Saravanakumar et al. 2016. *Int. J. Vehicle Structures & Systems*, 8(4), 246-249
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.8.4.11

International Journal of Vehicle Structures & Systems
Available online at www.maftree.org/eja

Determination of Damping Coefficient of a Bump Foil Squeeze Film Damper

G. Saravanakumar[a,b], L. Ravikumar[a,c], Raghu Yogaraju[a,d], C. Shravankumar[a,e] and V. Arunkumar[f]

[a]*Dept. of Mech. Engg., BMS College of Engg., Bangalore, India*
[b]*Corrresponding Author, Email: gurusaravana@yahoo.co.in*
[c]*Email: lrkmech@gmail.com*
[d]*Email: raghu.yogaraju@gmail.com*
[e]*Email: shravmechie@gmail.com,*
[f]*Dept. of Mech. Engg., Dr. Ambedkar Institute of Technology, Bangalore, India*
Email: arun51149@gmail.com

ABSTRACT:

The present investigation is aiming to get better squeeze film dampers which are normally used in high speed jet engines to minimize small amplitude large force vibrations. The investigation was started with squeeze film dampers employing conventional lubricating oils. Magneto rheological fluids are then used to enhance the viscosity characteristics of the fluid under the influence of magnetic fields in order to improve the damper performances. It is observed that the dynamic characteristics of the damper with magneto rheological fluids are enhanced. Further to improve the damper performance, few modifications in the damper assembly are carried out in this research work. A good amount of reduction in the amplitude of vibrations is observed in these modified squeeze film dampers coupled with magneto rheological fluids. This research work discusses dual and triple clearance squeeze film dampers and bump foil squeeze film damper, also subjected to variation in temperature. Dynamic characteristics are found to be decreasing as the viscosity of the fluid decreases with rise in temperature of the fluid.

KEYWORDS:

Bump foil squeeze film damper; Magneto rheological fluids; Temperature effect; Damping coefficients; Shims

CITATION:

G. Saravanakumar, L. Ravikumar, R. Yogaraju, C. Shravankumar and V. Arunkumar. 2015. Determination of damping coefficient of a bump foil squeeze film damper, *Int. J. Vehicle Structures & Systems*, 8(4), 246-249. doi:10.4273/ijvss.8.4.11

1. Introduction

The most commonly persistent problems in rotor dynamics are excessive steady state synchronous vibration levels and sub-synchronous rotor instabilities. The first problem may be addressed by improved balancing or by introducing modifications into the rotor-bearing system to shift the system critical speeds out of the operating range or by introducing external damping to limit peak amplitudes at traversed critical speeds. Sub-synchronous rotor instabilities may be circumvented by eliminating the instability mechanism or by rising the natural frequency of the rotor-bearing system as high as possible or by introducing damping to raise the onset speed of instability. Squeeze film dampers are therefore employed to mitigate these problems in rotating systems [1]. A squeeze film damper is a bearing assembly for the support of high speed rotors. It comprises of a bearing, generally a ball or roller, or a journal bearing and a damping oil film.

Fig. 1 is a schematic diagram of a typical squeeze film damper [1]. The rotor is supported by a ball bearing, while an anti-rotation pin prevents the outer race of the bearing from rotating. The basic design of a squeeze-film damper consists of an oil-filled annular cavity surrounding the outer race of a rolling element bearing. Wedge flow, also known as combined velocity and pressure induced flow occurs. The pressure distribution due to this wedge can sustain a radial load [2, 3].

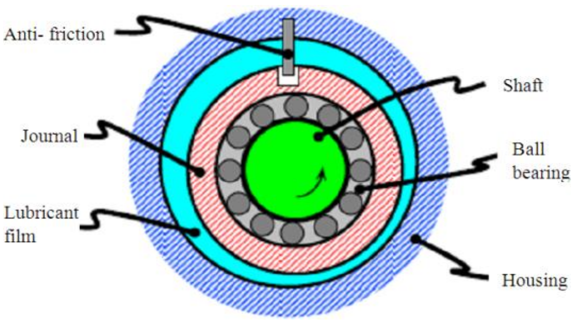

Fig. 1: Squeeze film damper with orbital motion

2. Mathematical analysis

The Reynolds equation for a journal bearing with polar co-ordinate system can be written as

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left[h^3\frac{\partial p}{\partial\theta}\right]+\left[h^3\frac{\partial^2 p}{\partial x^2}\right]=6\mu\omega\frac{\partial h}{\partial\theta}+\frac{\partial h}{\partial t} \quad (1)$$

The stiffness and damping co-efficient for the squeeze film damper with orbital motion using long bearing approximation are derived as follows,

$$K_{eq}=\frac{24\mu\omega r^3 Ln}{\left(2+n^2\right)\left(1-n^2\right)}\left[\frac{1}{c^3}\right] \quad (2)$$

$$B_{eq} = \frac{12\mu r^3 L\pi}{(2+n^2)(1-n^2)}\left[\frac{1}{c^3}\right] \quad (3)$$

Dubois and Ocvirk [4] considered the short bearing approximation (journal axis remains parallel to the bearing axis) for the steady state operating conditions. In such a case, the change in pressure in circumferential direction is small in relation to the pressure change in axial direction. For an open-end damper with a fully cavitated film, the stiffness and damping coefficients are conveniently determined through the short bearing approximation as follows,

$$K_{eq} = \frac{2\mu\omega r n L^3}{(1-n^2)^2}\left[\frac{1}{c^3}\right] \quad (4)$$

$$B_{eq} = \frac{\mu r L^3 \pi}{2(1-n^2)^{\frac{3}{2}}}\left[\frac{1}{c^3}\right] \quad (5)$$

It is apparent from these expressions that, for small eccentricities (e), stiffness increases approximately linearly with e and damping are approximately constant. For eccentricity ratios greater than about one-half, however, the $(1\text{-}e^2)$ term in the denominator of both expressions means that stiffness and damping now increase more rapidly as eccentricity increases. Theoretically, they increase without limit as C approaches 1[5]. It is this nonlinear property which is responsible for the undesirable characteristics of squeeze film damped rotors at high unbalance loads

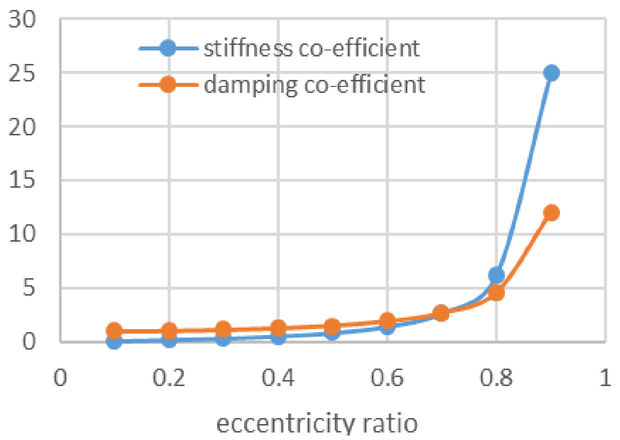

Fig. 2: Variation of stiffness and damping co-efficient with eccentricity ratio

2.1. *Squeeze film damper with a shim for dual clearance*

The quasi-linear range of a conventional squeeze film can be enlarged by increasing the damper clearance as shown in Fig. 3. This has the disadvantage of requirement for the damper to be longer. Furthermore, the added clearance results in less precise radial location of the rotor, possibly allowing rubbing of seal surfaces at engine start-up and shutdown. Centering springs (e.g., squirrel cage) are sometimes used in conjunction with squeeze film dampers. The stiffness of these springs is chosen for rotor dynamic purposes, however, and is frequently too low for purposes of shaft center line location. These disadvantages of the conventional squeeze film can be somewhat mitigated by the multiple-shim damper, but the total clearance required may still be larger than desirable. The dual clearance damper was designed in order to:

- Maintain close control of rotor radial location during normal operation when the rotor is well balanced.
- Maintain control of vibration amplitude and bearing load during operation above the critical speed with high unbalance.
- Allow safe deceleration through the critical speed with high unbalance.

It consists of two squeeze film dampers operating in series. During normal operation, the sleeve separating the two damper films is fixed in place by two or more shear pins. Only the inner film is active; behaviour is then identical to that of the single-film damper. The clearance is only as high as is required for the rotor unbalance likely to occur in normal operation; thus the rotor radial location can be closely controlled. In the event of rotor blade loss or some other occurrence which increases the unbalance, the damper load rises until the strength of the shear pins is exceeded. The pins shear, allowing the sleeve to move, and activating the outer damper film. The two films then operate in series; that is, the bearing load is transmitted first through the inner film; then through the sleeve and outer film to the machine structure. The outer film will generally have a larger clearance than the inner film in order to accommodate the larger amplitude of motion necessarily accompanying the higher unbalance. The inner and outer dampers will continue to operate together until the unbalance is corrected and new shear pins installed.

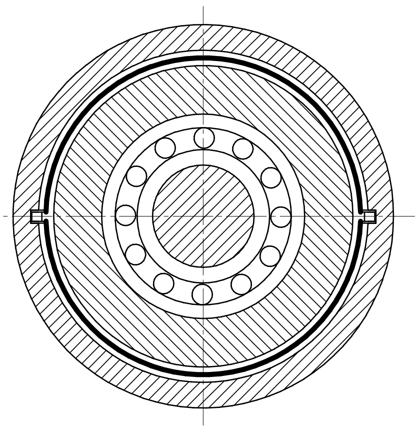

Fig. 3: Squeeze film damper with a shim

2.2. *Determination of dynamic coefficients of dual clearance squeeze film damper*

The shim divides the fluid film into two layers. The two fluid films on either side of the shim are in series in the radial direction. Let c_1 and c_2 are the radial clearance of inner and outer fluid films respectively. For two fluid films in series, the effective stiffness and damping co-efficient are derived as follows,

$$K_{eq} = \frac{K_1 K_2}{K_1 + K_2} \quad (6)$$

$$B_{eq} = \frac{B_1 B_2}{B_1 + B_2} \quad (7)$$

By substituting K_1 and K_2 in Eqns. (6) and (7),

$$K_{eq} = \frac{24\mu\omega r^3 Ln}{(2+n^2)(1-n^2)}\left[\frac{\frac{1}{c_1^3}\frac{1}{c_2^3}}{\frac{1}{c_1^3}+\frac{1}{c_2^3}}\right] = \frac{24\mu\omega r^3 Ln}{(2+n^2)(1-n^2)}\left[\frac{1}{c_1^3+c_2^3}\right] \quad (8)$$

$$B_{eq} = \frac{12\mu r^3 L\pi}{(2+n^2)(1-n^2)}\left[\frac{\frac{1}{c_1^3}\frac{1}{c_2^3}}{\frac{1}{c_1^3}+\frac{1}{c_2^3}}\right] = \frac{12\mu r^3 L\pi}{(2+n^2)(1-n^2)}\left[\frac{1}{c_1^3+c_2^3}\right] \quad (9)$$

Fig. 4 shows the variation of increment in stiffness and damping co-efficient with respect to the ratio of clearances. It is seen that the peak value occurs when both the clearances are equal. But the outer film generally has a larger clearance than the inner film in order to accommodate the larger amplitude of motion necessarily accompanying the higher unbalance.

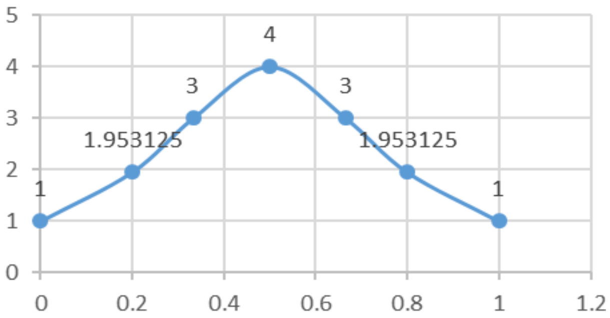

Fig. 4: Plot showing the variation of K_{eq}/K_D vs. c_1/c

2.3. Squeeze film damper with two shims for triple clearance

This is also a special type of damper similar to the squeeze film damper with a shim except that it has two shims. Three fluid film layers are sandwiched between two shims as seen in Fig. 5. The three fluid films are in series. Let c_1, c_2 and c_3 be the clearances of the three fluid films. These three fluid films are in series in the radial direction. The effective stiffness and damping co-efficient for this series combination of fluid films are given by:

$$K_{eq} = \frac{24\mu\omega r^3 Ln}{(2+n^2)(1-n^2)}\left[\frac{\frac{1}{c_1^3}\cdot\frac{1}{c_2^3}\cdot\frac{1}{c_3^3}}{\frac{1}{c_1^3\cdot c_2^3}+\frac{1}{c_2^3\cdot c_3^3}+\frac{1}{c_3^3\cdot c_1^3}}\right] \quad (10)$$

$$B_{eq} = \frac{12\mu r^3 L\pi}{(2+n^2)(1-n^2)}\left[\frac{\frac{1}{c_1^3}\cdot\frac{1}{c_2^3}\cdot\frac{1}{c_3^3}}{\frac{1}{c_1^3\cdot c_2^3}+\frac{1}{c_2^3\cdot c_3^3}+\frac{1}{c_3^3\cdot c_1^3}}\right] \quad (11)$$

Let $c_1 = c_2 = c_3 = c/3$, under this condition,

$$K_{eq} = \frac{24\mu\omega r^3 Ln}{(2+n^2)(1-n^2)}\cdot\frac{9}{c^3} \quad (12)$$

$$B_{eq} = \frac{12\mu r^3 L\pi}{(2+n^2)(1-n^2)}\cdot\frac{9}{c^3} \quad (13)$$

The stiffness and damping co-efficient have increased by 9 times as compared to conventional squeeze film dampers. The maximum stiffness and damping co-efficient of a double shim squeeze film damper is 2.25 times the maximum stiffness and damping co-efficient of a single shim squeeze film damper.

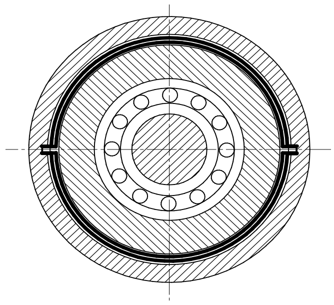

Fig. 5: Squeeze film damper with two shims

2.4. Bump foil squeeze film damper

Due to the bump foil's symmetrical nature, analysis on a portion of the bump foil is sufficient to understand its overall characteristics. Let c_1 and c_2 be the radial clearance on either side of the bump foil. Fig. 6 shows the variation of clearances along the angle θ. By use of trigonometric trial functions and the boundary conditions, c_1 and c_2 can be determined. From Fig. 7, the maximum stiffness and damping co-efficient of a bump foil squeeze film damper is determined to be 4 times of a conventional squeeze film damper. The maximum value of stiffness and damping co-efficient obtained is same as the maximum value obtained for a squeeze film damper with single shim. Therefore to improve the damper performance, free floating shims are added in the constrained space as shown in Fig. 8. These free floating shims divide the film thickness to a lower level to improve the dynamic coefficients.

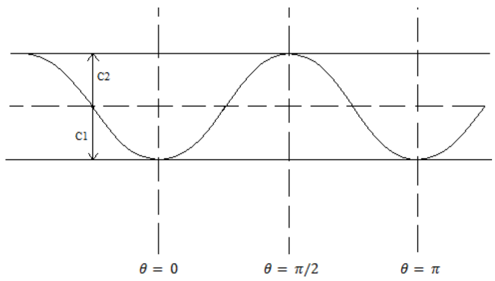

Fig. 6: Unwrapped bump foil

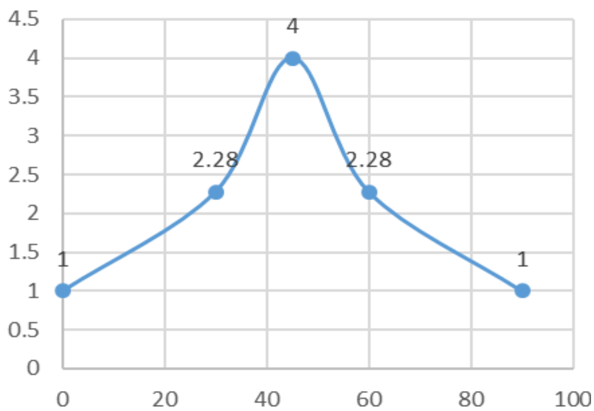

Fig. 7: Plot showing the variation of K_{eq}/K_D vs. c_1/c for bump foil squeeze film damper

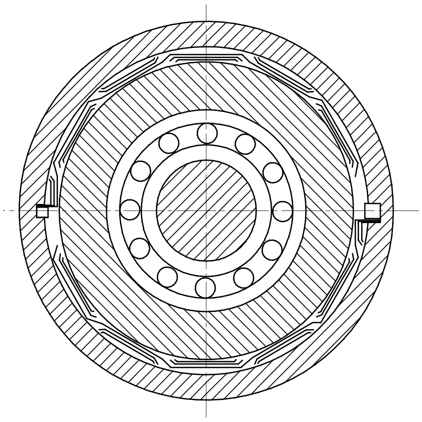

Fig. 8: Bump foil squeeze film damper with floating shims

3. Results and Discussions

The maximum stiffness and damping produced by the bump foil is equal to the maximum stiffness and damping of a dual clearance squeeze film damper. But unlike dual clearance damper due to physical contact between bump foil and journal there is damping due to friction. This type of damping is called coulomb damping. So the overall damping will be, Total damping = viscous damping + coulomb damping. Also, by doing the following design changes its characteristics can be further improved.

- By addition of shims in the bump foil.
- By giving perforations in the bump foil.

This modified bump foil will have different damping characteristics. The total damping offered by this modified bump foil will be equal to the sum of viscous damping with shim in the bump foil, damping due to friction and damping due to perforations. Total damping = viscous damping + coulomb damping + damping due to perforation. Effect of temperature on the viscosity and consequently on the damping characteristics of a magneto rheological fluid squeeze film damper is theoretically evaluated as shown in Figs. 9 and 10.

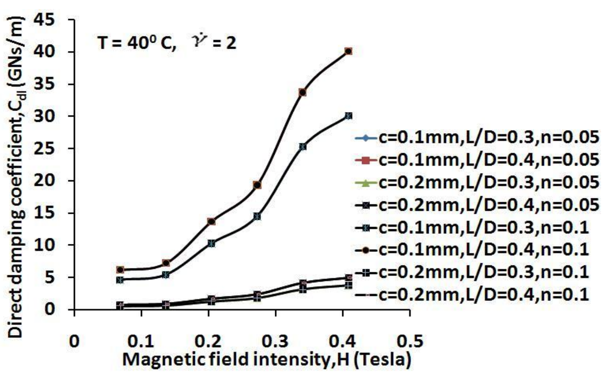

Fig. 9: Variation of damping coefficient for T = 40°C and strain rate = 2%

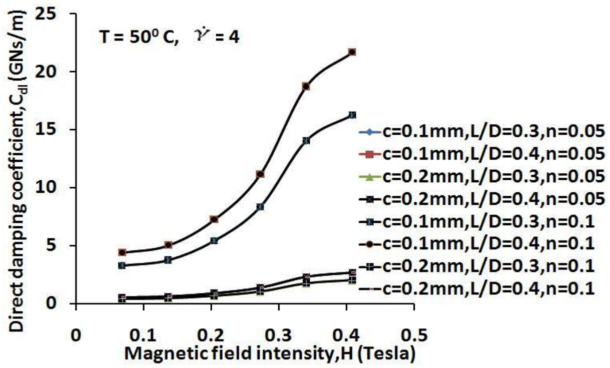

Fig. 10: Variation of damping coefficient for T = 50°C and strain rate = 4%

4. Conclusion

Stiffness and damping coefficients reduces with the rise in temperature as viscosity of the fluid falls. But use of magneto rheological fluid can compensate the loss of viscosity due to temperature by subjecting it under a magnetic field. Fluid film thickness can also be reduced by introducing shims in the space to enhance the damper properties.

ACKNOWLEDGEMENTS:

The authors would like to acknowledge the financial support received from Propulsion Panel, AR&DB, DRDO, New Delhi, Government of India for the R&D sponsored project vide File No: DARO/08/1041575/M/I /23-12-2010, and TEQIP-II of BMSCE for helping us to file a patent on this work.

REFERENCES:

[1] J.D. Carlson and K.D. Weiss. 1994. A growing attraction to magnetic fluids, *Machine Design*, 66, 61-64.

[2] C. Carmignani and P. Forte. 2001. Active squeeze film dampers in rotor dynamics, *Int. J. Applied Mechanics and Engg.*, 6(4), 1-8.

[3] P. Forte, M. Paterno and E. Rustishi. 2004. A magneto rheological fluid damper for rotor applications, *Int. J. Rotating machinery*, 10(3), 175-182. https:/doi.org/10.1155/S1023621X04000181.

[4] G.B. Dubois and E.W. Ocvirk. *Analytical Derivation and Experimental Evaluation of Short Bearing Approximation for Full Journal Bearings*, NASA Report 1157.

[5] J.S. Rao. 2006. *Rotordynamics*, New Age Int. (P) Limited, New Delhi.

[6] C. Guang and Y.S. Huat. 1996. *Research on MR Fluids and its Applications*, Applied Research Project Report, Nanyang Technological University, Singapore.