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TECHNOLOGY UTILIZATION REPORT

Technology Utilization Division

ELASTIC ORIFICES FOR GAS BEARINGS

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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ELASTIC ORIFICES FOR GAS BEARINGS

Prepared under contract for NASA by

STANFORD RESEARCH INSTITUTE
Menlo Park, California

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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NOTICE

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FOREWORD

The Administrator of the National Aeronautics and Space Administration has established a technology utilization program for "the rapid dissemination of information * * * on technological developments * * * which appear to be useful for general industrial application." From a variety of sources, including NASA Research Centers and NASA contractors, space-related technology is collected and screened; and that which has potential industrial use is made generally available. Information from the Nation's space program is thus made available to American industry, including the latest developments in materials, processes, products, techniques, management systems, and analytical and design procedures.

This report describes a device recently developed and its use in industrial applications.

The Director, Technology Utilization Division,
National Aeronautics and Space Administration.

CONTENTS

SUMMARY

This report describes an examination of the merits of an elastic orifice in externally pressurized gas bearings. The use of an elastic orifice in a test bearing demonstrated a threefold increase in bearing stiffness over a conventional fixed orifice when the two were compared at a common operating point as defined in this report. This increase should be welcomed wherever externally pressurized bearings are used. In addition, increased bearing stiffness may allow the use of these bearings in some applications formerly restricted to pressurized liquid and rolling contact bearings. Information is included to allow construction of working models of these orifices and duplication of experimental results.

INTRODUCTION

The development of the elastic orifice as a means of flow control may revolutionize the field of bearing design. Results obtained in preliminary experiments indicate that the elastic orifice may permit extensive use of gas-lubricated bearings in applications formerly restricted to pressurized liquid and rolling contact bearings.

The requirements of modern technology have resulted in many improvements over the simple, splash-fed hydrodynamic fluid bearing of the Model T Ford era. The fluid bearing was improved by using a pressurized supply to force the lubricant into the space between the bearing surface and the load.

However, the development of rolling contact bearings (such as ball and roller) greatly reduced the use of fluid bearings. Rolling contact bearings were preferred because of their lower cost and greater stiffness. (Stiffness, with respect to a bearing, is a measure of displacement of shaft with increased load.) Modern technology, however, demands performance characteristics that are difficult to achieve with rolling contact bearings. Liquid bearings are inherently capable of smoother operation, but they exhibit a high frictional resistance at high speeds and require lubricants that can withstand severe environments. Therefore, attention has been focused on fluid bearings using gas as the lubricant.

Gas-lubricated bearings can be operated at much higher speeds with extremely low friction, but they are characterized by low stiffness. Pressurized gas bearings permit low-friction stopping and starting but require a pressure supply.

In 1960, John H. Laub and Frank Batsch, of the California Institute of Technology's Jet Propulsion Laboratory, reported experiments with elastic orifices as a means of controlling fluid flow to a pressurized gas bearing. In summary, the innovators' work appeared to be a straightforward means of improving the operation of pressurized gas bearings by increasing the stiffness, thereby improving the competitive position of gas bearings with other bearing types. Subsequent investigation at Stanford Research Institute demonstrated the dependence of performance characteristics on elastic orifice geometry, especially orifice diameter and thickness of the diaphragm.

Externally pressurized (hydrostatic) gas bearings employ some form of flow-restricting element to control the flow of gas to a pressurized cavity which supports the load. Restricting the gas flow controls the bearing displacement under load and, thus, the relative stiffness of the bearing. A detailed analysis of flow control in gas-lubricated bearings is presented in appendix A.

SYMBOLS

The following symbols are used throughout this report in discussions of bearing technology:

Symbol	*Definition*	*Units*
D	Orifice diameter	in.
F	Bearing load force	lb
G	Mass flow rate	lb-sec/in.
h	Bearing gap	in.
k	Bearing stiffness	lb/in.
$\triangle P$	Pressure difference	lb/in.^{2}
P_P	Absolute bearing pool pressure	lb/in.^{2}
P_S	Absolute bearing supply pressure	lb/in.^{2}
T	Orifice thickness	in.

ELASTIC ORIFICES

The consumption of gas in hydrostatic gas bearings must be kept at a minimum, not only for economic reasons, but also to maintain operation within the laminar flow region in the gap and to avoid shock waves or negative pressure profiles with resulting impairment or loss of load-carrying capacity. For these reasons, the diameter of the orifice is kept small, of the order of a few mils. The lower limit of orifice diameter is determined by mechanical considerations, such as the size of dust particles, which may be carried by the gas stream and subsequently clog the orifice. Hole diameters ranging from 0.003 to 0.150 inch have been used for evaluating elastic orifices.

Experimental elastic orifices have been made of silicone rubbers (Dow-Corning or General Electric), prepared according to manufacturer's instructions and poured into suitable molds (fig. 1) that have been coated with a mold release and that contain central pins of appropriate diameter. Subsequent to curing, the molded elastomers can be cut to the desired thickness in a jig, such as is illustrated in figure 2.

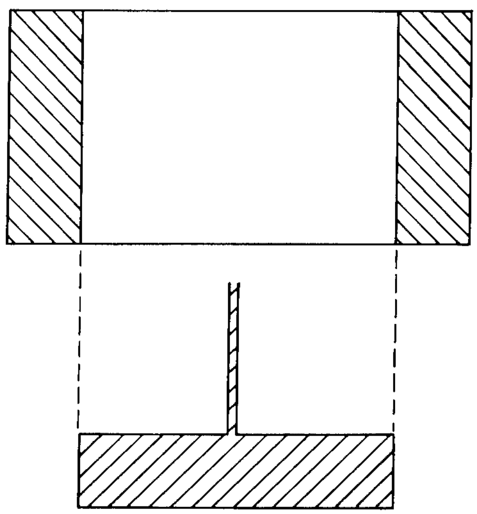

Figure 1.—Two-part mold.

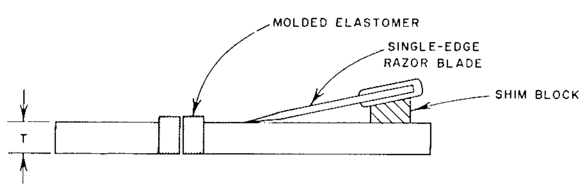

Figure 2.—Jig to cut molded elastomer to desired thickness.

A few examples of the various configurations that can be molded for elastic orifices having different properties are shown in figure 3. Examples appearing at the left are shown in the relaxed state when the upstream and downstream pressures are equal or nearly equal ($\Delta P=0$). The elastic deformation and closure of the orifice caused by increasing pressure ΔP is indicated in the examples shown on the right.

Five elastic orifices were operated in a test bearing in order to determine the ability of the elastic orifice to adjust flow as a result of increased pool pressure reflecting increased bearing load force. These test elastic orifices were constructed in accordance with the dimensions given in table I.

Table I.—Dimensions of Test Elastic Orifices

Orifice No.	T, inch	D, inch
4	0. 10	0. 01
6	0. 125	0. 01
11	0. 053	0. 015
13	0. 045	0. 015
14	0. 035	0. 015

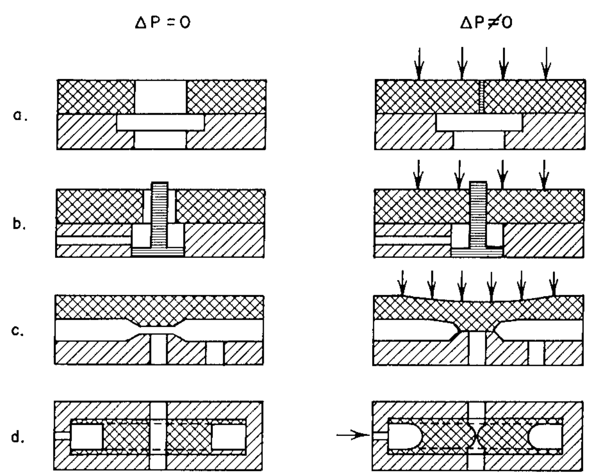

Figure 3.—Examples of elastic orifices.

The curves of mass flow rate versus pool pressure are shown in figure 4 while curves of bearing load force versus load displacement are shown in figure 5. The 10 numbered intersections of curves on these two graphs are the only valid points of comparison. At these points, the following factors are equal:

- supply pressure
- bearing area
- bearing load
- bearing gap
- pool pressure
- air flow rate
- pumping power

Thus, at these points, it can be said that all other factors are the same and that bearing **X** is stiffer or has a larger operating range around the nominal design load than bearing Y. For example, at point 5 in figure 4, where the G versus P_P curves for orifice 11 and 20 intersect, the stiffness of the bearings can be compared, since all other factors are equal.

The stiffness of the bearings around this operating point 5 can be determined from figure 5, where it can be seen that orifice 11 has a steeper curve (less displacement of the load for a given change in load force) than orifice 20. Similar comparisons can be made for the other nine common operating points.

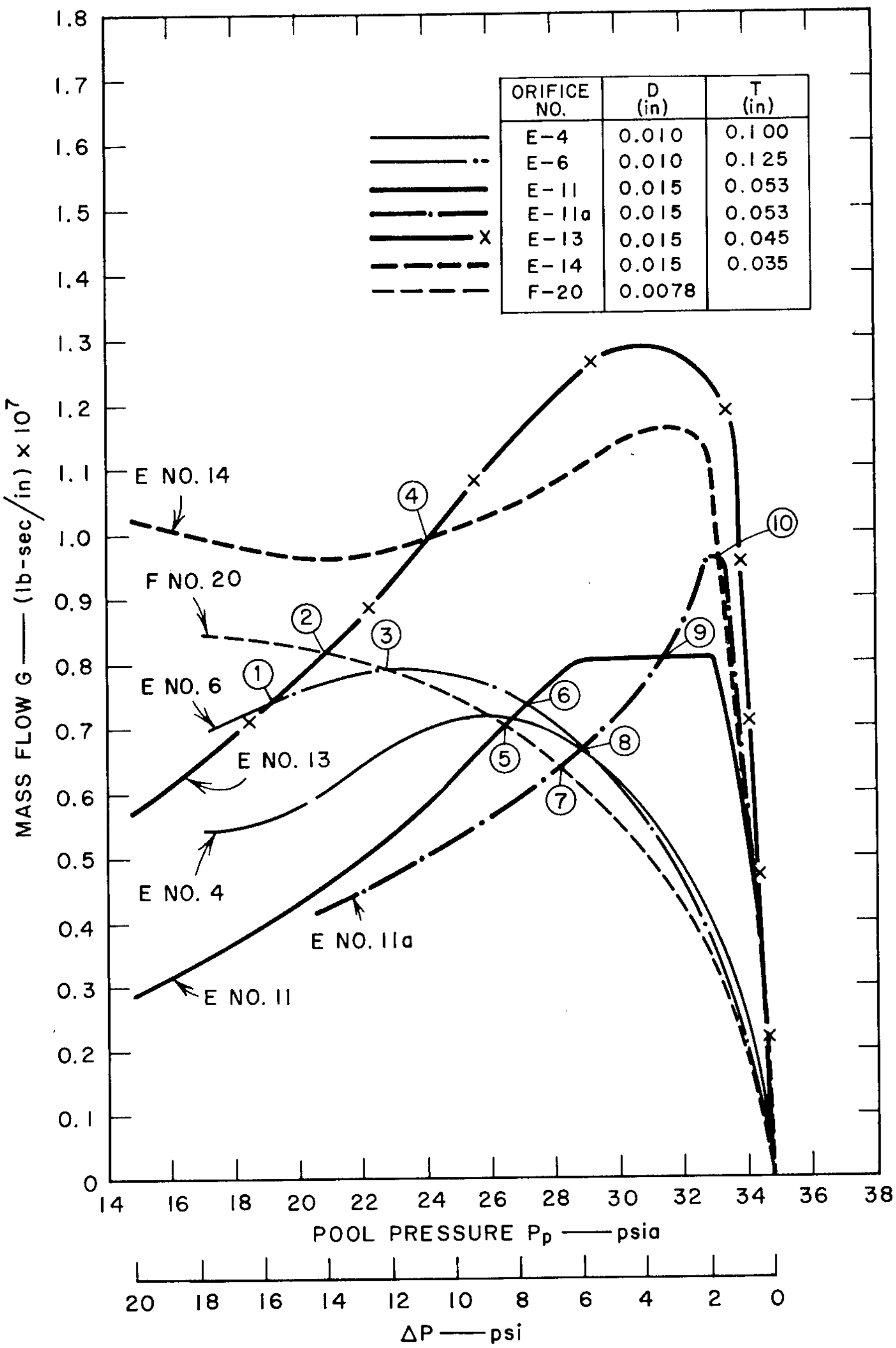

Figure 4.—Compensator characteristics.

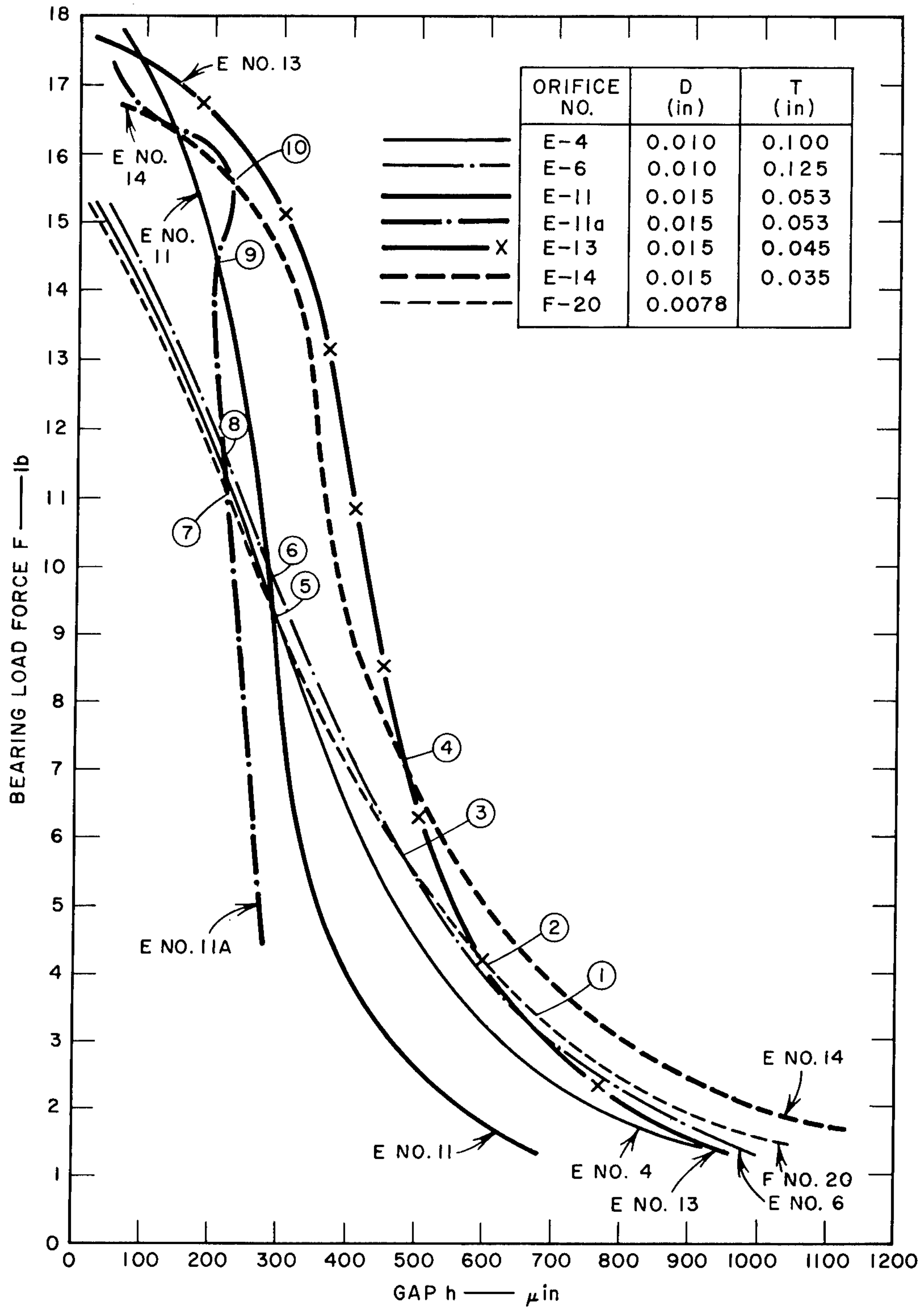

Figure 5.—Effect of gap on bearing load.

It should be noted that the use of an elastic orifice or any other flow control device will not, of itsef, affect flow rate or power consumption at the operating point. An elastic orifice can increase bearing stiffness or allow the construction of a more compact bearing with the same stiffness, but it cannot effect a reduction in pumping power.

The nonlinearity of the orifice characteristic curves (fig. 4) indicates that the power consumption is not constant. In a multipad journal bearing, a change in loading will shift the operating point and gas consumption of each diametrically opposed bearing pad. The orifice characteristic (G versus P_P) is a linear function, the sum of flow rates through the two bearing pads will remain constant during the load change. If the orifice characteristic curve is concave downward, total flow will be reduced. If it is concave upward, total flow will be increased. There is, then, a mechanism to reduce flow and power consumption, but it is independent of the elastic orifice concept of flow control. Many of the elastic orifice characteristic curves are concave downward, but so are those of a fixed orifice. No work has been done as yet to examine the extent to which this phenomenon can be exploited by manipulation of the nonlinearity of the orifice characteristic curve.

Figure 4 also illustrates some of the empirical design experience; for example, the result of enlarging orifice diameter is higher flow rate at high pool pressures. The elastic orifice at high pool pressures (low ΔP across the orifice) acts as a conventional orifice of the same hole size. Only when the ΔP increases does the orifice deflect and begin to reduce flow. In general, hole size determines the slope of the curve at the high pool pressure end.

Thickness of the elastic orifice is another basic geometric parameter that affects orifice performance. Because elastic orifice 4 is thinner than 6, it can deflect and reduce flow rates at lower ΔP values. As a general rule, a decrease in elastic orifice thickness moves the peak of the G versus P_P curve downward and toward higher pool pressures. At the same time, the slope of the curve at low pool pressures (high ΔP) is increased.

Manipulation of elastic orifice thickness and hole diameter permits freedom of adjustment to obtain a response curve with a desired slope at a desired operating condition of flow and pool pressure; this freedom of adjustment is not possible with conventional restrictors. However, practical limitations must be considered; for example, a flexible orifice of the plate configuration shown in appendix A, figure A–4, cannot be made too thin, or it will collapse under a large ΔP load. Orifices that are thick will not deflect and exhibit the desired flow area reduction. Variations in modulus of elasticity and accidental inclusion of small air bubbles have a pronounced influence on the load deflection characteristics.

The general rules stated above apply to the experimental elastic orifices, but variations in properties and orifice mounting make it difficult to predict the extent of change accomplished when geometry is modified. However, this variability should yield to design changes, such as mounting the orifice by bonding the edge to a rigid member to ensure uniform mounting conditions, and to more careful control of material properties and manufacturing techniques.

The curve shown as no. 11–A in figure 4 is not entirely produced by an elastic orifice function. In this case, an elastic orifice was positioned so that it would protrude into the bearing gap and could contact the movable bearing member. Thus, at low pool presure, the orifice was restrained from further deformation by air film lubrication, i.e., an air bearing within an air bearing. Although not a normally functioning elastic orifice, it does demonstrate the extent to which the slope of the G versus P_P curve can be increased by the choice of compensation.

Since quantitative analytical derivations of elastic orifice response have not been in agreement with experimental results, empirical methods must be employed to predict the effect of design changes. Before large-scale industrial application can be achieved, potential users will have to study several aspects of elastic orifice regulation in order to develop universal design criteria. Typical model test equipments used in the experimental evaluation of elastic orifices are described in appendix B.

Data concerning compensator response and bearing stiffness are presented in table II. It is obvious that an increase in compensator rate or response is always accompanied by an increase in bearing stiffness. This is true even to the extent that the very high response rate obtained with orifice 11 closing against the bearing plate (orifice 11–A) gives an infinite bearing stiffness and even an over-center response at one position. This represents a bearing that actively responds with an increased gap when loaded or, perhaps, a bearing that oscillates between two values of load when displaced, as in a confined journal bearing; however, this phenomenon could not be further investigated with the particulad test apparatus.

Table II.—Bearing Test Data

	Initial Conditions			I			II			III		
Operating point	Gap μin.	Load (lb)	Flow rate 10^7 lb-sec/in.	Orifice No.	dG/dP_P, 10^9 in.-sec	K, 10^{-4} lb/in.	Orifice No.	dG/dP_P, 10^9 in.-sec	K, 10^{-4} lb/in.	Orifice No.	dG/dP_P, 10^9 in.-sec	K, 10^{-4} lb/in.
1	670	3. 35	0. 74	E13	4. 45	1. 18	E6	2. 23	1. 0			
2	600	4. 20	0. 82	F20	−1. 11	1. 18	E13	5. 25	1. 47			
3	480	5. 80	0. 79	F20	−1. 59	1. 47	E6	0. 565	1. 82			
4	480	7. 20	0. 99	E14	1. 56	2. 22	E13	5. 7	5. 9			
5	290	9. 25	0. 71	F20	−3. 33	2. 44	E4	−0. 65	2. 78	E11	4. 76	10
6	288	9. 80	0. 74	E11	4. 76	10	E6	−3. 22	2. 63			
7	230	10. 70	0. 64	F20	−4. 55	2. 5	E11a	3. 85	10			
8	220	11. 65	0. 665	E6	−4. 76	2. 5	E11a	4. 35	10			
9	205	14. 40	0. 805	E11	0	4. 55	E11a	7. 15	−4			
10	225	15. 60	0. 96	E14	−47. 6	1. 11	E11a	0	2. 94			

Gas-lubricated bearings often tend to be unstable if certain design limits are not strictly observed. In classical gas bearing design, attempts to increase bearing stiffness by such methods as enlarging the pool are generally limited by instability; however, even the very stiff bearings used in the elastic orifice performance evaluations were quite stable in the load and gap range tested. A dynamic test was used to study this stable behavior: an impulse load on the bearing was simulated by striking the bearing case; time records of the bearing response were obtained by photographing the oscilloscope traces of displacement. Analysis of the displacement recordings indicated that three distinct frequencies are discernible, the lowest being about 60 cps, which corresponds to the frequency predicated for the bearing mass and the load pressure chamber acting as a pneumatic spring. The next highest frequency is of the order of 1000 cps, and it is believed that this is the natural frequency resulting from the bearing plate mass and the bearing stiffness. The highest frequency components are near 10,000 cps and correspond to expected stress wave reflections through the bearing case structure.

No obvious difference between the rate of decay of oscillations in elastic-orifice-regulated bearings and fixed-orifice-regulated bearings could be detected by qualitative inspection. There does appear to be a limiting frequency response around 1 kc; this results in the bearing with orifice characteristic 11–A having a lower dynamic stiffness than static stiffness. The cause of this limitation has not yet been determined, but either or both of two calculated factors may be responsible:

(1) Pool volume filling time is of the order of 1 msec.

(2) The inertia forces of an elastic orifice oscillating at 1 kc closely approach the value of pressure force available.

ALTERNATIVE DESIGNS FOR FLEXIBLE ORIFICES

The elastic diaphragm may be mounted between two rigid disks of similar diameter in such a fashion that pressure buildup on the upper disk will cause it to deform the elastic diaphragm and thus restrict the size of the orifice. A removable regulator assembly may also be provided by mounting the elastic diaphragm within an externally-screw-threaded plug positioned in the gas passageway.

When pressurized gas bearings must operate under conditions of extreme temperatures or radioactivity, the elastic diaphragm may be replaced by a metal disk of much larger diameter, surmounted by a frustro-conical member containing a number of small holes (fig. 6). Flow is controlled by movement of this member as the downstream pressure varies.

APPLICATIONS

Applications for elastic-orifice regulation of hydrostatic gas bearings would be the same as for fixed-orifice regulation with the additional advantages of increased bearing stiffness, or reduced size. Figure 7 illustrates the assembly of a gas-lubricated bearing regulated by an elastic orifice.

The combination of high speed of operation and extremely low friction has encouraged investigation of the application of gas bearings in machine tool components, especially grinding spindles.

Hydrostatic gas bearings are particularly suitable for instrument design where operations will require frequent starting and stopping, and for the pivot support of instruments, such as the gimbal axis of gyros and precision balances.

Hydrostatic gas bearings are used for vibration and shake tables to eliminate the cross-coupling between reference axes, as well as in three-degree-of-freedom bearings for attitude control platforms in simulated space vehicles. They also find use as a support for magnetic read heads.

In the above applications either stiffness or friction are critical requirements.

Elastic orifices have been shown to be applicable to many kinds of fluid flow regulation, for example, showerheads.

Operation of the elastic orifices as a logic device has been investigated; in this instance, it operates as a pneumatic switch activated by pressure pulses in the same fashion as does a relay or diode switch. However, this concept has not been shown to be very competitive.

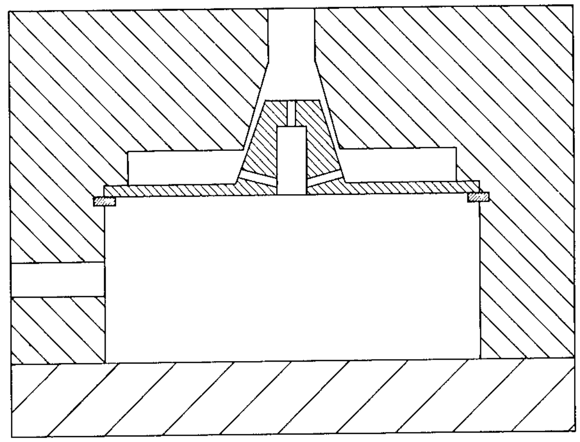

Figure 6.—Metallic flexible orifice.

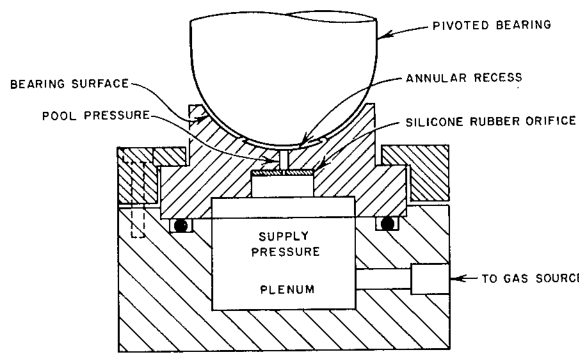

Figure 7.—Assembly of gas-lubricated bearing regulated by an elastic orifice.

APPENDIX A: FLOW CONTROL IN GAS-LUBRICATED BEARINGS

Externally pressurized (hydrostatic) gas bearings commonly employ some form of flow restricting element to control the flow of gas to a pressurized cavity which supports the load. A generalized schematic drawing of such a bearing is shown in figure A–1. Air flows at a rate G from supply pressure P_S through the fixed orifice into the pool. Pool pressure P_P sustains the load F. The bearing gap h adjusts to allow the flow G to exhaust to the surrounding pressure. The pressure drop ΔP (which is P_S–P_P) can be measured across the orifice.

As the load force is increased by some increment ΔF, producing a total load of $F+\Delta F$, supply pressure P_S is unchanged, but pool pressure must increase to support the larger load. An increase in pool pressure P_P reduces the pressure drop ΔP across the inlet orifice and causes a change in flow. Figure A–2 shows flow rate versus pool pressure for a conventional fixed ori-

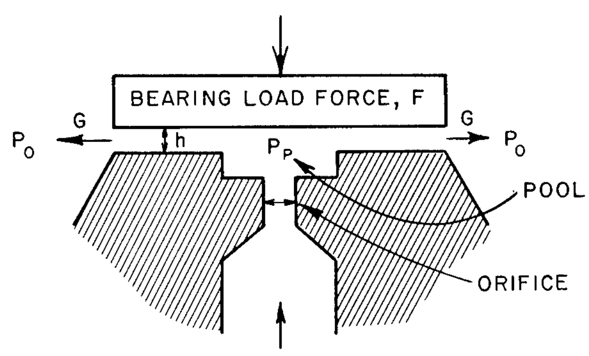

Figure A–1.—Externally pressurized gas bearing (fixed orifice).

fice. This curve illustrates that an increase in pool pressure causes a reduction in flow. This reduced flow causes a reduction in gap h, and the load is displaced.

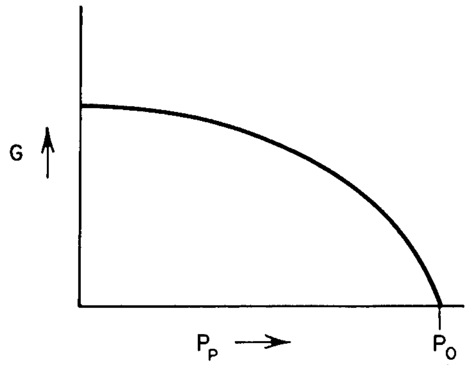

Figure A–2.—Effect of pool pressure on flow rate of conventional fixed orifice.

If the flow were increased rather than decreased, it would be possible to reduce or eliminate the bearing displacement and thus have a very stiff bearing. Figure A–3 shows a desirable relationship between flow and pool pressure that would accomplish this result. (The relationship may be determined empirically by controlling the gas flow manually with a valve and observing the value needed to eliminate displacement as load is changed.)

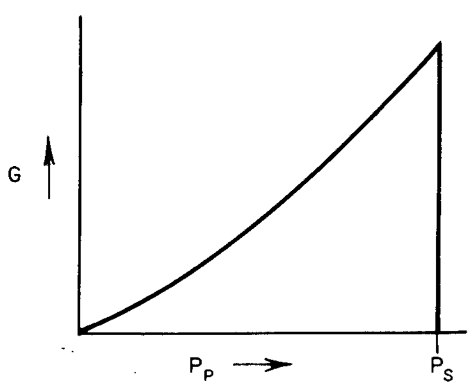

Figure A–3.—Effect of pool pressure on flow rate of ideal bearing.

Replacement of the fixed orifice shown in figure A–1 with an elastic orifice of the general configuration shown in figure A–4 provides an automatic means of adjusting air flow to compensate for variations in load. Under a certain initial load F, the orifice will deform within the

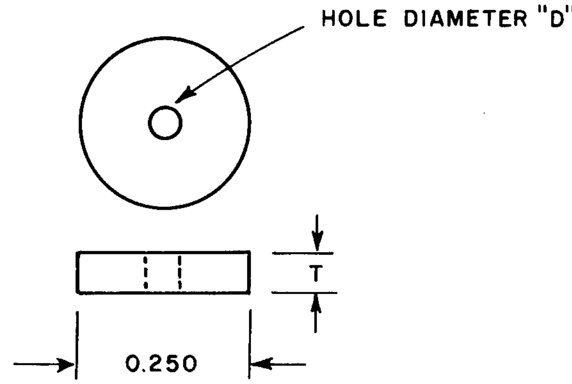

Figure A–4.—Elastic orifice.

bearing housing until it takes the general appearance shown in figure A–5(a). Increasing the load from F to $F+\Delta$ will tend to increase pool pressure P_P, thereby decreasing ΔP and changing the deformation of the orifice to resemble that shown in figure A–5(b). The increased orifice size permits more air to flow through the bearing, thereby redeflecting the load toward its initial position, that is, increasing the stiffness of the bearing. At $\Delta P=0$, there will be no deformation of the orifice; similarly, at some ΔP greater than the operating range of the unit, the orifice will be completely closed by supply pressure P_S (fig. A–5(c)).

The effect on the hole diameter of an elastic orifice caused by a pressure differential between the upstream and the downstream area is shown in figure A–6 (curve D). From a diameter of 3.5 mils in the "relaxed state" at zero pressure differential, the elastic orifice is deformed by increasing pressure differential so that the hole diameter is only 2.5 mils at 48 psig. The regulation of gas flow by the effect of increased pressure differential on the elastic orifice is also shown in figure A–6 (curve (Q_{el}); comparison is made with the performance of a fixed orifice of 3.5 mils (curve Q_{fi}). These results were obtained from measurements made as the upstream pressure was varied between 8 psig and 48 psig; downstream pressure was held constant at atmospheric pressure.

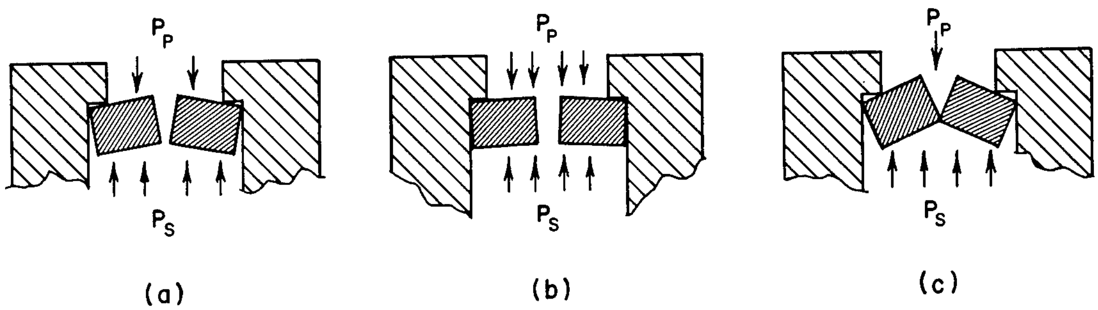

Figure A-5.—Operation of elastic orifice.
(a) Initial operating condition.
(b) High pool pressure (low ΔP).
(c) Low pool pressure (high ΔP).

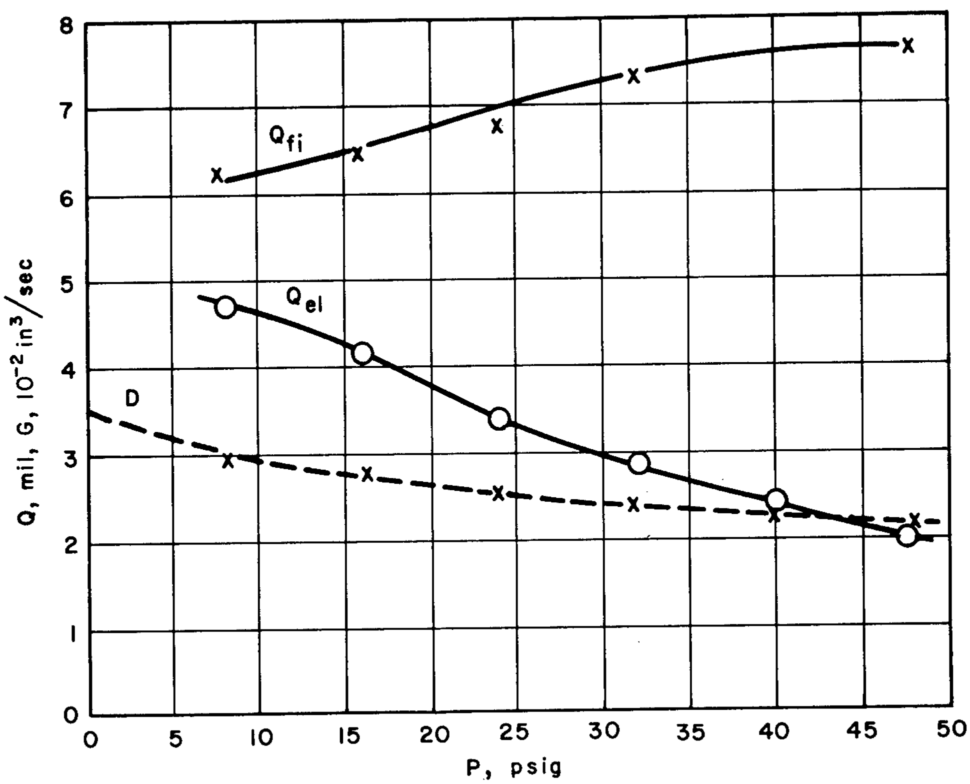

Figure A-6.—Comparison of fixed and elastic orifices.

APPENDIX B: ELASTIC ORIFICE MODEL TEST EQUIPMENT

The large-scale model test equipment illustrated in figure B-1 was used in the evaluation of the performance of some elastic orifice configurations. A silicone-rubber disk is molded into a metal form and can be pressurized independently downstream (P_1) and upstream (P_{PL}) by means of pressure regulators; if desired, the downstream side can be vented to atmosphere. The upstream and downstream pressures are measured by pressure gages; flow rate on the upstream side is measured by means of a variable-orifice-type flowmeter.

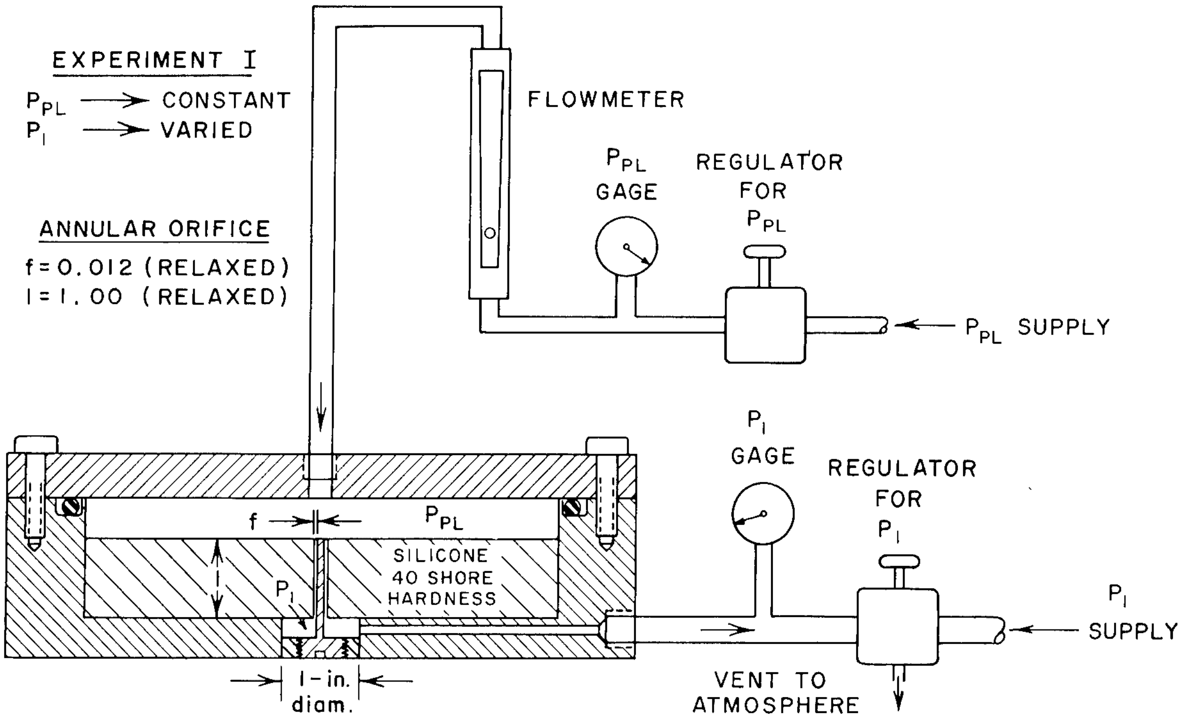

Figure B–1.—Large-scale elastic orifice model test arrangement.

A cross-sectional view of the test bearing used to evaluate flow rates, load displacements, and dynamic responses of elastic orifices is given in figure B–2. The device is a thrust bearing with a single, centrally-located lubricant inlet and pool. Bearing load is applied by introducing air pressure through inlet port (1) into a closed chamber over a rubber diaphragm (4) which is capable of supporting the pressure to give a maximum bearing load of about 20-lb force. The chamber has an effective piston area of 9.30 in ^{2}. Loading pressure is regulated manually with a pressure regulator and measured with a mercury manometer.

Displacement of the loaded bearing member is measured by a linear differential transformer (2) and oscilloscope unit capable of discerning 0.0001 inch of motion. Pool pressure is measured by a pressure transducer inserted in port (3). Lublication air is supplied through port (5) to a chamber below the orifice plate (6) and air flow is measured by means of a flowmeter, calibrated at supply pressure and temperature.

The orifice plate is removable, permitting simple changes of orifice. Lublicant supply pressure is measured externally in the supply line; since the orifice area is much less than supply line and fitting area, this measurement is an accurate representation of pressure below the orifice.

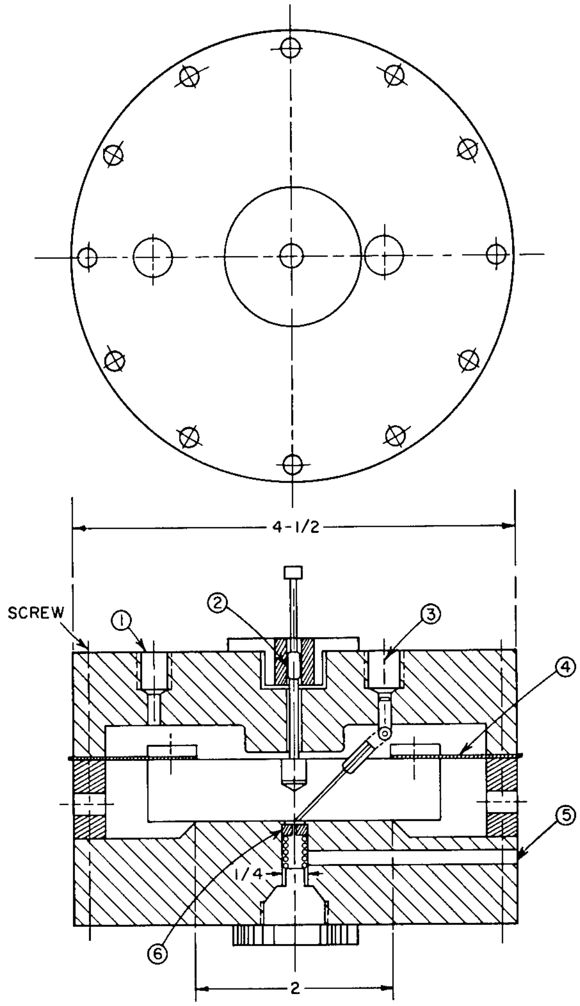

Figure B-2.—Test bearing.

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