PERFORMANCE ANALYSIS OF NON-CIRCULAR HYDRODYNAMIC JOURNAL BEARING USING CFD

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ABSTRACT

In high speed machine to improve the shaft stability and reduced power losses non-circular bearings are used extensively. Performance of non-circular bearings depends on operational and geometrical parameters such as offset factor and speed. In this present paper the CFD software has been used for analysis of non-circular hydrodynamic journal bearing. The Geometrical model and meshing has been prepared in GAMBIT and simulated by using ANSYS fluent software. It has been observed from the investigation that journal rotational speed and offset factor influence the maximum pressure generated in the clearance space of the bearing.

Keyword: CFD,Non circular hydrodynamic journal bearing, Offset factor, Speed

I. INTRODUCTION

Application of various type of noncircular journal bearing has increased in recent years. The noncircular journal bearings such as lobed and offset journal bearing provide better performance than circular hydrodynamic journal bearing. Plain circular journal bearings operate with only a single active oil film which results in high rise in Pressure and temperature. This is eliminated in noncircular hydrodynamic bearing as they operate with more than one active fluid film. These bearings also possess superior stiffness, damping and reduced temperature in oil film as compared to circular hydrodynamic journal bearing. Non-circular hydrodynamic journal bearings improve journal stability, decrease power losses and increase oil flow.This bearings are used in steam turbines, steam generators, gear boxes, connecting turbine. Amit Chauhan et al. [1] analysed offset-halves and elliptical bearings using different oils. They observed that the temperature rise in lower lobe is higher compared to upper lobe. Mahesh Aher et al. [2] presented the Pressure Distribution Analysis of Plain Journal Bearing with Lobe. They compare the pressure distribution and load carrying capacity of bearing at different speed.

Amit Chauhan et al. [3] analysed thermal effects in elliptical journal and lobed bearings.They conclude that the temperature rise in lower lobe of the elliptical bearing was higher than the upper lobe in every case. A.D. Rahmatabadi et al [4] analysed Micropolar lubricant effects on the performance of noncircular lobed bearings. They investigated effects of the coupling number and size of material characteristic length on the static performance of bearing. . Taylor et al. [5] performed the experimental investigation of thermal effect in circular and elliptical plain journal bearing. R. Sinhasan et al. [6] has studied the analysis of two-lobe porous hydrodynamic journal bearings. They conclude that porosity has important role for both static and dynamic characteristics of two-lobe porous hydrodynamic journal bearings. Mehta et al. [7] has been presented two lobe bearings for stability using couple stress fluid. It was observed that using the couple stress fluid increased the load bearing capacity. Gregory Patella [8] analysed pressure distribution in bearings of various profiles viz. Elliptical, offset-halves and circular bearings.They observed that circular journal bearings are subjected to oil whirl at increased rotational speeds, hence, elliptical and offset-halves bearings are used. Dinesh Dhande et al. [9] analysed circular journal bearing for static pressure distribution, stress in bearing and deformation in bearing using fluid structure interaction. The results obtained in this paper were used to compare the contour plot and static pressure values of circular bearing and thus validate the results obtained for elliptical bearings in this paper.

In the present paper the analysis of pressure distribution in the fluid film has been done using CFD software.The bearings has been modelled and meshed in Gambit and then imported into Ansys Fluent. The offset factors considered were 0.8, 0.9, 1.1, 1.2 and rotational speeds of journals were 1000 rpm, 2000 rpm and 3000 rpm. The results are validated by obtaining contour plot for static pressure distribution in circular hydrodynamic journal bearings corresponding to the same journal speeds and compare with available data.

II. ANALYSIS

When analysis of a bearing was done using Reynolds Equation, Reynolds equation governs the pressure distribution around the circumference of journal in the clearance space of fluid film bearing. It is used to plot the Pressure profile in the hydrodynamic fluid film theory.The coordinate system and the geometry of fluid film journal bearing is shown in Fig. 1. An equilibrium position under the external load is attended due to the hydrodynamic pressure generated by the lubricant film through journal rotates with an angular velocity ω.

$$\frac{\partial}{\partial x}(h3\frac{\partial p}{\partial x})+\frac{\partial}{\partial z}(h3\frac{\partial p}{\partial z})=6U\mu\frac{dh}{dx} \quad (1)$$

The assumption of the noncircular hydrodynamic journal bearingas it is rigid aligned bearing with steady state condition. The flow is Newtonian, isothermal, inviscous, incompressible and no inertia effect. Both Journal and bearing surfaces are smooth and a constant vertical load is only to be applied at journal centre. Lubricant pressure distribution as a function of journal speed, bearing geometry, clearance and lubricant viscosity is described by Reynolds equation.

The objectives of present work as follows:

1. To analyse the variation in pressure distribution in the oil film for various offset factors and rotation speeds of journal.
2. To compare the results of static pressure distribution in elliptical bearing with that ofcircular bearing.

III. METHODOLOGY

In order to analyse pressure distribution in the bearings using ANSYS Fluent software, meshing was done using Gambit. Gambit is pre-processor software used for engineering analysis. Edge meshing was carried out with interval count as 30 along the length and elliptical and circular profile. Face meshing and volume meshing carried out consequently.

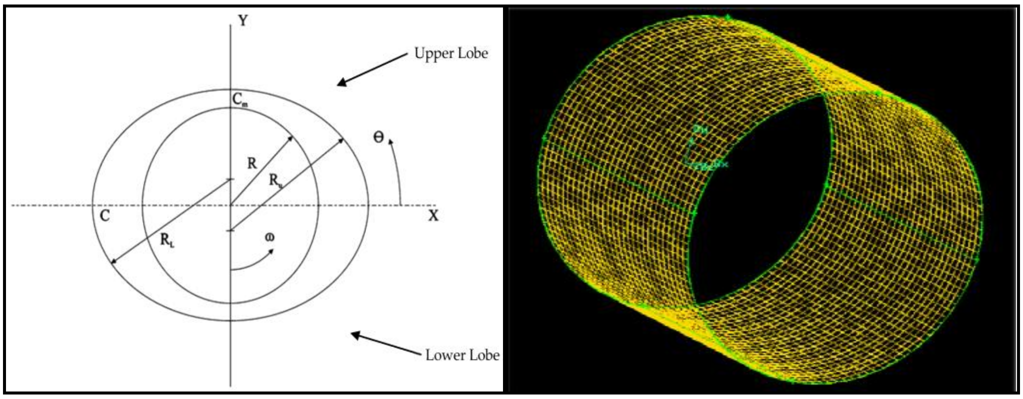


Fig.1Geometry and Mesh model of Non circular hydrodynamic journal bearing With offset factor =0.8

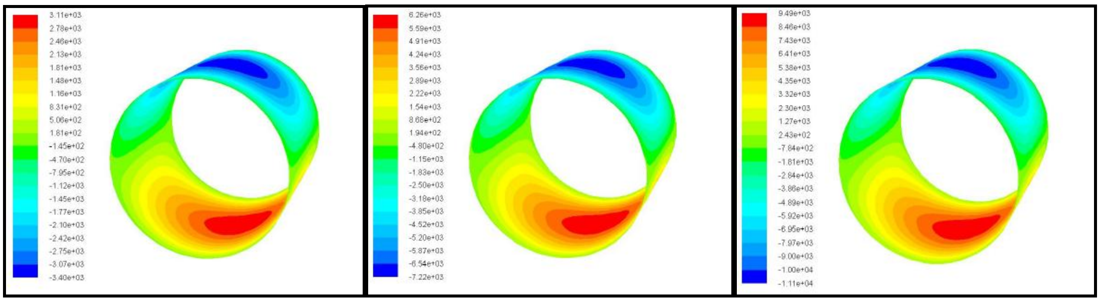
The mesh file imported into ANSYS Fluent to obtain the pressure contour plots. The bearing wall was considered as stationary and the journal was modelled as moving wall. The sides of the lubricant volume had been assigned with a zero pressure condition for lubricant to flow freely. The sides of the bearing were assigned pressure inlet and pressure outlet. The temperature of each of the entities i.e. bearing, journal, pressure inlet and outlet were taken as 40°C. The journal was given a rotational speed in rpm. At the pressure outlet, the pressure distribution was assumed to be radial.The solution scheme was taken as SIMPLE. The spatial discretization of pressure was taken as PRESTO.Standard initialization was used and was computed from the inner most layer i.e. the journal.

Table 1Geometrical and operational parameter of non circular journal bearing

Length of bearing	30 mm
Journal diameter	30 mm
Offset factor	0.8-1.2
Clearance size	50 µm
Fluid Density	780 kg/m^3
Thermal conductivity	0.149 w/mk
Specific heat	2090 j/kgk
Lubricant viscosity	0.0024 kg/ms
Speed	1000/2000/3000 rpm

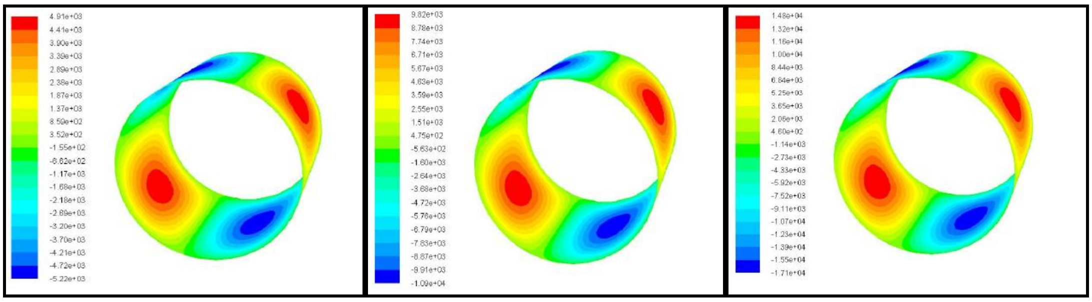
IV. RESULT AND DISCUSSION

The contour plots for noncircular hydrodynamic journal bearing with the offset factors (0.8 to 1.2) and rotational speeds of journal (1000, 2000 and 3000 rpm) areillustrated below. The pressure values are measured in Pascals (Pa).Results have been observed from the investigation that offset factor influence the maximum pressure generated in the clearance space of the bearing.It is observed from the static pressure contour plots that there are two pairs of pressure concentrations developed opposite to each other.



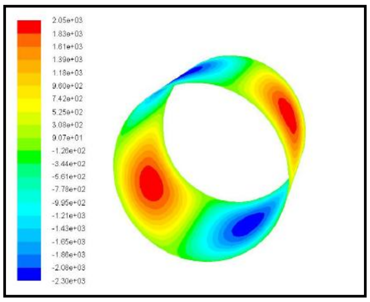
(a) N=1000rpm (b) N=2000rpm (c) N=3000rpm

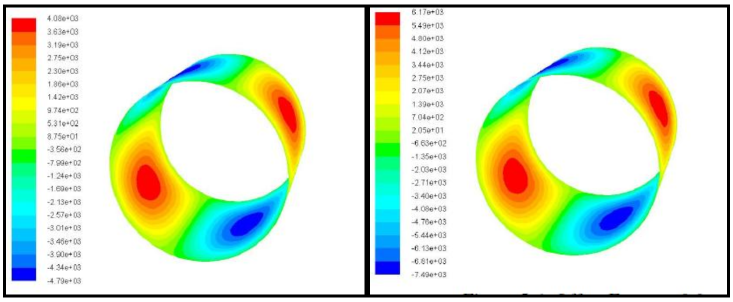
Fig.2: Pressure contour of circular hydrodynamic journal bearing(offset factor =1) with eccentricity ratio = 0.1



(a) N=1000rpm (b) N=2000 rpm (c) N=3000rpm

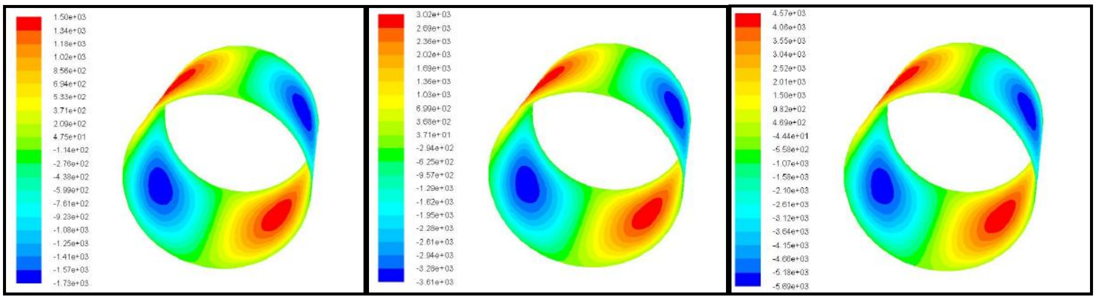
Fig.3: Pressure contour of noncircular hydrodynamic journal bearing with offset factor = 0.8





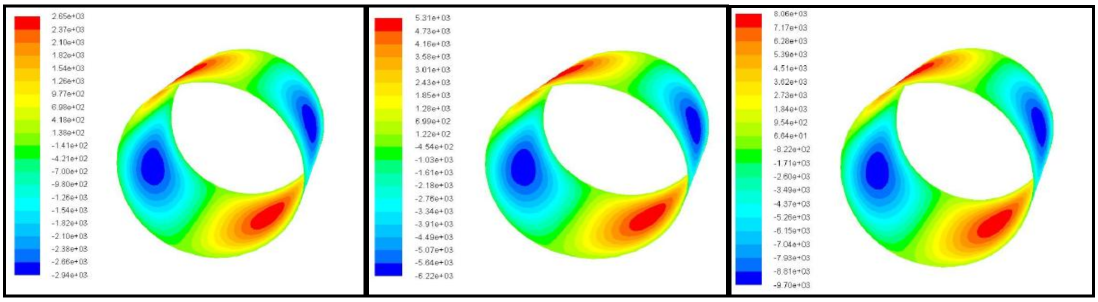
(a) N=1000rpm (b) N=2000rpm (c) N=3000rpm

Fig.4: Pressure contour of noncircular hydrodynamic journal bearingwith offset factor =0.9



(a) N=1000rpm (b) N=2000rpm (c) N=3000rpm

Fig.5: Pressure contour of noncircular hydrodynamic journal bearingwith offset factor =1.1



(a) N=1000rpm (b) N=2000rpm (c) N=3000rpm

Fig.6: Pressure contour of noncircular hydrodynamic journal bearingwith offset factor =1.2

The above method and results are validated by comparing the nature of static pressure contour plot of circular bearing.It can be observed that there are two pairs of maximum and minimum static pressure areas that are generated. The point where the area with maximum and minimum pressure will be generated is influenced by the offset factor. It can be observed from the contour plots of circular journal bearing that the two pressure areas lie on opposite sides of the profile. One of the areas exhibits maximum static pressure and on the side opposite exist an area which exhibits the minimum static pressure.These results in larger clearance andelliptical bearings are also susceptible to oil whip and whirl.Ideal operating conditions for any bearing refers to isothermal conditions. In actual operation there are changes in temperature while operation. Isothermal conditions do not prevail in actual operation of a bearing. If we consider a circular bearing, due to this change in temperature the profile of the bearing will again be changed, thus getting converted into a non-circular bearing. The heating due to temperature rise will cause a change in profile of the bearing.

The graph for the maximum pressure generated v/s offset factor for the three journal speeds was plotted and graph for maximum pressure generated v/s journal speed for various offset factors was also plotted.

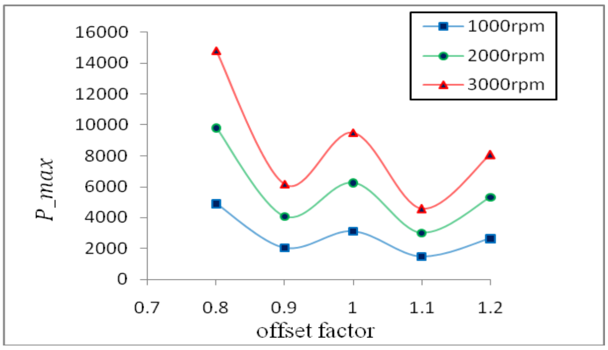


Fig.7 Maximum pressure generated v/s offset factor plot for Non circular hydrodynamic journal bearing with various Speed, Kerosene as a lubricant

It can be observed from the above graph that as the journal speed increases, for the same offset factor, values of maximum pressure generated increases. The maximum pressure generated in circular bearing is less than maximum pressure generated in the bearing with offset factor = 0.8.

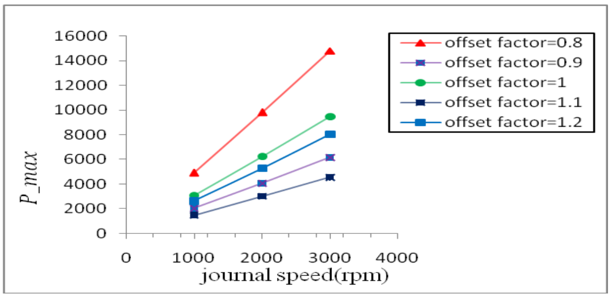


Fig.8 Maximum pressure generated v/s journal speed Of Non circular hydrodynamic journal bearing with different offset factor

As observed in the graph, the range of maximum pressure generated in the bearing with offset factor = 0.8 is the highest and the range of maximum pressure generated in the bearing with offset factor = 1.2 is the lowest.The maximum pressure generated in circular bearing is less than maximum pressure generated in the bearing with offset factor = 0.8

V. CONCLUSION

The point where the area with maximum and minimum pressure will be generated is influenced by the offset factor. From the contour plots of elliptical bearing it is observed that for offset factors more than 1 i.e. 1.1, 1.2, the area exhibiting maximum pressure concentration lies on the opposite side of the profile when compared to bearings with offset factor less than 1 i.e. 0.8, 0.9. The maximum pressure generated in circular bearing is less than maximum pressure generated in the bearing with offset factor = 0.8. As observed in the graph 2, the range of maximum pressure generated in the bearing with offset factor = 0.8 is the highest and the range of maximum pressure generated in the bearing with offset factor = 1.2 is the lowest.

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