Parametric studies on floating pad journal bearing for high speed cryogenic turboexpanders

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Abstract. Most modern medium and large capacity helium liquefaction/refrigeration plants employ high speed cryogenic turboexpanders in their refrigeration/liquefaction cycles as active cooling devices. The operating speed of these turboexpanders is in the range of 3000-5000 Hz and hence specialized types of bearings are required. Floating pad journal bearing, which is a special type of tilting pad journal bearing, where mechanical pivots are absent and pads are fully suspended in gas, can be a good solution for stable operation of these high speed compact rotors. The pads are separated from shaft as well as from housing by fluid film between them, and both these sides of pad are interconnected by a network of feed holes. The work presented in this article aims to characterize floating pad journal bearings through parametric studies. The steady state performance characteristics of the bearing are represented by load capacity, stiffness coefficients and heat generation rate of the bearing. The geometrical parameters such as bearing clearances, preload of pads, etc. are varied and performance characteristics of the floating pad journal bearing are studied and presented. The dependence of stiffness coefficients on rotational speed of shaft is also analyzed.

Table 1. Nomenclature.

| Symbol | Definition |
|--------|------------|
| R_s   | Radius of shaft |
| R_b   | Radius of bearing |
| R_p   | Radius of housing side curvature of pad |
| R_h   | Radius of housing |
| α'    | Total pad extent angle |
| α     | Effective pad extent angle |
| C_1   | Radial clearance between shaft and pad |
| C_2   | Radial clearance between pad and pad housing |
| ε_1   | Eccentricity of shaft |
| D_h   | Diameter of connecting holes |
| x     | X-direction coordinate |
| y     | Y-direction coordinate |
| l_e   | Length of an element in the mesh |
| θ     | Circumferential coordinate |
| z     | Axial coordinate |
| θ_p   | Angular extend of housing side surface of pad |
| η     | Viscosity |
| ω     | Rotational speed of shaft |
| ξ     | Attitude angle (angle between line of centre of shaft and pad leading edge) |
| P     | Pressure |
| P_a   | Operating ambient pressure |
| K_xy  | Stiffness coefficient in 'x' direction when shaft moves in 'y' direction |
| h     | Film thickness |
| W     | Load capacity |
| β     | Angle of tilt of pad |
| L     | Length of the bearing |
| F_j   | Load capacity of the bearing at eccentricity 'j' |
1. Introduction

Turboexpander is one of the most important components in a cryogenic helium liquefaction/refrigeration system as it is the sole device responsible for extracting energy from the high-pressure gas stream. According to the head-flow characteristics of cryogenic turboexpanders, the operating speed of these machines falls in the range of 3000 Hz to 5000 Hz. Due to high heat generation and wear and tear in rolling element and oil-lubricated bearings, coupled with chances of contamination of process [1], bearings lubricated by the process gas itself has been accepted as the only suitable option for such high speed applications in the field of cryogenics [2].

Over the years, from the need to provide rotor stability at high speeds, tilting pad journal bearing (TPJB) with fixed pivots have evolved as a popular choice [2]. Floating pad journal bearing (FPJB) is a specialized kind of TPJB, particularly suitable for small compact rotors used in cryogenic applications, in which mechanical pivots are absent and pads are suspended on gas film [3] [4]. These bearings were used for the first time in a cryogenic turboexpander in mid-seventies [5]. The work described in this article is an extension of earlier work [6] to include parametric studies of FPJB.

2. Description of bearing operation

The floating pad journal bearing considered for study in this article, consists of three pads placed symmetrically along circumferential direction. As shown in figure 2, the shaft side (A) and housing side (B) surfaces of the pad are connected by a network of three holes(one on 'A' side and two on 'B' side) present at two different axial planes. At the 'A' surface, the connecting hole is located at the geometric centre of 'B' side pad curvature. This point is the centre of pressure on 'A' and is also considered as an imaginary pivot around which the pad rotates. Formation of converging profile is necessary for generation of positive pressure between the shaft and each pad for which the centre of pressure should lie towards the trailing edge instead of centre of the pad. To incorporate this functionality, a wedge at the trailing edge is provided to indirectly shift the pivot position towards the trailing edge of the effective pad length. The rotation of the shaft due to nonslip condition and the viscous effect, moves gas into the gap between pad and the shaft forming the aerodynamic gas film. The tilting action of the pad forms a convergent profile generating pressures which support the shaft gas film. A fraction of the gas in each of the converging films is fed to the 'B' surface (via network of holes), where it is used to float the pad on gas film between the pad and the housing. The gas film thickness on 'B' side of the pad dictates the mass flow rate through holes, which indirectly affects the pressure on 'A' surface.

There are three types of forces that acts on the pad, namely, aerodynamic load and viscous force on 'A' side and load due to pressure on 'B' side. The total force and moment on pad are responsible for movement of pad. For a stable equilibrium of the pad, these forces and moments vanish. During this state, the pressure profile generated on 'A' surface is responsible for the force generated on the shaft.

![Figure 1. Schematic of FPJB [2].](image1)

![Figure 2. Features of a single floating pad [2].](image2)
3. Numerical Model
The steady state modeling of FPJB is done using long bearing approximation (LBA) in which the variation of pressure in the axial direction is neglected [7]. Reynolds equation [7] for compressible flow is used for the 'A' and 'B' bearing surfaces. Poiseuille's flow equation is used to model the pressure loss across the feed holes. Finite Element Method (FEM) is used to solve the nonlinear Reynolds equation [8]. The stable equilibrium position of pad is found out through computation of unbalanced forces and moments using Steepest Descent Method (SDM) [9]. When equilibrium position of pad is found, the radial and circumferential force that it generates on the shaft can be computed by equations (1) and (2). Load capacity of the pad is then calculated using equation (3). Force generated by all the pads are vectorially added to find the load capacity of the bearing. This procedure is repeated for different eccentricities of shaft-bearing system and the load capacities corresponding to different eccentricity values are obtained. The static stiffness of the bearing, which is defined as the ratio of change in load capacity to the change in eccentricity, can be calculated by equation (4).

\[
F_r = \sum_{\xi} P L l_e \cos \theta \\
F_\theta = \sum_{\xi} P L l_e \sin \theta \\
W = \sqrt{F_r^2 + F_\theta^2} \\
K_{ij} = \frac{(F_{ij+\Delta i} - F_{ij})i}{\Delta j}
\]
The preload of the pads is defined as the ratio of pad-shaft clearance to bearing-shaft clearance. It is a fixed value for a particular construct of the bearing. From figure 3, preload ratio is defined as:

\[
\text{preload ratio} = \frac{R_p - R_s}{R_b - R_s}
\]

(5)

An in-house computer code was developed using C++ to compute the pressure distribution, load capacity, static stiffness coefficients and heat generation rate in the bearing. Theoretical performance in terms of load capacity, stiffness coefficients and heat generation rate and their variation with different parameters have been computed and presented. Both direct and cross stiffness coefficients are computed for the analysis. But, it is found that cross stiffness coefficients are almost negligible (within computational error) when compared to direct stiffness. Hence, only direct stiffness coefficients are reported in the work presented in this article.

The present study is based on the analysis of FPJB for the turboexpanders to be used in modified Claude cycle based liquefier/refrigerator developed at BARC, India [10]. For the analysis, bearing inputs based on design of standard BARC turboexpanders have been considered and listed in table 2.

| Table 2. Geometrical and operational parameters. |
|--------------------------------------------------|
| Radius of shaft \((R_s)\)                        | 10 mm |
| Radius of housing side curvature of pad \((R_p)\) | 9.25 mm |
| Length of the pad \((L)\)                        | 20 mm |
| Viscosity of helium gas \((\eta)\)               | 20 μPa.s |
| Operating ambient pressure \((P_a)\)             | 8 bar |
| Speed of shaft                                  | 180,000 RPM |

4. Results and Discussions

From the working of FPJB, it can be concluded that movement of pad plays an important role in determining the bearing characteristics. Therefore, it is important to study the movement of pad with respect to changes in shaft positions. A schematic of the analyzed geometry is shown in figure 4, where pads are numbered in anticlockwise manner. The translation of pad in radial direction in response to changes in eccentricity ratio, is presented in figure 5. The positive value of translation represents displacement away from centre of shaft. At zero eccentricity, all the pads move towards the centre generating a thin film thickness between shaft and pad. As the shaft moves towards pad1, the pad starts to move away from centre but the movement is less than the movement of shaft and hence generates even lower film thickness. Meanwhile, pad2 and pad3 move further toward the centre, however, the film thickness between shaft and pads increases since the movement of the shaft is higher. These movements result in decrease in shaft side \((A)\) film thickness for pad1 and increase in the same for pad2 and pad3. Figure 6 presents the translation of pad in circumferential direction in response to changes in eccentricity ratio, with positive value representing movement in the direction of shaft rotation. At zero eccentricity, all pads move in the direction of rotation (i.e. away from leading edge). As eccentricity is increased towards pad1, it moves towards the leading edge, while pad2 and pad3 move away from it. The circumferential translation, however, is insignificant (of the order of 0.1μm) when compared to 10-20μm of radial translation. Figure 7 presents the variation of angle of tilt of the pad around the imaginary pivot of the pad, with counter-clockwise direction assigned as positive. When shaft is at the centre, all the pads move by almost 0.1° in counter-clockwise direction and hence forms a converging film profile. When shaft translates towards pad1, pad2 rotates anticlockwise while pad3 rotates clockwise maintaining their converging profile. As compared to pad2 and pad3, pad1 can maintain its converging profile in spite of not rotating much.
Aerodynamic bearing works only in the presence of relative motion between shaft and bearing and the pressure rise increases with speed due to increase in mass flow rate. The tilting of the pad is also responsible for increase in the pressure in a way similar to TPJB. In addition to these, the translation of pads towards shaft in FPJB, as seen in figure 5, results in decrease in effective film thickness between shaft and pad and thus bringing about increased pressure. The variation of load capacity with speed for different eccentricity ratios ($\epsilon_1/C_1$) is shown in Figure 8. Stiffness coefficient of the bearing also increases with speed of the shaft as shown in Figure 9. Since, the phenomenon of viscous heat generation is related to the viscous force, which increases with speed, the heat generation rate also increases with the speed of shaft. The pressure at the 'B' side is connected to the pressure at 'A' side. The net force generation on the pad is such that with increase in speed, it moves towards the shaft, decreasing the minimum film thickness between pad and shaft, as presented in figure 11.

**Figure 4.** Orientation of bearing for analysis.

**Figure 5.** Translation of pad in radial direction.

**Figure 6.** Translation of pad in circumferential direction.

**Figure 7.** Rotation of pad in counter-clockwise direction.

**Figure 8.** Variation of load capacity with operating speed of shaft.

**Figure 9.** Variation of stiffness coefficient with operating speed of shaft.
The clearances in the bearing are the main geometrical parameters which governs its performance. FJPJB is characterized by two different types of clearances, viz. shaft side clearance ($C_1$) and housing side clearance ($C_2$), as shown in figure 3. When $C_1$ is less than $C_2$, the equilibrium position of the pad could not be determined by the developed computer code. Hence, the results presented in this article are only for the cases when $C_1>C_2$. The variation of stiffness coefficient with $C_1$ and $C_2$ are shown in figures 12 and 13. From the figures, it is evident that the stiffness coefficient of the bearing is inversely proportional to both the clearances. The heat generation, as shown in figures 14 and 15, also decrease with increase in the clearance. Increase in clearance creates more space between shaft and pad and between pad and housing. The minimum film thickness increases with increase in $C_1$ (figure 16), but for higher clearances, the film thickness does not have much variation. Since the net distance between shaft and housing is increased, the film thickness on the housing side will be higher, allowing more mass flow through the feed holes. Thus, pressure at the connecting hole decreases leading to lower effective pressure on the shaft side for higher clearances. When the housing side clearance ($C_2$) is increased, the minimum film thickness between the pad and the shaft keeps on rising, as shown in figure 17; and since the stiffness is less for higher film thickness, the stiffness coefficient decreases with increase in clearance.
Figure 14. Variation of heat generation rate with shaft side clearance.

Figure 15. Variation of heat generation rate with housing side clearance.

Figure 16. Variation of minimum film thickness with shaft side clearance.

Figure 17. Variation of minimum film thickness with housing side clearance.

The variation of bearing characteristics with preload is shown in figure 18-21. Preload value of 1 signifies that radius of the bearing is same as the radius of the pad. If the preload value is less than 1, the default position of pad is pushed toward the centre of the shaft thus decreasing the bearing radius while maintaining the same pad radius. According to figures 18 and 19, when the pads are preloaded (i.e. decrease in the value of preload), the load capacity and stiffness coefficient increases. This phenomenon is a result of decreasing film thickness which is evident from figure 21. As the heat generation is also a function of the film thickness, it increases (figure 20) with lower film thickness.

Figure 18. Variation of load capacity with preload ratio of pad.

Figure 19. Variation of stiffness coefficient with preload ratio of pad.
Figure 20. Variation of heat generation rate with preload ratio of pad.

Figure 21. Variation of minimum film thickness with preload ratio of pad.

5. Conclusions
For the selected parameters of FPJB in the present work, the analysis shows that a converging fluid film between shaft and pad is ensured by movement of pad. The load capacity and stiffness coefficient increase with increase in the speed of shaft. The heat generation rate also increases with the speed of shaft due to increase in viscous force. The load capacity, stiffness coefficient and heat generation rate of FPJB increase with the preload of the pad. The load capacity, stiffness coefficient and heat generation rate is inversely related to the clearances on both sides of the pad. From the analysis presented, it has been observed that the load capacity, stiffness coefficient and heat generation rate are strongly dependent on the minimum film thickness between shaft and pad. The information generated in this paper would be useful in future design and development of new FPJB for cryogenic turboexpander rotors.

The variation of minimum film thickness and its effects on FPJB characteristics are observed and presented. However, the parameters like pivot circle radius (circle passing through imaginary pivot of all pads) and housing side film thickness can also be explored. The presented analysis was based on long bearing assumption where pressure variation along axial direction is neglected. For better accuracy, a 2-D analysis can be performed and variation of bearing characteristics for different length to diameter ratio can also be analyzed.

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