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Experimental Investigation on Hydrodynamic Journal Bearing using SAE 10W30 Multi Grade Oil

Paras Kumar and Ashish Kumar Gupta***

ABSTRACT

This paper presents an experimental study of pressure distribution on hydrodynamic journal bearing with SAE 10W30 multi grade oil. Hydrodynamic Journal bearing test rig (HJBTR) is used to test the 40 mm diameter and 40 mm long bearing (l/d = 1) made of Bronze. Test bearing is located between two antifriction bearings and loaded mechanically. The space between the shaft and the bearing is filled with SAE 10W30. A constant load of 800 N is applied at various journal rotational speeds of 1000, 1500, 2000 rpm. Various parameters like frictional torque, oil temperature and pressure at 10 different sensors along circumferential direction were recorded from Hydrodynamic Journal Bearing Test Rig (HJBTR). These results were used for experimental calculations and theoretical verification using Raimondi and Boyd charts for practical design. The experimental plot of pressure ratio vs sommerfeld number indicates that the working conditions are in the stable hydrodynamic regime. Also experimental results were following the same trend as MCKEE"s investigation curve.

Keywords: *Hydrodynamic Bearing; Pressure Distribution; Thick Film Lubrication; Journal Bearing; Multi Grade Oil.*

1.0 Introduction

Hydrodynamic Journal bearing is a critical power transmission component that carry high load in different machineries. It is essential to study the performance characteristics under the different loading and operating conditions. The behaviour of Hydrodynamic Journal Bearing is also dependent on lubricant used and can be studied experimentally with the help of a test rig.

Numerous studies of the operating conditions of Hydrodynamic Journal Bearing have been made during the last decades. Ma and Taylor [1] carried out a theoretical and experimental study of thermal effects in a plain circular journal bearing under steady load.

Baskar et al. [2] analysed the pressure distribution on hydrodynamic journal bearing under different Bio lubricants. Gethin [3] studied the thermal behavior of various types of high speed journal bearings and found good agreement with theoretical and the experimental results. Chun [4] examined the effect of Variable specific heat on maximum pressure, maximum temperature, bearing load, frictional loss journal. Now when the journal starts rotating, then at low speed and side leakage in high-speed journal bearing. Film pressure, load-carrying capacity attitude angle, end leakage flow rate, frictional coefficient and misalignment moment were calculated for different values of misalignment degree and eccentricity ratio.

It was found that there are obvious changes in film pressure distribution, the highest film pressure, film thickness distribution, the least film thickness and the misalignment moment when misalignment takes place. Sep [5] investigated the oil flow velocity, pressure and temperature distribution of a journal bearing with a two-component surface layer.

After all such studies, still the case is far from closed. There are a limited number of studies that carry out an in-depth examination of the true operating conditions of bearings in actual experiments upon the test rig with different oils.

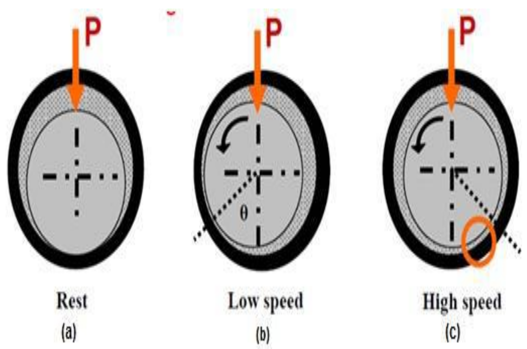
There is also a need for experimental studies to verify the theoretical ones.

**Corresponding Author: Department Of Mechanical Engineering, Delhi Technological University, Delhi, India (E-mail: paraskum007@rediffmail.com)*
***Department Of Mechanical Engineering, Delhi Technological University, Delhi, India*

2.0 Theory of Hydrodynamic Journal Bearing

Fig-1 shows the operation of Hydrodynamic Journal Bearing. When the journal is at rest, it touches the bush due to bearing load P at the lower most position and there is no oil film between the bush and the

Fig: 1. Operation of Journal Bearing



condition, with the load P acting, it has a tendency to shift to its sides as shown in the Fig-1 (b). At this equilibrium position, the frictional force will balance the component of bearing load. In order to achieve the equilibrium, the journal orients itself with respect to the bush.

The angle θ, shown for low speed condition, is the angle of friction. Normally at this condition either a metal to metal contact or an almost negligible oil film thickness will prevail.

At the higher speed, the equilibrium position shifts and a continuous oil film will be created as indicated in the Fig-1 (c). This continuous fluid film has a converging zone, which is shown in the magnified view.

It has been established that due to presence of the converging zone or wedge, the fluid film has sufficient pressure to support the heavy load. Hence, to build-up a positive pressure in a continuous fluid film, to support a load, a converging zone is necessary.

Moreover, simultaneous presence of the converging and diverging zones ensures a fluid film continuity and flow of fluid. The Pressure distribution in Hydrodynamic Journal Bearing is shown in Fig-2.

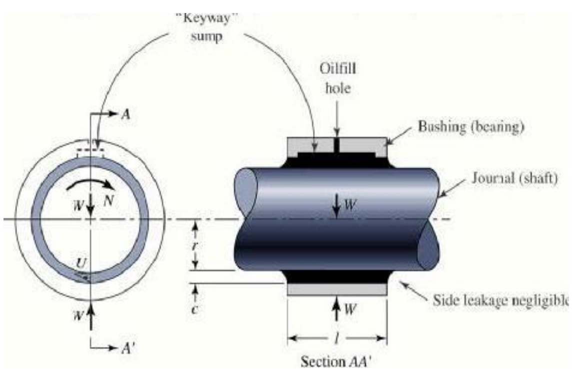
Fig: 2. Pressure Distribution [12]

The action of the rotating journal is to pump the lubricant around the bearing. The lubricant is pumped into a wedge shaped space. A minimum film thickness occurs, not at the bottom but slightly displaced. This is explained by the fact that the film pressures in the converging half reaches a maximum somewhere to the left of the bearing center. The wedge shaped fluid film applies a perpendicular force to the bearing, whose one component is vertical which supports the load [6].

2.1 Petroff 's equation

In 1883, Petroff published his work on bearing friction based on simplified assumptions.

Fig: 3. Petroff's Lightly Loaded Journal Bearing [9]



With reference to Fig-3, an expression for viscous friction drag torque is derived by considering the entire cylindrical oil film as the "liquid block" acted upon by force W.

From Newton"s law of Viscosity:

$$\tau = \mu \frac{U}{h} = \frac{2\pi r \mu N}{c} \quad (1)$$

The torque is the force times the lever arm and calculated as

$$T = (\tau A)(r) = \left(\frac{2\pi r \mu N}{c}\right)(2\pi r l)(r) = \frac{4\pi^2 r^3 l \mu N}{c} \quad (2)$$

Pressure (p)= W/2rl

Frictional torque is given by [7]

$$T = fWr = (f)(2rPl)(r) = 2r^2 flP$$

Substituting the value of torque from eq. (2) and (3) and the coefficient is expressed as

$$f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c} \quad (4)$$

When the journal is not concentric i.e. there is some eccentricity "e" between the journal and bearing, then Petroff's equation takes the form:

$$f = K_1 \frac{\pi^2}{30}\left(\frac{\mu N}{P}\right)\left(\frac{r}{c}\right) \quad (5)$$

where,

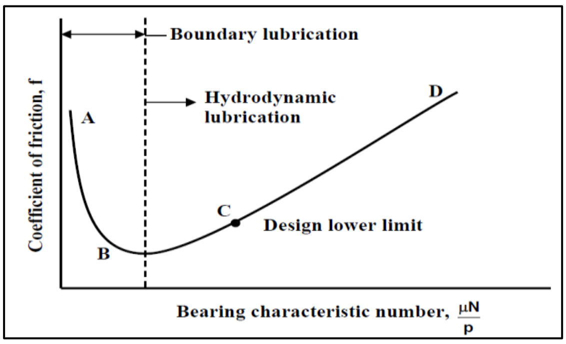
$$K_1 = \frac{1}{\sqrt{1-\left(\frac{e}{c}\right)^2}} \quad (6)$$

2.2 McKee's investigation

The different regimes of operation were further investigated by McKee brothers. Fig-4 shows a plot of variation of Coefficient of Friction (f) with Bearing Characteristic Number $\left(\frac{\mu N}{P}\right)$

The plot shows that initially there is a very high friction between the journal and bearing as there is no fluid film between the two. As the Journal speed is increased value of friction drops drastically. This region [From A to B] is referred as Boundary sometimes

Fig: 4. Mckee"S Investigation Curve [9]



termed as imperfect lubrication. Imperfect lubrication means that metal – metal contact is possible or some form of oiliness will be present. The portion from B to D is known as the hydrodynamic lubrication .The calculated value of bearing characteristic number should be somewhere in the zone of C to D. This zone is characterized as design zone.

For any operating point between C and D due to fluid friction certain amount of temperature generation takes place. Due to the rise in temperature the viscosity of the lubricant will decrease, thereby, the bearing characteristic number also decreases. Hence, the operating point will shift towards C, resulting in lowering of the friction and the temperature.

As a consequence, the viscosity will again increase and will pull the bearing characteristic number towards the initial operating point. Thus a self control phenomenon always exists. For this reason the design zone is considered between C and D. The lower limit of design zone is roughly five times the value at B. On the contrary, if the bearing characteristic number decreases beyond B then friction goes on increasing and temperature also increases and the operation becomes unstable.

Therefore, it is observed that, bearing characteristic number controls the design of journal bearing and it is dependent of design parameters like, operating conditions (temperature, speed and load), geometrical parameters (length and diameter) and viscosity of the lubricant.

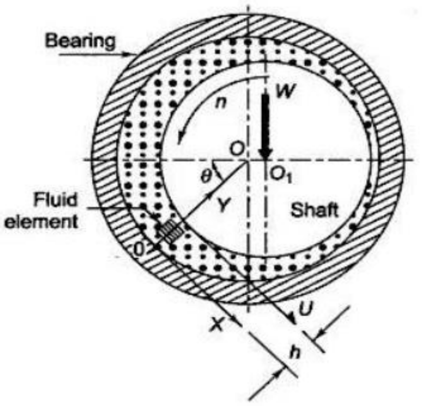
2.3 Reynols's equation

Theory of hydrodynamic lubrication is based on differential equation derived by Osborne Reynold.

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{U}{2}\frac{\partial h}{\partial x} \quad (7)$$

The left hand side of the equation represents flow under the pressure gradient. The corresponding right hand side represents a pressure generation mechanism. In this equation it has been assumed that the lubricant is incompressible and Newtonian [8].

Fig: 5. Fluid Element in X-Y Plane



Solutions to Reynolds equation were developed in first decade of 20th century and were applicable for long bearings and give reasonably good results for bearings with(l / d) ratios more than 1.5. Ocvirk‟s short bearing approximation on the other hand gives accurate results for bearings with (l /d) ratio up to 0.25 and often provides reasonable results for bearings with (l / d) ratios between 0.25 and 0.75. Raimondi and Boyd have obtained computerized solutions for Reynolds equation. (7) and reduced them to chart form which provide accurate solutions for bearings of all proportions [9].

3.0 Experimental Setup

Experimental test was carried out using a Hydrodynamic Journal Bearing Test Rig (HJBTR) TR-660 shown in Fig-6, (designed by Ducom instruments Ltd, Bengaluru). It is versatile equipment for easy operation and measuring the pressure distribution on journal and also designed to ensure laminar oil flow in the bearing. The rig was designed to enable operation close to the limit of stability to be investigated. The lubricant was supplied to the bearing at an inlet port on the vertical centre line of the bearing, in the cavitated region of the bearing clearance; the oil exited the bearing in an axial direction and was collected at drain positions before being returned to the oil supply tank. 10 no's of pressure sensors are mounted on the face of bearing to measure Pressure distribution of the oil along the bearing circumference. Also loading arm is pivoted to get 1:5 loading ratio, exerts pressure on bearing through a roller when dead weights are placed on pan in the longer end. A counter weight is fixed on short end of lever to balance lever. This tester mainly consists of a rotating journal held in a special bearings in horizontal condition, it is rotated by AC motor and speed controlled by variable frequency drive. A bearing having l/d ratio=1 & c/r ratio=0.00135 is inserted over the

journal and held in position by a loading lever. Bearing Pressure distribution on journal is displayed at different indexing angles on PC. When journal rotates, pressure created inside is measured & displayed on PC, depending on setting on PC the stepper motor rotates and indexes the bearing the value is measured and plotted on PC.

Fig: 6. Hydrodynamic Journal Bearing Test Rig [11]

The bearing performance characteristics were obtained for the following parameters:

1. Constant loading of 800N.
2. Journal speed variation from 1000 to 2000 rpm.
3. Lubricant used : Mobil 1 SAE 10W30
4. Ambient Temperature: 31℃-32℃.
5. Material for bearing : Bronze

The oil used contain viscosity modifier to stabilize its viscosity over a wide range of operating temperature. Typical properties are elaborated in the Table-1. From Hydrodynamic journal Bearing Test Rig (HJBTR), the values of frictional torque (T), oil inlet temperature, and pressure distribution were measured throughout the region. The measured parameters are shown in observation Tables- 2 and 3.

Table: 1. Typical Properties of Mobil 1 10W-30

S No.	Properties	Values
1.	SAE Grade	10W-30
2.	Viscosity @ 100ºC, cSt (ASTM D445)	10.1
3.	Viscosity, @ 40ºC, cSt (ASTM D445)	63.2
4.	Viscosity Index	146
5.	Sulfated Ash, wt% (ASTM D874)	0.8
6.	HTHS Viscosity, mPa.s @ 150ºC (ASTM D4683)	3.0
7.	Pour Point, ºC (ASTM D97)	-42
8.	Flash Point, ºC (ASTM D92)	232
9.	Density @15.6 ºC, g/ml (ASTM D 4052)	0.859

Table: 2. Observation Table

S No.	Load (N)	Journal Speed (rpm)	Frictional Torque (N-m)	Oil Inlet Temp. (℃)	Maximum Pressure (MPa)
1.	800	1000	0.72	38.30	0.999
2.	800	1250	0.77	39.60	0.998
3.	800	1500	0.78	39.90	0.994
4.	800	1750	0.82	41.60	0.989
5.	800	2000	0.84	42.90	0.984

Table: 3. Detailed Pressure Readings at 800 N Load

S No.	Journal Rotational Speed	Radial Pressure variation Distribution at 10 Different Sensors (MPa)									
	Rpm	P1	P2	P3	P4	P5	P6	P7	P8	P9	P10
1.	1000	0.876	0.781	0.768	0.800	0.866	0.971	0.999	0.896	0.742	0.734
2.	1250	0.875	0.775	0.748	0.783	0.860	0.968	0.997	0.892	0/740	0.735
3.	1500	0.878	0.766	0.739	0.765	0.858	0.960	0.994	0.890	0.742	0.737
4.	1750	0.877	0.766	0.737	0.762	0.855	0.951	0.991	0.890	0/739	0.734
5.	2000	0.876	0.764	0.735	0.759	0.848	0.942	0.984	0.888	0.738	0.737

5.0 Results and Discussion

The experimentation performed on the test rig with SAE10W30 oil at Constant load and varying rpm showed that the oil contains uniformly dispersed bubbles with a constant radius under atmospheric pressure. It was assumed that the bubbles remain spherical in the oil film and no interference, no break and combination between bubbles take place. Table 4 shows the comparison between theoretical (calculated by Raimondi and Boyd charts) and experimental (calculated by frictional torque obtained from test rig) values of coefficient of friction.

Table-4 Relative Error for Coefficient of Friction

S No.	Load (N)	Rpm variation	Coefficient of friction (experimental)	Coefficient of friction (theoretical)	% Error
1.	800	1000	0.0167	0.0121	29
2.	800	1500	0.0181	0.0135	31.8
3.	800	2000	0.0205	0.0162	23.4

Table 5 shows the comparison between theoretical and experimental values of pressure ratio (p/pmax.). The circumferential pressure distribution at different rpm is presented in Fig-7. Also variation of frictional coefficient (f) with bearing characteristic number (μn/p) is shown in Fig-8 and the variation of pressure ratio (p/pmax.) with sommerfeld number is shown in Fig-9. It is clear from the Fig-8 that with the increase in rpm, value of (f) increases throughout the region and thus it lies on the right side of the Mckee‟s curve. It also states a thick film lubrication and not any mixed or thin film lubrication. The value of torque (T) also increases with rpm and subsequently with coefficient of friction (f). The results obtained are in accordance with Raimondi and Boyd chart.

Table: 5. Relative Error for Pressure Ratio

S No.	Load (N)	Journal Rotational Speed (rpm)	Pressure ratio (P/Pmax.) (Experimental)	Pressure ratio (P/Pmax.) (Theoretical)	% Error
1.	800	1000	0.540	0.500	7.4
2.	800	1500	0.550	0.503	8.54
3.	800	2000	0.553	0.508	8.13

Fig: 7. Polar Pressure Distribution Curve at 800N Load and Different RPM

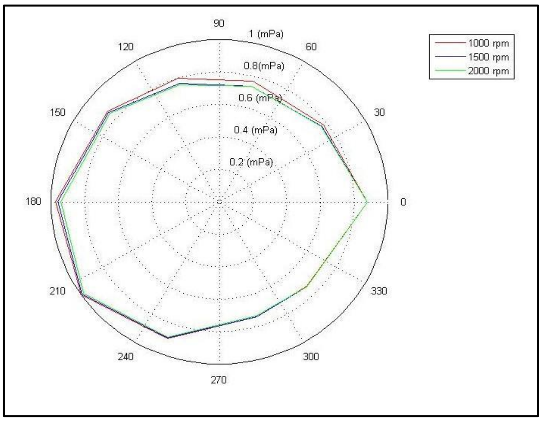
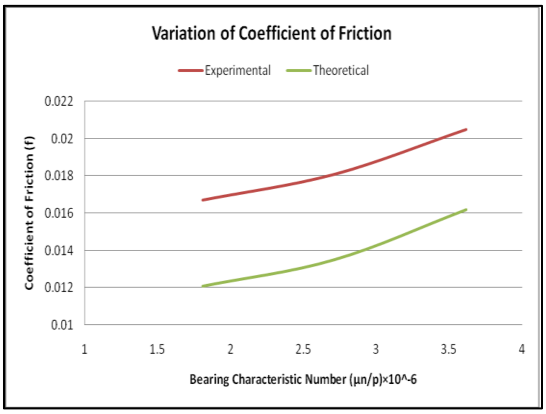


Fig: 8. Bearing Characteristic Number $\left(\frac{\mu N}{P}\right) \times 10^{-6}$ vs Coefficient of Friction (f)



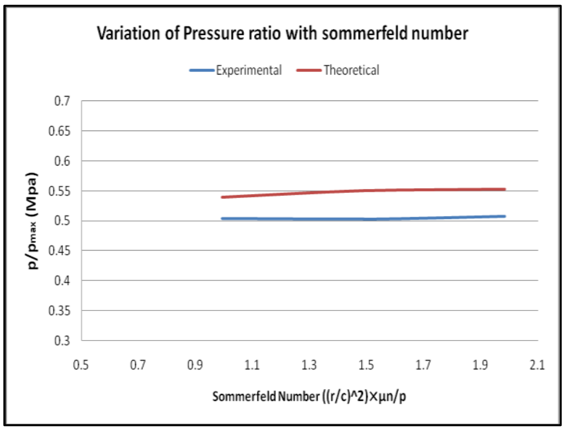


Fig: 9. Pressure Ratio vs. Summerfield Number

coefficient of friction (f) due to addition of more fluid layers in the wedge shaped region. The variation of pressure distribution was in accordance with Raimondi and Boyd charts and also the experimental values matches with the theoretical values with a good degree of closeness. Overall theoretical trend was even followed at actual working conditions and thus design criteria are safe. Further the work can be extended by performing the experimentation with different types of lubricants to obtain a comparative study and choose their suitability according to conditions.

6.0 Conclusion

The experiment conducted showed that, at all operating conditions of 800N Load and different rpm, the plot of (μn/p) vs (f) lies on the right side of the Mckee‟s curve thus ensuring thick film lubrication in the entire region. There was no thin film or mixed lubrication. Also it was observed that there was a slight variation of pressure with increase in rpm. This confirms the stability of fluid film in the bearing. Torque increases with increase in rpm or

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