A Brief Discussion Regarding Types of Cavitation in Squeeze Film Dampers and Cavitation Effects

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Abstract: *Squeeze film dampers (SFD) are probably the most used shaft control devices in aircraft jet engines; SFDs consist in oil films, elastic elements and various antirotational devices that tune the stiffness and damping of the shafts' supports and consequently adjust the lateral dynamics of the shaft. Fluid layers in SFDs are usually thin, hence the modeling can often be done using the Reynolds' theory,; however, some of the main features of the film, namely the behavior of the fluid in the divergent, negative squeeze area, where discontinuities may appear in the liquid, are still subject to intense research. This paper will discuss some aspects regarding the types of cavitation that appear in squeeze film dampers and some of the effects of cavitation on the SFDs.*

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

1. INTRODUCTION

Stability and appropriate behavior of turbomachinery is only possible when shaft's dynamics is adequate. The dynamics of rotors, on the other hand, depends upon the properties of the shaft, (mass distribution and elasticity) and upon the properties of the supports of the shaft.

Changes in stiffness and damping of the shaft supports can modify significantly the dynamic of the machine; turbomachinery that operate on land can be, theoretically at least, be equipped with a quite wide variety of bearings and shaft's mounts can be design without minimal weight concerns; moreover, since, in many cases, size requirements are not extremely severe, there are various methods of adding stiffness and damping, from which the designer can choose.

However, the rotors of the aircraft propulsion systems must be installed on ball bearings, because of safety concern related to failure of the bearings.

Moreover, both space and weight requirements for aircraft propulsion systems are very severe so design selections are limited. At present, squeeze film dampers, Fig.1, and mechanical devices within the squeeze film dampers are the only practical way to adjust the stiffness and damping of propulsion systems shafts.

A squeeze film damper basically consists in a layer of oil that separates two surfaces (like in any regular bearing), however, the spin of the surfaces is blocked, so the oil film is only affected by the precession and nutation of the shaft, while the spin rotation is supported by the ball bearings.

The spin of the surfaces separated by the oil film is prevented by various rigid or elastic devices, e.g. a pin.

SFD research has been reviewed, for example, in Refs. [1], [2]. The current paper presents a brief discussion regarding the cavitation in SFD and its effect upon the dynamic of SFD supported rotors.

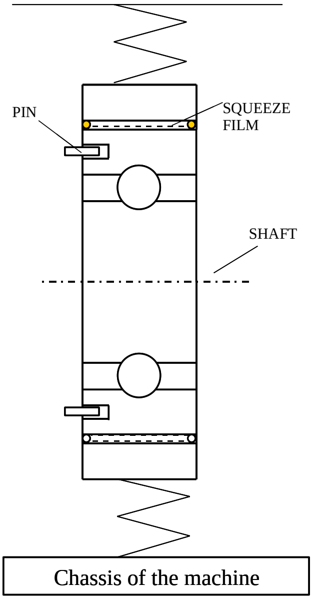

Fig. 1 SFD with springs

2. ANALYSIS, RESULTS AND DISCUSSIONS

The most important output parameter for dampers is the load and its variations with the operating conditions, most importantly the variations of the load with respect to displacement and velocity.

As it is well known, when classical lubrication conditions are valid, the pressures developed within the thin fluid films in SFDs are described by the Reynolds equation,

$$\frac{\partial}{\partial\theta'}\left[\left(1+\varepsilon\cos\theta'\right)^3\frac{\partial p}{\partial\theta'}\right]+r^2\frac{\partial}{\partial y}\left[\left(1+\varepsilon\cos\theta'\right)^3\frac{\partial p}{\partial y}\right]= 6\eta\left(\frac{r}{c}\right)^2\left(2\dot{\nu}\varepsilon\sin\theta'+2\dot{\varepsilon}\cos\theta'\right) \quad (1)$$

what matters is not that much the extent of the cavitation but the pressure in the divergent area of the bearing.

Many numerical methods have been used for solving Eq.(1), see for example; Refs. [3] – [6]. Theoretical pressure distribution typically shows that pressure increases in the positive squeeze domain, however, in the negative squeeze film domain pressure decreases significantly and it can reach negative values, which is practically impossible, because fluids cannot withstand tension.

Figure 2 shows a typical pressure distribution in a damper with 6 cm radius and 1 cm width, open to atmosphere at the side boundaries, for 0.25 relative eccentricity, 500 rad/s precession speed and for $\dot{\varepsilon}=12s^{-1}$.

Fluids have a quite reduced capacity to withstand negative pressures, therefore, in many cases, film ruptures appear in the divergent zone of bearings and vapors and gases occupy the area.

Many numerical algorithms can predict the extent of the gaseous domain within a bearing, see, for example, Refs. [4] and [7], however (at the design stage, at least), making an accurate estimation of the divergent area pressure has a much larger impact on the accuracy of the bearing's performance estimation than calculating the position of the oil film rupture and reformation boundaries with superior accuracy. This is especially valid for highly pressurized bearings.

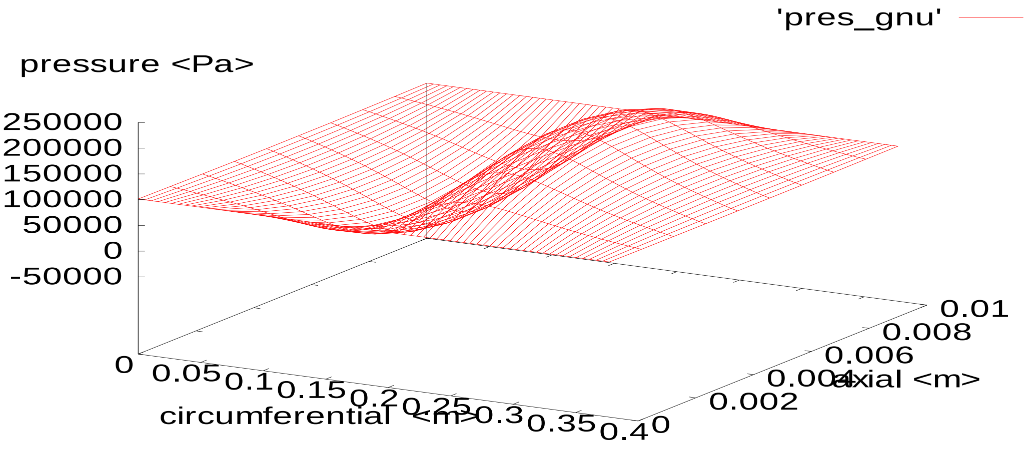

Fig. 2 Typical pressure distribution in a SFD

The divergent zone pressure in a bearing depends on the geometry, operating parameters, such as velocity and clearance, but also of the properties of the fluid, and, also, on a very large extent, it depends on the supply pressure, drain pressure and sealing. A detailed discussion of cavitation in bearings exceeds the space allocation of this paper; a few ideas will be briefly discussed below, however, details can be found in Refs. [1], [2], [4], [7], [8], [9] and [10].

Usually, when the bearing is not very pressurized (moderate supply pressure), but is very thigh sealed so no air entrainment is possible, the divergent area zone of the bearing is filled with very low pressure air (normally ingested in the oil) and in extreme cases, with saturate vapors of the lubricant.

This type of cavitation, called vapor cavitation, is extremely dangerous for bearings, because the cavitation area is unstable and the extremely large pressures appear when the vapor bubble collapses and typically destroy the bearing.

When, again, the bearing is not very pressurized (moderate supply pressure), but air entrainment is possible, the divergent area of the bearing is filled with gases (mostly air) at a pressure close (in many cases slightly below) to environmental pressure. This represents normal operating conditions for a bearing.

Rising significantly oil pressures can eliminate cavitation; this is especially valid when the drain pressure is also high and high amounts of oil are available in the pipes such as any potential pressure that decreases in the bearings divergent area can be rapidly compensated.

This type of operation is also safe, and it brings advantages, due to the combination of hydrodynamic and hydrostatic effects; however, it comes at the expense of more complex supply, drain and sealing systems.

The gaseous phenomena in SFD are not different from the gaseous phenomena in other types of liquid film bearings. What is perhaps specific to SFDs as compared to other types of bearings is the bubbly mixture operation occurring more frequently in SFDs. However, this appears mostly in dampers when the supply pressure and flow rate are rather low and environmental gas entrainment is significant; none of these, yet, is significant for the aircraft engines.

The subsequent part of this presentation will address the effects of the negative squeeze area pressure on the dynamics of rotors supported in SFD. Since the vapor cavitation is an undesirable operation, hence eliminated at the design stage, this condition will not be addressed.

The analysis will only deal with the cases of gaseous cavitation (with the negative squeeze pressure equal to the atmospheric pressure) and of pressurized bearings.

The SFD effects will be illustrated by the calculated dynamic response to the unbalance of a pinned rigid rotor, similar to the one presented in Refs. [11] to [13], and schematically presented in Fig. 3.

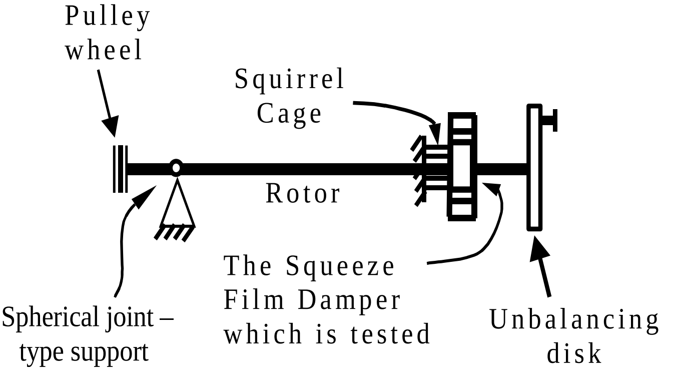

Fig. 3 – Experimental setup schematics

The equations of motion for such a fixed point rotor can derive using the moment of momentum equation, written with respect to the fixed point.

$$\frac{d\,\mathbf{H_0}}{dt} = \frac{\partial \mathbf{H_0}}{\partial t} + \boldsymbol{\omega} \times \mathbf{H_0} = \\ = \frac{dH_X}{dt}\mathbf{i} + \frac{dH_Y}{dt}\mathbf{j} + \frac{dH_Z}{dt}\mathbf{k} + \boldsymbol{\omega} \times \mathbf{H_0} = \mathbf{M_0} \tag{2}$$

where OXYZ are body-linked axes, and *XOY* is the longitudinal plane of symmetry of the shaft passing through the unbalancing weight so, $J_{XZ} = J_{ZX} = J_{YZ} = J_{ZY} = 0$ and the moment of momentum equations in terms of the body-fixed frame becomes

$$\begin{cases} J_X \dfrac{d\omega_X}{dt} + (J_Z - J_Y)\omega_Z\omega_Y - J_{XY}\left(\dfrac{d\omega_Y}{dt} - \omega_X\omega_Z\right) = M_X \\ J_Y \dfrac{d\omega_Y}{dt} + (J_X - J_Z)\omega_X\omega_Z - J_{XY}\left(\dfrac{d\omega_X}{dt} + \omega_Y\omega_Z\right) = M_Y \\ J_Z \dfrac{d\omega_Z}{dt} + (J_Y - J_X)\omega_Y\omega_X - J_{XY}\left(\omega_X^2 - \omega_Y^2\right) = M_Z \end{cases} \tag{3}$$

A fourth order Runge-Kutta solver was used to integrate Equations (3) and the kinematical equations. The unbalance response of the pinned rotor is displayed in Fig.4, for various values of the pressure in the negative squeeze area. The results show that the pressure in the divergent area of the bearing has a significant effect upon the dynamic of the shaft; moreover, for the rotor analyzed here, the effect drops significantly if the pressure falls below a certain limit; moreover, a maximum value seems to exists above which further increases of pressure no longer generate significant changes in the response.

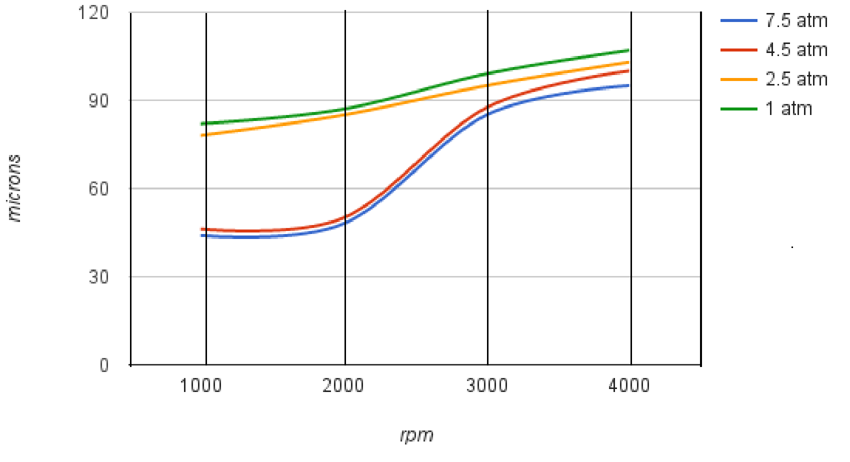

Fig. 4 – Unbalance responses

3. CONCLUSION

The forces developed in a SFD depend significantly on the pressure in the divergent area of the damper, also known as the “negative squeeze” area. Vapor and gaseous cavitation, as well as bubbly mixtures can appear in the divergent area of SFDs and the pressure in the negative squeeze area varies accordingly, from near vacuum to around 1 bar. Cavitation can be avoided by pressurizing the oil film, and when both inlet and outlet pressures are (significantly) above the surrounding atmospheric pressure, the oil film can be continuous, and the pressure in the divergent zone of the bearing is usually above 1 bar, the actual value being imposed to the supply-drain system, geometry and sealing. Pressure in the divergent area of the bearing has a significant effect upon the dynamic of the shaft; moreover, for the rotor analyzed here, the effect drops significantly if the pressure falls below a certain limit, moreover, a maximum value seems to exists above which further increases of pressure no longer generate significant changes in the response.

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REFERENCES

[1] D. F. Moore, A Review of Squeeze Films, *Wear*, Vol. **8**, pp.245-263, 1965.

[2] F. Zeidan, L. A. San Andres and J. M. Vance, *Design and Application of Squeeze Film Dampers in Rotating Machinery*, Proceedings of the Twenty-Fifth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College station, TX 169-188, 1996.

[3] S. Cioc, F. Dimofte and T. G. Keith, Application of the CE/SE Method to Wave Journal Bearings, *Tribology Transactions*, **46**, No. 2, pp.179-186,2003a.

[4] D. Vijayaraghavan and T. G. Jr. Keith, An Efficient, Robust and Time Accurate Numerical Procedure Applied to a Cavitation Algorithm, *ASME Journal of Tribology*, Vol. **112**, No.1, pp. 44-51, 1990.

[5] J. M. Vance, *Rotordynamics of Turbomachinery*, John Wiley and Sons, New York, 1988.

[6] V. N. Constantinescu, S. Dănăilă and S. Găletuşe, *Fluid Dynamics in Turbulent Regime*, The Publishing House of the Romanian Academy (in Romanian), 2008.

[7] A. F. Cristea, *Analysis of Thermal Effects in Circumferential Groove Journal Bearings with Reference to the Divergent Zone*, PhD Thesis, Universite de Poitiers and The "Politehnica" University of Bucharest, 2012.

[8] L. A. San Andres and J. M. Vance, Experimental Measurements of the Dynamic Pressure Distribution in a Squeeze-Film Bearing Damper Executing Circular-Centered Orbit, *ASLE Transactions*, Vol. **30**, No.3, pp.373-383, 1987.

[9] J. Ellis, J. B. Roberts and A. H. Sianaki, A Comparison of Identification Methods for Estimating Squeeze-Film Damper Coefficients, *ASME Jour. of Trib*., **110**, pp.119-127, 1988.

[10] F. Zeidan and J. M.,Vance, Experimental Investigation of Cavitation Effects on the Squeeze Film Force Coefficients, *Proceedings of the 12th Biennial ASME Conference on Mechanical Vibration and Noise*, Montreal, Canada, DE-18-1, pp.237-242, 1989a.

[11] D. P. Fleming, Improved Dual Clearance Squeeze Film Damper for High Loads, in *Proc. of the 3rd Int. Conf. on Motion and Vibration Control, Chiba, Japan*, September 1-6, 1996.

[12] L. Moraru, *Numerical Predictions and Measurements in the Lubrication of Aeronautical Engine and Transmission Components*, PhD. Dissertation, University of Toledo, Ohio 43606 (available from http://il.proquest.com/), 2005.

[13] L. Moraru, T. Keith, F. Dimofte, S. Cioc and D. P. Fleming, Dynamic Modeling of a Dual Clearance Squeeze Film Damper – Part I: Test Rig and Dynamic Model with one Damper, *STLE Trib Trans*, **46**, 2, pp. 170-178, 2003.