DYNAMIC CARACTERISTICS OF AN UNSEALED SQUEEZE FILM DAMPER WITH A CIRCUMFERENTIAL FEEDING GROOVE

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Abstract. Squeeze film dampers are widely used in aircraft gas turbines due to their capability to isolate vibrations, to reduce forces transmitted to structural components, and to improve the stability characteristics of rotor-bearing systems. Another important aspect of squeeze film dampers is related to the fact that they are not susceptible to subharmonic instabilities, such as oil whirl and oil whip. In aircraft gas turbines, it is very important to predict the dynamic behaviour of the rotating structure in order to avoid vibration problems. The behaviour of a rotor-bearing system is dependent upon the values of the stiffness and damping properties of the bearings. In this work, it is calculated the stiffness and damping properties of an unsealed squeeze film damper with a circumferential feeding groove. This squeeze film damper is to be used as the rear bearing of a single spool gas turbine. The circumferential groove is taken as a special damper, when the fluid forces are analysed. Therefore, the dynamic performance of the damper is attributed to two film lands and their interactions with the circumferential feeding groove. From this point of view, the dynamic effects in the groove are studied based on the linearized Navier-Stokes equations. The effects of variations of fluid velocity and pressure in the groove are taken into account to determine the fluid forces. As a result, the calculated stiffness and damping dynamic properties of the squeeze film damper can be used in a rotordynamics analysis to predict the dynamic behaviour of the single spool gas turbine.

Keywords: squeeze film damper, fluid forces, circumferential feeding groove, gas turbine.

1. INTRODUCTION

Squeeze film dampers are widely used in high speed turbo-machinery due to their capability to isolate vibrations, to reduce forces transmitted to structural components, and to improve the characteristics of rotor-bearing systems. The behaviour of a rotor-bearing system is dependent upon the stiffness and damping values of the bearings. Therefore, accurate theoretical models for calculating the dynamic properties of squeeze film dampers are very important to predict the rotor-system dynamic behaviour.

In a squeeze film damper, oil feeding is generally via a central circumferential groove, which effectively splits the damper width of L into two film lands, each one with an axial length of $L/2$. Experience has shown that theoretical models based on traditional lubrication theory are limited in their applicability and fail to explain the performance of squeeze film dampers over a wide range of operating conditions. In the studies carried out by (San Andrés, 1992) and (Arauz and San Andrés, 1994), it was shown that the main reason for such deviation is due to the fact that traditional lubrication theory neglects the pressure generated by the groove. Experimental evidence (San Andrés, 1992) has been found that the pressure produced in the groove relates to its depth, the shaft speed, the vibration amplitude at the squeeze film damper, and the oil supply pressure. In (Arauz and San Andrés, 1996), the dynamic pressures measured at the feeding groove are as large as those generated at the film lands. Therefore, the feeding groove affects the dynamic characteristics of squeeze film dampers and, consequently, the unbalance responses of the rotor-bearing system. An evaluation of various fluid film models which takes into account different types of grooves, lubricants and unbalance levels is performed in (Siew *et al*, 2002). Useful guidelines for designing a squeeze film damper are found in such work.

In this paper, it is calculated the stiffness and damping dynamic properties of a specially designed unsealed squeeze film damper with a circumferential feeding groove. This squeeze film damper is to be used as the rear bearing of a single spool gas turbine. Figure 1 shows a magnifying view of the rear bearing of the referred gas turbine. This squeeze film damper will be continuously lubricated with Aeroshell® 500 Turbine Oil.

The circumferential feeding groove of this squeeze film damper is taken as a special damper, when the fluid forces are analysed. Therefore, the dynamic performance of the whole damper is attributed to two film lands and their interactions with the circumferential feeding groove. The dynamic effects in the groove are studied based on the linearized Navier-Stokes equations. The effects of variations of fluid velocity and pressure in the groove are taken into account to determine the fluid forces. The calculated stiffness and damping dynamic properties of this squeeze film damper can be used in a rotordynamics analysis to predict the dynamic behaviour of the gas turbine rotating system.
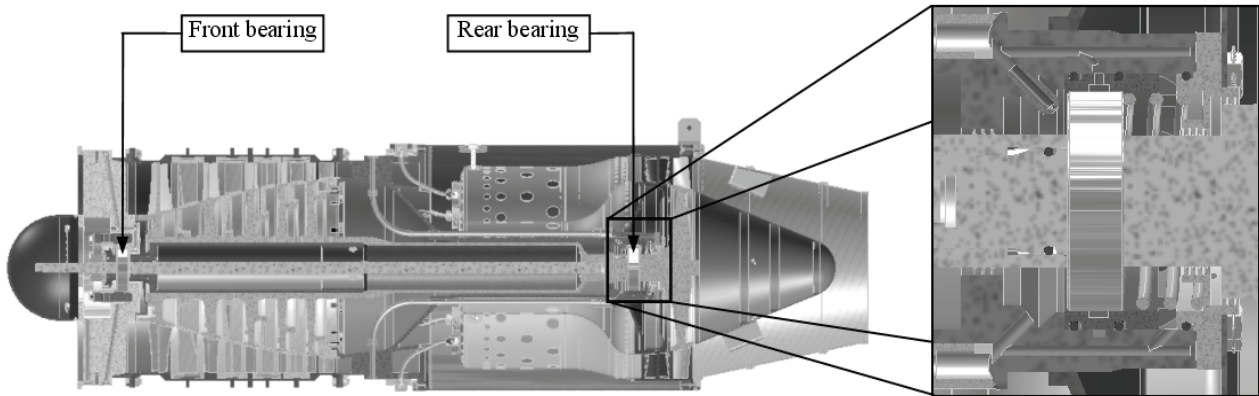
Figure 1. Magnifying view of the rear bearing of the presented single spool gas turbine.

2. ANALYSIS MODEL

The groove of the squeeze film damper is composed by a few radial inlets for oil supply, as recommended by literature up to now. It is assumed that the small diameter holes do not alter the pressure field in the groove. Therefore, from the point of view of fluid dynamics, the groove can be thought as a specific damper to analyse the fluid forces. Figure 2 shows a schematic diagram of an arbitrary squeeze film damper geometry. The geometric center of the displaced journal is located at O_J, and the geometric center of the bearing is located at O_B. R is the journal radius, h is the oil film thickness, θ is the angular position measured from the line of maximum film thickness, and ϕ is the attitude angle. The radial and tangential forces originated in the fluid film are denoted by F_r and F_t, respectively.

Other important properties shown in Figure 2 can be listed: C_r is the radial clearance between the journal and the bearing, C_g is the groove depth, d_h is the hole diameter for oil supply, l_g is one-side length of the groove, l is one-side length of the film land, L is the whole length of the damper, and p is the fluid pressure.
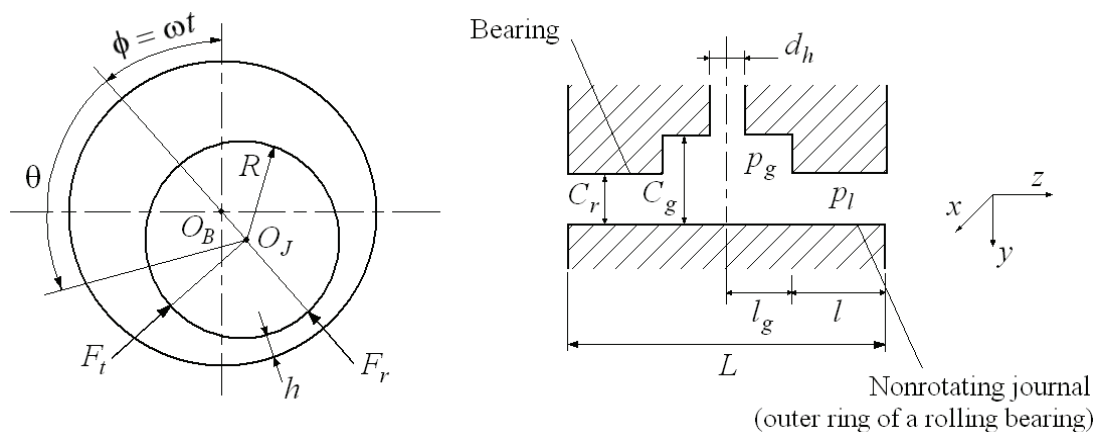

Figure 2. Schematic diagram of a squeeze film damper geometry.

The fluid forces in the groove can be analysed from a linearized Navier-Stokes equation. For film lands, the fluid forces are studied considering fluid inertia effects, according to (Tichy and Bou-Said, 1991). Therefore, the linearized Navier-Stokes equations can be expressed as:

$$\rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial x} + \eta \frac{\partial^2 u}{\partial y^2}, \qquad \frac{\partial p}{\partial y} = 0, \qquad \rho \frac{\partial w}{\partial t} = -\frac{\partial p}{\partial z} + \eta \frac{\partial^2 w}{\partial y^2}, \qquad (1)$$

with the following continuity condition:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad (2)$$

where u, v, and w are the fluid velocity components in the circumferential, radial and axial directions, respectively. η is the dynamic viscosity of oil and ρ is the oil density. For a circular-centered whirl with run-out e and precession rate $\partial \phi / \partial t = \omega$, the boundary conditions at the groove can be expressed as follows:

$$u = v = w = 0 \text{ at } y = 0, \qquad u = \text{Real}(e\omega e^{i\theta}), \qquad v = \text{Real}(ie\omega e^{i\theta}), \qquad w = 0 \text{ at } y = C_g, \tag{3}$$

where $\theta = x / R - \omega t$. The excentricity ratio with respect to the bearing is given by $\varepsilon = e / C_r$, and the pressure field in the damper with a central groove is expressed as:

$$p = \begin{cases} p_g \{-l_g \leq z \leq l_g\} & \text{for groove} \\ p_l \{-l_g \geq z \geq -L/2 \ \wedge \ l_g \leq z \leq L/2\} & \text{for film lands} \end{cases} \tag{4}$$

Thus, the radial and tangential forces acting on the journal can be expressed as (Tan $et\ al$, 1997):

$$F_r = \int_{-L/2}^{L/2} \int_0^{2\pi} p \cos\theta \, dz R d\theta \tag{5}$$

$$F_t = \int_{-L/2}^{L/2} \int_0^{2\pi} p \sin\theta \, dz R d\theta \tag{6}$$

For circular-centered whirl at constant velocity and π-film model in the squeeze film damper, the nondimensional forms of radial and tangential forces can be written based on (Tan $et\ al$, 1998) as:

$$f_r = \frac{2 + 3\varphi + 3A_1\varphi(1 + 2\varphi)}{2(1 + \varphi)^3} I_3^{11} - \text{Re} \frac{2 + 3\varphi}{2(1 + \varphi)^3} I_r \tag{7}$$

$$f_t = \frac{2 + 3\varphi + 3A_1\varphi(1 + 2\varphi)}{2(1 + \varphi)^3} I_3^{20} - \text{Re} \frac{2 + 3\varphi}{2(1 + \varphi)^3} I_t \tag{8}$$

where:

$$A_1 = \frac{\gamma^2(3 - 2\gamma)}{1 + \left(\dfrac{\text{Re}}{10\gamma}\right)^2}, \qquad \gamma = \frac{C_r}{C_g}, \qquad \text{Re} = \frac{C_r^2 \rho \omega}{\eta} \qquad \text{and} \qquad \varphi = \frac{l_g}{l}. \tag{9}$$

I_n^{lm} in the above expressions are defined by the Booker journal bearing integral form as:

$$I_n^{lm} = \int_{\vartheta_2}^{\vartheta_1} \frac{\sin^l \vartheta \cos^m \vartheta}{(1 + \varepsilon \cos \vartheta)^n} d\vartheta \tag{10}$$

$$I_3^{11} = \frac{2\varepsilon}{r^4}, \qquad I_r = \left(\frac{I_1^{02}}{12} + \frac{I_2^{21}}{5}\varepsilon\right) = \frac{\pi(1 - r)}{\varepsilon^2 r}\left[\frac{1}{12} - \frac{1}{5}(1 - r)\right] \tag{11}$$

$$I_3^{20} = \frac{\pi}{2r^3}, \qquad I_t = \left(\frac{I_1^{11}}{12} + \frac{I_2^{30}}{5}\varepsilon\right) = \frac{19}{60\varepsilon^2}\left(2\varepsilon - \ln\left|\frac{1 + \varepsilon}{1 - \varepsilon}\right|\right) \tag{12}$$

where $r = (1 - \varepsilon^2)^{1/2}$.

The dimensional forms of the radial and tangential forces can be written as:

$$F_r = -\left(\frac{RL^3\eta\omega}{C_r^2}\right)f_r\varepsilon \tag{13}$$

$$F_t = -\left(\frac{RL^3\eta\omega}{C_r^2}\right)f_t\varepsilon \tag{14}$$

Therefore, the stiffness and damping dynamic properties of the squeeze film damper, assuming circular-centered orbits and π-film model, can be calculated according to (Dai *et al*, 2001):

$$K_{SFD} = -\frac{F_r}{e} \qquad \text{(stiffness of the squeeze film damper)} \tag{15}$$

$$C_{SFD} = -\frac{F_t}{e\omega} \qquad \text{(damping of the squeeze film damper)} \tag{16}$$

3. RESULTS

A schematic diagram of the studied squeeze film damper is shown in Figure 3. It was selected an angular contact ball bearing for this squeeze film damper (Barden® 206HJH). The outer ring of this ball bearing does not rotate. A centralizing spring is used to assure that the raceways will be always in contact with the balls during the gas turbine operation. A preload force of 150 N is to be applied on the ball bearing by the centralizing spring. A distance δ is used to prevent any thermal expansion problem. The angular contact ball bearing will present small axial movements due to the high temperature loads of the gas turbine. These high temperatures loads give origin to relative movements between the rotating system and the static components. A safety clearance δ must be left at the bearing in order to avoid any problem during the thermal expansion behaviour.

The most important geometrical properties of the studied squeeze film damper are presented in Table 1. Three holes will be used for oil inlet. They will be separated from each other by 120°. The oil supply pressure is to be adjusted to approximately 20 psi. Table 2 shows the kinematics and dynamic viscosities of Aeroshell® 500 Turbine Oil.
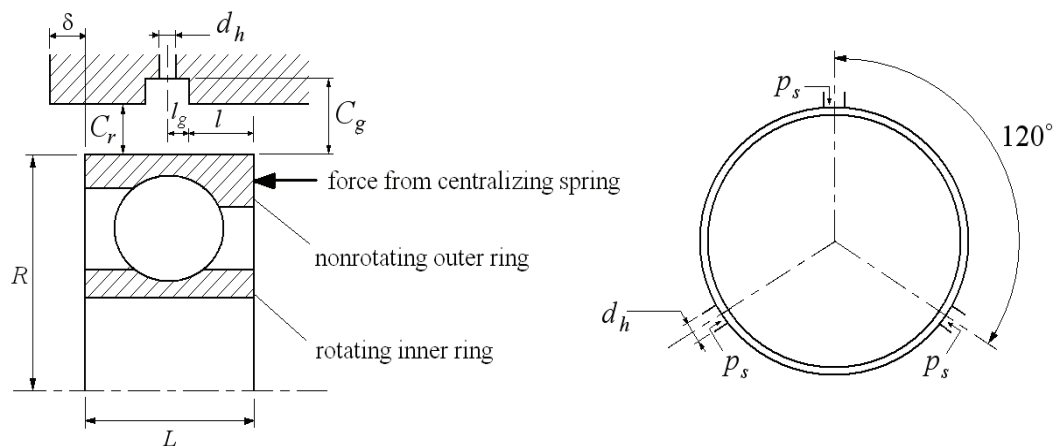

Figure 3. Schematic diagram of the designed squeeze film damper used as the rear bearing of the presented gas turbine.

Geometrical Properties [m]	
δ	0.0025
d_h	0.003
C_r	0.00012
C_g	0.0036965
l	0.005
l_g	0.003
L	0.016
R	0.031

Table 1. Geometrical properties of the studied squeeze film damper.

Rotor Speed	Oil Temperature	Kinematics Viscosity	Dynamic Viscosity
[rpm]	[°C]	[m²/s]	[Pa.s]
0	25	3.27555E-05	0.03291926
5000	60	1.97522E-05	0.019850971
10000	70	1.6037E-05	0.016117173
15000	80	1.23218E-05	0.012383376
20000	90	8.60655E-06	0.008649579
25000	95	6.74894E-06	0.006782681
30000	100	4.89133E-06	0.004915782

Table 2. Viscosities of Aeroshell® 500 Turbine Oil.

Considering the speed range and the geometric dimensions of the gas turbine rotor, the excentricity ratio ε for this application should not be larger than 0.43, according to (Zeidan *et al*, 1996). A balance quality grade G2.5 was selected from ISO 1940/1 to assure the balance quality of the rotor. So, a value of 0.2 was used for the excentricity ratio.

The stiffness and damping dynamic properties of the squeeze film damper were calculated using the theory presented in this paper. The values were obtained as a function of the rotational speed. The characteristics of the circumferential feeding groove were defined in order to provide good enough damping characteristics and a relatively small stiffness to the bearing. These characteristics are very important for aircraft gas turbines bearings. They allow a better isolation of vibrations, a more effective reduction of the forces transmitted to structural components, and an improvement in the overall system stability.

Figure 4 shows the calculated stiffness and damping properties of the studied squeeze film damper as a function of the rotational speed. The calculated stiffness values vary from 2.94×10^6 to 1.25×10^7 N/m. The calculated damping values vary from 2441.2 to 9719.7 Ns/m. The results are considered to be reasonable when compared to other works, (Dai *et al*, 2001) and (San Andrés and Santiago, 2004).
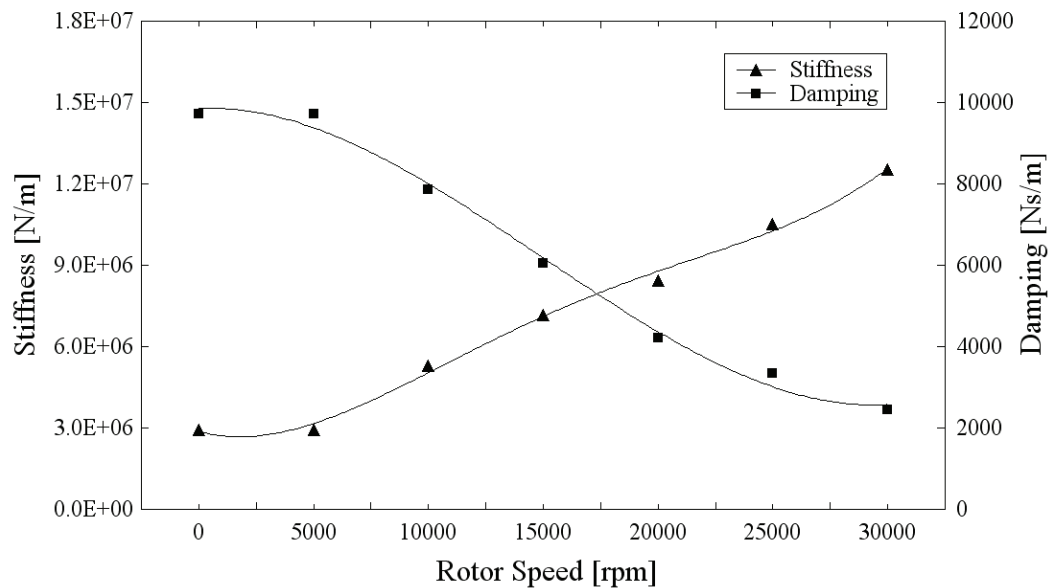

Figure 4. Calculated stiffness and damping dynamic properties of the studied squeeze film damper.

4. CONCLUSIONS

The stiffness and damping dynamic properties of the studied squeeze film damper were successfully calculated using the mathematical model presented in this paper. The dynamic performance of the squeeze film damper is attributed to two film lands and their interactions with the circumferential feeding groove. The geometrical properties of the circumferential feeding groove were carefully defined in order to provide good enough damping characteristics to the bearing. The groove contains a pressurized oil film which envolves the nonrotating outer ring of the angular contact ball bearing. This pressurized oil film is the main responsible for the stiffness attenuation.

Assuming that the z-axis of the main reference system is aligned to the rotor shaft, then, the calculated stiffness and damping values presented in this paper are related to the *x*- and *y*- directions, i.e., $K_{SFD}=K_{xx}=K_{yy}$ and $C_{SFD}=C_{xx}=C_{yy}$. Additionally, the stiffness and damping cross values, K_{xy}, K_{yx}, C_{xy} and C_{yx} are considered to be null.

From Figure 4, it can be observed that the calculated stiffness values increase with rotor speed but, on the other hand, the calculated damping values decrease with rotor speed. These behaviors can be explained by viscous losses in the lubricant. Additionaly, as the rotor speed increases, the temperature in the lubricant gets higher and the dynamic viscosity of oil decreases. The damping capacity of the bearing is reduced and its stiffness is increased.

It is very important to state that experimental measurements are necessary to validate the theoretical results presented in this paper. The mathematical modeling was carefully performed and the results obtained can be considered reasonable according to other author's works. Having the two bearings of the gas turbine properly characterized, a full rotordynamics analysis on the rotating system can be performed using the calculated stiffness and damping values. This way, the dynamic behavior of the rotating system can be predicted and vibration problems can be avoided.

ACKNOWLEDGEMENTS

REFERENCES

Arauz, G.L. and San Andrés, L.A., 1994, "Effect of a circumferential feeding groove on the dynamic force response of a short squeeze film damper", ASME, Journal of Tribology, Vol. 116, No. 2, pp. 369-377.

Arauz, G.L. and San Andrés, L.A., 1996, "Experimental study on the effect of a circumferential feeding groove on the dynamic force response of a sealed squeeze film damper", ASME, Journal of Tribology, Vol. 118, pp. 900-905.

Dai, X., Shen, Z., Wei, H., 2001, "On the vibration of rotor-bearing system with squeeze film damper in an energy storage flywheel", International Journal of Mechanical Sciences, Vol. 43, pp. 2525-2540.

San Andrés, L., 1992, "Analysis of squeeze film dampers with a central groove", Transactions of the ASME, Journal of Tribology, Vol. 114, No. 4, pp. 659-665.

San Andrés, L., Santiago, O., 2004, "Forced response of a squeeze film damper and identification of force coefficients from large orbital motions", ASME, Journal of Tribology, Vol. 126, pp. 292-300.

Siew, C.C., Hill, M., and Holmes, R., 2002, "Evaluation of various fluid film models for use in the analysis of squeeze film dampers with a central groove", Tribology International, Vol. 35, pp. 533-547.

Tan, Q., Chang, Y. and Wang, L., 1997, "Effect of a circumferential feeding groove on fluid force in short squeeze film dampers", Tribology International, Vol. 30, No. 6, pp. 409-416.

Tan, Q., Xaohua, L. and Dawei, Z., 1998, "Analytical study of the effect of a circumferential feeding groove on the unbalance response of a rigid rotor in a squeeze film damper", Tribology International, vol. 31, No. 5, pp. 265-270.

Tichy, T.A. and Bou-Said, B., 1991, "Hydrodynamic lubrication and bearing behaviour with impulsive loads", ASLE, Tribology Transactions, Vol. 34, No. 4, pp. 505-512.

Zeidan, F.Y., San Andres, L. and Vance, J.M., 1996, "Design and application of squeeze film dampers in rotating machinery", Proceedings of the 25th Turbomachinery Syposium, Gas Turbine Laboratory, A&M University College Station, Texas, pp. 169-188.

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