DESIGN STUDY FOR THE SUPPORT OF AN INERTIAL GUIDANCE TEST FACILITY ON GAS LUBRICATED COMPLIANT SURFACE SPHERICAL BEARINGS

By

J. T. McCabe and F. Kramberger

March 1969

Prepared Under
Contract No. NAS 12-2006
by The Franklin Institute Research Laboratories
Philadelphia, Pennsylvania

Electronics Research Center
National Aeronautics and Space Administration

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF FIGURES (Cont'd)

DESIGN STUDY FOR THE SUPPORT OF AN INERTIAL GUIDANCE TEST FACILITY ON GAS LUBRICATED COMPLIANT SURFACE SPHERICAL BEARINGS

By J. T. McCabe and F. Kramberger

The Franklin Institute Research Laboratories
Philadelphia, Pennsylvania

SUMMARY

A floating sphere approach is being considered for developement of a multi-purpose guidance and control test platform. All attitude capability, the elimination of requirements for both gimbals and angular base motion isolation, and the potential of low friction torque disturbances are factors which make the proposed platform developement attractive.

The objective of this contract was to develop and test compliant surface gas lubricated bearings for the spherical test facility.

Past experience with the application of analytical techniques for the determination of compliant surface gas bearing characteristics indicated that the "state of the art" was not sufficiently developed to justify a concentrated analytical effort in this design study. Consequently, the results presented in this report are almost exclusively. experimental.

The development approach here employed consisted of first examining and verifying the vulcanizing and bonding techniques required for manufacturing compliant surface bearing. Next, flat circular compliant bearing pads were constructed and parameters such as the number, size and location of orifices, the influence of peripheral capture of the rubber, and the methods of supplying pressurized gas to the bearings were investigated. During these tests the presence of a "dithering" lip at the rubber O.D. was observed suggesting comparison of a compliant surface air bearing to a "hover-craft" with its essentially constant interior pressure and an encircling, undulating curtain.

The results of the flat pad tests were used to deduce a design for the larger and more complex spherical compliant test pad. Fabrication techniques are described and experimental data are presented which both give test bearing performance and provide a basis for the preliminary design of other compliant bearings.

The primary objective, namely that of developing a compliant bearing for the support of a spherical test platform has been accomplished and the concept has been demonstrated to be practical.

INTRODUCTION

A floating sphere approach has been selected for development of a multi-purpose, multi-axis platform for guidance and control testing. Factors influencing the choice of the floating sphere approach were:

- complete all attitude capability,
- advantages of no gimbal design,
- low coulomb friction torque disturbances,
- no angular base motion isolation.

The floating sphere concept has been mechanized in several forms by several companies. The technology exists for application of this technique for inertial test usage.

The objective of this contract was to develop and test compliant surface gas lubricated bearings for support of a spherical platform.

The tasks of the contract were:

- Select and design a bearing concept suitable for supporting a hollow sphere with an approximate diameter of 40 inches.
- Predict bearing performance using currently available design data.
- Experimentally test the bearing design for friction moment and the load-flow-clearance relation. Determine the total friction moment, stiffness and natural frequency of the bearing support system.
- Provide a detailed design of the sphere, bearing system and support frame.

Compliant surface bearing design data, suitable for the spherical application, was not found in the current literature. Performance predictions of load capacity and flow for spherical bearings were extrapolated from flat pad experimental tests performed during the course of this study. All other tasks of the contract were completed as proposed.

DESIGN SUMMARY

- Static Load:

	Pounds
Steel shell 40" O.D. x 1.5" thick	2000
Closure flanges and support rings	550

Instrument housing	250
Balancing assembly, framing and miscellaneous	500
Electronics and test package	1250
TOTAL	4550
Preload on main support bearings	1200
TOTAL LOAD ON MAIN BEARINGS	5750

- Number of Main Bearings = 4 (See Fig. 9)
- Number of preload bearings = 1 (See Fig. 9)
- Number of pads per bearing = 3 (See Fig. 9)
- Mounting axis of main bearings = 45° to vertical*
- Bearing load (along mounting axis) = 2033 lb.
- Load per pad = 689 lb.
- Pad specifications:

See Dwg. 32G-C2275-01-A, (Fig. 1) for assembly and details.

See Fig. 2 for performance characteristics showing supply pressure, flow and pad rise vs. pad load. Pad rise is the variation in the distance between the metal sphere and the pad metal backing plate. The reference zero for static loading was taken with the test load on the bearing and with zero supply pressure.

At the 689 lb design load for the *main support pads*, the recommended supply pressure is 55 psig, the required flow 2.2 SCFM per pad, and the rise, 0.0026". The over-all stiffness of the pad is 500,000 lb/in.

At the 407 lb design load for the *preload pads*, the recommended supply pressure is 34 psig, the required flow 1.5 SCFM per pad and the pad-sphere separation, 0.0032". The over-all pivot axis stiffness of the pad is 500,000 lb/in.

For the design shown on Fig. 1, and the preliminary design of other compliant bearings, the experimental tests showed a minimum unit load to prevent flutter of the sphere on an excessively soft gas film is 14 psi per square inch of bearing area, a good design unit loading is 24 psi and the maximum recommended unit load is 32 psi. The maximum recommended loading did not result in touch-down. The limit was established by the on-set of a high pitch, high intensity noise coming from the lip at the OD of the compliant pad. Corresponding to the minimum, design and maximum loads were

* The four main bearing assemblies are spaced at 90° intervals on the test stand platform and seated on adjustable screws whose axes are 45° to the vertical and all intersect at the center of the sphere (Fig. 9).

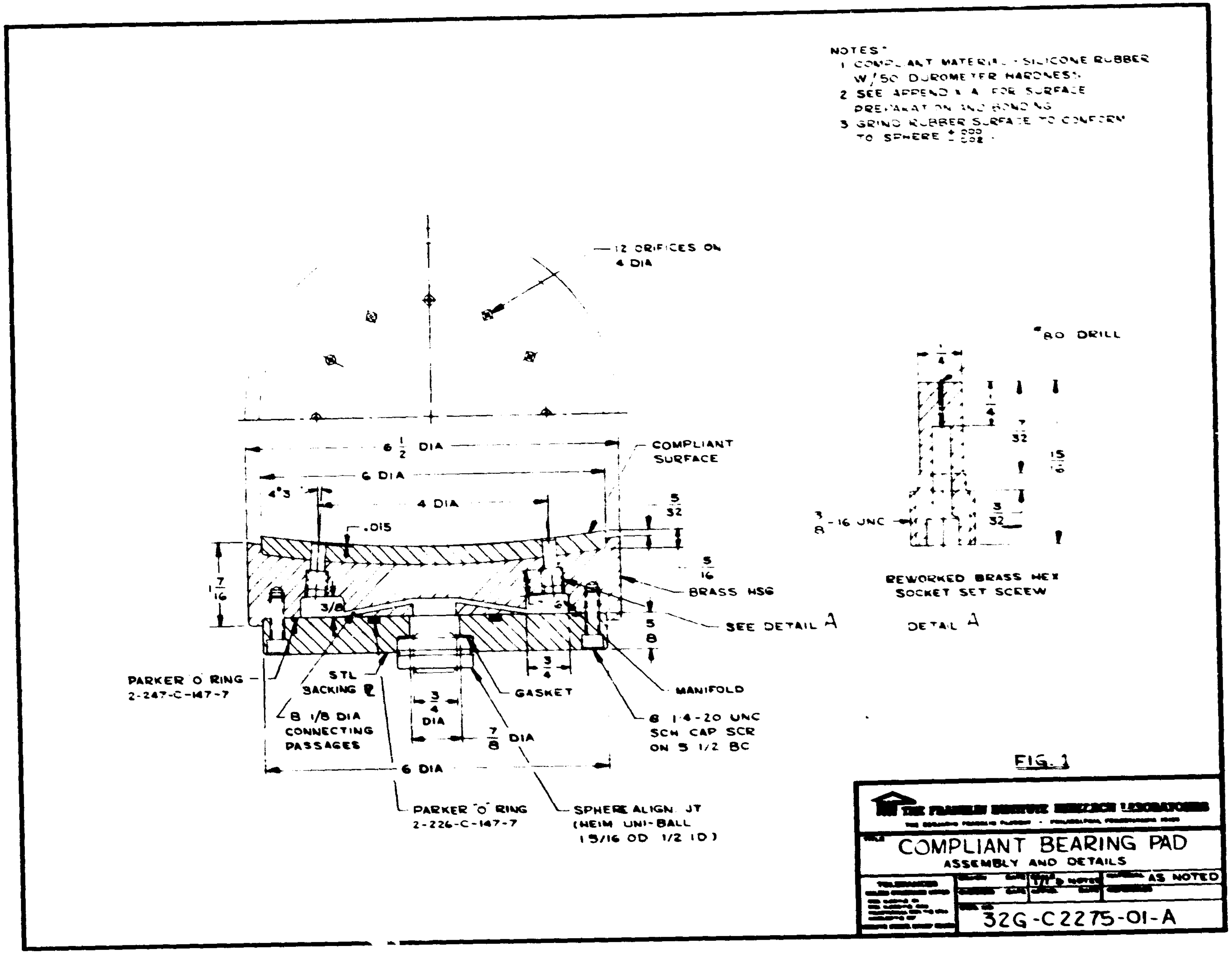


Fig. 1. Compliant Bearing Pad Assembly and Details

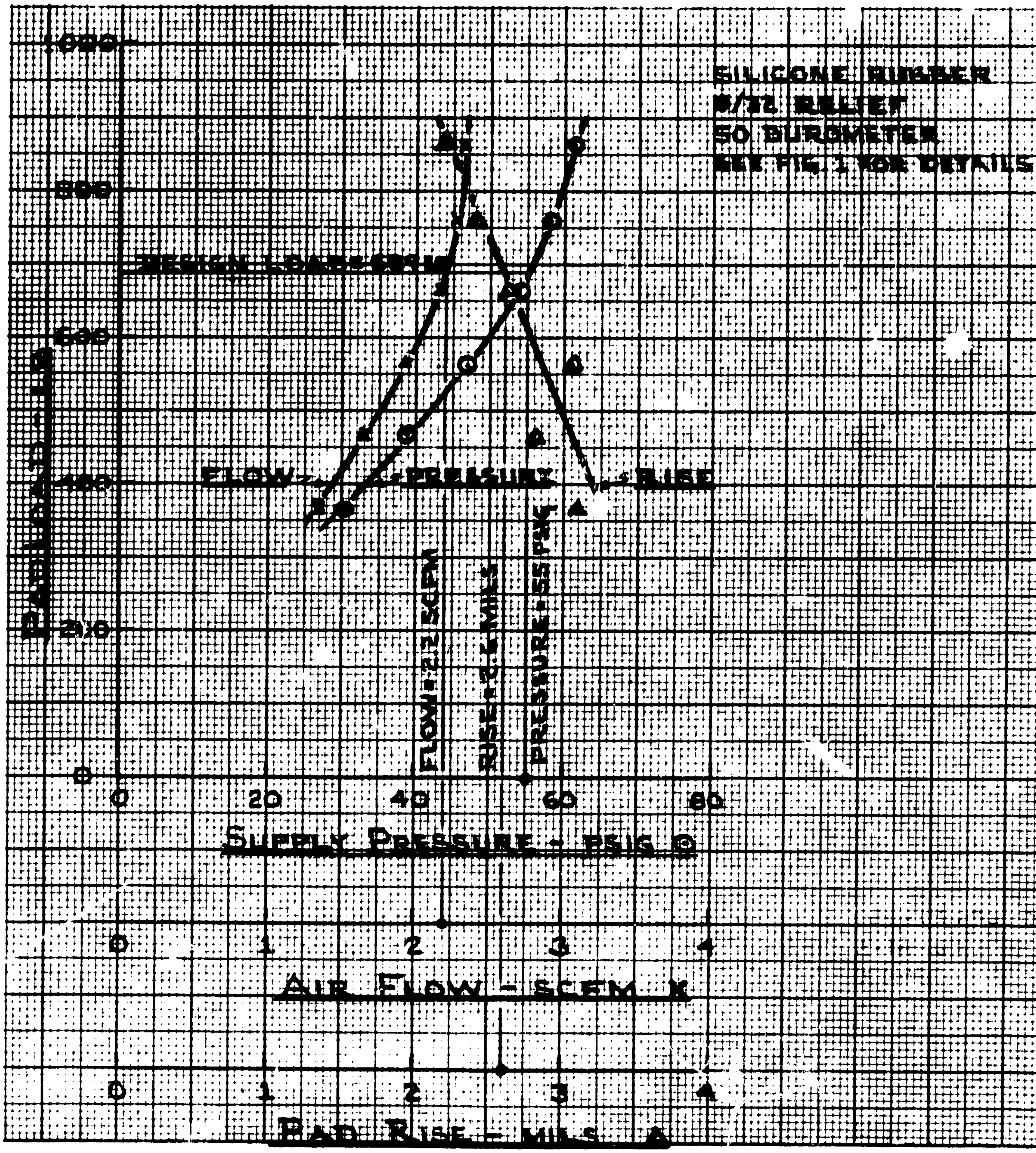


Fig. 2. Performance Characteristics for Compliant Surface Spherical Pad

flows of 0.0534, 0.0778 and 0.0832 SCFM per square inch of bearing area. Because of a relatively linear load-deflection characteristic, an approximate unit stiffness for estimating is 17,700 lb/in per square inch of bearing area.

- Bearing assembly specifications:

  See Fig. 3 (Dwg. 32-C2275-01-B) for assembly and details.

  At design conditions, each main support bearing has a load component of 2033 lb along the 45° mounting axis (center-line of hub, see Fig. 3). The total flow is 6.6 SCFM; the stiffness along the bearing hub axis is 1.45×10^6 lb/in.

- Bearing System Specifications

  Fig. 4 shows the performance characteristic for the sphere's preloaded gas bearing support system as a function of the weight of the sphere. See Appendix B for typical calculations.

  Using the estimated total sphere weight of 4550 lb given above, the gas bearing operating conditions given by Fig. 4 are:

  Supply pressure to 3 preload pads = 34 psig
  Supply pressure to 12 main pads = 55 psig
  Total flow = 30.8 SCFM
  Vertical rise of sphere (zero flow to full flow) = 0.0018"
  Critical frequency, vertical direction = 97 cps
  Critical frequency, horizontal direction = 80 cps

  Fig. 5 shows the same information without preload.

  Fig. 6 shows the relation between supply pressure and gas flow for two conditions. In one, the spherical surface was uninterrupted. In the other, the parting line between the joined hemispheres of the test facility was represented by a 1.3" deep by 0.001" wide slot in the surface of the metal sphere. The difference in flow was negligible.

  Fig. 2 shows that the pad pivot axis stiffness (taken as the ratio of a small change in load over the corresponding pad-sphere separation) is a constant over the applicable load range and equal to approximately 500,000 lb/in.

  The break-away torque to rotate the 40" diameter metal sphere at design conditions, is estimated at 60 in.-lb. A higher pad load (up to 900 lb) requires less break-away torque. Break-away torque was required because of the intermittant contact made between the load-formed lip at the OD of the pad and the surface of the sphere. The magnitude of the break-away torque could probably be reduced by selecting the

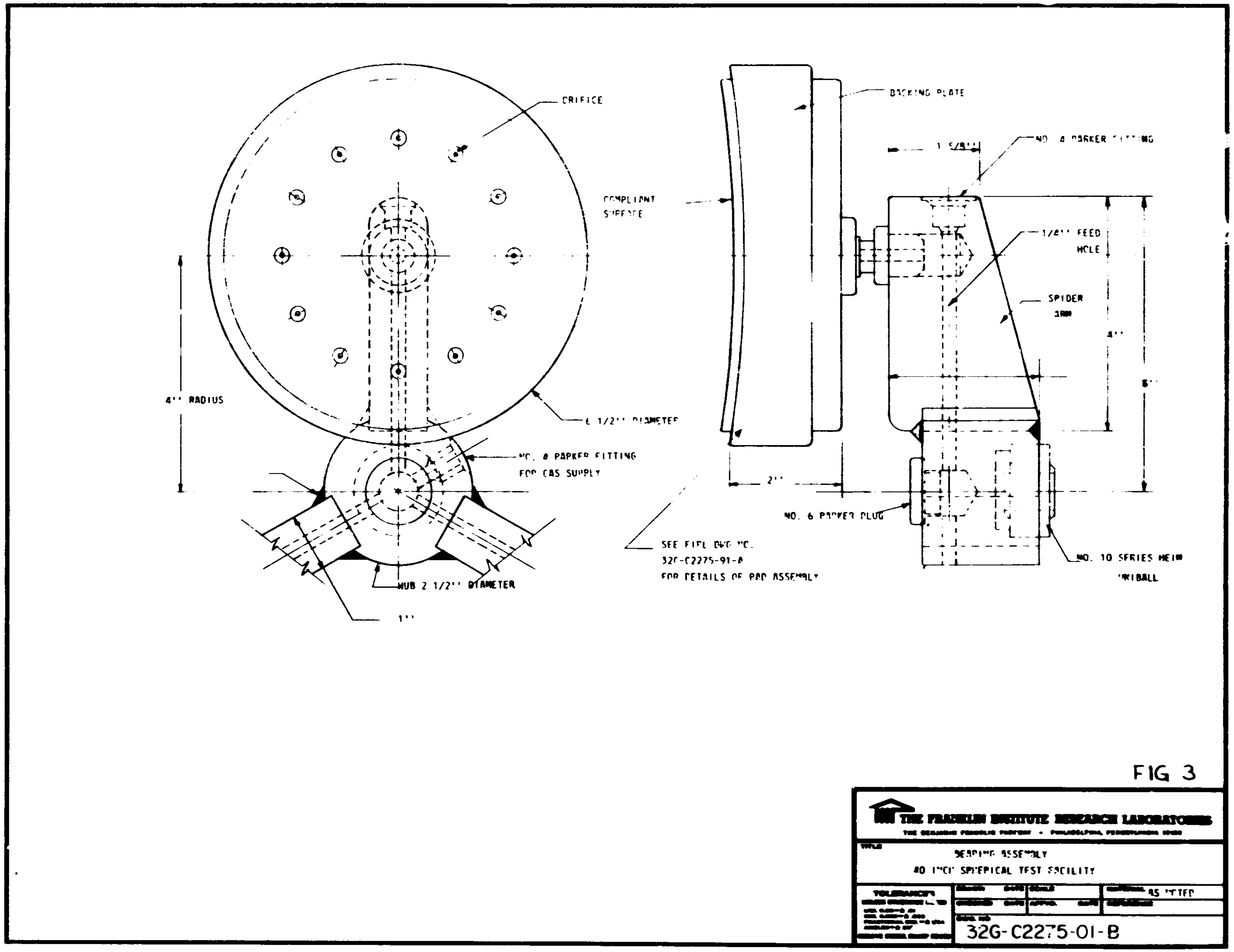


Fig. 3. Bearing Assembly 40" Diameter Spherical Test Facility

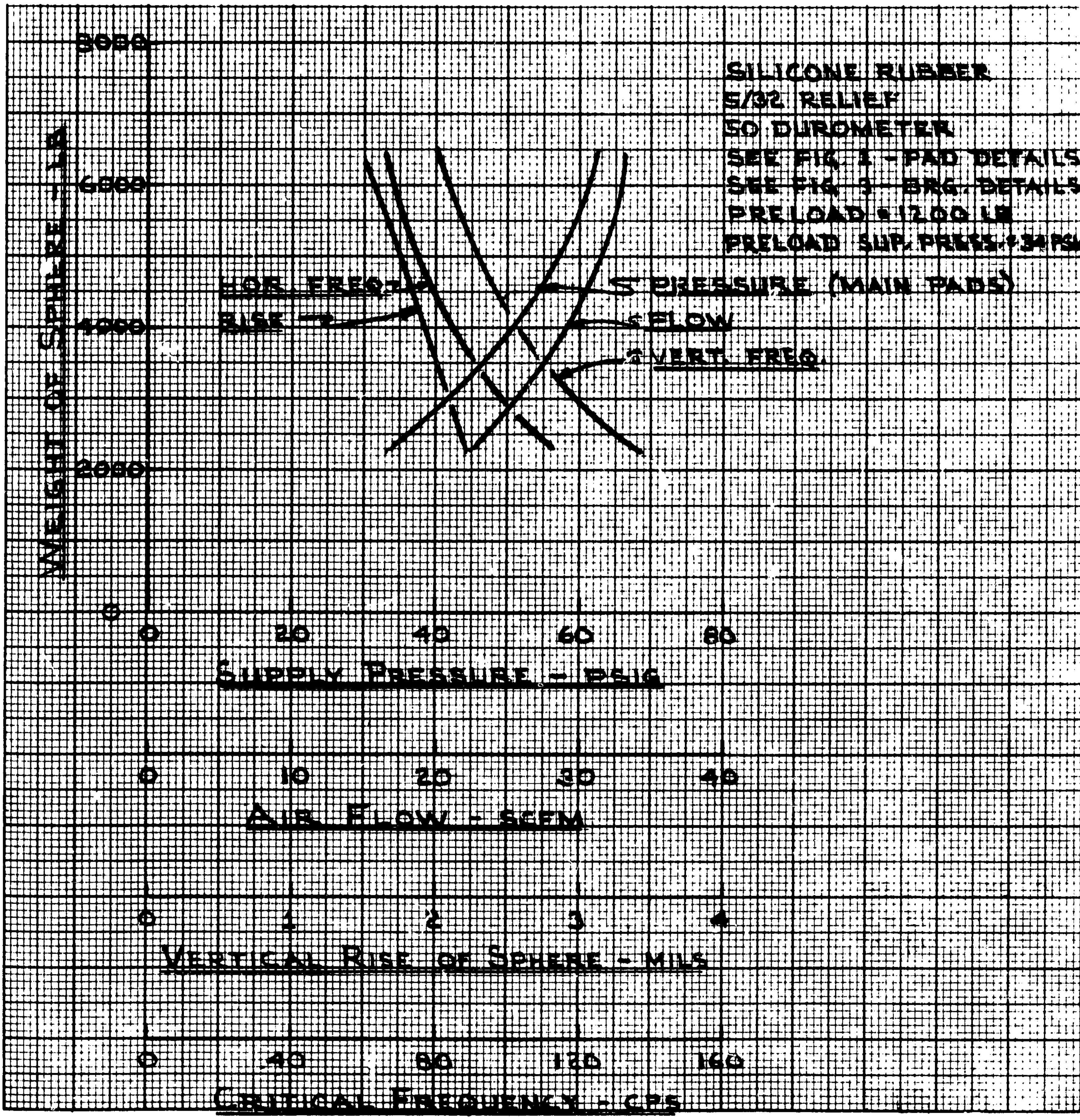


Fig. 4. Performance Characteristics for Compliant Surface Spherical Bearing Assembly with Preload

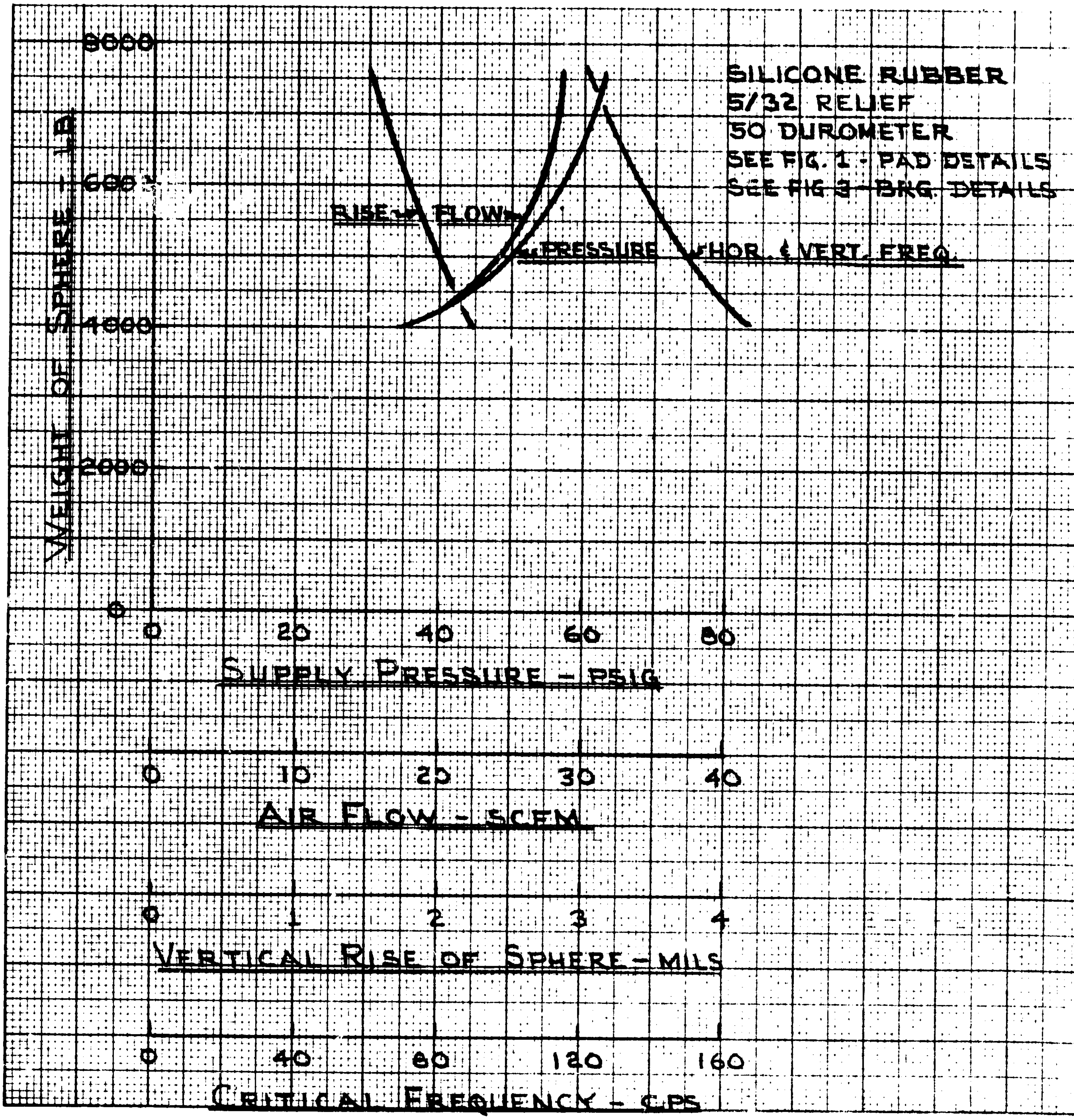


Fig. 5. Performance Characteristics for Compliant Surface Spherical Bearing Assembly Without Preload

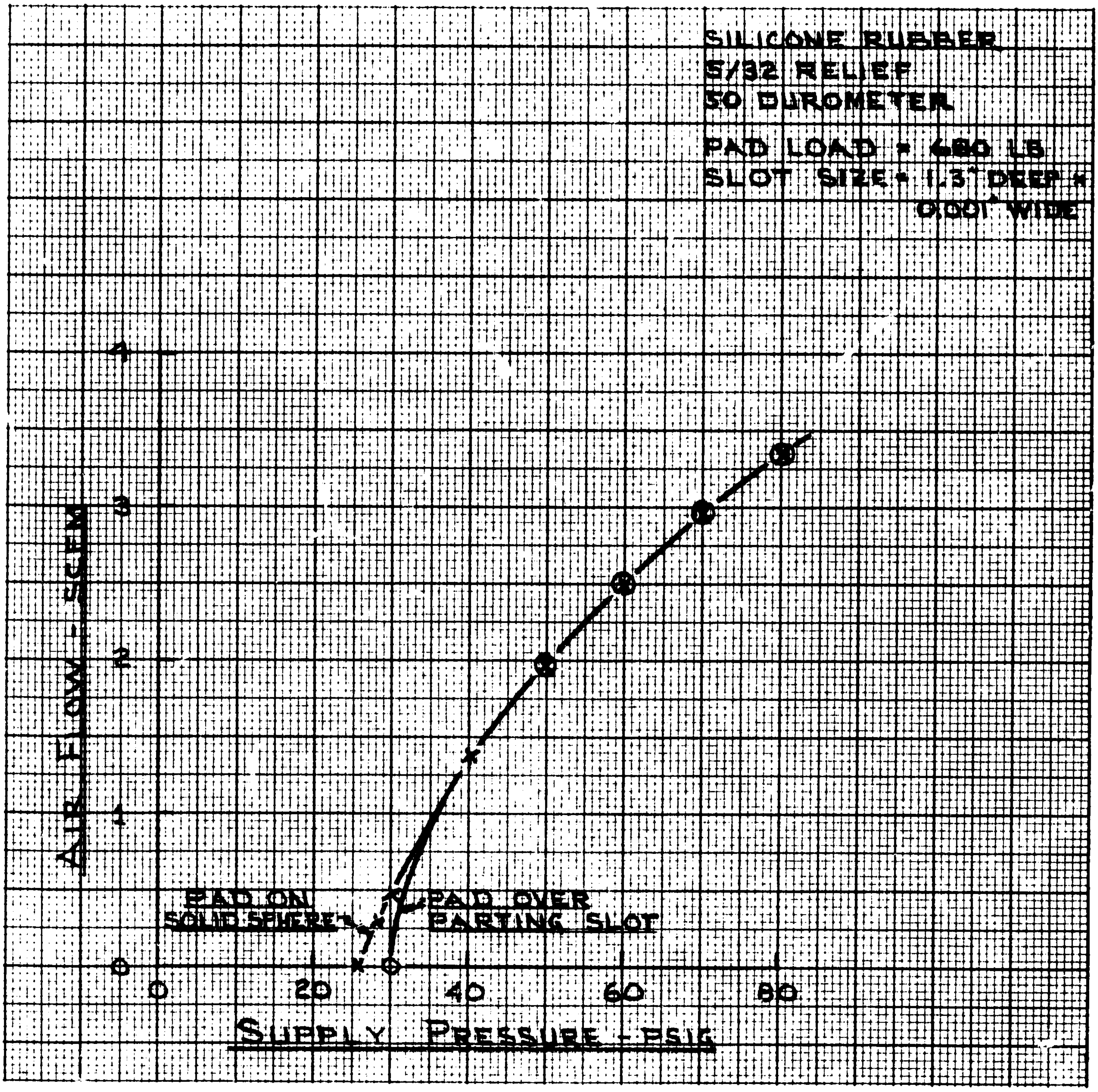


Fig. 6. Relation Between Supply Pressure and Gas Flow Showing Influence of Slot in Spherical Surface

rubber properties and pad shape to decrease lip contact.

The pad and bearing assembly spherical alignment joints are grease lubricated steel-on-brass with a friction coefficient of about 0.15. At design load, the break-away moment for the pad alignment joint is 50 in.-lb. The pad joint is not intended to allow the pad to follow the undulations of the spherical surface. The purpose is to avoid the machining requirements of a rigid mount and to accommodate spheres of different diameter.

Compressor requirements (for bearings & test stand hoist):

Air cooled and tank mounted,
100 psig rating,
Single stage,
1-1/2 hp motor,
Capacity at 100 psig = 57 cfm.
Storage tank capacity = 30 gallons
5 to 10 micron filter

Fig. 7 is a cross section of the spherical test facility showing some of the essential internal components; Fig. 8 shows details of the sphere fabrication. Fig. 9 shows an over-all conceptual view of the test facility.

FABRICATION OF COMPLIANT SURFACE BEARINGS

General

The fabrication of compliant surface bearings involves four principal steps:

- selection and preparation of compliant material,
- preparation of back-up plate,
- bonding,
- molding and vulcanizing.

Each step must be done with care to insure satisfactory results. Commonly incurred anomalies in bearing performance due to failure to observe the proper precautions in preparing a compliant surface are:

- assymmetric righting moment,
- high gas flow,
- touching of interior bearing surfaces.

Bearing righting moment capability is necessary to prevent edge dragging should the pad become cocked or pass over the equatorial joining

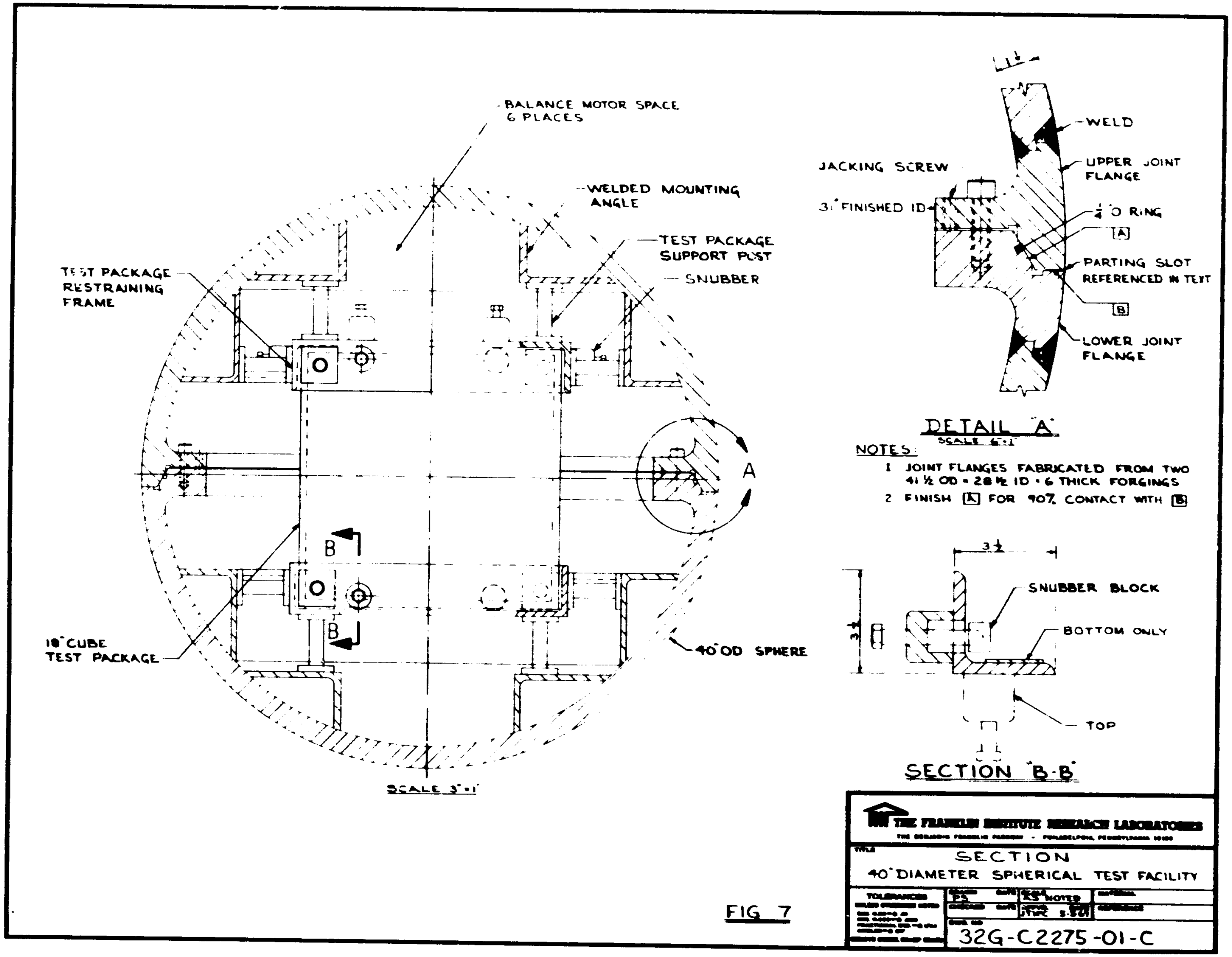


Fig. 7. Section 40" Diameter Spherical Test Facility

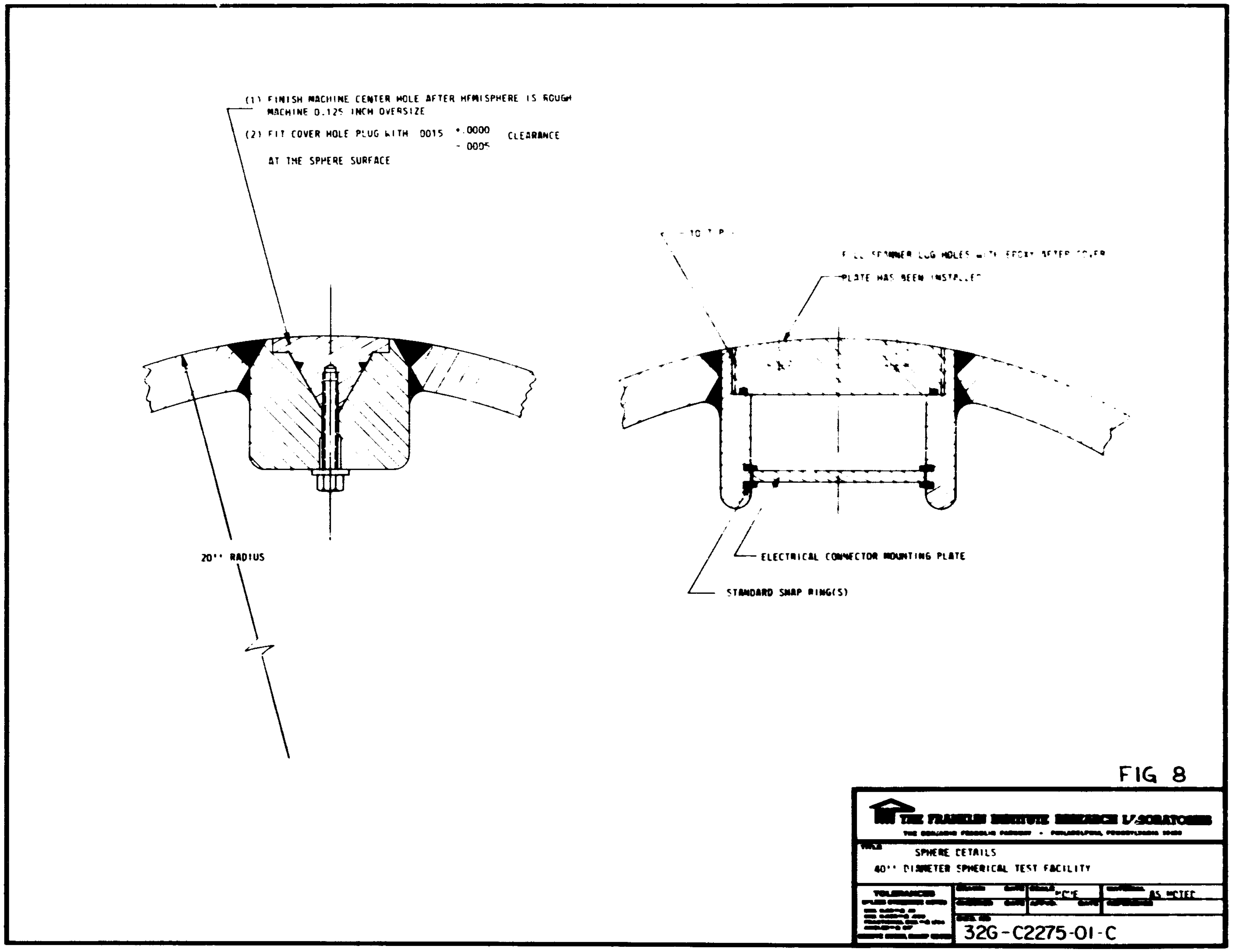


Fig. 8. Sphere Details 40" Diameter Spherical Test Facility

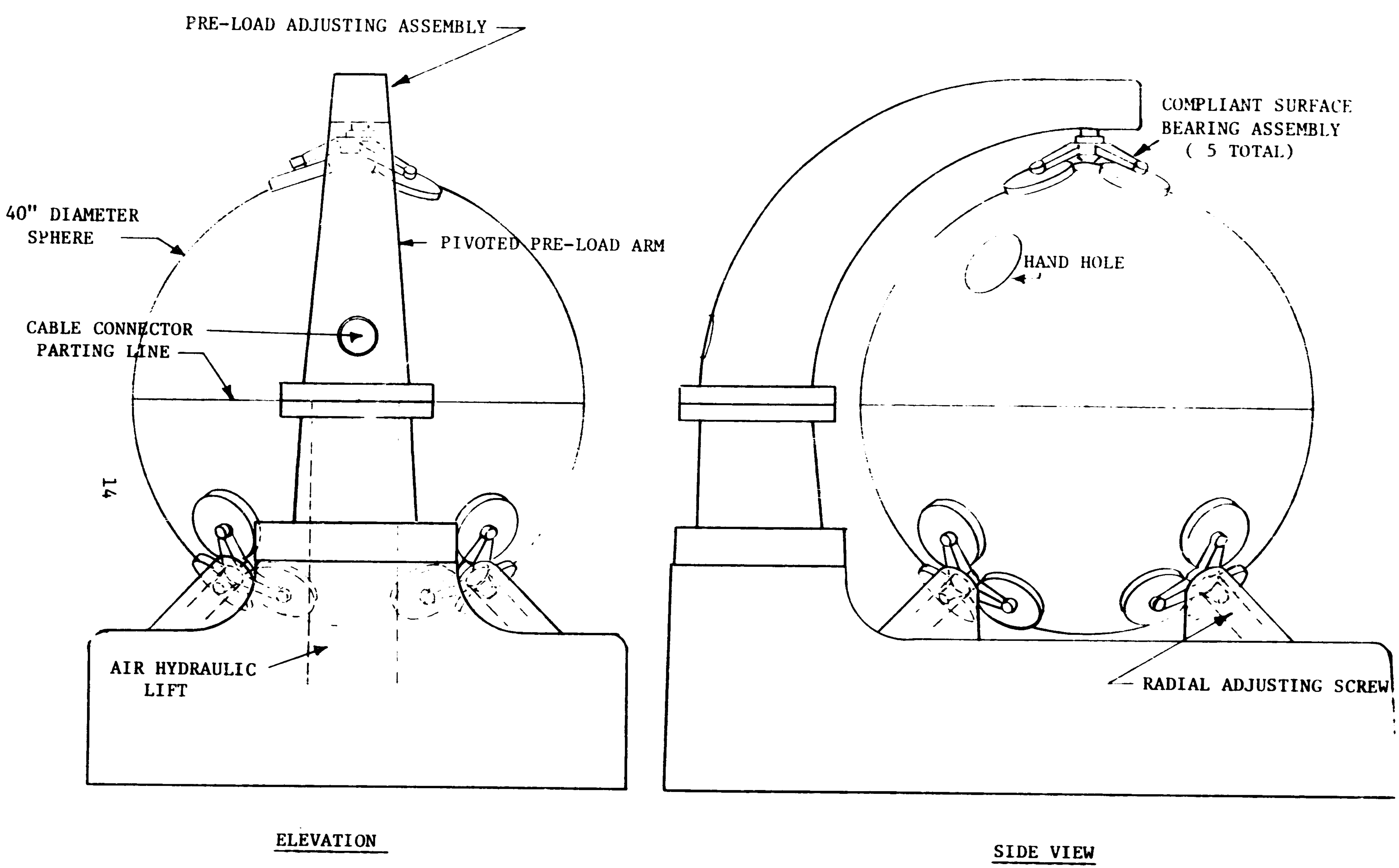


FIGURE 9 CONCEPTUAL ARRANGEMENT OF 40" DIAMETER SPHERICAL TEST FACILITY

slot between the two hemispheres. An assymetric righting moment can result from several causes most of which are related to nonhomogeneous rubber. Hard spots are often encountered when compliant surfaces are cut from cured rubber sheets or when partial spontaneous curing occurs due to prolonged storage of raw stock. In general, the stock should be kept in refrigerated storage and the shelf-life recommended by the manufacture observed.

High gas flow, which can be accompanied by an assymmetric moment response of the bearing pad, can be caused by the presence of a "channel" in the rubber surface that connects a gas supply orifice with the ambient. Non-uniform rubber shrinkage and improper surface grinding are potential causes of channels leading to excessive gas flow.

A troublesome aspect of compliant bearing fabrication is encountered when the rubber surrounds inclusions, such as orifice assemblies, that protrude from the backing-plate. Failure to maintain a complete bond over the surface of the insert usually results in a bulging of the rubber in the vicinity of the orifice causing local contact and an unduly large operating friction torque. Incomplete bonds cause high stress concentrations leading to a propagation of bond failure.

Selection and Preparation of Compliant Material

Best results are usually obtained when uncured stock is directly vulcanized to the backing-plate of the bearing pad. More uniform -- and thus more predictable -- bearing characteristics are obtained by this method as opposed to cutting the compliant surface from a precured sheet. Various compounds are available from various suppliers;* often these compounds are considered company proprietary. Thus, the user is forced to request a performance specification rather than a materials specification. In addition, the number of specifiable items is minimal. Among the more important of these are 1) a selection between organic and non-organic compounds and 2) the Durometer, or hardness of the rubber. In principle, silicone compounds should offer some advantages over most organic compounds. These advantages are a lower susceptibility to compression set,** more linear properties and a slower change in physical characteristics with time.

Insofar as rubber hardness is concerned, the experiments performed on this program indicated that 40 Durometer was too soft and 50 Durometer, the only other hardness tested, was suitable.

*For instance, Uniroyal, Passaic, New Jersey or Astro Molding, Inc., Old Bridge, New Jersey.

**A plastic deformation of the elastomer.

Preparation of the raw rubber stock for molding depends both on shelf time and storage conditions. If the uncured rubber is kept refrigerated and molding is done soon after blending (one month or less) special preparation need not be undertaken. If the raw rubber require softening before molding then mill shearing and calendar processing may be required.*

Preparation of Back-Up Plate

Brass is the most suitable material for a compliant surface back-up plate. It is easily machined, does not corrode in a lab atmosphere and provides an exceptionally good bonding surface. The surfaces to be bonded should be degreased by soaking in trichlorethylene for about 10 minutes and dried in a clean atmosphere. An acid bath consisting of (by volume):

sulfuric acid	44.4%
water	33.3%
nitric acid	22.2%
hydrochloric acid	0.1%

should then be used to etch the surface to a dull, matty appearance (about 1 minute). This should be followed by rinsing in distilled water for about 10 minutes and drying in a clean oven (150°F) for about 1/2 hour. Care should be taken to avoid touching the surface to be bonded. The backing plate should be allowed to cool in a clean atmosphere before applying the bonding agent.

Bonding

Organic rubber.--The organic rubber used in this program was an electrically conducting material called Uniroyal P35-2b. Bonding is done with a two-step adhesive called Chemlok 205 and 220. The prepared brass backing-plate is first given a thin coat of primer (205) which contains keystone as a solvent and takes about 1/2 hour to dry under stmospheric conditions. This step is followed by the application of the adhesive (220) in a thick coat directly on top of the primer. No adhesive or primer is applied to the rubber. Drying time is about 1/2 hour.

Silicone rubber.--The silicone rubber used in the program was compounded and supplied by Astro Molding, Inc. of New Jersey. The primer is Dow-Corning 4002 applied to all surfaces to be bonded and given 1/2

*"On Rubber-to-Metal Bonding Techniques with Reference to Compliant Surface Bearings" by Glenn K. Rightmire, Columbia University, Lubrication Research Lab., Dept. of Mech. Eng., Report No. 10, November 1967.

hour to dry. The adhesive is G.E. RTV-112. If a silicone pad is to be glued to the backing plate, then the pad should be wiped with Dow-Corning 400., coated with RTV-112, loaded with about 10 psi and let dry for 24 hours. The load should be removed and the piece placed under water for 12 hours.

Molding and Vulcanizing

Two types of molding were employed. One was compression molding of the organic rubber. Here, sheets of raw rubber were placed in a heated, press-mounted mold of the desired shape. A cure time of 90 minutes at 1500 to 2000 psi and 290°F was recommended for vulcanizing. This procedure was followed and the results were excellent.

The other method was transfer molding.* This technique did not result in as good a bond as was obtained by compression molding. One possible reason was that the rubber, flowing from the sprue hole to fill the mold cavity, scrubbed away some of the adhesive from sections of the backing-plate.

The experiences encountered in this program regarding the fabrication of compliant surface bearing lead to the following recommendations:

- vulcanize uncured stock directly to a brass backing-plate,
- use the compression molding technique.

PRELIMINARY TESTS WITH FLAT PADS

Purpose

Before fabricating the more expensive spherical bearing pads, tests were conducted with low cost flat pads for the following reasons:

- to test the rubber suppliers' recommended bonding and vulcanizing procedures,
- to determine if surface griding would be required,
- to screen potential orifice configurations and compliant surface layouts,
- to provide design data for sizing the spherical pad and estimating the flow requirements,
- to determine the conditions leading to contact between the pressure deformed rubber and the steel bearing surface.

---

*Raw rubber is forced by a plunger through a connecting tube to the sprue hole of the mold cavity.

Conclusions

- The bonding procedures described in the previous section result in successful bonds,
- Surface griding of the compliant layer is recommended,
- Righting moment capability is an important design consideration,
- The bearing gas supply for the spherical pad should come from orifices spaced relatively close together and on a pitch circle 2/3 to 3/4 of the pad OD,
- A suitable thickness for the finished compliant material is 1/4" to 5/16"; 50 Durometer is a suitable hardness,
- Assuming a supply pressure of 100 psig or more, a good design unit load (over projected area of pad) is about 30 psi.
- The amount of "relief"* is a critical design parameter,
- For the spherical pad, the relief should be between 50 and 75 percent of the compliant disc thickness.
- The rubber should be molded over the full face of the pad,
- Although "hammer" can be a design problem for compliant surface bearings, there are enough variable design parameters, such as rubber hardness, thickness and relief, to achieve a successful design,
- Unless the pad load is very low, an intermittant dynamic contact between the rubber and metal bearing surfaces can be anticipated at the ambient edges of the pad.

Description of Test Facility

The basic components of the test facility used in the flat pad studies consisted of a steel plate containing orifices, a means for loading the plate, a means for measuring gas flow, a flat compliant test pad, a means for measuring the distance between the test pad and the steel orifice plate and a spherical alignment joint between the test pad and ground. Load was provided by mounting the orifice plate assembly on a pneumatically operated air-bearing guided piston. The flow characteristics of the orifice plate and the calibration curve for the "variable area" meter used in the flow measurements are given in Appendix A. Separation between the orifice plate and the compliant surface backing-plate caused by hydrostatically pressurizing the bearing was measured by three "tenth" indicators.

Moment loading tests were accomplished by hanging weights from an extension arm fixed to the backing-plate.

---

*"Relief" is the distance the shoulder of the mold cavity is machined off in exposing the O.D. of the compliant disc (see Fig. 10).

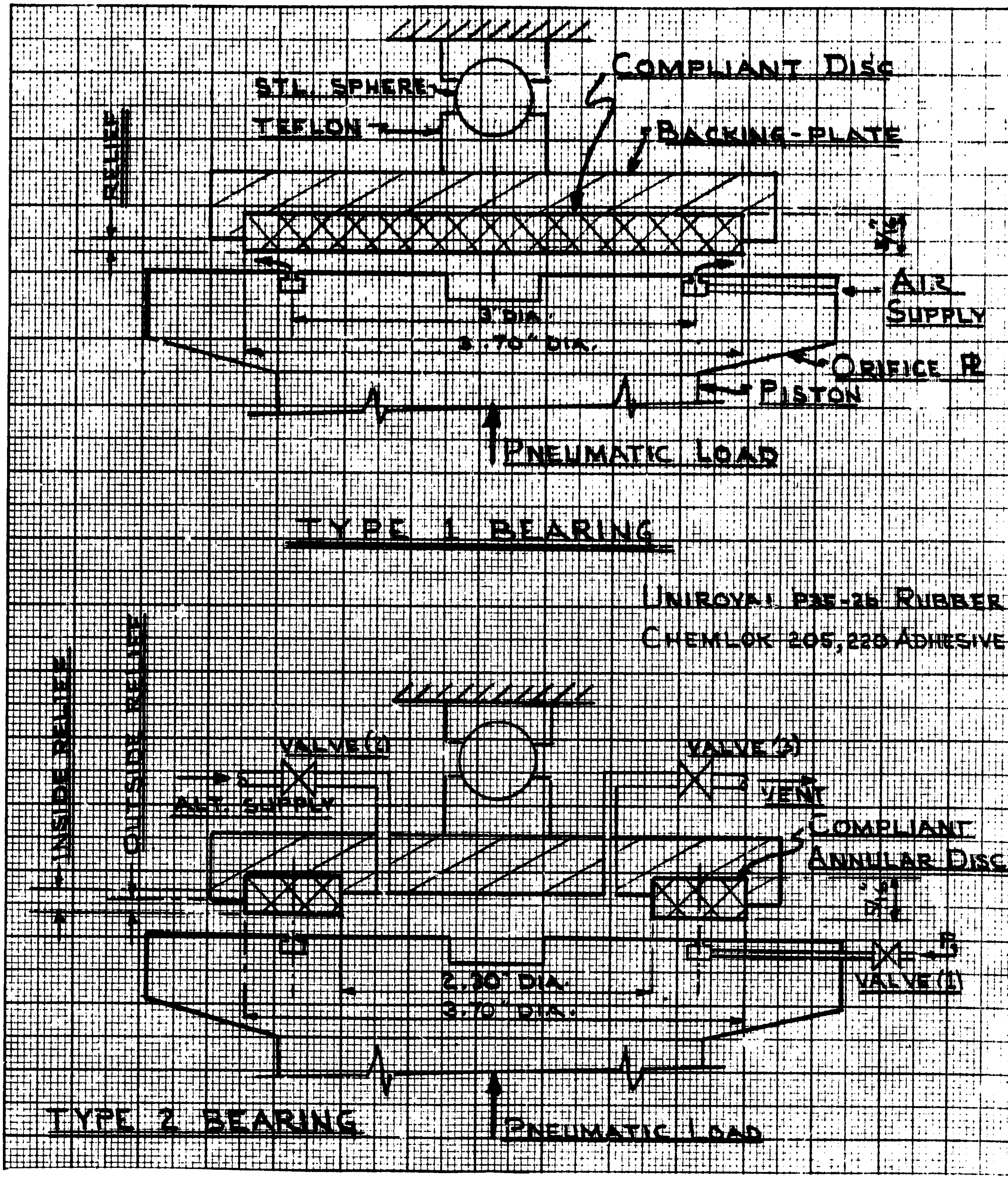


Fig. 10. Configurations Investigated During Flat Pad Tests

Configuration Screening

Fig. 10 shows the configurations investigated during flat pad testing. Essentially, two types of compliant pads were studied; one with a full face, the other with an annular face. Several methods of feeding pressurized gas to the annular pad were screened.

In all cases, surface grinding of the rubber to remove shrinkage distortion was found necessary. When surface grinding was not performed, light, localized bearing surface contact often existed and produced an undue increase in operating torque.

Annular bearing (type 2).--When gas was fed through the orifice plate and the center of the pad was vented to atmosphere, the pad resonated at low loads and hammered violently at higher loads. With the application of still higher loads, hammer stopped. Under these circumstances, the inside annular edge acted as a seal and the vent flow was measured as zero. The interior relief (Fig. 10) was 0.015" for this and all other tests of the Type 2 configuration.

A second scheme consisted of supplying gas to the pad interior (valve (2), Fig. 10) *without* a restrictor in the feed line. Thus, the pad was inherently compensated with the clearance at the ID of the compliant sill acting as the restrictor. The performance of this scheme as a bearing was poor because of excessive gas flow and low film stiffness over a wide load range.

A third scheme consisted of inserting a series of restrictor in the test pad feed line. In addition to the annular compliant pad (type 2 bearing), a geometrically equivalent steel pad was also constructed. Neither the compliant pad nor the limiting case of the rigid surface steel pad was capable of supporting a significant moment loading over the wide range of restrictor values employed. Thus, bearing righting moment deleted this scheme.

In one of the restrictor fed tests, the annular pad was hand-held and moment loaded with "finger" pressure against a ground glass surface. Even with a supply pressure of 50 psig, contact over 15% of the bearing area could be achieved easily. Also visible through the glass was the formation of both the interior cavity and a "lip" at the OD of the rubber.

The fourth feeding scheme used the orifice plate with the recess feed and vent line valves closed. Thus, the principal flow path was from the orifices to the pad OD and the bearing was similar to the full face case except for two additional fabrication considerations. These were (1), the removal of flashing from the inside edge and (2), molding to a shallow recess depth on the ID while still maintaining the desired relief on the OD. Since the full face pad avoids these problems and since no other feeding arrangement proved satisfactory, the annular configuration was disgarded.

Full bearing (type 1).--Two types of rubber "crush" tests were performed to determine the deflection of the rubber alone. These consisted in loading the orifice plate against the compliant pad without the presence of a gas film. In the first case the rubber and metal surfaces were clean and dry. These tests indicated an insensitive relation between deflection and relief due to adhesion between the rubber and metal. In the second case, the bearing surfaces were coated with a thin film of oil to simulate the negligible surface traction the rubber would realize when floating on a gas film. Fig. 11 shows the significant influence of the oil coating on the load-deflection-relief relation. Also shown on this figure is the same information with a gas film present, and when the crush is subtracted from the gross deflection, the result is shown on Fig. 12. Note the relative independence of deflection on relief up to about 1/4 to 1/3 of the compliant layer thickness and the change in character for the curve representing full relief.

Fig. 13 is taken from the slope of the curves on Fig. 11 at an axial load of 100 lbs. and shows the influence of relief on bearing stiffness. Note the maximum bearing stiffness (136,000 lb/in) is achieved for the special case of *zero relief* and that the stiffness is 63% of the film stiffness for an equivalent rigid surface bearing.

Fig. 14 shows a comparison of the load-displacement relation between the rigid surface and zero relief compliant pads. The zero relief compliant pad is similar to the Type 1 bearing shown on Fig. 10 except after vulcanizing, the shoulder of the backing-plate is chambered and ground flush with the compliant surface. At a relative deflection of about 4.2 mils, Fig. 14 shows the collapse of the rigid pad (zero flow and metal-to-metal contact). On the other hand, the compliant pad is capable of supporting a much greater load though at a lower stiffness where equal load comparisons are possible. Thus, there is high load capacity (and flow) relative to a comparable non-compliant pad.

The displacement of the rubber from the interior of the zero relief compliant cavity forms a "lip" around the OD of the pad. Tests with "Prussian Blue" and continuity checks with an Ohm meter indicate the lip exists and makes intermitant contact with the other bearing surface. It appears that, except possibly at very high load, the shoulder of the zero relief pad will *not* make metal-to-metal contact because of the formation of the lip. Most of the pressure drop appears to take place near the lip with a relatively constant pressure in the interior cavity.

The zero relief bearing is not suitable for the present application because of low compliance. However, this configuration should find many other applications because of its high load capacity per unit area, the elimination of the usually expensive gas bearing surface finish requirements and its relatively high stiffness (for a gas bearing).

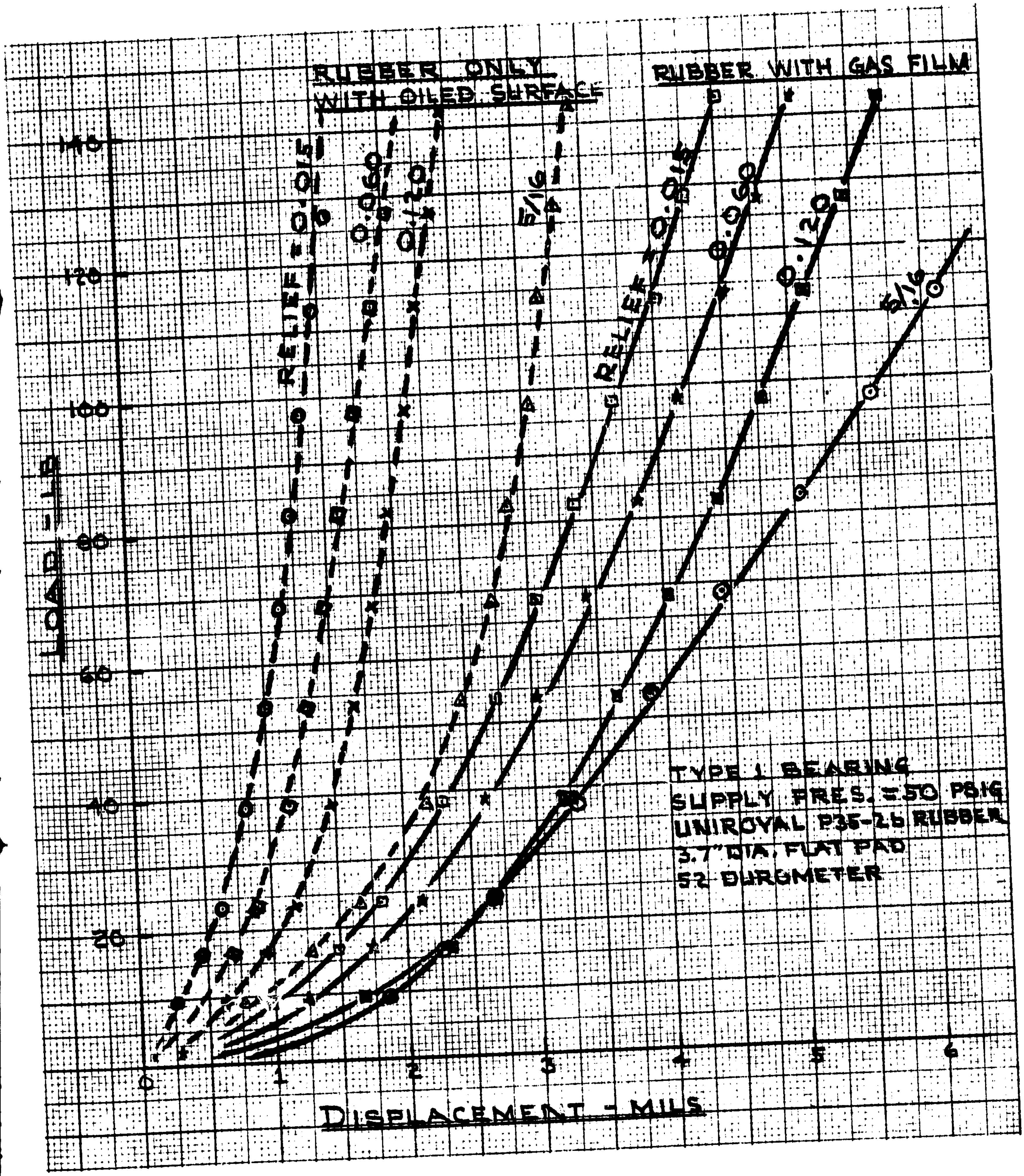


Fig. 11. Influence of Relief on Load vs Displacement With and Without Gas Lubrication

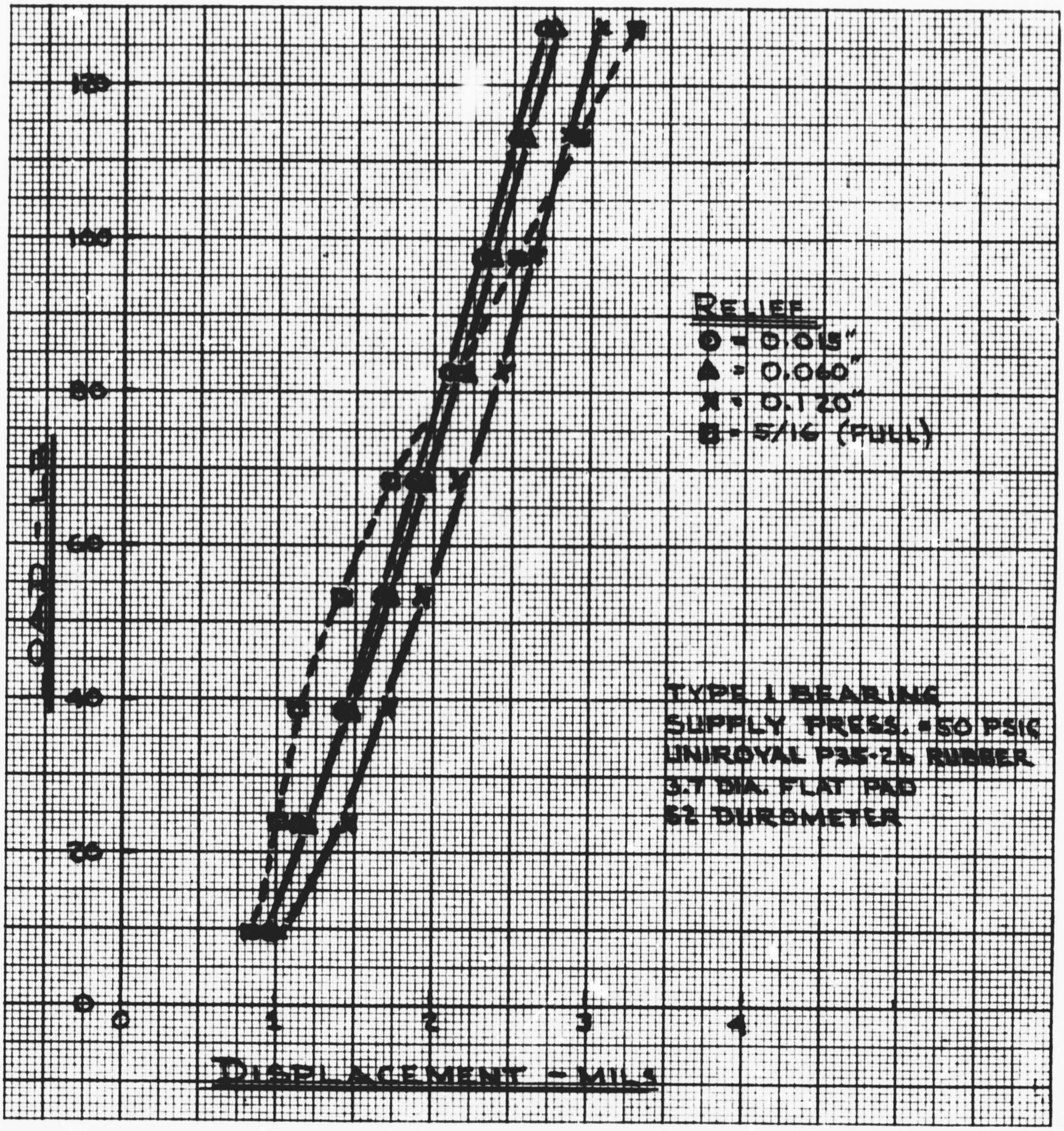


Fig. 12. Influence of Relief on Load vs Displacement With Crush of Rubber Substracted

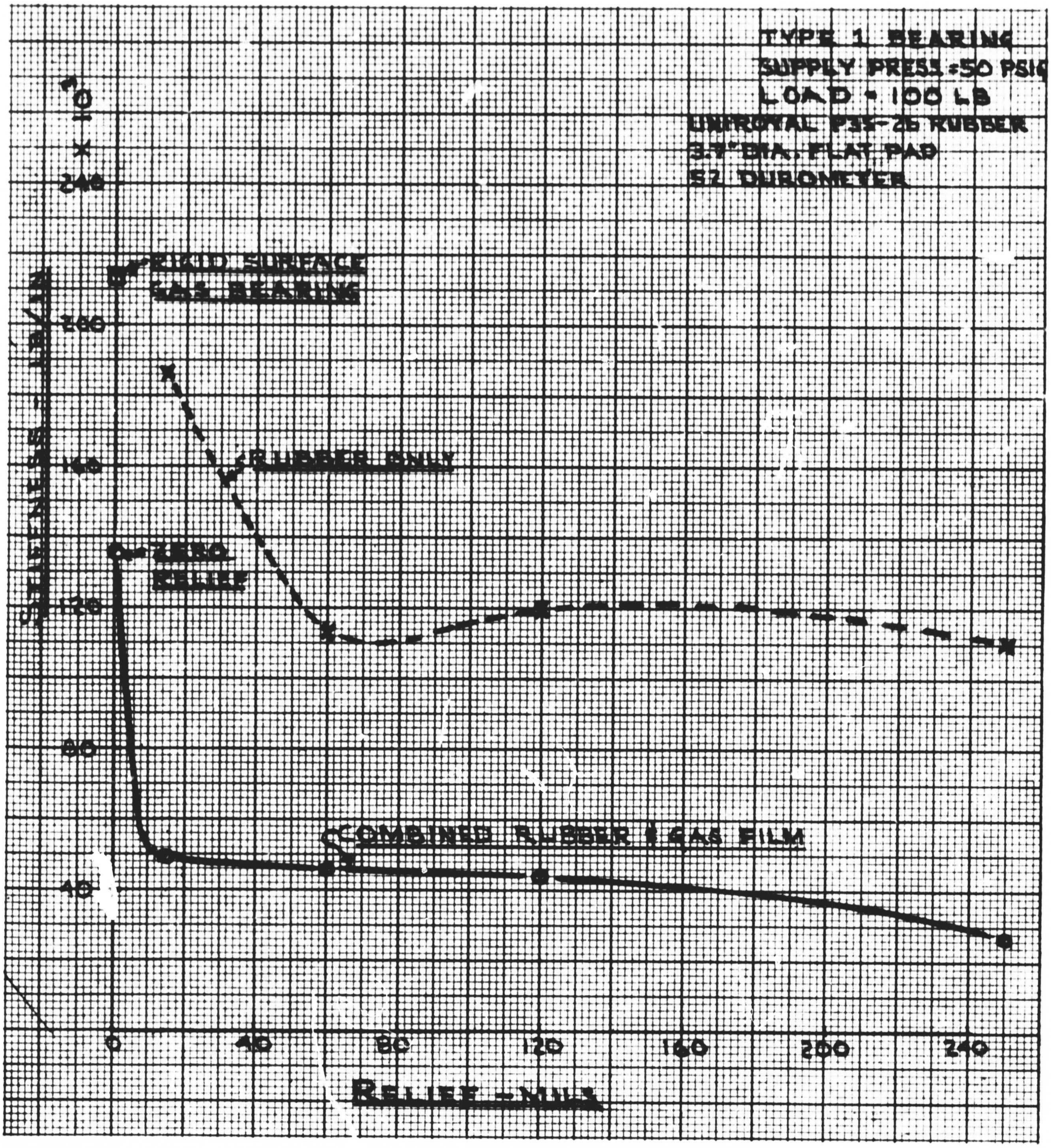


Fig. 13. Influence of Relief on Bearing Stiffness

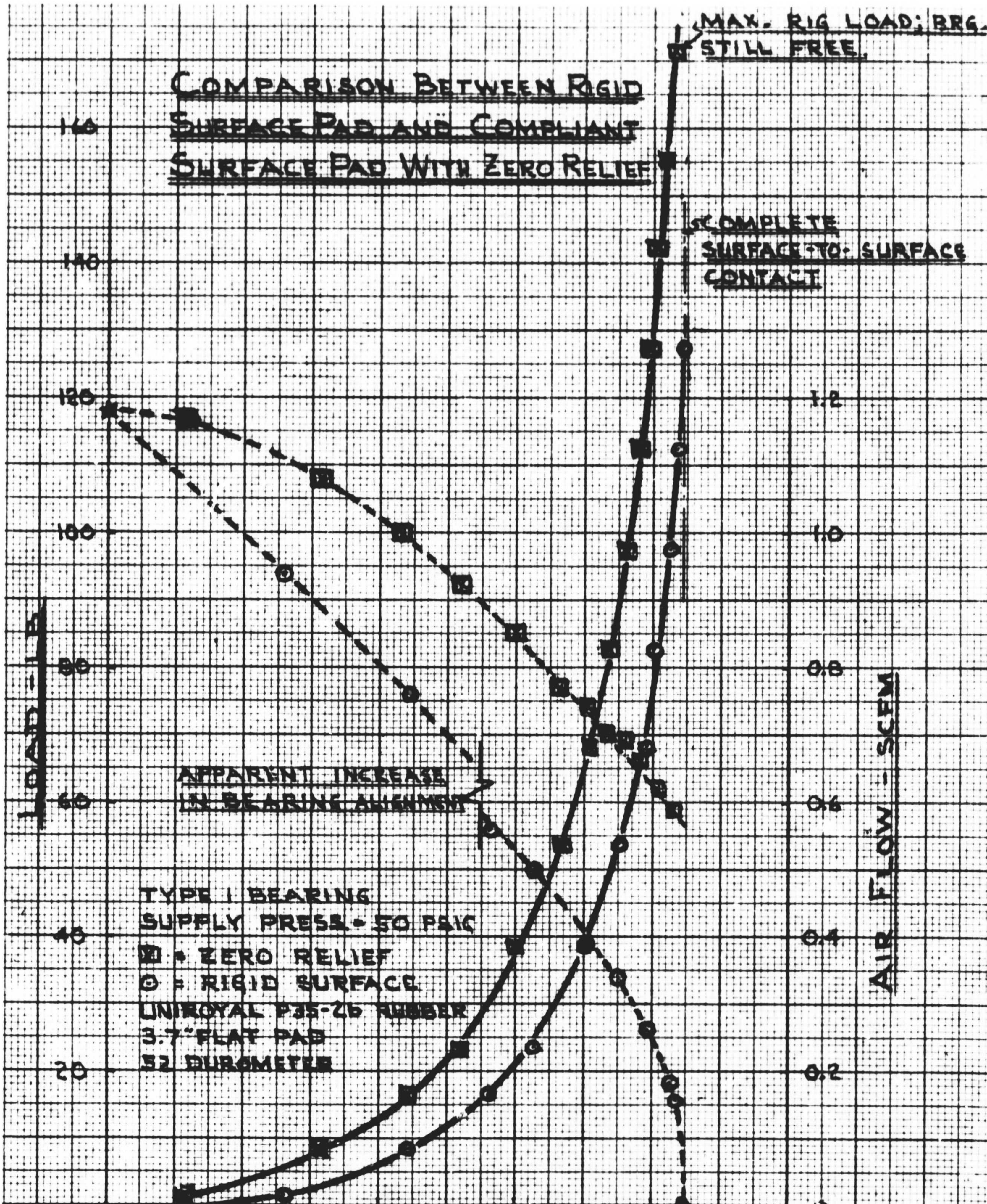


Fig. 14. Comparison Between Rigid Surface Pad and Compliant Surface Pad with Zero Relief

Fig. 15 shows the rotation of a pad due to an applied moment. The angular stiffness is seen to increase significantly with axial load (about 20,000 in-lb/radian at the light load of 53 lbs.).

The principal conclusions reached during the moment testing were:

- For axial load capacity, the supply orifices can be located at the center or even near the pad OD as long as the orifices discharge into the pressure formed compliant cavity.
- A compliant bearing with a negligible righting moment will not give satisfactory performance. Even if moment loading is not a principal design consideration, the bearing should have righting moment to prevent draging an edge.
- Righting moment can be satisfactorily provided by feeding the air through orifice restrictors located on a circle with a diameter equal to or about 2/3 to 3/4 of the bearing OD.
- The axial load threshold of pneumatic hammer is sensitive to moment loading.

Experimental Observations of Lip at Pad OD

The following observations were made concerning the character of a lip formed at the OD of the pad by the pressure displaced compliant material from the interior:

- When the orifice plate was smeared with Prussian Blue, (.0001" to .00015" deep), a sharp 360° contact circle was observed at the OD of the rubber, even under light load.
- Leak-teck (soap solution) indicated flow leakage from essentially all points of the perimeter.
- Resistance measurements using an Ohm meter indicated partial continuity increasing with increasing load (conducting rubber with 52 Durometer used).
- A circuit consisting of a 6 volt battery, both bearing members and an oscilloscope indicated partial contact between the rubber and the steel bearing plate for all test pads. The zero relief displayed "high frequency noise"; the fully relieved bearing showed a "DC" response level between the open circuit and closed circuit levels.
- When the compliant material was chamfered the lip formed at the ID of the chamfer. If the OD of the pad would wear due to rub, the lip might simply migrate to the interior and the effective bearing area be reduced. Although this does not appear to be a problem for the present spherical application, it may be a serious problem for other applications.

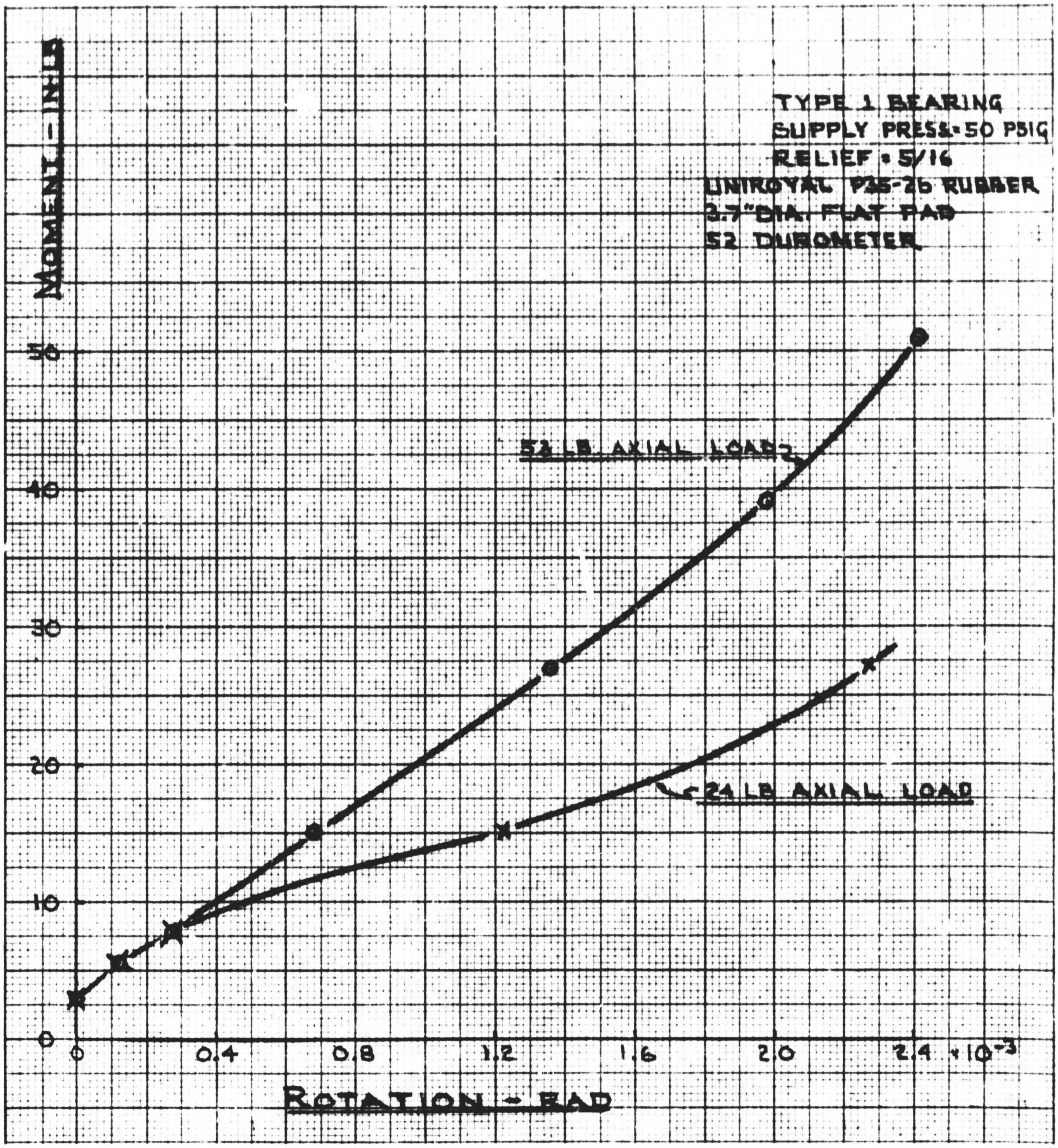


Fig. 15. Relation Between Moment Loading and Pad Rotation

When the fully relieved pad was rotated about the axis perpendicular to the plane of the orifice plate, the immediate response was an open circuit which gradually returned to the original "partial contact" response (two or three seconds). Also, the flow first decreased and then returned to normal. This variation occurred because in the one instance the orifice discharge was into a localized, pressure formed depression. In the next, the orifice discharged against a relatively undeformed surface. After several seconds, the depression reformed and the flow was re-established. This flow variation could be used to gage the compliance and compression set of the rubber.

The observations listed above were recorded during tests using the 52 Durometer conducting rubber, Uniroyal P35-2b. The magnitude and gas traping character of the lip would no doubt diminish with increasing Durometer since this feature is not present in rigid surface pads.

EXPERIMENTAL TESTS WITH SPHERICAL PADS

General

Test of spherical compliant surface pads were conducted for the following reasons:

- to demonstrate experimentally the ability of a compliant surface gas bearing to support a 40" diameter, turned metal sphere,
- to prove the spherical pad design extrapolations made during the flat pad test,
- to demonstrate the ability of a compliant pad to support a design load of 690 lbs., stably and with reasonable gas flow,
- to determine the response of a compliant when traversing the slot formed by joining two hemispheres,
- to determine the effects of relative velocity between the compliant and rigid surfaces.

Tests were conducted with 40 and 50 Durometer silicone rubber. These tests consisted of a dead weight or hydraulic jack load applied to a single pad while measuring pad rise and gas flow for a several supply pressures.

Fig. 16 shows the general arrangement of the spherical test pad. Pad variables investigated were the orifice size, the rubber hardness, the relief and the alignment joint. Two types of spherical surfaces were studied. One was solid; one could be parted, shimmed and re-bolted resulting in a slot between the joined pieces.

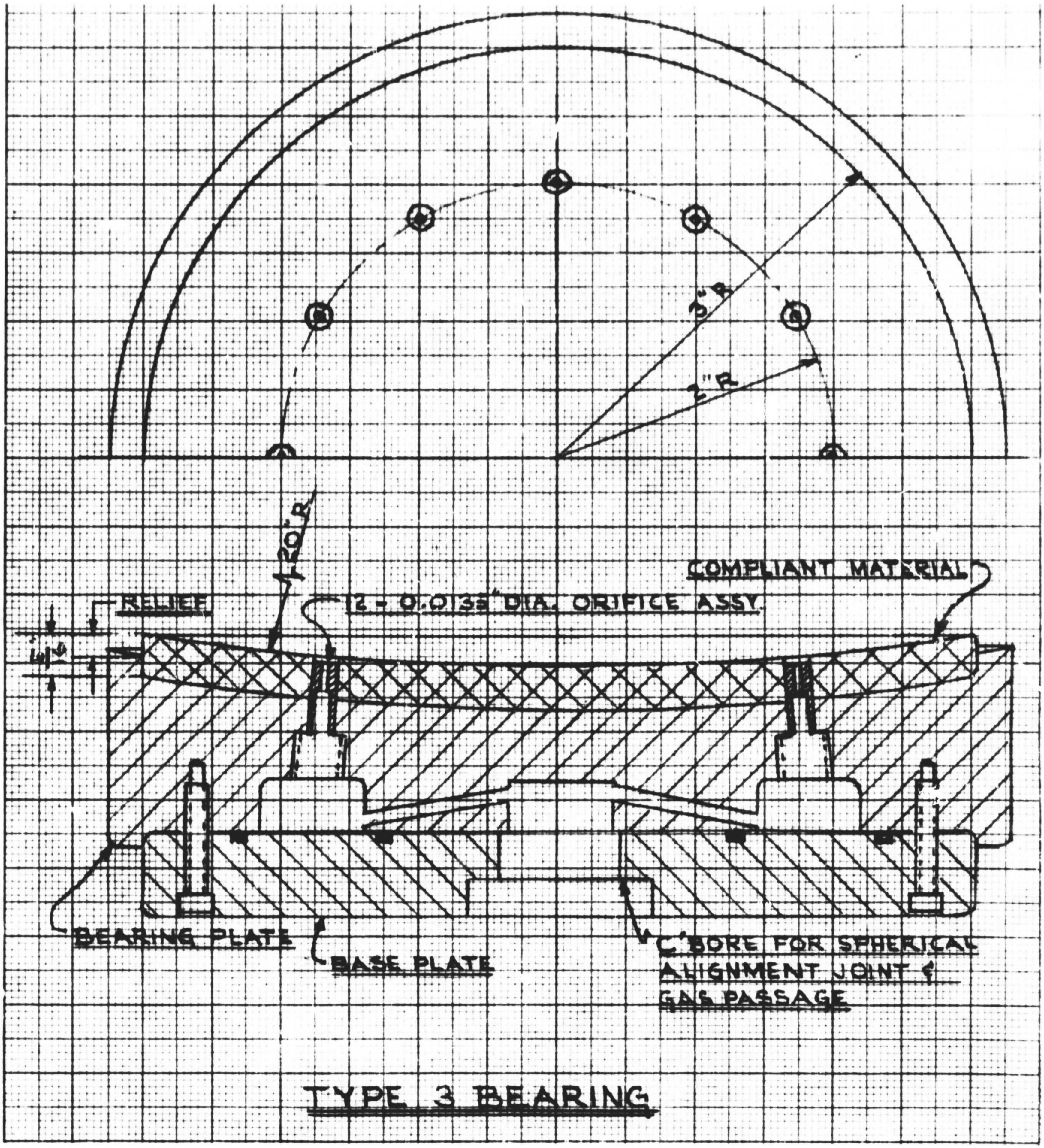


Fig. 16. General Arrangement of Compliant Surface Spherical Pad

Description of Test Facility

Fig. 17 shows a 6" diameter test pad bolted to the surface of a bed plate. In the left foreground is a brass spherical backing-plate blank; in the right foreground is a solid, 160 lb. segment of a 40" diameter steel sphere with a 14" cord.

Fig. 18 shows the spherical segment mounted on the compliant test pad. Two of the three indicators used to measure the rise of the spheres relative to the pad backing-plate are also visible. Bench test were accomplished by placing up to 900 lb. in weights on the flat face of the segment.

Rotational break-away torque was measured by applying a moment about the vertical axis of the test pad. Transverse break-away torque was determined by placing weights on an edge of the segment, plotting a moment-rotation curve and noting the value of the moment at the "knee" of the curve.

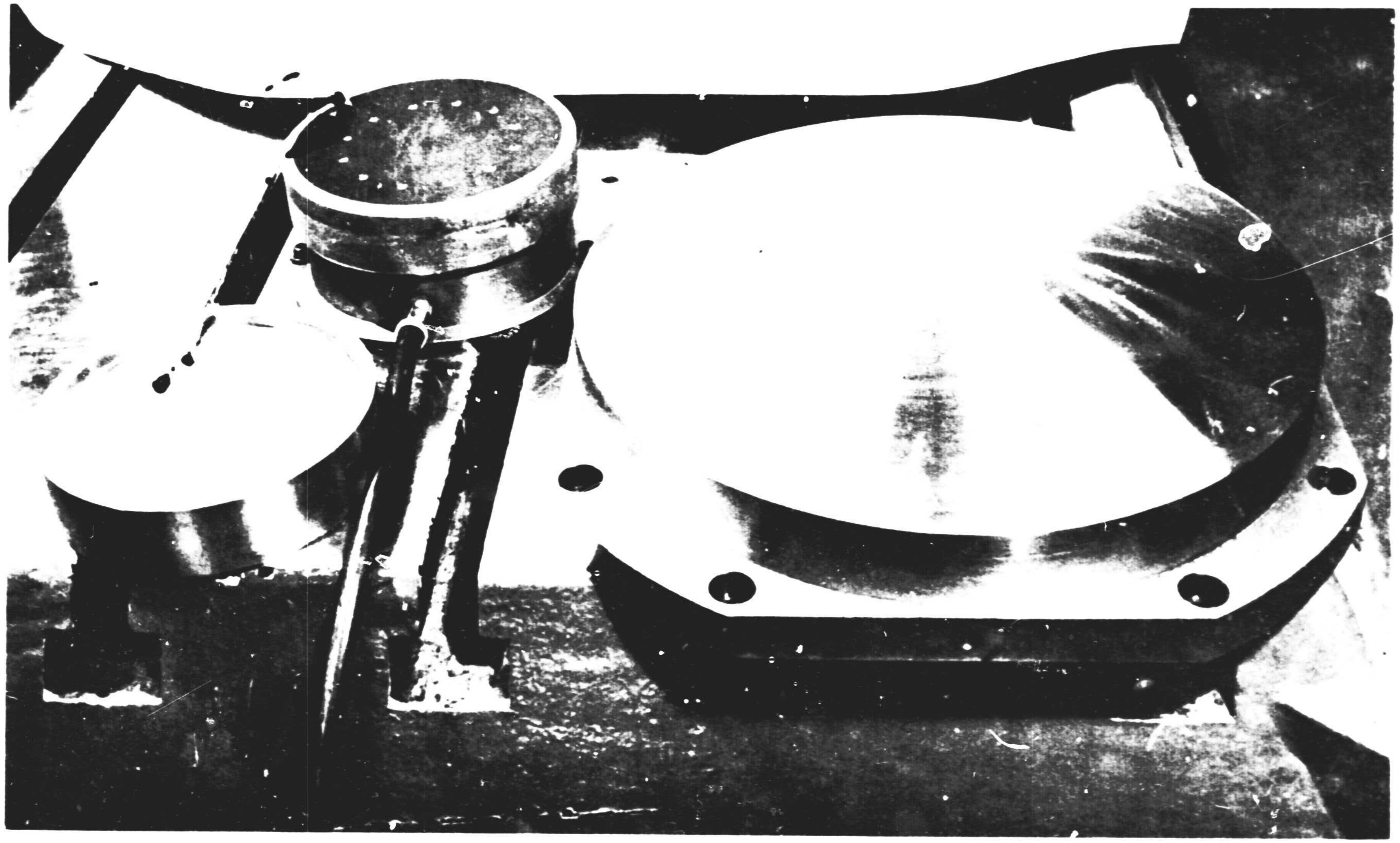
Fig. 19 shows a close-up of the compliant test pad mounted in the Sphere Simulator Rig A two-piece spherical segment, capable of simulating the anticipated hemispherical parting slot, is bolted to a pivot arm. The compliant pad, spherical joint, gas fitting connection, load cell and a 12" diameter thrust plate are mounted on the segment. Above the thrust plate is a 12" diameter gas lubricated isolator bearing and a hydraulic load piston.

Fig. 20 shows the general arrangement of the Sphere Simulator Rig. The pivot arm, a short length of beam welded to upper and lower mounting plates, is fixed to a shaft whose bearings are supported by the bed plate. The center of curvature of the spherical segment lies on the shaft axis. Thus, a rotation of the pivot arm simulates the rotation of a 40" diameter sphere. Relative motion between the pad and the spherical segment is accomplished manually. In operation, the test pad assembly is floated on one end by a compliant surface spherical gas bearing and on the other by a 12" diameter gas lubricated flat thrust bearing.

The instrument shown to the right in Fig. 20 is an amplifier-recorder for the load cell strain gage output.

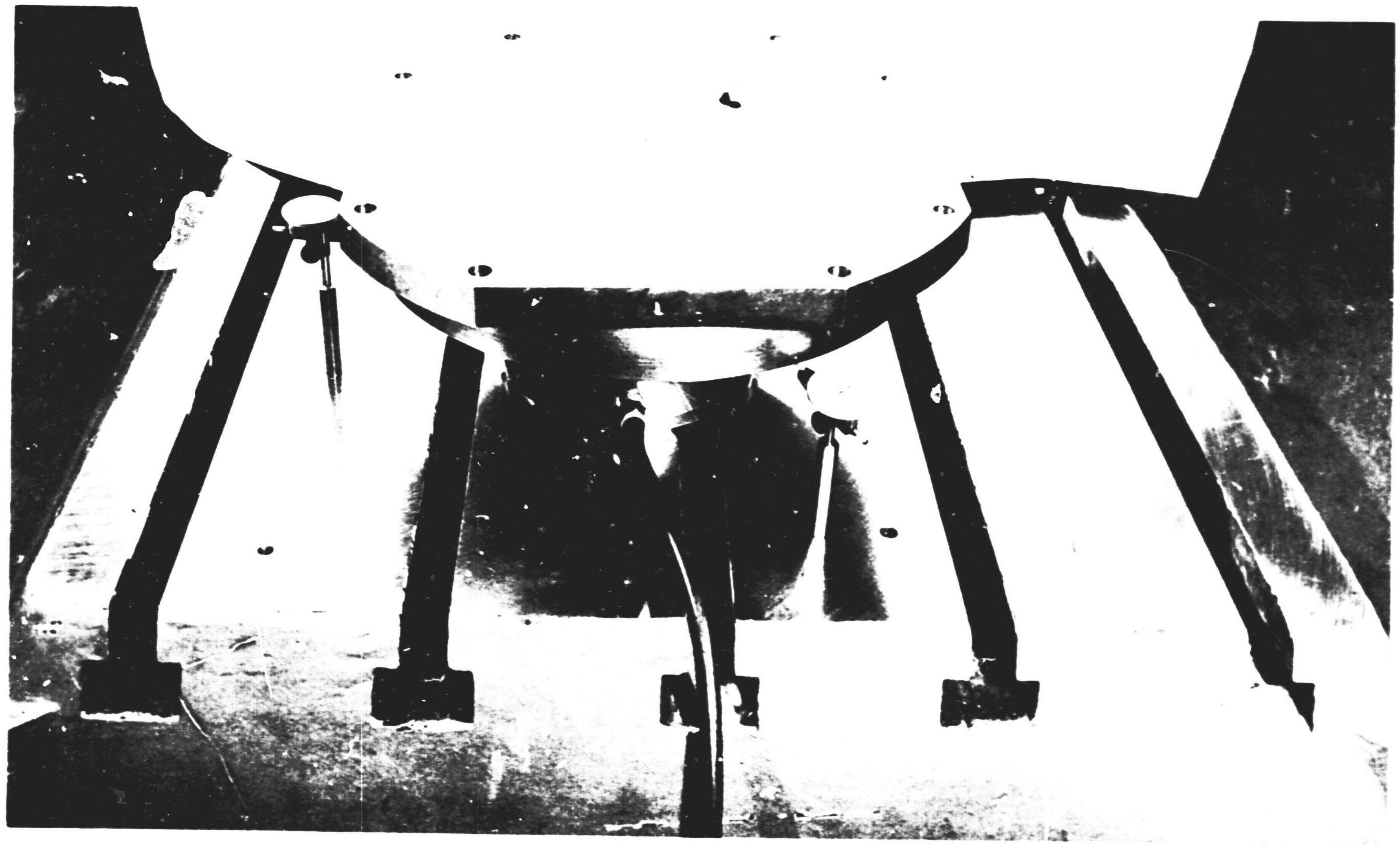
Test Results with 40 Durometer Silicone Rubber

Performance characteristics.--Initial tests were made with "orifices" consisting of a short length of 0.010" ID tubing. The gas flow was found to be insufficient to provide complete separation between the interior of the compliant pad and the metal spherical segment. When the orifice was opened with a #80 drill (0.0135" diameter) the pad performance was satisfactory.



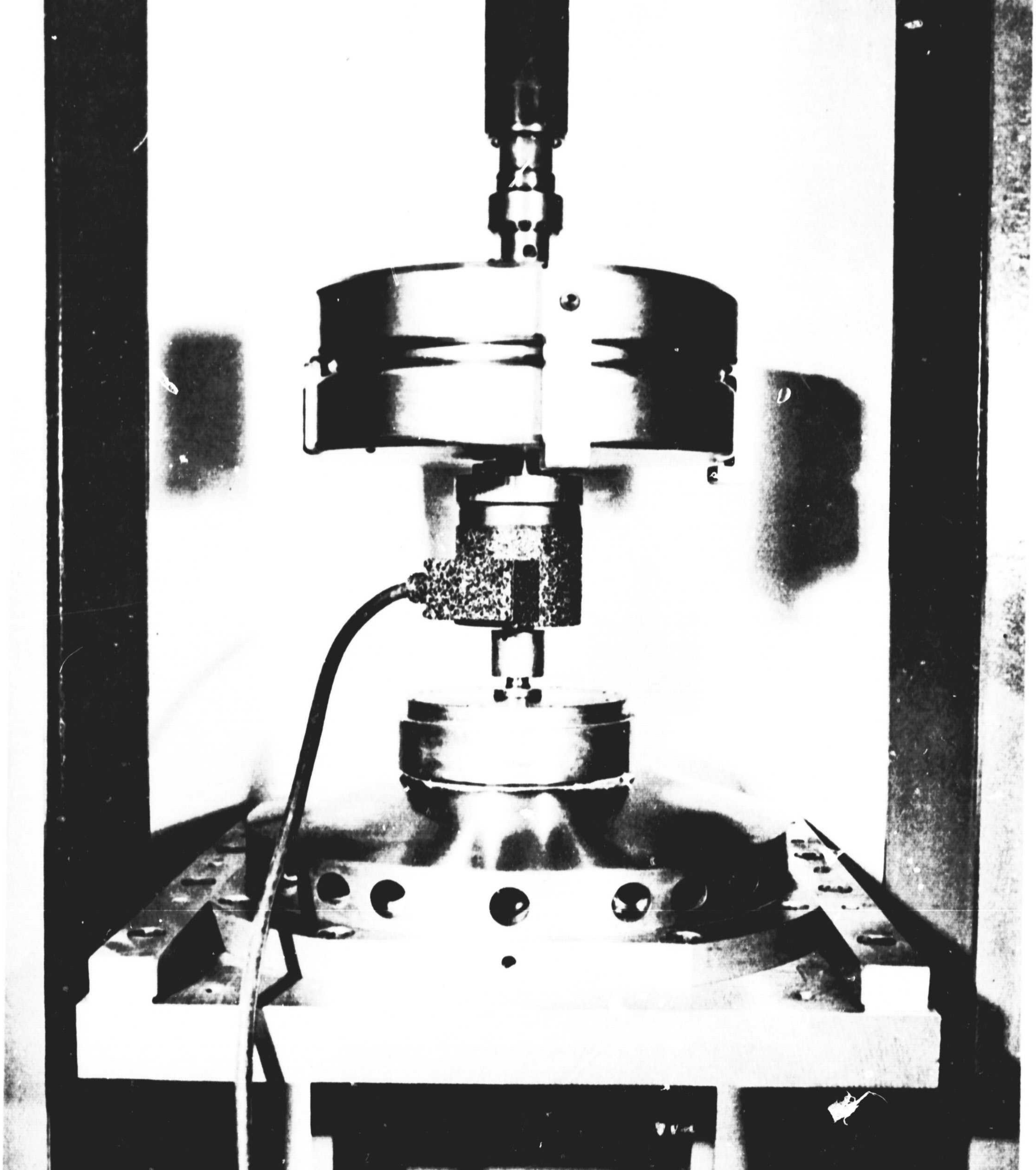
2062-1

Fig. 17. Spherical Compliant Test Pad, Backing-Plate and a Solid Steel Segment of a 40" Diameter Sphere



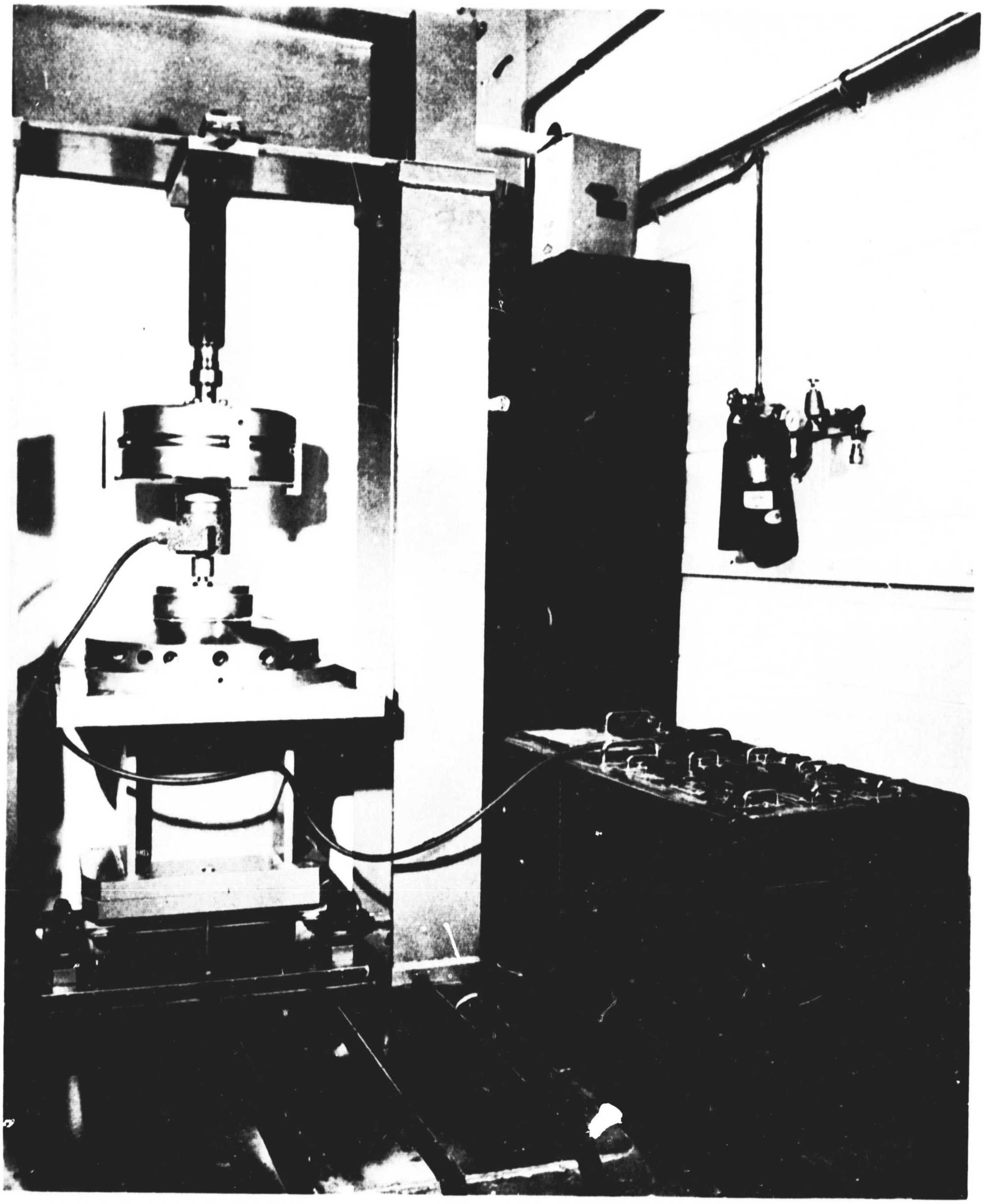
2062-2

Fig. 18. Test-Up for Dead Weight Loading of Spherical Complaint Pad



2062-3

Fig. 19. Close-up of Complaint Test Pad Mounted in Sphere Simulator Rig



2062-5

Fig. 20. General Arrangement of the Sphere Simulator Rig

Fig. 21 shows the pressure-flow-rise relation at break-away. Breakaway was determined when the spherical segment rotated freely under the influence of an 8 in-lb. moment. "Rise" is the separation between the metal spherical segment and the pad backing-plate. Rise measures both the rubber compression and the gas film thickness. "Zero" rise was taken as the metal-to-metal separation under load and without gas pressure.

Fig. 22 shows the relation between pressure, flow and rise at the on-set of hammer. Hammer manifests itself as a violent oscillation of the entire test assembly, thus restricting the design range for the present application. Fig. 21 represents the lower limit for pad operation; Fig. 22 gives the upper limit.

Fig. 23 shows design performance characteristics for the 40 Durometer Silicone pad. The recommended supply pressure shown on this graph is a minimum (resulting in minimum flow); the pressure shown on the hammer on-set curve (Fig. 22) is the maximum. Raising the supply pressure above the recommended design value results in a slight decrease in the lip contact force and, consequently, in the pad operating torque. Simultaneously, the pad stiffness would be lowered.

Fig. 24 is a cross-plot of the previous data showing the relation between load and rise for three different supply pressures. These curves terminate on the lower end because of hammer on-set and on the upper end because of surface contact. Also shown in Fig. 24 is a table, taken from the load-rise slope, giving the influence of supply pressure on stiffness at a constant load of 600 lb. Note, if the supply pressure is increased, the stiffness decreases.

Dynamic testing.--The following observations were made during testing in the Sphere Simulator Rig:

- Although the spherical segment showed a preferred direction of rotation under all test loads when rotated by hand, the motion did not persist. The lip resistance of the pads tested exceeded any force on the spherical surface that may have been present due to asymmetric gas flow. Thus, the self-sustained motion often associated with this surface force or "turbine torque" was not observed.
- When the motion of the sphere was transverse to the axis of symmetry of the pad, the drag between the sphere and the pad was much higher than when the rotation of the sphere was about the axis of symmetry of the pad. The action is the same as was observed with the flat pad test, namely, the lip of the pad lifts off with rotation; a transverse motion causes half of the lip to turn in and the other half to turn out.
- Pad righting moment was required to prevent excessive edge drag when the direction of sphere motion was reversed.
- Commercially available spherical alignment joints were suitable for the present application. The supply gas was readily fed

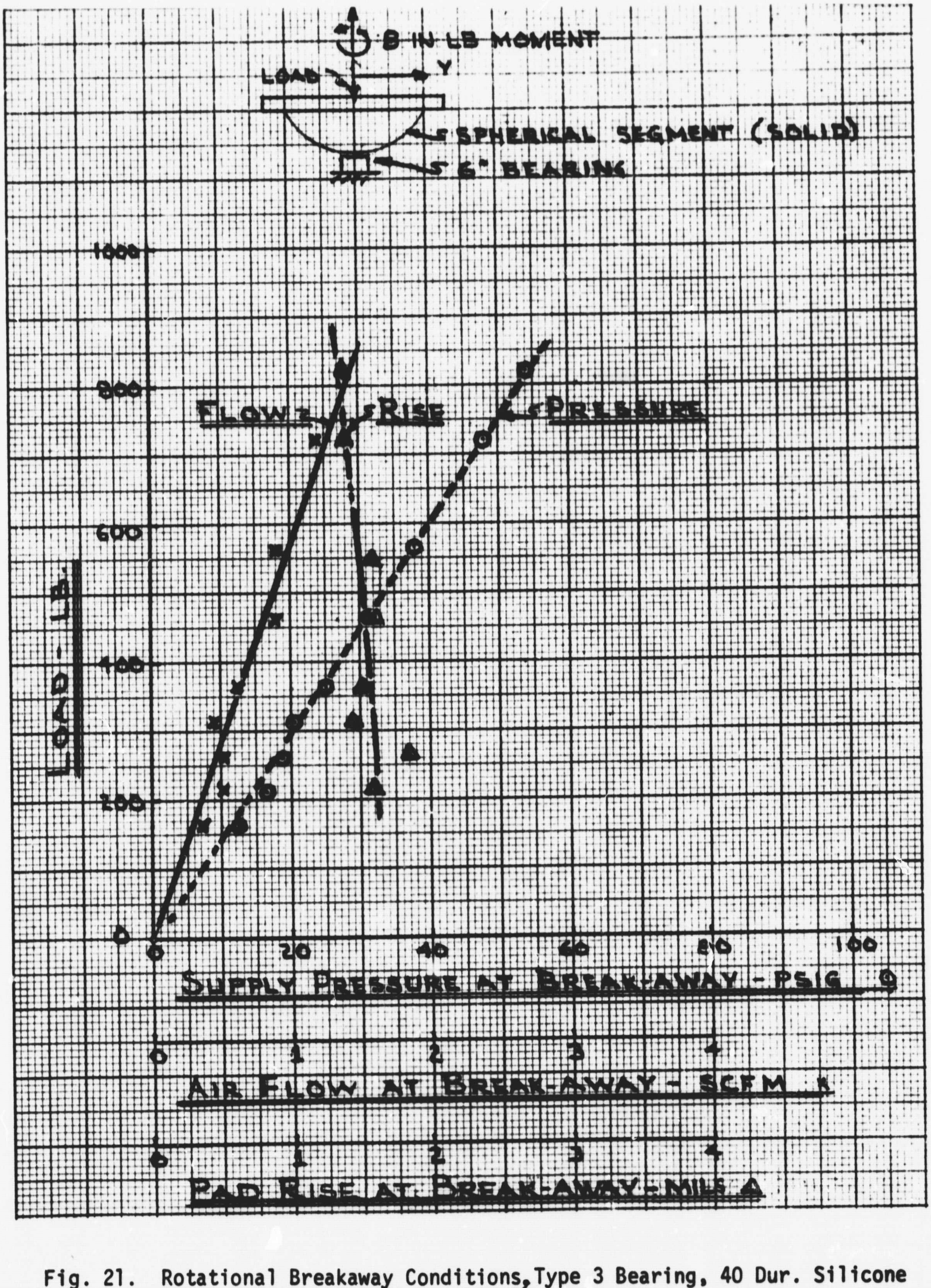


Fig. 21. Rotational Breakaway Conditions, Type 3 Bearing, 40 Dur. Silicone Rubber, 5/16 Thick, 5/32 Relief

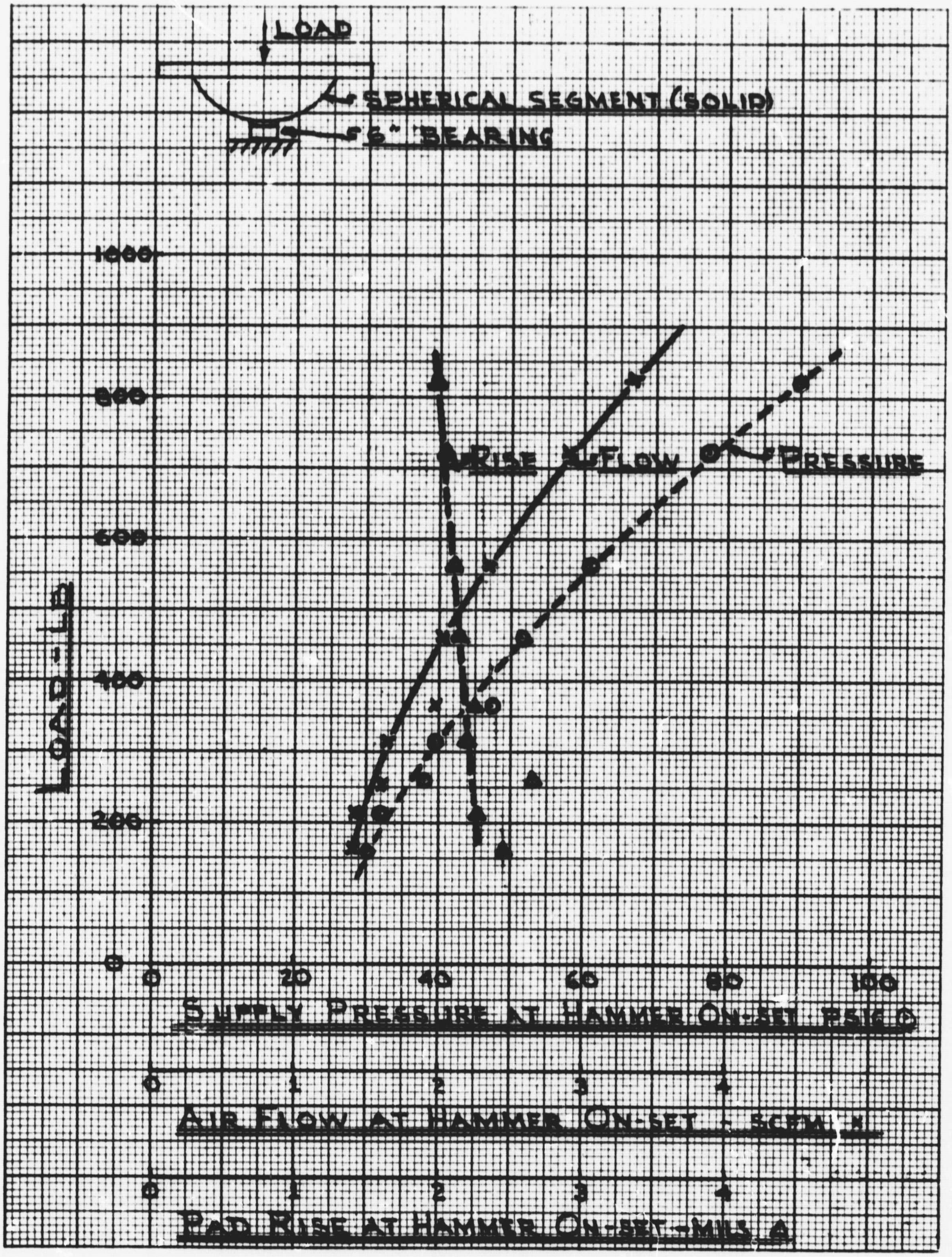


Fig. 22. Hammer On-Set Conditions, Type 3 Bearing, 40 Dur. Silicone Rubber, 5/16 Thick, 5/32 Relief

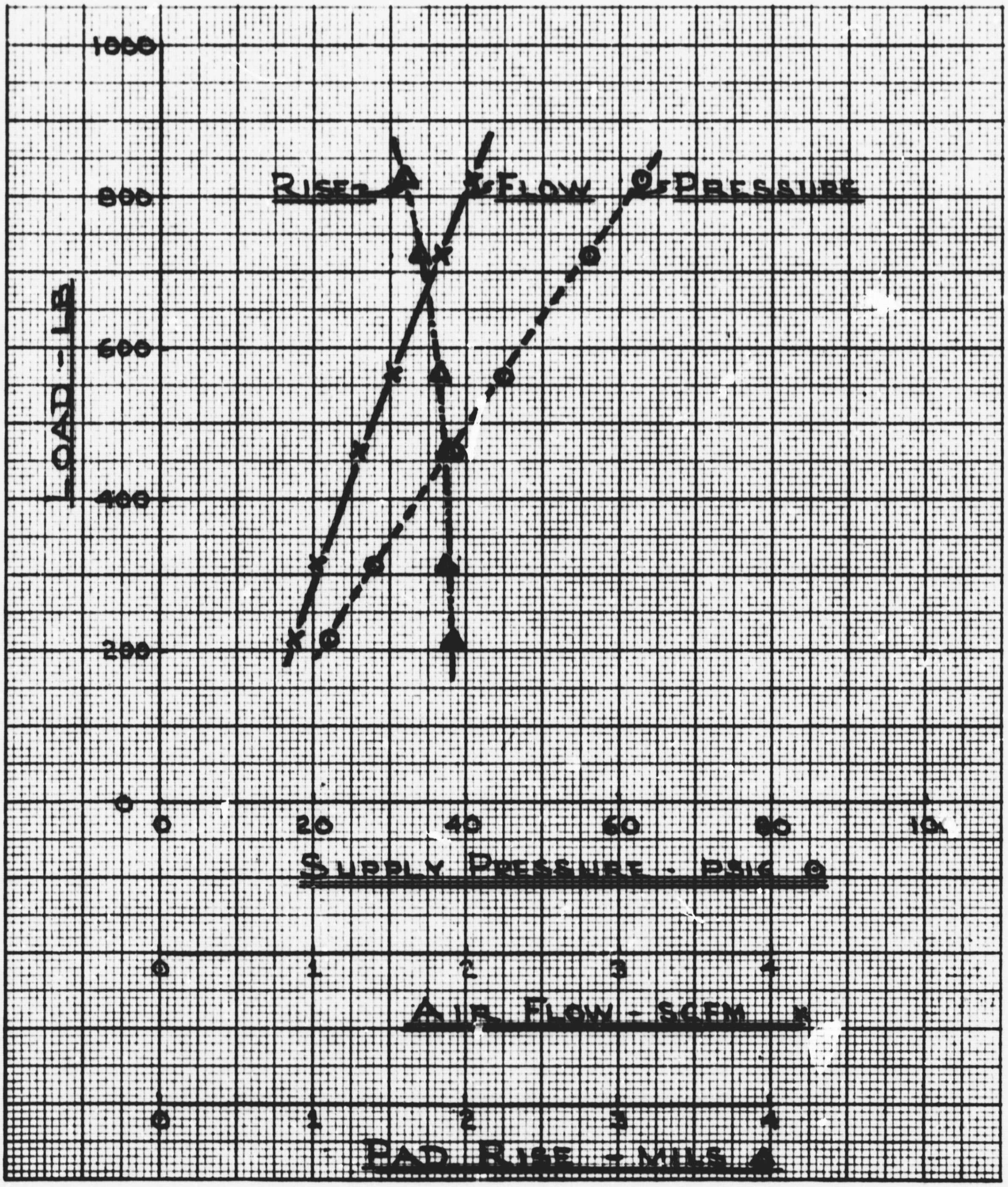


Fig. 23. Design Performance Characteristics, Type 3 Bearing, 40 Dur. Silicone Rubber, 5/16 Thick, 5/32 Relief

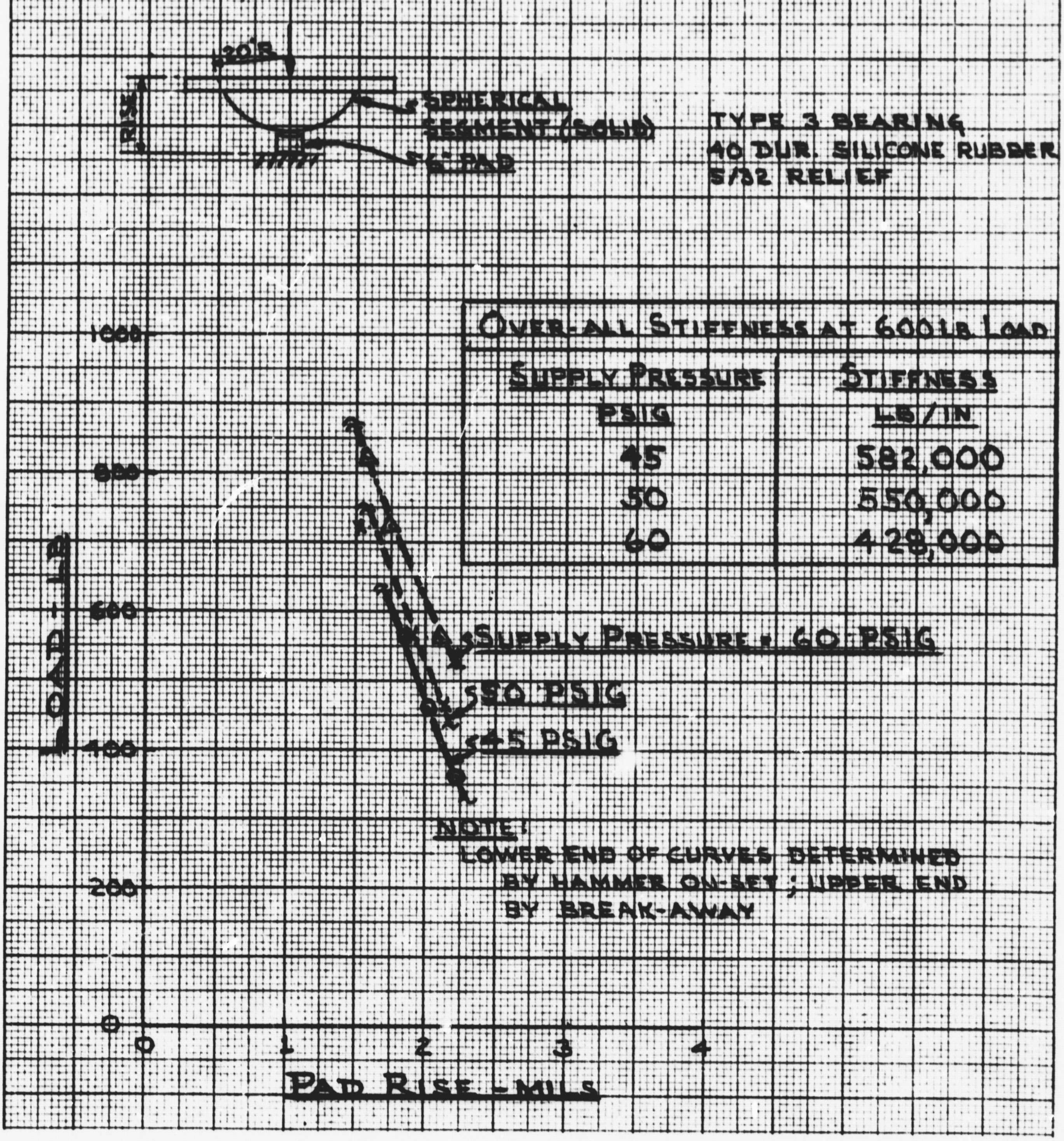


Fig. 24. Influence of Supply Pressure on Pad Rise and Stiffness

through the bore in the spherical joint and the righting moment of the pad was sufficient.

Dynamic testing with slotted segment.--Fig. 25 shows the results of testing with a split spherical segment. The split was formed by a standard shim (12" long by 1/2" wide) placed between the two halves of the spherical segment. In manufacture, the 20" radius segment was turned from two steel plates bolted firmly together through a tongue-and-groove joint. Thus, the shim interrupts the constant radius of curvature of the "spherical" segment.

Two tests were run, one with a 0.0025" shim and one with a 0.001" shim. The test with the 0.0025" shim indicated the following:

- no preferred direction of rotation,
- no pneumatic hammer in the supply pressure-load range investigated,
- high gas flow,
- low load capacity,
- very poor righting moment,
- a usable design load range between 150 and 200 lbs.
- The principal conclusion was that two hemispheres with a 0.0025" or larger parting slot *cannot* satisfactorily be supported by a 40 Durometer compliant pad.

Tests with a 0.001" shim indicated the following:

- the general performance of the bearing was considerably better than with the 0.0025" slot,
- pneumatic hammer was encountered,
- as the load increased, the separation between the minimum design supply pressure and hammer on-set pressure decreased. For example, at a load of 520 lb. the supply pressures were 80 and 88.5 psig respectively,
- minimum design pressure was determined when the parting slot caused asymmetry in the pad pressure distribution to such an extent that the low-pressure section of the pad collapsed and made contact with the spherical segment.
- Again, the principal conclusion was two hemispheres with a 0.001" parting slot *cannot* satisfactorily be supported by a 40" Durometer compliant pad.

General conclusions.--Both the static and dynamic tests indicated the 40 Durometer silicone rubber was *too soft* for the present application.

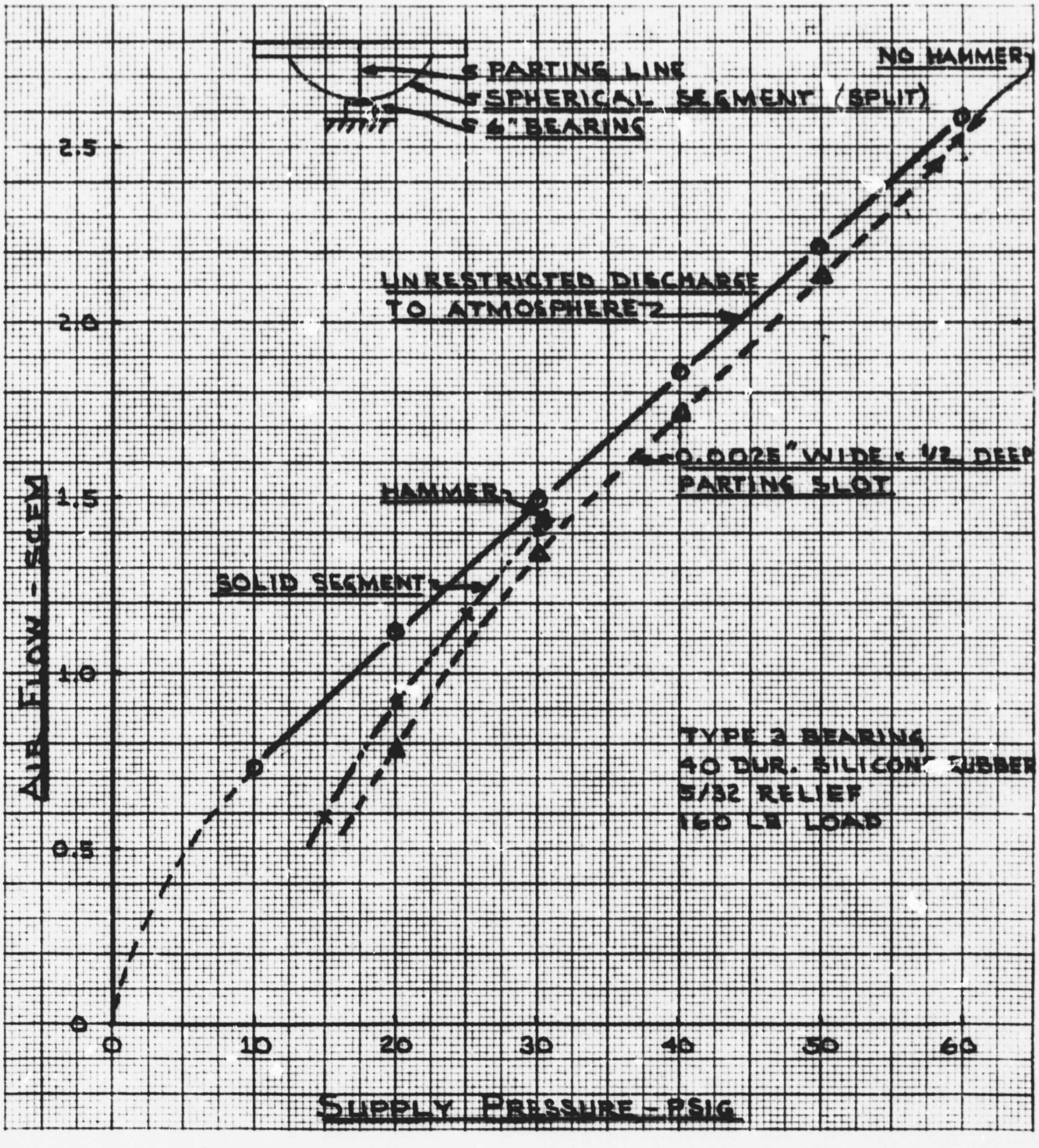


Fig. 25. Influence of Parting Line Slot on Pressure-Flow Relation

A good design load limit for the 40 Durometer pad was about 700 lb. The flow shut-off load was 968 lb.

The only way a 40 Durometer pad with a 5/32" relief can be used in combination with a slotted surface is to fill in the slot. This can be done by coating the edges of the hemispheres with silicone "O" ring compound, such as Silastic, before joining. A continuous surface can be attained after joining by trimming the flashing.

Test Results with 50 Durometer Silicone Rubber

Performance characteristics.--Fig. 26 and 27 shows the pressure-flow-rise relation to load at rotational breakaway and hammer on-set. Performance at the recommended design condition as shown in Fig. 2. Note the relative independence of hammer on-set from supply pressure for loads greater than about 500 lb. This pad exhibited the best performance characteristics of the three tested and is the configuration referred to in the Design Summary of this report.

Dynamic testing.--Similar to the previous dynamic tests, the turbine torque magnitude was less than the lip drag and the spherical segment, resting on the test pad, showed no tendency to rotate.

The righting moment capability of the 50 Durometer pad was higher than the 40 Durometer and sphere oscillation was more readily accommodated.

Dynamic testing with slotted segment.--A series of tests run with a 0.001" thick shim indicated no substantial differences between these tests and test run without the shim. This is contrary to the test results previously discussed for the 40 Durometer rubber. Thus, a *parting slot of at least 0.001" is a satisfactory design condition* of the 50 Durometer Silicone pad. Fig. 6 is typical of these results.

Reduction of relief to 5/64".--The bearing surface of the 50 Durometer pad was ground back 5/64" (half the original relief of 5/32). The principal differences with the 5/32" relief data were:

- a higher load capacity (up to 1000 lb.),
- a return of hammer on-set,
- more of the character of a rigid surface bearing.

Performance Comparison of Spherical Pads Tested

Table 1 shows a comparison of the test results for the three spherical pads. Typically, the designation 40-5/16-5/32 means 40 Durometer silicone rubber, 5/16" thick with 5/32" relief. The following

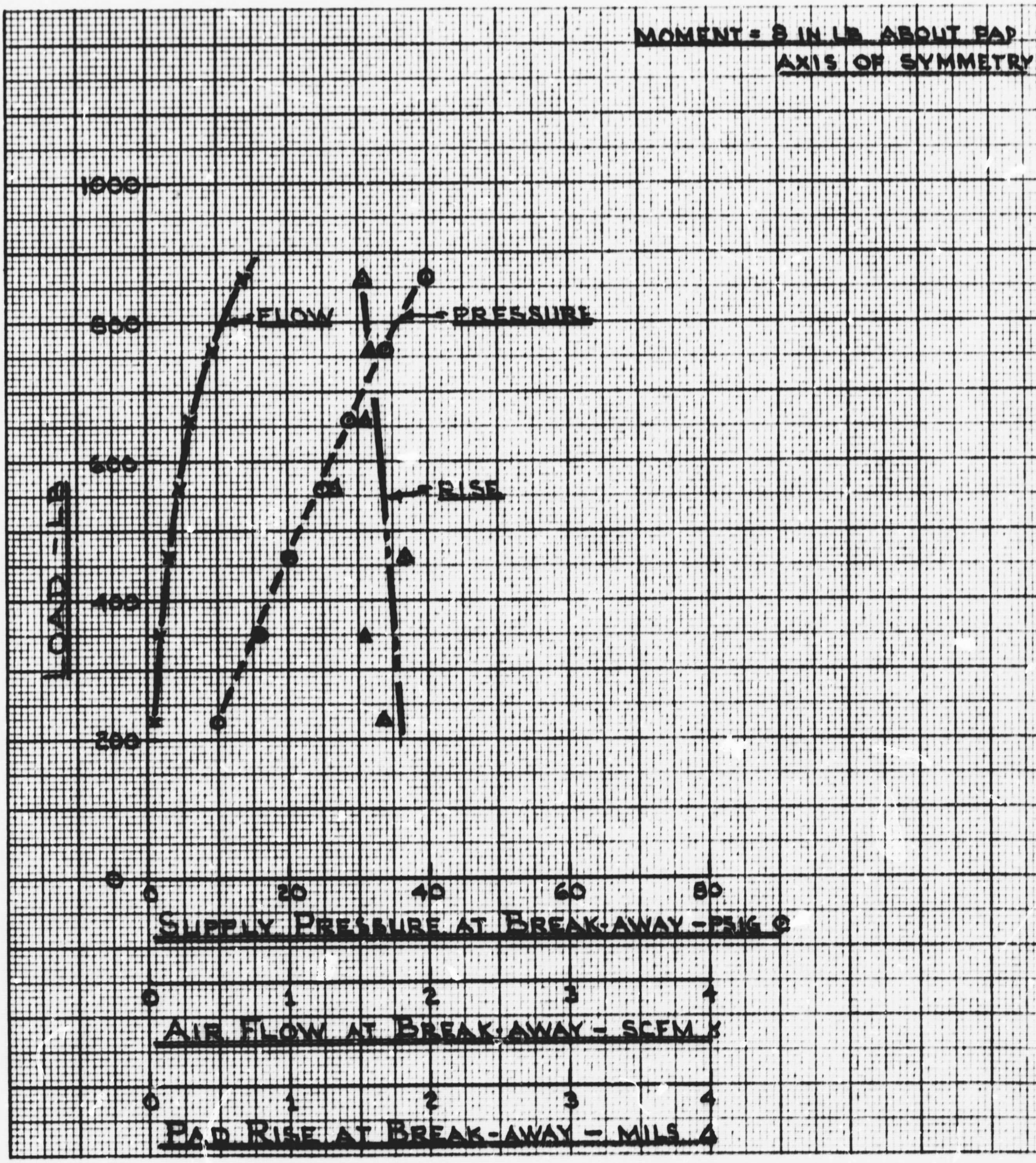


Fig. 26. Rotational Break-Away Conditions, Type 3 Bearing, 50 Dur. Silicone Rubber, 5/16 Thick, 5/32 Relief

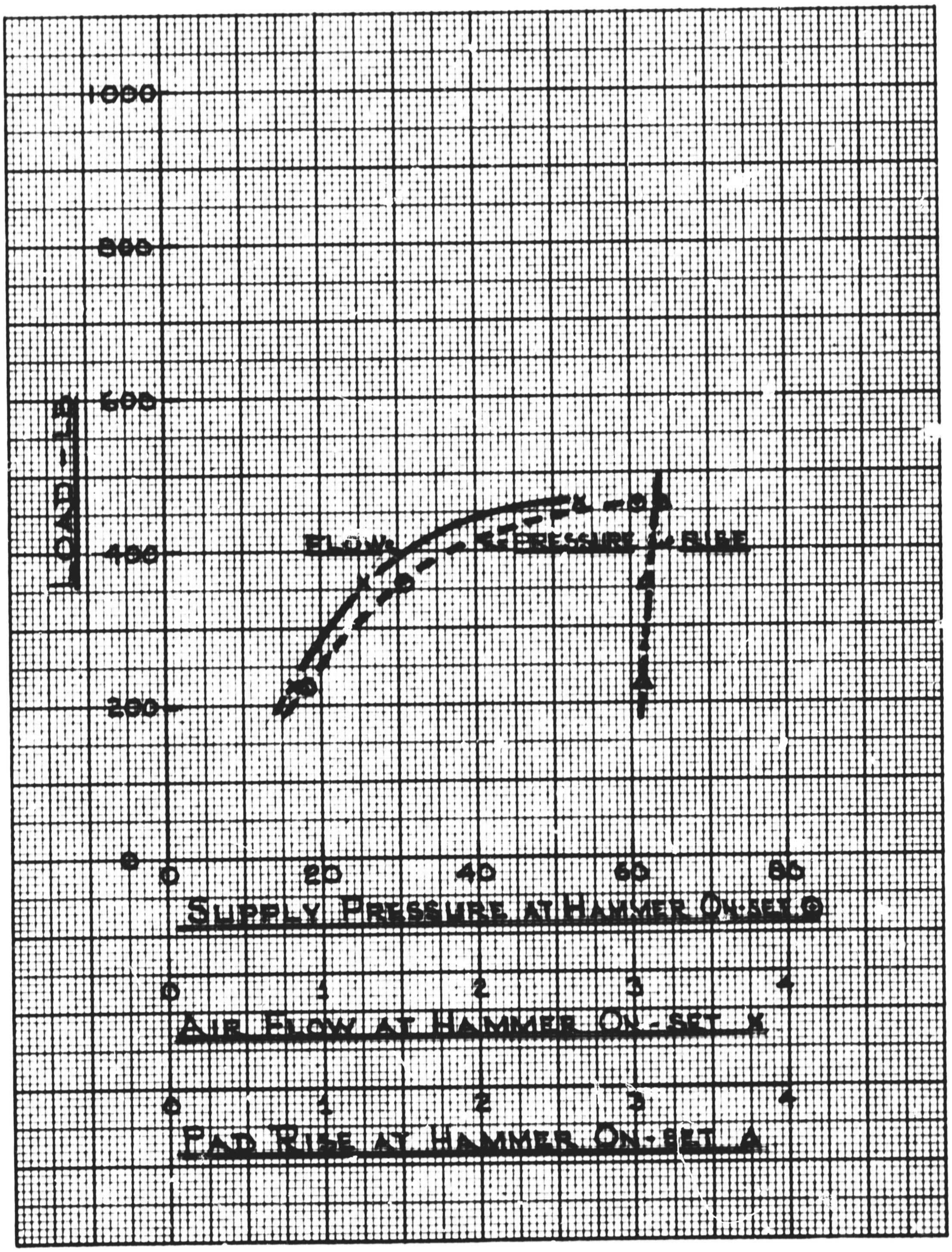


Fig. 27. Hammer On-Set Conditions Type 3 Bearing, 50 Dur. Silicone Rubber, 5/16 Thick, 5/32 Relief

TABLE I
COMPARISON OF EXPERIMENTAL DATA FOR COMPLIANT SPHERICAL TEST PADS AT THE DESIGN LOAD OF 680 LB

Pad Type	Rotational Breakaway			Hammer On-Set			Design Point		
	$40-\frac{5}{16}-\frac{5}{32}$	$50-\frac{5}{16}-\frac{5}{32}$	$50-\frac{15}{64}-\frac{5}{64}$	$40-\frac{5}{16}-\frac{5}{32}$	$50-\frac{5}{16}-\frac{5}{32}$	$50-\frac{15}{64}-\frac{5}{64}$	$40-\frac{5}{16}-\frac{5}{32}$	$50-\frac{5}{16}-\frac{5}{32}$	$50-\frac{15}{64}-\frac{5}{64}$
Supply Pressure psig	45	31	27	74	None	36	53	55	33
Air Flow scmf	1.20	0.40	0.30	2.75	None	1.15	1.74	2.2	0.77
Rise mils	1.42	1.58	1.46	2.05	None	2.20	1.75	2.58	2.06

observations can be made concerning the comparison of the selected pad (50-5/16-5/32) to the others at the design load:

- up to a supply pressure of 120 psig the selected pad did not hammer,
- the selected pad has a moderate lift-off supply pressure and a moderate design pressure,
- the recommended design pressure results in a large rise allowing for loose sphere tolerances,
- the selected pad showed good local compliance,
- the 50-15/64-5/64 pad reacted more like a rigid surface bearing than a compliant bearing showing undesirable hammer sensitive to mounting misalignment,
- the 40-5/16-5/32 pad reacted poorly to a parting slot.

RECOMMENDATIONS

- Build the spherical test facility. From a bearing standpoint, the project should be successful.
- Investigate the use of glass as a sphere material. There are many advantages in being able to visually observe the test package. Spun hemispherical castings of the size required for the test facility are available and the technology for installing hand-holes and flanges exists.*
- Obtain more experimental data on compliant bearing, particularly in the following areas:

  Measure relation between rubber thickness and relief for hardnesses up to 70 Durometer.

*Hand-holes can be installed by cutting and grinding a conical plug to fit a mating conical hole in the glass sphere. The plug should have the same curvature as the sphere OD, approximately the same thickness as the sphere and the apex at the center of the sphere. The plug can be held in place by an "o" ring and by external pressure when the interior of the sphere is operated at sub-ambient pressure. Metal mounting rings and sealing flanges can be attached to the glass sphere without introducing stresses from differential thermal expansion by machining the outside surfaces of the rings undersize and filling the gap between the metal and the glass with a compliant structural epoxy.

Measure accurate clearance and pressure profiles for the pad shown in Fig. 1. The purpose of this data would be to provide an experimental basis for checking analytical predictions for a practical pad.

- Obtain an "exact" theoretical solution for the pad shown in Fig. 1. The non-linear Reynolds' equation should be used for the gas film. The compliant disc could be treated as a 3D shell using an incremental theory to describe the pressure response of the rubber as a sequence of small strain problems. Finite element implementation of this approach is currently being studied at FIRL
- Check the theory with experimental measurements.
- Determine an approximate theory for the coupled fluid film-elasticity problem that would reduce the computer solution time required for an "exact" solution while still providing reasonable agreement with the experimental results.

About half of the recommendations listed above are analytical and concerned with advancing the state of the art in compliant surface bearings. They are not necessarily directly related to the construction of the spherical test facility. They are, however, a natural extension of the work performed under this contract. The extension should be undertaken because the demonstrated usefulness of the compliant surface is extrapolated readily to a wide range of applications. Analysis must be used to guide, limit and complement the experimental work necessary for application development.

APPENDIX A

FLOWMETER CALIBRATION CURVE
AND
FLOW CHARACTERISTICS OF ORIFICE RING USED
IN FLAT PAD TESTS

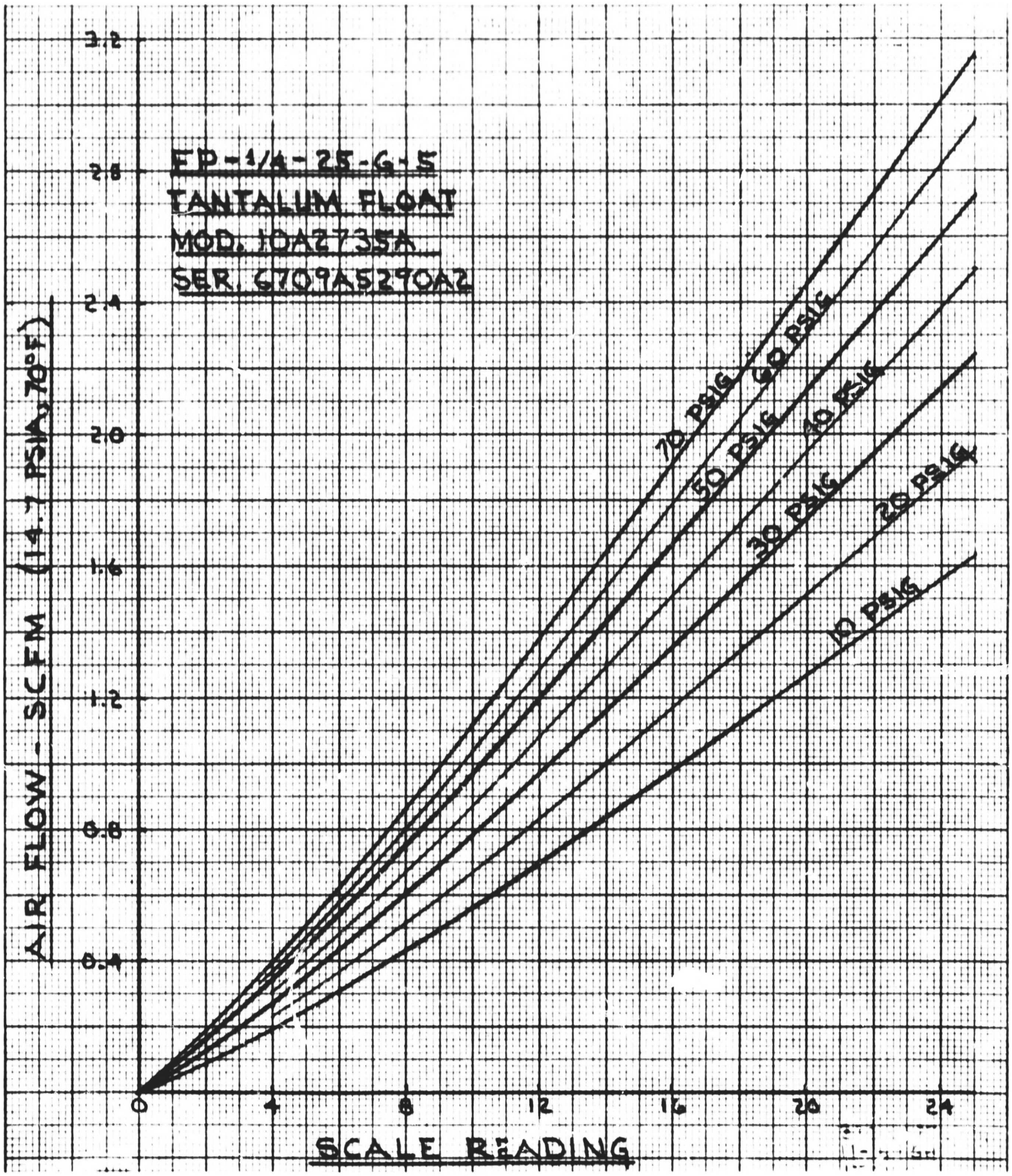


Fig. A1 - Calibration for Variable-Area Flowmeter

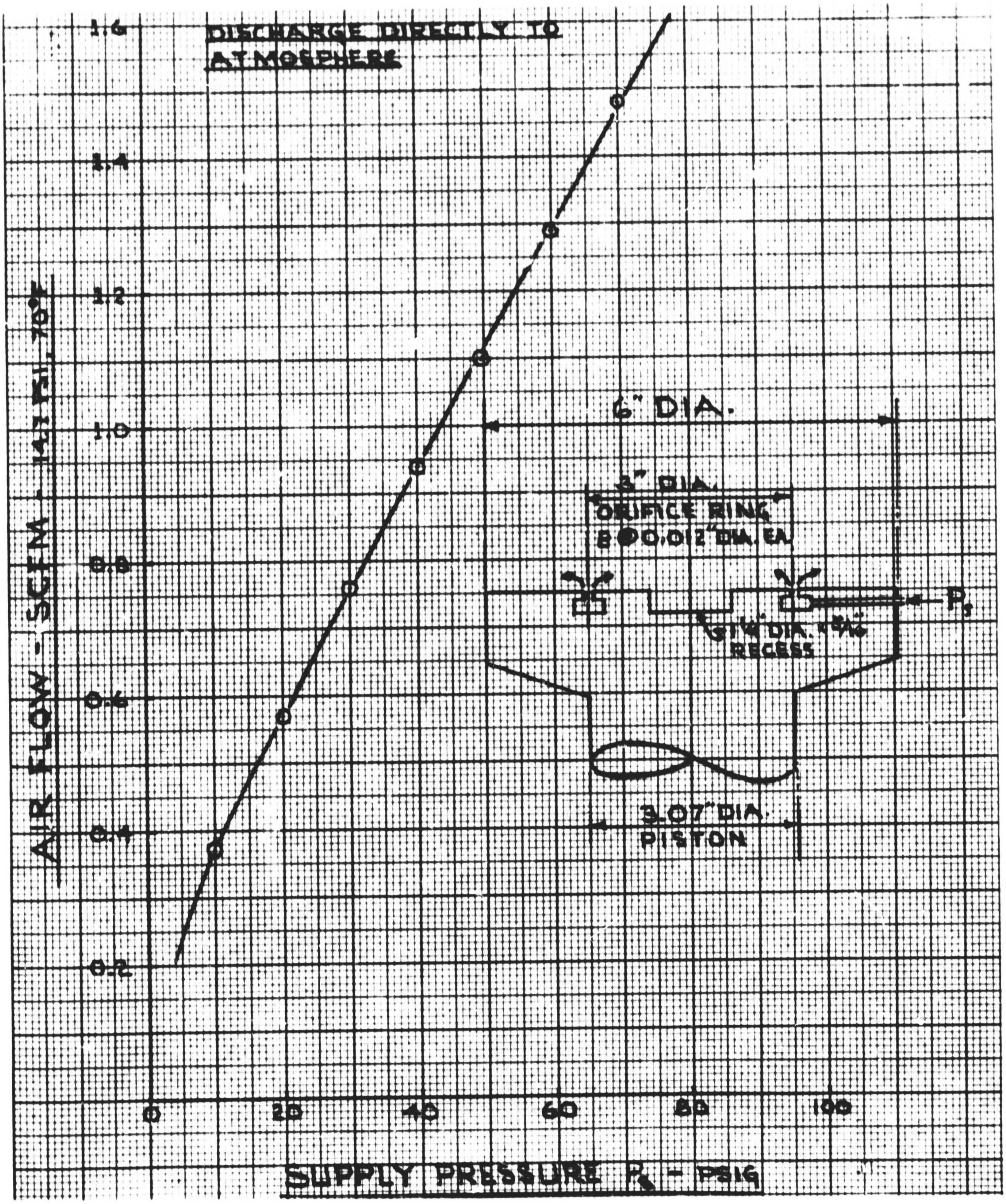


Fig. A2 - Flow Characteristic for SBTR Orifice Ring

APPENDIX B

TYPICAL CALCULATIONS OF SPHERE BEARING SUPPORT SYSTEM CHARACTERISTICS FROM PAD DATA

A typical set of points for Fig. 4, "Performance Characteristics for Compliant Surface Spherical Bearing Assembly with Preload", are computed as follows:

Let weight of sphe · = 4550 lb.
and the design preload = 1200 lb. (minimum recommended preload)

A radial line of the 40" dia. sphere that passes through the pad spherical alignment joint makes a 10.5° angle with the bearing mounting axis. The bearing mounting axis is also a radial line and is the center line of the hub shown of Fig. 3. The four main hub center lines are each 45° to a vertical line through the sphere.

The load per main bearing assembly, W_b, is given by:

$$W_b = \frac{(4550 + 1200) \text{ lb.}}{(4 \text{ bearing assemblies}) (\cos 45°)}$$

$$W_b = 2033 \text{ lb.}$$

The load per main bearing pad, W_p, is given by:

$$W_p = \frac{2033 \text{ lb.}}{(3 \text{ pads}) (\cos 10.5°)}$$

$$W_p = 689 \text{ lb.}$$

The load per preload pad, W'_p, is given by:

$$W'_p = \frac{1200 \text{ lb.}}{(3 \text{ pads}) (\cos 10.5°)}$$

$$W'_p = 407 \text{ lb.}$$

From Fig. 2, at a pad load of 689 lb.:

Flow per pad = 2.2 scfm

Supply pressure = 55 psig

Rise = 0.0026

Also, from Fig. 2, for a pad load of 407 lb.:

Flow per pad = 1.48 scfm

Supply pressure = 34 psig

The total rate of air flow, Q, is:

$$Q = (2.2 \text{ scfm/main pad}) (12 \text{ main pads}) + (1.48 \text{ scfm/preload pad}) (3 \text{ preload pads})$$

$$Q = 30.8 \text{ scfm}$$

The rise, R, of the main pads along the bearing mounting axis is:

$$R = 0.0026'' (\cos 10.5°)$$

$$R = 0.00256''$$

The vertical rise of the sphere, V, is given by:

$$V = R \cos 45° = (0.00256)\ (0.707)$$

$$\underline{V = 0.0018''}$$

The stiffness (for uniform pressure distribution) is 500,000 lb/in for both main and preload bearing pads. The bearing stiffness, k_b, is given by:

$$k_b = (500{,}000 \text{ lb/in per pad})\ (\cos^2 10.5°)\ (3 \text{ pads})$$

$$k_b = 1.45 \times 10^6 \text{ lb/in}$$

The vertical stiffness, k_v, is:

$$k_v = (1.45 \times 10^6 \text{ lb/in})\ (4 \text{ bearings})\ (\cos^2 45°) + 1.45 \times 10^6 \text{ lb/in preload stiffness}$$

$$k_v = (2.90 + 1.45) \times 10^6 \text{ lb/in}$$

$$k_v = 4.35 \times 10^6 \text{ lb/in}$$

The horizontal stiffness, k_h, is:

$$k_h = (1.45 \times 10^6 \text{ lb/in})\ (4 \text{ bearings})\ (\sin^2 45) + (500{,}000 \text{ lb/in})\ (3 \text{ pads})\ (\sin^2 10.5°)$$

$$k_h = (2.90 + 0.05) \times 10^6 \text{ lb/in}$$

$$k_h = 2.95 \times 10^6 \text{ lb/in}$$

The critical frequency in the vertical direction, F_v, is:

$$F_v = 1/2\pi \sqrt{\frac{k_v}{M}}$$

$$F_v = 1/2\pi \left[\frac{4.35 \times 10^6 \text{ lb/in}}{4550 \text{ lb}/(386 \text{ in/sec}^2)}\right]^{1/2}$$

$$\underline{F_v = 96.7 \text{ cps}}$$

The critical frequency in the horizontal direction, F_h, is:

$$F_h = 1/2\pi \sqrt{\frac{k_h}{M}}$$

$$F_h = 1/2\pi \left[\frac{2.95 \times 10^6 \text{ lb/in}}{4550 \text{ lb}/(386 \text{ in/sec}^2)}\right]^{1/2}$$

$$\underline{F_h = 79.7 \text{ cps}}$$

The quantities underlined above can read from Fig. 4 for Weight of Sphere = 4550 lb.

Fig. 4 was computed by repeating the above calculations for other sphere weights keeping the 1200 lb. preload constant.

Fig. 5 also repeats the above calculations except the preload is taken equal to zero.

2