5. LUBRICATING YOUR BEARINGS

A. Basic lubricant functions

Proper lubrication is essential to successful performance of any bearing. Making the best bearing selection includes considering the type of lubricant, the right amount of lubricant and the correct application of the lubricant on the bearing.

The three fundamental functions of a lubricant are as follows:
- To separate mating surfaces and reduce friction
- To transfer heat (with oil lubrication)
- To protect from corrosion and, with grease lubrication, from dirt ingress.

These functions include consideration of the lubrication and generated film thickness on the raceway (simulated according to elastohydrodynamic effects) and on rib/roller end contact.

1. Elastohydrodynamic lubrication

The formation of the lubricant film between the mating bearing surfaces is called the elastohydrodynamic (EHD) mechanism of lubrication (fig. 5-1). The two major considerations in EHD lubrication are:
- The elastic deformation of the contacting bodies under load
- The hydrodynamic effects forcing the lubricant film to separate the contacting surfaces while the pressure is deforming them.

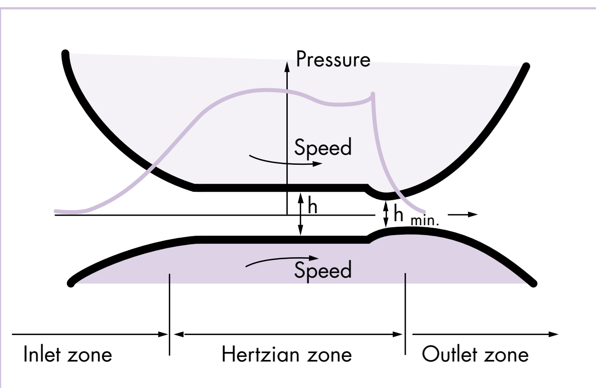

Fig. 5-1
Elastohydrodynamic (EHD) lubrication.

2. Film thickness on the raceway

EHD lubrication is important because lubricant film thickness between the two contacts can be related to the bearing performance. The thickness of the generated film depends on the operating conditions such as:

- Velocity
- Loads
- Lubricant viscosity
- Pressure/viscosity relationship.

Analytical relationships for calculating the minimum and the average film thickness have been developed:

Minimum film thickness (based on Dowson Equation):

$$h_{min} = K_D (\mu_o V)^{0.7} \alpha^{0.54} W^{-0.13} R^{0.43}$$

where:

h_{min} = minimum lubricant film thickness
K_D = constant containing moduli of elasticity
μ_o = lubricant viscosity at atmospheric pressure
V = relative surface velocity
α = lubricant pressure viscosity coefficient
W = load per unit length
R = equivalent radius

Average film thickness (based on Grubin Equation):

$$h = 0.039 (\mu V \alpha)^{0.728} (P/l)^{-0.091} (\Sigma 1/R)^{0.364}$$

where:

h = lubricant film thickness (µm)
μ = viscosity of lubricant
V = surface velocity
α = lubricant pressure viscosity coefficient
P = load between inner race and rollers
l = effective length contact between rollers and inner race
$\Sigma 1/R$ = sum of inverses of contact radii

The major factors influencing the lubricant film thickness are viscosity and speed. Load has less importance.
These thin EHD films are often not much larger than the surface roughness height.

Test group	Temperature		Viscosity @ test temperature	EHD Film (h_{min})		Life %
	°C	°F	mm^2/sec (cSt)	µm	µin	
A - 1	135	275	2.0	0.038	1.5	13 - 19
A - 2	66	150	19.4	0.264	10.4	100

Table 5-A
Relative bearing fatigue life vs EHD film thickness (Constant speed - variable temperature).

As shown in Section 3 "Calculating the performance of your bearings", the fatigue life of a bearing is related in a complex way to speed, load, lubrication, temperature, setting and alignment. The lubricant's role in this interaction is determined primarily by speed, viscosity and temperature. The effects of these factors on bearing fatigue life can be dramatic. In a test program, table 5-A, two bearing test groups were subjected to identical conditions of speed and load. Differing film thicknesses were achieved by varying operating temperature and oil grade, and thereby, oil operating viscosity. Life was dramatically reduced at higher temperatures with lower viscosity and thinner film thickness.

In another investigation, table 5-B, viscosity and load were held constant, but speed was varied producing results similar to those in table 5-A - higher speeds produce thicker film and longer lives.

The selection of the correct lubricant for any application requires careful study of expected operational and environmental conditions. Equations given in Section 3 permit the evaluation of the lubricant and these conditions relative to fatigue life in the form of a life adjustment factor for lubrication. The calculated L_{10} life is multiplied by this factor to obtain a life adjusted for lubrication effects.

Test group	Speed rev/min	EHD Film (h_{min}) µm	EHD Film (h_{min}) µin	Life %
B - 1	3600	0.102	4.0	100
B - 2	600	0.028	1.1	40

Table 5-B
Relative bearing fatigue life vs EHD film thickness.
(Variable speed - constant temperature)

3. Film thickness at rib/roller end contact

The contact between the large end of the roller and the inner race rib is described as an elastohydrodynamic contact or a hydrodynamic contact (elastic deformations are negligible). As the roller/rib loads are much lower than the roller/race loads, the film thickness at the rib/roller end contact is usually larger than the film thickness on the roller/race contact (approximately 2 times more).

Nevertheless, in severe conditions, scoring and/or welding of the rib/roller asperities can occur. This is related to speed, oil viscosity, load or inadequate oil supply to the rib/roller end contact. In these conditions, the use of Extreme Pressure (EP) lubricant additives may help prevent scoring damage.

B. Speed capabilities

The speed capability of a bearing in any application is subject to a number of factors including:

- Temperature
- Bearing setting
- Lubrication
- Bearing design

The relative importance of each of these factors depends on the nature of the application. The effect of each factor is not isolated – each contributes in varying degrees, depending on the application and overall speed capability of the design.

An understanding of how each of these factors affects performance as speeds change is required to achieve the speed capabilities inherent in tapered roller bearings.

1. Measuring speed

The usual measure of the speed of a tapered roller bearing is the circumferential velocity at the midpoint of the inner race large end rib (fig. 5-2), and this may be calculated as:

Rib speed:

$$V_r = \frac{\pi D_m n}{60\,000} \quad \text{(m/s)}$$

$$= \frac{\pi D_m n}{12} \quad \text{(ft/min)}$$

where:

D_m = Inner race rib diameter mm, in
n = Bearing speed rev/min

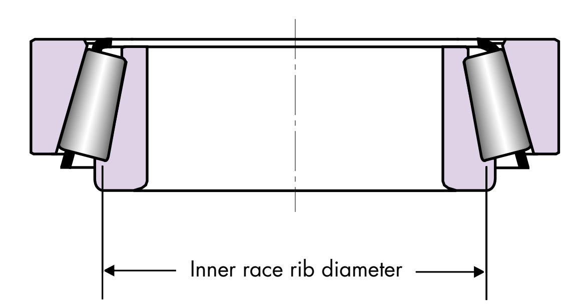

Fig. 5-2
Cone rib diameter. The inner race rib diameter may be scaled from a print.

The rib diameter at the midpoint of the roller end contact can be scaled from a drawing of the bearing, if available, or this diameter can be determined by contacting a Timken sales engineer or representative. The inner cone rib diameter can be approximated by taking 99% of larger rib OD.

DN values (the product of the inner race bore in mm and the speed in rev/min) are often used as a measure of bearing speed by other bearing manufacturers. There is no direct relationship between the rib speed of a tapered roller bearing and DN value because of the wide variation in bearing cross sectional thickness. However, for rough approximation, one meter per second rib speed is about equal to 16,000 DN for average section bearings. One foot per minute is equal to approximately 80 DN.

2. Speed capability guidelines

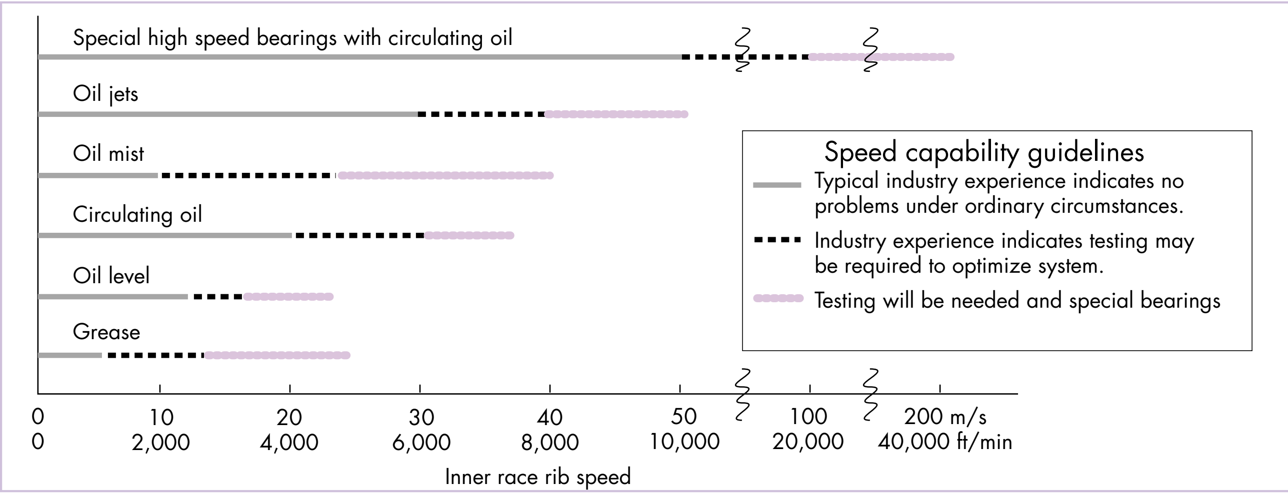

Fig. 5-3
Speed capability guidelines for various types of lubrication systems.

Fig. 5-3 is a summary of guidelines relating to speed and temperature based on customer experience, customer tests and research conducted by The Timken Company. Contact The Timken Company for questions regarding high speed capability.

3. Bearing material limitations

Standard bearing steel can operate continuously at temperatures up to 135°C (250°F) for extended periods of time without altering the hardness of the steel. For higher operating temperatures, special high temperature steels are available. Contact your Timken sales engineer or representative.

C. Guidance for oil/grease selection

1. Grease lubrication

The simplest lubrication system for any bearing application is grease. Lubricating grease as defined by the National Lubricating Grease Institute (NLGI) is "a solid to semi-fluid product of dispersion of a thickening agent in a liquid lubricant". Conventionally, greases used in Timken bearing applications are petroleum oils of some specific viscosity that are thickened to the desired consistency by some form of metallic soap. Greases are available in many soap types such as sodium, calcium, lithium, calcium-complex, and aluminium-complex. Organic and inorganic type non-soap thickeners are also used in some products.

1.1. Soap type

Calcium greases have good water resistance. Sodium greases generally have good stability and will operate at higher temperatures, but they absorb water and cannot be used where moisture is present. Lithium, calcium-complex and aluminium-complex greases generally combine the higher temperature properties and stability of sodium grease with the water resistance of calcium grease. These greases are often referred to as "multipurpose" greases since they combine the two most important lubricant advantages into one product.

TM = Trademark of The Timken Company

1.2. Consistency

The consistency (hardness) of a grease is defined as the depth (in tenths of a millimeter) to which a standard testing cone penetrates a grease sample under prescribed conditions. The NLGI has classified greases, table 5-C, according to consistency as measured by penetration.

NLGI number	Penetration range
000	445 - 475
00	400 - 430
0	355 - 385
1	310 - 340
2	265 - 295
3	220 - 250
4	175 - 205
5	130 - 160
6	85 - 115

Table 5-C
NLGI classification of greases.

Tapered roller bearings generally use NLGI No.2 or No.1 greases. No.3 and heavier greases are seldom used in Timken bearings because they tend to channel, causing lubricant starvation. NLGI No.0 or softer grease will circulate readily within the bearing chamber which accelerates softening and possible leakage.

1.3. Filling practices

When grease lubrication is chosen, the grease should be packed into the bearing making sure that it goes between the rollers and cage. When hand packing bearings, force grease through the bearing under the cage from the large end to the small end to ensure proper grease distribution.

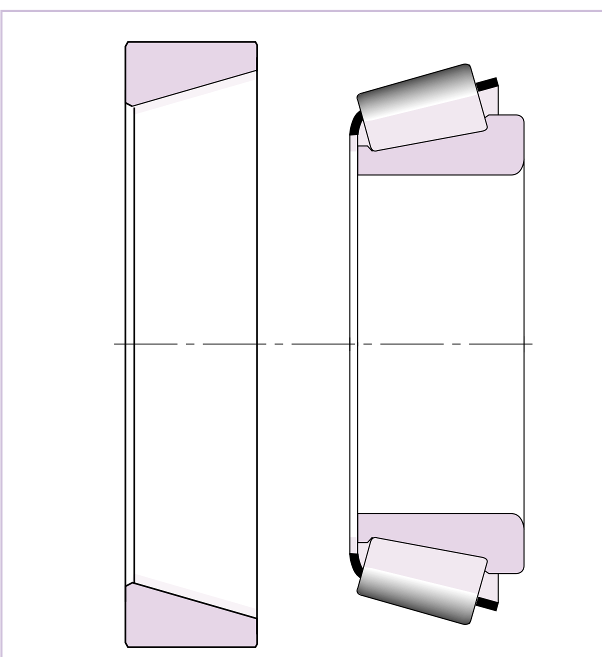

Fig. 5-4
Grease application at assembly.

A bearing completely filled with grease will purge itself of the excess when rotation starts. If provision is not made for grease exit from the cavity, the churning of the grease could cause excessive heat generation if rotational speeds are high. At initial assembly, it is advisable to smear grease on to the outer race. The area between the cage and the inner race should be filled (fig. 5-4). It may be advantageous to use internal closures to keep purged grease in the vicinity of the bearing (fig. 5-5). There is a separate Timken Company publication regarding workshop fitting practice, describing assembly procedures in more detail and showing various proprietary equipment to inject correct grease quantities into bearings automatically.

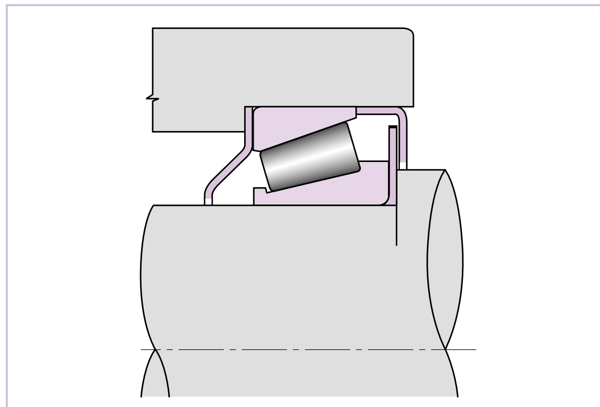

Fig. 5-5
Closures to keep purged grease in the vicinity of the bearing.

1.4. Required grease quantities

To ensure optimum lubrication, the required quantity of grease has to be packed into the bearing. This quantity is based on the volume of open space in the bearing calculated as follows:

$$V = \left[\frac{\pi}{4}(D^2 - d^2)\left(\frac{T + C}{2}\right)\right] - \frac{M}{7.85} \quad \text{(in cm}^3\text{)}$$

where:

V = volume of open space in the bearing
D = outer race O.D. (cm)
d = inner race bore (cm)
T = overall width (cm)
C = outer race width (cm)
M = bearing weight (g)

Depending on the application type, the quantity to fill the bearing will be approximately:

- 2/3 of V for conventional mineral grease
- 1/3 of V for synthetic grease.

To determine the corresponding weight of grease, approximate the grease density to 0.9 g/cm^3.

If speeds are very low or there is a dirty environment, it is suggested that the housing into which the bearing is fitted is completely filled. In high speed applications, overgreasing will generate excessive heat which can lead to lubricant degradation and bearing damage.

1.5. Regreasing cycle

The two primary considerations that determine the regreasing cycle on any application are operating temperature and sealing efficiency. Obviously, seal leakage will dictate frequent relubrication. Every attempt should be made to maintain seals at peak efficiency. It is generally stated that the higher the temperature, the more rapidly the grease oxidizes. Grease life is reduced by approximately half for every 10°C rise in temperature.
Therefore, the higher the operating temperature, the more often the grease must be replenished. In most cases, experience in the specific application will dictate the frequency of lubrication.

1.6. Typical grease lubrication guidelines

1.6.1. General purpose industrial grease

These are typical of greases that can be used to lubricate many Timken bearing applications in all types of standard equipment. Special consideration should be given to applications where speed, load, temperature or environmental conditions are extreme.

Suggested general purpose industrial grease properties

Soap type	Lithium 12-hydroxystearate, or equivalent
Consistency	NLGI No. 2
Additives	Corrosion and oxidation inhibitors
Base oil	Solvent refined petroleum oil
Base oil viscosity at 40° C	100 mm^2/s (cSt) to 320 mm^2/s (cSt)
Viscosity index	80 min.
Pour point	1–10°C max.

General purpose industrial grease should be a smooth, homogeneous and uniform premium-quality product composed of petroleum oil, a thickener, and appropriate inhibitors. It should not contain materials that are corrosive or abrasive to tapered roller bearings. The grease should have excellent mechanical and chemical stability and should not readily emulsify with water. The grease should contain inhibitors to provide long-term protection against oxidation in high-performance applications and protect the bearings from corrosion in the presence of moisture.
The suggested base oil viscosity covers a fairly wide range. Lower viscosity products should be used in high-speed and/or lightly loaded applications to minimize heat generation and torque. Higher viscosity products should be used in moderate to low-speed applications and under heavy loads to maximize lubricant film thickness.

2. Oil lubrication

Lubricating oils are commercially available in many forms for automotive, industrial, aircraft and other uses. Oils are classified as either petroleum types (refined from crude oil) or synthetic types (produced by chemical synthesis).

2.1. Petroleum oils

Petroleum oils are used for nearly all oil-lubricated applications of Timken bearings. These oils have physical and chemical properties that can help select the correct oil for any bearing application.

2.2. Synthetic Oils

Synthetic Oils cover a broad range of categories, which include polyalphaolefins, silicones, polyglycols, and various esters. In general, synthetic oils are less prone to oxidation and can operate at extreme hot or cold temperatures. Physical properties such as pressure-viscosity coefficients tend to vary between oil types and caution should be used when making oil selections.
The polyalphaolefins have a hydrocarbon chemistry which parallel petroleum oil both in their chemical structures and pressure-viscosity coefficients. Therefore, PAO oil is mostly used in the oil-lubricated applications of Timken bearings when severe temperature environments (hot and cold) are encountered or when extended lubricant life is required.
The silicone, ester, and polyglycol oils, however, have a oxygen based chemistry which is structurally quite different from petroleum oils and PAO oils. This difference has a profound effect on its physical properties where pressure-viscosity coefficents can be lower compared to mineral and PAO oils. This means that these types of synthetic oils may actually generate a smaller EHD film thickness than a mineral or PAO oil of equal viscosity at operating temperature. Reductions in bearing fatigue life and increases in bearing wear could result from this reduction of lubricant film thickness.

2.3. Selection of oils

The selection of oil viscosity for any bearing application requires consideration of several factors: load, speed, bearing setting, type of oil, and environmental factors. Since viscosity varies inversely with temperature, a viscosity value must always be stated with the temperature at which it was determined. High viscosity oil is used for low-speed or high-ambient temperature applications. Low viscosity oil is used for high-speed or low-ambient temperature applications.

2.4. Classification

There are several classifications of oils based on viscosity grades. The most familiar are the SAE classifications for automotive engine and gear oils. The American Society for Testing and Materials (ASTM) and the International Organization for Standardization (ISO) have adopted standard viscosity grades for industrial fluids. For reference purposes, fig. 5-7 shows a partial listing of the ISO/ASTM grades plotted on a viscosity-temperature chart. Fig. 5-8 shows the viscosity comparisons of ISO/ASTM with SAE classification systems at 40°C.

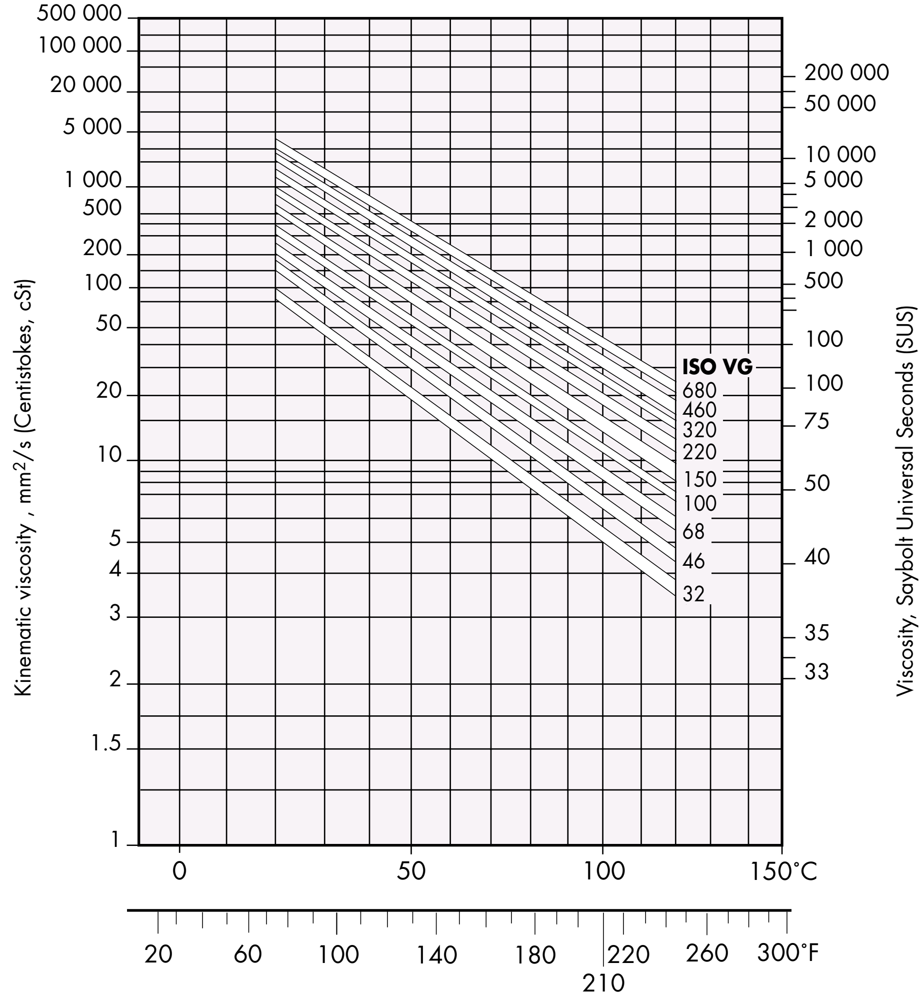

Fig. 5-7
ISO/ASTM viscosity system for industrial fluid lubricants (ISO 3448, ASTM D2442) assuming a viscosity index of 90 Kinematic viscosity, mm2/s (Centistokes, cSt).

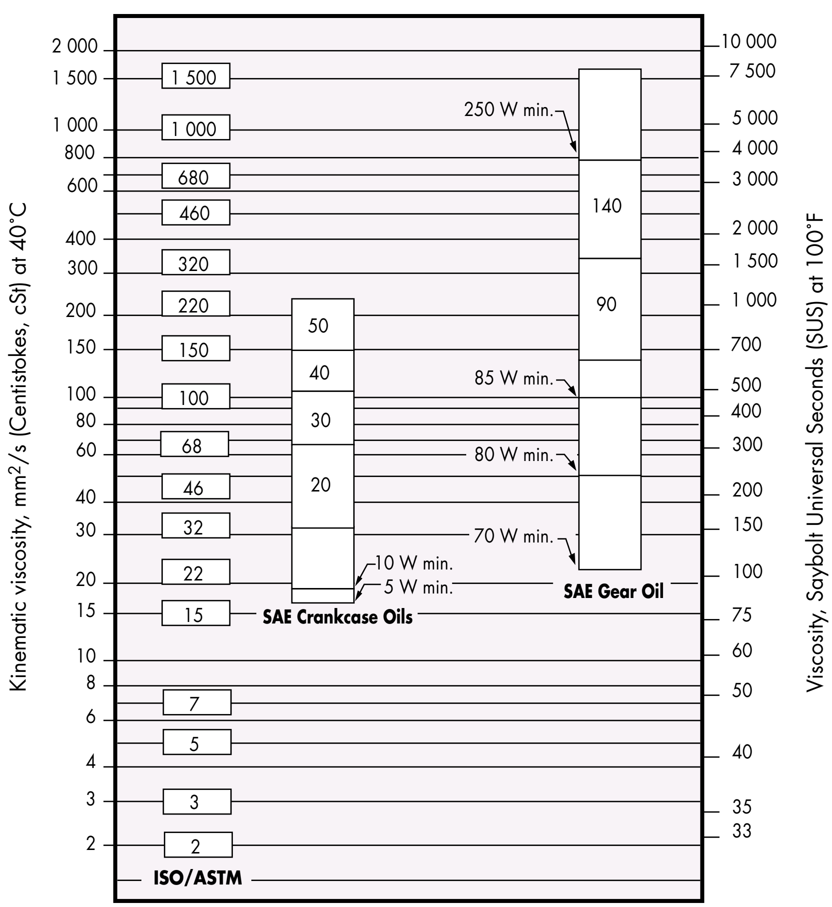

Fig. 5-8
Viscosity classification comparison between ISO/ASTM grades (ISO 3448/ASTM D2442) and SAE grades (SAE J 300-80 for crankcase oils, SAE J 306-81 for axle and manual transmission oils).

2.5. Typical oil lubrication guidelines

In this section, the properties and characteristics of lubricants for typical tapered roller bearing applications are listed. These general characteristics have resulted from long successful performance in these applications.

2.5.1. General purpose rust and oxidation lubricating oil

General purpose rust and oxidation (R&O) inhibited oils are the most common type of industrial lubricant. They are used to lubricate Timken bearings in all types of industrial applications where conditions requiring special considerations do not exist.

Suggested general purpose R&O lubricating oil properties

Base stock	Solvent refined, high viscosity-index petroleum oil
Additives	Corrosion and oxidation inhibitors
Viscosity index	80 min.
Pour point	–10°C max.
Viscosity grades	ISO/ ASTM 32 through 220

Some low-speed and/ or high-ambient temperature applications require the higher viscosity grades, and high-speed and/ or low-temperature applications require the lower viscosity grades.

2.5.2. Industrial extreme pressure (EP) gear oil

Extreme pressure (EP) gear oils are used to lubricate Timken bearings in all types of heavily loaded industrial equipment. They should be capable of withstanding heavy loads including abnormal shock loads common in heavy-duty equipment.

Suggested industrial EP gear oil properties

Base stock	Solvent refined, high viscosity index petroleum oil
Additives	Corrosion and oxidation inhibitors. Extreme pressure (EP) additive* - 15.8 kg(35 lb) min. “OK” Timken load rating
Viscosity index	80 min.
Pour point	–10°C max.
Viscosity grades	ISO/ ASTM 100, 150, 220, 320, 460

* ASTM D 2782

Industrial EP gear oils should be composed of a highly refined petroleum oil base stock plus appropriate inhibitors and additives. They should not contain materials that are corrosive or abrasive to tapered roller bearings. The inhibitors should provide long-term protection from oxidation and protect the bearing from corrosion in the presence of moisture. The oils should resist foaming in service and have good water separation properties. An EP additive protects against scoring under boundary-lubrication conditions. The viscosity grades suggested represent a wide range. High temperature and/or slow-speed applications generally require the higher viscosity grades. Low temperatures and/or high speeds require the use of lower viscosity grades.

2.6. Heat generation and dissipation

One of the major benefits of oil lubricated systems is that the heat generated by the bearings is carried away by the circulating oil and dissipated through the system.

2.6.1. Heat generation

Under normal operating conditions, most of the torque and heat generated by the bearing is due to the elastohydrodynamic losses at the roller/race contacts.

The following equation is used to calculate the heat generated by the bearing:

$$Q_{gen} = k_4 \, n \, M$$
$$M = k_1 \, G_1 \, (n\mu)^{0.62} \, (P_{eq})^{0.3} \quad \textbf{(1)}$$

where:

Q_{gen} = generated heat (W or Btu/min)
M = running torque N.m or lbf - In
n = rotational speed (rev/min)
G_1 = geometry factor from bearing data tables
μ = viscosity at operating temperature (cP)
P_{eq} = equivalent dynamic load (N or lbf)
k_1 = bearing torque constant
= 2.56×10^{-6} for M in N - m
= 3.54×10^{-5} for M in lbf - In

2.6.2. Heat dissipation

The heat dissipation rate of a bearing system is affected by many factors. The modes of heat transfer need to be considered. Major heat transfer modes in most systems are conduction through the housing walls, convection at the inside and outside surfaces of the housing, and convection by the circulating lubricant. In many applications, overall heat dissipation can be divided into two categories: heat removed by circulating oil and heat removed through the housing.

2.6.3. Heat dissipation by circulating oil

Heat dissipated by a circulating oil system is:

$$Q_{oil} = k_5 \; f \; (\theta_o - \theta_i) \quad \textbf{(2)}$$

If a circulating lubricant other than petroleum oil is used, the heat carried away by that lubricant will be:

$$Q_{oil} = k_6 \; C_p \rho f \, (\theta_o - \theta_i) \quad \textbf{(3)}$$

If lubricant flow is unrestricted on the outlet side of a bearing, the flow rate that can freely pass through the bearing depends on bearing size and internal geometry, direction of oil flow, bearing speed, and lubricant properties.
A tapered roller bearing has a natural tendency to pump oil from the small to the large end of the rollers. For maximum oil flow and heat dissipation, the oil inlet should be adjacent to the small end of the rollers.
In a splash or oil level lubrication system, heat will be carried by convection to the inner walls of the housing. The heat dissipation rate with this lubrication method can be enhanced by using cooling coils in the housing sump.

k_4	Dimensional factor to calculate heat generation rate in equation (1) k_4 = 0.105 for Q_{gen} in W when M in N-m = 6.73 x 10^{-4} for Q_{gen} in Btu/min when M in lbf.in	
k_5	Dimensional factor to calculate heat carried away by a petroleum oil in equation (2) k_5 = 28 for Q_{oil} in W when f in L/min and θ in °C = 0.42 for Q_{oil} in Btu/min when f in U.S. pt/min and θ in °F	
k_6	Dimensional factor to calculate heat carried away by a circulating fluid in equation (3) k_6 = 1.67 x 10^{-5} for Q_{oil} in W = 1.67 x 10^{-2} for Q_{oil} in Btu/min	
Q_{oil}	Heat dissipation rate of circulating oil	W, Btu/min
θ_i	Oil inlet temperature	°C, °F
θ_o	Oil outlet temperature	°C, °F
C_p	Specific heat of lubricant	J/(kg x °C), Btu/(lb x °F)
f	Lubricant flow rate	L/min, U.S. pt/min
ρ	Lubricant density	kg/m³, lb/ft³

2.6.4. Heat dissipation through housing

Heat removed through the housing is, in most cases, difficult to determine analytically. If the steady-state bearing temperature is known for one operating condition, the following method can be used to estimate the housing heat dissipation rate.
At the steady-state temperature, the total heat dissipation rate from the bearing must equal the heat generation rate of the bearing. The difference between heat generation rate and heat dissipation rate of the oil is the heat dissipation rate of the housing at the known temperature.

Heat losses from housings are primarily by conduction and convection and are, therefore, nearly linearly related to temperature difference. Thus, the housing heat dissipation rate is:

$$Q_{hsg} = C\,(\theta - \theta_{ambt}) \quad \textbf{(4)}$$

At the operating condition where the steady-state temperature is known, the housing heat dissipation factor can be estimated as:

$$C = \frac{Q_{gen} - Q_{oil}}{\theta - \theta_{ambt}} \quad \textbf{(5)}$$

2.7. Oil systems

There are 3 different types of oil lubricating systems commonly used in industry.

2.7.1. Oil level systems

Oil level systems where the bearings are partially submerged in a static oil reservoir are the simplest types of oil lubrication systems (fig. 5-9). The oil level system is generally only used for low and moderate speed applications because of the limited ability to transfer heat. Effective sealing is important to maintain the required oil level; sight gauges are often used to monitor the oil level.

Heat dissipation can be improved in an oil level system if the oil is splashed on the entire inner surface of the housing (fig. 5-10). Components such as gears, which rotate through the reservoir, splash oil onto housing walls, adjacent bearings and/or into catch troughs that distribute oil by gravity flow through channels to the bearings.

An example of splash lubrication is an automotive differential (fig. 5-11). Oil level systems with splash oil can be used at moderately high speeds if properly designed with a large oil reservoir and large cooling surface. Housing design can have a major influence on the degree of cooling provided.

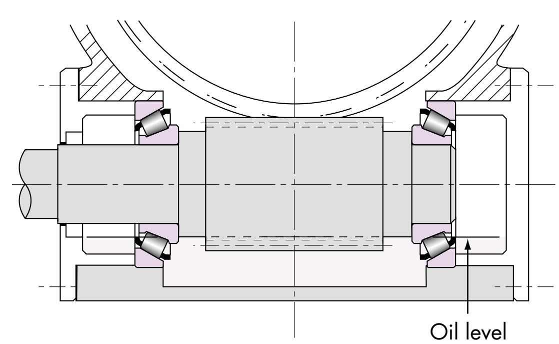

Fig. 5-9
Oil level system.

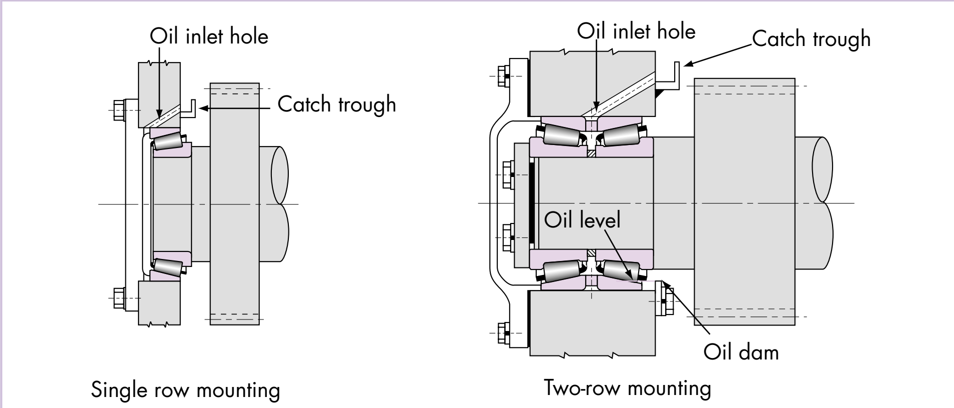

Fig. 5-10
Oil splash systems.

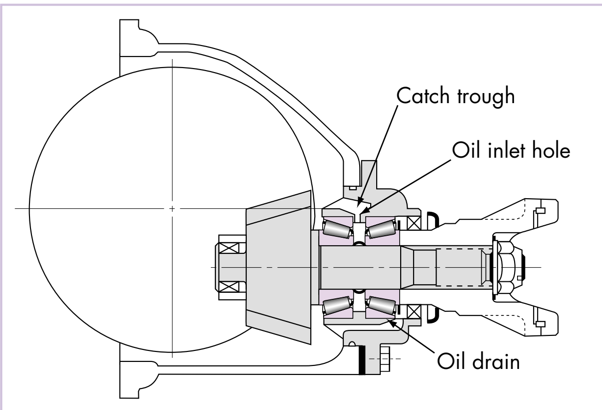

Fig. 5-11
Oil splash system - automotive differential.

2.7.2. Forced-feed oil systems

Forced-feed oil systems are more elaborate than static oil systems. In a typical system (fig. 5-12), oil is pumped from a central reservoir to each bearing. Oil is introduced at the small end of the bearing and drained away at the large end to take advantage of the natural pumping action of tapered roller bearings.

Circulating oil provides a continuous, regulated oil flow. This provides the advantages of maximum heat removal and washing action, which removes contamination or debris that could cause bearing wear. Heat exchangers can be included in a circulating system to reduce oil temperature and extend lubricant life. Filters should be used to remove debris which will cause bearing wear. Circulating oil systems are particularly beneficial on high-performance bearing applications where heat removal and long-term oil life are primary requirements.

Forced-feed oil systems with oil jets are used at higher speeds (fig. 5-13). The jets are positioned to direct oil to the space between the cage and the inner race at the small end of the roller. In addition, oil-jet orifices - usually about 2.5 mm (0.1 in) diameter - can be arranged around the circumference of the bearing to distribute oil at the small and sometimes at the large end of the rollers for maximum cooling efficiency.

Whenever large quantities of oil are used, it is important to balance the quantity of oil drained away with the oil directed into the bearing area. Large drain areas are necessary to prevent a build-up of oil in the bearing. If oil is not properly drained away, temperature will elevate because of excessive heat generated due to churning of the oil.

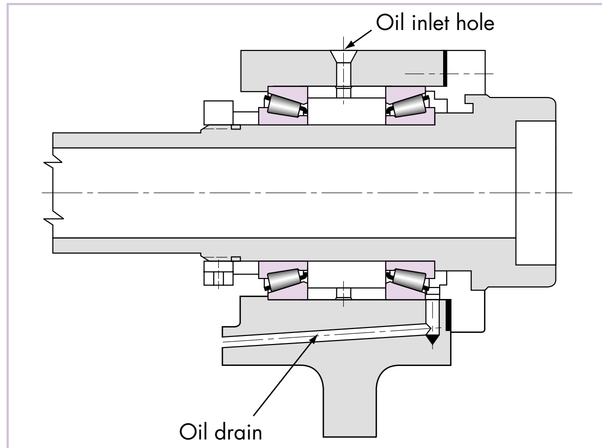

Fig. 5-12
Forced-feed oil system.

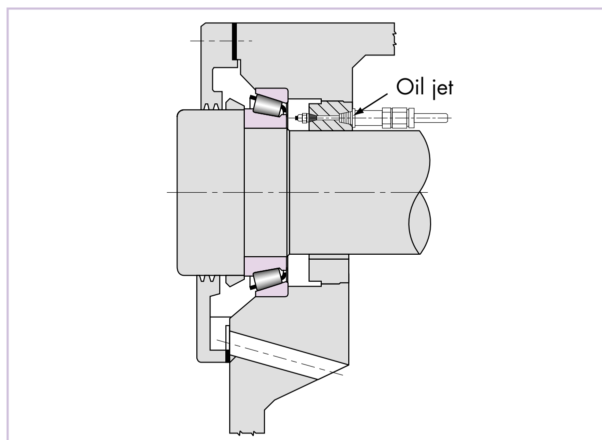

Fig. 5-13
Forced-feed oil system with oil jet.

2.7.3. Oil mist systems

Oil mist systems deliver very fine particles of oil suspended in a low-velocity, low-pressure air stream (fig. 5-14). Oil particles are wet-out in the bearing by reclassification nozzles and/or impingement on high-speed rotating bearing parts. Oil mist provides minimum cooling capacity because the air-flow rate and the specific heat of air are low. However, mist systems can be used on high-performance bearing applications, because heat generation in the equipment is minimized.

For additional information on lubrication systems relative to bearing speeds, see The Timken Company publication “Speed Capabilities”.

3. Lubricant additives

Additives are materials, usually chemicals, that improve specific properties when added to lubricants. Additives, when properly formulated into a lubricant, can increase lubricant life, provide greater resistance to corrosion, increase load-carrying capacity, and enhance other properties. However, additives are very complex, and therefore, should not be added indiscriminately in lubricants as a cure-all for all lubrication problems.

The more common lubricant additives include:

- Oxidation inhibitors for increasing lubricant service life
- Rust or corrosion inhibitors to protect surfaces from rust or corrosion
- Demulsifiers to promote oil and water separation
- Viscosity-index improvers to decrease viscosity sensitivity to temperature change
- Pour-point depressants to lower the pouring point at low temperatures
- Lubricity agents to modify friction
- Antiwear agents to retard wear
- Extreme pressure (EP) additives to prevent scoring under boundary-lubrication conditions
- Detergents and dispersants to maintain cleanliness
- Antifoam agents to reduce foam
- Tackiness agents to improve adhesive properties.

Inorganic additives such as molybdenum disulphide, graphite, and zinc oxide are sometimes included in lubricants. In most tapered roller bearing applications, inorganic additives are of no significant benefit; conversely, as long as the concentration is low and the particle size small, they are not harmful.
Recently, the effects of lubricant chemistry on bearing life (as opposed to the purely physical characteristics) have received much emphasis. Rust, oxidation, extreme pressure and antiwear additive packages are widely used in engine and gear oils. Fatigue testing has shown these additives may, depending on their chemical formulation, concentration and operating temperature, have a positive or negative impact on bearing life. Contact a Timken Company sales engineer or representative for more information regarding lubricant additives.

D. Contamination

1. Abrasive particles

When tapered roller bearings operate in a clean environment, the primary causes of damage is the eventual fatigue of the surfaces where rolling contact occurs. However, when particles contamination enters in the bearing system, it is likely to cause damage such as bruising which can shorten bearing life.

When dirt from the environment or metallic wear debris from some component in the application is allowed to contaminate the lubricant, wear can become the predominant cause of bearing damage. If, due to particle contamination of the lubricant, bearing wear becomes significant, changes will occur to critical bearing dimensions that could adversely affect machine operation.

Bearings operating in a contaminated lubricant exhibit a higher initial rate of wear than those not running in a contaminated lubricant. But, with no further contaminant ingress, this wear rate quickly diminishes as the contamination particles are reduced in size as they pass through the bearing contact area during normal operation.

2. Water

Either dissolved or suspended water in lubricating oils can exert a detrimental influence on bearing fatigue life. Water can cause bearing etching that can also reduce bearing fatigue life. The exact mechanism by which water lowers fatigue life is not fully understood. But it has been suggested that water enters microcracks in the bearing races that are caused by repeated stress cycles. This then leads to corrosion and hydrogen embrittlement in the microcracks which reduce the time required for these cracks to propagate to an unacceptable size spall.

Water-base fluids such as water glycol and invert emulsions have also shown a reduction in bearing fatigue life. Although water from these sources is not the same as contamination, the results support the previous discussion concerning water-contaminated lubricants.

The following chart (fig. 5-15) gives a good idea of the influence of water on bearing life. Based on Timken Research tests, it was determined that water content of 0.01% (100 parts per million) or less, had no effect on bearing life. However, greater amounts of water in the oil will reduce bearing life significantly.

3. Filtration

Many oil lubricated bearing applications are operated satisfactorily without the benefit of filters. However, where environmental contamination can have a detrimental effect on bearing performance, filtration equipment is recommended. Experience has shown that nominally rated 40 µm (0.0015 in) filters are satisfactory for most Timken bearing industrial applications.

E. Conclusion

The lubrication concepts and guidelines presented in this chapter are, by necessity, general in nature. These basic suggestions have been developed by the technical staff of The Timken Company based on knowledge gained from bearing application experience and specific laboratory work. However, the final selection of a lubricant for a particular application is the responsibility of the user.

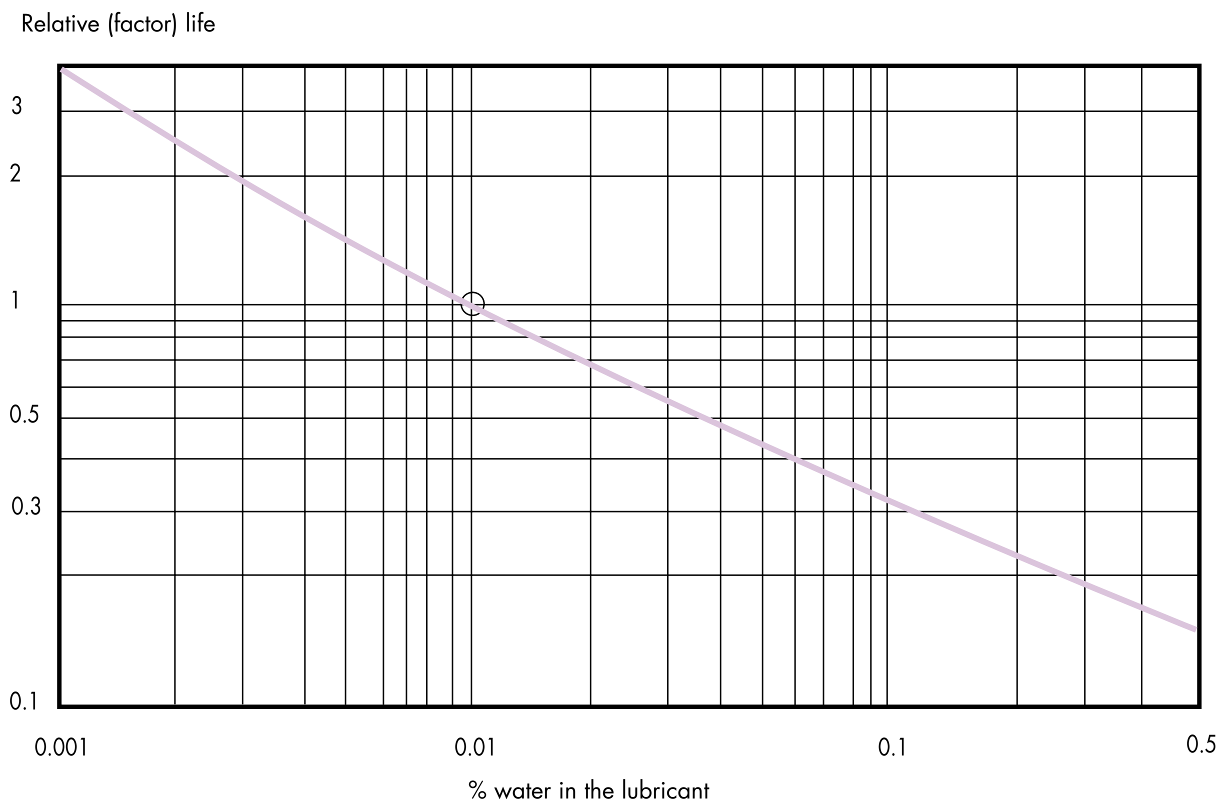

Fig. 5-15
Life reduction with water contamination.

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Notes