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Evaluation of Stiffness and Damping Coefficients of Multiple Axial Groove Water Lubricated Bearing Using Computational Fluid Dynamics

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Abstract**—**This research details a Computational Fluid Dynamics (CFD) approach to model fluid flow in a journal bearing with 8 equi-spaced semi-circular axial grooves. Water is used as the lubricant and is fed from one end of the bearing to the other, under pressure. The geometry of the bearing is modeled using a commercially available modeling software GAMBIT and the flow analysis is performed using a dedicated CFD analysis software FLUENT. The pressure distribution in the bearing clearance is obtained from FLUENT for various whirl ratios and is used to calculate the hydrodynamic force components in the radial and tangential direction of the bearing. These values along with the various whirl speeds can be used to do a regression analysis to determine the stiffness and damping coefficients. The values obtained are then compared with the stiffness and damping coefficients of a 3 Axial groove water lubricated journal bearing and those obtained from a FORTRAN code for a similar bearing.

Keywords**—**CFD, multiple axial groove, Water lubricated, Stiffness and Damping Coefficients.

I. Introduction

ENVIRONMENTAL concerns have bought about an increase in the use of water lubricated bearings. Oil and grease lubricated bearings pollute the water during oil tanker accidents, leakage and breakdown of mechanical equipment. Stringent regulations regarding pollution by oil spillage have encouraged the need to find certain alternate lubricating fluids which are less harmful to the environment. Even though water has viscosity about 30 to 40 times less than common lubricating oils (mineral oils), it can be used as an excellent coolant to remove the heat generated due to fluid friction. It can be conveniently used when the load to be supported is less. A significant amount of theoretical and experimental development is needed before the bearing can meet the requirements during service. Shelly and Ettles [1] analyzed the performance of journal bearings with oil grooves that were positioned at the maximum pressure location. They discovered that positioning the grooves at the maximum pressure will cause 30 to 70 percent reductions in the load capacity of the bearings. This results in the lowering of the film thickness between journals and bearing which would lead to journal bearing contact and increased wear. Majumdar, Pai and Hargreaves [3] studied the steady state and dynamic characteristics of 3 groove water lubricated journal bearings theoretically. The dynamic behavior in terms of stiffness and damping coefficients of film and stability were found using first order perturbation method. It has been shown that both load capacity and stability improve when smaller groove angles were used. Pai, Hargreaves and Brown [4] analyzed a 3 axial groove water lubricated bearing using Computational Fluid Dynamics. Results are presented for the circumferential and axial pressure distribution in the bearing clearance for different loads, speeds and supply pressures. Results are compared with experimentally measured pressure distributions. Vijayaraghavan and Keith [6] studied the effect of type and location of oil groove on the performance of journal bearing. They found out that, the provision of grooves should not be in the load carrying area and should be as close to the maximum film thickness which would least interfere with the hydrodynamic performance of the bearing. This would provide maximum performance. Thus it is important to note that improving one aspect of the bearing performance may have deleterious effects on other operating parameters. Therefore a complete bearing design must examine all aspects of the bearing performance [4]. The objective of this study is to use CFD approach to find the stiffness & damping coefficients of 8 equi-spaced semi-circular axial grooves and 3 axial groove water lubricated bearing & hence ascertain their stability.

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II. Nomenclature

C	diameteral clearance (m)
D	diameter of the bearing (m)
L	length of the bearing (m)
Φ	attitude angle (deg)
C_{rr}, $C_{\Phi\Phi}$	direct damping coefficient (Ns/m)
$C_{r\Phi}$, $C_{r\Phi}$	cross coupled damping coefficient (Ns/m)

F_r/Δ	bearing force along radial direction (N/m)
F_t/Δ	bearing force along tangential direction (N/m)
ω_p	whirl speed (rad/sec)
Δ	eccentricity (m)
ω	journal rotational speed (rad/sec)
K	direct stiffness coefficient (N/m)
k	cross stiffness coefficient (N/m)
C	direct damping coefficient (Ns/m)
c	cross damping coefficient (Ns/m)
$K_{r\Phi}$, $K_{\Phi r}$	direct damping coefficient (N/ m)
$K_{r\Phi}$, $K_{\Phi r}$	cross coupled stiffness coefficients (N/ m)

III. Fluid Film Modelling

The model shown in the figure 1 is the fluid film in the clearance space of the Journal bearing. The geometry of the bearing is modelled using a commercially available modelling software GAMBIT. The shaft diameter is 65mm. The length of the journal bearing is 200mm. The clearance between the journal and bearing is assigned as 0.86mm. Eight semicircular grooves of depth 3 mm and angular extent of 10˚ are placed at 45˚ diametrically to each other. The bearing shell was placed at the origin, X = 0, Y = 0, Z = 0. The rotational axis of the journal was set at eccentricity, which is X = 0.0957mm, Y = 0.414mm, Z = 0. All the data mentioned above, except the shaft diameter, angular extent, shape and the number of grooves was taken from the reference by Pai et al [4]. This type of bearing was modeled mainly to study the effect of varying the number of grooves and groove shape of the bearing on the stability of the system.

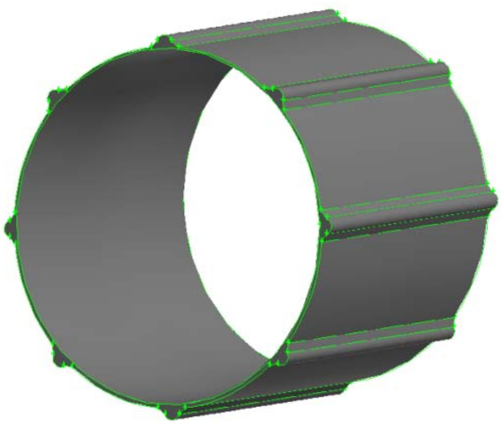

Fig. 1 Model of the clearance volume showing 8 axial groove bearing

IV. Grid Details

The figure 2 shows meshed clearance volume of 8 semicircular axial grooves bearing with 163825 nodes and 95634 hexahedral mesh elements with minimum skewness 0.6.

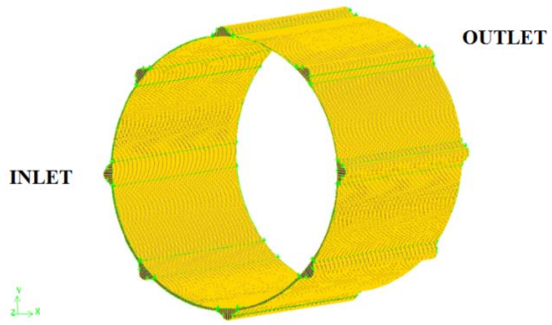

Fig. 2 Meshed clearance volume with the 8 semi-circular axial groove bearing

V. CFD Analysis

A software package FLUENT is used to obtain the pressure and velocity profiles in the bearing. The software uses four fundamental equations to solve the fluid related problems. The fluid film in the clearance space is modeled and meshed using this software package [4]. The meshed model of the bearing clearance will be subjected to appropriate boundary conditions and analyzed. The simulation is then performed and continuously monitored to determine whether a converged solution has been obtained. With this software it will be possible to visualize the flow situation in the bearing grooves as well as the land (non-grooved) area of the bearing. The CFD software enables us to visualize the flow situation in the bearing grooves as well as the land (non-grooved) area of the bearing. Water as it enters the clearance space of the bearing is subjected to velocity induced flow, as the shaft rotates and pressure induced flow, as the water is supplied under pressure from one end of the bearing to the other.

A. Boundary Conditions

Water is supplied through the inlet at a pressure of 50kPa.The inlet is at the front end of the model shown in Fig 2. The pressure is applied normal to the boundary at the inlet face. Similarly, the outlet pressure is set as 42kPa, normal to the boundary at the rear end of the model shown in the Figure 2. The bearing shell was modelled as a ‘moving wall’ with absolute motion of 0 rad/sec. 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 modeled as ‘moving wall’ with a motion relative to the adjacent cell zone at an angular speed of 108.91 rad/sec. The rotational axis origin for the journal was set at the eccentricity, which is X = 0.0957mm, Y = 0.414mm, Z = 0 and speed of 1000rpm.The rotation axis direction was set as X = 0, Y = 0, Z = −1. The water in the clearance volume was modelled as type- ‘fluid’, with rotation axis origin and direction same as that of the journal above. The rotational velocity was set at 108.91 rad/sec in the same manner as that for the journal. The translational velocity in all three coordinate directions was set equal to zero. The segregated solver was used for the solution and the flow was assumed to be laminar and steady.

The under-relaxation factors used for pressure, momentum, density and body forces are 0.3, 0.01,1 and 0.01 respectively for the solution. The discretisation used is ‘presto’ for pressure, 'quick' for momentum and 'simple' for the P-V coupling.

An interesting feature of the solution technique is that the angular velocity of the journal surface is varied in steps. The converged solution of the first step is used as the start point for the next step. The step size of the angular velocity is determined by halving the full angular velocity, which is again halved and so on. The halving process is stopped if it is determined that there is sufficient number of steps. If the step size is too small it may lead to unnecessary computations. It is necessary to arrive at a solution by varying the angular velocity in steps because of the curvature of the bearing surfaces. The solution is converged if the normalised residuals are of the order of $1x10^{-3}$ [4].

B. Stiffness and Damping Coefficients

In this paper, the stiffness and damping coefficients of multi axial groove journal bearing are evaluated using the CFD approach. The journal while rotating, executes small harmonic oscillations about its centre. Stiffness and damping coefficients gives us an idea of the stability of the bearing. It has been found that the fluid film, which supports the journal, can be interpreted as a spring and dashpot system.

Stiffness coefficients provides us with an idea of resistance offered by the lubricant (water) towards deformation, by the journal and subsequently, the damping coefficients gives us an idea of ability of water to reduce the oscillations created by the rotation of the journal at its centre.

C. Dynamic Pressure

Dynamic pressure is actually a term in the equation of energy conservation and is closely related to the kinetic energy of the fluid particle. After the solution is obtained from the FLUENT software, dynamic pressure is extracted from the software along with the face area in the radial and tangential direction at every node. The values are then exported to computing software MATLAB. The radial and tangential forces are obtained by multiplying dynamic pressure with face area in radial and tangential direction at each node respectively.

D. Solution Technique

Analysis using FLUENT software is carried out for various values of whirl speeds (ω_p), the bearing force of each case is calculated by integrating the pressure on the journal.

Assuming the effect of the inertia as negligible, the dynamic coefficients are derived from the following equations

$$F_r/\Delta = -K - c\omega_p + M\omega^2 \quad (1)$$

$$F_t/\Delta = -k - C\omega_p + m\omega^2 \quad (2)$$

The bearing shell is rotated in the opposite direction to that of the journal to extract the dynamic pressure values for different values of the whirl ratio and the speed of rotation of shaft and bearing were accordingly adjusted depending upon the whirl ratios. For this particular problem the journal speed was maintained constant at 108 rad/s and the whirl speed values of the bearing were maintained at **54**,**108**,**162** and **216** (rad/s) to obtain whirl ratios **0.5**, **1**, **1.5** and **2** respectively.

Fr/Δ and **Ft/Δ** values are obtained for various whirl ratios as **0.5, 1, 1.5,** and **2** and equation for the curve of best fit is found out using least square method. The curve of best fit (linear second order) is plotted using MATLAB computing software as illustrated in figure 7, 8, 9 and 10.

E. Validation

In order to validate the results obtained through the CFD software, 3 axial groove water lubricated bearing with 36º groove angle and L/D=1 [3] is modeled using GAMBIT and then analyzed with FLUENT Software. The solution technique adopted for the CFD analysis is similar to one described in the previous section. The values of direct stiffness (K_{rr}) and direct damping coefficients (C_{rr}) obtained are plotted against various eccentricity ratios (ε) of the bearing and are shown in figure 3 and 4.

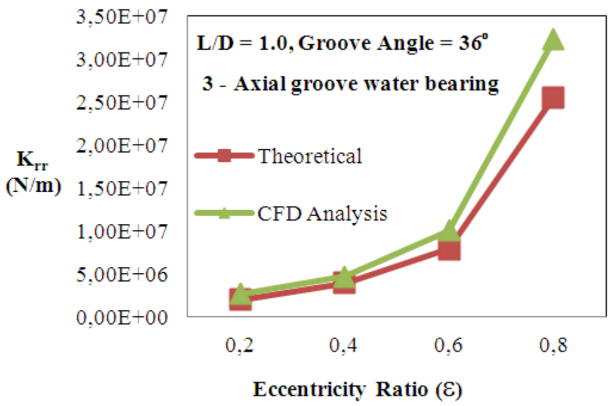

Fig. 3 Direct Stiffness coefficient (**K_{rr}**) vs Eccentricity ratio (ε)

The rotor dynamic forces predicted by Majumdar et al [3] are compared with the results of the CFD analysis. The stiffness and damping coefficients in the reference are found using first order perturbation method, which is based on the Reynolds equation, whereas the CFD analysis code uses full Navier stokes equation in providing the solution to the flow problem. The results obtained by the two approaches are therefore likely to differ. In this case it is observed that the difference is approximately 25%, which is well within the acceptable range.

VI. Results and Discussion

We know that flow in a journal bearing supply groove has an important role in determining the performance of the journal bearing, represented by its load carrying capacity and energy consumption. Maximum pressure region is observed where the clearance space of the bearing is least. This can be understood by the principle of converging and diverging wedge action.

The pressure contour obtained indicates that, the maximum pressure (red region) has slightly moved towards the outlet (rear end) side of the bearing. This is mainly because the water is supplied axially from the inlet to the outlet side of the bearing [3]. It is also observed from figure 6, that no flow takes place into the bearing at the inlet in the loaded region, because there is a positive pressure gradient. At the inlet, flow into the bearing takes place only in the unloaded region. At the outlet, flow takes place out of the bearing in the loaded region.

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The flow into and out of the bearing is maintained very much similar to that in a submerged bearing [4].

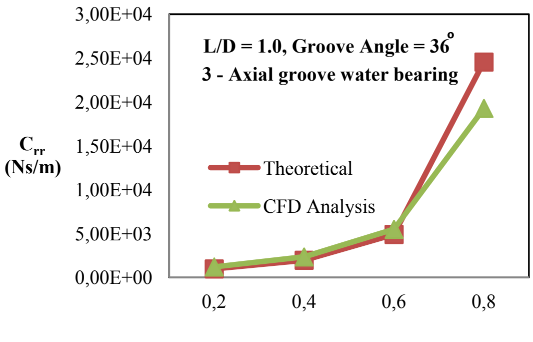

Fig. 4 Direct damping coefficient (C_{rr}) Vs Eccentricity ratio (ε)

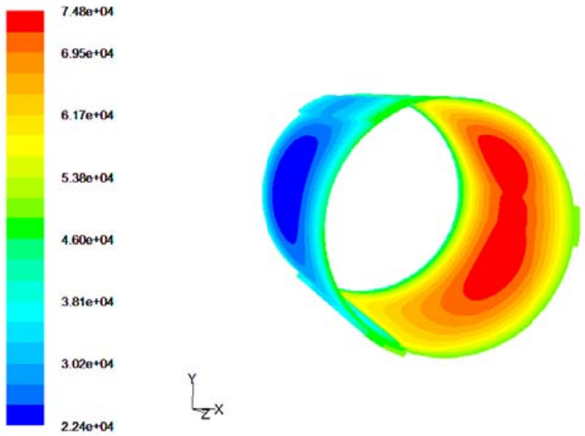

Fig. 5 Pressure contour of 3 axial Groove bearing with 36° groove angle, L/D = 1 and eccentricity ratio 0.8

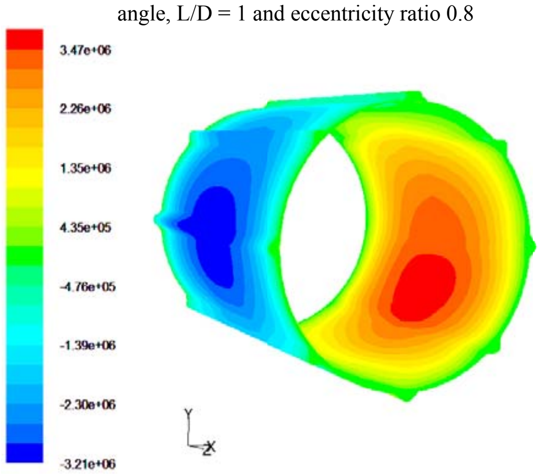

Fig. 6 Pressure contour of 8-semicircular groove bearing from fluent

Dynamic coefficients of both 3 axial groove and 8 semicircular axial groove bearing, derived from comparing equation of curve fit & equations (1) & (2) mentioned above are presented in the following table (Table I)

TABLE I
DYNAMIC CHARACTERISTICS OF BOTH THE 3 AXIAL AND 8 SEMI-CIRCULAR AXIAL GROOVE WATER LUBRICATED BEARING

Dynamic Coefficients	Number of Grooves: 3 rectangular axial groove	Number of Grooves: 8 semi-circular axial groove
Direct stiffness K_{rr} (N/m)	8.94E+05	1.86E+06
Direct damping coefficient C_{rr} (N-s/m)	- 4.54E+04	- 3.82E+04
Cross coupled stiffness $K_{r\Phi}$ (N/m)	1.80E+06	8.93E+06
Cross coupled damping coefficient $C_{r\Phi}$ (N-s/m)	-3.34E+04	-7.18E+04

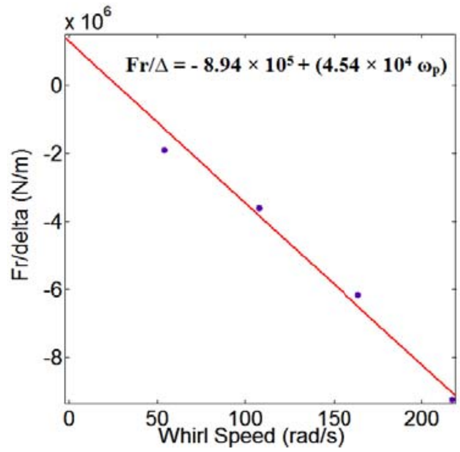

Fig. 7 F_r/Δ (radial direction) Vs. Whirl Speed (ω_p) from MATLAB of 3 groove bearing

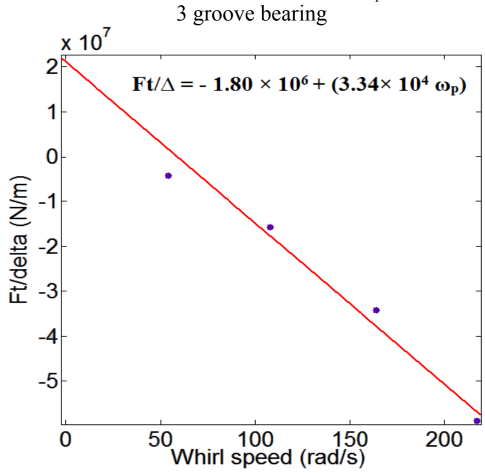

Fig. 8 F_t/Δ (tangential direction) Vs. Whirl Speed (ω_p) from MATLAB of 3 groove bearing

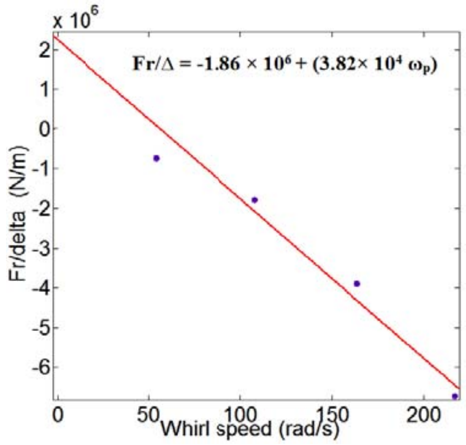

Fig. 9 F_r/Δ (radial direction) Vs. Whirl Speed (ω_p) from MATLAB of 8 groove bearing

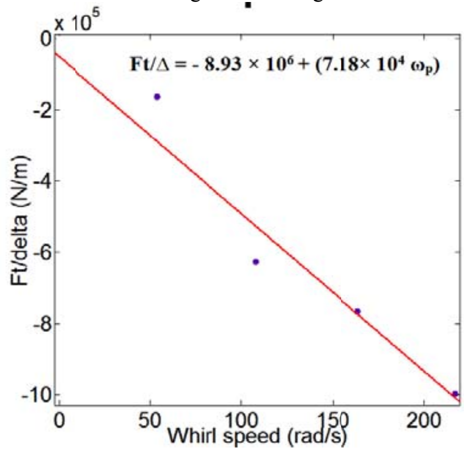

Fig. 10 F_t/Δ (tangential direction) Vs. Whirl Speed (ω_p) from MATLAB of 8 groove bearing

VII. Conclusion

The present study demonstrates that CFD approach can not only be used to determine the pressure and velocity contours but also to calculate the stiffness and damping coefficients. The coefficients obtained can be used to predict the stability of the bearing at the design stage itself.

The direct and cross stiffness values indicate the resistance offered by the fluid film to the deformation caused by the journal and the damping coefficients indicate the ability of the fluid film to reduce the oscillations created by the rotation of the journal.

It is observed that 8 semi-circular axial groove water lubricated bearing have higher value of direct stiffness coefficient, cross stiffness coefficient and cross damping coefficient than the 3axial groove water lubricated bearing (figure 11 and figure 12). We know that the stiffness and damping coefficients are responsible for the whirl instability of the bearing [3] and higher values of these coefficients indicate the stability of the whirl.

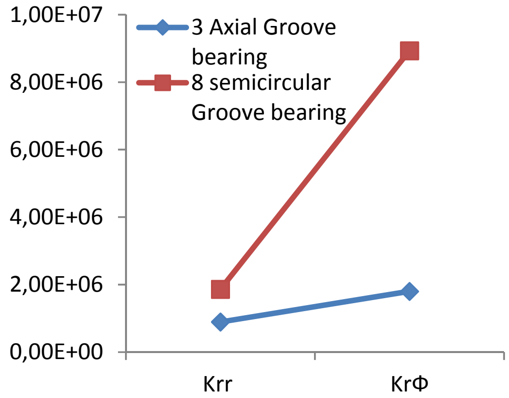

Fig. 11 Comparison of stiffness coefficient values for 3 axial groove and 8 semicircular groove bearing

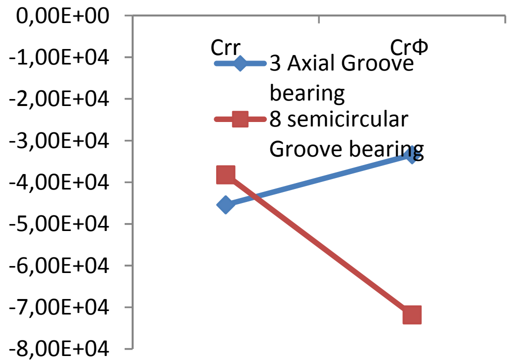

Fig. 12 Comparison of damping coefficient values for 3 axial and 8 semicircular groove bearing

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