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Dynamic Characteristics of a Squeeze Film Damper used as Rear Bearing in a Single Spool Aeronautic Gas Turbine

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***Abstract*— *Squeeze film dampers are widely used in aeronautic gas turbines because they effectively absorb vibrations and lessen the stresses on the structural components. In this study, we calculated the stiffness and damping dynamic coefficients of a squeeze film damper with open ends and a circumferential oil-feeding groove. This squeeze film damper was used as a rear bearing in an aeronautic gas turbine designed to generate 5-kN of thrust under ISA conditions. Three different radial clearances were investigated to determine the optimal bearing design configuration for the application because the radial clearance of a squeeze film damper is a crucial element in determining its dynamic stiffness and damping coefficients. To provide superior performance and avoid issues, a rotordynamic analysis using the calculated stiffness and damping dynamic coefficients can be conducted to predict the vibratory behavior of the entire rotating assembly.*

I. INTRODUCTION

Aeronautic gas turbines are internal combustion engines that have an operating power rating ranging from small (100 kW) to large (180 MW). Therefore, they are suitable for electrical power generation or aircraft propulsion. Their primary advantage is their small weight and volume compared with other types of heat engines. Low thermal inertia that allows a full load in a short time is another benefit [1].

The aeronautic gas turbine investigated in this study was a high-performance turbojet designed to generate 5-kN of thrust under ISA conditions [2]. For instance, they can be used in military unmanned aerial vehicle applications. The entire project was developed in partnership with the Department of Aerospace Science and Technology (DCTA), which is a military organization of the Brazilian Air Force Command, and TGM Turbinas Ltda (TGM), with the financial support of the Financier of Studies and Projects (FINEP) and the National Fund for Scientific and Technological Development (FNDCT). The nominal rotation speed of the rotor was 28,150 rpm, although the normal operating range was between 80% and 100% of the nominal speed. It features a single spool system, air intake duct, five-stage axial compressor, direct-flow annular-type combustion chamber, single-stage turbine disk, and exhaust nozzle. Figure 1 shows the aeronautic gas turbine mounted on the test rig. The user-defined specifications were: weight should be approximately 650 N; the length and diameter of the circular section must not exceed 1.5 m and 0.35 m, respectively.

The bearings of an aeronautic gas turbine must be carefully designed to provide the required stiffness and dynamic damping characteristics to the rotating assembly to prevent vibration issues. The front bearing of this aeronautic gas turbine is composed of a deep groove ball bearing and a vibration-absorbing element. The dynamic stiffness and damping coefficients of the front bearing were calculated previously [3]. The rear bearing is composed of an specific designed squeeze-film damper with open ends and a circumferential oil feeding groove.

Fig. 1: Illustration of the aeronautic gas turbine investigated in this study on test rig.

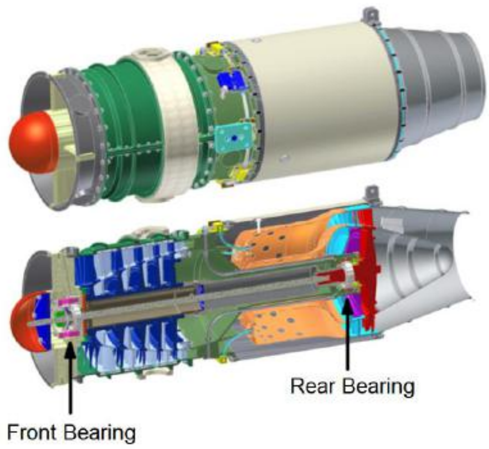

Fig. 2: Cross-section view of the studied aeronautic gas turbine with its main components.

Squeeze film dampers are continuously lubricated elements that support shafts and provide viscous damping in mechanical systems. In its simplest form, a squeeze film damper creates a lubricating oil film using a nonrotating bearing mounted in the housing with specified radial clearance. Owing to their capacity to suppress and isolate vibrations, lessen the stresses imparted to structural components, and eliminate non-synchronous instability, they are widely used in high-speed turbomachinery [4]. Figure 2 displays the cross-section of the aeronautic gas turbine, highlighting its front and rear bearings.

In this study, the dynamic stiffness and damping coefficients of the rear bearing of an aeronautic gas turbine were calculated. The rear bearing is composed of a squeeze film damper with open ends and a circumferential oil-feeding groove. The dynamic stiffness and damping coefficients have been evaluated using published mathematical models available in literature [5-7]. The circumferential feeding groove was considered a special damper in the analysis of fluid forces. The two adjacent oil film lands and their interactions with the circumferential feeding groove are responsible for the dynamic performance of the bearing. The calculations considered the effects of variations in pressure and fluid velocity within the groove. The radial clearances were analysed to determine the optimal bearing design configuration because the radial clearance of the squeeze film damper is the crucial factor that determines the dynamic stiffness and damping coefficients. The dynamic values of the stiffness and damping coefficients are subsequently used in a rotordynamic analysis to predict the vibratory behaviour of the aeronautic gas turbine rotating assembly to avoid issues and ensure superior performance [8].

II. LITERATURE REVIEW

Squeeze film dampers have been successfully developed to stabilize a wide variety of unstable rotating assembly units operating with tilt pad bearings. They are essential components in high-speed turbomachinery owing to their unique vibration energy dissipation advantages associated with the effective structural isolation of the mechanical components. Furthermore, squeeze film dampers can significantly increase the dynamic stability of rotating assemblies that exhibit instability due to their own configuration.

San Andres and Santiago have extensively studied the design and characterization of squeeze film dampers. They reported the experimental results of the damping and inertia coefficients considering various loading conditions [4]. An oil film's effective length was used to provide consistency between experimental and analytical results. Oil pressure field measurements reveal the appearance of the air intake, the effects of which increase the amplitude and frequency of dynamic bearing movements.

Siew et al. [5] compared several theoretical models for characterizing the properties of squeeze film dampers with circumferential feeding grooves. Four types of grooves and two types of lubricants were evaluated. The results were compared with experimental data to determine the suitable model for a particular circumferential groove configuration and lubricant type. Highly nonlinear fluid stiffness and damping coefficients were observed, and damping proved to be highly sensitive to oil viscosity and mass imbalance. A special model that considered the groove and the two oil lands proved to be efficient in predicting the vibratory behavior of a squeeze film damper with a shallow depth. However, the conventional two-land theory has proven suitable for squeezing film dampers with deep grooves.

Also, Siew et al. provided several useful instructions for the design and characterization of shallow- or deep-grooved squeeze film dampers.

Tan et al. [6,7] investigated a squeeze film damper model that was similar and applicable to the squeeze film damper used as the rear bearing of the aeronautic gas turbine examined here in this work. This squeeze film damper was composed of a circumferential feeding groove with open ends. They investigated the circumferential feeding groove as a special damper when analyzing the fluid forces. Therefore, the dynamic performance of the squeeze film damper is attributed to this special damper, the two adjacent oil lands and the interaction between the oil lands and groove. From this perspective, the dynamic effects in the groove were examined based on linearized Navier-Stokes equations. Variations in the velocity and pressure of the fluid in the feeding groove were considered in the calculation of the damping forces.

III. REAR BEARING MODELLING

Figure 3 shows schematic views of an arbitrary squeeze film damper with open ends and a circumferential feeding groove similar to the rear bearing of the aeronautic gas turbine investigated in this study. The theoretical model for determining the dynamic stiffness and damping coefficients of the squeeze film damper is based on the radial and tangential fluid forces acting on the bearing, according to the research carried out by Tan et al. [6] and [7]. The radial and tangential fluid forces are as follows:

$$F_r = \int_{-\frac{L}{2}}^{+\frac{L}{2}} \int_{\theta_1}^{\theta_2} p\cos\theta dz R d\theta \quad (1)$$

$$F_t = \int_{-\frac{L}{2}}^{+\frac{L}{2}} \int_{\theta_1}^{\theta_2} p\mathrm{sen}\theta dz R d\theta \quad (2)$$

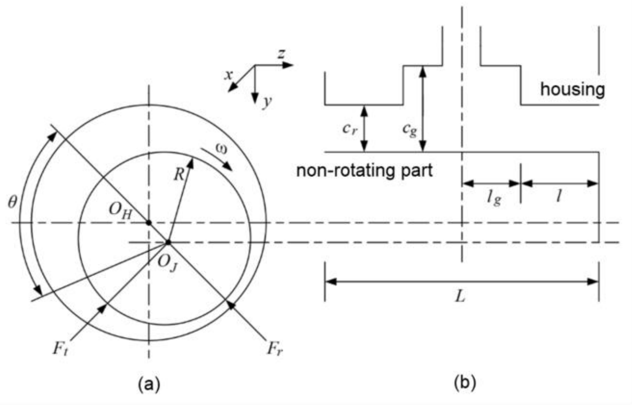

Fig. 3: Schematic of an arbitrary open-end squeeze film damper: (a) front view; and, (b) side view.

Assuming circular-centered orbits rotating at a constant speed, the radial and tangential fluid forces can be expressed using dimensionless force terms f_r and f_t, respectively.

$$F_r = B f_r \in \quad (3)$$

$$F_t = B f_t \in \quad (4)$$

where the dimensionless forces are given by

$$f_r = \frac{2+3\varphi+3C_1\varphi(1+2\varphi)}{2(1+\varphi)^3} I_3^{11} - Re\frac{2+3\varphi}{2(1+\varphi)^3}\left(\frac{I_1^{02}}{12}+\frac{I_2^{21}}{5}\in\right) \quad (5)$$

$$f_t = \frac{2+3\varphi+3C_1\varphi(1+2\varphi)}{2(1+\varphi)^3} I_3^{20} - Re\frac{2+3\varphi}{2(1+\varphi)^3}\left(\frac{I_1^{11}}{12}+\frac{I_2^{30}}{5}\in\right) \quad (6)$$

being that,

$$B = \frac{RL^3\eta\omega}{c_r^2}, \quad C_1 = \frac{\gamma^2(3-2\gamma)}{1+\left(\frac{Re}{10\gamma}\right)^2}, \quad \gamma = \frac{c_r}{c_g}, \quad \varphi = \frac{l_g}{l}, \quad Re = \frac{\rho\omega c_r^2}{\eta} \quad (7)$$

where R is the radius of the nonrotating bearing, L is the total length of the oil film, η is the absolute dynamic viscosity of the oil, ω is the precession rate, c_r is the radial clearance, c_g is the depth of the circumferential feeding groove, l_g is the length of half of the circumferential feeding groove, l is the length of an adjacent oil film, and ρ is the density of the lubricating fluid. In the expressions above, the terms I_n^{lm} are defined by Booker integrals:

$$I_n^{lm} = \int_{\vartheta 2}^{\vartheta 1} \frac{\mathrm{sen}^l\vartheta\cos^m\vartheta}{(1+\in\cos\vartheta)^n} d\vartheta \quad (8)$$

For the π-film model, that is, $\theta_1 = 0$ and $\theta_2 = \pi$, we have

$$I_3^{11} = \frac{2\in}{r^4}, \quad I_r = \left(\frac{I_1^{02}}{12}+\frac{I_2^{21}}{5}\in\right) = \frac{\pi(1-r)}{\in^2 r}\left[\frac{1}{12}-\frac{1}{5}(1-r)\right] \quad (9)$$

$$I_3^{20} = \frac{\pi}{2r^3}, \quad I_t = \left(\frac{I_1^{11}}{12}+\frac{I_2^{30}}{5}\in\right) = \frac{19}{60\in^2}\left(2\in - ln\left|\frac{1+\in}{1-\in}\right|\right) \quad (10)$$

where r = $(1-\in^2)^{\frac{1}{2}}$. Therefore, assuming circular-centred orbits and the π-film model, the dynamic stiffness and damping coefficients of the squeeze film damper can be calculated as:

$K_{SFD}=\frac{F_r}{e}$ (stiffness of the squeeze film damper) (11)

$C_{SFD}=\frac{F_t}{e\omega}$ (damping of the squeeze film damper) (12)

Considering circular-centered orbits, e is the eccentricity given by the length $O_H - O_J$ and the eccentricity ratio $\in$ is given by $\in=\frac{e}{c_r}$.

IV. RESULTS

As shown in Figure 4, the squeeze film damper analyzed in this study consists of a Barden 206(HJH) angular contact ball bearing and a centering spring. Lubrication was carried out using Aeroshell® 500 oil. The oil temperature reaches 100°C because this oil circulates close to the combustion chamber, where intense heat exchange occurs. The centering spring applied a preload of 150 N to the outer ring of the bearing, and the oil supply pressure, p_s, in the squeeze film damper was adjusted to approximately 20 psi, based on practical industrial experience.

The bearing inner ring was mounted with interference fit on the rotating assembly shaft. The axial centering spring ensured optimal contact between the balls and raceways. The outer ring acts as a nonrotating bearing, forming a squeeze film damper oil film. There is a radial clearance between the outer ring surfaces and housing that is filled with Aeroshell® 500 oil. The circumferential feeding groove acts as a circumferential oil supply that divides the total film into two adjacent film lands. Three feeding holes were used to supply oil to the squeeze film damper. The three oil-feeding holes were 120° apart. O-rings or sealing seals were not added to the lateral ends of the squeeze film damper because the rear bearing presented small axial slips owing to the thermal expansion behavior of the entire set. The geometric properties that characterize the studied squeeze film damper are listed in Table 1. The radial clearance was not specified in this table because three case studies were carried out, using three different values of the radial clearance to determine the optimal squeeze film damper design according to the verified characteristics.

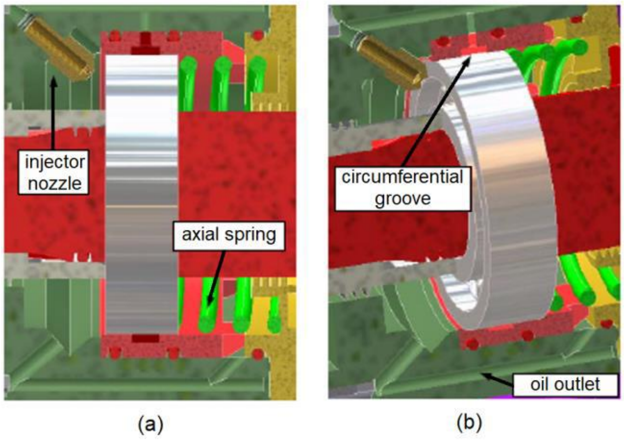

Fig. 4: Squeeze film damper used as rear bearing (a) side view; and, (b) perspective view.

Tab. 1: Geometric properties of the squeeze film damper.

Property	Value
Groove depth, c_g	0.003697 m
Adjacent oil film width, l	0.005 m
Half groove oil film width, l_g	0.003 m
Total oil film width, L	0.016 m
Non-rotating bearing radius, R	0.031 m

Radial clearance is one crucial factor that determines the stiffness and damping dynamic coefficients of the squeeze film damper, which provides optimal values depending on the type of application. The geometric properties listed in Table 1 were held constant for three clearance values of c_r: 0.08, 0.12, and 0.1905 mm, respectively. Using the ISO 1940/1 standard's balance quality grade, G1.0, for the manufacturing process of the rotating assembly, the eccentricity ratio of $\in$=0.2 was used in the calculations. For zero rotation speed, the same stiffness and damping values as for 5,000 rpm were considered to account for the values used in the rotordynamic analysis of the entire rotating assembly.

Table 2 lists the calculated stiffness and damping dynamic coefficients for the rear bearing, considering a squeeze film damper radial clearance of 0.08 mm. The stiffness and damping dynamic coefficients of the squeeze film damper vary according to the rotation speed and oil temperature. According to Table 2, for the radial clearance of 0.08 mm, the stiffness of the rear bearing ranged from 9.4270E+6 to 2.6193E+7 N/m, and the damping ranged from 8.111 to 32.668 Ns/m. For the radial clearance of 0.12

mm, according to Table 3, the stiffness of the rear bearing ranged from 2.935E+6 to 1.2518E+7 N/m, and the damping coefficients ranged from 2,441 to 9,720 Ns/m. Considering the radial clearance of the squeeze film damper of 0.1905 mm, according to Table 4, the stiffness coefficients of the rear bearing ranged from 8.282E+5 to 6.3821E+6 N/m, and the damping coefficients ranged from 635.9 to 2,449 Ns/m.

Tab. 2: Stiffness and damping dynamic coefficients of the studied squeeze film damper considering a radial clearance of 0.08 mm.

Rotation Speed [rpm]	Oil Temperature [°C]	Kinematics Viscosity [m²/s]	Dynamic Viscosity [Pa.s]	Stiffness Coefficients [N/m]	Damping Coefficients [Ns/m]
0	25	3.2755E-5	0.0329	9.4270E+6	32,668
5,000	60	1.9752E-5	0.0199	9.4270E+6	32,668
10,000	70	1.6037E-5	0.0161	1.6085E+7	26,418
15,000	80	1.2322E-5	0.0124	2.0129E+7	20,352
20,000	90	8.6065E-6	0.0086	2.1340E+7	14,137
25,000	95	6.7489E-6	0.0068	2.4363E+7	11,204
30,000	100	4.8913E-6	0.0049	2.6193E+7	8,111

As shown in Figure 5, the values of the calculated stiffness coefficients increased with the rotation speed of the rotating assembly for the three radial clearances evaluated. In contrast, in Figure 6, it is observed that the values of the damping coefficients decrease with the rotation speed of the rotating assembly for the three clearances. This is because of the viscous losses in the lubricant. As the speed of the rotating assembly increases, the temperature of the lubricant increases, and the dynamic viscosity of the oil decreases. The damping capacity of the bearing was thus reduced, and its stiffness increased. Assuming that the *-z* axis of the main reference system is aligned with the axis of the rotating assembly, the values of the calculated stiffness and damping coefficients are related to the *-x* and *-y* directions. That is, $K_{SFD} = K_{xx} = K_{yy}$ and, in the same manner, $C_{SFD} = C_{xx} = C_{yy}$. The cross-coupled terms K_{xy}, K_{yx}, C_{xy} and C_{yx} were considered to be null.

Tab. 3: Stiffness and damping dynamic coefficients of the studied squeeze film damper considering a radial clearance of 0.12 mm.

Rotation Speed [rpm]	Oil Temperature [°C]	Kinematics Viscosity [m²/s]	Dynamic Viscosity [Pa.s]	Stiffness Coefficients [N/m]	Damping Coefficients [Ns/m]
0	25	3.2755E-5	0.0329	2.935E+6	9,720
5,000	60	1.9752E-5	0.0199	2.935E+6	9,720
10,000	70	1.6037E-5	0.0161	5.303E+7	7,856
15,000	80	1.2322E-5	0.0124	7.159E+7	6,056
20,000	90	8.6065E-6	0.0086	8.440E+7	4,216
25,000	95	6.7489E-6	0.0068	1.0523E+7	3,352
30,000	100	4.8913E-6	0.0049	1.2518E+7	2,441

Tab. 4: Stiffness and damping dynamic coefficients of the studied squeeze film damper considering a radial clearance of 0.1905 mm.

Rotation Speed [rpm]	Oil Temperature [°C]	Kinematics Viscosity [m²/s]	Dynamic Viscosity [Pa.s]	Stiffness Coefficients [N/m]	Damping Coefficients [Ns/m]
0	25	3.2755E-5	0.0329	8.282E+5	2,449
5,000	60	1.9752E-5	0.0199	8.282E+5	2,449
10,000	70	1.6037E-5	0.0161	1.6905E+6	1,978
15,000	80	1.2322E-5	0.0124	2.6048E+6	1,529
20,000	90	8.6065E-6	0.0086	3.5563E+6	1,072
25,000	95	6.7489E-6	0.0068	4.8899E+6	859.6
30,000	100	4.8913E-6	0.0049	6.3821E+6	635.9

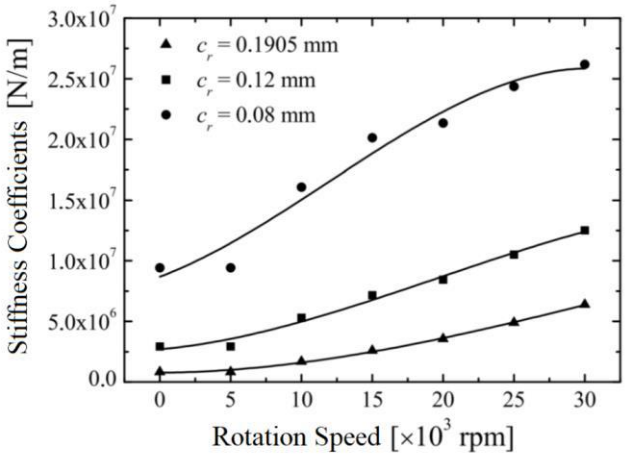

Fig. 5: Dynamic stiffness coefficients considering three values of radial clearance for the squeeze film damper.

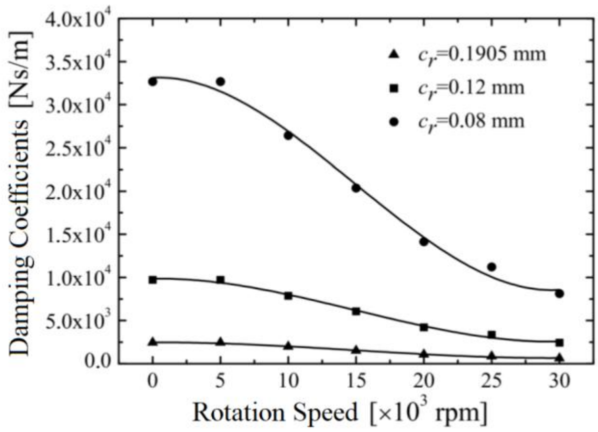

Fig. 6 Dynamic damping coefficients considering three values of radial clearance for the squeeze film damper.

V. CONCLUSIONS

It is observed that the radial clearance strongly affects the values of the dynamic stiffness and damping coefficients for the aeronautic gas turbine rear bearing. The stiffness coefficients increase with the increase in the rotation speed of the rotating assembly. In contrast, the damping coefficients decrease with the increase in the rotation speed of the rotating assembly. This is because of the viscous losses in the lubricant as the temperature of the lubricant increases with higher rotation speeds. The dynamic stiffness and damping cross-coupled coefficients were considered null for the application. For a zero-rotation speed, the same stiffness and damping values used for 5,000 rpm were adopted. The calculated stiffness and damping dynamic coefficients for the three different radial clearances can be used in a rotordynamic analysis of the entire rotating assembly to predict its vibratory behavior to ensure superior performance and prevent issues.

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