Parametric Study of the Unbalance Response of an Eccentric Squeeze Film Damper-Mounted Rigid Rotor*

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This paper presents a limited parametric study of the unbalanced response of a rigid rotor supported by an eccentric squeeze film damper bearing. The rotor unbalance response is approximated by a trigonometric series whose coefficients are determined by the collocation method. The results indicate that for small static misalignments and unbalances, nonsynchronous components are negligible and the rotor response is an offset ellipse. Nonsynchronous components increase as both the unbalance and the static misalignment are increased. However, the amplitude of the rotor response is independent of the static misalignment direction. There is a tendency for the rotor to self-center in the vicinity of the resonance. This self-centering effect is more pronounced as the unbalance is increased and it is reduced as the static misalignment is increased. For large unbalances and static misalignments, both main resonant jump and secondary resonant jump are predicted at speeds above twice the system critical speed.

Key Words: Squeeze Film Damper, Eccentric Operation, Trigonometric Series, Predictor-Corrector, Subharmonic and Superharmonic Components, Static Misalignment, Jump Phenomena, Main and Secondary Resonant Jump

1. Introduction

Squeeze film dampers have found wide applications in high-speed rotating machinery to attenuate resonant vibration and to improve the stability of rotor bearing systems^{(1),(2)}. A squeeze film damper can be designed such that the journal can find its own static position within the clearance circle or can be held centrally within the clearance by retaining spings which prevent meatal-to-metal contact (with subsequent wear) during start-up and run-down.

Cookson and Kossa⁽³⁾, using a direct numerical integration scheme, investigated the effectiveness of squeeze film dampers without retaining springs in reducing the rotor unbalance response. For squeeze film dampers with retaining springs, most of the previous studies have been based on the assumption of a centered, synchronous and circular orbit^{(4),(5)}, which only exists under idealized situations. In practice,

retaining springs are often used to support static loads and centralizing mechanisms are not always used for constructional simplicity.

Tonnesen⁽⁶⁾ experimentally studied the unbalance response of an eccentric squeeze film damper-mounted rigid rotor. He observed that the rotor followed an offset synchronous and elliptic orbit at low speeds and there was a tendency of self-centering and circulating of the orbit as the rotor speed and the unbalance were increased. Sykes and Holmes⁽⁷⁾ experimentally investigated the effect of the damper static misalignment on the unbalance response of a damper-mounted rigid rotor. They observed severe subharmonic motions at speeds above twice the system critical speed and they attributed the subharmonic motions to the damper static misalignment. For a better understanding of the eccentric operation of squeeze film dampers and the nature of the experimentally observed subharmonic motions⁽⁷⁾, it is necessary to investigate the effect of various system parameters on the system response.

Direct numerical integration has been commonly used to analyze the nonsynchronous and noncircular rotor response but it proves costly in terms of

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computing time. Pan and Tonnesen⁽⁸⁾ theoretically studied the eccentric operation of a squeeze film damper by treating nonsynchronous vibration components as perturbations of the synchronous component. Li and Taylor⁽⁹⁾, using a perturbation method, studied the effect of the damper static radial misalignment on the unbalanced response of a squeeze film damper-mounted rigid rotor. They concluded that the rotor motion is synchronous and circular for small static misalignments and the self-centering phenomenon only occurs below the critical speed. Nataraj and Nelson⁽¹⁰⁾ introduced the trigonometric collocation method for the study of nonlinear dynamic systems. Zhao et al.⁽¹¹⁾ combined the trigonometric collocation method with the arc-length continuation method for studying the eccentric squeeze film-damped rotor bearing systems with the stability of the approximate solutions determined by the Floquet transition matrix method.

It is the purpose of this paper to present the results of a limited parametric study of the unbalance response of an eccentric squeeze film damper-mounted rigid rotor. Solutions are approximated by a trigonometric series whose coefficients are determined by the collocation method⁽¹⁰⁾ together with the arc-length continuation method⁽¹¹⁾. The damper forces are determined using an incompressible lubricant model, a short bearing approximation and the ' π ' cavitation models. This analysis focuses on the effects of the damper static radial misalignment, the unbalance and the bearing parameter on the unbalance response.

Nomenclature

$a_{x0}, a_{x1}, a_{x2}, \dots, b_{x1}, b_{x2}, \dots; a_{y0}, a_{y1}, a_{y2}, \dots, b_{y1}, b_{y2}$: coefficients in the trigonometric series
 $AH(i)$: semi-major axis of i 'th harmonic component
 B : bearing parameter $= \mu RL^3 / m \omega_n c^3$
 $BH(i)$: semi-minor axis of i 'th harmonic component
 c : radial clearance of damper $= R - r$
 d : viscous damping coefficient
 D : $d / m \omega_n$
 e : eccentricity of rotor vibration $= \varepsilon c$
 F_r, F_t : nondimensional damper forces
 f_r, f_t : nonlinear damper forces in the radial, tangential directions
 $h(i)$: vector of real values
 k : retaining spring stiffness
 L : damper length; $\zeta = z/L$
 m : mass supported by damper
 N : order of the trigonometric series
 P : pressure in the damper
 O_b, O_j : geometric centers of bearing and journal
 P_v : minimum film pressure
 r, t : radial and tangential directions

s : ω / ω_n
 r, R : radii of journal and damper, respectively
 t : time; $\tau = \omega t$
 u : mass eccentricity
 U : u/c
 x, y, z : horizontal, vertical and axial coordinates
 x_0, y_0 : components of damper static eccentricity $X, Y = x/c, y/c$
 X_0, Y_0 : $x_0/c, y_0/c$
 α : phase angle of the synchronous vibration
 ε : eccentricity ratio $= \sqrt{X^2 + Y^2}$
 ε_0 : static eccentricity ratio $= \sqrt{X_0^2 + Y_0^2}$
 $\varepsilon_{\max}$: maximum eccentricity ratio
 $\gamma(i)$: ellipticity of i 'th ellipse $= BH(i)/AH(i)$
 θ : angular position (Figure 1)
 μ : dynamic viscosity
 φ : angular displacement of line $O_b O_j$
 $(\cdot)$ & $(')$: derivatives with respect to t & τ , respectively
 ω : rotor angular speed; $\omega_n = \sqrt{k/m}$

2. Approximate Analytical Solutions

Figure 1 schematically depicts a rigid rotor supported by a squeeze film damper in parallel with retaining springs. The following assumptions are made for this system :

- the retaining springs are radially symmetric ;
- the lubricant is incompressible and the Reynolds equation for constant lubricant properties is applicable ;
- the short bearing approximation is valid ; and
- the rotor speed is constant.

The nondimensionalized equations of rotor motion are :

Fig. 1 Cross section of a rigid rotor supported by a squeeze film damper

$$X'' + \frac{DX'}{s} + \frac{X}{s^2} = U \cos \tau + \frac{B}{s} \frac{(XF_r - YF_t)}{\varepsilon} + \frac{X_0}{s^2}, \quad (1)$$

$$Y'' + \frac{DY'}{s} + \frac{Y}{s^2} = U \sin \tau + \frac{B}{s} \frac{(XF_t + YF_r)}{\varepsilon} + \frac{Y_0}{s^2}. \quad (2)$$

Using the short bearing approximation and assuming supply and return pressures are negligible, the pressure distribution in the damper is given as:

$$P = \frac{6\mu\omega L^2 \zeta (\zeta - 1) (\varepsilon' \cos \theta + \varphi' \varepsilon \sin \theta)}{c^2 (1 + \varepsilon \cos \theta)^3}. \quad (3)$$

The resulting damper forces in the radial and tangential directions were determined by a two-dimensional integration at each step of nonlinear iterations, viz.:

$$(F_r, F_t) = \frac{c^2}{\mu L^2} \int_0^{2\pi} \int_0^1 P(\cos \theta, \sin \theta) d\zeta d\theta. \quad (4)$$

Various film extents were made possible by setting a film rupture pressure, P_v , such that if $P < P_v$, then $P = P_v$. For the ' π ' film model $P_v = 100$ kPa (absolute).

The steady-state rotor displacements in the X and Y directions were approximated by a trigonometric series, viz.:

$$X = a_{x0} + \sum_{i=1}^N a_{xi} \cos h(i)\tau + b_{xi} \sin h(i)\tau, \quad (5)$$

$$Y = a_{y0} + \sum_{i=1}^N a_{yi} \cos h(i)\tau + b_{yi} \sin h(i)\tau, \quad (6)$$

whose coefficients were determined by the collocation method⁽¹⁰⁾ together with the arc-length incremental continuation algorithm with initial values for nonlinear iterations being provided by a polynomial predictor-corrector⁽¹¹⁾.

In the following results, the damping coefficient D

was assumed to be zero and the ' π ' film model was used. Due to the presence of the static misalignment in addition to the unbalance, the steady-state rotor motion will generally be noncircular and nonsynchronous. For the parameters used, the rotor motion was approximated by a trigonometric series of $\{h\} = \{0.5, 1.0, 1.5, 2.0\}^T$ with $L = 2 \times 4 + 1 = 9$. Each of the frequency components in the trigonometric series forms an offset ellipse with a common offset of (a_{x0}, a_{y0}) and with principal axes rotated relative to the fixed frame of reference as shown in Fig. 2⁽¹²⁾. The lengths $AH(i)$ and $BH(i)$ of the semi-major and semi-minor axes are

$$\begin{aligned} AH(i)^2 &= 0.5(a_{xi}^2 + b_{xi}^2 + a_{yi}^2 + b_{yi}^2) \\ &\quad + \sqrt{0.25[a_{xi}^2 + b_{xi}^2 - a_{yi}^2 - b_{yi}^2]^2 + [a_{xi}a_{yi} + b_{xi}b_{yi}]^2}, \\ BH(i)^2 &= 0.5(a_{xi}^2 + b_{xi}^2 + a_{yi}^2 + b_{yi}^2) \\ &\quad - \sqrt{0.25[a_{xi}^2 + b_{xi}^2 - a_{yi}^2 - b_{yi}^2]^2 + [a_{xi}a_{yi} + b_{xi}b_{yi}]^2}, \end{aligned}$$

and the phase lag angle of the synchronous vibration with respect to the unbalance force is

$$\tan \alpha = \frac{a_{x2} + b_{y2}}{b_{x2} - a_{y2}}.$$

3. General Rotor Motion

For a static misalignment of $(X_0, Y_0) = (0.0, -0.3)$, a bearing parameter of $B = 0.072$ and an unbalance of $U = 0.1$, calculation was first carried out to show the general effect of the static misalignment on the rotor frequency response, and the results are given in Fig. 3 (a) and 3(b). For comparison purposes, the semi-major axes of the first-order approximation for $N = 1$ and $\{h\} = \{1.0\}$ and the undamped response for $B = 0$ are also given in Fig. 3(a)(3). In Fig. 3(b), the unit circles represent the bearing clearance circle.

It can be seen that the general rotor motion is an offset, nonsynchronous and noncircular orbit. The rotor orbit center basically remains on the vertical axis of the clearance circle with a slight offset in the X direction in the region of the static resonance of $s = 1.0$. It can be noted from Fig. 3(a)(3) that the damper is very effective in attenuating vibration around this resonance, where the vibration amplitude would be infinite without the damper.

The dynamic resonance is shifted to the right of the static resonance of $s = 1.0$ due to the hardening spring effect of the damper forces. As expected, at low and high speeds, the synchronous vibration lags behind the unbalance by 0° and 180° , respectively.

The frequency response curve of the 0.5-subharmonic component has two peaks. One peak, b , is in the region of $s = 1.15$ and represents a component with a frequency of approximately 0.575 superimposed upon the synchronous response. The other peak, c , is in the region of $s = 2.3$, which indicates that the rotor response contains a frequency component of 1.15 when

Fig. 2 The i 'th offset ellipse

the forcing frequency is 2.3. This region is a 0.5-subharmonic resonance.

The frequency response curve of the first superharmonic also has two peaks. The first peak, *d*, with $s=1.15$, represents a component with a frequency of approximately 2.3 superimposed upon the synchronous response. The second peak, *e*, with $s=0.575$, represents a component at a frequency of approximately 1.15 when the forcing frequency is $s=0.575$. This region is denoted as the first superharmonic resonance.

At speeds below $s=0.4$, the unbalance force is

small compared to the static force, and the damper forces to together with the static force restrain the rotor to oscillate around its static equilibrium position. When the rotor speed approaches the dynamic resonance ($s=1.15$ for this set of parameters), the orbit is lifted towards the bearing center. Beyond this resonance, the orbit is brought back towards its static equilibrium position and then above $s=2$, is lifted up again. This implies that a self-centering phenomenon exists below the resonance and beyond a given frequency above the resonance. This self-centering phenomenon agrees with the experimental results

(a) Components of the rotor frequency response for $B=0.072$, $U=0.1$ and $(X_0, Y_0)=(0.0, -0.3)$

(b) Rotor orbits at various rotor speeds for $B=0.072$, $U=0.1$ and $(X_0, Y_0)=(0.0, -0.3)$

Fig. 3

reported in Ref. (6).

The nonsynchronous components are generally small compared with the synchronous component, and the rotor orbit is very close to a circle at low and high speeds. The greatest distortion from a circular orbit is at speeds near $s=1$.

Comparing the first-order approximation to the fourth-order approximation shown in this figure, it can be concluded that the first-order approximation is

adequate except at speeds in the vicinity of the resonance where the nonlinear iterations failed to converge. Thus, one may assume that the rotor motion is an offset ellipse at speeds distinct from the resonance at least for a small static misalignment.

4. Parametric Study

To obtain an intuitive understanding of the effect of varying the system parameters, calculations were

(a) Components of the rotor frequency response showing the effect of increasing the damper static misalignment for $B=0.072$, $U=0.1$, $X_0=0$ and $Y_0=-0.3, -0.5, -0.7$

(b) Rotor orbits showing the effect of increasing the static misalignment for $B=0.072$ and $U=0.1$

Fig. 4

performed by varying one parameter with the remaining parameters unaltered.

4.1 Effect of increasing the static misalignment

The results obtained for a small static misalignment showed that the rotor motions at speeds far from the system resonance can be considered to be offset ellipses. To establish an upper limit for this approximation, further calculations were carried out to investigate the effect of increasing the static misalignment.

The static misalignments used were $(X_0, Y_0) = (0.0, -0.5)$ and $(0.0, -0.7)$. The other system parameters remained unchanged ($B=0.072$ and $U=0.1$). The results are given in Figs. 4(a) and 4(b) together with those obtained for $(X_0, Y_0) = (0.0, -0.3)$. In Fig. 4(a), arrows point in the direction of increasing static misalignment.

It can be seen that an increase in the static misalignment will generally result in a lower orbit position with a larger offset in the X direction. The self-centering effect is reduced as the static misalignment is increased. The amplitude of nonsynchronous components, the orbital distortion from a circle and the orbit maximum eccentricity ratio will increase dramatically as the static misalignment is increased. At the same time, the synchronous component decreases except at speeds in the region of $s=1.3$ to $s=3.0$. For large static misalignments, the rotor orbits exhibit multiple loops due to the significant subharmonic components.

The above results indicate that the speed region within which offset elliptic rotor motions can be assumed becomes smaller as the static misalignment is increased.

4.2 Effect of varying the static misalignment direction

In practice, the static misalignment may be in any direction and so calculations were carried out for static misalignments of a constant magnitude but different directions. The parameters used were $B=0.07$, $U=0.1$ and eight static misalignments with a constant magnitude of 0.3 and 45° apart considered. Some of the interesting results are given in Figs. 5(a) and 5(b).

It can be seen that the rotor orbit size and the orbit shape are independent of the static misalignment direction. Although the orbit center varies with the static misalignment direction, the distance from the orbit center to the bearing center (eccentricity ratio of the orbit center), $a_0 = \sqrt{a_{x0}^2 + a_{y0}^2}$, is independent of the static misalignment direction. This indicates that the rotor orbits will be located on a circle for the static misalignments having a constant magnitude. As a consequence, the orbit maximum eccentricity

ratio is also independent of the static misalignment direction.

4.3 Effect of increasing the bearing parameter B

The bearing parameter B has a crucial effect on damper performance. Figure 6(a) shows how the components of the frequency response vary with the bearing parameter B with arrows pointing in the direction of increasing B . Figure 6(b) shows the corresponding rotor orbits. The parameters used were $U=0.1$ and $(X_0, Y_0) = (0.0, -0.3)$ for three B values of 0.05, 0.072 and 0.1.

It can be seen that when the bearing parameter B is increased, the self-centering effect is reduced at speeds near the resonance, but it is increased elsewhere. An increase in the bearing parameter B will also result in a decrease in the magnitudes of all the frequency components and consequently a decrease in the maximum eccentricity ratio. Thus, an increase in the bearing parameter has a beneficial effect on the damper performance, until for very large bearing parameters, the damper is effectively locked up and begins to behave as a rigid support. There is a trade-off between minimizing the amplitude of resonant vibration and reducing the force being transmitted to the foundation at normal operating speeds above the resonance.

4.4 Effect of increasing the unbalance U

Figure 7(a) shows the effect of the unbalance U on the components of the frequency response with arrows pointing in the direction of increasing U . The corresponding rotor orbits are given in Fig. 7(b). The three unbalances used were $U=0.05$, 0.1 and 0.15 with $B=0.072$ and $(X_0, Y_0) = (0.0, -0.3)$.

It can be seen that an increase in the unbalance U results in an increase in the centralizing force and an increase in the offset in the X direction. Depending on the amount of the static misalignment, a minimum value of unbalance is required to lift the rotor from its static equilibrium position. For a sufficient large unbalance, centralized rotor motion will occur at speeds near the resonance.

As may be expected, an increase in the unbalance U increases the orbit size, as shown in Fig. 3(b). Both the synchronous and nonsynchronous components are increased, especially the subharmonic component at speeds near the subharmonic resonance of $s=2.3$, where the rotor orbit exhibits multiple loops. An increase in the unbalance U also results in a further shift of the resonance to the right due to the increased hardening spring effect of the damper forces at larger eccentricity ratios. The resonant shift to the right can be clearly seen from the phase angle of Fig. 7(a)(5).

5. Jump Phenomenon

The jump phenomenon observed in centralized damper operations has been attributed to the hardening spring effect of cavitated dampers at high eccentricity ratios. For eccentric damper operations, the presence of a static misalignment increases the non-linearity of the damper forces and introduces a radial asymmetry in the pressure profiles, and consequently

in the damper forces. Thus, one would expect dynamic behaviours which are more complex than those of centralized damper operations.

5.1 Main resonant jump

For an unbalance of $U=0.1$ and a bearing parameter of $B=0.072$, calculations were performed for static misalignments up to $\varepsilon_0=0.8$, and no jump was predicted. For a large unbalance of $U=0.3$, a bearing parameter of $B=0.072$ and a static misalignment of

(a) Components of the rotor frequency response showing the effect of varying damper static misalignment direction for $B=0.072$, $U=0.1$ and $\varepsilon_0=0.3$

(b) Rotor orbits showing the effect of varying the static misalignment direction for $B=0.072$, $U=0.1$ and $\varepsilon_0=0.3$

Fig. 5

$(X_0, Y_0) = (0.0, -0.2)$, multiple solutions were predicted in all the frequency components, as shown in Fig. 8(a). Figure 8(b) shows some of the orbits in the regions of multiple solutions with 'lower', 'middle' and 'upper' denoting the orbits on the lower, middle and upper branches of Fig. 8(a)(3) respectively.

During run-up, the large rotor orbit near resonance will be reduced suddenly by a jump down to a smaller orbit at $s_A = 2.175$. During run-down, the

small rotor orbit will be increased suddenly by a jump up to a larger orbit at $s_B = 1.622$. The difference in the jump speeds is the effective region of multiple solutions, and this difference in the jump speeds is due to the hysteretical behaviour of the system.

These jumps involve changes not only in the orbit size but also in the orbit shape and the relative orbit position in the clearance circle. Three possible solutions also exist in the speed range of $s = 4.642$ to $s = 10$,

(a) Components of the rotor frequency response showing the effect of increasing the bearing parameter for $B = 0.05, 0.072, 0.1$, $U = 0.1$ and $(X_0, Y_0) = (0.0, -0.3)$

(b) Rotor orbits showing the effects of increasing the bearing parameter B for $B = 0.05, 0.072, 0.1$, $U = 0.1$ and $(X_0, Y_0) = (0.0, -0.3)$

Fig. 6

where for a solution on the upper branch, a jump down on run-down will occur at $s_c=4.642$. From Fig. 8(a) it is seen that the solution on the upper branch in the second multiple-solution region can be considered as centralized, circular and synchronous.

A jump down is characterized by a jump in the phase angle of the synchronous vibration from about 110° to 180° . A jump up is characterized by a jump in the phase angle of the synchronous vibration from 180° to below 90° in the first jump region and to 100° in

the second jump region.

This jump phenomenon is referred to as the main (principal) resonant jump. For static misalignments up to $\varepsilon_0=0.5$, it was found that the jump speeds do not vary greatly with the static misalignment. Like that of centralized damper operations, the jump phenomenon reported here is due to the hardening spring effect at high eccentricity ratios and depends on the unbalance and the bearing parameter. However, it would be very difficult to locate the critical values of

(a) Components of the rotor frequency response showing the effect of increasing the unbalance U for $B=0.072$, $U=0.05, 0.1, 0.2$ and $(X_0, Y_0)=(0, 0, -0.3)$

(b) Rotor orbits showing the effect of increasing the unbalance U for $B=0.072$, $U=0.05, 0.1, 0.2$ and $(X_0, Y_0)=(0, 0, -0.3)$

Fig. 7

unbalance and bearing parameter due to the large amount of computational time involved.

5.2 Secondary resonant jump

When both the unbalance and the static misalignment are large enough, main and secondary resonant jumps are predicted as shown in Fig. 9(a) for $B=0.072$, $U=0.3$ and $(X_0, Y_0)=(0.0, -0.3)$. This figure exhibits the main resonant jump as discussed above, but it also exhibits a secondary resonant jump at speeds in the vicinity of the 0.5-subharmonic resonance.

There also exist three possible solutions in the

secondary resonant jump region. Of the three possible solutions there are significant differences in the orbit size, the orbit shape and the orbit position in the clearance circle, as shown in Fig. 9(b) with 'lower', 'middle' and 'upper' denoting the orbits on the lower, middle and upper branches, respectively. As shown in Fig. 9(a), the nonlinear iteration failed to converge as the speed was further increased from point U .

The secondary resonant jump phenomenon also has a hysteretical property with jump-up and jump-down occurring at different speeds. This secondary resonant jump is attributed to the large static

(a) Components of the rotor frequency response showing the main resonant jump for $B=0.072$, $U=0.3$ and $(X_0, Y_0)=(0.0, -0.2)$

(b) Rotor orbits showing three possible solutions at one rotor speed $B=0.072$, $U=0.3$ and $(X_0, Y_0)=(0.0, -0.2)$

Fig. 8

(a) Components of the rotor frequency response showing the main and secondary resonant jumps for $B=0.072$, $U=0.3$ and $(X_0, Y_0)=(0,0,-0.3)$

(b) Rotor orbits showing the secondary resonant jump phenomena for $B=0.072$, $U=0.3$ and $(X_0, Y_0)=(0,0,-0.3)$

Fig. 9

misalignment producing highly nonlinear and asymmetrical damper forces, and to the large unbalance which excites large-amplitude rotor oscillation. In the secondary resonant jump region, the 0.5-subharmonic component is significant, and the neglect of the subharmonic component in the trigonometric series will produce misleading results.

6. Conclusions

The results presented in this report offer a further

understanding of the dynamic behaviour of a rigid rotor incorporating an eccentrically operated squeeze film damper. Design objectives would include the avoidance of the main and secondary resonant jumps, the minimization of the rotor maximum eccentricity ratio over the operating speed range and the minimization of the nonsynchronous components. For the range of parameters investigated, the following conclusions can be drawn:

The periodic frequency response of a rigid rotor

supported by an eccentric squeeze film damper is offset, nonsynchronous and noncircular. Inadequate choice of the system parameters may result in occurrence of both the main and secondary jumps and significant undesirable nonsynchronous vibrations.

The rotor motions can be considered as offset ellipses at speeds far from the resonance. As static misalignment is increased, the speed range within which rotor motion can be assumed to be an offset ellipse is reduced. An increase in the static misalignment will result in a large offset in the misalignment direction and an increase in the system nonlinearity, and consequently large nonsynchronous orbits will occur. The rotor orbit size, orbit shape, orbit maximum eccentricity ratio and the eccentricity ratio of the orbit center are independent of the static misalignment direction. An increase in the unbalance will increase all the frequency components, resulting in orbits of increased magnitude.

A self-centering phenomenon occurs below the resonance and beyond a given speed above the resonance. This self-centering phenomenon depends on the relative magnitude of the unbalance and the static misalignment. For large static misalignments and small unbalances, centralized rotor motion may never occur.

A main resonant jump was predicted for an eccentric damper operation with a static misalignment of $\epsilon_0 = 0.2$, an unbalance of $U = 0.3$, and a bearing parameter of $B = 0.072$. This jump phenomenon is more complicated than that of centralized damper operation since it involves changes not only in the rotor orbit size but also in the orbit position, orbit shape and orbit orientation. However, the underlying mechanism is the same as the hardening spring effect of cavitated squeeze film dampers.

A secondary resonant jump phenomenon was predicted as a result of the combination of a large static misalignment and a large unbalance. In the region of this secondary resonant jump, the subharmonic component is significant, and to avoid the dangerous subharmonic vibration, the static misalignment must be closely controlled.

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