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A WAVED JOURNAL BEARING CONCEPT WITH IMPROVED STEADY-STATE AND DYNAMIC PERFORMANCE

Florin Dimofte*
NASA Lewis Research Center
Cleveland, Ohio

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ABSTRACT

Analysis of the waved journal bearing concept featuring a waved inner bearing diameter for use with a compressible lubricant (gas) is presented. A three wave, waved journal bearing geometry is used to show the geometry of this concept. The performance of generic waved bearings having either three, four, six, or eight waves is predicted for air lubricated bearings. Steady-state performance is discussed in terms of bearing load capacity, while the dynamic performance is discussed in terms of dynamic coefficients and fluid film stability.

It was found that the bearing wave amplitude has an important influence on both steady-state and dynamic performance of the waved journal bearing. For a fixed eccentricity ratio, the bearing steady-state load capacity and direct dynamic stiffness coefficient increase as the wave amplitude increases. Also, the waved bearing becomes more stable as the wave amplitude increases. In addition, increasing the number of waves reduces the waved 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 waved bearing design for a specific application. It is concluded that the, stiffness of an air 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
B_{ij} (i = x,y; j = x,y) = 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 = ν/Ω whirl frequency ratio
f_0 = ν_0/Ω unstable whirl frequency ratio
h = $\bar{h}/C$ dimensionless film thickness
$\bar{h}$ = film thickness, m
i = $\sqrt{-1}$, the imaginary unit
K_{ij} (i =x,y; j = x,y) = dynamic stiffness coefficient, N/m (N/μm)
$\bar{K}_{ij}$ (i =x,y; j = x,y) = 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
$\bar{M}_c$ = corresponding rotor mass, allocated to one bearing, required to make the bearing unstable, Kg (critical mass)
M_c = $\bar{M}_c(\nu_0^2 C)/(p_a LD)$ dimensionless critical mass
n_w = number of waves

*NASA Resident Research Associate at Lewis Research Center.

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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
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_{ij} (i = x,y; i = 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
ϵ = e/C, eccentricity ratio
ϵ_w = e_w/C, wave amplitude ratio
ϵ_0 = eccentricity ratio under static load
ϵ_1 = dimensionless radial whirl amplitude
$\epsilon_0\varphi_1$ = dimensionless tangential whirl amplitude
θ = angular coordinate originating at the line of centers
Λ = $(6\mu\Omega)(R/C)^2/p_a$, bearing number
μ = dynamic viscosity, Nsm^{-2}
ν = whirl frequency, rad/s
ν_0 = unstable whirl frequency, rad/s
τ = dimensionless time
Ω = rotation frequency, rad/s

INTRODUCTION

Hydrodynamic circular bearings can become unstable, generating a whirl motion whose frequency approximates half the rotation frequency of the shaft. However, hydrodynamic bearings can be made stable by changing the circular fluid film geometry to incorporate recesses, holes, steps, or lobes, although, these changes reduce the bearing's load capacity.

Recently, a new alternative to the plain circular hydrodynamic bearing, a waved journal bearing, was conceived [1]. The waved journal bearing features a waved inner bearing diameter. The numerical model of the waved journal bearing has shown significantly increased steady-state (stiffness) and dynamic (stability and dynamic coefficients) performance as compared to a circular bearing's performance.

The waved bearing concept is shown in figure 1 for a three wave journal bearing geometry. However, the waved bearing can have two, three, four, or more waves. Both three and four wave journal bearings are analyzed in order to show the influence of the number of waves. In addition, a truly circular bearing is analyzed and compared to a waved bearing, showing the performance improvements gained by the waved bearing. A compressible fluid (gas) is assumed as the lubricant. Both steady-state and dynamic bearing performance are predicted using a numerical code based on a perturbation solution of the complex form of the Reynolds equation [2, 3]. Waved bearing performance is computed assuming atmospheric air as the lubricant. The steady-state performance is discussed in terms of bearing load capacity, while the dynamic performance is discussed in terms of bearing stability and bearing dynamic coefficients. Both the wave amplitude ratio and the number of waves influence the waved journal bearing performance. Therefore, a waved 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. Both the rotor-bearing system critical speed and amplitude can be changed by changing the bearing dynamic coefficients via the wave amplitude.

WAVED BEARING CONCEPT

A three wave, waved journal bearing geometry is shown in figure 1. The mean diameter of the waved bearing (the diameter of the mean circle of the waves) is also the diameter of the truly circular bearing to which that the waved bearing is compared. 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 typically, of the radial clearance, C. The waved 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 wave amplitude, e_w, the number of waves, as well as the wave position angle, γ, are the basic design parameters of the waved journal bearing. The waved bearing performance is similar with the shaft rotating in either direction.

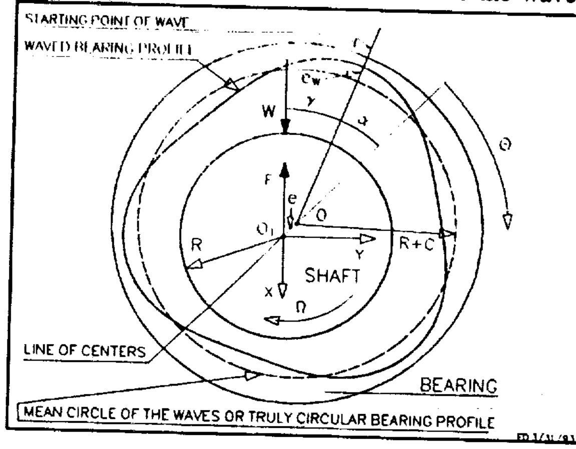

Fig.1 Three wave journal bearing geometry.

Any number of waves can be selected. In the present work the data for a three, four, six, and eight waves, waved journal bearing are compared to the data for a truly circular bearing.

ANALYSIS

When load, W (Fig. 1), 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 owing to both the rotation of the shaft and the variation in fluid film thickness along the circumference. This variation can be defined by:

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

where n_w is the number of waves, α is the angle between the starting point of the wave and the line of centers (Fig. 1), 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, the Reynolds equation has the following dimensionless form [4, 5, 6]:

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

$$+ \, i4f\Lambda\frac{\partial(ph)}{\partial\tau}$$

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 1 by the radial clearance, C.

The bearing number, Λ, (Eq. 4) is the main working parameter of the bearing. It reflects 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:

Both the steady-state and dynamic performance of the bearing can be determined using the small perturbation technique of the complex form of the Reynolds equation (Eq. 2) [2, 3]. Expanded in a Taylor series truncated to the first derivatives, the pressure can be written as:

$$p = p_0 + \epsilon_1 \exp(\tau)\ p_1 + \epsilon_0\varphi_1 \exp(\tau)\ p_2 \qquad (5)$$

where p_0 is the steady-state component and p_1 and p_2 are the dynamic components of the pressure. Each component can be calculated by numerically integrating the corresponding differential equation derived from the Reynolds equation (Eq. 2), [2, 3].

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_{ij}, and damping, B_{ij} (i =x,y; j = x,y), coefficients are calculated by integrating the dynamic pressure components, p_1 and p_2, respectively.

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 as is done, for example, in reference 7. In most practical situations, the amplitude of the shaft whirl is, of necessity, rather small. In these cases, a linearization of the bearing reaction is permissible [2]. Then it becomes possible to treat the bearing separately and represent the bearing reaction force components by means of bearing dynamic coefficients:

$$F_x = -K_{xx}x - B_{xx}\frac{dx}{dt} - K_{xy}y - B_{xy}\frac{dy}{dt}$$
$$F_y = -K_{yx}x - B_{yx}\frac{dx}{dt} - K_{yy}y - B_{yy}\frac{dy}{dt} \qquad (6)$$

Equation 6 is only valid when the journal motion is harmonic, and

$$x = \bar{x}\exp(i\nu t) = \bar{x}\exp(\tau)$$
$$y = \bar{y}\exp(i\nu t) = \bar{y}\exp(\tau) \qquad (7)$$

The equation 6 can be written in complex form as:

$$F_x = -Z_{xx}x - Z_{xy}y$$
$$F_y = -Z_{yx}x - Z_{yy}y \qquad (8)$$

where:

$$Z_{ij} = K_{ij} + i\,\nu\,B_{ij}$$
$$i = x,y; \quad j = x,y \qquad (9)$$

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.

It is also important to note that the bearing's dynamic reaction force components, F_x, and F_y, are functions of the bearing dynamic coefficients, as equation 6 shows. Consequently, these bearing dynamic coefficients influence the rotor-bearing system dynamic behavior. It will be shown that these coefficients depend on the wave amplitude ratio. This means that the rotor-bearing system behavior could be controlled by varying the bearing wave amplitude ratio, e_w.

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" instability [2, 3]. Half frequency whirl 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 [8]. To derive the equation for critical mass, in a simple manner, the rotor is considered rigid and symmetrical, and supported by two identical bearings [2, 3]. 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_{ij} (i = x,y; j = x,y), the motion equation can be written as:

$$\begin{bmatrix} (Z_{xx} - M\nu^2) & Z_{xy} \\ Z_{yx} & (Z_{yy} - M\nu^2) \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = 0 \tag{10}$$

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

$$Z = K_c + i\,\nu\,B_c, \quad K_c = M\,\nu^2 \tag{11}$$

the determinant equation can be solved to get:

$$Z = \frac{1}{2}\,(Z_{xx} + Z_{yy}) - \sqrt{\frac{1}{4}\,(Z_{xx} - Z_{yy})^2 + Z_{xy}\,Z_{yx}} \tag{12}$$

For stability calculations, only the solution with a negative sign in front of the square root proves to be of interest [2]. 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:

$$M_c = \frac{K_{c0}}{\nu_0^{\,2}} \tag{13}$$

If the calculated critical mass for the bearing (Eq. 13) is equal to or greater than one-half the actual rotor mass, then half-frequency whirl instability is likely to occur.

RESULTS AND DISCUSSION

In the present work, a generic bearing is used to better understand the waved 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). However, the data of 20,000 RPM running regime is shown in here. 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, $\epsilon_w = e_w/C$, varies from 0 (truly circular journal bearing) to 0.5. Two eccentricity ratios ($\epsilon = e/C = 0.2$ 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.

Bearing Load Capacity:

The waved journal bearing load capacity at each of the selected eccentricity ratios is strongly influenced by the wave amplitude ratio (Fig. 2). This remark is valid for all analyzed waved journal bearings. However, the three wave journal bearing's load capacity varies over a greater range (from 169 to 395 N at 0.2 eccentricity ratio, Fig. 2a, and from 380 to 1750 N at 0.4 eccentricity ratio, Fig. 2e), then do the other waved bearings(e.g. four waves from 169 to 315 N, Fig. 2b, and from 380 to 1275 N, Fig. 2f; the six waves from 169 to 255 N, Fig. 2c, and from 380 to 875 N, Fig. 2g; the eight waves from 169 to 223 N, Fig. 2d, and from 380 to 725 N, Fig. 2h; at both 0.2 and 0.4 eccentricity ratio, respectively).
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. As a direct

result, the influence of the wave amplitude on bearing performance can be maximized.Fig. 2 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.

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 the rest of the waved 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. 3 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. the direct dynamic stiffness of an eight wave bearing could be only up to 4 times greater than the truly circular bearing). The remaining dynamic stiffness coefficients (Fig. 3) 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.

Fluid Film Bearing Stability:

The fluid film stability of the waved journal bearing will be discussed in terms of "critical mass" (Eq. 13). The numerical results show that the critical mass of all analyzed bearings is dependent on the wave amplitude ratio (Fig. 4). All waved bearings are unconditionally stable at large wave amplitude ratios (as 0.5) for an eccentricity ratio of 0.4. However, Fig. 4 shows also that the stability of a wave bearing with fewer waves is 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 waved bearing critical mass, as the other bearing performance, is less sensitive to the wave position angle, γ if the number of waves is increased.

Actively Controlled Waved Bearing:

The rotor-bearing system behavior can be controlled by the wave amplitude ratio. The wave amplitude ratio influences the wave bearing dynamic coefficients, especially the direct stiffness. As a direct result, the rotor-bearing system critical speed can be changed by changing the bearing stiffness via the wave amplitude. The dynamic forces (Eq. 6) that the bearing applies to the rotor also depend on the wave amplitude ratio. These bearing dynamic forces combined with rotor dynamic forces will determine the rotor-bearing system dynamic behavior. Consequently, the rotor-bearing system dynamics can potentially be actively controlled by actively controlling the wave amplitude of bearings which support the rotor.

Note: This analysis shows that a six or eight wave bearing geometry could increase the hydrodynamic effect and the bearing steady-state and dynamic performance is improved due to this effect. Thus, the bearing stiffness could be up to 4 or 5 times greater and is stable. Using six to eight waves the bearing reacts almost uniformly as the applied load 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 truly circular bearing. It is important to note that the manufacturing tolerance for the wave amplitude are critical in establishing the performance of wave bearings for any application, although this analysis was only done for a compressible lubricant. However, if due to manufacturing tolerances the wave amplitude ratio decrease below 0.2 the bearing performance can become less than that of the truly circular bearing.

SUMMARY OF RESULTS

An analysis was performed to demonstrate the performance improvements achieved by using a waved bearing concept. The waved bearing concept was depicted using a three wave journal bearing geometry. The performance of three, four, six, and eight wave journal bearings operating with air was numerically predicted and discussed. The main conclusions are:

1. The bearing wave amplitude has an important influence on both steady-state (load capacity) and dynamic performance (fluid film bearing stability and dynamic coefficients) of the waved journal bearing. Thus, the waved bearing load capacity could be from 2 to more than 4 times greater than the load capacity of a truly circular bearing, the direct dynamic stiffness could be increased from 4 to 10 times, and the bearing could be turned into an unconditionally stable bearing.

2. The waved 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 can be varied 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.

3. 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. However, a small wave amplitude, less than 0.2 of the bearing radial clearance, the bearing performance can become less than that of the truly circular bearing, especially if the position of a three or four wave, bearing against the applied load is bad selected.

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REFERENCES

1. Dimofte, F., "Waved Journal Bearing with Compressible Lubricant; Part I: The Waved Bearing Concept and a Comparison with a Plain Circular Journal Bearing and Part. II: A Comparison of the Waved Bearing with a Waved-Grooved and a Lobed Bearing," presented to STLE 1993 Annual Meeting, May 17-20, 1993, Calgary, Canada, in printing process to STLE Tribology Transactions.
2. Dimofte, F., "Fast Methods to Numerically Integrate the Reynolds Equation for Gas Fluid Films," NASA Technical Memorandum 105415 (1992).
3. 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).
4. Constantinescu, V.N., "Gas Lubrication," ASME, New York (1969).
5. Szeri, A.Z., Fluid Film Lubrication," Hemisphere Publishing Corporation, Washington D.C. (1980).
6. Gross, W.A., "Fluid Film Lubrication," John Wiley & Sons, New York (1980).
7. Castelli, V., and McCabe, J. T., "Transient Dynamics of a Tilting Pad Gas Bearing System," ASME Journal of Lubrication Technology, Series F, Vol. 89, No. 4, Oct. 1967, pp. 499-509.
8. Dimofte, F., "Effect of Fluid Compressibility on Journal Bearing Performance," STLE Preprint No. 92-TC-1D-1 (1992). 11

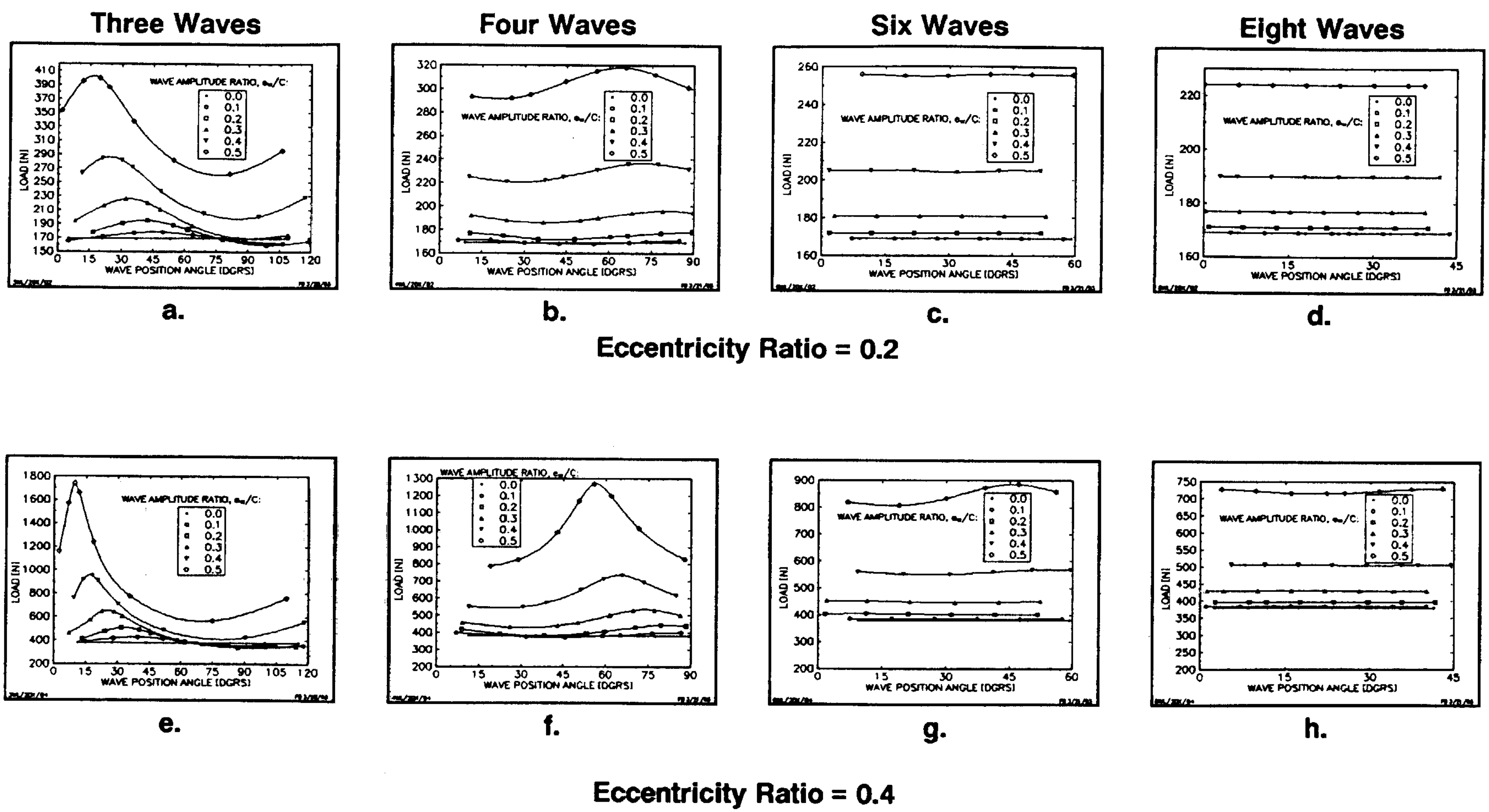
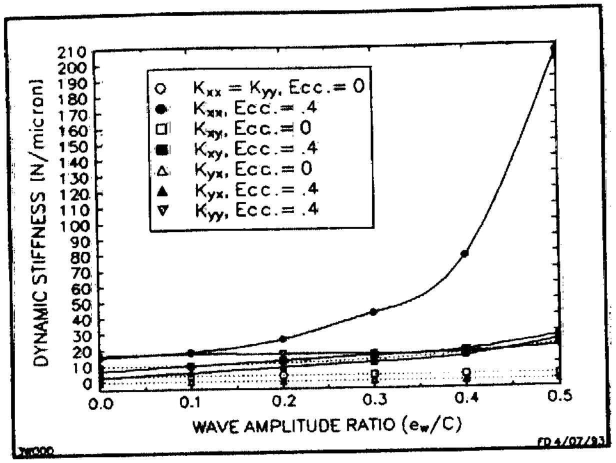
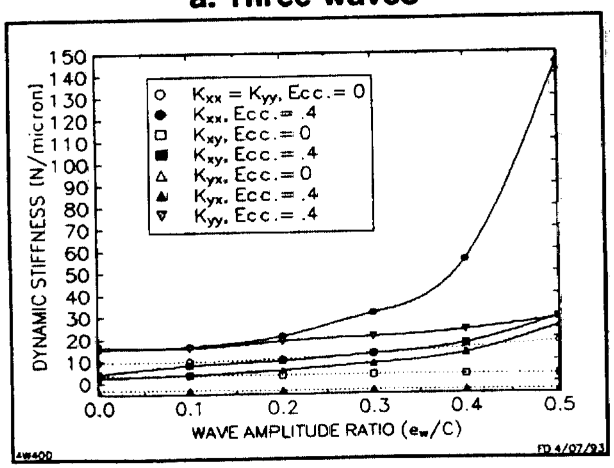


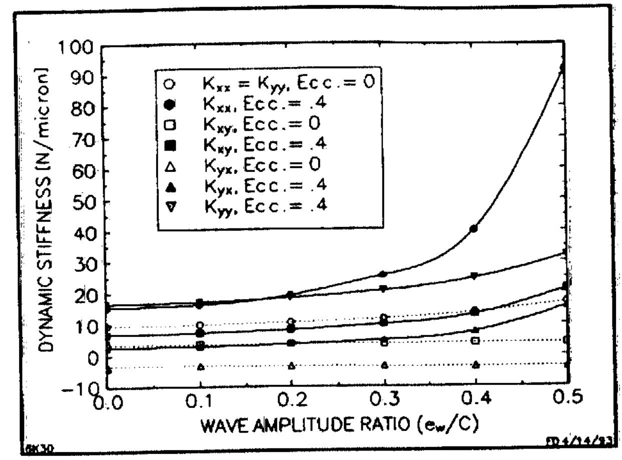
Fig. 2 Wave journal bearing load capacity vs. wave position angle; (D = 200 mm, L = 100 mm, C = 0.080 mm, N = 20,000 RPM)



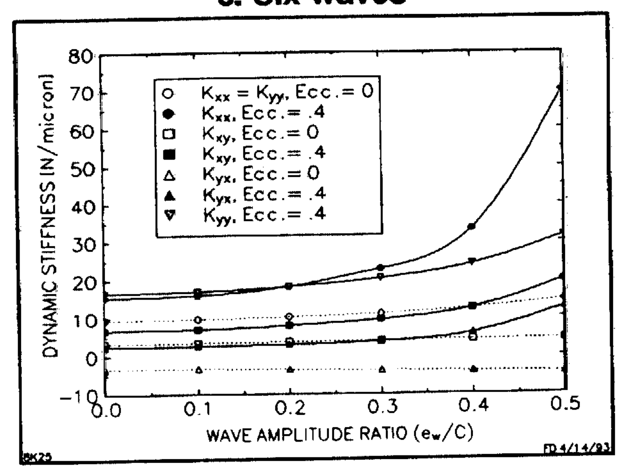
a. Three waves



b. Four waves



c. Six waves



d. Eight waves

Fig.3 Dynamic stiffness vs. wave amplitude ratio at eccentricity ratios 0.0 and 0.4 for a wave journal bearing with:D = 200 mm, L = 100 mm, C = 0.080 mm, N = 20,000 RPM, and various number of waves.

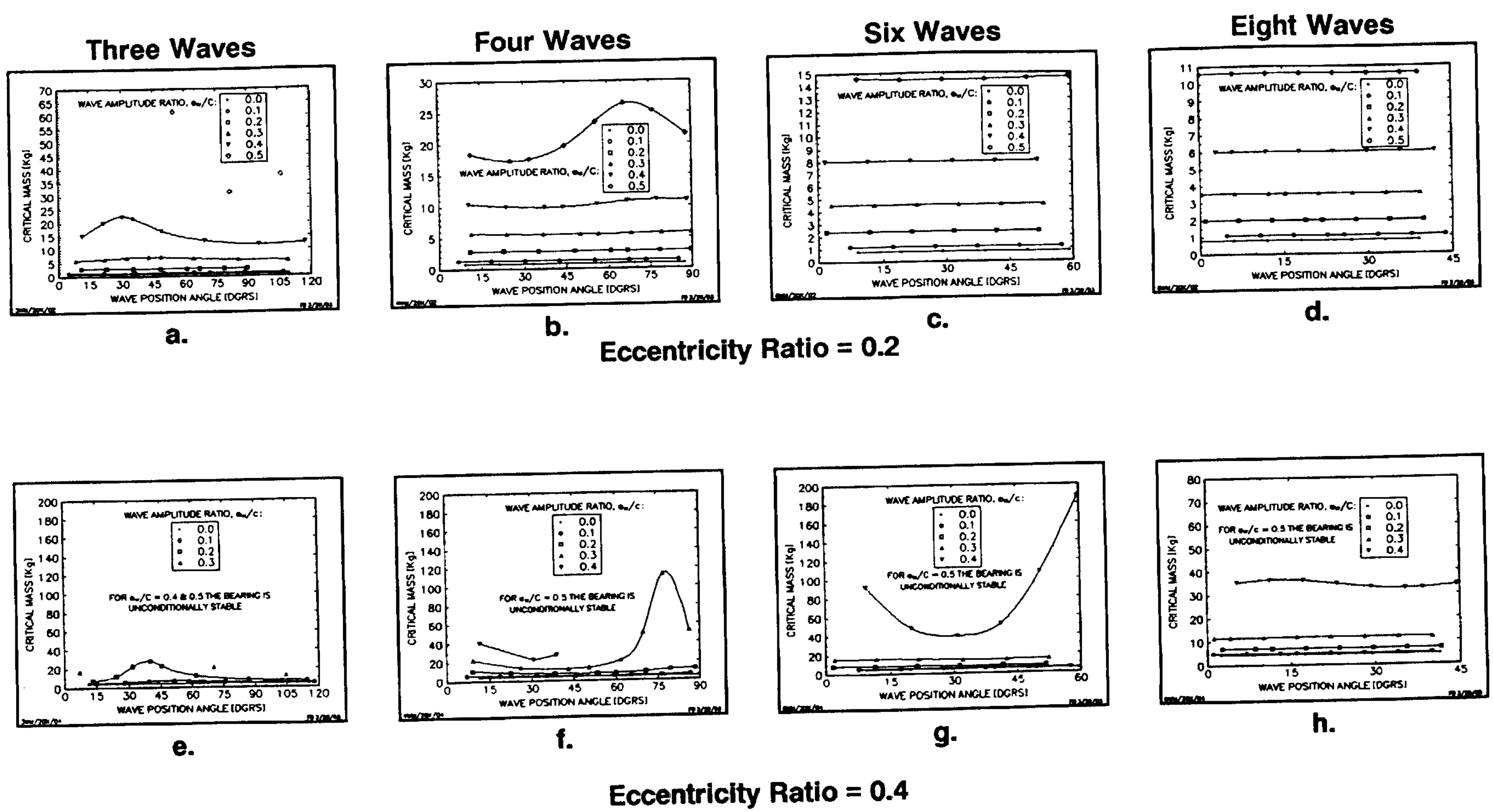


Fig. 4 Wave journal bearing critical mass vs. wave position angle; (D = 200 mm, L = 100 mm, C = 0.080 mm, N = 20,000 RPM)

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13. ABSTRACT ***(Maximum 200 words)***
The first rotordynamics workshop proceedings (NASA CP-2133, 1980) emphasized a feeling of uncertainty in predicting the stability of characteristics of high-performance turbomachinery. In the second workshop proceedings (NASA CP-2250, 1982) these uncertainties were reduced through programs established to systematically resolve problems, with emphasis on experimental validation of the forces that influence rotordynamics. In the third proceedings (NASA CP-2338, 1984) many programs for predicting or measuring forces and force coefficients in high-performance turbomachinery produced results. Data became available for designing new machines with enhanced stability characteristics or for upgrading existing machines. In the fourth proceedings (NASA CP-2443, 1986) there emerged trends toward a more unified view of rotordynamic instability problems and several encouraging new analytical developments. The fifth workshop (NASA CP-3026, 1988) supported the continuing trend toward a unified view with several new developments in the design and manufacture of new turbomachineries with enhanced stability characteristics along with new data and associated numerical/theoretical results. The sixth workshop report (NASA CP-3122, 1990) field experience and experimental results, and expanded the use of computational and control techniques with integration of damper, bearing, and eccentric seal operation results. The present workshop continues to report field experiences, numerical, theoretical, and experimental results and control methods for seals, bearings, and dampers with some attention given to variable thermophysical properties and turbulence measurements, and introduction of two-phase flow results. The intent of the workshop and this proceedings is to provide a continuing impetus for an understanding and resolution of these problems.

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