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EXPERIMENTS ON AN ULTRA-STABLE GAS JOURNAL BEARING

by

S.B. Malanoski

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EXPERIMENTS ON AN ULTRA-STABLE GAS JOURNAL BEARING

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MECHANICAL TECHNOLOGY INCORPORATED

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EXPERIMENTS ON AN ULTRA-STABLE GAS JOURNAL BEARING

1961

ABSTRACT

Shallow grooving in a herringbone pattern has been proposed to enhance the stability of both gas and liquid lubricated journal bearings. It has been shown theoretically that this possibility is particularly advantageous for unloaded journal bearings.

This paper describes corroborating experiments. The experiments included the running of an unloaded bearing up to speeds of 60,000 rpm and the collection of steady state load-displacement attitude angle data at intermediate speeds up to and including 60,000 rpm. No sign of bearing whirl instability was detected. There was good correlation between theoretical and experimental data. Design data is included as a guide for future designs. Author

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INTRODUCTION

In recent years during the development of gas bearing theory two phenomena became apparent: the plain, lightly-loaded journal bearing is unstable (Ref. 1) and the load capacity of a plain journal bearing approaches an asymptotic value with increasing speed (Ref. 2). Many theoretical and experimental investigators have verified the existence of these problems and have suggested a variety of solutions. In some instances these solutions proved very successful. For example, the application of an external load to a lightly-loaded bearing tends to raise the whirl threshold (Ref. 3). Also, machining axial grooves into a bearing raises the whirl threshold slightly at the expense of a decrease in the overall load capacity of the bearing (Ref. 4,5). Probably the most successful solution to the problem to date is the utilization of the tilting pad bearing (Ref. 6) which, when properly designed, possesses both anti-whirl properties and high stiffness characteristics. However, the bearing power loss can increase two-fold over the plain bearing, and the mechanical design of the tilting-pad bearing system is more difficult and costly.

Recently the herringbone-grooved (completely grooved) journal bearing has been studied both theoretically and experimentally (Ref. 7,8 and 9). Ref. 7 shows the completely-grooved bearing has a higher load capacity than the plain bearing at moderate and high values of Λ. Ref. 8 shows that the bearing has high stability characteristics. Ref. 9 gives a theoretical treatment of the completely-grooved bearing for the incompressible case and a presentation of some experimental results for bearing numbers as high as 4.8. Good correlation of theory and experiment was found.

This paper presents the results of experiments on the partially-grooved journal bearing and compares these experimental results with the theoretical results obtained from the analysis published in Ref. 7, 8. The partially-grooved bearing was selected for study because it is the optimum design for obtaining maximum radial stiffness. The analysis given in Ref. 8 indicates that the grooved bearing will be stable up to a certain speed for a given bearing geometry and rotor mass. In Ref. 7 it is shown that the load capacity of this bearing is not limited as is that of a plain journal bearing. These facts have been verified by the experiments reported herein, i.e., no sign of bearing instability was detected when an unloaded bearing was run up to a speed of 60,000 rpm. The measured load-displacement attitude angle characteristics were found to be in good agreement with theory.

The spiral-grooved journal bearing seems to possess the characteristics that make it a top candidate for all bearing system designs. For this reason design data is included in this paper for future design references.

EXPERIMENTAL APPARATUS AND INSTRUMENTATION

The basic elements of the test apparatus are shown in Figs. 1 through 3. Figure 1 depicts a 1.5 in. diameter journal, 8.75 in. long with 16 turbine blades on one end. The shaft also has two areas 1.5 in. long with spiral grooving. These two partially-grooved areas running with smooth sleeves comprise the test bearings. The bearing span is 5.5 in. The shaft may be lifted for start-up and shut-down by a hydrostatic lifter-loader bearing, Fig. 2, located between the two test bearings. This deep-pocketed partial-arc hydrostatic bearing is also used to load the rotating shaft upward. A nozzle ring in conjunction with the 16 blades on the shaft comprise an impulse-type drive turbine. Figure 3 provides an overall view of the test apparatus including the housing, housing rest and end plates. Each end plate has a carbon button, with one button being adjustable. These buttons provide a means for centering the shaft axially and for carrying a slight thrust load when necessary.

The shaft was driven to speeds of 60,000 rpm by the turbine drive. The speed was measured by a reflective method, i.e., an optical probe picks up the fluctuations of an alternately painted surface and the signal is indicated on a frequency meter, or counter.

The shaft displacement was measured by two horizontal and two vertical capacitance probes. Each test bearing has a set of two probes located inboard from the bearing itself, and mounted in the housing. The capacitance probes present a very clear picture on the oscilloscope and thus the total reading error is estimated at 10 microinch.

The lifter-loader bearing was calibrated by measuring the pocket pressure required to lift a certain dead-weight load while the shaft was non-rotating. Thus, the rotating shaft could be loaded by the hydrostatic bearing with a known load. Since the load-pressure calibration curve is an average curve obtained from a series of preliminary calibration tests, the load is estimated to be accurate within 4 percent. This calibration curve is shown in Fig. 4.

The entire test apparatus, including shaft and sleeves are made from AISI TY416 stainless steel. The sleeves have a 200μ-in. electrofilm coating on the bearing surface. The test bearings are designed for maximum radial stiffness at a bearing number, $\Lambda = 20$. The optimum proportions for this bearing with an L/D = 1 and with grooves on the rotating shaft are:

$$a_g/a_r = 0.54$$

$$h_g/h_r = 2.33$$

$$\beta = 25^o$$

$$\text{and } \bar{Y} = 0.46.$$

In order to provide the most theoretically perfect bearing, an infinite number of grooves is required. Because of the difficulty and cost of manufacturing spiral grooves, a compromise was made using thirty-six grooves. These grooves are of width .021 in. and axial length 0.396 in. The groove depth is between 500 to 600 μ-in. At zero speed the radial clearance is 495 μ-in. At 60,000 rpm the radial clearance has been calculated to be 435 μ-in. due to centrifugal growth. The sleeves were ground in line and have been measured to be less than 50 μ-in.out of line on the I.D. The shaft has a 30 μ-in. T.I.R. and an axial taper of 90 μ-in. between bearings. On the other hand, the sleeves are tapered 90μ-in. in the opposite direction of the shaft taper. The shaft was precision balanced to allow unbalance loading on the bearings of only fractions of a pound at 60,000 rpm.

EXPERIMENTS

Two particular experiments were performed:

a) load-displacement tests

and b) unloading of bearing to observe onset of stability.

The first series of tests provided steady state data at intermediate speeds up to 60,000 rpm. The second tests in which the bearings were completely unloaded to purposely try to cause a whirl instability showed positive results for the bearings, i.e. the bearings showed no sign of becoming unstable. Theoretically this bearing rotor system should be stable well above 60,000 rpm. At this speed, which corresponds to $\Lambda \sim 20$, the actual mass parameter is 0.2 (The critical mass parameter is 0.68 at 60,000 rpm).

THEORETICAL RESULTS

Fig. 5 shows typical comparisons of the total-load capacity and the radial stiffness versus bearing number between a lightly-loaded grooved journal (grooved member rotating, optimum proportions for $\Lambda = 20$) and a lightly-loaded plain journal bearing. The plain journal bearing has a total load capacity higher than the grooved bearing for Λ up to eight. At $\Lambda = 8$ both bearings are equal in total load capacity. For higher values of Λ the grooved bearing has a total load capacity higher than the plain journal bearing. The bearings are equal in radial stiffness at Λ approximately equal to "1.5" and "6". At values of Λ less than "1.5" the grooved bearing has a higher radial stiffness because its attitude angle is lower than that of the plain bearing. The grooved bearing has an attitude angle of 70 deg. at $\Lambda \to 0$ while the plain bearing has an attitude angle of 90 deg. at $\Lambda \to 0$. At values of Λ greater than "6" the grooved bearing has a higher radial stiffness than the plain bearing, which is already starting to level off to its asymptotic value of $\pi/2$.

Fig. 6 shows a stability map for the particular grooved bearing tested. A corresponding plot for an unloaded plain bearing does not exist, i.e., the plain bearing is unstable at all speeds. Notice that there are essentially two branches to this plot. The first branch, corresponds to the fractional-frequency whirl phenomenon, the second branch to pneumatic hammer. Along the first branch, the "critical mass parameter" decreases as Λ increases to a value of approximately 22, goes through a minimum point, and then increases until $\Lambda \sim 30$. At $\Lambda \sim 30$ the bearing is essentially infinitely stable. For $\Lambda > 30$, on this branch the "critical mass parameter" decreases rapidly and monotonically until $\Lambda \sim 42$. Beyond $\Lambda \sim 42$ the "pneumatic hammer" instability becomes predominant. (Instability occurs at a lower value of the critical mass.) This stability map is typical of grooved journal bearings. The criteria used in preparation of this data is based on the works of Ref. 8 and is summarized in the Appendix of this paper.

Fig. 7 represents a design chart of theoretical data for the design of the spiral-grooved journal bearing. This is included for future bearing design. It should be emphasized that the grooving proportions indicated on this chart are those proportions which give maximum radial stiffness at $\Lambda = 20$, for a bearing with an L/D = 1 and the grooved member rotating. If one is designing at other values of Λ and wishes to have the maximum radial stiffness possible at that particular Λ value (other than 20); then the grooving parameters would have to be changed accordingly. However, one should not expect more than approximately 20% improvement by varying the parameters. It is more important that the "number of grooves" be as large as possible in order to have the bearing behave as indicated by this design chart. Say, $\Lambda/2N < 1$.

If the design is to have the smooth member rotating, the steady state data of Fig. 7 can be used directly with little error. However, a stability map for this bearing has been prepared and is presented as Fig. 8. The method used for preparing this map is given in the Appendix.

EXPERIMENTAL-THEORETICAL DATA COMPARISON

Fig. 9 shows an experimental-theoretical data comparison of total radial stiffness for the two grooved bearings tested at various speeds to 60,000 rpm. Fig. 10 shows a comparison of the load at various speeds, and Fig. 11 gives an attitude angle comparison of theoretical and experimental data. As these curves are self-explanatory, no discussion is necessary. However, one point should be mentioned: the correlation is extremely sensitive to an accurate knowledge of the radial clearance. It is the most sensitive parameter and causes the greatest changes.

Fig. 12 gives a comparison of the shaft orbit size with speed. Fig. 13 shows a similar comparison at the same speeds but with a more precisely balanced shaft. Notice the importance of shaft balancing for high-speed running. Fig. 14 shows a comparison of the orbit size at both ends of the shaft at speeds of 50,000 and 60,000 rpm. And finally, Fig. 15 can be studied in conjunction with Fig. 14 to show the effect of extra-fine balancing. The main point that these orbit pictures bring out is that the shaft has not gone through any rigid body critical speed on its way up to 60,000 rpm. In other words, by rebalancing the shaft, the orbit sizes have been reduced, and there is no apparent enlarging and decaying of orbit size with increasing speed. On these photographs the total diametral clearance (radial play) of the bearing is represented by approximately 8.5 cm.

DISCUSSION AND CONCLUSIONS

Theoretically the spiral-grooved journal bearing appears to have excellent steady-state and stability characteristics. These characteristics have been observed and verified by experiment. Essentially the characteristics could be summarized as follows: the spiral-grooved journal bearing has a load capacity greater than the plain journal at $\Lambda > 8$, power loss less than the plain journal, and stability equivalent to the tilting -pad bearing. This bearing has two drawbacks. However, these will be overcome with time. They are the initial cost to set up a groove pattern, and the necessity to design a bearing-shaft alignment device.

A 1.5 in. diameter shaft has been driven by an air impulse-type turbine to 60,000 rpm. The shaft was mounted on two spiral-grooved, gas journal bearings, with an L/D=1 and a radial clearance at zero speed of 495μ-inch. The ambient gas was air at S.T.P. The bearings were loaded to a maximum of 12 lbs each at 40,000 rpm and 40 lbs each at 60,000 rpm. The eccentricity ratios at these conditions are respectively 0.28 and 0.63. Neither the loaded nor the unloaded bearing showed any signs of instability. A good correlation of theoretical and experimental data was found.

This particular test bearing-rotor system was designed from the theoretical design charts included in this report. It was designed to be a stable bearing system with all rigid body criticals above the top speed. The principal goals of these experiments were to collect steady-state data for comparison with theory and to prove by observation that the bearing-rotor system was stable for all speeds to 60,000 rpm. These goals were reached.

RECOMMENDATIONS

1. Extend theoretical steady-state data, and stability criteria if necessary, to large eccentricity ratios.
2. Design a spiral-grooved bearing-rotor system such that it would become unstable at a reasonable speed in order to permit experimental verification of the condition of marginal stability.
3. Collect large eccentricity experimental data and compare it with the theory developed in "1" above.
4. Determine theoretically and experimentally the critical speed behavior of this bearing.
5. Place the spiral-grooved bearing-rotor system on a shake table and observe the effects of dynamic loading at various frequencies and "g" levels.

ACKNOWLEDGEMENT

The author is grateful to Dr. C.H.T. Pan for his suggestions and criticisms, and to Mr. C.Y. Chow who has written the computer program from which the theoretical data had been generated.

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NOMENCLATURE

Symbol	Meaning
a_g	Groove width, in.
a_r	Ridge width, in.
C	Radial clearance, in.
D	Shaft diameter, in.
e	Radial displacement, in.
f_r	Frequency ratio of shaft whirl speed to shaft rotating speed, corresponding to the fractural frequency whirl phenomenon.
g	Gravitational constant, in./sec^2.
h_g	Groove clearance, in.
h_r	Ridge clearance, in.
K_r	Radial stiffness, lb/in.
K_t	Tangential stiffness, lb/in.
L	Bearing length, in.
m_o	Critical mass of rotor per bearing. lb.sec^2 /in.
M_c	Critical weight of shaft per bearing, lb.
N	Number of grooves per bearing.
p_a	Ambient pressure, psia.
R	Shaft radius, in.
T	Friction torque, in.lb.
W	Bearing load, lb.
W_T	Total bearing load for two bearings, lb.
$\overline{Y}$	Ratio of groove portion of bearing to L.

α'	Whirl ratio.
β	Spiral angle, deg.
Γ	Groove clearance to ridge clearance ratio.
$\varepsilon = e/C$	Eccentricity ratio.
$\Lambda = \frac{6\mu\omega}{p_a}\left(\frac{R}{C}\right)^2$	Bearing number.
μ	Absolute viscosity, lb.sec/in^2.
ϕ	Attitude angle, deg.
ω	Circular frequency of shaft, rad/sec.

APPENDIX

STABILITY CRITERIA FOR LIGHTLY-LOADED JOURNAL BEARINGS POSSESSING ROTATIONAL SYMMETRY*

The criterion requires an examination of the journal bearing steady-whirl characteristics. Accordingly, a critical mass of the rotor for the onset of instability may be determined at the points of neutral stability. Neutral stability points are such whirl frequency points (α_o') where the attitude angle (ϕ) or tangential force $\left(\frac{CK_t}{p_a LD}\right)$ vanishes and the centrifugal force due to whirl is exactly in equilibrium with the radial force component $\left(\frac{CK_r}{p_a LD}\right)$. Furthermore, an infinitesimal increase of the rotor mass from its critical magnitude (m_o) at the neutral stability point, would make the bearing-rotor system unstable if the derivative of $\frac{CK_t}{p_a LD}$ with respect to α' is negative, and conversely. A typical plot of a spiral-grooved journal bearing steady whirl characteristics is shown in Fig. 16.

The critical mass of the rotor is determined from the equation

$$m_o = \frac{1}{\omega^2(\alpha'_o)^2}\left(\frac{CK_r}{p_a LD}\right)\left(\frac{p_a LD}{C}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (A.1)$$

The critical mass parameter used in constructing Figs. 6, 7 and 8 is defined as:

$$\frac{Mc/g}{LD}\left(\frac{C}{R}\right)^5\left(\frac{Rp_a}{\mu^2}\right) = \frac{36}{\Lambda^2}\,\frac{1}{(\alpha'_o)^2}\,\frac{CK_r}{p_a LD} \quad \ldots\ldots\ldots\ldots\ldots \quad (A.2)$$

According to Ref. 7, also as shown in Fig. 16, the steady whirl data for a spiral-grooved journal bearing with smooth member rotating can be prepared from the data obtained for a bearing in which the grooved member is rotating, and vice-versa. Given the data for one type bearing, the data for the other type bearing is obtained as follows: the radial stiffness is the exact mirror image with the pivot point being

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*This appendix is a precis of the stability theorem previously given in Ref. 8.

$\alpha' = 0.5$, and the tangential stiffness is the negative mirror image with the pivot point being $\alpha' = 0.5$, i.e. the mirror image, plus an inversion about $\alpha' = 0.5$.

Examples (using Fig. 16)

	α'_o	$\frac{CK_r}{p_a LD}$	$\frac{Mc/g}{LD}\left(\frac{C}{R}\right)^5 \frac{Rp_a}{\mu^2}$	Slope	Stability Obtained With
Grooved Member Rotating	0.445	1.505	0.684	neg.	$M < M_c$
Smooth Member Rotating	0.555	1.505	0.440	neg.	$M < M_c$

LIST OF FIGURES

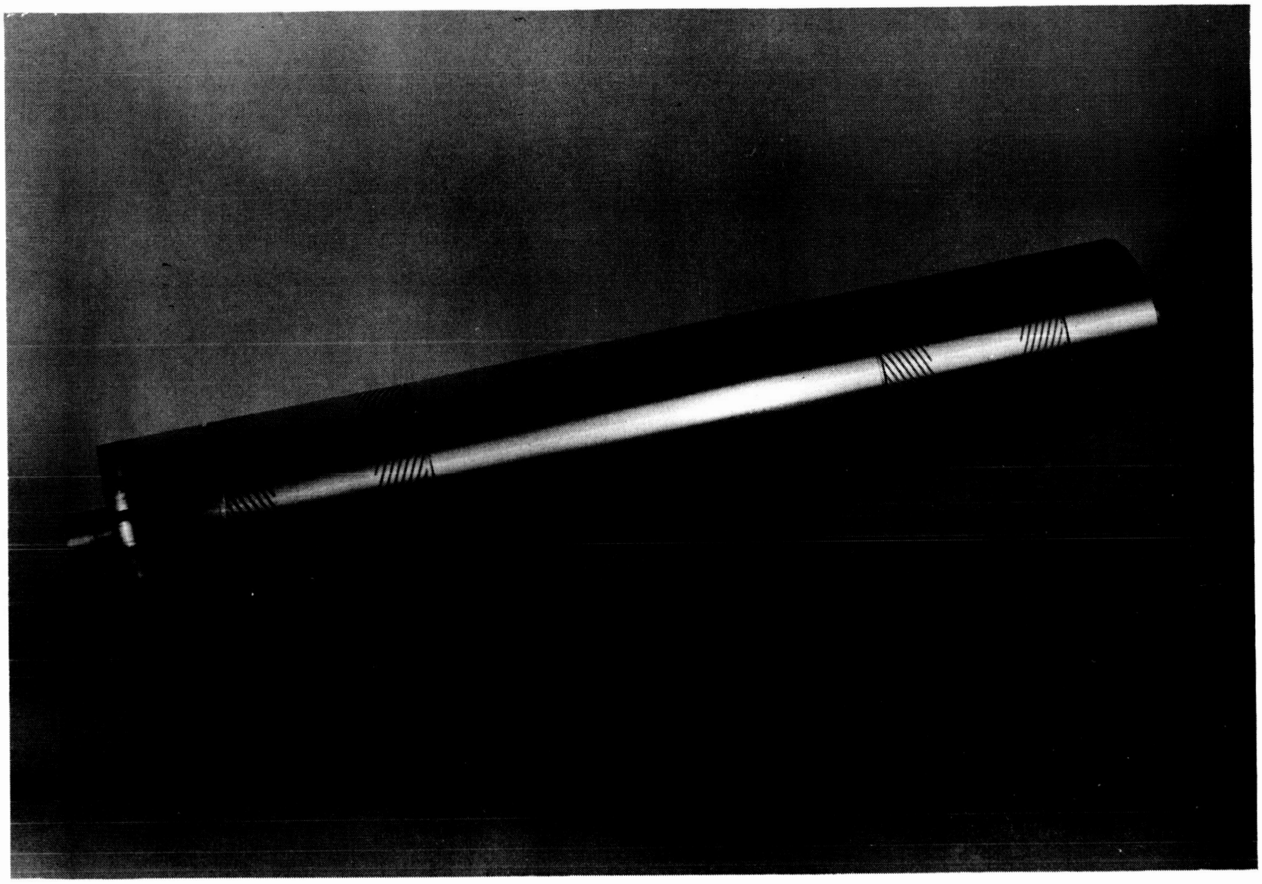


Fig. 1 Shaft-Illustrating Spiral Grooving and Turbine Blades

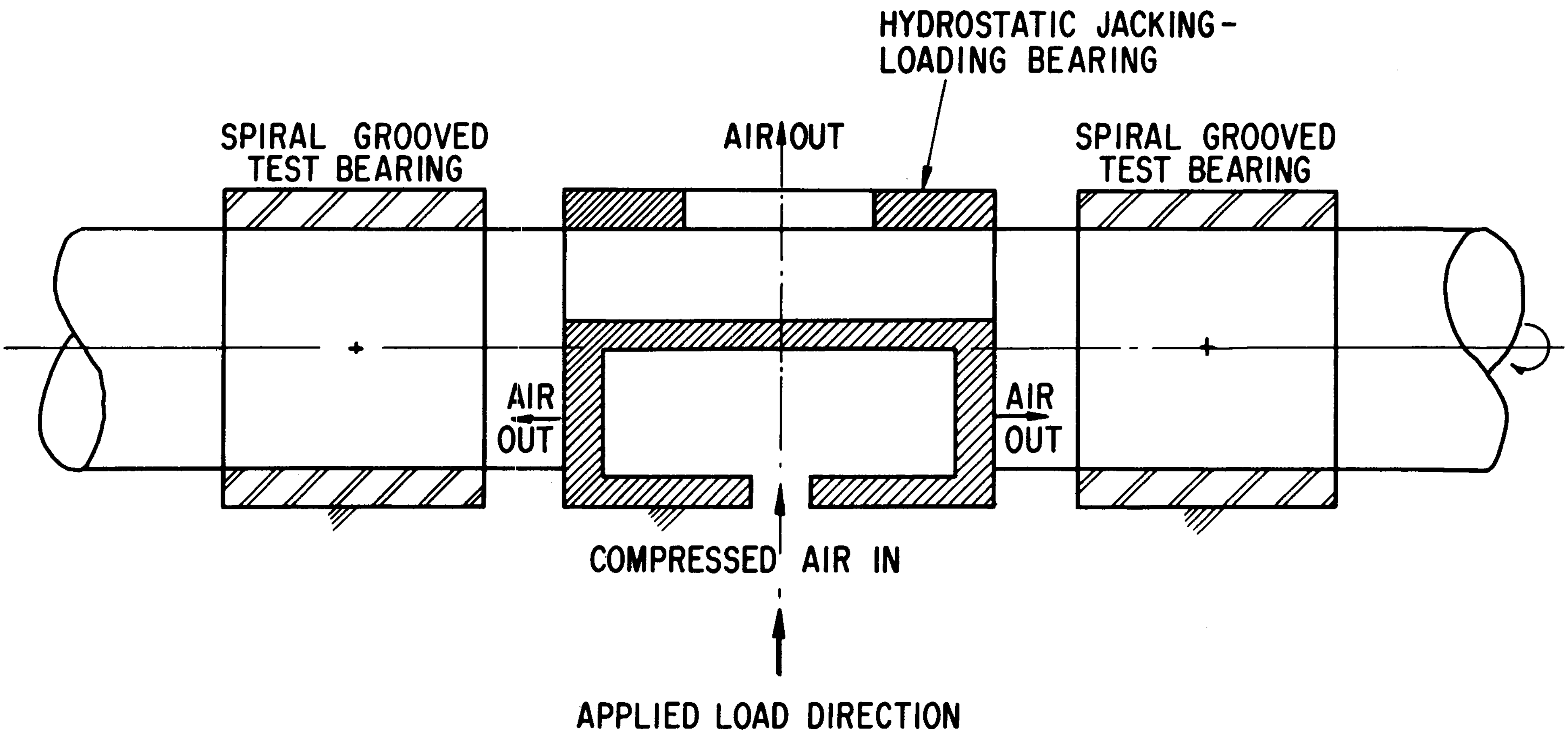


Fig. 2 Sketch of Test Bearings and Hydrostatic Jacking-Loading Bearing

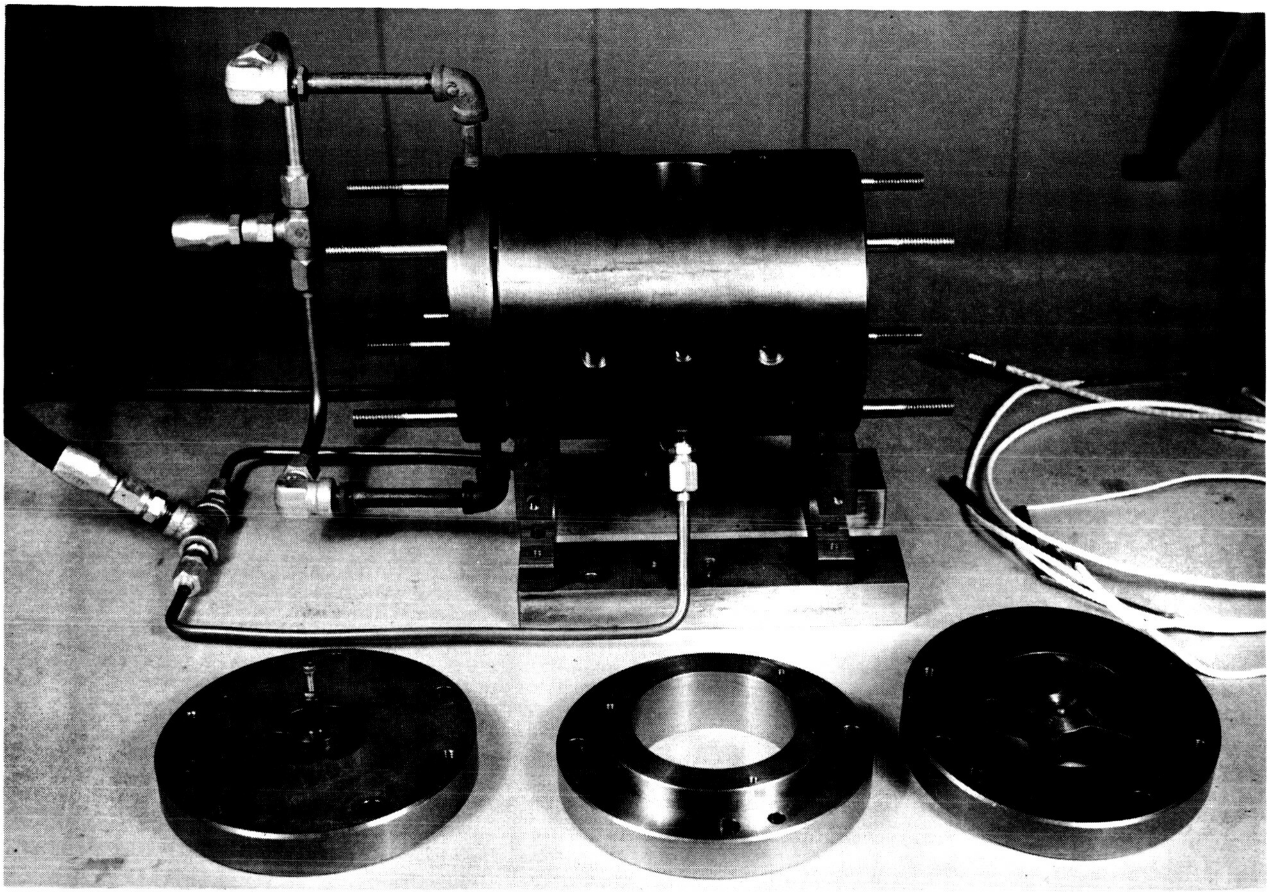


Fig. 3   Display of Housing, Housing Mounts, and End Plates

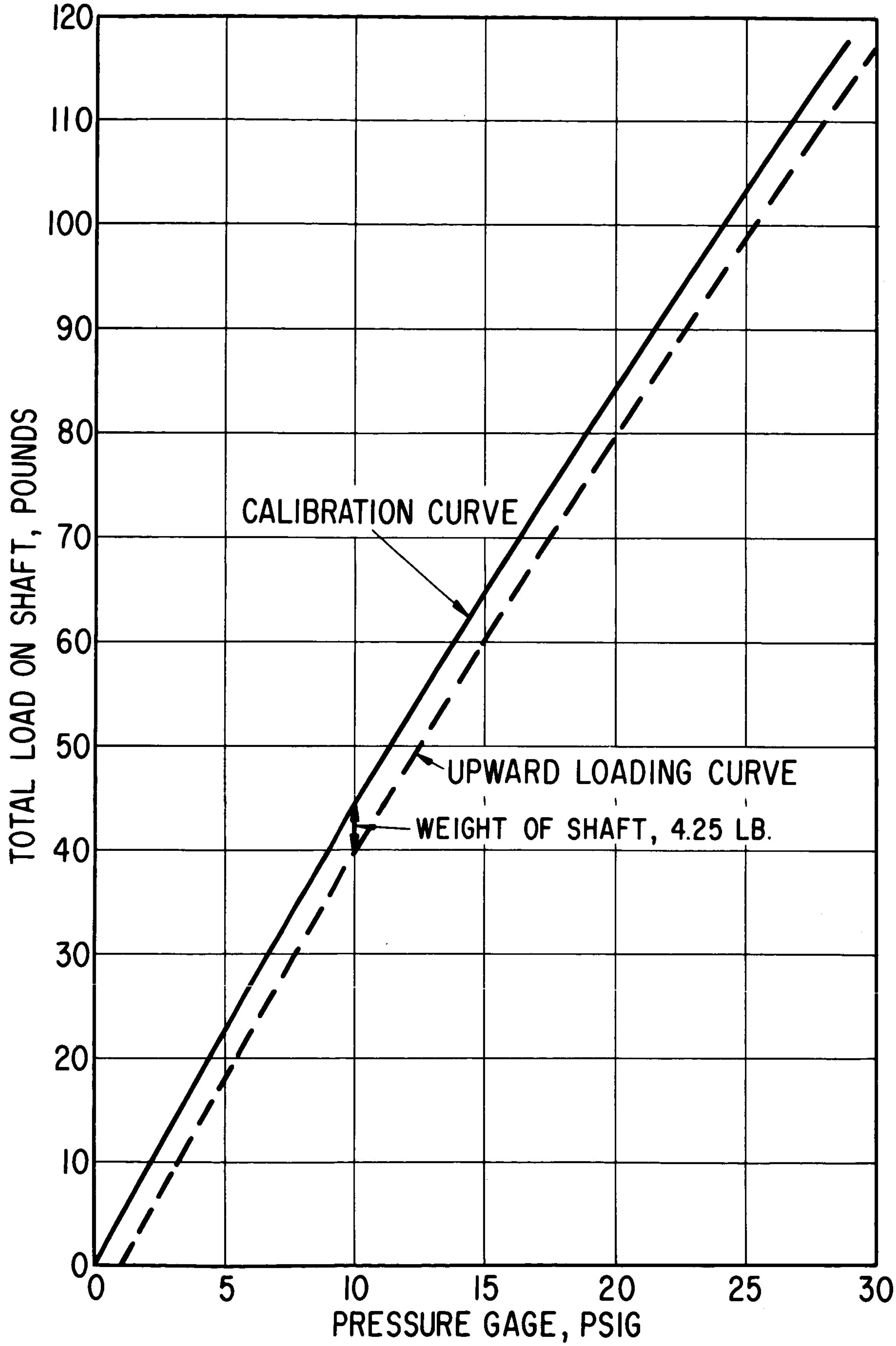


Fig. 4 Calibration Curve and Upward Loading Curve for Loading Bearing

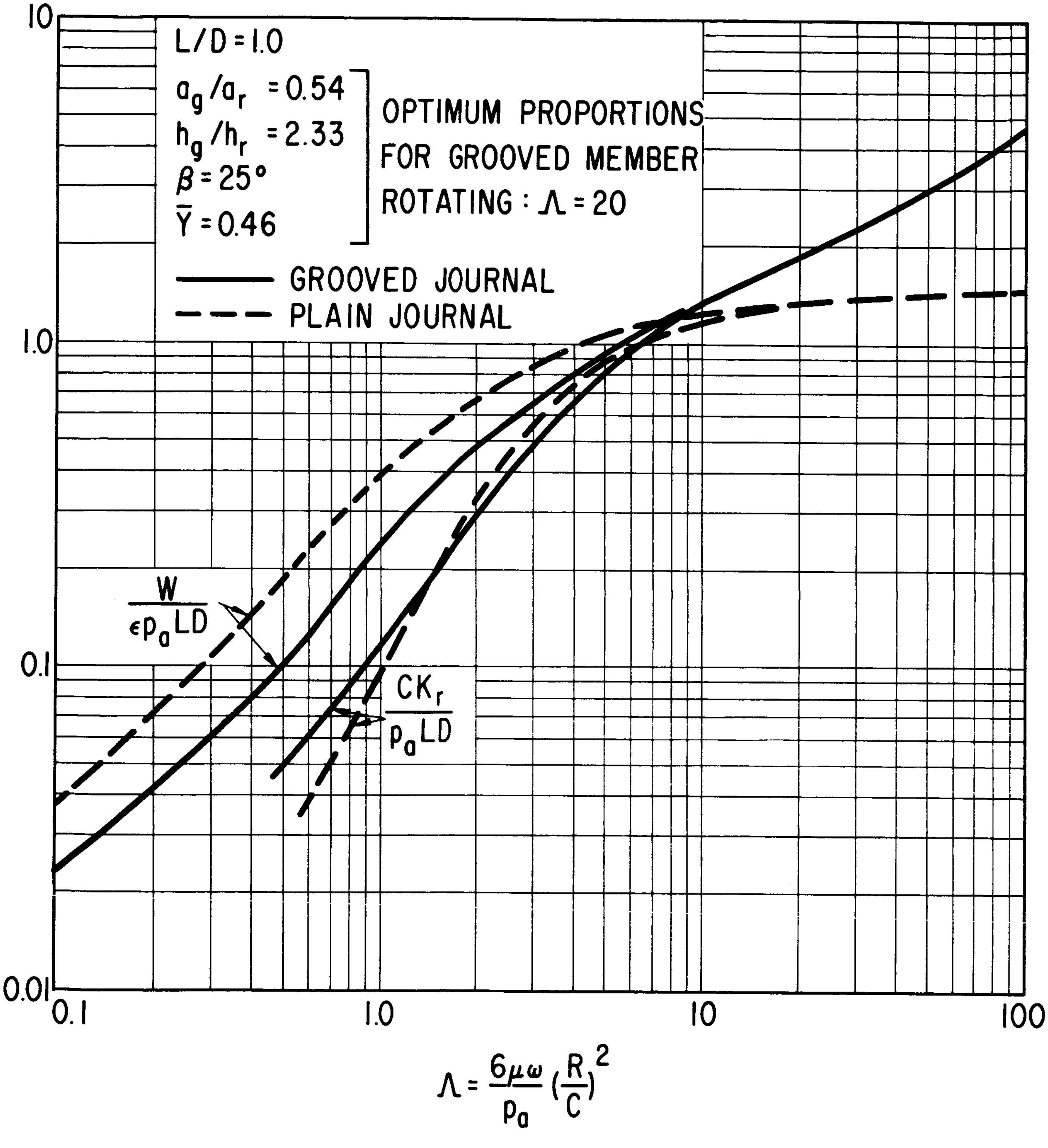


Fig. 5 Total Load and Radial Stiffness Versus Bearing Number

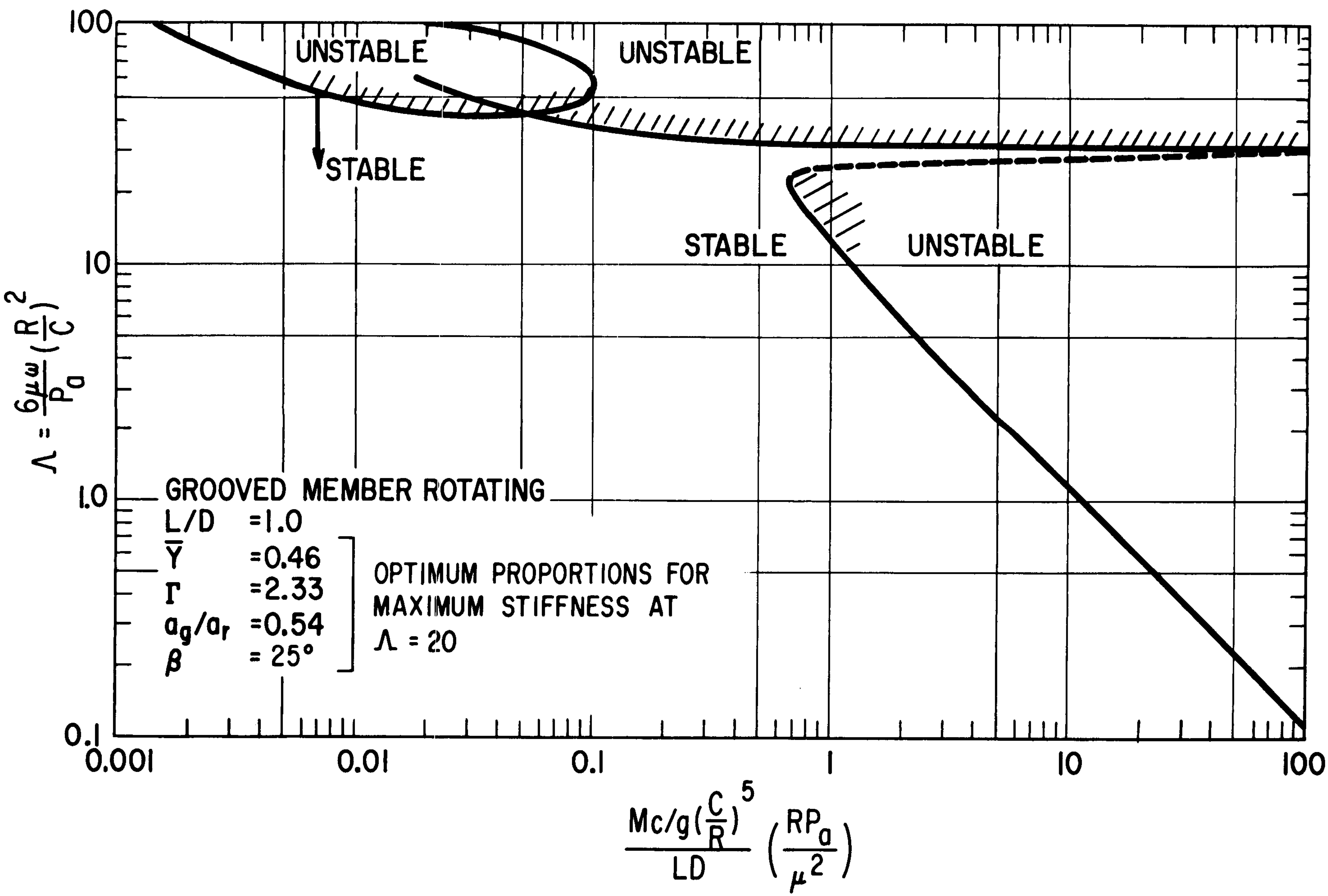


Fig. 6 Critical Mass Parameter Versus Bearing Number for Spiral-Grooved Journal Bearing - Grooved Member Rotating

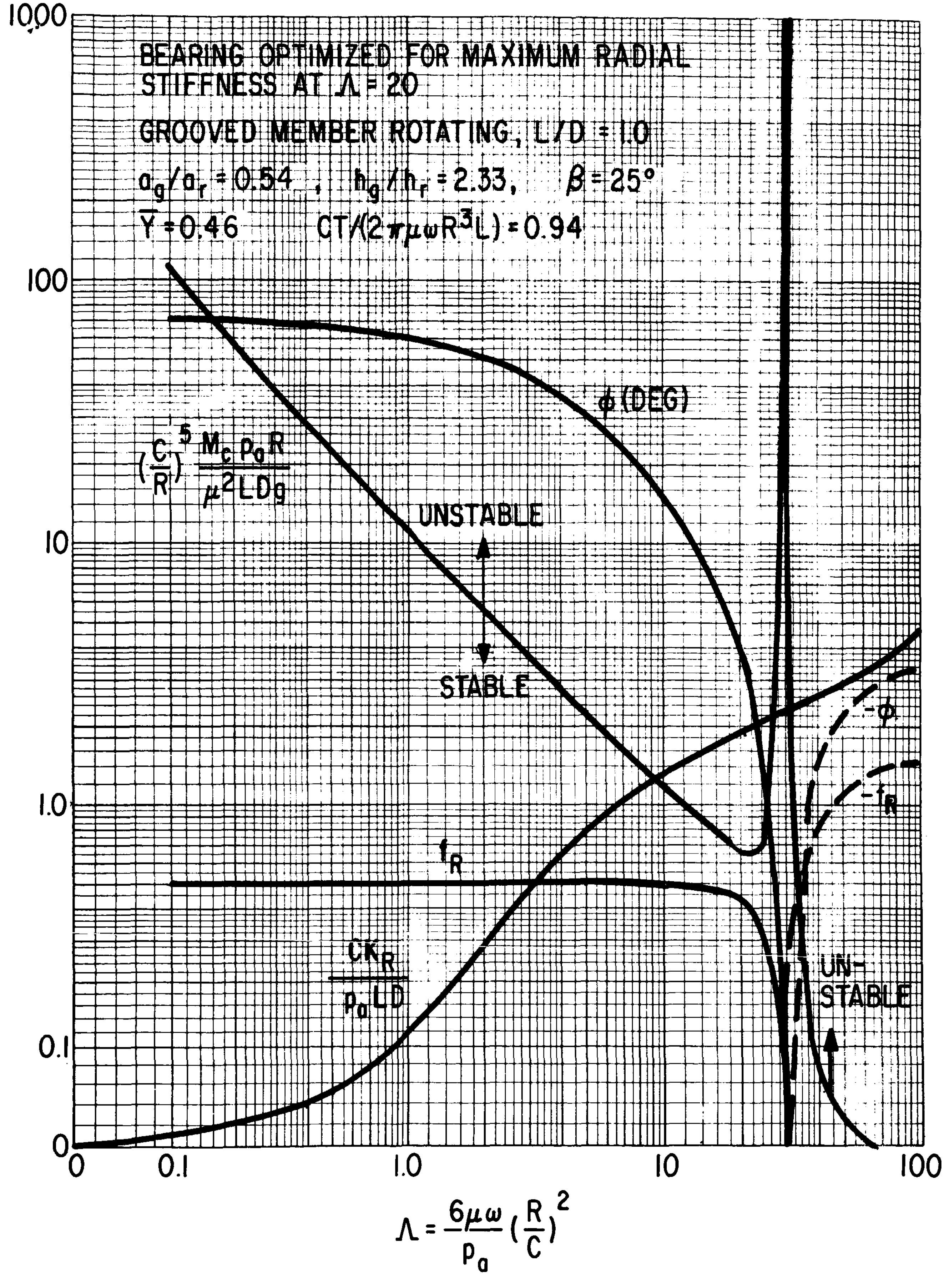


Fig. 7 Grooved Bearing Design Chart

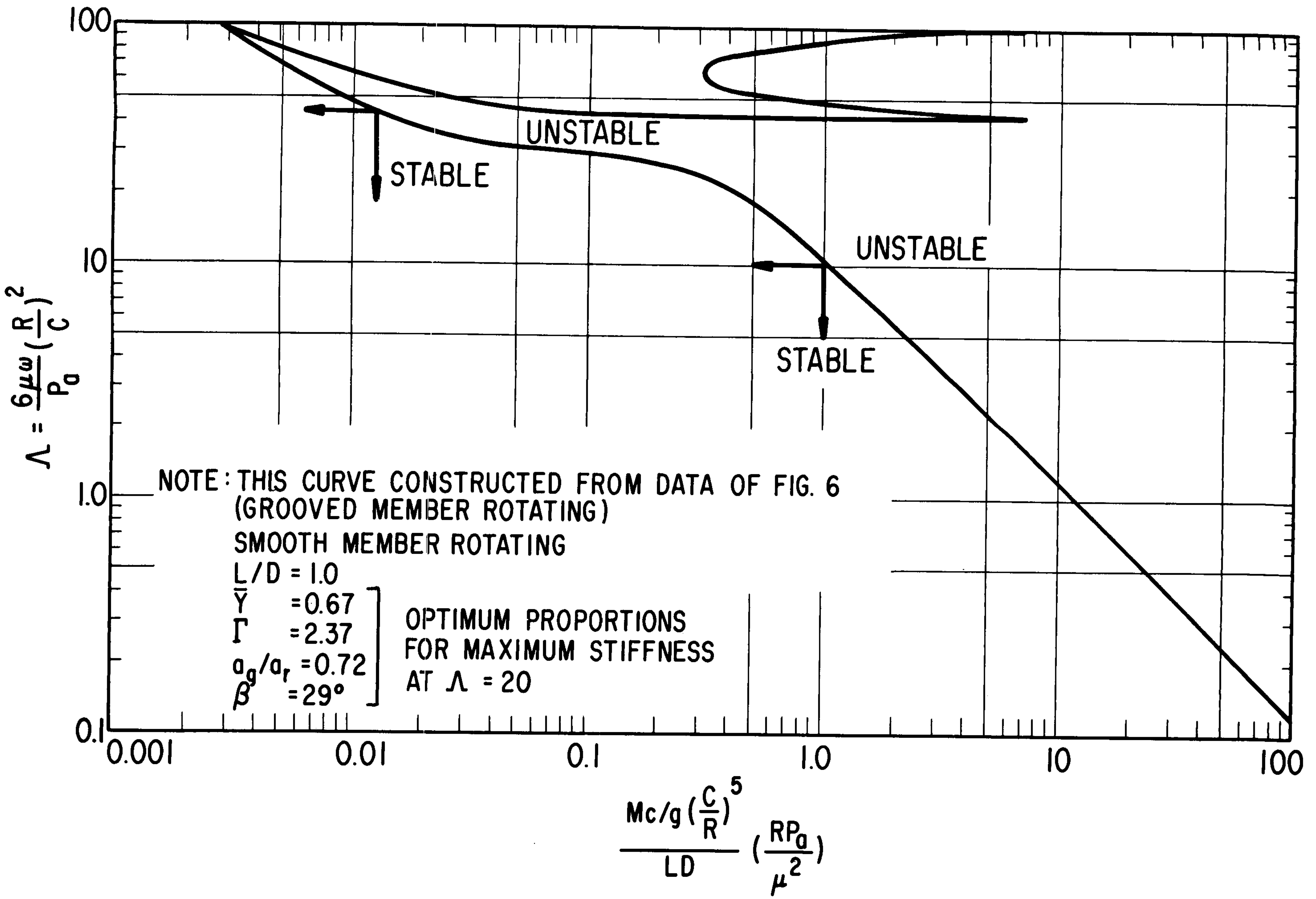


Fig. 8 Critical Mass Parameter Versus Bearing Number for Spiral-Grooved Journal Bearing - Smooth Member Rotating

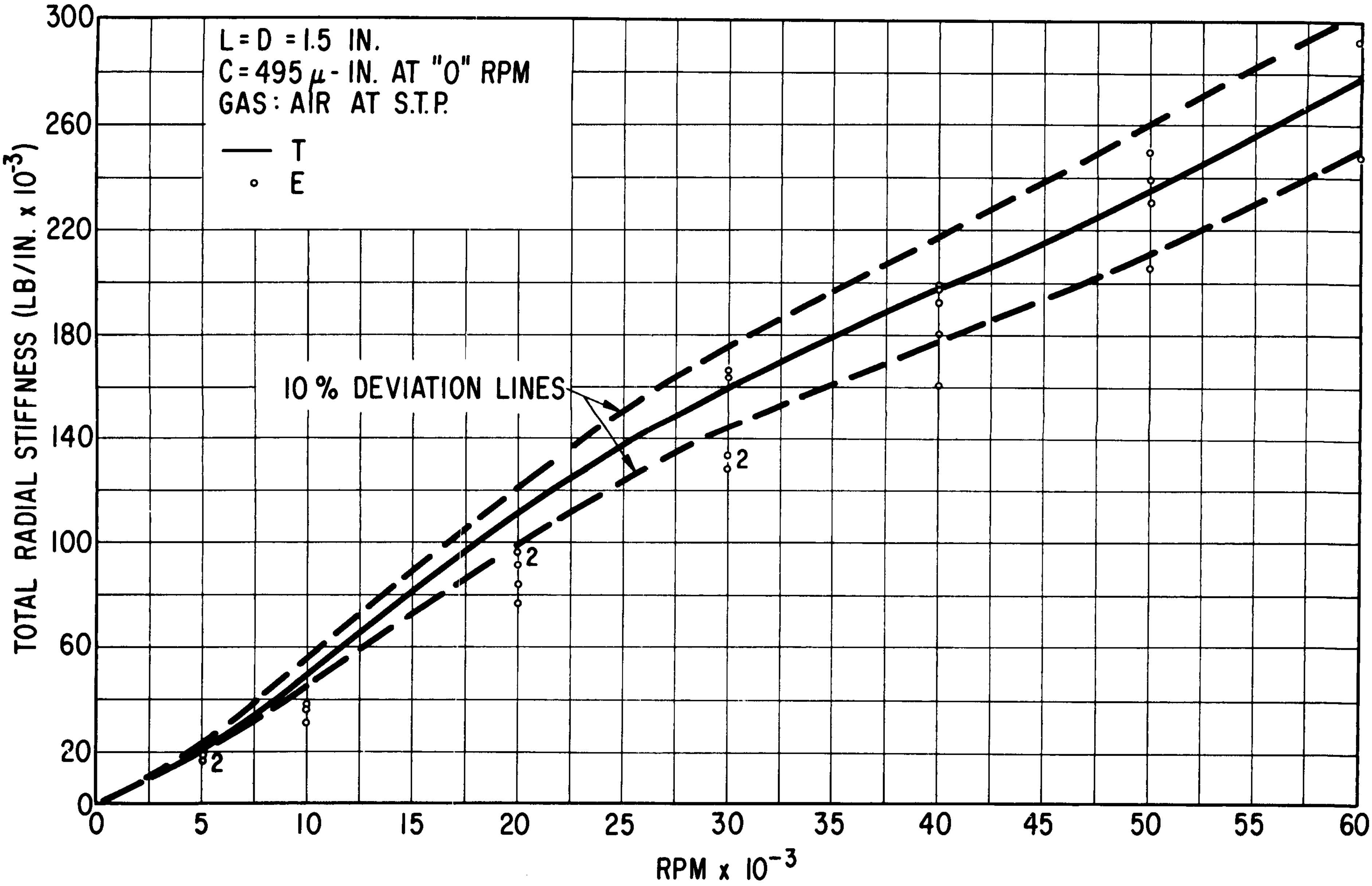


Fig. 9 Comparison of Theory and Experiment - Total Radial Stiffness Versus Speed

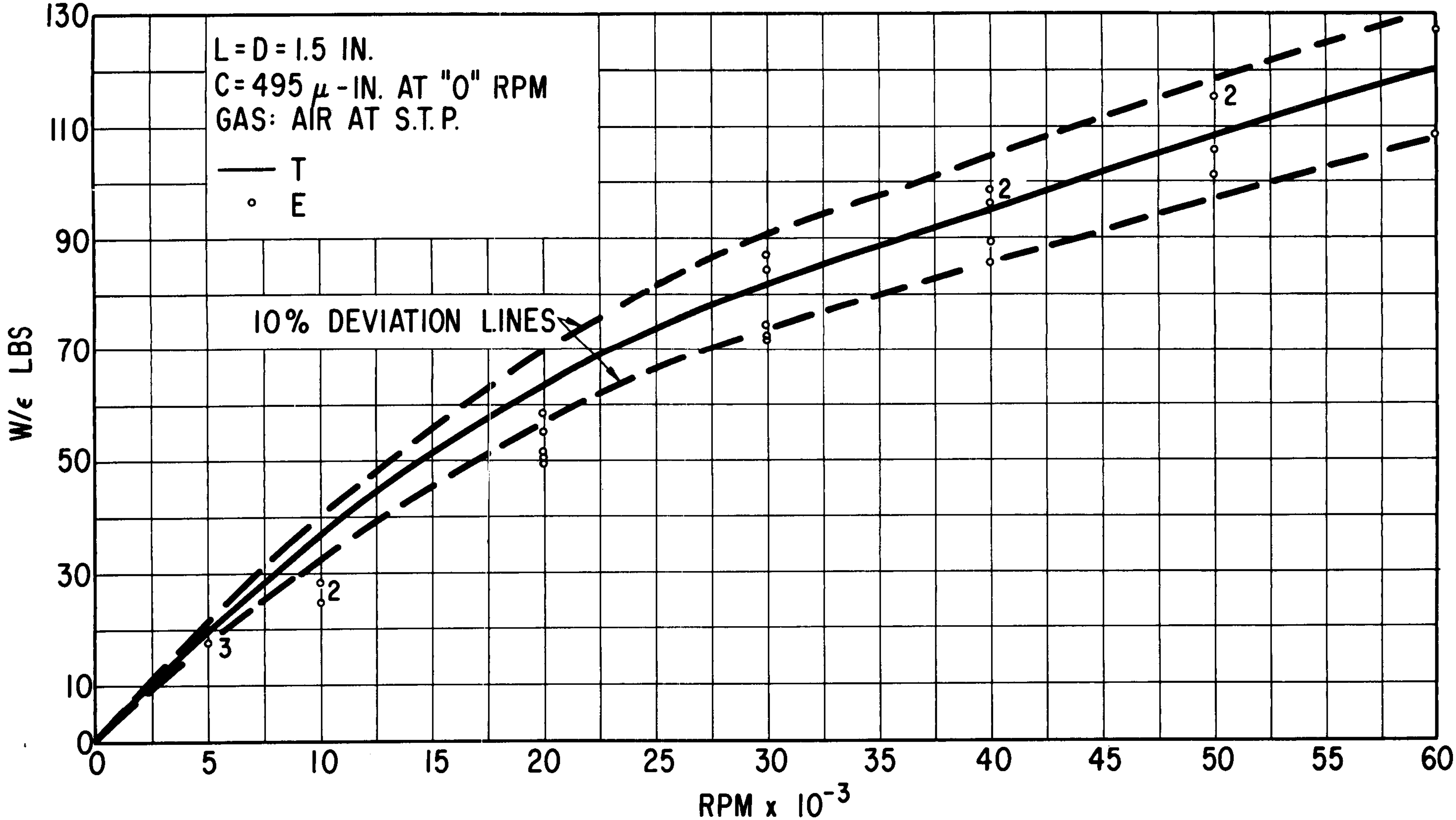


Fig. 10 Comparison of Theory and Experiment - (Total Load) / (Eccentricity Ratio) Versus Speed

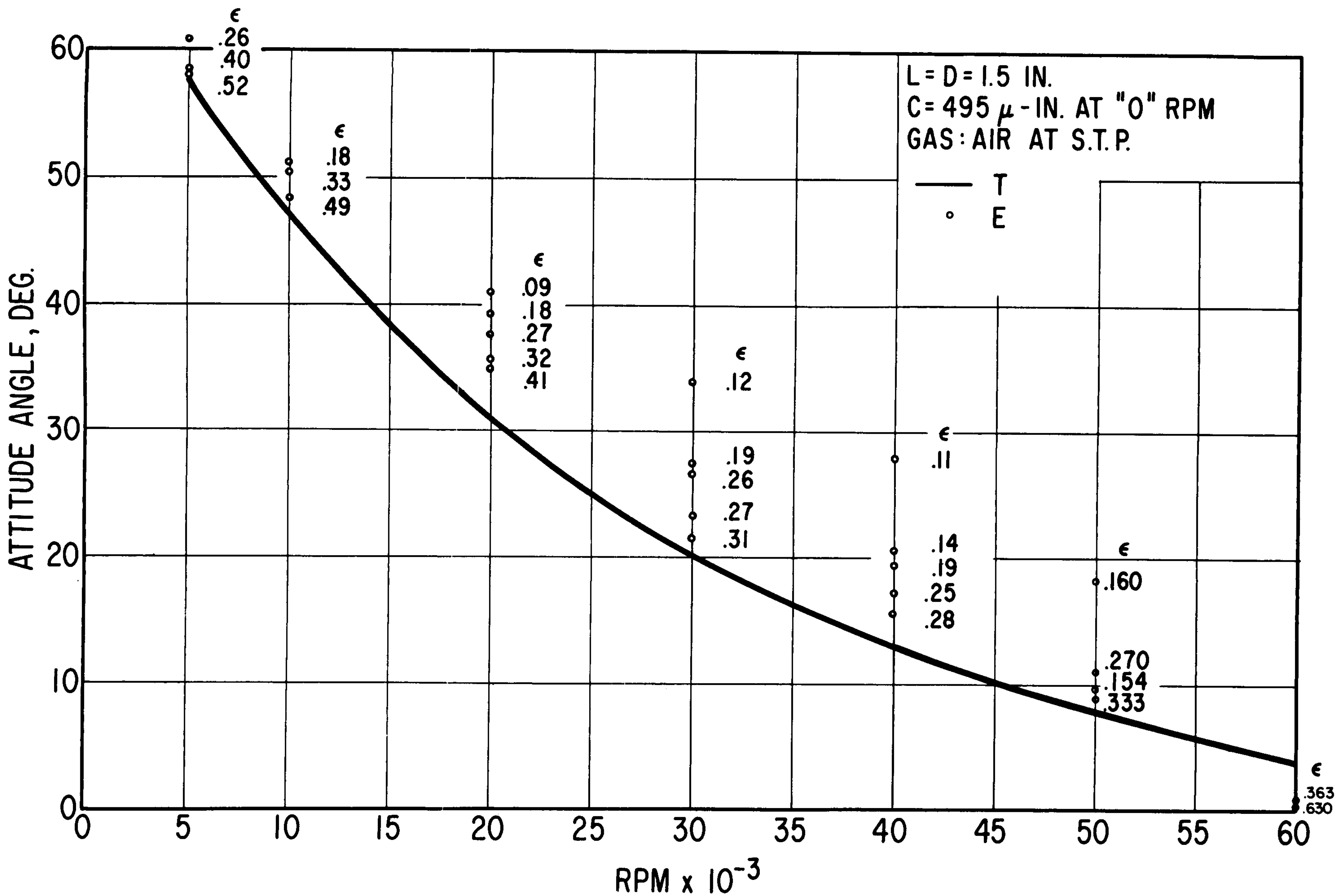
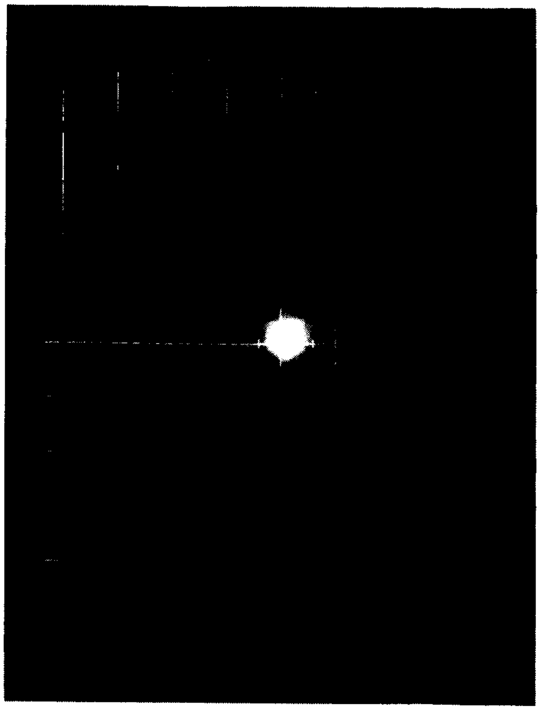
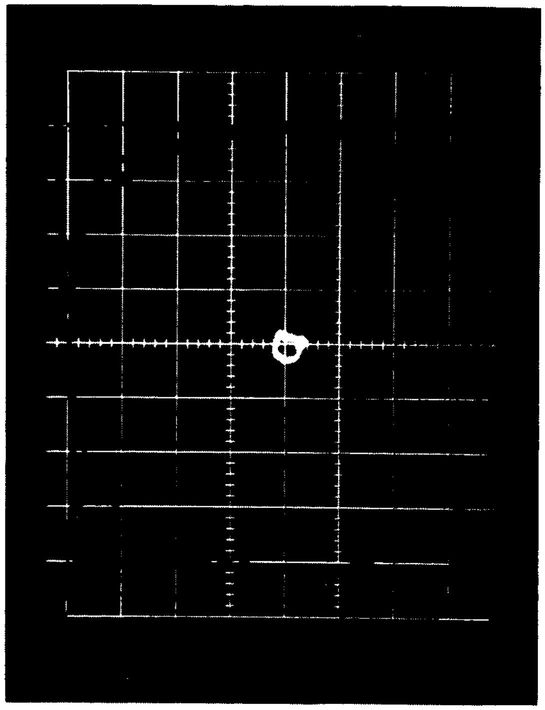


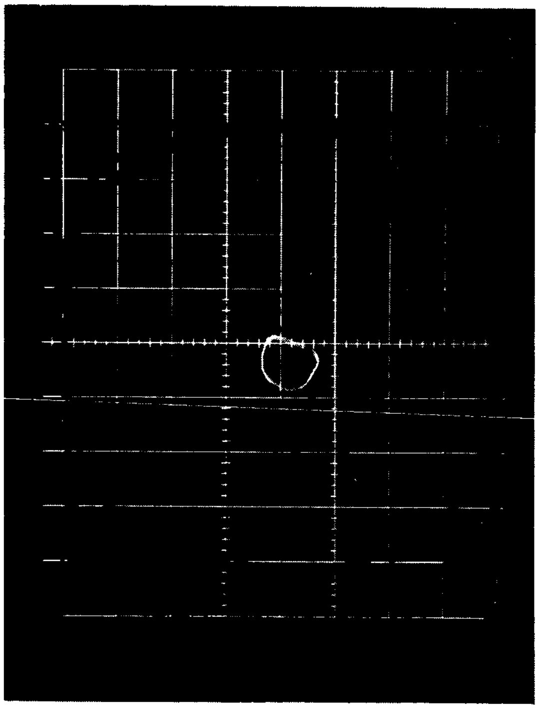
Fig. 11 Comparison of Theory and Experiment - Attitude Angle Versus Speed



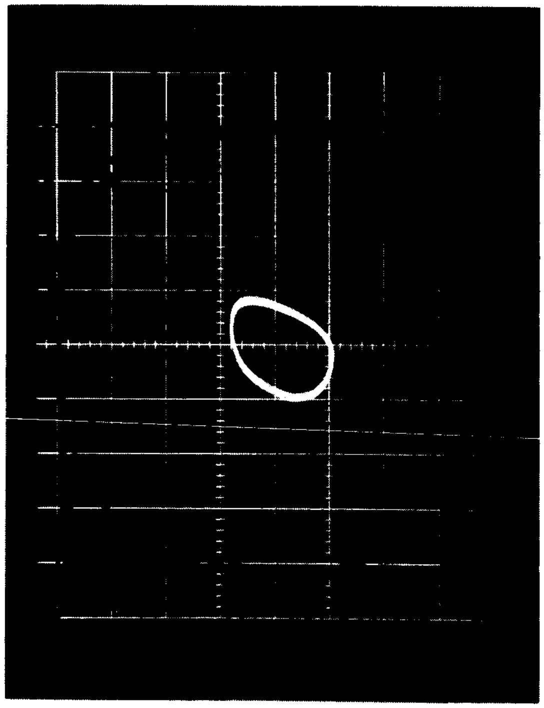
5,000 RPM



20,000 RPM



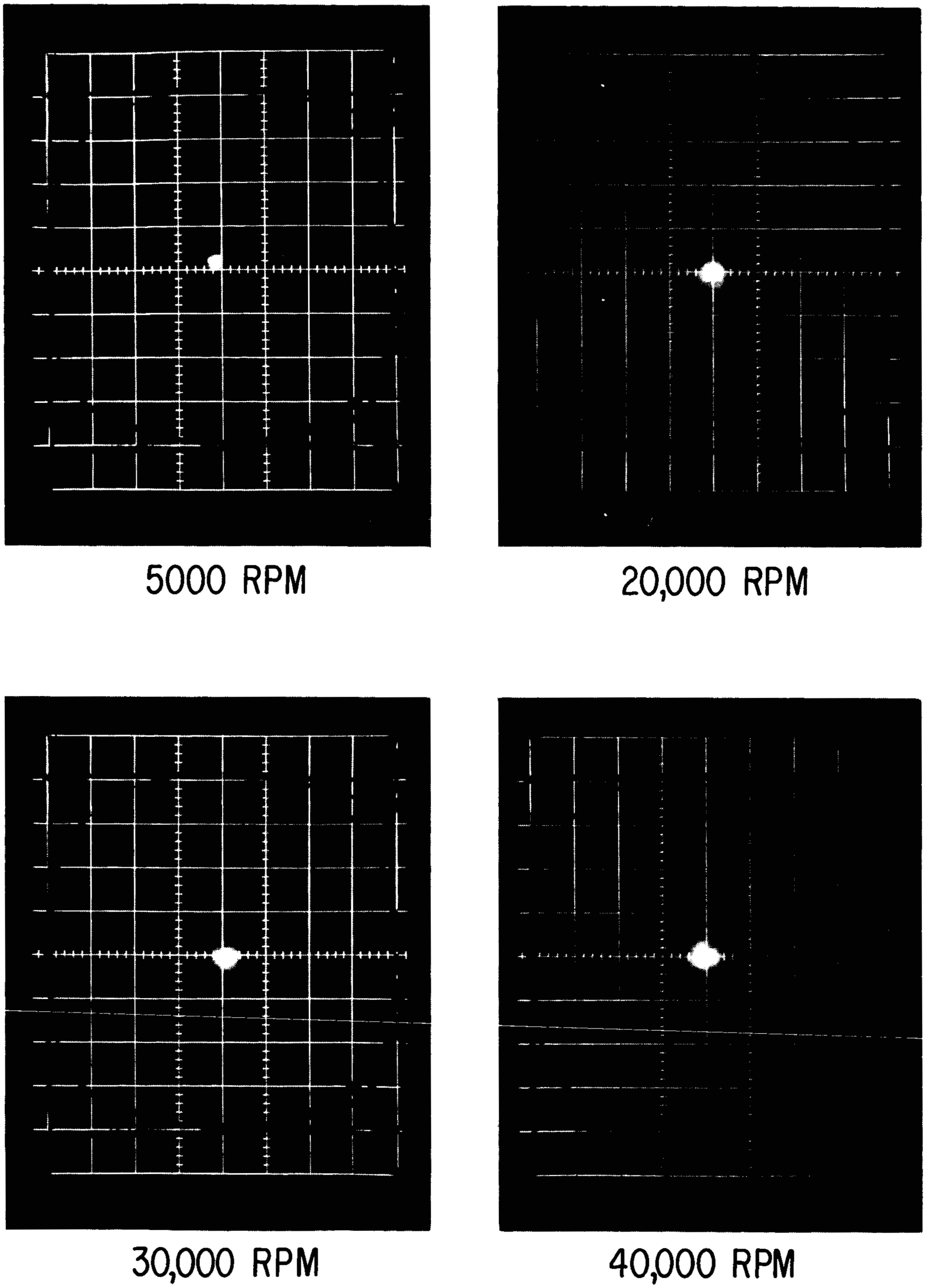
30,000 RPM



40,000 RPM

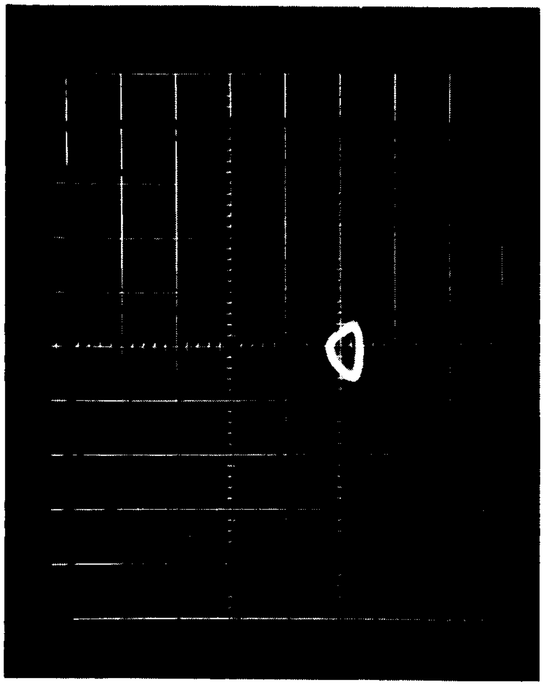
Comparison of Orbit Sizes: Coarse Balancing

Fig. 12 Comparison of Orbit Sizes - Coarse Shaft Balancing (5,000 to 40,000 rpm)

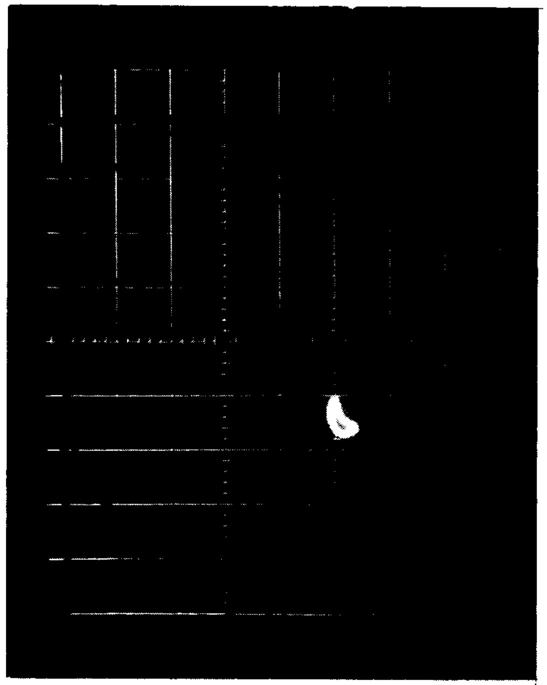


Comparison of Orbit Sizes: Fine Balancing

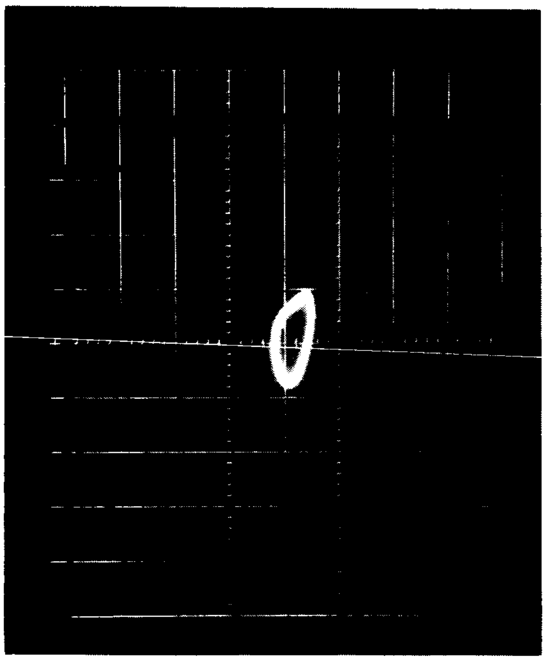
Fig. 13 Comparison of Orbit Sizes - Fine Shaft Balancing (5,000 to 40,000 rpm)



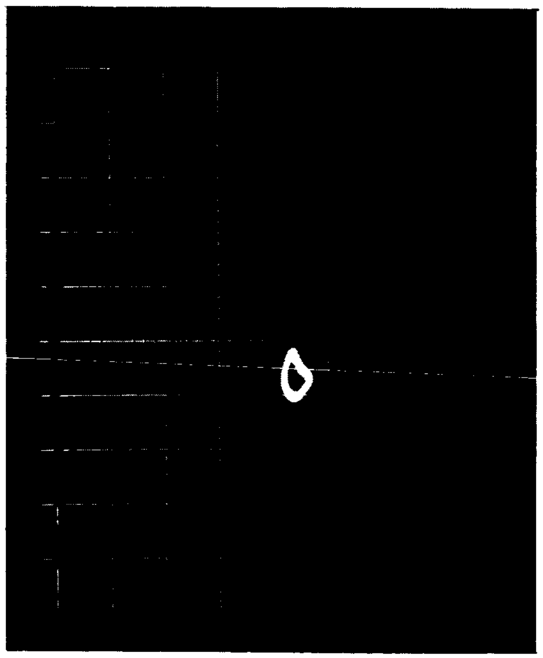
THRUST END
50,000 RPM



TURBINE END
50,000 RPM



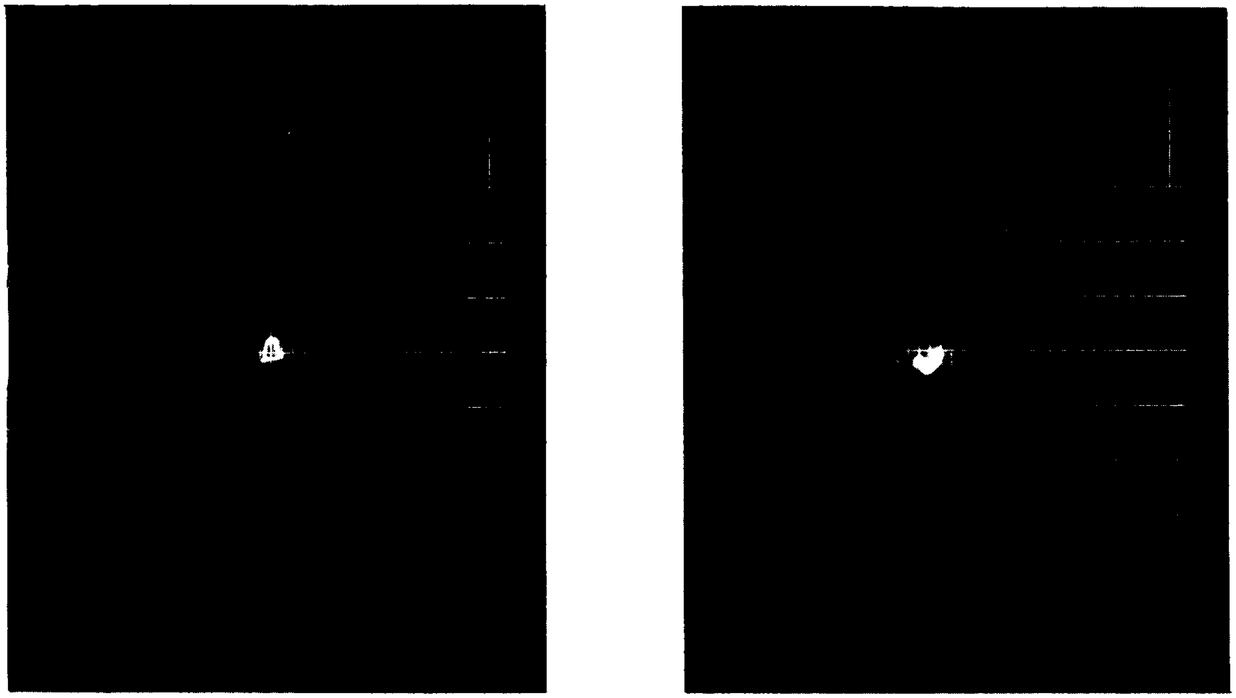
THRUST END
60,000 RPM



TURBINE END
60,000 RPM

Comparison of Orbit Sizes: Fine Balancing

Fig. 14 Comparison of Orbit Sizes - Fine Shaft Balancing (50,000 and 60,000 rpm)



50,000 RPM 60,000 RPM

Orbit Sizes: Extra Fine Balancing

Fig. 15 Comparison of Orbit Sizes - Extra Fine Shaft Balancing (50,000 and 60,000 rpm)

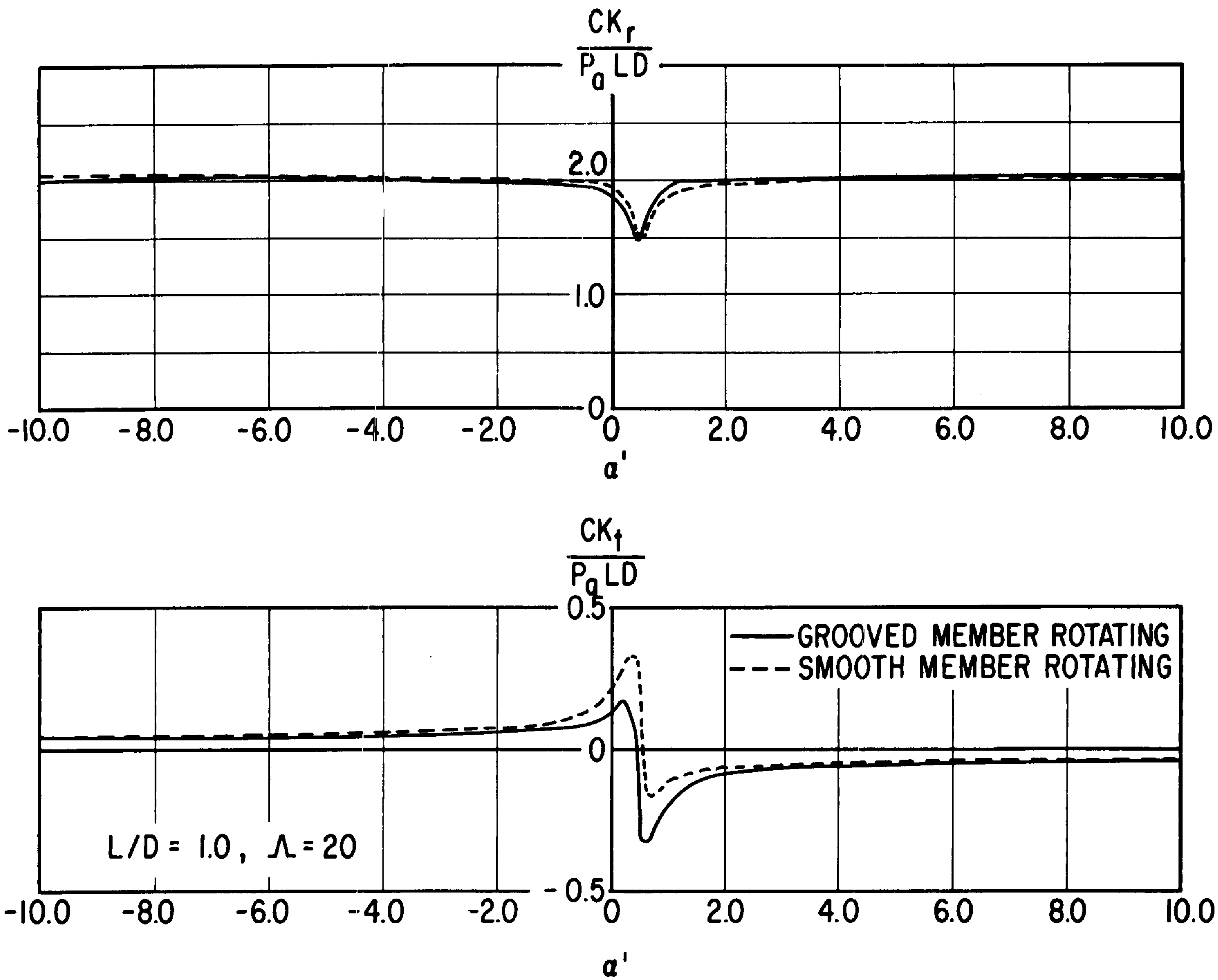


Fig. 16 Spiral-Grooved Journal Bearing Steady Whirl Characteristics

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