International Journal of Machine Design and Manufacturing
Vol. 2: Issue 2
www.journalspub.com

An Overview on Theoretical Analysis of Hydrodynamic Journal Bearing Considering Thermal Effects

Shubham R. Suryawanshi[*1], *Jayant T. Pattiwar*[2]
[1]Department of Mechanical Engineering, MET's IOE, BKC, SPPU, Nashik, India
[2]NDMVP Samaj's KBT COE, SPPU, Nashik, India

ABSTRACT
Hydrodynamic journal bearings are generally used to support the rotating shafts with the application of sliding motion. It sustains the radial as well as axial load in their respective directions. Whenever it runs at high operating conditions, temperature, which effects on lubricant viscosity plays a vital role as it varies due to the bearing surface. The main objective of this paper is to analyse the performance of the plain circular and non-circular journal bearings by using design parameters. An alternative approach to thermo-hydrodynamic analysis is presented assuming that all the heat that is generated by viscous shear in the fluid film is dissipated only in the fluid (no heat conduction through the boundaries). The effect of lubricant viscosity on the parameters such as eccentricity, Sommerfeld number, load carrying capacity, pressure distribution, frictional force, coefficient of friction, power loss and temperature distribution is presented. The analysis is done for a wide range of load, speed, lubricant viscosity. The results obtained theoretically can be validated by experimental methodology as mentioned in this paper. The setup prepared can be used for all types of bearings with different geometries. The analysis is carried out for various grades of a lubricant in which an effect of lubricant viscosity on the bearing temperature is identified. Comparative study is done by taking these lubricants. The study of geometry is also presented as it plays a vital role to overcome the thermal effects as well as instability.

Keywords: hydrodynamic journal bearing, thermal effect, whirl instability, lubricant temperature

[*]***Corresponding Author***
E-mail: shubhamsuryawanshi255@gmail.com

INTRODUCTION

Hydrodynamic bearings are familiar constituents of rotating machineries. Journal bearings endow with a cylindrical bearing face on which the shaft running through the bearing lies. The studies of journal bearings have concentrated on various features of engineering. The concept of Hydrodynamic Journal Bearing can be found in many areas like study of stress, material composition, behaviour of fluid, applied thermodynamics, vibration and instrumentation. The various researchers are carrying out the research on the stability analysis of Journal bearing system as it's the crucial area of the dynamics. Though, the concept of instability created by oil whirl is not absolutely understandable, this need of clear understanding of the oil whirl and whip phenomenon hampers the valuable fault identification of the system. In the transition phase of whirl instability, it was concluded that the value of whirling speed is more than half of the value of rotating speed with the help of analysis which was carried out by considering various parameters such as whirl speed, speed of rotation, inlet pressure, eccentricity ratio, load on bearing, thickness of oil film and pressure distribution in the different areas of film. As a result, in the convergent region of oil pressure change is varying from plus to minus value while in the divergent region, it changes from minus to plus again. This continuous fluctuation in film pressure creates the instability of oil whirl. This oil whirl instability can be identified by observing the variation in oil film pressure distribution. Oil whip phenomenon is very dangerous to the rotary system as it can't form a stable wedge in the film. It causes the chances of contact between metal surfaces between the shaft and bush. This structure causes an increase in the temperature in that respective area. The structure of hydrodynamic journal bearing due to this effect was observed in as shown in Figure 1 which was used for the steam turbine in thermal power plant situated in Eklahre, Nashik. It effects on the viscosity of the lubricant as it is depends upon the temperature. This action may lead to the damage to the system so as decrease in performance of the bearing. The geometry of bearing has a crucial impact on the whirl phenomenon, so it is necessary to analyse the performance characteristics for different geometries of bearing considering the thermal effects. Some of the researchers have provided the study based on the thermal effect and is elaborated in the literature review.

Fig. 1. *An effect of metal to metal contact due to whirl phenomenon*.

Nomenclature

D = Diameter of bearing (mm)
c = Radial clearance (mm)
L = Length of Journal bearing (mm)
ϵ = Eccentricity ratio
W = Load (N)
p = Bearing pressure (MPa)
Z = Kinematic viscosity (CP)
$P_{avg.}$ = Average pressure (MPa)
η = Side leakage factor
F_f =Frictional force (N)
P_f =Power loss per unit time (Watt)
T_{max} =Maximum temperature of oil (degree Celsius)
Q_s=Side flow rate (m^3/s)
θ= Angular coordinates (degree)
Ø=Attitude angle (degree)
C = Diametrical clearance (mm)
Dj = Diameter of Journal (mm)
e = Eccentricity (mm)
S = Sommerfield Number
N = Rotational speed of journal (r.p.m.)
μ = Dynamic viscosity (Pa-s)
h_0 = Film thickness (mm)

A = Load carrying factor for journal bearing
T_f= Frictional torque (Nm)
f=Bearing frictional coefficient
T_{in} =Inlet temperature of oil (degree Celsius)
Q=Flow rate (m^3/s)
τ =Shear stress, (N/m^2)
U=Journal surface velocity (m/s)
h_m=Minimum film thickness (mm)
W_x,W_y= Bearing load carrying capacity in X and Y directions (N)

LITERATURE REVIEW

Ram et al. [1–9] analysed the hybrid journal bearing with iterative method by using finite element method. Micro polar lubricant shows better performance than Newtonian lubricant as it influences the minimum film thickness effectively. The pressure in film, bearing flow, coefficient of friction, stiffness and damping coefficient in fluid film, threshold speed were also considered for the analysis. Lin et al.[2] analysed the different bearings with multi grooves (two, four, five and six) considering the cavitation phenomenon. FSI technique was used for the iterations. As the load increases; pressure in a film, temperature in a bearing and eccentricity also increases. The cavitation effect and temperature rise reduces effectively as the number of grooves increases. Gertzos et al. [3] carried out the simulation analysis using CFD as a medium. Fluent software was used to solve the equations. Various dimensionless parameters with different L/D ratio were resulted for Bingham lubricant to carry out comparative analysis. From the study of performance, Bingham lubricant showed better results than Newtonian lubricant as pressure in a film, frictional force, and load carrying capacity increase effectively. Deligant et al. [4] have done modelling for journal bearing using CFD as a tool to study an effect of frictional losses which is used in turbocharger. Simulated results showed good agreement with earlier experimental results as power, torque increases with an increase in speed of rotation for different inlet oil temperature. Bompos et al. [5] investigated the results for various parameters by simulation in journal bearing with magneto rheological fluid as a lubricant using CFD approach. Eccentricity ratio decreased effectively from L/D ratio from 0.5 to 2, whereas an attitude angle increased as the ratio increase for various Sommerfeld numbers. Chasalevris et al. [6] proposed a technique to solve Reynold's equation by an analytical method. A comparative analysis with various numerical methods with consideration of stiffness and damping coefficient were presented. An analytical method presented in this paper can give an exact solution to solve a Reynold's equation. Brito et al. [7] compared the performance of journal bearing with one and two groove experimentally. Flow rate and temperature rise reduces effectively for increase in load and angle respectively as the number of grooves increases. A study was carried out at upstream and downstream groove locations also. Papadopoulos et al. [8] identified the clearance in the system with the help of finite element technique. The matrix method was recommended to find out the stiffness and damping coefficients. It was found that both the coefficients were reduced as the rotational speed increased for various worn effects in bearing. Prasad et al. [10] solved the governing equations by mathematical modelling to find the pressure distribution, temperature effect at various angles. For non-Newtonian lubricant, the pressure and temperature effect showed the change in distribution compared to Newtonian lubricant. Montazeri [11] concentrated their study based on Ferro fluid as a lubricant using numerical and CFD technique. An effect of magnetic field on the performance of bearing system was investigated and attitude angle, load carrying capacity increased effectively whereas coefficient of friction decreased significantly. Ouadoud et al. [12] solved a modified Reynold's equation considering micro polar lubricant in respect to cavitation problem. Equations solved by numerical technique using alternating direction implicit (ADI) method. With the help of micro polar lubricant; load carrying capacity and temperature increase while frictional coefficient and side leakage flow reduces. Panday et al. [13] focussed their study based on elastohydrodynamic analysis considering the thermal effects, using CFS-FSI approach. As the speed increases, pressure and displacement in X direction increases whereas displacement in Y direction decreases. When an eccentricity increases, all the above mentioned parameters increase in a proportional manner. Boubendir et al. [14] worked on thin lubrication system in a journal bearing by solving the equations using CFD technique. Analysis for various L/D ratios was carried. As the ratio increases from 0.25 to 2, the pressure increase in a linear manner while wall shear stress and turbulent viscosity decreases apparently. Nicodemus et al. [15] carried their research work on hybrid journal bearing with micro polar lubricant. The study on influence of lubricant on stiffness coefficient, film damping coefficient with various recess shapes was carried in this paper. Rahmatabadi et al. [16] also focussed their study using micro polar lubricant. But researchers solved the basic fundamental equations using generalized differential quadrature (GDQ) method. Performance analysis was carried out for both circular as well as non-circular bearing and concluded that this method can be used as an effective tool rather than other techniques. Garg et al. [17] done thermohydrostatic analysis of hybrid journal bearing and they observed that temperature distribution values are in very good agreement with previously published results. The study of non-Newtonian behaviour of lubricant in relation with stiffness and damping coefficients were also presented. Bhaskar et al. [18] concentrated their focus on surface roughness phenomenon in the system. It was concluded that, as the surface roughness value increases the load carrying capacity increases by 52.7% in a sudden manner for various L/D values. Various contour plots were proposed for pressure curve with the help of ANSYS software. Mongkolwongrojn et al. [19] has also taken non-Newtonian lubricant for their research. The circular journal bearing was analysed for various parameters with respect to surface roughness phenomenon. In this work; stiffness and damping coefficients, whirl ratio with roughness amplitude were investigated for different direction with different material. Nair et al.[20] carried out comparative analysis for Newtonian and micro polar lubricants used in the system. They found that, at maximum eccentricity ratio the stability of a system disturbed for both the lubricants. Also oil whirl frequency decreased with an addition of some additives. Sharma et al. [21] presented a theoretical study for hole entry hybrid journal bearing using elastohydrodynamic effects. They summarized that pressure distribution in film, minimum film thickness, fluid flow, stiffness and damping coefficients increase as the values of deformation coefficient increases from 0 to 5. Nassab et al. [22] simulated the results with the help of CFD technique by considering the compressibility of lubricant. It was found that as the clearance ratio increases compressibility effect decreases apparently. Pressure distribution curve plotted for c/r ratio. Mishra et al. [23] studied the different contour plots for temperature in a non-circular (lobed) bearing. From the results it can be summarized that due to non-circular geometry of bearing the temperature reduces and cooling effect also produced. But at the same time pressure distribution hampers as in some extent it goes down. Kuznetsov et al. [24] presented an experimental outcome for the system considering liner deformation using different lining materials. PTFE material was recommended as it provides rise in load carrying capacity, oil film temperature, power loss with various combination of additives. Brito et al. [25] explained how performance is affected by the feed rate of lubricant in the journal bearing system. They have considered the bearings with one and two grooves for the analysis. With wide range of speed and load, feeding conditions were analysed. Twin axial groove bearing showed better results than single groove bearing. Ostayen et al. [26] studied the geometry of lemon bore bearing to carry the thermal analysis. Various parameters such as load capacity, pressure in film, power loss, and oil flow showed better results than plain circular bearing. Mathematical model was developed to find out these parameters.

Wang et al. [27] considered couple stress fluid as a lubricant to carry out thermohydrodynamic analysis. It was investigated that by using said lubricant, the load carrying capacity and frictional force increased whereas coefficient of friction reduces. It was been observed that side leakage flow remains constant with same lubricant. Suryawanshi et al. [28, 29] proposed the theoretical as well as experimental analysis of plain circular bearing considering the effect of whirl instability with the help of stiffness estimation and whirl frequency. [30]

MATHEMATICAL MODELING

A lubrication theory initially was proposed by Navier, Stokes and they have given the solution to the basic equations. Later Reynold extended the work and derived an equation known as Reynold's equation which is popularly used in most of the hydrodynamic theories.

The key parameters considered to achieve an equation are viscosity, density and film thickness of a lubricant. However many researchers proposed the different approaches mentioned in literature to get an accurate solution of the equation (Figures 2, 3).

Pressure Distribution in an Oil Film

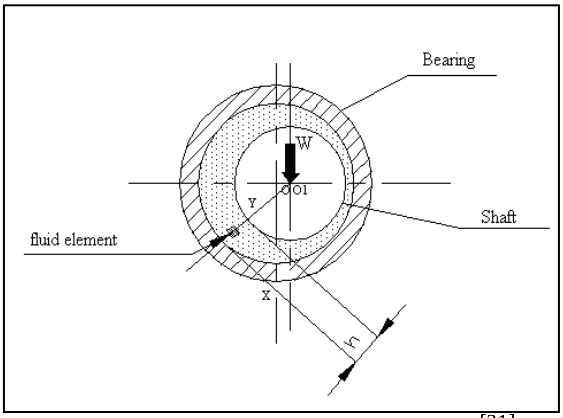


Fig. 2. *Fluid element in XY plane.*[31]

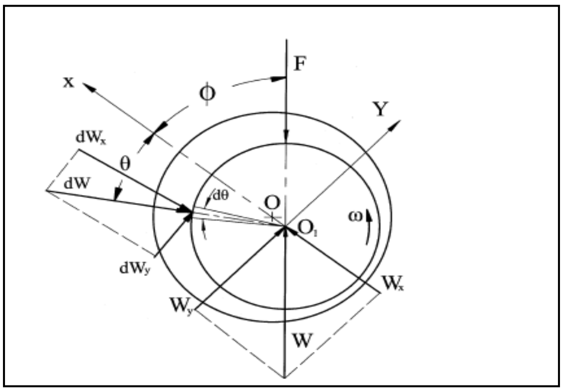


Fig. 3. *Hydrodynamic force bearing components.*[29]

Reynolds's equation in 2 dimensional form is derived as,

$$\frac{U}{2}\frac{\partial h}{\partial x}-\frac{1}{12\mu}\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)-\frac{1}{12\mu}\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=0 \quad (1)$$

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\mu U\frac{\partial h}{\partial x} \quad (2)$$

Now, if it is assumed that the bearing is infinitely long in the axial direction, implying no variation of pressure in the z-direction, the ($\partial p/\partial z$) term in the two dimensions. Reynolds equation can be dropped. The governing equation then will be,

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=6\mu U\frac{\partial h}{\partial x} \tag{3}$$

By using Reynold's boundary conditions, the above equation can be obtained as,

$$h_m=\frac{2C(1-\varepsilon^2)}{(2+\varepsilon^2)} \tag{4}$$

Substituting h_m from the equation, also reverting to θ co-ordinate from Y, the pressure distribution;

$$p=\frac{6\mu UR\varepsilon}{C^2}\frac{(2+\varepsilon\cos\theta)\sin\theta}{(2+\varepsilon^2)(1+\varepsilon\cos\theta)^2} \tag{5}$$

Load Carrying Capacity for Hydrodynamic Journal bearing

Figure 3 shows the load capacity, W, of a journal bearing and its two components, W_x and W_y. By solving the equation and applying integration by parts, the component will become as $W_x = 0$; therefore, for the Sommerfeld conditions, W_y is equal to the total load capacity, $W = W_y$:

$$W=\frac{12\pi\mu UR^2L}{C^2}\cdot\frac{\in}{\left(2+\in^2\right)\left(1-\in^2\right)^{1/2}} \tag{6}$$

Friction in a Journal Bearing

The bearing friction force, F_f, is the viscous resistance force to the rotation of the journal due to high shear rates in the fluid film.

$$F_f=\frac{T_f}{R} \tag{7}$$

Substituting the solution of the integrals, it results in the following expression for the friction force,

$$F_f=\frac{\mu URL}{C}\frac{4\pi\left(1+2\in^2\right)}{\left(2+\in^2\right)\left(1-\in^2\right)^{1/2}} \tag{8}$$

Let us recall that the bearing friction coefficient, *f* is defined as

$$f=\frac{F_f}{W} \tag{9}$$

Substitution of the values of the friction force and load in Eq. (9), it results in a relatively simple expression for the coefficient of friction of a long hydrodynamic journal bearing:

$$f=\frac{C}{R}\frac{1+2\in^2}{3\in} \tag{10}$$

Power Loss on Viscous Friction

The energy loss, per unit of time (power loss) P_f, is determined from the friction torque, or friction force, by the following equations:

$$P_f=T_f\omega=F_fU \tag{11}$$

where ω (rad/s) is the angular velocity of the journal. Substituting Eq. (8) into Eq. (11) yields,

$$P_f=\frac{\mu U^2RL}{C}\frac{4\pi\left(1+2\in^2\right)}{\left(2+\in^2\right)\left(1-\in^2\right)^{1/2}} \tag{12}$$

The friction energy losses are dissipated in the lubricant as heat. Knowledge of the amount of friction energy that is dissipated in the bearing is very important for ensuring that the lubricant does not overheat.

Temperature Rise in a Fluid Film

For design purposes it is sufficient to estimate the temperature rise of the fluid ΔT, from the point of entry into the bearing clearance (at temperature T_{in}) to the point of discharge from the bearing (at temperature T_{max}). This estimation is based on the simplified assumptions that it is possible to neglect the heat conduction through the bearing material in comparison to the heat removed by the continuous replacement of fluid. In fact, the heat conduction reduces the temperature rise; therefore, this assumption results in a design that is on the safe side, because the estimated temperature rise is somewhat higher than in the actual bearing.

The following equation is used to compute the temperature rise of oil in a journal bearing:

$$\Delta T = \frac{8.3P\left(\frac{fR}{C}\right)}{10^{6}\left(\frac{Q}{nRCL}\right)\left(1-0.5\frac{Q_s}{Q}\right)} \quad (13)$$

THEORETICAL ANALYSIS

The change in design procedure for an elliptical journal bearing as far as geometry is as follows (Figure 4).

Oil film thickness can be calculated as [23]

$$h = C_m[1+E_m+\varepsilon_1\cos(\theta+\phi-\phi_1)] \quad \text{for } 0^\circ \le \theta \le 180^\circ \quad (14)$$

$$h = C_m[1+E_m+\varepsilon_2\cos(\theta+\phi-\phi_2)] \quad \text{for } 180^\circ \le \theta \le 360^\circ \quad (15)$$

where

Eccentricity at upper lobe, $\in_1 = [E_m^{\,2} + \in^2 - 2E_m . \in . \cos(\phi)]^{1/2}$ (16)

Eccentricity at lower lobe, $\in_2 = [E_m^{\,2} + \in^2 + 2E_m . \in . \cos(\phi)]^{1/2}$ (17)

$$\phi_1 = \pi - \tan^{-1}\left[\frac{\phi}{E_m - \in . \cos\phi}\right] \quad (18)$$

$$\phi_2 = \tan^{-1}\left[\frac{\phi}{E_m + \in . \cos\phi}\right] \quad (19)$$

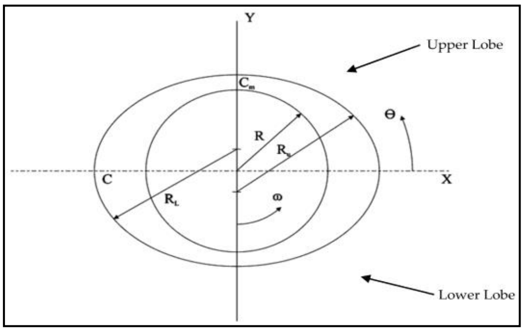


Fig. 4. *Geometry of elliptical journal bearing* [23]

Elliptical ratio,

$$E_m = \frac{C_h - C_m}{C_m}$$

where C_h is the horizontal clearance for elliptical journal bearing, C_m the minimum clearance when journal centre is coincident with geometric centre of bearing, and Ø is the attitude angle.

Operating Parameters for Analysis

(i) Diameter of Journal, D = 50 mm
(ii) Length of bearing, L = 50 mm
(iii) L/D Ratio = 1
(iv) Clearance, C = 0.05 mm
(v) Type of Lubricant = MOBIL DTE 24
(vi) Viscosity of Lubricant, μ = 28.1756×10^{-9} MPa-sec
(vii) Speed of Journal, N = 500, 750, 1000 r.p.m.
(viii) Load on Journal bearing, W = 1 kN
(ix) Clearance Ratio, C/R = 0.002
(x) Preload ratio for elliptical bearing, m = 0.5

Performance Analysis for MOBIL DTE 24 (For W=1000 N, N= 500, 750, 1000 r.p.m.)

Bearing Pressure

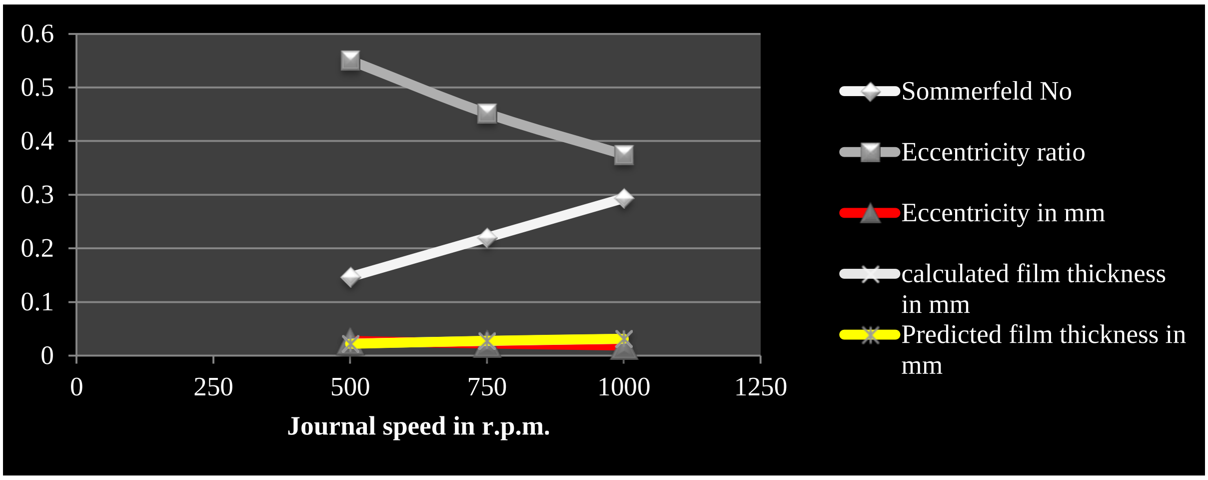
$$p = \frac{W}{2RL} = 0.4 \text{ N/mm}^2$$

Dependent Variables

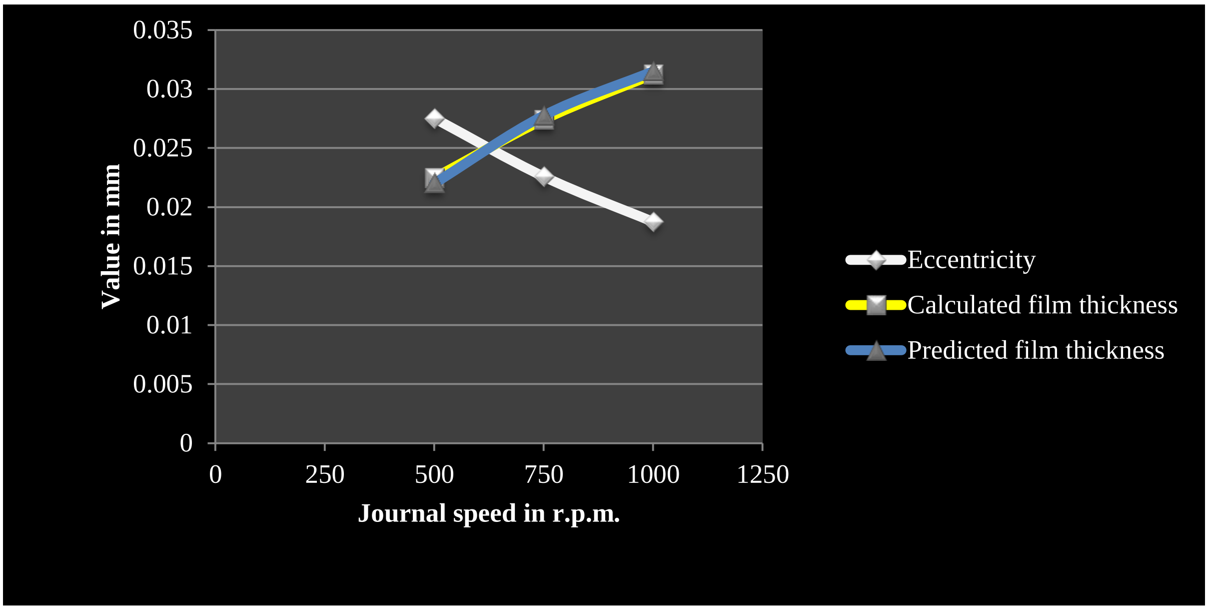
The various design variables are identified as mentioned in Table 1. It can be observed from the nature of graph obtained in Graphs 1, 2 that calculated film thickness value is in close agreement with predicted value.

Table 1. *Values of dependent variables for MOBIL DTE 24 at W=1000 N.*

Mobil DTE 24	**S**	**ε**	**e (mm)**	$h_{0(Calculated)}$ **(mm)**	$h_{0(Predicted)}$ **(mm)**
N=500 rpm	0.1467	0.5506	0.0275	0.0225	0.022
N=750 rpm	0.2201	0.4517	0.0226	0.0274	0.0278
N=1000 rpm	0.2934	0.375	0.0188	0.0313	0.0315



Graph 1. *Design variables for MOBIL DTE 24 at various speeds.*



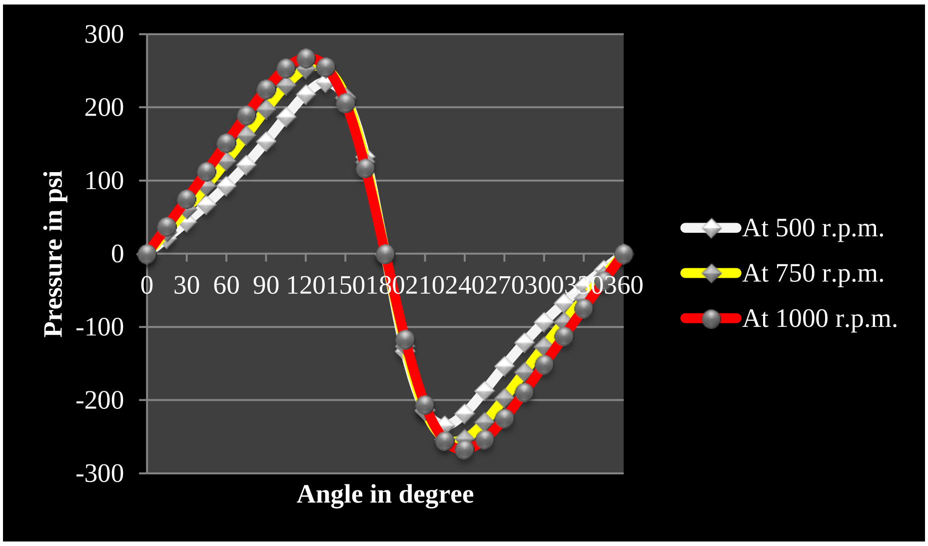
Graph 2. *Detailed view of eccentricity and film thickness for MOBIL DTE 24.*

Pressure Distribution in Oil Film

Table 2 shows pressure distribution values and Graph 3 shows the theoretical pressure distribution curve found by Reynold's equation at various speeds and it shows similar pattern for all speeds.

Table 2. *Pressure distribution values for MOBIL DTE 24 at various speeds.*

	N=500 rpm				N=750 rpm				N=1000 rpm			
θ	P (MPa)	P (KPa)	P (PSI)	P (Bar)	P (MPa)	P (KPa)	P (PSI)	P (Bar)	P (MPa)	P (KPa)	P (PSI)	P (Bar)
0	0	0	0	0	0	0	0	0	0	0	0	0
15	0.1477	147.705	21.423	1.47705	0.20789	207.89	30.152	2.0789	0.2554	255.399	37.0427	2.554
30	0.30033	300.326	43.5589	3.00326	0.42016	420.159	60.939	4.20159	0.51342	513.418	74.4656	5.1342
45	0.46295	462.954	67.1463	4.62954	0.64078	640.778	92.938	6.40778	0.77569	775.689	112.505	7.7569
60	0.64082	640.816	92.9433	6.40816	0.87247	872.473	126.54	8.72473	1.04138	1041.38	151.041	10.414
75	0.83857	838.567	121.625	8.38567	1.11484	1114.84	161.69	11.1484	1.30465	1304.65	189.225	13.046
90	1.05784	1057.84	153.428	10.5784	1.36029	1360.29	197.29	13.6029	1.55034	1550.34	224.86	15.503
105	1.29062	1290.62	187.19	12.9062	1.58637	1586.37	230.09	15.8637	1.74762	1747.62	253.473	17.476
120	1.50424	1504.24	218.173	15.0424	1.7437	1743.7	252.9	17.437	1.84314	1843.14	267.328	18.431
135	1.61536	1615.36	234.291	16.1536	1.74488	1744.88	253.08	17.4488	1.76101	1761.01	255.415	17.61
150	1.47172	1471.72	213.457	14.7172	1.47606	1476.06	214.09	14.7606	1.42406	1424.06	206.544	14.241
165	0.917	916.997	133	9.16997	0.86629	866.293	125.65	8.66293	0.80781	807.811	117.164	8.0781
180	4.7E-16	4.7E-13	6.7E-14	4.7E-15	4.3E-16	4.3E-13	6E-14	4.3E-15	4E-16	4E-13	5.7E-14	4E-15
195	-0.917	-917	-133	-9.17	-0.8663	-866.29	-125.65	-8.6629	-0.8078	-807.811	-117.16	-8.078
210	-1.47172	-1471.7	-213.457	-14.717	-1.4761	-1476.1	-214.09	-14.761	-1.4241	-1424.06	-206.54	-14.24
225	-1.61536	-1615.4	-234.291	-16.154	-1.7449	-1744.9	-253.08	-17.449	-1.761	-1761.01	-255.41	-17.61
240	-1.50424	-1504.2	-218.173	-15.042	-1.7437	-1743.7	-252.9	-17.437	-1.8431	-1843.14	-267.33	-18.43
255	-1.29062	-1290.6	-187.19	-12.906	-1.5864	-1586.4	-230.09	-15.864	-1.7476	-1747.62	-253.47	-17.48
270	-1.05784	-1057.8	-153.428	-10.578	-1.3603	-1360.3	-197.29	-13.603	-1.5503	-1550.34	-224.86	-15.5
285	-0.83857	-838.57	-121.625	-8.3857	-1.1148	-1114.8	-161.69	-11.148	-1.3046	-1304.65	-189.22	-13.05
300	-0.64082	-640.82	-92.9433	-6.4082	-0.8725	-872.47	-126.54	-8.7247	-1.0414	-1041.38	-151.04	-10.41
315	-0.46295	-462.95	-67.1463	-4.6295	-0.6408	-640.78	-92.938	-6.4078	-0.7757	-775.689	-112.51	-7.757
330	-0.30033	-300.33	-43.5589	-3.0033	-0.4202	-420.16	-60.939	-4.2016	-0.5134	-513.418	-74.466	-5.134
345	-0.1477	-147.7	-21.423	-1.477	-0.2079	-207.89	-30.152	-2.0789	-0.2554	-255.399	-37.043	-2.554
360	-1.4E-16	-1E-13	-2E-14	-1E-15	-2E-16	-2E-13	-3E-14	-2E-15	-2E-16	-2.4E-13	-3E-14	-2E-15



Graph 3. *Pressure distribution curve for MOBIL DTE 24 at various speeds.*

Evaluation of Performance Parameters

Table 3 shows the dimensionless parameters for Sommerfeld number.

Table 3. *Dimensionless parameters for characteristic number (MOBIL DTE 24).*

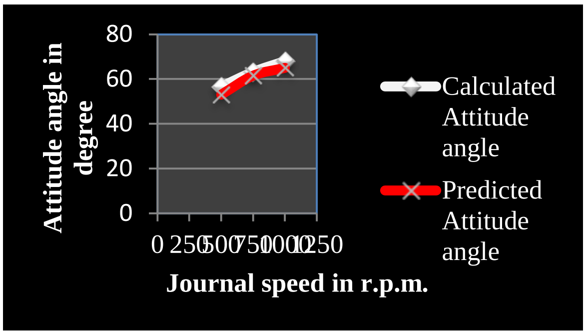
	h0/C	ϕ	P/Pmax	Q/RCLN	Qs/Q	ρCΔT/P
S= 0.1467	0.44	53	0.44	4.25	0.64	20.8
S=0.2201	0.555	61.5	0.475	4.08	0.545	28.9
S=0.2934	0.63	65	0.49	3.95	0.475	33.4

Table 4 shows the values of performance parameters for selected three speeds.

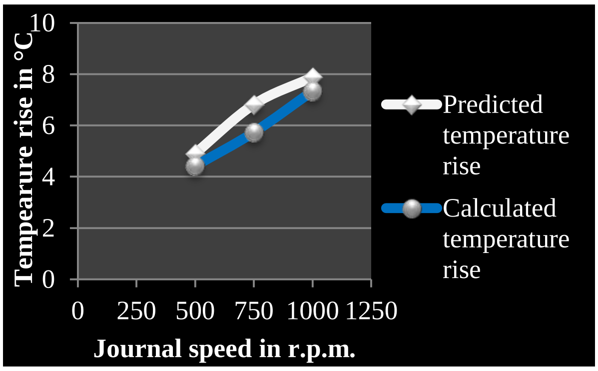
Table 4. *Performance parameters for MOBIL DTE 24 at various speeds.*

	W (N)	Ff (W)	f	Pf (W)	Pmax (Mpa)	T (Nm)	Q (lpm)	Qs (lpm)	ΔT(pre) (°C)	ΔT(cal) (°C)	ϕ
N=500 rpm	4971.3	9.6689	0.0019	12650.1	0.9091	0.0486	0.1328	0.085	4.907	4.418	56.59
N=750 rpm	5981.4	12.43	0.0021	24394.6	0.8421	0.052	0.1913	0.104	6.818	5.733	63.15
N=1000rpm	6560.8	14.944	0.0023	39103.5	0.8163	0.0569	0.2469	0.117	7.88	7.342	67.98

Graphs 4 and 5 show the comparative analysis of calculated temperature rise in a fluid film and the position of minimum film thickness or attitude angle with predicted values as per Table 3. It was found that, values are in good agreement with each other.



Graph 4. *Comparison of attitude angle for MOBIL DTE 24.*



Graph 5. *Comparison of temperature rise in a fluid film for MOBIL DTE 24.*

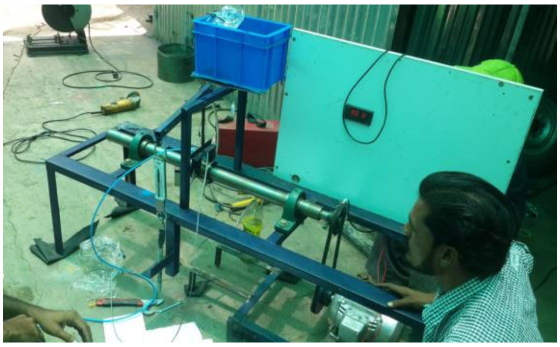


Fig. 5. *Experimental setup.*

EXPERIMENTAL METHODOLOGY

The apparatus for the experimentation is illustrated in Figure 5. It basically consists of a frame, bearing unit, loading, drive, lubrication, control and measuring system. It is having a base frame on which all components are mounted. It consists of bronze bearing mounted freely on a mild steel shaft which is considered as journal. This journal is fixed between two pedestal bearings and connected to motor shaft through the coupling. The speed of the A.C. motor shaft is converted into journal speed using v-belt drive through 1:1.5 ratios between pulleys. Speed of journal is varying using variable frequency drive.

The speed of journal is measured using proximity sensor and is displayed on speed indicator. The journal bearing has tapping at equal angles of 30° around its circumference to analyse pressure and temperature values through the sensors. The sensors used to measure for pressure and temperature are of SPD & k-type respectively. Oil film pressure and temperature are monitored on display through data acquisition card. Data acquisition card is connected to computer to observe the nature of parameters using Lab-View software. Reservoir is provided to store the lubricant which is further passed into the bearing through a pump. From the reservoir, oil enters the bearing for hydrodynamic phenomenon. One hole is provided for drain out purpose.

Different load can be applied on a journal bearing through spring loaded action. Dashboard is provided to monitor the values of speed and temperature. There are two types of the bearing construction solid and lined bushing. The solid bushing is made either by casting or machining from bar. They finished by grinding and reaming operations. A typical example of this type of bushing is Bronze bearings. A lined bushing consists of a steel backing with a thin lining of bearing material like Babbitt, brass. It is usually split into two halves and is provides with locking rug, which prevents the axial and radial displacement of bearing with respect to housing. [32, 33]

CONCLUSION

In this paper, theoretical analysis for plain journal bearing is carried out for grade of lubricants MOBIL. The lubricant category is selected as per the properties of lubricant are concern which is generally used in heavy rotating industries. Various dimensionless parameters and performance parameters are analysed and the nature for the values is in a good agreement with standard/predicted values. It is found that, lubricant viscosity plays a vital role on the performance of journal bearing. Selection of bearing, journal is done as per design process. Fabrication of experimental set up is in process which will be ready in all aspects in coming time to carry out trials.

ACKNOWLEDGEMENT

The author would like to express his sincere gratitude to his parents, family members, Dr. J. T. Pattiwar, Dr. V. P. Wani, colleagues, friends for their kind cooperation, motivation and support. Author is also thankful to Faculty of Engineering, Savitribai Phule Pune University.

REFERENCES

[1] Nathi R., Sharma S.C. Analysis of orifice compensated non-recessed hole-entry hybrid journal bearing operating with micro polar lubricants, *Tribol Int.* 2012; 52: 132–43p.
[2] Lin Q., Wei Z., Wang N., et al. Analysis on the lubrication performances of journal bearing system using computational fluid dynamics and fluid–structure interaction considering thermal influence and cavitation, *Tribol Int.* 2013; 64: 8–15p.
[3] Gertzos K.P., Nikolakopoulos P.G., Papadopoulos C.A. CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant, *Tribol Int.* 2008; 41: 1190–204p.
[4] Deligant M., Podevin P., Descombes G. CFD model for turbocharger journal bearing performances, *Appl Therm Eng.* 2011; 31: 811–9p.
[5] Bompos Dimitrios A., Nikolakopoulos Pantelis G. CFD simulation of magneto rheological fluid journal bearings, *Simul Model Pract Theory*. 2012; 19: 1035–60p.
[6] Chasalevris A., Sfyris D. Evaluation of the finite journal bearing characteristics, using the exact analytical solution of the Reynolds equation, *Tribol Int.* 2013; 57: 216–34p.

[7] Brito F.P., Miranda A.S., Claro J.C.P., et al. Experimental comparison of the performance of a journal bearing with a single and a twin axial groove configuration, *Tribol Int.* 2012; 54: 1–8p.
[8] Papadopoulos Chris A., Nikolakopoulos Pantelis G., Gounaris George D. Identification of clearances and stability analysis for a rotor-journal bearing system, *Mech Machine Theory*. 2008; 43: 411–426p.
[9] Sharma S.C., Nathi R. Influence of micropolar lubricants on the performance of slot-entry hybrid journal bearing, *Tribol Int.* 2011; 44: 1852–63p.
[10] Prasad D., Panda S.S., Subrahmanyam S.V. Non-Newtonian squeeze film lubrication of journal bearing with temperature effect, *Int J Eng Sci Adv Technol.* 2012; 2: 438–44p.
[11] Montazeri H. Numerical analysis of hydrodynamic journal bearings lubricated with ferrofluid, *Eng Tribol*. 2008; 222: 51–9p.
[12] Wanga X.-L., Zhu K.-Q. Numerical simulation CFD, FSI of a hydrodynamic journal bearing, *Tribol Int.* 2006; 39: 227–37p.
[13] Ouadoud A., Mouchtachi A., Boutammachte N. Numerical unsteady analysis of thin film lubricated journal bearing, *J Adv Res Mech Eng.* 2011; 2: 33–8p.
[14] Panday K.M., Choudhury P.L., Kumar N.P. Numerical study of the thermo-hydrodynamic lubrication phenomena in porous journal bearings, *Int J Eng Technol*. 2012; 4: 185–91p.
[15] Nicodemus R., Sharma S.C. Orifice compensated multirecess hydrostatic/hybrid journal bearing system of various geometric shapes of recess operating with micropolar lubricant, *Tribol Int.* 2011; 44: 284–96p.
[16] Rahmatabadi A.D., Mehrjardi M.Z., Fazel M.R. Performance analysis of micropolar lubricated journal bearings using GDQ method, *Tribol Int.* 2010; 43: 2000–9p.
[17] Garg H.C., Kumar V., Sharda H.B. Performance of slot-entry hybrid journal bearings considering combined influences of thermal effects and non-Newtonian behavior of lubricant, *Tribol Int.* 2010; 43: 1518–31p.
[18] Bhaskar S.U., Hussain M.M., Ali M.Y. Stability analysis on plain journal bearing with effect of surface roughness, *Int J Sci Eng Res*. 2013; 4(3): 1–8p.
[19] Mongkolwongrojn M., Aiumpronsin C. Stability analysis of rough journal bearings under TEHL with non-Newtonian lubricants, *Tribol Int.* 2010; 43: 1027–34p.
[20] Nair K.P., Nair V.P.S., Jayadas N.H. Static and dynamic analysis of elastohydrodynamic elliptical journal bearing with micropolar lubricant, *Tribol Int.* 2007; 40: 297–305p.
[21] Sharma Satish C., Kumar V., Jain S.C., et al. Study of hole-entry hybrid journal bearing system considering combined influence of thermal and elastic effects, *Tribol Int.* 2003; 36: 903–20p.
[22] Nassab S.A.G., Sohi H., Zaim E.H. Study of lubricant compressibility effect on hydrodynamic characteristics of heavily loaded journal bearing, IJST, *Trans Mech Eng.* 2011; 35: 101–5p.
[23] Mishra P.C., Pandey R.K., Athre K. Temperature profile of an elliptic bore journal bearing, *Tribol Int.* 2007; 40: 453–8p.
[24] Kuznetsov E., Glavatskih S., Fillon M. THD analysis of compliant journal bearings considering liner deformation, *Tribol Int.* 2011; 44: 1629–41p.
[25] Brito F.P., Miranda A.S., Claro J.C.P., et al.The role of lubricant

feeding conditions on the performance improvement and friction reduction of journal bearings, *Tribol Int.* 2014; 72: 65–82p.
[26] van Ostayen R.A.J., Beek A. Thermal modelling of the lemon-bore hydrodynamic bearing, *Tribol Int.* 2009; 42: 23–32p.
[27] Wang X.-L., Zhu K.-Q., Wen S.-Z. Thermohydrodynamic analysis of journal bearings lubricated with couple stress fluids, *Tribol Int.* 2001; 34: 335–43p.
[28] Michael S.A. Stability analysis of a turbocharger for marine diesel engine service, *M.S. Thesis.* State University, Blacksburg, Virginia, April, 2012, 13–9p.
[29] Suryawanshi S., Nagare P. Performance analysis of hydrodynamic journal bearing with the effect of whirl instability, *J Manuf Eng.* 2013; 8: 123–6p.
[30] Suryawanshi S., Pattiwar J. Study on influence of whirl instability on the performance of hydrodynamic journal bearing, *Int J Recent Res Sci Eng Technol.* 2015; 3(12).
[31] Harnoy A. *Bearing Design in Machinery*. New York: Marcel Dekker Inc.; 2002, 144–72p.
[32] Fuller Dudley D. *Theory and Practice of Lubrication for Engineers.* 2nd Edn, A Wiley Inderscience Publication; 1984, 271–82p.
[33] www.machinerylubrication.com/Read/470/journal-bearing-turbine.