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This months' installment is the second of three on gearbox lubrication. This column will cover issues related to the selection of gear and bearing lubricants

When available, the lubrication recommendations of the original equipment manufacturer should be followed. However, if problems are encountered with the OEM recommendations or operating conditions change, the following guidelines can be used.

The primary purpose of the lubricant is to minimize metal-to-metal contact between the mating surfaces. Since viscosity is the most important property that determines the lubricant film thickness, we need to first look at how to select the proper viscosity.

When gear teeth and rolling element bearings are loaded, the contact surfaces deform slightly and become essentially flat in the load zone. As the surfaces roll oil is drawn into the contact zone, and due to the pressure between the surfaces; the viscosity rises rapidly and is sufficient to separate the two surfaces. Increasing the viscosity grade of the oil or reducing the operating temperature will increase the resulting film thickness. The film thickness will also increase as the rolling speed is increased.

On a microscopic level, the contact surfaces consist of a series of peaks and valleys (see Figure 1). There are three basic regimes of lubrication. The "elastohydrodynamic" (EHD) regime is where the lubricant film is thick enough to keep the peaks on both surfaces fully separated. If the film thickness is thin and the peaks are in constant contact, then the contact is in the boundary lubrication regime. If the film thickness allows occasional contact between the peaks, then the contact is in what is called the "mixed lubrication" regime.

The parameter that is used to quantify the contact condition is the "specific film thickness ratio Lambda" (λ). It is defined as the minimum lubricant film thickness divided by the composite surface finish of the mating surfaces. AGMA 925-A03 "Effect of Lubrication on Gear Surface Distress" provides equations for calculating the minimum oil film thickness and Lambda ratio for gear meshes. It also provides equations for evaluating scuffing risk and the probability of wear-related distress.

Testing and field data has shown that the required minimum λ ratio to avoid wear-related distress varies with the pitch line velocity of the gear mesh. Figure 15 of AGMA 925-A03 shows the approximate relationship between the probability of wear, pitch line velocity, and λ ratio.

If the λ Lambda ratio cannot be determined, Appendix B of ANSI/AGMA 9005-E02 "Industrial Gear Lubrication" provides guidelines for the minimum oil viscosity based on the bulk oil temperature and pitch line velocity of the low speed gear set.

Some bearing manufacturers use κ ratio to quantify the lubrication conditions for bearings. It is defined as the ratio of the actual operating viscosity divided by the minimum viscosity required to provide adequate lubrication. If possible, a lubricant viscosity should be selected that results in a κ ratio greater than 1.0 for the bearings on the low speed shaft. If the κ ratio is less than 1.0, the theoretical L10 bearing life is reduced. Formulas for determining the κ ratio and the lubrication condition's effect on bearing life can usually be found in the bearing manufacturer's catalogs or on their Web sites.

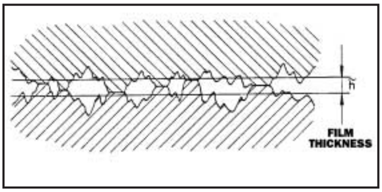

FIGURE 1

Once the desired minimum oil viscosity has been determined, consideration has to be given to the maximum oil viscosity that can be used. The pour point of the oil should be higher than the lowest anticipated cold starting temperature of the gearbox. If the gearbox has a pressurized lubrication system, the oil viscosity has to be low enough to avoid pump cavitation and prevent collapsing the filter. In splash lubricated systems with high-speed meshes the maximum oil viscosity has to be low enough to avoid excessive churning losses in the gears and bearings.

A synthetic oil may be required if the operating conditions are severe. Synthetic oils generally have a higher viscosity index, so their viscosity will be higher at elevated temperatures than a mineral oil of the same viscosity grade. In addition, they resist oxidation better, so the oil will last longer when running at elevated temperatures. The economic benefits of extended drain intervals versus higher initial cost needs to be evaluated. Synthetic oil pour points are low so that they can be used in applications requiring low temperature starts. The pressure viscosity coefficients of some synthetic oils are lower than mineral oils, so the calculated film thickness of each should be compared when deciding whether or not to use a synthetic oil.

After making an initial oil selection using the above criteria, the oil's compatibility with other components such as brakes, clutches, or backstops should be checked. In addition, the oil's affect on seals and interior paint needs to be determined. Some synthetic oils can damage Nitrile seals.

If the gear meshes must operate with an oil viscosity that results in low Lambda ratios and there is significant contact between surface peaks, then the oil should contain some EP and antiwear additives.

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