ALKALI METAL BEARING & SEAL DEVELOPMENT AT SPACE POWER AND PROPULSION SECTION

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ABSTRACT

SPPS engaged in three major programs associated with the development of Rankine cycle power conversion technology for space application: The low viscosity bearing stability investigation under NASA contract, the development of dynamic seals under Air Force contract and the preparations for a liquid metal bearing test, simulating the conditions of a space generator, on CIRP funding. All programs were supported by a corresponding analytical effort, presently geared towards predicting bearing/rotor response. Experimental results are expected in 1966, and 1967.

Screw seals, rotating channel and slinger squeeze seals were investigated in the dynamic seal program. Interface instability was identified as one of the major problems. It was resolved for the case of the slinger squeeze seal, which was successfully tested in potassium.

DISCUSSION

Alkali metal bearings and seals comprise two of the basic components necessary for reliable operation of Rankine cycle turbomachinery for space. As a result of the high temperatures and radiation fields in nuclear space power systems, and the general complexity of multi-fluid systems, present turbogenerator bearings and seals employ the primary thermodynamic fluid of the system in their operation. SPPS is engaged in the development of alkali metal bearings and seals for incorporation in a full turbogenerator system.

A typical 350 KWe turbogenerator for space power conversion - General Electric concept - is shown in Figure 1. It consists of a six-stage turbine driving an axial gap generator. Each component is supported on two radial bearings and a thrust bearing. The two components are connected by an elastic coupling. The design has been optimized for minimum rotor weight and distribution of this weight over four equal size pivoted pad bearings. Each component, turbine as well as generator, has its own thrust bearing primarily to fix the rotor position accurately, in relation to its stator. Thrust loads are minimized in the turbine by proper fluid design, and in the generator by carefully positioning the rotor in the middle of the stator. The generator cavity is

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open to space and is based on the availability of zero leakage seals. The generator is uncanned and its temperature distribution is so that no potassium vapor condensation or accumulation of condensate within the generator housing will take place.

Within SPPS, bearing and seal development is being carried out under four different programs, three of which are basic, while the last one will combine the knowledge gained in the three previous ones. Of the three basic programs, the first one is devoted to the study of bearing stability which implies the establishment of proper bearing geometries and the verification of bearing rotor response behavior. This program is carried out under NASA contract and uses water at 120°F as a lubricant (Appendix A and B).

The second program is devoted to the establishment and selection of suitable bearing materials. Under this program (Contract No. NAS 3-2534, Appendix C), accurate physical properties have been established for these materials, such as hot hardness, compressive strength, modulus of elasticity, thermal expansion, potassium wetting and dimensional stability at 1600°F. Also, corrosion resistance and compatibility in potassium. Friction testing is being carried out in two highly sophisticated test rigs, one operating in a vacuum and the other one in a potassium atmosphere. This program favors extremely hard and stable refractory materials primarily Tungsten and Titanium Carbides using special binders. This program will not be further discussed here because of the limitation in time. It covered three years and is close to its completion, so its results should be available in a reasonable time.

The third program was carried out under Air Force contract and was devoted to the development of zero leakage seals employing potassium as a sealant (Appendix D). Its main application will be in the electric generator for space.

The fourth program, which is based on the knowledge accumulated in the three aforementioned programs, is carried out under Company Research Funds and has as its objective the simulation of the bearing and seal arrangement in a space generator. This presents the more difficult problem than the space turbine due to rotor and containment materials involved. This program will comprise performance and endurance testing of a 70-pound rotor operating on large carboloy potassium bearings and a zero leakage seal exposed to vacuum (Appendix E).

Bearing Stability Program

The first portion of the Bearing Stability Program was carried out in 1961 through 1963 under NASA Contract No. NAS 3-2111. The test rig in this time was designed and manufactured by SPPS. Testing and test evaluation, however, was carried out in the Bearing and Lubricant Center in Schenectady by Gerry Fox and his staff. The program was continued in 1965 under a new contract, NAS 3-6479 (Appendix A and B).

Under the first contract, six different bearing configurations were compared in regard to their stability behavior, especially their capability of avoiding or suppressing partial frequency whirl in the turbulent operating regime. Out of this program, the pivoted pad and the three-lobe bearing emerged as the outstanding candidates for stable operation up to high speeds. These two bearing types are being further investigated under the present contract under aligned as well as purposely misaligned conditions. The intent is also to study the effects of seizure of pivoting points on the four-pad pivoted pad bearing by running tests with one or two pads fixed.

The test rig for this purpose had to be drastically modified. In addition, better and more instrumentation has been introduced. Shaft motions are monitored by four Bently gages per plane, arranged in four separate planes.

Opposed gages are used combining their signals in summing amplifiers to avoid errors due to differential thermal expansion. Gages are arranged on both sides of each bearing to be able to measure shaft bending under aligned as well as misaligned conditions. As a new venture, provisions have been made to measure bearing loads and to establish the load vector by size and angle during testing up to 500 cycles per second. The Bearing and Lubricant Center has had good success in making such measurements by using strain gaged buttons for shaft speeds up to 7,000 rpm. For high speed testing up to 30,000 rpm, originally similar strain gaged buttons had been considered; however, piezoelectric force gages were finally selected due to their high spring constant and their extreme capability of picking up fractions of a pound under preloads of several thousand pounds. Each bearing support is suspended on Kistler force gages in two planes, adequate to register the forces transmitted to the bearings.

Another innovation is introduced in this test program by using a digital data handling system for data taking. In high frequency testing, the normal tools are oscilloscopes and photography, as well as tape recorders. When it comes to the evaluation of test results, both of these methods are inadequate and it is usually an extremely time-consuming and difficult process to reduce tape recordings to printed language. In using a digital data handling system, special precautions have to be taken in measuring high frequency responses. This is done by using peak-to-peak detectors for forces and displacement and average level detectors for equilibrium position of the shaft. The other inputs, such as speed, torque, flows, temperatures and pressures, are straight forward. Accurate measurement of the phase angle constitutes a problem in itself and has been accomplished to within $\pm 2^{\circ}$F.

The test rig is shown in Figure 2. It comprises a 1.25-inch diameter shaft supported by two test bearings approximately 12 inches apart. The shaft is driven by an electric motor receiving its power from a variable frequency power supply through a quill shaft to speeds of 30,000 rpm. Side loads are imposed on the test shaft by two partial-arc loader bearings pneumatically operated. The lower bearing assembly can be adjusted both transversely and angularly to allow variation of bearing alignment. Measurements of shaft motion and force transmitted to the bearings are made in four planes, located on both sides of the test bearings as shown in detail in Figure 3. Shaft motion is monitored by Bently inductance gages in an assemblage shown in Figure 4. The gages have Teflon tips to protect them against water, as suggested by J. McHugh. To eliminate the effect of shaft inhomogeneities on gage readings, the shaft is silver plated in the zones of gage measurement

(Figure 5). A measuring accuracy within $\pm$ 50 micro-inch is an objective hard to meet. The alignment of the bearings prior to testing is accomplished optically by a high-precision telescope in conjunction with optical targets inserted in the bearings, Figure 6.

Figures 7 and 8 show the test rig under testing conditions at room temperature (open) as well as temperature controlled for 120°F operation (closed).

Figure 9 shows the instrumentation and control panel, subdivided into five compartments, one handling the lube flow, pressure and temperature control, another for power and frequency remote control of a variable frequency power supply arranged in another building. The third panel holds a speed counter, the proximity gage oscilloscopes, a temperature recorder and electronic equipment producing the signals for a digital recording system, also arranged in a separate building. The fourth panel is devoted to phase angle and force measurements. The fifth panel holds the equipment required for vibration and torque readout.

Figure 10 shows the pivoted pad bearing design incorporated in the present test program. It has four pads with gimbaling points at 55 percent partial arc. Figure 11 shows the calculated load carrying capacity of this bearing with oil and with liquid metal as lubricants. This graph brings out the weakness of the low viscosity bearing resulting in a load carrying capability of roughly 1/1000 of the oil lubricated bearing.

Computer programs are available which allow prediction of the dynamic response of a given bearing-rotor configuration. One such program was generated by G.E., Bearing and Lubricant Center. Another one was published by MTI under Air Force contract. SPPS has used both of these programs and has generated an additional program which combines VAST and LSE*, offering higher accuracy and flexibility. In the near future, a comparison of actual test results with those predicted will be possible. Figure 12 shows a typical predicted curve of bearing stiffness as a function of rotative speed, which in turn is used to calculate fluid film forces as shown in Figure 13. Figures 14 and 15 show typical amplitudes of shaft motion as a function of speed with two different levels of shaft unbalance in the easy-fluid test rig, calculated by using the MTI computer programs. In these curves, the occurrence of critical speeds can be identified with the high amplitude zones, and it can be seen that for the larger value of unbalance, the rotor amplitude actually exceeds the bearing clearance, predicting failure of the bearing. As a major test objective, generalized dynamic coefficients will be obtained for bearing rotor system operating conditions involving the practical problems of misalignment and varying support structure stiffness, both of which are difficult to preduct analytically.

Dynamic Zero Leakage Seal

A program was conducted by SPPS on the development of such a dynamic seal, Figure 16. Considerable testing in water was done to identify the proper seal concept, and to prove its hydrodynamic capabilities. Ultimately, the seal was successfully tested in potassium (Appendix D).

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* VAST - Vibration Analysis Systems
LSE - Linear Simultaneous Differential Equations

Figure 17 shows some of the seal concepts investigated. All of these share the common characteristic of being non-contacting hydrodynamic seals, which is a necessary requirement for long-time operation in space turbomachinery. Figure 18 shows the physical properties of potassium which assist the successful functioning of such a seal, where, because of low vapor pressure, the molecular diffusion from a free liquid surface to the vacuum of space is small.

A detailed configurational drawing of the dynamic zero leakage seal ultimately tested is shown in Figure 19. Liquid is injected into the rotating disc cavity, and sealing is accomplished by centrifugal force of the liquid. Any migration of liquid along stationary walls toward the vacuum side results in re-ingestion of the liquid into the seal before it escapes. Figure 20 is a photograph of the disassembled seal parts. Testing was accomplished in the liquid metal bearing and seal facility in Figure 21, which will be used in the forthcoming liquid metal bearing development program. Figures 22 and 23 show typical performance curves for seals of diameters of interest.

Figure 24 shows the liquid potassium which was accumulated in a vacuum tank during 173 hours of testing the dynamic seal. Most of this was accumulated during two startups and shutdowns.

<u>Liquid Metal Bearing Test Program (Appendix E)</u>

The liquid metal bearing test rig, shown in Figure 25, is similar to the previously discussed water test rig, in that two test bearings support a vertical shaft driven by an electric motor. The rotor weighs 70 pounds and simulates the mass and dynamic characteristics of an axial gap generator. Along with the very accurate manufacturing requirements of the bearing, the design of a bearing for the generator shaft constitutes a severe mechanical design requirement, since the rotor of the generator must be made from magnetic steel, the casing from stainless steel and the bearing is fabricated from refractory metal carbides having thermal coefficients of expansion of roughly one-third those of steels, Figure 26. A number of geometric solutions to the differential thermal expansion problem have been investigated, involving radial springs, radial faced teeth, and conical faces as shown in Figure 27, the latter approach having been selected as the most practical. The bearing detail is shown in Figure 28, and comprises four segmented pads, retained by spherical pins, allowing self-alignment of the bearing during operation. The pads are weight balanced so that their centers of gravity coincide with the pivot contact points to minimize fretting motions. The entire bearing region is enclosed by close-running shaft screw seals, and is flooded with potassium. A side load of 200 to 300 pounds can be applied to the rotor by generating a magnetic field with an induction coil. The test rig has other new features such as a high temperature induction gage measuring the potassium film thickness through an aluminum oxide window against an aluminum oxide ring which is part of the rotor. The lower shaft end is exposed to vacuum and sealed off by a zero leakage seal modified for a low power requirement. Dynamic sealing, in this case, is accomplished by a screw seal. An additional lift-off face seal provides sealing at standstill and during startup. Figures 29 and 30 show rotor and stator hardware. The test rig is presently being assembled and will undergo its easy-fluid checkout in November 1966. Liquid metal testing is scheduled for the first quarter of 1967.

Conclusions

The engineering know-how required to resolve the complex problems of a space turbogenerator operating on liquid metal bearings and employing zero leakage seals is progressing rapidly. As compared to eight years ago, when even the problems involved were poorly identified, much knowledge and much practical experience has been accumulated today. Computer programs are available and will eventually reach close agreement with actual test conditions. SPPS is prepared to contribute to this general realm of knowledge by coordinating computer predictions with actual test results, by establishing best qualified pairs of materials for long-time operation of liquid metal bearings and by developing seals which will combine the zero leakage feature with a minimum power consumption.

As a bearing configuration, SPPS presently favors the pivoted pad bearing not because of its resistance to partial frequency whirl, but also because of its self-adjustment capability to misalignments or structural distortions. It is realized that the pivoted pad bearing is complex and has an inherent weakness of fretting or seizing of pivoting points. A less complex stabilized sleeve bearing with equal resistance to partial frequency whirl and supported in a way providing equal self-adjustment to misalignments would be desirable.

The differential thermal expansion problem of the Tungsten carbide bearing in a steel environment is being actively attacked in our liquid metal bearing program. Answers and test results should be available in 1967.

APPENDIX A

TITLE OF CONTRACT:	JOURNAL BEARING INVESTIGATIONS
NASA CONTRACT NUMBER:	NAS 3-2111
DATE CONTRACT WAS ISSUED:	OCTOBER 10, 1961
DATE OF FINAL REPORT:	DECEMBER 31, 1963
NASA CONTRACTING OFFICE:	LEWIS RESEARCH CENTER
NASA PROGRAM MANAGER:	JOSEPH P. JOYCE

EXCERPT OF WORK STATEMENT

The Contractor shall conduct an investigation of hydrodynamically lubricated journal bearings having the capability to suppress shaft whirl while operating at high journal speeds and with zero bearing load.

The Contractor shall fabricate a test bearing rig with a vertically mounted shaft (1.25-inch diameter) and speed capability up to 35,000 rpm.

The test rig shall be designed with the facility to alter and to control accurately the following input variables:

1. Shaft speed
2. Bearing loads
   - Unit directional
   - Rotating
3. Inlet lubricant pressure
4. Inlet lubricant temperature
5. Lubricant flow rate

The test rig shall be so designed and instrumented to facilitate the determination of impending whirl speed and accurate readout of the following data:

1. Bearing load
2. Inlet lubricant pressure
3. Inlet and outlet lubricant temperature
4. Lubricant flow rate
5. Shaft speed
6. Friction torque
7. Shaft position in the bearing with respect to time
8. Eccentricity

APPENDIX A - (Cont'd)

The Contractor shall study analytically approximately eleven (11) different bearing configurations and shall then select approximately six (6) of these for experimental testing.

The Contractor shall determine experimentally the antiwhirl characteristics for each bearing configuration selected for testing over ranges of the following variables as specified.

1. Shaft speed: 0 - 35,000 rpm
2. Bearing load: Unidirectional - 0 - 200 lbs.
   Rotating - 0 - Maximum level as governed by maximum attainable speed or by bearing load capacity
3. Inlet lubricant pressure: 3 - 30 psi
4. Inlet lubricant temperature: 100 - 160^{o}F
5. Clearance: 0.0016" - 0.004"
6. Bearing length to diameter ratio (L/D): 0.75 - 1.5

The test lubricant for this test phase shall be distilled water.

APPENDIX B

TITLE OF CONTRACT:	INVESTIGATION OF STABILITY OF HYDRODYNAMIC JOURNAL BEARINGS
NASA CONTRACT NUMBER:	NAS 3-6479
DATE CONTRACT WAS ISSUED:	APRIL 29, 1965
DATE OF CONTRACT COMPLETION:	END OF 1966
NASA CONTRACTING OFFICE:	LEWIS RESEARCH CENTER
NASA PROGRAM MANAGER:	JOSEPH P. JOYCE

EXCERPT OF WORK STATEMENT

Specific Requirements

The Contractor shall extend the investigation of the stability of the two hydrodynamically lubricated bearings that have demonstrated under Contract NAS 3-2111 an ability to suppress half-frequency shaft whirl while operating at rotational speeds up to 21,000 rpm and with near zero radial loads. This study shall extend to rotational speeds of 30,000 rpm and shall include effects of bearing misalignment. The two bearing designs to be studied are the three lobe and the four pad pivot pad configurations. The test bearing dimensions are:

Inside Diameter	1.25 inches (nominal)
$\left(\frac{\text{Length}}{\text{Diameter}}\right)$	1.0
Diametrical Clearance	0.005 inch

The bearing test facility, its auxiliary equipment and instrumentation used on Contract NAS 3-2111 shall be modified and improved to demonstrate the capability of testing through the range of variables stated below:

	Variables	Range
1.	Shaft Rotational Speed	3,600 to 30,000 rpm
2.	Bearing Angular Misalignment	0 to 400 sec.

APPENDIX B - (Cont'd)

3.	Bearing Linear Misalignment	0 to 0.004 inches
4.	Combined Bearing Angular and Linear Misalignment	Same as Items 2 & 3
5.	Lubricant Supply Pressure	15 to 150 psia
6.	Bearing Mount Rigidity	Rigid & 50,000 pounds/inch
7.	Bearing Unidirectional Radial Load (Each Test Bearing)	0 - 60 pounds
8.	Bearing Synchronous Rotating Load (Each Test Bearing)	0 - 40 pounds at 30,000 rpm
9.	Bearing Reaction Force Vector	0 to 360^{o}

APPENDIX C

TITLE OF CONTRACT:	RESEARCH ON MATERIALS SUITABLE FOR POTASSIUM-LUBRICATED JOURNAL BEARING AND SHAFT COMBINATIONS
NASA CONTRACT NUMBER:	NAS 3-2534
DATE CONTRACT WAS ISSUED:	APRIL 22, 1963
DATE OF CONTRACT COMPLETION:	END OF 1966
NASA CONTRACTING OFFICE:	LEWIS RESEARCH CENTER
NASA PROGRAM MANAGER:	ROBERT L. DAVIES

EXCERPT OF WORK STATEMENT

The Contractor shall conduct an evaluation of materials suitable for potassium-lubricated journal bearings and shaft combinations destined for use in space system turbogenerators.

The Contractor shall provide capsule corrosion test facilities capable of subjecting single and/or multiple bearing material specimens to potassium at $400^{\circ}F$ to $1600^{\circ}F$. Test capsules shall be contained in a vacuum better than 1×10^{-7}torr pressure after initial startup and inception of stable operation of the tests. During the startup time, the vacuum shall be 1×10^{-6}torr or better. The capsule facility shall alter, control and give accurate readout of potassium temperature. Similar facilities shall be provided to determine dimensional stability characteristics of the bearing materials.

The Contractor shall provide a physical and mechanical properties test facilities capable of maintaining a vacuum in the range of 10^{-6}torr pressure, except for thermal expansion measurement apparatus which shall maintain an inert gas atmosphere. These facilities shall be fitted to determine material physical mechanical properties.

The Contractor shall provide a friction and wear test facilities capable of maintaining a vacuum in the range of 10^{-9}torr pressure or better. This facility shall be designed to alter, control, and read out accurately the following variables:

1. Test sample sliding speed from 0 to 1000 ft/min.

2. Test sample normal load to produce yield stress in contact materials.

APPENDIX C - (Cont'd)

3. Induced torque.

4. Test environment

   pressure, torr

   potassium level.

5. Test temperature 400^{o} to 1200^{o}F.

The Contractor shall provide facilities to evaluate the wetting behavior of the bearing materials with liquid potassium by the sessile drop method. The facilities shall be capable of being evacuated to pressures in the 10^{-9}torr range.

It is anticipated that about 14 materials and about 7 material combinations shall be chosen for test.

Corrosion capsule tests shall be conducted as outlined below:

1. At temperatures of 800^{o}, 1200^{o} and 1600^{o}F.
2. Continuous test duration of 1,000 hours at each temperature listed in 1. above.
3. Capsules shall be fabricated from Cb-1Zr Alloy.
4. Test specimen combinations shall be tested with equal surface areas exposed to potassium.

Dimensional stability - the amount of permanent dimensional change at constant temperatures shall be conducted at 800^{o}, 1200^{o}, and 1600^{o}F for 1000 hours.

The following physical and mechanical properties shall be determined:

1. Coefficient of thermal expansion at 400^{o}, 800^{o}, 1200^{o} and 1600^{o}F.
2. Compressive yield stress at room temperature, 400^{o}, 800^{o}, 1200^{o} and 1600^{o}F.
3. Ultimate compressive stress at room temperature, 400^{o}, 800^{o}, 1200^{o} and 1600^{o}F.
4. Modulus of elasticity at room temperature, 400^{o}, 800^{o}, 1200^{o} and 1600^{o}F.

APPENDIX C - (Cont'd)

5. Hardness at 400°, 800°, 1200° and 1600°F.

Coefficients of friction and wear rates shall be determined under the following conditions:

1. At temperatures of 400°, 800° and 1200°F.

2. At speeds to 1000 SFM.

3. At loads to 90 percent of the 0.2% yield strength or ultimate strength of the test materials.

4. At pressures in the 10^{-9} torr range and in liquid potassium.

APPENDIX D

TITLE OF CONTRACT:	DESIGN CRITERIA FOR ROTARY SEALS FOR A SPACE ENVIRONMENT
AIR FORCE CONTRACT NUMBER:	AF 33(657)-8469
DATE CONTRACT WAS ISSUED:	MARCH 1, 1962
DATE OF FINAL REPORT:	SEPTEMBER, 1965
AIR FORCE CONTRACTING OFFICE:	AERONAUTICAL SYSTEMS DIVISION
AIR FORCE PROGRAM MANAGER:	ROBERT J. SMITH

EXCERPT OF WORK STATEMENT

Objectives

The objective of this program is to acquire the techniques for completely sealing high speed rotating shafts under the operating conditions of high temperature liquid metals and vapors, the near-vacuum environments of space, and long life.

Requirements

Concepts of seals to meet the objective and the following conditions shall be designed, fabricated and experimented upon.

1. Fluid - The fluid to be sealed shall be potassium.
2. Temperature - The seals shall be operative at fluid temperatures from the melting point of the fluid selected to $800^{o}F$.
3. Pressure - The pressure on the fluid side of the seal shall be 15 psi and the external pressure shall be 10^{-6} mm Hg.
4. Speed - The speed of the rotating shaft shall be 24,000 rpm.
5. Life - The seal, or seal combinations, shall be designed for 10,000 hours of maintenance-free life.
6. Seal Lubricant - The working fluid, potassium, shall be used as the seal lubricant.

APPENDIX D - (Cont'd)

7. Leakage - The seal, or seal combinations, shall be capable of maintaining absolute zero leakage under all conditions of operation.

8. Size - The seals shall be designed for a 1.0-inch diameter shaft.

9. Zero G - The seals shall be capable of operation in a zero "g" environment.

Evaluation

The seal evaluation shall consist of the following:

1. Preliminary experiments

2. 100-hour operational screening test

3. Thermal-cycling test

4. 1000-hour life test

Test Conditions

The test conditions are potassium at $1400^{\circ}F$ and 15 psi on one side of the seal, a pressure of 10^{-6} mm Hg on the other side, and a shaft speed of 24,000 rpm.

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APPENDIX E

TITLE:	RESEARCH ON LIQUID METAL BEARINGS AND DYNAMIC SEAL SYSTEMS
FUNDING:	GENERAL ELECTRIC COMPANY INDEPENDENT RESEARCH FUNDING (CIRP) DA # L06-09F
PROGRAM INITIATED:	1965
PROGRAM TO BE COMPLETED:	MIDDLE OF 1967

Objectives

1. Establish feasibility of potassium lubricated bearings and seals for application in electric generator.
2. Employ optimum bearing materials as identified by Bearing Materials Program (NAS 3-2534).
3. Determine bearing stiffness.
4. Establish load carrying capability, dynamic as well as static.
5. Determine power requirements.
6. Determine rotor/bearing response.
7. Establish mechanical integrity
   a) During thermal cycling
   b) During long-time operation.
8. Establish low power DZL seal.

Specifications

Lubricant	Liquid Potassium
Temperature Level in Bearings and Seals	$600^{\circ}F$
Bearing Diameter	2.625 inches

APPENDIX E - (Cont'd)

Bearing L/D Ratio	1.0
Speed	18,000 rpm
Bearing Type	Four-Pad Pivoted Pad Bearing
Fixed Pivoting Points Preload Coefficient	0.5
Rotor Weight	70 pounds
Rotor Material	H-11 Tool Steel
Residual Unbalance	0.15 gr-in
Provisions for Rotational Loads up to 1000 pounds	By Unbalance
Electromagnetic Side Load	Up to 200 pounds
Containment Material	316 Stainless Steel
Bearing Material	Carboloy 907

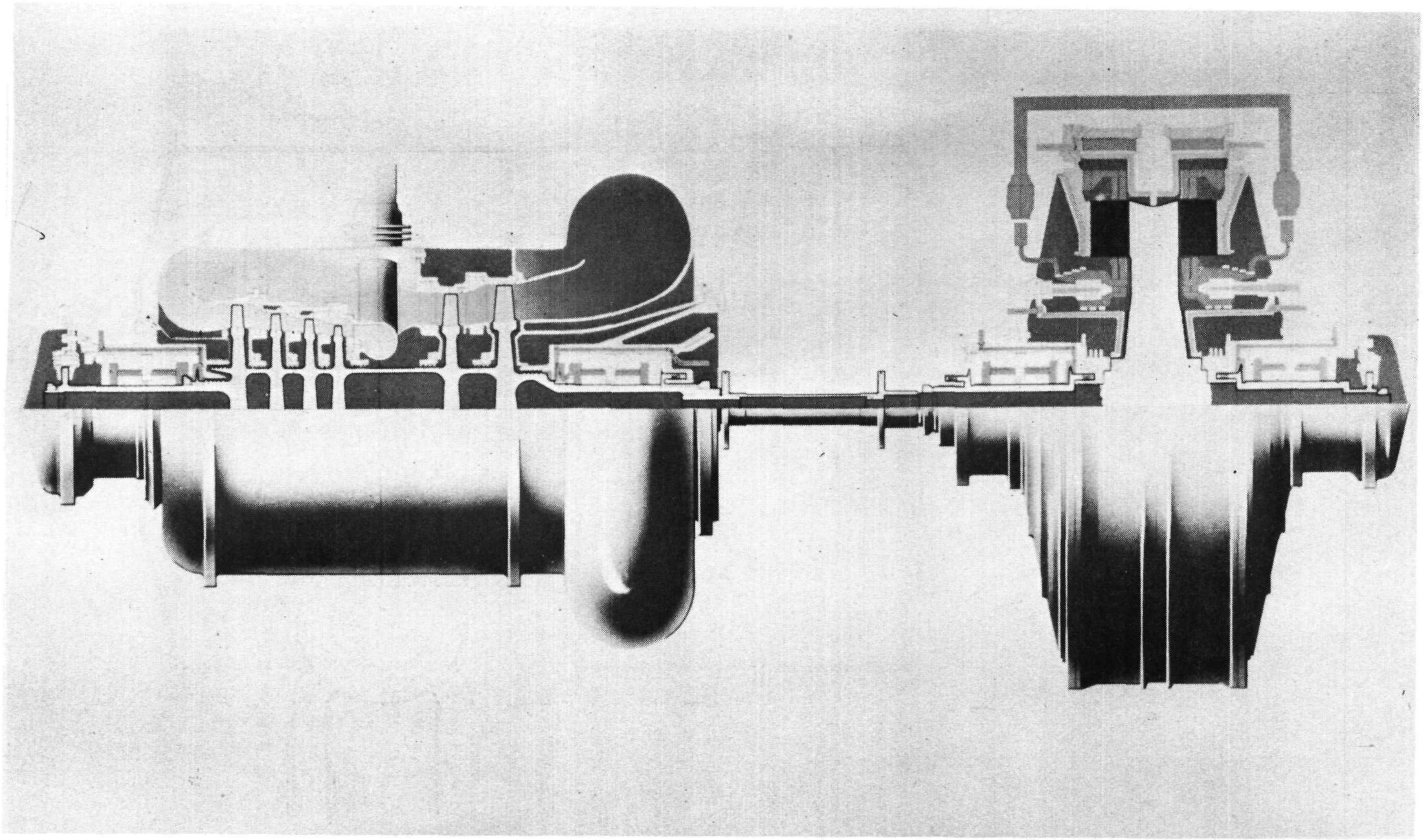

Fig. 1. Bearings and Seals in a Potassium Turbogenerator for Space.

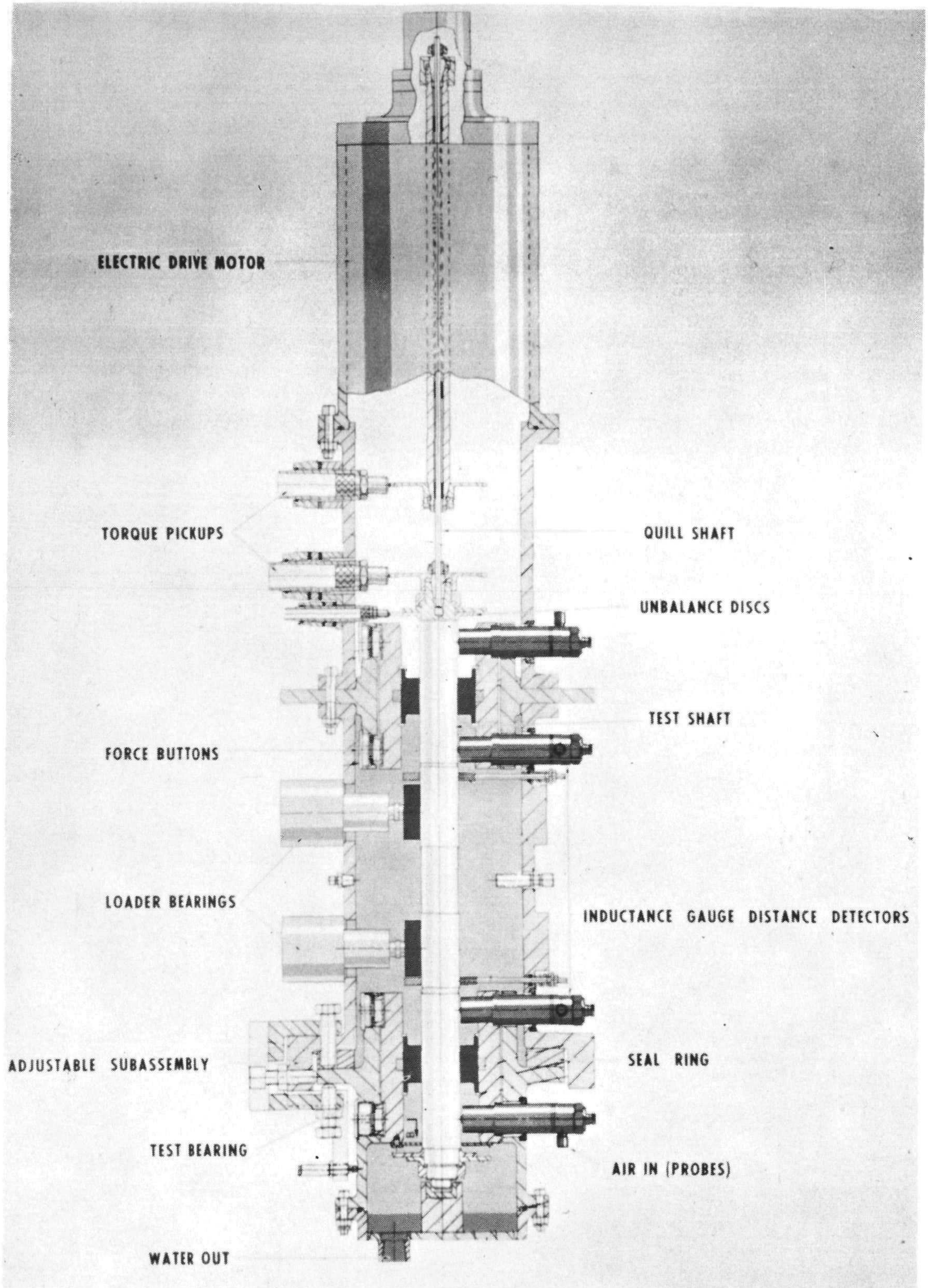

Fig. 2. Bearing Stability Test Rig.

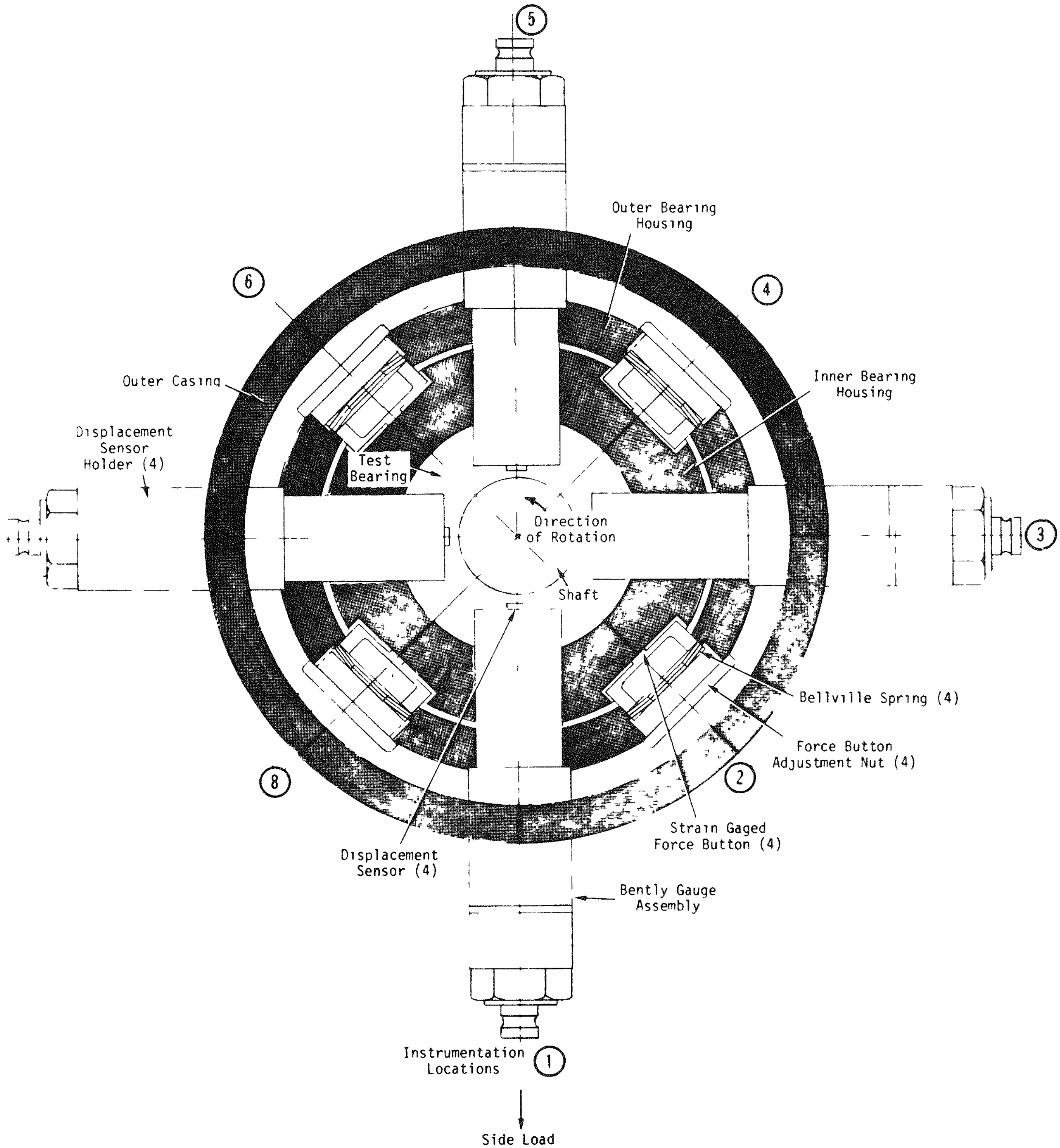

Fig. 3. Force Button and Proximity Gage Installation.

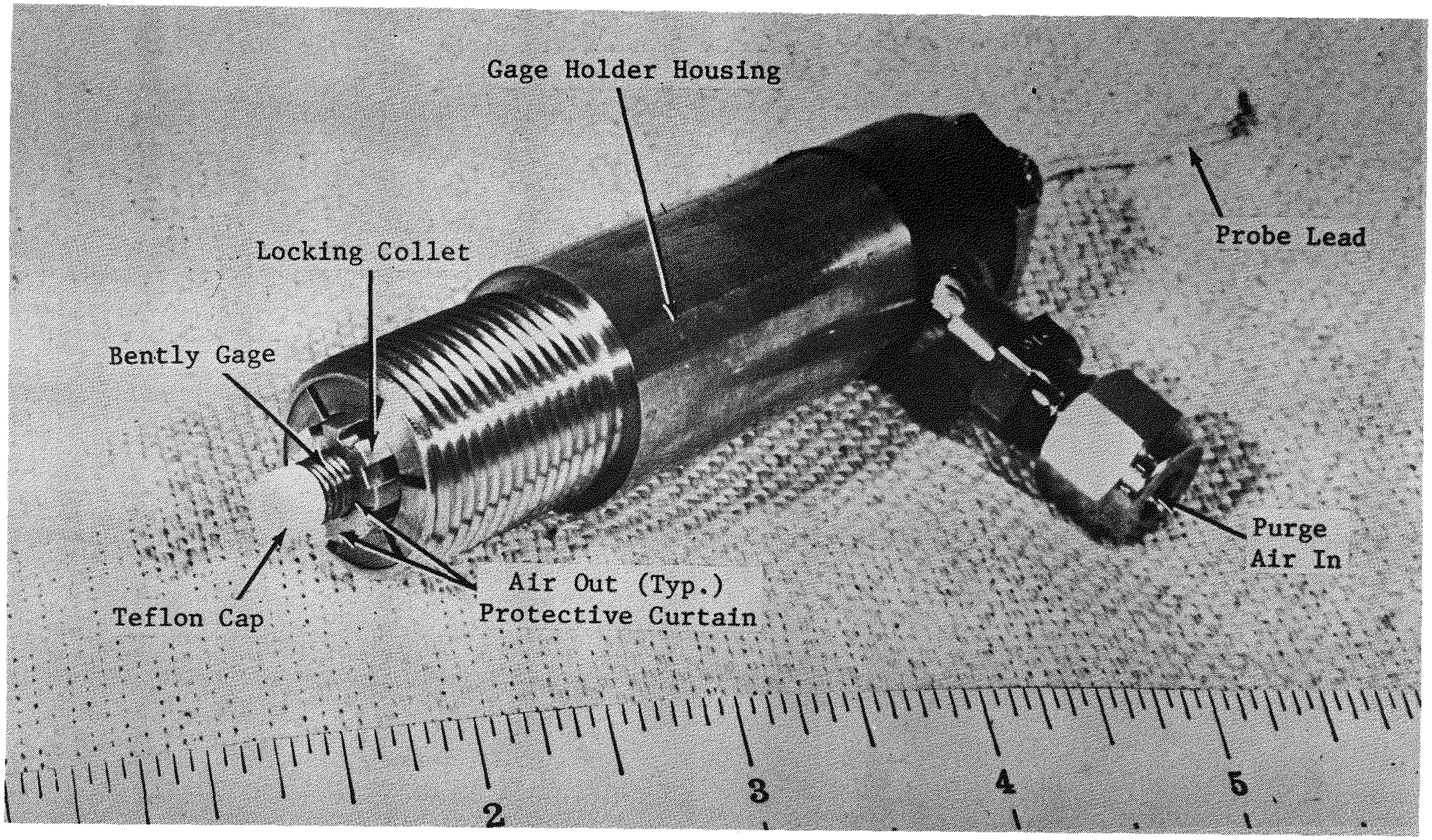

Fig. 4. Proximity Gage Holder.

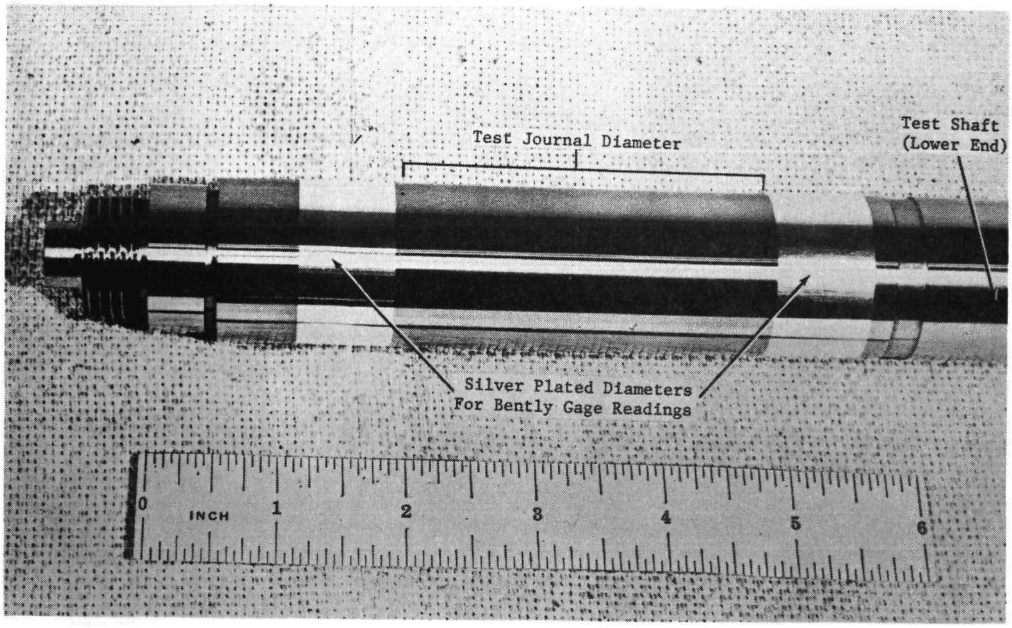

Fig. 5. Test Shaft.

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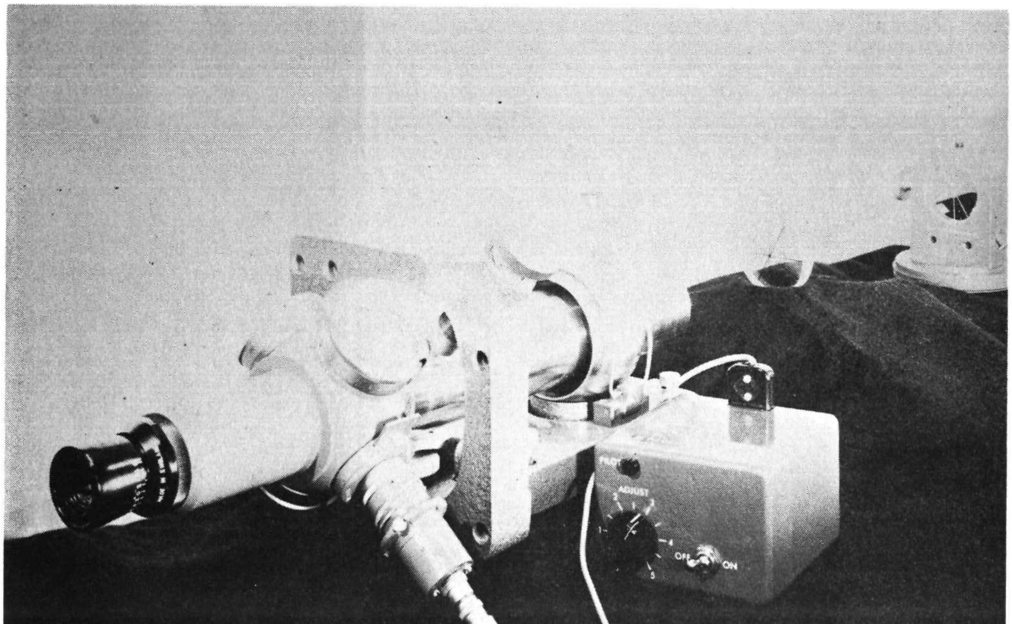

Fig. 6. Optical Alignment Measurement.

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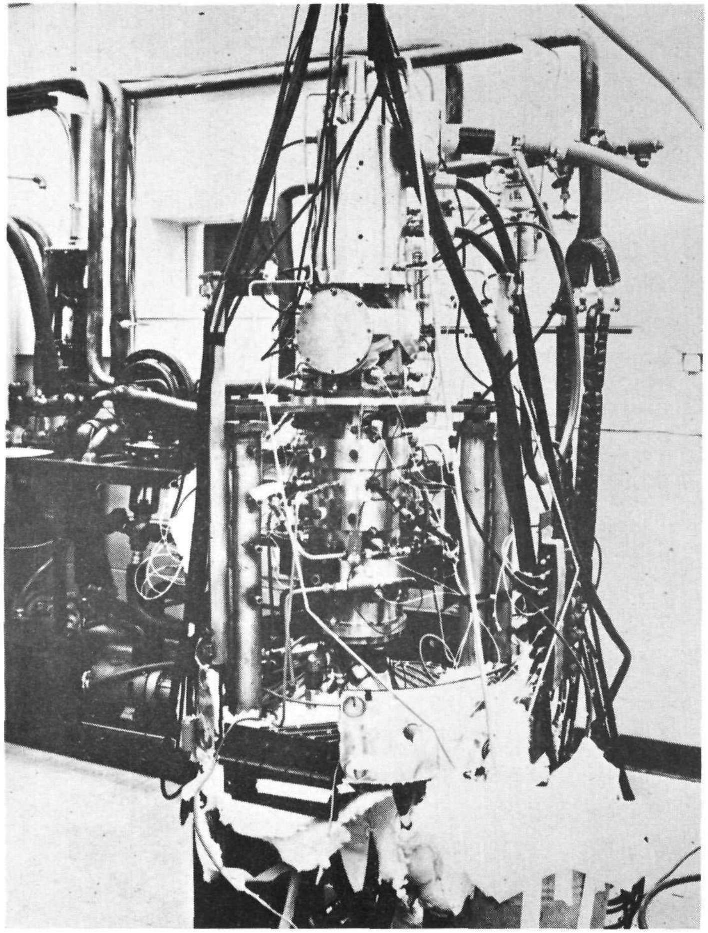

Fig. 7. Test Rig in Room Temperature Test.

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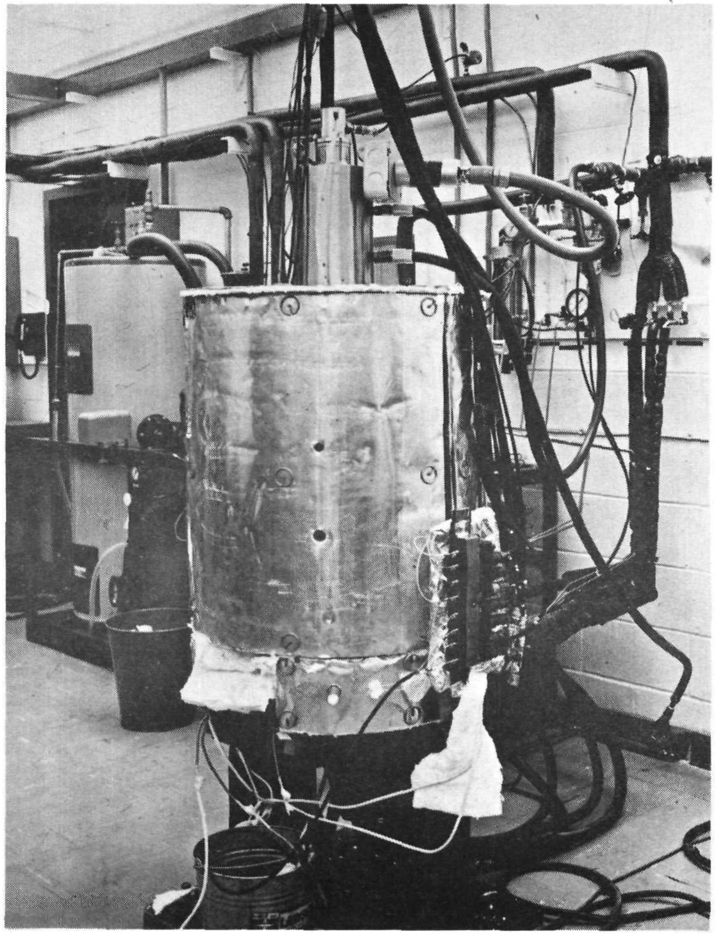

Fig. 8. Test Rig Temperature Controlled 120°F.

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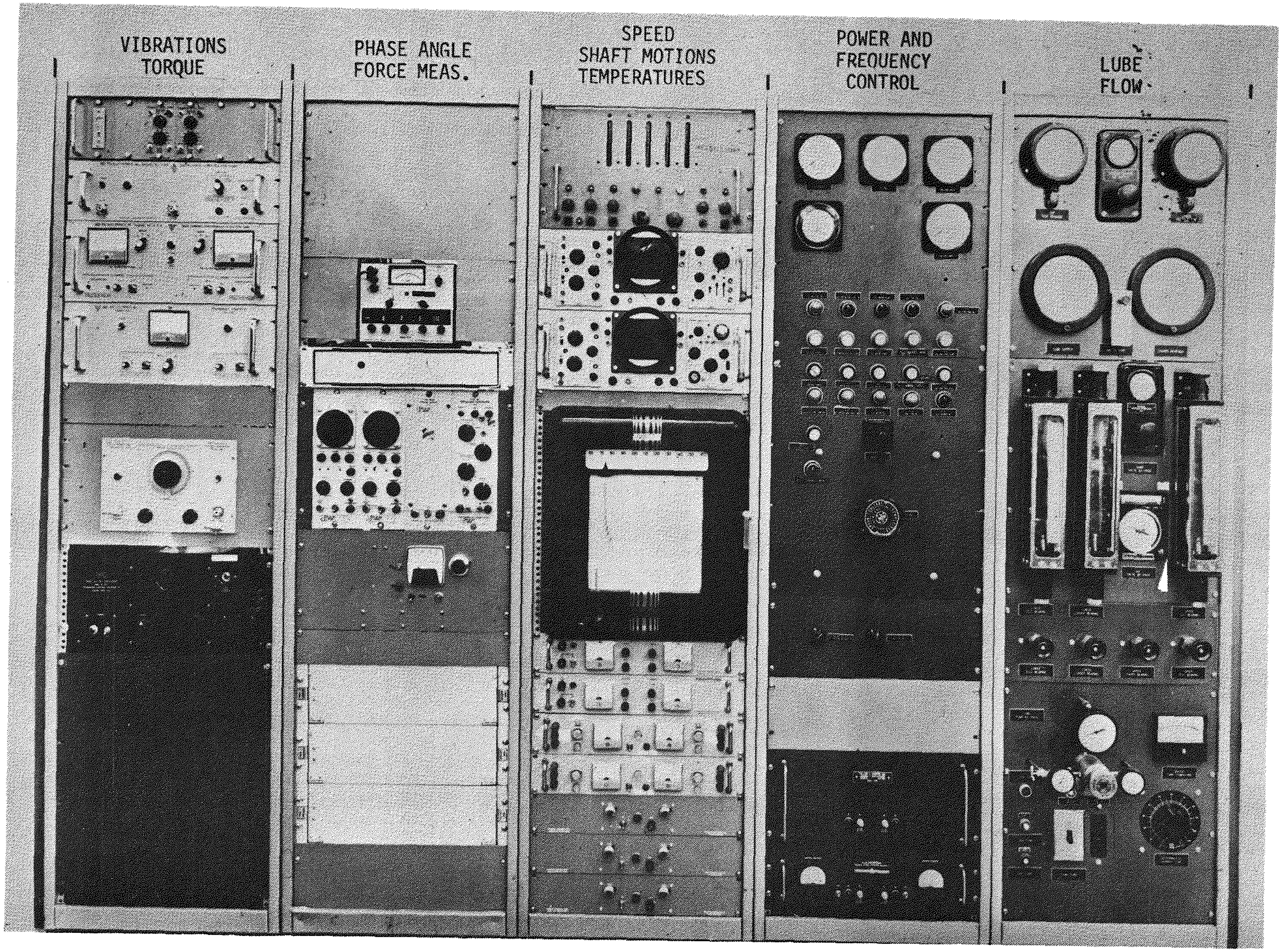
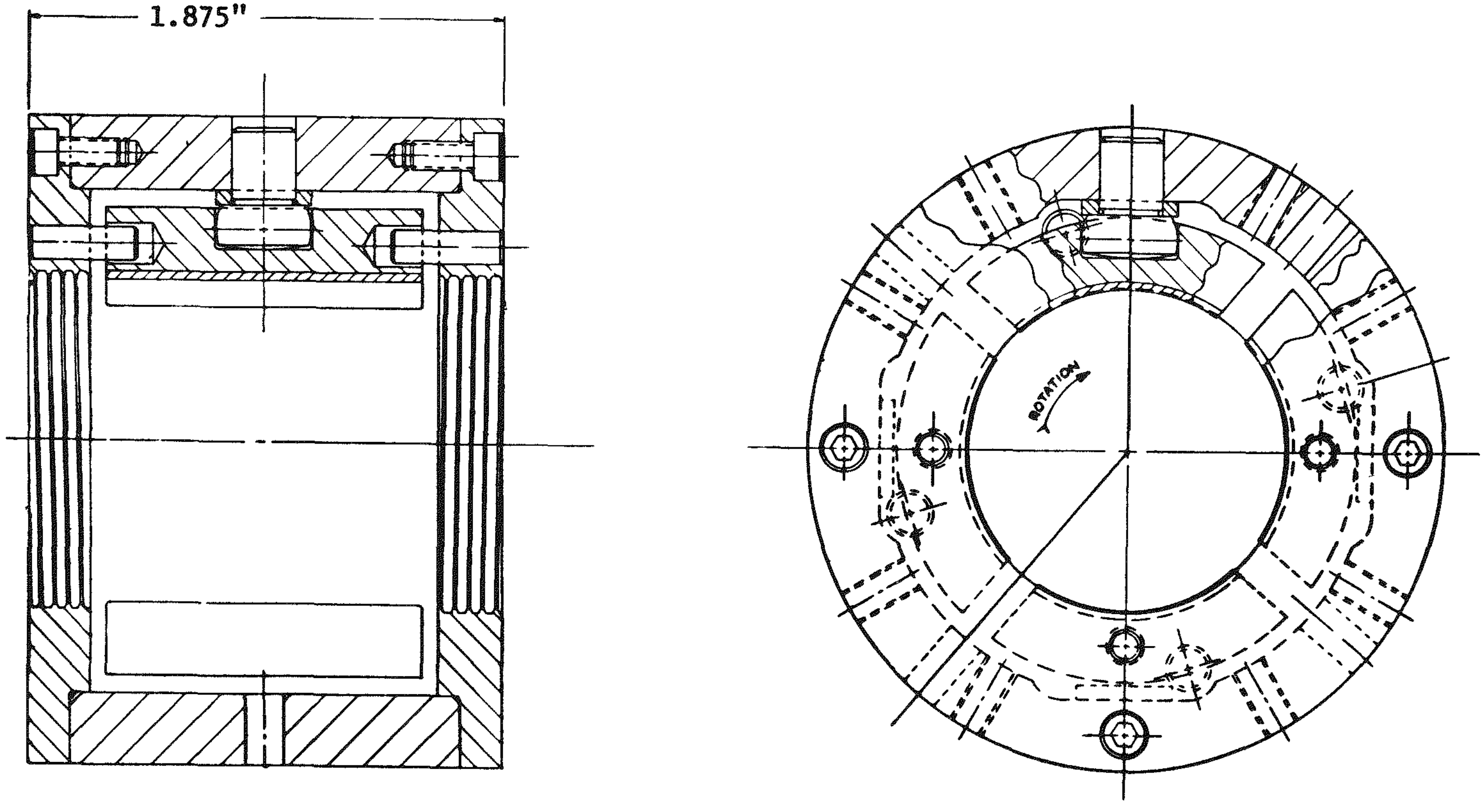


Fig. 9. Instrumentation Panel.



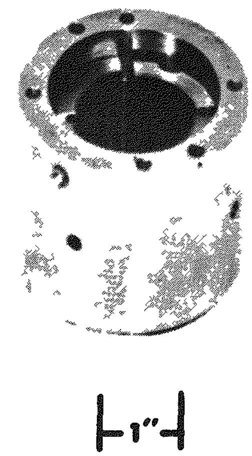


Fig. 10. Self-Aligning Pivoted Pad Bearing.

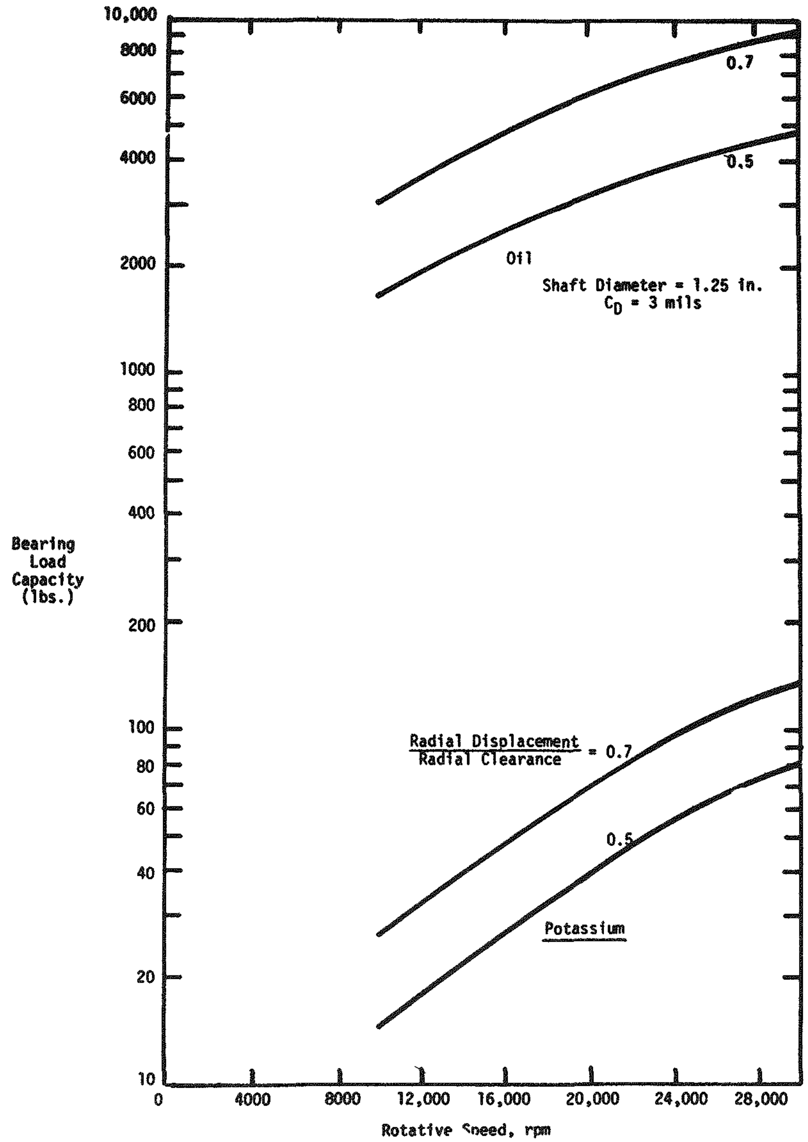


Fig. 11. Load Carrying Capacity (Pivoted Pad Bearing).

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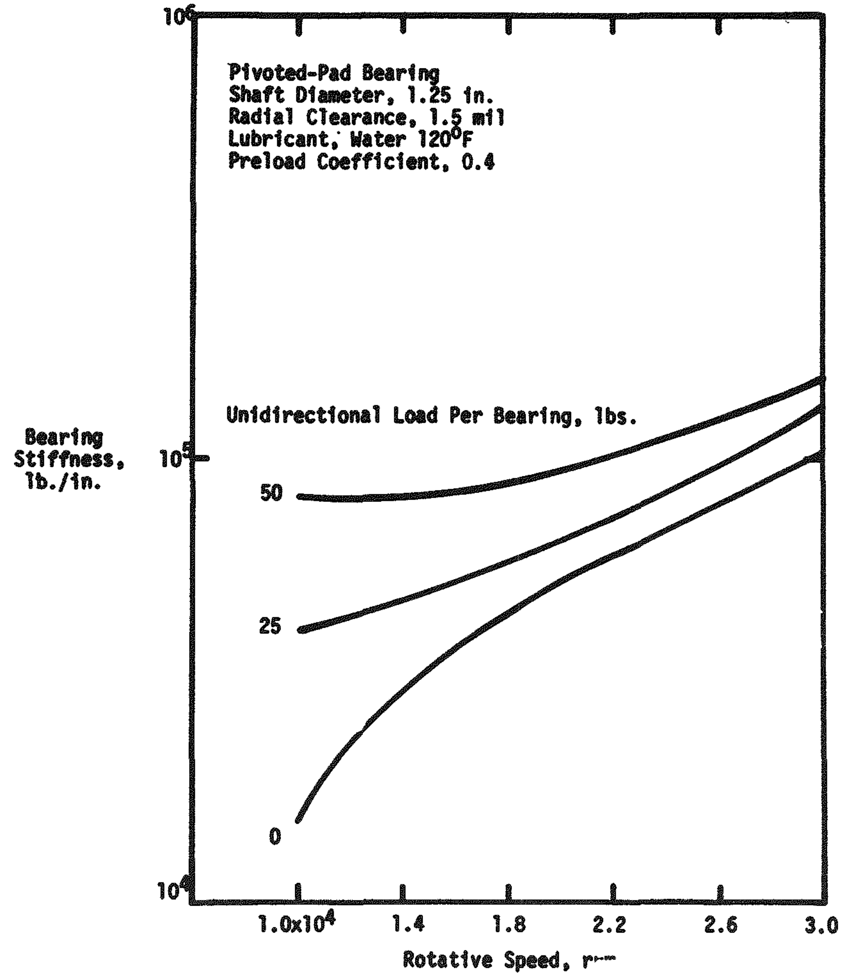

Fig. 12. Bearing Stiffness Vs. Speed and Side Load.

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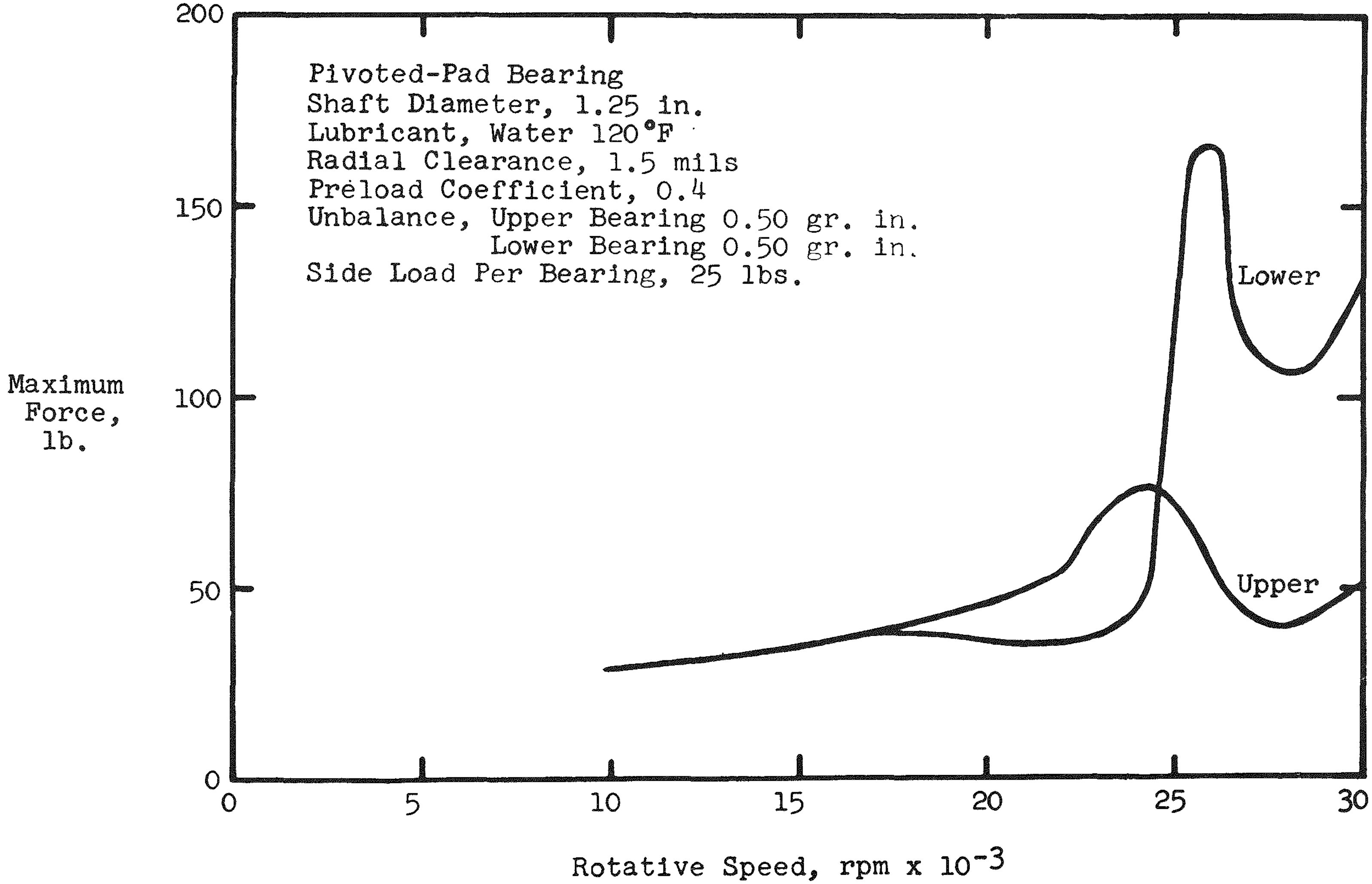

Fig. 13. Force Vs. Speed.

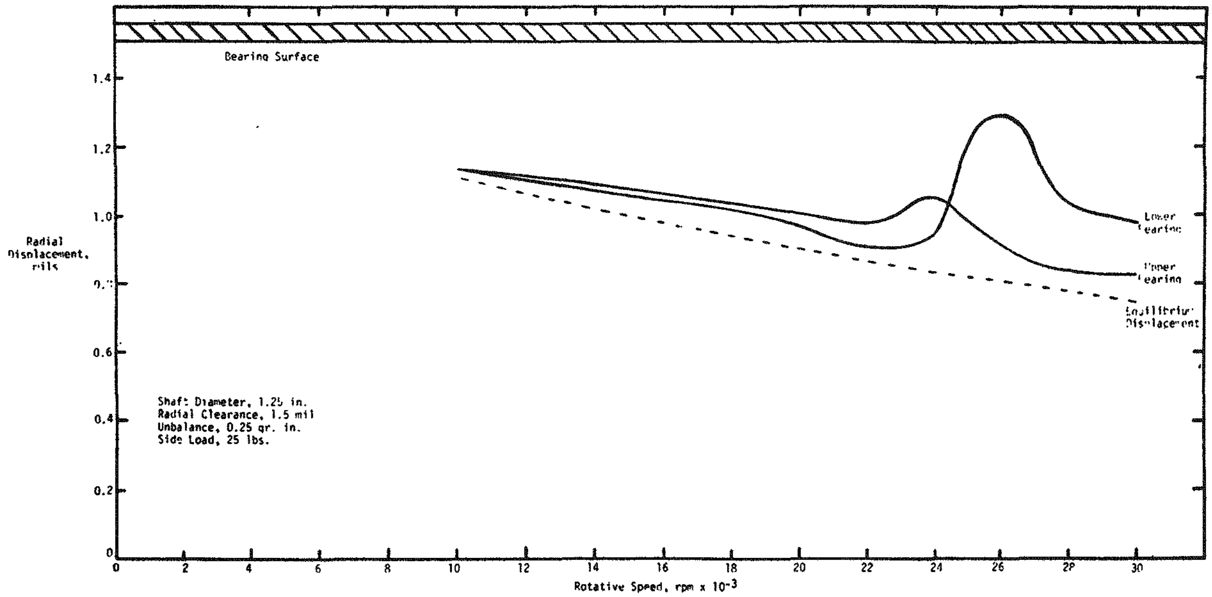

Fig. 14. Displacement Vs. Speed, 0.25 in-gr Unbalance.

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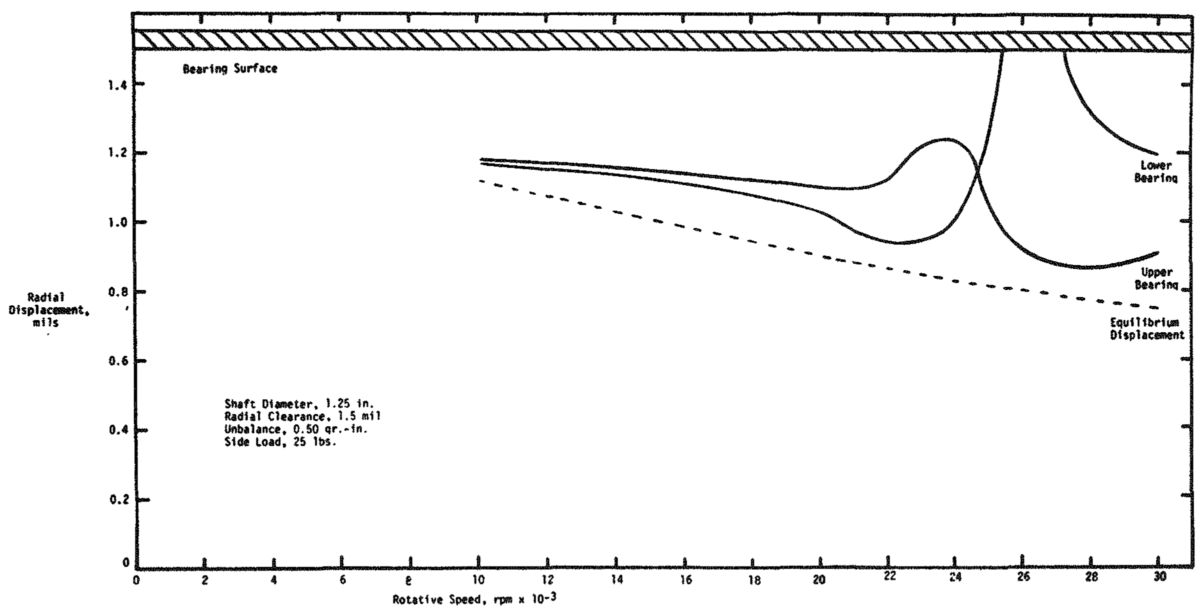

Fig. 15. Displacement Vs. Speed, 0.50 in-gr Unbalance.

MTI-2143

- **OBJECTIVE**

  ESTABLISH "ZERO LEAKAGE" SEAL FOR SPACE GENERATOR

- **SPECIFICATIONS**

  SEAL FLUID POTASSIUM AT 600°F

  SHAFT DIAMETER 1"

  SHAFT SPEED 24,000 RPM

  EXTERNAL: LOW VACUUM (10^{-6} TORR)

  LEAKAGE: 1 TO 10 LBS. IN 10,000 HOURS

Fig. 16. Dynamic Seal Development.

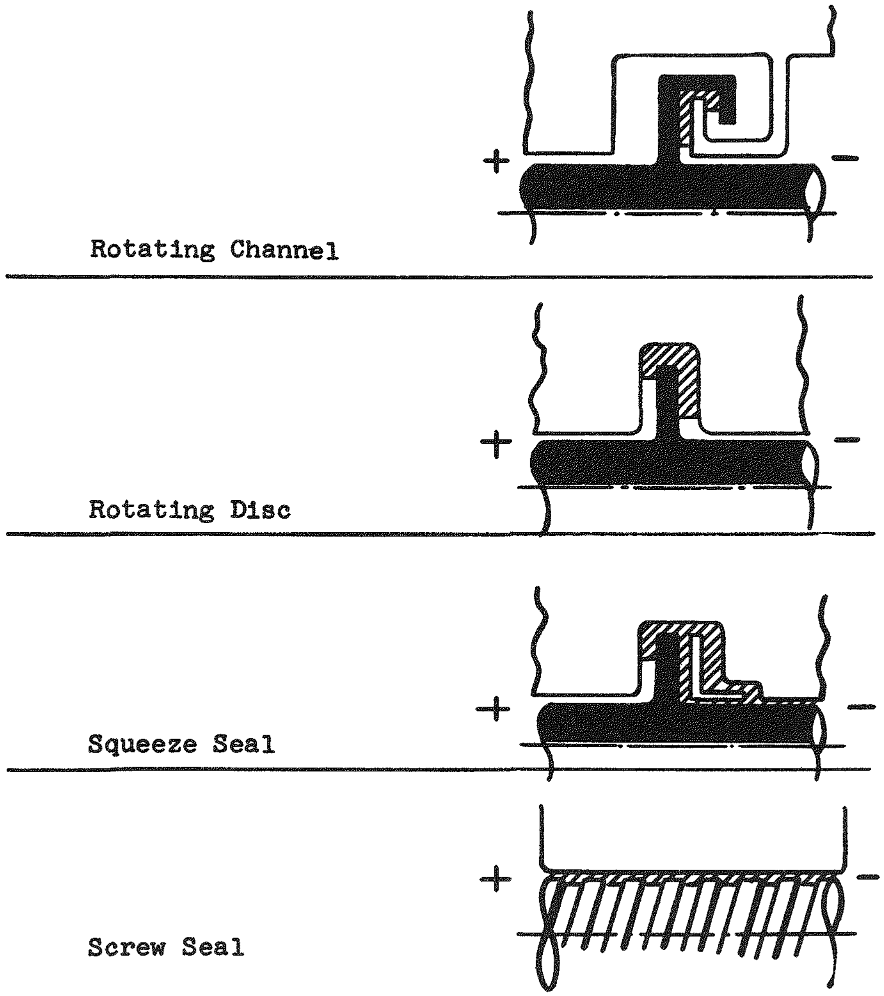

Fig. 17. The Seal Concept.

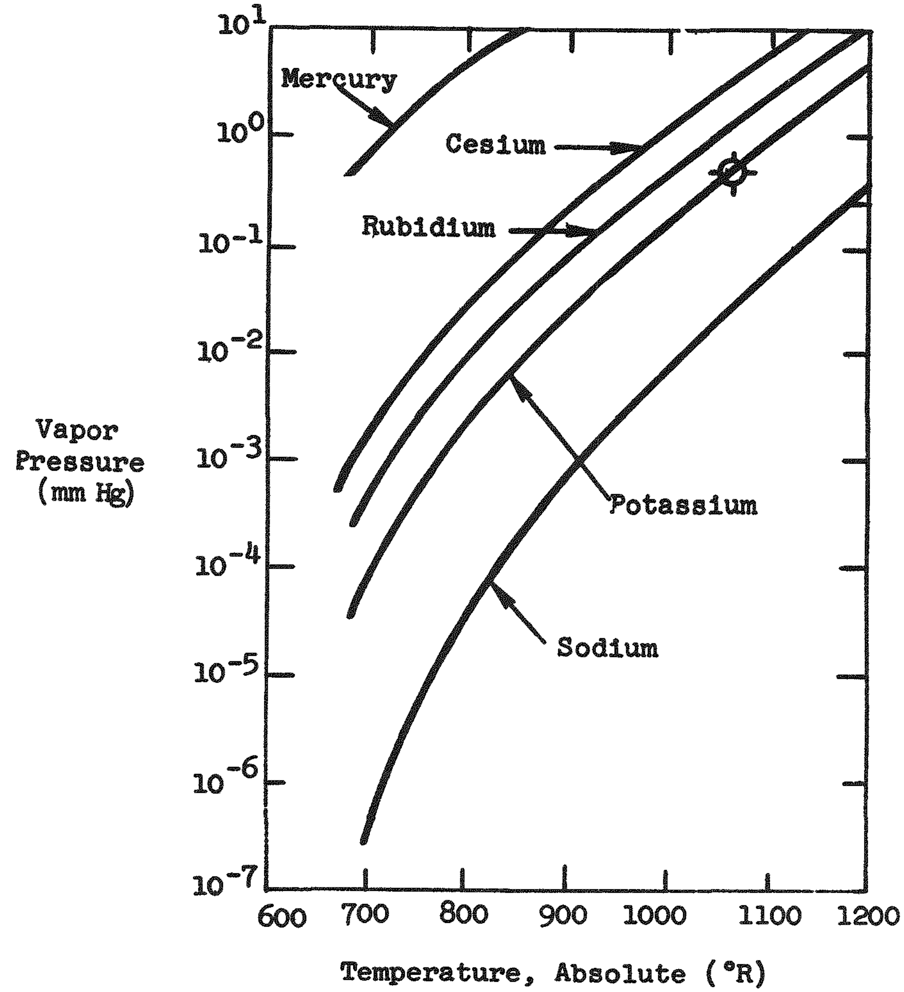

Fig. 18. Vapor Pressure of Liquid Metals.

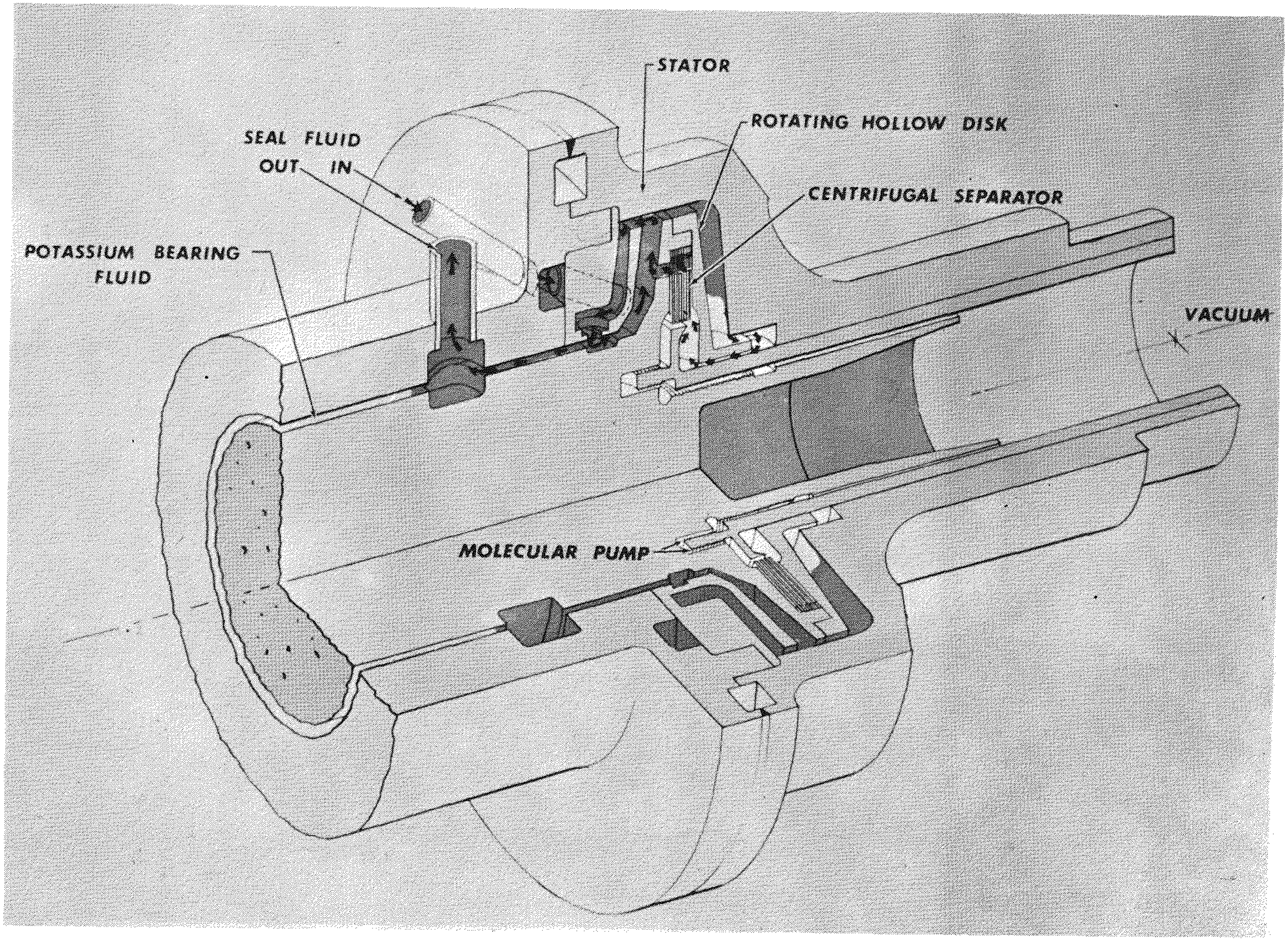

Fig. 19. Dynamic Zero Leakage Seal.

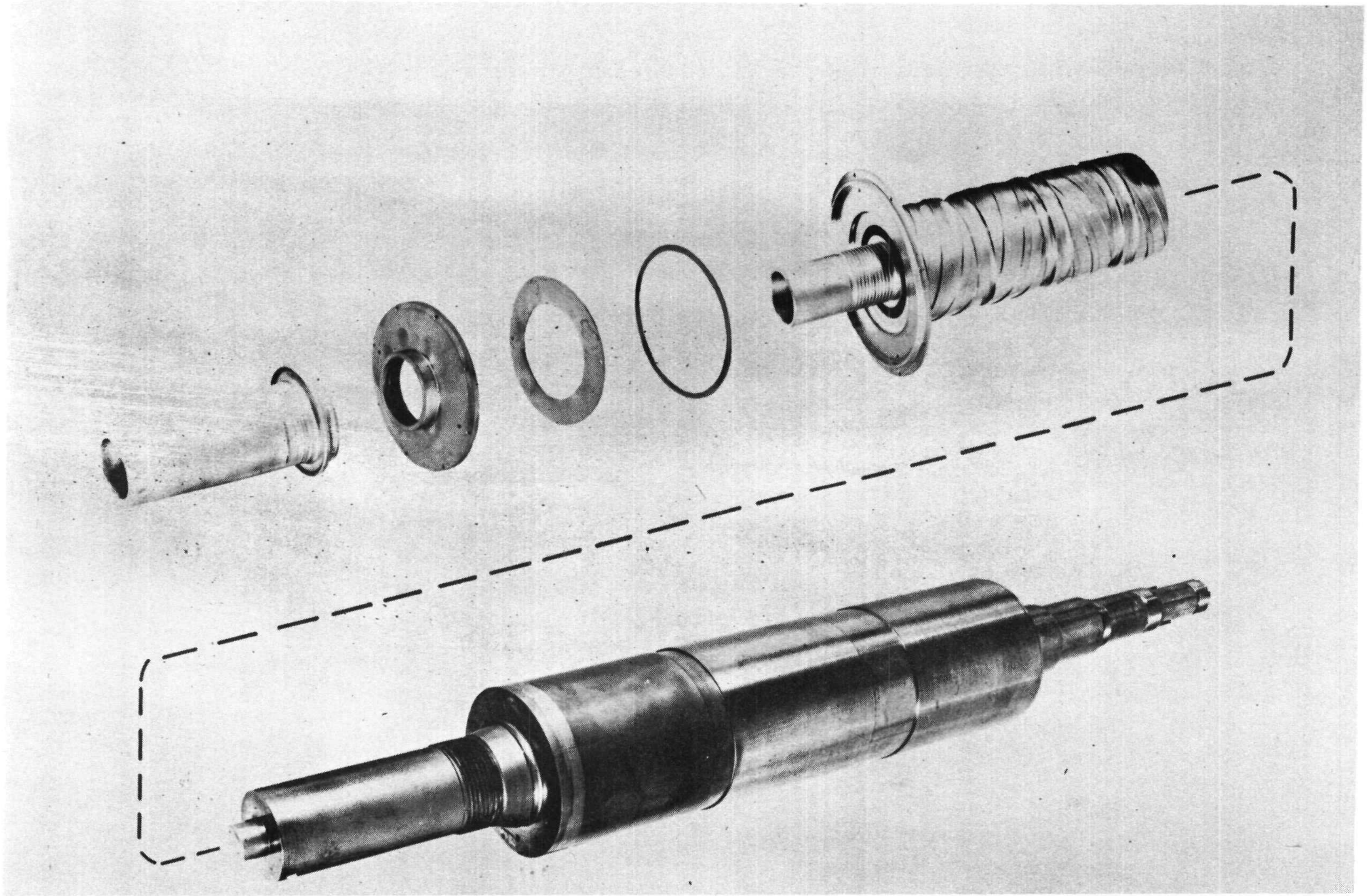

Fig. 20. Seal Hardware and Test Spindle.

6.2.37

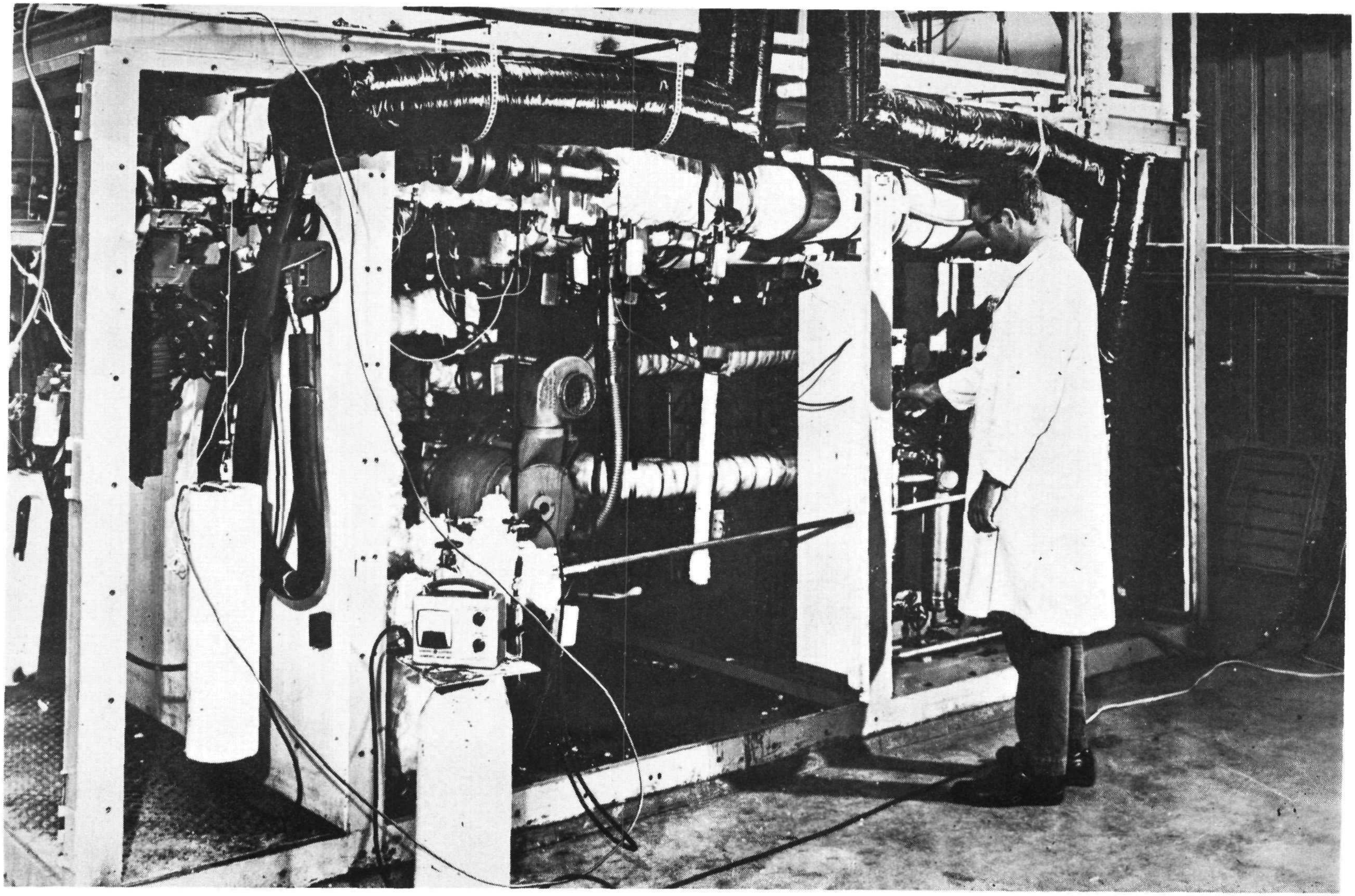

Fig. 21. Liquid Metal Bearing and Seal Facility.

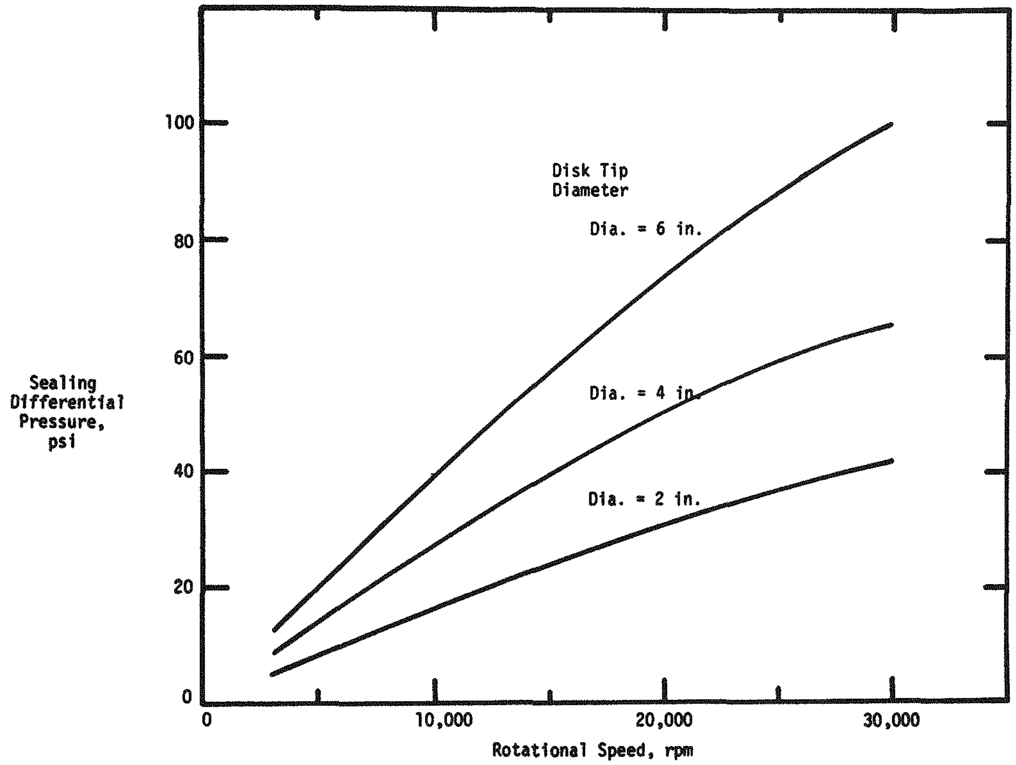

Fig. 22. Sealing Capacity of DZL Seal.

MTI-2150

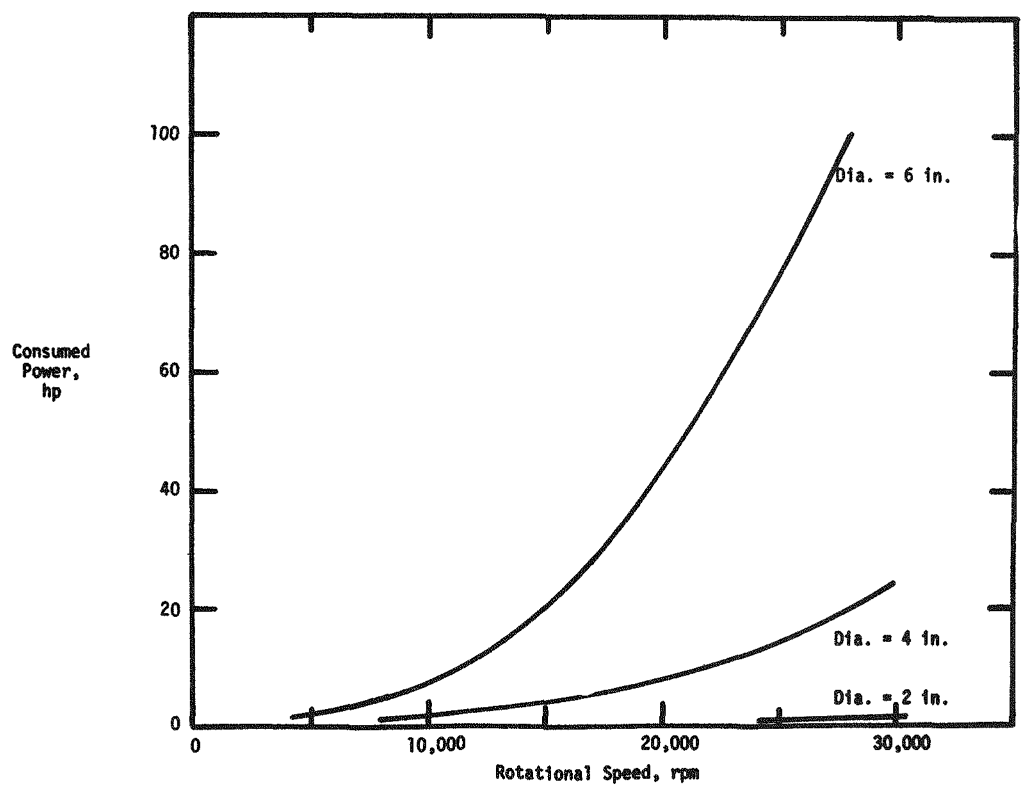

Fig. 23. Power Consumption of DZL Seal.

MTI-2151

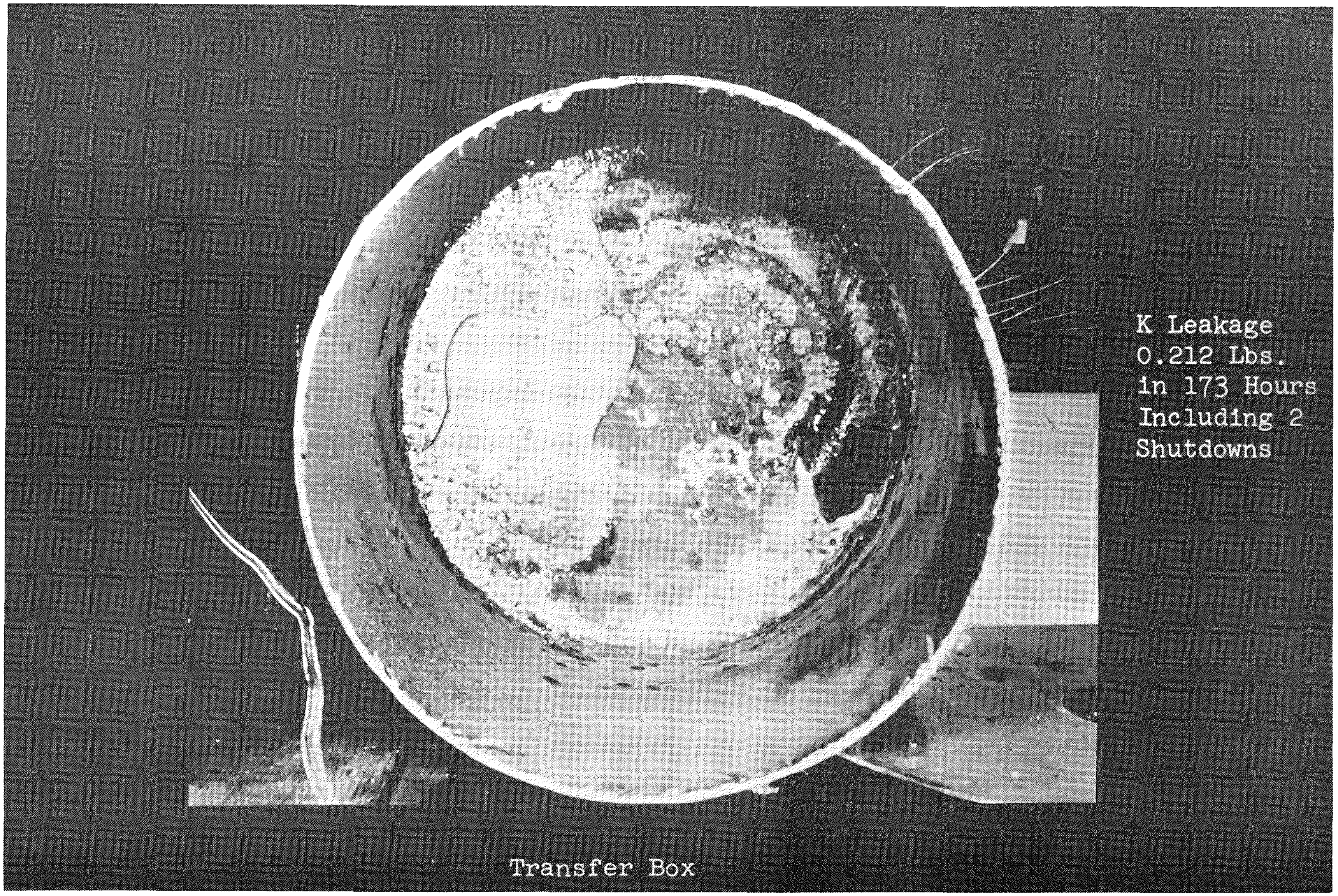

Fig. 24. Seal Leakage in 173 Hours of Testing.

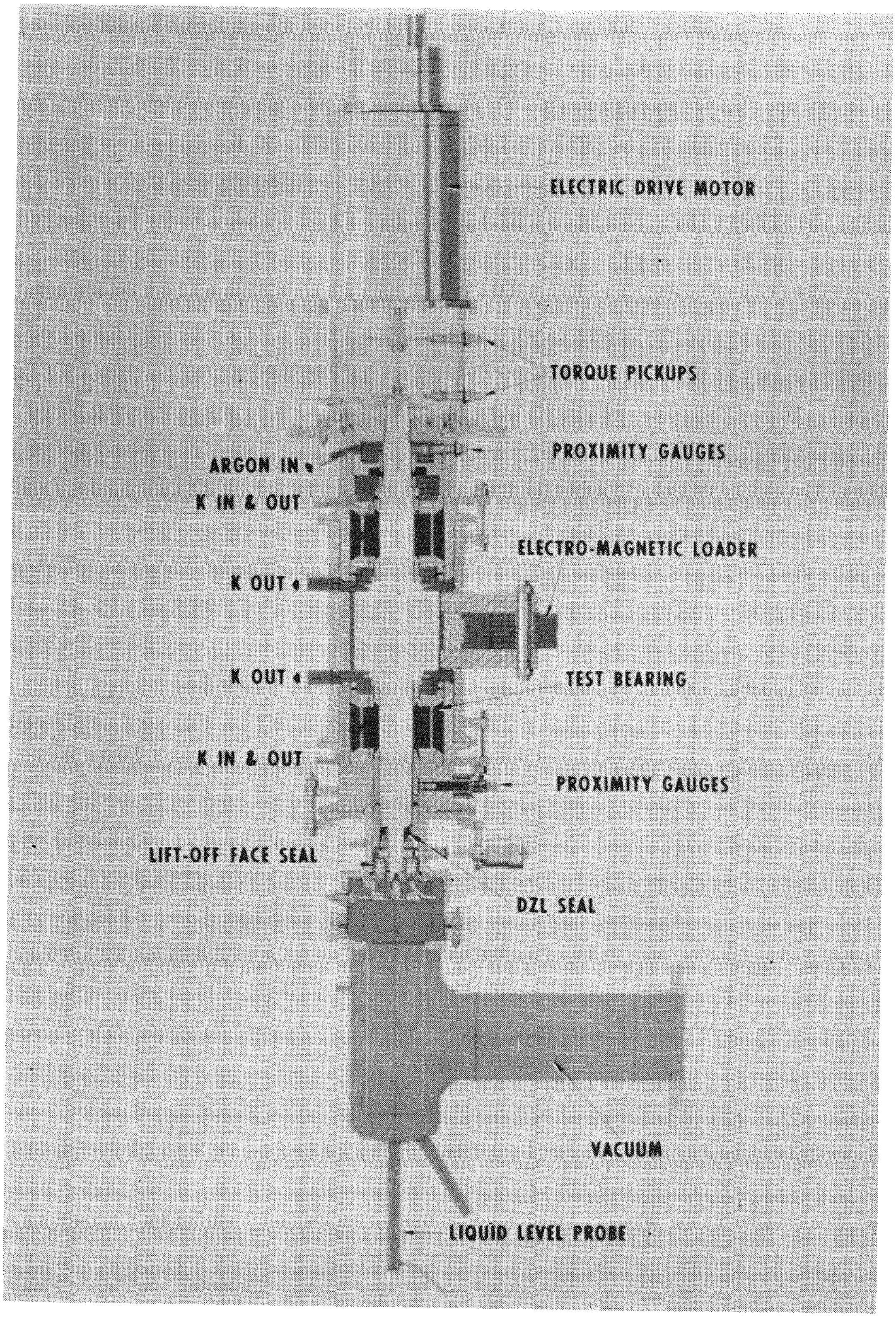

Fig. 25. Liquid Metal Bearing Test Rig.

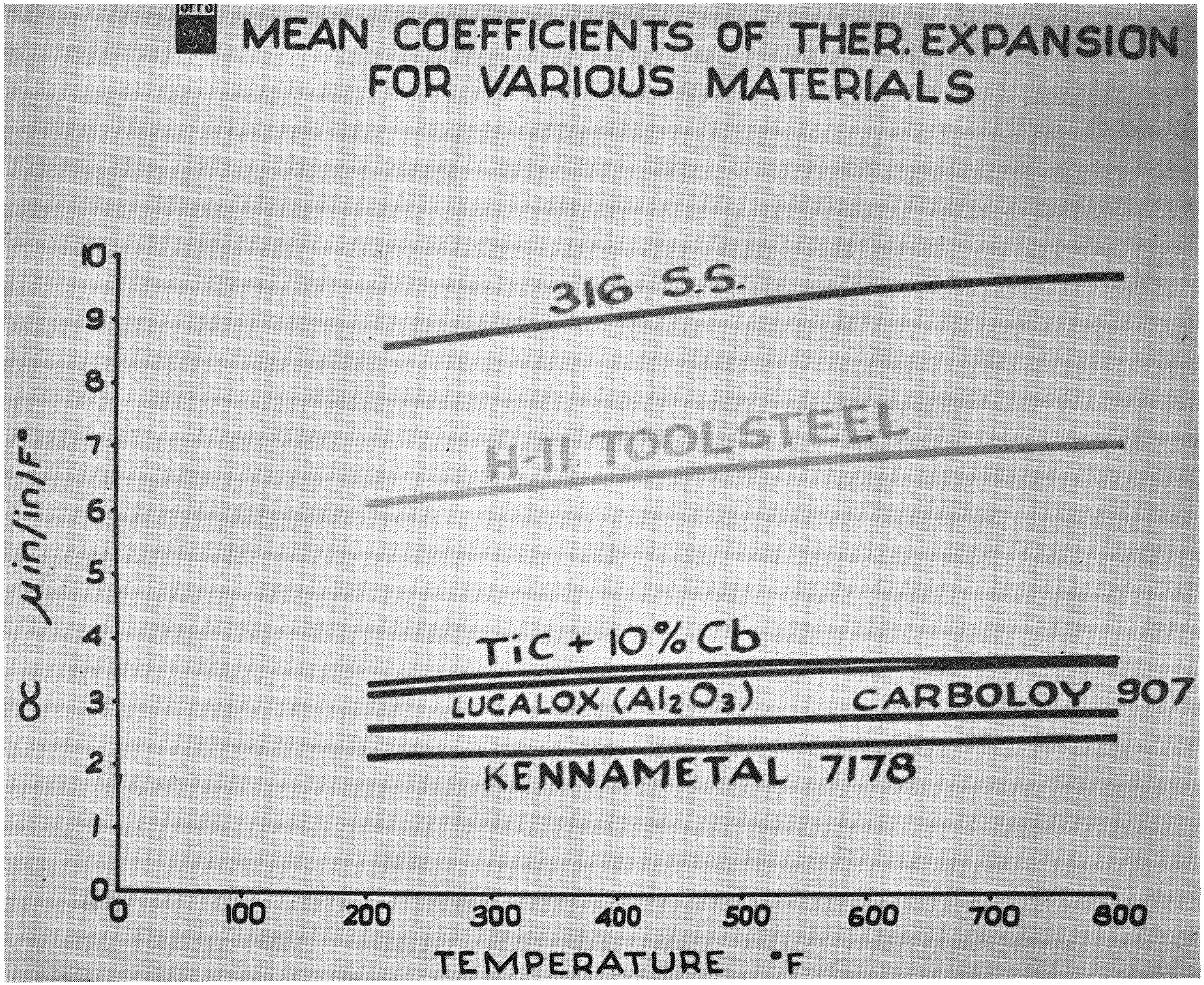

Fig. 26. Thermal Expansion Coefficients.

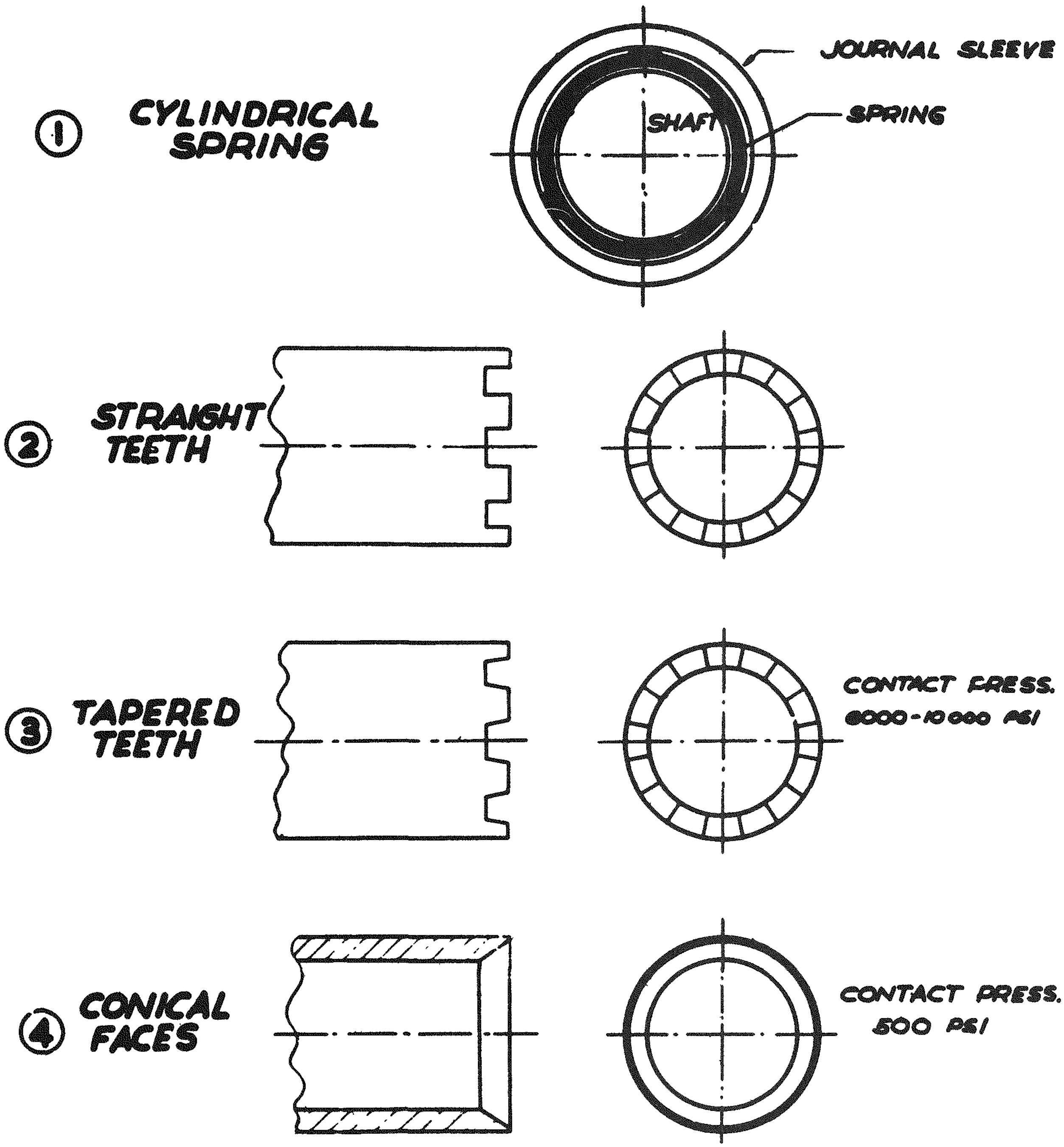

Fig. 27. Approach to Differential Thermal Expansion.

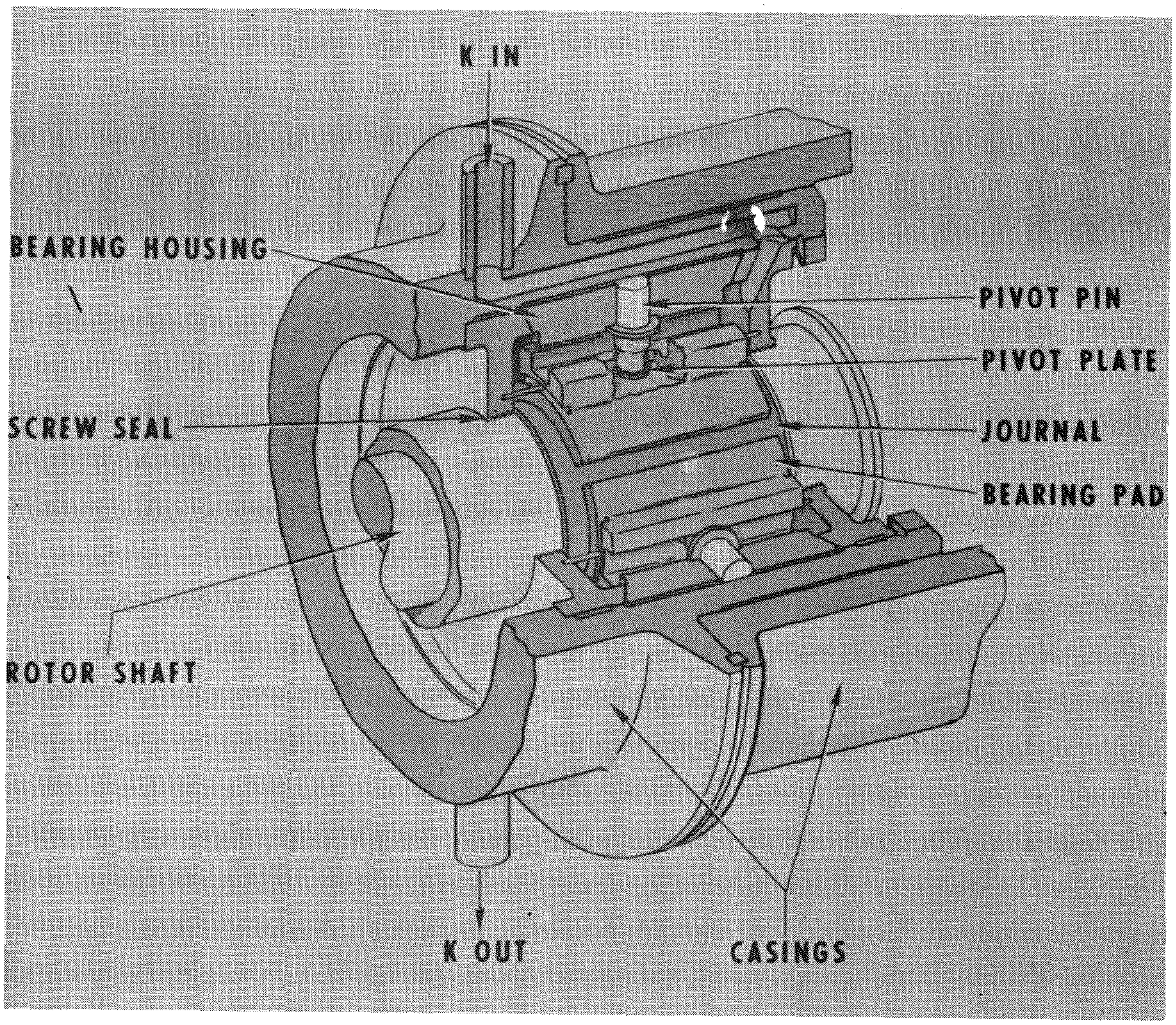

Fig. 28. Carboloy Tilting Pad Bearing.

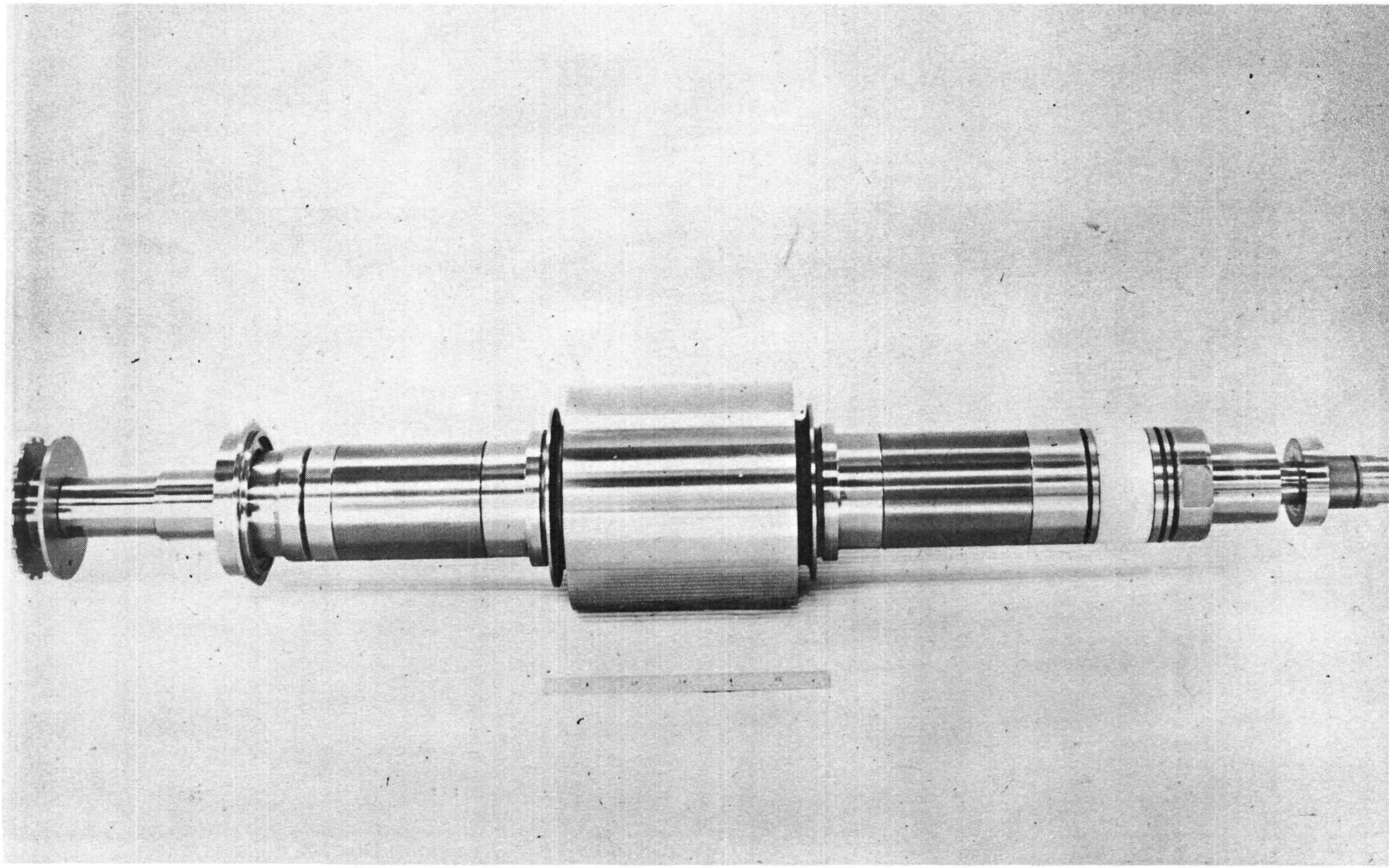

Fig. 29. Liquid Metal Bearing Test Rig Rotor.

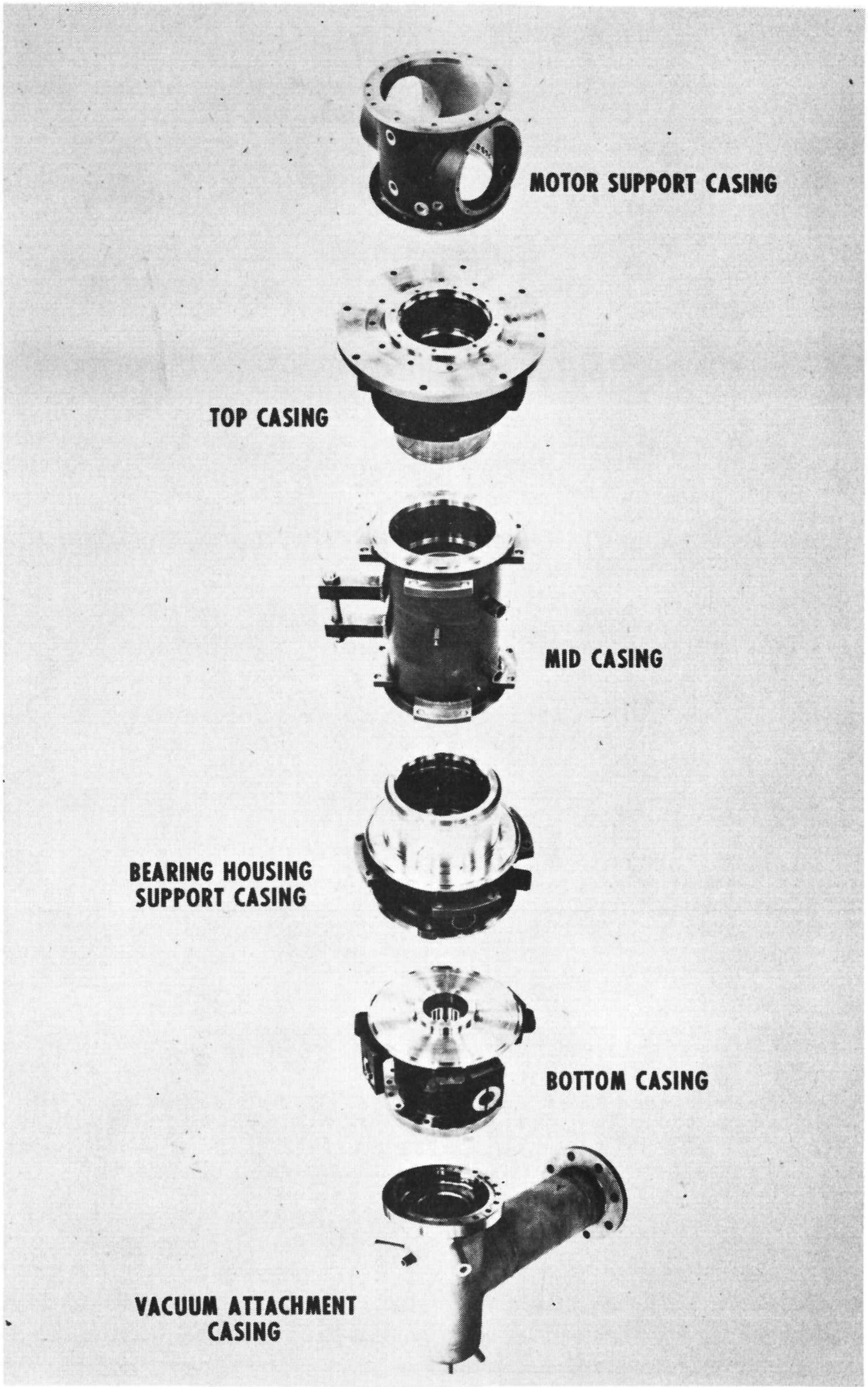

Fig. 30. Liquid Metal Bearing Test Rig Stator.