REVIEW OF JOURNAL BEARING OIL CLEARANCE AND LUBRICATION MANAGEMENT FOR TURBO GENERATORS

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ABSTRACT
The Review of Journal Bearing Oil Clearance and Lubrication Management for Turbo Generators is reported. The underlying tribological principles relating friction, wear and lubrication are considered. The dependence of lubricant film thickness on : Radial clearance between the leg journal and cone; Radius of the leg journal; Lubricant viscosity; Rotational velocity of the cone and Applied load is observed. The interplay between friction, wear and lubrication is observed.

Key words: Journal Bearings, Oil Clearance, Lubrication, Turbo Generators, Friction and Wear.

INTRODUCTION

1.1 BACKGROUND TO THE STUDY

The Nigerian power sector with an installed capacity of 13,308 MW had only 6 158 MW operational in 2014. Of this 6158MW, only 3000 MW to 4 500 MW were actually being generated due to unavailability of gas, breakdowns, water shortage and grid constraints. The poor performance of the power plants has led to acute shortage of power across the country. Altogether, up to 2 700 MW of power generation capabilities are regularly lost due to gas constraints in a country with one of the largest natural gas deposit in the world. [1]

Up to 500 MW are lost due to water management, while several hundred megawatts are regularly lost due to line constraints. Industry, commerce and private households are suffering from a severe shortfall in electricity generation. With the intention of incentivising private-sector investment in the power sector, the government has privatised the generation and distribution sections in two waves. The proceeds are sensibly being dedicated to infrastructure expansion and, in the case of the second wave, a large part of the revenue has been earmarked for expansion of the country's array of hydropower plants[1]. However, the process of privatisation is still ongoing. At present it is impossible to say with any certainty whether the independent power producers who now form the backbone of the Nigerian power sector will be commercially viable. As part of the process, however, the government has started to encourage investments in both renewable energy and energy efficiency.[1]

According to The Nigerian Economic Summit Group [2], One hundred million Nigerians, representing 60% of the country's population, have no access to grid electricity. Those who do have grid access experience

extremely unreliable supply. Also [3] posits that Turbine generator bearings failure are a leading cause of power plant unavailability and can cause serious damage not only to rotors, stators and nearby equipment. Consequent on the foregoing therefore, it is worthwhile to do a Review of Journal Oil Clearance (Gap) and Lubrication Management for Turbo Generators in order to ensure health operating regimes of the journal bearings.

1.2 STATEMENT OF PROBLEM

The bulk of the generating units of the National Grid is made up of turbo generators. Now the power output from these units is dismal. Since bearings failure is identified in Literature review as a leading cause of power plant unavailability, Review of journal oil clearance and lubrication is justified.

1.3 AIMS AND OBJECTIVE

i) To do an overview of Tribology and tribological problems, contact mechanics, friction theories, wear, lubricants and lubrication

ii) to review the application of the tribological factors to the design , operation and management of journal bearings.

iii) To draw inference and hence conclusion on the safe use of journal bearings governed by tribological factors.

LITERATURE REVIEW

Tribology is the science and engineering of interacting surfaces in relative motion. It includes the study and application of the principles of friction, lubrication and wear. Tribology is highly interdisciplinary(Wikipedia. 2019). The word *tribology* derives from the Greek root τριβ- of the verb tribo "I rub" in classic Greek, and the suffix -logy from logia "study of", "knowledge of". Peter Jost coined the word in 1966, in the eponymous report which highlighted the cost of friction, wear and corrosion to the UK economy[4].

Despite the relatively recent naming of the field of tribology, quantitative studies of friction can be traced as far back as 1493, when Leonardo da Vinci first noted the two fundamental 'laws' of friction. According to da Vinci, frictional resistance was the same for two different objects of the same weight but making contact over different widths and lengths. He also observed that the force needed to overcome friction doubles as weight doubles. However, da Vinci's findings remained unpublished in his notebooks. (Wikipedia. 2019). The two fundamental 'laws' of friction were first published (in 1699) by Guillaume Amontons, with whose name they are now usually associated, they state that:

1. the force of friction acting between two sliding surfaces is proportional to the load pressing the surfaces together
2. the force of friction is independent of the apparent area of contact between the two surfaces(Wikipedia. 2019).

To reduce friction between surfaces and keep wear under control, materials called lubricants are used.. Unlike what you might think, these are not just oils or fats, but any fluid material that is characterized by viscosity, such as air and water. Of course, some lubricants are more suitable than others, depending on the type of use they are intended for: air and water, for example, are readily available, but the former can only be used under limited load and speed conditions, while the second can contribute to the wear of materials[4].

What we try to achieve by means of these materials is a perfect fluid lubrication, or a lubrication such that it is possible to avoid direct contact between the surfaces in question, inserting a lubricant film between them. To do this there are two possibilities, depending on the type of application, the costs to address and the level of "perfection" of the lubrication desired to be achieved, there is a choice between:

1. Fluidostatic lubrication (or hydrostatic in the case of mineral oils) – which consists in the insertion of lubricating material under pressure between the surfaces in contact;
2. Fluid fluid lubrication (or hydrodynamics) – which consists in exploiting the relative motion between the surfaces to make the lubricating material penetrate[4].

The viscosity is high in fluids that strongly oppose the motion, while it is low for fluids that flow easily.

TABLE 1: VISCOSITY COEFFICIENTS μ FOR SOME FLUIDS

Fluid	$\mu(Pa \cdot s)$
CO_2	$1.5 \cdot 10^{-5}$

Air	$1.8 \cdot 10^{-5}$
Gasoline	$2.9 \cdot 10^{-4}$
Water (90 °C)	$0.32 \cdot 10^{-3}$
Water (20 °C)	$1.0 \cdot 10^{-3}$
Blood (37 °C)	$4.0 \cdot 10^{-3}$
Oil (20 °C)	0.03
Oil (0 °C)	0.11
Glycerin	1.5

SOURCE: [4]

Fluids can be divided into two types according to their viscosity:

1. Newtonian fluids, or fluids in which viscosity is a function of temperature and fluid pressure only and not of velocity gradient;
2. Non-Newtonian Fluids, or fluids in which viscosity also depends on the velocity gradient.

There are three main causes of temperature variation that can affect the behaviour of the lubricant:

- Weather conditions;
- Local thermal factors (like for car engines or refrigeration pumps);
- Energy dissipation due to rubbing between surfaces.

In order to classify the various lubricants according to their viscosity behaviour as a function of temperature, in 1929 the viscosity index (V.I.) was introduced by Dean and Davis. These assigned the best lubricant then available, namely the oil of Pennsylvania, the viscosity index 100, and at the worst, the American oil of the Gulf Coast, the value 0. To determine the value of the intermediate oil index, the following procedure is used: two reference oils are chosen so that the oil in question has the same viscosity at 100 °C, and the following equation is used to determine the viscosity index

$V.I. = [(L - O_{Test})/(L - H)]x\ 100$

This process has some disadvantages:

- For mixtures of oils the results are not exact;
- There is no information if you are outside the fixed temperature range;
- With the advancement of the technologies, oils with V.I. more than 100, which cannot be described by the method above.

In the case of oils with V.I. above 100 you can use a different relationship that allows you to get exact results

$V.I. = [10^{N-1})/(0.00715)] +100$

$N = Log(H) - Log(O_{Test}))/(Log\ (v))$

where, in this case, H is the viscosity at 100 °F of the oil with V.I. = 100 and v is the kinematic viscosity of the study oil at 210 °F. An increase in temperature leads to a decrease in the viscosity of the oil. It is also useful to note that an increase in pressure implies an increase in viscosity. to evaluate the effects of pressure on viscosity, the following equation is used: $\mu = \mu_0 \exp\{\alpha p)$

where μ is the pressure viscosity coefficient p, μ_0 is the viscosity coefficient at atmospheric pressure and α is a constant that describes the relationship between viscosity and pressure.

To determine the viscosity of a fluid, viscometers are used which can be divided into 3 main categories:

1. Capillary viscometers, in which the viscosity of the fluid is measured by sliding it into a capillary tube;
2. Solid drop viscometers, in which viscosity is measured by calculating the velocity of a solid that moves in the fluid;
3. Rotational viscometers, in which viscosity is obtained by evaluating the flow of fluid placed between two surfaces in relative motion.

The first two types of viscometers are mainly used for Newtonian fluids, while the third is very versatile.

This coefficient makes it possible to subdivide, depending on its size, the damage suffered by various materials in different situations, passing from a modest degree of wear, through a medium, to a degree of severe wear.

TABLE 2 CLASS VERSUS USAGE LEVEL

Class	T_{usury}	Usage level
0	$10^{-13} - 10^{-12}$	Moderate
1	$10^{-12} - 10^{-11}$	
2	$10^{-11} - 10^{-10}$	
3	$10^{-10} - 10^{-9}$	Medium
4	$10^{-9} - 10^{-8}$	
5	$10^{-8} - 10^{-7}$	
6	$10^{-7} - 10^{-6}$	
7	$10^{-6} - 10^{-5}$	Severe
8	$10^{-5} - 10^{-4}$	
9	$10^{-4} - 10^{-3}$	

Instead, to express the volume of wear V it is possible to use the Holm equation

- (for adhesive wear)
- (for abrasive wear)

where W / H represents the real contact area, l the length of the distance travelled and k and are experimental dimensional factors.

Wear measurement

This can be achieved through: 1) visual inspection 2) Radioactive tracer 3) Accelerating wear times (e.g high pressure contact tests).

Tribofilms are thin films that form on tribologically stressed surfaces. They play an important role in reducing friction and wear in tribological systems.

Stribeck curve

The Stribeck Curve shows how friction in fluid-lubricated contacts is a non-linear function of lubricant viscosity, entrainment velocity and contact load.

According to [5], after the power plant blackout, lubrication pumps of turbo-generator journal bearings were not turned on. In that moment the oil supply from the accidental spare oil tanks should be activated in all turbo-generator bearings. Increased temperature and vibrations registered by the turbo-generator control system were indicative of severe damage of some generator bearing. When the turbo-generator was stopped, the bearings were disassembled. The major damages were detected on the one of the generator bearings. A white metal layer of the sleeve was melted, journal surface was damaged, as well as the oil supply channel was blocked [5].

Also, a theoretical treatment of the problem of journal bearings modeling connected to electronic oil injection into the bearing gap is reported. The feasibility of influencing the static behaviour of hydrodynamic forces by means of such oil injection is investigated. The lubricant is injected into the bearing gap by two mechanisms of lubrication: the conventional hydrodynamic lubrication and through orifices distributed along the bearing surface (active lubrication in the radial direction). By controlling the pressure of the oil injection, it is possible to get large variations in the active hydrodynamic forces; such effects could be useful for reducing vibrations in rotating machines[6].

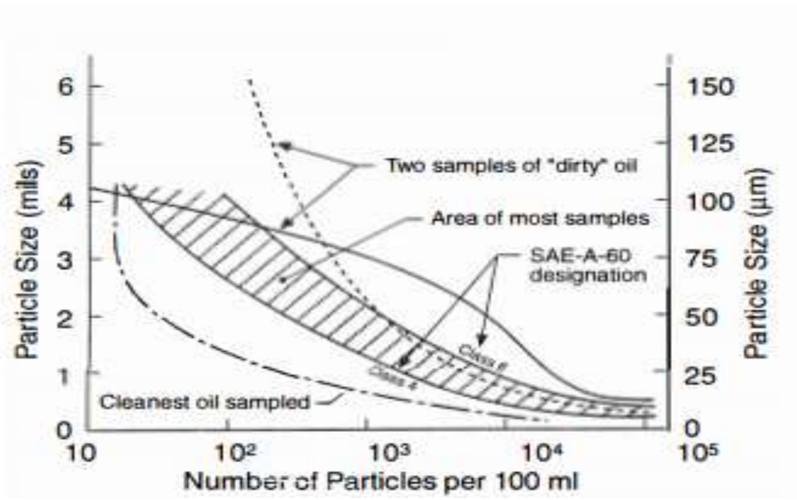


Figure 1. Typical Contamination Levels of Turbine Oils in Electric Utilities SOURCE: [3].

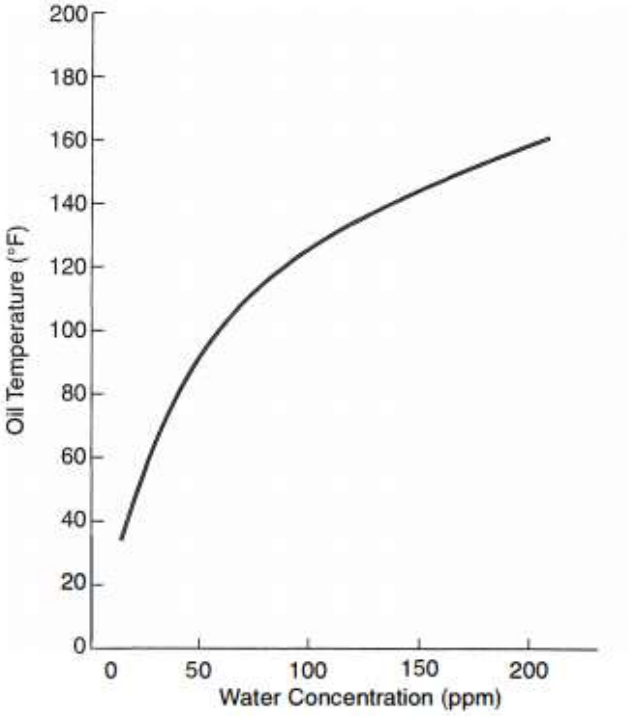


Figure 2. Equilibrium Water Content in a Turbine Oil at 40 Percent Humidity SOURCE:[3].

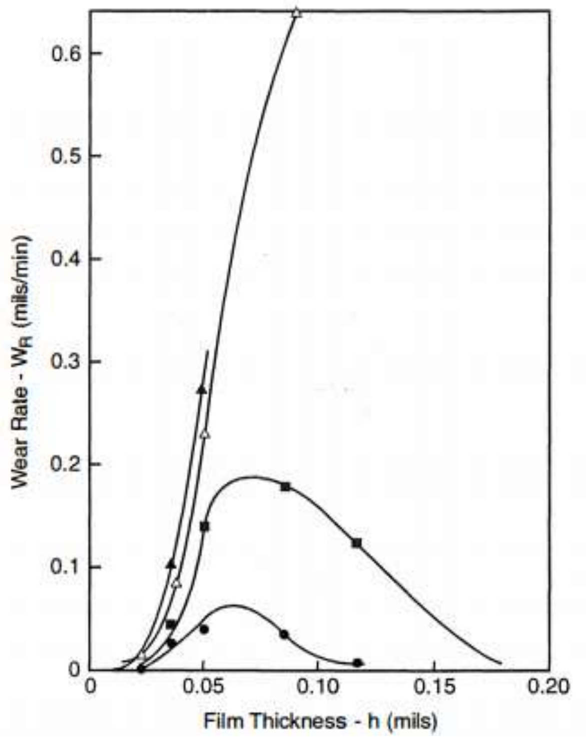


Figure 3. Wear Rate Vs Film Thickness for Different Voltages. SOURCE: [3].

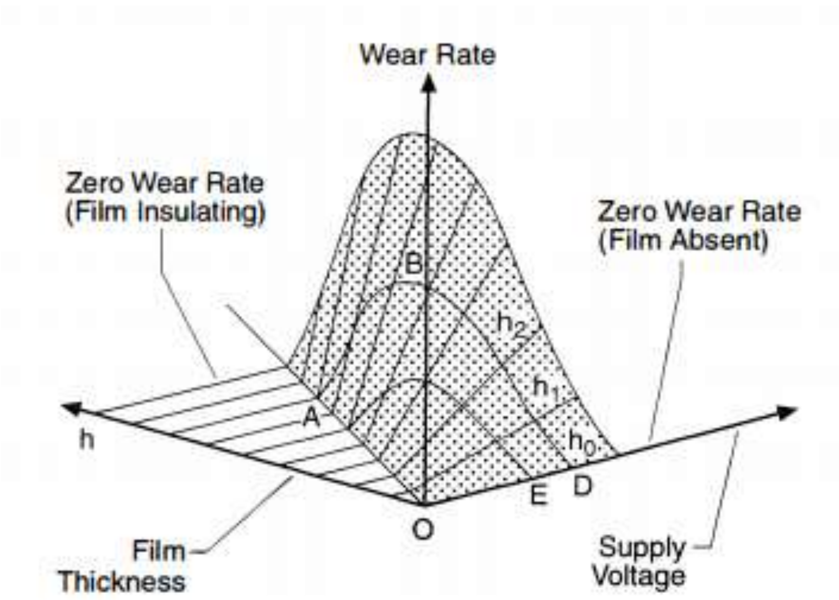


Figure 4. Effect of Film Thickness and Voltage on the Wear Rate Due to Electrical Pitting. SOURCE:([3].

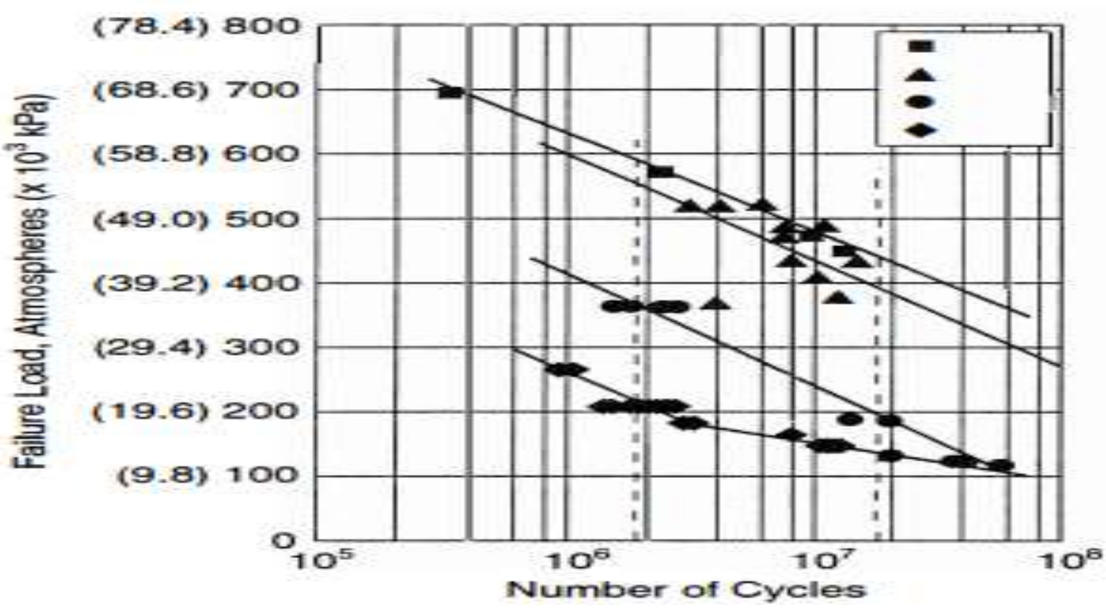


Figure 5. Failure Diagrams for the Four Modes of Loading. SOURCE:[3].

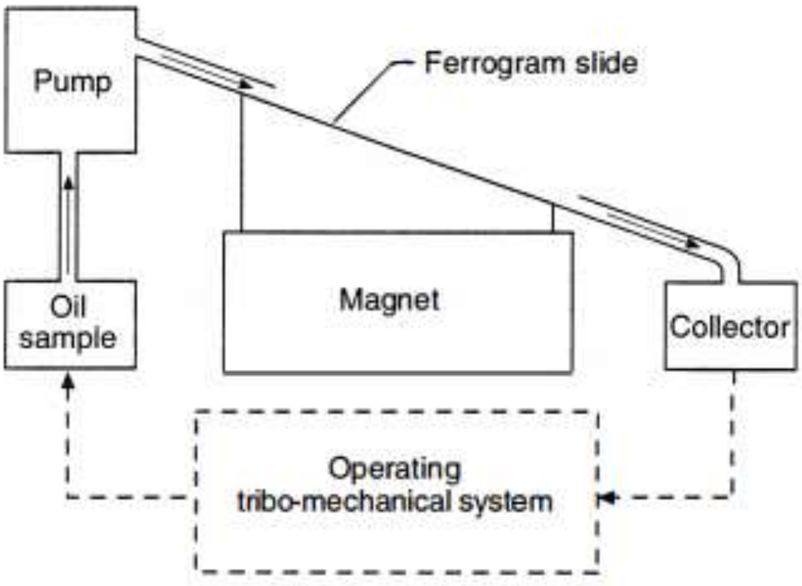


Figure 6. Schematic Representation of Ferrograph Analyzer. SOURCE:([3].

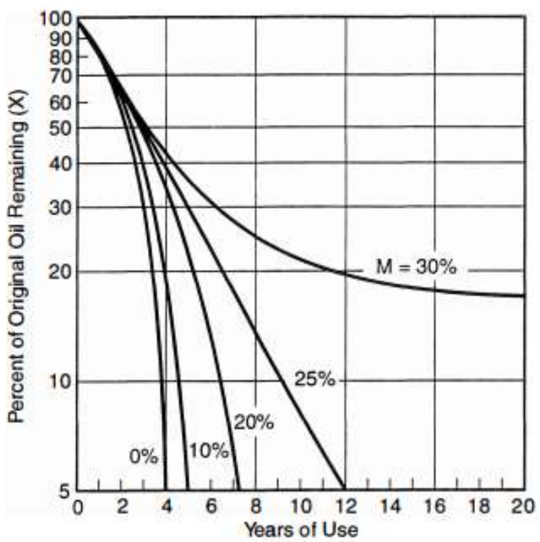


Figure 7. Effect of Makeup Rate on Oil Degradation (Turbine Severity, B = 25 percent per year. SOURCE:[3]

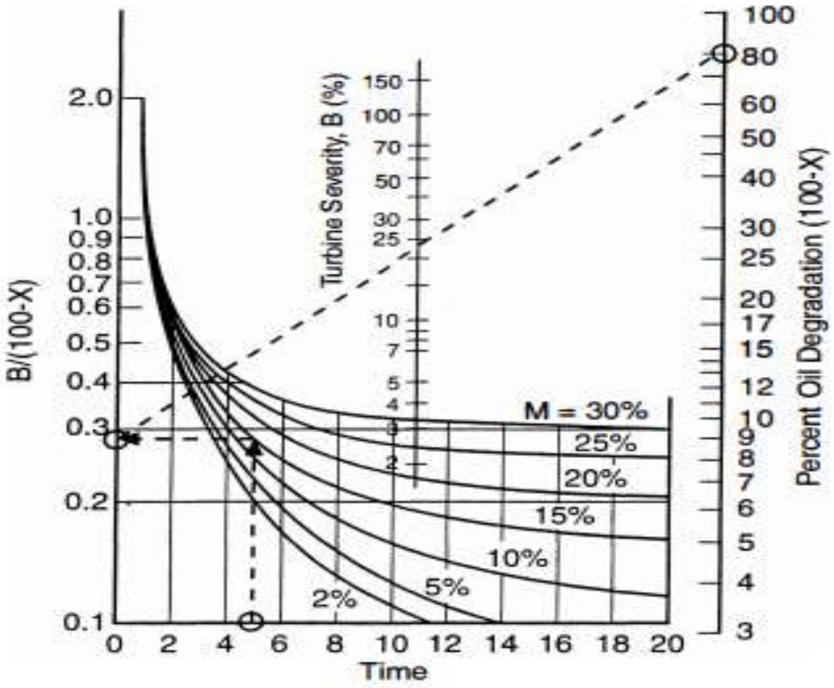


Figure 8. Effect of Turbine Severity (B) and Makeup Rate (M) on Oil Degradation. The dotted lines show the process for obtaining a value for turbine severity B. In the example, the turbine oil has been in service for five years, and the annual makeup rate is 15 percent. The oil has degraded from a rotary bomb test life of 1700 minutes initially to only 350 minutes, a loss of 79.5 percent. Starting at five years on the time axis, a point on the 15 percent makeup curve isis determined, and a line is projected left to the B/(100x) axis. A straight line between this point on the B/(100x) axis and a point at 79.5 percent on the percent oil degradation axis intersects the turbine severity scale at 22 percent per year. SOURCE:([3].

4. Information and an approach intended to help engineering and other plant personnel trouble shoot bearing problems and diagnose the mode of bearing failure are presented. Six leading mode of bearing failure (abrasion, corrosion, electrical pitting, fatigue, overheating and *wiping are covered in detail). Turbine generator bearing failures are a leading cause of*

power plant unavailability and can cause serious damage not only to bearing systems but also to rotors, stators and nearby equipment.

TABLE 3 - MODES OF BEARING FAILURE

MODE	OTHER NAMES
Abrasion	Gouging, Scoring, Scratching
Bond Fracture	Spalling
Cavitation Erosion	Cavitation
Corrosion	Chemical Attack
Electrical Pitting	Frosting
Erosion	Worm Tracts
Fatigue	
High Chromium Damage	Wire Wool, Black Scab
Non-homogeneity	Blistering, Porosity
Overheating	Mottling, Anisotropy, Ratcheting, Sweating
Seizure	
Structural Damage	
Surface Wear	Black Scale
Tin Oxide Damage	Wear

Wiping	Smearing, Polishing

Ferrography

In use at dozens of utilities, ferrography is one of the more recent methods of contaminant identification. Its principle of operation (Figure 32) consists of pumping a sample of oil at a slow, steady rate between the poles of a magnet. The fluid runs down an inclined microscope slide and the net effect of the viscous and magnetic forces acting on the particles is to sort them by size. The larger particles are deposited first, and the smaller particles are carried farther downstream. Information on the morphology of the deposited particles is obtained with the aid of a bichromatic microscope, which uses simultaneously reflected red light and transmitted green light. Metal particles as small as one micron reflect red light and bloc green light and thus appear red. Particles composed of compounds allow much of the green light to pass and appear green or, if they are relatively thick, yellow or green

• The pedestal housing the failed bearing had shifted downwards, which unloaded the lower half and imposed an unknown load on the upper half. • The unbalance load imposed a rotating force on both halves of the bearing; but with the static load diverted upwards, the upper half was loaded much more severely than the lower half. The upper half with its overshot groove had a much lower load capacity than the lower half. Load capacity is proportional to (L/D)2, where L is length, D, diameter. In this case the load capacity of the upper half relative to the lower half is given in Equation (1). Ratio = (L/D)2 • 2 = (1/3)2 • 2 = 22% (1)

Specifications

Specifications for new lubricating oil are formulated by equipment manufacturers, oil suppliers, and certain technical societies active in the field of turbine lubrication. A minimum specification guide has been issued by the American Society for Testing and Materials (ASTM) and titled "Standard Specification for Mineral Lubricating Oil Used in Steam and Gas Turbines (D 4304)."

Excessive or uncontrolled build-up of contaminants should alert the turbine operator to identify the source, take corrective action, and determine whether oil purification equipment or system filters are properly functioning. Analyses In-service monitoring of the condition of a turbine oil should focus on the following properties: Antirust protection Remaining oil life (oxidation stability) Viscosity Total acid number Cleanliness Foaming tendency Color/appearance Water content Flash point

Turbine Severity Level

Each turbine generator lubrication system is unique due to exclusive conditions that arise during construction and operation of the system. These conditions set the rate at which a new charge of fresh oil will lose its oxidation resistance. A property called turbine severity (B) level has been established which can be used to take these conditions into consideration when monitoring the remaining oxidation resistance of the oil during its service life (DenHerder and Vienna). "B" is defined as the percentage of fresh oil oxidation resistance lost per year due to oil reactions in the turbine generator lubrication system. "B" takes into consideration the following three factors: • Amount of make up oil added to the system to replenish the oil oxidation resistance • Time that the oil has been in use • Oxidation resistance that remains as determined by a RBOT, ASTM D 2274-84 Equation (2) determines turbine severity, B: B = M $(1-X/100)/(1-e^{-Mt/1oo})$ (2) where M = Amount of oil added as makeup into the system per year, expressed as a percentage of the total amount of oil originally placed in the system (percentage per year) X = Amount of oxidation resistance that remains in the oil, expressed as a percentage of the original oxidation resistance of the oil (percentage of fresh oil) = Amount of time the original oil has been in service in years The effect of makeup rate, M, on oil degradation for a turbine with a severity level of 25 percent per year is shown in Figure 36. The severity level for a particular lubrication system should be determined over a period of time beginning with initial operation or installation of a fresh oil charge. Keeping accurate records of the amount of oil makeup is essential, and RBOT should be conducted at three- to six-month intervals for one to two years. By knowing the oil makeup and degradation of the oil with time, the turbine severity for the oil can be found from Figure 37.

5. Hydrodynamic Lubrication

Journal bearings are able to function efficiently and avoid wear because of a hydrodynamically generated film of lubricant that separates the rotating surfaces. The thickness of this lubricant film depends on the following variables:

Radial clearance between the leg journal and cone

Radius of the leg journal

Lubricant viscosity

Rotational velocity of the cone

Applied load

So long as the hydrodynamic conditions exist to create a lubricating film between the two rotating surfaces, the life of a journal bearings is virtually limitless. If the film breaks down for any reason (e.g., high loads, low speeds, low viscosity), the bearing surfaces will come into contact with each other, and a phenomenon called boundary lubrications takes place. Film thickness decreases as rotational speed decreases or as bearing loading increases.

Inhydrodynamic lubrication, heat generated is a function of lubricant viscosity, applied load, and relative speed between cone and journal. As bearing temperature increases, lubricant viscosity and film thickness decrease creating a potential for journal bearing malfunction. Heat generated varies with the square of the speed. A twofold increase in bearing rotational speed produces a fourfold increase in the generation of heat in the bearings. In larger bits, heat generation can be significant and large bearing design must include provisions for removal of heat from the bearing area.

In another development, the optimum design procedure of high-speed, short journal bearings under laminar and turbulent flow conditions is developed based on three kinds of methods such as Successive Quadratic Programming, Genetic Algorithm and Direct Search. Applying the short bearing assumption to the modified turbulent Reynolds equation, simplified closed form design formulae are obtained for the eccentricity ratio, friction force, film temperature rise, supply lubricant quantity and whirl onset velocity as a function of design variables such as radial clearance, slenderness ratio and averaged viscosity of lubricants. Then, using these formulae, the design variables, which optimize the objective function given by a linear summation of temperature rise and supply lubricant quantity with respective weighting factor, are determined for a wide range of journal rotational speeds under various constraints.[7]

According to [8], Oil flow rate to tilting pad thrust bearings influences two important bearing operating characteristics; bearing power loss and pad operating temperatures. Reducing oil flow rates is desirable as this reduces the size and cost of the oil supply system and also can reduce bearing power losses. However, this can increase bearing operating temperatures, reducing the load capacity. Results of laboratory tests of thrust bearings with babbitt faced pads are reported in which oil flow rates were varied and losses and temperatures were measured[8].

Also, a newly developed experimental technique enabled the determination of lubricant film thickness distribution in elastohydrodynamic (EHD) point contacts with high accuracy and spatial resolution. This technique uses a computer-aided system developed for this purpose, which with the help of differential colorimetry makes the reconstruction of the lubricant film shape possible. Chromatic interferograms were obtained with the use of a conventional optical test rig. Obtained results were compared with a numerical solution of Newtonian isothermal compressible EHD lubrication of point contacts. Experimental and numerical three-dimensional representations of a lubricant film thickness and shape for various operating conditions are shown. Comparisons of obtained central and minimum film thickness values with Hamrock and Dowson equations are presented.[9].

[10] posits that Surface damage or failure of rolling element bearings is often due to inadequate lubricant replenishment of the contact. Our understanding of the mechanisms of starved elastohydrodynamic lubrication and the behaviour of thin fluid films has advanced significantly in recent years and, thus, provides a basis for a fundamental study of different bearing failure modes. In this paper a possible explanation for the surface damage observed in grease lubricated spherical roller bearings operating at low temperatures is advanced. This type of bearing failure is characterized by 'brown bands' that form identical patterns on every element of the bearing: the roller set, inner and outer ring. In this case damage is attributed to a single defect in the lubricant film, which is then propagated around the bearing. Such a defect might be initiated by local scraping, and removal, of the lubricant film by a bearing cage. It is the propagation of the film defect and the role of lubricant replenishment that is

studied in this paper. In the starved regime insufficient lubricant replenishment of the defect results in a locally depleted film which, combined with surface roughness, leads to surface damage and possibly brown band formation.

The numerical work studies the influence of a local lubricant defect on the central film thickness in the contact. An established starvation model is used to explore the relationship between oil inlet film thickness and the maximum defect size that can be replenished. In parallel experimental work a local defect has been deliberately created in the lubricant film in a rolling contact. The behaviour of the defect has been studied under different levels of lubricant replenishment. This work confirms qualitatively the effects predicted by the numerical study[10].

Energy losses in the spring-supported thrust bearings used in many large hydroelectric generator units were estimated to be small compared to the rate of energy (power) generation but, nevertheless, commercially significant. The purpose of the present study was examine the influence of oil viscosity on power loss. Experiments were performed using a test facility containing a thrust bearing of 1.2 m outer diameter and both power loss and temperature rise were measured for oils of ISO grade 32, 46 and 68, all at various oil pot temperatures. Power loss and temperature rise decreased as the viscosity of the oil in the oil pot decreased. Minimum oil film thicknesses were predicted with numerical analysis using a specialized software package (GENMAT). The accuracy of this calculation was supported by the good agreement between the temperature rise predicted by numerical analysis and the experimentally determined values. Provided film thickness were adequate to avoid any danger of wiping (10 - 15 µm), the present study showed clearly that significant energy savings could be realized in the large spring-supported thrust bearings and associated guide bearings by lowering oil viscosities [11].

The combined Effects of two sided purely longitudinal, transverse and isotropic roughness and shear thinning and viscoelasticity of engine oils on dynamically loaded finite journal bearings in mixed lubrication are analyzed, using Christensen's stochastic model of hydrodynamic lubrication of rough surfaces and considering the running-in effect on asperity height distribution. Shear thinning and viscoelasticity are characterized by the power law and the Maxwell fluid models respectively. Results show that the combined effect of roughness and non-Newtonian rheology on the bearing characteristics is closely tied up with the roughness texture and structure, features of nominal geometry, journal mass, and operating conditions. [12].

Particles present as contamination in a lubricant are known to increase the wear of components in rolling bearings. This wear leads to deterioration in running accuracy and vibration levels, which results in early replacement. Experiments have been conducted to investigate the nature and mechanism of the damage caused by hard particles (harder than either counterface) to bearing surfaces. Initially considering single particle scratching between two surfaces subject to sliding and rolling using a test rig to create a single elastohydrodynamic contact (Ball on Cylinder). The nature of an individual scratch is examined and then compared to the results from a test with more realistic concentrations of particles.

The results suggest that particle wear may often involve a more complex process than pure abrasion.[13].

The lubricating life of a grease in a rolling element bearing is reduced by operation at high temperatures and this can result in premature failure of the bearing. A grease experiences severe conditions in an operating bearing where the combination of high temperatures and sustained mechanical working result in gross physical and chemical changes to the grease.

These changes have a significant effect on the ability of the grease to replenish the contact and maintain a lubricating film, particularly under starved inlet conditions. Extended operation at high temperatures promotes evaporation of low molecular weight base oil components (1) and oxidation of one or both of the grease components (2)(3). The presence of small amounts of transition metals and their oxides can accelerate these processes and such material is commonly found as wear debris in bearings (4). It is difficult to disentangle these effects and this paper concentrates on the effects of thermal ageing on the lubricating ability of the grease.

Simple thermal ageing tests have been carried out on two lithium hydroxystearate greases and the resulting changes in their chemistry characterised by infrared spectroscopy. The lubricating performance of the aged greases has been assessed by measuring film thickness and oil release in a rolling contact under starved conditions. Results from infrared analysis have shown that the oxidation process is accelerated at a temperature of 120 ° C forming carboxylic acids and related species. The film thickness results showed that the aged greases give a lower equilibrium film thickness and this correlates with reduced oil release.[14].

Tribology has a part to play in the development of energy efficient technology at many different levels; in manufacture, in enabling components to operate with long life and low friction during use and in making practicable innovative, energy saving engines and transmissions. This paper considers five specific areas in which Tribology is challenged by the development of energy efficiency. These range from the reduction of friction in components, and consequent problem very thin hydrodynamic films, to the tribological implications inherent in the development of very high temperature adiabatic engines. In all of these areas, and also many others, it is concluded that Tribology has a key role to play in the introduction of energy efficient technology.[15].

Oil leaving the bearings is about 70°C. The oil system, like the hydraulic system, must be kept free of dirt and small particles.

The factors that affect lubricating oil are:

•Overheating and exposure to air result in oxidation. Fine metal particles from wear accelerate oxidation. This changes the oil composition and viscosity. Sludge may result. .•Water contamination from:•Leaking turbine and pump seals. •Condensation in humid air. Water leaks in heat exchangers. Emulsion forms when oil and water mix. Oxidation increases emulsification. Rust contamination. Rust particles. Catalyze the rate of oil oxidation. Scratch journals. Block small clearances in the governing system

Air contamination. Air bubbles cause sponginess in hydraulic controls. They reduce load capacity of oil films. Air accelerates oxidation. It can generate foaming.

Oil, for the turbine model and application in question, needs the correct:

Viscosity, Load carrying ability, Oxidation stability, Protection against rusting, Water separating ability, Foam resistance, Entrained air release, Fire resistance.

15. When bearings are lightly loaded (as in vertical machines), bearing pads are pre-loaded to provide a stabilizing load to the shaft and minimize dynamic motion. The pre-load is defined in Eq. (1.1).

Preload =(Rpad −Rbearing)/(Rpad −Rshaft) (1.1)

According to [16], typical turbine lubricants available and used today are based either on classic high quality petroleum base stocks or some form of synthetic base stock such as ester, poly-alkylene-glycol (PAG), or poly-alpha-olefin (PAO). Additionally, any number of additives are blended into the base oil to improve specific characteristics such as anti-oxidation additives which extend lubricant. life, extreme pressure additives to protect surfaces in high pressure contacts or anti-foaming additives to reduce the lubricant's tendency to trap air bubbles. Most important to this work are additives which improve the viscosity index (VI) of the lubricant. A lubricant's VI describes how the viscosity of a lubricant changes with variations in temperature. Generally, as temperature increases, the viscosity of a fluid decreases following logarithmic relationship as shown in Fig. 11. The five lubricants have equal viscosity at 100°C but very different viscosities at 40°C due to the differences in their VI

Because fluid flow losses in a system are directly coupled to the viscosity of the fluid and the force needed to shear it, Eq. (1.2), any reduction in viscosity can potentially result in direct power savings due to reduced flow losses.

F = μ*Area (Velocity/Film Thickness) 1.2)

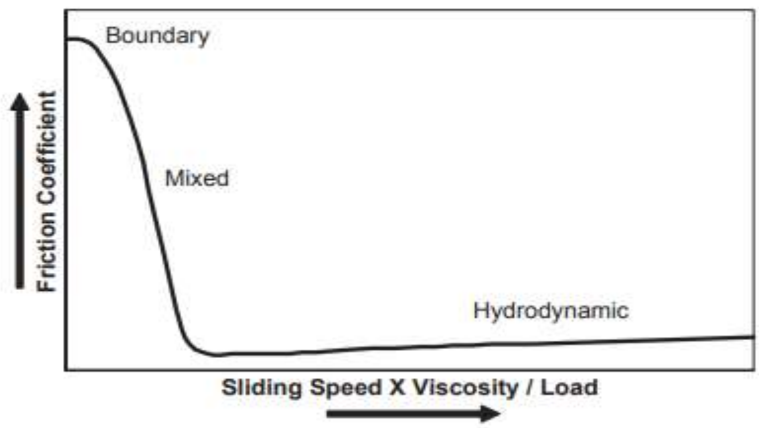


Figure 9. The Stribeck curve showing the transition from boundary to mixed and finally to hydrodynamic lubrication regimes with changing speed, viscosity or load on the contact. SOURCE: [16].

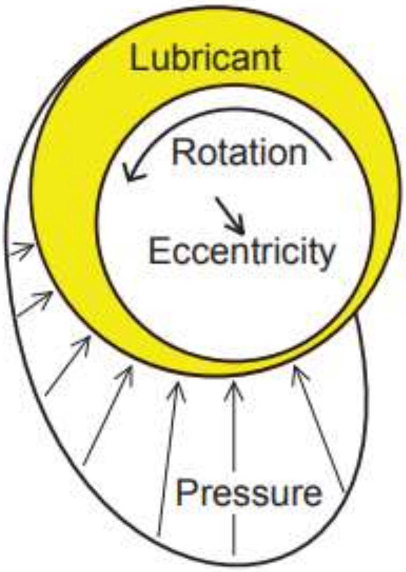


Figure 10. Hydro-dynamic pressure buildup in a journal bearing. Lubricant is nornally supplied to these bearings from the sides or top when the bearing operates with a horizontal shaft.With a vertical turbine, the bearing is often partly immersed in an oil bath and the pressure field rotates around the bearing with the shaft's motion. SOURCE: [16].

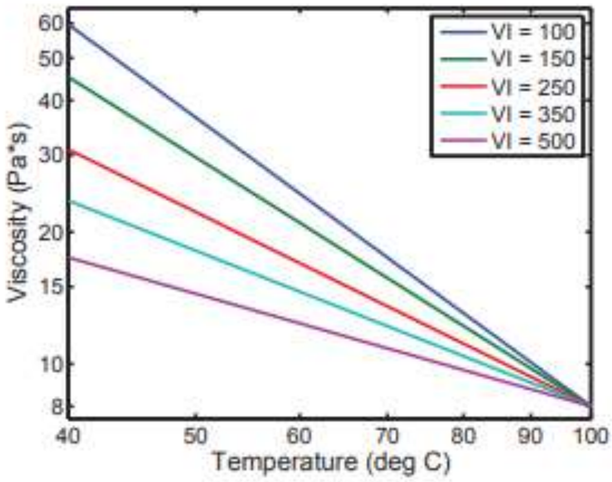


Figure 11. Variation in dynamic viscosity vs temperature for fluids with different viscosity indices. All fluids have equal viscosity at 100°C but viscosity at 40°C varies greatly SOURCE:[16].

A prototype of the next-generation, high-performance micro-turbine system was developed for laboratory evaluation. Its unique feature is its utilization of water. Water is the lubricant for the bearings in this first reported application of water-lubricated bearings in gas turbines. Bearing losses and limitations under usage conditions were found from component tests done on the bearings and load tests done on the prototype micro-turbine. The rotor system using the water-lubricated bearings achieved stable rotating conditions at a rated rotational speed of

51,000 rpm. An electrical output of 135 kW with an efficiency of more than 33% was obtained. Water was also utilized to improve electrical output and efficiency through water atomizing inlet air cooling (WAC) and a humid air turbine (HAT). The operation test results for the WAC and HAT revealed the WAC and HAT operations had significant effects on both electrical output and electrical efficiency [17].

Similarly, [18] reviews the problems arising in, and the system employed on, the lubrication of large modern steam-turbine electric generators. A description is given of the general layout, enumerating the various components and the reasons for their inclusion. The general qualitative relations governing the operation of film-lubricated bearings are briefly stated; published results of full-scale experiments on commercial types are compared; and a simple method is given for estimating the power absorbed, based on practical experience. Various types of oil pumps, and the principles adopted in fixing their capacity are described, the results on gear type pumps being shown by several curves[18]. The development of modern forms of oil-cooling apparatus is traced, attention being given to factors affecting the rate of heat transmission obtainable; a series of tests carried out on a modern type are described. Curves showing the relation between heat transmission rates and oil and water velocities obtained from these tests are reproduced. The disposition, design, and capacity of oil tanks are discussed, together with the practical aspects of piping layouts. The concluding section deals with the properties of lubricants, factors contributing to deterioration, and methods of purification commonly employed[18].

In another vein, a survey collects the efforts to understand the sources and consequences of damage to babbitted industrial bearings, which operate by means of a hydrodynamic, or hydrostatic, film. Major individual damage types are discussed in the context of major damage categories.. Hydrodynamic bearings support a rotating shaft with its associated loads, by means of a pressure field developed within a lubricant which separates the solid surfaces.

Generation of the separating pressure requires the presence of a converging film thickness and lubricant viscosity. With sufficient pressure over a bearing area, complete support of the loads is achieved avoiding contact between the rotating and stationary components. In theory, this hydrodynamic film prevents wear and degradation of the rotating and stationary components. For industrial equipment, the rotating shaft is more valuable than the supporting bearings. Therefore, the bearing inner diameter is lined with a sacrificial material to absorb damage and to protect the journal surface (for radial bearings) or thrust collar (for thrust bearings) from damage. To serve as a sacrificial material, this lining should be softer than the journal surface and accept contact from hard contaminants in the lubricant. As a consequence of this material combination, inspection of the lining surface, often a tin-based babbitt (whitemetal) , can provide understanding as to both the future operating integrity of the hydrodynamic film, as well as suggesting mechanisms in the machine, lubrication system or bearing which may be degrading [19].

. Most of the mechanisms which degrade the hydrodynamic film will be due to machine or lubricant conditions. A degraded hydrodynamic film can be defined from several perspectives including: Loss of adequate load capacity for normal operation. Loss of adequate load margin for anticipated overload. Hot operation leading to loss of oil film due to loss of lubricant viscosity Increases in machine vibrations due to bearing wear Increases in machine vibration due to changes in machine dynamics associated with wear of bearing degradation.

Three broad categories of observations can be established:

a) Loss of babbitt material, b) babbitt surface displacement and c)supporting structure degradation. Loss of material can be defined in terms of location and volume [19].

1. Bearings are used to support a rotating shaft in a machine. It is possible to reduce the weight of the machines, by increasing the speeds, it is not a common practice to go in for high-speed machines. These high-speed machines naturally need suitable bearings to withstand high dynamic forces and seven operating condition like misalignment, axial thrust etc. Lubrication also plays an important role in the bearing design and application. There are different types of bearings used for different speeds and applications.
2. The most important objectives of a bearing design is to reduce frictional losses, insure stable running of the rotor system in the extreme and normal condition ensuring long life of the bearing, and particularly no wear to the rotating journal etc. stationary bearing generates frictional losses, which are converted into heat energy leading to the temperature rise of the bearing [20].

If the specific bearing pressure is less than 5 kg/cm2, the bearing will be unstable. For journal bearings, the guide line for the limits of specific bearing pressure is about 8-25 kg/cm2. If specific bearing pressure is too high, the oil film will be very thin and there may be a chance of occasional metal to metal contact. If the specific pressure is too low, the oil film will be very thick and the rotor will be just floating on the oil film. The rotor will not be stable and for any small disturbance, the rotor becomes unstable. More over lightly loaded condition leads to a phenomenon called

HALF FREQUENCY WHRIL. D = is the diameter of the bearing in cm,

L = is the length of the bearing in cm

Where, Wb= is the load on the bearing in kg

P = (Wb) / (L x D) Kg/cm2 …………………………………………...........2.1

. This is the static load per unit area of the projected surface of the bearing

SPECIFIC BEARING PRESSURE

Generally in journal bearings, the L/D ratio of bearing is maintained in the range of 0.6-1.0. For bearings in the speed range about 3000 rpm, generally L/D ratio is taken as 0.8. Peripheral velocity: The peripheral velocity of the journal in the bearing is given by the equation u = (3.14 x D' x n)/60 meters/sec2 where, D' is the diameter of the journal in meters. N is the speed in rpm..[20].

L/D RATIO

From practical point of view (measurement point of view), half the value of horizontal clearance is called SIDE CLEARANCE. During assembly of the bearing and during inspections, when the journal is stationary, the journal will be resting in the bearing The bearing clearances are measured as TOP and SIDE CLEARANCES and compared with the design values

Horizontal clearance = Ch = Dh – Dj

Vertical clearance = Cv= Dv – Dj

In this case, the horizontal diameter of the bearing Dh is more than the vertical diameter of the bearing Dv. Generally the following values are maintained. (ii)Elliptical bearings (two-lobe bearings)

In cylindrical bearings, the diametrical clearance is generally maintained 0.15-0.2% of the journal diameter.

Cd = Db - Dj Diametrical clearance = Bearing inner diameter - journal diameter Cylindrical bearing.

BEARING CLEARANCES AND CLEARANCE RATIO:

Horizontal diameter -Vertical diameter)/ Horizontal diameter

CLEARANCE RATIO

This is the ratio of diametrical clearance to the bearing diameter. This is a non dimensional quantity. m =Cd/D. This acts as a guide line for selecting the clearance for different journal diameters of any particular type of bearing. ELLIPTICITY RATIO:

N = ((e)/(c/2)) = ((2e)/c) It is a dimensionless quantity, which is the ratio of eccentricity to the radial clearance.

ECCENTRICITY RATIO:

Eccentricity is the distance between the bearing centre and the journal centre. The eccentricity depends on the load on the bearing, viscosity and peripheral velocity of the journal. This is denoted by e.[20]

ECCENTRICITY AND ECCENTRICITY RATIO:

Ø N' = Speed of the journal in rps

Ø D' = diameter of the journal in meters.

Ø Wb =load on the bearing in kg

Ø u =coefficient of friction of hydrodynamic film

Where Frictional force on the rotor = u x Wb

P = {frictional force on the rotor} x {distance traveled by the rotor journal in one sec}

The frictional losses in a bearing are given by

-Temperature rise allowed for oil.θ)… Where, P -Frictional losses in KW r -Density of oil on gm/cm2 n -Specific heat of oil θ (iii) The temp rise allowed for the oil Flow rate requirement of lubricating oil =P/(r x n x (ii) The specific heat of oil (i) Heat generated due to frictional losses. The quantity of oil flow required shall be such that, the oil should be adequate to carry away the frictional heat generated in the bearing and keep the bearing temperature within the limits. Maximum allowable temperature for

Babbitt is 120ºC. In the process of cooling the bearing, the temperature of oil increases. The rate of flow is dependent on the Quantity of oil flow required

The representation of the rotor bearing system as mass, spring and damper elements . The rotor natural frequency (critical speed) is influenced by the combined stiffness of the oil film + support + foundation. In actual practice the journal is supported on an oil film of the bearing, and the bearing itself is supported bearing pedestal (or simply called the support) which in turn rest on the foundation. So in addition to the oil film stiffness, there are stiffness of support and stiffness of foundation in series [20].

BEARING STIFFNESS AND DAMPING COEFFICIENTS:

67% copper, 28% tin, 5% lead 75% lead, 10% tin 76% copper, 24% lead 80% lead, 15% antimony, 5% tin 89% tin, 7% antimony, 4% copper 90% tin, 10% copper . There are many Babbitt alloys in addition to Babbitt's original. Some common compositions are: When tin is used as the softer metal, friction causes the tin to melt and function as a lubricant, which protects the bearing from wear when other lubricants are absent. However, its structure is made up of small hard crystals dispersed in a softer metal, which makes it a metal matrix composite. As the bearing wears, the softer metal erodes somewhat, which creates paths for lubricant between the hard high spots that provide the actual bearing surface. Babbitt metal is most commonly used as a thin surface layer in a complex, multi-metal structure. Babbitt metal is soft and easily damaged, which suggests that it might be unsuitable for a bearing surface [20].

.Furthermore, a review of several different types of hydrodynamic journal bearings that are commonly found in turbo-machinery is presented. Emphasis is placed on the key geometric design parameters of each type. The discussion covers plain journal, axial groove, pressure dam, offset split, lemon bore, multi-lobe and titling-pad bearings. The application of the critical speed map and some basic non-dimensional bearing parameters as tools for preliminary bearing selection and comparison are discussed. These tools are applied to two case studies, which demonstrate the proper application of different bearing designs to industrial turbo-machinery. Industrial turbo-machinery, such as steam turbines, gas compressors, pumps and motors, contain a variety of different types of hydrodynamic journal bearings. The types of bearings most commonly found in turbo-machinery include: plain cylindrical, axial grove, pressure dam, lemon bore, offset pivot, three-lobe, four-lobe, tilting-pad. The reason for such a large selection of bearings is that each of these types has unique operational characteristics that render it more suitable for one application than another [21]. The fundamental geometric parameters for all journal bearings are diameter, pad arc angle, length-to-diameter ratio, and running clearance. For the bearing types consisting of multiple pads, there are also variation in the number of pads, preload, pad pivot offset angle, and orientation of the bearing (on or between pads). In addition to the geometric parameters, there are several important operating parameters. The key operating parameters are oil viscosity, oil density, rotating speed, gravity load at the bearing, and applied external loads. Volute loads in pumps and mesh loads in gear boxes are examples of external loads. The plain journal bearing, shown in Figure 12, is the most basic hydrodynamic journal bearing. As the name implies, this bearing has a plain cylindrical bore. A shaft rotating in a plain journal bearing is illustrated in Figure 13. The eccentric rotating [21].

Note that the plain journal, axial groove and pressure dam bearings have only one bore (the set bore is the same as the pad bore). The difference between the pad machined bore radius and the journal radius is the pad machined clearance. The difference between the bearing set bore radius and journal radius is the bearing set clearance. The set clearance is the same as the running clearance, which is often specified as a clearance ratio of mils (milli-inches) per inch of journal diameter. Some typical values of clearance ratio are between 1. 5 to 2.0 mils/in. Obviously, there are some applications where these values do not apply. The manufacturer will specify the recommended clearances for the particular bearing application.[21]. Slenderness ratio is also referred to as UD ratio. This is the ratio of the bearing length to the shaft diameter. This ratio typically varies between 0. 2 and 1. 0. However, some plain journal bearings have slenderness ratios above 1. 0. The bearing length affects the stiffness and damping characteristics of the bearing. In the selection of a bearing length, one must consider the bearing unit loading. The unit load is the bearing load divided by the product of the bearing length and the shaft diameter; therefore, the units are psi. Typical values of unit loading are between 150 and 250 psi. [21].

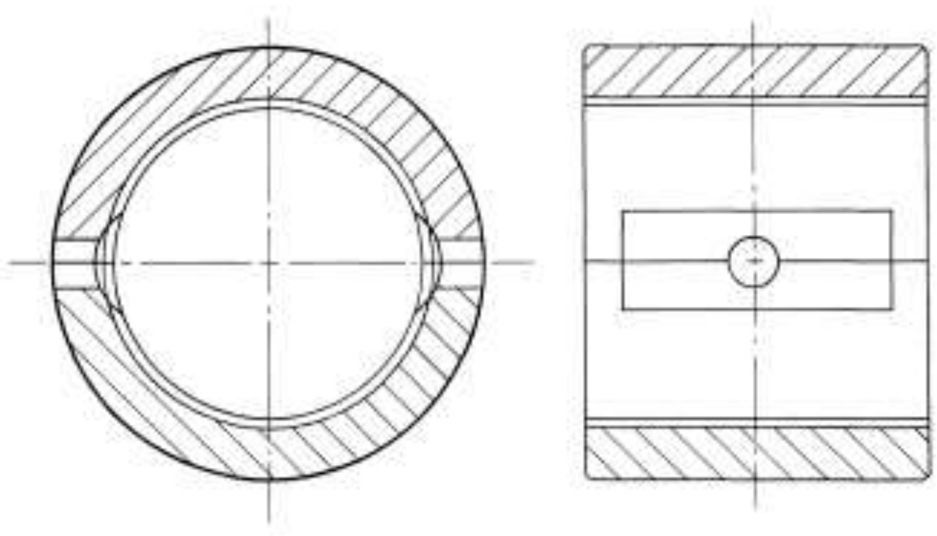


Figure 12. Plain Journal Bearing.

SOURCE: [21].

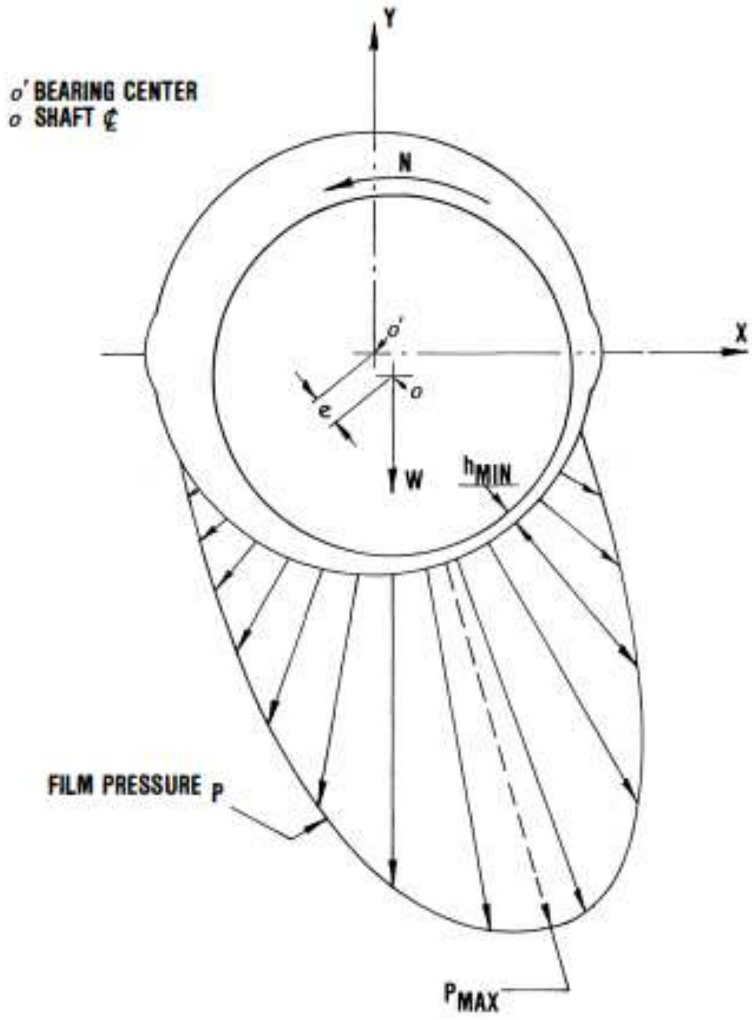


Figure 13. Hydrodynamic Bearing Preasure Profile

SOURCE [21].

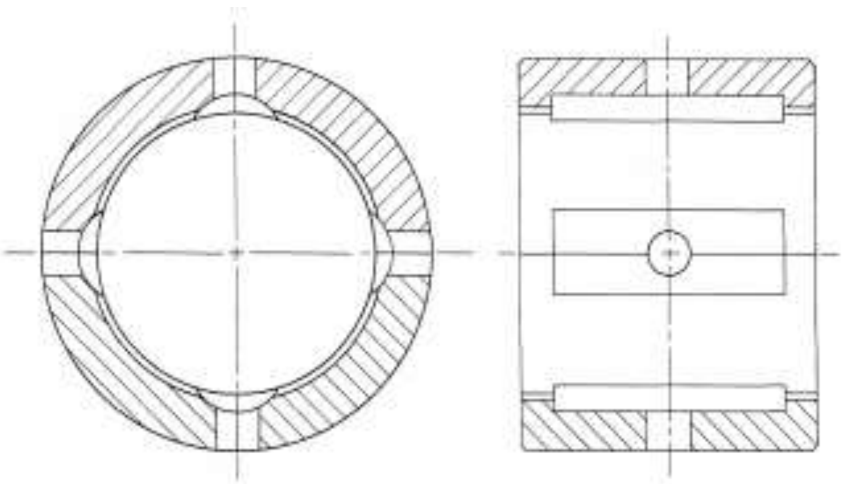


Figure 14. Four- Axial Groove Journal Bearing

SOURCE [21]

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.. 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. The types of machinery we are concerned with range from small high speed spindles to motors, blowers, compressors, fans, and pumps to large turbines and generators to some paper mill rolls and other large slow speed rotors [22].

WHEN TO USE FLUID FILM BEARINGS

There are applications where anti-friction bearings are the best choice. Commonly, smaller motors, pumps and blowers use rolling element bearings. Paper mill rolls often use large specialized spherical roller bearings. Clearly, anti-friction bearings are best for these applications. However, once the size of a pump (or fan or motor, etc.) gets large enough and fast enough, a gray area is entered. Here you will still find rolling element bearings used successfully but as speeds increase and temperatures rise, rotor dynamics often become a concern and critical speeds are encountered. This is when damping is required and fluid film bearings become increasingly necessary. My experience is that turbomachinery designers (and users) should consider using fluid film bearings if running above 3,000 RPM or the machine exceeds 500 HP. In my opinion, at 1,000 HP and up, all machines except very special cases should be on journal bearings specifically designed for that service. There are exceptions of course, and the decision where to apply what type of bearing is ultimately done for every machine individually based on good engineering practice and experience. Unfortunately, this decision is sometimes based on economics which keeps maintenance engineers and consultants employed [22].

ADVANTAGES OF FLUID FILM 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. In a pure sense, this is true, but other complications make this a secondary reason for using these bearings. During start up there is momentary metal-to-metal contact and foreign material in the lubricant or excessive vibration can limit the life of a fluid film bearing. For these reasons, special care must be taken when selecting and implementing a lubrication system and special vibration monitoring techniques must be applied. The most important aspects of the health and longevity of a fluid film bearing are proper selection, proper installation, proper lubrication, and the alternating hydrodynamic loads imposed on the bearing surface by relative shaft-to-bearing vibration [22].

Some of the primary advantages of fluid film bearings are:

a)Provide damping - Damping is required in order to pass through a critical speed. Damping is also required to suppress instabilities and sub-synchronous vibration.

b) Able to withstand shock loads and other abuse. c) Reduce noise. d) Reduce transmitted vibration. e) Provide electrical isolation of rotor to ground. f) Very long life under normal load conditions. g)Wide variety of bearing types for specific applications

The lubricant used provides these functions to all bearings: a) Remove heat generated in the bearing. b) Flush debris from load area. Some disadvantages to fluid film bearings are: a) Higher friction (HP loss) than rolling element type. b) Susceptible to particulate contamination. c) Cannot run for any length of time if starved for lubricant such as a lube system failure [22].

Lubricants can be any fluid, including gasses. In early reference books (1,2,3) some of the lubricants discussed are tallow, lard (animal fat), vegetable oils, and whale and fish oils. Obviously, sometimes you used what was available! Even water can be used under some conditions. Mineral oils from petroleum have evolved from straight distillates to complex formulations with special additives today. Synthetic lubricants have also been developed, primarily poylalphaolefins and esters. Silicones, glycols and other fluids are also used in special applications. There is no ideal or universal lubricant, all are compromises to fit any given situation. Applications range from heavy low speed loads to light high speed loads. At one extreme solid lubricants may be necessary and at the other, gas bearings may be required. Obviously, most applications fall in the middle where grease and oil lubricants are used. In this discussion of journal bearings we will limit ourselves to light oil lubrication found in the majority of turbo-machinery [22].

Bearing Nomenclature The shorthand that bearing analysts use with regards to journal bearings can be confusing and is certainly inconsistent from one analysis program to another. The terminology used in this paper is shown diagrammatically in figure 16. The symbolic notation and the definitions are as follows: Rj = Radius of Journal Rb = Radius of the Bearing Cb b j = Radial Clearance of the Bearing = R -R h = Radial clearance as a function of the angular position where the clearance is measured hmin = Minimum oil film clearance e = Eccentricity - the distance between the centre of the bearing and the centre of the shaft ecb = e/Cb = Eccentricity Ratio - if zero, shaft is centred; if 1 then shaft touches bearing Line of

Centres = Line connecting the centre of the bearing and the centre of the shaft ö = Attitude Angle = Angle from -Y axis to Line of Centres Ù or ù = Rotation Direction and Speed in RAD/SEC W = Gravity Load [22].

LUBRICATION

For fluid film bearings viscosity is the most important factor. Unfortunately, there are two forms of viscosity terminology and numerous units associated with these measures. Absolute or dynamic viscosity is the ratio of the shear stress to the resultant shear rate as a fluid flows. The more a fluid resists shear, the thicker it is and the higher the absolute viscosity. This is measured in Poise (or Centipoise) or Reyns. Some reference books use different terminologies, so read carefully. Kinematic viscosity is the absolute viscosity divided by the specific gravity. The most common unit of measure is the Centistoke, abbreviated cSt.

The friction factor (and thus the horsepower loss) is a function of viscosity, load, and speed. This was quantified by Richard Stribeck, a German engineer who did extensive friction testing in 1902. There was a great deal of research at that time trying to find the best combinations of materials and lubricants that would give the lowest coefficient of friction. Figure 21 is the original Stribeck curve. The friction factor is plotted as a function of ZN/P where Z is the viscosity, N is the speed and P is the load. This is a non-dimensional equation [22].

JOURNAL BEARING MATERIALS

Since the bearing is usually much less costly than the shaft, it is considered sacrificial if necessary. In addition, a low dry sliding friction coefficient between the shaft and bearing material is important for start-up and shutdown conditions. In the 17 century Robert Hooke advocated steel shafts with “bell metal”, essentially bronze, bushes to replace the practice of using wood blocks on cast iron. There has been extensive research on the best bearing material to use. In 1839 Isaac Babbitt patented several high tin and high lead alloys which are similar to modern formulations. Of course, his name is commonly used to describe many different bearing materials. Carbon, graphite, ceramics, plastics and composite materials are all used today in some applications. Lead alloys have been virtually eliminated in modern applications due to lower strength and environmental considerations. The most common bearing lining material used in turbo machinery today is high tin babbitt. The formulation and characteristics of the two most common babbitt alloys are given in table 5. There are other alloys with specialized properties and often proprietary composition.

TABLE 5 - BABBITT MATERIALS CHARACTERISTICS

PARAMETER	TYPE 2	TYPE 3
Tin	89%	84%
Antimony	7.5%	8%
Copper	3.5%	.8%
Liquidus Point	669^{o}F	792^{o}F
Brinell Hardness	24	27
Tensile Strength	12,600PSI @ 68^{o}F	10,000 PSI @ 68^{o}F
% Strength at 212^{o}F	52	52

SOURCE: [22]

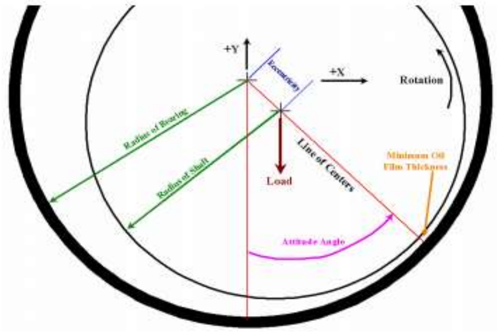
These formulations are applied to a base material, typically steel, by either molding in a stationary fixture or spin casting which is a messy and exciting thing to experience. Spin casting is supposed to create a better bond with the substrate but it has been determined that impurities in the molten babbitt can migrate to the babbitt-base metal bond line during this process and weaken that bond. While babbitt won't melt until the temperature is very high, it has lost almost half of its room temperature tensile strength at 212^{o}F.

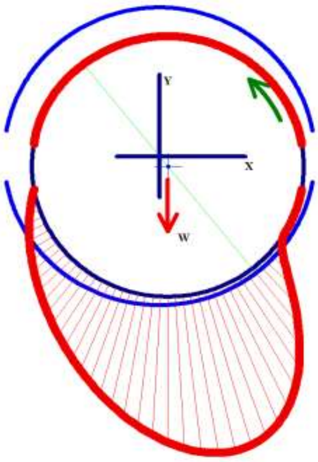
BEARING FAILURE MECHANISMS

There are many reasons a journal bearing might fail. A common mechanism is loss of lubricant and is not so much a bearing failure as a system failure. The next most common mechanisms is fatigue damage and one of the most important considerations for the material lining bearings is fatigue resistance. Thin babbit has significantly more fatigue resistance than babbitt thickness greater than 15 mils. [22]

Rotor imbalance and other conditions cause the journal to orbit in the bearing. This causes an oscillatory dynamic pressure to act on the babbitt surface. It is not uncommon for the peak static hydrodynamic pressure to approach 3 to 5 times the specific W/LD load. Oscillatory shaft motion adds an alternating pressure on top of this that impinges on the babbitt surface. Babbitt fatigues in a manner similar to the way potholes develop in a road surface Virtually all journal bearings cavitate. Typically oil contains dissolved gasses and in the divergent unloaded area of the oil film it is normal for streamers to form. As the pressure wedge re-forms, the gasses dissolve back into the oil. This is a gradual process and no damage to the bearing occurs. There are cases where rapid rupture of the oil film is normal such as with high speed reciprocating machinery. Here vapor bubbles can form and collapse explosively. These impacts may locally fatigue the bearing surface. Entrained free air will aggravate this problem. The most common failure mechanisms are a result of operating conditions outside of the intended design. These include foreign material (dirt) or contamination (water, etc.) in the oil, overload, lube oil viscosity degradation, or, far too often, lubrication failure.

Electrostatic and electromagnetic discharge across the oil film will erode the babbitt surface over time. Corrosion of the bearing babbitt is relatively rare especially if a regular oil analysis program is in effect. The steel backing can be attacked by water in the oil and copper alloys are more susceptible to corrosion from ammonia and other contaminates [22].





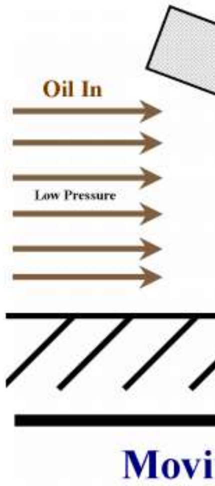


Figure 15. Basic Development of an Oil Wedge. SOURCE: [22] Figure 16. Bearing Nomenclature. SOURCE: [22] Figure 17. Pressure Profile in a Journal Bearing. SOURCE :[22].

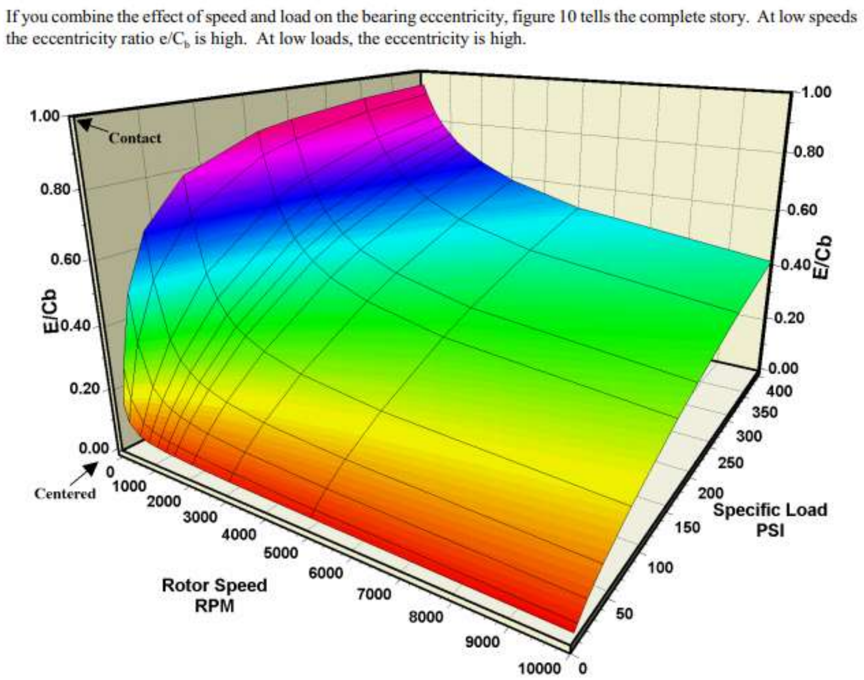


Figure 18. Effect of Load and Speed on Plain Journal Eccentricity. SOURCE: [22].

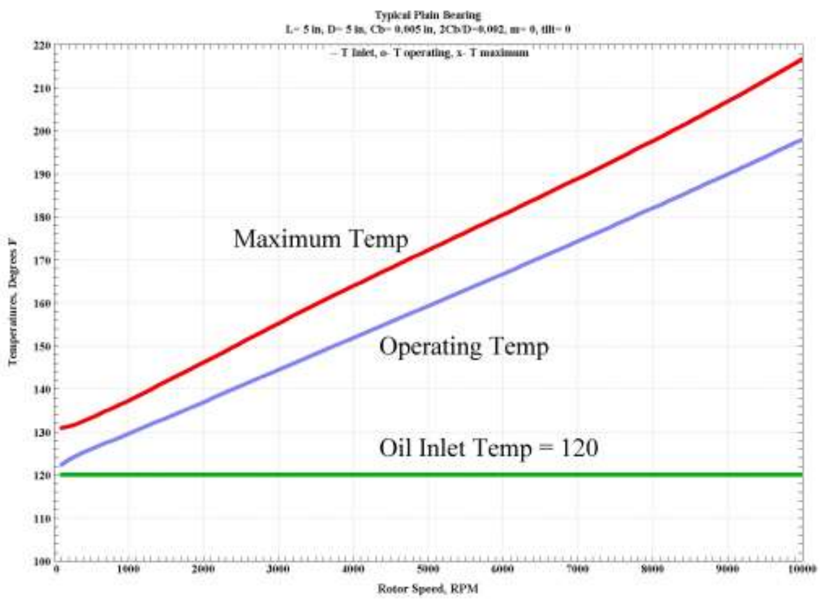


Figure 19. Temperatures in a Plain Bearing versus Speed at a Constant Load. SOURCE: [22].

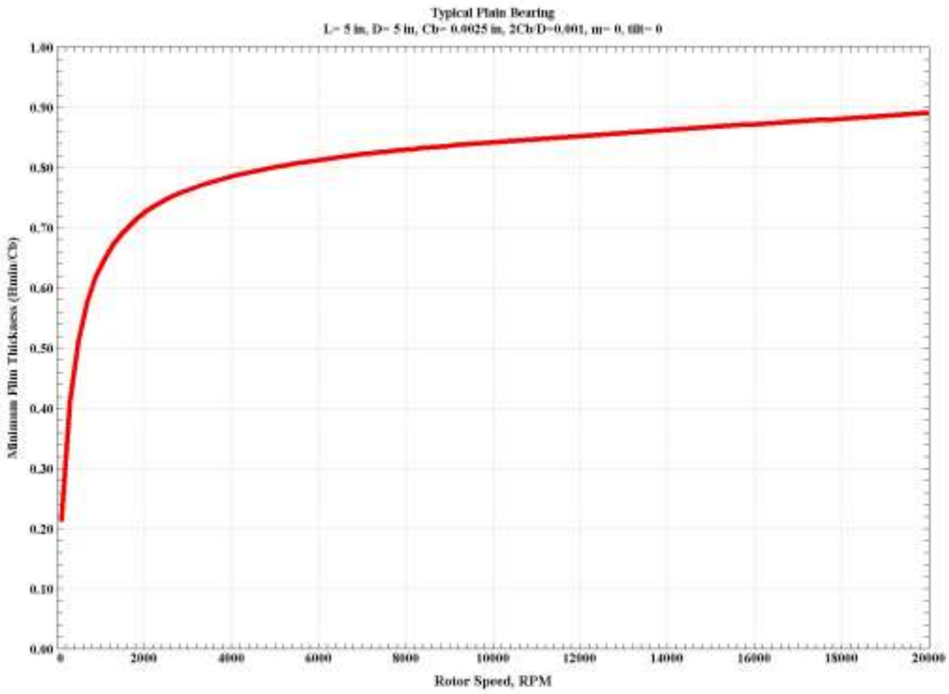


Figure 20. Minimum Oil Film Thickness in a Plain Bearing versus Speed at a Constant Load.

SOURCE: [22].

Table 4; Typical Viscosity Ranges for Various Applications.

Application	Viscosity Range, cSt
Clocks, Fine Instruments, High Speed Spindles	2 - 20
Turbomachinery - Turbines, Compressors, Etc.	4 - 30
Rolling Element Bearings	8 - 300
Low Speed, Heavy Load Journal Bearings	8 - 100
Reciprocating Engines and Pumps	10 - 300
High Speed Gears	50 - 150
Low Speed Gears	50 - 600
Worm Gears	200 - 1000

SOURCE: [22].

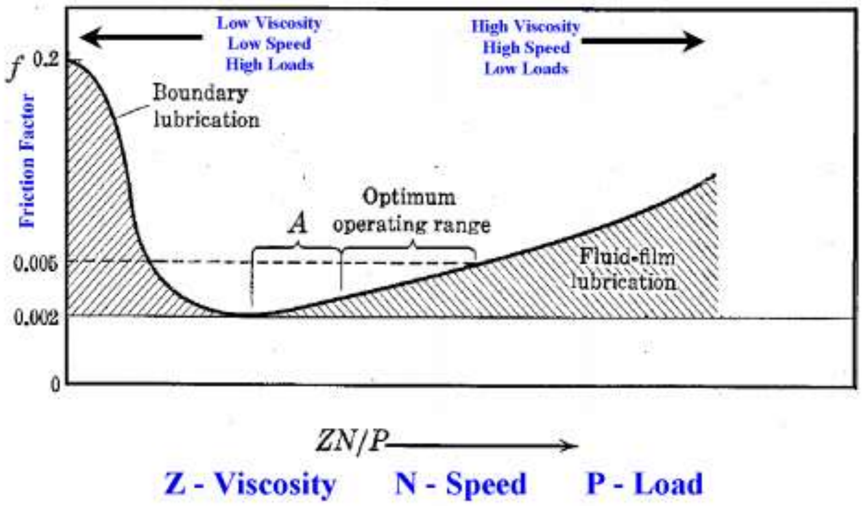
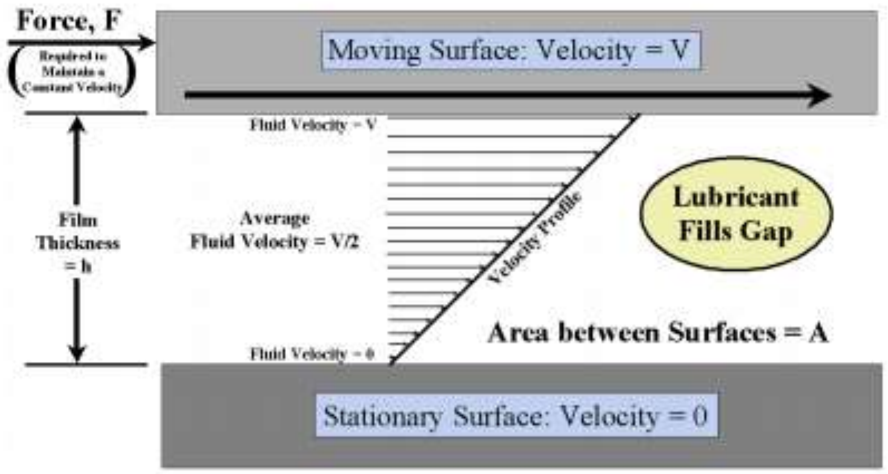


Figure 21. Stribeck Curve Relating Friction Factor to Viscosity, Speed and Load. SOURCE: [22].



$$\text{Shear Stress} = \frac{F}{A} = \frac{LB_f}{IN^2} \qquad \text{Shear Rate} = \frac{V}{h} = \frac{IN/SEC}{IN} = \frac{1}{SEC}$$

$$\text{Dynamic Viscosity} = \frac{\text{Shear Stress}}{\text{Shear Rate}} = \frac{F\ h}{V\ A} = \frac{\frac{LB_f}{IN^2}}{\frac{1}{SEC}} = \frac{LB_f - SEC}{IN^2} = 1\ \text{Reyn}$$

Or in SI units:

$$\text{Shear Stress} = \frac{F}{A} = \frac{Dynes}{CM^2} \qquad \text{Shear Rate} = \frac{V}{h} = \frac{CM/SEC}{CM} = \frac{1}{SEC}$$

$$\text{Dynamic Viscosity} = \frac{\text{Shear Stress}}{\text{Shear Rate}} = \frac{F\ h}{V\ A} = \frac{\frac{Dynes}{CM^2}}{\frac{1}{SEC}} = \frac{Dynes - SEC}{CM^2} = 1\ \underline{\text{Poise}}$$

$$\text{Kinematic Viscosity} = \underline{\text{Stoke}} = \frac{\text{Dynamic Viscosity}}{\text{Density}} = \frac{\frac{Dynes - SEC}{CM^2}}{\frac{Grams}{CM^3}} = \frac{Dynes - SEC - CM}{Gram} = \frac{CM^2}{SEC}$$

Figure 22. Defining Dynamic Viscosity - English and Metric Units. SOURCE: [22].

According to [23] most bearings are loaded radially. Sources of this loading are: 1) The weight of the machine parts themselves such as the pulleys, gears, or flywheels along with the weight of the shaft that supports them. 2) Tensions induced by the action of belt or chain drives. 3) The force between meshing gear teeth. 4) Centrifugal forces resulting from unbalanced masses. 5) Inertia forces accompanying the rapid acceleration and deceleration of machine members. The load magnitude will be affected by these factors: 1) Direction of the applied load on the shaft. 2) Point of application of the load on the shaft. 3) Distance between bearing centres.

SUMMARY OF LITERATURE REVIEW

Tribology is the science and engineering of interacting surfaces in relative motion. It includes the study and application of the principles of friction, lubrication and wear

Despite the relatively recent naming of the field of tribology, quantitative studies of friction can be traced as far back as 1493, when Leonardo da Vinci first noted the two fundamental 'laws' of friction

The two fundamental 'laws' of friction were first published (in 1699) by Guillaume Amontons, with whose name they are now usually associated, they state that:

1. the force of friction acting between two sliding surfaces is proportional to the load pressing the surfaces together
2. the force of friction is independent of the apparent area of contact between the two surfaces.

To reduce friction between surfaces and keep wear under control, materials called lubricants are used..

depending on the type of application, the costs to address and the level of "perfection" of the lubrication desired to be achieved, there is a choice between:

1. Fluidostatic lubrication (or hydrostatic in the case of mineral oils) – which consists in the insertion of lubricating material under pressure between the surfaces in contact;
2. Fluid fluid lubrication (or hydrodynamics) – which consists in exploiting the relative motion between the surfaces to make the lubricating material penetrate.

To determine the viscosity of a fluid, viscometers are used which can be divided into 3 main categories:

1. Capillary viscometers, in which the viscosity of the fluid is measured by sliding it into a capillary tube;
2. Solid drop viscometers, in which viscosity is measured by calculating the velocity of a solid that moves in the fluid;
3. Rotational viscometers, in which viscosity is obtained by evaluating the flow of fluid placed between two surfaces in relative motion.

The first two types of viscometers are mainly used for Newtonian fluids, while the third is very versatile.

[3] By controlling the pressure of the oil injection, it is possible to get large variations in the active hydrodynamic forces; such effects could be useful for reducing vibrations in rotating machines.

[4] Six leading mode of bearing failure (abrasion, corrosion, electrical pitting, fatigue, overheating and *wiping are covered in detail). Turbine generator bearing failures are a leading cause of power plant* unavailability and can cause serious damage not only to bearing systems but also to rotors, stators and nearby equipment.

Specifications for new lubricating oil are formulated by equipment manufacturers, oil suppliers, and certain technical societies active in the field of turbine lubrication. A minimum specification guide has been issued by the American Society for Testing and Materials (ASTM) and titled "Standard Specification for Mineral Lubricating Oil Used in Steam and Gas Turbines (D 4304)."

Analyses In-service monitoring of the condition of a turbine oil should focus on the following properties: Antirust protection Remaining oil life (oxidation stability) Viscosity, Total acid number,

0 Cleanliness Foaming tendency Color/appearance, Water content, Flash point

A property called turbine severity (B) level has been established which can be used to take these conditions into consideration when monitoring the remaining oxidation resistance of the oil during its service life (DenHerder and Vienna). "B" is defined as the percentage of fresh oil oxidation resistance lost per year due to oil reactions in the turbine generator lubrication system. "B" takes into consideration the following three factors: • Amount of make up oil added to the system to replenish the oil oxidation resistance • Time that the oil has been in use • Oxidation resistance that remains as determined by a RBOT, ASTM D 2274-84 Equation (2) determines turbine severity, B: $B = M(1\text{-}X/100)/(1\text{-}e^{-Mt/1oo})$ (2) where M = Amount of oil added as makeup into the system per year, expressed as a percentage of the total amount of oil originally placed in the system (percentage per year) X = Amount of oxidation resistance that remains in the oil, expressed as a percentage of the original oxidation resistance of the oil (percentage of fresh oil) = Amount of time the original oil has been in service in years The effect of makeup rate, M, on oil degradation for a turbine with a severity level of 25 percent per year is shown in Figure 36. The severity level for a particular lubrication system should be determined over a period of time beginning with initial operation or installation of a fresh oil charge

[5] Journal bearings are able to function efficiently and avoid wear because of a hydrodynamically generated film of lubricant that separates the rotating surfaces. The thickness of this lubricant films depends on the following variables:

Radial clearance between the leg journal and cone

Radius of the leg journal

Lubricant viscosity

Rotational velocity of the cone

Applied load

the optimum design procedure of high-speed, short journal bearings under laminar and turbulent flow conditions is developed based on three kinds of methods such as Successive Quadratic Programming, Genetic Algorithm and Direct Search.

[6] . Oil flow rate to tilting pad thrust bearings influences two important bearing operating characteristics; bearing power loss and pad operating temperatures. Reducing oil flow rates is desirable as this reduces the size and cost of the oil supply system and also can reduce bearing power losses. However, this can increase bearing operating temperatures, reducing the load capacity.

[15] Typical turbine lubricants available and used today are based either on classic high quality petroleum base stocks or some form of synthetic base stock such as ester, poly-alkylene-glycol (PAG), or poly-alpha-olefin (PAO). Additionally, any number of additives are blended into the base oil to improve specific characteristics such as anti-oxidation additives which extend lubricant. life, extreme pressure additives to protect surfaces in high pressure contacts or anti-foaming additives to reduce the lubricant’s tendency to trap air bubbles. Most important to this work are additives which improve the viscosity index (VI) of the lubricant

16. A prototype of the next-generation, high-performance micro-turbine system was developed for laboratory evaluation. The rotor system using the water-lubricated bearings achieved stable rotating conditions at a rated rotational speed of 51,000 rpm. An electrical output of 135 kW with an efficiency of more than 33% was obtained.

18. Major individual damage types are discussed in the context of major damage categories.. Hydrodynamic bearings support a rotating shaft with its associated loads, by means of a pressure field developed within a lubricant which separates the solid surfaces. Generation of the separating pressure requires the presence of a converging film thickness and lubricant viscosity. With sufficient pressure over a bearing area, complete support of the loads is achieved avoiding contact between the rotating and stationary components. In theory, this hydrodynamic film prevents wear and degradation of the rotating and stationary components. For industrial equipment, the rotating shaft is more valuable than the supporting bearings. Therefore, the bearing inner diameter is lined with a sacrificial material to absorb damage and to protect the journal surface (for radial bearings) or thrust collar (for thrust bearings) from damage.

21. The difference between the bearing set bore radius and journal radius is the bearing set clearance. The set clearance is the same as the running clearance, which is often specified as a clearance ratio of mils (milli-inches) per inch of journal diameter. Some typical values of clearance ratio are between 1. 5 to 2.0 mils/in.

Lead alloys have been virtually eliminated in modern applications due to lower strength and environmental considerations. The most common bearing lining material used in turbo machinery today is high tin babbitt.

There are many reasons a journal bearing might fail. A common mechanism is loss of lubricant and is not so much a bearing failure as a system failure. The next most common mechanisms is fatigue damage and one of the most important considerations for the material lining bearings is fatigue resistance. Thin babbit has significantly more fatigue resistance than babbitt thickness greater than 15 mils. (Malcolm, 2019).

DISCUSSIONS

By the definition of Tribology as the science and engineering of interacting surfaces in relative motion; and that it includes the study and application of the principle of friction, lubrication and wear; the three key words frication, lubrication and wear are at the heart of the dynamics of plant/equipment in motion. To reduce friction between surfaces and keep wear under control, materials called lubricants are used.

Now six leading modes of bearing failure as identified in literature review are: abrasion, corrosion, electrical pitting, fatigue, overheating and wiping. Also turbine generator bearing failures are a leading cause of power plant unavailability and can cause serious damage not only to bearing system but also to rotors, stators and nearby equipment. Based on the central role and importance of power output to the industrial sector and society, no action taken to prevent power plant outage is seen as over-emphasized.

A minimum specification guide has been issued by the American Society for Testing and Materials (ASTM) and titled "Standard Specification for mineral lubricating oil used in Steam and Gas Turbine (D4304)".

A property called turbine severity (B) level has been established which can be used to monitor the remaining oxidation resistance of the oil during its service life. Journal bearings are able to function efficiently and avoid wear because of a hydrodynamically generated film of lubricant that separates the rotating surfaces. The thickness of this lubricant film depends on the following variables: i)

Radial clearance between the leg journal and cone.
ii) Radius of the leg journal.
iii) Lubricant viscosity.
iv) Rotational velocity of cone.
v) Applied load.

Typical turbine lubricants available and used today based on classic high quality petroleum base stocks or some form of synthetic base such as ester, poly-Alkene-glycol (PAG) or poly-alpha-olefin (PAO).
The set clearance is the same as the running clearance, which is often specified as a clearance ratio of mils (milli - inches) per inch of journal diameter. The most common bearing lining used in turbo--machinery today is high tin babbitt. Thin babbitt has significantly more fatigue resistance than babbitt thickness greater than 15 mils. The optimal superintending over and control of these parameters guarantees the efficient and effective performances of journal bearings of Turbine Generators, and hence optimal performance of Turbo Generators in power delivery.

CONCLUSION.
By fervently implementing and controlling the five factors above, Journal Bearing Oil Clearance and lubrication management system would have been achieved. Hence an effective and efficient turbo-generator.

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