

A
Project
On

Analysis of Hydrodynamic Journal Bearing

By Group B03

Ms. Jaju Siddhi Manoj B28
Ms. Kuyate Pranjal Suresh B38
Ms. Shewale Ankita Vijay B64
Mr. Vishwakarma Aditya Chandrashekhar B69

Guided by
Prof. P.B. Surwade



Department of Mechanical Engineering

K.K. Wagh Education Society's

K.K. Wagh Institute of Engineering Education and Research,

Nashik, 422003

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C E R T I F I C A T E

This is to certify that *Ms. Jaju Siddhi Manoj, Ms. Kuyate Pranjal Suresh, Ms. Shewale Ankita Vijay, Mr. Vishwakarma Aditya Chandrashekhar* has successfully completed the seminar entitled “*Analysis of Hydrodynamic Journal Bearing*” under my supervision, in the partial fulfillment of Bachelor of Engineering-Mechanical Engineering of Savitribai Phule Pune University.

Date:

Place:

Prof. P.B. Surwade

Guide

Prof. M. B. Murugkar

Department Head

External Examiner

Dr. K. N. Nandurkar

Principal

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Ms. Jaju Siddhi Manoj B28

Ms. Kuyate Pranjal Suresh B38

Ms. Shewale Ankita Vijay B64

Mr. Vishwakarma Aditya Chandrashekhar B69

ABSTRACT

Journal bearings are widely applied in different rotating machineries. These bearings allow for transmission of large loads at mean speed of rotation. There are several types of journal bearing designs commonly used in machineries such as hydrodynamic journal bearing, which is based on hydrodynamic lubrication. Hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact. The aim of this project is to analysis various types of methods, equations and theories used for the determination of load carrying capacity, minimum oil film thickness, friction loss, and temperature distribution of hydrodynamic journal bearing.

Thus the main objective of this study is to analyze the pressure distribution on hydrodynamic journal bearing for various loading conditions and various operating parameters and do mathematical modeling of journal bearing and compare the results using MATLAB software. Predictions of these parameters are the very important aspects in the design of hydrodynamic journal bearings.

Keyword: Lubrication ,Mathematical Modeling, Analysis.

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1.INTRODUCTION

1.1 What is lubrication?

Lubrication is the process or technique of using lubricant reduce friction and wear and tear in a contact between two surfaces. The study of lubrication is a discipline in the field of tribology. Lubrication is the control of friction and wear by the introduction of a friction-reducing film between moving surfaces in contact. The lubricant used can be a fluid, solid, or plastic substance. Although this is a valid definition, it fails to realize all that lubrication achieves. Many different substances can be used to lubricate a surface. Oil and grease are the most common. Grease is composed of oil and a thickening agent to obtain its consistency, while the oil is what lubricates. Oils can be synthetic, vegetable or mineral-based as well as a combination of these. The primary purpose of lubrication is to reduce wear and heat between contacting surfaces in relative motion. While wear and heat cannot be eliminated, they can be reduced to negligible or acceptable levels. Because heat and wear are associated with friction, both effects can be minimized by reducing the coefficient of friction between the contacting surfaces. Lubrication is also used to reduce oxidation and prevent rust; to provide insulation in transformer applications; to transmit mechanical power in hydraulic fluid power applications; and to seal against dust, dirt, and water.

1.2 Types of lubrication

1.2.1 Hydrodynamic Lubrication

Hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal to metal contact. Hydrodynamic lubrication does not depend upon the introduction of the lubricant under pressure, though that may occur, but it does require the existence of an adequate supply at all times. The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create

the pressure necessary to separate the surfaces against the load on the bearing. Hydrodynamic lubrication is also called full-film, or fluid lubrication.

1.2.2 Hydrostatic Lubrication

Hydrostatic lubrication is obtained by introducing the lubricant, which is sometimes air, water or oil into the load-bearing area at a pressure high enough by a pump to separate the surfaces with a relatively thick film of lubricant. So, unlike hydrodynamic lubrication, this kind of lubrication does not require motion of one surface relative to another.

1.2.3 Elasto-hydrodynamic lubrication

Elasto-hydrodynamic lubrication is the phenomenon that occurs when a lubricant is introduced between surfaces which are in rolling contact, such as mating gears or rolling bearings. The mathematical explanation requires the Hertzian theory of contact stress and fluid mechanics.

1.2.4 Boundary Lubrication

Insufficient surface area, a drop in the velocity of the moving surface, a decreasing in the quantity of lubricant delivered to a bearing, an increase in the bearing load, or an increase in lubricant temperature resulting in a decrease in viscosity-anyone of these may prevent the buildup of a film thick enough for full-film lubrication. When this happens, lubricant films may separate the highest asperities only several molecular dimensions in thickness. This is called boundary lubrication. The change from hydrodynamic to boundary lubrication is not at all a sudden or abrupt one. It is probable that a mixed hydrodynamic- and boundary-type lubrication occurs first, and as the surfaces move closer together, the boundary type lubrication becomes predominant. The viscosity of the lubricant is not of as much importance with boundary lubrication as is the chemical composition.

1.2.5 Solid Film Lubrication

When bearings must be operated at extreme temperatures, a solid-film lubricant such as graphite or molybdenum disulfide must be used because the ordinary mineral oils are not satisfactory. Much research is currently being carried out in an effort to find composite bearing material with low wear rates as well as small frictional coefficients.

1.3 Lubricants – It's Function and Types

A lubricant is a substance, usually organic, introduced to reduce friction between surfaces in mutual contact, which ultimately reduces the heat generated when the surfaces move. It may also have the function of transmitting forces, transporting foreign particles, or heating or cooling the surfaces. The property of reducing friction is known as lubricity. In addition to industrial applications, lubricants are used for many other purposes. Other uses include cooking (oils and fats in use in frying pans, in baking to prevent food sticking), bio-applications on humans (e.g. lubricants for artificial joints), ultrasound examination and medical examination. It is mainly used to reduce friction and to contribute to a better and efficient functioning of a mechanism.

Types of lubricants

1.3.1 Gaseous lubricants

Gaseous lubricants belong to the simplest, lowest viscosity lubricants known and include air, nitrogen, oxygen, and helium. They are applied in aerodynamic and aerostatic bearings. Since the chemical properties and the aggregate state of most gases remain unchanged over a wide temperature range, gaseous lubricants offer several advantages over liquid lubricants. First, they can be applied at both very high and very low temperatures. Their chemical stability eliminates any risk of contamination of the bearing by the lubricant, important for the machinery used in many branches of industry, primarily in the food, pharmaceutical and electronic industries.

1.3.2 Liquid lubricants

Mineral oils: As the hydrodynamic behavior of plain bearings of plain bearings is totally dependent on the viscosity characteristics of the lubricant, typical liquid bearing lubricants are straight mineral oil raffinates of various viscosity grades. Mineral oil is any of various colorless, odorless, light mixtures of higher alkanes from a mineral source, particularly a distillate of petroleum, as distinct from usually edible vegetable oils. The name mineral oil by itself is imprecise, having been used for many specific oils over the past few centuries. Other names, similarly imprecise, include white oil, paraffin oil, liquid paraffin (a highly refined medical grade), paraffinum liquidum (Latin), and liquid petroleum. Baby oil is a perfumed mineral oil.

1.3.3 Synthetic lubricants

In practice, every synthetic oil of adequate viscosity and good viscosity temperature behavior can be used as a bearing lubricant, e.g. polyglycols are very good bearing lubricants for mills and calendars in the rubber, plastics, textile and paper industries. However, in most cases the synthetic oils specifically developed for lubricating particular equipment are also used to lubricate its bearings. Although synthetic oils do not form a lubricant film under pressure as well as mineral oils and may not be effective bearing lubricants despite their higher temperature viscosity.

1.3.4 Biodegradable products

Biodegradable products of vegetable or animal origin are also considered for liquid lubrication, e.g. the effects of sunflower oil added to base oil on the performance of journal bearings. The use of vegetable oils as lubricants is likely to increase due to environmental and government requirements and is becoming increasingly important.

1.3.5 Solid lubricants

General description: bearings used under vacuum, at very high temperatures or under very high radiation cannot be lubricated by liquid lubricants or greases. For these and many other cases, solid lubricants are used, deemed to be any solid material used to reduce friction and wear between two moving surfaces. In general, the solid material is interposed as a film between sliding and /or rolling surfaces.

Simply stated, an adequate solid material is required for the special lubrication requirements of extreme operating conditions, such as very high or very low temperatures over a wide range, e.g. 200 to 850oC, and corrosive atmospheres. Such materials normally have a layered crystalline structure which ensures low shear strength, thereby minimizing friction. The shear strength between the crystalline layers is weak and sets up a low and sets up a low friction mechanism by slippage of the crystalline layers under low shearing forces. Examples of layer lattice solids are molybdenum disulphide, graphite, boron nitride, cadmium iodide and borax. Solid lubricants are used mainly in the form of powders or as bonded solid films.

1.3.6 Functions of a Lubricant

1. The primary function of a lubricant is to prevent friction by creating a boundary layer between two surfaces
2. Dissipate heat from surfaces
3. Transport contaminants to filters
4. Protects from oxidization and corrosion
5. Power transmission.

1.4 Problem Statement :

To study the lubrication and impact of other factors on hydrodynamic journal bearing

1.5 Objective :

1. To understand lubrication and journal bearings.

2. To study the various factors affecting the performance of bearing.
3. To study mathematical modeling of journal bearing and compare the results using various software's.
4. To understand tribological behavior of journal bearing material under various conditions.
5. The main objective of this study is to analyze the pressure distribution on hydrodynamic journal bearing under different lubricants for various loading conditions and various operating parameters.
6. To study the heat transfer effect in hydrodynamic journal bearing.

1.6 Scope

The rapid industrialization in the world of increasingly sophisticated engineering and continuous demand of automotive industry globally are one of the major reasons for driving the lubricant market worldwide.

1. Further improvement in machine performance can be provided through the application of new lubricants in real machines.
2. A theoretical model should be developed for predicting minimum oil film thickness in a dynamic system with radial clearance as a time variant. Such a model would be helpful in developing an expert system for condition monitoring of machines operating in dusty environment.
3. A wider variety of anti-wear additives should be tested to characterize for the benefits of industrial users.
4. A study of reduction in friction due to anti-wear additives needs to be pursued, with regards to energy saving industrial applications.
5. To achieve super lubricity at larger scales is important for potential application in industries.

There should also be greater emphasis on cleaning and reuse of used lubricating oil in

Future.

6. Besides lubrication technology, developing and application of new composited material, antiwear coating, and surface modification is the trend for improvement of wear resistance of materials.
7. A number of similarities either from material or methodologies point of view can be found and this will help further develop research in bio tribology.

2.LITERATURE REVIEW

Lois j. Gschwender, brent k. Lok, shashi k. Sharma, carl e. Snyder, jr.mark l. Sztenderowicz did studied and explained that that a lubricant is a substance used to reduce friction between moving surfaces. The property of reducing friction is known as lubricity. Lubricating oil, sometimes simply called lubricant/lube, is a class of oils used to reduce the friction, heat, and wear between mechanical components that are in contact with each other. Lubricating oil is used in motorized vehicles, where it is known specifically as motor oil and transmission fluid. Further they explained about conventional lubricants and are defined as those that have historically been formulated with mineral oils derived from crude oil. The three common processes to improve the base oil process were clay treating, acid treating & so₂ treating. Bu by 1930 solvent processing emerged as a viable technology for improving base oil performance using a fairly safe, recyclable solvent. Over the next several decades, the solvent refining process did not change very much. Improvements in finished oil quality came mainly from the appearance of additives. Additives began to be widely used in 1947 when the api began to categorize engine oils by severity of service: regular, premium, and heavy duty. Additives were used to extend the life only in premium and heavy-duty oils. Hydro-treating was developed in the 1950s and first used in base oil manufacturing in the 1960s by amoco and others. It was used as an additional “clean-up” step added to the end of a conventional solvent refining process. Hydro-treating added hydrogen to the base oil at elevated temperatures in the presence of catalyst to stabilize the most reactive components in the base oil, improve colour, and increase the useful life of the base oil. Hydrocracking is a more severe form of hydro-processing. It is done by adding hydrogen at even higher temperatures and pressures than simple hydro-treating. In short they tried to explain that the conventional technologies evolved from ancient times till the medieval period.

1.Lubrication & lubricant's –(Nehal s. Ahmed & Amal m. Nassar -2013)

Author nehal s. Ahmed and amal m. Nassar studied various references and stated that the primary purpose of lubrication is to reduce wear and heat between contacting surfaces in relative motion. But wear and heat cannot be completely eliminated; they can be reduced to negligible or acceptable levels. Because heat and wear are associated with friction, both effects can be minimized by reducing the coefficient of friction between the contacting surfaces. Lubrication is also used to reduce oxidation and prevent rust; to provide insulation in transformer applications; to transmit mechanical power in hydraulic fluid power applications; and to seal against dust, dirt, and water. Adequate lubrication also helps to prevent foreign material from entering the bearings and guards against corrosion and rusting. Satisfactory bearing performance can be achieved by adopting the lubricating method that is most suitable for the particular application and operating conditions. Lubricant is consisting of either oil or grease. Most grease is from animal fats or vegetable lard. There are three major types of lubricants: gaseous lubricants e.g. Air, helium, liquid lubricants e.g. Oils, water and solid lubricants e.g. Graphite, grease, molybdenum disulphide etc. Liquid lubricant is the most commonly used lubricant because of its wide range of possible applications while gaseous and solid lubricants are recommended in special application.

2.Lubricating oil (22, 1937)

charles c. Swoope. & marn m. Sadlon did explained the improvements in the art of producing lubricating oils of low pour point of wax-containing lubricating oils by the use of pour inhibitors. They claimed that a low pour point lubricant, comprising a blend of two petroleum lubricating oils having average molecular weights differing by at least 150 and a pour inhibitor. A low pour point lubricating oil, comprising a blend of a straight cut oil and a minor quantity of a heavier petroleum lubricating fraction having an average molecular weight greater by not less than 200, together with a pour inhibitor. Low pour point lubricating oil, comprising spindle oil with from 3 to 50% of

a natural bright stock and a hydrocarbon pour inhibitor. A composition of matter comprising neutral oil having a viscosity below 400 seconds say bolt at 100°f with a minor proportion of a heavier natural oil having a viscosity in excess of 100seconds at 210 f. And a pour inhibitor. A composition of matter comprising a wax bearing neutral oil having a viscosity from about 60 to 400 seconds at 100°f with a minor proportion of a bright stock and a pour inhibitor. A composition of matter comprising spindle oil with a minor proportion of bright stock and a pour inhibitor of the type produced by the low temperature aluminum chloride condensation of an active wax derivative. An improved process for lowering the pour point of a narrow cut waxy oil comprising adding to such oil a substantial portion of another lubricant which has an average molecular weight higher by at least 150 than that of the said narrow cut oil, whereby the molecular weight range of the blend is greatly increased, and adding a pour inhibitor thereto. Process according to claim 10 in which the oil of higher molecular weight than the narrow cut oil is added in proportion of from about 3 to 50% of the blend. Process according to claim 10 in which the narrow cut oil is spindle oil and the other lubricant is a bright stock.

3. Lubricating oil composition (jan. 31, 2006)

Douglas e. Deckman, mullica hill, william l. Maxwell, pilesgrove, william h. Buck, mark d. Winemiller, clarksboro, david j. Baillargeon, cherry hill they all did invention and the invention provides for a lubricating oil composition comprising a base oil and the instant viscosity indexenhancing additives. It relates to the lubricating oil composition suitable for use in internal combustion engines. Engine oils contain base lube oil and a variety of additives. These additives include detergents, dispersants, friction reducers, viscosity index improvers, antioxidants, corrosion inhibitors, anti-wear additives, pour point depress sants, seal compatibility additives, anti-corrosion, and anti-foam agents. To be effective, these additives must be oil soluble or oil-dispersible. By oil-soluble, it is meant that the compound is soluble in the base oil or lubricating oil composition under normal blending or use conditions. In a first aspect

the invention relates to a viscosity index improver additive composition. Yet other aspect of this invention is a means to provide an unexpected increase in high-temperature high-shear (hths) viscosity when combining hydrocarbyl aromatics with olefin oligomers. Another aspect of this invention is that when the olefin oligomer is added to hydrocarbyl aromatics, pao, or hydro processed base stock, the resulting mixture surprisingly has newtonian high-temperature and low-temperature visco-metric properties, providing significant additional potential performance characteristics to the instant invention.

4. Lubricating oil composition (may 22, 1962)

Joseph arthur verdo & dolton their invention relates to the improvement of mineral oil compositions, and more particularly to the improvement of lubricating oils containing basic salts of oil soluble sulfonates by the addition thereto of an effective inhibitor-detergent type additive agent. They claimed that lubricating oil composition consisting essentially of a mineral lubricating oil containing minor detergent 50 amounts of a basic alkaline earth metal petroleum sulfonates and a minor amount of an oil-soluble inhibitor detergent agent selected from a member of the group consisting of a condensation product and the alkaline earth metal salts thereof obtained by reacting an alkyl-substituted phenol, in which the alkyl group contains from about 4 to 20 carbon atoms, and formaldehyde with an amine material selected from the group consisting of polymerized ethylene imines having a molecular weight of about 30,000 to 40,000 and an alkaline polyamine.

5. The reynolds centennial: a brief history of the theory of hydrodynamic lubrication (oscar pinkus- 1987)

This paper offers a brief review of the history of the theory of hydrodynamic lubrication. The crystallization of the concept started with nicolai petrov whose main

interest was in the area of friction. He proposed the hydrodynamic nature of friction in bearings. The geometry and operating condition of the first bearing was tested by beauchamp tower. Petrov, tower, and reynolds can be considered the founding fathers of the concept of hydrodynamic lubrication. Discovery was made by albert kingsbury that the fluid film does not have to be oil or a liquid, but that it can be a gas. Kingsbury went on to construct a special bearing 152.4 mm (6 in.) In diameter with a radial clearance of 0.02 mm (0.8 mils) carrying a load of 222.4 n (50 lb) which he ran on both air and hydrogen. He is also known as the inventor of tilting pad bearings.

6. Analysis of hydrodynamic journal bearing: a review (7, september - 2012)

Priyanka tiwari and veerendra kumar studied and presented a survey of important papers pertaining to analysis of various types of methods, equations and theories used for the determination of load carrying capacity, minimum oil film thickness, friction loss, and temperature distribution of hydrodynamic journal bearing. The bearing mostly used in various rotating machines such as pumps, generators etc. In journal bearing, a circular shaft called journal is made to rotate in a fixed sleeve called bearing. The clearance space between journal and bearing is assumed to be full of lubricant. The paper mainly focuses on the factors which affect the performance of hydrodynamic journal bearing. The parameters such as pressure gradient, presence of oil grooves, flexibilities of backing and housing materials, non-uniform housing supports and non-uniform temperature distributions have been individually considered as the sources of stresses in journal bearings. Two-dimensional and three-dimensional bearing have been modelled by using finite element modelling. Brinkman model is used to predict the influences of viscous shear stresses on the bearing. The oil film pressure is numerically calculated by the fourth runge-kutta method and this pressure is utilized to evaluate the load carrying capacity and the friction parameter. A comparison of the results between the darcy model and brinkman model is made to show the viscous shear effects provide an increase in the load capacity, as well as a decrease in the friction parameter. By solving

navier-stokes equation with the aid of the simpson rule, calculated the pressures, drags and load carrying capacities and predict their comparison at different viscosity ratio. The steady-state oil film pressures are obtained by using reynolds equation with the help of finite difference method. Hydrodynamic bearings are known for initial wear due to direct contact between bearing surfaces. This problem is overcome by assembling hydrodynamic bearing with rolling bearing, separated by a fixed clearance to form a journal-rolling hybrid bearing (jrhb). The journal is supported by the rolling bearing. The friction coefficient is increased, with increasing wear depth as well as misalignment and sommerfeld number. The friction coefficient and consequently the power loss are strongly dependent upon the misalignment angle and wear depth.

7. Lubrication of journal bearing during clockwise and counter-clockwise rotation (apr. 22, 2014)

A gear-turbofan engine consists of an epicycle gear system coupling the turbine to the fan. In this manner, both the fan and the turbine can operate at each components own optimum speed. The fan and the turbine may be coupled to one another through a gear train that is supported by a journal bearing system. In one embodiment, a gear system is disclosed, comprising: a shaft; and a lubrication system, the lubrication system comprising: a gear including a gear bearing surface, the gear operatively driven by the shaft; and a pump operatively driven by the gear, the pump including a first pump port and a second pump port wherein rotation of the gear below a predetermined operational speed range in either direction causes the pump to transfer lubricant to the gear bearing surface. In another embodiment, a turbofan engine is disclosed, comprising: a fan shaft operably coupled to the fan a gear including a gear bearing surface, the gear operatively driven by the fan shaft; and a pump operatively driven by the gear, the pump including a first pump port and a second pump port wherein rotation of the fan shaft in either direction causes the pump to transfer lubricant to the gear bearing surface when the engine is in a non-operational mode.

8. Hydrodynamic lubrication (Jaywant H Arakeri and K R Sreenivas 1996)

Friction plays a large and essential role in everyday life although we usually never think about it. For example, friction provides the support when we walk; without it we would not be able to move forward and indeed it would be impossible to stand up without additional support. When we grip an object and stop it from falling we again use friction. Friction between surfaces may be reduced by lowering the coefficient of friction; or it may be reduced by introducing a new substance a lubricant between the surfaces. Hydrodynamic lubrication is one method used extensively to support load and reduce friction. To understand the principal of lubrication consider a block sliding on horizontal surface. The force required to move the block is frictional force and is given by μw . If we now put a liquid film of thickness h between the block and the table surface the force required to move the block with constant velocity u is $p = \eta \frac{u a}{h}$; η is the coefficient of viscosity of the liquid and a is the bottom surface area of the block. The block, as it moves forward, drags the liquid into the gap. This liquid has to move into a gap which is narrowing and the pressure that builds up in the gap supports the load. There are two conditions under which hydrodynamic lubrication and the associated bearing action occur. One is that the reynolds number be small, i.e., when $\frac{\rho u h^3}{\eta} \sim 1$. Here, ρ is the density of the fluid in the gap. The second condition is that the angle of the narrowing passage must be low.

9. On the effects of surface roughness in the hydrodynamic lubrication theory of a short journal bearing (s. T. Tzeng and e. Saibel-1966)

The effects of surface roughness are important parameters in hydrodynamic lubrication. The effect of roughness of the surfaces of a journal bearing on the pressure development, load-carrying capacity, attitude and friction is studied. Roughness is treated as a random quantity, characterized by a probability density function which can be determined experimentally. An increase of the loading capacity occurs when the

roughness of the surface is taken into account. The frictional force also increases but less significantly than the loading capacity. As a consequence, the coefficient of friction decreases. But producing a very smooth surface leads to a less satisfactory bearing from a hydrodynamic point of view.

10. Understanding journal bearings (Malcolm e. Leader & durango)

This paper covers the basic aspects of journal bearings including lubrication, design and application bearings are used to prevent friction between parts during relative movement. Lubrication technology goes hand-in-hand with understanding journal bearings and is integral to bearing design and application. Since they have significant damping fluid film journal bearings have a strong impact on the vibration characteristics of machinery. There are applications where anti-friction bearings are the best choice. Commonly, smaller motors, pumps and blowers use rolling element bearings. The primary advantage of a fluid film bearing is often thought of as the lack of contact between rotating parts and thus, infinite life. If applicable, provide proper filtration to the lubricant. Understand the additive package in your lubricants to avoid potential conflicts with process fluids and/or component materials.

11. Synthetic lubricants in hydrodynamic journal bearings: experimental results (2011)

G.f. simmons and s.b. glavatskih did studied and constructed a new full-scale hydrodynamic journal bearing test rig to evaluate the behaviour of conventional and new bearing designs, synthetic lubricants and variations in operating parameters. The lubricants used in this study include standard mineral-oil based turbine oils (vg68 and vg32), as well as synthetic ester based oils with viscosity index improvers (se32, se22, sv22, and se15). The study results generally demonstrate that synthetic lubricants in combination with vi improvers provide improved bearing operation. They also

concluded that high vi iso vg15 synthetic ester lubricant offers an alternative to iso vg32 mineral oil at higher speeds such as for gas and steam turbines, providing reductions in power loss with equivalent lubricant film thickness. With further improvement in viscosity index improvers and base oil development, similar lubricants could provide better performance than the mineral based lubricants at both high and low speeds. High vi iso vg22 synthetic ester lubricant offers an alternative to iso vg32 mineral oil in low speed, high load applications such as hydroelectric power plants or shipping, providing reductions in power loss and equivalent film thickness. However, at higher speeds, this lubricant results in greater power loss due to the thicker film it provides.

12. Experimental analysis of pressure distribution of hydrodynamic journal bearing: a parametric study (2005)

Chaitanya K Desai and Dilip C Patel did study the pressure distribution in hydrodynamic journal bearing for various loading conditions and various operating parameters. The space between the shaft and the bearing is called lubrication gap and is filled with lubricant. They did an experiment using journal bearing test rig, which used 140 mm diameter and 70 mm long bearing. Test bearing is located between two antifriction bearings. The bearing is loaded mechanically. The bearing is tested under various parameters like type of lubricant, loading conditions, speeds etc. After conducting the experiment they concluded that that maximum pressure obtained where oil film thickness is minimum & zero pressure in cavitation zone. Also as the speed and load on the bearing increase, the pressure also increases. They further added that pressure at the periphery of bearing follows the sinusoidal form, theoretically as well as practically.

13.Tribological behaviour of journal bearing material under different lubricants (2014)

S. Baskar, g. Sriram did study of mechanics of friction and the relationship between friction and wear. The friction and wear behaviour of journal bearing material has been investigated using pin on disc wear tester with three different lubricating oils i.e. Synthetic lubricating oil (sae20w40), chemically modified rapeseed oil (cmro), chemically modified rapeseed oil with nano cuo. The bearing material (brass) lubricated with cmro + 0.5 w.% nano cuo has the lowest friction coefficient of 0.073. The frictional coefficient of bearing material lubricated with cmro is 0.13 and sae20w40 is 0.09. The frictional coefficient of cmro + 0.5 w.% nano cuo is 49 % lesser than cmro and 18 % lower than sae20w40. The wear of bearing material lubricated with sae20w40, cmro and cmro + 0.5 w.% nano cuo of 86.77, 136.34 & 82.07 mg. The wear value of bearing material lubricated with cmro + 0.5 w.% nano cuo has lowest wear and 39 % lesser than cmro. The wear value of bearing lubricated with cmro + 0.5 w.% nano cuo has 5 % lesser than sae20w40. It is also possesses superior tribological behaviour in chemically modified rapeseed oil with nano cuo than the other two lubricating oils. The above mentioned discussions are evaluated, it can be stated that among the three lubricating oils, one can contain nano cuo can be preferred for the lubrication purpose in journal bearing application.

14.Manufacturing process of lubricating oil

T. M. Aboul- fotouh, m. A. El-shafie studied the first process production of medium lubricating oil base stocks and the study of the effect of chemical composition on physico chemical characteristics of the prepared lube oil samples. In the first process, solvent dewaxing, the mixture of methyl ethyl ketone and methyl isobutyl ketone (mek/mibk) is used as a recent solvent in the dewaxing process to obtain a significant yield of low pour point products and to compare this mixture with mek/toluene which

is used in the industry process. Solvent dewaxing process is carried out at -10°C using the mixture of mek and mibk (40/60 by wt.) With solvent oil ratio of 3:1 for the selected raffinates, medium refined oils, from furfural and nmp extraction processes. The dewaxed oil samples were analysed to evaluate their physico-chemical characteristics by the standard methods outlined in astm, ip and uop books such as the standard methods for wax content, pour point, sulfur content and structural group analysis by n-d-m method. The results show that the use of mek/mibk mixture instead of mek/toluene at -10°C achieves an increase in filtration temperature with about 5°C with subsequent a reduction in the cooling rate of process. John j. Mcketta studied that many liquids including water have been used as lubricants to minimise friction, heat, and wear between mechanical parts in contact with each other. The two basic categories of lube oil are mineral and synthetic. Mineral oils are refined from naturally occurring petroleum or crude oil. Synthetic oils are manufactured polyalphaolefins, which are hydrocarbon-based polyglycols or ester oils. the advantage of mineral based lube oils is that they can be produced in a wide range of viscosities for diverse applications. They range from low-viscosity oils, which consist of hydrogen-carbon chains with molecular weights of around 200 atomic mass units (amu), to highly viscous lubricants with molecular weights as high as 1000 amu. Mineral based oils with different viscosities can even be blended together to improve their performance in a given application. First used in the aerospace industry, synthetic lubricants are usually formulated for a specific application. The raw material for producing lube oil is raw petroleum. Lube oils are just one of the many fractions or components that can be derived from raw petroleum, which emerges from an oil well as a yellow-to-black, flammable, liquid mixture of thousands of hydrocarbons.

15.Experimental research on the impact of lubricating oils on engine friction and vehicle fuel economy; yimin mo; 2015

The engine friction loss and vehicle fuel economy aiming at several kinds of energy-conserving engine oils with different quality standard, viscosity grade, friction modifier, and viscosity index improver were tested in this paper. Experimental results showed that the engine friction loss was reduced and vehicle fuel economy was improved by lowering the viscosity of engine oil and adding high-performance friction modifier and new-type viscosity index improver. Among which, the effect of energy-conserving engine oils dexos1 5w-20 adding 1% friction modifier and new type viscosity index improver was most significant with 12.45% engine friction reduction rate and 2.33% vehicle fuel economy improvement rate.

16.Experimental analysis of lubricating oil using nanoparticles as modifiers for tribological properties; mr. Nikam m.m; mar2018

Nanoparticles are proved to be useful as modifiers in the engine oil to enhance its tribological properties to reduce wear and friction of the engine. In this research molybdenum disulphide (mos_2) and copper oxide (cuo) nanoparticles are added to engine oil sae20w50 and tribological properties are analyzed. Test samples were made with varying percentage of mos_2 and cuo nanoparticles in engine oil (0.25, 0.5 and 1 wt. %). Redwood viscometer was used to carry out viscosity tests for different samples at three different temperatures. The flash point and fire point tests were carried on clever land open cup apparatus. The experimental observations show that mos_2 , cuo nanoparticles added in engine oil exhibits good friction reduction and antiwear properties and also decreased the coefficient of friction as compared with standard engine oil without mos_2 , cuo nanoparticles. This tribological behavior is closely related to the deposition of nanoparticles on the rubbing surfaces

17.Mathematical modeling of oil lubricants

The aim of this review paper is the presentation of the main ideas and recent advances in the modeling, mathematical analysis and numerical simulation of different processes taking place in lubrication technology. The mathematical modeling is mainly based on different versions of reynolds equation, which represents an asymptotic limit of fluid mechanics models for thin films. A naive deduction of the more classical linear reynolds equation from navier-stokes, the rigorous approach being mainly based on asymptotic expansions techniques. The scientific study of the three main aspects of tribology (friction, lubrication and wear) was initiated at very different dates. Thus, the basic laws of friction were first correctly deduced by da vinci (1519), followed by important advances provided by amontons (1699) and coulomb (1785). Later on, in 1886, the theory of lubrication starts with the work of reynolds [143], where he heuristically deduces the reynolds equation, which is the key point in the mathematical modelling of thin film lubrication processes. More recently, the origins of the theoretical study of wear are associated to archard's formula.

Hui-hui feng, chun-dong xu, and jie wan done mathematical modeling of the water-lubricated hydrostatic journal bearings.the water-lubricated bearings have been paid attention for their advantages to reduce the power loss and temperature rise and increase load capacity at high speed. To fully study the complete dynamic coefficients of two water-lubricated, hydrostatic journal bearings used to support a rigid rotor, a four-degree-of-freedom model considering the translational and tilting motion is presented.

18. Mathematical modeling of engine new and used oils ; yousef alhouli ;oct 2015

Engine is a complex machine; it consists of a hundreds of moving parts. The parts of engine are operated under a wide temperature ranges and pressure. The basic idea of using oil in any machine is reducing friction and wear, so the friction power drop in any machine depends on oil conditions and specifications, the most important property of oil is viscosity, viscosity shows the amount of forces between oil

particles. In this paper experimental apparatus is used to simulate the relation between oil viscosity and temperature, the experiment data are used to produce mathematical formula for different engine oils with different conditions. Engine oils 10w-40, 5w-30, and 20w-50 are selected for testing and simulation because these types are widely used in kuwait market for engines. The svm 3000 viscosity measuring unit is used to measure viscosity and density of engine oils 10w-40, 5w30, and 20w-50 in new and used conditions, the results show that there is no change in oil viscosity for new oil and used oil.

19.A review paper on performance analysis of hydrodynamic journal bearing with various types of lubricant for pressure distribution and cavitation ; arti singh, prof.s.s.waydande ; aug2017

Journal bearings are widely applied in different rotating machineries. These bearings allow for transmission of large loads at mean speed of rotation. There are several types of journal bearing designs commonly used in machineries such as hydrodynamic journal bearing, which is based on hydrodynamic lubrication. Hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact. The most important objectives of bearing design are to extend bearing life in machines, reduce friction energy losses and wear, and minimize maintenance expenses and downtime of machinery due to frequent bearing failure. From the literature review and previous investigation, it is found that the cavitation area in hydrodynamic journal bearing is very essential area of investigation because it reduces the load capacity and leads to a risk of material damages. When a bearing operates at maximum speed, the heat generated due to large shear rates in the lubricant film increase its temperature, which decrease the viscosity of the lubricant and it affects the performance of journal bearing. Different combination of nano particle concentration is studied and it had

been found that by addition of nanoparticle, results in high viscosity, high pressure distribution and ultimately high load carrying capacity of journal bearing. The determination of the cavitation boundaries are important to get a realistic model of a hydrodynamic journal bearing with various nanofluid lubricants such as CO_2 , TiO_2 , ZnO , SiC and Al_2O_3 . For theoretical analysis mostly Sommerfeld equation is used for finding the pressure distribution and Reynolds equation is used to get the cavitation location. From the previous research paper it is studied that various experimental analysis is done for finding the pressure distribution, cavitation location and cavitation shape.

20. Study of pressure profile in hydrodynamic lubrication journal bearing Chetan Mehra, Amar Singh Kokadiya; July 2014

-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 $\frac{1}{2}$ 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.

21. An experimental investigation of hydrodynamic journal bearing with different oil grades; Nour Marey; Sept 2019

Evaluating the behaviour of conventional and new bearing designs together with synthetic lubricants in operating parameters has been the main concern of the researchers Simmons, 2011. Synthetic lubricants were proved to be superior to their

mineral based counterparts characterized by higher viscosity grade. High vi iso vg32 synthetic ester lubricant has proved its effectiveness and success in replacing iso vg32 mineral oil. While keeping an equivalent lubricant film thickness, it could as well effect a reduction in power loss. The results obtained have ascertained the efficiency of such lubricants in improving performance if compared with the mineral based lubricants at both high and low speeds. It could reduce power loss and offered an equivalent film thickness. Also, greater power loss was noted at high speeds as a result of the thicker film it provided. (simmons, et al, 2011). Examining the tribological characteristics concerning journal bearings, under boundary and mixed lubrication conditions during shaft start-up, shutdown and low speeds has been the incentive that pushed forward the research presented by pickering, 2011. On tackling the issue of wear, it was proved that performance and pressure loads would negatively be affected in the existence of wear. To overcome the undesirable consequences of wear on performance, some suggestions have been made. Such solutions have involved surface hardening, polymer liners in addition to using lubrication fluids with additives. The potential solutions could ultimately help promote the product reliability and could as well optimize journal bearing design. The author stressed the need of a thorough investigation and a comprehensive understanding of that wear so that bearing life could be improved and costs could be reduced. Using lubrication additives as well as enhanced materials were just but a few of the solutions recommended on the road to reaching the aspired optimal working conditions.

Allmaier et al. 2011 have launched a beneficial and a systematic research with the aim of predicting friction in journal bearings. That was carried out in a reliable and an accurate way based on validated comparisons held between simulation results and experimental measurements. Two different loads were studied, namely, 41 and 70 mpa. The outcomes obtained have proved the effectiveness of optimum viscosity lubricants as opposed to other index lubricants in relation to helping ensuring friction

reduction. In other words, the results showed that reducing friction was, to a great extent, dependent on selecting a lubricant with reduced viscosity

22.Hydrodynamic journal bearing hooshang heshmat; dec 2000

A compliant hydrodynamic fluid film journal bearing for handling high loads. The bearing assembly includes a first thin smooth compliant sheet for facing a shaft, one or more corrugated foils, and a second sheet between the foils and the compliant sheet for supported underlying the compliant sheet for preventing sagging of the first sheet between ridges of the foils. A foil element is anchored at a position intermediate its ends to the bearing sleeve whereby both ends thereof are unanchored. A pair of radially outer and inner corrugated foils has outer foil ridges which underlie some inner foil ridges and outer foil furrows which underlie others of the inner foil ridges. A bearing assembly includes a plurality of rows of corrugated foil elements wherein ends of foil elements are offset circumferentially from ends of foil elements in adjacent rows symmetrically from the radial centerplane to provide a herringbone pattern effect. At least some ridges on at least one of inner and outer corrugated foils have truncated apex portions.

23.Study on the performance of journal bearings in different lubricants by cfd and fsi method with thermal effect and cavitation ; hulin li, yanzhen wang, ning zhong, yonghong chen and zhongwei yin ; 2018

This paper used a new transient computational fluid dynamics and fluid–structure interaction method to investigate the journal bearing performance with the effect of thermal and cavitation, to reveal the performance of journal bearing in different lubricants and to provide substitution references for bearings in different lubricants. Considering thermal effect, elastic deformation and cavitation, a detailed discussion was conducted to show the performance of plain journal bearings lubricated by water,

seawater, and lubricating oil by computational fluid dynamics (cfd) and fluid structure interaction (fsi) method. And the results in this work are compared with the published results. The variation of dimensionless load carrying capacity, maximum film pressure and temperature with eccentricity ratio are presented, which can provide reference for the design of bearings. Furthermore, a diagram is presented for journal bearings with different diameter, length-diameter ratio and lubricants, which can be used as a reference for the equivalent substitutions of bearings. The present research provides references as to the design of bearings and the substitutions of bearings by different lubricants.

24.Preliminary study of pressure profile in hydrodynamic lubrication journal bearing ; salmiah kaso;2018

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 $\frac{1}{2}$ 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.

25.The influence of bearing grease composition on friction in rolling/sliding concentrated contacts

There is very little published work on friction mechanisms in grease lubricated contacts. The studies focus on the effect of operating conditions, such as degradation

time, high or low temperature. Others have attempted to explain grease friction through rheometric. Only a few have explored the effect of grease chemical composition on its friction behaviour, on a thrust ball bearing, four-ball machine and ball-on-disc machine. Some of these studies have shown the importance of the base oil in grease formulation, with low viscosity and low pressure–viscosity oils suggested as preferred choice for low friction in ehl conditions. Furthermore, thickener type, composition and texture have also been shown to affect friction, with some thickeners able to reduce friction, particularly under boundary and high temperature conditions. However, it is clear that grease frictional behaviour cannot entirely be ascribed to one single component of grease, but rather it is determined by the interaction between all grease elements, which in turn strongly depends on the specific operating conditions. At present, the literature is missing a comprehensive investigation on the influence of grease composition on friction in concentrated ehl contacts operating under conditions pertinent to rolling bearings. In contrast, there is considerable work on grease ehl film thickness behaviour. Film thickness and friction are clearly related, and therefore, combined film thickness and friction analysis may help to explain some of the mechanisms of grease friction and suggest potential ways of improving frictional performance of bearing greases. This study aims to provide understanding of the role of the different grease components in determining friction and film thickness in non-conformal contacts pertinent to rolling bearings. As the focus of their search was lubricant composition rather than bearing factors, standard laboratory tests were used to measure friction coefficient and film thickness. The intention was to simulate the conditions (slide-roll ratio, contact pressure, temperature, speed, materials) in the contact between the rolling elements and the race way of a bearing. One of the fundamental factors determining lubrication and film formation with greases is the lubricant supply to the inlet of the contact [18]. Grease has a yield stress and, unless it is compelled by an external force, will not readily flow back into the race way after it has been displaced by passage of a rolling element, resulting in a set of lubricant

starvation, potential film thickness reduction and surface damage. Greased rolling bearings often operate in starved lubrication conditions, although these depend on operating, design and grease composition factors. In order to eliminate this critical variable, which greatly affects the repeatability of laboratory experiments, the tests described in this work were carried out in fully flooded conditions.

26. Effect of lubricating oil on tribological behaviour in pin on disc test rig

J.j. truhan et al. Have studied on the effect of oil condition and its effect on the friction and wear of piston ring and cylinder materials in a reciprocating bench test. Wear depth is good for wear measurement with compare to other method. M. Zheng et al. Have studied on the model for wear and friction in cylinder liners and piston rings. It is observed that there is good agreement between the predicted cylinder bore wear and measurement bore wear. J.j. truhan et al. Have studied on the laboratory test to evaluate piston ring and cylinder liner materials for their friction and wear behaviour in realistic engine oils. The result shows that jet a has higher wear & used 15w40 oil showed least wear. S.s. venetia et al. Has performed an experiment on pin and vee block test machine (falex) to measure coefficient of friction. Used engine oil has increased coefficient of friction at higher loads due to aging and contaminants. M.a. chowdhury et al. Have investigated and compared friction coefficient and wear rate of different steel material and observed that wear rate increases with increase of load and sliding velocity and mild steel offers highest wear rate. V. Laxshminarayana et al. Have study the influence of varying load on en6alloy steel when it is sliding against en31 alloy steel by using pin on disc apparatus. Behaviour of bronze and brass which have not large difference in micro hardness, the severe plastic deformation is also observed on the microstructure of the brass.

M. Laad et al. Investigate the tribological behaviour of titanium oxide (tio₂) nanoparticles as additives in mineral based multigrade engine oil by using pin-on-disc

tribo tester. For the smallest load 4 kg and minimum for the largest load 6 kg. With the use of cuo nanoparticles added with mineral oil, there was a significant reduction in both coefficient frictions (28.5 % approx.) And specific wear rate (70 % approx.). A.n. farhanah et al. Performed and experiment to investigate the performance of lubricants for an ic engine. Engine oil from three different manufacturers with the same sae viscosity grade (sae10w30) is used to compare the performance of lubricants. Experiments were performed by four ball wear tester for different temperatures (40 °c, 70 °c and 100 °c) and varied speed from 1000 rpm to 2500 rpm.

27.Pressure distribution analysis of plain journal bearing with lobe journal bearing

The current trend in industry is to run turbomachines at high speeds in order to make them compact and reduce mass. It is found that the performance of ordinary circular bearings is not very satisfactory. To improve the stability of these bearings, pressure dams are incorporated in these bearings. The analytical dynamic analysis has shown that the cylindrical pressure-dam bearings are found to be very stable. Also an experimental stability analysis of such types of bearings showed that the analytical stability analysis provides the general trends in the experimental data. The study of noncylindrical pressure-dam bearings such as finite elliptical, half elliptical, offset-halves, and three-lobe pressure-dam bearings have proved that the performance of the bearings is improved. Fredrick t. Schuller intended principally as a guide in the selection and design of anti-whirl bearings that must operate at high speeds and low loads in low-viscosity fluids such as water or liquid metals. However, the various fixed geometry configurations can be employed as well in applications where other lubricants, such as oil, are used and fractional-frequency whirl is a problem. The important parameters that affect stability were discussed for each bearing type, and design curves to facilitate the design of optimum-geometry bearings are included. J. Lund, a comparison of the stability of the different bearing configurations tested was

obtained. This volume treats three special bearing types selected for study because of their favourable stability characteristics and, hence, their potential for use in high speed rotating machinery applications. The three bearing types are, the three lobe journal bearing, the floating sleeve bearing with an incompressible lubricant, the floating sleeve bearing with a compressible lubricant. The volume gives extensive design data in form of charts and tables from which the bearing dimensions can be obtained for a given application. Data are given for bearing flow, friction power loss and the speed at which hydrodynamic instability sets in. In addition, two computer programs accompany the volume, and instructions and listings of the programs are included. The programs may be used to obtain data for cases not covered by the presented design data. G. Bhushan, multilobe bearings are found to be more stable than circular bearings. Rakesh sehgl, an experimental setup/rig was developed to investigate the behaviour of non-circular bearings. G. Bhushan, deal with a theoretical investigation of stability of four lobe bearing. Stanislaw strzeleck considered the rotors of turbo generators operate in 2-lobe journal bearings. These bearings can be designed with the same or different profile of upper and bottom lobe, e.g. The upper lobe has cylindrical and bottom one the offset profile. Chaitanya k desai and dilip c patel worked on the method of to analyze the pressure distribution in hydrodynamic journal bearing for various loading conditions and various operating parameters.

28.Oil lubrication on high-speed spindle bearing system

Oil lubrication on high-speed rolling bearing was examined by weck and koch . It was observed that the operational speed of spindle bearing systems with angular contact ball bearings is limited amongst other factors by the highest permissible rotary speed of the bearings. The revolution parameter limit of these bearings currently lies at about $n \cdot d_m = 1.5 \cdot 10^6 \text{ mm/min}$ if lubricated by the principle of minimum oil quantity. This paper will introduce how, by applying oil-air minimum quantity lubrication using a new oil-air supply, the revolutionary parameter limit was able to be increased to

approximately $n \cdot d_m = 1.8 \times 10^6 \text{ mm/min}$. The process of lubrication is highly complex which need to be investigated independently for different lubrication methods like grease lubricating, air lubricating and oil lubricating. A convenient way to account for starved lubrication was described in detail by jakobsson and flodeberg, elrod, chevalier, wijnant. Wedeven et al. Carried out the first experimental studies of starved contacts. In his work, he concluded that starvation in ehd contact is dependent on the location of the inlet boundary. He also stated that the central film thickness in the starved mode is meant as the function of position of the inlet meniscus from the hertz area. Wedeven et al. Observed in his experiments that the initial starvation is manifested by reduced pressure in hertz area and the lubricant film thickness decreases to zero level as the inlet meniscus approaches the hertz area. Chiu analysis showed that the degree of starvation depends mainly on the thickness of lubricant layer on the edge of the track.

29.Experimental analysis of hydrodynamic journal bearing under different biolubricants

Gy. Szota b. Kovács and f. J. Szabó have done their work on optimized gap shapes for sliding bearings on the basis of previously developed and published theoretical results of the authors an optimization process was elaborated for the calculation of the optimal lubricant film shape between lubricated sliding surface pairs. A theoretical and experimental study of thermal effects in a plain circular steadily loaded journal bearing was carried out by ma and taylor [2]. Dipl.-ing. Rolf lasaar, prof. Dr.-ing. Monika ivantysynova has done the work on gap geometry variations in displacement machines and their effect on the energy dissipation. The energy dissipation of displacement machines is mainly influenced by the design of individual lubricating gaps between parts, having relative motion to each other[3]. In comparing the global performance characteristics, theory and experiment exhibited an excellent agreement over a wide range of loads and rotational speeds. Gethin worked on the Thermal behavior of

various types of high speed journal bearings and found good agreement with theoretical and the experimental results. The effects of variable density and variable specific heat on maximum pressure, maximum temperature, bearing load, frictional loss and side leakage in high-speed journal bearing operation were examined.

30.Thermohydrodynamic lubrication analysis for a dynamically loaded journal bearing

Paranjpe and han proposed in 1994 a three-dimensional thd model that includes lubricant rupture and reformation phenomena by conserving the mass flowrate, conduction across solids, recirculation, and mixing of hot and cold fluids in the axial groove. They showed that only a little part of the heat is evacuated by conduction in the solids and 90 per cent of the heat is evacuated by the fluid film. In 1995 , they developed a transient thd model for journal bearings under sinusoidal loading, including the time scales for the film, bush, and journal. The finite-volume method is used in order to treat the energy equation of the film. By treating the bush and the lumped element journal in the quasi-steady state, the authors showed that the time scales for thermal transients in the oil film are of the same order as the period of the dynamic loading. Fillon et al. Observed an important breakdown temperature fall on the surface of a tilting pad bearing. This is near the side of most loaded pad because of the rupture film. When the bearing load or the preload conditions are high, the film is divergent near the exit edge of the bearing and the lubricant film is broken. In 1999, gomiciaga and keogh [4] developed a method for evaluating the asymmetric heat input in the synchronously vibrating journal of a hydrodynamic (hd) bearing. In 2000, piffeteau et al. Proposed a two-dimensional thermo Elasto hydrodynamic (tehd) model for a connecting-rod big end bearing under dynamic load. In this study, the elastic and thermal deformations of the solids are considered. In 2001, bonneau and hajjam elaborated an algorithm based on the jfo (jacobson, floberg, and olsson) model to

determine active and non-active zones in the films. In 2001, kim and kim analysed the tehd lubrication for connecting-rod bearings with a two-dimensional model. Fatu et al. [8] presented an original model to study the tehd lubrication in dynamically loaded journal bearings. Hoang et al. Compared the experimental results obtained for a connecting-rod bearing photoelastic device with the numerical results obtained by two-dimensional ehd model. Bukovnik et al. Proposed a comparison between different models, from simple to complex, for the simulation of non-stationary response of the journal bearings used in combustion engines.

31.Lubricant flow analysis for effective lubrication of tractor forward/reverse clutch

(lee and kim, 2011; kumar and manonmani, 2011). A test conducted by joo et al. (2011) revealed that the lubricant flow rate was the major determinant of the lifespan of the driving parts. In another study, the heat transfer was investigated with respect to the groove pattern of the clutch friction plate (bae et al., 2014). However, the findings of all these studies would only be fully relevant if the lubricant flow rate could be accurately calculated in advance. To accurately calculate the lubricant flow rate, a precise flow network analysis model is required. Various studies have been conducted to develop a precise flow network analysis model (lo, 1971; tran et al., 1987; mian, 1997; cho and yoon, 2005; cho, 2006; choi et al., 2012). This involved the use of elastohydrodynamic lubrication (ehl) analysis to obtain results that are more precise than those obtained by lubricant flow rate analysis using a one-dimensional (1d) analysis tool.

32.Lubricant for heavily loaded slow-speed journal bearing(Muzakkir) 2002,

Gears, and traction drives operate in a mixed lubrication regime. Under these conditions the surface material, lubricant viscosity, and additives play significant roles

in affecting the coefficient of friction and amount of wear. In journal bearings, wear of bearing surfaces increases the clearance, which affects the stability and durability of the system. Various researchers attempted to find the optimum set of tolerances on the journal and bearing bore that ensures correct operating performance of plain journal bearings. Optimization models were employed to determine the optimum values of the machine operating parameters, resulting in better out-of-roundness of finished parts .New methods of clamping the workpiece have been devised that reduce distortion of parts during machining The accuracy of centering during the measurement directly affects the measured value of out-of-roundness and methods have been proposed to compensate for these errors. It is also difficult to determine the correct value of the out-of-roundness because different measurement methods have different levels of accuracy

33.Lubrication oil condition monitoring and remaining useful life prediction with particle filtering

In comparison with vibration based machine health monitoring techniques, lubrication oil condition monitoring provides approximately 10 times earlier warnings for machine malfunction and failure (poley, 2012). The purpose of most research is, by means of monitoring the oil degradation process, to provide early warning of machine failure and most importantly extend the operational duration of lubrication oil in order to reduce the frequency of oil changes and therefore reduce maintenance costs. As stated by sharman and gandhi (2008), and many other researchers, the primary function of lubrication oil is to provide a continuous layer of film between surfaces in relative motion to reduce friction and prevent wear, and thereby, prevent seizure of the mating parts.

34.Study on lubrication performance of journal bearing with multiple texture distributions

Hamilton et al. Proposed a liquid lubrication theory applicable to parallel surfaces and explained the lubrication mechanism based on surface micro-irregularities. This theory is in reasonable qualitative agreement with experimental results. Etsion found that micro-dimples on the friction surface were able to significantly improve the load capacity, the wear resistance and the friction coefficient compared with the non-textured components. This beneficial effect was verified experimentally using the laser surface texturing (lst) technique. Cupillard studied the effect of the dimple distribution and position on the dynamic lubrication characteristics of bearings using the computational fluid dynamics (cfd) method. Brizmer studied the effect of distribution and size of spherical texture on the load capacity of a sliding bearing. The optimum parameters of the dimples and the favorable lst mode for achieving maximum load capacity were found. Tala-ighil et al. Studied the hydrodynamic effect of a few deterministic texture shapes of a journal bearing, and found that the parallelepiped texture shows advantages in enhancing the bearing performance compared with other geometries. Based on the reynolds equation, rahmani et al. Used the genetic algorithm to study the effect of the cross-section shape of concave/convex textures on the lubrication performance of an infinitely wide journal bearing. Brizmer et al. Presented a partial texture that was able to significantly increase the load capacity. Rao et al. Developed a theoretical model of a partially textured slip slider and coupled stress fluid lubricated journal bearing. In 2011, adatepe conducted numerous experiments to investigate the performances of plain and micro-grooved journal bearings, in which the micro-grooves were made by cutting micro-channels around and across the journal bearing surfaces. The results showed that the friction coefficients of the journal bearings with micro-grooved textures were higher than those without texture.

35.Comparative study –a mineral oil based lubricant and lubricant obtained from vegetable oil

It has been observed that approximately 50% of all lubricants which is sold worldwide gets released in to the environment through total loss applications such as volatility, spillage and accidents. Such environmental cause and loss enforce strictly the adoption and thereafter use of the following upgraded brands of lubricants in industry, forestry, agriculture, water treatment equipments such as moreover vegetable oils have excellent tribological properties such as high flash points, lower friction coefficient, lower evaporation and high viscosity index in comparison with mineral based lubricants (schneider 2006, erhan and asadankos 2000, miles 1996). Thermal oxidation is minor practical based problem for vegetable oils, though there were a few studies related to properties of oxidized vegetable oils, can overcome subsequently (fox & stachowiak 2003, castro et al 2006, mano et al 2009) however citation in this matter does not produce any anomaly taking tribology & environment both together. The use of denatured oil, impact on environment, poses serious threats afterwards, has not found anywhere (completely degradable).

36.Effect of methanol and ethanol on lubrication oil degradation of ci engine

Najafi et al. Used methanol diesel blend in 10:90 (m10), 20:80 (m20) and 30:70 (m30) in diesel engine. They observed 67%, 56% and 39% increment in bsfc for m10, m20 and m30 respectively at 2000 rpm . Qi et al. Tested the combine effect biodiesel (soybean diesel blend 50:50) with methanol 5% (bdm 5) and 10% (bdm 10). Bdm5 and bdm10 shows lower torque and power output compared to plain biodiesel blend, however there is dramatic reduction of smoke emission with bdm 5 and bdm 10. Prakash et al. Observed that castor oil is not suitable to directly use in ci engine, due to its high viscosity and water content. They prepared ternary fuel blend with castor oil, diesel and ethanol. Agarwal et al. Tested vegetable oil as a fuel for ci engine. Trans

esterification process is performed due to the high viscosity and low volatility of vegetable oil.. Better performance and 2.5% increase in thermal efficiency is observed in the case of 20% vegetable diesel blend compared to plain diesel fuel. Apra et al. Reported that the waste lubrication oil, tyres and plastics can be proceed to obtain diesel like fuel by removing impurities like dust, metal particles, carbon soot and others. Gopal and Raj studied the effect of lubrication oil under the use of biodiesel as fuel in ci engine. Biodiesel obtained from pongamia oil and methylester through transesterification process. Biodiesel is mixed with diesel fuel in 20:80 proportions. They observed large reduction in the kinematic viscosity and flashpoint of used lubricating oil in biodiesel fuelled engine. Agarwal tested the properties of used engine oil under the application of vegetable oil biodiesel. They observed lesser wear debris, metal particles and soot in biodiesel fuelled engine.

37.Surface quality improvement with solid lubrication in ceramic grinding

Research findings of shaji and radhakrisnan indicate that graphite is a successful solid lubricant for grinding hard and brittle materials. Venugopal and rao made preliminary investigation to study the effect of using a graphite lubricant on the performance of grinding silicon carbide material as compared to dry grinding. The most influential factors (all at three levels) affecting the grinding forces were studied by conducting the experiments with dry and graphite assisted grinding. The graphite powder was directed to the grinding zone by its own gravity and the rate of delivery of graphite was approx. Maintained constant.

38.Applications of industrial tribology

The interacting surfaces in relative motion are accompanied with friction. Friction between the surfaces leads to the dissipation of energy resulting in the loss of resources. The main cause of friction is the adhesion and deformation. Different types

of wear mechanisms include abrasive wear, adhesive wear, corrosive wear, erosive wear, fatigue wear. Corrosive wear occurs when there is a chemical reaction between a corroding medium and the material, which is a strong function of operating conditions. The friction and wear can be prevented by either providing a lubricating film between the surfaces or by developing new materials which are wear resistant. There are different types of lubrication which can be classified into a) hydrodynamic lubrication b) hydrostatic lubrication c) elasto-hydrodynamic lubrication (ehl) and d) boundary layer lubrication. The contact between such surfaces is called as counter formal contacts.

39.Re-refining of used lubricating oil.

Lubricant oils have been used primarily for reducing friction between moving parts of various machinery or equipment, minimize material wear, and improve the efficiency of equipment /machinery and for fuel and energy savings .access to lubricants is essential to any modern society and not only does lubrication reduce friction, but it also removes heat, keeps equipment clean, and prevents corrosion. one of its important applications includes gasoline and diesel engine oils .typically lubricants contain 90% base oil and less than 10% additives. a large number of additives are used to impart performance characteristics to the Lubricants. the main families of additives are: antioxidants • anti-wear Re-refining of waste lubricants could result in both environmental and economic benefits. re-refining of waste oil to manufacture base oil conserves more energy than reprocessing the waste oil for use as fuel. the energy required to manufacture refined oil from used oil is only one-third of the energy required to refine crude oil to produce virgin base oil. therefore, re-refining is considered by many as a preferred option in terms of conserving resource as well as minimizing waste and reducing damage to the environment.

40.Process for recycling of used lubricating oils.

The object of the invention is a process for recycling used lubricating oils. after drying and gasoline removal, the oil is distilled and then treated with sodium hydride in a finely-divided state at elevated temperature. this is followed by a further distillation. the oil fractions of low volatility being respectively distilled in a molecular distillation apparatus at below 2 mbar. according to the invention, a process is obtained in which used lubricating oil is reprocessed by treatment of the oil, pre-treated by drying and gasoline removal, with finely-divided sodium metal or sodium hydride at elevated temperature, and distillation under gentle conditions, such that the oil is distilled, before and after the treatment with sodium metal or sodium hydride, at a temperature of less than 300° c. this operating method makes it possible to reduce the amount of metallic sodium added to below 0.5 wt.%, based on the amount of distilled pre-treated oil. simultaneously, yields of over 90% of reusable re-refinates can be obtained.

41.Engineered synthetic engine oil and method of use.

A synthetic lubricant for gasoline and diesel engines having a viscosity ranging between about 14.5 and 16.5cs at 100c., the lubricant containing from about 55 to about 75 volume percent polyalphaolefin having a viscosity of about an ethylene-propylene copolymer diester, a packaged additive, a total base number as enhancer and a minor effective amount of an anti-foamant. references cited method of use is also provided whereby the subject lubricant is recirculated through an operating engine while periodically monitoring the total base number and adjusting total base number to a level of about 12.0 by the addition of a total base number enhancer such as calcium phenate. the lubricating oil disclosed herein is an engineered full pao synthetic oil specially tailored for use as a high performance lubricant in gasoline and diesel engines. engineered full synthetic oils are those made to the highest standards using the best pao base stock available and are the most expensive and highest performing of

the synthetic lubricating oils. these “full pao’ lubricants are designed rather than refined. as used herein, the term “full pao” base stock component, although viscosity improvers and minor amounts of other additives are used to further enhance the lubricant properties. such lubricants must have a product viscosity between about 12.9 and 16.7 centistokes (cs) over the requisite temperature range. the lubricants of the invention will desirably have a viscosity ranging between about 14.5 and 16.5 cs, preferably between about 15 and 16 cs, and most preferably, about 15.5 cs least about 25, and more preferably from about 27 to 29 or greater, to produce a viscosity ranging from about 14.5 to about 16.5, and more preferably from about 15 to about 16, in the resultant lubricant; from about 12 to about 15 volume percent of a commercially available lubricant additive package such as, for example, chevron oronite's oloa9061 to insure that the resultant lubricant meets all certification standards for an sae 5w40 motor oil; sufficient tbn enhancer to raise the tbn of the resultant lubricant to at least 10 and preferably to at least about 12, and, if needed, a minor effective amount of a compatible anti-foamant. furthermore, because the total volume of lubricant required is significantly lower than with mineral oil, the attendant expenses of transportation, storage and waste disposal are also reduced.

42.Industrial lubricant oils

Lubricants are products used mainly in engines to reduce friction among mechanical bodies. contrary to the majority of petroleum products which are identified through several parameters (the specs), lubricants are commonly identified only by their real performance, which can be tested only experimentally in specialized laboratories. the most important lubricants’ spec is the viscosity index (vi), a measure of viscosity variation at different temperatures. the oils mostly used are the mineral base oils quality strictly depends on the crude origin, also if it can be partially modified through refinery processes. paraffinic base oils arising from paraffinic crudes are the most widely used. naphthenic base oils are produced from a few crudes (typically from

venezuela) and are currently used in a few applications where low-temperature properties are required and the viscosity index is less important.

43.Lubrication

A lubricant consists of 70-100% base lubricants and up to 30% chemical compounds known as additives, which are fully mixed. the production of based oil consists of following steps distillation: the process of removing the components at a very low and very high boiling point, leaving the distillate in the boiling range of the lubricant. Removal of aromatics: it leaves the lubricating oil behind with a high proportion of saturated hydrocarbons, and improves its viscosity index and stability. de-waxing: it removes candles and controls the low temperature properties of the lubricant. Finishing: it removes the polar components, and improves the colour and stability of the base oil. the return on the base oil from the distillation column of the refinery depends on the proportion of desirable components in the range of the lubricant's boiling points. distillates of base oils derived from different oils have very different properties. the waste lubricant can be re-used and the process is called as re-refining. re-refining separates water, oil and asphalt products and through distillation it restores the lubricating oil (distillate) as in the refining process.

44.Study on lubrication performance of journal bearing.

The lubrication performance of journal bearings with different sorts of uniformly distributed micro-spherical textures are studied in this paper. geometries and dynamic models of journal bearings with pure concave/convex textures are developed. the validity of the proposed models is verified against the oil film pressure distribution from the literature. the effects of geometry parameters (the texture depth and the area density) on the load capacity and the friction coefficient of the bearing are analyzed and discussed. results indicate that: the bearing load capacity is reduced by the

concave spherical texture, but enhanced by the convex texture; both the concave and convex textures have a very slight influence on the friction coefficient. aiming to further improve the bearing lubrication performance, a novel concave-convex composite texture is proposed and modelled. Results show that the composite texture can significantly improve both the load capacity and the friction coefficient, because the concave spherical segments among the convex ones protect the oil film from rupture near the main load region. the oil film region is expanded by the composite texture as well. parameters on the bearing lubrication performance. a theoretical model of textured bearing is developed and validated. a novel convex-concave composite texture is proposed after discussing the effects of texture depth and area density on the load capacity and the friction coefficient of the bearing. the performance of the composite texture is then simulated and compared with the convex texture. the main conclusions can be drawn as follows:

- (1) the load capacity of the concave spherical textured bearing is lower than that of the plain bearing, and the friction coefficient is hardly affected by the concave texture. the convex spherical texture can increase the bearing load capacity and friction coefficient at the same time, but the reduction in friction coefficient is fairly slight.
- (2) the concave-convex composite spherical texture can improve the load capacity and the friction coefficient of the bearing. in contrast to the convex texture, the beneficial effect introduced by the composite texture on the friction coefficient is as significant as that on load capacity.
- (3) in addition to the load capacity and the friction coefficient, the oil film region is expanded in the composite texture case, as well. that is to say, the proposed convex-concave composite texture has significant potential for lubrication improvement of the bearing.

45.Self-lubricating journal bearing

A self-lubricating journal bearing includes an accurate bearing plate having a concave bearing surface. A plurality of lubricant plugs are positioned in apertures in the surface of said plate and are adapted to engage a rotating journal supported on said bearing surface for providing lubrication between said journal and said bearing plate. The lubricant plugs comprise an oil and polymer lubricating composition. The apertures and plugs are arranged to provide continuous lubrication over substantially the entire bearing surface as the journal rotates thereon, and coolant grooves cut into the bearing surface of said bearing plate in a generally shaped pattern for channeling liquid coolant between the journal and the bearing plate. In accordance with the foregoing objects and as shown in the drawings, the bearing plate is mounted in a pillow block. A similar cap structure can be provided. For providing lubrication between the bearing plate and a rotating journal supported thereon, a plurality of lubricant bearing plugs are inserted into apertures in the bearing plate and open into the bearing surface thereof for lubricating engagement with the journal. Coolant channels are cut into the bearing surface of the bearing plate for conducting coolant such as water when the bearing structure is to be utilized in high temperature applications such as supporting rolls in a rail mill. The lubricant bearing material may be a mixture of a hydrocarbon oil and polyethylene or other suitable oil and polymeric material lubricating.

46.Improved journal bearing

An improved journal bearing comprising in combination a non-rotatable cylindrical bearing member having a first bearing surface, a rotatable cylindrical bearing member having a confronting second bearing surface having a plurality of bearing elements, a source of lubricant adjacent said bearing elements for supplying lubricant thereto, each bearing element consisting of a pair of elongated relatively shallowly depressed surfaces lying in a cylindrical surface co-axial with the non-depressed surface and diverging from one another in the direction of rotation and obliquely arranged with respect to the axis of rotation of said rotatable member to cause a flow of lubricant

longitudinally along said depressed surfaces from their distal ends toward their proximal ends as said bearing members are rotated relative to one another, each depressed surface subtending a radial angle of less than 360.degree., and means for rotating said rotatable bearing member to cause the lubricant to flow across and along said depressed surfaces, the flow of lubricant being impeded by the non-depressed portions of said second bearing surface to cause an increase in the lubricant pressure.

47.Journal bearing design, lubrication and operation for enhanced performance - gregory f. Simmons (2013).

Simmons did study the basic of bearing and journal bearing test rig, and thus developing a small scale test arrangement. A journal bearing test machine was constructed to investigate a number of new synthetic lubricants and polymer bearing materials. These tests found that a significant reduction in power loss could be accomplished without significantly affecting the bearing's minimum film thickness by changing from a traditional mineral oil to a high viscosity index oil of much lower base viscosity grade. Results from the small scale studies then led back to the journal bearing test rig. His approaches thus lead to following conclusions:A] high viscosity index synthetic lubricants can greatly reduce power loss in journal bearings while maintaining safe machine operation. Select additives can both increase viscosity index to extremely high levels and reduce friction at start-up. B] start-up friction can further be reduced through the use of select polymer bearing materials. C] fabricating bearing tilting pads entirely of a polymer such as peek can lead to an increase in damping with a decrease in stiffness and an increase in oil temperature. D] full scale machine measurements demonstrate that the longer term transients in hydropower machines can significantly impact bearing performance.

48.Preliminary study of pressure profile in hydrodynamic lubrication journal bearing (2012)

Salmiah kasolang & mohamad ali ahmada did a study on pressure profile of journal bearing and stated that the pressure condition of the fluid is critical to ensure good performance of the lubricated machine elements such as journal bearings. They conducted an experiment to determine the pressure distribution around the circumference of a journal bearing and fluid frictional force of the bearing caused by shearing actions. They used a journal diameter of 100mm with a $\frac{1}{2}$ length-to-diameter ratio. The result 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. 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.

49.Recent advances in tribology: Argibay et al, 2017

Argibay et al. found that a self-lubricating DLC nano-composite film was in situ tri-biochemically formed from the ambient hydrocarbons of alcohols and alkanes on the nano-crystal line Pt–Au alloy surface. These films were extremely wear-resistant and underwent no obvious material removal even after 100,000 sliding cycles at a contact pressure of 1.1 GPa. Similarly, Wang et al. developed a multi-phase carbonaceous coating containing amorphous, fullerene-like, and Nano-crystalline carbons using magnetron sputtering method, which exhibited an ultralow friction coefficient of 0.05 and a low wear rate of about $10^{-8} \text{ mm}^3 \cdot \text{N}^{-1} \cdot \text{m}^{-1}$. Another research group synthesized graphite-like carbon (GLC) and fullerene-like carbon (FLC) films by different heating and cooling processes after plasma enhanced chemical vapor deposition (PECVD) . Both can bear quite high normal loads and lowering the friction to a super lubricity

state at higher contact pressure. Besides pure FLC, Wang et al. designed a fluorine-containing FLC (F-FLC) film, for which the bonding structure could be tailored from fullerene-like to amorphous. Another interesting lubrication system is the combination of nanostructured DLC with ionic liquid (IL) toward low friction and anti-wear interfacial behaviors for special applications. Shi et al. conducted comprehensive DFT calculations and demonstrated the effects of terminal states on friction behaviors of DLC in various gaseous environments. To monitor the tribo-reactions in real time, a MEMS holder developed by Sato et al. enabled in situ observation of the rolling and slipping events of DLC wear nanoparticles during lateral sliding. Recently, Kuwahara et al. using sliding experiments of ta-C/ta-C tri-bopairs showed that super lubricity with negligible wear can be achieved by lubrication with unsaturated fatty acids or glycerol. Zhang et al. using high-intensity pulsed ion beam irradiated WC-Ni surface against graphite under water lubrication also got significant friction reduction. A number of similarities either from material or methodologies point of view can be found and this will help further develop research in bio tribology.

3. HYDRODYNAMIC JOURNAL BEARING

Bearings are used to prevent friction between parts during relative movement. In machinery they fall into two primary categories: anti-friction or rolling element bearings and hydrodynamic journal bearings. The primary function of a bearing is to carry load between a rotor and the case with as little wear as possible. This bearing function exists in almost every occurrence of daily life from the watch on your wrist to the automobile you drive to the disk drive in your computer. In industry, the use of journal bearings is specialized for rotating machinery both low and high speed. This paper will present an introduction to journal bearings and lubrication. Lubrication technology goes hand-in-hand with understanding journal bearings and is integral to bearing design and application.

Hydrodynamic journal bearings are considered to be a vital component of all the rotating machinery. These are used to support radial loads under high speed operating conditions. In a hydrodynamic journal bearing, pressure of hydrodynamic lift generates thin film of lubricating oil which separates the shaft and bearing thus preventing metal-to-metal contact. Hydrodynamic type journal bearings are considered to be a vital component of all rotating machinery whose function is to support an applied load by reducing friction between the relatively moving surfaces. A journal bearing consists of a circular shaft, called the journal, is made to rotate in a fixed sleeve is called the bearing. The bearing and the journal operates with a small radial clearance of the order of $1/1000$ th of the journal radius. The clearance space between the journal and the bearing is assumed to be full of the lubricant. The radial load squeezes out the oil from the journal and bearing face and metal-to-metal contact is established. When the journal begins to rotate inside the bearing, it will climb the bearing surface and as journal speed is further increased; it will force the fluid into the wedge-shaped region. Since more and more fluid is forced into the wedge-shaped clearance space, which begins to exert pressure with increasing journal speed. At a particular speed, the

pressure becomes enough to support the load and the closest approach between journal and bearing where the oil film thickness is the minimum. A condition of perfect lubrication will exit when minimum oil film thickness is greater than the quantity dependent on the nature of the irregularities of the contacting surfaces. Journal bearing configurations are susceptible to large amplitude, lateral vibrations due to 'self-excited instability' known as oil whirl. Oil whirl is independent of shaft unbalance or misalignment. Forces generated in lubricating oil film due to hydrodynamic action cause a self-excited instability. During oil whirl shaft orbits in its bearing at a frequency approximately half the angular speed of shaft. If not controlled this non synchronous, self-excited orbiting motion will grow without bound and may lead to catastrophic failure. The stiffness of the shaft itself combined with the stiffness of bearing that support the journal determines several forms of natural frequencies of vibration called critical speed or threshold of whirl instability or stability of journal bearings. If the shaft is considered to be rigid mass in connections with the fluid film spring there will be natural frequency of vibration. There is also disturbing force coming from residual unbalance due to variation in load and thus speed in the system. Therefore the resonant vibration will be at shaft rotation speed called synchronous whirl and has been observed as a precession or orbiting of the center of shaft about the center of the bearing. Thus for rotor dynamic system synchronous oil whirl is a more serious issue than critical speed resonance. From different experimental work it has been observed that lubricant viscosity plays a major role in oil whirl instability. Under normal operating conditions the lubricant undergoes a significant change in viscosity and other bearing performance parameters such as minimum oil film thickness and load carrying capacity. Viscosity variations due to changes in temperature and oil film thickness variations will affect the stability of journal bearing system.

3.1 Advantages and Limitations

Some of the primary advantages of fluid film bearings are:

- Provide damping. Damping is required in order to pass through a critical speed. Damping is also required to suppress instabilities and sub-synchronous vibration.
- Able to withstand shock loads and other abuse.
- Reduce noise.
- Reduce transmitted vibration.
- Provide electrical isolation of rotor to ground.
- Very long life under normal load conditions.
- Wide variety of bearing types for specific applications
- The lubricant used provides these functions to all bearings:
- Remove heat generated in the bearing.
- Flush debris from load area.

3.2 Disadvantages of Fluid Film Bearings

- Higher friction (HP loss) than rolling element type.
- Susceptible to particulate contamination.
- Cannot run for any length of time if starved for lubricant such as a lube system failure.
- Radial positioning of rotor less precise.

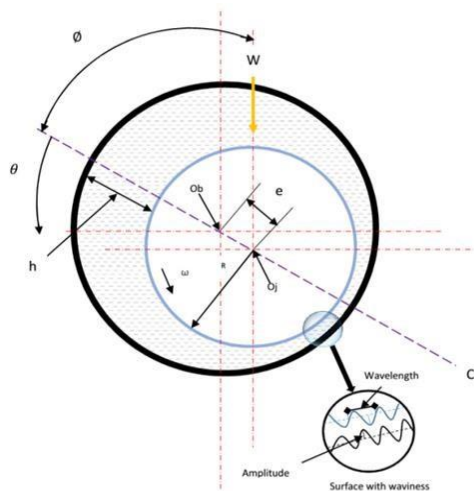


Fig 1.Schematic diagram of hydrodynamic journal bearing.

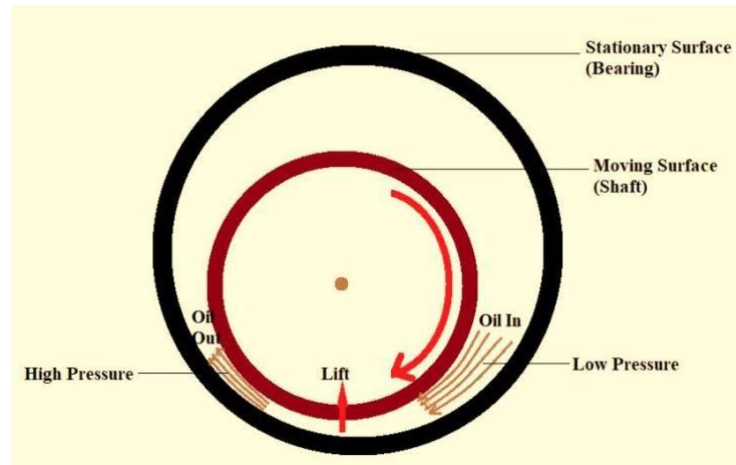


Fig 2. Basic Development of an Oil Wedge

3.3 Principle of Journal Bearing

When a journal bearing which has an adequate supply of lubricant is carrying a load it normally runs with the geometric centre's of the shaft and housing displaced so that a region of convergent flow is established. In this region large hydrodynamic pressures are set up within the oil film and these pressures when summated over the total bearing surface are found to completely support the load. If bearing conditions change; for instance the load may vary, the displacement or “attitude” of the centre's changes so that the new pressure distribution is sufficient to support the new load. Fig.3.3 illustrates the basic principles. Various non-dimensional load parameters are used to assess the performance of a bearing. The parameters are formed from terms such as the oil viscosity and the load and speed of the bearing. For a given bearing it is found that there is a critical value of load parameter at which the convergent film is unable to support the total bearing load and the surfaces touch. Under these conditions the friction torque suddenly starts to rise and boundary lubrication occurs in the region of minimum film thickness. Based on his theoretical investigation of cylindrical journal bearings, Professor Osborn Reynolds showed that oil, because of its adhesion to the journal and its resistance to flow (viscosity), is dragged by the rotation of the journal so as to form a wedgeshaped film between the journal and journal bearing .

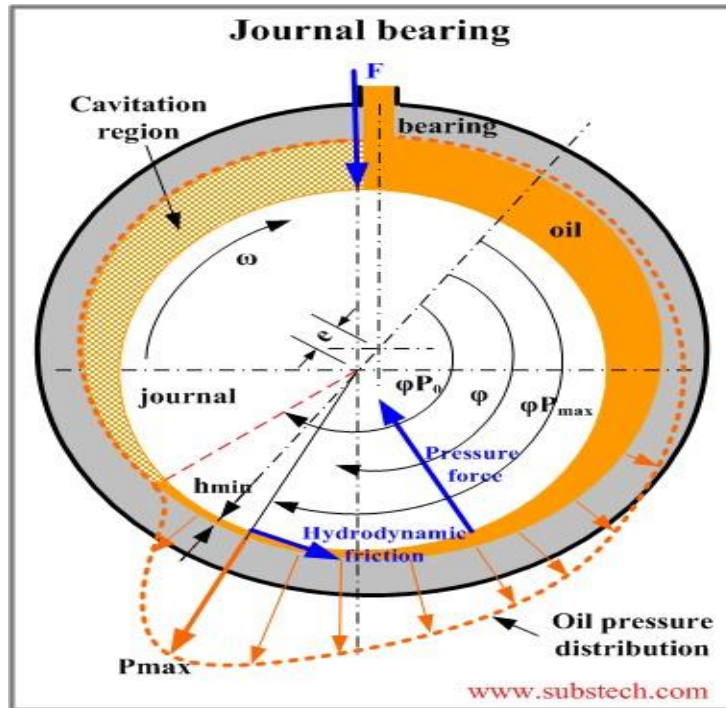


Fig 3. Principle of Hydrodynamic Journal Bearing

3.4 Materials

Hydrodynamic bearings utilize a protective layer to guard the shaft against damage at machine start up and shutdown. This protective layer has traditionally consisted of a thick coating of bronze or tin alloy known as Babbitt or white metal. Various forms of white metal have been widely used in sliding bearings since their invention in the 19th century. While this material has been tried and proven for over a century in sliding bearings, its use is not without drawbacks. Because white metal is especially susceptible to adhesion at break away, hydrostatic jacking systems are commonly used to transfer the load from the white metal layer's surface to lubricant under high pressure during machine start-up. These extra systems add complexity to the bearing system and start-up cycle which can reduce total machine reliability. Furthermore, because of white metal's relatively low melting point, white metal bearings are susceptible to softening at higher temperatures. This directly limits the maximum mean pressure during operation to between 2 and 3 MPa depending on the specific

geometry. Progression of material science has allowed for development of new materials and processing techniques not earlier imaginable. Polytetrafluoroethylene (PTFE), with its low friction characteristics, has been considered to be one of the better available materials for bearings. However, its wear rate is higher than desirable and PTFE can also have a tendency to creep under higher temperature loading and continuous use. Furthermore, when used in bearings, the insulating characteristics of most polymer materials can result in higher lubricant temperatures than for materials which more readily conduct heat away from the contact. This can result in differences in thermal expansion between the shaft and bearing which may lead to reduction of the bearing's clearance. Other polymer materials such as ultra high molecular weight polyethylene (UHMWPE) provide excellent friction and wearing characteristics, but are severely limited by their maximum operating temperatures. Additionally, most polymer materials for low friction applications are developed with the goal of self lubricated operation often with solid lubricant particles embedded into the material. These solid lubricant particles are not necessary in high speed machinery because liquid lubricant is always present in the contact for cooling and to allow for operation in the hydrodynamic lubrication regime where the lowest friction and highest surface speeds are possible.

Countless new materials as well as blends of PTFE have been developed which maintain the same low friction offered by PTFE while providing added strength and wear resistance. These materials are based on composites of PTFE or other polymers such as PEEK [18]. Applying these more advanced polymer materials to sliding bearing applications could potentially eliminate the issues faced by current materials while allowing for significant improvements in machine performance. Studies of a two-axial-groove journal bearing demonstrated that changes

in length and clearance in combination with changes in bearing material and lubricant could provide significant reductions in power loss in bearings while improving their dynamic performance.

3.5 State of the science

A tremendous amount of research effort has been given to the subject of hydrodynamic journal bearings over the course of the second half of the 20th century and the beginning of the 21st century.

Since Lund's ground breaking work in the understanding of journal bearing dynamics, understanding of the function of hydrodynamics and journal bearings has been steadily developed with at least a couple research groups around the world producing a steady flow of research results. The advent of improved computers has given much to simulations of journal bearings however with ever improving simulation software; the need for tangible experimental results becomes essential. Unfortunately, unlike much tribological testing, test equipment for journal bearings is neither standardized nor versatile. Further antagonizing this is the complexity of thermal and dynamic effects taking place in hydrodynamic journal bearings under operation, which extend over several length scales. This necessitates full scale or near full scale test equipment, which is in-turn large and expensive. The challenges associated with testing of journal bearings has led to the research being focused at several research centers around the world as opposed to broadly spread as is the case with some other fields of tribological research. As in most research fields, reviews of the research have been accomplished with the most recent being that of Swanson, and his following doctoral thesis, which well document experimental data for fixed geometry hydrodynamic journal bearings. The full range of bearings in terms of dynamics were covered more recently by Tiwari et al while methods and experimental data for extracting journal bearing dynamic coefficients were reviewed by Dimond et al including a detailed analysis of bearing test machines with dynamic testing capabilities.

4.JOURNAL BEARING DESIGN PARAMETERS AND NOMENCLATURE

The design of a journal bearing for a particular application involves many considerations.

4.1 Available space

Many machinery manufacturers purchase bearings from specific bearing companies. It is advantageous to use standard bearing form factors for cost reduction. A machine that was designed for thin shell bearing liners is not usually a good candidate for conversion to tilting pad bearings simply because there is insufficient radial room in the bearing housing. Sometimes new housings can be made, but this involves additional cost.

4.2 Friction and Heat Generation

The basic equation that is used to evaluate friction and heat generation is Petroff's equation which assumes a centred shaft in a plain bushing. Notice that journal load is not part of this calculation. Only the oil shear forces are used.

4.3 Specific Steady Load

A very important concept is the specific load, P , is defined as: $P = W/LD$ or simply the journal load divided by the active bearing length times the diameter. In English unit this is in PSI. The loading classification ranges for this variable are:

0 to 50 PSI – Very Light

50 to 100 PSI – Light

100 to 200 PSI – Moderate

200 to 300 PSI – Heavy

More than 300 PSI – Special Design

There are problems associated with bearings that are too lightly loaded, specifically oil whirl and instability. For heavily loaded bearings heat generation and Babbitt fatigue can become significant. One should always consider both the minimum film thickness and the peak hydrodynamic load when evaluating a design. Special evaluation is required if the peak pressure exceeds 1,000 PSI or the minimum film thickness is less than 1.0 mils.

4.4 Surface speed

Higher the surface speed the higher will be the journal diameter for a given RPM. To calculate feetper-minute (FPM) from

RPM the relationship is: (use inches for D) $FPM = \frac{RPM \times D}{12}$

($\pi D/12$) Some typical examples are:

600 RPM with 8 Inch Journal = 1,257 FPM

3,600 RPM with 4 Inch Journal = 3,770 FPM

12,000 RPM with 6 Inch Journal = 18,850 FPM

At some point, which depends mainly on the lubricant and clearance, the oil film becomes turbulent. Turbulence increases the friction, reduces oil flow and generates more heat. For typical turbo machinery, surface speeds above 15,000 FPM should receive special scrutiny.

4.5 Dynamic Load & start up load

Dynamic load is the result of shaft orbital motion in the oil film clearance space due to imbalance, misalignment, and other non-static forces. Thus, an alternating hydrodynamic pressure fluctuation is superimposed on the steady pressure being exerted on the Babbitt. It is this alternating force which fatigues the Babbitt. The easiest way to give guidelines for this is to limit maximum orbit vibration to less than 50% of the diametric clearance of the bearing. For example, if the bearing clearance is 6 mils, then the maximum acceptable displacement amplitude is 3 mils peak-to-peak

measured on the orbit from two orthogonal proximity probes. Once the orbit exceeds 75% of the clearance, short-term damage is almost always happening and bearing life is likely limited to a few hours if the orbit size exceeds 90% of the available clearance. The unit bearing pressure for starting conditions should not exceed 2N/mm^2 . The start-up load is the static load when shaft is stationary. It mainly consists of dead weight of shaft & its attachments.

4.6 L/D Ratio

The length to diameter ratio is one of the first things a bearing designer considers. Since the shaft diameter is often determined by other factors (torque and bending strength, for example) it is usually the active Babbitt axial length that is controlled. This factor is “tuned” to give sufficient steady and alternating load capacity. However, it also significantly affects the stiffness and damping characteristics of the bearing. Longer the bearing, lower the specific load and the lower the stiffness. A longer bearing also tends to have higher effective damping. Since damping is a frequency dependent stiffness, the speed of the machine must be considered as well. While, from a rotor dynamics standpoint, softer bearings are almost always a benefit; a too lightly loaded bearing may have stability problems. In the experience of this author, an L/D ratio less than 0.3 has poor damping and an L/D ratio greater than 0.75 usually shows little gain in effective damping. Heavily loaded bearings such as gears often exceed an L/D ratio of 1.0 and may need special care in aligning the bores to the shafting.

4.7 Clearance

The basic guideline universally used for diametric journal bearing clearance is 1.5 mils-per-inch of journal diameter. That is, a 4 inch diameter shaft would need about 6 mils of diametric clearance. Always check if the specifications you see are for diametric clearance. Some manufacturers specify *radial clearance* – which is really difficult to measure. If the application has atypical loads and/or speeds, then this

clearance rule may need to be adjusted. The bearing should be held in the housing with a zero to 1 mil interference fit. A loose bearing will have vibration problems and too much housing interference can reduce the clearance in the bearing. The issue of tolerances should also be considered. Clearance values are usually given as a range. The limits suggested here for minimum allowable clearance is 1.0 mil-per inch shaft diameter plus one mil. The maximum allowable is 2.0 mile-per-inch. Remember tolerance costs money.

4.8 Base Material

Normally steel is used as the backing material for Babbitt bearings and is preferred for strength. Cuprous alloys are about half the strength of steel and have virtually no endurance limit. Bronze or even copper may be used where extra heat conduction is needed for a successful design. Caution should be used with new machinery. API 612 (Special Purpose

Turbines) and API 617 (Centrifugal Compressors) both specifically disallow anything other than steel for a backing material. The reason for this is that it allows a relatively inexpensive and convenient path to fix a hot bearing or upgrade a design. If the original design includes copper pads, you are limited on the redesign.

4.9 Grooving

On some bearings grooves are cut into the surface of the Babbitt for a variety of reasons. The most often cited reasons for grooving are to direct lubricant to the loaded areas and for cooling. A welldesigned bearing with steady loads (not reciprocating) does not need any grooves and, in fact, the sole effect is to increase stiffness, reduce load capacity, and damping. In some cases a circumferential groove is used (called a *load* groove) to increase the specific loading and increase stability.

4.10 Minimum and maximum oil film temperature

The surface finish of a journal bearing is governed by the value of minimum oil film thickness selected by the designer and vice versa. There is a lower limit for the minimum oil film thickness, below which metal to metal contact occurs and hydrodynamic film breaks. This lower limit is given by, $h_{\min} = (0.0002) \cdot R$.

4.11. Nomenclature

c = Radical Clearance (mm^2)

c = Combined coefficient of radiation and convection ($\text{W/m}^2 \text{ K}$)

CFV = Coefficient of Friction value

d = Diameter of journal (mm)

D = Diameter of bearing (mm)

d_s = Shaft diameter (mm)

e = Eccentricity

ε = Eccentricity ratio

E_o = Young's Modulus

f = Coefficient of Friction

F_u = Friction force (KN)

F_u' = Friction force per unit area of bearing (MPa)

FV = Flow variable

h = Oil film thickness (mm)

h_{\min} = Minimum oil film thickness (mm)

h_{\max} = Maximum oil film thickness (mm)

H_d = Heat dissipating capacity

H_g = Heat generated in bearing (kJ/sec)

K = Leakage factor

l = Length of bearing (mm)

n = Journal speed (rps)

N = Speed in (rpm)

Analysis of Hydrodynamic Journal Bearing

P = Power (KW) or (HP)

p = Load per unit of the projected area

P_o = Atmospheric pressure (N/mm²)

P_1 = Intensity of Pressure (N/mm²)

P_i = Inlet pressure (N/mm²)

Q = Flow of Lubricant (mm³/sec)

r = Radius of journal (mm)

S = Sommerfeld number

Δt = Temperature rise (°C)

v = Surface speed of the journal (m/min)

Z' = Absolute viscosity (N-sec/mm²)

f = Coefficient of friction

θ = Circumferential coordinate

Φ = Angle of eccentricity / Altitude angle (degree)

Ψ = Angle between the load direction and centerline

5.MATHEMATICAL MODELING

Fluid mechanics is a physical science concerned with the behavior of fluid at rest and in motion. It combines the two separate approaches- the empirical hydraulics and the classical hydrodynamics. On the other hand, hydrodynamics is essentially a mathematical science dealing with flow analysis based on the concept of an ideal fluid in which both fluid viscosity and fluid compressibility are assumed absent. Although, the art of lubrication has been practiced from time immemorial, the analytical understanding of the lubrication phenomenon, which is basically, a part of an overall phenomenon of fluid mechanics, is a comparatively recent development dating back to the last quarter of nineteenth century. This is hardly surprising because the mathematical modeling of the bearing system is closely linked to the research developments in the field of fluid dynamic of real fluids which started in nineteenth century.

Mathematical modeling of a bearing system consists of various conservation laws of fluid dynamics such as conservation of mass, momentum, energy and equation describing various aspects characterizing the bearing problem such as constitutive equation of lubricant, viscosity dependence on pressure–temperature, equation of state, elastic deformations, surface roughness etc

5.1.Reynolds Equation

The differential equation which is developed by making use of the assumptions of hydrodynamic lubrication in equations of motion and continuity equation and combining them into a single equation governing lubricant pressure is called Reynolds Equation. The Reynolds equation when derived for more general situations like porous bearings or hydrodynamic bearings or bearings working with non-Newtonian or magnetic lubricant etc. is called generalized Reynolds type equation or modified

Reynolds equation. This equation is the basic governing differential equation for the problems of hydrodynamic lubrication.

Assumption

- The lubricant is assumed to be Newtonian; stress is proportional to the rate of the shear.
- The flow is considered to be laminar. A moderate velocity combined with a high kinetic viscosity gives rise to a low Reynolds number, at which flow essentially remains laminar.
- Lubricant film is assumed to be iso-viscous.
- Temperature changes of the lubricant are neglected.
- The bearing surfaces are assumed to be perfectly rigid so that elastic deformations of the bearing surfaces may be disregarded.
- When bearings work under the influence of electromagnetic field, it is assumed that the forces due to induction are small enough to be neglected.

Reynolds equation is given by

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(h^3 \frac{\partial p}{\partial z} \right) = 6\eta U \frac{\partial h}{\partial x}$$

Where,

h – local oil film thickness

η – dynamic viscosity of oil

p – local oil film pressure

U – linear velocity of journal

x - circumferential direction

z - longitudinal direction

5.2.Sommerfeld Equation

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

$$S = \left(\frac{r}{c}\right)^2 \frac{Z'N}{P}$$

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

- The surface is smooth,
- The fluid is Newtonian and the flow is laminar, and
- Inertia force resulting from acceleration of the fluid and body forces are small compared with the surface forced,
- It is interesting to study the pressure distribution in the hydrodynamic region of a fluid film bearing

5.3.Fluid Friction in Journal Bearing

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

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

where c is radial clearance, r is journal radius and ϵ is eccentric ratio. In this present study, the torque has been measured and later converted to frictional force and friction coefficient. For this preliminary work, the experimental frictional coefficient values were compared to the predicted values from Raimondi and Boyd charts for validation purposes. Comparison of experimental results with predicted values using equation to be addressed in future work.

5.4. Pressure distribution Equation

When so much heat is generated by hydrodynamic action that the normal lubricant flow is insufficient to carry it away, an additional supply of lubricant must be furnished under pressure. To force a maximum flow through the bearing and thus obtain the greatest cooling effect, a common practice is to use a circumferential groove at the center of the bearing, with an oil supply hole located opposite the load-bearing zone. The effect of the groove is to create two half bearings, each having a smaller l/d ratio than the original. The groove divides the pressure distribution curve into two lobes and reduced the minimum film thickness, but it has wide acceptance among lubrication engineers and carries load without overheating. To set up a method of solution for oil flow, we shall assume a groove ample enough so that the pressure drop in the groove itself is small. Initially we will neglect eccentricity and then apply a correction factor for this condition. The oil flow, then, is the amount which flows out of the two halves of the bearing in the direction of the concentration shaft. The pressure distribution of the hydrodynamic bearing can be calculated by the following equation

$$P = \left[\frac{20Z'Vde}{c^2} \right] \left\{ \frac{(2+e \cos \theta) \sin \theta}{(2+e^2)(1+e \cos \theta)^2 \theta} \right\}$$

Where,

Z' = absolute viscosity N-s/mm²

Analysis of Hydrodynamic Journal Bearing

V = surface speed of journal m/min

d = journal diameter in mm

e = eccentricity in mm

θ = circumferential co-ordinates in degree

5.5 Petroff's Equation

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

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

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

A vertical shaft rotating in the bearing is shown in fig.(a)

The following notations are used:

r = radius of the journal (mm)

l = length of the bearing (mm)

c = radial clearance (mm)

n_s = journal speed (rev/sec)

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

$$U = (2\pi r)n_s \dots\dots\dots (1)$$

Newton's law of viscosity and using Equation

$$P = Z'A\left(\frac{U}{h}\right) \dots\dots\dots (2)$$

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

P = tangential frictional force

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

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

Analysis of Hydrodynamic Journal Bearing

h = distance between journal and bearing surfaces = c

Substituting above values in Equation (2)

$$P = \mu(2\pi rl)(2\pi n_s)\left(\frac{1}{c}\right) = \frac{4\pi^2 r^2 l Z n_s}{c}$$

The frictional torque is given by,

$$(M_t)_f = P_r = \frac{4\pi^2 r^3 l Z n_s}{c}$$

Let us consider a radial force (W)' acting on the bearing as shown in fig.(b)

The unit bearing pressure (P) acting on the bearing is given by,

$$P = \frac{W}{\text{projected area of bearing}} = \frac{W}{2rl}$$

Or

$$W = 2pr$$

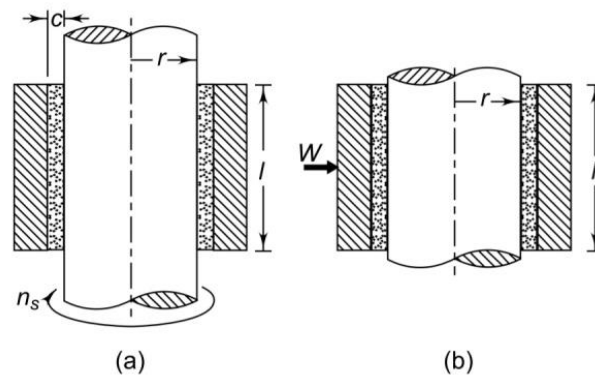


Fig no. 4-Vertical Shaft Rotating in the bearing

The frictional force will be ($f W$) and frictional torque will be ($f W r$).

Therefore,

$$(M_t)_f = f W r = f(2prl)r = f(2pr^2l)$$

Where f is frictional coefficient of friction.

From above

$$\frac{4\pi^2 r^3 l Z n_s}{c} = f(2pr^2l)$$

$$f = 2\pi^2 \left(\frac{r}{c}\right) \left(\frac{Z'n_s}{p}\right) \dots \dots \dots \textbf{Petroff's Equation}$$

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

6.PARTS OF HYDRODYNAMIC JOURNAL BEARING USING NX SOFTWARE

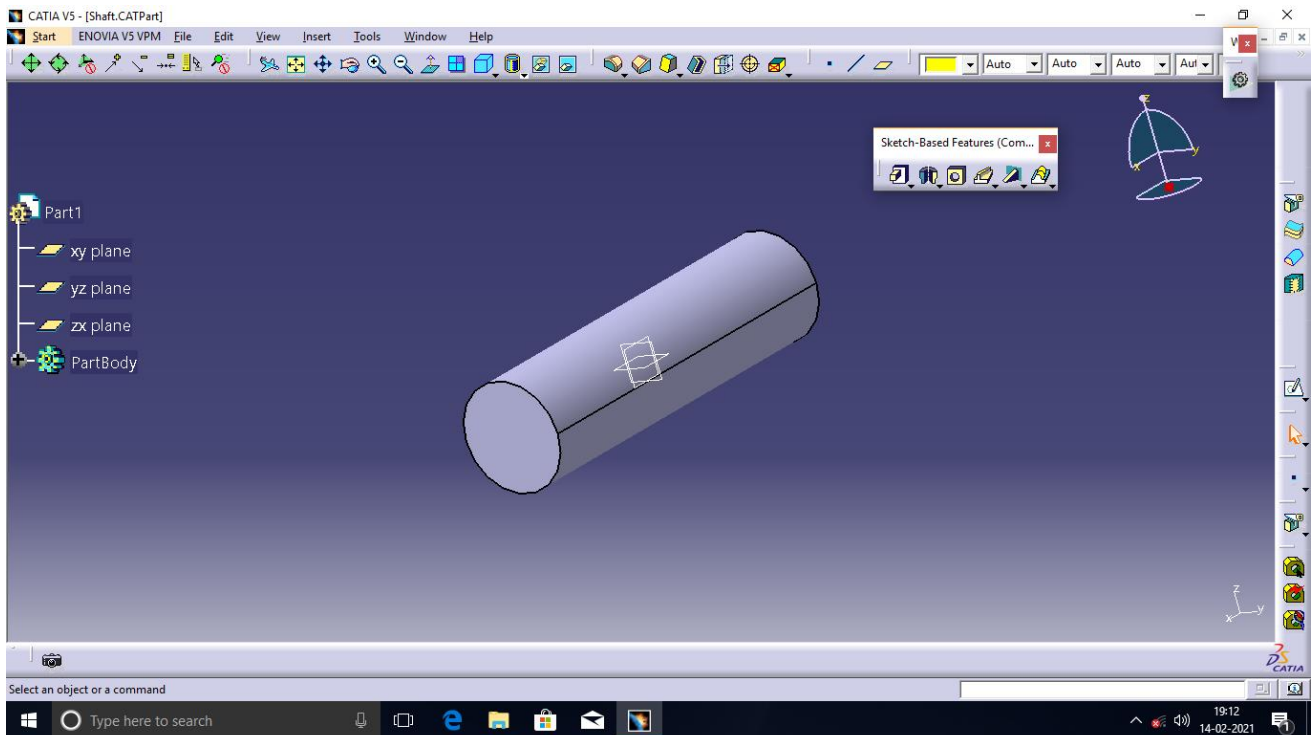


Fig No.5- Shaft

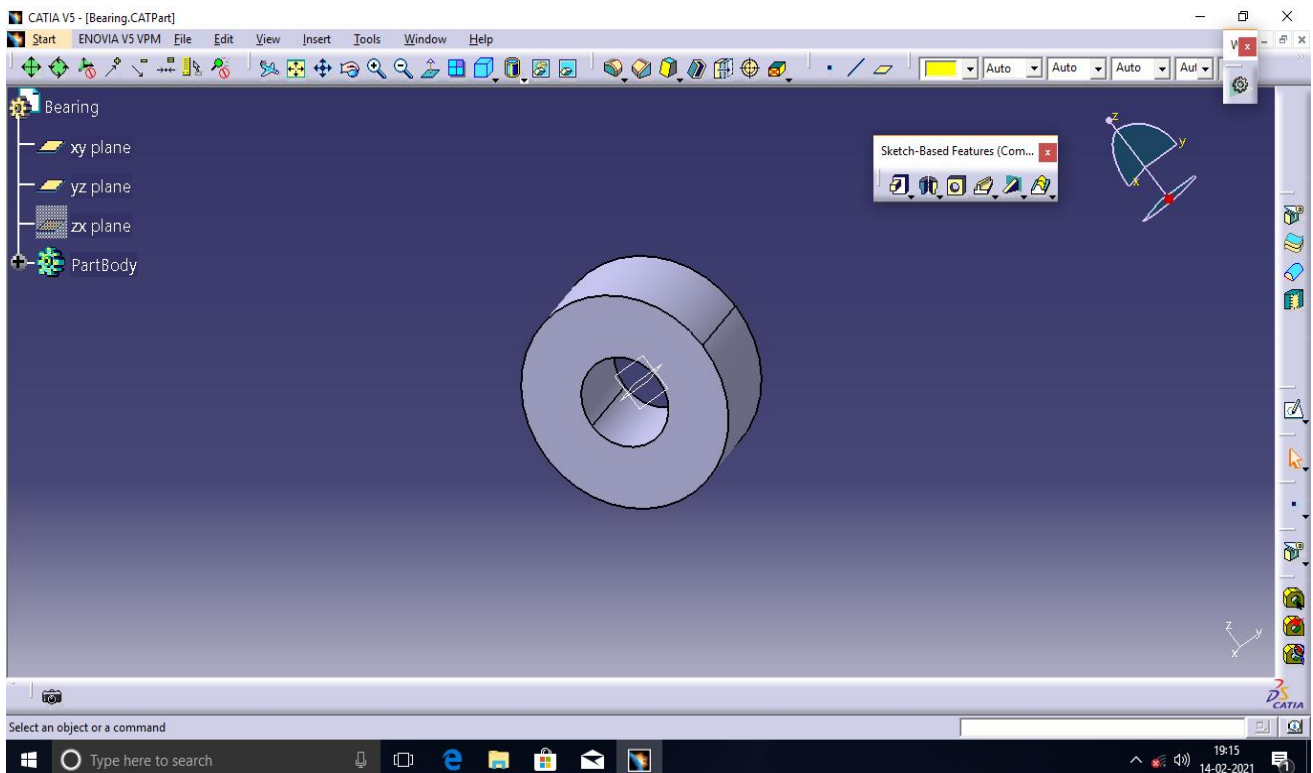


Fig No.6-Bearing

Analysis of Hydrodynamic Journal Bearing

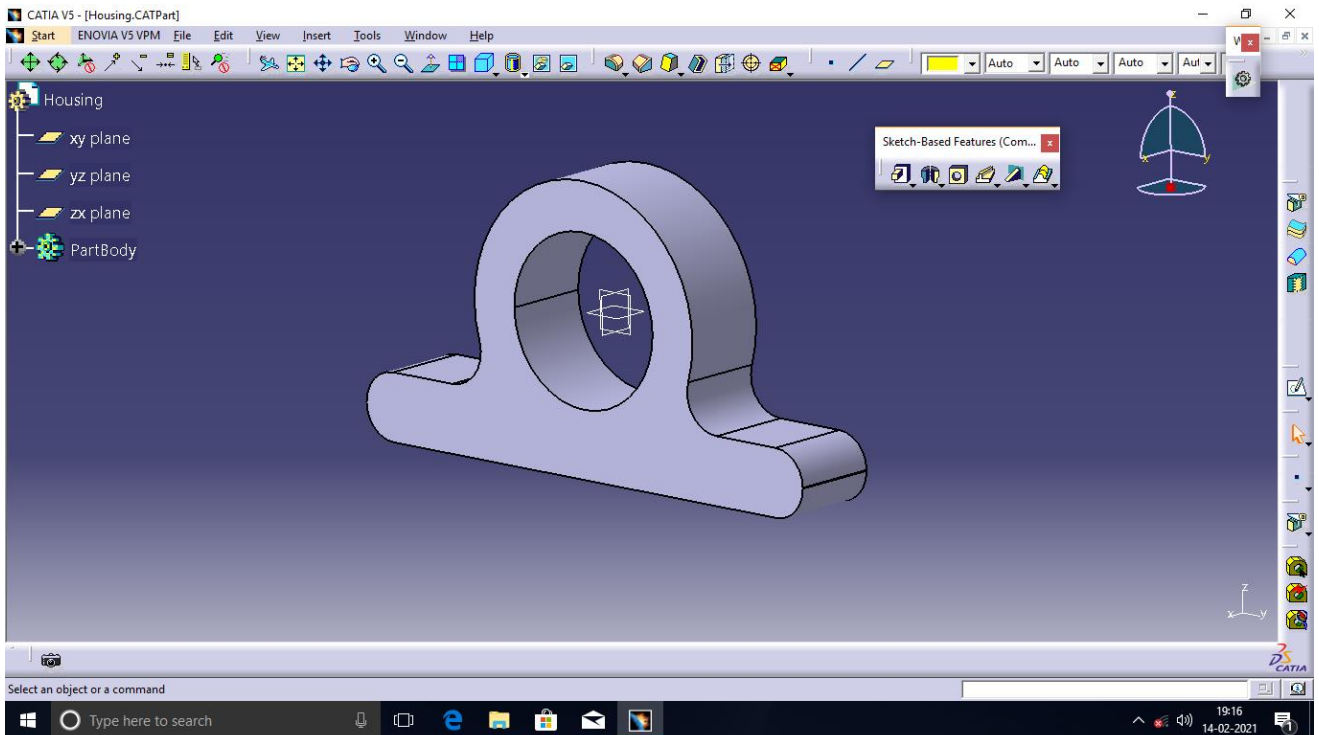


Fig No.8- Housing

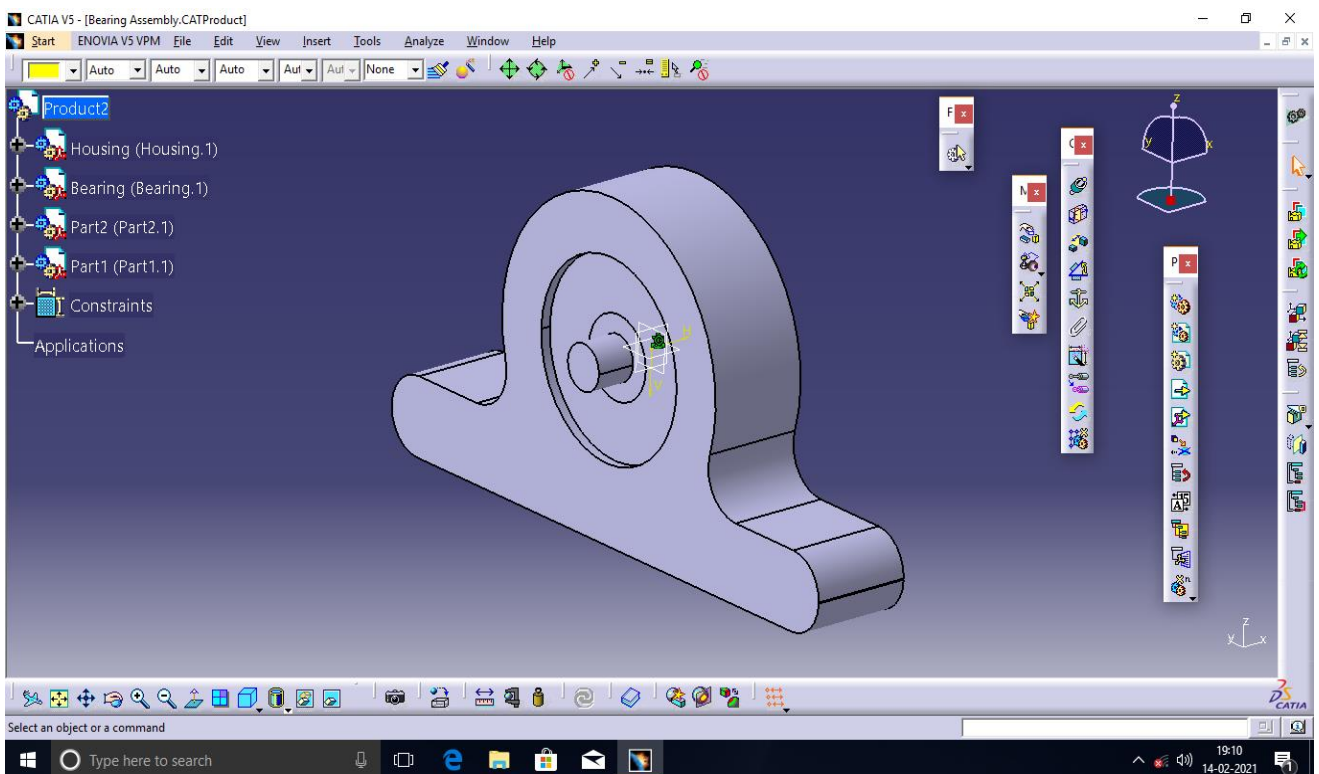


Fig No.9-Assembly

7.ANALYSIS :MATLAB PROGRAMS

7.1. PETROFF'S EQUATION

```
clc;
clear all;
%INPUT PHASE
r = input('Enter the value of shaft radius: ');
p = input('Enter the value of pressure: ');
u = input('Enter the value of absolute viscosity of oil: ');
c = input('Enter the value of radial clearance: ');
N = input('Enter the value of speed of shaft: ');
%Condition Check
[m,n] = size(N);
[s,t] = size(p);
S = zeros(1,n);
for i=1:n
    for j=1:t
        S(j,i) = 2*(3.142^2)*u*r*N(1,i)/(c*p(j)*60);
        while (u*N(1,i)/p(1,j))<= 1.7*10^-6
            fprintf('N = %f is not preferred for p = %f \n', N(1,i),p(1,j))
            break;
        endwhile
    endfor
endfor
disp(N)
disp(S)
plot(N,S(1,:), 'b--o')
title('Plot of Friction Coefficient against Shaft speed')
```

Analysis of Hydrodynamic Journal Bearing

```
xlabel('Shaft Speed(rpm)')
ylabel('Friction Coefficient')
hold on
plot(N,S(2,:), '--')
hold on
plot(N,S(3,:))
legend({'p=2.066 N/mm^2','p=4.132 N/mm^2','p=6.198
N/mm^2'}, 'Location','northwest')
```

OUTPUT

Enter the value of shaft radius: 110

Enter the value of pressure: [2.066 4.132 6.198]

Enter the value of absolute viscosity of oil: 8×10^{-8}

Enter the value of radial clearance: 0.275

Enter the value of speed of shaft: [40:400:8000]

N = 40.000000 is not preferred for p = 2.066000

N = 40.000000 is not preferred for p = 4.132000

N = 40.000000 is not preferred for p = 6.198000

Columns 1 through 15:

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

Columns 16 through 20:

6040 6440 6840 7240 7640

Columns 1 through 7:

0.000203878 0.002242660 0.004281442 0.006320224 0.008359006
0.010397787 0.012436569
0.000101939 0.001121330 0.002140721 0.003160112 0.004179503
0.005198894 0.006218285

Analysis of Hydrodynamic Journal Bearing

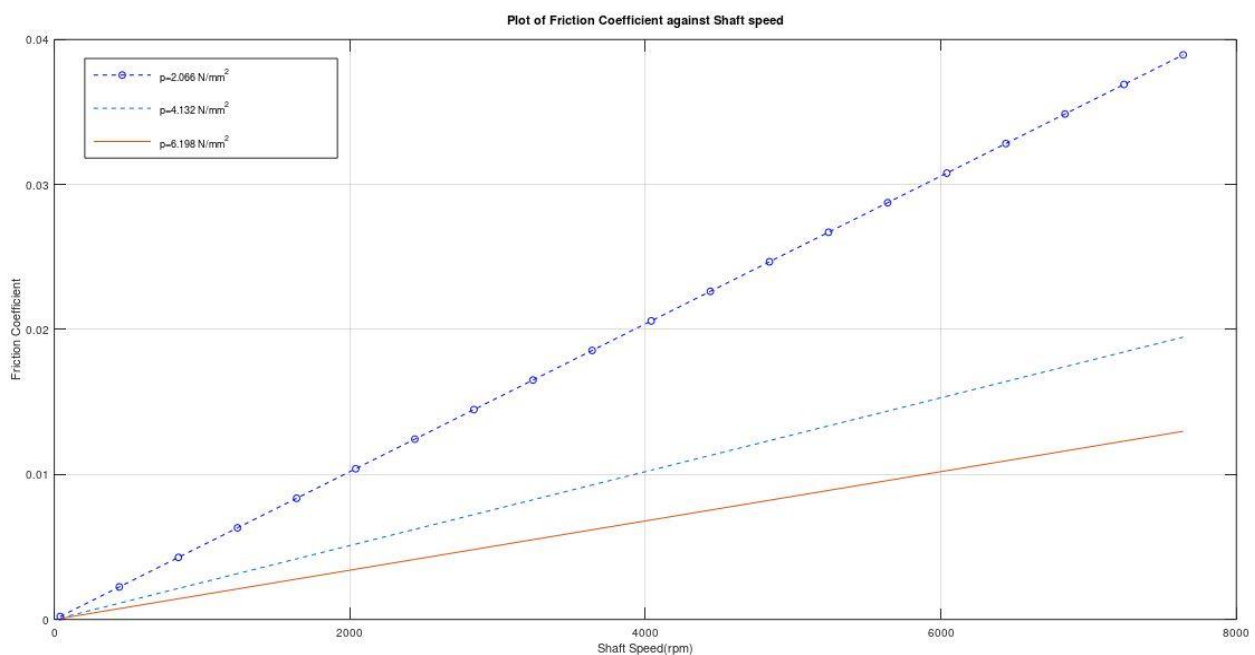
0.000067959 0.000747553 0.001427147 0.002106741 0.002786335
0.003465929 0.004145523

Columns 8 through 14:

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

Columns 15 through 20:

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



Graph No. 1 -Plot of Friction Coefficient Against Shaft Speed using MATLAB Software

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

1. As the shaft speed is increasing, the coefficient of friction also increases.
2. As we can see the graph is plotted for 3 different pressures which shows that for higher pressure the value of coefficient of friction for particular speed is also high.
3. The friction coefficient increases with the degree of misalignment at lower values of the eccentricity ratio.

7.2. SOMMERFELD EQUATION

```
clc;
```

```
clear all;
```

```
close all;
```

```
% Sommerfeld Equation Program
```

```
% BE Project
```

```
fprintf("\n \n \t SOMMERFELD EQUATION \n\n\n")
```

```
fprintf("\t S = (r/c)^2*u*N/p \n\n\n")
```

```
r = input("\t Enter the value of shaft radius: ");
```

```
p = input("\t Enter the value of pressure: ");
```

```
u = input("\t Enter the value of absolute viscosity of oil: ");
```

```
c = input("\t Enter the value of radial clearance: ");
```

```
N = input("\t Enter the value of speed of shaft: ");
```

```
fprintf("\n \n")
```

```
[m,n] = size(N);
```

```
[s,t] = size(p);
```

```
S = zeros(1,n);
```

```
for i=1:n
```

```
S(i) = (r/c)^2*u*N(1,i)/p;
```

```
if S < 0
```

```
print("Given value of N= %f is not feasible",N(1,i))
```

Analysis of Hydrodynamic Journal Bearing

$S = NaN$

endif

fprintf("\t For %d rps, Sommerfeld Number is %d \n\n",N(1,i),S(1,i))

endfor

plot(N,S(1,:), 'b--o')

title('Plot of Sommerfeld Number against Shaft speed')

xlabel('Shaft Speed(rps)')

ylabel('Sommerfeld Number')

Enter function: $S = (r/c)^2 * u * N / p$

OUTPUT

Enter the value of shaft radius: 110

Enter the value of pressure: 0.6

Enter the value of absolute viscosity of oil: $8 * 10^{-8}$

Enter the value of radial clearance: 0.275

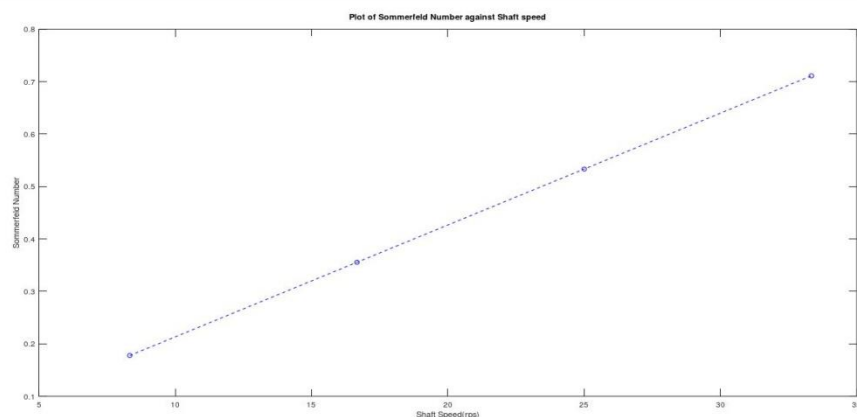
Enter the value of speed of shaft: [8.334 16.667 25 33.334]

For 8.334 rps, Sommerfeld Number is 0.177792

For 16.667 rps, Sommerfeld Number is 0.355563

For 25 rps, Sommerfeld Number is 0.533333

For 33.334 rps, Sommerfeld Number is 0.711125



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

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

1. As the shaft speed is increasing the Sommerfeld number is also increasing.
2. We can also see that for higher shaft speed the value of Sommerfeld number is also high.
3. Sommerfeld number is a dimensional quantity used extensively in hydrodynamic lubrication. It is very important as it contains all the variables normally specified by the designer.

7.3. PRESSURE DISTRIBUTION EQUATION

```
clc;
clear all;
close all;
% S = (20*z*v*d*e/c^2) * ((2+e*cosx*sinx)/(2+e))
f = inline(input('\n Enter function: ','s'), 'z', 'v', 'd', 'e','c', 'x');
d = input('\n enter the value of journal diameter: ');
e = input('\n enter the value of eccentricity: ');
z = input('\n enter the value of absolute viscosity of oil: ');
c = input('\n enter the value of diametrical clearance: ');
v = input('\n enter the value of surface speed of journal: ');
x = input('\n enter the value of circumferential co-ordinates: ');
S = f(z, v, d, e, c, x)
while S < 0
x = input('\n enter the initial guess again: ');
S = f(z, v, d, e, c, x)
end
fprintf('\n Value of function: S = %s', char(f));
```


OUTPUT

Enter function:

$$(20 * z * v * d * e / c ^ 2) (2 + e * \cos x * \sin x) / (2 + e)$$

Enter the value of journal diameter: 220

Enter the value of eccentricity: 0.023

Enter the value of absolute viscosity of oil: $0.08 * 10^{-6}$

Enter the value of diametrical clearance: 0.55

Enter the value of surface speed of journal: $1151.91 * 10^3$

Enter the value of circumferential co-ordinates:

[0 60 120 180 240 300 360]

For circumferential coordinate 0, Pressure = 0

For circumferential coordinate 60, Pressure = 0.0192

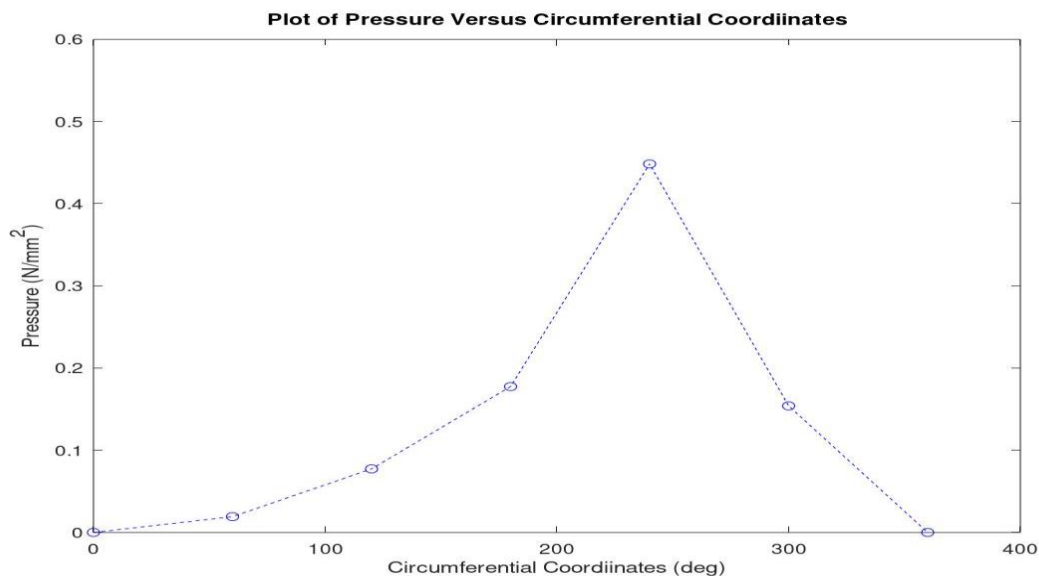
For circumferential coordinate 120, Pressure = 0.0773

For circumferential coordinate 180, Pressure = 0.1775

For circumferential coordinate 240, Pressure = 0.4484

For circumferential coordinate 300, Pressure = 0.1540

For circumferential coordinate 360, Pressure = 0



Graph No. 3- Plot of Pressure versus Circumferential Coordinate using

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

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

8.CALCULATION OF DESIGN OF HYDRODYNAMIC JOURNAL BEARING FOR CENTRIFUGAL PUMP :

Operating Temperature = 150°C

Load on bearing = 12 kN

Speed of journal = 1440 rpm

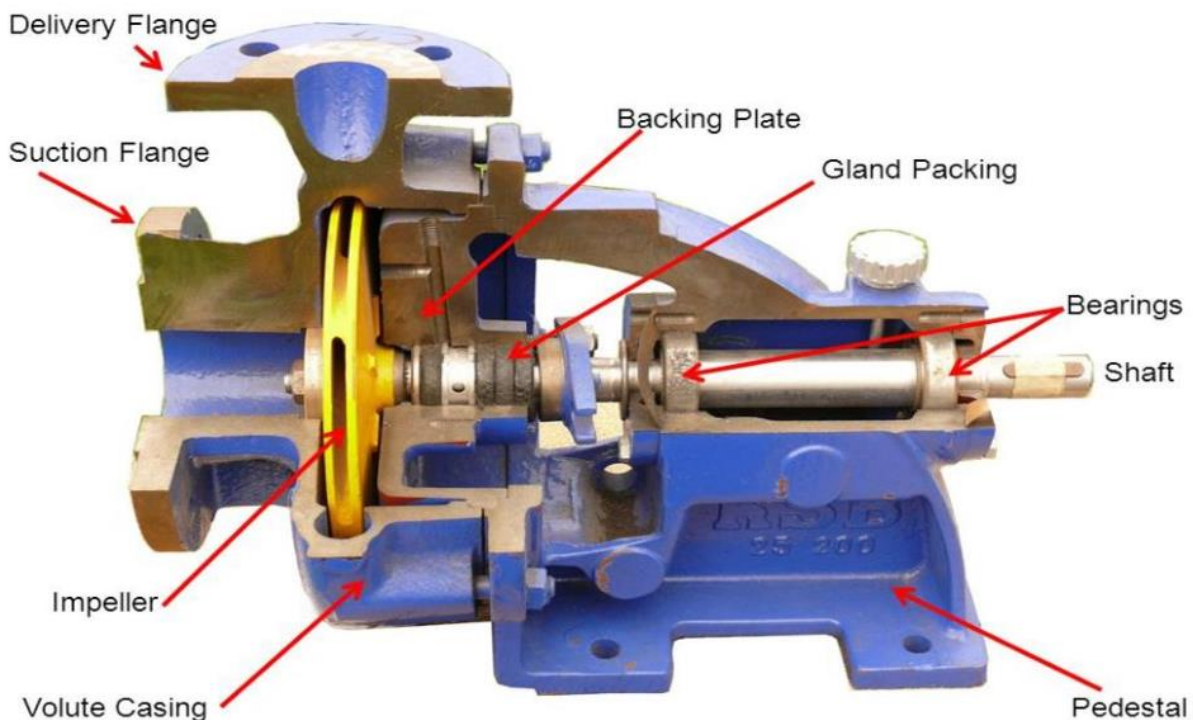


Fig No. 10- Journal Bearing mounted on Centrifugal Pump

8.1. Calculation of Diameter of journal and Length :

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

From Design Data Book Pg. 23.15, Table No. 23.2

1. The value for viscosity is 25 cP.
2. The value of $L/D = 1.0$ to 2.0

Analysis of Hydrodynamic Journal Bearing

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

$d = 94 \text{ mm}$

Diameter of journal = 94 mm

Radius of journal = 47 mm

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

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

Table No. 1 – Journal Bearing Design Practices

8.2. Radial Clearance :

Standard value of Radial clearance is given by ,

$$c = (0.0013)r = 0.12 \text{ mm}$$

8.3. Radius of Bearing :

$$R = c + r$$

$$= 0.12 + 47 = 47.12 \text{ mm.}$$

8.4. Minimum oil film thickness :

Standard value of Minimum oil film thickness is given by,

$$\frac{ho}{c} = 0.002 \times r = 0.094$$

8.5. Viscosity of Lubricant :

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

$$\frac{l}{d} = 1 \text{ and } \frac{ho}{c} = 0.8$$

For the above mentioned values,

$$S = 0.631$$

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

$$S = \left(\frac{r}{c}\right)^2 \frac{Z'Ns}{P}$$

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

$$Z' = 2.3 \times 10^{-8} \text{ N-sec/mm}^2_{(a)}$$

From Design Data Book Pg. 23.15, Table No. 23.2 the value for viscosity is 25 cP i.e

$$2.5 \times 10^{-8}$$

Z' calculated < Z' given

Hence the value selected is correct.

Dimensionless performance parameters for full journal bearings with side flow

Values of δ									
L/d ratio	0.25	0.5	1.0	∞					
For maximum load	0.27	0.43	0.53	0.66					
For minimum friction	0.03	0.12	0.3	0.6					
$\frac{L}{d}$	ε	δ	S	ϕ	$\frac{\mu}{\psi}$	$\frac{4Q}{\psi d^2 n' L}$	$\frac{Q_s}{Q}$	$\frac{\gamma c_{sp} T_0}{P}$	$\frac{P}{P_{max}}$
0.25	0	1.0	∞	(89.5)	∞	π	0	∞	—
	0.1	0.9	16.2	82.31	322.0	3.45	0.180	1287.0	0.515
	0.2	0.8	7.57	75.18	153.0	3.76	0.330	611.0	0.489
	0.4	0.6	2.83	60.86	61.1	4.37	0.567	245.0	0.415
	0.6	0.4	1.07	46.72	26.7	4.99	0.746	107.6	0.334
	0.8	0.2	0.261	31.04	8.80	5.60	0.884	35.4	0.240
	0.9	0.1	0.0736	21.85	3.50	5.91	0.945	14.1	0.180
	0.97	0.03	0.0101	12.22	0.922	6.12	0.984	3.73	0.108
0.5	1.0	0	0	0	0	—	1.0	0	0
	0	1.0	∞	(88.5)	∞	π	0	∞	—
	0.1	0.9	4.31	81.62	85.6	3.43	0.173	343.0	0.523
	0.2	0.8	2.03	74.94	40.9	3.72	0.318	164.0	0.506
	0.4	0.6	0.779	61.45	17.0	4.29	0.552	68.6	0.441
	0.6	0.4	0.319	48.14	8.10	4.85	0.730	33.0	0.365
	0.8	0.2	0.0923	33.31	3.26	5.41	0.874	13.4	0.267
	0.9	0.1	0.0313	23.66	1.60	5.69	0.939	6.66	0.206
1	0.97	0.03	0.00609	13.75	0.610	5.88	0.980	2.56	0.126
	1.0	0	0	0	0	—	1.0	0	0
	0	1.0	∞	(85)	∞	π	0	∞	—
	0.1	0.9	1.33	79.5	26.4	3.37	0.150	106	0.540
	0.2	0.8	0.631	74.02	12.8	3.59	0.280	52.1	0.529
	0.4	0.6	0.264	63.10	5.79	3.99	0.497	24.3	0.484
	0.6	0.4	0.121	50.58	3.22	4.33	0.680	14.2	0.415
	0.8	0.2	0.0446	36.24	1.70	4.62	0.842	8.0	0.313
∞	0.9	0.1	0.0188	26.45	1.05	4.74	0.919	5.16	0.247
	0.97	0.03	0.00474	15.47	0.514	4.82	0.973	2.61	0.152
	1.0	0	0	0	0	—	1.0	0	0
	0	1.0	∞	(70.92)	∞	π	0	∞	—
	0.1	0.9	0.240	69.10	4.80	3.03	0	19.9	0.826
	0.2	0.8	0.123	67.26	2.57	2.83	0	11.4	0.814
	0.4	0.6	0.0626	61.94	1.52	2.26	0	8.47	0.764
	0.6	0.4	0.0389	54.31	1.20	1.56	0	9.73	0.667
∞	0.8	0.2	0.021	42.22	0.961	0.760	0	15.9	0.495
	0.9	0.1	0.0115	31.62	0.756	0.411	0	23.1	0.358
	0.97	0.03	—	—	—	—	0	—	—
	1.0	0	0	0	0	0	0	∞	0

Key: Q_s = flow of lubricant with side flow, cm^3/s ; γ = weight per unit volume of lubricant whose specific gravity is 0.90 = 8.83 kN/m^3 (0.0325 lbf/in^3); c_{sp} = specific heat of the lubricant, kJ/NK ($\text{Btu/lbf}^\circ\text{F}$) = 0.19 kJ/NK (0.42 $\text{Btu/lbf}^\circ\text{F}$); T_0 = difference in temperature, $^\circ\text{C}$.

Source: A. A. Raimondi and J. Boyd, "A Solution for the Finite Journal Bearings and Its Applications to Analysis and Design" ASME, J. Lubrication Technol., Vol. 104, pp. 135–148, April 1982.

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

8.6. Coefficient of friction :

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

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

$$f = 0.00379$$

8.7. Temperature Rise:

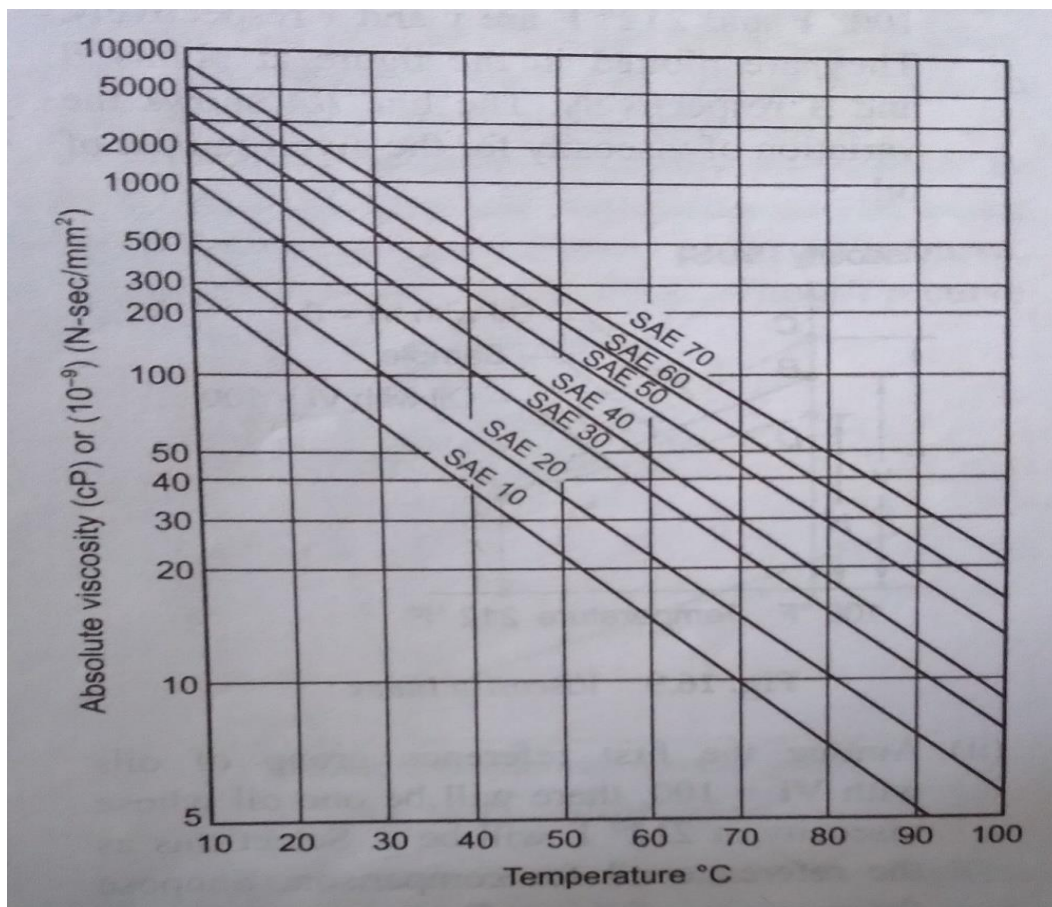
Referring to table 16.1 from Design of Machine elements (V B Bhandari) the coefficient of friction variable (CFV) is 12.8 and flow variable (FV) is 3.59

$$\Delta t = \frac{8.3p(CFV)}{FV} = \frac{8.3 \times 1.37 \times 12.8}{3.59}$$

$$\Delta t = 40.54^{\circ}\text{C}$$

$$T_{\text{avg}} = T + \frac{\Delta t}{2} = 85^{\circ}\text{C}$$

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



Graph No. 4- Viscosity-Temperature Relationship

8.8. Pressure Distribution :

$$P = \left[\frac{20Z'Vde}{c^2} \right] \left\{ \frac{(2+e \cos\theta)\sin\theta}{(2+e^2)(1+e \cos\theta)^2\theta} \right\}$$

For 0, Pressure = 0

for 60, Pressure = 0.01652

For 120, Pressure = 0.0773

For 180, Pressure = 0.1875

For 240, Pressure = 0.4584

For 300, Pressure = 0.1570

For 360, Pressure = 0

9. 3D MODELLING OF BEARING USING NX SOFTWARE

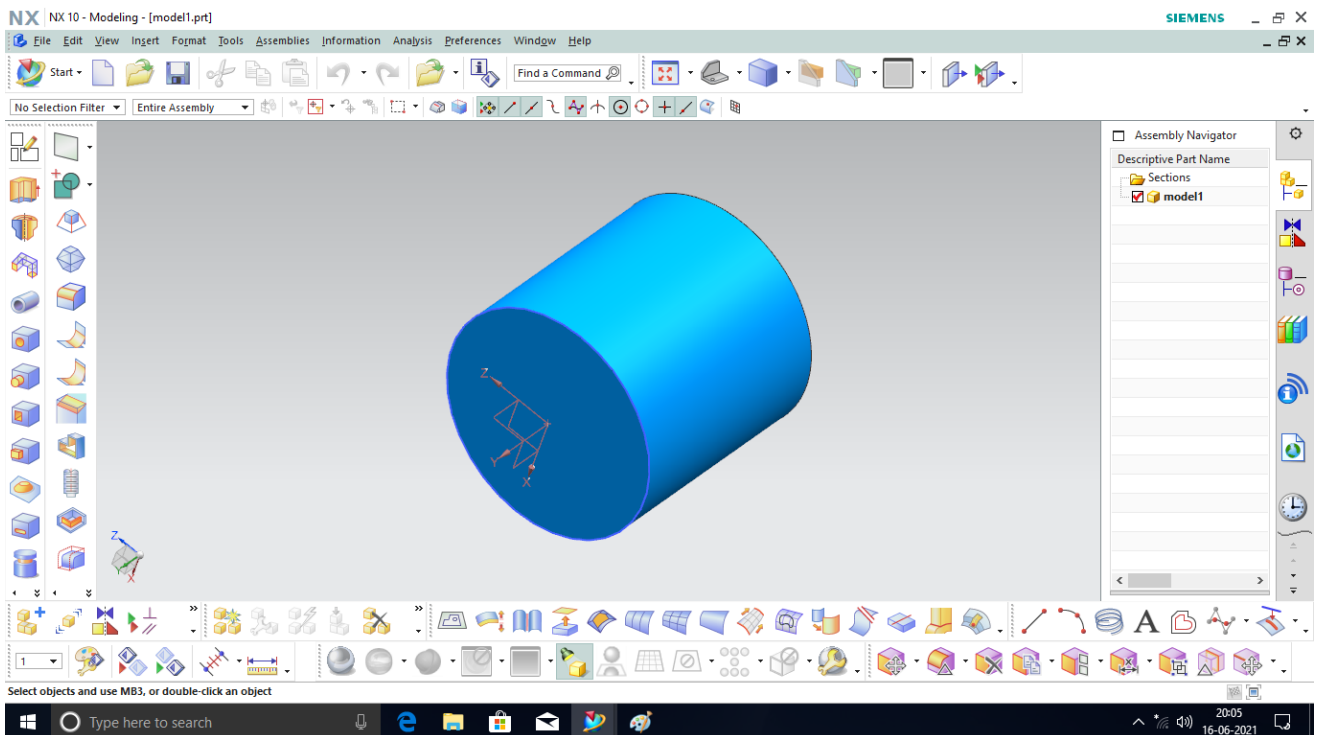


Fig No. 11-Shaft

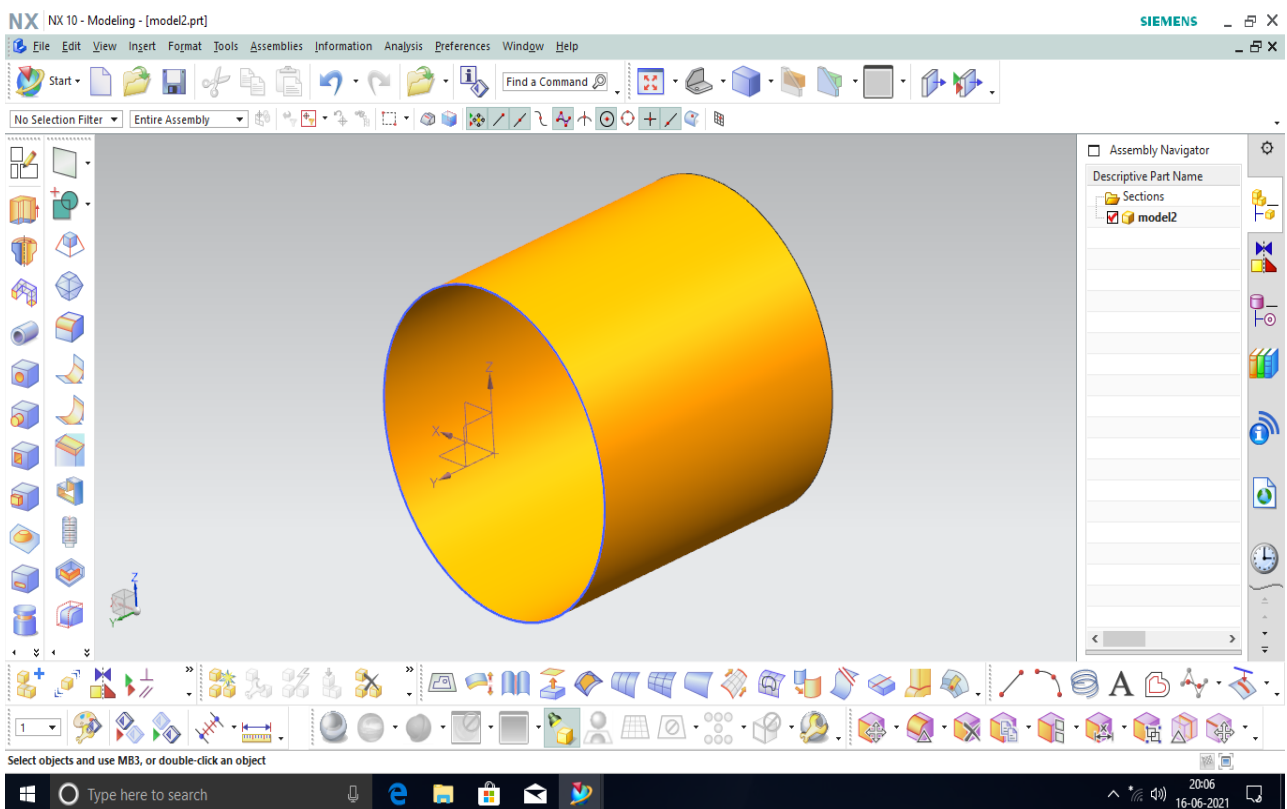


Fig No. 12-Lubricating Film

Analysis of Hydrodynamic Journal Bearing

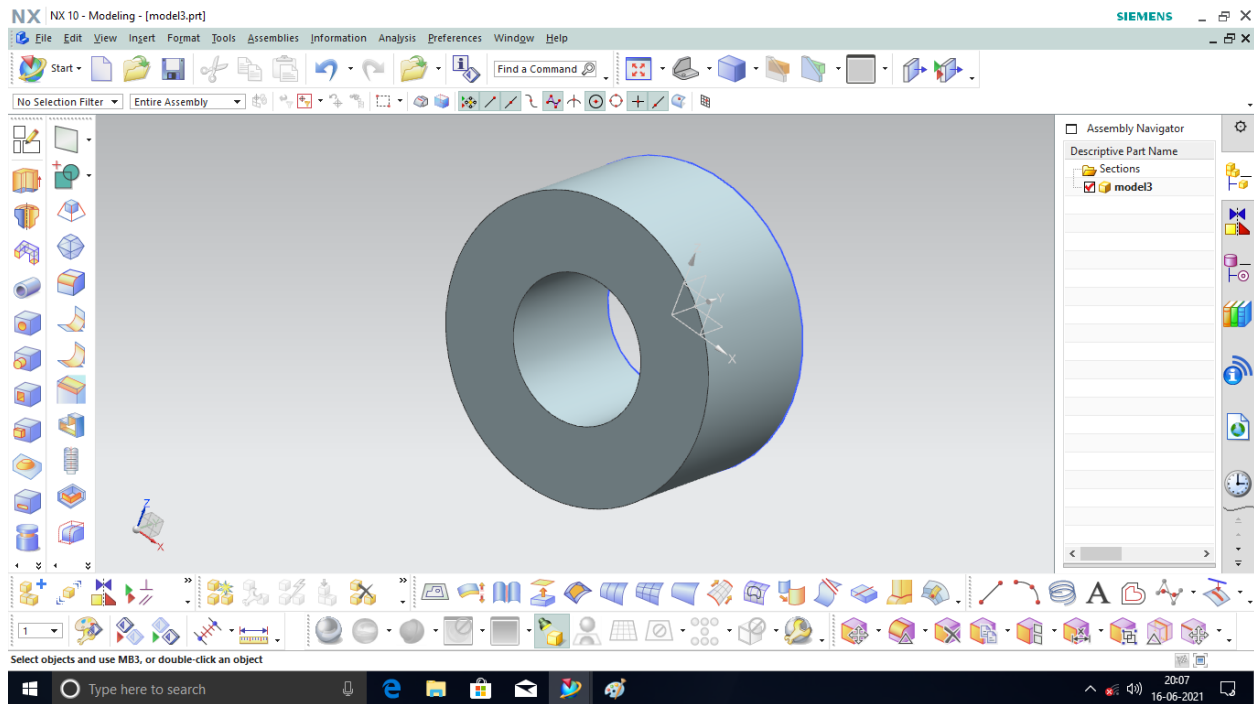


Fig No. 13-Bearing

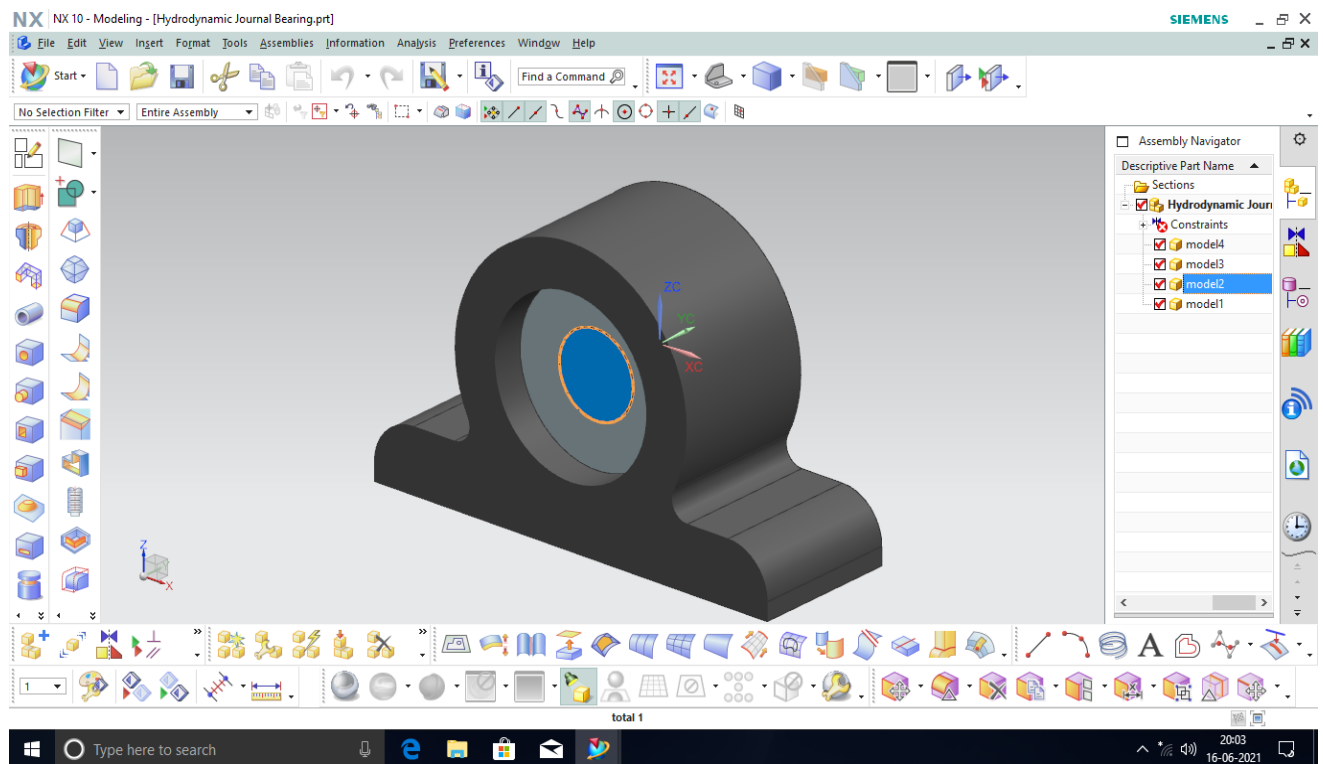


Fig No. 14-Assembly of Hydrodynamic Journal Bearing

9.1 Analysis on Ansys Software

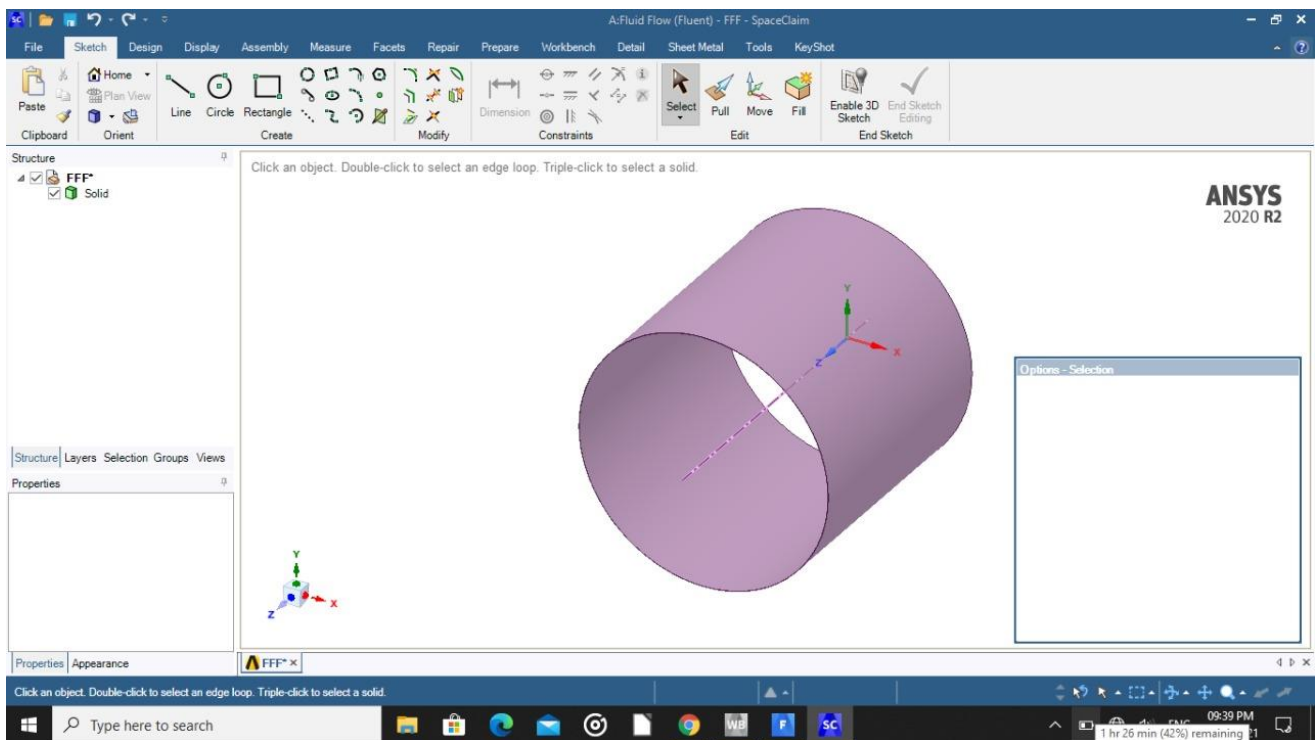


Fig No. 15

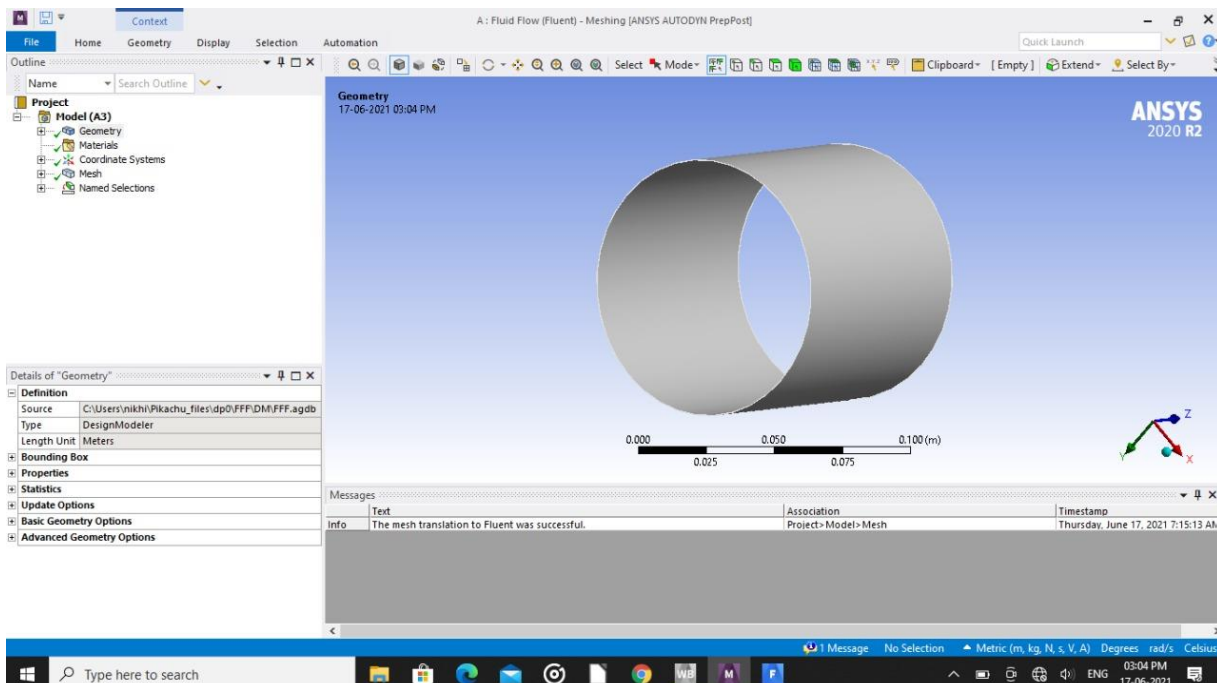


Fig No. 16

Analysis of Hydrodynamic Journal Bearing

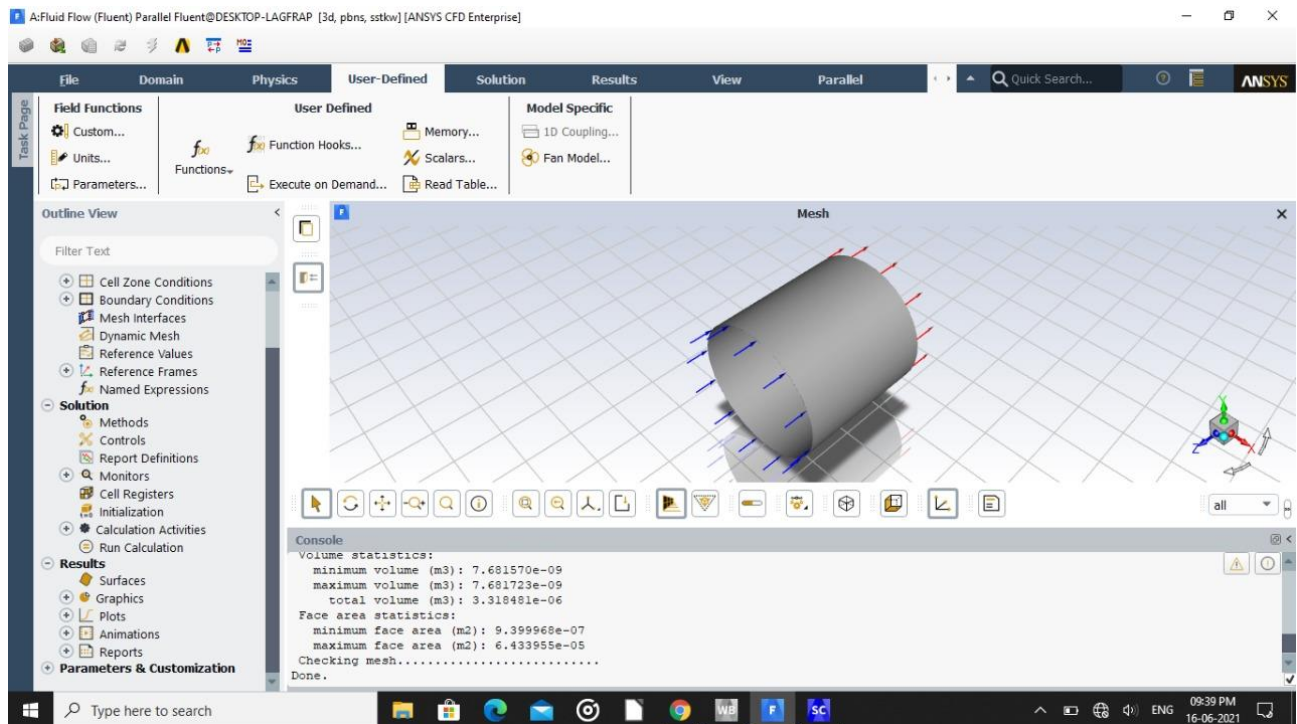


Fig No. 17

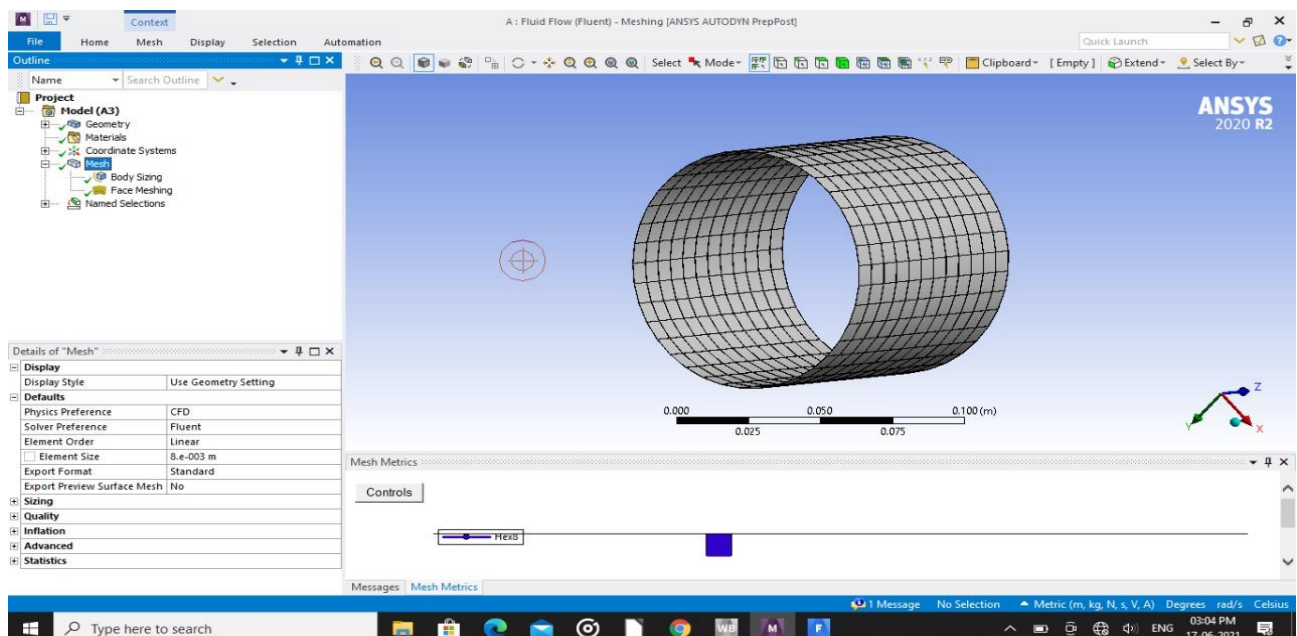


Fig No. 18

Analysis of Hydrodynamic Journal Bearing

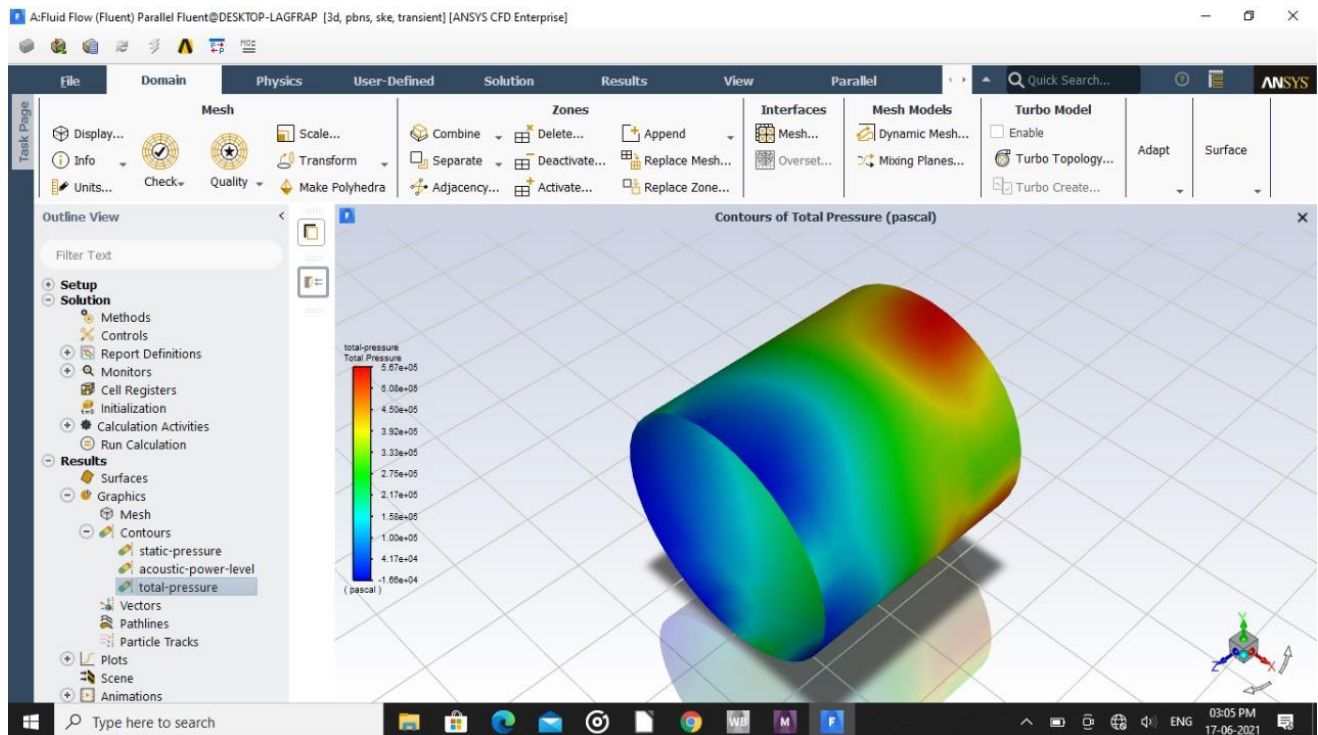


Fig No. 19

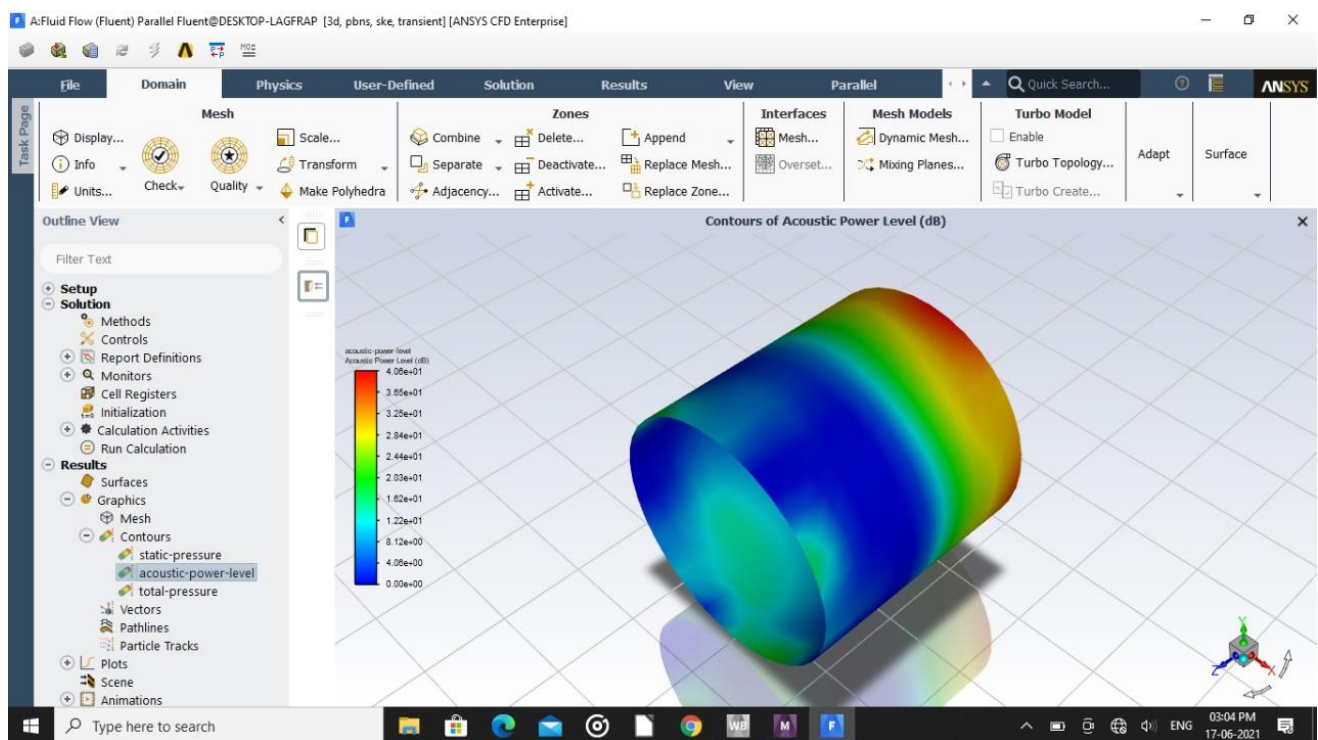


Fig No. 20

Analysis of Hydrodynamic Journal Bearing

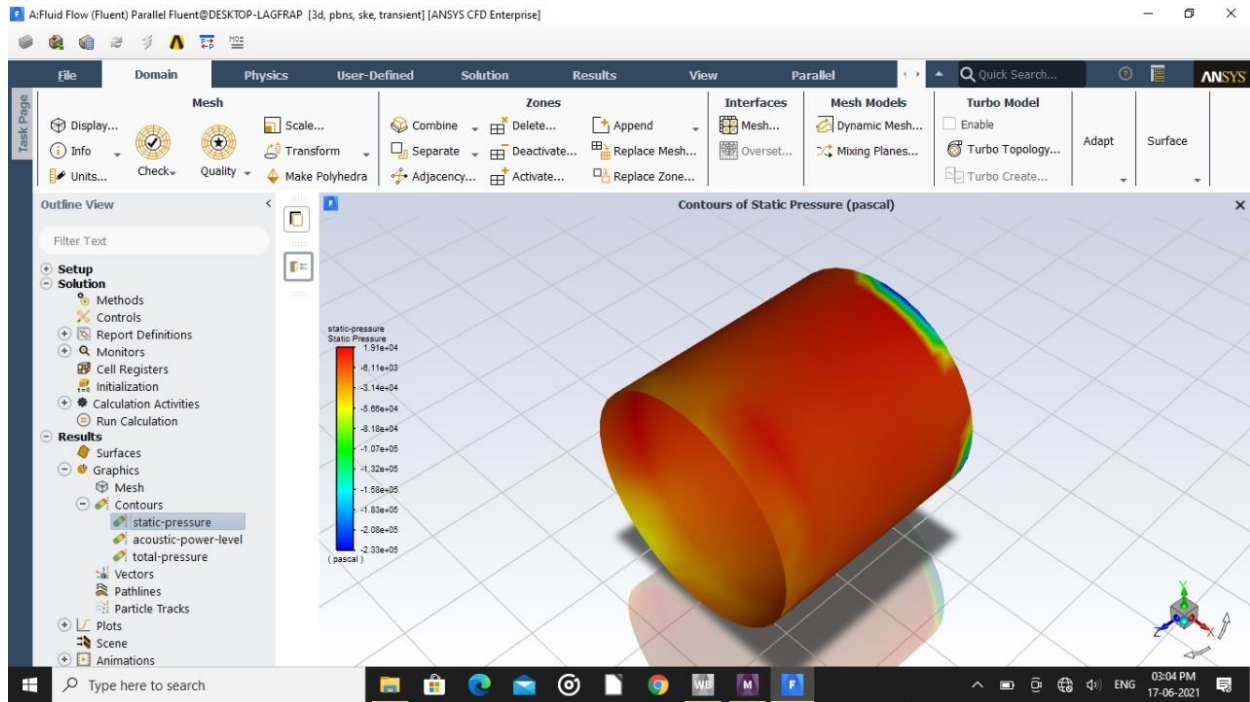


Fig No. 21

10.ANSYS RESULTS

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

11.BEARING FAILURES - CAUSES AND REMEDIES

Fatigue Failures are not common in general bearings unlike ball bearings.

The failures in bearings are mainly associated with insufficient lubricant, contamination of lubricant and faulty assembly. The Principal types of bearing failures are:-

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

12.CONCLUSION

The main objective of this project was to study the lubrication and impact of other factors on hydrodynamic journal bearing. On the basis of results from MATLAB software and discussions present in this report we can conclude that :

1. We can conclude that when the shaft speed is increasing the Sommerfeld number is also increasing.
2. The friction coefficient increases with the degree of misalignment at lower values of the eccentricity ratio.
3. The pressure distribution around the fluid film showed that the pressure variation along the circumference is varying greatly.
4. Also as the speed and the load on the bearing increases the pressure also increases.

13.FUTURE SCOPE

1. Pressure Measurement can be done over the bearing surface using experimental analysis.
2. Different lubrication can be studied to find the frictional torque between the bearing and the journal.
3. Different material can be used for journal to further reduce the frictional torque .
4. The effect of temperature on viscosity can be studied to examine the dependence of frictional torque on viscosity.

14. REFERENCES

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