Analysis of Grooves Profile and Performance of Hydrodynamic Bearing

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Abstract- *The present work deals with analyzed pressure distribution of journal bearing. By modifying the surface texture of bearing i.e. apply the different types of grooves on bearing and identify the differences occurs in the force acting on bearing as well as pressure distribution. This can be done with help of computational fluid Dynamic (CFD). It is observed that change of grooves profile affect the static pressure & force at center of bearing.*

I. INTRODUCTION

Now A Days, almost in all industries energy transfer take place from one machine component to another machine component or from one shaft to another shaft. Recently such industry never be exist where motion transfer or energy transfer component or device like electric motor, turbine, generator pump not present. This energy transfer or motion transfer take place by transmission device like shaft which get supported with the help of bearing.

These new machine design applications require high operating speed, higher power density, small size, and high load carrying capacity and to full fill this requirement it is necessary to consider transmission section. So one of the most important element to be considered for design is bearing. In a hydrodynamic journal bearing pressure of hydrodynamic lift is generated in thin lubricating oil film that separates the shaft and the bearing thus preventing metal to metal contact.

R.Pai, D.J.Hargreaves and R. Brown analysed [1] the performance of journal bearings with water grooves. The CFD results indicate that the maximum pressure zone in the bearing has moved towards the outlet and the pressure along the axial groove increased rapidly. And Shelly and Ettles [2] analyzed the performance of hydrodynamic bearing with oil grooves at maximum pressure location. They found that because of positioning it will cause 30 to 70 reduction in the load capacity of the bearing.

The objectives of this study were to:

1. Predict the pressure in the lubricant using computational fluid dynamics (CFD).
2. Measure the pressure in the lubricant film in the journal bearing operating with various loading conditions.

II. ANALYSIS

A FLUENT 14 was used to find or predict pressure distribution in the bearing. The computational fluid dynamic was use to solved equation for pressure and velocity

Fig 1: Schematic of 3 axial groove Journal bearing geometry

III. METHODOLOGY

Main objective in this research is to find out maximum static pressure for different load W.

IV. GEOMETRICAL MODEL

The bearing dimensions for schematic diagram fig 1 used in the present work are as given below:

Table I
JOURNAL BEARING PROPERTIES

Symbol	Quantity	Values
Rb	Journal Radius	50 mm
L	Bearing Length	80 mm
C	Radial Clearance	145 μm
Rl	Lobe height	3mm
	Angle between lobes	120^0
W	Load Range	2000- 5000 N
μ	Lubricant viscosity	0.0277 Pa-sec
ρ	Lubricant density	860 kg/m³
C_p	Lubricant Specific Heat	2000 J/kg °C

The model is drawn as one cylinder (journal) with a radius of Rb 50 mm and another one with a radius of 50.145 mm, i.e. bearing. The CFD analysis is done with ANSYS Fluent a hexahedral structure mesh is used. (Fig. 2) 100 divisions were taken along the length. So the total number of elements is 75840.The mesh was generated for different values load 2000N to 5000N. The mesh quality is always around 0.5 for all generated elements.

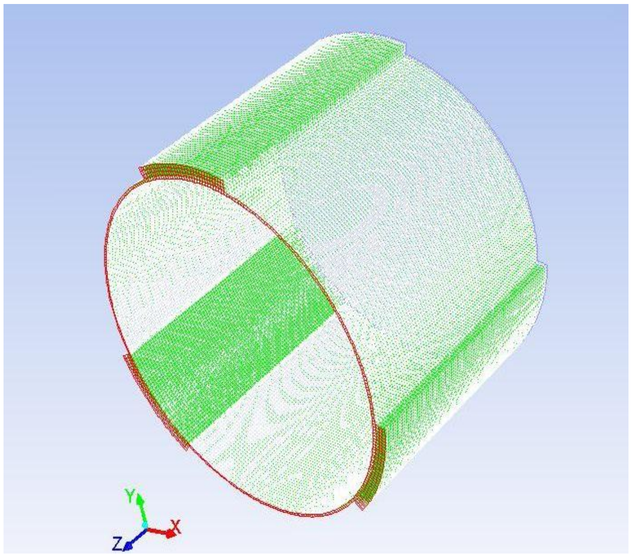

Fig 2: Meshed Model

V. BOUNDARY CONDITIONS

The inlet to the bearing is at the rear in figure 2. And was set as type- ‘pressure inlet’ with the supply pressure 50 kPa. The outlet of the bearing is at the front in Fig2.and was set as type- ‘pressure outlet’ with the outlet pressure 42 kPa. The bearing shell was modelled as a ‘moving wall’ with absolute motion of 0 rpm.The rotational axis origin was set at X=0, Y=0, Z=0 and direction of the axis was set as X=0, Y=0, Z=-1.The journal was modelled as a ‘moving wall’ with a motion relative to the adjacent cell zone at an rotational speed of 1000rpm.The rotational axis origin for the journal was set at the eccentricity, which is X=0.0664mm, Y=0.04149mm, Z=0 for a load of 2000 N and speed of 1000 rpm. The rotation axis direction was set as X=0, Y=0, Z=-1. The oil 142 in the clearance volume was modelled as type- ‘fluid’, with rotation axis origin and direction same as that of the journal above. The rotational speed was set at 1000 rpm in the same manner as that for the journal.The under-relaxation factors used for pressure, momentum, and density and body forces are 0.3, 0.7,1 and 1 respectively for the solution. The discretization used is ‘presto’ for pressure, ‘quick’ for momentum and ‘simple’ for the P-V coupling.

VI. RESULTS

Fig 3 shows static pressure contour plot neglecting negative pressures. L/D ratio = 0.8 with 1000rpm. And 2000N.

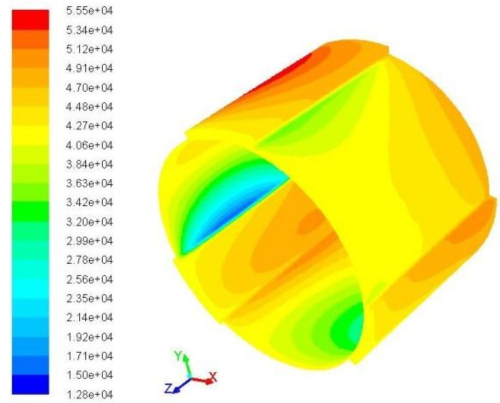

Fig 3: Pressure contours for 1000 rpm, W=2000 N L/D=0.8.

Fig 4 shows static pressure contour plot neglecting negative pressures. L/D ratio = 0.8 with 1000rpm & 3000N.

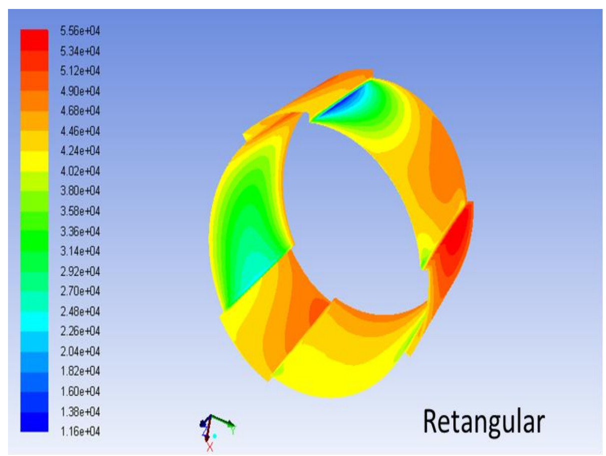

Fig 4: Pressure contours for 1000 rpm & W=3000 N, L/D=0.8

Fig 5 shows static pressure contour plot neglecting negative pressures. L/D ratio = 0.8 with 1000rpm & 4000N.

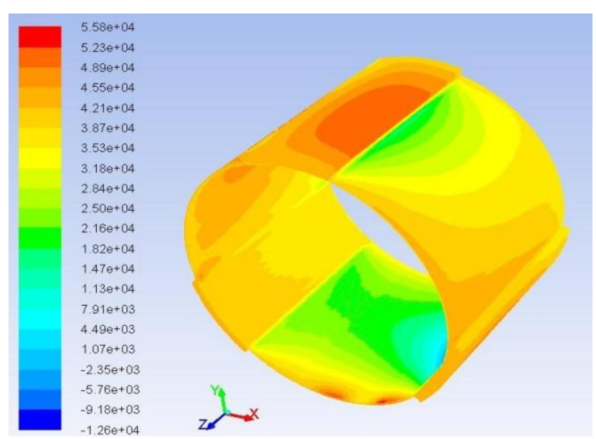

Fig 5: Pressure contours for 1000 rpm, W=4000 N L/D=0.8.

Fig 6 shows static pressure increases with increasing the load. L/D ratio = 0.8 with 1000rpm & 5000N.

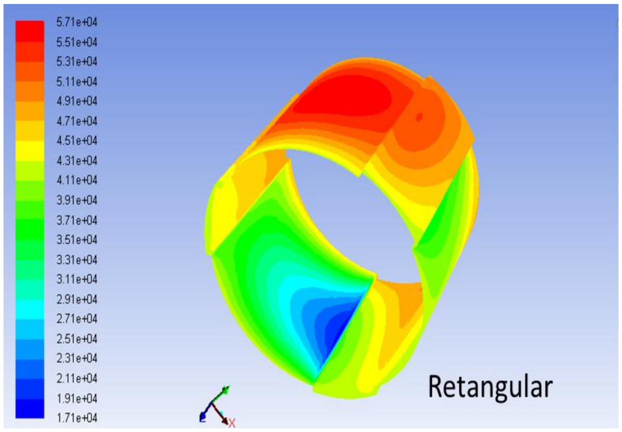

Fig 6: Pressure contours for 1000 rpm, W=4000 N L/D=0.8.

Fig 7 shows static pressure increases with increasing the load. L/D ratio = 0.8 with 1000rpm & 2000 N

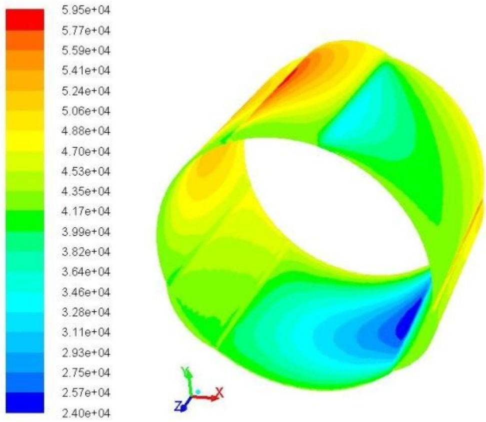

Fig 7: Pressure counter W=2000N L/D=0.8.
(Sinusoidal Groove)

Fig 8 shows eccentricity increases with increasing the load/force. L/D ratio = 0.8 with 1000rpm & 3000 N

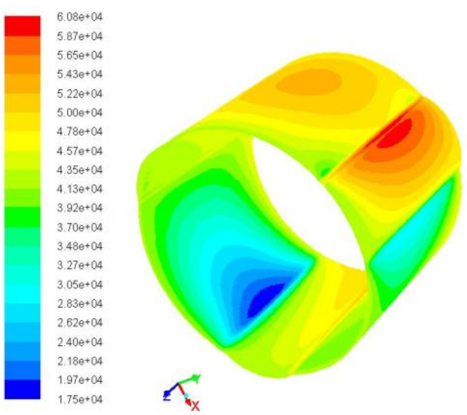

Fig 8: Pressure counter W=3000N L/D=0.8.
(Sinusoidal Groove)

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Fig 9 shows angle in degree increases with increasing the load/force. L/D ratio = 0.8 with 1000rpm & 4000 N

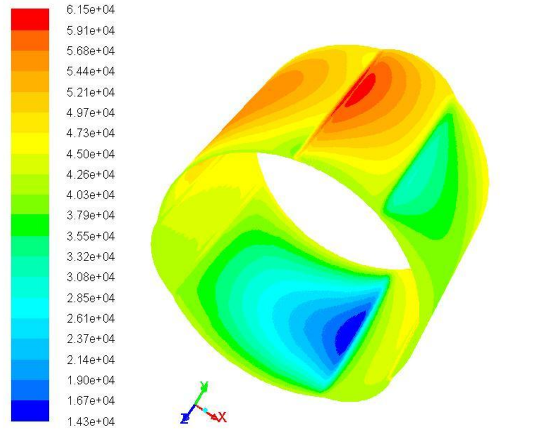

Fig 9: Pressure counter W=4000N L/D=0.8.
(Sinusoidal Groove)

Fig 10 shows ho increases with increasing the load/force. L/D ratio = 0.8 with 1000rpm & 5000 N

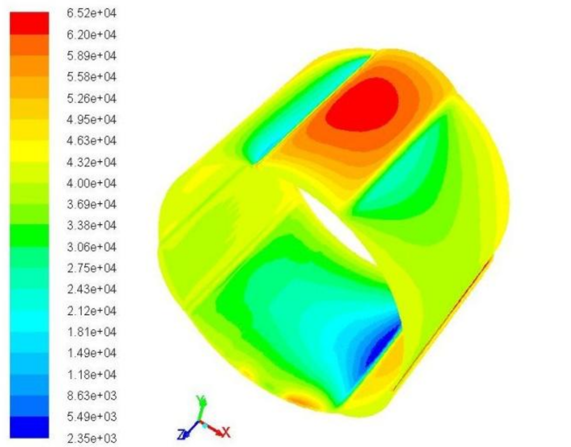

Fig 10: Pressure counter W=5000N L/D=0.8.
(Sinusoidal Groove)

VII. CONCLUSION

The static pressure distribution having maximum value in 3 rectangular grooves bearing than simple bearing As the static pressure increases it increase force at the center of bearing. It conclude that the load carrying capacity of groove bearing is more than the plain bearing and result show that presence of grooves highly effect the performance of bearing.

ACKNOWLEDGMENT

Authors are thankful to Mr.Chopde, CAE Engineer, TATA TECH. Pune for his help during meshing

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