Some Aspects Regarding the Modeling of Highly Pressurized Squeeze Film Dampers

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Abstract: Squeeze film dampers (SFD) have been used for many years to control vibration in gas turbines and other high-speed rotating machinery. The overall objective of this paper is to present the analysis and validation of a dynamic model of a highly pressurized squeeze film damper. Predictions based on the model are utilized in the rotor dynamic modeling of a rotor supported on a SFD.

Key Words: Squeeze film dampers (SFD), hydrodynamic bearings, rotor dynamics

1. INTRODUCTION

During the last decades, we are witnessing an ongoing race to improve the performances of the turbo machinery. Improving the dynamic performances and lowering the specific fuel consumption are permanent goals of engine designers and manufacturers. Turbo machines tend to be lighter and lighter and the clearances thinner and thinner and these led to increase vibration sensitivity. Much effort and resources have been spent for the development of improved balancing techniques, but sometimes it is not possible to achieve the desired low level of vibrations by balancing alone. Rotor assemblies, tough well balanced initially degrade with continuous service time. Consequently, continuous efforts have been continuously made to improve dampers’ design.

One of the commonly used types of dampers is the squeeze film damper (SFD). Many theoretical and experimental studies have been dedicated to this type of device but no “standard” satisfactory approach is yet available. This paper presents cases when the classical Reynolds theory was successfully applied to predict properties of a highly pressurized SFD. The main analytical solution of the short SFD is implemented on an experimental damper. The theoretical unbalance response of a rotor supported on the SFD is compared against the experimental results.

2. DESCRIPTION OF THE EXPERIMENTAL SET-UP. THE MATHEMATICAL MODEL OF THE SYSTEM. RESULTS AND DISCUSSIONS

SFDs have been intensively studied during the last decades and an extensive review is beyond the purpose of this paper. Review papers, like, for example, the studies of Moore 0 and Zeidan et. al. 0 present some key information in the field.

For design purposes it is very important to know the values of the forces developed in the damper. The pressures in the oil film can be determined from the fluid flow equations. For the classical lubrication conditions, the Navier Stokes equations and the continuity
equation can be simplified and combined into the Reynolds equation, which can be solved for the pressure distribution. Continuous efforts have been dedicated to this subject and new numerical methods have been adopted, Refs. 0, 0. The combined affects of fluid inertia and gaseous phenomena, is still subject of open research.

The order of magnitude of the inertia forces is indicated by the Reynolds number, which, for SFD lubrication is expressed as

$$\text{Re}^* = \frac{\rho \omega c^2}{\mu}$$  \hspace{1cm} (1)

Inertia of the oil can be neglected if Reynolds number is smaller than 1 and, consequently (if the gaseous phenomena are also negligible) the forces in the oil film can be obtained form the Reynolds equation.

The effects of fluid inertia have been studied for several decades, Refs. 0 - 0 and many researchers also confirmed the interrelation between the inertia effects and cavitation, Ref. 0. On the other hand, cavitation in hydrodynamic oil film bearings (the gaseous phenomena which often appears in the divergent are of the bearing) is quite complex and it depends upon the supply system, the sealing devices and the air content of the oil. Cavitation phenomena are still an open research field, and scientists direct their efforts both toward the experimental studies and toward the numerical simulation procedures, 0, however the calculations are quite complex and, for bearings with important cavitation and inertia effects, in many instances, the agreement between theory and experiments may be improved.

The research presented herein shows that for dampers with relatively low clearances (allowing the Reynolds number to remain low even for moderately high speeds) when cavitation was avoided using high pressures both in the supply and in the drain lines, the classical Reynolds theory provided an accurate simulation.

A squeeze film damper was tested, which consists of 2 lands, narrow enough to allow for the analytical short bearings solutions to be used without significant errors (for each of them the ratio between width and diameter is about 0.12) provided that the eccentricity is not very high and the pressure in the diverging area can be determined with sufficient accuracy.

However, as mentioned above, the pressure in the negative squeeze area, and, actually, the nature of the cavitation itself depends upon the local flow conditions within the damper; a complete and accurate calculation requires modeling of the supply and drain systems, sealing devices and the air content of the oil, it is still extremely time consuming and in many instances the agreement between the numerical results and the real values is not accurate enough.

In the experiments presented herein, the Reynolds number is below 0.3 and the pressure in the diverging area of the dampers is controlled via the supply and drain systems, which are both pressurized; the supply pressure was set to 760kPa (110psi) and the gage pressure in the drain line is 170kPa (25psi); moreover, the diameters of the pipes are quite large and they can rapidly provide the bearing with large amounts of oil. The supply orifice is located in the central groove which separates the two bearing lands and the drain holes are located in grooves at the two ends of the bearing.

Consequently, the radial and tangential components of the forces produced in the oil film are calculated according to the short bearings theory which gives 0,

$$F_r = \eta r \left( \frac{1}{c} \right)^2 \left[ 2\nu \frac{\epsilon^2}{(1-\epsilon^2)^2} + \frac{\pi}{2} \frac{\left( 1 + 2\epsilon^2 \right) \epsilon}{(1-\epsilon^2)^{1/2}} \right]$$  \hspace{1cm} (2)

and

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where, for the reasons explained above, the pressure in the diverging area of the bearing should be roughly constant in the circumferential direction and its average value should be approximately equal to the average of the supply and the drain pressures, so \( p_0 = 460 \text{kPa} \).

The test damper was installed on an experimental device dedicated to squeeze film damper testing. The test rig consists of a very rigid rotor (with no flexural critical speed in the operating domain) which is supported in the test damper at one end, while the other end is installed in a spherical joint-type support. The shaft is unbalanced by weights placed on a disk located at the damper end. The motion of the shaft is measured by non-contact proximity probes. A schematic view of the test rig is shown in Fig.1. Details regarding the test rig are given in Refs. 0-0.

![Fig. 1 – Test rig schematics](image)

The dynamic of the shaft is studied using the angular momentum theorem which is written with respect to the spherical joint (which is a fixed point of the rotor). The general moment of momentum expression with respect to the fixed point was written in terms of the body axes,

\[
H_0 = H_X i + H_Y j + H_Z k
\]

with

\[
\begin{align*}
H_X &= J_X \omega_X - J_{XY} \omega_Y - J_{XZ} \omega_Z \\
H_Y &= J_Y \omega_Y - J_{YZ} \omega_Z - J_{YX} \omega_X \\
H_Z &= J_Z \omega_Z - J_{ZX} \omega_X - J_{ZY} \omega_Y
\end{align*}
\]

Next, \( XOY \) is the longitudinal plane of symmetry of the shaft (passing through the unbalancing weight placed on the unbalancing disk) so,

\[ J_{XZ} = J_{ZX} = J_{YZ} = J_{ZY} = 0 \]

and

\[
\begin{align*}
H_X &= J_X \omega_X - J_{XY} \omega_Y \\
H_Y &= J_Y \omega_Y - J_{YX} \omega_Z \\
H_Z &= J_Z \omega_Z
\end{align*}
\]

where \( J_X, J_Y, J_Z \) and \( J_{XY} \) are the inertia moments and the inertia products with respect to the body-linked axes of the shaft, while the vector \( \omega \) is the angular speed of the shaft, with respect to a fixed frame.
Now, the general form of the theorem of the angular momentum in terms of the body-linked frame is

$$\frac{d \mathbf{H}_0}{dt} = \frac{\partial \mathbf{H}_0}{\partial t} + \mathbf{\omega} \times \mathbf{H}_0 = \mathbf{M}_0$$

where

$$\frac{\partial \mathbf{H}_0}{\partial t} = \frac{d H_x}{dt} \mathbf{i} + \frac{d H_y}{dt} \mathbf{j} + \frac{d H_z}{dt} \mathbf{k}$$

and

$$\frac{d H_x}{dt} = J_x \frac{d \omega_x}{dt} - J_{xy} \frac{d \omega_y}{dt}$$
$$\frac{d H_y}{dt} = J_y \frac{d \omega_y}{dt} - J_{xy} \frac{d \omega_x}{dt}$$
$$\frac{d H_z}{dt} = J_z \frac{d \omega_z}{dt}$$

so the moment of momentum equations in terms of the body-fixed coordinate system becomes

$$\begin{cases}
J_x \frac{d \omega_x}{dt} + (J_z - J_y) \omega_x \omega_y - J_{xy} \left( \frac{d \omega_y}{dt} - \omega_x \omega_z \right) = M_x \\
J_y \frac{d \omega_y}{dt} + (J_x - J_z) \omega_z \omega_y - J_{xy} \left( \frac{d \omega_x}{dt} + \omega_y \omega_z \right) = M_y \\
J_z \frac{d \omega_z}{dt} + (J_y - J_x) \omega_y \omega_z - J_{xy} \left( \omega_x^2 - \omega_y^2 \right) = M_x
\end{cases}$$  \(5\)

Equations (5) are next written in the canonical form and they are numerically integrated together with the kinematical equations to obtain the motion of the shaft. The moments of the damper forces acting on the shaft are calculated using equations (2) and (3). The other forces within the system are calculated using the data of the rig. The integration was performed using a 4th order Runge Kutta solver.

![Fig. 2 – Numerical and Experimental Results](image-url)
3. CONCLUSION

The values of the forces within a squeeze film damper depend upon cavitation and inertia effects which still rise serious computational difficulties. However, cavitation may be controlled by the pressure levels in the supply and drain lines and the classical Reynolds theory may still be successfully used for modeling when the Reynolds number is below unity as well.

This paper presented a study of a narrow SFD where pressure in the diverging area is controlled by the high pressure supply and drain lines; the unbalance responses were calculated using the analytical short bearing solution. The numerical solutions agree well with the measured data provided that the eccentricity of the bearing is below 40%, however, at higher eccentricities the force predictions based on the short bearing theory may not be accurate enough. Numerical calculations of the fluid film forces are believed to lead to better unbalance response predictions at higher bearing eccentricities.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$c$</td>
<td>Clearance of the bearing</td>
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<tr>
<td>$F_R$</td>
<td>Radial fluid film force</td>
</tr>
<tr>
<td>$F_T$</td>
<td>Tangential fluid film force</td>
</tr>
<tr>
<td>$l$</td>
<td>Bearing width</td>
</tr>
<tr>
<td>$p_0$</td>
<td>Parameter that describe the pressure in the diverging area</td>
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<tr>
<td>$r$</td>
<td>Radius of SFD journal</td>
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<tr>
<td>$\varepsilon$</td>
<td>Relative eccentricity</td>
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<tr>
<td>$\eta$</td>
<td>Oil viscosity</td>
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<tr>
<td>$\nu$</td>
<td>Precession angle of the shaft</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Angular speed of the shaft</td>
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REFERENCES