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EFFECT OF NUMBER OF LOBES AND LENGTH-DIAMETER RATIO ON STABILITY OF TILTED-LOBE HYDRODYNAMIC JOURNAL BEARINGS AT ZERO LOAD

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16. Abstract

Hydrodynamic journal bearing stability tests were conducted with 3.8-centimeter- (1.5-in.-) diameter tilted-lobe bearings. The bearings had three, five, and seven lobes and length to diameter (L/D) ratios from 0.2 to 1.0. They were tested in water and MIL-L-7808G oil at 294 K (70° F) at speeds to 5400 rpm with zero load. Stability was not appreciably affected by the number of lobes and decreased with a decrease in L/D ratio. However, a three-tilted-lobe bearing with an offset factor of 0.76 and an L/D of 0.5 was more stable than a three centrally lobed bearing with an offset factor of 0.50 and an L/D of 1.0. All data can be presented in terms of the parameter $\overline{M}(\Gamma^*)^{1.42} = 8.6$, where $\overline{M}$ is the dimensionless mass parameter and Γ^* is the optimized stability parameter.

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SUMMARY

A series of stability tests was conducted with 3.8-centimeter- (1.5-in.-) diameter tilted-lobe hydrodynamic journal bearings in water and MIL-L-7808G oil at 294 K (70° F) at speeds to 5400 rpm with zero load. Bearings with three, five, and seven tilted lobes and length to diameter (L/D) ratios from 0.2 to 1.0 were tested. The number of lobes did not appreciably affect the stability of the tilted-lobe bearings regardless of L/D ratio. Stability generally decreased with a decrease in L/D ratio for each configuration tested. However, a three-tilted-lobe bearing with an offset factor of 0.76 and an L/D of 0.5 was more stable than a three centrally lobed bearing with an offset factor of 0.50 and an L/D of 1.0. The useful range of the stability curves obtained from tests in water was extended by incorporating data from these same bearings run in oil. The tests in oil also confirmed the validity of the dimensionless mass and speed parameters which are used to determine stability. All data can be presented in terms of the parameter $\overline{M}(\Gamma^*)^{1.42} = 8.6$, where $\overline{M}$ is the dimensionless mass parameter and Γ^* is the optimized stability parameter.

INTRODUCTION

Investigations of the stability characteristics of various fixed geometry journal bearing configurations have shown that the most stable is the tilted-lobe bearing (refs. 1 to 6). There are applications in gas turbine engines where this type of bearing has been extremely successful. Reference 7 describes tests run on a three-tilted-lobe journal bearing which replaced a rolling element bearing in the compressor and turbine positions of a gas turbine engine. Full scale tests demonstrated vibration-free performance at 19 400 rpm with a 7.78-centimeter- (3.06-in.-) diameter shaft. In certain applications the tilted-lobe bearing can replace roller bearings to prevent catastrophic machine failures. When a roller bearing failure occurs, the rolling elements of the bearing can cause complete engine destruction, such as loss of compressor or turbine blades, which results from the loss of shaft position. Journal bearing failures are much less severe.

It would be highly desirable, in certain instances, to replace a rolling element bearing with a tilted-lobe bearing occupying the same space as the original bearing. This would mean that journal bearings of low length to diameter ratio would be required. Very little, if anything, is presently known about the effect of either length to diameter ratio or the number of lobes on the stability of tilted lobe bearings.

In the investigation reported herein, an attempt is made to fill this void by running stability tests on tilted-lobe bearings of three different numbers of lobes (three, five, and seven) and four different length to diameter ratios (1.0, 0.75, 0.50, and 0.20). Most of the tests were run with water as the lubricant. Some were run in MIL-L-7808G turbine oil. A comparison is made between the data obtained with both lubricants.

The objectives of this study were to (1) observe the effect of number of lobes on bearing stability, (2) determine the effect of length to diameter ratio on the stability of tilted-lobe bearings, and (3) test the validity of the stability curves for tilted-lobe bearings obtained from water tests over a wider range by substituting oil as the lubricant.

The test bearings had a nominal 3.8-centimeter (1.5-in.) diameter. They were submerged in water or oil at an average temperature of 294 K (70° F) and were operated hydrodynamically at zero load. The speed was increased to the onset of nonsynchronous whirl. The maximum journal speed was 5400 rpm and was limited by whirl.

SYMBOLS

- C bearing radial clearance, $R_{PC} - R$, mm (in.)
- C_D bearing diametral clearance, mm (in.)
- C_N capacity number, $S(L/D)^2$
- D bearing inside diameter, cm (in.)
- g gravitational constant, m/sec^2 ($in./sec^2$)
- L bearing length, cm (in.)
- L_1 lobe arc length, cm (in.)
- L_2 length of arc from leading edge of a lobe to a line passing through the lobe and journal centers at zero eccentricity, cm (in.)
- M rotor mass per bearing, W_r/g, kg ((lb)(sec^2)/in.)

$\overline{M}$ dimensionless mass parameter, $MP_a(C/R)^5/2\mu^2L$

N' journal speed, rps

N_W journal fractional frequency whirl onset speed at zero load, rpm

P' unit pressure on projected area, N/m^2 (psi)

P_a atmospheric pressure, N/m^2 abs (psia)

R journal radius, cm (in.)

R_P radius of lobe, cm (in.)

R_{PC} radius of pitch circle, cm (in.)

ΔR_L leading edge entrance wedge thickness, mm (in.)

S Sommerfeld number, $(\mu N'/P')(D/C_D)^2$

W_r total weight of test vessel, N (lbf)

α offset factor, L_2/L_1

Γ dimensionless speed parameter, $6\mu\omega R^2/P_aC^2$

Γ^o stability parameter, $\Gamma(D/L)$

Γ^* optimized stability parameter, $\Gamma(D/L)^{0.66}$

θ lobe wedge angle, deg

μ lubricant dynamic viscosity, (N)(sec)/m^2 ((lb)(sec)/in.^{2})

ω journal angular speed, rad/sec

APPARATUS

Test Bearings

Sector bearings having three, five, and seven tilted lobes and length to diameter ratios (L/D) from 0.2 to 1.0 were mounted in a solid housing. Figure 1 shows the geometry of one sector which forms a lobe. The lobes for each bearing were contour-milled according to the geometry shown in figure 1. The arc from the leading edge of each lobe to a point A (shown in fig. 2) formed a converging wedge with the journal outside diameter. The inside surface of the sector from point A to the trailing edge of the sector was concentric with the outside diameter of the journal. This resulted in bearings that produced a wholly convergent film portion followed by a constant film thickness portion in operation. This film contour differs somewhat from those of bearings tested in reference 2. In reference 2, some of the bearing sectors were tilted to result in a wholly converging wedge over the complete arc of the sector so that the minimum clearance occurred at the trailing edge of each sector when the journal was centrally located within the bearing. The remaining bearing sectors were tilted so that the minimum film thickness occurred at a point 60 percent of the arc length from the leading edge of each sector. This resulted in a converging-diverging wedge, which approximates the geometry of a centrally lobed bearing.

The offset factor α and lobe wedge angle θ (fig. 2) varied from 0.69 to 0.82 and 0.096° to 0.154°, respectively (table I). The various minimum radial clearances C (table I) were obtained by varying the outside diameter of the journals for each bearing tested. Circumferential profile traces were made of the internal surface of each sector in each bearing assembly in three radial planes along the length of the bearing to obtain an average leading edge entrance wedge thickness ΔR_L and arc lengths L_1 and L_2 from which the lobe wedge angle θ and offset factor α were calculated. Typical surface profile traces are shown in figure 3, which illustrates how L_1, L_2, and ΔR_L were obtained. Values for ΔR_L, α, and θ are listed in table I for each bearing tested.

The assembled bearings in all cases had a nominal 3.8-centimeter (1.5-in.) diameter. The inside surfaces of the bearing sectors were machined to a 0.8 micrometer (32 μin.) rms finish and the outside surface of the journals to a 0.1 to 0.2 micrometer (4 to 8 μin.) rms finish.

Bearing Test Apparatus

The test vessel and associated parts are shown in figure 4. The shaft is positioned vertically so that gravity forces do not load the bearing. The test vessel, which also serves as the test bearing housing, floats between upper and lower gas bearings.

In these experiments the motion of the bearing with its massive housing was monitored. The test shaft was mounted on two support ball bearings that were axially preloaded to about 890 newtons (200 lb) by a wave spring. This preload was necessary to insure a minimum amount of shaft runout. Thus, the journal axis was fixed while the bearing axis whirled. The validity of the stability data obtained in this manner was established in reference 8 where excellent correlation was obtained between theoretical and experimental data for a three-axial-grooved bearing run in water with a plain journal.

Movement of the test vessel during a test is measured by orthogonally mounted capacitance probes outside the test vessel. The output of the probes is connected to an X, Y-display on an oscilloscope where this motion can be observed. The orbital frequency of the test vessel motion was measured by a frequency counter. A more detailed description of the test apparatus and instrumentation is given in reference 9.

PROCEDURE

Test-shaft speed was increased in increments from 100 rpm in some tests to 1000 rpm in others depending on the anticipated whirl speed. The bearings were run at zero load throughout the entire test. The onset of whirl was noted by observing the bearing housing motion on the oscilloscope screen (ref. 8), and the shaft speed was recorded at this time. Damage to the test bearings due to fractional frequency whirl was prevented by reducing the speed immediately after observing the whirl pattern and photographing the amplitude-time trace on the oscilloscope screen.

RESULTS AND DISCUSSION

General

The results of 59 bearing stability tests with tilted-lobe bearings having L/D ratios from 0.2 to 1.0 are shown in table I and figures 5 to 12. The bearings had from three to seven lobes. Forty-eight tests were run in water, and the remaining eleven were run in MIL-L-7808G turbine oil. The bearings were run hydrodynamically while submerged in the lubricant which was at an average temperature of 294 K (70° F). The maximum speed attained without whirl was 5400 rpm.

The dynamic viscosity μ of water at 294 K (70° F) is 9.0×10^{-4} newton-second per square meter (1.3×10^{-7} lb-sec/in.^{2}) and that for MIL-L-7808G oil is 254.0×10^{-4} newton-second per square meter (37.0×10^{-7} lb-sec/in.^{2}).

The leading edge entrance wedge thickness ΔR_L is used when comparing data for bearings having identical numbers of lobes. However, when comparing data for bearings of different numbers of lobes the lobe wedge angle θ is employed. It is apparent from figure 2 that at a constant θ and α the ΔR_L value will be different for three, five, or seven lobed bearings since L_2 of necessity must vary with lobe number and ΔR_L depends on L_2. In reference 2, three-tilted-lobe bearings with wholly converging film geometry yielded maximum stability at ΔR_L equal to 0.066 millimeter (2600 μin.). This ΔR_L value yields a lobe wedge angle θ of 0.106°. It was originally intended to keep θ constant at 0.106° for the bearings tested in this present investigation, but, because of manufacturing difficulties, θ varied from 0.096° to 0.154°. This variation did not noticeably affect the results of these tests.

Stability Comparison of Three-Tilted-Lobe Bearings at Various Offset Factors

The curves in figure 5 show stability data for three-tilted-lobe water-lubricated bearings with an L/D ratio of 1.0 at various lobe offset factors. The experimental curves represent the stability limits of the bearings tested and indicate a zero-load threshold of stability. The area to the left of each curve represents those conditions that produce stable operation under zero load, while the area to the right of each curve represents those conditions that produce fractional frequency whirl. The theoretical stability analysis of a journal bearing (ref. 8) showed that the important parameters to consider are the dimensionless mass parameter $\overline{M}$ and the dimensionless speed parameter Γ as shown in figure 5. The dashed curve through the circular data points is the curve obtained from bearings run in this investigation. These bearings had an offset factor of 0.82, and their data curve lies, as expected, between previously reported experimental curves of three-tilted-lobe bearings with offset factors of 0.59 and 1.00 (ref. 2). These results confirm the trends shown in reference 2, which indicated that increased stability can be attained by using more of the arc of each lobe to build up pressure.

Figure 6 shows the effect of the number of lobes on stability. Only for bearings with an L/D ratio of 1.0 (fig. 6(a)) is there any effect on bearing stability due to the number of lobes, and only at the higher clearance values, 0.038 to 0.051 millimeter (1500 to 2000 μin.), is it significant. The effect is so slight for the bearings with L/D ratios of 0.75, 0.50, and 0.20 that only one experimental curve is drawn through the respective data points for three, five, and seven lobed bearings in figures 6(b), (c), and (d).

From the results of figure 6, it appears that a three-lobe bearing would be the most desirable to use in practical applications. Its stability characteristics are as good as or better than the five or seven lobe configuration, and it has the added feature of being much easier and less costly to fabricate than the bearings with more lobes.

Effect of Length to Diameter Ratio on Stability

Figure 7 shows the effect of L/D on stability for water-lubricated tilted-lobe bearings with three, five, and seven lobe configurations. Stability for each configuration generally decreases with a decrease in L/D. The incremental decrease in stability for each lobe configuration is least pronounced when going from an L/D of 0.75 to 0.50.

Even though some bearing stability is lost when L/D is reduced in a tilted-lobe configuration, the resultant stability threshold of a bearing with a low L/D ratio can be higher than that of other bearing configurations having greater L/D ratios but lower offset factor. An example of this is shown in figure 8 where the stability of a three-tilted-lobe bearing with an L/D ratio of 0.5 (solid curve) is compared with previously reported data (ref. 3) for a three centrally lobed bearing with an L/D ratio of 1.0 (short dashed curve). The stability of the tilted-lobe bearing, with an offset factor of 0.86, is greater than that of the centrally lobed bearing, with an offset factor of 0.50, even though the latter bearing had an L/D ratio twice that of the tilted-lobe bearing. Figure 8 also shows a stability curve for a tilted-lobe bearing with an L/D ratio of 0.2 (long dashed curve) which had better stability than the centrally lobed bearing (L/D = 1.0) up to a dimensionless speed value of approximately 1.8. These narrow bearings might therefore prove successful in applications where stability is a problem and axial space is limited.

In the approximate analytical solution of short plain bearings in reference 9, the Sommerfeld number S is replaced by the capacity or Ocvirk number C_N, where $C_N = S(L/D)^2$. From this relation a new variable is suggested instead of Γ for the stability data in figure 7. The data in figure 7 were therefore replotted in figure 9 using a new dimensionless speed, $\Gamma^o = \Gamma(D/L)$. These data are the combined data for bearings with three, five, and seven lobes and L/D ratios from 0.2 to 1.0, all shown on one curve. To minimize scatter, the log Γ against log L/D for the data of figure 7 is plotted in figure 10 at $\overline{M} = 0.2$, 0.6, 1.0, 5.0, and 10.0. The average slope of the resulting nine lines in figure 10 is 0.66. Therefore, the dimensionless speed variable for minimum scatter would be $\Gamma^* = \Gamma(D/L)^{0.66}$ or $\Gamma^* = 9.48\ \mu\omega R^{2.66}/P_a C^2 L^{0.66}$. The data of figure 11 are obtained by using the newly derived optimized stability parameter Γ^*. The resulting curve showed a minimum amount of scatter and has some merit as a means of generating a single stability line for all L/D ratios below 1.0. The data in figure 11 can be presented even more compactly. A straight line on log-log paper has the equation $\overline{M}(\Gamma^*)^B = A$. Thus, all data can be presented in terms of the simple expression $\overline{M}(\Gamma^*)^{1.42} = 8.6$, where B = 1.42 is the slope of the curve.

Stability Curve Range Extension

After the water tests were completed, MIL-L-7808G oil was substituted for water as the bearing lubricant. Eleven stability tests was run in oil using three-tilted-lobe bearings with L/D ratios of 0.5 and 1.0. The results of these tests are shown in table IV and figure 12. Figures 12(a) and (b) show the stability characteristics of a three-tilted-lobe bearing with L/D ratios of 0.5 and 1.0, respectively. The square symbols represent data from water tests on the bearing and the circles represent oil tests.

The oil tests were run to extend the range of the stability curves obtained in water. In figure 12(a) at a Γ value of 10.0 the bearing radial clearance C in water is 0.020 millimeter (800 μin.), whereas in oil at this same Γ value C is much larger, namely, 0.079 millimeter (3100 μin.). Therefore, in order to avoid running tests at extremely tight clearances in water, oil was used to obtain data at the higher values of Γ. This extended the range of Γ to approximately 300 in figure 12(a) and 150 in figure 12(b).

In figure 12(a) two of the oil data points are in close proximity to the water data curve and in figure 12(b) the same is the case for three oil data points. This indicates good correlation between water and oil data, and it confirms the validity of the dimensionless mass and dimensionless speed parameters for determining stability.

SUMMARY OF RESULTS

Forty-eight stability tests in water and eleven in MIL-L-7808G oil were performed on tilted-lobe bearings. The bearings had three, five, and seven tilted lobes with L/D ratios from 0.2 to 1.0. Radial clearances ranged from 0.017 to 0.102 millimeter (650 to 4000 μin.). The inside diameter of the bearings in all cases was 3.8 centimeters (1.5 in.). They were run hydrodynamically in the lubricant at an average temperature of 294 K (70° F). The following results were obtained:

1. Stability generally decreased with a decrease in L/D ratio for each of the three-, five-, and seven-tilted-lobe configurations tested.
2. A three-tilted-lobe bearing with an offset factor of 0.86 and an L/D ratio as low as 0.5 was more stable than a three centrally lobed bearing with an offset factor of 0.50 and L/D ratio of 1.0 over the complete range of dimensionless speeds tested.
3. The number of lobes, three, five, or seven, did not appreciably affect the stability of the test bearings, regardless of the L/D ratio.
4. The useful range of the stability curves obtained from water tests was extended by employing data from tests run in oil. The oil tests also confirm the validity of the dimensionless mass and speed parameters used to determine stability. All data can be presented in terms of the simple parameter $\overline{M}(\Gamma^*)^{1.42} = 8.6$.

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Cleveland, Ohio, December 4, 1974,
505-04.

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TABLE I. - TEST RESULTS FOR TILTED-LOBE BEARINGS

Bearing	Length to diameter ratio, L/D	Leading edge entrance wedge thickness, ΔR_L		Lobe wedge angle, θ, deg	Offset factor, α	Minimum radial clearance, C		Fractional-frequency whirl onset speed at zero load, N_w, rpm
		mm	μin.			mm	μin.	
Water-lubricated three-tilted-lobe bearings								
1	0.2	0.058	2300	0.125	0.78	0.020	800	400
						.034	1350	300
						.042	1650	200
						.055	2150	150
2	0.5	0.053	2100	0.117	0.76	0.022	850	2000
						.036	1400	600
						.043	1700	500
						.056	2200	360
3	0.75	0.053	2100	0.112	0.81	0.018	700	4800
						.032	1250	1400
						.039	1550	1080
						.052	2050	700
4	1.0	0.048	1900	0.096	0.82	0.018	700	5200
						.032	1250	3000
						.039	1550	2300
						.052	2050	900
Water-lubricated five-tilted-lobe bearings								
5	0.2	0.036	1400	0.130	0.80	0.017	650	580
						.029	1150	400
						.037	1450	200
						.050	1950	200
6	0.5	0.033	1300	0.126	0.80	0.019	750	2400
						.032	1250	900
						.039	1550	800
						.052	2050	300
7	0.75	0.031	1200	0.126	0.77	0.017	650	5000
						.029	1150	1600
						.037	1450	1000
						.050	1950	630
8	1.0	0.038	1500	0.154	0.71	0.017	650	5200
						.029	1150	3200
						.037	1450	2020
						.050	1950	700
Water-lubricated seven-tilted-lobe bearings								
9	0.2	0.020	800	0.111	0.69	0.017	650	600
						.029	1150	300
						.037	1450	180
						.050	1950	180
10	0.5	0.023	900	0.113	0.76	0.017	650	3300
						.029	1150	900
						.037	1450	800
						.050	1950	350
11	0.75	0.028	1100	0.139	0.76	0.017	650	4500
						.029	1150	1800
						.037	1450	1100
						.050	1950	600
12	1.0	0.028	1100	0.154	0.72	0.019	750	5200
						.032	1250	3000
						.039	1550	1500
						.052	2050	700
Oil-lubricated three-tilted-lobe bearings								
13	0.5	0.061	2400	0.130	0.78	0.030	1200	5100
						.043	1700	4200
						.060	2350	1700
						.074	2900	1100
						.086	3400	800
						.099	3900	600
14	1.0	0.056	2200	0.122	0.76	0.046	1800	5400
						.060	2350	5000
						.076	3000	3600
						.089	3500	2600
						.102	4000	1600

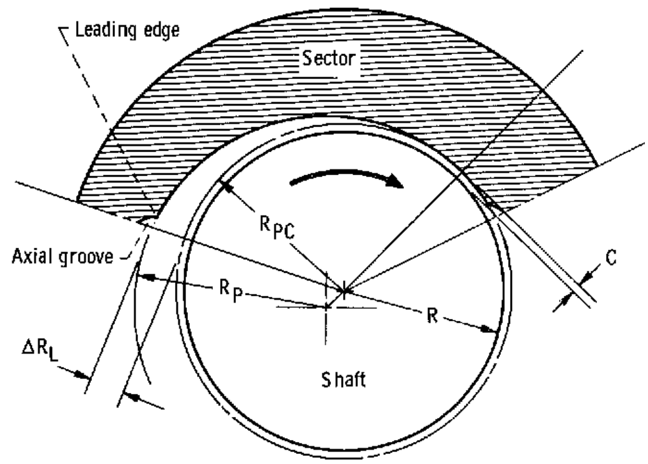


Figure 1. - Geometry of bearing sector or lobe.

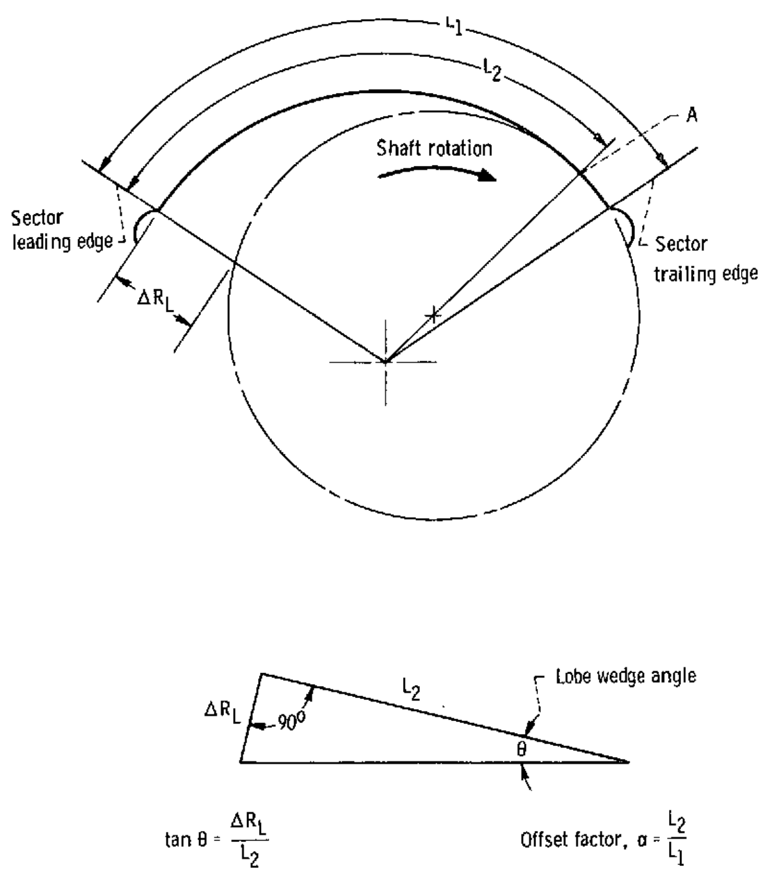


Figure 2. - Offset factor and lobe wedge angle determination.

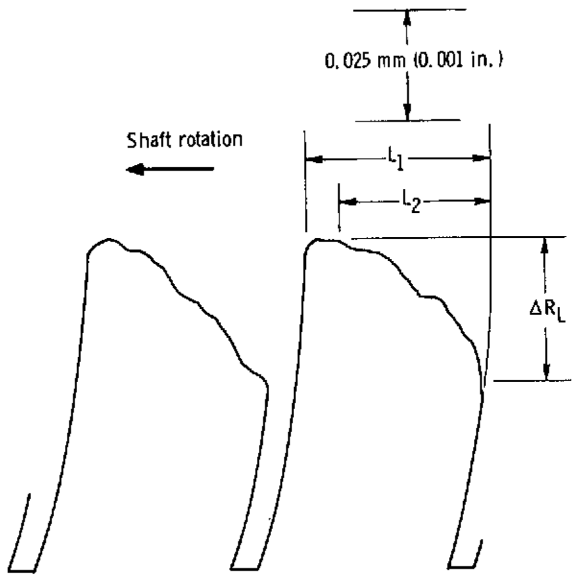


Figure 3. - Typical circumferential profile trace of inside surface of bearing sectors. Offset factor, L_2/L_1.

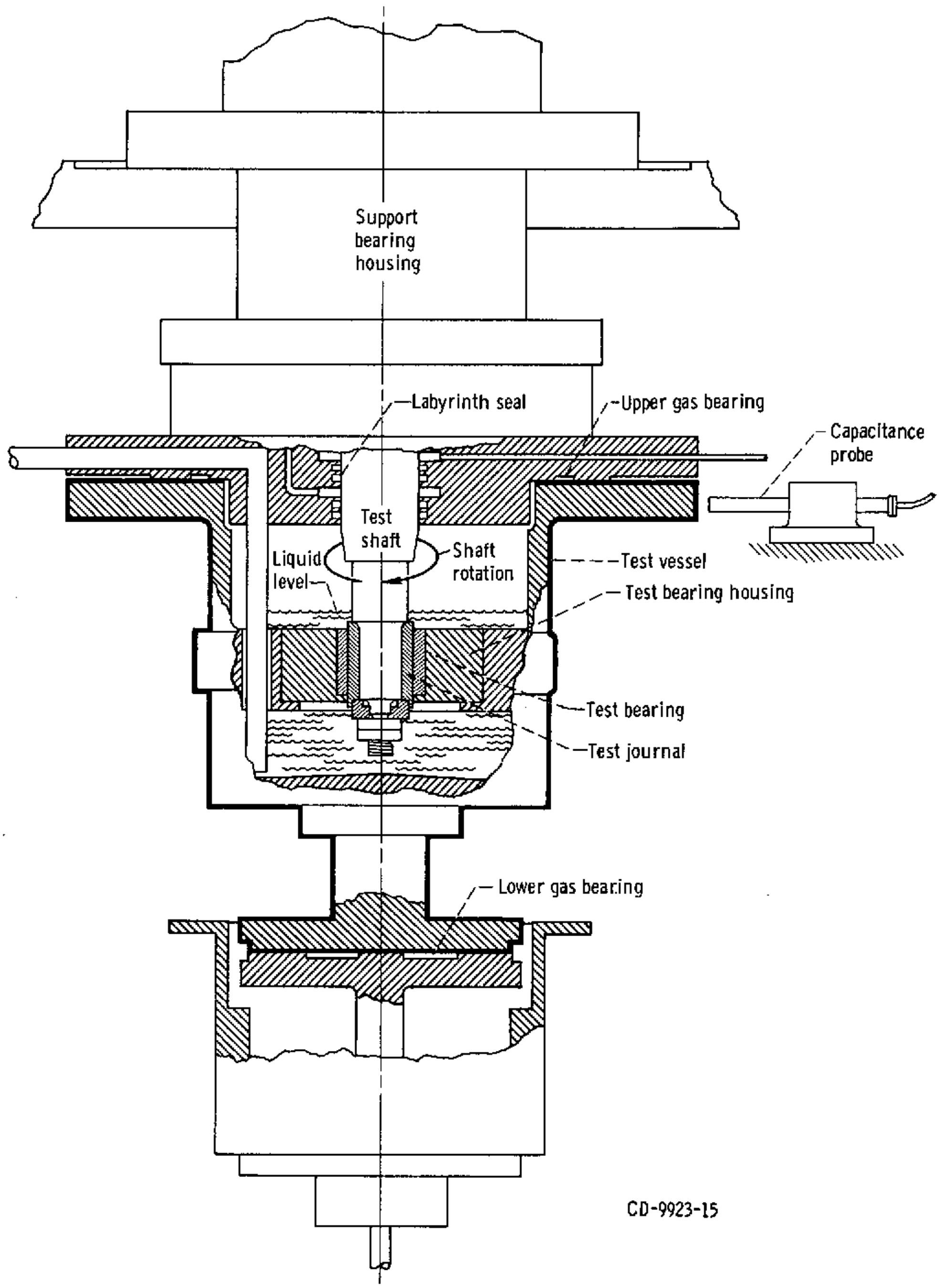


Figure 4. - Bearing test apparatus.

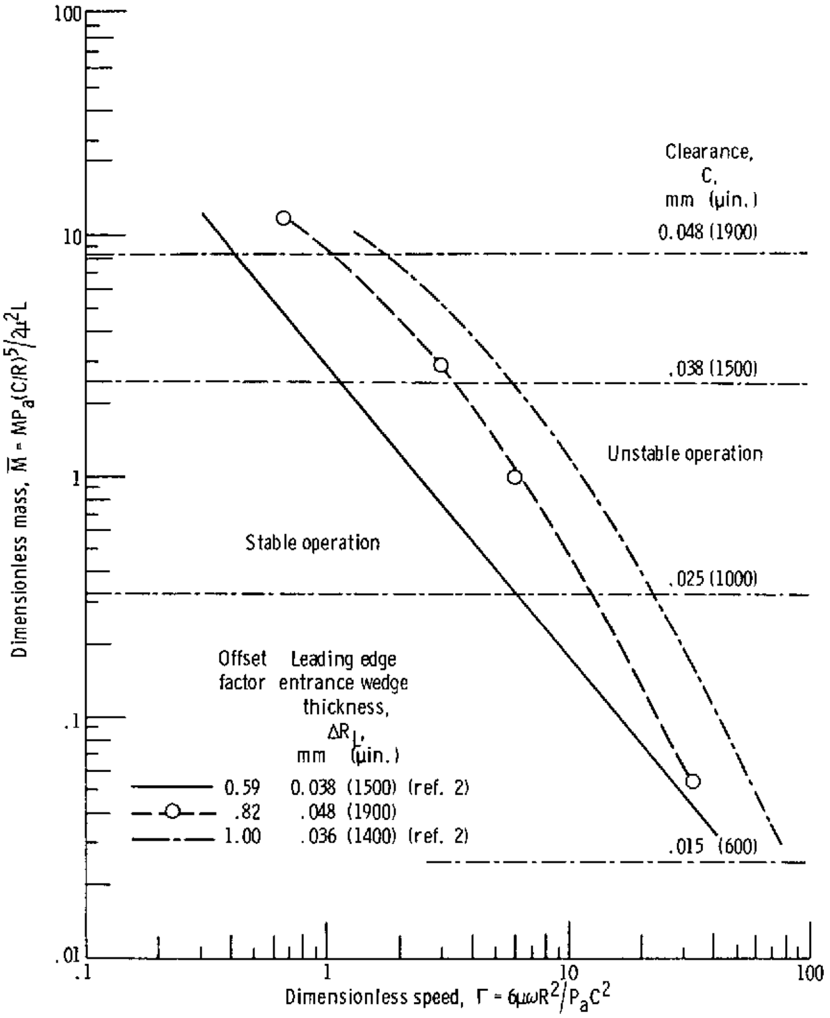


Figure 5. - Comparison of stability data for water-lubricated three-tilted-lobe bearings at various lobe offset factors. L/D = 1.0.

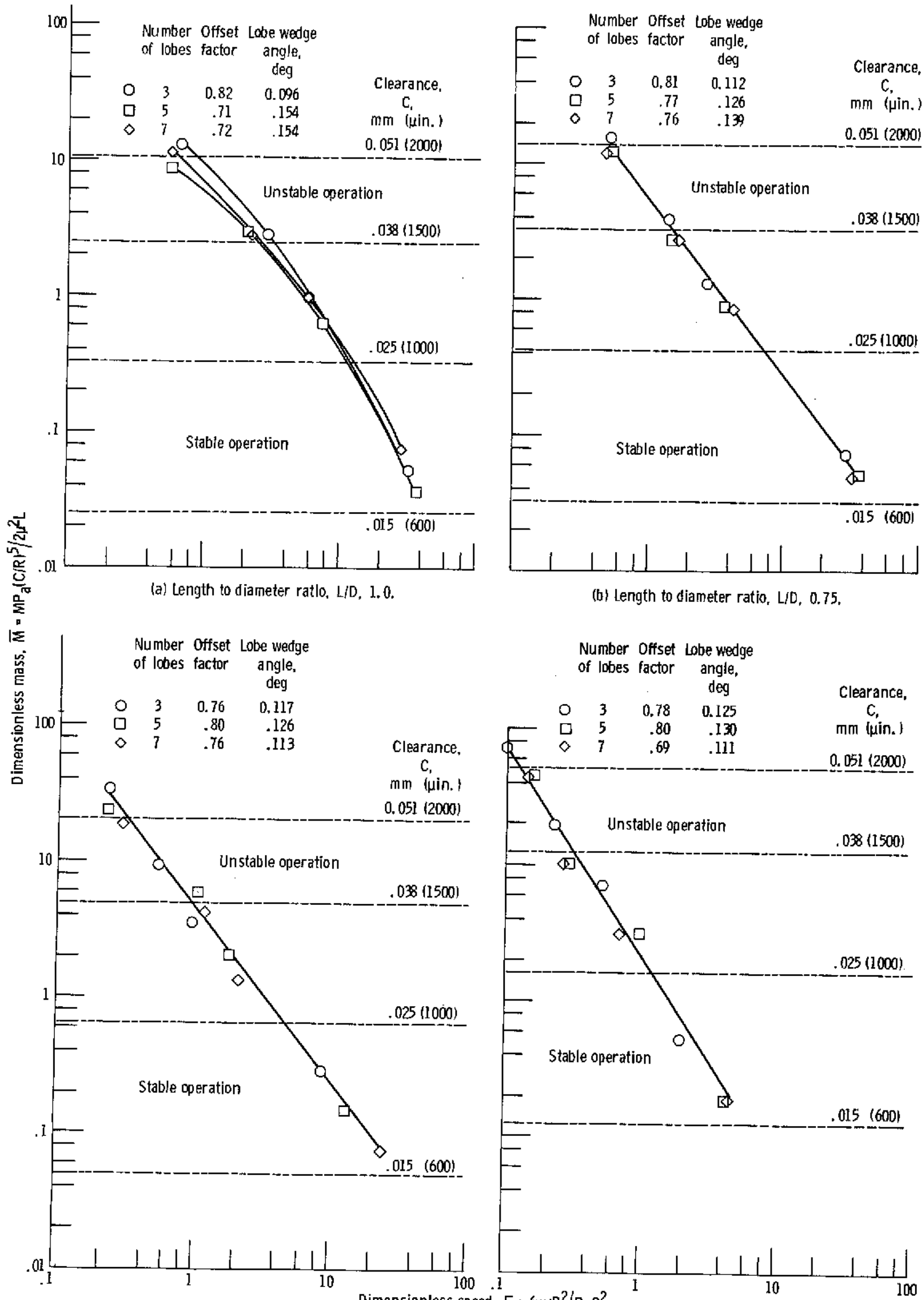
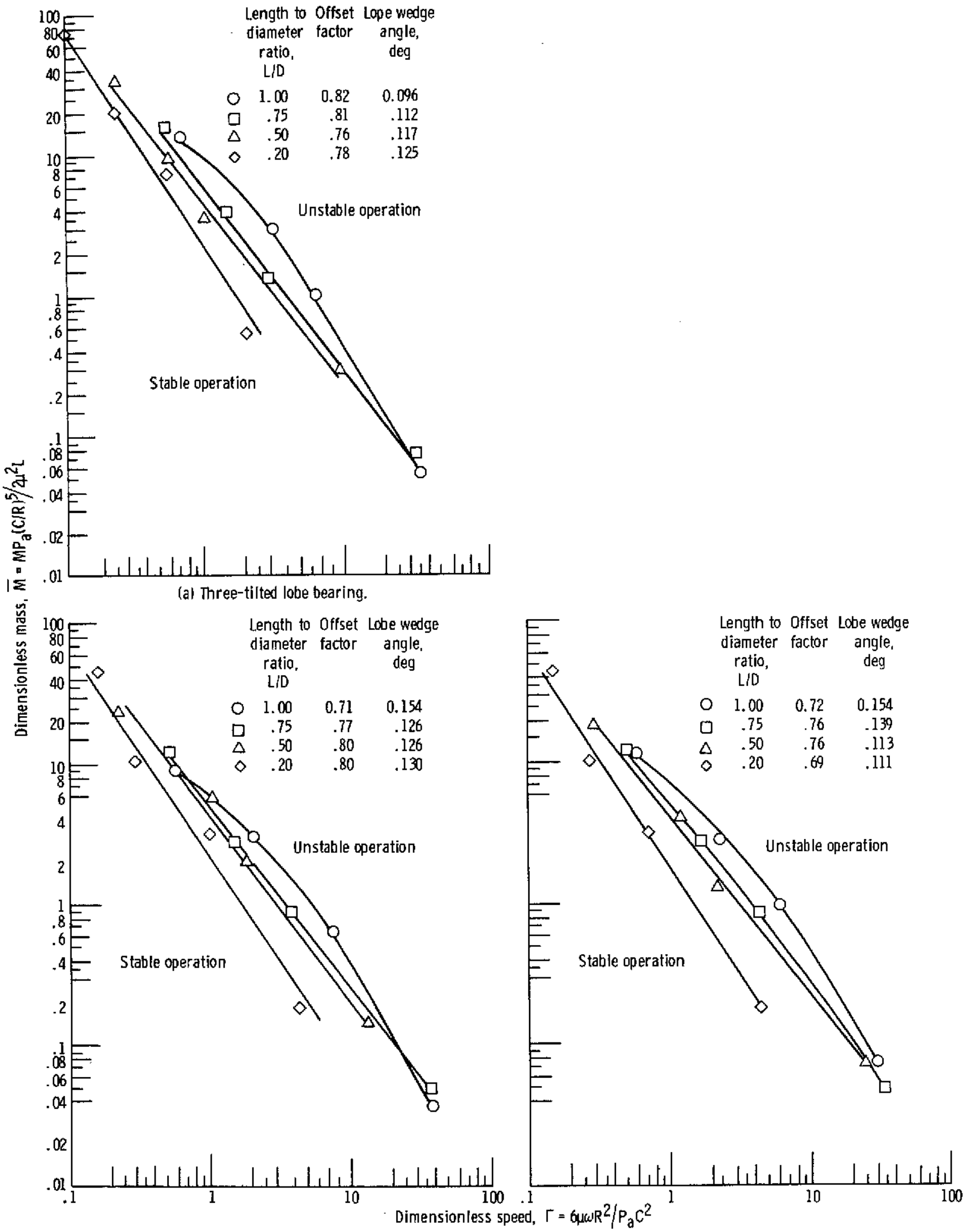


Figure 6. - Effect of number of lobes on stability for water-lubricated bearings.



(b) Five-tilted-lobe bearing.

(c) Seven-tilted lobe bearing.

Figure 7. - Effect of length to diameter ratio on stability for water lubricated bearings.

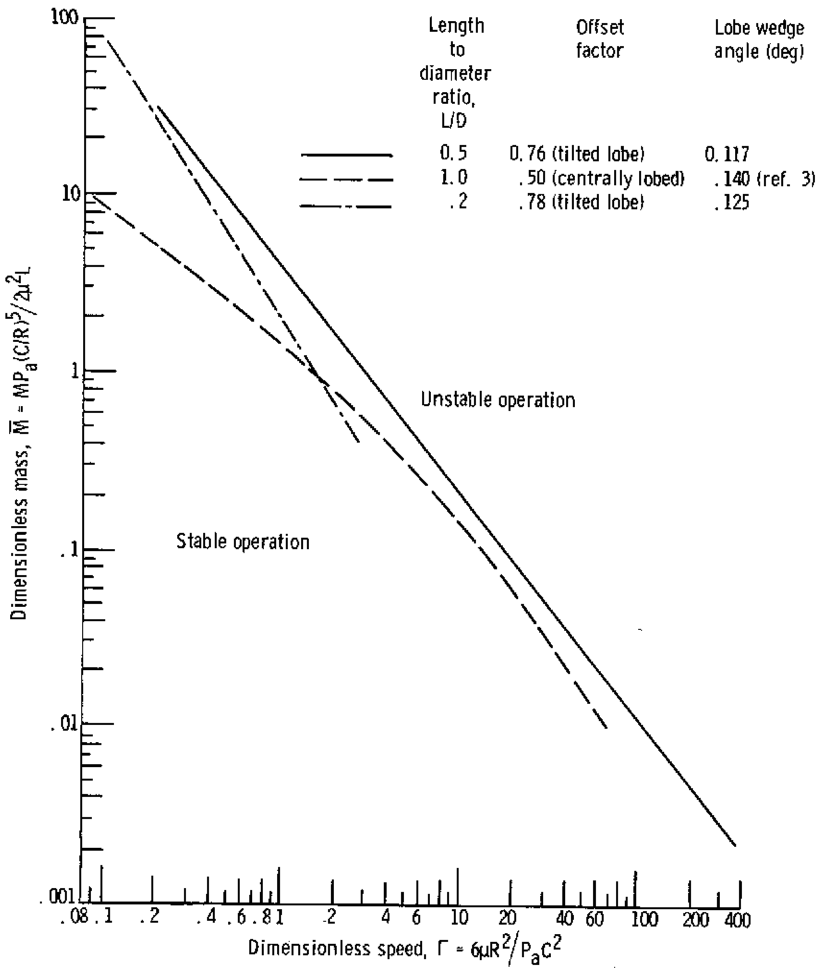


Figure 8. - Stability comparison of water-lubricated three-lobe bearing.

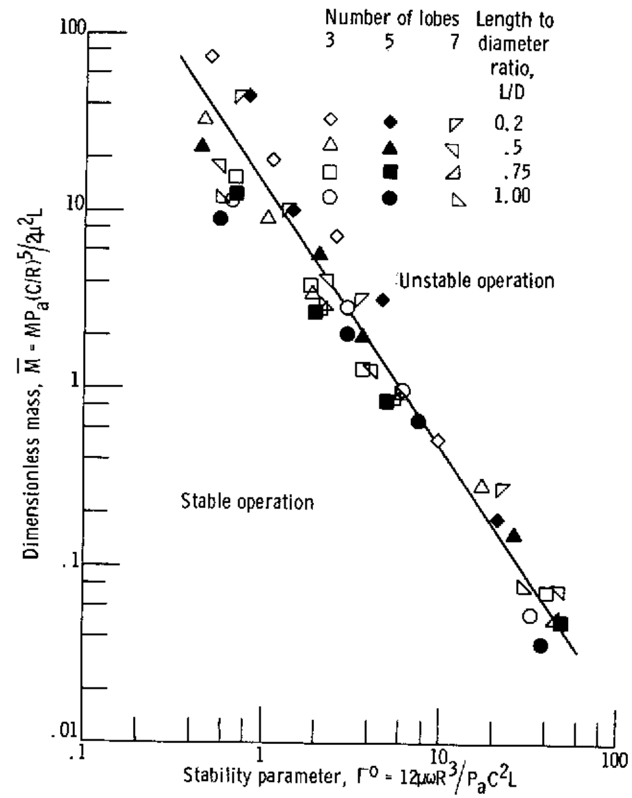


Figure 9. - Stability of water-lubricated tilted-lobe bearing with various numbers of lobes and L/D ratios. Average offset factor, $\alpha = 0.77$; average lobe wedge angle, $\theta = 0.125^0$.

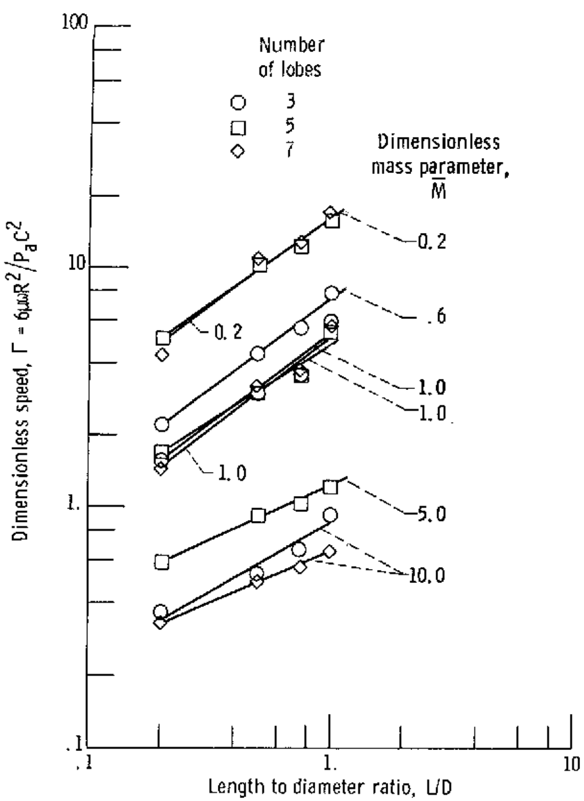


Figure 10. - Log Γ against log L/D ratio at various dimensionless mass values.

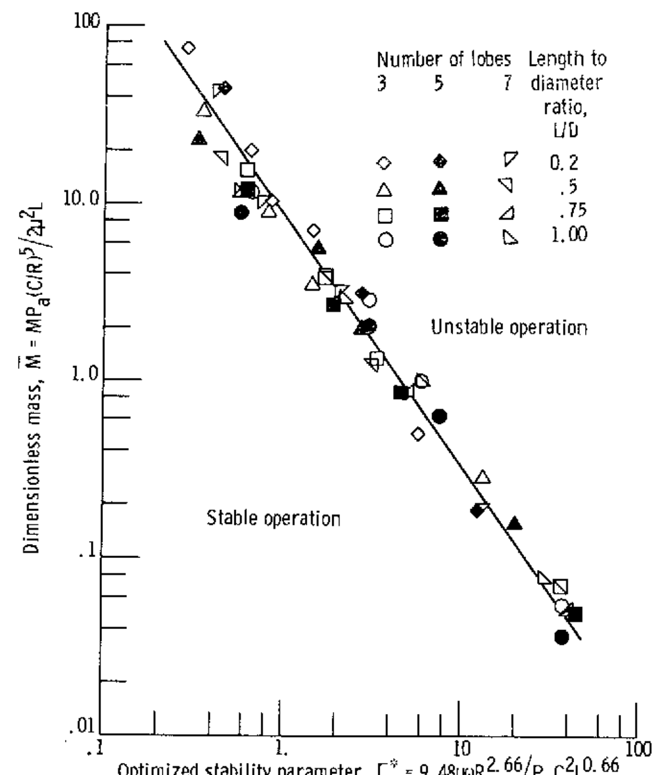
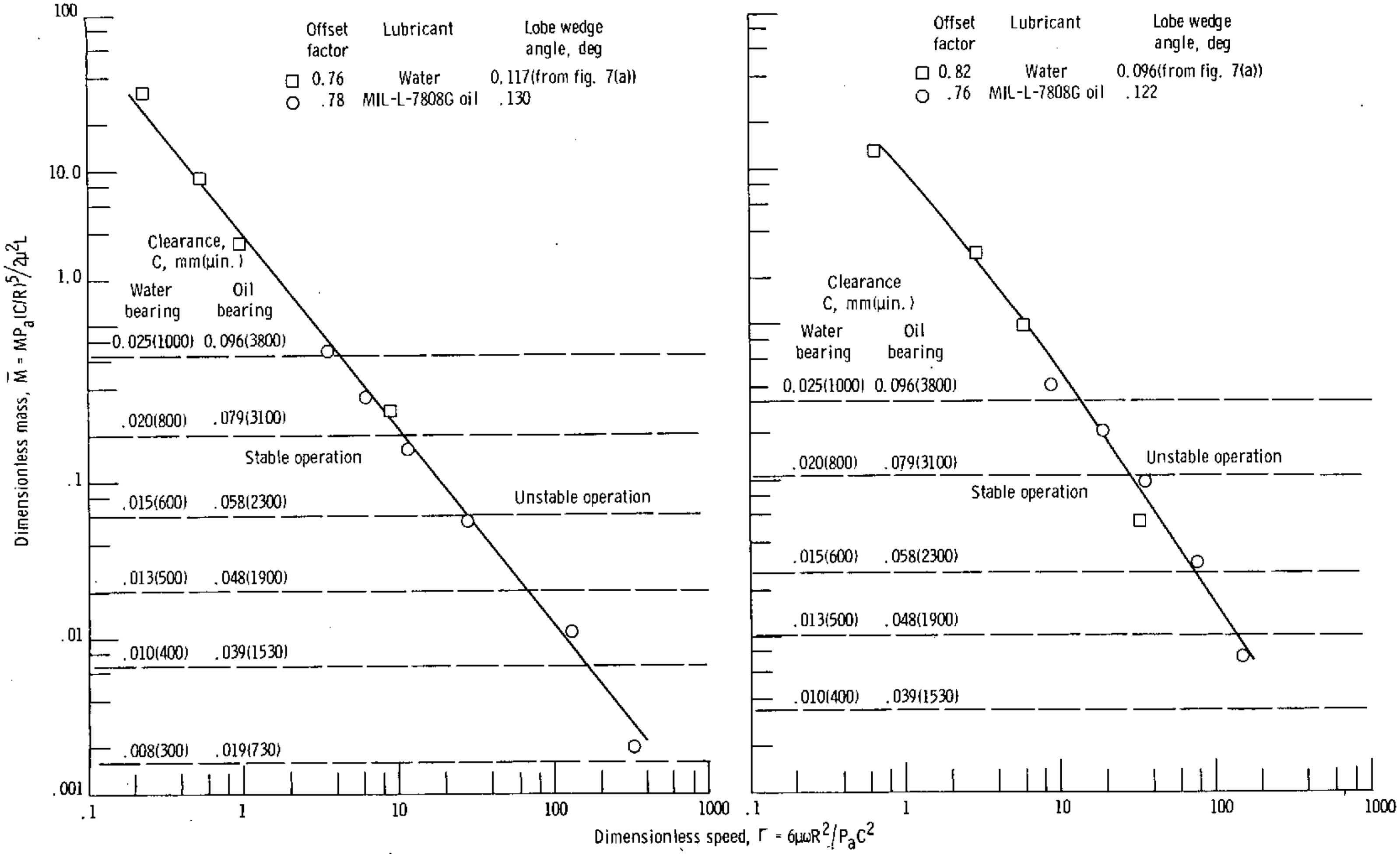


Figure 11. - Stability of water-lubricated tilted-lobe bearings with various number of lobes and length to diameter ratios. Offset factor, $\alpha = 0.77$; lobe wedge angle, 0.125^0.



(a) Length to diameter ratio, L/D = 0.5.

(b) Length to diameter ratio, L/D = 1.0.

Figure 12. - Comparison of stability data for three tilted-lobe bearing in water and MIL-L-7808G oil.

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