STUDY AND ANALYSIS OF DIFFERENT LUBRICANTS IN HYDRODYNAMIC BEARING

A project report submitted in partial fulfillment of the requirements for the award of the degree of

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IN

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(Affiliated to Andhra University, Accredited by NBA & NAAC)

SANGIVALSA, VISAKHAPATNAM-531162

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CERTIFICATE

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It Is A Privilege For Us To Submit Project Report On Study And Analysis Of Different Lubricants In Hydrodynamic Bearing Submitted To Mechanical Engineering Section In Partial Fulfillment For The Award Of Bachelor Of Technology In Mechanical Engineering From Anil Neerukonda Institute Of Technology And Sciences- Sangivalasa.

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ABSTRACT

Lubricants acts as an antifriction media, facilitating smoother working, reducing the risks of undesirable frequent failures and maintaining reliable machine operations among different rotating parts of machines. Today vegetable oils are finding their way into lubricants for industrial and transportation applications. Potential benefits include resource renewability and biodegradability, as well as providing satisfactory performance. Some applications where such oils are finding their niche include two stroke engines, chain saws, hydraulics, mold releases, open gears, and farming, mining and forestry equipment. These oils are also proved to serve as excellent greases and fuels. SAE Oils are widely used in lubrication purposes and they are non-degradable and they cause oil pollution. Vegetable oils are found to potential replacements of these SAE oils especially in lubrication purposes. Vegetable oils are found to have better properties like flash point, fire point, viscosity, better load carrying capacity. Hence it is important to study and analyse the pressure distribution in HYDRO DYNAMIC JOURNAL BEARING with mineral oils and vegetable oils.

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CHAPTER 1

INTRODUCTION

1.1 BEARINGS

Bearing is a mechanical element that permits relative motion between two parts, such as the shaft and the housing with minimum friction, the major functions of the bearings.

1) The bearing ensures free rotation of the shaft or the axle with minimum friction.
2) The bearing supports the shaft or axle and holds it in the correct position.
3) The bearing takes up the forces that act on the shaft or axle and transmits them to frame or foundation.

Bearings are classified in different ways, depending upon the direction of force that acts on them, bearings are classified based on types of friction between the shaft and bearing surface, they are

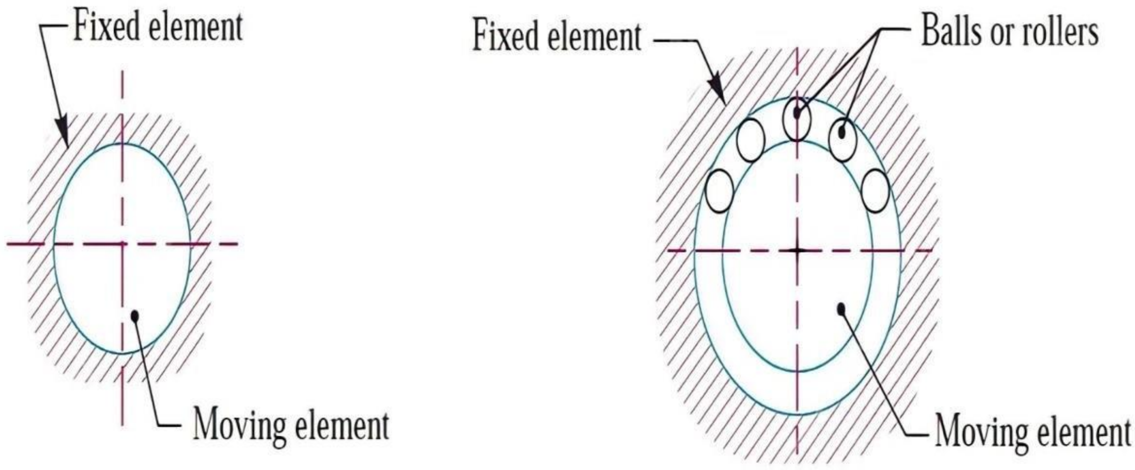
a) Sliding contact bearings
b) Rolling contact bearings

Sliding contact bearings are also called Plain contact bearings or journal bearings. In this case the surface of the shaft slides over the surface of bush resulting in wear and friction.

Rolling contact bearings are also called Antifriction bearings or Ball bearings. Rolling elements such as balls or rollers are introduced between the surfaces that are in relative motion.

Sliding contact bearings are used in the following applications:

1) crankshaft bearings in petrol and diesel engines;
2) centrifugal pumps;
3) large size electric motors;
4) steam and gas turbines and
5) concrete mixers, rope conveyors and marine installations.



(A) **(B)**

Figure 1.1 Sliding contact bearing (A) and Rolling contact bearing (B)

1.2 BASIC MODES OF LUBRICATION

Lubrication is the science of reducing the friction by application of suitable substance called lubricant, between the rubbing surfaces of bodies having relative motion.

The objectives of lubrication are:

a) To reduce friction
b) To reduce or prevent wear
c) To carry away the heat generated due to friction
d) To prevent the journal and bearing from corrosion.

The lubricants are classified into the following groups:

a) Liquid lubricants like mineral or vegetable oils
b) Semi-solid lubricants like grease
c) Solid lubricants like graphite

1.3 HYDRODYNAMIC LUBRICATION

Hydrodynamic lubrication is defined as the system of lubrication in which the load supporting fluid film is created by the shape and relative motion of lubrication which is shown in figure 1.2, initially the shaft is at rest (a) and it sinks to the bottom of the clearance space under the action of load W. The surfaces of the journal and bearing touch during 'rest'. As the journal starts to rotate, it climbs the bearing surface (b) and as the speed is further increased, it forces the fluid into the wedge-shaped region (c). Since more and more fluid is forced into the wedge-shaped clearance space, pressure is generated within the system. The pressure distribution around the periphery of the journal is shown in Fig. 1.2. Since the pressure is created within the system due to rotation of the shaft, this type of bearing is known as self-acting bearing. The pressure generated in the clearance space supports the external load (W)..

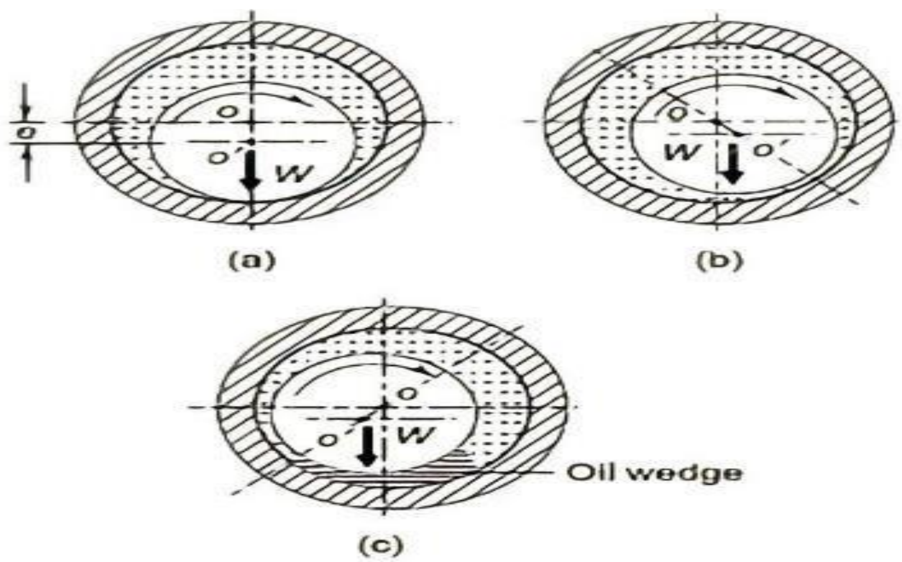


Figure 1.2 Hydrodynamic bearings

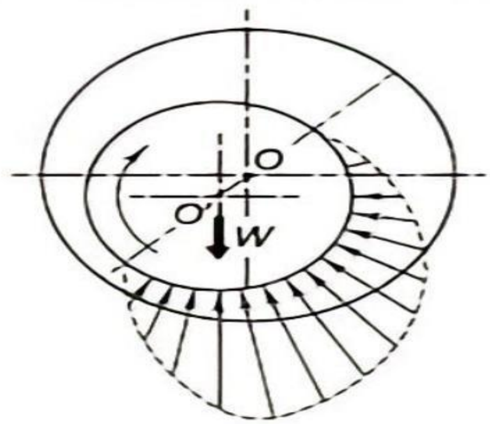


Figure 1.3 pressure distribution

In this case, it is not necessary to supply the lubricant under pressure and the only requirement is sufficient and continuous supply of the lubricant. This mode of lubrication is seen in bearings mounted on engines and centrifugal pumps. Frequently, a term 'journal bearing is used. A journal bearing is a sliding contact bearing working on hydrodynamic lubrication and which supports the load in radial direction. The portion of the shaft inside the bearing is called journal and hence the name 'journal' bearing.

There are two types of hydrodynamic journal bearings, namely, full journal bearing and partial bearing. In full journal bearing, the angle of contact of the bushing with the journal is 360°. Full journal bearing can take loads in any radial direction.

1.3.1 VISCOSITY

Viscosity is defined as the internal frictional resistance offered by a fluid to change its shape or relative motion of its parts. An oil film placed between two parallel plates is shown in Fig. 1.4. The lower plate is stationary while the upper plate is moved with a velocity U by means of a force P.

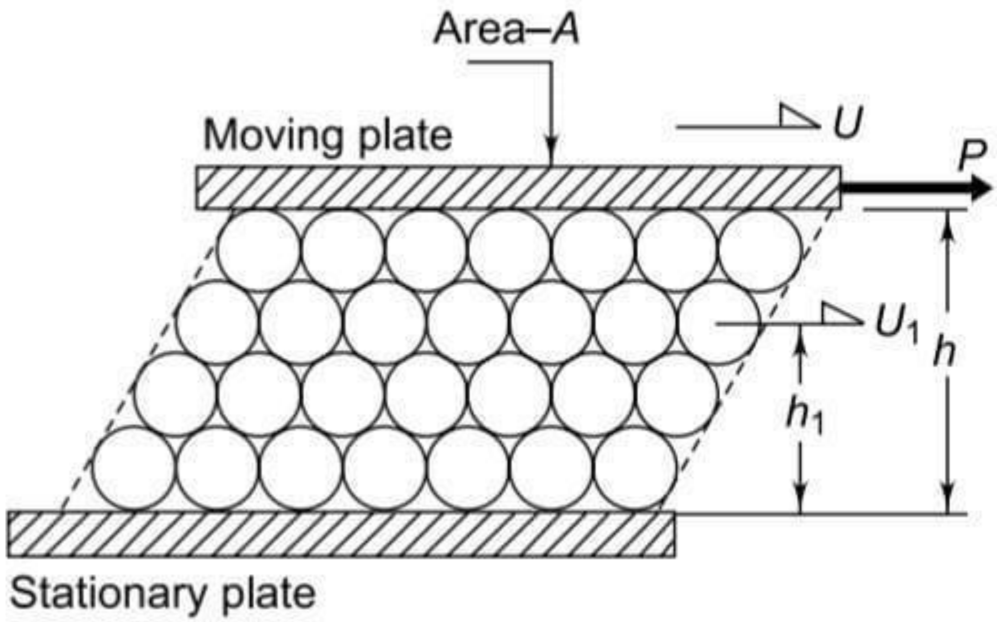


Figure 1.4 Newton's law of viscosity

1.4 PETROFF'S EQUATION

Petroff's equation is used to determine the
coefficient of friction in journal bearings. It is based on the following assumptions:

(i) The shaft is concentric with the bearing.

(ii) The bearing is subjected to light load. In practice, such conditions do not exist.. However, Petroff's equation is important because it defines the group of dimensionless parameters that govern the frictional properties of the bearing.

For a vertical shaft rotating in a bearing the following notations are used

r = radius of journal(mm)

l= length of the bearing(mm)

c= radial clearance(mm)

$n_s =$ speed of bearing (rev/sec)

The velocity at the surface of journal is given by

U= $(2\pi r)n_s$

P= tangential frictional force

A= area of journal surface = $(2\pi r)l$

U= surface velocity $(2\pi r)n_s$

Let us consider the radial force(W) acting on the bearing , the unit bearing pressure is given by

W= 2prl

The Petroff's equation is given by

$$f = (2\pi^2)\left(\frac{r}{c}\right)\left(\frac{\mu n_s}{p}\right)$$

Petroff's equation indicates that there are two important dimensionless parameters, namely, $\left(\frac{r}{c}\right)$ and that govern the coefficient of friction$\left(\frac{\mu n_s}{p}\right)$ and other frictional properties like frictional torque, frictional power loss and temperature rise in the bearing.

1.4.1 MCKEE'S INVESTIGATION

In hydrodynamic bearings, initially the journal. is at rest. There is no relative motion and no hydrodynamic film. Therefore, there is metal to metal contact between the surfaces of the journal and the bearing. As the journal starts to rotate, it takes some time for the hydrodynamic film to build sufficient pressure in the clearance space. During this period, there is partial metal to metal contact and a partial lubricant film. This is thin film lubrication. As the speed is increased, more and more lubricant is forced into the wedge-shaped clearance space and sufficient pressure is built up, separating the surfaces of the journal and the bearing. This is thick film lubrication. Therefore, there is a transition from thin film lubrication to thick film lubrication as the speed increases.

The transition from thin film lubrication to thick film hydrodynamic lubrication can be better visualized by means of a curve called μN/p curve. This curve is shown in Fig. 16.11. The μN/p curve is an experimental curve developed by McKee brothers.

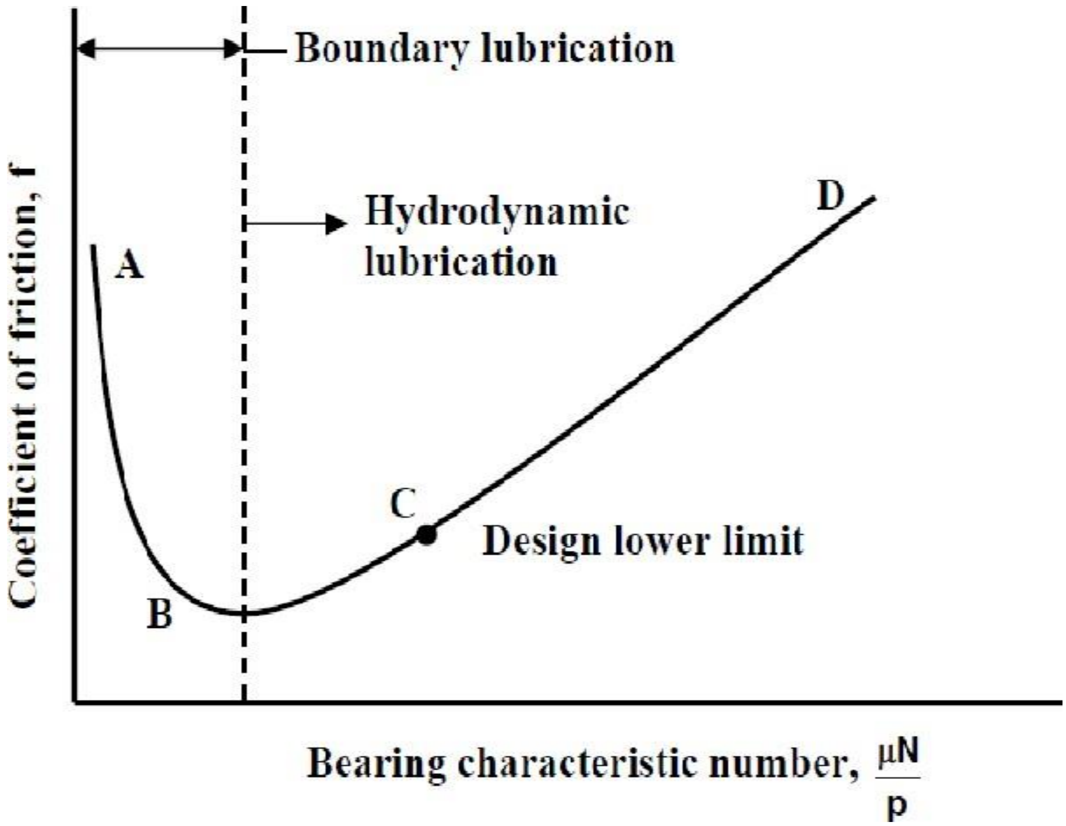


Figure 1.5 McKee curve

The bearing characteristic number is plotted on the abscissa. The coefficient of friction / is plotted on the ordinate. The coefficient of friction f is the ratio of tangential frictional force to the radial load acting on the bearing. As seen in Fig. 1.5, there are two distinct parts of the curve-BC and CD.

(i) In the region BC, there is partial metal to metal contact and partial patches of lubricant. This is the condition of thin film or boundary lubrication.

(ii) In the region CD, there is relatively thick film of lubricant and hydrodynamic lubrication takes place.

(iii) AC is the dividing line between these two modes of lubrication. The region to the left of the line AC is the thin film zone while the region to the right of the line AC is the thick film zone.

(iv) It is observed that the coefficient of friction is minimum at C or at the transition between these two modes. The value of the bearing characteristic number corresponding to this minimum coefficient is called the bearing modulus. It is denoted by K in the figure.

The bearing should not be operated near the critical value K at the point C. A slight drop in the speed (N) or a slight increase in the load (p) will reduce the value of $\mu N/p$ resulting in boundary lubrication. The guidelines for hydrodynamic lubrication are as follows:

(i) In order to avoid seizure, the operating value of the bearing characteristic number ($\mu N/p$) should be at least 5 to 6 times that when the coefficient of friction is minimum. (5 K to 6 K or 5 to 6 times the bearing modulus).

If the bearing is subjected to fluctuating loads or impact conditions, the operating value of the bearing characteristic number ($\mu N/p$) should be at least 15 times that when the coefficient of friction is minimum. (15 K or 15 times the bearing modulus).

It is observed from the ($\mu N/p$) curve that when viscosity of the lubricant is very low, the value. of ($\mu N/p$) parameter will be low and boundary lubrication will result. Therefore, if the viscosity of the lubricant is very low then the lubricant will not separate the surfaces of the journal and the bearing and metal to metal contact will occur resulting in excessive wear at the contacting surfaces.

The ($\mu N/p$) curve is important because it defines the stability of hydrodynamic journal bearings and helps to visualize the transition from boundary lubrication to thick film lubrication.

CHAPTER 2

LITERATURE REVIEW

Hughes and Osterle et al. have investigated the relation between viscosity as a function of temperature and pressure of the lubricant inside the journal bearing for adiabatic conditions.

Basri and Gethin et al. have investigated the thermal aspects of various non circular journal bearing using adiabatic model.

Gertzos et al. have investigated journal bearing performance with a Non-Newtonian fluid i.e. Bingham fluid considering the thermal effect.

Huiping Liu et al. studied hydrodynamic journal bearings with elastic insert and found that the elastic deformation of the bearing had a significant influence on the rotor–bearing system, particularly for the polymeric-based materials.

Nabhan et al. solved Navier-Stokes equation with the aid of Simpson rule and calculated the pressures, drags and load carrying capacities by taking binary fluid mixture with different viscosity ratio.

Hassan E. Rasheed et al. theoretically presented the effects of circumferential, axial and combined surface waviness on the performance of the hydrodynamic journal bearings by using Reynolds equation for Newtonian iso viscous lubricant. It was observed that when waviness number is approximately below nine, then circumferential waviness increases the load carrying capacity and decreases the friction variable. But the axial waviness is to always have an opposite effect on the load carrying capacity and friction variable.

S.k.guha et al. analized the effect of isotropic roughness on the steady-state characteristics of hydrodynamic journal bearings terms of load capacity, attitude angle, end leakage flow rate, misalignment moment and friction coefficient are estimated for different values of roughness parameter, eccentricity ratio and degree of misalignment at unit slenderness ratio. Finite difference method is also used to measure steady-state oil film pressures by using Reynolds equation.

NabarunBiswas and PrasunChakraborti et al. used physical properties of SAE-oil(20w40) lubricant for analysis purpose in journal bearing. They involve with six time steps 10, 30, 50, 70, 90, and 110 sec for unsteady analysis and found out that after 110 sec the flow becomes steady. It was also observed that maximum pressure is observed at minimum oil film thickness with increasing value of roughness.

Byoung-Hoo Rho et al. investigated acoustical properties of hydrodynamic journal bearing. The universal Reynolds equation is solved at each step of time using the finite difference method and the nonlinear transient motion of the journal centre is obtained by numerical integration of its acceleration using fourth order Runge-Kutta method.

Byoung-Hoo Rho et al. investigated the effects of design parameters on the noise of rotor-bearing system supported by oil lubricated journal bearing. The Reynolds equation for finite width bearing under unsteady condition is applied for calculating pressure. It was observed that the radial clearance, mass eccentricity of the rotor and the width of the bearing considerably affect the A-weighted sound pressure level of the bearing. .

Ron A.J. Van Ostayen et al. presented a mathematical optimization procedure to find the optimal film height distribution for a hydrodynamic bearing. Firstly this methodology is applied for a bearing with constant load and sliding speed. Then subsequently applied for a bearing with periodic load and sliding speed. Slider bearings with different shapes, loads and speeds are analyzed by new heuristic load optimization procedure along with Reynolds equation and found more efficient than general purpose optimization routine.

K. M. Panday et al. analysis thin film lubricated journal bearing with different L/D ratios such as 0.25, 0.5, 1, 1.5, and 2. It was observed that maximum pressure present at minimum oil film thickness. Also reported that shear stress is reduces on bearing and journal surface with increase in L/D ratio whereas turbulent viscosity of lubricant rises with increase in L/D ratio.

NacerTala-Ighil et al. developed a numerical model based on finite difference method by using Reynolds equation to study the cylindrical textures shape effect on the performance of hydrodynamic journal bearing. Based on geometric arrangement of textures on the bearing surface, a comparison of considered twenty five cases is conducted. It was found that the minimum oil film thickness increased approximately by 1.8% and friction torque is decreased approximately by 1.0%.

Peeyush vats et al. presented thermal analysis of journal bearing by using FEM analysis. Parameters like heat generated, temperature distribution and heat dissipation are studied. From results it is reported that difference between heat dissipated and heat generated in oil film was very large, which causes increase in temperature of the bearing and damaged the bearing pads.

CHAPTER 3

EXPERIMENTAL METHODOLOGY

3.1 INTRODUCTION TO EXPERIMENTAL SETUP

Journal Bearing Apparatus is designed on the bearing action used in practice. To formulate the bearing action accurately in mathematical terms is a more complex job. However, one can visualize the pattern of bearing pressure distribution due to the hydrodynamic action with the help of experimental rig. This helps to understand the subject properly. The experimental rig consists of a small journal bearing. This apparatus helps to demonstrate and study the effect of important variables such as speed, viscosity and load, on the pressure distribution in a Journal Bearing.

3.2 DESCRIPTION OF APPARATUS

The apparatus is illustrated in fig. It consists of a Brass bearing mounted freely on steel Journal shaft (A). This journal shaft is fixed directly on to a motor shaft (S). A Dimmer stat finely controls the speed of the DC motor. The Journal bearing has twelve (No.1 to 12) equispaced of 30° pressure tapings around its circumference, and two No, 13,14 additional axial pressure tapings are positioned on the topside of the journal bearing. The two sides of bearing are closed with two MS plates and sealed with gasket packing to avoid leakage. Balancing weights are provided to maintain the bearing in horizontal position while taking the readings. Both the weights can be adjusted freely along the rod. Oil film pressures are indicated in 14-tubes manometer frame and readings directly in head of oil. Clear flexible tubes are fixed on the manometer frame and connected to the tapings spaced around bearing and thus permit the bearing to turn freely. The oil reservoir can be adjusted at required height and is connected to the bearing by a flexible plastic tube. From this reservoir oil enters the bearing through this plastic tube.

Figure 3.1 Journal bearing apparatus

3.3 SPECIFICATIONS OF APPARTUS

1. Diameter of Journal = 2r = 52.5 mm.
2. Diameter of bearing = 2R=50mm (with 12 radial tapings and 2 axial tapings).
3. Bearing width (L) = 90mm.
4. Motor speed = 800 -1000rpm (variable speed – DC).
5. Manometer frame with 14 tubes of 240cm. Height with scales and adjustable oil supply tank.
6. Lubricating oil as per the test (SAE oil (OR) vegetable oil).
7. Supply required AC single phase 230v.50Hz stabilized.
8. r = Radius of Journal.
9. Radial clearance (R-r) = 0.0275mm

3.4 THEORY

The mathematical analysis of the behavior of a journal in a bearing fall into two distinct categories:

1. Hydrodynamics of fluid flow between plates.
2. Journal bearing analysis where the motion of the journal in the oil films is considered. According to the equation the Somerfield pressure function (when the velocity of the eccentricity and the whirl speed of the journal are both zero) is given by: (with seepage)

   So =(r/c) ^2 (μn/p) kw*10^-6

when KW correction factor for side leakage from graph. Where 'p' is the pressure of the oil film at the point measured anticlockwise from the line of common centers (00')

3.5 EXPERIMENTAL PROCEDURE:

1. Fill the oil tank by using SAE 15W40/SAE 20W40 / SAE10W30 / Rice bran oil / Canola oil under test and position the tank at the desired height (up to 1.5-liter oil).
2. Drain out the air from the tubes on the manometer by removing the tubes from manometer.
3. Check that some oil sea page is there (See page of oil is necessary for cooling purpose).
4. Check the direction of rotation and increase the speed of the motor slowly.
5. Set the speed and let the journal run for about 2 minutes until the oil in the bearing is warmed up and check the steady oil levels at various tapings.
6. Add the required loads and adjust the balancing weights, on the rod to maintain the horizontal levels position.
7. When the manometer levels are settled down, take the pressure readings on 1-14 manometer tubes. For circumferential and axial pressure distribution.
8. Repeat the experiment for various speed and loads.
9. After the test is over set dimmer to zero position and switch off the main supply.
10. Keep the oil tank at lower most position so that there will be no leakage in the idle period.

CHAPTER 4

RESULTS AND ANALYSIS

4.1 OBSERVATIONS

The readings obtained for **SAE 15W40** at three different speeds(500,700,900 rpm) are:

OIL : 15W40(SAE)				OIL: 15W40(SAE)				OIL: 15W40(SAE)			
SPEED : 500 rpm				SPEED: 700 rpm				SPEED:900 rpm			
Static head(Po) : 101 cm				Static head(Po):101cm				Static head(Po):101cm			
Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm	
Radial pressure distribution				Radial pressure distribution				Radial pressure distribution			
1	129	28		1	130	29		1	132.8	31.8	
2	114	13		2	112.5	11.5		2	116.3	15.3	
3	103.6	2.6		3	100.8	-0.2		3	104.9	3.9	
4	97.2	-3.8		4	95.2	-5.8		4	97.7	-3.3	
5	90.1	-10.9		5	84.8	-16.2		5	87.3	-13.7	
6	84.8	-16.2		6	79.2	-21.8		6	80.7	-20.3	
7	76.9	-24.1		7	74.1	-26.9		7	75.6	-25.4	
8	65.2	-35.8		8	63.2	-37.8		8	64.2	-36.8	
9	56.3	-44.7		9	55.3	-45.7		9	56.3	-44.7	
10	55.3	-45.7		10	97.2	-3.5		10	122.6	21.6	
11	125.9	24.9		11	150.8	49.8		11	148.5	47.5	
12	146.8	45.8		12	149.6	48.6		12	148	47	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	137.6	36.6		A	140.4	39.4		A	140.9	39.9	
B	149.8	45.8		B	149.6	48.6		B	148.3	47.3	
12	146.8	45.8		12	149.6	48.6		12	148	47	
C	138.9	37.9		C	141.7	40.7		C	140.7	39.7	
D	121.1	20.1		D	129.5	28.5		D	132	31	

Table 4.1 SAE 15W40 readings

The readings obtained for **SAE 20W40** for three different speeds (500,700,900 rpm) are:

OIL: 20W40(SAE)				OIL: 20W40(SAE)				OIL:20W40(SAE)			
SPEED : 500 rpm				SPEED: 700 rpm				SPEED:900 rpm			
Static head(Po): 103.5 cm				Static head(Po):103.5 cm				Static head(Po):103.5cm			
Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm	
Radial pressure distribution				Radial pressure distribution				Radial pressure distribution			
1	126.7	23.2		1	127.2	23.7		1	130	26.5	
2	111.2	7.7		2	110.2	6.7		2	114	10.5	
3	100.3	-3.2		3	98	-5.5		3	101.8	-1.7	
4	93.4	-10.1		4	89.9	-13.6		4	92.7	-10.8	
5	86.8	-16.7		5	81.5	-22		5	84	-19.5	
6	80.2	-23.3		6	73.6	-30.2		6	75.1	-28.4	
7	73.1	-30.4		7	69	-34.5		7	70.6	-32.9	
8	70.8	-32.7		8	68.5	-35		8	69.5	-34	
9	70.6	-32.9		9	69.5	-34		9	71.3	-32.2	
10	61.2	-42.3		10	103.8	0.3		10	129.2	25.7	
11	147.3	43.8		11	172.4	68.9		11	170.1	66.6	
12	150.3	46.8		12	152.9	49.4		12	151.3	47.8	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	140.9	37.4		A	143.7	40.2		A	144.2	40.7	
B	152.9	49.4		B	152.9	49.4		B	151.3	47.8	
12	150.3	46.8		12	152.9	49.4		12	151.3	47.8	
C	142.4	38.9		C	147	43.5		C	146.3	42.8	
D	124.7	21.2		D	132	28.5		D	134.3	30.8	

Table 4.2 SAE 20W40 readings

The readings obtained for **SAE 10W30** at three different speeds (500,700,900 rpm) are :

OIL:10W30				OIL:10W30				OIL:10W30			
SPEED:500 rpm				SPEED: 700rpm				SPEED:900rpm			
Static head(Po):106.5cm				Static head(Po): 106.5cm				Static head(Po): 106.5cm			
Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm	
Radial pressure distribution				Radial pressure distribution				Radial pressure distribution			
1	124.5	18		1	131.5	25		1	128.5	22	
2	112.5	6		2	116	9.5		2	114	7.5	
3	105.5	-1		3	105.5	-1		3	105.5	-1	
4	102.5	-4		4	101	-5.5		4	100.5	-6	
5	96	-10.5		5	95.5	-11		5	93.5	-13	
6	93	-13.5		6	91.5	-15		6	89	-17.5	
7	88.5	-18.5		7	85.5	-21		7	83.5	-23	
8	79.5	-27		8	76.5	-30		8	75	-31.5	
9	69.5	-37		9	58.5	-48		9	58	-48.5	
10	61	-45.2		10	62.5	-44		10	55	-51.5	
11	102.5	-4		11	112.5	6		11	129.5	23	
12	134	27.5		12	142.5	36		12	143.5	37	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	127	20.5		A	141.5	35		A	132	25.5	
B	137.5	31		B	143	36.5		B	143.5	37	
12	134	27.5		12	142.5	36		12	143.5	37	
C	130	23.5		C	136.5	30		C	136.5	30	
D	112.5	6		D	120.5	14		D	122.5	16	

Table 4.3 SAE 10W30 readings

The readings obtained for **RICE BRAN OIL** at three different speeds (500,700,900 rpm) are:

OIL: RICE BRAN OIL				OIL: RICE BRAN OIL				OIL: RICE BRAN OIL			
SPEED:500 rpm				SPEED:700 rpm				SPEED:900 rpm			
Static head(Po): 104.6cm				Static head(Po): 104.6cm				Static head(Po): 104.6cm			
Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm	
Radial pressure distribution				Radial pressure distribution				Radial pressure distribution			
1	121.9	17.3		1	124.7	20.1		1	128.2	23.9	
2	110.4	5.8		2	111.2	6.6		2	112.5	7.9	
3	103.3	-1.3		3	103.3	-1.3		3	103.6	-1	
4	99.8	-4.8		4	99	-5.6		4	98.8	-5.8	
5	97.5	-7.1		5	96	-8.5		5	94.9	-9.7	
6	95.2	-9.4		6	93.4	-11.2		6	91.9	-12.7	
7	90.9	-13.7		7	89.9	-14.7		7	88.6	-16	
8	88.6	-16		8	84.8	-19.8		8	81.5	-23.1	
9	81.2	-23.4		9	73.6	-31		9	68	-36.6	
10	60.7	-43.9		10	64	-40.6		10	56.3	-48.3	
11	63.2	-41.4		11	105.6	1		11	124.9	20.3	
12	140.7	36.1		12	145	40.4		12	149.6	45	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	132.5	27.9		A	135.1	30.5		A	138.4	33.8	
B	156.2	51.6		B	152.9	48.3		B	155.4	50.8	
12	140.7	36.1		12	145	40.4		12	149.6	45	
C	118.3	13.7		C	130.5	25.9		C	135.6	31	
D	107.9	3.3		D	114	9.4		D	118.1	13.5	

Table 4.4 Rice bran oil readings

The readings obtained for **CANOLA OIL** at three different speeds (500,700,900 rpm) are:

OIL: CANOLA OIL				OIL: CANOLA OIL				OIL: CANOLA OIL			
SPEED: 500 rpm				SPEED: 700 rpm				SPEED: 900 rpm			
Static head(Po): 104.1 cm				Static head(Po): 104.1 cm				Static head(Po): 104.1 cm			
Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm		Tube no	P cm	P-Po cm	
Radial pressure distribution				Radial pressure distribution				Radial pressure distribution			
1	120.6	16.5		1	125.4	21.3		1	124.9	20.8	
2	107.9	3.8		2	110.4	6.3		2	109.7	5.6	
3	101.6	-2.5		3	102.8	-1.3		3	101.6	-2.5	
4	98.8	-5.3		4	98.8	-5.3		4	97	-7.1	
5	96.5	-7.6		5	96	-8.1		5	94.4	-9.7	
6	94.4	-9.7		6	93.7	-10.4		6	91.6	-12.5	
7	92.7	-11.4		7	91.4	-12.7		7	89.4	-14.7	
8	89.4	-14.7		8	86.8	-17.3		8	81.2	-22.9	
9	83.5	-20.6		9	77.4	26.7		9	71.1	-33	
10	61.4	-42.7		10	45.4	-58.7		10	22.6	-81.5	
11	45.4	-58.7		11	77.4	-26.7		11	148.5	44.4	
12	147.3	42.7		12	152.1	48		12	151.8	47.7	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	142.7	38.6		A	144.5	40.4		A	142.2	38.1	
B	175	70.9		B	170.1	66		B	160.5	56.4	
12	147.3	43.2		12	152.1	48		12	151.8	47.7	
C	120.6	16.5		C	129	24.9		C	135.6	31.5	
D	105.6	1.5		D	110.7	6.6		D	118.1	14	

Table 4.5 Canola oil readings

4.2 ANALYTICAL CALCULATIONS

4.2.1 Analytical calculations of SAE 15W40

(a) speed in rpm = 500

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 500/60 = 8.33 rps

Viscosity of oil(μ) = 155cP = 155e-9 N-s/mm^2

Sommerfeld number is given by

$$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{500} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{155}{10^9} * \frac{8.33}{0.15}$$

S_{500} = 8.624

(b) speed in rpm = 700

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 700/60 = 11.33 rps

Viscosity of oil(μ) = 155cP = 155e-9 N-s/mm^2

Sommerfeld number is given by

$S = (\frac{r}{c})^2 * (\frac{\mu * ns}{p})$

$S_{700} = (\frac{27.5}{0.00275})^2 * \frac{155}{10^9} * \frac{11.33}{0.15}$

S_{700} = 12

(c) speed in rpm = 900

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 500/60 = 15 rps

Viscosity of oil(μ) = 155cP = 155e-9 N-s/mm^2

Sommerfeld number is given by

$S = (\frac{r}{c})^2 * (\frac{\mu * ns}{p})$

$S_{900} = (\frac{27.5}{0.00275})^2 * \frac{155}{10^9} * \frac{15}{0.15}$

S_{900} = 15.5

The Sommerfeld number obtained for SAE 15W40 at 500 rpm is 8.624.

The Sommerfeld number obtained for SAE 15W40 at 700 rpm is 12.

The Sommerfeld number obtained for SAE 15W40 at 900 rpm is 15.5.

4.2.2 Analytical calculations of SAE 20W40

(a) speed in rpm = 500

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 500/60 = 8.33 rps

Viscosity of oil(μ) = 120cP = 120e-9 N-s/mm^2

Sommerfeld number is given by

$$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{500} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{120}{10^9} * \frac{8.33}{0.15}$$

S_{500} = 6.664

(b) speed in rpm = 700

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 700/60 = 11.33 rps

Viscosity of oil(μ) = 120cP = 120e-9 N-s/mm^2

Sommerfeld number is given by

$$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{700} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{120}{10^9} * \frac{11.33}{0.15}$$

S_{700} = 9.328

(c) speed in rpm = 900

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 900/60 = 15 rps

Viscosity of oil(μ) = 50cP = 50e-9 N-s/mm^2

Sommerfeld number is given by

$$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{900} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{120}{10^9} * \frac{15}{0.15}$$

S_{900} = 12

The Sommerfeld number obtained for SAE 20W40 at 500 rpm is 6.664.

The Sommerfeld number obtained for SAE 20W40 at 700 rpm is 9.328.

The Sommerfeld number obtained for SAE 20W40 at 900 rpm is 12.

4.2.3. Analytical calculations of SAE 10W30

(a) speed in rpm = 500

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 500/60 = 8.33 rps

Viscosity of oil(μ) = 170cP = 170e-9 N-s/mm^2

Sommerfeld number is given by

$$S= \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{500} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{170}{10^9} * \frac{8.33}{0.15}$$

S_{500} = 9.44

(b) speed in rpm = 700

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 700/60 = 11.33 rps

Viscosity of oil(μ) = 170cP = 170e-9 N-s/mm^2

Sommerfeld number is given by

$S = (\frac{r}{c})^2 * (\frac{\mu * ns}{p})$

$S_{700} = (\frac{27.5}{0.00275})^2 * \frac{170}{10^9} * \frac{11.33}{0.15}$

S_{700} = 13.21

(c) speed in rpm = 900

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 900/60 = 15 rps

Viscosity of oil(µ) = 170cP = 170e-9 N-s/mm^2

Sommerfeld number is given by

$S = (\frac{r}{c})^2 * (\frac{\mu * ns}{p})$

$S_{900} = (\frac{27.5}{0.00275})^2 * \frac{170}{10^9} * \frac{15}{0.15}$

S_{900} = 17

The Sommerfeld number obtained for SAE 10W30 at 500 rpm is 9.44

The Sommerfeld number obtained for SAE 10W30 at 700 rpm is 13.21

The Sommerfeld number obtained for SAE 10W30 at 900 rpm is 17.

4.2.4. Analytical calculations of Rice bran oil

(a) speed in rpm = 500

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 500/60 = 8.33 rps

Viscosity of oil(μ) = 28.2cP = 28.2e-9 N-s/mm^2

Sommerfeld number is given by

$$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{500} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{28.2}{10^9} * \frac{8.33}{0.15}$$

S_{500} = 1.566

(b) speed in rpm = 700

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 700/60 = 11.33 rps

Viscosity of oil(μ) = 28.2cP = 28.2e-9 N-s/mm^2

Sommerfeld number is given by

$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$

$S_{700} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{28.2}{10^9} * \frac{11.33}{0.15}$

S_{700} = 2.192

(c) speed in rpm = 900

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 900/60 = 15 rps

Viscosity of oil(μ) = 28.2cP = 28.2e-9 N-s/mm^2

Sommerfeld number is given by

$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$

$S_{900} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{28.2}{10^9} * \frac{15}{0.15}$

S_{900} = 2.82

The Sommerfeld number obtained for RICE BRAN oil at 500 rpm is 1.566.

The Sommerfeld number obtained for RICE BRAN oil at 700 rpm is 2.192.

The Sommerfeld number obtained for RICE BRAN oil at 900 rpm is 2.82.

4.2.5. Analytical calculations of CANOLA OIL

(a) speed in rpm = 500

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 500/60 = 8.33 rps

Viscosity of oil(μ) = 40cP = 40e-9 N-s/mm^2

Sommerfeld number is given by

$$S = \left(\frac{r}{c}\right)^2 * \left(\frac{\mu * ns}{p}\right)$$

$$S_{500} = \left(\frac{27.5}{0.00275}\right)^2 * \frac{40}{10^9} * \frac{8.33}{0.15}$$

S_{500} = 2.221

(b) speed in rpm = 700

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 700/60 = 11.33 rps

Viscosity of oil(μ) = 40cP = 40e-9 N-s/mm^2

Sommerfeld number is given by

S= $(\frac{r}{c})^2 * (\frac{\mu * ns}{p})$

$$S_{700} = (\frac{27.5}{0.00275})^2 * \frac{40}{10^9} * \frac{11.33}{0.15}$$

S_{700} = 3.109

(c) speed in rpm = 900

Diameter of journal(d) = 55 mm

Radius (r) = 27.5 mm

Radial clearance(c) = 0.0275 mm

Length of journal (l) = 90mm

Taking permissible unit bearing pressure(p) as 0.15 N/mm^2

Speed in rev/sec (n_s) = 900/60 = 15 rps

Viscosity of oil(µ) = 40cP = 40e-9 N-s/mm^2

Sommerfeld number is given by

S= $(\frac{r}{c})^2 * (\frac{\mu * ns}{p})$

$$S_{900} = (\frac{27.5}{0.00275})^2 * \frac{40}{10^9} * \frac{11.33}{0.15}$$

S_{900} = 4.44

The Sommerfeld number obtained for CANOLA oil at 500 rpm is 2.221.

The Sommerfeld number obtained for CANOLA oil at 700 rpm is 3.109.

The Sommerfeld number obtained for CANOLA oil at 900 rpm is 4.44.

4.3 SUMMARY OF ANALYTICAL CALCULATIONS

S.NO	NAME OF OIL	SOMMERFELD NUMBER		
		500 RPM	700 RPM	900 RPM
1	SAE 15W40	8.624	12	15.5
2	SAE 20W40	6.664	9.328	12
3	SAE 10W30	9.44	13.21	17
4	RICE BRAN OIL	1.566	2.192	2.82
5	CANOLA OIL	2.221	3.109	4.44

Table 4.6 summary of analytical calculations

After calculating the Sommerfeld number of mineral oils and vegetable oils at three different speeds using Petroff's equation, it is evident that the values of Sommerfeld number are very low for vegetable oils which makes them much more favorable to use in Hydrodynamic lubrication, even at higher speeds the values of Sommerfeld number are low when compared to mineral oils, as Sommerfeld number is significant for the design of hydrodynamic bearings, these results conclude that vegetable oils can be useful for hydrodynamic bearings

4.4 RESULTS

The readings were tabulated and graphs are plotted for Radial pressure distribution and Axial pressure distribution at three different speeds for SAE 15W40, SAE 20W40, SAE 10W30, RICE BRAN & CANOLA oil

4.4.1 SAE 15W40

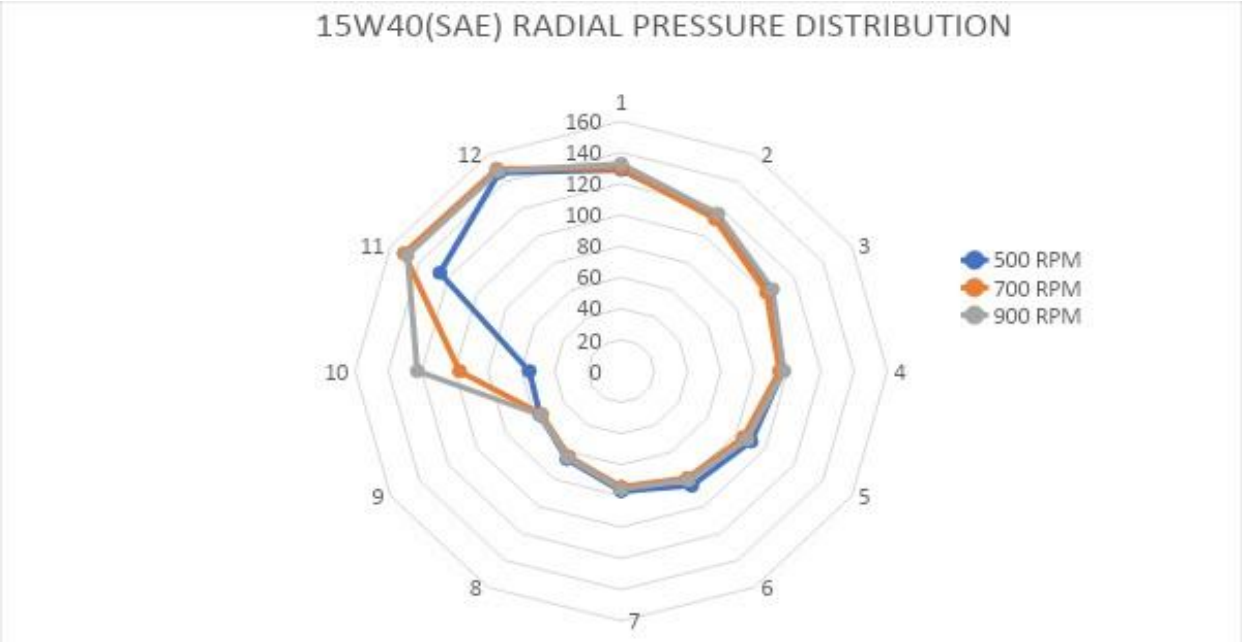


Figure 4.1 SAE 15W40 radial pressure distribution

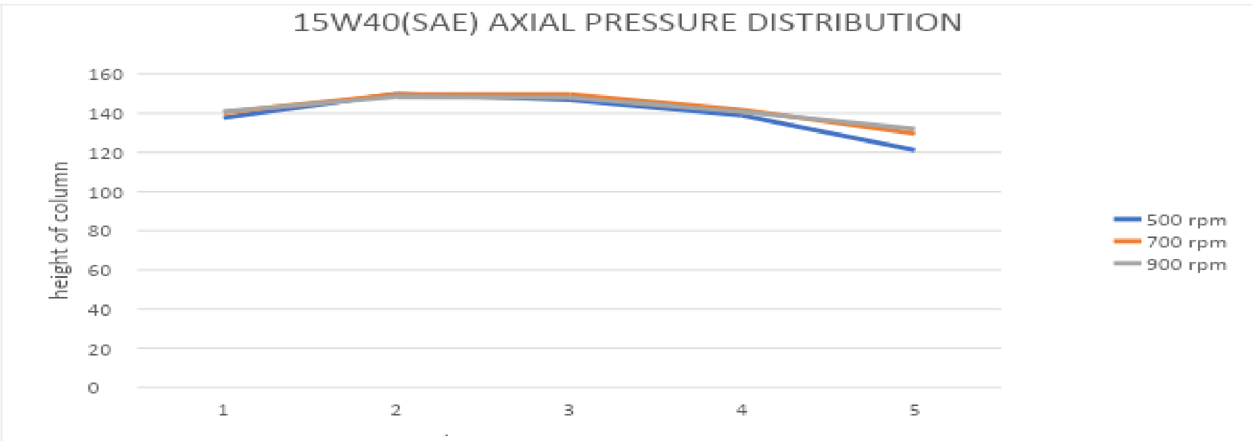


Figure 4.2 SAE 15W40 axial pressure distribution

4.4.2 SAE 20W40

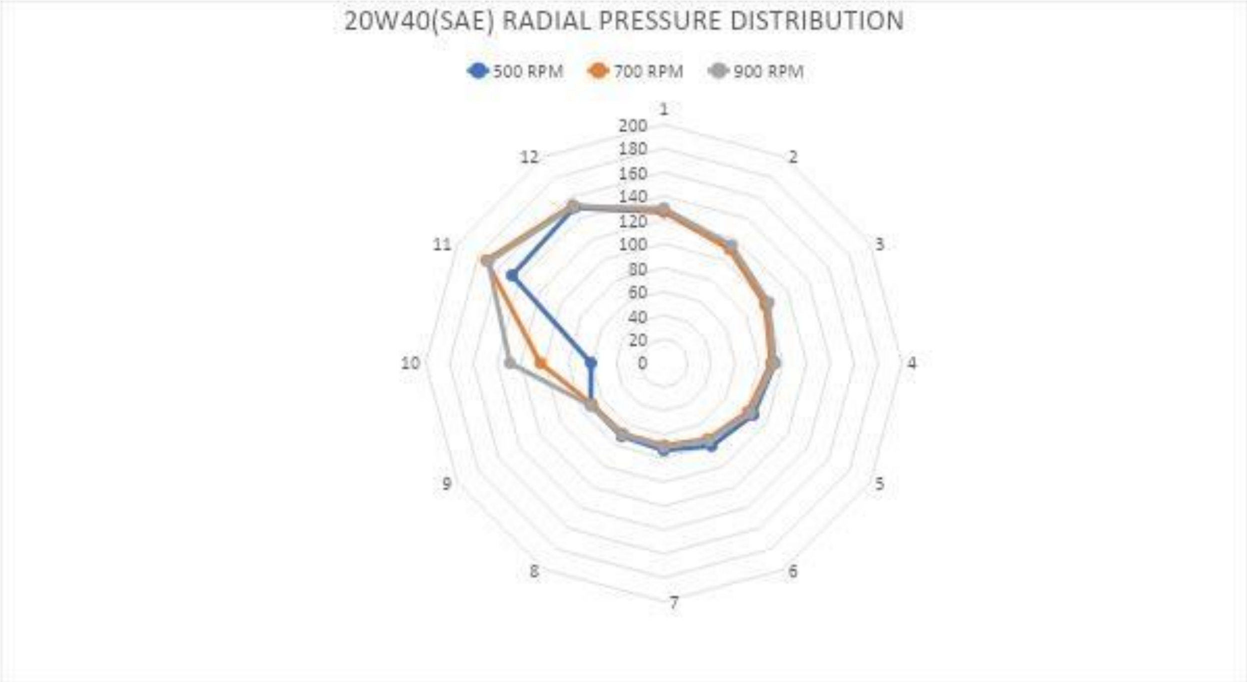


Figure 4.3 SAE 20W40 radial pressure distribution

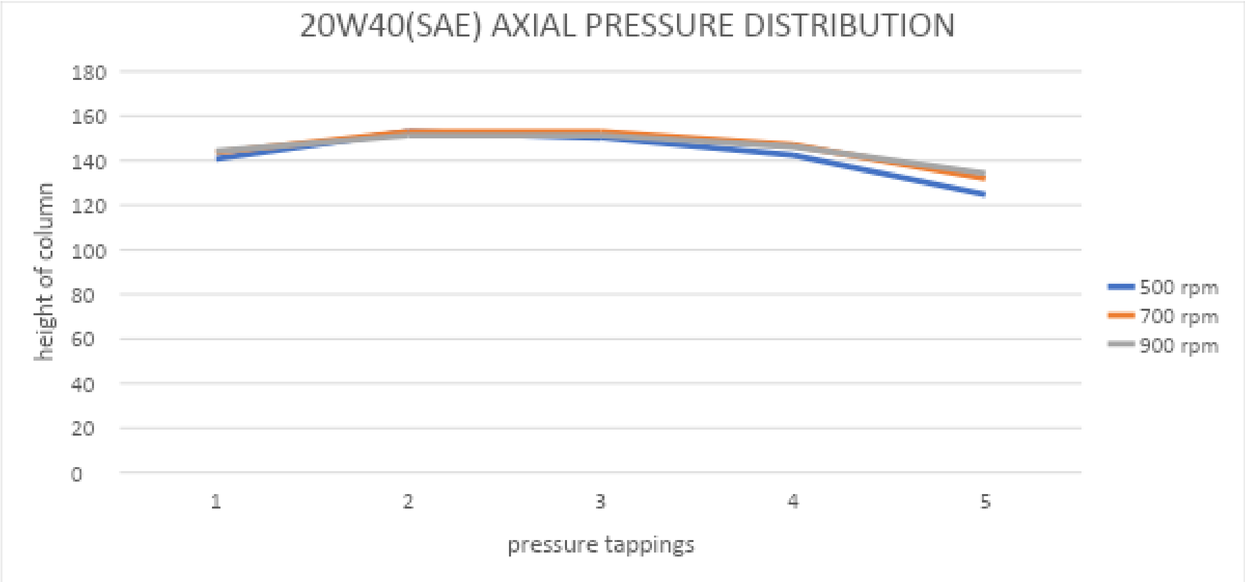


Figure 4.4 SAE 20W40 axial pressure distribution

4.4.3. SAE 10W30

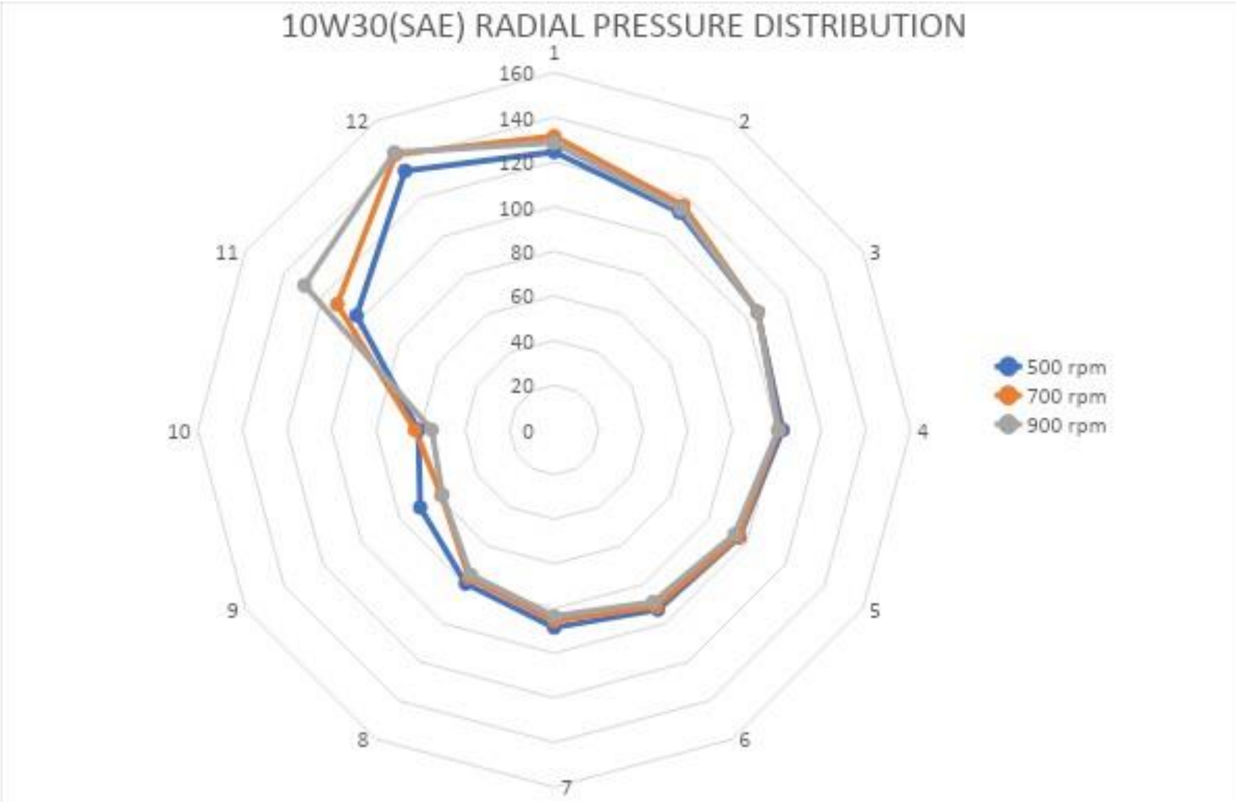


Figure 4.5 SAE 10W30 radial pressure distribution

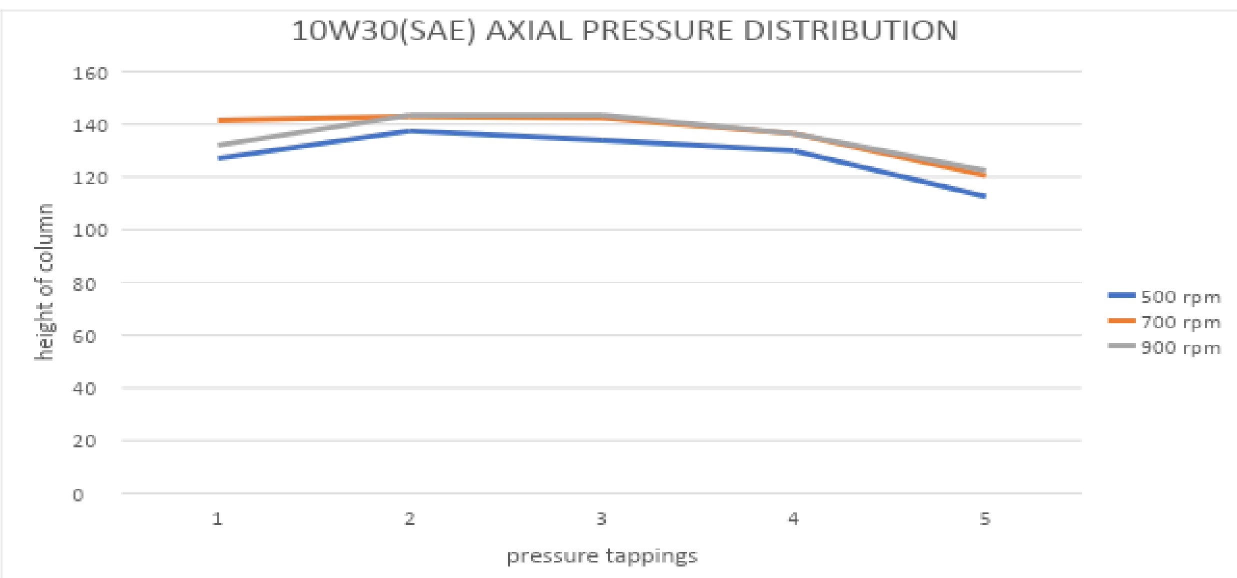


Figure 4.6 SAE 10W30 axial pressure distribution

4.4.4 RICE BRAN OIL

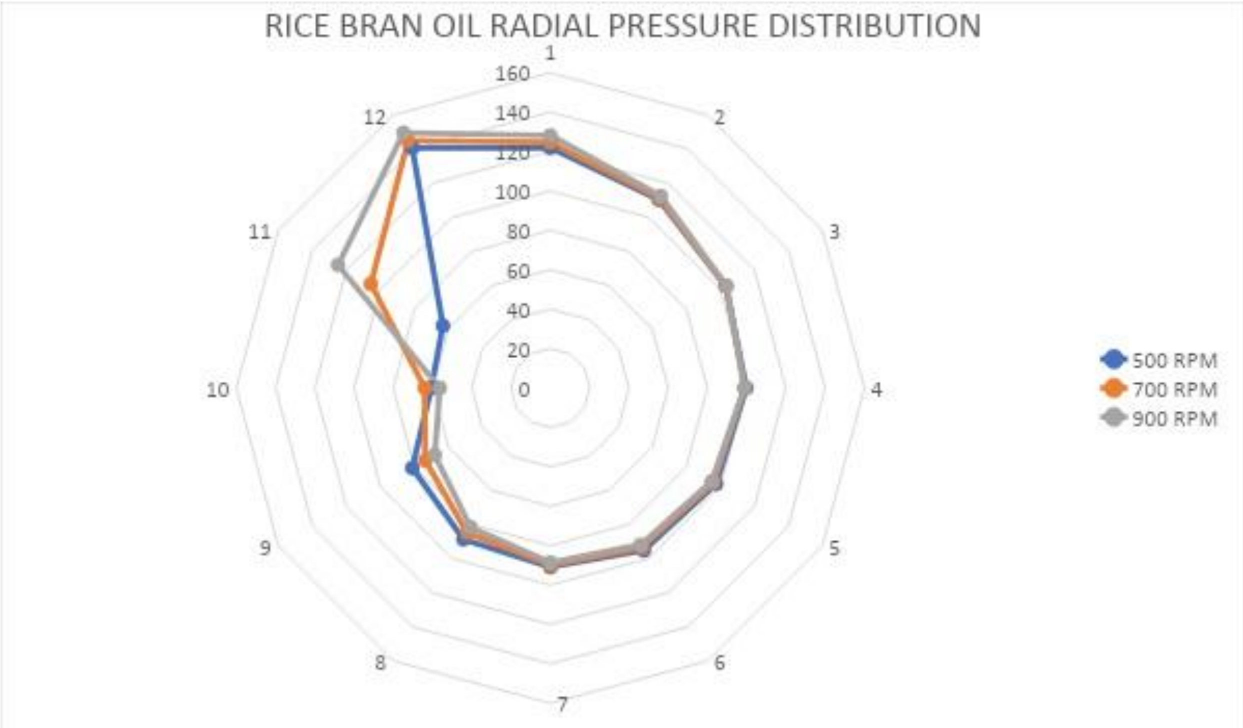


Figure 4.7 RICE BRAN radial pressure distribution

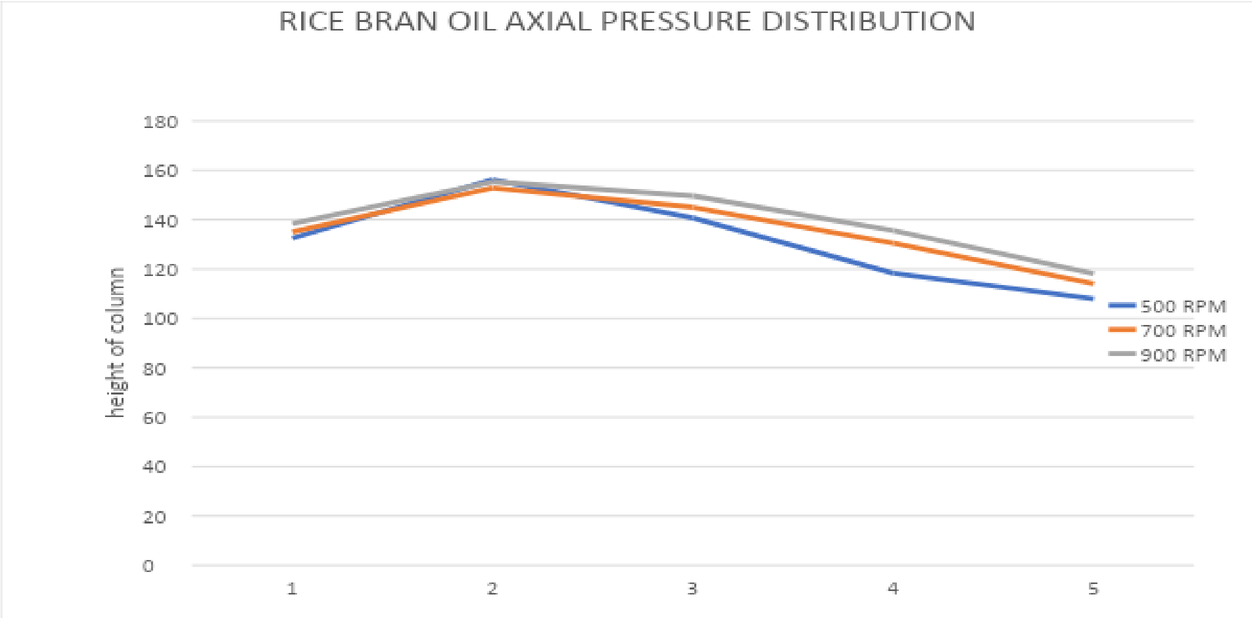


Figure 4.8 RICE BRAN axial pressure distribution

4.4.5 CANOLA OIL

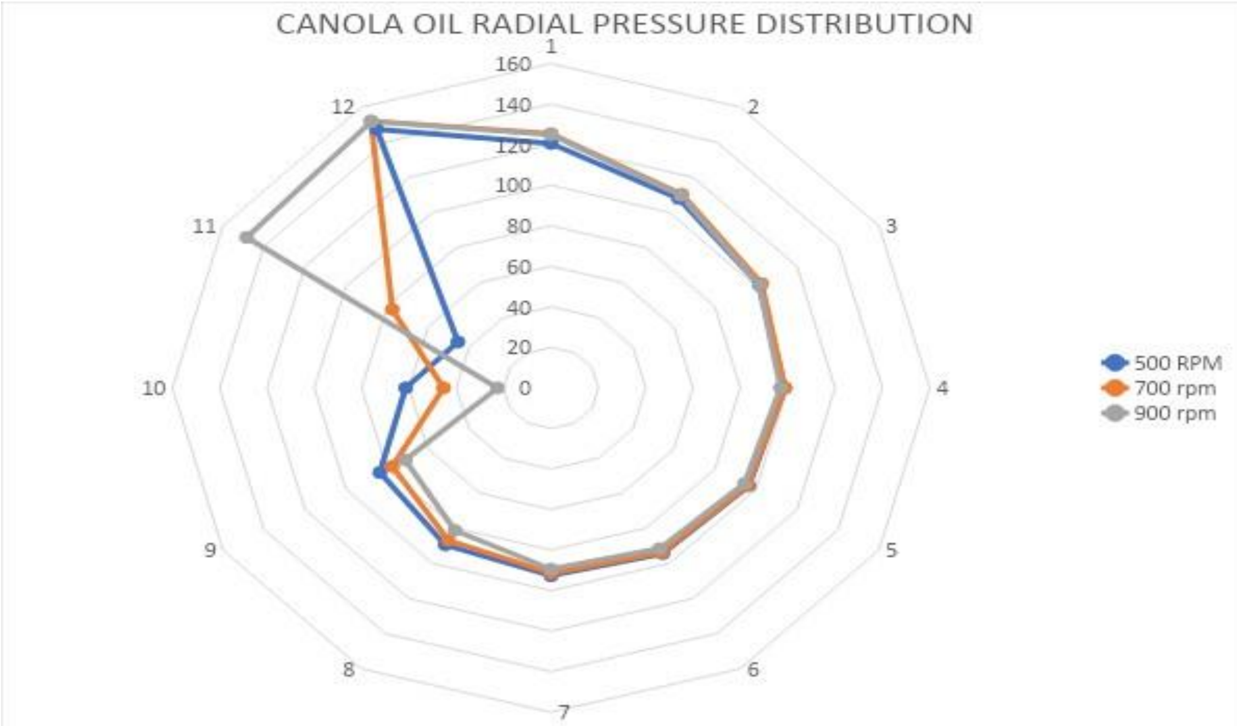


Figure 4.9 CANOLA OIL radial pressure distribution

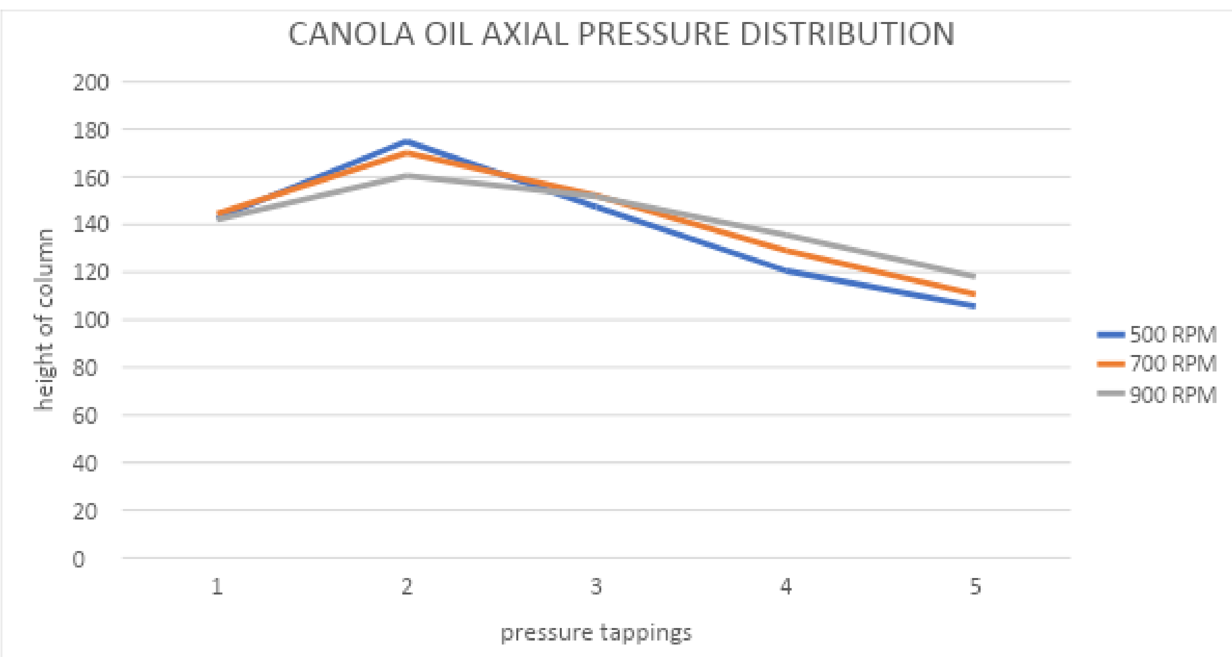


Figure 4.10 CANOLA OIL axial pressure distribution

CHAPTER 5

CONCLUSIONS & FUTURE SCOPE

5.1 ANALYSIS OF RESULTS

From the tabulated readings, the pressure profiles were plotted and Sommerfeld numbers is calculated for all the considered oils namely **SAE 15W40, SAE 20W40, SAE 10W30** are **mineral oils** and **two vegetable oils** namely **RICE BRAN & CANOLA oils** are analysed. The observations from the pressure profiles of both mineral oils and vegetable oils are

All three mineral oils exhibit similar behaviour in terms of pressure, all the radial pressures seem equal i.e., the oils exhibit higher pressures at certain locations and very low pressures at some certain location, whereas the change of speed does not show much higher impact on radial pressure, the values of pressure remain same or vary slightly for all three different speeds.

The behaviour of mineral oils varies for axial pressure distribution i.e., along the length of the journal. Each oil exhibit different behaviour at different speeds. SAE 10W30 has quite a similar behaviour at 700,900 rpm but exhibit a sudden drop in pressure towards the end of bearing but has gained pressure at the extreme pressure tapping whereas SAE 15W40 and SAE 20W40 has almost similar pressure distribution for all the speeds.

For vegetable oils i.e., RICE BRAN oil exhibits a different behaviour. For 500,700 and 900rpm it shows difference in pressure distribution but for canola oil the radial pressure distribution is somewhat same for 500,700 rpm and there is great difference for 900 rpm .

The axial pressure distribution of vegetable oils is dependent on the speed of rotor, higher the speed, higher the pressure of journal and the maximum axial pressure occurs at the middle of length of journal. For canola oil the pressure drop at the end tapping is low as compared to that of RICE BRAN oil at every speed and the pattern of axial pressure distribution is same for every speed for vegetable oils.

Vegetable oils are exhibiting better performance as that of mineral oils and can be the potential replacement of mineral oils in field of hydrodynamic lubrication.

5.2 CONCLUSION

- Mineral oils have been used tremendously in various engineering applications such as lubrication of bearings, as cutting fluids, coolants in cutting etc.
- But, as the pollution levels are increasing and these mineral oils are obtained from non-renewable energy sources, it is necessary to find a replacement for future use and reduce the pollution
- Few vegetable oils naturally have got the ability to replace mineral oils in terms of thermal, physical, chemical properties and act as potential replacement for mineral oils
- Few vegetable oils in compound with other vegetable oils have shown better characteristics.
- On a comparative study conducted between mineral oils and vegetable oils in Hydrodynamic lubrication to test whether the selected vegetable oils i.e., RICE BRAN AND CANOLA OIL can be perfect replacement for mineral oils like SAE 15W40, SAE 20W40, SAE10W30 which are commonly used in lubrication.
- The study is conducted by comparing and analysing pressure profiles, Sommerfeld number of all oils at different speeds i.e., 500 rpm, 700rpm, 900rpm.
- The study has shown that the vegetable oils have similar kind of values of Sommerfeld number as that of mineral oils.
- It is also found that the pressure profiles plotted for vegetable oils have much better and uniform profile than that of mineral oils
- Vegetable oils are biodegradable. Vegetable oils have a high friction coefficient compared to mineral oil. Also, the wear scars produced by vegetable oil is slightly lower than that of mineral oils. Vegetable oils have advantages like fire safety, low coefficient of thermal expansion, low humidity content and are easily available.
- As we know that everything have some pros and cons and we select one over the other i.e.: vegetable oil over mineral oil by considering its advantages and disadvantages. We prefer the one with more advantages and less disadvantages.
- By considering all these factors and after doing a lot of research and experimentation, finally we came to conclude that vegetable oils are best replacement of mineral oils in Hydrodynamic lubrication.

5.3 FUTURE SCOPE

This comparative study is done for 3 different mineral oils and 2 different vegetable oils namely SAE 15W40, SAE 20W40, SAE10W30, RICE BRAN, CANOLA oils. There are number of vegetable oils available and their application can be tested in hydrodynamic lubrication in different combinations. A comparative study can be done to study the performance of bearings using different theories and methods such as McKee's theory and Raimondi and Boyd method. A finite element analysis can be also done to study the performance of bearings under the affect of different lubricants using manual methods or by usage of computer aided tools such as MATLAB, Ansys etc.. and the pressure distribution in the bearing i.e., radial pressure distribution and axial pressure distribution can be studied through computational fluid dynamics (CFD) tools to analyse them in better way.

REFERENCES

1. W. Hughes and F. Osterle, “Temperature Effects in Journal Bearing Lubrication”.

2.S. Basri and D.T. Gethin, “A Comparative Study of The Thermal Behaviour of Profile Bore Bearings,” .

3. S. Cupillard, S. Glavatskih, M.J.Cervantes, “Computational Fluid Dynamics Analysis of A Journal Bearing with Surface Texturing,”.

4. K.P. Gertzos, P.G. Nikolakopoulos, C.A. Papadopoulos, “CFD Analysis of Journal Bearing Hydrodynamic Lubrication by Bingham Lubricant,”.

5. H. Liu, H. Xu , P. J. Ellison ,Z. Jin, “Application of Computational Fluid Dynamics And Fluid–Structure Interaction Method to The Lubrication Study of A Rotor–Bearing System,”.

6. J. Lin, C. Hwang and R. Yang, “Hydrodynamic Lubrication of Long, Flexible, Porous Journal Bearings Using The Brinkman Model,”.

7. M.B.W. Nabhan, G.A. Ibrahim and M.Z. Anabtawi, “Analysis of Hydrodynamic Journal Bearings Lubricated With A Binary Water-Based Lubricant,”.

8. H. E. Rasheed, “Effect of Surface Waviness on The Hydrodynamic Lubrication of A Plain Cylindrical Sliding Element Bearing,”.

9. S.K. Guha, “Analysis of Steady-State Characteristics of Misaligned Hydrodynamic Journal Bearings with Isotropic Roughness Effects,”.

10. Mr. Pramod, Akash.S. Patil, “Pressure distribution and friction coefficient of Hydrodynamic journal bearing.

11.A.Pratama, Jamari., “Hydrodynamic lubrication of textured journal bearing considering slippage”.

12.Arti Singh, “Performance of hydrodynamic journal bearing with various types of lubricant for pressure distribution and cavitation”.

13.S.A. Jagtap , M.J.Gitay ,“Performance of bio oil in journal bearing instead of synthetic oil”.