

## Lubrication and Lubricant Selection

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### ABSTRACT

*The actual contact of two surfaces is only at the peak of their asperities and great force is needed to separate them. The force which resists relative movement between two surfaces in contact is known as friction, a consequence of which heat is generated, energy consumed and wear. Any substance which is used to reduce the friction is known as a lubricant. Boundary and elastohydrodynamic lubrication are the two main regimes of lubrication which can be determined from Stribeck Curve. Type of motion, speed, temperature, load and operating environment are parameters of the tribological system that needs to be considered and analysed so that the best lubricant for a specific application can be selected. Usually the specifications of a lubricant are determined by the respective equipment manufacturers. Used oil analysis is crucial for preventive maintenance, equipment failure diagnosis, abnormal operating condition assessment, oil condition monitoring and checking for contamination. The tribology analysis shows that low viscosity lubricants are suitable for high speeds, high temperature and low pressures whereas high viscosity lubricants are good for low speeds, low temperature and high pressures.*

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### INTRODUCTION

Lubricant and lubrication are common issues in all palm oil mills, thus, it is useful to understand the theory of lubrication and proper selection of lubricant. The main function of lubricant is to provide hydrostatic, hydrodynamic or elasto-hydrodynamic lubrication by forming a physical barrier to keep moving parts apart so that friction, surface fatigue, formation of wear particles, heat generation, operating noise and vibrations could be reduced. Besides lubrication, liquid lubricant may also provide hydrostatic power transmission based on the Pascal's law or serves as a coolant (Wikipedia, 2009a).

A liquid lubricant usually with high specific heat capacity typically contain 90% base oil and 10% additives to impart desirable characteristics. Petroleum mineral oils, lanolin, vegetable oils and synthetic liquids such as hydrogenated polyolefins, esters, silicone, fluorocarbons and many others are used as base oils. Table 1 shows several types of lubricant base oils designated by the American Petroleum Institute (API). Additives such as friction modifiers that chemically bind to metal surfaces reduce surface friction even when there is insufficient bulk lubricant present for hydrodynamic lubrication. Detergent and dispersant additives could assist in debris and contaminant transport to the filter and their removal.



TABLE 1. AMERICAN PETROLEUM INSTITUTE LUBRICANT BASE OIL

Group	Specification
I	Less than 90% saturates and/or more than 0.03% sulphur, and SAE viscosity index of 80 to 120 such as 150 SN, 500 SN, and 150 BS.
II	More than 90% saturates and less than 0.03% sulphur, and SAE viscosity indexing of 80 to 120 which has superior antioxidation properties.
III	More than 90% saturates and less than 0.03% sulphur, and SAE viscosity index more than 120.
IV	Polyalphaolefins (PAO).
V	All others not included above such as naphthenics, PAG and esters.

Note: SAE - Society of Automotive Engineers.

Anti-wear additives improve the performance against wear and fatigue. *Table 2* shows the common additives and their functions. Sulphur impurities in fuels, biodiesel and 2-T oil added to fuels provide some lubrication properties to the fuel.

A good lubricant should cater for specific operational environment needs of a machine such as dry, wet, cold, hot, fire risk, high load, high or low speed, chemical compatibility, atmospheric compatibility, pressure or vacuum and various combinations. The usual thermal characteristics are indicated as SAE number at 100°C, like SAE 30 and SAE 40. On low temperature scale, it is given as SAE xxW. Both markings can be combined together to form for example SAE 0W-60. Viscosity index (VI) marks viscosity changes with temperatures, with the higher VI numbers being more temperature stable. Total base number does not measure the accumulation of oxidation products or antioxidants, rather, it measures the depletion of a detergents present in an engine oil for the purpose of neutralizing acidic blow by gases which occurs due to low levels of antioxidants within the oil. As the detergent is consumed during its role of neutralizing sludge and varnish, the base number decreases from its original new oil value. Monitoring this consumption allows one to

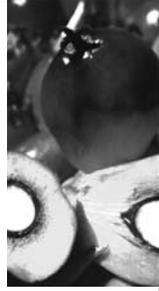
pro-actively replenish the oil through before the protection afforded by that additive is lost. Thus, high number in fact indicates a high detergent content that will keep the engine clean but the base oil can oxidize or breakdown faster than one with higher levels of antioxidants which will prevents acids by neutralizing the acids as opposed to cleaning up the by-products of the oxidized oil. Lubricants degrade primarily due to the presence of free radicals that will attack against the base oil chemical composition and change their physical properties such as viscosity (Wikipedia, 2009b).

Solid lubricants include grease, organic polymers such as teflon (PTFE), non-metal components such as graphite, hexagonal boron nitride, molybdenum disulphide and tungsten disulphide, and metal-alloy such as cadmium and gold used for electro-plating surfaces, lead, tin, zinc alloys and various bronze alloys used in bearings. Some of these solid lubricants are used as additives in grease. The generation of a compacted oxide glaze layer from metallic surfaces sliding against each other at high temperature is a phenomenon in relation to wear prevention and lubrication. The elimination of metallic contact and adhesion by the generated glaze layer will reduce friction and wear thus become self-lubricating.

TABLE 2. COMMON ADDITIVES AND FUNCTIONS

Additive	Typical compounds	Functions
Antifoamant	Silicone polymers, organic copolymers.	Reduces surface tension to speed collapse of foam.
Antioxidant	Zinc dithiophosphates, hindered phenols, aromatic amines, sulphurized phenols.	Decompose peroxides and terminate free-radical reactions.
Anti-wear and EP agent	Zinc dithiophosphates, organic phosphates, acid phosphates, organic sulphur and chlorine compounds, sulphurized fats, sulphides and disulphides.	Chemical reaction with metal surface to form a film with lower shear strength than the metal, thereby preventing metal-to-metal contact.
Corrosion and rust inhibitor	Zinc dithiophosphates, metal phenolates, basic metal sulphonates, fatty acids and amines.	Preferential adsorption of polar constituent on metal surface to provide protective film, or neutralize corrosive acids.
Detergent	Metallo-organic compounds of sodium, calcium and magnesium phenolates, phosphonates and sulphonates.	Chemical reaction with sludge and varnish precursors to neutralize them and keep them soluble.
Dispersant	Alkylsuccinimides, alkylsuccinic esters and mannich reaction products.	Contaminants are bonded by polar attraction to dispersant molecules, prevented from agglomerating and kept in suspension due to solubility of dispersant.
Friction modifier	Organic fatty acids and amides, lard oil, high molecular weight organic phosphorus and phosphoric acid esters.	Preferential adsorption of surface-active materials.
Metal deactivator	Organic complexes containing nitrogen or sulphur, amines, sulphides and phosphites.	Form inactive film on metal surfaces by complexing with metallic ions.
Pour point depressant	Alkylated naphthalene and phenolic polymers, polymethacrylates, maleate /fumerate copolymer esters.	Modify wax crystal formation to reduce interlocking.
Seal swell agent	Organic phosphates and aromatic hydrocarbons.	Chemical reaction with elastomer to cause slight swell.
Viscosity modifier	Polymers and copolymers of olefins, methacrylates, dienes or alkylated styrenes.	Polymers expand with increasing temperature to counteract oil thinning.

Source: <http://www.bobistheoilguy.com>



Lubrication is the process or technique employed to reduce wear of one or both surfaces in close proximity, and moving relative to each other, by interposing a substance called lubricant between the surfaces to carry or to help carry the load between the opposing surfaces. It can also be described as the phenomenon that result in the reduction of wear without human intervention. In the most common case, the applied load is carried by the pressure generated within the fluid due to the frictional viscous resistance to motion of the lubricating fluid between the surfaces. As the load increases on the contacting surfaces, three distinct situations can be observed with respect to the mode of lubrication, which are called regimes of lubrication.

- Fluid film lubrication is the lubrication regime in which the load is fully supported by the lubricant through viscous forces within the space or gap between the parts in motion relative to one another (the lubricated conjunction) and solid-solid contact is avoided.
- Hydrostatic lubrication is when an external pressure is applied to the lubricant in the bearing to maintain the fluid lubricant film where it would otherwise be squeezed out.
- Hydrodynamic lubrication is where the motion of the contacting surfaces, and the exact design of the bearing is used to pump lubricant around the bearing to maintain the lubricating film. This design of bearing may wear when started or stopped, as the lubricant film breaks down.
- Elastohydrodynamic lubrication. The opposing surfaces are separated but there occurs some interaction between the raised solid features called asperities, and there is an elastic deformation on the contacting surface enlarging the load bearing area whereby the viscous resistance of the lubricant becomes capable of supporting the load.

- Boundary lubrication. The bodies come into closer contact at their asperities; the heat developed by the local pressures causes a condition which is called stick-slip and some asperities break off. At the elevated temperature and pressure conditions, chemically reactive constituents of the lubricant react with the contact surface forming a highly resistant tenacious layer, or film on the moving solid surfaces (boundary film) which is capable of supporting the load and major wear or breakdown is avoided. Boundary lubrication is also defined as that regime in which the load is carried by the surface asperities rather than by the lubricant.

Lubrication theory can be seen as exploiting the disparity between two length scales. The first is the characteristic film thickness,  $H$ , and the second is a characteristic substrate length scale  $L$ . The key requirement for lubrication theory is to minimize the ratio  $\varepsilon = \frac{H}{L}$ . The Navier-Stokes equations are expanded in this small parameter, and the leading-order equations are then:

$$\begin{aligned} \partial p / \partial z &= 0 \\ \partial p / \partial x &= \partial^2 u / \partial z^2 \end{aligned} \quad (1)$$

where  $x$  and  $z$  are coordinates in the direction of the substrate and perpendicular to it respectively. Here,  $p$  is the fluid pressure, and  $u$  is the fluid velocity component parallel to the substrate (Wikipedia, 2009c).

## TRIBOLOGY

Tribology is the science and technology of interacting surfaces in relative motion includes the study and application of the principles of friction, lubrication and wear. When one material slides or rubs over another, it is affected by complex tribological interactions. Leonardo Da Vinci (1452-1519) was the first to enunciate two laws of friction where:

- the frictional resistance is the same for two different objects of the same weight

## NOMENCLATURE

		SI unit
$E$	Modulus of elasticity	$\text{N m}^{-2}$
$H$	Film thickness	m
$L$	Characteristic substrate length scale	m
$N$	Speed	$\text{M s}^{-1}$
$P$	Load projected on to the geometrical surface	N
$R$	Cylindrical radius	m
$d_i$	Inner diameter	m
$d_o$	Outer diameter	m
$h$	Lubricant film thickness	m
$h_u$	Uniform lubricant film thickness	m
$h_0$	The minimum lubricant film thickness	m
$k$	Specific viscosity	-
$m$	Total number of asperities	-
$p$	Fluid pressure	$\text{N m}^{-2}$
$q$	Reduced pressure	$\text{N m}^{-2}$
$r_1$	Roller radius	m
$r_2$	Inner ring raceway radius	m
$u$	Fluid velocity	$\text{m s}^{-1}$
$v$	Mean velocity	$\text{m s}^{-1}$
$x$	Direction coordinates	-
$z$	Direction coordinates	-
$\alpha$	Pressure-viscosity coefficient	$\text{m}^2 \text{N}^{-1}$
$\beta$	The deformation of the cylindrical surface	m
$\delta$	The normal relative displacement of distant points in the two surfaces	m
$\emptyset$	Poisson's ratio	-
$\varepsilon$	Thickness to length scale ratio	-
$\sigma$	Composite roughness of the two contact surfaces	m
$\lambda$	Specific film thickness	-
$\eta$	Dynamic viscosity	$\text{N m}^2 \text{s}^{-1}$
$\eta_0$	The pre-exponential viscosity coefficient	$\text{N m}^2 \text{s}^{-1}$

but making contacts over different widths and lengths; and

- the force needed to overcome friction is doubled when the weight is doubled.

Based on the results of Professor Richard Stribeck (1861 – 1950), the friction regimes for sliding lubricated surfaces are related to a dimensionless lubrication parameter  $\eta \frac{N}{P}$  known as Stribeck number where  $\eta$  is the dynamic viscosity,  $N$  is the speed of movement and  $P$  is the load projected on to the geometrical surface could be categorized into (i) solid/boundary friction, (ii) mixed friction, and (iii) fluid friction as shown in the Stribeck curve (Figure 1) (Wikipedia, 2009d).

There are five parameters of the tribological system to be considered and analysed so that the best lubricant for the specific application could be selected. However, the information obtained by defining the tribological system parameters also provides data for further in-depth technical analysis.

The first parameter of the tribological system involves the type of motion which may be sliding requires the hydrodynamic lubrication theory for its analysis, or rolling where elastohydrodynamic lubrication theory would be applied.

Speed is the second parameter on the tribological system which can be broken into the general ranges of fast, moderate and slow determined by the speed factor.

$$\text{Speed factor} = \eta \left[ \frac{(d_i + d_o)}{2} \right] \quad (2)$$

where  $d_i$  is the inner diameter and  $d_o$  is the outer diameter of a bearing.

The third tribological parameter is temperature. All lubricants have specific temperature ranges for optimal performance. There are some greases with synthetic hydrocarbon-based oil and barium-complex thickener that can operate at temperatures as low as  $-60^\circ\text{C}$  and greases with perfluorinated aliphatic ether base oil thickened with polytetrafluoroethylene (PTFE) that can lu-

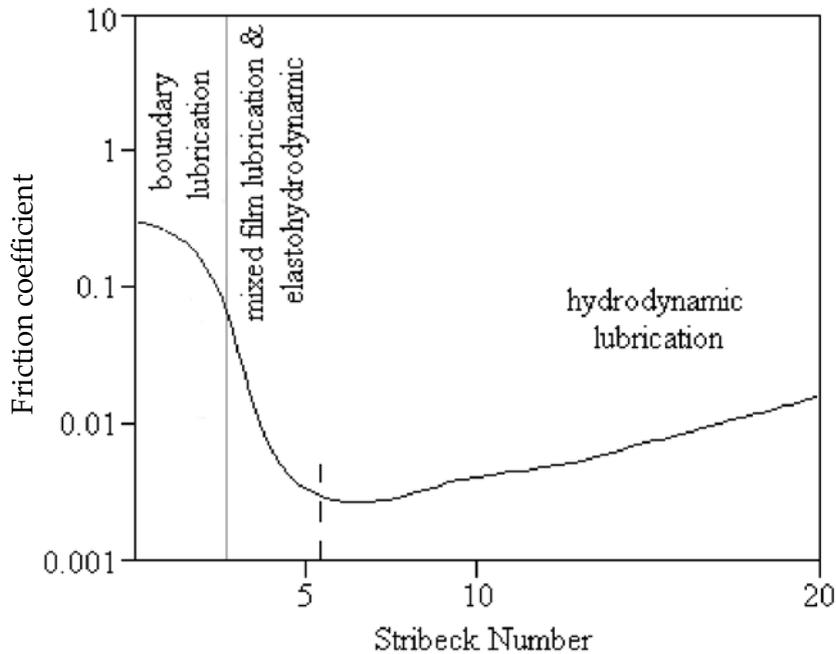


Figure 1. Stribeck curve.

bricate an oven chain bearing at 220°C for more than 15 000 hr. Knowing the temperature of the tribological system enables the engineer to select a lubricant that will provide optimum operating life and performance at the application temperature.

Load is the fourth parameter on the tribological system which is an important factor affecting the lubricant requirement.

The last parameter of the tribological system is the application's operating environment (Wikipedia, 2009d). Food safety issues need to be considered for lubrication points in food industries which could be critical control point. Federal Food and Drug Administration (FDA) defines ingredient and its concentration of lubricants that can contact with food in Group 21 CFR 178.3570 and US Department of Agriculture (USDA) later replaced by National Sanitation Foundation (NSF) since September 1998 classify approved food grade lubricants into two classes which are H1 – incidental food contact lubricant and H2 – no food contact possible (Shell, 2000).

### BOUNDARY LUBRICATED CONTACT MODELS

Lubricating layers as small as a single molecule are capable of producing significant improvements in reduced friction and wear which is the subject of boundary lubrication. Boundary friction and wear consists of a shear or adhesion component and a plowing or deformation component. The shear component predominates except when asperities sink too deeply into a boundary lubricant film or a soft opposing surface. When movement or sliding occurs, the shear friction force depends on the shear resistance per unit area of any boundary film in the real load-supporting area between asperities (George and Fennell, 2007). The total load support of a rough surfaces contact could be defined as:

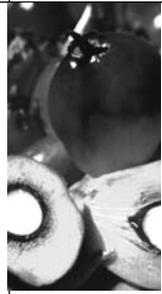
$$P = \sum_{i=1}^m p_i + \int_{area} p \, dA \quad (3)$$

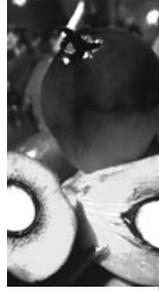
where  $m$  is the total number of asperities and  $A$  is the area subject to hydrodynamic fluid pressures.

Boundary lubrication is the formation and maintenance of a single or multi-molecular layer of lubricating material so as to prevent as much as possible the direct dry contact of the solid surfaces in the tribological couple. Many chemicals form films when certain organic compounds react with the metal surfaces. Intervening films such as oxide material in the film intimate contact with the metal surface underneath and sulphide layers are good examples of chemical film effective in reducing friction and wear. Combinations of pure paraffin oil and small amounts of fatty acid such as lauric acid can reduce friction effectively. The resulting metallic soap molecules formed at the surface perform well until the temperature becomes so high that soap melts and film breakdown. As a general rule, polar molecules such as straight chain organic molecules with one polar end, exhibit strong affinity for bare metal surfaces by physical adsorption, chemical adsorption or by chemical reaction, are ideal candidates as boundary lubricants. Longer chain molecules or the presence of more than one molecular layer of lubricating material leads to improve tribological performance. Surface coatings of materials with layered crystallographic structures, specifically graphite and molybdenum disulphide can also reduce friction and wear.

### ELASTOHYDRODYNAMIC LUBRICATION MODELS

Elastohydrodynamic lubrication theory is used to identify thickness of lubricant film in a rolling contact that both surfaces are assumed perfectly smooth. The analysis starts with the appropriate form of Reynolds equation to incorporate the pressure sensitivity of viscosity followed by the Hertzian contact concept which gives the stress and strain fields as well as the deformation of unlubricated surfaces in contact under load and combined with the viscous effect. A





contact area with an approximately uniform film thickness,  $h_u$ , is created while outside the uniform thickness zone the film thickness,  $h$ , varies approximately parabolic with distance in a pressurized liquid film. Consider now the analysis of pressure profiles in lubricating films incorporating the effect of elastic deformation of contacting solids. A common feature of full film hydrodynamic lubrication system is the presence of converging or diverging gaps around a minimum clearance point. Since Hertzian contacts contain converging or diverging wedges, lubrication behaviour can be expected to occur under dynamic conditions. Under the combined influence of solid elasticity and fluid viscosity pressures in the liquid, film may rapidly rise up to Hertzian solid contact levels. As a result, the film thickness becomes approximately constant and equal to  $h_u$  within the contact area. However, in order to maintain continuity and to compensate for the loss of viscosity towards the contact exit, a constriction of the gap down to size  $h_0$  is formed near the downstream exit. As a consequence, a pressure spike is formed and this is followed by a subsequent decay to values below the Hertz solution. The pressure on the upstream side lies also below the Hertzian value while it extends a greater distance towards the upstream direction. For line contact to two cylinders, the gradient in reduced pressure is given by:

$$\frac{dq}{dx} = 6v\eta_0 \frac{h - h_u}{h^3}; q \approx \frac{1}{\alpha} \int_{h_u}^{h_1} 6v\eta_0 \frac{h - h_u}{h^3} dx \quad (4)$$

where  $q$  is reduced pressure and  $\eta_0$  is the pre-exponential viscosity coefficient. Grubin solved the above equation numerically by assumed the surfaces have the deformed shape of an unlubricated contact but separated by a gap  $h$  given by:

$$h = \frac{x^2}{2R} + \beta - \delta \quad (5)$$

where  $R$  is cylindrical radius,  $\beta$  is the deformation of the cylindrical surface and  $\delta$  is the normal relative displacement of distant points in the two surfaces. The minimum film thickness,  $h_0$  in a rolling contact situa-

tion which forms the basis for elastohydrodynamic lubrication theory could be determined as (Lauer, 2009):

$$h_0 = \frac{0.007\eta v \alpha^{0.6}}{\frac{1}{r_1} + \left(\frac{1}{r_2}\right)^{0.43} \left(\frac{P}{L}\right)^{0.13}} \cdot \left[ \frac{E}{1 - \nu^2} \right]^{0.03} \quad (6)$$

where  $\alpha$  is pressure-viscosity coefficient,  $v$  is the mean velocity,  $r_1$  is roller radius,  $r_2$  is inner ring raceway radius,  $E$  is modulus of elasticity and  $\nu$  Poisson's ratio. It can be determined that if the pressure-viscosity coefficient is doubled, the film thickness increase by 51%. If the dynamic viscosity which can be directly related to the kinematic viscosity is doubled, the film thickness will increase by 62%. If the velocity of the roller bearing is doubled, the film thickness of the lubricant will be increased by 62% as well. Since surfaces are not perfectly smooth in reality, the elastohydrodynamic film thickness is not used directly even it is an important criteria for the lubricant selection. The specific film thickness,  $\lambda$ , is widely used which is defined as the ratio between the average or mean elastohydrodynamic film thickness and the composite surface roughness of the rolling contacts,  $\sigma$ , as shown in the equation below.

$$\lambda = \frac{h_0}{\sigma}; \sigma = \sqrt{\sigma_1^2 + \sigma_2^2} \quad (7)$$

If the specific film thickness is close to zero, there will be a dramatic increase in the metal-to-metal contact at the friction point and produces unacceptable wear. The transition from boundary lubrication into the mixed lubrication occurs at  $\lambda=1$  where the bearing will have only partially separated the metal asperities with some metal-to-metal contact. As the specific film thickness increases, metal-to-metal contact will decrease, thus, reduce in wear. When variable speeds and/or shock loading is present, specific film thickness greater than 4 are desirable, however, depending upon the relative bearing speed and oil viscosity, energy consumption and heat generation due to the internal fluid friction is unavoidable.

Appropriate lubricant is selected based on the specific viscosity,  $k$ , defined as the ratio of the actual viscosity of the selected lubricant to the minimum viscosity required to obtain separation of the moving surfaces which is determined based on the mean bearing diameter and the operating speed at the given operating temperature as shown in Figure 2. At  $k$  values below 1, it is generally accepted that EP additives will be required to mitigate the effects of boundary lubrication conditions. As  $k$  approaches 1, the bearing life approaches to DIN ISO 281 rate. At  $k$  values above 1, bearings life may exceed the rated value. If  $k$  values above 4, undesirable effects may occur such as increase fluid friction, viscous drag and ball skidding. For most bearing applications,  $k$  values range from 1 to 2.5 are optimal (Lauer, 2009).

### DISCUSSION AND CONCLUSION

The selection of lubricant is usually specified by equipment manufacturer. Used oil analysis is crucial for preventative maintenance,

equipment failure diagnosis, abnormal operating condition assessment, oil condition monitoring and checking for contamination. Different tests are carried out for different applications. Warning limits for various oil parameters have been defined as shown in Table 3 (Shell, 2000).

Viscosity is the most important property which affects friction and load carrying capacity. Viscosity change indicates the oil deterioration and contamination. Flash point indicates oil structure breakdown, safety hazard and contamination by fuel.

Total base number (TBN) applicable to diesel engine oils only measures the ability of the oil to neutralize strong acids.

Total acid number (TAN) measures the potential for corrosive wear caused by contamination with acidic products and organic acids from oil deterioration which indicates the rapid oil deterioration possibility and formation of sludge and lacquer.

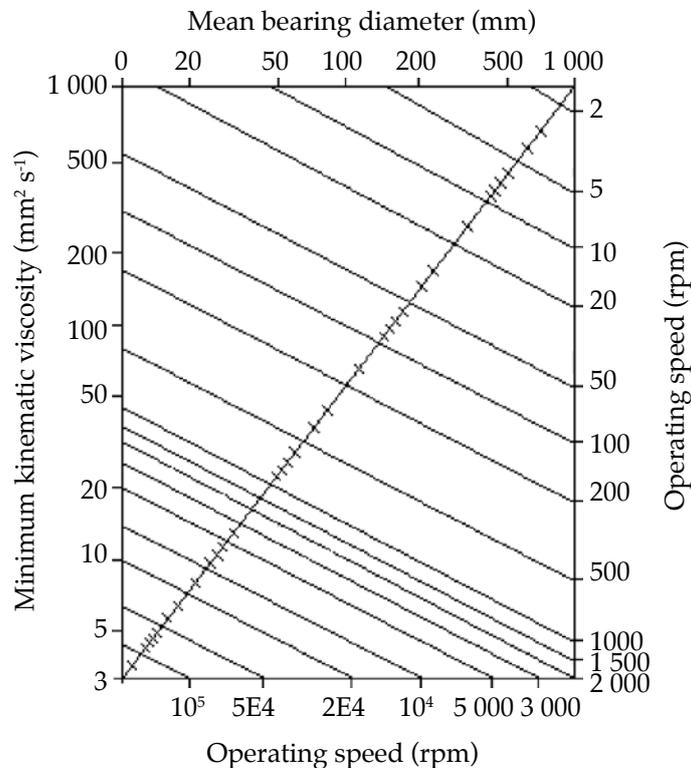


Figure 2. Minimum allowable viscosities for lubrication at operating temperature.



TABLE 3. WARNING LIMITS FOR VARIOUS SYSTEM OILS

Oil type	Viscosity change (%)	Water content (%)	Flash point (°C min)	TBN (mg KOH g <sup>-1</sup> )	TAN (mg KOH g <sup>-1</sup> )
Engine	30	0.30	160	2X fuel S%	-
Steam turbine	10	0.20	160	-	0.4
Hydraulic	10	0.20	160	-	0.2
Transformer	-	0.005	160	-	0.4
Gears	25	0.20	160	-	-
Compressors	25	0.20	160	-	0.4
Heat transfer	50	-	160	-	-

Water promotes oil oxidation and corrosion. Water contamination may originate from atmospheric condensation, leaking and by-product condensation of internal combustion engine operating at low temperatures.

Wear elements are elements that indicate a part or component is wearing out and may fail such as iron, copper, lead, tin, aluminium, chromium, nickel, sodium, silicon, silver, zinc, vanadium, etc. Faulty component could be identified from the type of wear metals thus catastrophic failure could be prevented.

One off sample in used oil analysis do not tell very much, thus, decision should be made based on trend analysis which are more important than absolute figures.

There is no single lubricant good for all purposes in the market. Lubricant selection is highly dependant on the system and application. The tribology analysis above shows that low viscosity lubricants are used for high speeds, high temperature and low pressures whereas high viscosity lubricants are good for low speeds, low temperature and high pressure.

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