STUDY AND ANALYSIS OF DIFFERENT LUBRICANTS IN HYDRODYNAMIC BEARING

A project report submitted in partial fulfillment of the requirement forthe award of the degree of

BACHELOR OF TECHNOLOGY IN MECHANICAL ENGINEERING

BY

PEDAPATI MVSK SAI SATYA GANESH (319126520L17)
RAVADA UPENDRA (318126520108)
KANCHIPATI SAI KUMAR (318126520085)
CHALLAPALLI DINESH RAJA (318126520070)

Under the esteemed guidance of

Mr. B.S. LAKSHMI PRASAD

M.Tech

ASST.PROFESSOR

PRAGNANAM BRAHMA
ANITS

DEPARTMENT OF MECHANICAL ENGINEERING

ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY & SCIENCES

(Permanently Affiliated to Andhra University, AICTE, Accredited by NBA, NAAC)

SANGIVALSA, VISAKHAPATNAM (District)- 531162

2018-2022

ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY AND SCIENCES

Autonomous status accorded by UGC and Andhra University (Approved by AICTE, Permanently Affiliated to Andhra University, Accredited and reaccredited by NBA, accredited by NAAC) Sangivalasa, Bheemunipatnam Mandal Visakhapatnam (District) - 531162

PRAGNANAM BRAHMA

ANITS

CERTIFICATE

This is to certify that the Project Report entitled "**STUDY AND ANALYSIS OF DIFFERENT LUBRICANTS IN HYDRODYNAMIC BEARING**" being submitted by **P M V S K Sai Satya Ganesh (319126520L17), R. Upendra (318126520108), K. Saikumar (318126520085), CH. Dinesh Raja (318126520070).** In partially fulfilment for the award of Degree of Bachelor of Technology in Mechanical Engineering. It is the work of bonafide, carried out under the guidance and supervision of **Mr. B S LAKSHMI PRASAD** Assistant Professor Department of Mechanical Engineering during the academic year 2018 to 2022.

26.5.22

PROJECT GUIDE
(Mr. B S LAKSHMI PRASAD)
Assistant Professor
Mechanical Department
ANITS, Visakhapatnam

HEAD OF THE DEPARTMENT
(Dr. B. NAGA RAJU)
Head of the Department
Mechanical Department
ANITS, Visakhapatnam

PROFESSOR & HEAD
Department of Mechanical Engineering
ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY & SCIENCES
Sangivalasa-531 162 VISAKHAPATNAM Dist A.P.

2

THIS PROJECT IS APPROVED BY THE

BOARD OF EXAMINERS

INTERNAL EXAMINER: 27.5.22

PROFESSOR & HEAD
Department of Mechanical Engineering
ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY & SCIENCE
Sangivalasa-531 162

EXTERNAL EXAMINER: 27/5/22

ACKNOWLEDGEMENT

We express our gratitude to our guide Mr. B.S. Lakshmi Prasad, Assistant Professor, Mechanical Engineering Department, Anil Neerukonda Institute of Technology and Sciences, for his inspiration, guidance and plentiful support in bringing out this project. In spite of his busy schedule, he shared his valuable time for fruitful discussion and guidance.

We respectfully acknowledge our gratitude to Prof. T.V Hanumantha Rao (PRINCIPAL), Dr. B. Nagaraju (Head of the Department), Anil Neerukonda Institute of Technology and Sciences for his dynamic counselling, encouragement, perennial approachability and for extending his help for completion of the project.

Finally, we sincerely thank all the staff members of the department for giving us their heart full support in all stages of this project.

PEDAPATI MVSK SAI SATYA GANESH	**(319126520L17)**
RAVADA UPENDRA	**(318126520108)**
KANCHIPATI SAI KUMAR	**(318126520085)**
CHALLAPALLI DINESH RAJA	**(318126520070)**

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

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.

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 are:

a) The bearing ensures free rotation of the shaft or the axle with minimum friction.
b) The bearing supports the shaft or axle and holds it in the correct position.
c) 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:

(i)crankshaft bearings in petrol and diesel engines;

(ii) centrifugal pumps;

(iii) large size electric motors;

(iv) steam and gas turbines; and

(v) concrete mixers, rope conveyors and marine installations.

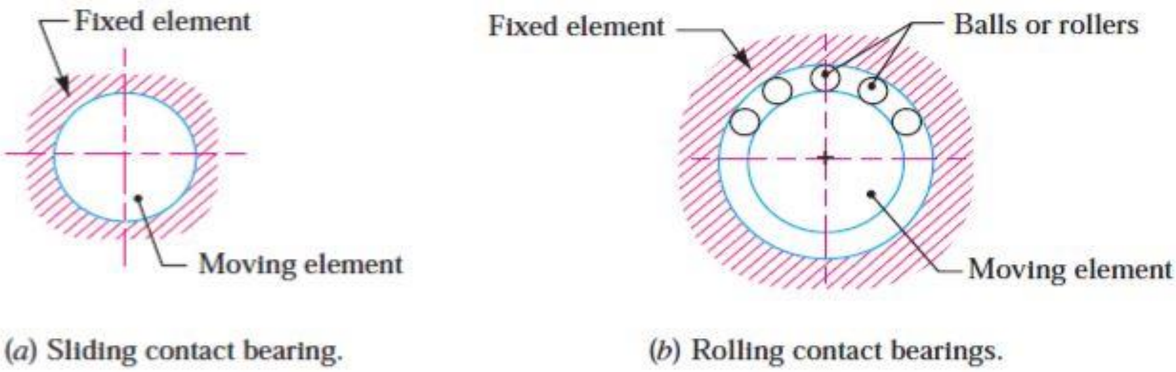


Figure1.1 Rolling contact bearing and sliding contact bearing

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 figure. 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).

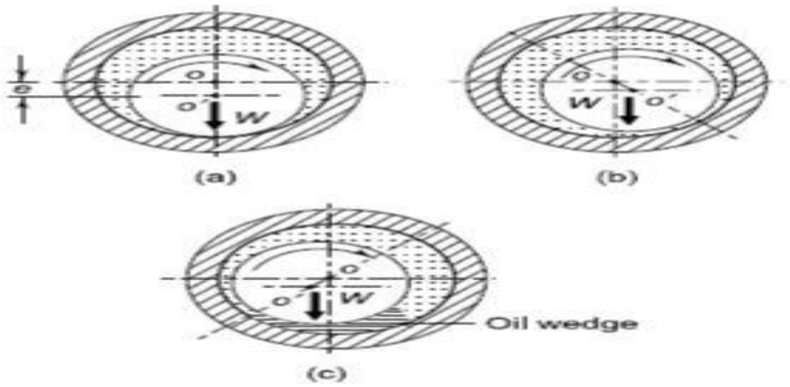


Figure 1.2 Hydrodynamic bearings

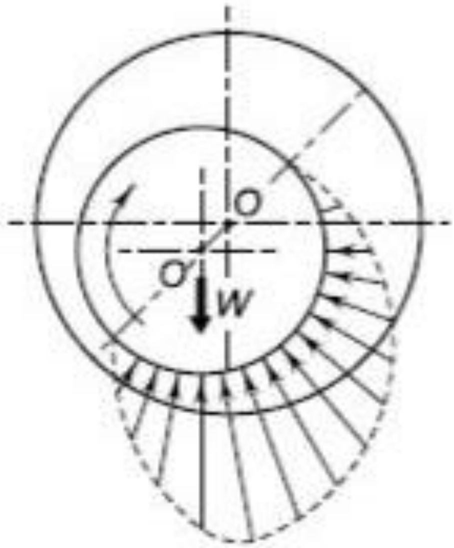


Figure 1.3 pressure distribution

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. The construction of full and partial bearings is illustrated in figure. 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 figure. The lower plate is stationary while the upper plate is moved with a velocity U by means of a force P.

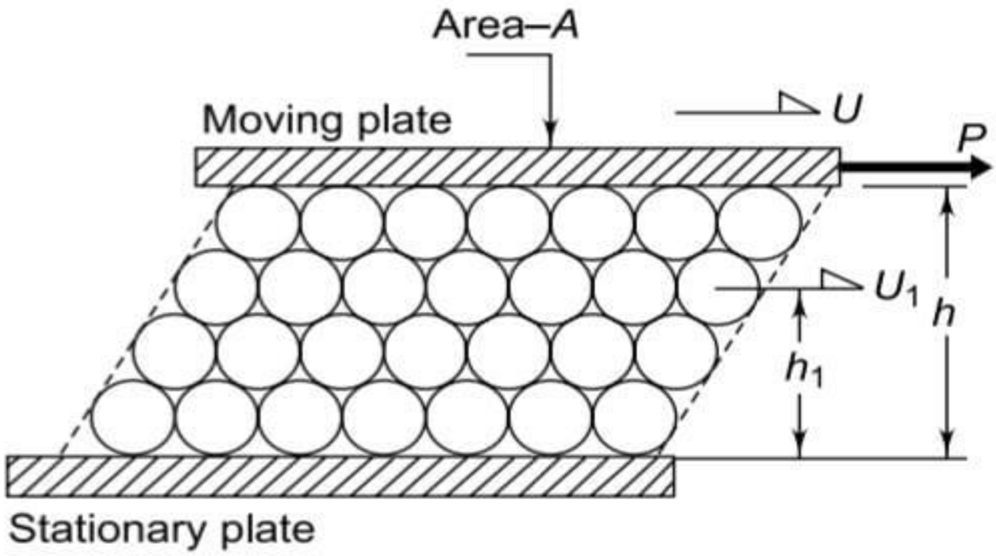


Figure 1.4 Newton's law of viscosity

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:

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

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

Petroff's equation indicates that there are two important dimensionless parameters, namely, $(\frac{r}{c})$ and that govern the coefficient of friction($\frac{\mu ns}{p}$) 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 are 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.

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. The μN/p curve is an experimental curve developed by McKee brothers.

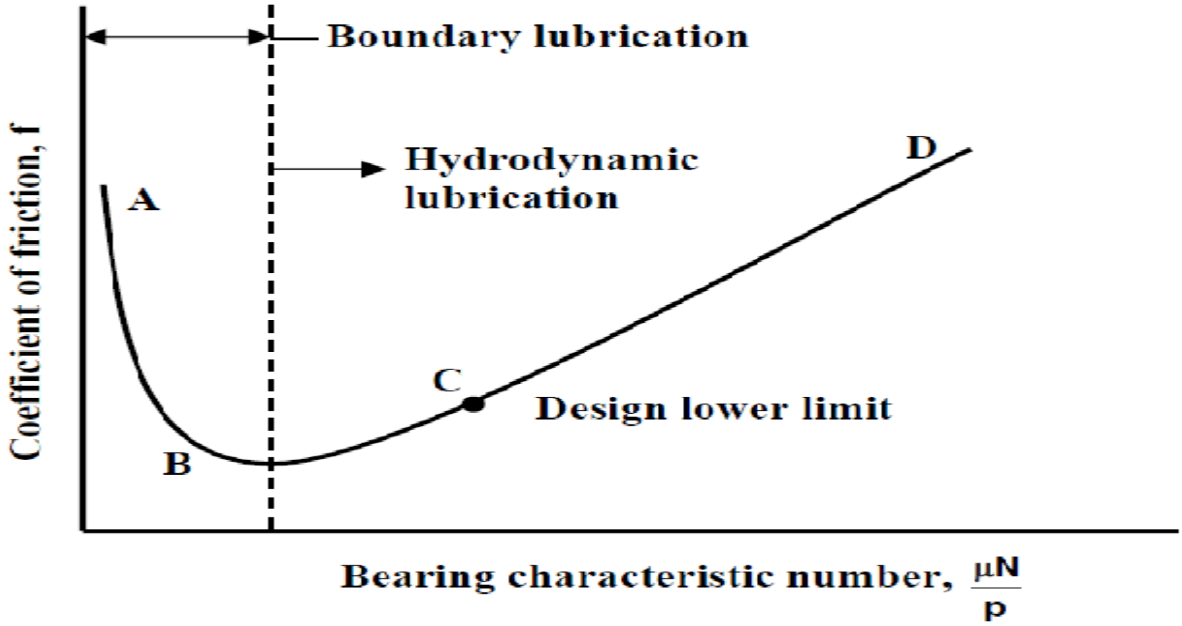


Figure 1.5 McKee's curve

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 figure, 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 μN/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 (μN/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 (μN/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 (μN/p) curve that when viscosity of the lubricant is very low, the value. Of (μN/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 (μN/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

Design and Development of Journal Bearing Experimental Setup for Determining the Pressure Distribution Due to Hydrodynamic Action

-Mr. Shantanu Sivananda Kulkarni

2.1 The learnings from this journal are

In hydrodynamic lubrication, the pressure condition of the fluid is critical to ensure good performance of the lubricated machine elements such as journal bearings. In the present study, an experimental work is conducted to determine the pressure distribution around the circumference of a journal bearing due to variation in Speed and Loads. It is concluded after observation that the location of the maximum pressure for the given operating conditions is close to the position where radial clearance between journal and casing is minimum. Our study deals with a design of journal bearing apparatus to determine the pressure distribution along the periphery and along the length of the journal bearing. By adding the weight also, we can take the readings at various speeds.

(a). Wear

Due to rotating motion of journal the bearing friction occurs. Due to friction wear of bearing and journal is occurred. To reduce wear hydrodynamic lubrication is used.

Figure 2.1 Wear in journal bearing

(b). Lubrication

Lubrication is the science (process and technique) employed to reduce friction and wear of one or both surfaces. Fig shows hydrodynamic lubrication. Left side figure shows static position of journal, middle figure shows starting position of journal and right-side figure shows Running position of Journal.

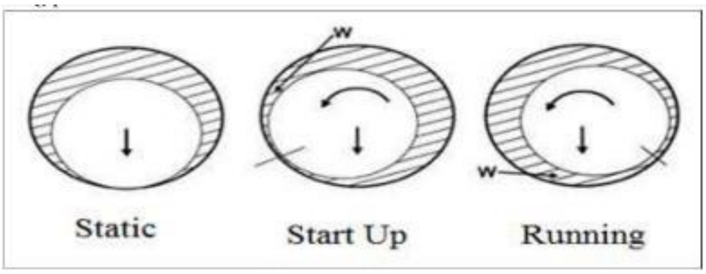


Figure 2.2Hydrodynamic lubrication

• Boundary – There is a very thin film and there is an effect of surface roughness of the contacting surfaces. In mixed lubrication there is asperity to asperity contact and a very thin fluid film.

• Solid-film – self-lubricating solid materials such as graphite are used in the bearing. This is used when bearings must operate at very high temperature.

• Hydrodynamic – The surfaces of the bearing are separated by a relatively thick film of lubricant (to prevent metal to metal contact). The fluid flow due to moving surface is in a converging gap which generates pressure separating the sliding surfaces

• Hydrostatic – The lubricant is forced into the bearing at a pressure high enough to separate the surfaces

• Electrohydrodynamic – The lubricant is introduced between surfaces that are in rolling contact (such as mating gears or rolling bearings). There is elastic deformation of the surfaces with a fluid film in between the contacting surfaces.

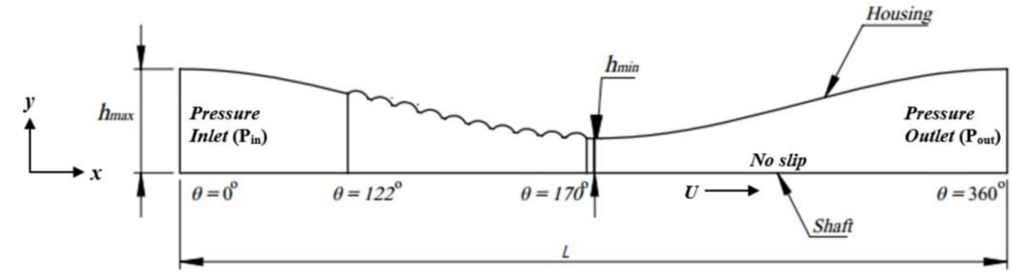
Hydrodynamic journal bearing which create load supporting fluid film according to shape and relative motion of the sliding surface, which avoid the metal-to-metal contact between the shafts and bearing. Hence, no friction will be occurred, and pressure profile is created to load action on journal bearing. Hydrodynamic bearing is suited for high load and high-speed condition particularly from considerations of long life also noise created by hydrodynamic journal bearing is lesser as compared to other bearing. Frictional loss is only at the starting condition and after that certain speed power loss due to friction is lower.

Hydrodynamic Lubrication of Textured Journal Bearing Considering Slippage: Two-dimensional CFD Analysis Using Multiphase Cavitation Model

-M.Tauviqirrahman,A.Pratama,Jamari, Muhammed

2.2 The learnings from this journal are

Partial texturing of the surface of journal bearings have been proven very beneficial in terms of friction coefficient. In the present work, the load support of the hydrodynamic textured journal bearing combined with artificial slippage is fully characterized by means of computational fluid dynamics (CFD) simulations based on the numerical solution of the Navier–Stokes equations for incompressible flow. In order to model slippage, the enhanced user-defined-function (UDF) code is developed. Realistic boundary condition is employed by implementing the mixture multiphase model to model a cavitation In the bearing. The numerical analysis is performed under the condition of different groove depths, eccentricity ratios and slippage placements along the textured area of bearing. The simulation results including hydrodynamic pressure and load support are gained and compared for conventional smooth parameters. A reference to determine optimal groove depths as well as best artificial slippage placement of textured bearing under different conditions of loading are proposed. Based on the present results, favorable slippage textured journal bearing design can be assessed.



Textured bearing modelled as a cosine profile.

Figure 2.3 pressure distribution of textured bearing

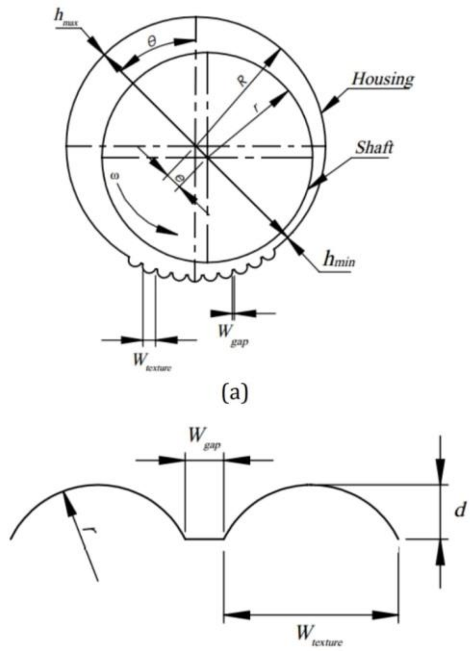


Figure 2.4 textured journal bearing

This study investigates the hydrodynamic performance of textured journal bearings considering the slippage. Large numbers of CFD analyses are conducted to find the influence of groove depth, slippage placement, and eccentricity ratio on ultimate load support and hydrodynamic pressure. The multiphase cavitation model used here revises the traditional single-phase cavitation model that neglects the phase change within the divergent zone. Based on the simulation results, the following conclusions can be drawn from the study: 1. The simulation results show that the higher the eccentricity ratio, the lower the positive effect of introduction of texturing. This prevails both for the purely textured bearing and the slippage textured one. 2. In the case of purely textured bearing, the optimum groove depth exists to a achieve the maximum load support. For a purely textured journal bearing with given eccentricity ratio, the groove depth which is equal to minimum film thickness results in the highest additional load support. 3. In the case of a combination of slippage and texturing, a well-chosen placement of the artificial slippage condition on textured zone of the bearing has more positive effect with respect to the load support, compared to purely textured surface. Introducing the slippage boundary on all groove edges of textured area leads to a significant enhancement in load support.

Performance of bio-oil on journal bearing instead of synthetic oil

- Mrs.S.A. Jagatap, Dr.L.B Abhang, Prof.M.J.Gitay

2.3 The learnings from this journal are

Rapid decrease in petroleum resources, we are in search of alternative sources for power generation and environmental hazards alarms to use eco-friendly alternative. Jatropha is a non-edible sourced Bio-lubricant shows excellent coefficient of friction, noble anti-wear capability, low environmental emission. Recent research states Jatropha have higher viscosity and improves the load carrying capacity. Comparative study of popular synthetic lubricant (i.e., 20W40, Turbinal XT46 oil) with Jatropha oil has been carried out. The friction forces and the hydrodynamic friction coefficients are calculated and compared. Rapid depletion of petroleum resources and environmental hazards alarms to use eco-friendly alternative. Jatropha is a non-edible sourced Bio-lubricant shows low coefficient of friction, anti-wear capability, low environmental hazardous.

Figure 2.5 Jatropha seed

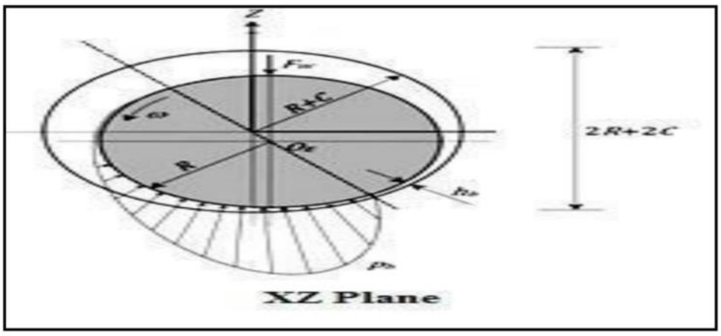


Figure 2.6 pressure distribution of jatropha oil in journal bearing

After testing of three oils that is SAE20W40, Turbinal XT 46 and Jatropha bio lubricant. We conclude that we got maximum tribological properties for Jatropha bio-oil as compared to XT 46 and 20W40, so it is beneficial to use. The main property like biodegradability of a bio-oil lubricant that's why bio-oil lubricant is ahead of other bio-oils with acts as non-pollutant for environment. Jatropha works on low operating temperature generates high torque, but power loss is high, this is because of high viscosity. Jatropha Bio lubricant shows better results for load carrying capacity as that of the 20W40 and Turbinal XT 46 and both theoretical and analytical results show enhance mantinada carrying capacity of the Jatropha bio-lubricant rises with increase in journal speed and eccentricity ratio. Jathropa be used as alternative bio lubricant for journal bearing because it has biodegradability property and increased load carrying capacity hence can be used as alternative bio lubricant for journal bearing application. Also, jatropha shows the higher-pressure distribution than SAE20W40and Turbinal XT46.

A comprehensive review on palm oil and the challenges using vegetable oil as lubricant base-stock

-Muhammad arif dandan, wan Mohamad aiman wan Yahaya

2.4 The learnings from this journal are

Many researchers have discovered that palm oil and most of the vegetable oils have a potential to be the main sources for environmentally favorable lubricant, due to its properties that showed excellent lubrication performance, biodegradability and renewability. Poor low-temperature behavior, low oxidation and thermal stability, and narrow range of available stabilities will create a boundary of bio-lubricant to be apply as a good lubricant. Bio-lubricant can be used in many types of applications such as automotive transmission fluids, metal working fluids, cold rolling oils, fire resistant hydraulic fluids, industrial gear oils, neat cutting oils and automotive gear lubricants. This review will address the challenge of using vegetable oil as a lubricant base-stock and the palm oil as a potential lubricant. There are many advantages and disadvantages using vegetable oil as a lubricant that will be discussed in this paper. The aim of this paper also to study the past, current and future of vegetable oil and the use of palm oil as a lubricant.

In summary, vegetable oil is an environmentally friendly lubricant which can be

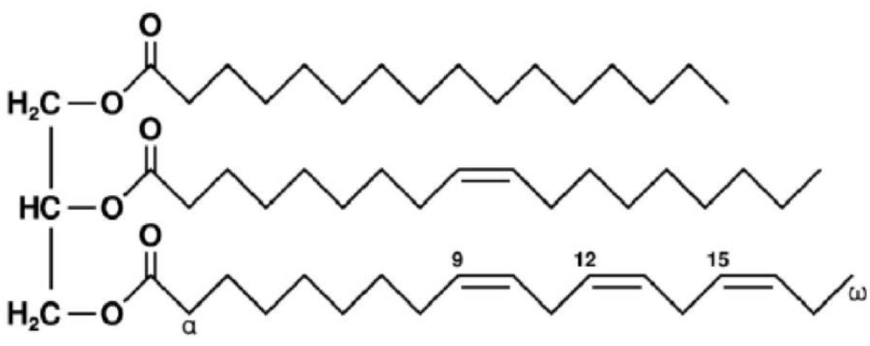


Figure 2.7 triglyceride molecule

used in any industry that involve lubrication with a little modification. It is proved by the researchers that the properties of the vegetable oil that is rarely found in the mineral oil is very useful in industrial application. If the vegetable oil is developed efficiently for lubricant purpose, it would benefit the environment and the local people by creating job

opportunity and provision of modern application carriers to rural people. Vegetable oil is facing several issues and challenge like the oxidation stability and poor low temperature behavior to be used in the industry, but there is much research that can been done in order to improve the quality of the vegetable oil.

Comparative study-a mineral oil-based lubricant and lubricant obtained from vegetable oil

-Dr.V.K. Chhibber, Mr. Sanket Kumar Saxena

2.5 The learnings from this journal are

Word petroleum is better recognized by the crude obtained through PETRA (Greek ROCK), the chemical nomenclature of oil generally is of C_nH_{2n+2}(alkane) type, of which petrol comes in the range of C5 to C8, diesel C9 to C16 and fuel oil and lubricating oil C>16.in its refined form. The thermal properties of lubricating oil obtained from hydrocarbons are not said to be environment friendly or eco-friendly due to the production of several irritative substances known as toxins and are said to be responsible for causing eye, ear and throat inflammation along with several type of allergies when they are released into atmosphere. Besides that, the thermal properties of lubricating oil obtained from vegetables are not so much toxic, harmful and accidentally disastrous has been found that the obtainable lubricating oil is far more eco-friendly or environment friendly and equally biodegradable comparatively. Because the vegetable based lubricating oil remains less oxidative, therefore causes a minute disadvantage in comparison to the mineral oil. The present paper discusses, describes and highlights some of the necessary and essential properties which are meant for a lubricating oil to be good and most suitable for proper functioning of an engine.

This comparative study concludes:

1. We can change the mineral based lubricating oil with the vegetable based lubricating oil.
2. Vegetable based oil are much more eco-friendly.
3. Though they are less thermoxidative yet biodegradable up to a maximum extent.
4. They can be considered as a nation economy booster.

Study of pressure profile in hydrodynamic lubrication journal bearing

- Chetan Mehra M.E Tribology & maintenance engg. SGSITS, Indore.

2.6 The learnings from this journal are

In hydrodynamic lubrication, the pressure condition of the fluid is critical to ensure good performance of the lubricated machine elements such as journal bearings. In the present study, an experimental work was conducted to determine the pressure distribution around the circumference of a journal bearing and fluid frictional force of the bearing caused by shearing actions. A journal diameter of 100mm with a ½ length-to-diameter ratio was used. Pressure results for 600 RPM speed at different radial loads were obtained. The experimental results were compared to predicted values from established Raimondi and Boyd charts. It was observed that the location of the maximum pressure for the given operating conditions is close to the predicted value.

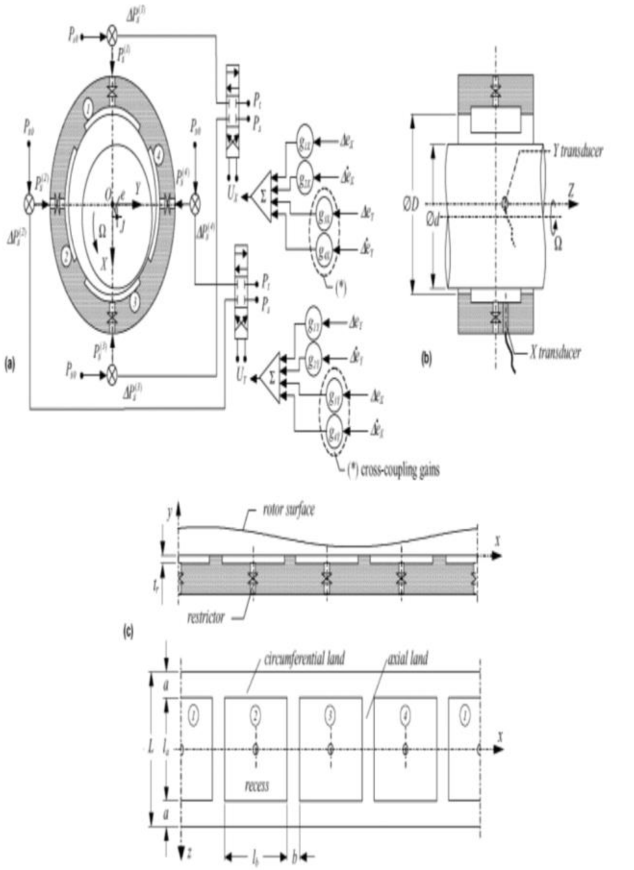


Figure 2.8 Hybrid journal bearing

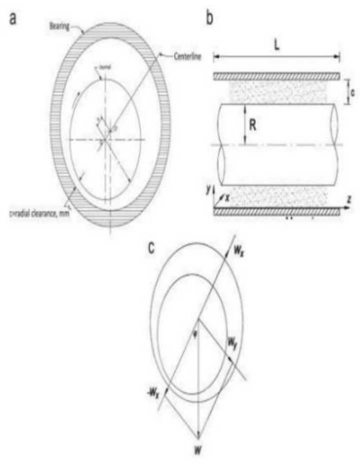


Figure 2.9 Journal bearings projections

The preliminary study of pressure profiles around a journal bearing under hydrodynamic lubrication were described and compared with theoretical profiles obtained from Raimondi and Boyd charts. From the experimental results, it was found that the experimental maximum pressure values were higher than the theoretical maximum pressure values. It was also observed that the position (i.e., angle) of the maximum pressure has not changed significantly with loads. Generally, the pattern of pressure distribution obtained was similar to those reported in other studies . However, the position of the minimum film thickness varied clearly with changes in loads. Friction coefficients of oil lubricant in this experiment decrease when the loads increase. This shows a similar trend as in Raimondi and Boyd chart. It was also observed that the experimental friction coefficient values are significantly higher than the predicted values.

Performance of vegetable oil as lubricant in extreme pressure condition

- S. Syahrullail, S. kamitani, A. Shakirin

2.7 The learnings from this journal are

Today, vegetable oils are being considered for their suitability as industrial lubricant. One of the main problems of vegetable oils is their poor performance when working at high temperature and pressure. They may oxidize and undergo changes to their chemical and physical composition. In the worst cases, the oxygen bond in vegetable oils can lead to metal oxidation and weaken the structure of the metal. In this paper, the performance of vegetable oils as a lubricant was tested using a fourball tribometer under extreme pressure conditions, which conforms to ASTM D2783. The test lubricants were commercial stamping oil, commercial hydraulic oil, jatropha oil, RBD palm olein and palm fatty acid distillate. The normal load used for this test was 126 kg. The results showed that vegetable oils have a high friction coefficient compared to mineral oil. Also, the wear scars produced by vegetable oil is slightly lower than those produced by mineral oil. It can be concluded that vegetable oils have potential as lubricants.

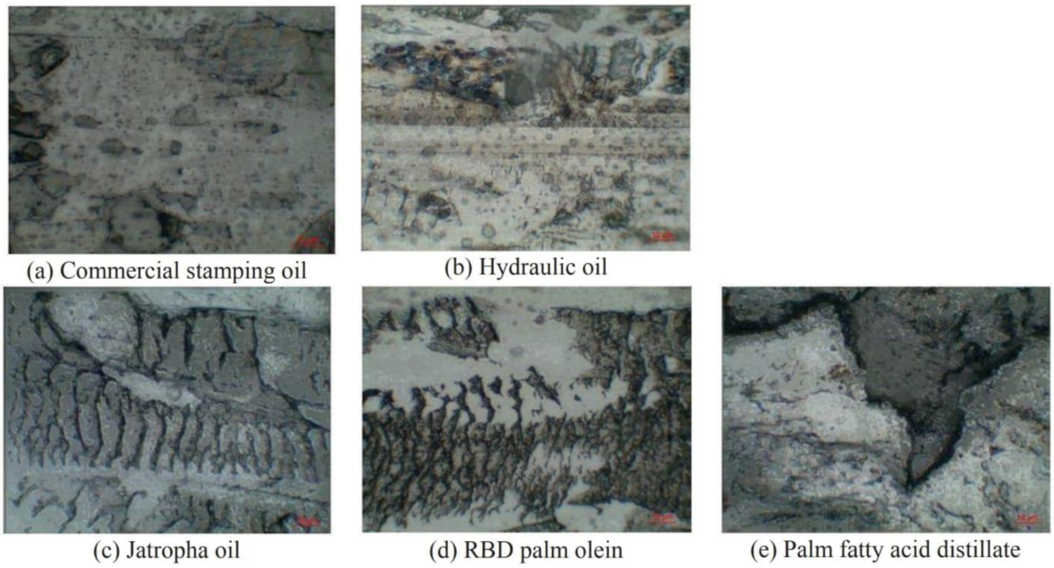


Figure 2.10 Microscopic view of lubricants

From this set of experiments, it can be concluded that vegetable oil shows potential for development as an industrial lubricant. Although the values of the friction coefficients and steel ball wear scar diameters were slightly larger compared to the results shown by the commercial stamping oil, this problem could be solved by adding proper additives. The study of suitable additives for vegetable oil is not in the scope of this research and will be investigated in the next experimental works.

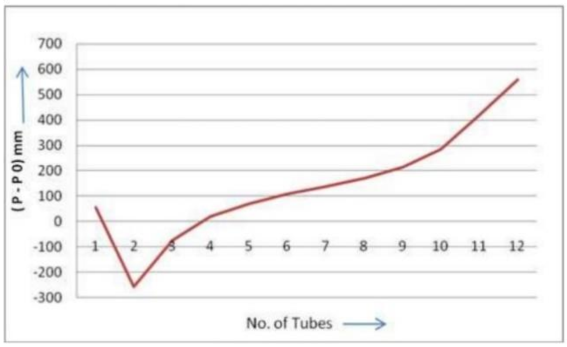


Figure 2.11 Linear graph

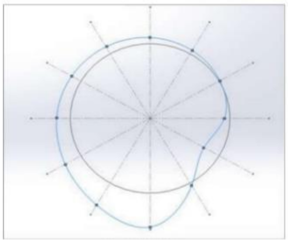


Figure 2.12 polar graph

Preliminary study of pressure profile in hydrodynamic lubrication journal bearing

- Muhammad Ali Ahmad, Rob-DwyerJoyce

2.8 The learnings from this journal are

In hydrodynamic lubrication, the pressure condition of the fluid is critical to ensure good performance of the lubricated machine elements such as journal bearings. In the present study, an experimental work was conducted to determine the pressure distribution around the circumference of a journal bearing and fluid frictional force of the bearing caused by shearing actions. A journal diameter of 100mm with a ½ length-to-diameter ratio was used. Pressure results for 600 RPM speed at different radial loads were obtained. The experimental results were compared to predicted values from established Raimondi and Boyd charts. It was observed that the location of the maximum pressure for the given operating conditions is close to the predicted value.

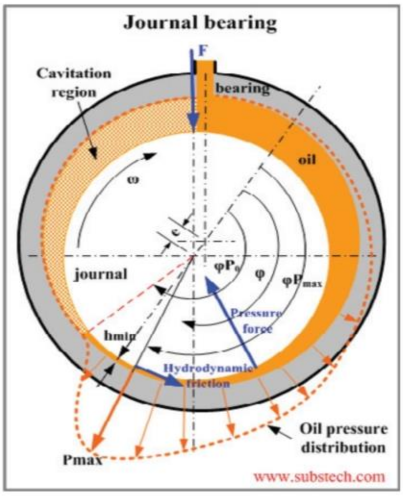


Figure 2.13 pressure distribution schematic

In this paper, the preliminary study of pressure profiles around a journal bearing under hydrodynamic lubrication were described and compared with theoretical profiles obtained from Raimondi and Boyd charts. From the experimental results, it was found that the experimental maximum pressure values were higher than the theoretical maximum pressure values. It was also observed that the position (i.e., angle) of the maximum pressure has not changed significantly with loads. Generally, the pattern of pressure distribution obtained was like those reported in other studies. However, the position of the minimum film thickness varied clearly with changes in loads. Friction coefficients of oil lubricant in this experiment decrease when the loads increase. This shows a similar trend as in Raimondi and Boyd chart. It was also observed that the experimental friction coefficient values.

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 = 50.055 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. Motor control. Electronic DC Controller for motor speed control.
6. Manometer frame with 14 tubes of 240cm. Height with scales and adjustable oil supply tank.
7. Recommended oil = Lubricating oil SAE 30.
8. Supply required AC single phase 230v.50Hz stabilized.
9. r = Radius of Journal.
10. δ = 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:

So =(r/c) ^2 (μn/p) *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 10/SAE 90 / SAE10W30 / Soyabean / Palm 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

4.1OBSERVATIONS

The pressure distributions for both mineral oils and vegetable oils are plotted for three different speeds are tabulated, the values are recorded according to the height of pressure column in the tube of journal bearing apparatus, the height of liquid in the column is directly proportional to the value of pressure at that tapping in journal bearing.

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

OIL : SAE 10				OIL: SAE 10				OIL: SAE10			
SPEED : 500 rpm				SPEED: 700 rpm				SPEED:900 rpm			
Static head(Po) : 106 cm				Static head(Po):106 cm				Static head(Po):106cm			
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	20		1	124	18		1	127	21	
2	112	6		2	112	6		2	113	7	
3	102.5	3.5		3	103	-3		3	102.5	-3.5	
4	96	-10		4	97	-9		4	95	-11	
5	88.5	-17		5	90	-16		5	86.5	-23.5	
6	81	-25		6	82.5	-23.5		6	77.5	-28.5	
7	69.5	-36.5		7	70.5	-35.5		7	63	-43	
8	52.5	-53.5		8	51	-55		8	42	-64	
9	31	-75		9	27	-79		9	26	-80	
10	81	-25		10	111	5		10	117	11	
11	156	50		11	164	58		11	163	57	
12	146	40		12	143.5	37.5		12	148	42	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	137.5	31.5		A	136	30		A	139	33	
B	146	40		B	143	37		B	146	40	
12	146	40		12	143.5	37.5		12	148	42	
C	100	-6		C	137.5	31.5		C	141	35	
D	125	19		D	126.5	20.5		D	129	23	

Table 4.1 pressure distribution of SAE10

The readings obtained for SAE90 for three different speeds (500,700,900 rpm) are:

OIL: SAE 90			OIL: SAE90			OIL:SAE90		
SPEED : 500 rpm			SPEED: 700 rpm			SPEED:900 rpm		
Static head(Po): 100.5 cm			Static head(Po):100.5 cm			Static head(Po):100.5cm		
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	125	24.5	1	126.5	26	1	125	24.5
2	110.5	10	2	111	10.5	2	111	10.5
3	99	-1.5	3	99	-1.5	3	98.5	-2
4	91.5	-9	4	90	-10.5	4	89.5	-11
5	82.5	-18	5	78.5	-22	5	78.5	-22
6	73.5	-27	6	66	-34.5	6	65	-35.5
7	60	-40.5	7	47	-53.5	7	45	-55.5
8	39.5	-61	8	23	-77.5	8	19	-81.5
9	19	-81.5	9	12	-88.5	9	29	-71.5
10	91.5	-9	10	130	29.5	10	163.5	63
11	157.5	57	11	168	67.5	11	169	68.5
12	146	45.5	12	148	47.5	12	146.5	45
Axial pressure distribution			Axial pressure distribution			Axial pressure distribution		
A	139	38.5	A	142	41.5	A	142	41.5
B	146	45.5	B	149	48.5	B	149	48.5
12	146	45.5	12	148	47.5	12	146.5	46
C	140	39.5	C	143	42.5	C	143	42.5
D	126	25.5	D	130	29.5	D	131	30.5

Table 4.2 pressure distribution of SAE90

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):105cm				Static head(Po): 105cm				Static head(Po): 105cm			
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	122.5	17.5		1	128	23		1	127.5	22.5	
2	111.5	6.5		2	114	9		2	113.5	8.5	
3	104	-1		3	105	0		3	104	-1	
4	99.5	-5.5		4	99	-6		4	98	-7	
5	95	-10		5	94	-11		5	92.5	-12.5	
6	91.5	-13.5		6	89.5	-15.5		6	88.5	-16.5	
7	86	-19		7	83.5	-21.5		7	82	-23	
8	78.5	-26.5		8	75	-30		8	73	-32	
9	68	-37		9	60	-45		9	57	-48	
10	59	-46		10	59	-46		10	53.5	-51.5	
11	103	-2		11	112	7		11	127	22	
12	134	29		12	141	36		12	142.5	37.5	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	125	20		A	139	34		A	130	25	
B	136	31		B	142	37		B	142	37	
12	134	29		12	141	36		12	142.5	37.5	
C	128	23		C	134.5	29.5		C	136	31	
D	114.5	9.5		D	118.5	13.5		D	120.5	15.5	

Table 4.3 pressure distribution of SAE10W30

The readings obtained for SOYABEAN oil at three different speeds (500,700,900 rpm) are

OIL: SOYABEAN OIL				OIL: SOYABEAN OIL				OIL: SOYABEAN OIL			
SPEED:500 rpm				SPEED:700 rpm				SPEED:900 rpm			
Static head(Po): 104cm				Static head(Po): 104cm				Static head(Po): 104cm			
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	16		1	123	19		1	125	21	
2	110.5	6.5		2	111.5	7.5		2	112	8	
3	114	10		3	104	0		3	104	0	
4	111	7		4	100	-4		4	99.5	-4.5	
5	98.5	-5.5		5	97	-7		5	96.5	-7.5	
6	97	-7		6	95	-9		6	94.5	-9.5	
7	95	-9		7	92.5	-11.5		7	91.5	-12.5	
8	92.5	-11.5		8	89.5	-14.5		8	88	-16	
9	88	-16		9	83	-21		9	79.5	-24.5	
10	72	-32		10	69	-35		10	43	-61	
11	71	-33		11	71	-33		11	87	-17	
12	116	12		12	128	24		12	135.5	31.5	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	114.5	10.5		A	122	18		A	127.5	23.5	
B	126	22		B	137	33		B	143	39	
12	116	12		12	128	24		12	135.5	31.5	
C	108	4		C	117	13		C	124	20	
D	101.5	-2.5		D	105.5	1.5		D	110.5	6.5	

Table 4.4 pressure distribution for SOYABEAN oil

The readings obtained for PALM oil at three different speeds (500,700,900 rpm) are:

OIL: PALM OIL				OIL: PALM OIL				OIL: PALM OIL			
SPEED: 500 rpm				SPEED: 500 rpm				SPEED: 500 rpm			
Static head(Po): 103 cm				Static head(Po): 103 cm				Static head(Po): 103 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	117.5	14.5		1	118.5	15.5		1	121	18	
2	109.5	6.5		2	109.5	6.5		2	110.5	7.5	
3	103.5	0.5		3	103.5	0.5		3	104	1	
4	100.5	-2.5		4	100	-3		4	100	-3	
5	97	-6		5	96.5	-6.5		5	96.5	-6.5	
6	94.5	-8.5		6	93.5	-9.5		6	94	-9	
7	91.5	-11.5		7	90	-13		7	90.5	-12.5	
8	86	-17		8	84	-19		8	84.5	-18.5	
9	78	-25		9	74.5	-28.5		9	74.5	-28.5	
10	70.5	-32.5		10	69.5	-33.5		10	68.5	-34.5	
11	105.5	2.5		11	115.5	12.5		11	116	13	
12	124.5	21.5		12	128	25		12	131	28	
Axial pressure distribution				Axial pressure distribution				Axial pressure distribution			
A	119.5	16.5		A	121.5	18.5		A	123.5	20.5	
B	126.5	23.5		B	129	26		B	132	29	
12	124.5	21.5		12	128	25		12	131	28	
C	121	18		C	123.5	20.5		C	125.5	22.5	
D	111.5	8.5		D	113.5	10.5		D	114.5	11.5	

Table 4.5 pressure distribution for PALM oil

4.2 ANALYTICAL CALCULATIONS

4.2.1 Analytical calculations of SAE10

(a) Speed in rpm = 500 rpm

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) = 8.33 rps

Viscosity of oil(μ) = 70cP = 70e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{500} = (\frac{27.5}{0.00275})^2 * \frac{70}{10^9} * \frac{8.33}{0.15}}$$

S_{500} = 3.88

(b) Speed in rpm = 700 rpm

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) = 70cP = 70e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{700} = (\frac{27.5}{0.00275})^2 * \frac{70}{10^9} * \frac{11.33}{0.15}}$$

S_{700} = 5.44

(c) Speed in rpm = 900 rpm

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) = 70cP = 70e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{900} = (\frac{27.5}{0.00275})^2 * \frac{70}{10^9} * \frac{15}{0.15}}$$

S_{900} = 7

The Sommerfeld number obtained for SAE 10 at 500 rpm is 3.88.

The Sommerfeld number obtained for SAE 10 at 700 rpm is 5.44.

The Sommerfeld number obtained for SAE 10 at 900 rpm is 7.

4.2.2 Analytical calculations of SAE90

(a) Speed in rpm= 500 rpm

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) = 50cP = 50e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{500} = (\frac{27.5}{0.00275})^2 * \frac{50}{10^9} * \frac{8.33}{0.15}}$$

S_{500} = 2.776

(b) Speed in rpm = 700 rpm

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) = 50cP = 50e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{700} = (\frac{27.5}{0.00275})^2 * \frac{50}{10^9} * \frac{11.33}{0.15}}$$

S_{700} = 3.886

(c) Speed in rpm= 900 rpm

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

$$\boldsymbol{S_{900} = (\frac{27.5}{0.00275})^2 * \frac{50}{10^9} * \frac{15}{0.15}}$$

S_{900} = 5

The Sommerfeld number obtained for SAE 90 at 500 rpm is 2.776.

The Sommerfeld number obtained for SAE 90 at 700 rpm is 3.886.

The Sommerfeld number obtained for SAE 90 at 900 rpm is 5.

4.2.3 Analytical calculations of SAE 10W30

(a) Speed in rpm = 500 rpm

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

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

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

S_{500} = 9.44

(b) Speed in rpm = 700 rpm

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

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

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

S_{700} = 13.21

(c) Speed in rpm = 900 rpm

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 $N\text{-}s/mm^2$

Sommerfeld number is given by

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

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

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 Soyabean oil

(a) Speed in rpm = 500 rpm

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) = 60cP = 60e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{500} = (\frac{27.5}{0.00275})^2 * \frac{60}{10^9} * \frac{8.33}{0.15}}$$

S_{500} = 3.332

(b) Speed in rpm = 700 rpm

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) = 60cP = 60e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{700} = (\frac{27.5}{0.00275})^2 * \frac{60}{10^9} * \frac{11.33}{0.15}}$$

S_{700} = 4.668

(c) Speed in rpm = 900 rpm

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) = 60cP = 60e-9 $N\text{-}s/mm^2$

Sommerfeld number is given by

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

$$\boldsymbol{S_{900} = (\frac{27.5}{0.00275})^2 * \frac{60}{10^9} * \frac{15}{0.15}}$$

S_{900} = 6

The Sommerfeld number obtained for SOYABEAN oil at 500 rpm is 3.332

The Sommerfeld number obtained for SOYABEAN oil at 700 rpm is 4.668

The Sommerfeld number obtained for SOYABEAN oil at 900 rpm is 6.

4.2.5 Analytical calculations of PALM oil

(a) Speed in rpm = 500 rpm

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) = 43cP = 43e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{500} = (\frac{27.5}{0.00275})^2 * \frac{43}{10^9} * \frac{8.33}{0.15}}$$

S_{500} = 2.38

(b) Speed in rpm = 700 rpm

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) = 43cP = 43e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{700} = (\frac{27.5}{0.00275})^2 * \frac{43}{10^9} * \frac{11.33}{0.15}}$$

S_{700} = 3.34

(c) Speed in rpm = 900 rpm

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) = 43cP = 43e-9 N-s/mm^2

Sommerfeld number is given by

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

$$\boldsymbol{S_{900} = (\frac{27.5}{0.00275})^2 * \frac{43}{10^9} * \frac{11.33}{0.15}}$$

S_{900} = 4.3

The Sommerfeld number obtained for PALM oil at 500 rpm is 2.38

The Sommerfeld number obtained for PALM oil at 700 rpm is 3.34

The Sommerfeld number obtained for PALM oil at 900 rpm is 4.3

S.No	Name of oil	Sommerfeld number		
		500 rpm	700 rpm	900 rpm
1	SAE 10	3.88	5.44	7
2	SAE90	2.776	3.886	5
3	SAE 10W30	9.44	13.21	17
4	SOYABEAN OIL	3.332	4.668	6
5	PALM OIL	2.38	3.34	4.3

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.

4.3 RESULTS

The readings were tabulated and graphs are plotted for Radial pressure distribution and Axial pressure distribution at three different speeds for SAE 10, SAE90, SAE 10W30, SOYABEAN, PALM oil

4.3.1 SAE 10

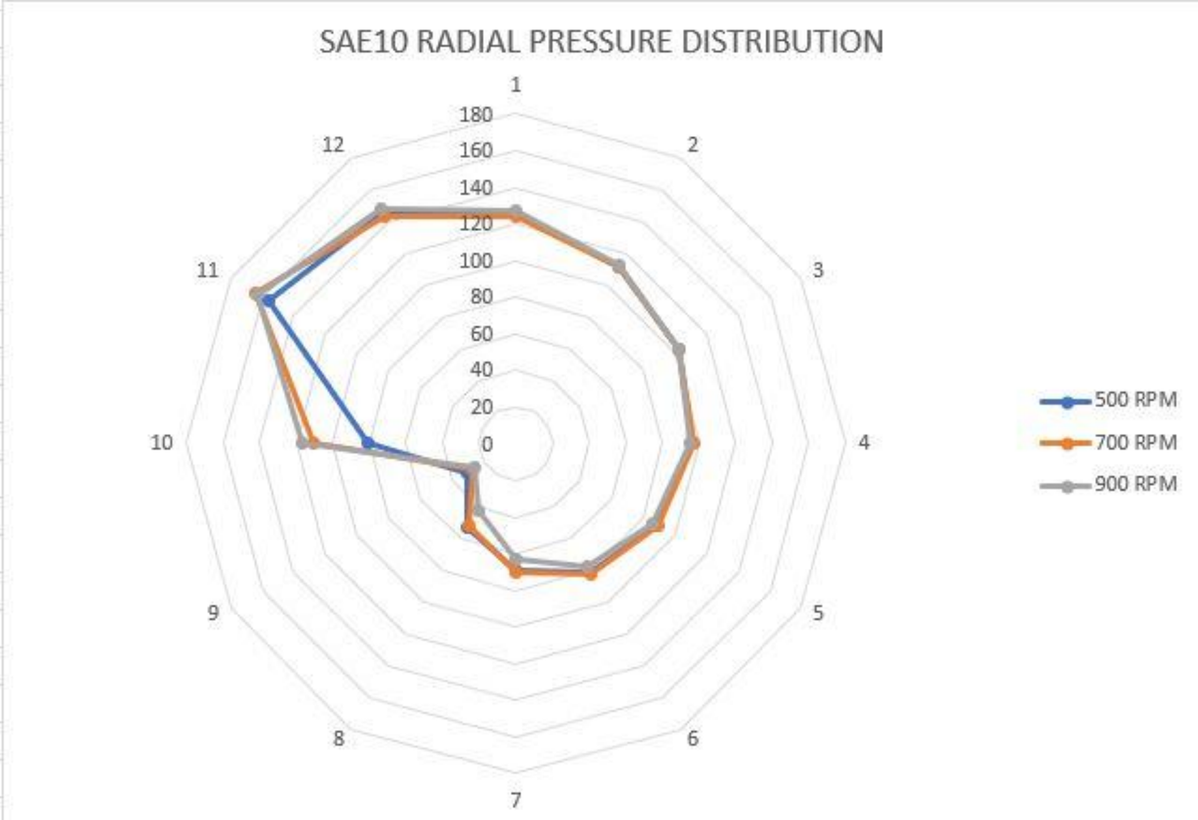


Figure 4.1 sae 10 radial pressure distribution

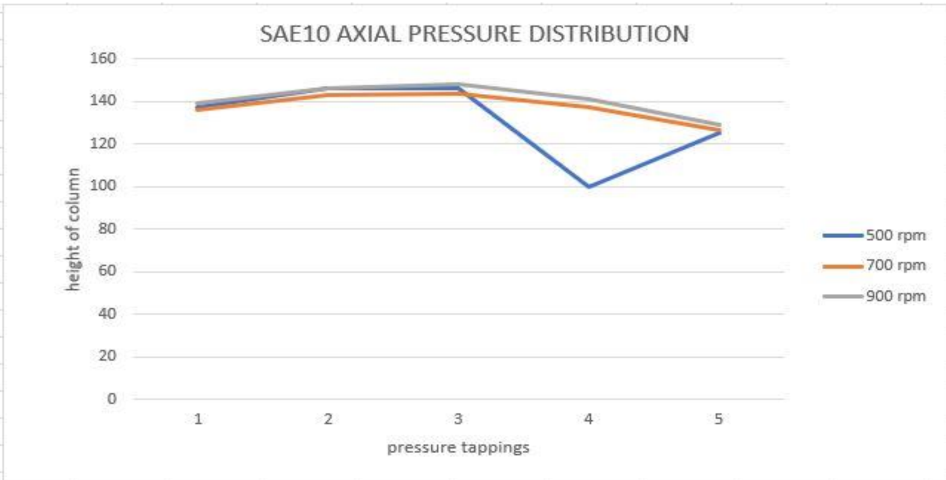


Figure 4.2 sae 10 axial pressure distribution

4.3.2 SAE 90

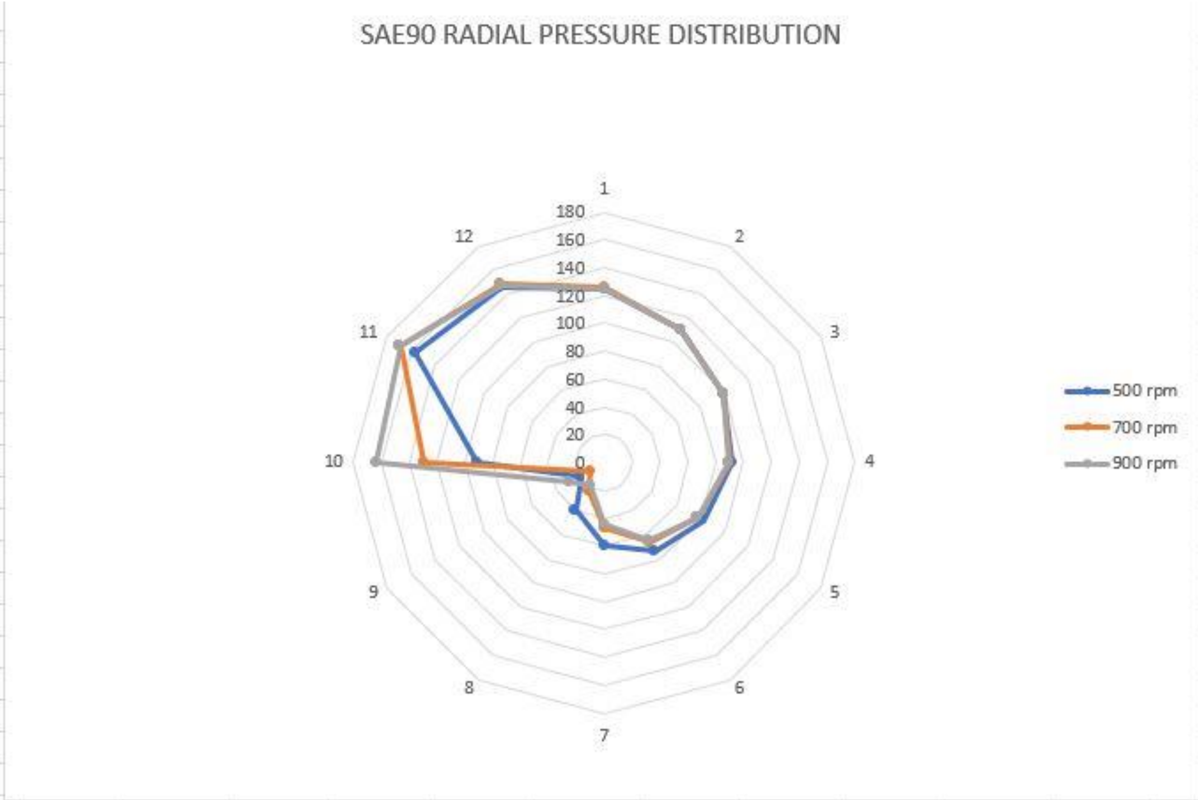


Figure 4.3 sae90 radial pressure distribution

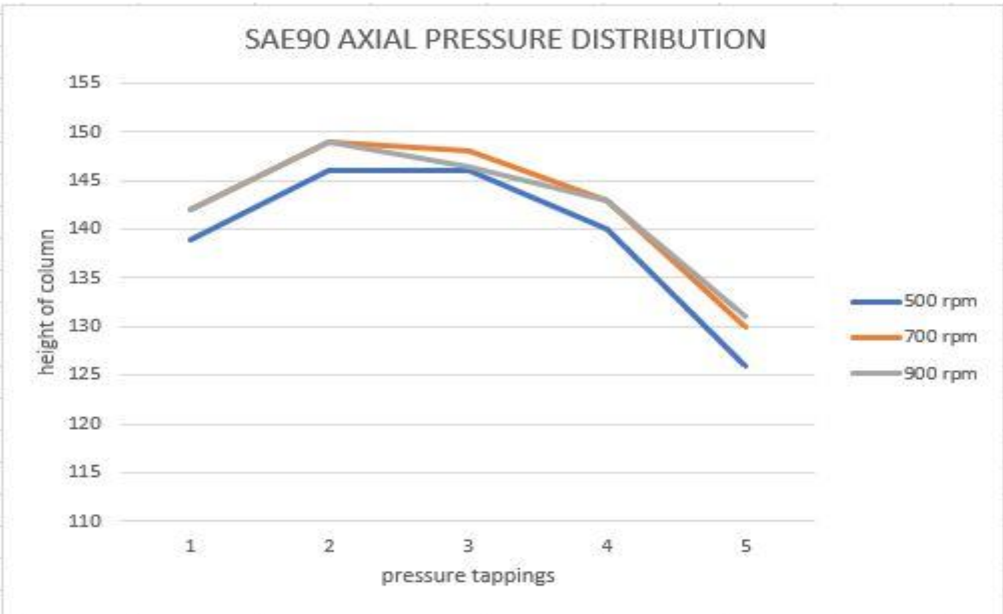


Figure 4.4 sae90 axial pressure distribution

4.3.3 SAE 10W30

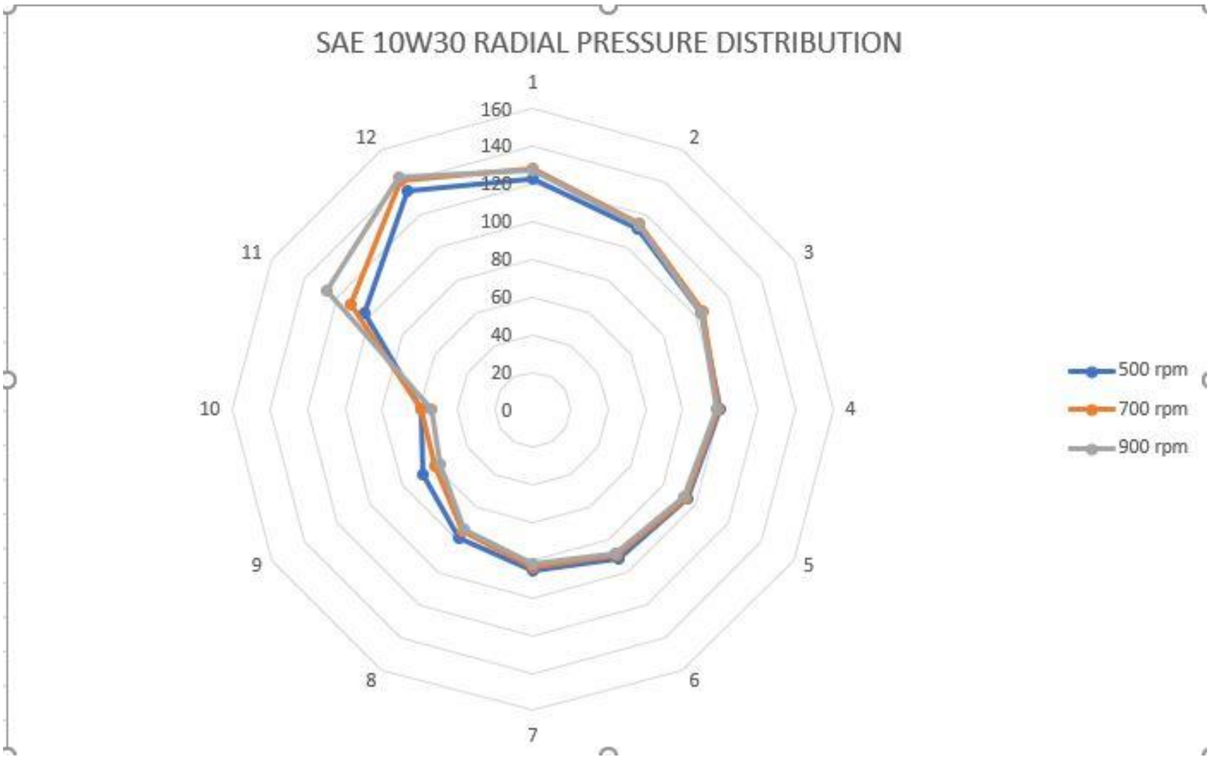


Figure 4.5 sae 10w30 radial pressure distribution

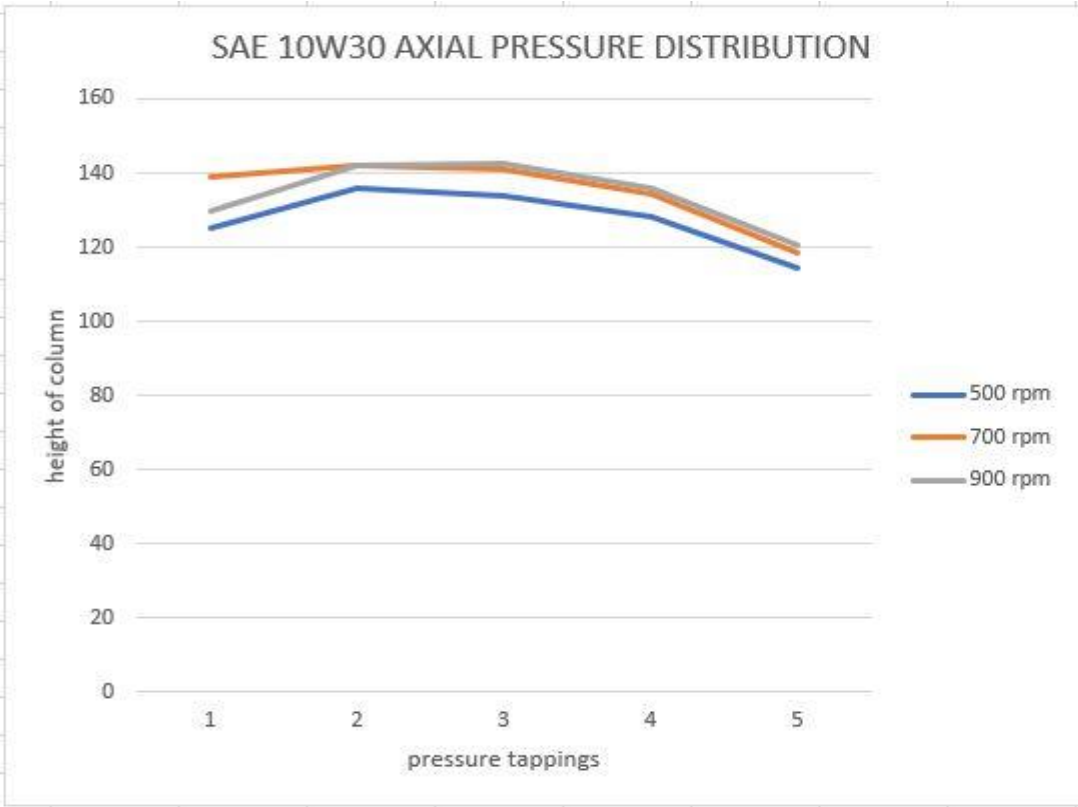


Figure 4.6 sae 10w30 axial pressure distribution

4.3.4 SOYABEAN OIL

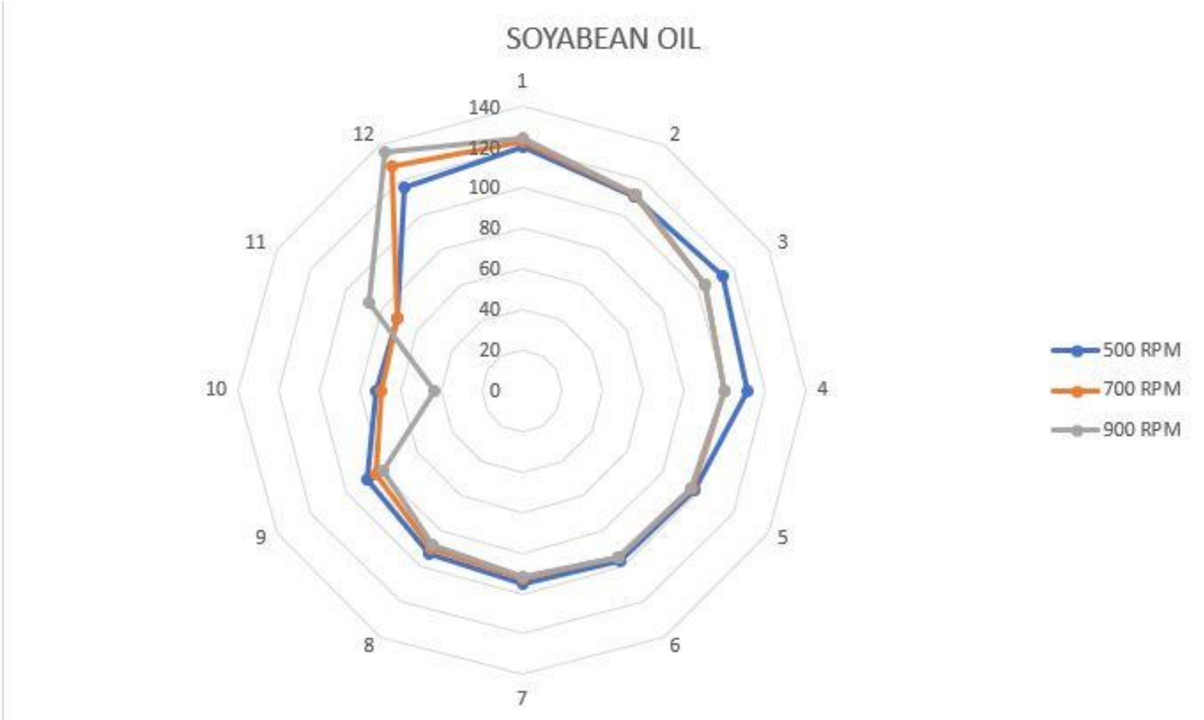


Figure 4.7 soyabean oil radial pressure distribution

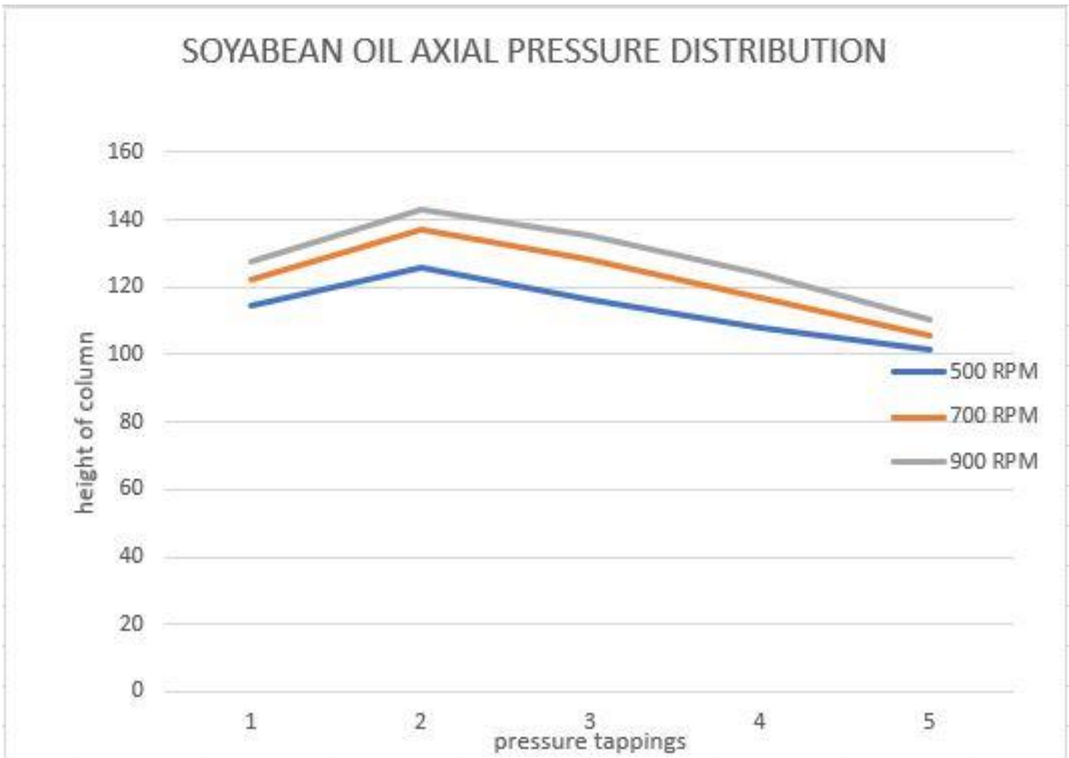


Figure 4.8 soyabean oil axial pressure distribution

4.3.5 PALM OIL

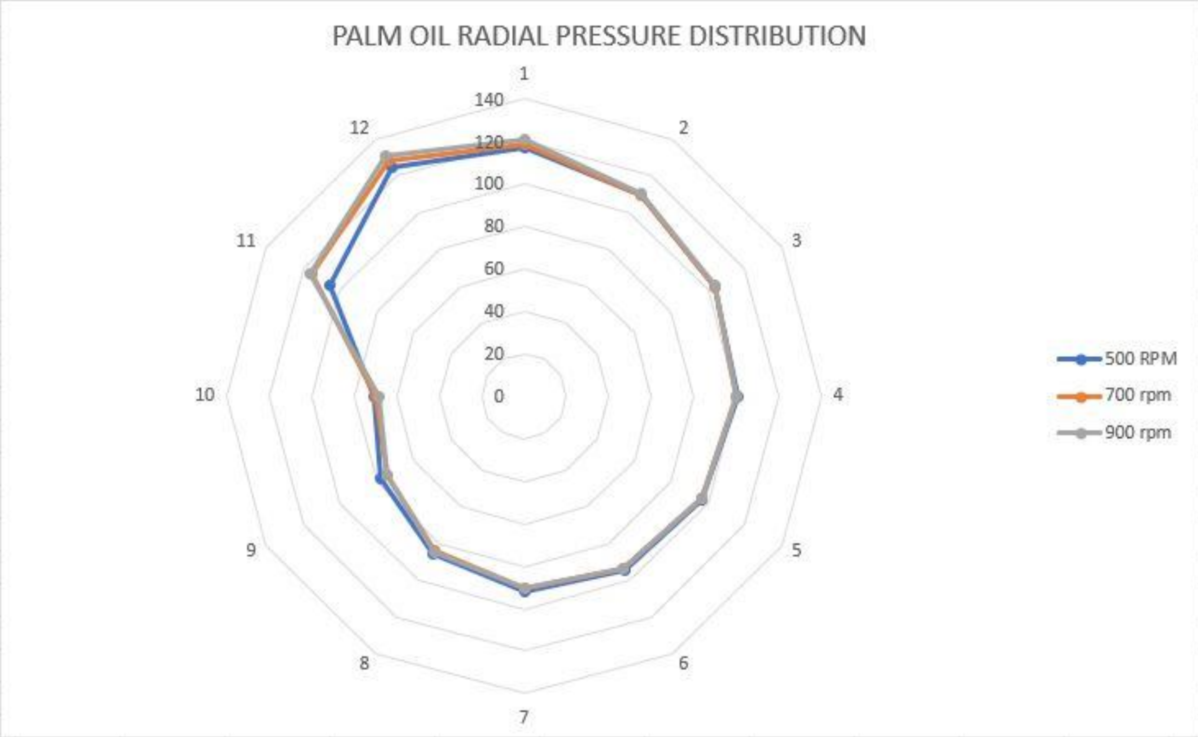


Figure 4.9 palm oil radial pressure distribution

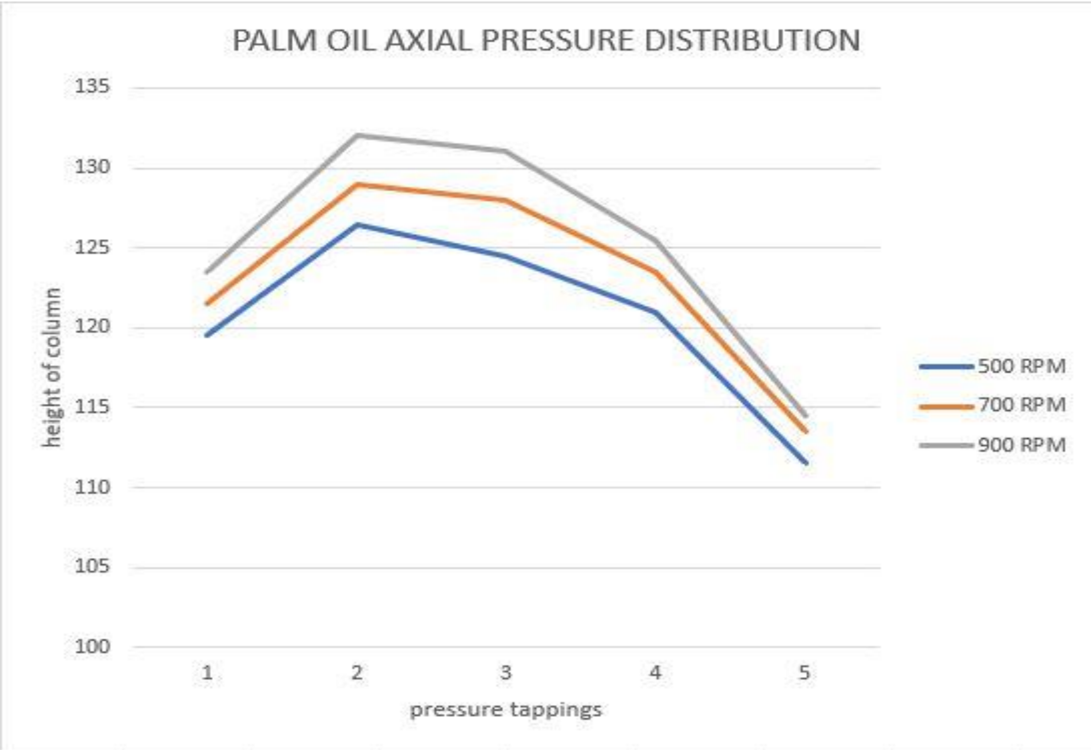


Figure 4.10 palm oil axial pressure distribution

4.4 ANALYSIS OF RESULTS

From the tabulated readings, the pressure profiles were plotted and Sommerfeld numbers is calculated for all the mineral oils and vegetable oils. All three mineral oils exhibit similar behaviors 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 behavior of mineral oils varies for axial pressure distribution i.e., along the length of the journal. Each oil exhibit different behavior at different speeds. SAE10 has quite a similar behavior 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 SAE90 and SAE 10W30 has almost similar pressure distribution for all the speeds. It is also observed that for SAE90 and SAE 10W30 the axial pressure is independent of speed of rotor the pressures vary slightly.

For vegetable oils i.e., Soyabean oil exhibits a different behavior, for 500 and 700 rpm it shows similar pressure distribution but it varies as it varies slightly but for Palm oil the radial pressure distribution is almost same for all the speeds and it is uniform throughout the radial part of journal for vegetable oils.

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 Soyabean oil the pressure drop at the end tapping is low as compared to that of Palm 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.

CHAPTER 5

CONCLUSIONS & FUTURE SCOPE

5.1 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., Soyabean and Palm oil can be perfect replacement for mineral oils like SAE10, SAE90, SAE10W30 which are commonly used in lubrication.
- The study is conducted by comparing and analyzing 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
- Hence, it is found that the vegetable oils analyzed in this study can be better replacement for mineral oils in Hydrodynamic lubrication.

5.2 FUTURE SCOPE

This comparative study is done for 3 different mineral oils and 2 different vegetable oils namely SAE10, SAE90, SAE10W30, Soyabean, Palm 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 effect 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 analyze them in better way.

REFERENCES

1. Mohamad Nor Musa, Syahurallai Samion, “A comprehensive review on Palm oil and the challenges using vegetable oil as lubricant base stock”.
2. Mr. Pramod, Akash.S. Patil, “Pressure distribution and friction coefficient of Hydrodynamic journal bearing.
3. A.Pratama, Jamari., “Hydrodynamic lubrication of textured journal bearing considering slippage”.
4. Arti Singh, “Performance of hydrodynamic journal bearing with various types of lubricant for pressure distribution and cavitation”.
5. S.A. Jagtap , M.J.Gitay ,“Performance of bio oil in journal bearing instead of synthetic oil”.
6. S.Kamitani and A.Shakirin ,“Performance of vegetable oil as lubricant in extreme pressure condition”.
7. Sanket kumar Saxena, V.K.Chibbler “Comparative study- a mineral oil based lubricant and lubricant obtained from Vegetable oil”.
8. Chetan Mehra, Amar singh kokadiya., “Study of pressure profile in hydrodynamic journal bearing”.