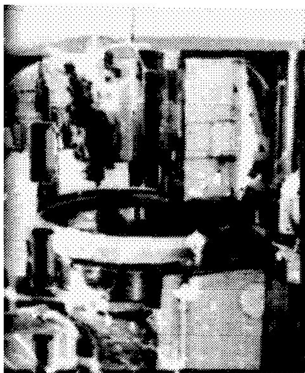

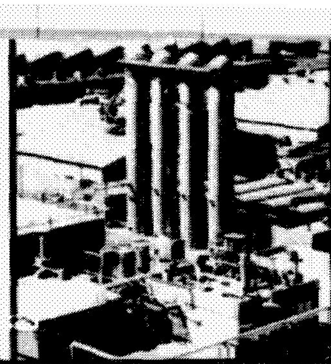

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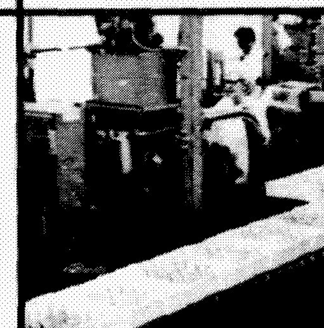

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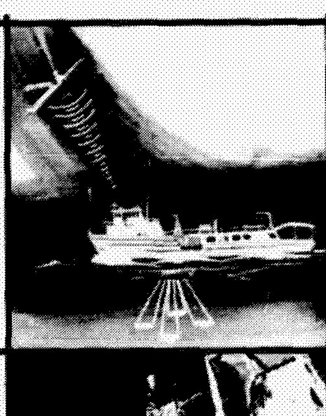

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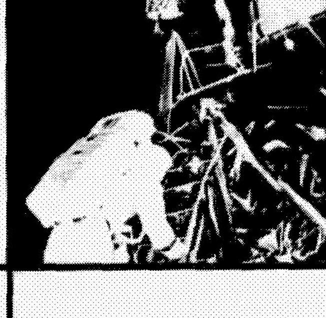

THE Marquardt COMPANY VAN NUYS, CALIFORNIA

Contract NAS9-12992
DRLT 776 Item 9
DRD MA-129TB

Report No. S 1296

SPACE SHUTTLE OMS HELIUM REGULATOR DESIGN & DEVELOPMENT

FINAL REPORT

May 31, 1974

Prepared for:

National Aeronautics and Space Administration
Lyndon B. Johnson Space Center
Houston, Texas 77058

The Marquardt Company
16555 Saticoy Street
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FOREWORD

This report is submitted by the Marquardt Company in accordance with the requirements of NASA Contract NAS 9-12992. The work was administered by the NASA Lyndon B. Johnson Space Center, Houston, Texas, with Mr. J. W. Griffin as the NASA Technical Project Manager.

This program was performed by the engineering department of the Marquardt Company at the Van Nuys facility. The program manager was Mr. H. Wichmann; project engineers were Mr. R. Lynch during the first part of the program and Mr. T. L. Kelly during the latter part of the program. Other contributors to this program were Messrs. G. Pond, E. Benz, W. E. Hensley, A. Malek, I. Dickens, T. Piercy, A. Marderian, and D. Slagle.

ABSTRACT

An integrated program of analysis, design, fabrication and design verification testing was conducted to determine the realistic technology level attainable for development of the helium pressurization regulator for the Space Shuttle OMS application. This program evaluated existing regulator concepts, identified their deficiencies, and generated new concepts which eliminated these deficiencies and added the features of long life and multi-mission use. The prototype regulator fabricated for this program was a single-stage design featuring the most reliable and lowest cost concept available.

A tradeoff study on regulator concepts indicated that a single-stage regulator with a lever arm between the valve and the actuator section would offer significant weight savings. Damping concepts, including pneumatic and mechanical, were evaluated by testing to determine the amount of damping required to restrict actuator travel to 0.002 inch during vibration. The disadvantage of each device-size and friction, respectively, is not present in the hydraulic damper.

The regulator was fabricated entirely of Inconel 718, except for the tungsten carbide seat/poppet and the Ni-span-c springs. Flanged joints for accessibility and an LVDT actuator position transducer were additional design features. Component design parameters such as spring rates, effective area, contamination cutting, and damping were determined by test prior to regulator final assembly. The unit was subjected to performance testing at flow rates from 44 to 340 scfm, temperatures from 310 to 610°R, and inlet pressures from 400 to 4000 psi, random vibration levels to 26 grms, slam starts, leakage and 15,000 cycles during the design verification test program.

A test plan for propellant compatibility and extended life tests to be conducted by NASA-JSC is included. It is recommended that a flightweight level arm regulator with hydraulic damping be developed during the follow-on program.

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1.0 SUMMARY

The objective of NASA Contract NAS 9-12992 was the development of pressure regulator technology for the orbital manuevering system of the space shuttle. Pressure regulators utilized in this propulsion system require longer service life, greater contamination tolerance, and greater propellant compatibility than currently available pressure regulators. The program included an assessment of the current state-of-the art of pressure regulators and the determination of the deficiencies of currently available pressure regulators, particularly those developed for the Apollo program. Based on this assessment and understanding more advanced pressure regulator concepts were defined. A single stage pressure regulator concept was selected from the four most promising designs thus defined and one prototype regulator of this concept was built and tested. The test program provided an excellent understanding of the concept selected and verified its suitability for the space shuttle OMS application.

The initial task during the program served to gain an understanding of the problems encountered during previous regulator developments and applications and to evaluate existing designs with respect to the space shuttle requirements of multi-mission, long service life. Particular emphasis was placed on the understanding the problems encountered with the pressure regulators used on the Lunar Module ascent propulsion system, Lunar Module descent propulsion system, and in the Apollo service module propulsion system since these components were of roughly the same size as those required for the space shuttle orbital manuevering system and since their performance during the Apollo program had been well documented. The primary deficiencies of these pressure regulators were determined to be a lack of propellant compatibility, contamination tolerance, and, particularly, as extrapolated to the space shuttle requirements, service life. The lack of propellant compatibility was, to a large degree, due to the fact that though specific propellant compatibility requirements were established during the development program, it was thought that the check valves utilized downstream of the pressure regulators would prevent propellant vapors from reaching the pressure regulators. However, all pressure regulators surveyed appeared sensitive to propellant residuals (particularly residuals from the nitrogen tetroxide or from the reaction of the two propellant vapors) and to other forms of contamination. These forms of contamination included self-generated contamination which resulted from sliding friction and the reaction of materials of construction with the propellant vapors. In the case of the three Apollo regulators analyzed the effects of contamination varied from one regulator to another. However, such features as small flow passages (as used in the pilot stages of these regulators), small clearances around moving elements, sliding dynamic seals, small control orifices, poppet/seat interfaces in the pilots, main stages, overpressure relief valves, etc., appeared most affected by the contamination. Primary regulator performance characteristic deficiencies identified were internal leakage, high lockup pressures, and wide open failure. To eliminate or at least minimize these deficiencies, all design concepts generated and analyzed during this program included such features as the elimination of all sliding fits and small clearances through the use of metallic

flexures, minimizing the number of potential internal leakage path by minimizing the number of sealing closures (ideally only a single poppet/seat interface such as in the single stage regulator), replacement of all sliding dynamic seals by hydroformed bellows, utilization of large flow passages wherever possible, employment of a contamination tolerant cutter sealing closure, and selection of fully compatible all metallic and ceramic materials (no plastics or elastomers). Preliminary designs of four pressure regulator concepts featuring the design features just mentioned were prepared and these designs were analyzed in detail to verify correct regulator sizing and regulator stability characteristics. This analysis effort included various manual analysis tasks as well as three analog computer programs, five APL programs, and experimental poppet/seat interface flow and force data from another pressure regulator technology program that was performed at Marquardt. The four regulator configurations were compared on the basis of cost and schedule, weight, envelope, stability characteristics, accuracy, and reliability. The concept receiving the most points in this comparison was a single stage regulator which constituted the simplest configuration of all of the four configurations analyzed.

Detail design drawings of a prototype version of the single stage regulator concept were subsequently prepared. The prototype regulator featured a pressure balanced poppet, friction-free flexure guidance, solid and pneumatic damping, and a bellows seal between the outlet cavity and the actuator sensing cavity. The regulator was made entirely of Inconel 718 except for the poppet/seat interface which was tungsten carbide and the reference springs which were Ni-span C. To make various internal components of the pressure regulator readily accessible the prototype regulator featured six flanged joints. They also included several additional pressure test ports and an LVDT position transducer for monitoring regulator actuator movement. Except for the teflon jacketed seals employed at the flanged joints the regulator was of completely metallic/ceramic construction. In addition to the prototype regulator detail design drawings were also prepared for a flow limiter. The intent of this flow limiter was to limit the maximum flow rate through the pressure regulator to approximately twice the nominal flow rate. The flow limiter employed a variable area venturi nozzle where the area was varied by a movable pintle as a result of the flow forces acting on the pintle. This flow limiter also featured all metallic construction and friction-free flexure guidance. One prototype pressure regulator and one prototype flow limiter were fabricated during this program.

The prototype pressure regulator was subjected to a design verification test program. Initial testing consisted of the verification of outlet pressure regulation accuracy and stability during flow bench tests at flow rates up to 20 cfm helium over an inlet pressure range of 400 to 4,000 psia and a temperature range of 150°F to -150°F. Excellent outlet pressure accuracy and stability characteristics were demonstrated during this program (±2 psia accuracy). As a result of facility malfunctions these flow bench tests also included conditions wherein ice and contaminants were introduced into the regulator. Even under these conditions good outlet pressure regulation characteristics were demonstrated. A major portion of the design verification test program consisted of vibration testing to the

OMS engine burn random spectrum and to the main shuttle engine burn random spectrum. These tests were performed at nominal helium flow rates and while both outlet pressure oscillations and regulator actuator movements were being monitored. Very stringent criteria was established which limited regulator actuator movement to less than 0.002 inch (even though outlet pressure under conditions of substantially higher movement oscillations are perfectly acceptable) to assure that the poppet would not strike the seat continually at the minimum stroke position. The pneumatic damper did not provide sufficient damping under this stringent criteria and was therefore replaced with a mechanical damper during later portions of the vibration test program. This mechanical damper did demonstrate the actuator movement limits set forth in the design criteria. Another design verification test performed consisted of life cycle testing the pressure regulator through 15,000 cycles. Maximum leakage rates measured during this test program were well below the stringent design goal of 100 scc per hour of helium (measured leakage rate was less than 15 scc per hour of helium). Other design verification tests included a slam start test and flow bench tests with relatively unstable check valves obtained from the Apollo program. Both of these tests were entirely successful since no overpressurization and no regulator instabilities were incurred.

The design verification test program successfully demonstrated the applicability of the single stage regulator concept to the requirements of the space shuttle orbital manuevering system. The program further demonstrated the use of completely propellant compatible long life materials and design concepts with greatly improved contamination tolerance. The program also showed that by employing design concepts which completely eliminate friction and by providing separate and distinct means of damping a very stable extremely long life pressure regulator can be developed.

2.0 INTRODUCTION

The pressure regulator technology program described in this report was performed in support of the Space Shuttle Program of the National Aeronautics and Space Administration. The space shuttle vehicle is designed to provide low cost transportation to earth orbit to support a variety of missions, including logistics resupply of a space station. To achieve maximum cost effectiveness, the space shuttle is being designed for up to 100 flights (reuses) over a ten year operational lifetime including the capability to relaunch within two weeks after earth landing. The system is being designed to minimize required postflight refurbishment, maintenance and checkout, and for simplicity and ease of maintenance when required. For translational maneuvers and to achieve attitude control the space shuttle employs two rocket propulsion systems called the Orbital Maneuvering System (OMS) and the Reaction Control System (RCS). Both of these propulsion systems are pressure fed rocket systems employing nitrogen tetroxide and monomethylhydrazine as the propellants and helium as the pressurant. Pressure regulator technology developed during this program was specifically intended for the OMS propulsion system; however, the possibility of making the pressure regulators common to both the OMS and RCS systems was also continually addressed during the performance of this program.

A need for the performance of the pressure regulator technology program described herein resulted from deficiencies inherent in other spacecraft pressure regulators, particularly those employed during the Apollo program, to satisfactorily perform during the long life multi-mission requirements of the space shuttle. These deficiencies generally consisted of a lack of sufficient propellant compatability and contamination tolerance and also included more limited cycle life. Therefore the pressure regulator design philosophy and accepted design practices of past applications had to be extrapolated and new approaches developed for the pressure regulators of the space shuttle OMS to achieve sufficient propellant compatibility, contamination tolerance, and life.

The space shuttle OMS helium regulator design and development program consisted of six principal tasks. These are:

- Task 1 - Analysis
- Task 2 - Design Definition
- Task 3 - Prototype Regulator Fabrication
- Task 4 - Design Verification Tests
- Task 5 - Post Test Evaluation and Refurbishment
- Task 6 - Final Report

Task 1 served to establish a firm technical basis upon which the development of the space shuttle OMS helium regulator designs could be founded. Based on Marquardt's in-house rocket system component experience and industry and government literature searches and surveys the state-of-the-art of regulators and regulator related components and elements relative to the OMS helium regulator requirements was determined. This included an evaluation of existing regulator designs, particularly those utilized during the Apollo program for potential application to the OMS requirements. A preliminary analysis program was prepared to appraise the suitability of various basic regulator concepts for the OMS application. These analyses efforts served to identify four candidate regulator concepts which were subsequently analyzed in detail during Task 2. During Task 2 the four candidate concepts were defined by preliminary design layout. Design parameters were developed, based upon meeting the performance requirements of the space shuttle OMS application. Steady state and dynamic performance projections were made for each configuration, employing digital and analog computer modeling programs developed specifically for this application. As a result of this effort, sufficient data was generated to make an objective evaluation of the four candidate regulator configurations and the selection of one configuration for further development.

During Task 3 the single stage pressure regulator selected during the design definition task as the concept most suitable for meeting the space shuttle OMS requirements was detail designed and one prototype unit was fabricated. The prototype regulator built included a number of flange joints as well as special provisions for instrumentation such as pressure and an analog position indicator to permit the performance of a flexible design verification test program. This test program was performed in support of Task 4 and included regulator performance over the required pressure and temperature range and at flow rates up to approximately 125% of nominal flow rate. In addition, the design verification tests included extensive vibration testing to vibration spectra anticipated during OMS engine burn, main shuttle engine burn, and liftoff. The first two of these vibration spectra were performed with the pressure regulator in an operating mode whereas the liftoff vibration spectrum was in a nonoperating mode. Life cycle testing was also performed as was the evaluation of system effects such as slam starts and check valve interaction. Certain regulator modifications were made during Task 6 to improve regulator performance and to get the prototype regulator ready for shipment to NASA-JSC for further extended propellant compatability testing. Task 6 served to identify the effort performed in writing this final report.

The Space Shuttle OMS Helium Regulator Design and Development Program served to demonstrate the feasibility of utilizing a single stage pressure regulator featuring component design characteristics greatly more tolerant to contamination and much more propellant compatible than existing spacecraft pressure regulators. This program prepared the way for the development of flightweight pressure regulators for the space shuttle orbital manuevering system.

3.0 TECHNICAL REQUIREMENTS

The technical requirements described by Exhibit A of the statement of work for this contract are presented. They include general requirements for the study, design, and development tasks and the technical guidelines for those tasks (3.1), specific analysis tasks (3.2), specific design tasks (3.3), specific development tasks (3.4), verification test (3.5), post-test evaluation and refurbishment (3.6), and final report (3.7). The sections of this report which correspond to the technical requirements are:

S.O.W. Paragraph	Final Report Section
3.2	4.0
3.3	5.0
3.4	6.0
3.5	7.0
3.6	6.0 and 7.0

3.1 GENERAL

3.1.1 STUDY REQUIREMENTS

The contractor will be required to develop two alternative approaches or concepts that are applicable to the fulfillment of the technical objectives set forth in this SOW. These alternatives will be the result of concept and feasibility investigations, trade-off analysis, engineering assessments and/or other specific identified investigations. Each alternative will (a) specify any evolving scientific and technological findings and requirements and (b) identify the impact that these requirements may have on gross schedules and costs. Based on the alternatives that have been presented, the contractor will be required to rank these alternatives in order of their desirability.

3.1.2 DESIGN REQUIREMENTS

The contractor will define in detail the concepts and theories emanating from the study effort. Environmental conditions under which the regulator(s) will satisfactorily operate and the performance and detailed characteristics of the equipment will be clearly specified.

3.1.3 DEVELOPMENT REQUIREMENTS

The contractor will specify those special factors that must be considered in translating the design data into tangible end items. The contractor should identify any problems which become evident and might potentially affect manufacturing processes and techniques. The solution to these problems should identify what must be developed in order to facilitate manufacturing of the end product. The contractor will conduct testing and prepare test documentation to verify that the performance/design requirements of the regulator(s) meet the requirements of this SOW.

3.1.4 TECHNICAL GUIDELINES

The following guidelines, with a few noted exceptions, are not to be considered firm requirements. They are intended as optimum design objectives and are subject to change in accordance with technology limitations and reliability considerations. One of the primary objectives of this contractual effort is to define the realistic and obtainable requirements that should be imposed on a reducer valve for the space shuttle and thus hopefully avoid development problems that may result from initially unrealistic performance requirements. Due to the undefined status of the OMS, the regulator requirements are presently quite flexible; for this reason, the contractor should identify parameters that would be considered "drivers" in selecting regulator concepts and that would result in significant design changes other than scaling.

3.1.4.1 Application

The regulator technology and design recommendations developed as a result of this contractual effort will be utilized in defining the requirements for helium regulators for the space shuttle orbital maneuvering system (OMS). This propulsion system will utilize earth storable propellants that will be pressure fed to the rocket engine(s) with gaseous helium. The function of the helium regulator will be to precisely control the flow of helium from a high-pressure source to the low-pressure propellant tanks. The arrangement will consist of two regulator units in parallel with each unit isolated from a common high-pressure source by a separate isolation valve. Each regulator unit shall consist of two independent integral pressure regulators in series. This will result in a total of four regulators, in two units, of which any single regulator could satisfy the requirements of helium flow to the propellant tanks. The pressure settings of the regulators will be capable of sufficient variation to insure that only one regulator of the four is operating at any one time. The outlet of each unit will be connected to a manifold that will supply pressurant to both the oxidizer and the fuel systems through separate propellant isolation valves. The nominal operating mode of the OMS will require that the helium isolation valves, upstream of each regulator unit, be cycled open and closed simultaneously with the engine valves.

1.4.2 Fluid Media Compatibility

The reducing valves for this program must be compatible for exposure to the following propellant vapors, liquids, and combinations of oxidizer and fuel vapors. The regulators will be protected from gross liquid exposure by propellant isolation valves, but unlimited vapor exposure and vapor condensation are firm design requirements that must be satisfied. The propellants will be nitrogen tetroxide (N_2O_4), hydrazine (N_2H_4), unsymmetrical dimethylhydrazine (UDMH), 50/50 blend of hydrazine and unsymmetrical dimethylhydrazine (50% N_2H_4 - 50% UDMH), and monomethylhydrazine (MMH). The regulators must also be compatible with freon, alcohol, water, and trichloroethylene-type flushing and cleaning fluids. The contractor will have conclusive compatibility data on each material recommended for usage. In evaluating propellant compatibility, the contractor will also evaluate propellant and moisture combinations since once a regulator is exposed to propellants it is unreasonable to assume that the unit will remain free of moisture for the remaining service life. The contractor will not consider propellant decontamination of components to extend the service life, since cleaning of hardware between missions is improbable and will result only when required to insure personnel safety during system repairs.

The contractor is encouraged to omit, i' not exclude, the use of polymer seal materials within the regulator unit. This will eliminate a major source of contamination and propellant incompatibility. The contractor will evaluate metal bellows and metal-to-metal seals as a potential alternate to the use of polymer seal materials.

3.1.4.3 Outlet Pressure

3.1.4.3.1 Steady State

To insure that the design resulting from this contractual effort will be applicable to the OMS, it should be adaptable to an outlet pressure requirement from a minimum of 172 psig to a maximum of 250 psig. As a design point for this study, 184 psig will be utilized. However, to avoid problems with "cross talk" between parallel regulator units and/or series regulators, the outlet pressure set point for the regulator shall be sufficiently variable to insure that none of the four helium regulators in each OMS (see section 3.1.4.1) will interfere with each other. The requirement for this set point difference should not exceed 20 psi. A design goal for the maximum deadband is ± 4 psi, but the contractor should examine this requirement t evaluate the effect on life, cost, and reliability if this requirement is relaxed.

3.1.4.3.2 Lockup

Under no conditions should the lockup pressure exceed the set point plus 15 psi.

3.1.4.3.3 Stability

For the initial two seconds of flow, a reasonable increase in the paragraph 3.1. deadband will be allowed, but after two seconds the unit shall operate with the deadban! defined in 3.1.4.3.1. In no case should the unit exhibit a tendency toward instability that could possibly result in damage to the unit or surrounding hardware.

3.1.4.3.4 Outlet Pressure Limitation

Since the OMS may incorporate propellant isolation systems downstream of the regulator that, because of human error or failure, could reduce the downstream ullage to an extremely small volume, it is a desirable design feature for the regulator outlet to be able to withstand the inlet pressure. This would allow overshoot and/or leakage into small volumes without unit damage.

3.1.4.4 Inlet Pressure

The regulator will perform in accordance to this SOW for all inlet pressures within the range of 4000 psig to 350 psig. Since the lower limit will depend on the regulated outlet pressure, the contractor should use a minimum inlet pressure of 150 psi greater than lockup pressure as a guideline for design. The maximum limit of 4000 psig will be considered a firm requirement although the maximum may be from 3500 psig to 3000 psig.

3.1.4.5 Flowrate

The flowrate requirement will be between 1.0 pound/minute to 6.0 pounds/minute of helium. When the OMS is fully defined the variation from maximum to minimum flow will be on the order of 3 pounds/minute within the previously specified band. To allow reasonable propellant tank relief valves and to insure against over-pressurization that may result from any regulator failure, the regulator will have a flow limiter that will under no circumstances allow more than 10 pounds/minute of helium flow. The limitation of maximum flow is a firm requirement.

3.1.4.6 Ullage

The regulator will be required to function in accordance with this SOW for downstream ullage volumes varying from one (1) cubic foot to a possible maximum of from 150 to 300 cubic feet.

3.1.4.7 Thermal Environment

The reducing valve will be required to function nominally for helium inlet temperatures varying from a maximum of + 150°F to a minimum of -150°F. The regulator will be required to conform to the requirement of this SOW for unit temperatures that will vary from +150°F to -100°F at the initiation of helium flow through the unit.

3.1.4.8 Leakage

3.1.4.8.1 Internal Leakage

The design goal for internal leakage with the regulator at lockup is a maximum of 100 std cc per hour of helium. However, internal leakage is of secondary importance to propellant compatibility, service life, operational stability, and contamination tolerance. Therefore, the contractor should attempt to meet the 100-std-cc-per-hour requirement, but leakages up to approximately 1000 std cc per hour of helium will be acceptable. The primary function of the regulator is to control propellant tank pressure during engine operation. Any requirement to lockup and hold this leakage requirement when the OMS is not operating will result only from a helium isolation valve failure or a requirement to make up pressure in the OMS propellant tanks due to external leakage.

3.1.4.8.2 External Leakage

The design goal for external leakage is a maximum of one (1) std cc per hour of helium. In meeting this goal, the contractor should maintain an awareness of the requirement for refurbishment and extended life that may limit the use of welding as an external seal. Dual seals or a sealing method of at least equivalent reliability will be required at points where the possibility of external leakage exists.

3.1.4.9 Contamination

Contamination tolerance will be a major design objective for this regulator. As a design goal, the design should be insensitive to particles of 150 microns and smaller. Limitation of self-generated contamination shall also be a primary design goal. The contractor shall take appropriate measure to limit self-generated contamination. These measures as well as the contamination sensitivity tolerance shall be documented in detail during this contractual effort. Contamination failures were a major failure mode during the Apollo program, and significant improvements in both component tolerance and self-generated contaminates will be required for the Space Shuttle Program.

3.1.4.10 Service Life and Refurbishment

A design goal is to obtain a regulator capable of a minimum shelf life of 7 years and a service life of 5 years with no maintenance other than adjustments or recalibration allowed. As a guideline the contractor shall assume that one year of service life consists of 520 minutes of flow time. This will consist of 1o minutes per mission for 20 missions and 10 minutes of ground checkout per mission. The design will be refurbishable and the possibility of critical subassembly replacement in the field should be evaluated.

3.1.4.11 Lubricants

Due to propellant compatibility, low-temperature operation, and the extended service life, only very limited use of lubricants will be allowed. The lubricants allowed will be for assembly purposes and not due to operational requirements. Total exclusion of lubricants is a desirable design goal.

3.1.4.12 Installation

The regulator will not be sensitive to orientation and will be capable of multiple braze cycles for installation purposes. This is not to imply that the regulator will be brazed into the OMS but rather that it should have the capability.

3.1.4.13 Reference Pressure

The regulator will use ambient pressure as a reference pressure, and the valve response and operational characteristics, other than outlet pressure, will not be affected by ambient pressures varying from sea level to space vacuum.

3.1.4.14 Weight and Envelope

Minimum weight and envelope are important design considerations not to be overlooked by the contractor.

3.1.4.15 Moisture Sensitivity

The contractor shall take appropriate design measures to minimize the sensitivity of the regulator to moisture and propellant vapor freezing. The valve shall be capable of heated vacuum drying to allow the removal of gross moisture.

3.1.4.16 Vibration

The regulator shall operate satisfactorily with the following random vibration input:

Orbiter Main Engine Burn — Acceleration spectral density constant at $0.025g^2/H_z$ from 20 to $280H_z$, 6db/octave increase to $0.15g^2/H_z$ at 700 H_z, constant $0.15g^2/H_z$ from 700 H_z to $2000H_z$.

OMS Engine Burn — Acceleration spectral density constant at 0.004 g^2/H_z from 20 to 340 H_z, 6 db/octave increase to 0.01 g^2/H_z at $500H_z$, constant 0.01 g^2/H_z from $500H_z$ to $2000H_z$.

The regulator shall not be damaged by repeated exposure to the following random vibration input for a non-operating condition:

Lift Off — Acceleration spectral density increasing at the rate of 3db/octave from $0.15g^2/H_z$ at $20H_z$ to $0.4g^2/H_z$ at $60H_z$, constant at $0.4g^2/H_z$ from $60H_z$ to $1000H_z$, 3db/octave decrease to 0.2 g^2/H_z at $2000H_z$.

Transonic — Acceleration spectral density increasing at the rate of 9 db/octave from 0.002 g^2/H_z at $20H_z$ to 1.0 g^2/H_z at 160 H_z, constant at 1.0 g^2/H_z from 160 H_z to 1000 H_z, 9 db/octave decrease to 0.12 g^2/H_z at 2000 H_z.

Max Q — Acceleration spectral density increasing at the rate of 9 db/octave from 0.001 g^2/H_z at 27 H_z to 0.2 g^2/H_z at $160H_z$, constant at 0.2 g^2/H_z from $160H_z$ to $1000H_z$, 9db/octave decrease to 0.0[illegible]5 g^2/H_z at 2000 cps.

Test duration for 500 missions: Lift-off 1.5 hours, transonic 1.5 hours, max Q 7.0 hours, orbiter main engine burn 67.0 hours, OMS engine burn 50.0 hours.

3.1.4.17 Plumbing Insensitivity

The regulator design should not be sensitive to downstream plumbing configurations. This requirement is necessary to insure that the regulator will be adaptable to the OMS when it is firmly defined, and due to the extended service life of the shuttle, it is reasonable to assume that the plumbing configuration may be varied on different shuttle vehicles.

3.1.4.18 External Environment

The reducing valve will be compatible with space environment as well as coastal area environments. The unit will also be able to withstand the shipping and storage environments that are commonly imposed on aerospace component hardware.

3.1.4.19 Filters

Due to inability to accurately monitor filter condition and the maintenance involved in filter changes for extended service vehicles, it is desirable to limit the use of filters in regulators. Filters that are placed in the valve should be removable for maintenance without requiring the removal of the regulator from the OMS or any recalibration or adjustment of the regulator.

3.1.4.20 Fabrication Limitations

In the process of designing a prototype regulator to satisfy the requirements of this SOW, the contractor should maintain an awareness of the design requirements that will be imposed on a "flight-type" design to insure that the prototype will be adaptable. Although not applicable to this contractual effort, the common aerospace limitations on snap rings, crimped joints, staking, etc., will be imposed on flight-type designs.

3.1.4.21 RCS Commonality Considerations

The reaction control system (RCS) for the space shuttle will have a requirement for a helium regulator with basically the same requirements that will be required for the OMS. The primary difference will be a firm requirement for a maximum leakage of 100 cc/hr of helium since the RCS helium isolation valves will be latched open for flight. If the OMS unit does not meet this requirement, the contractor will evaluate the adaptation required to achieve as much commanality between the RCS and OMS as possible.

3.2 ANALYSIS

3.2.1 PROBLEM DEFINITION

The contractor will conduct an industry and Government survey and literature search to define the failure modes, development problem areas, and operational problems in previous helium regulator development programs with emphasis on programs that imposed requirements for earth storable propellant compatibility, high-vibration environment, tight regulated outlet pressure bands, extended service life, low contamination sensitivity, and other critical areas as defined in 3.1.4. These problem areas will be divided into categories of technology deficiencies, procedural or handling sensitivity, design deficiencies, and materials.

The contractor will also prepare a compilation, based on his company and personnel experience, of theoretical or potential problem areas that will be combined with the survey data to establish a comprehensive definition of the potential problem areas to be encountered in the development of the OMS helium regulator. An approach with a reasonable number of alternatives will be developed to resolve or minimize each of these problem areas.

The contractor will present the analysis, historical background, and reasoning to support each of the proposed approaches as well as applicable data to support the definition of the problem area.

The contractor will maintain an awareness of these problem areas for the duration of this contractual effort and modify and supplement the solution approaches as design development proceeds.

3.2.2 STATE-OF-THE-ART DEFINITION

The contractor will analyze the guidelines presented in section 3.1.4 and, in conjunction with an industry and Government survey and literature search, determine the state-of-the-art in each of the requirement areas. The interrelation of the various requirements will be analyzed and a state-of-the-art set of design capabilities developed utilizing the various variable guidelines of section 3.1.4 as primary design drivers. The contractor will present the analysis, historical background, and reasoning to support the state-of-the-art definitions that he develops.

3.2.3 REGULATOR CONCEPTS AND ARRANGEMENT EVALUATION

The contractor will conduct an industry and Government survey and literature search to establish the regulator design concepts and arrangements that are promising for OMS shuttle application. The previous application, if any, of promising concepts will be documented to evaluate design confidence and "in-service" deficiencies. Each concept will be analyzed to determine the particular advantages and deficiencies of that design as well as the sensitivity to the design requirements as discussed in section 3.1.4. Regardless of their acceptability, the contractor will document all alternates considered along with sufficient schematics, drawings, historical data, analysis, and reasoning to provide a working knowledge of each concept and support the conclusions drawn. The contractor will utilize the data generated as a result of this task to select optimum design concepts for additional study as defined in section 3.3.

3.2.4 EXISTING HARDWARE EVALUATION

The contractor will survey the helium regulator industry to determine the availability of hardware that can potentially satisfy the significant requirements of section 3.1.4 with no or minimum modification to the basic design. The performance capabilities and deficiencies of the promising designs will be defined and the basic design changes will be identified. Regardless of their acceptability, the contractor will document all alternates considered, along with sufficient schematics, drawings, historical data, analysis, and reasoning to provide a working knowledge of the component and support the conclusions drawn. The contractor will utilize the data generated as a result of this task to select the optimum minimum modification design concepts for additional study as defined in section 3.3.

3.2.5 MATERIAL PROPELLANT COMPATIBILITY STUDY

Since propellant compatibility is the primary design goal of this contractual effort, the contractor will conduct a material study to determine the materials, metallic, non-metallic, and lubricants, that are acceptable for extended use in the referenced propellants in accordance to the requirements of section 3.1.4. Conclusive data will be supplied to support all materials approved for use, and all data sources will be documented.

3.3 DESIGN DEFINITION

Based on the results of the Analysis Phase (3.2) the contractor, with the concurrence of the NASA technical monitor, will define a minimum of four regulator designs for additional study and development. A minimum of one design will result from 3.2.4 (Existing Hardware Evaluation) and will be a compromise design that can satisfy the major requirements of section 3.1.4 with a minimum of modification and thus minimum cost and schedule. A minimum of three designs will be optimized designs that may be major modifications of an existing valve or entirely new designs that will satisfy all of the requirements of section 3.1.4 within state-of-the-art limitations.

The contractor will conduct analysis and sufficient design work on the designs to develop the following areas and prepare a specification for each design. At the conclusion of this contract phase a design review will be held at NASA-MSC to select a design for additional development as defined in sections 3.4 and 3.5.

3.3.1 CONFIGURATION

The contractor will define the design configuration schematically and in basic assembly drawings. Weight and envelope predictions will be prepared.

3.3.2 PERFORMANCE

The contractor will present the performance predictions for each design along with analysis, test data, and/or experience data to substantiate the predictions. The contractor will individually address each requirement in section 3.1.4 and define the design compliance with each requirement.

3.3.3 COST AND SCHEDULE

Based on each design concept, the contractor will prepare cost and schedule data for a regulator development program for an OMS [illegible]-qualified unit. The schedule and cost will be divided into design development, qualification, production, and refurbishment. The degree of risk involved with each unit will be included in the [illegible] specified as a separate item.

3.4 DESIGN DEVELOPMENT

3.4.1 REGULATOR DESIGN AND FABRICATION

The contractor will design and fabricate one prototype regulator stage and flow limiter of the configuration and performance potential selected by NASA-JSC design review team at the conclusion of section 3.3. As a part of the design development the contractor will perform sufficient design feasibility testing to resolve questionable design areas prior to final prototype fabrication and to develop confidence that the finished prototype will meet the design objectives.

3.4.2 TEST FIXTURE DESIGN AND FABRICATION

The contractor will design, develop, fabricate and/or subcontract the test fixtures and facilities required to test the regulator prototype units in accordance with section 3.5, Design Verification Test.

3.5 DESIGN VERIFICATION TEST

The contractor will utilize the regulator prototypes and test fixtures developed in phase 3.4 to conduct a design verification test program. The test program will verify the prototype compliance or degree of compliance with each of the requirements of section 3.1.4, with the exception of extended service life and long-duration compatibility. The contractor will prepare a test plan for a NASA-MSC test program that will include extended-service-life testing and long-duration compatibility as well as other tasks that may be defined as a result of the contractor Design Verification Test Phase. The contractor will, however, verify basic compatibility of the regulator unit to the extent that no reasonable doubt exists that the unit is acceptable for extended-duration propellant exposure. The function of the MSC test program is to conduct extended-duration testing and supplement, in questionable areas, the basic test data generated by the contractor. The contractor's design evaluation and conclusions will be independent of any MSC test activity. The design development phase and design verification phase of this contract shall have a sufficient scheduling overlap to allow minor redesign or modification of test units based on preliminary design verification testing. The contractor will devote special attention to the vibration testing of the prototype hardware and will perform in-depth analytical and test evaluation of the hardware to insure that no vibration sensitivity exists.

3.6 POST-TEST EVALUATION AND REFURBISHMENT

Following the completion of the Verification Test Phase, the contractor will perform a data evaluation and define the actual regulator performance as well as the potential performance that can be expected with design modifications. Two test units and/or spare regulator units, in the test unit configuration, will be refurbished, modified, and shipped to NASA-MSC for additional testing.

3.7 FINAL REPORT PREPARATION

The contractor will prepare a final report that will thoroughly document his entire contractual effort and provide the data, drawings, and analysis as required by this SOW.

4.0 ANALYSIS*

The establishment of a sound technology base, which enhances confidence, is a vital prerequisite to initiating the development design of an advanced helium pressure regulator for the Space Shuttle OMS application. Marquardt's initial effort, under NASA-MSC Contract NAS 9-12992, was directed toward the formulation of this technology base. This effort consisted of literature searches and surveys, industry and Government agency inquiries, preliminary analyses, and an evaluation of all accumulated information to define, with respect to the OMS Helium Regulator requirements, potential problem areas, technology deficiencies and state-of-the-art status, existing designs suitability for this application, candidate configurations for meeting the design requirements, and candidate materials of construction capable of withstanding the anticipated environments.

The results of this effort are presented herein and are organized into five (5) general categories as follows:

Problem Definition

A survey of failure reports of comparable regulators to determine most probable modes of failure and origins of failure. Recommended design approaches are presented to preclude these failures during development of the OMS Helium Pressure Regulator.

State-of-the-Art Definition

A detailed review of the technical requirements of the OMS Helium Pressure Regulator, with respect to current technology capability, as documented by prior achievement.

Regulator Concepts and Arrangements

General analytical methods employed to reduce technical specification requirements to corresponding design parameters.

Existing Hardware Evaluation

An evaluation of the Apollo pressure regulator designs for adaptability to the OMS application.

*From Report 5103-7-1 of October 1972

Material - Propellant Compatibility

A literature search and industry survey to document the compatibility of candidate materials of construction with the anticipated usage fluids and environments.

As a result of this analysis effort, it is concluded that the achievement of the design goals of the OMS Helium Regulator is consistent with the current state-of-the-art and technology. Currently there is no pressure regulator that will fulfill all of the requirements of this application and the required changes to upgrade existing regulators to the shuttle application requirements would negate the validity of prior development testing.

The data presented herein established the foundation for the design definition task, during which regulator concept designs were generated, and analog computer models formulated to project dynamic characteristics.

4.1 DISCUSSION

General Method and Approach to Task 1 Analysis

To obtain a sound background upon which an objective analysis and evaluation of current helium pressure regulator technology could be based, information was solicited from three basic sources to supplement resources available at The Marquardt Company:

a) A literature search by Defense Documentation Center.

b) Detailed information from industry members involved in the design and development of pressure regulators and subelements of pressure regulators.

c) Current data relative to recent manned spacecraft pressure regulators from the NASA-MSC.

Responses from these sources were integrated with information obtained from the Marquardt Technical Library, prior applicable Marquardt experience, and the past experience of key Marquardt personnel to establish the basis upon which the analyses and evaluations described herein were performed.

4.2 ANALYSIS SUBTASKS

4.2.1 Problem Definition

Apollo Failure Report data, made available by NASA-MSC are summarized in Table 4-I. Though the nature of the noted failures was limited almost entirely to leakage or out of specification regulated outlet pressure, and designs had matured to at least qualification configuration, a wide range of origins of failure is evidenced which is representative of current regulator capability, with respect to the Shuttle OMS requirements. In Table 4-II, this data is reorganized to delineate the frequency of occurrence of the various origins of failure and a general classification of the failure is assigned to discriminate between technology deficiencies, procedural or handling sensitivity, design deficiencies or material deficiencies.

The literature searches conducted in support of this analysis effort, and Marquardt's company and personnel experience, confirmed that this regulator failure origin compilation was representative of pressure regulator designs that had matured to the advanced development status. Prior to achieving this status, the incidence of design-deficiency-origin failures can be expected to be significantly higher, since analytical techniques generally assume "ideal" conditions and result in point design performance projections.

TABLE 4-I. APOLLO PROGRAM PRESSURE REGULATOR FAILURE SUMMARY

Regulator System Application	Manufacturer	Failure Mode	Assigned Cause of Failure	No. Occurrence of Failure	Remarks
Apollo S/M Propulson System	B. H. Hadley (Royal Industries)	Leakage	Exceeded design life	1	
			Sealing surface finish	4	
			Contamination	2	
			Assembly error	2	Loosening of threads
			Spec omission	1	Series regr. not tested individually to spec. leakage
		Regulated outlet Pressure out of Spec. limit	Test error	8	
			Exceeded design life	1	
			Contamination	3	
			System Interaction	3	
			Other	8	Out of spec test condition or usage
			Sliding seal surface finish	11	
			Burrs	3	
			Icing	3	
			Structure Failure	2	Orifice Pin
				1	Fatigue
			Assembly Error	6	
			Design Deficiency	1	
			Adjustment error	4	
LM Ascent Propulsion System	Sterer	Leakage	Cold Temp. stiction of seals	3	
			No failure investigation	9	Contract cancelled
			Contamination	4	Test introduced or by subsequent installation of reg. into system
				1	Inlet filer media migration
				1	Propellant vapor exposure
				2	Seat/poppet impact self generated
		Other		1	Loosening of external seal bolts
				5	Improper seal installation
				4	Over tested unit

TABLE 4-I. APOLLO PROGRAM PRESSURE REGULATOR FAILURE SUMMARY (continued)

Regulator System Application	Manufacturer	Failure Mode	Assigned Cause of Failure	No. Occurrence of Failure	Remarks
LM Ascent Propulsion System	Sterer	Regulated Outlet pressure out of spec. limits	Cold temp "stiction" of seals	11	
			Sensing diaphragm leak	1	
			Tolerance of test equipment	1	
			Unknown	1	
			Test error	1	
			No failure investigation	13	Contract cancelled
			Adjustment error	6	
			Diaphragm spring rate out of tolerance	1	
			Flow limiter design deficiency	1	
		Structural failure	Inadequate thread engagement at ports	2	
			No failure investigation	1	Contract cancelled
		Miscellaneous test errors and omissions	No failure investigation	19	Contract cancelled
LM APS	Fairchild	Leakage	Improper seal installation	1	
			Improper seal/surface finish	3	
			Test error	1	
			Contamination	2	Suspected only
			Icing	1	
			Seat errosion	1	
		Regulated outlet pressure out of spec limits	System interaction	11	
			Contamination	3	Suspected only
			Internal friction	17	
			Test error	7	
			Improper guidance or alignment	3	
			Icing	4	
			Pilot poppet overstroke	1	
			Tolerance stack-up	1	
			Adjustment error	1	
		Structural failure	Test error	1	Inlet tube bent

TABLE 4-I. APOLLO PROGRAM PRESSURE REGULATOR FAILURE SUMMARY (continued)

Regulator System Application	Manufacturer	Failure Mode	Assigned Cause of Failure	No. Occurrence of Failure	Remarks
LM Descent Propulsion System	Parker	Leakage	Contamination	1	Reverse flow during test
				2	
			Seal/surface finish	1	Impact caused
		Regulated outlet pressure out of spec. limits	Contamination	4	
			Icing	1	
			Assembly error	1	
			Test error	7	
		Structural failure	Mtg. bracket cracked	1	
			Outlet stub to body weld crack	1	Test handling error
	Sterer	Leakage	No failure investigation	4	Contract cancelled
			Seal failure	2	
			Contamination	2	Internally generated
			·	1	Impact generated
			Assembly error	3	
			Seal/surface finish	2	
		Regulated outlet pressure out of spec. limit	Adjustment error	2	
			Contamination	3	Externally introduced
			Icing	1	
			Design deficiency	5	
			No failure investigation	4	Contract cancelled
			Seal installation error	1	
			Test error	9	

TABLE 4-II

APOLLO PROGRAM PRESSURE REGULATOR FAILURE CLASSIFICATION

Failure Origin	Frequency of Occurrence	Classification*
Test error	54	B
Contamination	31	B, C, D
Seal-Surface Finish Degraded	24	C, D
"Stiction"	20	C, D
Design Deficiency	20	C
Cold Temp. Effects	14	C, D
Assembly Error	15	B
System Interaction	14	A
Adjustment Error	13	B
Icing	10	B
Exceeded Design Life	2	B

A = Technology Deficiency

B = Procedural or handling sensitivity

C = Design Deficiency

D = Materials Deficiency

In the following paragraphs, the failure origins of Table 4-II are discussed in more detail to establish a more comprehensive definition of potential problems which may be anticipated in the development of the OMS helium regulator, and Marquardt's recommendations and alternate approaches to resolving or avoiding the respective problem areas.

One-fourth of the failures of Table 4-I were the result of test errors. A majority of these were deviations from test specification pressure or temperature requirements and were significant only in that the test program objective was to achieve qualification status. With respect to the development of the Shuttle OMS regulator, this category of failure is considered to be a procedural and handling error and can be avoided, or at least minimized, by:

a) Defining test conditions and tolerances consistent with test facility capabilities

b) Instrumentation utilized should have an accuracy at least one order of magnitude greater than the desired accuracy of the measured parameter

c) Test systems should include monitor/alarm and/or safety reliefs to prevent out-of-test conditions

d) Instrumentation placement shall assure measurement of the appropriate parameter.

Contamination was the predominant origin of failures to meet design requirements. Though several occurrences of failure were attributed to externally introduced contaminants, this is indicative of the designs' sensitivity to contamination. Other failures due to contamination were manifested in self-generated particles due to rubbing action, spalling of protective coatings, or erosion of critical sealing surfaces.

Although detail design details of the various Apollo regulators of Table 4-I were not available, both hard and soft seat configurations were represented. General opinion is that the soft seat configuration is more insensitive to contamination than other seat configurations. Nevertheless, the regulator failures reported indicate the criticality of minimizing contamination sensitivity.

Marquardt's approach to minimizing contamination sensitivity was extensively described in Section 3.1.2.1 of TMC Proposal P-233, which was submitted in response to the RFQ for this program effort. In view of the searches and surveys performed as an integral portion of this analysis effort, the proposed approach has been reaffirmed. In summary, this approach was to:

a) Eliminate sliding fit sources of self-generated contaminants and eliminate close clearances by flexure guiding all moving elements.

b) Incorporate a hard seat and poppet interface of materials harder than any anticipated contaminant.

c) Use a "knife edge" seat/poppet interface to minimize the probability of encounter with particulates and enhancing the probability of cutting any contaminants which encounter critical areas during closure

d) Design seat loads which assure capability to "cut" hard particle contaminants

e) Clean and assemble the regulator and components to a level consistent with the usage environment

f) Invoke procedures and handling restrictions which assure the maintenance of the required cleanliness level

g) Perform testing using a test media representative of the usage media and with system filters consistent with the anticipated usage system

h) As required, incorporate integral filters to limit contaminants to a tolerable level.

Seal-surface degradation was a significant cause of failure of the Apollo regulators. Specific details were not available in the data to assess the nature of the degradation; however, several potential modes were hypothesized:

a) Erosion of seal surfaces by the flowing media and media-borne particulates

b) Improper seal surface finish at time of fabrication

c) Materials with inherent flaws

d) Materials that are incompatible with the environment, the media, or adjacent materials

e) Accelerated wear of surface due to improper materials selection or materials treatment

f) Scuffing or rubbing of mating surfaces

g) Poor lubricant properties

h) Manufacturing flaws such as burr and/or sharp corners and edges which preclude proper seal installation

i) Adhesion due to impact loading.

The resolution of many of the seal-surface degradation problems were directly related to the proposed corrective design action for contamination problems. Use of flexure guided moving elements and flexure alignment, virtually all scuffing and contact between mating surfaces were eliminated. With the exception of the poppet-seat seal, all other sealing functions were accomplished by metal bellows. A primary objective of the materials of construction of the Shuttle OMS regulator design was the use of only materials with documented compatibility with the media and environment. No special surface treatments were anticipated, since sliding contact surfaces were eliminated. Material quality to eliminate inherent flaws and imperfections were enhanced by the use of vacuum melt raw stock, to the greatest extent possible, and rigorous raw stock quality control procedures. The anticipated design restricted critical surface finish requirements to the poppet-seat interface area, and resident experience, expertise, and equipment for verification of the required surface finishes to achieve the leak rate goal were utilized to their fullest extent.

"Stiction" type failures which occurred during the Apollo regulator programs were related to sliding surfaces in guidance and alignment control areas or sliding seal areas. The elimination of the sliding surfaces in the Marquardt design should effectively eliminate any occurrence of this type of failure.

"Cold temperature effect" failures, although quite numerous, were reported only for the Sterer APS regulator configuration. A change of the elastomer seal material to a compound more resistant to cold temperature hardening was apparently adequate to overcome this deficiency. The elimination of all elastomers from the Marquardt design should avoid the occurrence of this type of failure. Other cold temperature effect failures, which must be considered in the design, are related to thermal compensation for thermal differential growth, blowdown effects on inlet pressurant temperature, and downstream overpressurization due to warm-up of the pressurant in the ullage volume.

Reported assembly error originating failures were generally related to seal installation wherein damage to the seal resulted. The occurrence of these types of failures is minimized by minimizing the number of seals, invoking significant test procedures at various stages of assembly to screen out these failures and logical design, with appropriate tooling and definitive procedures, to preclude this type of error.

System interaction failures resulted from feedback from downstream check valves or from other regulators within a redundant or quad redundant regulator package. Resolution of these failures resulted when check valves were replaced with actual usage system components or by adjusting the nominal set points of the respective regulators within the redundant package. The occurrence of these types of failures emphasizes the importance of matching the dynamic characteristics of the regulator with those of the usage system. To this end, Marquardt made extensive use of analog computer programs to model the various regulator designs and the intended usage system to optimize the regulator characteristics for minimum sensitivity to downstream components and interaction with the other regulators of the quad redundant package, in both the normal operating mode and in the various failure modes.

Adjustment errors were primarily attributable to instrumentation deficiencies or operator error, rather than any design deficiency, such as adjustment point shifts or drifting. Their elimination is achieved by rigid quality assurance procedures and appropriate training of test personnel to develop sensitivity for accuracy.

The formation of water crystals or "icing" in critical areas of the regulators resulted from inadequate control of the quality of test influents. The sensitivity of a regulator design to this effect is related to the number of close fit areas, wherein the crystals can lodge and inhibit normal function. Icing failures are, therefore, akin to contamination failures. The design considerations delineated for minimizing contamination sensitivity also reduce the probability of icing failures. In addition, proper control of test influents and appropriate handling precautions to preclude the introduction of moisture, minimize the propensity for this origin of failure.

The two failures attributed to exceeding the design life cannot be fully evaluated, since no data were available to assess the magnitude of over design life usage. It would have significant impact on the design's reliability rating if the failures had occurred within a narrow margin of the design life, and it is evident of an inherent design deficiency.

4.2.2 State-of-the-Art Definition

In Table 4-III, each section of Paragraph 3.1.4 of Exhibit A of the contract (NAS 9-12992) is presented along with an evaluation of the state-of-the-art of that particular design criteria. Also listed are references which contribute to establishing the defined state-of-the-art.

In the following paragraphs discussion is presented of several of the state-of-the art design features pertinent to the anticipated design approaches to meet the requirements of the Shuttle OMS Helium Pressure Regulator. It must be pointed out that no existing design of a pressure regulator incorporates a combination of all of the design features described as "state-of-the-art".

4.2.2.1 Hard Seat Technology

Achieving an effective seal in a fluid system is a formidable task because it involves an understanding of a large number of variables. These varaibles include such characteristics as: sealing surface materials, sealing surface condition, sealing surface

TABLE 4-III. STATE-OF-THE-ART DEFINITION

Ref. Para. of Exhibit A of NAS 9-12992	Topic	Requirement	State-of-the-Art	References
3.1.4.1	Application	Earth storeable pressurization Quad redundant arrangement	Demonstrated capability	Current spacecraft programs
3.1.4.2	Fluid Media Compatibility	Compatible with:		
		N_2O_4	Demonstrated capability	Section 3.2.5. of this report
		N_2H_4	" "	
		A-50	" "	
		MMH	" "	
		N_2O_4 + Amine fuel (Vapor combinations)	No documented test data Believed to be most severe environment but within state-of-the-art	
		Freon	Demonstrated capability	
		Alcohol	" "	
		Water	" "	
		Trichcoroethylene	" "	
3.1.4.3	Outlet Press. Steady state	172 to 250 @ ± 4 PSI	±1% deadband demonstrated over this set point range	(4)
		Minimize or eliminate "cross talk" by limiting 4 reg. set point range to > 20 PSI	>20 PSI set point spread compatible with ± 4 PSI dead band range; analog model will evaluate cross talk effect	(5) (6) TMC progress reports on JPL Contract #953383
3.1.4.3.2	Lock-up	<15 PSI above set point	Adequate to allow sufficient seat preload to meet leakage	(3) (4) (5) (6)
3.1.4.3.3	Stability	2.0 sec max. to achieve deadband after step flow demand	Demonstrated capability for similar flow rate reg.	(4)
3.1.4.3.4	Outlet Press. Limitation	Outlet side capable of withstanding full max. inlet press. (4000 PSIA)	Precludes use of metal bellows actuator which would jeopardize meeting other design drivers. Extreme weight and envelope penalty anticipated if metal bellows can be used at all.	Section 3.2.3. of this report.
3.1.4.4	Inlet Press.	4000 to 350 PSIG (Minimum to be outlet set pressure +150 PSI)	Demonstrated capability but meeting deadband with unchoked flow compromises other characteristics.	(4) (5) (6)
3.1.4.5	Flow Rate	1.0 to 6.0 lb/min. (3.0 lb/min max spread) 10 lb/min max.	10 lb/min flow limit compatible with 1.0 to 6.0 lb/min design flow	(7)
			Range of flows impacts on spring rates req'd to meet regulation deadband.	Section 3.2.3. of this report.

TABLE 4-III. STATE-OF-THE-ART DEFINITION (Continued)

Ref. Para. of Exhibit A of NAS 9-12992	Topic	Requirement	State-of-the-Art	References
3.1.4.6	Ullage	1 to 300 ft^3 ullage (Reg. to meet all performance requirements	Within state-of-the-art Ullage influences Dynamic characteristics	(4) (6)
3.1.4.7	Thermal Environment	+150 to -150°F helium +150 to -100°F environment	Demonstrated capability	(6)
3.1.4.8	Leakage	--	--	--
3.1.4.8.1	Internal Leakage	< 100 SCC/Hr helium	Demonstrated capability	(4) (5) (6) (7)
3.1.4.8.2	External Leakage	< 1 SCC/Hr helium	Demonstrated capability Welded joints and static seal joints	(4) (5)
3.1.4.9	Contamination	Insensitive to hard particles up to 150 microns	No documented demonstration, but independent studies and some test data confirm capability	(7), (8), (9) (10), (11), (12)
3.1.4.10	Service Life and Refurbishment	7 yrs shelf life	State-of-the-art for all metal construction. Goal of no required maintenance or refurbishment is realistic.	
3.1.4.11	Lubricants	Exclusion is goal minimum use on assembly only.	Demonstrated capability	(12)
3.1.4.12	Installation	Insensitive to orientation Capable of multiple braze installations.	Demonstrated capability	Apollo Program
3.1.4.13	Reference Pressure	Sea level to space vacuum ambient ref. pressure	State-of-the-art	(6)
3.1.4.14	Weight and Envelope	Minimize	N/A	
3.1.4.15	Moisture Sensitivity	Minimize sensitivity Vacuum Bake Compatible	Demonstrated capability with no sliding fit design	(12)
3.1.4.16	Vibration	Limit of insensitivity TBD	30g rms overall random 30g Sine	Apollo Program (Fig. 4-1)
3.1.4.17	Plumbing Insensitivity	Insensitive to downstream system configurations and components	Dynamic interaction with downstream plumbing can be anticipated and evaluated by computer modeling, interaction can be minimized by design.	Apollo Failure Reports (6)

TABLE 4-III. STATE-OF-THE-ART DEFINITION (Continued)

Ref. Para. of Exhibit A of NAS 9-12992	Topic	Requirement	State-of-the-Art	References
3.1.4.18	External Environment	Space and Coastal Environments Shipping and Storage Environments	Demonstrated Capability	
3.1.4.19	Filters	Limit use; replaceable, if used.	Demonstrated Capability Replacement, if used, may not be required.	(7)
3.1.4.20	Fabrication Limitations	Design for flight.	Demonstrated capability.	(4)
3.1.4.21	RCS Commonality Considerations	100 SCC/Hr. Helium Leak Variable flow demand	Demonstrated Capability	(4) (12)

RANDOM VIBRATION SPECTRUM REQUIREMENTS

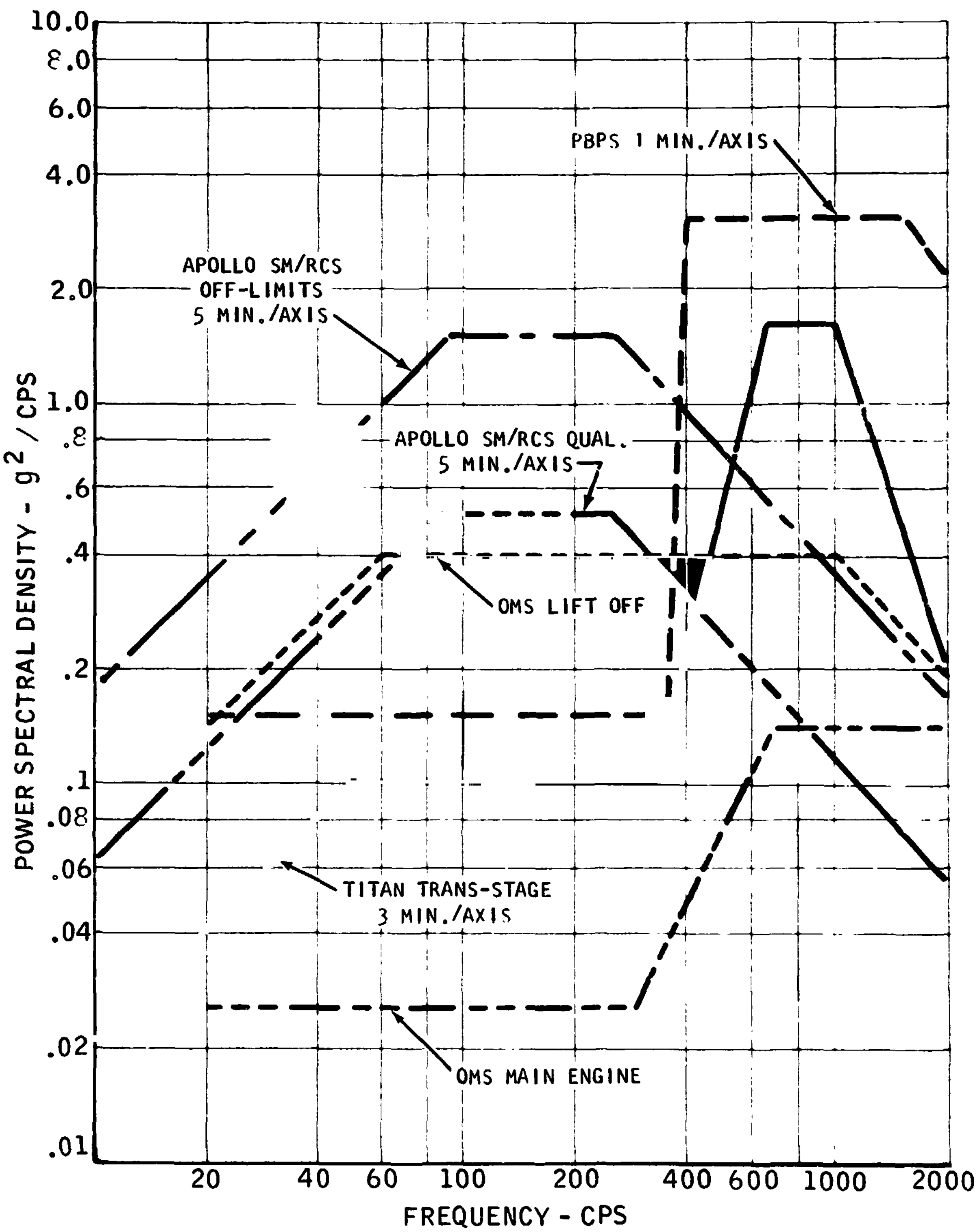


Figure 4-1

interface loads, properties of the fluid to be sealed, and all the environmental conditions such as temperature, pressure, shock and vibration loads, etc. Probably the greatest single problem area identifiable in the development of any propulsion system has been the problem of liquid or gas leakage. In recognition of this problem area, both the Air Force and NASA initiated the sponsorship of sealing technology programs approximately a decade ago. Initial research in this area was performed primarily by ITT Research Institute and the General Electric Company. Programs performed by these two organizations were of a more general nature in the area of sealing technology and were more oriented toward static seals and rotating dynamic seals rather than dynamic impact seals such as are encountered in the design of a poppet/seat interface.

Approximately eight years ago the Air Force Rocket Propulsion Laboratory initiated the funding of sealing closure technology specifically designed to solve the problem of poppet/seat interface leakage. Several years later, the NASA/Lewis Research Center also started to fund programs in this area. As a result of the sealing closure technology programs sponsored by AFRPL and NASA/Lewis, it became apparent that a primary cause of poppet/seat interface leakage was the presence of contamination and that criteria needed to be developed which would result in the definition of poppet/seat configurations which are compatible with reasonable levels of contamination. In recognition of this need, both AFRPL and NASA/Lewis, approximately four years ago, started funding programs which evaluated the contamination sensitivity of specific types of poppet/seat interfaces and also developed poppet/seat interface configurations which were more tolerant of contamination.

Nearly all of the sealing closure technology and contamination sensitivity programs released by AFRPL and NASA/Lewis were placed with three aerospace contractors: McDonnell Douglas West, the Rocketdyne Division of North American Rockwell, and The Marquardt Company. (Aerojet also received some funding but has not been active in this area for the past year.) Recently published reports pertinent to this area of technology and to the discussion presented herein are listed as References 6 through 12. The sealing closure technology work performed in recent years has resulted in the development of a good analytical leakage model which allows a reasonable prediction of gas leakage (certainly within 50%) for a particular poppet/seat interface once the required sealing closure characteristics are known. Analytical wear models of poppet/seat interfaces which consider the effects of cycling the poppet seat interface have also been prepared and wear data have been generated. However, the test data obtained in this area have been insufficient to date to permit the verification of a wear model which is accurate quantitatively; rather the wear model has been more suitable as a qualitative design tool. Finally, the programs concerned with the effects of contaminants upon sealing closure performance have given considerable insight into what happens when particles of various sizes and hardness are trapped between the sealing surfaces and some promising configurations which minimize these effects have been developed.

In reviewing the requirements of the Space Shuttle Orbital Maneuvering System Helium Regulator Design and Development Program, it is noted that the internal leakage requirements of the regulator units are indeed very severe. An allowable leakage rate of 100 scc's per hour of helium at a pressure differential of 3816 psi has been specified. This leakage rate must be met while the regulator is operating over a temperature range of -150 to + 150°F. In addition, the sealing closure is required to operate with contaminants of up to 150 microns in size present in the helium flow and for periods of up to 5 years. Finally, because of the pressurant system configuration, the materials of construction utilized in the sealing closure of the regulator unit must be compatible with liquid and vapor propellant as well as propellant/moisture combinations and various flushing and cleaning fluids.

The requirement for propellant compatibility limits the selection of sealing closure materials to teflon, metals, or ceramics. The long-term operation requirement, combined with the cold-flow characteristics of teflon and the zero maintenance requirements, raise considerable question regarding the applicability of the teflon as a dynamic sealing material. Consequently, the most promising sealing closure materials are the metals and the ceramics.

Poppet/seat technology programs have shown that high-cycle life of sealing closures is achieved only when the sealing closure interface stresses are kept below the endurance limit of the weaker of the two materials utilized for the sealing surfaces. To maintain the sealing closure interface stress levels low enough, and at the same time achieve low leakage rates, further requires the specification of very fine surface finishes. The importance of the surface finish is particularly evident from the analytical leakage model defined by Tellier and Caywood (Reference 49) as follows:

$$Q_u = \frac{100\, D_s\, H^3 \left(P_1^{\,2} - P_2^{\,2}\right)}{\mu L T \sigma^{2/3}}$$

$$Q_c = \frac{2 \times 10^4\, D_s\, h^3 \left(P_1^{\,2} - P_2^{\,2}\right)}{\mu L T \sigma^{3/2}}$$

Q_u = Leak rate, uni-directional lay - Scim

Q_c = Leak Rate, circular lay - Scim

D_s = Mean Seat Dia - in.

H = Sum of peak to valley heights - in.

P_1 = Inlet Pressure - psi

h = $H/2$

P_2 = Outlet Pressure

μ = Absolute Viscosity - lbm/in.-sec

L = Seat Land Width

T = Temperature - °R

σ = Stress - psi

These relationships predict the volumetric leakage rates through sealing closure interfaces featuring uni-directional and circular lay surfaces. (In practice, the measured leakage rates fall between these two predicted values, since the lapped surfaces feature a random lay rather than either the circumferential or uni-directional lay.) Now, it is evident from these relationships that the parameter designating surface finish (H or h) occurs to the third power. Thus it, more than any other parameter, influences leakage rate.

For the magnitude of leakage rate under consideration here, the range of surface finishes of interest is from approximately 4-AA to a fraction of 1-AA (Figure 4-2). The capability of measuring these fine surface finishes is essential to the performance of a sound sealing closure technology program. Because of the importance of being able to measure surface finishes and flatness of metal and ceramic surfaces, essentially all of the sealing closure technology work has been performed with flat poppet/seat interfaces. It is vastly easier to lap fine surface finishes and to inspect these surface finishes in a flat configuration than it is for conical or spherical surfaces. There is a second reason why a flat poppet/seat interface is generally preferred to conical or spherical interfaces for sealing closures requiring high cycle life (100,000 cycles or more) and this relates to the fact that the flat poppet/seat interface is subjected to much less interfacial scrubbing during each closure than the conical or spherical interfaces. Less scrubbing, in turn, means less wear and greater sealing closure life.

To evaluate the effects of wear on sealing closure life, The Marquardt Company, during its recent sealing closure technology work in support of Contract NAS 3-14349 for the NASA/Lewis Research Center, prepared an analytical wear model. This relationship has the form:

$$y_N = y^o + \frac{k_{AD}}{11,900} N \, s\sigma_I \left[\frac{1.4}{P_A} - 17.6 \times 10^4 \frac{y^o}{G_{AB}}\right]$$

where:

PREDICTED SURFACE FINISH REQUIREMENT

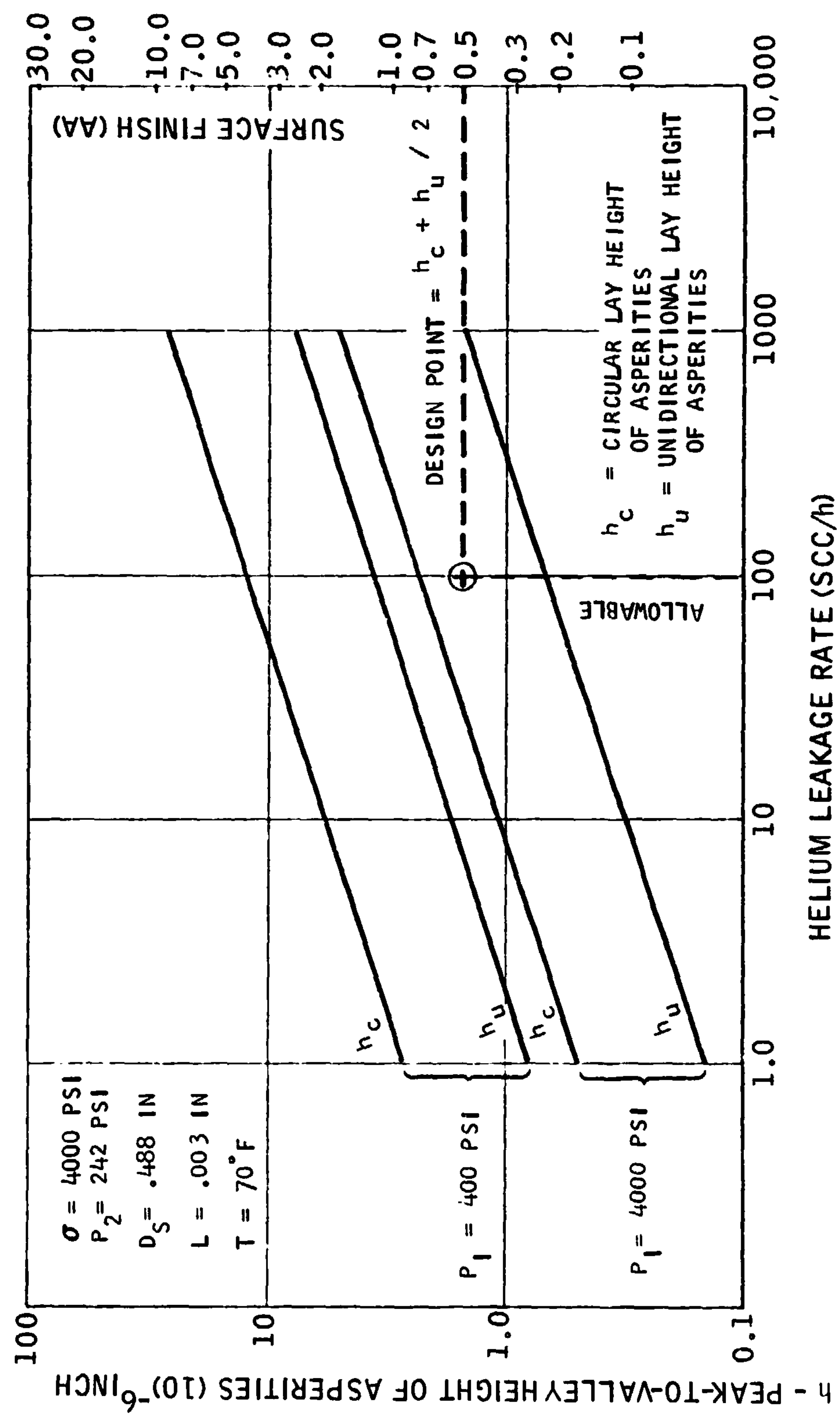


Figure 4-2

y_N = Surface finish after N cycles, in.

y^o = Original surface finish, in.

k_{AD} = Wear coefficient*

N = Number of cycles

s = Lateral Sliding component between poppet and seat (scrubbing distance), in.

σ_I = Poppet/seat impact stress, psi

P_A = Material hardness (softer surface), kg/mm^2

G_{AB} = Energy of adhesion, ergs/cm^2

The surface finish, y^o, in this equation is related to the peak to valley height, h, of the static analytical leakage model by the relationship $y^o = 1/3\ h$.

It is evident from this wear model that once a particular material has been chosen and its properties have been determined, the most effective way to minimize sealing closure wear is to minimize the scrubbing distance and the impact stress which occurs at the sealing surfaces during each closure. An example of techniques employed by The Marquardt Company to minimize scrubbing distances and impact stresses during closure may be seen by examining a large tungsten carbide seat which was successfully cycled for more than 100,000 cycles over a temperature range of -320 to +390°F without exceeding a leakage rate of 100 scc's per hour of helium at 450 psia inlet pressure. A photograph of this seat is shown in Figure 4-3 and a cross section in Figure 4-4. From Figure 4-4, it is evident that there is a second land incorporated into the tungsten carbide seat at a diameter slightly larger than the actual sealing land. The height of this second land is such that it is recessed with respect to the sealing land by 0.0001 inch. Thus, as the poppet approaches the seat during a closure motion, it will contact the sealing land only if the poppet surface is perfectly parallel to the sealing land surface. However, if any out-of-parallelism condition exists between these two surfaces, the poppet will first strike the outer land which is considered a bumper and which will then be subjected to essentially all of the scrubbing action that will occur between the poppet and the seat in turning the poppet to achieve perfect alignment between the sealing surfaces. Thus, the bumper serves as an alignment device between the poppet and the seat sealing surfaces and minimizes radial scrubbing.

*Rabinowicz (Reference 50)

TUNGSTEN CARBIDE SEAT
L 4682

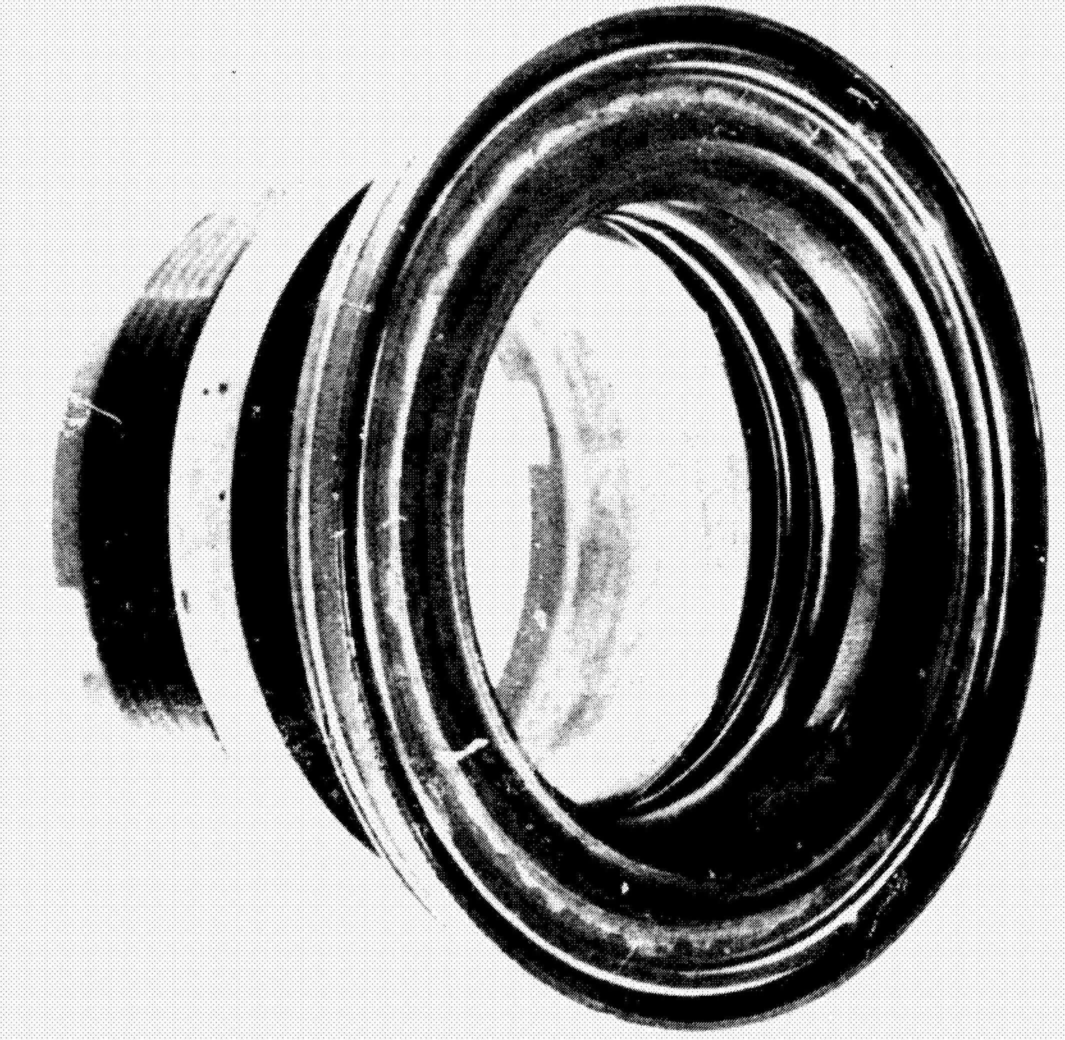


Figure 4-3

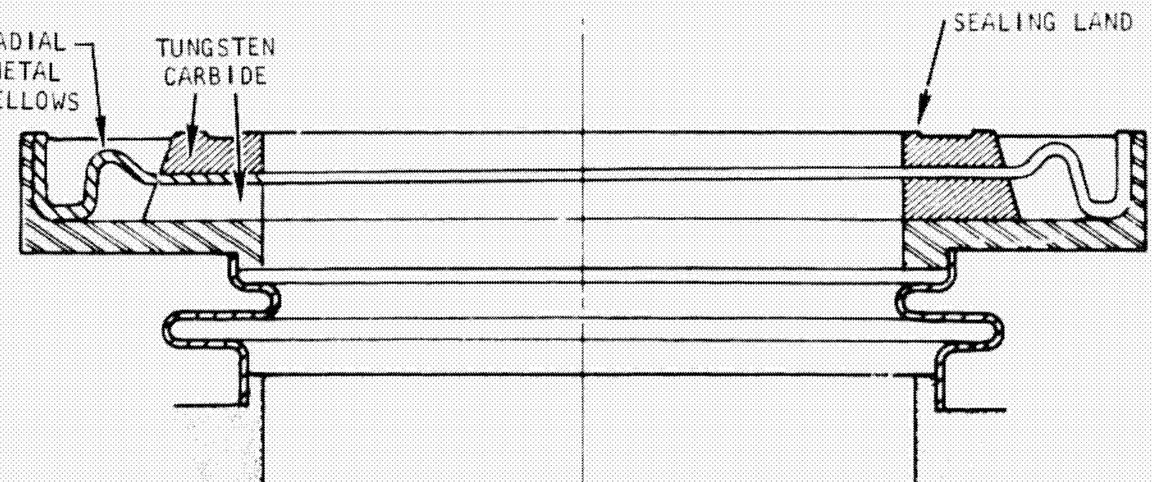


Figure 4-4

Another technique employed by The Marquardt Company to minimize scrubbing, which is not evident from this figure, is the use of metallic axial guidance flexures to guide the poppet during its closing motion. These flexures feature a very high radial spring rate and absolutely no radial clearance as is normally encountered in a sliding fit type guidance and therefore assure that the velocity vector of the poppet consists of only an axial component and no radial component when the poppet strikes the seat. Consequently, this guidance technique also minimizes scrubbing.

The approach utilized in the tungsten carbide sealing closure for minimizing impact loads consists of supporting the actual seat sealing ring on a single convolution bellows. Therefore, as the poppet strikes the seat during closure, the poppet is not stopped suddenly, but rather the bellows of the seat allow the poppet and the seat sealing ring to translate axially for some distance (0.006 inch for this particular design). At that point, the poppet strikes a separate stop (which is not evident in Figure 4-4) and imparts the kinetic energy of the moving poppet mass to the stop rather than to the seat sealing surface. In this manner, the impact stresses between the poppet sealing surface and the seat sealing surface are limited to those stresses required to accelerate the relatively low mass sealing ring to the closing velocity of the poppet. As evident from this discussion, the element of compliance (bellows) that permits reduction of impact stresses for this particular sealing closure has been incorporated into the seat; however, Marquardt has also developed designs where the element of compliance has been incorporated into the poppet/actuator part of the sealing closure.

Since nearly all of the sealing technology and wear technology programs have been concerned with flat poppet/seat interfaces, it was logical to also evaluate this type of interface for contamination sensitivity. The contamination sensitivity technology programs have generally approached this problem area from two directions. These are, what happens to particles and leakage characteristics when particles are trapped between the sealing surfaces, and how can the trapping of particles be avoided. A number of particle avoidance schemes have been evaluated and these have included such approaches as intentionally providing a flow cavity upstream of the interface where the particles were supposed to collect, rather than pass on through the sealing interface; secondary shutoff devices where an initial closure of the poppet/seat interface was achieved upstream of the sealing land such that only a very low flow rate of gas was permitted through intentionally provided small clearances at this first closure point to wash any particles located on the sealing land downstream without permitting the passage of additional particles towards the sealing land, and then closing the interface at the sealing land; and centrifugal devices which imparted a turning motion to the flow and were supposed to separate particles out into an area where they could be passed through the seat prior to the final closure of the poppet/seat interface.

The work with particle avoidance devices has resulted in the following conclusions:

- Particle dynamics are extremely complex.
- Analytical modeling of particle dynamics is not state-of-the-art at the present time.
- Particle avoidance devices incorporated into the sealing closures generally substantially complicated the sealing closures and resulted in other potential failure modes.
- The greatest reduction in particle hits on the sealing closure achieved to date has been a factor of only approximately 2.

Based on these conclusions, it must be stated that the state-of-the-art of contaminant particle avoidance devices is poor and that these devices do not appear practical at this time.

Elimination of the potential use of particle avoidance devices leaves the second area of sealing closure contamination sensitivity, namely, understanding what happens to entrapped particles, to be further explored and this area does offer considerable potential. Particle entrapment testing has been done with both soft and hard particles (R_c 62) and with sealing closures made from such materials as teflon, gold, copper, type 440C stainless steel, and tungsten carbide. The sealing closures utilized have featured land widths of from 0.0001 to 0.030 inch and have also featured multiple sealing lands. The advantages of multiple sealing lands are, of course, the same as of any redundant arrangement in that the probability of damaging all of the sealing lands due to particle entrapment is much more remote; however, this advantage is negated to some degree by the fact that the grooves between the various sealing lands tend to trap particles more readily so that the overall hit frequency for a multiple land seat is greater than the hit frequency for a single land seat of the same width. The particle entrapment tests have resulted in some important conclusions and these are briefly reviewed as follows:

- A particle which is harder than the softer of the two sealing closure materials is permanently embedded in the softer material.
- A particle which is softer than either sealing closure half is essentially flattened between the sealing surfaces and is most likely washed out during the following cycle. Also, the softer particle does not damage the sealing surfaces permanently.

- The effects of trapping a relatively hard or relatively soft particle between two sealing surfaces are essentially the same. If the particle is entrained in the sealing closure material, the material around the particle is disturbed and raised, whereas if the particle is not entrained, the particle itself simply spreads over a greater area. In either case, a diameter of disturbance has been identified as a function of the particle size and its relationship is shown in Figure 4-5.

- Hard or soft particles up to a size of approximately 40 microns can be squeezed between sealing surfaces featuring a land width greater than 0.010 inches without affecting leakage appreciably. Particles of this size will, however, damage (and cause leakage) sealing lands which are less than 0.005-inch wide and are made of a softer material than the particle.

- Particles were successfully cut by a very hard (ceramic) sharp-edged (0.0001 to 0.0004 inch wide) sealing closure without affecting leakage and without resulting in any damage to the sealing closure.

The above listed observations permit some conclusions as to what type of sealing closure interface is most contamination resistant and these are as follows: First of all, the harder the sealing closure material, the less likely it is to sustain any damage from entrapped particles. This suggests that the best sealing closure material is diamond, with a hardness of 7000 Knoop. Unfortunately, the use of diamonds in a size of 1/4 to 3/8 of an inch in diameter is not practical; consequently, the next best group of materials is the ceramics. Of all the ceramics generally available, boron carbide appears to feature the greatest hardness (over 3000 Knoop in a 99% of theoretical density material). Fortunately, The Marquardt Company has had substantial experience with a variety of ceramic materials, including boron carbide, for sealing closure applications. The presence of porosity is a characteristic of all ceramic materials and is due to the fabrication process which consists of sintering or hot pressing. The boron carbide material is exceptionally dense featuring in excess of 99% of theoretical density. This material was hot-pressed for Marquardt under specially controlled conditions by a laboratory which specializes in the fabrication of non-porous ceramics. Marquardt's experience has shown that the fabrication of non-porous ceramics requires the utmost in care and fabrication control and that ceramic materials are suitable for sealing closure fabrication only if this high degree of non-porosity has been achieved.

Another important conclusion that may be drawn from the particle entrapment program is that if the sealing closure land width is kept smaller than 0.005 inch and if the sealing closure interface loads are high enough to cause plastic deformation of the trapped

DIAMETER OF DISTURBANCE AS A FUNCTION OF CONTAMINANT PARTICLE SIZE

(FROM TEST DATA REFERENCE 9)

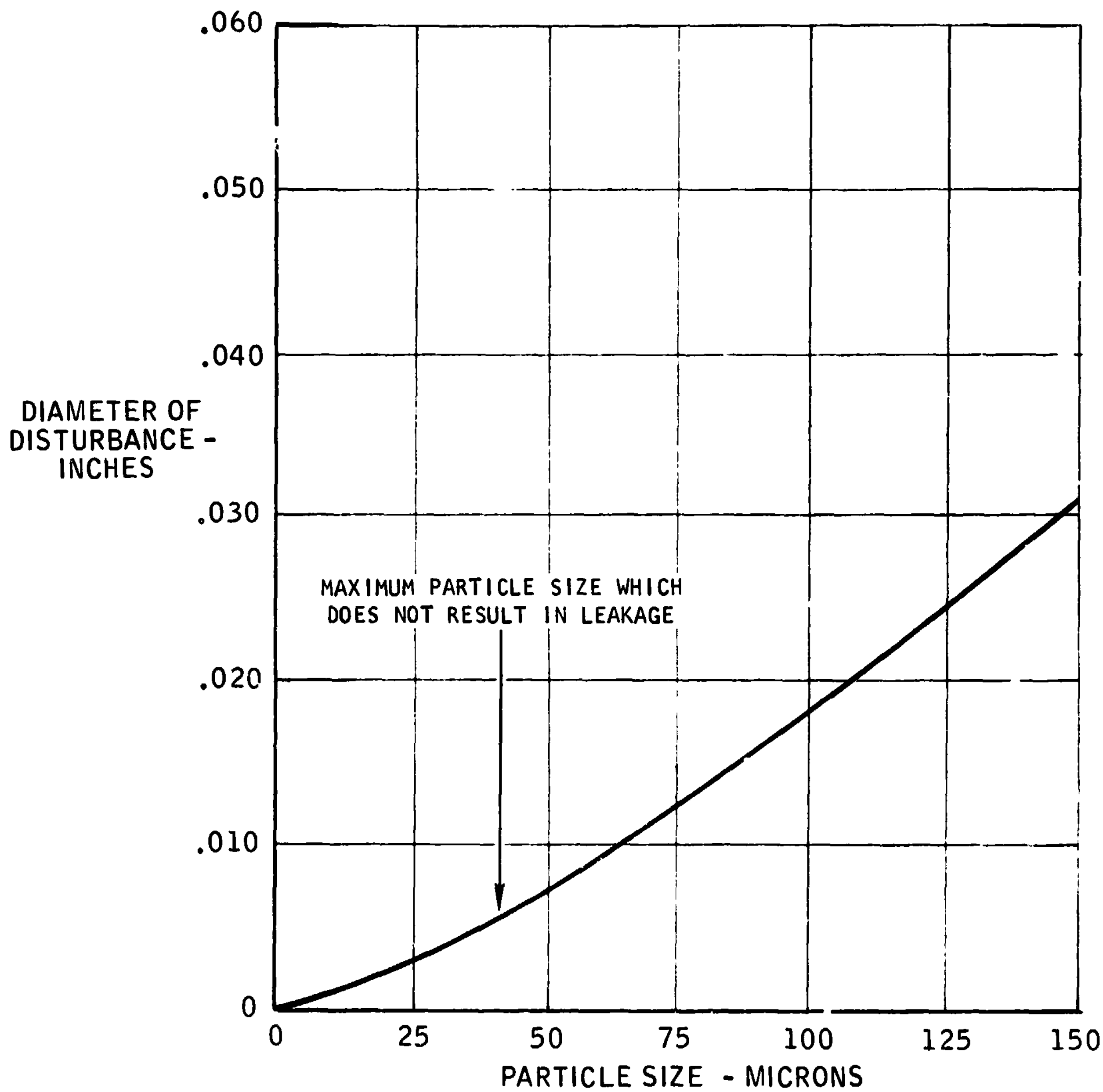


Figure 4-5

particle, it appears unlikely that sufficient particle mass can be trapped between the seat land and the poppet to cause any appreciable separation between the two sealing surfaces and to result in leakage. This is evident from Figure 4-5 which shows that it takes a diameter of disturbance greater than 0.005 inch to cause any appreciable leakage. Therefore, if a particle greater than 40 microns in size (0.0016" diameter) is trapped between the sealing surfaces of a sealing closure featuring less than 0.005 inch wide sealing land, the particle mass will be simply squeezed out adjacent to the sealing land and the particle mass finally remaining between the two sealing surfaces will be small enough not to cause any appreciable separation of the sealing surfaces. In other words, the relatively narrow sealing land will effectively cut the contaminant particle. This has, of course, been demonstrated with a tungsten carbide seat which featured a land width ranging from 0.0001 to 0.0004 inch. It can, therefore, be concluded that an effective and practical particle cutting sealing land width is between 0.0005 and 0.005 inch. Particle entrapment tests with sealing closure land widths between 0.0006 and 0.004 inch were conducted at Marquardt as reported in Section 7.1.8. Additional testing of contamination cutting configurations of selected materials is being conducted by Gil Tellier under Contract NAS 9-13882.

In summary, the current state-of-the-art of sealing technology, sealing closure wear, and contamination sensitivity, and the necessary analytical tools to design and specify a sealing closure interface which will meet the long life, low leakage, and 150 micron particle contamination tolerance requirements set forth in the Space Shuttle Orbital Maneuvering System Helium Regulator Design and Development Statement of Work is available. Furthermore, the ceramic material technology which appears to be essential to the performance of the proposed program is within the state-of-the-art.

4.2.2.2 Metal Bellows Technology

The combined effects of the external leakage requirement, operating temperature range and compatibility with the respective propellants, limit the selection of the regulated pressure sensing device to either metal bellows or a metal diaphragm. Several aspects of the regulator application result in favorable aspects for a metal bellows or diaphragm:

1. The desire to minimize spring rate and/or deflection over the regulation band usually results in small operating deflections relative to the bellows' or diaphragm's total stroke capability. Therefore, stresses resulting from deflection are relatively low. This also results in a stable and repeatable effective area.

2. With sensed pressure tap located downstream of the flow limiter, pressure surge effects are minimized.

3. The throttling characteristics of a regulator generally result in relatively slow "stroking" of the sensing device, thereby minimizing dynamic loads induced during normal operation.

For an anticipated operating stroke which is a small percentage of the maximum available bellows stroke capability, the cycle life of a metal bellows or diaphragm becomes primarily a function of the pressure generated stresses and the precompression stress. With the design goal of the regulator downstream volumes being capable of withstanding full inlet pressure, the normal operating pressure will generate relatively low stresses if the bellows or diaphragm can withstand full inlet pressure. Exclusive of the design goal, analytical methods are available to optimize pressure stresses and characteristics for achieving the required cycle life.

Marquardt has maintained an active interest in the extensive bellows and diaphragm study efforts of Battelle Memorial Institute under Contract No. AF 04(611)-10532. Marquardt's development of control components incorporating metal bellows and diaphragms has resulted in the development of analytical design techniques, substantiated by test, for long life metal bellows and diaphragms. These analytical methods are based primarily on the work reported in Reference 48. This development experience covered applications requiring effective areas up to 2.4 inches in diameter, temperatures from 200 to 850°R, operating pressures up to 500 psi and cycle life demonstrations in excess of 80,000 cycles. Both welded and hydroformed bellows which were fabricated of a wide variety of materials, have been employed. Many of these are applicable to the Shuttle OMS regulator.

Bellows with effective areas in the range of the seat diameter (approximately 0.5 inch) used to pressure compensate the poppet are well within the current state-of-the-art.

4.2.2.3 Guidance Devices

Reliable means for guiding the poppet during its mating with the seat are essential to the satisfactory performance of the regulator sealing closure. As pointed out previously, the poppet-to-seat alignment, which is substantially influenced by the way the poppet is guided, must be repeatable to minimize interfacial scrubbing during closure and to enhance regulator life. To the best of Marquardt's knowledge, all of the gas regulators built to date have featured some form of a sliding fit to guide the poppet to the seat. Sliding fit guidance has four distinct disadvantages. These are:

- Radial play, which results in poppet/seat interface scrubbing during closure.
- Inherent friction which tends to be destabilizing in the regulator dynamics model and which is difficult to predict.
- Sliding friction which tends to generate contamination, particularly in a high vibration environment.
- Limited life, since wear is occurring.

To eliminate the disadvantages of sliding-fit type guidance, The Marquardt Company has for the past five years been engaged in the development of axial guidance flexures. Axial guidance flexures have been demonstrated by Marquardt ranging in size from 0.625 to 8.375 inches in diameter. Photographs of four of Marquardt's flexures are shown in Figure 4-6. Three of the flexure types shown have been cycled extensively at cryogenic as well as ambient conditions. One flexure type has demonstrated two million cycles each in two test items. These devices are definitely state-of-the-art at Marquardt. The analysis techniques developed to size the axial guidance flexures have shown excellent correlation with experimental data.

Some typical performance characteristics that have been demonstrated with axial guidance flexures are as follows:

• Axial Spring Rates	30 - 600 lbs per inch
• Radial Spring Rates	10,000 - 300,000 lbs per inch
• Strokes	Up to 1 inch
• Cycle life	Over 2 million cycles

While the axial guidance flexures are all designed by The Marquardt Company, Marquardt has also employed pivot flexures developed by the Bendix Corporation. An artist's conception of a pivot flexure is shown in Figure 4-7. These pivot flexures take the place of a hinge or pin joint in the same manner that the axial guidance flexures take the place of sliding sleeve-type bearings. These pivot flexures have the same advantages as the axial guidance flexures; namely, elimination of radial clearances, unpredictable friction forces, and contamination generation. Extensive data on spring rates and flexure load capabilities and life is available from the Bendix Corporation.

4.2.3 Regulator Concepts and Arrangements

The literature search and responses to industry data solicitations did not result in the disclosure of any additional basic configurations of modulating pressure regulators than those presented in the Marquardt proposal (Ref. 7). The two-stage series (roughing and fine) configuration was deleted from consideration on the basis of its complexity and projected weight penalty and effort concentrated on three basic configurations.

a) Single stage direct acting

b) Single stage direct acting with lever arm

c) Integral pilot, dome loaded

AXIAL GUIDANCE FLEXURES

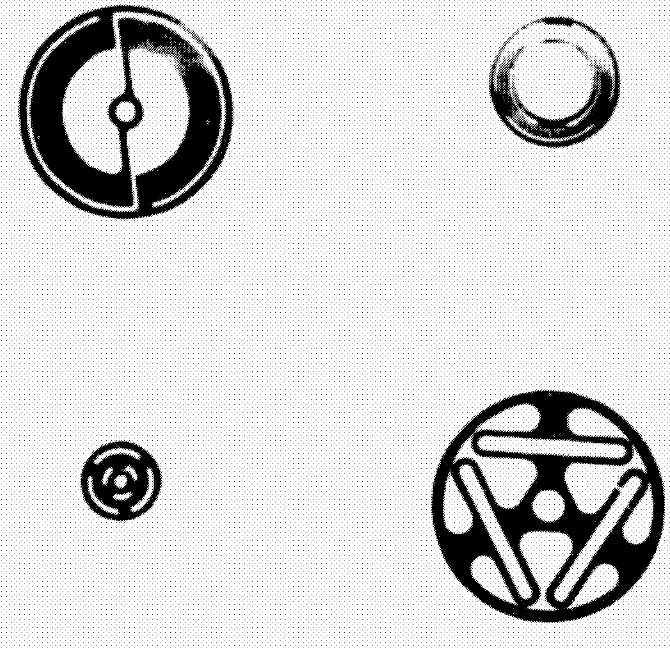


Figure 4-6

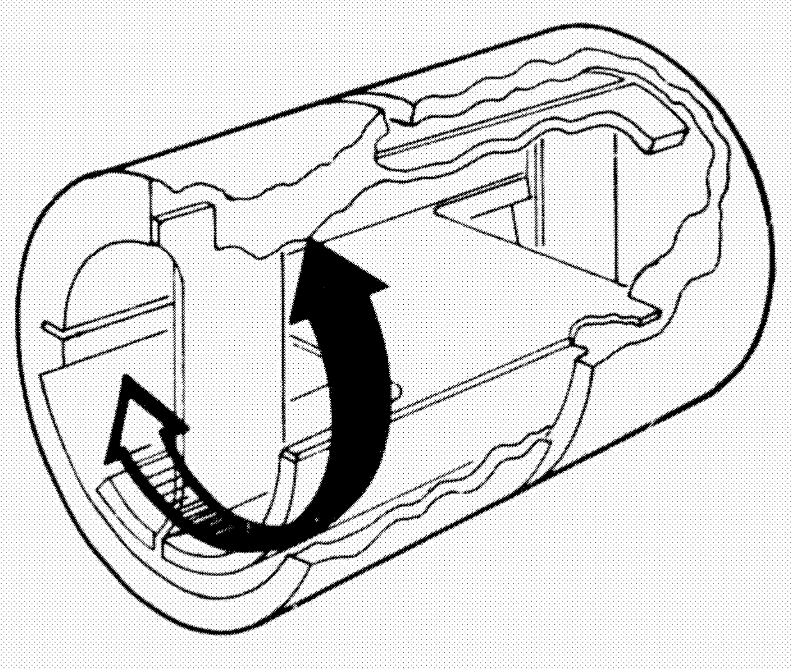


Figure 4-7

The establishment of nominal design conditions for meeting regulator performance characteristics was a necessary prerequisite to developing the analytical programs. OMS Propulsion System Thruster design criteria were used to develop the required helium flow rate. For the anticipated thrust level of 6000 LBF and O/F of 1.6

$$\dot{W} = \frac{F}{I_{SP}} = \frac{6000}{312} = 19.2 \text{ lb/sec}$$

$\dot{W}$ = propellant flow rate - lb/sec

F = thrust-L_{BF}

I_{SP} = specific impulse - sec

For O/F = 1.6 the volumetric flow rate of fuel equals the volumetric flow rate of oxidizer

$$\dot{W}_F = \text{fuel flow rate} = \frac{\dot{W}}{O/F + 1} = \frac{19.2}{2.6} = 7.4 \text{ lb/sec}$$

$$\dot{W}_{OX} = \text{ox. flow rate} = \dot{W}_F \times 1.6 = 7.4 \times 1.6 = 11.8 \text{ lb/sec}$$

For a propellant feed pressure of 250 psia and a pressurant temperature of -150 to -150°F, pressurant density at the temperature extremes are:

$$@ -150°F, \rho_{-150} = .0104 \frac{250}{14.7} \cdot \frac{528}{310} = .3 \text{ lb/ft}^3$$

$$@ +150°F, \rho_{+150} = .0104 \frac{250}{14.7} \cdot \frac{528}{610} = .153 \text{ lb/ft}^3$$

$$\dot{V}_{OX} = \dot{V}_F = \frac{\dot{W}_{OX}}{62.4 \times 1.46} \times 60 = 7.8 \text{ ft}^3/\text{min}$$

$$\dot{V}_{Total} = \dot{V}_{OX} + \dot{V}_F = 2\dot{V}_{OX} = 15.6 \text{ ft}^3/\text{min}$$

where $\dot{V}$ = volumetric flow rate ft^3/min

Pressurant mass flow rate $\dot{W}_{He} = \dot{V}_{Total} \times \rho_{\text{at temp}}$

$$@ +150°F \quad \dot{W}_{He} = 15.6 \times .153 = 2.39 \text{ lb/min}$$

$$@ -150°F \quad \dot{W}_{He} = 15.6 \times .300 = 4.68 \text{ lb/min}$$

The flow of helium gas through any kind of restriction (poppet/seat interface, orifice, or nozzle) is defined by the steady-state, one-dimensional, isentropic thermodynamic relationships as follows:

$$\dot{W} = \frac{P_{t_1} A C_m C_D}{\sqrt{T_t}}$$

The flow function, C_m, is defined from isentropic relationships as follows:

$$C_m = \sqrt{\frac{\gamma g}{R}} \; M \left(1 + \frac{\gamma - 1}{2} M^2\right)^{-\frac{\gamma + 1}{2(\gamma - 1)}}$$

The flow function for helium is shown in Figure 4-8 based upon numerical solutions of the thermodynamic functions published in Reference 45. The flow function is stated as follows:

$$C_m = f\,(P_{t_1}, P_2)$$

Thus, the flow function value is dependent upon the static-to-stagnation pressure ratio across the restriction in the pressure ratio range of 1.0 to 0.4867. The flow function in this range varies from 0 to 0.2098 lbm-$°R^{1/2}$/lbf-sec, respectively. At a pressure ratio of 0.4867, the flow is critical through the restriction, that is, the Mach number (M) is unity. This represents the maximum flow function value achievable, and the flow function remains constant for all pressure ratios of 0.4867 and less.

The discharge coefficient of the restriction is dependent upon the configuration and static-to-total pressure ratio. The discharge coefficient for the poppet/seat interface and a sharp-edged orifice is shown in Figure 4-9. The poppet seat interface test data is based upon experimental results obtained by Marquardt from flowing gaseous hydrogen and nitrogen at various pressure ratios through interfaces similar to the design proposed in the regulator. The test results indicate a maximum valve discharge coefficient of 0.943 is attained for pressure ratios less than 0.225. At higher pressure ratios, the discharge coefficient of the poppet/seat interface decreases until reaching a minimum extrapolated value of 0.544 at a pressure ratio of unity. The sharp-edged orifice experimental data is also shown in Figure 4-9 for comparison to the poppet/seat interface. The orifice data is taken from test results published in Reference 46. The sharp-edged orifice displays the same general trends. However, the discharge coefficient is lower at low pressure ratios and very slightly higher at high pressure ratios when compared to the poppet/seat interface. In addition to the discharge coefficient (C_D) being a function of pressure ratio, some dependence on the flow area ratio (annular radial flow area to axial flow area) is evidenced. This dependence on area ratio is also illustrated in Figure 4-9 and has been substantiated by experimental work at Marquardt.

HELIUM FLOW FUNCTION

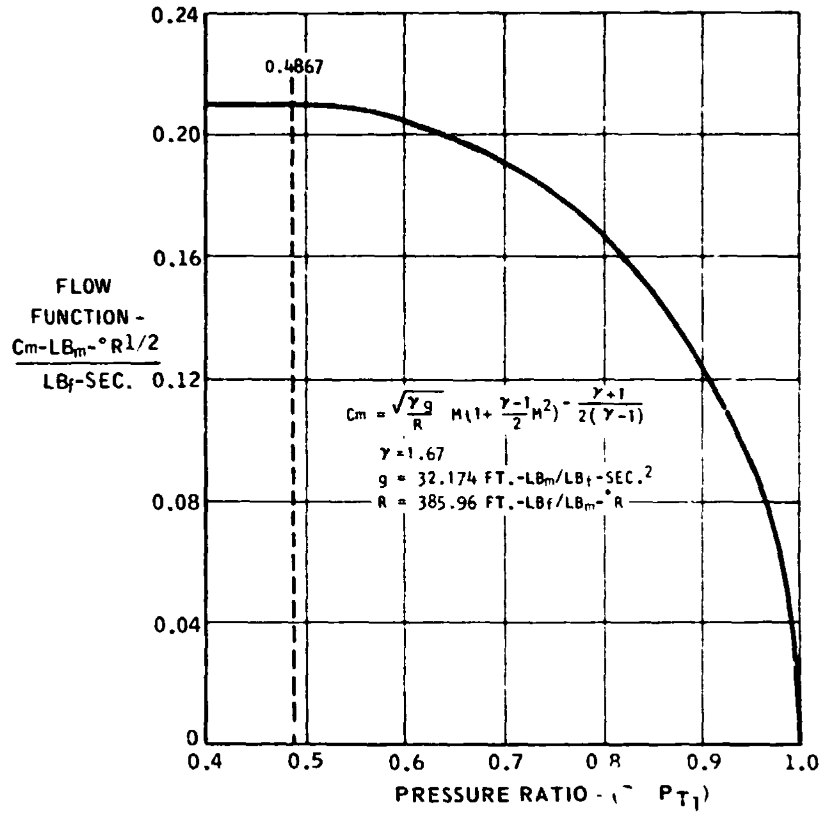


Figure 4-8

POPPET/SEAT INTERFACE DISCHARGE COEFFICIENT

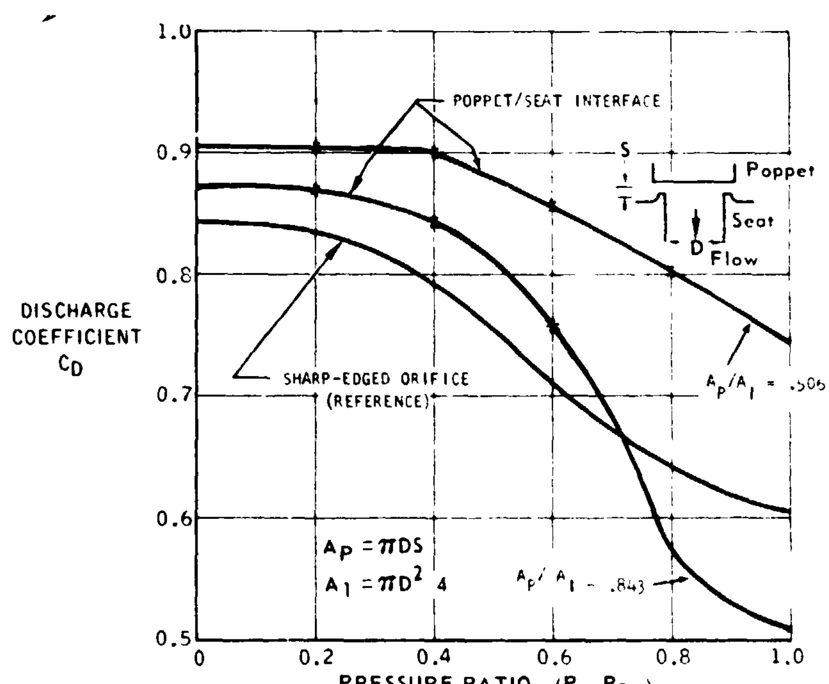
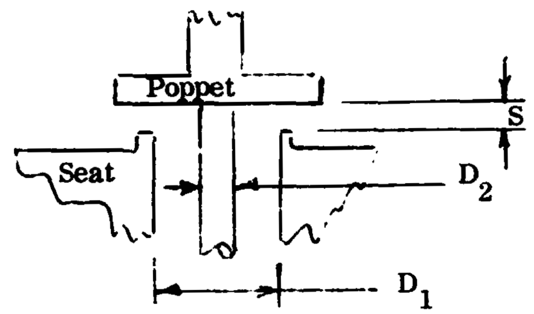


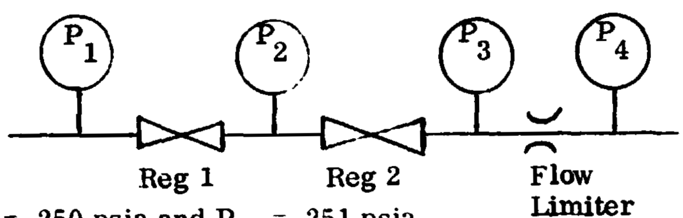
Figure 4-9

For purposes of establishing an initial regulator size, the following geometric relationships were assumed:



At max stroke(s) $\dfrac{\pi D_1 S}{\frac{\pi}{4}(D_1^2 - D_2^2)} = 0.3$

For two regulators, full open, in series and a flow limiter downstream of the second regulator with an assumed 1 psi pressure drop at max flow worst case conditions:



For nominal P_4 = 250 psia and P_3 = 251 psia

P_1 = 400 psia (minimum inlet pressure)

To determine: $A_{Reg\ 1} = \dfrac{\dot{W}\sqrt{T}}{P_1 C_M C_D}$

assume $P_1 - P_2 = P_2 - P_3$ (equal pressure drops across regulators)

Then

$400 - P_2 = P_2 - 251 \qquad P_2 = 325.5$

$\dfrac{P_2}{P_1} = \dfrac{325.5}{400} = .813$

From Figures 4-8 and 4-9:

$$C_m = .168$$

$$C_D \approx .82$$

$\therefore$ @ − 150°F

$$A_{Reg\ 1} = \frac{4.68\ \sqrt{310}}{60\ 400 \times .168 \times .82} = .0249\ in^2$$

$\therefore$ @ + 150°F

$$A_{Reg\ 1} = \frac{2.31\ \sqrt{610}}{60 \times 400 \times .168 \times .82} = .0172\ in^2$$

For Regulator No. 2

$$\frac{P_3}{P_2} = \frac{251}{325.5} = .772$$

From Figures 4-8 and 4-9:

$$C_m = .175$$

$$C_D = .83$$

$\therefore$ @ − 150°F

$$A_{Reg\ 2} = \frac{4.68\ \sqrt{310}}{60 \times 400 \times .175 \times .82} = .0239\ in^2$$

Add 10% flow area for contingency for assuming equal ΔP's.

For a maximum required flow area of $.0274\ in^2$ and the geometric relationships assumed,

$$\frac{\pi D_1 S}{\frac{\pi}{4}(D_1^2 - D_2^2)} = 0.3 \quad \text{and} \quad \pi D_1 S = .0274\ in^2$$

$$\frac{\pi D_1 S}{\frac{\pi}{4}\left[D_1^2 - (.5\ D_1)^2\right]} = \frac{\pi D_1 S}{\frac{\pi}{4}\ .75\ D_1^2} = .3$$

$$.0274 = .3 \frac{\pi}{4} .75 D_1^2$$

$$D_1^2 = .155 \qquad D_1 = .394$$

$$\pi D_1 S = .0274$$

$$S = \frac{.0274}{\pi x .394} = .0221 \text{ in.}$$

As design details are developed which may effect key dimensions in the poppet seat area, the analysis is repeated. However, for preliminary sizing, this analysis is considered valid. For various other area ratios, the seat diameter (D_1) and stroke (S) relationships are shown in Figure 4-10.

Flow limiting may be achieved by several methods. Schematics of three candidate devices which provide limiting of maximum flow are shown in Figure 4-11. They break down into two groups: active and passive. The first shown is the flow regulator which is an active device. Whenever the output flowrate of helium approaches or attains a value of 10 lbm/min. the differential pressure across the orifice increases to a preset value. The differential pressure acts on the effective area of the actuator to overcome the spring force. This action tends to close the poppet valve. The action of the flow regulator is to never allow the pressure drop across the orifice to exceed the value commensurate with a flow rate of 10 lbm/min. Under normal conditions, the flow regulator poppet is wide open and the device is inactive. Under flow limiting conditions, the orifice pressure drop is limited to the preset value. The flow regulator requires four (4) elements for its operation as a flow limiter: orifice, actuator, spring, and poppet/seat interface.

The temperature compensated nozzle is another active device which affects flow rate limiting. Under normal regulator operation, that is, flow rates of 1 to 6 lbm/min, the nozzle throat operates in an unchoked mode. However, as the maximum allowable flow is approached or attained (10 lbm/min), the nozzle throat operates in a critical mode (Mach number of unity) thereby limiting the flowrate. The limiting operation is described as follows:

$$\dot{W} = P_{t_1} K_T C_m C_D$$

where: K_T - Temperature compensation constant, in.^{2}/°R$^{1/2}$

PREDICTED SEAT DIAMETER & STROKE DESIGN POINT RANGE

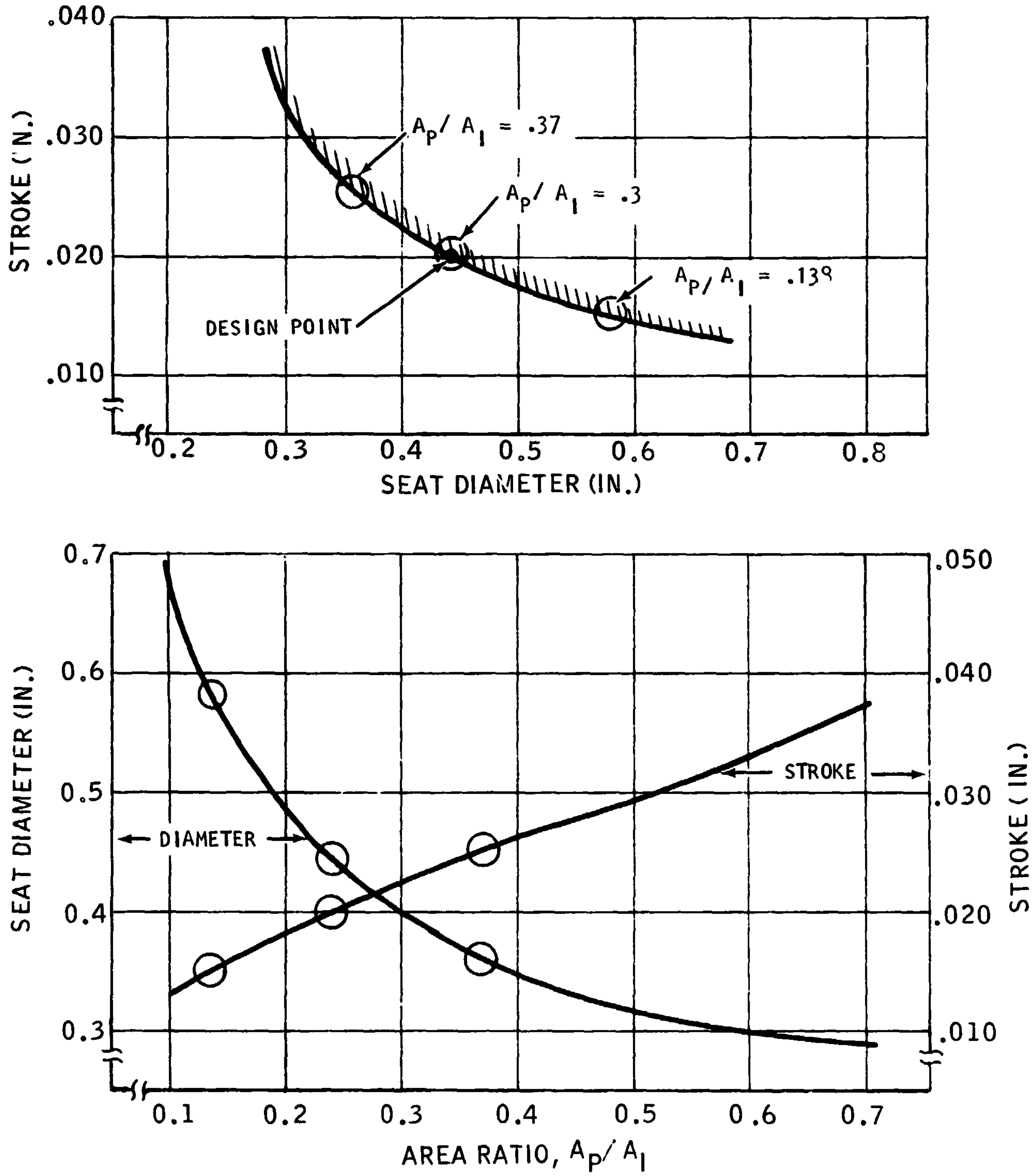
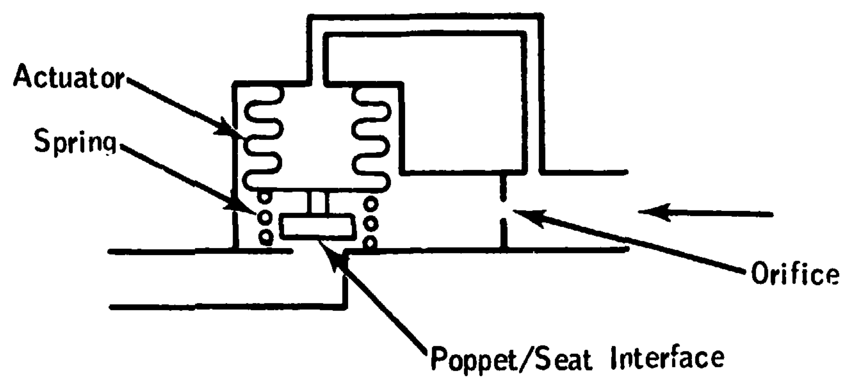


Figure 4-10

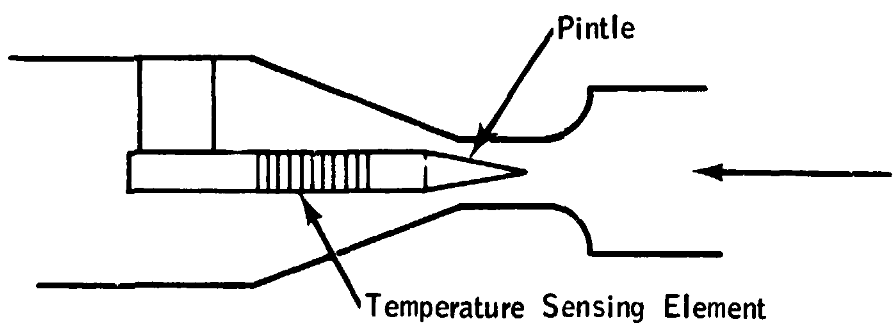
CANDIDATE FLOW LIMITER SCHEMATIC

• ACTIVE

FLOW REGULATOR



TEMPERATURE COMPENSATED NOZZLE



• PASSIVE

UNCOMPENSATED NOZZLE

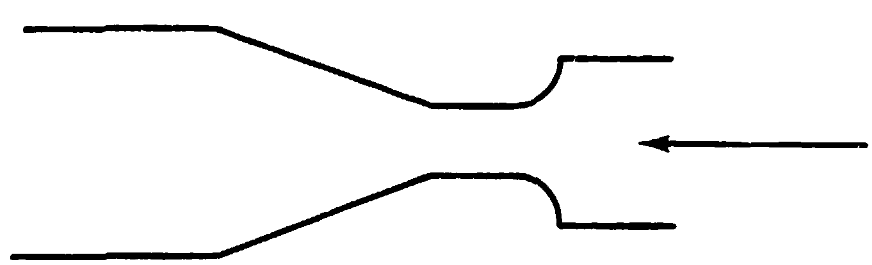


Figure 4-11

By sensing the temperature and varying the throat area of the nozzle by means of the pintle according to the following law the temperature compensation constant may be determined.

$$K_T = \frac{A^*}{1/\sqrt{T_t}}$$

The temperature compensated nozzle requires three (3) elements to operate as a flow limiter: nozzle, temperature sensing element, and pintle.

The nozzle discharge coefficient is shown in Figure 4-12, as a function of Reynolds number. The discharge coefficient of a small nozzle ($\frac{1}{8}$" throat diameter) is constant above a Reynolds number of 40,000 and decreases slightly for lower Reynolds numbers. The nozzle Reynolds number is defined as follows:

$$R_e = \frac{\rho^* V^* D^*_H}{\mu}$$

The Reynolds number of the nozzle is affected by the presence of the pintle in the throat. This effect is considered in the definition of the hydraulic diameter:

$$D^*_H = \frac{4A^*}{P_w}$$

Therefore, the nozzle discharge coefficient varies only slightly with Reynolds number and does not impair the operation of the nozzle when used as a flow limiter.

An uncompensated nozzle may be used as a flow limiter as shown in Figure 4-11. It is a passive device, has only one element (the nozzle), and has no moving parts. Under normal operation (flow rates of 1 to 6 lbm/min) the nozzle is unchoked, and the flow rate through the pressure regulator is dependent upon normal regulator operation and system propellant flow rates. As the flow limit is approached or attained (10 lbm/min), the nozzle throat operates in a critical mode thereby limiting the flow rate as follows:

$$\dot{W} = \frac{P_{t_1} A^* C_m C_D}{\sqrt{T_t}}$$

Since the helium temperature varies from 150°F to -150F, the limited flowrate is slightly less at 150°F than at -150°F. However, the limited flow rate at 150°F is always greater than the maximum normal flow rate (6 lbm/min) and less than the flow limit. Therefore, the uncompensated nozzle does not impair normal pressure regulator operation and always limits the flow rate to 10 lbm/min or less, with no moving parts.

NOZZLE DISCHARGE COEFFICIENT

REFERENCE 47

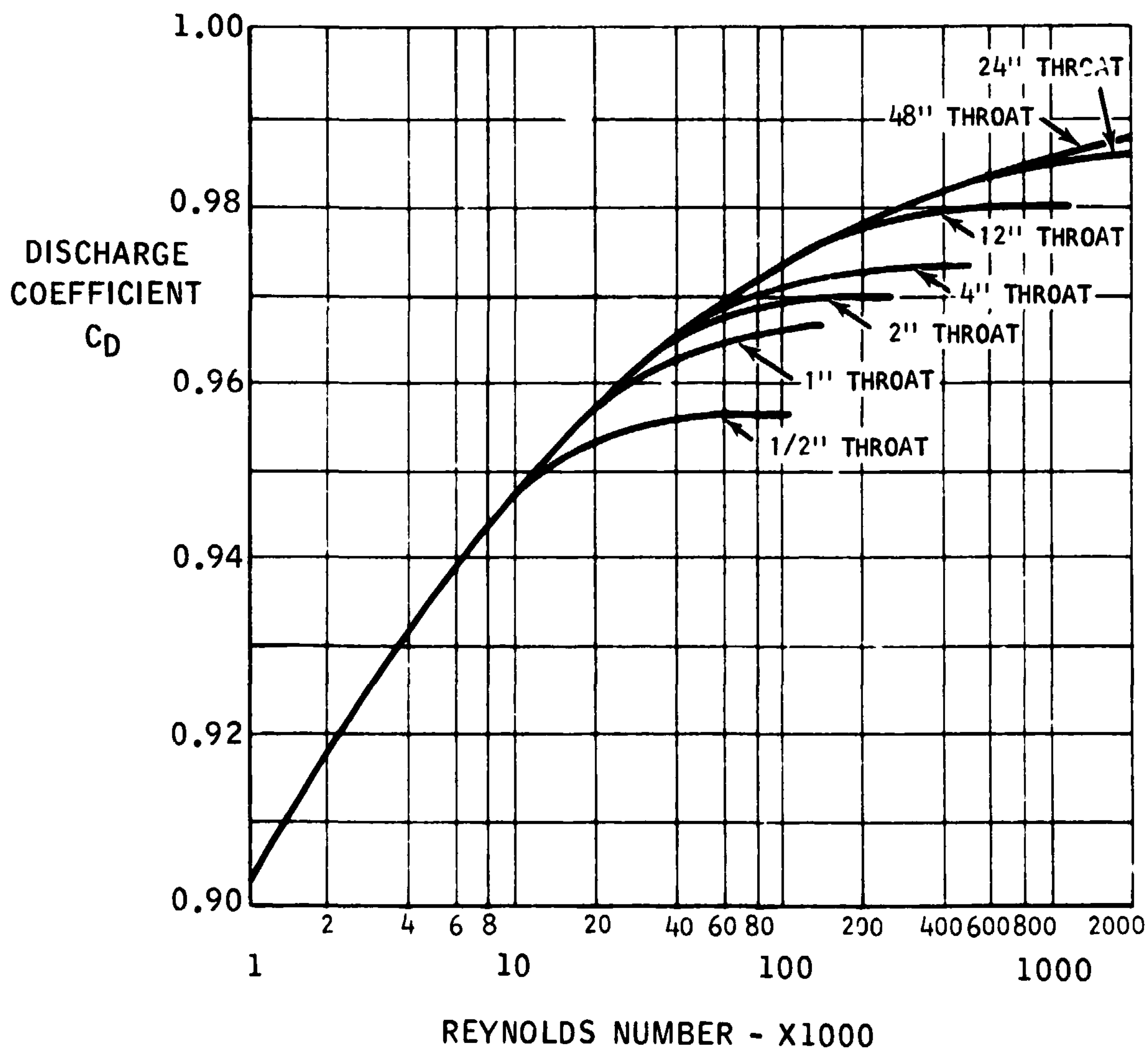


Figure 4-12

The deadband or regulator droop is a function of the system friction and apparent spring rate of the regulator. The apparent spring rate of the device is determined from several factors including mechanical spring rates and the effective spring rate caused by poppet flow forces. These flow forces are difficult to analyze and empirical data for the actual design configuration must be evaluated to establish precise mechanical spring rates. The friction in the proposed designs will have little influence on deadband due to the use of flexures for support and guidance and the use of bellows for sealing, both of which eliminate the sliding friction. It should be noted, however, that some damping is obtained from the bellows and spring inherent material characteristics to help in overall stability.

The allowable deadband of 8 psi for each regulator will be fully utilized to maximize overall spring rate, hence, highest possible mechanical natural frequency. The setup shown in Figure 1-13 represents the configuration that will be simulated on the computer to evaluate set points, system dynamics and failure mode effects.

The arrangement of set points and the influence of deadbands on crosstalk is shown in Figure 4-14. Regulators Nos. 1 and 2 are in series in one unit, and Regulators Nos. 3 and 4 are in series in the other unit. The two units are installed in parallel to comprise the system. Regulator No. 1 is the normally operating regulator and has a set point of 250 lbf/in^2, which is also the design set point of the system. Regulators 2, 3 and 4 have set points of 258 lbf/in^2, 238 lbf/in^2, and 246 lbf/in^2, respectively. For illustration, each regulator is assumed to have a deadband equal to the design goal, that is, ± 4 lbf/in^2. The following system requirements are met by this arrangement: (1) deadband, (±4 lbf/in^2); (2) set point difference, (20 lbf/in^2); and (3) minimum set point, (238 lbf/in^2). Also, problems with crosstalk are avoided by the system arrangement. Since Regulators 1 and 2 are in series, when Regulator 1 is operating, Regulator 2 should remain on the wide open stop. The range of outlet pressures of Regulator 1 is 254 lbf/in^2 to 246 lbf/in^2 (the set point plus and minus the maximum deadband). Regulator 2 has a range of outlet pressures from 256 lbf/in^2 to 262 lbf/in^2. Therefore, the operating ranges of Regulators 1 and 2 do not overlap, these two regulators do not both operate at the same time, and there is no crosstalk, since Regulator 2 always remains on the wide open stop while Regulator 1 is modulating. The same arguments and logic applies to Regulators 3 and 4. It is important to note, however, that there is passive crosstalk between regulators 1 and 4 under normal operating conditions. There is a 50% overlap in outlet pressures. However, Regulators 1 and 4 are in opposite parallel units. Therefore the crosstalk is passive, that is, when Regulator 1 is operating in the 24[illegible] to 2[illegible] lbf/in^2 range, Regulator 4 poppet stroke will be responding. However, Regulator 3 has that circuit shutoff at these conditions, and there is no effect on outlet pressure or flow rate.

REGULATOR SYSTEM BLOCK DIAGRAM

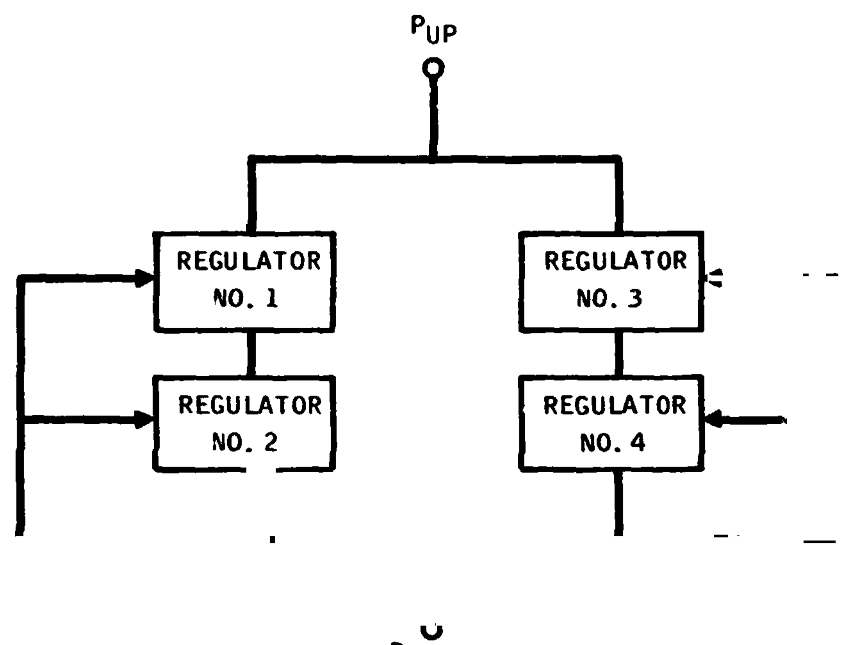


Figure 4-13

SET POINT AND DEADBANDS

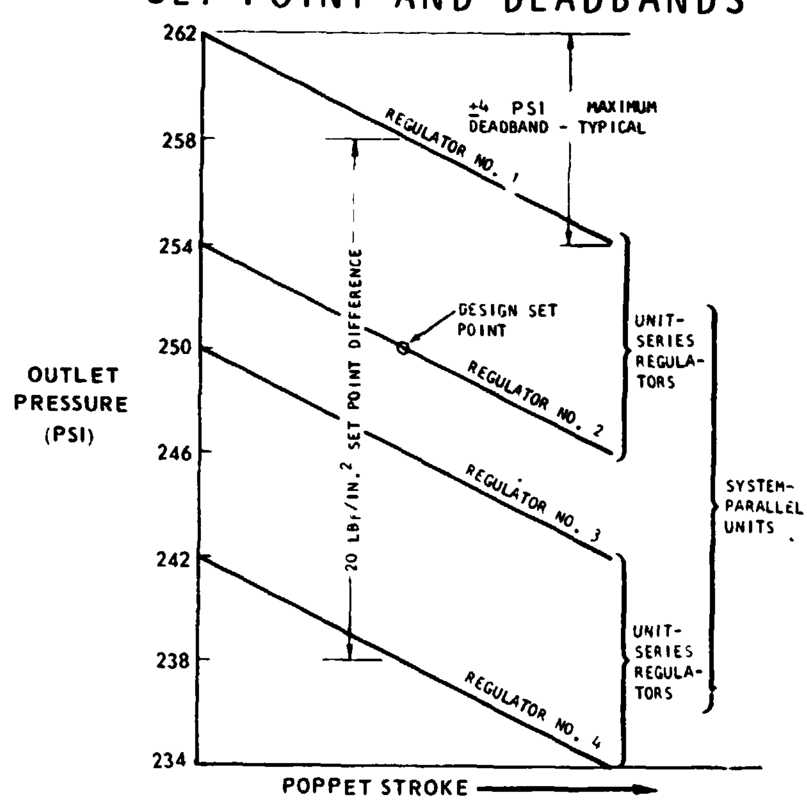
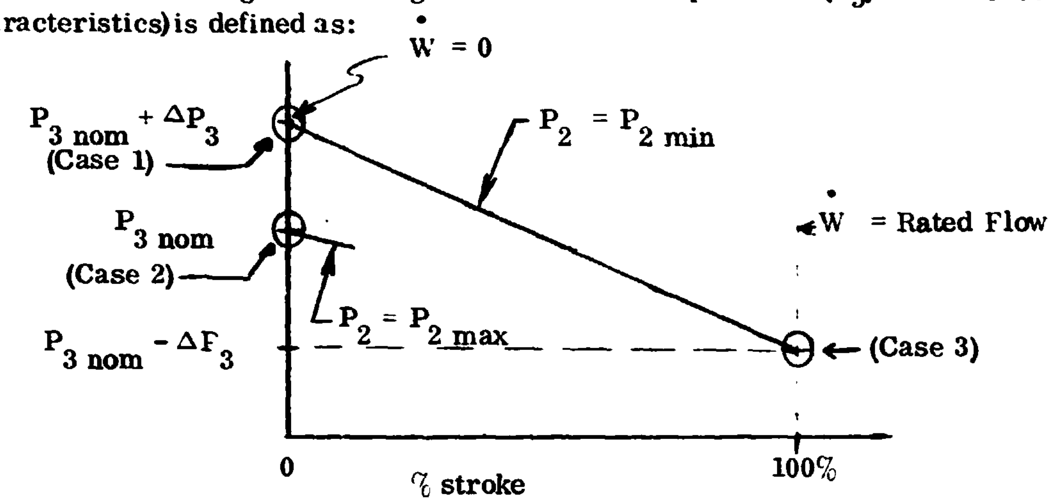


Figure 4-14

To resolve initial design criteria a static analysis, at specific conditions, is developed for the regulator configuration of Figure 4-15. Though this is a direct acting regulator with a lever arm, it is valid for the direct acting configuration, without lever arm, by reducing the lever arm ratio to 1 and also valid for the pilot operated configuration by replacing the force generated by the actuator spring with the pilot pressure, acting on the actuator bellows effective area.

For the regulator of Figure 4-15 the outlet pressure (P_3) versus poppet stroke (droop characteristics) is defined as:



ΔP_3 = 1/2 Allowable Deadband

Force balance equations for the 3 levels of outlet pressure (P_3) are as follows:

<u>(Case 1)</u>

$$\underline{P_3 = P_{3\,nom} + \Delta P_3, \quad \dot{W} = 0, \quad P_2 = P_{2\,min}}$$

$$\Sigma F = 0 = (P_{3\,nom} + \Delta P_3)\, A_{BE} - F_o + \frac{A_{SE}}{a/b}\left[P_{2\,min} - (P_{3\,nom} + \Delta P_3)\right]$$

$$- \left[\frac{P_{2\,min} - (P_{3nom} + \Delta P_3)\ (1 + \frac{\Delta A_S}{A_S})\ A_S}{a/b}\right]$$

$$\text{where } \frac{\pi}{4}\,(D_{BV})^2 = A_{BV} = \left(1 + \frac{\Delta A_S}{A_S}\right) A_S$$

$$\Delta A_S \text{ is } (-) \text{ when } \frac{\pi}{4}\, D_{BV}^2 < A_{SE}$$

SINGLE STAGE BALANCED POPPET REGULATOR

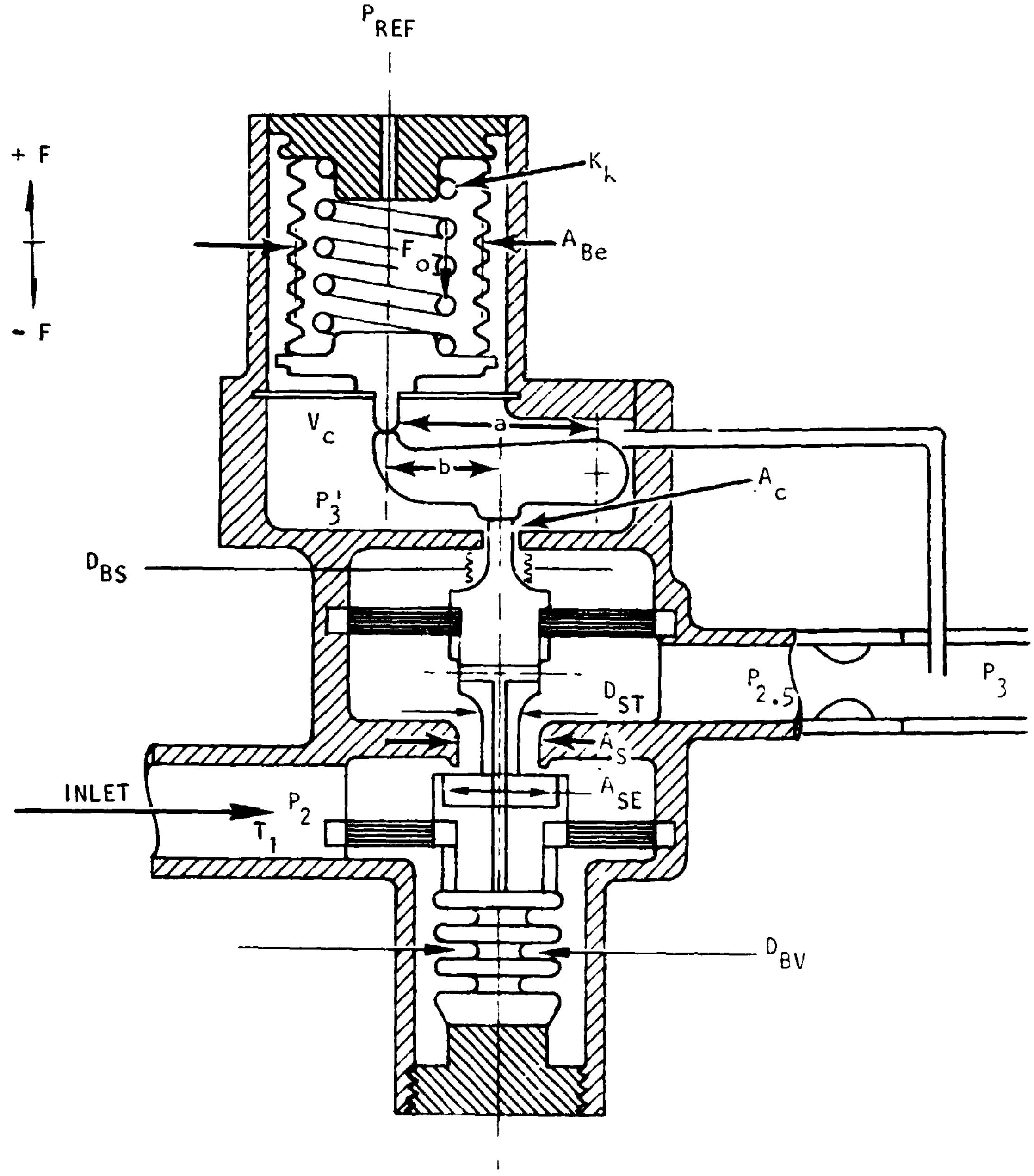


Figure 1-15

(Case 2)

$P_3 = P_{3\,nom}$, $\dot{W} = 0$

$$\sum F = 0 = P_3 A_{BE} - F_0 + \frac{A_{SE}}{a/b}\left[P_{2\,max} - P_3\right] - \frac{\left(P_{2\,max} - P_3\right)\left(1 + \frac{\Delta A_S}{A_S}\right) A_S}{a/b}$$

(Case 3)

$P_3 = P_{3\,nom} - \Delta P_3$, $\dot{W}$ = Rated Flow , $P_2 = P_{2\,min}$

$$\sum F = 0 = \left(P_{3\,min} - \Delta P_3\right) A_{BE} - F_0 + K_s\, a/b\, (X)\, max + \frac{\frac{\pi}{4} D_{BS}}{a/b}\left[P2.5 - \left(P_{3\,nom} - \Delta P_3\right)\right] + \frac{\left(P_{2\,min} - P2.5\right) A_S}{a/b} \text{ (Flow Force Function)}$$

$$- \frac{\left(P_{2\,min} - P2.5\right)\left(1 + \frac{\Delta A_S}{A_S}\right) A_S}{a/b}$$

where K_s = System spring rate (lb/in.), and

$$P2.5 = P_{3\,nom} - \Delta P_3 - \Delta P_{Flow\ Limiter}$$

from (Case 1) - (Case 2)

$$A_{BE} = \frac{- A_S \frac{\Delta A_S}{A_S}\left[P_{2\,max} - P_{2\,min} + \Delta P_3\right]}{a/b\, \Delta P_3}$$

Case 3 can now be solved for K_s where

$$K_s = \frac{F_0}{a/b\ X_{max}} \quad \frac{\left(P_{3\,nom} - \Delta P_3\right) A_{BE}}{a/b\ X_{max}} - \frac{\left(P_{2\,min} - P2.5\right) \frac{\pi}{4}\left(D_{BS}\right)^2}{a/b\ X_{max}}$$

$$- \frac{P_{2\,min} - P_{2.5}\; A_S}{a/b\; X_{max}} \quad \text{(Flow Force Function)}$$

$$+ \frac{\left(P_{2\,min} - P_{2.5}\right)\left(1 - \dfrac{\Delta A_S}{A_S}\right) A_S}{a/b\; X_{max}}.$$

A flow force function for a flat seat interface unbalanced poppet was developed from empirical data (Reference 6) and is shown in Figure 4-16, along with a plot of data from a TMC test. In both of these cases, great liberties were taken to achieve a linear function and the accuracy of the data, particularly in the low area ratio range, is questionable. Since the anticipated regulator operating area ratio $\left(\frac{A_P}{A_T}\right)$ is 0.0 to 0.3, a more rigorous analysis is warranted. Utilizing continuity, momentum and energy equations for compressible fluid flow, relationships were developed to determine aerodynamic properties of the fluid stream as it is throttled through the metering orifice. Pressure forces are then integrated over the respective areas to determine the net forces acting on a "balanced poppet." The analysis was adapted to the Marquardt APL computer terminal (Appendix A) to facilitate the iteration of dimensional relationships such that the optimum configuration can be established and the impact of tolerances on performance evaluated. This program has been iterated to evolve dimensional criteria which results in optimum dynamic characteristics.

The analytical approach to meeting the internal leakage and contamination tolerance requirements is expounded in section 4.2.2 of this report. From that analysis, a seat preload is determined which must exist at the seat/poppet interface at the regulator lock-up pressure. By specification definition, lock-up must occur at a pressure no greater than 15 psi above the nominal set point, or 11 psi above the maximum regulated pressure. From the analysis to develop the required actuator effective diameter (A_{BE}) to meet regulation deadband, the maximum seat preload capability of the regulator can be determined from the relationship:

$$F_{preload\,max} \quad A_{BE} \times 11 \text{ psia}$$

Since there is, theoretically, zero flow and zero stroke at the maximum regulated pressure, the full seat preload is transmitted to the actuator through the "push rod" mechanism. As outlet pressure rises, this preload must be transferred from the push rod to the seat/poppet interface, and is accomplished by outlet pressure acting on the actuator effective area. In the event $F_{preload\,max}$ does not exceed the seat preload required to meet the leakage and contamination tolerance criteria, a larger bellows effective area (A_{BE}) must be employed

FLOW FORCE RELATIONSHIP FOR UNBALANCED FLAT POPPET & SEAT

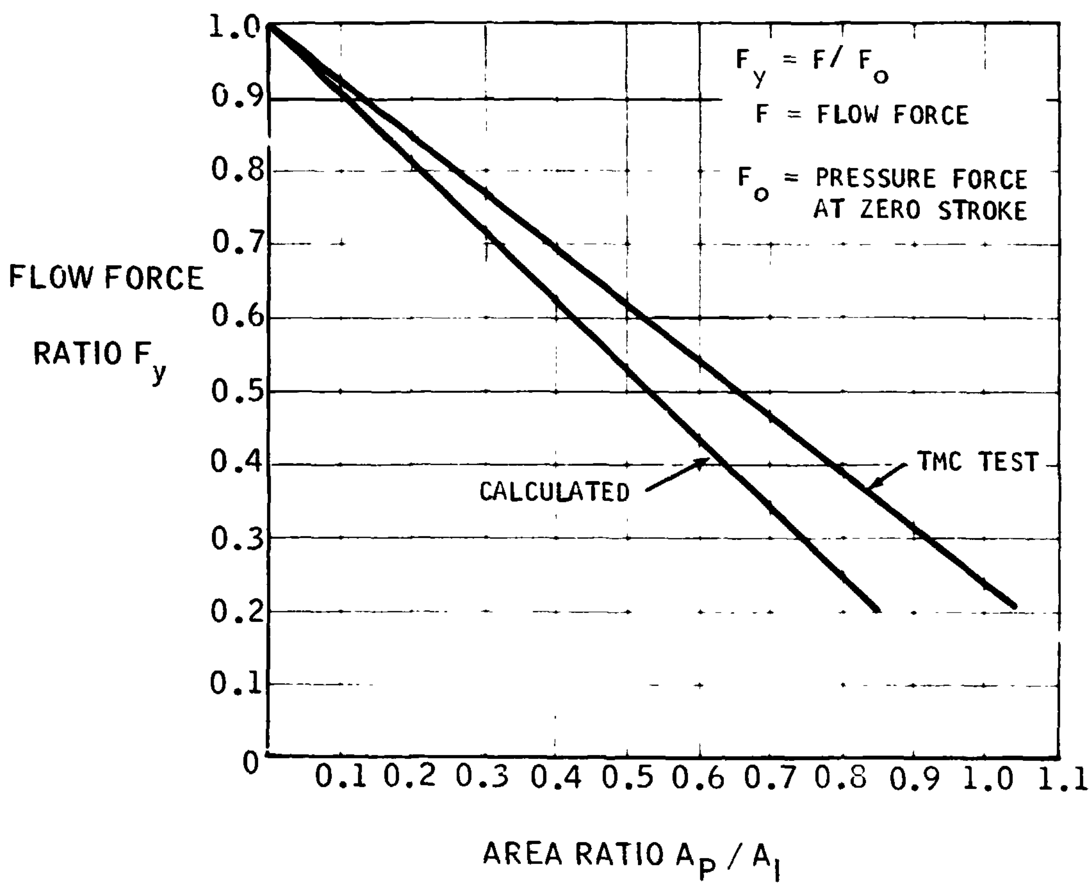


Figure 4-16

and the regulator design iterated, or the seat detail design modified to be compatible with a reduced preload.

The sizing of inlet and outlet port or line size is based upon maintaining a fluid velocity of no greater than Mach 0.2 at worst case conditions. From continuity relationships:

$\dot{W} = \rho V A C_d$

where $\dot{W}$ = mass flow rate lbm/sec

ρ = density lb/ft^2

A = cross-sectional area - ft^2

C_d = flow coefficient

V = flow velocity ft/sec

at Mach 0.20

$$V = 0.2 \sqrt{KgRT} \qquad \frac{ft}{sec}$$

where

K = specific heat ratio = 1.67 for helium

g = constant = 32.2 ft/sec^2

R = gas constant = 386 $\frac{ft\ lb}{lb\ °R}$

T = temperature - °R

at $\dot{W}$ = 4.68 lb/min. and -150°F

$$A = \frac{4.68 \times 14.7 \times 310 \times 144}{60 \times P \times 0.2 \sqrt{1.67 \times 32.2 \times 386 \times 310} \quad C_D \times 528 \times 0.0104}$$

$$= \frac{18.35}{P \times C_D} \text{ in.}^2$$

For a minimum inlet pressure of 400 psia and C_D = 0.7

A_{inlet} = 0.0656 $in.^2$

D_{inlet} = 0.289 in.

To allow for growth to 6 lb/min. flow capacity increase inlet minimum flow area proportionally

$$D_{\text{inlet min}} = \sqrt{\frac{6.00}{4.68}}\,(.289) = .326 \text{ in.}$$

Sizing the outlet diameter for the same criteria:

$$D_{\text{outlet min}} = \sqrt{\frac{400}{P_{\text{out min.}}}}\; D_{\text{inlet min.}} = \sqrt{\frac{400}{234}} \times .326$$

$$= .425 \text{ in.}$$

For the above derived inlet and outlet port sizes, fluid velocity, at a flow rate of 6.0 lb/min. and + 150°F fluid temperature, would be limited to Mach 0.31 at the respective minimum fluid pressures.

4.2.4 Existing Hardware Evaluation

Of the numerous pressure regulators developed for spacecraft, the most logical candidates, for evaluating their potential application to the Shuttle OMS System, are the various propulsion system pressurizing regulators employed in the Apollo Program. These regulators represent designs qualified for man rated systems and incorporate prevalent technology and state-of-the-art of the last decade. It must be noted, however, that the Apollo Program requirements were substantially less severe than those for the Space Shuttle. This is particularly shown by the total lack of requirements in the case of the Apollo Program for such environments as long term propellant compatibility, contamination tolerance to 150 micron size particles, disallowance of lubricants, and tolerance to icing conditions. In the following table, several design parameters of three of the Apollo regulators are tabulated, along with Shuttle OMS Regulator requirements for the same parameter.

Performance Parameter	LM APS Capability	LM DPS Capability	Apollo SPS Capability	Shuttle OMS Regulator Requirement
Inlet Pressure	3500 to 400 psi	1750 to 320 psi	4000 to 350 psi	4000 to 150 psi above outlet
Outlet Pressure	184 ± 4 psi	246 $\pm^{2}_{3}$ psi	186 ± 4 psi	250 ± 4 psi
Lock-up Pressure	203 psi max.	253 psi max.	200 psi max.	265 psi max.
Flow Rate nom.	1.45 lb/min He @ 30°F	5.5 lb/min He @ 0°F	6 lb/min He @ 60°F	4.68 lb/min He @ -150°F
max.	5.5 lb/min	19 lb/min	9 lb/min.	10 lb/min
Leakage internal	100 SCCH He	250 SCCH He	2400 SCCH He	100 SCCH He
external	1 SCCH He	6 SCCH He	1 SCCH He	1 SCCH He
Op. Temp. Range	-120 to +130°F	-20 to +100°F	-20 to +120°F	-150 to +150°F

On the basis of these performance parameters, the LM DPS regulator, designed and built by Parker Hannifin, was selected for closer evaluation. The LM DPS Regulator is shown schematically in Figure 4-17. As shown in the figure, a regulator unit consists of two regulators in parallel with a common outlet and separate inlets. The main stages and pilot stages are contained within a common cast metal housing. The regulator is of all-metal construction, including both the pilot and main stage pressure unbalanced poppets.

An active flow limiter is located in the regulator inlet. The spring loaded poppet responds to aerodynamic drag loads of high flow rates and moves proportional to the drag load to restrict the inlet flow passage and thus limit mass flow rate.

Although the flow rate, set point and regulation deadband of the DPS regulator appear compatible with the Shuttle OMS requirement, a review of the qualification Test Report (Reference 4) indicates several marginal areas where design deficiencies need resolution. In particular, extrapolation of the test data indicates a regulation deadband of 13 psi would result from increasing the inlet pressure range to 4000 to 400 psia, and operating temperature range to -150° to +150°F. The short term propellant compatibility test indicated potential long term exposure problems. Although the regulator incorporates an integral inlet filter, there is strong evidence to indicate a low contamination tolerance capability of the poppet/seat seal surfaces.

DESCENT PROPULSION SECTION - PRESSURE REGULATOR

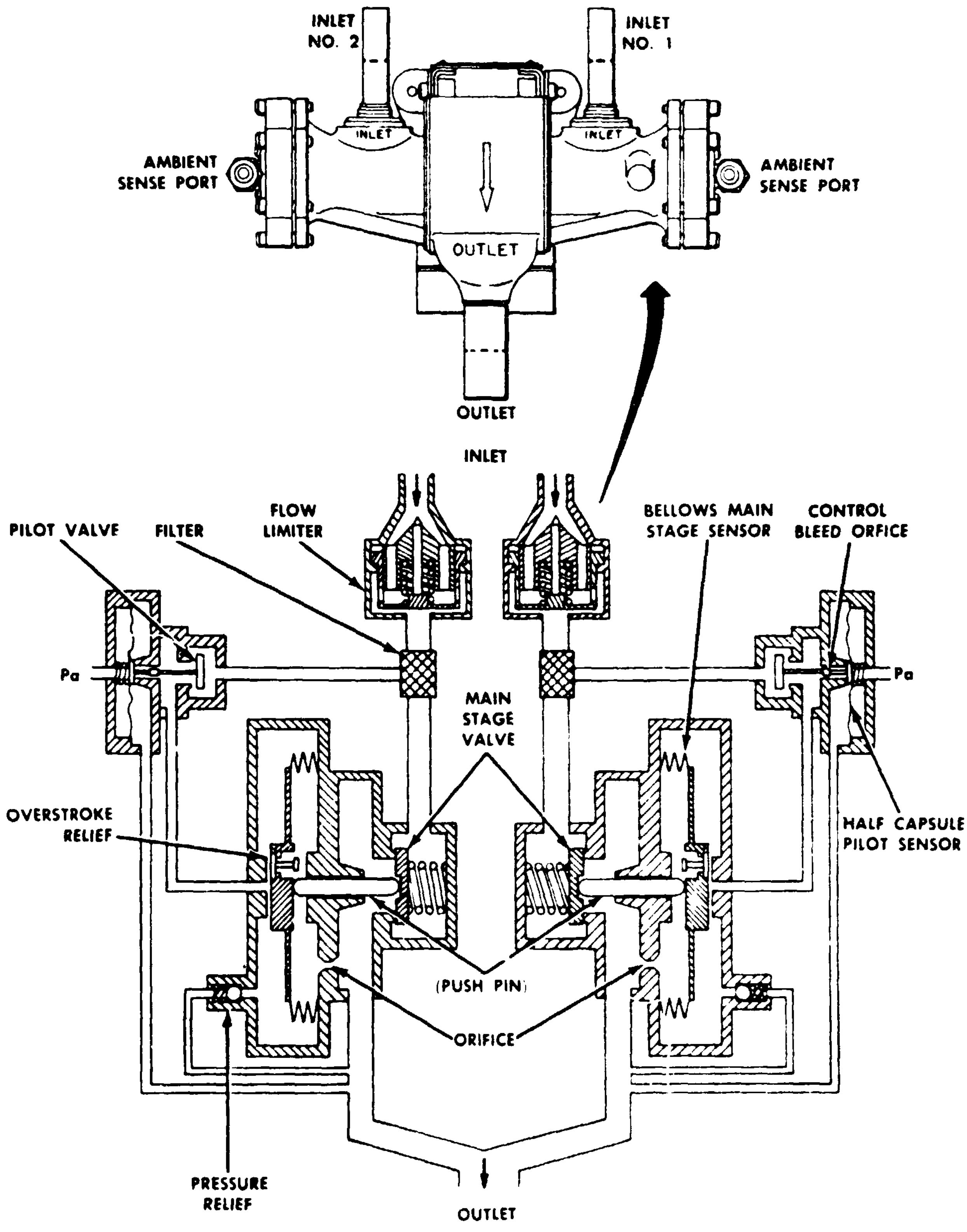


Figure 4-17

With respect to the long life multiple mission usage requirement, several other design deficiencies are apparent from the post qual test tear-down and inspection. In particular, the several areas of sliding fits, namely push rods and flow limiter guides, evidenced wear which would jeopardize regulator life. In addition to the degradation in performance which would result from changes in frictional characteristics as wear progresses, this wear is a source of self-generated contamination which could result in catastrophic failure of the regulator, or downstream components.

Though the qualification test program included compatibility testing in vapors of the respective propellants and vapor mixtures of the propellants, and the unit successfully passed these tests, the duration of these tests was approximately 3-1/2 days. Available data does not include a tabulation of the materials of construction of the DPS Regulator, therefore, an evaluation of their potential long term compatibility cannot be made at this time. The materials, however, may be a prime design driver in establishing the suitability of the DPS regulator for the Shuttle OMS application.

On the basis of the limited data available, it must be concluded that these existing regulator designs, and in particular, the DPS regulator, are not suitable for the Shuttle OMS helium regulator application for the following reasons:

1) Scaling to meet the Shuttle OMS regulator performance requirements would negate all prior development and qualification status.
2) Sliding fit guidance and/or alignment of moving elements is unacceptable.
3) Materials of construction of the existing designs are possibly unsuitable for long term propellant vapor exposure and an operating temperature range of -150° to +150°F.
4) Both hard and soft seats exhibited unacceptable contamination sensitivity.
5) Sensitivity to icing conditions was evident.
6) Sensitivity to propellant residue was evident (check valves in system did not preclude propellant residue from reaching the regulator).

4.2.5 Material Propellant Compatibility

A literature survey and study were conducted on the compatibility of materials with certain rocket engine fuels and oxidizers. More specifically, the materials of importance which were evaluated were those which were candidates for use in the Marquardt Space Shuttle OMS Helium Regulator. The regulator environment is helium, plus amounts of vapors of nitrogen tetroxide (N_2O_4), hydrazine (N_2H_4), unsymmetrical dimethylhydrazine (UDMH), monomethylhydrazine (MMH), and 50/50 blend of hydrazine or unsymmetrical dimethylhydrazine (50% N_2H_4 - 50% UDMH). The materials must be compatible for periods up to seven years and over a temperature range of -150°F to +150°F in the above environments.

All possible materials were evaluated; however, particular attention was paid to the materials in Table 4-IV, since these were the original materials selected in the proposal. As stated in the Marquardt Proposal (Ref. 7), aluminum alloys, even though they are excellent materials for shuttle environments, were not considered as candidate materials because the proposed designs require high strength and high modulus materials for the pressure sensing and sealing elements. The literature showed that in many instances the compatibility data for the remaining materials, because they are new materials, were not available. Secondly, if data exists, it is for shorter periods of time than seven years. Three years was about the longest time found for the materials of interest. For this application, the main environment, helium, has been shown to be basically inert (Ref. 14). In addition most of the investigations were conducted for propellant storage, perhaps a more severe condition than exists in the OMS regulator. In storage compatibility tests, metals that react with the propellants and cause them to decompose are considered incompatible. For the regulator, this would not be the case, because of the small amounts of propellants present, decomposition will not be a problem. The criterion used for the regulator was that to be compatible, the materials themselves must not corrode. Therefore, when it was necessary, engineering and metallurgical judgment was used to extrapolate the available data to select material to best satisfy contract requirements. The final material selections recommended are listed in Table 4-V. A more detailed discussion of these selections and the criteria leading to the selections follows.

Chemical Compatibility with N_2O_4

N_2O_4 is a very reactive oxidizer. A major problem is possible with long storage of N_2O_4 because of this relatively high reactivity and because of corrosion problems with certain metals. There are two grades or two specifications for propellant grade nitrogen tetroxide. These are differentiated by the amount of nitric oxide (NO) corrosion inhibitor contained in N_2O_4. If the (NO) content is less than 0.4%, the N_2O_4 is termed "brown" or Military Specification (Mil-P-26539A or B) N_2O_4. If the (NO) content is between 0.4%

and 0.8%, the N_2O_4 is termed "green" or NASA Specification (MSC-PPD-2A or B) N_2O_4. The terms "brown" and "green" arise from the colors of the liquid. Prior to 1966-1967, the Military Specification was used. At this time, problems of stress corrosion with N_2O_4 occurred. However, when the NASA Specification N_2O_4 propellant was used, the problems were eliminated. For this reason, the "green" NASA Specification N_2O_4 has replaced the "brown" Military N_2O_4 for aerospace applications since 1967. (Ref. 15)

According to AFRPL, Aerojet-General and Bell Aerosystems (Ref. 16, 17, and 18), most metals seem to be compatible with 60°F to 120°F N_2O_4 if the moisture content is small (0.1%). Moisture has been a cause for concern. Many of the earlier investigators believed that the presence of moisture would cause more corrosion due to the formation of nitric acid (HNO_3) when H_2O reacts with N_2O_4. According to Van Doehren (Ref. 16), carbon steel, aluminum, nickel, Inconel, and stainless steels are compatible with N_2O_4. However, if wet, 300 series stainless steel suitable for storage of HNO_3 should be used. Aerojet states that 300, 400 series 17-4PH, 17-7PH, AM350 and AM355 stainless steels and titanium are all good for wet N_2O_4 storage (Ref. 17).

Later investigations by Rocketdyne (Ref. 19), disproved conclusively the pr sumption of gross corrosion occurring to metals exposed to wet N_2O_4. The results of long-term (21 month) corrosion tests on 300 series stainless steels, AM350 steel, and aluminum alloys indicate that the actual corrosion rates with wet N_2O_4 (0.33 weight percent water) are far smaller than previously reported and do not differ from those obtair with dry N_2O_4 for most materials. The reason for this difference in the measured corrosion rates between this and previous programs is traceable to the difference in the duration of the tests. The calculation of a mil per year rate from a one-week test consists of determining the change in mils and multiplying by 52 as the assumption that the rate will remain constant over a long period. The long-term tests carried out by Rocketdyne prove that in the case of N_2O_4 corrosion, the rate of change in mils drops off drastically after a short period of exposure. Most of the data obtained showed that neither wet nor dry N_2O_4 was particularly corrosive at ambient temperatures. Therefore a selection of materials are available which are compatible with N_2O_4 and the presence of moisture.

Specifically, the following information was obtained concerning the compatibility of materials with N_2O_4 for the application for the Space Shuttle OMS Helium Regulator.

Inconel 718: Rocketdyne, Divisi a NAR, has tested and used this material successfully (Ref. 20 and 21). Huntington Alloys, supplier of the alloy, considers it to be compatible for the regulator application. There should be no problem with stress corrosion, pitting corrosion, or crevice corrosion (Ref. 22). Space Division NAR, is testing Inconel 718 in N_2O_4 at ambient temperatures and pressures for fifteen years. The tests have been running for six months and thus far results show no deleterious effects (Ref. 23). In addition, JPL tests on Inconel X-750 at 110°F for three years (Ref. 24) showed only 0.1 in./yr. corrosion. Inconel 718 and Inconel X-750 can be expected to perform in the same manner in N_2O_4 since they are chemically very close. It is concluded that Inconel 718 is compatible in NASA (MSC-PPD-2A or B) N_2O_4.

Inconel 625: No direct compatibility data for Inconel 625 in N_2O_4 could be found. However this material is similar to Inconel X-750 and should behave in the same manner. Thus the same comparisons and same conclusions can be drawn for this material as for Inconel 718.

6Al-4V Titanium: Titanium was one of the metals subject to stress corrosion with the Military Specification on N_2O_4. However, with the NASA Specification N_2O_4, titanium is compatible. 6Al-4V is rated class 1 in N_2O_4 (NASA) (Ref. 15). If 6Al-4V is used, even though it is a heat treatable alloy, it is recommended that it be used in the annealed condition. The position of this material in the galvanic series is the same as that of Inconel 718 and therefore should cause no galvanic corrosion problems in combination with nickel base alloys. Because of 6Al-4V's high strength to weight ratio and compatibility, it is recommended that it be considered for use.

Armco 21-6-9 SS: This is a new stainless steel considered a second generation successor to 304L stainless steel. As such, it has better corrosion resistance properties than its predecessor. However, no compatibility data exists for this material in N_2O_4. As mentioned earlier, the 300 series stainless steels have been used successfully in storage tanks (Ref. 16). The only problem that might arise is a problem of sludge formation. This is considered in more detail in a separate section. Since there is no direct test information on this material and there are other materials that are equal or better suited, such as Inconel 718, that have compatibility data, it is recommended that this material only be considered as a backup, and in addition, it is recommended that if possible, compatibility tests be conducted.

Braze Alloys:

Croniro 72Au-22Ni-6Cr: Although no test data exists for compatibility with N_2O_4, it is considered compatible based on Marquardt data. Marquardt has successfully used this material to braze poppet and seat materials. These were tested in CPF for six days. Examination showed no corrosion in the braze joint, and helium leakage was within limits. Consequently, this braze alloy was considered compatible with CPR (Ref. 12).

Nioro 82Au-18Ni: Boyd, et al, rate this material as compatible in N_2O_4. In addition, they rate gold as compatible (Ref. 12) in N_2O_4.

Ceramic Materials - K-801, K-96, and B_4C: K-801 and K-96 are tungsten carbide with 6% nickel added to K-801 and 6% cobalt added to K-96. No specific compatibility data were found for any of these materials in N_2O_4. Tungsten carbide was found to be compatible by Fish (Ref. 17). Marquardt has successfully tested these materials in CPF (Ref. 26). Additionally, (Ref. 27) carbides are inert in most chemical environments. Personal communication with Dr. Accountius (Ref. 28) revealed that no known data were available for N_2O_4 at this time, but it was his opinion that these materials should be compatible in N_2O_4. Compatibility data on these materials will become available as a result of a contamination resistant poppet sealing and check valve program (NAS 9-13882) being conducted by Rocketdyne.

Chemical Compatibility with N_2H_4

According to Martin Marietta (Ref. 15), hydrazine is a highly reactive propellent. It is considered thermodynamically unstable and exists in a state of continuous decomposition. The decomposition rate is a function of both the temperature and the presence of a catalyst. At ambient temperatures and in the absence of a catalyst, the average decomposition rate of N_2H_4 is minimal. In addition, according to Marquardt (Ref. 29), the decomposition of hydrazine was calculated to be only 0.002 gm. in 49 days at 70°F at 400 psig assuming maximum ullage fraction at 50%. The attack of storage materials is usually considered a problem only for non-metals since practically all metals show excellent corrosion resistance to N_2H_4. However, N_2H_4 has become corrosive to metals with certain contaminants such as CO_2 and Cl_2. An example of this was a problem at McDonnell Douglas (Ref. 30). Stress corrosion cracking was encountered with 400 series stainless steel. The problem was cured two ways. One way was to treat the hydrazine with barium oxide. The other was to procure hydrazine without CO_2. There was no problem with 6061 aluminum or 300 series stainless steels. Nickel base alloys were not tested.

In some instances the compatibility data for hydrazine is confusing, since the investigators will rate a material as being incompatible if it causes the fuel to decompose and the metal is unattacked. This is probably a good criterion for storage; however, for this regulator, only metal attack will be used to label a material as incompatible. Since the cycling conditions will purge the system, the problems of contamination and decomposition of fuel should not cause problems with the regulator.

The following information was obtained, specifically concerning compatibility of materials in N_2H_4 for application in the Space Shuttle OMS Helium Regulator.

Inconel 718: Rocketdyne reports (Ref. 21) that they considered Inconel 718 to be fully compatible with hydrazine. Space Division of NAR is conducting a fifteen year compatibility test program of materials in hydrazine. 718, along with 316 SS and 6Al-4V titanium are being tested at ambient temperatures and pressures. Results after six months show no corrosion problems for any of the materials. Huntington Alloys considers Inconel 718 to be compatible with hydrazine (Ref. 22). DMIC Bulletin and TRW rate Inconel X-750 as class 1 (Ref. 25 and 31) and Inconel 718 will perform in hydrazine environment like Inconel X-750. Tests by Carter (Ref. 32), show Inconel 718 to be compatible. All of these reports indicate that Inconel 718 is fully compatible with hydrazine for the proposed regulator, and it is therefore recommended for use.

Inconel 625: There is no direct information on this material with hydrazine. However, the similarity of Inconel 625 with the other Inconels, 718 and X-750, indicates that there should be excellent compatibility with hydrazine. However, it is recommended that compatibility tests be conducted to verify the analysis.

6Al-4V Titanium: All available sources show that 6Al-4V titanium is compatible with N_2H_4. United Aircraft Research Laboratories (Ref. 33) tested this material at 120°F and 160°F and showed full compatibility. Picatinny Arsnel (Ref. 34) showed compatibility at 160°F for two years. In addition, Martin Marietta (Ref. 15) and Boyd, et al (Ref. 25) agree that 6Al-4V titanium is compatible with hydrazine. From these results, this alloy can be considered compatible with the hydrazine requirements of the regulator design.

Armco 21-6-9 SS: As with N_2O_4 there is no compatibility data for this material with N_2H_4. However, there is ample data for the 300 series which show compatibility (Ref. 15 and 25). Since this material is a second generation extension of the 304 stainless steel, it can be deduced that 21-6-9 is fully compatible with hydrazine. However, there are other materials that are fully compatible and have been tested in N_2H_4; therefore, it is recommended that 21-6-9 be considered as a back-up material and that it be tested in hydrazine for exact evaluation.

Braze Alloys:

Croniro 72Au-22Ni-6Cr: Marquardt (Ref. 12) successfully operated valves that had components brazed with this material. These valves operated at 450 psig for 100,000 cycles in N_2O_4/MMH without damage to the braze material. Since this is a more severe environment than that of the regulator, it is believed that this material will be compatible for the regulator.

Nioro 82Au-18Ni: TRW and DMIC both rate gold as compatible in N_2H_4 (Ref. 23 and 13) However, AFRL (Ref. 36) rates Nioro as incompatible after 24 hours at 140°F. This unacceptable rating is due to the decomposition of the fuel, not because of an attack on the braze alloy. Therefore, for the use in the regulator, it could be considered compatible.

Ceramic Materials K-801, K-96, and B_4C: No specific compatibility data were found for these materials in N_2H_4. The references and information in the section for N_2O_4 for these materials also pertain to N_2H_4. They are recommended for use in the regulator.

4.2.5.3 Chemical Compatibility with MMH

Monomethylhydrazine, like hydrazine, is considered a highly reactive and toxic propellant. Its molecular structure is the same as hydrazine's except for having a hydrogen atom replaced by a methyl radical. Because of this, it shares many characteristics with N_2H_4. Metal corrosion is usually not a problem with MMH storage. MMH is generally not as active, or in other words is more stable than hydrazine. Materials showing compatibility with N_2H_4 will be either as compatible or more compatible in MMH (Ref. 15).

Tests were conducted at Martin Marietta at ambient temperatures and pressures for one year to determine the storage capability of several materials. The following were found to be compatible. No metal corrosion or MMH decomposition was observed (Ref. 37).

Stainless Steel: 304, 321, 347, 17-4PH, A-286, Carpenter 20Cb

Hastelloy C: 6Al-4V Titanium

Aerojet-General tested at 77°F and 150°F (Ref. 38). 347 stainless steel, maraging steel, and 6Al-4V titanium were not corroded after 24 weeks at 77°T and twelve weeks at 158°F. Martin and DMIC (Ref. 15 and 25) both show that 300 stainless steels, nickel base Inconel alloys, and titanium 6Al-4V are compatible. Because of the similarity of MMH to N_2H_4 and because MMH is less corrosive than N_2H_4, the specific recommendations made for materials in N_2H_4 will also pertain to MMH.

4.2.5.4 Chemical Compatibility with UDMH

Unsymmetrical dimethylhydrazine (UDMH) is, like hydrazine, a highly reactive and toxic propellant. Its toxicity and corrosivity are similar to hydrazine, but not so severe. According to DMIC (Ref. 25), UDMH affects materials in the same manner as hydrazine. They show that UDMH is compatible with aluminum alloys, 300 series stainless steels, 400 series stainless steels, 17-4PH, 17-7PH, and Carpenter 20 Cb. In addition, the nickel and nickel base Inconel alloys are compatible. The information shows that the recommendations made for materials in N_2H_4 will also pertain to UDMH.

4.2.5.5 **Chemical Compatibility with 50/50 UDMH and N_2H_4**

50/50 UDMH and N_2H_4 is a highly reactive and toxic propellant. Its toxicity and corrosivity are similar to N_2H_4. Most of the information on compatibility with metals is found in the DMIC and Titan II Storable Propellant Handbook (Ref. 25 and 18). This propellant is compatible with aluminum alloys, steel, stainless steel, nickel alloys, and titanium alloys. Because of the similarity to N_2H_4, the recommendation made of materials in N_2H_4 will also pertain to this propellant.

4.2.6 Special Considerations

In addition to chemical compatibility, other criteria are important to the successful operation of the OMS Helium Regulator. These are discussed individually in the following section.

4.2.6.1 Mechanical Property Effects: As mentioned earlier, high pressure helium does not adversely affect the mechanical properties of the proposed materials (Ref. 14). But the contamination of oxidizers and fuels to the helium must also be considered. From the standpoint of fracture mechanics, small amounts of N_2O_4 might be beneficial. If N_2O_4 breaks down and forms O_2, the formation of oxides at the point of a crack would tend to blunt the crack and thereby retard propagation. Fuels, on the other hand, might break down into nitrogen and hydrogen. Nitrogen, like helium, does not adversely affect mechanical properties at ambient temperatures. Hydrogen is a different story. The work at Rocketdyne shows that small amounts of hydrogen in helium are nearly as harmful as pure hydrogen (Ref. 39). In the presence of an oxidizer like N_2O_4, this hydrogen would be converted to water vapor which is less detrimental. It is believed that though the above conditions could exist, they are not likely to cause a problem for the following reasons. First, the amounts of fuel and oxidizers present in the system are likely to be very small. Second, at ambient temperatures or near ambient temperatures the propellants are stable and the chance that the above reactions might occur and cause problems is very remote.

4.2.6.2 Clogging Material: TRW (Ref. 4) and Rocketdyne (Ref. 41) conducted flow experiments to determine the formation and behavior of clogging materials. Nitrogen tetroxide (N_2O_4) was found to degrade the flow rate ($\nearrow$10%) in all tests. The critical parameters affecting flow degradation were flow rate and differential temperature of the propellant. It was postulated that the clogging material is a gel formation caused by the nucleation and solvation of colloidal or suspendable matter in the propellant. Analysis of the gel-like material included nickel, nitrates, and iron. Other impurities found were chromium, gold, manganese, tin, aluminum, copper, silver, and titanium.

Tests performed in hydrazine showed no evidence of flow clogging. Aluminum, titanium, and stainless steels were used for the flow tests.

The above information indicates that clogging could possibly occur. However, it is believed that the amounts of N_2O_4 present in vapor form will be so slight that clogging is unlikely to occur.

4.2.6.3 Galvanic Corrosion: Various metal couples show no galvanic corrosion in N_2O_4 at 55°F to 65°F. These couples are 2014-T6 Al and stainless 321SS, 2014-T6A1 and 303SS, silver and 347 SS, and nichrome and 347SS (Ref. 17). Specifically, for the materials being studied, nickel base materials Inconel 718 and Inconel 625 have an EMF of -0.15 volts. Titanium has an EMF of -0.15 volts. Armco 21-6-9 has an EMF of -0.20 volts. The gold containing brazing material has an EMF of -0.15 volts. The nickel brazing materials have an EMF of -0.15 volts. The ceramic materials have an EMF of +0.05 volts. Therefore, if the requirement of permitting no greater difference than 0.25 volts is to be met, (MSC Design and Procural Standard No. 63) a nickel base brazing material must be used to braze the ceramic seals to the nickel base Inconel 718 base. In addition, brazing should be done in a vacuum to eliminate fluxes which might cause severe galvanic corrosion.

4.2.6.4 Stress Corrosion, Crevice Corrosion, Pitting Corrosion: All of the materials selected are good or excellent in the resistance to these types of corrosion in the fuel and oxidizer environments. In addition, the design philosophy as stated in the proposal is to design components to eliminate problems like crevice corrosion.

4.2.6.5 Cryogenic Capabilities: Since one of the requirements of the contract is that the regulator must operate at -150°F, all material selections were made to eliminate any material that would lose ductility at cryogenic temperatures. Inconel 718, Inconel 625, and Armco 21-6-9 are austenetic or face centered cubic materials which do not transform at cryogenic temperatures to another crystal structure. Thus, they are not embrittled. The brazing materials, Croniro, Nioro and AMS4776 are likewise austenetic materials and not subject to low temperature embrittlement. Ceramic materials WC and B_4C have no low temperature phase change and no change in properties due to cryogenic temperatures.

4.2.6.6 Cleaning and Handling Requirements: The effects of residual contamination such as machining oils, vapor degreasing, solutions, non-destructive testing solutions, on compatibility of fuels and oxidizers are not clearly defined. 6A1-4V is incompatible with chlorine containing cleaning fluids and Freon MF. However, it is compatible with Freon TF. Other materials selected are compatible with chlorine containing cleaning solvents and Freon. Titan II Handbook (Ref. 17) states that the presence of organic compounds such as alcohols, acetones, and gasoline, are undesirable because of their reactivity with N_2O_4. Dr. Axworthy (Ref. 42) stated that failure to passivate tanks with hydrazine prior to storage doubled the pressure rise in the unpassivated tank. The evidence is that organic contamination can and will cause problems with fuel decomposition. Therefore, even though the amount of fuel and oxidizer in the regulator will be small, it is recommended that steps be taken to eliminate possible contamination prior to using the regulator. The exact procedure to be used will depend on the material used to fabricate the regulator.

However, the regulator components should be chemically passivated followed by passivation in the oxidizer and fuel prior to operation.

4.2.6.7 Combustion Products - Result of Mixing of Fuel and Oxidizer

The mixing of vapors of the oxidizers and fuels can result in a reaction product, amine nitrate. According to Yanizeski (Ref. 43), pure MMH nitrate is a crystalline material, but it can quickly absorb moisture and change to a viscous form. The MMH nitrate crystalline form melts at 104°F, it thermally decomposes at 455°F, and it is somewhat impact sensitive with a 136% TNT equivalence. Compatibility data are rather limited. However, there are definite indications that the amine nitrates are more corrosive than the amines themselves. Severe corrosion has been noted on stainless steels in contact with nitrated hydrazine (Ref. 12). The same source states that aluminum and titanium are the best materials for long time service with nitrated hydrazine mixtures. They conclude that all ferrous metals including stainless steels are totally unacceptable with nitrated hydrazine. Lee (Ref. 44) reports that Inconel X and Inconel are probably acceptable for limited servic with hydrazine - hydrazine nitrate water mixtures. From the information available, it is concluded that with amine nitrates, 6A1-4V titanium would be compatible, Inconel 718 and Inconel 625 are probably compatible, and Armco 21-6-9 would not be compatible based on stainless steel results. No data were found for the braze materials, but it is believed that they would be compatible based on the general corrosion resistance of these materials. Ceramics should not be attacked because of their generally excellent corrosion resistance. It is recommended that compatibility tests be conducted to verify the assumptions made above. It is possible that because of the small amount of amine nitrates which might be present, no problem will exist.

4.2.6.8 Welding Rod

If or when it is necessary to use weld filler materials, the same filler as the base material should be used whenever possible. For the materials selected, this is po sible. In this way, galvanic cell reaction is minimized. Thus, for Inconel 718, Inconel 625, Armco 21-6-9, and 6A1-4V titanium, matching fillers are available and these are the recommended ones to be used.

4.2.6.9 Thermal Compensation Materials

It may be necessary to use thermal compensation devices to account for temperature differences in the regulator. Such devices are commonly made from Invar and Ni-Span materials. Fish (Ref. 17) shows that both of these materials are compatible ir [illegible] in N_2H_4/UDMH. JPL test results (Ref. 24) show that Ni-Span is compatible [illegible] three years. The stress-corrosion, crevice corrosion, pitting corrosion compatibilities are unknown, but their chemical compositions are such that they should be resistant to these conditions. In addition, they should be compatible with the common cleaning solutions. Based on these findings, both Invar and Ni-Span should be compatible for Helium Regulator construction.

TABLE 4-IV

PROPOSED MATERIALS FOR REGULATOR CONSTRUCTION

MATERIALS	CERAMIC MATERIALS	VACUUM BRAZE ALLOYS
Inconel 718	K-801	Croniro
Inconel 625	K-96	Nioro
Armco 21-6-9	B_4C	AMS4776

TABLE 4-V. CANDIDATE MATERIALS FOR SPACE SHUTTLE OMS HELIUM REGULATOR

Item No.	Material	Principal Producers	Material Condition	Heat Treatment	Room Temp. Mech. Prop. F_{tu}	F_{ty}	% El	Rc	Weldability	N_2O_4 (Dry) Compatibility	[illegible] Compatibility	[illegible] Compatibility	MMH Compatibility	H_2O (deionized) Compatibility	Freon & Alcohol Compatibility	Trichlorethylene Compatibility	Amine Nitrates Compatibility	Amine Compatibility	Stress Corrosion Resistance	Crevice Corrosion Resistance	Pitting Corrosion Resistance	Galvanic Corrosion Potential EMF, Volts	Application	Remarks (1)
	Inconel [illegible] Ni Base	Special Metals, Huntington Alloy, Cabot Corp.	Solution Treat and Aged	1750°F 1 hr AQ to RT; 1350°F 8 hr FC to 1150°F; [illegible] 8 hr Air Cool to RT	180	150	[illegible]	[illegible]	F	C	C	C	C	[illegible]	F	[illegible]	TBD	C	In	F	F	0.25	Lever Arm, Push Rod, Flexure Poppet, Spring Actuator Flexure, Push Rod, Flexure Actuator, Bellows Poppet, Bellows Post Rod, Bellows Actuator	Excellent compatibility and fabricability. 1st choice
2	Inconel [illegible] Ni Base	Huntington Alloy, Cabot Corp.	Annealed	2100°F Solution Anneal	[illegible]	35	[illegible]		F	TBD	TBD	TBD	C	E	F	C	TBD	TBD	Im	E	F	0.15	Same as 1 above	Although no actual compatibility data available, should perform same as other Inconel alloys, i.e., excellent. Also excellent fabricability. 2nd choice
3	[illegible] Fe Base	Armco Steel	Annealed	1950°F, 1 hr, AQ to RT	100		40		F	TBD	TBD	TBD	TBD	E	F	C	In	TBD	S	S	S	0.20	Same as 1 above	Although no actual compatibility data available, should perform better than 300 series, which are compatible. Excellent fabricability. 4th choice
4	6Al-4V Ti Alloy	TIMET, Reactive Metals, Harvey Alum	Annealed	1400°F 1 hr Cool to RT	130	120	10		S	C	C	C	C	F	(B)	In	C	C	E	F	E	0.15	Same as 1 above, except Bellows	Excellent compatibility. Good fabricability. 3rd choice. No bellows made from this material. Not compatible with Cl containing solvents
	K96	Kennametal Inc.	Not Heat Treatable	Not Heat Treatable	[illegible]	[illegible] 490	Knoop Hardness [illegible]		(A)	TBD	TBD	TBD	TBD	E	E	C	TBD	TBD	Im	Im	Im	+0.05	Poppet, Seat	No actual data in N_2O_4 or amines. CPF compatibility acceptable. Material's chemically inert. Projected compatibility in N_2O_4 and amines acceptable
6	[illegible]	Man-Labs	Not Heat Treatable	Not Heat Treatable	[illegible]	[illegible]	[illegible]		(A)	TBD	TBD	TBD	TBD	F	F	C	TBD	TBD	In	Im	Im	+0.05	Poppet, Seat	Same as 5 above
	[illegible] Braze	Handy & Harman	Not Heat Treatable	Not Heat Treatable					NA	C	C	C	E	F	E	C	TBD	C	F	E	F	+0.15	Braze Poppet Seats, Flexures	This material has good compatibility and good brazing characteristics. However, the galvanic potential with the proposed materials is greater than 0.25 volts
8	Au-Ni Braze	Handy & Harman	Not Heat Treatable	Not Heat Treatable					NA	C	C	E	E	F	E	C	TBD	C	E	E	F	+0.15	Braze Poppet Seats, Flexures	Same as 7 above
9	AMS 4776 Ni Braze	Wall Colmonoy	Not Heat Treatable	Not Heat Treatable					NA	C	TBD	TBD	C	F	F	C	TBD	C	F	E	E	0.15	Braze Poppet Seats, Flexures	This material has not been used in these environments. However, compatibility should be the same as Inconel. No problem with galvanic potential.

LEGEND:

C	Compatible	S	Satisfactory
E	Excellent	TBD	To be determined
G	Good	(A)	Not weldable but brazeable
Im	Immune	(B)	Excellent in alcohol
In	Incompatible		Incompatible in Freon MF
NA	Not applicable		Compatible in Freon TB

Note (1) The compatibility data are for long terms, up to 3 years. No data available for 7-10 years.

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5.0 DESIGN DEFINITION*

As a result of the Analysis effort (Task 1), it was concluded that the unique requirements of the Shuttle OMS Regulator application, Figure 5-1, preclude the application of any existing man-rated spacecraft qualified pressure regulator, without major changes which would negate any prior test or usage history. A rigorous analytical design study was initiated to generate valid objective data to evaluate four candidate pressure regulator configurations for potential application to the shuttle OMS requirements:

I. Direct Acting Single Stage

II. Direct Acting Single Stage with Lever Arm

III. Pilot Operated

IV. Direct Acting Single Stage with Push-Pull Rod.

Design layouts and analytical studies were made, for each configuration, to generate design criteria for regulators to meet the performance requirements of Figure 5-2. Both digital and analog computer modeling were employed to make static and dynamic performance projections. Several design features, such as materials of construction, flexure guidance, poppet seat interface and metal bellows dyanmic seals, were incorporated into all the designs. These features represent workable solutions to such design parameters as long life capability, high contamination tolerance, and reuseability without maintenance.

This effort resulted in the design of the four candidate configuration regulators any one of which appeared capable of meeting the required performance parameters. The dynamic analyses confirmed the stability characteristics of the direct acting configurations but indicated a potential oscillation problem with the pilot operation (2-stage) design. Though outlet pressure oscillations were well within acceptable tolerances, the pilot operated configuration never achieved total stability and this continuous oscillation of the poppet represented a potential wear out mode leading to premature failure. The failure to achieve stable operation is felt to be inherent in this configuration due to lack of positive position feedback between the two stages.

At the completion of the design and analytical effort, all data was compiled, including development cost and schedule projections, for each configuration. The data was evaluated in an attempt to identify the optimum configuration for further design development. Marquardt concluded that the single stage (Configuration I) concept was the optimum choice for further development. This recommendation was concurred by the NASA Technical Monitor at the Design Review held at NASA-MSC on 15 November 1972.

*From Report 5103-7-2 of November 1972

UNIQUE SHUTTLE OMS REGULATOR REQUIREMENTS

- TOLERATE PARTICULATE CONTAMINATION UP TO 150μ
- OPERATING TEMPERATURE RANGE OFFERS POTENTIAL FOR VAPOR, LIQUID, AND SOLID PHASES OF PROPELLANTS AND WATER
- FIVE YEAR OPERATION LIFE
- MULTIPLE, MAN-RATED MISSIONS
- NO MAINTENANCE REQUIRED (GOAL)
- MULTIPLE APPLICATION POTENTIAL-OMS/RCS

Figure 5-1

PERFORMANCE REQUIREMENTS
FLOW LIMITER AND TWO REGULATORS IN SERIES

- INLET PRESSURE ____ 4000 TO 400 PSIA
- OUTLET PRESSURE ____ 250 ±4 PSIA (NO.1 REGULATOR CONTROLLING)
- LOCKUP PRESSURE ____ 265 PSIA (NO.1 REGULATOR CONTROLLING)
- DESIGN FLOW RATE ____ 4.68 LBS/MIN. He at -150 °F
- MAXIMUM FLOW RATE PER FLOW LIMITER ____ 10 LBS/MIN He at -150 °F
- LEAKAGE INTERNAL ____ 100 SCC/HR He
  EXTERNAL ____ 1 SCC/HR He
- OPERATING TEMPERATURE RANGE ____ -150 TO + 150 °F
- CONTAMINATION TOLERANCE ____ UP TO 150 MICRON PARTICLES
- SERVICE LIFE ____ 7 YEARS SERVICE LIFE
  ____ 5 YEAR SERVICE LIFE MAINTENANCE FREE
- ULLAGE ____ 1 TO 300 CUBIC FEET
- FLUID MEDIA COMPATIBILITY ____ N_2O_4 AND AMINE FUELS AND THEIR COMBUSTION PRODUCTS. FREON. ALCOHOL. WATER. TRICHLORO ETHYLENE

Figure 5-2

This section presents the design and performance definition of each candidate configuration, an evaluation and comparison of this information, and the rationale for the ultimate selection of the configuration for development.

5.1 DESIGN AND PERFORMANCE DEFINITION

Schematic presentations of the four candidate regulator configurations are shown in Figure 5-3 through 5-6 to illustrate the basic mode of operation and mechanisms of each. In the following sub-paragraphs, each configuration design is described and performance projections presented.

5.1.1 Configuration No. 1 - Direct Acting Single Stage

The direct acting, single stage concept, in a flightweight configuration, is shown in Figure 5-7. Flow from the inlet, through the valve seat, to the outlet port is controlled by the pressure balanced, flexure guided poppet. The poppet-seat interface is a flat controlled land width hard seat which contacts a lapped, flat, hard poppet-surface. This interface is more fully described in Paragraph 5.3. Poppet motion, hence media flow control, is modulated by a spring-loaded actuator which positions the poppet in response to the force balance of the reference pressure acting on the actuator effective area against the reference spring load. Movement of the actuator in response to this force balance is transmitted to the poppet through a flexure-guided push rod. A metal bellows shaft seal on the push rod isolates the actuator reference pressure cavity from the outlet pressure cavity.

The envelope and weight estimates for the configuration,as shown,are 4.0 inches diameter by 8.0 inches overall length, and 7.95 lbs based upon the selected materials of construction.

Configuration performance analyses were made utilizing several digital computer programs and dynamic characteristics were resolved by analog computer modeling.

A major analytical task was undertaken to develop a valid mathematical model of the forces that would act upon a balanced poppet as a function of its physical parameters and the characteristics of the flowing media. This analytical model was adapted to the Marquardt APL Computer and sufficient cases resolved to establish, with a high degree of confidence, the flow force characteristic. For the poppet/seat configuration of the candidate designs, Figure 5-8 illustrates dimensional parameters which were evaluated for impact on the flow force characteristic. An existing test model of a flat seat interface was employed to confirm the math model. Test model flow force measurements as a function of poppet stroke and inlet pressure were made and compared with the theoretical projections. Though the test model geometry could not be exactly duplicated by the math model, sufficient correlation was demonstrated to establish a high level of confidence in the analytical program. The program was then employed to generate flow force characteristics for the nominal dimensions of the configuration design.

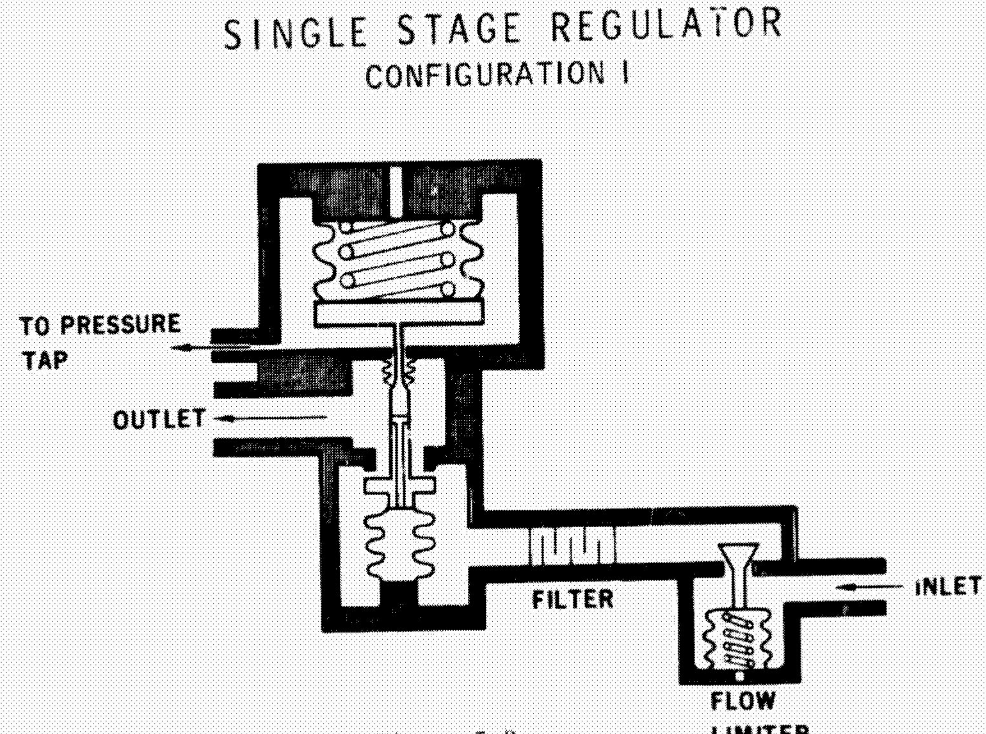


Figure 5-3

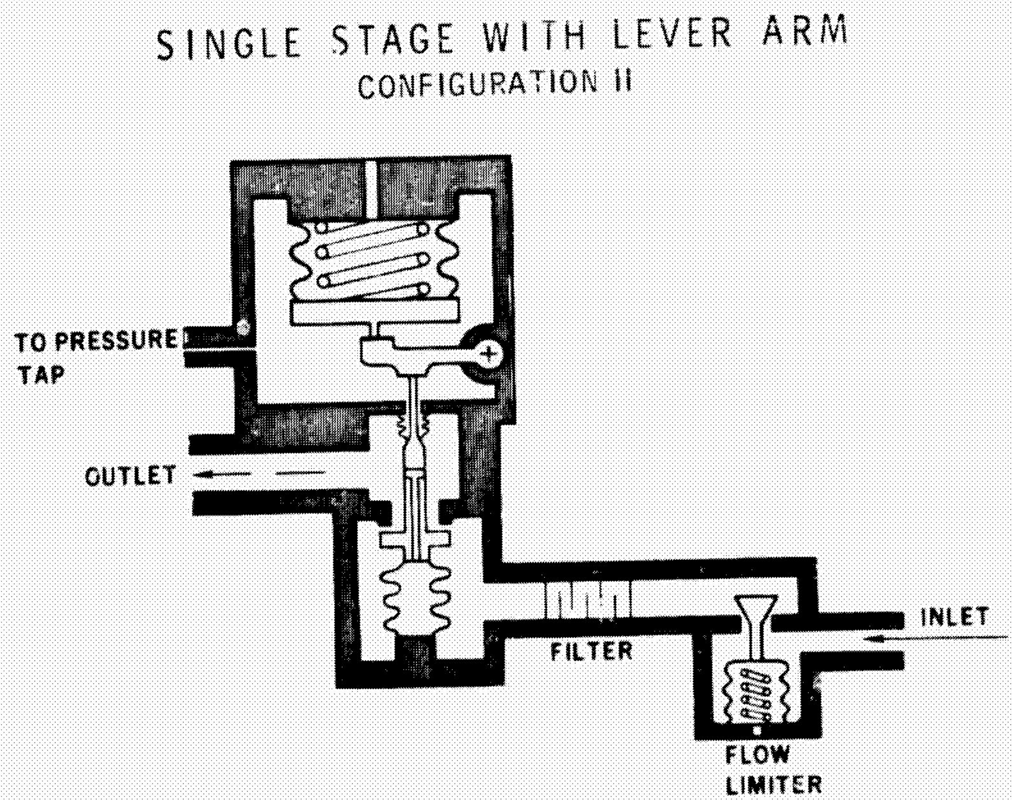


Figure 5-4

A74-4-612-53

PILOT OPERATED REGULATOR
CONFIGURATION III

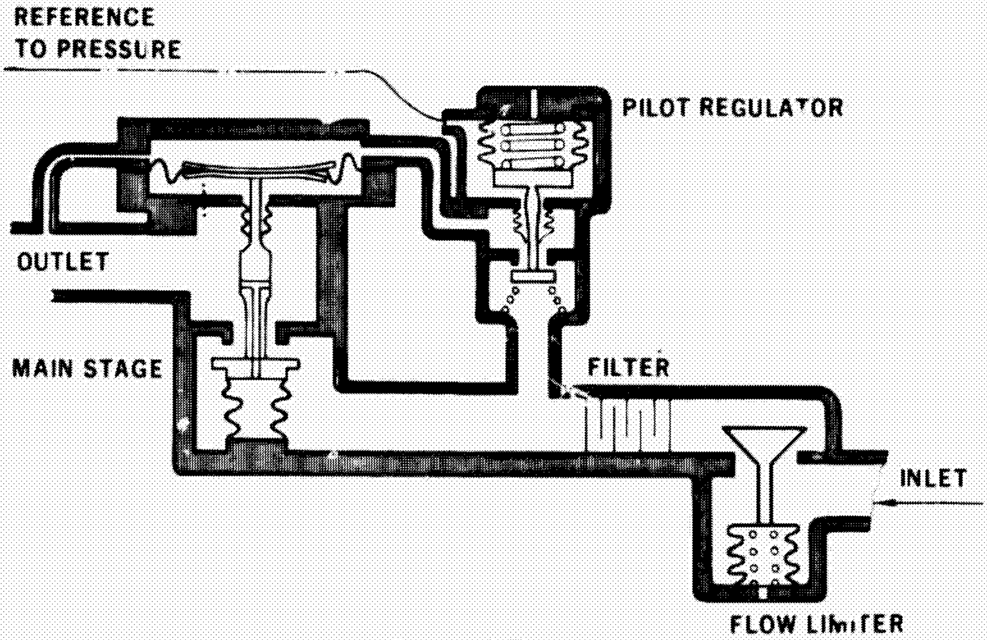


Figure 5-5

SINGLE STAGE PUSH-PULL REGULATOR
CONFIGURATION IV

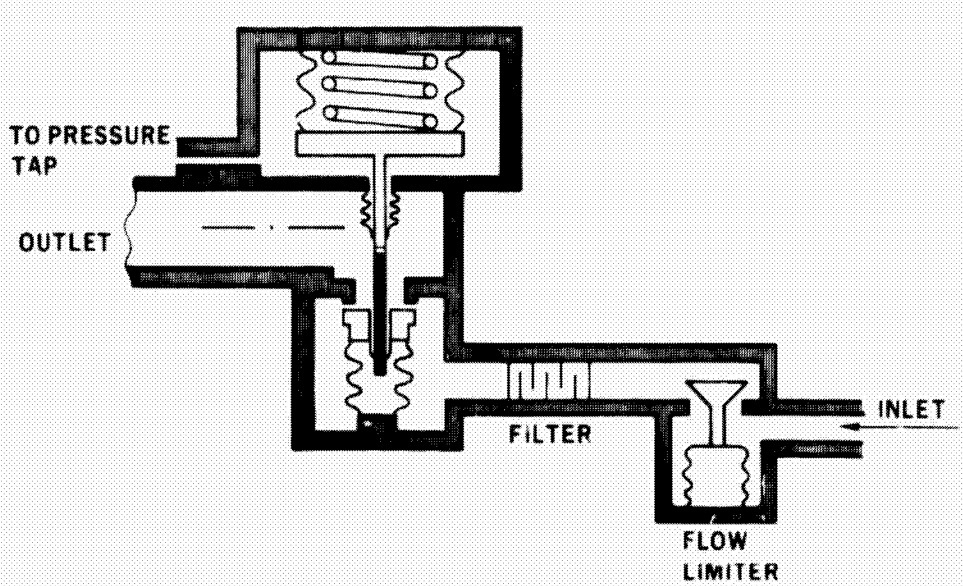
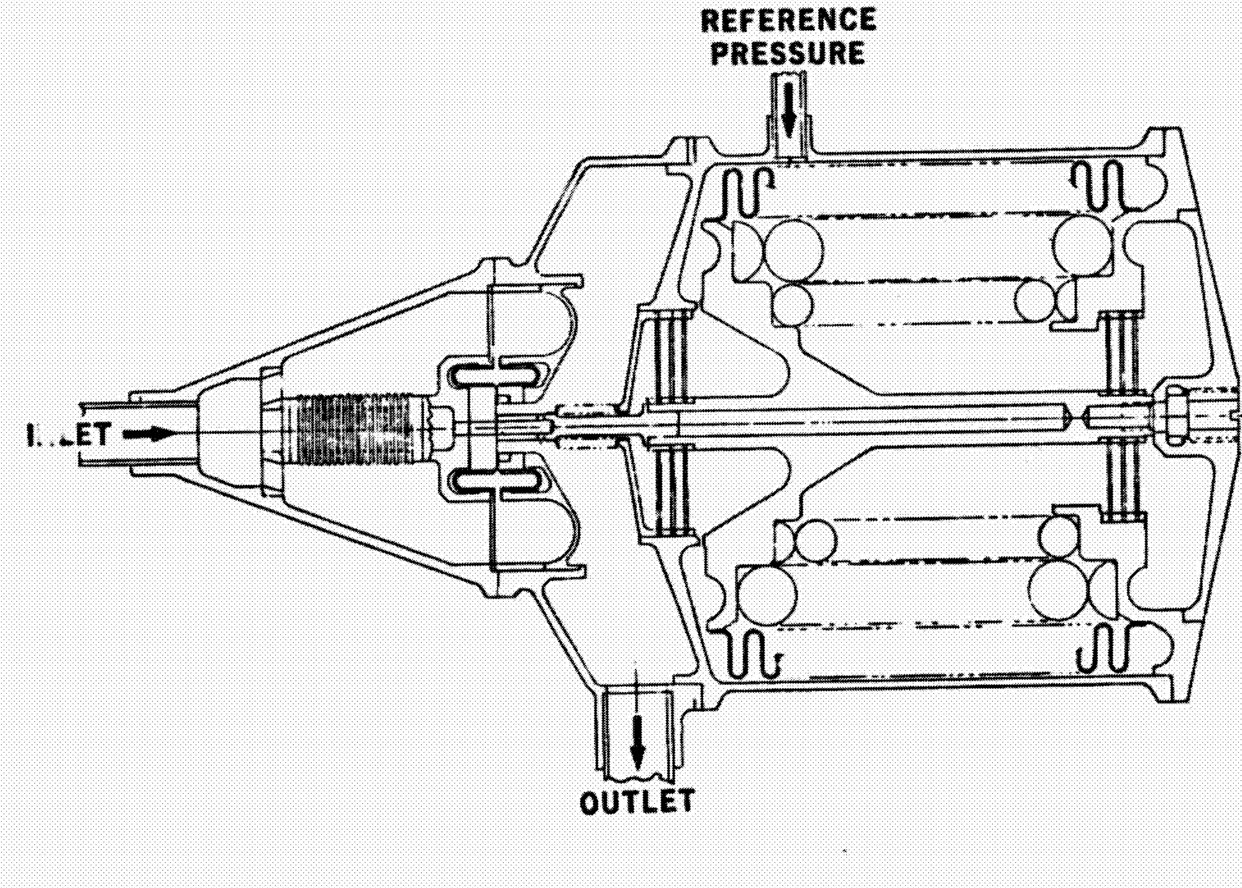


Figure 5-6

OMS REGULATOR

CONFIGURATION NO. 1



C-2

Figure 5-7

DEFINITION OF APL PROGRAM POP PARAMETERS

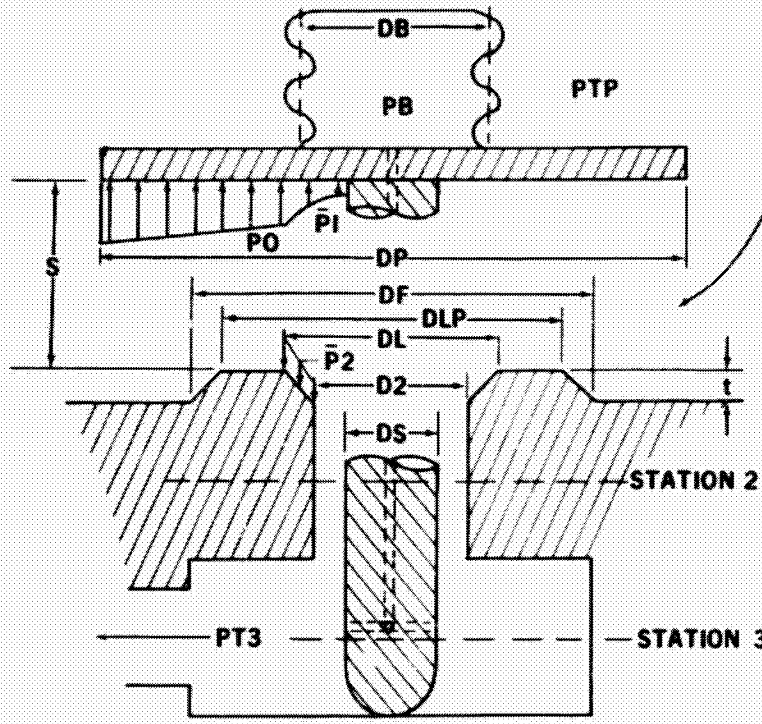


Figure 5-8

EFFECT OF POPPET BELLOWS MEAN AREA % TOLERANCE ON FLOW FORCE

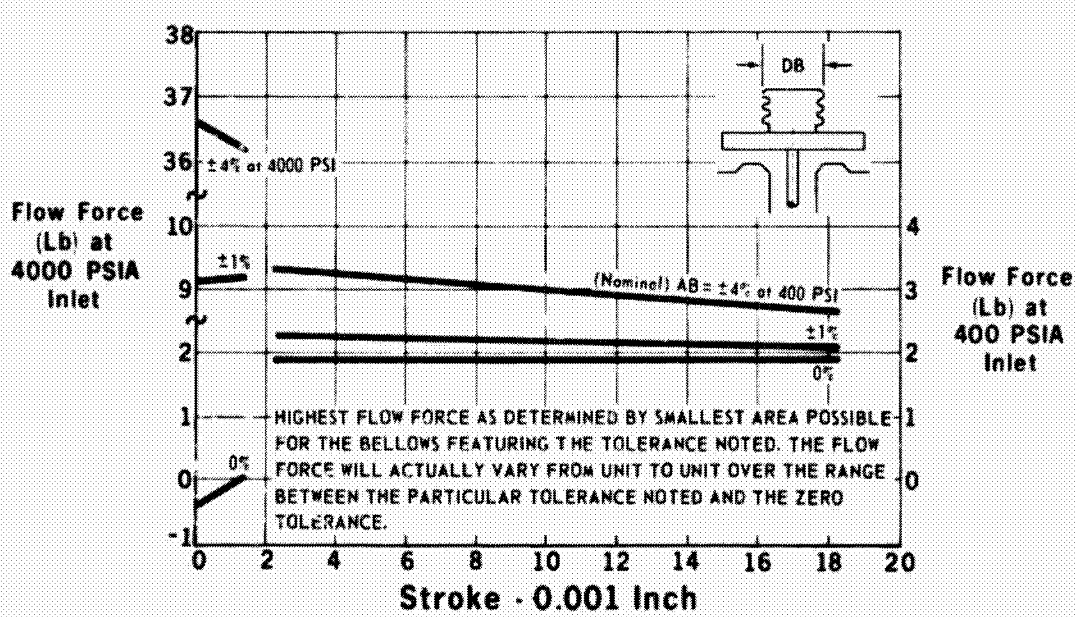


Figure 5-9

To assess the influence of key dimensional parameters on the flow force characteristics, several iterations of the "POP" Program (Appendix) were performed while changing only one variable. As is evidenced by Figures 5-9 through 5-13, the flow force characteristic can be fitted to a desired form by proper control of dimensional relationships.

The flow force characteristics for the design's nominal dimensions are plotted in Figure 5-14 for four cases of inlet pressure over the design range. Using the initial design criteria for this configuration, employing the APL Program (REGDES) described in Section 4.0 (Figure 5-15) and the flow force characteristic data, regulator performance projections can be made by solving for the design's force balance at various conditions.

In the quad-redundant arrangement anticipated, and shown schematically in Figure 5-16, the regulator outlet pressure of the upstream regulator of each leg (regulator No. 1 or regulator No. 3) will differ from the reference pressure by an amount equal to the pressure drop across the downstream regulator (regulator No. 2 or regulator No. 4, respectively). This pressure differential acts on the push rod shaft bellows and represents an input to the force balance. Since, by design, the downstream regulator is in the full open position when the upstream regulator is functioning, the downstream regulator can be represented by a fixed orifice; and its pressure drop can be resolved as a function of the flow rate, media temperature, and outlet pressure of the upstream pressure. This pressure differential, as a function of flow rate at -150°F, is plotted as Figure 5-17.

The force balance equation for any condition of inlet pressure, flow rate, stroke, and media temperature can then be solved for the reference pressure that results in a stable system. This analysis is summarized in Figure 5-18 and is expressed as:

EFFECT OF WIDTH L ON FLOW FORCES

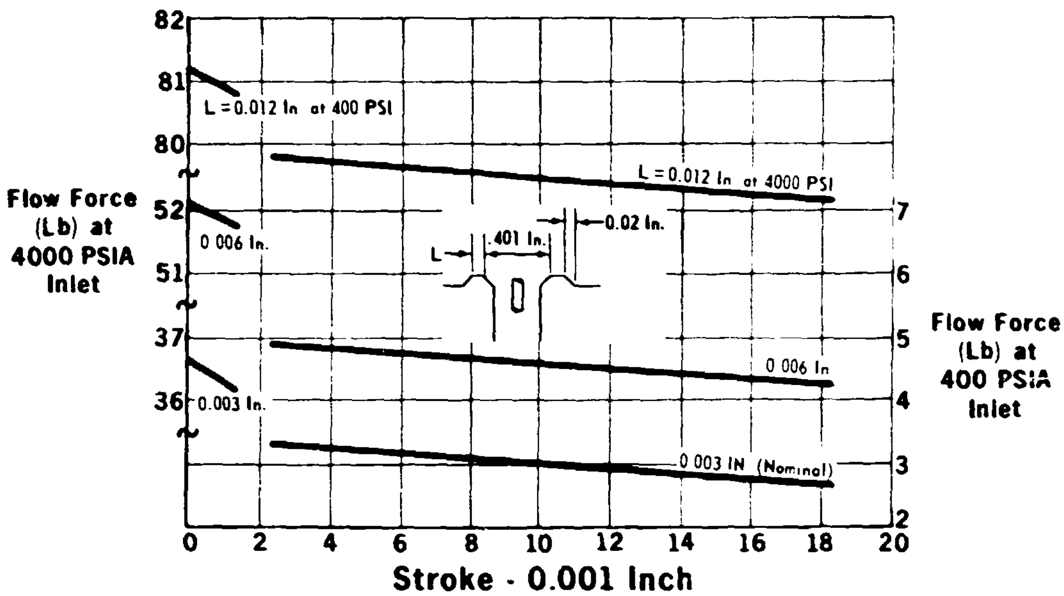
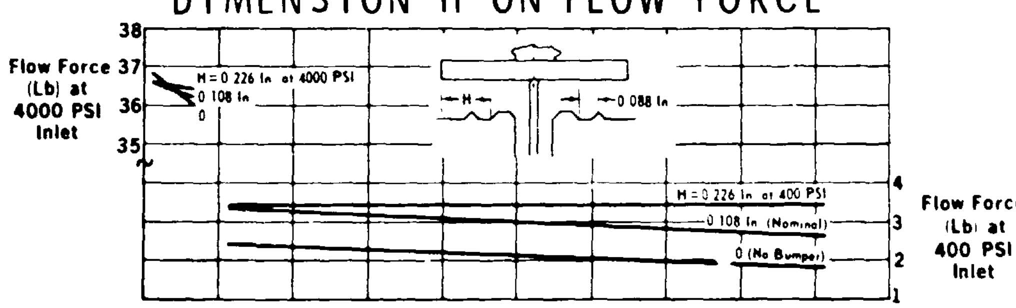


Figure 5-10

EFFECT OF POPPET OVERHANG DIMENSION H ON FLOW FORCE



EFFECT OF EXIT DIMENSION DIAMETER D2 ON FLOW FORCE

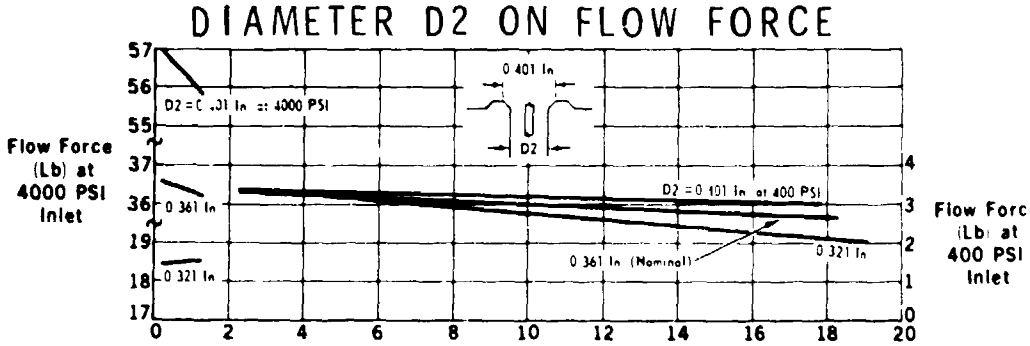


Figure 5-11

EFFECT OF SEAT RECESS DIMENSION t ON FLOW FORCE

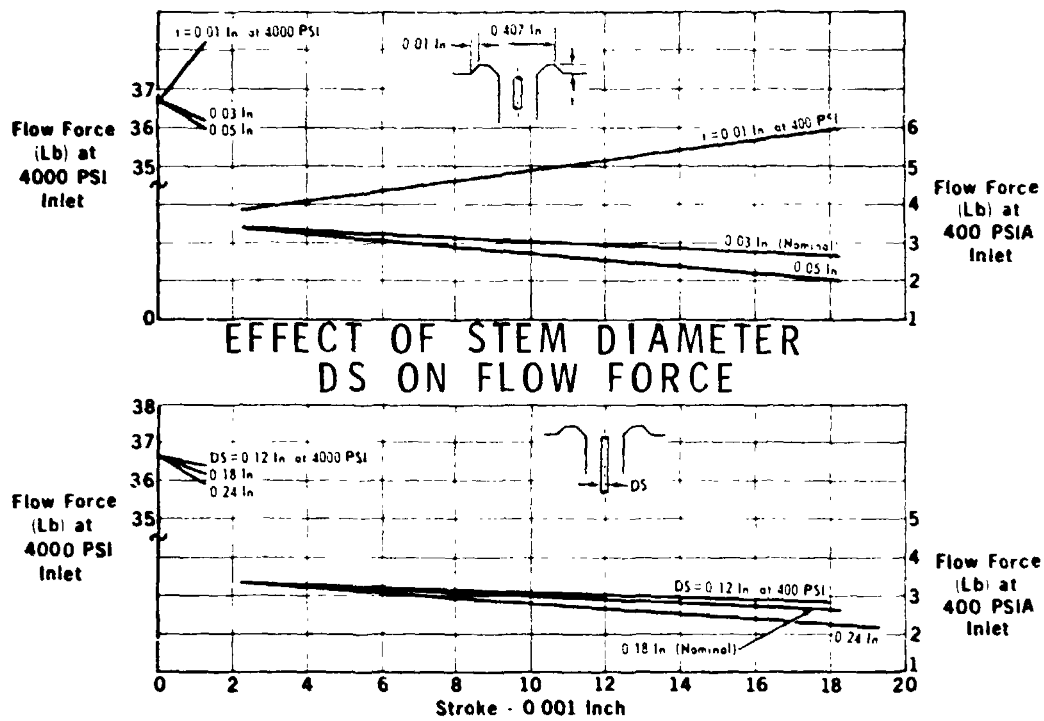


Figure 5-12

EFFECT OF OUTSIDE CHAMFER DIMENSION OC ON FLOW FORCE

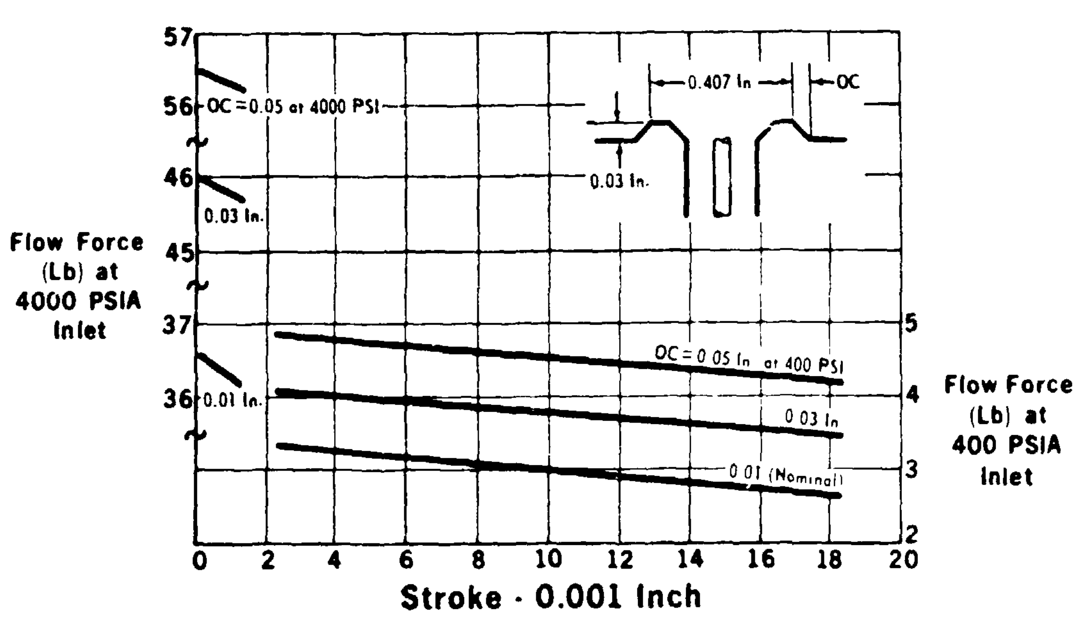


Figure 5-13

POPPET FORCE CHARACTERISTIC

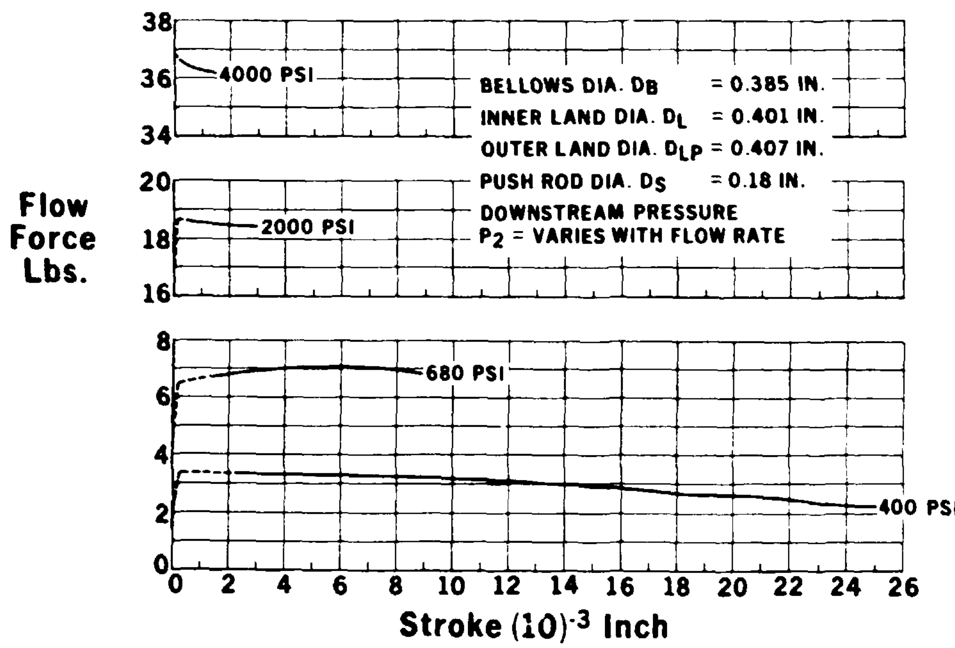
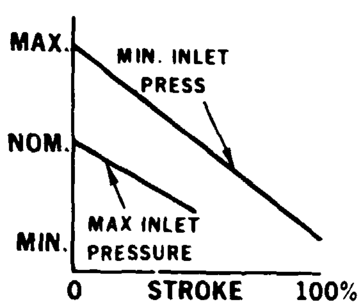


Figure 5-14

REGULATOR DESIGN PROGRAM

OBJECTIVE ●Establish Basic Regulator Design
●Parameters Based Upon Static Analysis

INPUT ●Minimum Inlet Pressure
●Maximum Inlet Pressure
●Nominal Output Pressure
●Minimum and Maximum Output Pressure
●Poppet Force vs Stroke Characteristic
(Flow Forces and Poppet Balancing Forces)
●Poppet/Seat Diameter and Stroke (Max) to
Meet Flow Requirements.
●Outlet Pressure vs Stroke Characteristic.



OUTPUT ●Actuator Sensing Area Required
●Preload Force Required
●Maximum Allowable System Springrate

Figure 5-15

REGULATOR SYSTEM BLOCK DIAGRAM

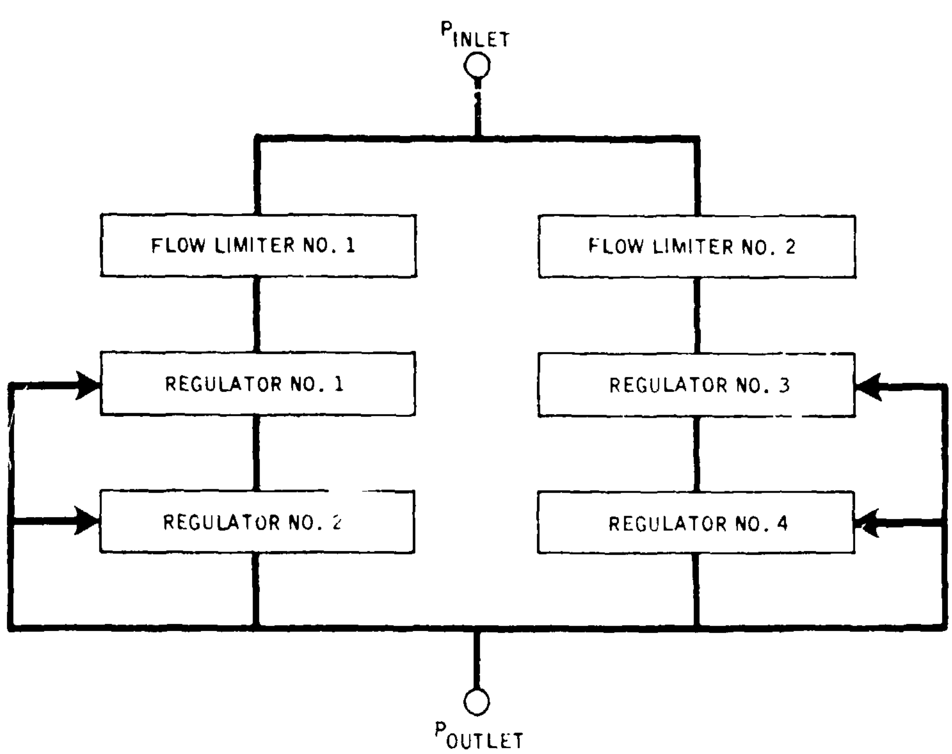


Figure 5-16

SECONDARY REGULATOR PRESSURE DROP CHARACTERISTIC

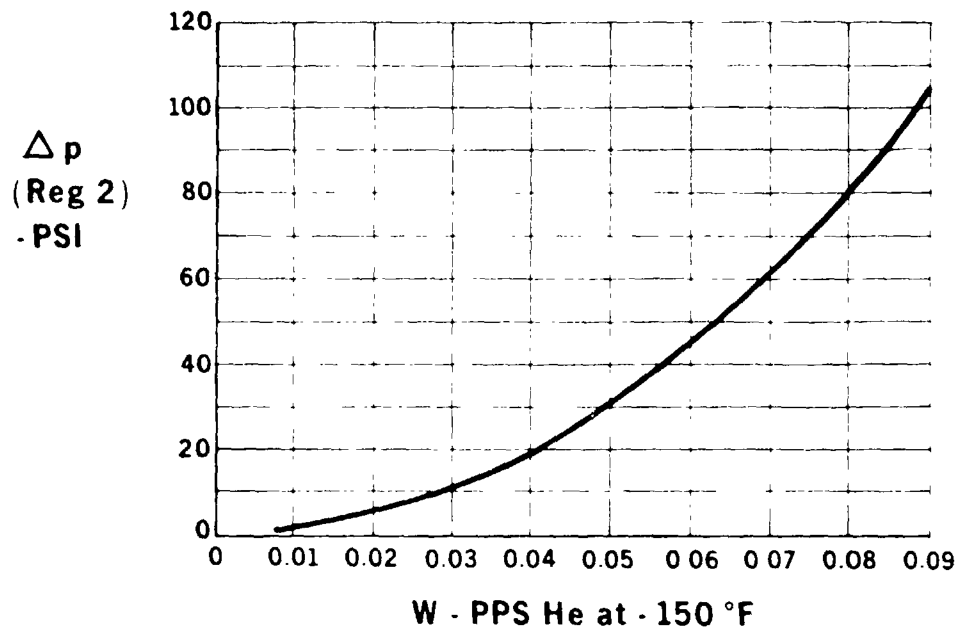


Figure 5-17

REGULATOR PERFORMANCE PROGRAM

CONFIGURATIONS I, II AND IV

Objective
- DETERMINE REGULATED PRESSURE VS STEADY STATE FLOW RATE CHARACTERISTICS OF DESIGN

Input ——
- ACTUATOR SENSING AREA AND DYNAMIC SEAL AREAS
- REGULATOR SYSTEM SPRING RATE
- NET PRELOAD
- LEVER ARM RATIO
- NET POPPET FORCE CHARACTERISTICS (FLOW FORCES AND POPPET BALANCING FORCES)
- ΔP VS FLOW CHARACTERISTIC FROM REGULATOR OUTLET TO REGULATED PRESSURE SENSING PORT
- DISCHARGE COEFFICIENT (CD) VS PRERSSURE RATIO AND AREA RATIO
- FLOW FUNCTION (CM) VS PRESSURE RATIO FOR FLOWING MEDIA
- INLET PRESSURE AND TEMPERATURE

Output ——
- REGULATED PRESSURE VS MASS FLOW RATE (LB/SEC) AT VARIOUS INLET PRESSURES AND TEMPERATURES
- REGULATED PRESSURE VS VOLUMETRIC FLOW RATE (FT^3/MIN) AT VARIOUS INLET PRESSURES AND TEMPERASTURES
- FLOW RATE VS POPPET STROKE AT VARIOUS INLET PRESSURES AND TEMPERATURES.
- REGULATOR OUTLET PRESSURE VS REGULATED PRESSURE AT VARIOUS INLET PRESSURES AND TEMPERATURES

Figure 5-18

$$P_3 = \frac{F_o - \left| K_S \overline{X} + \Delta p\, A_{BS} + F_{POP} \right|}{A_{BE}}$$

where:

P_3 = reference pressure - PSIA

A_{BE} = actuator effective area - IN^2

F_o = actuator net preload - LBS

K_S = lumped system mechanical spring rate - $\frac{LB}{IN}$

$\overline{X}$ = poppet stroke at required flow rate and inlet pressure - IN.

Δp = pressure drop across downstream regulator at required flow rate - PSI

A_{BS} = push rod shaft bellows effective area - IN^2

F_{POP} = flow force at required flow rate and inlet pressure - LBS

For the nominal dimensions of the Configuration 1 single stage direct acting regulator, the projected performance characteristic is shown in Figure 5-19 as a function of mass flow rate of helium at -150°F and in Figure 5-20 as a function of volumetric flow rate at -150°F and at 150°F. The performance characteristic of Figure 5-20 is considered more representative for evaluating performance in the anticipated OMS application since flow demand will be volumetric by virtue of expulsion of the liquid propellant at a relatively constant flow rate during thruster operation. As is evidenced by the plot of Figure 5-20 at the design flow rate of 15.8 FT^3/MIN, regulated pressure can be controlled to ± 2.25 psia over the range of inlet pressures and temperatures. As flow demand decreases, regulated pressure will tend to rise, achieving, in this case, a maximum value of 254.2 psia at zero flow. At this condition, the poppet stroke is zero; therefore, theoretically, flow is zero. However, there is no bearing load at the poppet/seat land interface. Additional increase in P_3 is required to generate this required seat load for sealing. Since this design preload is 18 LB, P_3 must increase $\frac{18\ LB}{A_{BE}} = \frac{18\ LB}{8.375\ IN^2}$ or 2.15 psi above the theoretical P_3 for zero flow to achieve lock-up.

REGULATOR PERFORMANCE

CONFIGURATION I

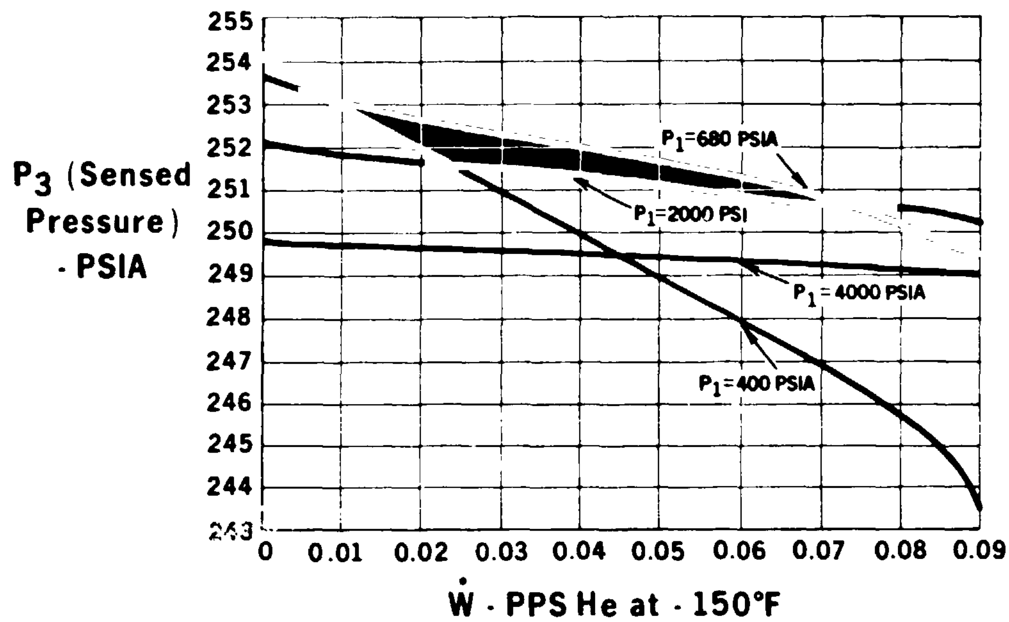


Figure 5-19

REGULATOR PERFORMANCE

CONFIGURATION I

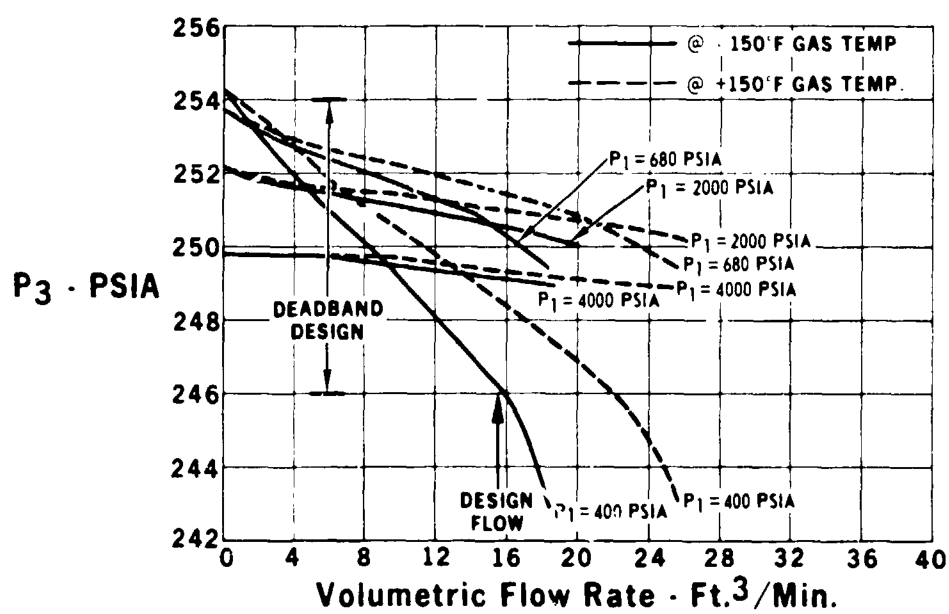


Figure 5-20

Dynamic characteristics of the design were analyzed by analog computer modeling the design and simulating worst case dynamic operation to assess damping characteristics and responses. The analog model schematic for the configuration No. 1 design is shown in Figure 5-21. This model gives consideration for moving masses, mechanical damping and gas damping. The operational mode, which Marquardt's experience has indicated as the most severe from the standpoint of dynamic loading, has been designated the "TWANG MODE". This is a hypothetical case in that it does not occur in actual operation, but it is a convenient tool for evaluating dynamic characteristics. The "TWANG MODE" consists of establishing the inlet, outlet, and poppet conditions for rated flow. The poppet is then artificially held closed, then instantaneously released at time zero, and key parameters monitored as the unit "stabilizes". Figure 5-22 and 5-23 present analog output traces of key parameters of the Configuration 1 design using three sizes of reference line to evaluate gas damping characteristics. All analog runs are performed with a simulated downstream ullage volume of one(1) cubic foot since this imposes the most severe dynamic loads. From the traces of Figures 5-22 and 5-23, stability of the regulator is achieved in less than 100 milliseconds; however, the pressure sensing orifice size has an influence on the amplitude of the oscillations resulting from the TWANG MODE operation. Stability is defined as no apparent motion of the poppet rather than in terms of a measured pressure. During computer runs, the ullage pressure was monitored and found to exhibit negligible variation (± .8 psi max.) when large amplitude oscillations of the poppet and flow rate were occurring. Since poppet movement represents a regulator life limiting parameter, stable poppet position was established as a more valid criteria for determining the dynamic stability of a design.

Schedule and cost projections, based on a comprehensive design verification and qualification test program, and production of 200 individual regulators (50 quad redundant units) were prepared. These projections are shown in Figures 5-47 and 5-48 for ease of comparison with the other candidate configurations.

5.1.2 Configuration No. 2- Direct Acting, Single Stage with Lever Arm

The addition of a lever arm, between the actuator and poppet push rod of a direct acting regulator allows modulation of the actuator force-position characteristic as it is transmitted to the poppet. This configuration, shown in Figure 5-24 allows the lever arm ratio to be selected such that an optimum actuator design can be employed. The design is identical, configuration-wise, with the configuration No. 1 design with the exception of the lever arm. The lever arm is a flexure pivot mounted element which transmits actuator motion, at the design mechanical advantage, to the poppet push rod. The poppet push rod, being separate from the actuator shaft is flexure guided to assure proper friction-free alignment. Within the design range of motion of the regulator some relative motion, at the rod ends in contact with the lever arm, will result as the lever arm rotates about its flexure pivot. Therefore, each rod end has a length of reduced diameter which allows minute deflections, without compromising column strength, such that the rod end can

ANALOG COMPUTER WIRING DIAGRAM

OMS REGULATOR CONFIGURATIONS I AND II

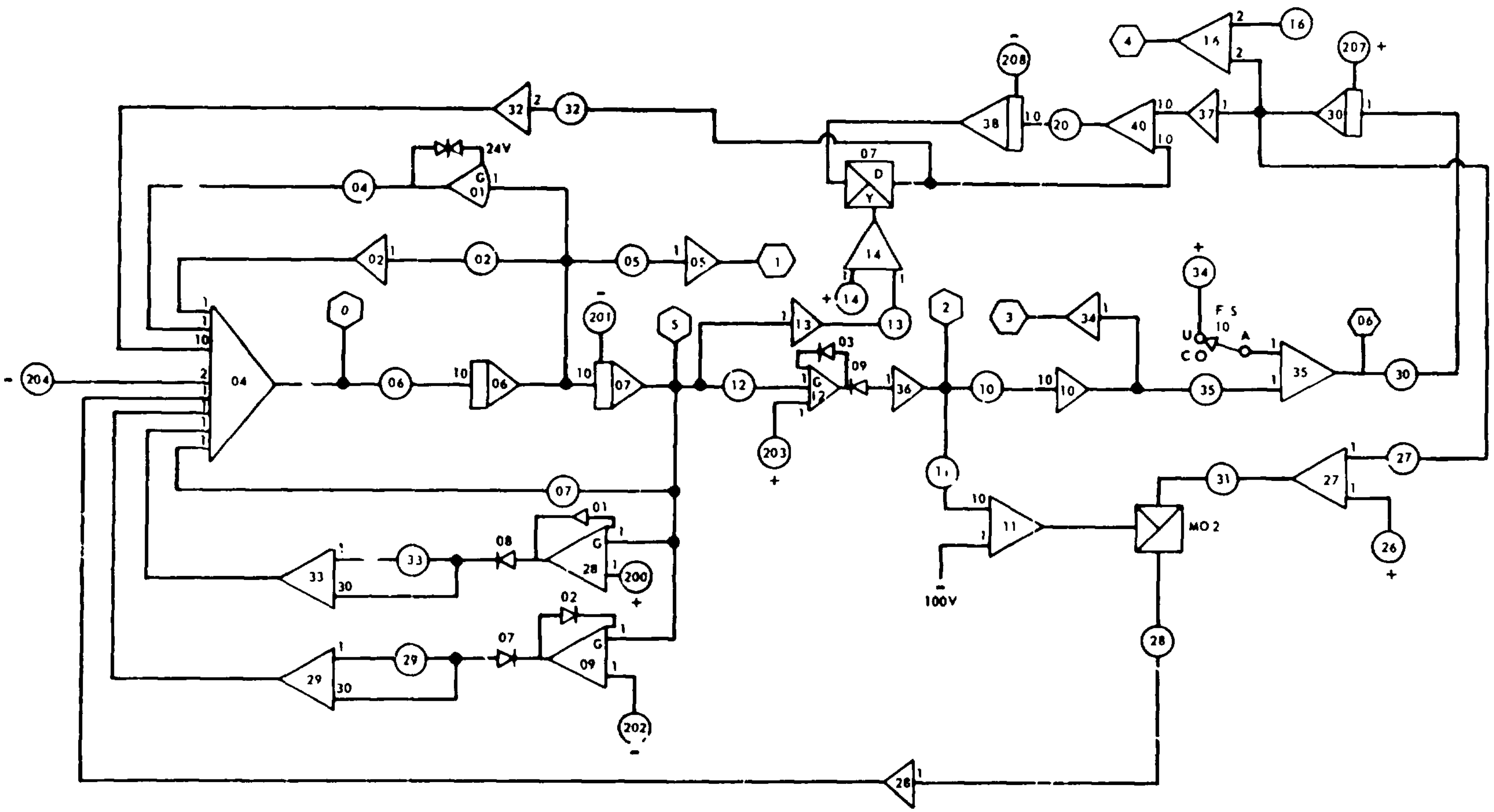


Figure 5-21

THE Marquardt COMPANY VAN NUYS, CALIFORNIA

A74-4 612-62

ANALOG COMPUTER PRINTOUT

CONFIGURATION I

Twang Mode P_{IN} = 400 PSI T = 310°R

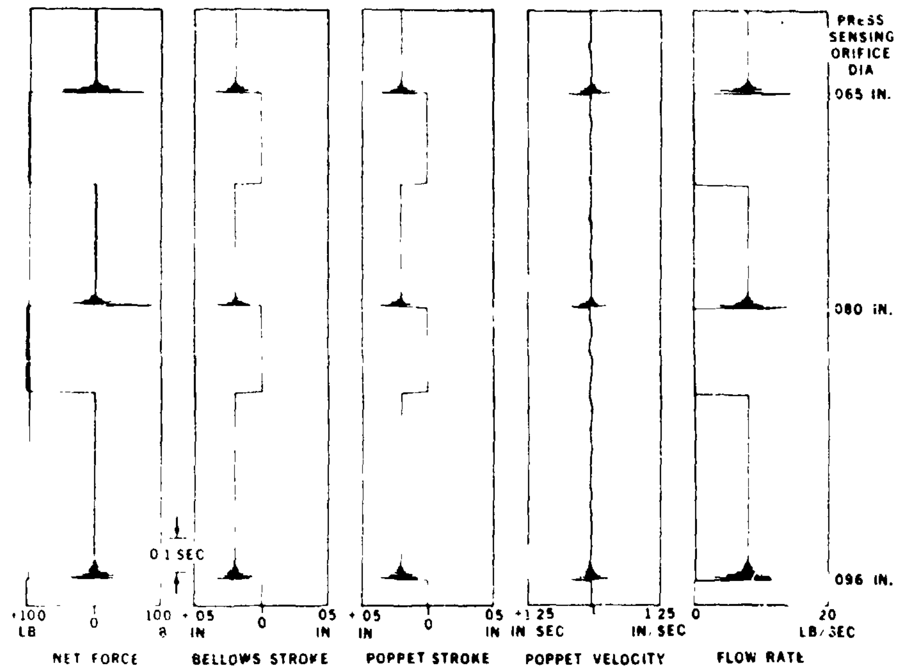


Figure 5-22

ANALOG COMPUTER PRINTOUT

CONFIGURATION I

Twang Mode PIN = 4000 PSIA T = 310°R

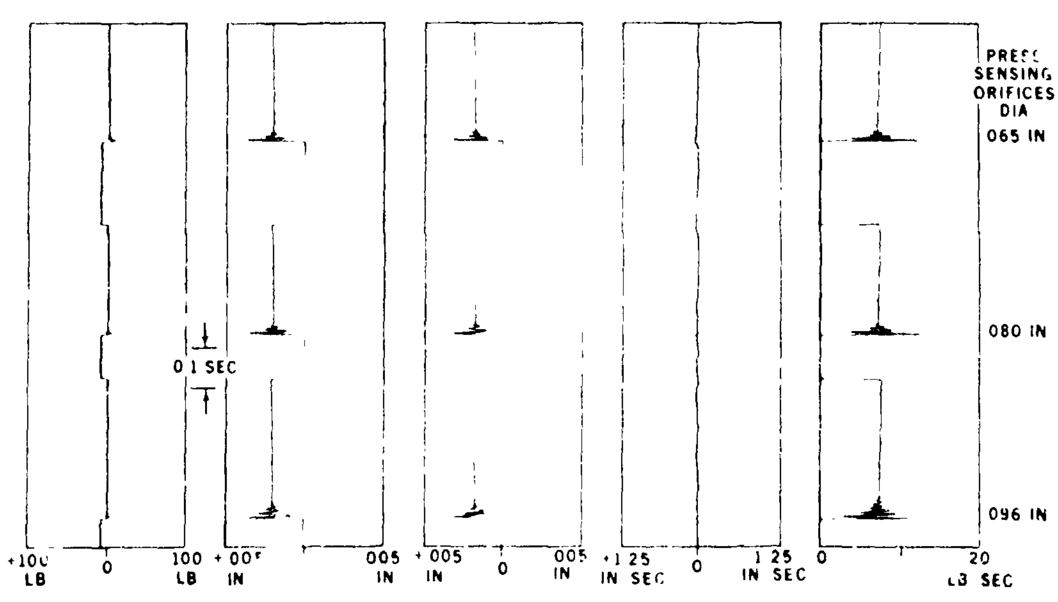


Figure 5-23

OMS REGULATOR / LEVER ARM DESIGN CONCEPT

CONFIGURATION II

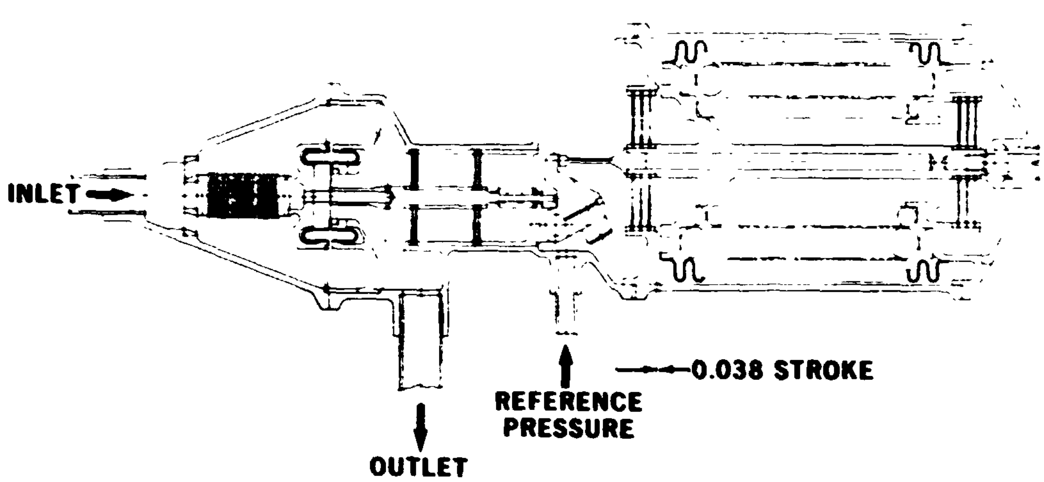


Figure 5-24

REGULATOR PERFORMANCE

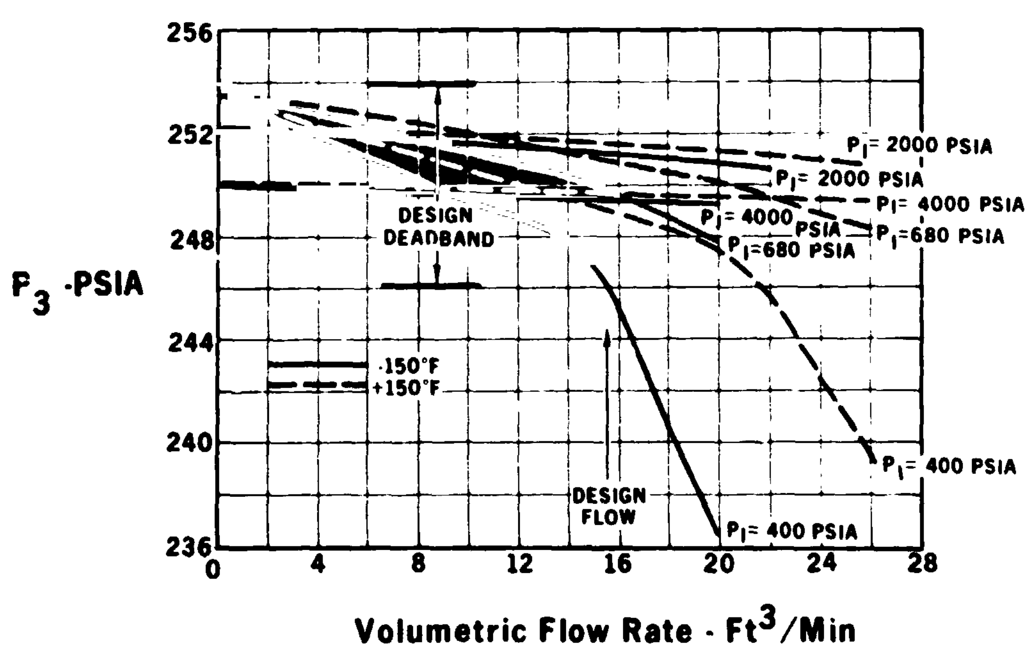


Figure 5-25

follow the lever arm contact point throughout its arc of travel, without relative motion at the interface.

The projected envelope and weight of the design, as shown is 11.0 x 3.5 x 3.2 inches and 4.95 lbs.

Performance characteristics for this design were projected by the analytical technique described in subsection 5.1.1. Consideration was given to the effect of downstream elements on the difference between unit outlet pressure and the reference or controlled pressure. The force balance equation was modified to accommodate the mechanical advantage effects of the lever arm. Performance of the design, as a function of volumetric flow rate is presented in Figure 5-25, and dynamic characteristics in Figures 5-26 and 5.27.

Cost and schedule projections for the design are shown in Figures 5-47 and 5-48 are based on attaining full flight qualification and production of 200 flight units.

5.1.3 Configuration No. 3 - Pilot Operated Regulator

As shown schematically in Figure 5-5, the Pilot Operated Regulator is a two-stage device. The pilot stage is a low capacity precision direct acting regulator. The output of the pilot regulator provides a reference load for the main stage, high flow element which provides flow in response to the pilot stage output. The pilot stage design is shown in Figure 5-28 and the main stage configuration is shown in Figure 5-29.

The pilot stage is a direct acting, spring loaded regulator employing an unbalanced poppet. The unbalanced poppet imposes some variation in outlet pressure as a function of inlet pressure, due to the change in pressure drop acting over the seat sealing area. The magnitude of this load variation is minute, due to the small seat area and is off-set by the large actuator effective area. As shown, the flow capacity of the pilot stage is designed for 5% of the design requirement (5% of 4.68 lb/min).

The outlet pressure from the pilot regulator is plumbed to the interior cavity of the main stage actuator bellows via orificed lines. The exterior cavity of the main stage actuator senses the main stage outlet pressure via an orificed line. The orificing in these lines is selected such that the pressure differential across the main stage actuator bellows is inversely proportional to the regulated pressure and this pressure differential, acting on the main stage actuator effective diameter generates a force which modulates the position of the main stage balanced poppet, relative to its seat. The pilot provides a nearly zero spring rate reference force to the main stage actuator, which minimizes the "droop" characteristic of the main stage over a wide range of flow rates. The pilot regulator amplifies the reference pressure error signal to create the magnitude of force necessary to operate the main stage. Since the main stage actuator bellows experiences only small pressure differentials, its design is not compromised by considerations for severe pressure vessel loads.

ANALOG COMPUTER PROGRAM PRINTOUT

CONFIGURATION II

Twang Mode (a/b = 2.45) PIN = 400 PSI T = 310° R

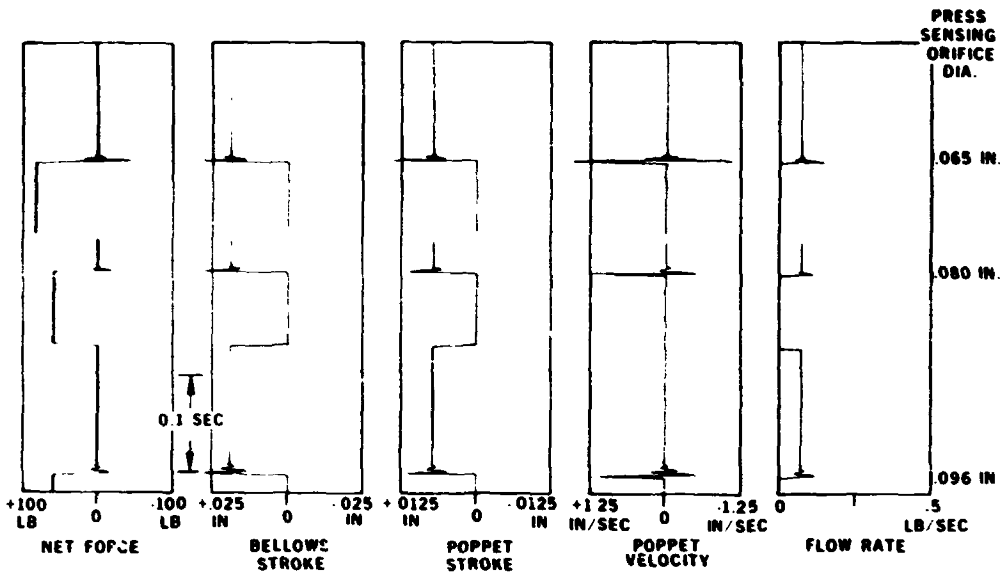


Figure 5-26

ANALOG COMPUTER PRINTOUT

CONFIGURATION II

Twang Mode (a/b = 2.45) P_{IN} = 4000 PSI T = 310°R

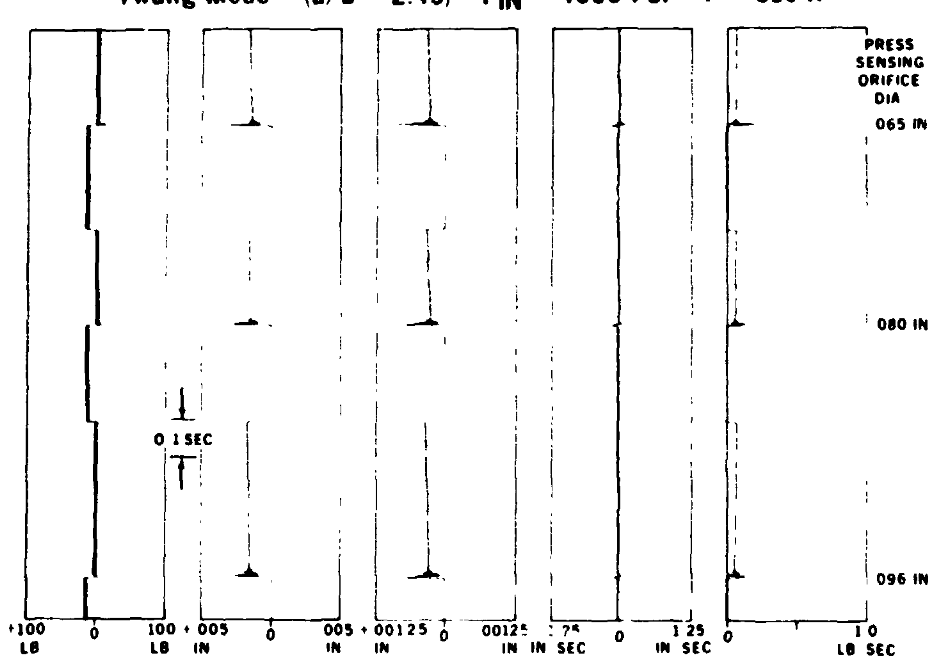


Figure 5-27

A74-4-612-66

OMS REGULATOR

CONFIGURATION NO. 3-PILOT STAGE

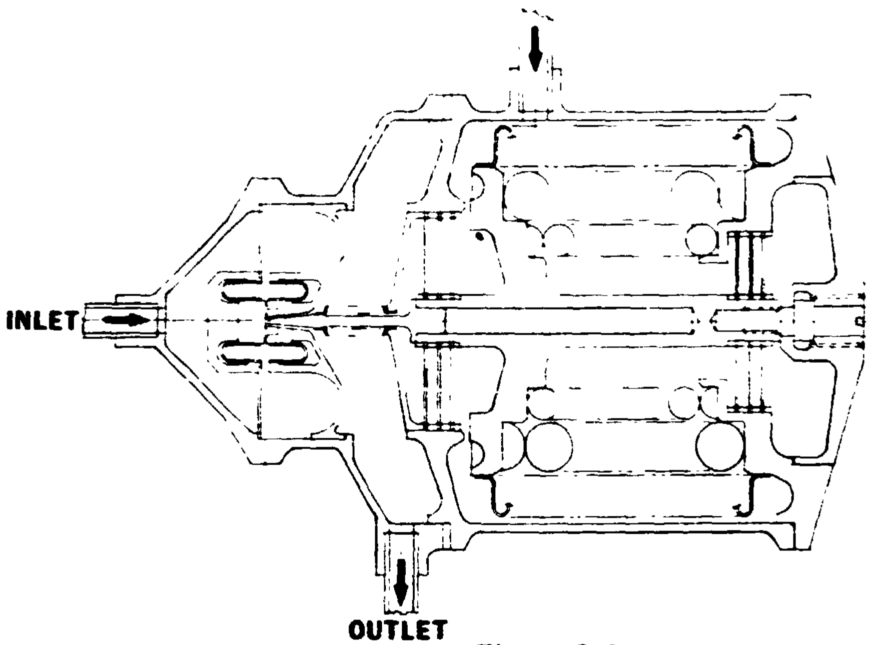


Figure 5-28

OMS REGULATOR

CONFIGURATION NO. 3 -MAIN STAGE

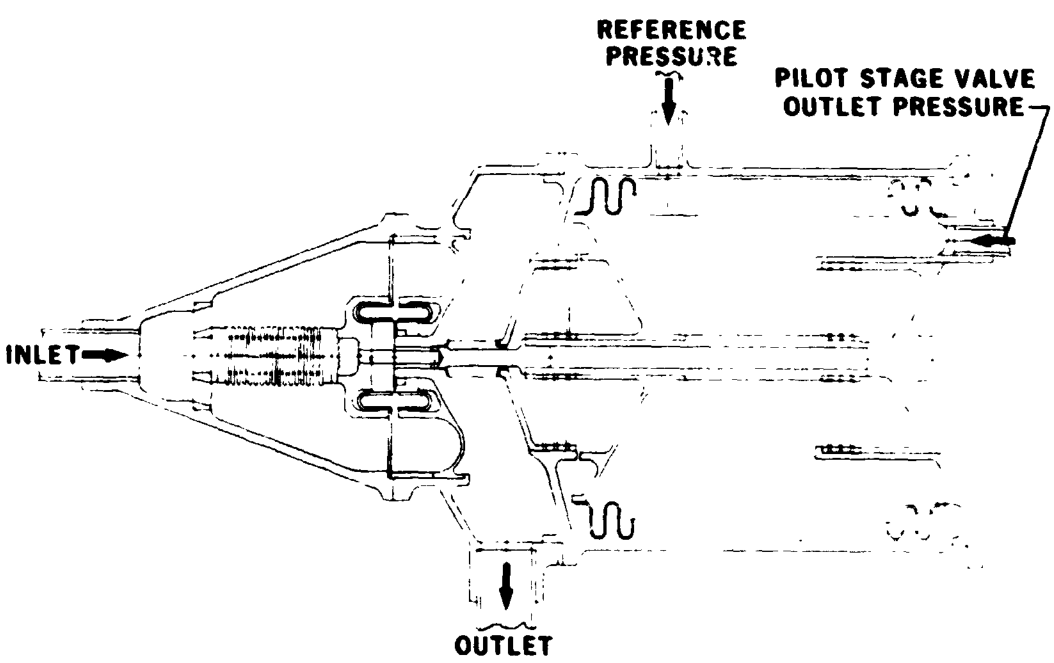


Figure 5-29

A unit weight of 6.86 lbs is projected for this configuration, and envelope requirements are 9.5 x 3.4 x 6.6 inches.

Performance analysis is a more complex problem for the pilot operated regulator, but makes use of the techniques previously described for the other configurations. The pilot regulator is analyzed as a single stage direct acting regulator similar to Configuration No. 1 but having a flow capacity of 5% of the rated flow and incorporating an unbalanced poppet. For each selected inlet pressure, the outlet pressure vs. reference pressure characteristic can be determined on the basis of the selected orificing of the respective interconnect lines. The main stage performance is then established on the basis of solving the force balance equations wherein the pressure differential between the pilot regulator outlet pressure and the main stage outlet pressure act on the main stage actuator effective area to generate the reference force. The performance characteristic of the nominal design is plotted in Figure 5-30. As is evidenced by this plot, the design exhibits little "droop" over the design flow rate range (0 to 15.8 FT^3/min.). This characteristic is attributed to the low system spring rate of the design, which is the predominant driver at low inlet pressures (400 psia).

Dynamic analysis of the pilot operated regulator is an order of magnitude more complex, as is apparent from the analog block diagram of Figure 5-31, than for the single stage design. Numerous analog computer runs were made for this configuration, with various flow passage orifice combinations, in an attempt to achieve stable "Twang Mode" operation. Though oscillations of the delivered pressure could be reduced to a low amplitude (less than ± .4 psia) the pilot and main stage poppets never achieved stability and these mechanical motions constitute a potential wearout mode which severely degrades reliability. Changes in system spring rates and spring rate allocations were also iterated but elimination of mechanical oscillation, in spite of "stable" performance, could not be achieved. The inability of the design to achieve absolute stability is attributable to the absence of positive feedback between the pilot and main stage poppets, so that poppet position errors are negated. The addition of positive poppet position feedback negates the basic design concept and results in a significantly more complex configuration. This lack of positive feedback and hence failure to be stable with respect to performance and mechanical oscillation is inherent in the basic pilot operated desing concept. Though the amplitude and frequency of oscillations can be minimized, to a great extent, this instability characteristic is a serious detriment for a multi-mission, high environmental load, high reliability application.

Cost and schedule projections for the pilot operated design are presented in Figures 5-47 and 5-48, based upon attaining full flight qualified status and production of 200 flight regulators.

REGULATOR PERFORMANCE

CONFIGURATION III

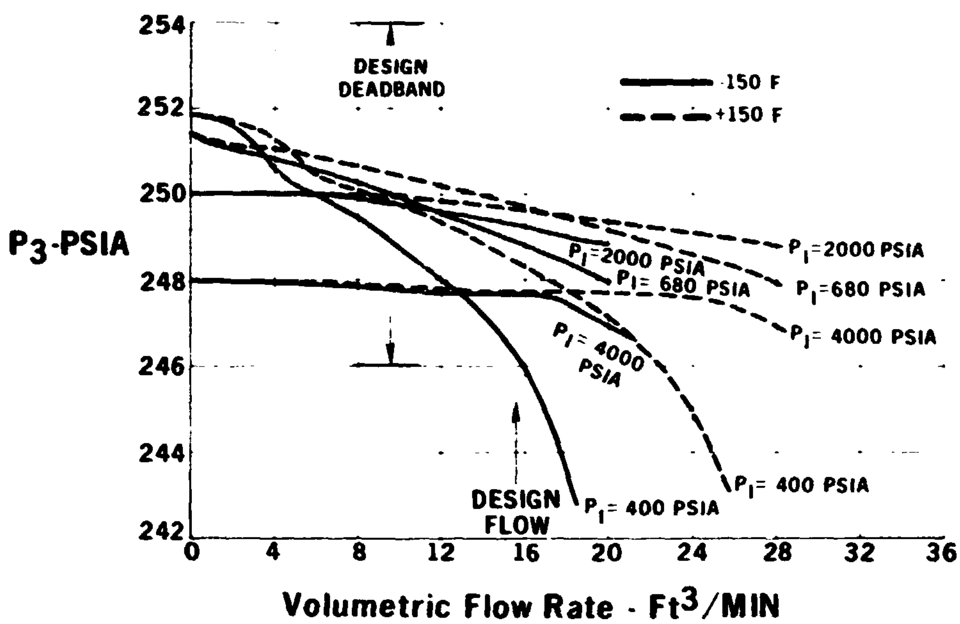


Figure 5-30

PILOT-DOME LOADED REGULATOR

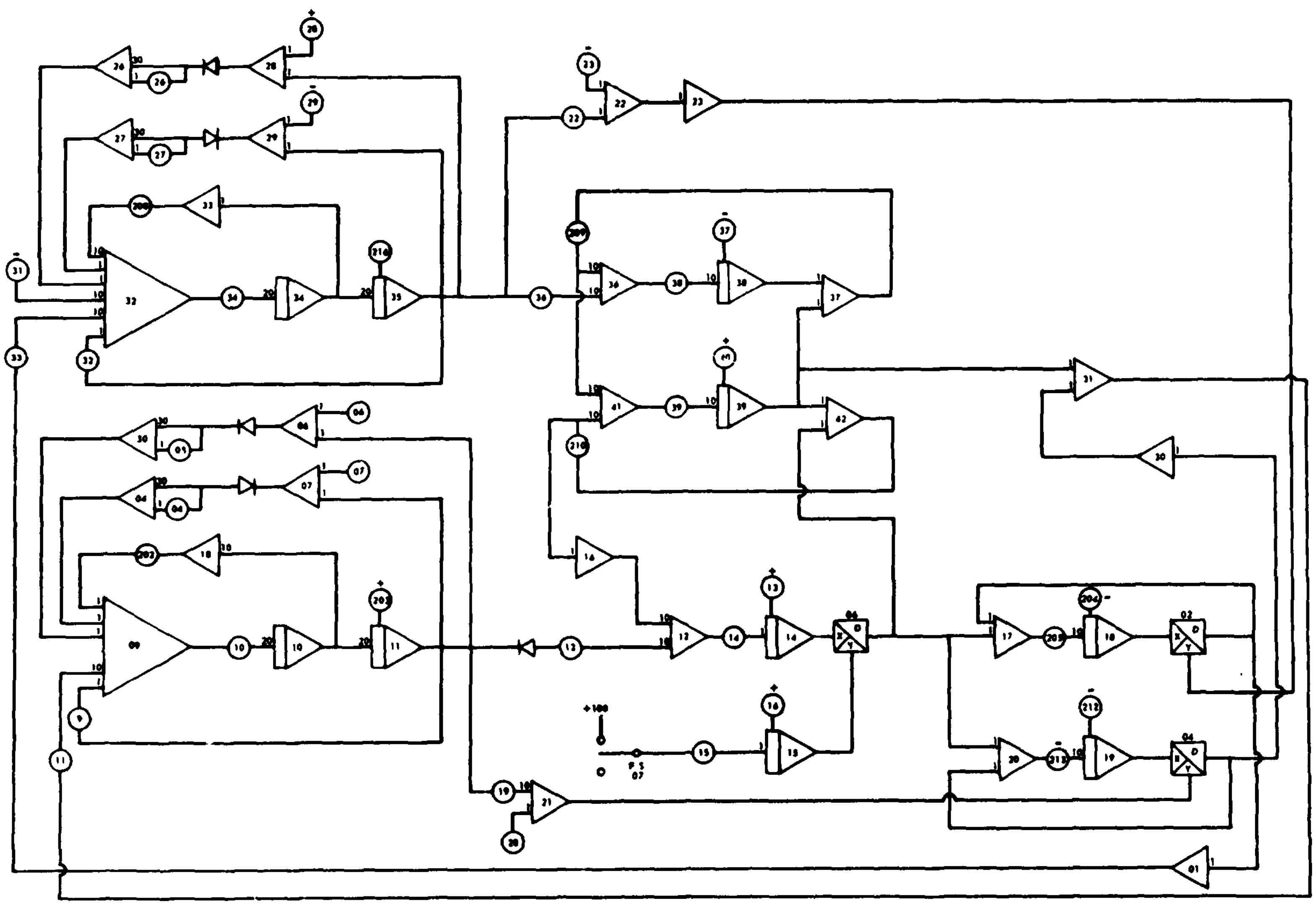


Figure 5-31

A74-4-612-68

5.1.4 Configuration No. 4 - Direct Acting Single Stage with Push-Pull Rod

Positive attachment of the actuator push rod to the metering poppet results in the single stage configuration shown in Figure 5-32. This layout also illustrates the installation of a replaceable integral inlet filter sized for 150 μ absolute filtration of regulator influent. Positive linking of the actuator and poppet of the single stage regulator allows the poppet balancing bellows to be installed without any preload and poppet/seat interface loads then become proportional to reference pressure, enhancing seat life by assuring that only the seat load necessary to preclude outlet pressure rise is applied at the interface. With the exception of the positive attachment of the push rod to the poppet, this design is identical with that of Configuration No. 1 in every respect.

The performance characteristics of this configuration are plotted in Figure 5-19 and 5-20 and are identical to the performance of Configuration No. 1 by virture of the similarity in design criteria which affect performance. Dynamic characteristics in "Twang Mode" operation are also identical to Configuration No. 1 as shown in the analog block diagram of Figure 5-21 and the traces of Figures 5-22 and 5-23.

Cost and schedule projections through flight qualification and production of 200 flight units are shown in Figures 5-48 and 5-49.

The envelope for this configuration is identical to that of configuration No. 1 (8.0 x 4.0 x 4.0 inches) and the weight estimate is somewhat higher (8.0 lbs) due to the additional elements required to make the positive attachment of the push rod to the poppet.

5.2 REGULATOR COMMON FEATURES

All of the configuration designs described in the previous subsections incorporate a number of features which are common to all designs and which are considered mandatory for any regulator design to meet the Space Shuttle OMS regulator requirements. These features are described in detail in the following subsections.

5.2.1 Flow Limiter

Initial efforts to define the design of a flow limiter that would limit regulator flow to no greater than 10 lb/min. were based on the assumption that the regulators or at least one regulator of a series unit would be functional. Consequently, flow limiter designs were located downstream of the regulators and assumed a relatively constant inlet pressure (regulator outlet pressure) and the concepts of Figure 5-33 were evaluated. The passive uncompensated nozzle offered the ultimate in reliability within the constraints of the assumption and did not compromise regulator design in the least.

OMS REGULATOR
CONFIGURATION NO. 4

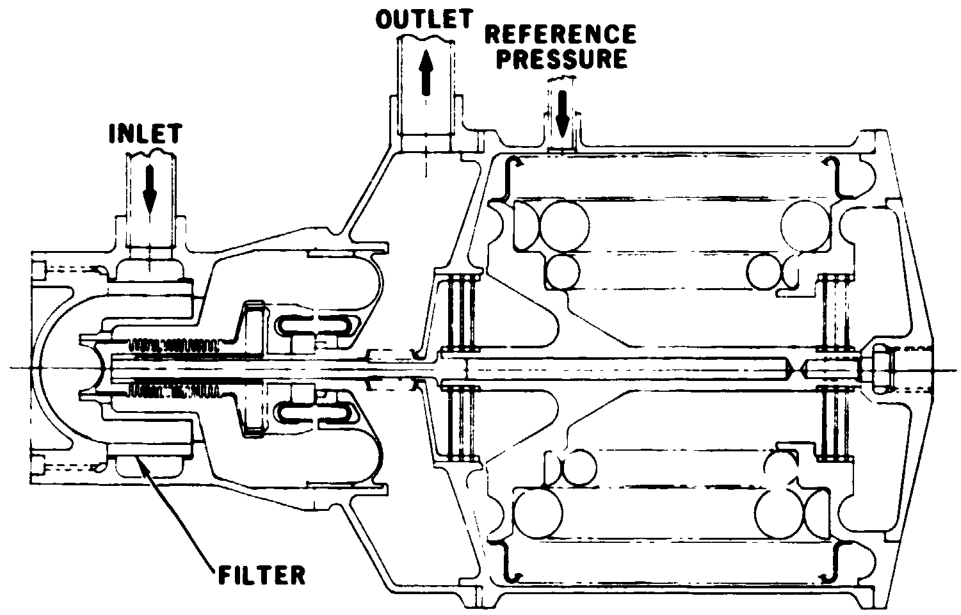


Figure 5-32

CANDIDATE FLOW LIMITER SCHEMATIC

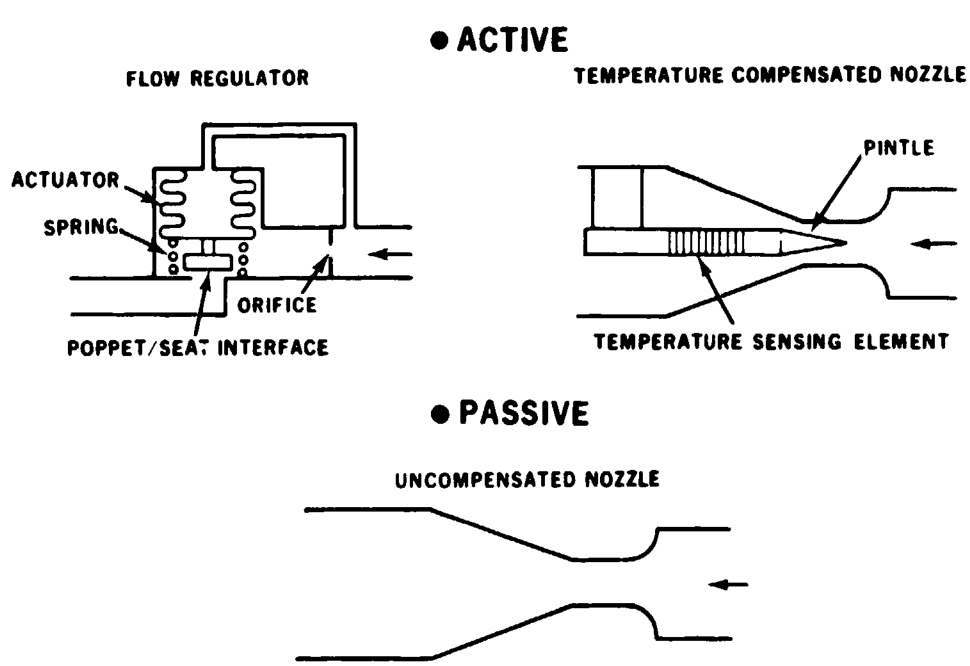


Figure 5-33

However, the primary function of the flow limiter was to limit flow in the event of a total regulator failure. This ground rule imposed criteria which dictated that an active, pressure compensated device was required which would minimize pressure drop at or below rated flow (4.68 lb/min) yet preclude flow rates in excess of 10 lb/min. The concepts of Figure 5-34 were deemed as feasible devices for performing this function and would be located in the inlet line to each series leg of a quad redundant regulator package.

The two pressure actuated devices are essentially area schedules which establish a flow area, as a function of inlet pressure, which effects choking at a flow rate between 7 and 10 lb/min., dependent upon gas temperature, at the respective sensed inlet pressure. Inlet pressure is sensed by a metal bellows with one side vented to ambient. The pressure differential creates a force, counter balanced by the device spring rate forces, such that a metering element is positioned to define the critical area for that inlet pressure.

The spring actuated venturi configuration represents a more complex design challenge though the design offers the distinct advantage of no dynamic seals to external leakage. The design challenge evolves from the low ratio of maximum allowable flow rate to nominal flow rate, the wide operational temperature range and wide range of inlet pressures. The design must, therefore, schedule flow area in response to the appropriate integration of drag loads and pressure drop such that critical flow results as mass flow approaches 10 lb/min. under any and all conditions. The elements must, therefore, incorporate contours which give the appropriate aerodynamic response.

In view of the magnitude of the design effort required to perfect the spring actuated venturi design, the pressure actuated poppet design was selected as the primary configuration on the basis of its simplicity and highest level of confidence in developing valid design criteria. A parallel effort to refine the spring actuated venturi configuration would be conducted prior to selecting a design for fabrication.

For the conditions of the OMS application, the flow limiter is shown in Figure 5-35. This configuration offers flow area scheduling in accordance with Figure 5-36 and projected flow limiting as a function of temperature as shown in Figure 5-37. The flexure guided pressure balanced poppet moves to schedule the critical flow area in response to the inlet pressure acting on the spring loaded bellows pressure sensor. Positive stops, at the poppet/seat interfaces establish minimum and maximum flow areas of the device and represent the only points of contact of the moving element with the body.

PRESSURE COMPENSATED FLOW LIMITER CANDIDATE SCHEMATICS

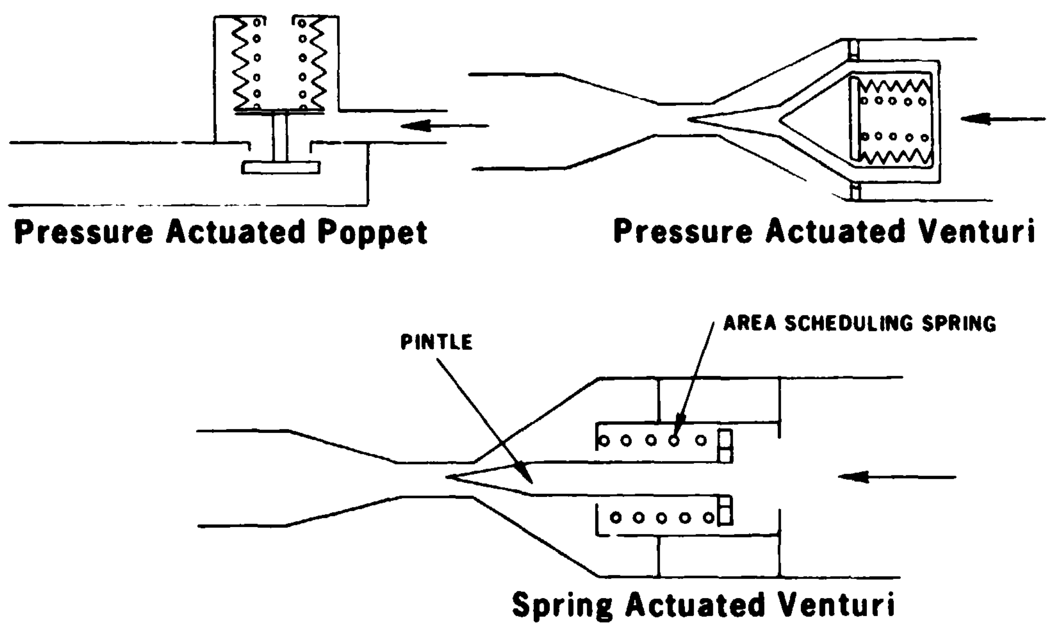


Figure 5-34

OMS FLOW LIMITER
INLET PRESSURE COMPENSATED

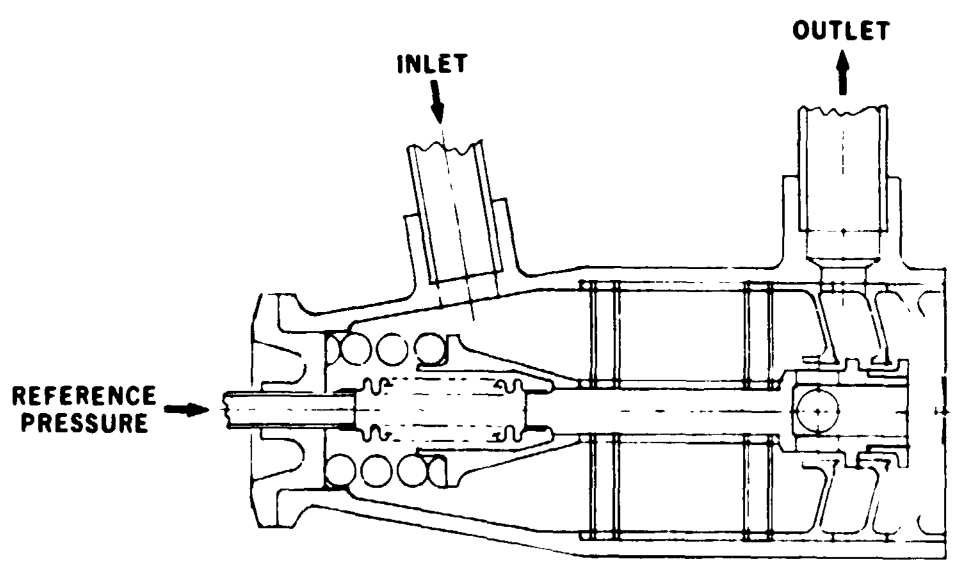


Figure 5-35

ACTIVE FLOW LIMITER AREA CHARACTERISTICS

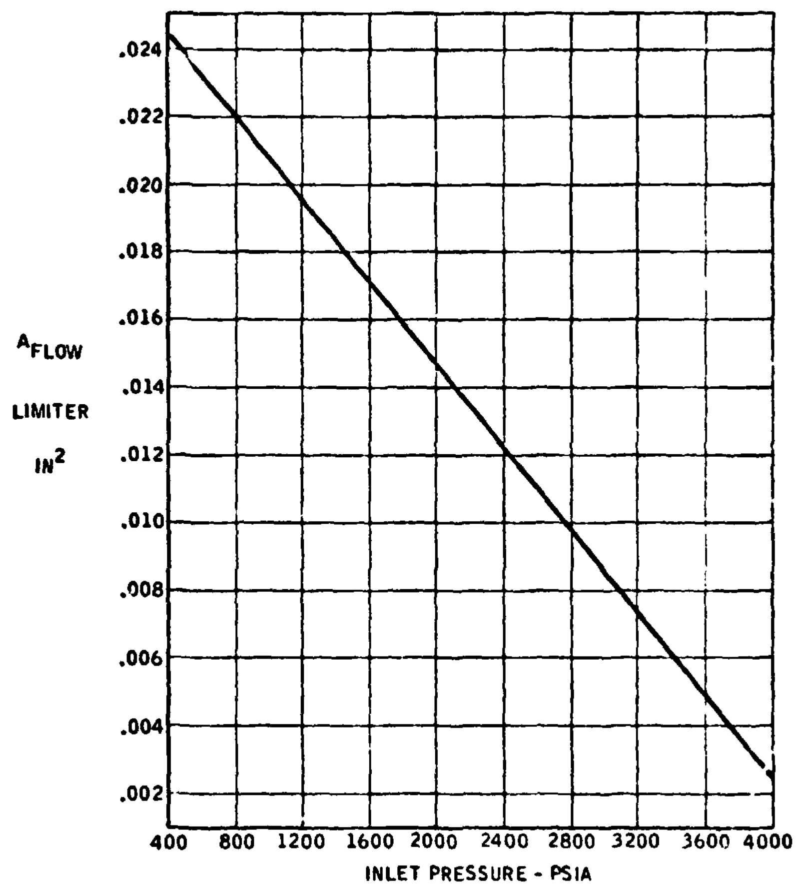


Figure 5-36

ACTIVE FLOW LIMITER

MAXIMUM FLOW VS TEMPERATURE

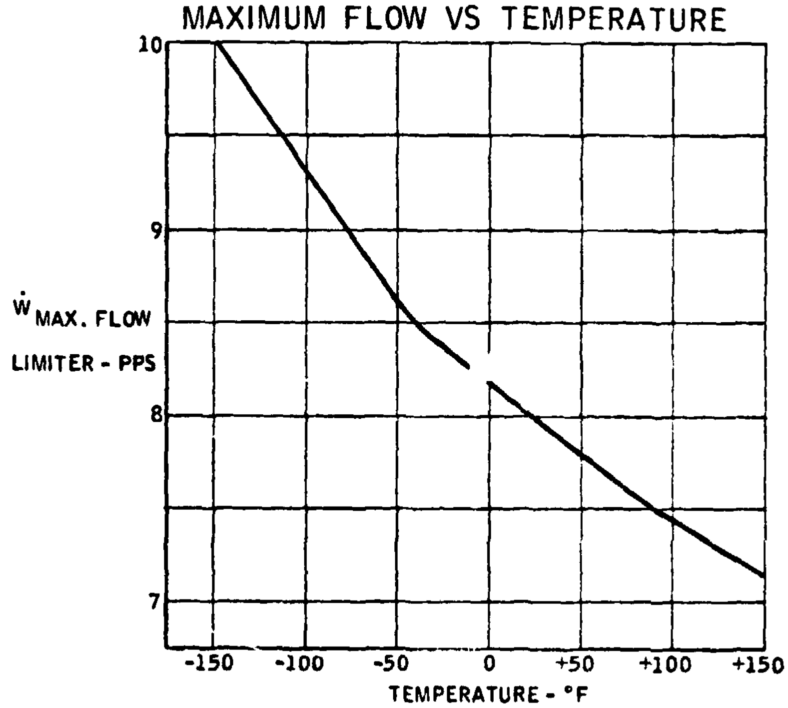


Figure 5-37

5.2.2 Inlet Filter

An integral cylindrical filter element, designed to fit within the inlet cavity of each configuration has been sized for the rated flow conditions and anticipated environment. The element consists of a 50 x 250 single Dutch Twill woven wire cloth layer supported on a perforated cylinder and welded to end rings which act as metal face seals in the installed position. The wire cloth provides 150 micron absolute filtration of influent and contributes less than 5 psi pressure drop at rated flow, 400 psia inlet pressure and -150°F fluid temperature. Installation of the filter element, as typically shown in Figure 5-32 include provision for replacement of the filter element in the event of plugging. The philosophy of element replacement is that if filter clogging occurs and replacement is warranted contamination level experienced was excessive and thorough inspection of the regulator seat and poppet seal surfaces is warranted as an integral part of filter element replacement. Post replacement regulator check out is also desireable to substantiate leakage characteristics. These procedures require removal of the regulator from the system and controlled area servicing.

5.2.3 Poppet/Seat Interfaces

Design philosophy and design parameters for achieving the contamination tolerance required for the OMS application were delineated in Reference 1 and have been reaffirmed by further analyses during this effort. The seat profile, illustrating the sealing land and the bumper land is depicted in Figure 5-38 and the installation into the regulator is shown in Figure 5-39. The knife edge sealing land and bumper land configuration were selected to achieve the following high contamination tolerance, long life objectives;

(a) Minimum scrubbing at seal interface

(b) Seal interface stress level compatible with achieveable surface finishes and leakage requirements

(c) 150 micron hard particle cutting capability

(d) True alignment of sealing interfaces.

Contaminant cutting, to achieve high particulate contaminant tolerance results from the poppet closing preload acting on a particle trapped between the seat knife edge sealing land and the poppet hard face. On the basis of creating a bearing stress of 300,000 psi on a particle, the cutting or crushing of the particle will be effected without jeopardizing seal interface integrity, using the ceramic materials selected for the respective parts. For the nominal bumper and seal land diameters of the designs, the relationship between crushing force and particle size is illustrated in Figure 5-40. The bumper to seal land spacing is taken into consideration as well as a worst case location of particle-to-bumper contact-to-point of force application.

CROSS SECTION OF SEAT SEALING LAND AND BUMPER

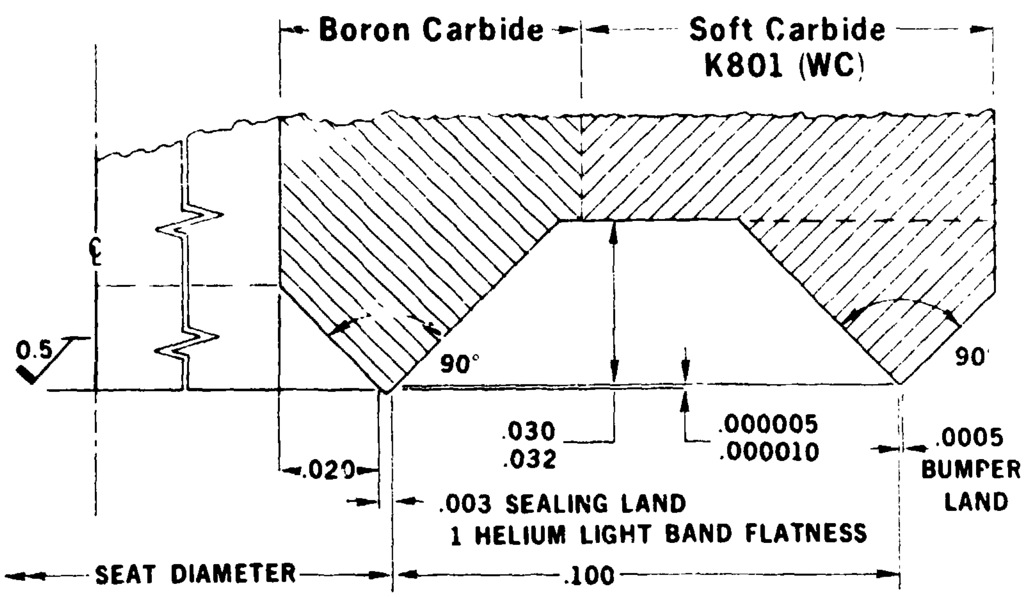


Figure 5-38

CROSS SECTION OF SEAT AND POPPET

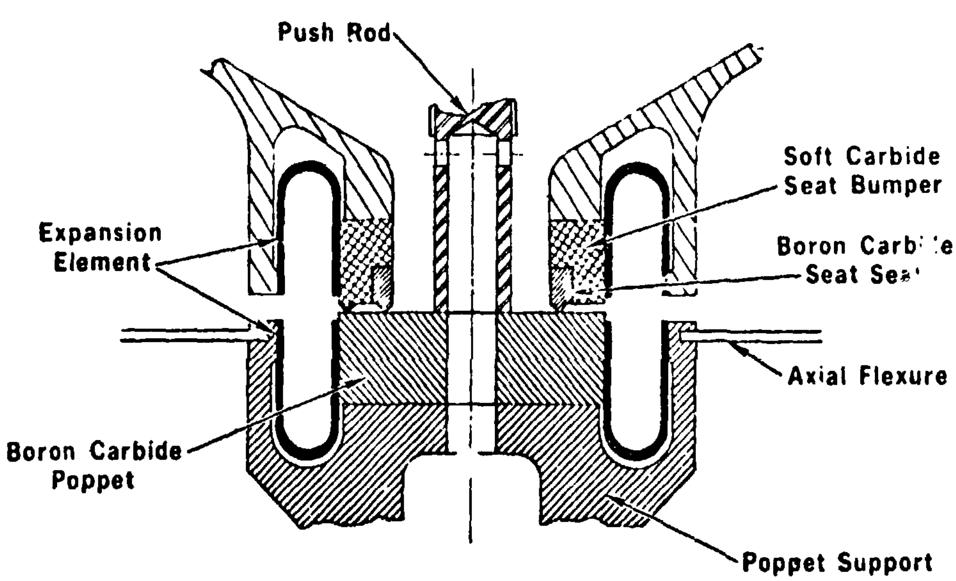


Figure 5-39

REQUIRED CONTAMINANT CRUSHING FORCE

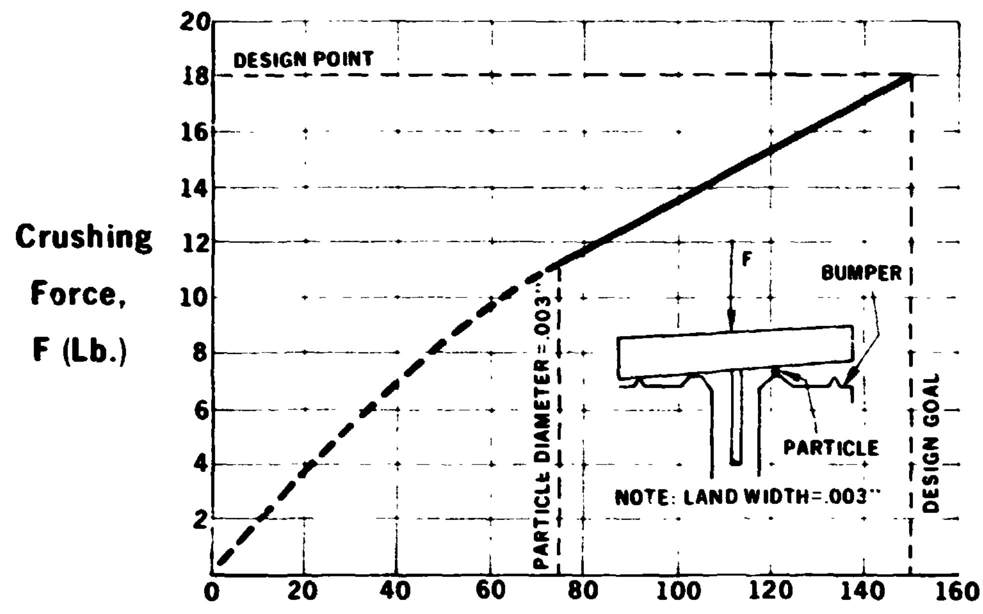
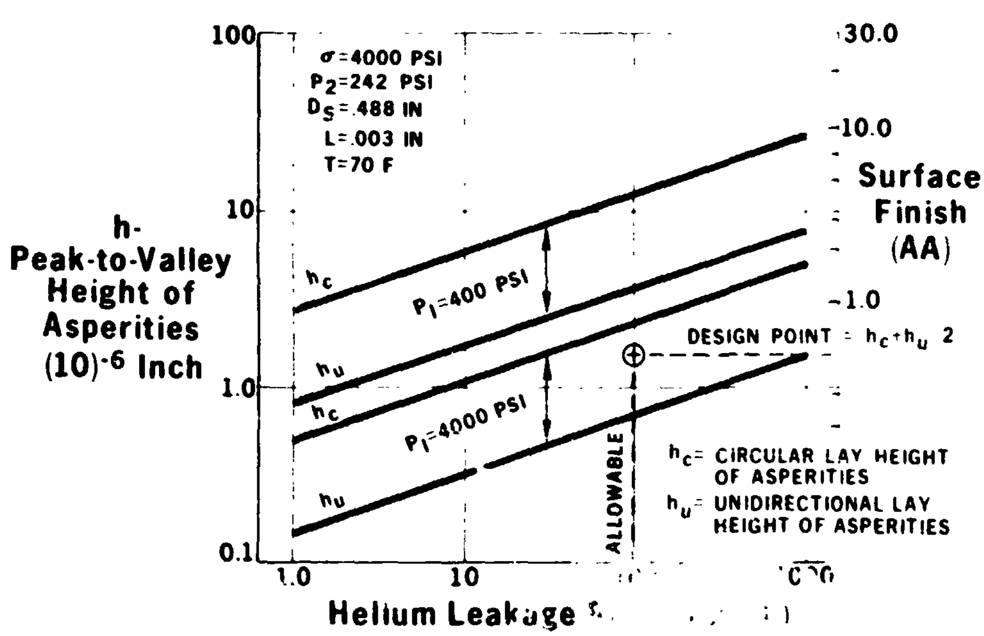


Figure 5-40

PREDICTED SURFACE FINISH REQUIREMENT



Figure

Surface finish requirements of the sealing surfaces were established on the basis of a leakage math model developed during prior Marquardt hard seat valve development. For the leakage rates required and the helium inlet pressure range, seal surfaces must be lapped to a finish of at least 1/2 AA. This quality of surface finish has been attained by Marquardt on previous hard seat seal surfaces and is compatible with the selected interface materials. The relationship between surface finish and leakage rate, as determined by the leakage model is shown in Figure 5-41. h_c and h_u refer to circular lay and unidirectional lay, respectively, of the finish. Since actual seal surface lapping results in a randum lay surface, the selected surface finish design point was chosen as an average of the finishes required to meet the 100 scc/hr leak rate.

The seat sealing diameter was selected on the basis of required flow area to meet flow requirements at minimum inlet pressure (400 psia) and minimum temperature (-150°F). Experimental data from prior valve development efforts were employed to incorporate allowance for pressure ratio and area ratio influence on flow coefficient (C_D). For the required flow rate, a range of poppet strokes and seat diameters will provide adequate flow area. This relationship is shown in Figure 5-42. The selected design points for the configurations are noted on the figure at their respective area ratios

$$\left(\frac{\text{flow area across minimum seat dia}}{\text{annular flow area downstream of seat}}\right).$$

This area ratio is a key input to the flow force analysis (POP program).

5.2.4 Bellows Types

Metallic bellows have been selected for all dynamic sealing functions required in each regulator configuration. These functions include poppet balancing, shaft seal, and actuator seal. Both welded and hydroformed bellows were evaluated for the respective applications. Though the welded bellows offer envelop advantages which impact on overall unit weight, single ply hydroformed bellows were selected on the basis of their cleanability, inspectability and validityof analytical techniques employed to establish life characteristics (200,000 cycles of maximum working stroke selected for this application).

A survey of hydroformed bellows fabricators was performed to establish available die sizes in the design range of interest for this application. At this time, fabricators were also solicited for normal tolerances on effective area, which may be expected over the operating pressure range and stroke range. Normal manufacturing tolerances could result in effective area repeatability of as much as ± 7%. For applications where effective area is critical (poppet balancing bellows) proper acceptance testing and matching of seat sealing diameter to bellows effective diameter should result in area matching capability of ± 1%.

PREDICTED SEAT DIAMETER AND STROKE DESIGN POINT RANGE

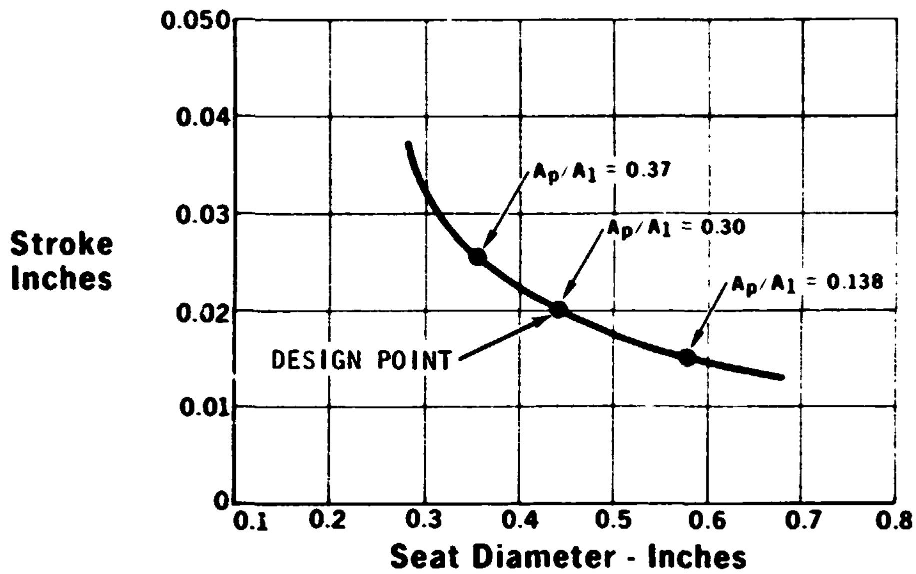


Figure 5-42

MATERIALS OF CONSTRUCTION

Element	Material
• Body & Housing	Inconel 718
• Bellows	Inconel 718
• Springs	Inconel 718
• Flexures	Inconel 718
• Push Rod	Inconel 718
• Support and Structural Elements	
• Seat	Boron Carbide (B_4C)
• Poppet Seal Surface	Boron Carbide (B_4C)
• Seat Bumper	Tungsten Carbide (K-801)
• Braze Alloy	Nioro (Au-Ni)

Figure 5-43

Design pressure capability for each bellows, shall be the applicable proof pressure (1.5 x operating pressure) for the respective application, without bellows degradation and a bellows burst pressure of at least 2.0 times the maximum operating pressure.

5.2.5 Materials of Construction

Based upon the materials-propellant compatibility study described in Section 4.0, materials of construction were selected which offer the highest probability of survival in the anticipated environment and also possess desirable mechanical properties. The selected materials of construction are summarized in Figure 5.43.

5.2.6 Flexure Guidance

A paramount objective of this design definition effort was the elimination of sliding fits, within the flow cavity, to enhance the units' multi-mission reliability. To achieve this goal, all moving elements. of each configuration, are supported on uniquely designed flexure elements. These flexures provide high radial stiffness but relatively low axial stiffness, thereby allowing uniaxial motion of the moving element. The configurations of flexure elements for this application, along with a summary of characteristics is presented in Figure 5-44.

The flexure is fabricated of sheet stock which is chemically milled to produce a precise pattern of beam elements. With the inner and outer annular rings constrained, axial movement of the inner ring, relative to the outer ring results in bending and torsional stresses in the beam elements. The poppet guidance flexure is a "three-lobe" design, thus a single flexure provides balanced radial support. The actuator guidance flexure is a "single-lobe" configuration. Three of these flexures, with spacers between flexures, are assembled into an element, such that the radial webs or spoke-like elements, are oriented 120° apart. This assembled element provides balanced guidance.

As installed in the regulators, the flexures allow all moving elements to have large radial clearances yet assure precise, repeatable motion with no rubbing friction.

5.2.7 Life and Maintenance Characteristics

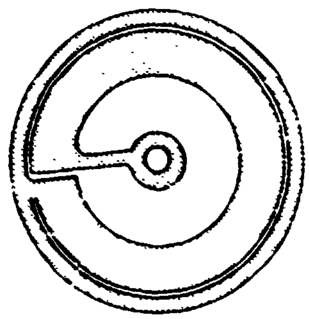
Each configuration design incorporates consideration for the life and maintenance requirements specified in the contract work statement. Particular features, that contribute to the achievement of these goals are summarized in Figure 5-45. In addition to these features, the materials of construction selection was based upon the projected compatibility with the anticipated environments. Operating stress levels will also be well below endurance limits on the basis of a detailed stress analysis.

FLEXURE GUIDENCE

Features —

- NO SLIDING FITS OR CONTACT BETWEEN MOVING PARTS
- CONTROLLABLE AXIAL SPRING RATE (LOW) AND RADIAL SPRING RATE (HIGH)
- INFINITE CYCLE LIFE
- CLEANABLE, INSPECTABLE, NO CONTAMINANT GENERATION

WIDE RANGE OF MATERIALS OF CONSTRUCTION



ACTUATOR GUIDANCE FLEXURE

Capability, Based on Demonstrated Usage

- AXIAL SPRING RATE 30 TO 600 LB/IN.
- RADIAL SPRING RATE 10,000 TO 300,000 LB/IN.
- AXIAL STROKE TO 1.0 INCHES
- CYCLE LIFE 2×10^6 CYCLES

POPPET GUIDANCE FLEXURE

Figure 5-44

LIFE AND MAINTENANCE REQUIREMENTS

Design Goals	Design Approach
● 7yr Storage Life	No Age Control Materials
● 5yr Operational Life	No Sliding Fits, No Lubricants, Materials Inert to Environment
● 200,000 Cycle Life	All Stresses Below 200,000 ~ Endurance Limit, No Sliding, Rubbing or Scuffing Action, Premium Quality Materials.
● No Scheduled Maintenance	No Lubricants, Purgeable, No Adjustments. *

* PERIODIC SETPOINT ADJUSTMENT DUE TO SPRING RELAXATION IS STILL UNDER EVALUATION.

Figure 5-45

5.3 EVALUATION

A primary goal of this effort is the generation of performance and physical characteristics data for each of the candidate designs such that a valid objective evaluation can be made to arrive at the selection of the optimum design concept for the Space Shuttle OMS Helium Regulator. The analytical and design studies described in the previous paragraphs generated this type of data. A summary of this data, for the four configurations, is tabulated in Figure 5-46.

In Figure 5.47, a schedule is presented which projects the development time required to evolve any of the four candidate designs to full flight qualified status. The schedule provides for a design verification phase, during which, compatibility with performance requirements and usage environments will be demonstrated on flight configurations. Qualification testing is then performed and flight units fabricated. No significant schedule disparities for development of any of the candidate designs, are anticipated.

In Figure 5-48, cumulative cost projections, for each of the four configurations, are plotted as a function of time. The time base is based on the development schedule defined in Figure 5-47. Cost differences are approximately proportional to the relative complexity of the four configurations.

As is evident from Figure 5-46 and previous discussions, the performance characteristics projections, for all configurations, are within the design envelope of the contract work statement. Minor differences result from differences in spring rates and dimensional relationships, but do not represent a significant basis for evaluating the four configurations.

Weight and envelope comparisons indicate wider disparities but these parameters, at this stage of the design development, do not represent the optimum values for the respective designs. However, it is anticipated that the relative weights of the optimum designs will be in approximately the same relationship as indicated by these projections.

The commonality of many features of the designs (Ref. Para. 5.1.2) minimzes the complexity of performing an objective trade-off. In Figure 5-49, a Summary Trade-Off matrix is shown which contains five categories of evaluation which can be appraised on the basis of the objective data of Figures 5-46 through 5-48. The sixth category, reliability, is more subjective in nature at this time since the designs have not matured sufficiently to perform a valid reliability prediction. For purposes of this appraisal, reliability was assummed to be proportional to design complexity and weighting was performed accordingly.

As is evident from the Trade-Off Summary (Figure 5-49), no one candidate configuration is indicated as being overwhelmingly the optimum selection. However, on the basis of this evaluation, the pilot operated design (Configuration III) appears to be the least desirable choice.

REGULATOR CHARACTERISTICS SUMMARY

CONFIG.	I	II	III	IV
Deadband at Rated Flow (Volumetric) -150 to +150 °F - PSI	5.5	5.8	3.8	5.5
Max Volumetric Flow for ±4 PSI Deadband -Ft^3/Min	15.6	15.6	17.7	15.6
Deadband 0 to Rated Flow PSIA	8.0	8.0	5.8	8.0
Unit Weight (Single Reg) Lb.	7.95	4.95	Pilot 3.83 Main 3.03 6.86	8.00
Lock-Up Pressure (Max) (Based on 18# Poppet/Seat Interface Load PSIA	256.45	257.66	258.20	256.45
Time to Stabilize (Max., Twang Mode, 400 or 4000 PSI Inlet)	50 Ms	20 Ms		50 Ms
Overshoot ($\dot{W}$ Max)	0.062 PPS	0.065 PPS		).062 PPS
Undershoot ($\dot{W}$ Max)	0.046 PPS	0.05C PPS		0.046 PPS
Inlet Pressure Sensitivity (Set Point Shift at 4.68 LB/Min. -150°F) 4000 to 400 PSI Inlet	4.6	3.8	2.9	4.6
Envelope Height	8.0	11.0	9.50	8.0
Envelope Width	4.0	3.5	3.4	4.0
Envelope Depth	4.0	3.2	6.6	4.0

Figure 5-46

SHUTTLE OMS HELIUM REGULATOR DEVELOPMENT SCHEDULE PROJECTION

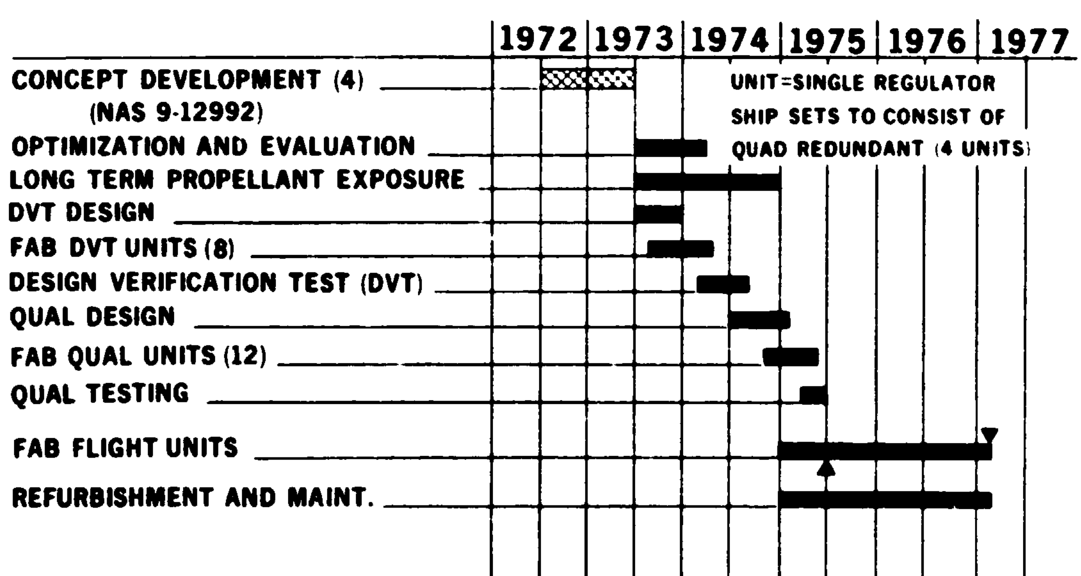


Figure 5-47

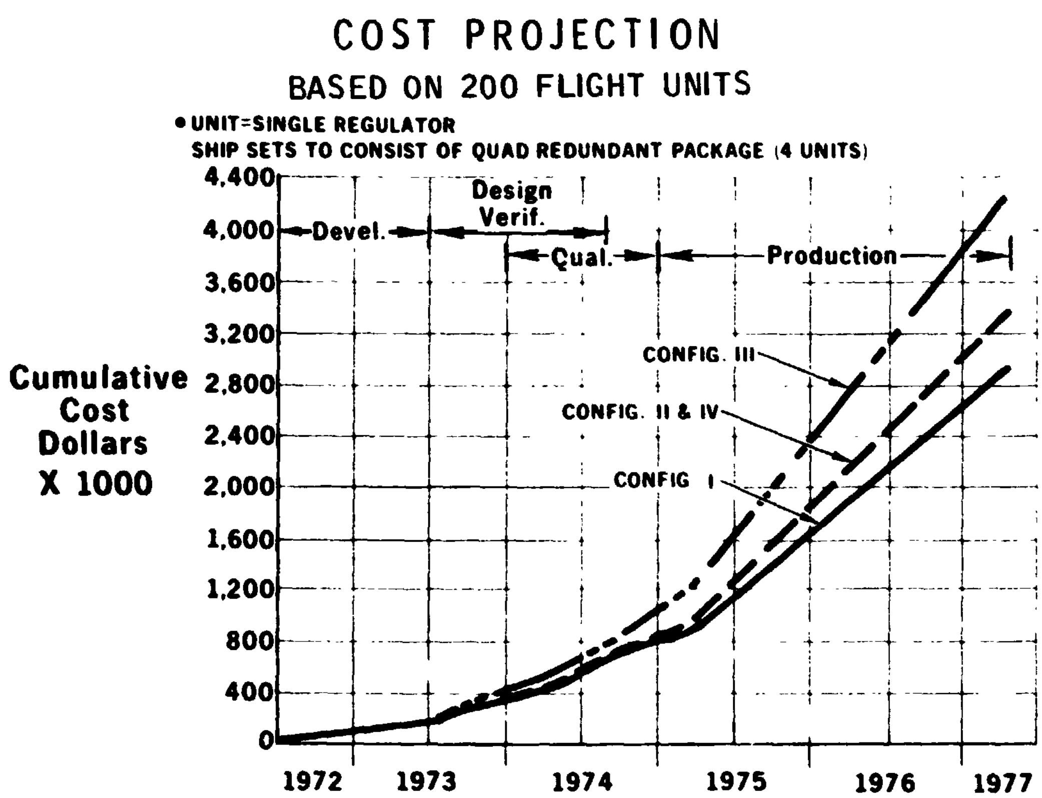


Figure 5-48

TRADE-OFF SUMMARY

	Configuration			
	I	II	III	IV
Cost and Schedule	10	9	6	9
Weight	8	10	9	8
Envelope	10	9	9	10
Stability Characteristics	10	10	7	10
Accuracy	9	9	10	9
Reliability	10	9	6	8
	57	56	47	54

All Ratings are Based on Maximum Possible 10 Points.

Figure 5-49

6.0 PROTOTYPE REGULATOR FABRICATION

6.1 FABRICATION AND QUALITY CONTROL APPROACH

The fabrication effort that was accomplished in support of this program was performed in accordance with Marquardt's experimental hardware fabrication and quality control procedures. The experimental hardware fabrication approach utilizes a liaison engineer as the key individual who decides where the parts are to be fabricated, who specifies critical fabrication techniques, and who defines fabrication and assembly sequences. In addition, the liaison engineer determines which of the detail part dimensions are critical to fit and function and obtains inspection confirmation of these dimensions.

The experimental hardware fabrication and quality control approach utilizes a logbook which contains a copy of the detail drawings of each part as well as copies of the purchase requisitions, material certifications, and all other directives relating to the manufacturer of the particular hardware. All detail part dimensions considered critical to the prototype regulator assembly are marked on these drawings and are subsequently measured by inspection. The actual measured dimension is then recorded by inspection immediately adjacent to the specified dimension on the detail parts drawing. In this manner, a record of all critical dimensions as built is maintained.

As a result of a very heavy workload in Marquardt's experimental shop and also to minimize fabrication costs, nearly all of the prototype regulator parts were subcontracted to local vendors. Major exceptions to this approach were the fabrication of the flow limiter, mechanical damper, and pneumatic damper which were made entirely in Marquardt's experimental shop. In addition, all assembly type work such as brazing, fit-ups for welding, and TIG welding were also accomplished at the Marquardt shop. All functional tests of the regulator components such as leak checks, springrate tests, bellows effective area tests, and proof pressure tests were also performed at The Marquardt Company.

6.2 HARDWARE DESCRIPTION

6.2.1 Regulator

Test hardware fabricated during this program included one prototype pressure regulator and one flow limiter. A cross section of the prototype pressure regulator is shown in Figure 6-1 and a photograph of this hardware in Figure 6-2. Some of the regulator components are shown in Figure 6-3. The prototype regulator is conceptually identical to the flight-type regulator identified as Configuration 1 in Section 5 of this report. This regulator is a single-stage regulator featuring a pressure balanced poppet, friction-free guidance by means of flexures, and pneumatic damping. Pressure balancing of the poppet is accomplished by means of a bellows which features essentially the same effective diameter as the seat and

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PROTOTYPE REGULATOR

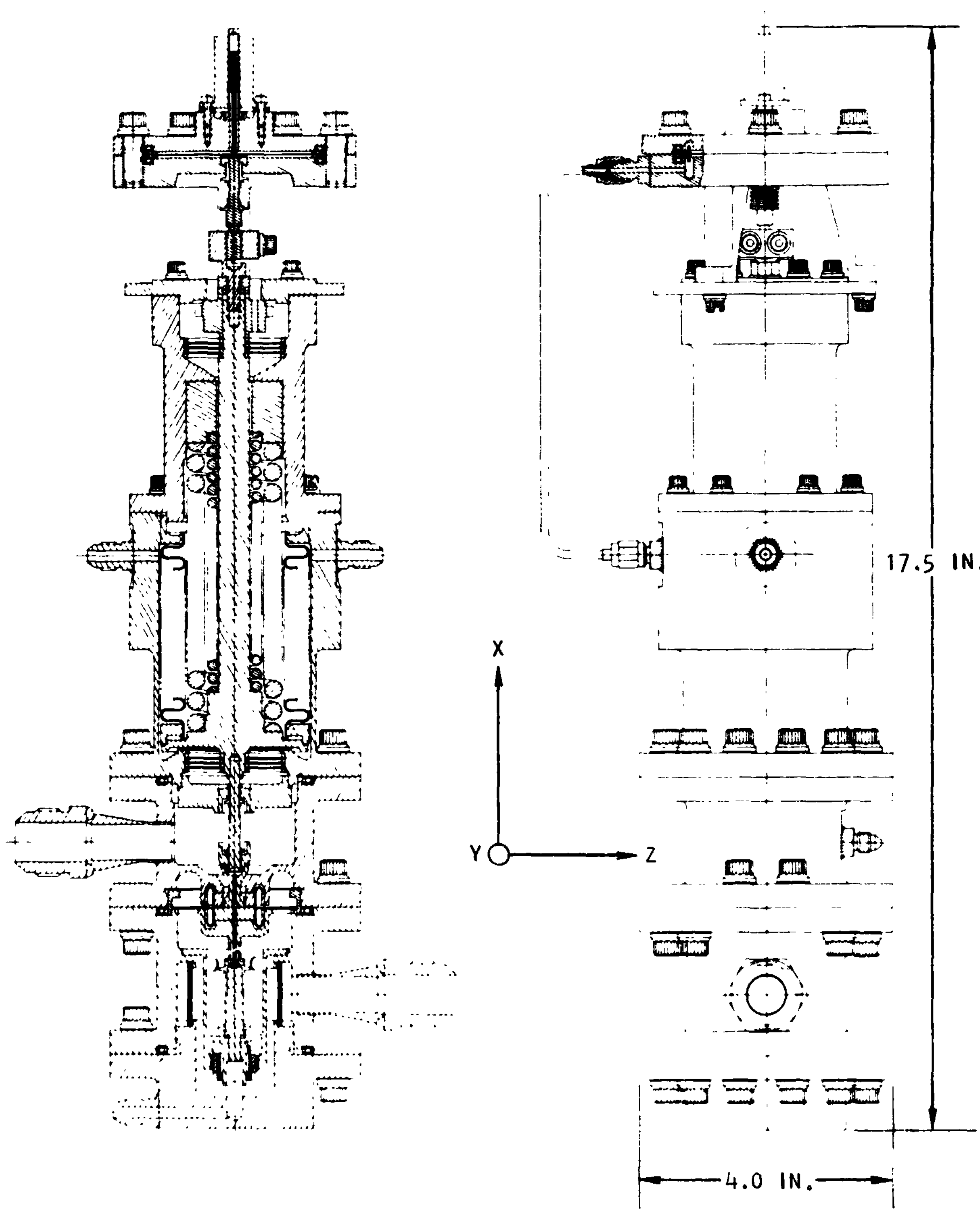


Figure 6-1

PROTOTYPE REGULATOR

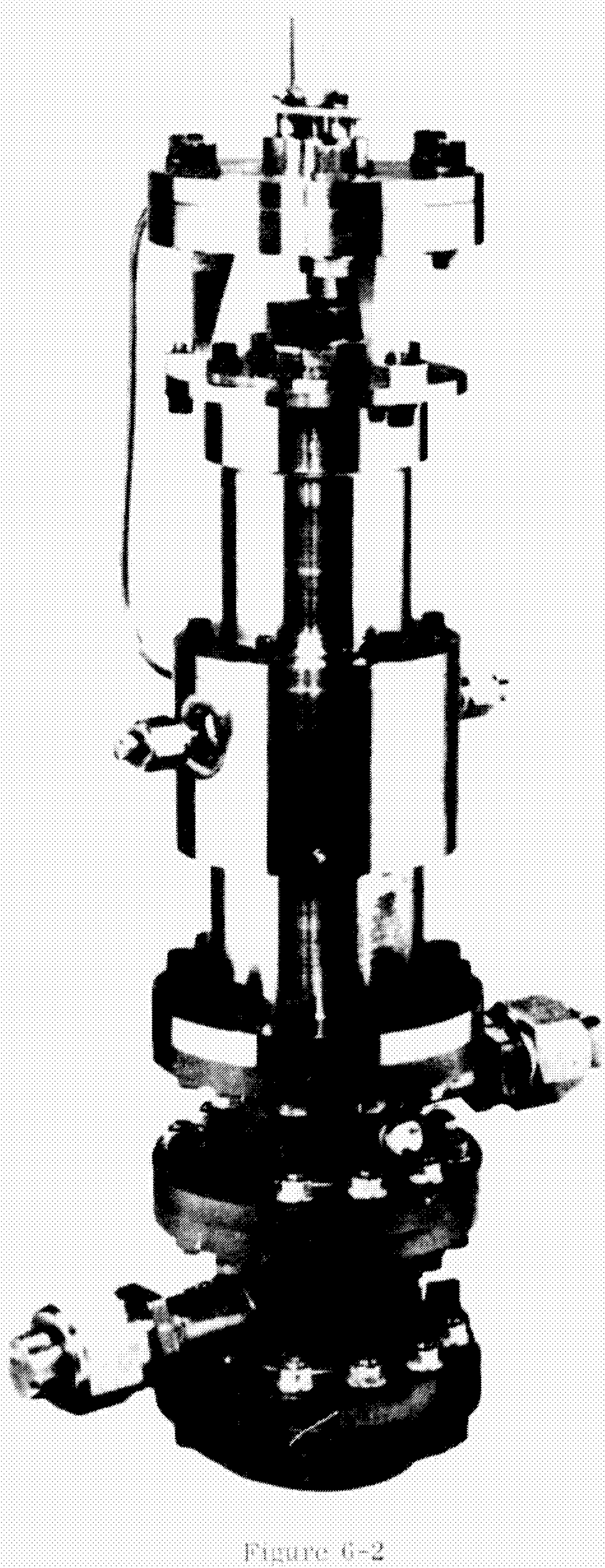
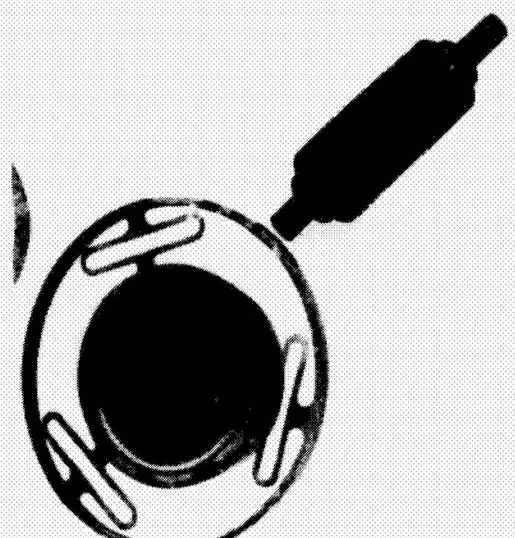


Figure 6-2

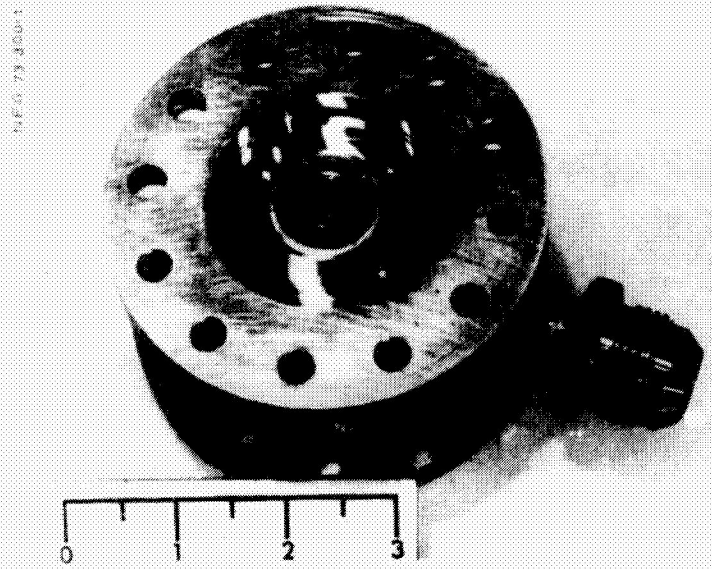
THE Marquardt COMPANY VAN NUYS, CALIFORNIA

A74-4-612-79

REGULATOR POPPET
WITH FLEXURE AND
PRESSURE BALANCING BELLOWS



REGULATOR SEAT



REGULATOR ACTUATOR
WITH SPARE ACTUATOR BELLOWS, SHAFT SEAL BELLOWS AND FLEXURE

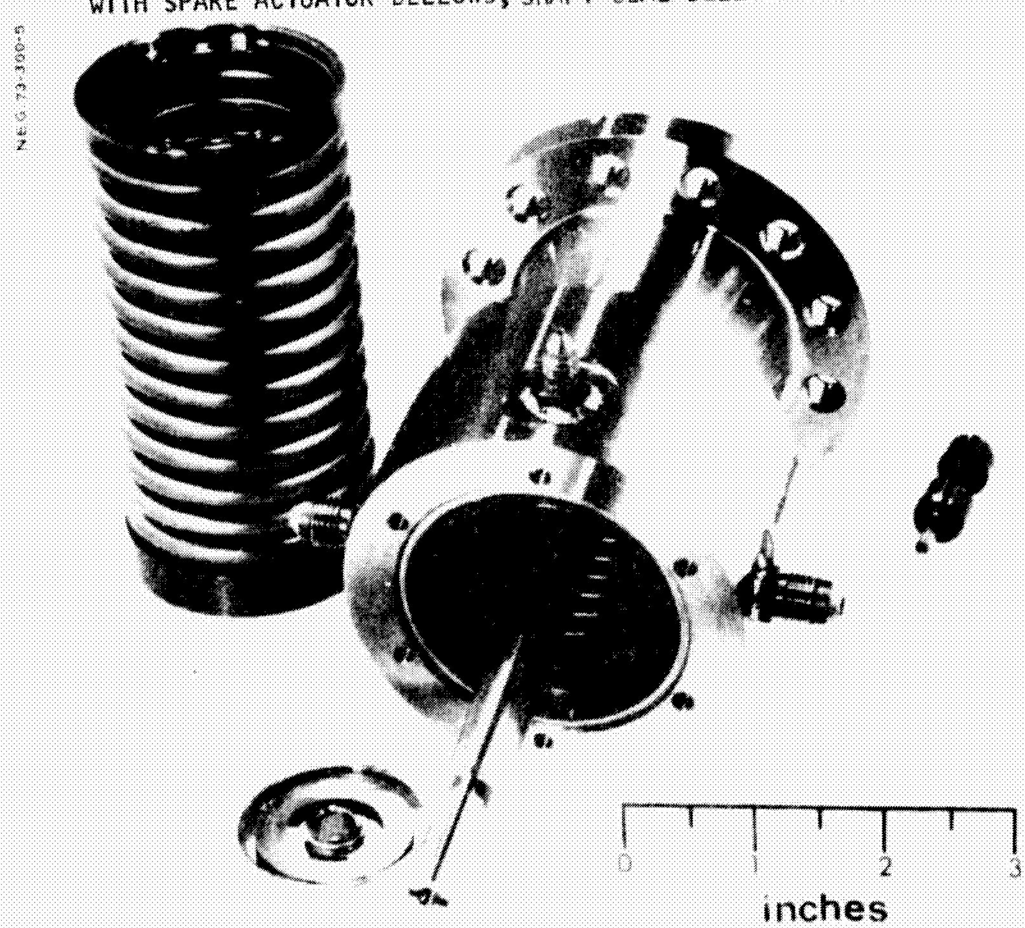


Figure 6-3

thereby eliminates pressure unbalance forces. The poppet is actuated by means of a push rod extending from the actuator into the regulator outlet cavity. A bellows seal at this push rod isolates the outlet cavity from the actuator sensing cavity. The actuator piston seal also consists of a bellows and the piston reference force is provided by means of two concentric coil springs. The pneumatic damper is located external to the main actuator body for accessibility and consists of a diaphragm with a control orifice across it. The pneumatic supply for the damper is obtained from the actuator sensing cavity through an external tube, and this tube includes an orifice which is an order of magnitude smaller than the pneumatic damping orifice in the diaphragm to effectively isolate the helium in the damper from that in the actuator. Sensing pressure to the actuator is supplied through a one-quarter inch diameter tube from the regulator outlet tubing.

To gain the maximum possible accessibility to the components of the prototype regulator, the regulator was designed and fabricated with several flanged joints. These joints permitted accessibility to the poppet/seat interface, the poppet bellows, the push rod bellows, the coil springs, the actuator stops, and the pneumatic damper. These various flanged joints are sealed by means of teflon-jacketed seals. Except for these seals, the prototype regulator was of an all-metal construction. To permit the monitoring of regulator position during the test program, the prototype regulator was also equipped with an LVDT transducer which was located on top of the pneumatic damper. Utilization of this position transducer during the pressure regulator development program is believed to be a unique approach by The Marquardt Company. The prototype pressure regulator also featured extra ports at the inlet cavity, outlet cavity, and actuator sensing cavity for instrumentation purposes.

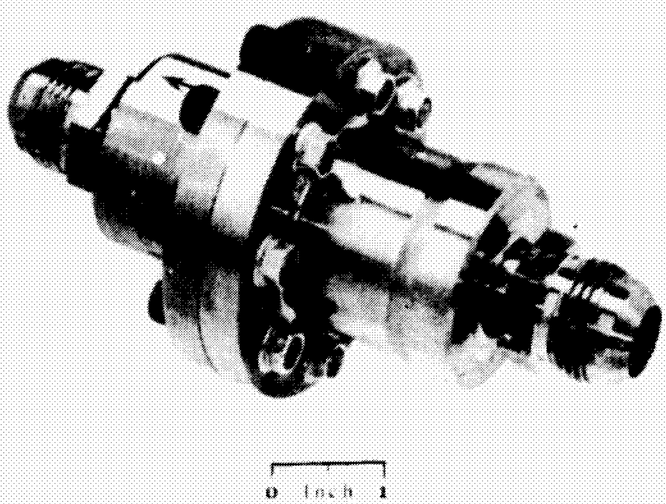
The prototype regulator was made from Inconel 718 except that the poppet and seat were tungsten carbide*; the reference springs were Ni-span C, the pneumatic damper bellows was 300 series stainless steel, and the LVDT position transducer employed various other materials of construction. Joining of various regulator components was accomplished primarily by means of electron beam welding except for the assembly of the flexures and the assembly of the poppet and seat to their support structures were accomplished by brazing.

6.2.2 Flow Limiter

A cross section and a photograph of the flow limiter is shown in Figure 6-4. The flow limiter is a relatively simple device consisting of a venturi nozzle with a variable position pintle such that the area through the venturi is varied as a function of helium flow rate. Movement of the pintle to accomplish this area variation is obtained as a result of the difference in the flow forces acting on the pintle and the reference forces provided by the coil spring. The flow limiter is again a friction-free device utilizing flexure plates for guidance of the pintle. The prototype flow limiter also included a flange at the center to permit access to the pintle and venturi as well as to allow stroke adjustments. Materials

*KZ801, tungsten carbide with 6% nickel binder

PROTOTYPE FLOW LIMITER



NEG. 73-300-7

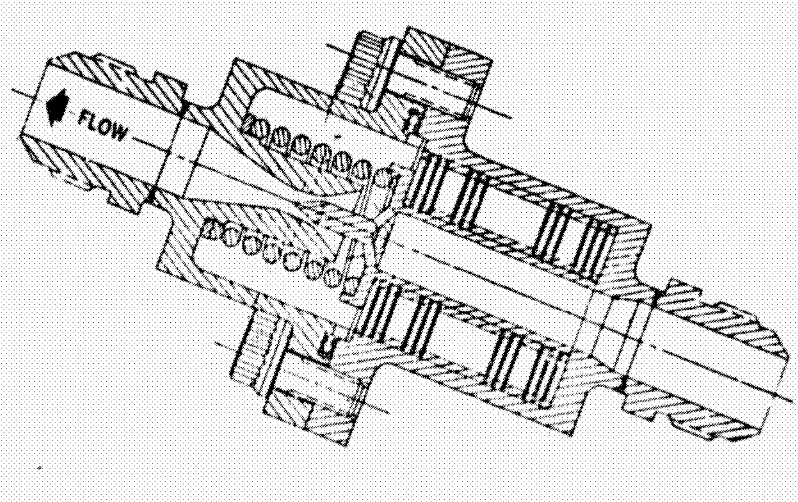


Figure 6-4

of construction were 300 series of stainless steel for all parts except the flexure reference coil spring. These were made from Inconel 718 and Type 302 stainless steel, respectively. The flexure assembly was a brazed assembly.

6.3 FABRICATION PROBLEMS

6.3.1 Actuator Bellows

During the fabrication of the prototype pressure regulator, several problems arose which significantly delayed the completion of the regulator. The first major problem developed when the actuator bellows was received from the supplier. This bellows was rushed to Marquardt by the supplier the day before the company went on strike and without having been subjected to inspection by the supplier. Receiving Inspection tests at The Marquardt Company disclosed that the bellows was both undersize in dimension (the inner diameter was smaller than specified) and that is was not capable of supporting the loads specified on the design drawing. Since the vendor was then on strike and it appeared that the strike would not be settled soon, Marquardt decided to redesign the coil springs that fit inside the bellows to make them compatible with the smaller diameter and to permit the use of the coil springs at a higher preload so as to compensate for the lesser preload of the bellows. New springs were subsequently procured from the spring supplier, causing a substantial program delay.

6.3.2 Poppet Bellows

Another problem arose when it was determined that the poppet bellows supplied by another bellows vendor failed to meet proof pressure requirements. To correct this problem, the vendor increased the material thickness from 0.008 inch to 0.011 inch and remade the bellows. Subsequent pressure tests were acceptable.

6.3.3 Poppet Brazing

Boron carbide was the selected poppet and seat material until it was determined that the braze material would not wet the surface. The boron carbide was even gold plated to act as a wetting agent without success. Tungsten carbide was then selected for the poppet-seat fabrication.

6.3.4 Seat Support

Initial acceptance tests of the regulator disclosed gross leakage at 3000 psi inlet pressure. This problem was traced to uneven deflection of the Inconel 718 support structure as described in Section 7.1.3. This problem was corrected by deleting the electron beam welded subsupport structure and brazing the seat directly into the main support structure.

6.4 REGULATOR MODIFICATIONS

As a result of the data obtained during the design verification test program, several regulator modifications/repairs were also made. One modification was the replacement of the seat as discussed in the preceding section. A second modification consisted of an attempt to increase the damping provided by the pneumatic damper. This was attempted by reducing the volume on each side of the diaphragm in the pneumatic damper by means of filling the existing volume partially with epoxy and by changing the orifice across the diaphragm. However, this modification still did not provide sufficient damping to satisfy the extremely stringent criteria setup for this particular regulator program. To permit the experimental determination of just how much damping was required to limit actuator movements to 0.002 inch during vibration, a mechanical damper featuring adjustable friction damping was designed and fabricated. A photograph of this damper installed in place of the pneumatic damper is shown in Figure 6-5. The damper consisted essentially of two friction pads which were placed against an extension of the actuator shaft. The force applied by these friction pads was variable by means of a screw adjustment. Friction pads made from polyimide, teflon, and brass were evaluated during mechanical damper component tests.

As a result of an assembly error the regulator actuator caused the diaphragm support to strike the damper housing during the closing motion rather than to limit this motion by the stop in the actuator and this resulted in the failure of the shaft connecting the damper diaphragm to the actuator shaft. Subsequent vibration testing with the completely undamped regulator resulted in structural damage to the actuator flexure assemblies and the push rod shaft seal (bellows). This damage was subsequently repaired and the design verification test program was continued successfully.

MECHANICAL DAMPER

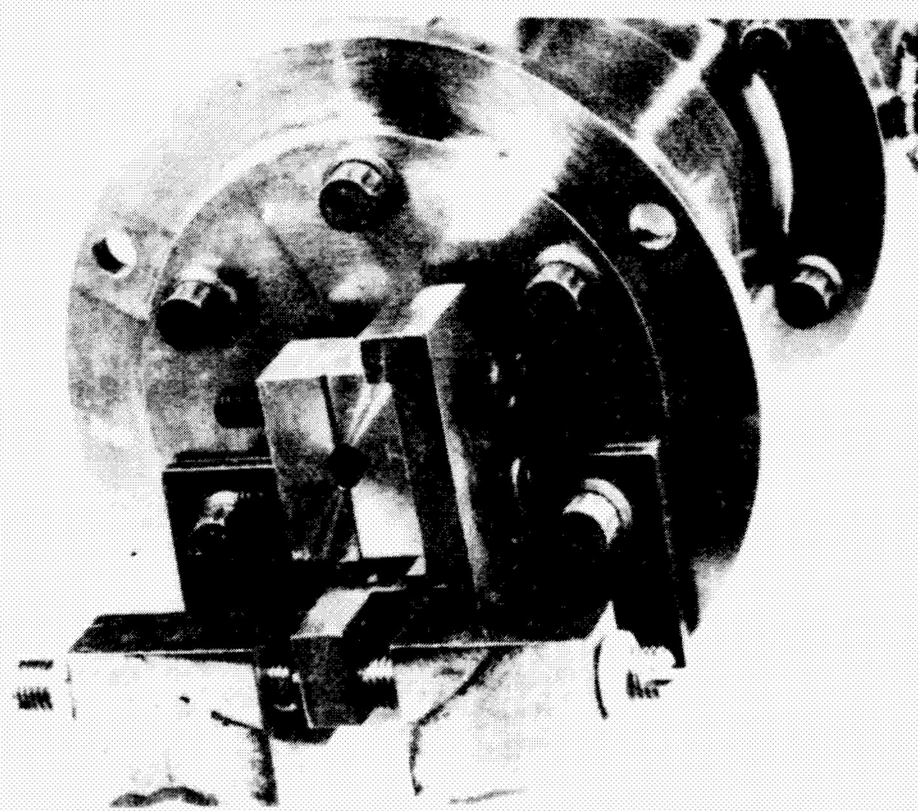


Figure 6-5

7.0 DESIGN VERIFICATION TEST

The regulator described in Section 6.0 was subjected to the verification test program of Marquardt Test Plan (MTP) 0213* to document the performance characteristics and verify compliance with the requirements of Section 3.1.4, excluding extended service life and long duration compatibility. The verification test program included component testing, regulator performance testing, and the preparation of a test plan for a NASA-JSC test program that includes extended service life and long duration compatibility testing. This report also includes additional vibration and life cycle testing conducted on this same regulator as defined in a follow-on statement of work.

A summary of each test, test conditions and results is presented in Table 7-I.

7.1 COMPONENTS TESTS

Tests were conducted on each of the components or design elements of the regulator to verify the design and/or material properties specified by the design. Mechanical tests were performed on the flexures, springs, bellows, dampers, and seat-poppet interface. Flow tests on the regulator and flow limiter to define component performance were also conducted. Flow force and flow limiter tests were conducted at the Ogden Technology Laboratories, Inc. facilities during August and September of 1973.

7.1.1 Flexures, Springs, and Bellows Spring Rates

The spring rates of the flexures, springs and bellows were determined in accordance with the sections A-1 through A-7 of Appendix A to MTP 0213. The spring rate is ratio of force required to compress or extend an elastic member, $k = F/X$ (lbs/inch). The data for each component tested is presented in Table 7-II. The average or linearized data, corrected for any preload condition on the part number selected for fabrication into the regulator is presented in Figure 7-1. The combination of bellows and diaphragm used in the pneumatic damper subassembly is not linear, as the diaphragm is extended outward. However, the combination is linear at $\pm$ 0.010 inch deflection as shown.

Each component was cycled through a load range representing its installed and operational forces and displacement. Spring rates were then determined over the operational range. The data tabulation indicates some significant variations between samples. The correlation of actual component values and the design points are summarized as follows:

*Included as Appendix B to this report.

THE Marquardt COMPANY VAN NUYS, CALIFORNIA

TABLE 7-I

DESIGN VERIFICATION TEST SUMMARY

Type of Test	MTP 0213 Paragraph	No. of Tests	Test Dates	Test Conditions: Pressure (psi)	Test Conditions: Temperature	Test Conditions: Flow (lb/min)	Results
Component							
Springrate	A-1 to A-7	17	June/Jul 73	Ambient	Ambient	N.A.	
Effective Area	A-8 to A-10	3	Jul/Aug 73	As required	Ambient	N.A.	
Flow Force / Flow Coefficient	B-1	22	8/22/73	305 to 3300	Ambient	0.4 to 3.3	
Pneu. Damper	(1)	25	Dec 73	0 to 300	Ambient	N.A.	
Mech. Damper	(1)	11	Dec 73	N.A.	Ambient	N.A.	
Flow Limiter	B-2	3	9/17/73	400 to 1800	100°F	2.5 to 13.8	
Cutting	(2)	34	Mar/Oct 73	N.A.	Ambient	N.A.	
Performance							
70°F	B-4.4	15	8/31/73	350 to 3950	Ambient	0.46 to 3.7	
70°F	B-4.4	4	9/17/73	380 to 3850	Ambient	2.5	
150°F	B-5	14	9/20/73	380 to 3960	130 to 161°F	0.38 to 3.1	
-150°F	B-6	11	9/29/73	380 to 1988	-125 to -169°F	0.49 to 6.6	Facility Failure
Slam Start	B-4.3	4	10/12/73	400 to 3800	Ambient	N.A.	0.040 orifice @ 3800
Vibration	B-7	2	10/15/73	4000	Ambient	15.6 cfm	Oms, X & Y
Vibration	B-8	2	10/16/73	4000	Ambient	15.6 cfm	Main engine X & Y
Vibration	B-8	6	10/20/73	400 to 3960	Ambient	15.6 cfm	Main engine X
Vibration	Sine sweep	2	10/20/73	3950	Ambient	15.6 cfm	160 Hz resonance
Vibration	B-9.1	1	10/20/73	400	Ambient	0	Liftoff X
Check Valve	(3)	5	11/20/73	460-3900	Ambient	15.6 cfm	2 w/o C.V.
Vibration		8	11/20/73	3400-3700	Ambient	15.6 cfm	1 w/o vib, 5 grms
Vibration		4	12/18/73	3900	Ambient	15.6 cfm	15.3 grms
Life Cycle	B-12	15,000	Feb 73	300-400	Ambient	10 cfm GN_2	Low leakage

Notes: (1) Damper tests not defined in MTP 0213.
(2) Static contamination tests in lieu of dynamic flow tests of B-10.

TABLE 7-II

SPRING RATE TEST

COMPONENT - ACTUATOR BELLOWS

S/N 3*		S/N 4	
F	X	F	X
17	.10	18	.10
34	.20	35	.20
50	.30	52	.30
66	.40	68	.40
81	.50	82	.50
95	.60	97	.60
110	.70	109	.70
122	.80	123	.80
137	.90	137	.90
150	1.00	150	1.00
k =	130 ppi	--	130 ppi

COMPONENT - PUSH ROD BELLOWS

S/N	10*	11	12	13	14	15
F	X	X	X	X	X	X
0	O	O	O	O	O	O
1	.0075	.0075	.0070	.0082	.0067	.0085
2	.0152	.0150	.0149	.0164	.0129	.0166
3	.0234	.0222	.0224	.0240	.0199	.0241
4	.0312	.0287	.0298	.0313	.0253	.0312
K =	127 ppi	137 ppi	134 ppi	126 ppi	156 ppi	127 ppi

*Serial Number selected for assembly.

TABLE 7-II (Continued)

COMPONENT - POPPET BELLOWS

S/N 1 (6/6)* F	S/N 1 (6/6)* X
O	O
5	.007
10	.0128
15	.0192
20	.0258
25	.0323
30	.0387
35	.0450
40	.0515
45	.0578
50	.0645
k =	775 ppi

DAMPER BELLOWS

F	S/N 1 X	S/N 2* X
O	O	O
.25	.0060	.0060
.50	.0121	.0126
.75	.0182	.0182
1.00	.0245	.0245
1.25	.0307	.0307
1.50	.0373	.0370
k =	40.4 ppi	40.6 ppi

COMPONENT - PUSH ROD FLEXURE (ACTUATOR)

S/N 2 (7/12)* F	S/N 2 (7/12)* X	S/N 3 (7/12)* F	S/N 3 (7/12)* X
.25	0	.25	0
.35	.0087	.35	.007
.45	.0162	.45	.0137
.55	.0210	.55	.0200
.65	.0252	.65	.0275
		.75	.0339
k =	15.8 ppi		14.5 ppi
	(Lower)		(Upper)

*Serial Number selected for assembly.
(Test Date)

TABLE 7-II (Continued)

COMPONENT - POPPET FLEXURE

S/N-(6/25) F	X	S/N 1(7/12)* F	X
.2	0	.6	0
.3	.009	.8	.0048
.4	.0187	1.0	.0095
.5	.0225	1.2	.0135
.6	.0285	1.4	.0178
.7	.0335	1.6	.0223
k =	149 ppi		45 ppi

COMPONENT - INNER SPRING / OUTER SPRING

Inner Spring S/N 2* F	X	Outer Spring S/N 1* F	X
4	.022	29	.105
17	.100	57	.195
36	.202	87	.295
55	.310	118	.402
76	.425	151	.512
93	.525	211	.720
110	.622	237	.802
125	.720	267	.902
141	.820	300	1.012
150	.890	332	1.115
154	.915	357	1.203
167	1.000	387	1.299
178	1.118	421	1.411
185	1.178	449	1.500
190	1.242	474	1.590
		479	1.504
k =	148 ppi	k =	299 ppi

*Serial Number selected for assembly.
(Test Date)

SPRINGS, FLEXURES AND BELLOWS SPRING RATES

DATA CORRECTED FOR ANY PRELOAD

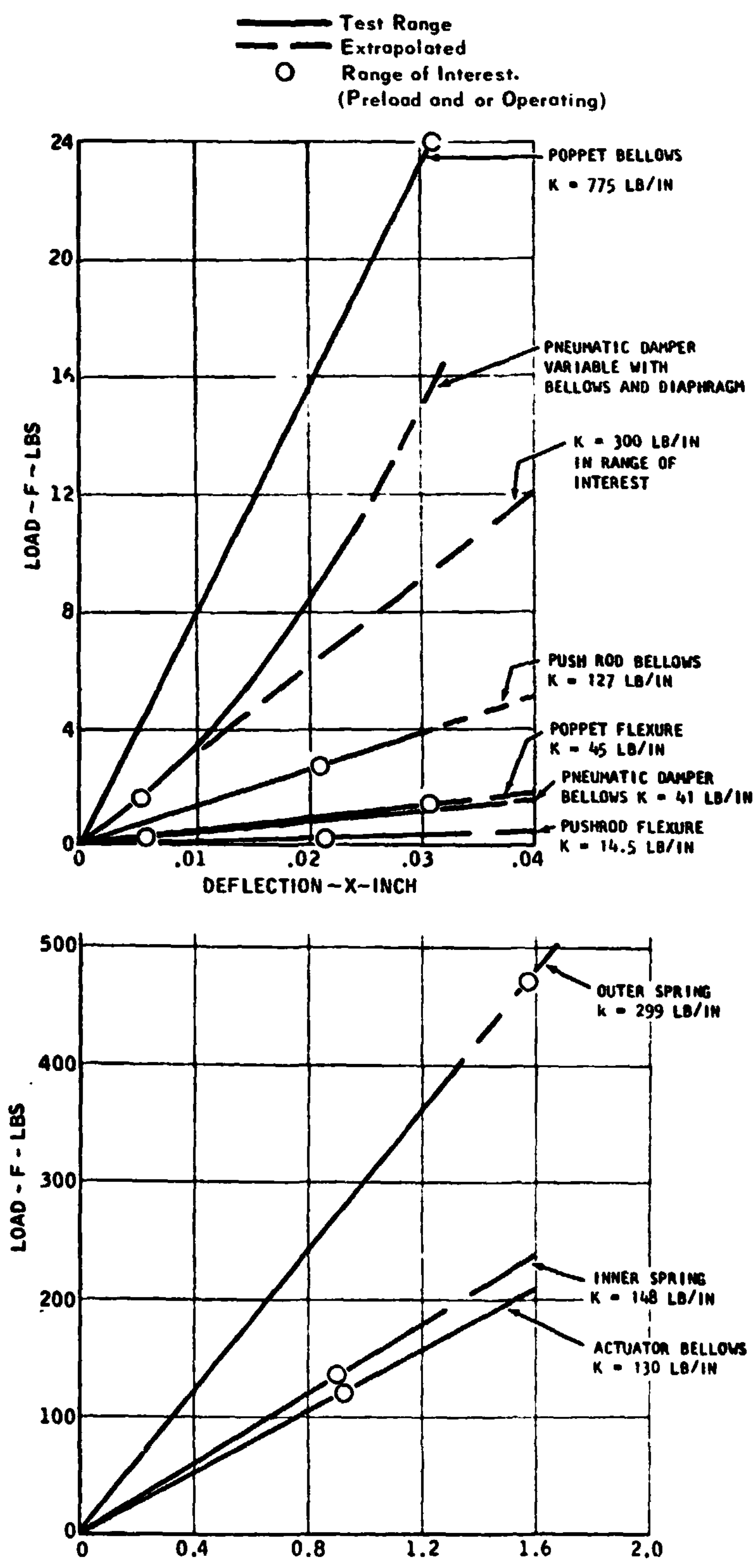


Figure 7-1

Item	Springrate (ppi) Design (±10%)	Actual
Push rod flexure	72**	30.3
Poppet flexure	75**	45
Inner spring	*	148
Outer spring	321	299
Actuator bellows	197	130
Push rod bellows	163	127
Poppet bellows	200**	775
Damper bellows	-	40.6

* The availability of spring wire and matching the spring diameter to the space determined by the actuator bellows resulted in the use of two springs, one nested inside the other. The springrate was increased to the actual values after other component design values were changed or were defined by the available manufacturer.

** The initial poppet bellows design could not meet the proof pressure requirements. The increased spring rate of the poppet bellows affected the allowable redistribution of springrates of the other components.

7.1.2 Bellows Effective Area

The effective area tests were conducted in accordance with Sections A-8, A-9 and A-10 of Appendix A to MTP 0213 with Section A-8 modified to measure force instead of deflection. The effective area is determined with the aid of a fixture which allows the bellows to deflect axially under an external pressure load. A fixture for each bellows was designed and fabricated to provide effective area measurements at the preload (operating) position and to check leakage at the proof pressure. These fixtures and a typical cross section are shown in Figure 7-2.

The relationship between the deflection and the applied pressure to calculate the effective area uses the springrate of the bellows as follows:

(1) $P = F/A$

(2) $K = F/X$

Solving for A gives

(3) $A = \frac{KX}{P}$ or $A = k/p/x$

where p/x is the slope of the pressure-deflection curve.

EFFECTIVE AREA TEST FIXTURES

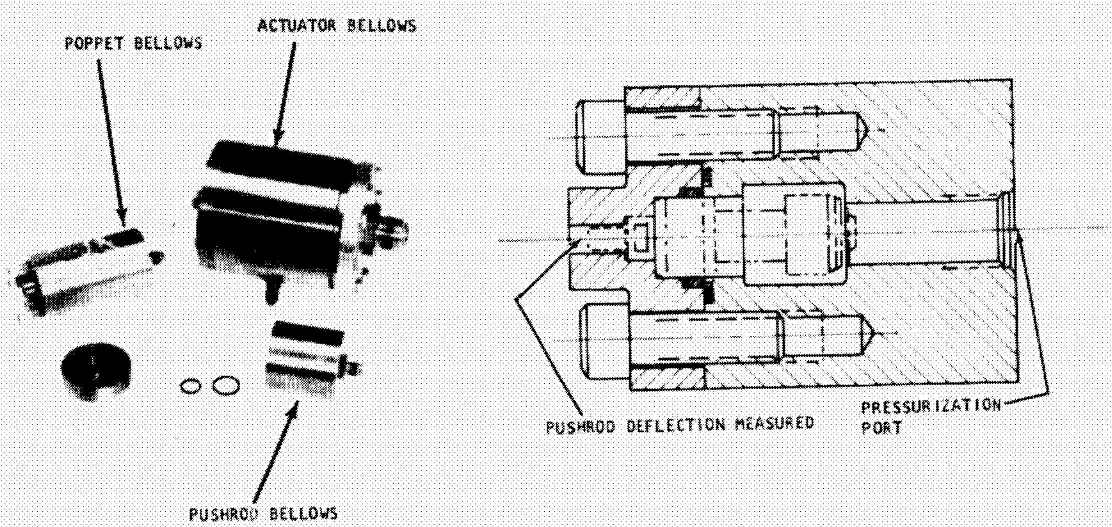


Figure 7-2

BELLOWS EFFECTIVE AREA

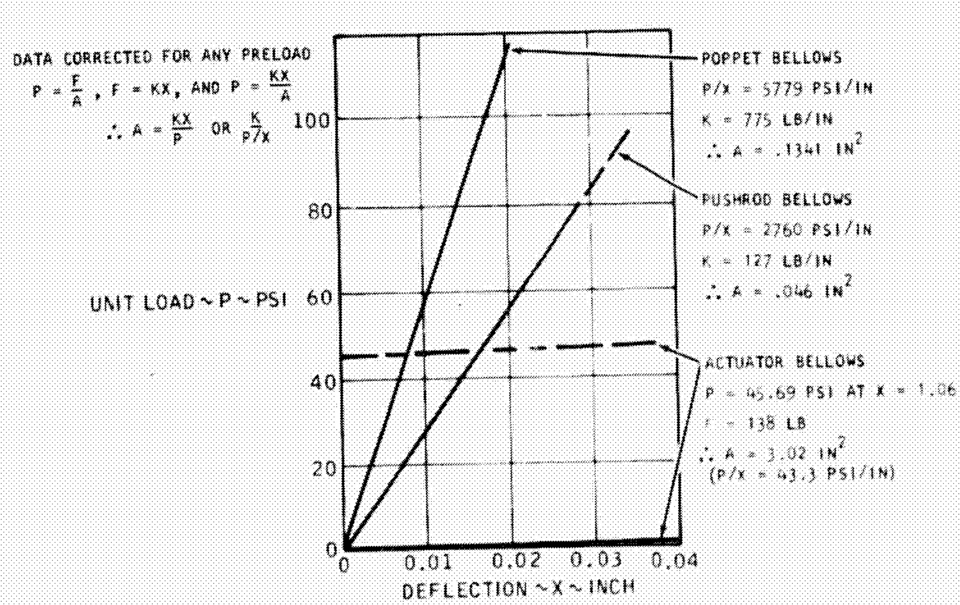


Figure 7-3

The effective area of the poppet and push rod bellows was calculated by measuring the slope of these curves and the springrates determined in 7.1.1. When the effective area is very large, such as in the actuator bellows, the pressure variation over the operating range is extremely small. Therefore, an alternate method was used consisting of measuring the force required to place the bellows in the preload condition and then measuring the pressure required to initiate deflection. In this case,

$$(4) \qquad A = \frac{F}{P}$$

The bellows effective area test data is presented in Table 7-III and plotted in Figure 7-3.

7.1.3 Flow Forces

The flow force test was conducted in accordance with Section B-1 of Appendix B to MTP 0213. The flow forces on the poppet were measured at various inlet pressures and flow rates using the fixture shown in Figure 7-4, installed in the test setup of Figure 7-5.

The fixture converted the regulator into a valve where the poppet stroke could be set to a fixed value, with the micrometer head. A load cell is added between the micrometer shaft and the regulator push rod, which makes contact with the poppet. A shutoff valve is connected to the inlet and is used to start and stop flow. The flow force is the difference between the load cell reading when flowing and at no-flow, minus the downstream pressure force acting on the push rod bellows. The no-flow load cell reading is made when the downstream pressure is zero gage.

The poppet stroke is corrected for the load cell and seat deflection. The load cell deflection versus load relationship is 0.0007-inch/100 lbs. The seat deflection was due to the pressure drop across the seat housing as shown in Figure 7-6. A comparison of the flow forces between the test model and the POP computer program model is shown in Figure 7-7. The test model results are based upon the data shown in Table 7-IV corrected for constant inlet and outlet pressures. Tests were not performed at inlet pressures greater than 3300 psig, due to facility limitations.

7.1.4 Flow Coefficient

The flow coefficient tests were conducted concurrently with the flow force test of 7.1.3.

TABLE 7-III

BELLOWS EFFECTIVE AREA TEST

Component:	Actuator		Push Rod		Poppet	
	S/N 1		S/N 10		S/N 1	
	P	F	P	X	P	X
	45.69	138	6.5	0.008	307.4	0.00080
	A =	3.02	15.2	0.011	311.3	0.0012
			23.9	0.015	315.7	0.0021
			30.1	0.017	322.2	0.0032
			36.8	0.019	327.8	0.0042
			51.9	0.024	333.6	0.0052
			60.0	0.026	340.5	0.0064
			45.1	0.020	344.6	0.0072
			30.2	0.015	A =	0.1341 in.^{2}
			A =	0.046 in.^{2}		

FLOW FORCE MEASUREMENT FIXTURE

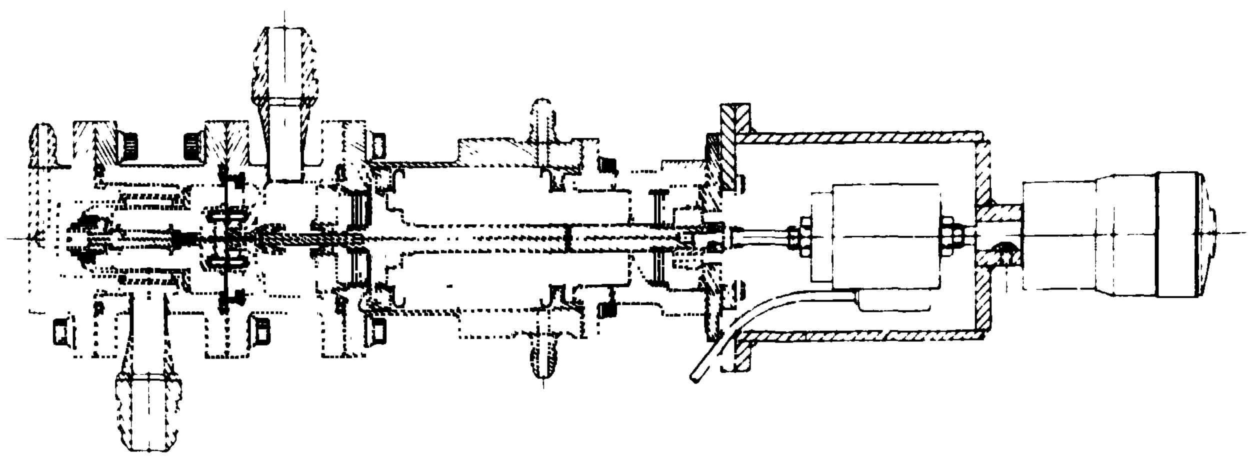


Figure 7-4

REGULATOR ΔP AND FLOW FORCE SIMULATOR TEST SETUP

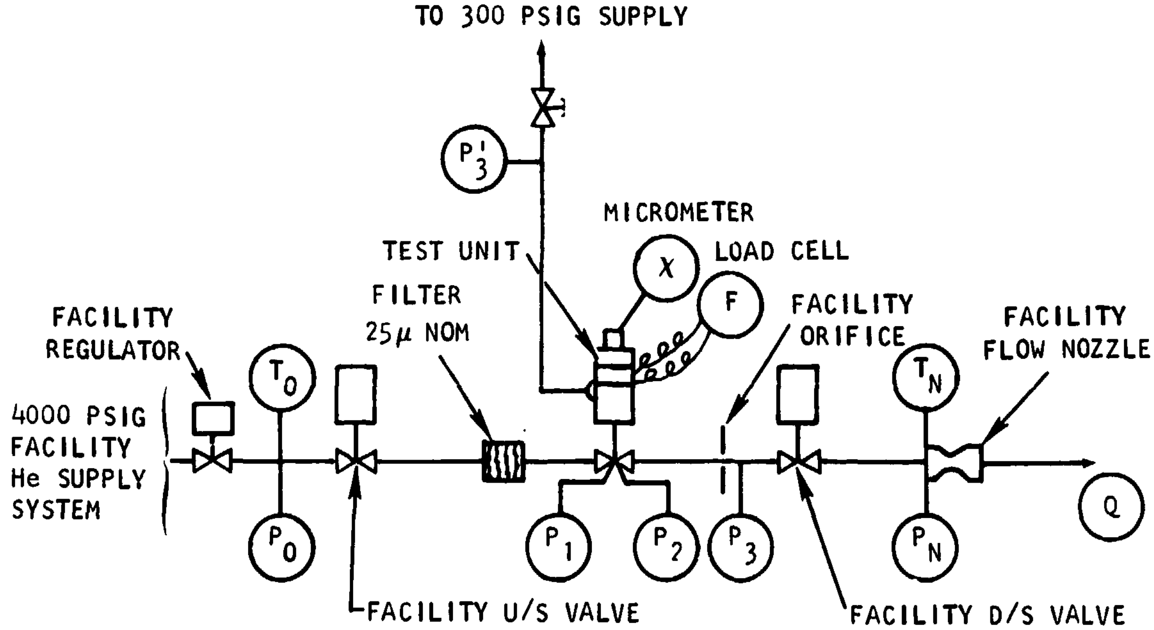


Figure 7-5

EFFECT OF PRESSURE LOADS ON SEAT SUPPORT STRUCTURE RESULTING IN SEAT DEFLECTION

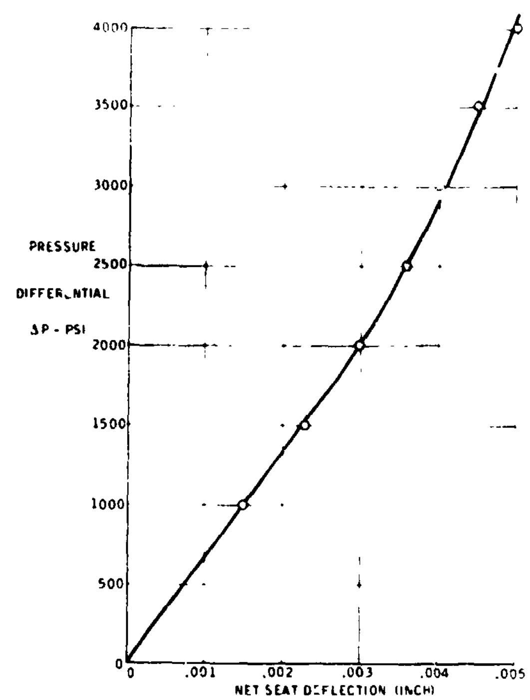


Figure 7-6

FLOW FORCE CORRELATION BETWEEN TEST DATA AND THEORETICAL MODEL

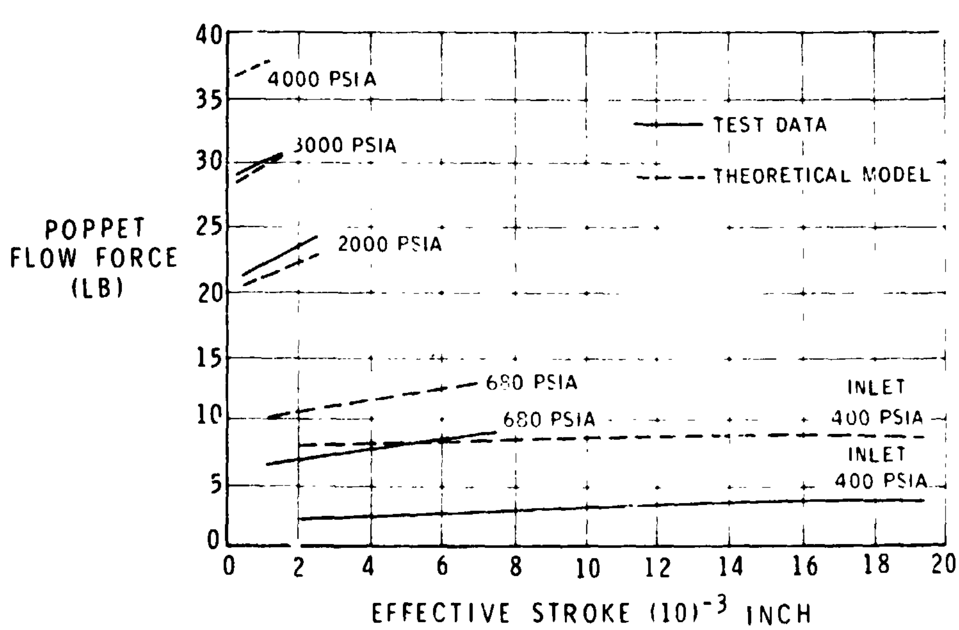


Figure 7-7

TEST DATA SHEET

MAC A 948

TABLE 7-IV

SHEET 1 OF 1

TEST OPERATOR	TEST ENGINEER	WITNESS	CLASSIFICATION
DICK KUNST	JOE HERNANDZ	G. POND	

SUBJECT FLOW FORCE TESTS MODEL X29267 SER. NO. 1

SHOP TRAVELER NO.

TEST NO. B-1 PER MTP0213

Tatm(DRY) °F

BAR. 29.89" AT 70 °F

FUEL SPEC.

FUEL SP. GR. AT °F

TEST DATE 8/22/73

$S_c = S + X_s - X_L$

$F_c = F - F_0 - A_2 P_2$

A_2 = PUSHROD BE[illegible] EFF AREA (.046 IN^2)

P_2 = D/S PRESS (PSIA)

RUN NO.	T_0 (°F) U/S ~~PT. NO.~~	T_N (°F) D/S ~~TIME~~	P_1 PSIG	P_2 PSIG	P_3 PSIG	Q_3 CFM	$\dot{m}$ LB/MIN	S IN (POPPET)	X_s IN (SEAT)	X_L IN (LOAD CELL)	S_c IN	F_0 LB	F LB	F_c LB	C_D
1A	-	105	438	437?	250	16.3	2.64	.0197	0	.0005	.0192	40.0	62.2	1.4	8.94?
2A	95	108	600	394	250	13.7	2.21	.0074	.00045	.0009	.00745	26	49.5	4.2	.726
3A	96	108	690	430	250	13.8	2.50	.0074	.00055	.00042	.00753	26.5	52.5	1.0	.695
4A	96	110	2030	261	108	22.3	1.64	-.0003	.0033	.0004	.0026	5.0	49.5	18.4*	.44
5A	-	113	1440	455	250	16.7	2.65	.0027	.0020	.00047	.00923	22.7	59.0	14.7	.615
6A	96	113	1980	410	250	14.6	2.33	.0007	.0030	.00048	.00322	17.2	60.0	22.8	.516
1B	82	88	370	315	235	17.3	2.71	.0167	.0001	.00047	.01633	38.5	57.0	3.3	.88
2B	86	88	370?	315	235	16.6	2.60	.0167	.0001	.00047	.01633	39.5	57.5	2.8	.849
3B	87	88	370	317	235	16.6	2.60	.0167	.0001	.00047	.01633	40.0	58.0	2.7	.869
4B	89	90	650	319	237	16.6	2.62	.0074	.0007	.00042	.00768	29.2	52.5	7.9	.725
5B	79	100	2000	315	237	16.7	2.57	.0007	.00315	.00047	.00338	18.5	59.0	25.3	.528
6B	80	100	3300	307	229	16.9	2.49	-.0014	.00475	.00053	.00282	7.0	66.5	32.4*	.373
7B	-	113	3290	397	294	18.0	3.34	-.0006	.00465	.0006	.00345	7.2	75.0	36.4*	.41
8B	79	10[illegible]	1960	310	93	50.7	3.33	.0015	.0031	.00048	.00912	22.5	60.5	23.0	.572
9B	83	104	710	420	300	18.4	3.55	.0097	.0006	.00048	.00982	32.5	60.6	8.1	.71
10B	-	105	650	369	243	21.3	3.39	.0097	.0006	.00046	.00984	32.0	57.7	8.0	.725
11B	95	105	400	330	167	30.2	3.34	.0187	.00015	.00049	.01836	42.0	61.3	3.4	.942
12B	84	102	305	203	200	3.11	.411	.003	.0002	.00028	.00292	2[illegible].5	34.5	1.0	.68
13B	84	102	420	235	232	2.75	.416	.002	.0004	.00029	.00211	22.5	36.0	2.0	.64
14B	-	102	680	212	209	3.04	.417	.001	.001	.00029	.00171	19.5	36.5	6.5	.493
15B	-	102	1520	259	259	2.50	.421	-.001	.00245	.00032	.00113	6.0	40.5	7.8*	.342
16B	-	105	2360	295	294	2.47	.448	-.0022	.0037	.00035	.00115	6.3	44.5	10.9*	.231

* FLOW FORCE CORRECTED FOR NEGATIVE STROKE AT PRE TEST CONDITION

NOTE: ACTUATOR BELLOWS BALANCING PRESSURE FOR EACH RUN = 51.6 PSIG

ORIGINAL PAGE IS OF POOR QUALITY

The flow coefficient for the poppet/seat interface was determined from the flow data in Table 7-IV and the results are shown in Figure 7-8. The flow coefficient C_D is normally calculated from the equation:

$$C_D = \frac{\dot{W}\sqrt{T_1}}{P_1 A C_M}, \text{ where}$$

$$C_M = \sqrt{\frac{\gamma g}{R}}\, M \left(1 + \frac{\gamma - 1}{2} M^2\right)^{-\frac{\gamma + 1}{2(\gamma - 1)}}, \quad \frac{\text{lbm} - {}^\circ\text{R}^{1/2}}{\text{lbf} - \text{sec}}$$

γ = 1.67

g = $32.174 \dfrac{\text{ft-lbm}}{\text{lbf - sec}^2}$

R = $386 \dfrac{\text{ft-lbf}}{\text{lbm-}^\circ\text{R}}$

M = Mach number, a function of static to stagnation pressure ratio across the poppet/seat interface

T_1 = Inlet Temperature, °R

$\dot{W}$ = Mass flow rate, $\dfrac{\text{lbm}}{\text{sec}}$

P_1 = Inlet pressure, psia

A = $\pi D S_c$ minimum poppet/seat flow area, in.^{2}

D = Seat ID = 0.4228 in.

S_c = Poppet stroke, in.

However, P_2 as measured in the regulator is not located at the minimum flow area, resulting in a 10-20 percent error in computing Mach number and the helium mass flow function (refer to discussion in Section 4.2.3). Also, P_1 is assumed to be P_{t_1} (less than 0.5% error).

The other significant parameter is the ratio of flow area (A_p) to the discharge area (A_1) downstream of the poppet seat interface. The analysis and reference data in Section 4.2.3 showed flow coefficients for fixed A_p/A_1 of 0.506 and 0.843. The values of A_p/A_1 for

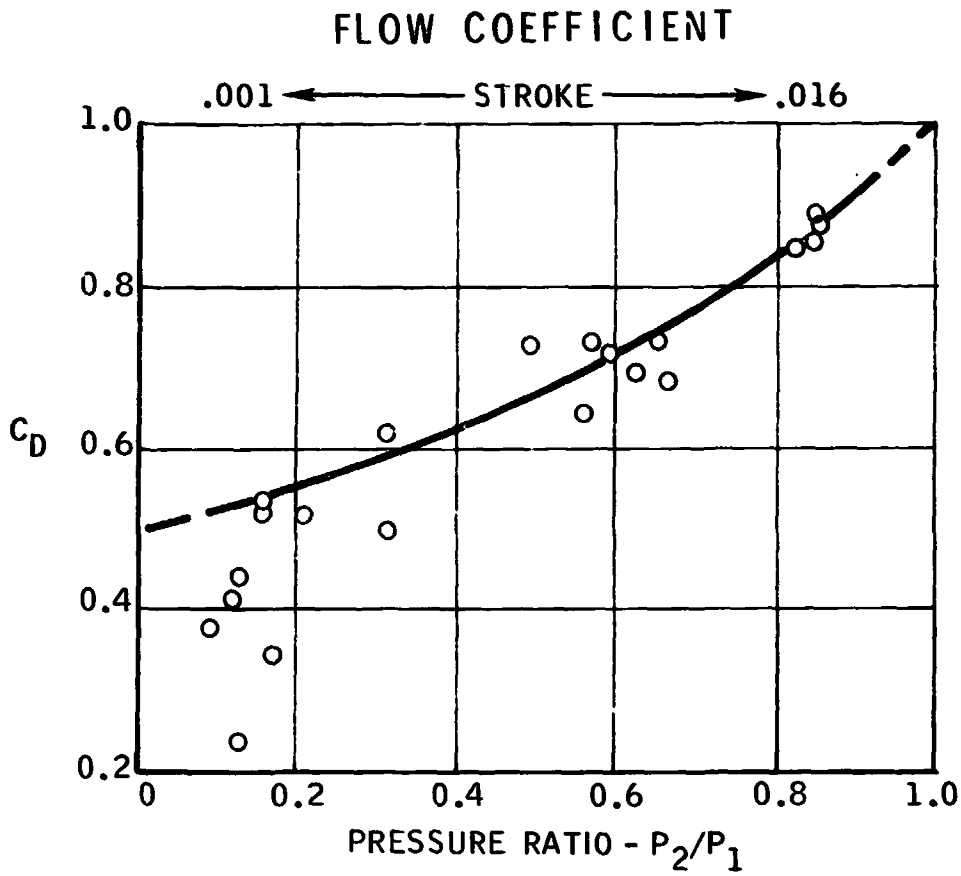


Figure 7-8

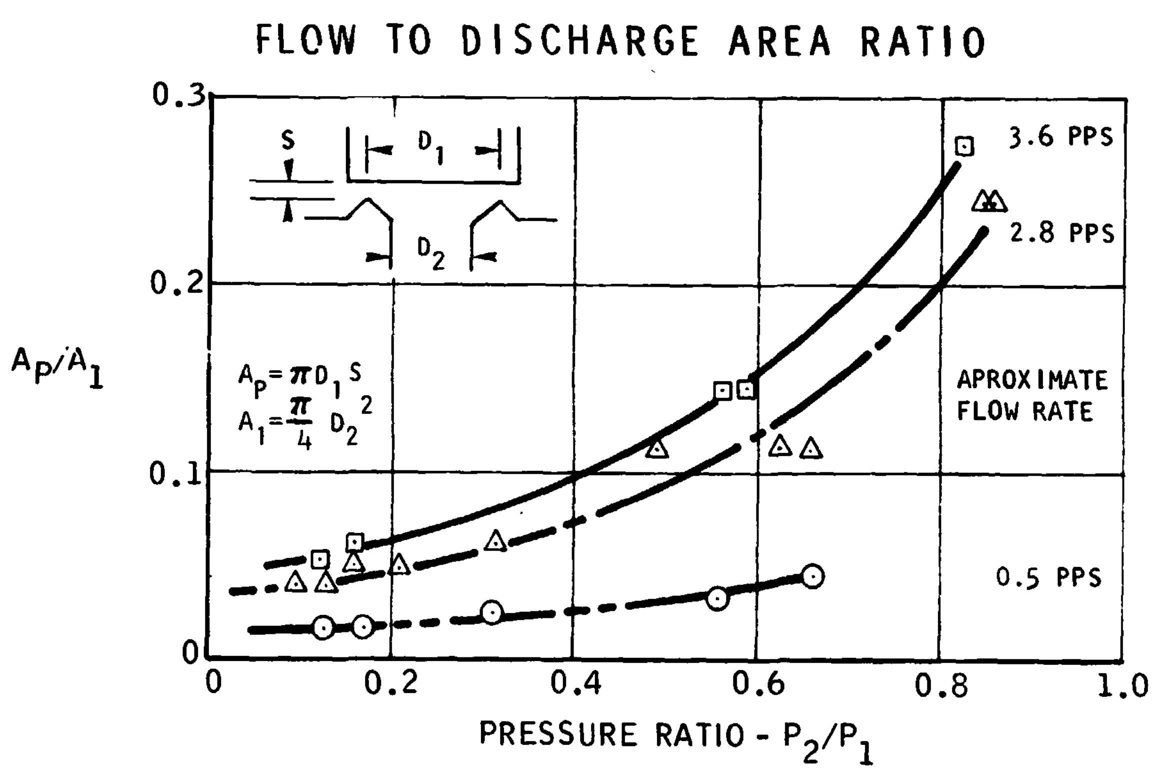


Figure 7-9

this configuration range from 0.017 to 0.274 as shown in Figure 7-9. Of this data, the value of C_D (Figure 7-8) determined for poppet strokes of 0.001 inch may be in error ±30%.

7.1.5 Pneumatic Damper

Under main engine vibration, the poppet was predicted (by the analog computer study) to vibrate with an amplitude greater than ±0.002 inch at the natural frequency of the poppet/spring combination. Damping coefficient of approximately 0.32 was found sufficient to keep the vibration-driven poppet motion under ±0.002 inch. A pneumatic damper which uses the helium pressurant gas as the damping medium was designed and fabricated.

The damper design shown in Figure 7-10 consists of a metallic diaphragm through which is placed an 0.013-inch diameter, 0.25-inch long capillary tube. As the poppet (lower stem) moves it deflects the diaphragm, thereby pumping the helium from one side of the diaphragm to the other. For test purposes, a linear variable differential transformer (LVDT) was attached to the top of the diaphragm to monitor valve poppet position.

During the vibration testing of the regulator and damper, vibration amplitudes greater than the target maximum amplitude of ±0.002 inch were observed when the regulator was operating. However, the off-target vibration was observed to occur at frequencies in excess of 300 cps. At the resonant frequency of the poppet/spring combination, the amplitudes were within the limits predicted by the computer.

Closer examination of the observed response to the imposed vibration and correlation of the frequencies with other modes of vibration of the springs revealed that the main springs were resonating within themselves (i.e., distributed parameter wave motion within the springs) in addition to the conventional spring-mass mode. These higher and more subtle vibration modes had not been simulated in the analog study and had not been included in the pneumatic damper design criteria.

Independent tests of the pneumatic damper were conducted to verify the design techniques. A photograph of the test setup is shown in Figure 7-11. The test procedure was to suddenly release a weight suspended just above (but in contact) with the damper shaft. The release of the weight acted as a force step input and the objective of the tests was to observe the response of the dashpot.

A typical response of the damper to the step input is shown in the oscilloscope photograph of Figure 7-12. The large test weight, in conjunction with the spring rate of the taut diaphragm, formed an oscillatory second order spring-mass system. The damping designed into the dashpot acts to damp the oscillations. Observation of the decay rate of the oscillations indicate that the damping coefficient was about 0.3, which is close to the design valve. The response and spring rate test data is presented in Tables 7-V and 7-VI.

PNEUMATIC DAMPER

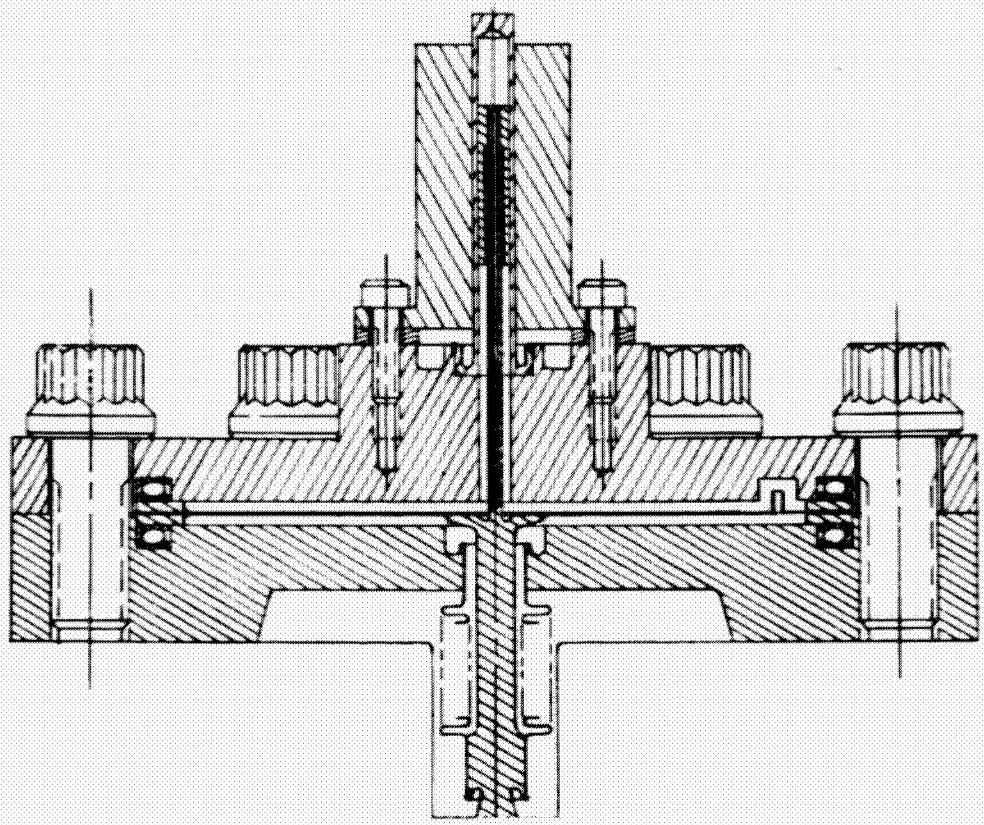


Figure 7-10

PNEUMATIC DAMPER TEST

TEST SETUP

Figure 7-11

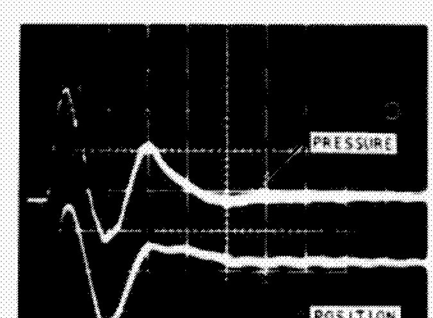


Figure 7-12

TABLE 7-V

PNEUMATIC DAMPER RESPONSE TEST

Test No.	Pneumatic Pressure (psig)	Drop Weight (lbs)	Valve Position	Orifice Size (in.)
1	250	15.25	Closed	0.009
2	250	17.26		0.009
3	250	13.24		0.009
4	300	13.24		0.009
5	300	15.25		0.009
6	250	15.25		0.013
7	250	13.24		0.013
8	300	15.25		0.013
9	300	15.25		0.013
10	300	17.26		0.013
11	300	17.26		0.013
12	200	17.26		0.013
13	200	17.26		0.013

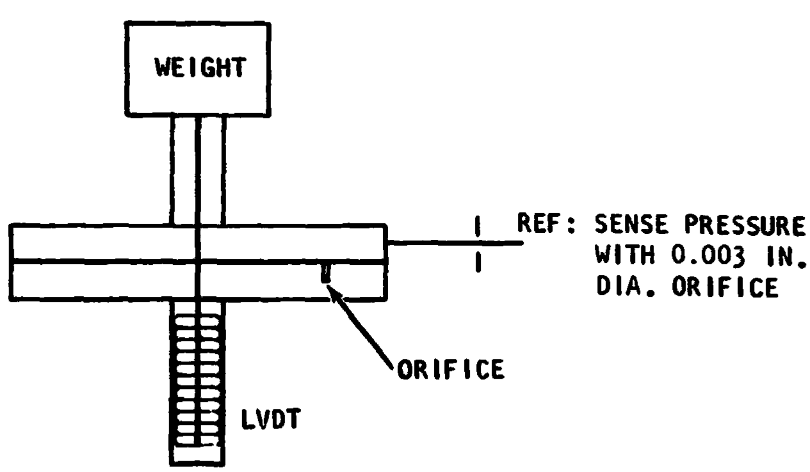


TABLE 7-VI

PNEUMATIC DAMPER SPRINGRATE TEST

Test No.	Force (lbf)	Displacement (inch)
1	0	0
	1.5	0.005
	3.3	0.010
	5.4	0.015
	8.4	0.020
	9.7	0.022
2	0	0
	1.7	0.005
	3.5	0.010
	7.2	0.015
	7.8	0.016
3	0	0
	1.6	0.005
	3.6	0.010
	5.5	0.0135
	6.5	0.015
	8.1	0.017
	8.9	0.018

However, since this amount of damping was inadequate to limit the vibration amplitude at frequencies higher than the fundamental resonance (150 cps), vibration testing was continued with a simple coulomb friction damper which was more easily adjustable than the pneumatic damper. The objective of the additional tests was to define how much damping was required to bring the poppet motion under imposed vibration to target amplitudes.

7.1.6 Mechanical Damper

A mechanical damper was designed and fabricated to determine the friction force that is required to reduce poppet amplitudes to ±0.002 inch under main engine vibration levels. The damper design allowed a range of friction forces by adjusting set screws connected to a set of flexures which produced a compressive load on the regulator shaft. A picture of the damper is shown in Figure 7-13.

Three friction materials (Teflon, polyimide, and brass) were tested to evaluate friction characteristics, and the results of these tests are shown in Figure 7-14. The polyimide material was selected because the difference between the breakaway friction and the sliding friction was not as great as for the Teflon and brass.

The spring rate for the support flexures is 435 lbs/inch.

The results of using the mechanical damper during the vibration tests is discussed in Section 7.2.

7.1.7 Flow Limiter

The flow limiter pressure drop tests were conducted in accordance with Paragraph B-2 of Appendix B to ITP 0213.

A flow limiter was designed and fabricated to limit flow to 10 lb/min in the event both series regulators failed in the open position. Of the flow limiter designs described in Section 5.2.1, the spring actuated venturi was selected for fabrication and test. The design features a venturi meter with a flexure mounted pintle which is pressure actuated by aerodynamic loads to vary the throat area. A cross-sectional view of the flow limiter is shown in Figure 7-15. The flow limiter is designed to be placed upstream of the series regulators to operate at Mach 1 when a failure occurs downstream and at low Mach numbers and pressure drops when operating at normal conditions, as shown in Figure 7-16.

The flow limiter was installed as shown in Figure 7-17 and was tested under a simulated failure condition. The results of Figure 7-18 indicated that the flow rate exceeded 10 lb/min because the flow limiter was not operating in a choked condition. Choked flow

MECHANICAL DAMPER

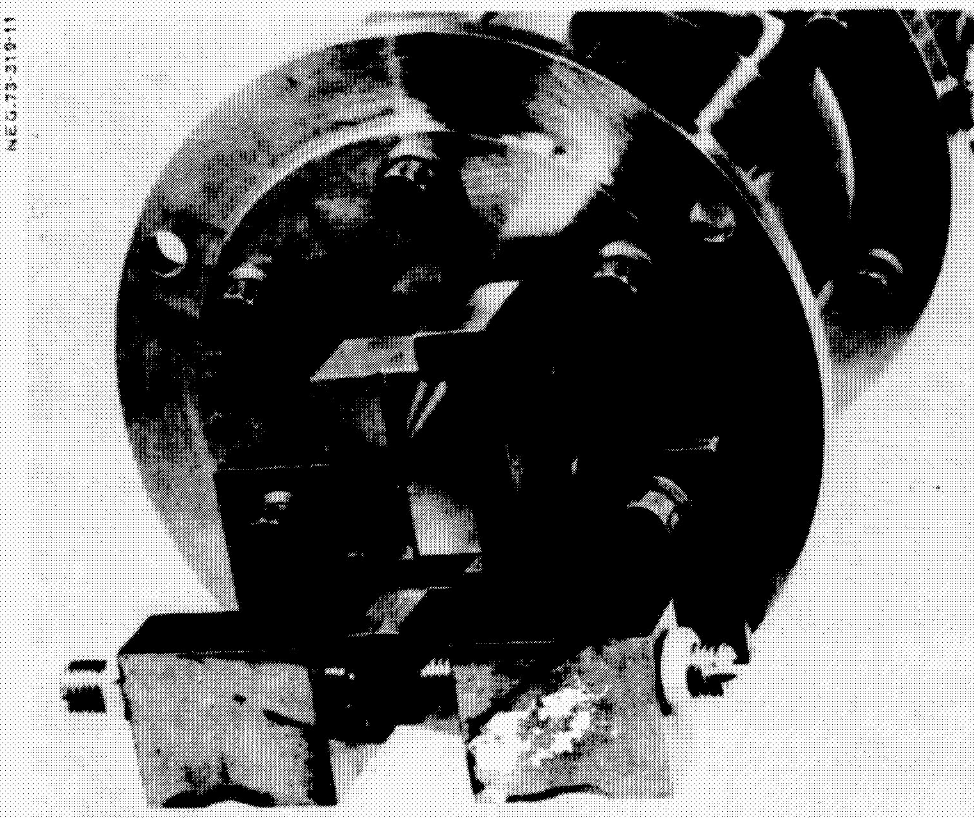


Figure 7-13

MECHANICAL DAMPER CHARACTERISTICS

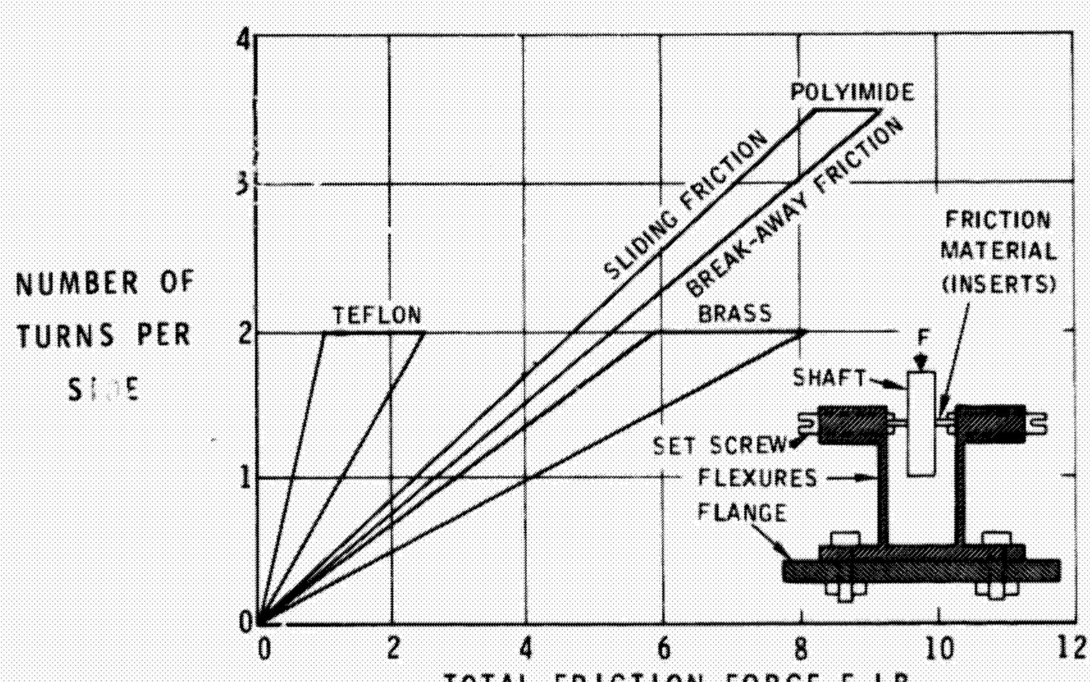


Figure 7-14

PROTOTYPE FLOW LIMITER

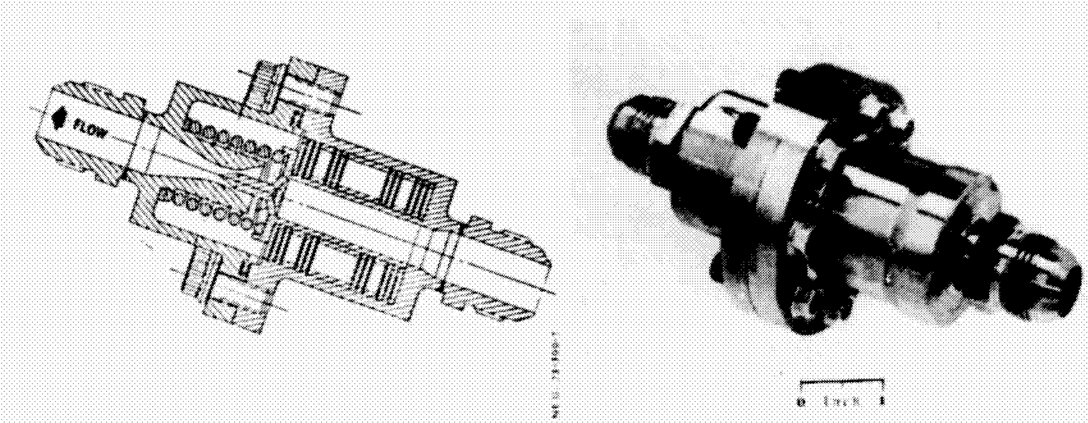


Figure 7-15

PREDICTED FLOW LIMITER PRESSURE LOSS

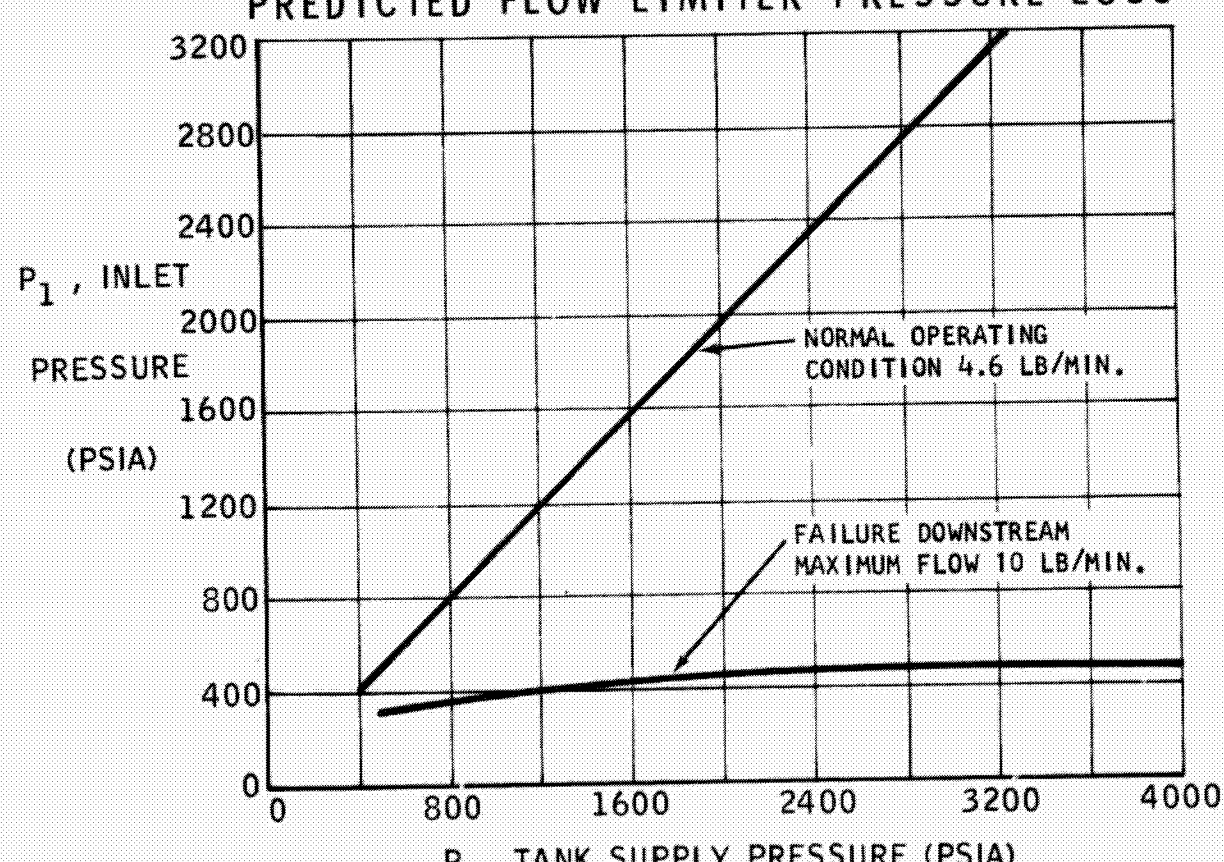


Figure 7-16

PREDICTED FLOW LIMITER MACH NUMBER VARIATION

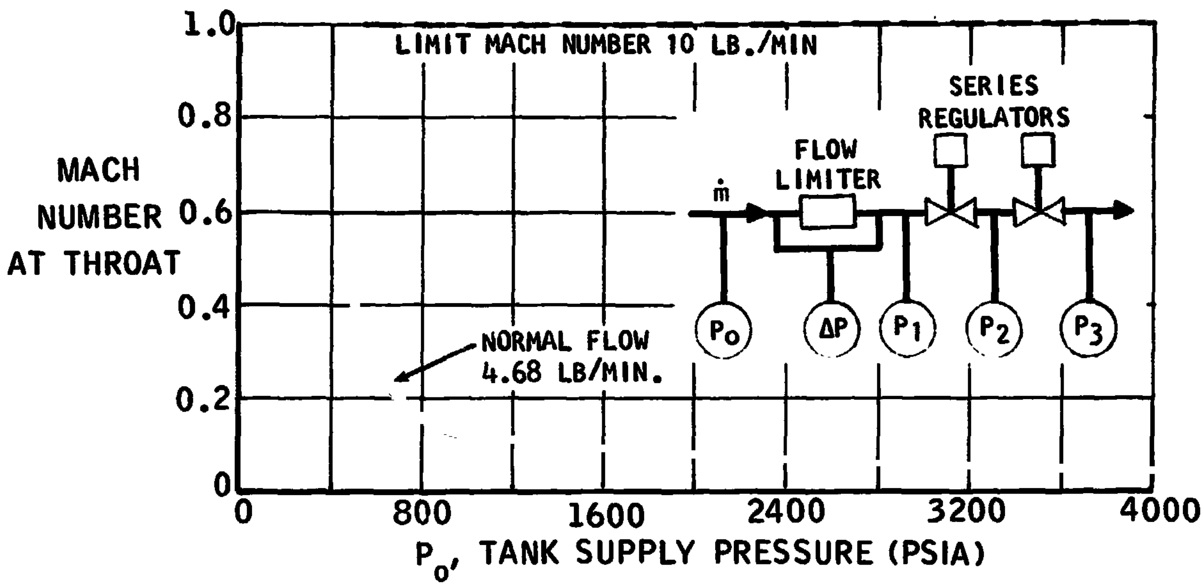


Figure 7-17

FLOW LIMITER PERFORMANCE

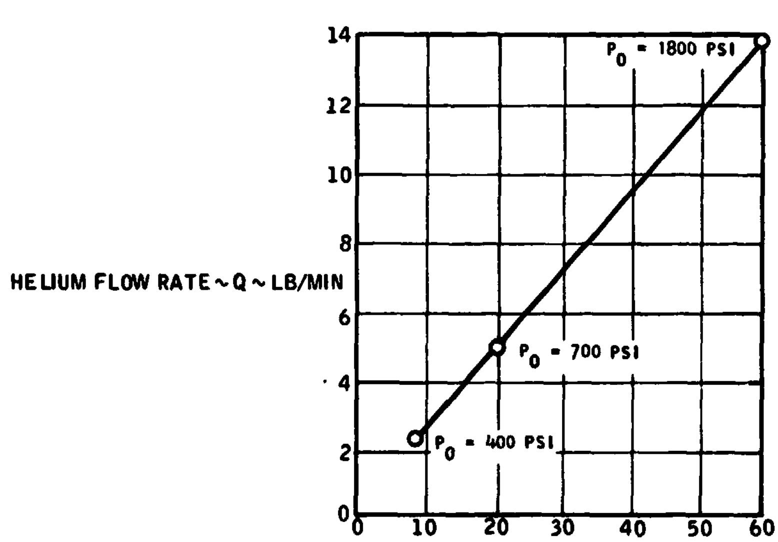


Figure 7-18

actually occurred at a component downstream of the flow limiter at a flow of 5 lb/min, resulting in an elevated flow limiter back pressure at subsequent higher flowrates.

This test was incomplete but suggests that the flow limiter must become active prior to any other subsystem component or the component could interact and cause failure of the flow limiter. An effective location for series regulators would be downstream of the regulator metering orifices.

TEST DATA Test Date: 9/17/73

Run No.	P_o psig	T_o °F	P_1 psig	P_2 psig	P_3 psig	ΔP F.L. psid	T_N °F	P_N psig	$\dot{m}$ ppm	ρ(3) lb/ft³	ΔP Reg1 psid	ΔP Reg2 psid	Stroke Inch
1	400	98	380	-	268	9.1	98	0.69	2.5	0.189	31	87	0.0229
2	-	104	760	680	285	20	105	2.70	5.0	0.198	70	325	0.0229
3	-	104	1800	1740	270	60	104	19.3	13.8	0.188	180	1290	0.0229

7.1.8 Contamination Cutting Tests

The design of the regulator sealing interface is a flat poppet and seat which offers low leakage for a long operating life. In addition, the seat is designed to cut away particles that normally cause excessive leakage. The cutting action is possible at low force levels by making the land width very narrow.

Tests were performed with land widths from 0.0006 to 0.004 inch to demonstrate the cutter seal design concept shown in Figure 7-19. Static load tests have indicated that the cutter seal effectiveness is a function of the land width, particle size and strength, load, and the included angle of the land. These tests were performed with stainless steel and copper wire of 0.001, 0.003, and 0.0063-inch diameters which simulated particle sizes of 25, 75, and 160 microns, respectively. The approximate yield strength of the stainless is 300,000 psi and the copper is 60,000 psi. Loads were varied up to 50 pounds to cut the wires and all tests were performed with a seat included angle of 120°. Test results indicated that the load required to effectively smash the wires to a smear thickness of approximately 30-millionth inch decreased as the land width decreased, as shown in Figure 7-20. Data is in Table 7-VII.

A typical top view of a smear is shown in Figure 7-21. In general, a 100% increase in load is required to reduce the smear thickness from 30 to 10-millionth inch, which is the order of magnitude that leakage significantly decreases as indicated from tests performed with a solenoid valve.

The solenoid valve tests were limited in scope because the maximum cutting force available was only 16.4 pounds; however, the valve effectively cut a copper and stainless wire at a .003 inch diameter. This data is presented in Table 7-VIII.

CONTAMINATION CUTTING TESTS

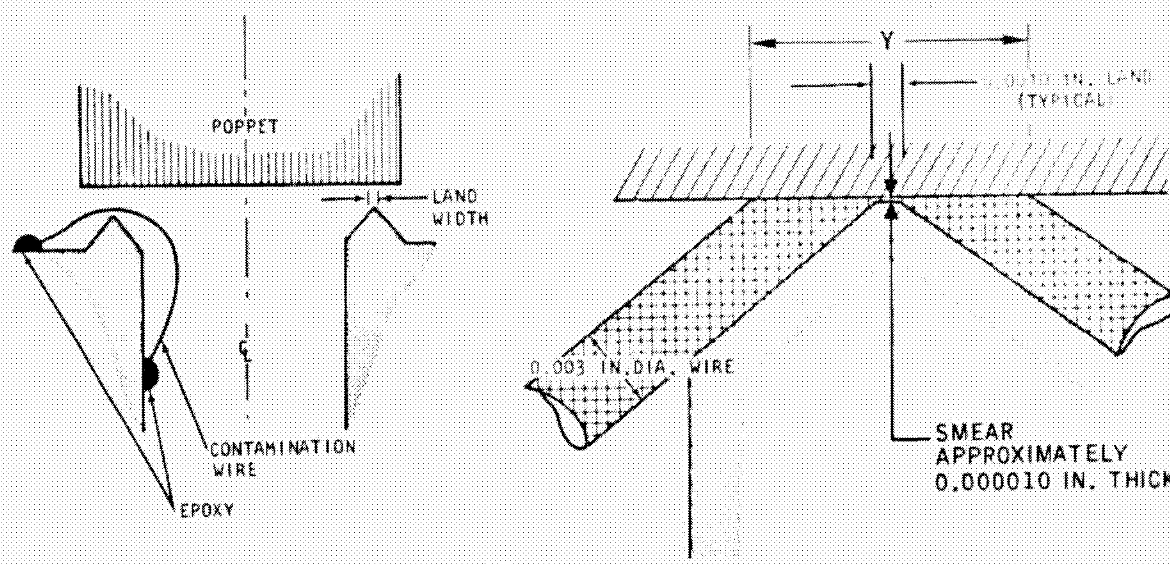


Figure 7-19

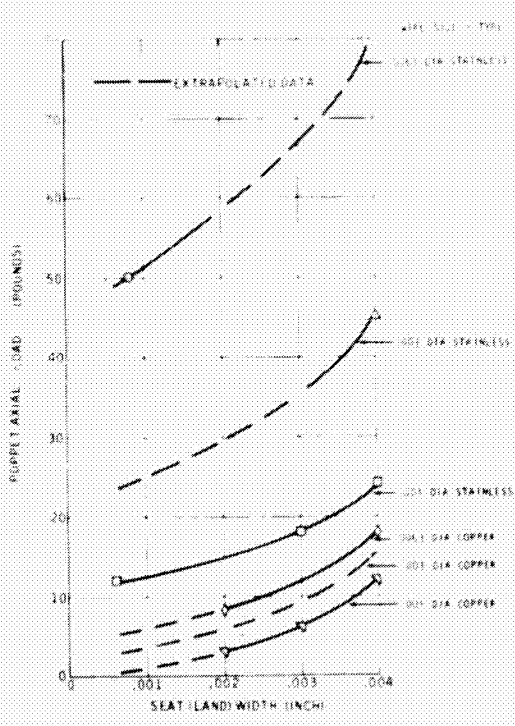
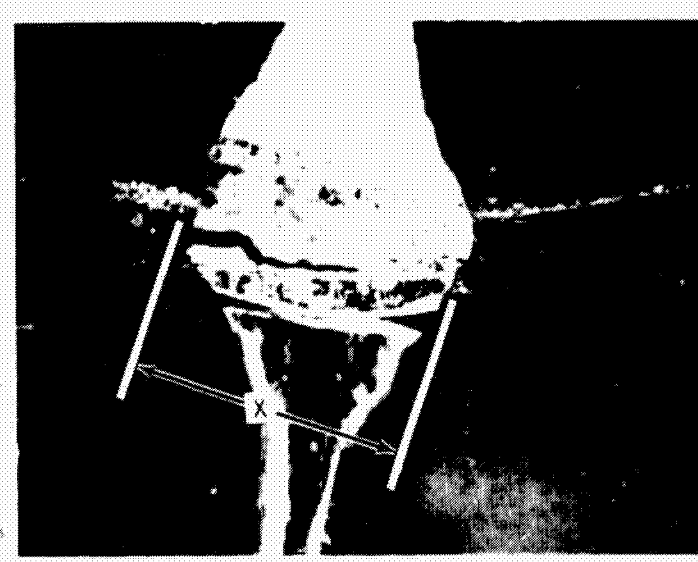


Figure 7-20



- COPPER WIRE DIAMETER-0.0063 (160μ)
- CUTTING FORCE-22 LBS. (TOTAL SEATING FORCE)
- LAND WIDTH-0.003 IN.

Figure 7-21

TABLE 7-VII

CONTAMINATION CUTTING TESTS WITH WEIGHTS

WC Cutter For 5/10 & 4/20 Tests, B_4C Cutter For 4/2 & 3/30 Tests

Date	Wire Material	D Wire Dia. 10^{-3}	L Land Width 10^{-3}	Force - Lbs			Remarks
				@ Min.* ΔX & ΔY	@ Start of Cut	@ Cut	
5/10	S/S	6.3	8			→	50% @ 28 lb, X = 6.4
		3		8	12	18	
		1		12	20	24	
	CU	6.3			18		
		3		12	18	22	
		1		12	18	22	
4/20	CU	6.3	2	8	20	24	X = 9.6 @ 8 lbs
		3			8		
		1		3		4	
	S/S	6.3		24			X = 6 @ 24 lb, X = 10.8 @ 50 lb
		3		28			X = 6.8 @ 28 & 50 lb
		1		28			X = 3.6 @ 28 & 50 lb
4/2	S/S	3	3	22			X = 8.8 @ 50 lb
		1		18	22		
	CU	6.3				19	
		3		18	20		
		1		6	8		
3/30	CU	1	4	12		24	X = 5.6 @ 12 & 24 lb
		3		18	24	24	
		6.3		18	20	20	
	S/S	1		24	40	50	
		3		45			
		6.3					X = 11.6 @ 50

X = wire width dimension in thousandths of an inch
X represents deformation in width, Y is deformation in length

*At first deformation of wire in either direction.

The results of these tests indicate the potential of contamination cutting seals. However, to reduce sealing loads and insure lower leakage rates, the included angle for the flight weight regulator has been reduced from 120° to 60°. A land width of approximately .001 inch is recommended to cut a 150 micron particle.

7.2 REGULATOR PERFORMANCE

The regulator was subjected to the expected inlet conditions and vibration environments of Section 3.0 between September and December of 1973. The unit was installed and tested in the helium flow and vibration facilities of Ogden Technology Laboratories, Inc. in Fullerton, California. The initial facility checkout was completed on 15 September, followed by the flow limiter test (reported in Section 7.1).

Regulator performance test at 70°F, +150°F and -150°F were completed, except for noted facility failures. Regulator stability and slam starts were also conducted. Initial vibration tests indicated additional damping was required. Stability tests with check valves and vibration tests with the mechanical damper completed testing at Ogden labs.

The regulator was refurbished, a new seat installed, and life cycle tested at Marquardt's flow facility during February of 1974.

7.2.1 Outlet Pressure Regulation

The regulator was installed in the test setup shown schematically in Figure 7-22. The facility capabilities included providing helium gas at pressures for 400 to 4000 psig at temperatures of -150°F to +150°F, and at flow rates of 44, 265 and 340 scfm (noted as 2.6, 15.6 and 20 cfm at 250 psia). These flow rates represent the nominal RCS, nominal OMS and maximum OMS requirements at this date (July 1973). A portion of the test setup showing the regulator and the immediate downstream plumbing is shown in Figure 7-23.

The selection of inlet pressures were determined by the minimum inlet pressure requirement (400 psig), the maximum inlet requirement (4000 psig), the critical pressure where the primary regulator is either subsonic at lower inlet pressures and sonic at inlet pressures above 680 psig, and an intermediate pressure (2000 psig).

The inlet temperature and pressure conditions are set with the upstream valve open. The flow rate is determined by the flow nozzle and regulated pressure at that flow rate is determined when the downstream valve is open. Lock-up pressures are recorded one minute after closing the downstream valve.

TABLE 7-VIII

CONTAMINATION CUTTING TESTS WITH SOLENOID VALVE

B_4C Cutter On 5/14 & 5/15, K-96 Cutter On 5/30, 6/1, & 6/4, K-801 Cutter on Remaining Tests

Date	Wire (type)	Size (dia) inch	Land (width) inch	Force - lbs		Leakage-scch		
				Start	Cut	50 psi	250 psi	400 psi
5/14	None	-	0.004	-	-	4	30	52
5/14	CU	0.001	0.004	7	-	1210	11,900	-
5/15	CU	0.003	0.004	7	-	177	860	-
5/30	CU	0.003	0.0005	-	8.0+	-	-	188
5/30	None	-	0.0005	-	-	12	-	162
5/31	SS	0.003	0.0005	6.8	-	102	-	405
5/31	None	-	0.0005	-	-	-	-	60
6/1	SS	0.0063	0.0005	4.9	-	-	-	Excessive
6/4	CU	0.0063	0.0005	-	-	-	-	Excessive
6/15	CU	0.0063	0.0005	-	15.2+	14	-	261
6/15	SS	0.003	0.0005	16.4	-	233	-	2080
10/10	None	-	0.002	-	-	-	14 & 15	45 & 39
10/10	CU	0.003	0.002	-	1[illegible].4+	-	-	555
10/10	CU	0.001	0.002	-	16.4+	-	-	Excessive
10/10	CU	0.006	0.002	16.4+	-	-	-	Excessive
10/10	None	-	0.002	-	-	-	-	555++

Reference leakage rate - no contaminants and 400 psig (in scch) 52, 162, 60, 45, and 39; Avg. = 71 scch

+Smear thickness not measured.

++Inspected and found hole in land.

REGULATOR TEST SETUP SCHEMATIC

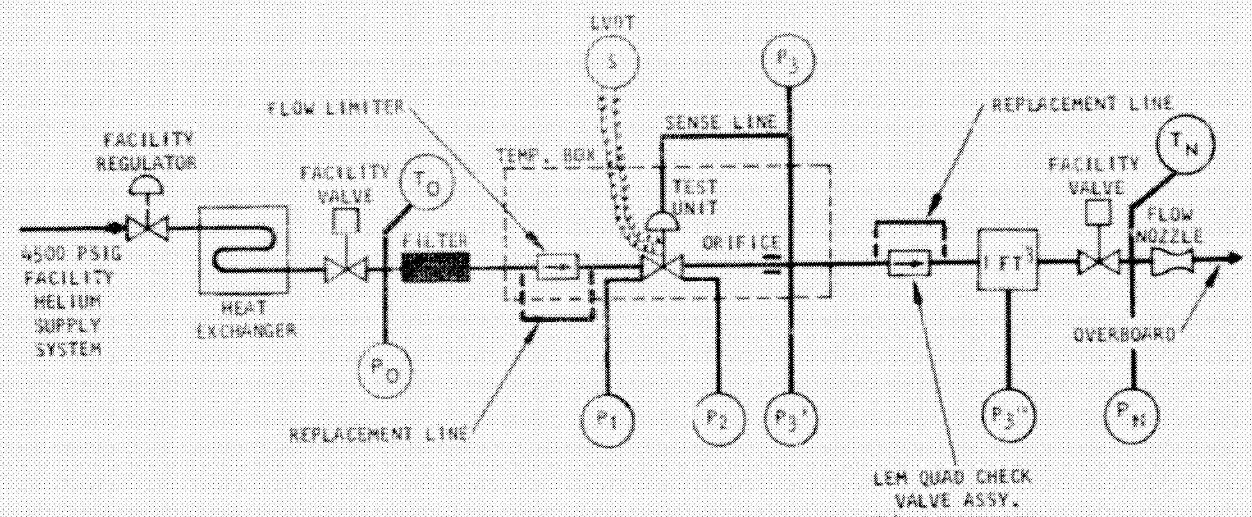


Figure 7-22

REGULATOR PERFORMANCE TEST SETUP

OGDEN TECHNOLOGY LABORATORIES

CHECK VALVE REMOVED

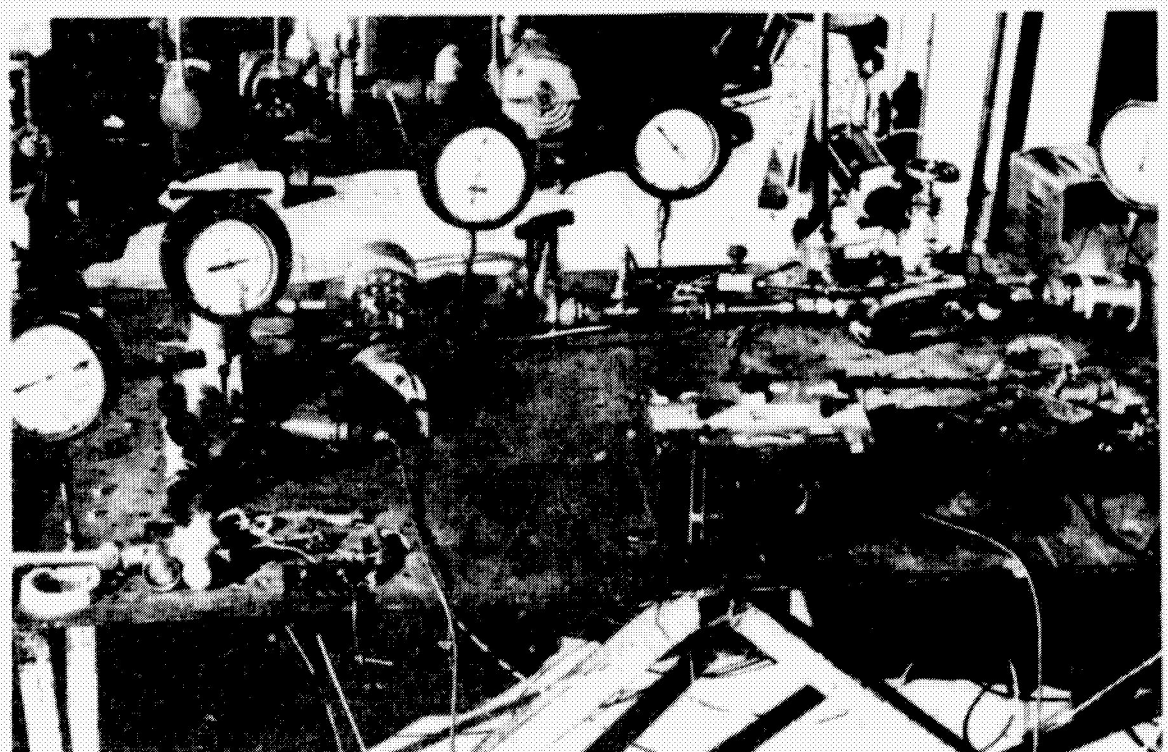


Figure 7-23

TEST DATA SHEET TABLE 7-IX

SHEET 1 OF 1

MAC A 948

TEST OPERATOR	TEST ENGINEER	WITNESS	CLASSIFICATION
DICK KONST	JOE HERNANDZ	G. POND & O. BENZ	

SUBJECT PERFORMANCE TEST @ 70°F MODEL X29200 SER. NO. 1

SHOP TRAVELER NO.

REF: FIG 7-22 FOR INSTRUMENTATION LOCATION

TEST NO. B-4 PER MTP0213

Tatm(DRY) °F

BAR. 29.80 AT 70 °F

FUEL SPEC.

FUEL SP. GR. AT °F

TEST DATE 8/31/73

RUN NO.	PT. NO.	TIME	P_1 PSIG	P_2 PSIG	P_3' PSIG	LOCKUP PSIG	U/S T_0 °F	D/S T_N °F	$\dot{m}$ LB/MIN	Q_3' CFM	
1 A			420	335	256.8	268.5	–	104	2.74	15.3	
1 B			360	332	253.1	–	64	64	2.84	14.9	
2 B			390	332	254A	–	62	62	2.83	14.7	
3 B			660	334	258.5	–	60	60	2.87	14.6	
4 B			1910	320	258.0	–	60	60	2.83	14.4	
5 B			3920	320	254.2	–	–	65	2.79	14.6	
6 B			3700	365	253.3	–	–	81	3.70	20.1	
7 B			1950	372	255.4	–	–	81	3.74	20.1	
8 B			560	380	255.0	–	–	81	3.73	20.0	
9 B			350	340	–	–	–	81	–	–	
10 B			480	380	253.5	–	–	81	3.70	20.1	
11 B			670	–	265.4	267.6	–	80	.459	2.47	
12 B			420	–	264.8	267.6	–	80	.462	2.48	
13 B			2020	–	264.6	267.4	–	80	.466	2.50	
14 B			3950	–	262.5	271.0	–	80	.461	2.50	
											9/17/73
1			380	335	237.8	251.4	84	85	2.54	14.5	
2			660	345	2420	251.4	85	87	2.54	14.5	
3			1990	340	241.0	251.4	89	92	2.49	14.4	
4			3850	340	240.2	254.2	–	99	2.47	14.6	

Dynamic performance was determined from data obtained from pressure transducers and LVDT indicating on an oscillograph recorder. Regulated and lock-up pressures were recorded from a precision pressure gage using a fused quartz pressure sensing element and optical transducer (Texas Instruments Model 145).

7.2.1.1 Performance at 70°F

Regulator performance tests were conducted at room temperature with helium at inlet pressures of 400, 680, 2000, and 4000 psig and flowrates of 2.6, 15.6, and 20 cfm for run durations up to 30 seconds in accordance with MTP 0213, Paragraph B-4.

Performance data is presented in Table 7-IX. This data is first plotted on working curves called regulator "blowdown" characteristics which reflect the effect of inlet pressure on regulated pressure for a constant flowrate condition. Data from these curves can then be used to describe the regulator droop characteristics shown in Figure 7-24. The results of test with 70°F pressurant indicate that the regulated pressure deadband for the nominal OMS flowrate of 15.6 cfm is ±2.2 psi.

Following the initial tests on August 31, the regulator was removed from the test setup and the spring spacer changed to reduce the lock-up pressure approximately 8 psi. However, when the tests were resumed on September 17, some difficulty was experienced in setting run conditions which resulted in overpressurizing the downstream system and operation of the relief valve set at 340 psig. The maximum pressure attained is not known, but the additional 8-9 (16-17 total) psi reduction in lock-up pressure suggests a permanent set in the actuator bellows spring rate. No further occurrences of overpressurization of the actuator were noted during the performance portion of the test program, nor were there any significant changes in lock-up pressure with temperature or pressure as shown in Table 7-X.

TABLE 7-X

LOCK-UP PRESSURE TEST SUMMARY

Test Date	Run Number	Pressure-psig Inlet	Pressure-psig Lock-Up	Nozzle Temp. °F	Test Date	Run Number	Pressure-psig Inlet	Pressure-psig Lock-Up	Nozzle Temp. °F
8/31/73	1A	420	268.5	104	9/17/73	4	3850	254.2*	99
↓	11B	670	267.6	80	9/20/73	7	490	248.6	164
↓	12B	420	267.6	80	↓	8	660	248.6	149
↓	13B	2020	267.4	80	↓	9	1970	249.2	151
↓	14B	3950	271.0*	80	↓	11	3960	252.4*	149
9/17/73	1	380	251.4	85	9/29/73	1	430	247.3	-132
↓	2	660	251.4	87	↓	2	600	248.2	-121
↓	3	[illegible]90	251.4	92	↓	3	1988	249.1	-109

*Indicated lock-up valve recorded during high seat leakage condition.

REGULATOR DROOP CHARACTERISTICS AT +70° F

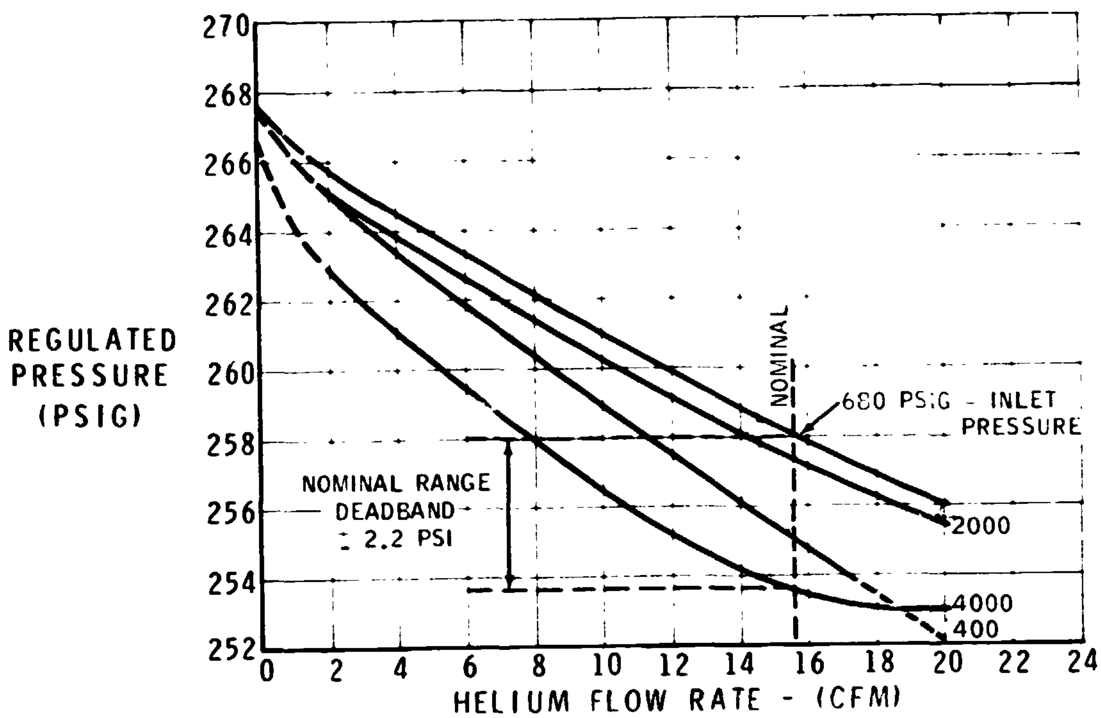


Figure 7-24

REGULATOR DROOP CHARACTERISTICS AT +150° F

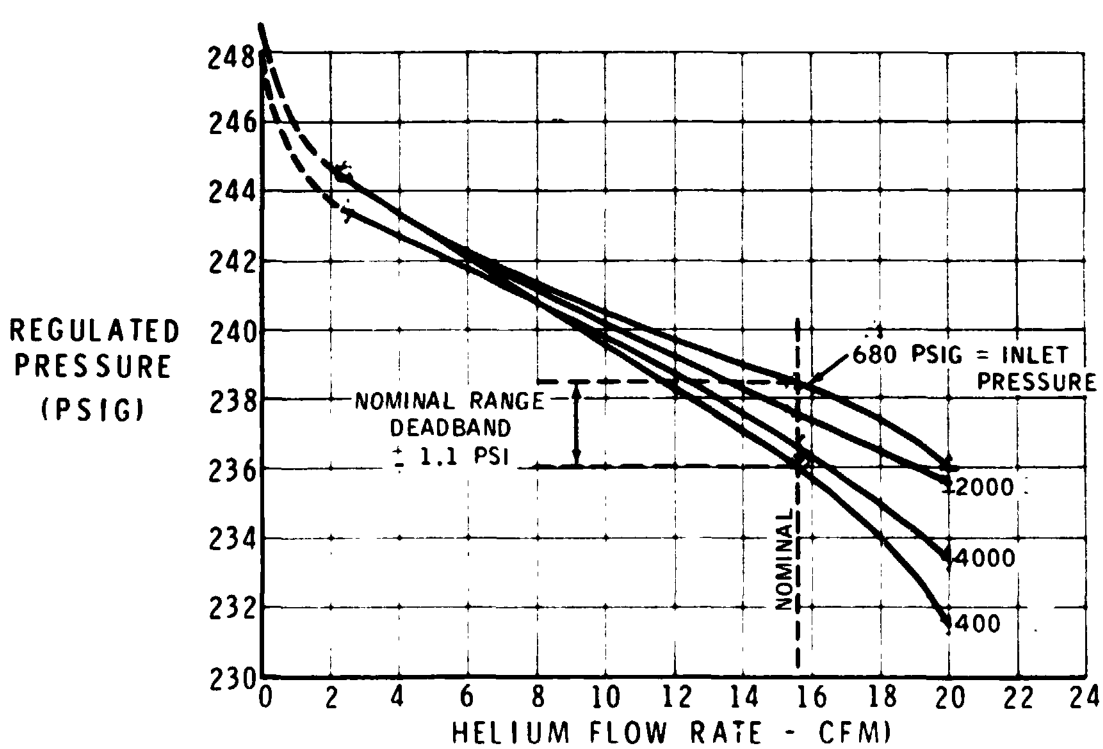


Figure 7-25

7.2.1.2 Performance at 150°F

Regulator performance tests were conducted at 150°F inlet and regulator temperature conditions as specified in MTP 0213, Appendix B, Paragraph B-5. A box with a heater was placed over the test item and an inlet line heat exchanger to control helium and regulator temperature. Fourteen runs were conducted on September 20 without incident. The data, tabulated in Table 7-XI was adjusted for temperature and pressure, using the "blowdown" working curves and cross-plotted in Figure 7-25 to present the droop characteristics. The regulated pressure deadband for the nominal OMS flowrate of 15.6 cfm is ±1.1 psi.

7.2.1.3 Performance at -150°F

Regulator performance tests were conducted at -150°F inlet and regulator temperature conditions in accordance with Paragraph B-6 of MTP 0213, Appendix B. Helium gas and the regulator were insulated and thermally conditioned with LN_2. Eleven runs were made on September 29 with some difficulty. The data presented in Table 7-XII indicates the loss of data on Run 5 and lock-up on Run 10 due to excessive leakage (contamination). The droop characteristics for the 150°F conditions are shown in Figure 7-26. The regulated pressure deadband is the same (±1.1 psi) as for +150°F operation.

The droop characteristics obtained from all test data (represented by each pressure at 150°F and 680 psi at -150°F) were correlated with the theoretical model as shown in Figure 7-27. The test data deadband is well within the limits suggested by the model. The model data may be conservative but appears to adequately describe the regulator characteristics when analyzed by single degree of freedom techniques.

Performance at an inlet pressure of 4000 psig was not determined due to facility limitations. In addition, Runs 10 and 11 were conducted after a leaking facility filter had been removed and the compressor repaired. During removal of the filter, the flow system became contaminated and iced up during the 4-hour (cold soak) interval. However, Runs 10 and 11 indicate that the regulator was still able to regulate under icing and contaminated conditions.

After Run 11, the regulator was disassembled and inspected. All the surfaces exposed to helium were covered with black oil and two insects were at the poppet outside diameter. Contaminates are shown in Figure 7-28 and the insects are shown in Figure 7-29.

7.2.1.4 Regulator Stability Considerations

Oscillograph traces of regulator stability at -150°F are presented for 15.6 and 20 cfm flow rates at the same inlet pressure (550 and 600 psi) and for two inlet pressures (600 and 2000 psi) at 15.6 cfm in Figures 7-30, 7-31, and 7-32. These traces are representative of regulator stability observed at all other temperature, pressure and flowrate conditions.

TEST DATA SHEET | TABLE 7-XI | SHEET 1 OF 1

MAC A 948

TEST OPERATOR	TEST ENGINEER	WITNESS	CLASSIFICATION
DICK KUNST	JOE HERNANDZ	O. BENZ	

SUBJECT PERFORMANCE TEST AT +150°F MODEL X29200 SER. NO. 1

SHOP TRAVELER NO.

REF: FIG 7-22 FOR INSTRUMENTATION LOCATION

TEST NO. B-5 PER MTP0213

Tatm (DRY) °F

BAR. 29.84 AT 70 °F

FUEL SPEC.

FUEL SP. GR. AT °F

TEST DATE 9/20/73

RUN NO.	PT. NO.	TIME	P_1 PSIG	P_2 PSIG	P_3' PSIG	LOCKUP PSIG	U/S T_0 °F	D/S T_N °F	$\dot{m}$ LB/MIN	Q_3' CFM	REG T_R °F
1			380	346	234.9	248.5	145	132	2.26	14.3	130
2			430	364	236.3	249.2	160	154	2.38	15.7	148
3			470	360	[illegible]	249.1	153	158	2.37	15.6	151
4			680	361	238.5	249.2	155	140	2.41	15.5	151
5			1960	364	[illegible]	251.0	165	158	2.38	15.6	145
6			3920	365	236.6	–	165	165	2.36	15.75	161
7			490	425	233.5	248.6	152	164	3.03	20.5	155
8			660	424	236.0	248.6	141	149	3.07	20.0	153
9			1970	425	235.7	249.2	146	151	3.07	20.0	156
10			3880	430	233.5	256.4	[illegible]	174	3.01	20.5	149
11			3960	250	243.4	252.9	141	149	.380	2.46	153
12			1950	252	244.7	248.9	142	130	.387	2.36	152
13			670	251	244.7	248.6	140	118	.391	2.33	148
14			390	251	244.6	248.3	145	120	.391	2.33	148

TEST DATA SHEET TABLE 7-XII

SHEET 1 OF 1

MAC A 948

TEST OPERATOR	TEST ENGINEER	WITNESS	CLASSIFICATION
DICK KUNST	JOE HERNANDZ	O. BENZ	

SUBJECT PERFORMANCE TEST AT -150°F MODEL SER. NO.

SHOP TRAVELER NO.

REF: FIG 7-22 FOR INSTRUMENTATION LOCATION

TEST NO. B-6 PER MTP0213

Tatm(DRY) °F

BAR. 29.91 AT 70 °F

FUEL SPEC.

FUEL SP. GR. AT °F

TEST DATE 9/29/73

RUN NO.	PT. NO.	TIME	P_1 PSIG	P_2 PSIG	P_3' PSIG	LOCKUP PSIG	U/S T_0 °F	D/S T_N °F	$\dot{m}$ LB/MIN	Q_3' CFM	REG T_R °F			
1			430	328	237.7	247.3	-164	-132	4.95	17.20	-169			
2			600	340	2?9.1	248.2	-142	-121	4.86	17.37	-125			
3			1988	343	238.2	249.1	-152	-109	4.78	17.70	-125			
4			550	400	234.8	248.6	-169	-156	6.60	21.4	-131			
5			—	—	—	—	—	—	—	—	—			
6			720	413	236.3	249.1	-135	113	6.15	22.8	-164			
7			1960	408	236.3	249.5	-130	-49	5.64	24.7	-130			
8			380	250	246.9	249.2	-178	-156	.856	2.66	-156			
9			670	250	247.3	249.3	-151	-117	.803	2.81	-143			
10			1960	245	243.2	—	-134	-34	.719	3.18	-125			
11			680	346	234.6	248.8	-160	-130	4.93	17.50	-8			

NOTES 1) FACILITY HELIUM COMPRESSOR PUMP FAILED AFTER RUN #9. ICING OCCURED AT REGULATOR INLET DURING REPAIR.

2) CONTAMINANTS FOUND IN REGULATOR AFTER RUN #11.

REGULATOR DROOP CHARACTERISTICS AT -150° F

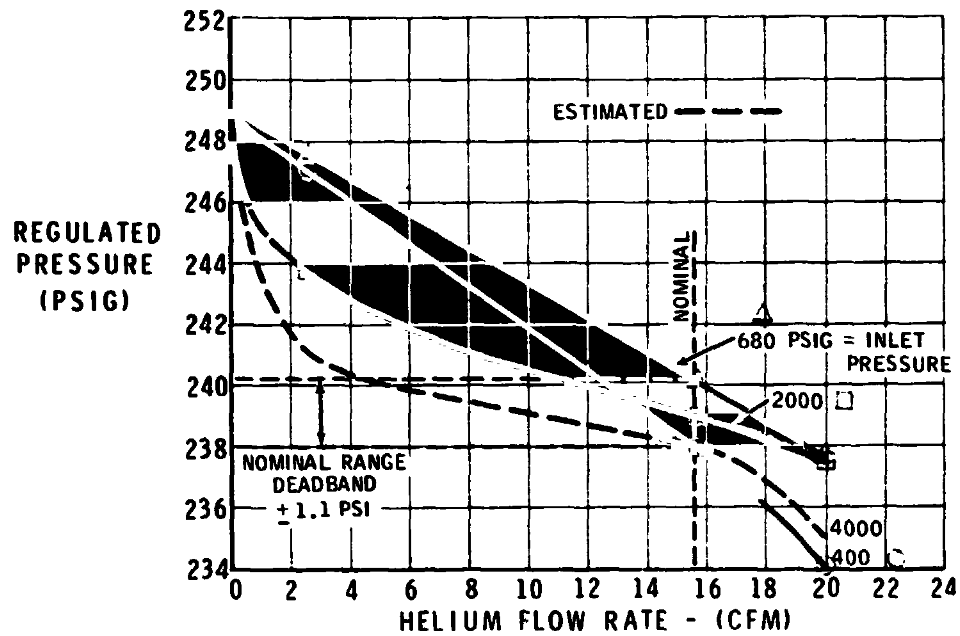


Figure 7-26

CORRELATION OF DROOP DATA WITH THEORETICAL PREDICTIONS AT 150° F

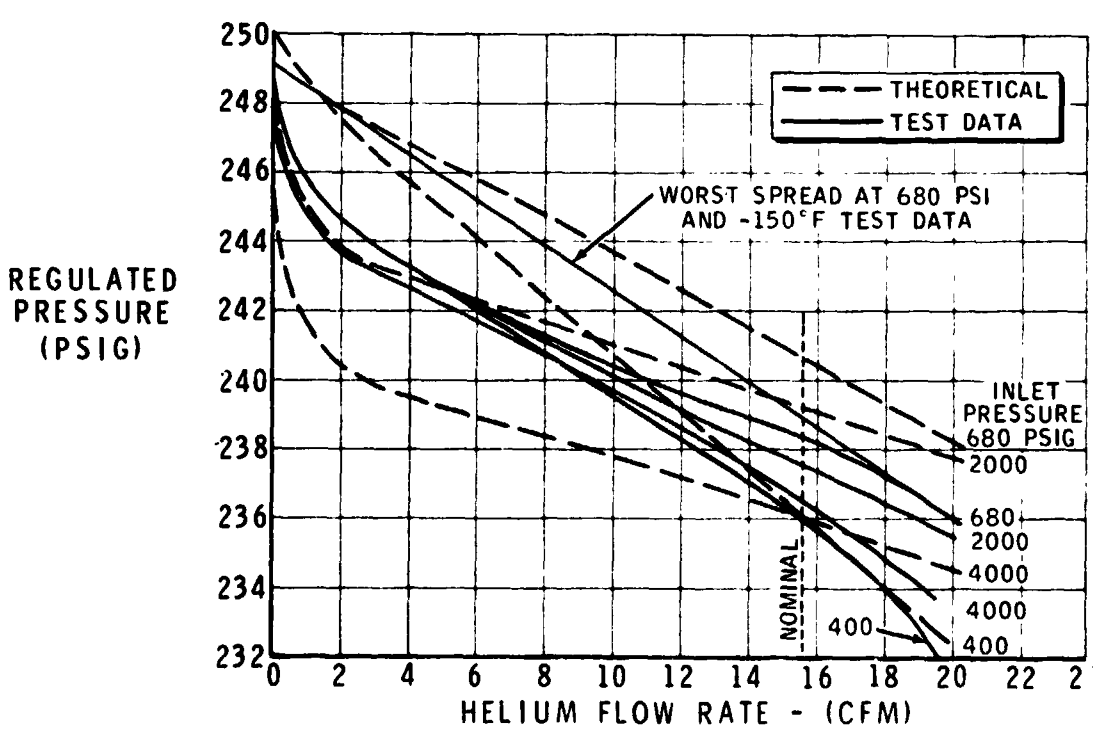


Figure 7-27

REGULATOR SEAT CONTAMINATED WITH OIL

INSPECTED AFTER RUN 11 9/29/73

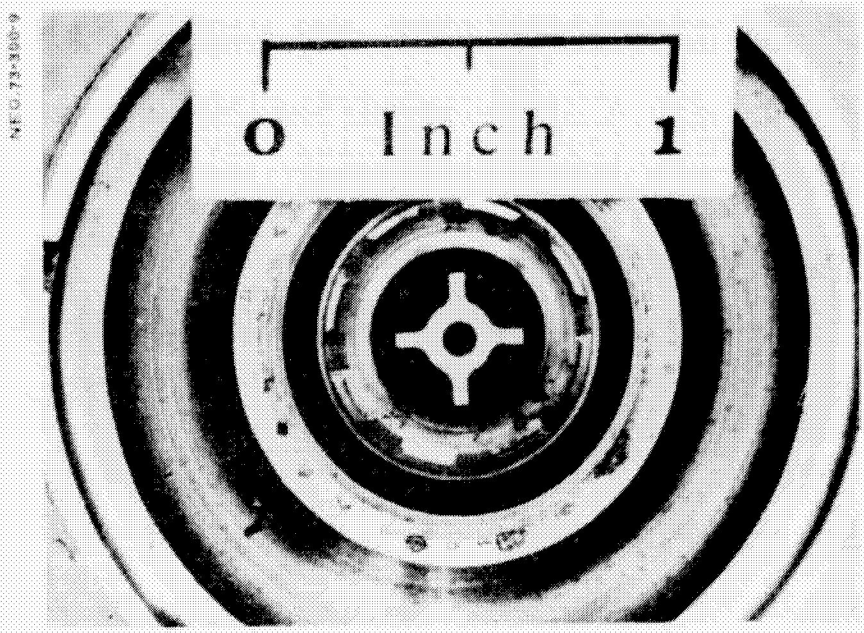


Figure 7-28

REGULATOR POPPET CONTAMINATED WITH INSECTS

INSPECTED AFTER RUN 11, 9/29/73

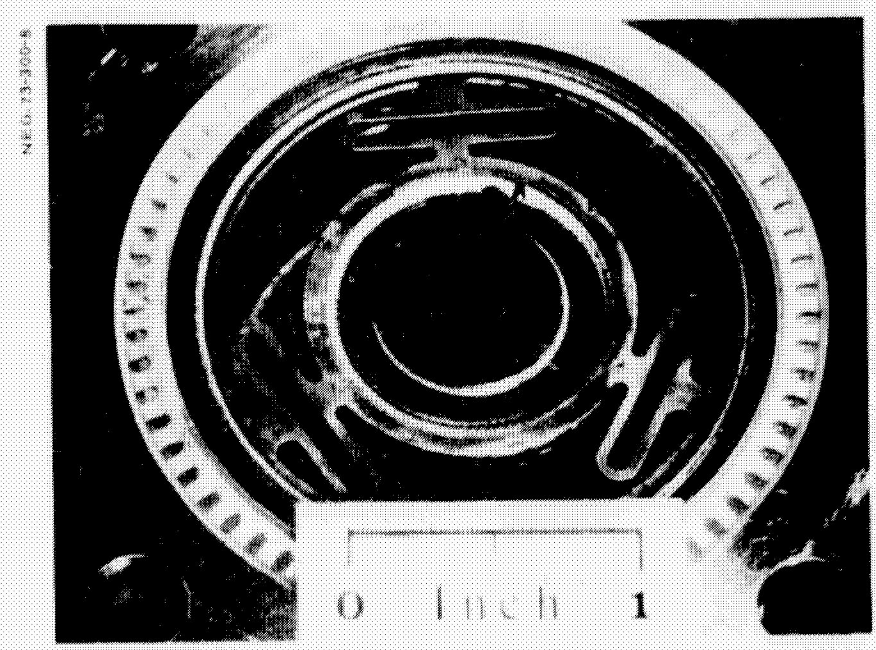


Figure 7-29

REGULATOR STABILITY AT 550 PSI INLET PRESSURE

-150°F AND 20 CFM

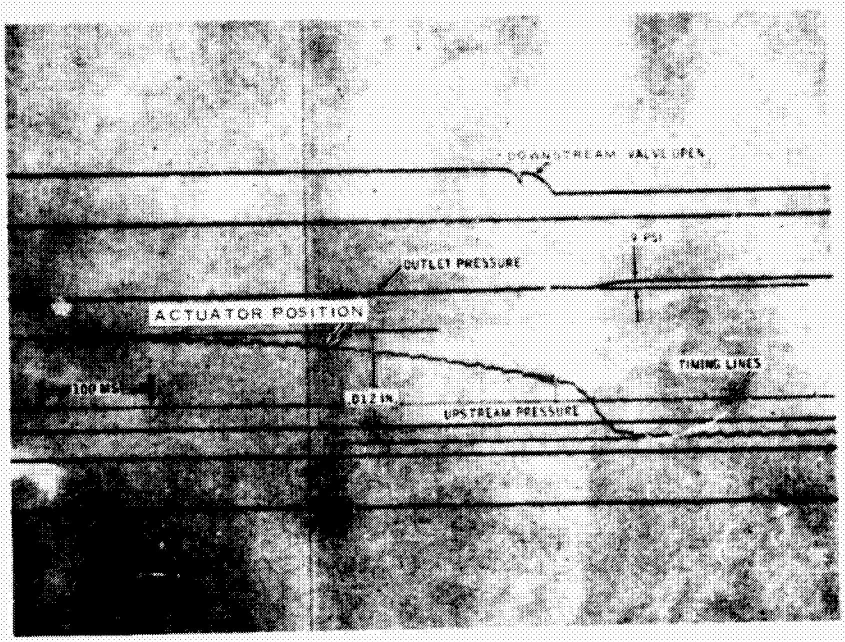


Figure 7-30

REGULATOR STABILITY AT 600 PSI INLET PRESSURE

-150°F AND 15.6 CFM

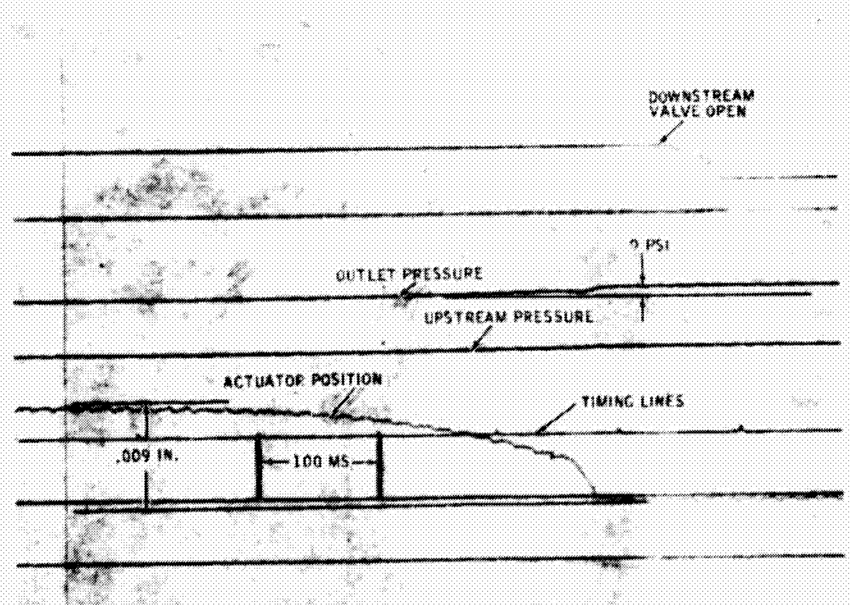


Figure 7-31

REGULATOR STABILITY AT 2000 PSI INLET PRESSURE
-150°F AND 15.6 CFM

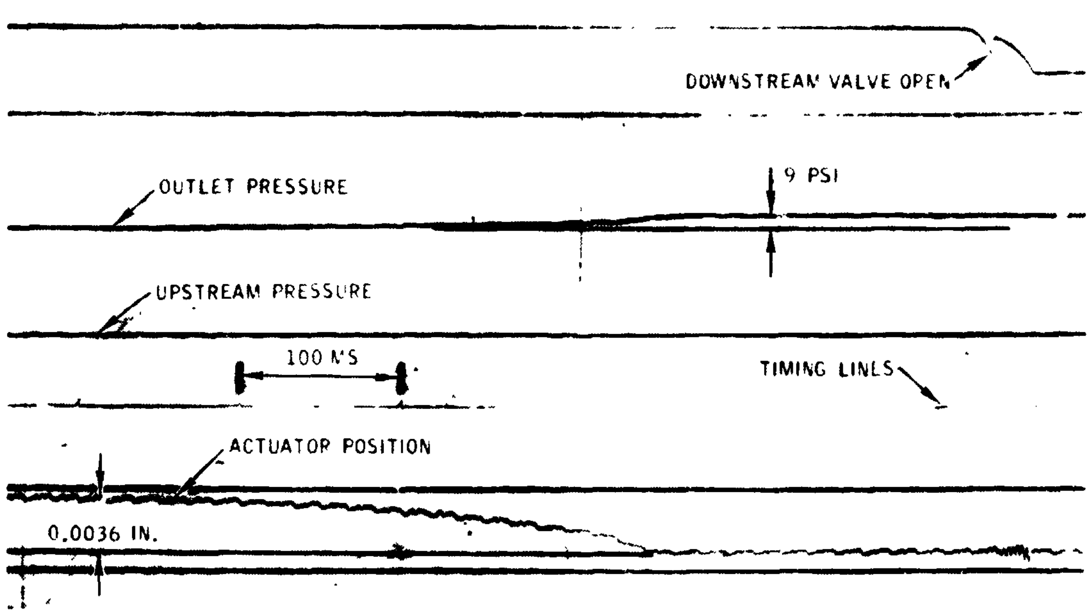


Figure 7-32

The conclusions drawn by oscillograph trace analysis are based on the typical readability of the magnitude of pressure and position traces.

Parameter	Span	Value	Readability (±.02 inch)
Inlet Pressure	1000 psi/inch	400-4000 psi	±20 psi
Outlet Pressure	100 psi/inch	0-250 psi	±2 psi
Actuator Position	.010 in/inch	0-.022 inch	±.0002 inch

Another factor considered in analyzing stability is the noise level or frequenc indicated by the parameter. The actuator position trace high noise level and oscillation remained unchanged prior to and during the run and was independent of the displacement. Similarly, the regulated outlet pressure signal width also remained constant during the regulating portion of the run.

7.2.2 Slam Start

Slam start tests were conducted as part of the ambient performance test requirements of Paragraph B-4. Two performance runs and two slam starts were conducted at ambient temperature conditions on October 12 following installation of a 0.10-inch diameter orifice to simulate the flow limiter rate of 980 scfm. The downstream flow volume of one cubic foot was reduced in pressure to one atmosphere, and the downstream valve closed. A 0.040-inch diameter orifice was placed in the sense line of the regulator actuator for one test and removed for the second. The slam start was simulated by opening the upstream facility valve and permitting the 3800 psig helium flow through the regulator and fill the 1 ft^3 ullage tank.

The purpose of the test was to determine the overshoot in regulated or lockup pressure resulting from an initial condition high inlet pressure (3800 psig) and a full open regulator. The oscillograph trace for the second slam start (Run 4) is presented as Figure 7-33. The regulator remained open for approximately 0.8 second as the ullage tank was pressurized to approximately 230 psig, at which time the regulator began to close. The regulator closes to 10 percent (0.002 stroke) within 130 to 150 milliseconds at a regulated pressure of 243 psig. Full lockup is attained approximately 0.9 second later at a pressure of 256 psig. This data is comparable to the ambient performance data obtained on 9/17 and 10/12. The test conditions and results are presented in Table 7-XIII.

TABLE 7-XIII

*Performance Runs

Run No.	P_1 psig	P_2 psig	P_3 psig	Lockup psig	T_o °F	T_N °F	$\dot{m}$ ppm	T_{reg} °F
1*	400	298	243.0	255.8	93	80	7.344	80
2*	660	302	246.2	255.9	92	77	7.450	-
3	3800	-	-	256	90	75	Slam Start	-
4	3800	-	-	256	89	72	Slam Start	-

7.2.3 Effect of Downstream Check Valves

A test to determine the sensitivity of the regulator to downstream component anomalies was not defined in MTP 0213. Midway through vibration testing, which revealed poppet oscillation, the check valve test was devised. This test consisted of installing the LM quad check valve assembly, P/N LSC270-817-3, S/N 197 (shown in Figure 7-34) into the downstream plumbing as shown in Figure 7-35. Performance tests were made at 15.6 cfm and ambient

SLAM START TEST

3800 PSI UPSTREAM PRESSURE, FLOW LIMITER AT 980 SCFM HELIUM, 1 FT.^{3} ULLAGE, AMBIENT TEMPERATURE AND PRESSURE

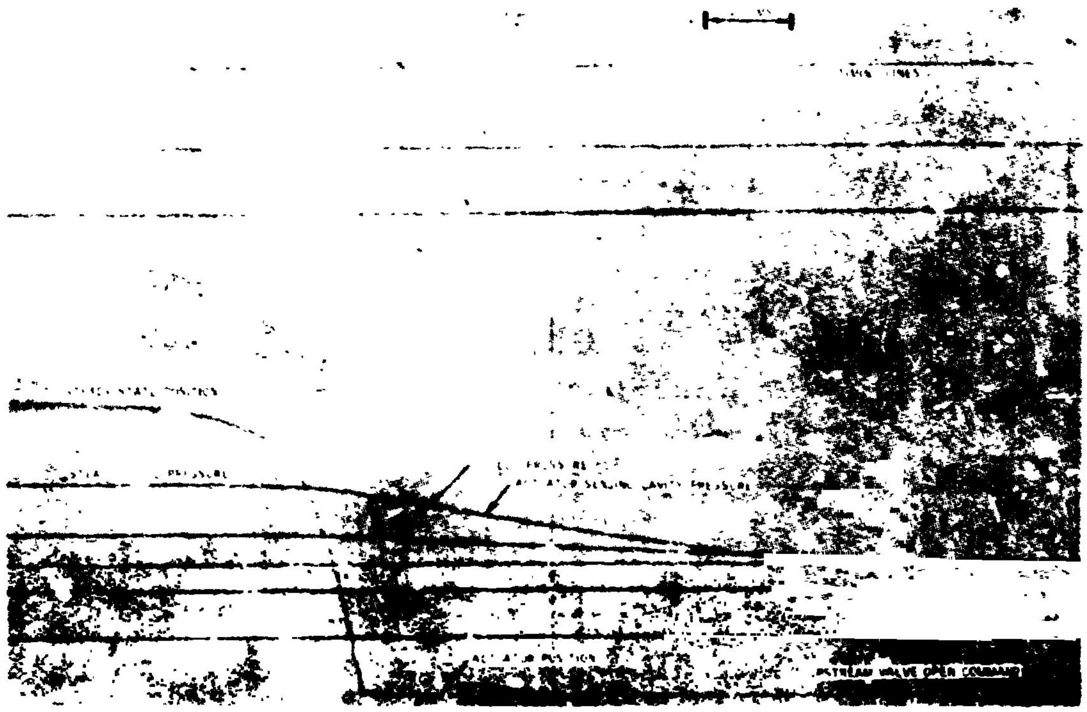


Figure 7-33

CHECK VALVE CONFIGURATION

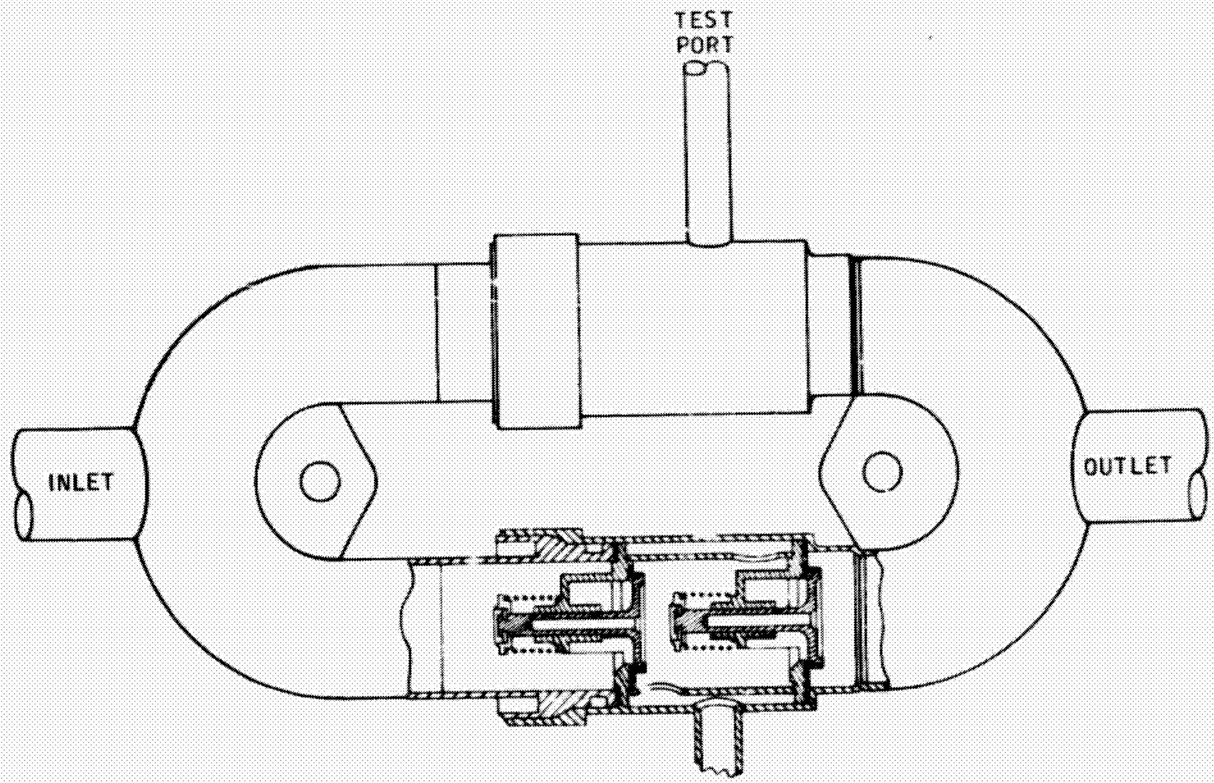


Figure 7-34

REGULATOR AND CHECK VALVE TEST SET UP

CHECK VALVE INSTALLED

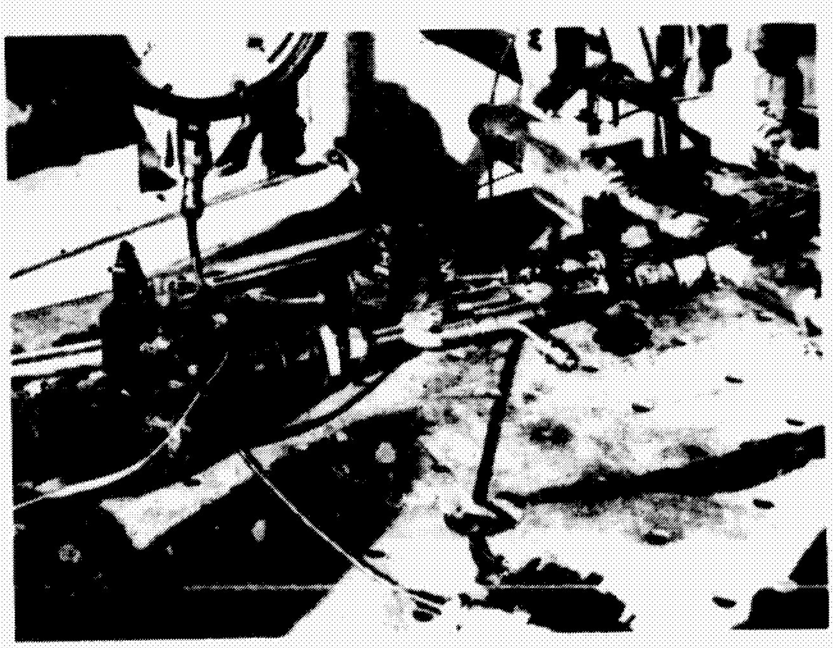


Figure 7-35

temperature at inlet pressures of 480 and 3900 psig. Three tests were made with the check valve in place and two without on November 20. The volume between regulator and check valve was matched to the volume of the bypass line when the check valve was removed.

A review of the test data presented in Table 7-XIV and typical oscillograph traces of regulated pressure without and with check valves is shown in Figures 7-36 and 7-37. The standard test system without check valves indicated nonoscillating regulated pressure and poppet position. With check valves installed in the test setup the regulated pressure - the sensing pressure measured upstream of the check valves - oscillated ±10 psi at a frequency of 120 Hz. However, the destabilizing tendency of the check valve did not alter regulator performance as evidenced by the stable poppet trace.

TABLE 7-XIV

Run No.	P_1 psig	P_2 psig	P_3 psig	Lockup psig	T_o °F	T_N °F	Notes
1	3900	340	290	288.9*	85	65	
2	3900	260 ±5	225 ±10	229.6**	80	60	With Check Valve
3	3800	275 ±7	220 ±10	230.9**	75	65	
4	480	280	225 ±10	230.5**	67	58	
5	460	280	225	230.6**	67	59	

* 0.040 sense line orifice inadvertently blocked.
**Lower lockup pressure following repair of pneumatic damper shaft.

7.2.4 Vibration

Vibration tests were conducted in accordance with Paragraphs B-7 (OMS Engine Run), 8 (Main Orbital Engine), and 9 (Lift-off) of MTP 0213, Appendix B. The regulator was installed on a shaker table and attached to the upstream and downstream flow system with flexible lines, 3/4-inch x 5-foot, as shown in Figure 7-38. The regulator was subjected to the random vibration spectrums shown in Figure 7-39. The procedure, tolerances, and duration of random vibration testing is specified in Table B-II of Appendix B to MTP 0213. Initial tests were conducted with a pneumatic damper designed to limit poppet travel during vibration. However, additional damping was required and the vibration tests were repeated with a mechanical damper. The test results using these two systems are described in the following sections. The axes of vibration are shown on Figure 6-1.

7.2.4.1 Vibration with Pneumatic Damper

The installation of the pneumatic damper was described earlier in Section 6.1.1. Tests were conducted in both X and Y axes. No tests were conducted in the Z axis. One test

REGULATOR TEST WITHOUT CHECK VALVE

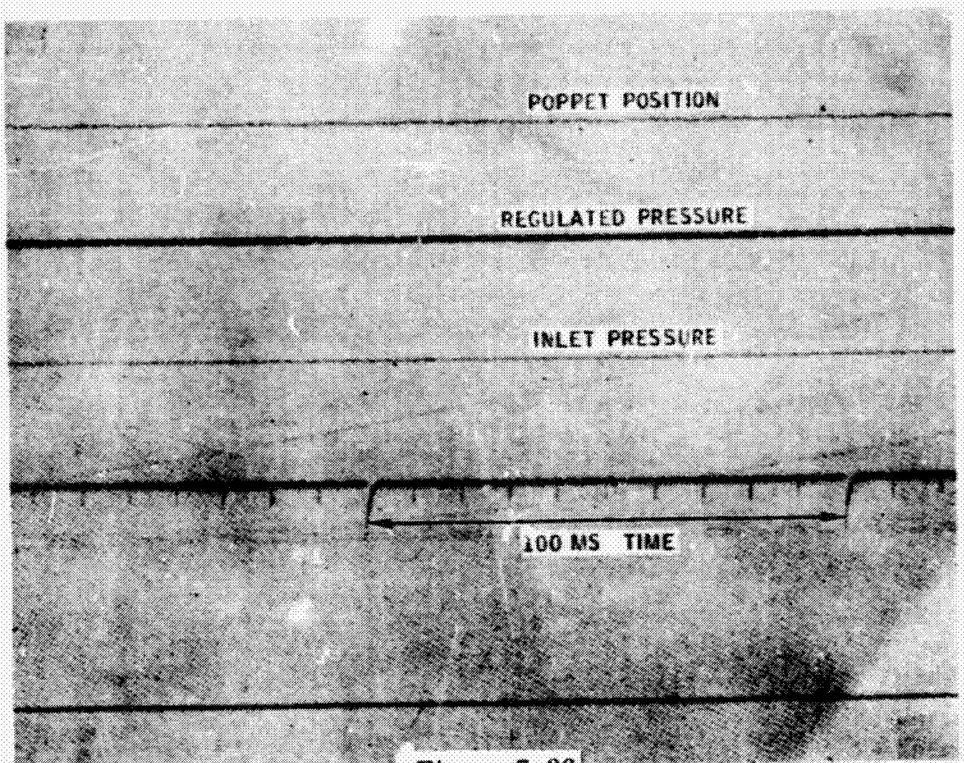


Figure 7-36

REGULATOR AND CHECK VALVE TEST

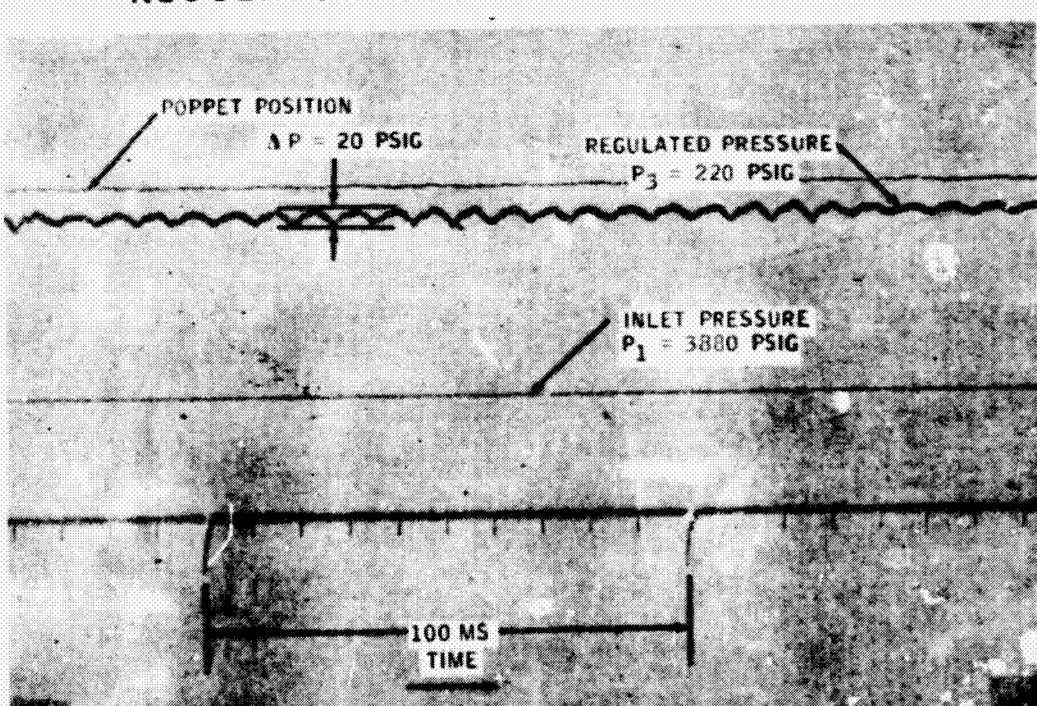


Figure 7-37

REGULATOR VIBRATION TEST SETUP SCHEMATIC

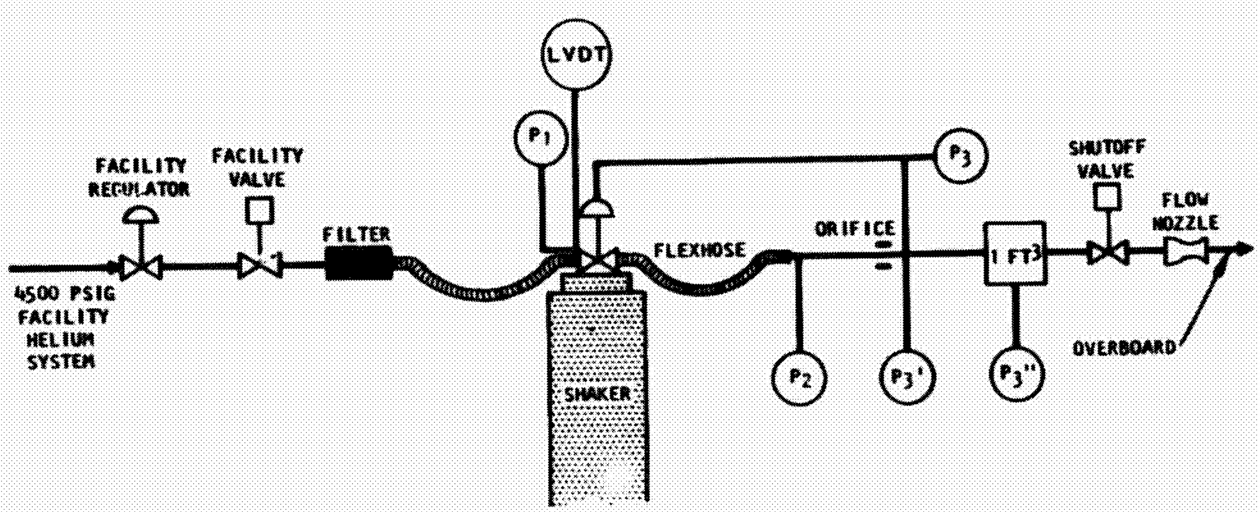


Figure 7-38

OMS VIBRATION SPECTRUM

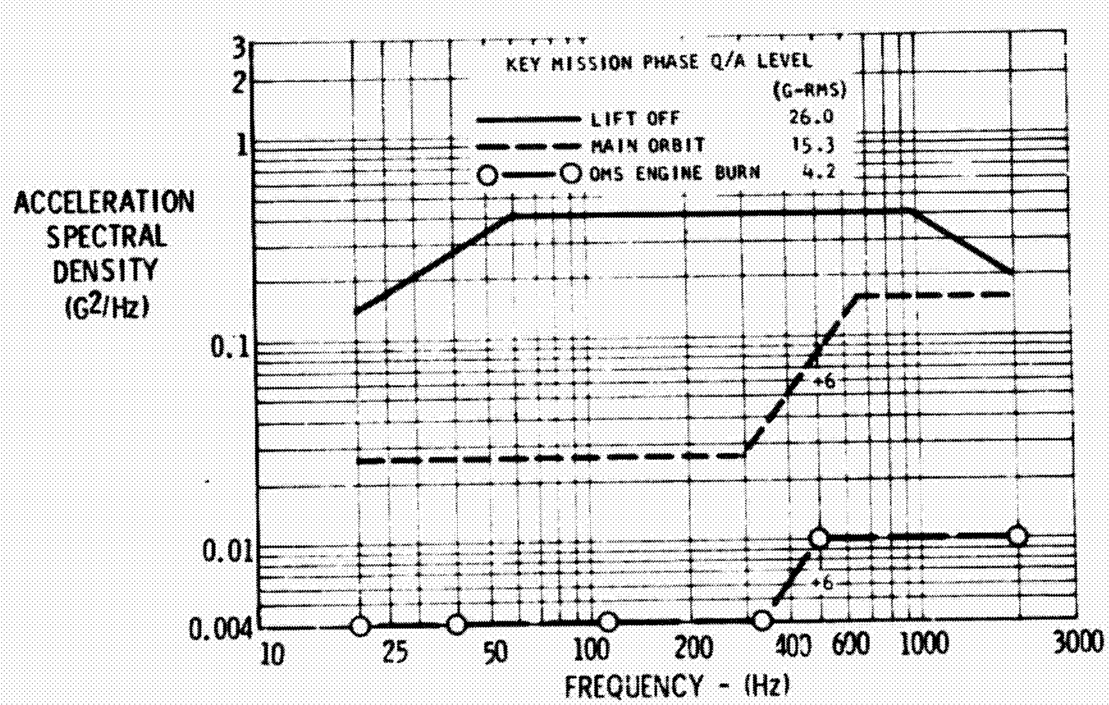


Figure 7-39

at the OMS engine burn level (4.2 grms) was conducted in each axis on October 15. A portion of the oscillograph trace of each run is shown in Figures 7-40 and 7-41. Poppet oscillation amplitude was greater than expected but acceptable (within the ±0.002-inch design goal)* One test at the Main Engine level (15.3 grms) was conducted in each axis on October 16. The oscillograph traces shown in Figures 7-42 and 7-43 indicate excessive poppet oscillation (±0.004 inch) in the X axis. Additional vibration tests were conducted in the critical X axis only to establish the effectiveness of the pneumatic damper at various damper pressures. Six random runs (all Main Orbital Engine level) and two sine sweeps were conducted on October 20. The results of the sine sweeps indicated a major resonance at 160 Hz and significant resonances at 50, 60, 90, 130, 135, 155, 240, 250, 300 and 480 Hz. The results of the damper tests, as reported in Section 7.1.5, were no better than for the tests on October 16. The specific design limits of the pneumatic damper were less than required for the vibration levels and the numerous resonances existent in the regulator.

The final vibration test conducted on October 20 consisted of the lift-off (nonoperating) level (26 grms) in the X axis. At the beginning of this test the shaft attaching the pneumatic damper to the poppet push rod broke. The actuator position indicated on the oscillograph traces of Figures 7-44, 7-45 and 7-46 are not the poppet or the actuator but the uncoupled shaft of the pneumatic damper with its amplitude determined by the impact imparted by the actuator movement.

The pneumatic damper was repaired and the orifice size changed to improve the damping. A performance run and seven vibration tests were conducted on November 20 following the check valve tests. The first five tests were conducted at a reduced Main Orbital Engine level of 5 grms and the last at full level. No significant improvement was noted.

7.2.4.2 Vibration With Mechanical Damper

The mechanical damper described in Section 7.1.6 was installed on the regulator and the regulator subjected to the Main Orbital Engine level in the X axis. These tests were sequenced to determine the damping force required to limit the poppet stroke to ±0.002 inch. Four runs were made on December 18. The first at three pounds friction force with no inlet pressure and the second with 3900 psi inlet pressure; the third and fourth runs were at 6.1 and 8.3 pounds friction. For these tests $P_{3''}$ was used to indicate regulated outlet pressure.

The results of these tests are presented in oscillograph traces shown in Figures 7-47, 7-48 and 7-49. A ±004-inch actuator movement at the three-pound friction level was reduced to ±0.003 inch with 6.1 pounds friction and to ±0.002 inch with 8.3 pounds friction.

A summary of vibration test conditions is presented in Table XV.

*Design goal of ±.002 inch poppet displacement for 15.3 grms level at resonant frequency.

REGULATOR OPERATION WITH PNEUMATIC DAMPER

OMS ENGINE BURN VIBRATION SPECTRUM - X AXIS

(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

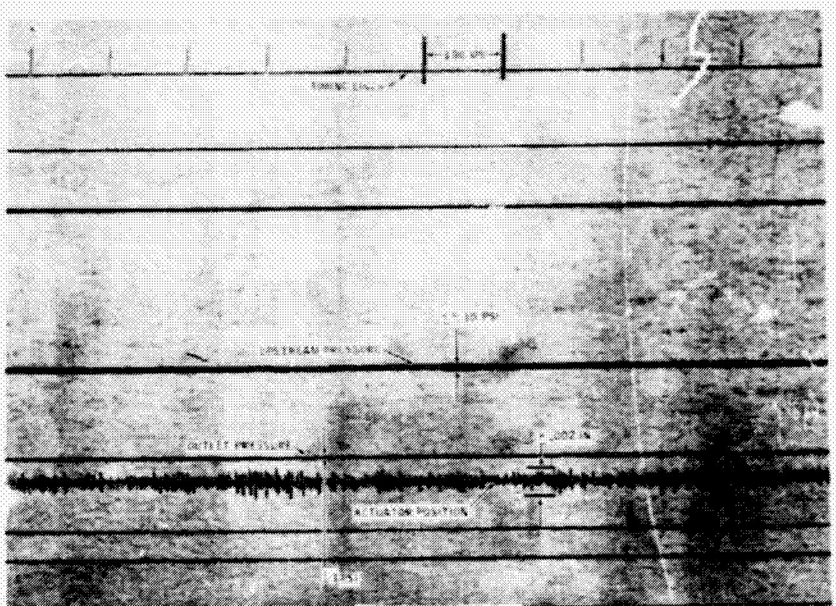


Figure 7-40

REGULATOR OPERATION WITH PNEUMATIC DAMPER

OMS ENGINE BURN VIBRATION - Y AXIS

(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

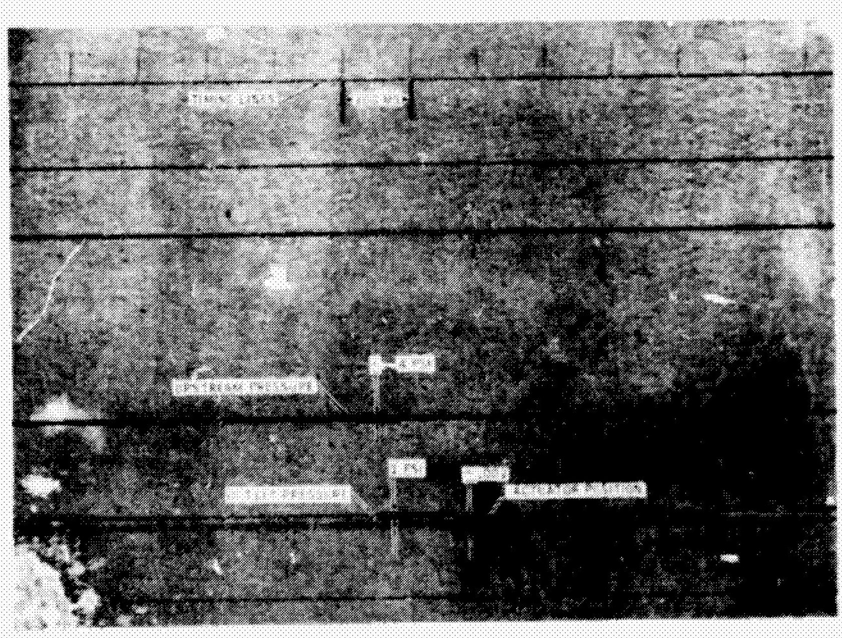


Figure 7-41

REGULATOR OPERATION WITH PNEUMATIC DAMPER

MAIN ENGINE BURN VIBRATION SPECTRUM - X AXIS
(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

Figure 7-42

REGULATOR OPERATION WITH PNEUMATIC DAMPER

MAIN ENGINE BURN SPECTRUM - Y AXIS
(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

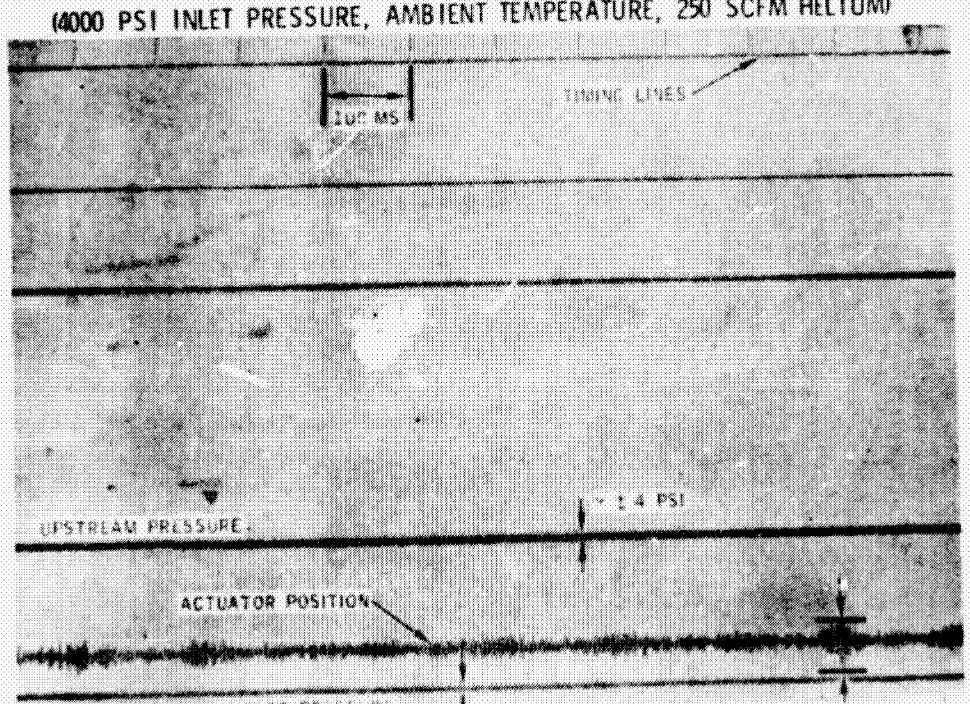


Figure 7-43

REGULATOR MOVEMENT WITH BROKEN DAMPER SHAFT

LIFT OFF VIBRATION SPECTRUM - X AXIS

(250 PSI LOCKUP PRESSURE, NO FLOW, START OF TEST)

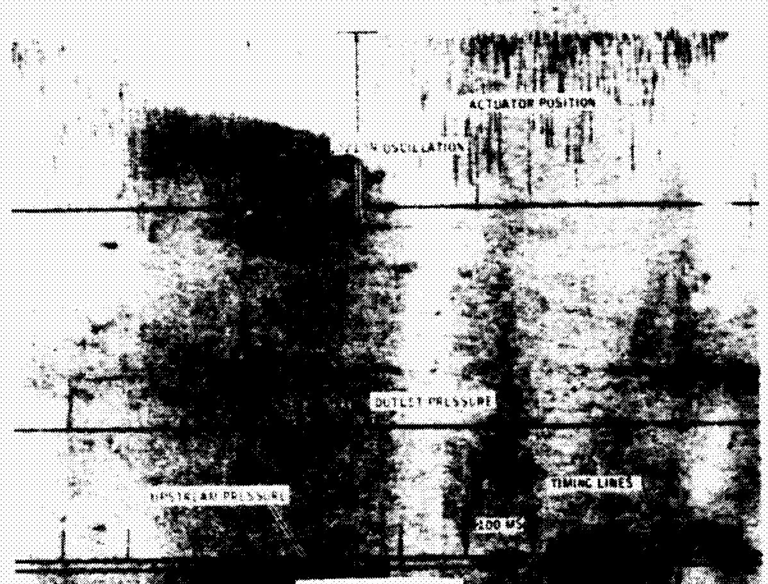


Figure 7-44

REGULATOR MOVEMENT WITH BROKEN DAMPER SHAFT

LIFT OFF VIBRATION SPECTRUM - X AXIS

(230 PSI OUTLET PRESSURE, PARTIALLY OPEN, NO FLOW)

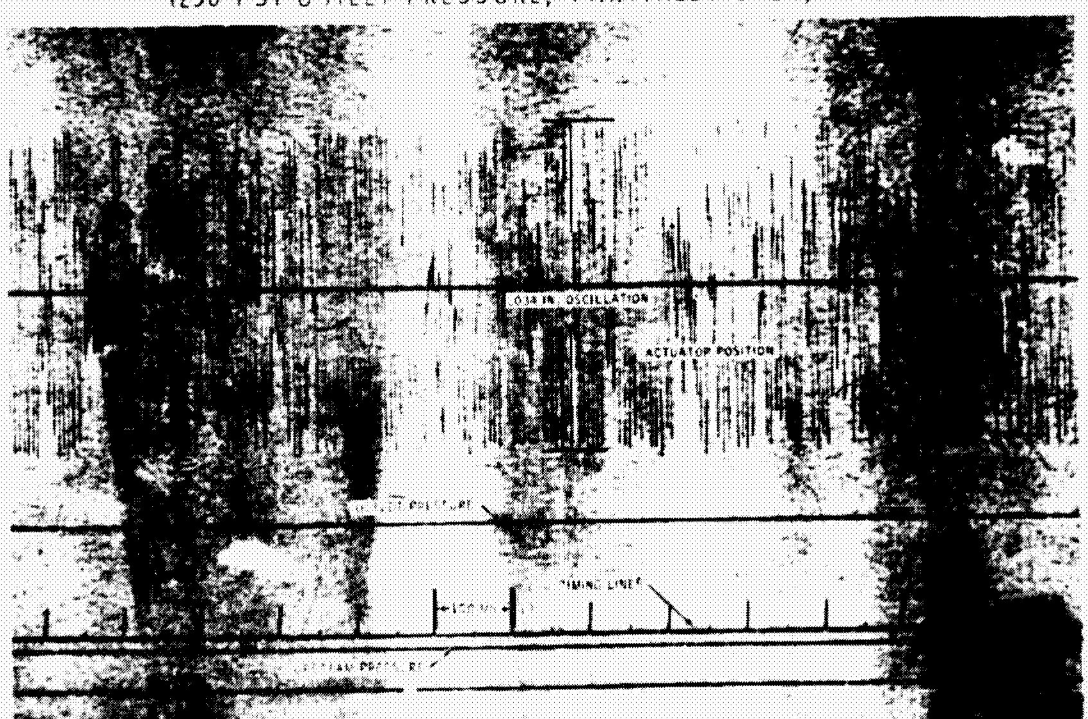


Figure 7-45

REGULATOR MOVEMENT WITH BROKEN DAMPER SHAFT

LIFT OFF VIBRATION SPECTRUM - X AXIS
(210 PSI DOWNSTREAM AND UPSTREAM,
REGULATOR FULLY OPEN, NO FLOW)

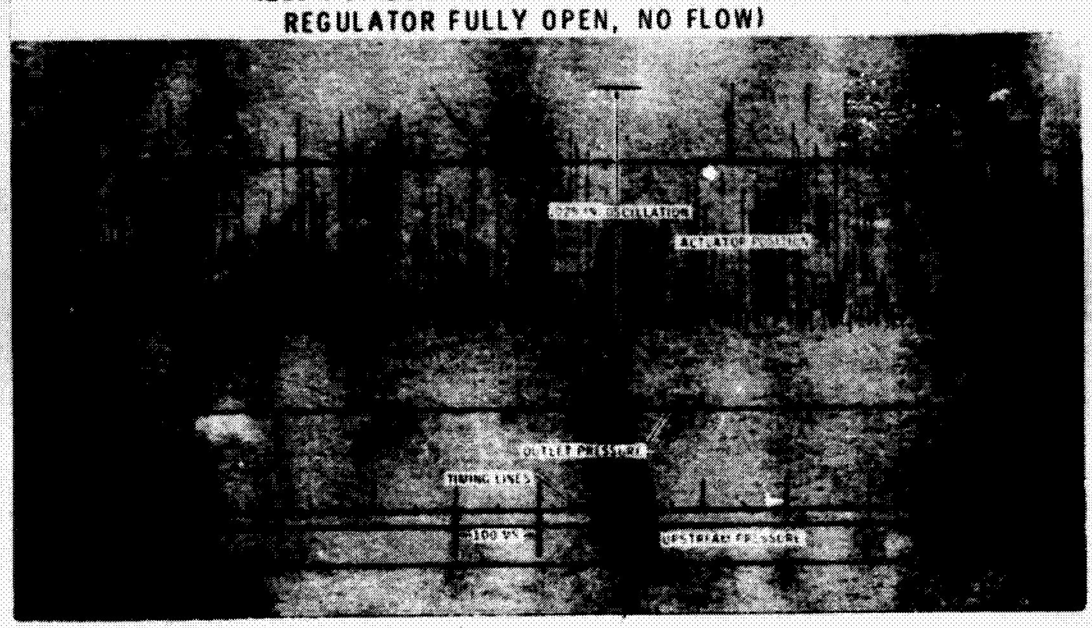


Figure 7-46

STABILITY CHARACTERISTICS OF REGULATOR WITH MECHANICAL DAMPER SET AT 3 LBS. FRICTION

MAIN ENGINE BURN SPECTRUM - X AXIS
(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

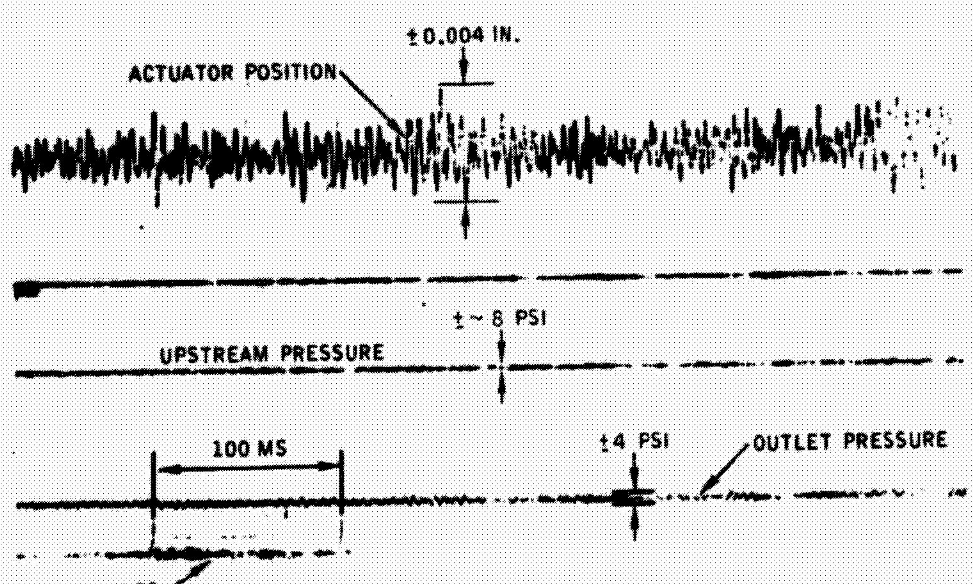


Figure 7-47

STABILITY CHARACTERISTICS OF REGULATOR WITH MECHANICAL DAMPER SET AT 6.1 LBS. FRICTION

MAIN ENGINE BURN VIBRATION SPECTRUM, X AXIS
(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

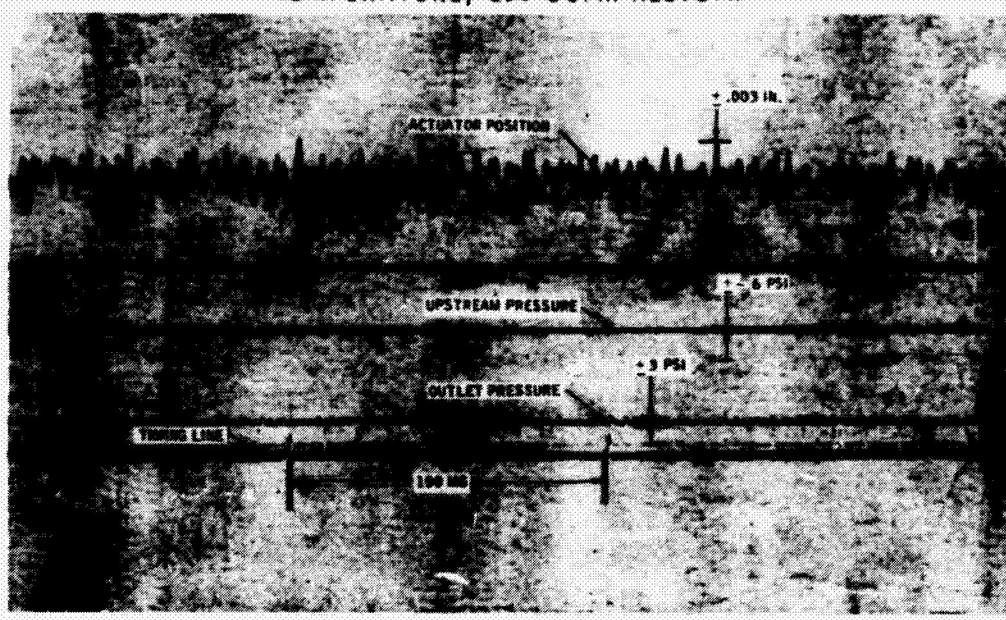


Figure 7-48

STABILITY CHARACTERISTICS OF REGULATOR WITH MECHANICAL DAMPER SET AT 8.3 LBS. FRICTION

MAIN ENGINE BURN SPECTRUM - X AXIS
(4000 PSI INLET PRESSURE, AMBIENT TEMPERATURE, 250 SCFM HELIUM)

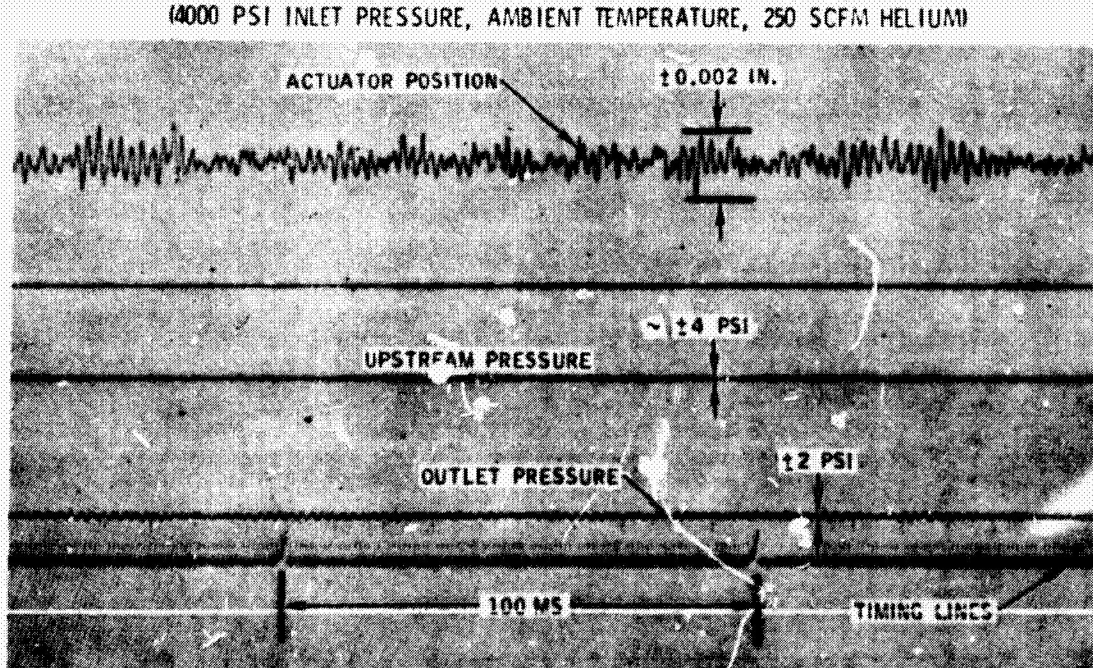


Figure 7-49

TABLE 7-XV

VIBRATION TEST SUMMARY

Run No.	Date	G_{rms} Level	Axis	P_1 (psi)	P_3 (psi)	Stroke (inch)	Remarks
1	10/15	4.2	X	4010	246 ±1	0.002 ±0.002	
2	10/15	4.2	X	3900	245 ±0.5	0.002 ±0.001	
3	10/16	15.3	Y	3950	240 ±0.5	0.003 ±0.002	
4	10/16		X	4000	240 ±1	0.003 ±0.004	
5	10/20			3920	250	-	
6				2950	250	-	
7				1980	260	-	
8				680	-	-	
9				400	-	-	
10				3910	-	-	
11		Sine		3960	240	±0.0005	All resonances as
12		Sine		3940	240	to ±0.0025	noted below
13		26		400-230	250-230	-	Damper shaft broke
14	11/20	-		3650	214.7	-	Preflow
15		5		3520	230 ±5	0.003 ±0.006	
16				3500	-	-	Lost data
17				3400	225 ±5	0.005 ±0.006	
18				3350	215 ±5	0.004 ±0.005	
19				3260	215 ±5	0.004 ±0.005	w/o 0.005 wire. Effective
20		15.3		3200	220 ±10	-	orifice dia. = .013
21				0	-	0.021 ±0	No pressure
22	12/18			0	10	0.0005	Resonance @ 60 Hz
23				3920	212^{+25}_{-5}	0.008 ±4	Resonance @ 400 & 200 Hz
24				3900	214 ±5	0.005 ±3	Resonance @ 466 & 180 Hz
25				3900	207 ±5	0.004 ±2	Resonance @ 500 & 450 Hz

7.2.5 Leakage and Life Cycle

Throughout performance and vibration testing of the regulator seat leakage was recorded. Following the successful completion of the proof pressure and external leakage tests of Paragraphs A-14 and A-15 of MTP 0213, Appendix A, the internal leakage test of Paragraph B-3 was conducted frequently, as indicated in Table 7-XVI.

7.2.5.1 Performance and Vibration Effects

The regulator configuration displayed significant seat leakage during performance testing due to the tilted "flexible" seat structure described in Section 7.1.3. Even with this anomaly, the leakage (sealing) characteristics improved during vibration as shown in Figure 7-50.

Some sealing improvement was noted after the seat was lapped and the bumper height lowered.

Following the performance and vibration tests previously described, the regulator was disassembled and a new seat (no deflecting housing and nonbumper configuration) was installed prior to life cycle testing.

7.2.5.2 Life Cycle Effects

Life cycle testing was conducted in accordance with MTP 0213, Appendix B-12, during the period of February 20-26, 1974. A total of 15,000 actuations was accumulated. Internal leakage tests were conducted periodically with indicated leakage rates in the range of 0-12 scch throughout the cycle life of 100 to 15,000 actuations.

To expedite the life cycle test, the test setup (Figure B-2 of MTP 0213) was modified to cause the regulator to complete a full open and full close cycle each second at inlet pressures of 300-400 psig. To accomplish this, the downstream facility throttle valve was cycled 0.5 second ON, 0.5 second OFF, with a nozzle flowrate of 10 cfm GN_2. The ullage system was reduced from 1.3 ft^3 to 15 in^3 and then to 45 in^3.

The leakage test setup consisted of disconnecting the regulator from the Figure B-2 setup and capping the outlet (P_2), holding a constant 300 psig GH_e to the actuator (P_3) and applying GH_e to the inlet (P_1) in amounts of 400, 800, 2000, and 4000 psig for five-minute intervals each. The outlet pressure was monitored for internal leakage by attaching a line to the P_2 pressure port and placing it at the base of a submerged water-filled graduated cylinder (50 ml). The sensitivity of this setup is determined by the readability of the graduated cylinder of 0.25 ml during the five-minute period (0.25 x 12 = 3 scch).

TABLE 7-XVI

INTERNAL LEAKAGE TEST

Test No.	Date	Intermediate Test-Type	Test Pressure psi/psi/psi...	GH_e Leakage Rate Scch/Scch/Scch...
1	8/29	Flow force	400/800/2000/4000	700/2220/2480/720,000
2	9/17	Flow Limiter	400/800/2000	140/480/80 (dust cap on P_2 port)
3	9/29	After 150°F	400/800/2000	1940/7280/14,460
4	10/15	After Slam	400/800/2000	1080/3000/2040
5	10/15	After OMS Vib (X)	400/800/2000	720/1720/360
6	10/16	After M. E. Vib (Y)	400/800/2000	400/1380/300
7	10/16	After M. E. Vib (X)	400/800/2000	400/1160/500
8	10/20	Prior to Liftoff (X)	400/800/2000	400/980/260
9	10/24	After Liftoff (X)	400/800/2000	1840/2400/2400
10	11/1	After Lapping	400/800/2000/4000	200/560/520/ 10,000
11	11/20	After All Vib	470/710/2000	200/240/440
12	12/18	Prior to Vib	400/800/2000	40/200/80
	2/18	New Seat Installed		
13	2/20	After 1 cycle	800/2000/3000/4000	0/30/96/156
14	2/20	After 10 cycles	3000/4000	60/60
15	2/20	After 20 cycles	4000	48
16	2/21	After 100 cycles	400/500/2000/4000	0/0/0/0
17	2/21	After 500 cycles	2000/4000	0/12
18	2/22	After 1000 cycles	400/800/2000/4000	3/0/3/10
19	2/22	After 2500 cycles	400/800/2000/4000	0/1/5/10
20	2/22	After 5000 cycles	400/800/2000/4000	0/0/2.4/7.2
21	2/22	After 10,000 cycles	400/800/2000/4000	18/12/12/12
22	2/26	After 15,000 cycles	400/800/2000/4000	0/0/10/12

REGULATOR LEAKAGE CHARACTERISTICS DURING VIBRATION TEST SEQUENCE

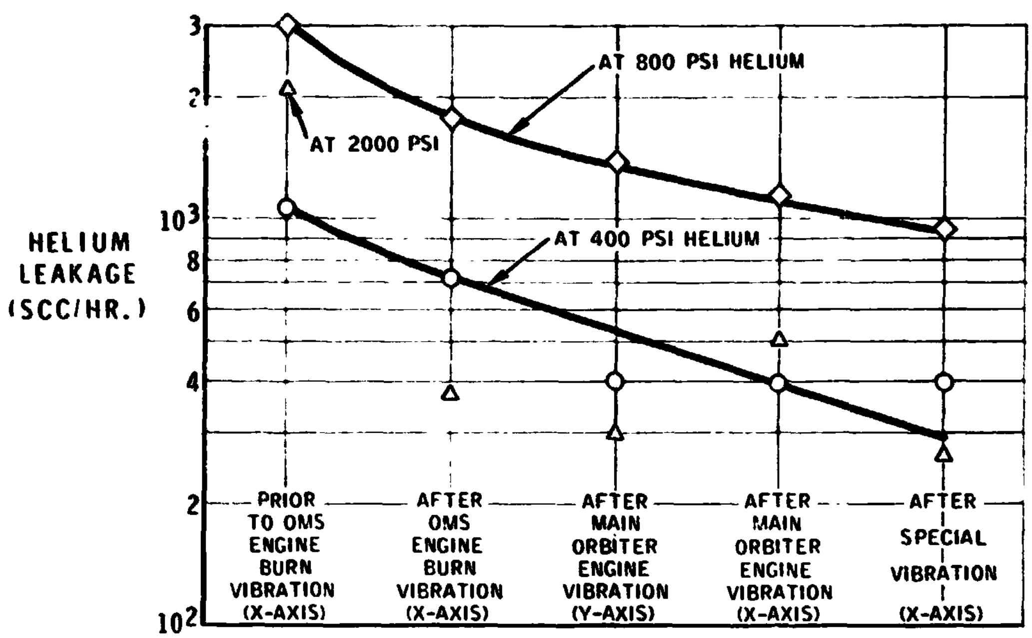
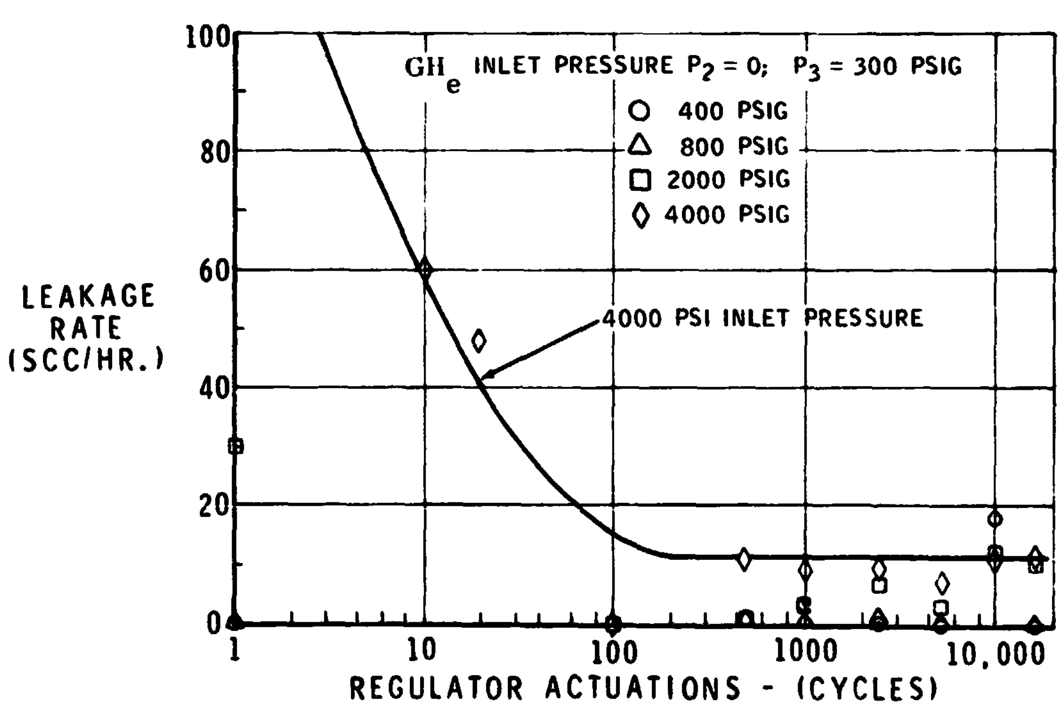


Figure 7-50

INTERNAL LEAKAGE DURING LIFE CYCLE TEST OF PROTOTYPE REGULATOR



C-3

Figure 7-51

Leakage tests were conducted after 1, 10, 20, 100, 500, 1000, 2500, 5000, 10,000, and 15,000 actuations. The results are presented in Figure 7-51.

Life cycle test instrumentation consisted of visual monitored pressure gages and gas temperature and visicorder record of throttle valve voltage and regulator stroke. Typical stroke traces for 1 cps actuations for 300 and 400 psig inlet pressures for the 45 $in.^3$ ullage setup are shown in Figures 7-52 and 7-53.

Observations made during setup modification of ullage volumes reflect the criteria for selection of 45 $in.^3$ as an acceptable ullage for life cycle testing. The .5 second flow demand on 1.3 ft^3 ullage required a relatively slow regulator response, resulting in a reduced stroke requirement. Also, with a 15 $in.^3$ ullage, the closing response of the regulator lagged the system, causing an overshoot in lockup pressure. The 45 $in.^3$ ullage was large enough to minimize overshoot and allow the regulator to obtain steady state open and closed conditions.

Ullage	P_3 (psig) Sensing Pressure Range	Comments
1.3 ft^3	228-233	Partial stroke only
15 $in.^3$	212-235	7 psi overshoot
45 $in.^3$	212-229	1 psi overshoot

7.3 NASA-JSC TEST PLAN

Marquardt Test Plan 0213, Revision A, was prepared to define the test procedures to be followed by NASA-JSC when conducting the propellant compatibility and extended life tests on the prototype regulator at the NASA facility. The flowrates and test conditions are the same as contained in the test report so that comparative data may be obtained. MTP 0213, Revision A, is attached as Appendix C to this report.

PULSING LIFE CYCLE - ACTUATOR POSITION

THROTTLE VALVE CYCLE 0.5 SEC OPEN, 0.5 SEC CLOSED

45 IN^3 ULLAGE VOLUME, 214 PSIG REGULATED PRESSURE

310 PSIG INLET PRESSURE

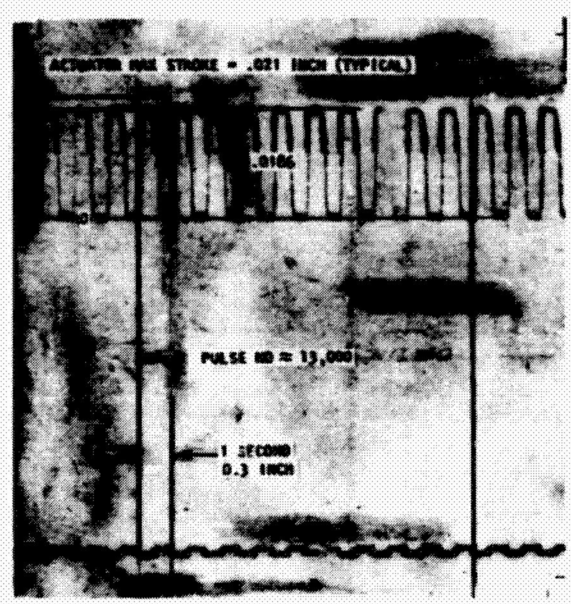


Figure 7-52

400 PSIG INLET PRESSURE

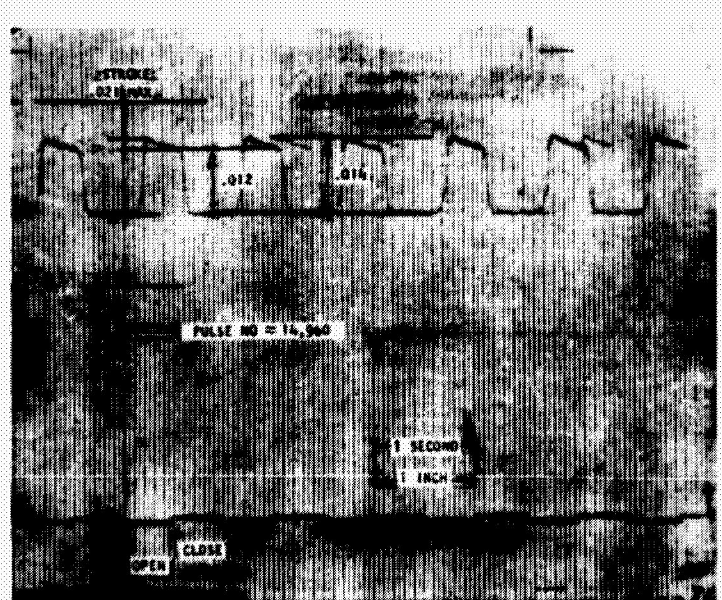


Figure 7-53

8.0 CONCLUSIONS AND RECOMMENDATIONS

The Space Shuttle OMS Helium Regulator Design and Development Program successfully demonstrated a prototype pressure regulator during flow bench tests over the required pressure, temperature and flow rate ranges, and during vibration and life cycle tests. In generating pressure regulator technology to permit this demonstration, the subject program evaluated existing pressure regulator concept, identified their deficiencies (small clearances and contamination sensitivity of piloted designs), and generated new concepts which eliminated these deficiencies and which hold a promise of satisfying the multi-mission, long life space shuttle requirements. The prototype pressure regulator demonstrated during this program employed a single-stage regulator concept which is believed to be the simplest and therefore the most reliable and also the lowest cost regulator concept applicable.

During the tradeoff portion of this program it was evident that although the most reliable and lowest cost regulator concept had been selected for fabrication and test evaluation, another single-stage regulator concept featuring a lever arm between the valve section and the actuator section promised to offer significantly lower weight. With the more recent heavy emphasis on minimum weight in the space shuttle program it now appears that this slightly more complex lever arm regulator may be a more desirable approach for the space shuttle orbital manuevering system. Consequently, it is recommended that a flight-weight, lever arm, single stage regulator be developed during a follow on program. Development of this lever arm regulator also has a side benefit in that the amount of damping required to meet the stringent regulator actuator movement criteria established during vibration is less than that required for the regulator concept developed during this program, particularly when the lever arm regulator is arranged in such a manner that the poppet motion is in the opposite direction to the actuator motion.

The amount of damping required to limit actuator travel to 0.002 inches during main shuttle engine random vibration was experimentally defined during this program by utilizing a mechanical damper. Also, early during this program three types of damping were compared: these included pneumatic damping, mechanical damping, and hydraulic damping. Of the three forms of damping pneumatic damping was selected originally because it offered a wider temperature operating range than hydraulic damping and because it did not feature the inherent static friction of mechanical damping. Based on the experimental data, however, it appears that pneumatic damping is not quite adequate or would at least require a significant increase in size of the pneumatic damper. Furthermore it has become evident that the minimum temperature requirement which was of concern with respect to the hydraulic damper is really only a transient condition which applies to the helium gas and not to the regulator environment. Consequently, it now appears that a hydraulic damper is quite applicable to the space shuttle OMS requirements and that indeed it is probably the best technical solution. Therefore it is recommended that the flightweight regulator be developed with a hydraulic damper.

During the fabrication of the prototype pressure regulator it became evident that working with Inconel 718 is a rather expensive proposition. The material is fairly difficult to machine since it work-hardens quite easily, and to take advantage of the high mechanical properties requires a rather long and therefore expensive heat treat cycle. These disadvantages suggest that a search for a equally propellant compatible but less expensive materials approach should be pursued. Based on the materials compatibility investigation performed during Task 1 of this program it appears that the judicious selection of materials such as titanium (6Al -4V), Armco 21-6-9, and type 304L stainless steel offer the greatest promise.

Finally, based on the experimental data verification of the various analysis programs utilized in sizing the pressure regulators it appears that a reoptimization would be timely to further improve the weight characteristics of the pressure regulator. This reoptimization should be combined with a reevaluation of the space shuttle OMS pressure regulator requirements since current indications are that the maximum inlet pressure requirement and maximum flow rate requirements will both increase. It is believed that the understanding and experience gained in the sizing of single-stage pressure regulators for flow requirements of this magnitude will significantly enhance the ability to achieve minimum pressure regulator weight and will thereby assure the utilization of a much more reliable single-stage pressure regulator concept for the space shuttle OMS as well as RCS systems.

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APPENDIX A

DEVELOPMENT OF EQUATIONS FOR FLOW FORCES ON POPPET VALVES

APL PROGRAM POP

DEFINITIONS

PRESSURE:

P	=	psi		
P_{TP}	=	PTP	=	supply pressure = variable
P_1	=	P1	=	pressure @ choke or min. area
P_{T3}	=	PT3	=	fixed outlet pressure to be delivered by valve
P_B	=	PB	=	bellows pressure (see Step V)
PBAR1			=	average pressure acting on $\frac{\pi}{4}\left(D_L^2 - D_s^2\right)$
P_2	=	P2	=	pressure at Station 2 $\equiv$ PT3
PBAR2			=	average pressure acting on $\frac{\pi}{4}\left(D_L^2 - D_2^2\right) = \frac{P1 + PT3}{2}$
P_o	=	PO	=	average pressure acting on $\frac{\pi}{4}\left(D_P^2 - D_L^2\right)$

AREAS:

A_P	=	AP	=	$\frac{\pi}{4} D_P^2$
A_B	=	AB	=	$\frac{\pi}{4} D_B^2$
A_L	=	AL	=	$\frac{\pi}{4} D_L^2$
A_1	=	A1	=	$\pi D_L S$
A_2	=	A2	=	$\frac{\pi}{4}\left(D_2^2 - D_S^2\right)$
A_S	=	AS	=	$\frac{\pi}{4} D_S^2$
K_3	=	K3	=	$\dfrac{\frac{\pi}{8}\left(D_L^2 - D_2^2\right)}{\frac{\pi}{4}\left(D_L^2 - D_S^2\right)}$ (Used in Step IIIA)

INPUTS WILL CONSIST OF:

Geometry: (See Figure)

D_P

D_B

D_L & D_L^1

D_2

D_S

D_F

t

Fluid Properties: γ, R, TTP (°R)

Flow Conditions:

$\dot{W}$ = WDOT = #/Sec Flow Rate

PTP = supply pressure

PT3 = outlet pressure

Force Equation

$$F = PTP\left(A_p - A_B\right) + PB\ (AB) - PBAR1\left(A_L - A_S\right) - PT3\ (AS) - PO\ (AP - AL)$$

F = + Net Closing Force

= - Net Opening Force

Momentum Equation

$$PBAR1 = \frac{P2 \cdot A2\left(1 + \gamma M_2^{\,2}\right) + \frac{\pi}{4}\ PBAR2\left(D_L^{\,2} - D_2^{\,2}\right)}{A_L - A_S}$$

Flow Equations

$$\frac{p}{P_T} = \frac{P}{PT} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{-\gamma}{\gamma - 1}}; \quad \gamma = \text{input} = 1.667 \text{ for Helium}$$

$$\frac{f}{p} = 1 + \gamma M^2$$

$$\dot{m} = mDOT = g\sqrt{\frac{\gamma}{R}}\ \ M\left[1 + \frac{\gamma - 1}{2} M^2\right]^{1/2}$$

g = 32.174

R = 12417.877 for Helium

$$k = \frac{p}{P_T}\ \dot{m}$$

Continuity

$$\dot{w} = wDOT = \frac{\dot{m}_1\ p_1\ A_1}{\sqrt{TTP}} = \frac{\dot{m}_2\ p_2\ A_2}{\sqrt{TTP}}$$

PROCEDURE

STEP

I. Assume min area A_1 is choked, solve for A_1, S, $\frac{A_1}{A_2}$

For $M_1 = 1.0, \; \dot{w} > 0$

$$A_1 = \frac{\dot{w}\sqrt{TTP}}{P_1 \; \dot{m}_1} \quad IN^2 \quad = \frac{\dot{w}\sqrt{TTP}}{PTP \cdot k_1}$$

$$S = \frac{A_1}{\pi D_L} \quad INCH$$

$$\frac{A_1}{A_2} = \frac{A_1}{\frac{\pi}{4}\left(D_L^{\;2} - D_S^{\;2}\right)}$$

$$k_1 = f\,(M_1)$$

NOTE:
For $\dot{w} \equiv o$, set
$p_1 \equiv PT3$
$M_1 \equiv O$
$A_1 \equiv O$

II. Solve for value of M_2 and related parameters

$$p_2 \equiv PT3 = p_1 \frac{\dot{m}_1}{\dot{m}_2} \left(\frac{A_1}{A_2}\right)$$

$$\therefore \; \dot{m}_2 = \frac{p_1 \dot{m}_1}{PT3} \left(\frac{A_1}{A_2}\right)$$

$$M_2 = f\left(\dot{M}_2\right) \qquad \text{(NOTE : } M_2 \gtreqless 1 \text{ for Step III)}$$

$$\left(\frac{f}{p}\right)_2 = 1 + \gamma M_2^{\;2} = f\,(M_2)$$

$$\left(\frac{p}{PT}\right)_2 = f\,(M_2) \quad ; \quad k_2 = \left(\frac{p}{PT}\right)_2 \quad \dot{m}_2 = f\,(M_2)$$

$$PT2 = \frac{p2}{\left(\frac{p}{PT}\right)_2} \quad \text{or} = PTP \, \frac{k_1}{k_2} \left(\frac{A_1}{A_2}\right)$$

$$K2 = \frac{P_2 A_2 \left(1 + \gamma M_2^{\;2}\right)}{A_L - A_S} \qquad \text{(Used in Step III A.)}$$

STEP

III. Solve for PBAR1

$$PBAR1 = \frac{P_2 A_2 (1 + \gamma M_2^2) + \frac{\pi}{4} PBAR2 \left(D_L^2 - D_2^2\right)}{A_L - A_S}$$

where:

$$PBAR2 = \frac{P_1 + PT_3}{2}$$

Check for condition $PBAR1 \leq P_1$

If $PBAR1 \leq P_1$ Proceed to Step IV

If $PBAR1 > P_1$ Go to Step IIIA; If $M_2 > 1.0$ all output should indicate notation: "Questionable Solution"; omit this notation if $M_2 \leq 1.0$

IIIA. If $PBAR1 > P_1$, then min area is not choked; condition to be met is $PBAR1 \equiv P_1$ for unchoked inlet.

Since the flow rate is input, and A_2 is known, conditions at station 2 will not change for unchoked inlet & $\therefore$ p2, $(f/p)_2$ are applicable from Step II.

$\therefore$

$$PBAR1 = P1 = \frac{P2A_2 (1 + \gamma M_2^2) + PT3 \times \frac{\pi}{8}\left(D_L^2 - D_2^2\right)}{\left(A_L - A_S\right)\left[1 - \frac{\pi}{8}\left(\frac{D_L^2 - D_2^2}{A_L - A_S}\right)\right]}$$

$$= \frac{K2 + K3 \times PT3}{1 - K3}$$

Then $M_1 = f\left(\frac{P1}{PTP}\right)$

$k_1 \quad f(M_1)$

$\dot{m}_1 = f(M_1)$

STEP

IIIB. Solve for A_1, S, $\frac{A_1}{A_2}$ for M_1 defined in Step III A

$$A_1 = \frac{\dot{w}\sqrt{TTP}}{P_1 \cdot \dot{m}_1}$$

$$S = \frac{A_1}{\pi D_L}$$

$$\frac{A_1}{A_2} = \frac{A_1}{\frac{\pi}{4}\left(D_2^{\,2} - D_S^{\,2}\right)}$$

IV. Solve for PO

$$AO = \text{Overhang Area} = \frac{\pi}{4}\left(D_P^{\,2} - D_L^{\,2}\right) = AO1 + AO2$$

$$A_{O1P} = \frac{\pi}{4}\left(D_{LP}^{\,2} - D_6^{\,2}\right)$$

$$AO1 = \frac{\pi}{4}\left(D_F^{\,2} - D_{LP}^{\,2}\right); \quad \Delta AO1 = \frac{AO1}{3} = 3 \text{ Equal Areas}$$

$$AO2 = \frac{\pi}{4}\left(D_P^{\,2} - D_F^{\,2}\right); \quad \Delta AO2 = \frac{AO2}{2} = 2 \text{ Equal Areas}$$

Determine Diameters Starting at D_L, Ending at D_P

$$D_n = \sqrt{\frac{4A_n}{\pi}} \qquad \text{where} \quad A_n = A_L + \sum_{1}^{n} \Delta AO$$

$$D_1 = D_{LP} \qquad A_1 = A_L$$

$$D_2 \qquad A_2 = A_L + \Delta AO1 = A_1 + \Delta AO1$$

$$D_3 \qquad A_3 = A_2 + \Delta AO1$$

$$D_4 = D_F \qquad A_4 = A_3 + \Delta AO1$$

$$D_5 \qquad A_5 = A_4 + \Delta AO2$$

$$D_6 = D_P \qquad A_6 = A_5 + \Delta AO2 = A_P = \frac{\pi}{4} (D_P)^2$$

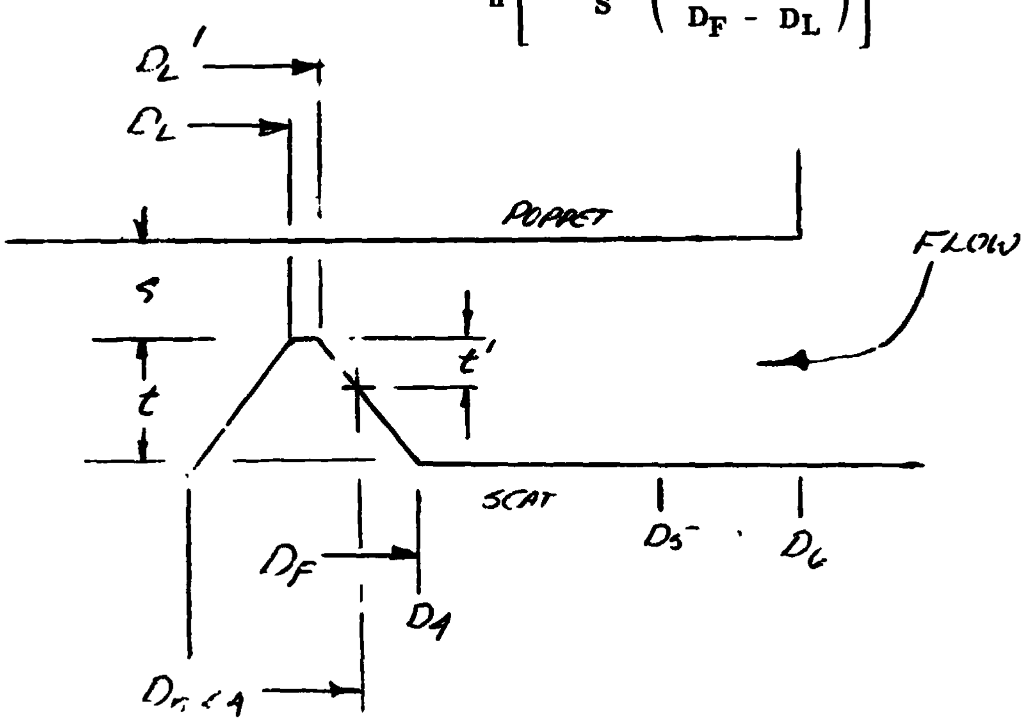
From Continuity $\qquad k_n = k_1 \frac{A1}{A_n} \qquad P_{T_n} = P_{T_1} = PTP$

k_1 from Step I or III A

From Geometry

For $\quad n \geq 4 \quad \frac{A_1}{A_n} = \frac{D_{LP}}{D_n\,(1 + t/S)}$

For $\quad n \leq 3 \quad \frac{A_1}{A_n} = \frac{D_L}{D_n \left[1 + \frac{t}{S} \left(\frac{D_n - D_L}{D_F - D_L} \right) \right]}$



For $D_n < D_F$

$$A_{flow} = \pi D_n (S + t')$$

$$t' = \left(\frac{D_n - D_L}{D_F - D_L}\right) t \; ; \quad D_n \leq D_F$$

$$A_{flow} = \pi D_n \left[S + t \left(\frac{D_n - D_L}{D_F - D_L}\right)\right]$$

$$A_1 = \pi D_L S$$

$$\frac{A_1}{A_n} = \frac{\pi D_L S}{\pi D_n \left[S + t \left(\frac{D_n - D_L}{D_F - D_L}\right)\right]} = \frac{D_L}{D_n \left[1 + \frac{t}{s} \left(\frac{D_n - D_L}{D_F - D_L}\right)\right]}$$

$$\therefore \quad k_n = k_1 \frac{D_{LP}}{D_n \left(1 + \frac{t}{S}\right)} \quad \text{for} \quad n \geq 4$$

$$= k_1 \frac{D_{LP}}{D_n \left[1 + \frac{t}{S} \left(\frac{D_n - D_{LP}}{D_F - D_{LP}}\right)\right]} \quad \text{for} \quad n \leq 3$$

$M_n = f\ (k_n)$ Subsonic Solution for each Diameter $D_n 1 \to 6$

$\left(\frac{p}{PT}\right)_n = f\ (M_n)$ " " " "

$p = \left(\frac{p}{P_T}\right)_n \times PTP$ " " " "

$pAVG_n = \frac{p_n + p_{n+1}}{2}$ for each successive pair of P_n

$\Delta F_n = pAVG_n \times \Delta AO$ " " " " " "

$$FAO = \sum_{n=1}^{6} \Delta F_n$$

$$PO = \frac{FAO}{AO}$$

STEP

V. Option for value of PB

Option (1) Input arbitrary value of PB

(2) PB $\equiv$ PT2 from Step II

VI. Solve for Net Force, Equation for F

$$F = PTP\,(A_p - A_B) + PB\,(A_B) - PBAR1\,(A_L - A_S) - PT3\,(A_S) - PO\,(A_p - A_L)$$

VII. Solve for Net Force, Equation for F_o, i.e. for $\dot{w} \equiv O$;

For this case PBAR1 $\equiv$ PT3

PO $\equiv$ PTP

Determine $\frac{F}{Fo}$ F from Step VI
Fo from Step VII

VIII. Solve for Net Force, Unbalanced Poppet, Fu

For this case PB $\equiv$ PTP

PO Value of Step IV

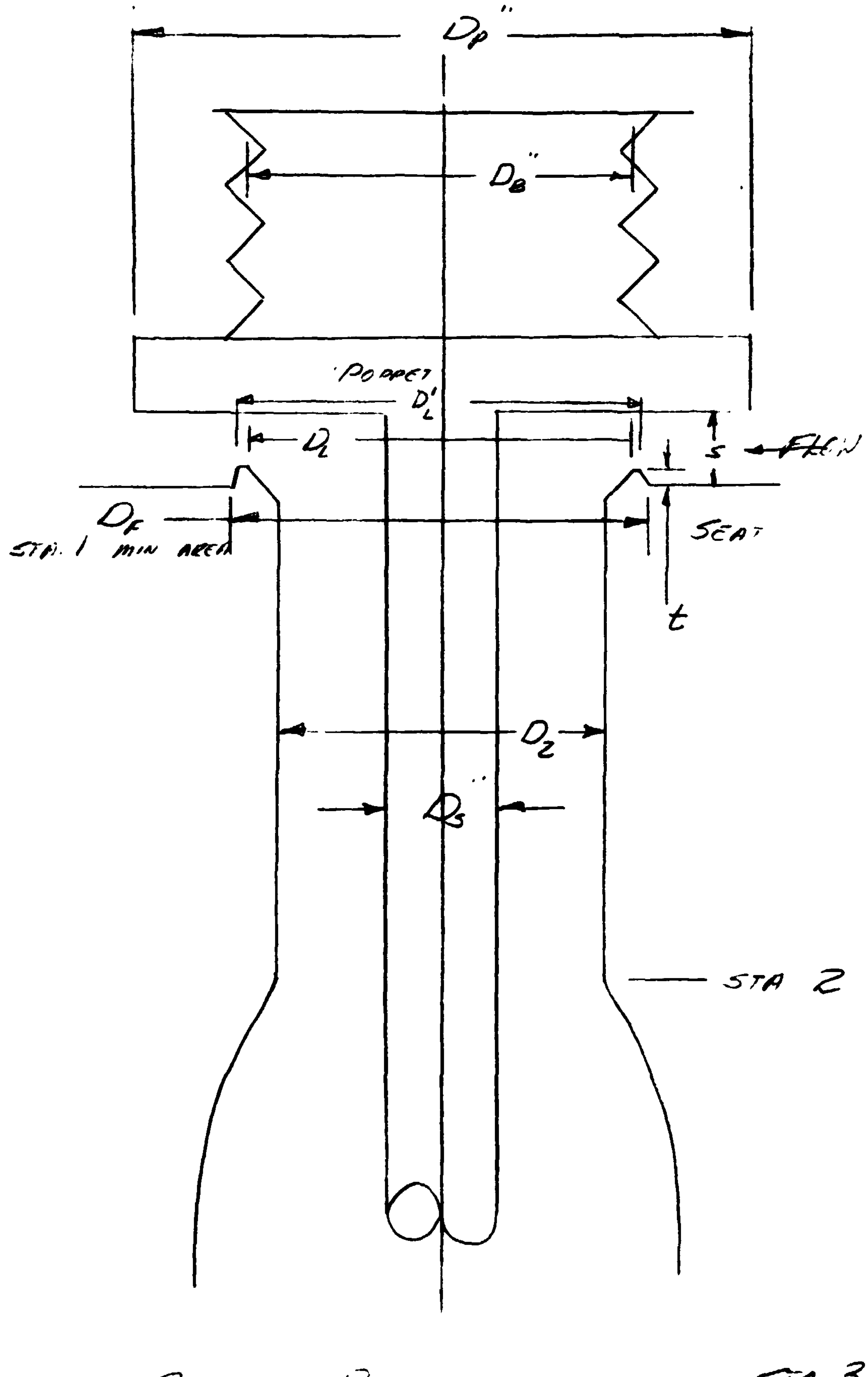
IX. Print Out Solution; Include "Questionable Solution" if appropriate from Step III.

PTP, PT3, PTP/PT3, PI, PBAR1, PBAR2, PB, PO, PT2

$\dot{w}$, TTP, M_1, M_2

F, Fo, F/Fo, Fu

S, A_1/A_2, A_1, A_B



BALANCED POPPET
NOMENCLATURE SCHEMATIC

Figure A-1

APPENDIX B

VERIFICATION TEST
SHUTTLE OMS HELIUM REGULATOR PROTOTYPE
MARQUARDT TEST PLAN 0213

VERIFICATION TEST - SHUTTLE OMS HELIUM REGULATOR PROTOTYPE

1.0 OBJECTIVE

The objective of this test program is to document performance characteristics of the prototype regulator relative to the design requirements of Section 3.0 of Exhibit A Statement of Work of Contract NAS 9-12992. Since this is the prototype design, the test program will allow wide latitude in the test requirements to permit the acquisition of a maximum of development data to refine the design, as required.

2.0 SCOPE

The scope of this test program shall be to document:

2.1 Regulator component design values.

2.2 Performance characteristics over the usage pressure, temperature, flow range, and vibration specifications.

2.3 Contamination tolerance of the poppet/seat interface.

2.4 Short term propellant compatibility.

3.0 DESCRIPTION OF TEST ITEM

The test program, defined herein, is applicable to the Prototype Shuttle OMS Helium Pressure Regulator, which is defined by Marquardt Drawing X29200. This regulator is a single stage, pressure balanced poppet, hard seat, all metal construction pressure regulator. While under test, the unit shall be evaluated as two regulators in series with a flow limiter located at the upstream unit inlet. The downstream regulator shall be simulated by an orifice.

4.0 REFERENCES

The following documents are applicable to this test program to the extent specified herein:

4.1 MIL-P-27407 (I) Propellant Pressurizing Agent, Helium

4.2 MIL-P-27401B Propellant Pressurizing Agent, Nitrogen

4.3 MSC-PPD-2A Propellant, Nitrogen Tetroxide, Inhibited

4.4 MIL-P-27404 Propellant, Monomethylhydrazine

4.5 ASTM D1193-56 Distilled Water

4.6 MSFC-SPEC-237 Freon TF

4.7 TMC Drawing X29200 OMS Pressure Regulator, Prototype Test

4.8 TMC Drawing X29267 OMS Pressure Regulator Simulator Valve

5.0 <u>GENERAL TEST CONSIDERATIONS</u>

5.1 Cleanliness

5.1.1 Handling and installation of the test unit shall be performed in a manner that will insure the cleanliness of the test item interior surfaces.

5.1.2 All test fluids are to be passed through a facility filter of a least 25 micron nominal rating, and the filter is to be installed upstream of the test unit.

5.1.3 The test units are to be kept in sealed, clean plastic bags during storage and transportation.

5. 2 Instrumentation

5.2.1 The accuracy of all measuring and recording devices used during the program shall be verified prior to their use.

5.2.2 Standard instrument inspection/calibration periods shall not be permitted to lapse during the subject test program.

5.2.3 Test instrumentation and required range and accuracy are shown in Tables A & B.

5.2.4 Test equipment description shall include the following minimum information:

5.2.4.1 Descriptive Name

5.2.4.2 Range

5.2.4.3 Accuracy

5.2.4.4 Date of Last Calibration

5.2.4.5 Date of Next Calibration

5.3 Facility

5.3.1 Final decision as to the adequacy of the test setup and conduct of the test, with the exception of the operation of the test facility, shall be at the discretion of the Project Engineer.

5.3.2 All liaison engineering concerning the test program shall be coordinated through the Project Engineer.

5.3.3 The facility test setup shall be of materials which are compatible with the test fluids being used.

5.3.4 All test unit actuations are to be recorded by the Test Engineer or his designated representative. The definition of a test unit actuation is each time the flow through regulator decreases to zero or each time the regulator locks up.

5.4 Documentation

5.4.1 Witnesses

5.4.1.1 The Project Engineer shall be informed prior to the start of all tests and at any time when an unanticipated situation affecting the test setup, method or test item occurs. The Project Engineer or his designated representative shall be present during the conduct of tests.

5.5 Test Data and Identification

5.5.1 The data recorded shall be marked with the information necessary to completely identify it. The following items are considered a minimum required test identification and will be the responsibility of the Project Engineer:

5.5.1.1 Unit part number and serial number being tested.

5.5.1.2 Type of test to be conducted, MTP No., and applicable appendix identification.

5.5.1.3 Type, range, and identification number of each measuring instrument used during the tests.

5.5.1.4 Identification of test operator, facility, time, date, and test witnesses.

5.5.1.5 Data sheets, or copies thereof, shall be incorporated in the applicable regulator log book.

5.6 Data Storage and Processing

5.6.1 The Project Engineer shall be responsible for all data retrieved from the tests as supplied by test operations personnel.

5.7 Testing

Detail test requirements for each test are defined in Appendices A &

APPENDIX A - REGULATOR COMPONENT DESIGN TEST

The purpose of Appendix A tests is to verify and document physical design values of critical regulator parts which effect regulator performance. Instrumentation requirements are specified in Table A for the following tests:

A-1.0 Actuator Outer Spring, Spring Rate Test

A-1.1 Record S/N and test results on Data Sheet A

A-1.2 Prior to testing, exercise spring by applying and removing a 400 lb. load three times.

A-1.3 Measure and record the free length l_o.

A-1.4 Apply a preload of 390 lbs. and measure and record the resulting deflection.

A-1.5 Apply additional 3-lb. load and measure and record the resulting deflection.

A-1.6 Remove the additional 3-lb. load and measure and record the resulting deflection.

A-1.7 Remove 390-lb. preload and measure and record the free length l_o'.

A-2.0 Actuator Inner Spring, Spring Rate Test

A-2.1 Record S/N, and test results on Data Sheet A.

A-2.2 Prior to testing, exercise spring by applying and removing a 175 lb. load three times.

A-2.3 Measure and record the free length l_o.

A-2.4 Apply a preload of 170 lbs. and measure and record the resulting deflection.

A-2.5 Apply additional 1.5-lb. load and measure and record the resulting deflection.

A-2.6 Remove the additional 1.5-lb. load and measure and record the resulting deflection.

A-2.7 Remove 170-lb. preload and measure and record the free length l_o'.

A-3.0 Actuator Bellows Spring Rate Test

A-3.1 Record S/N and test results on Data Sheet A.

A-3.2 Prior to testing, exercise bellows by applying and removing a 290-lb. load three times.

A-3.3 Measure and record the free length l_o'.

A-3.4 Apply a preload of 277 lbs. and measure and record the resulting deflection.

A-3.5 Apply additional 4-lb. load and measure and record the resulting deflection.

A-3.6 Remove the additional 4-lb. load and measure and record the resulting deflection.

A-3.7 Remove 277-lb. preload and measure and record the free length l_o'.

A-4.0 Poppet Bellows Spring Rate Test

A-4.1 Record S/N and test results on Data Sheet A.

A-4.2 Prior to testing, exercise bellows by applying and removing a 25-lb. load three times.

A-4.3 Measure and record the free length l_o.

A-4.4 Apply a preload of 18 lbs. and measure and record the resulting deflection.

A-4.5 Apply additional 5-lb. load and measure and record the resulting deflection.

A-4.6 Remove the additional 5-lb. load and measure and record the resulting deflection.

A-4.7 Remove 18-lb. preload and measure and record the free length l_o'.

A-5.0 Push Rod Bellows Spring Rate

A-5.1 Record S/N and test results on Data Sheet A.

A-5.2 Prior to testing, exercise bellows by applying and removing a 3-lb. load three times.

A-5.3 Measure and record the free length l_o.

A-5.4 Apply a 1-lb. load and measure and record the resulting deflection.

A-5.5 Apply additional 1-lb. load and measure and record the resulting deflection.

A-5.6 Apply additional 1-lb. load (3-lbs. total) and measure and record the resulting deflection.

A-5.7 Remove 1-lb. and measure and record the resulting deflection.

A-5.8 Remove 1 lb. and measure and record the resulting deflection.

A-5.9 Remove remaining 1 lb. load and measure and record the free length l_o'.

A-6.0 <u>Actuator Flexure Assembly Spring Rate</u>

A-6.1 Record S/N and test results on Data Sheet A.

A-6.2 Prior to testing, exercise flexure assembly by applying and removing a 1-lb. load three times in one direction and identify deflection direction.

A-6.3 Apply 1/2 lb. load and measure and record the resulting deflection.

A-6.4 Add another 1/2 lb. load and measure and record the resulting deflection.

A-6.5 Remove 1/2 lb. load and measure and record the resulting deflection.

A-6.6 Remove remaining 1/2 lb. load and measure and record resulting deflection.

A-7.0 <u>Poppet Flexure Spring Rate</u>

A-7.1 Record S/N and test results on Data Sheet A.

A-7.2 Prior to testing, exercise flexure by applying and removing a 2-lb. load three times in one direction and identify deflection direction.

A-7.3 Apply 1-lb. load and measure and record the resulting deflection.

A-7.4 Add another 1-lb. load and measure and record resulting deflection.

A-7.5 Remove 1-lb. load and measure and record resulting deflection.

A-7.6 Remove remaining 1-lb. load and measure and record resulting deflection.

A-8.0 Actuator Bellows Effective Area Test

A-8.1 Record S/N and test results on Data Sheet A.

A-8.2 Remove seals at closed end of bellows from test fixture X________.

A-8.3 Install the bellows and pressurize fixture to approximately 84 psig until the dial indicator reads a deflection less than .001" and greater than zero. Record pressure and resulting deflection.

A-8.4 Increase pressure approximately 0.6 psig and record pressure and total deflection.

A-8.5 Again increase pressure approximately 0.6 psig and record pressure and total deflection.

A-8.6 Decrease pressure approximately 0.6 psig and record pressure and total deflection.

A-8.7 Decrease pressure until dial indicator reads greater than zero and less than .001". Record pressure and total deflection.

A-9.0 Poppet Bellows Effective Area Test

A-9.1 Record S/N and test results on Data Sheet A.

A-9.2 Remove seals at closed end of bellows from test fixture X.

A-9.3 Install the bellows and pressurize fixture to approximately 138 psig until the dial indicator reads a deflection less than .001" and greater than zero. Record pressure and resulting deflection.

A-9.4 Increase pressure approximately 17 psig and record pressure and total deflection.

A-9.5 Again increase pressure approximately 17 psig and record pressure and total deflection.

A-9.6 Decrease pressure approximately 17 psig and record pressure and total deflection.

A-9.7 Decrease pressure until dial indicator reads greater than zero and less than .001". Record pressure and total deflection.

A-10.0 Push Rod Effective Area Test

A-10.1 Record S/N and test results on Data Sheet A.

A-10.2 Remove seals at closed end of bellows from test fixture X.

A-10.3 Install the bellows and pressurize fixture to approximately 31 psig and record pressure and resulting deflection.

A-10.4 Increase pressure approximately 31 psig and record pressure and total deflection.

A-10.5 Decrease pressure approximately 31 psig and record pressure and total deflection.

A-10.6 Decrease pressure to zero gage and record total deflection.

A-11 0 Time Preload Test

A-11.1 Install each actuator inner spring. outer spring, bellows, and poppet bellows in each Test Fixture X________ .

A-11.2 Load each test unit to the preload length per respective Data Sheet, and record time and date on respective Data Sheets.

A-11.3 Remove preload after one week and measure and record resulting free length l_o".

A-12.0 Contaminant Cutting Test

A-12.1 The ability of the poppet/seat interface to cut contaminants without damaging the interface is to be determined by placing wire between the poppet and seat. Tests are to be performed on both K801 poppet/seat and B_4D poppet/seat. Also, both copper and stainless steel wire of .001" (25 μ), .003" (76 μ), and .0063" (160 μ) diameter will be used to simulate the contaminant. Place the wire across the land and then install the seat on the Link Load Checker.

A-12.2 Measure and record the force required to cut each size wire on Data Sheet A. Inspect the land with the Wild Interference Microscope after each test for damage and record observations with photographs.

A-13.0 Propellant Compatibility Test

A-13.1 A static, one week propellant exposure test will be performed on materials that lack surface finish compatibility data for N_2O_4 and N_2H_4 propellants which will be contaminated with 5% water. The materials to be tested in each propellant are K801, B_4C, and a Croniro brazed B_4C sample. All samples are to be weighed within ± 10 milligrams and the surface finish of each sample is to be recorded.

A-13.2 Place each specimen in an ampule and pressurize to 250 psig with helium and maintain ampule at room temperature for one week.

A-13.3 Measure and record the weight and surface finish of each sample.

A-14.0 Proof Pressure Test

A-14.1 Regulator X29200 will be proof tested to ensure that the structural integrity of the design prior to any performance test. Record the S/N and test results on Data Sheet A.

A-14.2 Fill the inlet, outlet, and actuator sense port cavities with distilled water for safety reasons if necessary. Install the regulator in the test setup, Figure A-1.

A-14.3 Pressurize the actuator sense port to 500 psig at a rate less than 100 psi/sec. Hold for 5 minutes. Reduce the sense port pressure to 300 psig. Any evidence of permanent distortion or damage is unacceptable.

A-14.4 With the actuator sense port at 300 psig or regulator locked closed and the outlet port open, pressurize the inlet port to 6000 psig at a rate less than 100 psi/sec. Hold for 5 minutes and then reduce the inlet pressure to zero gage. Any evidence of permanent distortion or damage is unacceptable.

A-14.5 Reduce the actuator sense port pressure to zero gage to open the regulator. Close the outlet port and then pressurize the inlet port to 6000 psig at a rate less than 100 psi/sec. Hold for 5 minutes and then reduce pressures to zero gage. Any evidence of permanent distortion or diamage is unacceptable.

A-14.6 Remove the actuator from the test setup and drain any water from the cavities. Place the actuator in an oven for 1 hour at 250°F with all ports open to ensure that the regulator cavities are dry. Record proof test results on Data Sheet A.

A-15.0 External Leakage Test

A-15.1 Install the regulator in the test setup shown in Figure A-2 and record S/N and test results on Data Sheet A.

A-15.2 Pressurize the inlet port to 2000 psig with the outlet port closed. Next, pressurize the actuator sense port to 250 psig and hold pressures for 5 minutes to stabilize temperature.

A-15.3 Bubble check all regulator flanges for external leakage with "Snoop" for 5 minutes. Reduce all pressures to zero gage. Any evidence of bubbles is unacceptable.

TABLE A-1

INSTRUMENTATION REQUIREMENTS

PARAMETER	SYMBOL	RANGE	ACCURACY	PARAMETER TO BE RECORDED PER TEST #														
				1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
LOAD BALANCE SYSTEM	W_1	0-500 LB	.02%	X	X	X	X	X	X	X								
CALIPER	L	0-6"	±.001"	X	X	X	X	X			X	X	X	X				
DIAL INDICATOR	ΔX	0-.0300"	±.0001	X	X	X	X	X	X	X	X	X	X					
T.I. PRECISION PRESS. GAGE	P_3'	0-500 PSIG	.02%								X	X	X					
LINK LOAD CHECKER	W_2	0-50 LB	1%												X			
INTERFERENCE MICROSCOPE & CAMERA	M	-	-												X			
SCALE	W_3	0-100 GRAM	±.01 GRAM													X		
BENDIX PROFICORDER	-	-	±3 µin													X		
PRESS GAGE	P_3	0-500 PSIG	1%													X	X	X
PRESS GAGE	P_1	0-6000 PSIG	1%														X	X

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FIGURE A-1

PROOF PRESSURE TEST SETUP

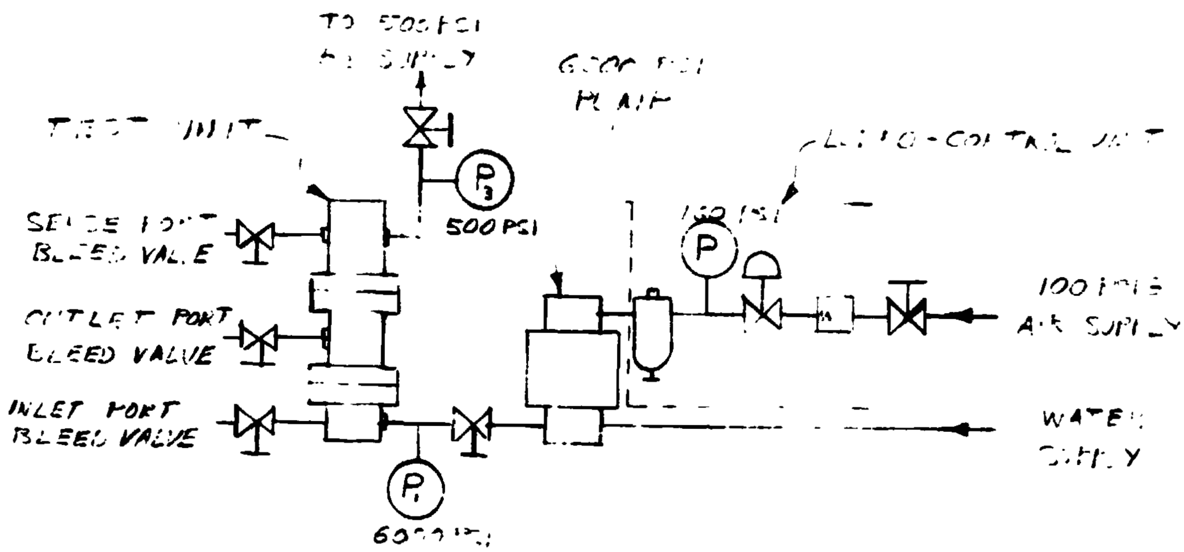
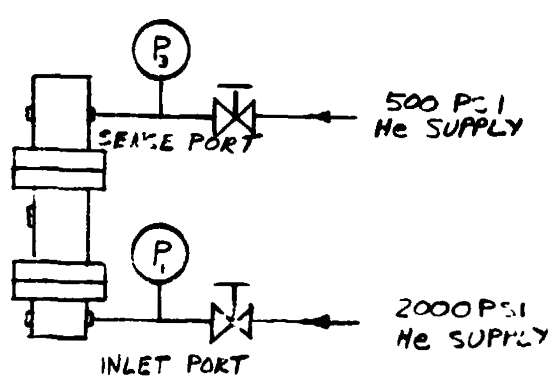


FIGURE A-2

EXTERNAL LEAKAGE TEST SETUP



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APPENDIX B - REGULATOR PERFORMANCE

The purpose of Appendix B tests is to verify regulator design performance characteristics. Instrumentation requirements are specified in Table B-I for the following tests:

B-1.0 REGULATOR FLOW FORCE AND PRESSURE DROP TEST

B-1.1 Install the regulator simulator valve X29267 in the test setup as shown in Figure B-1. The test fluid shall be helium. The facility U/S and D/S valves shall have an opening and closing response of 200 ms (max).

B-1.2 First, pressurize the sense port to balance the actuator preload force at a stroke set point of 0 to .0002". Record P_3' and maintain constant.

B-1.3 The tear force due to stroke shall be determined by measuring the load cell value at stroke set points of 0, .0005; .0010, .0020, .0050, .0100, .0150, .0200, and .0250" at zero differential pressure of inlet pressures (P_o) and outlet pressures (P_3) of 0, 250, 300, and 350 psig ± 5%.

B-1.4 The tear force due to static inlet and outlet differential pressures at "zero" stroke shall be determined by slowly pressurizing the inlet (P_o) to 400 psig ± 3% with the stroke set at zero. Next, crack open valve X29267 until (P_3) increases to 250 psig ± 5% (downstream facility valve closed) and then record load cell value. Next, slowly increase (P_3) to 300 psig ± 5% and then record load cell value. Finally, increase (P_3) to 350 psig ± 5% and record load cell value.

B-1.5 Repeat 1.4 except set inlet pressures (P_o) at 680 psig ± 3%.

B-1.6 Repeat 1.4 except set inlet pressure (P_o) at 2000 psig ± 3%.

B-1.7 Repeat 1.4 except set inlet pressure (P_o) at 4000 psig ± 3%

B-1.8 Next, three series of flow tests shall be performed. Each series of tests shall consist of four tests at inlet pressures (P_o) of 4000, 2000, 680, and 400 psia ± 3%, and at a constant back pressure (P_3) of 250 ± 3 psia. Three flow nozzle sizes or equivalent are required, one size for each series of tests: 2.6, 15.6, 20.0 CFM ± 5%. Use facility D/S throttle valve to maintain P_3 constant.

B-1.9 Prior to each flow test, set P_3' to balance the preload force and then set the stroke of the regulator simulator valve X29267 to a predetermined value which shall be recorded on the original data document for each test. The approximate values of strokes are as follows:

P_o (PSIG)	Q (CFM)	X (in.)
4000	20	.0018
2000	"	.0037
680	"	.0106
400	"	.0220
4000	15.6	.0145
2000	"	.0029
680	"	.0083
400	"	.0200
4000	2.6	.0002
200C	"	.0004
680	"	.0014
400	"	.0025

B-1.10 Set the inlet pressure (P_o) plus the flow pressure drop and then open the facility U/S and D/S valves. Record the following data after maintaining required pressures P_o and P_3 constant at least 5 seconds: T_o, P_o, P_1, P_2, P_3, T_N, P_3', P_N, X, and F. Flow data shall be sufficient to calculate mass flowrate within 3%. Pressure, temperature and force measurement accuracy shall be as specified in Table B-I. Terminate tests as soon as possible to conserve helium.

B-2.0 FLOW LIMITER PRESSURE DROP TEST

B-2.1 Install the Flow Limiter X 29500 in the test setup as shown in Figure B-2. The test fluid shall be helium. Lock test regulator X29200 open by maintaining one atmosphere in the sense port #3. The flow nozzle or equivalent shall be sized for 33.3 CFM ±5% at a test system back pressure (P_3) of 300 psia ± 5% and the flow nozzle shall have a range from approximately 15.6 - 33.3 CFM. The facility U/S and D/S valves shall have an opening and closing response of 200 ms (max).

B-2.2 Perform four(4) tests; one each at an inlet pressure (P_o) 400, 680, 2000, and 4000 psia. The inlet pressure (P_o) shall be maintained within ± 5% and the back pressure (P_3) shall be maintained at 300 psia ± 5% for each test. Use facility D/S throttle valve to maintain P_3 constant.

B-2.3 Prior to each test, charge the downstream plumbing to 300 psia ± 5% with the D/S throttle valve closed. Next, close the U/S facility valve and pressurize the inlet pressure (P_o) to the test set pressure plus the flow pressure drop.

B-2.4 Open the facility U/S and D/S valves within 200 ms. Record the following data after maintaining required pressures P_o and P_3 constant for at least 5 seconds: T_o, P_o, ΔP_F, P_1, P_2, P_3, T_N, & P_N. Flow data shall be sufficient to calculate mass flowrate within 3%. Pressure and temperature measurement accuracy shall be as specified in Table B-I. Terminate tests as soon as possible to conserve helium. The ΔP_F measurement is only required at P_o of 400 psia.

B-3.0 INTERNAL LEAKAGE TEST

B-3.1 Measure the internal leakage of the regulator by connecting the helium supply system to the regulator inlet port and sense port. Cap the outlet port and pressurize the sense port to 300 psig ± 5% so that the regulator is locked closed. Then pressurize the inlet port to 400 psig ± 5%. Measure and record internal leakage from the instrumentation outlet port B for 5 minutes minimum.

B-3.2 Repeat Test 3.1 except set inlet pressure at 800 psig ± 3%.

B-3.3 Repeat Test 3.1 except set inlet pressure at 2000 psig ± 3%.

B-3.4 Repeat Test 3.1 except set inlet pressure at 4000 psig ± 3%.

B-4.0 REGULATOR PERFORMANCE TEST AT ROOM TEMPERATURE

B-4.1 Install the regulator in the test setup shown in Figure B-2. The test fluid shall be helium. The facility U/S and D/S valves shall have an opening and closing response of 200 ms (max).

B-4.2 First, a facility flow test shall be performed to determine the dynamic stability of the test unit inlet pressure P_o at the nominal flowrate (15.6 CFM) and to check the regulator set point (P_3'). Install a 1000 HZ (min) response transducer at station P_o. With the U/S facility valve closed and the D/S facility valve open, pressurize the inlet pressure P_o to 4000 ± 50 psia, plus the flow pressure drop. Open the U/S valve, and then close the D/S valve after 5 second of constant flow as indicated by P_N not varying more than ± 1.0 psi. The inlet pressure P_o shall not drift more than ± 25 psig and the pressure oscillation shall not exceed 10 HZ and 20 psi (peak-to-peak) during the constant flow condition. Record pressure-time relationship of P_o and P_3 during the entire test beginning 5 seconds prior to opening the U/S valve and 5 seconds after closing the D/S valve. Also record steady state values of P_o, T_o, P_1, P_2, P_3', P_3, T_N, T_R, P_N, and S. The regulated steady state pressure P_3' and mass flowrate $\dot{M}_N$ shall be baseline values.

B-4.3 Prior to the first regulator test, one slam start shall be simulated by charging the flow volume between the facility U/S valve and D/S valve (closed) at an inlet pressure of 4000 psia ± 3%. The initial pressure of the volume shall be one atmosphere. Record the time-pressure relationship of the inlet and outlet pressures P_1 and P_3 during the entire test, beginning 5 seconds prior to opening the U/S valve and 5 seconds after regulator lock-up.

B-4.4 Next, one series of four regulator performance tests shall be performed at room temperature and at inlet pressures (P_o) of 4000, 2000, 680, and 400 psia ± 3%. With the facility U/S valve open and the inlet pressure P_o set, open the facility D/S valve to start flow. Record the pressure-time relationship of P_1 and

P_3 during the entire test beginning 5 seconds prior to opening the D/S valve and 5 seconds after regulator lock-up. Flow shall be terminated by closing the D/S facility valve after 5 seconds of constant flow as indicated by P_N not varying more than ± 1.0 psi. Also record steady state values of P_o, T_o, P_1, P_2, P_3', P_3, T_A, T_N, P_N, and S. Flow data shall be sufficient to calculate mass flowrate within 3%. Pressure and temperature measurement accuracy shall be as specified in Table B-I. Terminate test as soon as possible to conserve helium.

B-4.5 Perform internal leakage tests per Test 3.0 after completing each series of temperature performance tests.

B-5.0 REGULATOR PERFORMANCE TEST AT 150°F

B-5.1 Repeat Test 4.0 except heat the test fluid and test unit to 150°F. After the slam start test, allow temperatures to stabilize within 25°F before opening the facility D/S valve.

B-6.0 REGULATOR PERFORMANCE TEST AT -150°F

B-6.1 Repeat Test 4.0 except chill the test fluid and test unit to -150°F. After the slam start test, allow temperatures to stabilize with in 25°F before opening the facility D/S valve.

B-7.0 OMS ENGINE BURN VIBRATION TESTS

B-7.1 Vibrate the regulator per OMS Engine Burn Random Vibration Specification, Figure B-3 and Table B-II, in only the X-axis. The X-axis is defined as the axis parallel to the actuator shaft and the Y-axis as the axis perpendicular to the direction of flow and the X-axis. The regulator will be flowing helium at 15.6 CFM ±5% at an inlet pressure of 4000 psia ± 3%. A 1 ft^3 ± 10% tank shall be located D/S of the regulator and the regulated outlet pressure P_3 shall be recorded within 5% with a 1000 HZ (min) transducer. Record parameters per Table B-I.

B-7.2 The internal leakage shall be measured before and after vibrating the regulator in the X-axis per leakage Test 3.0.

B-8.0 MAIN ORBITAL ENGINE VIBRATION TEST

B-8.1 Vibrate the regulator per Main Orbital Engine Random Vibration Specification, Figure B-3 and Table B-II in the X-Y axes. The regulator will be flowing helium at 15.6 CFM ±5% at an inlet pressure of 4000 psia ± 3%. A 1 ft^3 tank shall be located downstream of the regulator and regulated outlet pressure P_3 shall be recorded within 5% with a 1000 HZ (min.) transducer. Record data per Table B-I.

B-8.2 The internal leakge shall be measured before and after each vibration test in each axis per leakage Test 3.0.

B-9.0 LIFT-OFF VIBRATION TEST

B-9.1 Vibrate the regulator per Lift-Off Random Vibration Specification, Figure B-3 and Table B-II in the X and Y axes. The regulator will be in the dry condition.

B-9.2 Record poppet stroke S to evaluate the effect of vibration on the position indicator.

B-9.3 The internal leakage shall be measured before and after each vibration test in each axis per leakage Test 3.0.

B-10.0 CONTAMINATION TOLERANCE TEST

B-10.1 Measure the regulator internal leakage per Test 3.0 to determine base line value.

B-10.2 Install the regulator in the test setup shown in Figure B-4. The nominal flowrate shall be 15.6 CFM of helium at a regulated outlet pressure P_3 of 250 psia. The D/S facility valve shall have an opening and closing response of 200 ms (maximum).

The contaminant injector shall be capable of injecting a range of particles from 25μ to 150μ at a rate of approximately 5 gm/sec at 400 psia.

B-10.3 Pressurize the inlet pressure (P_o) to 400 psia ± 3% with the facility D/S valve closed. Cycle the D/S valve 100 times at a rate of 1.0 cps and inject the 25μ sample at a predetermined rate in the flow stream at least 10 diameters upstream of the regulator inlet port. Record parameters per Table B-I.

B-10.4 Perform internal leakage test per Test 3.0.

B-10.5 Repeat Tests 7.3 and 7.4 until a total of 300 cycles have been accumulated.

B-10.6 Repeat Test 7.3 through 7.5 except inject the 150μ sample at a predetermined rate.

B-10.7 After completing contamination tests, repeat Test 4.4 in contamination test setup to check regulator performance. Delete parameters P_2 and P_3'.

B-11.0 TEMPERATURE PRELOAD TEST

The effect of temperature on the actuator preload will be determined by measuring the change in preload and deflection from room temperature to 150°F and -150°F using test fixture X which consists of a dial indicator thermally isolated from the pressure-preloaded actuator assembly. Prior to the test, record the ambient temperature T_A and preload pressure.

B-11.1 Heat the actuator assembly to 150°F in the test setup shown in Figure B-5 and allow temperature to stabilize within 10°F. Record stabilized temperatures, rod deflection X and preload pressure P_3'. Adjust and record preload pressure P_3' to obtain initial rod position.

B-11.2 Cool the actuator assembly to -150°F in the test setup shown in Figure 5 and allow temperature to stabilize within 10°F. Record stabilized temperatures, rod deflection X and preload pressure P_3'. Adjust and record preload pressure P_3' to obtain initial rod position X.

APPENDIX B ADDENDUM

B-12.0 <u>Life Cycle Test</u>

B-12.1 Install the test unit in the test setup, Fibure B-2, except connect the inlet of the test unit to a 400 psig gaseous nitrogen supply tank. Set the facility D/S throttle valve for an average flow rate of 2 cfm (2.5 lb/min.) at an inlet pressure (P_3) of 250 psia.

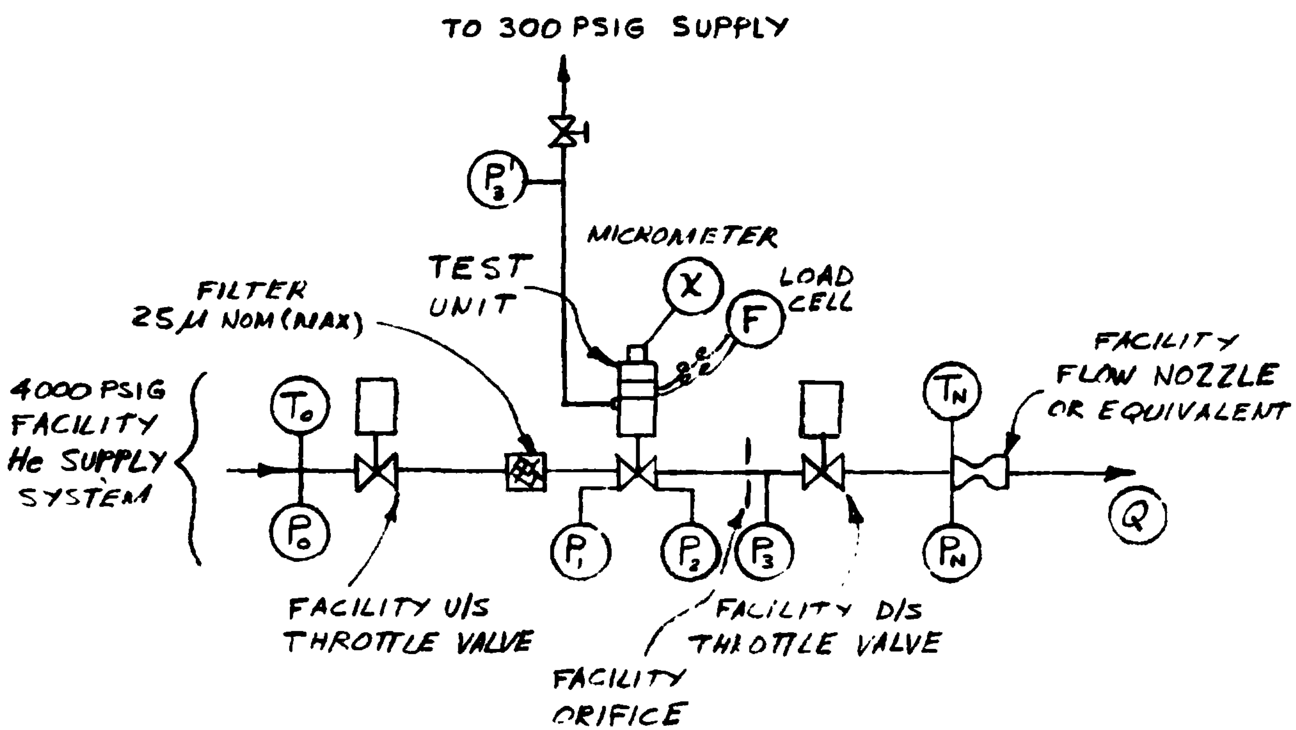
B-12.2 Cycle the test unit 100,000 times by cycling the facility D/S valve at an approximate rate of 1 cps. Start the cycle test at a regulator inlet pressure (P_1) of 400.

B-12.3 Measure the regulator internal leakage per Test B-3.0 after 100; 500; 1000; 2500; 5000; 10,000; 20,000; 40,000; etc. Also record the number of cycles accumulated in each successive 400 psi interval. The definition of a cycle is each time the regulator locks up or each time the flow through the regulator decreases to zero.

B-12.4 After completing the leakage test after 100,000 cycles, repeat Test B-4.4 with helium to check regulator performance at nominal conditions.

FIGURE B-1

SIMULATOR REGULATOR ΔP & FLOW FORCE TEST SETUP

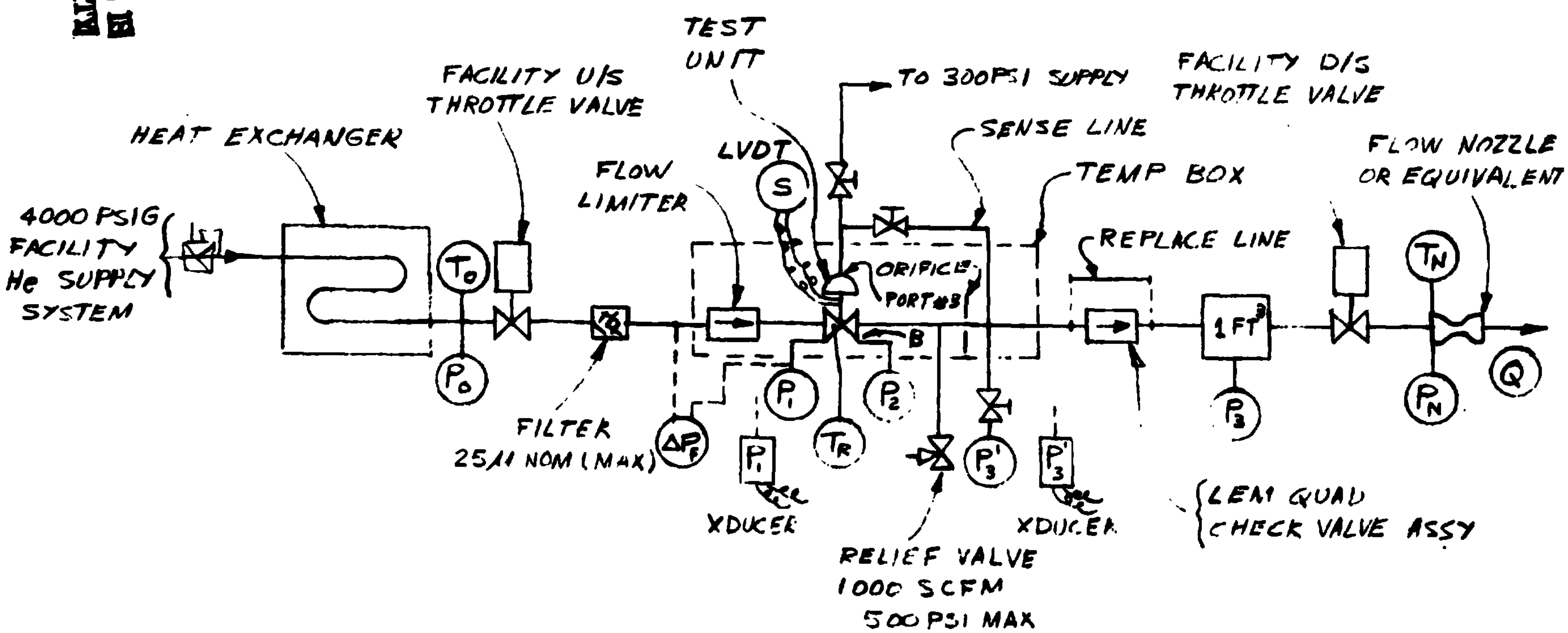


NOZZLE SIZE OR RANGE @ P_3 = 250 PSIA: Q_1 = 2.6 CFM
Q_2 = 15.6 CFM
Q_3 = 20.0 CFM

ORIFICE SIZE: Q = 15.6 CFM @ P_2 = 327 PSIA, P_3 = 250 PSIA, & T_2 [illegible] °F.

FIGURE B-2

REGULATOR & FLOW LIMITER PERFORMANCE TEST SET UP



NOZZLE SIZE: 20.0 CFM @ P_3 = 300 PSIA & 15.6 CFM @ P_3 = 250 PSIA

ORIFICE SIZE: 15.6 CFM @ P_2 = 327 PSIA, P_3' = 250 PSIA, & T_2 = -15[illegible] °F

FIGURE B-3

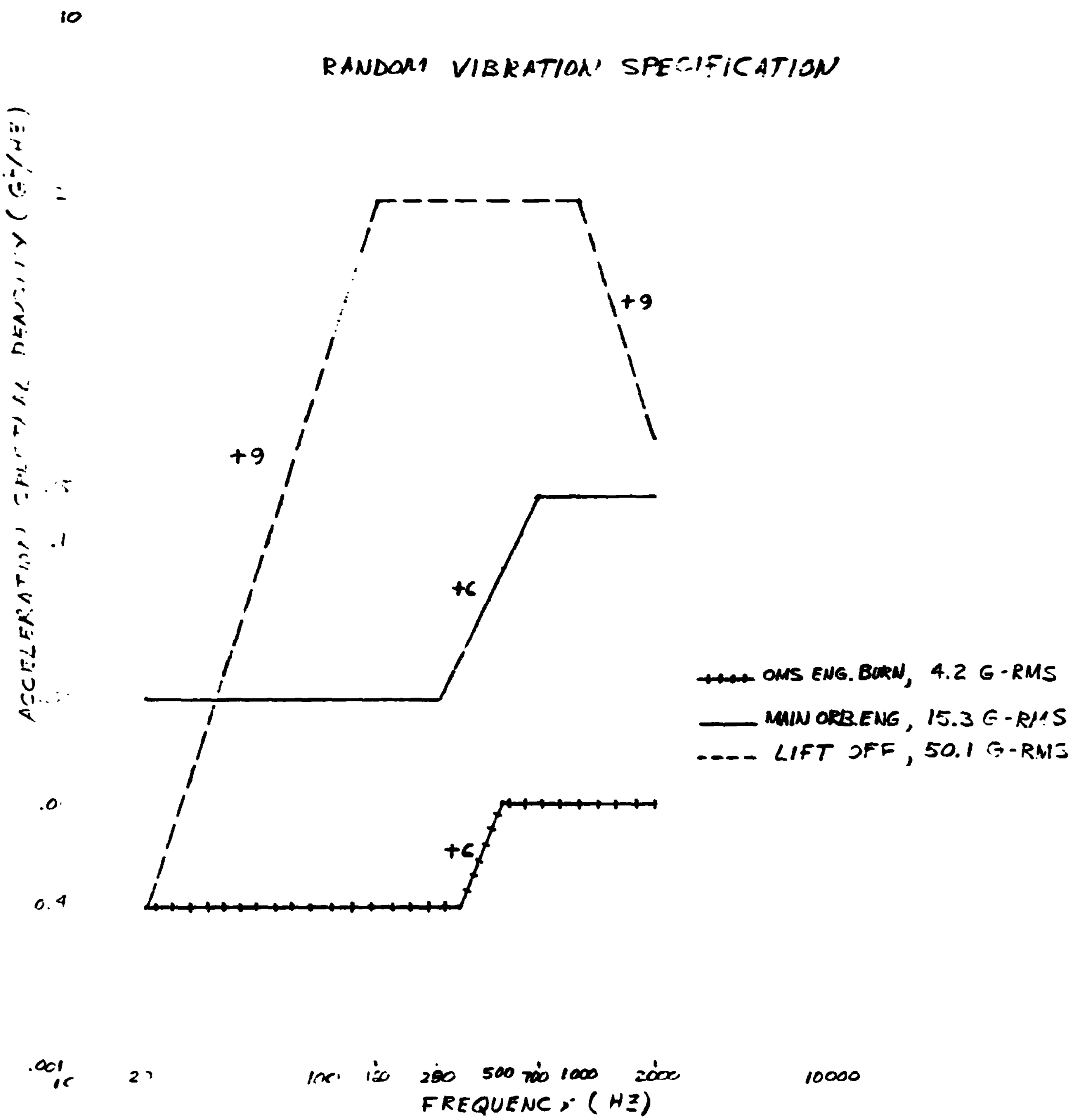
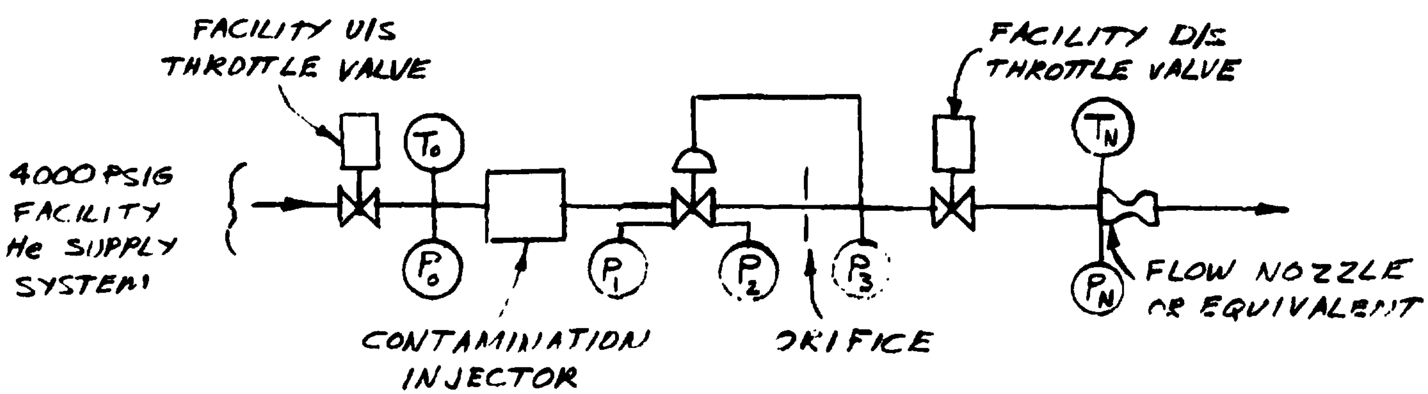

RANDOM VIBRATION SPECIFICATION
10
.1
.0
0.4
.001
+9
+9
+6
+6
OMS ENG. BURN, 4.2 G-RMS
MAIN ORB. ENG, 15.3 G-RMS
LIFT OFF, 50.1 G-RMS
500
1000
10000
FREQUENCY (HZ)

FIGURE B-4

CONTAMINATION TOLERANCE TEST SETUP



NOZZLE SIZE: 15.6 CFM @ P_3 = 250 PSIA

ORIFICE SIZE, 15.6 CFM @ P_2 = 327 PSIA, P_3 = 250 PSIA, & T_2 = -150 °F

FIGURE B-5

TEMPERATURE PRELOAD TEST SETUP

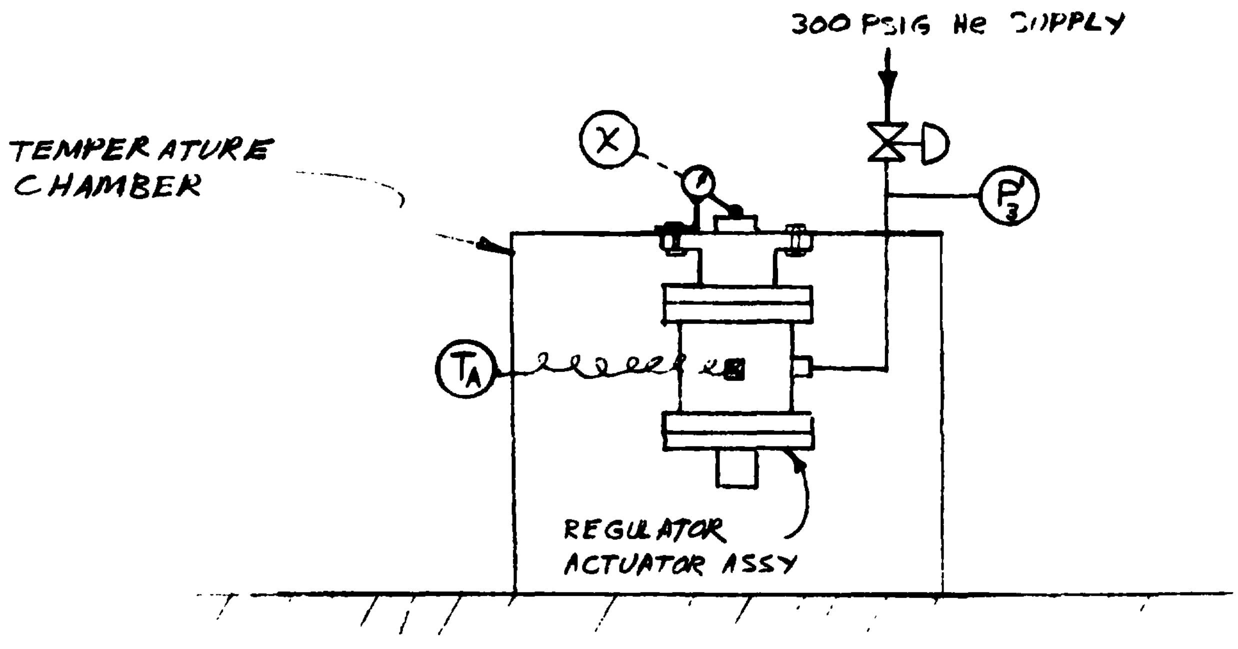


TABLE B-I
INSTRUMENTATION REQUIREMENTS

PARAMETER	SYMBOL	RANGE	ACCURACY	PARAMETER TO BE RECORDED PER TEST # 1	2	3	4	5	6	7	8	9	10	11
SUPPLY PRESSURE	P_o	0-4000 PSIG	1% F/S	X	X	X	X	X	X	X	X		X	
FLOW LIMITER ΔP	ΔP_F	50 ΔP 500 PSI INLET	1% F/S		X									
REG U/S PRESSURE	P_1	0-4000 PSIG	1% F/S	X	X		X	X	X					
REG U/S 1000 HZ XDUCER	P_1	0-4000 PSIG	5% F/S				X	X	X				X	
REG D/S PRESSURE	P_2	0-500 PSIG	1% F/S	X	X		X	X	X				X	
REG SENSE PRESS	P_3'	0-500 PSIG	.02% F/S				X	X	X	X	X			X
REG SENSE PRESS	P_3	0-500 PSIG	1% F/S	X	X		X	X	X	X	X		X	
REG SENSE 1000 HZ XDUCER	P_3	0-500 PSIG	5% F/S				X	X	X	X	X			
NOZZLE INLET PRESS	P_N	0-500 PSIG	1% F/S	X	X		X	X	X	X	X		X	
SUPPLY TEMP	T_o	150°F TO -150°F	3% F/S	X	X		X	X	X	X	X			
REG SKIN TEMP	T_R	150°F TO -150°F	3% F/S				X	X	X					
ACTUATOR AMB TEMP	T_A	150°F TO -150°F	3% F/S											X
INTERNAL LEAKAGE	Q_1	0-150 SCC/HR	3% F/S			X	X	X	X	X	X	X	X	
FLOW RATE	Q	SEE FIG'S	3%	X	X		X	X	X	X	X		X	
POPPET STROKE	χ	0-.022 IN	±.0001	X										X
POPPET STROKE 1000 HZ	S	20 V/IN 0-.022 IN	3% F/S				X	X	X	X	X	X		
FLOW FORCE	F	0-50 LB	.25% F/S	X										
NOZZLE INLET TEMP	T_N	150°F TO -150°F	1% F/S	X	X		X	X	X	X	X		X	

TABLE B-II

RANDOM VIBRATION TEST TOLERANCES & DURATION

a) Random Vibration Tolerances

Plus or minus 1-db on overall rms acceleration and plus or minus 3-db on acceleration spectral density (g^2/H_z) for the following bandwidths:

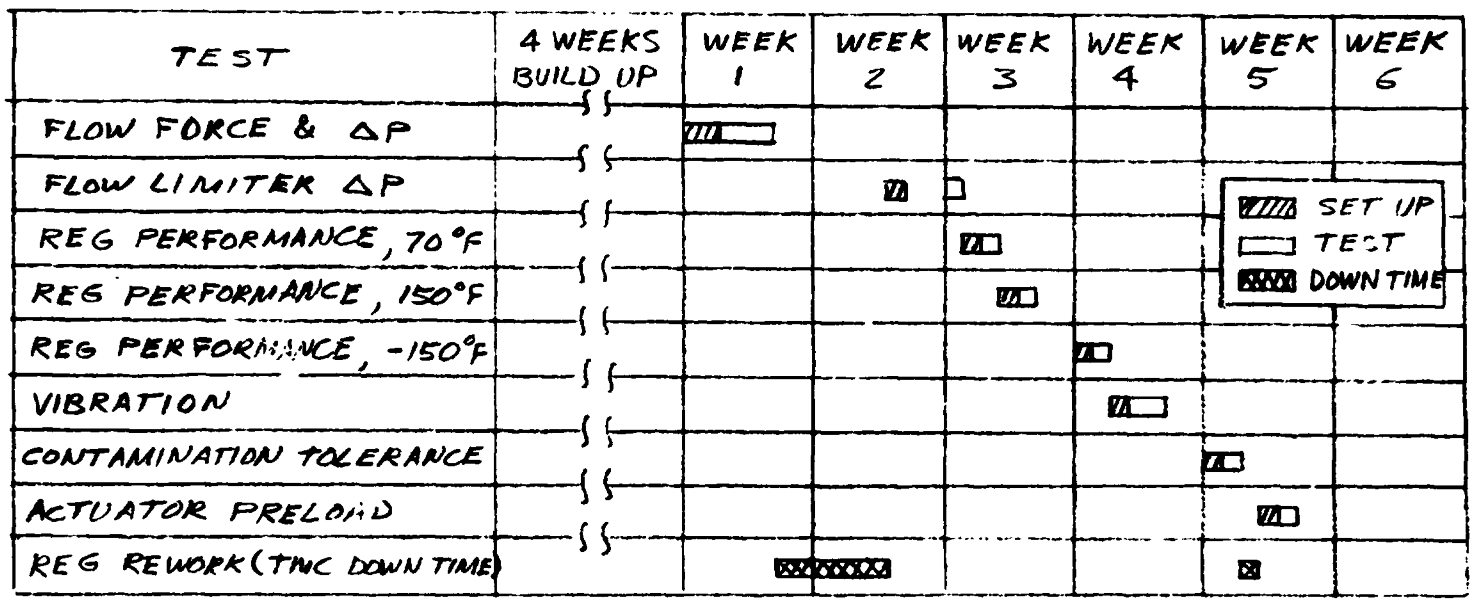
Frequency Range	Maximum Effective Bandwidth
10 to 100 H_z	6 H_z
100 to 500 H_z	12 H_z
500 to 2000 H_z	24 H_z

Analysis sample time (T) shall equal or exceed 50/BW, where BW is the effective bandwidth of the filter utilized. For swept filter analysis, analyzer filter scan rate (SR) shall not exceed BW/T (H_z/sec.). It is recommended the averaging be obtained by using linear integration with an integration time of T. However, if averaging is obtained by smoothing with an equivalent resistance capacitance (RC) low-pass filter, a time constant RC = T/2 shall be used. In this case, the scan rate shall be BW/4RC.

b) Random Vibration Duration

The vibration test duration shall be adequate to perform functional and/or continuity checkouts, but shall not be less than 3 minutes or greater than 5 minutes per axis in each of three orthogonal axe. Should reruns be required in any axis, the total accumulative vibration test time in that axis should not exceed 9 minutes.

OMS HELIUM REGULATOR TEST SCHEDULE



APPENDIX C

VERIFICATION TEST

SHUTTLE OMS HELIUM REGULATOR PROTOTYPE

SUPPLEMENTAL TEST PLAN FOR A NASA-JSC TEST PROGRAM

MARQUARDT TEST PLAN 0213, REVISION A

1.0 OBJECTIVE

The objective of this test program is to repeat those performance tests conducted at TMC for verification by NASA-JSC and to document propellant compatibility and its effect on extended life cycle performance. Additional test data on this prototype unit will contribute to the refinement of the regulator design.

2.0 SCOPE

The scope of this test program is to:

2.1 Verify regulated and lockup pressure at inlet conditions of 400-4000 psi and -150°F to 150°F.

2.2 Subject regulator to N_2O_4 and amine fuel vapors and moisture, individually and in combination for extended durations.

2.3 Verify regulated and lockup pressure (set point repeatability) at -150°F and 400-4000 psi during and following extended propellant exposure.

2.4 Document internal leakage throughout extended exposure and performance test period.

3.0 DESCRIPTION OF TEST ITEM

The test program defined herein is applicable to the Prototype Shuttle OMS Helium Regulator defined by Marquardt Drawing X29200. This regulator is a single stage, pressure balanced poppet hard seat configuration featuring all metal construction. A mechanical damper (active) and pneumatic damper (inactive) are also attached. A flow limiter, P/N 29500 and two quad redundant check valve assemblies, P/N LSC 270 817-3 S/N 465 and 467 are included should additional system testing be desired.

4.0 REFERENCES

The following documents are applicable to this test program to the extent specified herein:

4.1	MIL-P-27407 (I)	Propellant Pressurizing Agent, Helium
4.2	MIL-P-27401 (B)	Propellant Pressurizing Agent, Nitrogen
4.3	MSC-PPD-2A	Propellant, Nitrogen Tetroxide, Inhibited
4.4	MIL-P-27404	Propellant, Monomethylhydrazine
4.5	O-T-620, GRADE 1.1.1 INHIBITED	Cleaning Compound, Solvent, Trichloroethane
4.6	TT-I-735 Grade A	Isopropyl Alcohol
4.7	ASTM D1193-56	Distilled Water
4.8	MSFC-SPEC 237	Freon TF
4.9	TMC Drawing X29200	OMS Pressure Regulator, Prototype Test

5.0 GENERAL TEST CONSIDERATIONS

5.1 Cleanliness

5.1.1 Handling and installation of the regulator shall be performed in a manner to insure cleanliness of the interior surfaces of the regulator.

5.1.2 All test fluids should be passed through a filter of at least 25 micron nominal rating located upstream of the regulator.

5.1.3 Keep the regulator sealed in a clean plastic bag during storage and shipment.

5.1.4 Do not pass any cleaning fluid through the regulator.

5.1.5 Clean and lubricate (Krytox or Equiv) the threads of each regulator male fitting before installation.

5.2 Instrumentation and Facility

The instrumentation and vacility requirements will be specified with each test description. A general facility-test-item-flow diagram is shown in Figure 1.

5.3 Documentation, Data Identification, Storage and Processing

Provide TMC with a copy of test conditions, actuations and performance results. Follow NASA-JSC procedures to the extent required to obtain, identify and process the data sought by this test plan.

5.4 Testing

Detail Test requirements are defined in:

Appendix A Performance Test

Appendix B Propellant Exposure Test

Appendix C Extended Performance Test

Appendix D Internal Leakage Test

Appendix E Simulated Blowdown

5.5 Pressure Limits

5.5.1 Do not apply pressure to:

Inlet chamber in excess of 4000 psig

Outlet chamber in excess of 500 psig

Actuator chamber in excess of 300 psig

FACILITY - TEST ITEM FLOW DIAGRAM
TOTAL REQUIREMENT
DETAIL SCHEMATICS IN FIGURES A-1, B-1 & D-1

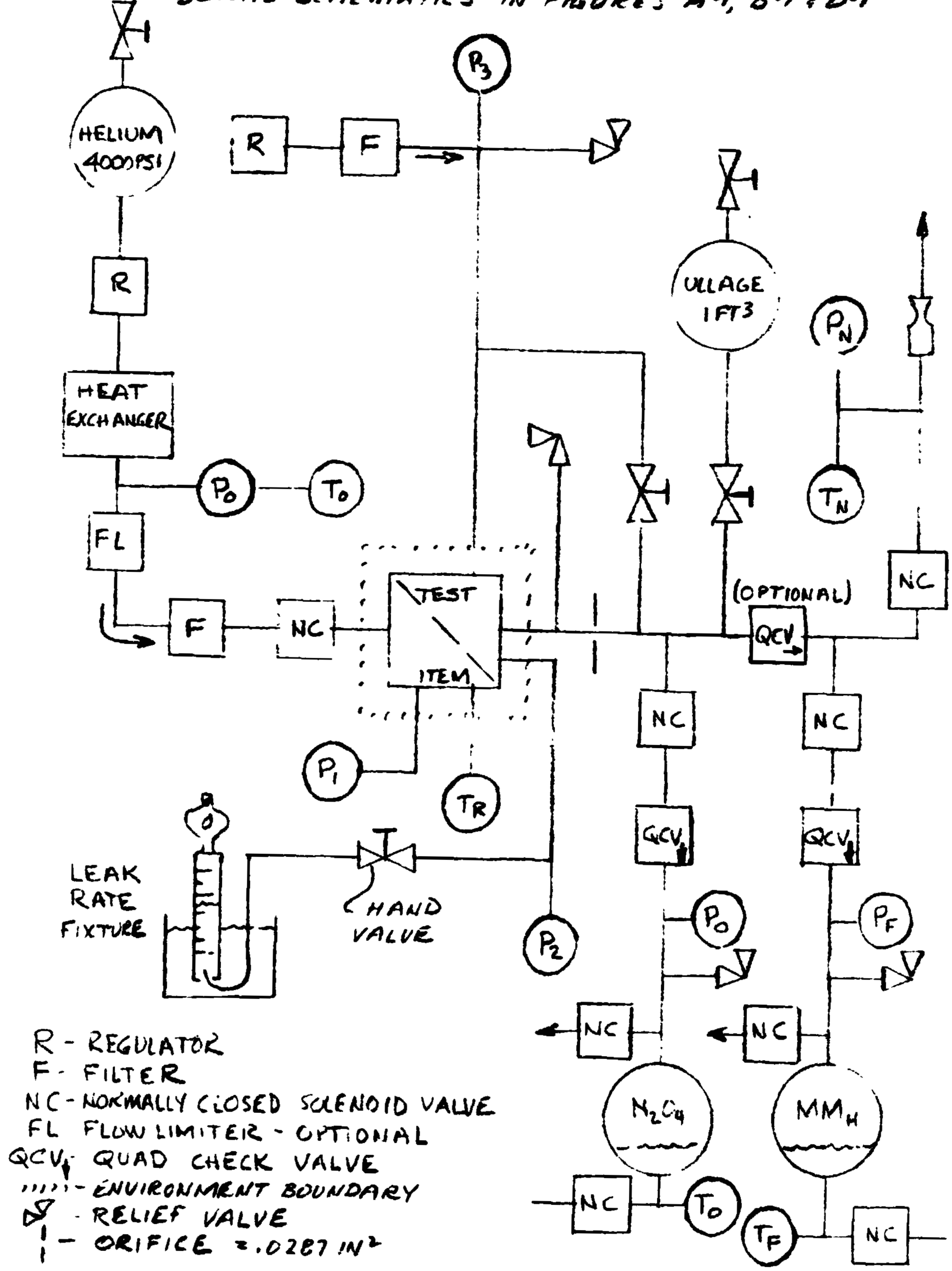


FIGURE 1

TMC A1683-1

APPENDIX A
PERFORMANCE TEST

A-1.0 TEST SETUP

Install the regulator as shown in Figure A-1. For the Slam Start tests at 4000 psi, the flow limiter or an orifice to simulate the flow limiter must be used (Q = 966SCFM) For other tests the flow limiter and/or check valve assembly is optional or may be an alternate setup to evaluate the flow limiter and check valve effects on the base line regulator performance.

The normally closed solenoid isolation valves should have opening and closing times less than 200 ms. Relief valves should be as large as possible consistent with the flow passage to which it is attached. The flow nozzle size will vary to yield steady state flow conditions of 44, 265, and 340 SFM. The test fluid shall be helium.

A-2.0 INSTRUMENTATION REQUIREMENTS

Parameter	Symbol	Range	Accuracy (F/S) - Type Instrument or Readout
Supply Pressure	P_0	0-4000 psig	±1% Gage
Inlet Pressure	P_1	0-4000 psig	±1% Gage; ±.3% 1000 Hz X-ducer*
Outlet Pressure	P_2	0-600 psig	±1% Gage; ±.3% 1000 Hz X-ducer*
Regulated Pressure	P_3	0-300 psig	±1% Gage; ±.3% 1000 Hz X-ducer*
(Sensing)		230-250	& ±.02% Precision Pressure Gage
Nozzle Pressure	P_N	0-400 psig	±1% Gage; ±.3% 1000 Hz X-ducer*
Supply Temperature	T_O	±200°F	±2% Thermocouple - Bristol
Regulator Skin Temperature	T_R	±200°F	±2% Thermocouple - Bristol
Nozzle Temperature	T_N	±200°F	±2% Thermocouple - Bristol
Pushrod Travel (Stroke)	S	0-.040 IN	±3% 1000 Hz LVDT* (1)

*Oscillograph recorders or equivalent. All recorders time correlated.

(1) Use Schaevitz Model CAS-2500RLT Signal Conditioning. TMC will furnish the transducer.

A-3.0 FACILITY FLOW CHECK

A facility flow test shall be performed to determine the dynamic stability of the test unit inlet pressure P_1 at the nominal flowrate (265 SCFM) and to check the regulator set point (P_3). With the upstream (U/S) isolation valve closed and both downstream (D/S) isolation valves open, regulate the supply pressure P_0 to 4000 ±50 psig. Open the U/S valve, and then close the D/S valve five seconds after constant flow has been established as indicated by P_N not varying more than ±1 0 psi. The inlet pressure P_1 shall not drift more than ±25 psi and the pressure oscillation shall not exceed 10 Hz and 20 psi (peak-to-peak) during the constant flow condition. Record pressure-time relationship of P_1 and P_3 during the entire test beginning five seconds prior to opening the U/S valve and five seconds after closing the D/S valve. Also record steady state values of P_0, T_0, P_1, P_2, P_3, T_N, T_R, P_N, and S. The regulated steady state pressure P_3 and mass flowrate $\dot{M}_N$ shall be baseline values.

A-4.0 SLAM START

Install an orifice in the supply line to simulate the flow limiter. Vent the system down to one atmosphere with upstream valve remaining closed. Close the last downstream valve. Simulate a slam start by opening the U/S valve and charging the volume at the nominal supply pressure (P_0) of 4000 ±50 psig. Record the time-pressure relationship of the inlet and outlet pressures P_1 and P_3 during the entire test, beginning five seconds prior to opening the U/S valve and five seconds after regulator lockup. A system charging time in excess of 1.0 seconds will verify the operation of the flow limiter orifice.

A-5.0 PERFORMANCE TEST

A-5.1 Conduct a series of regulator performance tests at ambient (70 ±25°F) temperature at supply pressures (P_0) of 4000, 2000, 680 and 400 psig ±3%.

A-5.1.1 With the facility U/S valve open, the inlet pressure P_0 set and the ullage volume pressurized, open the facility D/S valve to start flow. Record the pressure-time relationship of P_1 and P_3 during the entire test beginning five seconds prior to opening the D/S valve and five seconds after regulator lockup. Flow shall be terminated by closing the D/S facility valve after five seconds of constant flow as indicated by P_N not varying more than ±1.0 psi. Also record steady state values of P_0, T_0, P_1, P_2, P_3, T_B, T_N, P_N, and S. Flow data shall be sufficient to calculate mass flowrate within 3%. Pressure and temperature measurement accuracy shall be as specified in A-2. Terminate test as soon as possible to conserve helium.

A-5.1.2 Perform a leakage test in accordance with Appendix D-3.1.

A-5.2 Conduct a series of regulator performance tests at elevated (+150 ±25°F) fluid and test item temperatures at supply pressures (P_0) of 4000, 2000, 680 and 400 psig ±3%.

A-5.2.1 Perform a leakage test in accordance with Appendix D-3.3.

A-5.3 Conduct a series of regulator performance tests at cold (-150 ±25°F) fluid and test item temperatures at supply pressures (P_0) of 4000, 2000, 680, and 400 psig ±3%.

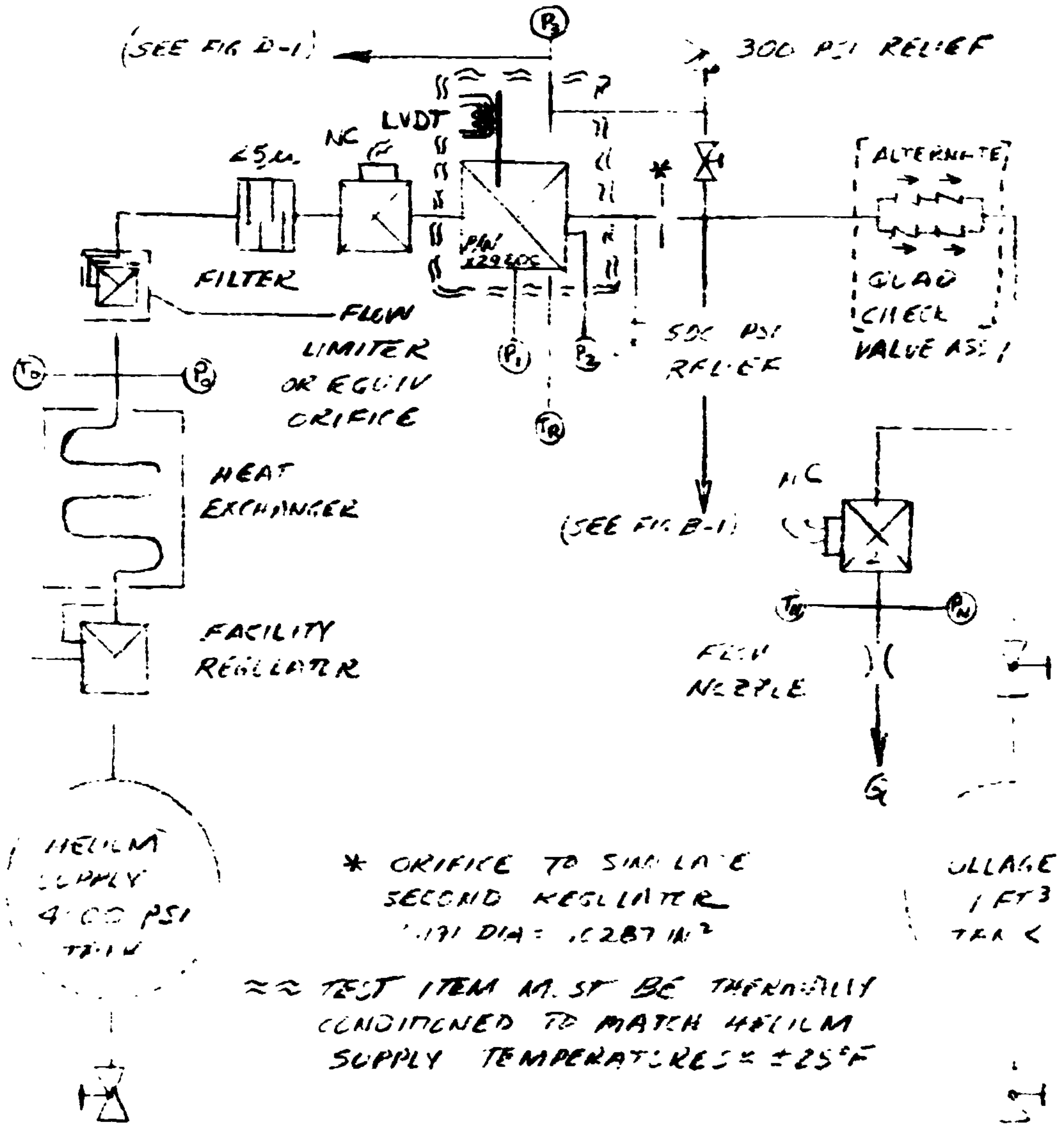
A-5.3.1 Perform a leakage test in accordance with Appendix D-3.2.

A-5.4 Replace the 265 with the 44 SCFM flow nozzle and conduct the tests of 5.1.

A-5.5 Replace the 44 with the 340 SCFM flow nozzle and conduct the tests of 5.1.

APPENDIX A

PERFORMANCE TEST SCHEMATIC



LINE SIZES

BETWEEN COMPONENTS	DIA (IN)	LENGTH (IN)
FILTER & TEST ITEM	.75	12-[illegible]
TEST ITEM & CHECK VALVES	.75	1 [illegible]
CHECK VALVES & ULLAGE & VALVE 2	.75	12 [illegible]
VALVE 2 & FLOW NOZZLE	[illegible]	[illegible]
ACTUATOR (P_3) & OUTLET LINE	.25	20-24

FIGURE A-1

APPENDIX B
PROPELLANT EXPOSURE TEST

B-1.0 TEST SETUP

Install the regulator as shown in Figure B-1. If this setup is used in the same area as the setup in Figure A-1 be sure that the upstream facility helium supply components are excluded to prevent their effects to long term propellant exposure from influencing the regulators. The system up to the first isolation valve is identical to the previous setup to permit alternate propellant exposure and performance testing to be accomplished without plumbing changes.

The propellant tankage system size should be minimized from a safety point of view and need only contain enough liquid propellant (approximately .63 lb) to maintain system vapor pressure at ambient (above 70°F) temperature conditions with an assumed vapor loss of 100 SCCH (72000 SCC per month plus 3280 CC tank volume). A controlled leakage rate device (an orifice of ≈.0007 inch diameter) should be used in lieu of the check valve assembly. Thermal condition only the test item to -150°F as specified in the test matrix while maintaining the propellant tanks above 70°F at all times during the exposure test. Use propellants with the maximum allowable water content.

B-2.0 INSTRUMENTATION REQUIREMENTS

Parameter	Symbol	Range	Accuracy (F/S) & Type Instrument or Readout
Oxidizer Pressure	P_0	0-100 psig	±1% Gage
Fuel Pressure	P_F	0-100 psig	±1% Gage
Regulator Pressure	P_3	0-20 psig	±.02% Precision Pressure Gage
Oxidizer Temperature	T_0	0-100°F	±2% Thermocouple - Bristol
Fuel Temperature	T_F	0-200°F	±2% Thermocouple - Bristol
Manifold Temperature	T_M	0-200°F	±2% Thermocouple - Bristol
Regulator Temperature	T_R	±200°F	±2% Thermocouple - Bristol

B-3.0 EXPOSURE TEST

B-3.1 AMBIENT SOAK

Connect the regulator to the loaded propellant system by opening the oxidizer isolation valve. Continuously monitor all temperatures and the regulator pressure (P_3) for two days under ambient (tanks above 70°F) temperature conditions. Open the fuel isolation valve (both vlaves now open) and monitor for three days. Repeat following each thermal cycle.

B-3.2 THERMAL CYCLE

After the one week ambient soak test, cool the regulator only to -150°F as indicated by T_R. Hold for 1-2 minutes and then allow to warm unaided. Repeat following each ambient soak cycle.

B-3.3 Repeat ambient and thermal tests for one month.

B-3.4 PERFORMANCE TEST

Conduct a series of regulator performance tests at cold (-150 ±25F) fluid and test temperatures at supply pressures (P_0) of 4000, 2000, 680 and 400 psig ±3% in accordance with the run procedure of Appendix A-5. - 1. Conduct this test following each month of propellant exposure.

B-3.5 LEAKAGE TEST

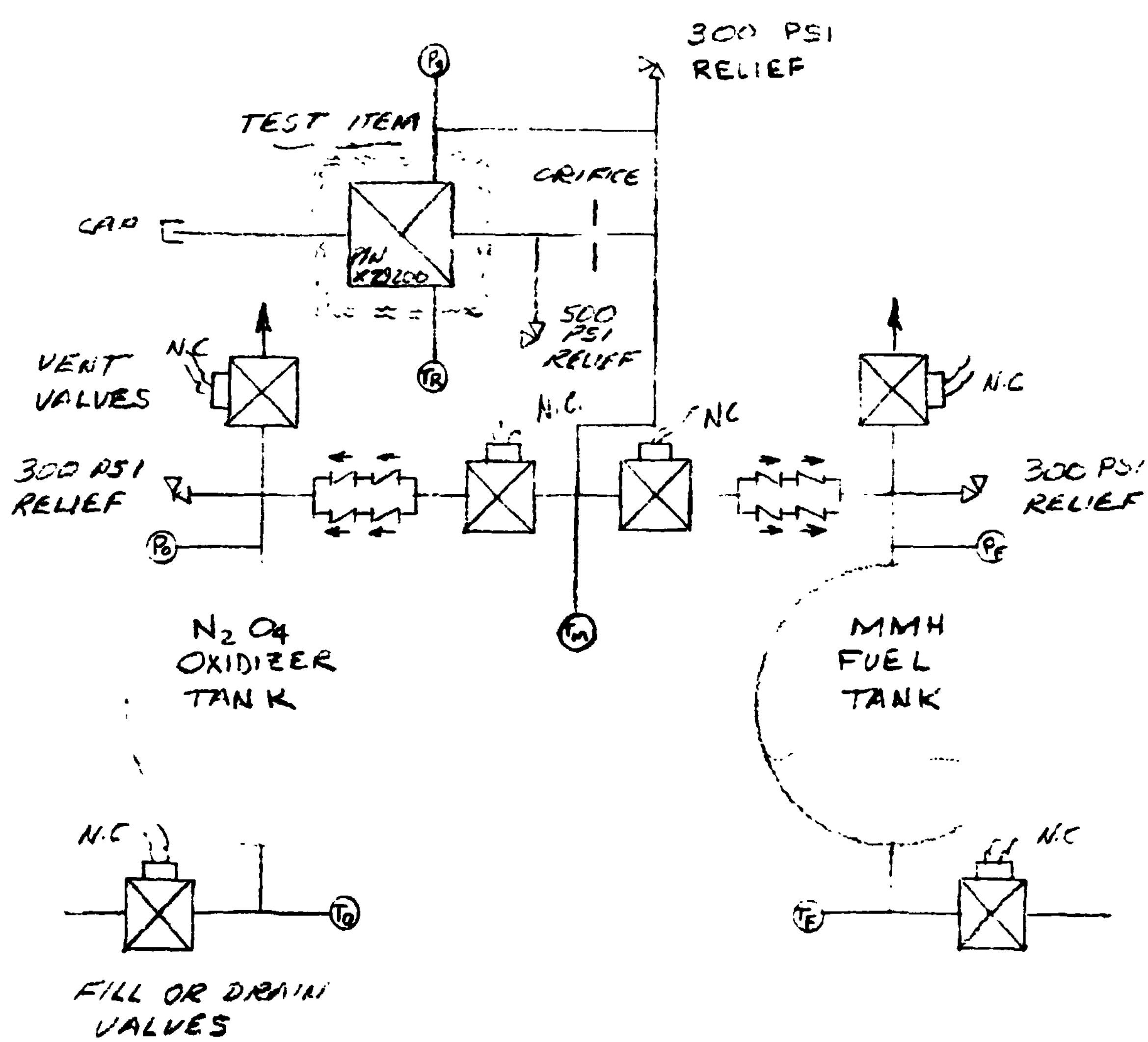
Perform a leakage test in accordance with Appendix D-3.2.

B-3.6 EXTENDED EXPOSURE

Repeat the tests of B-3.1 thru B-3.4 each month for a total of TBD months.

APPENDIX B

PROPELLANT EXPOSURE TEST SCHEMATIC



TANK VOLUME ≈ 200 IN^3
LINE VOLUME UPSTREAM OF REGULATOR ≈ 20 IN^3
LINE VOLUME DOWNSTREAM TO CHECK VALVES ≈ 200 IN^3

PROPELLANT TANK PRESSURIZATION NOT REQUIRED FOR TEST, BUT MAY BE ADDED TO FACILITATE FILL & DRAIN PROCEDURES
≈≈ TEST ITEM MUST BE THERMALLY CONDITIONED

FIGURE B-1

APPENDIX C
EXTENDED PERFORMANCE TEST

C-1.0 TEST SETUP

Use the same test setup shown in Figure 1A.

C-2.0 INSTRUMENTATION REQUIREMENTS

Use the same instrumentation specified in Appendix A.

C-3.0 EXTENDED PERFORMANCE TEST

C-3.1 Conduct a series of regulator performance tests at hot (+150 ±25°F) fluid and test item temperatures at supply pressures (P_0) of 4000, 2000, 680 and 400 psig ±3% in accordance with the run procedure of Appendix A-5.1.1.

C-3.2 Perform a leakage test in accordance with Appendix D-3.3.

C-3.3 Cycle the regulator 2000 times at 400 psig inlet pressure and hot 150 ±25°F temperature conditions.

C-3.4 Perform a leakage test in accordance with Appendix D-3.3.

C-4.0 Conduct the extended performance test each month if possible. If conducted less frequently increase the number of cycles of C-3.3 by the months between testing. Continue testing until 10,000 cycles have been accumulated.

APPENDIX D
INTERNAL LEAKAGE TEST

D-1.0 TEST SETUP

Install the regulator as shown in Figure D-1. The downstream isolation valve must remain closed. The downstream (outlet) common line to the sense port must be valved off. Two regulated helium sources are required. The actuator side must remain protected with a relief valve.

The leakage rate fixture may consist of any graduated cylinder up to 50 Ml and may or may not incorporate a suction bulb for refilling with water. The system should be readable to within 0.25 Ml.

The leakage rate fixture should not be connected into the regulator outlet instrumentation pressure line UNTIL the regulator has been closed - 280 ±15 psig on Actuator (P_3).

D-2.0 INSTRUMENTATION REQUIREMENTS

Parameter	Symbol	Range	Accuracy (F/S) & Type Instrument or Readout
Inlet Pressure	P_1	0-4000 psig	±1% Gage
Regulator (Sensing) Pressure	P_3	0-300 psig	±1% Gage
Regulator Temperature	T_R	±200°F	±2% Thermocouple - Bristol
Stop Watch	t	0-300 sec	±1 sec
Graduated Cylinder	ΔMl	0-50 Ml	±.12 Ml readability

D-3.0 INTERNAL LEAKAGE TEST

D-3.1.0 Ambient Temperature Test

D-3.1.1 Pressurize the actuator (P_3) to 280 ±15 psig. Pressurize the inlet (P_1) to 400 ±12 psig. Attach the leakage rate fixture to the outlet pressure (P_2) instrumentation port. Fill 80% of the graduated cylinder with water. Record starting time and fluid level. Record fluid level after five minutes. The change in fluid volume times 12 is the leakage rate in SCCH.

D-3.1.2 Without changing the actuator (P_3) pressure increase the inlet pressure to 1000 ±30 psig. Record leakage for a five minute interval.

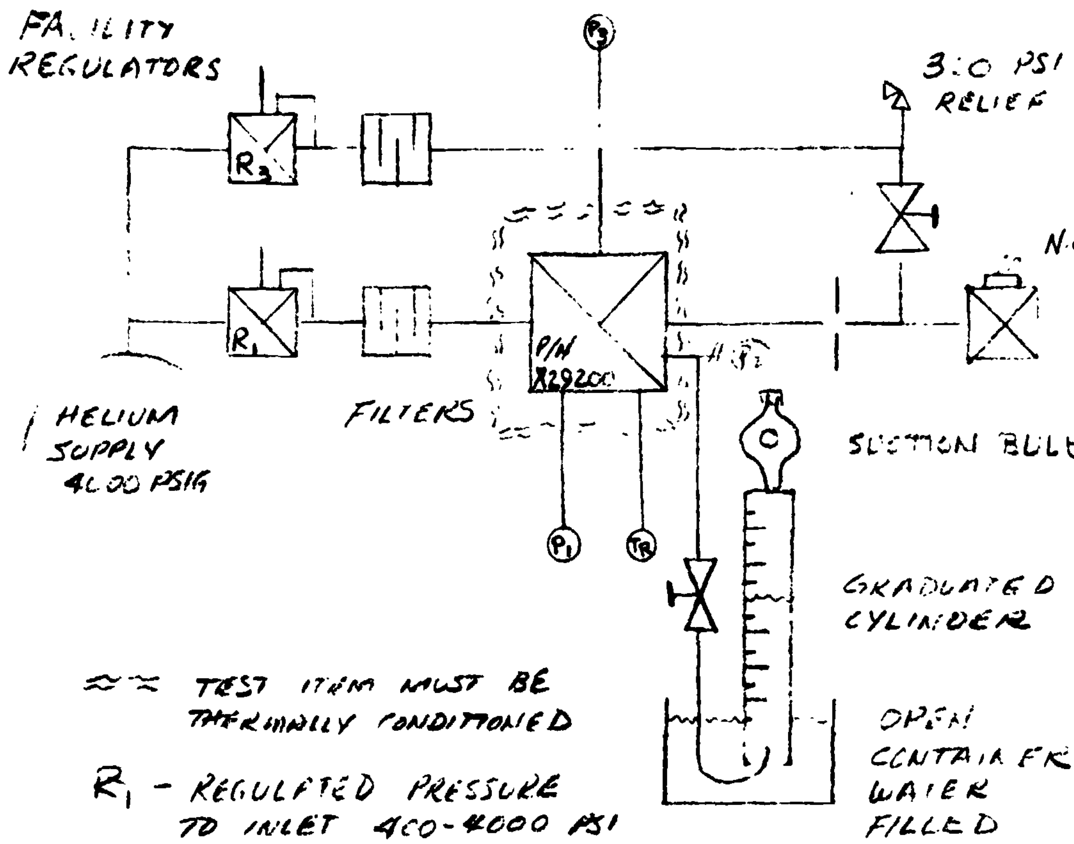
D-3.1.3 Without changing the actuator (P_3) pressure increase the inlet pressure to 2000 ±50 psig. Record leakage for a five minute interval.

D-3.1.4 Without changing the actuator (P_3) pressure increase the inlet pressure to 4000 ±100 psig. Record leakage for a five minute interval. Disconnect the leakage rate fixture, then discharge the inlet pressure regulator (R_1). Finally, discharge the actuator pressure regulator (R_3).

D-3.2 Conduct the leakage test of D-3.1 at a -150 ±25°F test item temperature.

D-3.3 Conduct the leakage test of D-3.1 at a +150 ±25°F test item temperature.

INTERNAL LEAKAGE TEST SCHEMATIC



≈≈ TEST ITEM MUST BE THERMALLY CONDITIONED

R_1 - REGULATED PRESSURE TO INLET 400-4000 PSI

R_3 - REGULATED PRESSURE TO ACTUATOR 280 ± 15 PSI

* R_3 MUST PRECEED R_1 AT START

R_1 MUST BE DISCHARGED BEFORE R_3 AT END OF TEST

* <u>CAUTION</u> - ATTACH LINE TO LEAKAGE RATE FIXTURE ONLY WHEN $R_3 \geq 250$ psig. ISOLATE BEFORE DISCHARGING EITHER R_1 OR R_3

FIGURE D-1

APPENDIX E
SIMULATED ABORT

E-1.0 TEST SETUP

This test can be accomplished with the test schematic shown in Figure A-1 of Appendix A with the following additional requirements:

The helium supply pressure and temperature are variable with time, i.e. pressure and temperature decrease as run time accumulates. The rate of change in pressure (4000 psig to 400 psig) and temperature (500°R to 270°R) must simulate a flight tank expulsion profile (Figure E-1). Change the constant flow nozzle to permit 343 SCFM flowrate.

D-2.0 INSTRUMENTATION REQUIREMENTS

The instrumentation defined in Appendix A is applicable. Continuously record the abort test.

E-3.0 SIMULATED ABORT

Open the downstream isolation valves and ullage tank inlet valve and vent atmospheric pressure with the upstream valve remaining closed. Set the upstream pressure at 4000 psig and proceed to open the upstream isolation valve. Reduce pressure (P_0) and temperature (T_0) in accordance with the time history plot and continue test until 400 psi and 270°R are attained.

PRESSURIZATION SYSTEM
ABORT PRESSURE & TEMPERATURE PROFILE

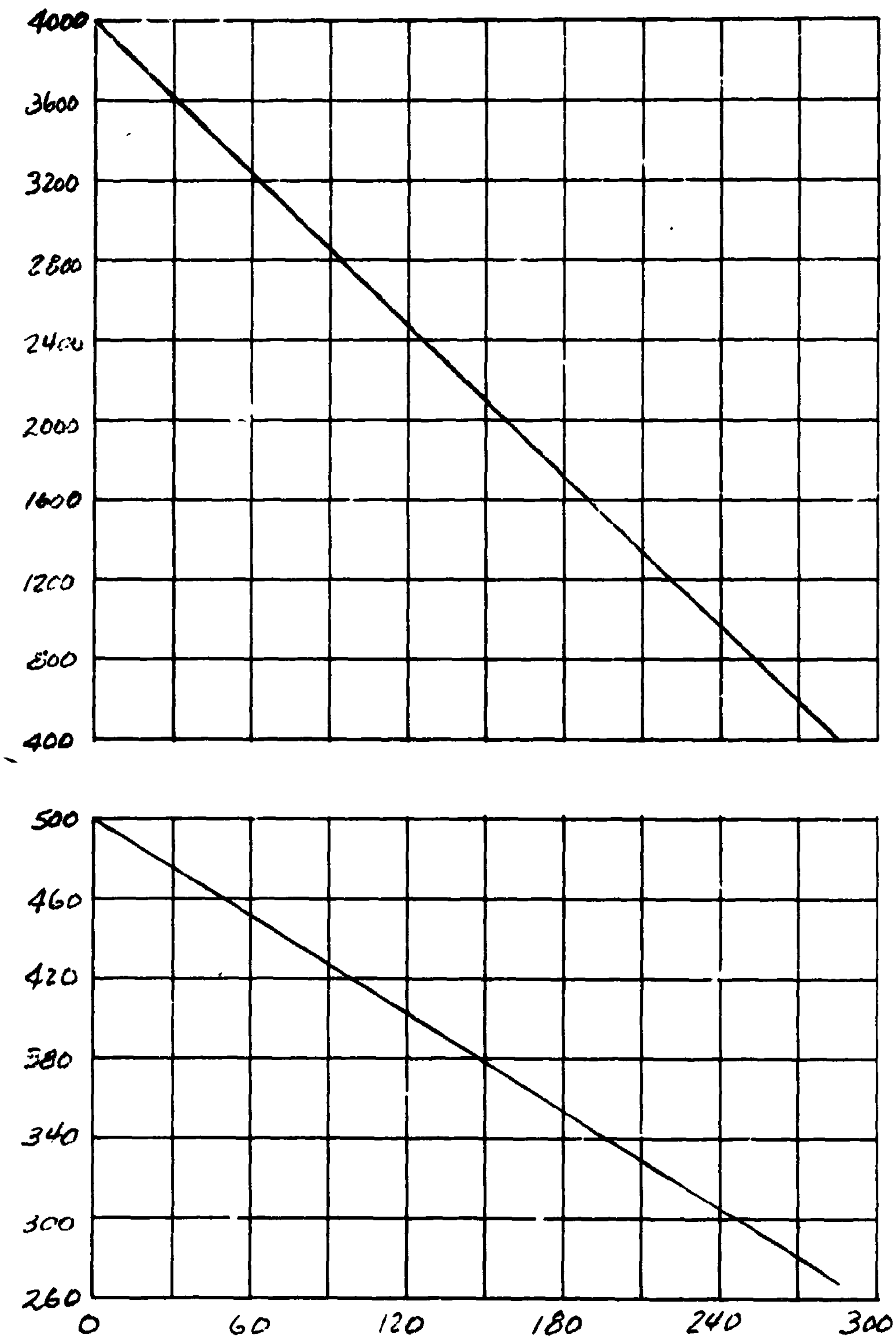


FIGURE E-1