Numerical Analysis of Three-Lobe Hydrodynamic Journal Bearing using ANSYS

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Abstract

Journal bearings are used on a variety of different rotating and reciprocating machines to support shafts that are rotating inside a bearing. In contrast to anti-friction type bearings where the primary mechanism is rolling action of contacting components, journal bearings operate by supporting the load of the shaft on a pressurized oil film. Journal bearing has a layer of lubricant (oil) separating two parts through fluid dynamic effects. Reynolds equation is used to obtain the pressure equation which can be integrated to obtain stiffness and damping coefficients. This pressure generates reaction forces on shaft and tries to keep it stable. The oil film in journal bearing provides stiffness and damping. The film of lubrication is analytically modeled in ANSYS workbench and by perturbation method the damping coefficients are obtained. Simple method to predict vibration characteristics of multi lobe journal bearing using FFT analyzer for vibration analysis is done in this study.

Keywords: stiffness coefficient, damping coefficients, Pressure profile, Hydrodynamic Journal Bearing

1. INTRODUCTION

[1] J.W.Lund, K.K.Thomsen has proposed the methodology for calculating stiffness and damping coefficient of oil- lubricated bearing. They found that the methodology is a finite difference solution of Reynold's equation, which obtained steady-state pressure distribution as well as the dynamic pressures produced by the small amplitude whirl of the journal centre. Film rupture is considered with the limit to the ruptured film zone controlled by an iterative procedure. An integration of the pressure yields the load carrying capacity, the four stiffness coefficients, and the four damping coefficients.

[2] David V. Taylor, Gregory J. Kostrzewsky, Ronald D. Flack has investigated the complete set of data for a three-lobe bearing having a relatively large preload factor of 0.75. The experiment is performed in which the Sommerfeld number range from 0.23 to 2.87 was varied by performing tests over different range of speeds and loads. In experimental test rig, the shaft is constraint to a fixed axis of rotation and the bearing is free to translate but not rotate.

[3] J. Moradi Cheqamahi, M. Nili-Ahmadabadi, S. Akbarzadeh developed the dynamic mesh method to solve the conservation equations of mass, momentum and energy. According to authors the main disadvantage of many analytical and numerical methods was their inability to analyse complex geometries. So they proposed the dynamic mesh method that shows smaller error compared to other techniques.

[4] H. N. Chandrawat, R. Sinhasan, presented a paper on elastohydrodynamic lubrication in a three-lobe journal bearing. Elastohydrodynamic (EHD) analysis that gives the appropriate results or effects of elastic deformation and pressure-sensitive viscosity on the operational characteristics of lubricated surfaces has now a days made essential for more realistic investigation. The theory of EHD lubrication is well

developed and has been extensively used in studies of point and line contact problems.

[5] Gengyuan Gao et al, investigated the relationship between eccentricity ratio and the Sommerfeld number based on hydrodynamic water-lubricated journal bearing. Considering the differences between water and oil, especially the difference in vapour pressure, the effect of cavitation on pressure distribution of water film is analysed based on three boundary conditions and a cavitation model by computational fluid dynamics (CFD). Eccentricity ratio ( ) was considered as an important factor for improving hydrodynamic performances of plain journal bearings.

[6]. Zenglin Guo, Toshio Hirano, has presented a paper in which the researcher evaluated the simulated results of hydrodynamic bearing conducted by using the CFX TASC flow. For grid generation the CFX-Build 4.4 was utilized to simulate complex flow geometry fluid-film bearing and damper design. The simulation was done for different time elapsed i.e. from some minutes to hours depends upon the intricacy of the model.

A theoretical model is proposed for the prediction of dynamic coefficients of three lobe hydrodynamic journal bearing where Reynolds equation is solved to obtain film thickness, pressure profile, reaction forces and there by leading to the equation to calculate the damping and stiffness coefficients. Analytical simulation is carried out in static structural and transient structural ANSYS simulation which is coupled with Fluent as there is fluid solid interface present. The obtained results are then substituted in the theoretically modeled equations to obtain the dynamic coefficients.

Many researchers have provided dynamic analysis of journal bearing using various methods. Here in this study an analytical method of determining dynamic coefficients of journal bearing on ANSYS work bench is proposed which is followed by experimental determination of the same parameters and their validation. The experimental method proposed is the easiest way to determine the dynamic coefficients. The analytical method used is less time consuming and very user friendly.

2. Mathematical Modeling

Referring following figure Cartesian coordinate system is introduced with the y-axis pointing vertically downwards (in the static load direction) and the x-axis horizontal. Thus at a given angular speed ω, the reaction force is a function of x, y, and $\dot{x}, \dot{y}$.

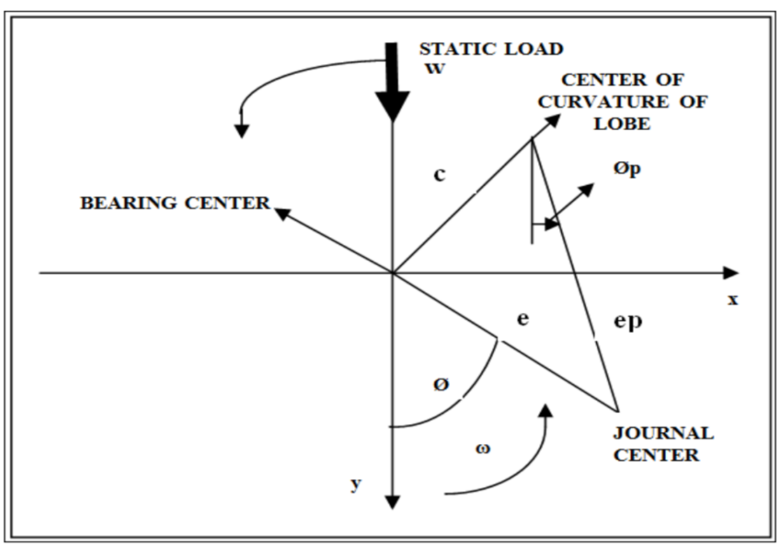

Fig. Journal and Bearing center in terms of various parameters

With components Fx and Fy in the x and y direction the force as a function of displacement and velocity may be written as:

$$F_x = F_x(x, y, \dot{x}, \dot{y}, \omega) \quad F_y = F_y(x, y, \dot{x}, \dot{y}, \omega) \qquad (1)$$

In static equilibrium condition, displacement x and y are zero, has coordinate x0 and y0. Let static eccentricity e, eccentricity ratio E and W is the static load.

$$F_{x0} = F_x(x0, y0, 0, 0, \omega) = 0$$

$$F_{y0} = F_y(x0, y0, 0, 0, \omega) = W \qquad (2)$$

The first order Taylor expansion for small amplitudes and in terms of dynamic reaction forces are,

$$\Delta F_x = F_x - F_{x0} = K_{xx}\,\Delta x + K_{xy}\,\Delta y + C_{xx}\,\Delta\dot{x} + C_{xy}\,\Delta\dot{y}$$

$$\Delta F_y = F_y - F_{y0} = K_{yx}\,\Delta x + K_{yy}\,\Delta y + C_{yx}\,\Delta\dot{x} + C_{yy}\,\Delta\dot{y} \qquad (3)$$

Where k are stiffness coefficient i.e. displacement coefficient and C's are damping coefficient i.e. velocity coefficients. These are gradient of reaction forces, evaluated in the equilibrium position:

$$K_{xx} = \left(\frac{\partial F_x}{\partial X}\right)0 \quad K_{xy} = \left(\frac{\partial F_x}{\partial Y}\right)0$$

$$C_{xx} = \left(\frac{\partial F_x}{\partial \dot{X}}\right)0 \quad C_{xy} = \left(\frac{\partial F_x}{\partial \dot{Y}}\right)0 \qquad (4)$$

Considering governing equation Reynolds equation in the bearing film,

$$\left(\frac{1}{R^2}\right)\frac{\partial}{\partial\theta}\left[\left(\frac{h^3}{12}\eta\right)\frac{\partial P}{\partial\theta}\right] + \frac{\partial}{\partial z}\left[\left(\frac{h^3}{12}\eta\right)\frac{\partial P}{\partial Z}\right] = \left(\frac{\omega}{2}\right)\left(\frac{\partial h}{\partial\theta}\right) + \frac{\partial h}{\partial t} \qquad (5)$$

And h film thickness is given by

$$h = C + e_p\cos(\theta - \phi_P) \qquad (6)$$

In static equilibrium condition eccentricity e_{op} and the altitude angle θ_{op} defines the journal position.Δx and Δy describe the amplitude under dynamic conditions of journal centre measured from static equilibrium position. Therefore film thickness become,

$$\mathrm{h} = h_0 + \Delta x\sin\theta + \Delta y\cos\theta \;; and \qquad (7)$$

$$h_0 = C + e_{op}\cos(\theta - \phi_{CP}) \qquad (8)$$

If amplitudes are assumed to be small then the first order equation is given by,

$$P = P0 + P_x\Delta x + P_y\,\Delta y + P_{\dot{x}}\Delta\dot{x} + P_{\dot{y}}\Delta\dot{y} \quad (9)$$

Where, Po is the film pressure under static equilibrium conditions.

$$\left(\frac{1}{R^2}\right)\frac{\partial}{\partial\theta}\left[\left(\frac{h^3}{12}\eta\right)\frac{\partial P}{\partial\theta}\right] + \frac{\partial}{\partial z}\left[\left(\frac{h^3}{12}\eta\right)\frac{\partial P}{\partial Z}\right]\begin{bmatrix} P0 \\ P_x \\ P_y \\ P_{\dot{x}} \\ P_{\dot{y}} \end{bmatrix} =$$

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$$\begin{bmatrix} \frac{1}{2}\omega\frac{\partial h0}{\partial\theta} \\ \frac{1}{2}\omega\, cos\theta - 3\left(sin\,\frac{\theta}{h_0}\right)\left(\frac{\partial h0}{\partial\theta}\right) - \left(\frac{h_0^3}{4\eta R^2}\right)\left(\frac{\partial P0}{\partial\theta}\right)\frac{\partial}{\partial\theta}\left(sin\,\frac{\theta}{h_0}\right) \\ -\frac{1}{2}\omega\, sin\theta - 3\left(cos\,\frac{\theta}{h_0}\right)\left(\frac{\partial h0}{\partial\theta}\right) - \left(\frac{h_0^3}{4\eta R^2}\right)\left(\frac{\partial P0}{\partial\theta}\right)\frac{\partial}{\partial\theta}\left(cos\,\frac{\theta}{h_0}\right) \\ sin\theta \\ cos\theta \end{bmatrix} \quad (10)$$

Up to this point further these equations are solved using boundary conditions. Considering the symmetry around the centre plane, only half of the bearing is considered i.e. boundary condition at z=-B/2, B is the effective length of bearing, are,

$$Z = 0;\ \frac{\partial p}{\partial z} = 0;\ \frac{\partial P0}{\partial z} = \frac{\partial P_x}{\partial z} = \frac{\partial P_y}{\partial z} = \frac{\partial P_x}{\partial z} = \frac{\partial P_y}{\partial z} = 0 \quad (11)$$

With the given boundary conditions, the five equations, equation (10) can be solved numerically (such as finite difference method) after which the pressures are integrated over the film domain to get the reaction forces from the film. By summing over all pads making up the bearing, the reaction forces are obtained as,

$$F_x = F_{x0} + K_{xx}\, x + K_{xy}\, y + C_{xx}\, \dot{x} + C_{xy}\, \dot{y}$$

$$F_y = F_{y0} + K_{yx}\, x + K_{yy}\, y + C_{yx}\, \dot{x} + C_{yy}\, \dot{y}$$

Furthermore above equations are solved by integration within the boundary conditions to give the solution in terms of Stiffness coefficient K_{xx} and Damping Coefficient C_{xx} which can be later calculated as,

$K_{xx} = \frac{\Delta F}{\Delta x}$; Dimensionless stiffness coefficients

$C_{xx} = \frac{\Delta F}{\Delta \dot{y}}$; Dimensionless damping coefficients, where C is radial clearance.

3. Ansys Simulation

Ansys 16 workbench simulation is used to carry out analytical simulation for dynamic coefficients identification, which provides a 3D platform for journal and bearing simulation. Pressure and reaction forces in bearing is obtained by fluent, and deformation in bearing is calculated by static structural module of Ansys. Optimisation technique is used to obtain the fluid solid interface using coupling of system.

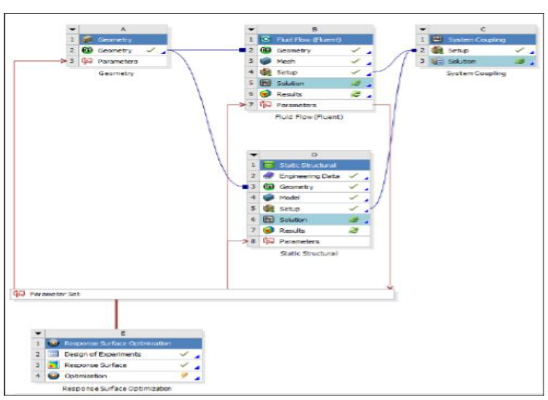

FIG1-ANSYS Fluent-Static structural Interface

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Parameter based geometry is modeled in terms of eccentricity ratio, attitude angle and minimum radial clearance. Following parameters are used to model geometry (Bearing, shaft, and oil film).

Geometry Size	Value
Bearing radius, Rb	25mm
Bearing thickness	10mm
Lobe radius, Rp	25.05mm
Minimum radial clearance, c	50µm
L/D	0.5
Preload factor	0.5

Origin X and Origin Y are used to move shaft from equilibrium position bearing which depends on the above parameters.

3.1 FLUENT SETUP- Static characteristics

Considering the instability in very thin oil-film (of order 10^{-6} m) generated between bearing and shaft, hexahedral meshing is used among the available option of triangular, tetrahedral and hexahedral meshing. Conformal meshing is generated for dynamic mesh setting. Virtual topology is used to obtain stable meshing and edge sizing, mesh sizing and mesh method is used to have control over meshing. 7 layers of mesh is generated using sweep method and multizone method. Mesh failure is avoided by recording meshing sequence of various parts of fluid film. At higher eccentricity ratio meshing of oilfilm fails and to avoid this sweep method is applied to thin film region and multizone to semi cylindrical inlet lobes. Keeping element size to 0.40mm, 211792 numbers of hexahedral cells are generated with minimum skewness of 0.0254 to maximum of 0.9 as good quality mesh. Following named sections are created with given constraints.

Region/Area on film geometry	Named Section	Constraint
Shaft outer surface	Shaft	Rotating @ rpm3000
Bearing inside surface	FSI wall bearing	Stationary
Three pressure inlets	Pressure inlet	101325pascal
Two pressure outlets	Pressure outlets	0 pascal
Overall meshing	Fluid	---------------------

Fluent setup is done with 8 processor and parallel processing. A pressure based solver is used for steady state calculations oil-vapour phase interaction is considered with cavitation model in the oil film. Primary phase – engine oil and secondary phase as oil-Vapour is defined with mass transfer mechanism. Following are the properties of oil-vapour used as input for fluent simulation.

Oil density	872 kg/m^3
Oil dynamic viscosity	0.015 kg/m^3
Oil vapour density	1.2 kg/m^3
Oil vapour viscosity	$2x10^{-5}$ kg/m^3

Cavitation properties of Zwart-Gerber-Belamri Cavitation model is considered.

UDF file is used to set outlet negative pressure to zero. Dynamic meshing is to be activated for moving shaft; smoothing mesh method with boundary distance parameter of 1.5 is set in diffusion method. Dynamic mesh zone is defined for pressure outlet as deforming and system coupling is used for force exchange between FSI wall bearing and shaft. X load and Y imbalance are taken in X and Y direction as

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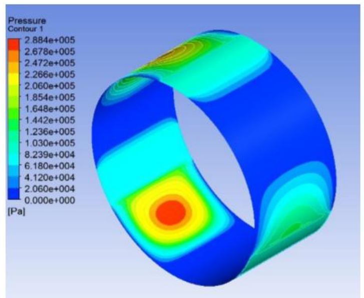
output parameters. Custom field functions, that is x-pressure force (pressure x X-face area) and y-pressure force (pressure x Y-face area) to be set, to define monitors such as mass flow rate, x force shaft and y force shaft. SIMPLEC solution method is used. A value of 0.4 and 0.5 are used for momentum and volume fraction solution control under relaxation factor. With standard method of initialization the obtained pressure profile is shown in figure.



The forces obtained on shaft as X-load and Y-imbalance are 63.1676 N and -6.99996 N respectively.

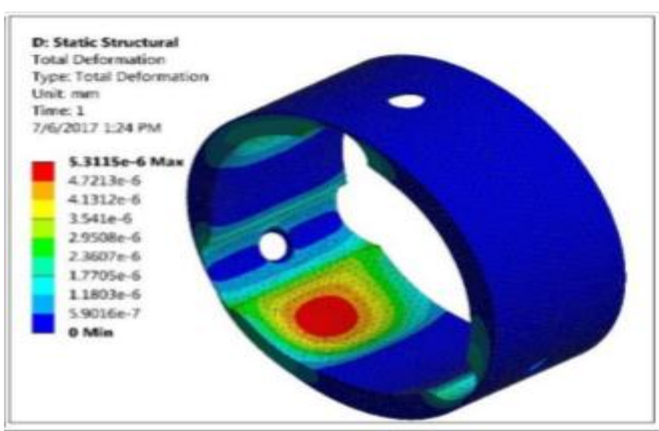
STATIC STRUCTURAL SETUP

Share geometry as shown in figure 1. Tetrahedral meshing is done on shaft and bearing with medium smoothing and course angle of span. Add support and constraints to bearing outer surface– Cylindrical support, shaft - fixed support, FSI wall bearing –fluid solid interface.

Following bearing material properties are used inn static structural simulation

PROPERTY	VALUE
Density	8490 Kg/m^3
Young's modulus	9.7 x 10^{10} Pa
Tensile ultimate strength	3.38 x 10^7 Pa
Poisson's Ratio	0.31

Pressure generated in fluid due to rotation of shaft in bearing causes deformation in bearing as shown in figure.



The obtained output from fluent and static structural is couples for 5 iterations between the bearing and journal region. Data transfer is set to interact forces from fluent to structural and structural to fluent setup.

Ten design points are selected to optimize the X load and Y imbalance taking attitude angle and eccentricity ratio as input parameter using MOGA algorithm. The optimized is selected as new input point and solved to obtain the X load and Y imbalance at 3000 RPM of shaft.

Considering small incremental displacement as 2µm in x and y direction, use small perturbation method to find stiffness coefficients. The shaft is moved to x and y direction respectively to obtain the value of Forces at that position as X load and Y imbalance at each point given in table ().

	Note	X+dx	X-dx	Y+dy	Y-dy	Eccentricity e [m]	Attitude angle [deg]	Eccentricity Ratio
Point 1	dy=0	9.0769E-06	--------	Y	Y	1.40917E-05	49.89956991	0.281834763
Point 2	dy=0	---------	5.0769E-06	Y	Y	1.19148E-05	64.77960105	0.238295501
Point 3	dx=0	X	X	1.2779E-05	--------	1.46077E-05	61.02269327	0.292154405
Point 4	dx=0	X	X	---------	8.779E-06	1.12762E-05	51.12712972	0.225524685

Following equation for unbalanced force and stiffness coefficients used to calculate the values of stiffness coefficients in magnitude.

Keeping dy=0, use Kxx=(Fxp-Fxeq)/dx; and Kyx=(Fyp-Fyeq)/dx

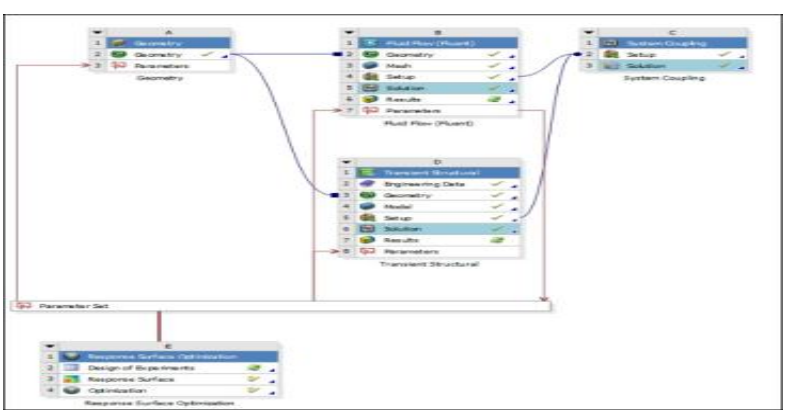
And when dx=0, use Kxy=(Fxp-Fxeq)/dy; and Kyy=(Fyp-Fyeq)/dy

Averaging the obtained value we get the answer for stiffness of oil film in magnitude as

Kxx = 2.71E6 N/m; Kyy= 3.37E6 N/m; Kyx= 9.3E6 N/m; Kxy= 8.55E6 N/m

FLUENT Setup- Dynamic characteristics

Dynamic characteristics ie damping coefficients of oil film is calculated only by considering motion of shaft. The project schematic of fluid fluent coupled with transient structural is shown in fig().



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The translation of shaft in time domain is recorded. Drag force (in x direction) and lift force(in y direction) are monitored on shaft for fixed time step. Run the simulation for 5 seconds with time step size of 0.01sec and for 500 steps.

Transient- Structural setup

Record the deformation inn the time step for 5 seconds, where number of steps are taken as 1, with current step number 1 and end time step at 5 second. Number of substeps is taken as 2 keeping time integration on. Direction deformation of bearing is recorded in x and y direction. For coupling apply four data transfer and coupling setting for 5 second duration with time step of 0.01 second for maximum of 5 second.

The values for calculation of drag force and lift force are taken from fluent setup result. Flow velocity v=0.88577002 m/sec, X-face area of bearing $Ax=0.1178m^2$, Y-face area $Ay=0.07853m^2$. the output data is available in the form of time step from 0 to 5 second (at 0.01 second of increment). The equation of damping coefficient interms of equilibrium forces and velocity is to be formed for any two time steps say t1=0.99sec and t2=1.04sec. the shaft is moved from perturbed position to equilibrium position with velocity $\dot{\Delta x}$ and $\dot{\Delta y}$.

So the incremental time step Δt=t2-t1=0.05sec and we have deformation in x and y direction at time t1 and t2. So we can obtain dx and dy as shown in table.

Time	X-deformation[m]	Y-deformation[m]
t1=0.99sec	5.0998E-3	3.0739E-2
t2=1.04sec	5.0996E-3	3.0795E-2
Δt=0.05sec	Δx=2E-7	Δy=5.6E-5

The velocity $\dot{\Delta x}$ and $\dot{\Delta y}$ are determined and the values of Kxx, Kyy, Kyx, Kxy are taken from previously calculated values. The drag and lift forces are calculated as

$Fd = Cd.(1/2)\, \rho\, v^2.\, Ax$ and $Fl = Cl.(1/2)\, \rho\, v^2.\, Ay$

Time	Drag Force Fd=Fxp[N]	Lift Force Fl=Fyp[N]
t1	43.3174	0.24723
t2	43.3109	0.23787

Calculation of damping coefficients is done by following equations:

1. If dx=0 and $\dot{\Delta x}$=0, shaft is moved in y direction
   Cxy= [(Fxp-Fxeq)-Kxy.dy]/ $\dot{\Delta y}$
   Cyy= [(Fyp-Fyeq)-Kyy.dy]/ $\dot{\Delta y}$
2. If dy=0 and $\dot{\Delta y}$=0, shaft is moved in x direction
   Cxx= [(Fxp-Fxeq)-Kxx.dy]/ $\dot{\Delta x}$
   Cyx=[(Fyp-Fyeq)-Kyx.dy]/ $\dot{\Delta x}$

Averaging the obtained value of damping coefficients of oil film in magnitude as Cxx=5.766E5 Ns/m, Cyy=1.687 Ns/m, Cxy=4.476E5 Ns/m, Cyx=5.367E5 Ns/m

4. Validation by Pressure Measurement

Taking eccentricity ration 0.25 and at 3000RPM and 45º attitude angle another reading of pressure is recorded by equating negative pressure generated in bearing to zero. This is only taken for validation of analytical reading with experimental reading. The graph shows pressure distribution at three lobe from ansys workbench.

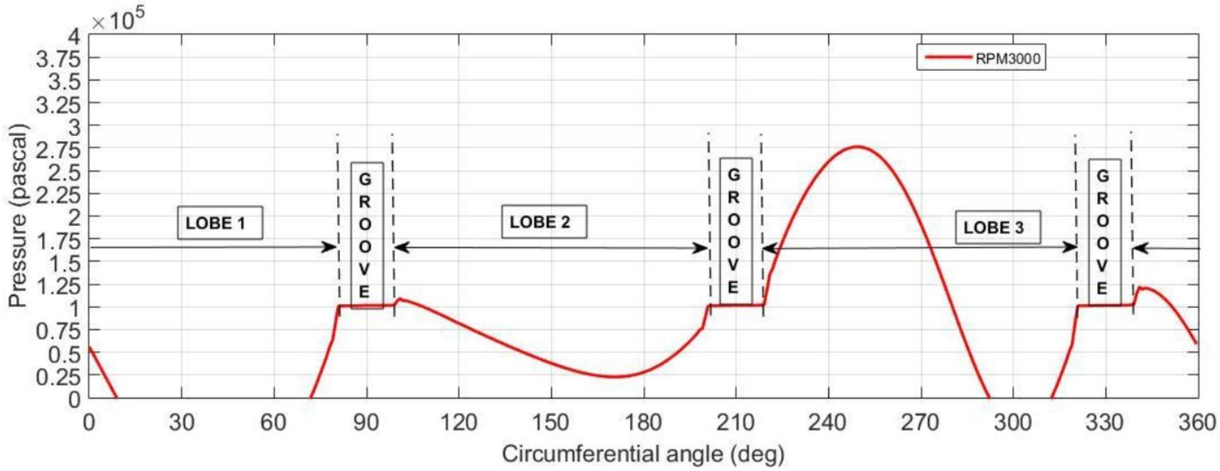

Fig. Pressure in fluid vs circumferential angle (ANSYS result)

Comparing these reading with the obtained experimental readings proves that the vales are close enough, and experimental values are less than the analytically obtained value. Closeness of these values validates the pressure generated by the experimental setup and ANSYS workbench.

Validation by Forces on shaft

David V. Taylor has done experimental measurement of highly preloaded three lobe journal bearing. By replicating the bearing in ANSYS workbench, and followed for simulations based on data presented by D Taylor, the horizontal and vertical shaft load is calculated. It is found that within 10% of accuracy. This validates the data generated by ANSYS workbench with experimental data on another experimental setup.

able- Validation by horizontal and vertical shaft load comparison

Test Case	Speed (rpm)	Sommerfeld Number	Calculated Shaft Load (N)		Taylors Shaft Load (N)		Supply Pressure (kPa)
			Horizontal	Vertical	Horizontal	Vertical	
1	900	0.231	462	43	668	15	30
2	900	0.320	447	16	471	15	31
3	900	0.570	292	25	264	14	32
4	900	2.660	67	49	54	15	33

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Validation by Stiffness Coefficient calculation

In the recent versions of ANSYS, the method of determination of stiffness coefficients is described on the basis of perturbation method. ANSYS presented the simulation of plain journal bearing for dynamic analysis. Here same bearing is modeled and solved in ANSYS workbench and the results of stiffness are compared in magnitude with the ansys obtained data. Since we need to validate the process again two points are taken into consideration for each stiffness coefficient component.

Table- Stiffness value comparison for validation

	ANSYS Workbench result [N/m] x E+7				Calculated results [N/m] x E+7			
Point	Kxx	Kyx	Kyy	Kxy	Kxx	Kyx	Kyy	Kxy
1	5.80	1.437			6.275	1.386		
2	5.52	1.437			5.030	1.622		
3			4.34	8.10			4.772	8.55
4			4.09	7.80			3.684	7.41

5. RESULT ANALYSIS

Keeping the complete focus of the study on the calculation of stiffness and damping coefficients of multilobe hydrodynamic journal bearing Ansys simulation and Experimental approach is being made. Perturbation method is used to move shaft from equilibrium position and reaction force identification lead to calculation of unbalance force in shaft in that direction.

Experimental investigation is done with the help of vibration analysis devise known as FFT analyser, although certain limitation of setup may lead to difference in values obtained experimentally. It is very difficult to predict the dynamic coefficients in journal bearing, but an analytical approach is made to do a complex study. With the mentioned bearing geometry and oil as lubricant in it, the dynamic analysis of fluid film ended to the approximate prediction of stiffness and damping values.

1. **ANSYS Workbench Resulted In following values of K and C**

Table- 1 ANSYS WB stiffness coefficient results

Kxx	Kyy	Kyx	Kxy
2.71E6 N/m	3.37E6 N/m	9.3E6 N/m	8.55E6 N/m

Table- 2 ANSYS WB damping coefficient results

Cxx	Cyy	Cyx	Cxy
5.766E5 Ns/m	1.687 Ns/m	4.476E5 Ns/m	5.367E5 Ns/m

2. Experimentally Predicted Result K and C Values

Table- 3 Experimental Stiffness coefficient results

Kxx	Kyy	Kyx	Kxy
2E6 N/m	2.8E6 N/m	7.6E6 N/m	7.9E6 N/m

Table- 4 Experimental Damping coefficient results

Cxx	Cyy	Cyx	Cxy
4.8E5 Ns/m	1.41 Ns/m	3.2E5 Ns/m	4.12E5 Ns/m

3. ANSYS Simulation Result of Pressure profile

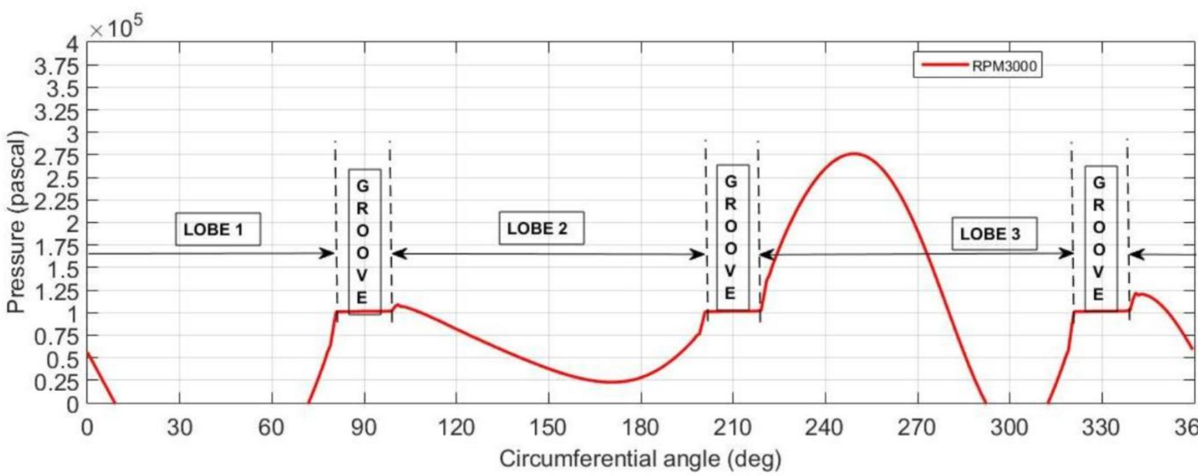

Fig. ANSYS simulation pressure profile result

Comparing these reading with the obtained experimental readings proves that the vales are close enough, and experimental values are less than the analytically obtained value. Closeness of these values gives the pressure generated by the experimental setup and ANSYS workbench.

4. Experimental Result of Pressure

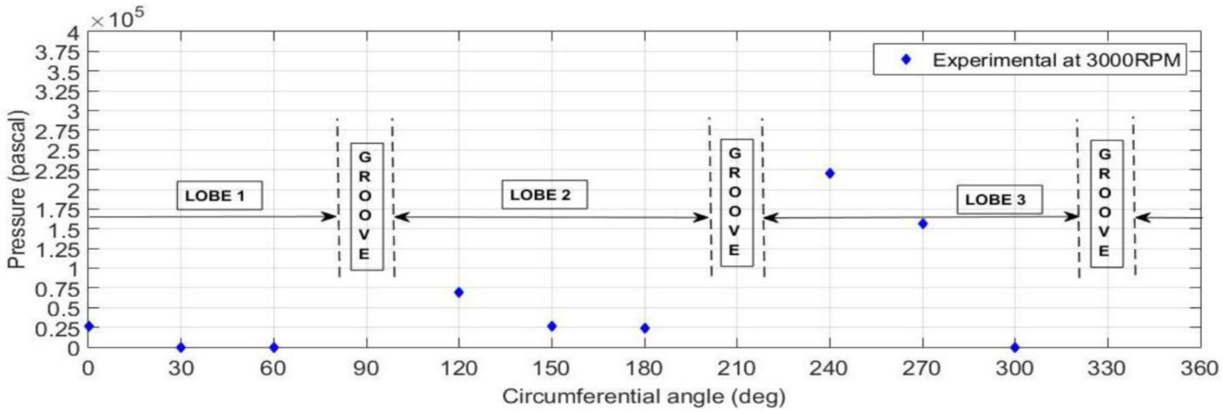

Fig.Experimentally obtained value of pressure

5. Conclusion

I. Maximum difference in the Kxx, Kyy, Cxx and Cyy obtained by simulation and experimentation are in the range of 30 percent and minimum of about 16 percent. 26 percent for Kxx and 16 percent for Kyy.

II. Maximum difference in the ansys and experimental value of Kxy, Kyx, Cxy and Cyx is 28 percent for Cxy and minimum of 7 percent for Kxy.

III. The values of stiffness and damping coefficients obtained for validation from ANSYS work bench in compared with the ansys provided result has minimum of 3 percent and maximum of 11 percent of deviation.

IV. In each lobe maximum and minimum pressure zone is generated due to eccentric centre and elliptical profile of lobe.

REFERENCE

1] J. W. Lund and K. K. Thomsen, A calculation method and data for the dynamic coefficients of oil-lubricated journal bearings, DYROBES rotor dynamics software, 1982.

2] David V. Taylor, Gregory J. Kostr zewsky, Ronald D. Flack and Lloyd E. Barrett, Measured performance of a highly preloaded Three-lobe Bearing – Part 1 static characteristics, Tribology Transactions, 25 March 2008, 38:3, 507-516.

3] J. Moradi Cheqamahi, M. Nili- Ahmadabadi, S. Akbarzadeh and M. Saghafian, Numerical Analysis of Turbocharger's Bearing using Dynamic Mesh, Journal of Applied Fluid Mechanics, 2016, Vol. 9, No. 5, pp. 2545-2557.

4] H. N. Chandrawat, R. Sinhasan, A study of Elastohydrodynamic lubrication in a Three-Lobe Journal bearing, Elsevier Sequoia, 30 August 1989, 0043- 1648/90.

5] Gengyuan Gao, Zhongwei Yin, Dan Jiang, Xiuli Zhang, Numeriacl Analysis of plain Journal Bearing under hydrodynamic lubrication by water, Elsevier, 12 March 2014, 31-38.

6] Zenglin Guo, Toshio Hirano, R. Gordon Kirk, Application of CFD Ana lysis for rotating machinery, Part 1: Hydrodynamic, Hydrostatic Bearing and Squeeze film damper, ASME, 2003.