Optimizing Plastic Gear Geometry: An Introduction to Gear Optimization

Richard R. Kuhr

There are numerous engineering evaluations required to design gear sets for optimum performance with regard to torque capacity, noise, size and cost. How much cost savings and added gear performance is available through optimization? Cost savings of 10% to 30% and 100% added capacity are not unusual. The contrast is more pronounced if the original design was prone to failure and not fit for function.

Development of the gear geometry is a critical part of the total design process. This article will summarize the design issues relating to optimizing gear geometry. All interrelated parameters that comprise the gear description are candidates for optimization. Those parameters include pitch, pressure angle, helix angle, addendum modification, root clearance, face width, root and tip radii, tooth thickness, center distance and profile modifications. One design parameter, total working depth, will be evaluated as an example of the potential benefit of gear optimization.

Gear Optimization Definition

An optimized gear design is the best possible gear arrangement, gear design and material selection that facilitates the lowest total cost for the performance and reliability required.

Optimization Cost Effects

The design benefits and cost savings of optimizing with practical solutions are substantial. Conversely, the costs associated with a non-optimized or poorly designed gear set are a real liability. The incremental cost of optimizing is minimal compared to the ongoing costs associated with an overdesigned gear arrangement using more costly material or larger gears and housing than required. The warranty costs soar as do the costs of lost user confidence if the gears are poorly designed or underdesigned.

The cost of optimization is only a small fraction of product introduction cost. Optimization does, however, require attention to the details of the gear geometry, gear accuracy, material selection, duty cycle and a detailed analysis of the mounting conditions of the application. Added cost savings are derived from using the most cost-effective material from a physical property perspective.

Then that material is configured in the most economical shape and size. The result is a very robust and cost-effective gear molded to the required accuracy level. The specific goals of optimization vary from application to application. What is required in one case may not meet the need in another. While optimization is application specific, the process can be applied to all applications.

Design Goals

The transmission of uniform motion under the operating load throughout the range of the operating environment is common to most gear applications. A number of design challenges arise from this basic goal of uniform motion under

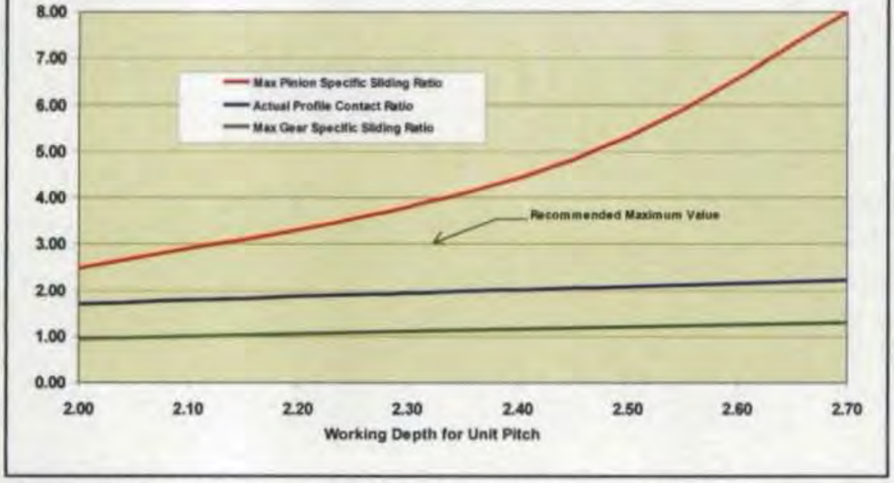

Figure 1—Tooth depth influence on sliding and contact ratios (equal addendums).

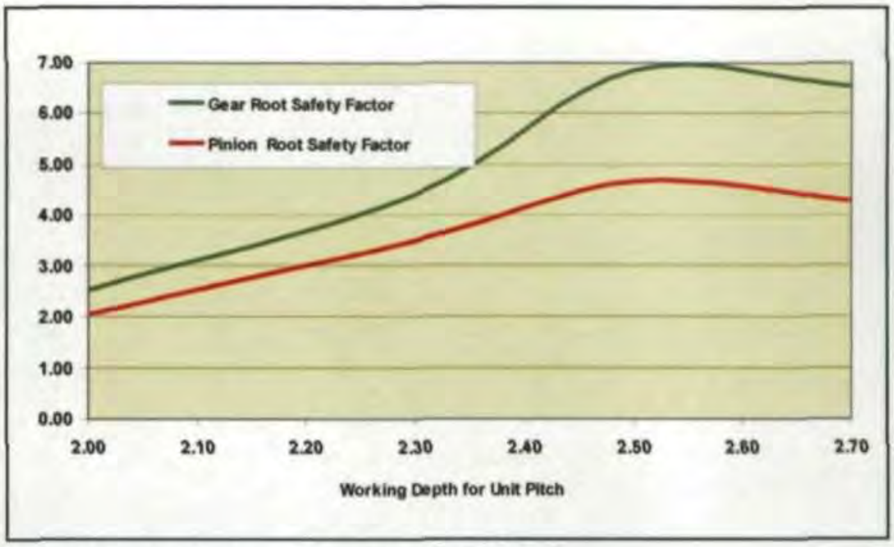

Figure 2—Root bending strength factors (equal addendums).

load. One of the challenges is the interdependence of the design parameters.

For example, a design change can have a positive influence on one set of design parameters and a negative influence on another set. It is not possible to have the optimum values for all the design parameters. It is necessary to determine which parameters are most important to the success of the gears operating in the specific application. A knowledgeable compromise is required to set the design limit of each gear geometry parameter.

One of the benefits of working with plastic is the potential of modified tooth proportions. The use of standard gear proportions will seldom yield optimization. Multiple design iterations are typically required to optimize new applications. Fully utilizing the optimization process will help in designing the best gear set for the performance and reliability the application requires.

Optimization Process Steps

To understand how this geometry optimization process fits into the total process, here is a summary of the steps in the plastic gear performance optimization process.

1. Define the specific application—including the processing accuracy—for the housing and gears.
2. Account for the extreme conditions of temperature, moisture, and tolerances.
3. Compute the loads and speeds over the entire duty cycle and the number of desired life cycles.
4. Select the appropriate combination of gear types—spur, helical, worm, bevel, face, crossed-axis helical, internal, external, planetary—their arrangement and the power path.
5. Calculate the ratios for minimum total volume if more than a single-stage drive.
6. Determine the required accuracy level and verify it matches the process capability.
7. Select the material for the application that provides the necessary strength and durability at the lowest possible cost.
8. Develop the gear geometry to meet the necessary limits of contact ratio, specific sliding ratios, root clearance, backlash, and deflection over the range of extreme conditions.
9. Design the minimum-weight structure that supports the gear teeth and provides the required stiffness, strength, and molded precision. This structure comprises the rim diameter that supports the teeth, strengthening rings, webs, ribs and hub diameter for the application.

Evaluation Conditions

Nominal Center Distance Evaluations. The gear designer needs to design the gear set to operate over the full range of extreme conditions. Note that acceptable design parameters at the nominal housing center distance will not assure acceptable parameters at the extreme conditions. The typical range of effective center distance can be three to five times that of the specified center distance tolerance of the housing. It is always desirable to minimize mounting and assembly tolerances to improve gear performance.

Minimum Effective Center Distance Check. Optimizing gear geometry requires that each design iteration be checked at two extreme conditions. The first extreme condition for external gears is the maximum material condition (MMC) at minimum effective center distance. The minimum effective center distance is the tightest mesh condition considering the location tolerances, bore-to-shaft clearances, total runout, gear and housing materials, temperature, humidity, and gear accuracy.

The primary design parameters that are evaluated at this closest mesh condition are backlash, root clearances, profile contact ratio, percent of recess action, and specific sliding ratios. Under heavy load conditions and higher temperatures, more backlash is required to avoid the coast side profiles from coming into contact as the teeth

Richard R. Kuhr

is global gear design manager for Enplas Corp., a worldwide injection molder with manufacturing facilities in seven countries for molding critical-to-function plastic components. He is responsible for customer gear design training, as well as development and optimization of Enplas commercial gear designs. He has more than 25 years of gear manufacturing and design experience. Kuhr has a bachelor's degree in mechanical engineering from the University of Illinois. He is vice chairman of the AGMA plastics gearing committee. He has provided gear design services for many industries, and he has presented numerous papers related to gear design and metrology at SME conferences.

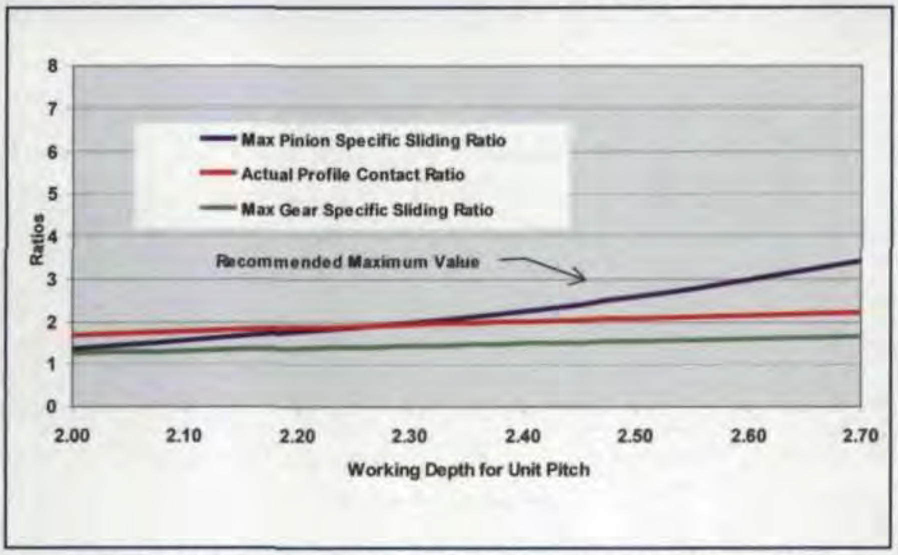

Figure 3—Tooth depth influence on sliding and contact ratios (addendum factors +/- 0.20).

are deflected. Another issue related to backlash determination is gear accuracy. More accurate gear teeth require less thinning to maintain the design backlash.

Sufficient root clearance is needed to avoid the risk of interference. Too much root clearance reduces the tooth working depth and the profile contact ratio. Higher profile-contact-ratio gear sets reduce noise levels and increase tooth strength by increasing the load sharing. It also reduces the deflection variation as the tooth loads are alternately distributed between one and two tooth pairs as they rotate.

For single-direction lubricated drives, the percentage of recess action should be greater than the percentage of approach action since the coefficient of friction in recess is less than approach. Long-addendum pinions increase the amount of recess action and also reduce the chance of undercutting the profile in the root area of the pinion.

Low specific sliding ratios promote surface durability and reduce operating temperatures. High specific sliding ratios can lead to overheating and a loss of capacity. Applying long-addendum modification and using enlarged operating center distances reduces the amount of sliding.

Maximum Effective Center Distance Check. The second extreme condition is at the least material condition (LMC) and the maximum effective center distance. The maximum effective center distance for external gears is the farthest apart the gears' centers can be considering the tolerances, bore-to-shaft clearances, component runout, temperature, humidity, and gear accuracy. The separating forces of the gear mesh influence this condition.

The profile contact ratio is the primary design parameter that is checked at this most widely spaced condition. If the effective contact ratio falls below 1.000, loss of smooth-motion transmission will result. Excessive wear and vibration will result because the radius on the driven tooth tip will be the first point of contact. This results in high load concentration and a loss of conjugate action.

Since mesh and mounting errors reduce the effective contact ratio, a design contact ratio larger than 1.0 is required. A typical minimum design profile contact ratio for spur gears is 1.200, although this is difficult to obtain with gear sets that have pinions with a low number of teeth. Designing gears for higher profile contact ratio is one of the goals of optimization.

Gear Geometry

Tooth Forms. Molded plastic gear design offers a striking potential for optimization because the mold is designed for a specific gear. The tooth form must be compatible with all mating gears but does not need to conform to any given standard. The AGMA standard *1006-A97, Tooth Proportions for Plastic Gears,* discusses four plastic tooth forms ranging in working depth from 2.000/(Diametral Pitch) to 2.700/(Diametral Pitch). The increasing tooth depth provides greater potential for higher profile contact ratios. These tooth forms have full fillet radii and represent sound engineering practice. Use of these forms alone does not produce optimum gear geometry; however, they do provide a very good start.

Tooth Depth Optimization Example

Here is an example of one of the design parameters—working tooth depth—evaluated for optimization. The specification of gear geometry includes number of teeth, pitch, pressure angle, helix angle, outside diameter, tooth depth, tooth thickness, face width, root and tip radii, along with any required tooth profile modification.

Equal Addendum Design. Equal addendum design demonstrates the positive and negative results of changing tooth depth. A common design goal is to design gears with the highest profile contact ratio and the lowest specific sliding ratio possible. The profile contact ratio increases as the working tooth depth is increased. That is a benefit for the design.

At the same time, the specific sliding ratio increases as the tooth depth increases. This is a detriment because of the increase in heat generated in the tooth mesh. The designer must balance the two opposing parameters to maximize the net benefit. For continuously running applications, a maximum specific sliding ratio of 3.0 is recommended. Exceeding 3.0 raises the probability of premature failure. Intermittent applications can tolerate higher values.

Figure 1 relates the influence of tooth depth on profile contact ratio and specific sliding ratio for pinion and gears that have equal addendum (Ref. 1). This example does not represent an optimized design and is only presented to illustrate the relative changes.

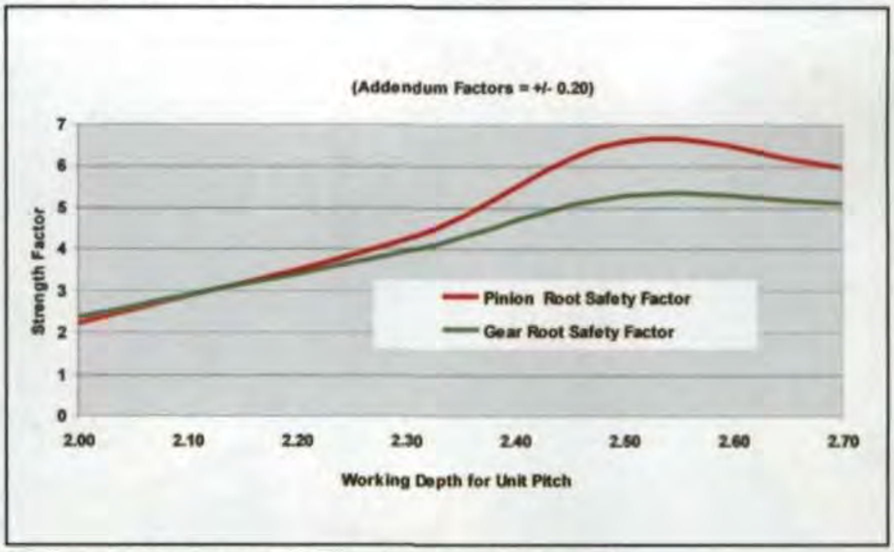

Figure 4—Root bending strength factors (addendum factors +/- 0.20).

The example is a 24-tooth pinion meshed with a 72-tooth gear having a 20° pressure angle. The tooth form is full fillet. The "Working Depth" axis is scaled to a unit measure of pitch. It would apply to either a 1.00 diametral pitch or a 1.00 module gear set.

Figure 1 indicates that the contact ratio increases as the tooth depth increases. There is also a marked increase in the specific sliding ratio. The increase in sliding raises the amount of heat generated in the mesh and will, at some point, reduce the bending strength. The effects are shown in Figure 2.

These strength factors apply to gear sets with equal addendums. The result is unbalanced root bending strengths. In this case, the gear member, with the larger number of teeth, is stronger than the pinion. Usually a stronger pinion is desired for equal life because it sees more contact cycles.

It is apparent in this example and the next that there is an increase in strength as the working tooth depth increases. This assumes that the teeth are accurate enough to share the load. This reduces the diameter of the lowest point of single tooth contact and reduces the bending stress. A peak value at the 2.55-unit working depth illustrates that the negative effects of the higher specific sliding ratio exceed the positive effects of the increased contact ratio as the tooth depth increases beyond that point. The next example shows a root strength balance in favor of the pinion.

Modified Addendum Design. If the pinion is made with a long addendum and the gear is made with a short addendum, the bending strengths are closer to a balanced condition. Figure 3 shows the positive effect of the addendum change on the specific sliding ratios. The pinion specific sliding ratio is reduced from 8.0 to 3.4, a 57% reduction.

As shown in Figure 4, the strength factor of the pinion is now increased 40%. This is a more favorable condition. At the optimum tooth depth, the pinion strength is higher than that of the gear. This comparison of a 24-tooth pinion represents a condition that readily allows for deeper tooth depths.

Often it is necessary to select a lower number of teeth for strength and/or for the increased depth to accommodate an acceptable profile contact ratio over the range in the effective operating center distance. As the numbers of teeth in the pinion get smaller, the outside diameter further restricts the tooth depth due to pointed teeth preventing larger tooth depths.

Working Depth Conclusions

The optimization example using the parameter of tooth working depth increased the strength factor by more than 100% for both equal-addendum and modified-addendum designs. This demonstrates that shallow depth or stub teeth are NOT stronger than teeth with a greater working depth as long as the tooth accuracy allows for load sharing. It points out that the gear profiles must be accurate to maintain the load sharing, assure smooth-motion transmission and deliver the performance required.

Other Parameters

All other gear specifications are also candidates for optimization. They include the pitch, pressure angle, helix angle, addendum modification, root clearance, face width, root and tip radii, tooth thickness, center distance and profile modifications. Every application has its own unique requirements. Every gear design can benefit from optimization. Plastic gears offer the greatest optimization opportunity because the gear design and the required tool are not constrained by tooth form standards.

The best time to apply the results of optimization is when a new tool is required. Optimization is an integral part of a robust design and tool manufacturing process. It represents a significant opportunity to reduce total costs, size, noise, and improve time-to-market with increased customer satisfaction. ○

Reference

1. Universal Technical Systems, TK Solver Program *60-610 Plastic Gear Geometry and Load Analysis* utilized.