Performance of Modified Hydrodynamic Journal bearings: A Review

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Abstract- *Hydrodynamic journal bearings are applicable in many areas, where the heavy loaded rotating devices are used. It is necessary to increase the load carrying capacity and decreases the different losses, for that there is a way to improve the characteristic of journal bearings is to modify the surface texture of bearing. When the hydrodynamic bearing operate under high speed, due to shearing of the lubricant and temperature rise of the lubricant fluid film on the bearing surface heat is generated within the oil film. It causes significant reduction of the viscosity of the lubricant, and at a lower value of minimum fluid-film thickness the bearing operates properly. Thus, the flow field of the lubricant becomes distorted and the bearing performance is affected. So to avoid the distortion of the fluid film inside the bearing apply the groove inside bearing and further analysis done on it by considering pressure factor. This paper shows a survey of important research papers which is used to determine load carrying capacity, minimum film thickness, friction losses and temperature distribution by modifying the hydrodynamic bearing.*

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

I. INTRODUCTION

The hydrodynamic bearing is used many industrial areas where the energy transfer takes place from one machine component to another machine component or from one shaft to another shaft.

The device like electric motor, turbine, generator and pump are the common applications of hydrodynamic 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. Bearing not only support rotor weight and operating speed but also influence on rotor dynamic behavior. For hydrodynamic bearings it is important that minimum film thickness never drops below a safety limit. Bearing operates at a lower value of minimum fluid-film thickness, when the hydrodynamic bearing operate at high speed, the temperature of lubricant get increases and viscosity get decreases. It causes reduction of fluid film thickness and its flow get distorted. For hydrodynamic bearings it is important that minimum film thickness never drops below a safety limit. The theoretical approach is to solve Reynolds equation and developing a CFD model to examine pressure and temperature profiles and the results are validated with the work done by experiment. Then validated approach is used to study the behavior of surface texture of bearings. Effect of pressure on bearing material can be finding out by CFD analysis and experiment. Hydrodynamic type journal bearings are considered to be a vital component of all rotating machinery. A journal bearing consists of a stationary cylindrical body (sleeve) separated from a rotating shaft by a layer of lubricant. Load carrying capacity of journal bearing is dependent on pressure in layer of lubricant during rotation of shaft or journal.

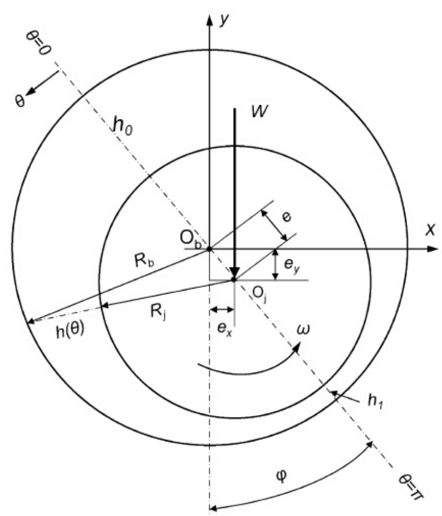

Figure 1. Schematic journal-bearing geometry [1]

In case of oil SAE 50, it is important to calculate Reynolds number to determine the case of laminar or turbulent. So Reynolds equation are as follows

$$Re = \frac{\rho \, v \, h^2}{\eta L}$$

If the Reynolds number is above 1 that indicate flow is turbulent flow and if less than 1 then its Laminar flow.[15]

Procedure for finding out Maximum Pressure and its location

All the parameters of journal bearing and oil properties are taken from Ferron et al case so with the help of this operating parameters it is possible to determine maximum pressure and its location in journal bearing.[16]

Pressure profile in terms of θ and z coordinate is given by

$$P = \frac{12\eta\, v\, Rj}{c^2 g_0}\left(\frac{L}{D}\right)^2 \left[\frac{\left\{\frac{1}{4} - \left(\frac{Z}{L}\right)^2\right\}\frac{\varepsilon \sin\theta}{(1+\varepsilon\cos\theta)^3}}{1 + \frac{2\, g_s \left(\frac{L}{D}\right)^2 (2+\varepsilon^2)}{g_o\,(1+\varepsilon\cos\theta)(2+\varepsilon\cos\theta)}\left\{\frac{1}{4} - \left(\frac{Z}{L}\right)^2\right\}} \right]$$

The coordinate system and the geometry of a journal bearing are shown in fig. The journal rotates with an angular velocity o and is in an equilibrium position under the external vertical load was well as the pressure of the lubricant film. The journal axis Oj is at distance e from the bearing axis Ob. The film thickness h(y) varies from its maximum value h0 at bearing angle y ¼ 0 to its minimum value h1 at y ¼ p. [1] It developed journal bearing with two axial groove and find out the maximum pressure by journal bearing test rig. The experimental and theoretical pressure find out of two axial groove of journal bearing and comparison made between them. It is obtained that the maximum pressure using the convention approach are found to be ~20% lower than the theoretical value.[2].CFD done to compare power friction loss due to hydrodynamic journal bearings. Speed and temperature factor is considered during CFD. The result of CFD compared with experiment, the friction losses of turbocharger bearing which seems closed to experimental results. It shows that by decreasing oil temperature the friction torque also decreases. Such study can find out heat transfer rate for different parts of turbocharger, compressor, turbine and bearing house holding.[3]

The compound dimple and simple dimple are made on hydrodynamic bearing. Performance comparison made between simple bearing and compound dimple bearing. The simple dimple bearing like rectangular shape and compound dimple bearing are like rectangular-spherical shape. The load carrying capacity of compound dimple bearing is more than simple dimple bearing. The result shows the lower value of friction coefficient. The severe cavitation of lubricant used for bearing in compound dimple hydrodynamic bearing.[4] for adequate dimple sizes, fully textured can minimize the thickness up to +27% and reduced the 3.4% friction coefficient reduction. Due to the textured area the distribution of pressure is better than plain bearing. The location of domain on the journal bearing is the main criterion for journal bearing performance enhancement. The shaft (journal) is supposed to be smooth and rigid while the bearing surface is partially or totally textured with cylindrical textured shape. Different arrangement of the textured are have been considered. The presence of the cavity (textured) increased locally the lubricant film thickness and decreases the friction force. In the complex case of a journal bearing with both convergent (hydrodynamic pressure) and divergent (cavitation) zones, partial texturing has minimal positive effect and fully texturing has a negative effect.[14] The water lubricant plain journal bearing are experimented and CFD also done on it. Its design by considering eccentricity ratio as main designing factor, 0.6 or 0.7 eccentric ratio is better for water as lubricant. The performance of hydrodynamic plain bearing is improve if the eccentric ratio is set to in proper range.[5] During CFD analysis of journal bearing hydrodynamic lubrication by Bhingham lubricant were used, The load carrying capacity, the film pressure and the frictional force of Bhingham solid are larger than those of the Newtonian fluid. The effect of yield stress To on the journal behavior is small.[10]

The tribological behaviors of non-groove and micro-grooved journal bearing under dynamic loading condition were investigated. The grooves are help to increase the roughness on the bearing for higher friction. The roughness was cut precisely in triangular and trapezoidal shape. The experiment shows that the lowest value of friction force was determine on the plain journal bearing. The experimentally obtained values increased progressively for the circumferential and the hearing bone micro-grooved bearing and finally highest frictional force distribution was exercised on the transversal micro-grooved bearing.[6] During the experimentation on plain and micro grooved journal bearing, the stribeck curves obtained experimentally for circumferential and micro-grooved bearing as like plain journal bearing. With comparing with plain bearing the highest value for the coefficient of friction are found out at multi-grooved journal bearing. As the load increases the coefficient of friction of micro-grooved journal bearing decreases as per theory of plain journal bearing. Also with increasing the load, the transmission speed increases for micro-grooved bearing whereas it remains constant for the plain journal bearings for all loading condition.[9]

An experimental work was conducted to determine the effect of oil groove location on the temperature and pressure in the hydrodynamic journal bearing. A journal with diameter of 100mm and a length to diameter ratio of 1:2 was used. The oil supply pressure was set at 0,20-0.25 Mpa. The groove was positioned at a seven different locations, namely -45°, -30°,-15°, 0°, +150°, +30° and +45°. At different radial load and speed the pressure were obtained. The temperature profile tends to decrease when the oil groove supply is located in the converging section near the minimum film thickness position.[7]

Stability analysis of hydrodynamic bearing with herring grooved sleeve, for this to obtained dimensionless Reynolds equation is solved by using the finite difference method. For finding the discontinuity of the oil film thickness Green's Theorem is used and Gaussian elimination method is used to solve the simultaneous equations. Analysis shows that the static load capacity W, the side leakage Qs, The circumferential flow rate Qc, and the total friction force Ff, for all kinds of bearings increase with increase in eccentric ratio ε.[13] performance of Twin axial groove bearing is better than single groove bearing. The experimental assessment of Journal bearing with one or two axial grooves located perpendicularly to the load line was performed. It was found that under a heavy loaded operation the tween groove configuration might actually deteriorated the bearing performance when compare with single groove arrangement due to uneven feed through each groove. After implementation of flow balancing strategies the feed flow rate through each groove can be used to improve bearing performance. Total flow rate in the bearing in single and twin groove bearing are nearly same, but for the twin grooved bearing the partial flow rate varied with increasing specific load. At high eccentricities flow entering through the upstream groove tends to increase up to a maximum , after which with increasing specific load it goes on decreasing consistently.at low eccentricity in twin groove bearing the flow rate at downstream also low.[8]

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. From the experiment result and the corresponding CFD analysis, it is observed that the pressure along the axial groove located in the loaded area of the bearing and supplied with water at one end increased rapidly to a value which depends on the load applied to the bearing remained constant along the length and then fell sharply to the outlet value. This is quite different to the hypothesis that there would be a linear change from the inlet pressure to the outlet value which was used in. The implication here is that lubricant is squeezed out of both ends of the groove that is, water clearly enters the bearing in the unloaded areas and is carried around the bearing by Coquette action.[2] The friction characteristics of a journal bearing with dimpled bushing manufactured by using the machining and the chemical itching techniques is investigated. A series experimental result is presented to examine the effect on the stribeck curve of journal bearing. Load, oil type, dimple size, depth and shape varied to explore their influence on the friction characteristic. Proper dimple size, shape and depth are essential to improve the friction performance.[12]The pressure distribution obtained from the experimental work of three lobe bearing has the maximum pressure value over the plain general bearing. The value of pressure is increased by 30% for lower load and 7-10% of higher load. It can be concluded that the load carrying capacity of the lobe manufactured bearing is higher than plain bearing. [11]

II. 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.

Symbol/ Notation	**Name of Symbol[16]**	**Units**
h	Film Thickness	m
C	Specific heat capacity of lubricant	J/kg.° C
D	Journal diameter	m
F	Friction force	N
F_h	Ratio of friction loss to viscosity	m2/s
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	m
N	Journal rotational speed	rpm
P	Dimensional film pressure	Pa
Pmax	Maximum pressure	Pa
Rj	Journal radius	m
Tmax	Inlet temperature, effective temperature, maximum temperature	°C
V	Journal surface velocity	m/s
W	Dimensional load capacity	N
We, Wf	Dimensional load capacity along and perpendicular to line of centers	N

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
Λ	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/m 3
Ω	Angular velocity	rad/s

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