GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

JANUARY/FEBRUARY 1992

DESIGN GUIDELINES FOR HIGH-CAPACITY BEVEL GEAR SYSTEMS

SHOT PEENING SPECIFICATION - PART II

RIGHT & WRONG OF MODERN HOB SHARPENING

PRODUCT LIABILITY DEFENSE

CLOCKWORK GEARING

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JANUARY/FEBRUARY, 1992

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SPECIAL FEATURES

DEPARTMENTS

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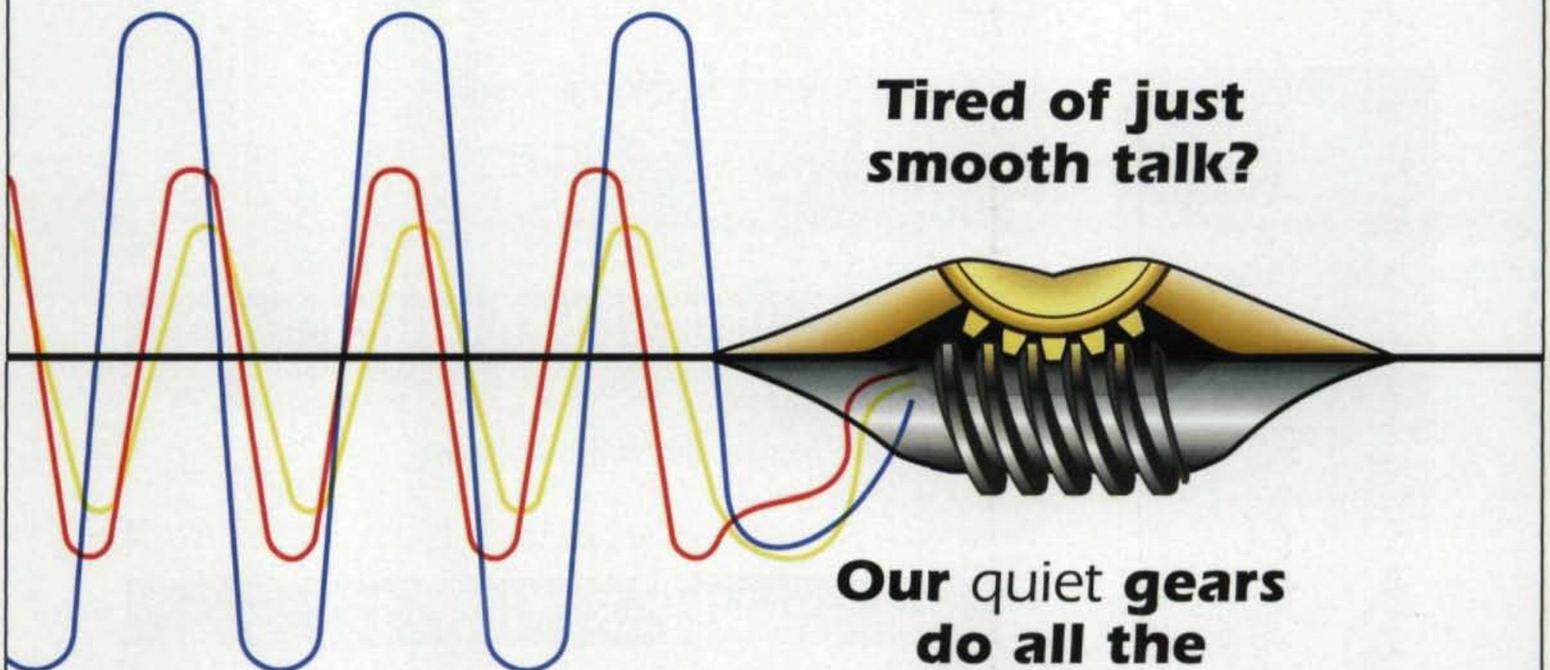
Tired of inadequate inspection capabilities?

Tired of promises and no action?

Tired of gear suppliers that make you look bad?



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Bourn & Koch Machine Tool Company, Rockford, Illinois Provides Unique Solutions to Solve Gear Manufacturing Problems

Problem - A small fractional horsepower motor manufacturer required quieter gears for his speed reducer gear boxes. These motors were used in office and computer products and were 100% inspected for noise level prior to shipment.

After the existing manufacturing equipment and method of processing was evaluated, Bourn & Koch determined that the BEST FIT SOLUTION was to better utilize the equipment by providing a gear manufacturing solutions seminar.

A one day seminar was presented with emphasis on controlling the design, processing, blank and hobbing quality, as well as the items that had an influence on gear noise transmission.

The seminar was presented to the operators, inspectors, engineers, and supervisors in a classroom type of setting. Each attendee received a text book for reference in solving future problems.

End Result - Bourn & Koch Best Fit Solution provided a better understanding of gear manufacturing and the factors most important on producing quieter gears. Class AA hobs are now being sharpened to Class AA tolerances, and blanks being controlled by gear quality requirements.

Problem - A portable electric tool manufacturer with many mechanical Barber-Colman Hobbers required more accurate change gear set ups that worked the first time and fit the gear boxes without trial and error.

Helical pinions and gears required "hobbing to green leads" due to heat treat distortion after hobbing.

On differential type hobbing machines, an accurate method of calculating ratios with four gears, using change gears from 20 to 100 teeth was also necessary.

A BEST FIT SOLUTION was to provide four computer software programs which provided the set ups and the processing information with the routings.

End Result - customer now has several people trained to use the gear software for change gear set ups, production estimating. The customer can also calculate four gear ratios which will achieve specific accuracies using a standard complete set of change gears.

Application determines the BEST FIT SOLUTION. Not every application can be addressed by new single purpose or new universal products. Even though Bourn & Koch has a full line of CNC and manual hobbing and gear shaping machines, we evaluate the application and let the solution determine whether new equipment or better utilization of existing equipment is the best solution.

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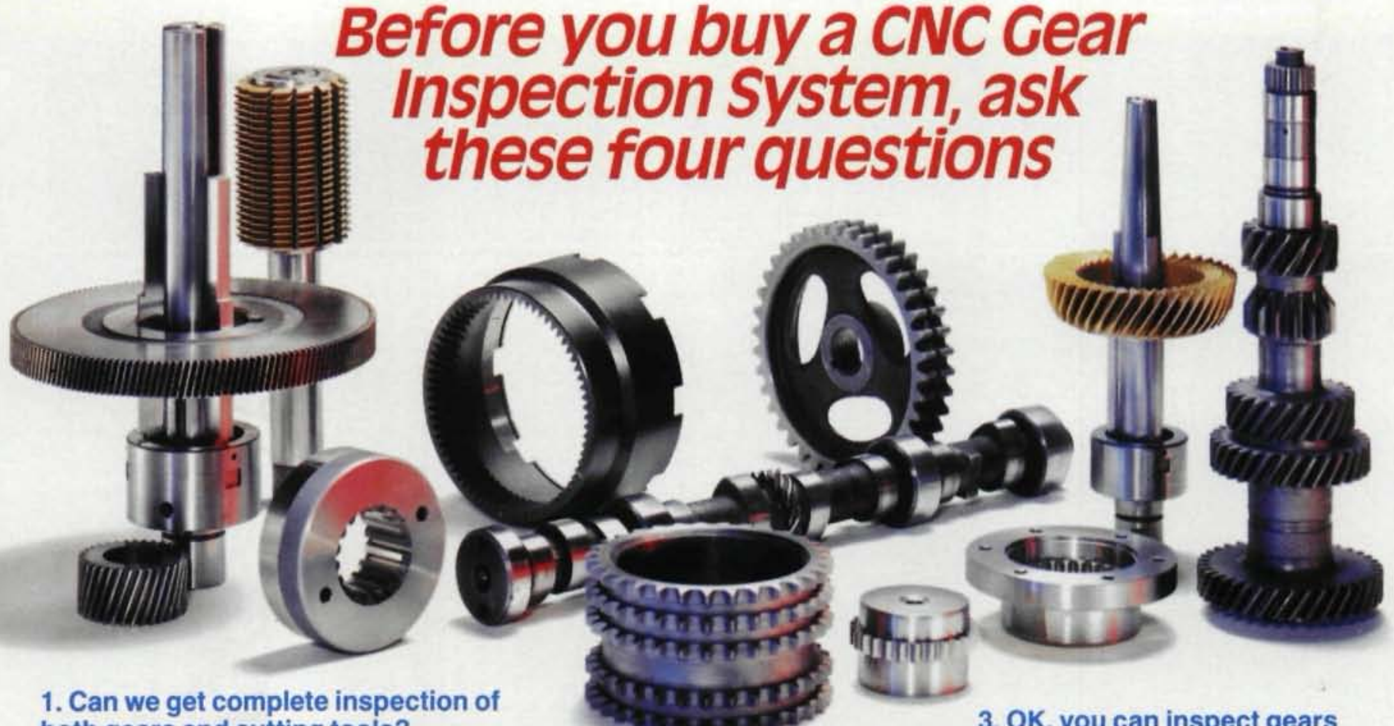
Before you buy a CNC Gear Inspection System, ask these four questions



1. Can we get complete inspection of both gears and cutting tools?

With the M & M QC 3000 System, the answer is yes. The system will test lead and involute profile characteristics of internal or external gears and splines. It is the only universal gear tester available which provides true index testing without expensive attachments. In addition, we have fully-developed software for checking hobs, shaver cutters, shaper cutters, and other cylindrical parts such as worms and cams.

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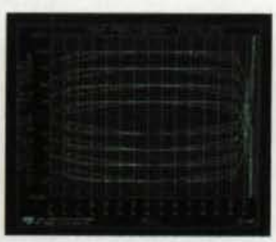
Line/Curve Fitting



Modified K-Chart Analysis

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SPC Run Chart

Topological Map

4. Do you have the technical support team and installation experience to back up the hardware and software provided?

At M & M Precision, we have a technical team with over 45 man-years of experience in developing CNC gear inspection hardware and software. All software for our QC 3000 System has been developed in-house. In addition, we have working installations at these leading companies:

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- TRW
- Ford Motor
- Warner Gear
- Cincinnati Milacron
- Chrysler
- Pratt & Whitney
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The Niña, The Pinta, And The American Gear Industry

Next year will be the 500th anniversary of Christopher Columbus' famous "discovery" of America. Poor Columbus has fallen on hard times of late, what with revisionist historians smacking their lips over his more notable failures and reminding us that American natives have a vastly different point of view on this Great American Success Story. But before we relegate the Great Navigator to the scrap heap of trashed-over heros, let's take one last look at some of the positive lessons to be learned from the Columbus experience - ones that could be instructive to our current situation in the American gear industry.

Whatever other complex factors motivated Columbus and his fellow explorers to travel, and the various kings, governments, and bankers of Europe to finance their adventures, one undeniable goal was the opening of new markets. Our European ancestors got into the exploration business to make money. They were looking to open up trade routes to Asia. What they found, of course, was something else entirely, but those trade routes were what they were looking for to begin with.

The decision to finance Columbus' voyage was a business decision too. It was based on the time-honored rule that risk and reward tend to be proportional - the bigger the risk, the bigger the possible reward at the end.

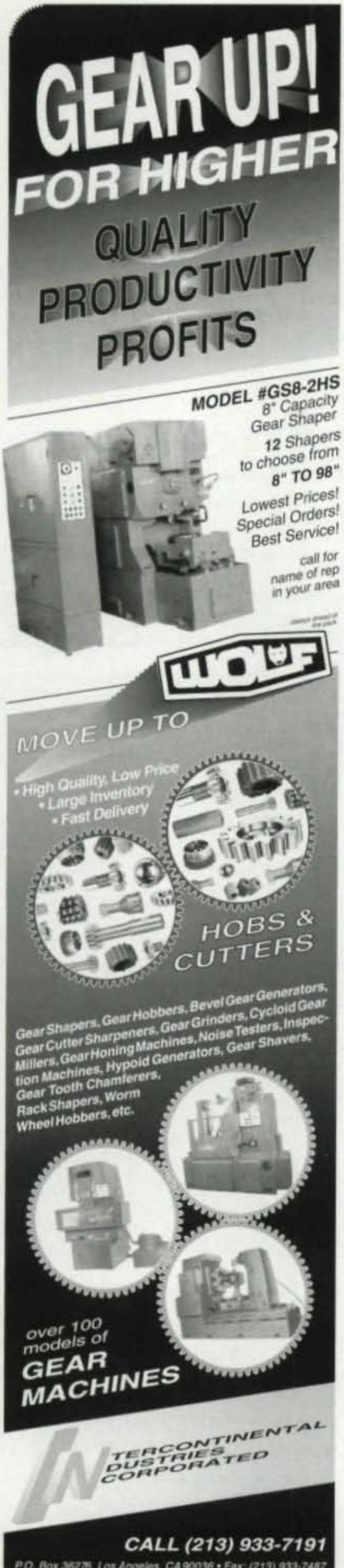
And never doubt the nature of the risks. It wasn't just that no one in Europe was certain of what lay in the Western Atlantic, and that most contemporary maps showed a big, blank space there and the ominous warning, "Here there be dragons." Nor was it just that even a state-of-the-art ship of the day was one most of us would be reluctant to take for an afternoon sail in a mild chop off Oak Street Beach in Chicago.

Some of the risks would sound quite familiar to us today. The "bean counters" in the Spanish court were "bottom line" men too. They were being asked to pay out a lot of money with only a dubious promise of something in return. And they weren't all that familiar with the markets they were trying to break into. Contemporary reports were vague, closer to fairy tales than hard data. All they knew for sure was that vast wealth was to be had by the person willing to overcome the obstacles, but that these were formidable. There were different languages and customs that could get the unwary into trouble-if not eaten-and sophisticated, knowledgeable traders who would strike a hard bargain and were reluctant to let newcomers share their markets. Even if one survived the voyage and the dragons, the competition could be, literally, killing.

The more cautious among them had other good reasons for not straying far from their home markets. It was certainly easier and safer and cheaper to do business with their familiar neighbors. They already had established customers. The older trade routes maybe weren't the best, but they were familiar. The risks were at least known ones.

PUBLISHER'S PAGE

And what happened if Columbus was wrong? What if he got eaten by those dragons? What if the earth really did have an edge, and he and his ships sailed over it into nothingness? What if those "Indians" wouldn't sell them their spices, and who knew how to speak Chinese anyway? Sure, there might be all the riches of a vast, wealthy continent if they succeeded, but what if they



lost their investment?

The decision to enter new markets hasn't changed. It's still a voyage into unknown territory, full of risks and unforeseen problems. The dragons may have changed the color of their scales, but they're still waiting out there, and nobody's investment is guaranteed. But the rewards haven't changed either. For the courageous, patient, and persistent, they can more than cover the risks and expenses.

In the Spanish court, the risk-takers won out, and the rest, as they say, is history. True, the course of success was not smooth. Columbus was lost for much of his voyage to the New World. After he landed, he thought he was someplace else. He never did find the markets he was really looking for. He and his followers made a lot of expensive, tragic mistakes in opening the markets they did find. But they persevered. They learned. And in the end, the success was of epic proportions.

Our situation in the American gear industry is not unlike that of those bean counters in Queen Isabella's court. For every powerful motivation we have for seeking out new markets overseas, there seems to be an equally good reason for sticking close to home. Money is tight. The economy is shaky. The foreign competition is stiff. The economies in some of our most likely markets are complete wrecks, and the paperwork necessary to get into foreign markets could reforest a small, third-world country. Can we afford to spend the money we have on risky ventures that might take years to pay off or might not pay off at all?

Can we afford not to?

What the visionaries in the Spanish court in 1492 knew was that, risky or not, they really had no option. The push to explore was on. Economic and social pressures all over Europe were demanding the widening of vision to include the rest of the world. If the Spaniards didn't reach for the new markets, one of their neighbors - Portugal, France, England - would.

The situation is the same today. The pressures of history demand that we broaden our horizons again. Like it or not, we live in a global village, and plenty of our neighbors are already aggressively exploring the opening markets of Eastern Europe, Asia, and South America. With our own markets at home shrinking and our industry growing smaller and smaller, can we afford not to develop new markets both at home and overseas?

Much of the American gear industry seems to me to be stuck in a timid, conservative mind-set where we are reluctant to take the risk of marketing ourselves aggressively - either at home

PUBLISHER'S PAGE

or abroad. Like some of Queen Isabella's bean counters, we see the world as limited to our own corner and are content to leave it that way. Let someone else run the risk of sailing off the edge.

This timidity will be our undoing. In the tough economic climate we face today, the gear manufacturers who are succeeding are the ones willing to take the risks and face the unknowns. They are willing to explore new markets here at home or in Eastern Europe, Asia, and South America, and never mind the fact that the profit might not be instant, or that they might make some mistakes and have some colossal failures. They're gambling on their ability to find and develop these new markets.

It's a necessary gamble. We either get out our maps, load our ships, and get moving, or be left behind by the people who do.

Michael Goldstein

Michael Goldstein
Publisher/Editor-in-Chief

Technological Advances... a Major Benefit Of Gear Industry Mergers

To attendees of AGMA's Gear Expo '91 (Detroit) it was obvious that spiral bevel gear producers are due for major advances in gear making and testing equipment. Nowhere was it more evident than at Booth 101, where Klingelnberg and Oerlikon demonstrated some of their capabilities. The (March, '91) joint venture of West German-Klingelnberg with Swiss-Oerlikon consolidates some of the best gear equipment technology in the world.

Together, Klingelnberg and Oerlikon will optimize their technologies to benefit all gearmakers. Oerlikon brings savvy from high volume, automotive-type production of smaller diameter Spiral Bevels. Klingelnberg's universal cutting systems cater to the needs of smaller run, high quality, larger OD producers. Both have fine testing equipment. As the companies blend their product offerings, gearmakers will be the beneficiaries. The results will lie with shorter, more cost-effective run cycles; faster changeover times; higher quality levels; coordinated testing equipment.

KNC-60 Gear Generator produces Spiral Bevel gears up to 600 mm.

Testing and documentation is quicker and more accurate on a PNC 60.

At Expo '91, the combined companies demonstrated a KNC-60 Gear Generator, (Figure A) and numerous Gear Testers (Figure B). The new OPAL parallel axis gear grinding system also attracted buyers' attention (Figure C). The OPAL delivers exceptional accuracies and has CNC controlled dressing of either conventional or CBN grinding wheels. This added flexibility permits cost-effective production of small-to-medium quantities of Spur or Helical gears for the Automotive, Aircraft or Machine Tool industries.

The Klingelnberg-Oerlikon joint venture will also channel development energy on a common path for vibration control, noise reduction and software development. In today's climate of global competitiveness, the importance of technology has never been such a critical issue. The combined companies, working together, will set the pace for gear equipment innovation for years to come.

The OPAL 420 control system speeds grinding and changeover of Spur and Helical gears.

Housed in the Klingelnberg Gear Technology, Inc. headquarters at Strongsville (OH), the combined resources of Klingelnberg and Oerlikon offer a single source for sales, technical service and maintenance for all their product lines.

For further information, contact:
Klingelnberg Gear Technology, Inc.
15200 Foltz Industrial Parkway
Strongsville, OH 44136
Phone (216) 572-2100 FAX (216) 572-0985

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Product Liability Defense

Seven strategies for minimizing your risk

Lawrence M. Kohn

It's every gear manufacturer's nightmare. Your company has been named as a defendant in a product liability suit - one involving serious injuries and death. You're facing endless court appearances, monumental legal fees, and, possibly, seven-figure settlements out of your coffers. The very existence of your business could be on the line. The question is, how do you prevent this nightmare from becoming a painful reality.

Recently *Gear Technology* talked to Elliott Olson, a partner in the Los Angeles firm of Haight, Brown & Bonesteel, lawyers specializing in product liability defense, about protecting your firm from ruinous product liability suits. He recommends the following seven strategies: Indemnification, Compliance, Detachment, Commitment, Control, Warning, and Defense.

Indemnification. First try to get a contract with the final product manufacturer to indemnify you in any lawsuits that involve your product. If you have indemnification, he is responsible for the defense in the event of a suit. At the same time, that defense can be somewhat simplified; and the more complicated a suit is, the more protracted - and expensive - it will be.

The risk of such indemnification to the final product manufacturer is small, especially if you are a long-time supplier. It has its own inspection procedures and presumably knows your product as well as its own. The costs of adding your defense to its insurance is probably not significant, and may also enable the manufacturer to buy components at a lower price, since you will not have to add the cost of additional insurance to your price.

If such a deal can be arranged, the wording of the contract needs to be very specific. Olson warns, "It is a slight trick to properly draw up such a contract. The best way to do it is to get the final manufacturer to 'indemnify, defend, and insure against all claims, liability, etc., including those arising from the sole negligence or defect of the product manufactured by the gear company.'"

But what if you do not have or cannot get such a contract? Then how do you protect yourself?

Compliance. Make sure your product complies with all government regulations covering it and with any applicable voluntary standards, such as those of ANSI, SAE, or AGMA. This provides an external, objective indication of quality. A well-run quality control department is also an important defense. Olson says, "Obviously, the proper quality control is essential." Failure to comply with your own criteria can be fatal.

Detachment. A third defense is to establish an appropriate distance between your company and the final product manufacturer. You should be aware of the final product application and whether the it goes beyond the limits of its design criteria. Olson explains, "If the tractor or whatever was designed to pull a plow, and it's now being used as a battering ram, and the gear manufacturer knows that his gear is not adequate for that use, he should take steps to protect himself, either by warning the manufacturer or final user not to use it for that application, or by beefing up the gear."

However, since the likelihood of this prior knowledge is slim, you should include in the contract of sale a statement that the gear use should be limited to the specific purpose intended. "The gear manufacturer should at least have some specifications to which it is manufacturing the gear and make sure that the component is designed beyond the endurance limit of those specifications," says Olson.

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Lawrence M. Kohn

is a Los Angeles-based marking consultant, writer, and media personality. He specializes in business-related subjects and was the host of KIEV Radio's "It's Your Business". Among the subjects of his interviews has been former president, Ronald Reagan.

On the other hand, be wary of becoming too closely involved in the design of the final product. According to Olson, the danger is this: "Each entity in the stream of commerce of a product is liable if the product is defective. If the tractor has a characteristic which causes it to lose steering, yet the gears provided are adequate, the tractor manufacturer and all entities down the line, includ ing the distributor and the dealer, would be liable, but the gear manufacturer would not be, as long as the gear didn't fail or break, or unless the gear manufacturer participates in the enterprise in some way, such as recommending how the gears should be used, how the tractor should be used, or making recommendations as to the owners' manual, etc."

Commitment. Commitment to safety is the fourth defense. No one sets out to deliberately make a product that will kill or injure someone, but sometimes safety concerns are lost sight of or overridden by other considerations. Management must instill an overall sense of commitment to safety in all departments. Simply put, a balance has to be struck between putting out a product that's too expensive and one that will cost more in the long run because of some catastrophic failure in the field that will cost life and limb. Olson says, "One of the most expensive things you can do is kill somebody or make someone a quadriplegic." Investment in safety should come at the design stage, not later in the courtroom.

Control. A fifth strategy is to maintain careful control of documentation. Sloppily worded documents can be financial time bombs. Be wary of the wording of justifications for design changes. Phrases like, "change made to avoid field failures," or "change made in order to prevent lawsuits," are deadly. Either don't document the reason at all or give other valid reasons, such as cost cutting, maintenance simplification, or performance improvement. Don't admit to safety problems that don't exist.

Every company needs a realistic document retention program so that useless or potentially damaging papers are not kept around for long periods of time. One such document is what Olson calls the "God-help-us-if-something-goes-wrong" memo. Such documents, usually created in the heat of discussions over a projected change, are also potential disasters. Of them Olson says, "There is no need to create such a document or to keep it."

This is not meant to discourage discussion of safety questions, and unsafe products should never go to market. Where the problem arises is in the careless use of language. Olson recommends that all employees be made aware of the potential danger of the kinds of things they write. A careful program of education of employees in proper documentation can save a lot of headaches.

Warning. A sixth strategy is to be very careful about the wording of any product warnings, installation and operating instructions, etc. Don't fall into the trap of thinking that warning against every conceivable danger will protect you from suits. You cannot possibly warn against all the possible dangers, and overwarning degrades the value of the notice.

Warnings should cover the unexpected. Says Olson, "Don't warn against obvious things. Don't warn drivers not to run into people while driving. On the other hand, if you have a clear, odorless liquid that is a deadly acid, the bottle should contain some kind of warning."

Depending on the product, having it reviewed by a product liability lawyer, preferably one with product liability trial experience, an engineer with product liability experience, and, perhaps, a human factors engineer may be a wise move. Olson says, "Sometimes defects can be easily corrected if discovered by such a review."

Defend. Suppose you have taken all these precautions and still find yourself confronted with a product liability suit. Olson's advice in this case is to be prepared to aggressively defend yourself and preferably take control of the defense. Don't just turn the defense over to your insurance company, thinking, well, we pay a lot of money for this policy, so let them worry about it.

Many times insurance companies are tempted to take the safe way out, settle the case, and then later raise your rates or drop your coverage entirely. This could be unfair to you in a number of ways. Chances are, nothing is wrong with your product. Its design is based on sound engineering practice. It meets all the standards, codes, and regulations, and there's nothing wrong with it, except someone has misused it. Accepting a settlement in such a case can cost you money you shouldn't have to pay and can hurt your company's reputation.

The decision to launch an aggressive defense should be made long before the possibility of a law suit arises. It should be discussed when you are negotiating your liability insurance coverage. "That's the time to negotiate the right to identify your own counsel," says Olson.

Once you've made the decision to aggressively defend yourself in such suits, you must also be prepared to present witnesses and company personnel to assist in the preparation and defense of cases. But this investment may be well worth it in terms of a final settlement.

Protecting your company from product liability suits is a complex, ongoing process. It requires forethought, careful planning, and a long-term commitment, but ultimately, that extra planning will pay off in added protection when the worst happens. ■

For answers to questions about this article, circle Reader Service No. 79.

Design Guidelines for High-Capacity Bevel Gear Systems

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Boeing Helicopter, Philadelphia, PA

Abstract: The design of any gearing system is a difficult, multifaceted process. When the system includes bevel gearing, the process is further complicated by the complex nature of the bevel gears themselves. In most cases, the design is based on an evaluation of the ratio required for the gear set, the overall envelope geometry, and the calculation of bending and contact stresses for the gear set to determine its load capacity. There are, however, a great many other parameters which must be addressed if the resultant gear system is to be truly optimum.

A considerable body of data related to the optimal design of bevel gears has been developed by the aerospace gear design community in general and by the helicopter community in particular. This article provides a summary of just a few design guidelines based on these data in an effort to provide some guidance in the design of bevel gearing so that maximum capacity may be obtained. The following factors, which may not normally be considered in the usual design practice, are presented and discussed in outline form:

- Integrated gear/shaft/bearing systems
- Effects of rim thickness on gear tooth stresses
- Resonant response

Nomenclature

D = Harmonic index, integer D = 0, 1, 2...
f = Tooth mesh frequency and harmonics
k = Integer multiplier, k = 1, 2, 3...
n = Number of gear teeth
N = Rotational speed, revolutions per second
Q = Number of mesh points on gear
W_D = Weight of helicopter damping ring, pounds
W_G = Weight of gear rim, pounds
n = Natural frequency, Hertz

Integrated Gear/Bearing/Shaft Systems

Bevel gears are typically manufactured as blanks and then attached to their shafts by a variety of techniques including keys, bolts, splines, and the like. The existence of these joints increases the weight and complexity of the finished assemblies while reducing their reliability, accuracy, and effective load capacity. In spite of these limitations, the two-piece shaft and gear assembly remains the most common configuration because it is the easiest system to design and manufacture. In addition, bearing journals are provided for assembly of bearing races, resulting in a multipart gear/bearing shaft assembly with many potential faying surfaces.

In applications in which weight and reliability are critical considerations, these joints and faying surfaces have been the subject of considerable research. The primary operational problem with these joints is fretting. Under the combined conditions of high stress and a fretted surface, a crack can initiate at this joint and, if undetected, can progress to failure. This problem is significant for the typical helicopter application; thus, much effort has been devoted to coatings, platings, lubricants, and similar devices aimed at reducing the long-term effects of fretting and the ensuing corrosion. The obvious final solution to the problem is, of course, the elimination of the joints and faying surfaces altogether. This can be accomplished either by welding or by designing integral gear/bearing/shaft configurations. The former technique has worked well in many applications, but introduces potential new problems associated with the weld itself. The optimum solution is, therefore, integral design. Because of the complex geometry of bevel gears and the offset generating motions of the cutting and grinding machines, this is easier said than done, especially in the case of complex shafts with projections at both ends of the gear.

The original impetus behind the growth of this technology was the development of the CH-47D helicopter. A decision was made to eliminate all bolted bevel gear joints, either through integral design or welding. Integral design was selected because of lower developmental costs and greater inherent reliability.

The first step in the design of complex, integral-shaft spiral bevel gears is the definition of the path of the cutter with respect to the gear blank. We have developed a computer program which uses manufacturing machine settings to produce plots of the path of various points on the cutting and grinding tools, as Fig. 1 shows.

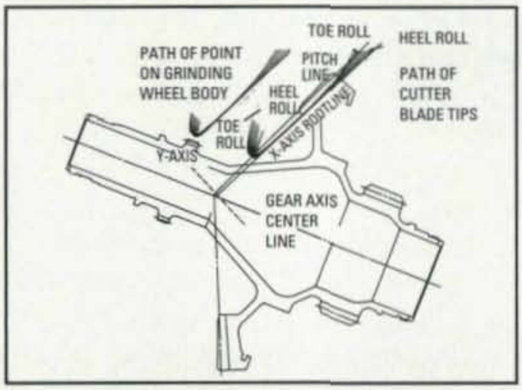

Fig. 1 - CH-47D forward sun/bevel gear with grinding wheel tip and body point plots.

Graphic layouts and model checks are used to ensure that a proper cutter and shaft design can be obtained in order to achieve an integral gear design. Machine and cutter modifications have been developed to achieve these results.[1]

Based upon these developments, all of the CH-47D spiral bevel gears were designed to eliminate all bolted or splined joints. A typical comparison of the CH-47C and CH-47D forward transmission sun/bevel gears is shown in Fig. 2. The concept of integral design was extended beyond the elimination of the bevel gear bolted joint to total integration. For this gear this included the integration of an accessory spur gear and a journal for bearings. The final design of the CH-47D sun/bevel gear incorporated three bearing journals and three gears (two spur and one bevel) into a single integral shaft/gear design.

Fig. 3 also shows the integral bevel gears for the aft and engine transmissions. The engine transmission bevel gear required a significant design change in the center portion in order to

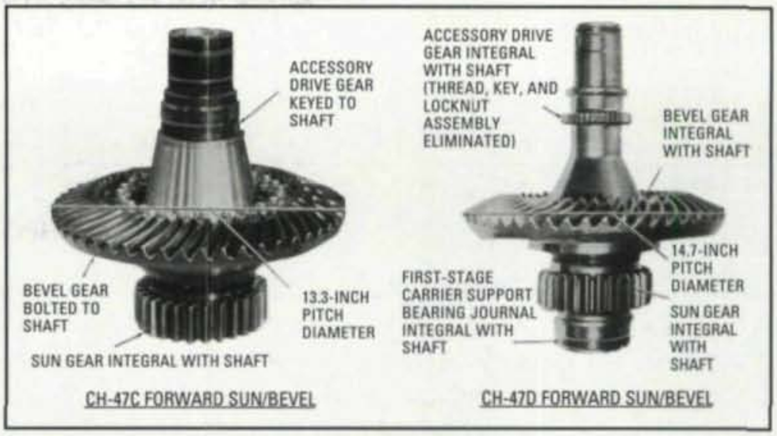

Fig. 2 - Comparison of CH-47C and CH-47D gears.

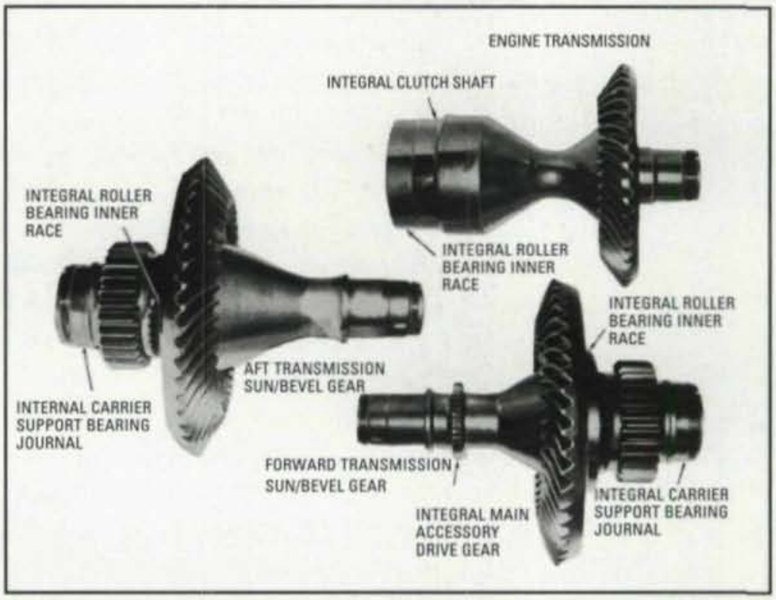

Fig. 3 - Representative example of a integral spiral bevel gears for the CH-47D helicopter.

eliminate cutter interference.

Furthermore, the power transmitted per pound of gear has been increased, as shown in Table 1, by at least 10% and as much as 20% for the growth power level of the CH-47D bevel gears (10,000-hp system) as compared to the CH-47C gears (6,000-hp system). The philosophy of integral design has been taken several steps further in the CH-47D transmissions, permitting a more compact (relative to power transmitted) and more reliable drive system than that of the CH-47C. All of these improvements have been made without significantly increasing the gear tooth stress levels over those of the CH-47C.

The projected production acquisition cost of the integral CH-47D gears is virtually the same as that for the conventional bolted shaft assembly, while the life-cycle cost will be significantly lower because of reduced rejections at overhaul for joint-related problems. The rejection rate is quite high for such problems; thus, eliminating the joint will greatly lessen rejects. This conclu-

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Table 1 Relative Weights of CH-47C Bolted Flange and CH-47-D Integral Bevel Gears			
Transmission	CH-47C (6,000-hp system) (hp/lb)	CH-47D (10,000-hp system) (hp/lb)	Percent Improvement
Forward	89	102	15
Aft	96	105	10
Mix	123	147	20
Engine	175	209	20

sion must be tempered somewhat, since with the integral gears a non-repairable defect on the gear or shaft will result in the entire component's being scrapped, whereas with separate parts, this is not the case. This is a small consideration within the context of the overall system and is far overshadowed by the improvements in weight, reliability, and life-cycle cost.

Effects of Rim Thickness on Gear Tooth Stresses

Many investigators have indicated that the accuracy and applicability of currently available equations for predicting gear tooth bending stress are limited in some applications. Testing and actual field experience have verified the fact that gears which are constructed in such a way that the rim supporting the gear teeth is less than rigid have tooth root stresses that differ substantially from those predicted by conventional theory.

In order to overcome the shortcomings of the existing analytical tools, several researchers have addressed the problem in a variety of ways. Some have proposed new approaches based on variations of simple beam theory, while others have used finite-element methods (FEM). In addition to these analytical efforts, many photoelastic and strain survey investigations have been conducted.

Unfortunately, the vast majority of the research (both analytical and experimental) has concentrated on the gear teeth themselves without regard to the blank configuration. This research has been demonstrated to be inadequate for lightweight, thin-rimmed gears. In order to better understand this phenomenon, a series of investigations was undertaken with a simple segment model. The results of this testing clearly indicated that the root stresses become much more significant as the gear rim thickness is reduced.

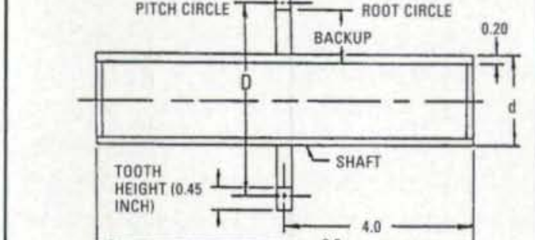

Fig. 4 - Photoelastic test specimen configuration.

In order to accurately evaluate the effect of rim thickness, three different pitch diameters combined with five rim thicknesses were evaluated, as summarized in Table 2, in an extensive series of photoelastic tests.[2-3] In order that the date from each test specimen could be directly compared, the gear tooth configuration was the same on all specimens, as shown in Table 3.

The gear blank construction was also designed to be representative of an actual blank, as shown in Fig. 4. The pitch diameter (D) and shaft bore diameter (d) were the parameters varied during the test. Radial support was provided for the model at each end of the shaft, and one end was clamped to provide torsional restraint.

In contrast to these test specimens, virtually all photoelastic gear testing done before this investigation was conducted with large tooth-segment specimens. For example, in their landmark work on stress concentration factors, Dolan and Broghamer used two-pitch, 14.5° and 20° pressure angle tooth segments. (Fig. 5).

The gear tooth form for each specimen was defined by a computer program developed as a preprocessor for a gear FEM analysis system.[4] This program provided accurate point-by-point coordinate data for both the profile and fillet areas. These data were transferred to a Gerber 4400 plotting system on magnetic tape. This computer-based system then scaled up the data by a factor of 10 and provided sequential tooth plots on Mylar.® The system enabled the plot accuracy to be controlled within a few thousandths of an inch. These plots were subsequently used to manufacture the actual specimens.

A line-master copy milling machine was used to cut the gear teeth on all specimens. The 10-times-size gear tooth plots were placed on the copy mill tracer table and converted optically back into digital coordinate data. The machine then reduced the digital data to actual size and used the signal thus produced to guide an end-milling cutter which cut the actual tooth profile. The speed of the cutter was kept high and the feed rate low, so that the resultant parts had good accuracy and finish; the accuracy of the finished parts was about AGMA Quality Class Q 11.

Two-dimensional models were fabricated from annealed polycarbonate plastic for the 8:1 through 2:1 specimens. These gears were then bonded to aluminum-filled epoxy shafts which were modulus-matched to the plastic blank material (solid aluminum shafts were, however, used for the 4- and 8-inch pitch diameter, 7.5:1 and 3.8:1 backup gears, respectively, to prevent excessive lateral shaft deflection).

Three-dimensional models were required for all the 0.45:1 specimens because the rim thickness was only slightly greater than the shaft wall thickness. The 3-D specimens were cast from an epoxy-based resin system employed phthalic anhydride and amine hardeners; the castings were then machined to the appropriate dimensions. In order to facilitate cutting the teeth on the copy mill, the 3-D shafts were parted near the gear tooth area and rebonded after the teeth were cut. Fig. 6 shows a variety of the specimens used.

The 2-D model assembly - model, load arm, and model restraint - was placed in the optical path of a transmission polariscope, which permits the analysis of the isochromatics (lines of constant color) through the use of circularly polarized light. The shaft was restrained radially at both ends and torsionally at one end, while the load was applied to the central tooth of the five-tooth cluster. Load was applied at three separate locations along the tooth profile: low point of single-tooth contact (LPSTC), pitch line (PL), and high point of single-tooth contact (HPSTC). An isochromatic color photograph (35mm slide) was obtained at each load location for subsequent analysis and documentation. The model was then modified to the next configuration by removing the gear from the shaft, enlarging the gear inside diameter, and bonding to a shaft of larger diameter (2:1 backup ratio).

The 3-D photoelastic models were loaded as shown in the test schematic in Fig. 7. All three positions, LPSTC, PL, and HPSTC, were loaded simultaneously to conserve time because of the "stress-freezing" procedure, which involves a detailed heating and cooling cycle. Subsequent to the "stress-freezing" procedure, the model was sliced by machining off the ends of the shaft so that a constant-thickness, 2-D model through the gear tooth section remained. The slice was placed in the polariscope and analyzed exactly as the 2-D specimens were.

Table 2
Photoelastic Test Specimen Configuration Gear

Dia. D (in.)	Shaft Dia. d (in.)	Backup Ratio	Dash No.	Model Type
16	8.70	8:1	1	2D
16	14.00	2:1	2	2D
16	15.40	0.56:1	3	3D
8	0.75	7.5:1	4	2D
8	6.10	2:1	5	2D
8	7.40	0.56:1	6	3D
4	0.75	3.8:1	7	2D
4	2.10	2:1	8	2D
4	3.40	0.56:1	9	3D

Table 3
Test Gear Tooth Geometry

Diametral Pitch	5.000
Pressure Angle	25.000
Addendum	0.200
Dedendum	0.245
Fillet Radius	0.062
Tooth Thickness	0.310
Tooth Form	Involute
Root Configuration	Circular fillet with no undercut

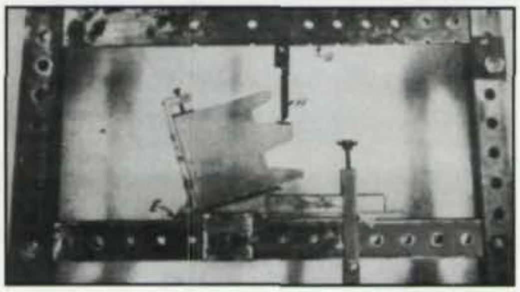
Fig. 5 - Dolan and Broghamer test setup.

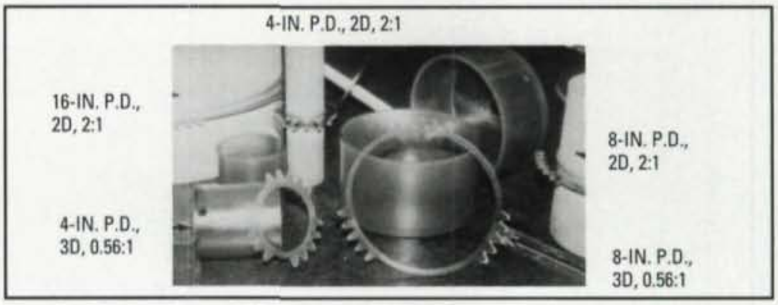

Fig. 6 - Comparison of 2-D and 3-D test specimens.

The photoelastic stress analysis was performed by the color-matching technique, whereby colors of known magnitude are matched to those of the projected image. The ability to discern partial fringe orders is typically ± 0.1 fringe, assuming an initial fringe order of approximately 0.7 or more. Therefore, the accuracy of this technique is directly related to the magnitude of the observed fringes. Throughout this test program, the majority of analyzed fringes were 2 or more, which resulted in an accuracy of ± 5% or better.

The data obtained from the experiments were reduced and converted into stresses. The magnitudes of the stresses of and by themselves are of little consequence, since we are primarily seeking trends. Furthermore, since the models were plastic, the loads and the resultant stresses were quite low relative to those which one would expect on a metal gear of the same size. Despite these facts, it is always interesting to compare experimental data to some known points. We have already indicated that for gears with relatively large rim thicknesses, the conventional equations predict the maximum fillet tension stress with reasonable accuracy. In order to verify this observation, measured fillet tension stresses for the test specimens with the largest backup ratios were plotted as functions of load position. This plot, Fig. 8, shows that the tension stress at the highest point of single-tooth contact (HPSTC) is quite close to the predicted by the conventional equations. The actual deviation varies from about + 8% to + 13%, which is reasonably good correlation.

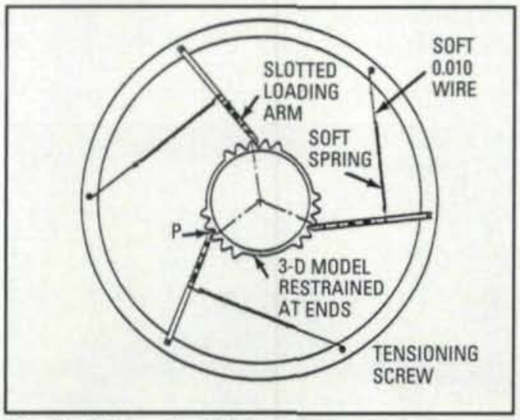

Fig. 7 - Schematic of 3-D test setup.

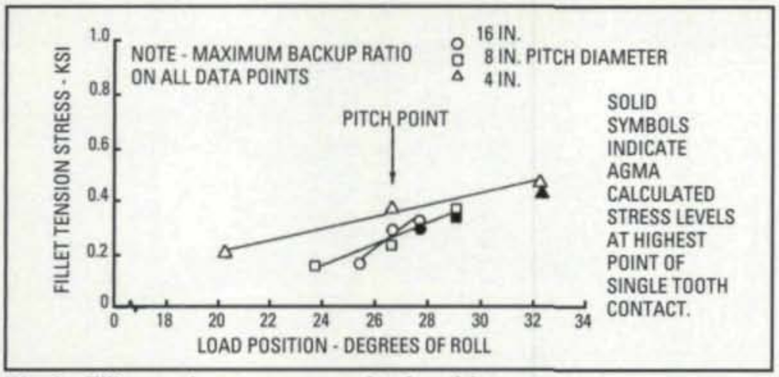

Fig. 8 - Fillet tension stress versus load position.

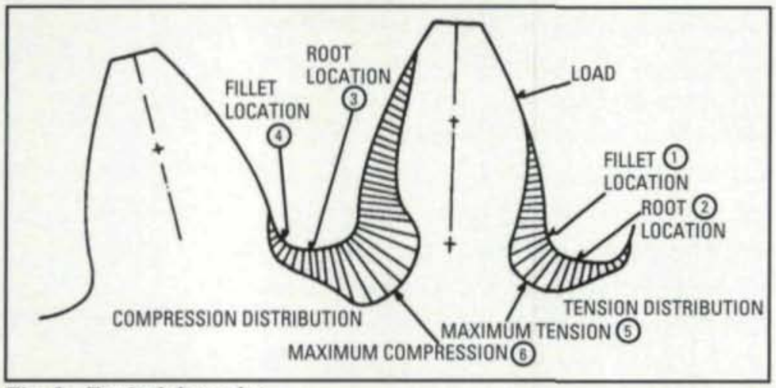

Fig. 9 - Typical data plot.

The stress index calculated with the conventional AGMA equations is based on the maximum tension stress. In fact, however, it is the maximum alternating stress which determines the fatigue load capacity of a gear tooth. With this in mind, most of our ensuing discussion will center on alternating stress levels. Before discussing the actual data, it would be wise to establish some conventions for naming the stress of interest. All of the raw data were reduced to produce plots of tensile and compressive stress as shown in Fig. 9 (which is the fringe data plot for the 4" pitch diameter, 2:1 backup specimen, loaded at HPSTC) for each combination of pitch diameter and backup ratio. The points identified as fillet locations correspond approximately to the conventional AGMA critical bending section. When we speak of fillet stresses in the ensuing discussion, they have been measured at this location. Similarly, when we speak of root stresses, they have been measured at the root location shown in Fig. 9. This point corresponds to the location of the maximum alternating stress in the root. Generally, it is quite close to the centerline of the space, usually on the tension side.

Clearly, for unidirectional loading, the maximum alternating stress does not necessarily occur at the same location as either maximum tension or maximum compression. For bidirectional loading (e.g., idler gear), the maximum alternating stress is usually the result of the combination of maximum tension and maximum compression.

Although we measured stresses with the load applied at several points along the tooth profile, the maximum alternating stresses occurred when the load was applied at the highest point of single-tooth contact. Our discussion will thus be limited to this load case for all models.

One of the first results which became apparent was that, while the trend was apparent, we could not define a firm diameter effect. Our attempts to define a rim thickness effect were much more successful. A large amount of data resulted from this testing, and the interested reader may consult Ref. 3 for more detailed information.

In our analysis of these data, many parameters were plotted and evaluated. Perhaps the most significant are the fillet and root alternating stress and the alternating stress for fully reversed loading shown in Figs. 10-12.

Although many subtle effects may also be noted, the main trend shown in these plots is quite apparent: the stress increases dramatically as the rim thickness-to-whole depth ratio drops below 2:1. The conventional AGMA equations show no such effect. It is also interesting to note that the measured stresses remain reasonably constant as the backup ratio increases above 2:1.

Another parameter which is of interest is the alternating stress that occurs when the gear is subjected to a fully reversed load, as is the case for an idler gear. As expected, the alternating stress level is substantially higher than for either the root or fillet locations when subjected to unidirectional loading. For a given allowable stress level, the load capacity is thus lower when a gear is used as an idler. This fact is by no means a startling revelation. Most designers incorporate a reduction factor when calculating idler gear load capacities. AGMA standards recommend that the allowable stress be reduced to 70% of its unidirectional value for bidirectional loading. Actually the allowable stress does not change; rather, the magnitude of the alternating stress changes. Thus, applying the correction to the allowable index stress in accordance with the AGMA standards is a convenience rather than a factor. In order to define the actual reduction to be expected, we have plotted the ratio of root alternating stress to the fully reversed alternating stress in Fig. 13.

Based on these data, it appears that the 70% suggested by the AGMA standards is a bit optimistic for small diameters, but is certainly reasonable overall.

We have clearly established the fact that the measured stresses are quite different from the calculated stresses. The calculated stresses are, however, stress indexes rather than exact stress levels. The difference between calculated and measured would then be much less significant if a constant relationship existed between the two. It should be obvious that this is not the case, since we have already demonstrated that the measured stresses vary substantially with backup ratio, while the calculated stresses do not.

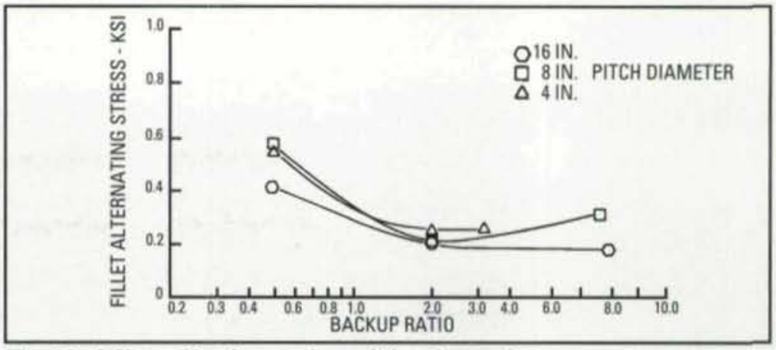

Fig. 10 - Effect of backup ratio on fillet alternating stress.

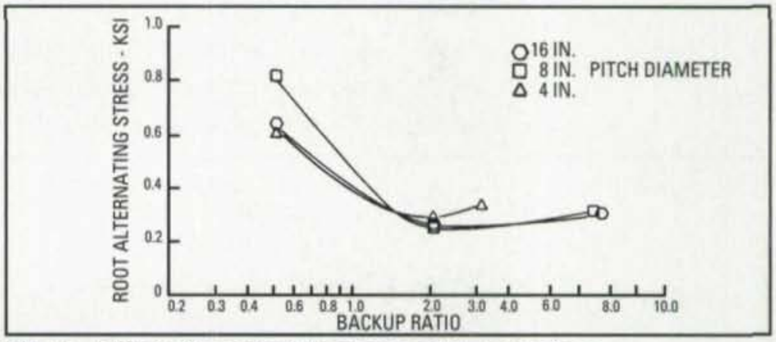

Fig. 11 - Effect of backup ratio on root alternating stress.

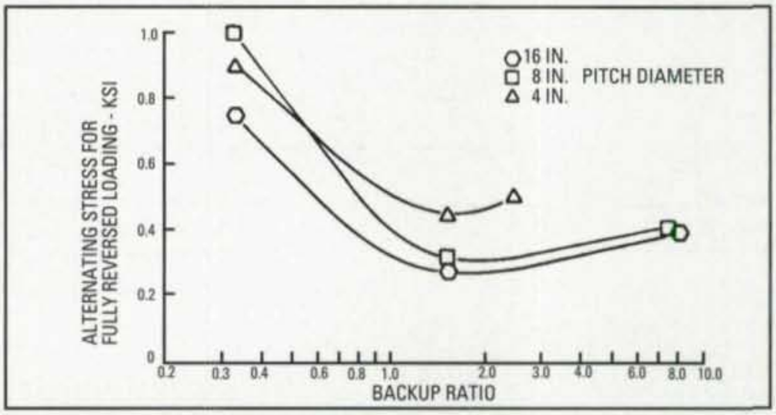

Fig. 12 - Alternating stress for fully reversed loading.

Plots of test data clearly indicate that a constant relationship does not exist, at least backup ratios below 2, between the actual stress levels and the conventionally calculated index stresses. These data may, however, be used to develop a rim thickness factor, K_B, which can be used to improve the accuracy of the conventional equations. Fig. 14 shows this factor with data from both the photoelastic test program and an investigation which used the FEM analysis to be discussed in the next section superimposed on the curves. The development and use of this factor are covered in more detail in Ref. 5, and the reader should consult this paper before applying K_B in design.

While these data have been developed with spur gear models, the basic trends apply to all types of gears. Great care must be exercised when applying these results to bevel gears with complex blanks, particularly those with T-shaped sections. The hard point over the center section can act as a concentration point for root stresses, further aggravating a bad situation.

Resonant Response

Gear resonance is one of the most insidious and destructive of all gear failure modes. It generally occurs quite suddenly and with catastrophic results. Since a gear often fails by the separation of large fragments from the blank, the damage is usually extensive, especially when high rotational speeds are involved.

The prevention of this type of failure is relatively simple once resonance has been identified as the causative factor. Either of two approaches may be taken: The gear can be redesigned so that its natural frequencies do not coincide with any operational excitation condition; or the blank can be damped so that even if excited at its natural frequency, the response of the gear is small enough to preclude failure.

Resonance phenomena in gears, while not widely treated in the literature, can be of considerable consequence in the design of high-speed, lightweight gearing. Should the blank be excited at a frequency sufficiently close to one of its resonant frequencies, deflections and, hence, stresses can increase to levels where failure can occur within a relatively short time. Fig. 15 shows the increase in stress resulting from operation near a resonant frequency. Stress plots were obtained from telemetered strain-gage readings during dynamic transmission tests. Strain gages were located on the flange of a spiral bevel gear. It is interesting to note that although the torque remained constant, a change in shaft speed of less than 9% resulted in an increase in stress level of over 73%. In this example, a further increase in speed of about the same magnitude would cause the stress level to fall again to the 3,800 psi range. The narrow speed band sensitivity is characteristic of gear resonance problems.

The origin of resonant failure is at or near the outermost part of the blank in the gear tooth root, where the combined tooth-bending and blank resonance stresses are maximum. The crack progresses through the blank as shown in Fig. 16 and returns to the gear outer diameter, thus removing a wedge-shaped fragment from the blank. Gears which suffer this failure mode are usually, but not always, operating at high rotational speed; thus the high-energy fragment causes considerable damage, as Fig. 16 demonstrates. Because of the catastrophic nature of the

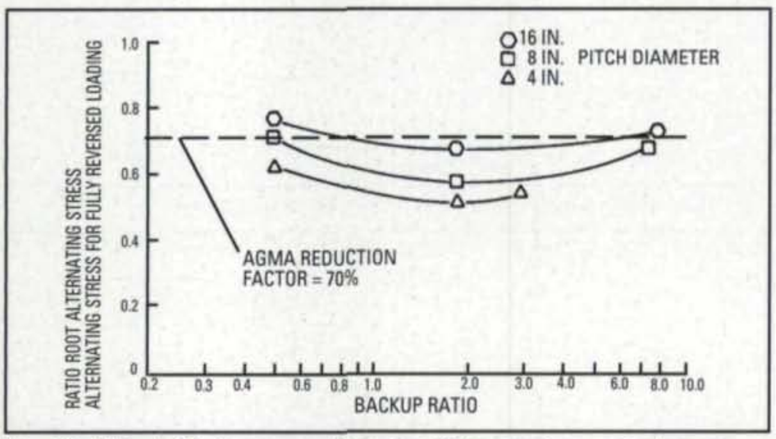

Fig. 13 - Effect of fully reversed bending (idler gear) on root stresses.

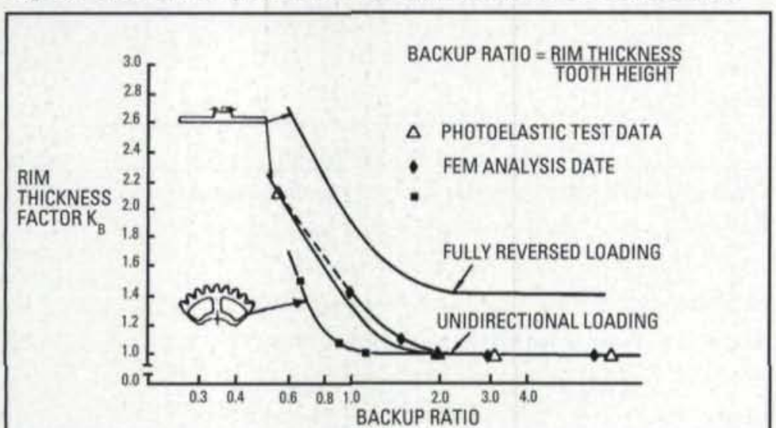

Fig. 14 - Proposed rim thickness factor.

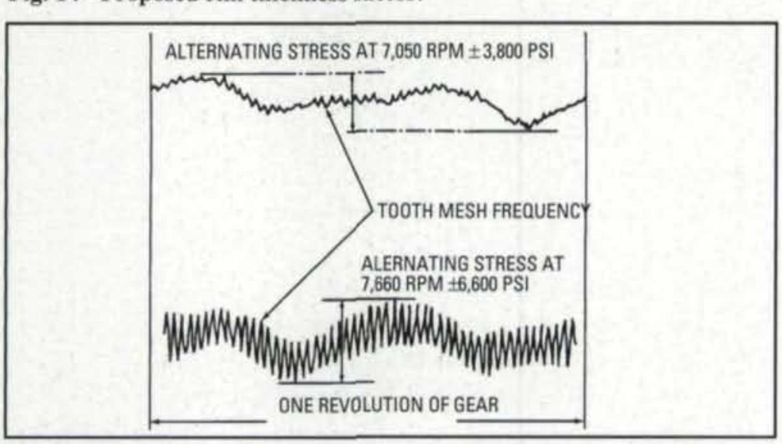

Fig. 15 - Increase in alternating stress resulting from speed change to excite resonant response.

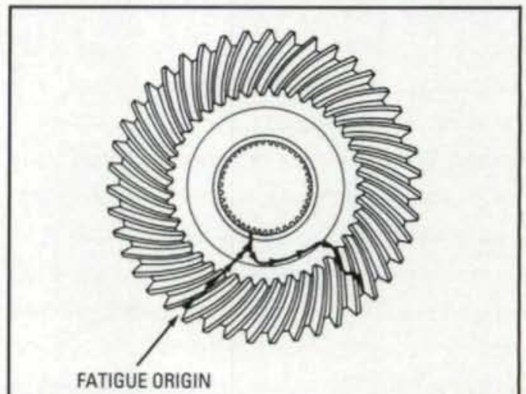

Fig. 16 - Typical resonance failuer.

failures associated with this phenomenon, it is possible that it will not be recognized as the causal agent, and a general beefing-up of all parts involved is undertaken as a fix for the problem. In many cases, this will solve the problem, since the resonant frequencies will also be altered; however, the resulting design is usually heavier and more costly than necessary. Had the resonant problem been recognized and dealt with, the final configuration would have been more cost- and weight-effective.

Mode Shapes - When a body is excited at its natural frequency, the body will assume an oscillating deformed shape called a mode shape. For a body of revolution these mode shapes can be generally classified by the number of nodal diameters (diameters exhibiting zero displacement) occurring during the vibration deformation cycle. The number of nodal diameters contained in a mode shape is referred to as the harmonic index of the mode. Classification of modes by their characteristic harmonic index is helpful, not only as an aid in visualizing the mode shape, but also in the determination of excitation frequencies. In theory, a body of revolution is capable of an infinite number of mode shapes for each value of harmonic index. These mode shapes are all unique and occur at unique characteristic natural frequencies. The mode shapes consist of various combinations of circumferential or ring nodes, i.e., circumferential rings of zero displacement. Fig. 17A is a sketch of the deformed mode shape of a thin-walled cylinder (solid line) superimposed on the undeformed shape of the cylinder. This particular mode has a harmonic index of 2, as evidenced by the number of nodal diameters. Note the circumferential ring node occurring at section B-B. Fig. 17B shows the cylinder again responding in a harmonic index = 2 mode, but at a higher frequency and with ring nodes at sections E-E and G-G.

Speed/Frequency (Campbell) Diagrams - It is important to note that the mode shapes shown in Fig. 17 represent the displacements at an instant in time; at 1/2 cycle or 1/2 ω_n seconds later, the displacements of the structure will be the negative (with respect to the undeformed shape) of the displacements shown. With this in mind, it can be seen that for a stationary (nonrotating) circular disk or cylinder, a rotating, steady (non-oscillating) force will induce constructive reinforcement of the displacement waves at a rotation speed of

$$N = \pm \omega_n/D \qquad (1)$$

By a simple change of reference system it can be seen that this is also the case for a rotating circular structure subjected to a force stationary in space. This relationship is valid for values of harmonic index D = 1, 2, 3... For the case of D = 0, no nodal diameters (purely axisymmetrical mode shape), this criterion does not apply, since a nonoscillating rotating force will not excite this type of displacement mode shape. The excitation forces for gear tooth meshing forces are not steady, but oscillate with respect to a reference system fixed to the rotating gear. The frequency of oscillation of the mesh forces and their harmonics is

$$f = knN \qquad (2)$$

for k = 1, 2, 3... Hence, the criterion to produce constructive reinforcement of a displacement mode shape by a rotating, oscillatory, exciting force is

$$\omega_n = knN \pm DN = (kn \pm D)N \qquad (3)$$

A convenient method of displaying the rela-

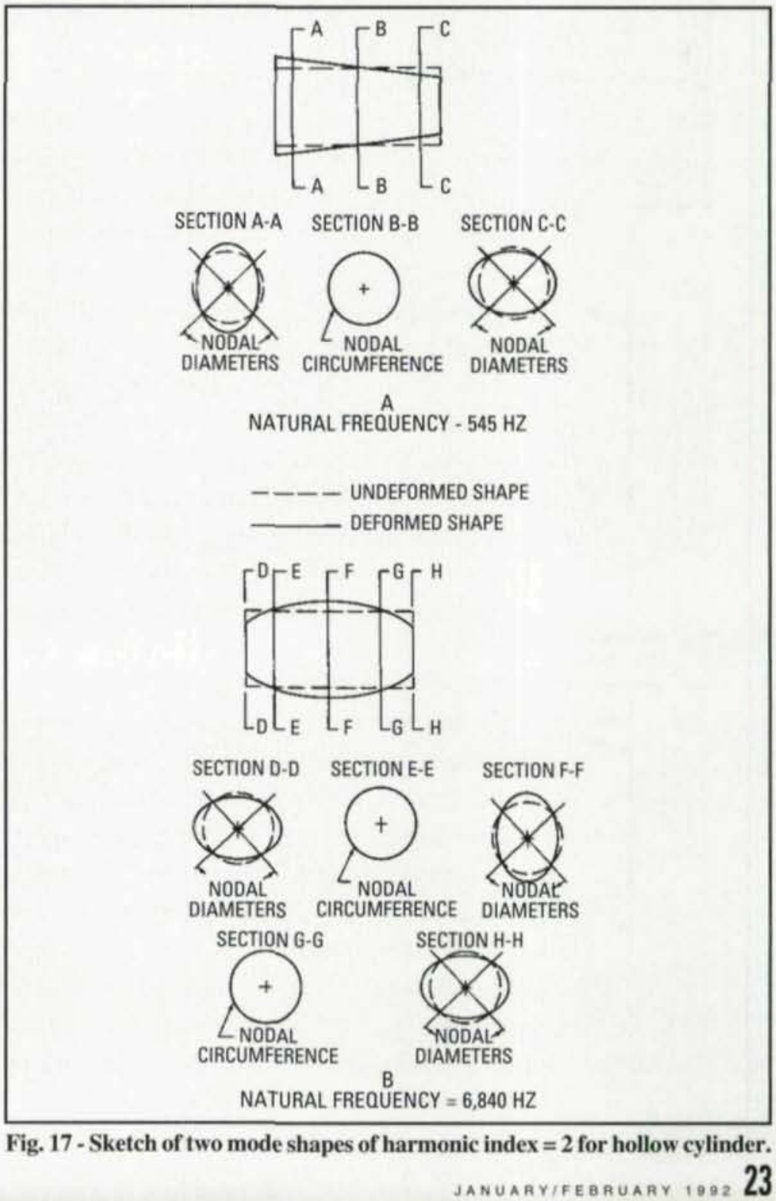

Fig. 17 - Sketch of two mode shapes of harmonic index = 2 for hollow cylinder.

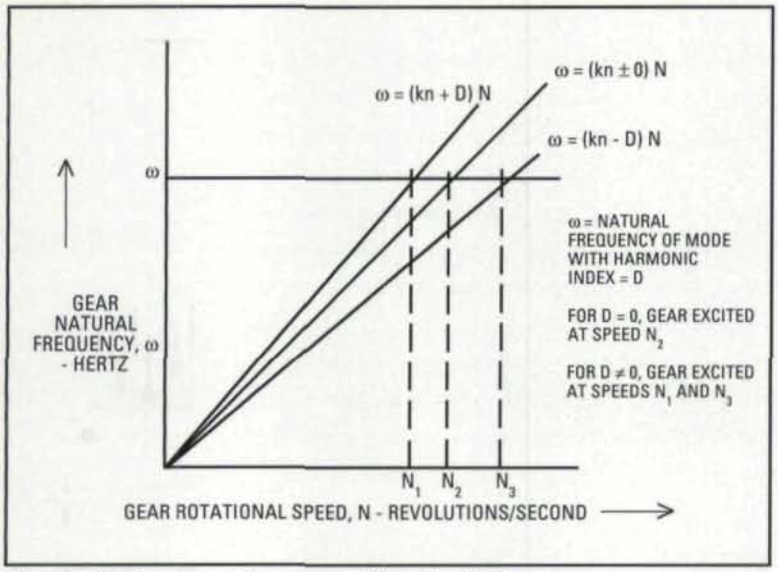

Fig. 18- Typical speed/frequency (Campbell) diagram.

tionship between gear natural frequencies and transmission system excitations is with a frequency-versus-speed diagram. Excited frequencies are plotted on the vertical axis and rotational speed is plotted on the horizontal axis as shown in Fig. 18. Such charts are frequently referred to as Campbell diagrams.

Natural frequencies from bench test or analysis are plotted as horizontal lines. The lines ω_n = (kn ± D)N are then plotted for the values of D corresponding to the mode shapes of the natural frequencies under consideration. The points where the horizontal ω_n lines intersect the ω_n = (kn ± D)N lines indicate the speeds at which the gear resonance will be excited. Should these speeds occur within the operating speed range, a potential gear vibration problem will exist. Resonant conditions can also be excited by harmonics of the tooth mesh frequency; therefore, n = (kn ± D)N should be plotted for several of the lowest values of k. In addition, other potential sources of excitation must be considered. One which is of particular importance in the design of epicyclic gear systems is the planet passage frequency. Note that for modes with a harmonic index of 1 or greater, resonance conditions are excited at two speeds for each tooth mesh harmonic. The case of a harmonic index = 0 mode is excited only at the basic tooth mesh frequency and its harmonics. In all cases the frequencies at which the gear vibrates are its natural frequencies with respect to a reference frame fixed to the rotating gear. It should be noted that the apparent stiffness of the rotating gear can be slightly altered by centrifugal effects, thereby causing n to be a function of rotational speed. If this function can be determined either by test or analysis, the foregoing is still applicable. In this case the ω_n horizontal lines would become ω_n = f(N) curves, and the intersection points would be determined as described previously. For gear sizes and speeds typically encountered in vehicle power gearing, these centrifugal effects have been found to be negligible.

Evaluation of Resonance Behavior - The problem of evaluating the resonance behavior of a specific gear may be approached in many ways, however, these methods can be broadly classed as either experimental or analytical. Experimental techniques provide very accurate data regarding response frequencies and mode shapes; however, actual hardware (or at least prototype gears) is required. This limits the correction of any problem areas identified to either redesign or add-ons. The use of analytical techniques, on the other hand, allows these problems to be addressed and corrected in the design stage. The basic data sought by either method are the response (resonant) frequencies, the mode shapes, and the level of response; i.e., the stresses induced by operating at a resonant frequency. Both methods can provide the first two pieces of the puzzle, frequency and mode shape; however, the level of response can usually be obtained only thorough the use of sophisticated strain-gage instrumentation during an actual operational transmission run. Some progress is being made in the analytical definition of response level. Although it would seem that analytical techniques are preferable in all cases, advantages and disadvantages to both methods will become clear as we proceed thorough the next two sections. One of the biggest unknown factors is the amount of damping that occurs because of bolted joints, supports, and the like.

Experimental Methods - There are many methods for experimentally defining the resonant frequencies and mode shapes of gears. The specific method which should be used in any particular case is a function of component size, mass, and the dollars available for test.

In order to evaluate the response of a part, an excitation source and some means of detecting the response and defining the mode shape must be provided.

The testing is generally conducted as follows: The gear is mounted in such a way that it is isolated from its surroundings (elastomeric pads, cords, etc.) and excited with a variable-frequency source. A sweep through the frequency range of interest is performed while the gear response is monitored. Frequencies which produce substantial responses are noted. After the frequency sweep is completed, the gear is again excited at each of the noted frequencies while the gear is searched or probed to define the mode shape.

The determination of frequencies to be searched can be enhanced by performing a simple bang test if frequencies have been calculated previously by some analytical method, or should a frequency spectrum analyzer be available. Fig. 19 shows a typical spectrum-analyzer oscillograph from gear bang testing. Note the number of frequencies which display high relative response; these frequencies represent the excitation frequencies near which gear resonant problems may occur. Note also the large number of resonant frequencies. A bang test consists simply of hitting the gear once and evaluating the resulting response with the aid of a spectrum analyzer. A gear thus excited will ring down at all its natural frequencies; with a spectrum analyzer, these frequencies can be separated and identified.

This experimental technique actually determines the free-vibration characteristics of the gear; however, the natural frequency of free vibration has been an excellent indicator of the installed and operating resonant frequencies of the gear and shaft. Fig. 20 is a Campbell diagram showing the free-vibration resonant frequencies of an integral spiral bevel gear and shaft, as determined from air siren testing versus shaft speed. Also plotted on this diagram are the frequencies at which increased response was noted for the component installed and operating under design loadings. These operating data were obtained from telemetered strain-gage readings recorded during a transmission dynamic strain survey. Note that the forced-response frequencies of the operating gear and shaft agree closely with the free-vibration natural frequencies determined in siren tests.

The four main variables to be addressed in an experimental investigation of the resonant response of a gear (or any other part) are the method of excitation, the method of detecting when a resonant condition has been encountered, the method of identifying the mode shape of a particular response, and the stress level associated with a particular response. With the

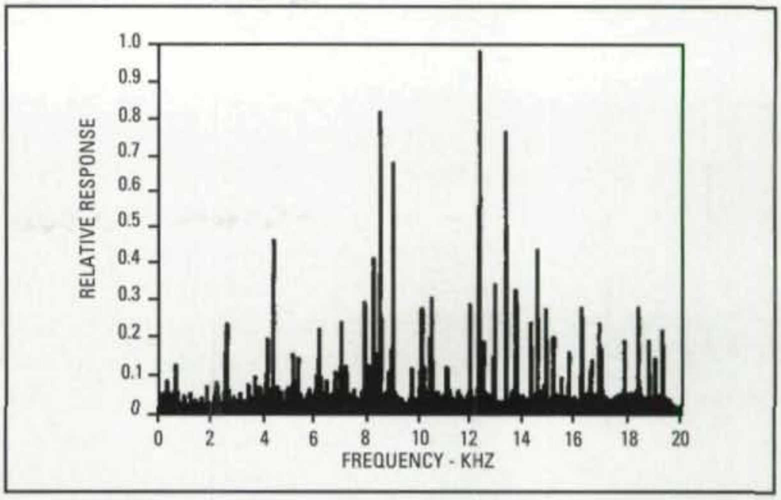

Fig. 19 - Typical plot of relative response amplitude versus frequency from gear bang test.

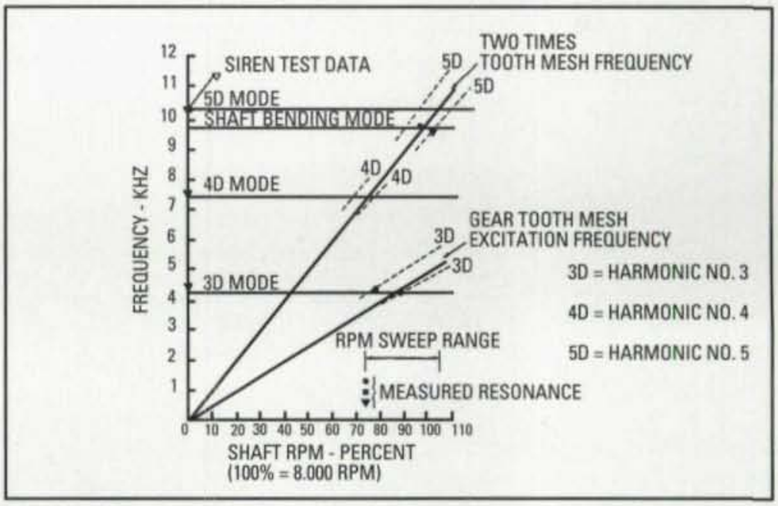

Fig. 20 - Campbell diagram showing siren test and operating resonant frequency response.

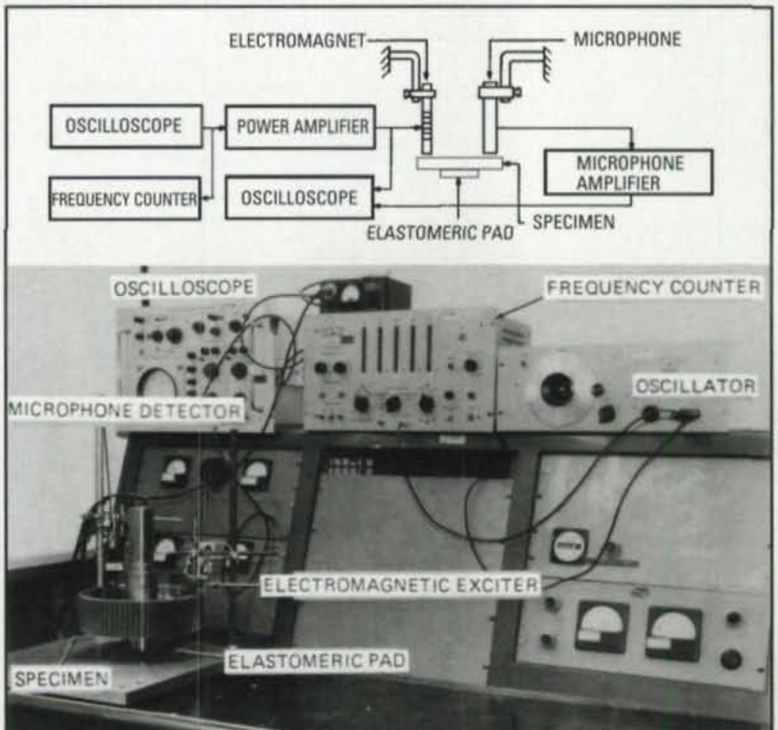

Fig. 21 - Electromagnetic excitation with detection by microphone and oscilloscope.

exception of the stress level, each parameter may be addressed separately since, in theory at least, any excitation method can be used with any combination of detection and identification techniques. However, considerations of experimental setup and procedure will limit the combinations which are practical. Complete descriptions of the various techniques can be found in Ref. 6. An excitation source, such as the electromagnetic setup shown in Fig. 21 or the air siren shown in Fig. 22, is the first piece of equipment required. A means of detecting the existence of a resonant frequency, such as the microphone shown in Fig. 21 or the accelerometer shown in Fig. 22, is also required. Finally, the mode shape itself must be defined through the use of sand or glitter patterns (Fig. 23), holography, or by hand-probing with a microaccelerometer (Fig. 24).

If a strong excitation source is used, hand-probing (Fig. 24) with a microaccelerometer can identify nodes and antinodes and, thus, mode shapes. This technique does require some operator skill and practice on simple shapes. It can be very useful in the evaluation of complex modes on complex gears.

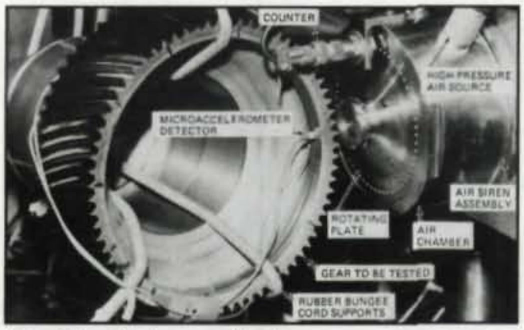

Fig. 22 - Air siren excitation source.

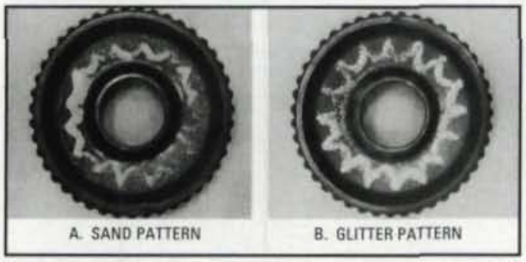

Fig. 23 - Mode shape detection on a flat spiral bevel gear (17,939 Hertz).

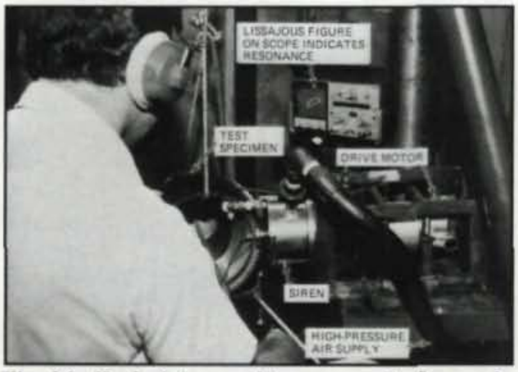

Fig. 24 - Technician probing manually for mode shapes.

Analytical Methods - The high costs associated with the redesign of complex gears to improve their dynamic characteristics after initial prototype fabrication and testing make necessary accurate analytical techniques with which to predict resonance. This would allow the design engineer to predict the dynamic characteristics of a particular gear design while it is still on the drawing board, thus enabling any changes to be incorporated before fabrication and test.

Because of geometrical complexity and the generality of configuration, the finite-element method (FEM) was chosen as the most suitable analytical technique. Recent experience using several general-purpose FEM computer programs and modeling techniques indicates that excellent correlation with experimental data from prototype resonant testing is possible. The most efficient of these techniques has been an axisymmetrical modeling approach. This method assumes the gear to be symmetrical about its rotational centerline and, hence, requires that only a single cross section be modeled.

The model definition required for sufficient-ly accurate results (that is, within about 5-10% overall) is quite coarse relative to that required for tooth or blank stress analysis. The teeth themselves, for example, are not modeled as separate entities; rather, only their mass is simulated. This is done by increasing the root diameter to the midtooth diameter and then modeling the gear teeth as a solid ring of this thickness; in most cases this modeling system yields excellent results. For many lower modes (which are the most important anyway) the accuracy is far better than that at higher modes; in most cases the error is less than 5%.

Some of the difference between bench and dynamic rig test results may be because of centrifugal stiffening of the blank at normal operating speed. However, when we consider the good overall correlations obtained at low to moderate speeds, this effect can be considered negligible. This may not be the case for very high-speed, large-diameter gears.

Most programs include the ability to produce a plot of each mode shape, which can be quite helpful in determining which areas of the gear require modification to obtain the desired dynamic characteristics. One such plot is shown in

Fig. 25. Each mode shape has an associated harmonic index. This index represents the integer number of nodal diameters characteristic of the nodal displacement shape of the gear. Fig. 25 also shows an axial view of the deformation of the integral spiral bevel gear shaft, indicating that the harmonic index of this mode is 3. The analytically predicted and measured resonant frequencies are also compared. The correlation in this particular case is exceptionally good; in general, the predicted values for the lower frequencies do not differ more than 5% from the measured siren test frequencies.

Modifying Resonance Behavior- Once the resonant frequencies have been determined, it must be ascertained whether the frequencies lie in the operating excitation range. If resonances do occur in the operating frequency range, the design engineer is faced with a decision: He may either modify the design (change mass and/or stiffness of the gear), or he may provide some form of damping to decrease the response of the gear near the resonant frequency.

Both approaches have inherent advantages and disadvantages. A careful evaluation of the alternatives is required in order to decide which method is applicable to any given situation.

Detuning - Modifying the geometry of the gear in an effort to change its mass and/or stiffness characteristics is frequently referred to as detuning. The ideal approach is to design all gears so that their natural frequencies are far from any potential excitation sources, especially gear mesh and its multiples.

Although this approach is not practical in all cases, it can be eminently successful, particularly if the blank is relatively simple and the Campbell diagram is uncluttered; such a case is shown in Fig. 26. The gear mesh frequency at overspeed (3,682 Hertz) is quite close to a natural frequency (3,764 Hertz). Adding material as shown increased this natural frequency beyond the overspeed excitation frequency. In this instance the next lower frequency did not increase enough to bring it into proximity with either the idle or 100%-speed excitation sources. In cases such as this, plots of the mode shape are an invaluable aid in determining where and how the gear must be modified.

Since each problem is unique, no hard-and-fast guidelines can be given on the methods to be used in accomplishing such modification, but it is generally best to make changes which increase stiffness and, thus, raise the natural frequencies, rather than to add mass alone, such as adding lumps to the gear bore. This method has been shown to be the most weight-effective for lightweight gearing.

This discussion is aimed at small perturbations of a basically fixed design. If an analysis is performed early enough in the design cycle, it may be possible to make a major change which will effectively improve the situation.

Damping - If detuning is not practical or not successful, damping is a viable alternative. Damping causes the amplitude of and, hence, stress response to the excitation forces to be reduced in the installed and operating forced - response environment. If the proper amount and type of damping are applied at the appropriate places on the gear blank, this approach is almost always successful. There are many ways in which damping can be introduced into the blank. In addition, the construction of the gear itself may contribute to its damping characteristics. If it is manufactured as a separate part and is bolted, keyed, or splined to its shaft, the joint will provide a small amount of damping through the rubbing action (Coulomb damping) which is inevitable at such joints. Unfortunately, this motion, which is beneficial in a damping sense, can be very detrimental over the long term, since it may give rise to fretting. Fretted joints can be the source of cracks in either gear or shaft which lead ultimately to structural failure. An integral shaft and gear design eliminates both the damping and

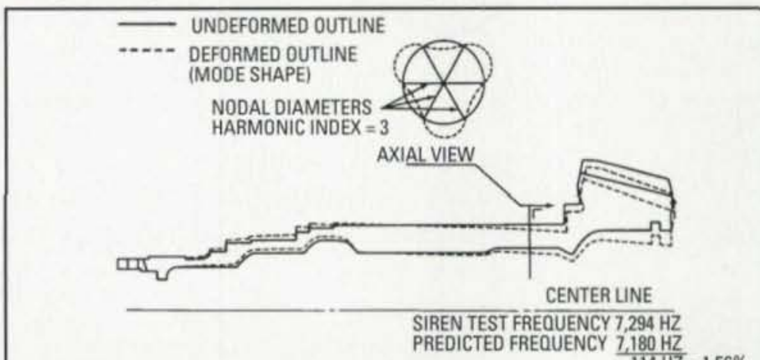

Fig. 25 - Predicted gear resonant frequencies.

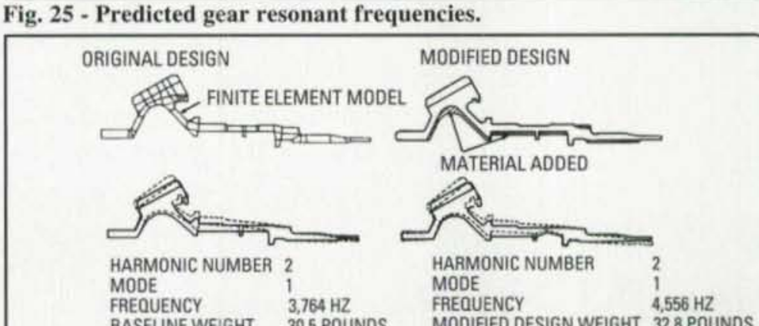

Fig. 26 - Comparison of original and modified spiral bevel gear designs; modification improved resonant frequency characteristics.

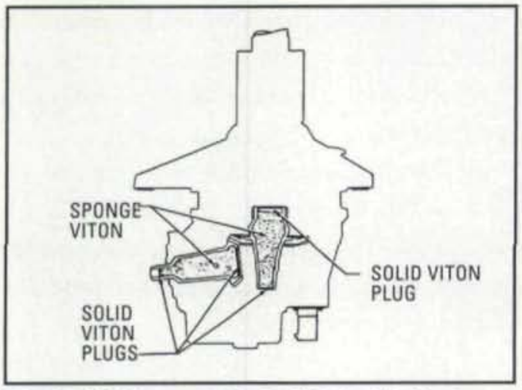

Fig. 27 - Hollow gear shafts filled with elastomer and plugged.

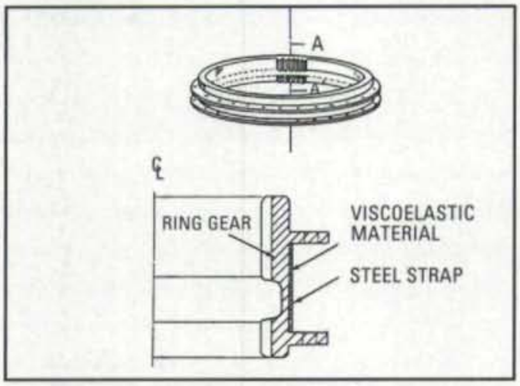

Fig. 28 - Thin elastomeric coating on gear web *with steel constraint.*

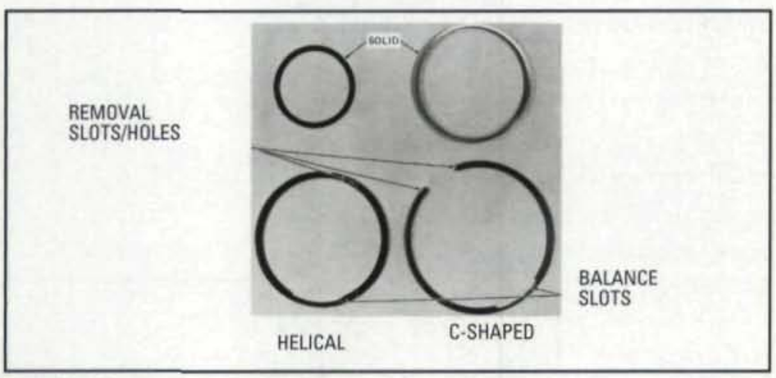

Fig. 29 - Types of damping rings.

fretting conditions and is preferable, since fretting can be a major problem, and there are other less severe and more effective methods of applying damping. Two of the simplest methods are coatings and rings.

Coatings - *Coatings can be applied to nonfunc*tional surfaces of the gear blank to reduce the vibration level. Both hard and soft coatings have been used with varying degrees of success. Soft coatings, such as bonded layers of an elastomer, have been used in a number of applications. Because they are exposed to the hot oil environment, they generally must be either trapped as shown in Fig. 27 or constrained as shown in Fig. 28. The effectiveness of either method depends upon maintaining the bond between the elastomer and the gear blank. Other methods of coating, such as plasma-spray coating or electrochemically plating relatively heavy metals like molybdenum or copper compounds on the appropriate surfaces, are more permanent and less likely to separate from the gear.

Any of these methods will work, particularly if the response is not overly energetic, and if the gear *design incorporates large, flat web areas; however,* our testing has shown that while the noise level is reduced in many cases, the damping provided is not sufficient to reduce the vibratory stresses to acceptable levels.[8]

Rings - Damping rings are a much more effective method of providing damping. Solid, C-shaped, and helical rings (Fig. 29) have been used successfully. Solid rings are inserted into an internal groove machined under the gear rim by heating the gear and cooling the ring. Helical rings, which are actually modified common retaining rings, are turned into a similar groove in the gear rim. Both dissipate energy through a mechanism which is believed to be largely Coulomb friction at the sides and, especially, the bottom of the groove. The helical rings provide considerably more *damping (about 2.5 times more, according to* Ref. 8) than solid rings, and both provide an order-of-magnitude more damping than most coatings, though they are generally not as effective in reducing noise as some coatings appear to be. The solid rings have less retention surface (limited by the amount of gear expansion and ring contraction at installation); thus, they are more likely to fall out of their grooves and cease their damping function because of wear than are helical rings, which typically have relatively deep retention grooves. In addition, the force exerted on the groove by a solid ring does not increase appreciably with rpm because of centrifugal force (CF). Since helical rings are not closed loops, they tend to unwind with increasing CF and exert greater force and, thus, greater damping on the ring groove. *In a similar manner, the damping effectiveness of* helical or C-shaped rings is not affected by wear, since CF will keep opening the ring so that contact is maintained with the groove. After solid rings wear, CF only contributes to increased ring stress and does not maintain contact pressure with the groove; thus, damping effectiveness may decrease substantially. In order to yield maximum effectiveness, the proper amount of ring weight must be provided. The results of extensive dynamic strain survey testing have been analyzed in an effort to determine the ring weight required in the general case. Based on this information, the following empirical relation between ring weight and gear weight has been developed:[9]

$$W_D = 0.06W_G 0.75Q \qquad (4)$$

Equation 4 has been shown to work well for lightweight aircraft-type gears in the size range of 6" to 33" pitch diameter and at pitch line velocities up to approximately 28,000 fpm.

The importance of the proper choice of ring weight and placement can be seen in Fig. 30. The final stress level has been reduced by 50%, even though the transmitted torque increased by more than 80%.

One of the most significant problems associated with damping rings is the wear that occurs both on the ring and its groove. Although the groove wear alone is almost never severe enough to cause the gear to fail, slivers which can cause indications on chip detectors, indicating screens, and other debris monitors can be generated. In addition, the ring wears excessively when its cross section is reduced, and the possibility of its failure arises. These problems can be reduced by hardening the groove and using multiple-wrap rings. The wear appears to be caused more by the ring being forced to rotate in its groove by the gear-load-induced deflections under the mesh point than by its vibration in the groove because of its response to gear natural frequencies. It seems likely that the wear problem could be reduced by decreasing these deflections.

The addition of damping rings or coatings does not change the natural frequencies of the gear to any measurable extent; this has been verified by electromagnetic and air siren testing as well as full-scale transmission testing.

Changing Excitation Frequency - Our discussion thus far has been limited to modifications of the response of a gear to a given excitation source. It is also possible in some cases to change the excitation frequency. This can be done in several ways. The easiest but usually the least practical method would be simply to alter the system speed so that it does not coincide with any of the natural gear frequencies. Relatively large changes in excitation frequency can also be made by changing the diametral pitch (module) of the gear set to increase or decrease the number of teeth, thus changing the mesh frequency without changing shaft speed. Any change in pitch should be carefully evaluated in order to insure that the bending-fatigue capacity of the gear set is not compromised. Several design techniques have been proposed to employ gearing

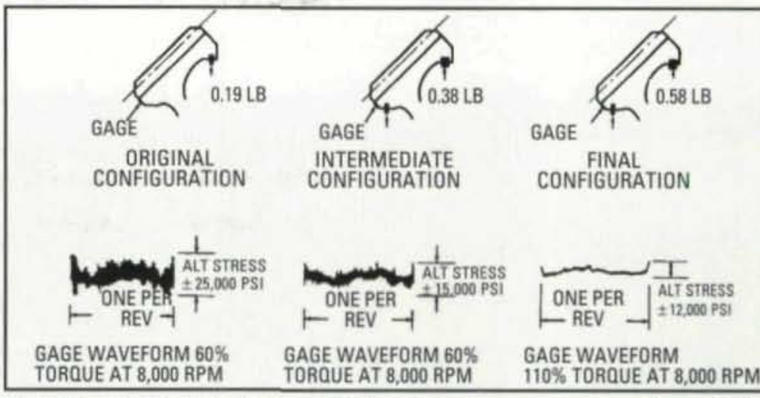

Fig. 30 - Reduction of transmission bevel gear response through increased damping.

with either more or fewer teeth than would ordinarily be used on a given sized gear.

Other alternatives, which would require redesign, are to alter the number of planets in an epicyclic system or to change the relative ratios used in each stage of a multipass system.

Conclusions

Three areas of significant concern to the designers of spiral bevel gearing have been identified and discussed. Each area is outside the normal design procedures for such gears, but can be very important in the overall success of specific designs.

General design guidelines have been presented and references for further and more detailed studies have been provided. ■

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Optimum Shot Peening Specification - II

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Summary: Following is the second part of an article begun in our last issue. The first part covered basic shot peening theory, shot peening controls, and considerations that should go into developing a shot peening specification. Part II covers optional peening methods and the relationship of shot peening specifications to the drawings.

Optional Peening Methods and Additional Considerations

Some additional peening methods and considerations which the reader may want to include as part of any general specification are shown below.

Strain Peening or Stress Peening - This technique is applied when parts are stressed in one direction only and longer fatigue life is desired than that obtained by conventional methods. The part is shot peened in a stressed or loaded condition, and compressive stresses produced by the peening can be as high as the compressive yield stress of the material itself. This technique has been used heavily in numerous industries.

Dual Intensity Peening - This technique also can be used when substantially longer fatigue life is required. Research done on carburized steel indicates that dual peening, which is high intensity shot peening, followed by lower intensity shot peening with smaller shot, increases the magnitude of surface compressive residual stress.[1] Additional testing on other materials have confirmed this data.[2,3] (See Figs. 1 and 2.)

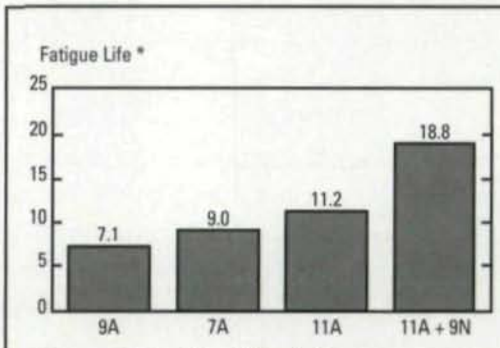

Fig. 2 - Dura Peening Titanium 6AI-4V Blades.[3]

Plating and Salvage Methods - If machining discrepancies in production are sometimes salvaged by building up an area for remachining by the use of chrome or other plating techniques, or if plating is used for a wear or protective surface, shot peening prior to plating can be used. The shot peening will prevent crack propagation from microcracks in the plating to the parent metal if the part is subjected to a cyclical

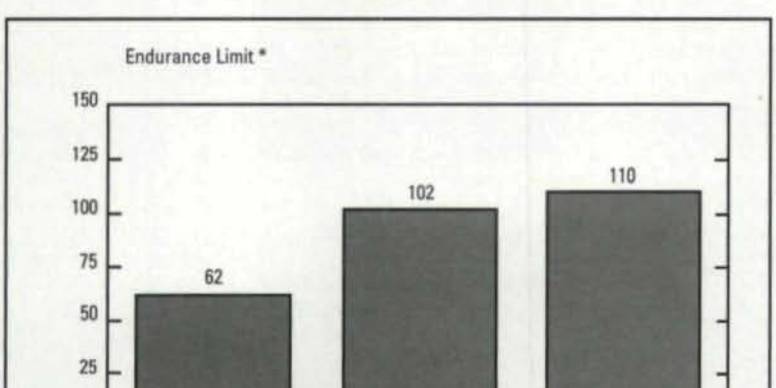

Fig. 1 - Dura Peening Rene 95 specimens.[2]

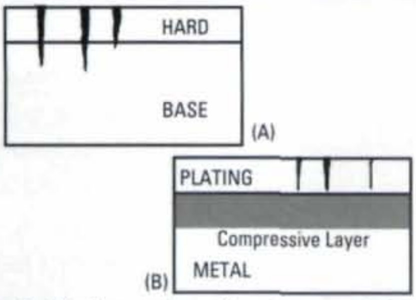

Fig. 3 - Plating cracks will not propagate into pre-stressed base.

load. Cracks will not propagate into layers of compressed stress. (See before and after shot peening illustrations in Fig. 3.) In addition, significant increases in fatigue strength closely approximating original unpeened surfaces are shown in Fig. 4. In some cases, shot peening prior to plating may be required by other contractual agreements. Specifications, such as Federal Specifications QQ-C-320 and MIL-C-26074A, require shot peening on steel parts that are chrome or electroless nickel plated.

An added side effect is the prevention of hydrogen embrittlement by shot peening of this parent metal prior to the plating operation. Since atomic hydrogen is extremely mobile and able to penetrate and interact with metal easily, the metal's ductility and ability to withstand cyclic loads is reduced. Peening has been proven effective in retarding the migration of hydrogen through metal.[5] (See Fig. 5.)

Contour Correction - Just as it is possible to create a desired curvature and shape to components by shot peening, it is also possible to correct the shape and form of parts. The shot peening process avoids the unfavorable (tensile) residual stresses produced by other straightening methods and instead produces favorable (compressive) residual stress. (See Fig. 6.)

Increasing Wear Due to Work Hardening - In the discussion on material considerations, considerable space was given to whether a material

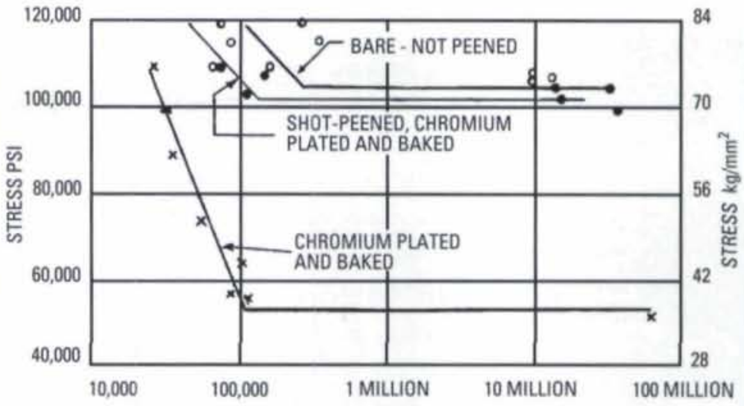

Fig. 4 - Cycles to failure 4340 steel, 52-53 Rc rotating beam fatigue.[4]

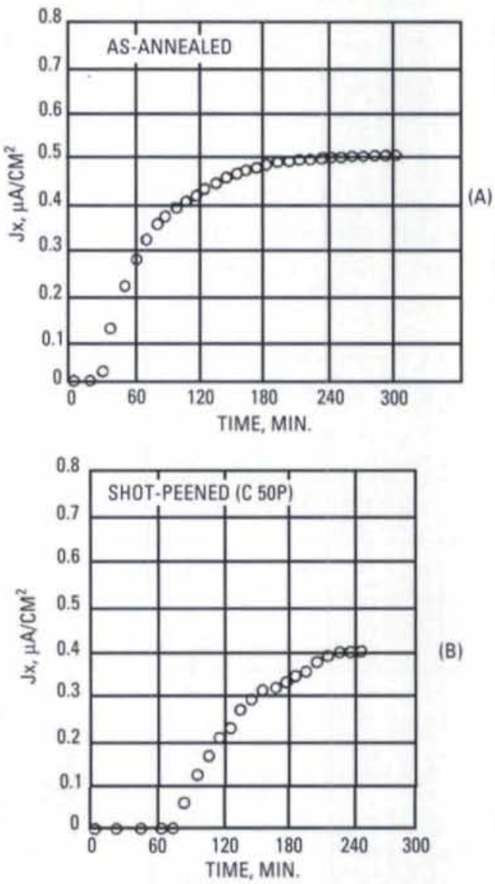

Fig. 5 - Effect of shot peening on hydrogen migration. J[8], A/CM[2] hydrogen permeation rate. A) Non-treated surface. B) Shot peened surface.[5]

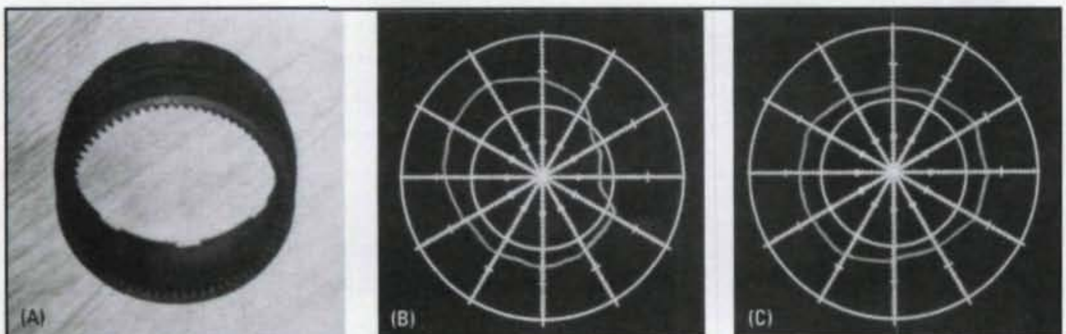

Fig. 6 - Contour correction. A) Ring gear. B) Before peenforming. C) After peenforming.

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would readily work harden. If this factor is a major consideration, then it should be addressed in the specification. (See Fig. 7) For materials which cannot be heat treated, but require wear resistance, shot peening should be considered.

Porosity - When this is a concern, porosity should be reviewed as a specification option. Typically this is not utilized for the compressed stress benefits, but rather is used to compact the surface[7] or reveal some sub-surface porosity prior to machining. It can, therefore, be utilized as an inspection tool before machining of questionable castings.

Salvage; Grinding; Before and After Shot Peening - In cases where a severe grinding operation has developed resultant residual tensile stresses and surface brittleness, shot peening of the surface after grinding should be considered. Fig. 8 reveals S-N curves for a part originally designed for an endurance limit with a gentle grind, the resultant lowered endurance limit after grinding, and improved endurance limit of the severely ground surface followed by shot peening.

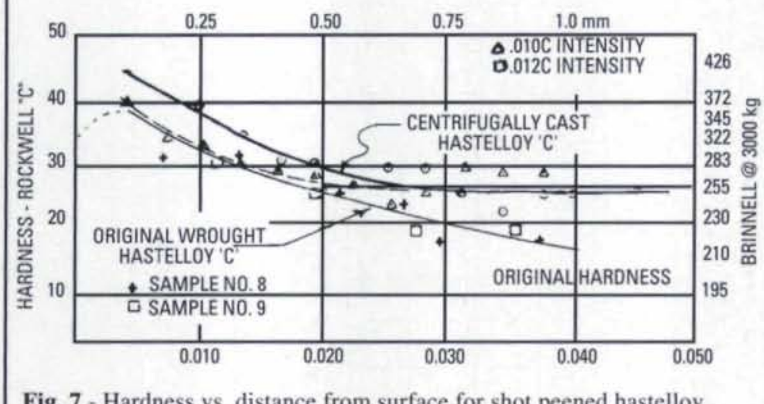

Fig. 7 - Hardness vs. distance from surface for shot peened hastelloy "C" samples.[6]

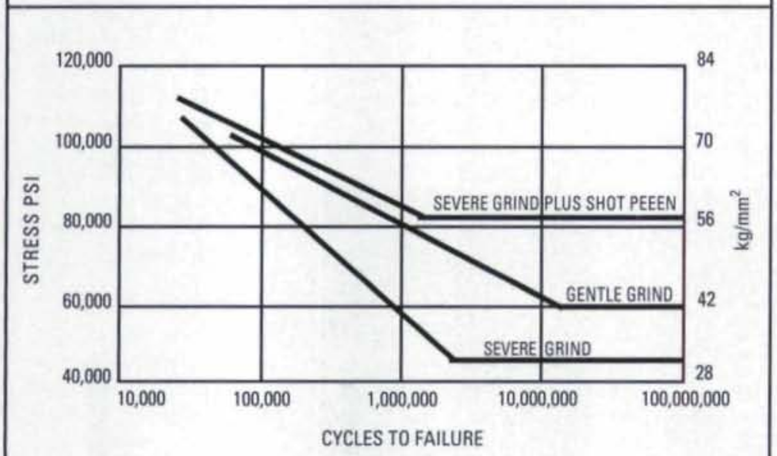

Fig. 8 - Shot peening improves endurance limits of ground parts. Reversed bending fatigue of flat bars of Rockwell hardness C45.[8]

Another technique that can be used, especially on particularly difficult grinding operations or materials, is to shot peen prior to grinding to prevent grinding cracks. Since grinding of carburized gears can produce high tensile stresses, these stresses can initiate cracks in the tooth surface. Shot peening prior to grinding will greatly reduce this tendency. Peening here is used to prevent crack propagation from the grinding and not to increase bending fatigue strength.

Stress Corrosion Cracking - In particularly hostile environments where a material being used may be affected adversely by general corrosion coupled with residual or applied tensile stresses, shot peening may be a consideration. The peening will change the surface residual tensile stresses to compressive stresses, which will eliminate the conditions needed to promote stress corrosion cracking.

The Part Drawing

Once a satisfactory in-house specification has been established which addresses the particular needs of a company, it is still necessary to translate this information to particular gears. The general specification should assist the design professional regarding the necessary steps to properly select an optimum drawing specification. The information must then be transferred to the manufacturing drawing.

In specifying shot peening requirements on part drawings, the following parameters should be identified:

1. Areas to be shot peened
2. Areas to be masked
3. Optional areas
4. Areas where shot peening fades out
5. Shot size, hardness, and material
6. Locations for intensity verification and intensity range at each location
7. Coverage requirements for all areas to be peened, including the method used for coverage determination
8. Applicable shot peening specification.

Fig. 9 provides a theoretical example of a gear with a suggested specification. Utilizing the above points, the analysis of this specification is as follows:

1. Areas To Be Shot Peened - These are noted by DIM "A", and further critical areas are identified by "XXX." Five primary areas require the proper intensity. These are at the tooth root fillet, the gear pitch line, two shaft fillet transi-

tion areas, and the main shaft body. Since only one peening operation is to be performed, the shot selection would indicate that the apparent geometric limiting factor of the shot is the fillet radii of the gear teeth. Most likely the main shaft is being peened because the shaft may also experience problems with fatigue. It is possible that some machining may occur on the shaft body after shot peening, so rather than mask this area, peening is being allowed. The gear pitch line area is noted because pitting of the gear tooth may occur.

2. Areas To Be Masked - These are noted by DIM "B" and DIM "C". Most likely the O.D. of the gear has limitations on the potential of burring at the top land. This is costly and should be avoided unless alternate ways are not available. A potential alternative solution may be to break or radius all sharp edges in the areas prior to peening. This can minimize or eliminate the potential to burr. The threads at the shaft end do not require peening and must be masked as optional peening could damage these.

3. Optional Areas - Noted by DIM "D," these are the holes in the gear body.

4. Areas Where Shot Peening Fades Out - Not applicable to this example.

5. Shot size, Hardness, and Material - MI 110 shot, intensity 6-10A; the MI 110 designation defines a cast steel shot.

6. Location for Intensity Verification and Intensity Range at Each Location - Only one intensity is specified and is marked by "XXX." If other intensities or shot sizes are used, additional callouts and symbols are required.

7. Coverage Requirements For All Areas To Be Peened, Including the Method Used for Coverage Determination - 125% coverage verified by Peenscan.®

8. Applicable Shot Peening Specification - MIL-S-13165B.

This drawing specification clearly denotes a proper shot peening requirement and should easily be accomplished by the manufacturing group or vendor. This specification should readily coincide with the company in-house peening specification. Note, however, that this specification may not, and most likely will not, work on parts similar to this part. It is strongly encouraged that each gear requiring shot peening be first evaluated based on the general in-house specification prior to definite

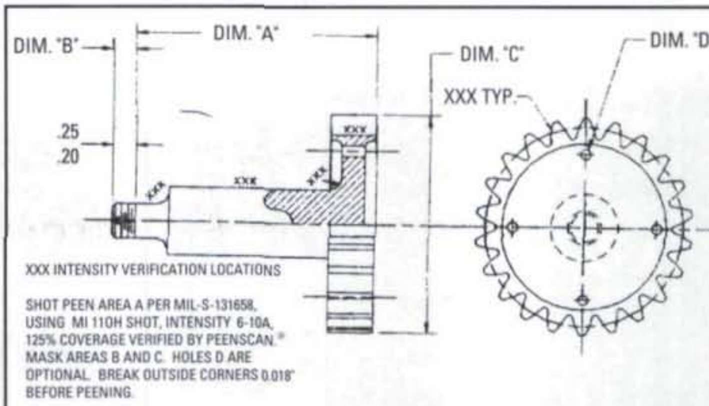

Fig. 9 - Theoretical example of a gear with a suggested specification.

shot peening callouts being made on a manufacturing drawing.

Summary

Confusion and some misunderstanding in properly specifying shot peening can cause difficulties in the manufacturing process. Concise in-house specifications covering considerations for shot peening, coupled with accurate manufacturing drawing callouts, can optimize the use of this effective tool. With the in-house specification addressing the particular needs of the manufacturing company's gearing requirements, and the correct specification on the manufacturing drawing conveying this to the vendor, shot peening can be utilized to its fullest advantage.■

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The Right and Wrong of Modern Hob Sharpening

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Precision gears play a vital role in today's economy. Through their application, automobile transmissions are more compact and efficient, ships sail faster, and diesel locomotives haul more freight. Today great emphasis is being placed upon the reduction of noise in all gear applications and, to be quiet, gears must be accurate.

For this reason, a great deal of engineering attention has been devoted to the various factors involved in precision gear production. The need for accurate machines and cutting tools, together with careful, precise set-up procedures, is well understood.

Not enough notice, however, has been given to hob sharpening, a necessary operation wherever gears are produced by hobbing and fully as important as original hob accuracy in its effect on gears produced. Even the finest class AA precision ground hob will produce poor gears if improperly sharpened.

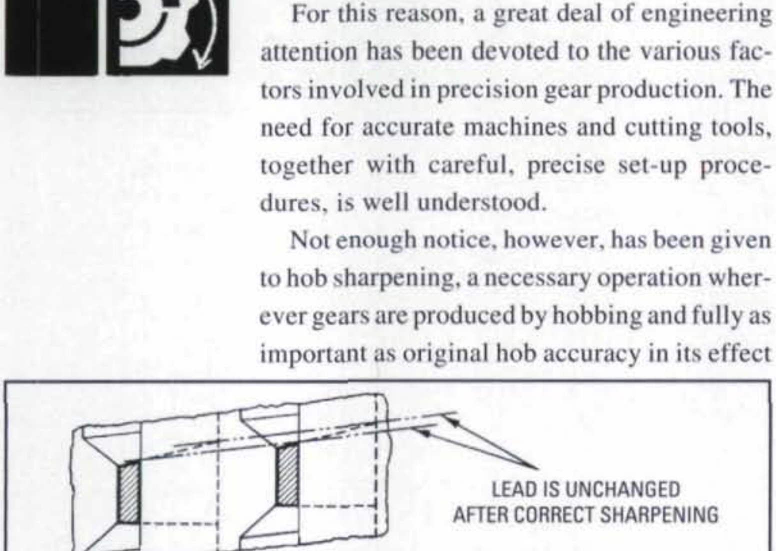

Fig. 1 - Correct tooth profile and lead are maintained with careful attention to sharpening methods and accuracy.

The purpose of this article is to discuss the various effects of hob sharpening on hob performance and proper sharpening methods.

Hobbing is a generating process in which involute gear teeth are formed by a sequence of cuts made by successive hob teeth in a continuously rotating gear blank. In order for the gear tooth form to be accurate it is essential that:

1. The cutting edges of the hob teeth have the correct form or pressure angle.

2. The cutting edges of the hob teeth lie along a helix of the correct lead.

Hob manufacturers hold these important elements to very close tolerances, and it is possible to sharpen a hob repeatedly to the very end of its useful life without impairing the accuracy of the gear produced. (See Fig. 1.) But sharpening must be done properly - otherwise both pressure angle and lead will be adversely affected.

Hobs are sharpened by grinding the face of all flutes until no trace of a worn surface is visible on the tops or sides of the hob teeth. Grinding should extend deep enough to blend in with the bottom of the flute. The flute elements affected by sharpening are:

1. Adjacent spacing
2. Non-adjacent spacing
3. Rake (A radial flute has zero rake.)
4. Lead (A straight flute is parallel to axis and has an infinite lead.)

All of these elements must be held within the tolerances listed in Table 1 if the original accu-

The figures in the table represent total indicator variation in ten thousandths of an inch.												
Diametral Pitch		1 thru 1.99	2 thru 2.99	3 thru 3.99	4 thru 4.99	5 thru 5.99	6 thru 8.99	9 thru 12.99	13 thru 19.99	20 thru 29.99	30 thru 50.99	51 and finer
Adjacent Flute Spacing	AA Precision Ground*	—	—	20	15	10	8	8	6	6	6	6
	A Precision Ground	40	30	25	20	15	10	10	10	10	10	10
	B Commercial Ground	50	45	40	30	20	15	15	10	10	10	—
	C Certified Unground	50	45	40	30	20	15	15	10	10	10	10
	D Commercial Unground	60	60	50	50	30	25	25	20	17	17	—
Non-Adjacent Flute Spacing	AA Precision Ground*	—	—	40	35	25	15	15	15	15	15	15
	A Precision Ground	80	60	50	40	30	30	30	25	25	20	20
	B Commercial Ground	100	90	80	60	50	50	50	40	35	30	—
	C Certified Unground	100	90	80	60	50	50	50	40	35	30	30
	D Commercial Unground	120	120	100	100	80	80	70	60	50	40	—
Rake to Cutting Depth	AA Precision Ground*	—	—	10	8	6	5	5	3	3	3	3
	A Precision Ground	30	15	10	8	6	5	5	3	3	3	3
	B Commercial Ground	50	25	15	10	8	7	7	5	5	5	—
	C Certified Unground	50	25	15	10	8	7	7	5	5	5	5
	D Commercial Unground	100	75	50	40	30	20	20	15	15	10	—

CUTTING FACE WIDTH						
		Up to 1"	1.001 to 2	2.001 to 4	4.001 to 7	7.001 & up
Flute lead over Cutting Face Width	AA Precision Ground*	8	10	15	20	20
	A Precision Ground	10	15	25	30	50
	B Commercial Ground	10	15	25	30	50
	C Certified Unground	10	15	25	30	50
	D Commercial Unground	15	23	38	45	75

* Single thread only

Table 1 - Sharpening Tolerances for Single and Multiple Thread Hobs

racy of the hob tooth form and lead is to be maintained. These tolerances, standardized by the Metal Cutting Tool Institute, are maintained by all hob manufacturers. When a hob is sharpened within these tolerances, the gear tooth accuracy will not be impaired.

Flute Spacing Error

Fig. 2 shows the effect of sharpening with unequal angular spacing of the cutting faces. Every lead variation caused by such faulty sharpening shows up in approximately a one-to-one ratio as an error in the gear tooth profile. The lead variation is roughly 1/16 of the flute spacing error. A spacing error of .016 would produce about .001 error in the lead and in the gear tooth profile.

The most common reasons for excessive flute spacing errors in sharpened hobs are:

1. Excessive runout of the hob during the sharpening process. This may be caused by loose-fitting or eccentric arbors, non-parallel collars, excessive tightening of the nut, or runout of the machine work spindle.

2. Use of worn index plates or pawls in

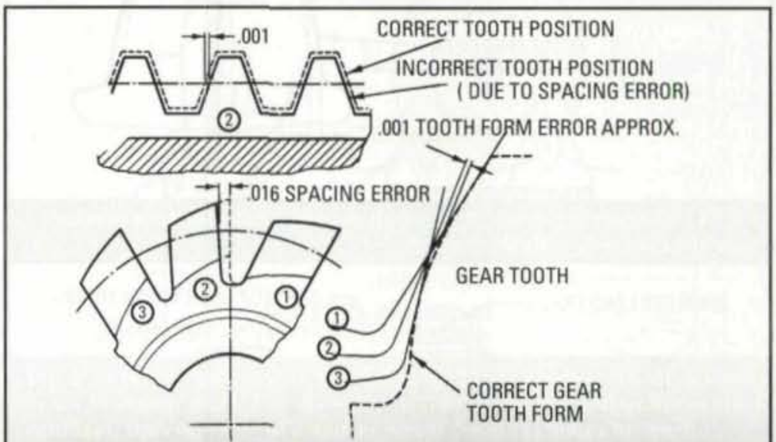

Fig. 2 - Serious hob lead errors result from sharpening with unequal spacing of cutting faces.

the sharpening machine.

3. Not "sparking out" the grinding wheel.

Under no circumstances should the tool sharpener remove more material from any one flute just because it shows greater wear. This would create a large flute spacing error and inaccurate gear tooth profile. All cutting faces must be ground back the same amount that is needed to sharpen those showing the greatest amount of wear.

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is the training manager at ITW, Illinois Tools. He has over 35 years' experience in gearing and is the author of numerous books and articles on gearing subjects.

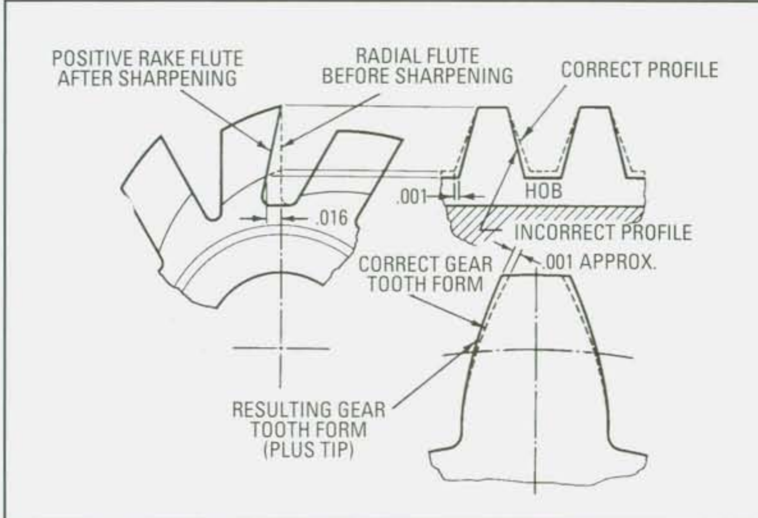


Fig. 3 - Sharpening radial flutes with a positive rake leads to production of gear teeth with thick tips.

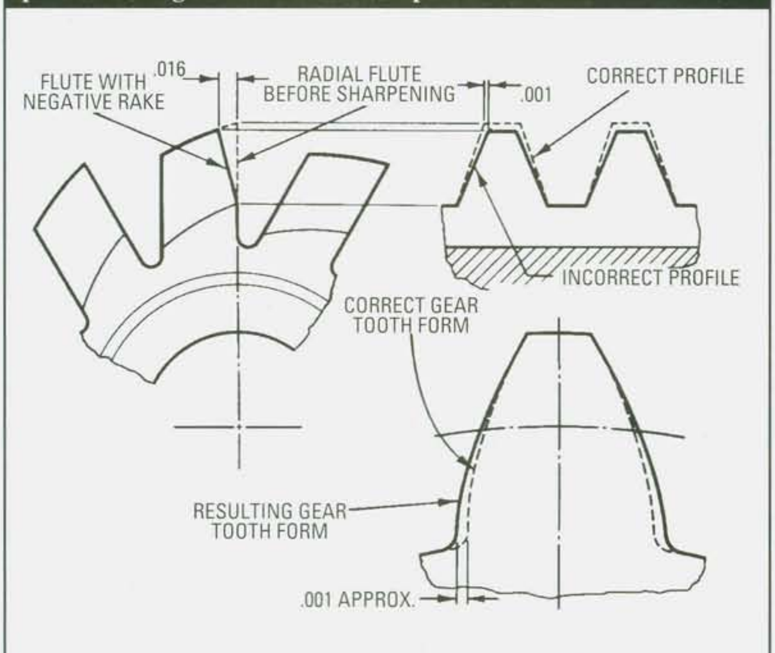


Fig. 4 - Sharpening radial flutes with negative rake leads to production of gear teeth with thick roots.

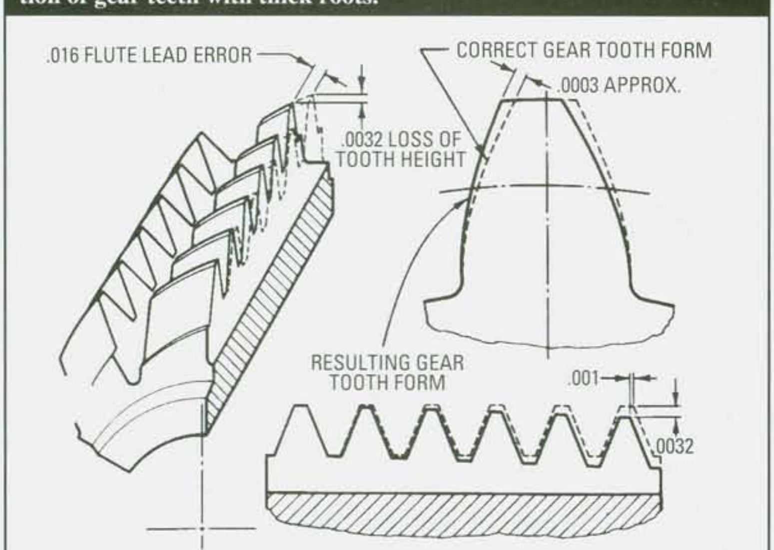


Fig. 5 - Lead errors in sharpening tend to produce distorted "leaning" teeth on the finished gear.

Flute Rake Error

Fig. 3 shows the effect of sharpening the flute with a positive rake when it should be radial. Because of the side clearance, this sharpening error makes the base of the hob tooth narrower, and as a result, the gear tooth becomes thicker at the tip. Here again, an error of .016 from radial will cause approximately .001 of error in the gear tooth profile.

Occasionally hobs are designed with a positive rake. In such cases the hob tooth form (or pressure angle) is designed to correspond to the positive rake flute. Such hobs must be sharpened with the same positive rake. On such hobs the rake angle and rake off-set are marked on the end of the hob.

Error in the rake of the flute is caused by improper setup of the grinding wheel in relation to the hob axis.

Fig. 4 shows the effect of sharpening the flute with a negative rake. Because of the side clearance, this sharpening error makes the tip of the hob tooth thin, and as a result, the gear tooth is thicker at the root. Again, an error of .016 from radial will cause about a .001 error in the gear tooth profile.

This sharpening error is also usually caused by incorrect setting of the grinding wheel in relation to the hob axis.

Flute Parallelism or Lead Error

Fig. 5 shows the effect of sharpening helical flutes with incorrect lead or straight flutes non-parallel with the hob axis. The result is an error in the lead and form of the hob teeth. In fact the lead on one side of the teeth becomes longer than the theoretical lead, and the lead on the other side becomes shorter. This produces a "leaning" gear tooth; one side plus, the other minus on involute.

However, a flute lead or parallelism error would have to be much larger than a flute spacing or rake error in order to affect the gear tooth profile to the same degree. This is true because only about two convolutions of hob teeth finish the gear tooth profile. A .016 flute lead error would result in about .001 lead error over all convolutions of hob teeth. Assuming the hob had eight convolutions, the gear tooth would be affected by two over eight or one-fourth of the total lead error of .001. The resulting tooth profile error would then be about .00025.

It is interesting to note that a flute lead error, because of the cam relief on the hob teeth, creates a tapered hob. The flute lead error is approximately 2 1/2 times the amount of taper in the hob diameter. Since the taper can be easily measured, this affords a quick easy way of measuring the flute lead error.

A sharpening error in flute lead may be caused by an incorrect sine bar setting, excessive backlash, worn machine parts, misaligned centers, or failing to "spark out."

How Hobs Are Sharpened

Machines designed and built solely for hob sharpening are on the market. These machines have automatic indexing provisions and can sharpen hobs with helical as well as straight flutes.

Hobs can also be sharpened in a cutter sharpening machine. However, in this case, the backs of hob teeth in straight-fluted hobs must be ground to provide accurate indexing surfaces for a steel supporting finger, such as is commonly used in cutter sharpening. (See Fig. 6.)

Hobs and helical flutes that are to be sharpened in a cutter sharpener require a guide bar. This bar must have the same number of equally spaced grooves of the same lead as the flutes of the hob. This guide bar and the hob are mounted on the same arbor and placed between centers on the cutter sharpener table. A guide finger mounted on the machine and engaging a groove on the guide bar rotates the hob the correct amount in relation to the table travel past the grinding wheel. (See Fig. 7.)

A saucer-shaped grinding wheel is used for hob sharpening. For straight-fluted hobs, either the flat or cone side of the wheel may be used. However, the cone side is preferable because of more uniform pressures between the wheel and the hob. With a flat wheel the area of contact between the wheel and hob is small at each end, but large in the center. (See Fig. 8.) If a heavy cut is being taken, more stock is removed at the ends and less in the center. To correct this more passes must be made through each flute.

The flat side of the wheel cannot be used for helical flutes because of interference at the root and top of the flute, resulting in a convex, nonradial cutting face. (See Fig. 9.)

The cone side of the wheel works well for all helical flute hobs, excepting those with large thread angles, such as are found in multiple thread worm gear hobs. In such cases interference becomes noticeable, and the wheel must be

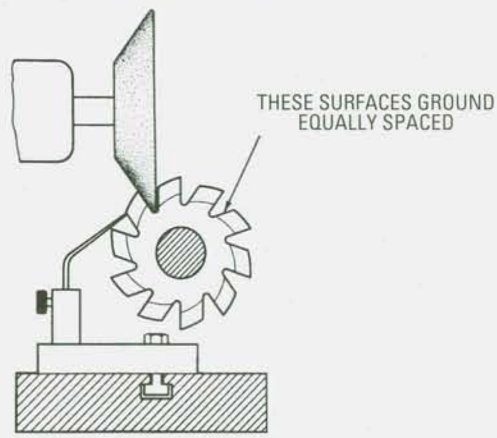


Fig. 6 - Backs of the teeth in straight-fluted hobs must be ground to provide an accurate indexing surface when sharpening is done on a cutter sharpener.

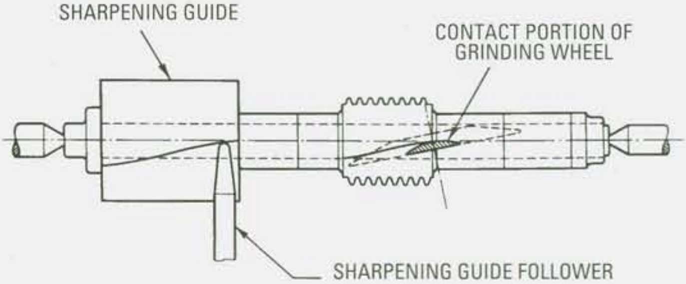


Fig. 7 - For sharpening hobs with helical flutes on a cutter sharpener, a guide bar with grooves of the same helical lead is required.

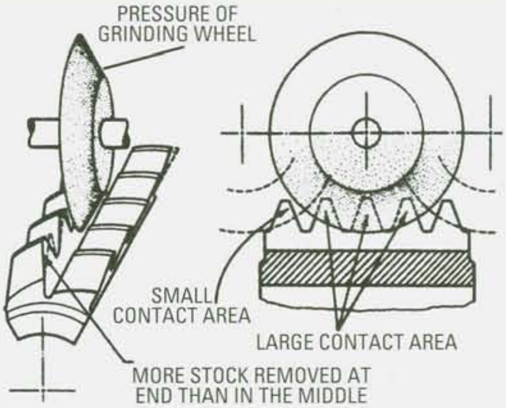


Fig. 8 - Flute grinding with the flat side of the grinding wheel as shown is not recommended. For uniform cutting pressures and uniform stock removal, the cone side of a saucer-shaped wheel should be used.

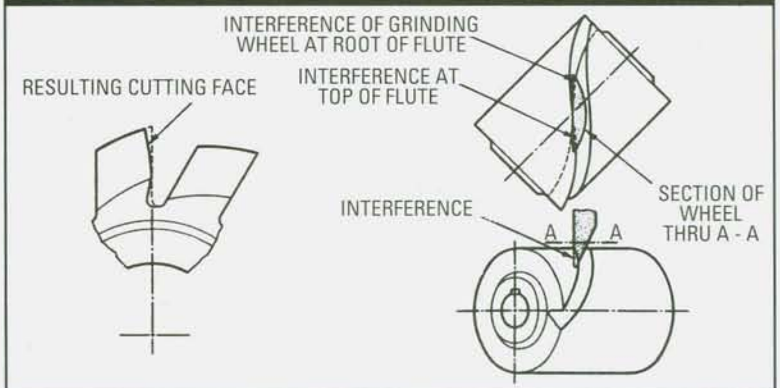


Fig. 9 - In sharpening hobs with helical flutes, the cone side of the wheel must be used to avoid interference.

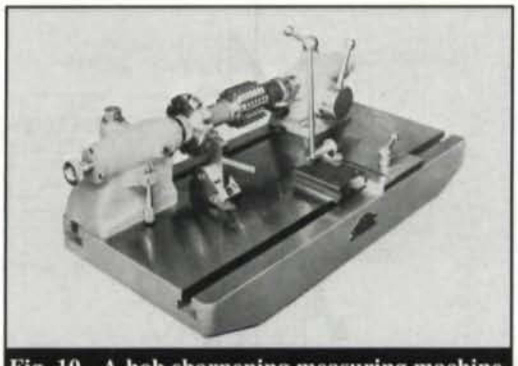


Fig. 10 - A hob sharpening measuring machine.

dressed to a curve that eliminates interference.

Effect of Sharpening Technique on Hob Life

In sharpening, care should be taken to avoid excessive heat. It is easily possible to create enough heat to soften the cutting edges and greatly increase the rate of hob wear. Another danger of excessive heat is the propagation of tiny cracks at the base of the hob teeth, which often results in tooth breakage. The "high" flute should be located and "kissed" before feeding in, or runout and unequal spacing conditions, when combined with the feed, may result in too heavy a grinding cut and excessive heat on one or more flutes.

It has been proved over and over again that more gears can be cut per sharpening when the grinding finish on the cutting faces of the hob teeth is good. It definitely pays to sharpen hobs with a good finish. The wheel should be dressed before the finish cut and should be allowed to "spark out", not only for accuracy's sake, but also for improved finish.

Need for Sharpening Inspection

It is obvious from the preceding that sharpening errors can result in gear tooth inaccuracies which prevent satisfactory gear performance. It is essential that hobs be sharpened accurately.

Sharpening errors, however, will occur. Mistakes will be made in the setup, machines will wear or become misaligned, worn arbors will be used, nuts will be tightened excessively, grinding wheels will not be allowed to "spark out", dirt and chips will get between collars, etc. And sharpening errors, even though large enough to cause intolerable gear errors, usually cannot be seen or measured with the naked eye.

Since it is only common sense that hobs with excessive sharpening errors be prevented from reaching the hobbing machine and cutting gears that have to be scrapped, the need for hob sharpening inspection is evident. Special measuring machines have been designed for this purpose, which measure directly, by means of a dial indicator, errors in flute spacing, rake, and parallelism of straight flutes. (See Fig. 10.) Flute lead errors are obtained indirectly through measurement of the taper in the hob diameter which results from an error in flute lead. The flute lead error is approximately 10 times the taper on diameter or approximately 5 times the difference in indicator readings, taken over the high points of the end teeth in the flute. These indicator readings can be obtained in the hob sharpening inspection machine.

All preceding remarks have been directed mainly to gear hobs, but they apply with equal force to all hobs, including sprocket hobs, spline hobs, and worm gear hobs. Fig. 11 shows the effect of sharpening errors on parallel key splines. Such inaccuracies may prevent proper fit.

In worm gear hobs, the high pressure angles and greater side clearance angles often used aggravate the bad effects of sharpening errors on the worm gear tooth form. Large lead angles are common in these hobs, and special wheel dressing is needed to avoid interference when sharpening.

Summary

1. Hobs sharpened outside of standard tolerances shown in Table 1 cut inaccurate gear teeth. This leads to unsatisfactory gear performance and often to early gear failure.

2. Sharpening with the cone side of the wheel is preferred for straight flute hobs and is essential on helical-flute hobs.

3. Excessive heat in sharpening and poor grinding finish shortens hob life.

4. Hob sharpening accuracy is easily checked with a hob sharpening measuring machine. ■

Presented at the AGMA 19th Annual Gear Manufacturing Symposium, April 7-9, 1991, Chicago, IL. Reprinted with permission. The opinions, statements, and conclusion presented in no way represent the position or opinion of the AGMA.

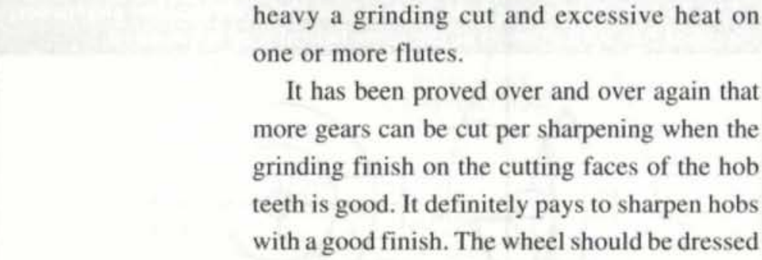


Fig. 11 - Parallel key spline profile inaccuracies resulting from improper spline hob sharpening.

A Clockwork Gear

William L. Janninck

Question: Could you explain what is meant by "horological gearing"? I never heard of this before, although I understand it has something to do with watches. Could you also explain the meaning of a "going gear train"?

Horology is the science of measuring time or the art of constructing instruments that indicate time. In earlier days this was as simple as tracking the sun using a sundial; today we use the vibration of a quartz crystal. For ultra-precise timing, the atomic clock uses a resonant frequency of Cesium 133 and is accurate to one second in 250,000 years. Not quite what you need around the house for telling time!

Although sundials can get quite sophisticated, they utilize no gears; neither do most modern digital timers and watches. But for the several centuries in between, clocks, watches, timers, fuses, and other such devices were prolific users of gears. An entire field of design and manufacture of these mechanical gear elements developed, and all the gears used in clocks, watches, and other timing devices, whether for driving the pointers, setting the hands, winding up springs, or driving the escapement mechanism and pendulum, were called "horological gearing".

Many books were written that covered the special concerns involved in this application of gearing, and the serious student could even enroll in correspondence courses in "horological science."

The problems of design faced by watch builders, especially considering the very small size of some mechanical watches, were totally different from those experienced by power train designers in transportation, construction, machine tools, and general machinery applications.

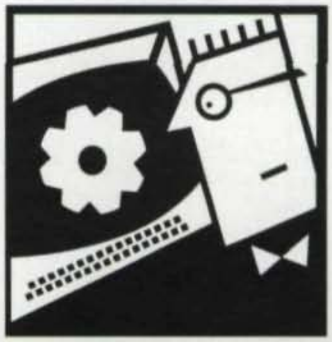
Some of the problems facing the horological gear designer come from the necessity of using multiple-stage, high-ratio, step-up gear sets. They are the opposite of the more usual application problems of step-down or speed reducing drives faced by most designers. In addition is the absolute need for low friction and high efficiency.

In a mechanical watch or clock, the driving energy is stored in a spring-wound drum and is slowly released as minute pulses through the step-up drive train and into an escapement mechanism, with the gear train starting and stopping completely with each cycle of the escapement wheel. The friction through the gear train must be minimal for the successful management and release of the available energy.

One of the first areas of concern in step-up drives is gear tooth friction. To reduce surface contact friction, surface finishes must be extremely good, and frequently the pinion teeth are polished. The reduced friction experienced in the arc of recession, or exit path out of mesh, rather than the arc of approach, or entry path into mesh, is used. To accomplish this, a cycloidal tooth form is used, and to further reduce the approach action, the cycloidal tooth form on the pinion addendum is modified, concentrating the action near the pitch line of the gear set.

For compactness, pinions of as few as 6, 7, or 8 teeth and step-up ratios of as much as 12:1 in a single mesh are used. The teeth can be as fine as 250 diametral pitch with depths as little as .012". To avoid friction losses from side thrust, the teeth are always spur.

Many tooth forms and proportions have been established suitable for horological purposes. Some of these used in step-up drives are the Ogival Form, British Standard 978, Black Forest Clock Standard, the Swiss Cycloidal, the Prescot, and the Circular Arc. Another one frequently encountered



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Address your gearing question to our panel of experts. Write to them care of Shop Floor, Gear Technology, P.O. Box 1426, Elk Grove Village, IL 60009, or call our editorial staff at (708) 437-6604.

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does gear and tool design and consulting. He has been involved with gears and gear manufacturing for 45 years, 40 of them with Illinois Tools - ITW Inc. He is the author of numerous articles on gear-related topics.

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is the Wickenburg form.

Fig. 1 shows a typical cycloidal form pinion and gear in mesh. The flanks on both members are radial lines below the pitch line. A special type of pinion having six teeth or more and used in step-up drives is called a lantern pinion and is shown in Fig. 2. A tooth layout is shown in Fig. 3. The teeth are formed of polished pins or wires set in a pair of end plates. These pinions are not recommended for reduction drives in horological applications.

The number of the tip modifications used on clock gears may cause one to question the functionality of these gears, since the tooth form departs from true conjugacy. Cycloidal gear ing is that one exception to the rule requiring a contact ratio of at least 1.0 for a pair of gears to be used successfully. Contact ratios of less than 1.0 on involute gearing are a signal of problems and, with an involute profile, indicate a possible damaging edge contact and lack of proper uniform transmis-

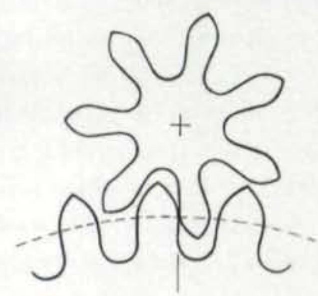

Fig. 1 - Cycloidal gear set.

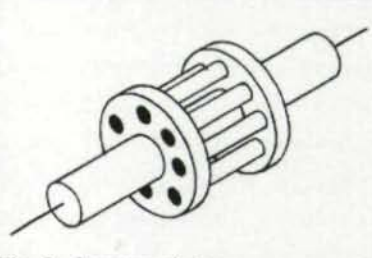

Fig. 2 - Lantern pinion.

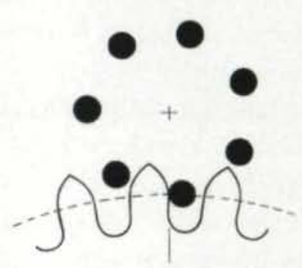

Fig. 3 - Lantern pinion gear set.

sion of rotation. This is not the case with cycloidal profiles, and edge contact is rare.

A step-up drive in a clock is one application where efficiency of energy transmission, rather than uniform transmission of rotation, is the primary objective. If these gears were inspected by the single flank method, the results would show a tooth-to-tooth rotational error, but this is not a concern for clock and watch gearing.

Not all the gears used in horological gearing have the cycloidal tooth form. Many applications can and do use involute gearing, such as those for winding drives, time setting, minute-

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hour hand synchronizing, or other motion transfer. In electric clocks using a synchronous electric motor power source, the gearing is all in reduction stages, and most use involute gearing. Electric meters used in measuring energy may use some special high-reduction worm drives and then end up with involute gearing for the dial recording drives. Liquid and gas meters also tend to use involute gearing.

It might be helpful to diagram the gear trains used in a typical, hand-wound chiming mantel clock, where several clock gearing applications can be shown and explained. Fig. 4 shows the main clock drive used for precise time keeping and display. Starting with the spring drum gear there are five step-up cycloidal tooth form stages located on six axes. The last stage

(Continued on page 48)

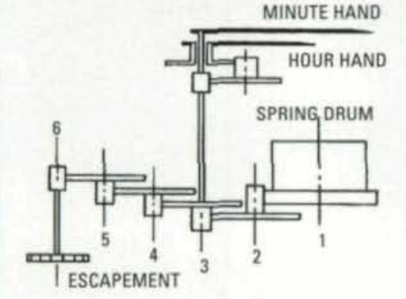

Fig. 4 - Main clock train.

Double Flank Grinding can increase your Productivity up to 50 %

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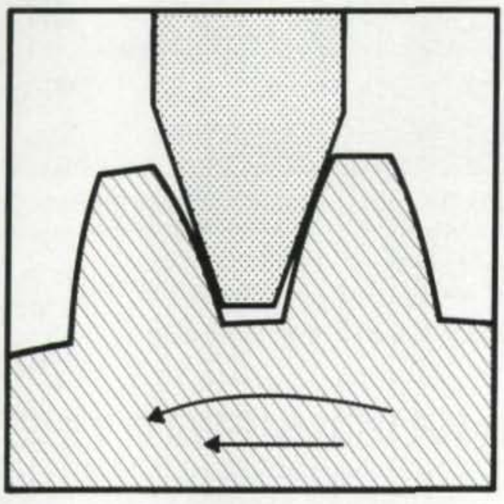
Successful companies like LUFKIN INDUSTRIES, INC. have been relying on the second to none accuracy and reliability of BHS-HÖFLER gear grinding machines for many years. BHS-HÖFLER is now excited about introducing their double flank grinding method which makes their machines unbeatably fast while maintaining the high level of accuracy BHS-HÖFLER is famous for. In some cases double flank grinding is even double as fast as single flank gear grinding!

The double flank grinding method grinds both flanks of a tooth gap simultaneously in one generation direction. A standard grinding wheel is used which is dressed to specs directly on the grinding machine with a CNC-grinding wheel dresser.



The following examples illustrate the actual time saved using double flank grinding as compared to single flank grinding. One pinion and one helical gear were selected for these examples to be ground on a BHS-HÖFLER H 1000 E.

Example 1: Pinion:

Number of teeth:	27
DP:	5.6
Pressure angle:	20 degr.
Helix angle:	13,5 degr.
Gear width:	4"
Outside diameter:	5.3"
Stock per flank:	.008"
Hardness:	HRC 62

AGMA Quality 14:
3 passes, 26 enveloping cuts per involute, intermediate dressing of the grinding wheel after 9-14-0 teeth.

Total time single flank grinding mode:
43,5 min.
Total time double flank grinding mode:
27,4 min.

AGMA Quality 13:
2 passes, 21 enveloping cuts per involute, intermediate dressing of the grinding wheel after 7-0 teeth.

Total time single flank grinding mode:
25,8 min.
Total time double flank grinding mode:
17,1 min.

Example 2: Gear:

Number of teeth:	92
DP:	3.18
Pressure angle:	20 degr.
Helix angle:	7 degr. R
Gear width:	4.1"
Outside diameter:	29.8"
Stock per flank:	0.01"
Hardness:	HRC 62

AGMA Quality 14:
3 passes, 32 enveloping cuts per involute, intermediate dressing of the grinding wheel after 8-16-0 teeth.

Total time single flank grinding mode: 132,1 min.
Total time double flank grinding mode: 92,9 min.

AGMA Quality 13:
2 passes, 28 enveloping cuts per involute, intermediate dressing of the grinding wheel after 8-0 teeth.

Total time single flank grinding mode: 86,9 min.
Total time double flank grinding mode: 59,7 min.

Workpiece mounting was accomplished in both cases with an efficient and accurate gear mounting system. The pinion was mounted between centers and the gear was setup on a newly developed pallet loading system.

The exclusive BHS-HÖFLER pallet loading system is the first high accuracy pallet system tailored to the requirements of the gear grinding industry. It reduces setup- and downtimes to an absolute minimum. Gears are clamped with pallet on the setup station outside of the gear grinding machine while it is in operation. During the time that one workpiece is being ground, the operator can true-up the next gear and prepare it for loading. When the previous workpiece is finished it is lifted with its pallet off the machine table and placed temporarily on a parking station. The trued-up gear can now be lifted with its pallet from the setup station and loaded onto the grinder. Once in place, the workpiece and pallet are clamped into position on the worktable to an accuracy of ± 1 micrometer (±40 millionths of an inch). Grinding can start immediately and the next workpiece can be trued-up on the setup station. This system can also be used in multi-machine operation whereby the trued-up pallets and gears can be stored at remote buffer stations.

These are just a few examples of the innovations you can expect from BHS-HÖFLER. If you would like to know more about these or other BHS-HÖFLER developments and products, please contact one of the following addresses at your earliest convenience. BHS-HÖFLER is looking forward to hearing from you.

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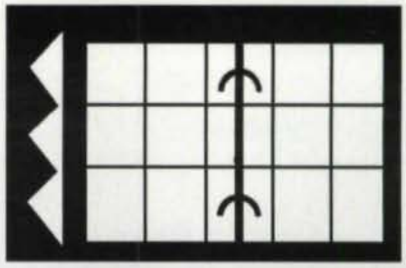

CALENDAR

FEBRUARY 12-14, 1992

3rd World Congress on Gearing and Power Transmissions. Palais de Congrès, Paris, France. Sponsored by major international organizations in the power transmission field and in the U.S. by AGMA and ASME-GRI. Topics include gear geometry, wormgears, transmission elements, chains, lubrication, materials, noise and vibration, testing, and applications. Contact: MCI (CMET) Tel: 33 1 42.94.27.67. FAX: 33 1 42.93.29.67.

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National Association of Manufacturer's National Manufacturing Week. Six interrelated trade events and NAM annual meeting focusing on American manufactured exports. All events at McCormick Place Complex, Chicago, IL. Contact Laura Brown, (202) 637-3087.

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CALL FOR PAPERS

For AGMA 1992 Fall Technical Meeting. Papers accepted on design, analysis, manufacturing, and application of gears, gear drives, and related products, as well as related processes and procedures. Authors should be prepared to meet the following schedule: **Jan. 2** - Abstracts due at AGMA Headquarters. For more information, contact Bill Daniels, (703) 684-0211.

pinion ending on the sixth axis drives the time controlling mechanism, consisting of the escapement wheel and a rotary pendulum, while axis three is the take-off to directly turn the minute hand. This also drives another two stages in reduction to synchronize and drive the hour hand.

Fig. 5 diagrams the second segment of the clock, the quarter-hour chiming section. It also has five stages of step-up cycloidal gears running from the spring drum to an air paddle type speed governor. From axis number three, motion is taken off and passed through a chain of four idler gears to drive a set of chime cams. These idler gears are conventional and utilize typical involute gears.

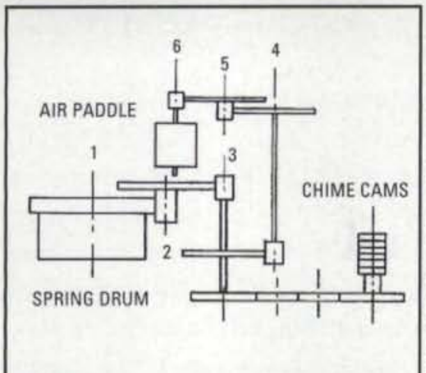

Fig. 5 - Quarter-hour chime train.

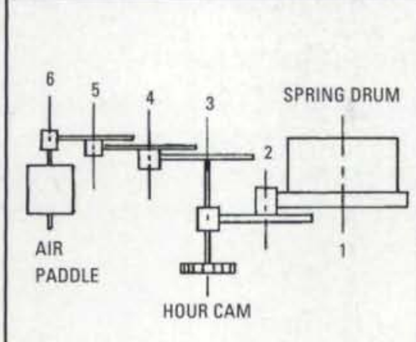

Fig. 6 - Hour-striking train.

In Fig. 6 the third section of the clock, used for striking the hours, is shown. It too uses a five-stage, step-up cycloidal train, starting with the drum and ending with another air paddle speed governor. There is a take-off from axis number three, and it turns the hour striking cam.

All in all, excluding cams, ratchets, and escapement parts, there are 38 gears in this clock.

Another name for the step-up drive described above is a going gear train. As shown in our dissection of the mantel clock, the going trains are five stages of step-up gearing using driven pinions of very small numbers of teeth.

In the above case, the energy was stored in hand-wound springs, but this can be accomplished by other means. A small, geared electric motor can wind a smaller, lighter spring, or an electrical solenoid and a ratchet can do the same thing. Another interesting design uses a sealed air chamber, which expands and collapses with atmospheric pressure fluctuations to do the winding. One of the oldest methods uses gravity by means of a weight, a rope, and a pulley to store the energy.

Another interesting application is seen in the traditional mechanical aneroid altimeter that is required on all aircraft, at least as a back-up instrument. It uses cycloidal gearing in a step-up train. The gears are of higher quality or precision and usually use highly polished pinions. The small movements of the aneroid chamber are magnified through the gearing, driving the indicating pointers.

Pressure measuring gages driven by a Bourdon tube sense the flexure of the tube to indicate the pressure. The driving force is quite high, and a pair of step-up stages will suffice. Those gages I have seen utilize involute gearing.

Mechanical dial indicators usually use geared step-up drives, and these too are of involute form.

Except for the very small teeth and small size of the gears, the usual manufacturing methods are used. Small hobbing machines using small cycloidal hobs and automated form milling machines using miniature rotary form milling cutters are employed. Most of these machines are of Swiss, German, or English origin.

Inspection, particularly for the elements of profile, tooth thickness, and adjacent tooth spacing, of these very fine-pitched, cycloidal, profiled gears and pinions, can best be done by optical means, using magnified projected images and by comparing the shadow against enlarged tooth layouts. Any visible bumps or hollows are cause for concern, especially in the immediate area of the pitch line. If the inspection layout includes several teeth, then any adjacent spacing errors can be judged. Surface finish can also be assessed by using microscopes and comparison samples.

In the clock gear field, backlash also requires special consideration. The dust and fiber particles that can get into gear teeth are larger in comparison to the small teeth in watch applications. Provisions must be made for space for this debris to settle without binding. Oils and greases catch and hold the debris and are not normally used on the very fine pitches.

The suggestion has been made that all of this special technology and procedure are not really necessary and any difference between clock gearing, and the more common involute gearing might be small. I personally became aware of the significant difference between them many years ago. We had a damaged drive gear set in a timing device which we replaced on an emergency basis, cutting the gears ourselves, using an available fine-pitch, involute hob. After all, we thought, a gear is a gear as long as the tooth ratio is correct. We could not make this apparatus work no matter how much buffing, polishing, oiling, and prodding we did. Finally, in utter frustration, as a last, desperate resort, the original tooth forms were copied, at some expense, and upon assembly, the mechanism took off running. We were pretty well convinced. ■

To address questions to Mr. William L. Janninck, please circle Reader Service No. 78.