Designing Reliability Into Industrial Gear Drives

Peter A. Mayo

Introduction

The primary objective in designing reliable gear drives is to avoid failure. Avoiding failure is just as important for the manufacturer and designer as it is for the end user. Many aspects should be considered in order to maximize the potential reliability and performance of installed gearing.

This article is intended to provide some insight into the important elements applied to the design and production of industrial gearing and how the reliability of the gear and drive train is influenced from such measures. Fortunately these days, the gear designer and gear manufacturer have some very sophisticated tools at their disposal to achieve these goals.

There are many gear design codes in use worldwide, including AGMA & DIN standards. The long-awaited ISO standard for gears has recently been approved after more than 20 years in the making. While the gear design codes provide formulas for the determination of various parameters, these equations do not yield a unique or definitive solution.

The actual design process proceeds by the intuitive selection of parameters by the experienced gear designer, who then applies the design code to establish compliance with certain criteria. Regardless of the standard employed, the gear design codes share the common objective of assessing the ability of the teeth to resist surface pitting and cracking when subjected to cyclic loads. The standards also provide guidelines to avoid surface damage to the active tooth flanks by scoring due to inadequate lubrication.

Tooth distress due to pitting is a manifestation of excessive contact (Hertzian) stress. Even more significant is the development of cracks in the critical tooth root fillet region when the tooth bending stress exceeds the endurance strength.

Gears may fail by other means such as wear, plastic flow, case crushing, quench cracks and corrosion, but these modes are not so readily derminant or predictable. Also, more than just the tooth design affects the reliability of the gearing. Other factors influencing reliability include lubrication, construction, the characteristics of the prime mover, bearings, application, assembly and maintenance.

Theory

The power capacity of gears is most often referred to as the gear rating. In order to understand the fundamental design criteria, a brief explanation as to the origins of the rating equations is appropriate. The following derivation should serve as an adequate introduction to the subject, but the reader should refer to Ref. 1 for more information.

For spur and helical gears, the basic equation for assessing the pitting resistance of two engaging teeth is based on the simple analogy of two cylinders of length F pressed together under load W_t, as shown in Fig. 1.

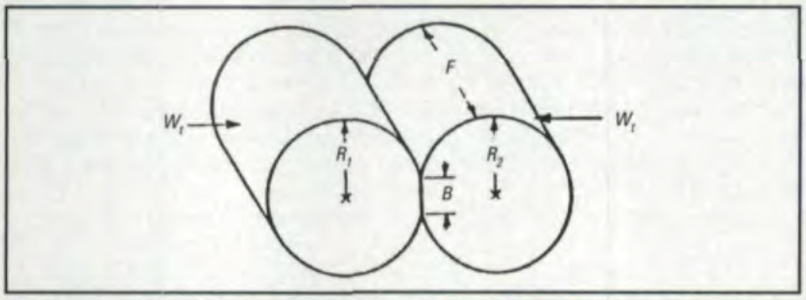

Fig. 1 — Parallel cylinders in contact and heavily loaded. Courtesy of Technomic Publishing (Ref. 1).

The Hertzian stress for the band of contact is given by

$$B = \sqrt{\frac{16\, W_t (K_1 + K_2) R_1 R_2}{F\,(R_1 + R_2)}}$$

where

$$K_1 = \frac{1 - \nu_1^2}{\pi E_1}$$

$$K_2 = \frac{1 - \nu_1^2}{\pi E_2}$$

The maximum compressive stress is

$$S_c = \frac{4\, W_t}{F \pi B}$$

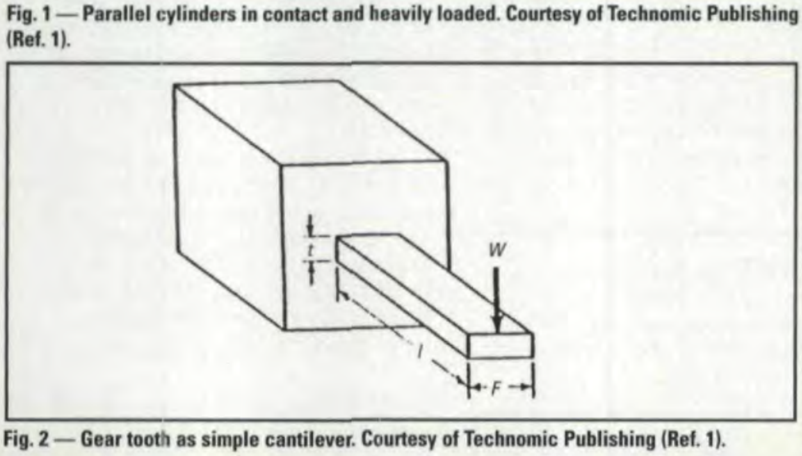

Fig. 2 — Gear tooth as simple cantilever. Courtesy of Technomic Publishing (Ref. 1).

These two equations may be combined with Poisson's ratio as $\upsilon = 0.3$ to give

$$S_c = \sqrt{0.35 \frac{W_t}{F} \frac{(1/R_1 + 1/R_2)}{(1/E_1 + 1/E_2)}}$$

The radius of curvature ρ at the pitch circle d is

$$\rho = \frac{d \sin \phi}{2}$$

Radius of curvature for the pinion is $R_1 = \frac{d \sin \phi}{2}$ and for the gear

$$R_2 = m_G R_1$$

The compressive stress at the pitch line of a pair of spur gears is therefore:

$$S_c = \sqrt{\frac{0.70}{(1/E_1 + 1/E_2) \cos\phi \sin\phi}} \quad \sqrt{\frac{W_t}{Fd}\left(\frac{m_G + 1}{m_G}\right)}$$

Which is the same as the fundamental rating formula for pitting from AGMA Standard 2001-1995 except for the addition of some derating factors to take account of the uncertainties that prevail in the real world situation over that of the theoretical.

$$S_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d\ F} \frac{C_f}{I}}$$

From AGMA Standard 2001-1995, the pitting resistance power rating is given by:

$$P_{ac} = \frac{n_p F}{126\,000} \frac{I}{K_o K_v K_s K_m C_f} \left(\frac{d\, S_{ac}}{C_p S_H} \frac{Z_N C_H}{K_T K_R}\right)^2$$

where

P_{ac} is the allowable transmitted power for pitting
n_p is the pinion rotational speed (rpm)
F is the net face width of the narrowest member
I is the geometry factor for tooth pitting resistance
K_o is the overload factor
K_v is the dynamic factor
K_s is the size factor
K_m is the load distribution factor
K_R is the reliability factor
K_T is the temperature factor
C_f is the surface condition factor
C_H is the hardness ratio factor
C_p is the elastic coefficient factor
d is the operating pitch circle diameter of the pinion
Z_N is the stress cycle factor for pitting resistance
S_{ac} is the allowable contact stress number
S_H is the safety factor

Similarly, the tooth bending strength considers the teeth as essentially short cantilever beams. The strength of gear teeth was conceived by Wilfred Lewis in 1893 by inscribing a parabola inside the outline of a gear tooth as shown in Fig. 3. By doing this, the stress along the surface of the parabola is constant.

The location (a) on the tooth where the largest inscribed parabola is tangent to the root fillet region determines the position of maximum stress.

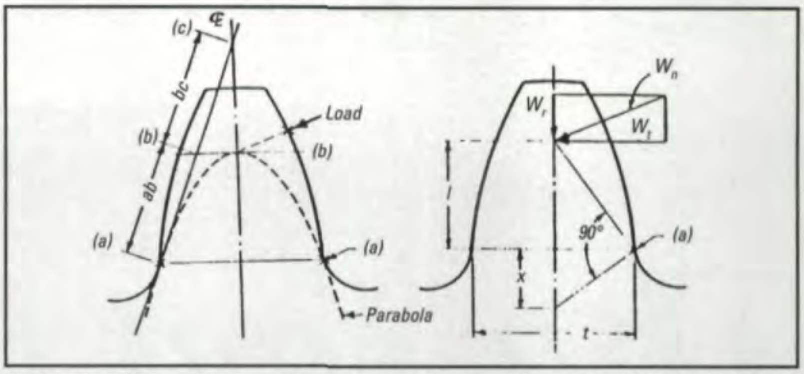

Fig. 3 — Inscribed parabola on gear tooth section. Courtesy of Technomic Publishing (Ref. 1).

The tensile stress at the base of a cantilever is:

$$S_t = \frac{6Wl}{Ft^2}$$

From Figure 3:

$$x = \frac{t^2}{4l}$$

By substitution:

$$S_t = \frac{W}{F(2x/3)}$$

The circular pitch may be introduced into the equations to give:

$$S_t = \frac{Wp}{F(2x/3)p}$$

The term $2x/3p$ was called y by Lewis. This was a paramater that could be determined from a layout of the tooth. With the addition of a stress concentration factor and other aspects, the term is now referred to as the geometry factor J. The latter can be determined with very little effort nowadays using various gear geometry software programs.

The Lewis Formula is written as

$$s_t = \frac{W}{Fpy}$$

Replacing y with Y/π and then diametral pitch P_d for π/p, the Lewis equation becomes

$$s_t = \frac{WPd}{Fy}$$

which is the fundamental rating formula for bending strength from AGMA 2001 (with the addition of some derating factors).

The fundamental rating formula for bending strength from AGMA 2001 is as follows:

$$s_t = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J}$$

Accordingly, the bending strength power rating from AGMA 2001 is given by:

$$P_{at} = \frac{n_p\, d}{126000\ K_o K_v} \frac{F}{P_d} \frac{J}{K_s K_m K_B} \frac{s_{at} Y_N}{S_F K_T K_R}$$

where

P_{at} is the allowable transmitted power for bending strength
n_p is the pinion rotational speed (rpm)
d is the operating pitch circle diameter of the pinion
F is the net face width of the narrowest member

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J is the geometry factor for tooth bending strength
K_o is the overload factor
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K_R is the reliability factor
K_T is the temperature factor
S_{at} is the allowable bending stress number
Y_N is the stress cycle factor for pitting resistance
S_F is the safety factor for tooth bending strength

From these two fundamental rating formulas, the important elements can be considered in terms of dynamic effects, material properties, loading characteristics and tooth geometry. There are also some basic differences between straight spur gears and helical gears to consider.

The way these attributes are chosen, applied and controlled during design and manufacture can significantly influence the reliability of the gears. Further elaboration of these aspects is therefore most relevant to the subject at hand.

Gear Selection

Single helical gearing provides significant advantages over spur gears. It is the meshing of the helical teeth along multiple contact lines inclined at an acute angle to the pitch line that contributes most to the ability of helical gears to transmit more load than straight spur gears.

Moreover, smooth transfer of load occurs gradually and uniformly by a combined sliding and rolling action as successive teeth come into contact along the engaging helicoidal surfaces. This situation is usually referred to as helical overlap. The helical overlap ratio is equal to the ratio of the face width to the axial pitch. Similarly, the ratio of the length of action (length of engagement) of the meshing gear teeth to the transverse base pitch is the transverse contact ratio.

The choice of helix angle for low speed gears is usually a balance between minimizing the axial thrust and maintaining a helical overlap ratio of at least 1.1. This means that the face width of the gear is at least 10% wider than the axial pitch and ensures that before a tooth begins to leave the mesh, the next tooth has already begun to take some share of the load.

Spur gears, on the other hand, rely totally on the conjugacy and contact ratio of the meshing involute tooth forms for the smooth transmission of the load. The average number of teeth in mesh (transverse contact ratio) is usually about 1.2 to 1.7 for both spur and helical gears. Helical gears have typically twice this average amount of teeth in mesh since the overall contact ratio of helical gears consists of the helical overlap ratio plus the transverse contact ratio.

Herringbone or double helical gears typically have helix angles from 20° to 30°. This results in a larger number of teeth in mesh in any given instant and hence high helical overlap ratio. The gears then operate smoothly and are much more tolerant of tooth variations.

Double helicals can, however, be sensitive to variations in accumulative pitch between the two helices that are not synchronized. The floating member otherwise tends to shuttle back and forth, or due to its inertia, the dynamic tooth loads become amplified to the detriment of the gear drive.

The critical tooth bending load for spur gears can occur either at the tip or close to it at the highest point of single tooth contact, but for normal helical gears, the load is designed to be distributed evenly over the oblique helical contact line (or in reality, finite width contact band due to elastic material properties) which extends from the bottom of the tooth to the tip as shown in Fig. 4.

By virtue of their load sharing ability, helical gears can have approximately 50% greater load carrying capacity than the equivalent spur gear of the same physical size when rated to AGMA standards.

Materials

The type and choice of materials obviously plays a vital role in the design and performance of gears. Steel of one type or another tends to be favored for gear materials because it has a high strength to cost relationship. Since gear teeth are subjected to cyclic loads, the fatigue strength rather than the normal mechanical strength determines the allowable design stresses.

The allowable stresses (S_{ac} and S_{at}) are influenced by many factors including hardness, chemistry, cleanliness, residual stress, microstructure, quality, heat treatment, processing practices and number of stress cycles.

There has been a trend over the years towards ever increasing gear tooth hardness. The reason for this is simple. Generally, the harder the material and tooth surface employed, the greater the resis-

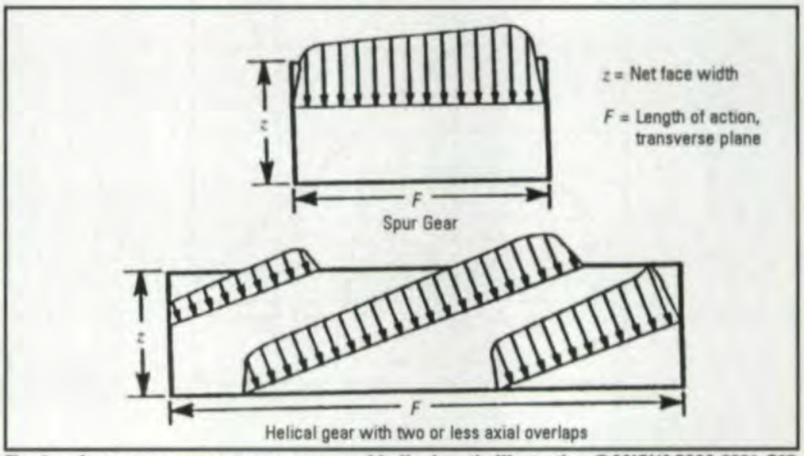

Fig. 4 — Instantaneous contact on spur and helical teeth. Illustration ©ANSI/AGMA 2001-C95.

tance to pitting and tooth bending fatigue. This allows gears to be made smaller in size for a given torque. A reduction in diameter reduces pitchline speeds, and reductions in face width increase the prospects for good gear tooth alignment.

Due to the nonlinearity of the allowable stresses as a function of hardness, the pitting resistance of through hardened gearing varies exponentially with hardness to the power of 1.6. On the other hand, the tooth bending strength varies to the power of only 0.6. The latter explains why increased tooth bending strength is usually achieved by a larger tooth size or greater face width.

Gear steels typically have carbon contents in the range of 0.3% to 0.5%, and they are alloyed to enable the desired hardness on the flanks and roots of the teeth. Below 0.3% carbon the gears have poor wear resistance. The 0.3% carbon alloy steels produce the greatest toughness compared with higher carbon steels, but 0.4% carbon has higher hardness potential. Above 0.5% the toughness of these steels tends to be quite low.

The AGMA gear rating codes assign allowable design stresses based on the verification undertaken to confirm the cleanliness, quality, homogeneity and integrity of the materials employed. It should be noted that for a given hardness, the selection of allowable stress must be made from a design range.

The lower values for allowable stress might be appropriate for castings which are relatively free of harmful defects, but which may contain some innocuous defects such as discrete gas holes and porosity, but not cracks and shrinkage.

Intermediate values for allowable stress may be applied to commercial grade forgings and maximum values for extra high quality steel forgings, such as those offered by the Electro Slag Refined steel making process. Vacuum degassing has also been a steel making process of significant benefit to improving gear steel quality.

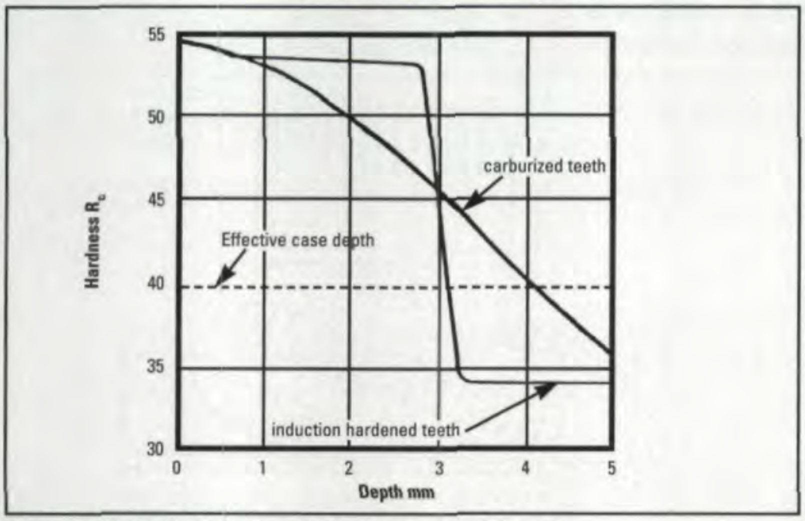

Fig. 5 — Induction hardened & carburized case core relationship.

The most important part of a gear contains the teeth, and so it often pays in the long run to invest in the best material available. For cyclically loaded components such as bearings, it has been well established that cleaner steels improve the life expectancy. The ESR steels boast extremely clean microstructures, which are inherently very resistant to the initiation of fatigue.

Lower alloy steels can be employed where low hardness will suffice or when the teeth can be gashed prior to heat treatment. The structure of the gear should also maximize the potential of the heat treatment. In order to encourage quicker quenching rates, the sides of large solid gears or pinions can also be dished to reduce the mass effect.

Even with high hardenability alloy steels, the teeth of coarse pitch, through hardened gears benefit significantly when practical from pregashing the teeth prior to heat treatment. This ensures uniform and effective hardening to flank and root when the hardness required is greater than 300 HB.

These days, foundry facilities and expertise have been developed to the point where even large gear castings can now be water quenched to produce a very tough and much improved fatigue resistant martensitic microstructure not previously possible by air hardening (normalizing), which produces a bainitic microstructure.

Surface Hardened Gearing

The pursuit of ever harder gears has seen the emergence and now dominance of case carburized gears in gearboxes. Typical hardnesses for such gears would be 58 to 62 Rc, so the teeth are invariably precision ground after hardening. The gears therefore boast high accuracy, high surface load carrying capacity and smoother operation.

As a consequence of surface hardening gear teeth, beneficial residual compressive stresses are induced into the surface layer. This effectively reduces the tensile bending stresses in the critical tooth root fillet region. It is most important therefore not to grind the roots of the teeth as the grinding can remove the residual compressive stress. Shot peening, however, can be employed to restore the residual stress if grinding becomes necessary to remove excess distortion.

Where the gearwheel itself might be too large to be case hardened, a significant benefit can still be gained by the use of a case hardened pinion meshing with a through hardened gearwheel. This has evolved even though pinions typically need only 20% to 40% hardness differential over and above that of the gearwheel to account for the greater number of stress cycles experienced by the pinion. The tooth hardness of the gearwheel might typically range from 300 HB up to 400 HB.

The benefit of this combination may be explained in terms of the ability of alloy steel gear teeth to strain harden under the influence of cold work by the much harder surface of the engaging pinion teeth. AGMA Standard 2001 recognizes this phenomenon and applies the hardness factor C_H to achieve a rating gain corresponding to the subsequent increased hardness. Actual measurements would suggest that alloy steels can increase the active tooth surface hardness by as much as 15% in this action.

Even though the carburized teeth can develop beneficial residual compressive stresses in the surface layer, the drawback experienced with producing carburized gear teeth is the volumetric expansion that occurs when the tooth surface is enriched with carbon and subsequently transformed to martensite when quenched. The associated distortions require further finishing in order to correct these distortions.

These movements can be quite uneven. Single helical teeth tend to unwind slightly, and long pinions can develop an hourglass shape. Therefore, subsequent finishing removes more material than expected, and thin localized case depths can occur.

Testing of the surface hardened layer is an important issue. Some tests, such as hardness, can correlate the success of the hardening process on the actual gear teeth with the test piece. Most of the testing, though, must be done on test samples independent of the actual work piece. It is therefore essential that the test samples be totally representative of the gear.

Induction hardening offers an alternative means to provide high surface hardness and increased strength in the root of gear teeth. Hardnesses typically 50 to 56 Rc can be readily achieved depending on the carbon equivalent, the type of steel and the quenching rates employed. The advantage of induction hardening is considerably less distortion than with carburizing.

However, induction hardening requires a heavier case thickness. This is because the induction hardened case has a more abrupt transition from case to core in contrast to the more gradual transition of carburized surface layers. This is illustrated in Fig. 5. Induction hardened case depth is also measured differently.

In specifying the case depth, it is important to ensure that the maximum subsurface shear stresses do not occur at this transition zone.

Nitriding also provides a very useful surface hardening technique where very high hardnesses are needed. It is limited, though, where only a thin surface layer can be used such as that found in small and high speed gears.

Using a very hard pinion is contrary to the practice of using a sacrificial pinion with particularly soft teeth. The disadvantage of allowing the pinion teeth to become worn and misshapen is uneven wear on the gearwheel teeth.

Very hard pinion teeth do not yield or suffer so easily from the usual perils of through hardened surfaces such as pitting, spalling and scoring (assuming adequate lubrication). As a result, the pinion can then be expected to always maintain its true involuted shape and in turn help to maintain the tooth profile of the gearwheel. The gearwheel teeth might otherwise suffer if the pinion tooth form became distorted for any one of the forgoing reasons.

Dynamic Considerations

The dynamic factor K_v takes into account the internally generated gear tooth loads induced by non-conjugate action (non-uniform motion) of the meshing teeth. Dynamic forces arise from the relative accelerations between the gears as they vibrate in response to excitation referred to as "transmission error."

Ideally, gears should have uniform transfer of motion from input to output gear. It is impossible to produce perfectly true gears, and it is the departures from the ideal geometry that contribute to non-uniform motion and transfer of load. These deviations can be caused by many factors including residual stresses in materials, variations in material metallurgy and hardness, poor machine tool condition, inaccurate tooling, poor machine setting practice and many more. Stiffness of the teeth and mesh also contribute to non-uniform motion.

The common source of transmission error occurs from variations in the elemental tooth parameters. These individual elemental parameters include the tooth spacing (pitch), involute profile, tooth alignment (helix) and runout (eccentricity).

Gear metrology is used to identify and quantify the various elemental parameters of the teeth and gear body. The somewhat peculiar geometry of gear teeth requires some unique measuring methods and facilities. In addition to verifying compliance with allowable tolerances, gear metrology can be used as a diagnostic tool to identify the source and cause of the deviations.

The objective is therefore to minimize the deviations by taking appropriate action during design and manufacture. In some situations, the deviations can be removed by subsequent processes such as grinding. In many cases, though, the deviations remain in the installed gearing. By improving gear accuracy, the dynamic load induced by non-conjugate meshing of the gear teeth can be reduced.

Fig. 6 provides some indication of the accuracy required as a function of pitchline speed.

The measurement precision needs to be better than that of the component tolerance by a factor of approximately 10. Gear tolerances vary from 50 to 100 μm right down to only several microns. Therefore, gear measuring equipment needs to have an accuracy of just 1 to 2 microns when measuring a component having a tolerance of 0.01 mm.

High accuracy, particularly in gears, can incur considerable manufacturing cost. The desired gear accuracy should therefore by chosen carefully so as to obtain maximum benefit without excessive cost.

Load Conditions

The load distribution factor K_M accounts for less than uniform load across the width of the teeth and from one tooth to the next. Ideally, the load should be uniform over the full width of the teeth and the full working depth.

Many factors can affect this condition, which in turn greatly affects the reliability and performance of the gears. Poor load distribution can occur due to misaligned shafts, excessive clearance in bearings, deflections in teeth, shafts and gear structures. Obviously, misalignment of shafts must be minimized or avoided. The deflections, on the other hand, are somewhat unavoidable since they invariably result from the applied loads.

Traditionally, the supporting structures of gears (webs, stiffeners, diaphragms and tubes) have relied almost totally on empirical or intuitive techniques. As a consequence of the ability of modern gearing to transmit high loads from the use of very hard tooth surfaces, higher achieved accuracy and higher pressure angles, the imposed loads increase the stress and strain on the supporting structures.

Empirical data becomes somewhat scarce for such situations, and so to improve the potential reliability of these high specific load (load per mm of face width) gear wheels failing to perform satisfactorily from lack of strength or rigidity, the method of finite element analysis (FEA) for stress distribution (and strain) provides a valuable tool for the evaluation of the loads in the structure of gearwheels.

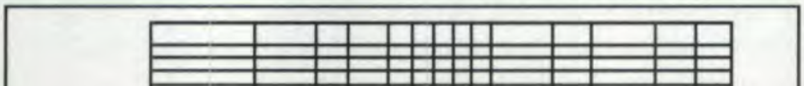

Fig. 6 — Gear quality as function of pitchline speed.

Accurate stress analysis also permits the elimination of redundant material, which contributes to reductions in unnecessary cost, weight and inertia effects. However, it has been found from these studies that it is in fact the need for stiffness rather than the level of stress that most often dictates the selection of structural dimensions. The reason is that uniform load distribution and load sharing between adjacent teeth is significantly influenced by the deflections of teeth, rim and supporting structure.

The dynamic gear alignment can be very much determined by the accuracy of manufacture. During gear cutting, significant movement can occur in the blank due to the release of residual stress. The circularity of a gear can be affected by as much as 10 to 12 mm, which would be considerably outside the permissible runout tolerance.

Residual stress in the form of tensile hoop stresses induced by improper welding practices or simply by the excessive interference with shrink fitted gear rims onto a hub can create a parasitic stress condition that combines with the normal (tensile) tooth bending fatigue stresses, which in turn often leads to broken teeth.

The dynamic load distribution can also be affected by elastic tooth deflections. Since the deflected tooth is slightly behind where it should be, the approaching tooth engages with an impact. Tip relief and sometimes root relief is applied to account for this interference. It is imperative that the tip relief, especially with spur gears, does not reduce the contact ratio below 1.0, though. A minimum contact ratio of 1.2 ensures conjugate action (uniform motion) is maintained.

The correct amount of tip relief is a function of the tooth stiffness and applied load. Tip relief can be imparted to the teeth by modifications incorporated into the gear cutting tool. The actual tip relief produced in the teeth is a function of the diameter and addendum modification.

For standardized tooling, gearwheels tend to receive a generous amount of tip relief, but pinions tend to acquire little or none. Fortunately, in speed reducing drives, the tips of the gearwheel teeth engage first. The interference of the deflected tooth (Ref. 3) can be seen in Fig. 7.

Uniform load distribution can likewise be affected by the elastic deflections of long and slender pinions. Such pinions can suffer from excessive bending and torsional deflections. It is especially important with case hardened gearing

to apply a small amount of end relief to avoid loading the ends of the teeth.

Rim Proportions

The rim factor K_B considers the reduction in ratings when the rim thickness is less than 1.2 times the tooth depth. Excessive deflections of the rim can seriously impair the reliability of the gear to perform properly (Ref. 4).

While AGMA suggests a factor of 1.2 times tooth depth as a safe minimum, the gears with higher specific load and those gearwheels with very wide face widths require proportionally greater rim thickness or other means of support to counter these deflections.

Tooth Geometry

The tooth shape or geometry also plays a major role in the overall performance of gearing. The constant angle between the line of action of involute gear teeth and the common tangent to the kinematic pitch circles at the pitch point is called the pressure angle.

Standard pressure angles evolved to rationalize the tooling required to produce gears. The Australian standard AS 2938-1995 for gearing promotes the use of ISO 53 tools with 20° pressure angle. Various other standard pressure angles are used, though.

The operating pitch line speed of the gears tends to govern the choice of pressure angle for a given application. At one extreme, pressure angles of 14.5°, 15°, 16° or 18.5° are employed to minimize noise and vibration excitation at high pitch line speed by virtue of a greater number of teeth in mesh (higher contact ratio) at any particular instant with such pressure angles.

With higher loads and slower speeds, it becomes more important to maximize tooth bending strength and pitting resistance. Pressure angles of 20°, 22.5°, 23° or 25° can be used on gears with high specific tooth loads and low pitch line speeds.

The disadvantage of high pressure angles is the reduction in transverse contact ratio (number of teeth in mesh) and a narrower top land thickness. Despite this, the use of a 25° pressure angle for low speed, high torque drives has achieved wide acceptance in many industries following its success in the gearing of draglines used in open-cut coal mines.

The geometry factors I for pitting and J for tooth bending strength both benefit from pressure angles larger than the traditional 20°. This is because for a given diameter, as the pressure angle increases, a lower radius of curvature on the teeth can be obtained, and so in turn, the Hertzian stress is reduced.

Similarly, a gain in tooth bending strength results from the increased tooth thickness at the base of the tooth. The 25° tooth form will carry about 20% more torque than the 20° nominal tooth form, all other factors being equal.

Additionally, the tooth geometry and effective pressure angle can also be influenced by addendum modification. Positive addendum modification is considered crucial for low numbers of teeth (typically < 17) where the base circle, from which the involute originates, intrudes on the active portion of the tooth profile.

Positive addendum modification alleviates the undercut in the root fillet region that ensues from this intrusion. Undercut can seriously affect the tooth bending strength by reason of the narrower (undercut) tooth thickness at the base. The contact ratio can also be adversely affected from the reduced length of active tooth profile.

The gearset likewise benefits from an increase in the relative radius of curvature on the pinion tooth, thus improving the pitting resistance. The maximum sliding velocities occurring at the extreme points of engagement can also be optimized with judicial selection of addendum modification.

The gears will not perform very well or at all without proper lubrication. Besides reducing friction and preventing wear, the lubricant is also relied upon to remove heat from the tooth surfaces.

In the case of the total loss spray systems used on the open gears of metalliferous grinding mills, cement mills, kilns and sugar mills, the air blast used to purge the lubrication system should not be so great that it displaces the grease from where it has just been deposited on the pitchline of the teeth. This can occur particularly with the low base oil viscosity grades or the latest nonchlorinated solvent group of lubricants.

A reasonable surface finish is necessary to ensure the contact between meshing teeth. While a smooth surface finish is desirable, it is sometimes the bane of manufacturers to achieve a satisfactory surface finish with the materials employed. As the purity of gear steel increases, the machinability of such materials tends to decrease. The cleaner steels improve fatigue resis-

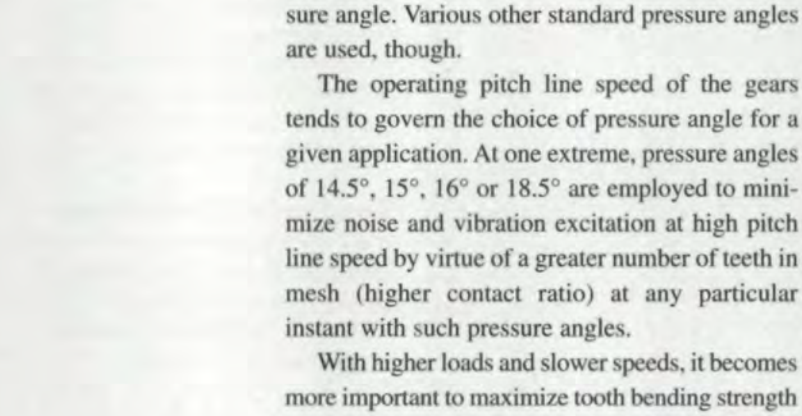

Fig. 7 — Tooth deflection and the need for tip relief.

tance but some trade-off sometimes occurs with surface finish. To date, none of the international gearing standards specify limits on surface finish other than the benefit of smooth pinion surfaces to encourage work hardening.

In addition to maintaining an acceptable surface finish on the teeth, the root fillet region deserves special attention. In this region, removal of lineated tool marks aligned with the axis of the tooth by finishing or shot peening can eliminate undesirable stress raisers.

Installation and Commissioning

Upon completion of the individual gears, the two components can be meshed together in the workshop to show how well the gears might ultimately perform. A simple mesh test with the gear and pinion set at the correct center distance and planar relationship can verify satisfactory compliance of runouts, profile, backlash and load distribution rather than taking for granted that these will be correct upon installation.

Dynamic testing methods that simulate the operating conditions can be employed to provide further information. A no-load test can be used to verify satisfactory gear design, manufacture and assembly in terms of potentially acceptable noise and vibration levels, satisfactory gear tooth alignment and bearings and lubrication operation.

A load test using either a brake or back-to-back test can simulate the actual service conditions for a much better validation. The back-to-back test can only be done with a pair of mirror-image gearboxes and not without incurring some additional cost.

The assembly of gearboxes has been found to directly affect their reliability when put into service. Since the gear case will have been machined to suit the bearing race size, final assembly must maintain the proper fit. The overzealous use of sealing compound on the gear case halves or simply a gap between top and bottom will allow the outer bearing races to spin, causing premature failure of the bearings and damage to the gear case.

In the case of rotating shafts, the fit of bearings to the shaft is also critical. The current design of some roller bearings uses reduced race thickness. The consequence can be increased hoop stress culminating in breakage of the race, particularly if the shaft size is at the upper tolerance or greater.

The installation can obviously have a direct bearing on the reliability of the gearset. The gear alignment and uniformity of load distribution across the width of large gears can be determined from non-contact temperature measurement using an infrared pyrometer (thermometer). The foundations and gear support structure must also be adequate and stable to maintain these alignments.

In bolting down the gearbox, an uneven foundation will distort the gear case. Even just a small amount of twist in the gearcase can significantly affect the gear alignments. This aspect can sometimes be used to advantage to achieve proper gear alignment when the gearbox bores are not machined just right.

Couplings play a vital function in the success of a gear drive. A case in point is herringbone or double helical gearing, which has the special need where the couplings must not restrict the axial movement of the floating member. The choice of couplings may also need to consider the necessity to isolate the gear elements from sources of resonant vibrations (or the coupled components from the gear mesh excitation).

Proper care and maintenance practices form an essential ingredient in order that the potential or intended reliability will be realized in service. The benefit of a regular inspection program should not be underestimated. Condition monitoring methods such as vibration levels, temperature measurements, oil analysis, visual inspections and nondestructive testing provide valuable information to assure the long-term performance and reliability of any critical or important gear elements.

In conclusion, the reliability of any gear set is influenced by many factors. The production of gears involves some very sophisticated machine tools and specialized processes and procedures, so gears tend to be rather expensive items. The designer plays a vital role in the success of these gearsets, but so too do those involved with the manufacture, installation and maintenance.

The whole of life cost should be considered since mediocre gear quality gearing rarely proves worthwhile, since the downtime can also be very costly. A considerable effort is expended in producing both good and not-so-good gearing, but hopefully the preceding information provides some insight into the many aspects that contribute to achieving the best and most reliable gearing. ○

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