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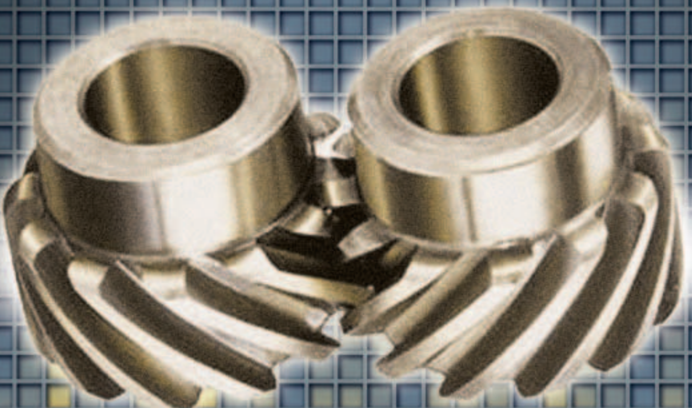

by William S. Rouverol

Less noise and increased power density are just two advantages of this new type of helical gearing.

Introduction

A 25 to 35 percent increase in power density and a markedly lower operating noise are two major performance advantages that a new type of helical gearing, called "OptiGearing," offers over conventional helical gearing. These advantages are a result of three novel features incorporated in OptiGearing:

1) A major change in the type and amount of tooth working surface relief, called "Function-Specific Modifications" (FSM).
2) A new way to proportion the field of contact that reduces the transmission error, called "Integral Virtual Field" (IVF).
3) A method of minimizing noise and vibration by providing for simultaneous loading and unloading of the teeth, called "Synchronized Cross Modifications" (SCM).

Function-Specific Modifications

OptiGearing has three forms of modification: profile, lead, and cross. Conventional gearing usually only has two forms of modification: profile and lead. That the addition of cross modification gives to OptiGearing a 25 to 35-percent advantage in power density came as a major surprise. Essentially this increase in power density results from the greatly increased possibilities for eliminating the excess profile relief that is locked into conventional gear designs.

Why is excess relief a concern? Because every cubic microinch of working surface that is removed reduces the load that that particular cubic microinch would otherwise carry. And this incremental reduction in load does not disappear. It is simply transferred to all other areas of the working surface, and since one of these areas is a critical area for gear failure, it can be stated unequivocally that every cubic microinch of excess relief produces a measurable loss in torque capacity, even if this region of excess relief is nowhere near the critical area for gear failure.

In view of this detrimental effect of excess relief, it is obviously desirable to reduce it as much as possible. Figures 1, 2, and 3 are provided to show how three different types of relief introduce different amounts of relief volume, by which we mean the amount of surface material removed by shaving or grinding inside the pure involute helicoid. Figure 1 shows a field of contact for a conventional helical gear pair, shaded to show the area where the tooth loads are reduced by tip and root profile relief. (Although most power train gears will also have lead modification, it would presumably be the same for all three figures, so it is not germane to the comparison.)

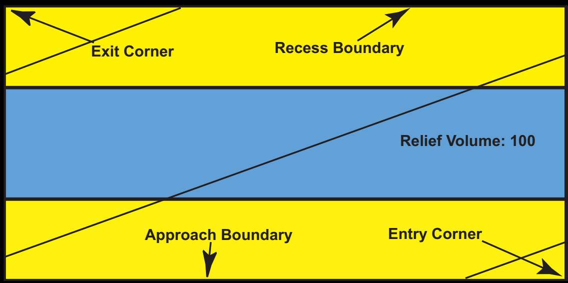

Figure 1 — Typical field of contact for conventional helical pair, showing the area having profile relief in a two-relief-form system.

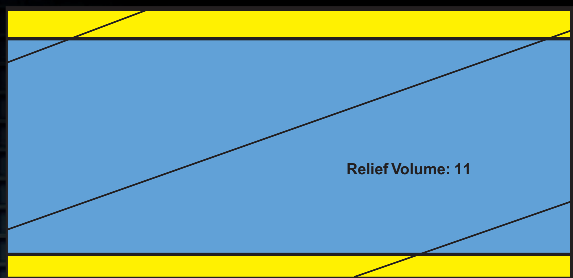

Figure 2 — Field of contact for an OptiGearing pair, showing area having gear profile relief area in a three-relief-form system.

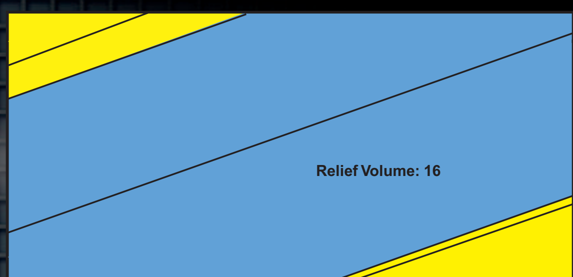

Figure 3 — Field of contact for an OptiGearing pair, showing relief area produced by cross modification on pinion.

To generalize the particular case illustrated in Figure 1, the relief volume has been normalized to a value of 100. Like all conventional profile relief, the transverse cross sectional area is uniform for the full length of the tooth. In order for the total relief, including the lead modification, to be at least as great as the mesh deflection at peak load, the depth and width of the profile relief at the entry and exit corners of the field is typically three or four times as great as the amount needed to alleviate the "buttress cusp" shown in Figure 4 and explained below. Since the factor of at least three applies to both depth and width of the base area of the "relief volume," that volume is only one ninth as great (11 percent) as the profile relief volume needed to provide sufficient initial working surface separation at the entry and exit corners of the field of contact.

If two thirds of the profile relief provided at the entry and exit corners of the field of contact has been eliminated by making the shaded areas of Figure 1 narrower, and representing a shallower relief by two thirds, how will adequate initial entry and exit corner separation be obtained? The answer to this question is shown in Figure 3: cross modification. When this additional form of modification is introduced, since it focuses its main relief depth at the entry and exit corners, it is ideally suited to its job: to provide working surface separation at the entry and exit corners of the field of contact sufficient to prevent impact loading and unloading.

If the "relief volume" of the cross modification shown shaded in Figure 3 is calculated by methods analogous to those used for the relief forms shown in Figures 1 and 2, the relief volume for the normalized cross modification by itself works out to be 16. Combining this with the relief volume of Figure 2, the total normalized relief volume is 27. This means that the total relief volume for an OptiGearing pair is only about one fourth of that for a comparable conventional helical gear set (100).

The ramifications of this startling increase in power density attainable by eliminating excess relief are far-reaching. What it means is that the two-relief form system of modifications for power train gearing has been made obsolete, that it is 25 to 35 percent too large, too heavy, and too expensive. Could the computer simulation program used to calculate the tooth stresses be wrong? The program used for calculating stresses in both the two- and three-relief form gear types was Ohio State's well-known "LDP" Load Distribution Program, modified only slightly so that it could handle both types of relief systems. Gear designers all over the world have used the LDP to design thousands of gear pairs, all of which have proven in innumerable applications to function exactly according to the LDP predictions.

Figures 1, 2, 3, and 4 show why it is a mistake to design helical gearing that has profile relief but no cross modification, or has cross modification but no profile relief. Quite a few researchers have studied cross modification to try to find out if it is better or worse than conventional profile relief (References 4-9). These studies have not been conclusive. The studies asked the wrong question, which should have been: "What mix of profile modification and cross modification maximizes the power density?"

Cross modification was invented in England in 1953 (Figure 3 of Reference 6) in a disclosure that called it "end relief." Figure 4 in this same reference also introduced the concept of combining tip relief with this "end relief," but in a manner that sacrificed most of the advantages of the cross modification relief form. In the 1953 disclosure the cross modification was given the shape of a truncated triangle in tandem with a part-length profile relief strip. This construction was not only unsuited to computer analysis of load distribution but also made the amount and shape of the two forms of relief interdependent instead of independent so that the combined relief form could be "function specific." In OptiGearing, the two relief forms are not in tandem, but are instead in contiguous layers, one on the pinion working surface and one on that of the gear, so proportioning to achieve an "optimum buttress effect" becomes possible.

The "function specific" concept is the key to the comparative advantage of a three-relief form system over a conven-

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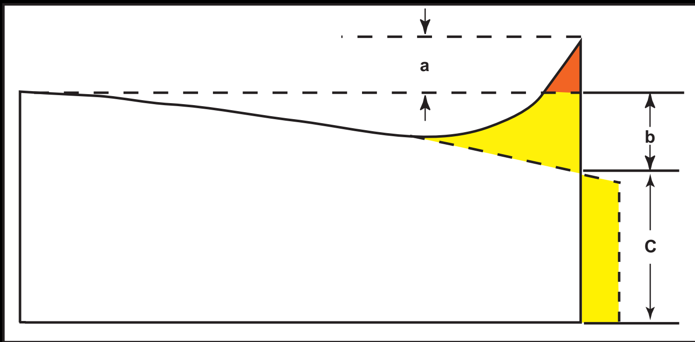

Figure 4 — Typical plot of unit pressure along a contact line that intersects an approach or recess field boundary, showing (a) stress cusp top that must be removed by profile relief, and (b) stress cusp bottom that acts to increase effective length of field of contact in load distribution and torque capacity calculations.

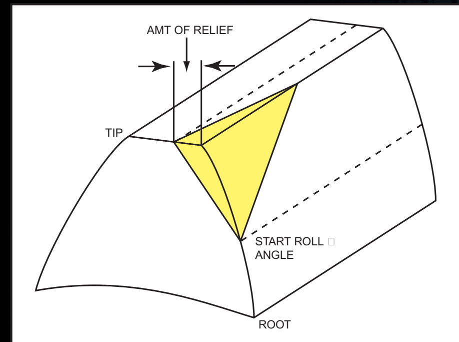

Figure 5 — A schematic view of a gear tooth with cross modification of the working surface of one tooth corner. The pyramidal form focuses relief at an entry or exit corner.

tional two relief-form system. This is because gear pair relief has three discrete jobs, so the conventional two-relief form is inherently inadequate. The three essential corrections are for: 1) misalignment and lead error; 2) alleviating the buttress cusp, and; 3) providing adequate initial separation at the field of contact entry and exit corners. Since conventional two-relief form modifications seek to treat both corrections (2) and (3) with a single relief form (profile relief), and correction (3) is typically three or four times as great as (2), this double-duty assignment automatically introduces a major amount of excess relief, and with it a major loss of torque capacity.

The "buttress effect" is a local stress cusp that is produced at tooth portions adjacent to the approach or recess boundaries of the field of contact in response to the stiffening of the tooth by shear support provided by tooth portions immediately before or beyond those boundaries. The shape and amount of the stress cusps can be controlled by varying the amount of tip or root relief at the approach and recess boundaries (see also Figures 7, 10, and 15 of Reference 10). It appears that the "optimum buttress effect" is one that is the result of a profile modification that lops off only enough of the cusp peak to insure that the local stress is no greater than the peak stress elsewhere along the tooth. It is this modification that affords the relief volume of value 11 shown on Figure 2.

Integral Virtual Field

In the ongoing battle of gear designers and researchers against gear noise and vibration, there have been many proposals as to how best to minimize mesh stiffness variations, which is generally accepted to be the primary cause of gear noise. One of the best-known proposals was made in 1961 by Edward Wellauer (Reference 11), who recommended the use of an integer face contact ratio because it caused the combined length of the several discrete contact lines to remain constant as these lines traversed the field of contact. Critics of the Wellauer thesis contend that profile and lead modifications greatly reduce the usefulness of the Wellauer concept.

A more recent analysis of gear noise and vibration, in 1979 (References 12 and 13), by William Mark, concluded that the product of the profile and face contact ratios should be an integer, at least to a "first approximation." This first

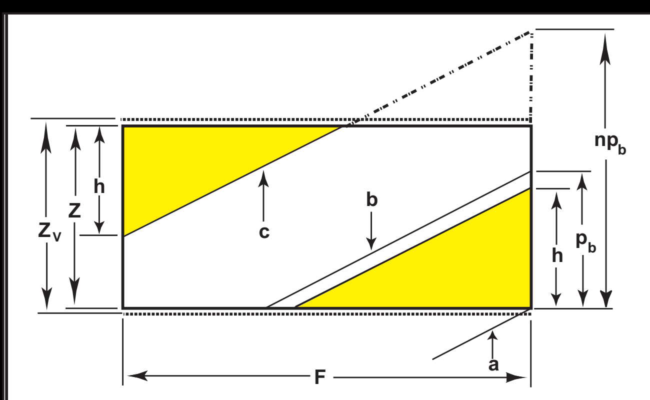

Figure 6 — Field of contact for helical gear pair that has synchronized cross modification.

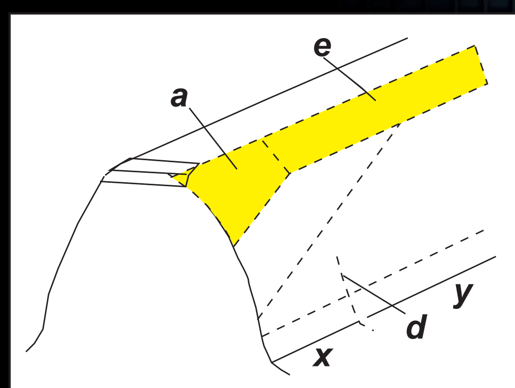

Figure 7 — A reproduction of a figure from U.K. Patent No. 741,376 (1953) showing a helical gear tooth provided with a truncated combination of profile and cross modification.

approximation becomes a "close approximation" if the product involves the virtual profile contact ratio instead of the actual profile contact ratio. The virtual profile contact ratio takes into consideration a buttress effect based on the effective tooth length which is 5 to 10 percent greater than the actual tooth length. The model for this concept may be seen in Figure 4 where a small increment, shown in broken line at the right end of the stress plot, takes into effect the bottom part of the buttress cusp so calculations of load distribution and tooth stress can be more accurate. The multiplier used to calculate the virtual profile contact ratio is of course dependent on the amount of profile relief used in the design, since this controls how much of the cusp top is lopped off. When the profile relief amount is as diagrammed in Figure 2, the buttress effect multiplier is about 1.10. Whenever design time allows, LDP runs with three or four different buttress correction values should be done in order to find the low point of the transmission error curve.

Synchronized Cross Modification

One special form of OptiGearing is diagrammed in Figure 5, which shows the cross modification areas at the entry and exit corners of the field of contact. (The narrow bands of profile relief have been omitted in the interest of clarity.) The narrow "buttress effect" extensions at the approach and recess boundaries are indicated in broken line.

It will be seen in Figure 5 that three lines of contact a, b, and c, are spaced apart in the transverse direction (vertical) by one base pitch and are moving upward at a common velocity. At the particular moment illustrated, contact line a passes through the lower right corner (the "entry corner") of the field, while contact line c coincides with the starting line (hypotenuse) of the cross modification triangle that contains the exit (upper left) corner of the field as its vertex. Because contact lines a and c are separated by exactly two base pitches, the unloading of line c will be in perfect synchronization with the loading of line a.

Happily there are enough arbitrary gear design inputs to allow a designer to incorporate both the Integral Virtual Field concept and the above-described Synchronized Cross Modifications in the same gear set. Both expedients are powerful reducers of transmission error, and are completely independent of each other. Probably a few years of service history will be needed to determine which of these methods of reducing mesh stiffness variation is the most effective and produces the most robust gearing, or whether the use of both methods of noise reduction will be favored.

LDP Predictions for a Sample Design

To put OptiGearing into perspective with respect to gearing with only two forms of relief (lead and profile), it is useful to make computer simulation runs of comparable designs.

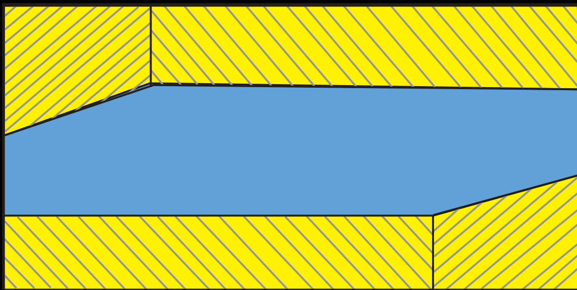

Figure 8 — The field of contact of the gear tooth shown in Figure 7. The lack of overlaid crosshatching indicates the relief type is "pre-emptive."

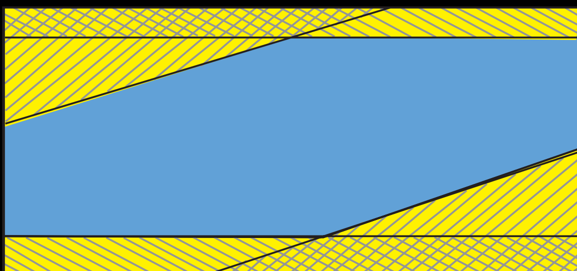

Figure 9 — The field of contact of a helical gear tooth that has additive cross modification and reduced area profile modification as shown by the overlapping crosshatching. The two forms of relief are additive because they are on different members of the pair.

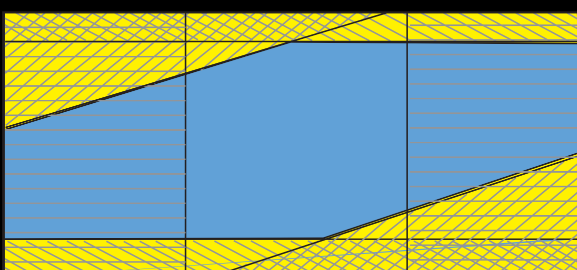

Figure 10 — The field of contact of a tooth such as shown in Figure 9, but with crowning added to the tooth working surface that has the profile modification.

Analysis of any helical gear pair will disclose the main differences between OptiGearing and two relief-form control gearing. The gear pair chosen for the comparison is a 17T-35T set with a 13.8544-degree helix angle. In order to make the comparison as meaningful as possible, the control gearing had all relevant design parameters identical to those of the OptiGearing aside from the modifications. Both sets had the same input pinion torque test values of 1000, 2000, 3000, 4000 and 5000 in-lbs (546 N-m).

To make the most of the Figure 6 curve sheet, it is necessary to select the design material early on, so that the line a-a′ may be drawn representing the allowable value of the contact stress. The intersection of line a-a′ with the maximum contact stress plots discloses the allowable pinion torque for each of the two sets. If the allowable contact stress is, say, 188,000 psi (line a-a′), the allowable pinion torque loads are 3,600 in-lbs for the control gears and 5,000 in-lbs for the OptiGearing. This gives a ratio of 1.32, so the OptiGearing set may be said to have a power density that is 32 percent greater than that of the two-relief form control gear set.

This same kind of comparison should be carried out with respect to the maximum tooth root stress. The advantage of the OptiGearing will not ordinarily be exactly 32 percent, but will generally fall between 25 and 35 percent.

Another plot of an important comparative feature of the OptiGearing and the control gear pair is shown in Figure 7. It will be evident that when each of the two sets is transmitting its design torque (5,000 in-lbs for the OptiGearing, and 3,600 in-lbs for the control set), the transmission error (circled points) for the OptiGearing, at 13.8 µin, is approximately half that of the control gears.

ABOUT THE AUTHOR:

William S. Rouverol, Ph.D., is one of America's more versatile inventors, with many U.S. and foreign patents. He has taught mechanical engineering at UC Berkeley and Oxford University in England, and has a long history as a gear noise consultant to Ford, GM, John Deere, Harley-Davidson, and other gearmakers. He is also the designer of the first Votomatic voting machine, which is now in the Smithsonian. At 85 he is still active, with four new patents pending, including three on gearing and one on a chad-free voting machine. He is the inventor of the gears currently being used in the Segway Human Transporter and is the cofounder of OptiGearing LLC. He can be reached at (510) 841-2474, and e-mail can be sent to Franz Ross at franz@optigearing.com. The company's Web site is [www.optigearing.com].

ACKNOWLEDGMENTS

Readers of the foregoing article will be aware that whatever contributions it may contain are very much the result of building on the work of earlier researchers, as is usually the case with most engineering and scientific advances. The IVF concept, for example, came out of two brilliant papers by Dr. William Mark. Prof. Houser, director of the Ohio State Gear Lab and father of the LDP, and his assistant Jonny Harianto, who modified the LDP so it could handle three-relief form modifications, both made important program contributions without which the new concepts would have been forbiddingly difficult to analyze.

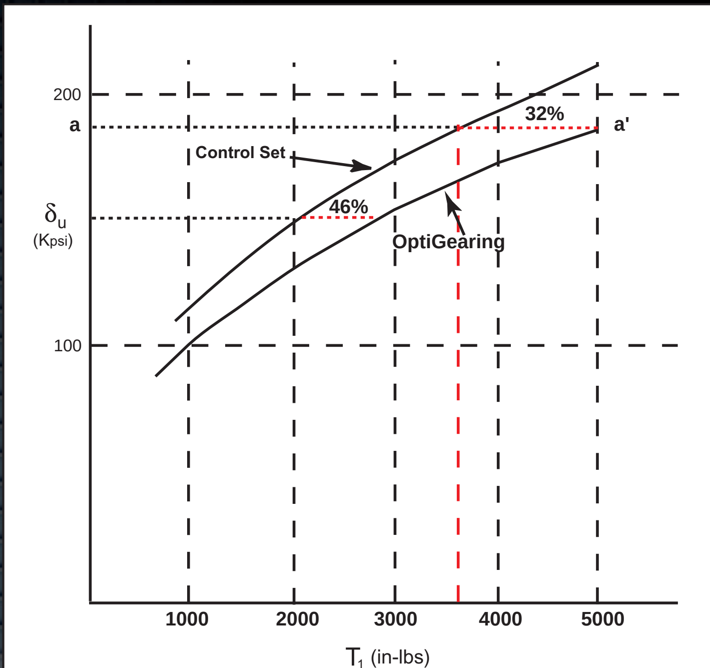

Figure 11 — Plot of maximum contact stress for an OptiGearing set and a control gear set having the same geometry except for the modifications.

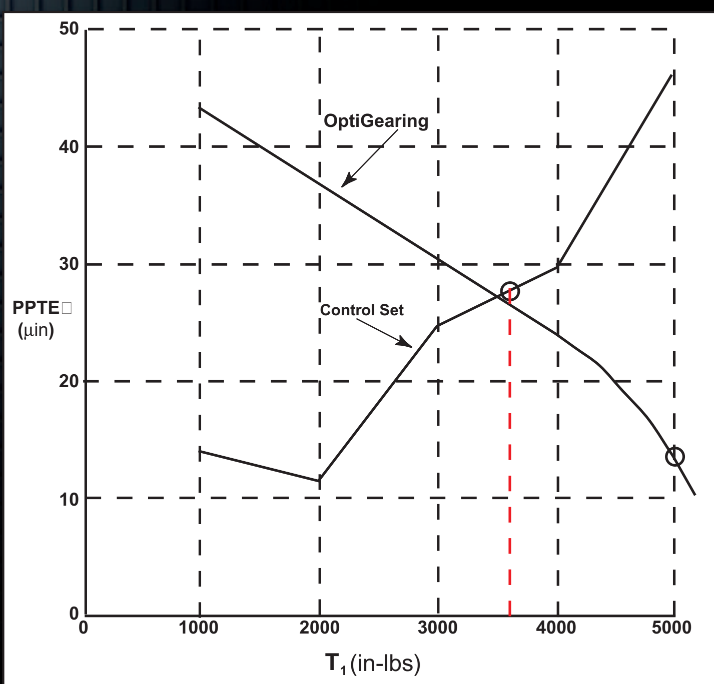

Figure 12 — Plot of Peak-to-Peak Transmission Error of an OptiGearing set and a control set having the same geometry except for the modifications.

REFERENCES:

1. Rouverol, W. S., “High Performance Gearing,” U.S. Patent Application No. 10/425,853, April 2003
2. Rouverol, W. S., “Gearing With Function-Specific Modifications,” U.S.Patent Application 10/438,382, May 2003
3. Rouverol, W. S., “Gearing with Synchronized Cross Modifications,” U.S.Patent Application 10/438,459, May 2003
4. Sundaresan S., Ishii, K., and Houser, D.R., “A Procedure that Accounts for Manufacturing Errors in the Design Minimization of Transmission Error in Helical Gears,” AGMA paper 90FTM3
5. Kiess, W., and Price S., “Noise Reduction Through Generated Engagement Relief Modification,” AGMA Paper 96FTM3
6. Cleff, P.H., and Newman, A.D., “Improvements in and Relating to Helical Tooth Gearing,” United Kingdom Patent No. 741,376, Feb. 1953
7. Sundaresan, S., Ishii, K., and Houser, D.R., “Design of Helical Gears,” JSME MPT ’91; pp. 1-6 Nov. 1991
8. Bayside Controls, Inc., brochure, “Stealth Planetary Gearhead.” 1998
9. Umezawa, K., “Low Vibration Design on a Helical Gear Pair,” *Gear Technology*, Jan-Feb 2000
10. Schmidt, G.R., Pinnacamp, W. and Winder, A., “Optimum Tooth Profile Correction of Helical Gears,” ASME paper 80-CI/DET-110, 1980
11. Wellauer, E. J., “Nature of the Helical Gear Oblique Contact Line,” Industrial Mathematics, Vol. XI, Part 1, 1961
12. Mark, W. D., “Analysis of Vibratory Excitation of Gear Systems: Basic Theory,” J. Acoust. Soc. Am. 63 (5) May 1978
13. Mark, W. D., “Analysis of Vibratory Excitation of Gear Systems II: Tooth Error Representations, Approximations, and Application,” J. Acoust. Soc. Am. 66 (6) Dec. 1979
14. Houser, D. R., and Harianto, J., “Design Robustness and its Effect on Transmission Error and Other Design Parameters,” *Gear Technology*, March-April 2003, pp. 18-25.