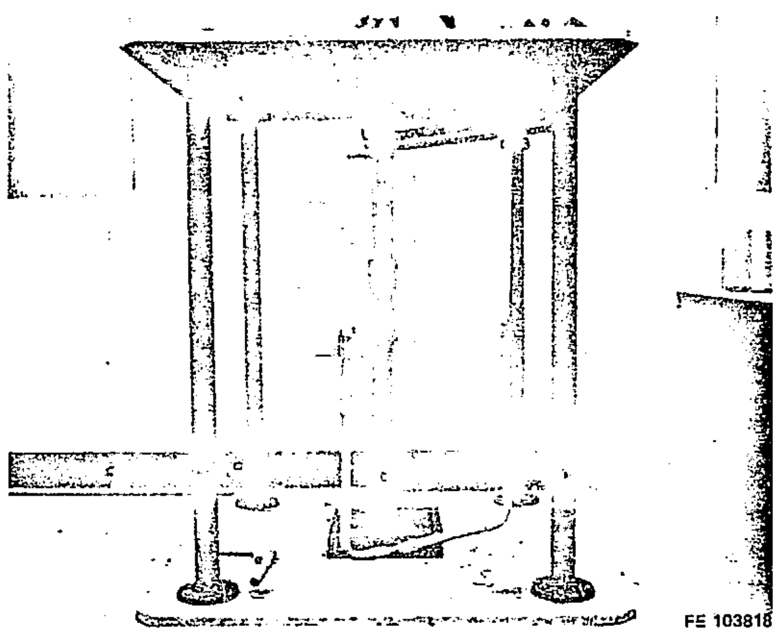

Figure IV-6. Gimbal Deflections Quickly Determined on Motion Frame FD 52390

After the actual design loads were determined, it was found that the internal ball strut design shown in figure IV-4 was best for the fuel duct, and the external pin design shown in figure IV-5 would handle the higher torsion loads of the liquid oxygen duct.

Test data have shown that no hydrogen degradation occurs in either Inconel 625 or 718 at the fuel duct design point conditions of 45°R at 285 psia. Degradation apparently does not occur at temperatures below 150°R based on testing performed to date. Therefore, both of the materials are used in the fuel duct construction.

Two high-pressure fuel pump inlet designs were evaluated. They are shown in figure IV-3. The first bend is a constant area annulus. Its design was based on NACA Technical Note 3995. The second design is an elbow with turning vanes. Pressure drop through the first bend is about 0.5 psi less than through the elbow with turning vanes, but the added fabrication cost and increased weight more than offset this advantage. The P&WA fuel duct incorporates the turning vane inlet as a part of the duct assembly.

SECTION V
FLUID COMPONENTS

A. FLUID INTERFACE LINES AND COMPONENT INTERCONNECTS

1. Introduction

The P&WA fluid interface lines have been group routed. Qualification and development testing cost is thereby reduced, installation improved, and dynamic induced stresses damped.

Multiple bellows and reduced flow velocities have been used to avoid flow-induced fatigue stress problems in the flexible sections.

The P&WA interface flanges improve maintenance and installation. One common bolt size, free-ring flanges, and bleed ports have been specified. Therefore, installation, with no possibility of cross-connecting, will be easier and leak checkout more rapid.

2. Design Description

The fluid interface lines, shown in figure V-1, have been group routed. As a result, the qualification and development testing of these flexible lines will simulate actual engine conditions. Group-routed lines can be qualified on a fixture such as shown in figure V-2, which was used for the J-2 fluid lines. Such a device allows flow and pressure checking of the lines during gimbaling. This group testing will also reduce the cost of qualification.

Other than the two main propellant inlets, only seven lines are required to transfer fluids between our engine and the stage. This is two less lines than the ICD specifies because the P&WA engine does not require hydraulic fluid from the vehicle, although space has been allowed at the interface for the hydraulic lines in the event the vehicle contractors find it more convenient to route their thrust vector control hydraulic lines through this interface.

The seven interface lines and their ICD interface numbers are: 6 hydrogen tank pressurization, 7 oxidizer tank pressurization, 8 fuel drain, 9 engine ground standby purge supply, 10 engine helium supply, 20 fuel recirculation, and 21 liquid oxygen recirculation.

Angulation of the flexible hose sections has been reduced by locating the hose section on the Y and Z axes near the gimbal plane. Since flexible hose sections can accommodate parallel offset, only two sections per line are required. These interface lines are designed to withstand all of the relative motion between the stage-mounted interface panel and the engine. Motion from gimbaling, tolerances, and thermals were all considered.

From a purely maintenance viewpoint, the fluid panel location should be near the outer edge of the engine envelope. This would probably be feasible on the booster vehicles because there is structure available to support the panel. Unfortunately, the orbiter vehicles of North American-Rockwell and McDonnell Douglas Aircraft Corporation do not have structure in that outboard area.

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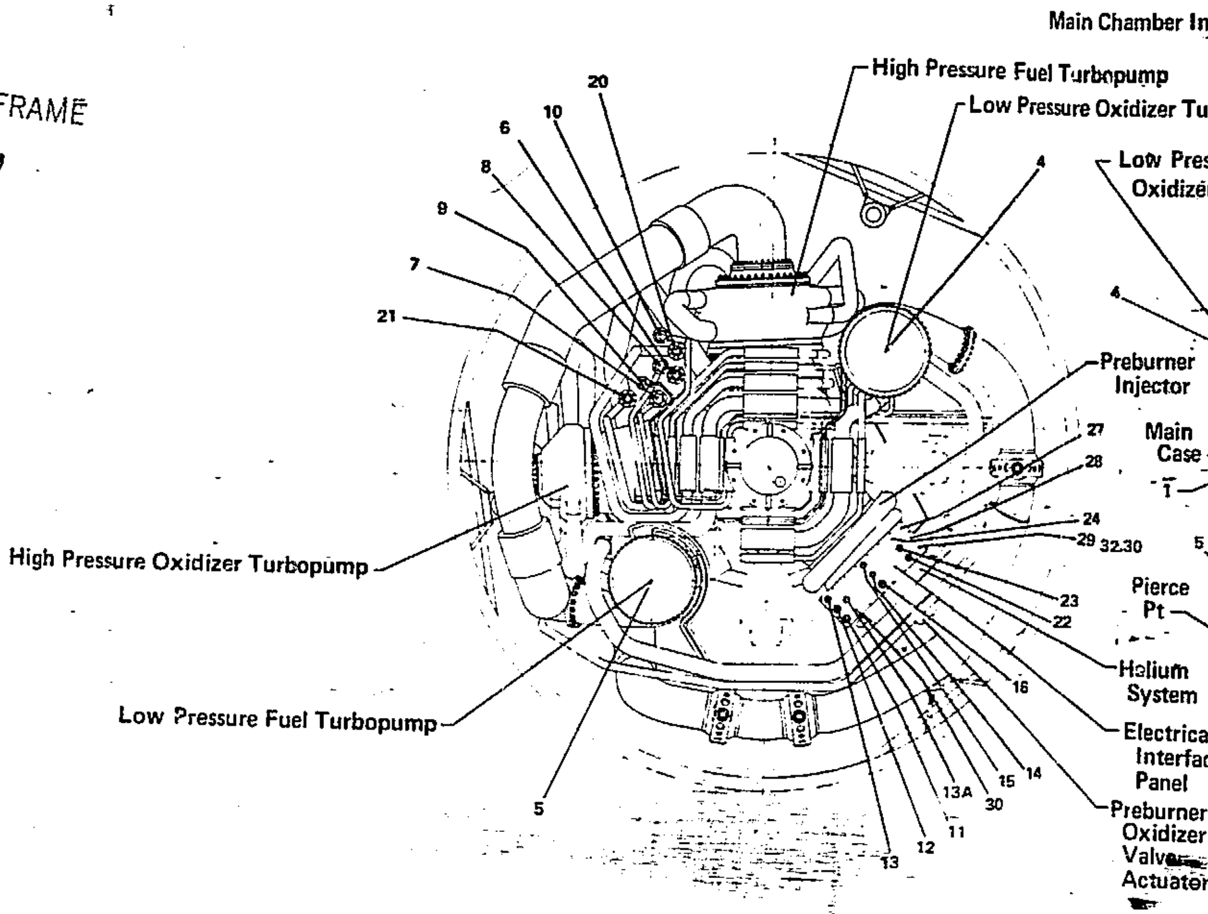
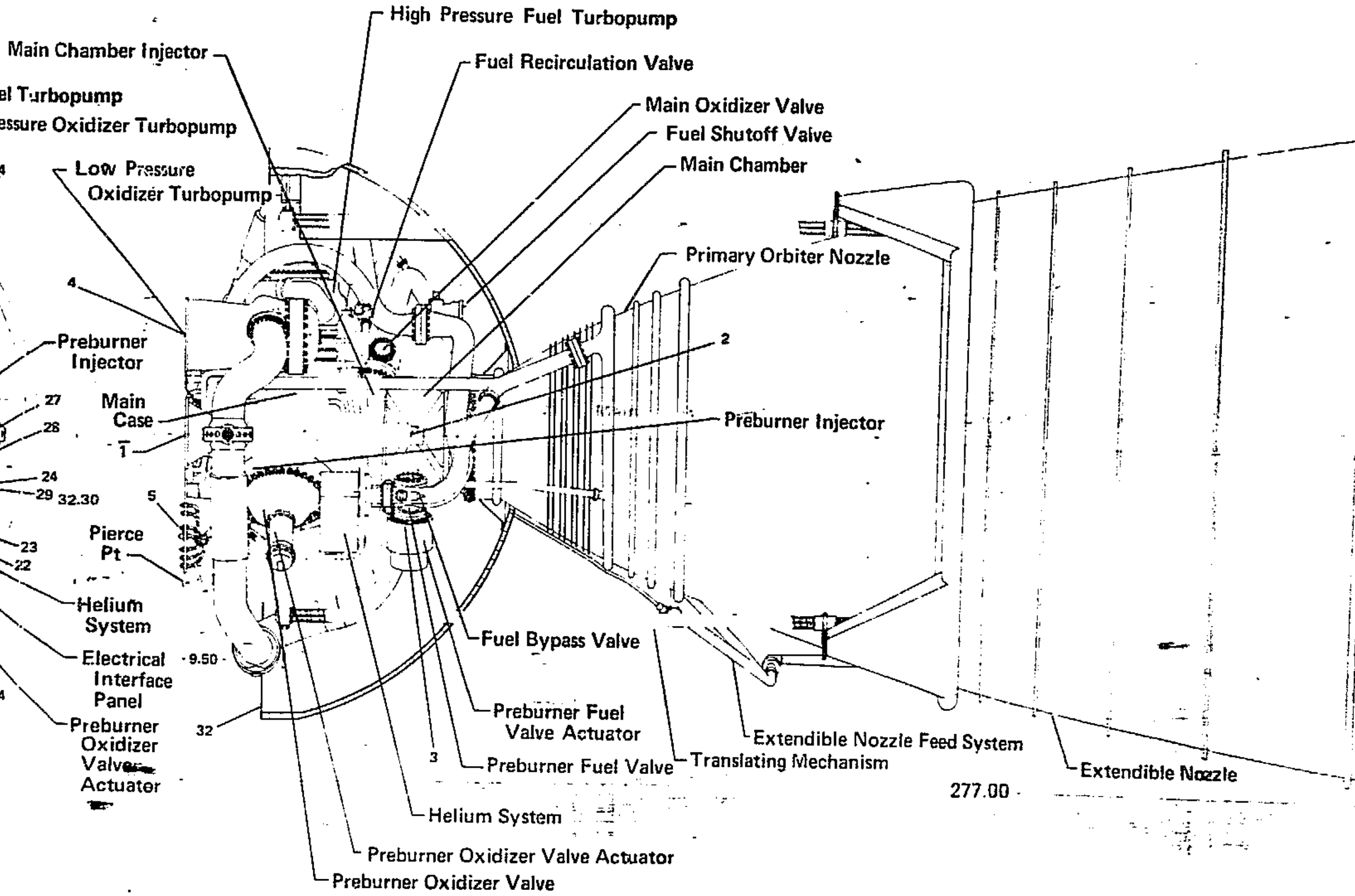


Figure V-1. Fluid Interface Lines Are Group Routed Near Gimbal Plane



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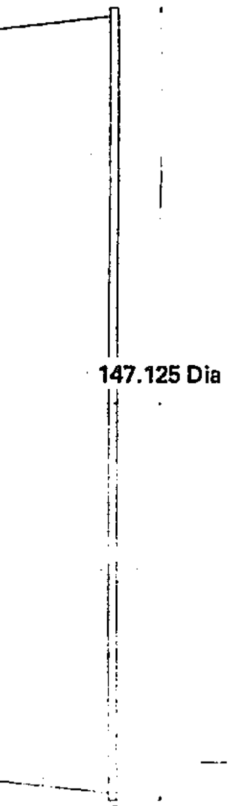

Vehicle Connections - Interface Points

Interface No.	Description
1	Gimbal Bearing
2	Gimbal Actuator Attach Point
3	Gimbal Actuator Attach Point
4	Lox Engine Inlet
5	Fuel Engine Inlet
6	Hydrogen Tank Pressurant Outlet
7	Oxidizer Tank Pressurant Outlet
8	Fuel Drain
9	Engine GN_2 Standby Purge Supply
10	Engine Helium Control/Purge Supply
11	Power, DC
12	Power, DC
13	Power, DC, Orbiter Only, ECU Heater
13A	Power, DC, Orbiter Only, ECU Heater
14	Data Bus
15	Data Bus
16	Instrumentation
17	Not Used
18	Not Used
19	Not Used
20	Fuel Recirculation
21	Oxidizer Recirculation
22	Power, AC
23	Power, AC
24	Power, AC, Optional, Fuel Recirculation
25	Not Used
26	Not Used
27	Power, AC, Optional, Fuel Recirculation
28	Power, AC, Optional, Oxidizer Recirculation
29	Power, AC, Optional, Oxidizer Recirculation
30	Vehicle/Engine Electrical Ground
31	Booster Heatshield (Not Shown)
32	Orbiter Heatshield

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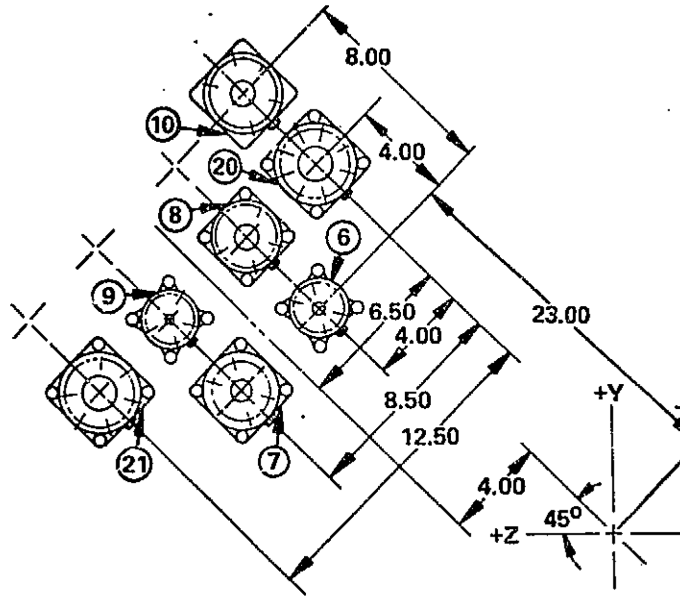
Figure V-2. Group Qualification Reduces Procurement Cost FD 52225

Therefore, the panel location shown in figure V-3 was selected primarily to satisfy the orbiter vehicle configurations. As specified in the ICD interface drawing 3.2.2, the panel is located at the YZ plane and in the (+Z, +Y) quadrant.

Another advantage of the group routing is the opportunity it provides for line-to-line bracketing. Such brackets can dampen dynamic loading and offer support against torsional loading. This is important because the flexible hose sections are already highly stressed from being pressurized and flexed simultaneously. In the past, flexible hose sections have failed when adequate vibration damping was not provided. More recently, the work of Dr. Garlach and his associates at the Southwest Research Institute has shown that often the vortex shedding frequency of the fluid flow across the bellows can couple with the natural mechanical frequency of a bellows convolutions. When such a condition exists, premature fatigue failure of the bellows will result. To avoid this occurrence, fluid velocities in the line have been kept below a Mach number of 0.3. Also, using Dr. Garlach's techniques, the first and second resonant frequencies were calculated for each line. Where required, lines were resized to avoid these resonant frequencies. The multiple construction of each flexible metal bellows offers further damping and provides an added margin of safety against pin hole leaks.

Because of possible hydrogen degradation problems, Inconel 625 and 718 were eliminated as materials for the hydrogen tank pressurization and fuel drain lines. AISI 347 stainless has been specified for these lines, although the investigation of other alloys such as 304N is continuing as a possible replacement. Inconel 625 will be used for the remaining interface lines. Inconel 625 has good fatigue life and shortens the length of hose required for a given deflection when compared with AISI 347.



Point	Line
6	LH_2 Tank Pressurant
7	LO_2 Tank Pressurant
8	Fuel Drain
9	Nitrogen Supply
10	Helium Supply
20	Fuel Recirculation
21	Oxidizer Recirculation

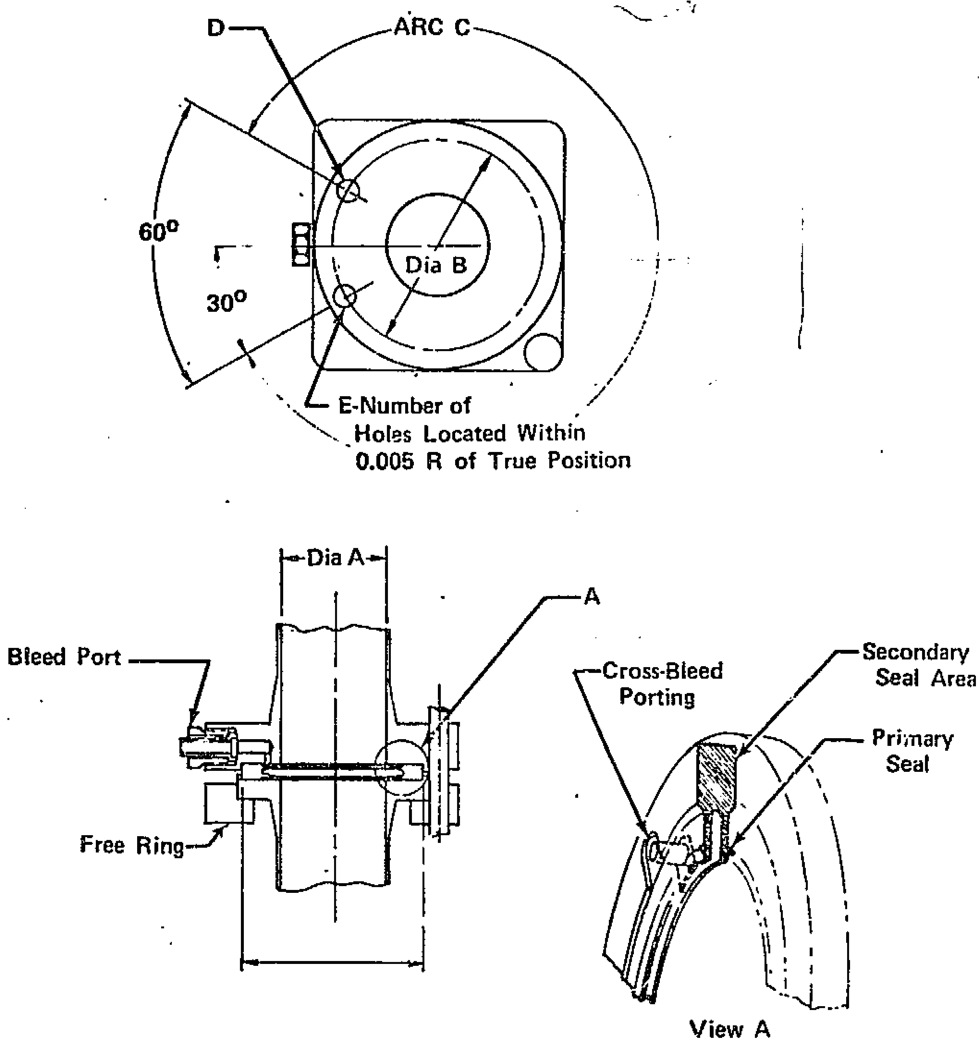
Figure V-3. Interface Flange Location FD 52234

After checking for maintenance clearances, it was concluded that the ICD flange-to-flange spacing was adequate. Also, consideration was given to whether this spacing would allow for increasing line diameters without re-spacing. In this respect, the spacing was also found adequate. All of the interface flanges use 0.250-in. diameter bolts to further prevent error.

Bleed ports are provided at each interface flange to reduce the time needed for leak check when mating the engine to the vehicle. Both the engine and vehicle sides of the interface connection have been designed as shown in figure V-4. A pressure activated seal is the key to this quick pressure check-out. Each side of this dual static seal has two sealing surfaces, a primary and a secondary. An internal vent hole vents the low pressure cavities between the two sealing surfaces. An external bleed port, designed as an integral part of the flange, is used to collect and monitor seal leakage.

Further reduction in the time required to connect the interface lines to the panel has been achieved by specifying that a free ring flange be used on the engine side. This will allow the group of lines to remain bracketed together when installed, thereby eliminating any chance of putting torsion or additional misalignment into the lines by making unneeded time-consuming secondary adjustments.

Toroidal segment seals have been specified for the engine end of the interface lines. Since these flanges are not disconnected as frequently as those at the interface, it was decided to take advantage of their better sealing capability even though leak check may prove more difficult.



	A	B	C	D	E
Lox Recirculation	1.32	2.700	300°	0.281	7
Lox Tank Pressure	1.777	3.300	300°	0.281	12
Fuel Recirculation	1.32	2.700	300°	0.281	8
Fuel Tank Press	1.430	2.900	300°	0.281	10
Fuel Drain	0.960	2.400	300°	0.281	5
Helium Supply	1.000	2.400	300°	0.281	6
Nitrogen Supply	0.335	1.890	300°	0.281	5

Figure V-4. Pressure Check Ports and Free Ring Improved Interface Maintenance FD 52223

One interface line, the fuel recirculation, is vacuum jacketed. The jacket is required to reduce the power needed for fuel recirculation and adequately prevents the formation of liquid air on this line.

Many of the design problems for the interface lines are common to each, but in sizing these lines, quite different considerations are needed for each line. Engine Plumbing and Miscellaneous Hardware Design Calculation, PWA FR-4430, contains several pressure drop versus weight curves that were used in optimizing the line sizes. All were optimized for the flow conditions specified in table 5.1.2 in the ICD.

3. Design Requirements

The proposed design is in compliance with all requirements of CEI Specification CP2291. Specific CEI specification requirements that influence the concept selection and design elements are met as follows:

1. The engine gimbaling capability shall be ±7.5 deg for the orbiter and ±10.5 deg for the booster. This includes 0.5 deg for snubbing and over-travel. In addition, 0.5 deg shall be allowed for engine misalignment in accordance with CEI paragraphs 3.1.4 and 3.3.3 and paragraph 4.4.2 of the ICD.

   Compliance - The fluid interface lines each have two flexible hose sections to accommodate motion resulting from the ±11 deg gimbaling.

2. The engine shall prevent the formation of liquid air that might come into contact with the vehicle structure, in accordance with paragraph 3.2.11.

   Compliance - Vacuum insulation of the fuel recirculation line is capable of maintaining the average maximum heat transfer rate at 8 Btu/sq ft/hr and preventing the formation of liquid air.

3. Mechanical connections shall be designed to preclude inadvertent cross connection, in accordance with paragraph 3.5.1.

   Compliance - All of the flanges have either different numbers of bolts or different bolt circle diameters. Also, the brackets provided for group routing the interface lines, fix the relative positions of the flanges to each other.

4. The engine contractor shall have the responsibility for specifying the design and sealing requirements for the stage half of the fluid connect flange in accordance with paragraph 3.5.1.4.

   Compliance - Swivel-type flanges with pressure activated seals as previously shown in figure V-4 are provided at each interface.

5. Flexible metal ducts of two-inch inside diameter or less shall be of multiple-ply construction. The average velocity of the fluid at any station in the duct shall be less than that corresponding to 0.3 Mach number. These requirements are in accordance with paragraph 3.7.7.2.

   Compliance - All six flexible interface lines have multi-ply bellows. Details for each line are given in Engine Plumbing and Miscellaneous Hardware Design Calculations, PWA FR-4430.

The maximum Mach number for each of the seven interface lines are:

Interface No.	Description	Mach No.
6	Hydrogen tank pressurization (Not a Flexible Line)	0.33
7	Oxidizer tank pressurization	0.25
8	Fuel drain	0.14
9	Engine ground standby purge supply	0.16
10	Engine helium supply	0.06
20	Fuel recirculation	0.122
21	LOX recirculation	0.002

6. The design of flexible metal ducts shall be such that the mechanical resonant frequency of the bellows system in its operating environment is not coincident with the vortex shedding frequency range and shall be capable of meeting the test requirements of 20M02360 unless otherwise approved. These requirements are in accordance with paragraph 3.7.7.2.

   Compliance - Determination of the mechanical resonant and vortex shedding frequencies by analytical procedures shows that they do not coincide in any of the seven lines. Negotiations are being made with Southwest Research Institute to verify the predictions by actual test of each line.

7. For engine checkout and maintenance purposes, the engine shall be designed to allow leak check capability of either external or internal seals. All separable connections shall not exceed an allowable gas leakage of 1×10^{-4} scc/sec of helium at operating or leak check pressures in accordance with paragraph 3.7.12.

   Compliance - Each interface flange has a bleed port to facilitate pressure checkout as shown in figure V-4. Pressure-activated seals are used at the interface panel and toroidal segment seals at the engine connections.

8. The flange joint design shall meet requirements of paragraph 3.7.13.1.

   Compliance - Standard nuts and bolts and through-bolt holes are used at the interface. A minimum number of bolts are used at the interface to improve maintenance.

C2

9. The engine shall be designed to withstand the operational and nonoperational gimbal duty cycles imposed for the life of the engine. The gimbal duty cycle requirements shall be met in accordance with paragraph 3.5.1.6.

Compliance - Fatigue criteria of SSME Structural Design Criteria, PWA FR-4449, was followed to verify life. All lines have at least twice the required life.

4. Design Substantiation

The P&WA motion study frame shown previously in figure IV-6 was used to determine the amount of angulation and parallel offset required by each flexible hose section. Torsion loads were predicted analytically. This information, in addition to the flow requirements for each line, were supplied to Anaconda, who in turn designed the metal bellows flex sections. Anaconda's design included a prediction of the locked-in resonant flow velocity range plus all of the physical bellows data. The P&WA analytical group also checked for flow-induced stresses using the physical bellows data supplied. The correspondence with Anaconda and other data can be found in Engine Plumbing and Miscellaneous Hardware Design Calculations, PWA FR-4430.

Free-ring flanges at the interface were designed using the data in AFRPL-TR-69-97. This was a study conducted at the Battelle Memorial Institute entitled, "Development of AFRPL Flanged Connectors for Rocket Fluid Systems." Flanges at the engine end are designed in accordance with Plumbing Design Criteria, PWA FR-4455. Discussion of the various analytical techniques is given in Vol. I, Sec. VIII.

5. Design Capability

The P&WA flexible interface lines have been designed for a total engine gimbal angle of ±13 deg. This two deg margin over the maximum CEI specification value of ±11 deg allows for unknown vehicle location tolerances, thermals, and deflections at the fluid panel. The lines were designed to full gimbal duty cycle at ±13 deg, therefore they have additional cycle life capability if these tolerances prove to be small.

The P&WA flow-induced stress analysis was performed using velocities 10% greater than those possible by using the flow data of ICD table 5.1.2. Each line was determined to have adequate cycle life even at the increased velocities. By using this velocity margin, future changes in density, pressure, flow, or fluid temperature are possible without redesign.

Leak check bleed ports are located on the vehicle side of the interface. This will allow leakage monitoring lines to be installed prior to engine installation. These lines can also remain attached during engine removal without interferring or causing delays. Automatic ultrasonic leak detection systems are being investigated. Dynamatec Corporation has supplied technical and cost information on this system. (Refer to Plumbing Design Criteria, PWA FR-4455.)

B. PLUMBING

1. Introduction

Pressures in excess of 6500 psi at line diameters up to 4 in. require sophisticated design procedures to maintain the integrity of the engine system and to provide the necessary flexibility for thermal growth and tolerance conditions.

The P&WA single preburner powerhead concept provides a unique solution for ducting of all preburner products (5500 psi at 2200°R) within the main case of the engine, thereby eliminating the need for external plumbing to accommodate this flow. This also allows use of slip joints for these ducts which eliminates the problems associated with thermal growth and greatly alleviates the manufacturing and assembly problems resulting from tolerance stackup control which would be required with a low leakage rate connection. With the P&WA powerhead configuration, hot gas leakage is not a problem because all ducting takes place within the main case of the engine.

The problem of designing the remaining high pressure plumbing was minimized by selecting high strength tube material wherever possible, not only for the lowest weight, but to ensure tube walls thin enough to realize a useful amount of tube flexibility with reasonable tube lengths. The 800°R hydrogen lines of the low pressure turbopump turbine drive supply and return lines are fabricated from PWA 1053 material. The materials for oxidizer, inert gas, and cryogenic fuel lines were selected to be thermally compatible with the component connections and with the environmental conditions of the fluid. Table V-1 and figure V-5 tabulate the major rigid plumbing lines and define the fluid contained, pressure, temperature, flow rate, and material selected.

2. Design Description

The fact that the large diameter, heavy wall plumbing used on the engine could produce high flange loads, pipe moments, and excessive reaction loads on major engine components has required that each line be thoroughly analyzed. To ensure design consistency, a comprehensive set of ground rules and special analytical computer routines were established. These design guides control the flanged joint dimensions, bolt size and quantity, bolt assembly preloads, minimum bolt load during operation, minimum allowable tube wall thickness, and external pipe moment allowable. The use of these ground rules and routines are covered extensively in Plumbing Design Criteria, PWA FR-4455. This Design Manual has been published and distributed to all SSME designers and is complied with in the design of all plumbing lines on the SSME.

The Design sequence employed in a specific line design is outlined below:

1. A determination of line requirements is made from the engine cycle (flow, pressure, temperature, and environment). Line size, wall thickness, and materials are determined on a preliminary basis.

Table V-1. Major Rigid Plumbing Lines

Line No.	Line	From	To	Fluid	P, psi	T, °R	lb/sec	Material	Inside Diameter, in.
1	Fuel High Pressure Pump Discharge Collector Lines	2nd Stage Discharge	Fuel Shutoff Valve	H_2	6570	130	207	Inconel 718	2.63
2	Preburner Injector Fuel Supply Line	Fuel Shutoff Valve	Preburner Fuel Valve	H_2	6450	131	145	Inconel 718	3.69
3	Primary Nozzle Coolant Supply Line	Preburner Injector Fuel Supply Line	Primary Nozzle Inlet Manifold	H_2	6440	131	41	Inconel 718	2.62
4	Main Case Coolant Supply Line	Transpiration Coolant Supply Line	Main Case	H_2	6268	326	3.6	PWA 1053	0.875
5	Extendible Nozzle Coolant Supply	Primary Nozzle Coolant Discharge Line	Nozzle Inlet Manifold	H_2	6330	131	6.3	Inconel 718	0.9
6	Fuel Start Transient Bypass Line	Primary Nozzle Coolant Supply Line	Preburner Injector Fuel Manifold	H_2	6180	690 135	NA	PWA 1053	2.60
7	Transpiration Coolant Supply Line	Primary Nozzle Tap-Off	Transpiration Coolant Valve	H_2	6308	325	8.7	PWA 1053	1.25
8	Oxidizer Low Pressure Turbine Supply Line	Fuel Low Pressure Turbine Exit	Oxygen Low Pressure Turbine Inlet	H_2	4530	651	26	PWA 1053	3.00
9	Oxidizer High Pressure Pump 1st Stage Discharge Lines	1st Stage Discharge	Oxidizer Interstage Y-Block	O_2	5697	209	1228	Inconel 718	3.38
10	Oxidizer High Pressure Pump 2nd Stage Inlet Line	Oxidizer Interstage Y-Block	2nd Stage Inlet	O_2	5560	210	224	Inconel 718	3.10
11	Main Chamber Oxidizer Supply Line	Oxidizer Interstage Y-Block	Main Oxidizer Valve	O_2	5560	210	1010	Inconel 718	3.90
12	Preburner Injector Oxidizer Supply Line	Oxidizer Discharge Y-Block	Preburner Oxidizer Valve	O_2	6725	216	163	Inconel 718	2.50

*Conditions specified do not constitute a compatible set for any unique operating point.

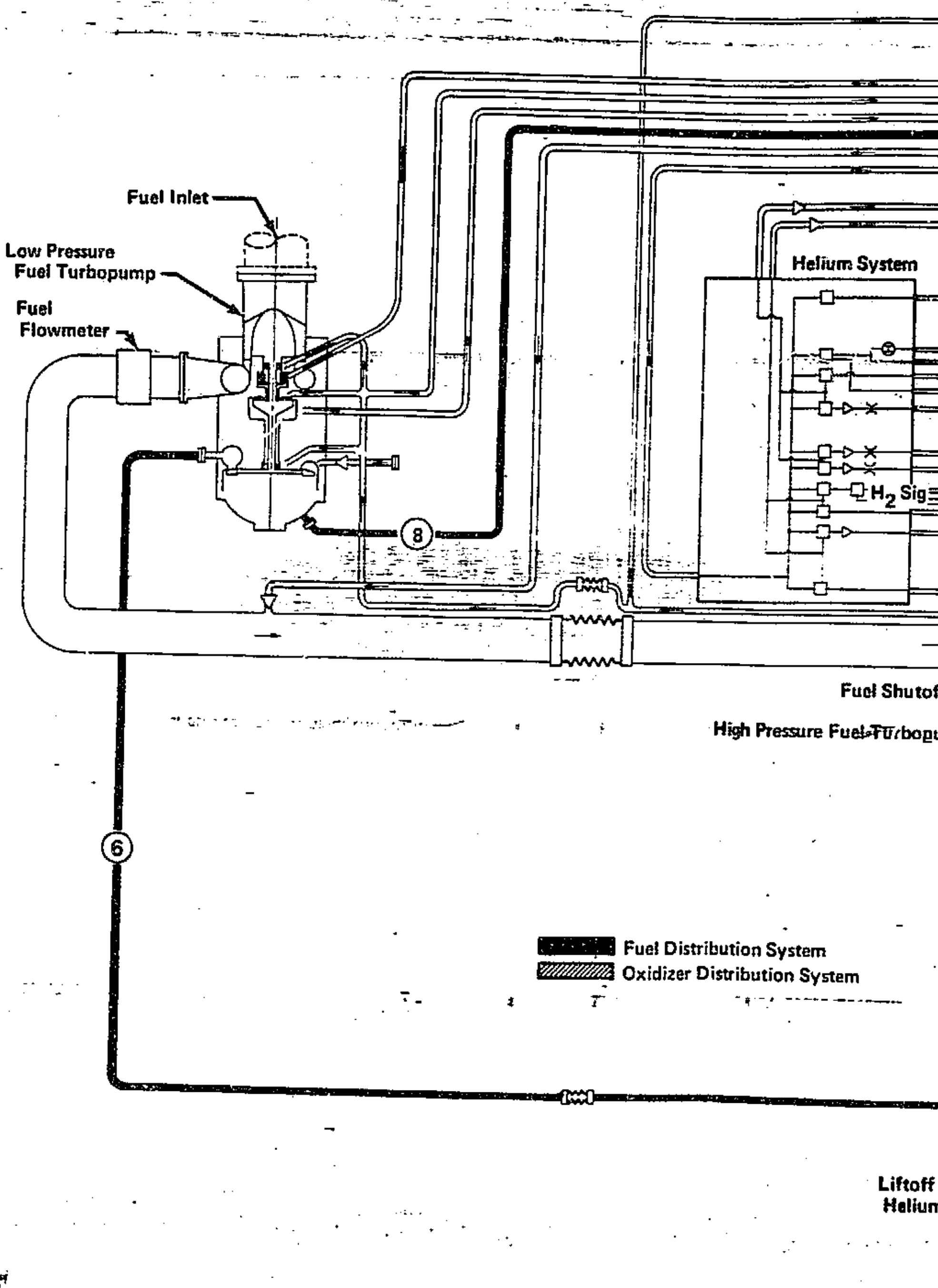


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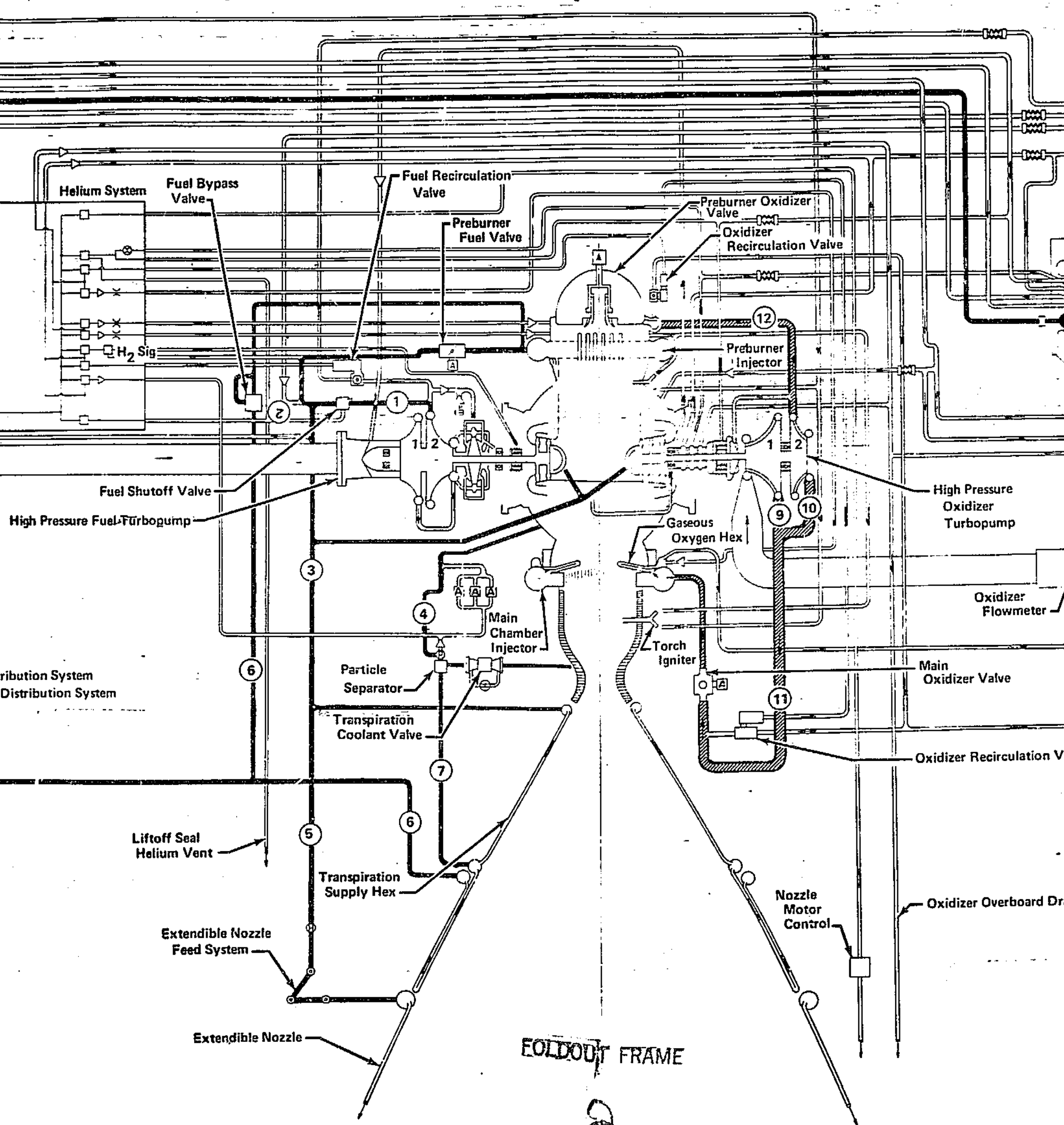
Figure V-5. Major Plumbing Comprised in Two Distribution Systems



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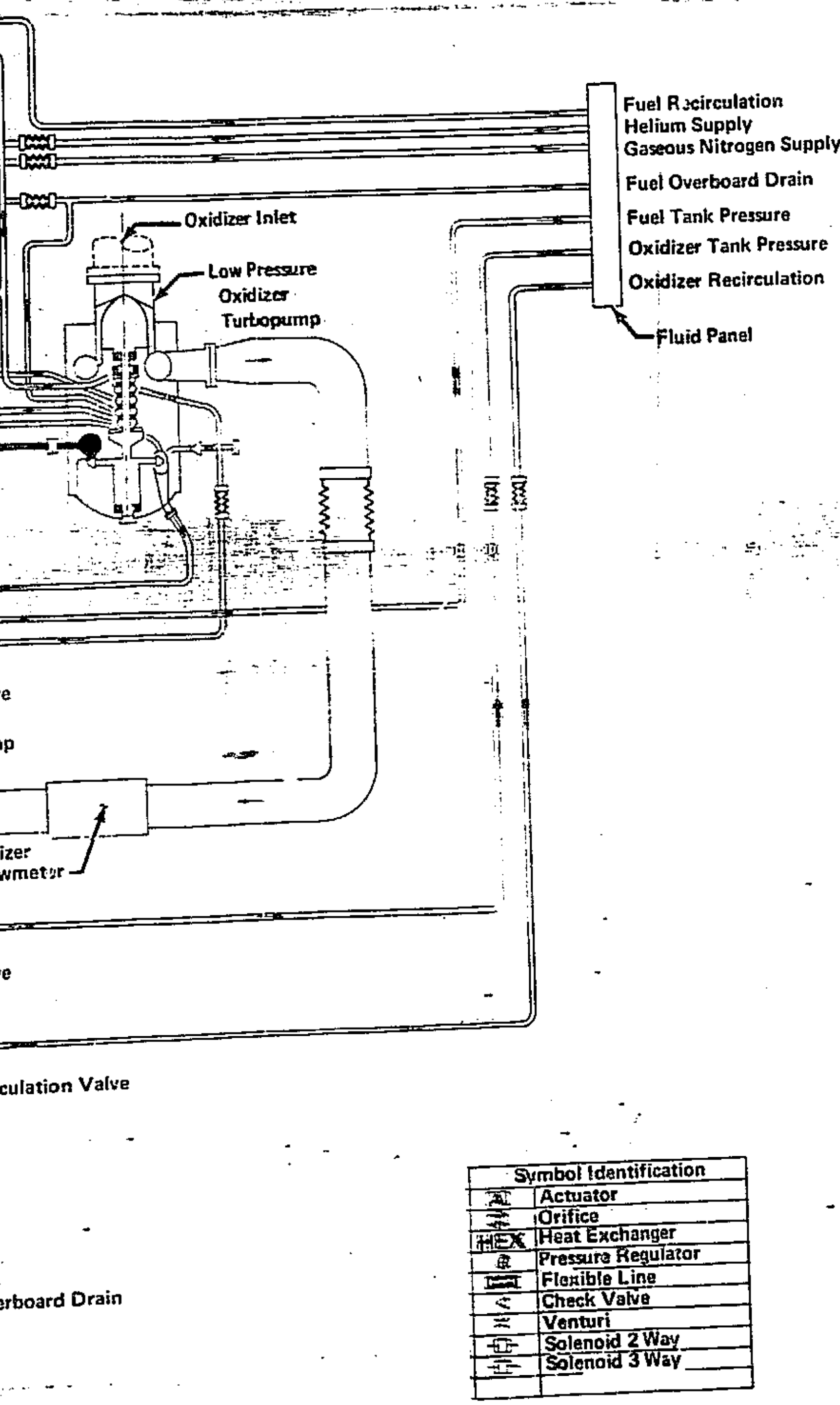

Fuel Recirculation
Helium Supply
Gaseous Nitrogen Supply
Fuel Overboard Drain
Fuel Tank Pressure
Oxidizer Tank Pressure
Oxidizer Recirculation
Fluid Panel
Oxidizer Inlet
Low Pressure Oxidizer Turbopump
e
p
izer
wmeter
e
culation Valve
erboard Drain
Symbol Identification
Actuator
Orifice
HEX Heat Exchanger
Pressure Regulator
Flexible Line
Check Valve
Venturi
Solenoid 2 Way
Solenoid 3 Way

2. Preliminary line routing is established, based on the engine configuration drawings and mockup. Routing considers flexibility, assembly, and maintenance requirements.

3. Preliminary analysis of the engine and cycle data determine plumbing deflections resulting from thermals, operating pressures, assembly, and manufacturing tolerance stackup.

4. Preliminary analysis of the engine/plumbing loop, by means of the computer techniques, establishes stress levels and intermediate deflections.

5. A recycle is performed if necessary until structural criteria is met (reroute line, relocate component and points, revise support brackets, etc.).

6. Flanges or couplings are positioned. Locations to impact minimal flange loads and to simplify assembly and maintenance are considered.

7. Analysis of engine/plumbing loop is performed to verify stress levels and deflections under all conditions.

8. Recyle if necessary to meet stress criteria, operational requirements, clearance, assembly, and maintenance requirements.

Flange designs are accomplished by application of the analytical methods defined in Plumbing Design Criteria, PWA FR-4455. The routine generates a practical low-weight cantilever, or raised face, flange that ensures the flange deflection at the seal contact point does not exceed a specified value under maximum bolt preloading at 120% of maximum operating pressure.

Any flange analysis must be preceded by a bolt load determination study. The various design rules presented in the Plumbing Design Critieria, PWA FR-4455, establish the minimum bolt preload value necessary at engine assembly. A 20% spread between minimum and maximum preload has been established and verified through test programs at FRDC.

The coupling criteria rules generate the following equation that mathematically describes the various bolt loads which constitute the total required minimum preload:

$$F_B = \frac{PA}{N} + F_T + F_{Rotation} + F_{Moment} + \frac{F_{Seal}}{N}$$

The terms in the equation are:

F_B = Minimum load per bolt required at assembly

P = Maximum engine cycle pressure

A = Area enclosed by seal contact diameter

F_{seal} = Total seal preload

N = Number of bolts in bolt circle

F_T = Load lost per bolt caused by temperature lag (if any)

$F_{Rotation}$ = Load reduction per bolt caused by flange rotation (as determined from flange deflection computer analyses)

F_{Moment} = Bolt preload required to provide external pipe moment capability.

In the above equation definitions, external moment is related to bolt preload by the basic equation:

$$F_{Moment} = \frac{2\,M}{R\,N}$$

where

R = bolt circle radius

N = number of bolts

M = external tube moment in lb-in., selected by overall considerations of tube section "moment of inertia", allowable tube bending loads, and tube routing limitations.

The tube wall thickness determination was accomplished by calculating a minimum allowable wall based on setting the maximum hoop stress at 0.2% yield stress at operating temperature and 120% of maximum operating pressure, unless other considerations (e.g., fracture mechanics) require the tube to be designed with more stress margin. This established a minimum wall thickness specification allowed in any tube bend. The nominal tube wall thickness was established by increasing this minimum wall value by a factor of up to 30%, to allow for thinning during fabrication if necessary. The remaining tube design criteria determines the material required for all sizes down to 0.375 in. outside diameter. Inconel 625, Inconel 718 and PWA FR-1053 welded and drawn tubing is used for these larger lines. For 0.3125 and smaller diameter tubing, AISI 347 (AMS 5512) material was used as seamless tubing stock.

All tube assemblies were analyzed to establish the end loads and moments imposed by end point deflections that occur because of component tolerances and strains caused by thermal and pressure effects.

The structural analysis involves the use of a computer program that was generated for use with pressurized tubes. This program is used to predict tube stress, reaction end loads, and moments at the end points. The program handles any multiple flowpath tube routing, including bends and interconnecting lines.

The primary fuel distribution system, as shown in figure V-6, is illustrated as a typical case. The engine/plumbing loop was analyzed to determine the maximum strain imposed at point "A". All deflections due to thermal, pressure, thrust, manufacturing tolerances and external loading, were considered. Spring rates for the various components and tubes composing the engine/plumbing loop as well as the deflections were then analyzed by means of computer to determine the resulting strain at any number of points within the loop, such as point "B". The strain difference between these points defines the operational and assembly deflections required for the plumbing segment between point "A" and "B" and provides the basis of design. Figure V-7 showing the preburner injector fuel supply line illustrates a typical plumbing segment which has been designed using this technique.

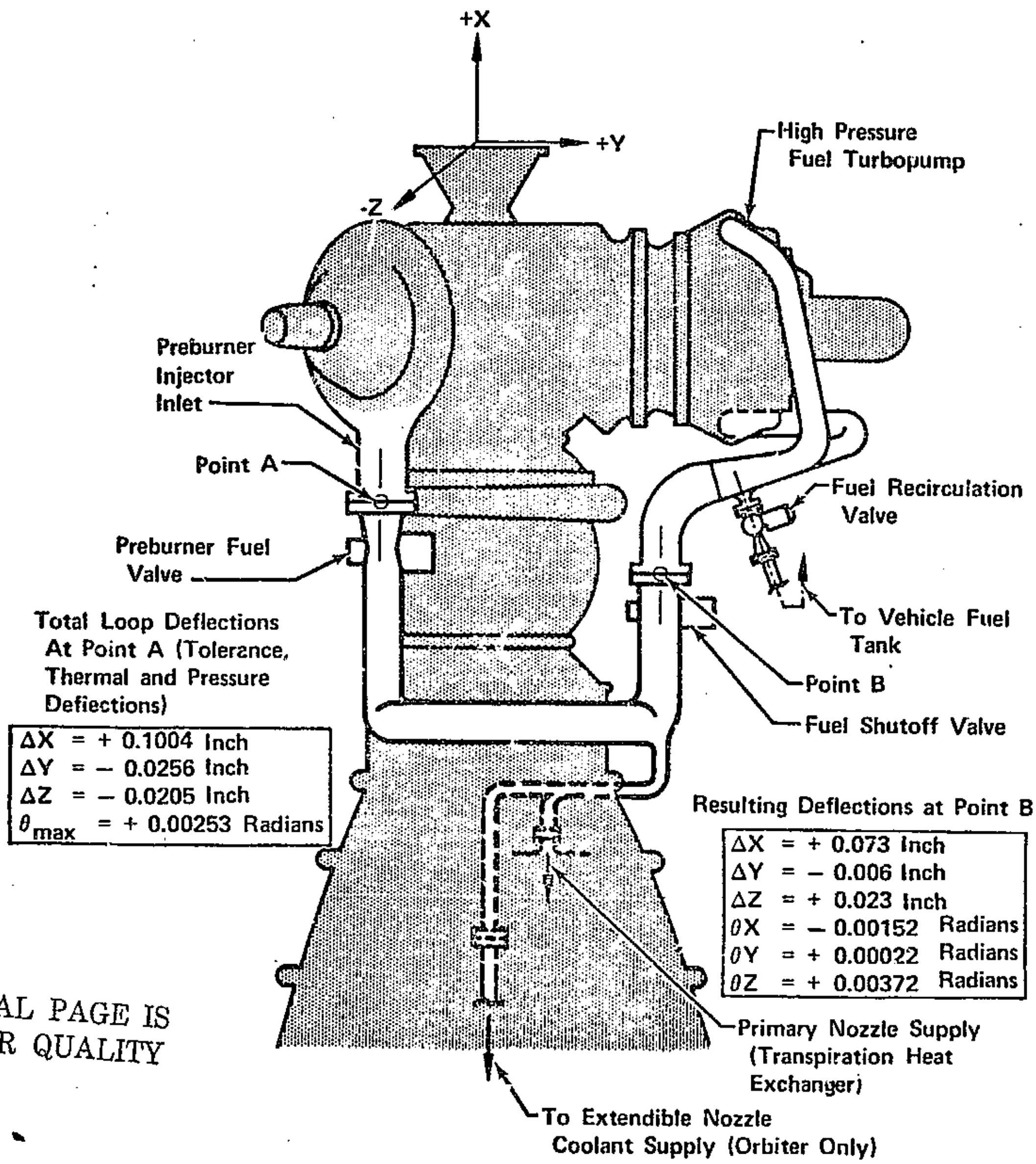
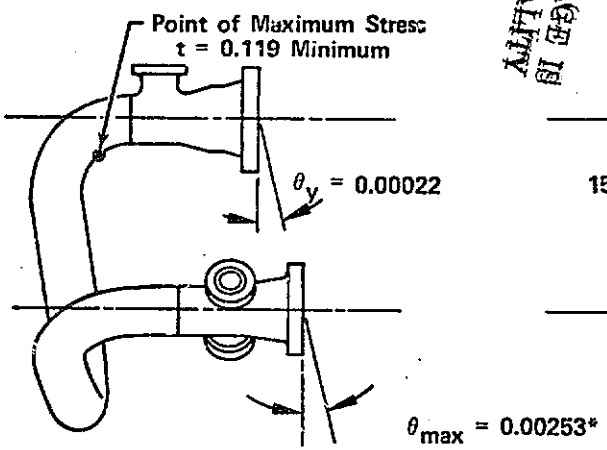
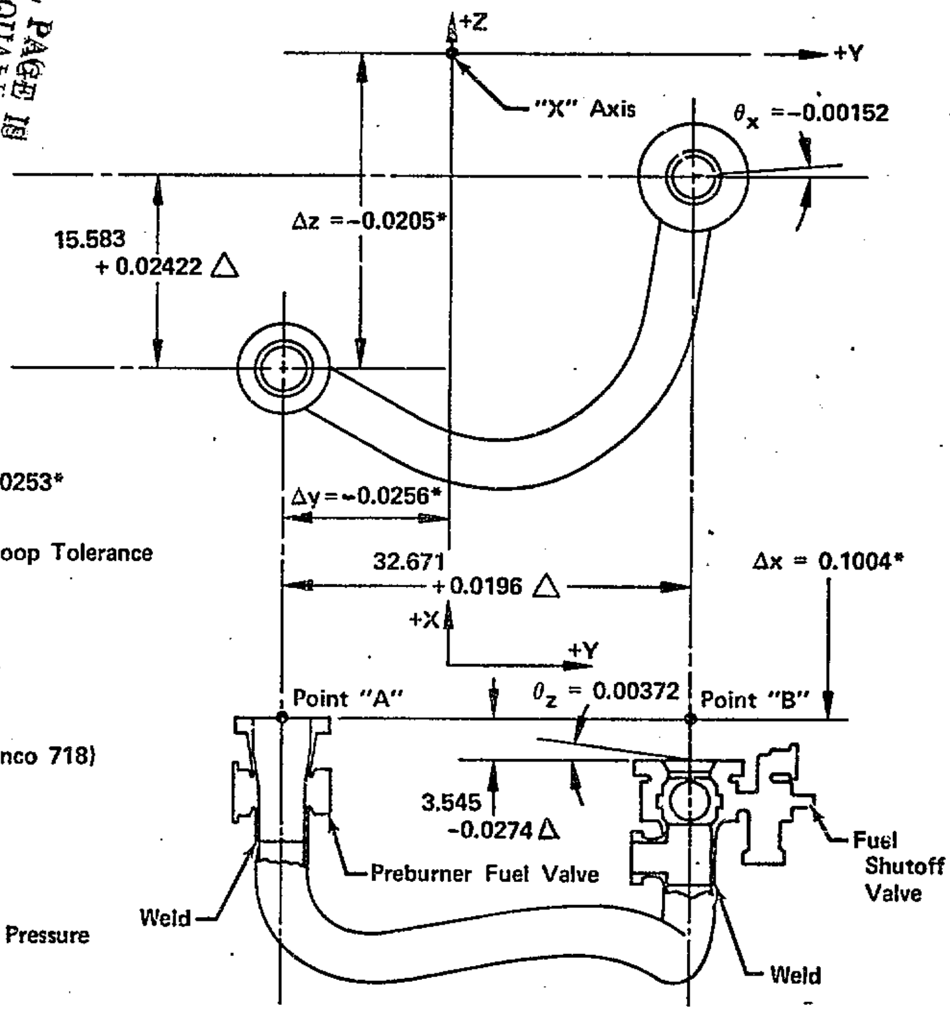


Figure V-6. Primary Fuel Distribution System FD 52323A





*Total Loop Tolerance

Type of Fluid Conducted	-	H_2
Limit Pressure	-	7260 psia
Design Point Flow Rate	-	157.19 lb/sec
Design Point Temperature	-	130.2°R
Line Inside Dia	-	3.69 in.
Material	-	PWA 1065 (Inco 718)
Nominal Wall Thickness	-	0.120 in.
Calculated Pressure Drop (Including Valves)	-	712 psid
Maximum Tube Hoop Stress	-	141,406 psi
Maximum Effective Stress	-	124,783 psi
Proof Pressure	-	1.34 x Limit Pressure

Figure V-7. Preburner Injector Fuel Supply Line

FD 50439

Tube routing configurations were established on the engine mockup, within specified envelope restraints in conjunction with interim analyses of initial designs. All plumbing lines are identified in accordance with MIL-STD-1247 with either tape or metal bands depending on environmental requirements for the particular line.

3. Design Requirements

All SSME plumbing fully meets the requirements of the CEI Specification. Sections of the CEI Specification CP2291 which deal specifically with the plumbing are paragraphs 3.7.6.2 and 3.7.12. Compliance with these paragraphs is outlined below:

1. The engine shall be permanently marked to indicate all connections shown on the installation drawing for instrumentation, propellant, and other fluid connections. All fluid lines shall be marked in accordance with MIL-STD-1247 and paragraph 3.7.6.2.

   Compliance - All plumbing lines and interfaces are identified and marked per MIL-STD-1247, using either tape or the metal band method as dictated by environmental considerations.

2. External or internal leakage of engine propellants or fluids shall not occur in such a manner as to impair or endanger proper function of the engine or vehicle. All welded, brazed, or hermetically sealed enclosures shall not exceed an indicated leakage of 1×10^{-6} scc/sec of helium at operating or leak check pressures. These requirements are met in accordance with paragraph 3.7.12.

   Compliance - All plumbing lines will utilize seals and connections as described in Section V.c. All plumbing will be X-ray inspected and pressure tested to ensure that the specified leakage rate of 1×10^{-6} scc/sec of helium at operating or leak check pressures is not exceeded.

3. In addition to the above paragraphs, the plumbing is designed to comply with all requirements of the specification pertaining to pressure vessels and engine subcomponents.

4. Design Capability

The plumbing line designs are based on providing minimum weight plumbing necessary to meet the CEI specification requirements. In the majority of cases, line size and wall thickness are a function of operational pressures and flow rates. In these cases, there is no capability over the design points of the SSME. In the case of small lines and some larger lines, the wall thickness is governed by vibration, assembly, or handling considerations, or stock tubing sizes available. In these instances there is a possibility of increasing operational conditions without plumbing revision. However, each case will require individual investigation. No estimate was made of overall capability of the plumbing in excess of present limits.

5. Design Substantiation

The basis for the plumbing design procedure used is a result of the experience gained from the XLR129-P-1 and RL10 programs as well as numerous turbine engine programs. The issuance of the Plumbing Design Criteria, PWA FR-4455, ensures consistency in design and adherence to a proven design concept. The use of specific design procedures provide the tools to incorporate the various plumbing lines into the engine as an integral part, rather than as a necessary pipe to get fluid from one place to another. The influence of the plumbing is considered at the initiation of the component design. Some of the factors considered are tolerance control to minimize line deflections, inlet and outlet locations, assembly requirements, structural requirements necessary to support line loads and to provide integrity at flanges. The use of a design procedure which allows the plumbing routing to be established and verified on a preliminary basis, early during the program, provides the component designer with accurate information as to the extend of influence the plumbing will have on the component design. This information will enable incorporation of these requirements into the design at a relatively early date. It also provides the means of recognizing and resolving "hard points" in the plumbing/engine integration prior to final component design when necessary changes become more difficult and compromises may result.

C. STATIC SEALS

1. Introduction

The static seals used on the SSME are one of the most critical areas of the engine design because of the possibility that explosive mixtures will collect in the engine compartment if there is excessive leakage at the propellant line connections. The number of flanges and seals are minimized to reduce the probability of leakage and avoid any unnecessary weight. The interconnecting lines of the SSME must serve to transport the various gases and fluids throughout the engine system at operating pressures as high as 6750 psia and temperatures ranging from 40°R to 700°R. Most of these lines will have one or more mechanical connections and must retain the integrity of the line throughout its operating life regardless of strains, loads, pressures, and thermal gradients, while maintaining the leakage rate as specified by CEI Specification No. CP2291.

Previous tests and studies made at P&WA during the XLR129 program indicate that none of the commercially available seals tested were capable of fully meeting the leakage requirements desired for the relatively large diameters encountered on the SSME. The toroidal segment seal, shown in figure V-8, developed by P&WA specifically for use in the XLR129 high-pressure rocket engine has been proved capable of meeting the specification requirement of 10^{-6} scc/sec leakage rate and is specified for use in all high pressure connections in excess of 1-in. in diameter. (Refer to SSME Related Data, PWA FR-4460.)

For static seals of 1-in. diameter and below, the Resistoflex Corporation's Dynatube seal is specified and is shown in figure V-9. This seal has been tested extensively at pressures up to 5000 psia and at temperatures of 140°R to 960°R with a leakage rate of $<10^{-7}$ scc/sec, being maintained through

500 assembly cycles. (Refer to Plumbing Design Criteria, PWA FR-4455.) The Dynatube seal offers an integral, reusable seal which is ideal for use in smaller size lines where frequent disassembly is a possibility. For these reasons, it was selected for use on the SSME. It is P&WA's intent to work with the Resistoflex Corporation to verify the capability of this seal during the development phase of the engine. In the event there are unpredictable problems with the Dynatube seals, P&WA is prepared to make use of the Battelle "Bobbin" seal, shown in figure V-10. This seal has proved capability for meeting all requirements of the SSME, however, it will require sacrifice in terms of weight, cost, and maintainability.

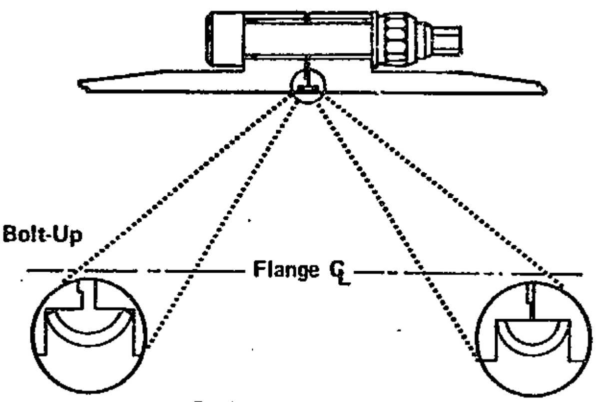
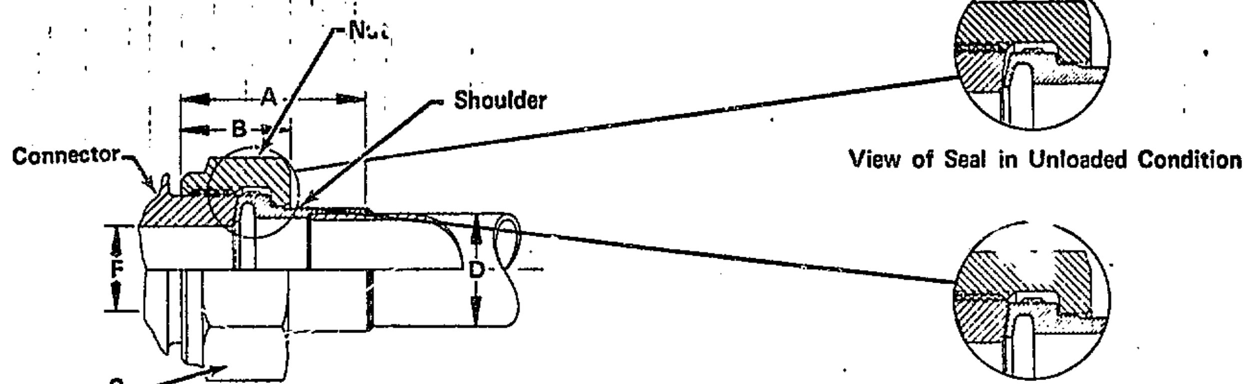


Figure V-8. Toroidal Segment Detail Part, Fit and Installed Shape FD 46260

The type of separable flange used is the raised face flange. The design requirement is to provide a flange of minimum weight necessary to maintain the integrity of the engine and to meet the leakage rate criteria of 10^{-4} scc/sec at the seal. The flange is designed to minimize flange deflection at the seal and thereby eliminate any relative motion between the seal and the seal gland. Here again, the selection is based upon development and test programs conducted at FRDC in conjunction with the XLR129 program to establish seal and flange technology capable of meeting the leakage rate requirements necessary for successful manufacture of high pressure rocket engines. This technology is directly applicable to the SSME.

2. Description

The toroidal segment seal, as previously shown in figure V-8, is used in all applications over 1-in. in diameter. It is a result of extensive seal testing conducted for the Air Force Rocket Propulsion Laboratory during the XLR129-P-1 program.



Size	A	B	C	D	F
-03	0.680	0.420	7/16	3/16	0.159
-04	0.709	0.454	9/16	1/4	0.189
-05	0.714	0.459	5/8	5/16	0.237
-06	0.733	0.518	11/16	3/8	0.302
-08	0.745	0.510	7/8	1/2	0.403
-10	0.906	0.571	1	5/8	0.518
-12	0.942	0.637	1 1/8	3/4	0.654
-16	1.427	0.705	1 3/8	1	0.868

View of Seal in Loaded Condition Showing:

1. Self-Energizing Metal to Metal Seal
2. Hydraulic Boost in Seal. (Seal Efficiency Increases With Higher Pressure.)
3. Tightening Produces Structural Forces Only.
4. Loaded Seal Provides Self-Locking Action.

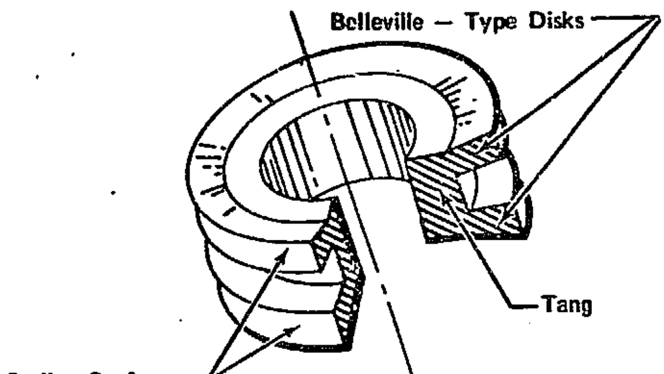
Nominal Installation Torque Values

Size	-03	-04	-05	-06	-08	-10	-12	-16
ft-lb	7	12	13	20	35	48	60	82
in.-lb	84	144	156	240	420	576	720	984

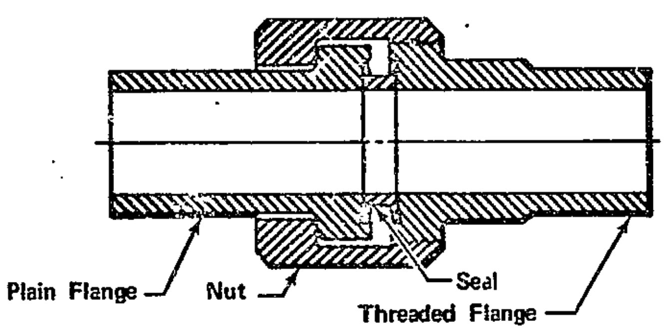
Figure V-9. Dynatube Fitting

FD 52559

The toroidal segment seal configuration provides reasonably consistent performance throughout the entire 0 to 7000 psig pressure range. This seal utilizes axial compression to produce the radial deflection and high unit loads necessary for effective sealing at low operating pressures. Sealing at high pressure is effected by the additional sealing forces produced by the ΔP across the seal. The seal groove depth, cross section size, and thickness are varied as required to meet the load requirements over a large range of coupling diameters. Complete seal size information is shown in figure V-11.



b. Construction of Bobbin Seal



a. Connector Assembly

Figure V-10. Connector and Bobbin Seal FD 34605

Testing during the development phase of the SSME will be used to confirm the seal capability at specific locations. Slight modification to seal cross section or groove design may be necessary, depending on environmental factors at the individual application; however, this is easily accomplished due to the versatile design of the seal.

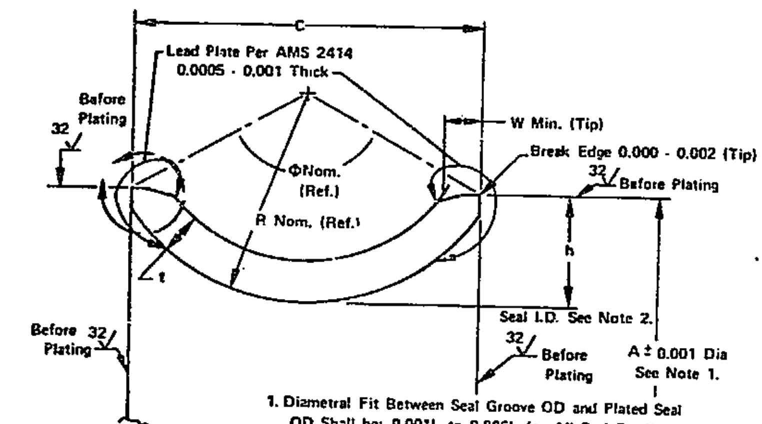
Primary sealing is accomplished by a radial sealing force developed at installation, as shown in figure V-12. By keeping the radial contact width of the sealing legs equal to 0.010 in. or less, a minimum value of radial sealing stress can be set at 2000 psi. This value of radial stress is sufficient to yield the lead plating on the seal. The addition of pressure increases this force. Therefore, seal pressure increases with increasing internal pressure and the integrity of the seal is maintained.

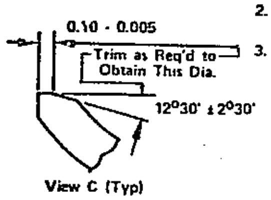
As the seal is installed, it is axially compressed by an amount which causes a radial growth of the seal, ΔR_O. If ΔR_O is greater than the radial clearance between the seal and the seal groove, then the seal is in effect subjected to hoop compression by the groove wall, as shown in figure V-12.

The radial deflection of each point along the seal cross section is plotted for various values of free seal angle, σ. Since the seal forms a closed ring, the deflection plots are equivalent to hoop stress plots across the seal cross section. To maintain equilibrium, the hoop compression must equal the hoop tension on the seal cross section. The plots are thus used to relate the radial deflection ΔR_o to ΔV.

Using this relation, radial sealing force can be expressed as:

$$H = \frac{ER\phi t}{R_o^2} \left[\Delta V \frac{\Delta F(\phi)}{F_2(\phi)} - \text{clearance} \right]$$





1. Diametral Fit Between Seal Groove OD and Plated Seal OD Shall be: 0.001L to 0.006L for All Seal Configurations. These Fits Are Prior to Ass'y.
2. Fits Between Pilot OD and Seal ID for Seal Configuration 9 Shall Be 0.002L to 0.014L Prior to Ass'y.
3. Seal Dia. Given are Prior to Plating Unless Otherwise Noted.

Seal Dimensions							
Seal Config. No.	A Dia Min and Max	C	h	t	ΦNom (Ref.)	R Nom (Ref.)	W Min.
1	1.000 to 2.000	0.107 0.109	0.036 [illegible]	0.011 0.013	124°	0.064	0.010
2	2.000 to 3.000	0.107 0.109	[illegible] [illegible]	0.018 0.021	120°12'	0.067	0.019
3	3.000 to 4.250	0.160 0.164	0.056 0.057	0.023 0.026	126°30'	0.096	0.053
4	4.250 to 8.500	0.211 0.217	0.074 0.070	0.023 0.026	130°36'	0.122	0.023
5	8.500 to 14.500	0.265 0.265	0.090 0.090	0.033 0.039	125°12'	0.158	0.033
6	14.500 to 16.500	0.265 0.271	0.090 0.084	0.039 0.044	123°	0.162	0.039
7	16.500 to 18.000	0.265 0.271	0.040 0.084	0.046 0.051	120°36'	0.166	0.046
8	18.000 to 19.000	0.267 0.271	0.090 0.084	0.046 0.051	120°12'	0.166	0.046
9	Over 19.000	0.211 0.217	0.074 0.070	0.016 0.019	131°48'	0.120	0.016

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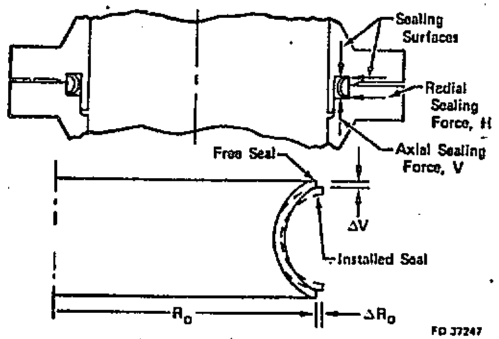
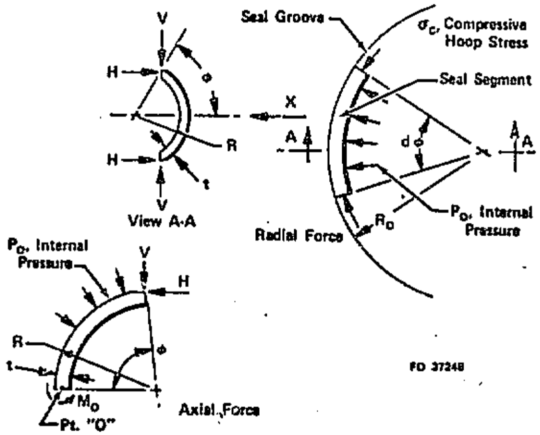
Figure V-11. Toroidal Segment Static Seal

FD 49146



Sealing Surfaces On Typical Installation



Radial and Axial Force Computation

Figure V-12. Determination of Sealing Forces

(1) Radial Sealing Force

Summing forces in the x direction:

$$P_oR_oR \sin \phi \, d\theta + \sigma_c R\phi t \frac{2d\theta}{2} = HR_o d\theta$$

Let $\sigma_c = E \dfrac{\Delta R_o'}{R_o}$ and $\Delta R_o' = (\Delta R_o - \text{clearance})$

$$\therefore H = P_oR \sin \phi + E \frac{\Delta R_o'}{R_o^2} R\phi t$$

at installation $P_o = 0$

$$\therefore H = E \frac{\Delta R_o'}{R_o^2} R\phi t$$

(2) Axial Force

$$\sum M_o = 0$$

$$VR(1 - \cos \phi) + P_oR^2 (1 - \cos \phi) = HR \sin \phi + M_o$$

$$\therefore V = \frac{H \sin \phi + M_o/R}{1 - \cos \phi} - P_oR$$

Definition of terms

$$P_oR_oR \sin \phi \, d\theta + \sigma_c R\phi t \frac{2d\theta}{2} = HR_o d\theta$$

$$\sigma_c = E \frac{\Delta R_o'}{R_o}$$

R_o = radius to radial sealing face, in.

P_o = internal pressure, psi

R = seal cross section radius, in.

ϕ = half segment angle of seal cross section

$d\theta$ = seal segment

σ_c = hoop stress in seal, lb/in.^{2}

t = seal thickness, in.

H = radial force per inch on seal leg, lb/in.

E = modulus of elasticity, lb/in.^{2}

$\Delta R_o' = (\Delta R_o - \text{clearance})$

Clearance = initial radial clearance between seal and groove prior to installation, in.

ΔR_o = unrestrained radial growth of seal due to ΔV, in.

$VR(1 - \cos \phi) + P_oR^2 (1 - \cos \phi) = HR \sin \phi + M_o$

V = axial force on seal leg, lb/in.

M_o = distributed moment at point 0, in.-lb/in.

ΔV = axial deflection of seal - in.

FD 52424

The Dynatube seal, which has been selected for use for static seal applications of 1-in. and under, is shown in its various configurations in figure V-13. It provides an integral, reusable seal which is required for small lines such as instrumentation or control lines where more frequent disassembly may be encountered.

The operation of the Dynatube seal is illustrated in figure V-13. Initial sealing force is provided by the controlled deflection of the seal lip engaging the seal face on the mating component. Additional sealing force is then a function of internally applied pressure of the fluid. The initial sealing force, developed by deflecting the lip, and the intimate contact resulting between the deflected lip and the conical seal seat, provides the sealing capability necessary to maintain the low leakage requirement at the lower pressure values.

The increase of internal line pressure produces an additional pressure/area force behind the seal, due to the undercut section, and thereby provides additional seal load between the seal lip and the mating face. The result is a seal which is capable of maintaining its low leakage rate requirement throughout the operating pressure spectrum experienced in the SSME.

The fact that the seal is an integral part of the coupling will ensure that the seal is not omitted or misassembled. Also, the reusability of the seal will allow reduction in maintenance time and cost by reducing the number of replacement seals required.

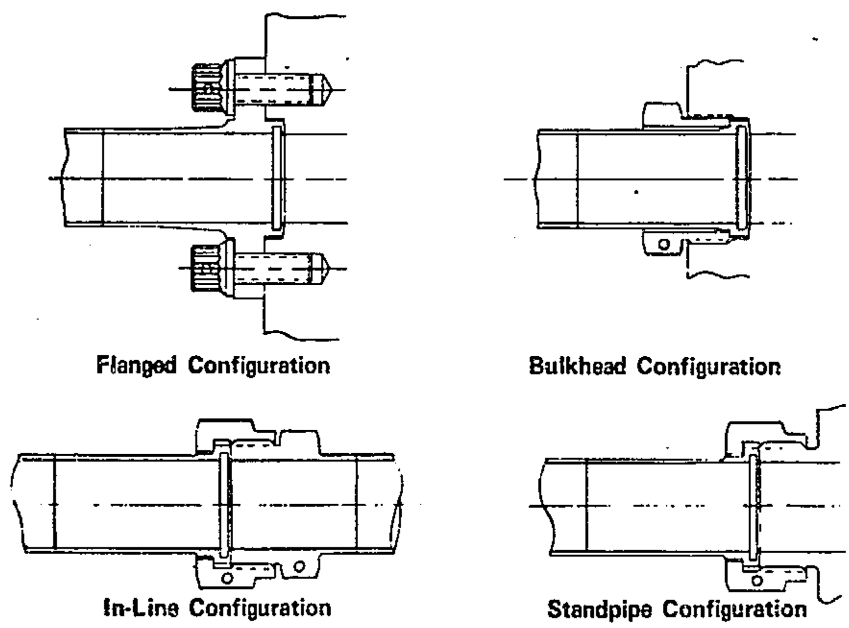


Figure V-13. Dynatube Seals FD 46261

All conventional straight line connections will employ the raised-face (cantilever) type flange shown in figure V-14. Basic design of the raised-face flange requires that the distance between the static seal and the bearing face of the flange be held to a minimum. Bolt loads and blowoff loads are located as close as possible to reduce the overturning moment in the flange ring, which produces bending of the flange as shown in figure V-15. This ensures that the deflection of the flange in the area of the seal surface will be small as the flange load is applied and the flange is deflected. The flange itself is designed to be relatively stiff in resistance to an overturning moment. In designing the flange, the radial thickness is held to a minimum by employing a large number of the smallest diameter bolts possible, to reduce the overturning moment in the flange and to provide sufficient preload to prevent separation of the flange under applied operational loads. The flange thickness is then computed, with the aid of a finite element computer analysis, to limit the rotational deflection at the seal contact point to a specific value as required for the particular application.

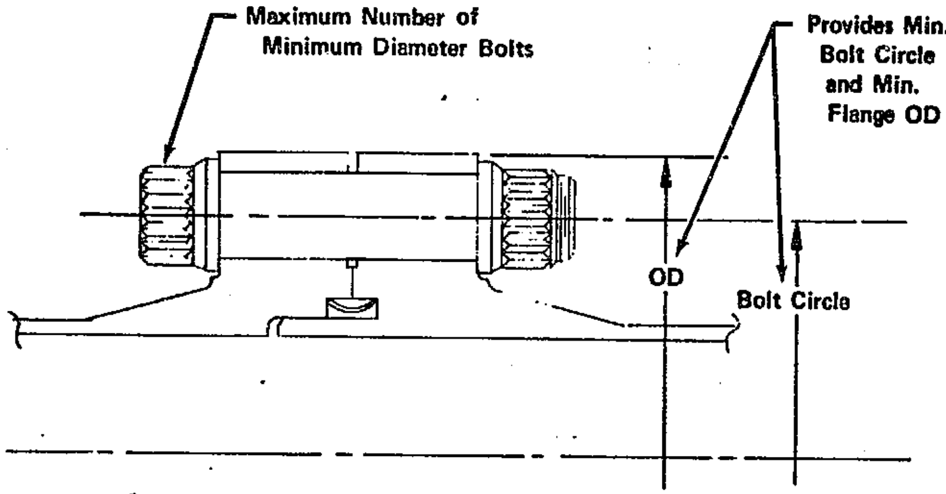


Figure V-14. Raised Face Flange Provides Seal Integrity With Minimum Weight FD 52443

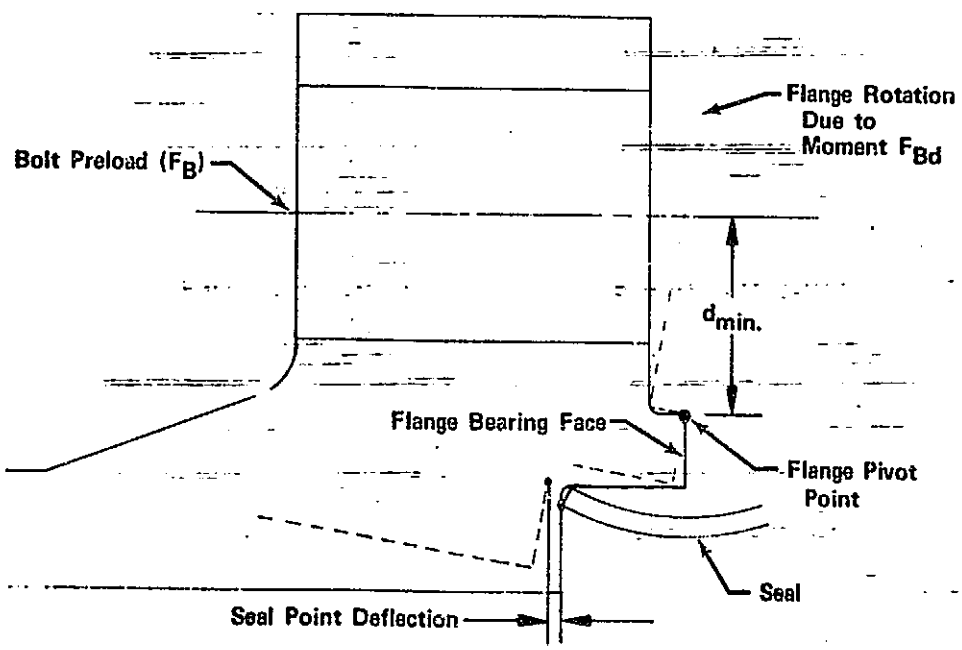


Figure V-15. Sealing Surface Deflection Minimized by Limiting Rotational Effect of Flange Loading FD 46300

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By minimizing the radial thickness of the seal, bearing surface, and bolt diameter stackup, the overturning moment in the flange is minimized and the flange material is utilized to its fullest advantage. The result is to minimize both the weight and size of the flange required to meet the requirements.

To ensure conformity and consistency in all components, Plumbing Design Criteria, PWA FR-4455, has been published and distributed to all designers. This design manual completely describes the step-by-step procedure to be followed to ensure a proper flange design of minimum weight. The design manual is based on the use of a minimum flange sized for a specific deflection at the sealing point, under the combined loads resulting from pressure, thermal gradient, bolt preloads, together with other external loads from the assembly and operating conditions, and utilizing the finite element computer analysis techniques which have been proven during the previous XLR129 program.

3. Design Requirements

Requirements of CEI Specification No. CP2291 which pertain particularly to the seals and separable connections are specified in paragraphs 3.7.12 and 3.7.13.1.

1. Seal leakage rate as specified in paragraph 3.7.12 shall not exceed 1 x 10^{-4} scc/sec of helium at operating or leak check pressure. P&WA is complying with this specification by providing, at all separable connections, static seals and connections which have proven capability of maintaining leakage below specified rates. Seal selection was based upon prior testing which indicates compliance and additional testing will continue throughout the engine development phase to assure leakage rate within the specification requirement. In conjunction with this leakage requirement, P&WA has enlisted the services of J. L. Pearce and Associates; Dynamatec Corporation of Cocoa Beach, Fla., to investigate the development of an automatic ultrasonic leak detection system for the SSME. Dynamatec has supplied technical information and cost estimates for continuing the investigation with hardware development in Phase C/D (Plumbing Design Criteria, PWA FR-4455).

2. Physical requirements described in paragraph 3.7.13.1 to be followed for design of flanges, and provide that threaded type connections may only be used where specific approval is given by the procuring agency. The flange design criteria are as follows:

    a. Minimum bolt circle diameter - Compliance is mandatory in the P&WA design and is included in Plumbing Design Criteria, PWA FR-4455, to ensure minimum bolt circle diameters.

    b. Standard strength/type/lubricated bolts for applications where weight savings do not justify the use of high strength fasteners. High-strength bolts with special lubrication are used where weight savings result. Special provisions may be made for bolt loading to close tolerances. P&WA design requires use of high-strength bolts where weight savings are accomplished as a result. In areas where no weight advantage exists, standard strength bolts are used.

c. Through-bolt holes, except where significant weight savings or design considerations justify the use of studs and/or blind tapped holes. P&WA design requires the use of through-bolts wherever possible. Where weight or design considerations require studs, the studs are equipped with torquing provision to prevent stud rotation.

d. Utilize identical bolt diameters and lengths for adjacent flanges. P&WA design requires the use of identical bolts, in size, length, and material, for all adjacent connections in order to prevent misassembly. In areas where adjacent flanges occur with varying environmental requirements which would preclude the use of bolts with identical materials, the bolt diameters are varied to prevent possible misassembly.

e. Utilize standard washers that do not require unidirectional installation. Washers requiring directional installation are not used.

f. Standard locknuts, high strength nuts with special lubrication may be used where weight savings result. P&WA utilizes high strength nuts as required to meet the conditions governed by bolt size and material selection as stated in subsections (b) and (d).

g. Utilize nonlocking inserts if through-holes are not possible. P&WA has incorporated the use of nonlocking helical inserts for all threaded bolt holes.

h. Seal-joint tolerance stackup layouts shall be utilized. P&WA design procedure for analysis of all plumbing lines requires the definition of all end point deviation due to thermal and pressure effects as well as mechanical tolerance. Plumbing lines are analyzed on the basis of multibranch distribution systems where necessary in order to establish the effect of interaction. All deflections and tolerance requirements are identified on layouts of individual plumbing lines.

P&WA has complied with all aspects of these specifications by incorporating them as requirements in the design procedure where possible as outlined above. By familiarization of the Design Group with the specification and by redundant checking procedures required prior to final layout approval, compliance is assured. In addition, P&WA has submitted for the NASA approval, a list of applications for threaded connections to be used, with substantiation data for the type selected. In the event the use of these threaded connections are not approved, the connections will be revised to accommodate flange connections as specified.

4. Design Capability

The static seals used on the SSME engine, both the toroidal-segment type and the Dynatube type, are sized primarily on the basis of diameter. Variations in operating pressures at an individual seal will require no revision unless the pressure requirements are beyond the present maximum requirement of the SSME.

The sealing capability of the toroidal-segment seal, as demonstrated by tests at FRDC, is sufficient to meet the goal of $<10^{-6}$ scc/sec of propellant gas at the operating or leak check pressures. Test reports from the manufacturer and various other agencies indicate that the Dynatube sealing capability is better than the goal by a wide margin (10^{-8} scc/sec helium at 5000 psia).

The Dynatube seal is reusable without degradation to its sealing capability. Up to 500 assembly operations with an individual seal have been demonstrated. The seal face for the Dynatube is easily maintained and may be refinished with hand tools. Refinishing without component removal may be possible but will depend upon specific determination.

The flanged connectors used on the SSME are designed to meet the specific requirements for an individual connection. This design is based on providing minimum weight of the assembly and therefore precludes the use of the individual connections at conditions more severe than those considered. The design is concerned primarily in maintaining the leakage below the specified rate by minimizing the deflection of the flange in the area of the seal. If conditions of pressure, temperature, or external loads at the flange are increased such as to increase the flange deflection at the seal, the leakage rate would be affected and flange redesign would be necessary to limit leakage to the presently specified rate.

The threaded connections, when used in conjunction with the Dynatube seal, have demonstrated a capability of wide latitude in torque variation while maintaining satisfactory leakage rates and structural integrity. Typical torque requirements indicate variation of ±50% of nominal specified torque will provide satisfactory operation of the seal and connector.

Seal drains will be incorporated into the individual connections as required. Typical drains are illustrated in figure V-16. Requirements for taps or drains at connections, for purposes of leakage monitoring on nonflight engines, will be accomplished by the addition of appropriate equipment for the particular connection or flange being monitored. Flight engine hardware will include ultrasonic leak detection monitoring capability if the feasibility of the system is demonstrated during engine development.

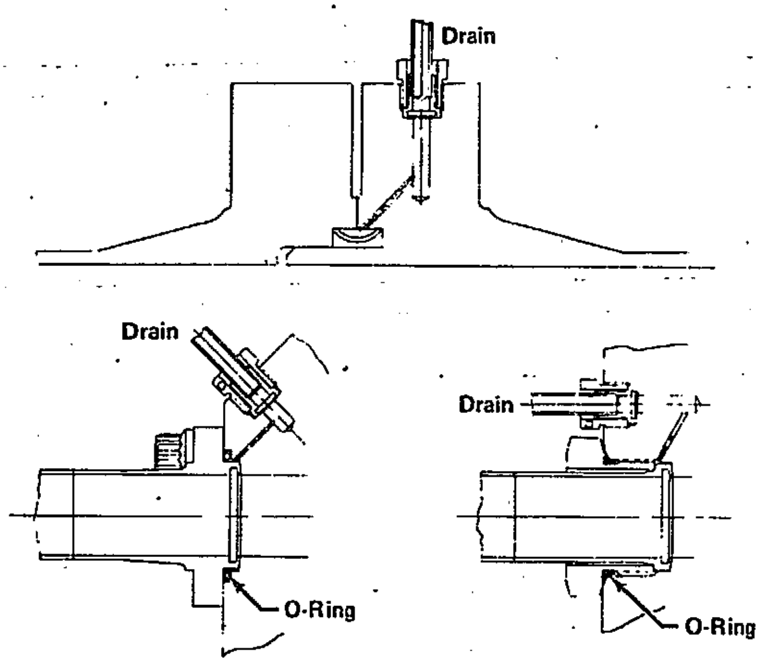

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Figure V-16. Typical Drain Configurations for Flight Hardware FD 46298

5. Design Substantiation

a. Flanges

During the XLR129-P-1 demonstrator engine program, extensive high pressure coupling rig design analyses were conducted. Conventional analysis methods proved inadequate for lightweight, high pressure coupling deflection and weight optimization predictions. Specialized computer analysis programs were developed to assist the designer. A hydrostatic coupling rig was designed and tested for stress and deflection at XLR129 operating pressures. A finite element disk analysis computer program was then modified to provide deflection prediction correlation with the test results, as shown in figures V-17 and V-18. This program was used to investigate bolted-flange coupling and static seal designs suitable for use in the XLR129-P-1 engine. (Refer to Plumbing Design Criteria, PWA FR-4455.)

The variables affecting flange weight for any given seal deflection are bolt circle diameter, flange thickness, bolt load, seal point diameter, taper height, taper length, and web configuration between bolt holes. Flange thickness, bolt circle diameter, and required bolt load have the greatest influence on flange weight and the study concentrated on these factors. All the factors except flange thickness and total bolt load were held constant during the analysis.

The minimum bolt-circle diameter was determined by the minimum requirement for wrench clearance and was used for all but the loose-ring design where the bolt circle was limited by other geometric considerations, such as minimum bearing surface and a taper height consistent with other flanges. A bolt load of 15,000 lb/bolt was used throughout the analysis while the maximum permissible bolt load was 17,750 lb/bolt using 15, 1/2-20 UNJF, Inconel 718 bolts. The difference was used as an allowance for bolt bending.

Raised face and simple beam flange seal loading schemes were then analyzed using the computer program. This analysis indicated that the raised-face design was lighter than the simple beam designs for the same deflection limit.

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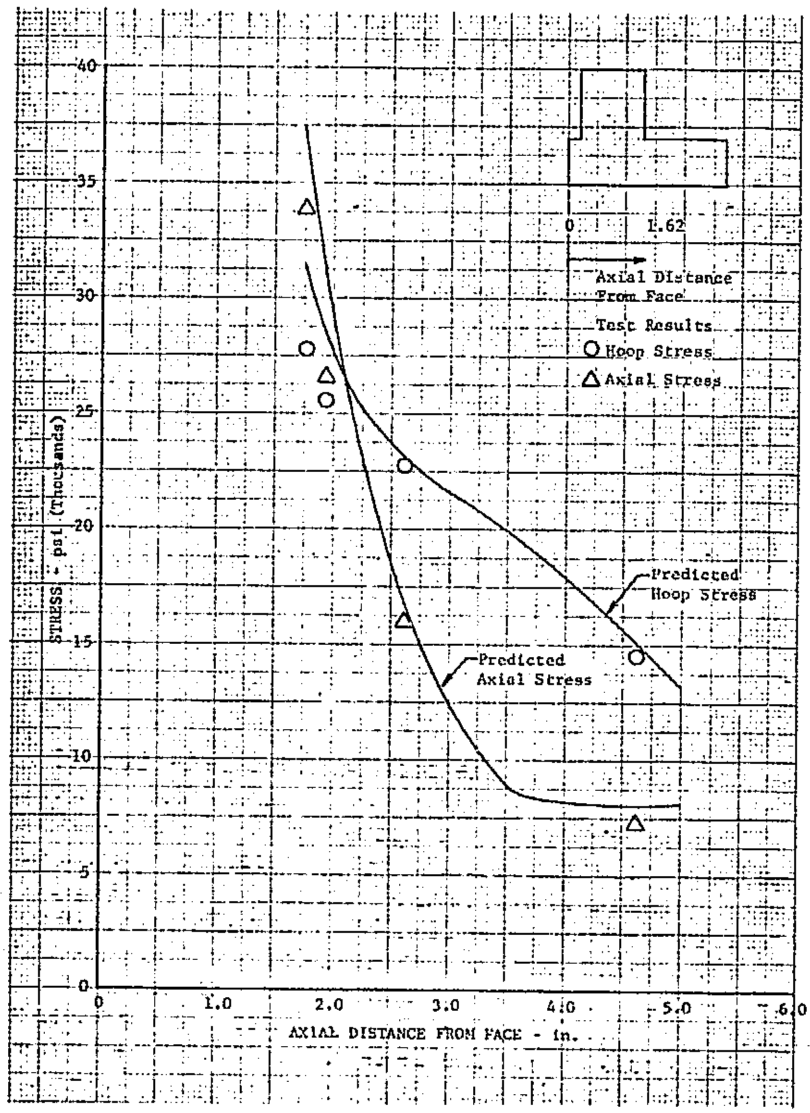

Figure V-17. Comparison of Predicted and Measured Stresses on OD Wall, Rig 35120-3 DF 68869

A weight and size comparison was performed for four couplings having a total axial deflection of 0.002 in. at the seal point, three of which are shown in figure V-19. The raised-face flange proved to be lightest while the undercut flange proved heaviest. The flat-face flange was eliminated because it had no advantage over the undercut flange. The distance from the theoretical pivot point to the sealing point was found to be a major influence on flange size because very little bending occurs in the flange.

Deflection tests conducted during the seal pressure cycle endurance and leakage test program confirmed the validity of the finite element computer analysis technique. The computer program model was kept current during the test series. This provided updated stress and deflection analysis capability for the demonstrator engine flanges, the results of which are shown in figures V-17 and V-18.

The flange configuration selection was based on the results obtained from the program described above and from subsequent component rig tests during the XLR129 program, including a full size hot firing of the preburner and fuel pump in a powerhead configuration. The raised face (cantilever) flange is used in all applications for high pressure plumbing flanges on the SSME.

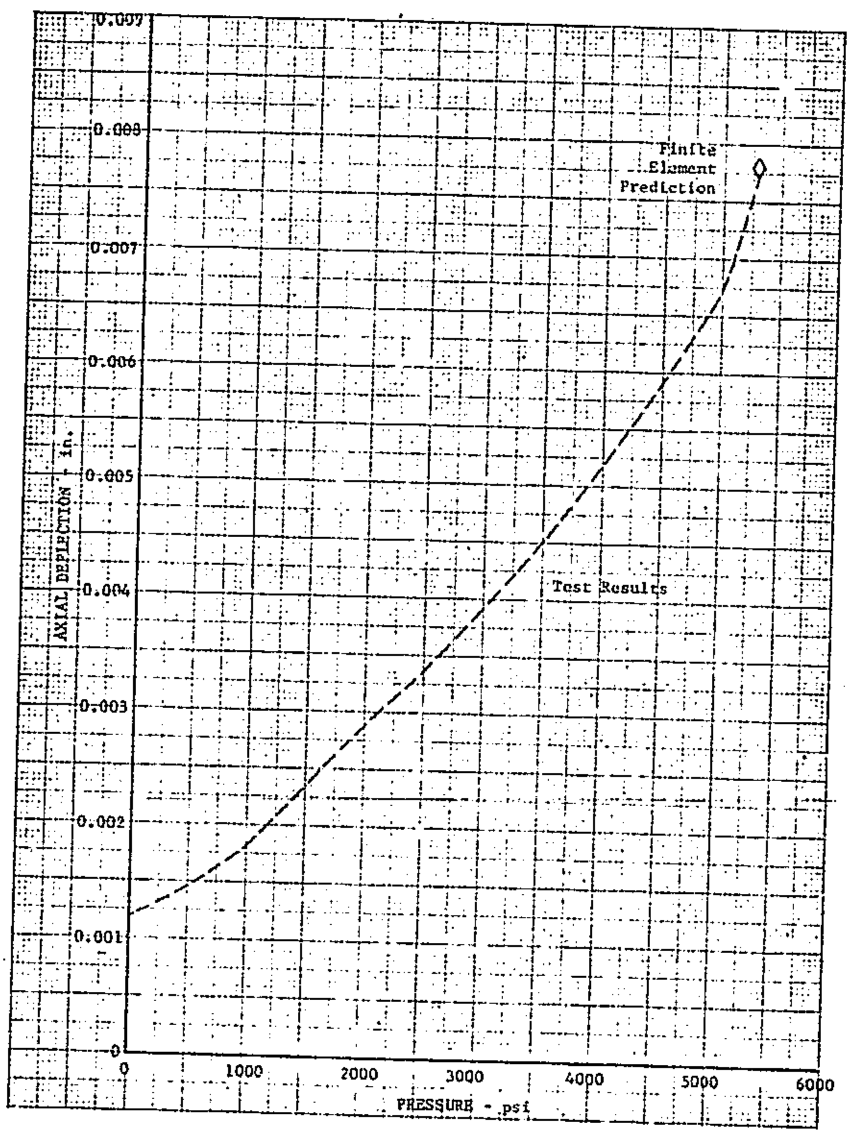

Figure V-18. Axial Deflection at ID of Flange, Rig 35120-3

DF 68870

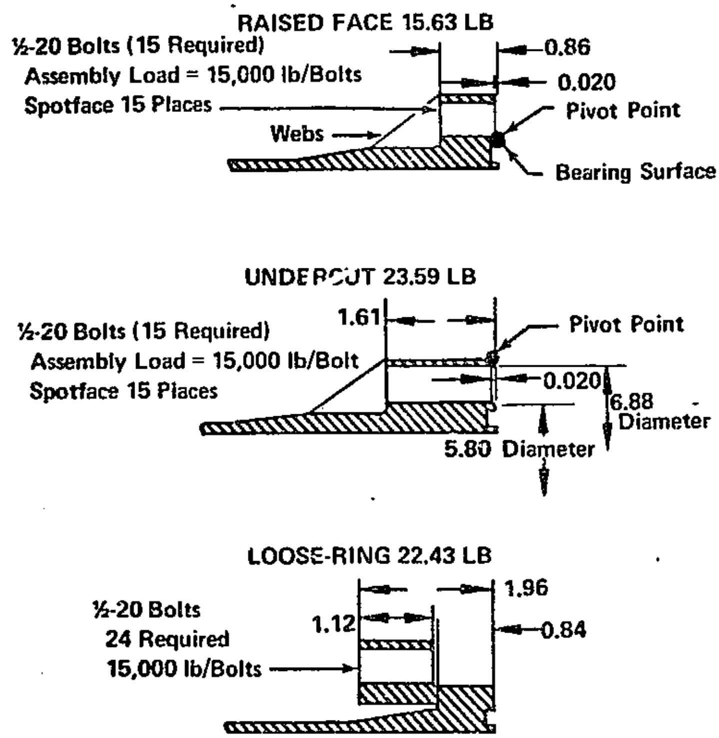

Figure V-19. Static Seal Rig 0.002 inch Deflection Flanges FD 46259

b. Seals

During seal rig, component, and staged combustion rig testing conducted in the Phase I portion of the XLR129-P-1 program (XLR129 Final Report, PWA FR-4460), excessive overboard static seal leakage was experienced. Dual static seals were incorporated into the rig couplings. The measured primary static seal leakage at maximum thrust during the staged combustion rig test firings was equivalent to an impulse loss of approximately 2 sec, and additional uncontrolled overboard leakage was visible.

Static pressure tests with the main chamber oxidizer valve indicated the leakage problem was aggravated by inadequate static seals.

A 5-in. seal test rig was designed using the finite element computer program. It was to have 0.002 in. (total) deflection at the sealing diameter with 7000 psig internal pressure at an LN_2 temperature of 140°R. Maximum acceptable leakage rate was set at 10^{-4} scc/sec per inch of seal circumference of gaseous nitrogen, as required by the XLR129 specification. Deflection tests proved the program good.

Eighteen cryogenic seals were investigated, the results of which are shown in table V-2. Many were rejected on the basis of excessive size and other obvious deficiencies. Six commercial cryogenic seals with claimed "zero leakage" and 0.002 in. deflection capability were selected for test in the rig.

Two hydrostatic pressure/deflection and 29 cryogenic pressure cycle endurance tests were completed with the seal test rig. Nineteen configurations of eight basic seal designs were tested, including the particular configuration recommended for the inclusion in the SSME demonstrator engine design.

Table V-2. Applicable Commercial Seals

Size Category	Common Name	Manufacturer	Cross Section	Gland Dimension Width (in.)	Gland Dimension Depth (in.)	Seal Material	Vendor Recommended Seal Coating	Maximum Axial Deflection Capability	Remarks
Low Deflection Capability (Group A)	"V" Seal	Parker Seal Co.	P	0.166	0.1[illegible]0	Inconel X-750	Teflon TFE	0.002	
	Metal "O" Ring	United Aircraft Products Co.	P	0.154	0.08[illegible]	AISI 321	Teflon TFE	0.002	
	"C" Ring	Pressure Science, Inc.	P	0.119	0.[illegible]16	Inconel X-750	Indium	0.002	
	Bi-Metal Gasket	Del Mfg. Co.	P	0.096	0.100	Inconel X-750	Teflon TFE	0.002	
	Apex Seal	Servotronics, Inc.	P	0.170	0.106	Inconel X-750	Lead	0.002	
	Omega Seal	Servotronics, Inc.	P	0.170	0.106	Inconel X-750	Lead	0.002	
Intermediate Deflection Capability (Group B)	"E" Ring	Pressure Science, Inc.	P	0.127	0.120	Inconel X-750	Indium	0.006	Has high axial deflection capability for small profile seal
	K-Seal	Harrison Mfg.	P	0.235	0.125	Inconel X-750	Teflon TFE or Nickel-Lead	0.010	
	Maflex Seal	Navan Inc.	P	0.293	0.175	Inconel 718	Teflon TFE	0.003	
	Dryerseal	W. S. Shamban & Co.	P	0.488	0.230	Inconel or SST	None - Is Teflon Filled	NA	Maximum operating temperature is 500°F
	Spring Gasket	Donaldson Co., Inc.	P	0.194	0.107	Inconel X	Silver	NA	
	Servoflex	The DSD Company	P	NA	NA	Inconel X	None	NA	Low axial deflection capability slender profile
	Lo-Load Seal	The Advanced Products Co.	P	0.169	0.088	Inconel 718	Indium Over Lead	0.005	Low installation load required - 25 lb/in.
Radial Seals (Group C)	Bobbin Seal	Battelle Memorial Institute	P	NA	NA	Inconel 718	None	Per Seal Design	
	Radial "C" Ring	Pressure Science, Inc.		0.127	0.125	AISI 304	Indium	Per Flange Geometry	0.002 maximum radial deflection
	Conoseal	Aeroquip Corp.		0.462	0.335	AISI 321	None	Per Flange Geometry	Vendor quotes 0.040 maximum axial deflection
	Bal-Seal	Bal-Seal Engr. Co.		0.127	0.197	Teflon TFE	None	Per Flange Geometry	8 rms gland finish required, maximum temperature = 250°F
	Muco Seal	National Utilities Corp.		NA	NA	A-286	NA	Per Flange Geometry	Information requested from vendor - not received 10-10-68

NA - Not Available

On the basis of the seal rig testing described above and the subsequent successful use of the toroidal segment seal in component and staged combustion tests during the XLR129-P-1 program, this seal has been incorporated into the design of the SSME.

Based on the numerous tests by the manufacturer as well as other independent agencies, including the NASA-MSFC, North American Rockwell Corporation, and the Naval Air Engineering Center, the Dynatube seal has been selected for use in applications up to 1-in. in diameter. These tests have indicated the leakage rates of 10^{-6} sccs of He, the environmental considerations of cryogenic and gaseous propellants, and the mechanical integrity and other operational requirements of the SSME, can be met and surpassed by this seal. In addition, the reusability and the integral feature of the seal provide definite advantages in the areas of cost reduction and maintainability.