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SEMI-ANNUAL REPORT NO. 6

DEVELOPMENT OF COMPRESSOR END SEALS STATOR INTERSTAGE SEALS, AND STATOR PIVOT SEALS IN ADVANCED AIR BREATHING PROPULSION SYSTEMS

prepared for
National Aeronautics and Space Administration

CONTRACT NAS3-7605

Pratt & Whitney Aircraft DIVISION OF UNITED AIRCRAFT CORPORATION

EAST HARTFORD • CONNECTICUT

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19 July 1968

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PREFACE

This report describes the progress of the work conducted between 1 January 1968 and 30 June 1968 by the Pratt & Whitney Aircraft Division of United Aircraft Corporation, East Hartford, Connecticut on Contract NAS3-7605, Development of Compressor End Seals, Stator Interstage Seals, and Stator Pivot Seals in Advanced Air-Breathing Propulsion Systems, for the Lewis Research Center of the National Aeronautics and Space Administration.

The authors wish to acknowledge the assistance of Messrs. Gordon D. Pfiefer and Phillip W. Tayntor in the preparation of material for this report.

Roger M. Hawkins is Project Manager for Pratt & Whitney Aircraft for this program.

The following National Aeronautics and Space Administration personnel have been assigned to the project:

Contracting Officer	J.H. DeFord
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SUMMARY

This report describes the work accomplished during the sixth six-month period of an analytical, design, and experimental program directed at developing compressor end seals, stator interstage seals, and stator pivot seals for advanced air-breathing propulsion systems.

Work under Tasks I and III was completed during the first two and one-half years of this contract, and is discussed in the first five semiannual reports. For Task I, this work consisted of feasibility analyses of the one-side and two-side floated shoe seals, the OC diaphragm seal, the semirigid seal, and the flexure-mounted shoe seal. For Task III, the single-bellows vane pivot seal and the spherical-seat vane pivot seal were analyzed. No work on Tasks I and III was carried out during this reporting period, but very brief summaries of the two tasks are included in this report for general background information.

Most of the work under Task II has been concerned with the procurement of the one-side floated-shoe end and interstage seals, the OC diaphragm end seal, and the semirigid interstage seal. However, some thermal analysis of the one-side floated-shoe interstage seal has been performed in order to improve the accuracy of earlier analyses.

The work under Task IV has been devoted to procurement and inspection of the Task IV seals and to checking out the test rig. Two preliminary checkout runs have been made on the rig, and the system for measuring leakage flow past the test seals has been modified.

Milestone Charts are presented at the end of this report.

SEMIANNUAL REPORT NO. 6

DEVELOPMENT OF COMPRESSOR END SEALS STATOR INTERSTAGE SEALS, AND STATOR PIVOT SEALS IN ADVANCED AIR-BREATHING ENGINES

by

R.M. Hawkins and A.H. McKibbin

ABSTRACT

The compressor end and interstage seals are described and procurement of the seals is discussed. There is also a brief discussion of the effects of heat shielding and heating on the one-side floated-shoe interstage seal. In addition, procurement of the stator vane pivot seals and modifications to their associated test apparatus is discussed.

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INTRODUCTION

Modern high-performance multistage axial-flow compressors built with state-of-the art features incorporate several air leakage paths which are detrimental to compressor performance. Elimination or significant reduction of these leaks would result in a compressor of higher efficiency and possibly smaller size. Some typical areas of leakage paths with estimates of percent air loss and potential effect on compressor performance are:

	Air Loss	Effect on Compressor Efficiency
End Seal	0.6%	1.0%
Interstage Stator Seals (ten stages)	0.9%	1.0%
Vane Pivot Seals (variable stator)	0.2% per stage	0.2% per stage

Increases in compressor efficiency are traditionally sought by means of compressor geometry redesign. A few extra points in efficiency often mean the difference between a successful of an unsuccessful design. These increases as a result of geometry change are always very expensive and not always successful. On the other hand, the efficiency losses resulting from air leaks are strikingly large, and real gains are within reach at a relatively low cost. The gains in efficiency, however, must be balanced against any detrimental effect that improved sealing may have on the engine, such as lower reliability or increased weight.

This program will provide for a research, analytical, and test program having as its goal the development of compressor end seals, stator interstage seals, and vane pivot seals which exhibit lower air leakage rates than those currently in use. This will be accomplished using components of such size, materials, and designs as to be considered applicable to compressors for engines capable of supersonic aircraft propulsion.

I. TASK I - CONCEPT FEASIBILITY ANALYSIS PROGRAMS FOR COMPRESSOR END SEALS AND FOR COMPRESSOR STATOR INTERSTAGE SEALS

A feasibility analysis was conducted on compressor end seals and stator interstage seal concepts for advanced air-breathing propulsion systems. The first phase of this program consisted of a preliminary analysis and screening of various seal concepts prior to the selection of concepts for the detailed feasibility analysis. The analytical effort included all calculations, analyses, and drawings necessary to establish the feasibility of the selected concepts, and a comparison of the concepts to current practice. This analytical program was subcontracted to Mechanical Technology Incorporated of Latham, New York, and was monitored by Pratt & Whitney Aircraft as required under the terms of the NASA contract.

Work under the Task I program was completed on the one-side floated-shoe compressor end seal and stator interstage seal concepts. Pratt & Whitney Aircraft submitted the final feasibility designs of these seals to NASA on 19 May 1966 requesting approval to start final design under Task II. Approval was granted in a letter from NASA dated 31 May 1966.

Contract Amendment Number 2, effective 15 December 1966, was received from NASA for the feasibility analysis of an OC diaphragm thin-strip seal design. As a backup effort, the contractor also performed a feasibility analysis on a semirigid one-piece seal concept. The results of these analyses indicated that both seals will yield satisfactory performance. Pratt & Whitney Aircraft subsequently selected the OC diaphragm thin-strip seal for the compressor end seal application and the semirigid one-piece seal for the interstage application.

Task I work under Contract Amendment Number 2 was completed in 1967. The final feasibility design concepts of the OC diaphragm end seal and the semirigid interstage seal were submitted to NASA on 3 October 1967, and the contractor requested approval to start final design under Task II. Approval was granted in a letter from NASA dated 6 November 1967.

II. TASK II - COMPRESSOR END SEAL AND STATOR INTERSTAGE SEAL EXPERIMENTAL EVALUATION

This phase of the program provides for final design and fabrication of compressor end seals and stator interstage seals, design and fabrication of a test rig, and experimental evaluation of the compressor seals.

The final design of the four compressor seal concepts selected for experimental evaluation includes all calculations, material determinations, analyses, and drawings necessary for seal optimization, procurement, and experimental evaluation. A test rig has been designed and fabricated to evaluate the selected compressor end seals and stator interstage seals under simulated compressor operating conditions. The test apparatus will simulate the last stages of a full-scale compressor, including supporting members and bearing system in order to faithfully duplicate structural flexibility and thermal gradients.

The compressor end seals and stator interstage seals will be calibrated in incremental steps at room-temperature static conditions, room-temperature dynamic conditions, and subsequently over the full speed, pressure, and temperature operating ranges. The seals will then be subjected to endurance testing and finally undergo a take-off and cruise cyclic test.

Final design layouts and detailed drawings have been completed for the one-side floated-shoe end and interstage seals and for the full-scale test rig in which the seals will undergo experimental evaluation. Hardware procurement is progressing for the seals, and the rig has been fabricated.

A. ONE-SIDE FLOATED-SHOE END SEAL

The one-side floated-shoe end seal is a face seal consisting of a ring of segments acting against a rotating surface attached to the compressor rotor. Figure 1 is a schematic drawing of the seal. The rotating surface is flat, and the leakage flows radially inward through the seal. The primary seal is between the stationary ring of shoes and the rotating face. The secondary seals are between the shoes and the carrier ring, and between the carrier ring and the mounting ring.

Work on the one-side floated-shoe end seal was largely confined to procurement and assembly during the reporting period. By technical direction, one change was made in the design of the seal. The change deleted the pressure probes in the seal face, because it was felt that they would unbalance the shoes and cause rubbing.

Fabrication of the seal continued without notable incident until April 1968, when the aluminum oxide hardcoat on the Rayleigh pads was chipped during final machining. By May, however, the pads had been recoated and remachined, and by June the seal was being assembled. The seal was delivered to the contractor at the end of the report period. It is shown as a complete assembly in Figures 2 and 3. Testing will be initiated on this seal as soon as the rig and stand systems have been checked out.

B. ONE-SIDE FLOATED-SHOE INTERSTAGE SEAL

The one-side floated-shoe interstage seal is very similar to the one-side floated-shoe end seal shown in Figure 1. The only differences between the seals are those caused by the slightly different mechanical requirements for sealing in the end and interstage positions.

Both analytical and procurement activities were carried out during the reporting period. The analytical activities were concerned with a detailed thermal analysis of the seal.

The thermal analysis of the seal included a more accurate determination of the convection and conduction paths within the seal parts than had been available for the preliminary analysis. It appears from the thermal map shown in Figure 4 that the temperature gradient across the seal carrier will be on the order of 100 degrees Fahrenheit per inch axially, which is considered to be excessive. To reduce this gradient and to better simulate an engine environment, the contractor is planning to use a heat shield with Calrod heating. The shield will provide a more nearly uniform temperature by keeping the 800-degree Fahrenheit cooling air for the disk from coming into contact with the interstage seal parts. With this scheme of providing a shield and Calrod heating, it appears that the thermal gradients can be significantly reduced, or even reversed, as shown in Figures 5 and 6.

The heat shield for this application is being procured as a one-piece unit which will be cut into three pieces for assembly into the rig. After mounting into the test rig case, the cuts will be welded to make a joined ring.

Four electric heating rods of 11 KW capacity will be welded to the shield to provide a capacity of more than 50 percent over the amount deemed necessary by the analytical studies. Additional rods will be placed around the mounting flange to provide 10 KW of heat. This is 100 percent greater than anticipated analytically. All heating rods will have Inconel sheathing capable of sustaining temperatures as high as 1800 degrees Fahrenheit.

C. OC DIAPHRAGM END SEAL

The OC diaphragm end seal (shown in Figure 7) employs a thin, flexible one-piece strip as the primary seal element, providing a high degree of conformity to runner distortion. The thin strip is supported by three C-shaped semitoroidal diaphragms mounted on a floating secondary seal carrier. The secondary carrier provides for axial travel relative to the main engine structure, and a piston-ring seal is used between the carrier and the engine structure. One of the C diaphragms forms a seal between the high-pressure and the low-pressure areas. The other two C diaphragms face each other and form a chamber to which the high-pressure air is admitted. This design, therefore, permits direct balancing of the moments on the thin strip. The OC design does not use the T-section strip required to properly balance moments in the thin-strip designs previously considered. It also permits a decrease in the thickness of the strip relative to other thin-strip designs. Thus, it offers the opportunity of generating a section with increased flexibility and better tracking capability. Furthermore, the moment balance is achieved with methods which are more nearly independent of angular displacements of the strip, making low residual moment imbalance easier to achieve.

Work on the OC diaphragm end seal has been concentrated on acquiring raw materials and bids for the finished seal, and on refining the welding techniques required for manufacturing the seal.

The welding experiments have been carried out with small Inconel test specimens which are about the same thickness as the seal parts. Techniques for electron-beam welding these specimens have been improved, and the investigation has shown that these welds are feasible.

All raw materials for the seal have been received. Bids for fabrication of the seal have been received and evaluated. NASA approval is required before a purchase order can be placed.

D. SEMIRIGID INTERSTAGE SEAL

The semirigid interstage seal (shown in Figure 8) operates on the same basic principle as the OC diaphragm end seal: leakage is controlled by controlling clearance. Primary sealing is accomplished at the seal face, which consists of a single land. This land has a spiral-groove inherently compensated orifice profile, and acts as a bearing and seal combination. The angular stiffness of a single-land bearing is very low. As a result, the seal ring must be rigid enough to absorb residual bending moments without appreciable deformation of the seal face. To accomplish this, substantial seal-ring length is required. Moreover, the seal ring must also serve as a housing for the piston ring required for secondary sealing, one of the secondary seals being formed by contact between the side of the piston ring and the seal ring. It should be noted that the combination of seal length and piston-ring contact is conducive to the generation of high thermal gradients. Thermal gradients in turn cause the seal surface to deform, and through this deformation they may seriously affect seal performance. In order to minimize the extent of thermal gradients and seal deformations, the following steps have been taken:

- The seal material was selected to provide high thermal conductivity in combination with a low coefficient of thermal expansion.
- The piston ring was insulated through the inclusion of a thermal barrier in the form of a radial annular slot.
- The seal's cross section was designed to minimize thermal distortions through the addition of a relatively constant temperature ring on the seal's outer edge close to the seal face.
- The seal's tracking performance characteristics indicate tolerance to some degree of distortion. Since slight distortion may be beneficial, the seal was designed so that sufficient distortion occurs, the net result of which is an increase in film clearance at high pressure ratios, reduction in heat generation at the seal face, and an increase in leakage flow. The increased air flow also carries more heat away from the seal face, leaving less heat to be dissipated by the seal, and therefore, lower thermal gradients.

The seal ring is preloaded with 24 helical coil springs to ensure contact at start and to permit the development of separating air films at relatively low speeds. The seal face and runner materials were also selected with regard to compatibility at high temperature and resistance to wear. They are identical to the ones used in the OC diaphragm seal.

All raw materials for the seal have been received. Bids for the fabrication of the seal have been received and evaluated. NASA approval is required before a purchase order can be placed.

E. TEST RIG

The test rig has been assembled, balanced, and installed in the test stand. The rig shown mounted on the dolly in Figure 9. In assembling the test rig, the rotor seal runner surfaces were kept to within 0.0035 inch total runout. This was considered to be acceptable, since the seal analysis had allowed up to 0.005 inch total runout in the rig rotor. When the rig was assembled, it was discovered that the connecting links were binding on the case, restricting its full axial travel. This condition will be remedied in the next build, since the first build is only intended to familiarize the stand personnel with the operation of the rig and stand systems and to check out the stand systems. For the same reason, the first build of the rig contains baffle plates instead of the test seals, in order to ensure that no seals will be damaged by stand failures.

In order to control vibration in the test rig, the rotor was assembled in three stages. In the first stage, the individual parts were statically balanced, and the location and magnitude of the residual imbalance was marked on the parts. In the second stage, the parts were assembled victorially in order to ensure that all or most of the residual imbalance would not be stacked up in the same direction. This step was intended to reduce the size of the final dynamic balance correction and to reduce the dynamic loads within the rotor, since such loads would tend to force the rotor out of shape at operating speed. In the third stage, the assembled rotor was dynamically balanced, bringing the residual imbalance to an amount considered acceptable by design and analytical studies.

Before the rig was mounted on the transporter stand, the stand was machined on the top and bottom to ensure that the mating surfaces would be flat and parallel. The stand was also reinforced to keep the parallel surfaces from deflecting during the mounting procedure.

The parts of the test rig before assembly are shown in Figures 10 through 21. Figure 10 shows the front disk, which is located just behind the interstage seal, and is the main support for the seal runner. Figure 11 shows the rig seal. The function of this seal is to force the cooling air through the rotor after it enters the rig. Figure 12 is a frontal view of the rear disk, and shows the spacer element of the disk with its instrumentation grooves. Figure 13 is a rear view of the interdisk spacer. The surface of the interstage seal runner can also be seen in this photograph. The front hub of the rotor is shown in Figure 14 with the rig air seal attached. Figure 15 shows the rear hub with balance weights attached. Figure 16 presents a view of the end seal disk with the seal face up, while Figure 17 shows the same part with the seal down. The front rig case is shown in Figure 18. The large cut-outs in the case allow passage of the mounting legs to attach the bearing supports. The rear case and sliding support are shown in Figures 19 and 20. The front bearing supports for the main rig thrust bearing are shown in Figure 22.

F. TEST STAND

Construction of the test stand has been completed. Figure 22 and 23 show the stand without the rig and transporter in place. All valving and instrumentation has been checked out.

The rig was mounted on the transporter stand and a dial indicator was mounted in such a way as to indicate distortion of the transporter stand. With the indicator zeroed out, the transport stand was bolted to the test stand rails. When all bolts were tightened, the indicator showed some distortion. The distortion was corrected by appropriate shimming between the transport stand and the mounting rails. This procedure was used to ensure that no additional bending moments would be imposed on the test rig and to ensure that the rotor bearing supports at each end of the rig would be properly aligned.

Stand plumbing lines and manifolds have been connected to the rig. An initial stand checkout revealed an instability in the alcohol burner unit used to provide air at 1200 degrees Fahrenheit to the rig. This problem was being investigated at the end of the report period. The checkout also revealed a high degree of sensitivity in the air-flow control and a need for greater flexibility in controlling the back pressure in the hot-air bypass line. Sensitivity was decreased by drilling an extra bleed hole in the bleed control area. Control over back pressure was improved by converting a shutoff valve to a control valve. With the exception of the alcohol burner, the lubrication, fuel, and air systems are now satisfactory, and can be used without further rework.

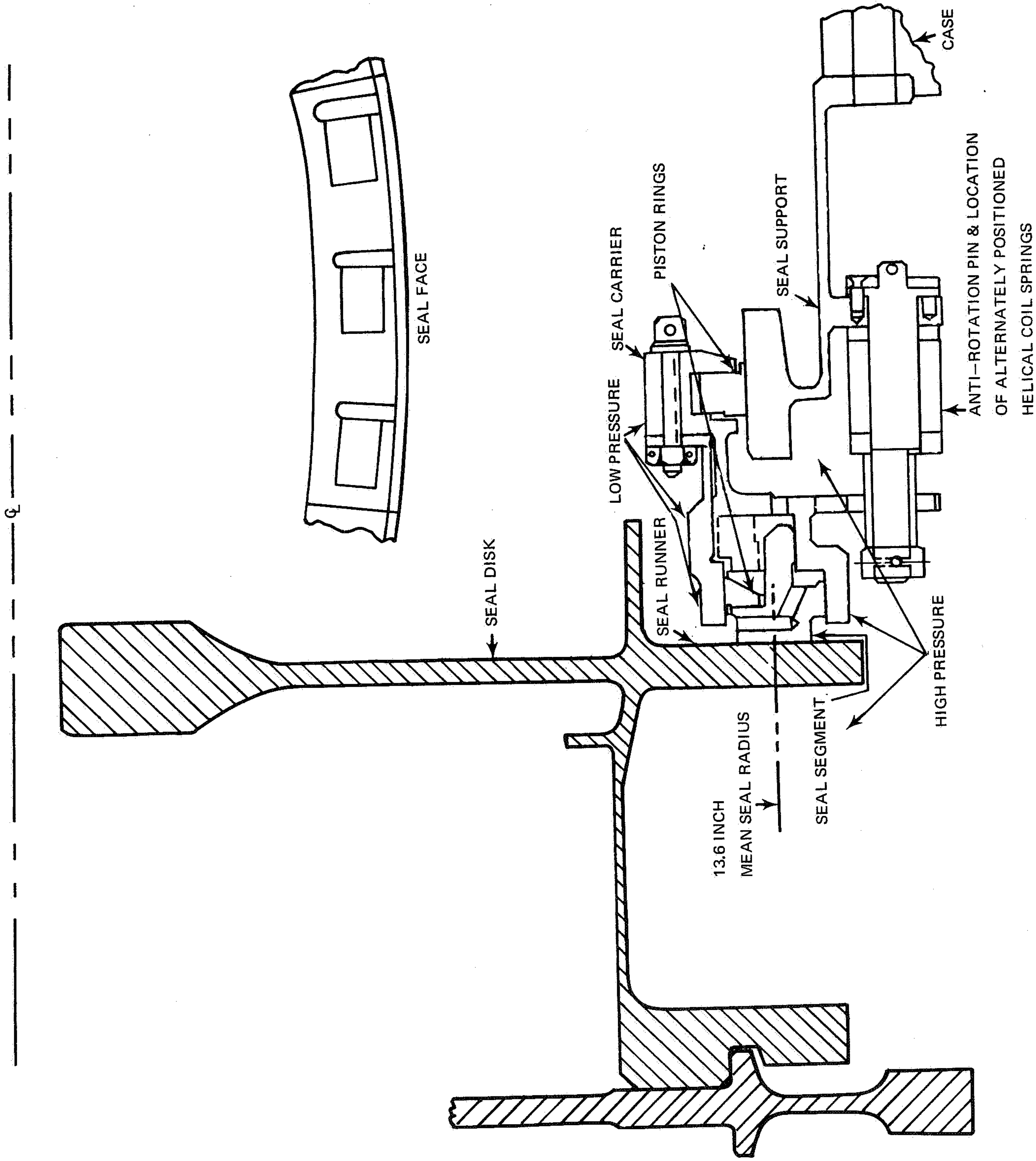


Figure 1 Schematic of the One-Side Floated Shoe End Seal

Figure 2 One-Side Floated Shoe End Seal. (CN-14385)

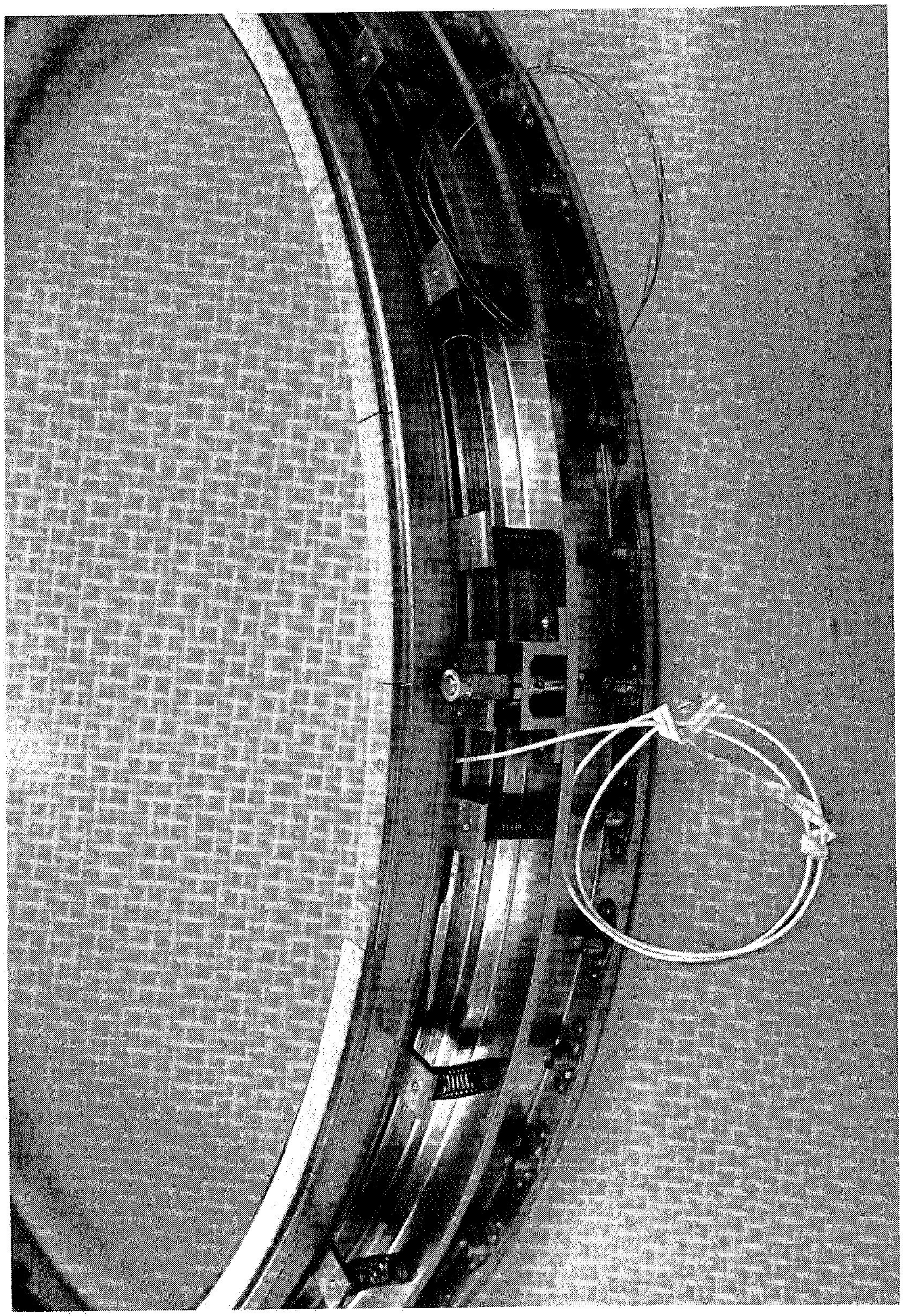


Figure 3 Side View of the One-Side Floated Shoe End Seal (CN-14386)

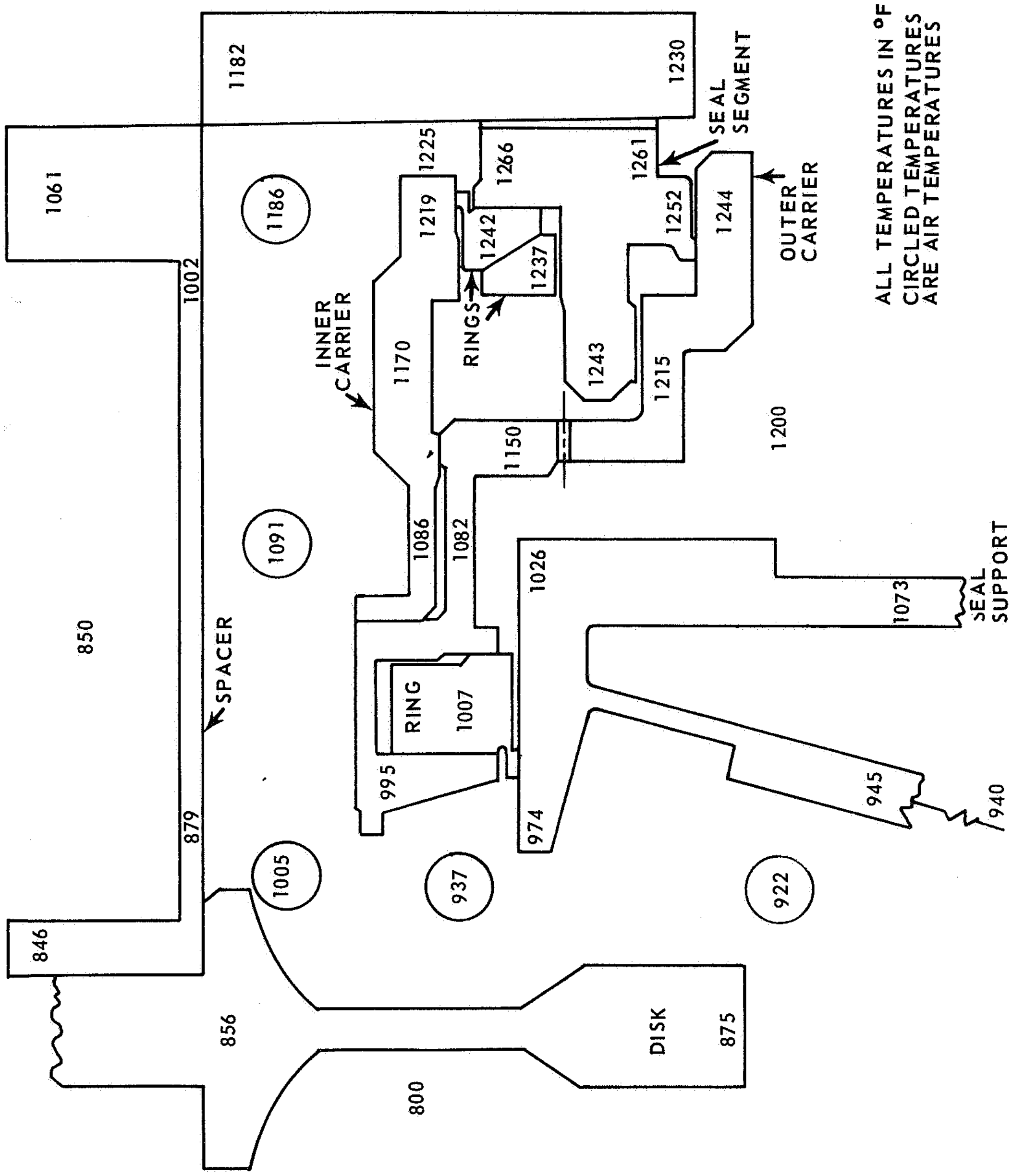


Figure 4 Thermal Map of the One-Side Floated-Shoe Interstage Seal at Cruise Conditions without Thermal Shield and Calrod Heaters

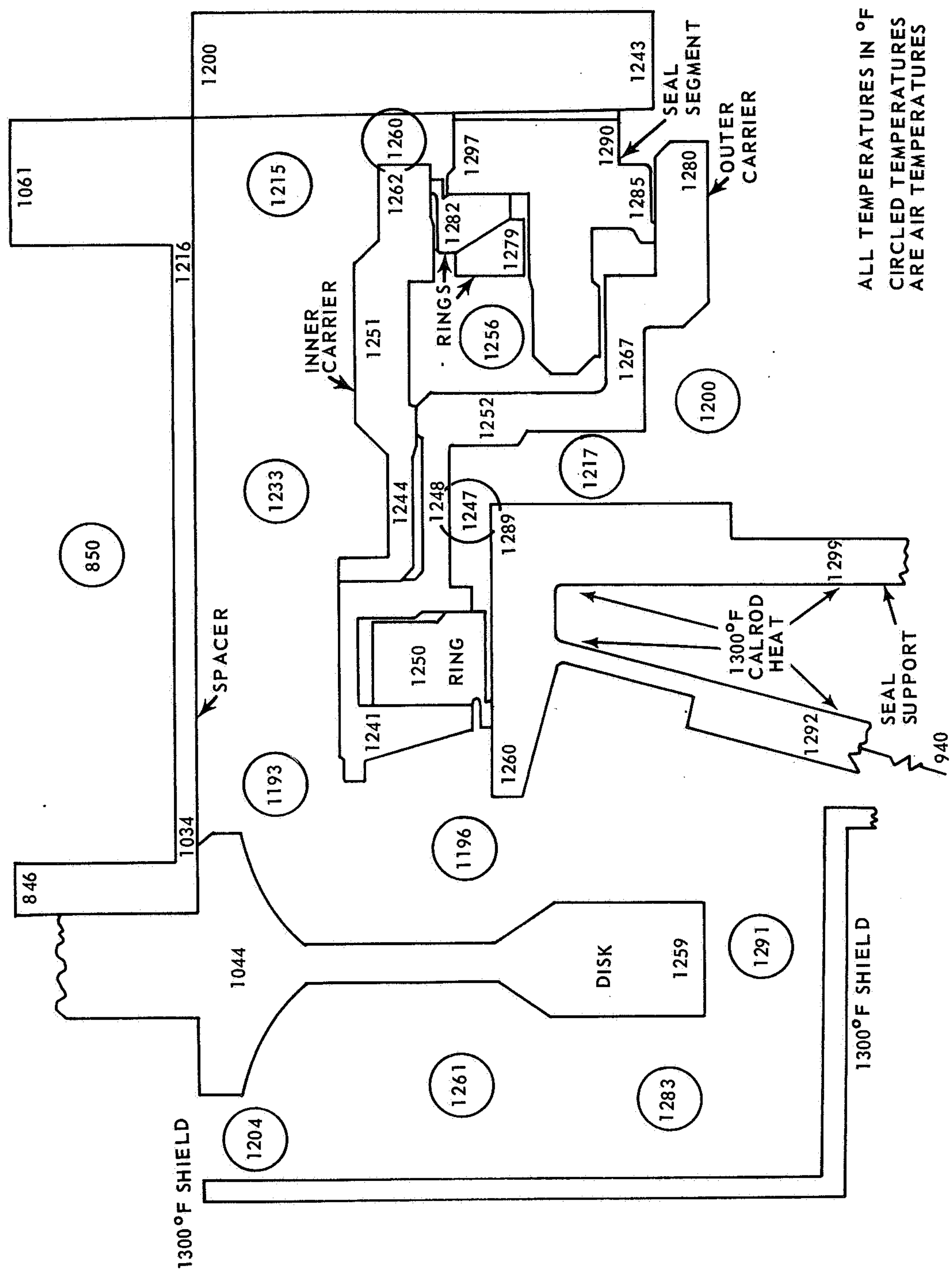


Figure 5 Thermal Map of the One-Side Floated-Shoe Interstage Seal at Cruise Conditions with Thermal Shield and 1300°F Calrod Heating

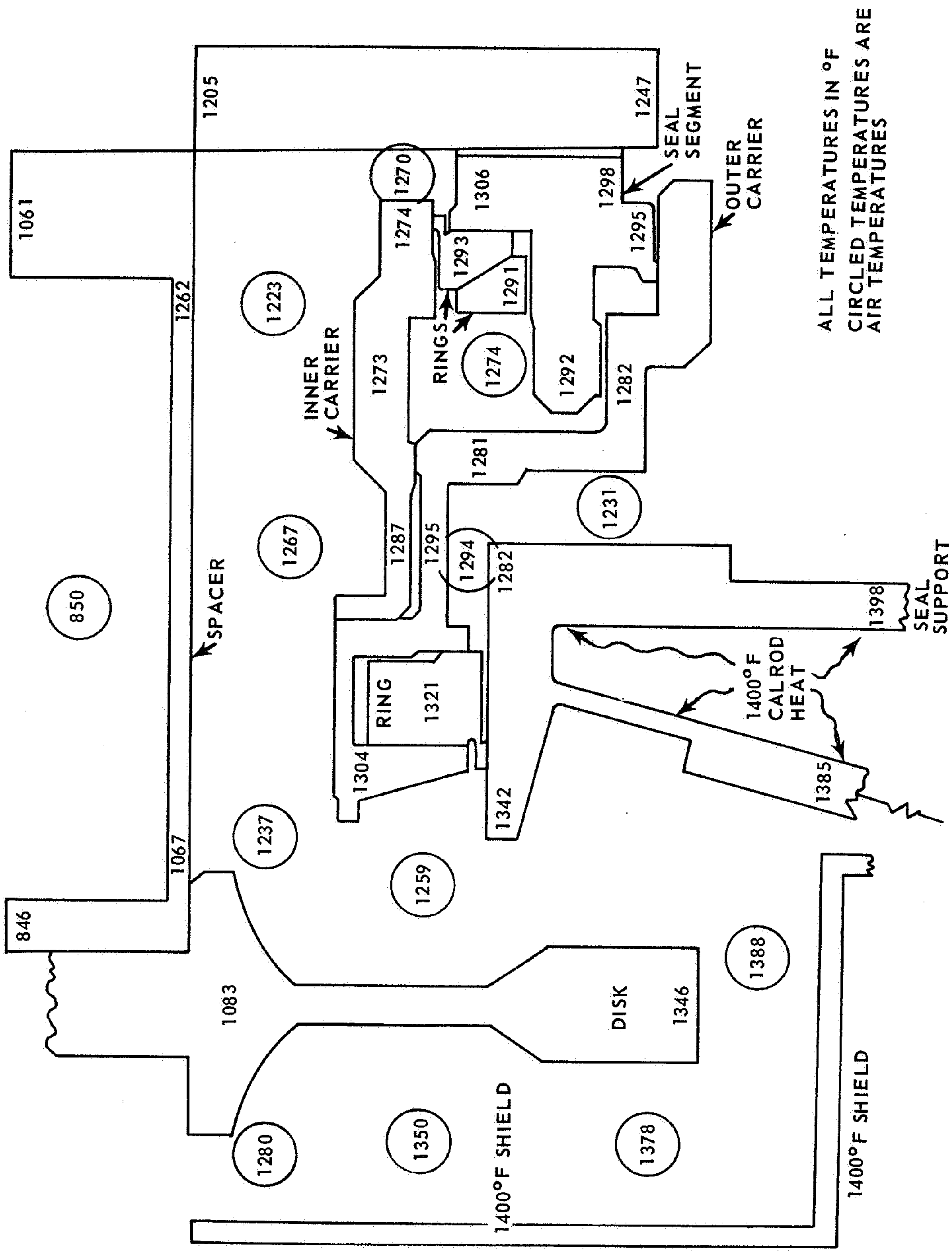


Figure 6 Thermal Map of the One-Side Floated-Shoe Seal at Cruise Conditions with Thermal Shield and 1400°F Calrod Heating

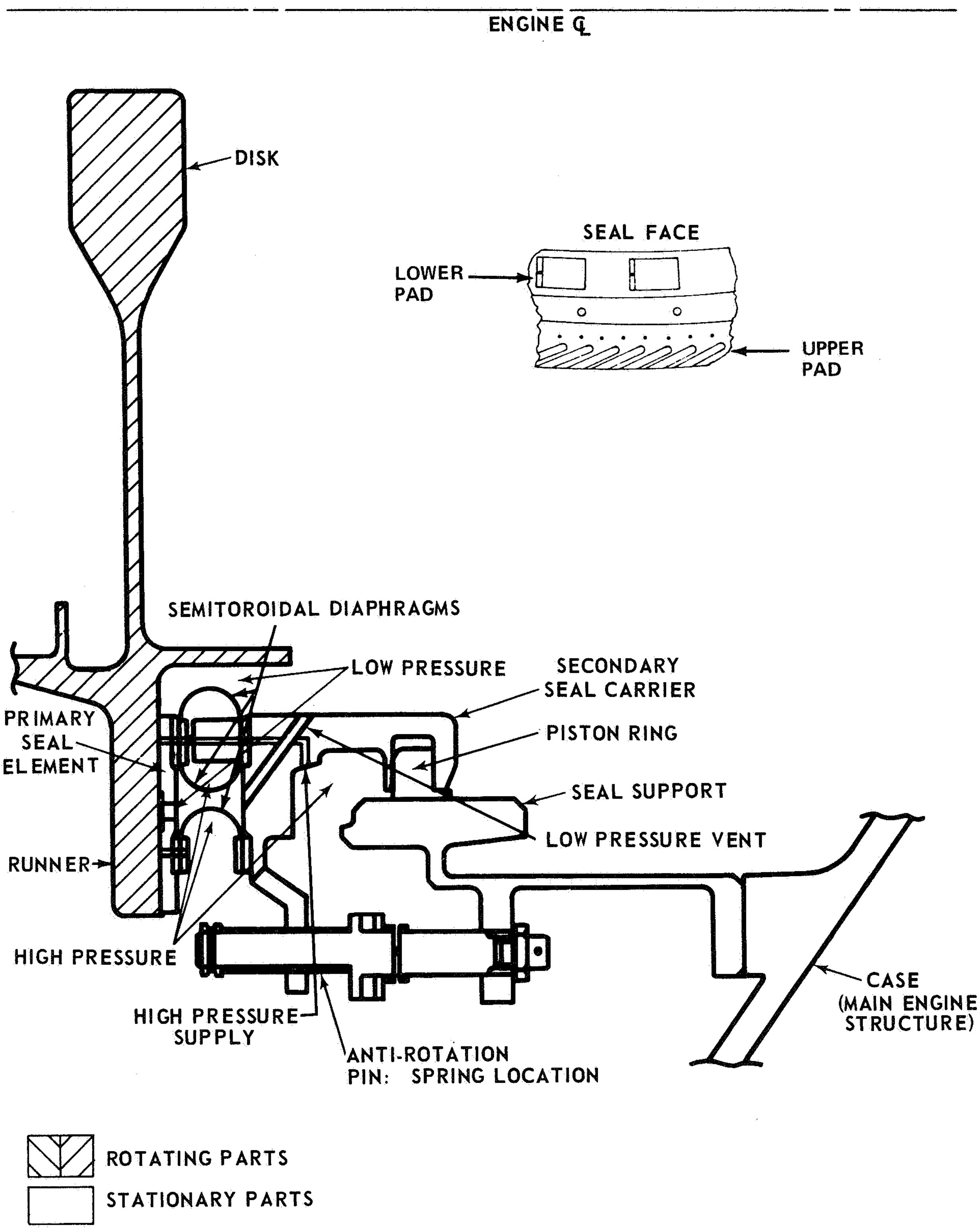


Figure 7 OC Diaphragm End Seal

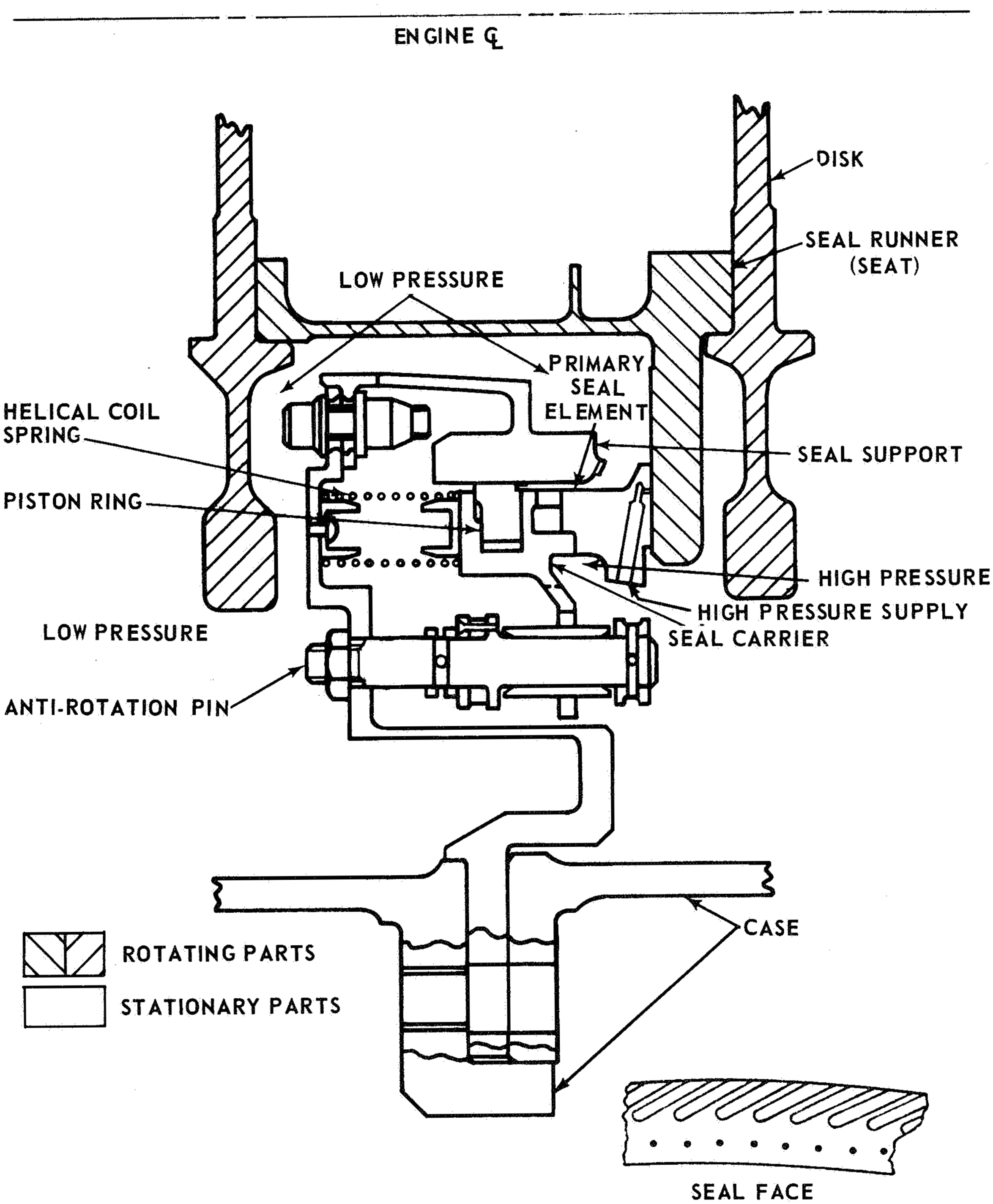


Figure 8 Semirigid Interstage Seal

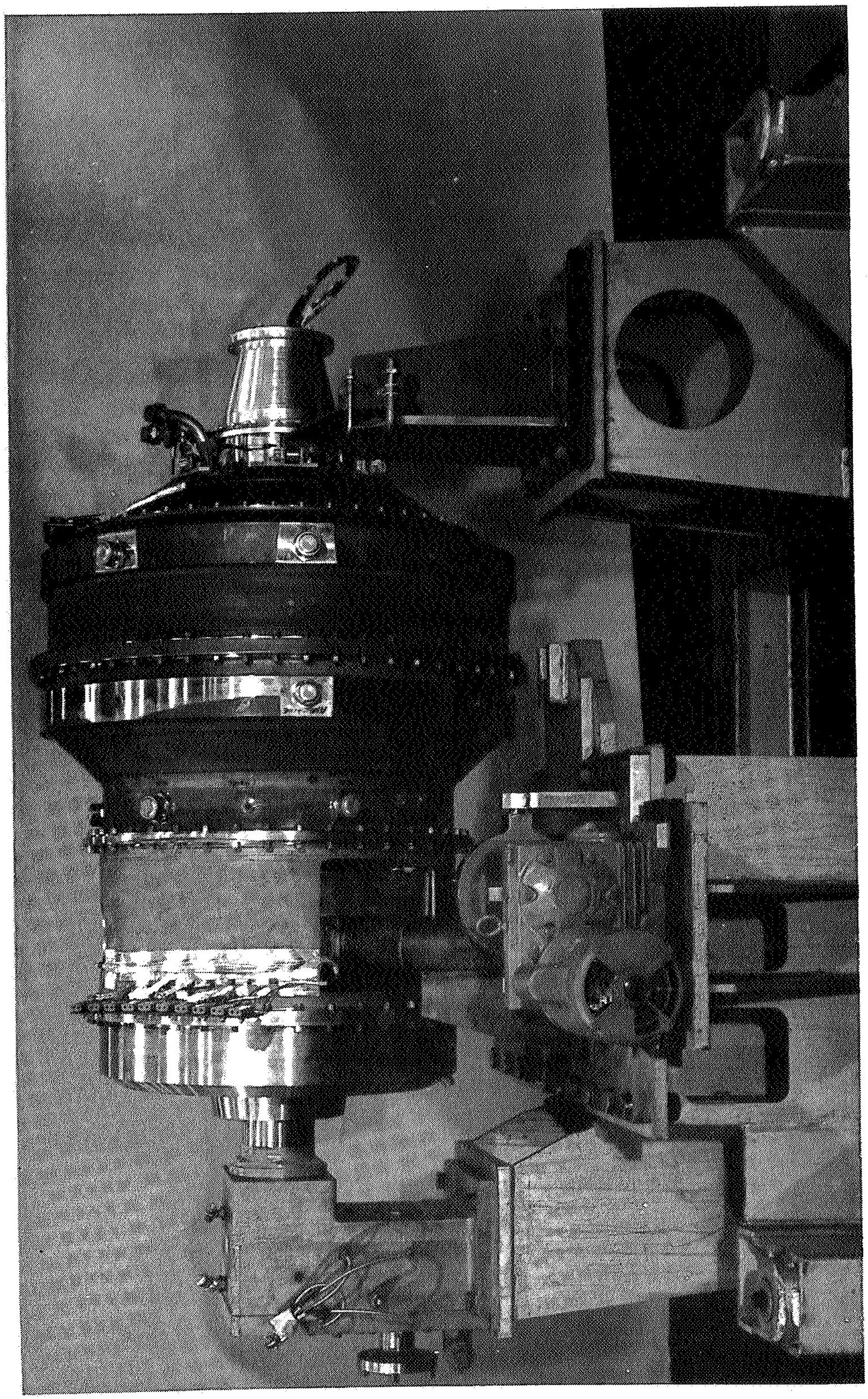


Figure 9 Compressor Seal Test Rig Mounted on Rig Transporter (CN-13744)

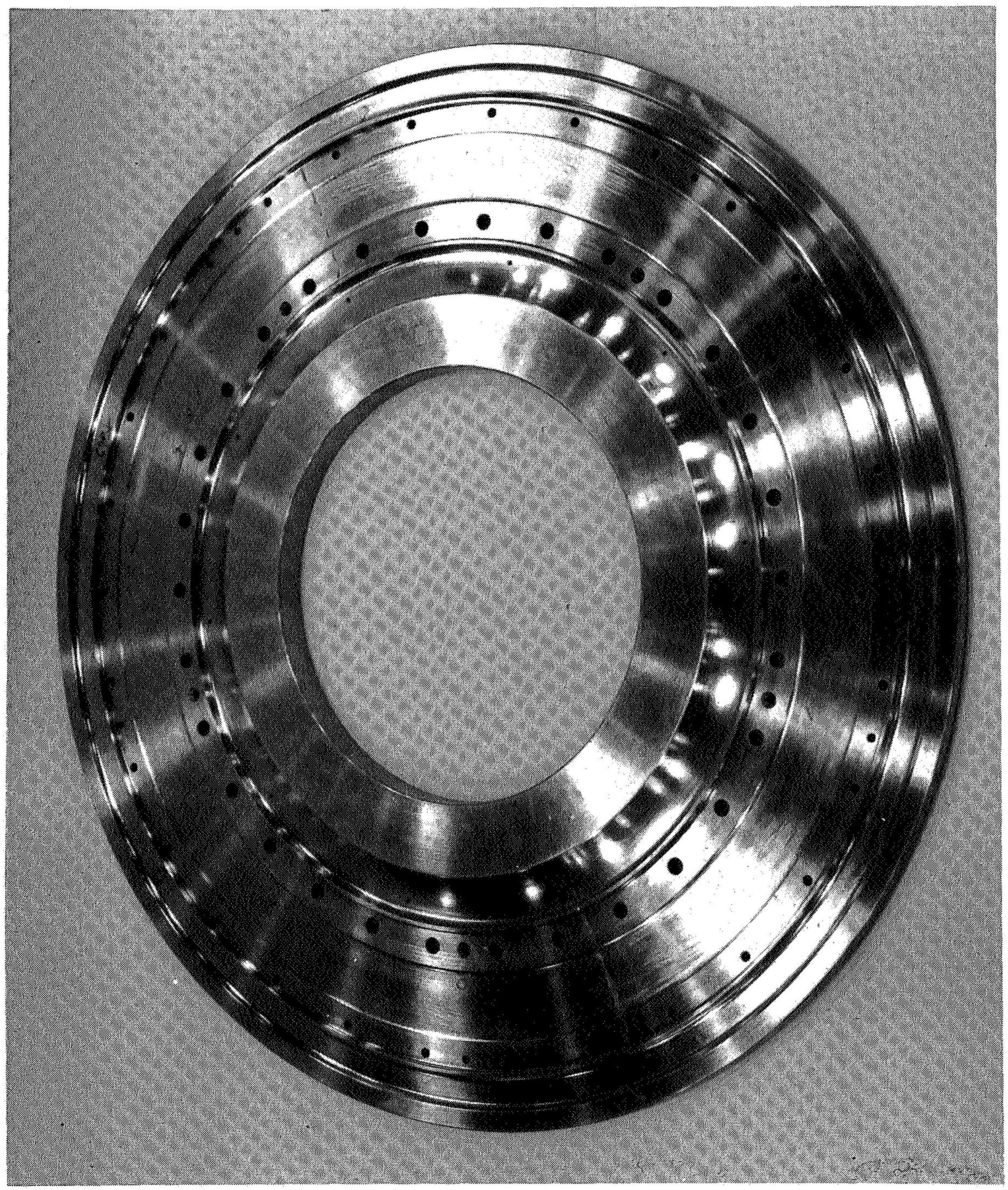


Figure 10 Test Rig Front Disk (XP-82831)

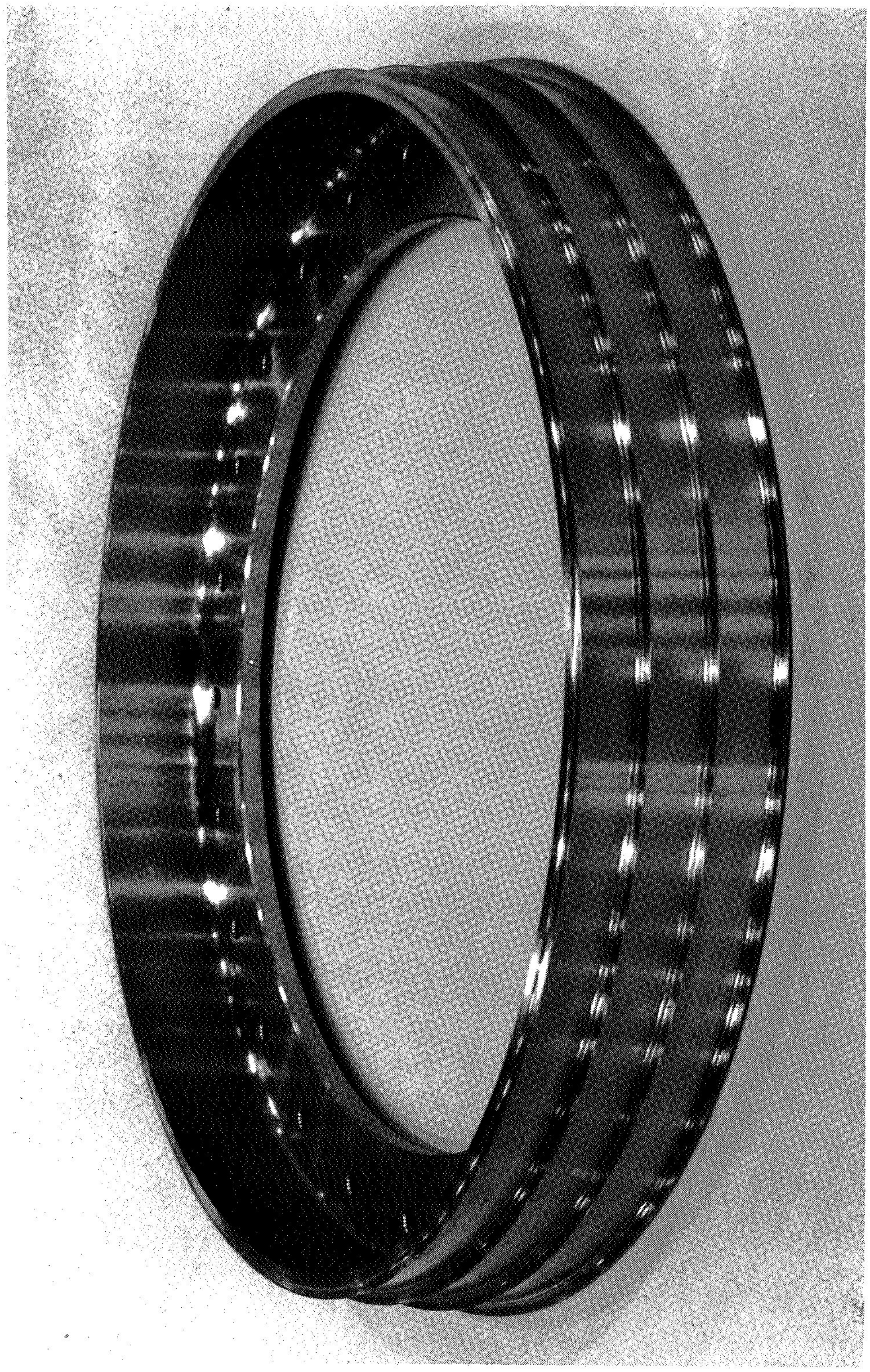


Figure 11 Test Rig Seal (XP-82832)

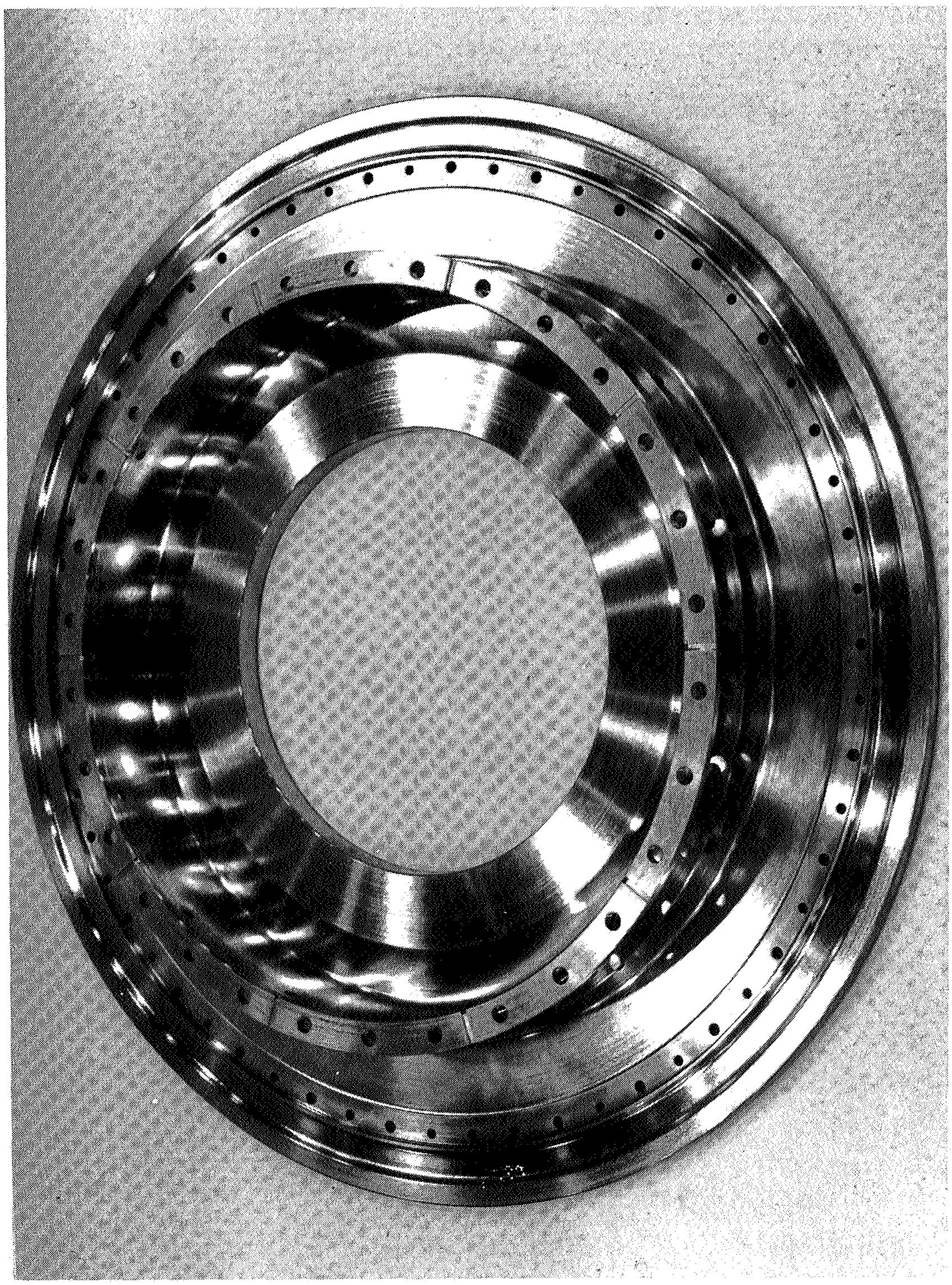


Figure 12 Test Rig Rear Seal (XP-82833)

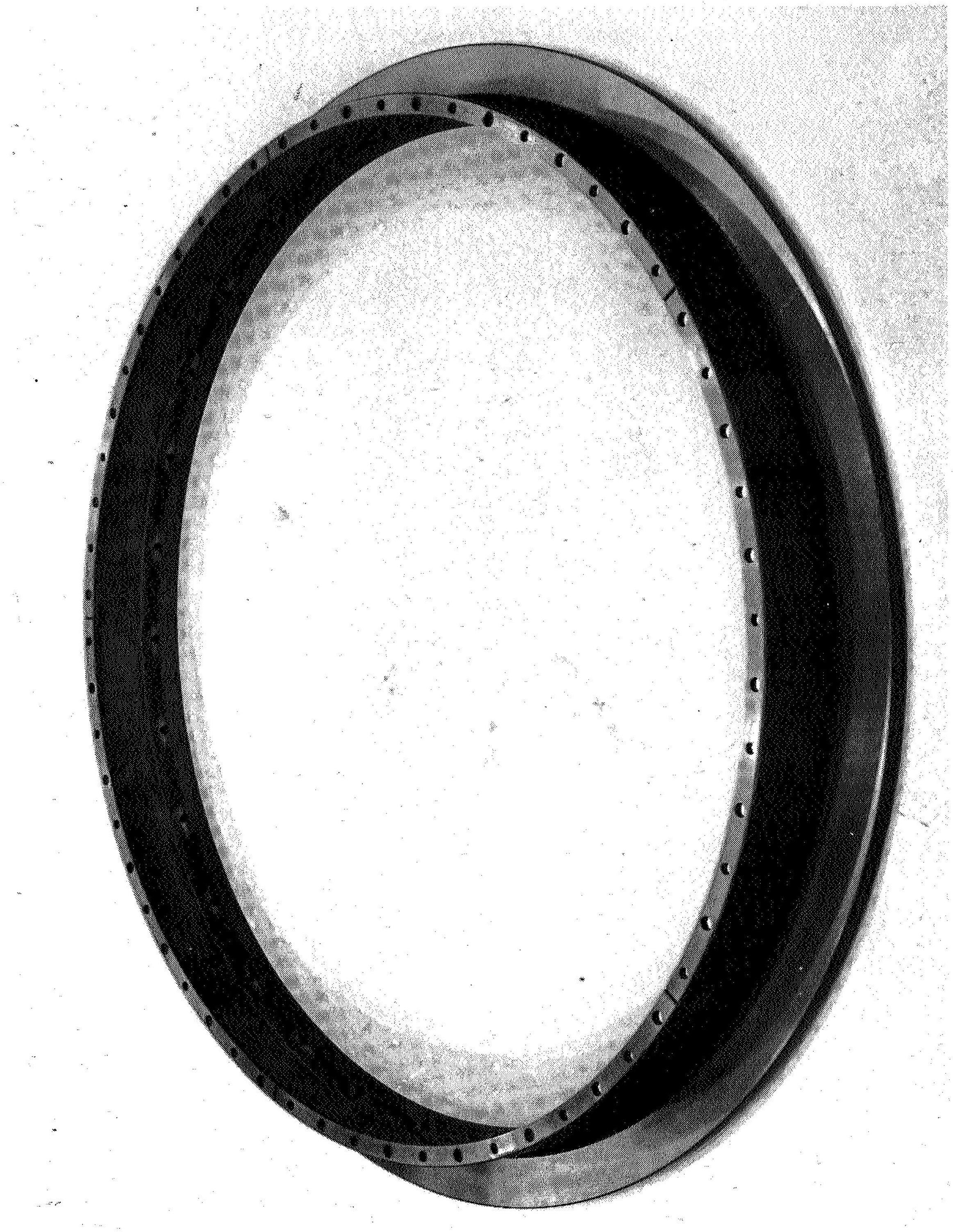


Figure 13 Test Rig Interdisk Spacer (XP-82834)

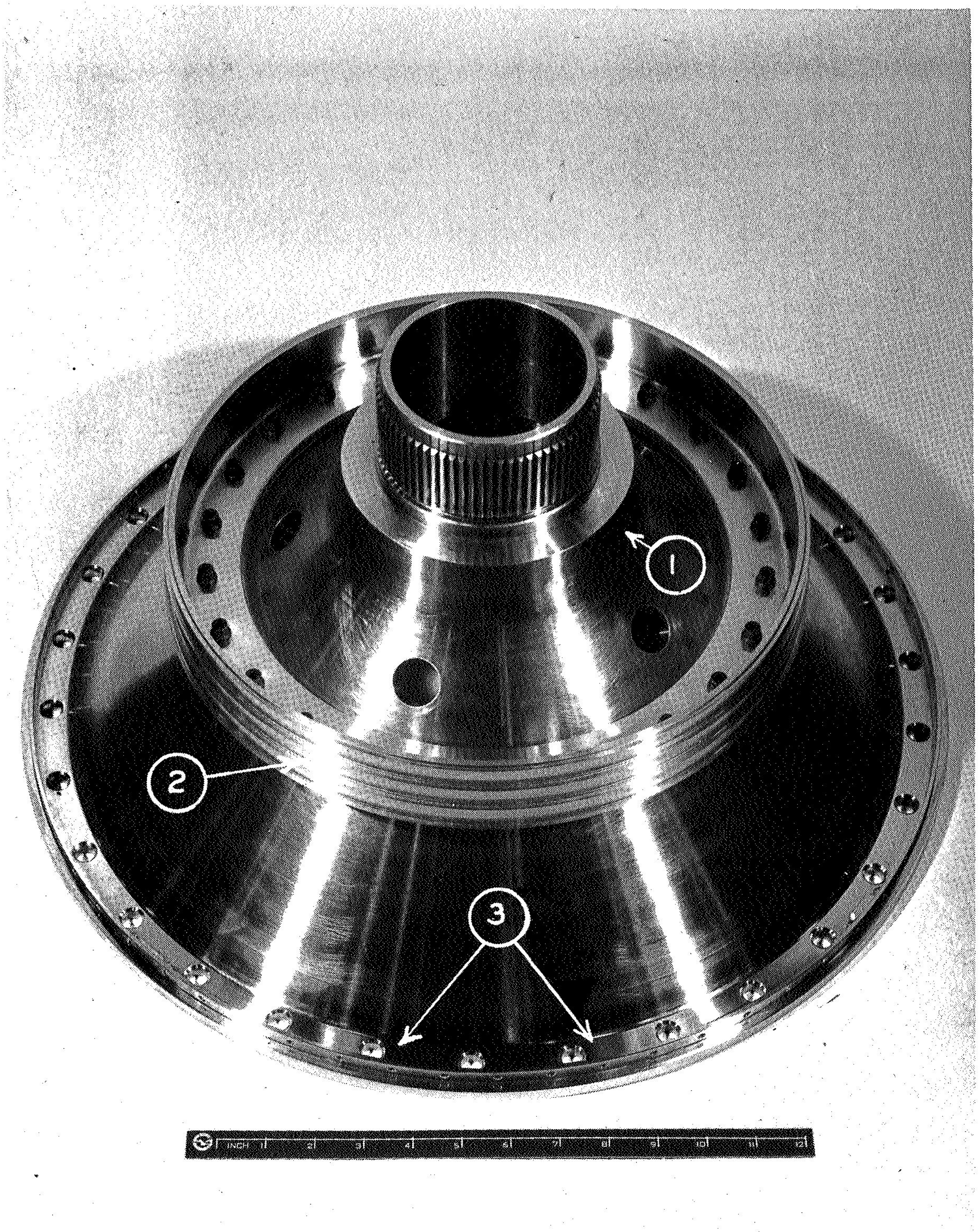


Figure 14 Test Rig Front Assembly (XP-83360)
1. Front Hub
2. Rig Seal
3. Counterweight

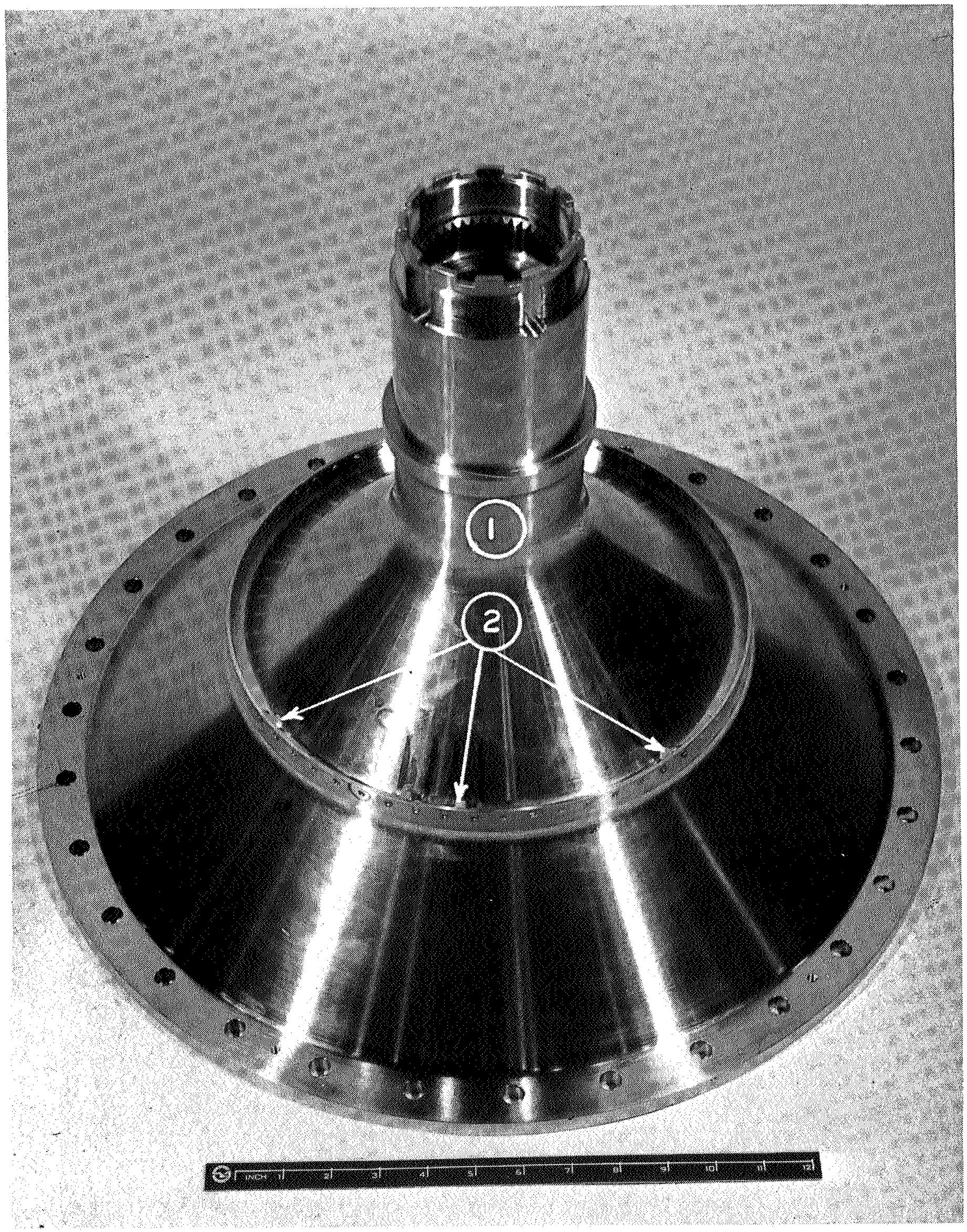


Figure 15 Test Rig Rear Assembly (XP-83361)
1. Rear Hub
2. Counterweight

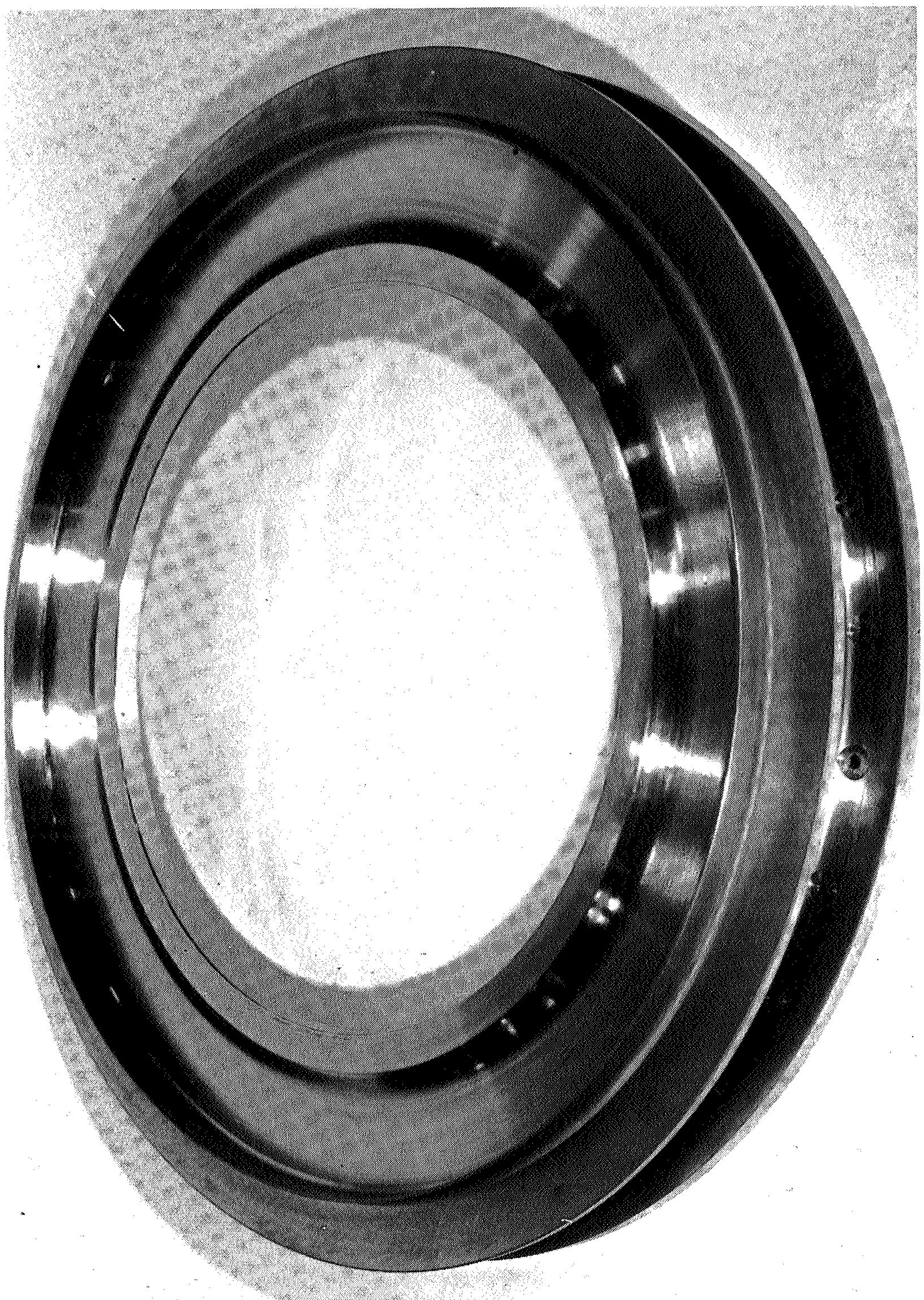


Figure 16 Test Rig End Seal Disk with Seal Face Up (XP-83362)

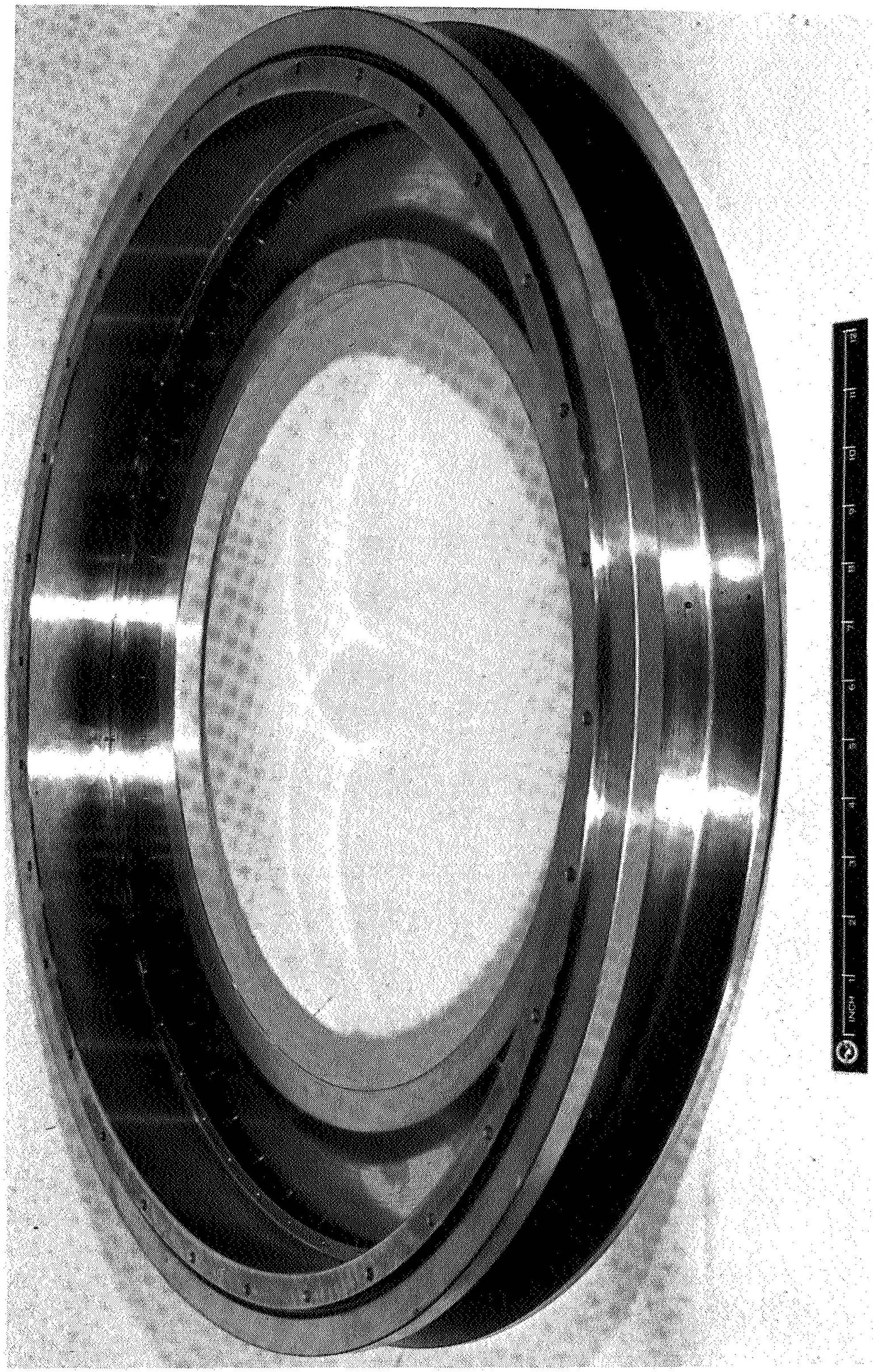


Figure 17 Test Rig End Seal Disk with Seal Face Down (XP-83363)

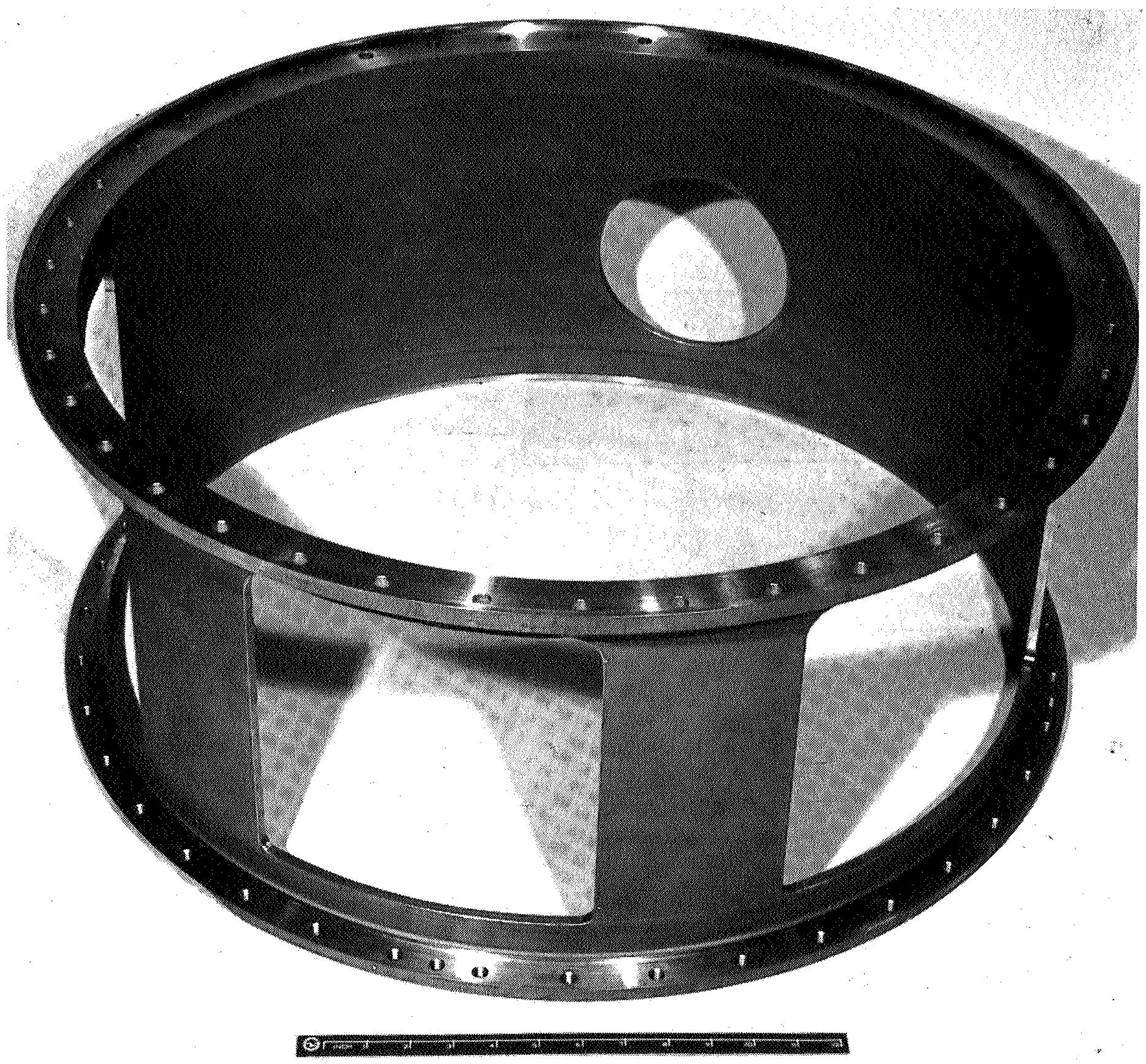


Figure 18 Test Rig Front Case (XP-83364)

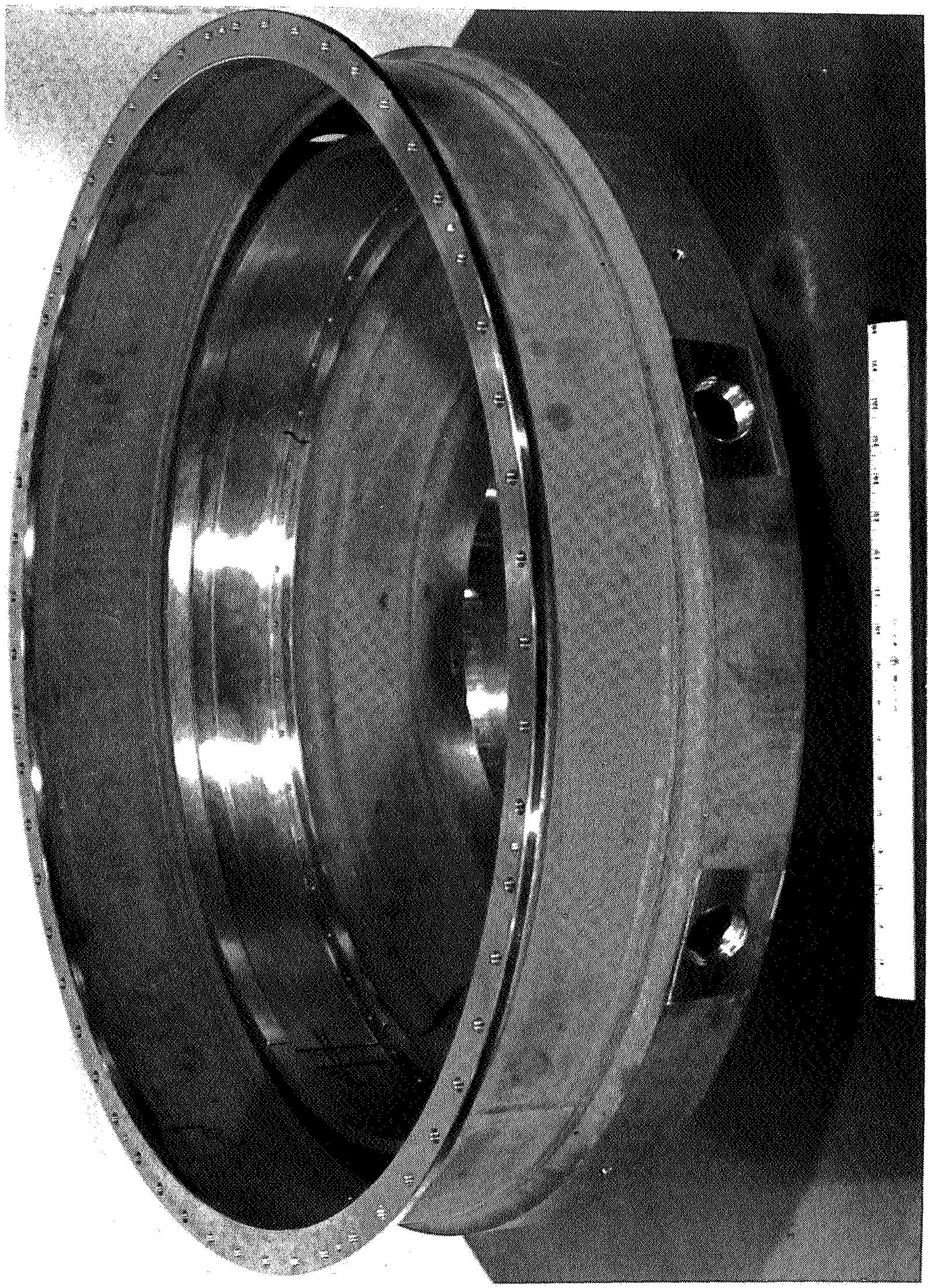


Figure 19 Test Rig Rear Case (XP-83876)

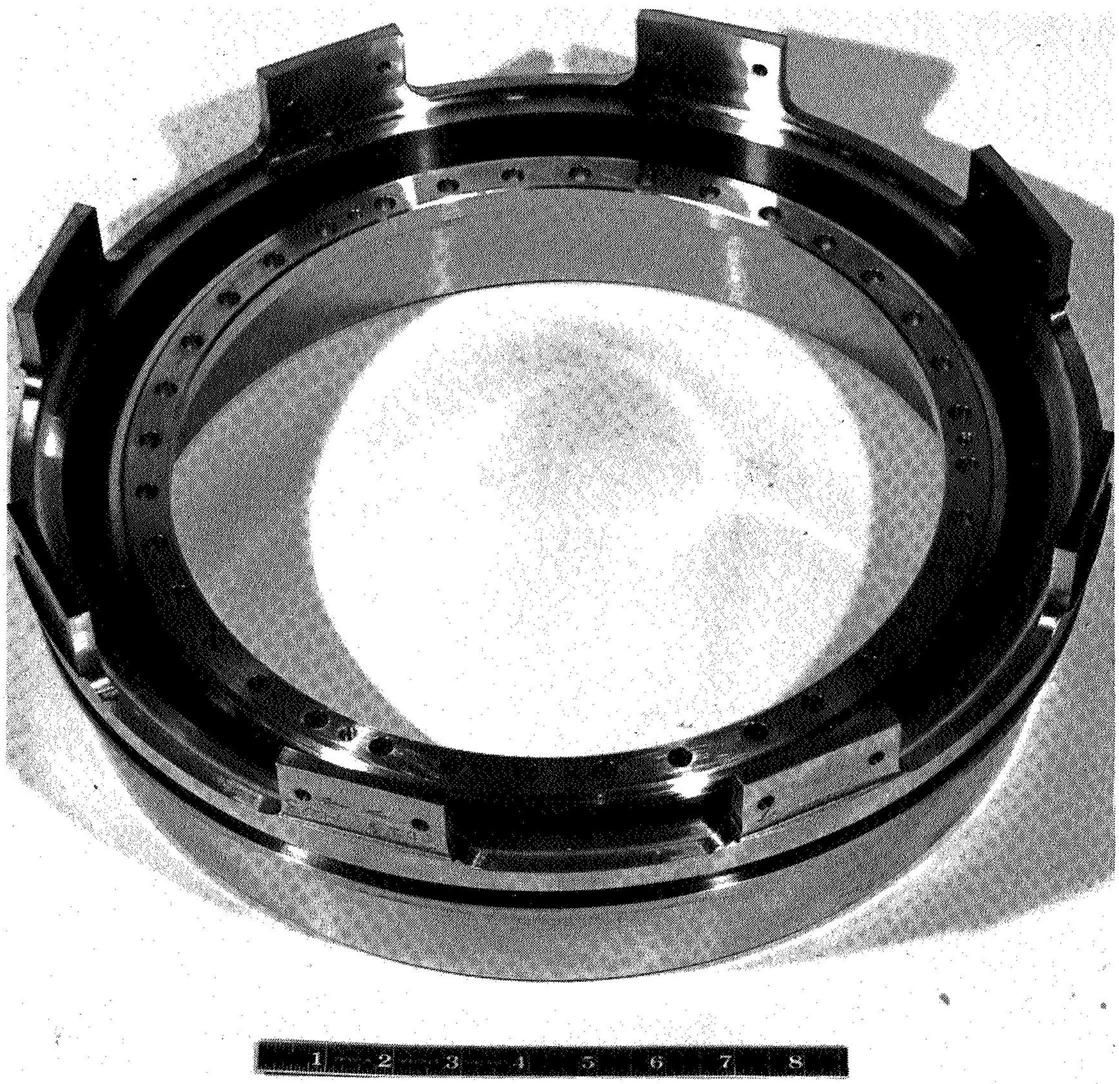


Figure 20 Sliding Support for Test Rig Case (XP - 83871)

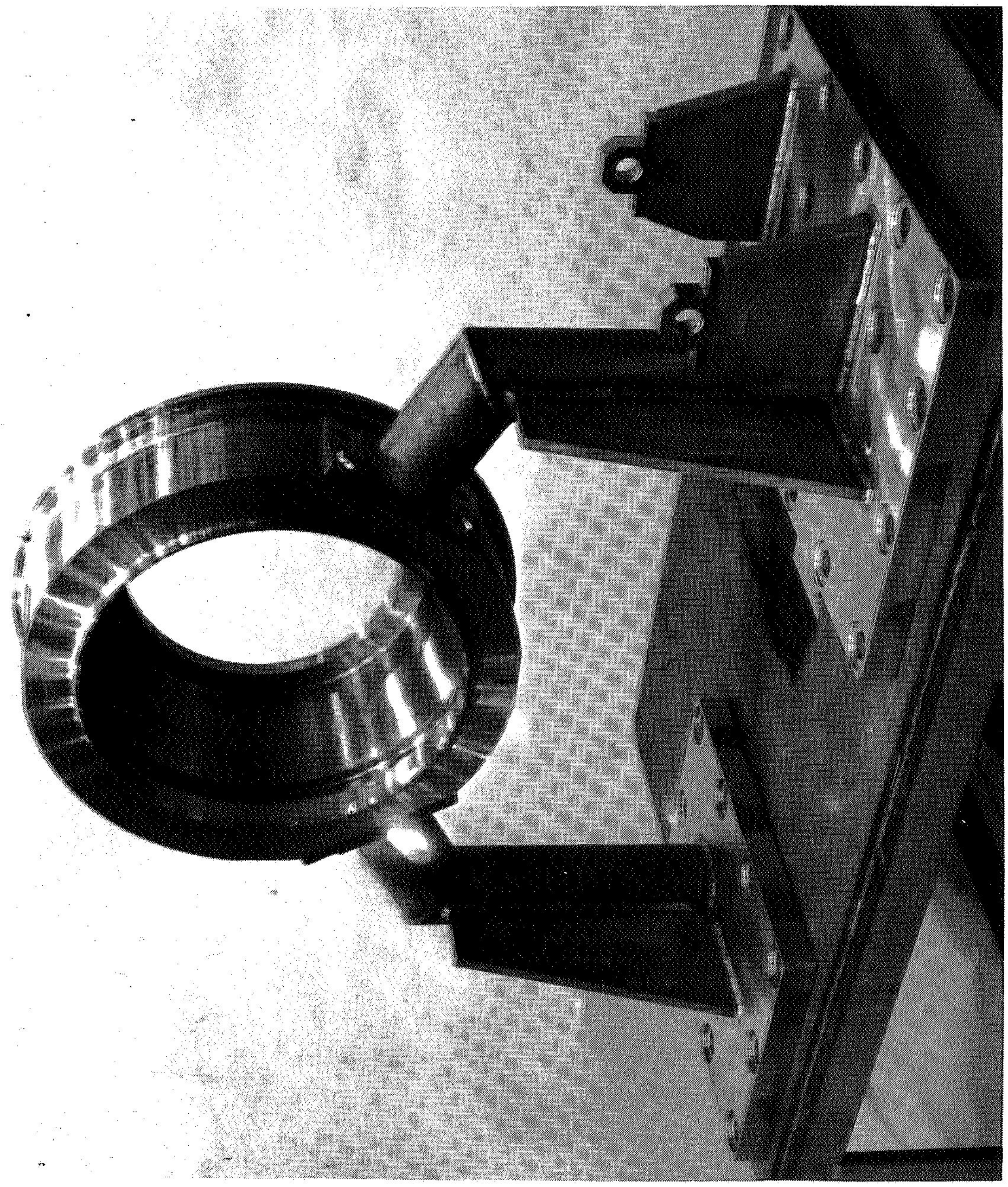


Figure 21 Front Bearing Support for Main Rig Thrust Bearings (XP-84232)

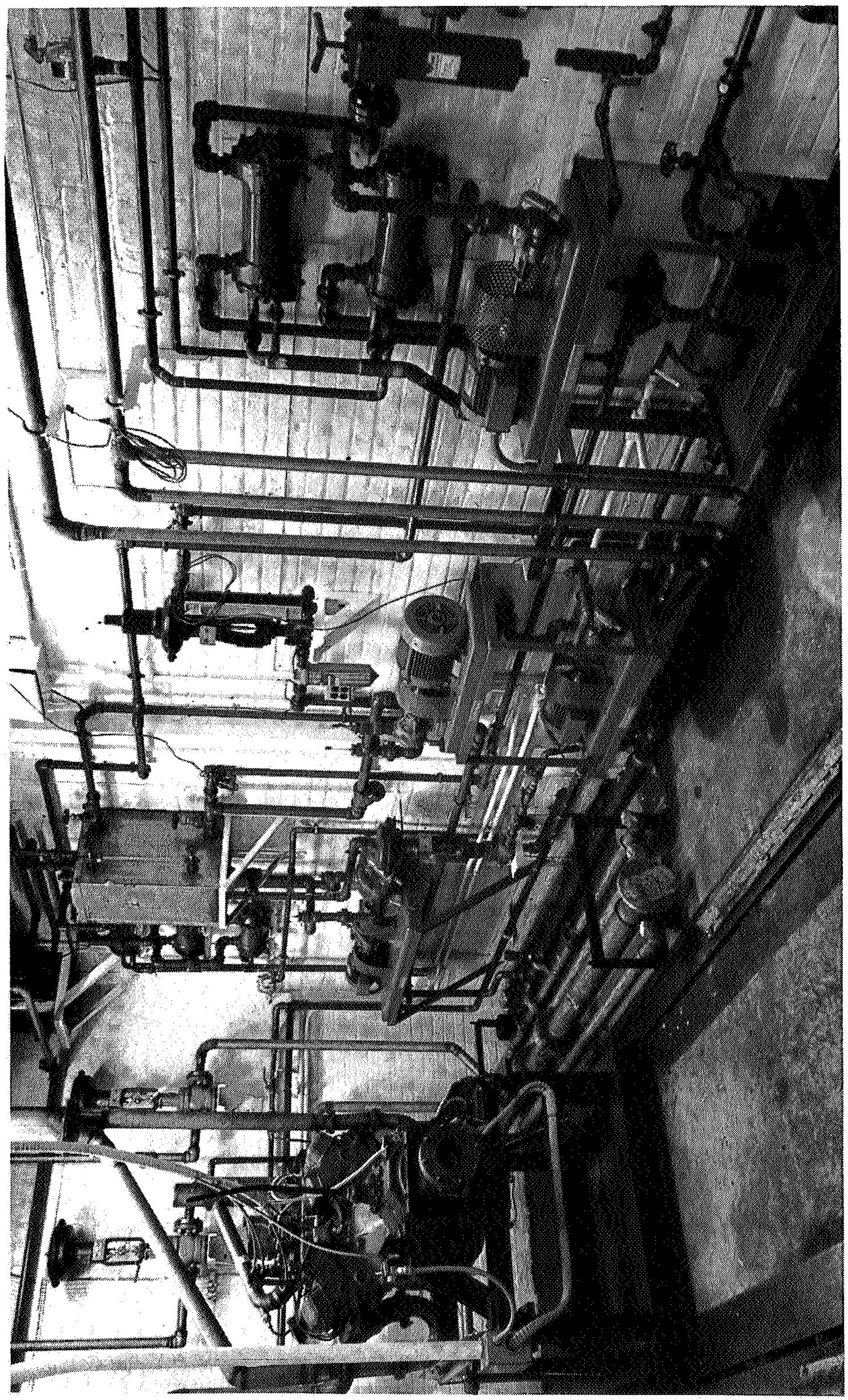


Figure 22 Interior View of the Test Stand for Task II Seals (CN-13756)

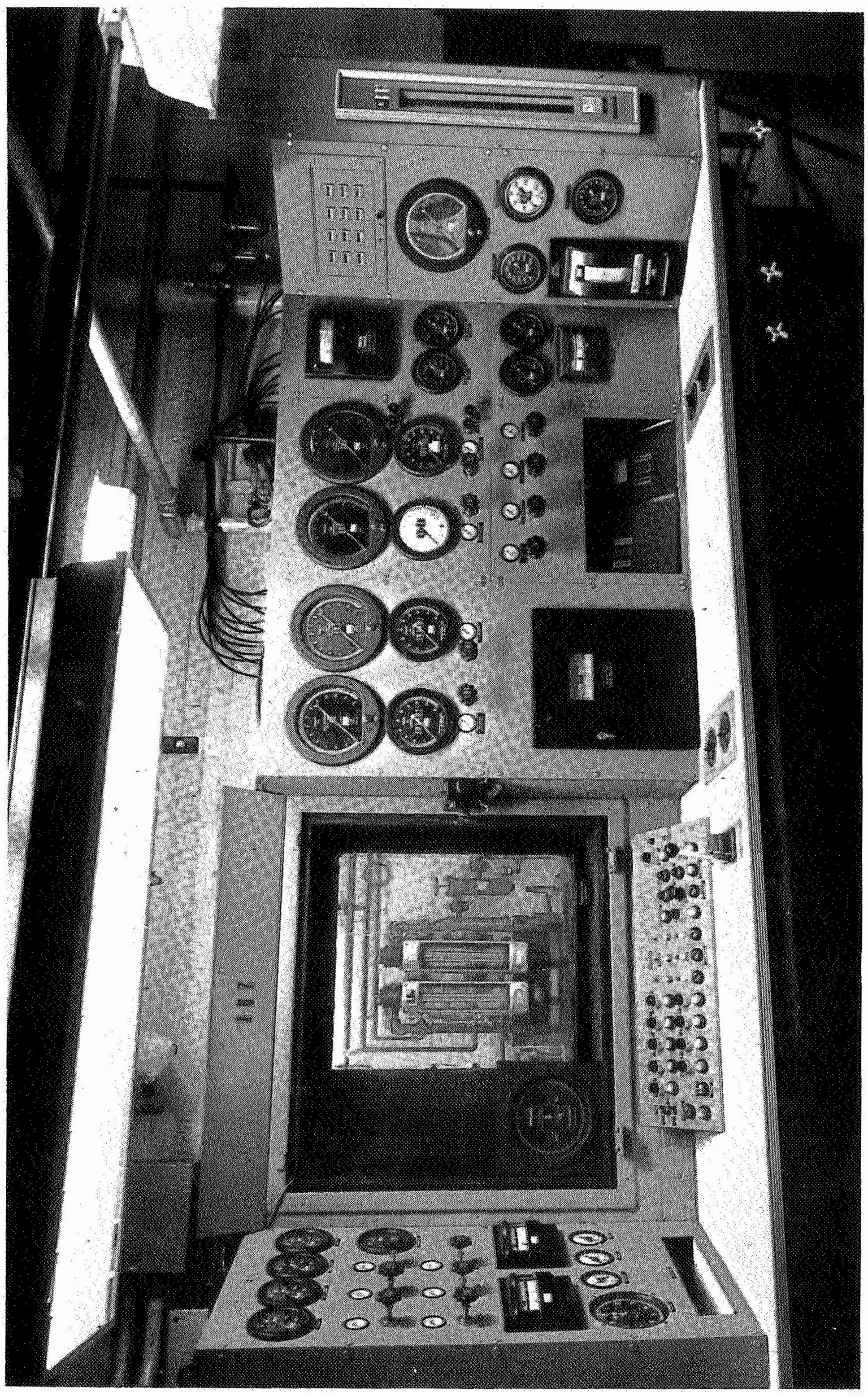


Figure 23 Control Panel for Test Stand (CN-13755)

III. TASK III - COMPRESSOR STATOR PIVOT BUSHINGS AND SEAL CONCEPT FEASIBILITY ANALYSIS

A feasibility analysis program was conducted on stator vane pivot bushing and seal concepts for application in compressors for advanced air-breathing propulsion systems. The first phase of this program consisted of a preliminary analysis and a screening of various seal concepts prior to the selection of concepts for the detailed feasibility analysis. The analytical effort included a comparison of the selected concepts to current practice and all calculations, analyses, and drawings necessary to establish the feasibility of these selected concepts. This analytical program was subcontracted to Mechanical Technology Incorporated of Latham, New York, and was monitored by Pratt & Whitney Aircraft as required under the terms of the NASA contract.

Work under Task III has been completed. Pratt & Whitney Aircraft submitted the latest feasibility designs of the single bellows and spherical seat vane pivot seals to NASA on 19 May 1966, requesting approval to start final design of these seals under Task IV. An effort was made to simplify the seal designs within practical limits without making major changes in the basic seal concepts shown on the MTI drawings. Approval was granted in a letter from NASA dated 31 May 1966.

IV. TASK IV - PIVOT BUSHING AND SEAL EXPERIMENTAL EVALUATIONS

A. INTRODUCTION

This phase of the program provides for final design and procurement of bushings and seals, design and fabrication of a test rig, and experimental evaluation of bushing and seal assemblies. The final design of the two selected concepts for experimental evaluation included all calculations, material determinations, analyses, and drawings necessary for pivot bushing and seal optimization, procurement, and experimental evaluation.

A single-vane test rig has been designed and fabricated to evaluate the two selected pivot bushing and seal designs under simulated operating conditions for the last compressor stage. The vane and actuating mechanisms are applicable to current advanced engine practice.

The pivot bushing and seal assemblies will be calibrated in incremental steps over the full pressure and temperature range, with a maximum pressure of 135 psi and a maximum temperature of 1200 degrees Fahrenheit.

The seals will be subjected to a cyclic endurance run of at least 40 hours duration following a test program which provides for simulation of take-off (20 hours) and cruise (20 hours) conditions typical of advanced engine designs through duplication of:

Compressor stage air temperatures
Supporting structure geometry
Supporting structure temperatures
Pivot movements as required for the vanes
Pivot loading (mechanical loading to simulate air loading is acceptable)
Compressor stage pressure drop.

The pivot movement will be a minimum of 13 degrees at 10 cycles per minute. The pivot loading will include a vibratory load at a convenient frequency superimposed on the steady load and equal to approximately ± 15 percent of the steady load.

B. TEST RIG

During the past six months, much of the effort on Task IV has been devoted to the test rig. As shown in Figure 24, the original design for the test rig called for measurements of airflow to be made upstream of the rig. This scheme required that all of the rig static seals and seals around instrumentation leads be capable of withstanding the relatively high pressure cycles and thermal cycles used in testing the vane pivot seals and bushings.

To improve the accuracy of measurements of leakage past the test seal, the test rig has been redesigned to allow these measurements to be taken downstream of the seal instead of being made upstream of the rig. A schematic of the new rig design is shown in Figure 25. The hardware used in the new method of measuring leakage is shown schematically in Figure 26, and a photograph of the new arrangement is shown in Figure 27.

C. SPHERICAL-SEAT VANE PIVOT SEAL

The design of the spherical-seat vane pivot seal is shown in Figure 28. It combines the thrust face and seal face and does not require a bellows. A spring is used to keep the two faces together at all times. The spherical geometry, combined with the lack of restraint on the seal seat permits it to seek its own alignment and therefore stay seated, even though there may be some shifting of the axis as a bending moment is applied to the vane. The seal is formed between the spherically concave seat located in the housing and the spherically convex seal held to the shaft. This surface is also the thrust bearing for the vane: the loading due to compressor pressure is taken by the seal, and the seat is not tightly confined perpendicular to the vane axis. This permits motion required to keep the sphere seated as a bending movement is applied to the vane. The high pressure is on the outer seal surface.

The first spherical-seat vane pivot seal has been calibrated and endurance tested with the old design of the test rig. The calibration portion of the test amounted to a little more than 11 hours of running time, and included a pressure range from 20 psi to 135 psi and a temperature range from room temperature to 1200 degrees Fahrenheit. Simulated cruise and sea-level take-off conditions accounted for the 40 hours of endurance testing. The endurance testing consisted of alternating periods of 5 hours each of cruise and take-off. At the end of testing, the test seal appeared to be in excellent condition with some wear evident on the interface between the seal seat and seal housing.

When the testing had been completed, the test seal leakage path was sealed off with a flat plate, and the rig was recalibrated. The results of this recalibration indicated leakage as great as that experienced with the test seal in place, which indicated that the static seals and brazed areas in the test rig were not satisfactory. Subsequently, the test rig was redisigned, as discussed above.

D. SINGLE-BELLOWS VANE PIVOT SEAL

The final design layout of the single-bellows vane pivot seal is shown in Figure 29. It has a flat face seal held in contact with its seat at all times by the spring action of a slightly loaded bellows. The seat against which the seal face rides is mounted as a separate piece to keep it free from distortion. The seal is formed against a seat fastened to the shaft and a face seal held to the housing by a bellows. The high pressure is on the outside of the bellows. The thrust caused by the internal pressure of the compressor is taken up on a thrust collar located at the compressor wall.

A single-bellows configuration was used for the first test of stator pivot seals under this contract. It was during this first build that many of the instrumentation sealing problems were first brought to light. It was decided at that time to sheathe the leads for the strain gauges and to braze them into holes in the seal housing. It was also decided to pass the leads for the thermocouples through stainless steel tubes, braze the tubes into the seal housing, and seal the tubes at the other end. The second build of the test rig (which used the spherical-seat seal configuration) included both of these changes.

In some areas of the single-bellows seal, the vendor has had difficulty in holding the required tolerances. Pratt & Whitney Aircraft has set up a continuing program to assist the vendor to reoperate those parts which require it.

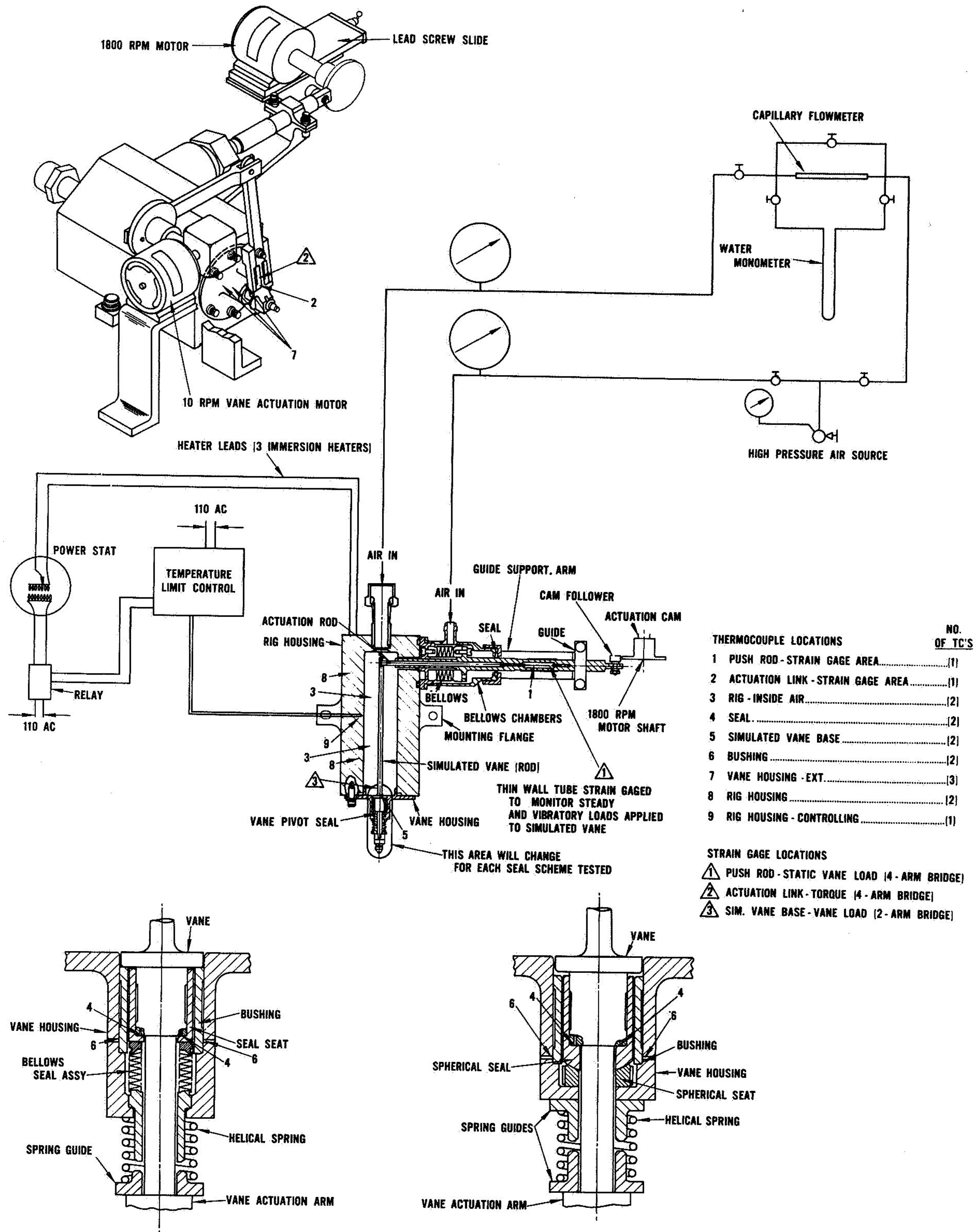


Figure 24 Schematic of the Original Design of the Stator Vane Pivot Seal Test Rig (XP-82625)

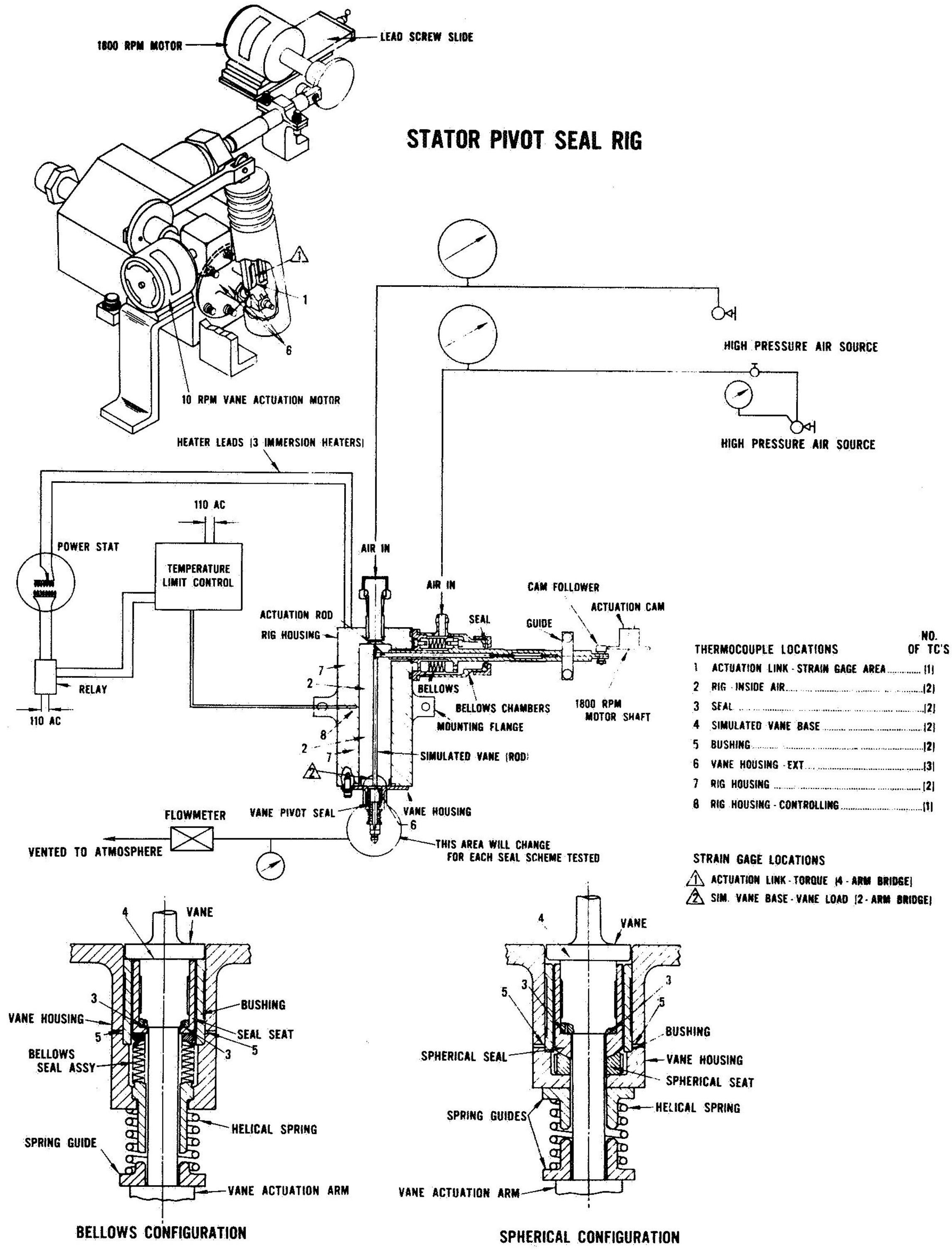


Figure 25 Schematic of the New Design of the Stator Vane Pivot Seal Test Rig

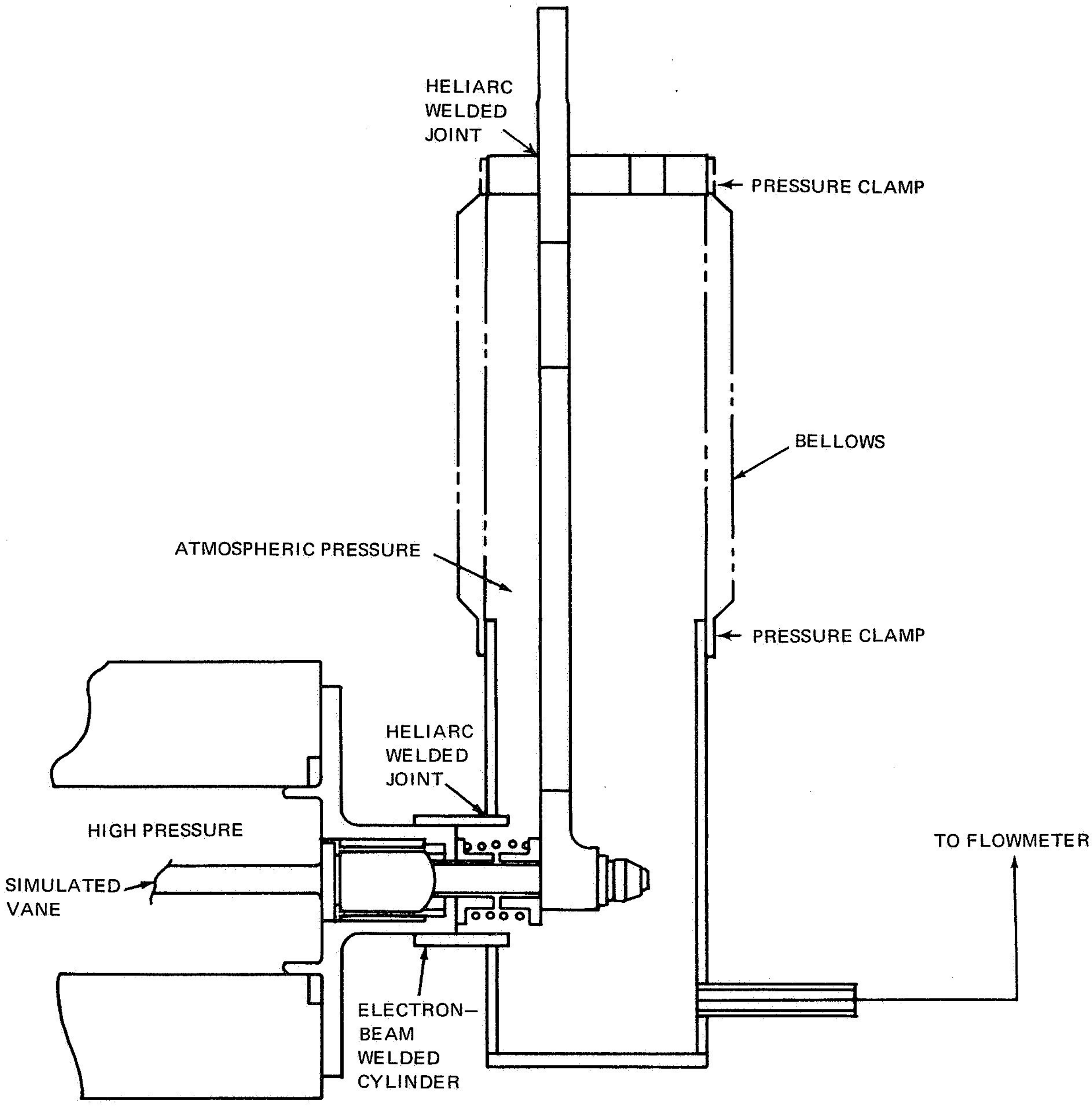


Figure 26 New Scheme for Measuring Leakage Past Task IV Seals

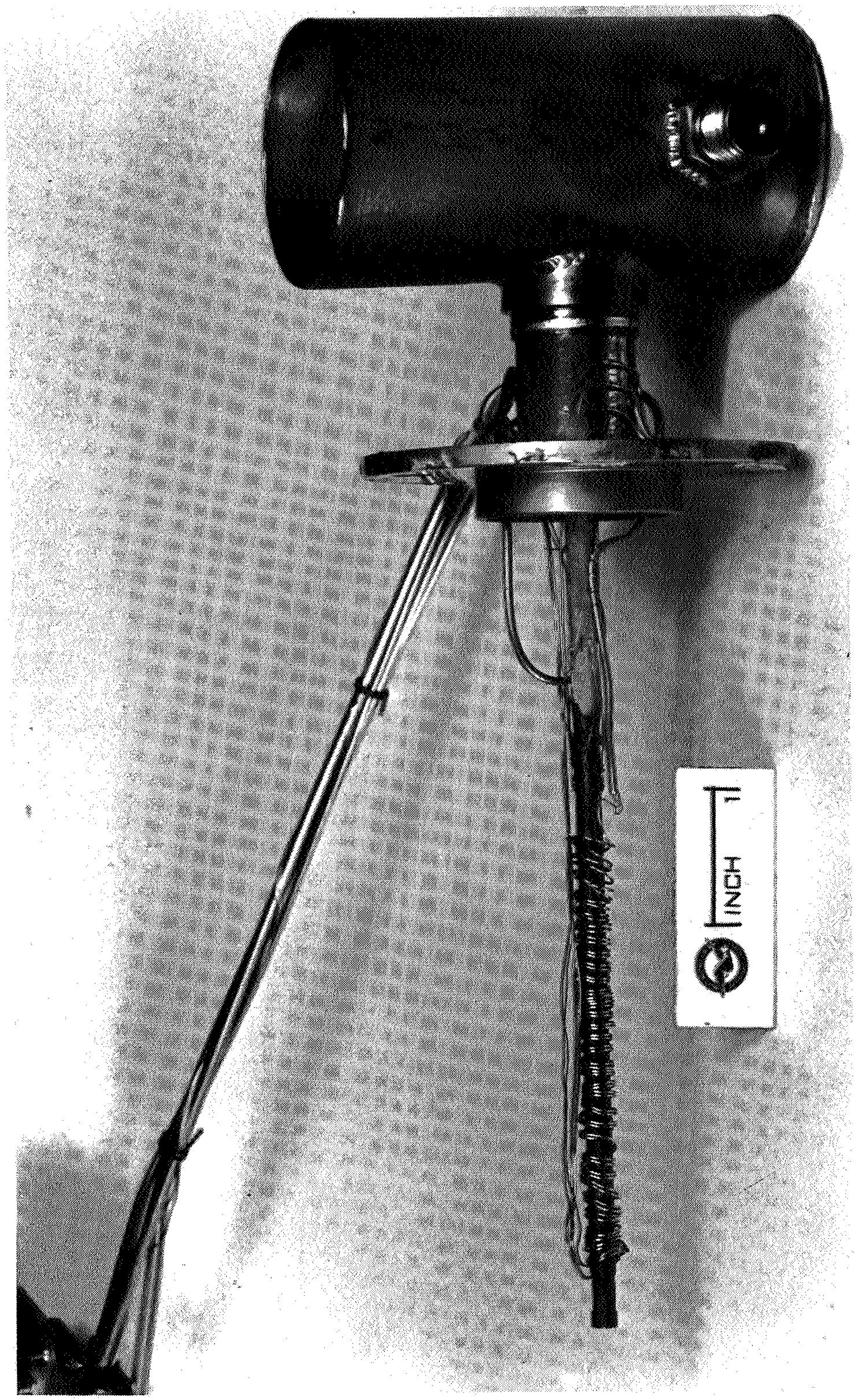


Figure 27 Seal Assembly with Leakage Collector (XP-87848)

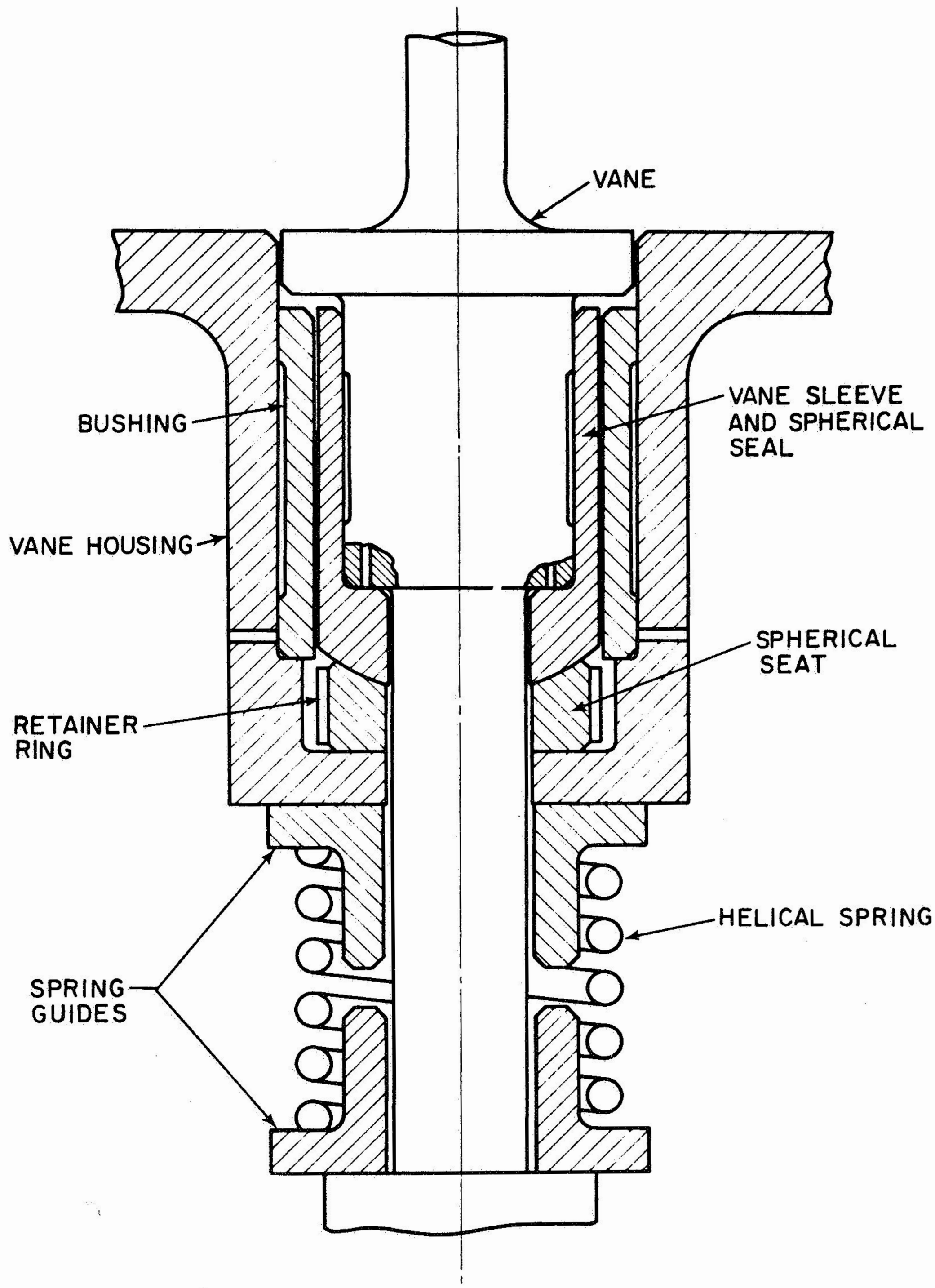


Figure 28 Schematic of Spherical-Seat Vane Pivot Seal

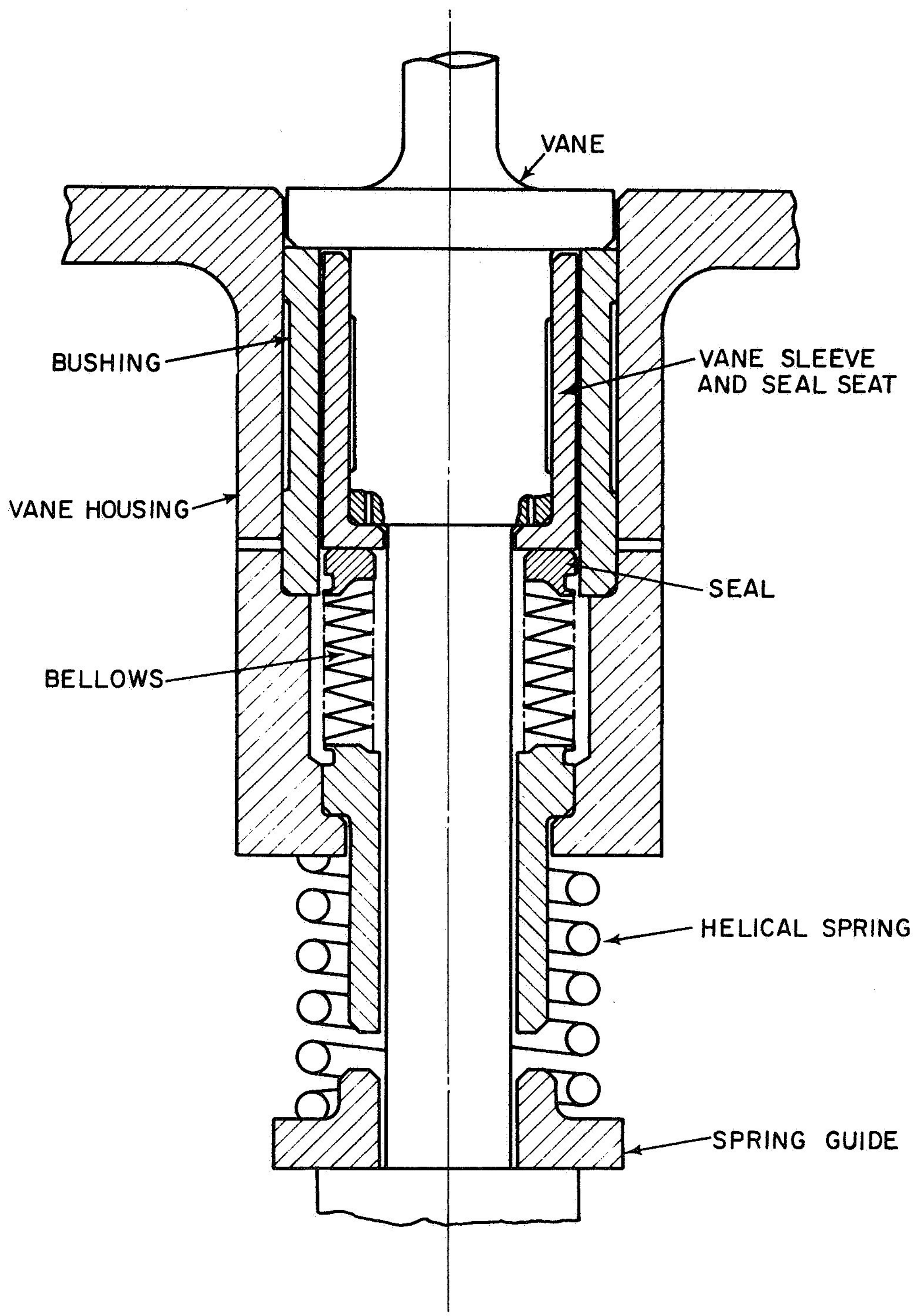
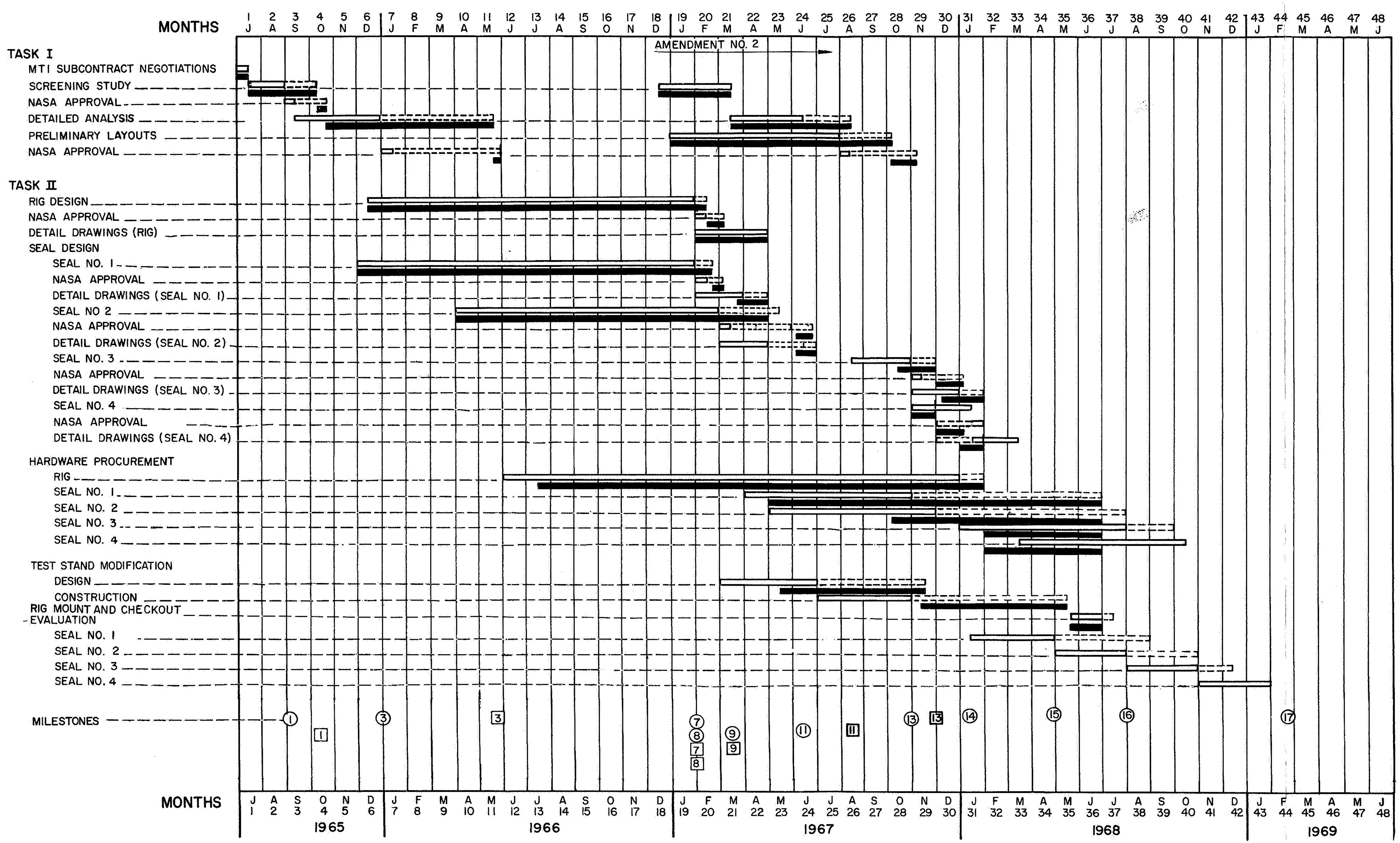


Figure 29 Schematic of Single Bellows Vaen Pivot Seal



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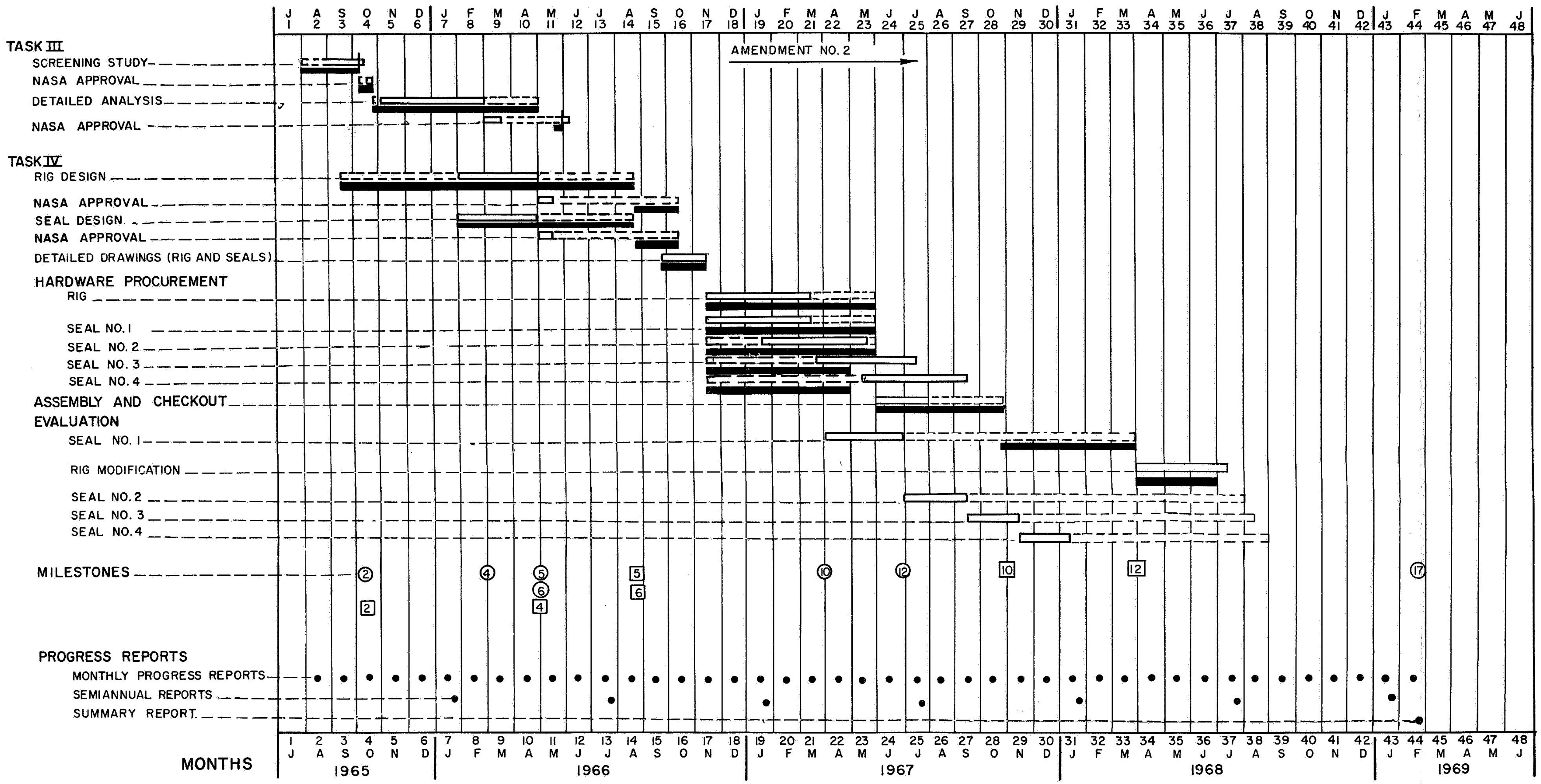

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