Analysis of Hydrodynamic Journal Bearing

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Abstract. The use of journal bearings is widespread in various rotating machinery applications, as they can efficiently transmit large loads at moderate rotational speeds. These bearings come in different designs, with the hydrodynamic journal bearing being a popular option based on hydrodynamic lubrication. This lubrication method involves the creation of a thick layer of lubricant between the load-bearing surfaces of the bearing to prevent metal-to-metal contact. In this project, the focus is on analyzing different techniques, equations, and theories used to determine crucial parameters such as load carrying capacity, minimum oil film thickness, friction loss, and temperature distribution of hydrodynamic journal bearings. The primary objective is to investigate the pressure distribution on hydrodynamic journal bearings under various loading and operating conditions and develop a mathematical model for journal bearings. The results will be compared using MATLAB software, as accurate predictions of these parameters are crucial for designing efficient hydrodynamic journal bearings.

Introduction

Lubrication is a critical process that involves the application of lubricant to reduce friction and wear between two surfaces. This field is a discipline in tribology that is concerned with controlling friction and wear through the introduction of a friction-reducing film between moving surfaces. Lubricants can come in various forms, such as fluids, solids, or plastic substances. The most common types of lubricants are oils and grease, which prevent friction by creating a boundary layer between two surfaces. Lubricants also dissipate heat from surfaces, transport contaminants to filters, protect against oxidation and corrosion, and facilitate power transmission.The primary aim of this work is to understand lubrication and journal bearings, study the various factors that affect bearing performance such as pressure, temperature, density, and viscosity of lubricant, and investigate the tribological behavior of journal bearing materials under conditions such as high speed and maximum pressure. Furthermore, this project aims to study the mathematical modeling of journal bearings and compare the results using MATLAB and ANSYS software. An additional objective is to analyze the pressure distribution on hydrodynamic journal bearings under different lubricants for various loading conditions and operating parameters and to investigate the heat transfer effect in hydrodynamic journal bearings. By achieving these goals, we can better understand lubrication and improve the design and performance of journal bearings in various applications.

Literature Review

Numerous researchers have conducted studies on liquid lubricants and lubrication, as well as the mechanics of friction and the relationship between friction and wear. For example, S. Baskar and G. Sriram [01] explored the mechanics of friction, while PriyankaTiwari et al. [02] focused on the determination of load carrying capacity, minimum oil film thickness, friction loss, and temperature distribution of hydrodynamic journal bearings. Chaitanya K Desai et al. [03] analyzed the pressure distribution in hydrodynamic journal bearings for various loading conditions and operating parameters. Arti Singh and Prof. S.S. Waydande [04] also conducted a study on journal bearings and their widespread application in rotating machinery. Additionally, Hulin Li, Yanzhen Wang, NingZhong, Yonghong Chen, and Zhongwei Yin [05] used a new computational fluid dynamics and fluid-structure interaction method to investigate the performance of journal bearings, with consideration of the effects of thermal and cavitation. In summary, lubrication technology is crucial to understanding journal bearings and plays a vital role in bearing design and application, as demonstrated by the works of these researchers.

Hydrodynamic Journal Bearing

Bearings are essential components used in machinery to prevent friction between parts during relative movement. There are two primary categories of bearings: anti-friction or rolling element bearings and hydrodynamic journal bearings. The main function of a bearing is to carry load between a rotor and the case while minimizing wear. This function is present in almost every aspect of daily life, from the watches we wear to the automobiles we drive to the disk drives in our computers. In industrial applications, journal bearings are commonly used in rotating machinery, both at low and high speeds. Understanding lubrication technology is crucial to comprehending journal bearings and is vital to their design and application. This paper aims to provide an introduction to journal bearings and lubrication, highlighting the crucial role that lubrication technology plays in bearing design and application.

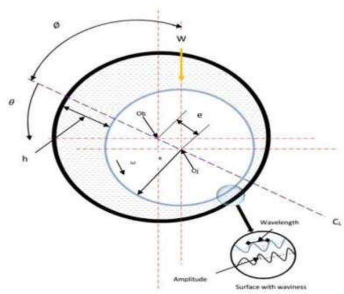


Figure 1 Schematic diagram of hydrodynamic journal bearing.

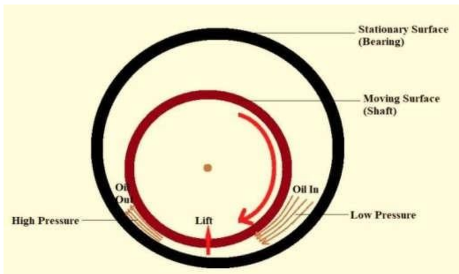


Figure 2 Basic Development of an Oil Wedge

Principle of Journal Bearing

The principle of a journal bearing is that it operates with an adequate supply of lubricant toreduce friction and wear between the shaft and housing. As the bearing carries a load, the geometric centers of the shaft and housing may become displaced, leading to a region of convergent flow. However, the pressure distribution will adjust to support the new load, ensuring that the bearing continues to function properly. Figure 3 provides a visual representation of these basic principles.

Materials

Hydrodynamic bearings are designed to protect the shaft from damage during machine start-up and shutdown by using a protective layer. This protective layer is typically made of a thick coating of bronze or tin alloy, commonly referred to as Babbitt or white metal. Since the invention of sliding bearings in the 19th century, various forms of white metal have been extensively used. However, despite being a tried and tested material for over a century, its use is not without drawbacks.

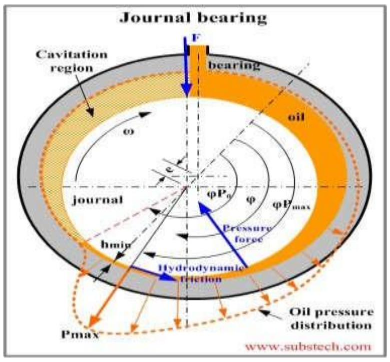


Figure3 Principle of Hydrodynamic Journal Bearing

Journal Bearing Design Parameters
1 The availability of space in the bearing housing is a crucial factor in determining whether a machine designed for thin shell bearing liners can be converted to tilting pad bearings. In most cases, there is insufficient radial room in the bearing housing for such a conversion.
2 Petroff's equation is used to evaluate friction and heat generation in journal bearings, assuming a centered shaft in a plain bushing. This calculation only takes into account the oil shear forces and not the journal load.
3 The specific load, P, is an important concept that refers to the journal load divided by the active bearing length times the diameter.
4 The surface speed of a journal bearing is directly proportional to the journal diameter for a given RPM. The calculation of feet per minute (FPM) from RPM is based on the relationship FPM=RPM (πD/12), where inches are used for D.
5 Dynamic load is caused by the shaft's orbital motion in the oil film clearance space due to non-static forces like imbalance, misalignment, and others.
6 The length-to-diameter (L/D) ratio is one of the first factors a bearing designer considers, as the shaft diameter is often determined by other factors like torque and bending strength.
7 The diametric journal bearing clearance is determined using a universal guideline of 1.5 mils per inch of journal diameter. For instance, a 4-inch diameter shaft would require approximately 6 mils of diametric clearance. It is important to verify if the specifications are for diametric clearance.
8 Steel is typically used as the backing material for Babbitt bearings due to its strength. Cuprous alloys have half the strength of steel and no endurance limit, and bronze or copper may be used for better heat conduction in certain designs. Caution should be exercised with new machinery.
9 Grooves are sometimes cut into the surface of the Babbitt for different reasons, such as directing lubricant to the loaded areas and cooling.
10 The minimum oil film thickness selected by the designer governs the surface finish of a journal bearing, and vice versa. Additionally, the maximum oil film temperature must be considered to ensure proper operation of the bearing.

Mathematical Modeling

Fluid mechanics is a physical science concerned with the behavior of fluid at rest and in motion. It combines the two separate approaches- the empirical hydraulics and the classical hydrodynamics.

1. Reynolds Equation

The differential equation which is developed by making use of the assumptions of hydrodynamic lubrication in equations of motion and continuity equation and combining them into a single equation governing lubricant pressure is called Reynolds Equation. Reynolds equation is given by

$$\frac{\partial}{x}\left(h^3\frac{\partial p}{x}\right) + -\left(h^3 -\right) = 6 \quad \frac{\partial h}{x} \qquad (01)$$

Where,
h – Local oil film thickness
η – Dynamic viscosity of oil
p – Local oil film pressure
U – Linear velocity of journal
z - Longitudinal direction

2. Sommerfeld Equation
Reynolds equation forms the foundation of fluid film lubrication theory. From this equation, relation between the geometry of the surface, relative sliding velocity, the property of the fluids and the magnitude of the normal load can be predicted. In this study, bearing length L over bearing diameter D ratio (L/D) is equal to 0.5. From this value, Sommerfeld number was calculated using equations

$$\square = \left(\frac{r}{C}\right)2\frac{Z'N}{P} \qquad (02)$$

Where, η is viscosity (Pa-s), N is speed (rps), r is journal radius (m), c is clearance (m) and P is radial load per unit of projected bearing area (N). Equation (1.1) was used to obtain the predicted values of eccentricity ratio, friction coefficient, maximum film pressure, position of maximum film pressure, and position of minimum film thickness from Raimondi and Boyd chard.

3. Fluid Friction InJournal Bearing
Friction is known as a resisting force that always opposes motion between mating parts. Friction is the resistance to motion during sliding or rolling when one solid body moves tangentially over another. In the case of journal bearing, fluid friction is generated in the fluid film when pressure induces shear. The mathematical models for predicting viscous shear force on journal and bearing surfaces have been derived. Coefficient of on the bearing surface can be calculated by,

$$F = \frac{C}{r} * \frac{(2+\epsilon^2)(1-\epsilon^2)}{3\epsilon} \qquad (03)$$

Where c is radial clearance, r is journal radius and İ is eccentric ratio In this present study, the torque has been measured and later converted to frictional force and friction coefficient.

4. Pressure Distribution Equation
When so much heat is generated by hydrodynamic action that the normal lubricant flow is insufficient to carry it away, an additional supply of lubricant must be furnished under pressure. The pressure distribution of the hydrodynamic bearing can be calculated by the following equation.[8]

$$P = \left[\frac{20Z'Vde}{C^2}\right]\left\{\frac{(2+e\,\mathrm{Cos}\theta)\sin\theta}{(2+e^2)(1+e\,\mathrm{Cos}\theta)^2\theta}\right\} \qquad (04)$$

where,
Z’= absolute viscosity N-s/mm2
V= surface speed of journal m/min
d= journal diameter in mm

e= eccentricity in mm
θ= circumferential co-ordinates in degree

5. Petroff's Equation

Petroff's Equation is used to determine the coefficient of friction in journal bearing. It is based on the following assumption:

1. The shaft is concentric with the bearing.
2. The bearing is subjected to light load.

Petroff's equation is important because it defines the group of dimensionless parameters that governs the frictional properties of bearing.

A vertical shaft rotating in the bearing is shown in figure 4a

The following notations are used:

r = radius of the journal (mm)
l=length of the bearing (mm)
c=radial clearance (mm)
ns= journal speed(rev/sec)

The velocity at the surface of the journal is given by,

$$U= (2\pi r)ns \qquad (05)$$

Newton's law of viscosity and using Equation

$$P= Z'A\left(\frac{U}{h}\right) \qquad (06)$$

We will apply the above equation for viscous flow through the annular portion between the journal and the bearing in the circumferential direction.

P= tangential frictional force
A= area of journal surface = (2πr)l
U= surface velocity= (2πr)ns
h= distance between journal and bearing surfaces = c

Substituting above values in Equation (6)

$$P = \mu(2\pi rl)(2\pi rns)\left(\frac{1}{C}\right)=\frac{4\ {}^{2}r^{2}lZ'n_s}{C} \qquad (07)$$

The frictional torque is given by,

$$(Mt)f =Pr =\frac{4\ {}^{2}r^{3}lZ'n_s}{C} \qquad (08)$$

Let us consider a radial force (W)' acting on the bearing as shown in fig.(b) The unit bearing pressure (P) acting on the bearing is given by,

$$P= \frac{W}{\text{projeCted area of bearing}}=\frac{W}{2rl} \qquad (09)$$

Or

W = 2pr

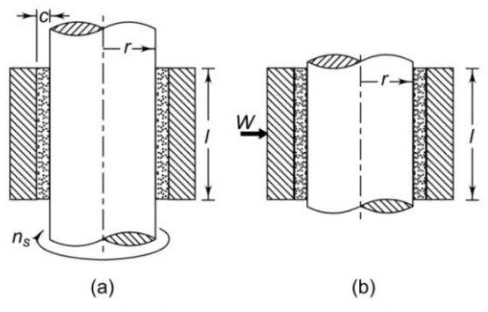


Figure 4Vertical Shaft Rotating In The Bearing

Drawing and Parts ofHydrodynamic Journal Bearing Using CatiaSoftware

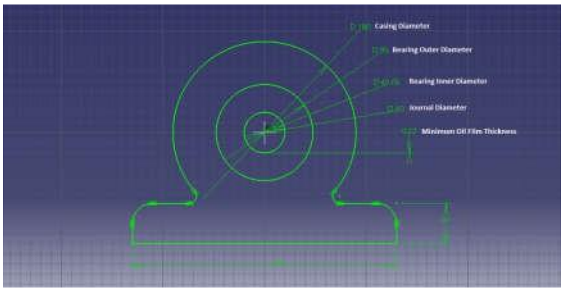


Figure 5 - 2D drawing of Hydrodynamic Journal Bearing

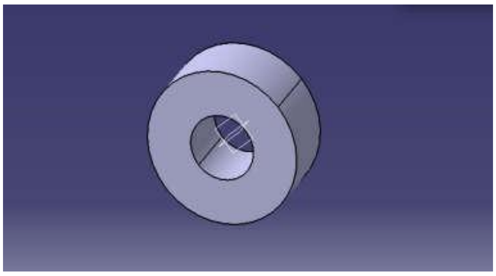


Figure 7-Bearing

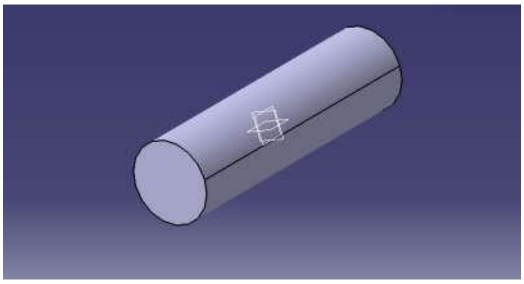


Figure 6- Shaft

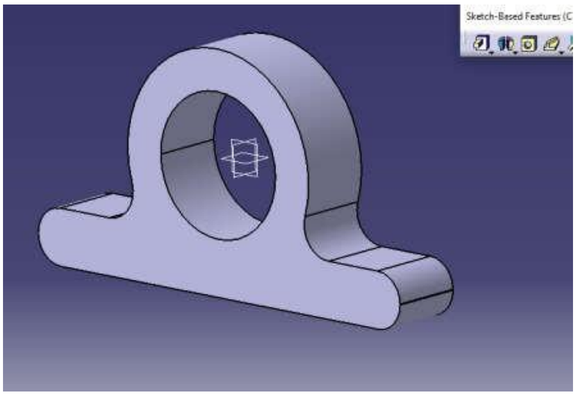


Figure 8- Housing

ANALYSIS OF BEARING USING MATLAB SOFTWARE

- Maximum load – 3000 N
- Loading ratio – 1:5
- Speed range – 40 rpm to 80000 rpm
- Pressure- 5 to 10 bar
- From design data book we selected the dimensions of the bearing as follow:

1. Maximum Load – 3000N
2. Pressure range – 5 to 10 bar
3. Speed Range – 40 to 80000 rpm
4. Shaft diameter = 22(+0.005 -0.0055)mm
5. Bearing inner and outer diameter = 40 & 95mm respectively
6. Radial clearance= 0.0275mm
7. Eccentricity = 0.022mm
8. Eccentricity ratio= 0.8
9. L/D ratio =1
10. C/R ratio= 0.00315

- Bearing material properties

1. Material –Lubricant properties Bronze
2. Density – 8719kg/m^3
3. Young's modulus – 103Gpa
4. Poisson's ratio – 0.34
5. Shear modulus & bulk modulus- 40GPa & 116GPa resp.

- Lubricant properties

1. Lubricant – SAE40

2. Temperature of fluid- 323K
3. Viscosity- 0.08kg/m-s & Density – 84kg/m^3

Thermal conductivity -0.1242 & specific heat – 1901.07j/kgk

1. Petroff's EquationOutput :-

N = 40.000000 is not preferred for p = 2.066000
N = 40.000000 is not preferred for p = 4.132000
N = 40.000000 is not preferred for p = 6.198000

Columns 1 through 15:

40 440 840 1240 1640 2040 2440 2840 3240 3640 4040 4440 4840
5240 5640

Columns 16 through 20:

6040 6440 6840 7240 7640

Columns 1 through 7:

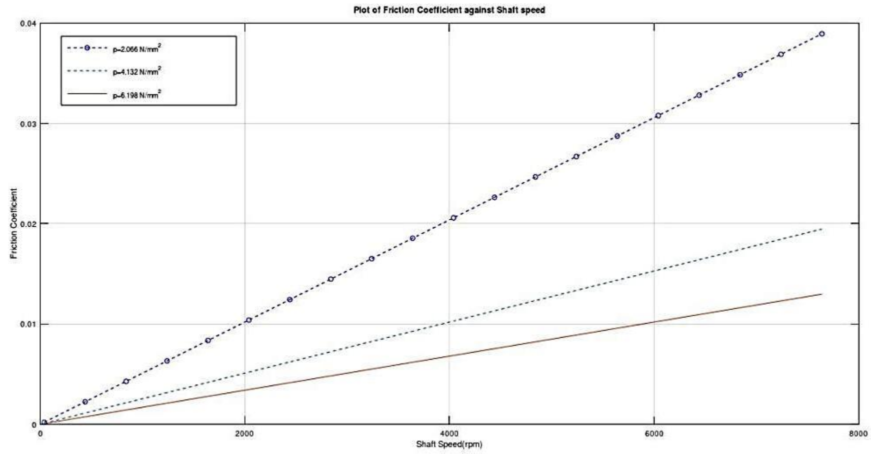
0.000203878 0.002242660 0.004281442 0.006320224 0.008359006 0.010397787
0.012436569
0.000101939 0.001121330 0.002140721 0.003160112 0.004179503 0.005198894
0.006218285
0.000067959 0.000747553 0.001427147 0.002106741 0.002786335 0.003465929
0.004145523

Columns 8 through 14:

0.014475351 0.016514133 0.018552915 0.020591697 0.022630479 0.024669260
0.026708042
0.007237676 0.008257067 0.009276457 0.010295848 0.011315239 0.012334630
0.013354021
0.004825117 0.005504711 0.006184305 0.006863899 0.007543493 0.008223087
0.008902681

Columns 15 through 20:

0.028746824 0.030785606 0.032824388 0.034863170 0.036901952 0.038940733
0.014373412 0.015392803 0.016412194 0.017431585 0.018450976 0.019470367
0.009582275 0.010261869 0.010941463 0.011621057 0.012300651 0.012980244



Graph No. 1– Plot of Friction Coefficient against Shaft speed using MATLAB Software

From the graph, which is Shaft speed vs. friction coefficient we can conclude that:-

- 1. As the shaft speed is increasing, the coefficient of friction also increases.
- 2. As we can see the graph is plotted for 3 different pressures which shows that for higher pressure the value of coefficient of friction for particular speed is also high.

- 3. The friction coefficient increases with the degree of misalignment at lower values of the eccentricity ratio.

2. Sommerfeld EquationOutput :-

For 8.334 rps, Sommerfeld Number is 0.177792

For 16.667 rps, Sommerfeld Number is 0.355563

For 25 rps,Sommerfeld Number is 0.533333

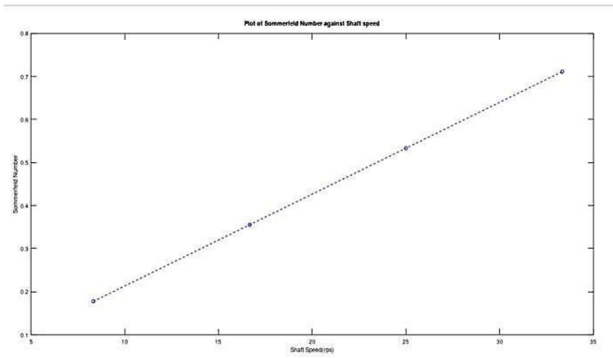
For 33.334 rps, Sommerfeld Number is 0.711125

From the above graph, that is shaft speed vs. Sommerfeld number we can conclude that,

1. As the shaft speed is increasing the Sommerfeld number is also increasing.

2. We can also see that for higher shaft speed the value of Sommerfeld number is also high.

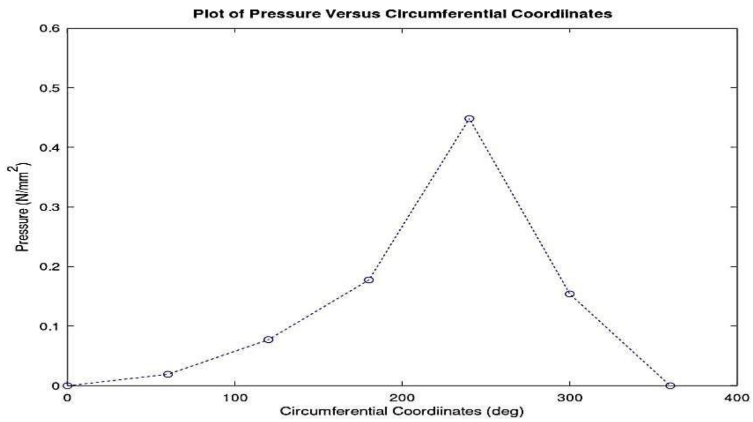
3. Sommerfeld number is a dimensional quantity used extensively in hydrodynamic lubrication. It is very important as it contains all the variables normally specified by the designer.



Graph No 2 -Plot of Sommerfeld Number against Shaft Speed using MATLAB Software

3. Pressure Distribution equationOutput :-

For circumferential coordinate 0, Pressure = 0
For circumferential coordinate 60, Pressure = 0.0192
For circumferential coordinate 120, Pressure = 0.0773
For circumferential coordinate 180, Pressure = 0.1775
For circumferential coordinate 240, Pressure = 0.4484
For circumferential coordinate 300, Pressure = 0.1540
For circumferential coordinate 360, Pressure = 0



Graph No. 3- Plot of Pressure againstCircumferential coordinate using MATLAB Software

From the above graph that is circumferential coordinate vs. pressure we can conclude that,

1. The pressure distribution around the fluid film showed that the pressure variation along the circumference is varying greatly.
2. We can also see that pressure is maximum near the minimum film thickness region.
3. At circumferential coordinate 0 and 360 degrees no pressure is acting in cavitation zone
4. Also as the speed and the load on the bearing increases the pressure also increases.

Case Study- Calculation of Design OfHydrodynamic Journal Bearing ForCentrifugal Pump

Operating Temperature = 150^{O}C

Load on bearing = 12KN

Speed of journal = 1440 rpm

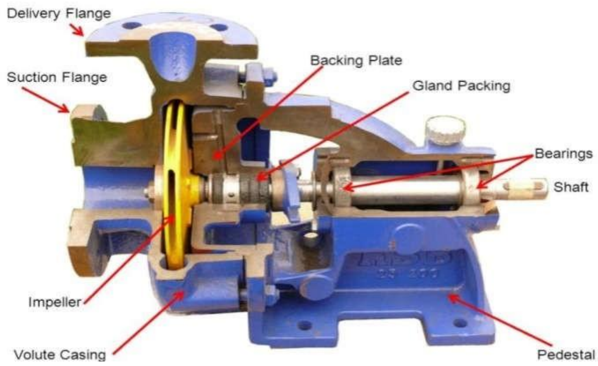


Figure 9- Journal Bearing mounted on Centrifugal Pump

1. Calculation of Diameter of journal and Length :

From Design Data Book Pg. 23.15, Table No. 23.2 the range of permissible bearing pressure in the application of Centrifugal Pump is from 0.69 to 1.37. We will assume permissible bearing pressure as 1.37 MPa.

From Design Data Book Pg. 23.15, Table No. 23.2
1. The value for viscosity is 25 cP.
2. The value of L/D = 1.0 to 2.0

$$P = \frac{W}{\text{projeCted area of bearing}} = \frac{W}{dl} = \frac{}{\text{\^{}2}}$$

d = 94 mm

Diameter of journal = 94 mm

Radius of journal = 47 mm

2. Radial Clearance :

Standard value of Radial clearance is given by ,

c = (0.0013)r = 0.12 mm

3. Radius of Bearing :

R = c + r
= 0.12 + 47 = 47.12 mm.

4. Minimum oil film thickness :

Standard value of Minimum oil film thickness is given by,

h_o = 0.002 x r = 0.094

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5. Viscosity of Lubricant :

Referring to table 16.1 from Design of Machine elements (V B Bhandari)

$\frac{l}{\ } = 1$ and $\frac{ho}{\ } = 0.8$

For the above mentioned values,

S = 0.631

$Ns = \frac{1440}{60} = 24$

$(-)\frac{2Z'NS}{\ }$

$0.631 = (\frac{47}{0.12})^2 \frac{Z\ \ 24}{1.37}$

Z' = 2.3 x 10^-8 N-sec/mm^2**(a)**

From Design Data Book Pg. 23.15, Table No. 23.2 the value for viscosity is 25 cPi.e 2.5 x 10^-8

Z' calculated <Z'given

Hence the value selected is correct.

6. Coefficient of friction :

$f = 2\pi^2(\frac{r}{C})(\frac{Z'ns}{p})$

$f = 2\pi^2(\frac{47}{0.12})(\frac{2.3 \times 10^{-8} \times 24}{1.37})$

f= 0.00379

7. Temperature Rise:

Referring to table 16.1 from Design of Machine elements (V B Bhandari) the coefficient of friction variable (CFV) is 12.8 and flow variable (FV) is 3.59

$$= \frac{8.3\ (\quad)}{\ } = \frac{8.3 \quad 1.37 \quad 12.8}{3.59}$$

Δt = 40.54®C

$Tavg = T + \frac{\Delta t}{2} = 85^0C$

Machinery	Bearing	Maximum pressure, P kgf/mm²	kpsi	MPa	Diameter clearance ratio $\psi = \frac{c}{d}$	Ratio $\frac{L}{d}$	Viscosity, η_1 cP	Viscosity, η Pa s × 10⁻³	Bearing modulus (minimum) $S'' = \frac{\eta_1 n}{P}$ USCSU	$S'' = \frac{\eta n'}{P}$ SI Units, ×10−9
Automobile and aircraft engines	Main	0.56–1.19	0.8–1.7	5.50–11.70	—	0.1–1.8	7	7	15	36.3
	Crankpin	1.06–2.47	1.5–3.5	10.40–24.40		0.7–1.4	to	to	10	24.2
	Wrist pin	1.62–3.62	2.3–5.0	15.00–34.80		1.5–2.2	8	8	8	19.3
Gas and oil engines (four-stroke)	Main	0.49–0.85	0.7–1.2	4.85–8.35	0.001	0.6–2.0	20	20	20	48.4
	Crankpin	0.90–1.27	1.4–1.8	8.80–12.40	<0.001	0.6–1.5	to	to	10	24.2
	Wrist pin	1.27–1.55	1.8–2.2	12.40–15.20	<0.001	1.5–2.0	65	65	5	12.1
Gas and oil engines (two-stroke)	Main	0.35–0.56	0.5–0.8	3.42–5.50	0.001	0.6–2.0	20	20	25	60.4
	Crankpin	0.70–1.06	1.0–1.5	6.85–10.40	<0.001	0.6–1.5	to	to	12	29.0
	Wrist pin	0.85–1.07	1.2–1.8	8.35–12.50	<0.001	1.5–2.0	65	65	10	24.2
Marine steam engines	Main	0.35	0.5	3.42	<0.001	0.7–1.5	30	30	20	48.4
	Crankpin	0.42	0.6	4.14	<0.001	0.7–1.2	40	40	15	36.3
	Wrist pin	1.06	1.5	10.40	<0.001	1.2–1.7	30	30	10	24.2
Stationary, slow-speed steam engines	Main	0.28	0.4	2.75	<0.001	1.0–2.0	60	60	20	48.4
	Crankpin	1.06	1.5	10.40	<0.001	0.9–1.3	80	80	6	14.5
	Wrist pin	1.27	1.8	12.50	<0.001	1.2–1.5	60	60	5	12.1
Stationary, high-speed steam engines	Main	0.17	0.25	1.66	<0.001	1.5–3.0	15	15	25	60.4
	Crankpin	0.42	0.6	4.14	<0.001	0.9–1.5	30	30	6	14.5
	Wrist pin	1.27	1.8	12.50	<0.001	1.3–1.7	25	25	5	12.1
Steam locomotives	Driving axle	0.39	0.55	3.72	0.001	1.6–1.8	100	100	30	72.5
	Crankpin	1.40	2.0	13.70	<0.001	0.7–1.1	40	40	5	12.1
	Wrist pin	2.82	4.0	27.60	<0.001	0.8–1.3	30	30	5	12.1
Reciprocating pumps and compressors	Main	0.17	0.25	1.66	<0.001	1.0–2.2	30	30	30	72.5
	Crankpin	0.42	0.6	4.14	<0.001	0.9–1.7	to	to	20	48.4
	Wrist pin	0.70	1.0	6.85	<0.001	1.5–2.0	80	80	10	24.2
Railway cars	Axle	0.35	0.45	3.42	0.001	1.8–2.0	100	100	50	120.9
Steam turbines	Main	0.07–0.19	0.1–0.275	0.69–1.87	0.001	1.0–2.0	2–16	2–16	100	241.8
Generators, motors, centrifugal pumps	Rotor	0.07-0.14	0.1-0.2	0.69-1.37	0.0013	1.0–2.0	25	25	200	483.5
Gyroscope	Rotor	0.60	0.85	5.90	0.0013	—	30	30	55	133.0
Transmission shafting	Light, fixed	0.08	0.025	0.17	0.001	2.0–3.0	25	25	100	241.8
	Self-aligning	0.106	0.15	1.04	0.001	2.5–4.0	to	to	30	72.5
	Heavy	0.106	0.15	1.04	0.001	2.0–3.0	60	60	30	72.5
Cotton mill	Spindle	0.0007	0.001	0.0069	0.005	—	2	2	10000	24177.5
Machine tools	Main	0.21	0.3	2.06	0.001	1.0–1.4	40	40	40	96.7
Punching and shearing machine	Main	2.82	4.0	27.80	0.001	1.0–2.0	100	100	—	—
	Crankpin	5.62	8.0	55.60	0.001	1.0–2.0	100	100	—	—
Rolling mills	Main	2.11	3.0	20.60	0.0015	1.1–1.5	50	50	10	24.2

Key: $\eta(\eta_1)$ = absolute viscosity, Pa s (cP); n = speed, rpm; n' = speed, rps; P = pressure, N/m² or MPa (psi); MPa = megapascal = 10^6 N/m²; Pa = Pascal = 1 N/m²; 1 psi = 6894.757 Pa; 1 kpsi = 6.89475 MPa; USCSU = US Customary System units.

Table No. 1 – Journal Bearing Design Practices

Dimensionless performance parameters for full journal bearings with side flow

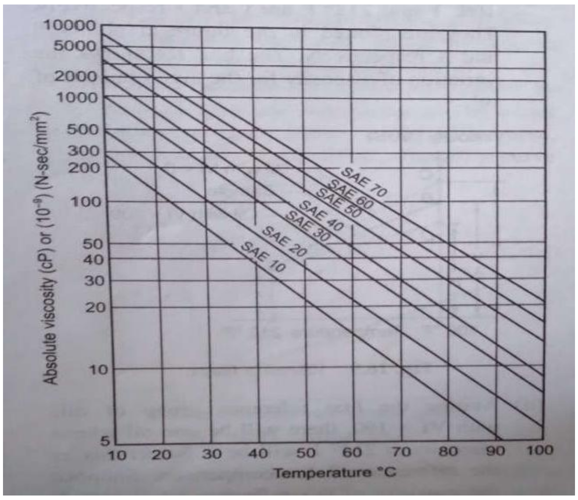
	Values of δ			
L/d ratio	0.25	0.5	1.0	∞
For maximum load	0.27	0.43	0.53	0.66
For minimum friction	0.03	0.12	0.3	0.6

$\frac{L}{d}$	ε	δ	S	ϕ	$\frac{\mu}{\psi}$	$\frac{4Q}{\psi d^2 n' L}$	$\frac{Q_s}{Q}$	$\frac{\gamma c_{sp} T_0}{P}$	$\frac{P}{P_{max}}$
0.25	0	1.0	∞	(89.5)	∞	π	0	∞	—
	0.1	0.9	16.2	82.31	322.0	3.45	0.180	1287.0	0.515
	0.2	0.8	7.57	75.18	153.0	3.76	0.330	611.0	0.489
	0.4	0.6	2.83	60.86	61.1	4.37	0.567	245.0	0.415
	0.6	0.4	1.07	46.72	26.7	4.99	0.746	107.6	0.334
	0.8	0.2	0.261	31.04	8.80	5.60	0.884	35.4	0.240
	0.9	0.1	0.0736	21.85	3.50	5.91	0.945	14.1	0.180
	0.97	0.03	0.0101	12.22	0.922	6.12	0.984	3.73	0.108
	1.0	0	0	0	0	—	1.0	0	0
0.5	0	1.0	∞	(88.5)	∞	π	0	∞	—
	0.1	0.9	4.31	81.62	85.6	3.43	0.173	343.0	0.523
	0.2	0.8	2.03	74.94	40.9	3.72	0.318	164.0	0.506
	0.4	0.6	0.779	61.45	17.0	4.29	0.552	68.6	0.441
	0.6	0.4	0.319	48.14	8.10	4.85	0.730	33.0	0.365
	0.8	0.2	0.0923	33.31	3.26	5.41	0.874	13.4	0.267
	0.9	0.1	0.0313	23.66	1.60	5.69	0.939	6.66	0.206
	0.97	0.03	0.00609	13.75	0.610	5.88	0.980	2.56	0.126
	1.0	0	0	0	0	—	1.0	0	0
1	0	1.0	∞	(85)	∞	π	0	∞	—
	0.1	0.9	1.33	79.5	26.4	3.37	0.150	106	0.540
	0.2	0.8	0.631	74.02	12.8	3.59	0.280	52.1	0.529
	0.4	0.6	0.264	63.10	5.79	3.99	0.497	24.3	0.484
	0.6	0.4	0.121	50.58	3.22	4.33	0.680	14.2	0.415
	0.8	0.2	0.0446	36.24	1.70	4.62	0.842	8.0	0.313
	0.9	0.1	0.0188	26.45	1.05	4.74	0.919	5.16	0.247
	0.97	0.03	0.00474	15.47	0.514	4.82	0.973	2.61	0.152
	1.0	0	0	0	0	—	1.0	0	0
∞	0	1.0	∞	(70.92)	∞	π	0	∞	—
	0.1	0.9	0.240	69.10	4.80	3.03	0	19.9	0.826
	0.2	0.8	0.123	67.26	2.57	2.83	0	11.4	0.814
	0.4	0.6	0.0626	61.94	1.52	2.26	0	8.47	0.764
	0.6	0.4	0.0389	54.31	1.20	1.56	0	9.73	0.667
	0.8	0.2	0.021	42.22	0.961	0.760	0	15.9	0.495
	0.9	0.1	0.0115	31.62	0.756	0.411	0	23.1	0.358
	0.97	0.03	—	—	—	—	0	—	—
	1.0	0	0	0	0	0	0	∞	0

Key: Q_s = flow of lubricant with side flow, cm^3/s; γ = weight per unit volume of lubricant whose specific gravity is 0.90 = 8.83 kN/m^3 (0.0325 lbf/in^3); c_{sp} = specific heat of the lubricant, kJ/NK (Btu/lbf °F) = 0.19 kJ/NK (0.42 Btu/lbf °F); T_0 = difference in temperature, °C.
Source: A. A. Raimondi and J. Boyd, "A Solution for the Finite Journal Bearings and Its Applications to Analysis and Design" ASME, J. Lubrication Technol., Vol. 104, pp. 135–148, April 1982.

Table No 2- Design Performance parameters for full journal Bearings with side flow.

From (a) and (b) it is observed that the lubricating oil should have minimum viscosity of 25Cp AT 85^0C . We will select SAE40 oil from graph 16.8 Viscosity-Temperature Relationship from Design of Machine Elements (V B Bhandari) which will satisfy the minimum viscosity of 25 Cp.



Graph No. 4- Viscosity-Temperature Relationship[2]

Pressure Distribution :

$$P = [\frac{20Z'Vde}{2}] \{\frac{(2+e\ cos\Theta)sin\Theta}{(2+\ ^2)(1+\qquad)^2}\}$$

For 0, Pressure = 0
for 60, Pressure = 0.01652
For 120, Pressure = 0.0773
For 180, Pressure = 0.1875
For 240, Pressure = 0.4584
For 300, Pressure = 0.1570
For 360, Pressure = 0

1. 3-D Modelling of Bearing using NX Software

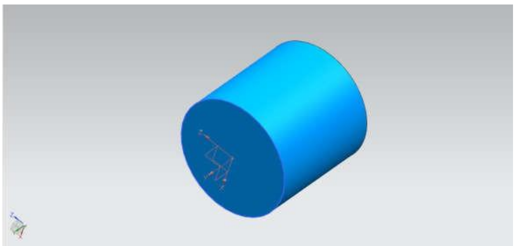


Figure 10-Shaft

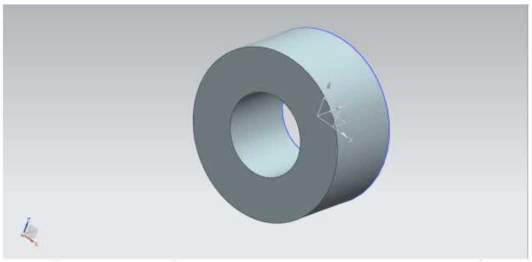


Figure 12-Bearing

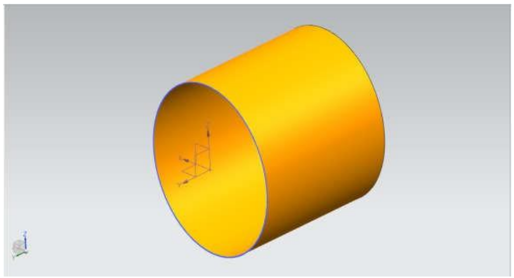


Figure 11-Lubricating Film

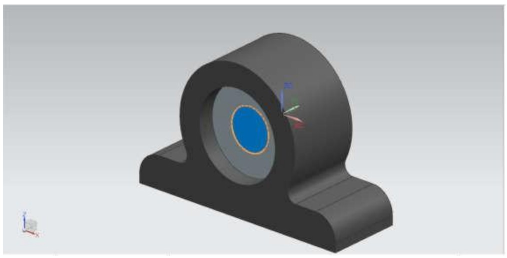


Figure 13-Assembly of Hydrodynamic Journal Bearing

2. Analysis on ANSYS Software

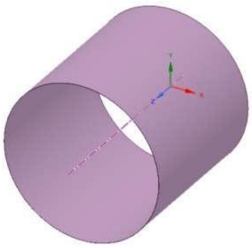


Fig No. 14 Geometry of lubricant film using ANSYS Fluent

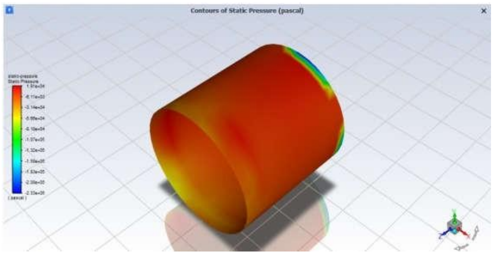


Fig No. 16 Contours of Static pressure using ANSYS Fluent

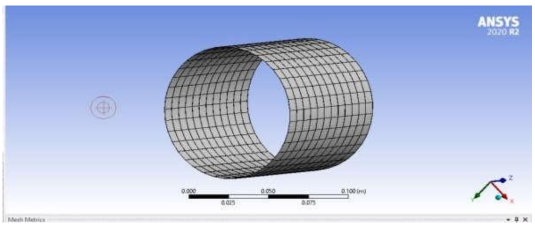


Fig No. 15 Mesh generation on lubricant film using ANSYS Fluent

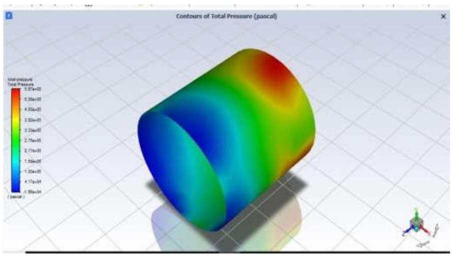


Fig No. 17 Contours of Total Pressure using ANSYS Fluent

3. ANSYS Results

1. The ANSYS showed pressure distribution around the fluid film varied along the circumference greatly. The pressure being maximum near the minimum thickness region. The result can further used to design bearing in such a way that it could bear the stress generated in various regions.
2. Pressure generated is less than 1.47 MPa,hence design is safe to optimum value.
3. Output results of MATLAB software approximately matches with results of Ansys software.

Bearing Failures - Causes and Remedies

Fatigue Failures are not common in general bearings unlike ball bearings. The failures in bearings are mainly associated with insufficient lubricant, contamination of lubricant and faulty assembly. The Principal types of bearing failures are:-

- **Abrasive wear** on the surface of the bearing is the common type of failure. It is in the form of scratches in the direction of motion often with embedded particles. Abrasive wear occurs when lubricating oil is contaminated with dust, rust or spatter. Proper enclosures for the bearing and the housing, cleanliness of lubricating oil and use of high viscosity oil are some of the remedies against this type of wear.
- **Wiping of bearing surface** when the rotating journal touches the bearing, excessive rubbing occurs resulting in the melting and smearing of the surface. This type of failure is in the form of surface melting and flow of bearing material. The main causes for this type of wear are inadequate clearance, excessive transient load, and insufficient oil supply. The remedy is to keep these factors under control.
- **Corrosion** of bearing surface is caused by the chemical attack of reacting agents that are present in the lubricating oil. These oxidation products corrode materials such as lead, copper and zinc. Lead reacts rapidly with all oxidation agents. The remedy is to use oxidation inhibitors as additive in the oil.
- **Distortion** misalignment and incorrect type of fit are the major sources of difficulties in journal bearings. When the fit is too high, Bore distortion occurs. When foreign Particles are trapped between the bearing and housing during the assembly local bore distortion occurs. Correct selection of the fir and proper assembly procedure is the remedy against this type of wear.[2]

Conclusion

The primary goal of the project was to investigate the lubrication mechanism and the effects of various factors on hydrodynamic journal bearings. Based on the findings obtained through the use of MATLAB software and the discussions presented in this report, several conclusions can be drawn:

- As the rotational speed of the shaft increases, the Sommerfeld number also increases.
- The friction coefficient increases with the degree of misalignment, particularly at lower values of the eccentricity ratio.
- The pressure distribution around the fluid film exhibits significant variation along the circumference.
- Increasing the speed and load on the bearing leads to a corresponding increase in pressure.

Future Scope

The following are some potential areas for further research and study regarding hydrodynamic journal bearings:

- Experimental analysis can be conducted to measure the pressure over the bearing surface to better understand the lubrication performance.
- Investigation of different lubrication methods can be carried out to study the impact of lubrication on frictional torque between the bearing and the journal.
- Utilization of different materials for the journal can be explored to evaluate the effects on frictional torque reduction.
- The relationship between temperature and viscosity can be studied to examine the influence of viscosity on frictional torque.

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