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II - Effects of Configuration and Groove to Ridge Clearance Ratio

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ABSTRACT

Hydrodynamic stability tests in water were performed on six 1.5-in. - (3.8-cm-) diameter herringbone-grooved journals having 10, 20, or 40 full or partial grooves. A maximum speed of 11 500 rpm with stable operation at zero load was attained.

The number and length of grooves did not appreciably affect the zero load stability limits of the herringbone-grooved journal bearings. Generally, the herringbone-grooved journals had maximum stability (a maximum fractional frequency whirl onset speed) when the groove to ridge clearance ratio was closest to 2.1 as predicted by incompressible flow theory.

EXPERIMENTS ON THE STABILITY OF WATER LUBRICATED HERRINGBONE-GROOVE JOURNAL BEARINGS

II - EFFECTS OF CONFIGURATION AND GROOVE TO RIDGE CLEARANCE RATIO

by Fredrick T. Schuller, David P. Fleming,
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SUMMARY

A series of 67 tests was performed to determine the zero load stability limits (as measured by fractional frequency whirl onset speed) of herringbone-grooved journal bearings of six different configurations. The bearings were totally immersed in water and run hydrodynamically. Three journals with 10, 20, and 40 full grooves and three 10, 20, and 40 partial grooves ranging in depth from 340 to 3350 microinches (0.009 to 0.085 mm) were evaluated. Bearing diameter was nominally 1.5 inches (3.8 cm), and the bearings had a length to diameter ratio of 1. A maximum speed of 11 500 rpm with stable operation at zero load was attained.

Within the range covered in this investigation, the number of grooves and groove lengths did not appreciably affect the zero load stability limits of the herringbone-grooved journal bearings.

The journal bearings with 10 partial grooves reached a maximum in stability at a groove depth of 1280 microinches (0.032 mm) as the depth was varied from 340 to 3350 microinches (0.009 to 0.085 mm).

The herringbone-grooved journals had a maximum stability (a maximum fractional frequency whirl onset speed) when the groove to ridge clearance ratio H was closest to 2.1 as predicted by incompressible flow theory.

INTRODUCTION

One of the prime requisites of a journal bearing in a power generation system for space vehicles is its ability to inhibit self-excited fractional frequency whirl. The whirl problem is compounded by the presence of several factors tending to produce this instability, namely, zero load, high speed, and low viscosity lubricants. In this type of whirl, the journal center has a tendency to orbit the bearing center at an angular velocity about half that of the journal around its own center. Tilting pad bearings are exceptionally stable (refs. 1 to 4) but are complex since they are composed of several parts. Externally pressurized bearings can be designed to inhibit whirl but require a continuous high pressure source of lubricant. This increases system complexity and may result in excessive power loss. A journal bearing of fixed geometry that exhibited good stability characteristics when lubricated with air and fair stability when lubricated with sodium is the herringbone-grooved bearing (refs. 1, 5, and 6).

Most of the theoretical and experimental work on herringbone-grooved bearings has dealt with compressible lubricants (refs. 5 to 7), with very little information available on incompressible lubricants (ref. 8). The experimental data reported herein is for incompressible lubricants.

Theoretical analysis of the herringbone-grooved bearing assumes an infinite number of grooves (ref. 7). It would be of interest to observe experimentally how bearing stability is affected by a variation in the finite number of grooves in such a bearing. Further, theory for incompressible flow predicts an optimum stability at a groove to ridge clearance ratio of 2.1 (ref. 7). Experimental verification of this prediction would be desirable. Groove to ridge clearance ratio can be varied by either maintaining constant groove depth while varying the radial clearance or by varying the groove depth while maintaining constant radial clearance. In these experiments, because of the strong influence of radial clearance on stability (ref. 8), both clearance and groove depth were varied so that the effects of varying groove to ridge clearance ratio could be studied independently of radial clearance.

The objectivies of this study were (1) to determine the zero load stability limits of six water-lubricated herringbone-grooved journal bearing configurations with various clearances and groove to ridge clearance ratios, (2) to observe the effect of changes in the length and number of grooves on bearing stability, and (3) to compare experimental results with theoretical predictions of the groove to ridge clearance ratio for maximum stability of herringbone-grooved bearings with an incompressible lubricant.

Bearings 1.5 inches (3.8 cm) in diameter by 1.5 inches (3.8 cm) long were submerged in water at an average temperature of 80° F (300 K) and operated hydrodynamically at journal speeds to 11 500 rpm.

APPARATUS

Test Bearings

Herringbone-grooved journal bearings of six configurations, three journals having 10, 20, and 40 full grooves and three having 10, 20, and 40 partial grooves (fig. 1), were evaluated. Grooving extended beyond the bearing sleeves at each end to ensure an adequate lubricant supply. The partially grooved journals had a circumferential land centrally located along the length of the bearing. The width of the circumferential land was one-third the length of the bearing (fig. 2). The fully grooved journals had herringbone grooves that met midway along the axial length of the bearing. Circumferential surface profile traces were made of each journal outside diameter in two planes along the length of the journal to obtain the average groove depths listed in table I. A typical surface profile trace is shown in figure 3. To reduce the number of new journals required, the outside diameter of the journals with large groove depths was ground after testing to reduce the groove depth, and the journal was reused. Surface profile traces were made after each grinding operation to determine the new average groove depth.

The herringbone journal helix angle and groove to ridge width ratio (table I) were set at those values which yield a maximum radial force at a compressibility number of zero for a compressible lubricant (ref. 7), which approximated the conditions existing with an incompressible lubricant. Because of manufacturing difficulties, the maximum number of grooves was set at 40 for this investigation.

The bearings in all cases had a nominal 1.5-inch (3.8-cm) length and diameter, and the bearing inside diameter and journal outside diameter were machined to a 4- to 8-microinch (root mean square) finish.

Bearing Test Apparatus

The test vessel and radial loading system are shown in figures 4(a) and (b), respectively. The shaft is positioned vertically so that gravity forces do not load the bearing. The test vessel, containing the test bearing, floats between upper and lower gas bearings. Radial load is applied by an air cylinder between two semicircular wheels, one of which pivots on a knife edge. Bearing torque can be measured, if desired, by a force transducer.

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 the motion of the test vessel can be observed. The orbital frequency of the test vessel motion was measured by a frequency counter. A

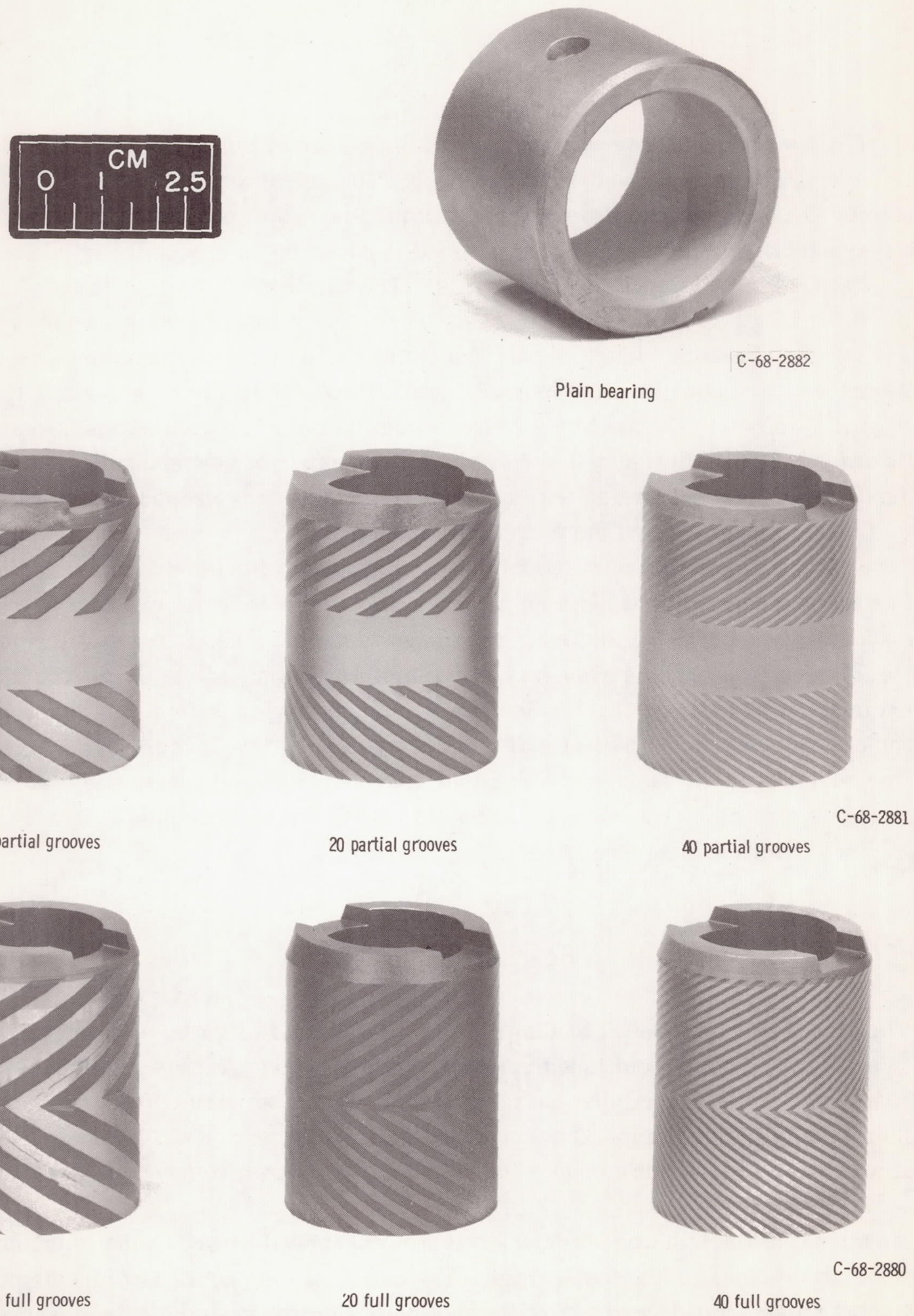


Figure 1. - Herringbone-grooved journal configurations.

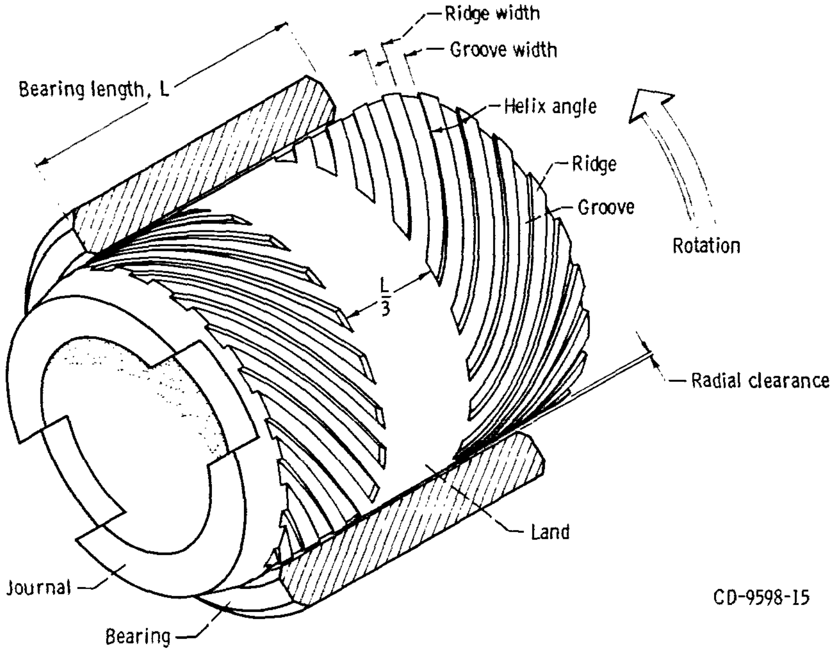


Figure 2. - Partially grooved herringbone journal bearing assembly.

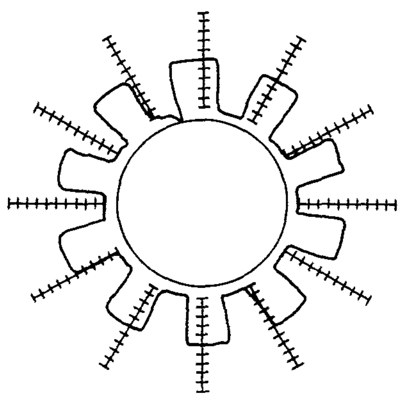


Figure 3. - Circumferential surface profile trace of journal J-14. Scale: 250 micro-inches (0.006 mm) per division.

TABLE I. - HERRINGBONE-GROOVED GEOMETRIES

[Helix angle, 33°.]

Journal	Groove depth, δ		Groove type	Number of grooves, n	Groove width, a_g		Ridge width, a_r	
	μin.	mm			in.	mm	in.	mm
K-15R	590	0.015	Partial	40	0.032	0.813	0.032	0.813
J-9R	620	.016	Full					
J-9	1380	.035	Full					
[a]K-15	1700	.043	Partial					
K-14R	410	0.010	Partial	20	0.062	1.626	0.062	1.626
K-6R	700	.018	Full					
[a]K-14	1400	.036	Partial					
[a]K-6	1640	.042	Full					
J-14RL	340	0.009	Partial	10	0.128	2.438	0.128	2.438
J-14R	700	.018	Partial					
J-12R	960	.024	Full					
J-14	1280	.032	Partial					
J-12	1420	.036	Full					
J-10	1350	.034	Full					
J-11R	2660	.068	Partial					
J-11	3350	.085	Partial					

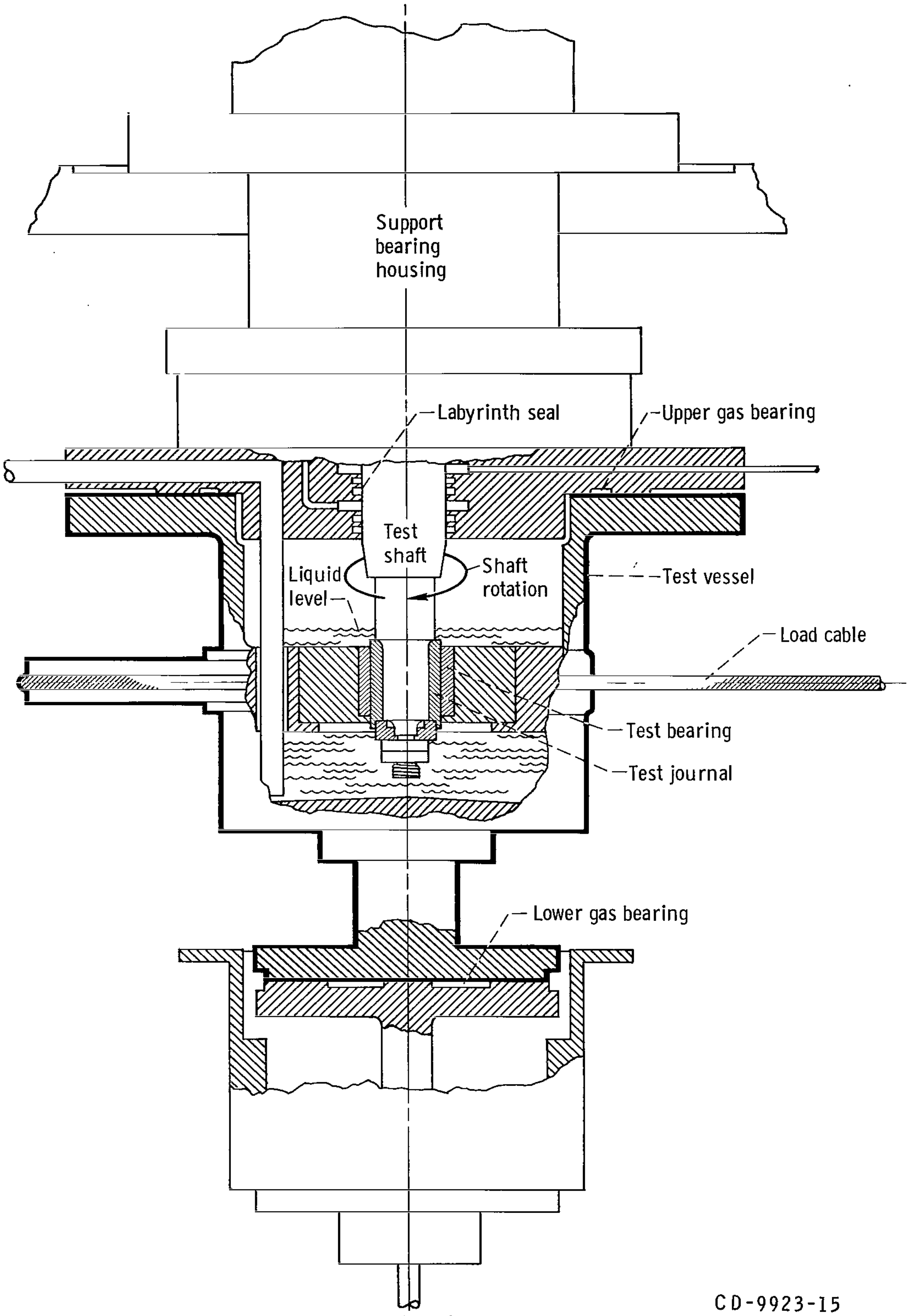
[a]Data from ref. 8.

more detailed description of the test apparatus and instrumentation is given in reference 8.

Procedure

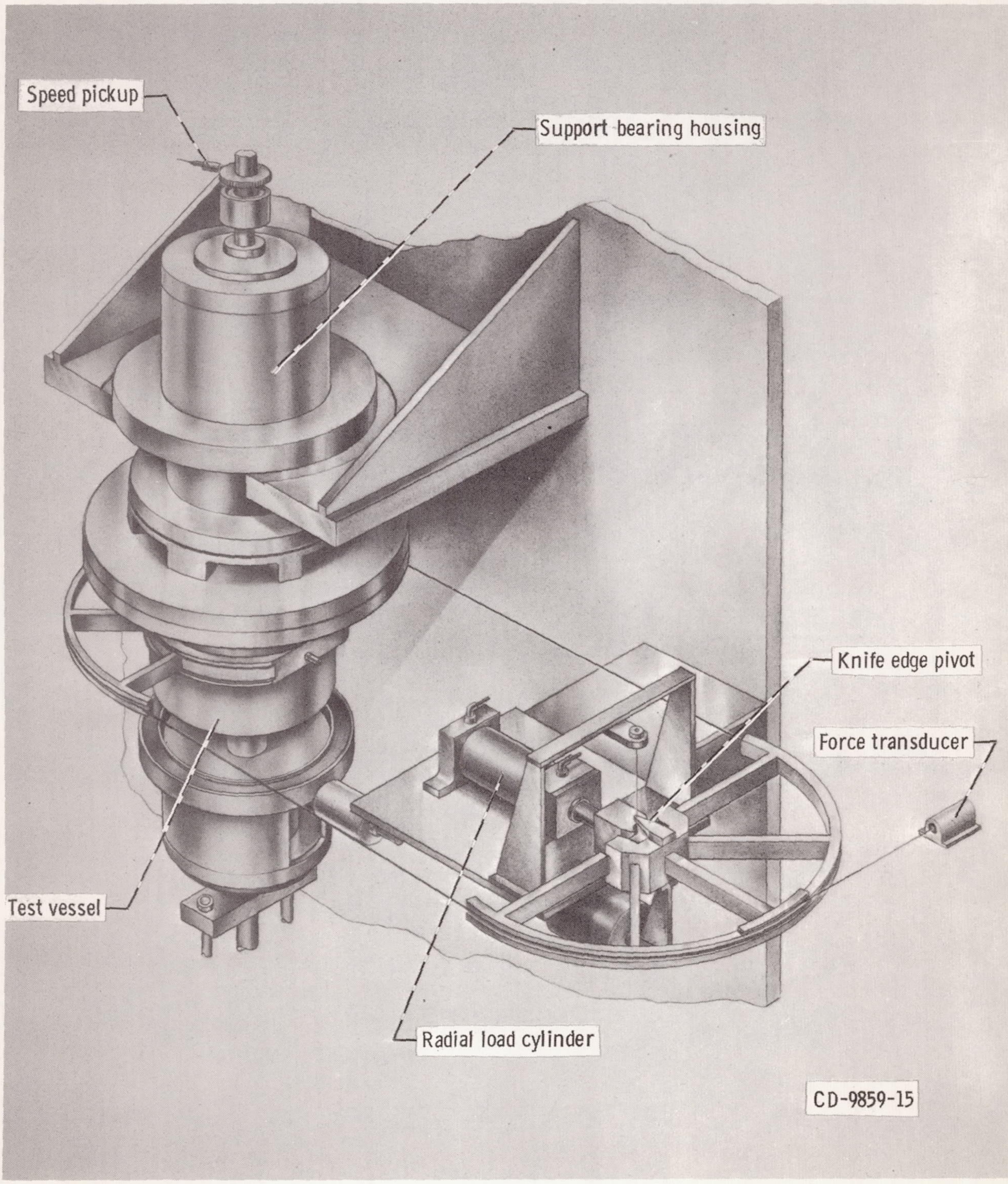
The test shaft speed was increased in increments of 1000 rpm from 0 to 11 500 rpm maximum. Radial load was kept at zero until a speed was reached where fractional frequency whirl occurred. The onset of whirl was noted by observing the bearing housing motion on the oscilloscope screen. Radial load was applied to suppress the whirl and prevent bearing failure. The time interval between speed changes varied but was of sufficient duration to allow the friction torque to stabilize.

In these experiments the motion of the bearing with its massive housing was monitored. 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.



(a) Detailed view.

Figure 4. - Bearing test apparatus.



(b) Loading mechanism.

Figure 4. - Concluded.

RESULTS AND DISCUSSION

General

Experimental results obtained with six journals having different groove geometries are shown in table II and figures 5 to 12. Fifteen different groove depths were investigated. These were obtained by regrinding the outside diameter of the original journals. The reworked journals were then used to obtain data for a journal having the same geometry as the original but of lesser groove depth. A total of 67 bearing stability tests was conducted at radial clearances ranging from 300 to 1800 microinches (0.008 to 0.046 mm), and groove depths ranging from 340 to 3350 microinches (0.009 to 0.085 mm).

The bearings were submerged in water at an average temperature of 80° F (300 K) and run hydrodynamically. Maximum speed attained without whirl was 11 500 rpm.

Effect of Number of Grooves on Stability

The experimental results obtained with herringbone journal bearings having 40, 20, and 10 partial grooves are shown in figure 5(a). In the area labelled stable operation to the left of the experimental curve, the bearings ran stably at zero load; to the right of the curve, fractional frequency whirl occurred. The experimental curve represents the stability limits of the bearings tested and indicates the zero load threshold of stability. The dimensionless parameters $\overline{M}$ and Γ were obtained from the theoretical stability analysis in reference 8. The stability of partially grooved herringbone journal bearings with grooves 1280 to 1700 microinches (0.032 to 0.043 mm) deep is not appreciably affected by the number of grooves, except at the higher $\overline{M}$ values where the 10-groove journal became less stable than the 40-groove journal.

Figure 5(b) shows the stability limits of herringbone journal bearings with 40, 20, and 10 full grooves 1380 to 1640 microinches (0.035 to 0.042 mm) deep. The number of grooves did not appreciably affect the stability of the fully grooved journal bearings, except at an $\overline{M}$ value of 0.04 where the 20-groove journal was somewhat less stable than the 40-groove journal.

Effect of Partial and Full Grooves on Stability

The partially grooved journals, because of the circumferential land centrally located along the length of the bearing, have a total groove length two-thirds that of the fully

TABLE II. - TEST RESULTS ON JOURNALS WITH 40, 20, AND 10 GROOVES

(a) 40 Grooves

Journal	Groove depth, δ		Groove type	Bearing	Radial clearance, C		Groove to ridge clearance ratio, H	Fractional frequency whirl onset speed at zero load, N_s, rpm
	μin.	mm			μin.	mm		
K-15R	590	0.015	Partial	B-15	550	0.014	2.1	9 000
				B-13	900	0.023	1.7	5 500
				B-18	1100	0.028	1.5	3 900
				B-11	1600	0.041	1.4	1 000
J-9R	620	0.016	Full	B-32	500	0.013	2.2	11 300
				B-33	600	0.015	2.0	7 500
				B-36	750	0.019	1.8	7 500
				B-29	1000	0.025	1.6	4 600
				B-28	1400	0.036	1.4	1 700
[a]K-15	1700	0.043	Partial	MP-12	500	0.013	4.4	7 000
				MP-10	800	0.020	3.1	6 000
				MP-5	1700	0.043	2.0	2 900
J-9	1380	0.035	Full	B-29	300	0.008	5.6	11 500
				B-28	650	0.016	3.1	11 200
				B-26	1100	0.028	2.3	4 800
				B-27	1450	0.037	2.0	2 000
				B-15A	1800	0.046	1.8	1 500

[a]Data from ref. 8.

TABLE II. - Continued. TEST RESULTS ON JOURNALS

WITH 40, 20, AND 10 GROOVES

(b) 20 Grooves

Journal	Groove depth, δ		Groove type	Bearing	Radial clearance, C		Groove to ridge clearance ratio, H	Fractional frequency whirl onset speed at zero load, N_s, rpm
	μin.	mm			μin.	mm		
[a]K-14	1400	0.036	Partial	MP-12	300	0.008	5.7	7800
				MP-10	700	0.018	3.0	6500
				B-14	950	0.024	2.5	5300
				MP-6	1250	0.032	2.1	4000
[a]K-6	1640	0.042	Full	B-3	650	0.016	3.5	6500
				B-5	800	0.020	3.0	6000
				B-2	1050	0.027	2.6	7000
				B-4	1350	0.034	2.2	3000
K-14R	410	0.010	Partial	B-13	450	0.011	1.9	7800
				B-18	650	0.016	1.6	6000
				B-19	950	0.024	1.4	3000
				B-16	1350	0.034	1.3	1200
K-6R	700	0.018	Full	B-24	750	0.019	1.9	9800
				B-23	1000	0.025	1.7	5000
				B-25	1150	0.029	1.6	4300
				B-20	1350	0.034	1.5	2000
				B-21	1600	0.041	1.4	1500

[a]Data from ref. 8.

TABLE II. - Continued. TEST RESULTS ON JOURNALS

WITH 40, 20, AND 10 GROOVES

(c) 10 Grooves

Journal	Groove depth, δ		Groove type	Bearing	Radial clearance, C		Groove to ridge clearance ratio, H	Fractional frequency whirl onset speed at zero load, N_s, rpm
	μin.	mm			μin.	mm		
J-11	3350	0.085	Partial	B-29	450	0.011	8.5	4300
				B-28	800	0.020	5.2	3200
				B-26	1250	0.032	3.7	2600
				B-27	1600	0.041	3.1	1300
J-11R	2660	0.068	Partial	B-30	450	0.011	6.9	5300
				B-32	800	0.020	4.3	4000
				B-29	1250	0.032	3.1	3000
				B-28	1650	0.042	2.6	1500
J-14	1280	0.032	Partial	B-29	450	0.011	3.8	7000
				B-28	800	0.020	2.6	6000
				B-26	1250	0.032	2.0	3800
				B-27	1600	0.041	1.8	1800
J-14R	700	0.018	Partial	B-30	350	0.009	3.0	7500
				B-34	550	0.014	2.3	7000
				B-32	750	0.019	1.9	6500
				B-29	1150	0.029	1.6	3300
				B-28	1550	0.039	1.4	1200

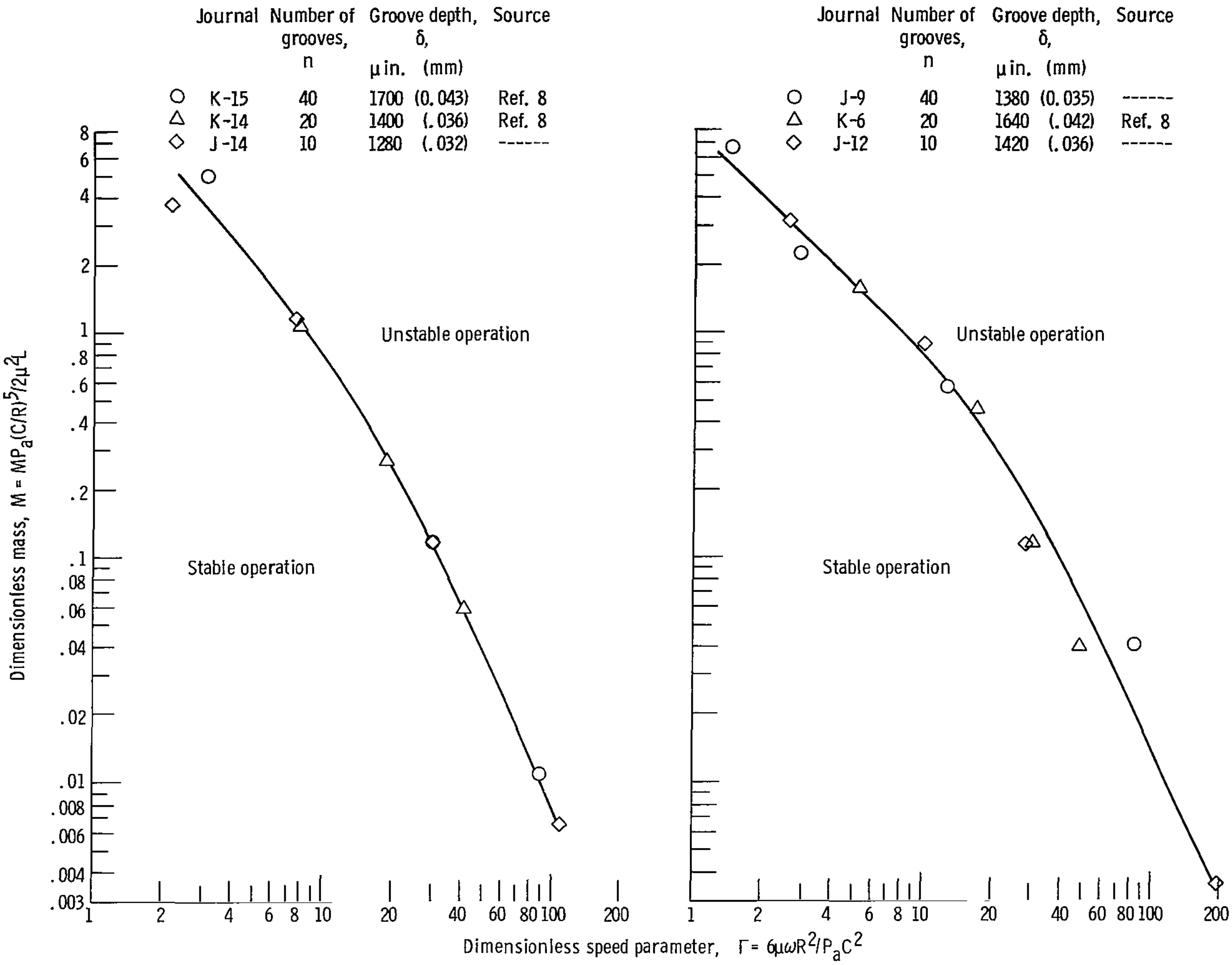
TABLE II. - Concluded. TEST RESULTS ON JOURNALS

WITH 40, 20, AND 10 GROOVES

(c) Concluded. 10 Grooves

Journal	Groove depth, δ		Groove type	Bearing	Radial clearance, C		Groove to ridge clearance ratio, H	Fractional frequency whirl onset speed at zero load, N_s, rpm
	μin.	mm			μin.	mm		
J-14RL	340	0.009	Partial	B-37	500	0.013	1.7	6 500
				B-38	950	0.024	1.4	4 000
				B-36	1300	0.033	1.3	1 000
[b]J-10	1350	0.034	Full	B-39	750	0.019	2.8	9 500
				B-31	1200	0.030	2.1	7 500
				B-40	1400	0.036	2.0	6 200
				B-35	1650	0.042	1.8	5 400
J-12	1420	0.036	Full	B-29	400	0.010	4.6	10 000
				B-28	800	0.020	2.8	5 800
				B-26	1200	0.030	2.2	4 600
				B-27	1550	0.039	1.9	2 000
J-12R	960	0.024	Full	B-32	500	0.013	2.9	7 500
				B-33	600	0.015	2.6	7 500
				B-36	750	0.019	2.3	7 000
				B-29	1000	0.025	2.0	5 300
				B-28	1400	0.036	1.7	1 600

[b]Bearing tests with one-third the housing mass of other bearings.



(a) Partially grooved journals.

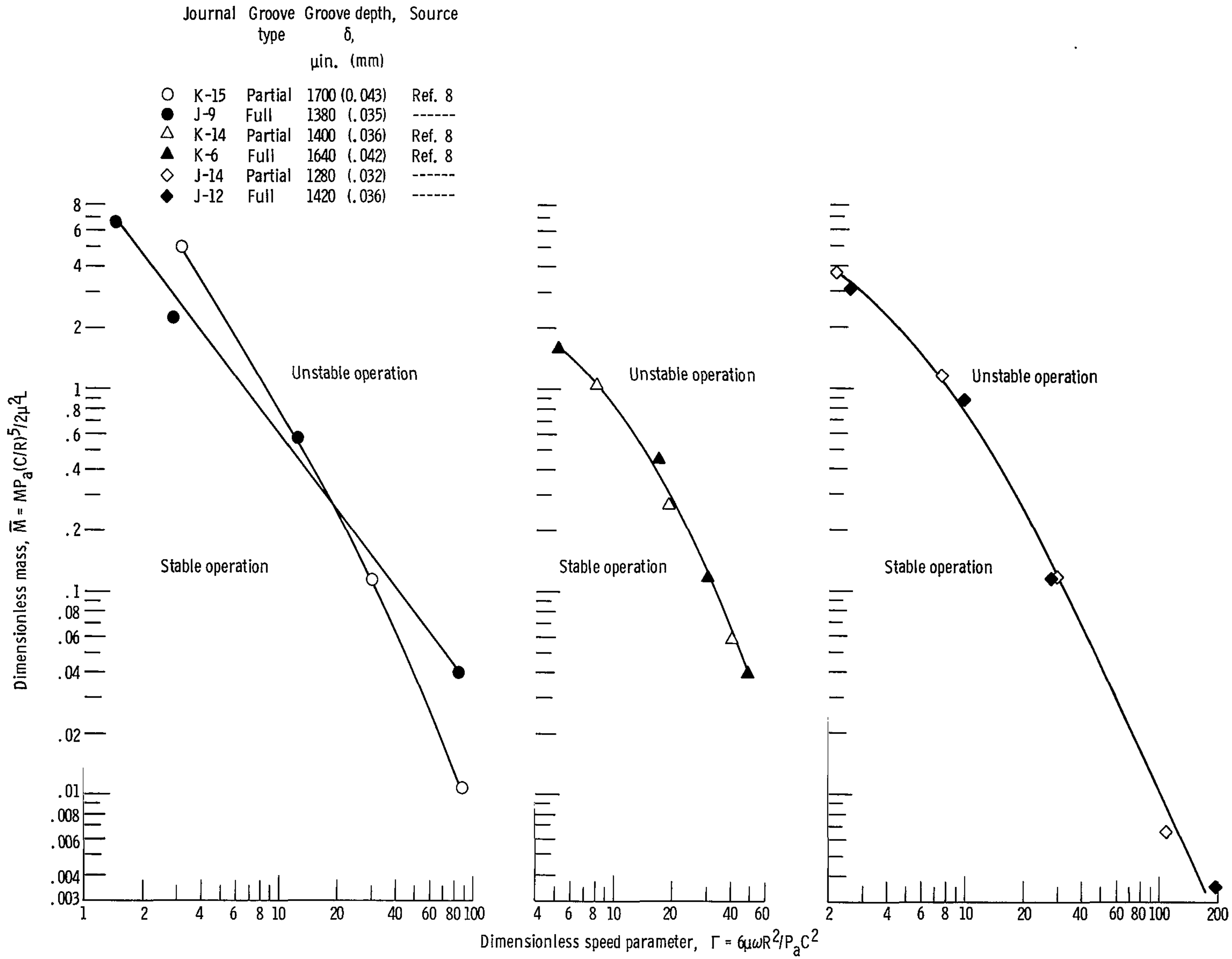
(b) Fully grooved journals.

Figure 5. - Effect of number of grooves on stability of water-lubricated herringbone-grooved journal bearings having 40, 20, and 10 grooves.

grooved journals. How this difference in groove length affects stability is shown in figure 6. A study of this figure shows that the stability characteristics of only the 40-groove journals are appreciably affected by groove length. In figure 6(a), at $\overline{M}$ values above 0.25, the journal with the shorter grooves (partially grooved) had better stability than the one with the longer grooves (fully grooved). At $\overline{M}$ values less than 0.25, however, the opposite was true.

For the journals with 20 and 10 partial and full grooves, the effect of groove length on stability was negligible (figs. 6(b) and (c)).

As has been pointed out, there are individual differences in the stability characteristics of herringbone-grooved journals when considering the effect of number of grooves and groove length. However, these differences are seen to be minimal when all the data



(a) Journal with 40 partial and 40 full grooves.

(b) Journal with 20 partial and 20 full grooves.

(c) Journal with 10 partial and 10 full grooves.

Figure 6. - Effect of partial and full length grooves on stability of water-lubricated herringbone-grooved journals.

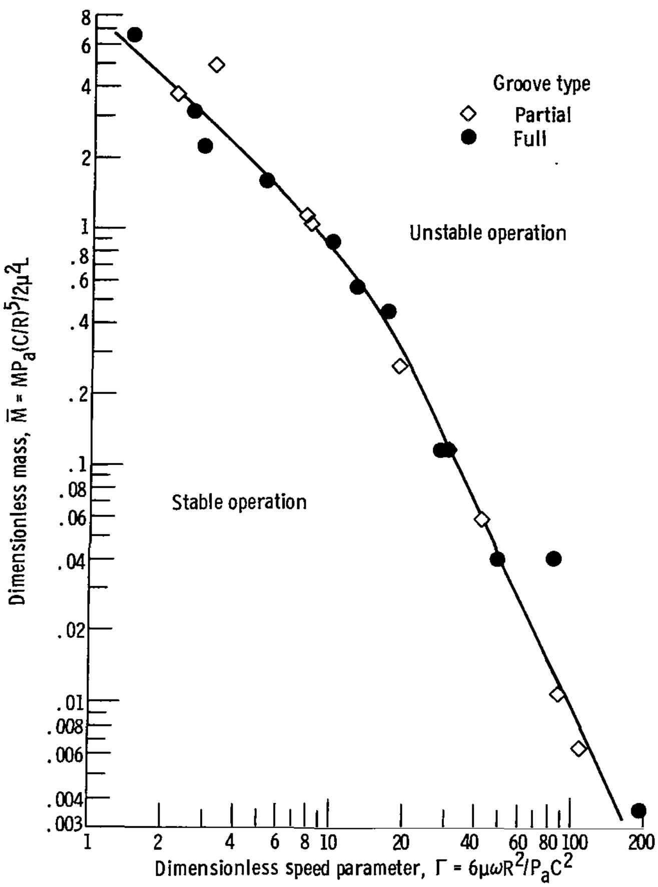
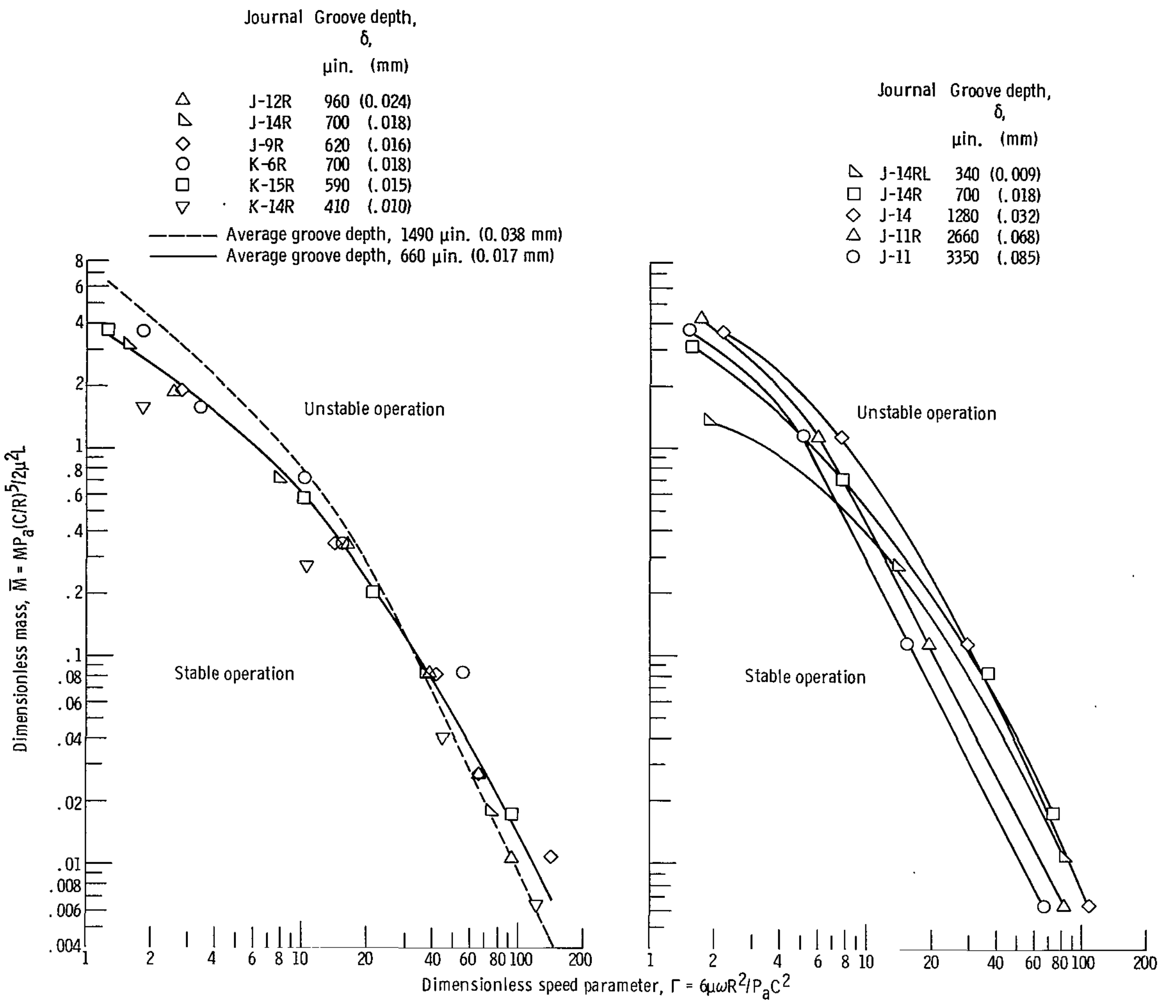


Figure 7. - Mean stability of herringbone journal bearings with 10, 20, and 40 partial and full grooves. Range of groove depth, 1280 to 1700 microinches (0.032 to 0.043 mm).

are grouped to form one stability plot. Such a curve is shown in figure 7, where a smooth curve has been drawn through the collective data for all journal bearings having 40, 20, and 10 partial and full grooves ranging in depth from 1280 to 1700 microinches (0.032 to 0.043 mm).

Effect of Groove Depth on Stability

The effect of groove depth on stability is shown in figure 8. Figure 8(a) shows the effect of reducing the groove depth for journals having 40, 20, and 10 partial and full grooves from an average of 1490 microinches (0.038 mm) to an average groove depth of 660 microinches (0.017 mm). The dashed curve is the curve in figure 7 and represents journals with an average groove depth of 1490 microinches (0.038 mm), and the solid



(a) Journals with 10, 20, and 40 partial and full grooves. (b) Journal with 10 partial grooves.

Figure 8. - Effect of groove depth on stability of water-lubricated herringbone-grooved journal bearings.

curve represents the mean stability of a group of herringbone-grooved bearings with groove depths ranging from 410 to 960 microinches (0.010 to 0.024 mm) averaging 660 microinches (0.017 mm). Stability decreased with a decrease in groove depth for bearings with $\overline{M}$ values above 0.12 and increased slightly with a decrease in groove depth for bearings with $\overline{M}$ values below 0.12.

In general, figure 8(a) shows that the optimum groove depth depends on clearance. The optimum groove depth changes with $\overline{M}$ and Γ both of which are dependent on the clearance; a larger clearance requires a larger groove depth for maximum stability.

Figure 8(b) shows the stability limits of a journal having 10 partial grooves at five different groove depths. As the groove depth was increased from 340 to 1280 microinches (0.009 to 0.032 mm), the stability increased. Then, as the groove depth was further increased from 1280 microinches (0.032 mm) to a maximum value of 3350 microinches (0.085 mm), the stability gradually decreased. Maximum stability occurred at a groove depth of 1280 microinches (0.032 mm).

Effect of Groove to Ridge Clearance Ratio on Stability

The herringbone journal helix angle and groove to ridge width ratio (table I) were set at those values which yield a maximum radial force as calculated from reference 7 at a compressibility number of zero for a compressible lubricant. This approximates the conditions existing with an incompressible lubricant. The optimum groove to ridge clearance ratio H is 2.1 for this condition (ref. 7). Figure 9 shows plots of H against

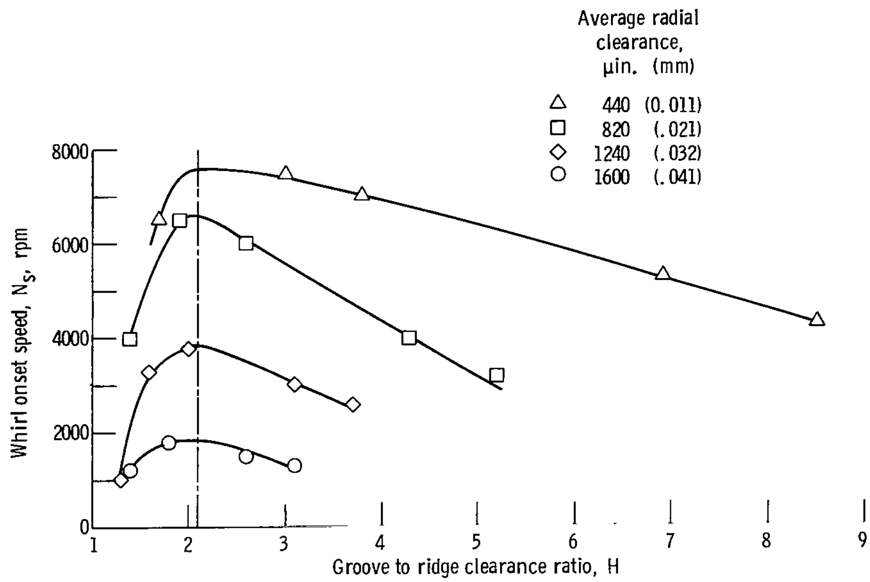


Figure 9. - Effect of groove to ridge clearance ratio on stability of herringbone journal bearing with 10 partial grooves in water (zero load).

zero load whirl onset speed at four different clearance values. The zero load whirl onset speed at each clearance reached a maximum at an H value very close to 2.1, indicating that this value of H is also optimum for stability. These curves were obtained from data on a herringbone journal having 10 partial grooves.

Figure 10 shows a dimensionless plot of H against the stability parameter $\overline{M\Gamma}$. The data from partially and fully grooved journals with 40, 20, and 10 grooves were combined to obtain the five short-dashed curves shown and represent five different clearances. Although the data show some scatter caused by minor variations in the stability

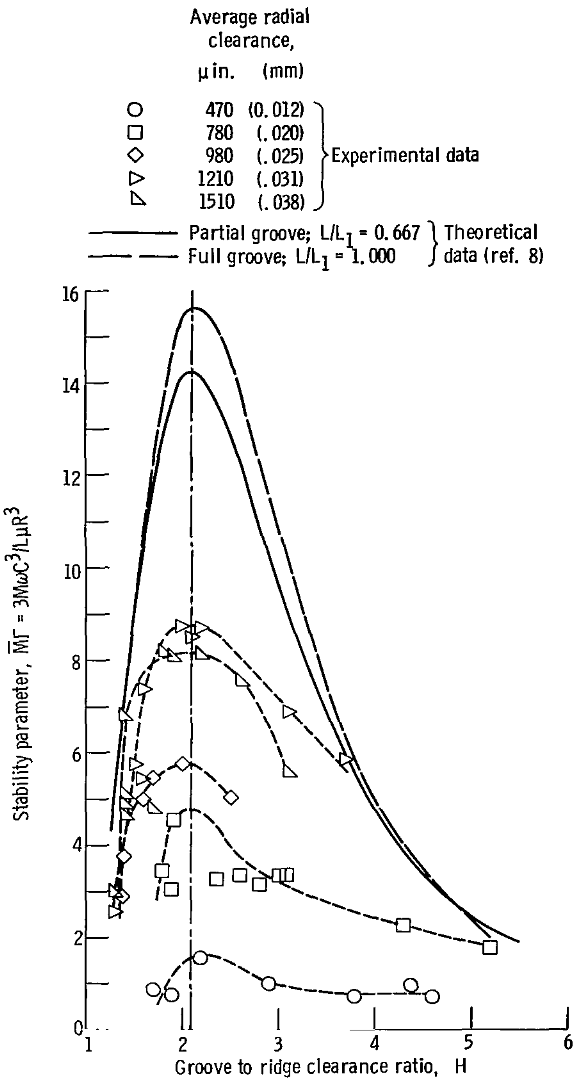


Figure 10. - Comparison of theoretical and experimental stability data using dimensionless parameters. Helix angle, 33°; groove width ratio, 0.5; length to diameter ratio, 1.

limits of the different journal configurations, the curves all tend to peak at a value of H very close to 2.1.

Once a value for radial clearance has been chosen for a particular herringbone-grooved journal bearing, the groove depth for maximum stability can be reliably determined from the H = 2.1 value for incompressible flow.

The theoretical curves for a herringbone journal bearing having partial and full grooves are also shown in figure 10 as the solid and long-dashed curves, respectively. Data for these curves were obtained from the computer program of reference 8. These

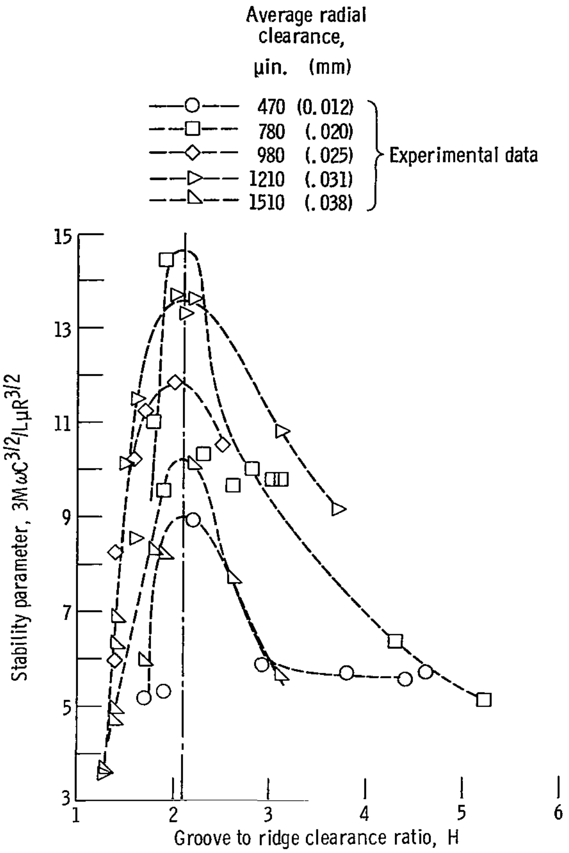


Figure 11. - Experimental stability data for herringbone-grooved bearings. Helix angle, 33°; groove width ratio, 0.5; length to diameter ratio, 1.

two curves also peak at an H value of 2.1. This indicates that maximum stability is predicted to occur at the same H value as the maximum radial force at a compressibility number of zero (for a compressible lubricant), namely, at H = 2.1 (ref. 7). Figure 10 also shows that theory predicts a larger range of stable operation than that observed experimentally. Similar results were reported in references 6 and 8, indicating that there may be an important parameter missing in the theoretical analysis.

The data in figure 10 are repeated in figure 11 using a different stability parameter $3M\omega C^{3/2}/L\mu R^{3/2}$ in an effort to bring the experimental curves closer together. The parameter $\overline{M\Gamma} = 3M\omega C^3/L\mu R^3$ was used in figure 10 because this was used in the theoretical stability analysis in reference 8.

Effect of Changes in Housing Mass per Bearing

The mass of the test housing M that was used in all but four stability tests in this investigation was 0.286 pound seconds^2 per inch (45.4 kg). The remaining four tests were run with a plastic test housing which resulted in a mass of 0.084 pound second^2 per inch (13.3 kg), or 30 percent of the mass of the heavier test housing. Stability curves are shown in figure 12 for a herringbone journal having 10 full grooves run at the two different mass conditions. The close correlation of the data for the two curves demonstrates the validity of the chosen dimensionless parameters $\overline{M}$ and Γ.

	Journal	Groove depth, δ, μin.	(mm)	Housing mass per bearing, (lb)(sec^2)/in.	(kg)
○	J-10	1350	(0.034)	0.084	(13.3)
◇	J-12	1420	(.036)	.286	(45.4)

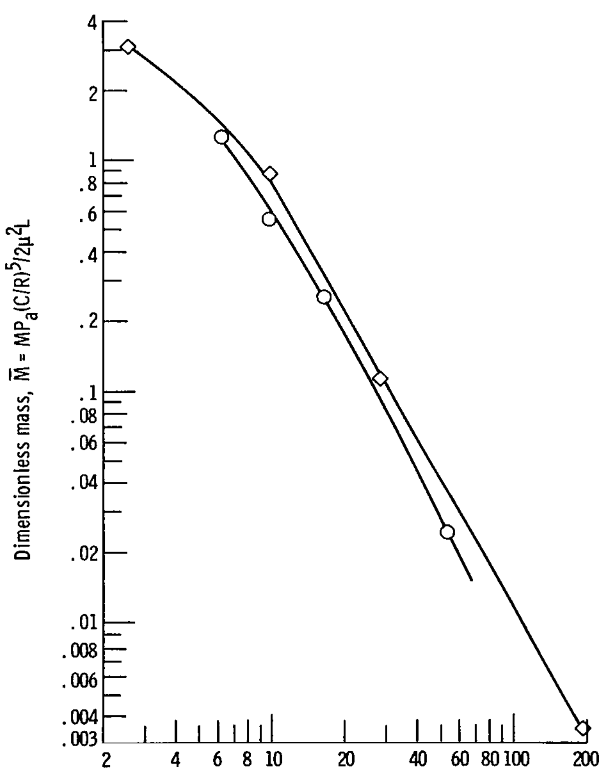


Figure 12. - Herringbone journal having 10 full grooves run with plain bearing at two different rotor mass values.

SUMMARY OF RESULTS

A total of 67 stability tests was performed on the herringbone-grooved journal bearings of six different configurations. Three journals with 10, 20, and 40 full grooves and three with 10, 20, and 40 partial grooves were evaluated. The groove depths ranged from 340 to 3350 microinches (0.009 to 0.085 mm). The bearings were run hydrodynamically in water at an average temperature of 80^{O} F (300 K) at speeds to 11 500 rpm. Bearing diameter was nominally 1.5 inches (3.8 cm), and the bearing had a length to diameter ratio of 1. The following results were obtained:

1. Groove length and number of grooves had no marked effect on stability when the data for journal hearings with 40, 20, and 10 partial and full grooves are taken as a whole. For individual bearing configurations, some effects are discernible.

2. The stability of the journal bearings with 40, 20, and 10 partial and full grooves was maximum when the groove to ridge clearance ratio was closest to 2.1 for a range of clearance values. This is as predicted by incompressible flow theory.

3. Maximum stability is achieved at the same groove to ridge clearance ratio as is the maximum radial force at a compressibility number of zero for a compressible lubricant, namely, at 2.1.

4. Theory predicts a larger range of stable operation for herringbone-grooved journal bearings than was observed experimentally.

5. The close correlation of stability data with two different housing mass values demonstrated the validity of the chosen dimensionless mass and speed parameters in the stability plots.

6. A maximum in stability was reached with a journal having 10 partial grooves at a depth of 1280 microinches (0.032 mm) as the depth was varied from 340 to 3350 microinches (0.009 to 0.085 mm).

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, March 14, 1969,
120-27-04-91-22.

APPENDIX - SYMBOLS

a_g width of helical groove, in.; mm

a_r width of helical ridge, in.; mm

C bearing radial clearance, in.; mm

g gravitational constant, in./sec^2; m/sec^2

H groove clearance to ridge clearance ratio, h_g/C

h_g groove clearance, $C + \delta$, in.; mm

L bearing length, in.; cm

M rotor mass per bearing, W_r/g, (lb)(sec^2)/in.; kg

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

N_s whirl onset speed at zero load, rpm

n number of grooves

P_a atmospheric pressure, lb/in.^{2} abs; N/m^2 abs

R journal radius, in.; cm

W_r total weight of housing (test vessel weight), lb; N

α groove width ratio, $a_g/(a_g + a_r)$

β helix angle, deg

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

δ groove depth, in.; mm

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

ω journal angular speed, rad/sec

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