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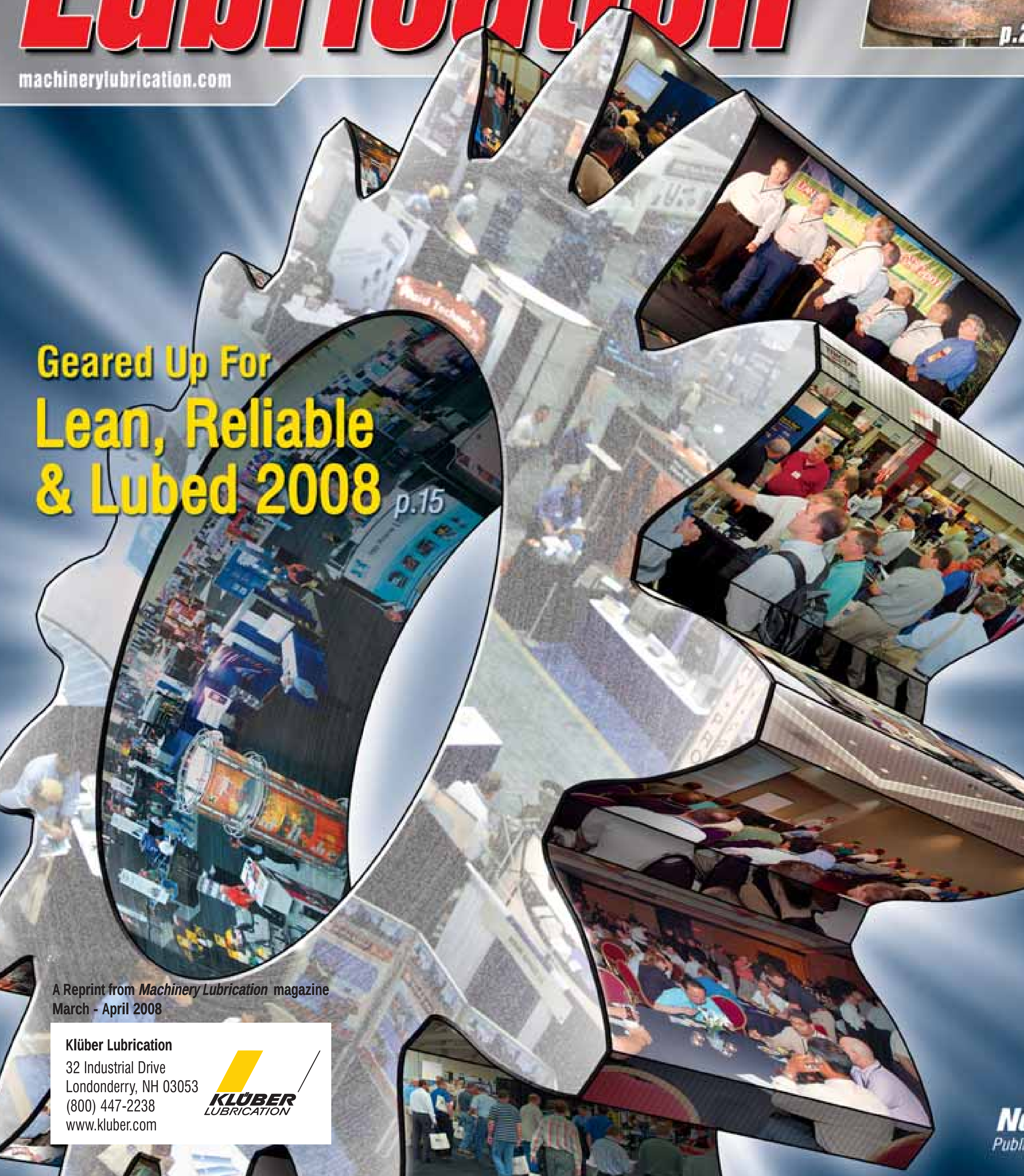

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Tribology: The Key to Proper Lubricant Selection

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In the past, the lubrication requirements for a specific application could be satisfied by using general-purpose lubricants. Lubricant selection was typically based on experience and knowledge. Today, this approach is no longer viable due to the requirements of the current demanding environments to run faster, longer and hotter. Today's lubricants must satisfy extreme requirements that are specific to each application.

Tribology - the study of friction, lubrication and wear - has become the basis for selecting lubricants. The lubrication requirements for a given application can be identified by examining the effects of tribological system parameters on lubricant chemistry.

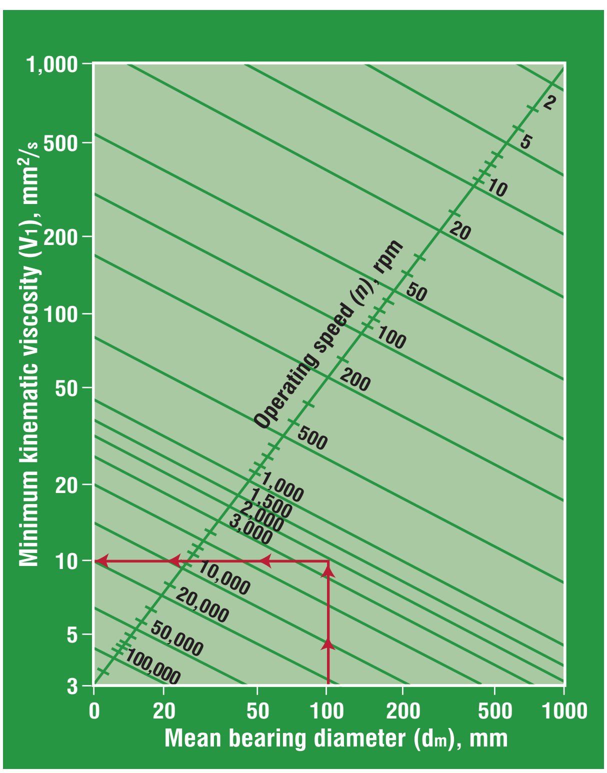

Figure 1. Minimum Allowable Viscosity for Lubrication of Rolling-element Bearings at Operating Temperature

Tribological System

Before the proper lubricant can be selected, the tribological system must be identified. This system includes the type of motion, speed, temperatures, load and the operating environment.

Once these system parameters are identified, the lubrication engineer (or tribo-engineer) can utilize different lubricant chemistries to select a lubricant that will optimize the performance of the application. Because each chemistry has advantages and disadvantages, it is important to choose the appropriate one to address each of the tribological system parameters.

In addition, the lubrication engineer must analyze the application based on the identified tribological system. This analysis includes elements such as speed factors, elastohydrodynamic (EHD) lubrication, bearing-life calculations, extreme-pressure lubrication, emergency lubrication and various special application requirements.

Type of Motion

The first parameter of the tribological system involves the type of motion. The motion may be sliding, which would require the hydrodynamic lubrication theory for its analysis, or rolling, in which case EHD lubrication theory would be applied.

Combined sliding and rolling is also a possible form of motion in certain rolling-element bearings including the tapered roller bearing. Sliding in the rib area can occur in this bearing, but the rolling elements roll on the raceway surfaces. Lubricant protection of these types of motion can be optimized with specific chemistries. Some lubricant chemistries are effective in sliding contacts but do not perform as well in rolling contacts.

Speed

Speed is the second parameter on the tribological system. Speed of roller element bearings can be broken into the general ranges: fast, moderate and slow. Specific ranges for each of these speed categories can be set by using the speed factor, as defined in Equation 1:

Speed factor = $n \cdot d_m$

n = operating speed, rpm

d_m = mean bearing diameter, mm

= (ID +OD)/2

D = inside diameter, mm

OD = outside diameter, mm[2]

Equation 1

The Stribeck curve is a graph showing the relationship between coefficient of friction and the dimensionless number $\eta n/P$, where η is dynamic viscosity, n is speed, and P is the load per unit of projected area. According to this curve, there is an optimum speed for the lubricated contact. Knowing the speed of the contact, a lubricant can be selected with the optimum physical attributes to minimize friction.

Temperature

The third tribological parameter is temperature. All lubricants have specific temperature ranges for optimal performance. Many lubricants have a broad operational temperature range; however, some lubricants are more suited for lower temperatures. For example, there are some greases with synthetic hydrocarbon base oil and barium-complex thickener that can operate at temperatures as low as -60°C.

Other lubricants are designed for high-temperature applications, such as greases with perfluorinated aliphatic ether base oil thickened with polytetrafluoroethylene (PTFE) that can lubricate an oven chain bearing at 220°C for more than 15,000 hours. Knowing the temperature of the tribological system enables the engineer to select a lubricant that will provide optimum operating life and performance at the application temperature.

Load

Load, the fourth parameter, is an important factor affecting the lubricant requirement. A light load may indicate the application is sensitive to frictional torque, and therefore a lubricant would have to be selected to minimize the fluid friction while still providing protection from metal-to-metal friction. On the opposite end is a heavily loaded application, which could require specific additives to help protect from pitting, galling and extreme wear.

Operating Environment

The last parameter of the tribological system is the application's operating environment. If the environment includes moisture or water, the lubricant must provide good anticorrosion properties as well as resistance to water washout or contamination. If the application operates in a vacuum or partial vacuum, the atmospheric pressure of the application must be within the operational limits of the lubricant and above its vapor pressure at the operating temperature.

If the application requires the presence of certain chemical liquids or vapors, the selected lubricant must be resistant to these chemicals. Even an ideal environment, such as a computer room or clean-room processing facility, could have specific requirements for noise-reducing lubricants in rolling-element or instrument bearings.

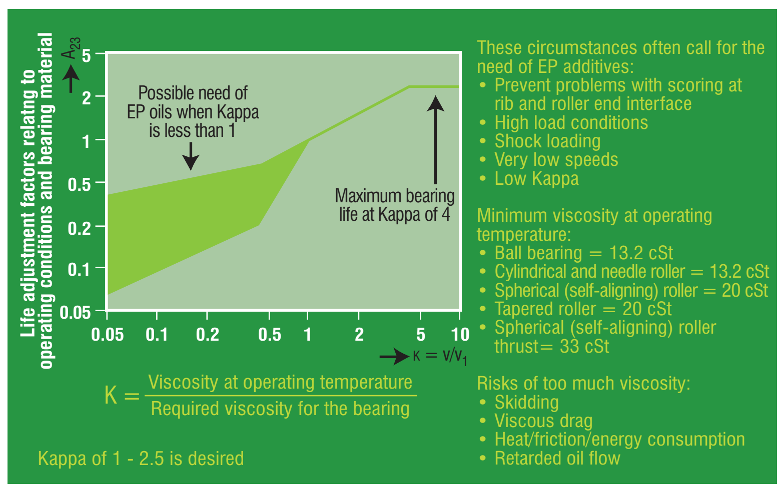

Figure 2. κ Values

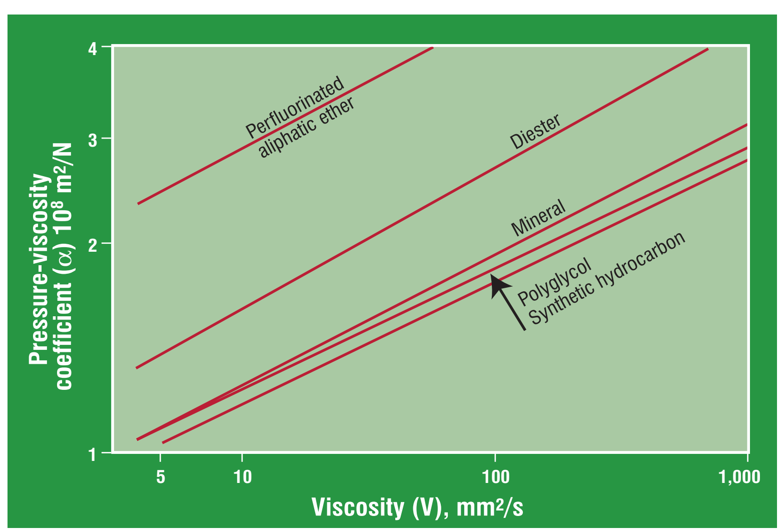

Figure 3. Pressure-viscosity Coefficients for Five Oils Over the Pressure Range 0 to 2,000 bar

Tribological Analysis Theories

The five parameters of the tribological system must be taken into consideration and analyzed for the best lubricant to be selected for the application. However, the information obtained by defining the tribological system parameters also provides data for further in-depth technical analysis.

EHD Lubrication Theory

An important type of analysis involves the lubrication theory for rolling-element bearings. Elastohydrodynamic (EHD) lubrication theory, sometimes referred to as EHL, is used to identify a lubricant's film thickness in a rolling contact. A roller and a raceway can illustrate the factors affecting the minimum film thickness, h0, in the area of rolling contact. Assuming that both surfaces are perfectly smooth, we can define the minimum film thickness, h0 in a rolling contact situation according to Equation 2 which forms the basis for EHD lubrication theory:

$h_0 \quad =([0.1 \times \alpha^{0.6} \times (\eta v)0.07])$

$/\{[(1/r_1) + (1/r_2)^{0.43}(Q/l)^{0.13}\})$

$\{E/[1-(1/m^2)]\}^{0.03}$

where

h_0 = minimum lubricating film thickness in area of rolling contact, µm

α = pressure-viscosity coefficient, mm^2/N

η = dynamic viscosity, mPa•s

v = $(v_1 + v_2)/2$, m/s

v1 = roller velocity, m/s

v_2 = raceway velocity, m/s

r_1 = roller radium, mm

r_2 = inner ring raceway radius, mm

Q = roller load, N

L = roller length, mm

E = modulus of elasticity, N/mm^2

= 2.08×10^5 for steel

1/m = Poisson's ration

Equation 2

Lubrication engineers use the EHD lubrication theory to select the proper viscosity of the lubricant. Each of the variables in Equation 2 has a specific impact on the ultimate film thickness. Most of these variables are under the control of the designer of the application, but several may also be under the control of the lubrication engineer. One of the lubrication engineer's primary interests is how a change in a specific variable will affect the magnitude of the film thickness.

From Equation 2, it can be determined that if the pressure-viscosity coefficient (α) is doubled, there is an increase in the film thickness by 51 percent. Knowing the pressure-viscosity coefficient of the different lubricant chemistries, the lubrication engineer can alter the film thickness by changing the lubricant chemistry. The remaining physical characteristics of the lubricant are not changed.

Another lubricant-related variable in Equation 2 is the dynamic viscosity, η. The dynamic viscosity can be directly related to the kinematic viscosity, and if it is doubled, will increase film thickness by 62 percent. By doubling the velocity of the roller bearing, the film thickness of the lubricant can again be increased by 62 percent. The lubrication engineer has no control over the speed of the application, but knowing how the speed affects the film thickness is important to the selection of a lubricant when the application has variable-speed capability. Additional variables have a lesser impact on the film thickness of the lubricant.

While the information in Equation 2 is important to the lubricant selection criteria, the EHD film thickness is not used directly because in reality, surfaces are not perfectly smooth. Instead of minimum film thickness, the lubrication engineer works with the specific film thickness, λ, defined as the ratio between the average or mean EHD film thickness, a, and the composite surface roughness of the rolling contacts σ (equation 3).

$\lambda = h0/\sigma$

where

λ = specific film thickness

σ = composite roughness of the two contact surfaces

$= (\sigma_1^2 + \sigma_2^2)^{½}$

σ_1	= rms roughness of raceway, μm
σ_2	= rms roughness of roller, μm

Equation 3

From Equation 3, it is clear that if the specific film thickness is close to zero, there will be a dramatic increase in the metal-to-metal contact at the friction point. This increased metal contact produces unacceptable wear. When λ=1, the bearing will have only partially separated the metal asperities and some metal-to-metal contact will still be occurring.

It is at this point that the transition from boundary lubrication into the mixed lubrication occurs. As the specific film thickness increases above 1 and approaches values greater than 3, there will be a decrease in metal-to-metal contact with an associated decrease in wear. When λ = 3 to 4 or greater, there is full metal separation between the peaks, indicating full fluid film lubrication and an absence of wear.

Lambda values greater than 4 are possible and sometimes desirable, particularly where variable speeds and or shock loading is present. However as λ increases beyond 4, the internal fluid friction may increase and generate excessive heating and energy consumption depending upon the relative bearing speed and oil viscosity.

Required Viscosity

Research sponsored by the bearing manufacturers has made it possible to calculate the minimum required viscosity of the lubricant V_1 to obtain separation of the moving surfaces. The nomograph in Figure 1 can be used to determine the value of V_1 from the mean bearing diameter, d_m, and the operating speed, n.

For example, enter the bottom of the chart with the average diameter of the rolling-element bearing (100 mm), then proceed vertically to the speed of the bearing (2,000 rpm). Drawing a line horizontally to the left then provides the required viscosity (10mm^2/s) to obtain λ = 1 in this specific application at the given operating temperature.

Knowing the required minimum viscosity, V_1, at operating temperature, the lubrication engineer can then select the appropriate lubricant based on the specific viscosity, κ, which is defined as the ratio of the actual viscosity, V, of the selected lubricant to the required minimum viscosity, V_1. Given this information, the engineer will attempt to select a lubricant to meet the full fluid film lubrication regime for the application. If this is not possible, then the next available option is to select a lubricant that provides the best protection to the application.

Figure 2 shows how κ relates to both lubricant selection and expected bearing life. At κ values below 1, it is generally accepted that EP additives will be required to mitigate the effects of boundary lubrication conditions. As κ approaches 1, the life expectancy approaches the rated life expectancy L of the bearing according to DIN ISO 281. At κ values above 1, it is in fact possible to exceed the L value of the bearings, perhaps by as much as 2.5 times.

As κ approaches 4, the bearing life approaches a maximum (all else being equal) while κ values above 4 may cause increase fluid friction, viscous drag, ball skidding and other undesirable effects. It is generally accepted that ê values in the 1 to 2.5 range are optimal for most bearing applications.

If the application must be operated in the boundary lubrication regime, then specific additives must be provided to protect the metal contact points. If the application is operating in a lubrication regime that approaches full fluid film lubrication, these extreme-pressure additives can be eliminated from the lubricant.

Points of Consideration

The EHD theory is a valuable tool in guiding the lubrication engineer toward selecting the proper lubricant. If the lubrication engineer uses the specific viscosity as part of the selection criteria, the assumptions used in the analysis must be taken into consideration, along with the controllable variables. Two specific points should be considered:

1. Calculation of film thickness (h_0), specific film thickness (λ), and the chart used to determine required viscosity (V_1) are all based on the pressure-viscosity coefficient (α) of mineral oil. Figure 3 shows the pressure-viscosity coefficient for mineral oil and four other oils over the pressure range zero to 2,000 bar.

2. The oil is considered to be the only lubricating component of the lubricant. In a grease, it is believed that the thickener system does not contribute to the lubricating film thickness. However, the thickener system can have a significant effect on film thickness.

Figure 4 documents research identifying the effect of a grease thickener system on the actual film thickness. The upper chart shows that a clay-thickened grease will reduce the film thickness of the application to almost 50 percent of the film thickness provided by the base oil alone. A grease thickened with a barium-complex soap has the opposite effect of almost doubling the film thickness relative to the base oil alone.

However, as shown in the lower chart in Figure 4, the advantage of increased load-carrying film thickness comes with the disadvantage of frictional torque. While a clay-thickened grease reduces the film thickness, it does have the benefit of reducing frictional torque in the bearing. The opposite effect is observed for grease thickened with a barium-complex soap.

Speed Factor and Bearing Design

Speed is another parameter of the tribological system that requires further detailed analysis. The calculated speed factor for a rolling-element bearing, which is defined as the product of bearing speed (n) and mean bearing diameter (d_m), is not consistent for all bearings with the same dimensions and speed. The lubrication engineer must apply a correction to the calculated speed factor, depending on the bearing design.

Lubricant Selection Criteria

Tribological System

Analysis of the tribological system for a given application is essential to the selection of the appropriate lubricant.

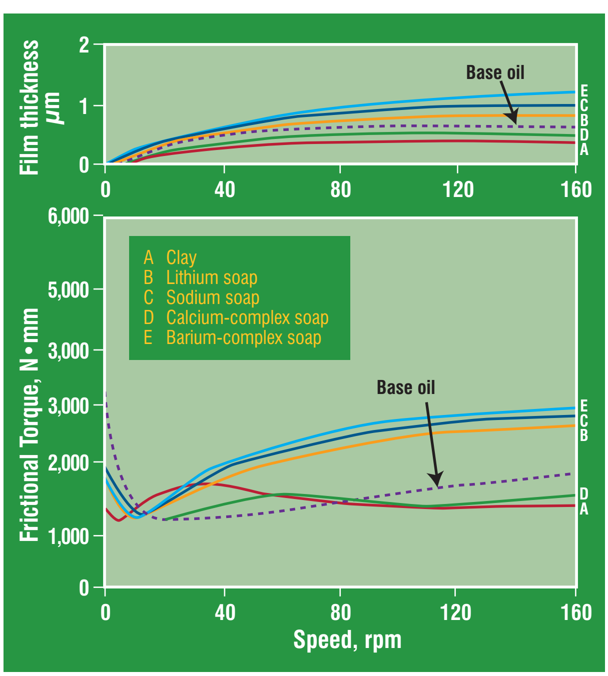

Figure 4. Film Thickness and Frictional Torque for Base Oil and Five Greases as Functions of Bearing Speed

- Service temperature range
- Speed factor ($n \cdot d_m$)
- Hydrodynamic lubrication
- Elastohydrodynamic lubrication
- Extreme pressure
- Emergency lubrication
- Fretting

Special Requirements

Many applications have special requirements that go beyond the tribological system that must be taken into consideration. Some applications are limited to oils, while others require a grease. Applications that involve the use of sintered bearings or special sealing arrangements will require additional analysis. Material compatibility is another important issue.

Other Requirements

Lubricant selection can also be affected by a variety of other specialized requirements:

- Design life
- Lubrication equipment
- Acceptable relubrication intervals
- Cost
- Special certifications such as NSF registration
- Biodegradability

Summary

Multipurpose lubricants cannot provide satisfactory service in current demanding environments. Lubricant performance must be optimized to meet the increasing demands of modern industry.

The first step in selecting the best lubricant for a given application is to define the tribological system. With a fully defined tribological system in place, the next step is theoretical analysis. Selection of a lubricant based on EHD lubrication analysis or analysis of any other discrete parameter is inappropriate, because such analyses focus only on a subset of the tribological system. ML

About the Author

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