# Lubrication of Gears and Bearings

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# Chapter 9
# Gears

## 9-1. General

*a.* Energy is transmitted from a power source to a terminal point, through gears that change speeds, directions, and torque. Gear lubricants are formulated and applied to prevent premature component failure, assure reliable operation, reduce operating cost, and increase service life. The important objectives accomplished by these lubricants include: reduction of friction and wear, corrosion prevention, reduction of operating noise, improvement in heat transfer, and removal of foreign or wear particles from the critical contact areas of the gear tooth surfaces.

*b.* Gears vary greatly in their design and in their lubrication requirements. Proper lubrication is important to prevent premature wear of gear tooth surfaces. When selecting a lubricant for any gear application the following issues must be considered: type and materials of gear; operating conditions, including rolling or sliding speed, type of steady load, and temperature; method of lubricant application; environment; and type of service. Enclosed gears -- those encased in an oil-tight housing -- usually require an oil with various additives, depending on the operating conditions. Rust, oxidation, and foam inhibitors are common. Extreme pressure (EP) additives are also used when loads are severe.

*c.* Worm gears are special because the action between the worm and the mating bull gear is sliding rather than the rolling action common in most gears. The sliding action allows fluid film lubrication to take place. Another significant difference is that worm gears are usually made of dissimilar materials, which reduces the chance of galling and reduces friction. EP additives usually are not required for worm gears and may actually be detrimental to a bronze worm gear. Lubrication can be improved by oiliness additives.

*d.* In open gear applications, the lubricant must resist being thrown off by centrifugal force or being scraped off by the action of the gear teeth. A highly adhesive lubricant is required for most open gear applications. Most open gear lubricants are heavy oils, asphalt-based compounds, or soft greases. Depending on the service conditions, oxidation inhibitors or EP additives may be added. Caution must be exercised when using adhesive lubricants because they may attract and retain dust and dirt, which can act as abrasives. To minimize damage, gears should be periodically cleaned.

## 9-2. Gear Types

*a. Spur gears.* Spur gears are the most common type used. Tooth contact is primarily rolling, with sliding occurring during engagement and disengagement. Some noise is normal, but it may become objectionable at high speeds.

*b. Rack and pinion.* Rack and pinion gears are essentially a variation of spur gears and have similar lubrication requirements.

*c. Helical.* Helical gears operate with less noise and vibration than spur gears. At any time, the load on helical gears is distributed over several teeth, resulting in reduced wear. Due to their angular cut, teeth meshing results in thrust loads along the gear shaft. This action requires thrust bearings to absorb the thrust load and maintain gear alignment.

*d. Herringbone.* Herringbone gears are essentially two side-by-side opposite-hand helical gears. This design eliminates thrust loads, but alignment is very critical to ensure correct teeth engagement.

*e. Bevel.* Bevel gears are used to transmit motion between shafts with intersecting center lines. The intersecting angle is normally 90 deg but may be as high as 180 deg. When the mating gears are equal in size and the shafts are positioned at 90 degrees to each other, they are referred to as miter gears. The teeth of bevel gears can also be cut in a curved manner to produce spiral bevel gears, which produce smoother and quieter operation than straight cut bevels.

*f. Worm.* Operation of worm gears is analogous to a screw. The relative motion between these gears is sliding rather than rolling. The uniform distribution of tooth pressures on these gears enables use of metals with inherently low coefficients of friction such as bronze wheel gears with hardened steel worm gears. These gears rely on full fluid film lubrication and require heavy oil compounded to enhance lubricity and film strength to prevent metal contact.

*g. Hypoid.* Hypoid gears are similar to spiral bevel gears except that the shaft center lines do not intersect. Hypoid gears combine the rolling action and high tooth pressure of spiral bevels with the sliding action of worm gears. This combination and the all-steel construction of the drive and driven gear result in a gear set with special lubrication requirements, including oiliness and antiweld additives to withstand the high tooth pressures and high rubbing speeds.

*h. Annular.* Annular gears have the same tooth design as spur and helical gears, but unlike these gears, the annular gear has an internal configuration. The tooth action and lubrication requirements for annular gears are similar to spur and helical gears.

## 9-3. Gear Wear and Failure

The most critical function provided by lubricants is to minimize friction and wear to extend equipment service life. Gear failures can be traced to mechanical problems or lubricant failure. Lubricant-related failures are usually traced to contamination, oil film collapse, additive depletion, and use of improper lubricant for the application. The most common failures are due to particle contamination of the lubricant Dust particles are highly abrasive and can penetrate through the oil film, causing “plowing” wear or ridging on metal surfaces. Water contamination can cause rust on working surfaces and eventually destroy metal integrity. To prevent premature failure, gear selection requires careful consideration of the following: gear tooth geometry, tooth action, tooth pressures, construction materials and surface characteristics, lubricant characteristics, and operating environment. The first four items are related to design and application, and further discussion is beyond the scope of this manual. These items may be mentioned where necessary, but discussions are limited to those aspects directly related to and affected by lubrication, including wear, scuffing, and contact fatigue. Refer to ANSI/AGMA Standard 1010-E95, and ASM Handbook Volume 18, for photographs illustrating the wear modes described in the following discussion.

*a. Normal wear.* Normal wear occurs in new gears during the initial running-in period. The rolling and sliding action of the mating teeth create mild wear that appears as a smooth and polished surface.

*b. Fatigue.*

(1) Pitting. Pitting occurs when fatigue cracks are initiated on the tooth surface or just below the surface. Usually pits are the result of surface cracks caused by metal-to-metal contact of asperities or

defects due to low lubricant film thickness. High-speed gears with smooth surfaces and good film thickness may experience pitting due to subsurface cracks. These cracks may start at inclusions in the gear materials, which act as stress concentrators, and propagate below and parallel to the tooth surface. Pits are formed when these cracks break through the tooth surface and cause material separation. When several pits join, a larger pit (or spall) is formed. Another suspected cause of pitting is hydrogen embrittlement of metal due to water contamination of the lubricant. Pitting can also be caused by foreign particle contamination of lubricant. These particles create surface stress concentration points that reduce lubricant film thickness and promote pitting. The following guidelines should be observed to minimize the onset of pitting in gear units:

- Reduce contact stresses through load reduction or by optimizing gear geometry.
- Steel should be properly heat-treated to high hardness. Carburizing is preferable.
- Gear teeth should have smooth surfaces produced by grinding or honing.
- Use proper quantities of cool, clean, and dry lubricant with the required viscosity.

(2) Micropitting. Micropitting occurs on surface-hardened gears and is characterized by extremely small pits approximately 10 μm (400 μ-inches) deep. Micropitted metal has a frosted or a gray appearance. This condition generally appears on rough surfaces and is exacerbated by use of low-viscosity lubricants. Slow-speed gears are also prone to micropitting due to thin lubricant films. Micropitting may be sporadic and may stop when good lubrication conditions are restored following run-in. Maintaining adequate lubricant film thickness is the most important factor influencing the formation of micropitting. Higher-speed operation and smooth gear tooth surfaces also hinder formation of micropitting. The following guidelines should be observed to reduce the onset of micropitting in gear units:

- Use gears with smooth tooth surfaces produced by careful grinding or honing.
- Use the correct amount of cool, clean, and dry lubricant with the highest viscosity permissible for the application
- Use high speeds, if possible.
- Use carburized steel with proper carbon content in the surface layers.

*c. Wear.*

(1) Adhesion.

(a) New gears contain surface imperfections or roughness that are inherent to the manufacturing process. During the initial run-in period, these imperfections are reduced through wear. Smoothing of the gear surfaces is to be expected . Mild wear will occur even when adequate lubrication is provided, but this wear is limited to the oxide layer of the gear teeth. Mild wear is beneficial because it increases the contact areas and equalizes the load pressures on gear tooth surfaces. Furthermore, the smooth gear surfaces increase the film thickness and improve lubrication.

(b) The amount of wear that is acceptable depends on the expected life, noise, and vibration of the gear units. Excessive wear is characterized by loss of tooth profile, which results in high loading, and loss of tooth thickness, which may cause bending fatigue.

(c) Wear cannot be completely eliminated. Speed, lubricant viscosity, and temperature impose practical limits on gear operating conditions. Gears that are highly loaded, operate at slow speeds, i.e., less than 30 m/min (100 ft/min), and rely on boundary lubrication are particularly subject to excessive wear. Slow-speed adhesive wear is highly dependent upon lubricant viscosity. Higher lubricant viscosities provide significant wear reduction, but viscosities must be carefully selected to prevent overheating.

(d) The following guidelines should be observed to minimize the onset of adhesive wear in gear units:

- Gear teeth should have smooth surfaces.

- If possible, the run-in period for new gear units should be restricted to one-half load for the first hours of operation.

- Use the highest speeds possible. High-load, slow-speed gears are boundary lubricated and are especially prone to excessive wear. For these applications, nitrided gears should be specified.

- Avoid using lubricants with sulfur-phosphorus additives for very slow-speed gears (less than 3 m/min, or 10 ft/min).

- Use the required quantity of cool, clean, and dry lubricant at the highest viscosity permissible.

(2) Abrasion. Abrasive wear is caused by particle contaminants in the lubricant. Particles may originate internally due to poor quality control during the manufacturing process. Particles also may be introduced from the outside during servicing or through inadequate filters, breathers, or seals. Internally generated particles are particularly destructive because they may become work-hardened during compression between the gear teeth. The following guidelines should be observed to prevent abrasive wear in gear units:

- Remove internal contamination from new gearboxes. Drain and flush the lubricant before initial start-up and again after 50 hours of operation. Refill with the manufacturer's recommended lubricant. Install new filters or breathers.

- Use surface-hardened gear teeth, smooth tooth surfaces, and high-viscosity lubricants.

- Maintain oil-tight seals and use filtered breather vents, preferably located in clean, nonpressurized areas.

- Use good housekeeping procedures.

- Use fine filtration for circulating-oil systems. Filtration to 3 µm (120 µ-in.) has proven effective in prolonging gear life.

- Unless otherwise recommended by the gear manufacturer, change the lubricant in oil-bath systems at least every 2500 hours or every 6 months.

! When warranted by the nature of the application, conduct laboratory analysis of lubricants. Analysis may include spectrographic, ferrographic, acid number, viscosity, and water content.

(3) Polishing. Polishing wear is characterized by a mirror-like finish of the gear teeth. Polishing is caused by antiscuff additives that are too chemically reactive. An excessive reaction rate, coupled with continuous removal of surface films by very fine abrasive particles in the lubricant, may result in excessive polishing wear. The following guidelines should be observed to prevent polishing wear in gearsets:

! Use less chemically active antiscuff additives such as borate.

! Remove abrasives from the lubricant by using fine filtration or by frequent oil changes.

*d. Scuffing.*

(1) General. The terms scuffing and scoring are frequently interchanged. The following definitions are provided to assist in correctly ascertaining the type of damage observed. The ASM Handbook Vol 18 defines scuffing as localized damage caused by the occurrence of solid-phase welding between sliding surfaces. It defines scoring as the formation of severe scratches in the direction of sliding. The handbook also stipulates that scoring may be caused by local solid-phase welding or abrasion, but suggests that minor scoring be considered as scratching. Gear scuffing is characterized by material transfer between sliding tooth surfaces. Generally this condition occurs when inadequate lubrication film thickness permits metal-to-metal contact between gear teeth. Without lubrication, direct metal contact removes the protective oxide layer on the gear metal, and the excessive heat generated by friction welds the surfaces at the contact points. As the gears separate, metal is torn and transferred between the teeth. Scuffing is most likely to occur in new gear sets during the running-in period because the gear teeth have not sufficient operating time to develop smooth surfaces.

(2) Critical scuffing temperature.

(a) Research has shown that for a given mineral oil without antiscuffing or extreme pressure additives, there is a critical scuffing temperature that is constant regardless of operating conditions. Evidence indicates that beyond the critical temperature, scuffing will occur. Therefore, the critical temperature concept provides a useful method for predicting the onset of scuffing. The critical scuffing temperature is a function of the gear bulk temperature and the flash temperature and is expressed as:

$$T_c = T_b + T_f \tag{9-1}$$

where the bulk temperature $T_b$ is the equilibrium temperature of the gears before meshing and the flash temperature $T_f$ is the instantaneous temperature rise caused by the local frictional heat at the gear teeth meshing point. The critical scuffing temperature for mineral oils without antiscuffing or extreme pressure additives increases directly with viscosity and varies from 150 to 300 °C (300 to 570 °F). However, this increased scuffing resistance appears to be directly attributed to differences in chemical composition and only indirectly to the beneficial effects of increased film thickness associated with higher viscosity. Examination of the critical temperature equation indicates that scuffing can be controlled by lowering either of the two contributing factors. The bulk temperature can be controlled by selecting gear geometry and design for the intended application. The flash temperature can be controlled indirectly by gear tooth smoothness and through lubricant viscosity. Smooth gear tooth surfaces produce less friction and heat while increased viscosity provides greater film thickness, which also reduces frictional heat and results in a lower flash temperature. Furthermore, judicious application of lubricant can cool the gears by removing heat.

(b) For synthetics and lubricants containing antiscuff additives, the critical temperature depends on the operating conditions and must be determined experimentally for each case. Antiscuff additives commonly used are iron sulfide and iron phosphate. These additives react chemically with the protected metal gear surface to form very strong solid films that prevent metal contact under extreme pressure and temperature conditions. As previously noted in the discussions of oil additives, the beneficial effects of extreme pressure additives are enhanced as the temperature increases.

(c) The following guidelines should be observed to prevent scuffing in gear units:

- Specify smooth tooth surfaces produced by careful grinding or honing.
- Protect gear teeth during the running-in period by coating them with iron-manganese phosphate or plating them with copper or silver. During the first ten hours of run-in, new gears should be operated at one-half load.
- Use high-viscosity lubricants with antiscuff additives such as sulfur, phosphorus, or borate.
- Make sure the gear teeth are cooled by supplying adequate amount of cool lubricant. For circulating-oil systems, use a heat exchanger to cool the lubricant.
- Optimize the gear tooth geometry. Use small teeth, addendum modification, and profile modification.
- Use accurate gear teeth, rigid gear mountings, and good helix alignment.
- Use nitrided steels for maximum scuffing resistance. Do not use stainless steel or aluminum for gears if there is a risk of scuffing.

## 9-4. Gear Lubrication

*a. Lubricant characteristics.* Gear lubricant must possess the following characteristics:

(1) General. The following characteristics are applicable to all gear lubricants. The lubrication requirements for specific gears follow this general discussion:

(a) Viscosity. Good viscosity is essential to ensure cushioning and quiet operation. An oil viscosity that is too high will result in excess friction and degradation of oil properties associated with high oil operating temperature. In cold climates gear lubricants should flow easily at low temperature. Gear oils should have a minimum pour point of 5 $^0$C (9 $^0$F) lower than the lowest expected temperature. The pour point for mineral gear oil is typically -7 $^0$C (20 $^0$F). When lower pour points are required, synthetic gear oils with pour points of -40 $^0$C (-40 $^0$F) may be necessary. The following equation from the ASM Handbook provides a method for verifying the required viscosity for a specific gear based on the operating velocity:

$$\nu_{40} = \frac{7000}{V^{0.5}} \tag{9-2}$$

where

$v_{40}$ = lubricant kinematic viscosity at 40°C (105°F) (cSt)

$V$ = pitch line velocity (ft/min) given by :

$$V = 0.262nd \tag{9-3}$$

where n is the pinion speed in rev/min and d is the pitch diameter (inches).

(b) Film strength. Good film strength helps prevent metal contact and scoring between the gear teeth.

(c) Lubricity (oiliness). Lubricity is necessary to reduce friction.

(d) Adhesion. Helps prevent loss of lubrication due to throw-off associated with gravity or centrifugal force especially at high speeds.

(e) Gear speed. The now superseded Industrial Gear Lubrication Standards, AGMA 250.04, used center distance as the primary criterion for gear lubricant selection. The new version of this standard, designated AGMA 9005-D94 Industrial Gear Lubrication, has adopted pitch line velocity as the primary selection criterion. As noted above, gear speed is a factor in the selection of proper oil viscosity. The pitch line velocity determines the contact time between gear teeth. High velocities are generally associated with light loads and very short contact times. For these applications, low-viscosity oils are usually adequate. In contrast, low speeds are associated with high loads and long contact times. These conditions require higher-viscosity oils. EP additives may be required if the loads are very high.

(f) Temperature. Ambient and operating temperatures also determine the selection of gear lubricants. Normal gear oil operating temperature ranges from 50 to 55 $^0$C (90 to 100 $^0$F) above ambient. Oils operating at high temperature require good viscosity and high resistance to oxidation and foaming. Caution should be exercised whenever abnormally high temperatures are experienced. High operating temperatures are indicative of oils that are too viscous for the application, excess oil in the housing, or an overloaded condition. All of these conditions should be investigated to determine the cause and correct the condition. Oil for gears operating at low ambient temperatures must be able to flow easily and provide adequate viscosity. Therefore these gear oils must possess high viscosity indices and low pour points.

(2) Open gears. In addition to the general requirements, lubrication for open gears must meet the following requirements:

(a) Drip resistance. Prevents loss of lubricant, especially at high temperatures which reduce viscosity.

(b) Brittle resistance. Lubricant must be capable of resisting embrittlement, especially at very low temperatures.

(3) Enclosed gears. In addition to the general requirements, lubrication for enclosed gears must meet the following requirements:

(a) Chemical stability and oxidation resistance. Prevents thickening and formation of varnish or sludge. This requirement is especially significant in high-speed gears because the oil is subjected to high operating oil and air temperatures.

(b) Extreme pressure protection. Provides additional galling and welding protection for heavily loaded gears when the lubricant film thickness fails. Extreme pressure lubricants are available for mild and severe (hypoid) lubricant applications.

*b. Types of gear lubricants*

(1) Oil. Refer to AGMA 9005-D94 for the specifications for the following lubricants.

(a) Rust and oxidation oils. These petroleum-based oils are frequently referred to as RO gear oils. RO oils are the most common gear lubricants and have been formulated to include chemical additives that enhance their performance qualities. RO lubricating oils have easy application properties for gear and bearings, good lubrication qualities, and adequate cooling qualities and they are economical to use. Disadvantages include restriction to enclosed gear applications to prevent contamination.

(b) Compounded gear lubricants. These oils are a blend of petroleum-based oils with 3 to 10 percent fatty or synthetic fatty oils. They are particularly useful in worm gear drives. Except as noted in the AGMA applicable specifications, compounded oils should comply with the same specifications as RO oils.

(c) Extreme pressure lubricants. These gear lubricants, commonly referred to as EP lubricants, are petroleum-based and specially formulated to include chemical additives such as sulfur-phosphorus or other similar materials capable of producing a film that provides extreme pressure and antiscuffing protection.

(d) Synthetic oils. Synthetic oils have the advantage of stable application over wide temperature range, good oxidation stability at high temperatures, high viscosity indices, and low volatility. Because gear oils must be changed periodically, the main disadvantage of synthetics is high cost, which can only be justified for applications at high temperature extremes where other lubricants fail. Another disadvantage of synthetics is possible incompatibility with seals and other lubricants. The equipment manufacturer should be consulted before using synthetic oils to ensure that exposed materials will not be damaged or warranties voided. Gear units should be flushed of all mineral oils before the filling with the final synthetic oil.

(e) Residual compounds. These are higher-viscosity straight mineral or EP lubricants that are mixed with a diluent to facilitate application. Viscosities range from 400 to 2000 $mm^2/s$ at 100 °C (mm sq/sec cST at 100 °C) without diluent. Once applied, the diluent evaporates and leaves a heavy residual lubricant coating on the treated surface.

(2) Special compounds and greases. These lubricants include special greases formulated for boundary lubricating conditions such as low-speed, low-load applications where high film strength is required. These lubricants usually contain a base oil, a thickener, and a solid lubricant such as molybdenum disulfide ($MoS_2$) or graphite. The gear manufacturer should be consulted before using grease. The primary disadvantage of using grease is that it accumulates foreign particles such as metal, dirt, and other loose materials that can cause significant damage if adequate maintenance is not provided.

Grease also has a tendency to be squeezed out of the gear tooth meshing zone, and it does not provide satisfactory cooling.

(3) Open-gear lubricants. Open-gear lubricants are generally reserved for slow-speed low-load boundary lubricating conditions. Due to the open configuration, the lubricants must be viscous and adhesive to resist being thrown off the gear teeth surfaces. The disadvantages of these lubricants are similar to those noted above for grease.

(4) Solid lubricants. The solid lubricants most commonly used in gear trains are molybdenum disulfide, graphite, polytetrafluoroethylene (PTFE), and tungsten disulfide ($WoS_2$). Because they are expensive to apply, use of these lubricants is reserved for special applications such as high and low temperature extremes where other lubricants fail to perform adequately.

*c. Applications.*

(1) Spur, helical, bevel, and hypoid gears. Spur, helical, and bevel gears have similar load and speed characteristics, and similar requirements for antiscuffing and viscosity.

(a) Spur and helical gears. Spur and helical gears usually require mineral oils with RO inhibitors. Low-viscosity RO oils, such as turbine oils, are commonly used in high-speed, low-load gear units. For high-speed, low-load gear applications, mineral oils without antiscuff/extreme pressure agents can be used successfully provided the oil viscosity is capable of maintaining the required film thickness. However, low-speed gears are usually heavily loaded so antiscuff/extreme pressure agents are necessary to ensure adequate protection.

(b) Hypoid gears. Hypoid gears combine the rolling action and high tooth pressure of spiral bevel gears with the sliding action of worm gears. These severe operating conditions result in high load, high sliding speeds, and high friction. Therefore, hypoid gears are very susceptible to scuffing. Mineral oils for this application must have high lubricity and high concentrations of antiscuffing/extreme pressure additives.

(2) Worm gears. Worm gears operate under high sliding velocity and moderate loads. The sliding action produces friction that produces higher operating temperatures than those that occur in other gear sets. Normal operating temperature for worm gears may rise to 93$^{o}$ C (200$^{o}$ F) and is not a cause for concern. Lubricants for worm gears must resist the thinning due to high temperatures and the wiping effect of sliding action, and they must provide adequate cooling. Mineral oils compounded with lubricity additives are recommended. Extreme pressure additives are usually not required for worm gears. However, when EP protection is required, the additive should be selected with caution to prevent damaging the bronze worm wheel.

(3) Gear combinations. Many applications use different gears in the same gear housing. For these applications the lubricant must be suitable for the gears with the most demanding requirements. Generally, the other gears will operate satisfactorily with such high-performance lubricants.

(4) Gear shaft bearings. Gear shaft bearings are frequently lubricated by gear oil. In most instances this condition is acceptable. Bearings in high-speed, low-load applications may operate satisfactorily with the gear oil. However, low-speed, heavily loaded gears usually require a heavy oil. For these conditions a low-viscosity EP oil may provide adequate lubrication for the gears and bearings. The low-viscosity oil

will adequately lubricate the bearings while the EP additive will protect the gear teeth from the effects of using a low-viscosity oil.

# Chapter 10
# Bearings

## 10-1. General

Bearings can be divided into two subgroups: plain bearings and rolling-contact bearings. Both have their place in the world of machines. Each type has some obvious advantages and disadvantages, but there are subtle properties as well that are often ignored. Each type of bearing can be found in a multiplicity of places, and each can be lubricated with either oil or grease. Some bearings are lubricated by water, and some are lubricated by air (as in the case of a dentist's drill).

## 10-2. Plain Bearings

Plain bearings consist of two surfaces, one moving in relation to the other. Plain bearings can be the journal type, where both wear surfaces are cylindrical; thrust type, where there are two planar surfaces, one rotating upon the other; and various types of sliding bearings where one surface slides in relation to the other. All depend upon a lubricating film to reduce friction. Unless an oil pump is provided to generate the oil film, these bearings rely on shaft motion to generate a hydrodynamic oil wedge.

*a. Advantages of plain bearings.*

(1) They have a very low coefficient of friction if properly designed and lubricated.

(2) They have very high load-carrying capabilities.

(3) Their resistance to shock and vibration is greater than rolling-contact bearings.

(4) The hydrodynamic oil film produced by plain bearings damps vibration, so less noise is transmitted.

(5) They are less sensitive to lubricant contamination than rolling-contact bearings.

*b. Types of plain bearings*.

(1) Journal (sleeve bearings). These are cylindrical with oil-distributing grooves. The inner surface can be babbitt-lined, bronze-lined, or lined with other materials generally softer than the rotating journal. On horizontal shafts on motors and pumps, oil rings carry oil from the oil reservoir up to the bearing. In the case of very slow-moving shafts, the bearings may be called bushings.

(2) Segmented journal. These are similar to the journal except that the stationary bearing consists of segments or bearing shoes. Each shoe is individually adjustable. This type of bearing is commonly found in vertical hydrogenerators and large vertical pumping units. This bearing is usually partially immersed in an oil tub.

(3) Thrust bearings. These bearings support axial loading and consist of a shaft collar supported by the thrust bearing, many times in segments called thrust shoes. The thrust shoes are sometimes allowed to pivot to accommodate the formation of the supporting oil wedges. There are many different configurations

of the thrust bearing aimed at equalizing loading and oil wedges. The bearing is immersed in a tub of oil. On large hydrogenerators and pumps an oil pump is sometimes used to provide an oil film at start-up.

(4) Self-lubricated bearings. These are journal (sleeve) bearings in which the bearing surface contains a lubricant, usually solid, that is liberated or activated by friction in the bearing. This type of bearing is gaining popularity as a wicket gate bearing or wicket gate linkage bushing.

*c. Plain bearing lubrication selection.*

(1) The most common lubricants for plain bearings are mineral and synthetic oils, and greases. Mineral oils are generally used except in extreme hot and cold temperature applications where synthetics provide superior performance. Oil is used for faster rotational speeds where the hydrodynamic oil wedge can be formed and maintained. It also is used in high-temperature conditions where grease may melt or degrade. Grease is used for slower rotational speeds or oscillating movements where the hydrodynamic oil wedge cannot form. It is also used in cases of extreme loading where the bearing operates in boundary conditions. Table 10-1 shows some of the important considerations regarding lubricant selection.

**Table 10-1**
**Choice of Lubricant**

| Lubricant | Operating Range | Remarks |
|---|---|---|
| Mineral oils | All conditions of load and speed | Wide range of viscosities available. Potential corrosion problems with certain additive oils (e.g., extreme pressure) (see Table 7.1). |
| Synthetic oils | All conditions if suitable viscosity available | Good high- and low-temperature properties. Costly. |
| Greases | Use restricted to operating speeds below 1 to 2 m/s (3.28 to 6.56 fps) | Good where sealing against dirt and moisture is necessary and where motion is intermittent. |
| Process fluids | Depends on properties of fluid | May be necessary to avoid contamination of food products, chemicals, etc. Special attention to design and selection of bearing materials. |

Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd., Oxford, England.

(2) The lubricating properties of greases are significantly affected by the base oil and type of thickeners used. Table 10-2 provides general guidelines for selecting the type of grease for bearing lubrications. In Table 10-2, speed factor (also referred to as speed index) is determined by multiplying the pitch diameter of the bearing by the bearing speed as follows;

$$d_n = n\,\frac{D + d}{2} \tag{10-1}$$

where $D$ is the bearing diameter (mm), $d$ is the bore diameter (mm), and $n$ is the rev/min. Speed factors above 200,000 are usually indicative of fluid film lubrication applications. The load column provides indications of the degree of loading on a bearing and is defined as the ratio of rated bearing load to the actual bearing load.

**Table 10-2**
**Bearing Lubrication Considering Speed Factor**

| Load, Rated Applied | Temperature, C° (°F) | Base Oil | Thickener | Additives |
|---|---|---|---|---|
| | | **Speed Factor Less than 100,000** | | |
| <10 | -56.6-17.7 (-70-0) | Mineral oil, synthetic, ester | Lithium | Graphite or $MoS_2$, rust oxidation |
| <10 | -17.7-176.6 (0-350) | Mineral oil | Lithium, calcium, barium, aluminum sodium | Graphite or $MoS_2$, rust oxidation |
| <10 | 176.6+ (350+) | Synthetic, ester | Sodium, clay, calcium, lithium, polyurea | Graphite or $MoS_2$, rust, oxidation |
| | | **Speed Factor 100,000 to 500,000** | | |
| <10 | -17.7-176.6 (0-350) | Mineral oil, synthetic, PAG, ester | Lithium, calcium, aluminum, barium, polyurea | Graphite or $Mos_2$, rust, oxidation |
| <10 (high)<br>>30 (low) | -17.7-176.6 (0-350) | Mineral oil, synthetic, PAG, ester | Lithium, calcium, aluminum, barium, polyurea | EP, rust, oxidation |
| >30 | 17-7-176.6 (0-350) | Mineral oil, synthetic, PAG, ester | Lithium, clay, polyurea, aluminum, barium, calcium | Antiwear, rust, oxidation |
| | | **Speed Factor Greater than 500,000** | | |
| >30 | -17.7-93.3 (0-200) | Mineral oil, synthetic, ester | Lithium, calcium, barium | Rust, oxidation |

(3) Viscosity is the most critical lubricant property for insuring adequate lubrication of plain bearings. If the viscosity is too high, the bearings will tend to overheat. If the viscosity is too low the load-carrying capacity will be reduced. Figure 10-1 is a guide to selection of viscosity for a given operating speed. For plain journal bearings the surface speed u is given by:

$$u = \pi dn, \quad \text{m/sec} \qquad (10\text{-}1)$$

and the mean pressure $p_m$ is given by

$$P_m = \frac{W}{ld}, \ kN/m^2 \qquad (10\text{-}2)$$

where

$n$ = shaft speed, rev/s

l = bearing width, m

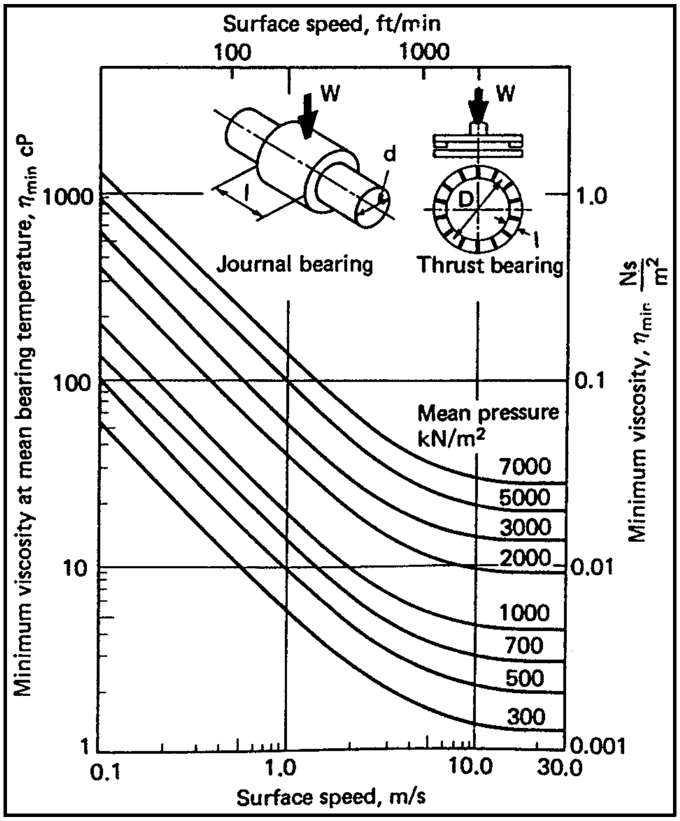


**Figure 10-1. Lubricant viscosity for plain bearings (Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd., Oxford England)**

$d$ = shaft diameter, m

$W$ = thrust load, kN

For thrust bearings, the surface speed $u$ is given by Equation 10-1. The mean pressure is given by

$$p_m = \frac{0.4W}{lD}, \quad kN/m^2 \qquad (10\text{-}3)$$

where $p_m$, and l are as previously defined, $W$ = thrust load, $kN$, and $D$ = mean pad diameter, m. Equations 10-1 through 10-3 are intended to provide a means for understanding Figures 10-1 and 10-2. Refer to Machinerys Handbook, 24th edition, for a detailed discussion and analysis of bearing loads and lubrication.) Figure 10-2 shows the relationship between temperature and viscosity for mineral oils.

(4) Table 10-3 identifies some of the methods used to supply lubricants to bearings. The lubricant should be supplied at a rate that will limit the temperature rise of the bearing to 20°C (68 °F).

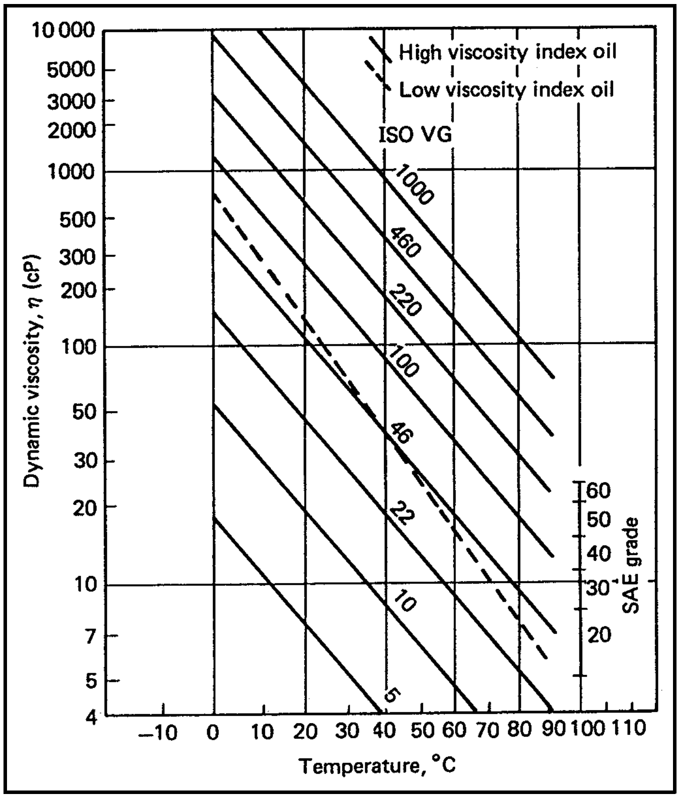


**Figure 10-2. Typical viscosity/temperature characteristics of mineral oils (Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)**

**Table 10-3**
**Methods of Liquid Lubricant Supply**

| Method of Supply | Main Characteristics | Examples |
|---|---|---|
| Hand oiling | Nonautomatic, irregular. Low initial cost. High maintenance cost. | Low-speed, cheap journal bearings |
| Drip and wick feed | Nonautomatic, adjustable. Moderately efficient. Cheap | Journals in some machine tools, axles |
| Ring and collar feed | Automatic, reliable. Efficient, fairly cheap. Mainly horizontal bearings | Journals in pumps, blowers, large electric motors |
| Bath and splash lubrication | Automatic, reliable, efficient. Oiltight housing required. High initial cost. | Thrust bearings, bath only. Engines, process machinery, general |
| Pressure feed | Automatic. Positive and adjustable. Reliable and efficient. High initial cost. | High-speed and heavily loaded journal and thrust bearings in machine tools, engines, and compressors |

Notes:
Pressure oil feed: This is usually necessary when the heat dissipation of the bearing housing and its surroundings are not sufficient to restrict its temperature rise to 20 °C (68 °F) or less.
Journal bearings: Oil must be introduced by means of oil grooves in the bearing housing.
Thrust bearings: These must be lubricated by oil bath or by pressure feed from the center of the bearing.
Cleanliness: Cleanliness of the oil supply is essential for satisfactory performance and long life.
(Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)

(5) Generally, oil additives such as those noted in Table 7-1 are not required in plain bearing applications. Some additives and contaminants may cause corrosion, so caution should be exercised when using bearing lubricants containing additives or when contaminants may be present. Table 10-4 identifies some of the most common bearing materials used, and their resistance to corrosion when subjected to the additives noted.

**Table 10-4**
**Resistance to Corrosion of Bearing Metals**

| | Maximum Operating Temperature, °C (F°) | Additive or Contaminant | | | | |
|---|---|---|---|---|---|---|
| | | Extreme-Pressure Additive | Antioxidant | Weak Organic Acids | Strong Mineral Acids | Synthetic Oil |
| Lead-base white metal | 130 (266) | Good | Good | Moderate/poor | Fair | Good |
| Tin-base white metal | 130 (266) | Good | Good | Excellent | Very good | Good |
| Copper-lead (without overlay) | 170 (338) | Good | Good | Poor | Fair | Good |
| Lead-bronze (without overlay) | 180 (356) | Good with good quality bronze | Good | Poor | Moderate | Good |
| Aluminum-tin alloy | 170 (338) | Good | Good | Good | Fair | Good |
| Silver | 180 (356) | Sulfur-containing additives must not be used | Good | Good - except for sulfur | Moderate | Good |
| Phosphor-bronze | 220 (428) | Depends on quality of bronze. Sulfurized additives can intensify corrosion. | Good | Fair | Fair | Good |
| Copper-lead or lead-bronze with suitable overlay | 170 (338) | Good | Good | Good | Moderate | Good |

Note:
Corrosion of bearing metals is a complex subject. The above offers a general guide. Special care is required with extreme-pressure lubricants; if in doubt refer to bearing or lubricants supplier.
(Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)

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## 10-3. Rolling-Contact Bearings

In rolling-contact bearings, the lubricant film is replaced by several small rolling elements between an inner and outer ring. In most cases the rolling elements are separated from each other by cages. Basic varieties of rolling-contact bearings include ball, roller, and thrust.

*a. Advantages of rolling-contact bearings.*

(1) At low speeds, ball and roller bearings produce much less friction than plain bearings.

(2) Certain types of rolling-contact bearings can support both radial and thrust loading simultaneously.

(3) Rolling bearings can operate with small amounts of lubricant.

(4) Rolling-contact bearings are relatively insensitive to lubricant viscosity.

(5) Rolling-contact bearings have low wear rates and require little maintenance.

*b. Types of rolling-contact bearings*.

(1) Ball bearing. This bearing has spherical rolling elements in a variety of configurations. It is able to carry both radial and moderate axial loads. A special type, called maximum-type ball bearings, can take an extra 30 percent radial load but cannot support axial loads.

(2) Roller bearing. The roller bearing has cylindrical rolling elements and can take much higher radial loads than ball bearings but can carry no axial loads.

(3) Tapered roller bearing. This type has truncated-cone shaped rolling elements and is used for very high radial and thrust loads.

(4) Double-row spherical. The bearing has a double row of keg-shaped elements. The inner surface of the outer race describes part of a sphere. This bearing can handle thrust in both directions and very high radial loads.

(5) Ball thrust. This type has ball elements between grooved top and bottom races.

(6) Straight roller thrust. This bearing has short segments of cylindrical rollers between upper and lower races. The rollers are short to minimize skidding.

(7) Spherical thrust. This type is also called a tapered roller thrust bearing. The lower race describes part of a sphere. The rolling elements are barrel-shaped and the outside has a larger diameter than the inside.

(8) Needle bearing. These bearings have rollers whose lengths are at least four times their diameter. They are used where space is a factor and are available with or without an inner race.

*c. Rolling-contact conditions.* The loads carried by the rolling elements actually cause elastic deformation of the element and race as rotation occurs. The compressive contact between curved bodies results in maximum stresses (called Hertzian contact stresses) occurring inside the metal under the surfaces involved. The repeated stress cycling causes fatigue in the most highly stressed metal. As a result, normal wear of rolling contact bearings appears as flaking of the surfaces. Lubrication carries away the excessive heat generated by the repeated stress cycles. While lubrication is necessary, too much lubrication--especially with grease lubrication--results in churning action and heating due to fluid friction.

*d. Rolling bearing lubricant selection.* In most cases, the lubricant type--oil or grease--is dictated by the bearing or equipment manufacturer. In practice, there can be significant overlap in applying these two types of lubricant to the same bearing. Often the operating environment dictates the choice of lubricant. For example, a roller bearing on an output shaft of a gearbox will probably be oil-lubricated because it is contained in an oil environment. However, the same bearing with the same rotational speed and loading would be grease-lubricated in a pillow block arrangement.

(1) Selection of lubricant. Table 10-5 provides general guidance for choosing the proper lubricant.

**Table 10-5**
**General Guide for Choosing Between Grease and Oil Lubrication**

| Factor Affecting the Choice | Use Grease | Use Oil |
|---|---|---|
| Temperature | Up to 120 °C (248 °F) - with special greases or short relubrication intervals up to 200/220 °C (392/428 °F) | Up to bulk oil temperature of 90 °C or bearing temperature of 200 °C (428 °F) - These temperatures may be exceeded with special oils. |
| Speed factor* | Up to *dn* factors of 300,000/350,000 (depending on design) | Up to *dn* factors of 450,000/500,000 (depending on type of bearing) |
| Load | Low to moderate | All loads up to maximum |
| Bearing design | Not for asymmetrical spherical roller thrust bearings | All types |
| Housing design | Relatively simple | More complex seals and feeding devices necessary |
| Long periods without attention | Yes, depending on operating conditions, especially temperature | No |
| Central oil supply for other machine elements | No - cannot transfer heat efficiently or operate hydraulic systems | Yes |
| Lowest torque | When properly packed can be lower than oil on which the grease is based | For lowest torques use a circulating system with scavenge pumps or oil mist |
| Dirty conditions | Yes - proper design prevents entry of contaminants | Yes, if circulating system with filtration |

* *dn* factor (bearing bore (mm) x speed (rev/min)).
Note: For large bearings (0.65-mm bore) and $nd_m$ ($d_m$ is the arithmetic mean of outer diameter and bore (mm)).
(Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)

(2) Grease.

(a) Grease is used for slower rotational speeds, lower temperatures, and low to medium loads. Grease is used in situations where maintenance is more difficult or irregularly scheduled. It can be used in dirty environments if seals are provided. Tables 10-6 and 10-7 provide guidance on method of application and environmental considerations when using grease.

**Table 10-6**
**Effect of Method of Application on Choice of a Suitable Grade of Grease**

| System | NLGI Grade No. |
|---|---|
| Air pressure | 0 to 2 depending on type |
| Pressure-guns or mechanical lubricators | Up to 3 |
| Compression cups | Up to 5 |
| Centralized lubrication | 2 or below |
| (a) Systems with separate metering values | Normally 1 or 2 |
| (b) Spring return systems | 1 |
| (c) Systems with multidelivery pumps | 3 |

Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England.

**Table 10-7**
**Effect of Environmental Conditions on Choice of a Suitable Type of Grease**

| Type of Grease | NLGI Grade No. | Speed Maximum (percentage recommended maximum for grease) | Environment | Typical Service Temperature Maximum °C | Maximum °F | Minimum °C | Minimum °F | Base Oil Viscosity (approximate values) | Comments |
|---|---|---|---|---|---|---|---|---|---|
| Lithium | 2 | 100<br>75 | Wet or dry | 100<br>135 | 210<br>275 | -25 | -13 | Up to 140 cSt at 37.7 °C (100 °F) | Multipurpose, not advised at max. speed or max. temperatures for bearings above 65-mm (2.5-in.) bore or on vertical shafts |
| Lithium | 3 | 100<br>75 | Wet or dry | 100<br>135 | 210<br>275 | -25 | -13 | | |
| Lithium EP | 1 | 75 | Wet or dry | 90 | 195 | -15 | 5 | | Recommended for roll-neck bearings and heavily-loaded taper-roller bearings |
| Lithium EP | 2 | 100<br>75 | Wet or dry | 70<br>90 | 160<br>195 | <br>-15 | <br>5 | 14.5 cSt at 98.8 °C (210 °F) | |
| Calcium (conventional) | 1, 2, and 3 | 50 | Wet or dry | 60 | 140 | -10 | 14 | 140 cSt at 37.7 °C (100 °F) | |
| Calcium EP | 1 and 2 | 50 | Wet or dry | 60 | 140 | -5 | 25 | 14.5 cSt at 98.8 °C (210 °F) | |
| Sodium (conventional) | 3 | 75/100 | Dry | 80 | 175 | -30 | -22 | 30 cSt at 37.7 °C (100 °F) | Sometimes contains 20% calcium |
| Clay | | 50 | Wet or dry | 200 | 390 | 10 | 50 | 550 cSt at 37.7 °C (100 °F) | |
| Clay | | 100 | Wet or dry | 135 | 275 | -30 | -22 | Up to 140 cSt at 37.7 °C (100 °F) | |
| Clay | | 100 | Wet or dry | 120 | 248 | -55 | -67 | 12 cSt at 37.7 °C (100 °F) | Based on synthetic esters |
| Silicone/ lithium | | 75 | Wet or dry | 200 | 390 | -40 | -40 | 150 cSt at 25 °C (77 °F) | Not advised for conditions where sliding occurs at high speed or load |

Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England

(b) Figure 10-3 shows approximate maximum bearing speeds for grease-lubricated bearings based on the bore diameter and series of grease. Figure 10-4 provides guidance on grease life expectancy for various operating speeds (given in percent) as a function of temperature. Correction factors for use with Figure 10-3 are shown in Table 10-8.

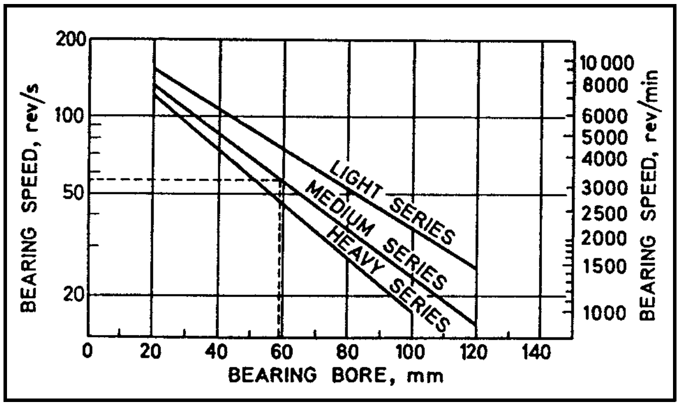


**Figure 10-3. Approximate maximum speeds for grease-lubricated bearings ((Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)**

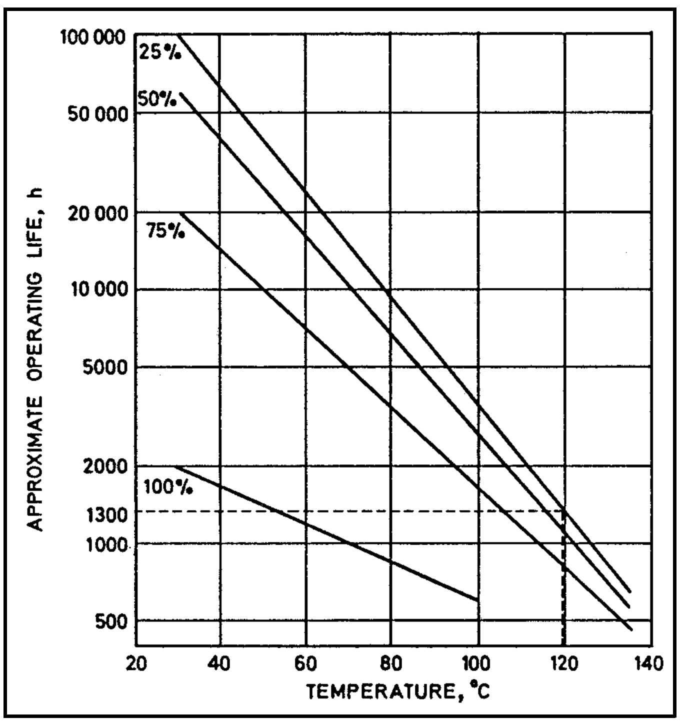


**Figure 10-4. Variation of operating life of Grade 3 lithium hydroxystearate grease with speed and temperature (Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)**

**Table 10-8**
**Correction Factors for Figure 10-3**

| Bearing Type | | Multiply Bearing Speed from Figure 10-1 by This Factor to Get the Maximum Speed for Each Type of Bearing |
|---|---|---|
| Ball bearings and cylindrical roller bearings | Cage centered on inner race | As Figure 10-1 |
| | Pressed cages centered on rolling elements | 1.5-1.75 |
| | Machined cages centered on rolling elements | 1.75-2.0 |
| | Machined cages centered on outer race | 1.25-2.0 |
| Taper- and spherical-roller bearings | | 0.5 |
| Bearings mounted in adjacent pairs | | 0.75 |
| Bearings on vertical shafts | | 0.75 |
| Bearings with rotating outer races and fixed inner races | | 0.5 |

Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England

(3) Oil.

(a) Oil is used for higher rotational speeds and higher operating temperatures. It is used in maximum loading situations and for bearing configurations where a high amount of heat generated in the bearing can be carried away by the oil. It is used in dirty conditions when the oil is circulated and filtered. For moderate speeds, the following viscosities are recommended:

| | |
|---|---|
| Ball and cylindrical-roller bearings | 12 cSt |
| Spherical-roller bearings | 20 cSt |
| Spherical-roller thrust bearings | 32 cSt |

(b) In general, oils will be the medium to high viscosity index type with rust and oxidation inhibitors. Extreme pressure (EP) oils are required for taper-roller or spherical-roller bearings when operating under heavy loads or shock conditions. Occasionally EP oils may be required by other equipment or system components.

(c) Figure 10-5 provides a means for selecting bearing oil lubricant viscosity based on the bearing operating temperature, bore diameter, and speed. The following example shows how to use this figure. Assume a bearing bore diameter of 60 mm (2.3 in.), speed of 5000 rev/min and an operating temperature of 65 °C. To select the viscosity, locate the bore diameter then move vertically to the required speed. At this intersection move left to intersect the operating temperature. Since the required viscosity falls between an S8 and S14 oil, select the oil with the higher viscosity (S14). The correct oil selection has a viscosity of 14 cSt at 50 °C. Table 10-9 provides guidance on applying oil to roller bearings.

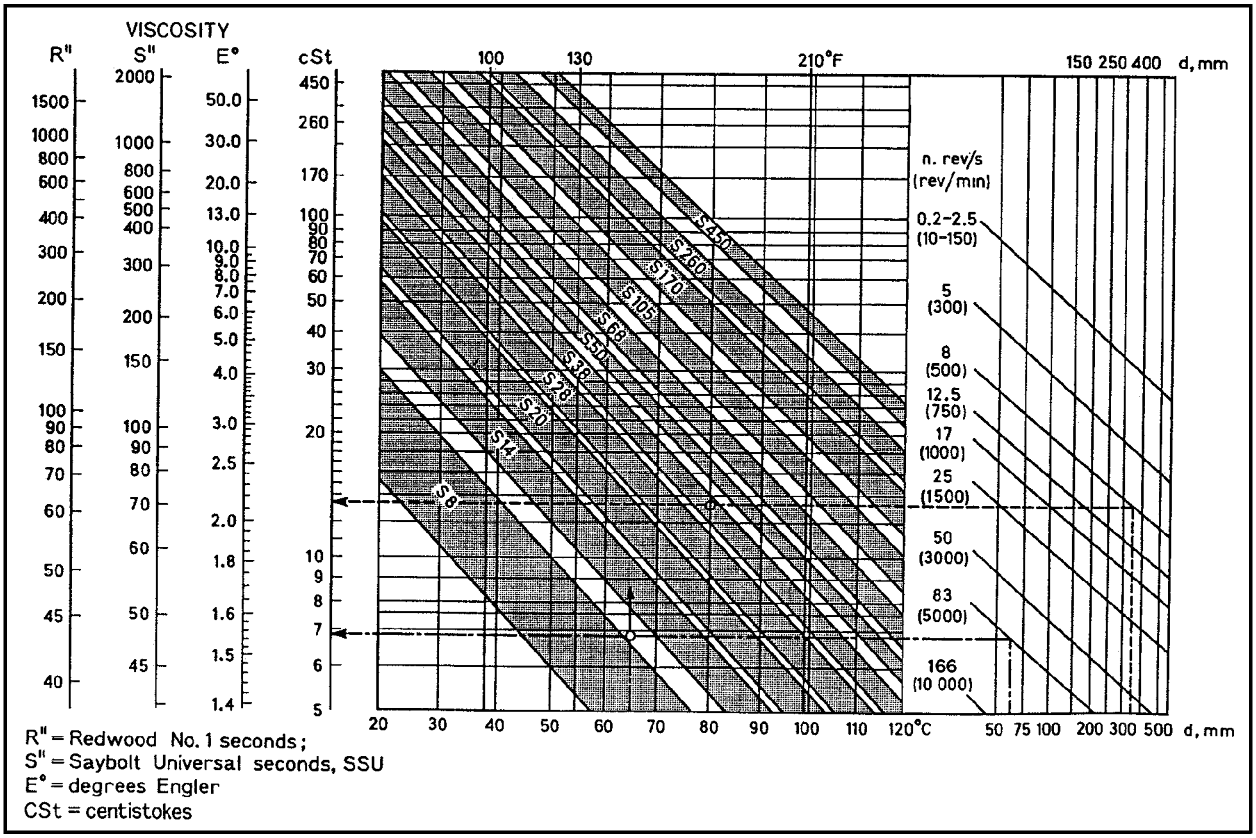


**Figure 10-5. Roller bearing oil selection (Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England)**

## 10-4 Calculation of Bearing Lubrication Interval

The following procedure for calculating lubrication intervals is extracted from Neale, “Lubrication: A Tribology Handbook” (see Appendix A for complete reference). The required interval is calculated using the data from Figures 10-3 and 10-4. The following example illustrates the procedure:

1. Given:

| | |
|---|---|
| Bearing type: | Medium series with 60-mm bore diameter |
| Cage: | Pressed cage centered on balls |
| Speed: | 950 rev/min |
| Temperature: | 120 °C (bearing temperature) |
| Position: | Vertical shaft |
| Grease: | Lithium grade 3 |
| Duty: | Continuous |

2. Determine the lubrication interval.

3. Procedure:

**Table 10-9**
**Application of Oil to Roller Bearings**

| System | Conditions | Oil Levels/Oil Flow Rates | Comments |
|---|---|---|---|
| Bath/splash | Generally used where speeds are low<br><br>A limit in *dn* value of 100,000 is sometimes quoted, but higher values can be accommodated if churning is not a problem. | Bearings on horizontal and vertical shafts, immerse half lowest rolling element<br><br>Multirow bearings on vertical shafts, fully immerse bottom row of elements | |
| Oil flingers, drip-feed lubricators, etc. | Normally as for bath/splash | Flow rate dictated by particular application; ensure flow is sufficient to allow operation of bearing below desired or recommended maximum temperature - generally between 70 °C and 90 °C (158 °F and 194 °F) | Allows use of lower oil level if temperature rise is too high with bath/splash |
| Pressure circulating | No real limit to *dn* value<br><br>Use oil mist where speeds are very high | As a guide, use: *0.6 $cm^3$/min $cm^2$ of projected area of bearing (o.d. x width) | The oil flow rate has generally to be decided by consideration of the operating temperature |
| Oil mist | No real limit to *dn* value<br><br>Almost invariably used for small-bore bearings above 50,000 rev/min, but also used at lower speeds | As a guide use: * 0.1 to 0.3 x bearing bore (cm/2.54) x no. of rows - $cm^3$/hour<br><br>Larger amounts are required for pre-loaded units, up to 0.6 x bearing bore (cm/2.54) x no. of rows - $cm^3$/hour | In some cases oil-mist lubrication may be combined with an oil bath, the latter acting as a reserve supply which is particularly valuable when high-speed bearings start to run |

* It must be emphasized that values obtained will be approximate and that the manufacturer's advice should be sought on the performance of equipment of a particular type.
Reference: Neale, M. J., Lubrication: A Tribology Handbook. Butterworth-Heinemann Ltd, Oxford, England.

*a*. From Figure 10-3, determine the speed for a 60-mm bore medium series bearing (3100 rev/min).

*b*. Maximum speed correction factor for cage centered bearing from Table 10.8. (1.5).

*c*. Maximum speed = 1.5 x 3100 = 4650 rev/min.

*d*. Obtain correction factor for vertical shaft mounting from Table 10.8 (0.75).

*e*. Corrected speed = 0.75 x 4650 rev/min = 3,488 rev/min (this is the maximum speed rating, i.e., 100 %).

*f*. Percent of actual speed to maximum speed = 100 x [950/ 3488] = 27 %.

*g*. Refer to Figure 10-4. Using 120 °C and the 25 % line, obtain the estimated operating life = 1300 hours.