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Studies on steady state response of floating pad journal bearing for high speed cryogenic turboexpanders

A Jain, M M Jadhav, S Karimulla and A Chakravarty
Bhabha Atomic Research Centre, Mumbai, India
ankitj@barc.gov.in

Tilting pad journal bearings are commonly used for high speed rotor applications such as cryogenic turboexpanders. Floating pad journal bearing is a special type of tilting pad journal bearing characterized by the absence of pivots altogether. The aim of this paper is to carry out the analysis of floating pad journal bearing (pivot-less tilting pad) to understand its working and characterize the journal bearings through parametric studies. A mathematical model is developed by using one-dimensional Reynolds Equation for compressible flow and numerical analysis is done using Finite Element Method. The studies involve steady state analysis for computing the stiffness coefficients and load capacity of the bearing. A comparison between standard tilting pad journal bearing and floating pad journal bearing on the basis of performance characteristics, is also carried out and presented.

Table 1. Nomenclature.

| Symbol | Description |
|--------|-------------|
| $R_s$  | Radius of shaft |
| $R_b$  | Radius of bearing |
| $R_p$  | Radius of housing side curvature of pad |
| $R_h$  | Radius of pad housing |
| $L$    | Axial length of the pad |
| $\alpha'$ | Total pad extent angle |
| $\alpha$  | Effective pad extent angle |
| $D_h$  | Diameter of connecting holes |
| $C_1$  | Radial clearance between shaft and pad |
| $C_2$  | Radial clearance between pad and pad housing |
| $e_1$  | Eccentricity of shaft |
| $\theta$  | Circumferential coordinate |
| $x$    | X-direction coordinate |
| $y$    | Y-direction coordinate |
| $z$    | Axial coordinate |
| $\omega$ | Rotational speed of shaft |
| $\beta$ | Angle of tilt of pad |
| $\theta_p$ | Angular extend of housing side surface of pad |
| $\eta$  | Viscosity |
| $Re$   | Reynolds no. |
| $\rho$  | Density of gas |
| $\xi$  | Attitude angle (angle between line of centre of shaft and pad leading edge) |
| $P$    | Pressure |
| $P_a$  | Ambient pressure |
| $h$    | Film thickness |
| $F_x$  | Unbalance force in X-direction |
| $F_y$  | Unbalance force in Y-direction |
| $M$    | Moment about imaginary pivot |

TPJB  Tilting Pad Journal Bearing
FPJB  Floating Pad Journal Bearing
1. INTRODUCTION
Most of the high speed devices like cryogenic turboexpanders employ gas lubricated bearings as most preferable option. Gas bearings, unlike rolling element or oil-lubricated bearings, offer advantages like low noise and less heat generation due to friction [1]. Gas bearings are important particularly in the cryogenic refrigeration/liquefaction system where it is necessary to keep the environment free from contamination due to conventional lubricants [2]. For supporting vertical rotors where the desired load capacity is not very large, aerodynamic bearing is the most preferable option owing to compactness and simplicity in installation and absence of gas pressurization system needed in aerostatic bearings. In the development of aerodynamic journal bearings, the plain full journal bearing is the most primitive. But this type of bearing is highly susceptible to self-excited half speed whirl which sets a limit to the maximum attainable rotor speed [1]. To avoid this instability, many innovative geometries have been invented such as axial groove, herringbone groove, tilting pad, foil, etc. Tilting pad journal bearings (TPJB), amenable to easy fabrication and compact installation, have found widespread use in the industry. Detailed studies pertaining to stability characteristics of these bearings can be found in the literature [3,4,5,6]. TPJB overcomes the instability of half speed whirl and also provides a wide range of design parameters. This bearing consists of a set of pads distributed symmetrically in circumferential direction, which are supported on pivots fixed to bearing housing. Each pad is allowed to tilt about its pivot to attain an equilibrium forming a converging film between shaft and pad. This converging film generates the pressures and is responsible for load carrying capacity of the bearing. Since the early days of TPJB, different designs have been developed and used [7,8,9]. Floating pad journal bearing (FPJB) is a specialized type of TPJB where pivots are absent [10]. The journal bearing has three floating pads and the fluid pressure between the shaft and pad generates the force to support the radial load. The pads in this type of bearing are free from any mechanical contact and hence free from misalignment issues. The aim of the work presented here is to analyse and understand characteristics of FPJB using numerical methods.

2. GEOMETRY AND WORKING
A typical FPJB consists of a housing and three pads placed symmetrically along the circumferential direction as shown in figure 1. Each pad has a shaft side that forms the bearing surface, a housing side, a network of three hole at two planes which connects shaft and housing side of the pad, and wedge at trailing edge (figure 2). In the shaft side, the connecting hole is located at the geometric centre of housing side curve of pad. This point is considered as imaginary/floating pivot (represented as IP in figure 3) in the presented analysis. The wedge at the trailing edge is provided to indirectly shift the pivot position towards the trailing edge because for maximum load, the optimum pivot location in standard TPJB is found to be around 40% of effective pad angle from trailing edge [5].

During shaft rotation, pressure is generated at the shaft side of the pad. The shaft side pressure is communicated to the housing side through the connecting holes [1]. A fraction of fluid stream passes through the connecting hole and is fed to the housing side of the pad. The pressure generated at the housing side of the pad is responsible for lifting of pad. The pad housing side cannot be considered as standard orifice-compensated aerostatic bearing because the pressure at entry of connecting hole and pressure at the exit of connecting hole are inter-dependent. There are three types of forces that acts on the pad: aerodynamic load on shaft side, aerostatic load on housing side, and viscous force on the pad surface. The sum total of forces and moments (as shown in figure 3(b)) on the pad are responsible for movement of the pad. A pad has three degree of freedom; it can translate in the radial direction, circumferential direction, and can rotate about its imaginary pivot. After achieving stable equilibrium position, the pressure profile generated on the shaft side is responsible for the load capacity of the bearing. The geometrical and operational parameters taken for the analysis are listed in table 2.
3. GOVERNING EQUATIONS
In analysing this problem, following equations are considered:
1. Generalised Reynolds Equation
2. Poiseuille's flow between two plates
3. Moment and Force balance

1) Generalized Reynolds Equation
The steady state pressure distribution between the shaft and pad, and between the pad and pad housing is assumed to be governed by Reynolds equation [11] for an isothermal film which can be expressed as:

\[ \left( \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( \frac{Ph^3 \partial P}{12 \eta \frac{\partial P}{\partial \theta}} \right) + \frac{\partial}{\partial z} \left( \frac{Ph^3 \partial P}{12 \eta \frac{\partial P}{\partial z}} \right) \right) = \frac{\omega}{2 \frac{\partial \theta}{\partial P}} (Ph) \]

The ideal gas equation for isothermal process which has been used in the analysis is:
\[ \frac{P}{\rho} = \text{constant} \]
For our analysis, long bearing assumption has been taken in which the variation of pressure in axial direction is negligible and hence this equation reduces to:

\[
\left( \frac{1}{R^2} \right) \frac{\partial}{\partial \theta} \left( \frac{P h^3}{12 \eta} \frac{\partial P}{\partial \theta} \right) = \frac{\omega}{2} \frac{\partial}{\partial \theta} (Ph)
\]  

(3)

The flow Reynolds number \( Re \), as applied to the bearing, is defined as

\[
Re = \frac{\rho V D}{\eta}
\]

(4)

where \( V \) is taken as the maximum velocity of gas (which is equal to shaft surface speed), and \( D \) is the characteristic length = twice the maximum possible film thickness.

For the problem under consideration, the Reynolds no. is computed to be 766.06 which is in the laminar flow range, thus justifying the use of the Reynolds equation.

2) Poiseuille's Flow between two plates

The flow through the feed holes involves the Poiseuille's equation [11] which is expressed as:

\[
Q_n = -\frac{\rho \pi D_h^4}{128 \eta} \frac{\partial P}{\partial n}
\]

(5)

where \( Q_n \) is mass flow rate in \( n \) direction and \( D_h \) is diameter of hole.

Analysis presented in this paper is based on one dimensional fluid flow in a two dimensional geometry. Hence equation of flow through connecting holes is converted for one dimensional flow as flow between two plates. Corresponding equation is [11]:

\[
Q_n = -\frac{\rho t^3 L}{12 \eta} \frac{\partial P}{\partial n}
\]

(6)

Where, \( t \) is thickness between plates and \( L \) is the axial length of the bearing, \( Q_n \) is the mass flow rate in \( n \) direction.

Equating the pressure loss and mass flow rate in equation 3 and equation 4, the relation between \( t \), \( D_h \) and \( L \) is found out for one dimensional model:

\[
t = \left( \frac{3 \pi D_h^4}{32 L} \right)^{1/5}
\]

(7)

According to the geometry of holes (i.e. \( D_h \) and \( L \)), corresponding thickness \( t \) can be obtained and used as an input for one dimensional analysis.

3) Moment and Force balance

From one equilibrium position, when the eccentricity ratio is changed, the film thickness distribution between shaft and pad gets altered, due to which the pressures on pad changes. It disturbs the equilibrium of the pad by generating unbalanced forces and moments. The new equilibrium position of pad is defined to be that corresponding to the minimum potential energy. This position is found using Steepest Descent Method [12]. In this method, minimum of a function can be found by taking steps proportional to the negative of gradient of that function until gradient becomes zero. In the case considered in this work, potential energy \( \Psi \) has to be minimized. Pad has been considered as a two dimensional body, hence, the forces and moments can be expressed as:

\[
F_x = - \left( \frac{\partial \Psi}{\partial x} \right); F_y = - \left( \frac{\partial \Psi}{\partial y} \right); M = - \left( \frac{\partial \Psi}{\partial \beta} \right)
\]

(8)

For equilibrium of the pad,

\[
\Sigma F_x = 0; \Sigma F_y = 0; \text{ and } \Sigma M = 0
\]

(9)

When the potential energy is minimum, pad is considered to be in a stable equilibrium. To find the stable equilibrium of pad, the movement of pad will be as follows:

\[
\Delta x = a_1 F_x; \Delta y = a_2 F_y; \Delta \beta = a_3 M
\]

(10)
where, coefficients $a_1$, $a_2$ and $a_3$ are adjusted to have convergence of solution towards stable equilibrium; and $\Delta x$, $\Delta y$ and $\Delta \beta$ are changes in orientation of pad in $x$, $y$ and $\beta$ respectively. When $F_x$, $F_y$ and $M$ are zero, the potential energy is minimum and the pad is said to be in a stable equilibrium.

4. SOLUTION METHOD

Finite element formulation of Reynolds equation is used for shaft and housing side of the pad [13]. Different meshes are created for shaft and housing side of pad. These two meshes are connected at the point where the feed holes are present. Poiseuille equation for flow between two plates is used to connect the two sides of holes. The boundary conditions used are:

i. Pressures at the leading and trailing edges of shaft side mesh are equal to ambient pressure.

ii. Pressures at both the circumferential ends of housing side mesh are equal to ambient pressure.

Pressure profile is calculated first for an initial orientation of pad. Then according to the unbalanced forces and moments, orientation of pad is changed repeatedly until a stable equilibrium is found. This process is repeated for all pads for different eccentricities and attitude angles. The pressure profile on shaft side of the pad is used to calculate the load on the shaft due to that pad. Load due to all pads are vectorially added to find the load capacity of full bearing. The load capacity is defined as the net force acting on the shaft at a particular eccentricity. The stiffness of the bearing is defined as the ratio of change in load capacity to the change in eccentricity of the shaft. Load capacity at different eccentricities can be obtained to compute the stiffness of the bearing. Flowchart for the computation and analysis is presented in figure 4 based on which a computer code is generated using C++.

![Figure 4. Algorithm used for analysis of FPJB.](image-url)
5. RESULTS AND DISCUSSIONS
A typical FPJB under investigation has three pads as shown in figure 1. Convergent-diverging film profiles often give rise to instabilities [1]; hence the results presented here are for fully convergent film profiles.

A typical FPJB with three pads has been analyzed and presented. The parameters taken for the analysis are mentioned in table 2.

A grid independence test has been carried out and it was found that after 200 elements, an increase in the number does not have any significant effect on the result. Hence for the analysis presented in the article, 200 elements mesh has been used.

At eccentricity ratio ($\epsilon_1/C_1$)=0.4 and attitude angle ($\xi$) =120° (i.e. towards centre of pad1), the film thickness profile on the shaft side and housing side of the pad is shown in figure 5. Here, it can be seen that at equilibrium, film thickness on the shaft side is converging in nature. The film thickness is less in case of pad1 to generate more pressure in comparison to other pads so that the net force acting on the shaft pushes the shaft towards centre. On the housing side of pad, from the film profile (figure 6), it can be said that the net translation of pad is towards the centre of shaft to attain the equilibrium. The symmetric nature of film profile indicates absence of any unbalanced force in the circumferential direction. At same eccentricity and attitude angle, the dimensionless pressure profiles generated on shaft and housing side of the pad are shown in figures 7 and 8 respectively. The trend of pressure profile on the shaft side is similar to that of standard TPJB except the dip in pressure as shown in figure 7. On the shaft side, at the location of the connecting hole, the pressure decreases and increases after attaining some minimum value. The reason behind decrease in pressure in that region is the flow of fluid through the connecting hole.

![Figure 5. Shaft side film thickness profile.](image1)

![Figure 6. Housing side film thickness profile.](image2)

Variation of load capacity of each pad with eccentricity ratio is shown in figure 9. The shaft is provided with an eccentricity in the direction of pad1 ($\xi$=120°). So, the pressure generated between shaft and pad1 increases due to decrease in film thickness thus bringing about an increase in load capacity due to pad1. On the other hand, the increase in film thickness between pad2 and shaft, and between pad3 and shaft, results in a decrease in pressure and consequently the load capacity.

When forces generated due to all pads are combined, the load capacity of the full bearing can be computed. Figure 10 represents the variation of load capacity of full bearing for different eccentricities and attitude angles. When shaft is at the centre, the net force due to all pads is zero. As the eccentricity increases, the load carrying capacity also increases. When the eccentricity is small, the load increases linearly, but at higher eccentricities, the increase in load with eccentricity is more. Hence, the force assumes non-linear characteristics at higher eccentricities. It can be seen that when shaft moves in the direction of the imaginary pivot (i.e. towards centre of pad, $\xi$=120°), the force generated is higher than that when the shaft moves at position between the two pads ($\xi$=60°).
The variation of stiffness coefficient with eccentricity is shown in figure 11. Bearing is more stiff when shaft moves towards centre of pad ($\xi=120^\circ$). Stiffness is comparatively less when shaft moves in between two pads ($\xi=60^\circ$).

Load capacity and stiffness of TPJB and FPJB are compared and presented in figures 12 and 13 respectively. The parameters taken for comparison of both the bearings are same as given in table 2. It can be observed that FPJB of same bearing area and same shaft geometry is stiffer than its TPJB counterpart and can generate larger force to withstand higher load disturbance in the shaft.
6. CONCLUSIONS

- The floating pad is lifted by aerostatic action on housing side. The aerodynamic side of the bearing also acts like a source pressure for the pad aerostatic action.
- Effective pressure on the shaft side reduces due to flow of gas to the housing side of the pad.
- A typical FPJB for cryogenic turboexpander applications, with shaft radius of 8 mm and radial clearance of 40 μm can sustain around 12 N of load and offers a stiffness of around 1.4x10^6 N/m at eccentricity ratio of 0.3.
- Load capacity and stiffness coefficients of FPJB is not symmetric, but a function of direction of eccentricity. However, the variation in load and stiffness coefficient is not very significant.
- For the pad bearing configuration studied, the load capacity and stiffness of FPJB is around three times higher than a TPJB of similar dimensions.

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