FINE PITCH, PLASTIC FACE GEARS: Design

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Since receiving his degree in mechanical engineering in 1985 from the University of Waterloo in Ontario, Canada, Reiter worked in the field of plastics part production for the automotive industry. His various responsibilities include developing tooling and directing the manufacture of molded plastic gears. As part of his engineering duties, he has acquired advanced skills in computer graphics and its application to gear geometry.

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Figure 1—Power tool application: 28 diametral pitch, 44:15 ratio face gear, 90° shaft angle.

Figure 2—A 90° shaft angle wheelchair access door opener application: 20 diametral pitch, 56:16 ratio.

Face gear technology is not widely recognized. If mentioned at all in gear-related editorial, it is described as merely one of a number of unusual gear geometries. Its beneficial applications are largely overlooked not only for coarse-pitch, high-power applications where gears are made from hardened steel, but also in fine-pitch, limited-power applications where gears are made from materials such as molded plastic. The information presented in this paper counters such oversight, particularly in the molded plastic applications where modeling of such gears is critical. An example of a fine-pitch plastic face gear that is used in a power tool application is shown in Figure 1.

Figure 2 shows a face gear and pinion which is used as a main drive gear in a commercial power swing door application. In this case, a cut steel pinion was used, although it would not be uncommon to use powdered metal or plastic pinions as well, as shown in Figure 3.

Introduction

To increase familiarity with this face gear technology, it is necessary to consider a number of subjects. These will start with a description of typical face gears and their combinations with mating gears. Since these combinations are always non-parallel shaft drives, comparisons to other such drives will

and Manufacture

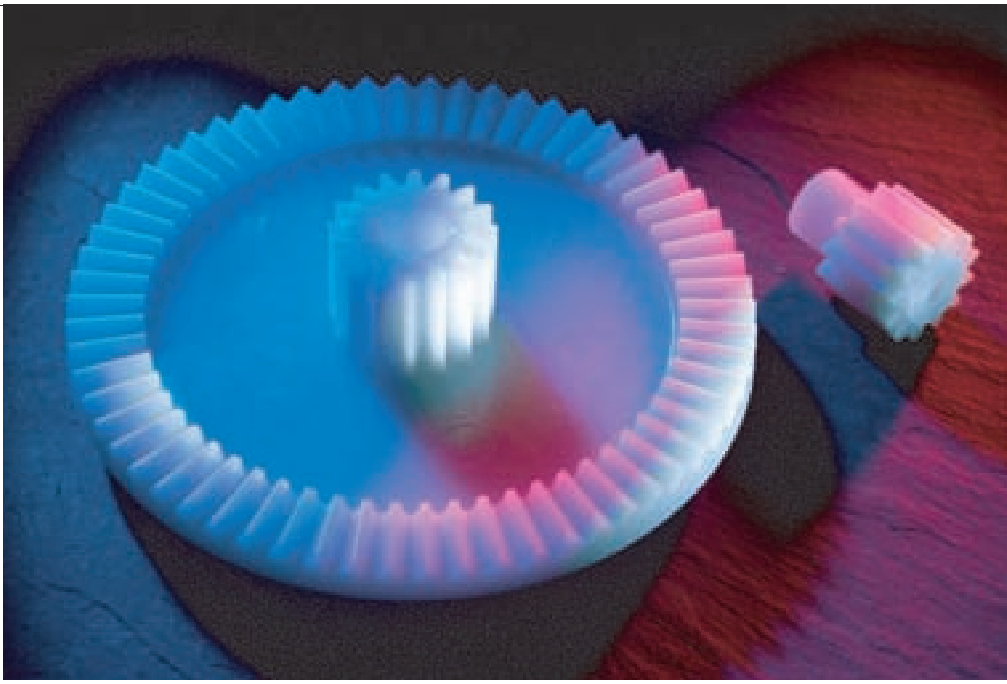

Figure 3—Example of a 0.8 module, 62:12 ratio face gear set, 90° shaft angle.

follow. Any discussion of gears would be incomplete without some consideration of their manufacture, which will be the next subject. This will lead into a discussion of design issues. Present day work in gear technology, both in manufacturing and design, has moved into graphic modeling. The treatment of the topic of graphic modeling for face gears may represent the first such treatment in published gear literature. The final subject, to begin to convey the flexibility of face gears, consists of a brief description of face gear configurations that go beyond the typical, and simplest versions described earlier.

Many of these descriptions will apply equally well to face gears made from materials other than molded plastic. This is especially true for general face gear geometry and many operating conditions. The discussion naturally becomes specific to plastic gears when manufacture is covered. There is also some consideration of certain operating conditions typically encountered in products containing plastic gears of any type.

Description of Face Gears and Meshing Action

In conventional gears, the gear teeth project radially from the outside rim of the gear blank. In common face gears, the teeth project axially from one of the faces of the gear blank, as can be seen in the CAD model in Figure 4.

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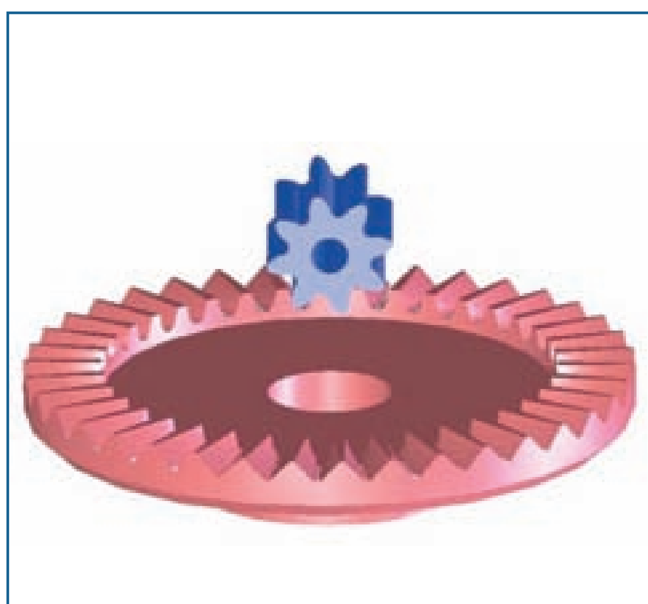

Figure 4—Mating face gear and pinion.

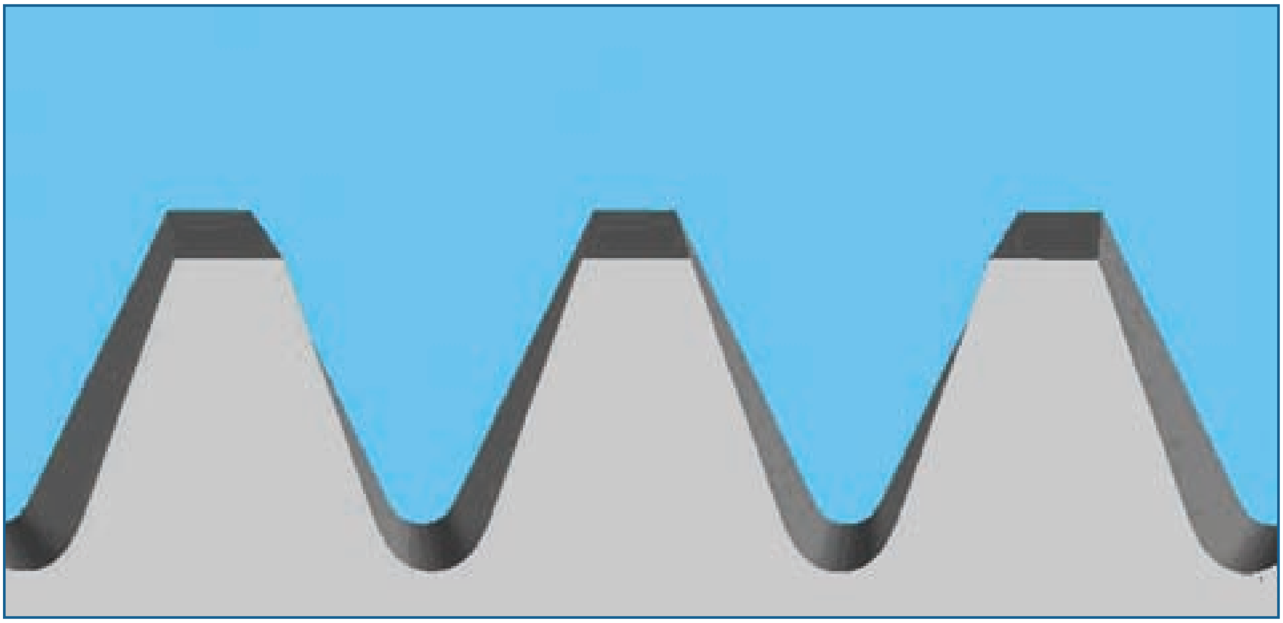

Figure 5—Changes in tooth cross section with radial positioning.

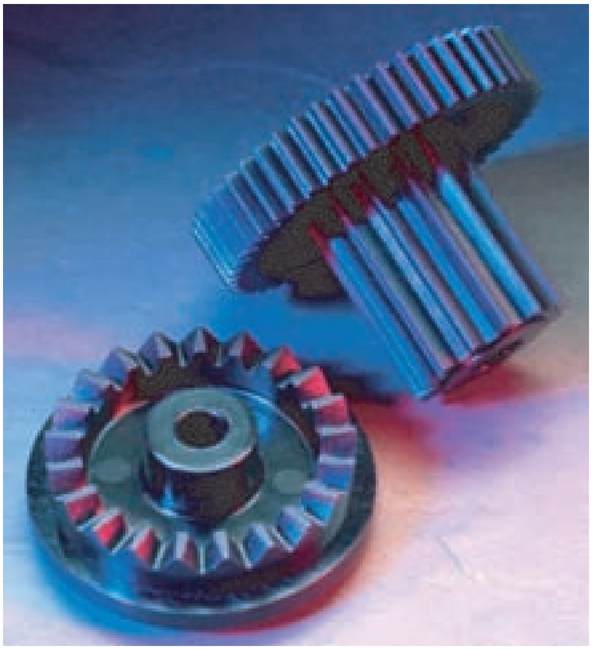

Figure 6—90° shaft angle automotive transfer case actuator application with 32 diametral pitch, 21.14 ratio face gear set.

The radial limits on these teeth are defined by inside and outside circles. Conditions that impose restrictions on the diameters of these circles then determine the net face width, or radial length, of each tooth. The tooth tips, or top lands, lie in a plane perpendicular to the gear axis. The tooth cross-section changes with its radial location, as can be better seen in Figure 5. Part of this change is reflected in the changing top land, which tapers to a reduced width with increasing radius.

A real example of this changing cross-section can be clearly seen in Figure 6. Note the change in the top land and the tooth form between the inside and outside surface.

The face gear is mated with a pinion, as shown in the cross-sectional view of Figure 7. In the most common type of face gear drive, the pinion is a spur pinion. (Less common versions of face gears and mating pinions are described later in this article.) The pinion may be of conventional design, or its tooth proportions may be modified to optimize the performance of the face gear drive. To take full advantage of what may be the limited face width of the face gear teeth, the face width of the pinion is made large enough and proportioned as to straddle the face gear teeth.

Contact between pinion and face gear tooth surfaces is ideally along a line extending the full width of the face gear tooth. These ideal lines are not exactly straight or parallel to the pinion axis, as seen in Figure 8. This line contact depends not only on the gear and shaft angle accuracy, but also on the axial position of the face gear. Deviation from the ideal axis position changes contact from a line to a nominal point. This contact will be either at the inner or outer end of the face gear tooth, depending on whether the axial position is too much either in the tight or loose direction. Despite such a shift in contact location, in a properly designed gear set, conjugate or smooth meshing action is maintained. Contact ratio is generally similar to that of spur gears of similar proportions, even when contact is localized at one end of the tooth.

Mating tooth action is essentially a combination of rolling and sliding as in spur gears. Due to the rotation of the face gear, there is some axial sliding on the pinion tooth with corresponding radial sliding on the face gear tooth. This sliding will be greater for gear sets of lower gear ratio, but for all gear ratios, sliding adds very little to the overall friction losses. As a result, face gear drives will have similar efficiency to spur gear drives, excluding possible differences in bearing losses.

The most common mounting has the axes of the two gears intersecting at right angles. In principle, this arrangement

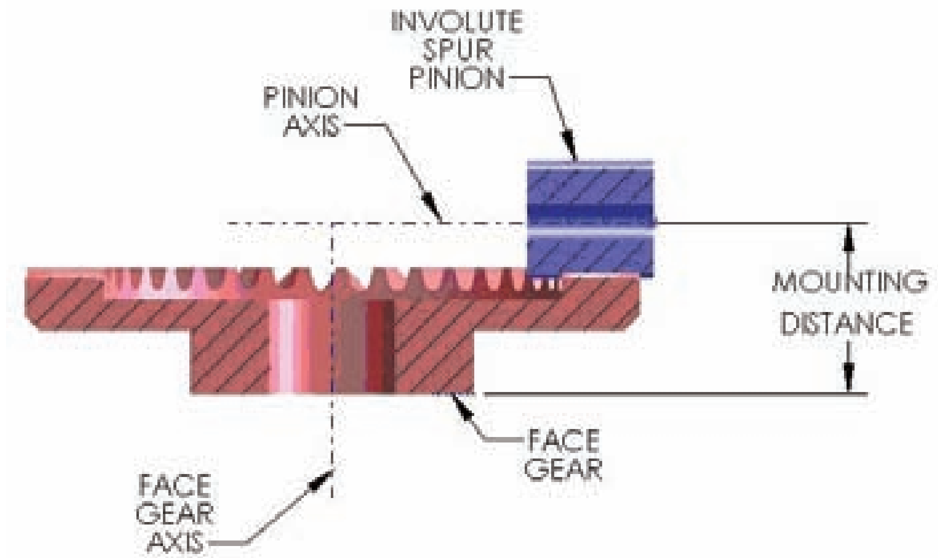

Figure 7—Cross-sectional view of mounted face gear and pinion.

should permit designs with the supporting bearings for each gear straddling the gear. However, practical design considerations lead to one or both gears being supported in an overhung arrangement.

Comparisons with Other Types of Non-Parallel Shaft Gearing

Such comparisons are best made with some description of typical operating conditions for molded plastic gears. They are rarely made to ideal levels of accuracy, rigidly supported in low clearance ball bearings and precisely positioned in ideally accurate housings. Instead, some eccentricity, out-of-round conditions and out-of-flat conditions are to be expected in the molded face gear. Supporting shafts may lack full rigidity and may be guided in journal bearings with generous clearances. The housings are most often of molded plastic, with distortions and other bearing location issues. Taken together, the gears of whatever type are expected to perform under less than ideal conditions. All this should enter into evaluating the comparisons.

The first comparisons are best made with some description of typical operating conditions for molded plastic gears. Face gears are commonly considered as substitutes for such gears. Bevel gears require careful, almost precise, positioning to avoid a rough mesh similar to mating gears of slightly mismatched module. With face gears, axial positioning of the pinion is not a factor in gear meshing. Axial positioning of the face gear remains a factor, but not in a way that is much more demanding than for a set of spur gears. Backlash requirements must be met with some extra care to achieve some degree of control over tooth length contact. As with bevel gears, this can be helped somewhat by the introduction of crowning, or adding of slight amounts of material on the tooth flanks at the preferred contact locations.

Bevel gears of very high gear ratios are often restricted by

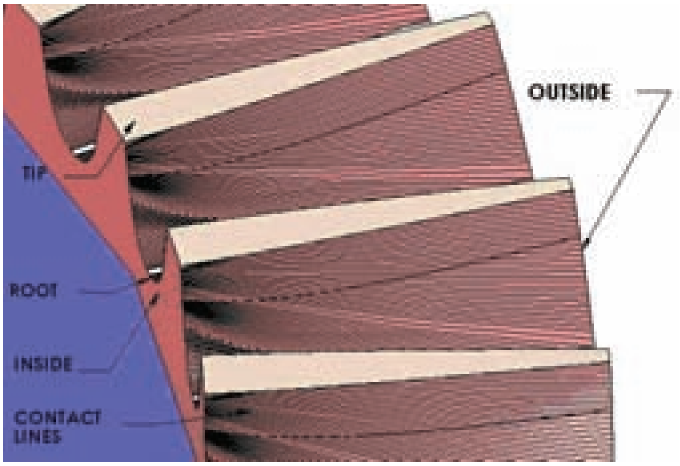

Figure 8—Lines of contact.

mechanical angle limitations on the gear cutting machines to be used in cutting the gears or the electrodes that will be needed to make the gear molds. Face gears do not have such a restriction. However, face gears have a gear ratio restriction on the other end. It is very difficult to design a face gear for a gear ratio under 2.0, or 1.5 at best (as was the case for the part in Fig. 6), thereby disqualifying face gears as miter gears with ratios of 1.0.

The major possible limitation of face gears in comparisons to bevel gears will be in load capacity. This results not so much in the selection of module, for which gears of similar size and numbers of teeth may have similar module values. Instead, it is the face width of the load carrying teeth. In face gears, this width may be only 15% or 20% of the face gear outside radius. In bevel gears, it may be 25% to 30% or even greater for low ratio gear sets. Of course, this advantage in bevel gears is lost if the gears cannot be positioned to ensure contact along the entire tooth width.

Face gears may also be compared to cross-axes helical gears,

Figure 9—Tooth shaper cutter for cutting face gears.

especially the common version in which the driving member is recognized as a worm and the driven member as the helical gear. Such gears are best able to adjust to variations in mounted position. Axial positioning is not a restriction as long as each gear has adequate length. Variations in shaft angle often have little effect on the gear mesh. Control over center distance is no more demanding than for parallel-shaft gearing.

These gears can accommodate the biggest range of gear ratios well beyond face gears for the very high ratios. This flexibility extends to low gear ratios, in which the driving member no longer resembles a worm, although this is rarely exploited.

In other respects, these cross-axes helical gear sets are clearly limited in comparison to face gears. Contact between the flanks of the two gears is nominally a point, which leads to local contact pressures. The result is excessive wear unless the loads are severely restricted. Furthermore, the gear meshing action introduces considerable axial sliding. The friction associated with this sliding materially reduces the gear set efficiency, placing it well below face gear efficiency.

Manufacture of Face Gears

Manufacture of the mating pinion does not require special attention. The pinion may be made of a different material than the face gear. It is not unusual for a sintered powder metal pinion or machined metal pinion to run against a plastic face gear. If machined, a pinion of optimized design may require special cutting tools.

The plastic face gear generally has machining in its history. Often, prototype parts are machined from plastic material for design evaluation before molds are built for production parts. Machining is generally required for the preparation of face-gear-shaped electrodes to be used for electrical discharge machining (EDM) of mold cavities.

Except as described later, this machining is performed on special gear shaping machines. In a gear shaper, a cutter in the shape of a gear with cutting edges on one face (Figure 9) is reciprocating along the width of the machined gear tooth, as can be seen in Figure 10 for shaping a face gear. The cutter and gear are rotated between or during strokes with the gear ratio needed to give the required number of machined teeth. In conventional machines, say for spur gears, the cutting stroke is parallel to the spur gear axis. For face gears, the cutting stroke is radial to the spur gear axis, from the outside diameter to the inside, in a direction determined by the orientation of the pinion axis to the face gear axis in the mounted assembly.

In conventional machining of spur gears, the number of teeth in the cutter is selected for convenience and is not tied to the number of teeth in the gear to mate with the machined spur gear. In face gear machining, however, there is a design connection to the pinion mate and the face gear. The cutter design is commonly of special proportions to make this design connection, although the pinion is sometimes designed to match an available cutter with an acceptable number of teeth.

The cutter must be of hardened and ground steel when many face gears of tough materials are to be machined. For a limited number of machined prototype plastic face gears or the few machined from electrode materials, the cutter is often made by machining from hard steel.

The mold cavities, as noted above, are commonly made from electrodes machined on gear shapers. A number of electrodes are needed for each cavity, some designed for more rapid roughing burns and others for finishing. A more recent method uses CNC machining from precise graphical models, in which the cutter is a tiny ball-shaped end mill. In another metal removing process, the end mill machining is replaced by the application of laser technology.

The mold cavity design must include allowances for shrinkage of the plastic material. These allowances may need to be adjusted to any non-uniform size change in the molding process. The electrode requires a further allowance for the "overburn," the small gap between the electrode and mold cavity surface.

Design Issues

The general objectives in design start with the specified gear ratio. The design must also conform to the specified size and space limitations. It must be compatible with anticipated manufacturing variations in gear dimensions and mounting locations. Tooth proportions and material selection must provide load and life capacity. There are likely to be further requirements relating to noise and vibration at specified speeds. In addition, the design must be compatible with the planned manufacturing processes.

The specified objectives apply not only to the face gear itself but to the mating pinion and, in some features, also to the shaper tool when that has a role in the face gear manufacturing. The tooth thickness of the face gear and pinion together must provide adequate backlash, even when the axial position of the face gear is subjected to manufacturing variations. The root diameter of the pinion must be adequate to ensure sufficient wall thicknesses over its bore. Even if there is no hole, the material inside the root diameter must be able to transmit the applied torque.

The whole depth of the pinion teeth and face gear teeth must each contribute to a suitable level of contact ratio. For pinions of low numbers of teeth, attention must be paid to the top lands. The location of the face gear tooth tips relative to the pinion center, together with the pinion root diameter, will determine if the root clearance in the pinion is adequate. The same face gear dimension and its whole depth together with the pinion outside diameter will define the face gear root clearance. To meet all these interlocking dimensional requirements, and those yet to be mentioned, may constitute a design challenge. This is most likely to be the case if the pinion, as is often the case, has a small number of teeth.

The proportions of the cutter designed to properly match the face gear features must be evaluated. Its machining position relative to the face gear is ideally connected to the design of the pinion. Proper position is needed to preserve ideal conjugate action between the pinion and face gear. This position, together with the selected face gear whole depth, determines the outside diameter of the cutter. With this diameter, the cutter must have adequate top land. An additional cutter design requirement is related to the location on the cutter flank, which will machine the face gear tooth near its tip. This location must be on the involute curve, preferably some distance outside the base circle, or any undercut, on the cutter.

The inside and outside diameters must be properly selected. Too small an inside diameter will introduce undercutting in the nearby tooth flanks. Too large an outside diameter will result in inadequate nearby top land. Adjustments in these diameters may in turn leave a tooth face width too small for load capacity. The design procedure may require continuous examination of these conditions.

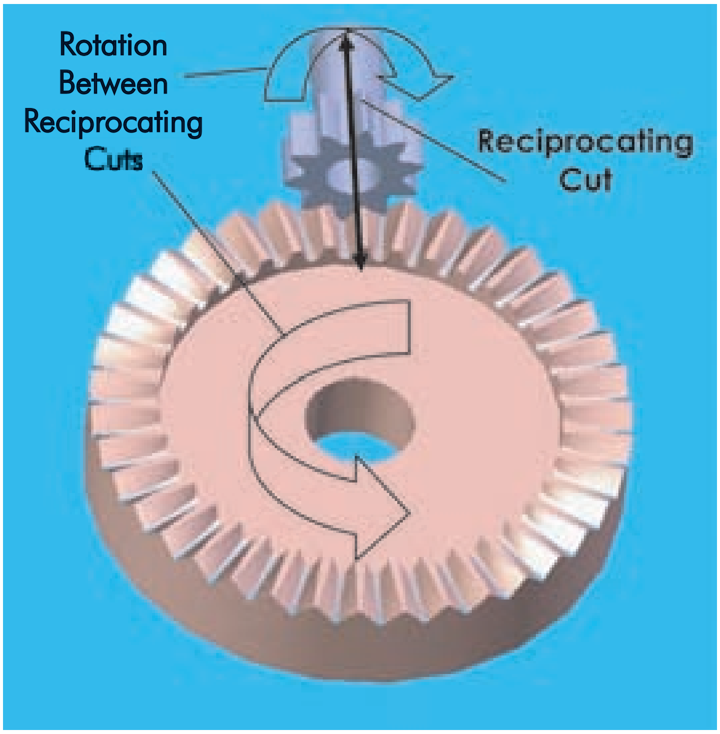

Figure 10—Machining of face gears.

Some documents and design software may offer solutions that meet all these requirements. However, added consideration of the role of manufacturing variations may be needed in evaluating these designs.

Earlier mention was made of the introduction of crowning to cope with various forms of less than ideal positioning of the assembled gears. This crowning may be introduced in one of three ways. The pinion may be crowned as is commonly available in machined pinions and, more recently, in molded plastic pinions. The face gear teeth may be crowned by introducing a cam-like cutter stroke into the shaper machining of the face gear electrodes. Alternatively, the crowning may also be introduced into the face gear teeth in an indirect fashion.

If the cutter has more teeth than the pinion, the resulting tooth surface as seen by the pinion will have a combination of lengthwise crowning and taper. A slight adjustment in the cutter stroke direction changing the shaft angle will correct for the taper and leave the crown. This adjustment is best accomplished with the aid of contact pattern inspection.

Graphic Modeling

It has already been suggested that graphic modeling may have a role in mold cavity manufacture. It may also serve as a test of design specifications by revealing conditions such as undercut

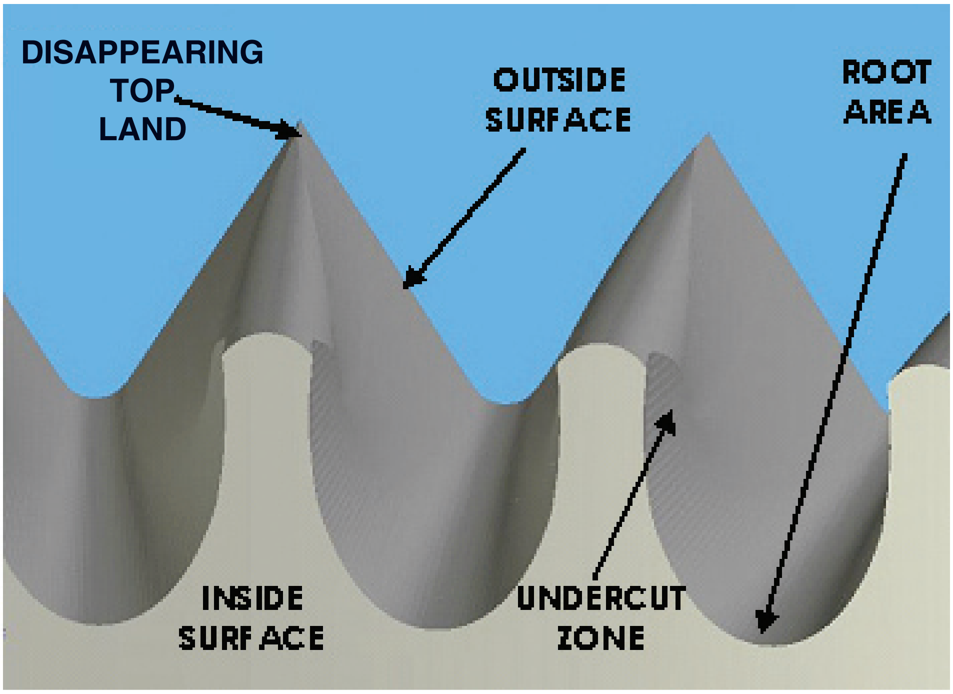

Figure 11—CAD-generated face gear model with undersized inside diameter and oversized outside diameter.

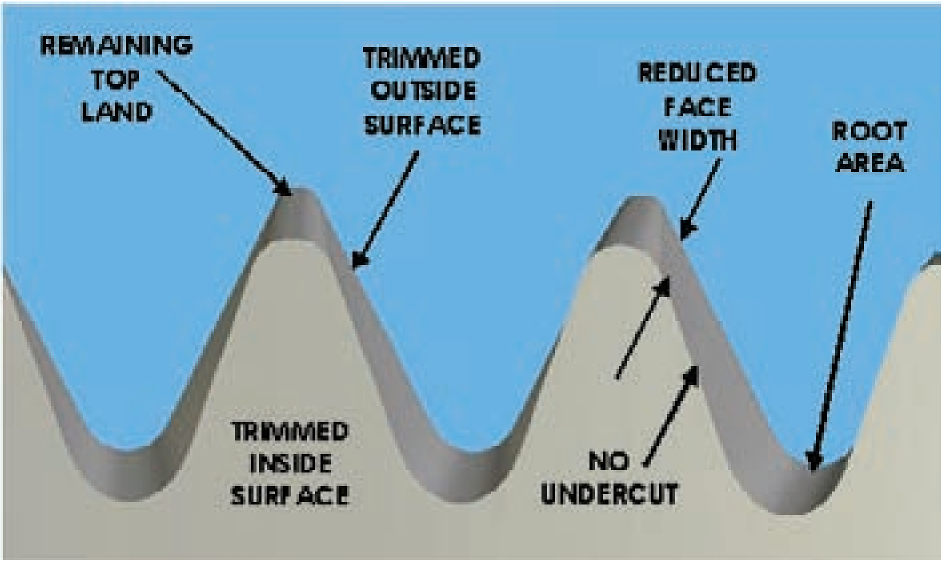

Figure 12—CAD-generated face gear model with trimmed inside and outside diameters.

and inadequate top land. It would also serve for the rapid prototyping of demonstration components or simply for creating a realistic representation for engineering drawings.

Two general methods have been used to create such models. One makes use of analytical methods for calculating points on the face gear tooth surface. These would be in sections located at multiple face gear diameters or in multiple sections parallel to the face gear tooth tips, or in both sets of sections. The computer graphics system would then join these points, first in lines and then as a surface.

The second method takes advantage of a more advanced computer graphics technique. This permits the description

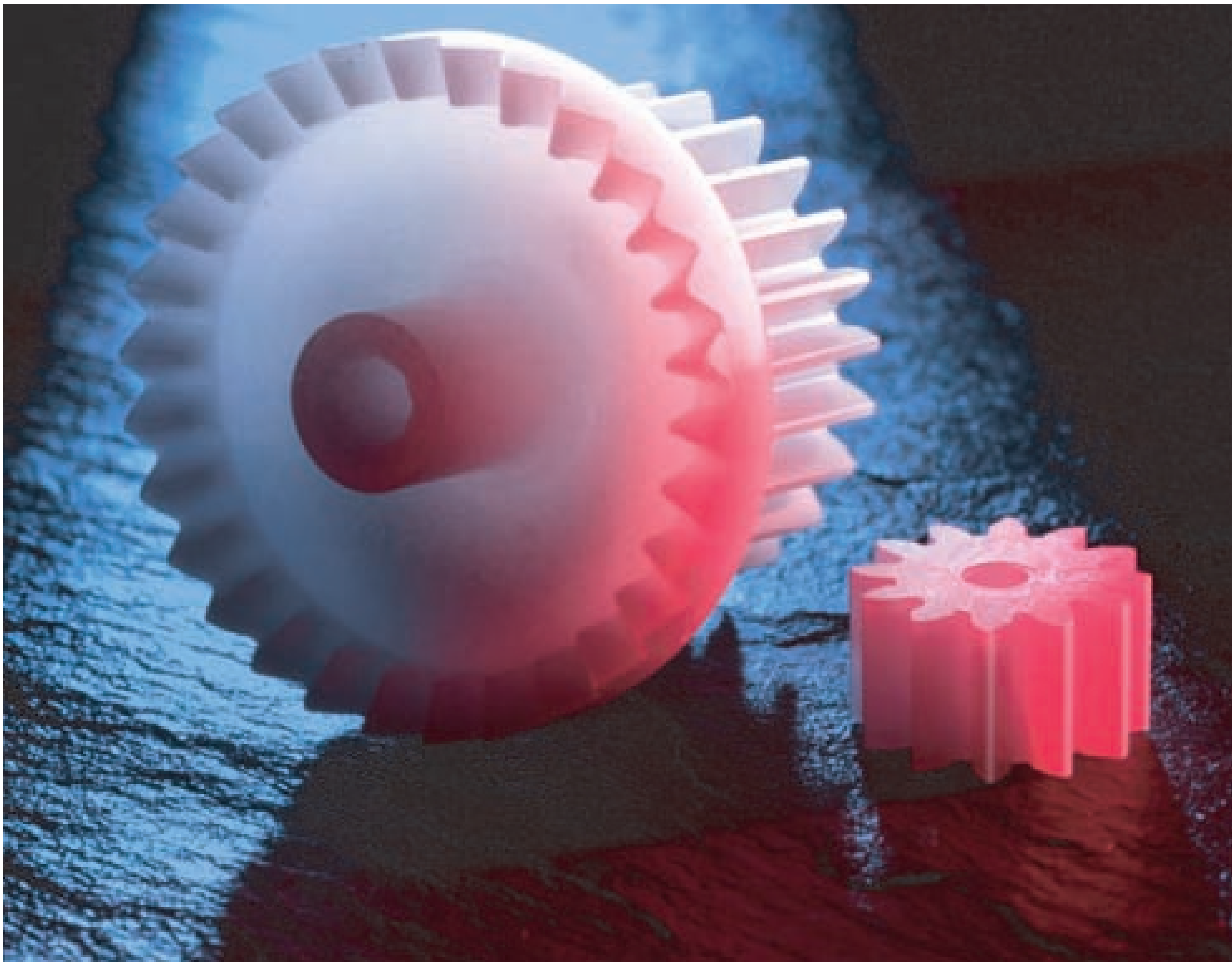

Figure 13—A 65° shaft angle automotive door lock actuator application: 40 diametral pitch, 30:12 ratio face gear set.

and placement of two intersecting, three-dimensional bodies. The volume where the two bodies interfere may then be removed from one of the bodies, giving it a new shape and size. This may be recognized as the same as a machining process. One body may be made to represent the tool, stretched by its cutting stroke. The other body represents the workpiece. The discarded intersecting volume will then represent the discarded machining chips.

This technique has been applied for face gear geometry. The imaginary shaper cutter is made to resemble a stretched spur gear. The pre-machined face gear blank is the second body. The first interfering volume is then removed. With the tool being rotated through some small angle and the face gear blank being rotated by the gear ratio-defined angle, a new chip can be removed. Repetition of this process will form the entire tooth, which may then be replicated to create the complete model of the face gear as seen in Fig. 11.

Undersized Inside Diameter and Oversized Outside Diameter

This figure shows the face gear with an undersized inside diameter. It therefore reveals an undercut condition at the inside and a disappearing top land at the outside. A "trimmed" face gear is shown in Figure 12.

Other Face Gear Configurations

The face gear configuration described above was the most common version based on a spur pinion whose axis was intersecting and perpendicular to the face gear axis. Other configurations have been implemented to meet special requirements.

When shaft orientations dictate an angle other than a right angle, the mating face gear can be made with teeth arranged along a conical surface instead of a plane surface. The automotive door lock actuator face gear seen in Figure 13 is mounted

Figure 14—A 30.5° helical angle on pinion printer application: 0.5 module, 69:11 ratio helical face gear with a 90° shaft angle.

on a 65° shaft angle. (Note the angled teeth in the photograph.) This arrangement can be concave or convex depending on the direction of the deviation from perpendicular.

Keeping with the spur pinion, its axis can be offset to the face gear axis. This may be indicated if shaft orientations must be skewed rather than intersecting. Performance is not marginally affected as long as the drive is in only the preferred direction and if the offset is limited. In other than the preferred direction, there will be an increase in lengthwise sliding and frictional power loss.

Another version replaces the spur pinion with a helical pinion, as shown in Figure 14. The objective is to achieve the anticipated benefits of a helical drive over a spur drive. Adding the helical overlap in the meshing action can increase the overall contact ratio with the potential of noise and vibration reduction. As with parallel-axes helical gear sets, such benefits may be limited by poor shaft orientation. However, if other features of gear accuracy are far less than ideal, there may be an overall benefit.

Conclusion

Face gears have a role to play in molded plastic gearing. By increasing general familiarity with face gears, this role is likely to expand. Graphic modeling of face gears will assist in this expansion of face gear application. ■