Numerical study of the influence of turbulent regime on the load capacity of a hydrostatic journal bearing

Nemchi N. and Bouzidane A.
Department of Mechanical Engineering,
Univ. Tiaret, Algeria
nemchi.amina@hotmail.fr
ahmed.bouzidane@gmail.com
Abed A.
lDepartment of Mechanical Engineering,
Univ. Chlef, Algeria
asma.raven@gmail.com

***Abstract*—The aim of the paper work is to investigate and predict the performance characteristics of four-pad hydrostatic journal bearings loaded between pads in both laminar and turbulent regimes. Linear modeling was performed using a numerical method to study the effects of Poiseuille Reynolds number, eccentricity, squeeze velocity and pressure ratio on the static and dynamic characteristics of hydrostatic journal bearings. The finite difference method has been used to solve Reynolds equation, based on Constantinescu's turbulent lubrication theory, governing the lubricant flow in film thickness of hydrostatic bearings. It assumed that the fluid flow is incompressible, isothermal, steady-state and inertialess fluid. The results presented in this paper are expected to be quite useful to bearing designers, in order to use it as a device for actively controlling rotors operating at high speeds.**

Keywords— hydrostatic journal bearings, laminar and turbulent regimes, squeeze velocity, eccentricity ratio and pressure ratio.

I- Introduction

The hydrostatic journal bearings have been selected as support elements in future cryogenic high-speed turbo machinery. They are often used in multi-application such as high speed turbomachinery; machine tools; machine control and the dispositive equipment test because of their capability of suppressing oil whirl that causes instability. Hydrostatic journal bearings have better dynamic characteristics due to their high stiffness; good damping at critical speeds their lubrication prevent contact of metal-metal even at zero speed.

Several researchers studied the performance of a hydrostatic journal bearing in the laminar regime [1-12]. The operation of high speed bearings and has a low viscosity meant that the flow regime is turbulent (the Reynolds number is above than 2000) as bearings for cryogenic applications operating at high speeds.

Several investigators [13-18] have proposed various theories to analyze the turbulent aspects applied in plain bearings. Of research efforts have been reported in the literature [19-27], focusing on the performance of multirecess hydrostatic journal bearings Hybrid operating in turbulent regime.

operating in a turbulent regime by considering various geometric shapes of recess. Their numerical simulation results indicate that the influence of turbulence is quite significant on the performance of a hybrid journal bearing system with different geometric shapes of recess However, the literature review revealed that the performance of a four-pad Hydrostatic Journal Bearing (HJB) loaded between pads in turbulent flow regime (Fig. 1) has not yet been investigated. This paper addresses this gap and presents an investigation of the effects of different flow regimes, squeeze velocity, eccentricity ratio and pressure ratio on the load-capacity, flow rate, damping ratio and dynamic characteristics of an orifice compensated four-pad HJB loaded between pads

II-Hydrostatic bearing description

Fig.1a shows a horizontal rigid rotor supported by a hydrostatic squeeze film damper composed of four-pad hydrostatic bearings. The indices 1, 2, 3 and 4 refer to the characteristics of the lower, left, upper and right hydrostatic bearing flat pad, respectively. Each pad is fed by an orifice restrictor through a recess, which is supplied with an external pressure P_s. All pad-geometries are identical and equally spaced around the journal. The calculation of the characteristics of the hydrostatic journal bearings can be obtained through the juxtaposition of four hydrostatic bearing flat pads. It is assumed that the fluid is incompressible and inertialess. The flow is laminar and the regime is steady state and isothermal.

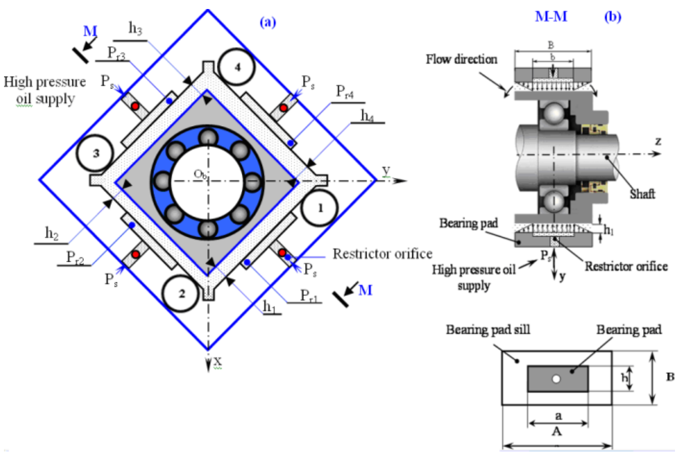

Fig1: Four-pad hydrostatic journal bearings loaded between pads

III-Reynolds Equation

Pressure distribution P_i (x_i, z_i,t), in the clearance space between runner and pad-bearing (Fig.2), can be governed by Reynolds equation. This equation can be solved numerically by applying the centered finite differences method, or analytically for specific cases such as infinitely long or short bearings. If we consider that there is no slip between the fluid and pad bearing, the boundary conditions associated with the speed will be as follows (Fig. 2):

- On flat pad: U_{1i} =0; V_{1i} = 0 and W_{1i} =0 (1)
- On runner: U_{2i} =0; V_{2i}= $\dot{h}_i$ and W_{2i} = 0 (2)

where $U_{1i};V_{1i}$ and w_{1i} are the speeds of the surface of the i^th hydrostatic bearing pad, and $U_{2i};V_{2i}$ and w_{2i} are the speeds of the surface of the runner; $\dot{h}_i$ is the squeeze velocity of the i^th hydrostatic bearing pad (i =1, 2 , 3 and 4);

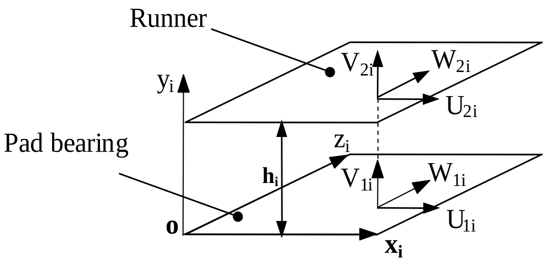

Fig.2: Boundary conditions of hydrostatic squeeze film dampers.

With these boundary conditions, and for an incompressible, isoviscous, and inertialess fluid, the modified Reynolds equation proposed by Constantinescu [15-19], for Poiseuille flow, may be written as:

$$\frac{\partial}{\partial x_i}\left(\frac{h_i^3}{\mu}G_x\frac{\partial P_i(x_i,z_i,t)}{\partial x_i}\right)+\frac{\partial}{\partial z_i}\left(\frac{h_i^3}{\mu}G_z\frac{\partial P_i(x_i,z_i,t)}{\partial z_i}\right)=\frac{\partial h_i}{\partial t} \quad (3)$$

Where: G_x and G_y are coefficients dependent on the Reynolds number. According to the preponderant flow, Constantinescu propounds the following expressions [17, 18, and 28]:

- For laminar flow:

$$K_p = 12; \quad \text{if } R_p < \Re_{pi}; \quad (4)$$

- For transition flow:

$$K_p = 12 + \frac{(K_{pm}-12)(R_p-\Re_{pi})}{\Re_{ps}-\Re_{pi}}; \quad \text{if } \Re_{pi} < R_p < \Re_{ps} \quad (5)$$

Where: $K_{pm} = a_p \Re_{ps}^{bp}$

- For turbulent flow:

$$K_p = K_{pm} = a_p R_p^{bp}; \quad \text{if } R_p > R_{ps} \quad (6)$$

Where:

$$\Re_{pi} = 1000; \Re_{ps} = 2\Re_{pi}$$

$$a_p = 0.197; b_p = 0.681 \quad \text{for } R_p \le 100000$$

$$R_p = \rho\frac{V_{pm}\,h}{\mu}$$

Where: a_p and b_p are coefficients dependent on the Poiseuille Reynolds number value; R_p is the Poiseuille Reynolds number based on the fluid mean velocity produced by the hydrostatic pressure gradients; V_{pm} is the fluid mean velocity and h is the film thickness.

Thus, due to hydrostatic pressure gradient, the modified turbulence coefficients G_x and G_z can be obtained by:

$$G_x = G_z = 1/K_p \quad (7)$$

III-1 Carrying Load Capacity

The integration of pressure over the bearing area yields the following load capacity result

$$W_{Pi} = \int_s P_i\,ds \quad (8)$$

For the calculation of the load considering the method by numerical integration or method called Trapeze

$$W = \frac{1}{4}\left[P_i(1,1)+P_i(1,N)+P_i(M,1)+P_i(M,N)\right]\Delta x\Delta z + \frac{1}{2}\left[\sum_{I=2}^{M-1}(P_i(I,1)+P_i(I,M))\right]\Delta x\Delta z + \frac{1}{2}\left[\sum_{J=2}^{N-1}(P_i(1,J)+P_i(M,J))\right]\Delta x\Delta z + \sum_{J+2}^{N-1}\sum_{I=2}^{M-1}P_i(I,J)\Delta x\Delta z \quad (9)$$

The fluid film forces on the journal may be written as:

$$\begin{cases} W_x = -\left(\left(F_{P1}+F_{P2}\right)-\left(F_{P3}+F_{P4}\right)\right)sin(\pi/4) \\ W_y = -\left(\left(F_{P1}+F_{P4}\right)-\left(F_{P2}+F_{P3}\right)\right)sin(\pi/4) \end{cases}$$

(10)

Flow rate requirement

The Recess Pressure is determined from the resolution of the flow continuity equation as follows:
For i=1 and 3

$$Q_{Oi} = Q_{OXi} + Q_{OYi} + Q_{Vi}$$

(11)
Where

$$Q_{OXi} = 2\int_0^B dZ \int_0^{h_i} U_{Xi}\,dy\,;\; Q_{OZi} = 2\int_0^A dX \int_0^{h_i} U_{Zi}\,dY\,;$$

(12)

$$U_{Xi} = \frac{1}{2\mu}\frac{\partial P_i}{\partial X}(y-h_i)y;\; U_{Zi} = \frac{1}{2\mu}\frac{\partial P_i}{\partial Z}(y-h_i)y$$

And for i=2 and 4

$$Q_{Oi} = Q_{OYi} + Q_{OZi} + Q_{Vi}$$

Where

$$Q_{OYi} = 2\int_0^B dZ \int_0^{h_i} U_{Yi}\,dX\,;\; Q_{OZi} = 2\int_0^A dY \int_0^{h_i} U_{Zi}\,dX\,;$$

(13)

$$U_{yi} = \frac{1}{2\mu}\frac{\partial P_i}{\partial Y}(y-h_i)y;\; U_{Zi} = \frac{1}{2\mu}\frac{\partial P_i}{\partial Z}(y-h_i)y$$

Where u_{xi}, u_{Yi} and u_{Zi} are the flow velocities in the x, y and z directions respectively.

$$Q_{Vi} = S_a \dot{h}_i$$

(14)
Where Q_{Vi} represents the squeeze flow of the i^{th} hydrostatic bearing pad (i = 1, 2, 3 and 4)
The flow through a capillary (Used when the flow is laminar) is governed by

$$Q_R = \frac{\pi d_C^4}{128\mu L_C}(P_S - P1)$$

(15)
The flow through an orifice is governed by:

$$Q_{Ri} = \frac{\pi C_d d_0^2}{4}\sqrt{\frac{2}{\rho}(P_S - P1)}$$

(16)
Where d_0 is the orifice diameter and C_d is the discharge coefficient.

Parameter C_d is a function of the Reynolds number, Re Typically, it varies in a nonlinear fashion from Cd=0.3 for Re=2 to C_d=0.7 at Re=100 and drops to about C_d=0.6 for higher Re values.
Flow rate requirement
The total volumetric flow rate that must be supplied to the hydrostatic squeeze film dampers is:

$$Q_T = \sum_{i=1}^{4} Q_{oi} = Q_{o1} + Q_{o2} + Q_{o3} + Q_{o4}$$

(17)

III-2 Dynamic characteristic

The single pad linearized stiffness and damping coefficients of the hydrostatic bearing pad may thus be obtained by

$$K_{P1} = -\left(\frac{\partial W_{P1}}{\partial h_1}\right)_0$$

(18)

$$C_{P1} = -\left(\frac{\partial W_{P1}}{\partial \dot{h}_1}\right)$$

(19)
A similar approach can be extended to the calculation of the other hydrostatic bearing pads:

$$K_{P2} = -\left(\frac{\partial W_{P2}}{\partial h_2}\right)_0, K_{P3} = -\left(\frac{\partial W_{P3}}{\partial h_3}\right)_0, K_{P4} = -\left(\frac{\partial W_{P4}}{\partial h_4}\right)_0$$

(20)

$$C_{P2} = -\left(\frac{\partial W_{P2}}{\partial \dot{h}_2}\right), C_{P3} = -\left(\frac{\partial W_{P3}}{\partial \dot{h}_3}\right), C_{P4} = -\left(\frac{\partial W_{P4}}{\partial \dot{h}_4}\right)$$

(21)

VI-Results and discussion
In this section, we study the influence of Poiseuille Reynolds number, eccentricity ratio, squeeze velocity and pressure ratio on the dimensionless static load capacity

- Dimensionless load capacity

$$\overline{W} = \overline{W}_X = W_x/(S_p\ P_s);\ \overline{W}_{yy} = 0$$

Fig. 3 shows the effect of Poiseuille Reynolds numbers and pressure ratio on the dimensionless load capacity for a squeeze velocity of 0.001 m/s and an eccentricity ratio of 0.2.

This figure indicates that the dimensionless load capacity increases with an increase in Poiseuille Reynolds numbers during the increase of pressure ratio.

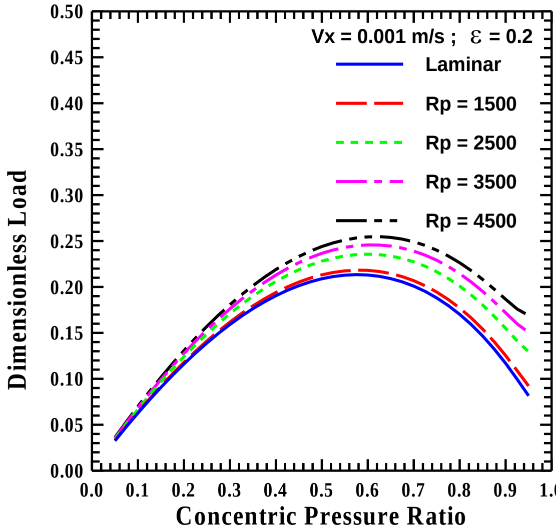

Fig3: Variation of load versus pressure ratio for different value of Reynolds number.

Fig.4 shows the effects of the eccentricity ratio and pressure ratio on the dimensionless static load capacity for a Poiseuille Reynolds number value of 2500 and a zero squeeze velocity. We note that the increase of eccentricity ratio increase the dimensionless static load capacity. This increase can be explained by the increase of hydrostatic forces due to the decrease of film thickness which increase the pressure in recess and land bearing.

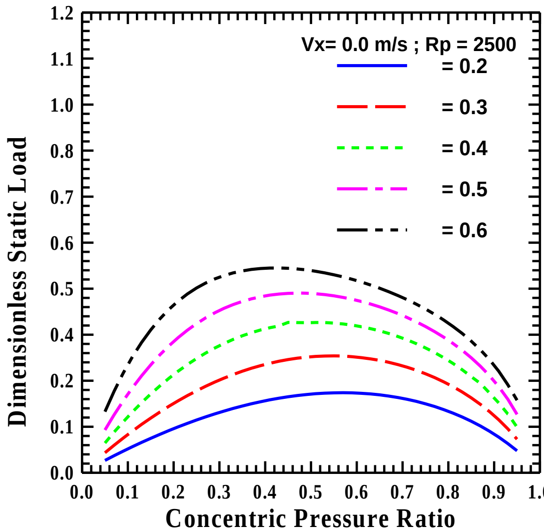

Fig4: Variation of load versus pressure ratio for different value of eccentricity ratio.

Fig.5 shows the effects of the squeeze velocity and pressure ratio on the dimensionless load capacity, for a Poiseuille Reynolds number value of 2500 and an eccentricity ratio of 0.2. We note that that the increase of squeeze velocity, from 0.001 to 0.01 m/s, increases significantly the dimensionless load capacity due to the increase of hydrostatic forces. It must be noticed that when the pressure ratio is higher than 0.9, the increase of the squeeze velocity results a very large dimensionless load capacity.

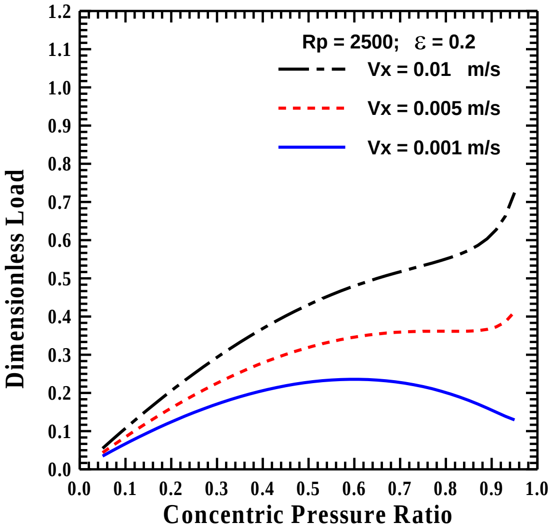

Fig5: Variation of load versus pressure ratio for different value of squeeze velocity.

V-Conclusion

The result obtained from a numerical model has been developed to study the effect of the Poiseuille Reynolds number; eccentricity and squeeze velocity on the dimensionless static load capacity of a hydrostatic journal bearing can be summarized as follows:

- An increase in Poiseuille Reynolds, eccentricity ratio and squeeze velocity lead to a significant increase of the load capacity due to the increase of pressure in recess and land bearing. However, there are no effects of Poiseuille Reynolds and squeeze velocity on the flow rate;
- It must be noticed that the load capacity have an optimum value with respect to pressure ratio;

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