Analysis of Load Locations and Magnitudes on Performance Parameters of Journal Bearing

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***Abstract*— In past few years a compact design of journal bearing has been used in most of the industrial applications. Hence there is need to develop the test bench to assess the performance characteristics. Where journal bearings are used under severe operating conditions, it is observed that, due to the high load imposed on the shaft, shaft starts deforming which causes change in oil film thickness, oil film temperature and subsequent misalignment in the bearings. Hence, experimental analysis was carried out for two load locations and having different magnitudes. One is for maximum center load of 100 N at 1440 rpm and second is maximum eccentric load of 100 N at 250 rpm, 350 rpm and 450 rpm. The results show that oil film thickness, oil film temperature depends on loading conditions. If load increases oil film thickness decreases, oil film temperature increases and misalignment increases. Using dimensionless analysis, a mathematical model is formulated which predicts the oil film thickness. The results obtained by mathematical model are in good agreement with experiential results.**

Keywords*— *Buckingham π method, oil film thickness, oil film temperature and misalignment.

I. INTRODUCTION

Journal bearing is most common hydrodynamic bearing which provide support and relative motion between rotor system. A journal bearing widely used in industrial applications such as heavy machinery, turbines and centrifugal pumps etc. The main problem of high rotating machinery is faulty bearing design or faulty assembly techniques, lead to excessive vibrations. To avoid these excessive vibrations in machinery, there is need to select appropriate operating parameters including misalignment of bearing as well. Sometimes journal bearing operate under severe operating conditions which causes increase in temperature and misalignment. Misalignment depends on manufacturing and assembly defect.

Several researchers have analyzed the misalignment aspect for journal bearing by experimental as well as analytical methods. Jun Sun et al,[1] presented analytical method to calculate film pressure, film thickness and misalignment, when shaft deformation occurred. They observed that misalignment has little influences on load capacity and coefficient of friction. J. Bouyer, M. Fillen, et al. [2] carried out experimental analysis to calculate minimum film thickness, misalignment maximum pressure and concluded that maximum pressure at mid plane decreased by 20 percent and minimum oil thickness was reduced by 80 percent. Further R. H. Buckholz, et al. [3] described non- Newtonian lubrication in partial arc journal bearing to calculate misalignment and results were compared with numerial analysis and experimental results. Chao Zhang, et al. [4] developed steady state mixed- THED (thermo-elssto-hydrodyanmic) model for journal bearing. They concluded that the shaft deformation and misalignment should considered in mixed lubrication analysis. Jun Sun, et al. [5] used experimental analysis to calculate oil film thickness, oil film temperature and misalignment in journal bearing and results shows that misalignment depend upon load factor, if load increases, misalignment also increase. Further J.O. Medwell et al. [6] used analysis to calculate pressure and temperature of journal bearing. They used Reynolds and energy equation to solve misalignment problem and concluded that bearing failure due to distortion of pressure and temperature field of misalignment shaft. B.Kucinschi et al. [7] carried out experimental analysis to determine temperature distribution in steady state journal bearing. The effect of journal speed on bearing temperature was analyzed. They observed that temperature was different while slow start-up as compare to rapid start-up. L.Costa et al. [8] studied experimentally using single groove-bearing to calculate shaft temperature, flow rate and bush torque. The result shows that characteristics of bearing are so sensitive. If groove position changes by 30^0 then the reduction occurred in temperature, pressure while flow rate increases. Monmousseau et al. [9] investigated rise in temperature of tilt-pad bearing and compared. They observed that while heat transfer in bearing elements realistic model should consider and also observed that if thermoelastic deformations are considered there is good agreement between experimental and theoretical temperature. Further Ravindra Mallya et al. [10] presented numerical analysis to calculate misalignment, load capacity, friction coefficient and side leakage of three axial groove bearing had groove angles 36° and 18°. The results show that load capacity and side leakage of 18° bearing is more than 36°, coefficient of friction of 18° bearing is less as compared to 36° bearing, misalignment depend on coefficient of friction. Boedo et al. [11] used numerical analysis to calculate misalignment in groove-less bearing. It was found that effects of bearing misalignment on load capacity was limited to end-plane journal eccentricity ratios close to 1. Further M.O.A Mokhtar et al. [12] explained adiabatic analysis to calculate misalignment of journal bearing. They observed that thermal effect was pronounced for journal bearing. ZS Zhang et al. [13] described numerical analysis (3-D Reynold equation) to identify effect of thermal

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on plain journal bearing. They conclude that thermal effect had not been ignored because their significant influence on oil film thickness and other parameters of bearing. J.Y. Jang et al. [14] explained comprehensive analysis of misalignment journal bearing, it was based on 3-D mass-conservative thermo-hydrodynamic model. The results presented in numerical solution were useful to examine misalignment in journal bearing.
Aim of this work is to study the effect of loads location and magnitudes on oil film thickness, oil temperature and misalignment of shaft by experimental approach. Further based on experimental results, the mathematical models are formulated by using dimensional analysis approach which predicts oil film thickness. Analytical results are compared with experimental results.

II. MATHEMATICAL MODELING

Dimensional analysis method is use to reduce complex dependence physical quantity into simplest form to analyses theoretical and experimental results. There are some basic physical quantities such as length (L), mass (M), time (T) and temperature(K) required for formation of dimensionless parameters. In this analysis Buckingham π method is used to construct the mathematical model. If there are 'n' variables in a system and these variables contain 'm' fundamental dimensions the equation relating all the variables will have (n-m) dimensionless groups.
In this application oil film thickness (t) is dependent variable while speed of journal (N), viscosity of oil (μ) , load applied on journal (W), temperature of fluid film (T) and clearance between journal and bearing (C) are independent variables. The functional equation between dependent and independent variables can be expressed as ;

$$t = f\,(N, \mu, W, T, C) \tag{1}$$

or

$$f\,(t, N, \mu, T, C) = 0 \tag{2}$$

In this problem, Number of variables = n = 6,
Number of fundamental dimensions = m = 4,
Thus, Number of dimensionless groups = n-m = 6-4 = 2
Therefore, equation can be expressed as;

$$f\,(\pi_1 , \pi_2) = 0 \tag{3}$$

Therefore,

$$\pi_1 = [\, N^{a_1}\, \mu^{b_1}\, W^{c_1}\, T^{d_1} \,]\ t \tag{4}$$

$$\pi_2 = [\, N^{a_2}\, \mu^{b_2}\, W^{c_2}\, T^{d_2} \,]\ C \tag{5}$$

Table 1 shows parameters and their SI unit and fundamental dimensions.

TABLE 1. SI unit and fundamental dimension.

Parameter	SI unit	Fundamental dimensions
Oil film thickness (t)	mm	[L]
Speed (N)	RPM	$[T^{-1}]$
Viscosity (μ)	$\frac{kg}{m.s}$	$[M\ L^{-1}T^{-1}]$
Load (W)	Kg	[M]
Temperature (T)	K	[K]
Clearance I	mm	[L]

Solving equations 3, 4 and 5 we get values of π_1 and π_2.
For π_1 ;

$[M^0L^0T^0K^0] = [[T^{-1a_1}]\ [M^{b_1}L^{-1b_1}T^{-1b_1}]\ [M^{c_1}]\ [K^{d_1}]]\ \ [L^1]$

Therefore,

$$M = 0 = b_1 + c_1, \qquad \therefore c_1 = -1 .$$
$$L = 0 = - b_1 + 1, \qquad \therefore b_1 = 1 .$$
$$T = 0 = -a_1 - b_1, \qquad \therefore a_1 = -1 .$$
$$K = 0 = - d_1, \qquad \therefore d_1 = 0.$$

Therefore,

$$\pi_1 = [\, N^{-1}\, \mu^1\, W^{-1}\, T^0 \,]\ t$$

$$\pi_1 = \frac{\mu\ t}{N\ W} \tag{6}$$

For π_2 ;

$[M^0L^0T^0K^0] = [[T^{-1a_2}]\ [M^{b_2}L^{-1b_2}T^{-1b_2}]\ [M^{c_2}]\ [K^{d_2}]]\ \ [L^1]$

Therefore,

$$M = 0 = b_1 + c_1, \qquad \therefore c_1 = -1 .$$
$$L = 0 = - b_1 + 1, \qquad \therefore b_1 = 1 .$$
$$T = 0 = -a_1 - b_1, \qquad \therefore a_1 = -1 .$$
$$K = 0 = - d_1, \qquad \therefore d_1 = 0.$$

Therefore,

$$\pi_2 = [\, N^{-1}\, \mu^1\, W^{-1}\, T^0 \,]\ C$$

$$\pi_2 = \frac{\mu\ C}{N\ W} \tag{7}$$

Now putting experimental results in equations 6 and 7, will get values of π_1 and π_2. Further plotting π_1 v/s π_2, and using curve fitting tool , we get a quadratic equation for center loading and eccentric loading. For this experiment viscosity of oil is 0.061 $\frac{kg}{m.s}$ at 50 to 60°C is used.

A. Theoretical calculations for center loading

For C = 0.2 at 1440rpm
Fig 1 shows graph of dimensionless oil film thickness (π_1) and dimensionless clearance (π_2).

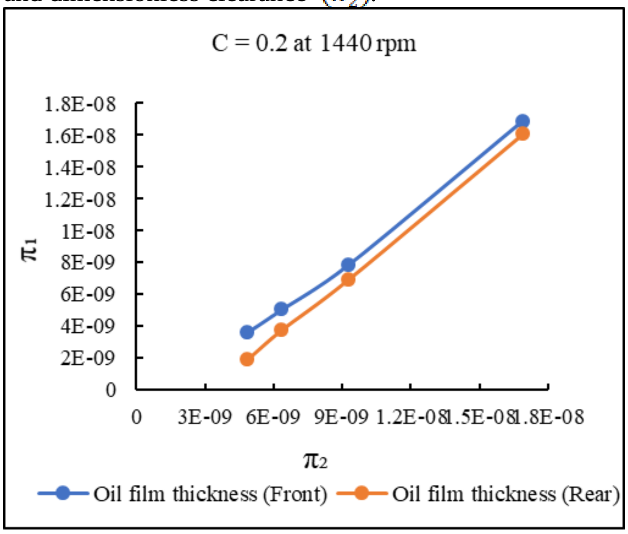

Fig. 1. Dimensionless oil film thickness (π_1) v/s dimensionless clearance (π_2).

From fig. 1 the linear quadratic equations in terms of π_1 and π_2 is developed by using curve fitting method and expressed as follows;

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Liner quadratic equation for oil film thickness (Front) is,

$$\pi_1 = 1.0547\ \pi_2 - 2E\text{-}09 \quad (8)$$

Liner quadratic equation for oil film thickness (Rear) is,

$$\pi_1 = 0.9001\ \pi_2 + 2E\text{-}09 \quad (9)$$

Fig. 2 shows graph of experimental and theoretical oil film thickness v/s Load (N) for C = 0.2 at 1440 rpm.

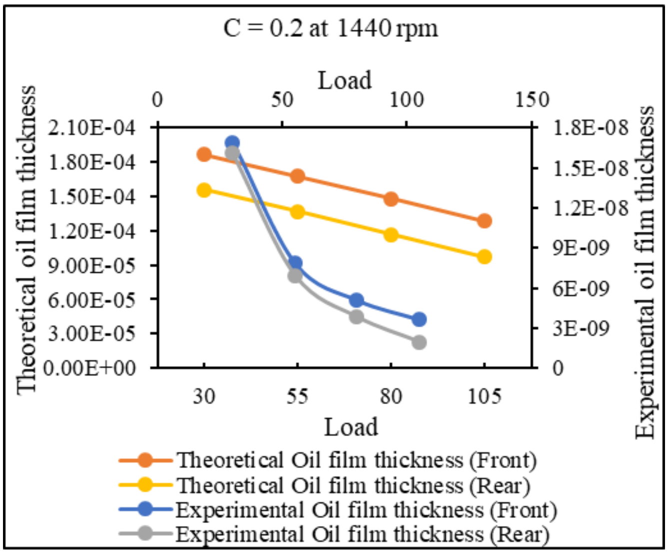


Fig. 2. Experimental and theoretical oil film thickness v/s Load (N) for C = 0.2 at 1440 rpm.

The graphs and equations of remaining parameters of center and eccentric loading have the similar trends hence these results are not presented in this paper.

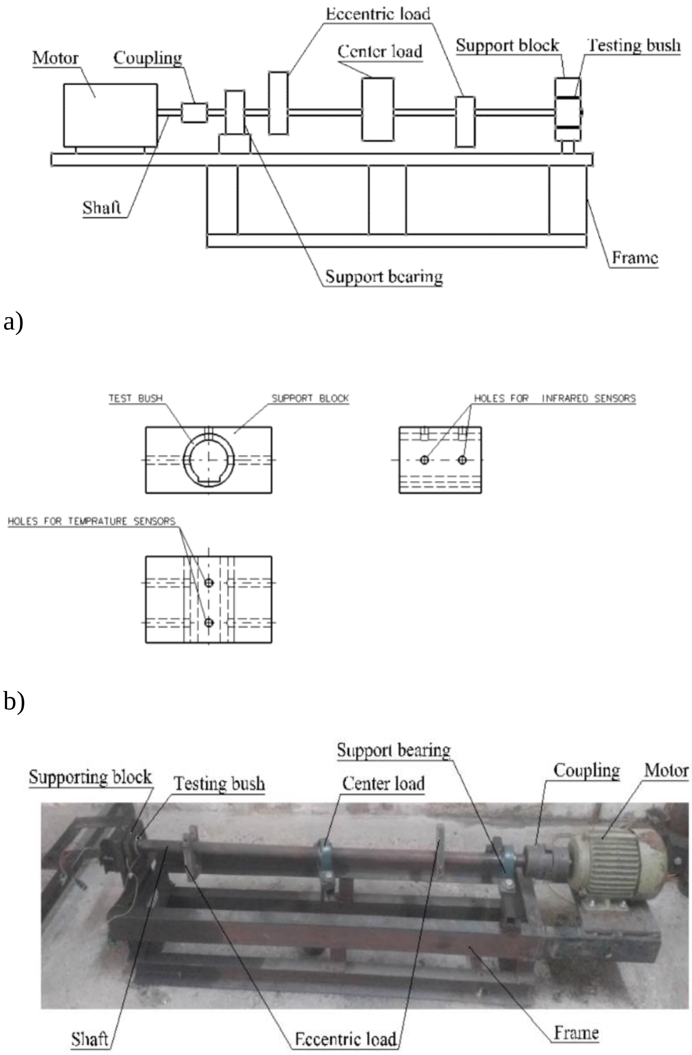
From above Figure 2 it concludes that theoretical oil film thickness are almost close with experimental oil film thickness.

III. EXPERIMENTAL ANALYSIS

The schematic of experimental test rig is shown in Figure 3 a) and a systematic view of test bush and sensor mounting assembly as shown in Figure 3 b) while actual photograph of test ring installed in laboratory is presented in Figure 3 c). The test ring is used to analyze the effect of load magnitudes and locations on performance parameters such as oil film thickness and temperature and, misalignment of journal bearing. For this experiment nylon 66 material bushes are used.

Analysis is carried out at two conditions; one is center loading and second one eccentric loading. In case of center loading motor speed of 1440 rpm is used and load is applied at the center of shaft. One end of shaft is coupled to motor via supporting bearing and at other end of shaft test bush is assembled, where sensors are embedded in lubricated test bearing at appropriate locations. Due to applying load at the center of the shaft, shaft start deforming which causes change in oil film thickness, oil film temperature and misalignment.

An oil film thickness is measured by IR sensor and oil film temperature is measured by temperature sensor and misalignment is calculated by theoretical approach. If load increases oil film thickness decreases, oil film temperature increases and misalignment increases. In case of eccentric loading above whole procedure is same but loading condition is different, where load is applied near to both bearings. When shaft starts rotating due to eccentric weight centrifugal force is developed. In case of eccentric loading we use 3-phase auto-transformer to reduce rpm for three different speeds such as 250 rpm, 350 rpm and 450 rpm.



c)

Fig. 3. a) Schematic diagram of experimental set-up; b) test bush and sensor mounting assembly; c) Photograph of test rig.

Table 2 shows parameters used for experimental analysis.

TABLE 2. Parameters of journal bearing, shaft, operating condition and lubricant properties

Bearing diameter D (mm)	I.D =30, O.D = 40
Bearing length l_b (mm)	67
Radial clearance C (mm)	0.2,0.4,0.5
Rotating speed (RPM)	1440
Lubricant used	SAE W30
Shaft length between two bearings l_s (mm)	1000
Shaft diameter d (mm)	30
Young Modulus for shaft material E (Gpa)	210

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IV. RESULTS AND DISCUSSIONS

In this section the results corresponding to effect of center and eccentric loads on test bearing and behavior of oil film thickness, oil temperature and misalignment angle are presented.

A. For Centre Loading

Fig. 6 shows that graph for oil film thickness v/s load for clearance C = 0.2, 0.4 and 0.5 at 1440 rpm.

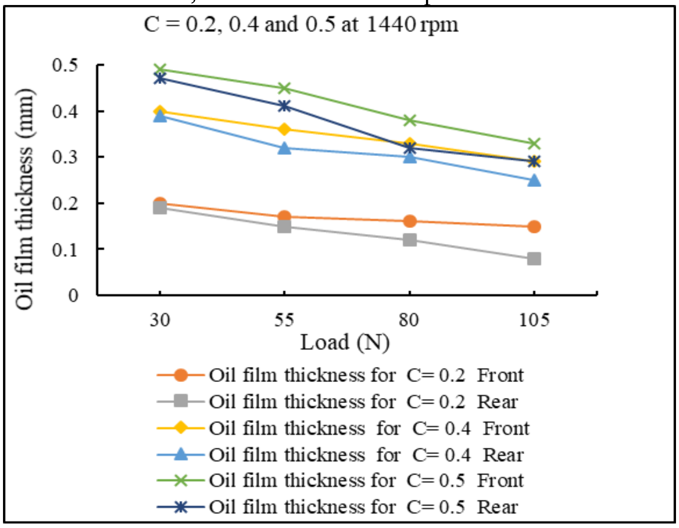

Fig. 6. Oil film thickness v/s Load for clearance C=0.2, 0.4 and 0.5 at 1440 rpm.

From above fig. 6, it concludes that oil film thickness decreases due to increasing load and decreasing viscosity of oil.

Fig. 7 shows graph for oil film temperature v/s load for clearance C = 0.2, 0.4 and 0.5 at 1440 rpm.

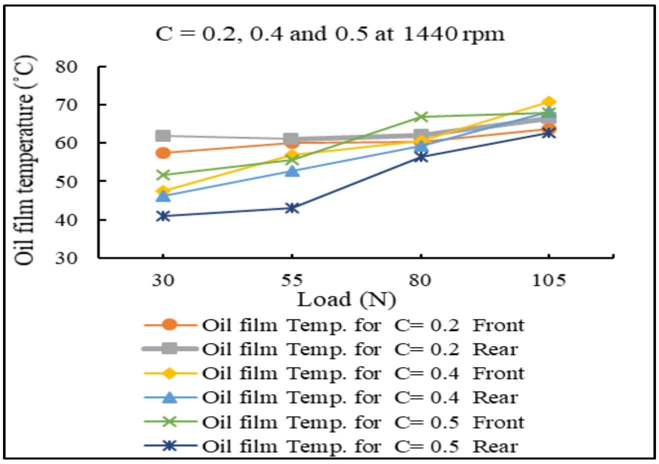

Fig. 7. Oil film temperature v/s Load for clearance C=0.2, 0.4 and 0.5 at 1440 rpm.

From above fig. 7, it concludes that oil film temperature increase due to increasing load.

B. Misalignment angle for center load

The force acting on the center of the shaft causes the shaft deformation; hence the journal misalignment occurred. Based on the calculations of deformation of beam acted by the concentrative force, the angle of journal misalignment in the bearing hole can calculated by following expression;

$$\text{Slope} = \tan\alpha = \frac{Pl^2}{16EI} \qquad (10)$$

Where,

α = Angle of misalignment in bearing hole, P = Force acting on the center of shaft, *l* = Length of shaft, E = Modulus of elasticity, *I* = Moment of inertia.

From fig. 8 it concludes that if load increases misalignment increases.

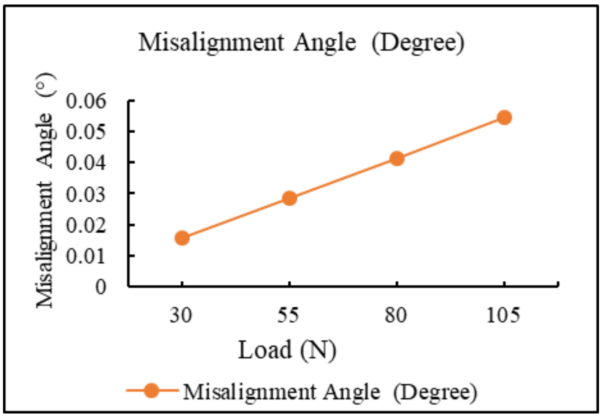

Fig. 8: Load v/s misalignment angle.

C. For Eccentric Loading

Fig. 9 shows that graph for oil film thickness v/s load for clearance C = 0.2, 0.4 and 0.5 at 250 rpm.

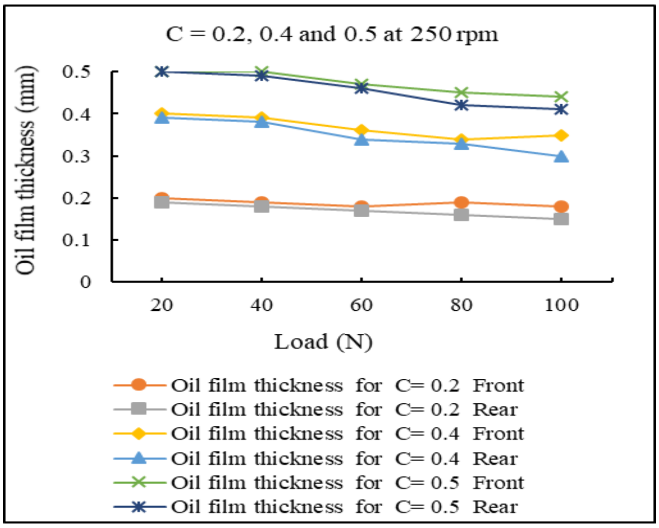

Fig. 9. Oil film thickness v/s Load for clearance C=0.2, 0.4 and 0.5 at 250 rpm.

Fig. 10 shows that graph for oil film thickness v/s load for clearance C = 0.2, 0.4 and 0.5 at 350 rpm.

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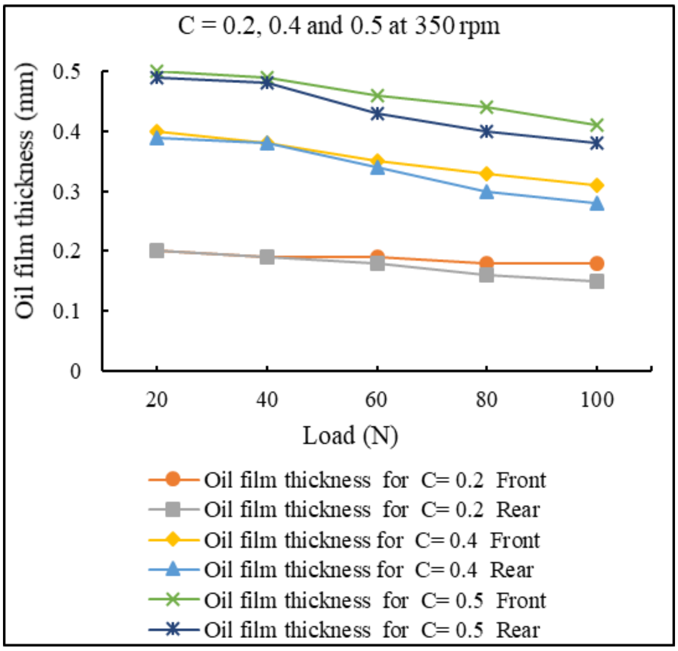

Fig. 10. Oil film thickness v/s Load for clearance C=0.2, 0.4 and 0.5 at 350 rpm.

Fig. 11 shows that graph for oil film thickness v/s load for clearance C = 0.2, 0.4 and 0.5 at 450 rpm.

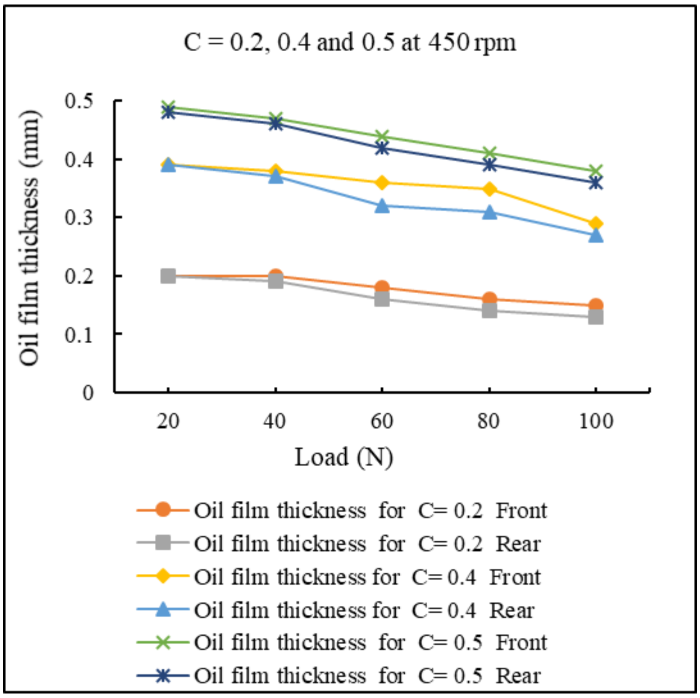

Fig. 11. Oil film thickness v/s Load for clearance C=0.2, 0.4 and 0.5 at 450 rpm.

Fig.12 shows that graph for oil film temperature v/s load for clearance C = 0.2, 0.4 and 0.5 at 250 rpm.

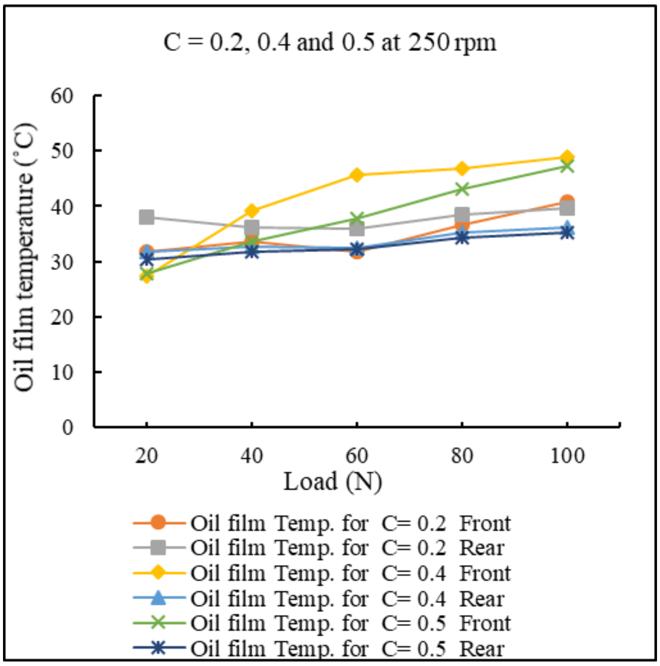

Fig. 12. Oil film temperature v/s Load for clearance C=0.2, 0.4 and 0.5 at 250 rpm.

Fig. 13. Shows that graph for oil film temperature v/s load for clearance C = 0.2, 0.4 and 0.5 at 350 rpm.

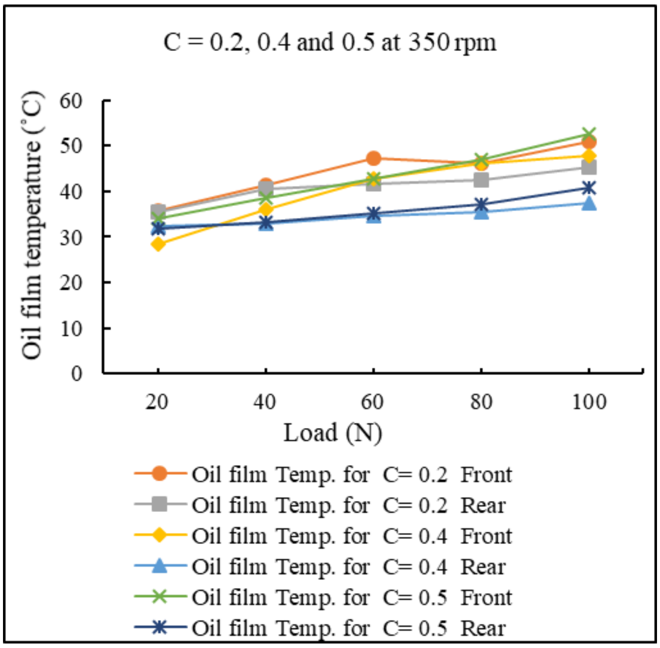

Fig. 13. Oil film temperature v/s Load for clearance C=0.2, 0.4 and 0.5 at 350 rpm.

Fig. 14 shows that graph for oil film temperature v/s load for clearance C = 0.2, 0.4 and 0.5 at 450 rpm.

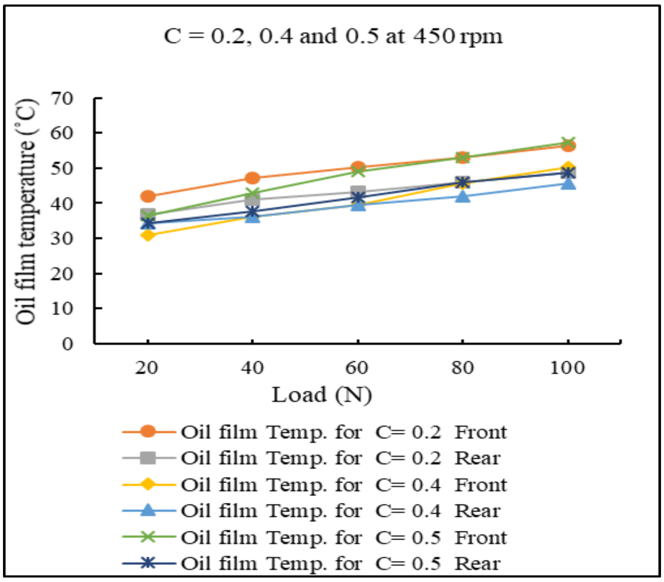

Fig 14. Oil film temperature v/s Load for clearance C=0.2, 0.4 and 0.5 at 450 rpm.

From figs 9, 10 and 11 it concludes that oil film decreases due to increasing load and decreasing viscosity of oil.
From figs. 12, 13 and 14 it concludes that oil film temperature increase due to increasing load.

D. Misalignment angle for eccentric load

The force acting on the eccentric of the shaft causes the shaft deformation; hence the journal misalignment occurred. According to Maxwell's theorem of reciprocal displacements if the two loads are now made equal, then

$$\alpha_1 = \alpha_2 \quad (11)$$

$$\text{slope} = \tan\alpha_1 = 0.062\,\frac{Pl^2}{EI} \quad (12)$$

Where,
α = Angle of misalignment in bearing hole, P = Force acting on the center of shaft, l = Length of shaft, E = Modulus of elasticity, I = Moment of inertia.

From fig. 15 it concludes that if load increases misalignment increase.

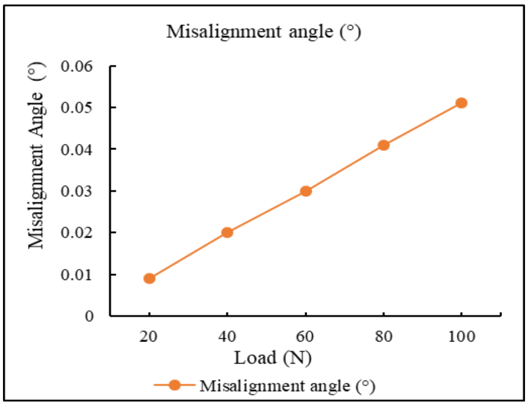

Fig. 15. Load v/s Misalignment angle.

V. CONCLUSIONS

Effect of load locations and magnitudes on journal bearing performance is analyzed experimentally. It is observed that as load increases oil film thickness depresses, misalignment increases and temperature increases for all loading conditions. However, still more number of experiments are required to obtain optimum values for satisfactory performance of the bearings. The mathematical models are constructed based on dimensional analysis approach and observed good correlation between experimental and analytical results.

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