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STABILITY OF INTERSHAFT SQUEEZE FILM DAMPERS

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Intershaft squeeze film dampers have been investigated for damping of dual rotor aircraft jet engines. Initial investigations indicated that the intershaft dampers would attenuate the amplitude of the engine vibration and decrease the force transmitted through the intershaft bearing, thereby increasing its life. Also it was thought that the intershaft damper would enhance the stability of the rotor-bearing system. Unfortunately, it was determined both theoretically and experimentally that the intershaft squeeze film damper was unstable above the engine's first critical speed. In this paper, a stability analysis of rotors incorporating intershaft squeeze film dampers is performed. A rotor model consisting of two Jeffcott rotors with two intershaft squeeze film dampers is investigated. Examining the system characteristic equation for the conditions at which the roots indicate an ever growing unstable motion, result in the stability conditions. The cause of the instability is identified as the rotation of the oil in the damper clearance. The oil rotation adds energy to the forward whirl of the rotor system above the critical speed and thus causes the instability. Below the critical speed the oil film removes energy from the forward rotor whirl. It is also shown that the backward whirl of the rotor system is always stable. Several proposed configurations of intershaft squeeze film dampers are discussed, and it is shown that the intershaft dampers are stable supercritically only with a configuration in which the oil film does not rotate.

INTRODUCTION

The ability of a squeeze film damper to attenuate the amplitudes of engine vibration and to decrease the dynamic forces transmitted to the frame of gas turbine engines, make it an attractive rotor support. In a single spool application, the squeeze film damper consists of an oil film in an annulus surrounding a rolling element bearing whose outer race is constrained from rotating, usually by a squirrel cage. Thus the spinning of the rotor does not reach the oil, and only when the rotor whirls does the oil film act to damp the motion.

In some applications it may be necessary to use intershaft differential rolling element bearings in a two-spool engine configuration. The use of the intershaft bearing reduces the rub between the two rotors, decreases the shaft deflections, and eliminates the static support structure in the aerodynamic flow path. Thus the engine's performance, efficiency and reliability are improved. But the intershaft bearing may carry a large load, thus decreasing the life of the bearing (ref. 1), and will also provide a direct path for the transmission of vibration between the inner and outer rotors.

Hibner *et. al.*, in reference 2, proposed using a squeeze film damper in an intershaft configuration to decrease both the intershaft bearing loading and the vibration of the engine. They proposed a damper configuration in which the oil film rotates at the speed of the inner (LP) rotor. They found out that, for their specific engine, by using the intershaft squeeze film damper, the intershaft bearing loads could be reduced to less than half the level achieved by the incorporation of squeeze film dampers on the other bearings of the rotor system. Furthermore, the cyclic stresses on the bearing supports and the engine mount structure were also substantially reduced (ref. 2). However, they also found out that the engine became unstable above its first critical speed. They were able to demonstrate both theoretically and experimentally that this instability is due to the intershaft squeeze film damper. To control this instability, they had to increase the engine's critical speed, and they proposed adding a spring in parallel to the damper to achieve this goal (ref. 3).

Courage (ref. 4) did an experimental study of an intershaft squeeze film damper which was quite similar to that of Hibner *et. al.* (ref. 2,3), i.e. the oil film rotated at the speed of the inner (LP) rotor. He mentioned that another possible damper configuration would be one in which the oil film rotates at the speed of the outer (HP) rotor. His experimental rig was also unstable above the first critical speed. He suggested, as a solution to the instability problem, a damper configuration in which there are two squeeze films, one rotating at the speed of the inner (LP) rotor, and one rotating at the speed of the outer (HP) rotor, but he did not determine the feasibility of such a design.

Recently, Li *et. al.* (ref. 5,6) used more advanced numerical methods (namely the transfer matrix method and the component mode synthesis method) to determine the steady state and transient responses of a dual rotor system incorporating intershaft squeeze film dampers. The oil film in the intershaft damper they studied was also rotating at the speed of the inner (LP) rotor. They also were able to determine that the system became unstable above the engine's critical speed. They suggested that the operating speed of the engine should be below this critical speed. Their results indicated that, below the critical speed, the use of an intershaft squeeze film damper in a rotor system is effective in reducing amplitude response and bearing loading, especially for the bearing loading at the location of the intershaft damper and adjacent rotor bearing supports.

In reference 7, Alderson conducted an experimental investigation on a two spool rig with an intershaft squeeze film damper. The oil film in his damper was also rotating at the speed of the inner (LP) rotor. He also determined that the system was unstable above the first critical speed. He was able to show that this unstable whirl was nonsynchronous with the speed of any of the rotors, and that the instability was driven by destabilizing hydrodynamic forces in the intershaft damper that develop when the system tends to whirl at a speed less than the damper spin speed. To remove the instability, he replaced the intershaft squeeze film damper with a spring cage.

From the above discussion, one may conclude that intershaft squeeze film dampers are desirable in high performance dual rotor turbomachinery, if their inherent instability above the engine's first critical speed could be overcome. In this paper, we study the stability of two Jeffcott rotors with two intershaft squeeze film dampers. It will be shown that the instability of the intershaft squeeze film damper is caused by the rotation of the oil in the previously proposed damper configurations. If the oil is rotating and the rotor is whirling, then the rotating oil will tend to damp the whirling motion below the engine's critical speed. Above the critical speed the rotating oil tends to enhance the forward whirl of the rotor thus causing the instability. A new design (ref. 8,9) to eliminate the oil film rotation in the intershaft squeeze film damper is discussed. It is shown that this design is stable for all speeds.

PREVIOUSLY PROPOSED INTERSHAFT DAMPERS

Figure 1 illustrates the two previously proposed intershaft squeeze film damper configurations. Figure 1 (a) shows the configuration in which the intershaft damper's oil film is rotating with the speed of the inner (LP) rotor. This is achieved by constraining the inner race of the intershaft rolling element bearing to rotate with the speed of the LP rotor by using a squirrel cage (a dog mechanism is sometimes used). The oil film is confined between two rings that are fitted between the inner race of the ball bearing and the LP rotor. This is the configuration (or variations of it) that was experimentally tested in references 2,4 and 7. Figure 1 (b) shows the configuration in which the intershaft damper's oil film is rotating with the speed of the outer (HP) rotor. This is achieved by constraining the outer race of the intershaft rolling element bearing to rotate with the speed of the HP rotor by using a squirrel cage. The oil film is confined between two rings that are fitted between the outer race of the ball bearing and the HP rotor.

Another configuration proposed earlier (ref. 4) consists of two intershaft rolling element bearings and two oil films, one rotating at the speed of the inner rotor and one rotating at the speed of the outer rotor. This configuration is just the combination of the configurations shown in figure 1. It will be shown later that such a configuration would also be unstable above the engine's fundamental critical speed.

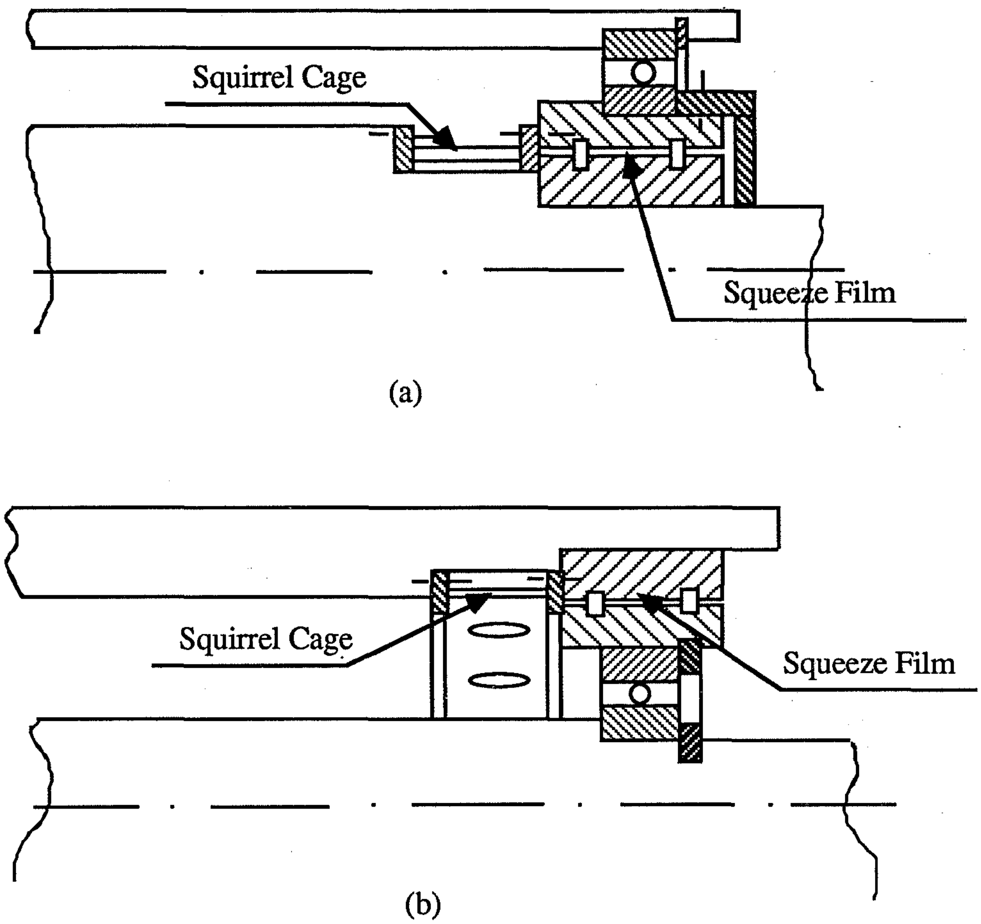

Figure 1 Previous designs

HYDRODYNAMIC FORCES IN INTERSHAFT DAMPERS

Figure 2 shows the nomenclature used in an intershaft squeeze film damper. At any angle θ measured from the whirling (r , t, z) coordinate system, which is centered at the center of the bearing but whirling with the journal (with z perpendicular to the plane of the paper), the film thickness h is given by

$$h = c - e \cos \theta \quad (1)$$

where c is the clearance, and e is the eccentricity of the journal in the bearing.

The forces generated in a squeeze film damper are usually calculated based on Reynolds equation†. Although more elaborate analytical models of the forces in squeeze film dampers exist (ref. 11), which include the effects of fluid inertia, they will not be used here since the instability can be completely described based on the more simple model based on Reynolds equation. Also for values of squeeze

† For a derivation of Reynolds equation see, for instance, reference 10.

Reynolds number significantly less than 10, the effects of fluid inertia can be neglected (ref. 12), but the instability would still exist.

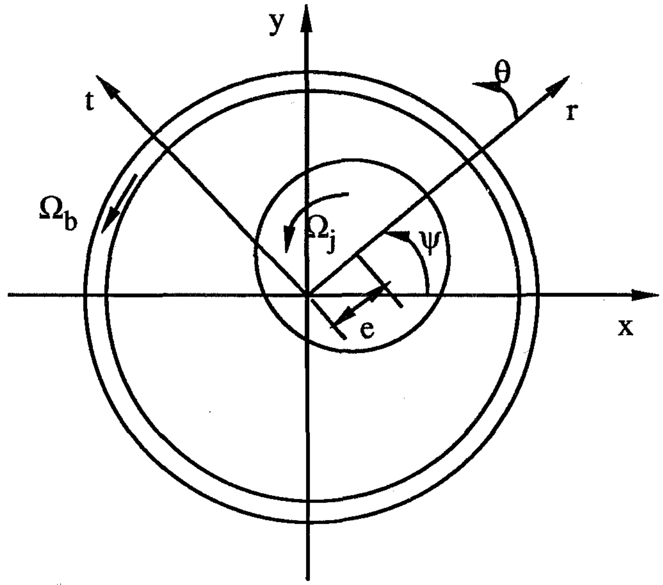

Figure 2 Intershaft Squeeze Film Damper

Reynolds equation is the differential equation that the pressure in the oil film bearing has to satisfy. For an intershaft squeeze film damper, Reynolds equation is

$$\frac{1}{R}\frac{\partial}{\partial\theta}\left(\frac{h^3}{\mu R}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=-12\,(e\,\dot{\psi}\sin\theta+\dot{e}\cos\theta)\;+6\,(\Omega_b+\Omega_j)\frac{\partial h}{\partial\theta} \tag{2}$$

where R is the radius of the bearing, μ is the viscosity of the oil, p is the pressure, $\dot{e}$ and $e\dot{\psi}$ are the radial and tangential velocities of the journal with respect to the bearing, respectively, and Ω_b and Ω_j are the angular velocities of the bearing and the journal respectively. For the intershaft damper of figure 1 (a), we have $\Omega_b = \Omega_j = \Omega_1$, where Ω_1 is the rotational speed of the inner (LP) rotor, while for the intershaft damper of figure 1 (b), we have $\Omega_b = \Omega_j = \Omega_2$, where Ω_2 is the rotational speed of the outer (HP) rotor. We will study the forces in the damper of figure 1 (a), that is we have $\Omega_b = \Omega_j = \Omega_1$, but the results can be applied to the damper of figure 1 (b) just by replacing Ω_1 with Ω_2.

For short dampers, with length to diameter ratio less than 0.25, the short bearing approximation (ref. 10) is usually used, which implies that the oil flow in the damper is predominantly axial and thus the pressure gradient in the θ-direction is much smaller than the pressure gradient in the z-direction and can be neglected in equation (2). In this case, Reynolds equation can be integrated with respect to z subject to the boundary conditions that the pressure is atmospheric at z = L/2 and at z = - L/2, then the pressure becomes

$$p=\frac{6\,\mu}{h^3}\left(\frac{L^2}{4}-z^2\right)[\,\dot{e}\cos\theta+e\,(\dot{\psi}-\Omega_1)\sin\theta\,] \tag{3}$$

The forces acting on the journal are obtained by integrating the pressure over both θ and z. Thus the radial and tangential forces are given by

$$F_r = -\int_{\theta_1}^{\theta_2} \int_{-\frac{L}{2}}^{\frac{L}{2}} p \cos\theta \; R \, d\theta \, dz$$

$$F_t = -\int_{\theta_1}^{\theta_2} \int_{-\frac{L}{2}}^{\frac{L}{2}} p \sin\theta \; R \, d\theta \, dz$$

respectively, where θ_1 and θ_2 represent the extent of the film. The forces in the damper depend heavily on cavitation, but since the instability under investigation is not caused by cavitation, and will, in fact, occur for both cavitated and uncavitated dampers, thus, for simplicity, we are going to assume an uncavitated damper and we have a full film and $\theta_1 = 0$ and $\theta_2 = 2\pi$. This would be the case if the damper was highly pressurized. In this case the damper forces become

$$F_r = -\frac{\mu R L^3}{c^3} \frac{\pi (1 + 2\varepsilon^2)}{(1 - \varepsilon^2)^{5/2}} \dot{e} \tag{4}$$

$$F_t = -\frac{\mu R L^3}{c^3} \frac{\pi}{(1 - \varepsilon^2)^{3/2}} e(\dot{\psi} - \Omega_1) \tag{5}$$

where $\varepsilon = e/c$, is the eccentricity ratio. The tangential force F_t acts in the direction opposite of the whirl if $\dot{\psi} > \Omega_1$, while if $\dot{\psi} < \Omega_1$ then F_t acts in the direction of the whirl. This is the case in which the oil is rotating at the speed of the LP rotor Ω_1 (fig. 1 (a)). For the intershaft squeeze damper of figure 1 (b) in which the oil is rotating at the speed of the HP rotor Ω_2, the tangential force F_t acts in the direction opposite of the whirl if $\dot{\psi} > \Omega_2$, while if $\dot{\psi} < \Omega_2$ then F_t acts in the direction of the whirl.

It should be noted that the radial force acting on the journal, equation (4), is a damping force that is proportional to the radial velocity of the journal, however, it is a nonlinear damping force because of the dependence on the eccentricity of the journal in the damper. For small oscillations of the journal in the damper, the damping force can be linearized, and the linear damping coefficient C will be

$$C = \frac{\pi \mu R L^3}{c^3} \tag{6}$$

Similarly, it can be shown that, for small oscillations, the linearized damping coefficient for the tangential force is also given by equation (6). Thus, if we consider the plane of the damper in figure 2 to be a complex plane, then the force acting in the damper, for small oscillations, considering equations (4)-(6), is

$$F_d = -C[\dot{Z}_e - i\Omega_1 Z_e] \tag{7}$$

where F_d is the complex linearized damping force acting in the damper, Z_e is the position vector of the journal in the damper in the complex plane, and $i = \sqrt{-1}$.

STABILITY ANALYSIS

Consider a rotor system consisting of two Jeffcott rotors of masses m_1 and m_2, as shown in figure 3. The inner rotor, which represents the low pressure (LP) rotor in a two spool gas turbine, is mounted on two identical rolling element bearings (which are assumed to be rigid and massless), and is rotating at Ω_1, and has a stiffness K_1. The outer rotor, which represents the high pressure (HP) rotor, is mounted on two intershaft squeeze film dampers (similar to those in figure 1 (a)), and is rotating at Ω_2, and has a stiffness K_2.

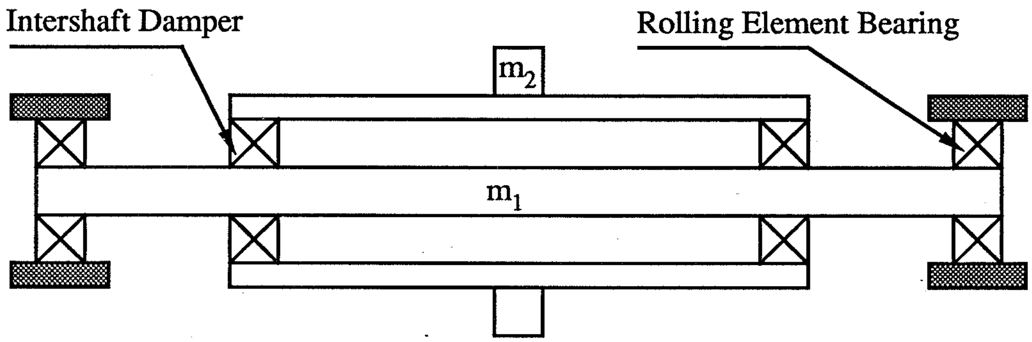

Figure 3 Schematic of the rotor system under investigation

Since the rotor system shown in figure 3 is symmetric, then we can use the planar rotor model introduced by Crandall in reference 13, shown in figure 4, to model this system. The outer massless ring of figure 4 represents the journal of the LP rotor rotating at Ω_1 in the ball bearings. The mass of the LP rotor is represented by the second outer ring which has mass m_1 and is rotating at Ω_1. The flexibility of the LP rotor is represented by the springs of stiffness $K_1/2$ connected between the two outermost rings. The two intershaft squeeze film dampers are represented by the rotating dashpots of linearized coefficients C, and they are rotating between the ring representing the mass of the LP rotor and the outer race of the ball bearing in figure 4 which represents the inner race of the ball bearing of figure 1(a) and is constrained to rotate at Ω_1. The mass of the HP rotor is represented by the disk of mass m_2 in the center of figure 4. The flexibility of the HP rotor is represented by the springs of stiffness $K_2/2$. It should be noted that we did not incorporate in this model the stiffness of the retaining spring, that is we are assuming that the intershaft squeeze film damper is constrained to rotate at the speed of the LP rotor by using a dogging mechanism rather than by a squirrel cage. This will not affect the stability analysis, since the instability is caused by the hydrodynamic forces in the oil film, however, if a squirrel cage is used then it will affect the critical speeds of the rotor system.

Considering figure 4 to be a complex plane, and neglecting gyroscopic effects, then the equations of motion can be obtained directly from figure 4, as

$$m_1 \ddot{Z}_1 + K_1 Z_1 = 2 F_d \tag{8}$$

$$m_2 \ddot{Z}_2 + K_2 Z_2 = K_2 Z_3 \tag{9}$$

$$K_2 (Z_3 - Z_2) = - 2 F_d \tag{10}$$

where F_d is given by equation (7) with

$$Z_e = Z_1 - Z_3 \tag{11}$$

We have used here Z to represent the position vectors in the complex plane, i.e. $Z = x + iy$, with the subscripts 1 for the LP rotor, 2 for the HP rotor, and 3 for the position of the intershaft rolling element bearing. Equations (8) and (9) are the equations of motion of the LP and HP rotors, respectively, and equation (10) represents the force balance at the intershaft bearing.

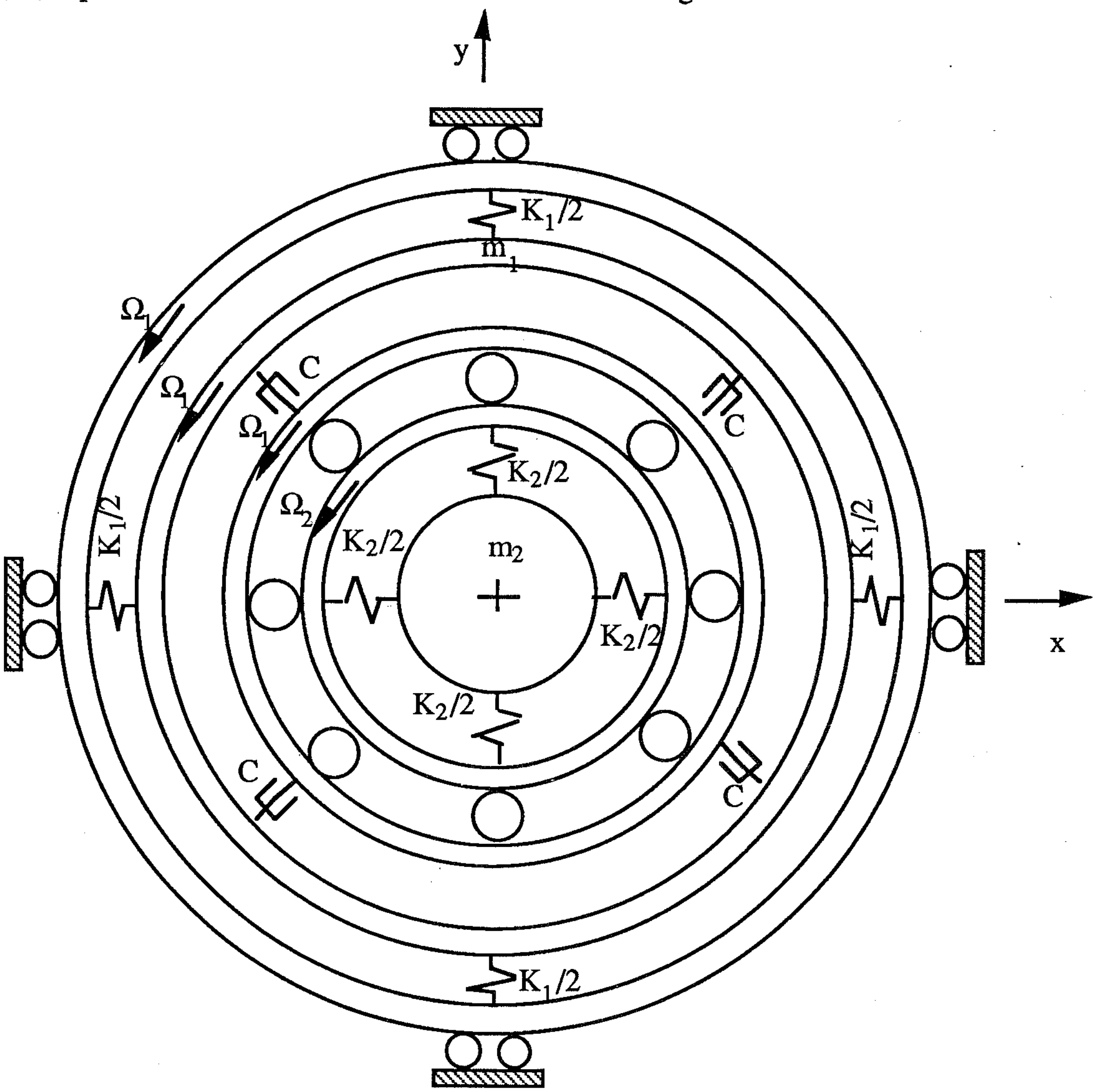

Figure 4 Planar rotor model

We begin by studying the undamped case. If the oil film was rigid, then the system would be undamped, as can be easily visualized in figure 4, if the dashpots become rigid. That would be the case if the damping coefficient became infinite. In this case the system can be thought of as the two-mass-two-spring system of figure 5, as the motions in the x and y directions are uncoupled. Then equations (8) and (9) reduce to

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{bmatrix} K_1 + K_2 & -K_2 \\ -K_2 & K_2 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \tag{12}$$

in the x-direction, with similar equations in the y-direction. The motion in the complex plane has exactly the same natural frequencies for the motion in both the x and y directions. It is useful to think of these natural frequencies as forward and backward whirling motions. Thus this system exhibits two forward whirls and two backward whirls corresponding to the four degrees of freedom the system possesses.

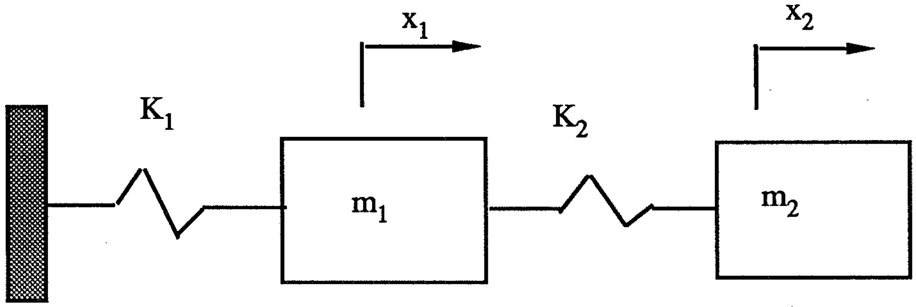

Figure 5 Undamped model of system

The characteristic equation for the system described by equation (12) is

$$\omega_n^4 - [\, \omega_{11}^2 + (\mu + 1)\, \omega_{22}^2 \,]\, \omega_n^2 + \omega_{11}^2\, \omega_{22}^2 = 0 \tag{13}$$

where $\omega_{11}{}^2 = K_1/m_1$, $\omega_{22}{}^2 = K_2/m_2$, $\mu = m_2/m_1$, and ω_n is a natural frequency of the system. ω_{11} is the natural frequency of the LP rotor alone, and ω_{22} is the natural frequency of the HP rotor alone. The natural frequencies of the system can be obtained by solving the quadratic of equation (13) and we get

$$\omega_{n2,1}^2 = \frac{1}{2} \{\, [\, \omega_{11}^2 + (\mu + 1)\, \omega_{22}^2 \,] \pm \sqrt{D} \,\} \tag{14}$$

where

$$D = \omega_{11}^4 + 2\,(\mu - 1)\, \omega_{11}^2\, \omega_{22}^2 + (\mu + 1)^2\, \omega_{22}^4 \tag{15}$$

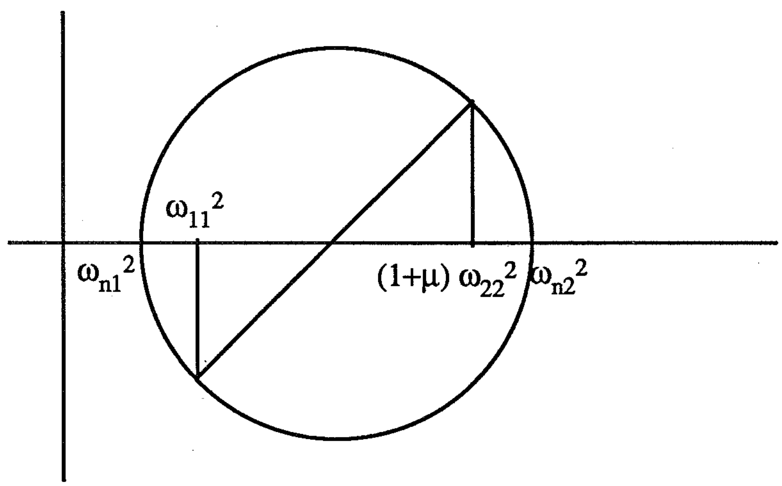

Figure 6 Mohr's circle representation in the ω^2-plane

ω_{n1} is the first natural frequency and ω_{n2} is the second natural frequency of the system. It should be emphasized that there are two forward whirls: one at ω_{n1} and the other at ω_{n2} , and there are two backward whirls one at $-\omega_{n1}$ and the other at $-\omega_{n2}$. It will prove beneficial to know the relative magnitudes of ω_{n1},

ω_{n2} and ω_{11}. To do so, we use the Mohr's circle representation of equations (13) and (14) (ref. 14), which is shown in figure 6, and from which it can be seen that

$$\omega_{n2}^2 > \omega_{11}^2 > \omega_{n1}^2 \tag{16}$$

To study the stability of the damped system, we assume that Z_1, Z_2 and Z_3 have $e^{i\omega t}$ time behavior, where ω is a complex quantity. If the imaginary part of ω is negative, then the system is unstable. Substituting equations (7), (9), (10) and (11) into equation (8), and after some manipulation, we get

$$\omega^4 - [\, \omega_{11}^2 + (\, \mu + 1)\, \omega_{22}^2 \,]\, \omega^2 + \omega_{11}^2\, \omega_{22}^2 + \frac{i\, \eta\, \omega_{11}\, \omega^2\, (\omega^2 - \omega_{11}^2)}{(\, \Omega_1 - \omega\,)} = 0 \tag{17}$$

where $\eta = K_2/(2\, C\, \omega_{11})$ is a nondimensional parameter. It should be noted that as $\eta \rightarrow 0$, i.e. as $C \rightarrow \infty$, equation (17) reduces to equation (13) for the undamped system. This would be the case if the oil film becomes rigid. Solving the above quadratic, we get

$$\omega^2 = \frac{1}{2} \left\{ [\, \omega_{11}^2 + (\, \mu + 1)\, \omega_{22}^2 \,] \pm \sqrt{ D - 4\, \frac{i\, \eta\, \omega_{11}\, \omega^2\, (\omega^2 - \omega_{11}^2)}{(\, \Omega_1 - \omega\,)} } \right\} \tag{18}$$

We can solve either equation (17) or (18) numerically to obtain the characteristic roots for this system. However, we need to find analytically an expression to determine the effect of the intershaft damper on the stability of the system, and also we need an indication of the regions of instability. We can consider the effect of relaxing the rigid damper for the undamped system, and introduce some damping into the undamped system, and study its effects on the stability. This corresponds to studying the system for small values of η. Thus, expanding the square root in equation (18) and neglecting terms of $O(\eta^2)$, equation (18) becomes

$$\omega^2 = \frac{1}{2} \left\{ [\, \omega_{11}^2 + (\, \mu + 1)\, \omega_{22}^2 \,] \pm \sqrt{D} \left(1 - \frac{i\, 2\, \eta\, \omega_{11}\, \omega^2\, (\omega^2 - \omega_{11}^2)}{(\, \Omega_1 - \omega\,)\, D} \right) \right\} \tag{19}$$

The real part of equation (19) represents the square of the undamped natural frequencies ω_{n1}^2 and ω_{n2}^2. To obtain the damped frequencies ω^2, we have to iterate on equation (19) in the neighborhood of ω_{n1}^2 for the first mode, and in the neighborhood of ω_{n2}^2 for the second mode. Note that there is a sign change in the imaginary part of ω in equation (19) when we are studying the motion in the neighborhood of ω_{n1}^2, from that when we are studying the motion in the neighborhood of ω_{n2}^2. Thus we have two equations to iterate on, for the first mode

$$\omega^2 = \omega_{n1}^2 + \frac{i\, \eta\, \omega_{11}\, \omega^2\, (\omega^2 - \omega_{11}^2)}{(\, \Omega_1 - \omega\,)\, \sqrt{D}} \tag{20}$$

and for the second mode

$$\omega^2 = \omega_{n2}^2 - \frac{i\,\eta\,\omega_{11}\,\omega^2\,(\omega^2 - \omega_{11}^2)}{(\,\Omega_1 - \omega\,)\sqrt{D}} \tag{21}$$

Iterating on equation (20) in the neighborhood of ω_{n1} we get (neglecting terms of $O(\eta^2)$)

$$\omega = \omega_{n1} - \frac{i\,\eta\,\omega_{11}\,\omega_{n1}\,(\omega_{11}^2 - \omega_{n1}^2)}{2\,(\,\Omega_1 - \omega_{n1}\,)\sqrt{D}} \tag{22}$$

for the forward whirl in the first mode. Note that, by equation (16), $\omega_{11}^2 > \omega_{n1}^2$, and thus the quantity $(\omega_{11}^2 - \omega_{n1}^2)$ is always positive. Equation (22) shows that the forward whirl in the first mode will be stable as long as Ω_1, the speed of the LP rotor, is less than ω_{n1}, the fundamental natural frequency of the system. If $\Omega_1 > \omega_{n1}$, the imaginary part of ω will be negative and the forward whirl in the first mode will be unstable.

Iterating on equation(20) in the neighborhood of $-\omega_{n1}$, we get (neglecting terms of $O(\eta^2)$)

$$\omega = -\,\omega_{n1} + \frac{i\,\eta\,\omega_{11}\,\omega_{n1}\,(\omega_{11}^2 - \omega_{n1}^2)}{2\,(\,\Omega_1 + \omega_{n1}\,)\sqrt{D}} \tag{23}$$

for the backward whirl in the first mode. Equation (23) shows that the backward whirl in the first mode is always stable, irrespective of the speeds of the rotors, since the imaginary part of ω is always positive.

For the forward whirl in the second mode, we iterate on equation (21) in the neighborhood of ω_{n2}. Thus (neglecting terms of $O(\eta^2)$)

$$\omega = \omega_{n2} - \frac{i\,\eta\,\omega_{11}\,\omega_{n2}\,(\omega_{n2}^2 - \omega_{11}^2)}{2\,(\,\Omega_1 - \omega_{n2}\,)\sqrt{D}} \tag{24}$$

Also, by equation (16), $\omega_{n2}^2 > \omega_{11}^2$, and thus the quantity $(\omega_{n2}^2 - \omega_{11}^2)$ is always positive. Equation (24) shows that the forward whirl in the second mode is stable if the speed of the LP rotor Ω_1, is less than ω_{n2}, the second natural frequency of the system. If $\Omega_1 > \omega_{n2}$ the forward whirl in the second mode will become unstable, as the imaginary part of ω in equation (24) would be negative.

Iterating on equation (21) in the neighborhood of $-\omega_{n2}$, we get (neglecting terms of $O(\eta^2)$)

$$\omega = -\,\omega_{n2} + \frac{i\,\eta\,\omega_{11}\,\omega_{n2}\,(\omega_{n2}^2 - \omega_{11}^2)}{2\,(\,\Omega_1 + \omega_{n2}\,)\sqrt{D}} \tag{25}$$

for the backward whirl in the second mode. It can be seen from equation (25) that the imaginary part of ω is always positive, for the backward whirl in the second mode, and thus the backward whirl in the second mode will always be stable, irrespective of the speeds of the rotors.

Thus, we have shown that, for the intershaft squeeze film damper of figure 1(a), the backward whirls are always stable, while the forward whirls become unstable if the speed of rotation of the LP rotor surpasses the natural frequencies of the system. Thus, the intershaft squeeze film damper of figure 1(a) will become a destabilizing device if the speed of the LP rotor surpasses the fundamental critical speed. For the intershaft squeeze film damper of figure 1(b), as mentioned earlier, all the above stability analysis will be valid, if we replace Ω_1 by Ω_2, the speed of the HP rotor, in all of the equations. Thus, for the intershaft squeeze film damper of figure 1(b), the backward whirls are always stable, while the forward whirls become unstable if the speed of rotation of the HP rotor surpasses the natural frequencies of the system. Also, we conclude that the intershaft squeeze film damper of figure 1(b) will become a destabilizing device if the speed of the HP rotor surpasses the fundamental critical speed.

DESCRIPTION OF THE INSTABILITY

If a tangential force acts in the same direction as the whirl, then it would be adding energy to the whirl, and thus results in the system spiralling outwards with an ever growing whirl amplitude (ref. 15). In the preceding sections, we had shown that the intershaft squeeze film damper of figure 1 (a) produced a tangential force acting on the journal in the same direction as the forward whirl if $\dot{\psi} < \Omega_1$. Thus it is pumping energy into the forward whirl and causing the instability. This energy is obtained from the kinetic energy of the spinning of the LP rotor. The intershaft squeeze film damper of figure 1 (b) produces a tangential force acting on the journal in the same direction as the forward whirl if $\dot{\psi} < \Omega_2$. Thus it is pumping energy into the forward whirl obtained from the kinetic energy of the spinning of the HP rotor, and causing the instability.

On the other hand, if the tangential force is acting in the opposite direction of the whirl, then it would be removing energy from the whirl. This is the damping mechanism of squeeze film dampers in general. For the intershaft squeeze film damper of figure 1 (a) this occurs when $\dot{\psi} > \Omega_1$, while for the intershaft squeeze film damper of figure 1 (b) this occurs when $\dot{\psi} > \Omega_2$. Also, this occurs for the backward whirls, the tangential force is always opposing the whirl.

From the above analysis it can be seen that the threshold of instability of intershaft squeeze film dampers is when the whirl frequency $\dot{\psi}$ is equal to the speed of rotation of the oil (Ω_1 for the intershaft squeeze film damper of figure 1 (a), and Ω_2 for the intershaft squeeze film damper of figure 1 (b)). For free vibration, the whirl frequency $\dot{\psi}$ of the rotor system is its natural whirling frequency. Thus rotor systems incorporating intershaft squeeze film dampers of the configuration shown in figure 1 (a) become unstable if the speed of rotation of the LP rotor Ω_1 is larger than the lowest natural whirling frequency of the system, while for rotor systems incorporating intershaft squeeze film dampers of the configuration shown in figure 1 (b), they become unstable if the speed of rotation of the HP rotor Ω_2 is larger than the lowest natural whirling frequency of the system. It is important to interpret Ω_1 and Ω_2, for the intershaft dampers of figure 1(a) and figure 1(b), respectively, as being the speed of oil rotation in the damper, in the absence of vibration. This oil film rotation is due to the spinning of the journal and the bearing at the speed of either the LP or the HP rotor.

The design proposed by Courage (ref. 4) incorporating two oil films, one rotating at Ω_1 and one rotating at Ω_2 will also be unstable if both Ω_1 and Ω_2 are above the lowest natural whirling frequency (first critical speed) of the rotor system, since the tangential forces in both of the oil films will act to enhance the whirl. If one of the rotors is operating above the critical speed and the other below it, then the stability of the rotor system will depend on the amount of energy being pumped into the whirl by the unstable damper and the amount of energy removed from the whirl by the stable damper.

A pictorial representation of the instability is shown in figure 7. The outer ring in figure 7 represents the bearing while the inner ring represents the journal. If the oil in the gap is rotating at a speed

Ω, then as the journal whirls in the bearing at a speed ω, the oil film will produce a drag force on the journal opposite to the whirl if ω is larger than Ω (figure 7 (a)), and thus acts to damp the whirl. While if Ω is larger than ω then a drag force acts on the journal in the direction of the whirl (figure 7 (b)), and thus acts to enhance the whirl. If the oil film is stationary, then the oil film will always produce a drag force opposing the whirl (figure 7 (c)), and thus will always be stable.

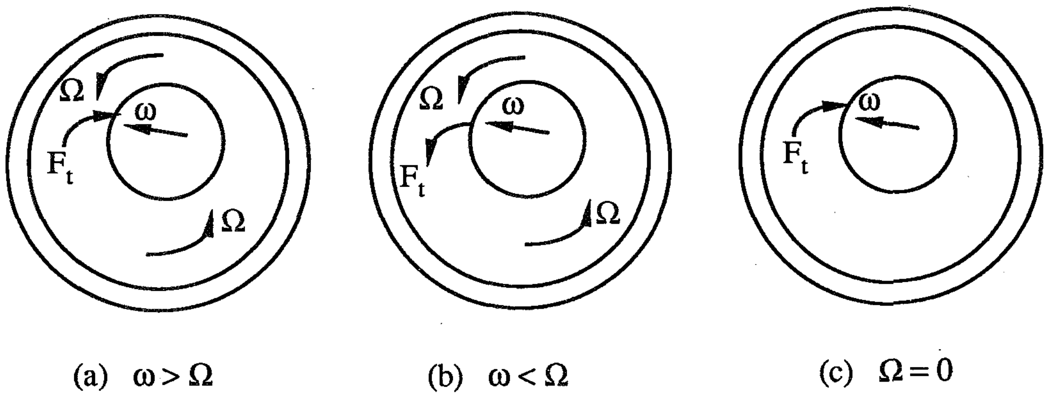

Figure 7 Instability mechanism

STABLE INTERSHAFT DAMPER

If the rotor system is operating above a natural frequency it is said to be operating supercritically. Usually high performance turbomachinery operate supercritically, above several critical speeds, but because of the instability described above, rotating systems incorporating either of the intershaft squeeze film dampers shown in figure 1 cannot operate supercritically.

From the discussion in the preceding section, it can be seen that the reason for the instability is the rotation of the oil film. The oil film is rotating because the journal and the bearing are both rotating, at Ω_1 for the damper of figure 1 (a) and at Ω_2 for the damper of figure 1 (b). It can also be seen that if the oil film was stationary (figure 7 (c)) then the intershaft squeeze film damper would be stable irrespective of the speed of either the LP or the HP rotor. A stationary oil film can only be achieved if both the journal and the bearing are not rotating.

To show that the intershaft squeeze film damper will be stable if both the journal and the bearing are not rotating, let us find the forces acting on the journal. In this case, Ω_j and Ω_b are both zero, and if we substitute in Reynolds equation, equation (2), and solve for the pressure, and then integrate the pressure over θ and z, as we did before, we will find that the tangential force acting on the journal is

$$F_t = -\frac{\mu R L^3}{c^3} \frac{\pi}{(1-\varepsilon^2)^{3/2}} e \dot{\psi} \qquad (26)$$

Thus from equation (26) it can be seen that the tangential force F_t, for an intershaft damper with a stationary oil film, always acts in the opposite direction of the whirl, and thus it is always removing energy from the whirl. Also, if we substitute for $\Omega_1 = 0$ in the stability analysis equations, it can be seen that the intershaft damper with a stationary oil film is always stable.

A new design of intershaft squeeze film dampers (ref. 8,9) to eliminate the instability by incorporating a stationary oil film is shown in figure 8. In this design two intershaft rolling element bearings are used. The outer race of the outer bearing is rotating with the speed of the HP rotor, and the inner race of the inner bearing is rotating with the speed of the LP rotor. The inner race of the outer bearing and the outer race of the inner bearing are both held stationary by two squirrel cages connected to the engine frame. The oil film is confined in an annulus between the nonrotating outer race of the inner bearing and the nonrotating inner race of the outer bearing. Thus neither the rotation of the LP rotor nor the rotation of the HP rotor will reach the oil film, which will act only to damp the whirling vibration of the rotor system. The squirrel cages, which are light springs, aside from keeping the oil film stationary, provide a means for centering the squeeze film damper.

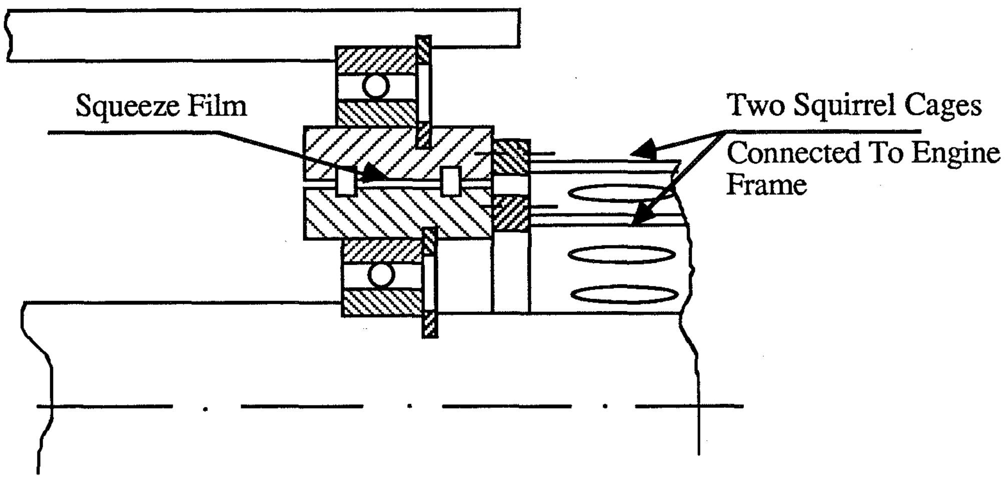

Figure 8 Stable intershaft squeeze film damper

The main advantage of this design is that intershaft squeeze film dampers can be used in rotating machinery that operate supercritically without becoming unstable. The other advantages of using intershaft squeeze film dampers are also retained, namely, the reduction of bearing loading, the reduction of cyclic stresses both in the bearings and in the support structure, and the reduction of the amplitude of vibration (ref. 2,5). Also the use of intershaft bearings results in the decrease of rubbing between the rotors and the reduction of shaft deflections (ref. 1).

The disadvantage of this design of intershaft squeeze film dampers is that the damper must be located close to the support structure. Thus one of the advantages of using intershaft bearings is lost, namely the obstruction of the aerodynamic flow path. Also, because the damper must be located close to the support structure, it may not be always possible for the engine designer to locate the damper at the best possible locations to damp the engine modes effectively. If the damper is located near a node of one or more of the important modes of the engine, then it will provide very little damping to these modes. The use of long squirrel cages may be a remedy for such a situation, but since the squirrel cages act as soft springs, then having unnecessary flexibility at the intershaft bearing location may be detrimental to the overall dynamics of the engine. Thus such a solution should be assessed carefully.

Another disadvantage may be the cost of an additional ball bearing. The stable design of the intershaft squeeze film damper requires two intershaft ball bearings instead of one. Usually engine designers try to minimize the number of bearings in the engine. If the designer had the luxury of adding another ball bearing to the engine, then he may choose to place it in a non-intershaft location, and this may well reduce the bearing loadings. But again this may not always be possible, since this would require that the support structure be close to the rotor in yet another location.

CONCLUSION

A stability analysis of rotors incorporating intershaft squeeze film dampers was presented. A rotor model consisting of two Jeffcott rotors with two intershaft squeeze film dampers was investigated. Examining the system characteristic equation for the conditions at which the roots indicate an ever growing unstable motion, result in the stability conditions. The reason for the instability of the previously proposed intershaft squeeze film dampers above the engine's first critical speed, is shown to be the rotation of the oil film. If the oil film is rotating at a speed above the critical speed of the engine, then it will be adding energy to the whirl from the spinning rotor, thus causing the instability.

A new design of intershaft squeeze film dampers in which the oil film is stationary is described. This design is shown to be stable irrespective of the speeds of rotation of the rotor system. The advantages and disadvantages of such a design are also discussed, and it is pointed out that although it may not be always possible to place the damper, with this design, in the best possible location to damp the engine modes effectively, this is the only stable intershaft squeeze film damper.

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