Performance Analysis of Triangular Grooved Hydrodynamic Journal Bearings

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Abstract- *Hydrodynamic journal bearings are modified by introducing the grooves on it. The present work deals with the analysis and performance of semicircular and triangular grooved journal bearing. During the experimentation on modified hydrodynamic journal bearing can identify the pressure distribution and the result can compare with CFD Computational Fluid Dynamic analysis. The semicircular grooves and triangular grooves help to maintain the fluid film thickness. For hydrodynamic bearings it is important that minimum film thickness never drops below a safety limit. Pressure created in the fluid film help to avoid friction between journal and bearing.*

Keywords- CFD, Load carrying capacity, Fluid film thickness, Reynolds Equation.

I. INTRODUCTION

Knight et al the CFD results indicate that the maximum pressure zone in the bearing has moved towards the outlet. The pressure contours obtained, can be used to understand the flow characteristics of the bearing.[25]

Gertzos et al done “CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant”. It shows that the load carrying capacity, the film pressure, and the frictional force of the Bingham solid are larger than those of the Newtonian fluid.[15]

G.Descombes et al finding about “ CFD model for turbocharger journal bearing performances”, the paper try to access the heat transfer for different parts of the turbocharger, such as compressor, turbine and bearing housing and done analysis by CFD and Experiment for pressure and temperature.[12]

The purpose of study were to:

1. To find out the pressure in the lubricant using computational fluid dynamics (CFD).
2. Pressure impact at different speed of Journal.

II. ANALYSIS

For modeling CATIA V5 R20 and analysis purpose a FLUENT 14 was used to find out the pressure distribution in the bearing. The CFD was used to solved equation for pressure.

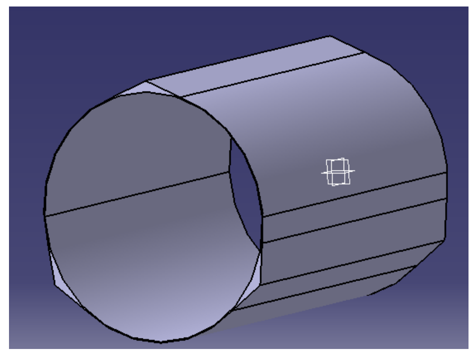

Figure 1 Conceptual model for triangular 3-Groove Bearing

III. METHODOLOGY

Main objective I research is to find out maximum static pressure for different speed.

IV. GEOMETRICAL MODEL

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

Table 1 Journal Bearing Properties

Symbol	Quantity	Values
Rb	Journal Radius	22.5 mm
L	Bearing Length	67.5 mm
C	Radius Clearance	145 μm
Rl	Lobe height	3mm
θ	Angle between lobes	120°
W	Load Range	1000N
μ	Lubricant Viscosity	0.0277 Pa-sec

The model is drawn as one cylinder (journal) with a radius of Rb. 22.5 mm another one with a radius of 22.645 mm, i.e bearing. The CFD analysis is done with ANSYS fluent a hexahydral structure mesh is used.(Fig. 2) 100 divisions were taken along the length. So the total number of the elements is 75840. The mesh was generated for different values speed 1000 rpm to 4000 rpm. At constant load 1000 N. The mesh quality always around 0.5 for all generated elements.

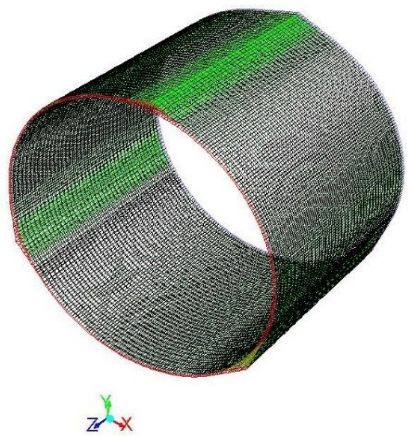

Figure 2 Meshed Model

Grid details
Number of mesh elements= 75840
Minimum skewness= 0.5

Boundary Conditions

The supply pressure for the ‘Pressure Inlet’ is 50 kPa. The inlet of bearing is rear side of bearing in fig.2 and outlet of bearing is front side. ‘Pressure outlet’ is 42 kPa. The bearing shell was modelled as a ‘moving wall’ with absolute motion of 0 rpm. The rotational axis origine was set at X=0, Y=0, Z=0 and direction of axis was set as X=0, Y=0, Z=-1. The Journal was modelled as moving wall with a motioin relative to the adjustant cell zone at an rotational speed 1000 rpm. The rotational axia for Journal was set at eccentricity, which is X= 0.0664mm, Y=0.04149mm, Z=0 for a load of 2000 rpm at 1000N load.The rotation axis direction set as X=0,Y=0,Z=-1.The oil 142 in the clearance volume was modelled as a type ‘fluid, with the rotation axis origin and direction same as that of journal above. The rotational speed was set as 1000 rpm to 4000 rpm. The load constant 1000N for all conditions.

V. RESULTS

Fig 3 Shows static pressure contour plot neglecting negative pressure with 1000 rpm and 1000N with Plain bearing

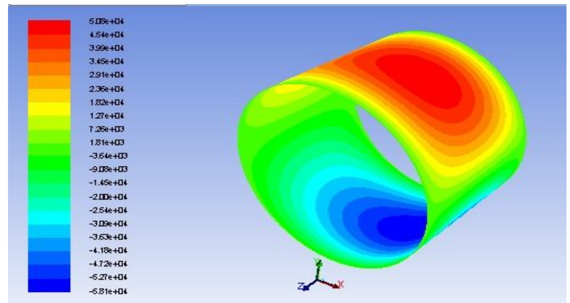

Figure 3 Pressure contours for Plain bearing for 1000 rpm at 1000 N

Fig.4 Shows Pressure contour of bearing for 1000 rpm and 1000N with Triangular Groove

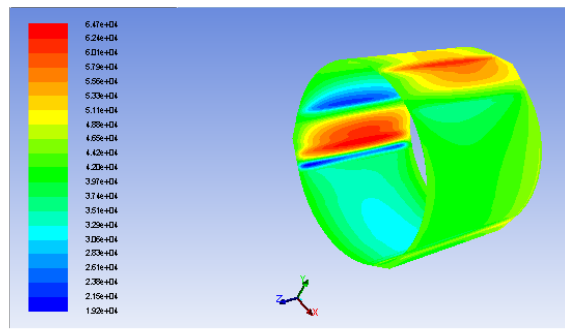

Figure 4 Pressure contours for Triangular grooved bearing for 1000 rpm at 1000 N

Fig.5 Contour plot for pressure variation in Semicircular bearing at 1000rpm for Semicircular groove bearing

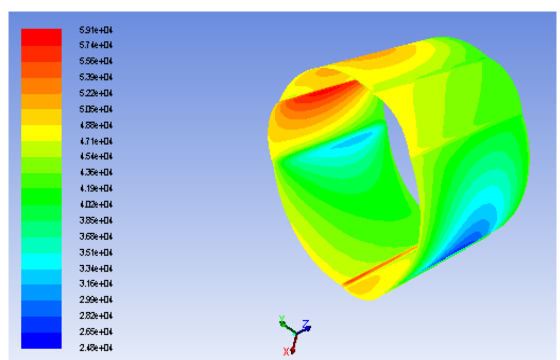

Figure 5 Pressure contours for Semicircular grooved bearing for 1000rpm at 1000 N

Table 2 CFD and Experimental Max. Pressure values at different speed

Speed	Max. Pressure for Plain Bearing		Max. Pressure for Triangular Groove		Max. Pressure for Semicircular Groove	
RPM	CFD	EXP	CFD	EXP	CFD	EXP
1000	0.5	0.48	0.65	0.61	0.59	0.56
2000	0.51	0.49	0.66	0.63	0.6	0.57
3000	0.52	0.5	0.67	0.64	0.6	0.57
4000	0.54	0.52	0.69	0.67	0.62	0.59

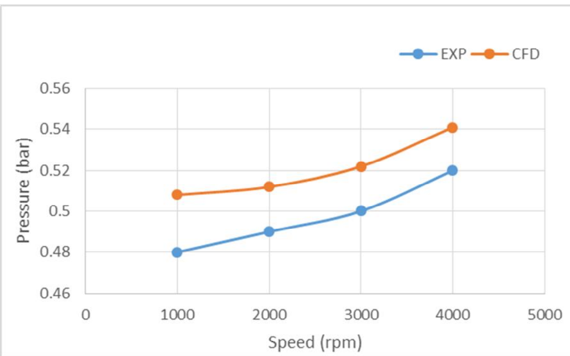

Figure 6 Scatter Plot for pressure variation along speed of journal at 1000 rpm and 1000N for Plain bearing

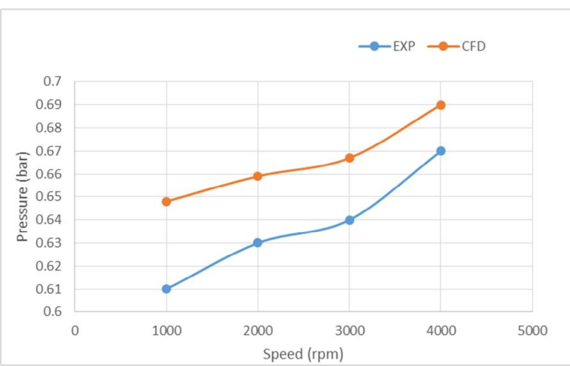

Figure 7 Scatter plot for pressure variation along speed of journal at 1000 rpm and 1000N for Triangular groove bearing

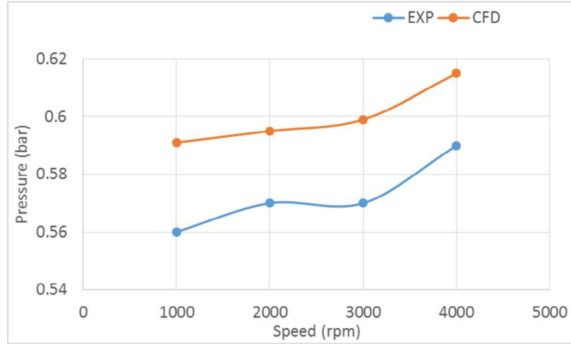

Figure 8 Scatter Plot for pressure variation along speed of journal at 1000 rpm and 1000N for Semicircular groove bearing

VI. CONCLUSION

The hydrodynamic journal bearing are modified in different ways such as by making dimples, lobes, grooves, in different shapes on journal bearing for improvement of performance. Due to textured surface of journal bearing, it get affected on fluid film, load carrying capacity, maximum pressure at the circulating fluid, temperature of lubricant. It can be find out by analytical method or experimental or CFD software. The comparative values are nearly same. The values from theoretical may be up to 20% greater than experimental value.[2] Finally it can conclude that if the hydrodynamic bearing modified by different texture of different shapes, its performance get improved as compare to plain journal bearing.

APPENDIX

Symbol/Notation	**Name of Symbol[16]**
h	Film Thickness
C	Specific heat capacity of lubricant
D	Journal diameter
F	Friction force
F_h	Ratio of friction loss to viscosity
g_O, g_S	Pressure correction factors for Ocvirk's And Sommerfeld bearings
H, Hpmax	Non-dimensional film thickness, film thickness at maximum pressure location
L	Bearing length
N	Journal rotational speed
P	Dimensional film pressure
Pmax	Maximum pressure
Rj	Journal radius
Tmax	Inlet temperature, effective temperature, maximum temperature
V	Journal surface velocity
W	Dimensional load capacity
We, Wf	Dimensional load capacity along and

	perpendicular to line of centers	
W_η	Ratio of dimensional load capacity to viscosity	m2/s
$W_{\varepsilon\eta}$, $W_{\text{Ø}\eta}$	Ratio of W_ε, $W_{\text{Ø}}$ to viscosity	m2/s
ʌ	Slenderness ratio (L/D)	
E	Eccentricity	
ε	Eccentricity ratio	
Ø	Attitude angle	radian
H	Viscosity coefficient of lubricant	Pa.s
Θ	Co-ordinate in circumferential direction	radian
θ_{Omax}, θ_{Smax}	Location of maximum pressure for Ocvirk's and Sommerfeld bearings	radian
P	Density of lubricant	kg/m3
		rad/s

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