COOLING AND LUBRICATION OF HIGH-SPEED HELICAL GEARS

by

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ABSTRACT

Centrifugal compressor and turbine manufacturers are building machines that operate at increasingly higher speeds to reduce the size and cost of their equipment. Helical gears are commonly used to step down the speed of these machines to match nominal motor and generator speeds. The demand for higher compressor and turbine speeds has resulted in an increase in the maximum pitch line velocities of helical gears.

Gear manufacturers have also increased the allowable tooth loading to reduce the size and cost of their equipment. The increase in pitch line velocities and tooth loadings has not been without problems. Several new and updated American Gear Manufacturers Association (AGMA) standards have been published to educate end users on the issues with high-speed helical gears and to provide consistent criteria for rating gears. One of the key issues is lubrication and cooling. The purpose of this paper is to present some of the current methods for evaluating the lubrication and cooling requirements of high-speed helical gears to assist end users with selecting new gears and to provide a tool for solving gear problems.

INTRODUCTION

Proper lubrication and cooling are critical to the reliable operation of the high-speed helical gears. As pitch line velocities and gear tooth loading have increased, so have the lubrication and cooling requirements. Figure 1 shows how the maximum pitch line velocities of helical gears have increased over the last 20 years.

The primary purposes of the lubricant are to prevent wear of the gear tooth surfaces and to remove the heat generated by friction of the mating gear teeth. Although there is no universal standard on how to design for proper lubrication and cooling of high-speed helical gears, within the last few years gear manufacturer associations, such as American Gear Manufacturers Association (AGMA),

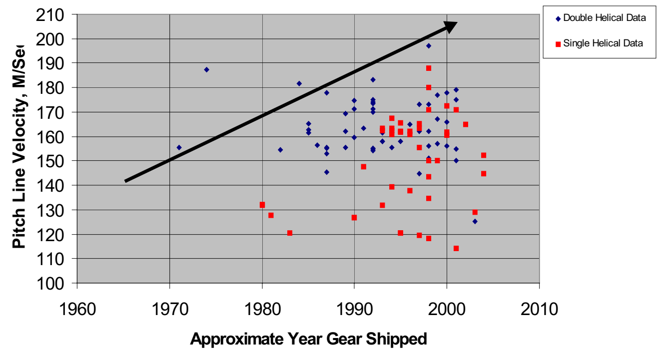

Figure 1. Increase in Helical Gear Pitch Line Velocities.

have published new and updated standards that provide guidance on how to perform critical gear calculations. Many technical papers have also been published on this subject.

In this paper the current critical design calculations used to evaluate lubrication and cooling of high-speed helical gears will be presented and explained. Guidance on how to interpret the results will also be provided. In addition, some unique design concepts used by different gear manufacturers will be described. In this paper, high-speed helical gears are defined as those gears that operate pitch line velocities greater than 120 m/s (394 ft/s). The topics include:

- Key gear parameters
- Lubricant properties
- Gear surface distress
- Thermal problems
- Gear spray arrangements
- Special designs
- Modern oils

The information presented in this paper came from several AGMA standards, books, and published papers. Additional information was also obtained from discussions, design reviews, surveys, and the author's observations of gear designs from the following companies: BHS Getriebe, Cooper Compression, Flender Graffenstaden, Lufkin, MAAG, Philadelphia Gear, Renk AG, and Siemens AG.

KEY GEAR DESIGN PARAMETERS

When gears mesh, the tooth surfaces roll and slide against each other. This meshing creates enormous contact and shear stresses. Oil is used to lubricate the mating gear teeth and prevent scuffing, wear, and pitting damage to gear tooth surfaces due to metal-to-metal contact. Oil is also used to cool the gear teeth and prevent excessive tooth temperatures and overheating of the oil. The key gear design parameters that are used to evaluate the lubrication and cooling requirements of high-speed helical gears include the pitch line velocity, helix angle, axial meshing velocity, face width, pressure angle, surface roughness, power, and the lubricant properties.

Early gears were nothing more than one cylinder driving another cylinder by friction. Today's gears are toothed wheels, but these toothed wheels can still be modeled as two cylinders. Based on the distance between the centers of the gears and the number of gear teeth of both gears, the theoretical equivalent diameters of these two cylinders can be determined. These theoretical diameters are known as the pitch diameters. The pitch diameter is somewhere between the root circle of the gear teeth and the outside diameter (OD) of the gear (Figure 2). It cannot be measured directly on a gear, but it can be calculated for a meshed pair of gears. The following equations apply:

$$r_1 = \frac{a_w}{u + 1} \tag{1}$$

$$u = \frac{z_2}{z_1} = \frac{r_2}{r_1} \tag{2}$$

where:
a_w = Center distance, mm
r_1 = Pitch radius of pinion, mm
r_2 = Pitch radius of gear, mm
u = Gear tooth ratio
z_1 = Number of pinion teeth
z_2 = Number of gear teeth

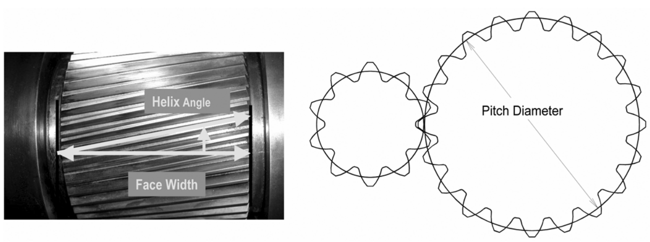

Figure 2. Pitch Diameter, Face Width, and Helix Angle.

Pitch line velocity is the linear speed of a point on the pitch diameter of a gear. For a meshed set of gears, the pinion and the gear operate at the same pitch line velocity. It is equal to rotational speed (n_1 for the pinion) multiplied by the pitch radius (r_1).

$$v_t = \frac{n_1 * \pi * r_1}{30{,}000} \tag{3}$$

where:
v_t = Pitch line velocity, m/sec
n_1 = Pinion speed, rpm

The bottom of the tooth on the gear that is the driver first contacts the top of the mating-driven gear tooth. These teeth roll and slide as they mesh. The sliding velocity on the driver gear tooth is away from the pitch line (toward the root of the gear teeth) when the meshed teeth first contact. The sliding velocity slows to zero at the pitch line and then increases in the opposite direction, away from the pitch line (toward the OD of the gear) until the gear teeth separate. The direction of the sliding velocity is just the opposite for the driven gear tooth (Figure 3). This is an important concept because many issues with surface distress are a function of the sliding velocity, which is zero at the pitch line and maximum at either end of the gear tooth.

The line of action is the line formed by the point of contact of the gear and pinion along the length of a tooth. For a gear tooth on the gear that is the driver, the line of action starts at the bottom of the tooth where it first contacts the driven gear tooth, and it extends across the pitch circle to the point to the top of the tooth where the meshed gear teeth separate.

The helix angle is the angle between the tooth and the axial axis of the gear at the pitch circle. Helical gears can be single or double helical types (refer to Figures 2 and 4). Most single helical gears

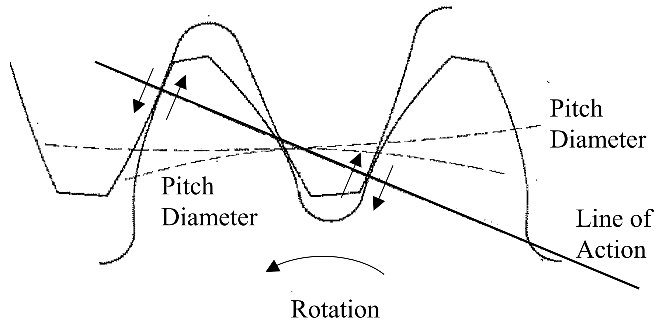

Figure 3. Direction of Sliding.

are made with helix angles between 5 and 20 degrees. Most double helical gears are made with helix angles between 15 and 30 degrees. The advantage of double helical gears is that there is no axial gear force, but double helical gears are much more expensive to manufacture. For single helical gears, the larger the helix angle, the greater the axial thrust.

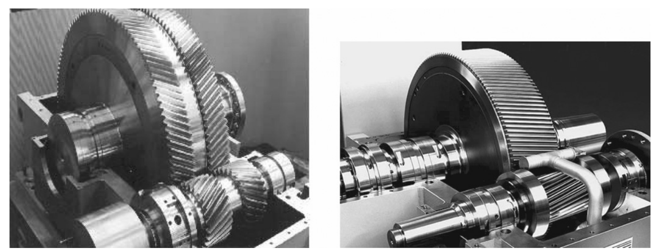

Figure 4. Double Versus Single Helical Gear.

The lower the helix angle, the higher the axial meshing velocity. Trapped air and oil get pumped across the gear mesh at this speed. So, as the axial meshing velocity increases, so does the frictional heating of this trapped air and oil mixture. This effect causes higher gear tooth surface temperatures at the oil discharge end of the gear teeth and can lead to thermal problems, especially with wide gears.

$$Axial\ meshing\ velocity = \frac{\pi * r_1 * n_1}{30{,}000 * \tan(\beta)} \tag{4}$$

where:
β = Helix angle, degrees

The face width is the length of the gear teeth measured on a line parallel to the axial axis. The wider the gears, the greater the distance the oil and air travel and the hotter the trapped air and oil become, which makes face width another key component in evaluating the potential for thermal problems.

The pressure angle is the slope of the gear tooth at the pitch circle (Figure 5). For high-speed helical gears, the most common pressure angle is 20 degrees. At lower angles the tooth load capacity is reduced, the calculated tooth temperatures are higher, and the calculated oil film thickness is less. At higher-pressure angles the gear meshing is not as smooth and there will be more noise.

Gear tooth finish is a critical factor in controlling scuffing and pitting of gear tooth surfaces. The rougher the gear surface, the greater the friction between the mating gear teeth. Greater friction means higher tooth temperatures, which reduces the viscosity of the oil. Lower viscosity increases the risk of wear and scuffing.

Power affects the cooling and lubrication requirements for high-speed helical gearing. In general, the frictional losses for a single pinion and gear combination are approximately 1.5 percent to 2 percent of the total power. This includes bearing and gear losses. For compressors with multiple pinions, the total frictional power losses will be higher since there are multiple gear combinations.

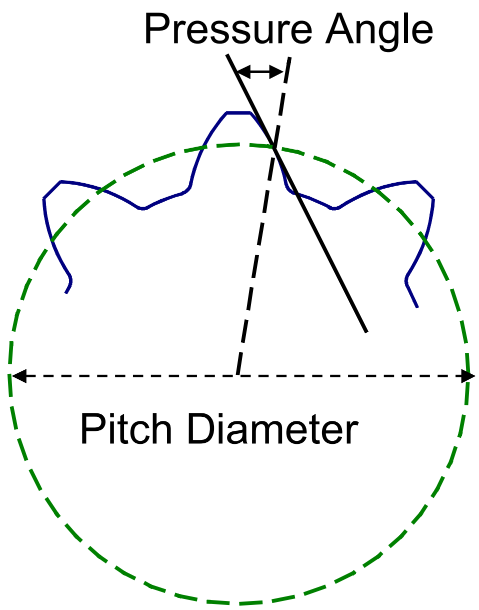

Figure 5. Pressure Angle.

LUBRICANT PROPERTIES

AGMA 9005-E02 (2002) provides guidance on selecting an oil for a gear application. The key lubricant properties are the class of oil, the viscosity, and the pressure-viscosity coefficient. Oils used in high-speed gear applications are generally classified as either inhibited or antiscuff/antiwear oils. Inhibited oils are formulated with highly refined mineral or synthetic base oils and contain additives that enhance oxidation stability, provide corrosion resistance, and inhibit foam. Antiscuff/antiwear oils contain additional additives that provide protection against wear and scuffing. The viscosity is the oil's ability to resist shear. Oil viscosity decreases with increasing temperature. The rate of change of viscosity with temperature is generally less with synthetic oils, which can be an advantage in some cases. The pressure-viscosity coefficient is used to determine how the viscosity changes with pressure.

Oil can separate the mating gear teeth despite the enormous stresses because oil viscosity increases with pressure. In the gear teeth contact band (also known as the Hertzian contact band), Errichello (1991) states that the trapped lubricant cannot escape because the viscosity increases to the point where it is essentially a rigid solid.

The mating surfaces of gear teeth are not perfectly smooth. The surfaces are full of small asperities, such as marks due to machining. This is what is referred to as surface roughness. If the oil film is thick enough, the asperities from the mating gear teeth do not contact one another. Based on elastohydrodynamic lubrication (EHL) theory, lubrication falls into one of three regimes. In Regime 1, there is essentially no EHL oil film even though the tooth surfaces appear wet with oil. The asperities contact one another in the Hertzian band contact areas. These conditions exist with slow-speed, high-load gears and rough gear teeth. A typical application is a winch. In Regime 2, there is a partial EHL oil film thickness and there are areas where the asperities contact one another and other areas where there is sufficient film thickness so that there is no contact. These conditions exist in medium-speed, high-load gears operating with thick oil. A typical application is an automotive gear set. In Regime 3, full EHL oil film is developed and the asperities in the tooth surfaces do not contact one another. A typical application is a high-speed gear used in a centrifugal compressor.

GEAR SURFACE DISTRESS

Gear tooth distress includes scuffing, wear, and surface fatigue (micropitting and macropitting). ANSI/AGMA 1010-E95 (1995) provides excellent descriptions and pictures of these types of damage.

Scuffing

Scuffing is damage to a tooth surface due to welding and tearing of the tooth surface by the flank of the mating tooth. It is characterized as radial scratch lines (Figure 6). Scuffing occurs when the oil film thickness is less than the composite roughness of the pinion and gear and metal-to-metal contact occurs. Most manufacturers use a scuffing criterion and/or a comparison of the calculated contact temperature to the scuffing temperature to evaluate the risk of scuffing.

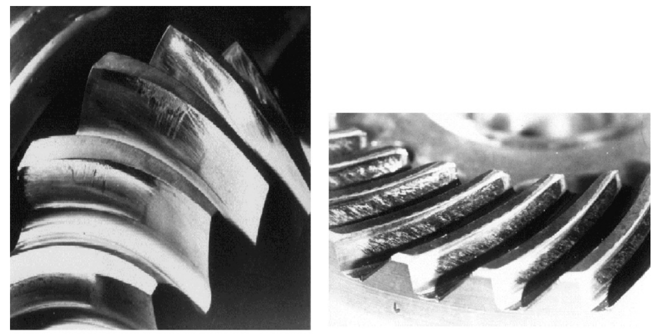

Figure 6. Scuffing. (Courtesy ANSI/AGMA 1010-E95, 1995)

The procedure in AGMA 925-A03 (2003) can be used to determine the risk of scuffing. Per this standard, "The basic mechanism of scuffing is caused by intense frictional heat generated by a combination of high sliding velocity and high contact stress." Scuffing will normally not start at the pitch line because the sliding velocity is zero. Thus, scuffing generally starts in either the top or bottom half of the teeth. The theory used to evaluate scuffing is based on Blok's (1937) contact temperature theory, which states that scuffing will occur when the contact temperature reaches a specific temperature based on the oil. The contact temperature is equal to the flash temperature plus the tooth temperature before entering the mesh. The flash temperature is a function of the geometry, load, friction, velocity, and material properties during operation. It varies along the line of action. For helical and spur gears, the equation for flash temperature based on Blok's (1937) theory is:

$$\theta_{fl(i)} = \frac{31.62\ K\mu_{m(i)} X_{\Gamma(i)} \omega_n{}^* \left| v_{r1(i)} - v_{r2(i)} \right|}{b_{H(I)}^{0.5} \left(B_{M1} v_{r1(i)}^{0.5} + B_{M2} v_{r2(i)}^{0.5} \right)} \tag{5}$$

where:

- $\theta_{fl(i)}$ = Flash temperature, °C
- K = Numerical factor for a semi-elliptic distribution of heat
- $\mu_{m(i)}$ = Mean coefficient of friction
- $X_{\Gamma(i)}$ = Load sharing factor
- $v_{r1(i)}$ = Rolling tangential velocity of pinion, m/s
- $v_{r2(i)}$ = Rolling tangential velocity of gear, m/s
- $b_{h(i)}$ = Semi-width of the Hertzian band contact, mm
- B_{M1} = Thermal contact coefficient of the pinion material, N/[mm $s^{0.5}$K]
- B_{M2} = Thermal contact coefficient of the gear material, N/[mm $s^{0.5}$K]

The subscript (*i*) is used to identify a point along the line of action of the gear teeth. AGMA 925-A03 (2003) provides the basic equations for calculating the rolling tangential velocities of the gears, the normal unit load, and the semi-width of the Hertzian contact band. The flash temperature is calculated along the entire line of contact of the gear teeth.

The semi-width of the Hertzian band contact is a function of the load, radius of curvature of the teeth, and material properties of the gears. The thermal contact coefficient is a function of the material and for most steels it is approximately 13.6 N/[mm $s^{0.5}$K].

The load sharing factor corrects the tooth loading along the line of action since the mating teeth do not handle the full load when they first make contact. The load sharing factor is 1.0 in the middle of the tooth and less than 1.0 at the bottom and top of the tooth.

The standard also provides some guidance on determining the mean coefficient of friction. It is assumed that the coefficient of friction is constant along the line of contact.

$$\mu_{m(i)} = 0.6 * C_{Ravgx} \quad (6)$$

$$1.0 \leq C_{Ravgx} = \frac{1.13}{1.13 - R_{Ravgx}} \leq 3.0 \quad (7)$$

$$R_{avgx} = \frac{R_{a1x} + R_{a2x}}{2} \quad (8)$$

where:
C_{avgx} = Surface roughness constant, mm
R_{avgx} = Average surface roughness of pinion and gear, µm
R_{a1x} = Average surfaced roughness of pinion, µm
R_{a2x} = Average surface roughness of gear, µm

Equation (6) is based on lubrication for Regime 2. Since high-speed gears normally operate in Regime 3, $\mu_{m(i)}$ tends to be lower than the value calculated in Equation (6). For high-speed gears, it is normally less than 0.01.

The calculated tooth temperature is the sum of the tooth temperature before entering the meshing zone plus the flash temperature. Since the flash temperature varies across the line of action, the tooth temperature will vary across the entire line of action. The maximum flash temperature does not normally occur at the pitch line because the sliding velocity at the pitch line is zero. It may occur in the upper or lower half of the tooth. For low-speed helical gears with spray lube, the temperature of the teeth before entering the meshing zone is typically 1.2 times the oil supply temperature. However, in high-speed helical gears, the temperature of the teeth before entering the meshing zone may be much higher. For example, Errichello (1991) reported a case where the temperature of the teeth before entering the meshing zone was 171°C (340°F) higher than the oil supply temperature. In another example, testing done by Akazawa, and others (1980), on a gear unit running at 25,000 rpm and at a pitch line velocity of 200 m/s (656 ft/s) resulted in tooth temperatures that were 81°C (178°F) higher than the oil supply temperature plus flash gas temperature.

At high pitch line speed there are several sources that can cause higher tooth temperatures. Per AGMA 925-A03 (2003), "For pitch line velocities above 80 m/s, churning loss, expulsion of oil between meshing teeth, and windage loss become important heat sources that must be considered." The other factor is how effective the oil spray is in cooling the teeth. When spraying the gear mesh, it is important for the oil to cover the tooth surfaces for a slight time interval before being flung off. At high pitch line speeds, the teeth may be moving so fast relative to the oil velocity that not all the teeth get covered with oil. Per Dudley (1984), the oil can also be deflected away by the high wind produced by gear rotation at high speed. These effects can cause overheating and blackening of some of the gear tooth surfaces.

Scuffing is a function of the material-lubricant system of a gear pair. AGMA 925-A03 (2003) provides the following equations for scuffing temperature:

$$\theta_s = 63 + 33 \ln v_{40} \ (\textit{for nonantiscuff mineral oils}) \quad (9)$$

$$\theta_s = 118 + 33 \ln v_{40} \ (\textit{for antiscuff mineral oils}) \quad (10)$$

where:
θ_s = Scuffing temperature, °C
v_{40} = Kinematic viscosity at 40°C, mm^2/sec

For gears operating with a typical ISO 46 turbine oil, the calculated scuffing temperature is 191°C (376°F). If the high-speed helical gear teeth approach this temperature, there will be additional problems other than just scuffing.

The calculated scuffing temperature may be modified further by using data from test gears and an empirical welding factor. Per AGMA 925-A03 (2003),

$$\theta_s = X_w \theta_{fl,\max test} + \theta_{M,test} \quad (11)$$

where:
X_W = Welding factor
$\theta_{fl,\ max\ test}$ = Maximum flash temperature of test gears, °C
$\theta_{M,\ test}$ = Tooth temperature of test gears, °C

Table 1 shows the welding factor for various materials.

Table 1. Welding Factors.

Material	X_W
Through hardened steel	1.00
Phosphated steel	1.25
Copper-plated steel	1.5
Bath or nitrided steel	1.5
Hardened carburized steel	
- Less that 20% retained austenite	1.15
- 20 to 30% retained austenite	1.00
- Greater than 30% retained austenite	0.85
Austenite steel (stainless steel)	0.45

The scuffing risk is then determined using a probability analysis that is described in AGMA 925-A03 (2003). In general, the greater the margin of tooth temperature to scuffing temperature, the lower the risk of scuffing. Other methods of determining the risk of scuffing include the integral temperature method, the pressure-volume-temperature (PVT) method, Borscoff scoring factor method, and the simplified scuffing criteria for high-speed gears. Gear manufacturers do not use these methods as commonly as Blok's (1937) contact theory.

Wear

Wear is defined as the removal or displacement of metal from gear tooth surfaces. Damage due to scuffing or pitting is not normally considered wear. Wear reduces tooth thickness and can change the contour of the teeth. Wear can result from mechanical, chemical, or electrical action. For gears that operate in lubrication Regime 1 and 2, wear is inevitable. For well-designed gears that operate in Regime 3, the rate of wear should be so slight that the original machining marks should still be visible after a year or more of service. This low rate of wear also assumes that the supply oil is well filtered and that there is no abrasive wear due to hard particles suspended in the oil. Recall that in lubrication Regime 3, there is no metal-to-metal contact because the oil film thickness is greater than the asperities on the gear tooth surfaces. The specific film thickness is the ratio of the oil film thickness divided by the composite roughness of the contacting gear teeth. So, all gears that operate in Regime 3 have a specific film thickness greater than 1.0, and the larger the specific film thickness, the lower the risk of wear. However, due to complex interactions between the metal surfaces, oil, oil additives, and the atmosphere, wear can still occur even if the specific film thickness is greater than 1.0. It is difficult to determine the risk of gear wear from a single parameter. However, AGMA 925-A03 (2003) provides a curve based on pitch line velocity and specific film thickness that provides some guidance on the risk of wear.

$$\lambda_{(i)} = \frac{h_{c(i)}}{\sigma_x} \quad (12)$$

where:
$\lambda_{(i)}$ = Specific oil film thickness
$h_{c(i)}$ = Oil film thickness, µm
σ_x = Composite surface roughness, µm

The composite surface roughness can be calculated from roughness measurements on gear tooth surfaces. The equation for oil film thickness dates back to testing that was done in the 1950s and 1960s. Per AGMA 925-A03 (2003), it takes into account "…the exponential increase of lubricant viscosity with pressure, tooth geometry, velocity of the gear teeth material elastic properties, and the transmitted load." The oil film thickness will vary across the line of action, and the equation given in AGMA 925-A03 (2003) is:

$$h_{c(i)} = \frac{3060\rho_{n(i)}\left(\mathit{Materials\ parameter}\right)^{0.56}\left(\mathit{Speed\ parameter}\right)_{(i)}^{0.69}}{\mathit{Load\ parameter}_{(i)}^{0.10}} \tag{13}$$

where:
$\rho_{n(i)}$ = Normal relative radius of curvature, mm

Unless the gears are very highly loaded, the unit loading does not have a large impact on the oil film thickness. The rolling velocities and radius of curvature have the biggest impact. Because most high-speed gears operate at high speeds (large speed parameter) and low to moderate loads (low load parameter), the oil film thickness is normally quite large and the specific film thickness is much greater than 1.0. In most cases the probability of wear-related distress is very low.

Micropitting

Micropitting is a high rolling contact fatigue incident that occurs in the Hertzian contact band area. It is a function of combined rolling and sliding velocities, load, temperature, specific film thickness, and the lubricant itself. Per AGMA 925-A03 (2003), the pits are typically 10 to 20 µm (.0004 to .0008 inch) deep by about 25 to 100 µm (.001 to .004 inch) long and 10 to 20 µm (.0004 to .0008 inch) wide. Because of the size of the pits, micropitting can be difficult to see with the unaided and/or untrained eye. If micropitting progresses far enough, the micropits will coalesce and the surface will take on a dull, matte appearance (Figure 7).

Figure 7. Micropitting. (Courtesy ANSI/AGMA 1010-E95, 1995)

Micropitting can lead to a reduction in gear tooth accuracy, which can increase gear tooth loading, vibration, and noise. Micropitting can lead to macropitting and gear tooth breakage. Per Cardis and Webster (2000), the material from the pits themselves may also be small enough to get past the oil filter but large enough to cause abrasive wear to gear tooth surfaces.

One problem is that there is a basic lack of understanding of the mechanism of micropitting. One theory is that micropitting starts when the asperities on gear tooth surfaces carry a significant portion of the load. These asperities then deform, which produces local residual tensile stresses. The cyclical loading is then high enough to cause local fatigue cracks that take on the form of small pits. Thus, surface roughness is a big factor in the risk of micropitting. Per AGMA 925-A03 (2003), there have been cases where micropitting was eliminated when the gear tooth surfaces were finished to a mirror-like finish.

Micropitting can occur anywhere on the gear tooth surface. However, per Cardis and Webster (2000), it generally starts in areas associated with high sliding velocity, which is in the bottom or top of the tooth profile, not at the pitch line where the sliding velocity is zero.

Lubricants also play a key aspect in the risk of micropitting. Studies done by Cardis and Webster (2000) have shown that micropitting is more apt to occur in gears that use oils with antiscuffing additives. Also, micropitting resistance tends to decrease with higher gear tooth temperatures, but it has been reported that other additives actually improve micropitting resistance at higher temperatures. The only true method to understand micropitting is to run a micropitting test at the actual gear tooth temperature.

Some gear manufacturers use their own criteria for determining the risk of micropitting. Other manufacturers have not because they have not had any problems with micropitting. The test reported in FVA (the German Research Association for Drive Technology) Information Sheet 54/I-IV has gained acceptance from some gear manufacturers and end users, but this test is very controversial.

Macropitting

Macropitting is also a surface fatigue phenomenon. These pits are typically on the order of 0.5 to 1.0 mm (.02 to .039 inch) in diameter and are large enough to be seen by the unaided eye (Figure 8). Macropits typically occur if there are high asperities and metal-to-metal contact occurs between meshing teeth. However, in high-speed gears, the surface finish is typically very smooth and the oil film is typically thick enough to prevent metal-to-metal contact. In these cases, the cause of the macropits is generally an inclusion or small void in the material that provides the initiation point for the crack, and the subsurface shear stresses propagate the crack. The key is that macropits in well-designed, high-speed gears are typically a result of some material defect or operating the gear set at power levels greater than design. ANSI/AGMA 2101-C95 (1995) provides a calculation method for determining the pitting resistance power rating.

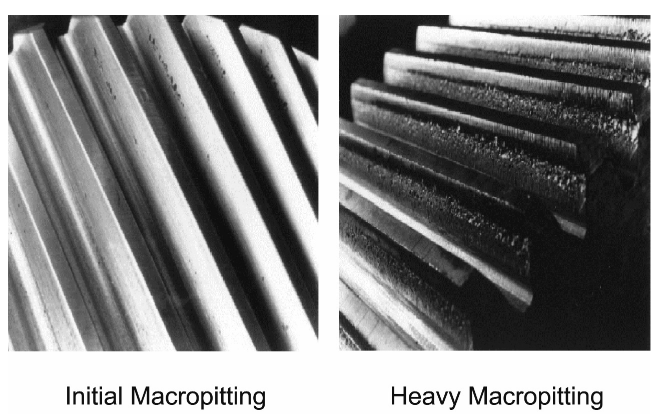

Figure 8. Macropitting. (Courtesy ANSI/AGMA 1010-E95, 1995)

THERMAL PROBLEMS

One of the other purposes of lubricating oils is to cool the gear tooth surfaces. Testing done by Martinaglia (1972) showed that high-speed helical gear teeth could distort due to nonuniform temperature distribution along the face width of the gearing. In many cases the working flanks of the pinion and gear have to be modified to account for this distortion. At very high pitch line velocities, problems can also occur with overheating of the air/oil mixture trapped in the gear mesh. Varnish and carbon deposits can also form on gear tooth surfaces. These deposits can close up the backlash and cause local overloading and gear tooth failure.

Dudley (1984) proposed using axial meshing velocity as criteria for evaluating the risk of thermal problems. He suggested that thermal distortions could typically be managed at axial meshing velocities up to about 700 m/s (2297 ft/s) if the gears are not too big and the tooth contour is properly corrected. Martinaglia (1972) reported that thermal distortions become a significant problem at pitch line velocities above 120 m/s (394 ft/s), powers of at least a few megawatts, and a face width-to-pitch diameter ratio of 0.8 of more.

The author's company has experience with excessive varnishing and gear tooth failures with a gear set that operates at a pitch line velocity of 148 m/s (486 ft/s), an axial meshing velocity of 667 m/s (2188 ft/s), and a face width-to-pitch diameter ratio of 1.41. The company also performed some crude testing with this gearing. This gear set is installed in an operating integral gear compressor. The gearing is not fitted with temperature detectors in the gear teeth. After shutting down the compressor, the pinion tooth temperatures were measured with an infrared detector. The first temperature could not be taken until the machine stopped rotating and the oil pump shut off. So, the first reading was taken 550 seconds after shutdown. By extrapolating back, the hottest gear tooth temperature during operation was estimated to be 107°C (225°F). This is well below the temperature where varnishing would be expected. The conclusion was that the varnishing was due to overheating of the air oil mixture, not excessive tooth temperatures. This particular gear had a higher face width-to-pitch diameter ratio than other gears made by the manufacturer at similar pitch line velocities and axial meshing velocities. So, understanding a supplier's experience can be a crucial step in evaluating the risk of thermal problems.

GEAR SPRAY ARRANGEMENTS

Oil is typically sprayed into the gear mesh from a spray bar. The spray bar usually extends across the width of the gears and is fitted with small nozzles that are used to create small oil jets that spray oil into the mesh. Based on a survey of several gear manufacturers, there is no universal agreement on whether oil should be sprayed into the in-mesh, the out-mesh, or both. It depends on the experience of the gear manufacturer. It is only important that the gear teeth are wetted with oil for proper lubrication and that enough oil gets into the tooth spaces for sufficient cooling.

Some manufacturers spray into the in-mesh at low pitch line velocities and into the in-mesh and out-mesh at high pitch line velocities. One theory is that oil spray into the in-mesh is better for lubrication and into the out-mesh is better for cooling. Generally, if oil is sprayed into the in-mesh and out-mesh, only 15 to 25 percent is sprayed into the in-mesh. Based on the testing done by Akazawa and others (1980), the best method is to only spray oil into the out-mesh. Based on their tests, spraying oil into the out-mesh resulted in 10°C (50°F) lower tooth temperatures than spraying oil into the in-mesh with a gear set operating at a pitch line velocity of 140 m/s (459 ft/s). Spraying oil into the in-mesh and out-mesh resulted in the same tooth temperatures as spraying oil into the out-mesh, but with this arrangement the power losses were higher, presumably due to additional churning losses of the oil.

Some gearbox manufacturers use special baffles to reduce windage losses inside gearboxes. Air inside the gearbox is accelerated because of the high rotational speed of the gears. The energy that it takes to accelerate the air is a loss. By fitting plates close to the sides of the gears, the amount of air that is accelerated is limited and this loss is reduced. These plates can also be used as shields to prevent oil that is squeezed out of bearings from hitting the gears. This oil would also be accelerated, which would result in an additional loss. Finally, some gear manufacturers use a false bottom in the gearbox. This is typically a perforated plate that is fitted between the gears and the bottom of the gearbox. It is used to help the oil drain by preventing the oil in the bottom of the gearbox from being lifted back up and reaccelerated.

The effectiveness of gearbox windage baffles/shields and false bottoms depends on the specific gearbox design. Some manufacturers have found these devices to be effective in improving gearbox efficiency and others have not. The author's company experimented with these devices on a gearbox in LaSalle, Illinois. The result was a small reduction in bulk oil return temperature but higher gear tooth temperatures. Without knowing the specific oil flow, it was impossible to calculate or predict the exact reduction in gearbox power, but it was small enough that this effort was discontinued.

SPECIAL DESIGNS

Greiner and Langenbeck (1991) performed testing with separate lubrication and cooling oil supply to understand the influence of this arrangement on gear temperatures (scuffing load capacity) and on efficiency. One set of nozzles was used to spray either the in-mesh or out-mesh for lubrication and another set of nozzles was used to spray the gear webs for cooling. Temperature, power loss, and wear measurements were taken for a multitude of spray arrangements, oil flows, and flow splits. Based on the extensive testing, the separation of lubrication and cooling oil supply led to a 60 percent reduction in oil flow (compared with the original recommended oil flows), lower gear tooth temperatures, and a slight increase in efficiency.

One company also has a patented gearbox that operates in a vacuum and virtually eliminates windage losses. This invention has reportedly lowered the gearbox power loss by up to 50 percent in comparison to conventional gearboxes. One documented case was a comparison of two 90 MW gearboxes: one with conventional gearing and one with the vacuum gearbox. The calculated losses in the conventional gearbox at full load were 1407 kW, while the losses in the vacuum gearbox were measured at 628 kW. This is a difference of 779 kW, or about 1 percent of the rated power.

Several companies also manufacture very high-speed helical gears with special axial grooves such as those shown in Figure 9. These grooves provide a path for the hot air/oil mixture to escape before overheating and allow fresh, cool oil to be supplied to the gear mesh at an intermediate point in the mesh. Typically, these grooves are added on the bullgear and sometimes on the bullgear and the pinion. Although Figure 9 shows only a single groove, some designs have two or three rows of grooves.

Some very high-speed gearboxes are also equipped with a special high-pressure/high-velocity-type spray arrangement that is fitted close to the gear mesh (Figure 10). As stated above, for proper cooling it is important for the oil to cover the tooth surfaces for a slight time interval before being flung off. At high pitch line speeds, higher velocity oil may be needed to get sufficient cooling oil into the tooth spaces before being stuck by the next tooth and flung out of the gear mesh. Generally, much more oil is required for cooling than for lubrication. So, while there may be enough oil on the gear tooth surfaces for lubrication, there might not be enough for cooling.

MODERN OILS

While most high-speed helical gears can operate reliably for many years with conventional turbine oils, there are some cases

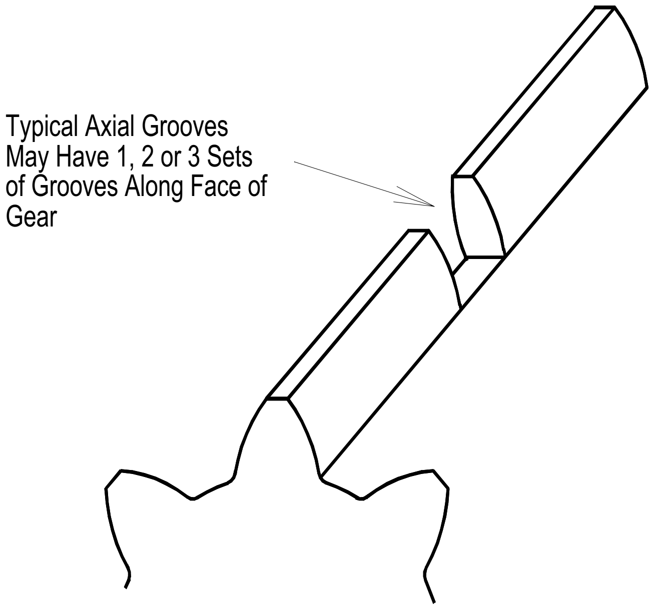

Figure 9. Axial Grooves.

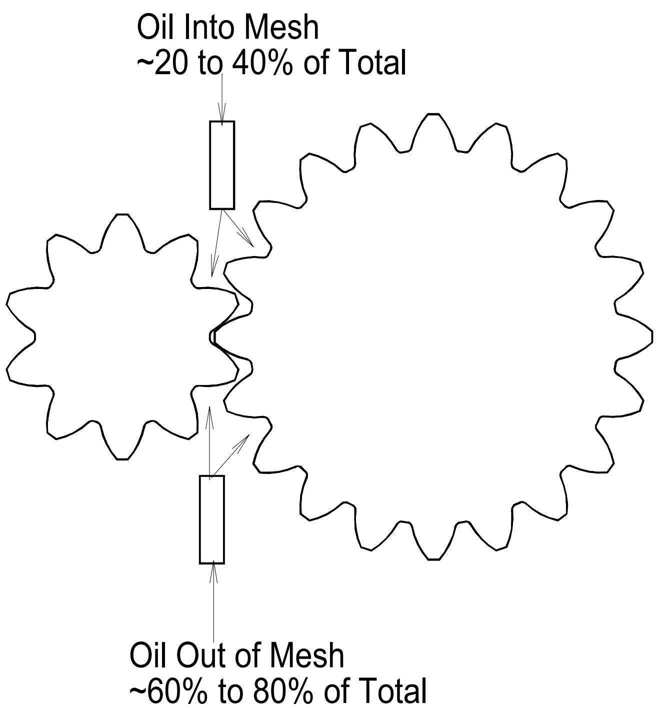

Figure 10. High Pressure Gear Spray.

where special oils can help. If gear teeth show low to moderate varnishing, it may be possible to correct the problem by changing to synthetic oil with a higher oxidation temperature.

Operating gears above design power can also cause problems such as micropitting. The author's company has operated a number of high power gears 20 percent above the original design power levels to increase plant capacity. On some of these gears, the beginning of micropitting was discovered. Changing to synthetic oils with antiwear additives appears to have stopped the progression of this pitting.

Oil plays a key part in the design and reliability of a gearbox system. While most of the time conventional turbine oils are more than capable of providing the required reliability, there are cases in which special oils can be used to solve unique problems.

CONCLUSIONS

There are many good sources available to end users to assist in the evaluation of high-speed helical gear designs. Although lubrication and cooling problems are rare, these problems can be very difficult to solve. When evaluating high-speed helical gears, it is important to understand the manufacturer's experience. Special attention should be paid to the design and experience of any high-speed helical gear whenever the pitch line velocity is above 155 m/s (509 ft/s), the face width-to-diameter ratio exceeds 0.8, and the power level is above 1 MW. Axial meshing velocity and face width need to be considered together when evaluating a gear design. As discussed, lubrication and cooling are complex issues and a number of parameters need to be considered.

NOMENCLATURE

a_w = Center distance, mm
b = Face width, mm
$b_{h(i)}$ = Semi-width of the Hertzian band contact, mm
B_{M1} = Thermal contact coefficient of the pinion material, N/[mm $s^{0.5}$K]
B_{M2} = Thermal contact coefficient of the gear material, N/[mm $s^{0.5}$K]
C_{avgx} = Surface roughness constant, mm
$h_{c(i)}$ = Oil film thickness, µm
K = Numerical factor for a semi-elliptic distribution of heat
n_1 = Pinion speed, rpm
n_2 = Gear speed, rpm
r_1 = Pitch radius of pinion, mm
r_2 = Pitch radius of gear, mm
R_{avgx} = Average surface roughness of pinion and gear, µm
R_{a1x} = Average surface roughness of pinion, µm
R_{a2x} = Average surface roughness of gear, µm
u = Gear tooth ratio
X_W = Welding factor
z_1 = Number of pinion teeth
z_2 = Number of gear teeth
β = Helix angle, degrees
$\lambda_{(i)}$ = Specific oil film thickness
σ_x = Composite surface roughness, m
$\theta_{fl(i)}$ = Flash temperature, °C
θ_s = Scuffing temperature, °C
$\theta_{fl,\ max\ test}$ = Maximum flash temperature of test gears, °C
$\theta_{M,\ test}$ = Tooth temperature of test gears, °C
$\rho_{n(i)}$ = Normal relative radius of curvature, mm
$\mu_{m(i)}$ = Mean coefficient of friction
v_t = Pitch line velocity, m/sec
$v_{rl(i)}$ = Rolling tangential velocity of pinion, m/sec
$v_{r2(i)}$ = Rolling tangential velocity of gear, m/sec
v_{40} = Kinematic viscosity at 40°C, mm^2/sec
ω_n = Normal unit load, N/mm
$X_{r(i)}$ = Load sharing factor

REFERENCES

AGMA 925-A03, 2003, "Effect of Lubrication on Gear Surface Distress," First Edition, American Gear Manufacturers Association, Alexandria, Virginia.

AGMA 9005-E02, 2002, "Industrial Gear Lubrication," First Edition, American Gear Manufacturers Association, Alexandria, Virginia.

ANSI/AGMA 1010-E95, 1995, "Appearance of Gear Teeth—Terminology of Wear and Failure," First Edition, American National Standards Institute, Washington, D.C./American Gear Manufacturers Association, Alexandria, Virginia.

ANSI/AGMA 2101-C95, 1995, "Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth—Metric Edition," First Edition, American National

Standards Institute, Washington, D.C./American Gear Manufacturers Association, Alexandria, Virginia.

Akazawa, M., Tejima, T., and Narita, T., 1980, "Full Scale Test of High Speed, High Powered Gear Unit—Helical Gears of 25,000 PS and 200 m/s PLV," ASME Paper Number 80-C2/DET-4.

Blok, H. 1937, *Theoretical Study of Temperature Rise at Surfaces of Actual Contact Under Oiliness Lubricating Conditions*, Institution of Mechanical Engineers, London, United Kingdom.

Cardis, A. B. and Webster, M. N., September/October 2000, "Gear Oil Micropitting Evaluation," *Gear Technology*, pp. 30-35.

Dudley, D. W., 1984, *Handbook of Practical Gear Design*, New York, New York: McGraw-Hill Book Company.

Errichello, R., May/June 1991, "The Lubrication of Gears—Part II," *Gear Technology*, pp. 18-22.

Greiner, J. and Langenbeck, K., 1991, "Separation of Lubrication and Cooling in Oil-Jet Lubricated Gears," AGMA Paper Number 91 FTM 13.

Martinaglia, L., 1972, "Thermal Behavior of High Speed Gears and Tooth Corrections for Such Gears," ASME/AGMA International Symposium on Gears and Transmissions, San Francisco, California.

ACKNOWLEDGEMENT

The author wishes to thank Chip Gallagher and Ken Beckman for their assistance in reviewing this paper. The author would also like to thank Ed Martin and Klaus Gaiser for their contributions to this paper.