NASA Contractor Report 195431

Wave Journal Bearing

Part I: Analysis

Florin Dimofte
University of Toledo
Toledo, Ohio

January 1995

Prepared for
Lewis Research Center
Under Grant NAG3–1370

National Aeronautics and
Space Administration

(NASA-CR-195431-Pt-1) WAVE JOURNAL BEARING. PART 1: ANALYSIS Final Report (Toledo Univ.) 10 p

N95-22404

Unclas

G3/37 0040650

WAVE JOURNAL BEARING; PART I: ANALYSIS

Florin Dimofte
University of Toledo
at
NASA Lewis Research Center
Cleveland, Ohio.

ABSTRACT

A wave journal bearing concept features a waved inner bearing diameter of the non-rotating bearing side and it is an alternative to the plain journal bearing. The wave journal bearing has a significantly increased load capacity in comparison to the plain journal bearing operating at the same eccentricity. It also offers greater stability than the plain circular bearing under all operating conditions. The wave bearing's design is relatively simple and allows the shaft to rotate in either direction.

Three wave bearings are sensitive to the direction of an applied stationary side load. Increasing the number of waves reduces the wave bearing's sensitivity to the direction of the applied load relative to the wave. However, the range in which the bearing performance can be varied decreases as the number of waves increases. Therefore, both the number and the amplitude of the waves must be properly selected to optimize the wave bearing design for a specific application. It is concluded that the stiffness of an air journal bearing, due to hydrodynamic effect, could be doubled and made to run stably by using a six or eight wave geometry with a wave amplitude approximately half of the bearing radial clearance.

NOMENCLATURE

B_c = symbolic damping coefficient used in stability analysis
$\bar{B}_{jk}$ (j = x,y; k = x,y) = dynamic damping coefficient, Ns/m
$B_{jk} = C\Omega\bar{B}_{jk}/(p_a LD)$ = dimensionless dynamic damping coefficient
C = journal bearing radial clearance, m
D = journal bearing diameter, m
e = eccentricity, m
e_w =wave's amplitude, m
$F = \bar{F}/(p_a LD)$ dimensionless load capacity
$\bar{F}$ = load capacity, N (the resulting force of the pressure distribution)
$f = \nu/\Omega$ whirl frequency ratio
$\bar{h}$ = film thickness, m
$h = \bar{h}/C$ dimensionless film thickness
$i = \sqrt{-1}$, the imaginary unit
$\bar{K}_{ij}$ (j =x,y; k = x,y) = dynamic stiffness coefficient, N/m (N/μm)
$K_{ij} = C\bar{K}_{jk}/(p_a LD)$ = dimensionless dynamic stiffness coefficient
K_c, K_{c0} = symbolic stiffness coefficient used in stability analysis
L = bearing length, m
M = rotor mass allocated to one bearing; for a symmetric rotor M is half of the rotor mass, Kg
$\bar{M}_c$ = rotor mass, allocated to one bearing, required to make the bearing unstable, Kg (critical mass)
$M_c = \bar{M}_c(\nu_n^2 C)/(p_a LD)$ dimensionless critical mass
n_w = number of waves
O = center of the bearing
O_1 = center of the shaft
p = dimensionless pressure
$\bar{p}$ = pressure, Pa
p_a = ambient pressure, Pa
p_0 = steady state component of the pressure p
p_1, p_2 = perturbation components of pressure p
R = journal bearing radius, m
R_m = mean circle radius, m
t = time, s
W = bearing load, N
x = x-direction (direction of the load on bearing)
y = y-direction perpendicular to x
$x_{1,3}$ = fluid film coordinates
z = axial coordinate parallel to rotor axis
Z_{jk} (j = x,y; k = x,y) = impedance for translatory motion α = angle between the starting point of the wave and the line of centers, dgrs

γ = wave position angle = angle between the starting point of the wave and the direction of the load, dgrs
ϕ = attitude angle = angle between the load direction and the line of centers, dgrs
$\varepsilon = e/C$, eccentricity ratio
$\varepsilon_w = e_w/C$, wave amplitude ratio
ε_0 = eccentricity ratio under static load
ε_1 = dimensionless radial whirl amplitude
$\varepsilon_0\varphi_1$ = dimensionless tangential whirl amplitude
θ = angular coordinate originating at the line of centers
$\Lambda = (6\mu\Omega)(R/C)^2/p_a$, bearing number
μ = dynamic viscosity, Nsm^{-2}
ν = whirl frequency, rad/s
ν_0 = unstable whirl frequency, rad/s
ν_n = synchronous frequency (rotational frequency of the shaft), rad/s
τ = dimensionless time
Ω = rotational frequency, rad/s

INTRODUCTION

The wave journal bearing is an alternative to the plain circular hydrodynamic bearing. It features a waved inner bearing diameter to improve hydrodynamic journal bearing steady-state and dynamic performance.

Previous work [1] based on a numerical model of the wave journal bearing has shown that the wave bearing has a significantly increased load capacity in comparison to the plain circular bearing operating at the same eccentricity. The wave bearing also offers a higher degree of stability than the plain journal bearing under all operating conditions. The analysis was done over a broad range of bearing working parameters (bearing numbers from .01 to 100). The load capacity of the plain circular and wave bearing was compared at the eccentricity ratio of 0.3, while the stability of these bearings was compared at both 0.0 and 0.3 eccentricity ratios. Thus, the results of [1] are applicable over a wide range of wave journal bearing's operating conditions for compressible lubricant (gas).The lubricant was assumed to be compressible, and the lubricating film was assumed to be continuous.

To identify the potential advantages of the wave journal bearing, a three wave journal bearing was compared to a three lobe bearing [2]. Both bearing types analyzed have significantly improved performance in comparison to a plain (truly) circular bearing. The analysis done in [2] establishes when it is beneficial to use each of the bearing geometries based on bearing performance. Thus, the wave bearing shows a better load capacity than the lobed bearings at any applied load and running regime. The stability threshold of the lobe bearing is greater than the that of the wave bearing if the applied load is small. If the applied load increases, the wave bearing dynamic performance is competitive with the lobe bearings.

The wave bearing can have two, or more waves. A three, four, six, and eight wave journal bearing was analyzed in [3] to show the influence of the number of waves. Both the wave amplitude ratio and the number of waves influence the wave journal bearing performance. Therefore, a wave bearing configuration (the number of waves and the wave amplitude) can be optimized for a specific application. The type of load applied to the bearing (major side load or dynamic rotating load) and the bearing dynamic coefficient values required to control the rotor-bearing system behavior are important for an optimal design selection of a wave bearing [3].

Both the rotor-bearing system critical speed and amplitude can be changed by changing the bearing dynamic coefficients via the wave amplitude. Consequently, the rotor-bearing system dynamics can be actively controlled by actively controlling the wave amplitude of bearings which support the rotor [4].

In the present study, the wave bearing concept and the analysis of this bearing are described as a review of the pervious work [1 to 4]. The main advantages of the wave journal bearing compared to the plain (truly circular) journal bearing are presented based on the three wave journal bearing performance. However, the number of waves is of equal importance to the other wave bearing parameters, therefore, the influence of the number of waves is also discussed.

The ratio of bearing length to shaft diameter is assumed to have little influence over the difference between a plain circular and a wave bearing. Thus for this work, all computations were done for an L/D ratio of 1.

ANALYSIS

Wave Bearing Concept:
A three wave journal bearing geometry is shown in Fig. 1.

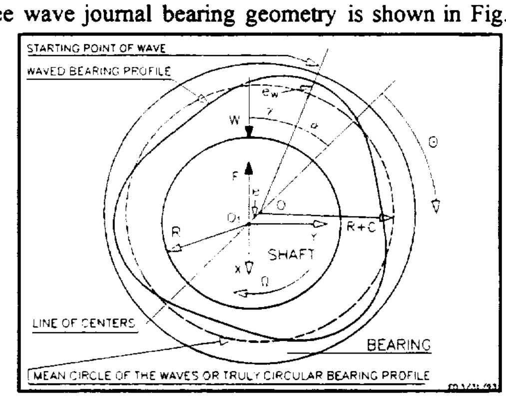

FIGURE 1. WAVE AND PLAIN JOURNAL BEARING GEOMETRY

The mean diameter of the wave bearing (the diameter of the mean circle of the waves) is also the diameter used in this analysis of the plain (truly circular) journal bearing. The radial clearance, C, is the difference between the mean circle radius and the radius, R, of the shaft. The clearance, C, and the wave's amplitude, e_w, are greatly exaggerated in figure 1 so that the concept may be

visualized. The radial clearance, C, is typically less than one thousandth of the journal radius, R, and the wave amplitude, e_w, is typically a fraction, 0.2 - 0.6, of the radial clearance, C.
The wave bearing performance depends on the position of the waves relative to the direction of the applied load (W). This position can be defined by the wave position angle, γ, which is the angle between the starting point of the waves and the direction of the applied load. The amplitude of wave, e_w, the number of waves, as well as the wave position angle, γ, are the basic design parameters of the wave journal bearing. The wave bearing produces similar performance with the shaft rotating in either direction.

When load, W, is applied to the shaft, the shaft must find an equilibrium position at an eccentricity, e, such that the load capacity of the bearing, F, balances the applied load, W (Fig. 1). The load capacity, F, is a result of the pressure generated in the fluid film due to both the rotation of the shaft and the variation in fluid film thickness along the circumference. This variation can be defined by equation:

$$\bar{h} = C + e\cos\theta + e_w\cos[n_w(\theta + \alpha)] \qquad (1)$$

where n_w is the number of waves, α is the angle between the starting point of the wave and the line of centers, and θ is the angular coordinate starting from the line of centers.

The pressure generated in the fluid can be calculated by integrating the Reynolds equation. Assuming a compressible lubricant with isothermal behavior (p/ρ = constant [5, 6, and 7], the Reynolds equation has the following dimensionless form [8]:

$$\frac{\partial}{\partial\theta}\left(h^3\frac{\partial p^2}{\partial\theta}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p^2}{\partial z}\right) = 2\Lambda\frac{\partial(ph)}{\partial\theta} + i4f\Lambda\frac{\partial(ph)}{\partial\tau} \qquad (2)$$

where:

$$p = \frac{\bar{p}}{p_a},\ \theta = \frac{x_1}{R},\ z = \frac{x_3}{R}, \qquad (3)$$

$$\tau = i\,\nu t,\ (i = \sqrt{-1})$$

$$\Lambda = \frac{6\,\mu\,\Omega}{p_a}\left(\frac{R}{C}\right)^2 \qquad (4)$$

The film thickness, h, is made dimensionless by dividing equation (Eq. 1) by the radial clearance, C.

The bearing number, Λ (Eq. 4), is the main working parameter of the bearing. It reflects the bearing working parameters including: dynamic viscosity of the fluid, μ, the ambient operating pressure, p_a, the rotational speed of the shaft, Ω, and the bearing main geometry parameter, (R/C).

Bearing Steady-State and Dynamic Performance:
Bearing steady-state and dynamic performance can be determined by using the small perturbation technique on the complex form of the Reynolds equation (Eq. 2) [8 or 9]. Letting the journal center have a static eccentricity and a corresponding attitude angle, as shown in Fig. 1, the position is perturbed by a small amplitude harmonic motion keeping the shaft and bearing housing axes parallel:

$$\epsilon = \epsilon_0 + \epsilon_1\exp(\tau),\ \phi = \phi_0 + \phi_1\exp(\tau) \qquad (5)$$

Expanded in a Taylor series truncated to the first derivatives, the corresponding perturbed forms of dimensionless film thickness and pressure can be written as:

$$p = p_0 + \epsilon_1\exp(\tau)\,p_1 + \epsilon_0\varphi_1\exp(\tau)\,p_2 \qquad (6)$$

$$h = h_0 + \epsilon_1\exp(\tau)\cos\theta + \epsilon_0 h_\phi \qquad (7)$$
$$where:\ h_\phi = \sin\theta + \epsilon_w n_w\sin(n_w(\theta + \alpha))$$

where h_0 and p_0 are the steady-state component of the film thickness and the pressure, respectively, and p_1 and p_2 are the dynamic components of the pressure. Each component of the pressure (p_0, p_1, and p_2) can be calculated by numerically integrating the corresponding differential equation derived from the Reynolds equation (Eq. 2) as was done in references 8 or 9.

The bearing steady-state and dynamic characteristics can be obtained by integrating the pressure components, p_0, p_1, and p_2, over the whole bearing fluid film. The steady-state load capacity, F, is calculated by integrating p_0, while both the dynamic stiffness, K_{jk}, and damping, B_{jk} (j =x,y; k = x,y), coefficients are calculated by integrating the dynamic pressure components, p_1 and p_2.

Bearing dynamic reaction force:
Under dynamic conditions, the journal (shaft) center whirls in an orbit around its static equilibrium position. The corresponding bearing dynamic reaction force is actually a nonlinear function of the whirl amplitude and depends implicitly on time. In a thorough analysis it is necessary to consider the rotor and the bearing simultaneously. In most practical situations, the amplitude of the shaft whirl is, of necessity, rather small. In these cases a linearization of the bearing reaction force is permissible [8]. Then it becomes possible to treat the bearing separately and represent the bearing reaction force components by means of bearing dynamic coefficients:

$$F_x = -K_{xx}x - B_{xx}\frac{dx}{dt} - K_{xy}y - B_{xy}\frac{dy}{dt} \qquad (8)$$
$$F_y = -K_{yx}x - B_{yx}\frac{dx}{dt} - K_{yy}y - B_{yy}\frac{dy}{dt}$$

Equation (Eq. 8) is only valid when the journal motion is harmonic, and:

$$x = \bar{x}\,\exp(i\nu t) = \bar{x}\,\exp(\tau)$$
$$y = \bar{y}\,\exp(i\nu t) = \bar{y}\,\exp(\tau) \tag{9}$$

The equation (Eq. 8) can be written in complex form as:

$$F_x = -Z_{xx}x - Z_{xy}y;\ F_y = -Z_{yx}x - Z_{yy}y \tag{10}$$

where:

$$Z_{jk} = K_{jk} + i\,\nu\,B_{jk};\ j = x,y;\ k = x,y \tag{11}$$

are the bearing impedance coefficients. For a given bearing geometry, the dynamic coefficients are functions of the static load on the bearing and the rotor speed. The dynamic coefficients also depend on the whirl frequency, and they are actually impedances of the gas film. Note, also, that the x-axis (Fig.1) was chosen along the direction of the steady-state load.

Bearing Stability:
In a bearing stability calculation, it is necessary to evaluate the bearing coefficients over a frequency range, usually around one half of the rotating frequency. On this basis, a stability analysis can be performed in order to calculate the critical mass. The critical mass, M_c, is used to help determine whether the bearing will run free of "half frequency whirl" movement. Half frequency whirl movement is an instability of the fluid lubricant film of the bearing. It appears as a whirling orbiting motion of the shaft and its frequency or speed, ν_0, is often close to one-half the running frequency or shaft speed. This phenomenon is more likely to occur when the shaft center is close to the center of the bearing (near zero eccentricity). This frequency, ν_0, can be much lower than one-half of the running frequency when the value of eccentricity is large [10]. To derive the equation for critical mass, in a simple manner, the rotor is considered rigid and symmetrical, and supported by two identical bearings [8, and 9]. This means that each bearing carries one-half of the rotor mass. If M is the rotor mass supported by the each bearing (M = 1/2 of the rotor mass) and the bearing is represented by its four impedance coefficients, Z_{jk} (j = x,y; k = x,y), the motion equation can be written as:

$$\begin{bmatrix} (Z_{xx} - M\,\nu^2) & Z_{xy} \\ Z_{yx} & (Z_{yy} - M\,\nu^2) \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = 0 \tag{12}$$

The threshold of instability occurs when the determinant of the matrix is zero. Noting:

$$Z = K_c + i\,\nu\,B_c,\ K_c = M\,\nu^2 \tag{13}$$

the determinant equation can be solved to get:

$$Z = \frac{1}{2}(Z_{xx} + Z_{yy}) - \sqrt{\frac{1}{4}(Z_{xx} - Z_{yy})^2 + Z_{xy}Z_{yx}} \tag{14}$$

At the threshold of instability Z must be real. The imaginary part of Z can be evaluated over a range of frequencies to find the frequency value, ν_0, causing the imaginary part of Z to be zero. The corresponding mass, is the mass required to make the bearing unstable under the selected working conditions and is shown by equation 15. This critical mass (Eq. 15) is a threshold and there are three possibilities: i) the critical mass is positive which means that the bearing could run free of half frequency whirl movement if its actual allocated mass is less than the critical mass; ii) the threshold of instability was not found which means that the

$$M_c = \frac{K_{c0}}{\nu_0^2} \tag{15}$$

bearing could run free of half frequency whirl movement at any value of its actual allocated mass; iii) the critical mass is negative which means that the half frequency whirl movement is most likely to occur for any value of the rotor mass.

RESULTS AND DISCUSSION

Comparison of the Wave Bearing to the Plain Journal Bearing

Bearing Load Capacity:
The load capacities of the wave and the plain circular bearings were compared after setting their eccentricity ratios to 0.3.
The load capacity is plotted in figures 2 and 3 versus the bearing number, Λ, using a dimensionless load, F:

$$F = \frac{\bar{F}}{p_a\,L\,D} \tag{16}$$

The wave bearing load capacity depends on the wave's position relative to the direction of the applied load. Varying this position through 120°, for a three wave bearing, results in a maximum and a minimum value of the three wave bearing load capacity. Thus, the maximum and the minimum wave bearing load capacity for each bearing number, Λ, are plotted in figure 2 and 3, respectively.

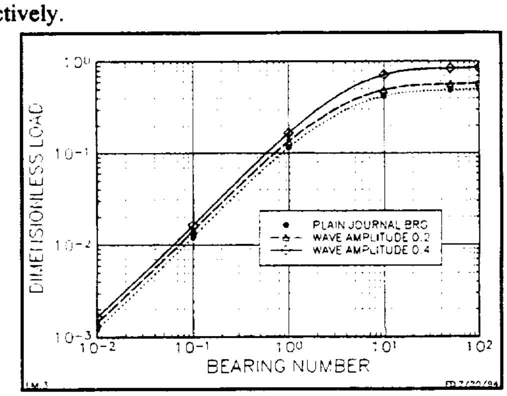

FIGURE 2. MAXIMUM DIMENSIONLESS LOAD CAPACITY AT AN ECCENTRICITY OF 0.3

It is known that the lubricant fluid film compressibility limits the load capacity of the plain circular bearing at values of the bearing number greater than 10 [5 and 12]. This effect is also shown in the wave bearing case (Fig. 2 and 3). Thus, both maximum and minimum wave bearing load capacity are limited if the bearing number is greater than 10. However, both minimum and maximum limits of the wave bearing load capacity at high bearing numbers are greater than the load capacity of the plain circular bearing. The maximum load capacity of the wave journal bearing is also greater than the load capacity of the plain circular bearing at any value of the bearing number. At high bearing numbers (over 10) this difference could be up to 60% for wave amplitude ratio as high as 0.4 (Fig. 2). The wave bearing minimum load capacity is slightly less than the plain circular bearing load capacity at bearing numbers less than 2 (Fig. 3).

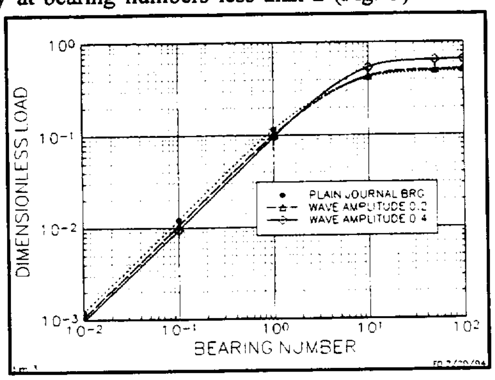

FIGURE 3. MINIMUM DIMENSIONLESS LOAD CAPACITY AT AN ECCENTRICITY OF 0.3

Bearing Stability:
The wave journal bearing's dynamic characteristics is discussed in terms of the bearing critical mass. A dimensionless form for critical mass is used in this discussion and is defined as :

$$M_c = \overline{M}_c \frac{v_n^2 C}{p_a LD} \qquad (17)$$

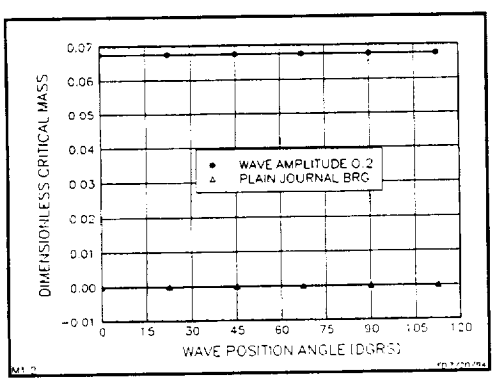

FIGURE 4. DIMENSIONLESS CRITICAL MASS AT AN CONCENTRIC POS. OF THE SHAFT (Λ=1.)

Figures 4 and 5 shows plots of critical mass versus wave position angle for a bearing number, Λ, equal to 1 and two eccentricity ratios (0 and 0.3) for both the wave journal bearing with an amplitude ratio of 0.2 and the plain circular bearing. It can be seen at zero eccentricity ratio (unloaded bearing) that the waves allow the bearing to run free of half frequency whirl motion if its actual allocated mass is less than the critical mass (Fig. 4). The plain journal bearing, at concentric position, starts to whirl with half of its running frequency at any allocated mass. The plain journal bearing load capacity is zero with half frequency motion as described in [5]. Therefore, when the half frequency whirl movement occurs the shaft will touch the bearing surface due any residual radial load (as residual unbalance, rotor weight component, etc) because of a low gas bearing damping performance and failure is likely.

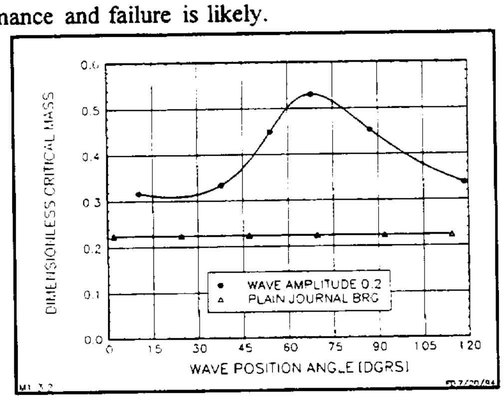

FIGURE 5. DIMENSIONLESS CRITICAL MASS AT AN ECCENTRICITY OF 0.3 (Λ=1.)

At an eccentricity ratio of 0.3 and for bearing numbers equal toor greater than 1, the wave bearing has a greater threshold of stability (critical mass) than the plain circular bearing (e.g. Fig.5). This advantage of the wave bearing can be maximized by properly fixing the wave position angle.

The Influence of the Number of Waves on the Bearing Performance

A generic bearing is used to better understand the wave bearing performance. The selected generic journal bearing has a mean diameter of 200 mm, a length of 100 mm, and a radial clearance of 0.080 mm. The bearing performance was determined at 5,000, 20,000, and 100,000 RPM (the corresponding bearing number, Λ, are 0.89, 3.56, and 17.38, respectively). The bearing is lubricated by atmospheric air. Generic bearings having three, four, six, and eight waves are considered. For each bearing the wave amplitude ratio, $\varepsilon_w = e_w/C$, varies from 0 (truly circular journal bearing) to 0.5. Two eccentricity ratios ($\varepsilon = e/C = 0.2$ and 0.4) are specified as input data to the numerical code. To evaluate the influence of the wave position angle, γ, on the bearing performance, the three, four, six and eight wave bearings are rotated over a range of angles from 0 to 120, 90, 60, and 45 degrees, respectively. However, the analysis at 20,000 RPM for a three and six wave bearings is shown in here. More details can be found in reference [3].

Effect of the Number of waves on the Bearing Load Capacity:
The analysis shows that the wave journal bearing load capacity at

each of the selected eccentricity ratios is strongly influenced by the wave amplitude ratio. Furthermore, it shows that the three wave journal bearing's load capacity varies over a greater range (from 380 to 1750 N at 0.4 eccentricity ratio, Fig. 6), than do the other wave bearings (e.g. the six waves from 380 to 875 N, Fig. 7).

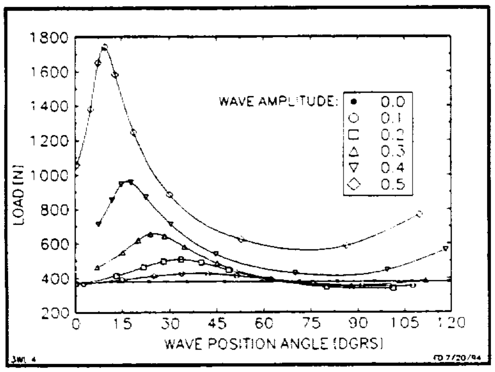

FIGURE 6. THREE WAVE BEARING LOAD CAPACITY AT AN ECCENTRICITY OF 0.4

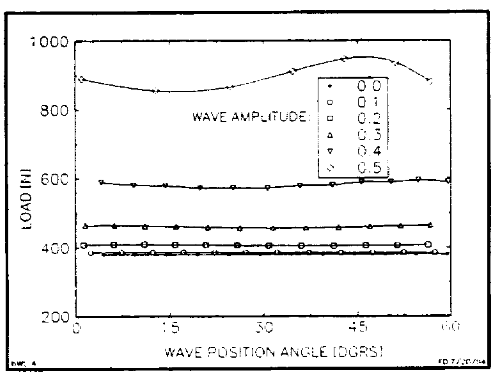

FIGURE 7. SIX WAVE BEARING LOAD CAPACITY AT AN ECCENTRICITY OF 0.4

Thus, a low number of waves such as three waves should be selected if the predominant load on the bearing is a steady-state side load and the bearing (waves) position can be properly fixed. Figure 7 also shows that bearing load capacities are less sensitive to the orientation of applied load to the waves as the number of waves increase.

Consequently, a large number of waves (four or more waves) is required if the direction of the steady-state load varies or a predominant rotating dynamic load is applied to the bearing.

Effect of the Number of Waves on the Bearing Dynamic Coefficients:

Only the direct dynamic stiffness coefficient, $\bar{K}_{xx}$, of all analyzed bearings is strongly influenced by the wave amplitude ratio while

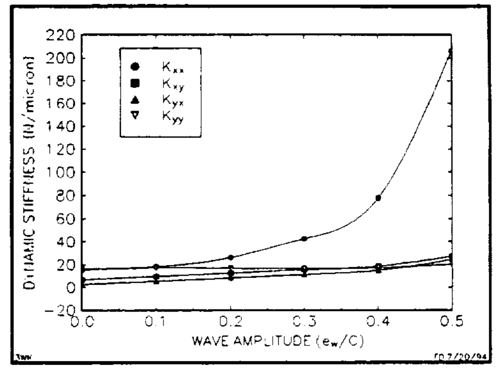

FIGURE 8. THREE WAVE BRG DYNAMIC STIFFNESS AT AN ECCENTRICITY OF 0.4

the rest of the wave bearing dynamic coefficients are almost constant with respect to wave amplitude ratio. The direct dynamic stiffness, $\bar{K}_{xx}$, significantly increases with increasing wave amplitude ratio, especially at large amplitude ratios. Fig. 8 shows that the direct dynamic stiffness, $\bar{K}_{xx}$, could be up to 10 times greater than the truly circular bearing stiffness in the case of a three wave bearing with a wave amplitude ratio of 0.5, at a large eccentricity ratio such as 0.4. This effect decreases if the number of waves increase (e.g. see figure 9 for a six wave bearing). The remaining dynamic stiffness coefficients as well as the dynamic damping coefficients are less sensitive to the wave amplitude ratio than the direct dynamic stiffness coefficient. This physically explains the stabilizing effect of the waves. The shaft reaction forces align more closely with the applied load and the effects of the cross-coupling, destabilized forces become less important as the wave amplitude increases.

The Effect of the Number of Waves on the Bearing Stability:

The fluid film stability of the wave journal bearing is discussed in terms of critical mass [3 and 4]. The numerical results show that the critical mass of all analyzed bearings is dependent on the wave amplitude ratio. All wave bearings are unconditionally stable at large wave amplitude ratios (as 0.5) for an eccentricity ratio of 0.4. However, the stability of a wave bearing with fewer waves can be enhanced, exceeding the stability of a wave bearing with a large number of waves, if the orientation between the wave position and the applied load (the wave position angle) is properly selected. In addition, the wave bearing critical mass is less sensitive to the wave position angle, γ if the number of waves is increased.

A six wave bearing geometry could significantly increase the bearing steady-state and dynamic performance is improved due tothis effect. Thus, the bearing stiffness could be up to 4 or 5 times greater than a plain journal bearing and can run free of the half frequency whirl movement. Using six waves the bearing reacts almost uniformly as the applied load direction changes, and as a consequence, the position of the bearing is not critical. However, the wave amplitude ratio must be greater than 0.2 to allow the bearing performance to be greater than that of the plain journal bearing. It is important to note that the performance of a journal bearing having a profile like a two or three wave bearing due to manufacturing tolerances can become less than that of the

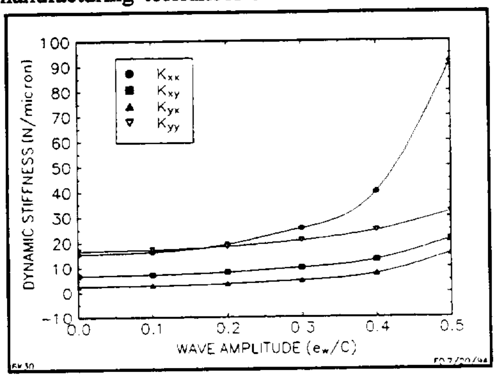

FIGURE 9. SIX WAVE BRG DYNAMIC STIFFNESS AT AN ECCENTRICITY OF 0.4

plain journal bearing if the wave amplitude ratio is below 0.2. Therefore, a cheaper manufacture bearing design should be with large tolerances so that the manufactured journal bearing includes distortions from 0.2 to 0.4 of the bearing clearance. These distortions will make the bearing to perform better than a very precise circular journal bearing.

SUMMARY OF RESULTS

A numerical analysis, reviewed herein, was undertaken to determine: i) the advantages offered by the wave journal bearing as compared to the plain journal bearing using a compressible lubricant; ii) the influence of the number of waves on the bearing performance by using a three, four, six, and eight wave generic bearings. The main conclusions are:

1. The wave journal bearing has greater load (stiffness) and is more stable than a plain journal bearing under most operating conditions. Furthermore, the wave journal bearing offers stability under running conditions where the circular bearing is unstable.

2. Three wave bearings are more sensitive to the direction of the applied load which respect to the waves than the other analyzed bearings especially at low and intermediate bearing numbers (from 0.01 to 1). Therefore, a careful selection of the wave position angle has to be done to maximize its performance. The performance of a two or three wave bearing with wave amplitude of less than 0.2 of the bearing radial clearance can become less than that of the plain journal bearing, especially if the position of a three or four wave, bearing against the applied load is bad selected.

3. The wave bearing is less sensitive to the direction of the applied load relative to waves if a greater number of waves is used. However, the range over which the bearing performance varies decreases as the number of waves increases. Therefore, the actual number of waves must be selected based on the actual rotor-bearing system particularities to optimize the bearing.

4. Stiffness of any air journal bearings, due to hydrodynamic effect, could be doubled and made to run stably by using a six or eight wave geometry with a wave amplitude approximately half of the bearing radial clearance.

ACKNOWLEDGMENTS

The present paper reports work conducted at NASA Lewis Research Center, Cleveland, Ohio, by the author, who is sponsored under grant NAG-1370 awarded to the University of Toledo.

REFERENCES

1. Dimofte, F., "Wave Journal Bearing with Compressible Lubricant; Part I: The Wave Bearing Concept and a Comparison with a Plain Circular Journal Bearing", presented to STLE 1993 Annual Meeting, May 17-20, 1993, Calgary, Canada, to be published in STLE Tribology Transactions.

2. Dimofte, F., "Wave Journal Bearing with Compressible Lubricant; Part. II: A Comparison of the Wave Bearing with a Wave-Groove and a Lobe Bearing," presented to STLE 1993 Annual Meeting, May 17-20, 1993, Calgary, Canada, in printing process to STLE Tribology Transactions.

3. Dimofte, F., "A Wave Journal Bearing Concept with Improved Steady-State and Dynamic Performance," presented to The Seventh Workshop on Rotordynamic Instability Problems in High-Performance Turbomachinery, May 10-12, 1993, Texas A&M University, College Station, TX.

4. Dimofte, F., "A Waved Journal Bearing Concept - Evaluating Steady-State and Dynamic Performance with a Potential Active Control Alternative", presented to the ASME 14th Biennial Conf. on Mechanical Vibration and Noise, September 19-22, 1993, Albuquerque, NM, Paper No.: RS-43, Proceedings DE-Vol.60 Vibration of Rotating Systems, edited by Wang, K.W, and Segalman, D, pp. 121-128.

5. Constantinescu, V.N., "Gas Lubrication," ASME, New York (1969).

6. Szeri, A.Z., "Fluid Film Lubrication," Hemisphere Publishing Corporation, Washington D.C. (1980).

7. Gross, W.A., "Fluid Film Lubrication," John Wiley & Sons, New York (1980).

8. Dimofte, F., "Fast Methods to Numerically Integrate the Reynolds Equation for Gas Fluid Films," NASA Technical Memorandum 105415 (1992).

9. Lund, J.W., "Calculation of Stiffness and Damping Properties of Gas Bearings," ASME Journal of Lubrication Technology,

Series F, 90, 4, pp 793-808 (1968).

10. Dimofte, F., "Effect of Fluid Compressibility on Journal Bearing Performance," STLE Tribology Trans. Vol. 36, 3, pp. 341-350, (1993).

11. Constantinescu, V.N., et al. "Sliding Bearings," Allertin Press Inc., New York, 1985.

12. Raimondi, A.A., "A Numerical Solution for the Gas-Lubricated Bearing of Finite Length," ASLE, Vol. 4, pp 131-155 (1961).

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)

2. REPORT DATE
January 1995

3. REPORT TYPE AND DATES COVERED
Final Contractor Report

4. TITLE AND SUBTITLE
Wave Journal Bearing
Part I: Analysis

5. FUNDING NUMBERS
WU-778-32-21
C-NAG3-1370

6. AUTHOR(S)
Florin Dimofte

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
University of Toledo
Toledo, Ohio 43606

8. PERFORMING ORGANIZATION REPORT NUMBER
E-9423

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135-3191

10. SPONSORING/MONITORING AGENCY REPORT NUMBER
NASA CR-195431

11. SUPPLEMENTARY NOTES
Prepared for the Energy and Environmental Expo 95 sponsored by the American Society of Mechanical Engineers, Petroleum Division, Houston, Texas, January 29—February 1, 1995. Project Manager, James Walker, Space Propulsion Technology Division, NASA Lewis Research Center, organization code 5310, (216) 977-7465.

12a. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified - Unlimited
Subject Categories 18, 20, 07, and 37

This publication is available from the NASA Center for Aerospace Information, (301) 621-0390.

12b. DISTRIBUTION CODE

13. ABSTRACT ***(Maximum 200 words)***

A wave journal bearing concept features a waved inner bearing diameter of the non-rotating bearing side and it is an alternative to the plain journal bearing. The wave journal bearing has a significantly increased load capacity in comparison to the plain journal bearing operating at the same eccentricity. It also offers greater stability than the plain circular bearing under all operating conditions. The wave bearing's design is relatively simple and allows the shaft to rotate in either direction. Three wave bearings are sensitive to the direction of an applied stationary side load. Increasing the number of waves reduces the wave bearing's sensitivity to the direction of the applied load relative to the wave. However, the range in which the bearing performance can be varied decreases as the number of waves increases. Therefore, both the number and the amplitude of the waves must be properly selected to optimize the wave bearing design for a specific application. It is concluded that the stiffness of an air journal bearing, due to hydrodynamic effect, could be doubled and made to run stably by using a six or eight wave geometry with a wave amplitude approximately half of the bearing radial clearance.

14. SUBJECT TERMS
Bearings; Journal bearings; Analysis; Hydrodynamic analysis; Dynamic analysis

15. NUMBER OF PAGES
10

16. PRICE CODE
A02

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102