EVALUATION OF A GAS BEARING PIVOT
FOR A HIGH AMPLITUDE
DYNAMIC STABILITY BALANCE

By

A. E. Hodapp, Jr.

von Kármán Gas Dynamics Facility

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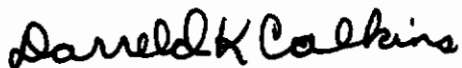
December 1962

FOREWORD

Recognition is justly given to Mr. J. M. Epley for his effort in the design of the bearing and to Mr. R. L. Ledford for his work in developing the angular transducer.

ABSTRACT

The results of a study of the load capacity and damping level of a gas journal bearing are presented. Experimental data indicate that the bearing is capable of supporting radial loads in excess of 300 lb and that the damping parameter has some inverse relationship to frequency which is determined by the radial load. It is shown that the damped motion of the gas bearing can be approximated adequately with the viscous-damping equation.

PUBLICATION REVIEW

This report has been reviewed and publication is approved.

Darreld K. Calkins
Major, USAF
AF Representative, VKF
DCS/Test

Jean A. Jack
Colonel, USAF
DCS/Test

CONTENTS

ILLUSTRATIONS

NOMENCLATURE

a	Distance from the center of rotation to the centroid of the offset mass, in.
b	$b = f(L)$, Fig. 7
C_1	$C_1 = f(L)$, Fig. 7, ft-lb/rad
C_{y_R}	Cycles to damp to a given amplitude ratio R, cycles
d	Bearing core diameter, in.
E	$E = f(p)$, Ref. 2
f	Frequency, cycles/sec
g	Acceleration due to gravity, ft/sec^2
h	Radial clearance, in.
I	Moment of inertia, slug-ft^2
i	$\sqrt{-1}$
L	Radial load, lb
ℓ	Length of the bearing outer ring, in.
ℓ_n	Natural logarithm
M_θ	Angular restoring-moment parameter $= mga$, ft-lb/rad
$M_{\dot{\theta}}$	Angular viscous-damping-moment parameter, $\dfrac{\text{ft-lb sec}}{\text{rad}}$
m	Mass, slugs
p	Plenum pressure, psig
P_{cr}	Critical plenum pressure, psig
R	Ratio of the amplitude of a damped oscillation after a given number of cycles to the initial amplitude
t	Time, sec
ϵ	Eccentricity, deflection per unit clearance Δ_h/h
θ	Angular displacement, rad or deg
θ_o	Maximum angular displacement, rad
$\dot{\theta}$	Angular velocity, rad/sec
$\ddot{\theta}$	Angular acceleration, rad/sec^2
ω	Angular frequency of oscillation $= 2\pi f$, rad/sec

this model is then used to develop a design procedure for a full journal bearing. The equation for radial load,

$$L = \frac{E\pi\ell d\epsilon p}{2}$$ (1)

developed as part of the theory in the design procedure (Ref. 2), can be used to predict the performance of a gas bearing.

Because of the nature of the present application of the gas bearing, its design deviated from that of Ref. 1 in the following ways: (1) the plenum was located in the fixed center portion of the bearing to supply gas through the orifices to float the outer moving ring of the bearing. The design in Ref. 1 had a manifold located in the outer fixed portion of the bearing to supply gas through the orifices to float the inner moving core; and (2) plates were used at the ends of the bearing, whereas the design in Ref. 1 was used with gas exhausting along the journal.

For proper operation it was necessary that the bearing be self-centering; therefore, end plates were required to extend beyond the outer rotating ring. End plates for centering the bearing would allow the exhaust gas of the bearing to expand radially outward. Since the bearing is to be contained within a model this characteristic is undesirable, and therefore provisions were made to cancel this effect. The shields shown in Fig. 1 are part of the end plates; their function is to direct the exhaust gas toward the center of the bearing, thus reducing the radial flow of the gas. During the design it was anticipated that the end plates would have some effect on bearing load capacity; therefore, the bearing was designed with removable shims (Fig. 1) to allow variation of end clearance.

2.2 ANGULAR TRANSDUCER

An angular transducer, shown in Fig. 2 and described in detail in Ref. 2, was developed to obtain a continuous time history of the oscillatory motion of the bearing. The variable reluctance transducer (Fig. 2) consists of a housing and an eccentric. The housing which contains the two "E" cores is mounted on the fixed portion of the bearing. The eccentric, mounted on the outer movable ring of the bearing, has no physical connection to any component of the transducer.

When the coils contained in the "E" cores are excited, magnetic paths are established through the "E" cores, their adjacent air gaps, and the eccentric ring. As the eccentric is rotated the reluctance of the magnetic paths through the "E" cores and eccentric remain constant while the reluctance of the air gap changes. This change in reluctance

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produces an analog signal proportional to the angular displacement of the outer ring of the bearing. The output is amplified by a carrier amplifier system and recorded on a direct-writing oscillograph. The present transducer output is linear over a ±16-deg range, and with minor modifications, this range can be extended.

2.3 CALIBRATION APPARATUS

To determine the load-carrying capacity of the gas bearing, radial loads were suspended below the bearing by means of a loading bracket. The radial clearance, plenum pressure, and gas temperature were monitored by means of a dial gage, pressure gage, and thermocouple, respectively.

Gas bearing damping level was determined with the aid of the "U" bracket and the two radial arms, shown in Fig. 3. The "U" bracket connects directly to the bearing and supports the offset-mass and the radial arms. Calibration weights can be attached at either the axial arms or to the bolts located in the slots on the radial arms. The calibration weights are arranged so that their weight vector acts through the center of rotation regardless of the angular displacement. This allows variation of radial load without affecting the restoring moment of the system. Moment of inertia for the system may be changed either by varying the distance to the offset-mass or by arranging the calibration weights at varying distances about the center of rotation. These controls on the moment of inertia allowed the frequency to be varied or held constant over a range of radial loads.

3.0 PROCEDURE AND RESULTS

3.1 STATIC LOADING

The load-carrying capacity of the bearing was determined by varying the end clearance and plenum pressure over a range of radial loads. For each particular end clearance the plenum pressure was raised to the design maximum pressure of 500 psig, the dial reading was recorded, and the bearing was loaded. The plenum pressure was then slowly reduced until the critical pressure was reached, i. e., the dial gage indicated an eccentricity, ϵ, of approximately 0.5. This procedure was repeated over a range of total end clearances from 0.002 to 0.010 in. for loads in increments of 25 lb up to a load of 300 lb.

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A family of curves of critical pressure versus radial load showing the effect of end clearance and the theoretical capacity of the bearing are presented in Fig. 4. The experimental results show that optimum performance of the bearing in terms of static load capacity was obtained for total end clearances of 0.005 and 0.006 in. It is also interesting to note that even though the bearing design as used in the theory was modified to incorporate end plates, the theoretical performance closely agrees with the optimum performance obtained.

3.2 DAMPING MEASUREMENTS

The free-oscillation technique using an offset-mass was employed in this investigation because it is the most accurate method for obtaining the gas bearing damping coefficient. The damped oscillatory trace resulting from this technique is conducive to studying the characteristics of the damped motion.

The test procedure was to displace the offset-mass from its equilibrium position and release it. The resulting motion was recorded by a direct-writing oscillograph used in conjunction with the angular transducer.

By controlling the moment of inertia of the system as outlined previously, trends of bearing damping were obtained at constant radial load with varying frequency and at constant frequency with varying radial load. Results obtained in this manner are shown in Figs. 5 and 6. For both the static loading and damping measurements nitrogen was used as the working gas.

Oscillograph traces showing the damped motion of the gas bearing revealed that the frequency for a given oscillation remained constant as the motion damped out and that a plot of the natural logarithm of the envelope curve against cycles of oscillation was approximately linear over a range of amplitudes. For the amplitude variation to be exactly linear, the motion must be damped exponentially. In some cases, nonlinearities were observed over larger amplitude ranges, confirming deviations from true exponential damping; however, for the small amplitude range of concern, the amplitude variation could be approximated adequately with a linear curve.

These variations are significant since they indicate that the bearing damping characteristics over small amplitude ranges may be approximated with the basic viscous-damping relations. This becomes apparent in observing the solution of the basic differential equation describing

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one-degree-of-freedom viscous-damped motion which is given by

$$I\ddot{\theta} - M_{\dot{\theta}}\dot{\theta} - M_{\theta}\theta = 0 \tag{2}$$

For damping less than critical, the roots of the characteristic equation for the motion defined by the above differential equation are

$$-\left(-\frac{M_{\dot{\theta}}}{2I}\right) \pm i\sqrt{\left(\frac{-M_{\theta}}{I}\right) - \left(\frac{-M_{\dot{\theta}}}{2I}\right)^2}$$

The damping parameter of the system is $-M_{\dot{\theta}}/2I$, and the frequency of oscillation, ω, is

$$\sqrt{\left(\frac{-M_{\theta}}{I}\right) - \left(\frac{-M_{\dot{\theta}}}{2I}\right)^2}$$

The quantity $(-M_{\dot{\theta}}/2I)^2$ in the frequency term is negligibly small relative to the quantity $(-M_{\theta}/I)$; hence the frequency is adequately given by $\sqrt{-M_{\theta}/I}$. From the above pair of complex roots a solution of the viscous-damped equation can be written as

$$\theta = \theta_o e^{-\left(\frac{-M_{\dot{\theta}}}{2I}\right)t}\left[\left(\sin\sqrt{\frac{-M_{\theta}}{I}}\,t\right)\right] \tag{3}$$

The above solution for the viscous-damped system confirms that the envelope curve of a damped oscillation varies exactly exponentially with time and that the frequency is constant.

The coefficient of the damping term (Eq. (2)), the angular viscous-damping-moment parameter, can be written as

$$M_{\dot{\theta}} = \frac{(\ell n\ R)(-M_{\theta})}{2\pi^2\ C_{y_R}\ f} \tag{4}$$

Therefore, for equivalent viscous-damped motion, it is only necessary to determine the frequency of oscillation and the cycles to damp to a given amplitude ratio to evaluate the damping parameter.

In true viscous-damped motion, the damping moment $(M_{\dot{\theta}}\dot{\theta})$ is defined as a moment proportional to the angular velocity of motion; therefore, $M_{\dot{\theta}}$ is constant for all values of amplitude and frequency. One would expect the damped motion of the gas bearing to be defined exactly by the viscous damping theory; however, this is not the case. The damping parameter for damped motion of the gas bearing departs from

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true viscous damping because there exists some inverse relationship with frequency, as demonstrated in Figs. 5 and 6. At a constant radial load this inverse relationship of damping with frequency remains approximately fixed; however, changing the radial load causes a marked change in the relationship between frequency and damping.
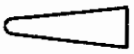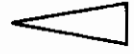
The effect of radial load on the relationship between the damping parameter and frequency is best demonstrated by the following empirical equation:

$$M_{\dot{\theta}} = C_1 (f)^b \tag{5}$$

Figure 7 shows the dependence of C_1 and b on radial load, thus exhibiting the effect of radial load on the relationship between the damping parameter and frequency.

The offset-mass technique used to evaluate the damping has the disadvantage that the frequency range is restricted; however, it is the only technique that can be successfully used to measure the low damping level of the bearings. Any effort to extend the frequency range to include higher frequencies would necessitate the introduction of an external restoring moment into the system. The damping contribution of this restoring moment, being at least an order of magnitude higher than that of the bearing, would certainly tend to mask the damping levels and/or trends of the data. The frequencies obtained were lower than the anticipated operating range in the wind tunnel; therefore, Eq. (5) may be used to extrapolate existing data to the desired frequency range.

The following table, used to demonstrate the expected performance of the gas bearing, shows that for two typical re-entry configurations at Mach numbers 8 and 10, the bearing damping is negligible compared to the aerodynamic damping. Values of aerodynamic damping are based on Newtonian flow (Ref. 3), and values of bearing damping were determined using Eq. (5).

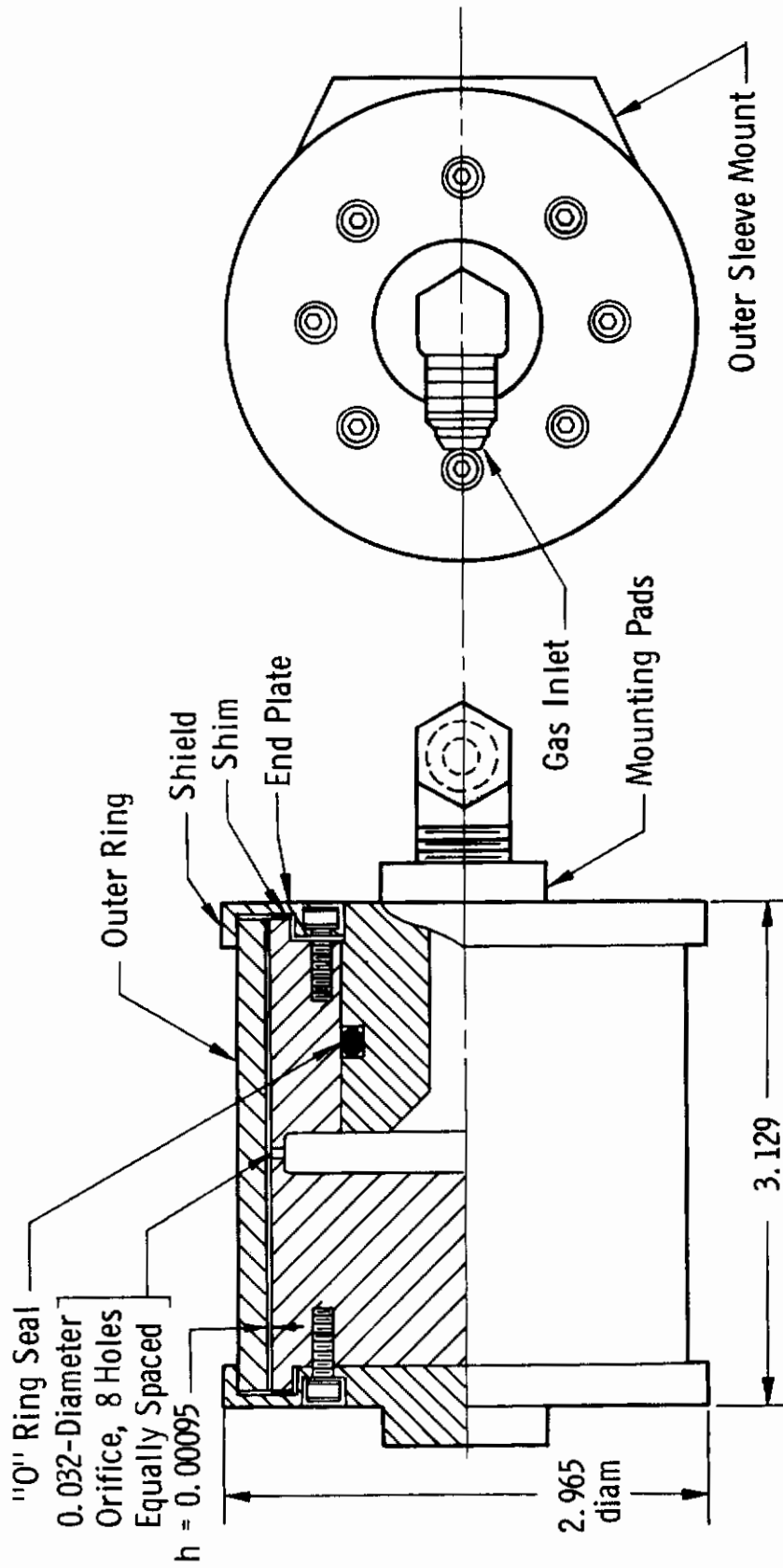| Configuration	Mach No.	$M_{\dot{\theta}}$ (Aerodynamic)	$M_{\dot{\theta}}$ (Bearing)	Frequency, f, cps
◁	10	-0.72×10^{-2}	-0.55×10^{-4}	3.65
◁	8	-1.75×10^{-2}	-0.28×10^{-4}	6.75

4.0 CONCLUSIONS

A gas bearing has been developed for use in free-oscillation dynamic stability balances. Based on data obtained during the evaluation of the bearing, the following conclusions were reached:

1. End clearance has an appreciable effect on the load-carrying capacity, with the 0.005 and 0.006 in. total end clearances giving the optimum load capacity.

2. The load-carrying capacity for the 0.005- and 0.006-in. total end clearances agrees well with theory, and the bearing is capable of satisfactory operation at loads in excess of 300 lb.

3. The damping parameter of the gas bearing varies inversely with frequency, and the magnitude is determined by the radial load.

4. The damped motion of the gas bearing can be approximated adequately with the equation for viscous-damped motion.

5. The performance of the bearing indicates that it can be successfully applied in a dynamic stability balance.
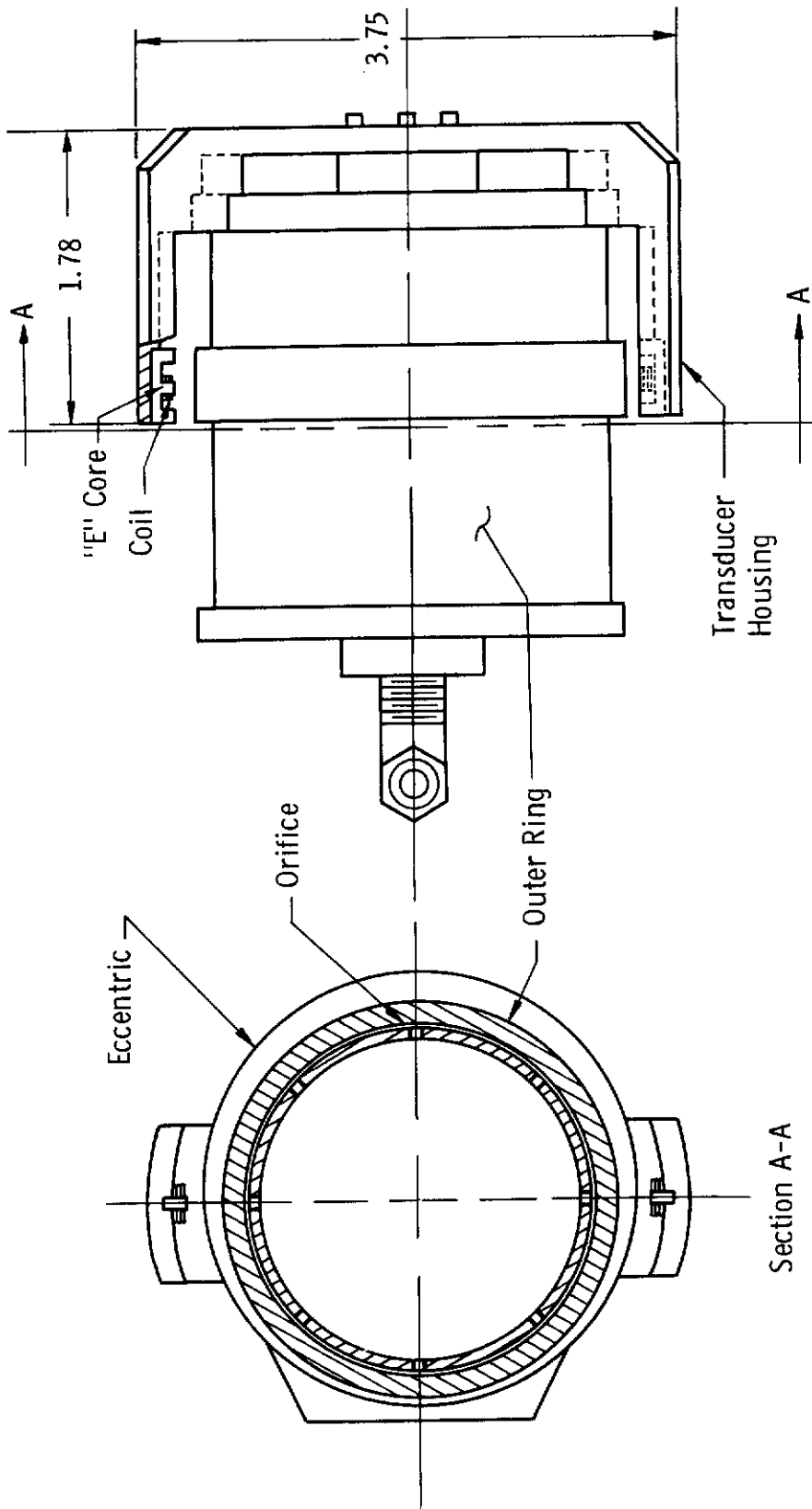
REFERENCES

1. Grinnel, S. K. and Richardson, H. H. "Design Study of a Hydrostatic Gas Bearing with Inherent Orifice Compensation." Transactions of the ASME, Vol. 79, No. 1, January 1957, pp. 11-21.

2. Welsh, C. J., Ledford, R. L., Ward, L. K., and Rhudy, J. P. "Dynamic Stability Tests in Hypersonic Tunnels and at Large Model Amplitude." AEDC-TR-59-24, December 1959.

3. Tobak, Murray and Wehrend, William R. "Stability Derivatives of Cones at Supersonic Speeds." NACA TN 3788, 1956.

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Fig. 1 Sketch of the Gas Bearing

All dimensions in inches

Outer Sleeve Mount

Gas Inlet

Mounting Pads

Shield

Shim

End Plate

Outer Ring

"O" Ring Seal

0.032–Diameter
Orifice, 8 Holes
Equally Spaced

h = 0.00095

3.129

2.965
diam

All dimensions in inches

"E" Core
Coil

Transducer
Housing

Orifice

Eccentric

Outer Ring

Section A-A

Fig. 2 Angular Transducer Bearing Assembly

3.75

1.78

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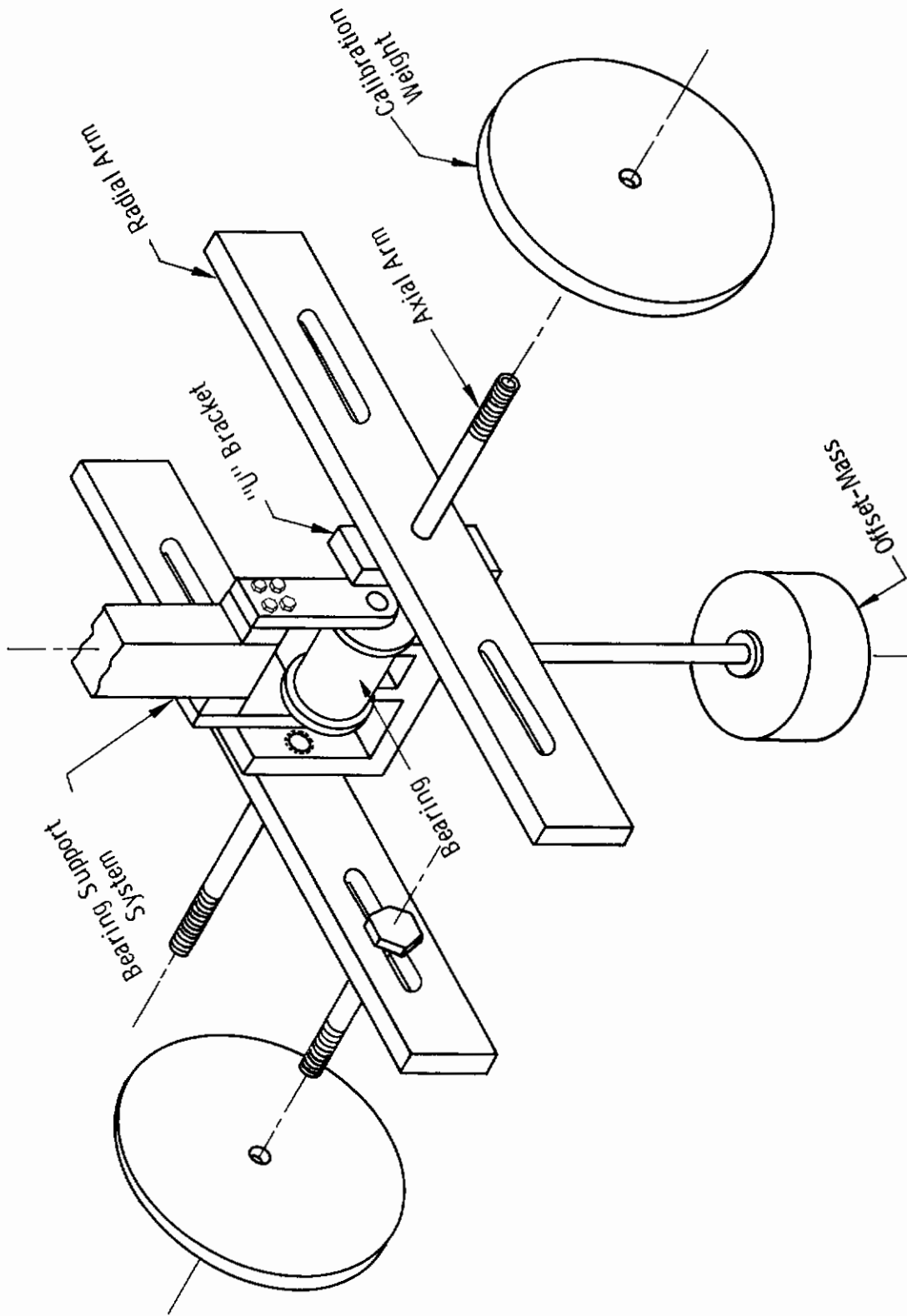
Fig. 3 Damping Calibration Equipment

Calibration Weight

Radial Arm

Axial Arm

"U" Bracket
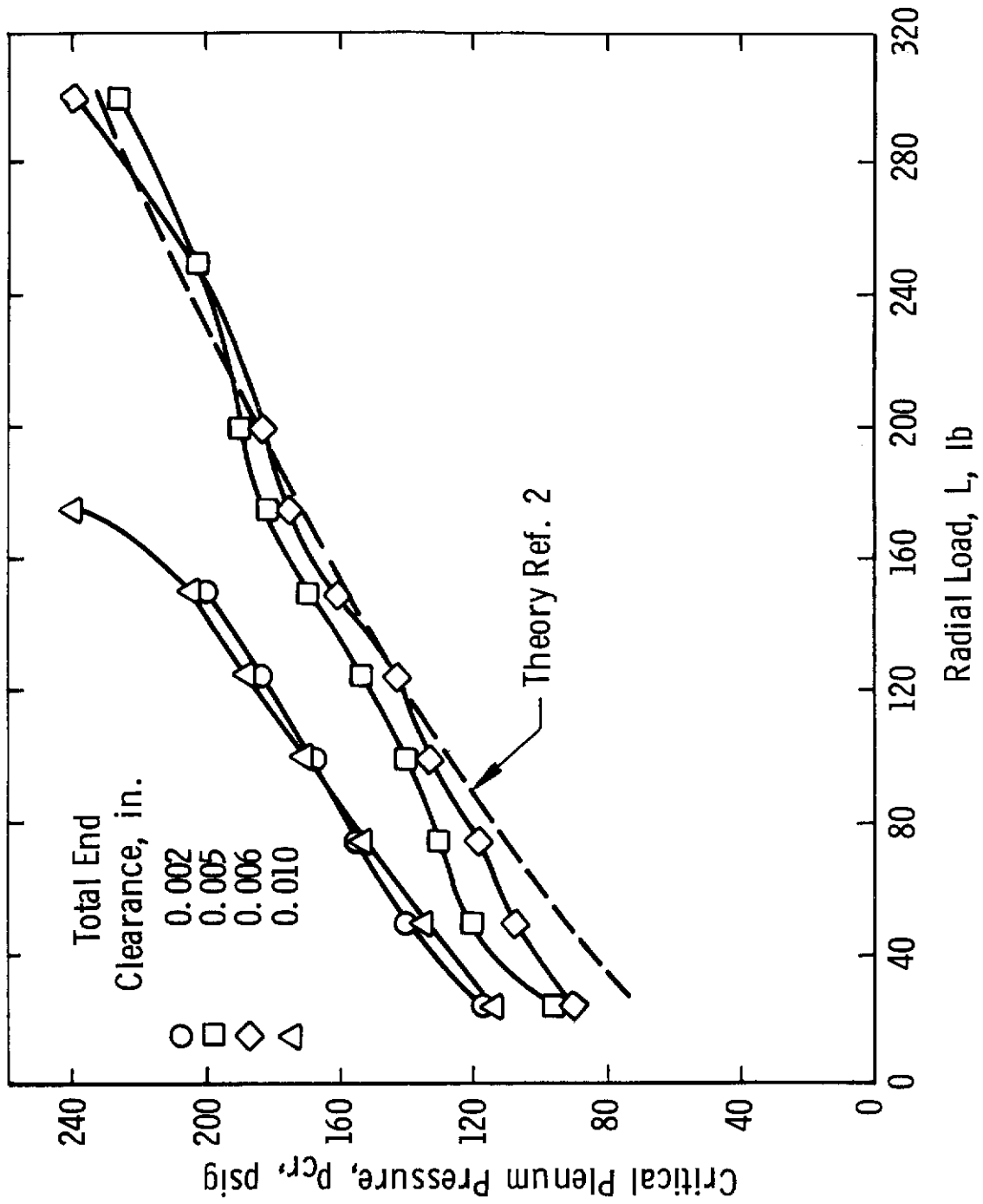
Offset-Mass

Bearing

Bearing Support System

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Fig. 4 Static Loading Characteristics of the Gas Bearing
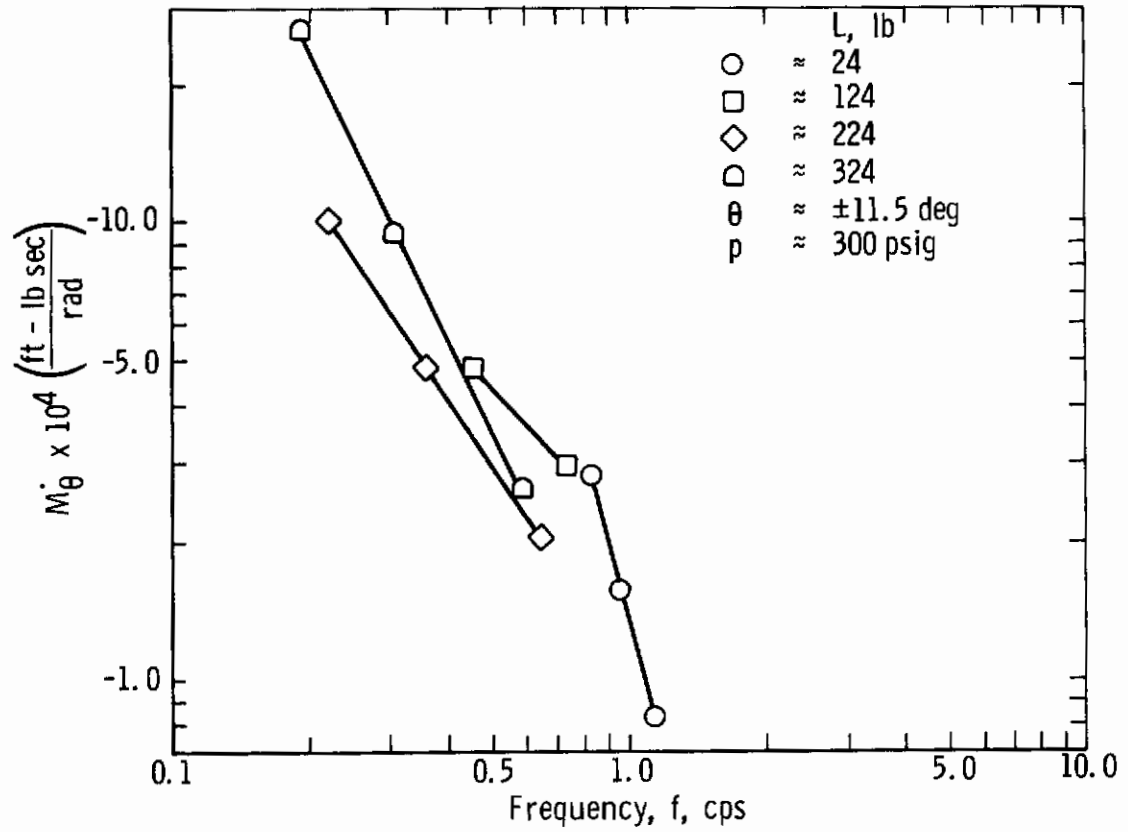
Fig. 5 Variation of Damping Parameter with Frequency, Constant Radial Load

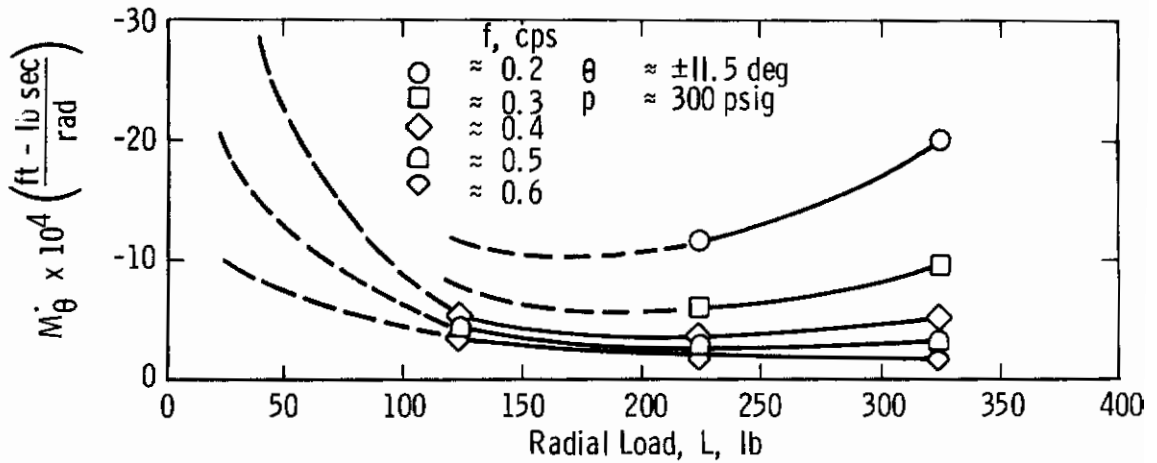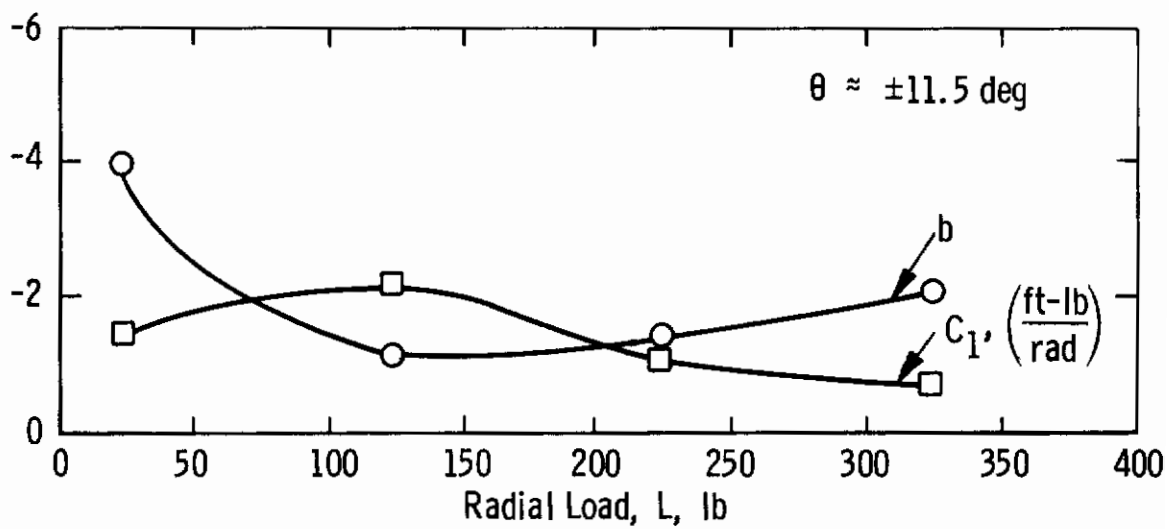**Fig. 6 Variation of Damping Parameter with Radial Load, Constant Frequency**

Fig. 7 Variation of b and C_1 with Radial Load