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EXPERIMENTS ON THE STABILITY OF WATER-LUBRICATED RAYLEIGH STEP HYDRODYNAMIC JOURNAL BEARINGS AT ZERO LOAD

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16. Abstract

Tests were conducted with four different bearing configurations. The bearings, with nominal 3.8-centimeter (1.5-in.) diameter and length, were tested at 300 K (80° F) at speeds to 6300 rpm. In this report a pad is defined as a combination of feed groove and step, and ridge regions. A segment is defined as an independently acting 360° region; it may incorporate one or more pads. The four stepped-bearing configurations tested, in order of diminishing stability were: one-segment, three-pad shrouded; one-segment, one-pad shrouded; one-segment, three-pad unshrouded; and three-segment, one-pad shrouded. An increase in stability can be gained with a shrouded over an unshrouded step bearing. Stability becomes more sensitive to the ratio of step to ridge film thickness as clearance is increased. Five fixed geometry bearings were rated in order of diminishing stability as follows: (1) three-lobe bearing with minimum film thickness at trailing end of each lobe, (2) herringbone-groove bearing, (3) one-segment, three-pad shrouded Rayleigh step bearing, (4) three-lobe bearing, centrally lobed, and (5) three-segment, one-pad shrouded Rayleigh step bearing.

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EXPERIMENTS ON THE STABILITY OF WATER-LUBRICATED RAYLEIGH STEP HYDRODYNAMIC JOURNAL BEARINGS AT ZERO LOAD

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SUMMARY

A series of stability tests was conducted with 3.8-centimeter- (1.5-in.-) diameter, 3.8-centimeter- (1.5-in.-) long bearings at 300 K (80° F) at speeds to 6300 rpm. A pad is defined as a combination of feed groove and step, and ridge regions. A segment is defined as an independently acting 360° region; it may incorporate one or more pads. The four stepped bearing configurations tested, in order of diminishing stability were: (1) one-segment, three-pad shrouded, (2) one-segment, one-pad shrouded, (3) one-segment, three-pad unshrouded, and (4) three-segment, one-pad shrouded. An increase in stability can be gained with a shrouded over an unshrouded Rayleigh step bearing of the same configuration. Stability becomes more sensitive to the ratio of step to ridge film thickness as clearance is increased.

Five fixed geometry bearings including two Rayleigh step bearing configurations can be generally rated in order of diminishing stability as follows: (1) three-lobe bearing with minimum film thickness at the trailing end of each lobe, (2) herringbone-groove bearing, (3) one-segment, three-pad shrouded Rayleigh step bearing, (4) three-lobe bearing, centrally lobed and (5) three-segment, one-pad shrouded Rayleigh step bearing.

INTRODUCTION

Fractional frequency whirl, more than any other factor, limits the usefulness of fluid film bearings at low loads and high speeds when lubricants of low viscosity, such as water or liquid metals, are used. It is of prime importance that journal bearings capable of inhibiting this self-excited whirl be employed in these applications.

The tilting pad configuration has excellent stability characteristics in low viscosity fluids such as sodium but is complex since it is composed of several parts, and may be subject to pivot surface damage (ref. 1). To avoid this complexity, a more practical bearing would be one with a fixed geometry. It is known that discontinuous films, such as those produced by steps and grooves, tend to stabilize a bearing (ref. 2). One such bearing that showed good stability properties is the herringbone-groove bearing (refs. 3 to 6). Another fixed geometry bearing that has shown promise is the lobed bearing (refs. 7 to 10), which has a marked similarity to the tilting pad bearing except that its pads (lobes) are fixed in one position. References 7 and 10 contain analytical investigations of centrally lobed bearings. Reference 8 is an experimental investigation of a three-lobe bearing with lobes tilted on their trailing edges which resulted in a bearing that was generally more stable than the herringbone-groove configuration. Reference 9 compares the experimental stability data of centrally lobed bearings with those having lobes that are tilted at their trailing edges.

There is one fixed geometry journal bearing configuration, the Rayleigh step bearing, that has had very little investigation as far as stability characteristics are concerned. For this reason and because analysis has shown that the single step Rayleigh step journal bearing will carry a substantial load even though the bearing is in a concentric position (ref. 11), its stability characteristics will be investigated experimentally. In this report a pad is defined as a combination of feed groove and step and ridge regions. A segment is defined as an independently acting 360° region; it may incorporate one or more pads.

A Rayleigh step bearing with only a single pad in its circumference results in maximum load capacity (ref. 12). However, it has the disadvantage that the load capacity varies considerably with the direction of the applied load. In spite of this disadvantage, the stability of this type of bearing (see fig. 1) was investigated. Two other Rayleigh step bearing configurations were also chosen for testing. One utilizes the single-pad configuration and is composed of three circumferential segments side by side (fig. 2). Each segment, with a length one-third the entire bearing length, contains one pad and is displaced 120° circumferentially with respect to the other segments. The symmetry thus achieved largely eliminates the disadvantage of anisotropy. The other configuration consists of three pads extending the entire length of the bearing as shown in figure 3. The ridge to pad arc ratio ψ is defined as the ratio of the arc subtended by the ridge, to the arc subtended by the pad (fig. 4). The trisymmetrical arrangements were selected mainly to enable direct comparison of stability data with previously tested three lobe bearings in references 8 and 9.

The objectives of this study were (1) to observe the effect of film thickness ratio and ridge to pad arc ratio on stability, (2) to compare the stabilities of four different Rayleigh step journal bearing configurations: (a) the three-segment, one-pad shrouded, (b) the one-segment, three-pad shrouded, (c) the one-segment, one-pad shrouded and (d) the one-segment, three-pad unshrouded, (3) to generate design curves to facilitate the design of optimum-geometry Rayleigh step bearings; and (4) to compare the stability of Rayleigh step bearings of different geometries with three-lobe and herringbone-groove bearings.

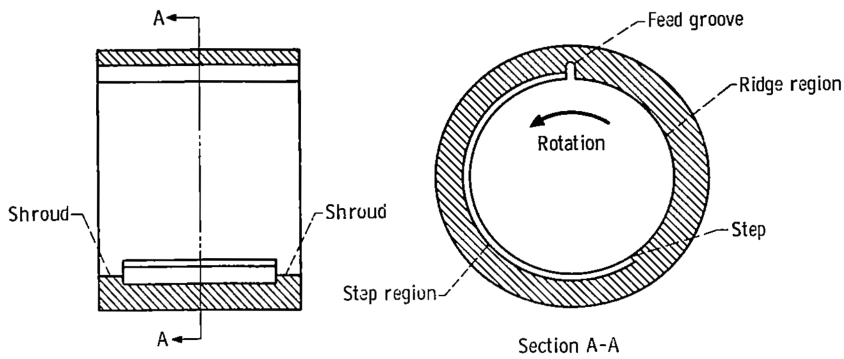


Figure 1. - One-segment, one-pad shrouded Rayleigh step journal bearing.

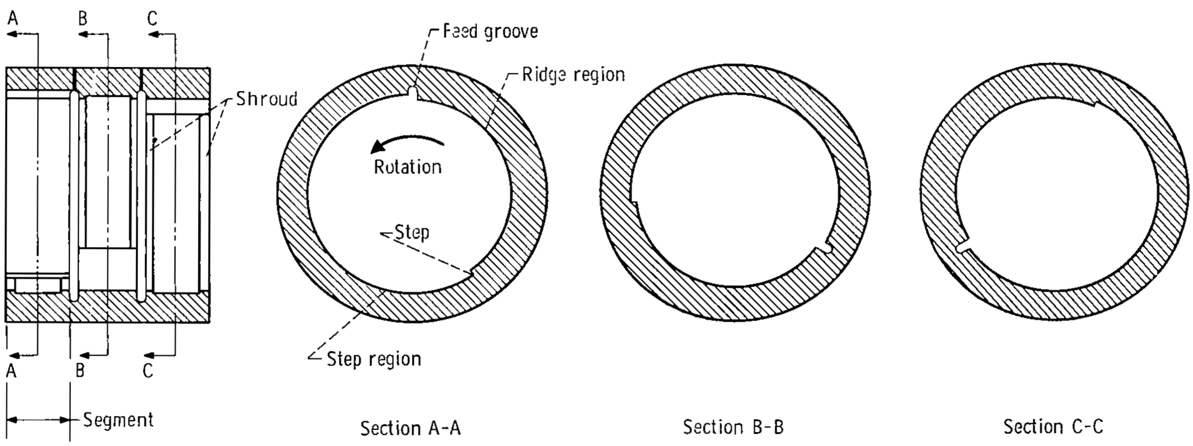


Figure 2. - Three-segment, one-pad shrouded Rayleigh step journal bearing.

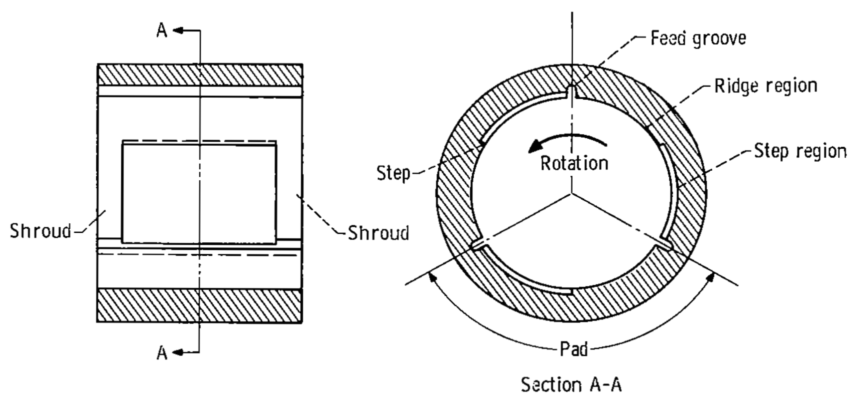


Figure 3. - One-segment, three-pad shrouded Rayleigh step journal bearing.

The bearings had a nominal 3.8-centimeter (1.5-in.) diameter and were 3.8 centimeters (1.5 in.) long. They were submerged in water at an average temperature of 300 K (80° F) and were operated hydrodynamically at journal speeds to 6300 rpm at zero load.

SYMBOLS

C	radial clearance in ridge region when bearing and journal are concentric, mm; in.
C_s	radial clearance in step region when bearing and journal are concentric, $C + S$, mm; in.
D	journal diameter, cm; in.
e	eccentricity, mm; in.
g	gravitational constant, m/sec^2; $in./sec^2$
k	film thickness ratio, C_s/C
L	bearing length, cm; in.
M	supported mass per bearing, W_r/g, kg; $(lb)(sec^2)/in.$
$\overline{M}$	dimensionless mass parameter, $MP_a(C/R)^5/2\mu^2 L$
N_w	fractional-frequency-whirl onset speed at zero load, rpm
P_a	atmospheric pressure, N/m^2; psia
R	journal radius, cm; in.
S	depth of step, mm; in.
W_r	total weight of test vessel, N, lb
β	angle subtended by step, deg
Γ	dimensionless speed parameter, $6\mu\omega R^2/P_a C^2$
γ	angle subtended by ridge, deg
δ	angle subtended by lubrication groove, deg
ϵ	eccentricity ratio, e/C
μ	lubricant dynamic viscosity, $(N)(sec)/m^2$; $(lb)(sec)/in.^2$
ψ	ratio of ridge angle to total angle of ridge-step-groove combination (ridge to pad arc ratio), $\gamma/(\gamma + \beta + \delta)$
ω	journal angular speed, rad/sec

DEFINITIONS

Configuration - relative disposition of pads and segments (figs. 1 to 3)

Feed groove - axial groove that runs the length of the segment and feeds lubricant to the bearing (fig. 3)

Geometry - arc lengths and step depths that describe the step and ridge regions (fig. 4)

Pad - a combination of feed groove and step and ridge regions (fig. 3)

Ridge region - raised area of a pad not including the shrouds (fig. 3)

Segment - an independently acting 360° region; it may incorporate one or more pads (fig. 2)

Shroud - side rails at the axial ends of the segment in the step region that act as a dam (fig. 3)

Step region - relieved area of a pad (fig. 3)

APPARATUS

Test Bearings

Rayleigh step journal bearings of four different configurations were evaluated (figs. 1 to 3). One consisted of one segment running the length of the bearing with one pad in its circumference as shown in figure 1. Another consisted of three circumferential segments of equal length, side by side, separated from each other by grooves as shown in figure 2. Each segment contained one pad that was displaced 120° circumferentially with respect to the others to achieve symmetry in the bearing. The length of each segment was one-third the total length of the bearing. One set of such bearings had a ridge to pad arc ratio ψ of 0.40 and another set of bearings had a ψ of 0.49. These ridge to pad arc ratios were selected on the basis of the load capacity results of reference 12. Since the load capacity curve of reference 12 has a rather flat peak, they both represent conditions which yield maximum load capacity for a single sector Rayleigh step bearing at the small eccentricity ratios ($\epsilon = 0$ to 0.3), that can be expected with a bearing under zero load.

The third Rayleigh step bearing configuration that was evaluated is shown in figure 3. It consisted of three pads extending the entire length of the bearing. One set of such bearings had a ridge to pad arc ratio ψ of 0.27 and another set had a ψ of 0.45. The ratios chosen were based on analytical work on load capacity for three-pad Rayleigh step bearings at an eccentricity ratio of 0.1 reported in reference 13.

All bearings tested were shrouded Rayleigh step bearings, except bearing RS5AM (table IV) which was the fourth configuration tested. After tests were completed with bearing RS5A with shrouds (table II(b)), the shrouds were machined off and the bearing number was changed to RS5AM and rerun without shrouds. The width of the shroud for the three-segment, single-pad bearing was 1.5 millimeters (0.06 in.) and 4.8 millimeters (0.19 in.) for the one-segment, three-pad and one-segment, one-pad bearings.

The various radial clearances (see tables I to IV) were obtained by varying the outside diameter of the journals for each bearing tested. The assembled bearings in all cases had a nominal 3.8-centimeter- (1.5-in.-) length and diameter. The inside surface of the bearings in the ridge area and the journal outside diameter were machined to a 0.1- to 0.2-micrometer (4- to 8-μin.) root mean square finish. The bottom corners of the steps in the bearings were not sharp but had a machined radius of approximately 0.97 centimeter (0.38 in.).

Figure 4 shows the parameters that define the Rayleigh step bearing geometry. The angle γ is defined as the angle subtended by the ridge region; β, the angle subtended by

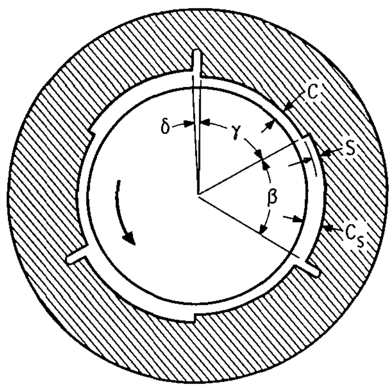


Figure 4. - Rayleigh step bearing geometry. Ridge to pad arc ratio $\psi = \gamma/(\gamma + \beta + \delta)$; film thickness ratio $k = C_s/C$.

the step region; and δ, the angle subtended by the lubrication groove. The latter angle δ is included since it was used in the analysis of references 11 and 12. Although this angle is normally small and may not be important for the one- and three-step configurations employed in this investigation, it can become important in an analysis of a Rayleigh step bearing with a large number of pads. The symbol C in figure 4 is the radial clearance between the journal and the ridge region and C_s is the radial clearance in the step region when the journal is in a concentric position in the bearing. The depth of the step S is then $C_s - C$.

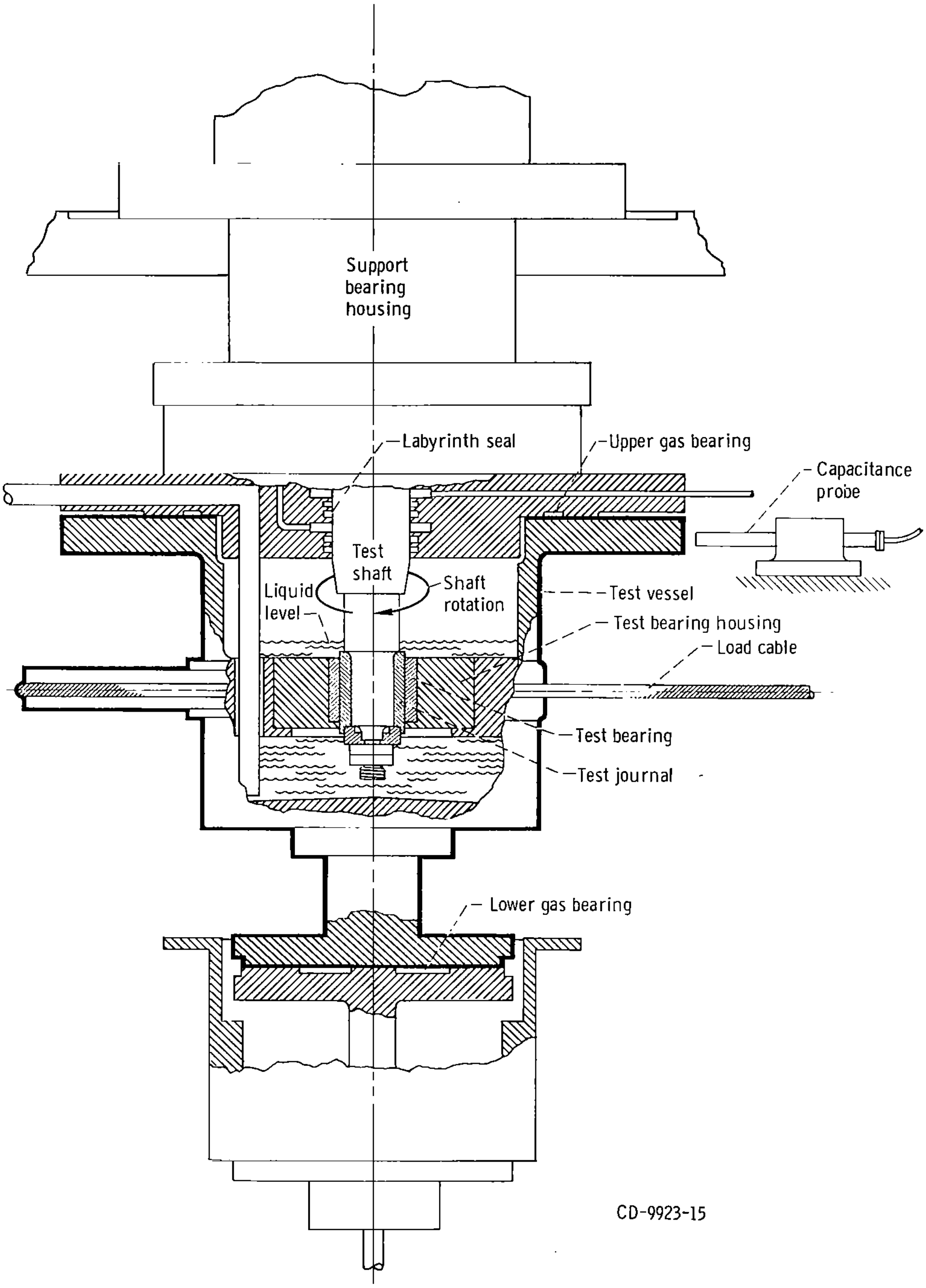


Figure 5. - Bearing test apparatus.

Bearing Test Apparatus

The test vessel and associated parts are shown in figure 5. 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 the upper and lower gas bearings. Bearing torque can be measured, if desired, by a force transducer attached to the floating test vessel.

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 test vessel 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 3.

Procedure

Details of the test procedure are given in reference 3. 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. 3), 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 photographing the whirl pattern on the oscilloscope screen.

In these experiments, the motion of the bearing with its massive housing was monitored. 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 3, where excellent correlation was obtained between theoretical and experimental data for a three-axial-groove bearing run in water with a plain journal.

RESULTS AND DISCUSSION

General

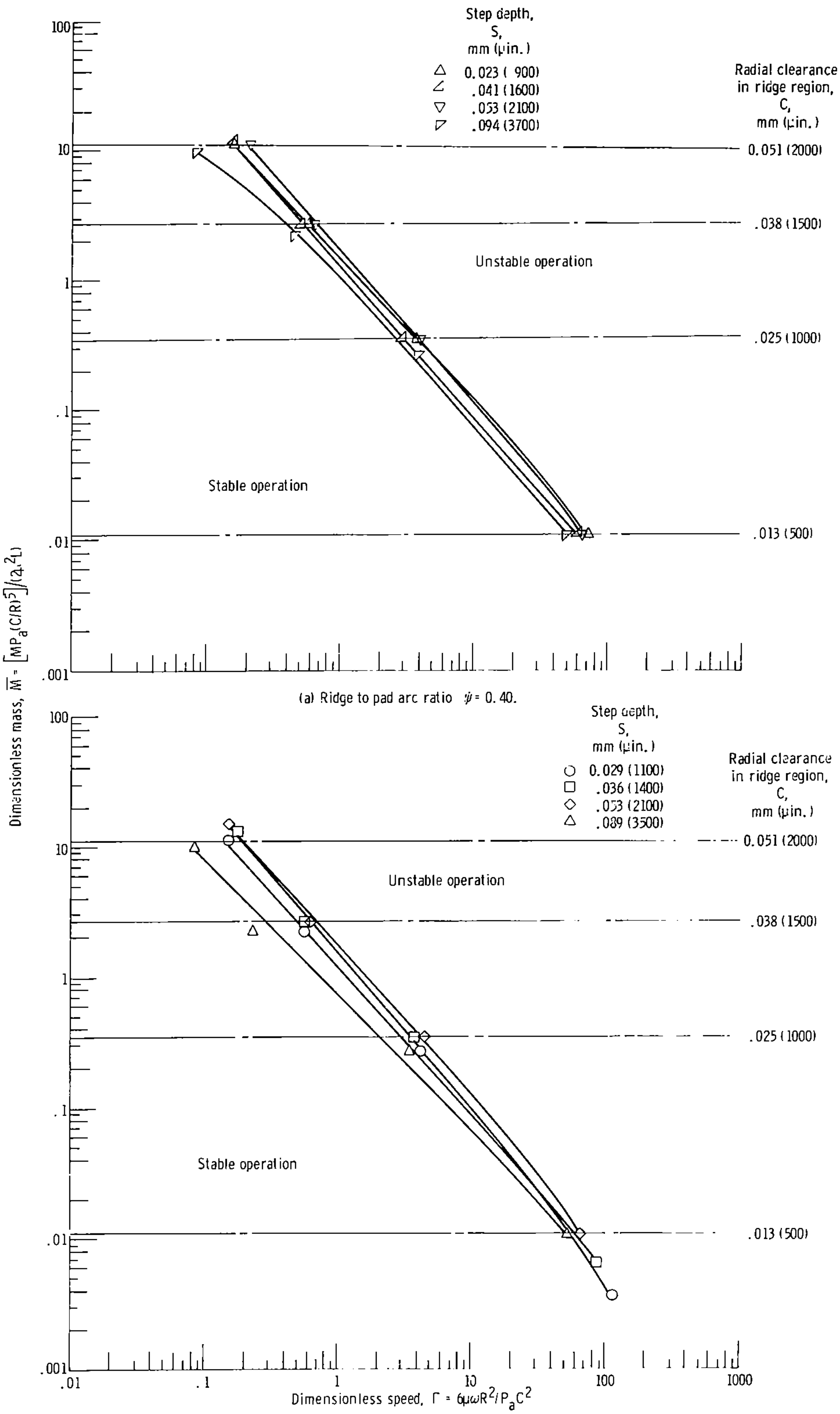
Experimental results obtained with three-segment, one-pad shrouded Rayleigh step bearings (fig. 2) with ridge to pad arc ratios ψ of 0.40 and 0.49 are shown in table I. Tests results for one-segment, three-pad shrouded bearings (fig. 3) with ψ of 0.27 and 0.45 are shown in table II. Table IV lists the results of a similar bearing with ψ of 0.45 but without shrouds. Test results for a Rayleigh step bearing with a single segment extending the length of the bearing with one pad in its circumference and $\psi = 0.4$ are listed in table III. A total of eighty bearing stability tests was conducted at radial clearance C ranging from 0.010 to 0.052 millimeter (400 to 2050 μin.) and step depths S ranging from 0.015 to 0.094 millimeter (600 to 3700 μin.).

The bearings were submerged in water at an average temperature of 300 K (80° F) and run hydrodynamically. Maximum speed attained without whirl was 6300 rpm.

Effect of Step Depth on Stability

The experimental results obtained with three-segment, one-pad shrouded Rayleigh step bearings are shown in figure 6. In the area labeled stable operation, to the left of the experimental curves, the bearings ran stably at zero load; to the right of the curves, fractional-frequency whirl occurred. The experimental curves represent the stability limits of the bearings tested and indicate the zero load threshold of stability. The theoretical stability analysis of a journal bearing in reference 3 showed that the important parameters to consider are the dimensionless mass parameter $\overline{M}$ and the dimensionless speed parameter Γ, as shown in figure 6. Figure 6(a) shows the stability limits of a three-segment, one-pad shrouded Rayleigh step bearing with a ridge to pad arc ratio of 0.40 at four different step depths. Maximum stability occurred at a step depth of 0.053 millimeter (2100 μin.), as the step depth was varied over a range of 0.023 to 0.094 millimeter (900 to 3700 μin.). This was also the case for similar bearings with ψ of 0.49 (fig. 6(b)) where maximum stability occurred at a step depth of 0.053 millimeter (2100 μin.), as the step depth was varied over a range of 0.029 to 0.089 millimeter (1100 to 3500 μin.).

The experimental results obtained with one-segment, three-pad shrouded Rayleigh step bearings are shown in figure 7. Threshold of stability curves for the bearings with ψ of 0.27 are shown in figure 7(a). The stability characteristics remained essentially the same for bearings with a range of step depths from 0.015 to 0.064 millimeter (600 to 2500 μin.), at $\overline{M}$ values below 5.0. A noticeable drop in stability occurred when the step depth was increased to 0.091 millimeter (3600 μin.). This was also true for similar bearings with ψ of 0.45 (fig. 7(b)) where stability was little affected by increased step depth at $\overline{M}$ values below 5.0, until a value of 0.089 millimeter (3500 μin.) was reached. Experimental data points for dimensionless mass $\overline{M}$ values above 5.0 for the bearings with ψ of 0.27 (fig. 7(a)) and ψ of 0.45 (fig. 7(b)) represent bearings with large clearances ranging from 0.048 to 0.052 millimeter (1900 to 2050 μin.). There is a greater variation in stability at these large clearances for both bearing types, ψ of 0.27 and 0.45 as the step depths are varied over their respective ranges.



(b) Ridge to pad arc ratio ψ = 0.49.

Figure 6. - Effect of step depth on stability of three-segment, one-pad shrouded Rayleigh step bearing.

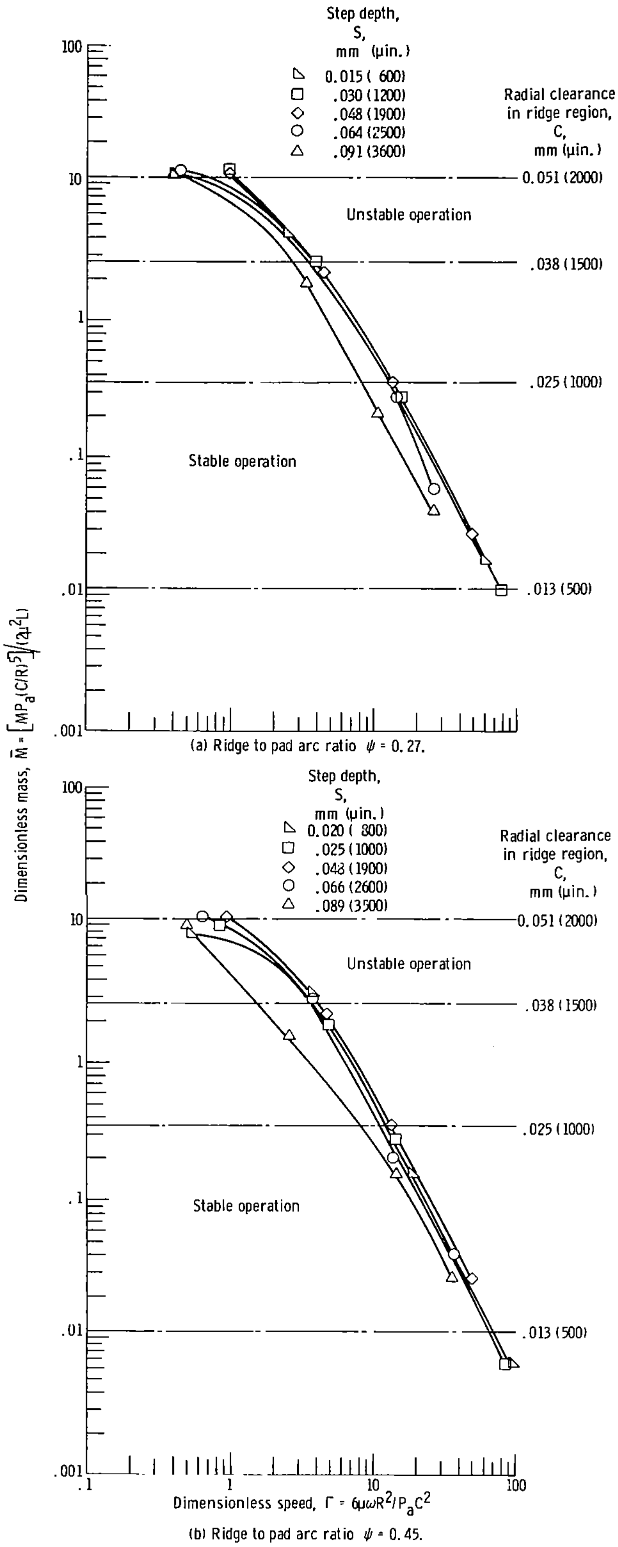


Figure 7. - Effect of step depth on stability of one-segment, three-pad shrouded Rayleigh step bearing.

Effect of Ridge to Pad Arc Ratio on Stability

The effect of ridge to pad arc ratio ψ on stability of the three-segment, one-pad shrouded Rayleigh step bearing is shown in figure 8. The solid and dashed curves were obtained from figures 6(a) and (b), respectively, and represent the stability characteristics of a three-segment, one-pad shrouded bearing at two different ψ values (0.40 to 0.49) but at like step depths of 0.053 millimeter (2100 μin.). A study of figure 8 shows that the stability of the three-segment, one-pad shrouded Rayleigh step bearing was not appreciably affected by a change in ridge to pad arc ratio ψ of 0.40 to 0.49, since the two curves are in such close proximity to one another.

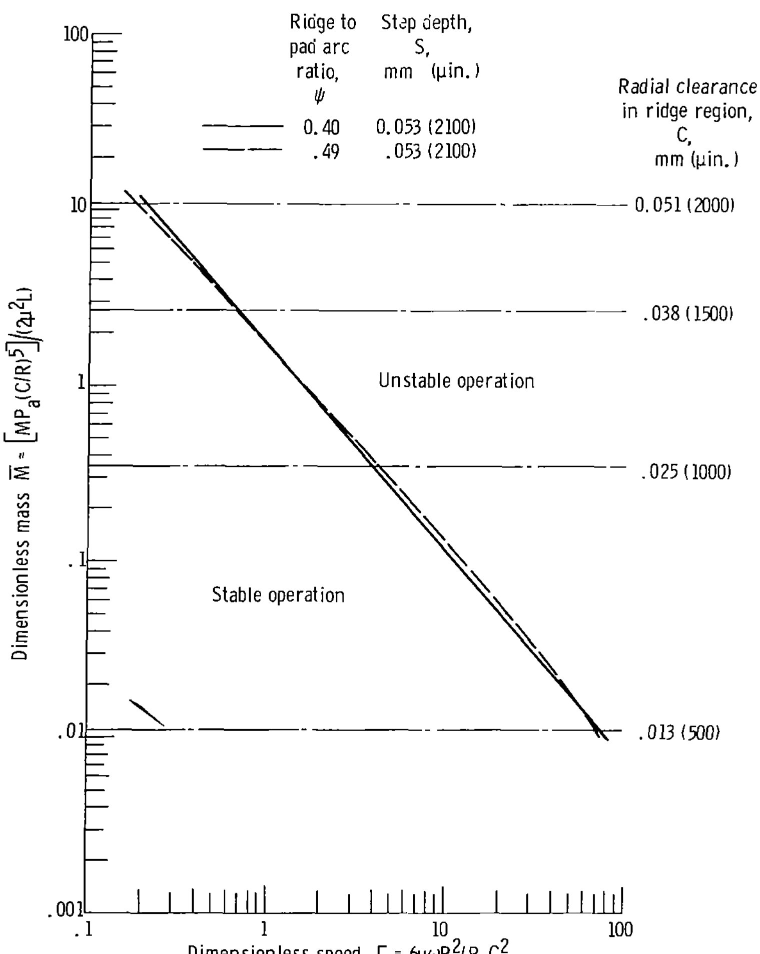


Figure 8. - Effect of ridge to pad arc ratio ψ on stability of three-segment, one-pad shrouded Rayleigh step bearing.

The two curves shown in figure 9 for a one-segment, three-pad shrouded Rayleigh step bearing were obtained in a similar manner to those in figure 8. The solid and dashed curves represent the stability characteristics of a one-segment, three-pad shrouded bearing at ψ of 0.27 and 0.45, respectively, both at a step depth of 0.048 millimeter (1900 μin.) obtained from figures 7(a) and (b). As in the case of the three-segment, one-pad bearing, the stability characteristics of the one-segment, three-pad bearing were not appreciably affected by a change in ψ even though this change in ψ

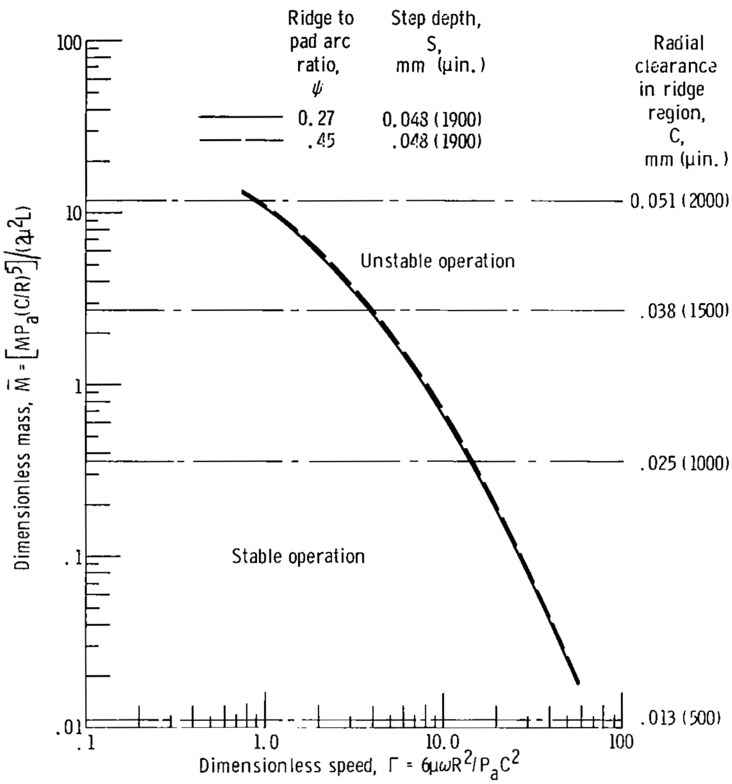


Figure 9. - Effect of ridge to pad arc ratio ψ on stability of one-segment, three-pad shrouded Rayleigh step bearing.

(0.27 to 0.45) was much greater than that for the three-segment, one-pad bearing (0.40 to 0.49).

Effect of Bearing Configuration on Stability

Comparison of the stability curves for the three-segment, one-pad bearing (fig. 8) with those for the one-segment, three-pad bearing (fig. 9) shows that the latter bearing configuration is the more stable of the two. For example, at a dimensionless mass $\overline{M}$

of 1.0, the dimensionless speed Γ is 1.6 for the three-segment, one-pad bearing (fig. 8) and 7.8 for the one-segment, three-pad bearing (fig. 9) indicating a substantial gain in stability with the one-segment, three-pad configuration. However, it must be remembered that the single-pad bearing is composed of three separate bearings (segments), each with a length to diameter ratio L/D = 1/3, whereas the three-pad bearing has an L/D = 1. It may be that this difference in L/D ratio between the two bearing configurations accounts for the main difference in their stability characteristics.

The main purpose for testing the one-segment, one-pad bearing (fig. 1) was to determine if it had any stability whatsoever. The results of the tests showed that this config-

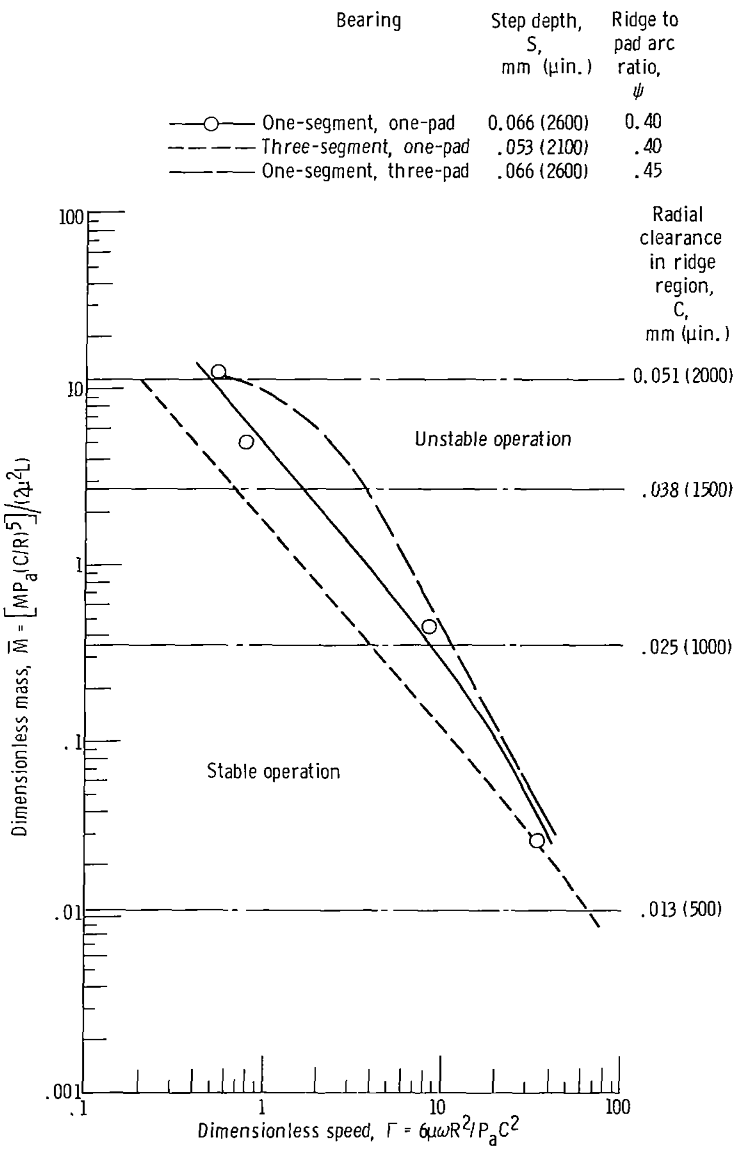


Figure 10. - Comparison of stability of shrouded Rayleigh step bearings of three different configurations.

uration does have very good stability characteristics even though it has the disadvantage of a preferential loading direction. Figure 10 shows a comparison of stability characteristics of the one-segment, one-pad bearing with that of the three-segment, one-pad and the one-segment, three-pad configuration. Bearings with approximately the same step depth S and ridge to pad arc ratio ψ were employed for this comparison. Figure 10 shows that the one-segment, three-pad configuration is the most stable of the three followed by the one-segment, one-pad bearing with the three-segment, one-pad bearing the least stable. The increase in stability of the one-segment, one-pad and the one-segment, three-pad over the three-segment, one-pad bearing in all probability is due to the larger L/D ratio of the first two bearing configurations.

Comparison of the Stability of a Shrouded And Unshrouded Rayleigh Step Bearing

The two curves in figure 11 show the threshold of stability of a shrouded and unshrouded one-segment, three-pad bearing with a ψ of 0.045. The data for both were ob-

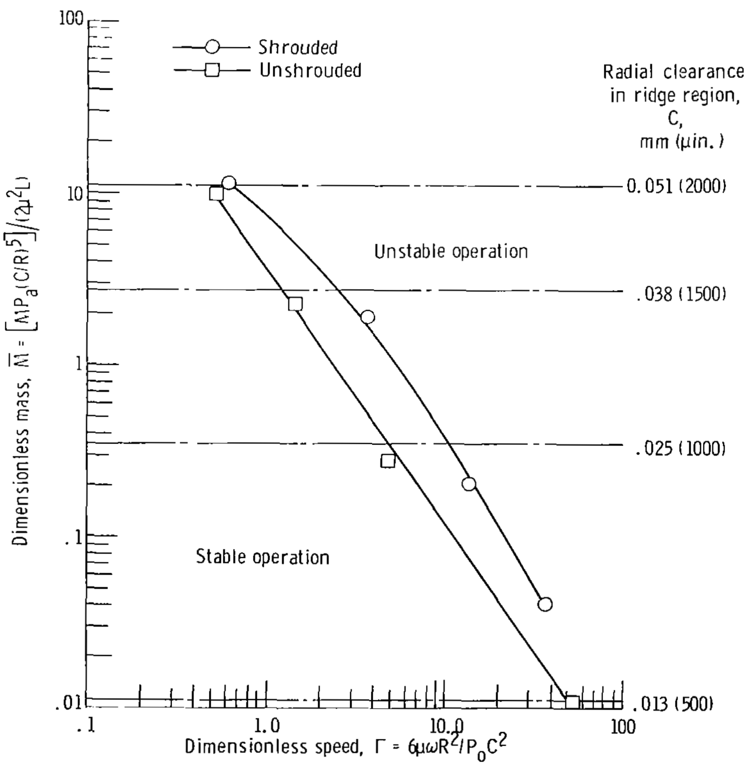


Figure 11. - Comparison of stability of shrouded and unshrouded one-segment, three-pad Rayleigh step bearing. Ridge to pad are ratio ψ = 0.45; step depth S = 0.066 millimeter (2600 μin.).

tained from the same bearing (number RS5A, table II(b)) which was first run with shrouds. The shrouds were then machined off and the bearing was renumbered RS5AM (table IV) and rerun without shrouds. Figure 11 shows that a substantial increase in stability can be gained by using shrouds since the dimensionless mass $\overline{M}$ for the shrouded bearing is always greater than that for the unshrouded bearing over the entire range of dimensionless speeds Γ tested.

Effect of Clearance on Stability

Figure 12 is a plot of whirl speed against film thickness ratio at four different clearances. At the larger clearances, 0.025 to 0.048 millimeter (1000 to 1900 μin.), the one-

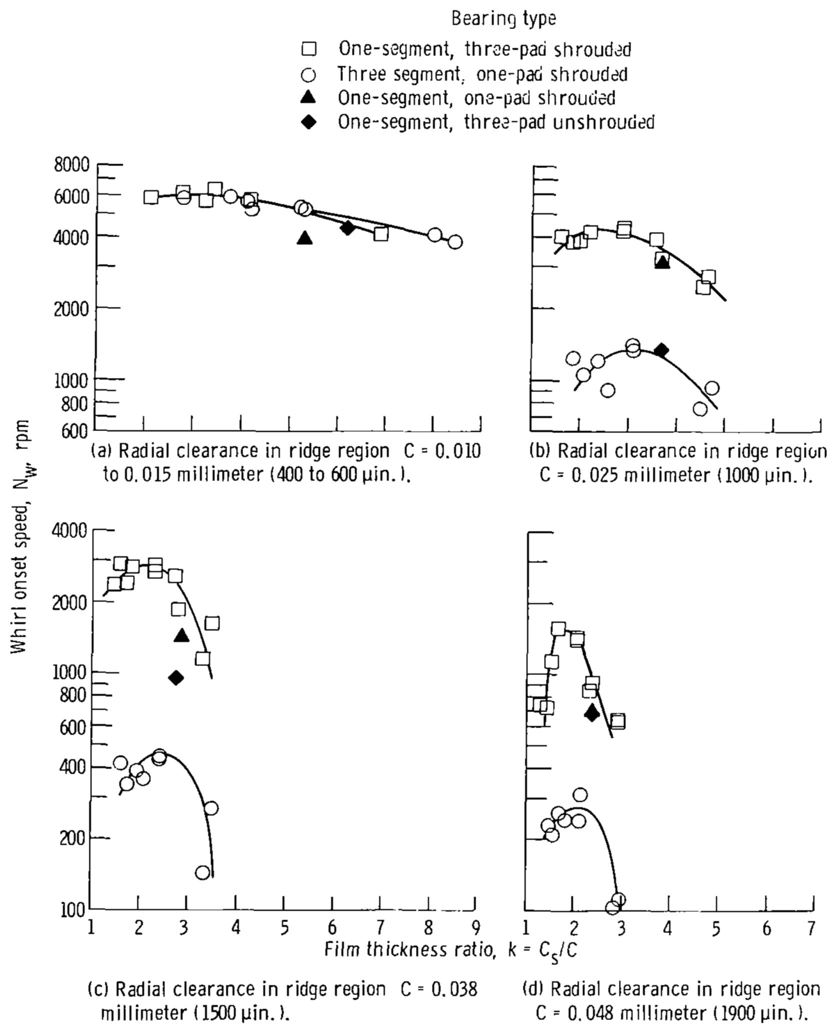


Figure 12. - Whirl onset speed as a function of film thickness ratio at various radial clearances.

segment, three-pad shrouded bearing, is clearly the most stable of the four Rayleigh step bearings tested. However, at the smallest clearance value, 0.010 to 0.015 millimeter (400 to 600 μin.), the stability of all four configurations tend to approach equal magnitude. In fact, the three-segment, one-pad shrouded bearings, show greater stability than the other three configurations, at high film thickness ratios (above 5.2), which is the opposite of their showing at the larger clearance value, 0.025 to 0.048 millimeter (1000 to 1900 μin.). This is possibly due to some slight angular misalinement of the bearing axis with the journal axis which tends to preload the bearings giving them a stability which might not be present under more ideal alinement conditions (see ref. 14). Reference 9 reports a plain bearing and journal running stably in water at a radial clearance of 0.009 millimeter (350 μin.) to a speed of 6200 rpm due apparently to preloading of the bearing by misalinement. For the Rayleigh step bearings (in fig. 12) at clearances above 0.015 millimeter (600 μin.), this preload is too small to affect the bearings resulting in a more accurate measure of their true stability. In light of the preceding discussion, the four Rayleigh step bearing configurations investigated herein can be rated in order of diminishing stability as follows: (1) one-segment, three-pad shrouded bearing, (2) one-segment, one-pad shrouded bearing, (3) one-segment, three-pad unshrouded bearing and (4) three-segment, one-pad shrouded bearing.

Design Curves

The data for the one-segment, three-pad shrouded Rayleigh step bearing with a ψ of 0.27 was replotted in slightly different form in order to facilitate the design of optimum-geometry bearings. Because of the close similarity of the stability data between the bearings with a ψ of 0.27 and 0.45 (fig. 9) the following design curves can be applied to either. Design curves for the three-segment, one-pad shrouded bearing were not considered here because of their relatively low stability. The one-segment, one-pad configuration was not considered because of its limitation of preferential loading direction and the fact that insufficient stability data were available.

Whirl onset speed is plotted against radial clearance in figure 13 for five different values of step depth for the one-segment, three-pad shrouded bearings. The values of the film thickness ratio $k = C_s/C$ (table II(a)) are given for each data point and represent the ratio of the clearance at the step region to the clearance at the ridge region of each bearing when the shaft is concentric in the bearing (see fig. 4). The four vertical dashed lines, drawn at arbitrary clearance values of 0.018, 0.025, 0.038, and 0.051 millimeter (700, 1000, 1500, and 2000 μin.), intersect the stability curves at various values of whirl onset speed. The curves in figure 14 were obtained by cross plotting the data in figure 13 at the four different values of clearance using whirl speed N_w and film thickness ratio

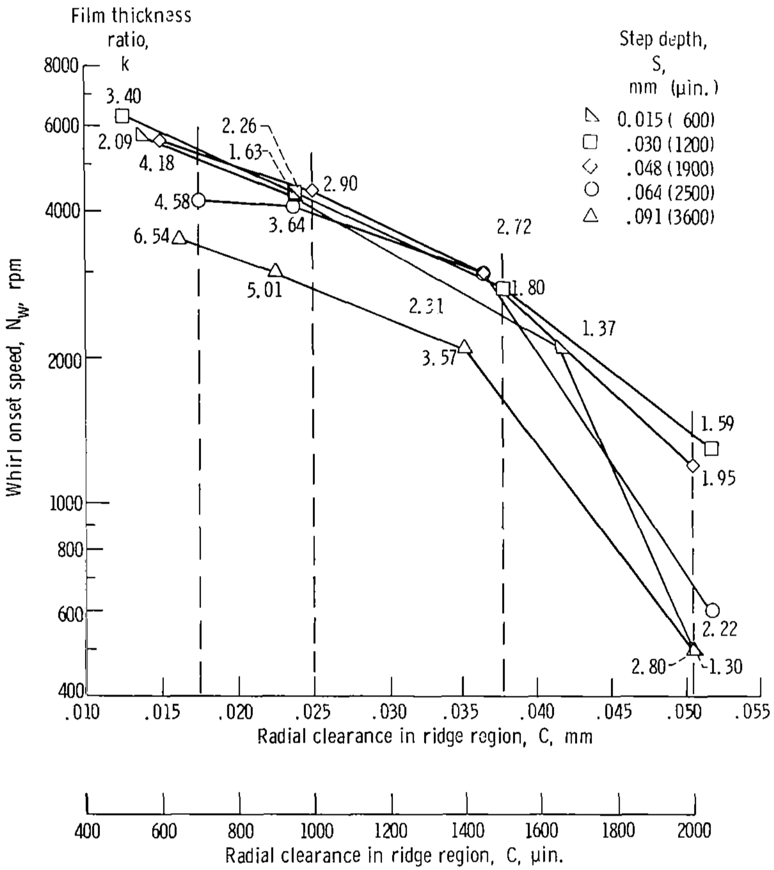


Figure 13. - Whirl onset speed as a function of radial clearance for one-segment, three-pad shrouded Rayleigh step bearing at various film thickness ratios. Ridge to pad arc ratio $\psi = 0.27$.

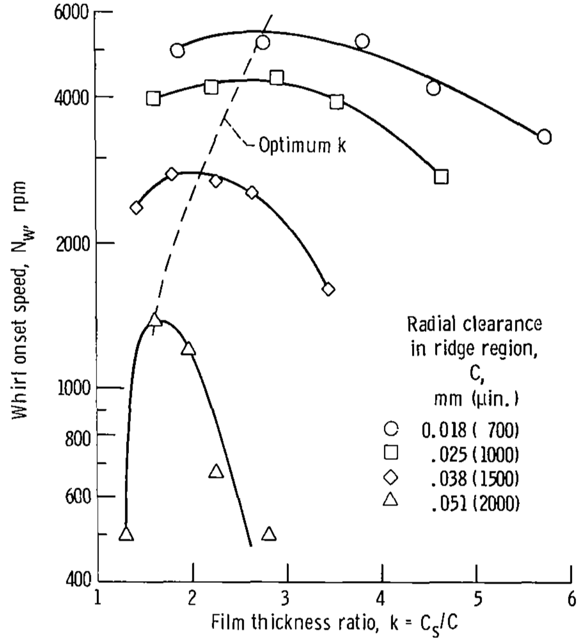


Figure 14. - Whirl onset speed as a function of film thickness ratio for one-segment, three-pad shrouded Rayleigh step bearing. Ridge to pad arc ratio $\psi = 0.27$.

k as variables. Simple straight line interpolation was used for the cross plot. In figure 14 there is an optimum value of k at any given C and this optimum is a function of C. Figure 14 shows that stability becomes more sensitive to k as C increases. It also shows that the optimum film thickness ratio k decreases from 2.8 to 1.6 as the clearance increases from 0.018 to 0.051 millimeter (700 to 2000 μin.). For eccentricity ratios less than 0.2, an optimal film thickness ratio of 1.7 was reported in reference 12, when using the criterion of load capacity. The optimal k value is therefore considerably higher (1.6 to 2.8) when using the criterion of stability as opposed to load capacity.

Figure 15 is essentially a repetition of figure 14 except that the ordinate is the dimensionless speed Γ rather than the dimensional speed N_w. The C/R ratios as well as C have also been listed in an attempt to make this figure of wider applicability in the

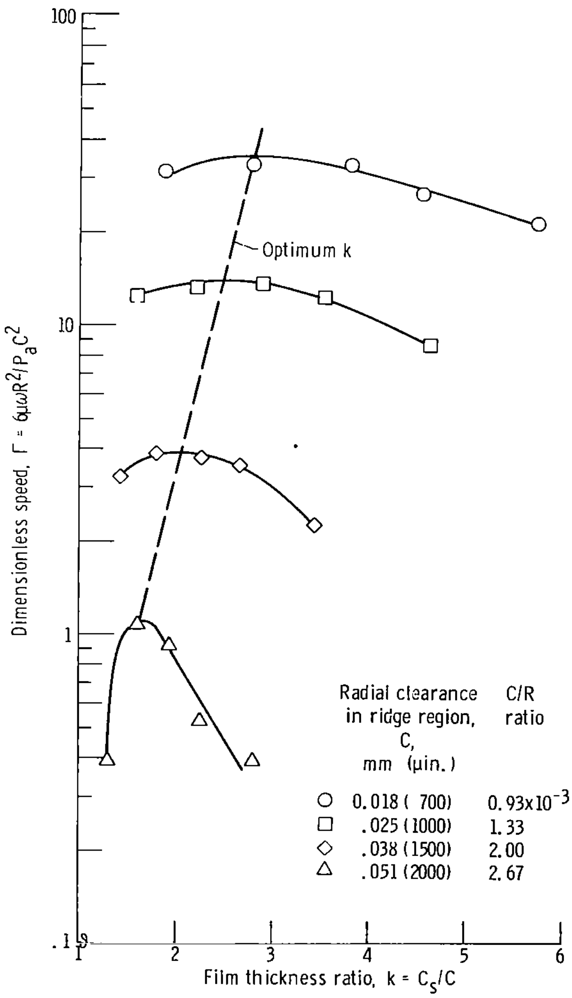


Figure 15. - Dimensionless speed as a function of film thickness ratio for one-segment, three-pad shrouded Rayleigh step bearing. Ridge to pad arc ratio ψ = 0.27.

design of a Rayleigh step bearing of optimum stability.

Stability Comparison of Rayleigh Step Bearings With Three Other Fixed Geometry Types

Whirl speed is plotted against radial clearance in figure 16 for five different fixed geometry journal bearings. The data points for the one-segment, three-pad shrouded Rayleigh step bearings correspond to the points of maximum stability (optimum locus curves) obtained from figure 14. The points for the other four bearing types were obtained in a similar manner from this work and from the data in references 6, 8, and 9. Figure 16 shows that the five fixed geometry bearings considered can be generally rated in order of diminishing stability as follows: (1) three-lobe bearings with the minimum film thickness at the trailing end of each lobe (offset factor of 1.0), (2) herringbone-groove bearings, (3) one-segment, three-pad shrouded Rayleigh step bearings, (4) three-

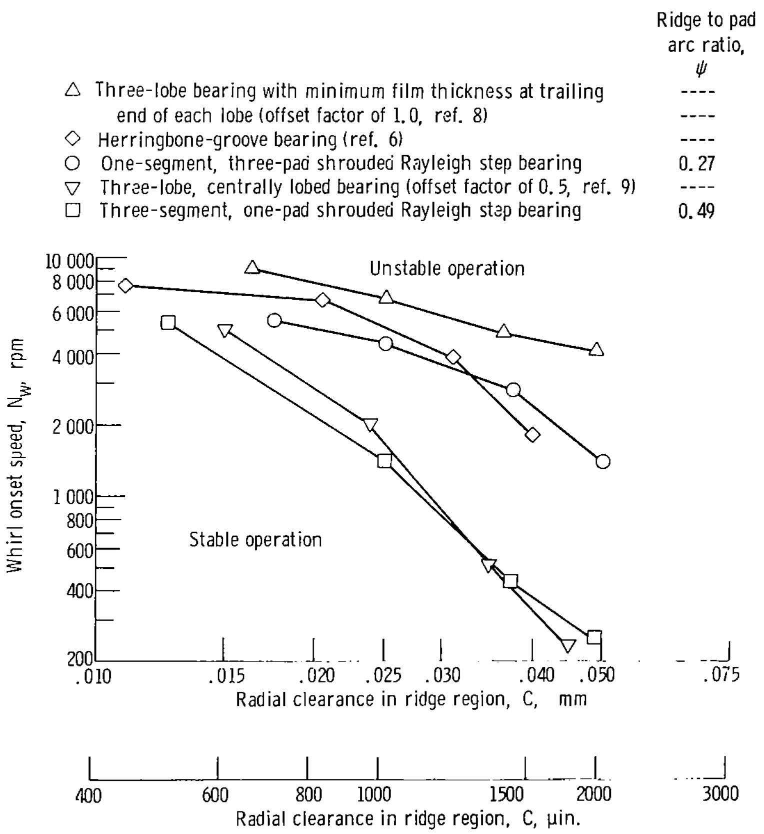


Figure 16. - Stability of five fixed geometry bearings.

lobe, centrally lobed bearings (offset factor of 0.5), and (5) three-segment, one-pad shrouded Rayleigh step bearings. The last perhaps should not be rated with the other four since it comprises three separate bearings with an L/D ratio of one-third whereas the others have an L/D ratio of 1. However, the three segments taken as a unit configuration have an L/D of 1 and result in a bearing of least stability when compared with the other four. It is interesting to note that the commonly used centrally lobed bearing has relatively poor stability, close to that of the poorest, the three-segment, one-pad shrouded Rayleigh step bearing.

The one-segment, three-pad shrouded Rayleigh step bearing is less stable than the herringbone-groove bearing (diamond symbols) at clearances up to 0.033 millimeter (1300 μin.). However, at clearances greater than 0.033 millimeter (1300 μin.) the Rayleigh step bearing has the advantage in stability. This type of cross-over pattern in the stability curves of various bearing types makes it difficult to rate one above the other. The ratings almost certainly require a qualifying range of clearance to make them worthwhile.

SUMMARY OF RESULTS

Eighty stability tests were conducted on Rayleigh step journal bearings of four different configurations: (1) three-segment, one-pad shrouded, (2) one-segment, three-pad shrouded, (3) one-segment, one-pad shrouded, and (4) one-segment, three-pad unshrouded. Each configuration was run at four different clearances at various step depths. The clearances ranged from 0.010 to 0.052 millimeter (400 to 2050 μin.), and the step depths ranged from 0.015 to 0.094 millimeter (600 to 3700 μin.). The test bearings had a nominal diameter and length of 3.8 centimeters (1.5 in.) and were run hydrodynamically in water at an average temperature of 300 K (80° F) at speeds to 6300 rpm at zero load. The following results were obtained:

1. The Rayleigh step bearing configurations tested ranked in the following order of diminishing stability for clearances above 0.015 millimeter (600 μin.): one-segment, three-pad shrouded; one-segment, one-pad shrouded; one-segment, three-pad unshrouded; and three-segment, one-pad shrouded.

2. At clearance values below 0.015 millimeter (600 μin.), the stability of all four Rayleigh step bearing configurations approach equal magnitude presumably because of an effective preload on the bearings caused by slight angular misalinement of the bearing and journal axes.

3. The stability of a three-segment, one-pad shrouded and a one-segment, three-pad shrouded bearing was not appreciably affected by a change in ridge to pad arc ratio from 0.40 to 0.49 and 0.27 to 0.45, respectively.

4. A substantial increase in stability can be gained with a shrouded compared to an unshrouded Rayleigh step bearing.

5. For a one-segment, three-pad shrouded bearing there is an optimum value of film thickness ratio at any given clearance and this optimum is a function of clearance. Stability becomes more sensitive to film thickness ratio as clearance increases.

6. The optimum film thickness ratio for a stepped journal bearing is higher when stability rather than load capacity is used as the criterion.

7. From the data reported herein and from previously reported data, five fixed geometry bearings can be generally rated in order of diminishing stability as follows: (1) three-lobe bearing with the minimum film thickness at the trailing end of each lobe (offset factor of 1.0), (2) herringbone-groove bearing, (3) one-segment, three-pad shrouded Rayleigh step bearing, (4) three-lobe, centrally lobed bearing (offset factor of 0.5), and (5) three-segment, one-pad shrouded Rayleigh step bearing.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 1, 1971,
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TABLE I. - TEST RESULTS FOR THREE-SEGMENT, ONE-PAD SHROUDED

RAYLEIGH STEP JOURNAL BEARINGS

(a) Ridge to pad arc ratio $\psi = 0.40$

Bearing	Depth of step, S		Radial clearance at ridge, C		Radial clearance at step, $C_s = C + S$		Film thickness ratio, $k = C_s/C$	Fractional-frequency-whirl onset speed at zero load, N_w, rpm
	mm	μin.	mm	μin.	mm	μin.		
3RS4	0.023	900	0.013	500	0.036	1400	2.80	5700
			0.025	1000	0.048	1900	1.90	1230
			0.038	1500	0.061	2400	1.60	420
			0.051	2000	0.074	2900	1.45	200
3RS5	0.041	1600	0.013	500	0.053	2100	4.20	5200
			0.025	1000	0.066	2600	2.60	920
			0.038	1500	0.079	3100	2.07	360
			0.052	2050	0.093	3650	1.73	210
3RS6	0.053	2100	0.013	500	0.066	2600	5.20	5300
			0.025	1000	0.079	3100	3.10	1320
			0.038	1500	0.091	3600	2.40	450
			0.051	2000	0.104	4100	2.05	280
3RS4R	0.094	3700	0.013	500	0.107	4200	8.41	3800
			0.024	950	0.118	4650	4.90	1100
			0.037	1450	0.131	5150	3.55	300
			0.050	1950	0.144	5650	2.90	100

TABLE I. - Concluded. TEST RESULTS FOR THREE-SEGMENT, ONE-PAD SHROUDED RAYLEIGH STEP JOURNAL BEARINGS

(b) Ridge to pad arc ratio $\psi = 0.49$

Bearing	Depth of step, S		Radial clearance at ridge, C		Radial clearance at step, $C_s = C + S$		Film thickness ratio, $k = C_s/C$	Fractional-frequency-whirl onset speed at zero load, N_w, rpm
	mm	μin.	mm	μin.	mm	μin.		
3RS1	0.029	1100	0.010	400	0.038	1500	3.75	5840
			0.024	950	0.052	2050	2.16	1200
			0.037	1450	0.065	2550	1.76	370
			0.051	2000	0.079	3100	1.55	190
3RS2	0.036	1400	0.011	450	0.047	1850	4.12	5600
			0.025	1000	0.061	2400	2.40	1200
			0.038	1500	0.074	2900	1.93	390
			0.052	2050	0.088	3450	1.68	230
3RS3	0.053	2100	0.013	500	0.066	2600	5.20	5300
			0.025	1000	0.079	3100	3.10	1410
			0.038	1500	0.091	3600	2.40	440
			0.052	2050	0.105	4150	2.03	200
3RS1R	0.089	3500	0.013	500	0.102	4000	8.00	4100
			0.024	950	0.113	4450	4.69	1000
			0.037	1450	0.126	4950	3.41	150
			0.050	1950	0.138	5450	2.80	100

TABLE II. - TEST RESULTS FOR ONE-SEGMENT, THREE-PAD SHROUDED

RAYLEIGH STEP JOURNAL BEARINGS

(a) Ridge to pad arc ratio $\psi = 0.27$

Bearing	Depth of step, S		Radial clearance at ridge, C		Radial clearance at step, $C_s = C + S$		Film thickness ratio, $k = C_s/C$	Fractional-frequency-whirl onset speed at zero load, N_w, rpm
	mm	μin.	mm	μin.	mm	μin.		
RS3A	0.015	600	0.014	550	0.029	1150	2.09	5700
			0.024	950	0.039	1550	1.63	4300
			0.042	1650	0.057	2250	1.37	2100
			0.051	2000	0.066	2600	1.30	500
RS1	0.030	1200	0.013	500	0.043	1700	3.40	6300
			0.024	950	0.055	2150	2.26	4400
			0.038	1500	0.069	2700	1.80	2800
			0.052	2050	0.083	3250	1.59	1300
RS3	0.048	1900	0.015	600	0.064	2500	4.18	5600
			0.025	1000	0.074	2900	2.90	4400
			0.037	1450	0.085	3350	2.31	3000
			0.051	2000	0.099	3900	1.95	1200
RS1A	0.064	2500	0.018	700	0.081	3200	4.58	4200
			0.024	950	0.088	3450	3.64	4100
			0.037	1450	0.100	3950	2.72	3000
			0.052	2050	0.116	4550	2.22	600
RS2A	0.091	3600	0.017	650	0.108	4250	6.54	3500
			0.023	900	0.114	4500	5.01	3000
			0.036	1400	0.127	5000	3.57	2100
			0.051	2000	0.142	5600	2.80	500

TABLE II. - Concluded. TEST RESULTS FOR ONE-SEGMENT, THREE-PAD SHROUDED RAYLEIGH STEP JOURNAL BEARINGS

(b) Ridge to pad arc ratio ψ = 0.45

Bearing	Depth of step, S		Radial clearance at ridge, C		Radial clearance at step, $C_s = C + S$		Film thickness ratio, $k = C_s/C$	Fractional-frequency-whirl onset speed at zero load, N_w, rpm
	mm	μin.	mm	μin.	mm	μin.		
RS4A	0.020	800	0.011	450	0.032	1250	2.78	6000
			0.022	850	0.042	1650	1.95	4300
			0.039	1550	0.060	2350	1.52	2800
			0.050	1950	0.069	2700	1.42	600
RS6	0.025	1000	0.011	450	0.037	1450	3.23	5600
			0.024	950	0.050	1950	2.05	4100
			0.036	1400	0.061	2400	1.71	3000
			0.050	1950	0.075	2950	1.51	1020
RS4	0.048	1900	0.015	600	0.064	2500	4.18	5700
			0.025	1000	0.074	2900	2.90	4200
			0.037	1450	0.085	3350	2.31	3200
			0.051	2000	0.099	3900	1.95	1200
RS5A	0.066	2600	0.017	650	0.083	3250	5.00	5000
			0.023	900	0.089	3500	3.89	3700
			0.036	1400	0.102	4000	2.86	2300
			0.051	2000	0.117	4600	2.30	800
RS6A	0.089	3500	0.015	600	0.104	4100	6.84	4100
			0.022	850	0.110	4350	5.12	3300
			0.034	1350	0.123	4850	3.59	1500
			0.050	1950	0.138	5450	2.79	600

TABLE III. - TEST RESULTS FOR ONE-SEGMENT, ONE-PAD SHROUDED

RAYLEIGH STEP JOURNAL BEARING

[Ridge to pad arc ratio $\psi = 0.40$.]

Bearing	Depth of step, S		Radial clearance at ridge, C		Radial clearance at step, $C_s = C + S$		Film thickness ratio, $k = C_s/C$	Fractional-frequency-whirl onset speed at zero load, N_w, rpm
	mm	μin.	mm	μin.	mm	μin.		
50	0.066	2600	0.015	600	0.081	3200	5.33	3900
			0.027	1050	0.093	3650	3.47	3000
			0.043	1700	0.109	4300	2.53	720
			0.052	2050	0.118	4650	2.27	700

TABLE IV. - TEST RESULTS FOR ONE-SEGMENT, THREE-PAD

UNSHROUDED RAYLEIGH STEP JOURNAL BEARING

[Ridge to pad arc ratio $\psi = 0.45$.]

Bearing	Depth of step, S		Radial clearance at ridge, C		Radial clearance at step, $C_s = C + S$		Film thickness ratio, $k = C_s/C$	Fractional-frequency-whirl onset speed at zero load, N_w, rpm
	mm	μin.	mm	μin.	mm	μin.		
RS5AM	0.066	2600	0.013	500	0.079	3100	6.20	4300
			0.024	950	0.090	3550	3.74	1400
			0.037	1450	0.103	4050	2.79	1000
			0.050	1950	0.116	4550	2.33	650

NASA-Langley, 1971 —— 15 E-6277