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AN ANALYTICAL STUDY OF DAMPING CHARACTERISTICS OF HYDROSTATIC BEARINGS

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ABSTRACT

A theoretical analysis is presented to predict the dynamic performance characteristics of a circular thrust bearing with a central recess and integrated with a restrictor of finite fluidic resistance. The governing equations of flow are solved in closed forms to obtain pressure distributions in the bearing clearances. Assuming a sinusoidal variation in the oil film thickness, the damping load, flow rate and damping coefficient are evaluated for squeezed disks. Results are presented in dimensionless terms describing the cyclic variation of the different squeeze parameters and the different geometrical aspects. They show the non-linearities of these parameters especially at high amplitude ratios.

INTRODUCTION

One of the advantages of hydrostatic bearings is their high damping involved which results from the squeeze actions [1] . On the other hand, stability problems may arise when such systems are subjected to cyclic vibrations due to either dynamic loads or base excitations. The main objective of this paper is to deduce the basic relations describing the non-linear damping behavior of an actual circular thrust bearing with a central recess. The analysis considers the different kinds of restrictors, usually of finite fluidic resistance(Z_c) and through which the lubricant is fed to jack up the loads. Koenigsberger[2], Brown [3] and Hunt [4] , presented different closed form expressions for normal damping coefficients for circular pads. They considered the effect of the compensating elements in terms of an extra route to the squeezed flow. Their expressions are found to disagree with each other as was clearly stated in reference [2]. In this study proper boundary condition is used to solve Reynolds equation for thin film incompressible flows. The two limiting cases where Z_c=o and $Z_c = \infty$ (circular disks with open and closed recesses respectively) are analyzed in details. In addition, a generalized expression for the total operating loads of both hydrostatic and dynamic nature is deduced using a control volume approach for which the continuity equation is applied for an actual film lubricated thrust bearing.

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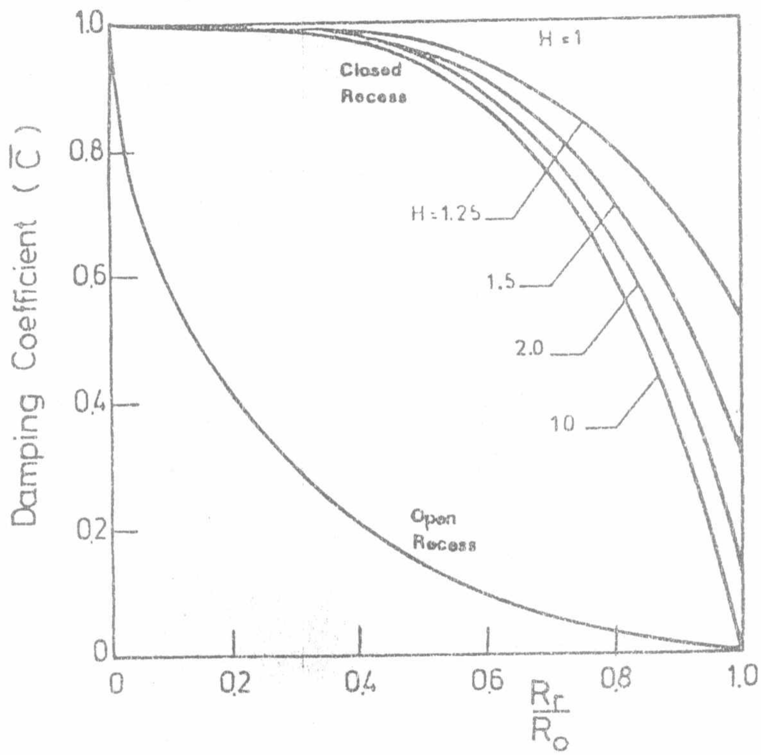

Fig. 2 Variation of the Dimension less Damping Coefficient with Recess Dimensions.

ANALYSIS

The bearing configurations shown in Fig.1 illustrate the design sequence leading to the actual thrust bearing systems shown in Fig. 1-d. These include a pressure supply unit feeding the lubricant via a restrictor of finite fluidic resistance (Z_c) for control purposes. For incompressible and isothermal flow, Reynolds equation for circular axisymmetric bearings can be written such as:

$$\frac{d^2P}{dr^2} + \frac{1}{r}\frac{dP}{dr} = - \frac{12\mu \dot{h}}{h^3} \qquad (1)$$

Where the RHS represents the squeeze term. For the three simple cases shown in Fig.1 (a-b-c), the problem reduces to a one dimensional boundary value problem along any radial direction. The solution may be obtained by close form integrations using proper boundary conditions as follows:

For a flat disk:

$$p = 0 \text{ at } r = R_o$$
$$dp/dr = 0 \text{ at } r = 0 \qquad (2a)$$

For a disk with open recess: ($Z_c = 0$)

$$p = 0 \text{ at } r = R_r$$
$$p = 0 \text{ at } r = R_o \qquad (2b)$$

Where $$Q_c = \frac{\lambda}{(\mu)^{\gamma}} \; h_r^{\alpha} \; (P_s - P_r)^{\beta} \quad (7)$$

which is the flow to the CV through the restrictor [8], and α, β, γ are exponents, with their values depending on the type of restrictor such as:

For capillary tube:	$\alpha = 0$,	$\beta = 1$,	$\gamma = 1$
For sharp edged orifice:	$\alpha = 0$,	$\beta = 0.5$,	$\gamma = 0$
For proximity orifice:	$\alpha = 1$,	$\beta = 0.5$,	$\gamma = 0$
For constant flow valve:	$\alpha = \beta$	$= \gamma$	$= 0$

The inflow squeezed from the annular sill is: (Table 1)

$$Q_{ss} = A\dot{h} \; (\xi^2 - 2\xi^2 \ln\xi - 1)/2\ln \xi \quad (8)$$

The flow out the recess via the sill, [1] .:

$$Q_H = (\pi/\mu \; 6\ln \xi) \; P_r \; h_s^3 \quad (9)$$

And the squeezed volume of the CV is

$$Q_{\dot{h}} = A \; \xi^2 \; \dot{h} \quad (10)$$

After substitutions and rearanging terms Eq.(7) reads

$$\frac{h_s^3}{\mu} \left[\frac{\lambda H^{\alpha}}{\mu \gamma - 1} \; h_s^{\alpha - 3} \; (P_s - P_r)^{\beta} \; - \; \frac{\pi}{6 \ln\xi} \right] + A\dot{h} \left[\frac{\xi^2 - 2\ln\xi - 1}{2 \ln\xi} - \xi^2 \right] = 0$$

or

$$(P_s - P_r)^{\beta} - \frac{\pi \; \mu^{\gamma - 1}}{6\lambda \; H^{\alpha} \; h_s^{\alpha - 3} \; \ln\xi} \; P_r = \frac{A\dot{h} \; \mu}{\lambda H^{\alpha} \; hs^{\alpha}} \left(\xi^2 - \left(\frac{\xi^2 - 2\xi^2 \ln \xi - 1}{2 \ln\xi} \right)\right) \quad (11)$$

For specific pad dimensions (A, ξ, H), restrictor type ($\lambda, \alpha, \beta, \gamma$), oil viscosity ($\mu$) and supply pressure ($P_s$) the recess pressure at any instant within a cycle can be obtained by solving Eq.(11) numerically.

Hence, the total load may be obtained by superposition as shown in Fig.1-d and may be expressed as:

$$W = P_r A \; K_1 + W_f \; K_2 \quad (12)$$

Where $K_1 = (\xi^2 - 1) / 2 \ln \xi$

$K_2 = 1 - \xi^4 + (\xi^2 - 1)^2 / \ln \xi = \overline{W}$ (disk with open recess)

or

$$\overline{W} = \frac{W}{W_f} = \frac{P_r A}{W_f} \; K_1 + K_2 \quad (13)$$

Table 2 Cyclic Functions of Squeezed Circular Bearings.

Description	Expression	Cy lic Functions
Dimensionless Pressure	$\tilde{P} = \frac{P}{\mu\omega} \left(\frac{h_o}{D_o}\right)^2$	$\frac{3}{4}\left[\frac{H_v \cos \omega t}{(1+H_v \sin \omega t)^3}\right] \cdot \bar{P}$
Dimensionless Pressure Gradient	$\tilde{P}' = \frac{P' h_o^2}{\mu\omega D_o}$	$\frac{3}{2}\left[\frac{H_v \cos \omega t}{(1+H_v \sin \omega t)^3}\right] \cdot \bar{P}'$
Dimensionless Load	$\tilde{W} = \frac{W}{\mu A\omega} \left(\frac{h_o}{D_o}\right)^2$	$\frac{3}{8}\left[\frac{H_v \cos \omega t}{(1+H_v \sin \omega t)^3}\right] \cdot \bar{W}$
Dimensionless Damping Coefficient	$\tilde{C} = \frac{C}{\mu A} \left(\frac{h_o}{D_o}\right)^3$	$\frac{3}{8}\left[\frac{1}{(1+H_v \sin \omega t)^3}\right] \cdot \bar{C}$
Dimensionless Flow Rate	$\tilde{Q} = \frac{Q}{A \omega h_o}$	$[H_v \cos \omega t] \cdot \bar{Q}$
Dimensionless Damping Flow Rate	$\tilde{Q}_D = \frac{QA\mu}{W h_o^3}$	$\frac{2\pi}{3} (1+H_v \sin \omega t)^3 \cdot \frac{\bar{Q}}{\bar{W}}$

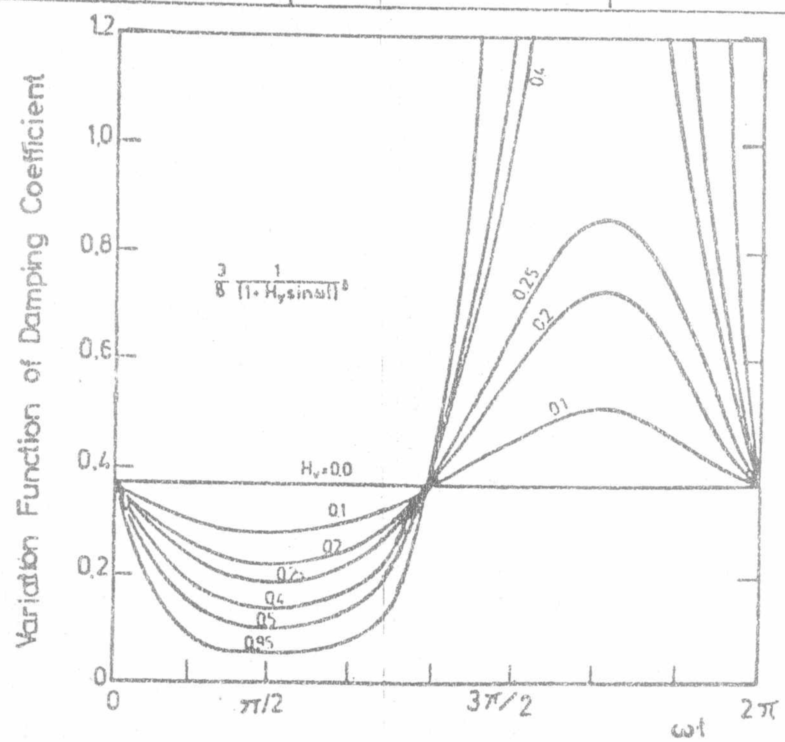

Fig. 3 Cyclic Variation of the Damping Coefficient for a Flat Disk.

RESULTS AND DISCUSSION

Referring to tables 1 and 2 both geometrical and cyclic variation functions are listed for an annular disk with open recess and a disk with sharp edged central closed recess. The dimensionless damping coefficients $\bar{C}$ are plotted in Fig.2 for both dampers against ξ. It is clearly shown that a flat disk provides higher damping with a decreasing value of $\bar{C}$ as ξ increases. The cyclic variation of C may be obtained from Eq.(5) by using the multiplier cyclic function shown in Fig.3. The C values are shown to increase sharply as H_v increase and when the bearing gap closes ($\omega t = 3\pi/2$). The attained expression for C for the opened recess case, table 1,comes in complete conformity with earlier predictions by Archibald [5]. Meanwhile, its value equals 0.5737 and 0.0171 for ξ values of 0.1 and 0.9 respectively when compared to Koenigsberger expression (appendix I) which gives 0.9079 and 0.0029 for the same two values. However, results have been asserted by analysing the squeezed flows at both inner and outer edges. Figs.5 and 6 show the variation of pressure gradients P and rate of flows Q at $\bar{R}=1$ and $\bar{R}=\xi$. The algebraic sum of both flows equals the squeezed volume $\pi \dot{H} R_o^2(1-\xi^2)$, a condition which satisfies continuity instantaneously. For the actual bearing with a specific resistor resistance, the load parameter $\bar{W}$ shows linear dependency on the variable recess pressure with the coefficients K_1 and K_2 being functions of ξ as illustrated in Fig.7.

CONCLUSIONS

1. The cyclic variation of squeezed circular disks has been presented in compact dimensionless forms as multiplier functions.
2. The variation of the recess pressure in an actual hydrostatic thrust bearing subjected to squeeze has been formulated. The analysis considered a generalized flow equation for different restrictors.

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