SETTING VIBRATION CRITERIA FOR TURBOMACHINERY

by

James D. McHugh

Engineering Consultant

Schenectady, New York

James D. McHugh is an Engineering Consultant with particular interest in fluid film bearings, seals, and turbomachinery vibrations and with numerous publications in these areas. He has more than 37 years of experience in research, design, and development with General Electric, Carrier Corporation, and Du Pont.

Mr. McHugh is a Fellow of the ASME, Executive Vice-President of the Vibration Institute, and a past chairman of the ASME Tribology Division. He is a member of the ISO/TC108 committee which develops International Standards for machinery vibration. He was selected in 1987 for the GE Turbine Business Operation award for "Outstanding Technical Contribution."

He holds 13 U.S. patents granted during his 28 year career at GE. Mr. McHugh received a B.S. degree Mechanical Engineering from the University of Rochester and an M.S.M.E. degree from Cornell University.

ABSTRACT

Vibration of a journal within a fluid film bearing produces a dynamic load on the babbitted bearing surfaces. A rational vibration criterion can be based on a consistent magnitude of this dynamic load among widely different machines rather than a simple equality of shaft mils of vibration.

An analysis and curves are provided for some common types of turbomachinery bearings which can be used to correlate mils of vibration with bearing dynamic load. The analysis shows that the percent of bearing clearance consumed by a vibrating journal, together with its steady load, are two important factors in establishing the dynamic load magnitude. These factors can be used to provide a simple estimate of allowable shaft vibration for a specified dynamic load criterion at acceptable, alarm or trip levels.

INTRODUCTION

It is now a common practice to measure shaft vibrations on important turbomachinery in industrial use throughout the world. There has not, however, been a widely accepted consensus on the significance of measured shaft vibration levels. While some criteria have existed for a number of years, as described by Eshelman [1], or have been proposed more recently [2], the basis for them typically is unknown or highly subjective.

The American Petroleum Institute (API) specifies a simple equation, with shaft vibration limits for acceptance varying inversely as the square root of shaft speed [3]. It does not, however, offer any guide for assessing vibration severity for field operation when the acceptance limits are exceeded. Neither does it make a distinction between large and small diameter turbomachinery shafts with correspondingly different clearances.

International standards for interpreting the severity of shaft vibrations have been under consideration by the International Standards Organization (ISO) for several years [4]. A consensus among the member countries is beginning to emerge, but published ISO shaft vibration standards are still some time away.

ISO explicitly recognizes that kinetic (dynamic) load on the machine's bearings provides a useful measure of vibration severity. API implicitly recognizes this same fact by its requirement for shaft vibrations to be measured at, or close to, the bearing journals. Neither standard, however, describes how shaft vibration can be quantitatively translated into bearing load.

The purpose herein is to show how this can be done and how dynamic load can be used as a rational and consistent basis for setting shaft vibration limits.

DYNAMIC BEARING LOAD AS A VIBRATION CRITERION

In the great majority of cases, proximity probes are used to measure the shaft or journal displacements relative to the bearings. The journals of most turbomachines are typically supported on fluid film bearings, such as those pictured in Figures 1, 2, 3, 4, and 5. Motion of the shaft journal relative to the bearing housing displaces the oil and causes a dynamic load to be transmitted to the bearing housing liner.

The magnitude of this dynamic load is a complex function of the bearing geometry, the size and shape of the orbit of the journal, and its frequency. The relationship between dynamic load and the shaft orbit may be calculated using the spring and damping coefficients of the fluid film bearings, pictured schematically in Figure 6.

These coefficients are widely used by rotordynamicists to predict critical speeds, response to unbalance, etc., for turbomachines. Fluid film bearing computer programs are now readily available which are capable of producing values for these coefficients in either a dimensional for dimensionless form. For

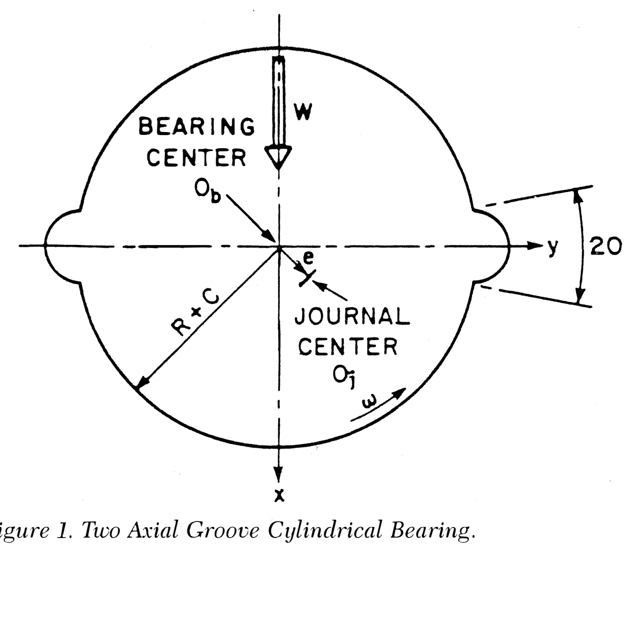

Figure 1. Two Axial Groove Cylindrical Bearing.

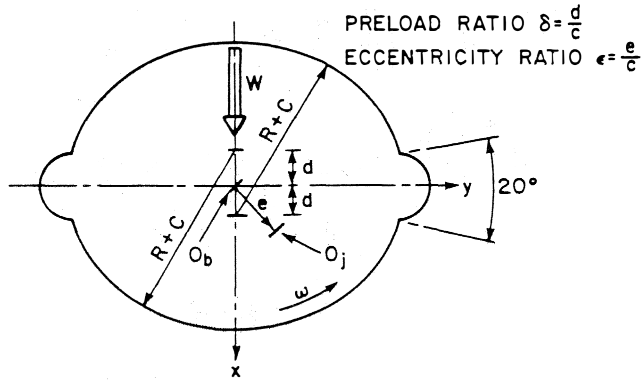

Figure 2. Elliptical Bearing.

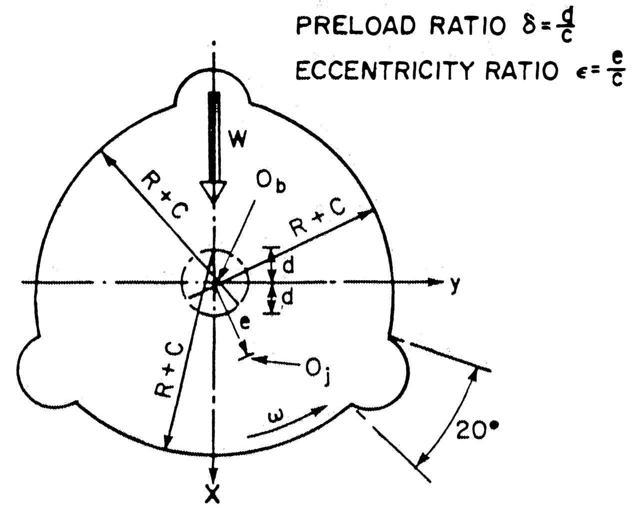

Figure 3. Three-Lobe Bearing.

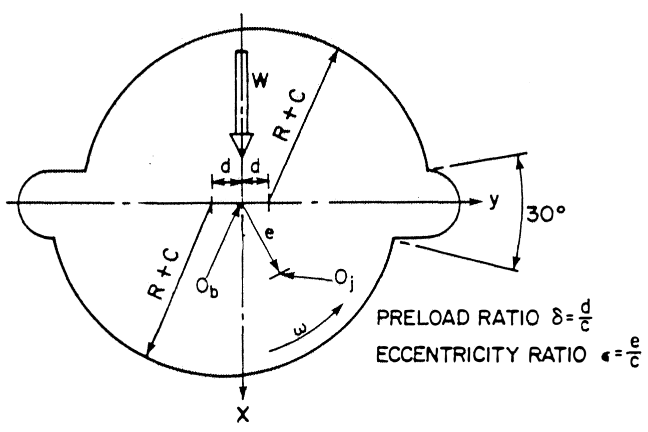

Figure 4. Offset Cylindrical Bearing.

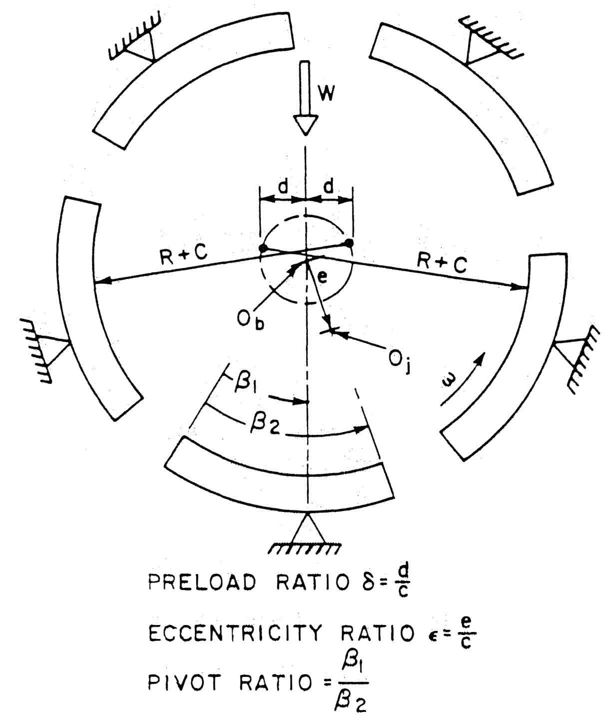

Figure 5. Tilting-Pad Journal Bearing, on Pad Loading.

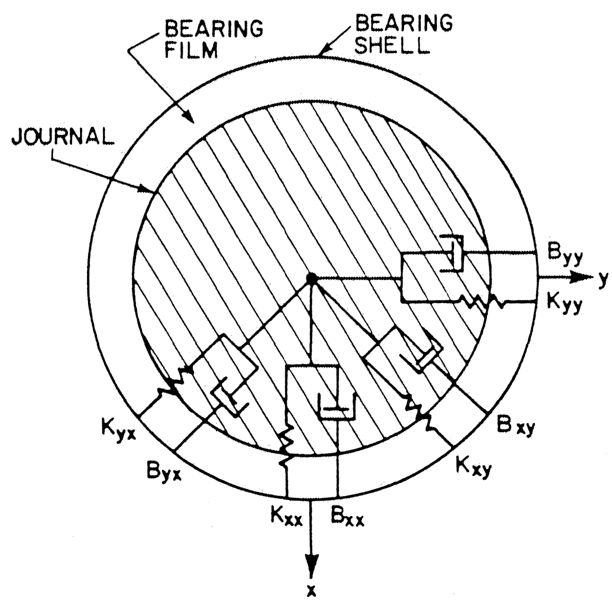

Figure 6. Dynamic Representative of a Bearing by Spring and Damping Coefficients.

the purpose of this presentation, however, the coefficients have been obtained from the published data [5, 6].

Different rotors clearly may exhibit different responses to a given level of distributed disturbing force such as unbalance. The sensitivity of a rotor to these forces requires a rotordynamics analysis with a complete characterization of the rotor, stator, and bearings. In contrast, the present study considers only the result of these forces in producing dynamic bearing loads; it is concerned, therefore, only with the journal orbits within the bearings.

The dynamic load produced by an orbiting journal, however, is often of paramount importance in establishing the maximum allowable shaft vibration. The reason is that an orbiting journal always produces a time-varying dynamic load on the bearing liner. This can lead to high alternating stresses in the soft and relatively weak babbitted surfaces used in most turbomachinery bearings. These stresses may lead to babbitt fatique with the potential for destruction of the whole rotor.

In contrast with the bearing situation, whirling of a shaft may produce little or no alternating bending stresses in the rotor. The strength of the materials used on rotors far exceeds that of babbitt; bearing babbitt fatigue failure can, therefore, be expected to precede a catastrophic rotor failure due to its alternating bending stresses.

ANALYSIS ASSUMPTIONS AND LIMITATIONS

The fundamental assumption is that bearing dynamic load is the useful measure of vibration severity. A secondary assumption is that the fluid film bearing behavior can be adequately represented by a set of eight linearized spring and damping coefficients. A third, implicit assumption is that proximity probes are used to measure shaft vibration relative to the bearing liner, rather its absolute value in space.

Possible shaft orbits can range from a simple ellipse produced by rotor unbalance to the very complex pattern produced when the shaft vibration contains both synchronous and nonsynchronous components.

The analysis first considers the most common case of a synchronous, elliptical orbit. The maximum dynamic load is calculated for an arbitrary elliptical orbit shape and orientation with respect to an axis parallel to the static load vector. While the dynamic load may be calculated from the ellipse parameters and the bearing coefficients, it is convenient to consider the limit case when the major and minor axes of the ellipse are equal. The ellipse then becomes a circular orbit. The number of parameters entering the dynamic load equation is significantly reduced. It becomes possible, therefore, to make a simple plot of a useful amplitude severity parameter for a given bearing over its entire operating range.

The analysis may be extended to cover nonsynchronous orbits, either above or below the shaft rotational frequency. Again, it is necessary to make some simplifying assumptions in order to reduce the number of possible parameters and provide a simple estimate of the dynamic load.

It is assumed that the nonsynchronous component also produces a circular vibration orbit at its own frequency, resulting in its own contribution to the dynamic load. The total load is then taken as the simple sum of the maximum synchronous and nonsynchronous components.

In principle, this method of estimating dynamic load can be used for various nonsynchronous frequencies. In practice, two particular cases of nonsynchronous vibration are more common and are of particular interest. These are the cases when the nonsynchronous vibration is at or close to one-half the shaft rotational frequency, and when it is at twice this value. Curves are provided herein for different bearing geometries which can be used to estimate dynamic load resulting from synchronous vibrations alone, or those combined with either half-per-rev or two-per-rev frequency components.

The curves shown herein are for the indicated bearing length to diameter ratio L/D. As shown in previous studies [7, 8], however, there is little change in the amplitude severity parameter for a range of L/D from 0.5 to 1.0. The values presented in this paper for amplitude severity should be realistic within this range.

It must be remembered, however, that dynamic bearing load is not the sole or even best indicator of vibration severity for all machines under all circumstances. Clearly, if the bearings are located at shaft nodes and the rotor is highly flexible, the possibility of mid-rotor rubs may be a more significant concern than bearing damage. In such a case it is far more useful to have proximity probes closer to the potential rub sources than at the bearings.

With some exceptions, therefore, it is believed that the use of bearing dynamic load as a vibration criterion can provide a useful and consistent measure of vibration severity for different types and sizes of turbomachinery.

DYNAMIC LOAD PRODUCED BY SYNCHRONOUS SHAFT ORBITS

The equations relating bearing dynamic load, the bearing fluid film spring and damping coefficients, and an arbitrary elliptical shaft orbit are developed completely in a previous study [7] and will only be summarized herein.

The elliptical shaft orbit is pictured in Figure 7. The origin of the coordinate axes coincides with the center of the ellipse. Physically, the origin corresponds to the undisturbed position of the shaft center in the absence of any dynamic loads. The ellipse axes are rotated an amount α from the positive x direction, which corresponds to the static load vector direction. The shaft angular orientation is also pictured in Figure 7 for four discrete positions of the shaft center on the orbit traced during one shaft revolution.

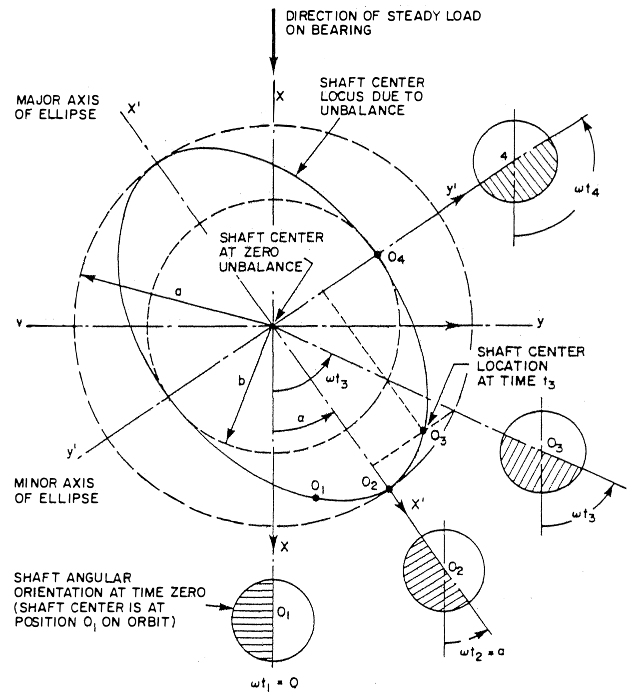

Figure 7. Coordinate References Axes for Shaft Center Orbit (with shaft angular orientation pictured for four orbit positions).

The steps taken in the analysis are:

- Express the shaft orbit in terms of the parametric equations for an ellipse.
- Relate the x and y coordinate displacement measurements to the ellipse parameters.
- Write equations for the time-varying forces in the x, y directions, e. g.,

$$F_x = K_{xx}x + B_{xx}\dot{x} + K_{xy}y + B_{xy}\dot{y}$$
$$F_y = K_{yy}y + B_{yy}\dot{y} + K_{yx}x + B_{yx}\dot{x}$$

- Combine the forces F_x and F_y to find the time-varying radial force F_r.

- Find the maximum value of the radial force F_r.
- Express the resulting force F_r in dimensionless terms.

When these steps are taken, the resulting equation is written as:

$$\frac{\sqrt{2}(F_r)_{max}c}{Wa\cos^2\alpha}=\{\bar{A}^2+\bar{B}^2+\bar{C}^2+\bar{D}^2+[(\bar{A}^2+\bar{B}^2)^2+(\bar{C}^2+\bar{D}^2)^2-2(\bar{A}^2-\bar{B}^2)(\bar{D}^2-\bar{C}^2)+8\bar{A}\bar{B}\bar{C}\bar{D}]^{1/2}\}^{1/2} \quad (1)$$

where

$$\bar{A}=\bar{K}_{xx}(1+b/a\tan^2\alpha)+\bar{B}_{xy}(\tan^2\alpha+b/a)+(\bar{B}_{xx}+\bar{K}_{xy})(1-b/a)\tan\alpha \quad (2)$$

$$\bar{B}=(\bar{K}_{xx}-\bar{B}_{xy})(1-b/a)\tan\alpha-\bar{B}_{xx}(1+b/a\tan^2\alpha+\bar{K}_{xy}(\tan^2\alpha+b/a) \quad (3)$$

$$\bar{C}=(\bar{K}_{yy}+\bar{B}_{yx})(1-b/a)\tan\alpha+\bar{B}_{yy}(\tan^2\alpha+b/a)+\bar{K}_{yx}(1+b/a\tan^2\alpha) \quad (4)$$

$$\bar{D}=\bar{K}_{yy}(\tan^2\alpha+b/a)-\bar{B}_{yx}(1+b/a\tan^2\alpha)+(\bar{K}_{yx}-\bar{B}_{yy})(1-b/a)\tan\alpha \quad (5)$$

and the terms $\bar{K}_{xx}$, $\bar{B}_{xy}$, etc. are dimensionless spring and damping terms for the fluid film, i.e.:

$$\bar{K}_{xx}=K_{xx}\,c/W \qquad \bar{K}_{yy}=K_{yy}\,c/W$$
$$\bar{B}_{xx}=\Omega B_{xx}\,c/W \qquad \bar{B}_{yy}=\Omega B_{yy}\,c/W \text{ etc.} \quad (6)$$

where

W = total static load on the bearing

c = bearing radial clearance (the difference between the journal radius and the radius of curvature of the bearing liner, sometimes referred to as the "ground" clearance.)

It will be seen that the right-hand side of Equation (1) depends solely on the dimensionless constants $\bar{A}$, $\bar{B}$, $\bar{C}$, $\bar{D}$. These constants are a function of three factors, i. e.:

- The dimensionless spring and damping coefficients. These coefficients are a function of bearing geometry and operating conditions; for a given bearing at a known eccentricity ratio, the coefficients are fixed values.
- The aspect ratio of the elliptical orbit formed by the journal center, i. e., the ratio a/b of major to minor axes of the ellipse.
- The orientation angle α of the shaft major axis with respect to the vertical load vector.

The left-hand side of Equation (1) is a dimensionless quantity which contains the maximum bearing dynamic force, $F_{r,\,Max}$, the static force W on the bearing, the bearing clearance c, the ellipse orientation angle α and the major ellipse orbit axis a. For a fixed value of the right-hand side of Equation (1), if the angle α and the static load $F_{r,\,Max}$ is specified, the ratio (a/c) is then fixed. That is, the maximum vibrational amplitude, a, for a bearing with clearance c can be specified.

For the case of the elliptical shaft orbit, the constants $\bar{A}$, $\bar{B}$, $\bar{C}$, $\bar{D}$ of Equation (1) may have an infinite number of values for a given eccentricity ratio. Although the dimensionless spring and damping coefficients are fixed by the eccentricity ratio, different variables of ellipse attitude α and aspect ratio may be arbitrarily assigned in computing these constants $\bar{A}$, $\bar{B}$, etc. Hence, a plot of Equation (1) would require selected parameters of a/b and α.

To reduce the number of such plots and present a simple method for estimating allowable shaft motion, assume that the shaft orbit is circular. In this case, the ratio a/b = 1 and α = 0. Hence, the constants $\bar{A}$, $\bar{B}$, $\bar{C}$, $\bar{D}$, from Equations (2, 3, 4, and 5), to be used in Equation (1) become:

$$A_1=\bar{K}_{xx}+\bar{B}_{xy} \quad (7)$$
$$B_1=-\bar{B}_{xx}+\bar{K}_{xy} \quad (8)$$
$$C_1=\bar{B}_{yy}+\bar{K}_{yx} \quad (9)$$
$$D_1=\bar{K}_{yy}-\bar{B}_{yx} \quad (10)$$

where the subscript 1 indicates the particular case of a circular orbit and frequency Ω_1.

When a/b = 1 and $\alpha = 0°$, the left-hand side of Equation (1) is reduced to:

$$\frac{\sqrt{2}(F_r)_{max}c}{Wa\cos^2\alpha}=\frac{\sqrt{2}(F_r)_{max}c}{Wr_1} \quad (11)$$

where r_1 = radial amplitude of the circular shaft orbit.

DYNAMIC LOAD PRODUCED BY NONSYNCHRONOUS SHAFT ORBITS

The equations for estimating the dynamic load produced by a combined synchronous and nonsynchronous shaft vibration are developed in a 1986 article [8], and again, will only be summarized here. Although the article [8] is concerned specifically with the contribution to bearing load produced by a half-per-rev shaft vibration, the analysis can also be applied to shaft vibrations above the running speed.

It is assumed that the nonsynchronous component of shaft vibration also produces a circular orbit, but at a radius r_2 and a frequency Ω_2. The equation for estimating dynamic lead produced by this nonsynchronous component alone is identical in form to that of Equation (1). Each of the values for the terms of Equations (7, 8, 9, and 10), however, must be modified slightly to account for the nonsynchronous frequency Ω_2. Thus,

$$A_2=\bar{K}_{xx}+\Omega_2/\Omega_1(\bar{B}_{xy}) \quad (12)$$
$$B_2=\bar{K}_{xy}-\Omega_2/\Omega_1\,\bar{B}_{xx} \quad (13)$$
$$C_2=\bar{K}_{yx}+\Omega_2/\Omega_1\,\bar{B}_{yy} \quad (14)$$
$$D_2=\bar{K}_{yy}-\Omega_2/\Omega_1\,\bar{B}_{yx} \quad (15)$$

where the subscript 2 indicates that the quantity refers to the nonsynchronous value.

For a given bearing operating at a fixed eccentricity ratio, the dimensionless spring and damping coefficients of Equations (7, 8, 9, and 10) are constant. Hence, for a circular orbit of radius r_1 and a synchronous frequency Ω_1, the right-hand side of Equation (1) is a constant P for a given bearing at a given eccentricity.

The dimensionless load for a circular, synchronous orbit may be written as:

$$\frac{\sqrt{2}\,F_{r1}\,c}{Wr_1}=P \quad (16)$$

where $P=f(A_1, B_1, C_1, D_1)$ and A_1, B_1, etc. are given by equations (7-10).

Similarly, for a circular orbit of radius r_2 at a frequency Ω_2, the right-hand side of Equation (1) is a constant Q at a given bearing eccentricity ratio.

The dimensionless load for the circular, nonsynchronous orbit may be written as:

$$\frac{\sqrt{2}\, F_{r2}\, c}{W r_2} = Q \qquad (17)$$

where $Q = f(A_2, B_2, C_2, D_2)$ and A_2, B_2, etc. are given by equations (12-15).

Combining equations (16) and (17) and rearranging,

$$\frac{\sqrt{2}(F_{r1}+F_{r2})}{W} = P\, r_1/c + Q\, r_2/c \qquad (18)$$

The quantity $(F_{r1} + F_{r2})$ represents the total force F_t, resulting from the synchronous and nonsynchronous components of shaft vibration.

The total force F_t may be written as the product of the dynamic unit pressure P_d and the projected bearing area LD. Similarly, the static load W may be written as the product of the static unit pressure P_s and the projected bearing area LD. Thus, Equation (18) may be inverted and written as:

$$\frac{P_s\, r_1}{P_d\, c} = \frac{\sqrt{2}}{P[1+(r_2/r_1)\, Q/P]} \qquad (19)$$

VIBRATION AMPLITUDE SEVERITY CURVES FOR SELECTED BEARINGS

Equation (19) is the basis for the curves of amplitude severity *vs* bearing eccentricity ratio for the bearings shown in Figures 1, 2, 3, 4, and 5.

The left-hand side of Equation (19) is a dimensionless vibration severity parameter. It is the product of two ratios, i. e., static to dynamic bearing pressure, and shaft vibration amplitude to bearing clearance. It has a value defined by the quantities P, Q, and the ratio of the nonsynchronous to synchronous shaft vibration amplitudes r_2/r_1.

For a given bearing at a given eccentricity ratio, and with a circular, one-per-rev shaft orbit, the quantity, P, is a constant. For a given bearing, the value of, P, will vary as the eccentricity ratio changes. Hence, a plot can be made for each bearing of the amplitude severity parameter versus eccentricity ratio. This corresponds to the case when a nonsynchronous vibration is absent and, therefore, $r_2/r_1 = 0$.

If a nonsynchronous vibration is present along with a once-per-rev shaft orbit, both the ratio r_2/r_1, and the quantity Q have finite values. At a particular eccentricity ratio for a given bearing, the quantity Q is dependent only on the ratio of the nonsynchronous to synchronous frequency Ω_2/Ω_1. If a radius ratio r_2/r_1 is assumed, all the quantities on the right-hand side of Equation (19) are known for the given eccentricity ratio.

The process can be repeated for different eccentricity ratios. A family of curves can therefore be generated for a given bearing and for a known or assumed frequency ratio Ω_2/Ω_1. The curves in each plot represent a constant ratio of nonsynchronous to once-per-rev shaft orbit, i. e., r_2/r_1 equals the value indicated on the curve. In all cases, the top curve reflects the case when the amplitude ratio $r_2/r_1 = 0$.

This process has been carried out for the bearings shown in Figures 1, 2, 3, 4, and 5. The results are shown in Figures 8, 9, 10, 11, and 12 for the case of a half-frequency, subsynchronous vibration ($\Omega_2/\Omega_1 = 0.5$), and in Figures 13, 14, 15, 16, and 17 for the case when the nonsynchronous vibration is twice that of the shaft rotational frequency ($\Omega_2/\Omega_1 = 2$). The top curve for each bearing, however, applies to the purely synchronous case, when the amplitude ratio $r_2/r_1 = 0$.

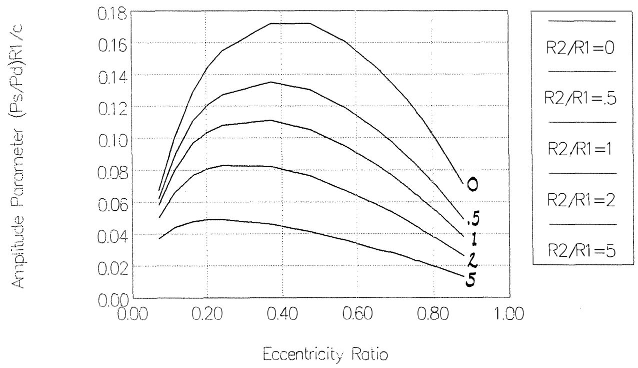

Figure 8. Amplitude Parameter vs *Eccentricity Ratio for Two Axial Groove bearing, L/D = 0.5, and Combined One-Per-Rev, and Half-Per-Rev Circular Orbits.*

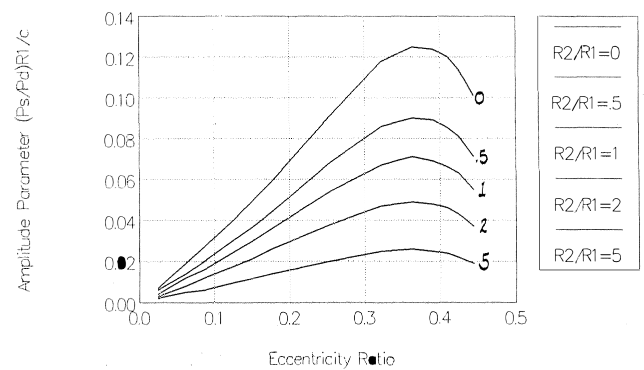

Figure 9. Amplitude Parameter vs *Eccentricity Ratio for Elliptical Bearing, 50 percent Preload, L/D 0.5, and Combined One-Per-Rev and Half-Per-Rev Circular Orbits.*

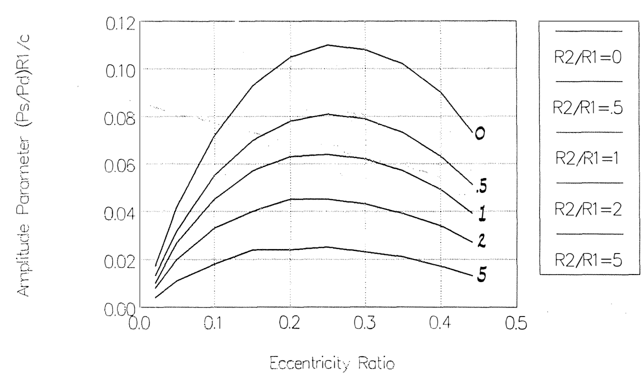

Figure 10. Amplitude Parameter vs *Eccentricity Ratio for Three Lobe Bearing, 50 Percent Preload, L/D 0.5, and Combined One-Per-Rev and Half-Per-Rev Circular Orbits.*

The use of these curves can perhaps be best illustrated by examples:

Example 1: For a two axial groove bearing, Figure 1, find the maximum, one-per-rev shaft vibration permitted, as a fraction of the bearing clearance, if the dynamic load, P_d, is not to exceed 400 psi, and the following conditions exist:

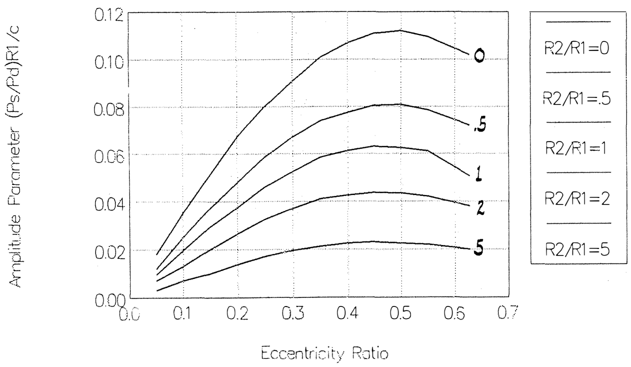

Figure 11. Amplitude Parameter vs *Eccentricity Ratio for Offset Cylindrical Bearing, 50 Percent Preload, and Combined One-Per-Rev and Half-Per-Rev Circular Orbits.*

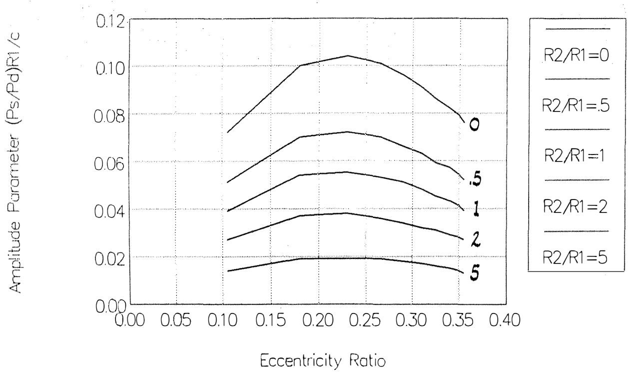

Figure 12. Amplitude Parameter vs *Eccentricity Ratio for Tilt-Pad Journal Bearing, 50 Percent Preload, on Pad Load, and Combined One-Per-Rev and Half-Per-Rev Circular Orbits.*

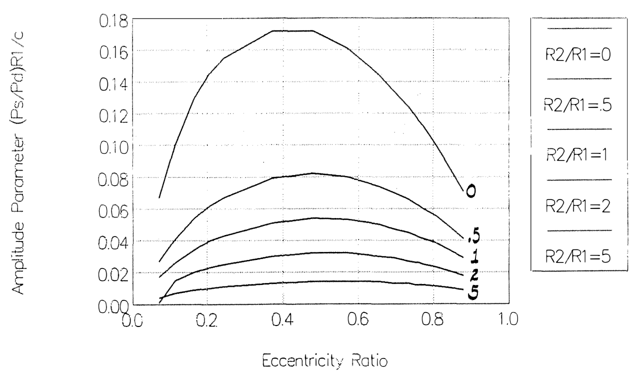

Figure 13. Amplitude Parameter vs *Eccentricity Ratio for Two Axial Groove Bearing, L/D 0.5, and Combined One-Per-Rev and Two-Per-Rev Circular Orbits.*

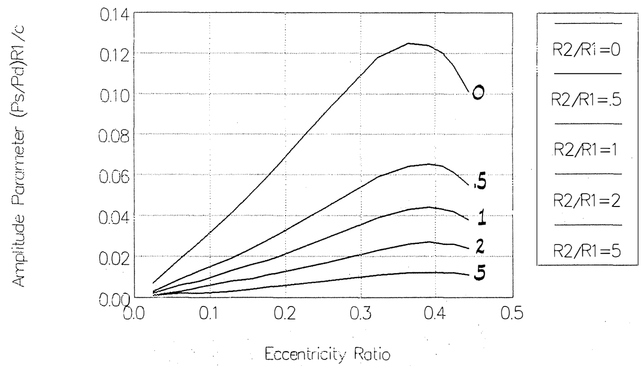

Figure 14. Amplitude Parameter vs *Eccentricity Ratio for Elliptical Bearing, 50 Percent Preload, L/D 0.5, and Combined One-Per-Rev and Two-Per-Rev Circular Orbits.*

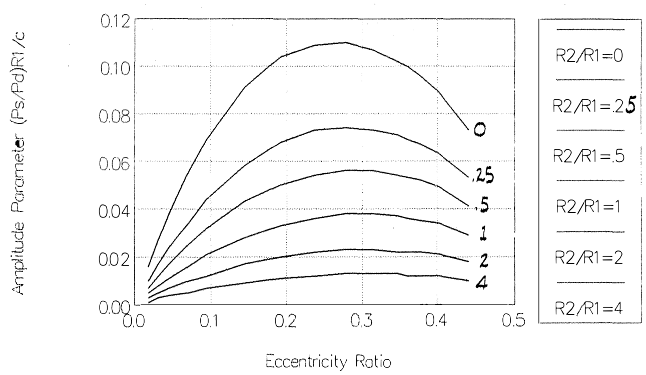

Figure 15. Amplitude Parameter vs *Eccentricity Ratio for Three Lobe Bearing, 50 Percent Preload, L/D 0.5, and Combined One-Per-Rev and Two-Per-Rev Circular Orbits.*

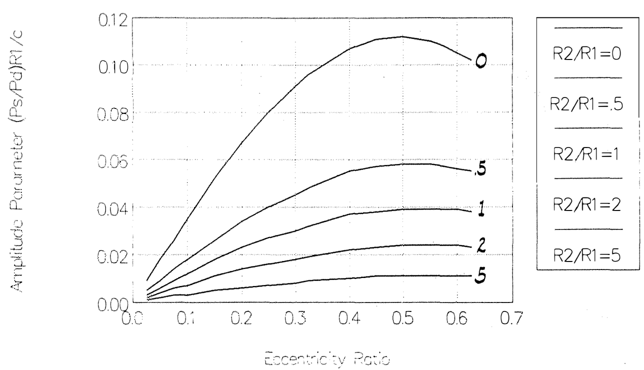

Figure 16. Amplitude Parameter vs *Eccentricity Ratio for Offset Cylindrical Bearing, 50 Percent Preload, L/D 0.5, and Combined One-Per-Rev and Two-Per-Rev Circular Orbits.*

eccentricity ratio = 0.6

static bearing pressure P_s = 400 psi.

From Figure 8 at an eccentricity ratio of 0.6, and $r_2/r_1 = 0$

$$\frac{P_s\, r_1}{P_d\, c} = 0.16 \quad , \quad P_s/P_d = 1, \text{ hence, } r_1/c = 0.16$$

Example 2: For an elliptical bearing, Figure 2, estimate the maximum dynamic load P_d for the case when:

r_1 = 1 mil 0-peak, one-per-rev
Ω_2/Ω_1 = 0.5 half frequency vibration
c = 10 mils bearing radial clearance
r_2 = 1 mil o-peak, sub-synchronous vibration
P_s = 140 psi static bearing pressure
E = 0.35 eccentricity ratio

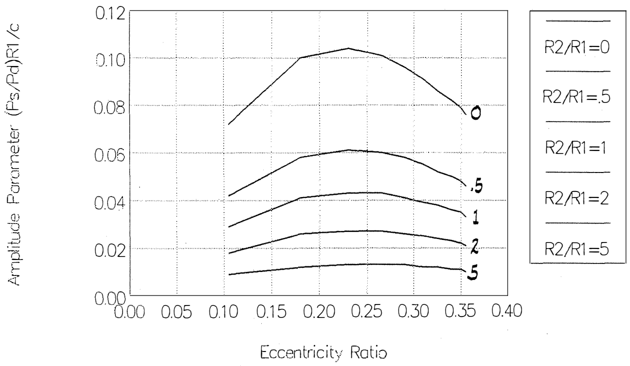

Figure 17. Amplitude Parameter vs *Eccentricity Ratio for Tilt-pad Journal Bearing, 50 Percent Preload, on Pad Load, and Combined One-Per-Rev and Two-Per-Rev Shaft Circular Orbits.*

From Figure 9, at $E=0.35$ and $r_2/r_1=1$

$$\frac{P_s\, r_1}{P_d\, c}=0.07$$

Calculate $P_d=(P_s/0.07)\,(r_1/c)=200$ psi

SIMPLIFIED VIBRATION CRITERION FOR ONE-PER-REV SHAFT ORBITS

As shown in Figures 8, 9, 10, 11, 12, 13, 14, 15, 16, and 17, the amplitude severity parameter varies with the eccentricity ratio at which the journal is operating in the particular bearing. If the turbomachine operates at a fixed set of conditions, with a resulting fixed eccentricity ratio, the amplitude severity parameter may be determined from the applicable curve. The allowable vibration for a given dynamic load may then be established as shown by the prior examples.

If the machine operates over a wide range of conditions, the amplitude severity parameter will vary. In principle, the parameter could be evaluated for each bearing under each new set of conditions. As an alternative, however, it can be convenient to consider a single representative value for this parameter. This can lead to a simplified expression for allowable shaft vibration as a function of the bearing's static load, allowable dynamic load, and bearing clearance.

If the value selected for this parameter is representative of that existing among different bearing types, then a simple, single expression can cover a variety of situations. Vibration levels may be set easily for different machines with different bearings, and operating under variable conditions. Some accuracy is sacrificed for the sake of simplicity, but there still exists a rational basis for the vibration levels.

Summarized data are presented in Table 1 for the amplitude severity parameter for all the bearings of Figures 1, 2, 3, 4, and 5, from which a representative value may be selected.

In order to compare the values for the non-preloaded, axial groove bearing with that of the other bearings which have a 50 percent preload, the values for the parameter must be placed on a consistent basis. This can be done by expressing the value of the bearing lobe clearance c in terms of the assembled radial clearance c′ and the preload factor δ. Thus,

$$c'=c(1-\delta) \quad (20)$$

and for a bearing with a 50 percent preload factor, i.e., $\delta=0.5$,

Table 1. Comparison of Average and Maximum Amplitude Severity Parameters for Five Bearing Types.

Bearing Description		Amplitude Severity Parameter $(P_s/P_d)\,(r_1/c')=K/(1-\delta)$		
Fig. No.	Type of bearing	L/D	Average	Maximum
1	2 Axial Groove	0.5	0.143	0.172
		1.0	0.140	0.181
2	Elliptical	0.5	0.157	0.250
		1.0	0.171	0.276
3	3 Lobe	0.5	0.175	0.22
		1.0	0.178	0.22
4	Offset Cylindrical	0.5	0.173	0.224
		1.0	0.183	0.238
5	5 Pad, Tilt Pad on Pad Load	0.5	0.179	0.208

$$c'=c(1-0.5)=0.5c$$

In figures (8-16) the amplitude severity parameter is expressed in terms of the lobe clearance c, i.e.,

$$\frac{P_s\, r_1}{P_d\, c}=K \quad (21)$$

and substituting for c in equation (21) from eg (20), we obtain:

$$\frac{P_s\, r_1(1-\delta)}{P_d\, c'}=K$$

or

$$\frac{P_s\, r_1}{P_d\, c'}=\frac{K}{(1-\delta)} \quad (22)$$

The amplitude parameters for the indicated bearings on the basis of the assembled clearance c′ are compared in Table 1. Thus, the values for the amplitude parameter of the curves of the bearings with 50 percent preload have been multiplied by 2, as required by Equation (22). Values for both the average and maximum severity are shown in the table for each of the bearings considered herein. Considering the broad geometrical differences among the five bearing types, it will be seen that the values of the severity parameter are not greatly different among them.

The average value among the five bearing types is 0.167 for the average parameter, and 0.221 for the maximum. Assuming a representative value of 0.2 for the severity parameter, based on assembled clearance c′, then

$$\frac{P_s\, r_1}{P_d\, c'}=0.20 \quad (23)$$

or the ratio of allowable vibration to bearing clearance is therefore given as:

$$\frac{r_1}{c'_1}=0.20\,\frac{P_d}{P_s} \quad (24)$$

where clearly the ratio of P_d/P_s is restricted to values less than 5.

Within this restriction, the double amplitude of vibration d = 2r in terms of the total assembled clearance 2c′ is therefore:

$$d' = 0.20 \frac{P_d}{P_s} \cdot (2c')$$

If the level of dynamic load, P_d, is to be kept at one-half the static load, P_s, the allowable vibration would be 10 percent of the diametral assembled clearance.

In order to set limits for vibration, the corresponding values of dynamic load must be set. As discussed in a previous publication [7] a trip level for P_d is the value at which babbitt fatigue damage may occur. A maximum dynamic load of 500 psi is suggested for the trip level, 250 psi for the alarm level, and 125 psi for the acceptable level.

Shaft vibration measurements often include the effect of mechanical and electrical runout due, for example, to a noncircular probe target surface, and measurements at other than the mid-plane of the bearing. To provide an allowance for this effect, a value of 1 mil is added to the acceptable level based on a dynamic load of 125 psi.

The simplified equations for classifying shaft vibration severity are then given in mils of vibration (peak to peak) as:

$$d' = \frac{100}{P_s}(2c') \quad \text{trip level} \qquad (26)$$

$$d' = \frac{50}{P_s}(2c') \quad \text{alarm level} \qquad (27)$$

$$d' = \frac{25}{P_s}(2c')+1 \quad \text{acceptable level} \qquad (28)$$

SUMMARY AND CONCLUSIONS

The dynamic load imposed on a fluid film bearing can be related to the size and shape of the shaft orbit and the spring and damping coefficients of the bearing film. For the case when the shaft orbit is circular, the equations are simplified. A dimensionless vibration severity parameter can be plotted for a given bearing geometry against eccentricity ratio ranges. The maximum dynamic load can be computed easily from the severity parameter value, the static load on the bearing, and the bearing clearance.

The analysis can be extended to include nonsynchronous journal orbits as well as one-per-rev, circular orbits. Curves are presented for five types of bearings which can be used to estimate dynamic loads for a journal operating with a combined synchronous and half-per-rev or two-per-rev circular orbit.

If geometrically similar bearings are operated at identical eccentricity ratios, the allowable shaft vibration (for the same limiting dynamic pressure) will be proportional to bearing size. Larger vibrations will be permitted for larger machines.

Simplified equations are proposed for estimating the allowable shaft vibrations at acceptable, alarm, and trip levels, based on dynamic bearing unit load.

NOMENCLATURE

a, b,	major and minor axes of ellipses (Figure 7)
$\bar{A}, \bar{B}, \bar{C}, \bar{D}$,	dimensionless coefficients, Equations (2-5) and Equations (12-15)
$\bar{B}_{xx}, \bar{B}_{yy}, \bar{B}_{yx}, \bar{B}_{xy}$	dimensionless direct and cross-coupled damping coefficients (Figure 6) $\bar{B}_{xx} = \Omega B_{xx} c/W$, etc.
c	bearing radial lobe clearance, i. e., the difference between the lobe radius and the shaft radius in Figures (1-5)
c′	bearing radial assembled clearance, i. e., the minimum gap between the bearing and the centered shaft
D	journal diameter
F_x, F_y	amplitude of dynamic force transmitted through the bearing film in the x, y directions
F_r	radial force resultant on the bearing
$\bar{K}_{xx}, \bar{K}_{yy}, \bar{K}_{yx}, \bar{K}_{xy}$	dimensionless bearing direct and cross-coupled spring coefficients (Figure 6) $\bar{K}_{xx} = K_{xx} c/W$, etc.
K	value of dimensionless amplitude severity parameter, equation (21)
L	bearing axial length
P_s	static, average bearing pressure = W/DL
P_d	alternating dynamic bearing pressure $(F_r)_{max}$/DL
r_1, r_2	radial amplitude of synchronous and non-synchronous shaft circular orbits, respectively
W	static load on bearing
α	orientation angle of elliptical shaft orbit (Figure 7)
δ	bearing preload ratio d/c (Figures 1, 2, 3, 4, 5)
d	distance between bearing lobe center and axis of centered shaft, Figures 1, 2, 3, 4, 5, for determining preload ratio δ
d′	peak to peak shaft vibrational amplitude, equations (25 to 28)
Ω_1, Ω_2	synchronous and non-synchronous shaft orbit frequencies, respectively

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