

TRIBOLOGICAL DESIGN GUIDE

PART 2: LUBRICATION

Institution of
**MECHANICAL
ENGINEERS**



Tribology Group

The IMechE Tribology Group has produced this guide as Part 5 of a series of guides on Tribological Design which it wishes to make freely available for student use in connection with their studies. Part 1 is on Bearings, Part 2 covers Lubrication, Part 3 discusses Contact Mechanics and Part 4 focuses on a Wear Analysis Process and Part 5 on Wear; copies may be obtained from:

Institution of Mechanical Engineers, 1 Birdcage Walk, Westminster, London, SW1H 9JJ

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TRIBOLOGICAL DESIGN GUIDE

PART 2: LUBRICATION

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Tribology Group
The Institution of Mechanical Engineers
2nd Edition

FOREWORD

The design of machines elements involves consideration of:

- Kinematic function
- Strength
- Mechanical efficiency
- Required life

Friction and wear directly affect mechanical efficiency and may also undermine kinematic function and strength to the point of premature failure. Wear directly limits life at acceptable performance level.

Tribological considerations in machine element design are no less important than considerations of kinematic function and strength.

Kinematics and strength are comprehensively covered as core subjects in the education and training of engineers and scientists are commonly addressed in the practice of Engineering Design. The subject of Tribology is much more variably covered and, in consequence, tribological considerations are often overlooked in the subject of Design.

In view of its importance, the Tribology Group of the Institution of Mechanical Engineers is anxious to encourage the inclusion of tribological considerations in the practice of Design in the education of mechanical engineers. To this end, the Tribology Group has prepared a collection of Tribological Design Guides to offer to students of engineering in connection with their design studies. The hope is that, by making such data readily available, awareness in tribological design will be encouraged. The data presented will not, of itself, permit complete tribological design but references are included to more comprehensive sources of data and detailed design procedures.

It is the hope of the Tribology Group that those involved with the education of engineers and scientists will find it useful to reproduce this document for distribution to students or for incorporation into their own in-house produced Design Data Handbooks.

TRIBOLOGY AND THE NEED TO LUBRICATE

Tribology is the science of interacting surfaces. It encompasses the study of friction, lubrication, and wear. An integral part of design engineering is a consideration of what happens at the interface between two touching components. When the surface of one component moves over another there is always a resisting frictional force. If the surfaces are in close proximity then the peaks of the surface roughness (called asperities) interact, increasing friction, and may cause surface damage. The primary purpose of a lubricant is to separate these contacting surfaces and thereby reduce friction and wear. They may, in addition, act as a cooling medium or as protection from corrosion.

Tribological Failure Mechanisms

All too often in industrial machinery the process of lubrication fails, frequently through poor design. An incorrect choice of lubricant may have been made (e.g. inadequate viscosity or additive package). The lubricant may not be suited to the operating conditions (e.g. high speeds, vacuum, or temperature extremes), or there may have been insufficient lubricant supply (e.g. low flow rate or blockages). Table 1 shows some of the tribological failure mechanisms which can result from unsatisfactory lubrication.

Table 1: Tribological failure mechanisms associated with inadequate lubrication

Failure	Nature of Failure	Remedy
Adhesive Wear	<p>Surface contact occurs and asperities locally weld.</p> <p>Junctions are pulled apart.</p> <p>Material removal from one or both of the surfaces.</p>	<p>Increase film thickness (e.g. more viscous oil, lower load, higher speed).</p> <p>Boundary additives.</p> <p>Increase hardness.</p>
Abrasive Wear (Two Body)	<p>Asperities of a hard surface plough through the opposing softer surface.</p> <p>Material is removed from the softer surface.</p>	<p>Increase film thickness.</p> <p>Design for sacrifice of the softer surface.</p> <p>Increase material hardness.</p>
Abrasive Wear (Three Body)	<p>Solid particles in the lubricant become trapped in the contacts and remove material when sliding occurs.</p>	<p>Increase film thickness.</p> <p>Remove particulates by filtration.</p> <p>Increase material hardness.</p>
Corrosive Wear	<p>Material removal by oxygen or water attack at the surfaces.</p> <p>Some lubricant contaminants or additives may chemically attack surfaces.</p>	<p>Ensure protective lubricant film on components.</p> <p>Remove contaminants.</p> <p>Anti-corrosion additive.</p>
Scuffing	<p>Heat generation causes viscosity to drop and film thickness to reduce.</p> <p>This in turn results in more heating, eventually leading to film collapse, local welding and seizure.</p>	<p>Increase lubricant film thickness.</p> <p>Increase cooling.</p> <p>Extreme pressure additives.</p>
Contact Fatigue	<p>Repeated contacts induce cyclic stressing.</p> <p>Fatigue cracks may initiate, particularly if there is a defect present (e.g. roughness, surface dent, or material defect).</p>	<p>Reduce contact loads/cycles.</p> <p>Increase lubricant film thickness.</p> <p>Remove stress concentrations.</p>

TYPES OF LUBRICANT

There are five main classes of lubricant; oils (mineral or synthetic), emulsions, greases (oils with a thickening agent), solid lubricants, and gases.

Mineral Oils

Obtained as the heavier fractions from the distillation of petroleum. Typically hydrocarbons of 25 to 45 carbon atoms in the chain. Table 2 summarises the three mineral oil classifications.

Table 2: Classification of mineral oil, structure, and properties

Mineral oil	Molecular structure	Typical properties		
		VI ¹	Density	Pour point ²
Paraffinic	Saturated (straight or branched) chain molecules	High >80	High	Low
Napthenic	Saturated ring molecules	Low 30-80	Low	High
Aromatic	Carbon atoms in ring with alternating single and double bonds	Very low <30	Very low	Very high

A mineral oil will usually be a mixture of all three of these hydrocarbons and small quantities of compounds containing oxygen, sulphur, phosphorous, and nitrogen (sometimes known as asphaltenes). Greases consist of a base oil with 5-20% dispersed thickener and selected additives. Table 3 lists some of the more commonly encountered base oils, thickeners and additives.

Table 3: Common grease base oils, thickeners, and additives

Base oils	Mineral oil Synthetic hydrocarbon Di-esters Silicones Phosphate esters Fluorinated oils
Thickeners	Lithium and aluminium complexes Lithium, sodium, calcium, and aluminium soaps Polyurea, diurea Bentonite clay Polymers, PTFE
Additives	Antioxidants EP additives Corrosion inhibitors Tackiness additives Water repellants

¹ Viscosity Index V1 see page 8

² Pour point – see page 8

The thickener forms a soft solid matrix which holds the oil. The quantity of thickener influences the consistency of the grease; the type of thickener influences the grease properties. The consistency of a grease does not vary greatly with temperature until the 'drop point' is reached (typically 100-300°C depending on the thickener); then the fibrous structure breaks down and the grease becomes liquid.

Synthetic Oils

To obtain a greater range of lubrication properties, lubricants are also synthesised. Table 4 summarises some of the more commonly used.

Table 4: Common synthetic lubricants and their uses

Fluid	Features	Typical applications
Synthetic hydrocarbon	Higher VI than most mineral oils.	Most uses
Phosphate esters	Fire and radiation resistant, low temp stability, attack plastics.	Aircraft hydraulic systems
Organic esters	Temperature stable, may be corrosive, high VI.	Gas turbines
Silicones	Chemically inert, repel water, poor boundary lubrication.	Rolling bearings
Polyglycols	Decompose cleanly, miscible with water.	Hydraulic and brake fluids
Fluoro ethers	Thermal stability, chemically inert, resistant to solvents, costly.	Chemical plant, space
Polyphenyl ethers	Temperature stable, radiation resistant, expensive.	Aircraft hydraulic systems

Lubricant Additives

Usually chemical compounds are added to the lubricant base stock to improve certain of its properties. Table 5 contains a description of some the more commonly used additives.

Table 5: Lubricant additives and their function

Lubricant additive	Purpose
Anti-oxidants	Reduce oxidation of the lubricant.
Anti-wear (AW)	Reduce wear at moderate temperature. Adsorb on the surface to form thin protective films.
Basic additives	Neutralise acidic contaminants in the lubricant (sulphuric acid, oxidation products etc.)
Corrosion inhibitors	Protect certain non-ferrous metals (copper, lead, silver, aluminium etc.) by reacting with the surface. Form a protective film.
Detergents	Prevent the build-up of deposits in hot running engines.
Dispersants	Prevent the coagulation of carbon deposits which may form a sludge and block oilways.
Extreme pressure (EP)	Prevent scuffing of components operating under severe conditions React with the metal surfaces producing a thin protective film.
Foam inhibitors	Long chain polymers or silicones which act to destabilise surface foam.
Friction reducers	Reduce friction under boundary lubrication conditions. Form an organic surface film on the component surfaces.
Pour point depressants	Solubilise the wax structure that forms in paraffinic oils at low temperatures.
Rust inhibitors	Adsorb onto ferrous surfaces to protect against corrosion
Viscosity index (VI) improvers	Polymer chains uncoil to thicken the lubricant at elevated temperatures.

A typical commercial lubricant will be a formulation of 90 to 95% base oil with 5 to 10% dissolved additives. Table 6 shows some common formulations for different machine applications.

Table 6: Typical lubricant formulations for various machine applications

Application	Base oil	Additives
Gasoline engine crankcase oil	Mineral oil, synthetic hydrocarbon	Anti-oxidant, corrosion inhibitor, VI improver, AW, detergent, dispersant
Diesel engine oil	Mineral oil	Anti-oxidant, AW, corrosion inhibitor, basic additive
Gas turbine oil	Di-ester, polyol-ester	Anti-oxidant, AW, corrosion inhibitor
Industrial gear oil	Mineral oil	Anti-oxidant, EP or AW, anti-foam
Rolling bearing grease	Mineral oil and lithium soap	EP, AW, anti-oxidant
Hydraulic systems	Mineral oil, phosphate ester, emulsions	Anti-oxidant, AW, corrosion inhibitor, VI improver

Emulsions

Dispersions of oil droplets in water (or water in oil 'invert emulsions'). The mixture is stabilised with 1-10% surfactant (usually a detergent or soap). Emulsions have good cooling properties and low flash points. They are frequently used as metal cutting fluids, cold rolling lubricants, and fire retardant hydraulic fluids.

Gases

The viscosities of gases are low so externally pressured lubrication is usually employed. Resulting bearings designs typically have small clearances but very low frictional resistance. The lubricating gas is usually air, although process gases and others which do not decompose or attack the component materials may be used.

Solid Lubricants

Solids of low shear strength may be interposed between contacting components. Low shear strength, non corrosive materials which adhere well to the surfaces make good solid lubricants. The three most common are graphite, molybdenum disulphide, and PTFE. All three have a layered structure; they can thus support high loads perpendicular to the layers, but also shear easily parallel to the layers. Low strength metals, such as lead, gold, or tin are sometimes used; they tend to have higher friction coefficients but have the ability to withstand greater temperature ranges.

PROPERTIES OF LUBRICANTS

In hydrodynamic lubrication the most important property of the lubricant is its viscosity. It is this which is the principal factor controlling the lubricating film thickness. For applications covering a wide temperature range the variation of viscosity with temperature is critical. In the elastohydrodynamic lubrication regime the variation of viscosity with pressure is also important. In boundary lubrication the surface film is dependent on the presence of active additives.

Viscosity

The most important property of a lubricant is its viscosity. This is a measure of its resistance to flow and is defined in two different ways.

The dynamic (or absolute) viscosity, η is defined as the shear stress divided by the strain rate (which is equal to the velocity gradient) in the relation:

$$\tau = \eta \dot{\gamma} = \eta \frac{du}{dy}$$

Note: The SI unit of η is the Pascal second (Pa s). It is also expressed in poise or centipoise (where 1 cP = 10^{-3} Pa s), and Reyn (where 1 Reyn = 6895 Pa s).

The kinematic viscosity ν , is a composite of the dynamic viscosity and the density:

$$\nu = \frac{\eta}{\rho}$$

It is measured by timing the flow under gravity of the lubricant through some form of capillary tube. The SI unit of ν is m^2/s . It is also expressed in Stokes or centistokes (cSt), where 1 cSt = $1 \text{ mm}^2/\text{s} = 10^{-6} \text{ m}^2/\text{s}$. Other scales for the measurement of kinematic viscosity include: Redwood (No.1 and No.2) seconds, Saybolt universal seconds, Saybolt Furol seconds, and Engler degrees.

In most tribological calculations the dynamic viscosity is the parameter used. The kinematic viscosity is of lesser importance. For comparison, Table 7 shows typical dynamic viscosities, η in Pascal seconds of several fluids at 20°C.

Table 7: Typical dynamic viscosities of some fluids at 20°C

Fluid	η (Pa s)
Bitumen	$10^2 - 10^6$
Honey	2
SAE 50 motor oil	0.8
Glycerine	0.5
SAE 30 motor oil	0.3
Olive oil	0.1
SAE 10 motor oil	0.07

Fluid	η (Pa s)
Clock oil	$0.5 \times 10^{-2} - 1 \times 10^{-2}$
Mercury	1.6×10^{-3}
Turpentine	1.5×10^{-3}
Water	1×10^{-3}
Petrol	0.6×10^{-3}
Ether	0.2×10^{-3}
Air	1.8×10^{-5}

Viscosity Variation with Temperature

As the temperature of a liquid increases the molecules move further apart, the intermolecular forces decrease and the viscosity falls very rapidly. Figure 1 shows the viscosity temperature characteristics of various oils. The data for most lubricating oils is linear on a double natural logarithm against natural logarithm plot. This is true for both kinematic and dynamic viscosities.

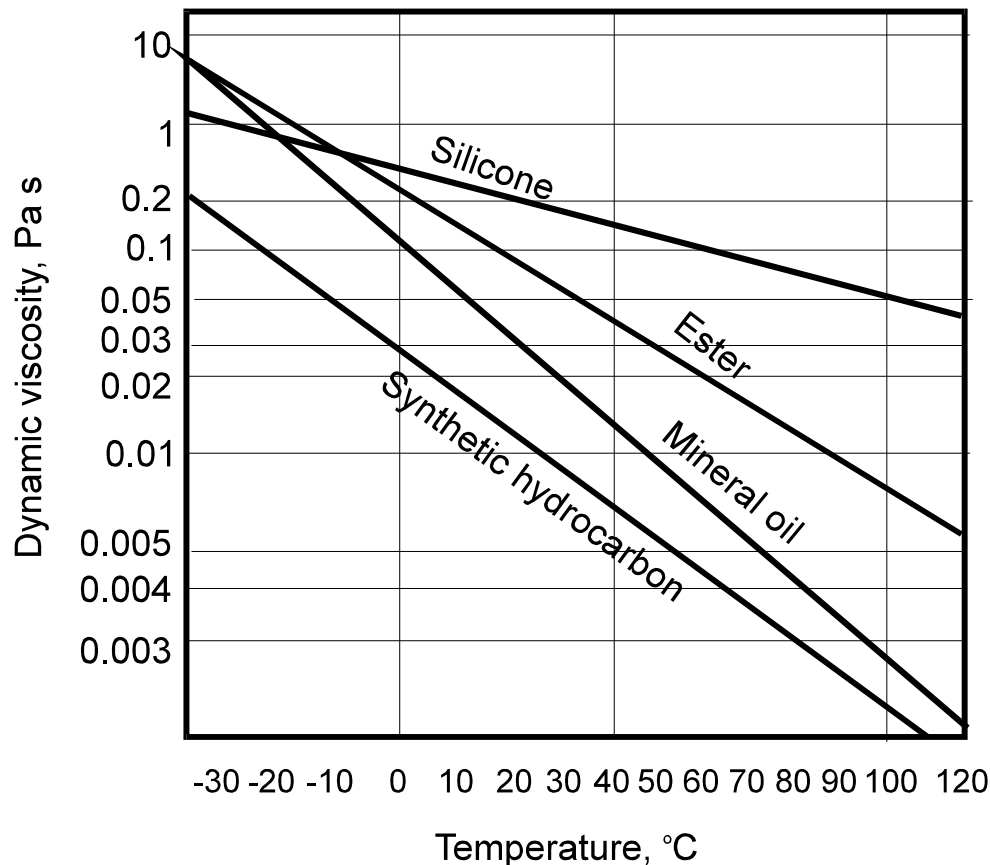


Figure 1: Viscosity variation with temperature for several oils. [Note the use of natural logarithmic scales; the axis are $\ln(\ln \eta)$ against $\ln T$.]

The Viscosity Index (VI) is used to describe the variation of viscosity with temperature. Lubricants with a low viscosity index show a relatively greater rate of decrease of viscosity with temperature. Thus, for applications with a wide temperature range, a high viscosity index lubricant is desirable.

Long chain polymers are frequently added to oils (known as VI improvers) to increase the viscosity index. As the lubricant warms up the polymer chains uncoil and restrict the flow of the lubricant thus slowing the rate of viscosity decrease.

The calculation used to obtain the VI is given on the next page. Table 8 details typical dynamic viscosities and VI's for mineral and synthetic oils. The nomograph included as Figure 2 may be used to determine the VI from the kinematic viscosities at the two reference temperatures.

Table 8: Typical dynamic viscosities and VI's for mineral and synthetic oils

Lubricating oil	Dynamic Viscosity (mPa s, cP)		Viscosity Index
	40°C	100°C	
Light naphthenic oil	20-40	2-4	45
Heavy naphthenic oil	100-150	5-10	43
Light paraffinic oil	30-40	3-5	98
Heavy paraffinic oil	70-120	8-10	95
Cylinder oil	600-800	20-30	95
Phosphate esters	10-40	2-4	0-100
Organic esters	10-80	2-10	120-180
Silicones	50-70	15-25	175-200
Polyglycols	20-80	4-10	75-150
Fluorocarbons			-25
Polyphenyl ethers	60-80	5-10	-60

The viscosity of the test oil at 40°C is compared with two standard oils, one with a VI of 0 and the other with a VI of 100. The standard oils are chosen such that they have the same viscosity as the test oil at 100°C. Then, $VI = 100(L-U)/(L-H)$; where L, H, and U are the kinematic viscosities at 40°C of the 0 VI oil, the 100 VI oil, and the test oil respectively. Many modern synthetic oils now have VIs which fall outside of these standard values (i.e. give values of VI which are negative or greater than 100).

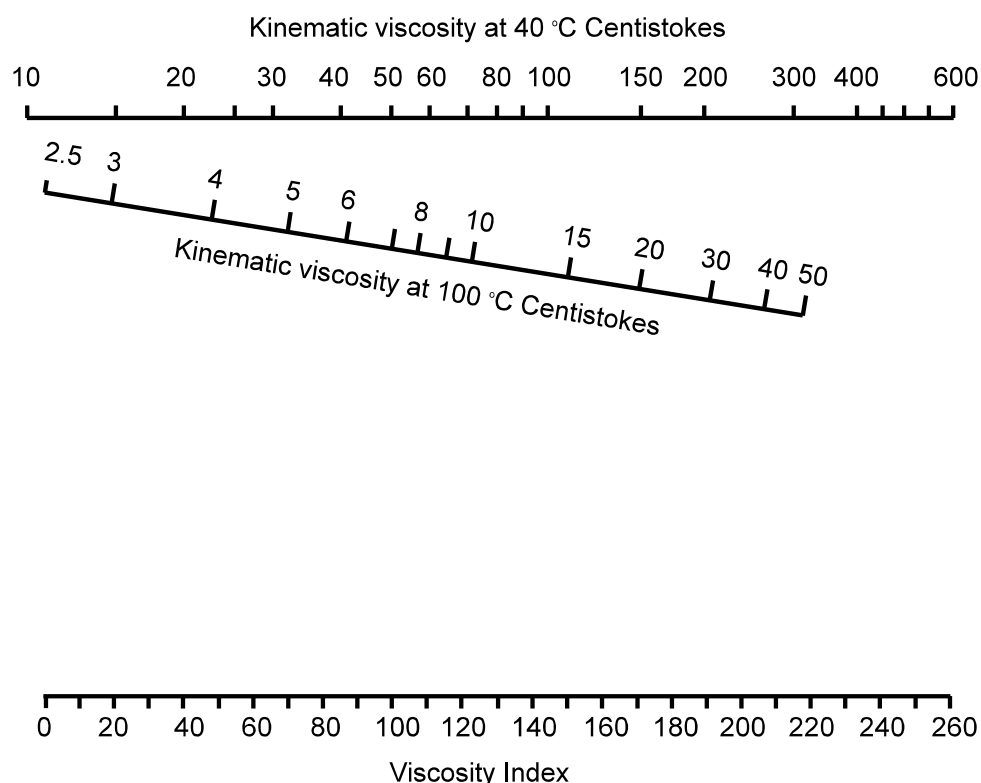


Figure 2 Nomograph relating the lubricant viscosity index to kinematic viscosities at reference temperatures of 40°C and 100°C.

SAE Viscosity Classification

A widely used viscosity grading system has been produced by the Society of Automotive Engineers. The classification defines the minimum operating viscosity at a high temperature, and the maximum viscosity at a low temperature (the suffix 'W' denotes a lubricant for winter use). Multigrade oils (say 10W40) achieve both the higher and lower SAE specifications.

Table 9 gives details of the classification for engine (crankcase) oils. Table 10 gives the classification for transmission oils (where the 'W' grade is specified as the temperature at which the viscosity reaches 150,000 cP).

Table 9: SAE Classification for engine oils

SAE Viscosity Grade	Dynamic viscosity at 100°C (cSt)	Kinematic viscosity at -18°C (cP)
5W	<1250	>3.8
10W	<2500	>4.1
15W	<5000	>5.6
20W	<10,000	>5.6
20	-	5.6<v<9.3
30	-	9.3<v<12.5
40	-	12.5<v<16.3
50	-	16.3<v<21.9

Table 10: SAE Classification for transmission oils

SAE Viscosity Grade	Minimum temperature for viscosity of 150,000 cP	Kinematic viscosity at 100°C (cSt)
75W	-40°C	>4.1
80W	-26°C	>7.0
85W	-12°C	>11.0
90	-	13.5<v<24.0
140	-	24.0<v<41.0
250	-	>41.0

Pressure Viscosity Coefficient

Most lubricating oils undergo an increase in viscosity when subjected to high pressure (this is important in the generation of elastohydrodynamic and squeeze lubricating films). An approximate relation for this variation is:

$$\eta_p = \eta_0 e^{\alpha p}$$

where η_0 and η_p are the dynamic viscosity at atmospheric pressure and at pressure p respectively. The property α is known as the pressure viscosity coefficient (usually expressed in the units GPa^{-1}). Typically mineral oils have pressure viscosity coefficients in the range $20 \text{ GPa}^{-1} < \alpha < 30 \text{ GPa}^{-1}$.

Viscosity with Shear Rate

Most simple structure fluids (water, benzene, light oil) are Newtonian (ie their viscosity is independent of shear rate). More complex fluids may be non-Newtonian (for example, thixotropic and dilatant fluids show respectively decreasing and increasing viscosities with increased shear rate). A common case of non-Newtonian behaviour is a mineral oil with VI additives; the viscosity falls by as much as half at shear rates $>10^4 \text{ s}^{-1}$.

Thermal Capacity and Thermal Conductivity

Both should be as high as possible to provide effective cooling by the lubricant. Mineral oils typically have thermal capacities in the range 1800 to 2200 J/kg K, thermal conductivities in the range 0.12 to 0.13 W/m K.

Pour Point

The temperature at which waxes dissolved in the oil separate. Below the pour point the oil becomes so viscous that it can no longer be poured. Mineral oils typically have pour points in the region of -50 to -60°C .

Flash Point

The temperature at which the vapour of the fluid will ignite when exposed to a naked flame. For fire safety a high value is desirable. Mineral oils typically have flash points in the range 60 to 120°C .

HOW LUBRICANTS WORK

Lubrication Regimes

Most of the failure mechanisms, described in Table 1, are a result of the surfaces coming into too close contact. An important role of a lubricant is to ensure this contact is minimised. The load between the components must be supported in some way; and the motion must either not remove the lubricant film or must act to entrain it. There are several mechanisms by which this process is achieved. They are commonly known as lubrication regimes and are summarised in Table 11 and Figure 3.

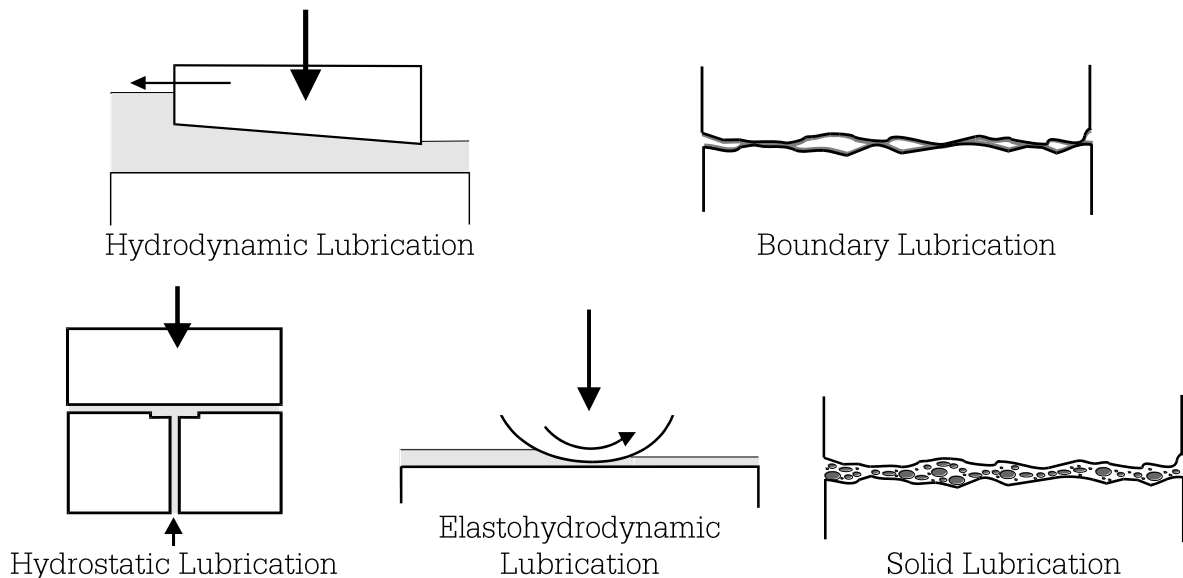


Figure 3: Schematic diagrams of the lubrication regimes

Table 11: Typical film thickness, friction coefficients, and mechanisms for the various lubrication regimes.

Regime	Film Thickness (μm)	Typical Friction Coefficient	Mechanism of Film Formation	Typical Applications
Hydrodynamic Lubrication	1 – 100	0.01 – 0.03	Lubricant is dragged into wedge between components. The lubricant pressure increase supports the applied load.	Journal bearings Machine slideways Piston ring/liner
Boundary Lubrication	0.001 – 0.05	0.1 – 0.3	Surfaces not being fully separated. Thin chemical layers reduce the tendency of the asperities to adhere.	Metal cutting Bearing start-up or shutdown
Hydrostatic Lubrication	1 – 100	0.01 – 0.03	Lubricant pumped into the interface to separate surfaces.	Machine Tool spindles
Elastohydrodynamic Lubrication (EHL)	0.1 – 1.0	0.001 – 0.01	As hydrodynamic, but high local pressure causes increase in viscosity and elastic deformation.	Rolling element bearings Gears, cams and tappets
Solid Lubrication	-	>0.05 – 0.3	Low shear strength solid separates surfaces. Shears more easily than the component materials.	'Dry' bearings, vacuum, graphite, PTFE, MoS ₂

The Stribeck Curve and Lambda Ratio

Figure 4 shows the friction coefficients obtained for three regimes of lubrication which occur in sliding pad and journal type bearings. The co-ordinate is known as the Stribeck parameter (where η is the lubricant dynamic viscosity, V is the contact speed, and P is the pressure).

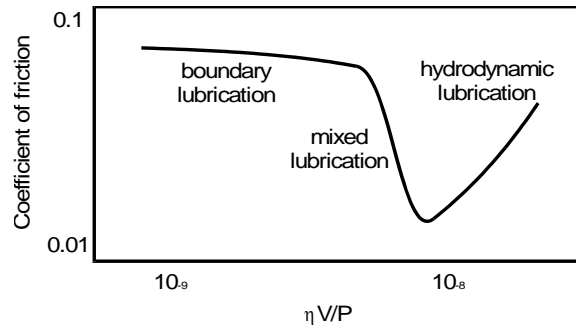


Figure 4: The Stribeck Curve: A plot of friction coefficient against the parameter, $\eta V/p$. The regimes of boundary, mixed, and hydrodynamic lubrication are shown.

At low speeds and lubricant viscosity, a thin film is generated. This is not sufficient to separate the asperities and some surface contact occurs resulting in appreciable friction (this is the region of boundary lubrication). As the Stribeck parameter increases the lubricant film thickness increases, hence the degree of contact drops and the friction reduces (this is known as mixed lubrication). At the trough in the graph the surfaces are fully separated and the friction is at a minimum. At higher speeds and viscosity, thick films are generated (hydrodynamic lubrication).

As the Stribeck parameter increases further the film thickness increases. The speed is increasing and so therefore is the velocity gradient; this results in an increase in lubricant shear stress and hence friction increases.

Another useful parameter is the lambda ratio, λ given by;

$$\lambda = \frac{h}{\sigma_c}$$

where h = lubricant film thickness
 σ_c = composite rms roughness of the two surfaces

$$\sigma_c = \sqrt{\sigma_1^2 + \sigma_2^2}$$

This lambda ratio has been found to correlate with many of the surface contact failure mechanisms. The lower the lambda ratio the greater the probability of surface contact and the likelihood of wear, fatigue, or scuffing failures. Typically, designs should satisfy $\lambda > 1.5$ to minimise surface contact.

LUBRICANT SELECTION FOR MACHINE COMPONENTS

Many varieties of lubricant are commercially available. Each manufacturer will supply a whole range of mineral oils, synthetics, greases, and metal working lubricants. Many components will satisfactorily operate with a variety of lubricants. The following section is intended only as a basic guide to lubricant selection. More complicated applications or severe environments will require more detailed design.

Fluid Film Plain Bearings

Plain bearings (journal or thrust types) are usually lubricated with a mineral oil. Where a greater temperature range is required, synthetics may be employed. Greases are less common and restricted to surface speeds below 1-2m/s. The most important property of the lubricant is its viscosity (over the required operating temperature range). If the viscosity is too low the surfaces may not be adequately separated; if too high then power loss will be unnecessarily high. Figure 5 may be used to determine the required oil viscosity, at the operating temperature. The surface speed and mean pressure are calculated from:

$$u = \pi d n$$

$$\bar{p} = \frac{0.4W}{bd}$$

(Thrust bearing)

$$\bar{p} = \frac{W}{bd}$$

(Journal bearing)

where d = journal/ mean thrust pad diameter (m)
 b = bearing width (mm)
 n = shaft speed (rpm)
 W = applied load (N)

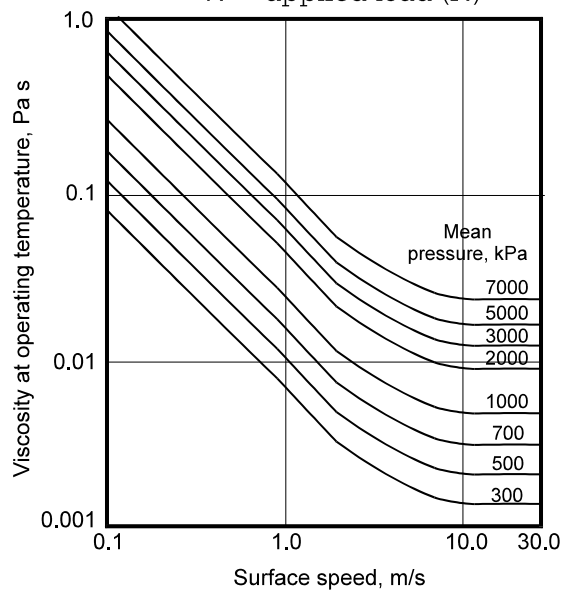


Figure 5: Viscosity selection for plain bearings.

Once the viscosity at operating temperatures³ has been determined, this is converted to the viscosities at reference temperatures (using Figure 1 or VI data) for lubricant selection. Additives are not usually required, unless dictated by other components in the system (i.e. likelihood of lubricant contamination or oxidation).

Rolling Bearings

For low speeds ($d_m N < 500,000$)⁴ and low loads grease lubrication is common, since it may be sealed within the bearing cavity. Soft greases should be used with faster bearings to minimise churning losses. For higher loads and speeds oil lubrication is necessary. Figure 6 may be used to select the required oil viscosity at the operating temperature⁵ for a given bearing bore and speed (most bearing manufacturers publish similar charts in their catalogues). The chart is intended as a guide only; increases in bearing life can be achieved by using higher viscosities than those quoted, but may result in greater heating or pumping problems.

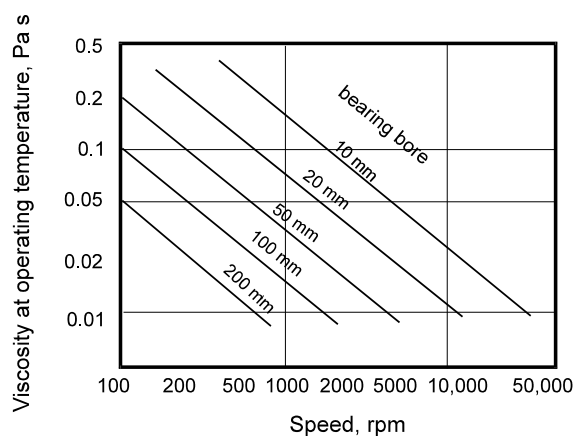


Figure 6: Viscosity selection for rolling element bearings.

Typically rust and oxidation inhibitor additives are included. Extreme pressure (EP) additives may also be required for highly loaded bearings, typically where $C/P < 4$ for ball bearings; $C/P < 6.5$ for roller bearings.⁶

Gears

³ Typically plain bearings operate at around 60 - 80°C.

⁴ $d_m N$ factor; the product of the bearing bore, mm and the shaft speed, rpm. See the IMechE Tribological Design Guide Part 5: Bearings.

⁵ Typically rolling bearings operate at around 40-60°C.

⁶ Where C is basic load rating of the bearing and P is the equivalent bearing load. See the IMechE Tribological Design Guide Part 5: Bearings.

Gears are usually lubricated with plain mineral oils. At high speeds, lower viscosity oils with anti-oxidants must be used. Figure 7 can be used to select the oil viscosity at the operating temperature⁷ for a given gear pitch line speed.

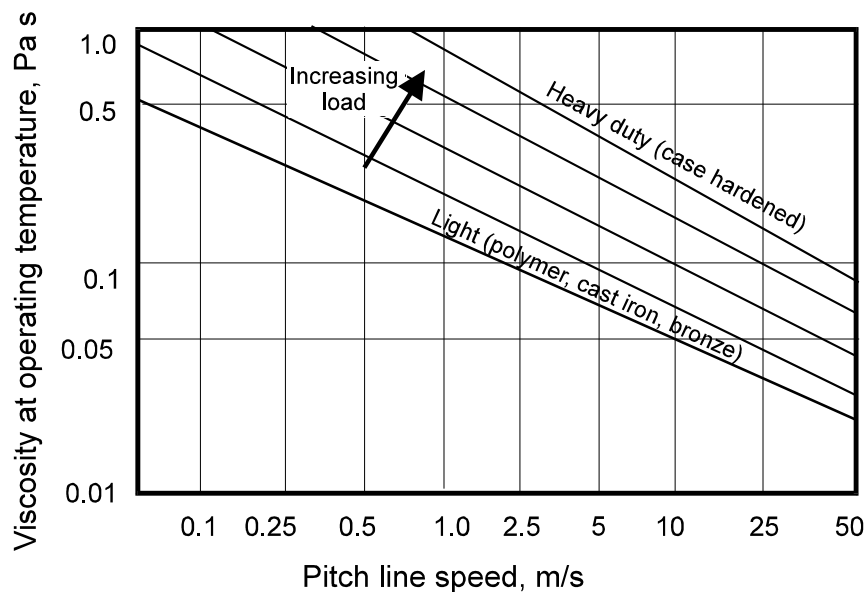


Figure 7: Viscosity selection chart for gears.

Hypoid gears and heavily loaded spiral bevel gears require the presence of an EP additive. For open gears (i.e. not enclosed in a gear box or chamber) it is important to ensure the oil is not thrown off; 'tackiness' additives are necessary.

Other Machine Elements

Slides. Usually a plain grease or mineral oil containing some boundary additive. The slideways, since they are frequently exposed during operation, must be kept free from contamination.

Wire ropes and chains. Mineral oils are normally used for the lubrication of ropes and chains (synthetics and greases tend to be too costly to be used in the bulk required for these applications). To ensure the lubricant is not thrown off at high speed or sudden accelerations, viscous oils or those containing 'tackiness' additives are used.

Flexible couplings (gear, spring, or chain type). Usually filled with a semi-fluid mineral base oil grease. At higher speeds continuous oil lubrication may be required.

Hinges, latches, and locks. For these types of simple components the choice of lubricant is usually not critical. A light 'general purpose' mineral oil applied directly to the contacting areas at regular intervals is usually sufficient.

Metal machining (cutting, drilling, turning etc.). Mineral oil-in-water emulsions are used to provide cooling and lubrication. Additives are usually included to reduce corrosion, toxicity, foaming, and for EP action.

Metal rolling. Low viscosity lubricants such as paraffin or emulsions.

⁷ Typically industrial gearing operates at around 70-90°C.

Cold drawing and extrusion. A coating of sodium soap is often applied to the material before working. For more severe operations polymeric or soft metal coatings are applied.

Special Lubrication Problems

Contaminated Environments. Where there is a risk from external contamination, a sealed-in grease system may be employed. Alternatively, a circulation system with filters (for solid contaminants) or separators (for fluid contaminants) is used.

High Vacuum. Employ solid lubricants and dry bearings. Otherwise use a high vacuum oil or grease.

Large Temperature Range Lubrication. Use an oil with a high VI, e.g. synthetic silicone oil.

High Load (Extreme Pressure) Lubrication. Above a contact pressure of 0.8 MPa gas lubrication is probably not possible, and an oil or grease must be employed. Above 5 MPa boundary lubrication additives (anti-wear or EP) may be required. Above 8 MPa a solid lubricant may be necessary.

Extreme Temperature Lubrication. For temperatures up to 260°C an oil circulation system with cooling is acceptable. For greater than 260°C then solid or gaseous lubricants need to be employed. For temperatures below -60°C a solid lubricant or cryogenic liquid/gas must be used.

Table 12: Typical maximum and minimum operating temperatures for mineral oils, synthetics, and greases.

Lubricant	Minimum Operating Temperature (°C)	Maximum Operating Temperature (°C)
Mineral Oils	0 to -50	150 – 200
Phosphate Esters	-57	90 – 120
Organic Esters	-35 to -65	250 - 300
Silicones	-30 to -70	200 - 280
Polyglycols	-20	200 - 260
Fluoro Ethers	-50	200 – 380
Polyphenyl Ethers	-7	300 – 380
Lithium Grease Mineral Base Oil	-40	100 – 150
Clay Grease Mineral Base Oil	-30	150 – 200
Lithium Grease Silicone Base Oil	-55	150 – 200
Complex Soap Grease Silicone Base Oil	-55	200 – 250
PTFE		200 – 250
Graphite		400 – 500 ⁸
Molybdenum Disulphide		300 – 350

Incompatibility Problems. Lubricating oils chemically attack some non-metallic materials used commonly in seals and hoses. For example, mineral oil will attack natural rubber, polyurethanes, and plasticized plastics. Care should be taken to ensure any materials used in the machine design are compatible with the selected lubricant.

⁸ Poor lubricant above 150°C.

LUBRICANT SUPPLY SYSTEMS

Once a lubricant has been selected it must be supplied to the contacting components. The purpose of the supply system is, at minimum, to provide an adequate flow of lubricant. The system may also be required to provide; lubricant cooling, pre-heating, or removal of contaminants. The following is list of the most common types of lubricant supply system. All but the last two are known as 'total loss' systems where no attempt is made to collect or recycle the lubricant.

Direct Application

An oil can or a grease gun is used to feed lubricant directly onto the components.

Refillable Reservoir

An oil can or a grease gun is used to recharge a reservoir, usually fitted with a sight glass, located near the component to be lubricated. The lubricant is usually drip fed onto the component. A common alternative for the lubrication of lightly loaded journal bearings is the wick or pad feed; where lubricant is fed from the reservoir to the bearing via the capillary action of a wick. Figure 8a shows a typical drip feed lubricant reservoir.

Centralised Total Loss System

Where there are a large number of components requiring the same lubricant it may be cost effective to have a single reservoir from which the lubricant is pumped to each component through individual flow lines. The system may consist of a single pump supplying lubricant to a dividing manifold (indirect system) or a multi-piston pump supplying several outlets simultaneously (direct system). In both cases it is important to monitor supply to each point, using sight glasses or flow meters. These centralised lubrication systems are usually manufactured and installed by specialist companies. Figures 8b and 8c show two types of centralised total loss systems, the first using multi-outlet pump, the second with a divider manifold.

Mist Systems

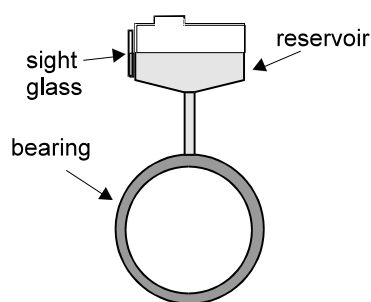
Oil is dispersed as a mist in a clean air supply and passed through flow lines to the lubrication points. The mist then passes through a fine nozzle to increase the stream speed. The oil droplets then adhere to the surfaces they strike. The advantage of this system over the conventional centralised total loss system is that it provides some cooling action and is less wasteful of lubricant.

Oil Bath Systems

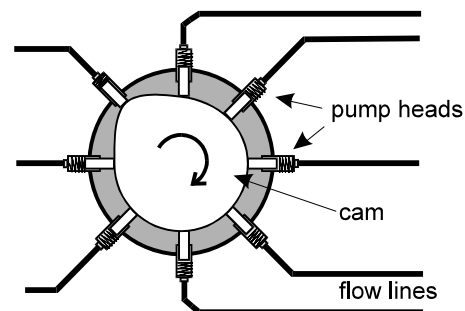
The contacting components are sealed in a chamber part filled with oil. The submerged parts then pick up oil and splash over the contacting parts. If the oil level is too high then power is lost heating the oil by churning. The oil level is thus such that the components are partly submerged. The effectiveness of the splashing action is then dependent on the geometry of the components. The action may be assisted by including rings, disks, or splash wheels on the rotating shaft. Seals are required where the shafts penetrate the lubricant chamber.

Circulation Systems

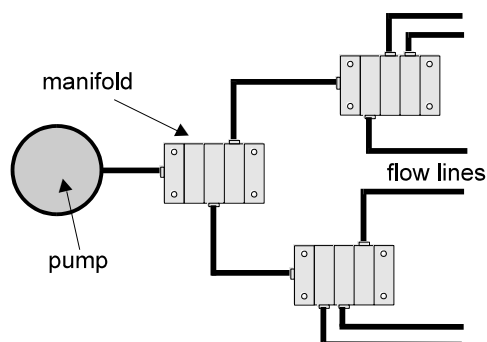
Where there is a requirement for cooling, filtration, or oil recycling then a circulation system is employed. A simple circulation system (Figure 8d) consists of a reservoir, pump, feed lines to the lubrication points, and return lines. More complex systems may include; flow meters, pressure gauges, pre-heaters to improve flow during cold starts, coolers to remove excessive heat during normal running, in-line filters and strainers to remove solid contaminants, water or gas separators to remove fluid contaminants, and sample points and chip detectors for condition monitoring.



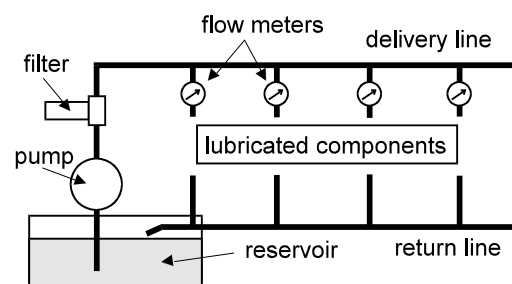
Refillable Reservoir and Drip Feed



Multi-Outlet Pump Centralised Total Loss System



Single Outlet Pump Centralised Total Loss System



Simple Circulation System

Figure 8: Schematic diagrams of lubrication systems

Table 13: Advantages and disadvantages of various lubricant supply systems (L=low, M=medium, H=high).

Lubrication system	Operating Costs			Flow control	Cooling	Reliability	Applications
	Initial	Labour	Maintenance				
Direct application	L	H	L	None	No	Variable	Few easily accessible components
Refillable reservoir	L	M	L	Some	No	Variable	Few less accessible components
Centralised total loss	H	L	H	Good	No	High	Many close less accessible components
Oil mist/fog	H	L	M	Good	Yes	High	Few close components requiring cooling
Oil bath	M	L	L	None	Some	High	Individual systems, requiring cooling and contamination removal
Circulation	H	L	H	Good	Yes	Yes	Many components needing effective cooling and contamination removal

OIL CHANGING, DISPOSAL, AND RE-USE

The life of a lubricant is generally limited by oxidation (increases viscosity and acidic contaminants), contamination (by solids, liquids, gases, or bacteria), or when the additives have been exhausted. Lubricant supply system manufacturers generally recommend oil-change periods; although some form of lubricant monitoring is useful. An oil supply should be checked for viscosity increase, discolouration, presence of contaminants, and corrosion of metal parts. Table 14 shows some guidelines for the frequency of oil change. A useful rule of thumb is that the oil life is halved for every 10°C increase in temperature. When a change is indicated, the old oil must be fully drained and flushed out of the system.

Table 14: Guidelines for the frequency of oil change.

Lubrication system	Operating temperature	Oil-change period
Oil-bath	<70°C	1 year
Circulation system	<50°C	>2 years
Circulation system	60-70°C	1 year
Circulation system	>70°C	3 months

Disposal of the waste oil largely depends on the quantities and the quality involved. For large volumes of contaminated but not seriously degraded lubricant, 'laundering' using filters or centrifuges may be cost effective. Waste oils which have become degraded (but not contaminated with light fuels or solvents) may be mixed with fuel-oil supplies for burning. Some specialist companies re-refine waste lubricant, which requires removal of contaminants and oxidation breakdown products, and then replenishment of the additive package.

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EXAMPLE PROBLEMS

- Q1. A lubricant has a kinematic viscosity of 30 cSt at 40°C and a viscosity index of 80.
- (a) Determine the kinematic viscosity at 100°C. Plot this data using appropriate axes.
 - (b) Using the graph plotted in (a), determine the viscosity of the lubricant at 70°C.
- A1. Using the nomograph of Figure 2. The viscosity at 100°C is approximately 5 cSt. The data should be plotted on a graph of $\ln \ln$ viscosity against \ln temperature (similar to Figure 1). The resulting viscosity at 70°C is found to be approximately 10.5 cSt.
- Q2. Select a lubricant for the following applications:
- (a) A deep groove ball bearing with bore diameter 50 mm, outer diameter 90 mm, radial load 3 kN, operating speed 1500 rpm, and L_{10} life 10,000 hours.
 - (b) A lightly loaded spur gear operating with a pitch line speed of 10 m/s.
 - (c) A journal bearing with journal diameter 76 mm, bearing length 38 mm, load 10 kN and operating speed 3000 rpm.
- A2.
- (a) Using the chart included as Figure 6, the viscosity of the lubricant should be at least 0.02 Pas at the bearing operating temperature. The ratio C/P was calculated as 11.7 in the previous example. Thus an EP additive would not be required. The $d_m N$ value is calculated as 105,000; grease lubrication would therefore be acceptable.
 - (b) From Figure 7 a viscosity of 0.05 Pas is suggested. Spur gears generally do not have the same high degree of sliding or loading as hypoid or spiral bevels; an EP additive is therefore unlikely to be required.
 - (c) The mean pressure is calculated as 3460 kPa, and the surface speed as 12 m/s. Using Figure 5 the viscosity at operating temperature is estimated as 0.012 Pas. Lubricant additives are not usually required for journal bearing applications.
- Q3. The rolling bearings in a high speed conveyor system have a 50 mm bore size and operate at 1000 rpm. Experience suggests that the bearings operating at around 40°C above the atmospheric temperature. Select an appropriate lubricant and supply method.
- A3. From Figure 6 it can be seen that a lubricant with a viscosity of 0.03 Pas is likely to be suitable. Depending on the temperature of the environment the lubricant could reach 70°C. To achieve this viscosity at such temperatures would require the use of a costly synthetic oil (see Figure 3). It may be cheaper, particularly if there are many bearings in the assembly, to supply a mineral based oil through a circulation system with cooling. If the conveyed products are likely to contaminate the lubricant (for example coal, sand etc.) then filtration and condition monitoring units should also be included.