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FINAL REPORT

CONTRACT NAS8-27568

LONG-LIFE, SPACE-MAINTAINABLE NUCLEAR STAGE REGULATORS AND SHUTOFF VALVES

TO GEORGE C. MARSHALL SPACE FLIGHT CENTER

MARCH 1972

AEROJET NUCLEAR SYSTEMS COMPANY
A DIVISION OF AEROJET-GENERAL

FOREWORD

The design study described in this report was conducted by the Aerojet Nuclear Systems Company under NASA contract NAS 8-27568. Mr. Kenneth Anthony of the National Aeronautical and Space Administration, Marshall Space Flight Center, Alabama, 25812 was the NASA Project Manager.

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I. SUMMARY

The overall objective of the program was to generate sketches during Phase I of the most promising concepts of nuclear stage regulators and shutoff valves that can be designed with characteristics which render them suitable for being space maintainable and capable of operation under the conditions set forth in the RFQ. The six most promising valve, regulator and remote coupling concepts were then selected and subjected to further investigation during Phase II.

During Phase I, following a literature review, specific design criteria was established as the basis for designing and selecting valve and regulator concepts. The concepts generated took into consideration the problems associated with the selection of materials and design configurations (including tolerances) to ensure that the designs are capable of meeting the performance requirements under specified environmental conditions. Problems associated with leakage, wear and operating life, contamination, storage life, temperature, pressure, radiation, and vibration environments, and remote in-space maintenance were solved for each concept.

During the Phase I effort, concept design sketches of eleven valves, six pressure regulators and five remote couplings were completed. These sketches ranged from radical design concepts to current "state-of-the-art".

For the Phase II effort, three valves, one pressure regulator and two remote couplings were selected for preliminary design. This selection represented the more radical design concepts from the Phase I package of twenty concepts.

The three valves utilized unique methods of performing the shutoff function. One valve design has no moving parts because shutoff sealing is accomplished by an electromagnetic field which ionizes the flowing fluid. This principle has been demonstrated by electromagnetic pumps in a liquid metal system (SNAP-8 Program). Another valve uses liquid metal to obtain sealing. In the third valve, high sealing forces are generated by heating and expanding trapped hydrogen.

The pressure regulator selected is an electronically controlled, electro-mechanically operated, single stage valve. This has been made possible by advancements of radiation hardened solid state miniature electronics components. Thus, the complexity is in the electronic circuitry rather than in the mechanical devices. Compared to a conventional regulator system the ANSC design results in

less weight, increased reliability, increased performance flexibility, and multipurpose application. The flexibility and multipurpose capability results because the valving can be programmed to various positions; thus, the tank pressure can be varied in flight, the unit can be shut off, or the flow can be reversed.

The two remote couplings selected feature the minimization of weight and mechanical complexity for the flight article. The objective was to put the complexity into the manipulator and manipulator tooling. One concept uses a low melting temperature metal alloy which is injected into the joint cavity. Upon solidification, the alloy provides a seal and a structural joint. The second concept is based upon the differential thermal expansion of the coupling mating parts. At thermal equilibrium, over the operating temperature range, there is a predetermined interference between the male and female parts. Sealing is achieved by the interference loading.

Since the Phase II designs represent the more radical design concepts, it was concluded that a fabrication/test program would be required to quantify the variable parameters and to establish feasibility of each of the six concepts.

This report provides all of the information developed on the program including all drawings and the detail calculations.

II. TECHNICAL DISCUSSION

This section covers the work done in two parts. Phase I includes the work leading to concept selection, and Phase II covers the preliminary design of those selected concepts.

A. PHASE I, CONCEPT SELECTION

1. Requirements Analysis

The objective of the requirements analysis task was to establish the performance, environmental, and storage requirements which must be met and those design criteria (e.g., materials, producibility, and service) which produce guidelines for design. The information needed to obtain this objective was extracted from existing literature, evaluated, and then combined with the pertinent information from the RFQ "scope of work" into one suitably organized document for the valve, the pressure regulator, and the remote couplings. These design goals were originated and then revised following a meeting with MSFC. They were used as the requirements for the design sketches formulated during Phase I and are presented as Appendixes A, B, and C.

2. Literature Review

A brief synopsis of major references reviewed was prepared with respect to shutoff valves, gaseous pressure regulators, remote coupling and uncoupling devices, seals and sealing techniques. The synopsis of the literature review is shown in Appendix D. Many other references such as supplier bulletins, supplier drawings and text books have been reviewed but not noted as references.

A review was made of the nuclear shuttle system definitions studies by LMSC, MDC and NAR. The review showed that the MDC final report (G-2134, Part B, Class 3RNS Book, Volume II, System Definition) presented the most comprehensive information on valve requirements such as valve size, description and design conditions. The information permitted a more judicious selection of the kinds of valves to be studies during Phase I. Appendix E presents information taken from the MDC report. Appendix F presents the propulsion system schematic proposed by MDC.

3. Problem Areas

The following section presents a summary of the most important problem areas considered during the Phase I effort.

a. Seal Leakage and Wear

The paramount determinant of valve success is the rate of leakage. Thus, the cyclic life must be considered in terms of resultant leakage. There are many inter-related factors that determine the leakage rate that will be obtained with any sealing arrangement. These must be identified and their relationship structured for proper evaluation.

(1) <u>Shutoff Seals</u>

The shutoff seal is the heart of the valve. Attainment of 1×10^{-7} sccs helium leakage per circumferential inch of seal is a particularly critical problem. High sealing-interface loads are required to achieve this leak rate; therefore, careful consideration must be given to friction and wear parameter "tradeoffs" to assure that the long life requirements are also achieved. To avoid the high sealing-interface loads, positive sealing by forming a bond between the valve element and the seat were considered.

Factors that determine the rate of leakage are: 1) pressure differential across the sealing interface, 2) the fluid being sealed, 3) physical material properties of the seal and seat, 4) interface contact length, 5) interface contact width, and 6) surface finish. For the shutoff seal, other factors must also be considered; e.g., interface geometry, constraint of the seal material, peak stresses from impact, and seal stress levels in areas other than the contact interface.

(2) <u>Static Seals</u>

Attainment of low leakage rates requires minimization of the number of flow paths. One way this can be accomplished is to use semi-separable and nonseparable seals at all interfaces other than the shutoff seal. The most critical problem which this approach presents is in the area of remote "changeout" of the component and component parts.

One solution is to use automatic orbital welding of the valve to the inlet and outlet lines to minimize exit and entrance joint leakage. The main problem area is in the detachment and subsequent reattachment of a replacement module while maintaining system cleanliness. The solution to the problem lies in development of a remote capability for cutting and welding.

b. Thermal

Thermal gradients during chilldown transients, as well as during nominal steady-state conditions, are potential problem areas because of thermal stresses induced in the components. Although all component parts are subjected to thermal stress, the component housing or body in most designs is the most severely affected. Fluid heating rates, solar heating, nuclear-heating, "soak-backs" from adjacent hardware, nonsymmetrical housings, supports, and mechanical restraints all contribute to the magnitude of thermal stresses. These stresses can be sufficiently high to cause severe distrotion which may result in seal-seat misalignment, binding of moving parts, and/or opening of leakage paths past seal assemblies.

Solution of the thermal-gradient problem cannot be defined in specific terms. Each design requires evaluation, and the correct approach for one component may not be entirely applicable to another.

c. Deep-Space Pressure (Cold Welding)

One of the primary problem areas of a space environment is cold welding of contacting parts that must have relative movement during use. The adhesion mechanisms are accentuated by increased temperatures, surface-film instability, and increased surface loading. Oxygen and other gas species contribute to the prevention of adhesion of contacting parts designed for relative movement. Adhesion is prevented by the integrity of the surface film upon the contact surface. The sliding of surfaces and other mechanisms disrupt the residual contamination effects of the gas species. Thus, sliding surfaces are more subject to adhesion thatn "push-pull" contact surfaces. The solid-film lubricants generally used to prevent adhesion have a finite life; however, spacecraft experience justifies the use of MOS_2 and WS_2 for surface preparation. The life of the solid-film lubricant used is dependent on a combination of mechanical, physical, and chemical processes.

The primary approach in finding the general solution to this problem will be to select hard materials and/or coatings that have complete insolubility for the contact areas. Pressure welding can be avoided by designing these mating parts with a contact stress less than 50% of yield strength. The one specific area where this approach is most difficult is the shutoff seat and seal required to meet th 1×10^{-7} sccs helium leakage requirements. However, seal design and test experience has demonstrated that sealing interfaces consisting of gold-plated surfaces in contact with electrolized surfaces will prevent surface adhesion.

d. Radiation

Radiation effects upon fluid control system components present both unique problem areas and some possible beneficial aspects with respect to increased strength. The phenomena can be categorized into areas of gamma dosage or nuclear heating and fast neutron dosage (nvt).

Considerations involving aerospace fluid control system configurations must take into account the results of expected radiation dosage. While gamma rates do not degrade physical properties of metals directly, the resultant molecular excitation increases the bulk temperature of the component or part. Sufficient gamma rates, combined with other sources of heating, can create severe problem areas for sophisticated hardware (e.g., loss of material strength at elevated temperatures, or an excessive expansion of parts, inducing leakage and binding). Large neutron flux dosages (10^{15} - 10^{22} nvt) have a definite effect upon material properties of most metal alloys and organic materials. In general, physical properties of metal alloys (e.g., yield strength, ultimate strength, and shock sensitivity) increase while ductility and fatigue strength decrease, which is very similar to work-hardening of common stainless steels.

Degradation of commonly used elastomers (e.g., Teflon, Kel-F, and Viton-A) has shown these materials to be relatively unsatisfactory in extended radiation environments. However, a comparatively new class of polyimide plastics have shown stable properties at dosages of 10^8 to 10^9 Rads, indicating promise for possible seal and bearing applications. Since the component location, radiation level, and exposed duration have not been established, the conservative approach of all metal construction was selected for the Phase I design effort.

e. Line-Size Variability

Nearly all design parameters and considerations are directly affected by the size of component parts. Therefore, it is of major importance to select component configurations that are not too sensitive to changes in line size. The Phase I approach was to design the components to meet the specified requirements at the maximum size established from the literature survey. Fewer problems will be encountered when scaling down to a smaller size than when scaling up to a larger size because in most design considerations the size-effects decrease as the part size decreases. For example: (1) distrotion and relative motion between mating parts because of thermal cycling, pressure loading, and external loading is greater in a larger valve, (2) meeting the internal and external leakage requirements is more difficult in larger valves where sealing surfaces are larger and distortion is greater, (3) large components require greater actuation requirements to overcome inertia, pressure, and frictional forces, and (4) vibration considerations are more significant in the larger component because the natural frequency of a part decreases as the size increases. However, these design problems can be overcome by careful consideration of the particular concept configuration and by sufficient engineering analysis.

f. Storage Life

Environmental conditions during the storage life of component parts are usually less severe than those imposed during operating life. However, each component should be capable of satisfying all requirements of its controlling specification after being stored ten years in warehouse facilities, provided that the component has been adequately designed and proper packaging and preservation procedures have been followed.

A long storage life requirement makes it mandatory that the component design incorporate features that will not be affected during long inoperative periods. Two major problem areas that will require design effort are creep properties and galvanic corrosion.

Creep may be defined as an increase in strain in a material under a constant static load at a given temperature. The total amount of creep varies with time, while the rate of creep is a function of temperature and stress level. Under less severe conditions, material relaxation (i.e., microcreep) can cause yielding in various critical areas, subsequently resulting in malfunction of the component. For example, the valve seals may be degraded because of a loss of bolt preload at a flange seal or loss of spring preload at a poppet seal.

Solutions of the potential creep problems rely upon stress analysis in conjunction with heat transfer analysis and a knowledge of material characteristics. General guidelines concerning allowable stress levels were used for preliminary design work; however, each design will require further detailed analysis to ensure that desired performance can be maintained for the designed storage life of the component.

Galvanic corrosion occurs when two dissimilar metals are coupled in the presence of an electrolite. Galvanic corrosion also occurs when electrodes of the same metal contact different electrolites, or the same electrolite, but at different strengths. The extent of galvanic corrosion depends upon the type of metal, the electrical resistance of the electrolite, and the duration of exposure. Thus, a material combination which is acceptable for a one or two year lifetime may be unacceptable for a ten-year lifetime. The use of dissimilar metals will be avoided wherever practicable. Where it is not possible, metals were selected as close together as possible in the galvanic series. During operation and space storage, both liquid water and oxidizer are excluded. In fact, a high concentration of hydrogen in the environment will tend to retard corrosion. However, protection must be provided during storage. The usual method is to employ a more-or-less impermeable seal around the contact interfaces to prevent liquid from reaching the metal surface. Additionally, a coating (which also prevents cold welding and increases wear life) can be selected to be less subject to corrosion than the base metal.

g. Vibration

Vibration can be introduced in several ways when a component is in use. Flow-induced vibration from cavitation collapse, and vortex shedding must be considered with vibration induced through the structure. Although vibration may not be significant in short term applications, the effects over a three-year operational life may be detrimental to component functioning. In addition, vibration levels which may be acceptable in a 1/4-inch shutoff valve or regulator may be detrimental in a 20-inch shutoff valve or a 6-inch regulator. Experience has established that more components suffer reduced performance as a result of vibration testing than in any other form of environmental testing. Common causes of component failures under vibration are: (1) fretting of two interfaces surfaces, such as a poppet against a seat seal, (2) internal resonance of springs or structural supports, such as sleeve-gate supporting webs, (3) rubbing between springs, (4) physical impact between parts, such as a seal mass supported by a bellows impacting against the mating sealing surface, and (5) flow induced vibration, such as chattering of flow control elements in a regulator.

Based upon previous experience, design of vibration sensitive components can be minimized and vibration problems can be avoided through dynamic flow and vibration analyses. When necessary, the component stiffness and configuration was modified to reduce vibration sensitivity. Examples of methods considered were: (1) use of unsymmetrical parts, such as five as opposed to four sleeve element supporting webs to reduce the effects of part resonance, (2) increase stiffness, if required, to increase resonant frequency, (3) use of nested springs to prevent the load from resonating at the same frequency, and (4) the use of parallel stacked Belleville springs where friction between parallel discs can be utilized to provide spring damping.

h. Remote In-Space Maintenance

As a design goal, the valves or regulators were designed for long life so maintenance would not be required during the storage life of ten years or an operating life of three years. This would be accomplished by selecting materials which are least affected by nuclear radiation or aging and by providing adequate margins for the as-aged or irradiated condition. The NERVA materials evaluation program has made available an extensive bank of data on the properties of many materials in cryogenic and radiation environments.

Maintenance consists of replacement of the entire valve or regulator or of a component. To obtain an optimum design, the requirements for maintenance or replacement should be considered throughout the program from conception to application. Consideration of maintainability early in the design process produces an integrated design with an inherent space maintenance capability.

i. Contamination

Contamination of some form and degree is alwyas present in fluid systems. To attain a reliable flight system, fluid systems are cleaned to the most stringent cleanliness level within practical and economical constraints for a given system. This is based upon the numerous variables associated with fluid contamination and upon the lack of adequate information concerning specific effects of contamination upon fluid system components. Only a small amount of data was found to exist on the sealing characteristics of seals and seats in contaminated environments. Most of these data are for a specific system and are based upon broad assumptions and/or limitations in the existing state-of-the-art.

Particulate contaminants found in space vehicle fluid systems originate from sources that are internal and external to the system. The various sources of these contaminants have been divided into four categories: (1) manufacturing process residuals, (2) operating fluids, (3) system generated, and (4) environmental.

Contaminants remaining from manufacturing operations (e.g., metal chips, weld splatter, and lapping compounds) are very detrimental to fluid systems. However, these types of contaminants can be minimized by employment of rigid cleaning and inspection methods during the manufacturing cycle. Fluids used for final testing of a product, as well as operating fluids, are also a source of contamination if adequate filtration has not been provided. Many times test fluids are overlooked as a source of contaminants because they are assumed to be clean or the filter in the system is assumed to be clean. Particles and various contaminants generated within a system as a result of friction, wear, and fluid deterioration are another source of system contamination. These system-generated contaminants are difficult to control since their origin includes the attrition and breakdown processes of all parts of the system after it has been designed, assembled, and tested. The cleanliness of a system can only be as clean as the environment in which it was assembled. Inevitably, airborne contamination will enter a system whenever the system is not adequately protected. A means of controlling this type of contamination is to provide a "clean" room where the air has been filtered to a specified level of cleanliness. The degree of cleanliness required varies for each application, and the contaminants from all four categories must be controlled within limits specified to achieve the desired results.

4. Concepts Considered

During the Phase I effort, concept design sketches of eleven valves, six pressure regulators and five remote couplings were completed. A complete listing of the concepts completed is presented in Tables I, II, and III. In this section each design is described and sketch included as a figure. The "Concept No" refers to a method used to identify the concept in the tradeoff matrices discussed in Section II.A.5.

a. Valves

Each of the eleven sketches depict solutions to the problems of meeting the 1×10^{-7} standard cubic centimeters per second leakage

TABLE I

VALVES

ANSC Drawing Number	Concept No.	Drawing Title	Drawing Depicts
1139890	1	Solenoid operated and shutoff valve with filled tank removal capability.	Remote removal of valve from manifold.
1139892	2	8-inch in-line poppet tank valve motor operated - remote replacement valve from a pressurized system.	Spherical metal shutoff seal.
1139893	3	In-line poppet valve, motor operated.	Gaseous hydrogen energized poppet valve, shutoff seal.
1139894	4	Solenoid operated multiple poppet valve.	Solenoid operated valve with multiple poppet valve arrangement for redundant module step control package.
1139895	5	3/4-inch electromagnetic in-line valve - no moving parts.	Since valve has no moving parts, valve can be welded into the line.
1139896	6	90° balanced poppet with electromagnetic skirt seal.	On-Off valve. Position control valve.
1139897	7	Balanced poppet with liquid metal skirt and poppet seals - motor operated.	On-Off valve. Low leakage capability. Similar to 1139893 except for seals.
1139899	8	90° poppet valve - motor operated with spherical metal shutoff seal.	Low leakage.
1139012	9	8-inch in-line poppet tank valve, motor operated.	Remote replacement of the actuator and ball screw drive mechanism.
1139930	10	Poppet valve - remote actuator removal.	Remote removal of actuator. Rack and pinion. Low leakage poppet shutoff seal, hard metal seal, 304 CRES on 7075 AL.
1139931	11	10-inch visor valve - linear seal withdrawal - motor operated.	Spherical metal main shutoff seal.

TABLE II

PRESSURE REGULATORS

ANSC Drawing Number	Concept No.	Drawing Title	Drawing Depicts
1139923	1	Regulator - flow control.	Pressure regulator.
1139924	2	Regulator - pressure or flow control.	Pressure regulator concept. Shutoff valve concept. Remote replacement of actuator and subcomponent concept.
1139925	3	Regulator - pressure or flow control.	Pressure regulator concept. Shutoff valve concept.
1139926	4	Regulator - dome load.	Pressure regulator concept. Shutoff valve concept.
1139928	5	Regulator - dome load.	Upgraded "state-of-the-art" regulator.
*1139929	6	Regulator, pressure	Pressure regulator concept with split element to provide tank shutoff capability during coast.

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* Late concept presented to MSFC but not covered in the Matrix or Ranking effort of Appendixes G through J.

TABLE III

REMOTE COUPLINGS

ANSC Drawing Number	Concept No.	Drawing Title	Drawing Depicts
1139920	1	Remote coupling - structural joint seal.	Remote coupling of a line to a line, or to a manifold. Remote coupling of a component to a line or to a manifold.
1139921 (Sheet 1 and Sheet 2)	2	Low melting alloy structural coupling and seal.	Remote coupling and sealing of line to a line or to a manifold. Remote coupling and sealing of a component to a line or to a manifold.
1139922	3	Thermal interference joint.	Remote coupling of a line to a line or to a manifold. Remote coupling of a component to a line or to a manifold.
1139927	4	Remote coupling - in-line.	Remote coupling of a regulator to a line. Remote coupling of a shut-off valve to a line.
1139891	5	Concept - In-line remote coupling.	

rate, the remote coupling and uncoupling of the valve, or the remote changeout of components and component parts of the valve. The sketches ranged from radical design concepts to current "state-of-the-art."

1139890, Concept No. 1, Figure (1) - This is a solenoid shutoff valve that could be removed from a manifold without draining the inlet or outlet cavities. The secondary valve performs a quick disconnect function to prevent loss of propellant during removal of the valve. This concept is applicable only to the smaller valve sizes.

1139892, Concept No. 2, Figure (2) - This concept is similar to 1139012 but with the capability of removing the valve from the tank without depressurizing or draining the tank. The concept contains an extra poppet that seals tank fluid under pressure prior to final removal of the basic valve. The design is readily adaptable to a side mounted actuator that could be replaced without removing the valve from the tank. The concept also incorporates the Aerojet developed spherical metal seal. A large amount of effort has already been expended to develop this seal for low helium and hydrogen leak rates for use in fluorine and hydrogen applications. Helium leak rates as low as approximately 1×10^{-7} standard cubic centimeters per second had been observed using a 2.0 inch diameter seal fabricated from phosphor bronze. Other seals fabricated from gold plated stainless steel have also been tested.

1139893, Concept No. 3, Figure (3) - This concept features a seal that traps LH_2 or GH_2 in a cavity and expands the gas by heat to generate large sealing forces against a poppet in the closed position. The poppet is placed in position by a motor driven ball screw that is designed with a lead of approximately 12 degrees so that it is not backdriveable. When the valve is opened, the bellows trapped high pressure fluid is vented downstream. A check valve system allows the bellows surrounded cavity to be recharged prior to the next closure.

1139894, Concept No. 4, Figure (4) - This multiple poppet concept is a means of achieving very low leakage by utilizing four small poppets rather than one large poppet. This is because small precision hard poppets and seats are more readily obtainable from a manufacturing standpoint than are large hard poppets and seats. A certain amount of redundancy is also offered since the four poppets are in parallel. An additional multiple poppet valve could be installed in series for additional redundancy. Modular manifolding

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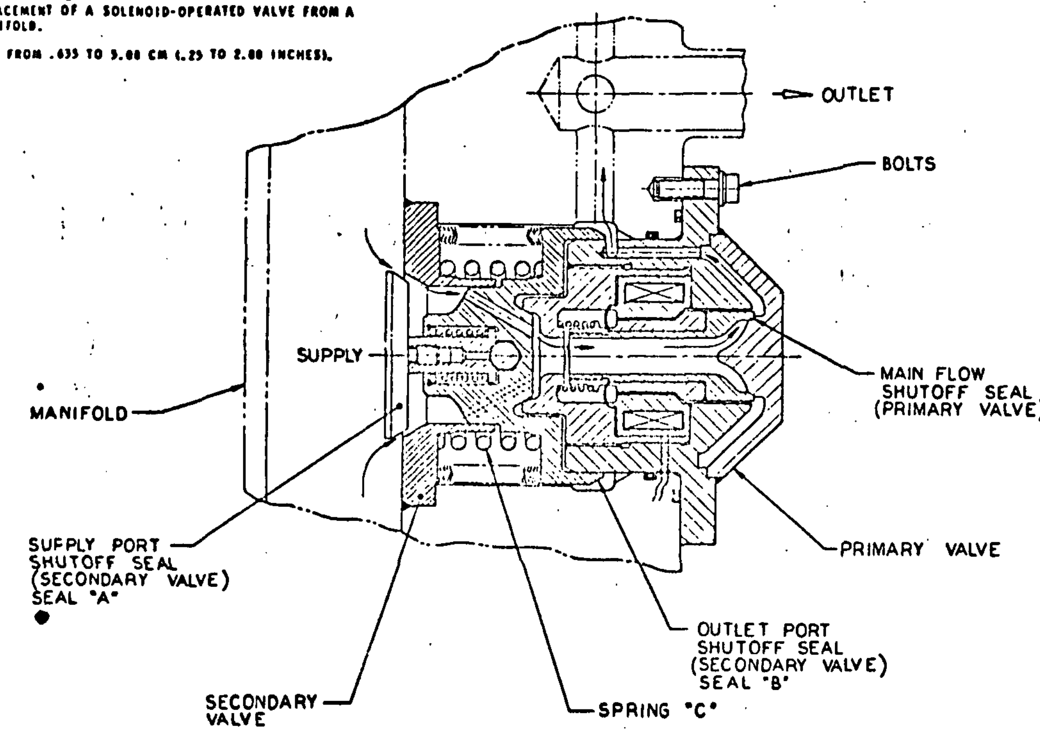
NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET "GOALS FOR REMOTE COUPLING AND UNCOUPLING OF VALVES FOR SPACE MAINTAINABILITY" DATED 30 JULY 1971.
3. DRAWING DEPICTS:

   REMOTE REPLACEMENT OF A SOLENOID-OPERATED VALVE FROM A PRESSURIZED MANIFOLD.
4. CONCEPT IS VALID FROM .635 TO 5.08 CM (.25 TO 2.00 INCHES).
5. DESCRIPTION OF OPERATION:

   TO REMOVE THE PRIMARY VALVE, UNSCREW THE BOLTS AND LIFT THE VALVE OUT OF THE MANIFOLD. AS THE VALVE IS BEING REMOVED, SEAL "A" AND "B" OF THE SECONDARY VALVE IS CLOSED BY THE SPRING "C" THUS PREVENTING LOSS OF FLUID FROM THE SUPPLY AND OUTLET LINE.



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CONTRACT NO. NAS 8-27568

DRAWN P. BIANCHI 9-17-71

TITLE: SOLENOID OPERATED SHUTOFF VALVE WITH PRESSURIZED SYSTEM REMOVAL CAPABILITY

SIZE D CODE IDENT NO 34632 DWG NO 1139890

1139890

FIGURE 1

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET VALVE DESIGN GOALS, DATED 30 AUGUST 1971, FOR THE REMOTE REMOVAL OF VALVES FROM THE SYSTEM AND TO MEET INTERNAL LEAKAGE REQUIREMENTS OF 1×10^{-7} SCCS GHe.
3. DRAWING DEPICTS:
   A. THE REMOTE REPLACEMENT OF A TANK VALVE FROM A PRESSURIZED SYSTEM.
   B. THE CONCEPT OF A SPHERICAL METAL SHUTOFF SEAL TO MEET THE 1×10^{-7} SCCS GHe INTERNAL LEAKAGE REQUIREMENT.
4. CONCEPT IS VALID FROM 12.7 TO 50.8 CM (5.00 TO 20.00 INCHES).
5. DESCRIPTION OF OPERATION:

   WITH THE POPPET IN THE CLOSED POSITION AND THE STATIC SEAL JOINT DISCONNECTED THE VALVE CAN BE LOWERED OUT WITH PRESSURE IN THE TANK.

   AS THE VALVE IS BEING REMOVED, THE COMPRESSION SPRING FORCES THE BACKUP POPPET AND THE SUPPORTING WEB DOWN UNTIL THE SHUTOFF SEAL OF THE BACKUP POPPET IS MADE. THE BACKUP FLANGE SEAL IS REQUIRED DURING THE TRANSITION PERIOD OF VALVE REMOVAL TO PREVENT EXTERNAL LEAKAGE.

   THE MAIN SEAL CONFIGURATION FEATURES A FLEXIBLE GOLD-PLATED METAL LIP-SEAL WHICH SEATS ONTO THE VALVE BODY.

   AN ALTERNATE SEAL CONFIGURATION SHOWN IN VIEW "A" IS THE REVERSE APPLICATION OF THE PRIMARY CONFIGURATION.
6. THE VALVE CONFIGURATION SHOWN IN THIS DRAWING IS SIMILAR TO THE ONE SHOWN IN DRAWING 1139012.

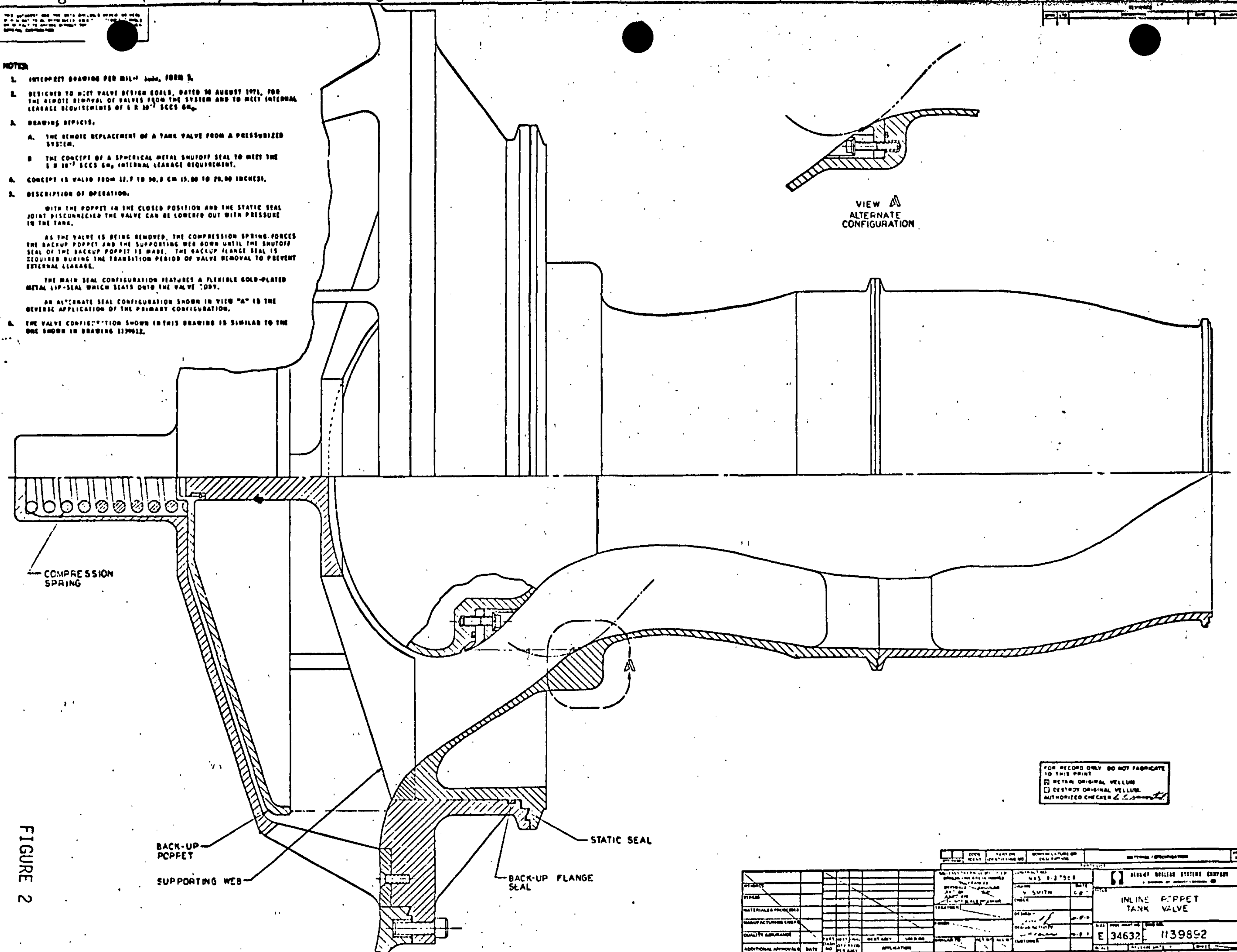

FIGURE 2

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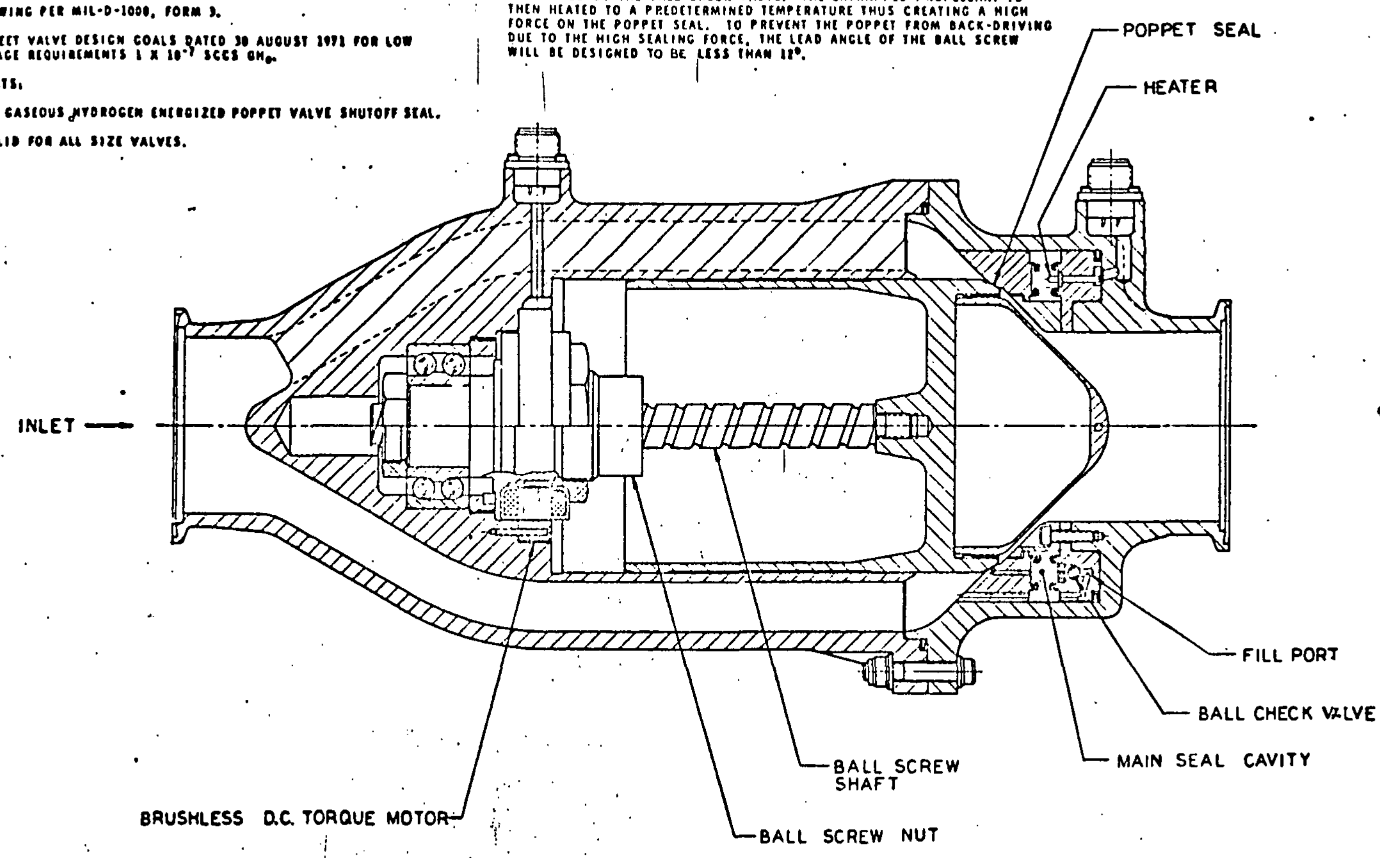
NOTES

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET VALVE DESIGN GOALS DATED 30 AUGUST 1971 FOR LOW INTERNAL LEAKAGE REQUIREMENTS 1 X 10^{-7} SCCS GH_2.
3. DRAWING DEPICTS:
   LIQUID TO GASEOUS HYDROGEN ENERGIZED POPPET VALVE SHUTOFF SEAL.
4. CONCEPT IS VALID FOR ALL SIZE VALVES.
5. DESCRIPTION OF OPERATION:

   THE CONCEPT SHOWS A METHOD OF TRAPPING AND HEATING GH_2 OR LH_2 WITHIN A SEAL CAVITY TO CREATE A HIGH SEALING FORCE ON THE POPPET.

   TO EFFECT PRIMARY SEALING WHEN THE VALVE IS CLOSED GH_2 OR LH_2 FLOWS INTO THE MAIN SEAL CAVITY THRU THE FILL PORT TO BECOME ENTRAPPED BY THE BALL CHECK VALVE. THE ENTRAPPED PROPELLANT IS THEN HEATED TO A PREDETERMINED TEMPERATURE THUS CREATING A HIGH FORCE ON THE POPPET SEAL. TO PREVENT THE POPPET FROM BACK-DRIVING DUE TO THE HIGH SEALING FORCE, THE LEAD ANGLE OF THE BALL SCREW WILL BE DESIGNED TO BE LESS THAN 12°.



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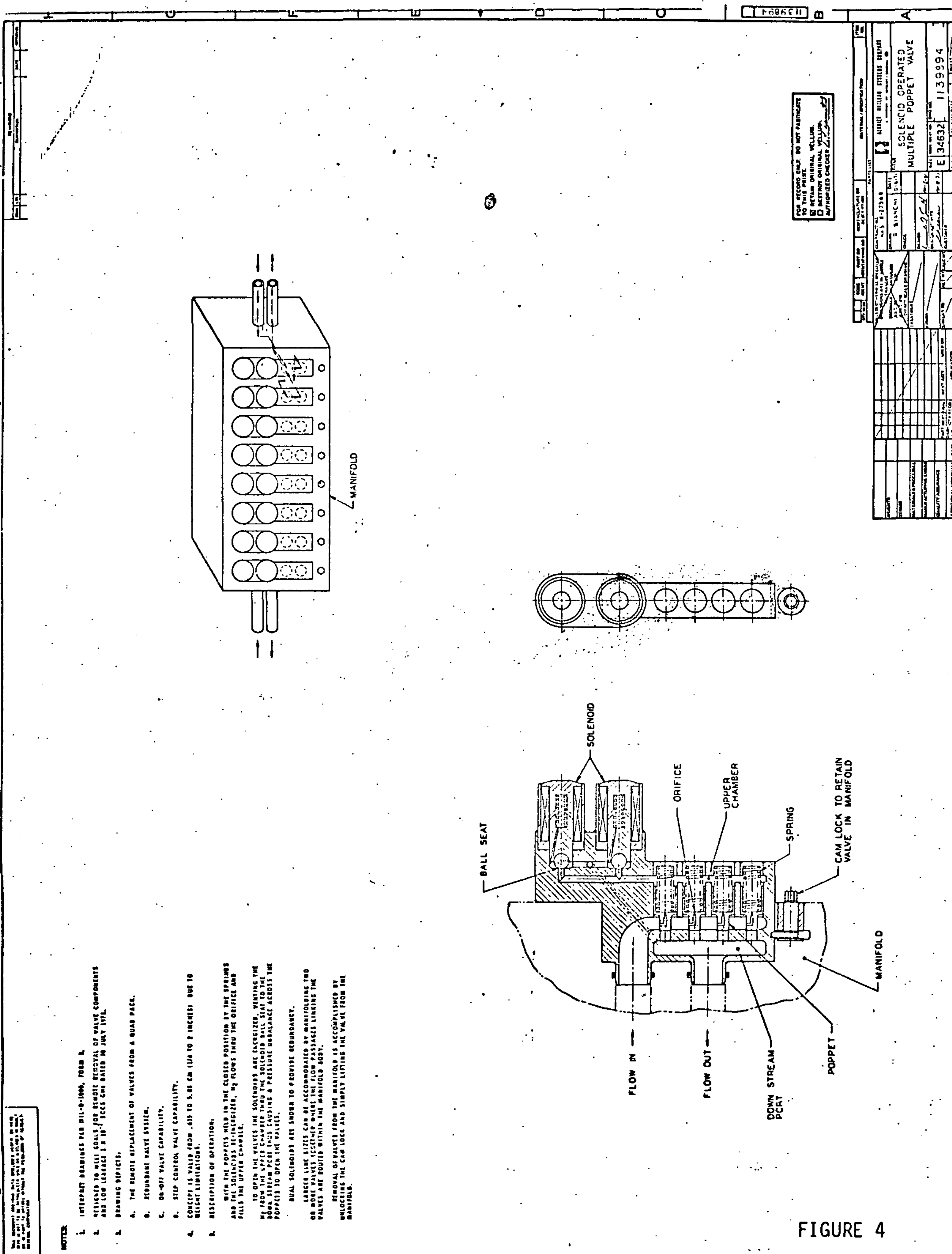
CONTRACT NO. NAS 8-27568
DRAWN R. BIANCHI DATE 10-5-71

TITLE: INLINE POPPET VALVE-MOTOR OPERATED

SIZE D CODE IDENT NO 34632 DWG NO 1139893

FIGURE 3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SOLENOID OPERATED MULTIPLE POPPET VALVE
E 34632 1139994
MANIFOLD
SOLENOID
ORIFICE
UPPER CHAMBER
SPRING
CAM LOCK TO RETAIN VALVE IN MANIFOLD
BALL SEAT
MANIFOLD
FLOW IN
FLOW OUT
DOWN STREAM PORT
POPPET
NOTES:
1. INTERPRET DRAWINGS PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET GOALS FOR REMOTE REMOVAL OF VALVE COMPONENTS AND LOW LEAKAGE 1 X 10 SCCS GHe DATED 30 JULY 1970.
3. DRAWING DEPICTS:
A. THE REMOTE REPLACEMENT OF VALVES FROM A QUAD PACK.
B. REDUNDANT VALVE SYSTEM.
C. ON-OFF VALVE CAPABILITY.
D. STEP CONTROL VALVE CAPABILITY.
4. CONCEPT IS VALID FROM .635 TO 5.08 CM (1/4 TO 2 INCHES) DUE TO WEIGHT LIMITATIONS.
5. DESCRIPTION OF OPERATION.
WITH THE POPPETS HELD IN THE CLOSED POSITION BY THE SPRINGS AND THE SOLENOIDS DE-ENERGIZED, N_2 FLOWS THRU THE ORIFICE AND FILLS THE UPPER CHAMBER.
TO OPEN THE VALVES THE SOLENOIDS ARE ENERGIZED, VENTING THE N_2 FROM THE UPPER CHAMBER THRU THE SOLENOID BALL SEAT TO THE DOWN STREAM PORT THUS CAUSING A PRESSURE UNBALANCE ACROSS THE POPPETS TO OPEN THE VALVES.
DUAL SOLENOIDS ARE SHOWN TO PROVIDE REDUNDANCY.
LARGER LINE SIZES CAN BE ACCOMMODATED BY MANIFOLDING TWO OR MORE VALVES TOGETHER WHERE THE FLOW PASSAGES LINKING THE VALVES ARE ROUTED WITHIN THE MANIFOLD BODY.
REMOVAL OF VALVES FROM THE MANIFOLD IS ACCOMPLISHED BY UNLOCKING THE CAM LOCK AND SIMPLY LIFTING THE VALVE FROM THE MANIFOLD.

FIGURE 4

for larger line sizes is an additional feature. This concept can be considered when striving to develop shutoff valves with very low leakage requirements because small poppets and seats are less vulnerable to thermal distortions, deflections and tolerances.

1139895, Concept No. 5, Figure (5) - Because of the electromagnetic seal concept, it is possible to design a valve with no moving parts. The design is based on the principal of electromagnetic liquid metal pumps and would work in a liquid metal system since the media is a good conductor. Thus, feasibility in a hydrogen system cannot be assured until development hardware is fabricated and tested. This concept, as shown, is applicable to small line sizes.

1139896, Concept No. 6, Figure (6) - This concept is an extension of the electromagnetic seal principle to large line sizes. Diametrical sealing between the body and a large diameter poppet is accomplished by the electromagnetic seal principle. Several possible variations exist for this design. Two electromagnetic seals could be utilized to seal the skirt clearance of a sleeve for a shutoff valve (i.e., sleeve valve). Also, electromagnetic input power could be modulated to produce a regulator if the sealing principle is workable. This concept should not be pursued until concept 1139895 is proven and design variable parameters quantified.

1139897, Concept No. 7, Figure (7) - Positive shutoff sealing is accomplished by liquid metal which is heated for valve actuation and frozen by the hydrogen for sealing. This concept is temperature limited and would require hardware testing to define the variable parameters.

1139899, Concept No. 8, Figure (8) - This uses the NERVA Cooldown Shutoff and Control Valve approach which is a soft spherical metal seal seating into a hardened cone. This seal satisfactorily completed 1000 liquid hydrogen flow cycles with 600 micron contamination in the fluid. The seal and seat guides were designed for contamination resistance and protection.

1139012, Concept No. 9, Figure (9) - This concept is a large tank or line mounted shutoff valve with actuator and ball screw removal capability. The actuator and ball screw are mounted in the central structure of the valve and benefit from the cooling effect of the hydrogen flow. The valve design is conventional but does show the problems and complexity resulting from an actuator/valve element changeout requirement.

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET GOALS FOR LOW INTERNAL AND EXTERNAL LEAKAGE REQUIREMENTS 1×10^{-7} SCCS GHe DATED 30 AUGUST 1971.
3. DRAWING DEPICTS:
   - A. A VALVE WITH NO MOVING PARTS.
   - B. A VALVE TO BE WELDED INTO THE LINE.
   - C. BI-DIRECTIONAL ON-OFF VALVE.
   - D. CONTROL VALVE
4. CONCEPT IS VALID FOR COMPONENT-TO-LINE OR COMPONENT-TO-MANIFOLD.
5. CONCEPT IS VALID FROM .635 TO 1.90 CM (1/4 INCH TO 3/4 INCH) DUE TO WEIGHT LIMITATIONS.

6. **DESCRIPTION OF OPERATION:**

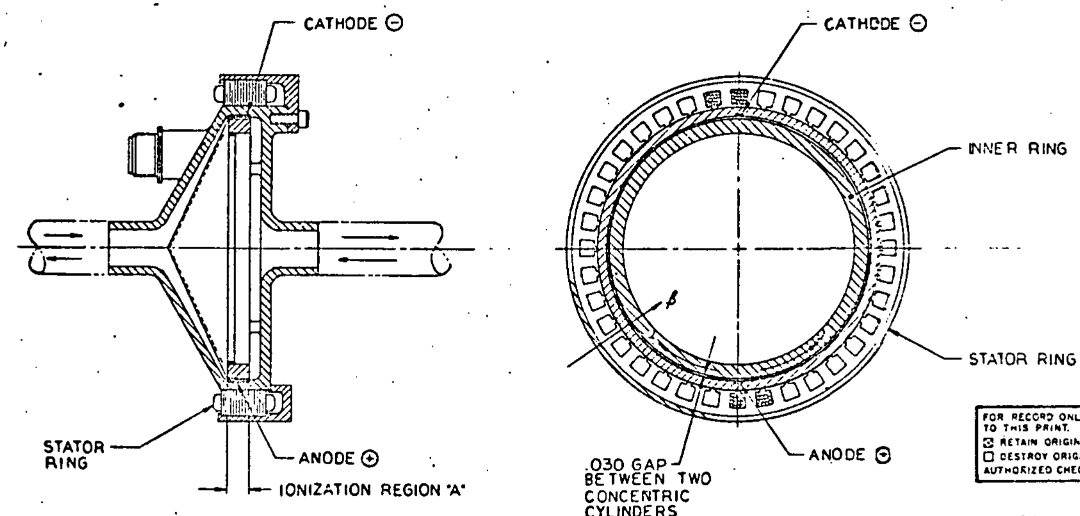
THE VALVE OPERATES WITH NO MOVING PARTS, BY UTILIZING THE PRINCIPAL OF IONIZATION OF A GAS IN A MAGNETIC FIELD WHICH STATES THAT A CONDUCTOR (GAS PARTICLE) MOVING PERPENDICULAR TO A MAGNETIC FIELD EXPERIENCES A FORCE (F) PERPENDICULAR TO BOTH THE CURRENT (α) IN THE CONDUCTOR AND THE FIELD (β).

THE SEALING PRINCIPAL INVOLVES THE LOCALIZED IONIZATION OF GH_2 BETWEEN TWO CONCENTRIC CYLINDERS .030 INCHES APART.

THE IONIZATION OF THE GAS IS PRODUCED IN THE REGION "A" WHERE THE CURRENT IS CONDUCTED WITHIN THE ANULAR GAP AND FLOWS FROM THE ANODE TO THE CATHOD.

THE FORCE EXERTED ON THE CONDUCTING GAS IS PRODUCED BY THE MAGNETIC FIELD OF THE STATOR RING WHOSE FIELD IS DIVERTED IN A RADIAL DIRECTION BY THE SOFT IRON CENTER CORE TO EFFECT THE HIGHEST FORCE POSSIBLE BETWEEN THE TWO VECTORS(α) AND(β).

THE FORCE PRODUCED BY THE FIELD IS DISTRIBUTED ALONG THE ANULAR AREA OF THE CONDUCTING PATH THUS PRODUCING THE PRESSURE REQUIRED TO OVERCOME THE SYSTEM PRESSURE.



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CONTRACT NO. NAS 3-27568
DRAWN V SMITH 10-7-71
TITLE 3/4 INCH INLINE VALVE-ELECTROMAGNETIC SEAL
SIZE D 34632 DWG NO 1139395
SCALE 1/1

FIGURE 5

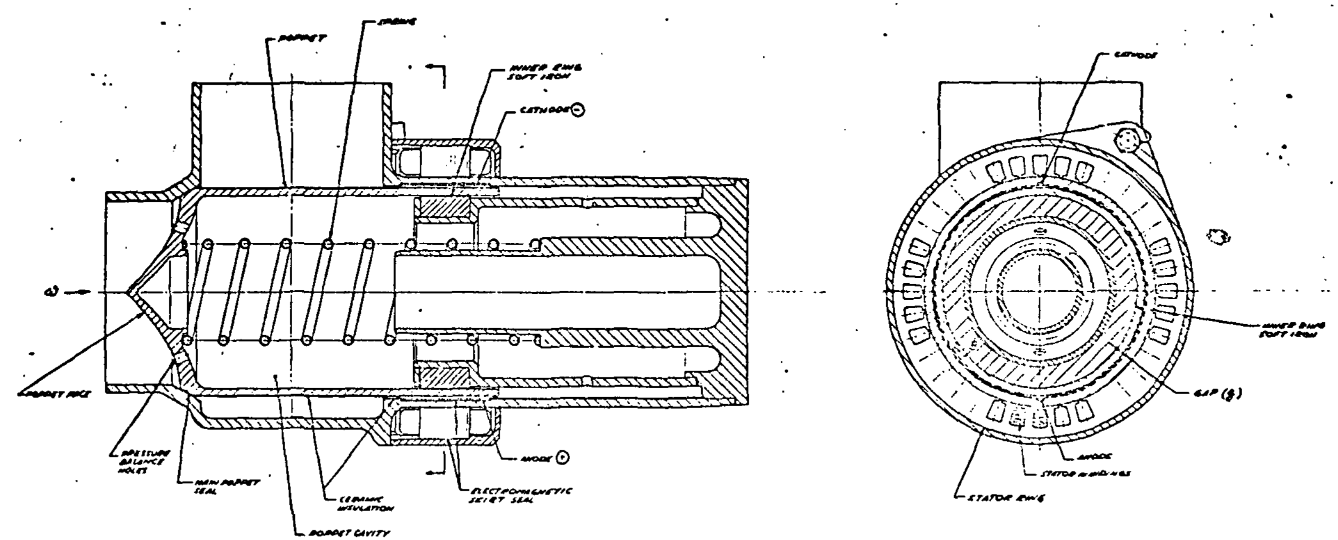
NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET VALVE GOALS FOR LOW INTERNAL AND EXTERNAL LEAKAGE, REQUIREMENTS 3 X 10^{-7} SCCS GH_2 DATED 18 AUGUST 1972.
3. DRAWING DEPICTS:
   A. A BALANCED POPPET
   B. A BI-DIRECTIONAL ON-OFF VALVE
   C. A CONTROL VALVE
4. CONCEPT IS VALID FOR COMPONENT-TO-LINE OR COMPONENT-TO-MANIFOLD.
5. CONCEPT IS VALID FOR ALL VALVE SIZES.
6. DESCRIPTION OF OPERATION:

   WITH THE POPPET HELD CLOSED BY THE SPRING THERE ARE TWO SEALING POINTS; THE MAIN POPPET SEAL AND THE ELECTRO-MAGNETIC SKIRT SEAL WHEREBY THE SKIRT SEAL MUST BE ENERGIZED TO PREVENT LEAKAGE PAST THE ANULAR GAP.

   TO OPEN THE VALVE THE POWER TO THE ELECTRO-MAGNETIC SKIRT SEAL IS TURNED OFF THUS ALLOWING THE H_2 TO FLOW OUT OF THE POPPET CAVITY FASTER THAN H_2 CAN BE ADMITTED THRU THE BALANCING HOLES CREATING AN UNBALANCED PRESSURE DISTRIBUTION ACROSS THE POPPET FACE CAUSING THE VALVE TO OPEN.

   BY PROPER VOLTAGE ADJUSTMENT AT THE SKIRT SEAL, MIDWAY POSITIONING OF THE POPPET CAN BE ACHIEVED THUS CONTROLLING FLOW.



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90° BALANCED POPPET ELECTROMAGNETIC SKIRT SEAL

E 34632 1139896

FIGURE 6

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NOTES:

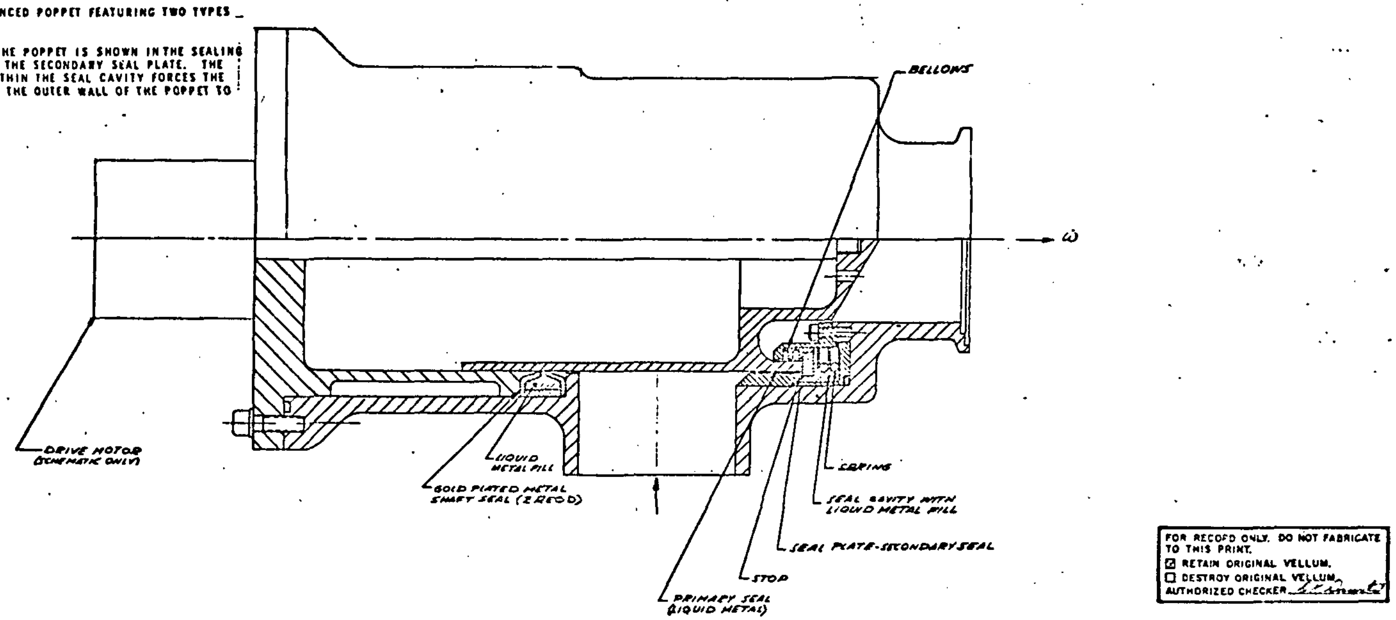
1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET VALVE DESIGN GOALS DATED 30 AUGUST 1971 FOR LOW INTERNAL LEAKAGE REQUIREMENTS 1×10^{-7} SCCS GHe.
3. DRAWING DEPICTS METHODS OF USING LIQUID METAL SEALS FOR CRYOGENIC OR AMBIENT TEMPERATURE RANGES.
4. CONCEPT IS VALID FOR ALL SIZE VALVES.
5. DESCRIPTION OF OPERATION:

THE CONCEPT SHOWS A BALANCED POPPET FEATURING TWO TYPES OF LIQUID METAL SEALS:

MAIN SHUTOFF SEAL: THE POPPET IS SHOWN IN THE SEALING POSITION MAKING CONTACT WITH THE SECONDARY SEAL PLATE. THE DISPLACEMENT OF THE POPPET WITHIN THE SEAL CAVITY FORCES THE LIQUID METAL TO FLOW UP ALONG THE OUTER WALL OF THE POPPET TO CREATE A PRIMARY SEAL.

AS THE POPPET IS REMOVED FROM THE SEAL CAVITY THE LIQUID METAL IS FORCED BACK INTO THE SEAL CAVITY BY THE SYSTEM PRESSURE AT THE SAME TIME THE SPRING FORCES THE SEAL PLATE UP AGAINST THE STOP TO RETAIN THE LIQUID METAL. THE BELLOWS PREVENTS THE LOSS OF THE LIQUID METAL SEALANT.

SHAFT SEAL: THE SHAFT SEAL CONSISTS OF TWO METAL LIP SEALS INSTALLED FACE-TO-FACE TO ENTRAP THE LIQUID METAL SEALANT. THE SEAL IS APPLICABLE TO A ROTARY OR LINEAR SHAFT MOTION.



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AEROJET NUCLEAR SYSTEMS COMPANY

CONTRACT NO NAS 9-27568

TITLE: POPPET VALVE - LIQUID METAL SEALS.

SIZE D — CODE IDENT NO 34632 — DWG NO. 1139897

FIGURE 7

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET GOALS FOR LOW INTERNAL AND EXTERNAL LEAKAGE REQUIREMENTS 1 X 10^{-7} SCCS GHe DATED 30 AUGUST 1971.
3. DRAWING DEPICTS:
   A. *SPHERICAL METAL SHUTOFF SEAL.*
   B. MOTOR OPERATED VALVE FOR ON-OFF AND/OR FLOW CONTROL APPLICATION.
4. THE CONCEPT IS VALID FOR COMPONENT-TO-LINE.
5. CONCEPT IS VALID FOR ALL SIZES OF VALVES FROM .635 TO 50.80 CM (1/4 TO 20 INCHES).
6. DESCRIPTION OF OPERATION:

   THE VALVE IS A MOTOR DRIVEN, PLANETARY GEAR TRANSMISSION, DIGITAL OR ANALOG POPPET TYPE, WITH A SPHERICAL METALLIC SHUTOFF SEAL.

   ACTUATION IS ACCOMPLISHED BY CONVERTING ROTARY MOTION OF THE MOTOR TO LINEAR MOTION, BY A CAM ARRANGEMENT.

   THE MAIN FEATURE OF THE DESIGN IS THE SPHERICAL METAL SHUTOFF SEAL.

   ANALYTICAL METHODS FOR DETERMINING THE MECHANICAL PERFORMANCE OF THE SEAL HAVE BEEN PROGRAMMED ON A COMPUTER TO DETERMINE ITS BEHAVIOR UNDER COMPLEX LOADING CONDITIONS.

   THE LOADING CONDITIONS CONSIDERED BY THE PROGRAM INCLUDE VALVE CLOSING, PRESSURIZATION, DIFFERENTIAL THERMAL CONTRACTION AND ASSEMBLY RADIAL CLEARANCE EFFECTS.

   THESE PARAMETERS CAN READILY BE INVESTIGATED FOR THEIR EFFECTS ON SEALING PERFORMANCE.

⚠ SEAL SHOWN SATISFACTORILY COMPLETED OVER 2200 CYCLES OF WHICH 200 CYCLES WERE WITH LH_2 AT 30 PSIA AND 2.0 PPS.

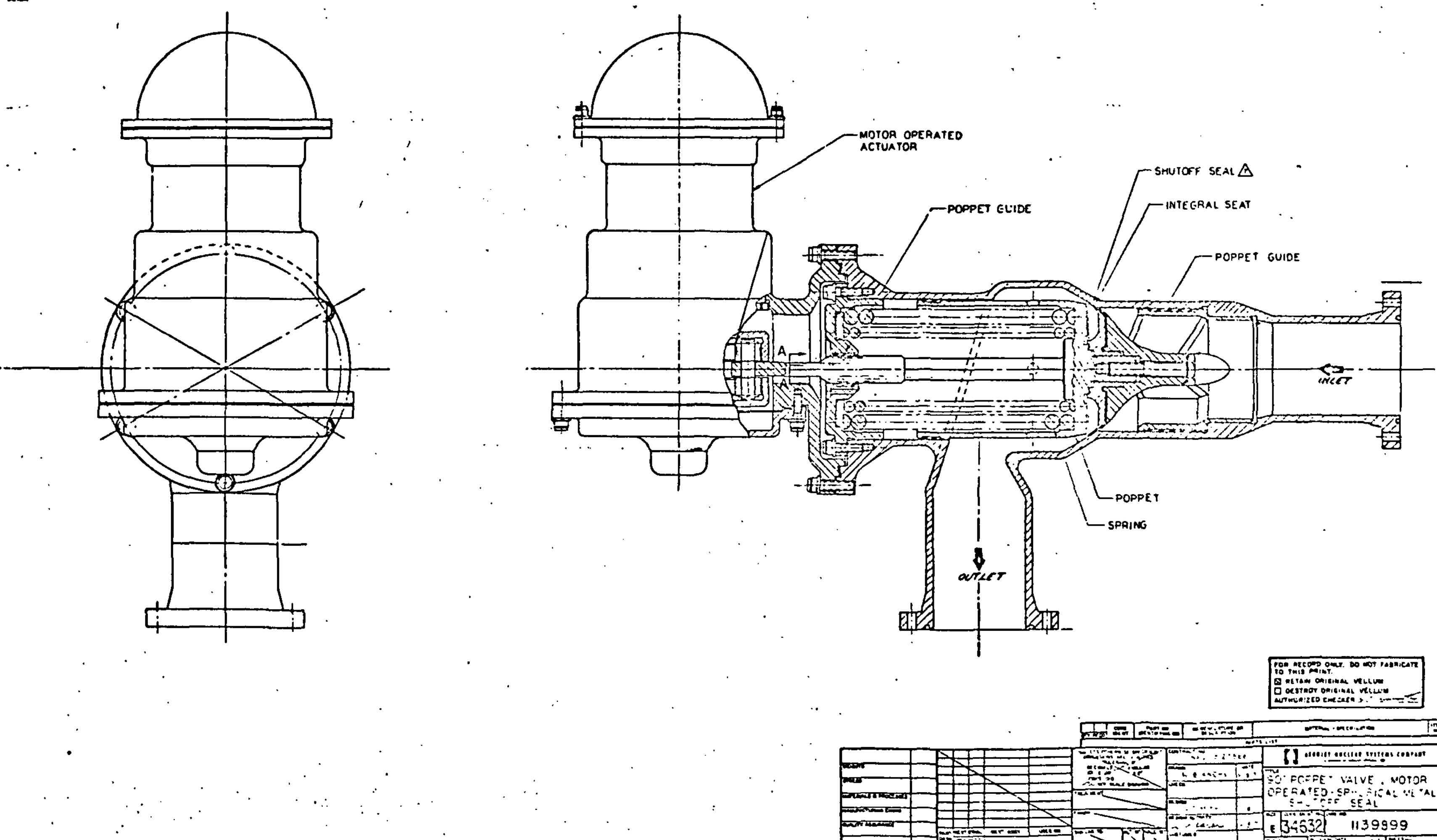


FIGURE 8

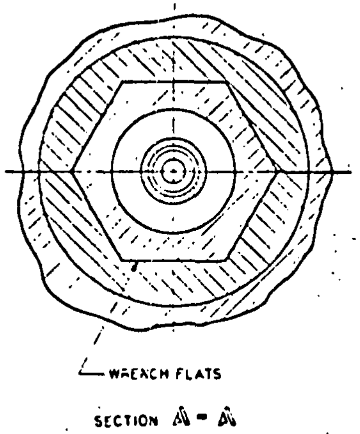
NOTES:

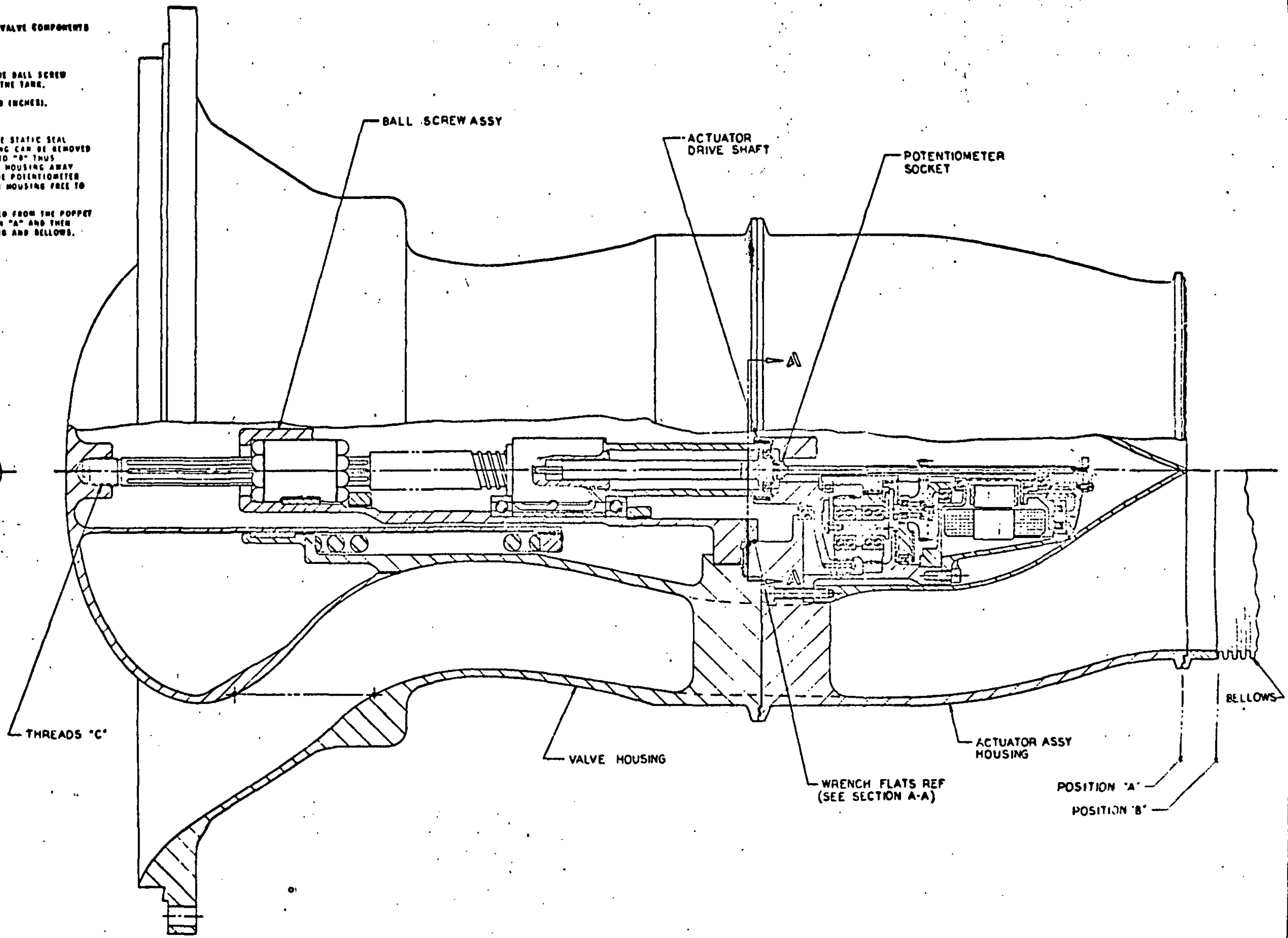
1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET GOALS FOR THE REMOTE REMOVAL OF VALVE COMPONENTS DATED 30 JULY 1971.
3. DRAWING DEPICTS:

   THE REMOTE REPLACEMENT OF THE ACTUATOR AND THE BALL SCREW DRIVE MECHANISM FROM THE VALVE WITH PRESSURE IN THE TANK.
4. CONCEPT IS VALID FROM 12.7 TO 50.8 CM (5.00 TO 20.00 INCHES).
5. DESCRIPTION OF OPERATION:

   WITH THE POPPET IN THE CLOSED POSITION AND THE STATIC SEAL JOINTS DISCONNECTED, THE ACTUATOR ASSEMBLY HOUSING CAN BE REMOVED BY DEPRESSING THE LINE BELLOWS FROM POSITION "A" TO "B" THUS ALLOWING CLEARANCE TO DROP THE ACTUATOR ASSEMBLY HOUSING AWAY FROM THE MAIN VALVE HOUSING. THIS DISCONNECTS THE POTENTIOMETER AND ACTUATOR DRIVE SHAFT, AND LEAVES THE ACTUATOR HOUSING FREE TO BE REMOVED FROM THE LINE.

   THE BALL SCREW ASSEMBLY MAY THEN BE UNSCREWED FROM THE POPPET BY ROTATING THE WRENCH FLATS AS VIEWED FROM SECTION "A" AND THEN REMOVED THRU THE OPENING BETWEEN THE VALVE HOUSING AND BELLOWS.





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E 34632

1139012 A

FIGURE 9

1139930, Concept No. 10, Figure (10) - This depicts how a side mounted, external electromechanical actuator could be used with a tank shutoff valve so that the actuator could be removed by a remote manipulator.

1139931, Concept No. 11, Figure (11) - This is a visor shutoff valve with a retractable seal similar to the existing S-IC 17-inch prevalve. The design differs from the S-IC valve in that it incorporates the Aerojet spherical shell seal and has a more positive seal retraction and sequencing mechanism.

b. Pressure Regulators

The six regulators completed during Phase I included "state-of-the-art", upgraded "state-of-the-art", and closed loop control valves driven by electromechanical actuators which are being developed as part of the NERVA program.

The electronically controlled electromechanically operated single stage regulators offered many benefits over the conventional mechanical pneumatic devices. The electromechanically operated servo regulators with the appropriate inputs and feedback loops can be designed to control tank pressure on a programmed basis to minimize the loss of propellant due to venting. This can be done by starting a mission with a low tank pressure, sufficient to maintain the minimum NERVA NPSP. This requires programing the tank pressure to a predetermined schedule throughout each burn rather than using a constant single level tank pressure throughout the entire mission. The electrical command for positioning the actuator is a digital input signal. With the advancements of radiation hardened solid state miniature electronic components and techniques, it more reliable with less weight to put the complexity into electronics rather than into mechanical devices. This way, regulating devices with two or more sensing bellows or diaphragms, with their attendant fatigue, temperature compensations, vibration, creep, manufacturing and adjustment problems are avoided. The performance of the digital electronic servo controlled regulator is predictable. It is also possible to calibrate and check out the electromechanical stage in space with simulators without flowing large amounts of GH_2, which is generally lost. This would simplify the logistics problem Other benefits in some of the regulator concepts presented is the ability to positively shut off on command, ability to hold reverse pressure (tank pressure) during coast periods, and the ability to

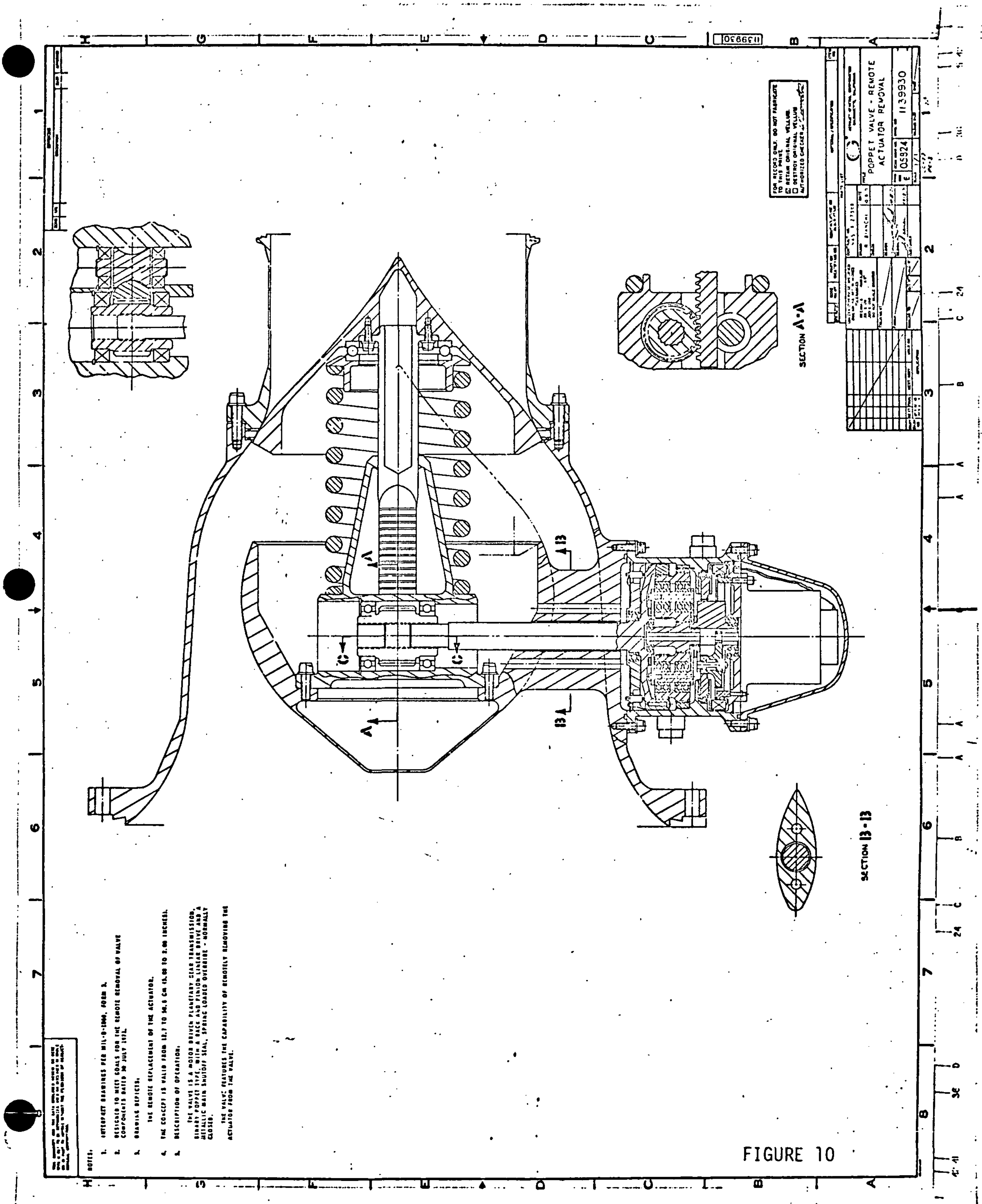
POPPET VALVE - REMOTE ACTUATOR REMOVAL
1139930
05924
SECTION A-A
SECTION B-B

FIGURE 10

REVISIONS

ZONE	LTR	DESCRIPTION	DATE	APPROVED

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET GOALS FOR LOW LEAKAGE REQUIREMENTS 1 X 10^{-7} SCCS GHe DATED 30 AUGUST 1971.
3. DRAWING DEPICTS:

   A TANK MOUNTED OR IN-LINE MOUNTED VALVE MOTOR-OPERATED VISOR VALVE WITH A LINEAR SEAL WITHDRAWAL.
4. THE CONCEPT IS VALID FOR COMPONENT-TO-LINE AND COMPONENT-TO-TANK.
5. CONCEPT IS VALID FOR 12.7 TO 50.8 CM (5.00 TO 20.00 INCHES), BASED ON VALVE FUNCTION IN THE SYSTEM - SUCH AS A THROTTLE AND BLOCKING VALVE FOR THE PROPELLANT FEED SYSTEM OR A SPRAY BYPASS VALVE FOR ORBIT REFUELING.
6. DESCRIPTION OF OPERATION:

   MOTOR DRIVEN PLANETARY GEAR TRANSMISSION, FULL FLOW BINARY VISOR TYPE, METALLIC MAIN GATE SEAL, OPEN MOTION-LINEAR SEAL, WITHDRAWAL, 90° ROTATIONAL, MECHANICAL SPRING LOADED, OVER-RIDE NORMALLY CLOSED.

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1139931

QTY REQD	SYM	CODE IDENT	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	ZONE	ITEM NO.

LIST OF MATERIALS

	DATE
STRESS	
MATERIALS & PROCESSES	
MANUFACTURING ENGRG.	
PRODUCT ASSURANCE	
ADDITIONAL APPROVALS	

PART DASH NO.	NEXT FINAL	QTY REQD PER ASSY	NEXT ASSY	USED ON

APPLICATION

DRAWING LEVEL

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCE ON DECIMALS ANGULAR
.XX ± .03 ± 2°
.XXX ± .010
DO NOT SCALE DRAWING

TREATMENT

FINISH

SIMILAR TO | ACT. WT | CALC WT

CONTRACT NO. NAS 8-27568

		DATE
DRAWN	R. BIANCHI	10-8-71
CHECKED		
DESIGN	V. SMITH	10-8-71
DESIGN ACTIVITY	Scheuer	10-8-71
CUSTOMER		

AEROJET-GENERAL CORPORATION
SACRAMENTO, CALIFORNIA

TITLE VISOR VALVE, LINEAR SEAL WITHDRAWL-MOTOR OPERATED

DWG SIZE	CODE IDENT NO.	DRAWING NO.
C	05824	1139931

SCALE: 1/1 | RELEASED: 12-11-71 | SHEET: 1 OF 2

CC-77
RC-3

FIGURE 11
Page 1 of 2

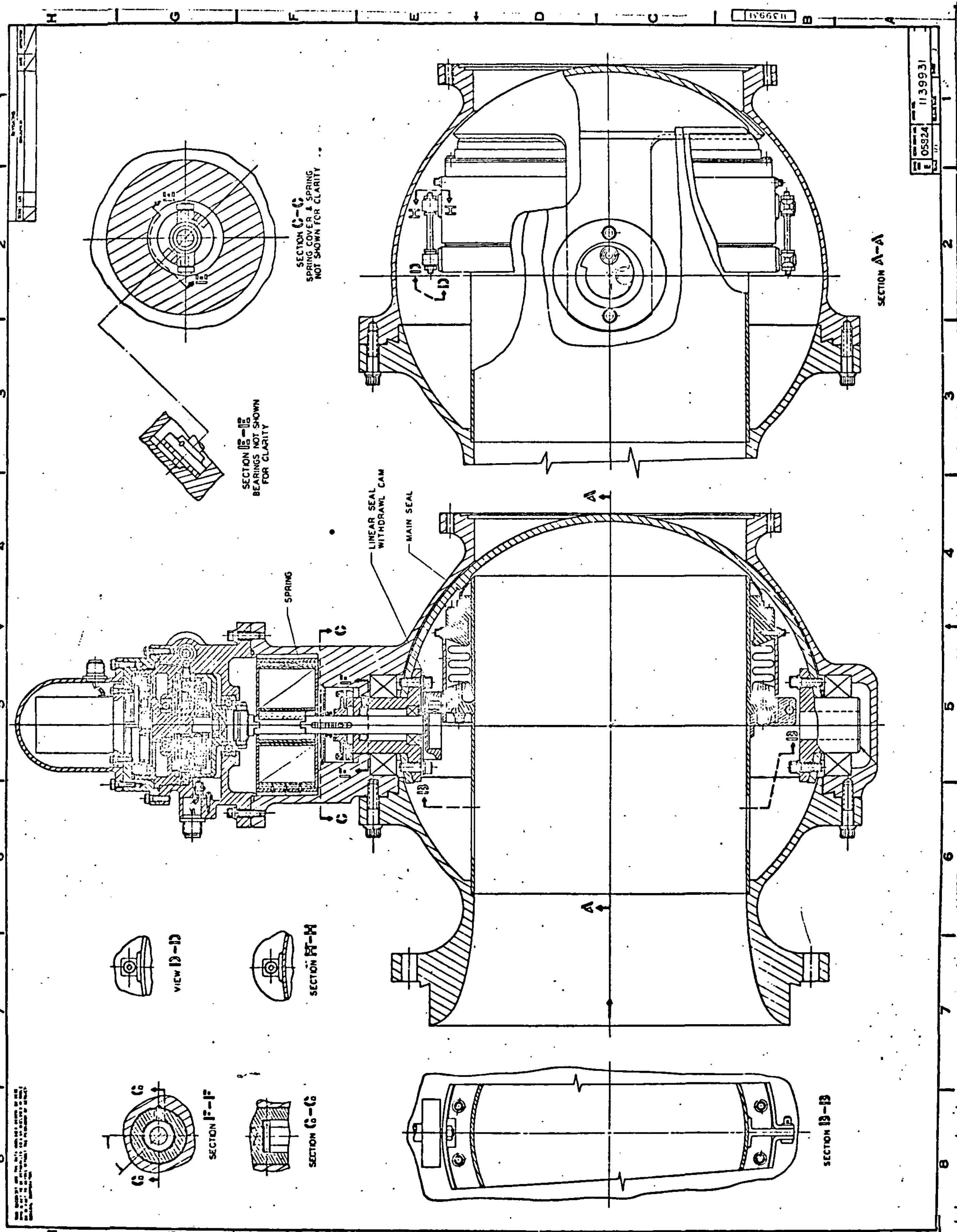


FIGURE 11
Page 2 of 2

act as relief or vent valves with a separate command. This latter feature can be accomplished by permitting flow through the engine during filling of the tank and during coast.

<u>1139923, Concept No. 1, Figure (12)</u> - This includes the ANSC electrical system concept described above and is essentially identical to the NERVA Bypass Control Valve (BCV) electromechanically operated visor regulator. In this concept the valve element or visor is rotated through a large angle. The angular rotation includes visor to seal shutoff area, flow control contour and then full open area for minimum pressure drop. Thus, the valve has a combined control and shutoff capability.

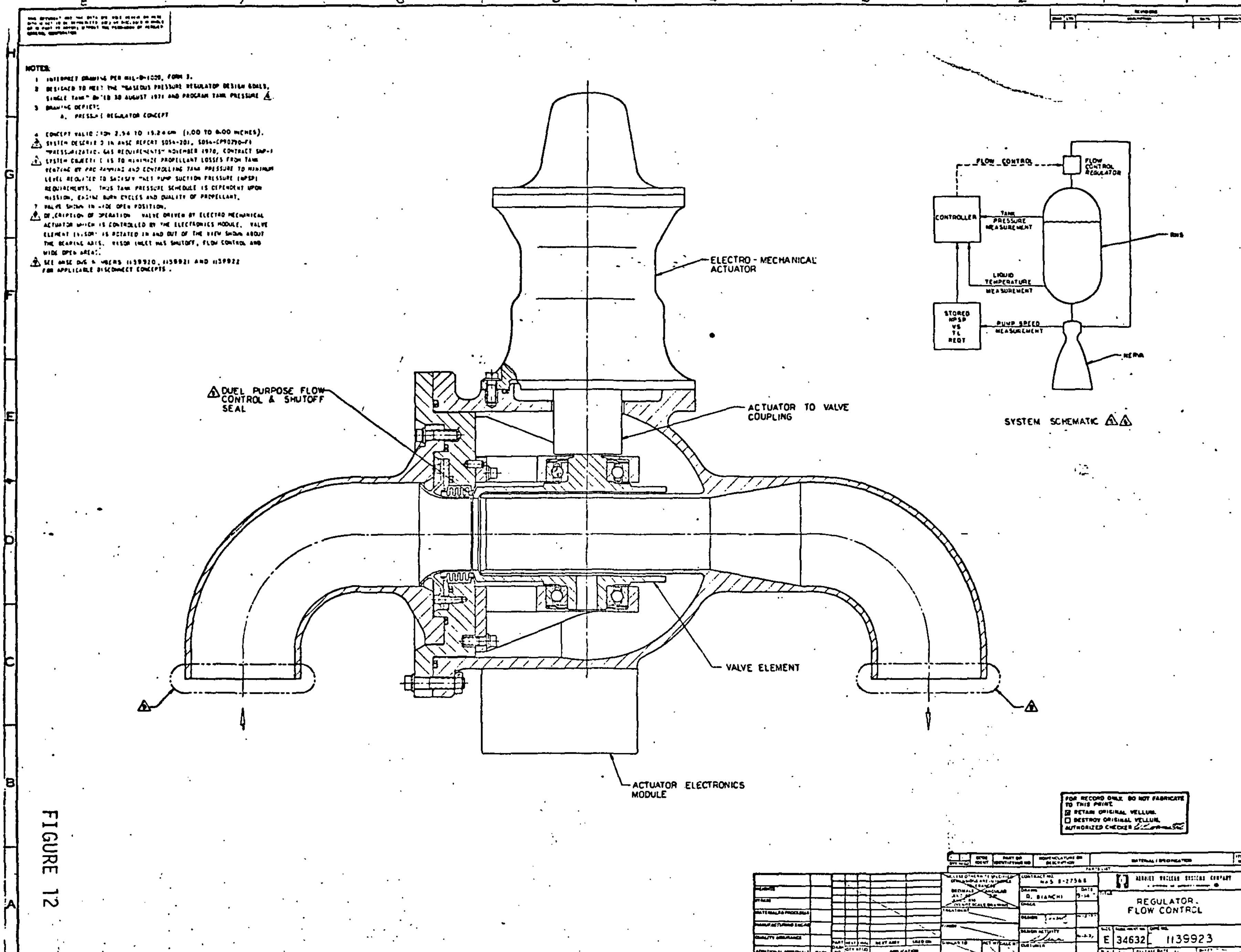
<u>1139924, Concept No. 2, Figure (13)</u> - This includes the ANSC electromechanical system concept described above. The valving element is a floating shear plug. This design is especially adaptable to modification with reversible sealing to hold tank pressure during coast. The shear plug regulating element is an optically flat element operating pressure unbalanced and intimate contact with an optically flat seat.

<u>1139925, Concept No. 3, Figure (14)</u> - This includes the ANSC electromechanical system concept described above. The valving element is an optically flat rotary shear plate in intimate contact with an optically flat stator. The metering area is exceptionally fine because the metering area extends for 270 degrees rotation plus the shutoff angle. The design is limited to smaller sizes. The unit can be used in-line or as a pilot for a large regulator.

<u>1139926, Concept No. 4, Figure (15)</u> - This includes the ANSC electromechanical system concept previously described in conjunction with a metering valve (similar to that described on Drawing 1139925 above). The metering valve programs pressure into the dome of a normally closed dome loaded shear seal regulator. The predetermined pressure determines the dome loaded regulator position.

<u>1139928, Concept No. 5, Figure (16)</u> - This concept is an all mechanical design with a spring loaded bellows sensing pilot regulator pressurizing the piston of a large dome loaded shear seal regulator. The design is not capable of programing tank pressure and requires temperature compensation.

<u>1139929, Concept No. 6, Figure (17)</u> - This includes the ANSC electromechanical system concept described above. The valving element is a floating shear plug which seals in both directions. The concept is the next iteration of 1139924 describe above.


NOTES:
1 INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2 DESIGNED TO MEET THE "GASEOUS PRESSURE REGULATOR DESIGN GOALS, SINGLE TANK" DATED 30 AUGUST 1971 AND PROGRAM TANK PRESSURE
3 DRAWING DEPICTS
A. PRESSURE REGULATOR CONCEPT
4 CONCEPT VALID FROM 2.54 TO 15.24 cm (1.00 TO 6.00 INCHES).
5 SYSTEM DESCRIBED IN ANSC REPORT S054-201, S054-CP90290-F1 "PRESSURIZATION GAS REQUIREMENTS" NOVEMBER 1970, CONTRACT SNP-1
6 SYSTEM OBJECTIVE IS TO MINIMIZE PROPELLANT LOSSES FROM TANK VENTING BY PRE PLANNING AND CONTROLLING TANK PRESSURE TO MINIMUM LEVEL REQUIRED TO SATISFY THE PUMP SUCTION PRESSURE (NPSP) REQUIREMENTS. THUS TANK PRESSURE SCHEDULE IS DEPENDENT UPON MISSION, ENGINE BURN CYCLES AND QUALITY OF PROPELLANT.
7 VALVE SHOWN IN WIDE OPEN POSITION.
8 DESCRIPTION OF OPERATION VALVE DRIVEN BY ELECTRO MECHANICAL ACTUATOR WHICH IS CONTROLLED BY THE ELECTRONICS MODULE. VALVE ELEMENT (VISOR) IS ROTATED IN AND OUT OF THE VIEW SHOWN ABOUT THE BEARING AXIS. VISOR INLET HAS SHUTOFF, FLOW CONTROL AND WIDE OPEN AREAS.
9 SEE ANSC DWG NUMBERS 1139920, 1139921 AND 1139922 FOR APPLICABLE DISCONNECT CONCEPTS.
ELECTRO-MECHANICAL ACTUATOR
DUEL PURPOSE FLOW CONTROL & SHUTOFF SEAL
ACTUATOR TO VALVE COUPLING
VALVE ELEMENT
ACTUATOR ELECTRONICS MODULE
FLOW CONTROL
FLOW CONTROL REGULATOR
CONTROLLER
TANK PRESSURE MEASUREMENT
LIQUID TEMPERATURE MEASUREMENT
STORED NPSP VS TL REQT
PUMP SPEED MEASUREMENT
RNS
NERVA
SYSTEM SCHEMATIC
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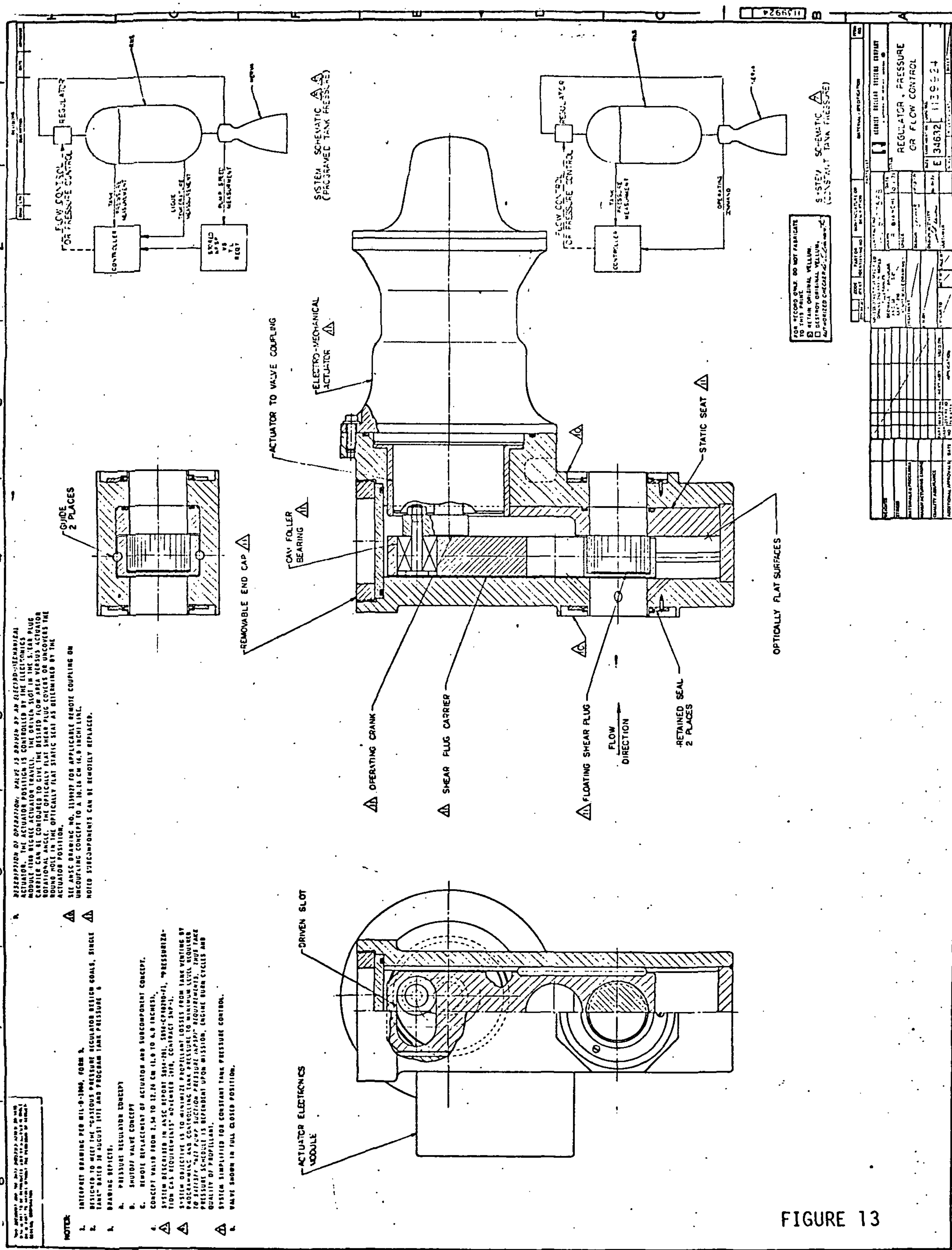
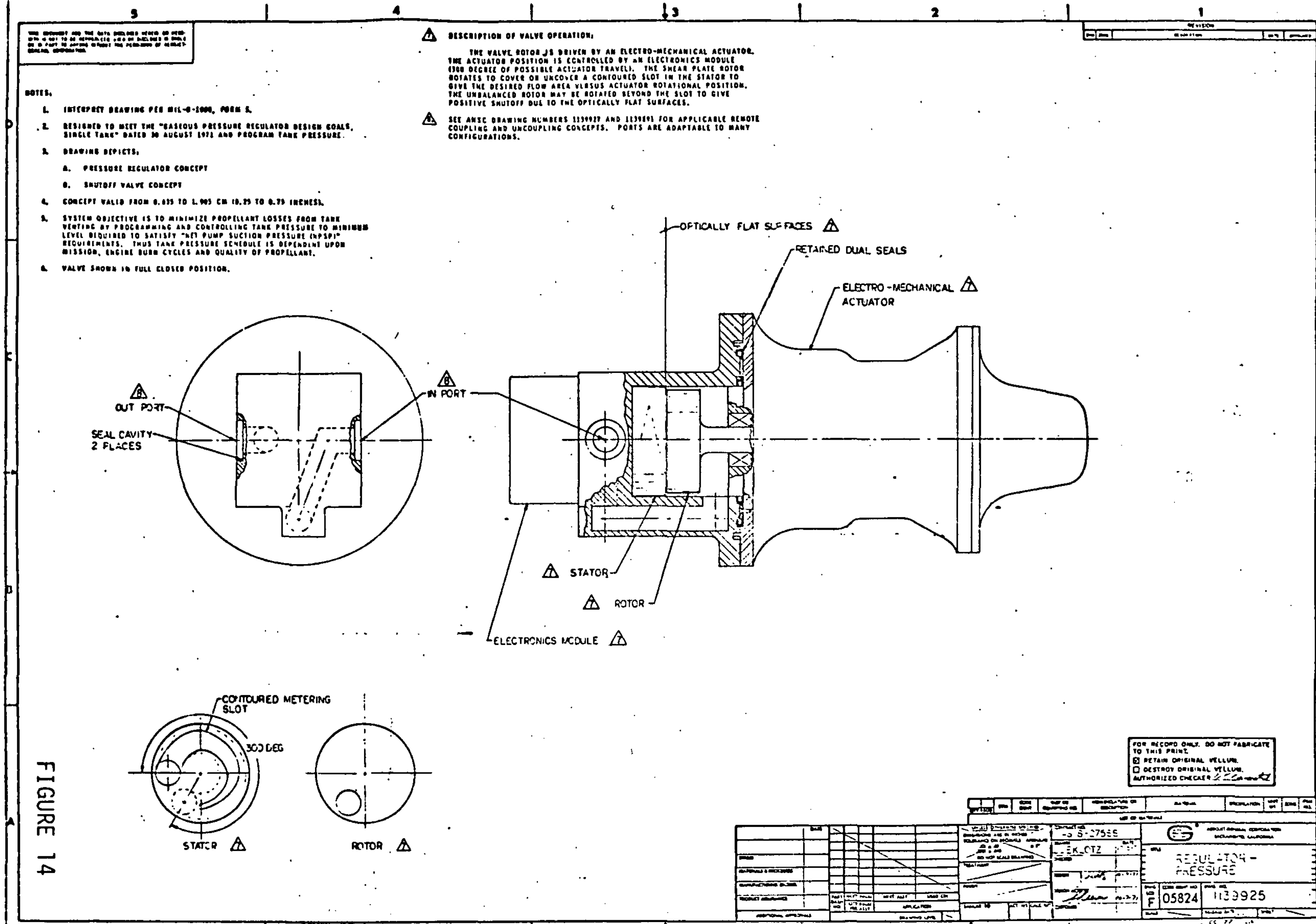
NAS 8-27566
R. BIANCHI
REGULATOR, FLOW CONTROL
E 34632 1139923

FIGURE 12



SYSTEM SCHEMATIC (PROGRAMMED TANK PRESSURE)
SYSTEM SCHEMATIC (CONSTANT TANK PRESSURE)
REGULATOR
FLOW CONTROL OR PRESSURE CONTROL
CONTROLLER
ACTUATOR TO VALVE COUPLING
ELECTRO-MECHANICAL ACTUATOR
STATIC SEAT
CAM FOLLER BEARING
REMOVABLE END CAP
GUIDE 2 PLACES
OPTICALLY FLAT SURFACES
OPERATING CRANK
SHEAR PLUG CARRIER
FLOATING SHEAR PLUG
FLOW DIRECTION
RETAINED SEAL 2 PLACES
DRIVEN SLOT
ACTUATOR ELECTRONICS MODULE
REGULATOR, PRESSURE OR FLOW CONTROL
34632

FIGURE 13


NOTES:
1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET THE "GASEOUS PRESSURE REGULATOR DESIGN GOALS, SINGLE TANK" DATED 30 AUGUST 1971 AND PROGRAM TANK PRESSURE.
3. DRAWING DEPICTS:
A. PRESSURE REGULATOR CONCEPT
B. SHUTOFF VALVE CONCEPT
4. CONCEPT VALID FROM 0.635 TO 1.905 CM (0.25 TO 0.75 INCHES).
5. SYSTEM OBJECTIVE IS TO MINIMIZE PROPELLANT LOSSES FROM TANK VENTING BY PROGRAMMING AND CONTROLLING TANK PRESSURE TO MINIMUM LEVEL REQUIRED TO SATISFY "NET PUMP SUCTION PRESSURE (NPSP)" REQUIREMENTS. THUS TANK PRESSURE SCHEDULE IS DEPENDENT UPON MISSION, ENGINE BURN CYCLES AND QUALITY OF PROPELLANT.
6. VALVE SHOWN IN FULL CLOSED POSITION.
7 DESCRIPTION OF VALVE OPERATION:
THE VALVE ROTOR IS DRIVEN BY AN ELECTRO-MECHANICAL ACTUATOR. THE ACTUATOR POSITION IS CONTROLLED BY AN ELECTRONICS MODULE (300 DEGREE OF POSSIBLE ACTUATOR TRAVEL). THE SHEAR PLATE ROTOR ROTATES TO COVER OR UNCOVER A CONTOURED SLOT IN THE STATOR TO GIVE THE DESIRED FLOW AREA VERSUS ACTUATOR ROTATIONAL POSITION. THE UNBALANCED ROTOR MAY BE ROTATED BEYOND THE SLOT TO GIVE POSITIVE SHUTOFF DUE TO THE OPTICALLY FLAT SURFACES.
8 SEE ANSC DRAWING NUMBERS 1139927 AND 1139891 FOR APPLICABLE REMOTE COUPLING AND UNCOUPLING CONCEPTS. PORTS ARE ADAPTABLE TO MANY CONFIGURATIONS.
OPTICALLY FLAT SURFACES 7
RETAINED DUAL SEALS
ELECTRO-MECHANICAL ACTUATOR 7
OUT PORT 8
IN PORT 8
SEAL CAVITY 2 PLACES
STATOR 7
ROTOR 7
ELECTRONICS MODULE 7
CONTOURED METERING SLOT
300 DEG
STATOR 7
ROTOR 7
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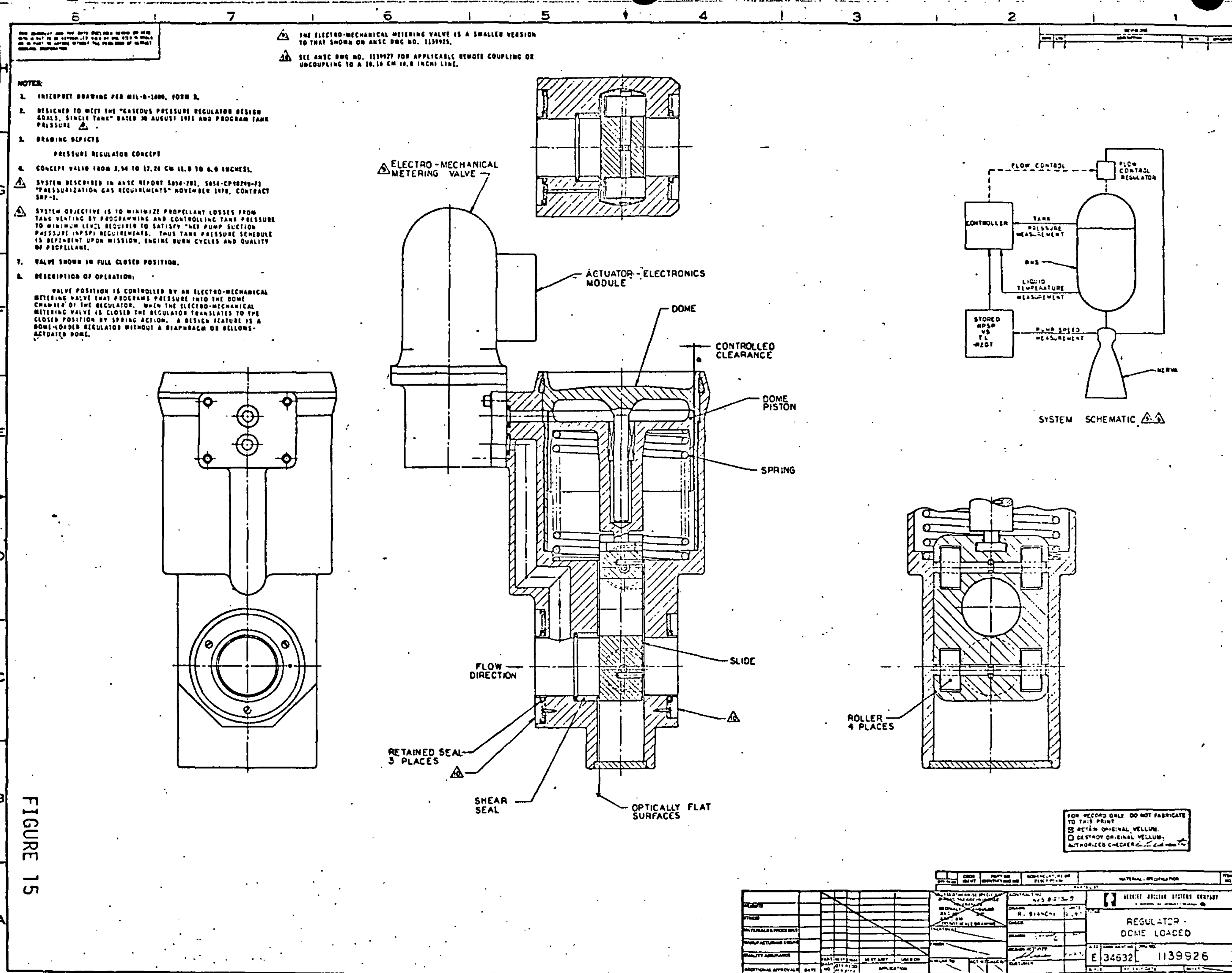
AUTHORIZED CHECKER
REGULATOR - PRESSURE
F 05824 1139925
1139925

FIGURE 14


NOTES:
1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET THE "GASEOUS PRESSURE REGULATOR DESIGN GOALS, SINGLE TANK" DATED 30 AUGUST 1971 AND PROGRAM TANK PRESSURE 6.
3. DRAWING DEPICTS
PRESSURE REGULATOR CONCEPT
4. CONCEPT VALID FROM 2.54 TO 12.70 CM (1.0 TO 6.0 INCHES).
5 SYSTEM DESCRIBED IN ANSC REPORT 5054-201, 5054-CP10290-F1 "PRESSURIZATION GAS REQUIREMENTS" NOVEMBER 1970, CONTRACT SNP-1.
6 SYSTEM OBJECTIVE IS TO MINIMIZE PROPELLANT LOSSES FROM TANK VENTING BY PROGRAMMING AND CONTROLLING TANK PRESSURE TO MINIMUM LEVEL REQUIRED TO SATISFY "NET PUMP SUCTION PRESSURE (NPSP) REQUIREMENTS. THUS TANK PRESSURE SCHEDULE IS DEPENDENT UPON MISSION, ENGINE BURN CYCLES AND QUALITY OF PROPELLANT.
7. VALVE SHOWN IN FULL CLOSED POSITION.
8. DESCRIPTION OF OPERATION:
VALVE POSITION IS CONTROLLED BY AN ELECTRO-MECHANICAL METERING VALVE THAT PROGRAMS PRESSURE INTO THE DOME CHAMBER OF THE REGULATOR. WHEN THE ELECTRO-MECHANICAL METERING VALVE IS CLOSED THE REGULATOR TRANSLATES TO THE CLOSED POSITION BY SPRING ACTION. A DESIGN FEATURE IS A DOME-LOADED REGULATOR WITHOUT A DIAPHRAGM OR BELLOWS-ACTUATED DOME.
9 THE ELECTRO-MECHANICAL METERING VALVE IS A SMALLER VERSION TO THAT SHOWN ON ANSC DWG NO. 1139925.
10 SEE ANSC DWG NO. 1139927 FOR APPLICABLE REMOTE COUPLING OR UNCOUPLING TO A 10.16 CM (4.0 INCH) LINE.
ELECTRO-MECHANICAL METERING VALVE
ACTUATOR - ELECTRONICS MODULE
DOME
CONTROLLED CLEARANCE
DOME PISTON
SPRING
SLIDE
FLOW DIRECTION
RETAINED SEAL 3 PLACES
SHEAR SEAL
OPTICALLY FLAT SURFACES
ROLLER 4 PLACES
FLOW CONTROL
FLOW CONTROL REGULATOR
CONTROLLER
TANK PRESSURE MEASUREMENT
LIQUID TEMPERATURE MEASUREMENT
STORED NPSP VS TL REQT
PUMP SPEED MEASUREMENT
SYSTEM SCHEMATIC
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RETAIN ORIGINAL VELLUM.
DESTROY ORIGINAL VELLUM.
AUTHORIZED CHECKER
R. BIANCHI
REGULATOR - DOME LOADED
E 34632 1139926
1139926

FIGURE 15

NOTES:

1. INTERPRET DRAWING PER MIL-D-[illegible], FORM 3.
2. DESIGNED TO MEET THE "GASEOUS PRESSURE REGULATOR DESIGN GOALS, SINGLE TANK" DATED 30 AUGUST 1971
3. DRAWING DEPICTS:

   PRESSURE REGULATOR CONCEPT
4. CONCEPT VALID FROM 2.54 TO 15.24 CM (1.0 TO 6.0 INCHES).
5. MAIN REGULATOR SHOWN IN FULL CLOSED POSITION WITH PILOT REGULATOR OPEN.
6. DESCRIPTION OF OPERATION:

   VALVE POSITION IS CONTROLLED BY A PILOT REGULATOR WHICH IS SET TO REGULATE A PRESET PRESSURE INTO THE MAIN REGULATOR DOME. THE PILOT REGULATOR IS OPEN WITH NO PRESSURE APPLIED TO THE SYSTEM. THE PILOT REGULATOR REFERENCES AMBIENT PRESSURE THROUGH THE USE OF A SPRING-LOADED BELLOWS. WHEN PRESSURE IS APPLIED TO THE SYSTEM, THE PILOT REGULATOR METERS FLOW INTO THE DOME AND THE MAIN REGULATOR OPENS. DOWNSTREAM PRESSURE (REGULATED PRESSURE) IS REFERENCED ACROSS THE DOME PISTON AREA. THE DIFFERENTIAL PRESSURE, BETWEEN PILOTED REGULATOR PRESSURE AND DOWNSTREAM PRESSURE, THEN POSITIONS THE MAIN REGULATING ELEMENT TO THE REQUIRED POSITION. AS DOWNSTREAM PRESSURE INCREASES THE DOME DIFFERENTIAL PRESSURE DECREASES AND THE MAIN ELEMENT METERS CLOSED DUE TO SPRING ACTION. LIKEWISE THE MAIN ELEMENT METERS OPEN AS THE DOWNSTREAM PRESSURE DECREASES, THUS MAINTAINING A CONTROLLED REGULATED PRESSURE ZONE.
7. SEE ANSC DWG. NO. 1139927 FOR APPLICABLE REMOTE COUPLING FOR UNCOUPLING TO A 10.16 (4.0 INCH) LINE.

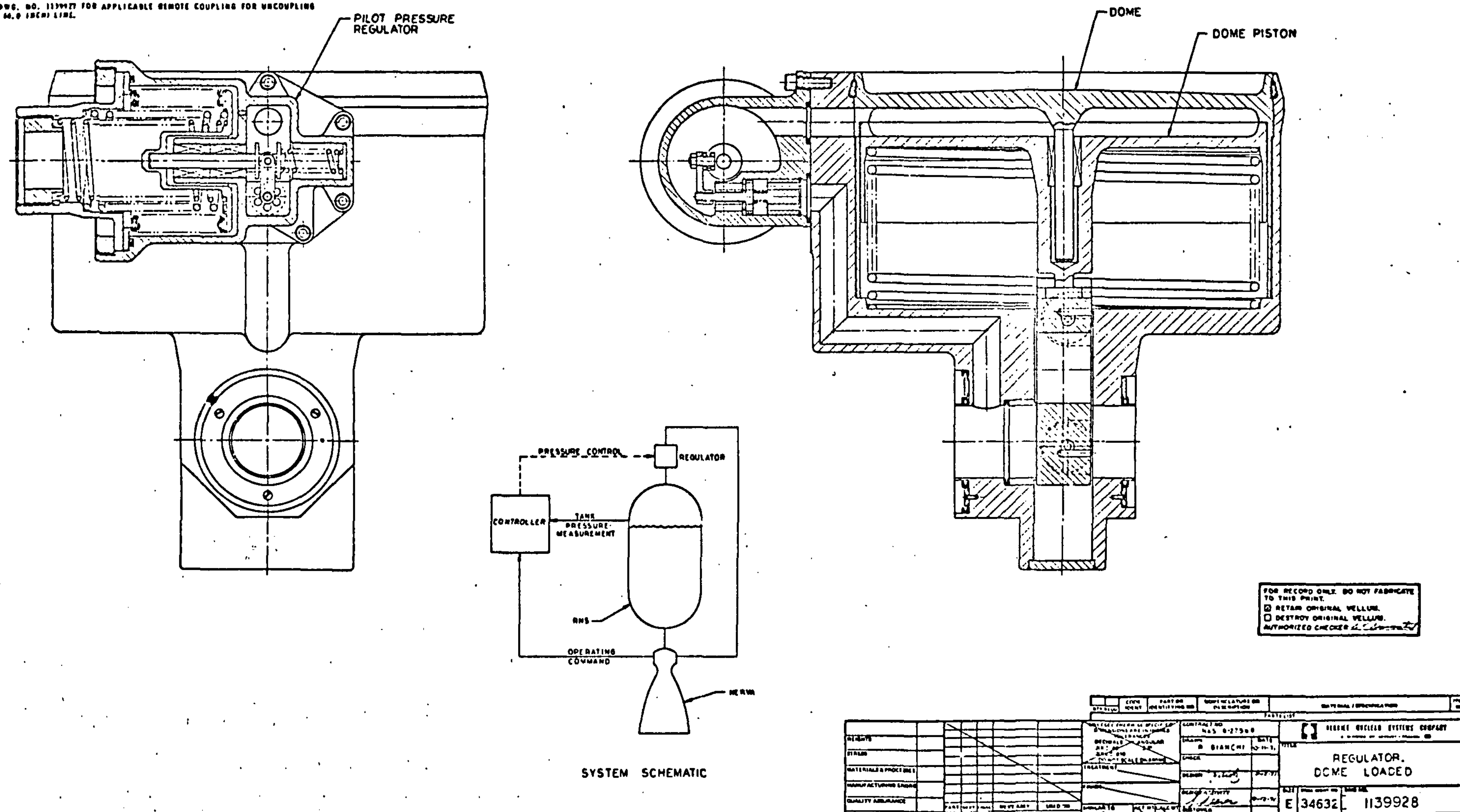
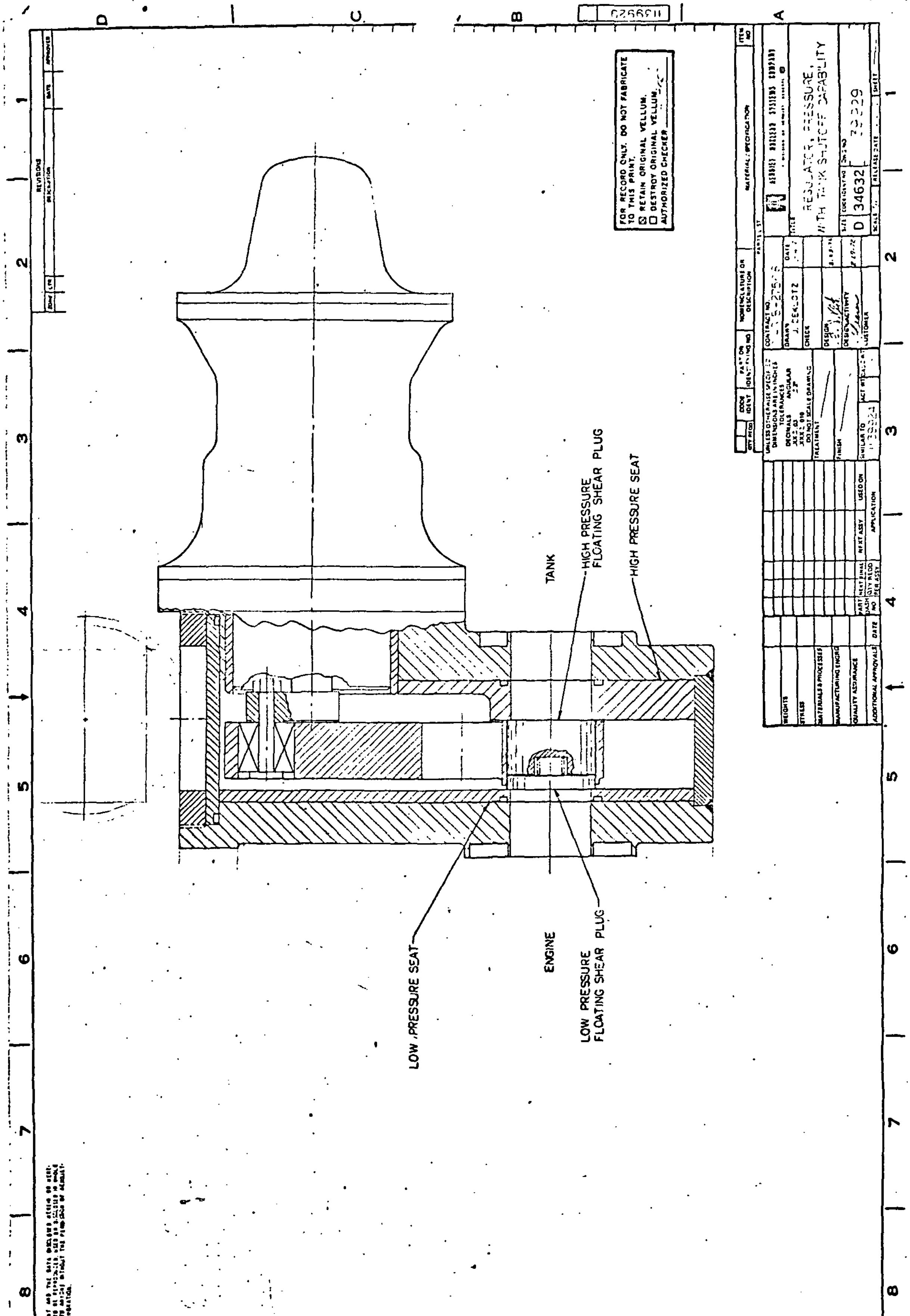


FIGURE 16


TANK
HIGH PRESSURE FLOATING SHEAR PLUG
HIGH PRESSURE SEAT
ENGINE
LOW PRESSURE SEAT
LOW PRESSURE FLOATING SHEAR PLUG
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DESTROY ORIGINAL VELLUM.
AUTHORIZED CHECKER
REGULATOR, PRESSURE, WITH TANK SHUTOFF CAPABILITY
D 34632

FIGURE 17

c. Remote Couplings

The five remote couplings completed during Phase I include three mechanical, one thermal interference and one low melting alloy concepts. These concepts are described in this section.

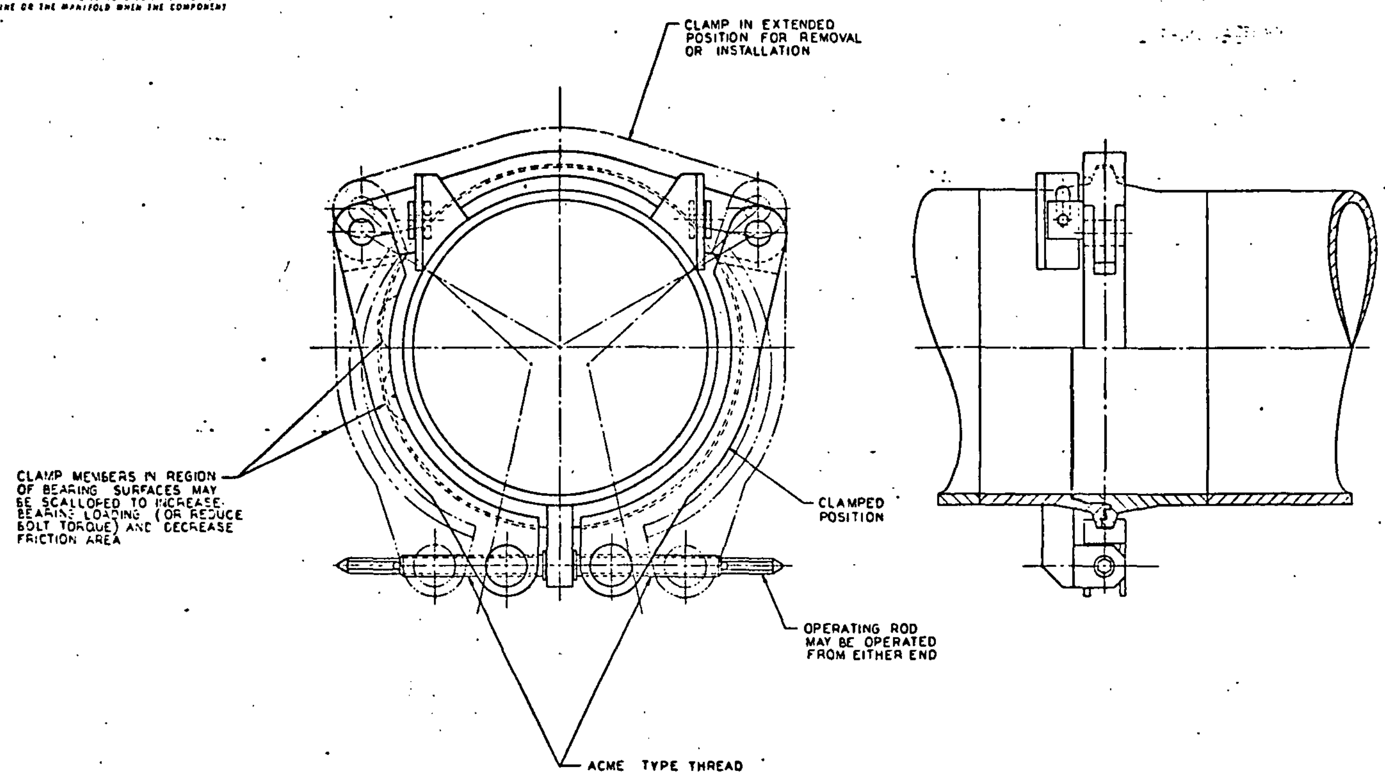
1139920, Concept No. 1, Figure (18) - This design is a recent approach to coupling component packages on NERVA. It varies from the existing Vee couplings in that it utilizes a three-segment device with two operating points that must be operated in synchronization. The coupling features two operating points either of which could be used for greater accessibility.

1139921, Concept No. 2, Figure (19) - The low melting alloy concept is a seal and a structural joint. After the liquid low melting alloy is injected into the joint cavity and then allowed to solidify, it structurally resists pressure area loads in shear. Sealing is accomplished due to expansion on cooling and due to pressure area loads acting on the inclined plane (re-entrant cavity). The concept provides a simple joint without the nuts and bolts approach. NASA-Lewis has used joints similar to this concept with success in laboratory pressure vessels. The barrier seals are provided to prevent the liquid metal from migrating internally or externally during injection or ejection. For injection, the integral cal-rod heaters are activated. The liquid metal supply is attached to a port in the joint and pumped into the joint cavity until full. A similar port is located in the joint opposite the supply port to allow excess liquid to be collected into a porous catch receptacle that is impervious to liquid. Space vacuum could aid in the pumping operation. Ejection would be accomplished by pumping the liquid out of the joint cavity pneumatically and collecting it in a porous catch receptacle. The metal thus can be salvaged for future use or disposed of as desired.

1139922, Concept No. 3, Figure (20) - The thermal interference joint is a simple shrink fit where a thin wall tube, initially fabricated to a predetermined interference with its mating male part, is expanded to a predetermined clearance by heating. The mating male part is then inserted and the tube is allowed to cool and shrink around the male part thus generating high contact stresses which could provide good sealing. The seal requires structural support from an additional mechanical device. Surface finishes, materials, dimensions, and plating must be studied and evaluated in order to determine the effectiveness of this sealing joint.

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET "GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY" DATED 30 JULY 1971.
3. DRAWING DEPICTS:
   - A. REMOTE COUPLING AND SEALING OF A LINE TO A LINE OR A MANIFOLD.
   - B. REMOTE COUPLING AND SEALING OF A COMPONENT TO A LINE OR TO A MANIFOLD.
4. CONCEPT VALID FROM 0.635 TO 50.800 CM (0.25 TO 20.00 INCH).
5. CONCEPT VALID FOR LINE-TO-LINE JOINTS OR COMPONENT-TO-LINE OR TO MANIFOLD JOINTS.
6. *DESCRIPTION OF OPERATION:* TO COUPLE, THE OPERATING ROD IS ROTATED TO LOCATE THE MECHANISM IN THE EXTENDED POSITION. THE TWO PARTS ARE THEN MATED AND THE OPERATING ROD IS ROTATED TO MOVE THE MECHANISM TO THE CLAMPED POSITION. FINAL TORQUE PROVIDES THE STRUCTURAL JOINT AND ESTABLISHES THE SEAL. A DESIGN FEATURE IS THAT THE REMOTE OPERATION IS ACCOMPLISHED BY APPLICATION OF THE MANIPULATOR TOOL AT ONLY ONE OPERATING ROD POINT. TWO OPERATING ROD POINTS ARE PROVIDED (EITHER OF WHICH CAN BE USED), THUS ENHANCING ACCESSIBILITY. AN ADDITIONAL FEATURE IS THAT THE CLAMP REMAINS ATTACHED TO THE LINE OR THE MANIFOLD WHEN THE COMPONENT TO BE REPLACED IS REMOVED.



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REMOTE COUPLING - STRUCTURAL JOINT AND SEAL

R. BIANCHI

NAS 8 27568

E 34637 1139920

FIGURE 18

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET "GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY" DATED 30 JULY 1971.
3. DRAWING DEPICTS:
   A. REMOTE COUPLING AND SEALING OF A LINE TO A LINE OR A MANIFOLD.
   B. REMOTE COUPLING AND SEALING OF A COMPONENT TO A LINE OR TO A MANIFOLD.
4. CONCEPT VALID FROM 0.635 TO 50.800 CM (0.25 TO 20.00 INCHES).
5. CONCEPT VALID FOR LINE-TO-LINE JOINTS OR COMPONENT-TO-LINE OR TO MANIFOLD JOINTS.
6. DESCRIPTION OF OPERATION: TO COUPLE, THE TWO MATING PARTS ARE MATED WITH THE BARRIER SEALS IN PLACE. THE THERMOSTATICALLY CONTROLLED HEATING ELEMENTS ARE ACTIVATED TO HEAT THE LOW MELTING ALLOY TO THE LIQUID STATE. THE LIQUID LOW MELTING ALLOY IS THEN INJECTED INTO THE JOINT CAVITY UNTIL FULL. THE HEATER IS THEN DEACTIVATED AND THE LOW MELTING ALLOY IS ALLOWED TO COOL AND SOLIDIFY AND THUS FORMS A STRUCTURAL JOINT (DUE TO SHEAR AREA) AND A PRESSURE-AIDED SEAL (DUE TO INCLINED PLANE). THE JOINT CAN BE UNCOUPLED BY HEATING THE LOW-MELTING ALLOY TO THE LIQUID STATE AND REMOVING IT BY AN EJECTION SYSTEM. CANDIDATE INJECTION AND EJECTION SYSTEMS ARE SHOWN SCHEMATICALLY.
7. DESIGN PROOF PRESSURE OF JOINT SHOWN IS 917,035 N/m^2 (133 LBF/IN^2) BASED ON THE FOLLOWING VALUES:

SHEAR DIAMETER	0.1143 m	(4.5 IN.)
SHEAR WIDTH	0.0254 m	(1.0 IN.)
ALLOWABLE SHEAR STRENGTH	1,034/250 N/m^2	(150 LBF/IN^2)
TEMPERATURE	294°K	(70°F)
APPROXIMATE LINE SIZE	0.0254 m	(4.0 IN.)

8. ESTIMATED INJECTION OR EJECTION PRESSURE - 10,343 N/m^2 (15 PSI).
9. ENVIRONMENTAL COMPATIBILITY OF LOW-MELTING ALLOY TO BE ESTABLISHED (SEE TABLE 1).
10. SOME TYPICAL LOW-MELTING ALLOYS TO BE CONSIDERED AND THEIR PERTINENT PROPERTIES ARE SHOWN IN TABLE 1.
11. THE INJECTION AND EJECTION SYSTEMS ARE REMOVED FROM THE JOINT WHEN NOT IN USE.

TABLE 1 (9) (10)

PROPERTIES	EUTECTICS		NON-EUTECTICS	
	Cerrobase*	Cerrotru*	Cerromatrix*	Cerrocast*
Melting Temperature (°F) Range (°F)	255 255-255	281 281-281	(see yield temp. 218-440	281-338
Yield Temp (°F)	255	281	240	302
Weight Lb/In.^{2}	.380	.315	.343	.296
Specific Gravity @ 20°C	10.3	8.7	9.5	8.2
Tensile Lb/In.^{2} (70°)	6400	8000	13000	8000
*Elongation in 2" Slow Loading %	60-70	200*	Less than 1%	200*
Brinell Hardness #	10.2	22	19	22
*Specific Heat Liquid	.042	.045	.04	.047
*Specific Heat Solid	.03+	.045	.045	.047
*Latent Heat--Fusion Btu/Lb.	7.2	20		22
*Coefficient of Thermal Expansion	.000021/°C	.000015/°C	60°-130F. = .000022/°C; 130°-218/F Zero	.000015/°C
Thermal Conductivity (Solid)Cal/Cm^2/°C/Sec .94 = Copper	*.04	*.05	—	*.09
Conductivity (Electrical) Compared with Pure Copper	1.75%	5.00%	2.57%	7.77%
Resistivity, OHMS based on volume standard (Meter, MM^2)	.8825	.3445	.6696	.2219
▲Maximum Load--30 Seconds Lb/In.^{2} - YTS	8000	15000	16000	15000
▲Maximum Load--5 Minutes Lb/In.^{2} - YTS	4000	9000	10000	9500
▲Safe Load—Sustained Lb/In.^{2} - YTS	300	500	300	500
Volume Change (Liquid to Solid)	-1.5%	+0.77%	-1.5%	*+0.5%
Volume Change (Linear growth after solidification)	0.5%	0.05%	0.5%	*0%
▲Approximate ROOM TEMPERATURE VALUES				
Compositions (%)				
Bismuth	55.50	58.00	48.00	40.00
Lead	44.50		28.50	
Tin		42.00	14.50	60.00
Antimony			9.00	
Nuclear Properties	TBD	TBD	TBD	TBD
Working Temperature Range (°F)	-453 TO +70	-453 TO +70	-453 TO +70	-453 TO +70

* Data extracted from Cerro Sales Corp. bulletin.

* Trade mark of Cerro Sales Corp.

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CONTRACT NO. NAS 9-27568
DRAWN R. BIANCHI DATE 10-4-71

TITLE: LOW MELTING ALLOY STRUCTURAL COUPLING & SEAL

SIZE D CODE IDENT NO. 34632 DWG NO. 1139921

FIGURE 19

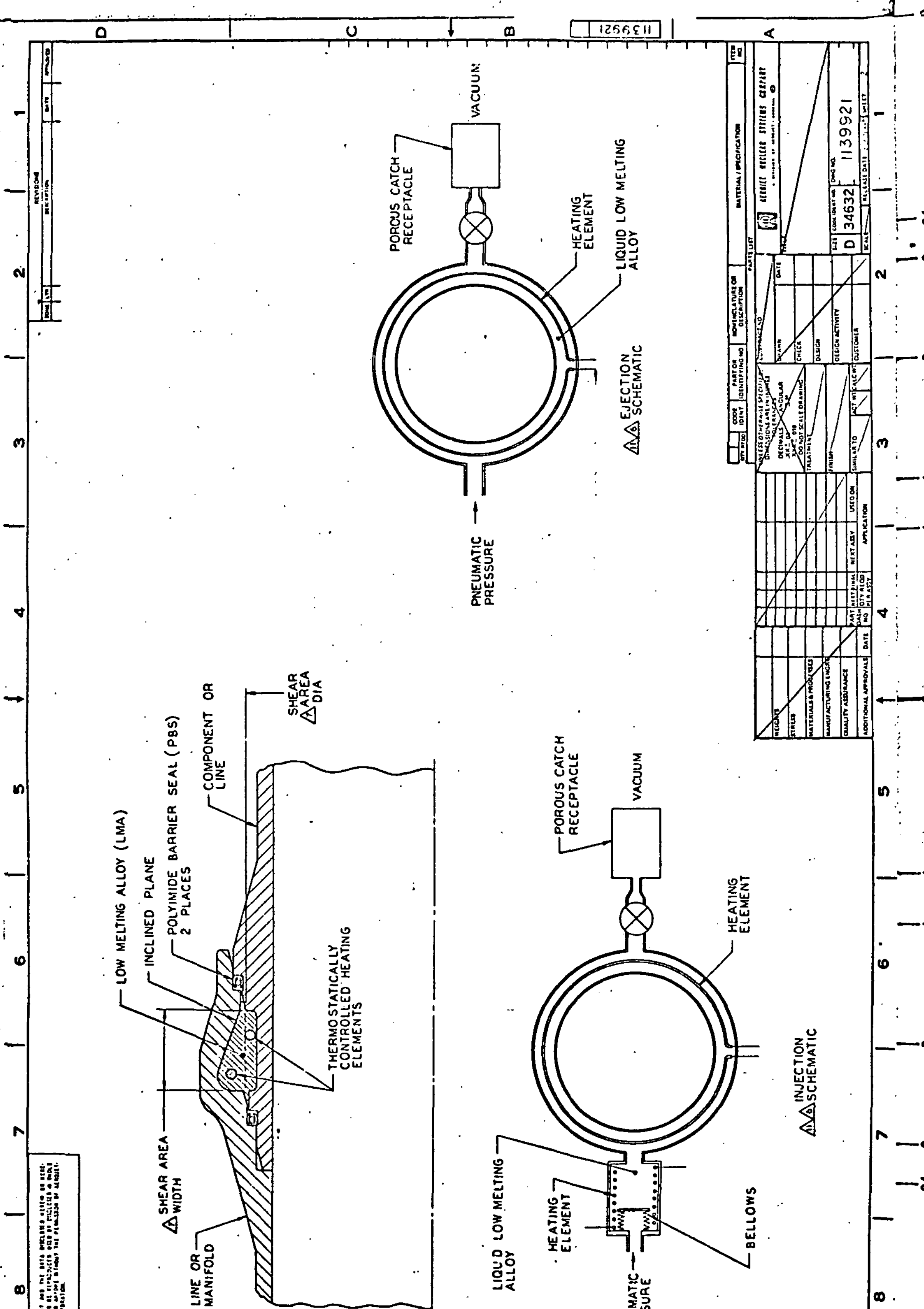
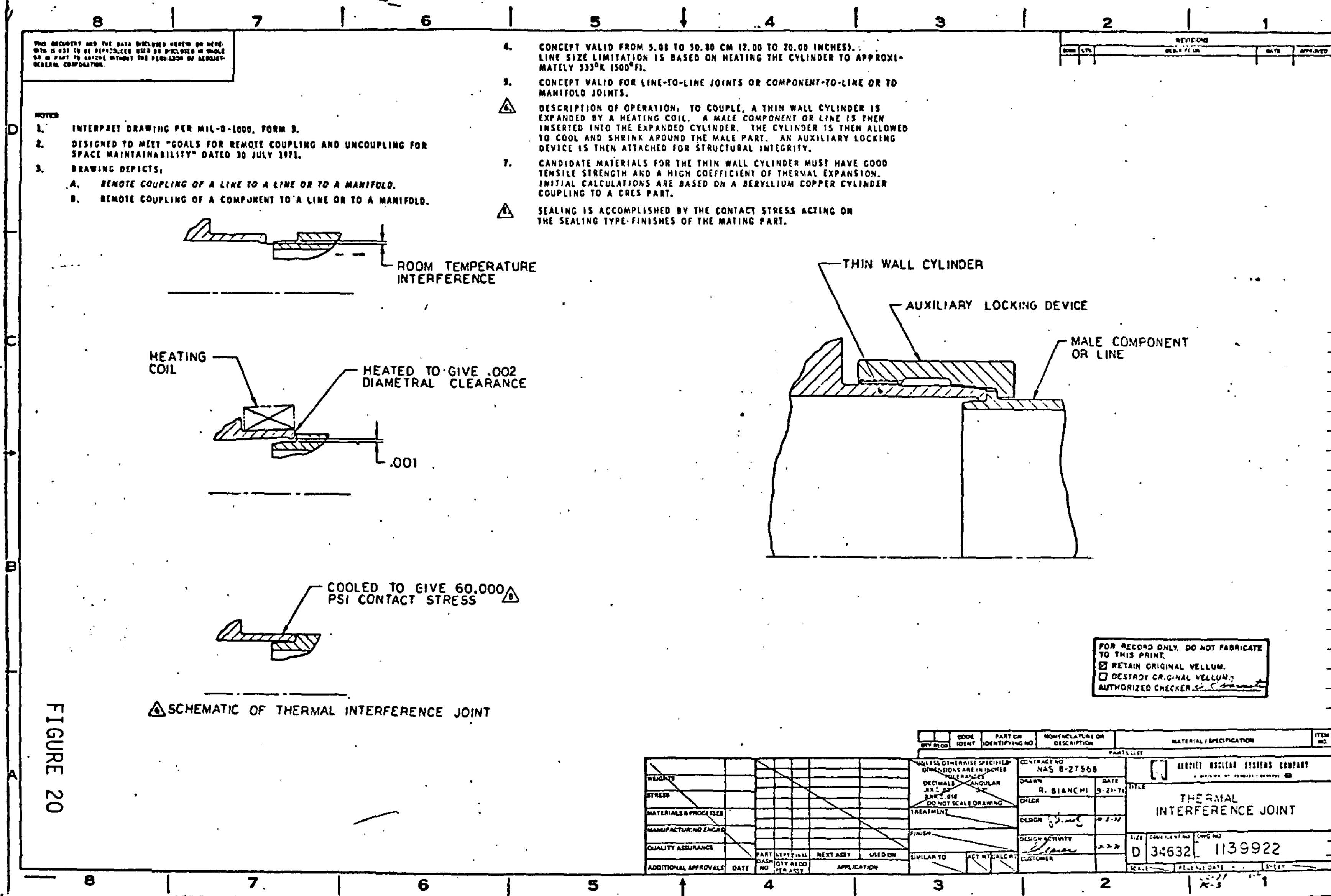


FIGURE 19

Page 2 of 2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NOTES
1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET "GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY" DATED 30 JULY 1971.
3. DRAWING DEPICTS:
A. REMOTE COUPLING OF A LINE TO A LINE OR TO A MANIFOLD.
B. REMOTE COUPLING OF A COMPONENT TO A LINE OR TO A MANIFOLD.
4. CONCEPT VALID FROM 5.08 TO 50.80 CM (2.00 TO 20.00 INCHES). LINE SIZE LIMITATION IS BASED ON HEATING THE CYLINDER TO APPROXIMATELY 533°K (500°F).
5. CONCEPT VALID FOR LINE-TO-LINE JOINTS OR COMPONENT-TO-LINE OR TO MANIFOLD JOINTS.
6. DESCRIPTION OF OPERATION. TO COUPLE, A THIN WALL CYLINDER IS EXPANDED BY A HEATING COIL. A MALE COMPONENT OR LINE IS THEN INSERTED INTO THE EXPANDED CYLINDER. THE CYLINDER IS THEN ALLOWED TO COOL AND SHRINK AROUND THE MALE PART. AN AUXILIARY LOCKING DEVICE IS THEN ATTACHED FOR STRUCTURAL INTEGRITY.
7. CANDIDATE MATERIALS FOR THE THIN WALL CYLINDER MUST HAVE GOOD TENSILE STRENGTH AND A HIGH COEFFICIENT OF THERMAL EXPANSION. INITIAL CALCULATIONS ARE BASED ON A BERYLLIUM COPPER CYLINDER COUPLING TO A CRES PART.
8. SEALING IS ACCOMPLISHED BY THE CONTACT STRESS ACTING ON THE SEALING TYPE FINISHES OF THE MATING PART.
ROOM TEMPERATURE INTERFERENCE
HEATING COIL
HEATED TO GIVE .002 DIAMETRAL CLEARANCE
.001
COOLED TO GIVE 60,000 PSI CONTACT STRESS 8
6 SCHEMATIC OF THERMAL INTERFERENCE JOINT
THIN WALL CYLINDER
AUXILIARY LOCKING DEVICE
MALE COMPONENT OR LINE
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1139922
CONTRACT NO NAS 8-27568
DRAWN R. BIANCHI
AEROJET NUCLEAR SYSTEMS COMPANY
THERMAL INTERFERENCE JOINT
D 34632 1139922

FIGURE 20

1139927, Concept No. 4, Figure (21) - This coupling concept was designed for most of the in-line regulator concepts but is adaptable to many types of in-line components. The coupling is a gear driven device, with two manipulator operating points, either of which can be used for accessibility. This coupling requires a line mounted expansion joint or elbow for coupling or uncoupling. Face seal surfaces are protected from damage during installation or removal of the regulator or other component.

1139891, Concept No. 5, Figure (22) - This design is a remote coupling that does not require an additional expansion joint in the upstream or downstream lines. Two manipulator operating points are provided, either of which can be used for greater accessibility. Component replacement is relatively simple. A good mechanical advantage in the ball screw is available for high force loading for the two face seals. The two bellows are protected from torsion loads because of an indexing key that prevents rotation of the mechanism during operation of the ball screw.

5. Design Selections for Phase II

At the completion of the Phase I design work, ANSC recommended three regulators and three valves for further effort during Phase II. The basis for this selection was a design selection matrix comparing each concept to each requirement using a weighted numerical scoring system. The selection matrix is presented in Appendix G. The results were then summarized in Appendixes H, I, and J. The ANSC recommendations are shown in Tables IV and V.

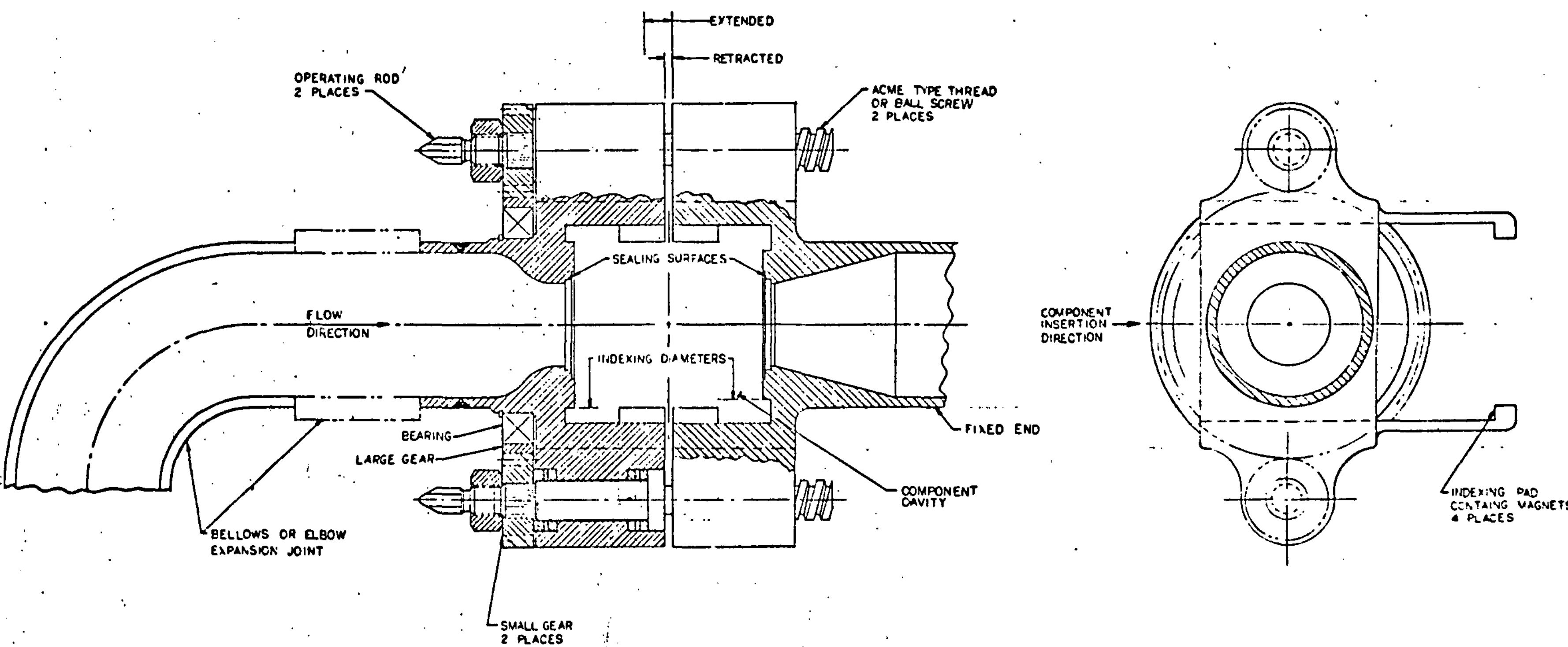
The drawings, selection material, and other Phase I material was then reviewed in a meeting at the George C. Marshall Space Flight Center (MSFC) Huntsville, Alabama in Building 4610, on 19 October 1971.

Attendees:	Kenneth Anthony	S&E - ASTN - ENC
	Jack Potter	" " "
	L. D. Johnson	Aerojet Nuclear Systems Company
	E. A. Shearer	" " " "
	V. A. Smith	" " " "
	J. E. DeKlotz	" " " "

Authority to proceed into Phase II was received in a letter dated 17 November 1971 from the contracting officer, William J. McKinney. The design concepts to proceed with, throug Phase II, were:

NOTES:

1. INTERPRET DRAWING PER MIL-[illegible], FORM 3.
2. DESIGNED TO MEET "GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY" DATED 30 JULY 1972.
3. DRAWING DEPICTS:
   A. REMOTE COUPLING FOR A REGULATOR TO A LINE.
   B. REMOTE COUPLING FOR A SHUTOFF VALVE TO LINE.
4. CONCEPT VALID FROM 0.635 TO 50.800 CM (0.25 TO 20.00 INCHES).
5. CONCEPT VALID FOR LINE-TO-LINE JOINTS OR COMPONENT-TO-LINE OR TO MANIFOLD JOINTS.
6. DESCRIPTION OF OPERATION. TO COUPLE A COMPONENT, THE OPERATING ROD IS ROTATED TO LOCATE THE MECHANISM IN THE EXTENDED POSITION. A COMPONENT SUCH AS SHOWN ON DRAWINGS 1139974 OR 1139976 IS INSERTED IN THE DIRECTION SHOWN AND PLACED AGAINST INDEXING PADS (INDEXING PADS AND BOTTOM OF COMPONENT CAN CONTAIN MAGNETS FOR ZERO G INDEXING). THE OPERATING ROD IS THEN ROTATED TO THE RETRACTED POSITION, THUS ESTABLISHING THE STRUCTURAL JOINT AND FINAL SEAL. A DESIGN FEATURE IS THE REMOTE OPERATION IS ACCOMPLISHED BY APPLICATION OF THE MANIPULATOR TOOL AT ONLY ONE OPERATING ROD POINT. TWO OPERATING ROD POINTS ARE PROVIDED (EITHER OF WHICH CAN BE USED SINCE THEY ARE GEARED TOGETHER), THUS ENHANCING ACCESSIBILITY. THE COUPLING CAVITY IS DESIGNED SO THAT THE COMPONENT CANNOT BE INSTALLED BACKWARD. THE SEALING SURFACES ARE RE-ENTRANT SO THEY WILL NOT BE DAMAGED WHEN INSTALLING OR REMOVING THE COMPONENT.



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REMOTE COUPLING - INLINE

E 34632 1139927

FIGURE 21

NOTES:

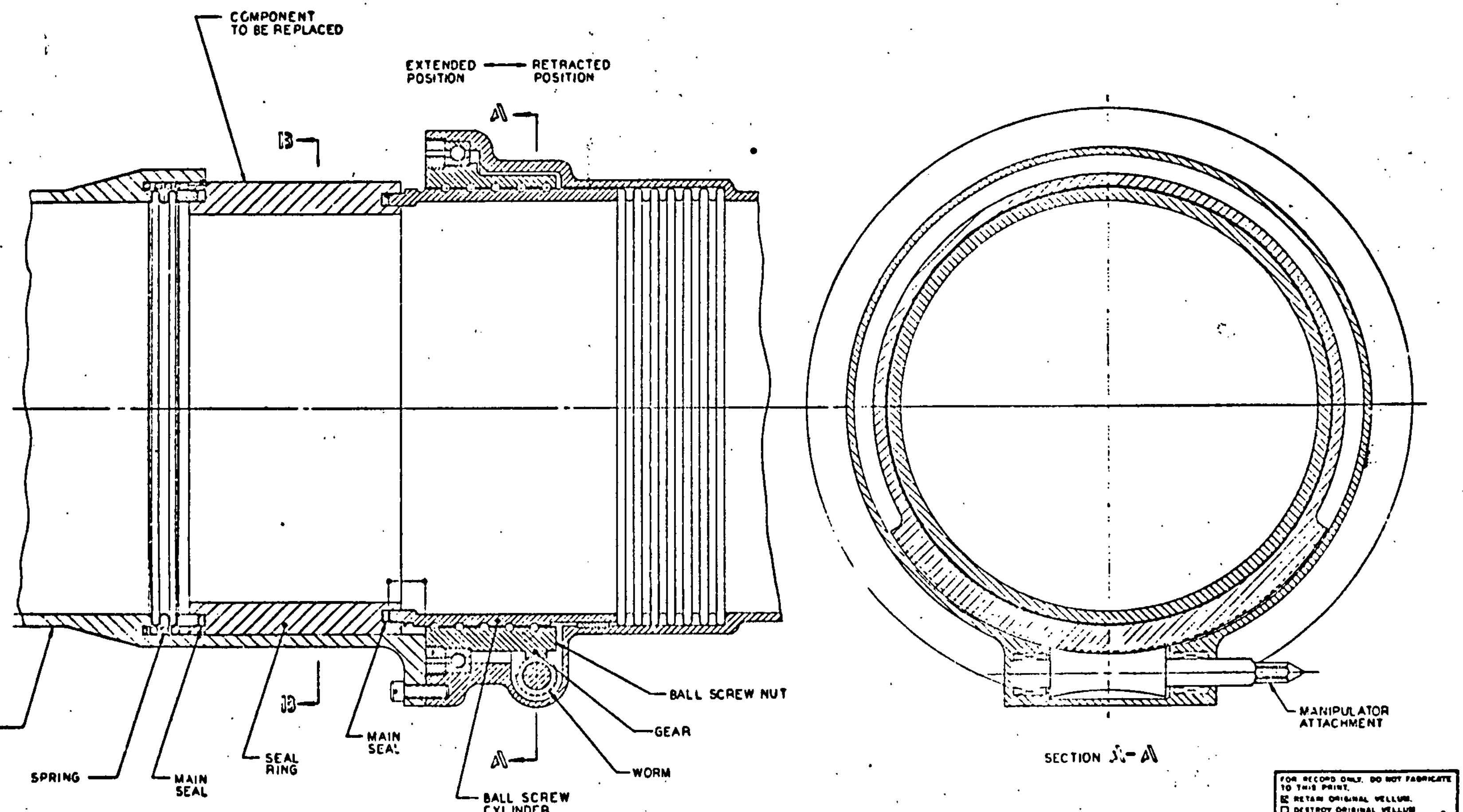
1. INTERPRET DRAWING PER MIL-[illegible], FORM 3.
2. DESIGNED TO MEET "GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY" DATED 30 JULY 1971.
3. DRAWING DEPICTS:

   REMOTE COUPLING AND UNCOUPLING OF A LINE TO A LINE AND/OR A COMPONENT TO A LINE.

4. CONCEPT IS VALID FROM .635 TO 50.80 CM (1/4 TO 20 INCHES).
5. DESCRIPTION OF OPERATION:

   TO INSTALL THE SEAL RING, VALVE OR COMPONENT, THE BALL SCREW CYLINDER MUST FIRST BE ACTUATED INTO THE RETRACTED POSITION BY DRIVING THE WORM GEAR. THE COMPONENT IS THEN GUIDED THRU AN OPENING IN THE HOUSING UP TO THE STOP GUIDES (SEE SECTION B-B) & CLAMPED INTO POSITION BY DRIVING THE BALL SCREW CYLINDER FORWARD.

   TO REMOVE THE SEAL RING THE BALL SCREW CYLINDER IS ACTUATED INTO THE RETRACTED POSITION TO ALLOW THE SPRING TO PUSH THE SEAL RING COMPONENT INTO POSITION FOR REMOVAL.



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CONTRACT NO. NAS 8-27560
V. SMITH

CONCEPT - INLINE REMOTE COUPLING

E 34632 1139891

FIGURE 22
Page 1 of 2

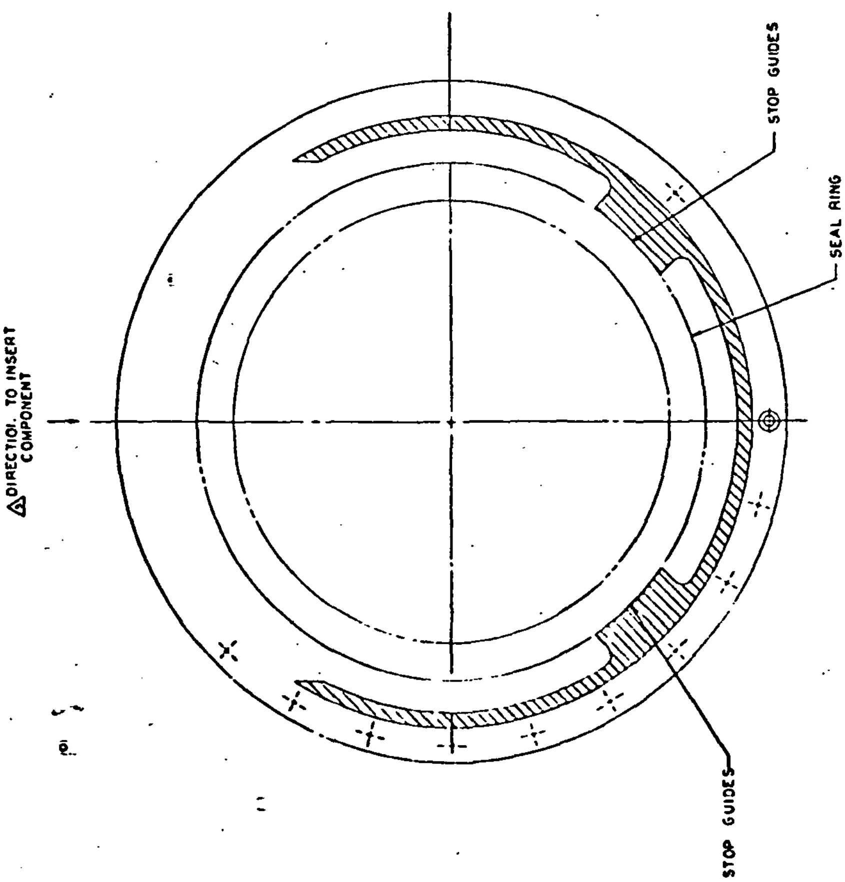


FIGURE 22
Page 2 of 2

TABLE IV

Contract NAS 8-27568

RECOMMENDATIONS
SHUTOFF VALVES

Rating	Concept No.	Dwg. No.	Description	Reason for Recommendation	Requirement/Source
2	5	1139895	Similar to sketch shown. Will have capability of a parametric test bed to measure variables involved.	If the concept is feasible there can be no simpler valve.	Size to optimize test flexibility and minimize costs.
1	4	1139894	Basic design shown.	Optimize the design of one valve, then by building blocks, determine how many applications and sizes can be satisfied. Compare results with conventional valves.	McDonnell-Douglas Nuclear Shuttle definition study G-7134, Phase III, Final Report, Vol. II.
	2	1139892	Basic design as shown except removable angle drive actuator. See 1139930.	Provide a space maintainable tank shutoff valve which can be replaced wholly or in part with a filled LH_2 tank.	10 inch: tank valve or vehicle fill and drain valve.

TABLE V

Contract NAS 8-27568

RECOMMENDATIONS

PRESSURE REGULATORS

Rating	Concept No.	Dwg. No.	Description	Reason for Recommendation	Requirement/Source
2	2	1139924	Basic design shown	Low leakage, long life potential of a shear plug metering element - good control range.	Approximate line O.D. 4.00 inches Inlet pressure range 723 psia to 219 psia (1) Inlet temperature range 238°R to 415°R (2) Minimum flow rate 4.00 lbm/sec (3) Main tank pressurization regulator - single tank.
1	3	1139925	Basic design shown	Low leakage, long life potential of rotary shear plate metering element. Broad control range.	Approximate line O.D. 1.00 inch Inlet pressure range (1) (4) 723 psia to 547 psia 850 psia to 400 psia (5) Inlet temperature range 238°R to 415°R (2) Minimum flow rate 0.82 lbm/sec (4) 0.60 lbm/sec (5)
3	1	1139923	Basic design shown	Long life, broad control range.	Same as Concept 2

(1) 723 psia to 319 psia - ANSC N4110:0067.
319 psia -100 psia = 219 psia (due to assumed line drop)

(2) 238°R to 315°R - ANSC N4110:0067.
315°R +100°R = 415°R (due to line temperature rise)

(3) LMSC A984555, Vol. VII - Sec. 3, 1 May 1971.

(4) Minimum pressure based on 415°R and 0.82 lbm/sec - 0.83 lbm/sec based on (3).

(5) 850 psia based on McDonnell-Douglas MDC G2134, Vol. II, Part B, Book 2 - Multiple Tank System, May 1971 - minimum pressure based on 415°R and 0.60 lbm/sec.

A. VALVE CONCEPTS

1. ANSC 1139893, In-Line Poppet Valve, Motor Operated.
2. ANSC 1139895, 3/4-Inch In-Line Valve, Electromagnetic Seal.
3. ANSC 1139897, Poppet Valve - Liquid Metals Seals.

B. REMOTE COUPLING CONCEPTS

1. ANSC 1139921, Low Melting Alloy Structural Coupling and Seal.
2. ANSC 1139922, Thermal Interference Joint.

C. REGULATOR CONCEPT

1. ANSC 1139924, Regulator, Pressure, or Flow Control.

B. PHASE II, PRELIMINARY DESIGN

The Phase II design effort was initiated with the revision of the design goals which are included herein as Appendixes K, L, and M. These goals serve as preliminary design specifications and contain all of the major parameters required for the RNS application. Each of the valve and regulator designs was designed to accept the remote couplings which were also designed during this period. The valve line sizes were selected from the MDAC RNS studies to cover a broad range of valve sizes. The magnetohydrodynamic (MHD) valve was designed as a tester to evaluate the variable parameters pertinent to the ionization principle.

1. <u>Valves</u>

The three valve concepts studied feature unusual methods of performing the shutoff sealing function. One valve has no moving parts and utilizes the principle of gas ionization of the hydrogen to retard flow. The other two valves use liquid metal and the heating of contained hydrogen.

a. MHD Valve Concept

The magnetohydrodynamic (MHD) valve concept shown in Figure 23, Drawing 1140003, is a valve design in which (MHD) forces are used to modulate gaseous flow or completely impede the flow.

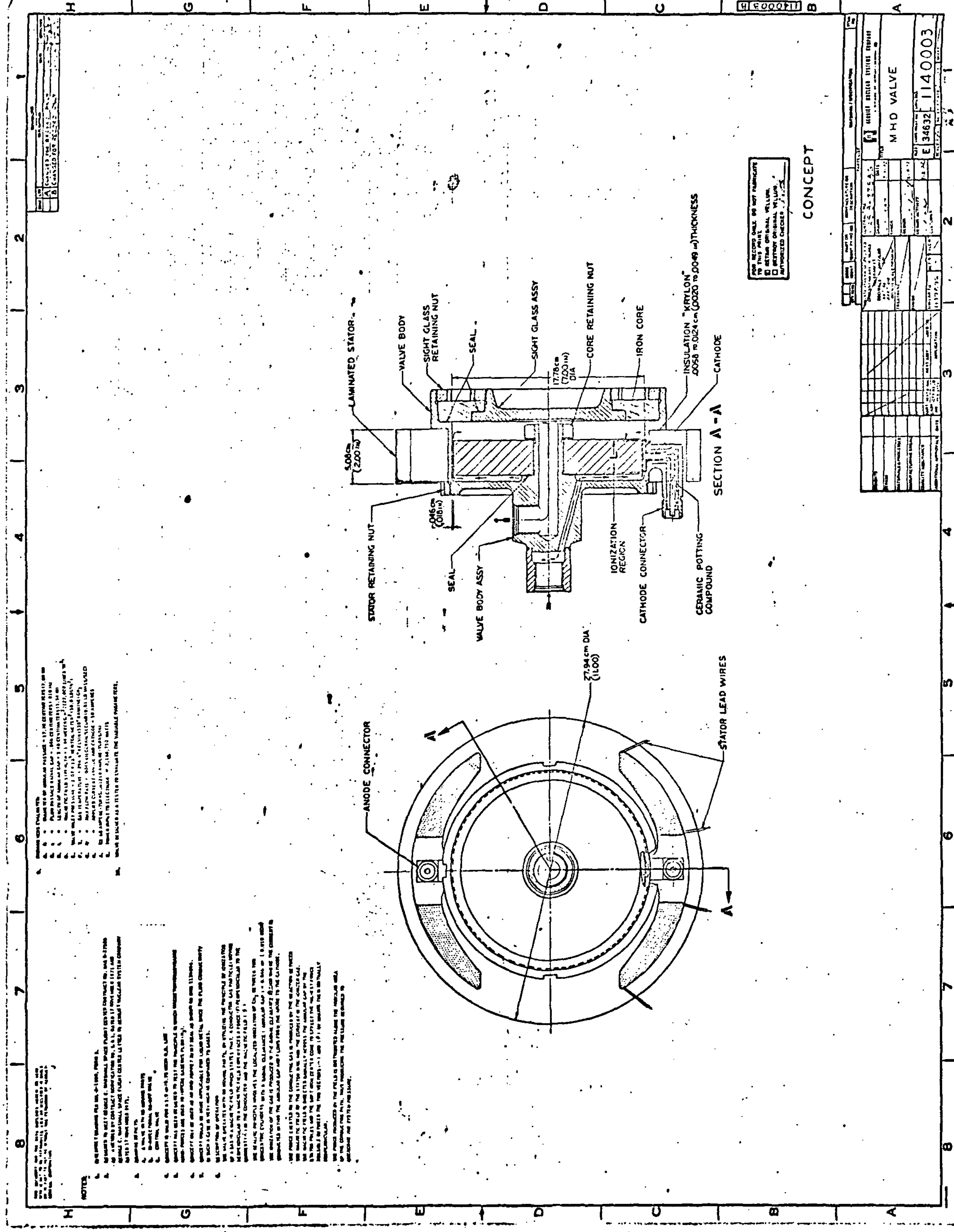

MHD VALVE
E 34632 1140003
CONCEPT
SECTION A-A
LAMINATED STATOR
VALVE BODY
SIGHT GLASS RETAINING NUT
SEAL
SIGHT GLASS ASSY
17.78cm (7.00 in) DIA
CORE RETAINING NUT
IRON CORE
INSULATION "KRYLON" .0058 TO .0124 cm (.0020 TO .0049 in) THICKNESS
CATHODE
5.08cm (2.00 in)
.046cm (.018 in)
STATOR RETAINING NUT
SEAL
VALVE BODY ASSY
IONIZATION REGION
CATHODE CONNECTOR
CERAMIC POTTING COMPOUND
27.94 cm DIA (11.00)
ANODE CONNECTOR
STATOR LEAD WIRES
NOTES:

FIGURE 23

The principal of the concept is based on the ionization of a gas within a magnetic field, whereby a conductor (ionized gas particle) moving perpendicular to the magnetic field experiences a force (F) perpendicular to the current (i) and the magnetic flux field (β). The magnetic field vectors are so aligned that the force may be expressed as follows:

$$F = i\beta$$

The sealing principal (Figures 24 and 25) involves the localized ionization of the gas between two concentric cylinders with an 0.0457 cm (0.018 inch) radial clearance identified as g in Figure 25. The ionization of the gas is produced in the radial clearance region, where the current is conducted within the annular gap and flows from the anode to the cathode. The force exerted on the ionized gas is produced by the magnetic field of the stator ring whereby the field is diverted in a radial direction by the soft iron center core. The vectors are thus aligned in order to effect the highest possible force between the two vectors (i and β). The force produced by the field is distributed along the annular area of the conducting path, thus producing the back pressure necessary to overcome the system pressure.

A visual sight glass has been designed into the housing to observe the glow discharge characteristics for several values of volt-ampere, flux density and pressure parameters. The following equations have been utilized in, or to establish the basic concept, as shown on ANSC drawing 1140003. Both U. S. Customary and System Internal (S.I.) units have been used in these calculations. However, the final results of important parameters are always given in S.I. units. Equations are extracted from "Marks Handbook" and other common sources except as noted in Footnote 1. Calculations are based on room temperature conditions.

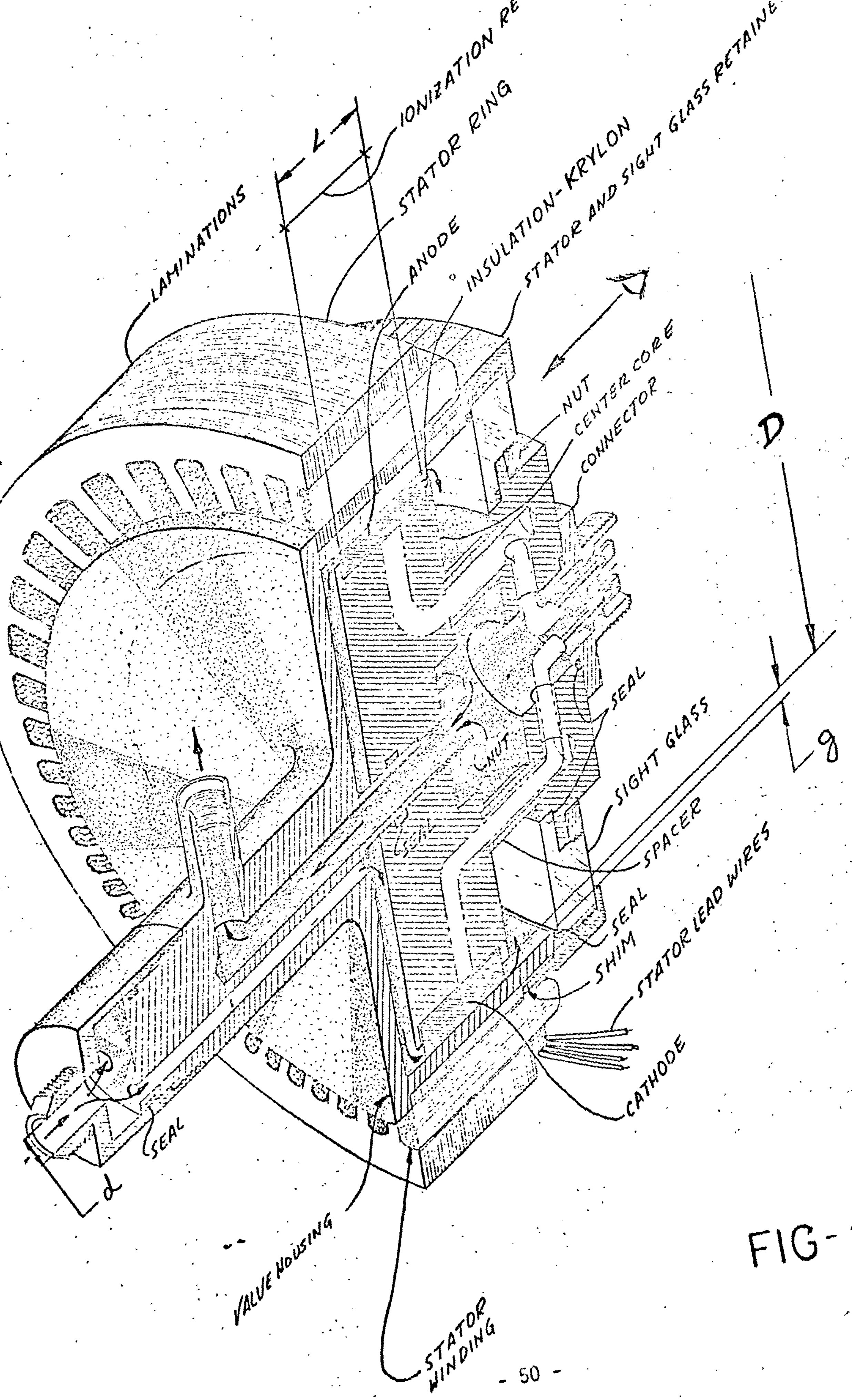
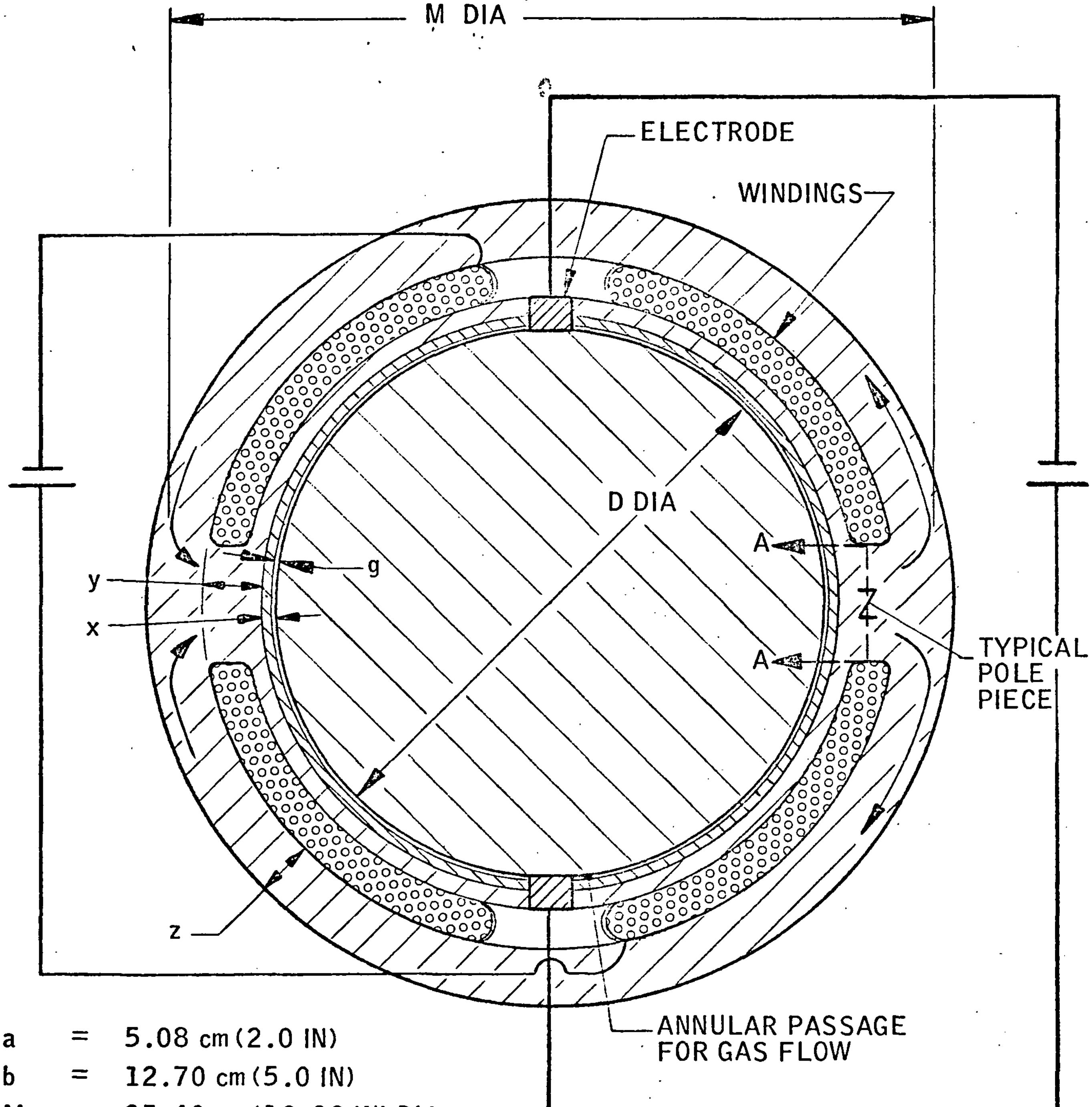
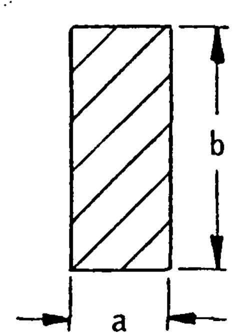

IONIZATION RE
STATOR RING
LAMINATIONS
L
ANODE
INSULATION-KRYLON
STATOR AND SIGHT GLASS RETAINE
NUT
CENTER CORE
CONNECTOR
D
SEAL
SIGHT GLASS
g
SPACER
SEAL
SHIM
STATOR LEAD WIRES
CATHODE
NUT
SEAL
SEAL
d
VALVE HOUSING
STATOR WINDING

FIG-24



a = 5.08 cm (2.0 IN)
b = 12.70 cm (5.0 IN)
M = 25.40 cm (10.00 IN) DIA
D = 16.80 cm (7.00 IN) DIA
x = 0.457 cm (0.18 IN)
y = 1.905 cm (0.75 IN)
z = 1.778 cm (0.7 IN)



SECTION A-A

FIG - 25

β = Flux Density = 125,000 lines/in.^{2} (this flux density was arbitrarily assigned for wrought iron from Figure 3 since it yields the highest flux density for a reasonable magnetic field intensity) <u>Item 1</u>

i = Current (ampere) flowing through the conducting gas <u>Item 2</u>

ℓ = Circumference of the conducting path of the gaseous GH_2 = in. and $\ell = \pi D$. <u>Item 3</u>

The force acting on the conductor of circumference (ℓ) may be expressed as

$$F = \beta \ell i \, (8.85 \times 10^{-8}) \qquad \text{Eqn 1}$$

where 8.85×10^{-8} is a conversion factor for the appropriate units.

Hence, the pressure (P) excerted by the field can be stated as

$$P = \frac{F}{\pi D g} \qquad \text{Eqn 2}$$

where

P = Pressure - lbf/in.^{2} <u>Item 4</u>

F = Force = lbf <u>Item 5</u>

D = Mean diameter of conducting path = in. (see Figures 1 and 4) <u>Item 6</u>

g = The radial clearance of the annular flow path between the center core and the outer housing = in. (see Figures 24 and 25) <u>Item 7</u>

By substituting F from Eqn 1 in Eqn2

$$P = \frac{\beta \ell i\ (8.85 \times 10^{-8})}{\pi D g} \qquad \underline{\text{Eqn 2A}}$$

but

$$\ell = \pi D \text{ (from Item 3)}$$

Therefore

$$P = \frac{\beta i \pi D\ (8.85 \times 10^{-8})}{\pi D g}$$

or

$$P = \frac{\beta i\ (8.85 \times 10^{-8})}{g} \qquad \underline{\text{Eqn 3}}$$

which is the expression for the retardation pressure. Based on β selected in Item 1 which gives an H value = 210 ampere-turns/in. for wrought iron with a β = 125,000 lines/in.^{2} (Figure 26), Eqn 3 may be stated as follows:

$$P = \frac{0.011\ i}{g} \qquad \underline{\text{Eqn 4}}$$

Figure 27 was then plotted with current (i) versus various selections of radial clearance (g). The pressure (P) was established as a constant = 30 lbf/in.^{2} which was assigned for the design shown on ANSC drawing 1140003, (Figure 23).

The radial clearance (g) was selected as 0.018 inches for further calculations. From Figure 2 for a design pressure (P) = 30.0 lbf/in.^{2}

g = 0.018 in. (0.0457 cm) (Figure 27) <u>Item 8</u>

i = 50.0 amperes (Figure 27) <u>Item 9</u>

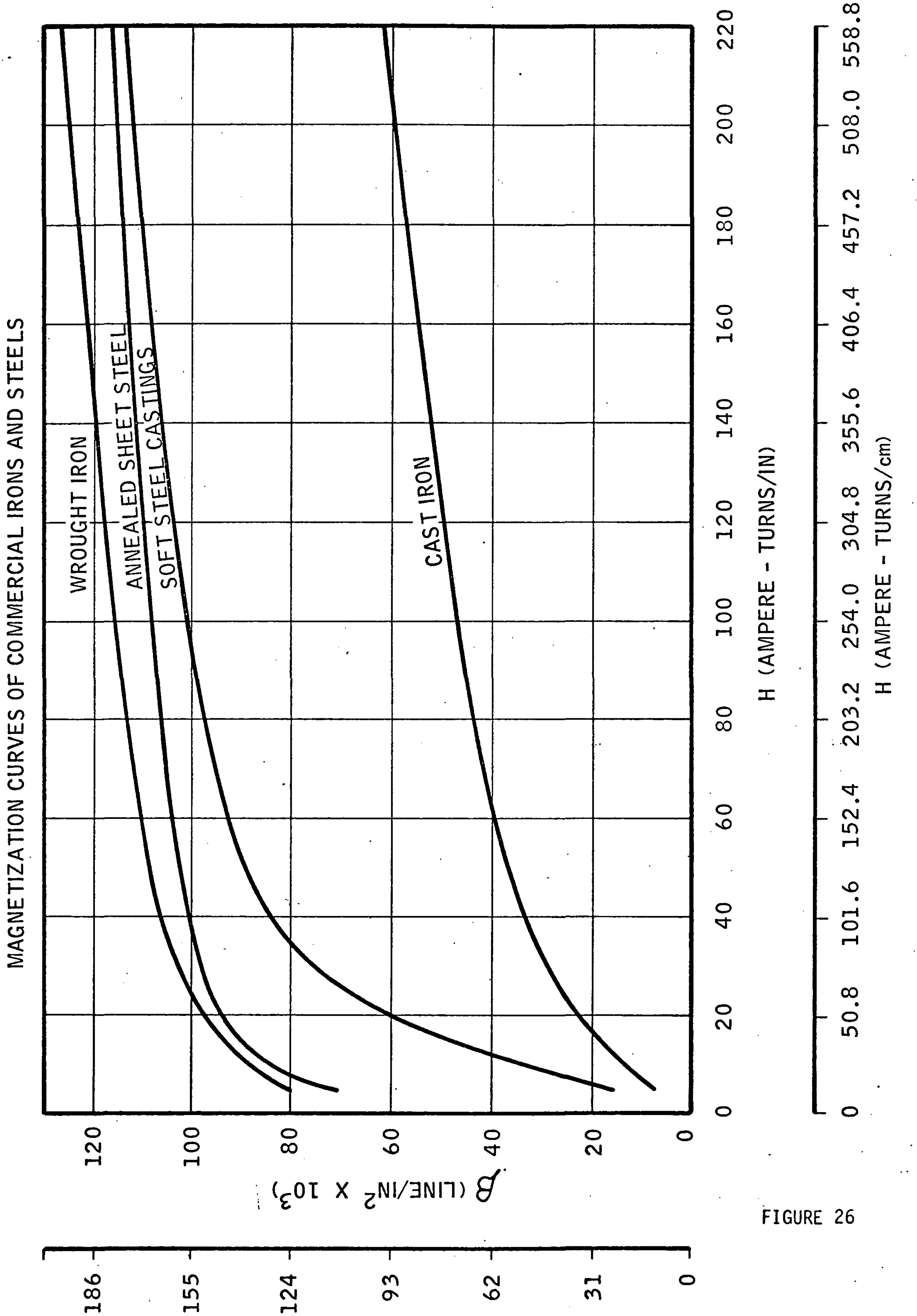

MAGNETIZATION CURVES OF COMMERCIAL IRONS AND STEELS
WROUGHT IRON
ANNEALED SHEET STEEL
SOFT STEEL CASTINGS
CAST IRON
β (WEBERS/m^2 X 10^{-4})
0
31
62
93
124
155
186
β (LINE/IN^2 X 10^3)
0
20
40
60
80
100
120
0
20
40
60
80
100
120
140
160
180
200
220
H (AMPERE - TURNS/IN)
0
50.8
101.6
152.4
203.2
254.0
304.8
355.6
406.4
457.2
508.0
558.8
H (AMPERE - TURNS/cm)

FIGURE 26

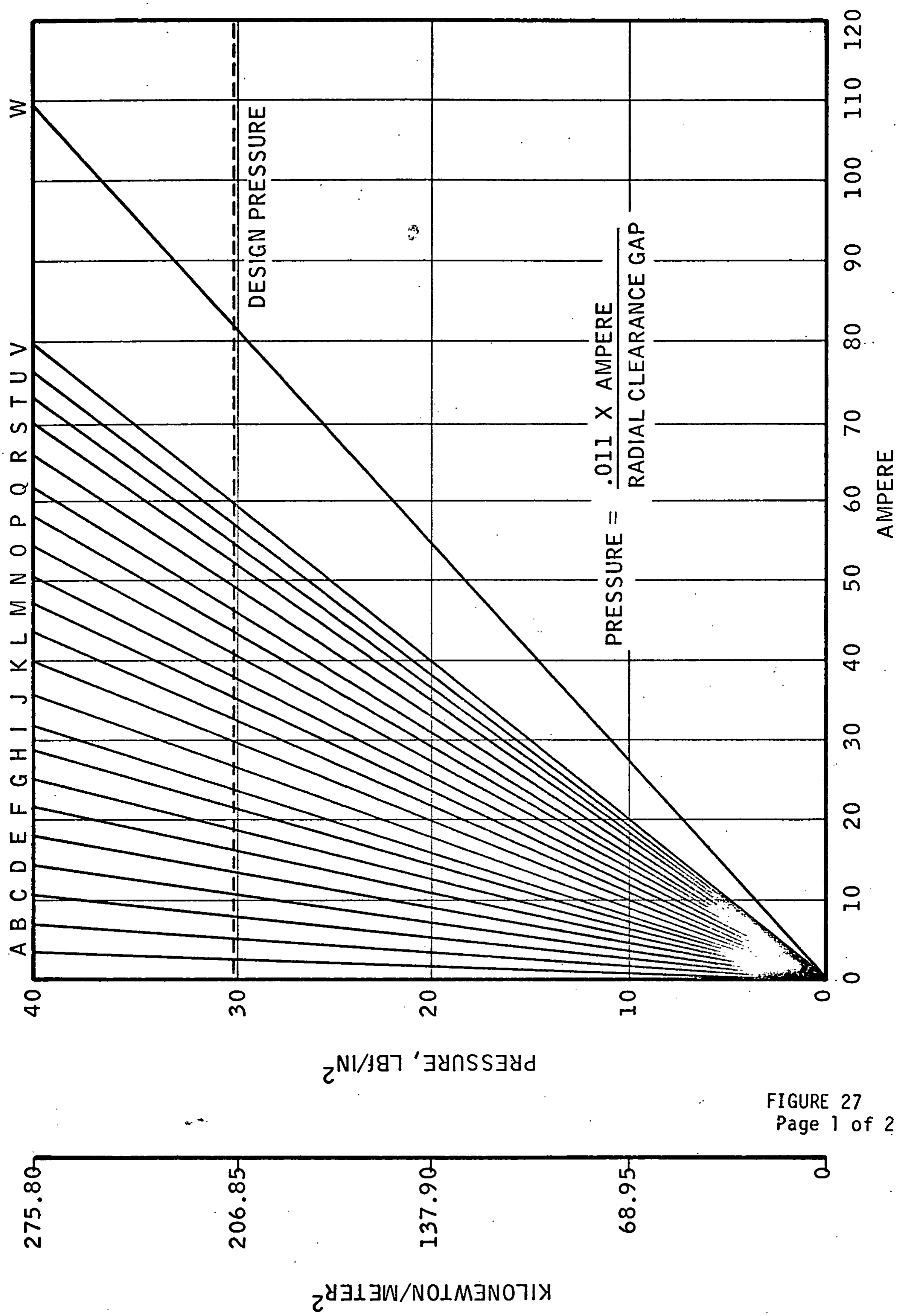


FIGURE 27
Page 1 of 2

g = RADIAL CLEARANCE GAP	INCHES	CENTIMETERS
A	0.001	0.00254
B	0.002	0.00508
C	0.003	0.00762
D	0.004	0.01016
E	0.005	0.01270
F	0.006	0.01524
G	0.007	0.01778
H	0.008	0.02032
I	0.009	0.02286
J	0.010	0.02540
K	0.011	0.02794
L	0.012	0.03048
M	0.013	0.03302
N	0.014	0.03556
O	0.015	0.03810
P	0.016	0.04064
Q	0.017	0.04318
R	0.018	0.04572
S	0.019	0.04826
T	0.020	0.05080
U	0.021	0.05334
V	0.022	0.05588
W	0.030	0.07620

FIGURE 27
Page 2 of 2

The cross sectional area of the radial clearance annular gap was then determined as

$$A = \pi D g$$

A = area = in.^{2}

D = 7.0 inches (Figures 24 and 25)

g = 0.018 inches (Item 8)

$A = 0.396$ in.^{2} <u>Item 10</u>

Therefore, the flow passage area approximates the area of a 1.905 cm (0.75 in.) O.D. line.

Using the reference shown in Footnote 1, the normal voltage drop for both the cathode and the anode can be stated as:

$$V_n = \frac{2\bar{\beta}}{\bar{A}} \ln \left(1 + \frac{1}{r}\right)$$ <u>Eqn 5</u>

where, for hydrogen

V_n = Normal cathode plus anode voltage drop-volts D.C.

$\bar{\beta}$ = 130.0 volts/cm/mm Hg = the Stoleton constant from Footnote 1 for hydrogen gas <u>Item 11</u>

$\bar{A}$ = 5.0 volts/cm/mm Hg (based on the Townsend ratio of the ionization coefficient (E) to the pressure (P) from Footnote 1 <u>Item 12</u>

r = 0.05 = coefficient of electron emission by positive ion bombardment for hydrogen gas and was selected for a copper cathode in hydrogen from Footnote 1 <u>Item 13</u>

therefore

V_n = 475 volts D.C. <u>Item 14</u>

Now based on the voltage and the current, the power required to cause a glow discharge within the ionization region of the radial clearance gap can be expressed as

$W = i\ (V_n + V)$ <u>Eqn 6</u>

W = power - watt

i = 50.0 ampere (Item 9)

V_n = 475 V D.C. (Item 11)

V = 43,300 V D.C. (Appendix N) = the voltage drop across the conducting H_2 gas at room temperature with no temperature rise in the gas and is a worse case condition. As pointed out in Appendix N, this voltage decreases with a gas temperature increase, and decreases considerably with other media such as He and liquid metals. <u>Item 14A</u>

$V_T = V_n + V$ = 43,775 Volts D.C.

W = 2,188,750 Watt. <u>Item 14B</u>

Footnote 1: "Gaseous Conductors Theory and Engineering Applications", by James Dillon Cobine.

Now the length of the radial clearance gap must be calculated by the following formula

$$L = \frac{i}{\theta_n \; g} \qquad \underline{\text{Eqn 7}}$$

L = Length of radial gap - cm (see Figure 24).

However, both (i) and (g) have been previously determined and it is necessary to find θ_n using the following equation from Footnote 1 in terms of system pressure.

$$\theta_n = \frac{5.92 \times 10^{-14} \bar{A} \bar{B}^2 (K_p P) (1 + r) P^2}{\ell n [1 + \frac{1}{r}]} \qquad \underline{\text{Eqn 8}}$$

$\bar{A}$ = 5.0 volts/cm/mm Hg (see Item 12)

$\bar{B}$ = 130.0 volts/cm/mm Hg (see Item 11)

r = 0.05 (see Item 13)

K_p = 6.38 - 4.89 cm^2/volt = mobility.

p = 30.0 lbf/in.^{2} = 1551.3 mm Hg using Items 8 and 9
from Figure 2 and the design pressure of the valve

therefore

θ_n = 32.17 ampere/cm^2 <u>Item 15</u>

and L in Eqn 7 can then be calculated using Items 8, 9, and 15

L = 3.4 cm (1.34 in.) <u>Item 16</u>

A summary of the major parameters established thus far are as follows (see Figures 24 and 25).

$P = 2.07 \times 10^5$ Newton/meter^2 (30.0 lbf/in.^{2})

$g = 0.0457$ cm (0.018 in.)

$i = 50.0$ ampere

$\beta = 1.94$ Weber/meter^2 (125,000 lines/in.^{2})

$L = 3.4$ cm (1.34 in.)

$D = 17.78$ cm (7.00 in.)

$V_T = 43{,}775$ volts D.C.

$T = 294.4°K$ (530°R)

$W = 2{,}188{,}750$ watt

$A = 1.000$ cm^2 (0.396 in.^{2})

Line Size = 1.90 cm (0.75 in.) O.D.

$H = 82.24$ ampere turns/cm (210.0 ampere turns/in.)

The next step is to size the stator: First, the ampere-turns required to establish the desired flux density across the radial clearance must be determined. Two wrought iron pole pieces are used:

β = 125,000 lines/in.^{2} Item 1)

H = 210.0 ampere-turns/in. (using β and Figure 26) Item 17

Then the number of lines (ϕ) is given by

$\phi = \beta$ a b Eqn 9

β = 125,000 lines/in.^{2} (from Item 1)

a = 2.0 in. (determined from ANSC drawing 1140003 and Figure 25) Item 18

b = 5.0 in. (determined from ANSC drawing 1140003 and Figure 25) Item 19

ϕ = 1,250,000 lines Item 20

and since the same number of lines (ϕ) must also pass through the armature:

NI = 2.0 H Y Eqn 10

2.0 = number of pole pieces (Figure 25) Item 21

H = 210.0 ampere-turns/in. (Item 17)

Y = 0.75 in. (arbitrarily determined from ANSC drawing 1140003 and Figure 25) Item 22

NI = 314.0 ampere-turns Item 23

and

$$\bar{\bar{\beta}} = \frac{\phi}{2.0\ D}$$ Eqn 11

$\bar{\bar{\beta}}$ = Flux density of each pole piece

ϕ = 1,250,000 lines (Item 20)

D = 7.0 inches (ANSC drawing 1140003 and Figures 24 and 25) Item 24

2.0 in. (from Item 21)

$\bar{\bar{\beta}}$ = 89,000 lines/in.^{2} Item 25

and

$\overline{NI} = \overline{H}\ D$

$\overline{H}$ = 11.0 ampere-turns/in. (using $\bar{\beta}$ and Figure 26 for wrought iron) Item 26

$\overline{D}$ = 7.0 inches (Item 24)

$\overline{NI}$ = 77.0 ampere-turns Item 27

Now for the radial clearance gap (g)

$$\overset{*}{\bar{\bar{\beta}}} = \frac{\phi}{\pi\ D\ a}$$ Eqn 12

ϕ = 1,250,000 lines (Item 20)

D = 7.0 inches (Item 24)

a = 2.0 inches (Item 18)

$\overset{*}{\bar{\bar{\beta}}}$ = 28,500 lines/in.^{2} Item 28

now

$$\overline{\overline{NI}} = \frac{2.0\ \overset{*}{\bar{\bar{\beta}}}\ \bar{\ell}}{1.26}$$ Eqn 13

2.0 in. (from Item 21)

$\overset{*}{\bar{\bar{\beta}}}$ = 28,500 lines/in.^{2} (from Item 28)

$\bar{\ell}$ = 0.394 (reluctance of a in.^{3} of air for both gaps)

1.26 = conversion factor for the appropriate units

$\overline{\overline{NI}}$ = 17,800 ampere turns (for both pole pieces) Item 29

The total ampere-turns required for the stator windings for both pole pieces is shown as follows:

$$NI_T = \overline{\overline{NI}} + \overline{Ni} + NI$$

$\overline{\overline{NI}}$ = 17,800 ampere-turns (Item 29)

$\overline{NI}$ = 77.0 ampere-turns (Item 23)

NI = 314.0 ampere-turns (Item 23

NI_T = 18,191.0 ampere-turns <u>Item 30</u>

or

NI_T = 9095.5 ampere-turns for each pole piece <u>Item 31</u>

Brown and Sharp copper wire gage No. 20 (0.031 inch diameter) wire was arbitrarily selected in order to generate a feel for the size of the winding and the heat losses in the windings. Future designers experimentating with the MHD valve concept may select any desirable wire size.

$$P = 2.0\ (a + b)$$ <u>Eqn 14</u>

P = winding circumference for each coil = in./turn (see Figure 4)

a = 2.0 in. (see Item 18)

b = 5.0 in. (see Item 19)

P = 14.0 in./turn <u>Item 32</u>

and

$$N = \frac{NI_T}{i}$$ <u>Eqn 15</u>

N = Number of turns/pole piece

NI_T = 9095.5 ampere turns per pole piece (see Item 31)

i = 10.0 ampere (arbitrarily used as a trial to minimize heat losses) <u>Item 33</u>

$N = 909.55$ turns Item 34

then

$\bar{L} = PN$ Eqn 16

$\bar{L}$ = Total length of wire - in.

$N = 909.5$ turns (see Item 34)

$P = 14.0$ in./turn (see Item 32)

$\bar{L} = 12{,}734.0$ inches Item 35

and

$R = \frac{10\ \Omega\ \bar{L}}{12{,}000}$ Eqn 17

R = Resistance of the wire = OHM

$\bar{L} = 12{,}734.0$ inch (see Item 35)

$\frac{10\ \Omega}{12{,}000}$ = Resistance (OHMS)/12,000 in. of 0.031 in. diameter copper wire

$R = 10.61$ OHM Item 36

now

$V = i\ R$

V = Applied voltage = volts

$i = 10.0$ ampere (see Item 35)

$R = 10.61$ OHM (see Item 36)

$V = 106.1$ volts Item 37

and

$\alpha = \frac{\pi}{4}(d)^2$

α = Cross sectional area of the wire = in.^{2}/turn

$d = 0.031$ in. diameter

$\alpha = 0.00075$ in.^{2}/turn Item 38

now

$$\bar{\alpha} = \alpha N$$ Eqn 18

$\bar{\alpha}$ = Space area for wire = in.^{2}

α = 0.00075 in.^{2}/turn (see Item 38)

N = 909.55 turns (see Item 34)

$$\bar{\alpha} = 0.6864 \text{ in.}^2$$ Item 39

and using a space factor = 3.0 for the windings

$$\bar{\alpha}_T = 2.06 \text{ in.}^2$$ Item 40

now for the heat loss for each coil

$$W = i^2R$$ Eqn 19

W = Heat loss = watt

i = 10.0 ampere (see Item 35)

R = 10.61 OHM (see Item 36)

$$W = 1061.0 \text{ watt}$$ Item 41

These calculations were made without any wire insulation considerations. The heat losses through the outer frame were not determined since they were considered negligible as far as the total number of ampere turns for the entire circuit. The selection of a larger wire size would also reduce the heat losses in the circuit.

Appendix N is a theoretical "evaluation of a MHD" concept including suggestions for a modified design approach.

b. Electromechanically Actuated In-Line Valve with High Pressure Heat Energized Poppet Seal

General

The in-line balanced poppet valve, shown on ANSC drawing 1140002, Figure 28, features a high pressure energized poppet seal. LH_2 in the annular cavity is heated by a cal-rod type heater (submerged in the cavity) when the poppet is closed. The heat expands the LH_2 into GH_2 to produce high seat loads on the poppet.

The concept also shows an in-line shutoff valve design capable of being remotely coupled and uncoupled through the use of remote coupling designs shown on ANSC drawings 1140005 and 1140006, (Figures 39 and 40).

ANSC drawing 1140002 (Figure 28) consists of two sheets. Sheet 1 contains performance notes for the valve, a main view of the valve with the thermal interference remote coupling ports compatible with 1140006-3. This sheet also shows estimated area versus stroke, K (flow resistance coefficient) versus stroke and torque versus stroke curves. Sheet 2 depicts the cross section of the valve with materials and with low melting allow remote coupling ports compatible with 1140005-3, (Figure 39).

Discussion

The objective of this concept was to produce a unidirectional flow shutoff valve design with an extremely low liquid hydrogen leakage capability. The shutoff valve shown utilizes an unbalanced poppet. The valve poppet is actuated by a D.C. torque motor to drive a ball screw system which in turn either opens or closes the valve. This method of actuation was selected because of the simplicity of a direct drive between the torque motor and the ball screw. The design also incorporates an anti-backdriveable, spragging type, locking device to mechanically prevent the poppet from moving away from the seat in the closed position, since in the closed position extremely high seat loads are generated by the expansion of LH_2 to GH_2 in the annular cavity.

The shutoff seal configuration consists of the main seal assembly with a bellows, a ball check valve, and a cal-rod type heating element.

With the poppet in the closed position and filled with liquid hydrogen, the seal operates as follows: The hydrogen flows into the total pressure pickup probe, past the ball check valve, and into the annular cavity for

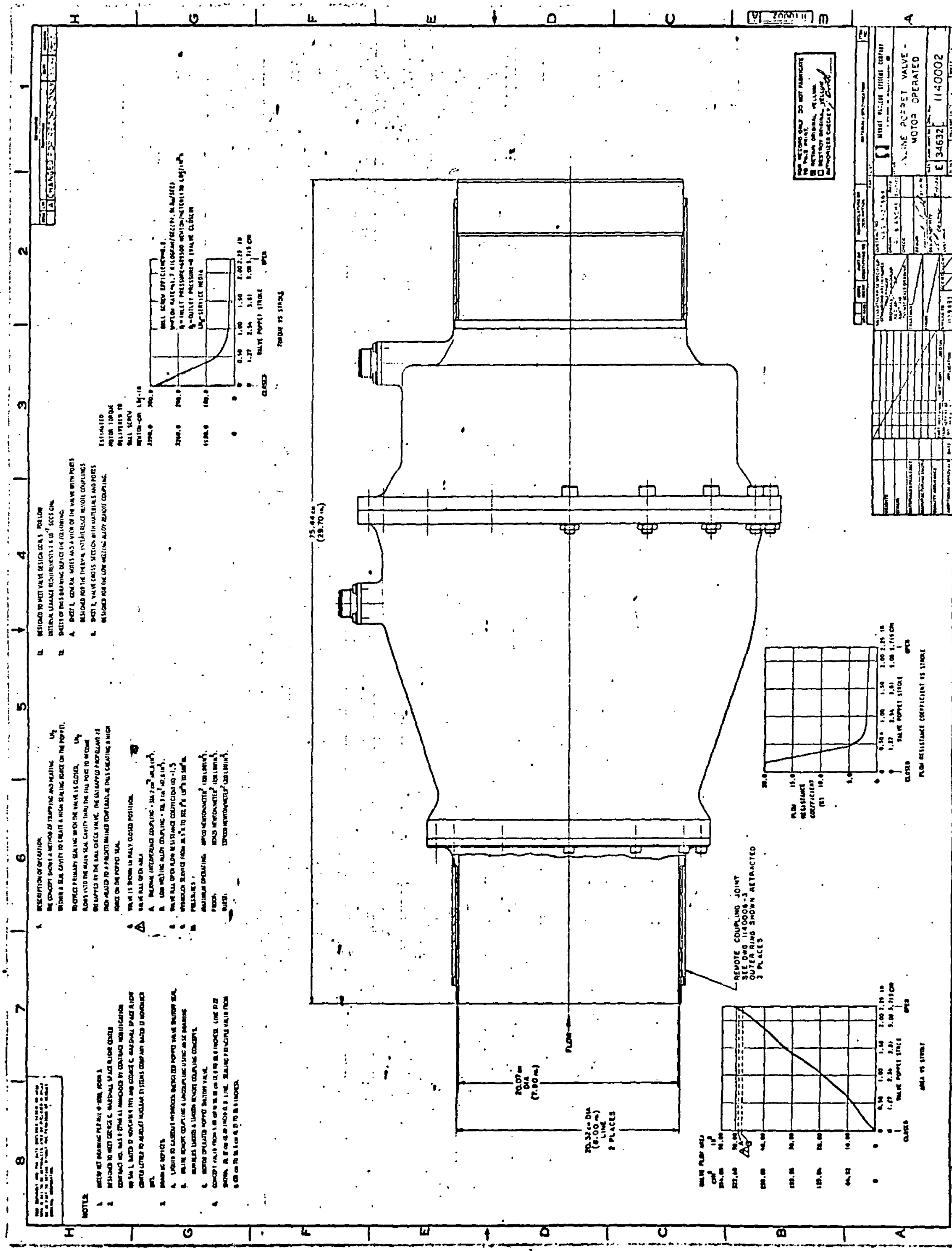


FIGURE 28
Page 1 of 2

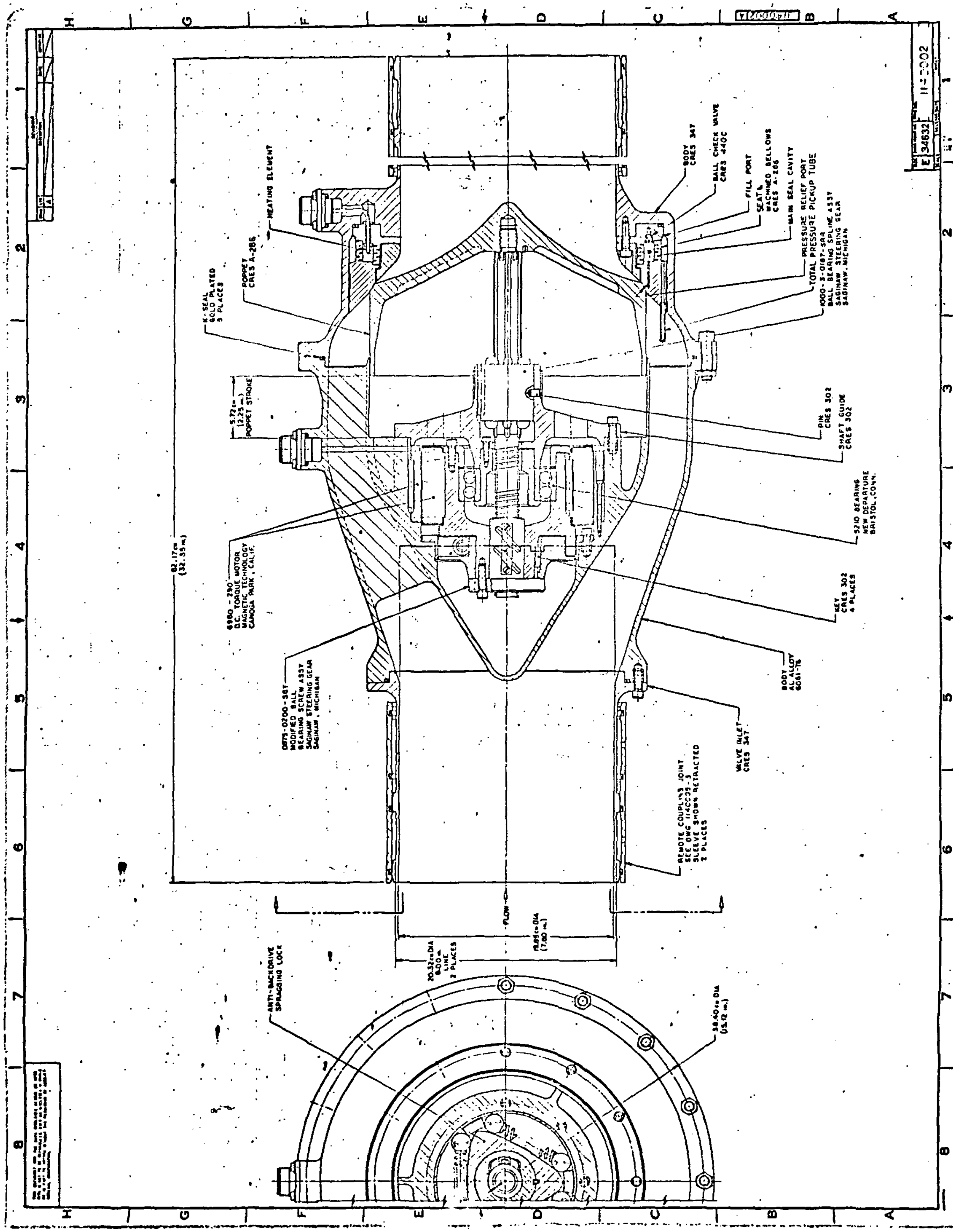


FIGURE 28
Page 2 of 2

filling. Power is then applied to the heater to raise the temperature of the hydrogen and expand it from a liquid to a gas. The pressure developed in the annular cavity by the heated and entrapped gas can be extremely high and will force the seat in the mechanically locked poppet by flexing the bellows. The pressure developed in the annular cavity is clearly dependent on the heat generated by the heater and can be varied for different leakage requirements.

The high pressure developed in the annular cavity acts on the cross sectional area of the annulus to create a high seat force against the mating poppet conical area. Thus, a high unit compressive load (pressure) is created between the two conical seating surfaces.

During the opening of the poppet the action of the torque motor will release the spragging locking device and move the poppet to the open position. The high pressure entrapped in the annular cavity is then vented through the pressure relief orifice, located in the seat, and allows the seat to return to its open position stop.

The sealing capability of this concept should be extremely effective because of the high unit loading that can be developed between the mating conical sealing surfaces. Surface finish and conical accuracy of the two sealing surfaces will be important factors in the Phase III design.

Although the design is rated from 20.5°K to 322.2°K (37°R to 580°R), it is essentially a liquid hydrogen valve. The power required to expand LH_2 into GH_2 (temperature rise) is small for high pressures in the annular cavity compared to the power requirements and very large temperature rises necessary to expand GH_2 to generate the large pressures for very low leakage sealing.

Following is an analysis that will aid the designer in determining seat poppet configurations, in conjunction with various cross sectional areas of the annular cavity.

SCHEMATIC OF POPPET AND SEAT ARRANGEMENT
SHOWN ON ANSC DWG. 1140002 (FIGURE 28)

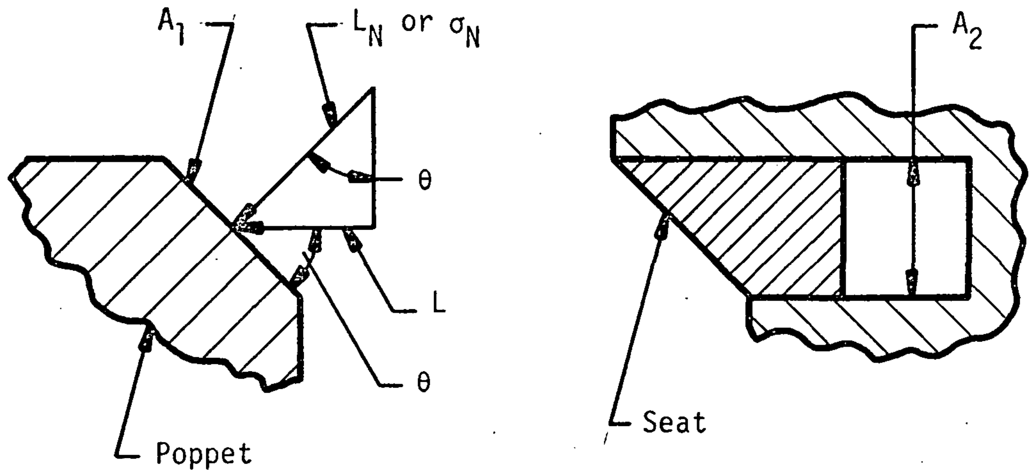


FIGURE 29

$L_N = \sigma_N A_1$ <u>Eqn 1</u>

L_N = Normal load on conical poppet area (see Figure 29) <u>Item 1</u>

$\sigma_N = F_{cy}$ (compressive yield stress of conical poppet and seat materials) or any desired contact stress level required by the designer (see Figure 29) <u>Item 2</u>

A_1 = Conical poppet contact area that will be in intimate contact with the seat when the poppet is closed (see Figure 29) <u>Item 3</u>

Now

$L = L_N \sin\theta$ <u>Eqn 2</u>

L_N (see Item 1)

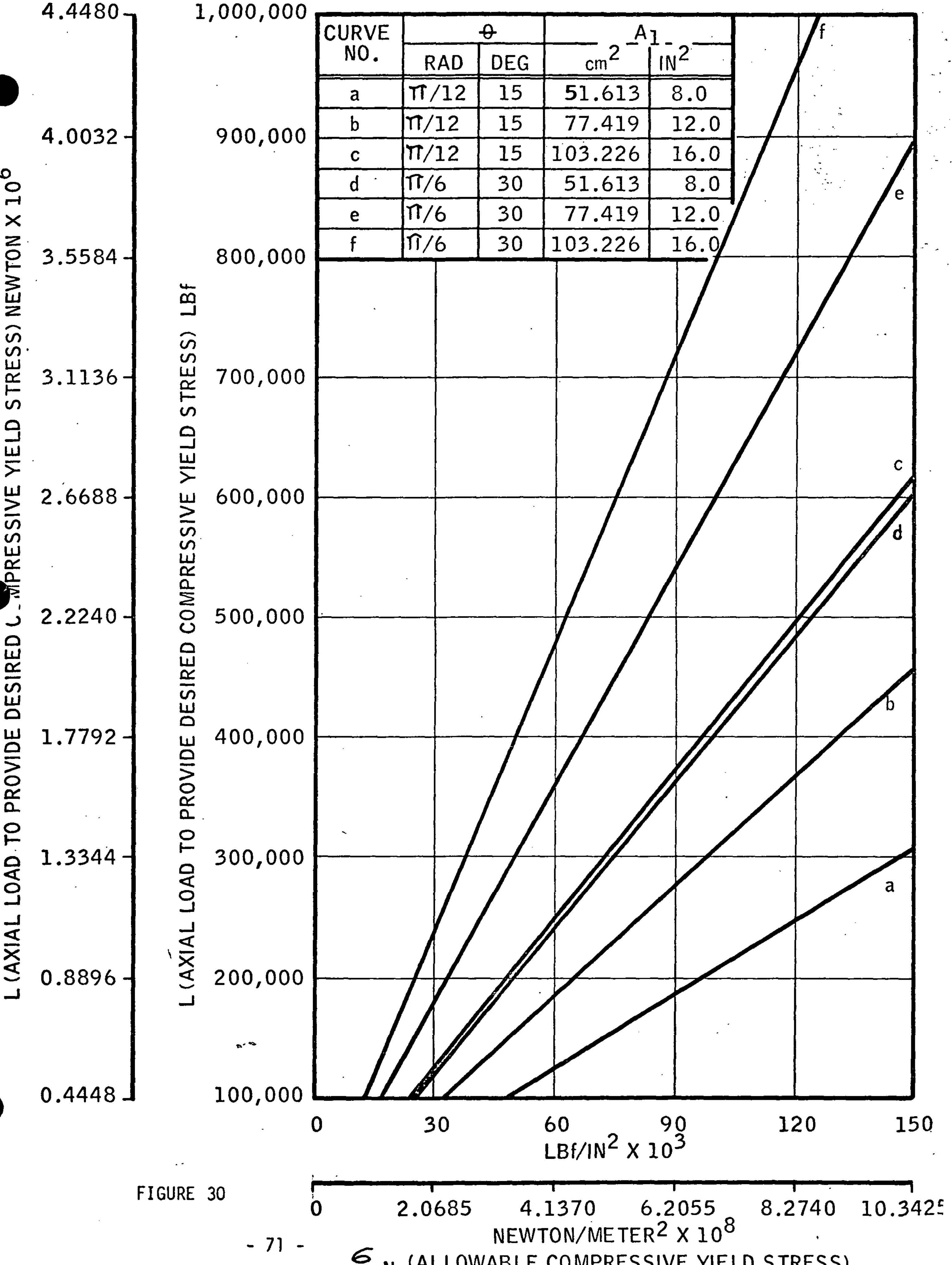
$\sin\theta$ = The poppet and seat conical half angle (see Figure 29) <u>Item 4</u>

L = The axial component of L_N (see Figure 29) <u>Item 5</u>

Then

$L = \sigma_N A_1 \sin\theta$ <u>Eqn 3</u>

Figure 30 is a graph of Eqn 3 for σ_N versus L for various A_1 and θ values. Curve e closely approximates the poppet and seat configuration shown on ANSC Drawing 1140002, (Figure 28).



CURVE NO.	θ		A_1	
	RAD	DEG	cm^2	IN^2
a	$\pi/12$	15	51.613	8.0
b	$\pi/12$	15	77.419	12.0
c	$\pi/12$	15	103.226	16.0
d	$\pi/6$	30	51.613	8.0
e	$\pi/6$	30	77.419	12.0
f	$\pi/6$	30	103.226	16.0

FIGURE 30

The dotted lines shown on Figure 30 reflect a $\sigma_{N_6} = 8.067 \times 10^8$ Newton/meter^2 (117,000 lbf/in.^{2}) and gives an L = 3.1136×10^6 Newton (700,000 lbf). This value was selected for the F_{cy} of the A-286 material of the poppet and seat shown on ANSC drawing 114002, (Figure 28). The F_{cy} was selected from the ANSC "Materials Property Data Book", Volume 2, Page 431, using room temperature properties.

Now

$$P = \frac{L}{A_2} \qquad \underline{\text{Eqn 4}}$$

L (see Item 5)

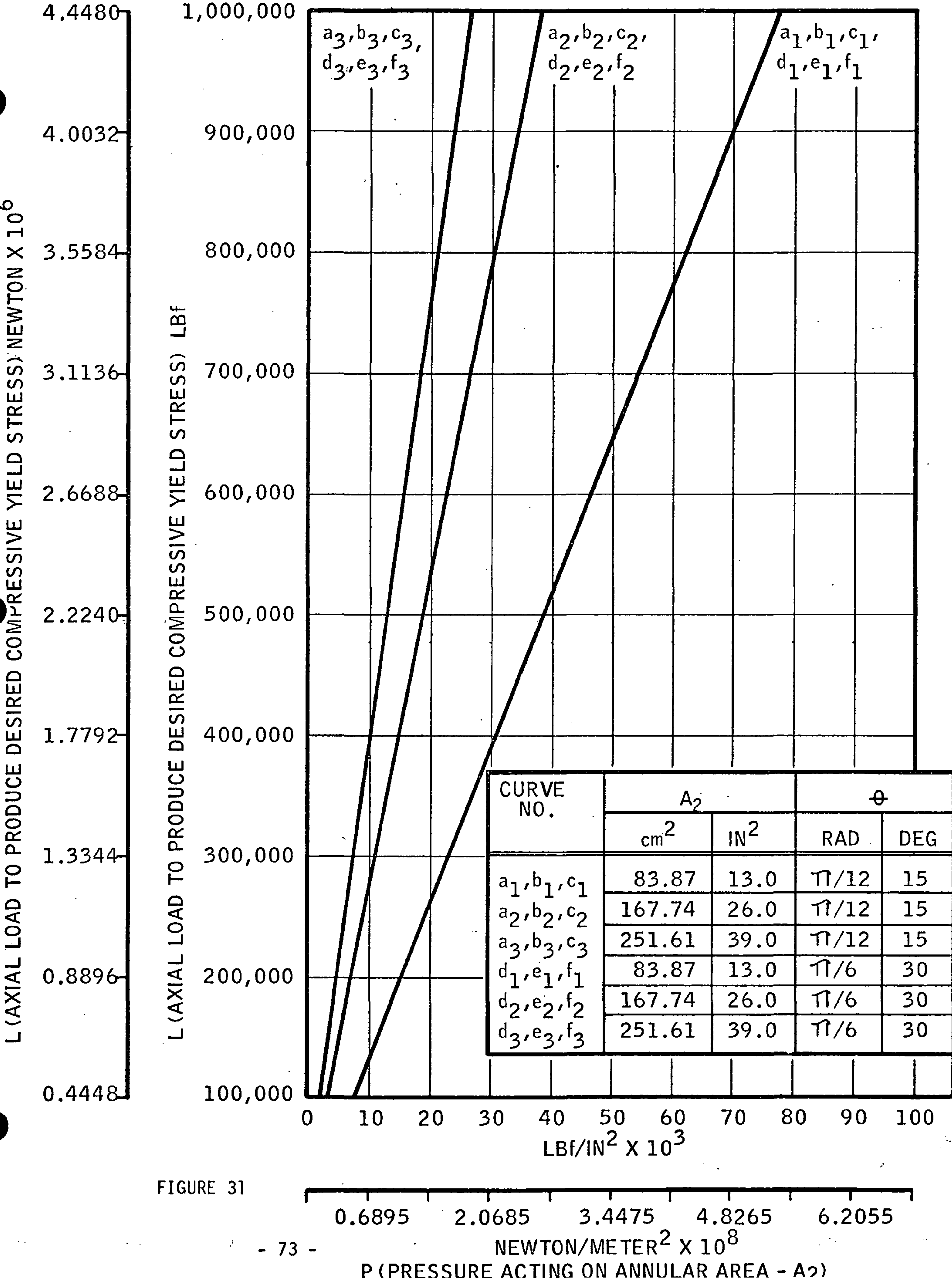
A_2 = Annular LH_2 expansion area to produce the load (L) on the poppet and seat (see Figure 29) — Item 6

P = The pressure acting on A_2 to produce the load (L). — Item 7

Figure 31 is a graph of Eqn 4 for P versus L for various values of A_2 and Curve d_1, e_1, and f_1 closely approximates the A_2 shown on ANSC drawing 1140002. The design as shown, could be expected to produce a poppet and seat load L = 3.1136×10^6 Newton (700,000 lbf) with LH_2 expanded to P = 3.7233×10^8 Newton/meter ^{2} (54,000 lbf/in.^{2}).

Using Figures 29 and 30 for the size valve shown on ANSC drawing 1140002, (Figure 28) the designer can select any values of P, L, A_2, σ_N, Θ or A_1 that he chooses. For other size valves the designer can use Equations 1, 2, 3 and 4 to generate his own Figures 29 and 30.

P is a function of the initial LH_2 pressure in the A_2 annular cavity, the cavity volume and the temperature rise in the cavity due to the heat produced by the cal-rod type heater. Power requirements for heating the LH_2 (GH_2) to various P values are shown in Appendix O.



CURVE NO.	A_2		θ	
	cm^2	IN^2	RAD	DEG
a_1, b_1, c_1	83.87	13.0	$\pi/12$	15
a_2, b_2, c_2	167.74	26.0	$\pi/12$	15
a_3, b_3, c_3	251.61	39.0	$\pi/12$	15
d_1, e_1, f_1	83.87	13.0	$\pi/6$	30
d_2, e_2, f_2	167.74	26.0	$\pi/6$	30
d_3, e_3, f_3	251.61	39.0	$\pi/6$	30

FIGURE 31

c. Electromechanically Actuated In-Line Poppet Valve with Liquid Metal Seals

General

The in-line balanced poppet valve shown on ANSC drawing 1140001, Figure 32, features an electromechanically actuated valve with two types of liquid metal filled seals. In addition, the concept shows an in-line valve design capable of being remotely coupled and uncoupled through the use of remote coupling designs as shown on ANSC drawings 1140005 and 1140006, (Figures 39 & 40).

ANSC drawing 1140001 (Figure 32) consists of two sheets. Sheet 1 contains performance notes for the valve, a main view of the valve with the thermal interference remote coupling ports compatible with 1140006-2. This sheet also shows estimated area versus stroke, K (flow resistance coefficient versus stroke, and torque versus stroke curves. Sheet 2 depicts the cross section of the valve with materials and with the low melting alloy remote couplings compatible with 1140005-2.

Discussion

The objective of this concept was to produce a uni-directional flow shutoff valve design with a low leakage capability that could be used with GH_2, GN_2 or He in temperature ranges from 216.7°K to 366.7°K (390°R to 660°R).

The concept shown is a balanced poppet valve featuring two types of main poppet seals. For the main poppet seal, liquid metal is the primary seal. The secondary main poppet seal consists of an optically flat poppet in intimate contact with an optically flat seat.

The poppet skirt shaft seal is a liquid metal filled double metal lip seal cavity arrangement. The valve poppet is actuated by a D.C. torque motor to drive a ball screw system which in turn either opens or closes the valve. This method of actuation was selected because of the simplicity of a direct drive between the torque motor and the ball screw.

The seal main shut off consists of a reservoir containing a liquid metal volume that is enclosed on one side by a bellows and the other side by an optically flat seat plate. The seat plate is forced against the seal stop by the pre-load of the bellows with the poppet in the open position, in order to retain the liquid metal.

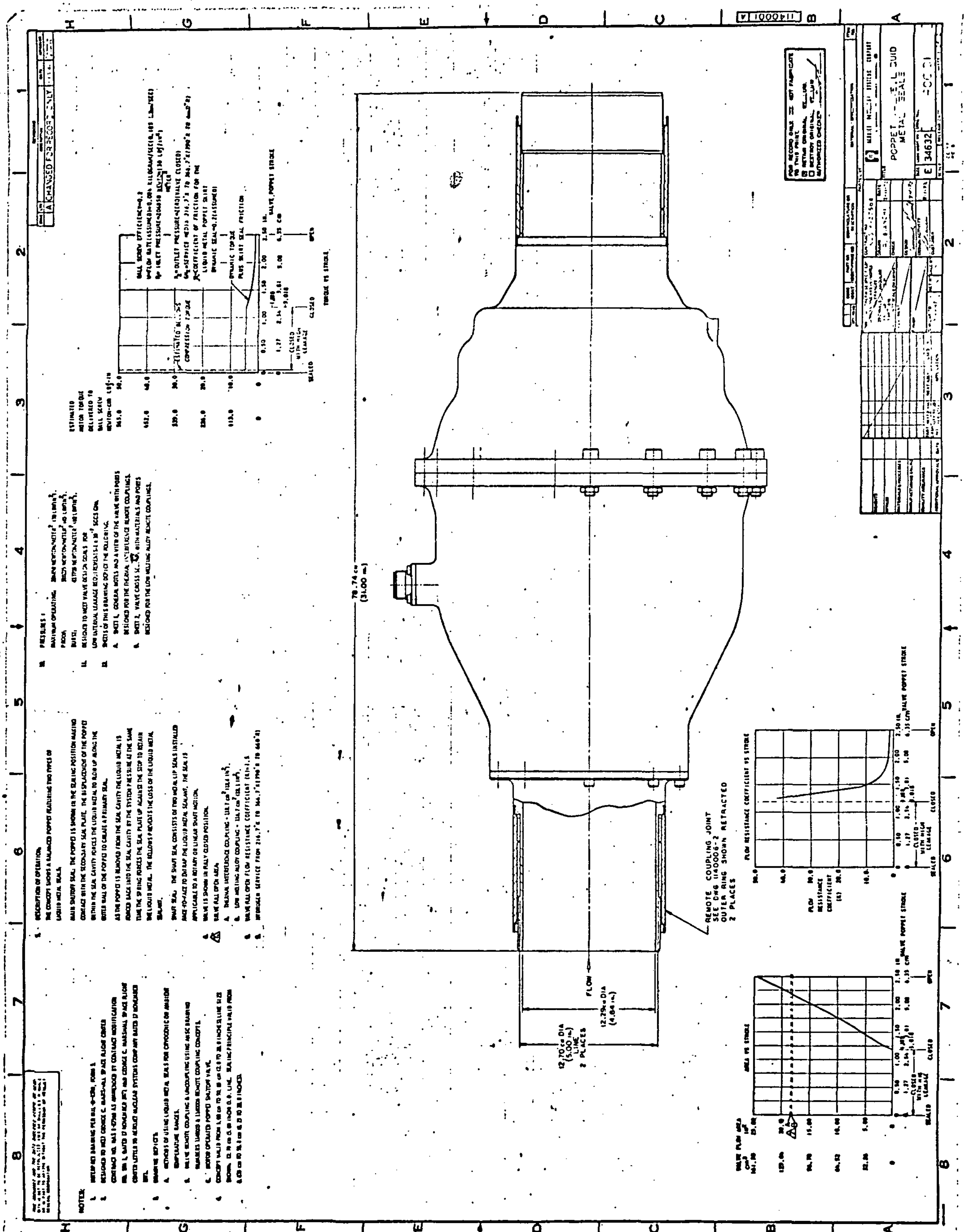


FIGURE 32
Page 1 of 2

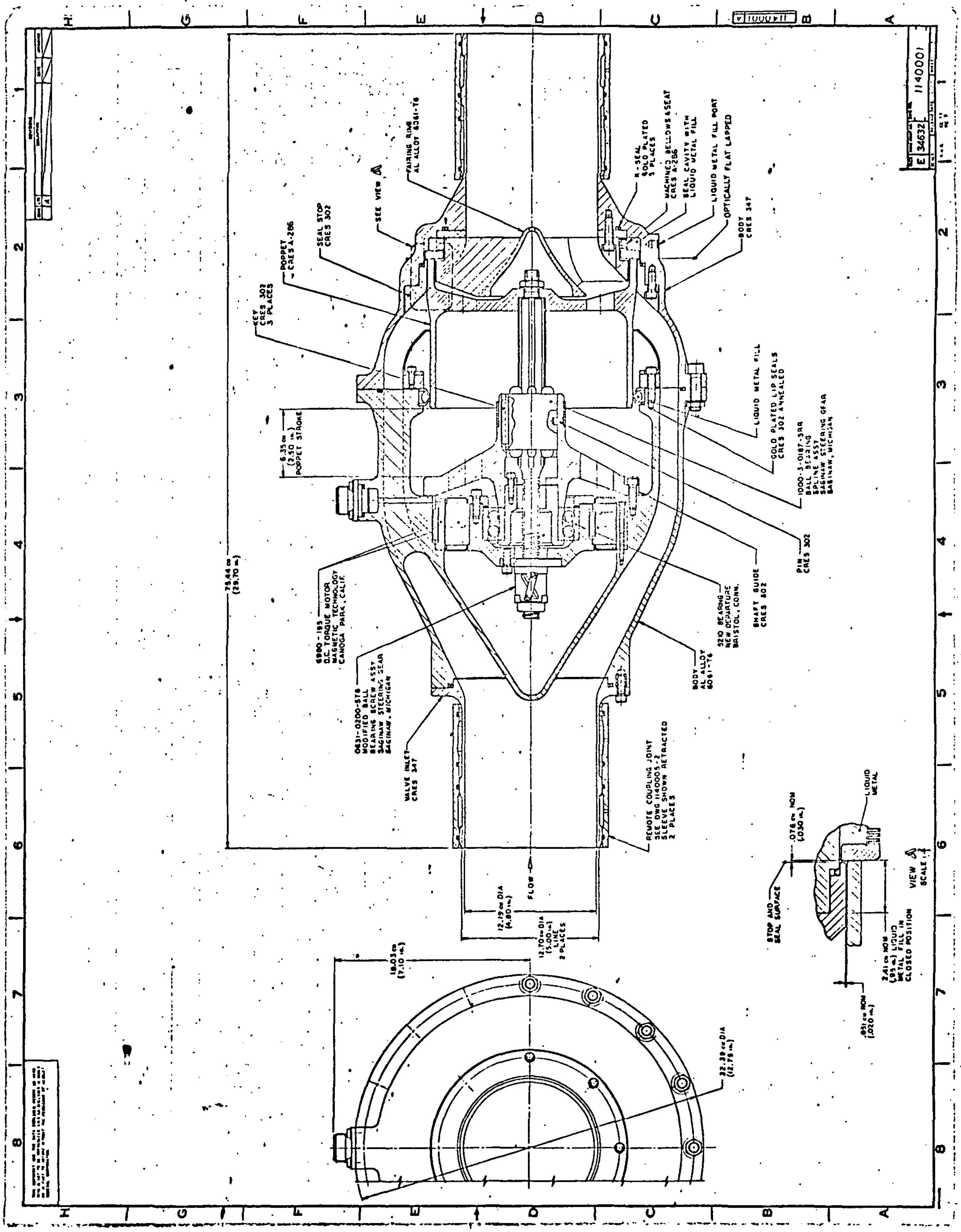


FIGURE 32
Page 2 of 2

To energize the seal, the poppet must actuate the seat plate to a stroke of .076 cm (.030 in.). This action displaces a predetermined volume of the liquid metal, to a calculated height of 2.41 cm (0.95 in.) between the walls of the poppet and the seat plate. View A of Figure 32 depicts the energized state of the seal.

During valve opening, as the poppet is moved away from the seat plate, the liquid metal sealant is forced back into the seal cavity by the pressure differential across the liquid column. Simultaneously, the preload of the bellows forces the seat plate against the stop to again retain the liquid metal sealant. Sealing is accomplished by the column of liquid metal entrapped between the walls of the poppet and seat plate. Since the sealing capability is dependent on the liquid metal volume, any major loss of the metal would destroy the effectiveness of the seal.

In order to prevent leakage of the liquid metal sealant between the poppet face and the seat plate, the sealing surfaces must be optically flat lapped to a surface finish of approximately 5.08 μ cm (2.0 μ in.). Optical flatness will be on the order of 1/8 of a helium light band.

The poppet skirt dynamic seal consists of metal lip containment seals installed back to back with a liquid metal sealant entrapped between them. The sealant is contained on the dynamic surface by the poppet skirt. The surface of the poppet skirt that is in contact with the liquid metal seal is closely mated or fitted to the liquid metal containment lip seals.

Reference 14, Volume 1,of Appendix D offers an excellent study and references of liquid metals and the wet seal concept.

The following data, Figure 33, and information was used from the reference noted.

"When two highly polished surfaces are placed in contact, or in near contact, a small volume of porous structure can be visualized as existing between the two contacting surfaces. Numerous leakage paths result from the microscopic nonuniformity between the two surfaces.

Utilizing this concept, sealing by a liquid metal can then be visualized as the closing of the capillary leakage paths by the liquid metal. The force necessary to overcome the retention of the liquid or break down the seal is directly related to the capillary forces.

The capillary force between a liquid-solid interface depends upon the manner in which they meet. Between a liquid and a solid, the contact angle is the angle between the solid and the tangent to the liquid at the point of contact. The surface tension of the solid and liquid phases and interfacial tension between these phases determined the magnitude of the contact angle and thus the capillary force. In a confined volume the contact angle results in the formation of a meniscus. The contact angle is directly related to wettability of the liquid which is described at equilibrium and neglecting gravitational effects as:

$$\lambda_{SV} = \lambda_{LS} + \lambda_{LV} \cos \theta$$

where: λ_{SV} = surface tension solid: vapor

λ_{LS} = interfacial tension liquid: solid

λ_{LV} = surface tension liquid: vapor

For complete wettability $\theta = 0°$ so that:

$$\lambda_{SV} = \lambda_{LS} + \lambda_{LV}$$

The general equation relating the pressure difference across a curved liquid surface is:

$$\Delta P = -\lambda \left(\frac{1}{R_1} + \frac{1}{R_2}\right)$$

where is the interfacial tension of the liquid surface film and R_1 and R_2 are the radius of curvature of a curved liquid surface.

For closely space parallel plates

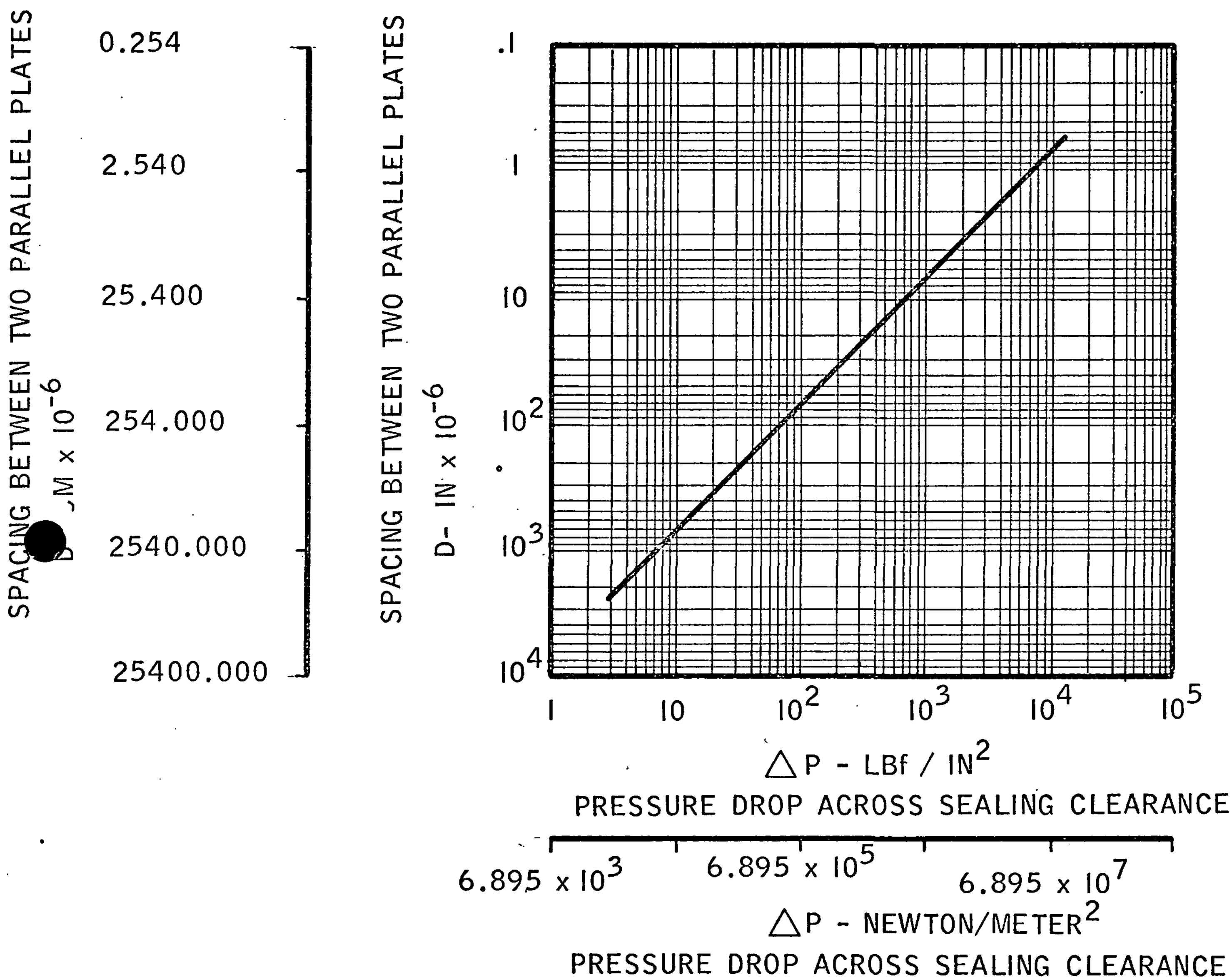
$R_2 = \infty$ and $R_1 = D/2$; the above equations reduces to:

$$\Delta P = -\lambda 2/D$$

where perfect wetting is assumed ($\cos \theta = 1$).

This equation is plotted in Figure 33.

CAPILLARY PRESSURE ΔP VERSUS SPACING D BETWEEN PARALLEL PLATES, BASED ON SURFACE SURFACE TENSION OF λ = 0.005 NEWTON/CM (500 DYNE/CM) $\Delta P = 2 \lambda / D$



NOTE: (extracted from Volume I of Reference 14, Appendix D and modified to include both U.S. customary units and System International units.)

FIGURE 33

Figure 33 illustrates the significance of surface finishes for wet seals. Surface finish is important in reducing the spacing between parallel plates. For both mercury and gallium alloy liquid metal systems at room temperature, the surface tension is greater than 0.005 Newtons/cm (500 + dynes/cm^2). Any contaminants present in the liquid phase or on the surface of the solid will, in general, tend to degrade wettability by lowering the surface tension.

While more rigorous mathematical solutions are available for treatment of idealized geometries, their contribution does not provide more direction to the engineering problem than this simple model.

The mercury-indium-thallium (Hg-In-Tℓ) alloy was tentatively selected for the liquid metal sealing concepts. This alloy possesses the following properties and the lowest melting point now known for a liquid metal alloy:

Liquid Metal System:	Hg-In-Tℓ
Melting Point:	214.4°K (386°R)
Surface Tension: (Room Temperature)	0.00532 Newton/cm (532 Dyne/cm)

The mercury-thallium alloy is highly reactive with oxygen and will rapidly degrade to form thallium oxides or hydroxides. Purified hydrogen can reduce the absorbed oxygen and prevent or retard additional reaction. It is the formation of the thallium oxide which probably produces the widespread wettability of this liquid metal system for most materials. For sealing purposes the presence of oxides is not detrimental. The mercury-thallium system does require special care and handling, although precautions are not prohibitively complex. The alloy system is reported to have incurred no apparent degradation after nuclear exposure to 1.35 x 10^{16} n/cm^2 greater than 1 Mev, 4.75 x 10^{15} n/cm^2 greater than 2.9 Mev, and 2.9 x 10^8R gamma.

This alloy was also selected since the work statement (Reference 1 of Appendix D) did not specify oxygen service."

Based on a maximum operating pressure differential of $206850 \frac{\text{Newton}}{\text{meter}^2}$ (30 lbf/in.^{2}) and utilizing Figure 33, it can be seen that the radial clearance between the metal lip seal containment system and the poppet skirt diameter cannot exceed 0.00056 cm (0.00022 in.) so that the selected liquid metal cannot be lost due to the insufficient surface tension.

For the optically flat poppet and seat plate seal, the capillary pressures can be much greater. By doubling the D values (due to two contacting parallel surfaces)

$$D = 10.16 \times 10^{-6} \text{ cm } (4.0 \times 10^{-6} \text{ in.})$$

and from Figure 33

ΔP across the seal can be greater than 6.895×10^6 Newton/meter^2 (1000.0 lbf/in.^{2}) in order to effect a seal.

2. Electromechanically Actuated Pressure or Flow Control Regulator

a. General

The pressure regulator shown in Figure 34, ANSC drawing 1140007, combines the features of the electromechanically actuated regulators originally shown on ANSC drawings 1139924 and 1139929. The concept, as shown, is an in-line design capable of being remotely coupled and uncoupled through the use of the remote coupling designs as shown on ANSC drawings 1140005 and 1140006. These remote couplings are described in subsequent paragraphs of this report.

ANSC drawing 1140007 consists of four sheets. Sheet 1 contains performance notes for the valve, motor and electronics and transmission. This sheet also contains a flow area curve, a valve torque curve, a sine wave response curve, a step response curve and a motor torque and current curve. Sheet 2 shows the cross section of the regulator along with materials and low melting alloy remote coupling ports compatible with 11400005-1 and -5. Sheet 3 shows the plan view of the regulator with the thermal interference remote coupling ports compatible with 1140006-1. This sheet also contains the actuator electrical schematic and two simplified RNS tank pressurization system schematics (one for a constant tank pressure system and the other for a programed tank pressure system). Sheet 4 contains the electronics module block diagram schedule.

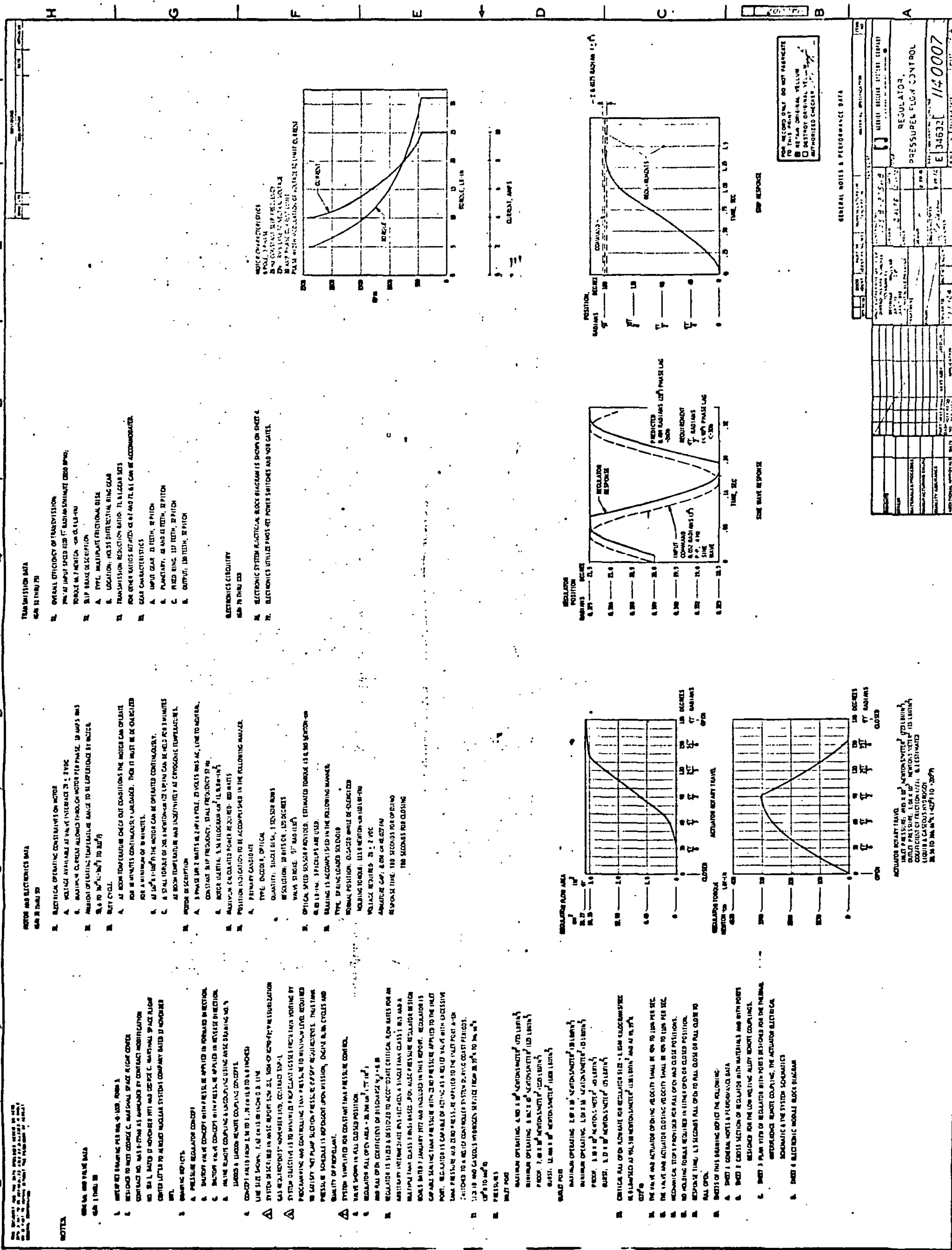


FIGURE 34
Page 1 of 4

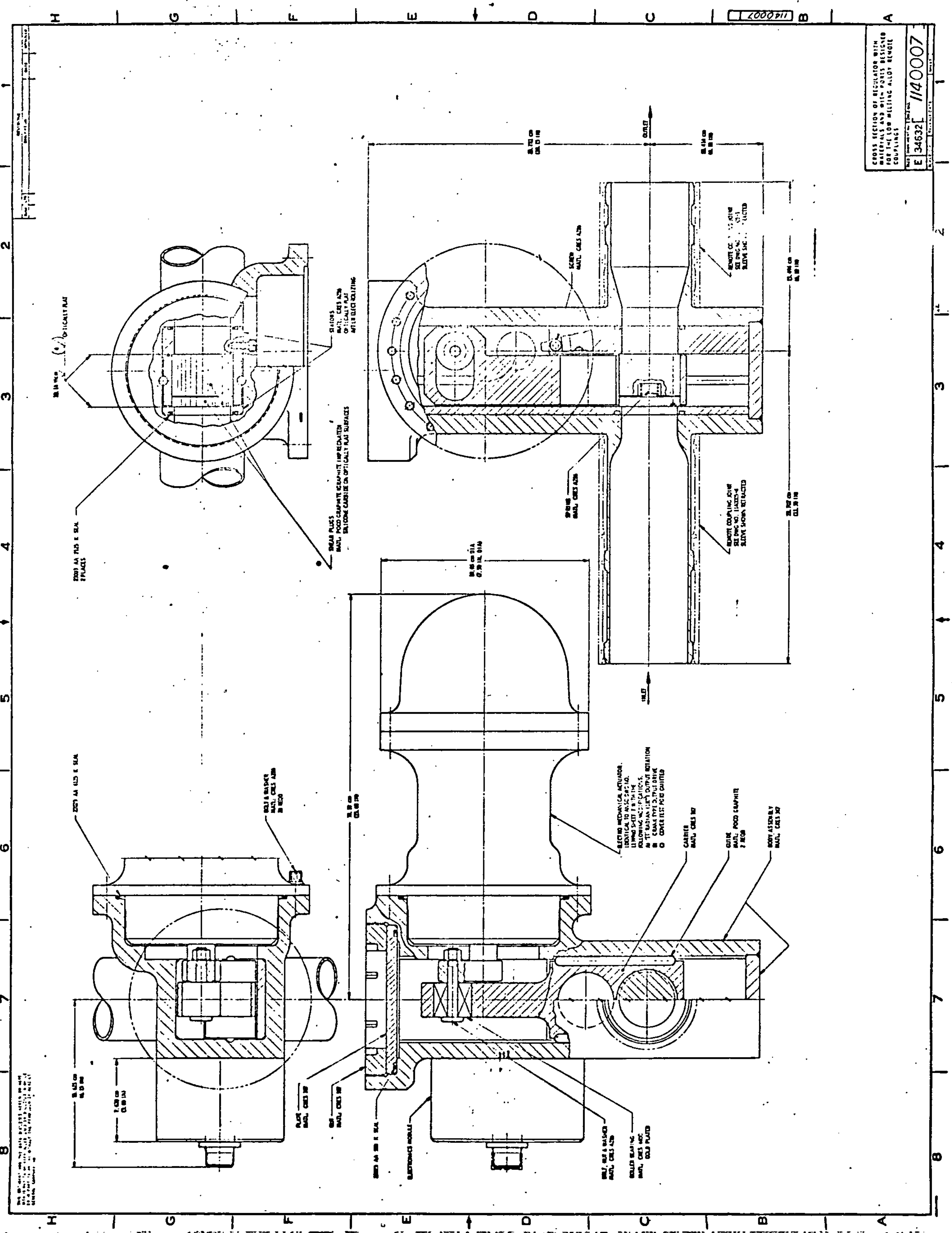


FIGURE 34
Page 2 of 4

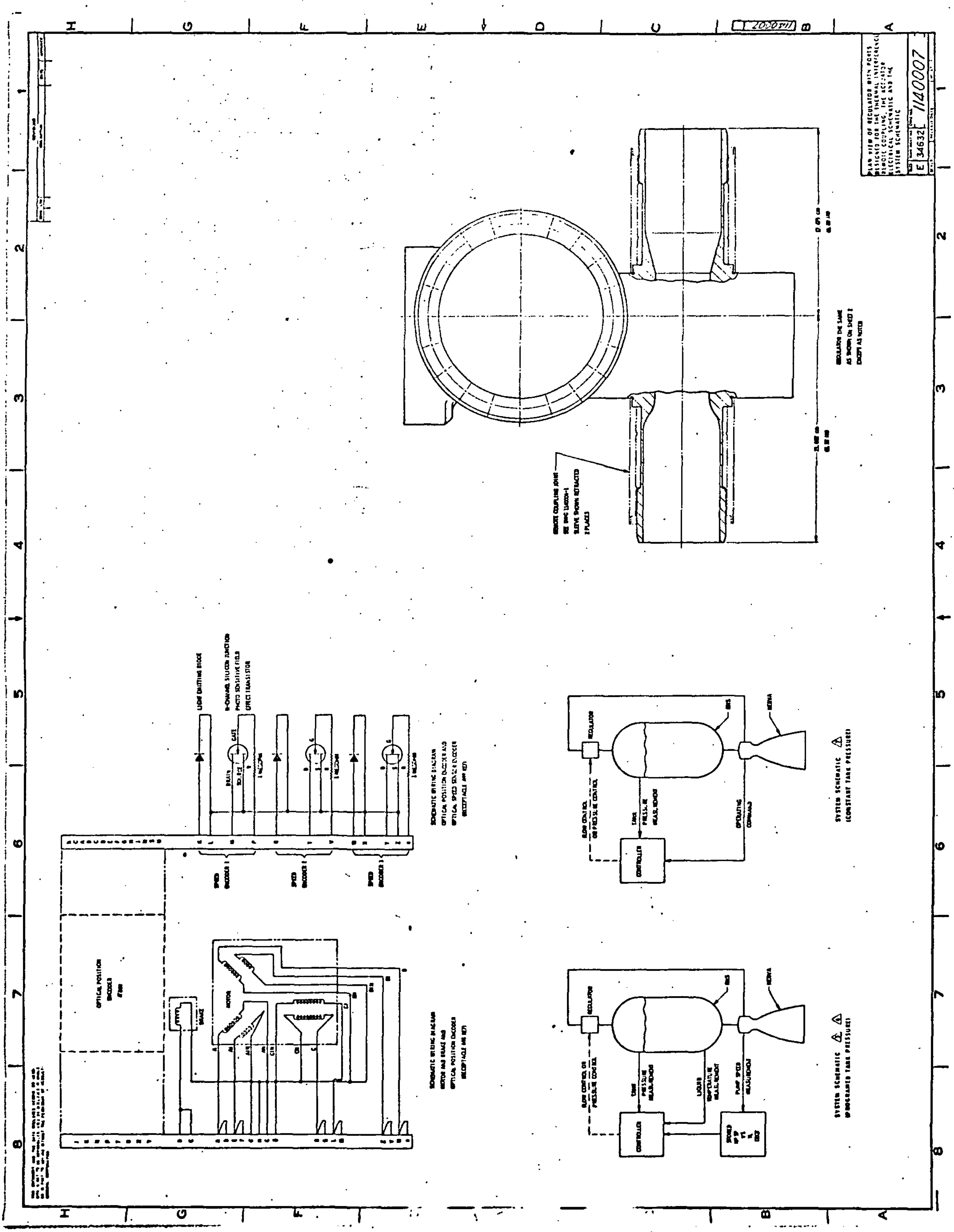


FIGURE 34
Page 3 of 4

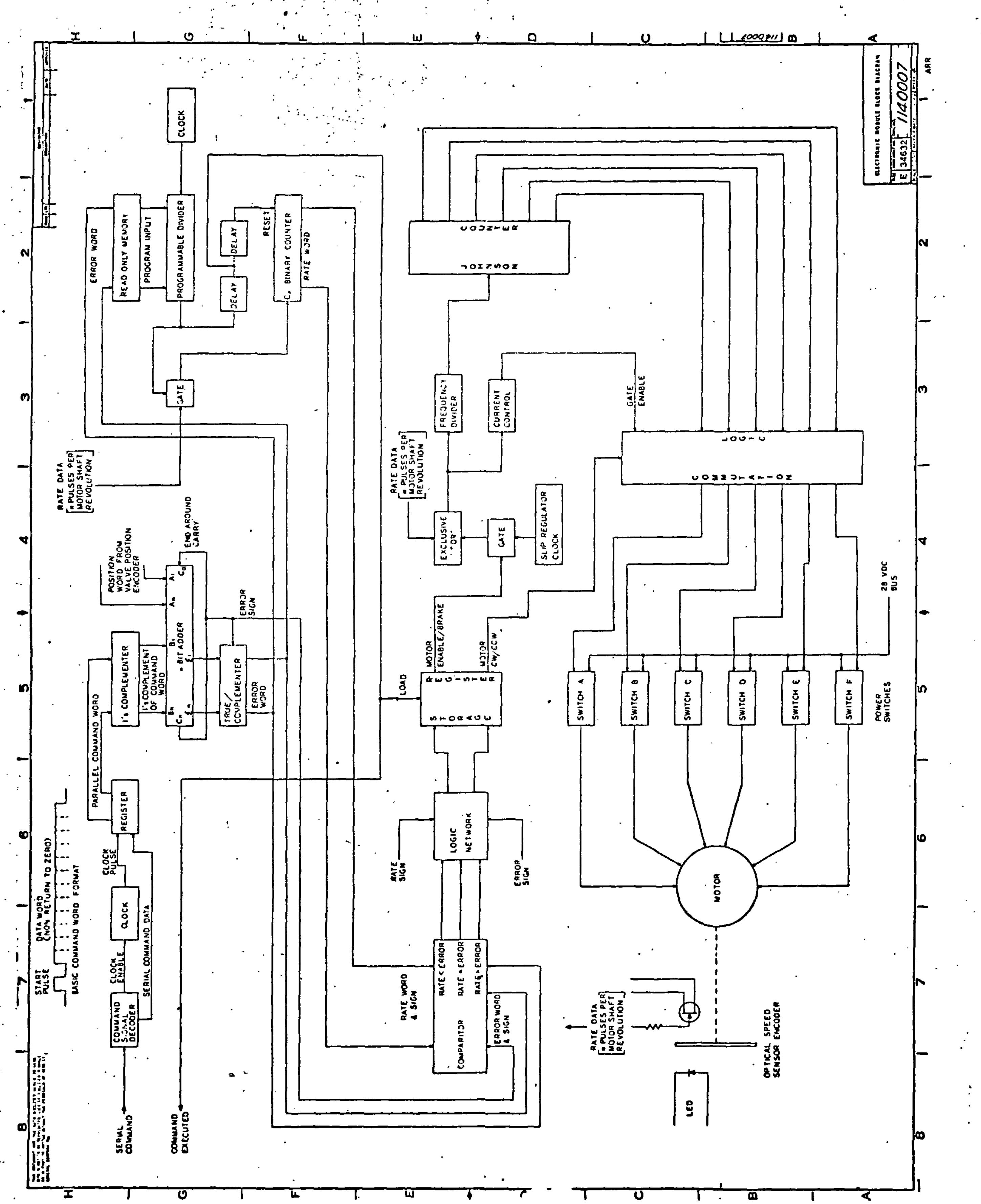


FIGURE 34
Page 4 of 4

b. System Discussion

The 1140007 regulator control function design was patterned after the ANSC 1139960 bypass control valve for the NERVA engine, scheduled for use as the propulsion system for the reuseable nuclear shuttle (RNS). Reference 16 recommends the autogenous tank pressurization for the single tank RNS for the lunar missions. The report concludes that large increases in payload can be realized if the mission is started with the propellant load at 15 psia saturation pressure, and the tank pressure programed to levels that satisfy minimum engine NPSP. This technique minimizes propellant losses due to minimal venting of tank pressurant. The regulator, which is controlled by programming from a digital controller, modulates to program tank pressure as a function of burn schedule, burn number, type of mission and engine start up time. The digital controller computes the tank pressure required based on liquid hydrogen temperature, actual tank pressure and the stored NPSP versus liquid temperature requirement combined with a turbopump speed measurement.

The unique electromechanical regulation system is capable of fast, accurate positioning of loads with optimized start, run, braking and reverse drive characteristics.

According to Reference 16 the maximum turbine bleed gas requirements occur during NERVA engine bootstrap startup in a malfunction mode (one TPA operating - 80% thrust - during second burn of ALU(unmanned lunar mission). Reference 16 was based on a single 10.06 meter (33 foot) diameter liquid hydrogen tank. The most recent tank configuration for RNS appears to be multiple tank system consisting of eight 4.88 meter (16 foot) liquid hydrogen tanks. Using Figure 35 it can be seen that the peak start up flow rate is 9.98 kilogram/sec (22.0 lbm/sec) occurring at approximately 26.0 seconds after start up for the single large tank.

Extrapolating for the small tank results in a flow rate of approximately 1.81 kilograms/sec (4.0 lbm/sec) at approximately 26.0 secs. Figure 35 can be rescaled for a peak flow rate of 1.81 kilograms/sec (4.0 lbm/sec).

Figure 36 shows turbine discharge bleed temperatures and pressures versus start up time. Temperatures and pressures versus time were assumed to remain constant regardless of the size or configuration of the RNS. Therefore, an analysis was performed of start up time for regulator CA (coefficient of discharge times regulator flow area) for a multiple tank system.

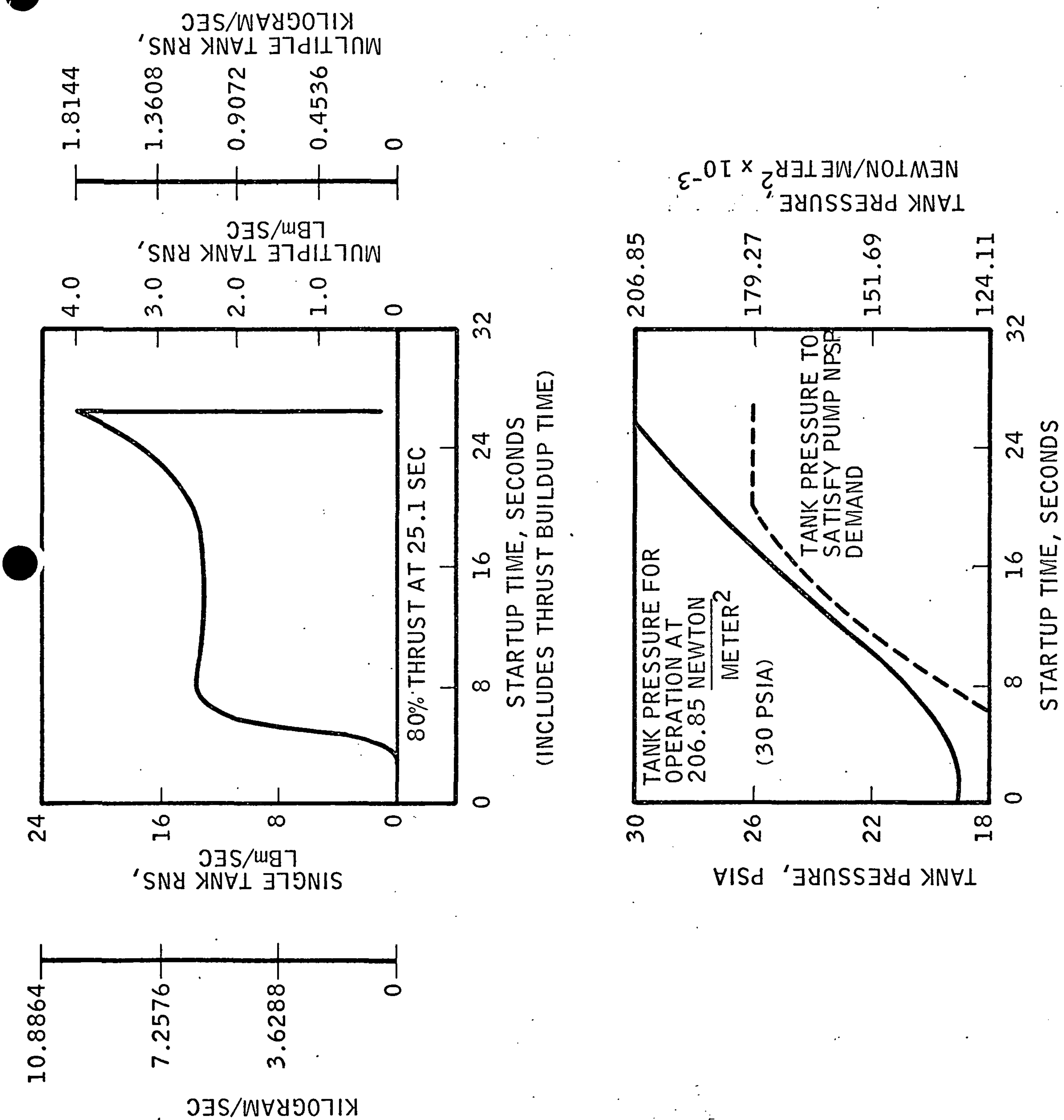
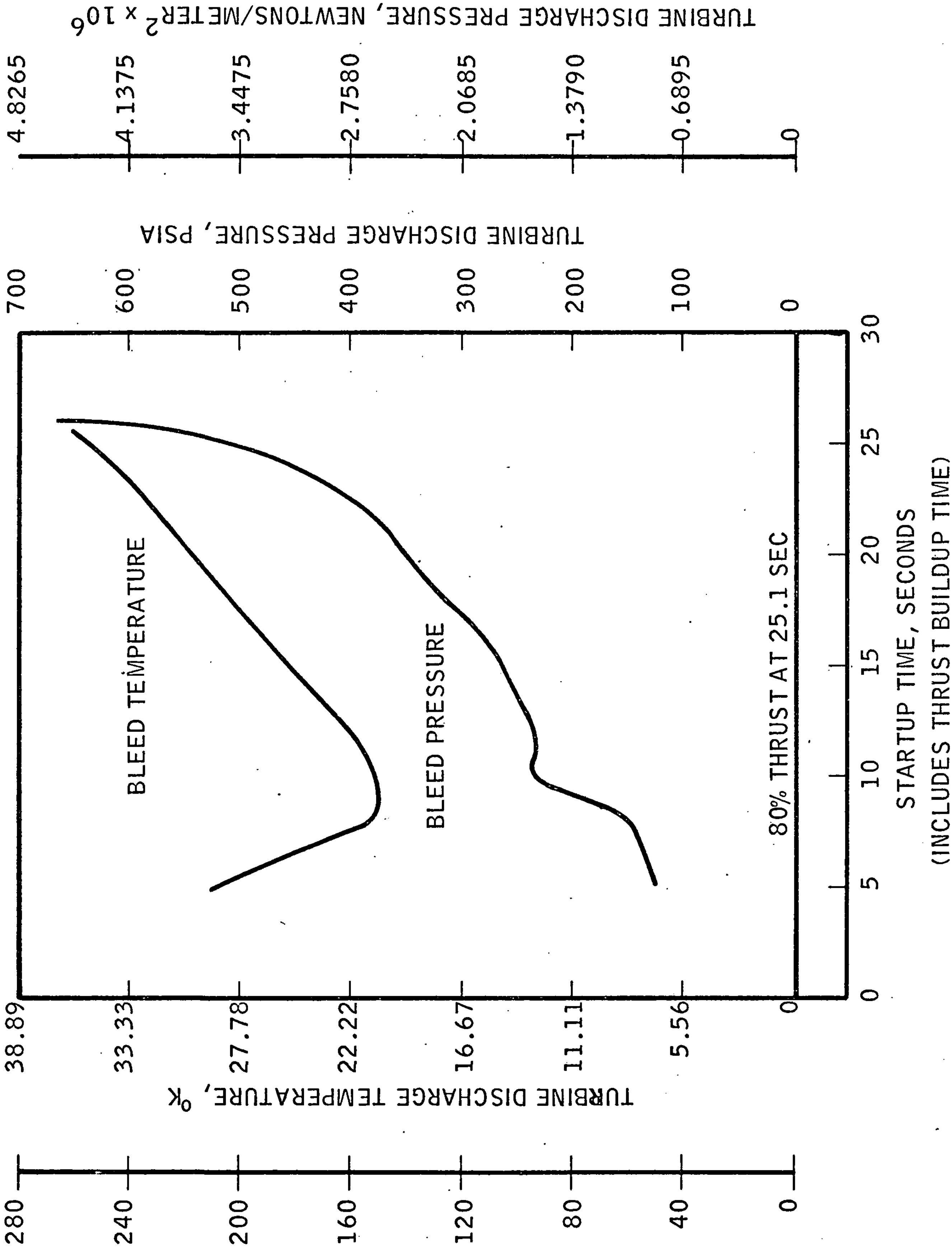


FIGURE 35
(REFERENCE: FIGURE 19/2 OF REFERENCE 16)(APPENDIX D)

MALFUNCTION MODE STARTUP FOR SECOND BURN OF UNMANNED LUNAR MISSION



MALFUNCTION MODE STARTUP FOR SECOND BURN OF UNMANNED LUNAR MISSION

FIGURE 36

(REFERENCE: FIGURE 19/1 OF REFERENCE 16) (APPENDIX D)

This analysis appears in Appendix P and uses the flow rates of Figure 35 and the pressures and temperatures of Figure 36. The critical CA versus start up time was thus determined for regulator sizing. The analysis was accomplished disregarding temperature rise or pressure drop from the engine bleed point (state point 36 of Reference 17) (Appendix D) to the regulator inlet (due to the unknowns of regulator location. line lengths and sizes and line heat leaks). Calculations start with the smallest size regulator (8 multiple tanks). The critical CA of the single tank regulator was also determined and an arbitrary intermediate size regulator was established as the final Phase II regulator concept. The tank configurations being studied for RNS vary between the two extremes noted above and it was deemed desirable to design an intermediate size regulator that could be scaled up or down depending on the final RNS tank configuration. Engine steady-state data (from Reference 17, (Appendix D) was required in order to evaluate the regulator turn down ratios, since it may be necessary to consider the use of a large and a small size regulator for certain tank configurations. Steady-state data appears in Appendix P and should remain constant regardless of the stage tank configuration.

As shown in Appendix P, the critical CA of the regulator occurs at approximately 7.0 seconds after the initiation of start up. This determines the minimum throat size of the regulator. In the case of the multiple tank regulator the fully open CA ≈ 10.84 cm^2 (1.68 $in.^2$) at approximately 7.0 seconds after initiation of start up and the CA = 4.51 Cm^2 (0.6986 $in.^2$) at approximately 26.0 seconds. The CA for the steady-state operating point extremes lies between 0.0463 to 0.6497 cm^2 (0.0718 to 0.1007 $in.^2$) regardless of the tank size. The CA for throttling mode is not known. However, the flow rate for the throttle hold mode is estimated to be approximately 60% of steady-state flow rates. The CA position accuracy from the start of the throttling mode through cooldown and pulse modes is expected to be liberal with the regulator possibly shut off early during temperature retreat. It must be recalled that the regulator position is controlled by the onboard computer that maintains tank NPSP as a function of tank pressure and propellant temperature. The CA turn down ratio for the multiple tank regulator from start of bootstrap through steady-state operation is on the order of 24.3 to 1.0. For the large single tank regulator the fully open CA = 59.61 cm^2 (9.24 $in.^2$) at approximately 7.0 seconds and the CA at approximately 26.0 seconds is 24.79 cm^2 (3.842 $in.^2$).

The turn down ratio for the large single tank regulator from the start of bootstrap to the end of steady-state operation would be on the order of 128.7 to 1.0 and may require the use of a large size and small size regulator (see Appendix P for the NERVA engine operational phases). It should be noted that the NERVA engine requires large flow rates at bootstrap start up (as a function of propellant and tank pressure condition) and very small flow rates during steady-state operation.

The regulator design, as shown on ANSC dwg. 1140007, (Fig.34), was arbitrarily scaled up to incorporate a throat area of 20.27 cm^2 (π $in.^2$) or 5.08 cm (2.00 in.) diameter to allow for some line pressure drop and line temperature rise. This would also allow for some requirement fluctuations until the RNS configuration is finalized.

$$\text{Flow Area} = 20.27\ cm^2\ (\pi\ in.^2)$$

Assume a coefficient of discharge (c) = 0.85 for a reasonable rounded entrance orifice. Therefore, for the ANSC 1140007, (Figure 34 regulator

$$CA = 17.23\ cm^2\ (2.67\ in.^2)$$

The CA turndown ratio is 37.2 to 1.0 from the start of engine bootstrap operation to the end of steady-state engine operation.

The inside diameter of the line for the regulator was sized at approximately two times the full open geometric area and yielded a 7.62 cm (3.00 in.) outside diameter line. This gave a reasonable wall thickness of the line of nominally 0.254 cm (0.10 in.). Hoop stresses in the line are compatible with the allowable strengths of the AISI 347 CRES material selected.

c. The Regulator Mechanism (ANSC 1140007, Sheet 2)

The main regulator metering element consists of an optically flat cylindrical shear plug operating pressure unbalanced on an optically flat stator. In the closed position the shear plug acts as a seal for positive shutoff of flow to the tank. An auxiliary optically flat shear plug also operates pressure unbalanced on a second optically flat stator to provide positive shutoff from the outlet port (tank side) to the inlet port (NERVA engine side) in the closed position with zero pressure supplied from the NERVA engine. The auxiliary shutoff feature protects the tank from significant propellant tank pressure loss during the RNS coast periods. The metering and shutoff elements are floating in a linear carrier that contains a straight slot for the actuator output crank. The output crank which is attached to the electro-mechanical actuator translates the carrier as the actuator rotates. This translation positions the shear plug in the desired flow area position as a function of the controller input command signal and the position feedback and rate feedback in the servo loop.

Presently, both shear plug metering elements are constructed of POCO graphite (see Footnote 1) and are gas pulse impregnated with carbon (see Footnote 2) which should yield permeability leak rates of less than 10^{-7} scc/cm^2/sec of helium. The sealing surface is then given a silicon carbide conversion coating (see Footnote 3) for extreme hardness, wear resistance and low coefficient of friction. The sealing surfaces are then optically flat lapped to approximately 5.08 μ cm (2.0 μ in.) surface finish. Optical flatness will be on the order of 1/8 of a helium light band.

The stators for both shear plug stators would be fabricated from A-286 material that has been electrolized (see Footnote 4) and optically flat lapped to a surface finish and flatness the same as the sealing surfaces of the shear plugs. The basic valve body is constructed from an AISI 347 CRES forging.

Sheet 1 of ANSC drawing 1140007 reflects a curve of flow area versus actuator rotation. This curve is basically linear for the major protion of the open area. It is based upon a straight slot in the carrier and π radians (180°) of actuator rotation. Planimeter techniques were used in determining the flow area versus actuator rotation. The moment arm of the actuator was established at 2.794 cm (1.1 in.) for the straight slot configuration and with the shear plug and full open flow area shown. The flow area versus actuator rotation curve can be varied by changing slot contours, slot lengths, crank arm lengths and actuator rotational angles.

---

1. POCO Graphite AXF - product of POCO Graphite, Inc.
2. Gas Pulse Impregnation of Graphite with Carbon-Nuclear Applications and Technology, Volume 8, June 1970.
3. Pyto-Tech PT300, A Silicon Carbide Graphite Conversion - Product of ULTRA Carbon Corporation.
4. Electrolize - A Dense Chrome Plate Process of Electrolizing, Inc.

Sheet 1 of ANSC drawing 1140007 also shows a curve of regulator torque versus actuator rotation. This curve was based on shear plug friction forces that are assumed constant from the fully closed position to the $\frac{\pi}{2}$ radians (90°) actuator rotational position. At this point the friction forces are assumed to vary linearly to zero friction at 2.723 radians (156°) (the point in the actuator rotation where the shear plug edge coincides with the flow passage hole edge). The friction force then remains at zero under further rotation to the fully open position.

Torque can then be expressed as

$$T = (F_f)\ (R)\ (\cos\theta)$$

where (see Figure 37)

T = Torque = Newton-cm (lbf-in.)

F_f = Friction force = Newton (lbf)

$F_f = F_n\ \mu$

F_N = Normal Force = Newton (lbf)

μ = Coefficient of friction = 0.1 assumed conservative for the materials selected

$F_N = P\ A$

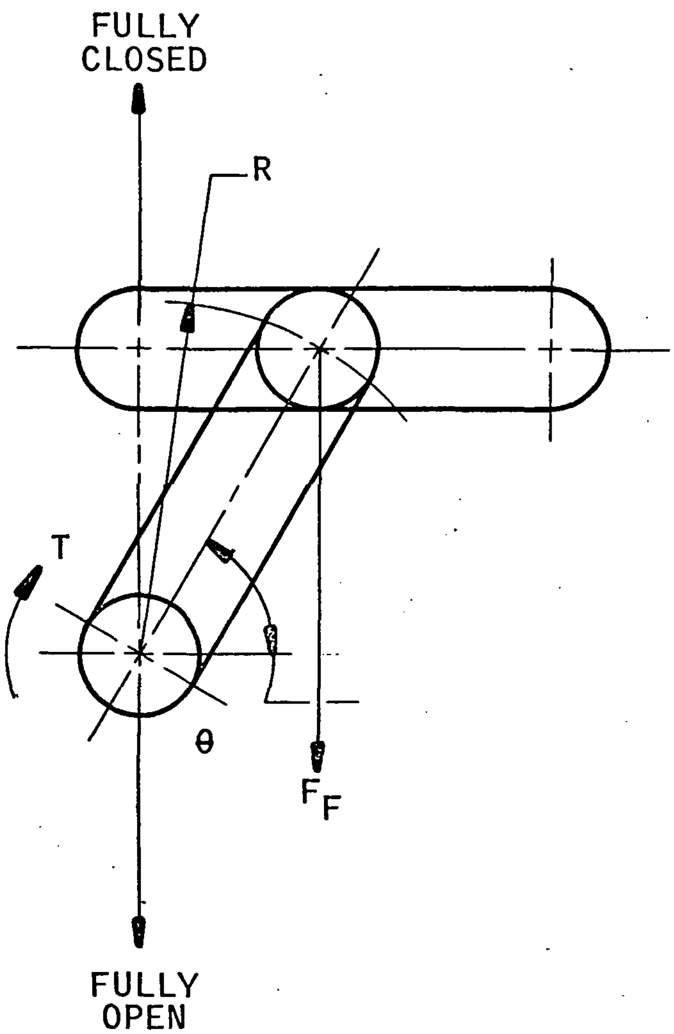
P = Inlet port pressure = $4985 \times 10^3 \frac{\text{Newton}}{\text{meter}^2}$ $(723 \frac{\text{lbf}}{\text{in.}^2})$ constant and then varying as noted above.

A = Area of the shear plug = $9.655\ \text{cm}^2$ $(3.8\ \text{in.}^2)$

R = Moment arm of crank = 2.79 cm (1.1 in.)

θ = Actuator crank rotation from zero to π radians (zero to 180°)

Using the above equation, regulator torque versus actuator rotation was then calculated and plotted for a series of θ values.



PRESSURE REGULATOR TORQUE ARM

FIGURE 37

d. The Electromechanical Actuator and Electronic Module Discussion

The electromechanical actuator is essentially identical to the one shown on ANSC Dwg. 1139960, Sheets 1 and 7, with the following modifications.

(1) π radian (180°) output rotation in lieu of 2.15 radians (123°).

(2) Crank type output drive in place of the direct, on center drive.

(3) The test port on the actuator end has been omitted.

A 3-phase squirrel cage induction motor of custom design is operated in a variable frequency controlled slip mode. The frequency input to the fixed stator is a function of rotor speed. As a function of the control mode, a frequency, called slip frequency, is either added to or subtracted from the rotor speed and the total frequency is fed to the stator. Such control yields high starting torque as constant horsepower, optimum induction regenerative braking and maximum braking torques. This is achieved with controlled starting current. Reversing and forward torques are identical. For the same current drain, full electric braking torques can be on the order of four times that of starting. The systems uses DC power. The digital logic and power units can be replaced on a modular basis. Both rotary position feedback and rotary rate feedback are obtained through the incorporation of two separate optical encoders in the electromechanical actuator. The all-digital solid-state electronics are packaged in environment-resistant controller and power modules. The controller module can be used to control one or more drives of the same or different HP ratings. The regulator actuator and the electronics module are of nuclear quality. The all digital electronic components and circuitry is furnished radiation hardened. The nominal output torque of the actuator is 3842 Newton-centimeters (340 lbf-in.) which is more than sufficient to accommodate the regulation element peak torque of 3418.25 Newton-centimeters (302.5 lbf-in.). The transmission consists of a compound planetary gear system and utilizes a gear reduction of 71.8:1. The electromechanical actuator and electronics module could be available for Phase III design and hardware modification.

3. Remote Couplings

The two coupling concepts studied during Phase II use a low melting alloy and a thermal interference to obtain sealing.

a. Low Melting Alloy Remote Coupling Joint and Seal

General

The low melting alloy remote coupling and seal, as shown on ANSC drawing 1139921, was reviewed in order to assess its most efficient adaptability to the following ANSC design concepts:

New No.	Original No.	Nomenclature	New Line Size cm	in.
1140001	1139893	Shutoff Valve	12.70	5.0
1140002	1139897	Shutoff Valve	20.32	8.0
1140007	1139924	Regulator	7.62	3.0

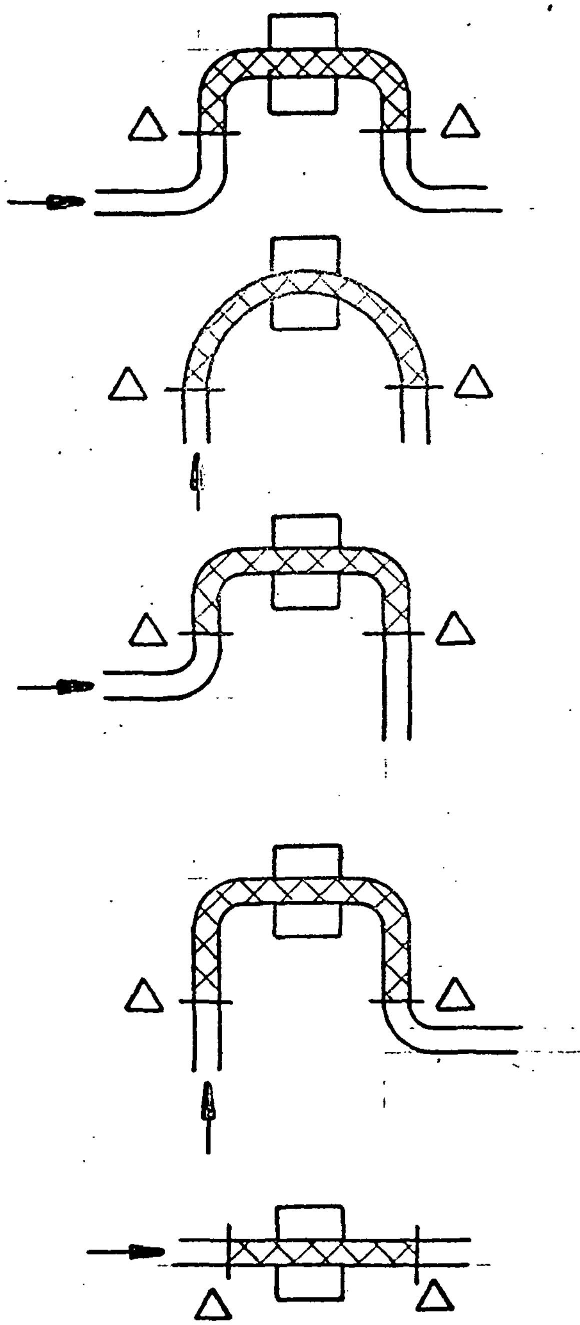
Concept drawings 1139893 and 1139924 reflect in-line ports. Originally, concept drawing 1139897 utilized right angle ports but has been redesigned to incorporate in-line ports for straight through flow and low pressure drop. The low melting alloy joint was therefore redesigned so that the three valve and regulator concepts can be installed in-line as shown on ANSC drawing 1140005, (Figure 39). Figure 38 shows the various line and component routing possibilities for the low melting alloy remote coupling, including the in-line approach. The design, as shown in Figure 39, ANSC drawing 1140005, requires certain line axial deflections so that the pressure area loads can produce pure shear on the solidified alloy. A similar low melting alloy structural joint and seal has been used successfully by National Aeronautics and Space Administration, Lewis Research Center (Reference 25), (Appendix D).

ANSC drawing 1140005, (Figure 39) consists of four sheets. Sheet 1 contains general notes and the material properties of the 55.5% Bismuth/44.5% lead low melting alloy selected. Sheet 2 shows the assembled design and materials for the selected sizes of couplings. Sheet 3 reflects tabulated dimensions for these coupling sizes. Sheet 4 establishes a suggested semi-schematic approach to a manipulator hand injection system for a 12.70 cm (5.00 inch) outside diameter line as well as showing injection and ejection system schematics.

VARIOUS LINE AND COMPONENT ROUTING APPROACHES

FOR

REMOTE COUPLING



△ INDICATES REMOTE JOINT

⋀⋀⋀ INDICATES COMPONENT

FIGURE 38

NOTES:

1. INTERPRET DRAWING PER MIL-D-1000, FORM 3.
2. DESIGNED TO MEET GEORGE C. MARSHALL SPACE FLIGHT CENTER CONTRACT NO. NAS 8-27568 AS AMENDED BY CONTRACT MODIFICATION NO. S.A. 1, DATED 17 NOVEMBER 1971 AND GEORGE C. MARSHALL SPACE FLIGHT CENTER LETTER TO AEROJET NUCLEAR SYSTEMS COMPANY DATED 17 NOVEMBER 1972.
3. DRAWING DEPICTS
   A. REMOTE COUPLING AND SEALING OF A LINE TO A LINE.
   B. REMOTE COUPLING AND SEALING OF A COMPONENT TO A LINE.
4. CONCEPT VALID FROM 0.635 TO 50.800 CM (0.25 TO 20.00 INCHES).
5. CONCEPT VALID FOR LINE-TO-LINE JOINTS OR COMPONENT-TO-LINE.
6. DESCRIPTION OF OPERATION: AN INLINE VALVE OR REGULATOR, CONTAINING THE RETRACTED SLEEVES (SEE ANSC DRAWINGS 1140001, 1140002 AND 1140003), IS LOCATED BETWEEN THE OPEN LINE PORTS. THE SLEEVES ARE THEN EXTENDED AND WELDED OVER THE LINE PORTS. A HEATER-INJECTION SYSTEM, WHICH IS A PART OF THE MANIPULATOR HAND, CLAMPS ON TO THE SLEEVE AND AUTOMATICALLY HEATS THE SLEEVE AND MELTS THE LOW MELTING ALLOY TO APPROXIMATELY 422°K (300°F) TEMPERATURE. THE SPRING LOADED PISTON RESERVOIR PUMPS THE MOLTEN ALLOY INTO THE ANNULAR CAVITIES IN THE COUPLING JOINT. WHEN THE ANNULAR GROOVES ARE FULL OF MOLTEN ALLOY, THE EXCESS FLOWS INTO THE POROUS CATCH RECEPTACLE. AT APPROXIMATELY 422°K (300°F) TEMPERATURE, THE EXCESS MOLTEN ALLOY ACTUATES A TEMPERATURE SWITCH WHICH TURNS OFF POWER TO ALL HEATERS AND DE-ENERGIZES A SOLENOID VALVE THAT SHUTS OFF FLOW INTO THE CATCH RECEPTACLE. THE LOW MELTING ALLOY IS THEN ALLOWED TO SOLIDIFY AND FORM BOTH A STRUCTURAL COUPLING AND SEAL. FOR EJECTION A SIMILAR HEATER SYSTEM IS USED WHICH MELTS THE ALLOY. AT APPROXIMATELY 422°K (300°F) TEMPERATURE A SWITCH ACTUATES, ENERGIZING AND OPENING A SOLENOID VALVE WHICH ALLOWS THE MOLTEN ALLOY TO BE PNEUMATICALLY PUMPED INTO THE POROUS CATCH RECEPTACLE.

   WHEN THE TEMPERATURE AT THE ENTRANCE TO THE CATCH RECEPTACLE DROPS TO APPROXIMATELY 397°K (255°F) TEMPERATURE DUE TO PNEUMATIC FLOW, A SECOND TEMPERATURE SWITCH TURNS OFF POWER TO THE HEATERS AND DE-ENERGIZES THE SOLENOID VALVE. THE MANIPULATOR HAND IS THEN REMOVED AND THE SLEEVES ARE RETRACTED. THE VALVE OR REGULATOR CAN THEN BE REPLACED. THE POLYIMIDE BARRIER SEALS PREVENT MIGRATION OF THE MOLTEN ALLOY.
7. DASH NUMBERS SHOWN COINCIDE WITH THE TABULATED DASH NUMBERS.
8. THE CANDIDATE LOW MELTING ALLOY AND PERTINENT PROPERTIES ARE SHOWN IN TABLE 1.
9. ESTIMATED INJECTION OR EJECTION PRESSURE 103,425 N/m² (15 PSI)
10. SIMILAR TO ANSC DRAWING 1139921.
11. SHEETS OF THIS DRAWING DEPICT THE FOLLOWING:
    A. SHEET 1 GENERAL NOTES & MATERIAL PROPERTIES OF THE SELECTED LOW MELTING ALLOY
    B. SHEET 2 GENERAL CONSTRUCTION OF THE 4 SIZES OF LOW MELTING ALLOY REMOTE COUPLINGS
    C. SHEET 3 TABULATED DIMENSIONS FOR THE 4 SIZES OF LOW MELTING ALLOY REMOTE COUPLINGS
    D. SHEET 4 SEMI-SCHEMATIC OF 12.70 cm (5.00 IN) SIZE MANIPULATOR HAND HEATER-INJECTION SYSTEM AND SCHEMATICS OF BOTH MANIPULATOR HAND HEATER-INJECTION AND HEATER-EJECTION SYSTEMS

TABLE 1

PROPERTIES	CERROBASE* EUTECTIC U.S. CUSTOMARY UNITS	*TRADE NAME OF CERRO SALES CORP SI UNITS
MELTING TEMPERATURE	255°F	397°K
RANGE	255-255°F	397-397°K
YIELD TEMP	255°F	397°K
SPECIFIC GRAVITY	10.5 AT 20°C	10.5 AT 293°K
TENSILE STRENGTH	6400 LB/IN² AT 70°F	44,120 kN/m² AT 294°K
* ELONGATION (SLOW LOADING)	60-70%	60-70%
BRINELL HARDNESS NO.	10.2	10.2
* SPECIFIC HEAT LIQUID	0.042 BTU/LB-°F	175.73 J/kg-°K
* SPECIFIC HEAT SOLID	0.031 BTU/LB-°F	129.52 J/kg-°K
* LATENT HEAT - FUSION	7.2 BTU/LB	16,736 J/kg
* COEFFICIENT OF THERMAL EXPANSION	0.000021 IN/IN-°C	0.000021 m/m-°K
THERMAL CONDUCTIVITY (SOLID) % COPPER	0.000323 BTU-IN/FT²-SEC-°F	0.1676 W-m/m²-SEC-°K
CONDUCTIVITY ELECTRICAL COMPARED WITH PURE COPPER	1.79%	1.79%
* MAXIMUM LOAD - 30 SECONDS - YTS	8000 LB/IN²	55,160 kN/m²
* MAXIMUM LOAD - 5 MINUTES - YTS	4000 LB/IN²	27,580 kN/m²
* SAFE LOAD - SUSTAINED - YTS	300 LB/IN²	2,068 kN/m²
VOLUME CHANGE LIQUID TO SOLID	-1.5%	-1.5%
VOLUME CHANGE LINEAR GROWTH AFTER SOLIDIFICATION	0.3%	0.3%
*APPROXIMATE ROOM TEMP VALUES		
COMPOSITIONS		
BISMUTH	55.50%	55.50%
LEAD	44.50%	44.50%
NUCLEAR PROPERTIES (ATOMIC NO.)		
BISMUTH	83	83
LEAD	82	82
WORKING TEMPERATURE RANGE	-423 TO 70°F	20 - 294°K

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RETAIN ORIGINAL VELLUM
DESTROY ORIGINAL VELLUM
AUTHORIZED CHECKER

CONTRACT NO. NAS8-27568
VIC ROMANI
AEROJET NUCLEAR SYSTEMS COMPANY
COUPLING, REMOTE - LOW MELTING ALLOY SEAL
E 34632 1140005

FIGURE 39
Page 1 of 4

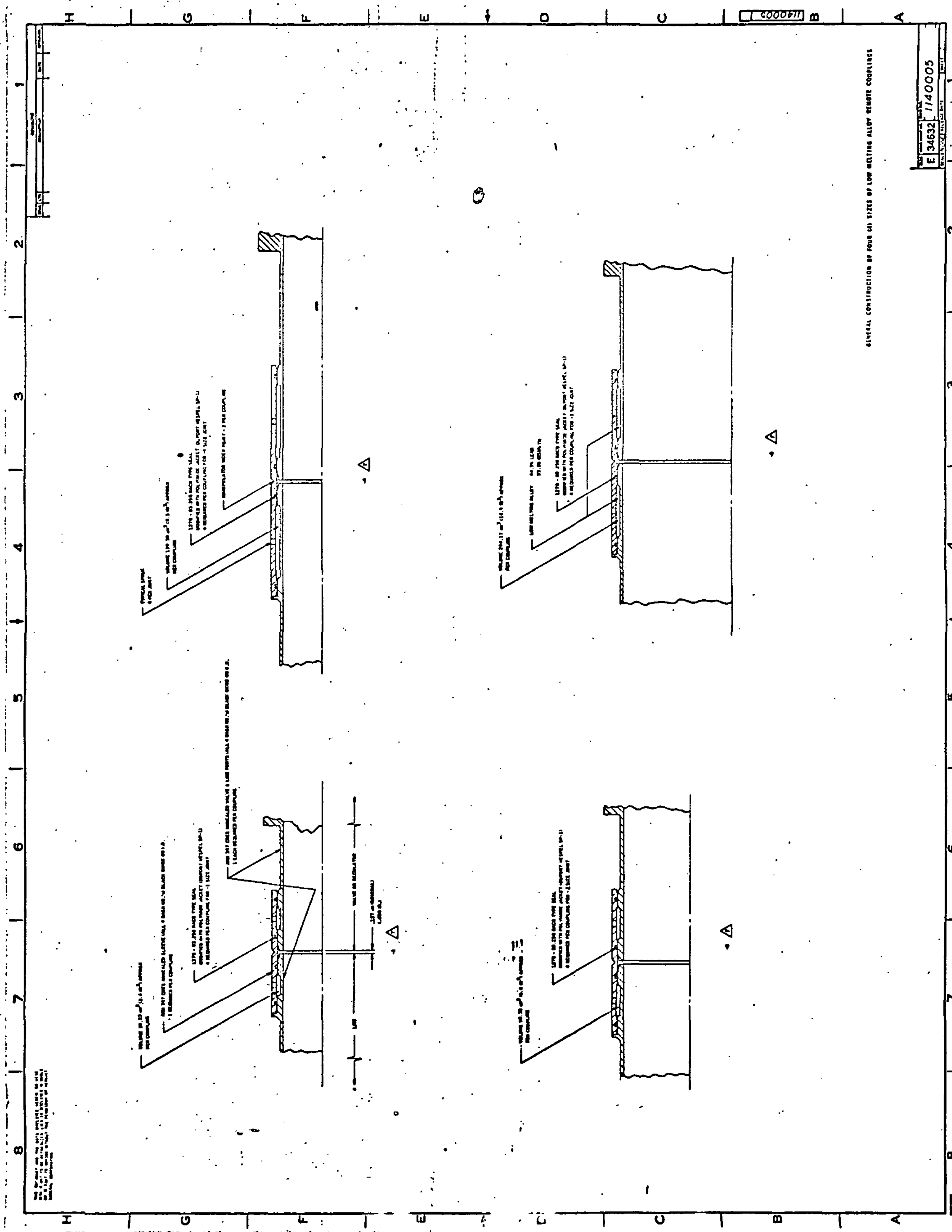


FIGURE 39
Page 2 of 4

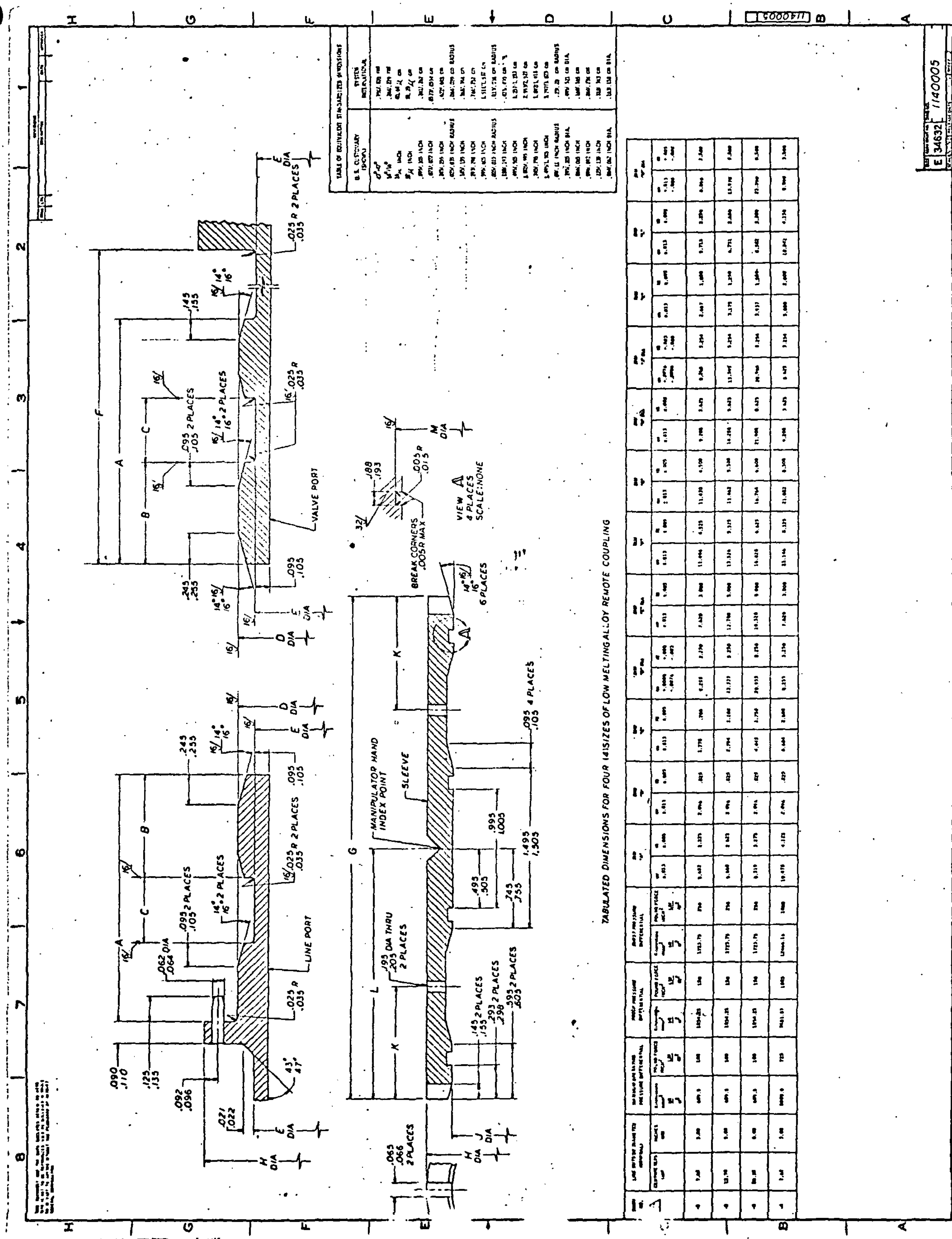


FIGURE 39
Page 3 of 4

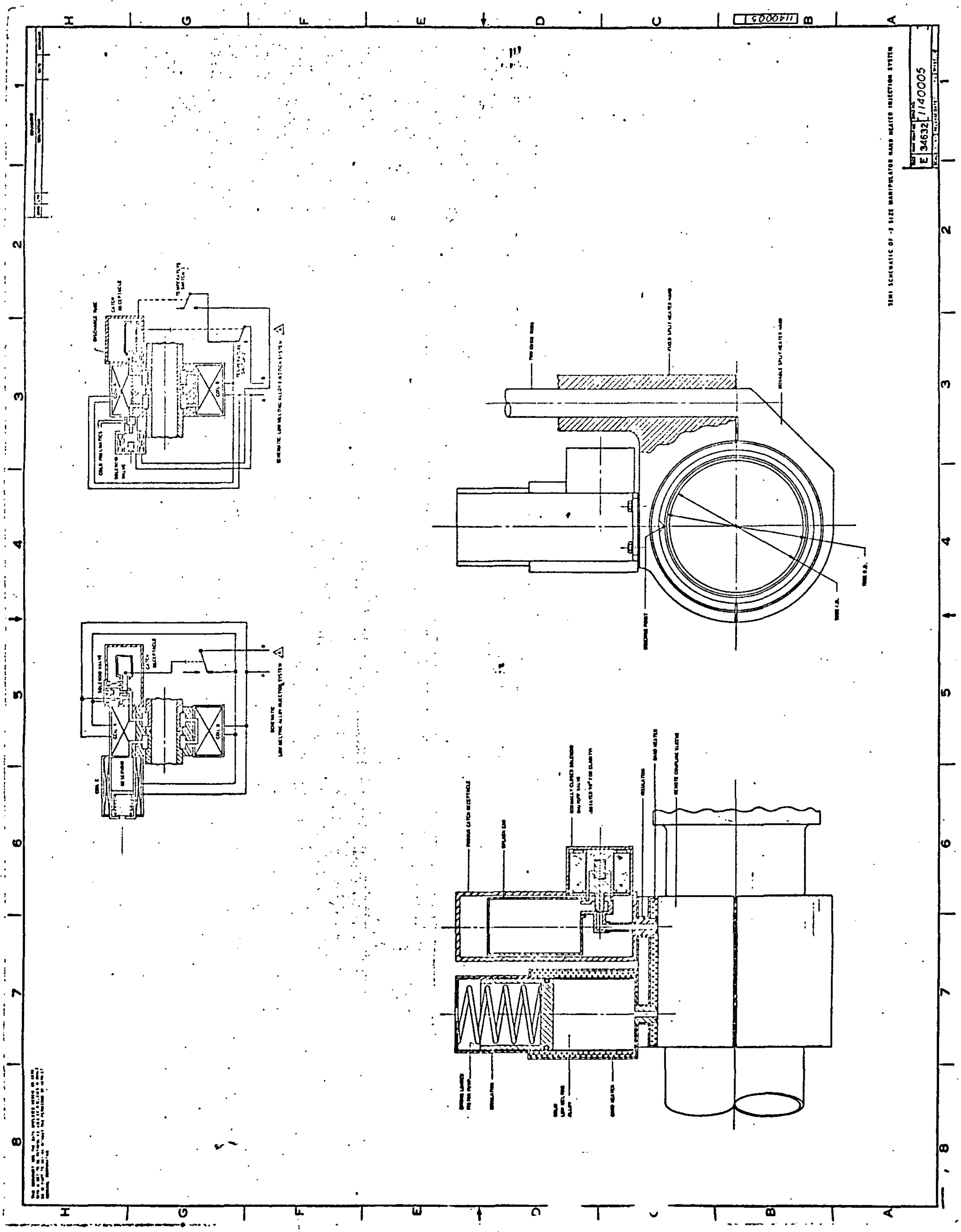


FIGURE 39
Page 4 of 4

The concept was developed from the original concept shown on ANSC drawing 1139921. The heater elements were removed from the flight portion of design and integrated with the manipulator hand to decrease flight weight and increase flight reliability. The sleeve was made integral with the valves and regulator so that sleeves and polyimide barrier seals can be inspected and replaced along with the component, if required, at the space depot. The integral manipulator hand heater injection and heater ejection systems can also be inspected, repaired, and recharged with, or purged of, the low melting alloy. The low melting alloy selected expands on solidification and produces surface compression on the joint surfaces. The difference in thermal expansion between the low melting alloy and its surrounding AISI 347 CRES cavity is more than offset by the expansion during solidification.

Installation of In-Line Components

An in-line valve or regulator component containing the retracted sleeves (reference Figures 28, 32, and 34) is located between the open line ports. The component can be indexed on the center of the line by the use of two tapered locating pins which mate with two tapered holes that are a part of the line or stage structure. As the tapered pins engage the tapered holes, an electrical connection is also made. Magnets can be added to the tapered pins or holes to positively locate the component and cancel any impact reaction forces.

The sleeves are then extended and indexed over the open line ports, thus forming cavities ready to receive the low melting alloy. The low melting alloy injection system can be a fully atuomatic device that is a part of a special manipulator hand. The general type of hand suggested is shown in Figure III of Reference 18, (Appendix D). The basic components associated with hand are as follows:

- A spring loaded piston type low melting alloy reservoir charged with the solidified alloy. The entire reservoir portion is surrounded by a band type electrical heater.

- A split band type heater system with each half attached to each portion of the manipulator hand. The heaters are insulated from the frame work of the hand to minimize conduction and radiation losses. Four seals for the four ports in the sleeve are incorporated in the inside diameter of the split band heater system.

- An overflow catch receptacle that consists of a porous outer housing. The porous design would be such that gas and vapor would penetrate its walls, but the receptacle walls would be impervious to molten alloy penetration. The catch receptacle would include an internal splash can, a normally closed solenoid shutoff valve, and a temperature sensor probe.

The manipulator heater injection hand indexes on the sleeve and then is maneuvered to lightly clamp the sleeve. The heaters are then automatically activated and the solenoid valve is energized open. The sleeve heaters lead the reservoir heaters so that when the alloy melts in the reservoir at approximately 397°K (255°F), the coolest annular cavity areas are at a minimum of 422°K (300°F).

The inside diameter of the sleeve and the outside diameter of the valve and line ports are black bodies for maximum radiation emissivity, due to the radial clearance between the mating members. As the molten alloy starts to flow into the annular cavities, thermal conductivity across the members starts to predominate and joint filling time decreases. This factor is not accounted for in the power requirement analysis in Appendix Q and will result in more conservative power requirements.

When the molten alloy has filled the annular cavities and starts to overflow into the catch receptacle, a temperature switch is actuated at approximately 422°K (300°F) temperature. The actuation of the switch shuts off power to all heaters and de-energizes (closes) the solenoid shutoff valve; stopping flow of the molten alloy. The molten alloy is then allowed to cool and solidify.

The manipulator hand is then expanded, severing the four sprues on the outside diameter of the sleeve, and is then maneuvered away from the coupled joint. The low melting alloy reservoir can be sized to join more than one remote coupling joint prior to recharging at the space depot.

The barrier seals prevent migration of the molten alloy to undesirable areas. The black oxided surfaces of the annular cavities prevent surface adhesion (wetting) of the molten alloy.

Removal of In-Line Components

The low melting alloy ejection system can be a fully automatic device that is a part of a special manipulator hand similar to the injection system described above. The basic components associated with the manipulator hand are as follows:

- A normally closed solenoid shutoff valve that either blocks or allows pneumatic flow from the manipulator.

- A split band type heater system with each half attached to each portion of the manipulator hand. The heaters are insulated from the frame work of the hand to minimize conduction and radiation losses. Four seals for the four ports in the sleeve are incorporated in the inside diameter of the split band heater system.

- A catch receptacle that consists of a porous outer housing. The porous design would be such that gas or vapor would penetrate its walls, but the receptacle walls would be impervious to molten alloy penetration. The catch receptacle would include an internal splash can and two temperature sensor probes.

The manipulator heater ejection hand indexes on the sleeve and then is maneuvered to lightly clamp the sleeve. The heaters are automatically activated while the solenoid shutoff valve remains de-energized (closed). When the molten alloy discharge tube reaches a pre-determined temperature of approximately 422°K (300°F), temperature switch No. 1 is actuated; causing the solenoid shutoff valve to energize and open. At this point the alloy in the annular cavities is molten. Low pressure pneumatics on board the manipulator then pumps the molten alloy into the catch receptacle. When the alloy is completely pumped from the annular cavities the temperature at the discharge tube drops due to the lower pneumatic temperatures, thus actuating temperature switch No. 2. This action shuts off all power to the heaters and de-energizes (closes) the solenoid shutoff valve.

The manipulator hand is then expanded and is maneuvered away from the uncoupled joint. The sleeves are then retracted and the component removed. The catch receptacle can be sized to uncouple more than one remote coupling joint prior to being purged at the space depot.

Strength Analysis

The basic load on the low melting alloy joint is a shear load produced by pressure area loads. Figure 40 defines the nomenclature that is used to determine the shear diameter length.

$$\sigma_s = \frac{P \pi D^2 K}{4 \pi D}$$

$$\sigma_s = \frac{K P D}{4 \ell}$$

or

$$\ell = \frac{K P D}{4 \sigma_s}$$

when

$K = 0.8$ (see Figure 40)

then

$$\ell = \frac{0.2 PD}{\sigma_s}$$

and

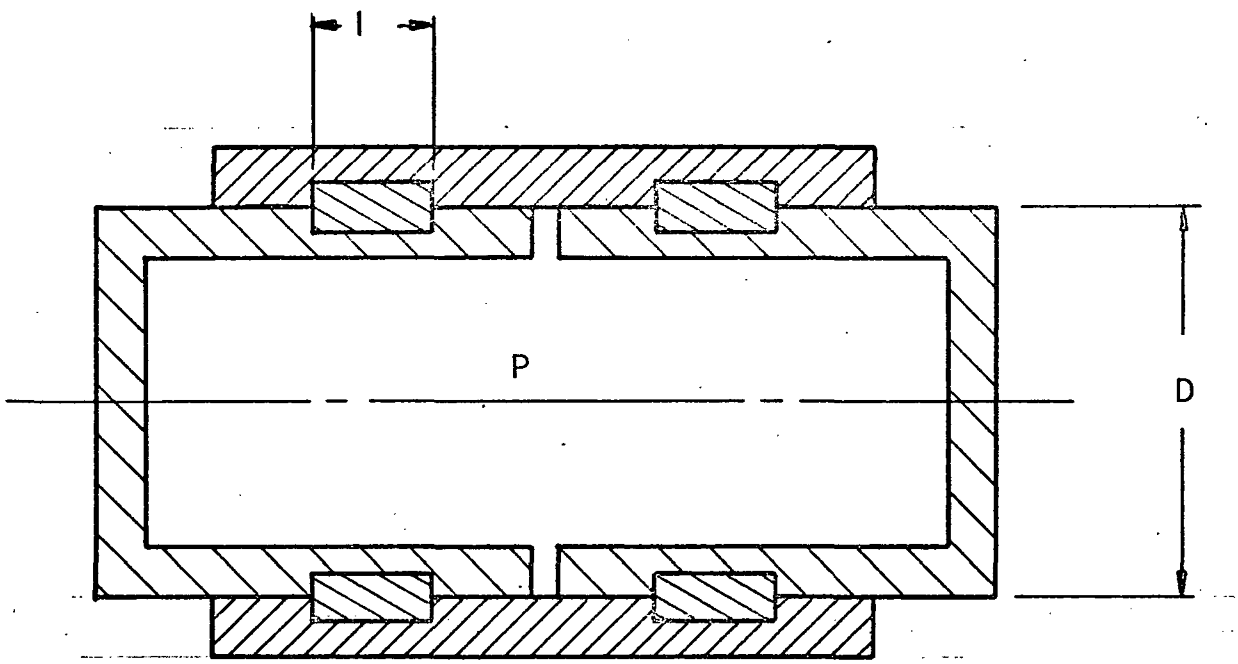
$$\sigma_s = 0.5 \sigma$$

σ = Safe Tension Load (reference ANSC drawing 1140005, Sheet 1), (Figure 39)

From this technique the shear dimensions, as shown on ANSC drawing 1140005, Sheet 3, were determined. Other dimensions are somewhat arbitrary for standardization. However, hoop stresses for critical sections were checked for safety at proof and burst pressures. It was not necessary to stress the shear area of the low melting alloy for proof and burst pressure since the ratio of the maximum load for five minutes to the safe load (sustained) is equal to 13.3:1 (reference ANSC drawing 1140005, Sheet 1) which is greater than the ratios of 1.5:1 and 2.5:1 normally used for proof and burst pressures respectively.

LOW MELTING ALLOY REMOTE COUPLING

ANALYTICAL SKETCH



l = SHEAR DIAMETER LENGTH

D = SHEAR DIAMETER

P = PRESSURE

σ_s = SHEAR STRESS

K = SAFETY FACTOR = 0.8 (ASSUMED)

FIGURE 40

Electrical Power Analysis

An electrical power versus heatup time analysis for the low melting alloy remote coupling is shown in Appendix Q. The analysis includes heatup times in space and on earth for three different power inputs for each of the four basic dash numbers shown on Figure 39. The power requirements and power applied durations appear to be reasonable for the low melting alloy remote coupling concept. This analysis is for the installation of a component and the making or coupling of a joint and seal. A power requirements analysis to uncouple a joint was not included since uncoupling power would be much less severe than during the coupling operation.

b. Thermal Interference Remote Coupling Joint and Seal

General

The thermal interference remote coupling and seal, as shown on ANSC drawing 1139922, was reviewed in order to assess its most efficient adaptability to the following ANSC design concepts:

New No.	Original No.	Nomenclature	New Line Size cm	New Line Size in.
1140001	1139893	Shutoff Valve	12.70	5.0
1140002	1139897	Shutoff Valve	20.32	8.0
1140007	1139924	Regulator	7.62	3.0

Concept drawings 1139893 and 1139924 reflect in-line ports. Originally, concept drawing 1139897 utilized right angle ports but has been redesigned to incorporate in-line ports for straight through flow and low pressure drop. The thermal interference joint was therefore redesigned so that the three valve and regulator concepts can be installed in-line as shown in Figure 41, ANSC drawing 1140006. The design requires both lines to be fixed or restrained from axial deflections so that the pressure area loads cannot produce axial loads on the thermal interference joint.

ANSC drawing 1140006 (Figure 41) consists of two sheets. Sheet contains general notes and the assembled design and materials for the selected sizes of couplings. Sheet 2 reflects tabulated dimensions for these coupling sizes.

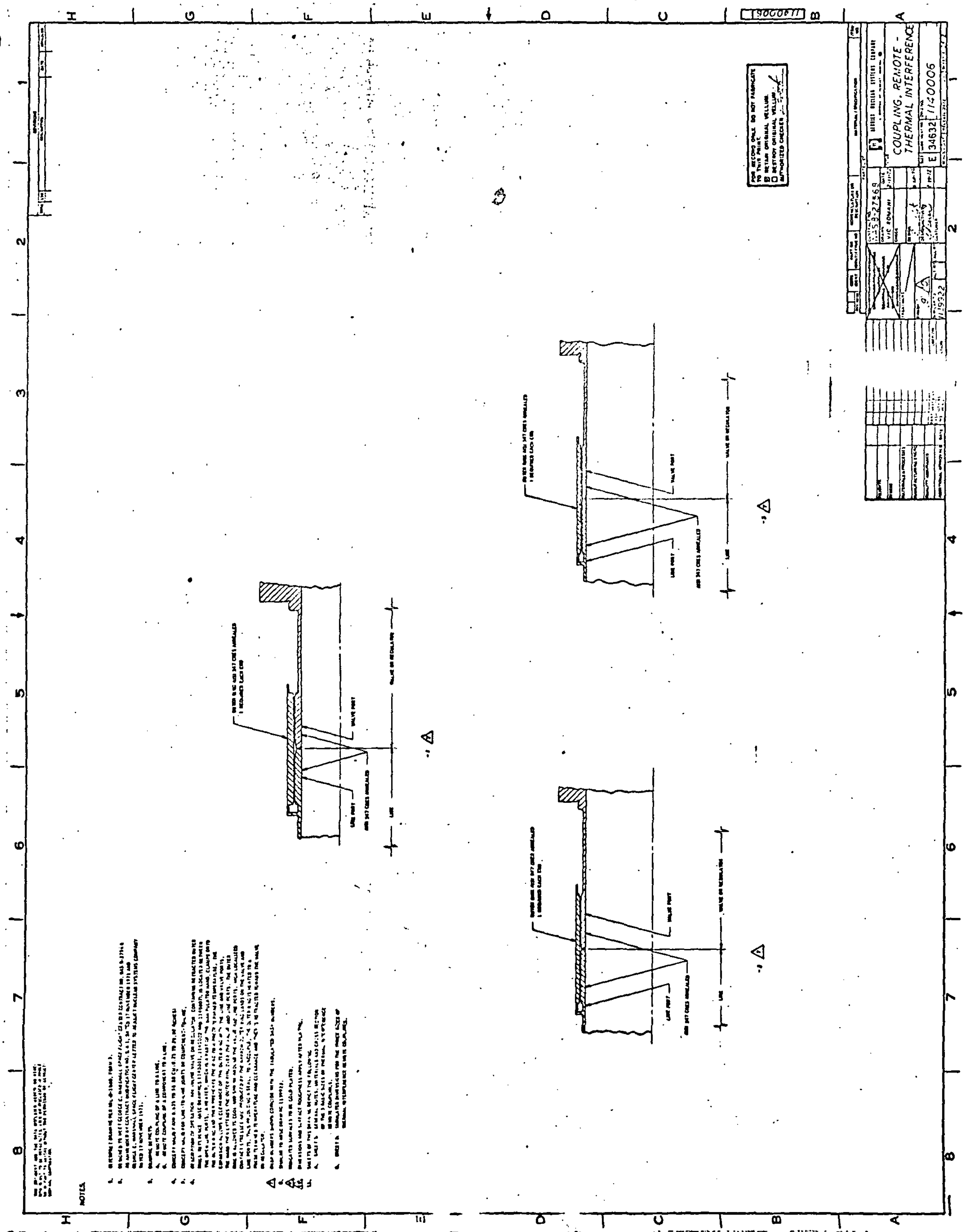


FIGURE 41
Page 1 of 2

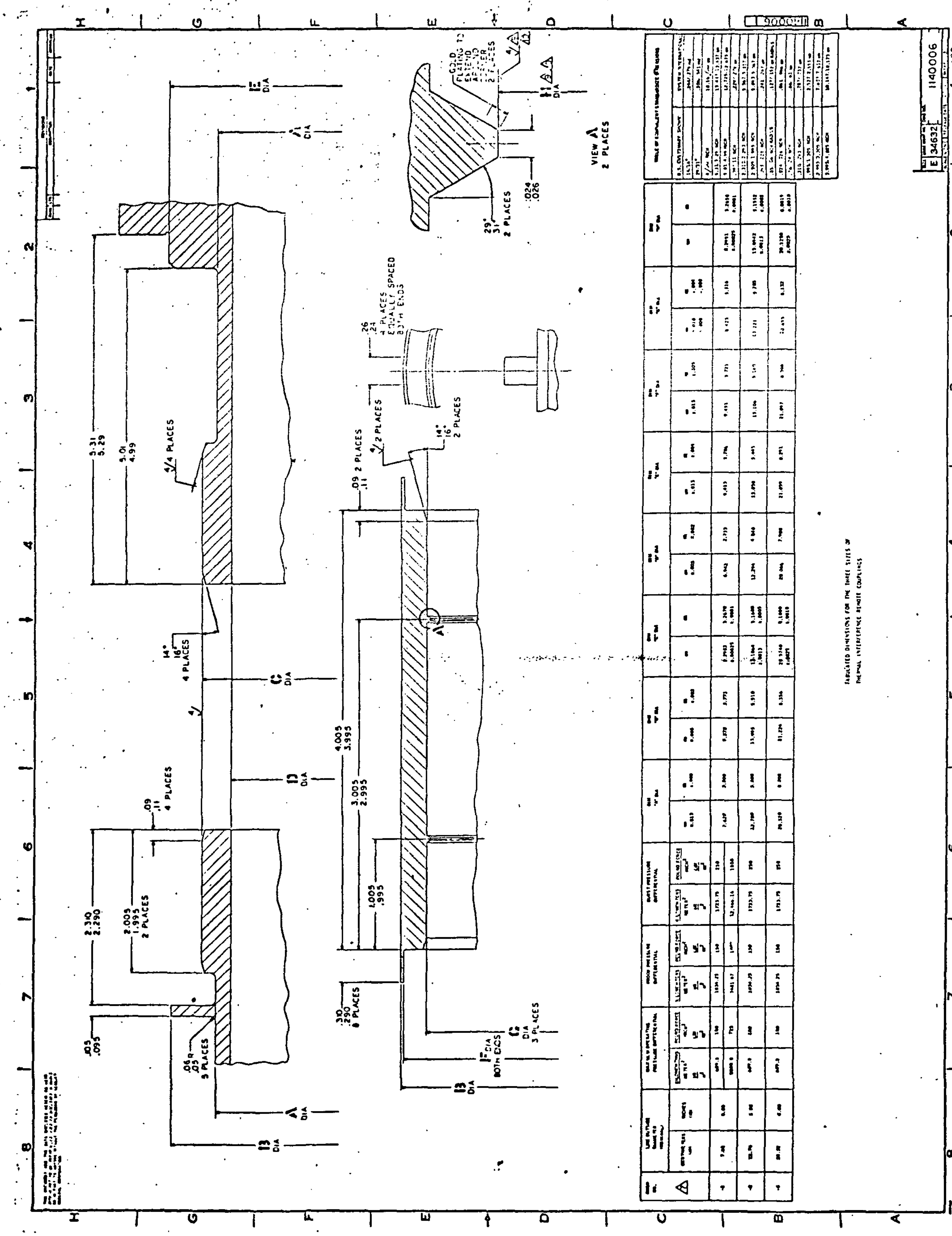


FIGURE 41
Page 2 of 2

The concept was changed slightly from the original concept shown on ANSC drawing 1139922. The outer rings, each containing two sealing lands on the outer ring I.D., are integral with the valves and regulator. One sealing land of each outer ring clamps on the O.D. of the line port and the other clamps on the O.D. of the valve or regulator port. The materials for the outer rings, line ports, and valve ports are AISI 347 CRES to provide the same coefficient of thermal expansion over a broad temperature range. The clamping or contact stress of the lands, when coupled is nominally $413.7 \times 10^6 \frac{\text{Newtons}}{\text{meter}^2}$ (60,000 lbf/in.^2). This stress is sufficient to provide asperity conformance on finely finished and round mating surfaces for zero leakage (1×10^{-7} sccs He). The I.D. of the outer ring lands are also gold plated to provide dissimilar surfaces to avoid cold welding in a space vacuum. The outer rings were made integral to the valves and regulator so that they can be inspected and replaced along with the component, if required, at the space depot. The manipulator hand, which contains the outer ring heater for both coupling and uncoupling the joint can also be inspected and/or repaired at the space depot.

Installation of In-Line Components

An in-line valve or regulator component, containing the retracted outer rings (reference Figures 28, 32, and 34), is located between the open line ports. The component can be indexed on the center of the line by the use of two tapered locating pins which mate with two tapered holes that are a part of the line or stage structure. As the tapered pins engage the tapered holes, an electrical connection is also made. Magnets can be added to the tapered pins or holes to positively locate the component and cancel any impact reaction forces.

The manipulator heater hand consists of a split band type heater system with each half attached to each portion of the manipulator hand. The manipulator heater hand would be similar to the hand shown on Sheet 4 of ANSC drawing 1140005, (Figure 39) with the following major exceptions. It would not contain a spring loaded reservoir, ports for flowing molten metal, and a porous catch receptacle. It would contain electrical circuitry and switches that accomplish the following operations through the use of on-board manipulator electrical power.

The manipulator heater hand indexes on the outer ring, and then is maneuvered to lightly clamp the outer ring. The clamping action would operate a switch that turns on electrical power to the heaters. The outer ring would then be heated to a pre-determined temperature that would generate a radial displacement for the required contact stress plus an additional radial clearance of 0.0635 cm (0.002 in.) for ease of extension. The manipulator hand would then be maneuvered to extend the outer ring along the axis of the line until the outer ring end segments contact the shoulder on the line. At this time the manipulator hand is expanded which unclamps the outer ring and shuts off electrical power to the heaters. The manipulator hand can then be maneuvered away from the joint. The outer ring then cools, allowing the sealing lands to shrink on the outside diameters of the valve and line ports, thus producing the desired contact stresses.

Removal of In-Line Components

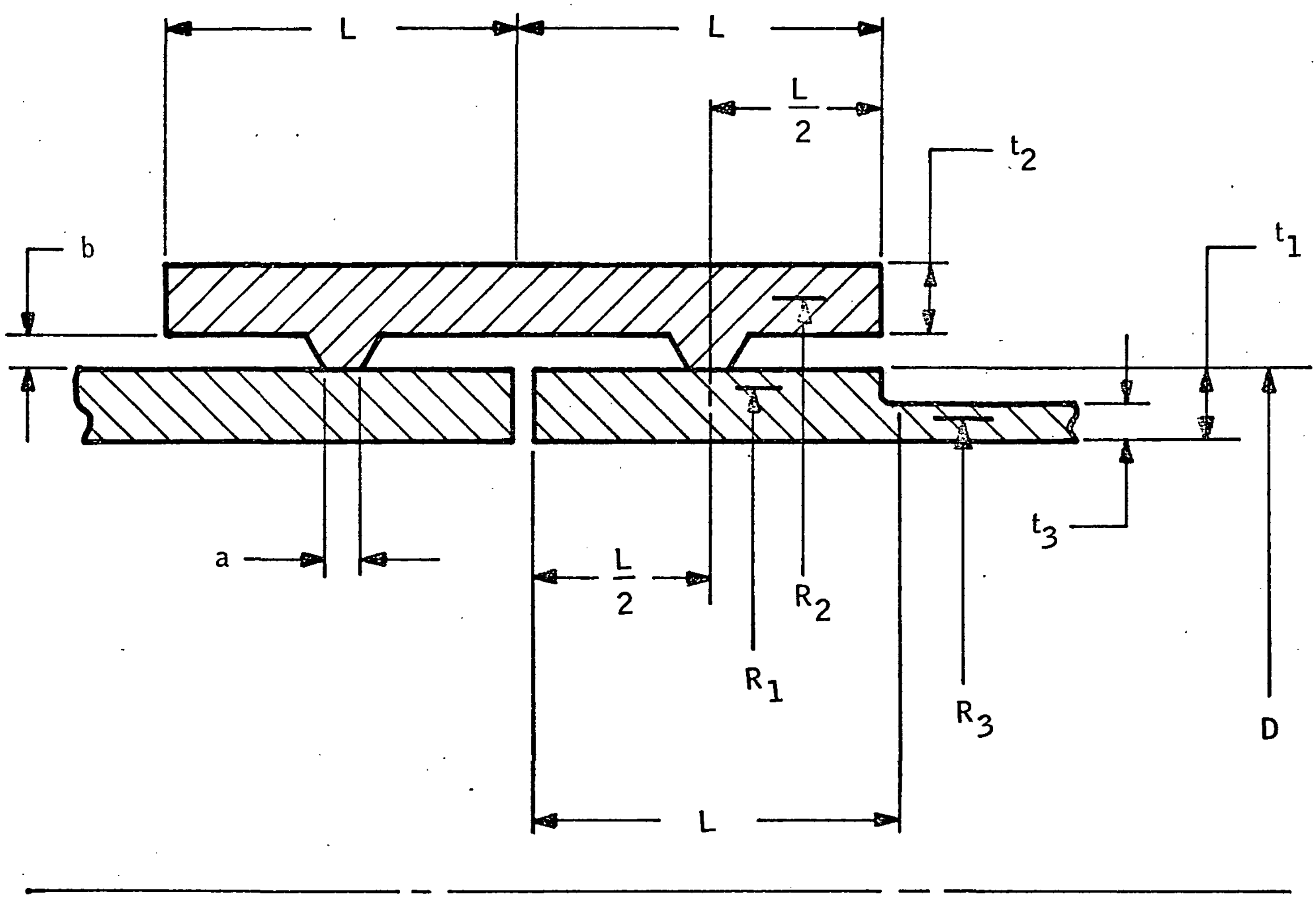
The uncoupling of an in-line component is similar to the coupling of an in-line component described above. The manipulator heater hand is identical to the one used for coupling and in-line joint.

The manipulator heater hand indexes on the outer ring, and then is maneuvered to lightly clamp the outer ring. This clamping action would operate a switch that turns on electrical power to the heaters. The outer ring would then be heated to a predetermined temperature that would generate a radial displacement, to loosen the shrink fit, plus producing an additional radial clearance. The manipulator hand would then be maneuvered to retract the outer ring along the axis of the line until the outer ring end segments contact the shoulder on the valve or component.

At this time the manipulator hand is expanded, which unclamps the outer ring and shuts off electrical power to the heaters. The manipulator hand can then be maneuvered away from the uncoupled joint. The valve or regulator component is then removed to the space depot after both joints are uncoupled.

Stress and Dimensional Analysis

A contact stress of 413,700 Kilo Newtons/meter^2 (60,000 lbf/in.^{2}) was selected as the nominal interface stress between the small internal lands on the outer ring and the outside diameter of the two tube ends. Using the Case 1 and Case 8 formulas from the 1954 edition of "Formulas for Stress and Strain" by Roark, wall thickness, radial displacements and hoop stresses were determined. These dimensions are shown on Sheet 2 of ANSC drawing 1140006. (Figure 41). The formulas are itemized as follows with respect to Figure 42.



ALL MATERIALS ARE AISI 347 CRES ANNEALED
b = 0.0635 cm (0.025 IN) ALL THREE DASH NO.'s

FIGURE 42

$$\Delta R = \frac{\pm FR^2 \lambda C}{Et} \qquad \text{Eqn 1}$$

$$\sigma_1 = \frac{\pm FR \lambda}{2t} \qquad \text{Eqn 2}$$

$$F = P_1 a \qquad \text{Eqn 3}$$

$$\Delta T = \frac{\Delta R_T}{R\alpha} \qquad \text{Eqn 4}$$

$$\sigma_2 = \frac{P_2 R}{t} \qquad \text{Eqn 5}$$

- Compression or inward direction

\+ Tension or outward direction

ΔR_1 = Inward radial displacement for nominal contact stress

ΔR_2 = Outward radial displacement for nominal contact stress

F = Load per unit of circumference on D diameter

R_1 = Mean circumferential radius of line ports

R_2 = Mean circumferential radius of outer ring

R_3 = Mean circumferential radius of tube

E = The modulus of elasticity of AISI 347 CRES = $206.85 \times 10^9 \frac{\text{Newton's}}{\text{Meter}^3}$ (30.0×10^6 lbf/in.^{2})

t_1 = Wall thickness of line port

t_2 = Wall thickness of outer ring

t_3 = Wall thickness of line

P_1 = Contact stress between outer ring lands and outside diameter of line ports = 413,700 Kilo Newton's/Meter (60,000 $lbf/in.^2$)

a = Land width = 0.0635 cm (0.025 in.)

$\Delta R_T = \Delta R_1 + \Delta R_2 + 0.00508 \text{ cm} = \Delta R_1 + \Delta R_2 + 0.002 \text{ in.}$

Where

0.00508 cm (0.002 in.) = Radial extension or retraction clearance of outer ring over line ports.

ΔT = Temperature increase to either install or retract the outer ring over the line ports = °K (°F)

α = Coefficient of thermal expansion of AISI 347 CRES from = 14.4×10^6 cm/cm/°K (8.0×10^6 in./in./°F)

σ_2 = Hoop tension on tube wall due to internal pressure P_2

P_2 = Internal pressure = $689.5 \times 10^3 \frac{\text{Newton's}}{\text{Meter}^2}$ (100 $lbf/in.^2$) for -1, -2 and -3 sizes and also $5000.0 \times 10^3 \frac{\text{Newton's}}{\text{Meter}^2}$ (725 $lbf/in.^2$) for -1 size.

$$\lambda = \left[\frac{3\,(+\,r^2)}{R^2\ t^2} \right]^{1/4}$$

Where

r = Poisson's ratio (assumed = 0.3 for AISI 347 CRES

And

$\lambda = 1.00 \text{ in.}^{-1}$ for $Rt = 0.4 \text{ in.}^2$ Eqn 6

For short cylinders of $\lambda L = 2.00$

$L = 2.00$

And C = Correction factor for $\Delta R = 1.2$.

U. S. customary units were used for the basic calculations and the results were converted to SI units when applicable.

The R_1 of the line and valve ports was established as 50 percent of the line outside diameter with $t = 0.4/R$ inches.

F was computed using Equation 3 and Figure 42.

$F = 10.342 \times 10^6 \frac{\text{Newton's}}{\text{Meter}}$ (1500 lbf/in.).

ΔR_1, σ_1 and t_1 were established using Equations 1, 2 and 6 and are shown in Table VI.

TABLE VI

VALVE & LINE PORTS PARAMETERS

Dash No.	Line O.D.		R_1		t_1		ΔR_1 (-)		σ_1 (-)
	cm	in.	cm	in.	cm	in.	cm	in.	Newton's/Meter^2 / lbf/in.^{2}
-1	7.62	3.00	3.81	1.50	0.678	0.267	0.0012852	0.000506	29.052×10^6 / 4213.50
-2	12.70	5.00	6.35	2.50	0.406	0.160	0.006187	0.002436	80.800×10^6 / 11718.75
-3	20.32	8.00	10.16	4.00	0.254	0.100	0.0243840	0.009600	$206.85.00 \times 10^6$ / 30,000.00

The inside radius (I.R.) of the outer ring was then established by adding .0635 cm (0.025 in.) to the outside radius of the line and valve ports. Using Eqn 6,

$$R_2 \, t_2 = 0.4 \text{ in.}^2$$

Or

$$t_2 = \frac{0.4}{R_2}$$

Then

$$R_2 = \text{I.R.} + \frac{t_2}{2}$$

$$R_2 = \text{I.R.} + \frac{0.2}{R}$$

$$R_2^{\,2} - (\text{I.R.})\, R - 0.2 = 0.$$

R_2 for this equation is solved quadradically, and is shown in Table VII. ΔR_2, σ_1, and t_2 were established using Equations 1, 2 and 6 and are shown in Table VII.

TABLE VII

OUTER RING PARAMETERS

Dash No.	Line O.D.		R_2		t_2		ΔR_2		σ_1 Newtons/Meter^2 / lbf/in.^{2}
	cm	in.	cm	in.	cm	in.	cm	in.	
-1	7.62	3.00	4.500	1.772	0.5766	0.227	0.00210	0.00083	40.37×10^6 / 5854.6
-2	12.70	5.00	6.807	2.680	0.3810	0.150	0.00723	0.00287	92.39×10^6 / 13,400.0
-3		8.00	10.485	4.128	0.2540	0.100	0.02164	0.00852	213.47×10^6 / 30,960.0

$$\Delta R_T = \Delta R_1 + \Delta R_2 + 0.00508 \text{ cm}$$

$$= \Delta R_1 + \Delta R_2 + 0.002 \text{ in.}$$

ΔR_T of the outer ring is tabulated based on the above relationship and ΔT is found by use of equation 4 for extension and retraction of the outer ring over the line and valve ports with 0.0635 cm (0.025 in.) radial clearance. Values are based on α data from room temperature to 589°K (600°F). This data is shown in Table VIII.

TABLE VIII

OUTER RING PARAMETERS

Dash No.	Line O.D.		R_2		ΔR_T		ΔT	
	cm	in.	cm	in.	cm	in.	°K	°F
-1	7.62	3.00	4.500	1.772	0.0081	0.0032	492.0	225.7
-2	12.70	5.00	6.807	2.680	0.0173	0.0068	431.7	317.1
-3	20.32	8.00	10.485	4.128	0.0511	0.0201	593.7	608.6

Hoop stress was checked for the outer ring and the line by using equation 5 and is shown in Table IX for the various pressures involved.

TABLE IX

OUTER RING HOOP STRESS

Dash No.	Line O.D.		Operating Pressure Differential	Outer Ring			Line		
				R_2	t_2	σ_2	R_3	t_3	σ_2
			Newtons/Meter^2	cm	cm	Newtons/Meter^2	cm	cm	Newtons/Meter^2
	cm	in.	lbf/in.^{2}	in.	in.	lbf/in.^{2}	in.	in.	lbf/in.^{2}
-1	7.62	3.00	689.5×10^3	4.500	0.5766	5.38×10^6	3.980	0.338	8.124×10^6
			100	1.772	0.227	780.6	1.567	0.133	1,178.2
-1	7.62	3.00	5000.0×10^3	4.500	0.5766	39.02×10^6	3.980	0.338	58.897×10^6
			725	1.772	0.227	5659.0	1.567	0.133	8,542.0
-2	12.70	5.00	689.5×10^3	6.807	0.3810	12.32×10^6	6.452	0.203	$21,892 \times 10^6$
			100	2.680	0.150	1786.7	2.540	0.080	3,175.0
-3	20.32	8.00	689.5×10^3	10.485	0.2540	28.5×10^6	10.224	0.127	55.505×10^6
			100	4.128	0.100	4128.0	4.025	0.050	8,050.0

All compressive stresses are within the allowable F_{cy} of AISI 347 CRES annealed of $372.3 \times 10^6 \frac{\text{Newtons}}{\text{meter}^2}$ (54,000 lbf/in.^{2}) The hoop stresses are well within the allowable F_{ty} of $206.8 \times 10^6 \frac{\text{Newtons}}{\text{meter}^2}$ (30,000 lbf/in.^{2}).

Electrical Power Analysis

An electrical power versus heatup time analysis for the low melting alloy remote coupling is shown in Appendix R. The analysis includes heatup times in space and on earth for three different power inputs for each of the three basic dash numbers shown on Figure 39. The power requirements and the power applied durations appear to be reasonable for the thermal interference remote coupling. This analysis is for the removal of a component and the uncoupling of a joint and seal. A power requirements analysis to couple a joint was not included since coupling power would be less severe than during the uncoupling operation.

III. CONCLUSIONS

In general, it can be concluded that each of the designs if feasible dependent upon the requirements of the application. The conclusions pertaining to each of the six designs is listed separately in this section.

A. VALVES

1. MHD Valve Concept

Based on the analysis presented, the magnetohydrodynamic (MHD) valve concept is feasible; however, large amounts of power will be necessary (2.2 MW) to operate the valve due to the low conductive gas.

It is possible, however, that the initiating power may be reduced once ionization of the gas has been achieved, since it has been demonstrated experimentally that the ionization of a gas can be maintained with a voltage considerably lower than the ionization potential.

The amount of power that can be reduced in this case must be determined experimentally.

It should be noted that the concept is sound, though, with better conducting fluids the power requirements would be within more reasonable levels. For example, helium would be expected to have a stronger interaction with the MHD forces since He has a lower ionization potential than H_2.

In addition, this concept would be particularly attractive for liquid metal, such as sodium application, since the fluid conductivity in such a case is very high.

It should be noted further, that the concept presented is limited to small size valves 1/4" to 1". However, it can be used as a skirt seal for a balanced poppet valve to handle large flows, by using the MHD principle to pump out the fluid in the poppet cavity. Thus, a pressure differential across the poppet causes the poppet to open when the MHD portion is energized.

2. Electromechanically Actuated Inline Poppet Valve with High Pressure Energized Poppet Seal

Based on the analysis presented, the seal design present is very feasible for controlling very low poppet seat leakages for LH_2 shutoff valves. Although the design can be used for GH_2 service, it is limited powerwise and materialwise for the temperature rises necessary to generate poppet to seat unit loading. The design also requires an anti-backdrivable system to prevent the poppet from moving with the seat when the poppet is in the closed position with the heater activated. Also based on Appendix O, very high LH_2 boil off rates occur upstream of the valve seat.

3. Electromechanically Actuated Inline Poppet Valve with Liquid Metal Seals

ANSC concludes that the liquid metal seal concepts are feasible for both the optically flat poppet seal and the poppet skirt seal. However, the poppet skirt dynamic seal will be far more troublesome to develop than the optically flat poppet seal since radial clearances, surface finishes, concentricities, etc. are by far more difficult to obtain and control for round mated parts than optically flat surfaces in intimate contact. Thermal distortions are also a consideration. Since the surface finish is important to the sealing effectiveness of liquid metals, reactions between the liquid metal and the wetted surface which would produce roughening are of concern. Roughening could occur in the form of etching, pitting, or material buildup through amalgamation and alloying. The Aluminum Company of America summarizes the compatibilities of gallium as readily alloying with all metals as elevated temperatures and forming alloys with tin, zinc, cadmium, aluminum, silver, magnesium, copper, and others at room temperature. Material properties of the substrate materials can also be reduced by liquid metal embrittlement. In addition to the alteration of mechanical properties, the possibility of liquid metals in contact between two substrate materials may result in welding or bond formation between the solid components and may be a serious consideration.

B. THE PROGRAMED PRESSURE AND FLOW CONTROL REGULATOR

ANSC 1140007 regulator has the capability of programing RNS tank pressure by metering turbine discharge bleed gas from state point 36 of the NERVA engine in order to maintain minimum required engine NPSP. This technique results in large increases in RNS payload capability because it minimizes propellant losses by minimizing the venting requirement of tank pressurant.

In addition, the regulator design features complete shutoff on command. This action isolates the pressurization system from the tank and precludes the necessity of a separate shutoff valve in series with the regulator. During coast periods, when the NERVA engine is not operating, it features reverse flow shutoff, through the use of an auxiliary shutoff feature that prevents tank leakage or pressure loss from the tank through the NERVA engine. By incorporating a separate controller, it could be switched to this controller and act as a tank pressure relief or vent valve during coast periods by relieving excessive tank pressure through the NERVA engine. Thus the regulator could be utilized for three to four functions greatly simplifying the overall system and enhancing reliability. The regulator can be tested and checked out in space for proper operation, without the use of any kind of costly gaseous flow by the use of a simulator controller.

C. REMOTE COUPLINGS

Of the two remote couplings studied, the low melting alloy which provides a structural, as well as a sealing joint, appears the most promising. This is especially true as the line size and/or pressure increases. The joint would be a very good candidate for a large joint such as a pressure vessel end cover or a joint where a marmom type clamp performs the structural function and the alloy is used as a seal.

1. Low Melting Alloy Remote Coupling

The low melting alloy remote coupling is a workable and proven coupling and seal approach that should yield zero leakage (1×10^{-7} SCCS He) at low pressures for long life in a space vacuum and a nuclear environment. Although somewhat bulky in appearance, the concept has the capability of weight and size refinements during the final detail design phase when a more refined analysis can be utilized. The flight or stage hardware for the coupling is simple and reliable, leaving the more complex heater injection and heater ejection systems as part of specially designed manipulator hands that could be serviced at the space depot. The remote coupling design should produce reliabilities similar to the reliabilities of welded and brazed joints and couplings.

2. Thermal Interference Remote Coupling

The thermal interference remote coupling is a workable approach, based on a localized shrink fit that produces contact stresses of sufficient magnitudes to achieve zero leakage (1×10^7 SCCS H_e) at low pressures for long life in a space vacuum and in a nuclear environment due to its all metal construction. The general configurations can be somewhat varied by manipulating the stress and deflection formulas, contact stresses and heater powers. The joint, as it is now shown, requires axial restraint of the two line ports to counteract pressure area loads to prevent slippage of the shrink fit sealing points.

IV. RECOMMENDATIONS FOR PHASE III

Each of the designs studied is feasible depending upon the requirements of the application. Since the designs presented represent "radical" concepts rather than "state-of-the-art" it is generally recommended that a fabrication/test program be conducted to quantify the variable parameters and to establish feasibility of each of the six designs. The recommendations pertaining to each of the six designs is listed separately in this section.

A. VALVES

1. Magnetohydodynamic Valve Concept

The valve should be built and tested. The Phase III design analysis should include a parametric computer analysis to determine and define the optimum combination of valve dimensions, magnetic flux, current and voltage to result in minimum power. Phase III should also consider the alternate design approach discussed in Appendix N. Tests should be conducted with GH_2, GHe, GN_2, Hg and H_2O at various levels of pressures, temperature, voltage, and currents.

The tests should also be conducted with gases that are seeded which will increase the conductivity to usable levels and reasonable temperatures. Examples of good seed material are cesium, potassium, sodium and lithium, and mercury vapor.

The tests should also be conducted by varying the gap (g) distance to establish optimum gap distances for a given power input. Tests should include, but not be limited to, the following for the various gases, pressures and temperatures noted above.

- Pressure drop
- Internal leakage
- Power requirements to determine exact glow discharge region
- Regulation capability.

2. <u>Electromechanically Actuated Inline Poppet Valve with High Pressure Energized Poppet Seal</u>

A simple tester unit should be fabricated and evaluated. Experimental work should include techniques for matching the poppet and seat conical surfaces and determining surface finish requirements. Different conical angles and conical seating areas should be tested in conjunction with annular cavity cross sectional areas and pressures for the most effective sealing.

The circulation system that fills the annular cavity with LH_2 should be evaluated and modified, as required, to ensure that the annular cavity is always full of LN_2 when the poppet is seated. The effectiveness of the total pressure pickup circulation system should be determined with various poppet strokes.

Thermal gradients should be determined by LH_2 tests so that heater power requirements can be coordinated with the power analysis, and to support the thermal analysis. Also based on the thermal analysis shown in Appendix O, the trapped volume in the annular cavity must be thermally insulated from the valve seat or must be located at a large distance from the seat in order to decrease the LH_2 boil off rate upstream of the valve seat.

3. <u>Electromechanically Actuated Inline Poppet Valve with Liquid Metal Seals</u>

It is recommended that further research be performed in liquid metal alloy technology in the following areas:

- Lower melting temperature alloys.
- Compatibility of liquid metal alloys with various structural materials and service fluids.
- Problems of assembling and packaging liquid metal valving elements for use on RNS missions.
- Evaporation rates of liquid metal alloys in a space environment.

It is also recommended that the liquid metal poppet skirt seal development during Phase III be secondary to the main poppet optically flat liquid metal seal development. The liquid metal skirt seal can be eliminated by utilizing an unbalanced unidirectional valve design similar to the one described in Section II.B.1.

ANSC strongly recommends a review and extension of the research work performed in Reference 14, Volume 1 of Appendix D.

B. THE PROGRAMMED PRESSURE OR FLOW CONTROL REGULATOR

ANSC recommends that 1140007 regulator be pursued into final detail design phase with hardware testing. The electromechanical actuator and the associated electronics module could be available as hardware with minor modifications from another NASA program. Major hardware items to be fabricated could include primarily the mechanical portion of the regulator. Major test parameters would include the following items:

1. Coefficient of discharge versus shear plug position using sonic nitrogen or air flow at room temperature.

2. Regulator torque versus shear plug position using sonic nitrogen or air flow followed by torque determination using sonic hydrogen flow from temperatures slightly above the liquid phase to liquid nitrogen temperatures. This would allow a reasonable correlation between the shear plug coefficient of friction with hydrogen to that of air or nitrogen.

3. Helium leakage tests over the temperature range and at various pressures. This could be accomplished during a life cycle test with periodic leakage tests.

4. The determination of regulator position accuracy and response with various controller input command signals.

C. REMOTE COUPLINGS

1. Low Melting Alloy Remote Coupling

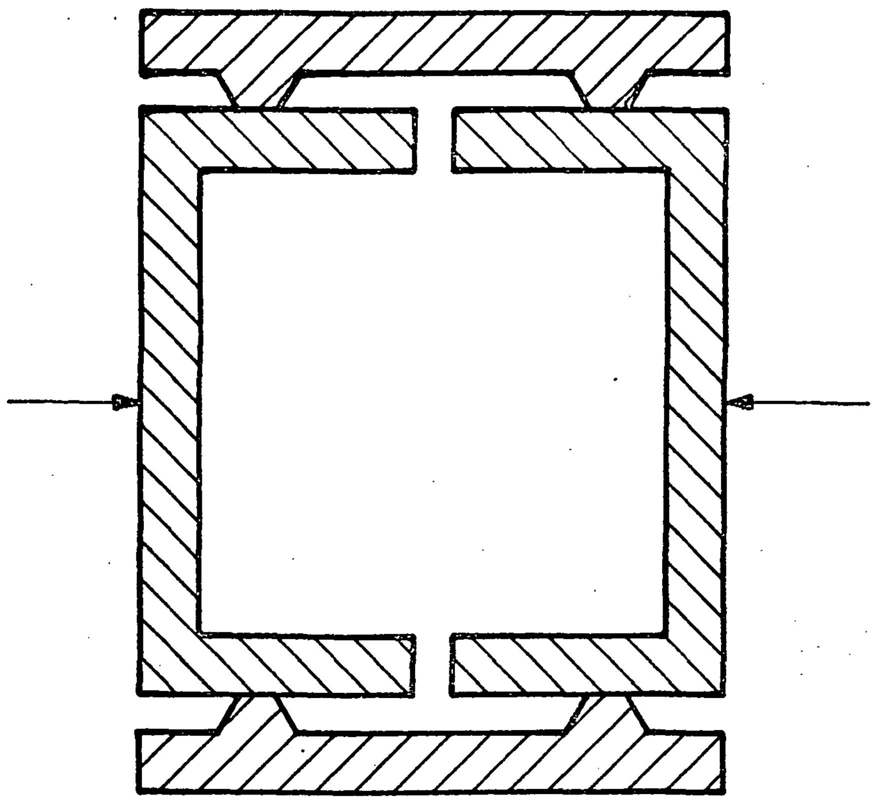
The low melting alloy remote coupling design lends itself to a simple hardware tester similar to the design shown in Figure 42. It could be developed and tested as a joint independent of regulator or shutoff valve development.

Little information is available on the material properties of the 55.5% Bismuth/44.5% lead low melting alloy at temperatures other than room temperature. ANSC recommends the initiation of a test program to determine shear, creep and thermal expansion properties in the temperature range of 20.6°K (-423°F) to 366.7°K (200°F). This effort could be followed by designing and developing a refined tester design similar to that shown in Figure 42.

2. Thermal Interference Remote Coupling

The thermal interference coupling design lends itself to a simple hardware tester similar to the design shown in Figure 43. It could be tested as a joint independent of regulator or valve development.

Contact stresses could be varied in order to determine threshold stresses for various leak rates per unit of sealing circumference for various fluids. Surface finishes can also be varied in order to determine their affect on leak rates.



SIMPLE HARDWARE TESTER

FIGURE 43

APPENDIX A

VALVE DESIGN GOALS

VALVE DESIGN GOALS

OPERATIONAL REQUIREMENTS

Nominal Line Size (meters)	.0064 to .508
(inches)	1/4 to 20
Valve Type	TBD
Pressures	
Maximum system working, $(\frac{\text{Newton's}}{\text{meter}^2}$ absolute)	6.895×10^5
(psia)	100
Minimum system working, $(\frac{\text{Newton's}}{\text{meter}^2}$ absolute)	6.895×10^{-5}
(psia)	1×10^{-8}
Maximum differential, $(\frac{\text{Newton's}}{\text{meter}^2}$ absolute)	6.895×10^5
(psid)	100
Proof, $(\frac{\text{Newton's}}{\text{meter}^2}$ absolute)	10.343×10^5
(psia)	150
Burst, $(\frac{\text{Newton's}}{\text{meter}^2}$ absolute)	13.79×10^5
(psia)	200
Flow resistance coefficient (K)(7)	TBD
Temperature Range, (°K)	366.48 to 20.37
(°F)	+200° to -423°
Internal Leakage, SCCS He	1×10^{-7} (2)
External Leakage, SCCS He	1×10^{-7} (2)
Internal Leakage, SCCS He	1×10^{-3} (4)

OPERATIONAL REQUIREMENTS (cont'd)

Media	LH_2m GH_2m GN_2 & GHe	
Purge Pressurization & Checkout Media(3)	He & GN_2	
Maximum Flow Rate(3) (LH_2)	Equivalent to 18.23 meter/sec Equivalent to 60 ft/sec	
Fluid Density, ($\frac{\text{Kilograms}}{\text{meters}^3}$) ($LH_2$)	70.00	
(lb/ft^3)	4.37	
Maximum Flow Rate (GH_2)(7)	TBD	
Storage Life, years	10	
Operating Life, cycles(4)	sizes 1" thru 20"	sizes 1/4" thru 3/4"
Grand Total	242	1000
(1) Ambient temperature and pressure		
a. Acceptance	38	160
b. Pre-installation checkout	12	50
c. Pre-launch checkout	50	200
Total	100	410
(2) Cryogenic		
a. Acceptance	19	70
b. Pre-installation	6	25
Total	25	95
(3) Space Operation		
a. During engien operation	72	300
b. Checkout at space station	45	195
Total	117	495

OPERATIONAL REQUIREMENTS (cont'd)

Nuclear Environment(5)	3×10^{6} (6)	Gamma KERMA Rate: Rads (carbon)/hr.
	3×10^{11}	Fast Neutron Flux: n/cm^2-sec (En > 0.9 MeV)
Opening Time, secs	TBD	
Closing Time, secs	TBD	
Mode of Actuation	TBD	

(7) Flow Resistance Coefficient, "K", is defined by the equation:

$$K = \frac{2g_c \; \rho A^2 \;\; P}{\dot{w}^2}$$

where: g_c = Dimensional conversion factor = 386.4 $\frac{\text{lbm}}{\text{lbf}} \frac{\text{in.}}{\text{sec}^2}$ or $\frac{m}{S^2}$

ρ = Mass density of fluid at inlet flow conditions lbm/in.^{3} or $\frac{kg}{(m/S^2)/m^2}$

A = Reference area, in.^{2} or m^2

ΔP = Pressure loss through the valve, lbf/in.^{2} or $\frac{kg}{m^2}$

$\dot{w}$ = Fluid weight flow rate through the valve, lbm/sec or $\frac{kg}{S}$

FUNCTIONAL REQUIREMENTS

1. Valves shall not be overly sensitive to vibration or acceleration in any axis.

   Launch-regulator not functioning - no structural failure

	(8) X Axis	Y Axis	Z Axis	Time
Sinusoidal vibration	±3.0 g 3 to 35 Hz	±4.5 g 0.1 to 15 Hz	±4.5 g 0.1 to 15 Hz	5 min.
Acceleration	5.2 g	0	0	(9) 5 min.

OPERATIONAL REQUIREMENTS (cont'd)

Space operation - regulator functioning

	(8) X Axis	Y Axis	Z Axis	Time
Random vibration	TBD	TBD	TBD	60 min.
Acceleration	1.0 g	0.8 g	0.5 g	60 min.

2. Investigate replacement of the valve by remote, in space, handling equipment.
3. Investigate replacement of the valve element or actuator by remote, in space, handling equipment.
4. Provide valve position indication.

---

(1) May discharge to space vacuum or be exposed internally to space vacuum.

(2) Per inch of sealing diameter.

(3) Interpretation or assumption by ANSC.

(4) Based upon 1 cycle per burn.

(5) At engine to stage interface per ANSC Memo N4340:6397M, E. A. Warman to A. D. Cornell, dated 22 January 1971, Subj: "Perturbed NERVA Engine Flux and Isokerma Rate Maps". See LMSC-A984555, Final Report, Volume VIII. Dose rates are also specified.

(6) Ten hours run time for total dosage.

(7) Gas flow rates will be determined based on temperatures, pressure ratios, density and line sizes. Values to be established after R.N.S. requirements have been determined.

(8) Longitudinal axis of R.N.S.

(9) Increases from 1.0 g to 5.2 g in five minutes.

APPENDIX B

GASEOUS PRESSURE REGULATOR

DESIGN GOALS

GASEOUS PRESSURE REGULATOR DESIGN GOALS[1]

OPERATIONAL REQUIREMENTS	FUNCTION: Tank Pressurant Regulator — Multiple Tank	Single Tank
Approximate Line Sizes (meters)	6.033×10^{-2}	9.525×10^{-2}
(inches)[4]	2.375	3.750
Regulator Type[1]		Pressure Reducing
Regulator Performance[2]	See Figure 1	See Figure 2
Maximum System Inlet Pressure ($\frac{Newtons}{meter^2}$ absolute)	4.985×10^6	4.985×10^6
(psia)[3]	723	723
Minimum System Inlet Pressure ($\frac{Newtons}{meter^2}$ absolute)	2.199×10^6	2.199×10^6
(psia)[3]	319	319
Inlet Proof Pressure ($\frac{Newtons}{meter^2}$ absolute)	7.481×10^6	7.481×10^6
(psia)[12]	1085	1085
Inlet Burst Pressure ($\frac{Newtons}{meter^2}$ absolute)	12.466×10^6	12.466×10^6
(psia)[12]	1808	1808
Outlet Proof Pressure ($\frac{Newtons}{meter^2}$ absolute)	3.103×10^5	3.103×10^5
(psia)[13]	45	45
Outlet Burst Pressure ($\frac{Newtons}{meter^2}$ absolute)	5.171×10^5	5.171×10^5
(psia)[13]	75	75
Minimum non-operational Pressure ($\frac{Newtons}{meter^2}$ absolute)		6.895×10^{-5}
(psia)		1×10^{-8}
Flow Rate ($\frac{Kilogram}{second}$)	0.4536	1.8144
(lbm/sec)[3][4], Minimum	1.00	4.00
Lockup Leakage, SCCS He[5]	TBD	TBD
External Leakage, SCCS He[6]	1×10^{-7}	

Ambient Pressure Range (°K)	1.014×10^5 to 5.895×10^{-5}	
(psia)	$14.7 \times 1 \times 10^{-8}$	
Ambient Temperature Range (°K)	20.37 to 366.48	
(°F)	-423 to +200	
Inlet and Outlet Media and Temperatures (1)		
GH_2 for service (°K)	20.37 to 366.48	33.15 to 366.48
(°F)	-423 to +200	-400 to +200
GN_2 for purge media (°K)	88.70 to 366.48	88.70 to 366.48
(°F)	-300 to +200	-300 to +200
GHe for purge media (°K)	20.37 to 366.48	20.37 to 366.48
(°F)	-423 to +200	-423 to +200
Storage Life (seconds)	3.1536×10^8	
(years)	10	
Operational Life (seconds)	9.4608×10^7	
(years)	3	
Operating Life, cycles (7)		
Modulating - grand total	253,000	253,000
Lockup - grand total	110	110
1. Ambient temperature and pressure		
a. Acceptance		
Modulating	1,000	1,000
Lockup	10	10
b. Pre-installation checkout		
Modulating	500	500
Lockup	5	5
c. Pre-launch checkout		
Modulating	200	200
Lockup	2	2

2. Cryogenic		
a. Acceptance		
Modulating	500	500
b. Pre-installation checkout		
Modulating	300	300
Lockup	3	3
3. Space Operation		
a. During engine operation		
Modulating	250,000	250,000
Lockup	80	80
b. Checkout at space station		
Modulating	500	500
Lockup	5	5

Nuclear Environment[8]	3×10^6 [9]	GAMMA KERMA Rate: RADS (carbon)/hour.
	3×10^{11}	Fast Neutron Flux n/cm^2-sec (En > 0.9 MeV)

FUNCTIONAL REQUIREMENTS

1. Valves shall not be overly sensitive to vibration or acceleration in any axis.

Launch-regulator not functioning - no structural failure

	[10] X Axis	Y Axis	Z Axis	Time
Sinusoidal vibration	± 3.0 g	±4.5 g	±4.5 g	300 seconds (5 min.)
	3 to 35 Hz	0.1 to 15 Hz	0.1 to 15 Hz	
Acceleration	5.2 g	0	0	300 seconds [11] (5 min)
Space operation - regulator functioning				
Random vibration	TBD	TBD	TBD	3600 seconds (60 min.)
Acceleration	1.0 g	0.8 g	0.5 g	

2. Investigate replacement of the regulator by remote, in space, handling equipment.

3. Investigate replacement of regulator working remote, in space, handling equipment.

---

(1) Gaseous pressure reducing regulators are the most common type and are used extensively on all space booster and space vehicle stages. Typical applications are for propellant tank pressurizations and pressure regulation for reaction control systems.

(2) Typical pressure regulator performance analogues are shown in Figures 1 and 2. These analogues describe the major performance considerations for a stable pressure reducing regulator.

(3) The maximum inlet pressure at the regulator was based on 100 psia* and was a requirement of NASA MSFC contract NAS 8-27568. Most pressure regulators operate with inlet pressures greater than 100 psia*. In the case of the RNS, the ANSC NERVA supplies propellant tank pressurant GH_2 to a regulator with steady-state inlet pressures at the engine varying from 319 psia* to 723 psia* and with temperatures varying from 238°R* to 315°R*. (Reference: ANSC N4110:0057, 26 February 1971, "State Points for the 1137400/Revision E Reference Engine"). Both the multiple and single tank pressurization regulators are based on these values. A temperature rise of 100°R* and a pressure drop of 100 psi* was considered since the regulator was assumed to be located remotely from the source gas. Pressures and temperatures for a typical RCS regulator are yet to be determined.

(4) Flow rate is based on down stream flow demand of the system being pressurized. The full open regulator elements were sized to give a geometric flow area approximately 50 percent of the geometric flow area of the inlet line with a coefficient of discharge of 0.5. Flow rates approximate the requirements for an optimized multiple tank stage and for an optimized single tank stage for the tank pressurant regulators. (Reference LMSC - A984555, Volume VII, dated 1 May 1971, Section 3). Regulated pressures also approximate the above reference. Flow rates and regulated pressures for a typical RCS regulator are to be determined.

(5) Lockup leakage will be determined by the lockup pressure requirement (TBD). Usually, regulators are accompanied by shutoff valves located upstream of the regulator inlet port.

(6) Per inch of sealing diameter or per 2.54×10^{-2} meters of sealing diameter.

---

* 100 psia = 6.895×10^5 Newton's/Meter^2 absolute

319 psia = 2.199×10^6 " " "

723 psia = 4.985×10^6 " " "

100 psi = 6.895×10^5 Newton's/Meter^2

238°R = 132.222 Degree Kelvin

315°R = 175.000 " "

100°R = 55.556 " "

(7) Life cycles of state-of-the-art regulators today are said to be 250,000 cycles with a two-year life. Modulating cycles are defined as the number of pulses of the main regulator metering element in either direction (without seating) due to varying downstream flow demand or changing inlet pressures and temperatures. Lockup cycles are the number of seating cycles of the main regulator metering element due to the lack of downstream flow demand. The 80 lockup cycles for space operation are based on 10 space station/moon/space station missions for RNS with a maximum of 8 burns per mission. The 250,000 modulating cycles are a ROM estimate based on 10 hours of engine operation with 80 burns.

(8) At engine stage interface per ANSC Memo N4230:6397M, E. A. Warman to A. D. Cornell, dated 22 January 1971, Subj: "Perturbed and Isokerma Rate Maps". Also, see LMSC Report A98455, Final Report, Volume VIII, for dose rates at stage tank bottom.

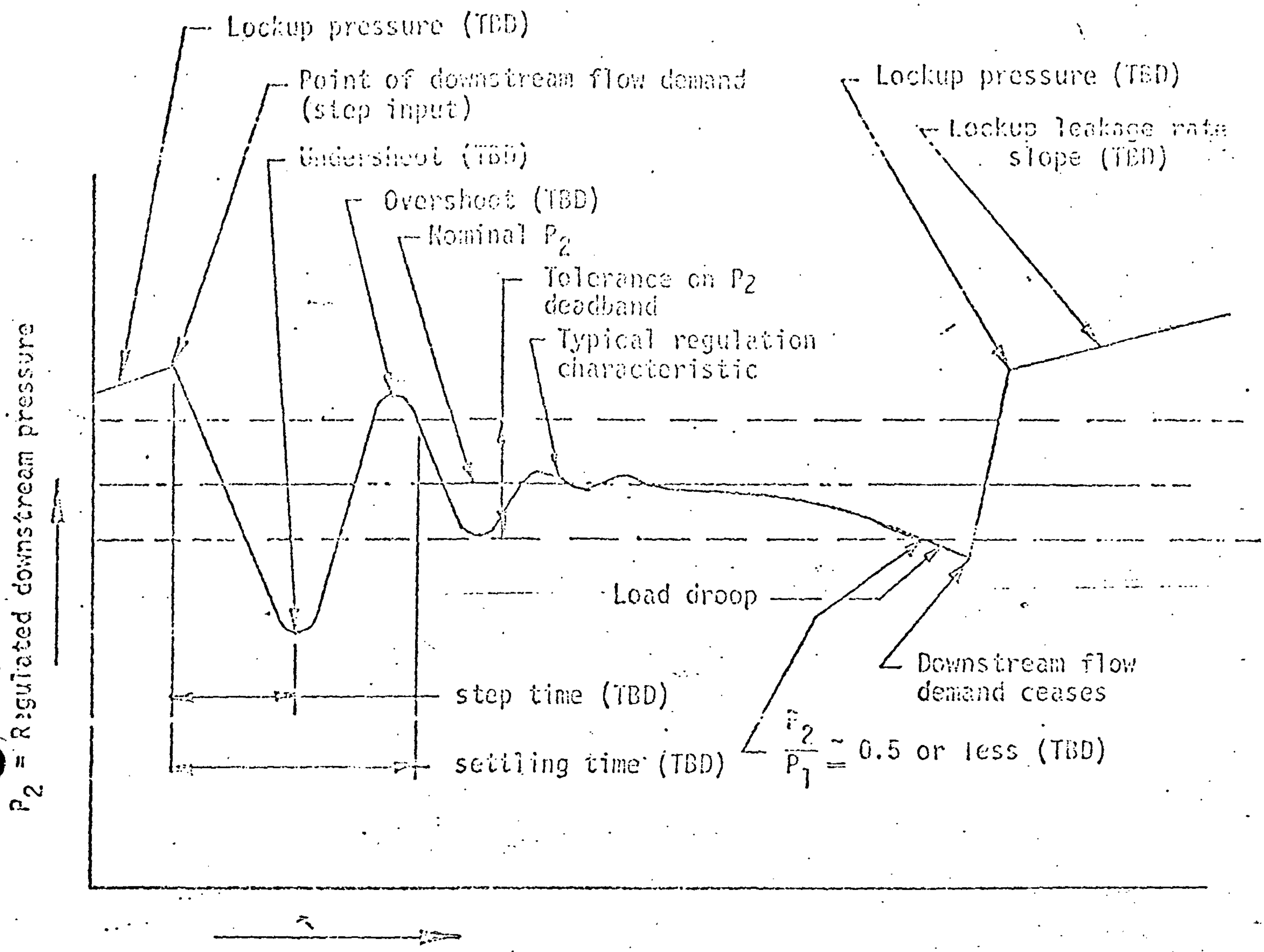
(9) Ten hours run time for total dosage.

(10) Longitudinal axis of RNS.

(11) Increases from 1.0 g to 5.2 g in 300 seconds (5 minutes).

(12) Proof and burst pressures are respectively the product of 1.5 and 2.5 times the maximum system inlet pressure.

(13) Proof and burst pressures are based on a maximum tank working pressure of 2.685×10^5 Newton's/Meter^2 absolute (30 psia) and are respectively the product of 1.5 and 2.5 times this pressure.



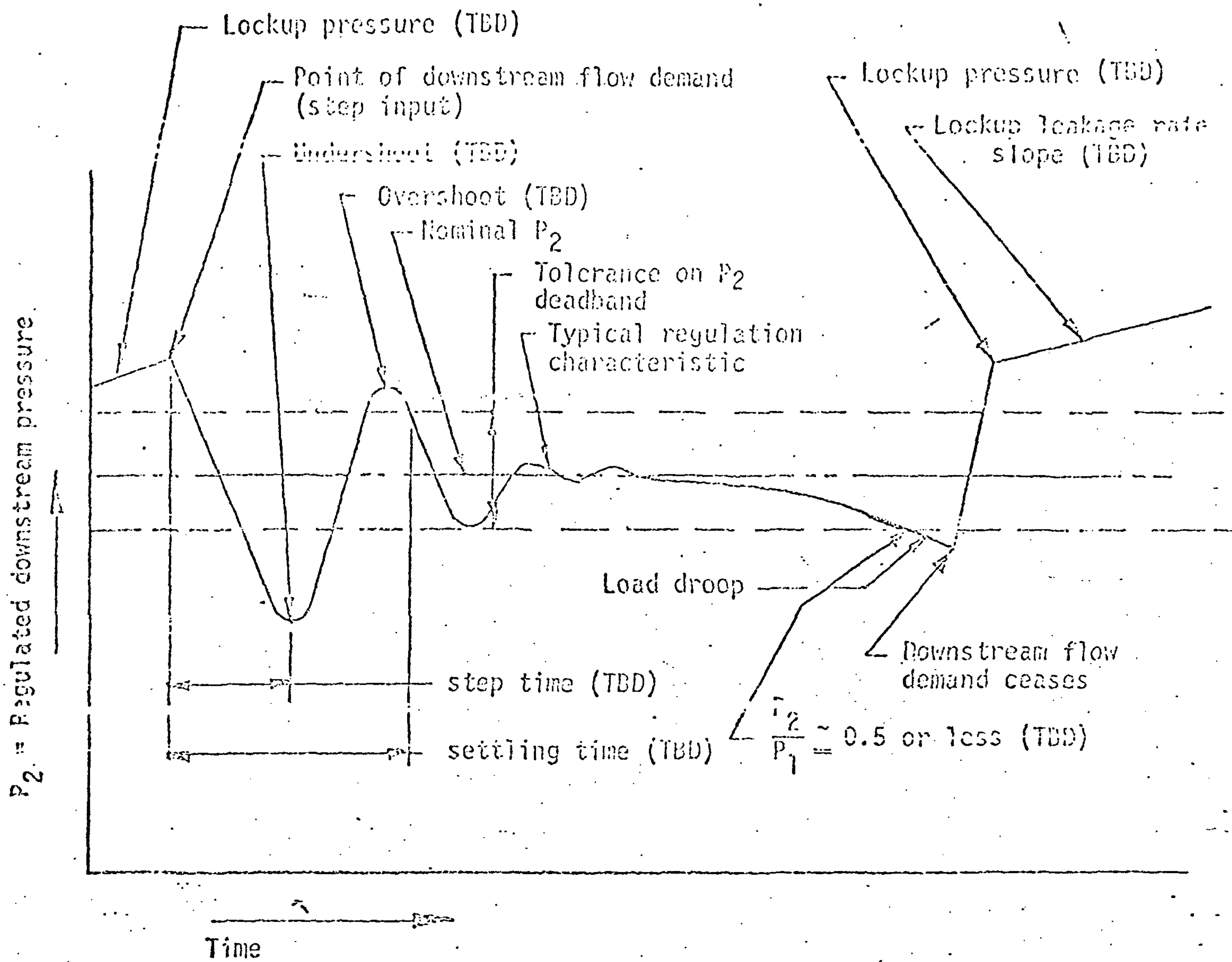
P_1 = inlet or upstream pressure (15.10×10^5 to 49.85×10^5 Newton/Meter^2 absolute) (219 to 723 psia).

$\frac{P_2}{P_1} \cong$ 0.5 or less to remain within deadband and is dependent on the sonic pressure ratio of the gaseous media, upstream system geometry, downstream system geometry, type of system and many other variables.

Nominal $P_2 = 1.62 \times 10^5 \frac{\text{Newton's}}{\text{Meter}}$ absolute (23.5 psia).

Tolerance on P_2 Deadband = $\pm 1.034 \times 10^4 \frac{\text{Newton's}}{\text{Meter}^2}$ ($\pm$ 1.5 psia)

FIGURE 1



P_1 = inlet or upstream pressure (15.10×10^5 to 49.85×10^5 $\frac{\text{Newton's}}{\text{Meter}^2}$ absolute) (219 to 723 psia).

$\frac{P_2}{P_1} \cong$ 0.5 or less to remain within deadband and is dependent on the sonic pressure ratio of the gaseous media, upstream system geometry, downstream system geometry, type of system and many other variables.

Nominal $P_2 = 1.724 \times 10^5$ $\frac{\text{Newton's}}{\text{Meter}^2}$ absolute (25.0 psia).

Tolerance on P_2 Deadband = $\pm$ 1.034×10^4 $\frac{\text{Newton's}}{\text{Meter}^2}$ ($\pm$ 1.5 psi).

FIGURE 2

APPENDIX C

GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY

GOALS FOR REMOTE COUPLING AND UNCOUPLING FOR SPACE MAINTAINABILITY

1. Ports in parallel planes for valves and regulators having two or more ports so that the components can be moved directly away from, or towards, the decoupled interface.

2. Proper location of the valves and regulators so that they may be removed directly away from, or towards, the interface without requiring zig-zag motion to avoid collision with adjacent structure.

3. Uncoupling or coupling components in pressurized systems with minimum loss of fluid through the use of secondary sealing devices.

4. Uncoupling or coupling fluid components with electrical connections in a single operation in lieu of separate operations for fluid connections and electrical connections.

5. Use of line mounted manifolds so that plug-in components can be uncoupled or coupled from the manifold without disconnecting lines.

6. Where two or more components are adjacently located, consideration should be given to mounting these units in a common line mounted manifold for more efficient component removal. Examples of this item would be a quad pack system or a shutoff valve and regulator system. This would allow the use of common point for indexing the remote manipulator.

7. Indexing devices for fool proofing so that components cannot be installed in wrong orientations. This item could be expanded to include slightly different mounting patterns so that regulators and valves of the same size cannot be installed in wrong locations.

APPENDIX D

BRIEF SYNOPSIS OF MAJOR REFERENCES

BRIEF SYNOPSIS OF MAJOR REFERENCES

The following listing covers the primary references read during the literature review portion of Phase I, Contract NAS 8-27568. Additional references will be needed and used during the design phases of the program. The complete listing will be published in the final report.

Reference 1

NASA Marshall Space Flight Center Contract No. NAS 8-27568 - titled "Design of Long Life Space Maintainable Nuclear Stage Regulators and Shutoff Valves". Basic contract with design goals.

Reference 2

"Aerospace Fluid Component Designers Handbook", Volume 1 and Volume 2, Revision D, dated February 1970 (RPL-TDR-64-25).

This handbook describes many types of regulators and shutoff valves as well as other types of components and offers mathematical approaches for the design of aerospace components down to subcomponent level. Major coverage is given in sections designated Heat Transfer, Fluid Mechanics, Fluid Systems, Fluid Components, Modules, Dynamic Analysis of Fluid Components, Computers, Specifications, Contamination and Cleaning, Reliability, Materials, Environments, Stress Analysis, Component Testing, and Fluidics. An extensive reference bibliography is also given.

Reference 3

Anon and not released, "Design Criteria Monograph, Liquid Propellant Regulators, Relief Valves, Check Valves, Burst Discs and Explosive Valves", NASA Lewis Research Center, Cleveland, Ohio.

The monograph treats the design of direct operated, dome loaded and pilot operated gaseous pressure regulators as well as the design of other types of fluid components. It also includes precautions and techniques gained by experience for the design and testing of these components. Section 4.0 cites a number of references applicable to the monograph.

Reference 4

R. L. Kenyon, "Design, Development and Testing of Advanced Helium Pressure Regulator, Part No. 551302", Volume 1 and Volume 2, AFBMD-TR-60-74, July 1960, under contract AF 04(647)-160 to Rocketdyne, Canoga Park, California.

This report deals with the design, development and qualification testing of a pilot operated helium missile propellant tank pressurization regulator. The regulator consists of bleed regulator, a controller and a main metering element with its attendant actuator. Major porting consisted of a supply pressure port, and outlet port, a tank pressure sensing port and an overboard exhaust port. The report includes a rigorous analysis including digital computer techniques leading to the successful development and testing of regulator hardware. Major parameters for the regulator are as follows:

Inlet port size (inches)	5/8
Outlet port size (inches)	2.0
Tank pressure sensing port (inches)	1/4
Fluid temperature range (°F)	-320 to +550
Ambient temperature range (°F)	-320 to +212
Inlet pressure range (psig)	230 to 5000
Outlet pressure (psig) (adjustable)	33 to 35
Flow rate (lbm/sec) at 530°F and 230 psig	0.11
CA of main metering element (in.^{2})	0.054

The regulator design features excellent regulation tolerance, temperature compensation and vibration, and acceleration compensation.

Reference 5

LMSC, Nuclear Shuttle Systems Definition Study, Phase III, LMSC A 984 555, Volume VII, dated May 1, 1971, "RNS Tank Pressurization Analysis".

This report optimizes propellant tank pressures for four tank configurations used in the optimization are as follows:

NAR Single Tank

LMSC Single Tank

LMSC Multiple Tank

MDC Hybrid Tank.

Optimized tank pressures vary from 22.55 to 26.50 psia and flow rates vary from 0.81 to 3.92 lbm/sec of GH_2 depending on tank configuration, mission, engine startup time and engine burn number.

Reference 6

ANSC Technical Proposal RN 71007, Folder 1, March 1971, "Design of Long Life, Space Maintainable Nuclear Stage Regulators and Shutoff Valves"

This proposal describes the ANSC approach and program for the design of long life, space maintainable nuclear stage regulators and shutoff valves. Now under NASA Contract NAS 8-27568. The contract specifies that the purpose of the program is to develop zero leakage, long life components with characteristics that render them suitable for being space maintainable and emphasizes that critical areas to be stressed are radical departures from current state-of-the-art in the field of remote coupling and uncoupling, as well as remote change out of components and component parts.

Reference 7

Robertshaw Fulton, "Analysis, Design and Development of High Flow Helium Pressure Regulator", Volume 1 and Volume 2, AFBMD-TR-60-72 (I) and AFBMD-TR-60-72 (II), June 1960, Contract NO. AF 04(647)-161, under contract to Robershaw Fulton Controls Company, Anaheim, California.

This report describes the analysis, design and development of a pilot operated helium regulator. Computer analysis techniques as well as comprehensive thermodynamic analysis of gaeous flow through orifices. The regulator elements consisted of a main metering element and its attendant actuator, a pilot valve, a controller element and a flow limiter.

Reference 8

J. C. DuBuisson, "Long Life Reliability Problems and Solutions for Valves under Spatial Environment", AIAA Paper No. V II N.1, June 17-20, 1969.

This paper deals with the reliability problems associated with valves and regulators employed in long duration space missions of up to ten years in a hard vacuum. An extensive literature survey was performed followed by a survey of twenty users and producers of valves and regulators. The paper points out reliability problems such as cyclic life, contamination and fluid compatibility for these components. Conclusions and recommendations are given. It emphasizes that extreme care should be exercised in selecting regulator suppliers since only a few suppliers are designing and developing regulators for long term spatial use. Non-piloted regulators are desired over piloted types for reliability reasons. The paper states that Futurecraft Corporation is supplying hardware to TRW for a 5-year life and 250,000 cycle test of a solenoid valve and a pressure regulator. Also, Royal Industries is developing a long life pressure regulator for North American Rockwell under open contract.

Reference 9

NASA, "Advanced Valve Technology", NASA SP-5019, February 1967.

Many subjects are discussed in this report which consists of 14 chapters. Many types of valving elements, control elements, seals and sealing devices are discussed as well as the problems of contamination, the hard vacuum problems of material sublimation, cold welding, and propellant and temperature compatibility of materials. Of special interest are the discussions on wet seals, labyrinth seal, the all-metal valve, the indium seated valve and the thermoelectric freeze valve described in Chapter 13.

Reference 10

TRW Space Technology Laboratories, "Valve Study", Volume 1, Report No. 8651-6032-SU000, 19 July 1964, under NASA Contract No. NAS 7-107, Washington, D. C.

The objective of this program was to advance the state-of-the-art of valves used in liquid propulsion space craft engines. The various sections are identified as valve analysis chart, advanced valve study, luquid propellant study, space maintenance study, meteroidal impact, valve qualification testing and new concepts. Of special interest is the liquid metal seal study for helium leak rates of 10^{-8} scc/sec.

Reference 11

Anon and not released, "Liquid Rocket Valve Assemblies Design Criteria Monograph", NASA Lewis Research Center, Cleveland, Ohio.

The monograph treats existing state-of-the-art for many types of valves and gives recommended practices from the standpoint of experience for the design and testing of these valves. Section 4.0 cites a number of references applicable to this monograph.

Reference 12

Anon and not released, "Liquid Rocket Valve Component Design Criteria Monograph", NASA Lewis Research Center, Cleveland, Ohio.

The monograph treats existing state-of-the-art for many types of valving components and gives recommended practices from the standpoint of experience for the design of these components. Section 4.0 cites a number of references applicable to this monograph.

Reference 13

TRW Space Technology Laboratories, "Advanced Spacecraft Valve Technology", Final Report No. 12411-6011-R000, July 1970 under NASA Contract NAS 7-717, Jet Propulsion Laboratory, Pasadena, California.

Both mechanical and non-mechanical controls are treated in this report. Mechanical controls treat seal and valve development studies, including testing of an energized seal with helium leak rates as low as 3.0×10^{-9} scc/sec, with seat stresses of 12,000 psi. Non-mechanical controls are intended for liquid flow control. Also included are metal-to-metal interference seals and thin film seal studies.

Reference 14

TRW Space Technology Laboratories, "Advanced Space Craft Technology Compilation", Volume 1 (Mechanical Controls) and Volume II (Non-Mechanical Controls), Report No. 12411-6012-R000, July 1970, under NASA Contract NAS 7-717, Jet Propulsion Laboratories, Pasadena, California.

Volume 1 (Mechanical Controls) contains a great deal of data on environmental, functional and operational considerations for aerospace controls. Materials, including liquid hydrogen are discussed. A lengthy review of valves is also presented and includes valve state-of-the-art, valve design considerations, conceptual valve designs, zero G vent valve study and in tank design study. Conceptual valve designs include the semi-toroidal diaphragm valve, radial shutoff valve, valves with thermally generated seat stress, thermally actuated valves, piezo electrically actuated valves, electro fluid interaction valves, electro-seal valve, electromagnetic capillary valves, capillary relief and check valves, radioisotope heated valves, fusion valves and diffusion valves. Sealing technology discussed teflon seal improvements, wet metal seal study, metal-to-metal seals, thin film studies, and energized seal development. Volume 2 (Non-Mechanical Controls) contains a large amount of data on fluid devices and fludic systems. Both gaseous pressure regulator and liquid flow control devices are treated. It was concluded that the hybrid (mechanical first stage) fluidic gaseous pressure regulator was the best candidate for wide ranges of supply pressures. Other fluidic concepts and demonstrations were concerned with vented jet evaluation, electrical interfaces, piezo electric E-F transducer, E-F digital output selector, fluidic power supplies and cavitating venturi sump.

Reference 15

TRW Systems, "Advanced Valve Technology", Volume I (Mechanical Controls) and Volume II (Non-Mechanical Controls), Final Report No. 06641-6023-R000, January 1969, under NASA Contract NAS 7-436, Washington, D. C.

Volume I (Mechanical Controls) contains valve component rating analysis charts, valve application studies, valve conceptual studies, valve actuator studies, instrumentation and measurement, and technology support. Valve concepts considered were the thermally stressed sealed valve, the thermally stressed shutoff valve, the 2-way thermally actuated micro valve, the 3-way thermally actuated micro valve, the snap acting thermally actuated valve, and the piezo electric actuated valve. Actuators discussed were the thermal expansion actuator, the thermal state change actuator, electro dynamic actuators, piezo electric actuators and the use of super conductivity. Seal technology support was in the area of thin film coated valve seats and ultra sonic effects on valve leakage. A design analysis of a piezo electric actuator is included. Volume II (Non-Mechanical Controls) contains a fluidic component analysis chart, fluidic application studies for a pressure regulator, hybrid pressure regulator designs, subsystem trade-off studies and electrical interfaces for fludics. Fluidic power supplies are also discussed.

Reference 16

ANSC, "Pressurization Gas Requirements, NERVA Program", Report No. S054-201, November 1970, under NAS Contract SNP-1.

This report recommends the autogenous tank pressurization for the single tank RNS for the lunar missions. The report concludes that large increases in payload can be realized if the mission is started with the propellant load at 15 psia saturation pressure, and the tank pressure is programed to levels that satisfy minimum engine NPSP.

Reference 17

ANSC, "State Points for the 1137400/Revision E Reference Engine", Memo N4110:0067, 26 February 1971.

This memorandum gives state point pressures, temperatures, and flow rates for the engine components and component junctions for the Revision E engine. The state point data are tabulated for extreme conditions for start of life (100% thrust), end of life (100% thrust), start of life - 1 TPA (80% thrust), end of life - 1 TPA (80% thrust), start of life (60% thrust), end of life (60% thrust), start of life - 1 TPA (60% thrust), end of life - 1 TPA (60% thrust) and emergency operating point.

Reference 18

ANSC, "Design for Remote Handling", Report No. 2307, August 1962, under NASA Contract No. SNP-1.

This manual discusses problems with remote handling and suggests design details for components, assemblies, and systems of NERVA engines which will greatly aid remote maintenance.

Reference 19

AeroQuip Corp/Marman Division, "Conoseal Tube Joints, Pipe Joints and Fittings", Catalogue No. 819 and associated bulletins.

This supplier bulletin describes several remote coupling designs and seals for remote joints. These remote couplings are used in the nuclear field and have been used extensively on the NERVA program. They are utilized in high vacuum work, cryogenic applications and liquid metal work.

Reference 20

Gray Tool Company, "Remotely Operated Connections, Disconnect Couplings", Graylock Bulletin No. 66-2.

This supplier bulletin describes some mechanical remote coupling designs that are in present industrial and nuclear reactor use.

Reference 21

MDC, "Nuclear Shuttle System Definition Study", Phase III, Final Report, Volume II, Concept and Feasibility Analysis, Part B, Class 3 RNS, Book 2, Systems Definition, Report No. MDC G-2134, May 1971, under NASA Contract NAS 8-24714.

This report includes proposed methods of remote coupling and uncoupling and sealing various fluid subsystems and flange joints for a multiple tank RNS. The report also includes valve sizes and types for various fluid systems and subsystems.

Reference 22

ANSC, "Valves Suitable for Long Term Operation in Space", ANSC Document No. 7U2417.

This document reflects actual helium leak rates of a poppet seal as low as 5.98×10^{-6} scc/sec after 1000 cycles with 45 psig applied. The flexible metal poppet seal was fabricated from phosphor bronze with a 2.5 inch diameter. The document also gives the test results of interference fit pyro technique shutoff valve in 1/2-inch and 3/4-inch line sizes. The initial leak rates, with 400 psig of helium applied, were as low as 1×10^{-10} scc/sec and 5×10^{-8} scc/sec for the 1/2-inch line size valve and 3/4-inch line size respectively.

Reference 23

NASA, "Seals and Sealing Techniques", NASA SP 5905(02), 1970.

This compilation briefly describes many sealing innovations including translating and rotary dynamic seals as well as static seals down to hard cryogenic temperatures. Of special interest are heated eutectic metal vacuum seals for rotating shaft and static applications and a laminated metal polyimide cryogenic poppet seal.

Reference 24

NASA, "Valve Technology", NASA SP 5927(01), 1970.

This compilation treats relief valves, cryogenic applications, valves for extreme conditions, safety valves, special applications and leak proof designs. A double seated shutoff valve, a thermally actuated shutoff valve and a self-cleaning metal-to-metal seated poppet valve are of special interest.

Reference 25

NASA, "A Technique for Joining and Sealing Dissimilar Materials", NASA SP 5016, 1965.

This document describes a technique for joining and sealing an end cap head to a pipe forming a sealed end of a cryogenic pressure vessel. A special eutectic alloy was used that expands on cooling. The design has been used to seal and withstand structural loads with fluid temperatures from 70°F to -452°F and pressures up to 2000 psig. The design features ease of assembly and disassembly, reusability of components, sealing effectiveness, minimum induced stresses, capability of joining dissimilar metals and reliability. It is also a candidate for remote coupling and uncoupling of a component.

Reference 26

AEC Research and Development Reports "Survey of Candidate Static Sealing Mechanisms for Fast Flux Test Facility Closed Loop and Driver Fuel Duct Applications", BNWL-689, dated March, 1968 and "Sealing Mechanisms for FFTF Closed Loop and Open Test Position Closures", BNWL-1069 (UC80), dated October 1969 by Pacific Northwest Laboratory, Richland, Washington.

These reports survey the field of remote couplings and joints for nuclear test and power reactors. Many types of joints and seals are reviewed for long life, sealing and remote coupling and uncoupling.

APPENDIX E

DATA EXTRACTED FROM THE NUCLEAR SHUTTLE SYSTEM DEFINITION STUDY G-2134 PHASE III FINAL REPORT

DATA EXTRACTED FROM THE

NUCLEAR SHUTTLE SYSTEM DEFINITION STUDY G-2134

PHASE III FINAL REPORT

Volume II System Definition by McDonnell Douglas Astronautics Company

1. PROPULSION MODULE HARDWARE TREE

Description	Design Conditions/ Requirements
Main Propulsion	
Propellant Feed System	91.9 lb/sec LH_2, flow at 30 psia
° 8 in. dia throttle valves 2 reqd, motor-drive (120 Vac 400 Hz), planetary gear transmission, full flow analog visor type, metallic main gate seal opening motion - linear seal withdrawal 90° rotational.	
Orbit Refueling	
° 4 in. dia spray bypass valve - 4 required, planetary gear transmission, full flow visor type, metallic main gate seal, open motion - linear seal, withdrawal, 90° rotational, mechanical spring loaded, override - normally closed, ac motor driven (120 vac, 400 Hz).	3,000 gpm p = 40 psia T ≐ 40°R
° 4 in. dia check valve flapper type (2 sections).	
Flight Vent	
° 2-1/2 in. dia pilot operated quad blowdown valves - 4 reqd.	0.6 lb/sec GH_2 at 29 psia T ≡ 40°R

PROPULSION MODULE HARDWARE TREE

Description	Design Conditions/ Requirements
Main Propulsion	
Integrated Childown System	
° 2 in. dia check valves - flapper type (split design) 5 reqd.	2 lb/sec LH_2
° 2 in. dia shutoff valves - normally closed, pilot operated solenoid type - 4 reqd.	2 lb/sec LH_2
Refill System	
° 3.25 in. dia check valves - flapper type (split design) 2 reqd.	5 lb/sec
Auxiliary Propulsion	
Reaction Control System	Module stabilization during assembly and maintenance
° 1/2 solenoid valves direct acting - 2 reqd.	
° 1/2 in. relief valve	
° 1/4 in. nozzle control solenoids direct acting- 24 reqd.	

II. CORE PROPELLANT MODULE HARDWARE TREE

Description	Design Conditions/ Requirements
Main Propulsion	
Propellant Feed System	91.9 lb/sec LH_2 30 psia
° 8 in. dia throttle valves - 2 reqd 12-in. dia blocking valve - motor-driven (120 vac, 400 Hz), planetary gear transmission, full flow binary visor type, metallic main gate seal, opening motion - linear seal withdrawal, 90° rotational.	

CORE PROPELLANT MODULE HARDWARE TREE

Description	Design Conditions/ Requirements
Main Propulsion	
Orbit Refueling	
° 4 in. dia spray bypass valve - 4 reqd, motor driven (120 vac, 400 Hz), planetary gear transmission, full flow binary visor type, metallic main gate seal, open motion - linear seal, withdrawal, 90° rotational mechanical spring loaded override - normally closed.	3,000 gpm p = 40 psia T = 40°R
° 4 in. dia disconnect, normally open.	
° 4 in. dia check valve - 2 sectioned flapper type.	
Ground and Emergency Vent	
° 6 in. vent and relief (pneumatically actuated) valve	11 lb/sec GH_2 at 29 psia 3,000 gpm
° 6 in. poppet disconnect - normally closed.	
° 4 in. (poppet balanced) relief valve	2 lb/sec GH_2 at 29 psia
° 1/4 in. 3-way direct acting solenoid valve.	475 psig
° 1/4 in. quick disconnect coupling normally closed.	475 psig
Flight Vent	
° 2 1/2-in. dia pilot operated quad blowdown valves - 4 reqd.	0.6 lb/sec GH_2 at 29 psia T = 40°R

III. TANDEM PROPELLANT MODULE HARDWARE TREE

Description	Design Conditions/ Requirements
Main Propulsion	
Propellant Feed System	91.9 lb/sec LH_2 30 psia
° 8 in. dia throttle valves - 2 reqd.	
° 12 in. dia blocking valve, motor-driven (120 vac, 400 Hz), planetary gear transmission, full-flow, binary, visor-type metallic main gate seal opening motion - linear seal withdrawal, 90° rotational.	
Orbit Refueling	
° 4 in. dia spray bypass valve - 4 reqd, motor driven (120 vac, 400 Hz), planetary gear transmission full-flow, binary, visor-type, metallic main gate seal open motion - linear seal, withdrawal, 90° rotational, mechanical spring loaded, override-normally closed.	3,000 gpm
° 4 in. dia check valve - 2 sectioned flapper type.	
Ground and Emergency Vent	
° 6 in. vent and relief valve (pneumatically actuated)	11 lb/sec GH_2 at 29 psia
° 6 in. poppet disconnect - normally closed	3,000 gpm
° 4 in. (poppet balanced) relief valve	2 lb/sec GH_2 at 29 psia
° 1/4 in. 3-way direct acting solenoid valve	475 psig
° 1/4 in. quick disconnect coupling normally closed.	475 psig
Flight Vent	
° 2-1/2 in. dia pilot operated quad blowdown valves - 4 reqd.	0.6 lb/sec GH_2 at 29 psia T = 40°R

IV. OUTBOARD PROPELLANT MODULE HARDWARE TREE

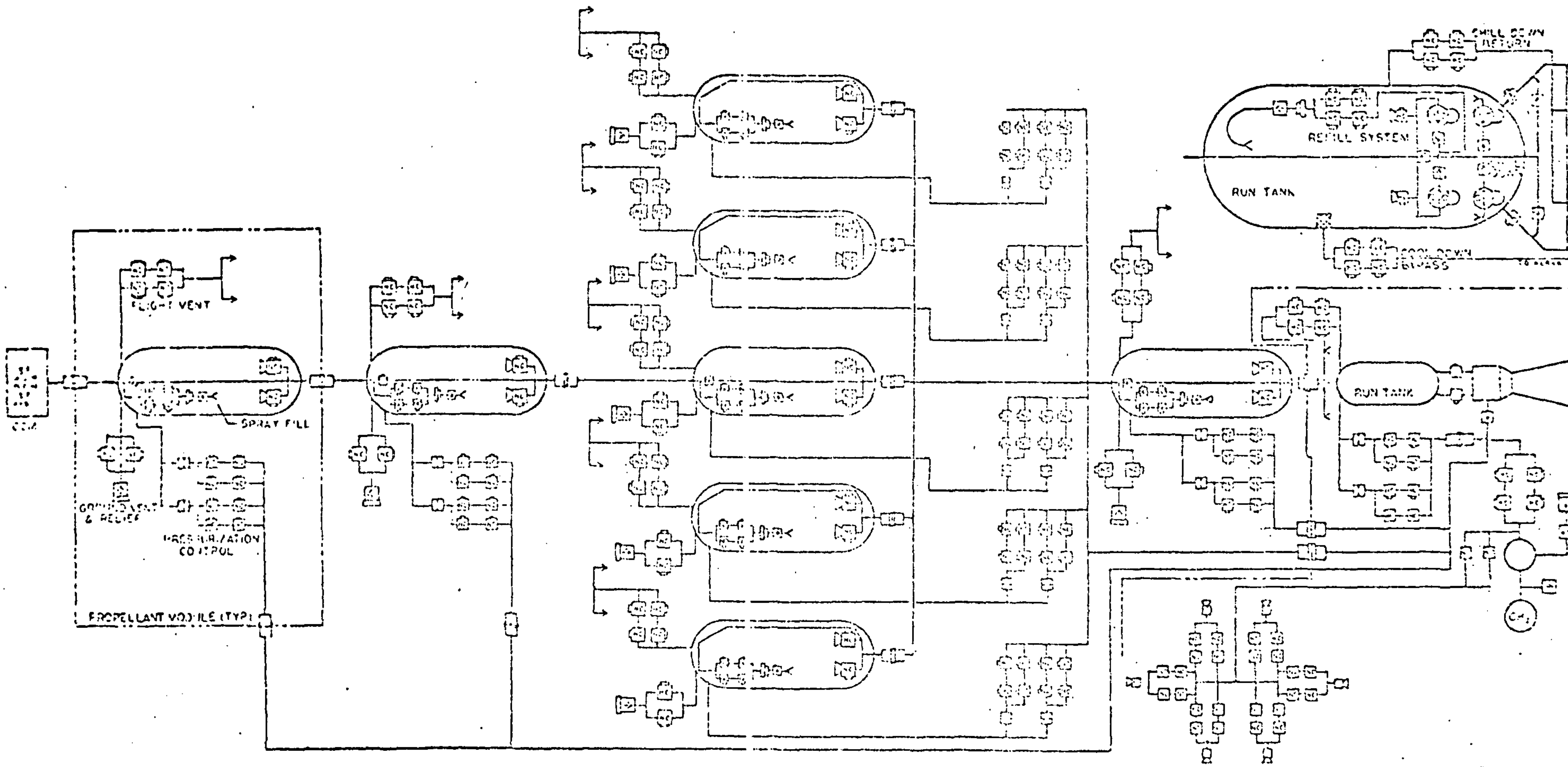
Description	Design Conditions/ Requirements
Main Propulsion	
Propellant Feed System	91.9 lb/sec LH_2 30
° 8 in. dia throttle valves - 2 reqd, motor-driven (120 vac, 400 Hz), planetary gear transmission, full flow binary visor type, metallic main gate seal, opening motion - linear seal withdrawal, 90° rotational.	
Orbit Refueling	
° 4 in. dia spray bypass valve - 4 reqd.	3,000 gpm p = 40 psia T = 40°R
° Motor driven (120 vac, 400 Hz), planetary gear transmission, full flow binary visor type, metallic main gate seal, open motion-linear seal, withdrawal, 90° rotational, mechanical spring loaded, over-ride - normally closed.	
° 4 in. dia normally open disconnect.	
° 4 in. dia check valve - 2 sectioned flapper type.	
Ground and Emergency Vent	
° 6 in. vent and relief valve - (pneumatically actuated)	11 lb/sec GH_2 at 29 psia
° 4 in. (poppet balanced) relief valve.	3,000 gpm 2 lb/sec GH_2 at 29 psia
° 1/4 in. 3-way direct acting solenoid valve.	475 psig
° 1/4 in. quick disconnect coupling - normally closed.	475 psig
Flight Vent	
° 2-1/2 in. dia pilot operated quad blowdown valves - 4 required.	0.6 lb/sec GH_2 at 29 psia T = 40°R
° 2-1/2 in. dia x 0.016 in. stainless steel ducting.	p = 29 psia

V. COMMAND AND CONTROL MODULE HARDWARE TREE

Description	Design Conditions/ Requirements
° Thrustor Control Valves, direct acting solenoids, normally closed - 48 reqd.	
° Squib Operated Closed, plunger type isolation valves - 12 reqd.	
° Regulators - 4 reqd.	
° Normally Closed Direct Acting Solenoid Valves - 32 reqd.	
° Normally Open Direct Acting Solenoids - 2 reqd.	
° Burst Disk Relief Valves - 2 reqd.	
° Check Valves - 4 reqd.	

APPENDIX F

PROPULSION SYSTEM SCHEMATIC



PROPULSION SYSTEM SCHEMATIC

EXTRACTED FROM THE NUCLEAR SHUTTLE SYSTEM DEFINITION STUDY G-2134

PHASE III FINAL REPORT VOLUME II SYSTEM DEFINITION

by McDonnell Douglas Astronautics Company

APPENDIX G

MATRIX - DESIGN SELECTION CHART
(VALVES)

13 [illegible] 1971

MATRIX - VALVE CONC[illegible] DESIGN SELECTION CHART

(Contract NAS 8-27568)

Rating Factors

3 = Excellent
2 = Good
1 = Fair
0 = Poor

Highe[illegible]ssible Score = [illegible] points.

Concept/Description		Replacement Maintenance	In-Line Maintenance	Leakage (1×10^{-7} SCCS)	Cycle Life	Storage Life (10 years)	Operating Life (3 years)	Scaling	Vibration	Contamination	Pressure Drop	Performance	Availability of Design Information	Total Points	Relative Ranking	Remarks
No. 1 Solenoid Shutoff - Removal Capability	1139890	3	3	3	3	2	2	1	2	2	0	3	3	27	2	Can remove valve without draining tank and line.
No. 2 Inline Poppet Tank Valve	1139892	3	3	2	3	2	2	2	2	2	3	1	2	27	2	Can remove valve without draining tank
No. 3 Inline Poppet - High Pressure Shutoff Seal	1139893	0	0	3	1	2	2	3	3	3	3	1	1	22	7	
No. 4 Multiple Poppet Solenoid Valve	1139894	3	3	3	3	2	2	1	2	2	1	3	3	28	1	
No. 5 Inline Valve - Electromagnetic Seal	1139895	3	1	1	3	3	3	1	3	3	2	3	1	27	2	No moving parts
No. 6 Poppet - Electromagnetic Seal	1139896	2	2	1	3	3	3	3	2	2	2	1	1	25	4	Self Actuation
No. 7 Poppet - Liquid Metal Seal	1139897	2	0	2	0	2	2	3	1	2	2	0	1	17	9	Liquid metal seal
No. 8 Poppet - Spherical Metal Seal	1139899	2	1	3	3	2	2	3	1	2	2	3	2	26	3	Demonstrated Shutoff Seal
No. 9 8" Inline Poppet Tank Valve	1139012	1	2	2	3	2	2	2	2	2	3	1	2	24	5	Replaceable and buried actuator.
No. 10 Poppet - Remote Actuator Removal	1139930	1	1	2	3	2	2	2	2	1	3	2	2	23	6	Remote Removal Actuator
No. 11 Visor - Linear Seal Withdrawal	1139931	1	1	1	2	2	2	2	1	2	3	2	2	21	8	Similar to Saturn IV Lox Pre Valve

MATRIX - VALVE CONCEPT DESIGN SELECTION CHART

(Contract N. 8-27568)

Rating Factors

3 = Excellent
2 = Good
1 = Fair
0 = Poor

Highest Possible Score = 39 points.

Concept/Description		Programed Regulated Pressure Capability	Replacement Maintenance	In-Line Maintenance	Leakage	Complexity	Cycle Life	Storage Life (10 years)	Operating Life (3 years)	Scaling	Vibration	Contamination Resistance	Performance	Availability of Design Information	Total Points	Relative Ranking	Remarks
No. 1 Motor Operator Visor Regulator	1139923	3	1	2	1	2	2	2	2	2	3	2	2	2	26	3	Will be designed by NERVA. Can shut off on command.
No. 2 Motor Operated Shear Plug Regulator	1139924	3	2	1	3	2	3	3	2	2	3	3	2	2	31	2	Can shut off on command.
No. 3 Motor Operated Shear Plate Regulator	1139925	3	2	2	3	2	3	3	2	1	3	3	3	2	32	1	Can shut off on command.
No. 4 Motor Operated Metering Valve Control Dome Load Shear Seal Regulator	1139926	2	2	1	2	1	2	2	1	2	2	2	1	2	22	4	Can shut off on command.
No. 5 Piloted Pressure Dome Loaded Shear Seal Regulator	1139928	0	3	0	0	0	1	1	0	2	1	2	1	3	14	5	Cannot shut off on command. Contains balancing springs and bellows.

13 [illegible]971

MATRIX - REMOTE COUPLING [illegible]IGN SELECTION CHART

(Contract NAS 8-27568)

Rating Factors
3 = Excellent
2 = Good
1 = Fair
0 = Poor

Highest Possible Score = 36 points.

Concept/Description		Simplicity of Manipulator Movement	Accuracy Required of Manipulator	Accessibility Requirements	Seal/Sealing Surface Protection	Dead Flight Weight	Storage Life (10 years)	Operating Life (3 years)	Scaling	Vibration	Contamination Generation	Availability of Design Information	Complexity	Total Points	Relative Ranking	Remarks
No. 1 Remote Coupling Structural Joint and Seal	1139920	3	3	3	2	2	2	1	2	1	2	2	3	26	1	May be designed by NERVA.
No. 2 Low Melting Alloy Structural Coupling and Seal	1139921	1	3	2	3	3	1	3	3	2	0	0	1	22	4	May be investigated by NERVA
No. 3 Thermal Interference Joint	1139922	1	1	2	1	2	0	0	2	3	3	0	3	18	5	
No. 4 Remote Coupling - In-Line	1139927	3	3	2	3	1	2	2	1	2	2	2	1	24	3	
No. 5 In-Line Remote Coupling	1139891	3	3	2	2	1	3	2	1	1	2	2	2	25	2	

APPENDIX H

SUMMARY - DESIGN CONCEPT SELECTION (VALVES)

Contract: NAS 8-27568

Date: 13 October 1971

SUMMARY
Design Concept Selection
(VALVES)

Item	Concept No. 1	Concept No. 2	Concept No. 3
	Dwg. No. 1139890 Description: Solenoid shutoff removal capability	Dwg. No. 1139892 Description: In-line poppet tank valve	Dwg. No. 1139893 Description: In-line poppet - high pressure shutoff seal
Rating	2	2	7
Summary	Pro: (1) Remote replacement of valve from pressurized manifold. (2) Good replacement maintenance. (3) Provides good shut off sealing. (4) Actuator is down stream of main shutoff seal. Con: (1) High pressure drop. (2) Two leak paths to outlet side when valve is closed. (3) Size limitati n.	Pro: (1) Remote replacement of valve from pressurized tank. (2) Remote replacement of actuator and ball screw. (3) Actuator is downstream of main shutoff seal. Con: (1) Added pressure drop from back-up poppet. (2) Valve must be removed from tank to replace main seal. (3) Long envelope.	Pro: (1) Provides good shut off seal. (2) Can be designed for bi-directional flow. (3) Direct drive actuator - no gears. Con: (1) Difficult to control high pressure in seal cavity. (2) Actuator is upstream of main shutoff seal. (3) Power required during coast to maintain sealing.

SUMMARY
Design Concept Selection
(VALVES)

Item	Concept No. 4 Dwg. No. 1139894	Concept No. 5 Dwg. No. 1139895	Concept No. 6 Dwg. No. 1139896
Description	Solenoid operated - multiple poppet valve	3/4-in. in-line electromagnetic seal	Balanced poppet - Electromagnetic skirt seal
Rating	1	2	4
Summary	Pro: (1) Remote replacement of valve from manifold. (2) Built in parallel and series redundancy. (3) Can be used for bang-bang regulator. (4) Provides good shutoff seal. (5) Larger valve sizes can be accommodated by manifolding to, or more valves together. Con: (1) Interface seal design problem. (2) Weight. (3) Line size limitation in larger sizes.	Pro: (1) No moving parts. (2) Unique in design. (3) Valve does not have to be removed from line to be service overhauled. Con: (1) Limited to smaller size valves, 1/4 - 3/4. (2) Uncertain of leakage control to goal of 1×10^{-7} SCCS He. (3) Power required to provide sealing.	Pro: (1) Unlimited in size. (2) No actuator required. (3) Can be used as a flow regulator. Con: (1) Needs much development for proper pressure balance of poppet during opening. (2) Uncertain of leakage control to goal of 1×10^{-7} SCCS He. (3) Power required to provide sealing.

Contract: NAS 8-27568

Date: 13 October 1971

SUMMARY

Design Concept Selection

(VALVES)

Item	Concept No. 7 — Dwg. No. 1139897 Description: Poppet valve - liquid metal seal	Concept No. 8 — Dwg. No. 1139899 Description: Poppet valve - motor operated - spherical metal shutoff seal	Concept No. 9 — Dwg. No. 1139012 Description: 8" poppet tank valve
Rating	9	3	5
Summary	Pro: (1) Provides good shutoff seal. Con: (1) Loss of liquid metal in hard vacuum. (2) Contamination of propellant system. (3) Temperature range limited.	Pro: (1) Provides good shutoff seal. (2) Can be bi-directional valve. (3) Mechanics of sealing understood. Con; (1) Cantelever actuator mount.	Pro: (1) Provides good shutoff seal. (2) Provides capability of remotely removing actuator and ball screw subassembly. (3) Bi-directional flow valve. (4) Low pressure drop. Con: (1) Long envelope. (2) Buried actuator.

Appendix H

Contract: NAS 8-27568

Date: 13 October 1971

SUMM[illegible]

Design Concept Selection

(VALVES)

Item	Concept No. 10 Dwg. No. 1139930 Description: Poppet valve - remote actuator removal	Concept No. 11 Dwg. No. 1139931 Description: Visor valve - linear seal withdrawal motor operated	Concept No. Dwg. No. Description:
Rating	6	8	
Summary	Pro: (1) Provides good shutoff seal. Con: (1) Actuator is upstream of main shutoff seal.	Pro: (1) Low pressure drop. (2) Can be designed for bi-directional flow. (3) Linear lift-off seal. (4) Compact envelope. Con: (1) Cantelever actuator mount. (2) Lift-off seal mechanism.	

APPENDIX I

SUMMARY - DESIGN CONCEPT SELECTION (PRESSURE REGULATORS)

Contract: NAS 8-27568

Date: 13 October 1971

SUMMARY

Design Concept Selection

(PRESSURE REGULATORS)

Item	Concept No. 1 Dwg. No. 1139923	Concept No. 2 Dwg. No. 1139924	Concept No. 3 Dwg. N 113992
	Description: Motor operated visor	Description: Motor operated shear plug	Description: Motor operated shear plate
Rating	3	2	1
Summary	Pro: (1) Low torque. (2) Shutoff on command. (3) Good availability of design information. (4) Possible to production test and checkout without flowing gas. (5) Low pressure in main valve cavity. (6) Actuator directly coupled to metering element.	Pro: (1) Simplest sealing mechanism. (2) Metering rotation of 180° possible. (3) Positive shutoff and low leakage. (4) Possible to production test and checkout without flowing gas.	Pro: (1) Least complex. (2) Metering rotation of 270 degrees possible. (3) Positive shutoff and low leakage. (4) Possible to production tes and checkout without flowi gas. (5) Actuator directly coupled to metering element.
	Con: (1) Not designed for direct single point installation. (2) High leakage. (3) Requires electrical connections.	Con: (1) High torque. (2) Requires electrical connections. (3) High pressure in main valve cavity.	Con: (1) Scaling limited. (2) Requires electrical connections. (3) Bulky metering elements. (4) High pressure in main valv cavity.

Appendix

Contract: NAS 8-27568
Date: 13 October 1971

SUMMARY

Design Concept Selection

(PRESSURE REGULATORS)

Item	Concept No. 4 Dwg. No. 1139926 Description: Motor operated metering Dome loaded shear seal	Concept No. 5 Dwg. No. 1139928 Description: Piloted pressure Dome loaded shear seal	Concept No. Dwg. No Description:
Rating	4	5	
Summary	Pro: (1) Low operating force. (2) Valving element adaptable to motor operation. (3) Shutoff on command. Con: (1) Contains long stroking springs. (2) Electronics feedback loop is most complex for constant or programmed pressure regulation. (3) Relatively poor in vibration.	Pro: (1) Low operating force. (2) No electrical connections. (3) Best availability of design information. Con: (1) Most complex. (2) Contains long stroking springs and long stroking bellows. (3) Constant pressure regulation - not adaptable to program tank pressure. (4) Poor in vibration. (5) Cannot shut off on command.	

APPENDIX J

SUMMARY - DESIGN CONCEPT SELECTION (REMOTE COUPLINGS)

SUMMA

Design Concept Selection

(REMOTE COUPLING)

Item	Concept No. 1 Dwg. No. 1139920 Description: Remote coupling Structural joint and seal	Concept No. 2 Dwg. No. 1139921 Description: Low melting alloy Structural joint and seal	Concept No. 3 Dwg. No. 1139922 Description: Thermal interference joint
ating	1	4	5
ummary	Pro: (1) Single operating point in two locations for accessibility. (2) Uses conoseal which has good experience on NERVA. (3) Small envelope with minimal increase in "dead" flight weight. Con: (1) Unknown torque requirements due to unknown friction coefficients in hard vacuum. (2) Unknown cold welding of mating materials in a hard vacuum. (3) Some sliding motion.	Pro: (1) No bolted, threaded or clamped joints. (2) Pressure assists sealing. (3) Lowest leakage of all five concepts. Con: (1) Complex coupling or uncoupling. (2) Unknown strength and creep properties of low melting alloy at cryogenic temperatures. (3) Unknown evaporation rates in a hard vacuum. (4) Undeveloped techniques for injecting and ejecting the low melting alloy for a reliable joint in a zero "G" envrionment.	Pro: (1) High sealing stress. (2) Simple design. (3) Dead flight weight minimized. Con: (1) Close manufacturing tolerances. (2) Large threaded backup coupling. (3) Possible creep during long term storage. (4) Unknown cold welding of mating materials in a hard vacuum. (5) May damage sealing surfaces.

Contract: NAS 8-27568

Date: 13 October 1971

SUMMARY

Design Concept Selection

(REMOTE COUPLING)

Item	Concept No. 4 Dwg. No. 1139927 Description: In-line remote coupling	Concept No. 5 Dwg. No. 1139891 Description: In-line remote coupling	Concept No. Dwg. No. Description:
Rating	3	2	
Summary	**Pro:** (1) Single operating point in two locations for accessiblity. (2) Mechanical drive provides pure compression on static seal - no sliding seal engagement. (3) Adaptable to many types of static seals. **Con:** (1) Unknown torque requirements due to unknown friction coefficients in hard vacuum. (2) Unknown cold welding of mating materials in a hard vacuum. (3) Requires axial expansion joint.	**Pro:** (1) Single operating point for accessibility (can readily be modified for two locations). (2) Mechanical drive provides pure compression on static seal - no sliding seal engagement. (3) Does not require axial expansion of line to couple or uncouple. (4) Adaptable to many types of static seals. (5) Minimal weight in large line sizes. **Con:** (1) Unknown torque requirements due to unknown friction coefficients in hard vacuum. (2) Unknown cold weld of mating materials in a hard vacuum. (3) Ball screw may be expensive.	

Appendix J

APPENDIX K

VALVE DESIGN GOALS

8" DIAMETER POPPET VALVE

VALVE DESIGN GOALS
8" DIAMETER POPPET VALVE
P/N 1140002

OPERATIONAL REQUIREMENTS

Nominal Line Size (meters)	20.320×10^{-2}
(inches)	8.00
Valve Type	Poppet
Pressures	
Maximum system working, ($\frac{\text{Newton}}{\text{meter}^2}$ absolute)	6.895×10^{5}
(psia)	100
Minimum system working, ($\frac{\text{Newton}}{\text{meter}^2}$ absolute)(1)	6.895×10^{-5}
(psia)	1×10^{-8}
Maximum differential, ($\frac{\text{Newton}}{\text{meter}^2}$ differential)	3.447×10^{4}
(psid)	5 @ full flow condition
Proof, ($\frac{\text{Newton}}{\text{meter}^2}$ absolute)	1.032×10^{6}
(psia)	150
Burst, ($\frac{\text{Newton}}{\text{meter}^2}$ absolute)	1.397×10^{6}
(psia)	200
Flow resistance coefficient (K)(7)	1.5
Temperature Range (°K)	-19.99 to +322
(°F)	-423 to +120
Internal Leakage, SCCS He	1×10^{-7}(2)

External Leakage, SCCS He	1×10^{-7} (2)
Internal Leakage, SCCS He	1×10^{-3} (4)
Media	LH_2 & LN_2
Purge Pressurization & Checkout Media (3)	He & GN_2
Maximum Flow Rate(3) (LH_2) ($\frac{\text{Kilogram}}{\text{second}}$)	41.69
(lbm/sec)	91.9
Storage Life, (seconds)	3.1536×10^8
(years)	10
Operating Life, cycles(4)	242

(1) Ambient temperature and pressure

a. Acceptance	38
b. Pre-installation checkout	12
c. Pre-launch checkout	50
Total	100

(2) Cryogenic

a. Acceptance	19
b. Pre-installation	6
Total	25

(3) Space Operation

a. During engine operation	72
b. Checkout at space station	45
Total	117

Nuclear Environment[5]	3×10^6 [6]	Gamma KERMA Rate: Rads (carbon)/hr.
	3×10^{11}	Fast Neutron Flux: n/cm^2-sec ($E_n \geq 0.9$ MeV)
Opening Time, secs		5
Closing Time, secs		5
Mode of Actuation		Electric Motor

---

(7) Flow Resistance Coefficient, "K", is defined by the following equation using U.S. customary units:

$$K = \frac{2g_c \, \rho A^2 \, \Delta P}{\dot{w}^2}$$

where: g_c = Dimensional conversion factor = 386.4 $\frac{\text{lbm}}{\text{lbf}} \frac{\text{in.}}{\text{sec}^2}$

ρ = Mass density of fluid at inlet flow conditions lbm/in.^{3}

A = Reference area, in.^{2}

ΔP = Pressure loss through the valve, lbf/in.^{2}

$\dot{w}$ = Fluid weight flow rate through the valve, lbm/sec.

FUNCTIONAL REQUIREMENTS

1. Valves shall not be overly sensitive to vibration or acceleration in any axis.

Launch-valve not functioning

	(8) X Axis	Y Axis	Z Axis	Time
Sinusoidal vibration	±3.0 g	±4.5 g	±4.5 g	5 min.
	3 to 35 Hz	0.1 to 15 Hz	0.1 to 15 Hz	
Acceleration	5.2 g	0	0	(9) 5 min.

Space operation - valve functioning

Random vibration	TBD	TBD	TBD	3600 seconds (60 min.)
Acceleration	1.0 g	0.8 g	0.5 g	3600 seconds (60 min.)

2. Valve to be capable of replacement by remote, in space, handling equipment.

3. To be compatible with low melting alloy and thermal interference remote couplings.

4. Provide valve position indication.

DESIGN REQUIREMENTS

1. Inline poppet valve similar to valve concept P/N 1139893.

---

(1) May discharge to space vacuum or be exposed internally to space vacuum.

(2) Per inch of sealing diameter.

(3) Interpretation or assumption by ANSC.

(4) Based upon 1 cycle per burn.

(5) At engine to stage interface per ANSC MEMO N4340:6397M, E. A. Warman to A. D. Cornell, dated 22 January 1971, Subj: "Perturbed NERVA Engine Flux and Isokerma Rate Maps". See LMSC-A984555, Final Report, Volume VIII. Dose rates are also specified.

(6) Ten hours run time for total dosage.

(7) Gas flow rates will be determined based on temperatures, pressure ratios, density and line sizes. Values to be established after R.N.S. requirements have been determined.

(8) Longitudinal axis of R.N.S.

(9) Increases from 1.0 g to 5.2 g in 5 minutes.

APPENDIX L

VALVE DESIGN GOALS

5" DIAMETER BALANCED POPPET

VALVE DESIGN GOALS
5" DIAMETER BALANCED POPPET
P/N 1140001

OPERATIONAL REQUIREMENTS

Nominal Line Size (meters)	12.700×10^{-2}
(inches)	5.00
Valve Type	Poppet
Pressures	
Maximum system working, ($\frac{newton}{meters^2}$ absolute)	2.0685×10^{5}
(psia)	30
Minimum system working, ($\frac{newton}{meters^2}$ absolute)(1)	6.895×10^{-5}
(psia)	1×10^{-8}
Maximum differential working, ($\frac{newton}{meters^2}$ differential)	1.379×10^{4}
(psid)	2
Proof, ($\frac{newton}{meters^2}$ absolute)	3.103×10^{5}
(psia)	45
Burst, ($\frac{newton}{meters^2}$ absolute)	4.137×10^{5}
(psia)	60
Flow resistance coefficient(K)[7]	1.5
Temperature Range, (°K)	+ 336.48 to 232.75
(°F)	+ 200 to -40
Internal Leakage, SCCS He	1×10^{-7} (2)
External Leakage, SCCS He	1×10^{-7} (2)

Internal Leakage, SCCS He	1×10^{-3} (4)
Media	GH_2, GN_2 & GHe
Purge Pressurization & Checkout Media(3)	He & GN_2
Maximum Flow Rate(3) (LH_2) (Kilogram/sec)	4.989
(lbm/sec)	11 lb/sec
Maximum Flow Rate (GH_2)(7)	TBD
Storage Life, (seconds)	3.1536×10^8
(years)	10
Operating Life, cycles(4)	1000

(1) Ambient temperature and pressure		
a. Acceptance		160
b. Pre-installation checkout		50
c. Pre-launch checkout		200
	Total	410
(2) Cryogenic		
a. Acceptance		70
b. Pre-installation		25
	Total	95
(3) Space Operation		
a. During engine operation		300
b. Checkout at space station		195
	Total	495

Nuclear Environment[5]	3×10^{6} [6]	Gamma KERMA Rate: Rads (carbon)/hr.
	3×10^{11}	Fast Neutron Flux: n/cm^2-sec (En > 0.9 MeV)
Opening Time, secs		TBD
Closing Time, secs		TBD
Mode of Actuation		TBD

---

(7) Flow Resistance Coefficient, "K", is defined by the following equation using U.S. customary units:

$$K = \frac{2g_c \, \rho A^2 \, P}{\dot{w}^2}$$

where:

g_c = Dimensional conversion factor = 386.4 $\frac{\text{lbm}}{\text{lbf}} \frac{\text{in.}}{\text{sec}^2}$

ρ = Mass density of fluid at inlet flow conditions lbm/in.^{3}

A = Reference area, in.^{2}

ΔP = Pressure loss through the valve, lbf/in.^{2}

$\dot{w}$ = Fluid Weight flow rate through the valve, lbm/sec.

FUNCTIONAL REQUIREMENTS

1. Valves shall not be overly sensitive to vibration or acceleration in any axis.

Launch-regulator not functioning - no structural failure

	[8] X Axis	Y Axis	Z Axis	Time
Sinusoidal vibration	± 3.0 g 3 to 35 Hz	± 4.5 g 0.1 to 15 Hz	± 4.5 g 0.1 to 15 Hz	5 Min.
Acceleration	5.2 g	0	0	[9] 5 Min.

Space operation - regulator functioning

Random vibration	TBD	TBD	TBD	60 Min.
Acceleration	1.0 g	0.8 g	0.5 g	60 Min.

2. Valve to be capable of replacement by remote, in space, handling equipment.

3. To be compatible with low melting alloy and thermal interference remote couplings.

4. Provide valve position indication.

---

(1) May discharge to space vacuum or be exposed internally to space vacuu.
(2) Per inch of sealing diameter.
(3) Interpretation or assumption by ANSC.
(4) Based upon 1 cycle per burn.
(5) At engine to stage interface per ANSC Memo N4340:6397M, E. A. Warman to A. D. Cornell, dated 22 January 1971, Subj: "Perturbed NERVA Engine Flux and Isokerma Rate Maps". See LMSC-A984555, Final Report, Volume V Dose rates are also specified.
(6) Ten hours run time for total dosage.
(7) Gas flow rates will be determined based on temperatures, pressure ratios, density and line sizes. Values to be established after R.N.S. requirements have been determined.
(8) Longitudinal axis of R.N.S.
(9) Increases from 1.0 g to 5.2 g in 5 minutes.

APPENDIX M

GASEOUS PRESSURE REGULATOR

DESIGN GOALS

GASEOUS PRESSURE REGULATOR DESIGN GOALS /
TANK PRESSURANT REGULATOR
FOR MULTIPLE TANK RNS
P/N 1140007

OPERATIONAL REQUIREMENTS

Approximate Line Sizes (meters)	7.620×10^{-2}
(inches)(4)	3.00
Regulator Type	Programmed Pressure Reducing
Regulator Performance(1)	See Figure 1
Maximum System Inlet Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	4.985×10^6
(psia)(2)	723
Minimum System Inlet Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	8.62×10^5
(psia)(2)	125
Inlet Proof Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	7.481×10^6
(psia)(10)	1085
Inlet Burst Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	12.466×10^6
(psia)(10)	1808
Outlet Proof Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	3.103×10^5
(psia)(11)	45
Outlet Burst Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	5.171×10^5
(psia)(11)	75

Minimum non-operational Pressure ($\frac{\text{Newtons}}{\text{meter}^2}$ absolute)	6.895×10^{-5}
(psia)	1×10^{-8}
Flow Rate ($\frac{\text{Kilogram}}{\text{second}}$)	1.8144
(lbm/sec)[2], Minimum	4.00
Lockup Leakage, SCCS He[3]	TBD
External Leakage, SCCS He[4]	1×10^{-7}
Ambient Pressure Range ($\frac{\text{Newtons}}{\text{meter}^2}$)	1.014×10^{5} to 6.895×10^{-5}
(psia)	$14.7 \times 1 \times 10^{-8}$
Ambient Temperature Range (°K)	20.37 to 366.48
(°F)	-423 to +200
Inlet and Outlet Media and Temperatures	
GH_2 for service (°K)	33.15 to 366.48
(°F)	-400 to +200
GN_2 for purge media (°K)	88.70 to 366.48
(°F)	-300 to +200
GHe for purge media (°K)	20.37 to 366.48
(°F)	-423 to +200
Storage Life (seconds)	3.1536×10^{8}
(years)	10
Operational Life (seconds)	9.4608×10^{7}
(years)	3

Operating Life, cycles[5]

Modulating - grand total	253,000
Lockup - grand total	110
1. Ambient temperature and pressure	
a. Acceptance	
Modulating	1,000
Lockup	10
b. Pre-installation checkout	
Modulating	500
Lockup	5
c. Pre-launch checkout	
Modulating	200
Lockup	2
2. Cryogenic	
a. Acceptance	
Modulating	500
b. Pre-installation checkout	
Modulating	300
Lockup	3
3. Space Operation	
a. During engine operation	
Modulating	250,000
Lockup	80
b. Checkout at space station	
Modulating	500
Lockup	5

Nuclear Environment[6]	3×10^6 [7]	GAMMA KERMA Rate: RADS (carbon)/ hour.
	3×10^{11}	Fast Neutron Flux n/cm^2-sec (En > 0.9 MeV)

FUNCTIONAL REQUIREMENTS

1. Valve shall not be overly sensitive to vibration or acceleration in any axis. Launch-regulator not functioning - no structural failure

	[8] X Axis	Y Axis	Z Axis	Time
Sinusoidal vibration	± 3.0 g	± 4.5 g	± 4.5 g	300 seconds (5 min.)
	3 to 35 Hz	0.1 to 15 Hz	0.1 to 15 Hz	
Acceleration	5.2 g	0	0	300 seconds [9] (5 min.)
Space operation - regulator functioning				
Random vibration	TBD	TBD	TBD	3600 seconds (60 min.)
Acceleration	1.0 g	0.8 g	0.5 g	

2. Investigate replacement of the regulator by remote, in space, handling equipment. The regulator design must be compatible with the remote coupling concepts shown on ANSC Drawings 1139921 and 1139922.

(1) A typical regulator performance analogue is shown in Figure 1.

(2) In the case of the RNS, the ANSC NERVA engine supplies propellant tank pressurant GH_2 to the regulator with steady state inlet pressures at the engine supply point varying from 310 psia* to 723 psia* and

with temperatures varying from 238°R* to 315°R (Reference: ANSC N4110:0067, 26 February 1971, "State Points for the 1137400/Revision E Reference Engine"). Steady state flow rates vary from 0.20 lbm/sec* at 319 psia* to 0.60 lbm/sec* at 723 psia*. However, critical turbine discharge bleed flow rates for regulator sizing occur during engine bootstrap startup in a malfunction mode (one TPA operating during the second burn of an unmanned lunar mission) (Reference: ANSC Report S054-201, November 1970, "Pressurization Gas Requirements"). The peak or critical flow rates, temperatures and pressures in this report were based on a single 33.0 foot* diameter tank. Peak bleed flow rates were approximately 22 lbm/sec* at approximately 26.0 seconds after startup. The startup flow rates for the 16.0 foot* diameter multiple tank RNS (8 tanks) have been extrapolated from this report and appear to be on the order of 4.0 lbm/sec* at approximately 26 seconds after startup. The bleed temperatures and pressures versus startup time are assumed to be the same regardless of tank size. Disregarding temperature rise or pressure drop from the engine bleed point to the regulator inlet (due to the unknows of regulator location, line lengths and line heat leakage), the extremes of maximum regulator CA (coefficient of discharge times area) are shown in the following table. This table precludes an arbitrary intermediate size RNS regulator. This size was selected for the Phase II regulator design and will be shown on the revised ANSC Drawing 1139924. This size will allow for some changes in RNS performance parameters and will also allow for scaling up or down to finalized RNS tank configurations.

Tank Configuration	Fully Open Regulator CA At 7.0 Seconds After Startup	Flow Rate (Note A)	Partially Open Regulator CA at 26.0 Seconds After Startup (Note B)	Flow Rate (Note C)
Single Tank Class 1 RNS	59.6128 cm^2 9.2400 $in.^2$	6.16 Kg/s 13.57 lbm/sec	18.2322 cm^2 2.8260 $in.^2$	10.0 Kg/s 22.0 lbm/sec
Multiple Tank Class 3 RNS	7.9045 cm^2 1.2252 $in.^2$	1.11 Kg/s 2.45 lbm/sec	3.3155 cm^2 0.5139 $in.^2$	1.8 Kg/s 4.0 lbm/sec
Arbitrary Intermediate RNS	17.2258 cm^2 2.6700 $in.^2$	1.78 Kg/s 3.92 lbm/sec	7.2258 cm^2 1.1200 $in.^2$	2.9 Kg/s 6.4 lbm/sec

NOTES:

A. Conditions: 951,510.0 N/M^2 (138 psia)
95.55 °K (172°R)

B. 26.0 seconds after bootstrap startup = end of thrust buildup

C. Conditions: 4,585, 175.0 N/M^2 (665 psia)
147.22 °K (265°R)

---

*319 psia = 2.199 x 10^6 Newton's/Meter^2 absolute

723 psia = 4.985 x 10^6 " " "

100 psi = 6.895 x 10^5 Newton's/Meter^2

238°R = 132.222 Degree Kelvin

315°R = 175.000 " "

0.20 lbm/sec = 0.0907 Kilograms/sec

0.60 " " = 0.2722 " "

22.0 " " = 9.9792 " "

4.0 " " = 1.1844 " "

33.0 feet = 10.0584 meters

16.0 feet = 4.8768 "

(3) Lockup leakage will be determined by the lockup pressure requirement (TBD). Usually, regulators are accompanied by shutoff valves located upstream of the regulator inlet port.

(4) Per inch of sealing diameter or per 2.54×10^{-2} meters of sealing diameter.

(5) Life cycles of state-of-the-art regulators today are said to be 250,000 cycles with a two-year life. Modulating cycles are defined as the number of pulses of the main regulator metering element in either direction (without seating) due to varying downstream flow demand or changing inlet pressures and temperatures. Lockup cycles are the number of seating cycles of the main regulator metering element due to the lack of downstream flow demand. The 80 lockup cycles for space operation are based on 10 space station/moon/space station missions for RNS with a maximum of 8 burns per mission. The 250,000 modulating cycles are a ROM estimate based on 10 hours of engine operation with 80 burns.

(6) At engine stage interface per ANSC Memo N4230:6397M, E. A. Warman to A. D. Cornell, dated 22 January 1971, Subj: "Perturbed and Isokerma Rate Maps". Also, see LMSC Report A98455, Final Report, Volume VIII, for dose rates at stage tank bottom.

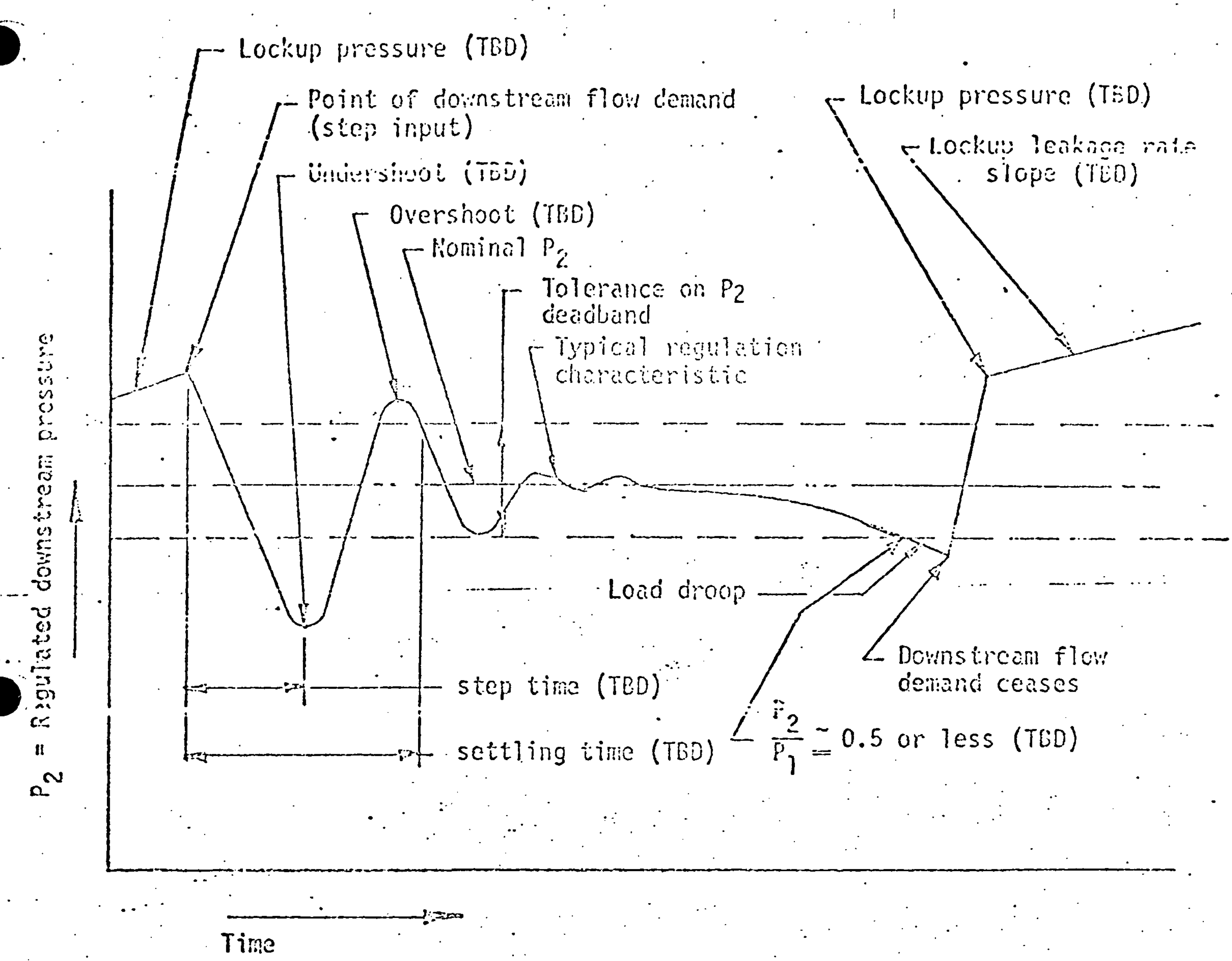
(7) Ten hours run time for total dosage.

(8) Longitudinal axis of RNS.

(9) Increases from 1.0 g to 5.2 g in 300 seconds (5 minutes).

(10) Proof and burst pressures are respectively the product of 1.5 and 2.5 times the maximum system inlet pressure.

(11) Proof and burst pressures are based on a maximum tank working pressure of 2.685×10^5 Newton's/Meter^2 absolute (30 psia) and are respectively the product of 1.5 and 2.5 times this pressure.



P_1 = inlet or upstream pressure 8.62×10^5 to 49.85×10^5 Newton/Meter^2 absolute) (125 to 723 psia).

$\frac{P_2}{P_1} \cong$ 0.5 or less to remain within deadband and is dependent on the sonic pressure ratio of the gaseous media, upstream system geometry, downstream system geometry, type of system and many other variables.

Nominal P_2 = $1.62 \times 10^5 \frac{\text{Newton's}}{\text{Meter}^2}$ absolute (23.5 psia) and may be programmed from 1.034×10^5 to $2.068 \times 10^5 \frac{\text{Newton's}}{\text{Meter}^2}$ absolute (15.0 to 30 psia)

Tolerance on P_2 Deadband = $\pm\ 1.034 \times 10^4 \frac{\text{Newton's}}{\text{Meter}^2}$ ($\pm$ 1.5 psia)

FIGURE 1

APPENDIX N

EVALUATION OF A MHD VALVE CONCEPT

WITH THEORETICAL CALCULATIONS

I. VALVE DESCRIPTION

A valve design in which magnetohydrodynamic (MHD) forces are used to modulate the flow was evaluated. This work was done under MSFC contract NAS 8-27568. In a typical design as shown in Figure 1, an ionized gas flows axially through an annular passage, a current passes circumferentially through the gas from the top electrode to the bottom electrode, and a radial magnetic field is produced by the outer windings. The current and magnetic field vectors are so aligned that the Lorentz force, $\overline{F}_L = \overline{j} \times \overline{B}$ acts in the axial direction opposing the flow. By increasing the current input, the retarding Lorentz force is increased reducing the flow through the valve.

The specific parameters used in this evaluation were:

D, diameter of annular passage - 7.0 inches

h, annular gap height - 0.018 inches

L, length of annular gap - 1.0 inches

B, magnetic field strength - 1.94 tesla (webers/m^2)

p, valve inlet pressure - 30 psia

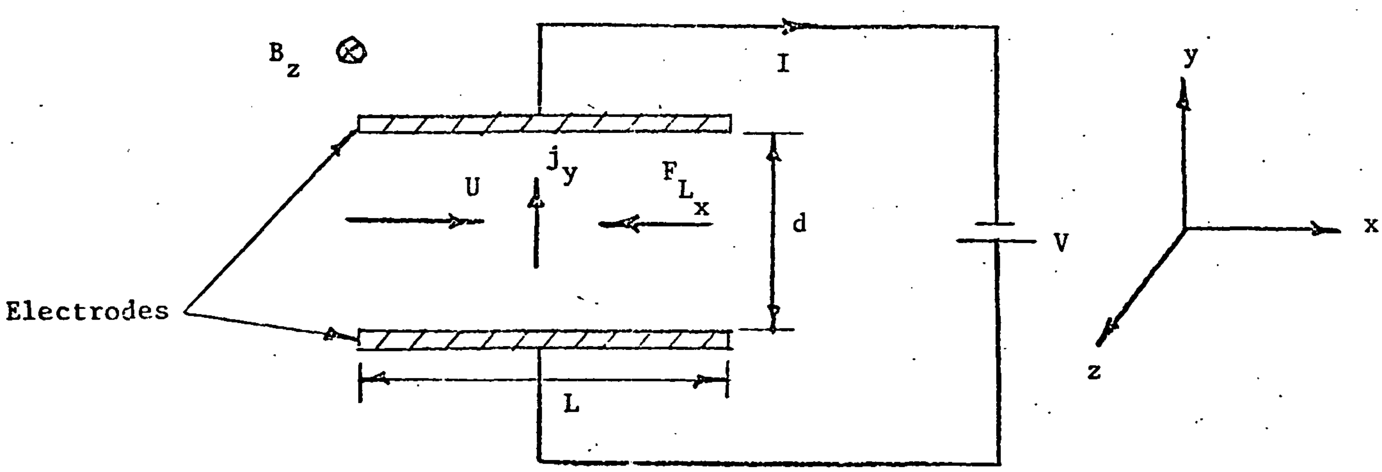
T, gas temperature - 530^{O}R

gas - hydrogen

$\dot{w}$, maximum flow rate - 0.01 lbm/sec

II. GOVERNING MHD RELATIONS

A schematic of the valve is shown below.



The electrodes are assumed to be continuous, thus shorting out the axial electric field, E_x. The electrode separation is the circumferential distance

$$d = \frac{\pi D}{2} = 11.0 \text{ inches or } 0.28 \text{ m.}$$

A. REQUIRED CURRENT

The axial Lorentz force per unit volume (newtons/m^3) is

$$F_{L_x} = j_y B_z \tag{1}$$

where j_y = the circumferential current density (amps/m^2)

B_z = the radial magnetic field strength (tesla)

Assuming F_{L_x} to act uniformly throughout the gap region, the equivalent pressure difference (newtons/m^2) is

$$\Delta p = \frac{F_{L_x} (AL)}{A} = j_y B_z L \tag{2}$$

where A is the cross-sectional flow area (m^2) and L is the gap length (m.). For conversion to the usual engineering units, 1 newton/m^2 = 0.000145 psi. To get a Δp of 30 psi (for complete shut-off, if possible) with B_z = 1.94 tesla and L = 1.0 inches, a current density of $j_y = 4.20 \times 10^6$ amps/m^3 is required. Assuming a uniform current density throughout the gap, the maximum circumferential current flow for the valve geometry being considered would be 48.6 amps.

B. REQUIRED VOLTAGE

To determine the input voltage V, Ohm's law for conducting gases must be used. From Reference (a) for $E_x = 0$, Ohm's law for the j_y component of current density is

$$j_y = \frac{\sigma}{1 + \beta^2} (E_y - UB_z) \tag{3}$$

where σ = the gas conductivity (mhos/m)

β = the Hall parameter, (cyclotron frequency/collision frequency)

E_y = the applied electric field - V/d, (volts/m)

U = the axial gas velocity, (m/sec)

UB = the induced electric field - $\vec{v} \times \vec{B}$, (volts/m)

Equation (2) is the equivalent form of I = V/R for gases, where R is the electrical resistance. In this case the resistance of the gas is considered in terms of the conductivity σ, which is the key to determining the applied voltage required to obtain the desired current density. For the conditions specified for this valve, the maximum induced field (i.e., UB at full flow) is 205 volts/m. Solving Equation (2) for E_y (with B_z acting in the negative z direction) gives

$$V = E_y d = 0.28 \frac{(1 + \beta^2)}{\sigma} j_y - 5.75 \tag{4}$$

The applied voltage can thus be obtained for a given j_y if the electrical properties of the gas (σ and β) are known. These properties are functions of the particular gases considered and the temperature of the gases. Since normal gases only ionize significantly at very high temperatures, a small amount ($\sim$ 1.0 mole percent) of an easily ionized substance, or seed, is usually added to the gas in most MHD applications to increase the conductivity to usable levels at reasonable temperatures. Examples of good seed material are cesium, potassium, sodium, and lithium, which have much lower ionization potentials than regular gases.

The sensitivity of the gas conductivity to temperature is illustrated in the table below for hydrogen at 30 psia seeded with 1 mole percent of cesium:

T(°R)	σ (mhos/m)	β
1000	$.341 \times 10^{-12}$	.645
2000	$.382 \times 10^{-3}$	.913
3000	.450	1.115
5000	117.4	1.149
10,000	1030	.680
15,000	1060	1.120

These values were calculated using the simplified method described on Page 25 of Reference (b), which considers only the electron current. In most MHD applications, the bulk of the current is carried by the electrons since due to their very low mass they react more readily to electric and magnetic fields than do the positive ions.

Note in the above table that at temperatures around 1000°R the gas conductivity, even when seeded, is extremely low. This would seem to rule out any possibility of operating with a gas at 530°R. However, these results hold for thermal equilibrium with all the constituents of the gas at the same temperature, and it is possible to have a non-equilibrium condition in which the electrons are at an effectively higher temperature than the gas. This higher electron temperature, and thus higher gas conductivity, can be achieved by such methods as the application of an electric field or radiation of the gas. The former is the most common and is used in such practical applications as the fluorescent light, where a discharge is maintained for low gas temperatures.

The effectiveness of an electric field in increasing the gas conductivity is strongly dependent of the gases used. For monatomic gases such as argon, neon, and mercury vapor, the collisions between the electrons and neutral atoms are very nearly elastic, and thus the electron loses little energy in traversing the gas flow field. However, for diatomic gases, the electron-molecule collisons are more inelastic since some of the collision energy can be absorbed by the diatomic molecule in vibrational or rotational modes. The difference in electron temperatures for monatomic and diatomic gases is illustrated in Figure 2 with information obtained from Reference (c).

Figure 2 shows that the elevation in electron temperature is a function of the parameter E/p. For hydrogen, an E/p value of 1.0 volt/cm-mm H_g would produce electron temperatures 9 times the gas temperature, which would give a reasonable conductivity (as shown in Table I). At the pressure of 30 psia or 1550 mm H_g, however, an electric field strength of 1550 volt/cm is required. For the circumferential gap length of the valve in Figure 1, a voltage of 43,300 volts would be necessary to attain this field strength. Hence, to attain reasonable conductivity values in hydrogen, very high voltages, which combined with the 48.6 amps. current requirement mean very high power input, must be used.

C. POWER INPUT

The external power required by the valve is

$$P = \vec{E}\cdot\vec{J}\ (\mathrm{Vol.}) = IV \tag{5}$$

where the current density $\vec{j}$ is determined by the desired Δp (Equation (2)), the field strength $\vec{E}$ by the conductivity (as discussed above), and the volume of the gap Vol. by the valve geometry. Note that the power input is independent of the orientation of $\vec{E}$ and $\vec{j}$.

III. THEORETICAL RESULTS

Using the idealized relations derived in the previous section, the effective Δp produced by the MHD forces was calculated as a function of power input. The results are shown in Figure 3 for various initial gas temperatures. The range of operation for the value in Figure 1 with hydrogen is bracketed by two limiting cases. The lower bound is the curve corresponding to a constant gas temperature of 530°R, which assumes none of the input energy has affected the gas temperature. The upper bound is the dashed curve that was generated by assuming an initial gas temperature of 530°R that is increased by the amount of electrical energy input to the gas. Since the gas temperature would actually be varying along the length of the electrodes, the valve would be expected to operate somewhere between these two bounds.

The primary results shown by Figure 3 is the extremely high power inputs required to get significant Δp levels. As previously discussed, this is due to the high electric field required to obtain reasonable gas conductivity values in room temperature hydrogen.

IV. LIMITATIONS OF ANALYSIS

The relations used to calculate the results shown in Figure 3 hold only for a highly idealized case in which the flow and electrical parameters are assumed to be constant and uniform throughout the annular gap. The effect of non-uniformities and the various loss mechanisms present in such a device would be to further increase the excessive power input levels.

In reality, it is most probable that an arc discharge would be produced in the gap between the electrodes, and thus the actual current density distribution would be highly non-uniform. At the relatively high pressures used in the valve and the high current levels required, Cobine in Reference (d) indicates that an arc discharge would definitely be expected. The arc is undesirable for this particular valve application since all the current is concentrated in a small diameter rod-like volume, and hence the MHD forces act only on a small portion of the flow. With an arc, the applied voltage is much lower because of the high gas conductivity in the arc region due to the very high local heating of the gas. Short distances away from the arc, however, the gas rapidly returns to the thermal equilibrium temperature determined by the energy input of the arc. Even within the small high temperature portion of the arc itself, Reference (d) shows that current densities in hydrogen at 1 atmosphere pressure are around 6×10^6 amps/m^2, which is of the same order as the current density required throughout the volume of the gap to get the required 30 psia Δp.

V. CONCLUSIONS

Based on an idealized representation of the interaction of MHD forces with a gas flow, it can be concluded that the valve shown in Figure 1 cannot effectively control the flow rate of room temperature hydrogen without the expenditure of excessive amounts of power. Deviations from the idealized case serve only to further increase power requirements. Changes in valve geometry and magnetic field strength might reduce the power an order of magnitude, but even greater reductions are necessary to reach acceptable levels. It is possible, however, that some other means might be available for ionizing the gas before it enters the valve, which could greatly reduce the power input to the valve. It is likely, though, that such means would also require significant power inputs.

It should be noted that the feasibility of the valve configuration shown in Figure 1 has been evaluated only for room temperature hydrogen. The valve concept is sound, though, and with better conducting fluids the power requirements may be within reasonable levels. For example, helium would be more subject to nonequilibrium ionization since it is monatomic, and thus it would be expected to have a stronger interaction with the MHD forces. Also, this type of valve would be particularly attractive for liquid metal applications since the fluid conductivity in such cases is very high.

An alternate concept that might be considered for using MHD forces to control flow is a vortex valve. The valve geometry shown in Figure 1 could be readily adapted to a vortex valve by aligning the $\vec{j}$ and $\vec{B}$ vectors to produce a circumferential Lorentz force in the annular gap. In this case, only a small Δp need be generated by the MHD forces to start the flow swirling. The swirl is increased in the chamber downstream of the gap as the flow approaches the exit at a smaller radius. The over-all Δp across the valve is a direct function of the swirl at the valve exit. A computer code described in Reference (e) is available at ANSC for calculating the valve exit flow as a function of the initial swirl of the flow entering the valve. Since a much smaller Lorentz force is required by the vortex valve concept, it is possible the probable arc discharge, which requires lower voltages and hence lower power input, might produce enough interaction with the over-all flow to make the valve feasible from a power input standpoint.

MHD VALVE DESIGN

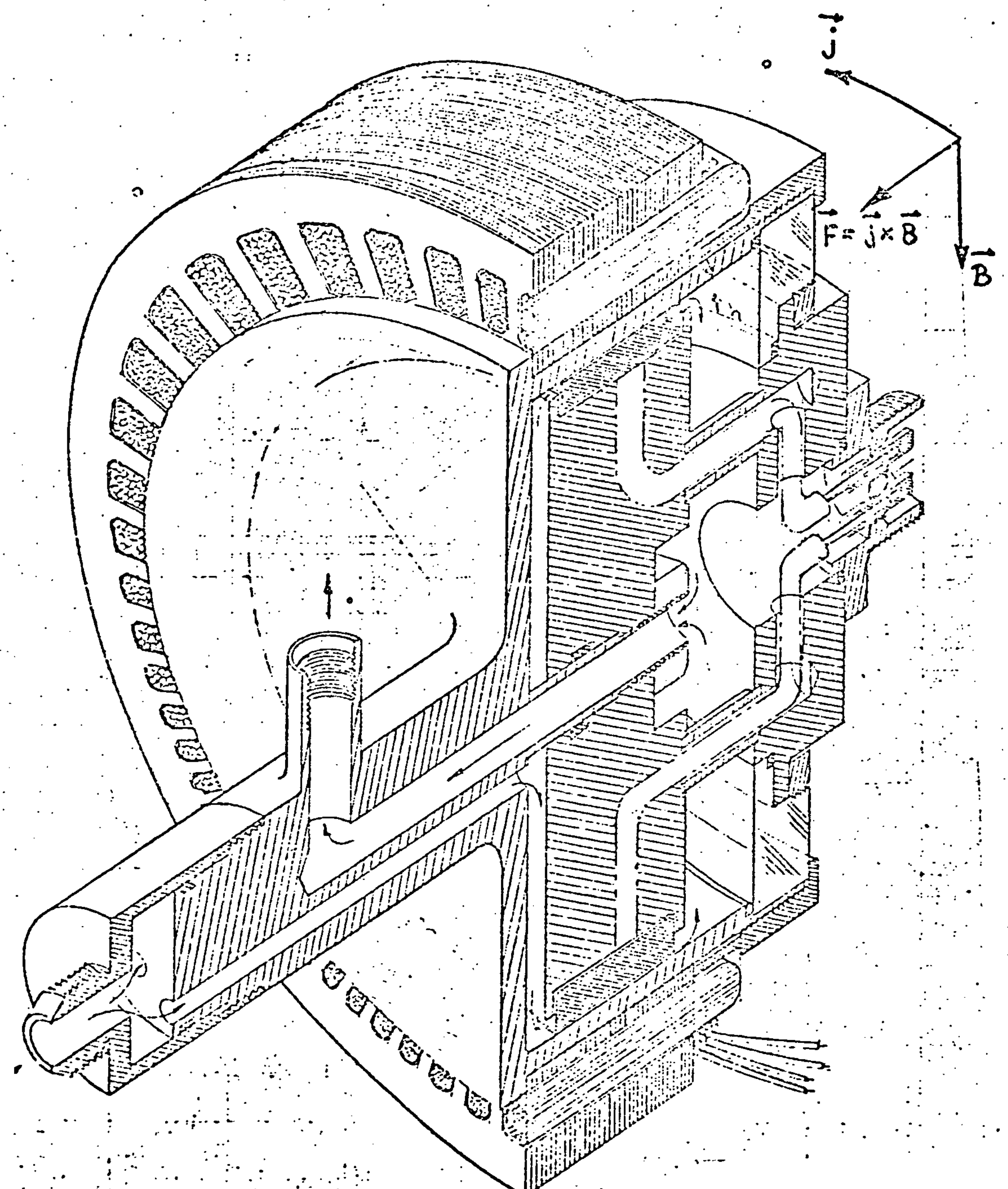


FIGURE 1

ELECTRON TEMPERATURE IN VARIOUS GASES

REF: S.C. BROWN, BASIC DATA OF PLASMA PHYSICS

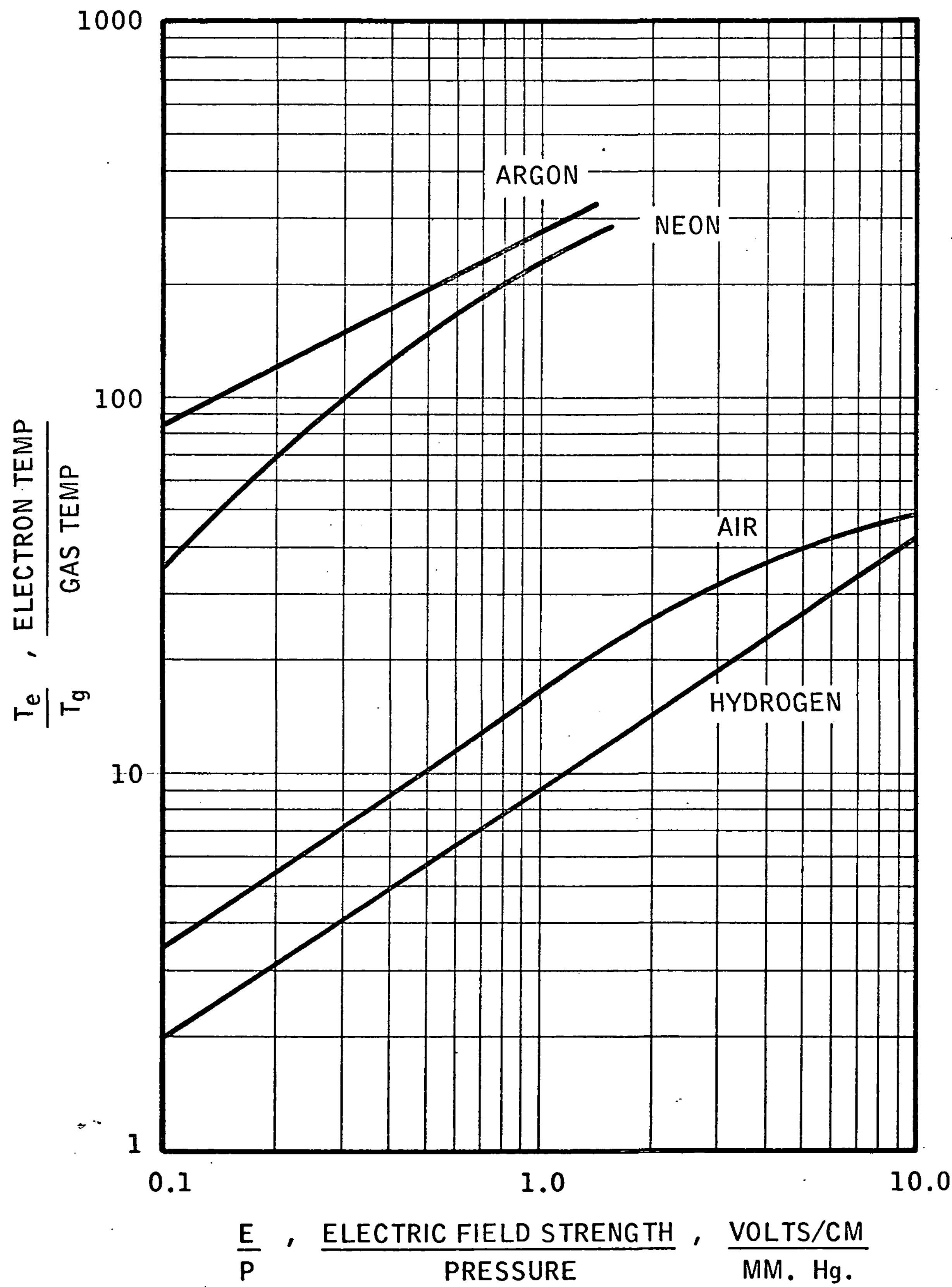


FIGURE 2

CALCULATED POWER REQUIREMENTS FOR MHD VALVE

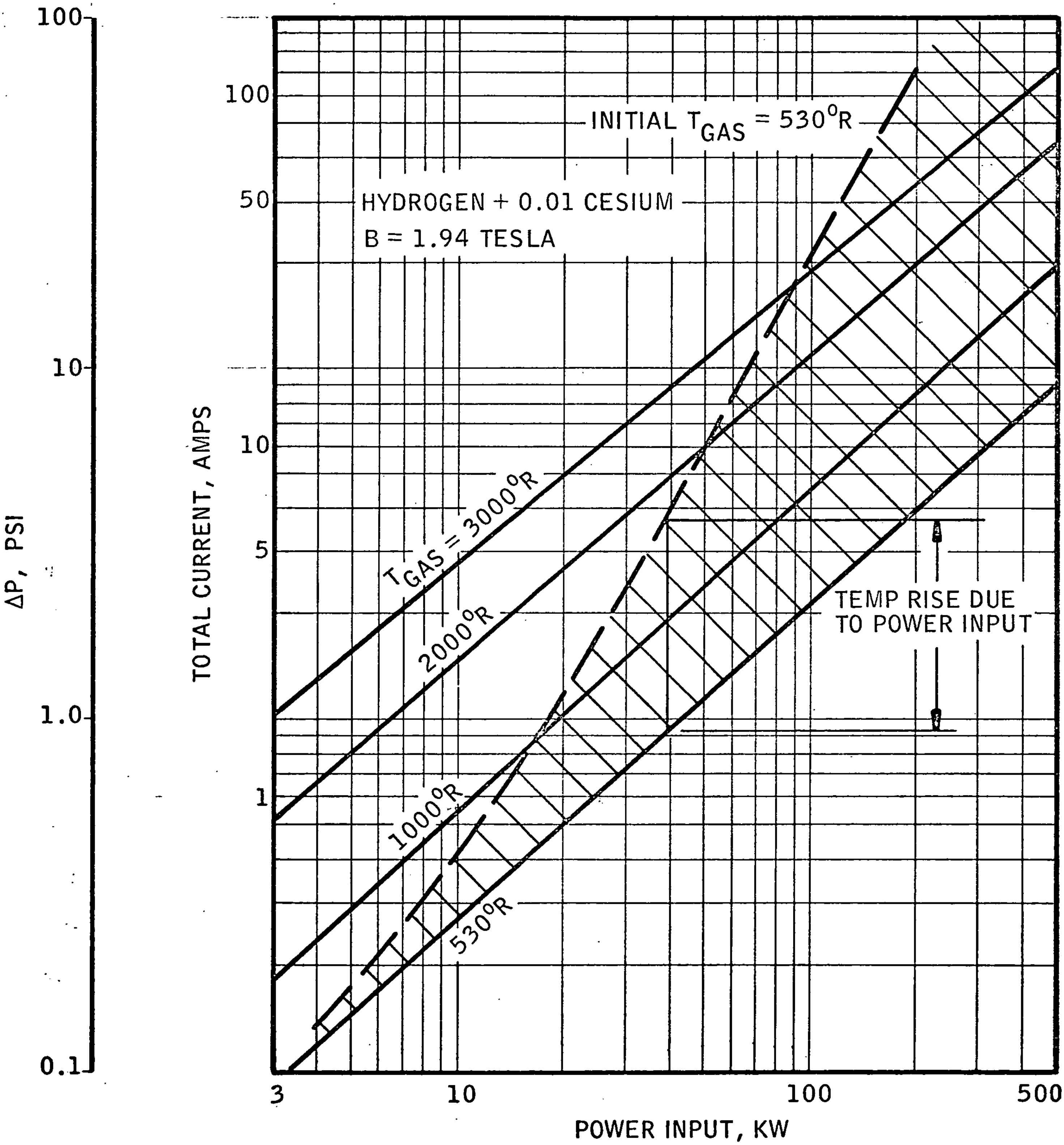


FIGURE 3

APPENDIX O

HEATER SIZING ANALYSIS FOR THE ELECTROMECHANICALLY ACTUATED POPPET VALVE WITH HIGH PRESSURE ENERGIZED SEAT

HEATER SIZING ANALYSES FOR THE ELECTROMECHANICALLY ACTUATED POPPET VALVE WITH HIGH PRESSURE ENERGIZED SEAT

I. INTRODUCTION

This report documents the heater sizing analyses for the Electromechanically Actuated Poppet Valve with High Pressure Energized Seat. This work was done under MSFC Contract NAS 8-27568 to provide support analyses for the evaluation of this valve seal concept.

II. SUMMARY/CONCLUSIONS

This LH_2 poppet valve concept has the ability to generate a very high load at the valve seat at the expense of a large boil-off rate of the LH_2 upstream of the valve seat. In order to decrease this heat load to the LH_2, the trapped volume must either be thermally insulated from the valve seat or must be located at a distance from the valve seat. The latter appears to be the only feasible solution since the member between the trapped volume and the valve seat must be a high strength material.

III. DISCUSSION

A steady-state thermal analysis of the poppet valve was performed using the D12207 finite element method program. The two-dimensional thermal model was generated in cylindrical coordinates with the axis of rotation at the valve centerline as shown on Figure 1. This model has 88 nodes and 64 elements. It was assumed that the trapped fluid and the fluid upstream of the valve seat is liquid hydrogen at 22.2°K (40°R) and that a vacuum exists downstream of the valve.

The thermal conductivities of the valve materials (CRES 302 and CRES 347) and the trapped liquid hydrogen were assumed to have constant values as shown below:

$$K_{SS} = 1.875 \times 10^{-2} \frac{\text{joule}}{\text{sec-cm-}^{\circ}\text{K}} \; (.5 \times 10^{-4} \text{ Btu/sec-in-}^{\circ}\text{R})$$

$$K_{LH_2} = 9.375 \times 10^{-4} \frac{\text{joule}}{\text{sec-cm-}^{\circ}\text{K}} \; (.25 \times 10^{-5} \text{ Btu/sec-in-}^{\circ}\text{R})$$

The heat transfer coefficient from the valve to the liquid hydrogen upstream of the valve seat of 1.0×10^{6} joule/hr-meter^{2}-°K (1.0×10^{-4} Btu/sec-in^{2}-°R) was used based on the film boiling data given in Reference 1.

IV. RESULTS

Shown on Figure 2 are the results of this analysis. The heater element, mean fluid and the maximum value metal steady-state temperatures are plotted as a function of the heater power into the trapped fluid. This heat is conducted through the valve body to the fluid upstream of the valve seat. Shown on the same Figure is the hydrogen boil-off rate as a function of the heater power.

The pressures generated by the trapped fluid were computed based on the data given in Reference 2 (reproduced on Figure 3). It was assumed that the initial fluid was at atmospheric pressure at 22.2°K (40°R). When the trapped LH_2 is heated as constant volume, it became more supercooled moving away from the saturated liquid line (see Figure 3). Also shown on Figure 2 is the pressure generated by the trapped LH_2 as a function of heater power. Since the Reference 2 data only extend to 10,000 psi, the extrapolation of this data is shown with a dotted line on Figure 2.

V. REFERENCES

1. "Boiling Heat Transfer for Oxygen, Nitrogen and Hydrogen Helium", NBS Technical Note No. 317, September 20, 1965

2. "Properties of Principal Cryogenics", Report No. 9050-111-65, Liquid Rocket Operation Aerojet-General Corp., Nov. 1965

POPPE[illegible] ALVE MODEL

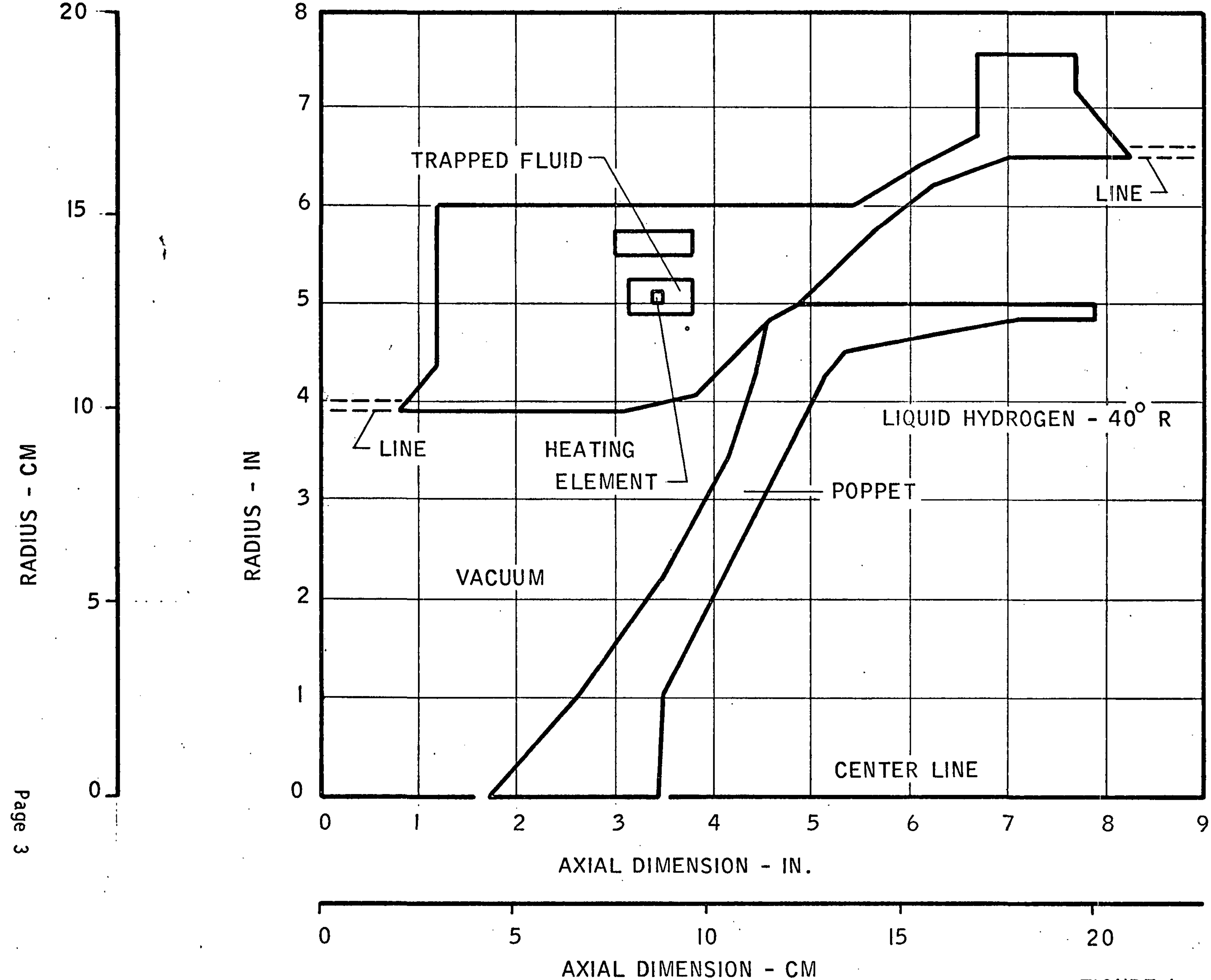


FIGURE 1

POPPET VALVE THERMAL ANALYSIS

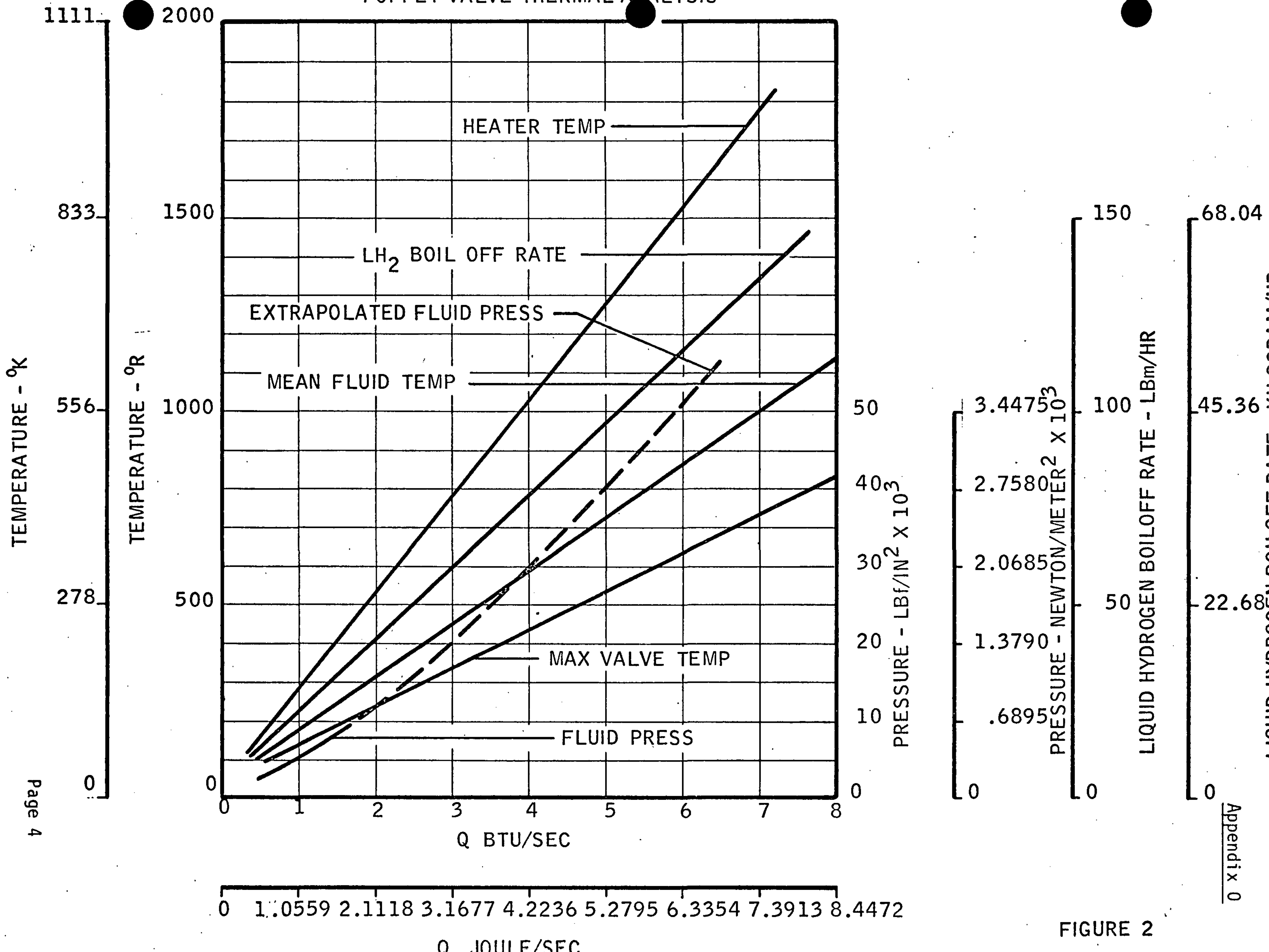
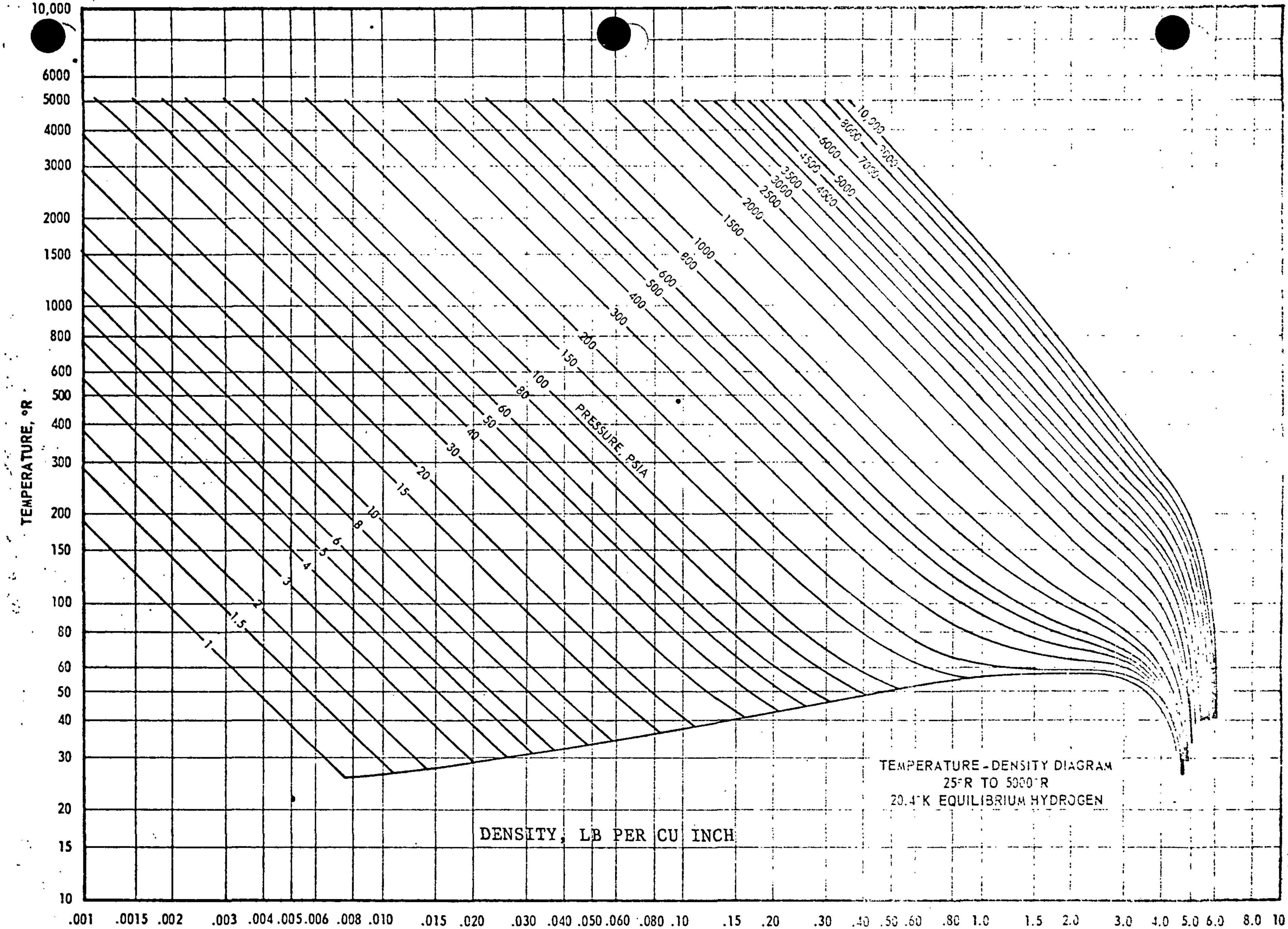


FIGURE 2



To Convert From	To Obtain	Multiply By
PSIA ($lbf/in.^2$)	$Newton/meter^2$	6895.0
°R	°K	5/9
LB PER CU FT (lbm/ft^3)	$Kilogram/meter^3$	16.018

FIGURE 3

APPENDIX P

CA CALCULATIONS FOR THE ANSC 1140007 ELECTROMECHANICALLY ACTUATED REGULATOR

U.S. customary units were used for the basic CA calculations. Final results were then converted to S.I. units where applicable.

The standard gas orifice flow equation was used as follows:

$$\frac{\dot{W}}{P_1} = \frac{8.02\ CA\ X}{R^{1/2}\ T^{1/2}}$$

where

$\dot{W}$ = Flow rate = lbm/sec

P_1 = Regulator inlet pressure - lbf/in.^{2} absolute

C = Coefficient of discharge = dimensionless

A = Regulator throat area = in.^{2}

R = Universal gas constant = $766.8\ \frac{\text{ft-lbf}}{°\text{R - lbm}}$ for GH_2

$R^{1/2}$ = 27.691

T = Inlet temperature of gas at regulator = °R

$$X = \left[\frac{K}{K-1}\left(\frac{P_1}{P_2}\right)^{2/K} - \left(\frac{P_1}{P_2}\right)^{\frac{K+1}{K}}\right]^{1/2} = .484 \text{ for sonic } GH_2$$

K = Ratio of specific heats = 1.4 for GH_2

P_2 = Regulator downstream pressure = $P_2 \leqq .532\ P_1$

Then for sonic flow across the regulator

$$\dot{W} = 0.14\ \frac{P_1\ CA}{T^{1/2}} \qquad (1)$$

or

$$CA = 7.1337\ \frac{\dot{W}\ T^{1/2}}{P_1} \qquad (2)$$

Steady state conditions were then tabulated in Table 1 using state point 36 of Reference 16. State point 36 is located downstream of the turbine discharge state point. Table 2 was then established using turbine discharge flow rates, temperatures and pressures versus startup times extrapolated from Figures[35] and [36] (Section II.B.2) for the multiple tank RNS. The CA values versus startup times shown in Table 2 were then computed using equation 2 and the extreme NERVA steady state design points of Table 1. The extreme steady state design points are also shown in Table 2. Startup time versus regulator CA was plotted in Figure A for the multiple tank RNS. An examination of Figure A shows that the maximum CA demand on the regulator occurs approximately 7.0 seconds after the initiation of engine startup and the lowest CA demand occurs during the extremes of steady state operation. The curve from approximately 4.0 to 26.0 seconds can be rescaled by the following equation for any RNS configuration assuming temperatures and pressures remain constant.

$$\frac{\dot{W}_1}{\dot{W}_2} = \frac{CA_1}{CA_2} \tag{3}$$

The maximum turndown ratios of the regulators can be determined by the ratio of the maximum CA to the minimum CA such as

$$\frac{CA_{max}}{CA_{min}} = \text{Turn down ratio} \tag{4}$$

Thusly, Table 3 parameters was established for three different regulator sizes using Equations 1, 2, 3 and 4 data from Tables 1 and 2, Figure A and data from Section II.B.2. The single tank RNS and the multiple tank RNS dictate the extremes in tank pressurant regulator size. The regulator size shown on ANSC drawing 1140007 (Figure 34) and as discussed in Section II.B2 is for an arbitrary intermediate size RNS. This regulator is designed for a full open geometric flow area of 20.27 cm^2 (π $in.^2$) and a 7.62 cm (3.00 in.) line size. Further analysis of Table 3, would indicate that for the Class 1 RNS (large single tank), a turndown ratio of 128.7:1 will possibly require two sizes of tank pressurant regulators.

Figure B shows the NERVA engine operational phases and Figure C shows the NERVA engine schematic. These figures are included for general reference.

TABLE 1

STEADY STATE CONDITIONS

(REFERENCE: STATE POINT 36 OF REFERENCE 36)

U.S. CUSTOMARY UNITS			
NERVA Engine Design Point	Flow Rate lbm/sec	Absolute Pressure $lbf/in.^2$	Absolute Temperature °R
Start of Life - Normal	$0.60\,^{0.66}_{0.54}$	$705\,^{718}_{693}$	$255\,^{266}_{244}$
End of Life - Normal	$0.60\,^{0.66}_{0.54}$	$710\,^{723}_{698}$	$265\,^{275}_{254}$
Start of Life - 1 TPA 80%	$0.40\,^{0.46}_{0.34}$	$574\,^{588}_{560}$	$271\,^{284}_{258}$
End of Life - 1 TPA 80%	$0.40\,^{0.46}_{0.34}$	$577\,^{591}_{564}$	$282\,^{295}_{268}$
Start of Life - 60%	$0.31\,^{0.37}_{0.25}$	$441\,^{452}_{430}$	$296\,^{306}_{286}$
End of Life - 60%	$0.31\,^{0.37}_{0.25}$	$443\,^{454}_{432}$	$305\,^{315}_{295}$
Start of Life - 1 TPA 60%	$0.31\,^{0.37}_{0.25}$	$440\,^{451}_{429}$	$295\,^{305}_{285}$
End of Life - 1 TPA 60%	$0.31\,^{0.37}_{0.25}$	$443\,^{454}_{432}$	$305\,^{315}_{295}$
Emergency Operating	$0.20\,^{0.22}_{0.18}$	$330\,^{341}_{319}$	$248\,^{258}_{238}$

SYSTEM INTERNATIONAL UNITS			
NERVA Engine Design Point	Flow Rate Kilogram/sec	Absolute Pressure $Newton/Meter^2 \times 10^6$	Absolute Temperature °K
Start of Life - Normal	$0.272\,^{0.300}_{0.245}$	$4.861\,^{4.951}_{4.778}$	$141.67\,^{147.78}_{135.55}$
End of Life - Normal	$0.272\,^{0.300}_{0.245}$	$4.895\,^{4.985}_{4.813}$	$147.27\,^{152.78}_{141.11}$
Start of Life - 1 TPA 80%	$0.181\,^{0.209}_{0.154}$	$3.958\,^{4.054}_{3.861}$	$150.55\,^{151.78}_{143.33}$
End of Life - 1 TPA 80%	$0.181\,^{0.209}_{0.154}$	$3.978\,^{4.075}_{3.889}$	$156.67\,^{163.89}_{148.89}$
Start of Life - 60%	$0.141\,^{0.168}_{0.113}$	$3.041\,^{3.117}_{2.965}$	$164.44\,^{170.00}_{158.89}$
End of Life - 60%	$0.141\,^{0.168}_{0.113}$	$3.055\,^{3.130}_{2.979}$	$169.44\,^{175.00}_{163.89}$
Start of Life - 1 TPA 60%	$0.141\,^{0.168}_{0.113}$	$3.034\,^{3.110}_{2.958}$	$163.89\,^{169.44}_{158.33}$
End of Life - 1 TPA 60%	$0.141\,^{0.168}_{0.113}$	$3.055\,^{3.117}_{2.965}$	$169.44\,^{175.00}_{163.89}$
Emergency Operating	$0.090\,^{0.100}_{0.081}$	$2.275\,^{2.351}_{2.200}$	$137.78\,^{143.33}_{132.22}$

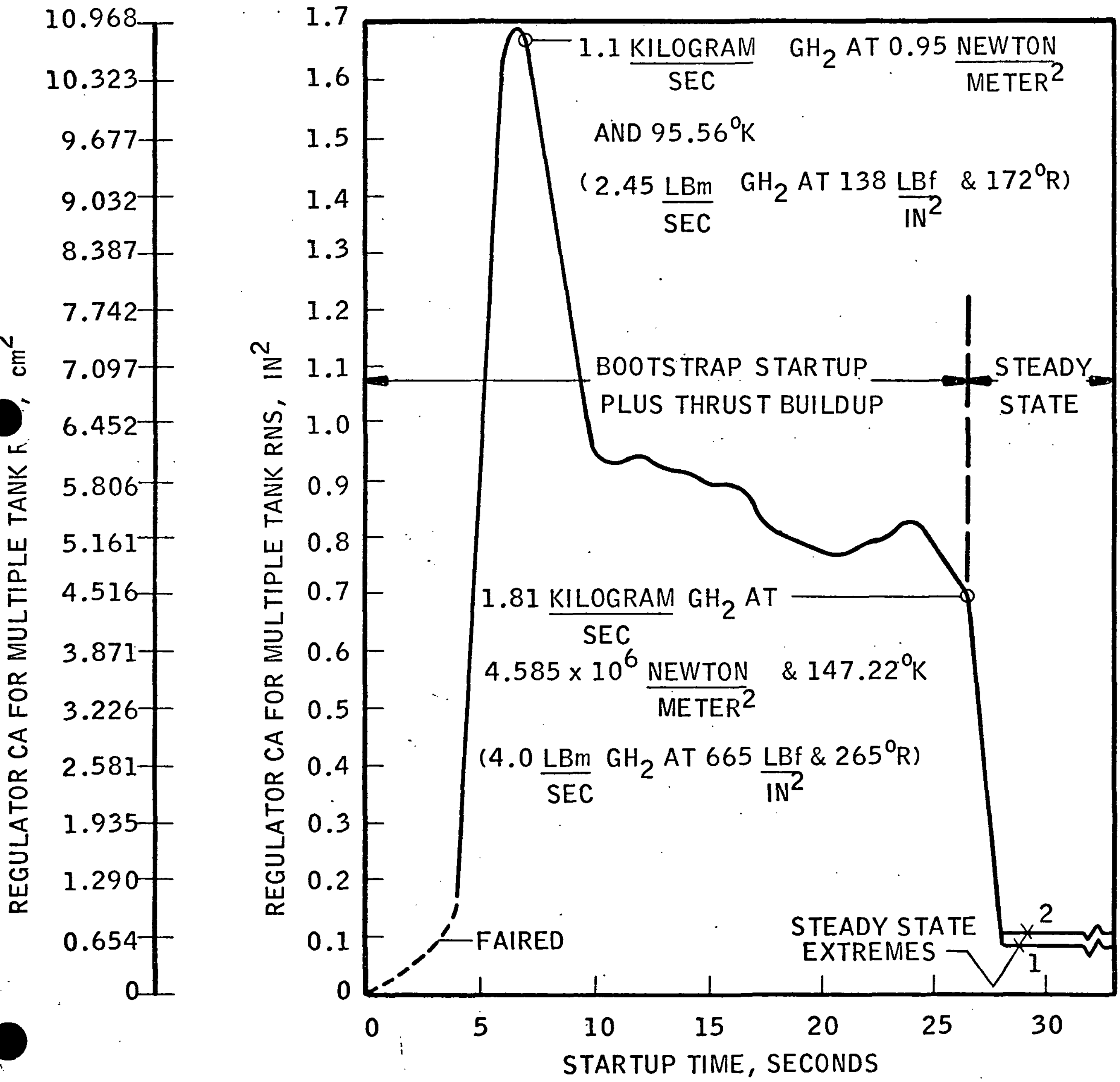

REGULATOR CA FOR MULTIPLE TANK R, cm2
10.968
10.323
9.677
9.032
8.387
7.742
7.097
6.452
5.806
5.161
4.516
3.871
3.226
2.581
1.935
1.290
0.654
0
REGULATOR CA FOR MULTIPLE TANK RNS, IN2
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1.1 KILOGRAM/SEC GH2 AT 0.95 NEWTON/METER2
AND 95.56°K
(2.45 LBm/SEC GH2 AT 138 LBf/IN2 & 172°R)
BOOTSTRAP STARTUP PLUS THRUST BUILDUP
STEADY STATE
1.81 KILOGRAM/SEC GH2 AT
4.585 x 10^6 NEWTON/METER2 & 147.22°K
(4.0 LBm/SEC GH2 AT 665 LBf/IN2 & 265°R)
FAIRED
STEADY STATE EXTREMES
2
1
0
5
10
15
20
25
30
STARTUP TIME, SECONDS

FIGURE A

TABLE 2

NERVA Startup Time (Seconds)	Flow Rate ($\dot{W}$) (lbm/sec)	Absolute Temperature (T) (°R)	$(T)^{1/2}$ $(°R)^{1/2}$	Upstream Absolute Pressure (P_1) (lbf/in.^{2})	CA (in.^{2})	CA (cm^2)
U.S. CUSTOMARY UNITS						SI UNITS
4	.16	229	15.13	116	.1489	0.9606
5	1.00	210	14.49	125	.8629	5.5671
6	2.15	192	13.86	130	1.6352	10.5497
7	2.45	172	13.15	138	1.6654	10.7445
8	2.50	152	12.33	150	1.4660	9.4580
9	2.52	148	12.17	186	1.1762	7.5884
10	2.50	150	12.25	234	.9336	6.0232
11	2.47	152	12.33	234	.9285	5.9903
12	2.48	156	12.49	234	.9443	6.0922
13	2.45	164	12.81	242	.9252	5.9690
14	2.45	172	13.15	252	.9120	5.8838
15	2.45	180	13.42	262	.8952	5.7755
16	2.42	186	13.64	264	.8920	5.7548
17	2.48	194	13.93	290	.8498	5.4826
18	2.50	200	14.14	312	.8083	5.2148
19	2.56	208	14.42	334	.7885	5.0871
20	2.60	215	14.66	350	.7769	5.0122
21	2.64	222	14.90	365	.7688	4.9600
22	2.80	228	15.10	383	.7875	5.0806
23	3.00	235	15.33	410	.8002	5.1626
24	3.30	243	15.59	444	.8226	5.3071
25	3.50	252	15.87	500	.7925	5.1129
26	3.84	264	16.25	620	.7180	4.6322
26.5	4.00	265	16.28	665	.6986	4.5071
1	0.20	258	16.06	319	.6986	0.4632
2	0.60	266	16.31	693	.1007	0.6497

1 Emergency steady state operating point (Table 1)

2 Start of Life steady state normal operating point (Table 1)

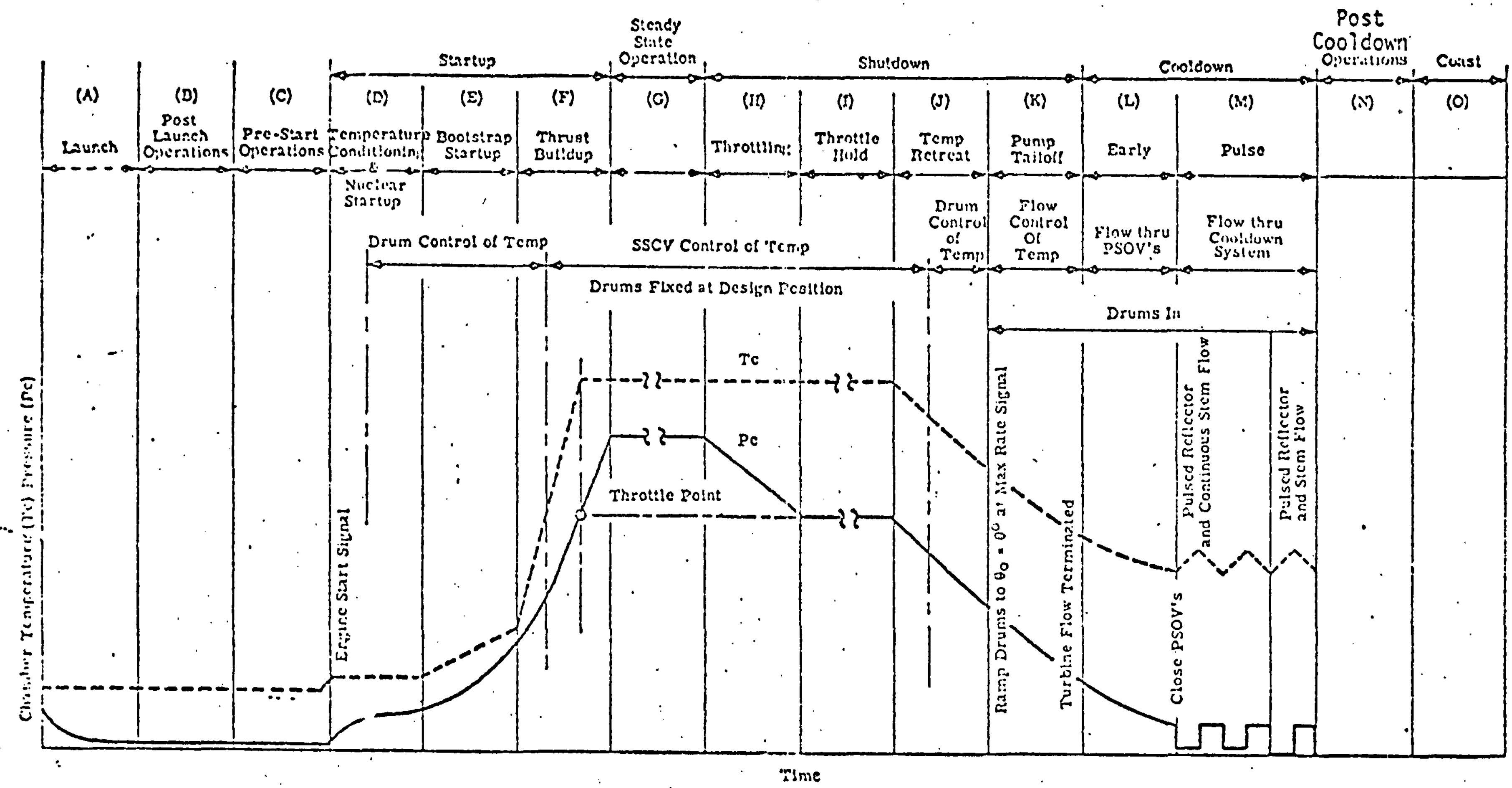
TABLE 3

U.S. CUSTOMARY UNITS

RNS Tank Configuration	Fully Open Regulator CA ($in.^2$) at Approx. 7.0 Seconds	Flow Rate lbm/sec @ 138 psia & 172°R	Partially Open Regulator CA ($in.^2$) at Approx. 26.0 Seconds	Flow Rate lbm/sec @ 665 psia & 265°R	Turn Down Ratio
Single Tank Class 1 RNS	9.2400	13.57	2.8260	22.0	128.7:1.0
Multiple Tank Class 3 RNS	1.2252	2.45	0.5139	4.0	24.3:1.0
Arbitrary Intermediate RNS	2.6700	3.92	1.1200	6.4	37.2:1.0

SYSTEM INTERNATIONAL UNITS

RNS Tank Configuration	Fully Open Regulator CA (cm^2) at Approx. 7.0 Seconds	Flow Rate Kilogram/sec @ 0.95 X 10^6 $\frac{Newton}{Meter^2}$ & 95.56°K	Partially Open Regulator CA (cm^2) at Approx. 26.0 Seconds	Flow Rate Kilogram/sec @ 4.585 X 10^6 $\frac{Newton}{Meter^2}$ & 147.22°K	Turn Down Ratio
Single Tank Class 1 RNS	59.6128	6.16	18.2322	9.98	128.7:1.0
Multiple Tank Class 3 RNS	7.9045	1.11	3.3155	1.81	24.3:1.0
Arbitrary Intermediate RNS	17.2258	1.78	7.2258	2.90	37.2:1.0



NERVA Engine Operational Phases

FIGURE B

AEROJET NUCLEAR SYSTEMS COMPANY

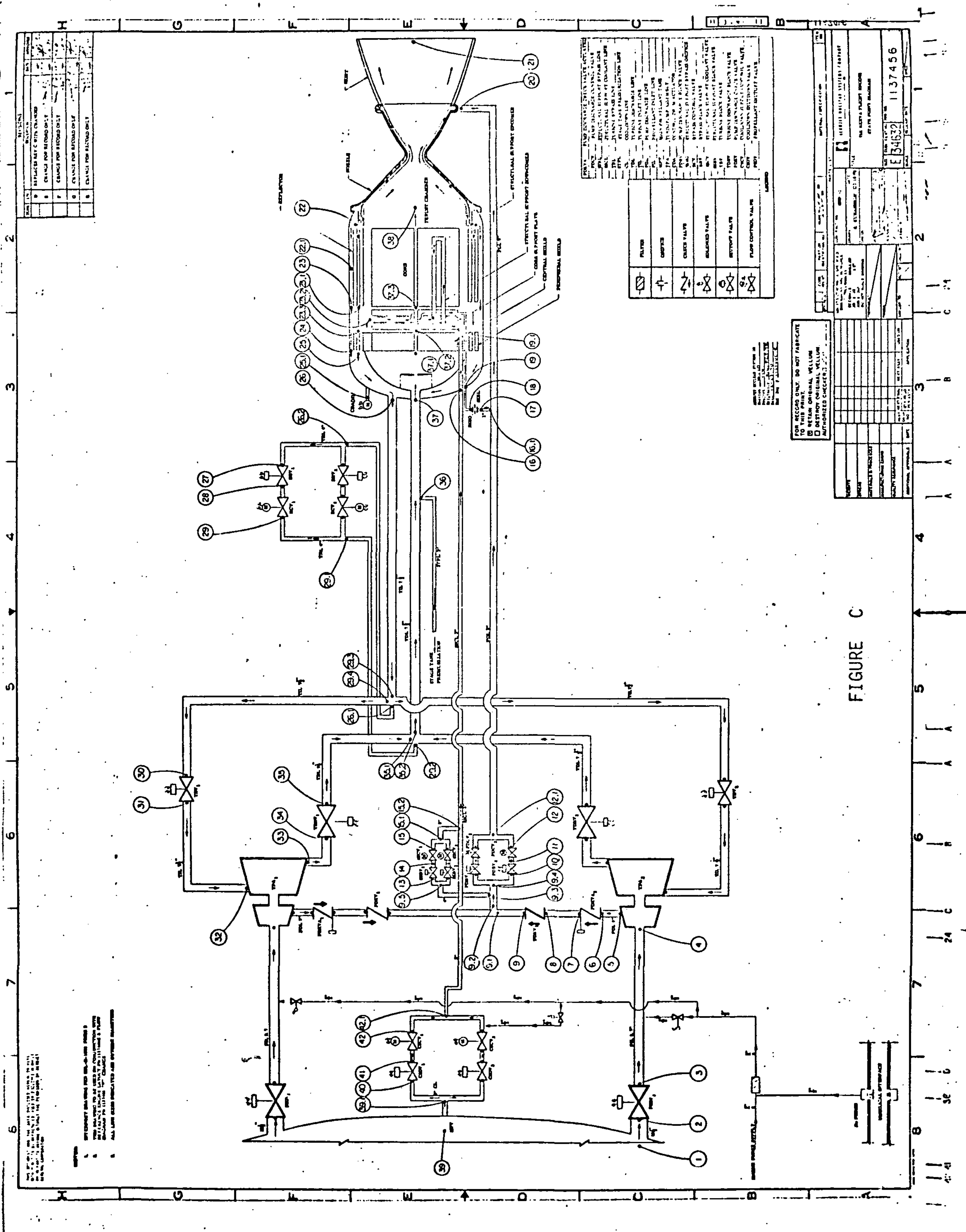


FIGURE C

APPENDIX Q

LOW MELTING ALLOY
REMOTE COUPLING
ELECTRICAL POWER REQUIREMENT VERSUS HEATUP TIME ANALYSIS
FOR
ANSC 1140005 LOW MELTING ALLOY REMOTE COUPLING

I. INTRODUCTION

The purpose of this report is to present a parametric analysis for sizing this heater for the Low Melting Alloy Remote Coupler. This work was done under MSFC Contract NAS 8-27568 to provide thermal support analyses for the evaluation of this remote coupling concept.

II. SUMMARY/CONCLUSIONS

Thermal transient analysis of one of the four remote coupling designs was performed on a detailed analytical model. The heater power requirements and predicted heat up times of the other three designs were estimated based on geometric similarities in the four design configurations. The heat transfer rate from the sleeve to the line is limited to thermal radiation, which is a poor mechanism at low temperatures. The maximum temperature of the line coupler is limited by the material of the barrier seals between the sleeve and line to approximately 644°K (700°F). A typical heat up time is 45 minutes at a heater power flux of approximately 1.55 watts/cm^2 (10 watts/in.^{2}).

III. DISCUSSION

A. METHOD OF ANALYSIS

The thermal transient analysis of the remote coupling heater and the stored alloy heater were performed using the D12207 finite element method program. The two dimensional thermal models were generated using cylindrical coordinates with the axis of rotation at the centerline of the joint to be coupled.

The remote coupling heater thermal model is shown on Figure 1. In outer space, the only means of transferring heat from the sleeve to the line is by thermal radiation. It was assumed that the surface emissivity of the line and sleeve was 0.9 in this analysis. A strap heater is in contact with the sleeve O.D.

The contact pressure is supplied by a massive clamp through 0.254 cm (0.1 inch) of resilient insulator pad. It was assumed that all exposed surface ($\varepsilon = 0.5$) radiates to outer space at 0°K.

This same remote coupling thermal model was also used to predict the thermal transient response on earth. The model was modified to include air in the 0.013 cm (5 mil) gap between sleeve and the line. The natural convection coefficient on the outside of the line and clamp was taken as 40,916 $\frac{\text{joule}}{\text{hr-meter}^2\text{-°K}}$ (2.0 BTU/hr-ft^2°R).

The thermal model of the alloy storage container is shown on Figure 2. The alloy is contained in a cylinder with a strap heater on the outside. It was assumed that the exposed surface ($\varepsilon = 0.5$) radiates to outer space at 0°K.

B. MATERIAL PROPERTIES

The nominal values of thermal conductivity and heat capacitance of AISI 347 reported in Reference 1 were used in this analysis. The thermal conductance and capacitance of the strap heater was assumed to be one-half the AISI 347 values.

The thermal properties of the aluminum clamp were taken to be independent of temperature as follows:

$$K = 1.50 \frac{\text{joule-cm}}{\text{sec-cm}^2\text{-°K}} \qquad \left(0.002 \frac{\text{Btu-in.}}{\text{sec-in.}^2\text{-°R}}\right)$$

$$\rho\, C_p = 1.16 \frac{\text{joule}}{\text{cm}^3\text{-°K}} \qquad \left(0.01 \frac{\text{Btu}}{\text{in.}^3\text{-°R}}\right)$$

The thermal properties of the rubber insulation were taken as independent of temperature as follows:

$$K = 0.0015 \frac{\text{joule-cm}}{\text{sec-cm}^2\text{-}^\circ\text{K}} \qquad \left(0.000002 \frac{\text{Btu-in.}}{\text{sec-in.}^2\text{-}^\circ\text{R}}\right)$$

$$\rho\, C_p = 2.32 \frac{\text{joule}}{\text{cm}^3\text{-}^\circ\text{K}} \qquad \left(0.02 \frac{\text{Btu}}{\text{in.}^3\text{-}^\circ\text{R}}\right)$$

The thermal properties of the alloy CERROBASE was obtained from the Cerro Copper Brass Company data sheet as follows:

$$K = 0.172 \frac{\text{joule-cm}}{\text{sec-cm}^2\text{-}^\circ\text{K}} \qquad \left(0.00023 \frac{\text{Btu-in.}}{\text{sec-in.}^2\text{-}^\circ\text{R}}\right)$$

$$\rho\, C_p = 1.32 \frac{\text{joule}}{\text{cm}^3\text{-}^\circ\text{K}} \qquad \left(0.0114 \frac{\text{Btu}}{\text{in.}^3\text{-}^\circ\text{R}}\right) \text{ solid}$$

$$\text{Heat of Fusion} = 16{,}760 \frac{\text{joule}}{\text{kilogram}} \qquad \left(7.2 \frac{\text{Btu}}{\text{lbm}}\right)$$

$$\rho\, C_p = 1.86 \frac{\text{joule}}{\text{cm}^3\text{-}^\circ\text{K}} \qquad \left(0.016 \frac{\text{Btu}}{\text{in.}^3\text{-}^\circ\text{R}}\right) \text{ liquid}$$

Melting point = 397°K (255°F) eutectic

During the 2-phase melting process, a modified high thermal capacitance value of $31.32 \frac{\text{joule}}{\text{cm}^3\text{-}^\circ\text{K}}$ (0.27 Btu/in.^{3}-°R) was used between the temperatures of 394 and 400°K (250 and 260°F) to account for the heat of fusion.

IV. RESULTS

Shown on Figure 2 are the results of the alloy heating thermal transient analysis. The heater power was 268 watts and the volume of CERROBASE alloy was 303.2 cm^3 (18.5 in.^{3}). The heat flux of the heater on the outside of the containment cylinder was 1.29 watts/cm^2 (8.32 watts/in.^{2}). The time required to heat up the solid alloy from 144°K (-200°F) to liquid at 422°K (+300°F) was 24 minutes. Figure 2 shows that the maximum temperature difference between maximum heater temperature at Node #18 and minimum alloy temperature at Node #5 is less than 55.6°K (100°R) throughout the heat up transient. For estimating purposes, it can be assumed that the alloy heat up time is inversely proportional to heat power and the maximum temperature difference between the heater and alloy is directly proportional to the heater power.

Shown on Figure 3 are the predicted sleeve-line thermal response in outer space. This analysis was performed at three different heater powers of 2.51, 1.88 and 1.26 watts/cm^2 (16.2, 12.15 and 8.1 watts/in.^{2}). The heater power and the corresponding heat up time to 422°K (760°R) (line) of 4 different sleeve-line designs were required. Since all four designs had the same thickness of sleeve, pipe, heater and clamp only one of the designs was modeled and analyzed. Since the material thicknesses are all the same, the thermal transient response is a function of heat flux of the heater. Shown below are the dimensions of the four remote coupling designs:

Coupler Design	Line Diameter		Sleeve Length	
	cm	in.	cm	in.
-1	7.62	3	11.43	4.5
-2	12.70	5	13.46	5.3
-3	20.32	8	16.76	6.6
-4	7.62	3	18.54	7.3

Tabulated on Figures 3 and 4 are the heater power of the four remote couplings designs at the 3 heater power flux values.

Shown on Figure 3 are the predicted thermal transient response of the line coupler at the same power fluxes in air. In all 3 heat flux cases, essentially steady-state temperatures are reached in one hour. This equilibrium temperature of the heater is a strong function of the natural convection heat transfer coefficient to ambient air. The value used in this analysis was $40{,}916 \dfrac{\text{joule}}{\text{hr-meter}^2\text{-}^\circ\text{K}}$ (2 Btu/hr-ft^2-$^\circ$R). If the actual natural convection coefficient is only $20{,}458 \dfrac{\text{joule}}{\text{hr-meter}^2\text{-}^\circ\text{K}}$ (1 Btu/hr-ft^2-$^\circ$R), the equilibrium temperature difference between the heater and ambient would be doubled.

V. REFERENCE

(a) Materials Properties Book, ANSC Report No. 2275

LINE HEATER MODEL

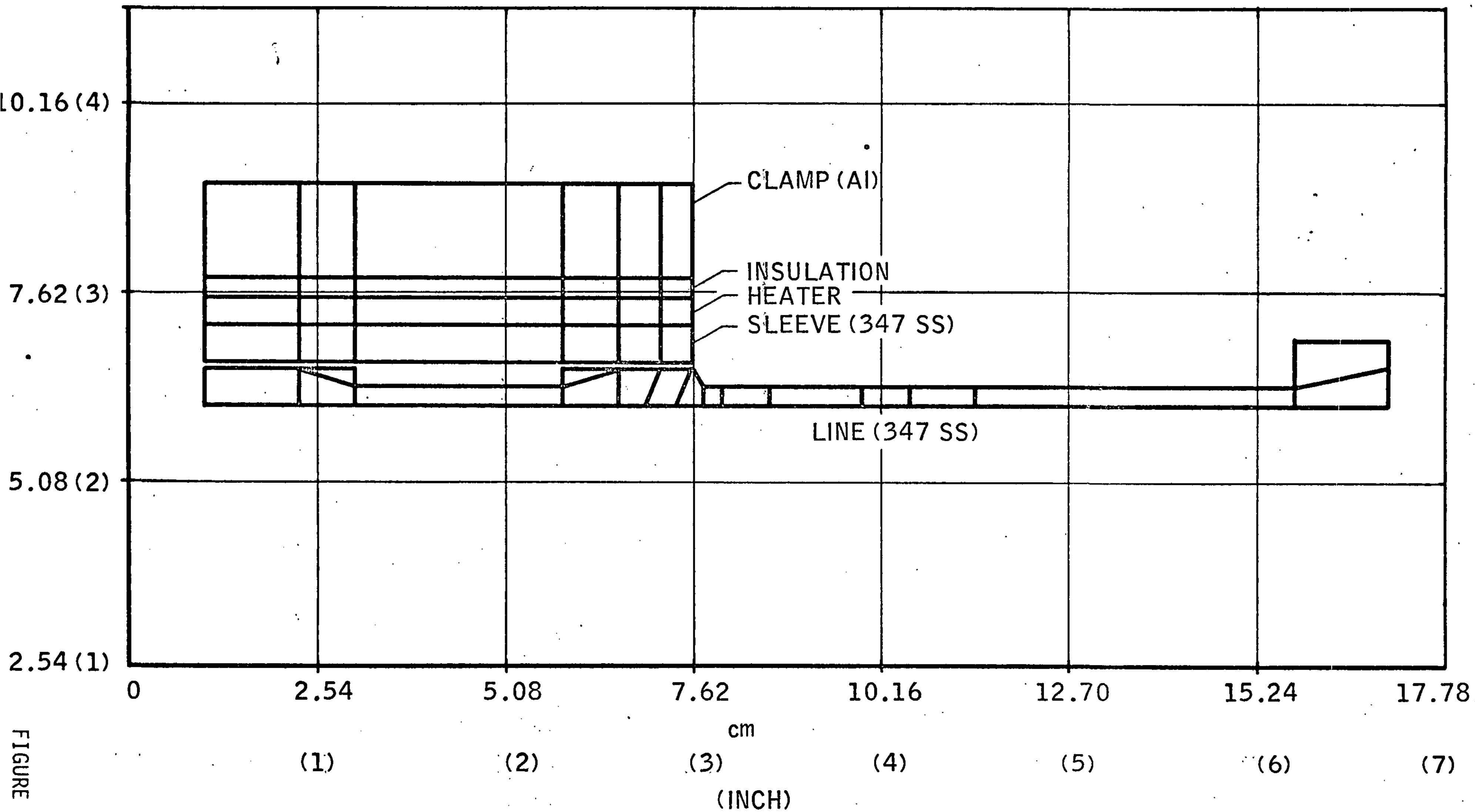


FIGURE 1

ALLOY STORAGE CYLINDER
THERMAL TRANSIENT

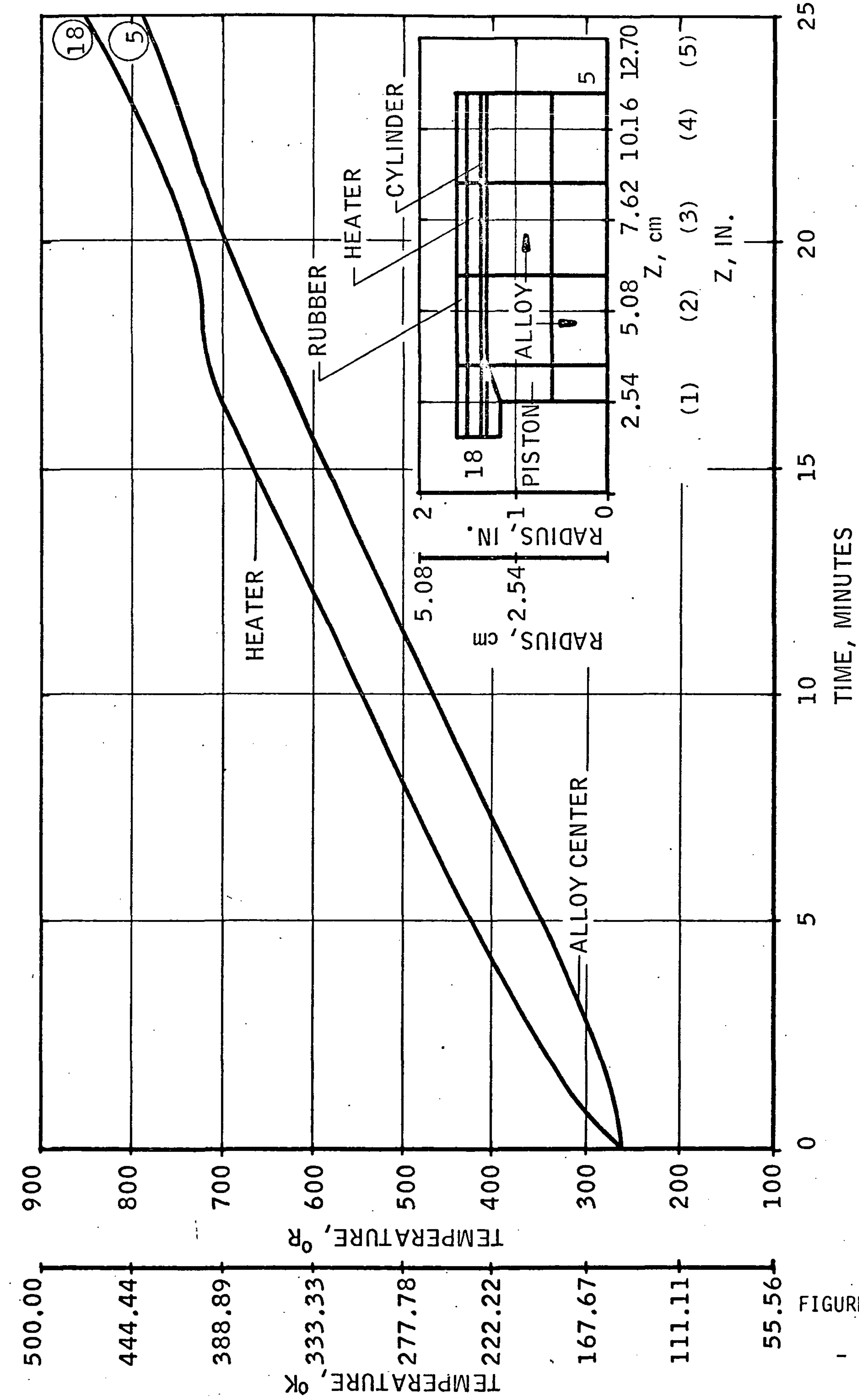


FIGURE 2

LINE HEATER THERMAL TRANSIENT IN SPACE

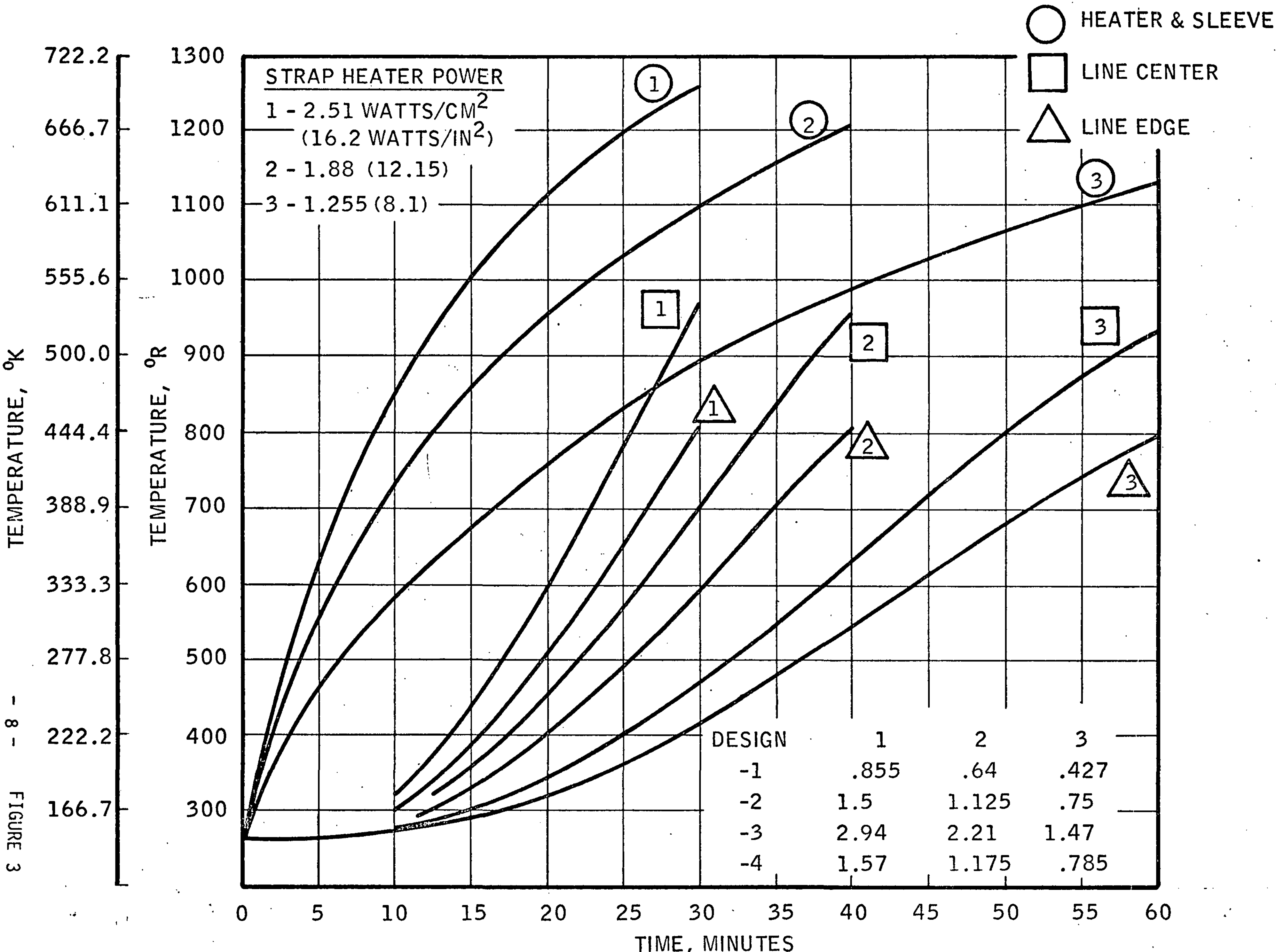


FIGURE 3

LINE HEATER THERMAL TRANSIENT IN AIR

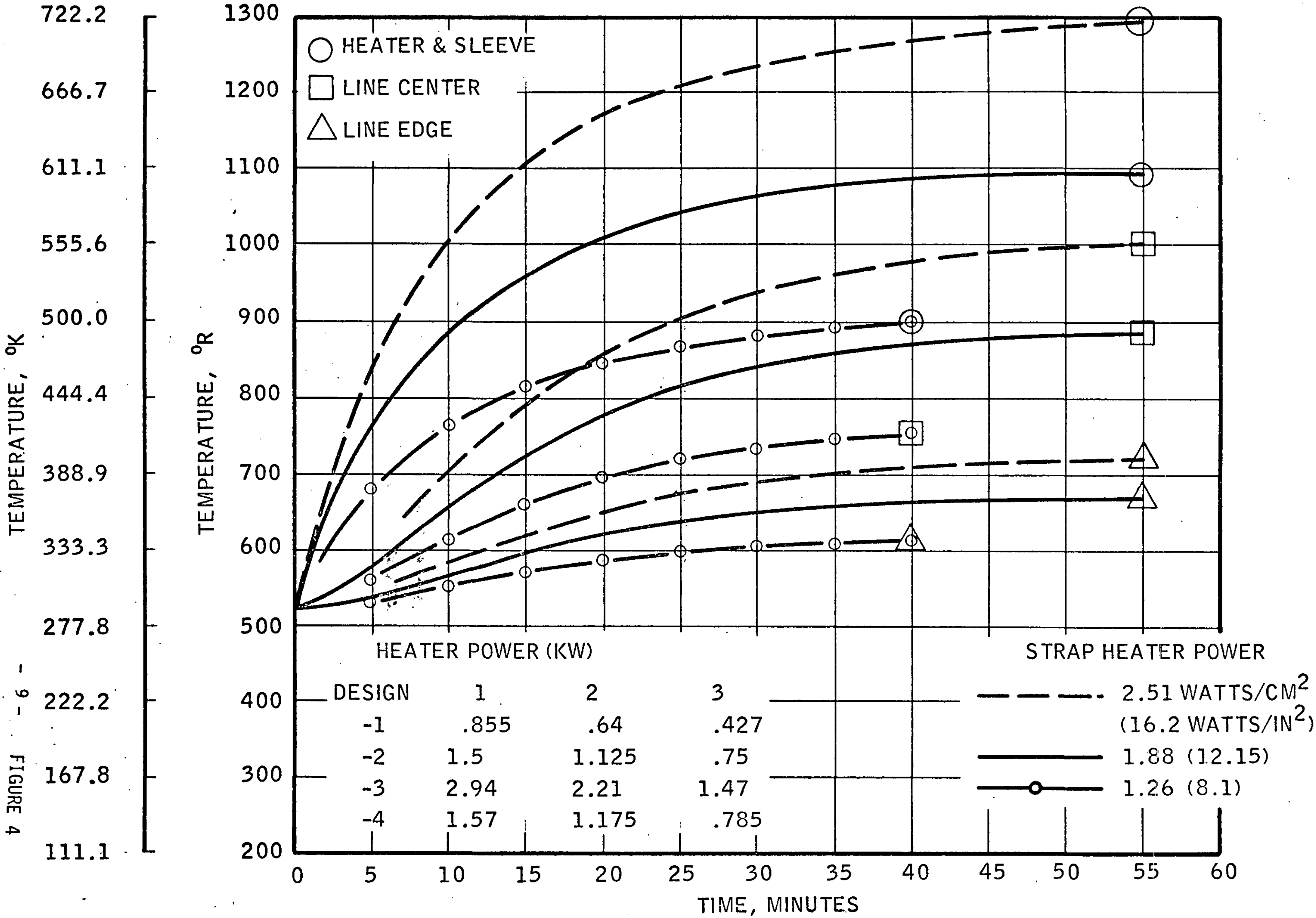


FIGURE 4

APPENDIX R

HEATER SIZING ANALYSIS FOR ANSC 1140006

THERMAL INTERFERENCE REMOTE COUPLING

I. INTRODUCTION

This report documents the heater sizing parametric analysis performed for the Thermal Interference Remote Coupling Design Concept. This work was done under MSFC Contract NAS 8-27568 to provide support analyses for the evaluation of this remote line coupler concept.

II. SUMMARY/CONCLUSIONS

Transient thermal analyses of the three sizes of thermal interference remote coupling designs were performed on a detailed analytical model. In general, the analyses indicated that fairly high heater power fluxes ($\sim$7.44 watt/cm^2) are required and that the heat-up of the sleeve (to $\sim$ 500°K) must be accomplished in a few minutes in order to achieve the desired temperature difference between the sleeve and line.

III. DISCUSSION

The thermal transient analyses of the remote couplings were performed using the D12207 finite element method program. The two dimensional thermal models were generated in cylindrical coordinates with the axis of rotation at the center line. The axial and radial dimensions of the three models are shown on Figure 1. This model assumes an 0.0635 cm (0.025 in.) wide-line contact between the pipe line and the sleeve. A 0.381 cm (0.15 in.) thick strap heater is in contact with the sleeve O.D. over the entire surface. The heat from the sleeve is transferred to the line through the small contact surface area and by thermal radiation across the sleeve-line gap. The emissivity value used for these surfaces was 0.5.

The nominal values of thermal conductivity and heat capacitance for AISI-347 as reported in Reference (a) were used in this analysis. The thermal conductance and capacitance of the strap heater were assumed to be one-half the AISI-347 values.

IV. RESULTS

For each of the three sizes of the thermal interference remote coupling designs, three heater power flux values were used. In each case the lowest power flux case could not generate sufficient temperature difference between the sleeve and line.

Shown on Figures 2 through 4 are the predicted heat-up thermal transient results for the three line size cases. Shown on Table 1 are the heat-up times and the maximum sleeve temperatures when the required temperature difference between the sleeve and line is achieved.

V. REFERENCES

(a) Material Properties Data Book, ANSC Report No. 2275

TABLE 1

Coupler Design	-1		-2		-3	
ΔT required	125.4°K (225.7°R)		176.17°K (317.1°R)		381.1°K (608.6°R)	
Heater Power Flux	T_{max} °K (°R)	θ Minutes	T_{max} °K (°R)	θ Minutes	T_{max} °K (°R)	θ Minutes
2.48 watt/cm^2 (16 watt/in^2)	-	-	-	∞	-	-
4.96 " (32 ")	-	∞	489 (880)	3	-	∞
7.44 " (48 ")	364 (655°)	1-2/3	400 (720)	1-1/3	589 (1060)	1-5/6
9.94 " (64 ")	333 (600)	1.0	-	-	540 (975)	1-1/6

DIFFERENTIAL THERMAL EXPANSION LINE COUPLER MODELS

RADIUS

-1 cm	-1 IN.	-2 cm	-2 IN.	-3 cm	-3 IN.
5.220	(2.055)	7.591	(2.91)	11.049	(4.35)
4.839	(1.095)	7.010	(2.76)	10.668	(4.20)
4.331	(1.705)	6.786	(2.67)	10.464	(4.12)
4.204	(1.655	6.629	(2.61)	10.414	(4.10)
4.191	(1.650)	6.502	(2.56)	10.287	(4.05)
4.064	(1.600)	6.350	(2.50)	10.211	(4.02)
3.556	(1.400)	6.096	(2.40)	10.033	(3.95)

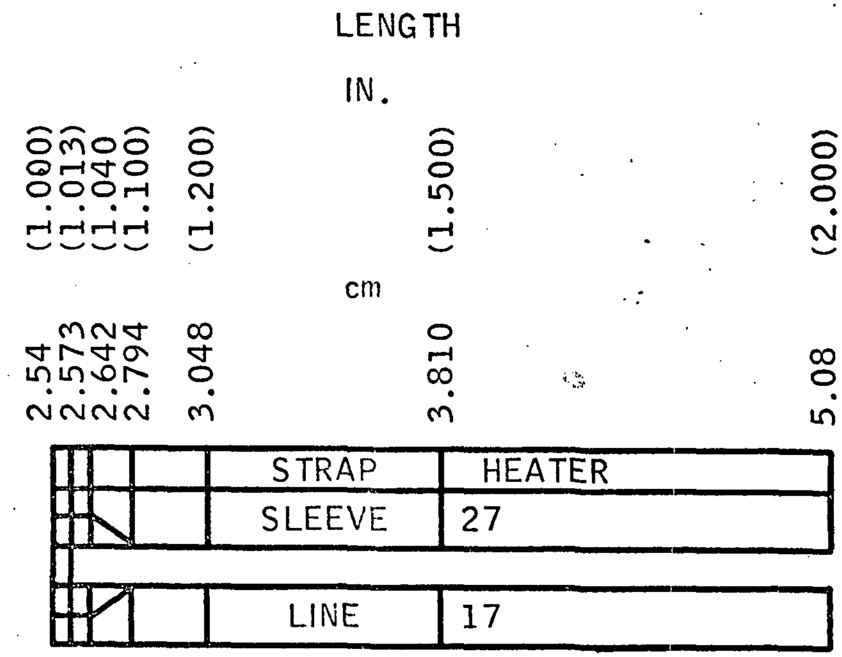
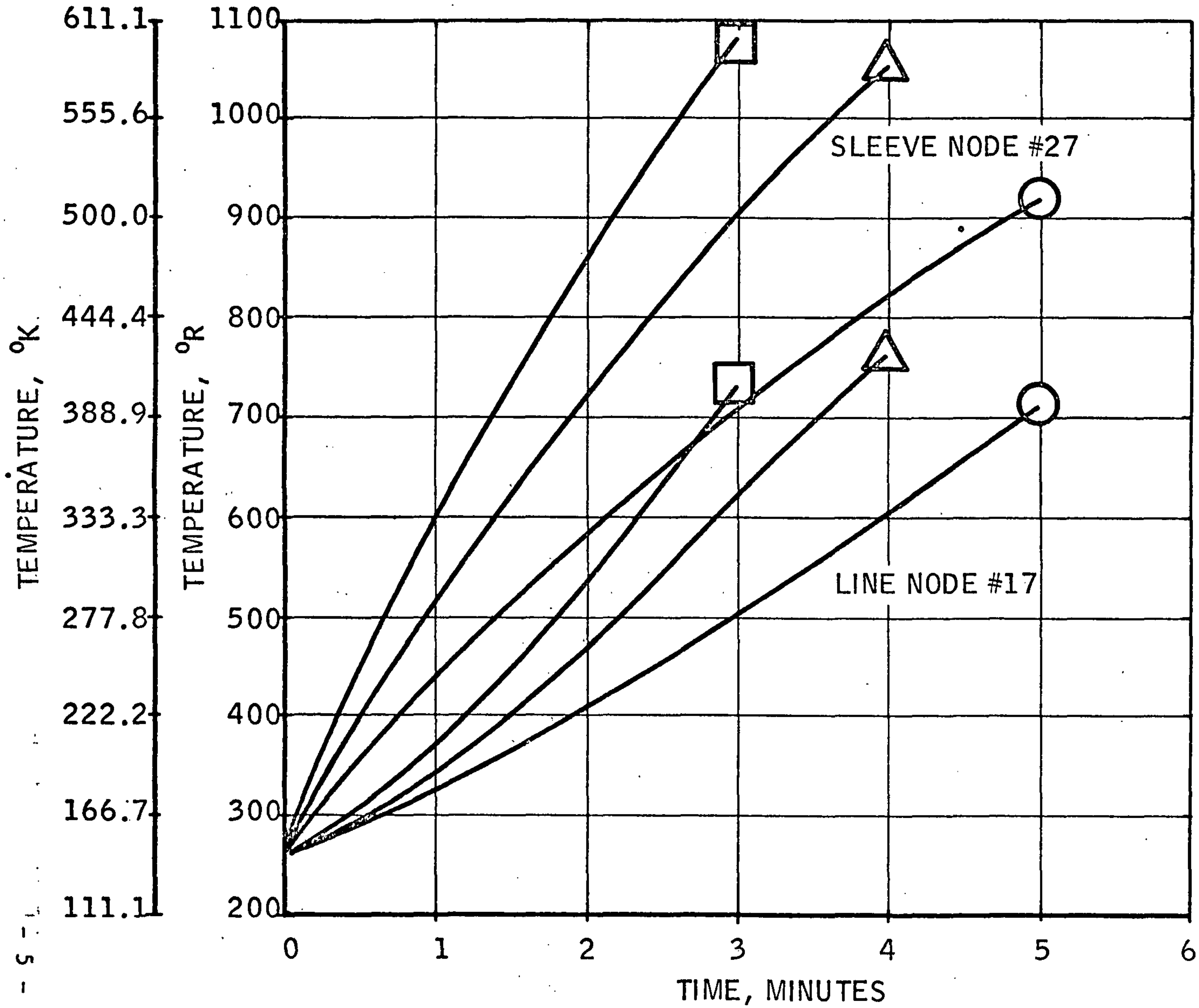


FIGURE 1

DIFFERENTIAL EXPANSION LINE COUPLER
THERMAL TRANSIENT ANALYSIS

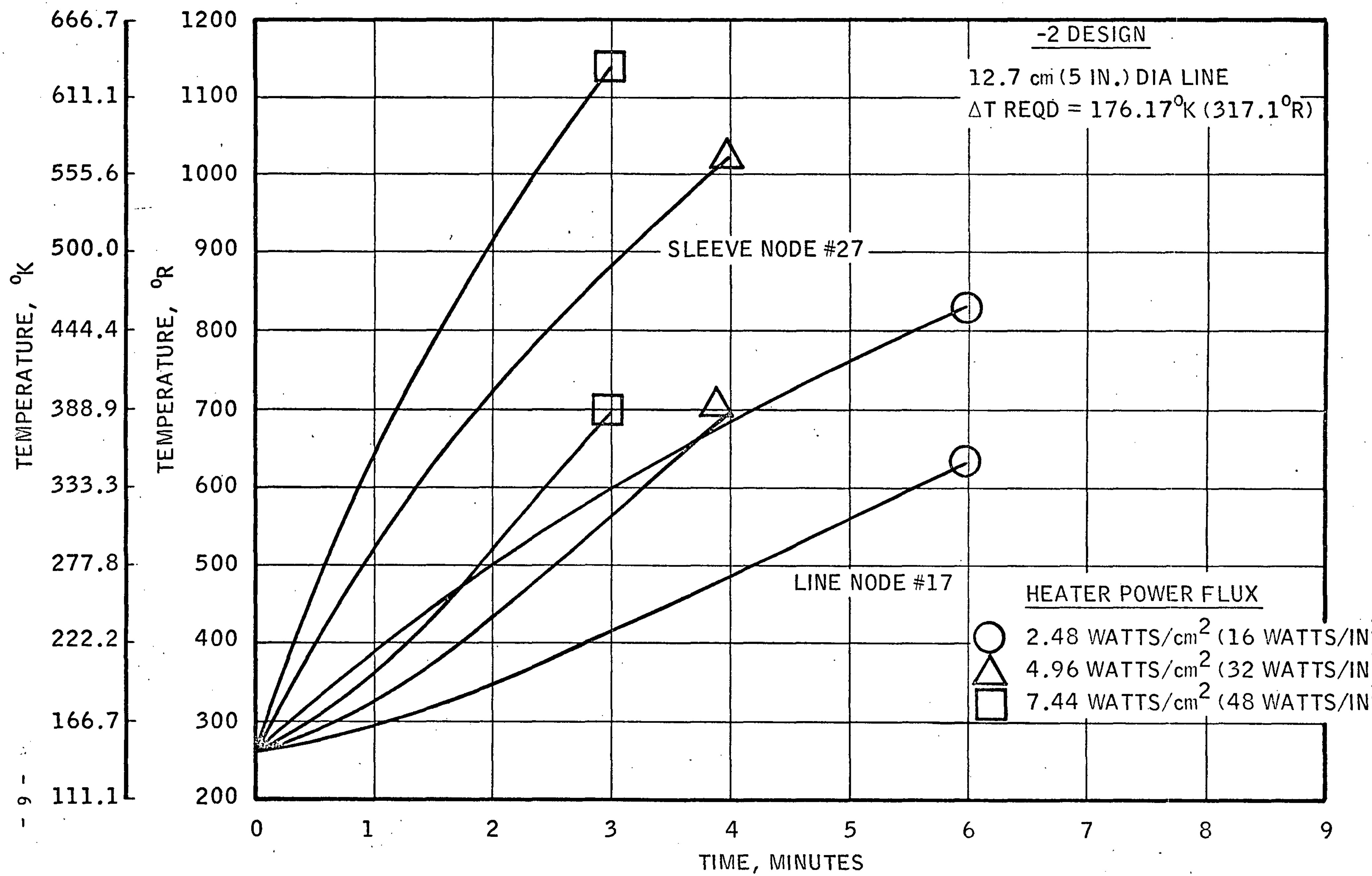


-1 DESIGN

7.62 cm (3 IN.) DIA LINE
ΔT REQD = 125.4^oK (225.7^oR)

HEATER POWER FLUX
○ 4.96 WATTS/cm^2 (32 WATTS/IN^2)
△ 7.44 WATTS/cm^2 (48 WATTS/IN^2)
□ 9.94 WATTS/cm^2 (64 WATTS/IN^2)

DIFFERENTIAL EXPANSION LINE COUPLER THERMAL TRANSIENT ANALYSIS



DIFFERENTIAL EXPAN[illegible]ON LINE COUPLER
THERMAL TRANSIENT ANALYSIS