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OPTIMIZATION OF SELF-ACTING HERRINGBONE JOURNAL BEARINGS FOR MAXIMUM RADIAL LOAD CAPACITY

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ABSTRACT

A computer program was developed to determine optimal self-acting herringbone journal bearings for maximum radial load capacity. Design curves enable one to find the optimal heeringbone journal bearing for a wide range of operating conditions. These range from incompressible lubrication to a highly compressible condition, for either smooth or groove members rotating, and for length to diameter ratios of 1/4, 1/2, 1, and 2. The analysis is valid for small displacements of the journal center from the fixed bearing center, and for a large number of grooves.

SUMMARY

A computer program was developed to determine the optimal herringbone journal for maximum radial load capacity. The design curves shown in this report enable one to find the optimal herringbone journal bearing for a wide range of operating conditions. These include:

1. Incompressibly lubricated to highly compressible condition
2. Smooth or grooved member rotating
3. Length to diameter ratios of 1/4, 1/2, 1, and 2.

The analysis is valid for small displacements of the journal center from the fixed bearing center, and for a large number of grooves.

Some of the findings of the work presented in this report are:

1. For length to diameter ratios of one and two, and small dimensionless bearing numbers a plain journal bearing has a greater radial load capacity than any herringbone configuration. However, for the limiting case of incompressible lubrication the plain journal is not optimal but the herringbone journal is.

2. For $\Lambda \rightarrow 0$, the incompressible case, the optimal configuration is the same whether the smooth or grooved member is rotating. However, as the bearing number increases the optimal configuration differs appreciably depending on whether the smooth or groove member is rotating.

3. At high bearing numbers the radial load capacity is appreciably higher for the case when the smooth member is rotating.

INTRODUCTION

More than any other factor, self-excited whirl instability and low load capacity limit the usefulness of gas lubricated self-acting journal bearings. The whirl problem is the tendency of the journal center to orbit the bearing center at an angular speed less than or equal to half that of the journal about its own center. In many cases the whirl amplitude is large enough to cause destructive contact with the bearing surfaces.

The low load capacity of self-acting gas lubricated journal bearings is also a serious concern in many applications. Unlike liquid lubricants, a gaseous lubricant changes its density as it passes through the bearing. This so-called compressibility effect results in a "terminal" load condition. That is, the load capacity does not increase indefinitely with speed, but quickly approaches a fixed value.

In quest of a bearing which would overcome the two problems of self-excited whirl instability and low load capacity, Vohr and Chow (1) theoretically investigated a herringbone grooved journal bearing. They obtained a solution for bearing load capacity valid for small displacements of the journal center from the bearing center. An additional assumption was that the number of grooves was large enough that local pressure variations across a groove-ridge pair would be ignored. One of the conclusions obtained from the Vohr and Chow analysis is that in contrast to a plain bearing the load capacity of a herringbone grooved journal bearing continues to increase with increase in speed. Furthermore the herringbone grooved journal bearing may not suffer from the self-excited whirl instability that is normally associated with unloaded plain bearings. Malanowski (2) and Cunningham et al. (3) and (4) experimentally verified the above conclusion of Vohr and Chow.

Therefore, it has been shown that the self-acting herringbone journal bearing has highly desirable characteristics, namely that of high load capacity and that of operating in a whirl free condition. A remaining problem which is not in the literature is that of obtaining optimal herringbone journal bearing configurations for a wide range of bearing operating conditions. Therefore, the objective of the present report is to develop an optimization program, utilizing the analysis of Vohr and Chow (1) to determine groove configurations to maximize radial load capacity. Results are to be applicable for operating conditions ranging from an incompressible solution to a highly compressible solution, and for bearing length-diameter ratios of 1/4 to 2.

SYMBOLS

b_1 = width of groove

b_2 = width of ridge

C_2, C_4, C_7 = dimensionless coefficients in differential equation of Vohr and Chow analysis (1)

D = diameter of journal

e = eccentricity of journal

f_r = radial load capacity of herringbone journal bearing

$\bar{f}_r$ = radial load capacity of plain journal bearing

$F_r = \dfrac{f_r}{\epsilon\, p_a LD}$ = dimensionless radial load capacity of herringbone journal bearing

$\bar{F}_r = \dfrac{\bar{f}_r}{\epsilon\, p_a LD}$ = dimensionless radial load capacity of plain journal bearing

h_{10} = film thickness in groove region when journal is concentric

h_{20} = film thickness in ridge region when journal is concentric

$H_o = \frac{h_{10}}{h_{20}}$ = film thickness ratio

L = length of journal

L_1 = total axial length of groove

N = number of grooves

p_a = ambient pressure

R = radius of journal

U = velocity

w = total load capacity of herringbone journal bearing

w_D = total load capacity of plain incompressibly lubricated journal bearing obtained from Donaldson (6)

w_R = total load capacity of plain compressibly lubricated journal bearing obtained from Raimondi (7)

$W = \frac{w}{\epsilon p_a LD}$ = dimensionless load capacity of herringbone journal bearing

$W_D = \frac{w_D}{\epsilon p_a LD}$ = dimensionless load capacity of plain incompressibly lubricated journal bearing

$W_R = \frac{W_R}{\epsilon p_a LD}$ = dimensionless load capacity of plain compressibly lubricated journal bearing

$\alpha = \frac{b_1}{b_1 + b_2}$ = groove width ratio

β = groove angle

$\gamma = \frac{L_1}{L}$ = groove length ratio

$\epsilon = \frac{e}{h_{20}}$ = eccentricity ratio

θ = angular coordinate

$\lambda = \frac{L}{D}$ = length to diameter ratio

$\Lambda = \frac{6\mu UR}{p_a h_{20}^2}$ = dimensionless bearing number

μ = viscosity of fluid

σ = -1 grooves member rotating

σ = 1 smooth member rotating

BEARING DESCRIPTION

Figure 1 shows the bearing to be studied. Note that the bearing has angled, shallow grooves in the journal surface. The grooves can be partial as shown or extend the complete length of the bearing. Also, the grooves can occur in the rotating or non-rotating surfaces. The purpose of these grooves is to pump fluid toward the center of the bearing thereby increasing the lubricant pressure in the bearing. Load is directly related to the pressure distribution. This self pressurization can increase the load capacity over that of a smooth bearing. The bearing shown in Figure 1 is uni-directional (e.g., it pumps inwardly for only one direction of rotation).

From Figure 1 the groove region is where the film thickness is h_{10} and the ridge region is where the film thickness is h_{20}. Also, the groove width is defined as b_1 and the ridge width is defined as b_2. The analysis of reference (1) indicates that the groove parameters to be optimized are:

1. <u>The film thickness ratio</u> (H_o) which is equal to the film thickness in the groove region divided by the film thickness in the ridge region when the bearing is concentric ($H_o = h_{10}/h_{20}$)

2. The groove width ratio (α) which is equal to the width of the groove region divided by the width of the groove-ridge pair $\left(\alpha = b_1/(b_1+b_2)\right)$

3. The groove angle (β)

4. The groove length ratio (γ) which is equal to the length covered by grooves divided by the overall length of the bearing ($\gamma = L_1/L$)

In Figure 1 the number of grooves is six. However, the Vohr and Chow analysis (1) assumes essentially an infinite number of grooves. Reference (5) developes the following criterion for minimum number of grooves such that the infinite groove analysis yields valid results

$$\frac{\Lambda}{N} < \frac{\left[(1-\alpha)H_o^3+\alpha\right]\left\{H_o^3+\alpha(1-\alpha)(H_o^3-1)^2\sin^2\beta\right\}}{2\pi\left\{H_o^3+\alpha(1-\alpha)(H_o^3-1)^2\right\}\alpha(H_o-1)(1-\alpha)\sin^2\beta} \qquad [1]$$

where N = number of grooves

$\Lambda = \frac{6\mu UR}{p_a h_{20}^2}$ = dimensionless bearing number

The numerical value of the right side of [1] is typically between 5.5 and 8.0. Therefore, the minimum number of grooves placed around the journal can be represented conservatively by

$$N \geq \frac{\Lambda}{5} \qquad [2]$$

VERIFICATION OF EQUATIONS

A digital computer program was written to solve the equations presented in reference (1). The equations for the dimensionless load capacity for incompressible lubrication are identical to the compressible equations with the exception that the dimensionless coefficients C_2, C_4, and C_7 in reference (1) are equal to zero. The equations for the dimensionless load capacity where found to be a function of the groove parameters H_o, α, β and γ discussed in the Bearing Description section as well as the following bearing operating parameters:

1. $\Lambda = \frac{6\mu UR}{p_a h_{20}^2}$ = dimensionless ratio

2. $\lambda = \frac{L}{D}$ = length to diameter ratio

3. $\sigma = -1$ smooth member rotating

   $\sigma = 1$ grooved member rotating

It should be recalled that perturbation on the eccentricity was assumed, thereby restricting the results to small eccentricity ratios.

Table I shows that the equations developed are indeed valid. The table compares the dimensionless load capacity of a plain journal with that of a herringbone journal bearing when it is made to approach a plain journal bearing. That is the table shows the following:

$$\text{Herringbone Journal when} \left\{ \begin{array}{l} H_o \rightarrow 1 \\ \alpha \rightarrow 0 \\ \beta \rightarrow 0 \\ \gamma \rightarrow 0 \end{array} \right\} \longrightarrow \text{Plain Journal Bearing}$$

Table I uses plain journal bearing results from Donaldson (6) for the incompressible results and Raimondi (7) for the compressible results. These results are for an eccentricity ratio of 0.1. From Table I it is seen that the herringbone bearing loads when $H_o \to 1$ and $\alpha \to \beta \to \gamma \to 0$ are within 3 perdent the plain journal bearing results.

Table I also shows that for incompressible lubrication the dimensionless load increases 6.8 times as the length to diameter ratio changes from 0.5 to 2.0. For the compressible case of $\Lambda = 6$ the dimensionless load increases only 2.1 times over this range. This implies that side leakage is much greater in incompressible lubrication.

OPTIMIZING PROCEDURE

The problem as defined in the Introduction is to find the optimal herringbone journal bearing for maximum radial load capacity for various bearing parameters. (The radial load capacity is the component in the direction of journal displacement of the total load capacity.) Therefore, the basic problem is to optimize the film thickness ratio H_o, the groove width ratio α , the groove angle β , and the groove length ratio γ for maximum radial load F_r given in dimensionless bearing number Λ , length to diameter ratio λ , and whether the grooved or smooth member is rotating. Mathematically this is expressed as:

GIVEN: Λ , λ , and σ

FIND : H_o, α , β , and γ which satisfy the following:

$$\frac{\partial F_r}{\partial H_o} = \frac{\partial F_r}{\partial \alpha} = \frac{\partial F_r}{\partial \beta} = \frac{\partial F_r}{\partial \gamma} = 0$$

The method used is the Newton-Raphson method of solving simultaneous equations. This method is described in Scarborough (8); it was previously used in optimizing the step thrust bearing (9).

DISCUSSION OF RESULTS

Tables II and III give optimal herringbone parameters (H_o, α , β , γ) for maximum radial load capacity (F_r). The difference between these tables is that Table II the grooved member is rotating (σ = -1) and in Table III the smooth member is rotating (σ = 1). The tables cover a wide range of bearing parameters; from incompressible lubrication ($\Lambda \rightarrow 0$) to a highly compressible situation (Λ = 160) as well as for length to diameter ratios of 1/4, 1/2, 1, and 2. The tables also contain the radial load capacity of a plain journal bearing $\bar{F}_r$.

The following general observations can be made from Tables II and III:

1. For length to diameter ratios (λ) of two and dimensionless bearing number (Λ) of one the optimal configuration is a plain journal bearing.

2. At high values of dimensionless bearing numbers (Λ) the radial load capacity of an optimal herringbone journal bearing is considerably higher than that of a plain journal bearing.

3. For incompressible lubrication the results are exactly the same whether the smooth or grooved member is rotating.

4. As the dimensionless bearing number (Λ) increases the optimal configuration differs appreciably depending on whether the smooth or groove member is rotating.

5. For $\lambda = 1/4$ there is little change in optimal configuration over the complete range of bearing number. However, for $\lambda = 2$ there is considerable change in the optimal configuration.

Figures 2 through 6 are directly obtained from data presented in Tables II and III. In the "a" part of each figure the grooved member is assumed to be rotating ($\sigma = -1$); in the "b" part of each figure the smooth member is assumed to be rotating ($\sigma = 1$). Figures 2 through 5 show the effect of Λ on optimal configuration parameters for $\lambda = 1/4$, 1/2, 1, and 2. In all these figures it is observed that for $\lambda = 1$ and $\lambda = 2$ and small dimensionless bearing number the results change rapidly and tend toward a plain bearing as optimal. However, it should be pointed out that for the limiting case of $\Lambda \rightarrow 0$, incompressible lubrication, the plain bearing is not optimal but the herringbone configuration is as shown in Tables II and III. For some cases in the low dimensionless bearing number range ($\Lambda < 24$), additional data other than tha shown in Tables II and III had to be obtained to produce the curves of Figures 2 through 6.

Figure 6 shows the effect of Λ on dimensionless radial load capacity of an optimal herringbone journal bearing for length to diameter ratios of 1/4, 1/2, 1, and 2. It is observed that the dimensionless radial load capacity increases with Λ and does not approach any fixed value as is the case for a plain journal bearing. It is seen from Figures 6(a) and 6(b) that the dimensionless radial load capacity is appreciably higher for the case when the smooth member is rotating ($\sigma = 1$).

CONCLUDING REMARKS

A computer program was developed to determine optimal self-acting herringbone journal bearings for maximum radial load capacity. The design curves shown in the report enable one to find the optimal herringbone journal bearing for a wide range of operating conditions. These range from incompressible lubrication to a highly compressible condition, for either smooth or groove members rotating, and for length to diameter ratios of 1/4, 1/2, 1, and 2. The analysis is valid for small displacement of the journal center from the bearing center, and for a large number of grooves.

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TABLE I. - COMPARISON OF THE DIMENSIONLESS LOAD CAPACITY OF A PLAIN JOURNAL BEARING WITH A HERRINGBONE JOURNAL AS IT APPROACHES A PLAIN JOURNAL BEARING

λ \ Λ	Incompressible lubrication ($\Lambda \to 0$)	$\Lambda = 0.24$	$\Lambda = 0.6$	$\Lambda = 1.2$	$\Lambda = 3.0$	$\Lambda = 6.0$	$\Lambda = 12.0$	$\Lambda = 24.0$
$\lambda = 0.5$	$W/\Lambda = 0.1190$ $W_D/\Lambda = .1205$	$W = 0.0286$ $W_R = .0288$	$W = 0.0713$ $W_R = .0719$	$W = 0.142$ $W_R = .143$	$W = 0.344$ $W_R = .348$	$W = 0.625$ $W_R = .634$	$W = 0.961$ $W_R = .974$	$W = 1.19$ $W_R = 1.22$
$\lambda = 1$	$W/\Lambda = 0.3745$ $W_D/\Lambda = .3776$	$W = 0.0897$ $W_R = .0902$	$W = 0.221$ $W_R = .223$	$W = 0.425$ $W_R = .429$	$W = 0.851$ $W_R = .862$	$W = 1.13$ $W_R = 1.16$	$W = 1.28$ $W_R = 1.31$	
$\lambda = 2$	$W/\Lambda = 0.8137$ $W_D/\Lambda = .8170$	$W = 0.193$ $W_R = .194$	$W = 0.458$ $W_R = .462$	$W = 0.786$ $W_R = .794$	$W = 1.18$ $W_R = 1.20$	$W = 1.33$ $W_R = 1.36$		

TABLE II. - OPTIMAL HERRINGBONE GROOVE PARAMETERS FOR MAXIMUM DIMENSIONLESS RADIAL LOAD CAPACITY FOR VARIOUS DIMENSIONLESS BEARING NUMBERS AND LENGTH TO DIAMETER RATIOS WHEN

GROOVED MEMBER IS ROTATING

λ \ Λ	Incompressible solution (Λ → 0)	Λ = 0.1	Λ = 1.0	Λ = 10.0	Λ = 20.0	Λ = 40.0	Λ = 80.0	Λ = 160.0
$\lambda = \frac{1}{4}$	H_o = 2.592	H_o = 2.592	H_o = 2.583	H_o = 2.548	H_o = 2.507	H_o = 2.471	H_o = 2.489	H_o = 2.552
	α = .5006	α = .5002	α = .4973	α = .4773	α = .4703	α = .4823	α = .5234	α = .5604
	γ = .9410	γ = .9410	γ = .9408	γ = .9201	γ = .8779	γ = .7961	γ = .7015	γ = .6286
	β = 19.26°	β = 19.26°	β = 19.25°	β = 19.40°	β = 19.88°	β = 20.91°	β = 22.31°	β = 23.69°
	F_r/Λ = .0380	F_r = .0038	F_r = .0381	F_r = .3864	F_r = .7579	F_r = 1.396	F_r = 2.337	F_r = 3.625
	$\bar{F}_r/\Lambda$ = .0	$\bar{F}_r$ = .0000	$\bar{F}_r$ = .0008	$\bar{F}_r$ = .0734	$\bar{F}_r$ = .2503	$\bar{F}_r$ = .631	$\bar{F}_r$ = 1.021	$\bar{F}_r$ = 1.222
$\lambda = \frac{1}{2}$	H_o = 2.381	H_o = 2.379	H_o = 2.355	H_o = 2.241	H_o = 2.317	H_o = 2.454	H_o = 2.567	H_o = 2.617
	α = .5044	α = .5021	α = .4820	α = .3577	α = .3799	α = .4416	α = .4552	α = .4253
	γ = .8653	γ = .8653	γ = .8636	γ = .7463	γ = .6247	γ = .5483	γ = .5146	γ = .5327
	β = 23.26°	β = 23.25°	β = 23.21°	β = 22.82°	β = 22.68°	β = 23.31°	β = 23.88°	β = 23.83°
	F_r/Λ = .0669	F_r = .0067	F_r = .0689	F_r = .7362	F_r = 1.295	F_r = 1.985	F_r = 2.913	F_r = 4.482
	$\bar{F}_r/\Lambda$ = .0	$\bar{F}_r$ = .0001	$\bar{F}_r$ = .0107	$\bar{F}_r$ = .5855	$\bar{F}_r$ = 0.990	$\bar{F}_r$ = 1.211	$\bar{F}_r$ = 1.318	$\bar{F}_r$ = 1.393
$\lambda = 1$	H_o = 2.219	H_o = 2.203	H_o = 2.051	H_o = 2.294	H_o = 2.486	H_o = 2.587	H_o = 2.616	H_o = 2.630
	α = .5228	α = .5089	α = .3529	α = .2951	α = .3816	α = .3913	α = .3725	α = .3632
	γ = .7607	γ = .7567	γ = .6857	γ = .4213	γ = .4397	γ = .4785	γ = .5503	γ = .6032
	β = 28.62°	β = 28.59°	β = 27.94°	β = 23.38°	β = 23.77°	β = 23.71°	β = 23.18°	β = 22.62°
	F_r/Λ = .1007	F_r = .0103	F_r = .1234	F_r = 1.334	F_r = 1.876	F_r = 2.691	F_r = 4.229	F_r = 7.367
	$\bar{F}_r/\Lambda$ = .0	$\bar{F}_r$ = .0011	$\bar{F}_r$ = .0980	$\bar{F}_r$ = 1.160	$\bar{F}_r$ = 1.302	$\bar{F}_r$ = 1.388	$\bar{F}_r$ = 1.444	$\bar{F}_r$ = 1.482
$\lambda = 2$	H_o = 2.147	H_o = 2.085	H_o = 1.0	H_o = 2.404	H_o = 2.503	H_o = 2.566	H_o = 2.624	H_o = 2.661
	α = .5671	α = .5071	α = 0.0	α = .3531	α = .3571	α = .3548	α = .3626	α = .3704
	γ = .6796	γ = .6241	γ = .0	γ = .3954	γ = .4974	γ = .5790	γ = .6108	γ = .6213
	β = 35.37°	β = 34.63°	β = .0	β = 25.81°	β = 24.60°	β = 23.46°	β = 22.73°	β = 22.33°
	F_r/Λ = .1138	F_r = .0126	F_r = .3557	F_r = 1.792	F_r = 2.580	F_r = 4.172	F_r = 7.417	F_r = 13.95
	$\bar{F}_r/\Lambda$ = 0.0	$\bar{F}_r$ = .0049	$\bar{F}_r$ = .3557	$\bar{F}_r$ = 1.357	$\bar{F}_r$ = 1.434	$\bar{F}_r$ = 1.479	$\bar{F}_r$ = 1.507	$\bar{F}_r$ = 1.53

TABLE III. - OPTIMAL HERRINGBONE GROOVE PARAMETERS FOR MAXIMUM DIMENSIONLESS RADIAL LOAD CAPACITY FOR VARIOUS DIMENSIONLESS BEARING NUMBERS AND LENGTH TO DIAMETER RATIOS WHEN

SMOOTH MEMBER IS ROTATING

λ \ Λ		Incompressible solution ($\Lambda \to 0$)	Λ = 0.1	Λ = 1.0	Λ = 10.0	Λ = 20.0	Λ = 40.0	Λ = 80.0	Λ = 160.0
$\lambda = \frac{1}{4}$	H_o =	2.592	2.592	2.589	2.561	2.539	2.537	2.605	2.747
	α =	.5006	.5003	.4979	.4804	.4739	.4859	.5246	.5640
	γ =	.9416	.9411	.9420	.9443	.9360	.9096	.8766	.8591
	β =	19.26°	19.26°	19.24°	19.26°	19.65°	20.79°	22.42°	23.73°
	F_r =	(F_r/Λ =) .0380	.0038	.0381	.3889	.7777	1.4959	2.693	4.621
	$\overline{F}_r$ =	($\overline{F}_r/\Lambda$ =) 0.0	.0000	.0008	.0734	.2503	.6309	1.021	1.222
$\lambda = \frac{1}{2}$	H_o =	2.381	2.378	2.350	2.205	2.288	2.485	2.721	2.950
	α =	.5044	.5025	.4854	.3810	.4236	.4967	.5327	.5326
	γ =	.8653	.8655	.8674	.8480	.8166	.8146	.8328	.8575
	β =	23.26°	23.25°	23.23°	24.11°	25.47°	26.64°	26.75°	25.60°
	F_r =	(F_r/Λ =) .0669	.0067	.0686	.7544	1.390	2.319	3.792	6.438
	$\overline{F}_r$ =	($\overline{F}_r/\Lambda$ =) 0.0	.0001	.0107	.5855	.9900	1.211	1.318	1.393
$\lambda = 1$	H_o =	2.219	2.201	2.022	2.113	2.338	2.557	2.723	2.828
	α =	.5228	.5112	.3570	.3013	.4104	.4521	.4643	.4695
	γ =	.7607	.7564	.6773	.5754	.6897	.7678	.8113	.8330
	β =	28.62°	28.58°	27.81°	28.70°	30.27°	29.69°	28.36°	27.36°
	F_r =	(F_r/Λ =) .1007	.0102	.1212	1.363	2.061	3.264	5.567	10.15
	$\overline{F}_r$ =	($\overline{F}_r/\Lambda$ =) 0.0	.0011	.0980	1.160	1.302	1.388	1.444	1.482
$\lambda = 2$	H_o =	2.147	2.077	1.0	2.218	2.389	2.501	2.565	2.602
	α =	.5671	.5159	0.0	.3626	.4015	.4157	.4221	.4261
	γ =	.6796	.6179	0.0	.5492	.6287	.6765	.7028	.7151
	β =	35.36°	34.45°	0.0	32.84°	31.62°	30.31°	29.44°	28.90°
	F_r =	(F_r/Λ =) .1138	.0125	.3557	1.906	2.946	4.976	9.020	17.11
	$\overline{F}_r$ =	($\overline{F}_r/\Lambda$ =) 0.0	.0049	.3557	1.357	1.434	1.479	1.507	1.526

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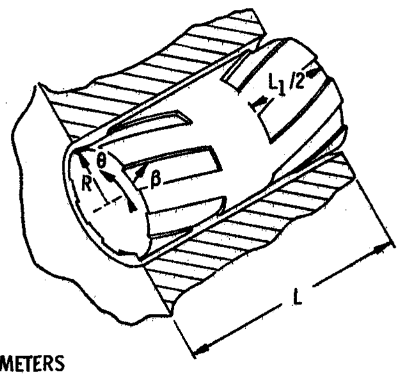

BEARING PARAMETERS

1. $\lambda = \frac{L}{2R}$
2. $\Lambda = \frac{6\mu UR}{p_a h_{20}^2}$

GROOVE PARAMETERS

1. $H_0 = \frac{h_{10}}{h_{20}}$
2. $\alpha = \frac{b_1}{b_1 + b_2}$
3. β
4. $\gamma = \frac{L_1}{L}$

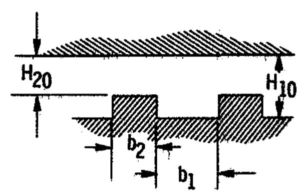

Figure 1. - Schematic of a concentric herringbone journal bearing.

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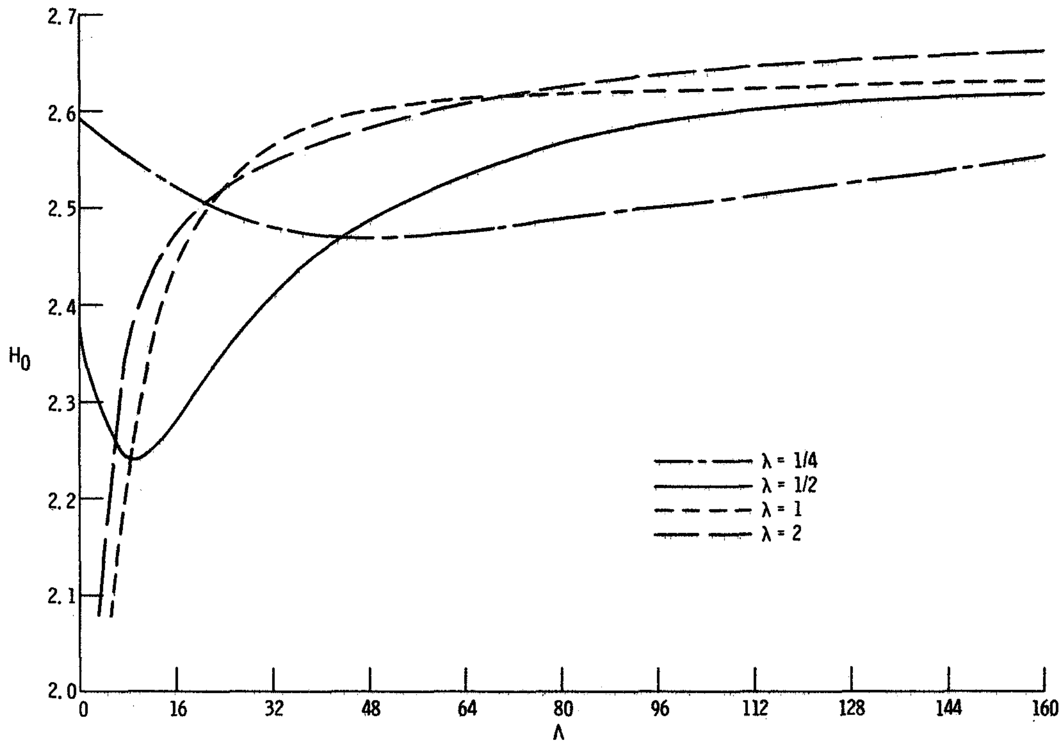

Figure 2(a). - Effect of dimensionless bearing number on optimal film thickness ratio for various length to diameter ratios when the grooved member is rotating.

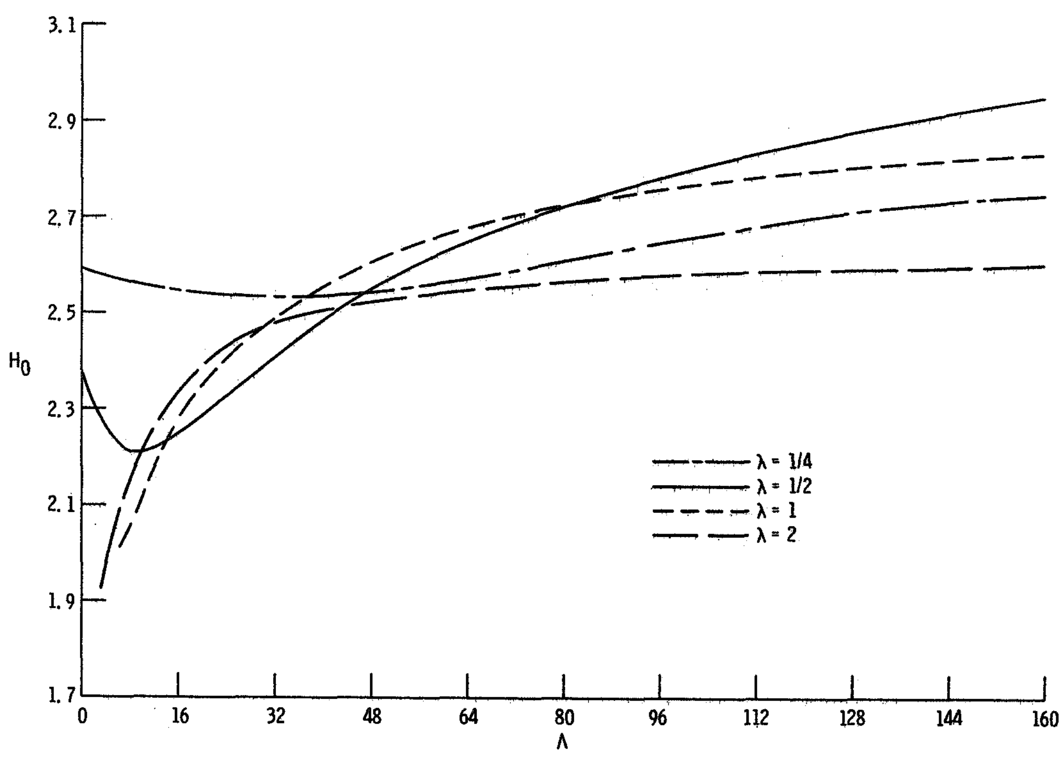

Figure 2(b). - Effect of dimensionless bearing number on optimal film thickness ratio for various length to diameter ratios when the smooth member is rotating.

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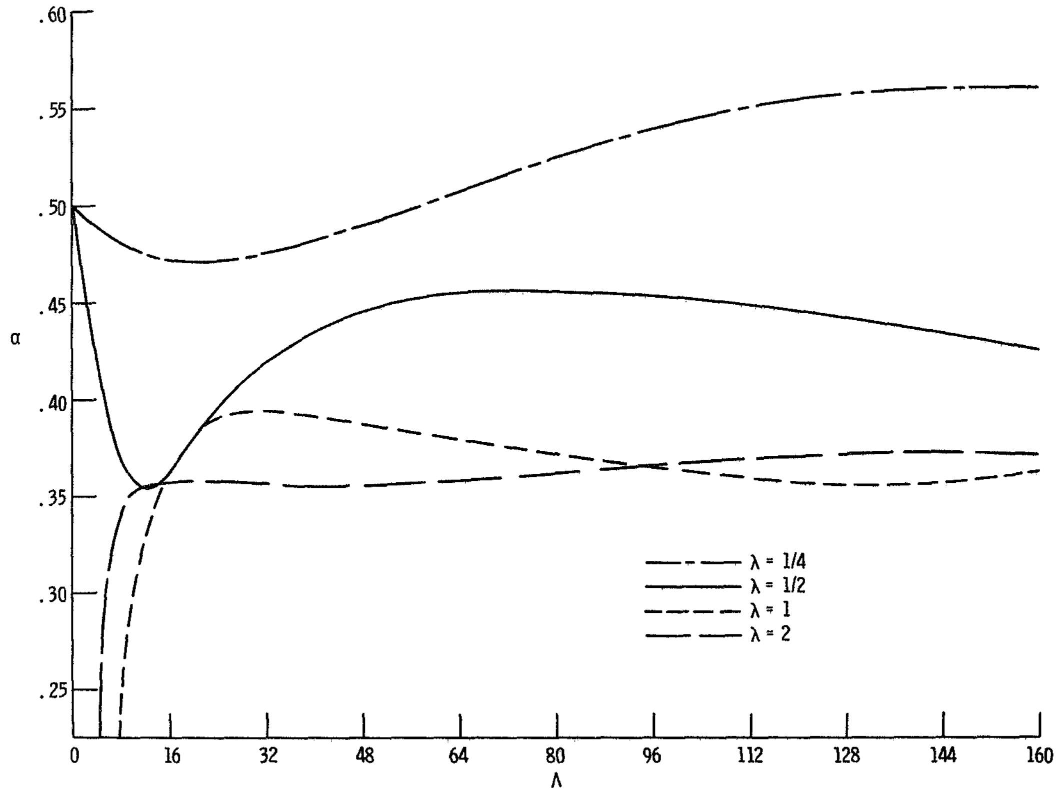

Figure 3(a). - Effect of dimensionless bearing number on optimal groove width ratio for various length to diameter ratios when the grooved member is rotating.

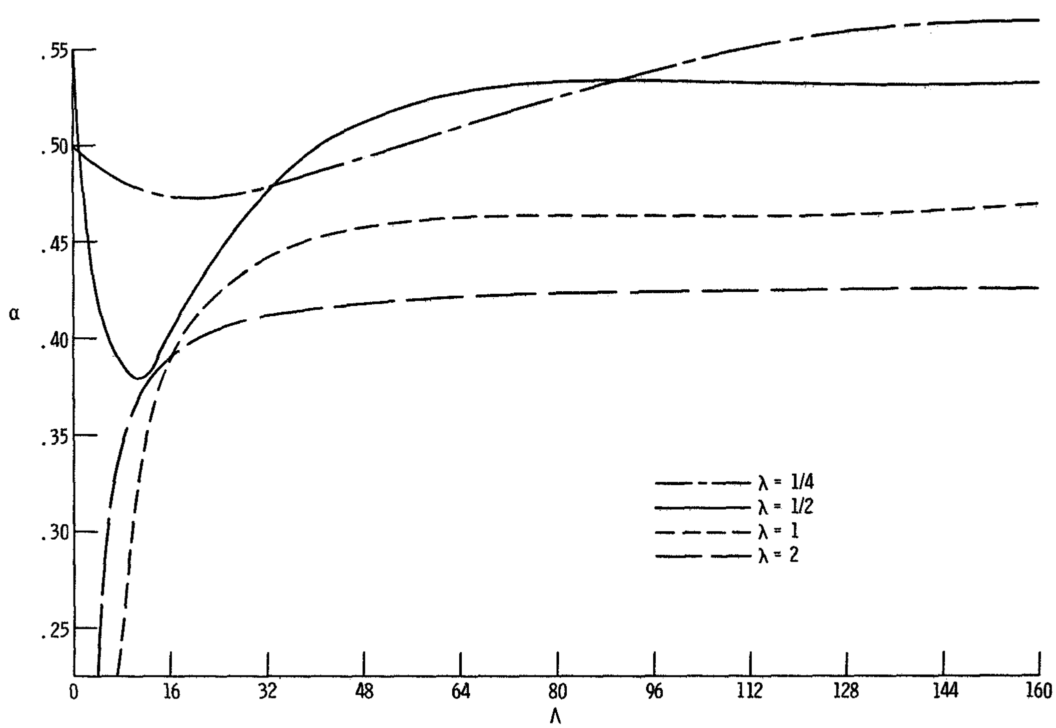

Figure 3(b). - Effect of dimensionless bearing number on optimal groove width ratio for various length to diameter ratios when the smooth member is rotating.

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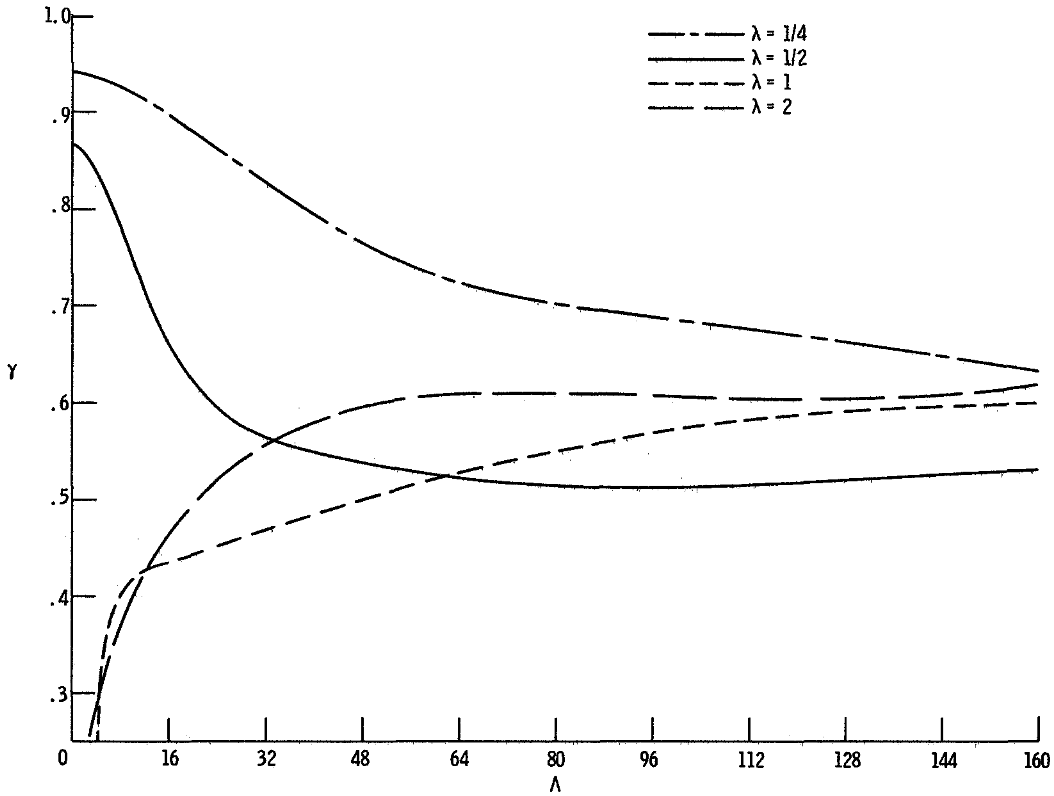

Figure 4(a). - Effect of dimensionless bearing number on optimal groove length ratio for various length to diameter ratios when the grooved member is rotating.

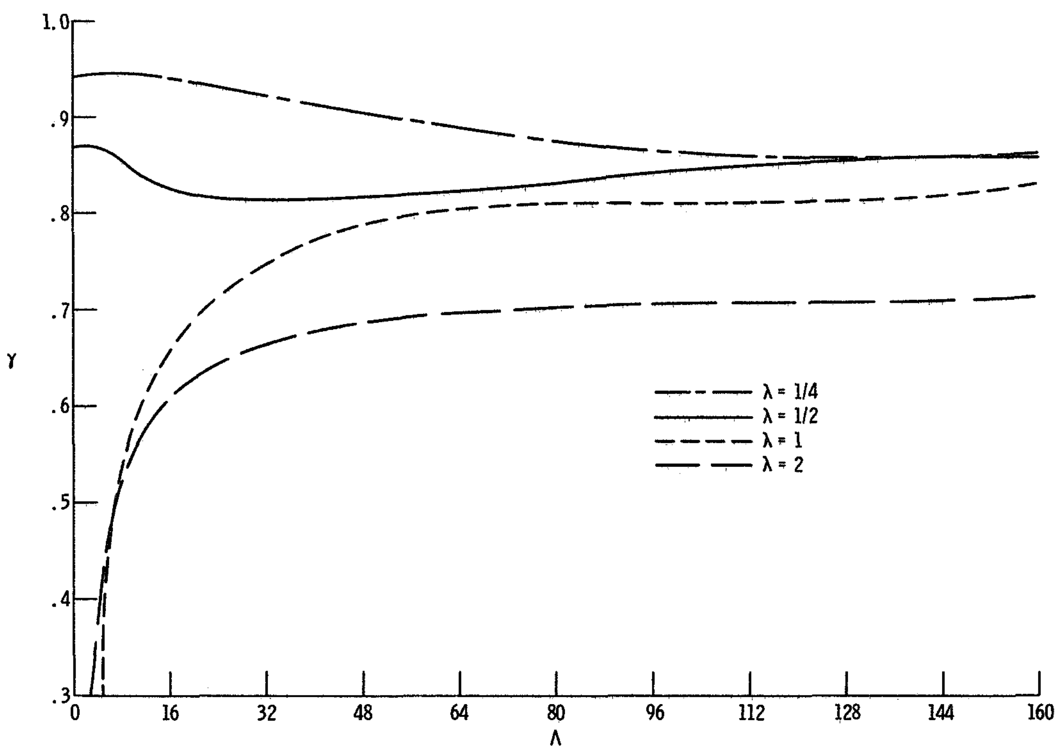

Figure 4(b). - Effect of dimensionless bearing number on optimal groove length ratio for various length to diameter ratios when the smooth member is rotating.

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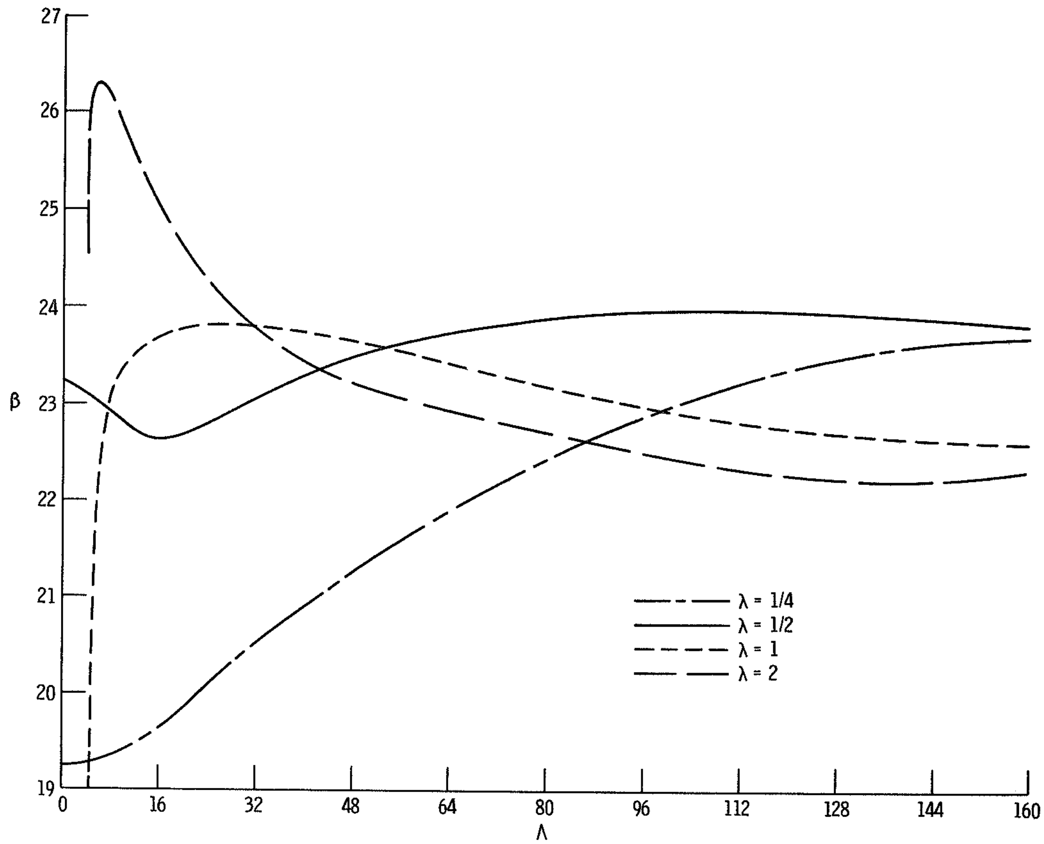

Figure 5(a). - Effect of dimensionless bearing number on optimal groove angle for various length to diameter ratios when the grooved member is rotating.

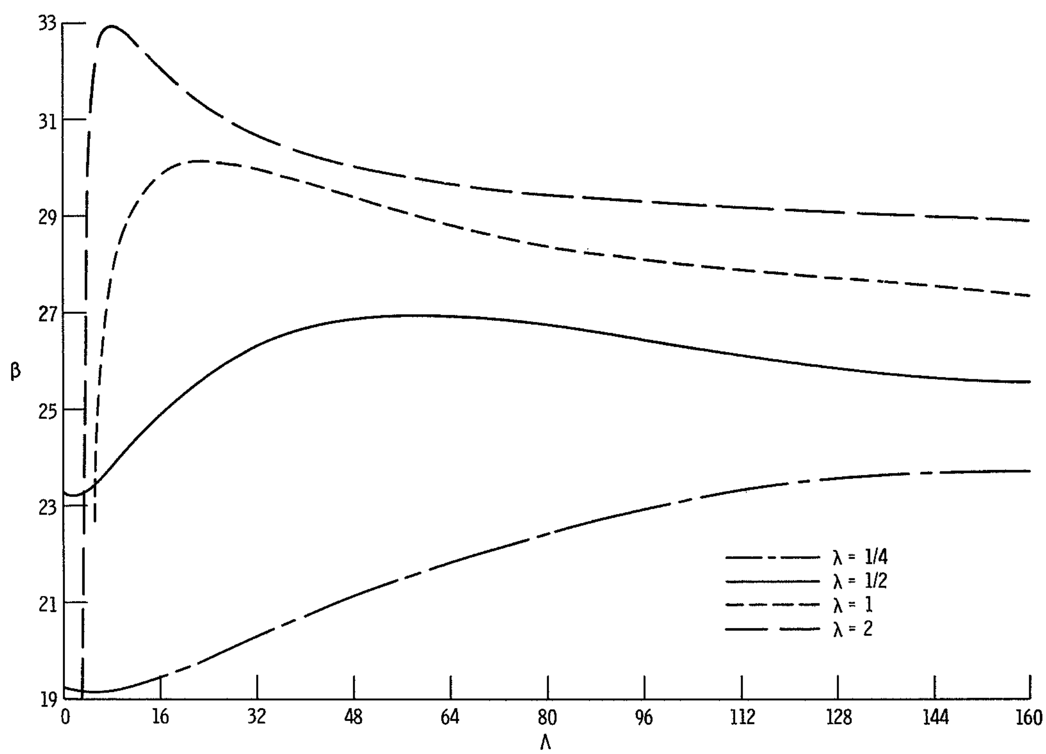

Figure 5(b). - Effect of dimensionless bearing number on optimal groove angle for various length to diameter ratios when the smooth member is rotating.

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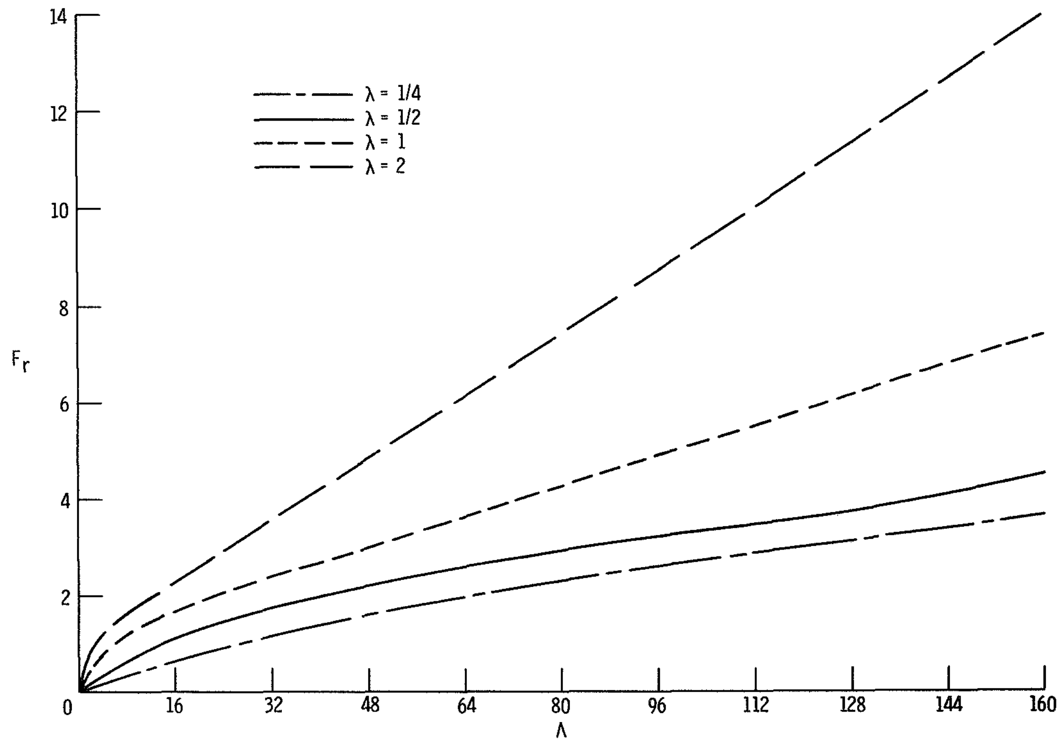

Figure 6(a). - Effect of dimensionless bearing number on maximum dimensionless radial load capacity for various length to diameter ratios when the grooved member is rotating.

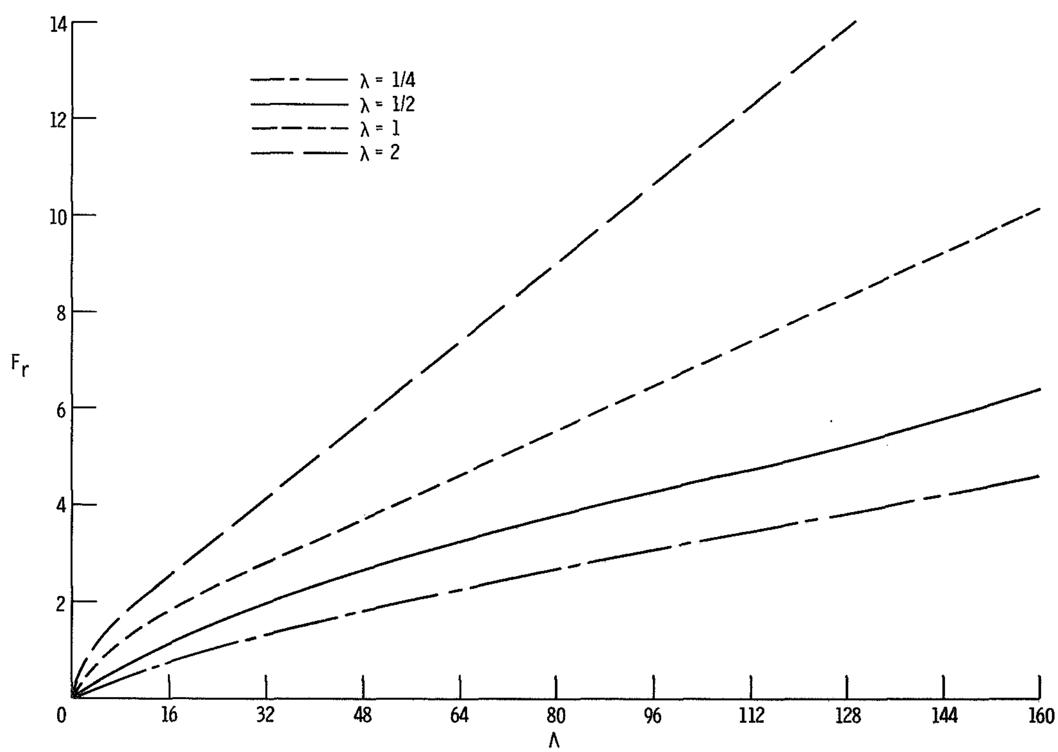

Figure 6(b). - Effect of dimensionless bearing number on maximum dimensionless radial load capacity for various length to diameter ratios when smooth member is rotating.