GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

NOVEMBER/DECEMBER 1991

DESIGINING MINIMUM-WEIGHT GEARS

ESTABLISHING AN OPTIMUM SHOT PEENING SPECIFICATION - PART 1

RECENT DEVELOPMENTS IN GEAR METROLOGY

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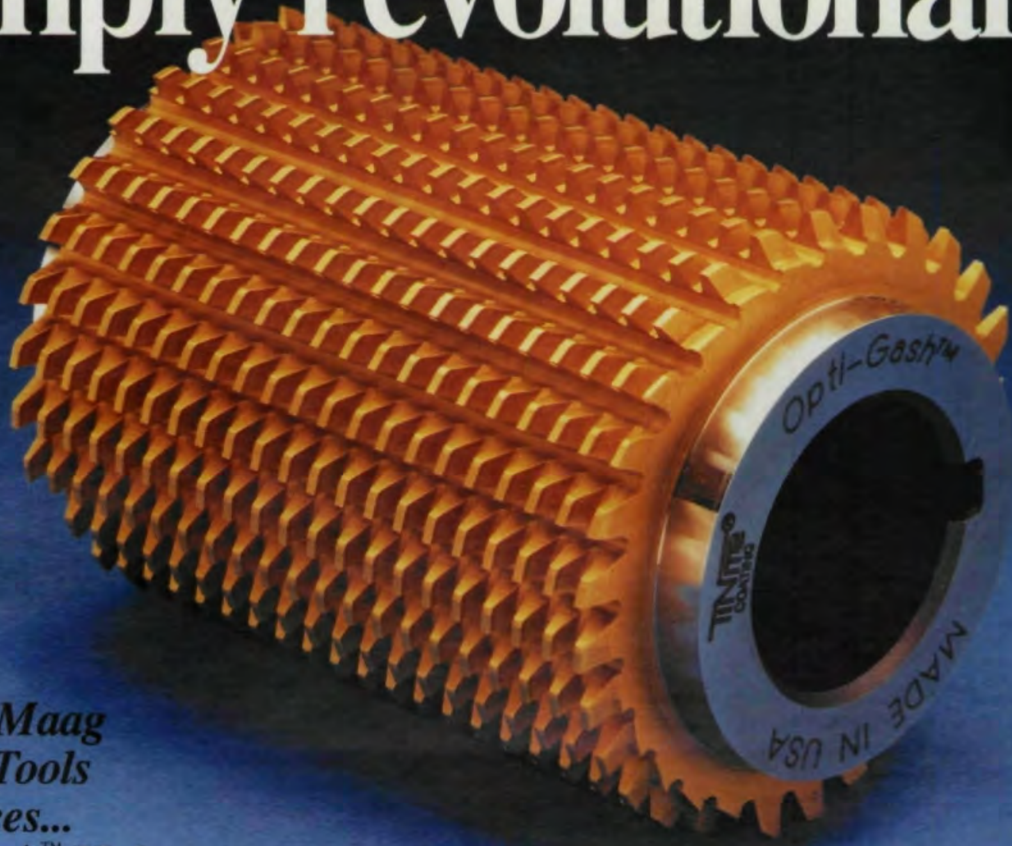

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CONTENTS

NOVEMBER/DECEMBER, 1991

FEATURES

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SPECIAL FEATURES

DEPARTMENTS

GEAR TECHNOLOGY

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To Market, To Market

A few months ago at an AGMA management seminar, I was surprised by the feverish note taking that went on at a presentation on marketing. The sight reminded me that while many of us in the gear industry are good engineers, designers, and managers, we are often not as familiar - or comfortable - with less concrete concepts, such as marketing.

We tend to like things neat and quantifiable, and marketing isn't always that way. Marketing doesn't have convenient standards or specifications that, when plugged in, give "right" answers. Marketing solutions are not available on trig tables. But if we are to compete effectively in today's grueling business climate, we have to overcome our aversion and our ignorance of this subject and learn to use it effectively. Good marketing can often help you take the lead over your competitors when all other factors are equal.

PUBLISHER'S PAGE

But what exactly is marketing? In its broadest sense, marketing involves every aspect of your company's operations - production, delivery, research and development, advertising, sales and promotions, customer relations. It is the total combination of functions - the products you make, the segment of the market you're trying to reach, the messages conveyed by your advertising and sales materials, your responses to customer inquiries - that taken together produce your company's image in the marketplace. Done right it makes for those most valuable commodities, company reputation and name recognition, which in turn, make for increased sales.

Some parts of marketing may seem far removed from building gears. Unfamiliar terminology like "repositioning," "corporate image," "product research," and "media relations" are apt to crop up in marketing discussions. But image or perception are important in the marketplace. What good does it do to build the best gears in the world if customers don't know that, or, worse yet, think that you don't? And no one is better able or more willing to spread the word about your company and products than you are. If you're not committed to developing a favorable image for your company, who will be?

(Continued on page 36)

VIEWPOINT

Dear Editor:

In Mr. Yefim Kotlyar's article "Reverse Engineering" in the July/August issue, I found an error in the formula used to calculate the ACL = Actual lead from the ASL = Assumed lead.

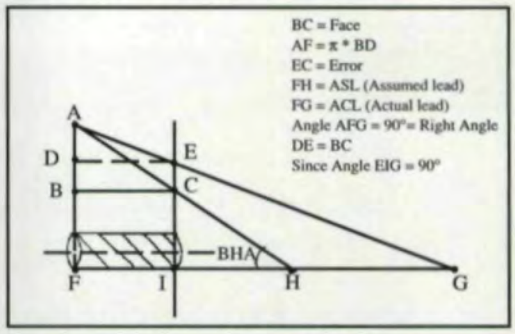
The correct formula should read ACL = ASL*(1 + error/(face*tan(BHA) - error)), instead of ACL = ASL +*(1+ error/ face*tan(BHA)).

The formula is derived as follows:

1) FG = AF*DE/AD = AF*BC/AD

2) AD - AB - BD = AB - EC, since DE/ /BC array AB = BC*tanBHA substituting in the identity (2) gives AD = BC*tan BHA - EC = Face*tanBHA - error.

Therefore, from (1) we derive:



3) FC = AF*BC/(BC*tanBHA - EC) = π*BD*/(Face*tanBHA - error)

4) AF = FH*tanBHA or π*BD = ASL*tanBHA.

Substituting in (3) π*BD we get

FG = ASL*tanBHA*Face/ (Face*tanBHA - error) = ASL*Face*tanBHA/ (Face*tanBHA - error)

This equation can be arranged in the forms:

FG = (ASL*Face*tanBHA - error + error)/(Face*tanBHA - error)

which gives the wanted equation.

5) ACL = ASL*(1 + error/ (Face* tanBHA - error)).

I am sure that Mr. Kotlyar has his formula right, and the error is the result of a typo, therefore, he will pardon my intrusion.

Sante Basili, V.P.
SU America, Inc.

Mr. Kotlyar replies:

Dr. Basili is right: The correct formula for determining an actual lead should be

ACL = ASL*(1 + Error/ Face*Tan(BHA) - Error)).

The formula used in the article, ACL = ASL*(1 + Error(Face*Tan (BHA))), is an approximation formula which has some degree of inaccuracy. For example, let's assume a gear of 25° helix angle, 1" face width, and with an error detected during lead check of 0.01".

Results calculated by the correct formula are

ACL = ASL*(+ .01/(1*tan(25) - .01)) = ASL*1.02191.

Results calculated by the approximation formula are:

ACL = ASL*(1 + .01/(1*tan(25))) = ASL*1.02144.

As you can see, the inaccuracy for the lead correction multiplier appears only in the fourth digit after the decimal point. If the lead error detected on the inspection machine is .001", then the inaccuracy would be even smaller, appearing only in the sixth digit after the decimal point.

Nevertheless, it is conceivable that the use of Dr. Basili's more precise formula would result in a fewer number of iterations for determining actual lead.

Dear Editor:

May I reinforce what my namesake Robert E. Smith said in your September/ October issue about the use of single flank testing?

Single flank testing is essential for worm and bevel gears if noise or vibration is a problem, and also for parallel axis gears if noise is a problem in a critical installation or if accuracy matters, as for printing problems.

In Britain some companies already use 100% checking of S.F. and find that it pays because the rogue gears can be intercepted before they are built into equipment. Statistical checking is not normally used, partly because the objective is to weed out the 5% of poor gears instead of having to tighten tolerances expensively, and partly because S.F. testing is now so fast, it is relatively cheap.

Cycle times for testing can easily be faster than 2/minute, and if many gears are being tested, a special purpose tester can be set up at 20% of the cost of a test which does "everything".

An additional advantage of S.F. testing is that, unlike double flank testing, it can be done at full load quite cheaply, either on a development rig or on a production test rig, and this gives additional information about the effects of torque on load sharing between teeth and on the effects of alignment between teeth on noise levels.

Some British firms have already found that they need a "commercial" S.F. tester as a production control and an additional cheap "high speed" set for use in their development test shop. On a long and complicated gear train, such as that on a large printer, it is a great advantage to be able to measure accuracy between shafts 30 meters apart directly at any speed up to full speed.

Although AGMA has not yet established tolerances for S.F. tests, DIN specs have listed them. In practice, however, specifications are little help, but a couple of sets of gears built into equipment rapidly establishes allowable tolerances on a production line.

The great advantage of S.F. testing is that, for noise control, it is so much faster (cheaper) and easier than using separate pitch, involute, and helix check, that it is by far the most economical way of keeping a check on subcontractors. In time it is the economics which will lead to much greater use.

J. Derek Smith
Cambridge University

A Rational Procedure for Designing Minimum-Weight Gears

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GEARTECH
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Abstract: A simple, closed-form procedure is presented for designing minimum-weight spur and helical gearsets. The procedure includes methods for optimizing addendum modification for maximum pitting and wear resistance, bending strength, or scuffing resistance.

Introduction

Gear design is a process of synthesis where gear geometry, materials, heat treatment, manufacturing methods, and lubrication are selected to meet the performance requirements of a given application. The designer must design the gearset with adequate pitting resistance, bending strength, and scuffing resistance to transmit the required power for the design life. With the algorithm presented here, one can select materials and heat treatment within the economic constraints and limitations of manufacturing facilities, and optimize the gear geometry to satisfy constraints on weight, size, and configuration. This article assumes that the gear ratio is known. Methods already exist[1] for choosing the ratio of each gearset in a multistage gearbox to minimize overall weight. The gear designer can minimize noise level and operating temperature by minimizing the pitch line velocity and sliding velocity. This is done by specifying high gear accuracy and selecting material strengths consistent with maximum material hardness, to obtain minimum-size gearsets with teeth no larger than necessary to balance the pitting resistance and bending strength.

Gear design is not the same as gear analysis. Existing gearsets can only be analyzed, not designed. While design is more challenging than analysis, current textbooks do not provide procedures for designing minimum-weight gears. They usually recommend that the number of teeth in the pinion be chosen based solely on avoiding undercut. This article will show why this practice or any procedure which arbitrarily selects the number of pinion teeth will not result in minimum-weight gearsets. Although there have been many technical papers on gear designs (see Refs. 2-3, for example) most advocate using computer-based search algorithms which are unnecessary. Tucker[4] came the closest to an efficient algorithm, but he did not show how to find the optimum number of teeth for the pinion.

Optimum Number of Pinion Teeth

The optimum number of pinion teeth maximizes the load capacity of a gearset. Fig. 1 shows that load capacity is limited by surface fatigue, bending fatigue, or scuffing, depending on the number of teeth in the pinion. Also, there is a lower limit to the number of teeth, below which undercut occurs. The shaded zone in Fig. 1 is bounded by all three failure-mode curves and the undercut limit. It applies to a homologous class of gears with a specific combination

Nomenclature

(N_p)optimum = optimum number of pinion teeth	**K_c** = pitting resistance constant
(x1)min = minimum addendum modification coefficient to avoid undercut	**K_D** = combined dreating factor
	K_L = bending strength life factor
ϕ_c = normal generating pressure angle	**K_t** = bending strength constant
ϕ_s = transverse generating pressure angle	**m_a** = aspect (F/d) ratio
ψ_s = standard generating helix angle	**m_G** = gear ratio ($m_G \geq 1$)
C_1 = distance to SAP (See Fig. 2)	**N** = number of load cycles
C_5 = distance to EAP (See Fig. 2)	**n_c** = pitting resistance safety factor
C_6 = distance between interference points (See Fig. 2)	**N_p** = number of teeth in pinion
	n_p = pinion speed (rpm)
Ca = application factor	**n_t** = bending strength safety factor
C_D = combined dreating factor	**P** = transmitted horsepower
C_L = pitting resistance life factor	**P_n** = normal diametral pitch
Cm = load distribution factor	**Sac** = allowable (uncorrected) contact stress
C_p = elastic coefficient (Cp = 2300 for steel)	**Sat** = allowable (uncorrected) bending stress
Cs = size factor	**Snc** = contact strength
Cv = dynamic factor	**Snt** = bending strength
d = operating pitch diameter of pinion	Subscripts/sign convention
F = net face width	**p** = pinion
H_B = Brinell hardness	**1** = pinion, **2** = gear
I = pitting resistance geometry factor	(±) = upper sign external gearsets, lower sign internal gearsets
J = bending strength geometry factor	

of gear geometry, material properties, and application requirements. The relative positions of the curves change as these parameters change. This is not a disadvantage to the gear designer because the algorithm presented here directly solves for the optimum number of pinion teeth, making it unnecessary to draw Fig. 1, which is shown strictly for demonstrating the concept of the optimum number of pinion teeth. The curve marked "Surface Fatigue", representing the pitting resistance of the gearset, is relatively flat, being only weakly influenced by the number of pinion teeth. In contrast, the curve marked "Bending Fatigue", representing the bending strength, depends strongly on the number of pinion teeth, and it drops rapidly as the number of teeth increases. Maximum load capacity occurs at point "A", where the pitting resistance and bending strength are balanced. For more pinion teeth (to the right of point "A"), load capacity is controlled by bending fatigue, while for fewer teeth (to the left of point "A"), load capacity is controlled by surface fatigue.

The two failure modes are quite different. Surface fatigue usually progresses relatively slowly, starting with a few pits, which may increase in number and coalesce into large spalls. As the tooth profiles deteriorate with pitting, the gears generate noise and vibration, which warns of the surface fatigue failure. In contrast, bending fatigue may progress rapidly as a fatigue crack propagates across the base of a tooth, breaking the tooth with little or no warning. Hence, surface fatigue is often less serious than bending fatigue, which is frequently catastrophic.

Considering the differences between pitting fatigue and bending fatigue, it is prudent to select the number of pinion teeth somewhat to the left of point "A" (shown by the vertical line marked (N_p)optimum in Fig. 1), where surface

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fatigue controls rather than bending fatigue. With this design approach, not much load capacity is lost because the surface fatigue curve is relatively horizontal, while a margin of safety against bending fatigue is gained. This practice should not be carried to extremes, because pinions with few large teeth (with high specific sliding ratios) are prone to scuffing (See point "B" on curve marked "Scuffing Failure" in Fig. 1).

Some textbooks recommend using a number of teeth for the pinion equal to the minimum required to avoid undercut. This gives gearsets with less than optimum load capacity, which are prone to scuffing (see point "C" in Fig. 1). A pinion tooth number near (N_p)optimum provides a good balance between pitting resistance and bending strength, while good scuffing resistance is also obtained because the teeth are not larger than necessary.

Design Algorithm

There is no need for cut-and-try procedures for gear design if one exploits the near independence of pitting resistance and the number of pinion teeth. The following algorithm first solves for the diameter and face width of the pinion based on surface fatigue, and then solves for the optimum number of pinion teeth by simultaneously satisfying the surface fatigue and the bending fatigue constraints. It is derived from equations given in AGMA 218.01,[5] and is limited to steel. For alloys other than those shown, substitute allowable stresses, Sac and Sat from Ref. 5. Because it is necessary to approximate the geometry factors I and J, the final design must be verified using Ref. 5.

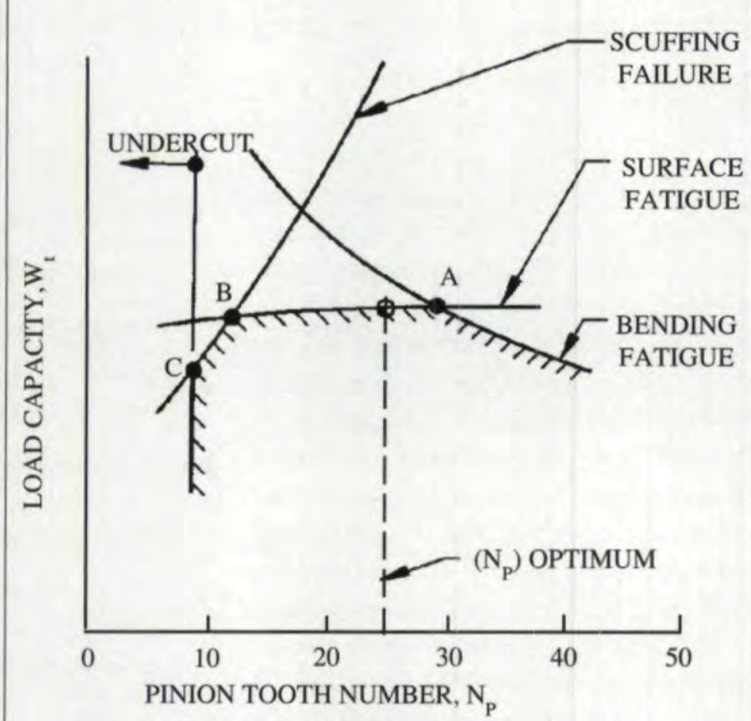

Fig. 1

Allowable (uncorrected) stresses for through-hardened steel:

$$Sac = 26000 + 327*H_B$$
$$Sat = -274 + 167*H_B - 0.152*H_B^2$$

Allowable (uncorrected) stresses for carburized steel:

$$Sac = 180,000$$
$$Sat = 55,000$$

Life factors:

$$C_L = 2.4660*N^{-0.0560}$$
$$K_L = 1.6831*N^{-0.0323}$$

Contact strength:

$$Snc = C_L*Sac$$

Bending strength:

$Snt = K_L*Sat$ (for reversed bending, multiply Snt by 0.7.)

The contact strengths and bending strength are calculated for both the pinion and gear, and the minimum values of Snc and Snt are used in the following equations.

Combined derating factor:

$$C_D = K_D = \frac{Ca*Cs*Cm}{Cv} \qquad (1)$$

Geometry factors for spur gears:

$$I = \frac{\sin\phi_c*\cos\phi_c}{2}\left(\frac{m_G}{m_G \pm 1}\right) \qquad (2)$$

$$J = 0.45$$

Geometry factors for helical gears:

$$I = \frac{1 + 0.00682*\phi_c}{4.0584}\left(\frac{m_G}{m_G \pm 1}\right) \qquad (3)$$

$$J = 0.50$$

Pitting resistance constant:

$$Kc = \frac{126000*P*C_D}{I*n_p}\left(\frac{C_p*n_c}{Snc}\right)^2 \qquad (4)$$

Bending strength constant:

$$Kt = \frac{126000*P*K_D}{J*n_p}\left(\frac{n_t}{Snt}\right) \qquad (5)$$

Aspect (F/d) ratio:

$$m_a = \frac{m_G}{m_G + 1} \quad \text{(for spur and single helical)} \qquad (6)$$

$$m_a = \frac{2*m_G}{m_G + 1} \quad \text{(for double helical)}$$

Pinion diameter:

$$d = \left(\frac{Kc}{m_a}\right)^{1/3} \qquad (7)$$

Face width:

$$F = d*m_a \qquad (8)$$

Optimum number of pinion teeth:

$$(N_P)\text{ optimum} = \frac{Kc}{Kt} \qquad (9)$$

(Round to integer)

Addendum Modification

Once the diameter, face width, and optimum number of teeth for the pinion are determined with the design algorithm, routine methods are used to select the number of teeth in the gear, diametral pitch, and operating center distance. However, the gear design is not complete until the addendum modification has been selected considering the following criteria:

- •avoiding undercut
- •balanced specific sliding
- •balanced bending fatigue life
- •balanced flash temperature
- •avoiding narrow toplands.

Avoiding undercut. The design algorithm usually gives a number of pinion teeth considerably larger than the number to avoid undercut. Conditions which lead to small numbers of teeth are high material hardness, short design life, large gear ratios, and high bending fatigue safety factors. With reasonable selections of these parameters, (N_p)optimum is usually greater than 20. In any case, the minimum addendum modification coefficient (to avoid undercut) for the pinion is given by:

$$(x_1)_{min} = 1.1 - N_p\left(\frac{\sin^2 \phi_s}{2* \cos \psi_s}\right) \qquad (10)$$

Balanced Specific Sliding. Maximum pitting and wear resistance is obtained by balancing the specific sliding ratio at the ends of the path of contact. This is done by iteratively varying the addendum modification coefficients of the pinion and gear until the following equation is satisfied:

$$\left(\frac{C_6}{C_1} \mp 1\right)\left(\frac{C_6}{C_5} \mp 1\right) = m_G^2 \qquad (11)$$

where:

C_1 = distance to SAP (See Fig. 2)
C_5 = distance to EAP (See Fig. 2)
C_6 = distance between interference points (See Fig. 2)

Balanced bending fatigue life. Maximum bending fatigue resistance is obtained by iteratively varying the addendum modification coefficients of the pinion and gear until the ratio

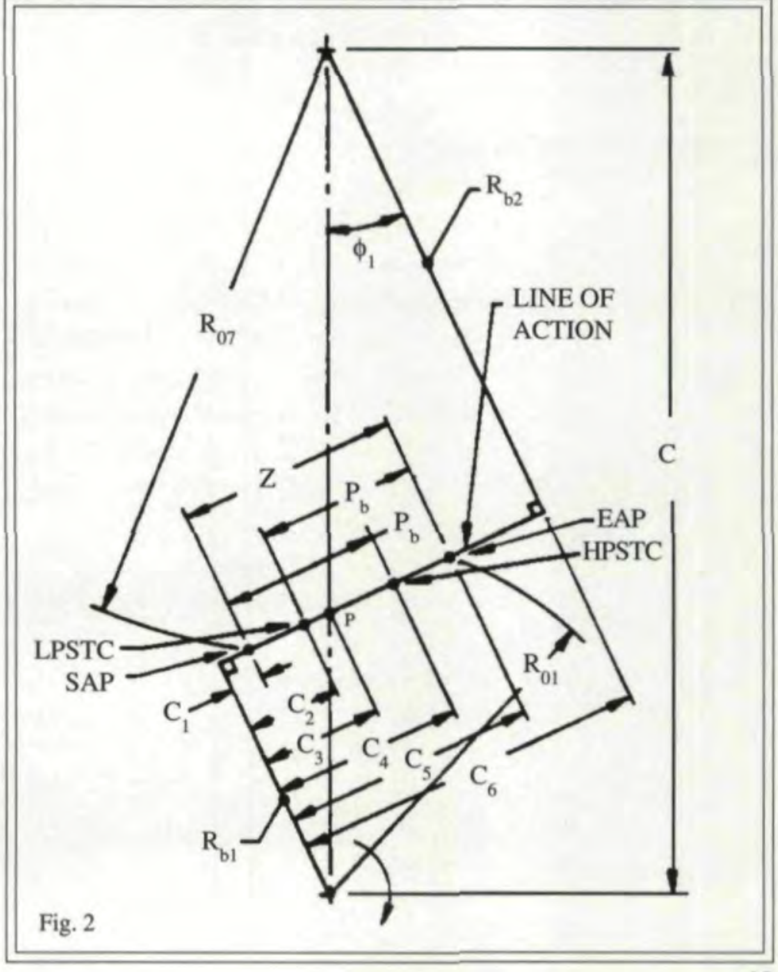

Fig. 2

of the bending strength geometry factors equals the ratio of bending strengths, i.e.,

$$\frac{J_1}{J_2} = \frac{Snt_2}{Snt_1} \tag{12}$$

Balanced flash temperature. Maximum scuffing resistance is obtained by minimizing the contact temperature. This is done by iteratively varying the addendum modification coefficients of the pinion and gear, while calculating the flash temperature by Blok's equation, until the flash temperature peaks in the approach and recess portions of the line of action are equal. The flash temperature should be calculated at the points SAP, LPSTC, HPSTC, EAP, and at several points in the two pair zones (between points SAP and LPSTC and between points HPSTC and EAP. (See Fig. 2.)

Avoiding narrow toplands. The maximum permissible addendum modification coefficients are obtained by iteratively varying the addendum modification coefficients of the pinion and gear until their topland thicknesses are equal to the minimum allowable (usually $0.3/P_n$).

Design Audit

With the addendum modification selected, the gear design is complete. It is necessary to audit the design by analyzing the stresses and lives (using Ref. 5) because approximate values were used for I and J. The only change that is usually required to meet the design life is a small adjustment of the face width. Although it is beyond the scope of this article, the selection of the lubricant type and viscosity should be verified by calculating the film thickness and flash temperature to ensure that they are within allowable limits.

Example

Using the data from Ref. 3, Example 1:

$Snc = 200{,}000$ psi
$Snt = 60{,}000$ psi
$P = 20$ hp
$n_p = 1260.5$ rpm
$\phi_c = 20°$
$m_G = 5$
$m_a = 0.25$
$n_c = n_t = 1.0$
$C_D = K_D = 1.0$

The design algorithm gives;

$I = 0.134$
$J = 0.450$
$K_c = 1.973$
$K_t = 0.074$
$d = 1.991"$
$F = 0.498"$
(N_p)optimum = 27
$P_n = 13$

Ref. 3 obtained essentially the same results after an extensive computer search.

Conclusions

1. Maximum load capacity or minimum-weight gearsets are obtained by selecting the optimum number of teeth for the pinion. (N_p)optimum, which balances the pitting resistance and the bending strength.

2. (N_p)optimum is easily found from a simple, closed-form design algorithm.

3. Addendum modification is designed to obtain maximum pitting and wear resistance, bending strength, or scuffing resistance by balancing specific sliding, bending strength, geometry factors, or flash temperature.

4. Any design procedure that selects the number of teeth in the pinion based solely on avoiding undercut or which arbitrarily selects the number of pinion teeth, will not result in gearsets with optimum load capacity. Such procedures usually give gearsets with low pitting resistance and low scuffing resistance. ■

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Optimum Shot Peening Specification-I

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Abstract: Shot peening is widely recognized as a proven, cost-effective process to enhance the fatigue characteristics of metal parts and eliminate the problems of stress corrosion cracking. Additional benefits accrue in the areas of forming and texturizing. Though shot peening is widely used today, the means of specifying process parameters and controlling documents for process control are not widely understood. Questions regarding shot size, intensity, and blueprint specification to assure a high quality and repeatable shot peening process are continually asked by many design and materials engineers.

This article should answer many of the questions frequently asked by engineering professionals and to further assist companies interested in establishing a general shot peening specification.

Many existing internal company specifications are adequate, but many are not because they have not been updated to coincide with the many improvements in shot peening technology over the past years. Companies considering creation of an in-house specification or interested in revising an existing specification should consult a knowledgeable shot peening authority.

For smaller companies and those who less frequently specify the shot peening process, good specifications, which can be used as a reference, are readily available. Two of these are Military Standard MILS-13165-B and AMS 2430.

We have assumed that the reader of this article understands the basics of shot peening and its effect on gearing and realizes that shot peening is an effective tool for combating problems of fatigue and stress corrosion cracking, as well as for assisting in forming and correction of shape. A brief discussion of the theory of shot peening is provided, but the reader should consult Refs. 1-4 for a more in-depth review.

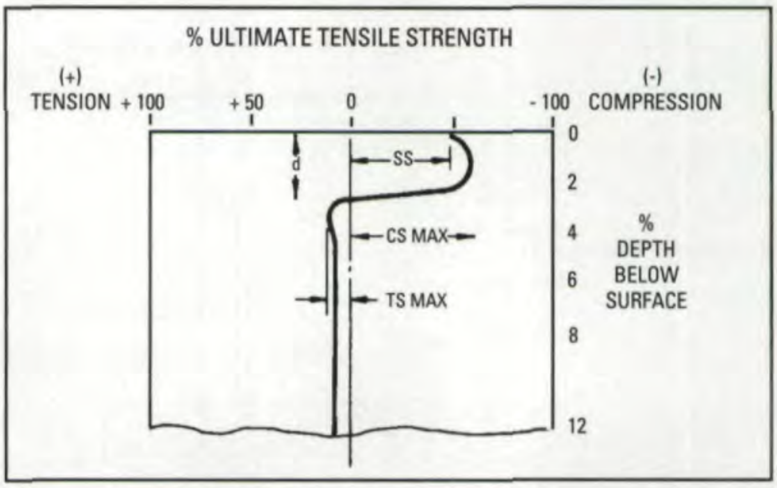

Fig. 1 - Example of residual stress profile created by shot peening.

Shot Peening Theory

Shot peening, by definition, is the bombardment of a surface of a material by small spherical media (the shot) to produce a thin layer of high magnitude residual (or self) compressive stress. This residual or self stress is introduced into a material prior to any actual application of loads to a component. The magnitude and depth of these compressive stresses are predictable. As shown in Fig. 1,[5] the maximum compressive stress usually occurs at some distance below the peened surface, which is represented by the top horizontal line. Typically, this magnitude of compressive stress (CS MAX) is approximately 50-60% of the ultimate tensile strength of the material, as shown in Fig.2.[6] (Dual intensity peening can move the CS MAX closer to the surface of the peened material.) Since this method is not typically employed, Fig. 1 is adequate for this discussion. The "d" represents the depth of compression or the point at which

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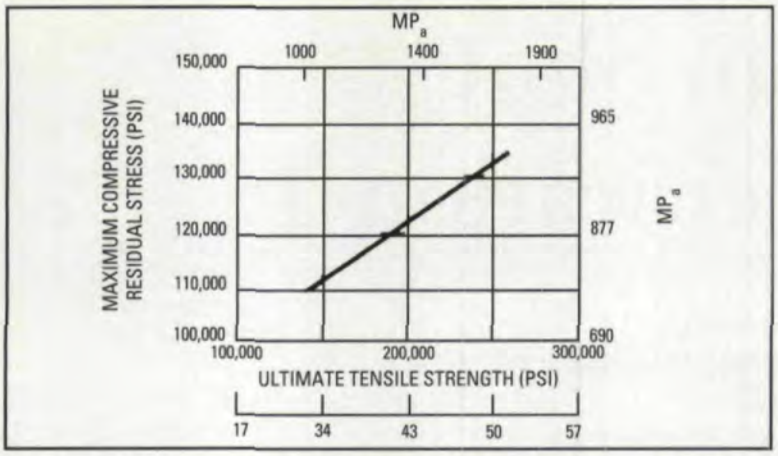

Fig. 2 - Residual stress produced by shot peening vs. tensile strength of steel.

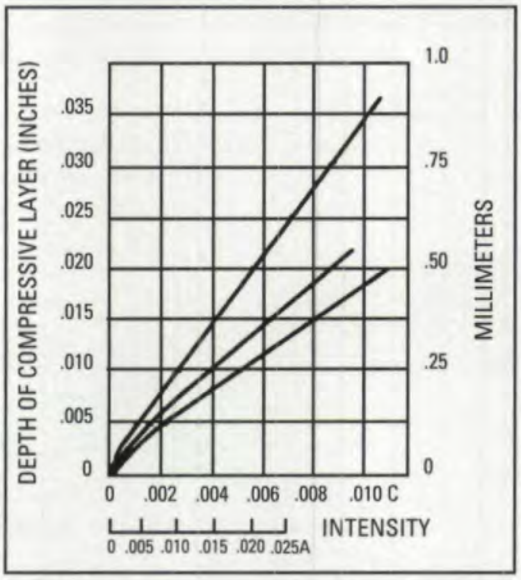

Fig. 3 - Depth of compression vs. Almen arc height.

the compressive stress induced by the peening crosses the neutral axis and becomes tensile.

The depth, as shown in Fig. 3, is dependent on the hardness of the target material and the mass and velocity of the shot. Essentially, the softer the target material, the deeper the depth of compression at a given intensity. For example, at a 15A Almen intensity on a $52R_c$ material, the depth of compression will be about .008". Additional curves for various materials are available in detail in Ref. 5.

The purpose of introducing compressive stresses into a part is to prevent fatigue failures, which are typically propagated through a component in regions of tensile stress. Changing tensile stresses to compressive stresses at the surface of a component where fatigue cracks typically occur limits their propagation.

Residual tensile stresses can decrease the fatigue life of a component. Compressive stresses, however, tend to increase fatigue life. Some machining processes will introduce unwanted tensile stresses into a part prior to any applied loading. If these factors are not taken into consideration, premature component failure can occur.

Shot Peening Controls

Certain basic controls must be introduced into any in-house company specification on shot peening. *Specifications, such as AMS 2430* and Military Specification MILS-13165-B, deal extensively with these. Since the intent of this article is to better enable the design engineer to properly specify other areas, we will touch briefly on the controls. The reader should consult reference sources which more thoroughly explain these points.

To assure proper shot peening the engineer must: A) determine the intensity; B) maintain and control the integrity of the shot; C) assure that coverage is complete; and D) ascertain whether computer-controlled equipment or automated equipment will be used. Without proper shot peening controls, repeatability and desired product reliability will not be maintained. The process will then degenerate into nothing more than a blasting operation, as used in cleaning, potentially leading to severe damage to the fatigue properties of a part.

INTENSITY - Intensity is determined by application of a shot stream to a metal strip known as an Almen strip. Three gauges of these strips exist: The "N" strip, the "A" strip, and the "C" strip. The "N" strip is used for light-intensity peening, the "C" strip for high-intensity peening, and the "A" strip for medium-range peening. The proper strip is selected, mounted in an Almen block, and a shot stream is applied to the exposed surface. After proper exposure time, the strip is removed from the block (Fig. 4a). The strip deflects upward toward the peened surface (Fig. 4b), and the arc height is measured by the use of an Almen gauge (Fig. 4c). The arc height of the strip and the amount of time the strip was exposed to the shot stream are noted.

Additional strips of the same type are then exposed to shot streams for increasingly longer periods of time. The information from all of these strips is then used to plot a saturation curve (Fig. 5). The saturation curve assures that the equipment setup has been properly made to assures repeatability of the desired intensity.

SHOT INTEGRITY - Numerous controls go

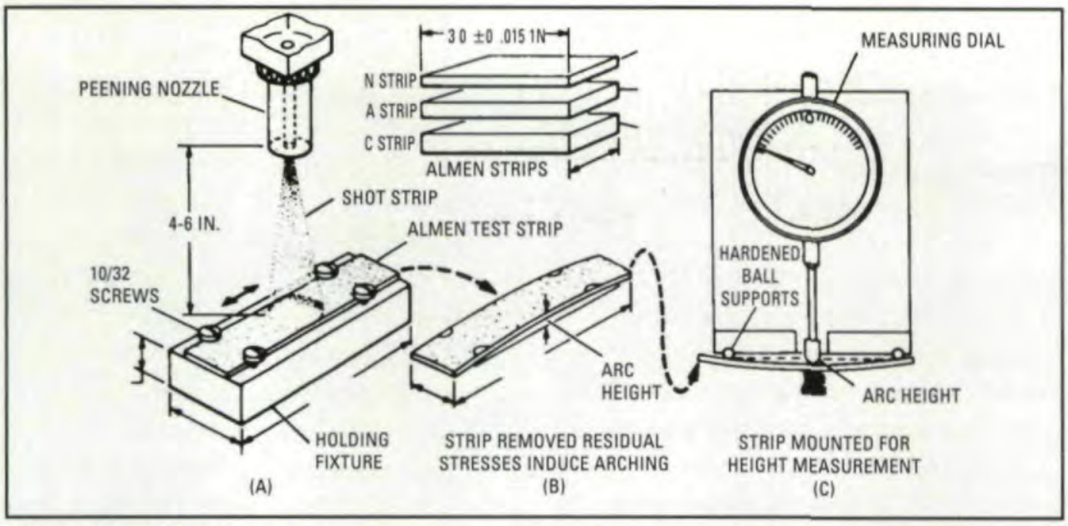

Fig. 4 - The Almen strip system.

into this aspect of the peening process. The primary purpose of maintaining proper shot control is to prevent degeneration of the round shot into broken shot that would typically be used in a blasting operation. Left in an improperly controlled state, shot of the latter type could produce unacceptable surfaces, as shown in Fig. 6a. Properly peened surfaces produced by controlled shot should appear as shown in Fig. 6b. The primary equipment used to assure shot integrity is a classifier, which not only segregates improperly sized shot from good shot, but also segregates irregularly shaped shot from the desired round peening media.

In addition to use of the classifier, techniques to qualify the shot prior to use should include methods to determine porosity and means to determine breakdown of shot, as well as a method to confirm proper shot hardness and metallurgy. To neglect this aspect of controls could hasten degeneration of the process into blasting rather than peening. This would be analogous to striking the surface of the material with the claw end of a hammer rather than the ball end, which should be used for peening.

COVERAGE - A properly peened surface should have many overlapping dimples, referred to as an "orange skin" or "orange peel" effect. (See Fig. 7b.) Fig. 7a represents a partially covered surface and should never been seen. Proper coverage can be determined by the use of a ten power (10x) magnifying glass or by the Peenscan® process. The Peenscan process is a method of viewing coverage of a surface by ultraviolet light after it has been treated with a material similar to a dye penetrant, which is

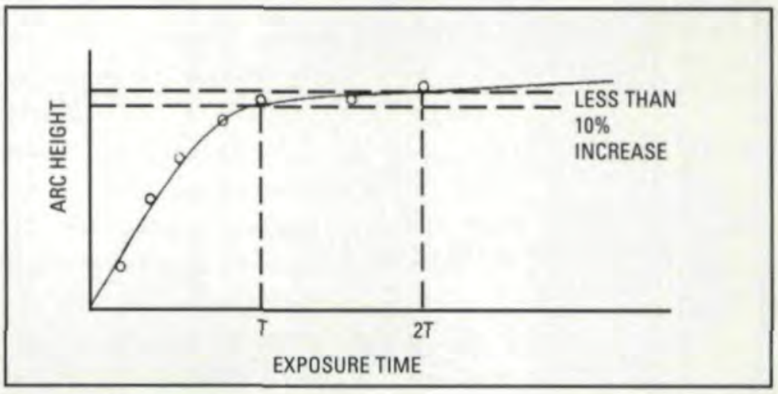

Fig. 5 - Saturation curve.

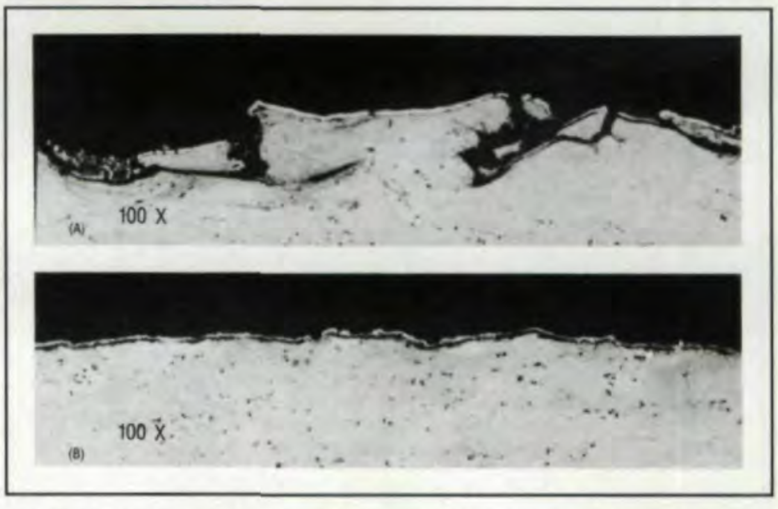

Fig. 6 - A) Surface damage caused by poor shot control. B) Acceptable surface accomplished through good shot control.

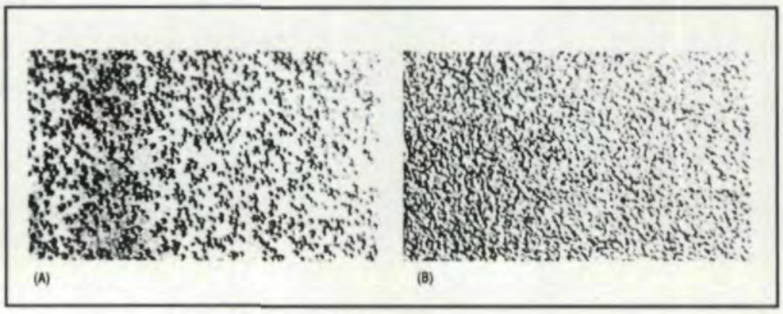

Fig. 7 - Shot peening coverage. A) Partial. B) Full.

removed by a peening operation. Areas that have not been peened properly will glow under a black light.

EQUIPMENT TO BE USED - The engineer must determine whether the equipment to be used is computer-controlled or automated without computer control. Computer-controlled equipment will typically be used for more sophisticated parts and where repeatability and computer printouts for the monitoring of process variables are required. This is the most sophisticated (and usually most expensive) peening method. A sample of a software path flow diagram is shown in Fig. 8. Primary monitoring points are shown on the left-hand column.

Automated machinery without computer control typically employs manual load and unload of equipment. The machine will automatically peen a part for a set cycle without computer monitoring or operator involvement. The majority of parts are peened in this manner.

Considerations for a Shot Peening Specification

Now that we have briefly discussed the theory of shot peening and the necessity for good controls to assure repeatability, the following considerations should be applied to any gearing. These items should be reviewed in any general specification before any shot peening specification is made. They include, but are not limited to, the following:

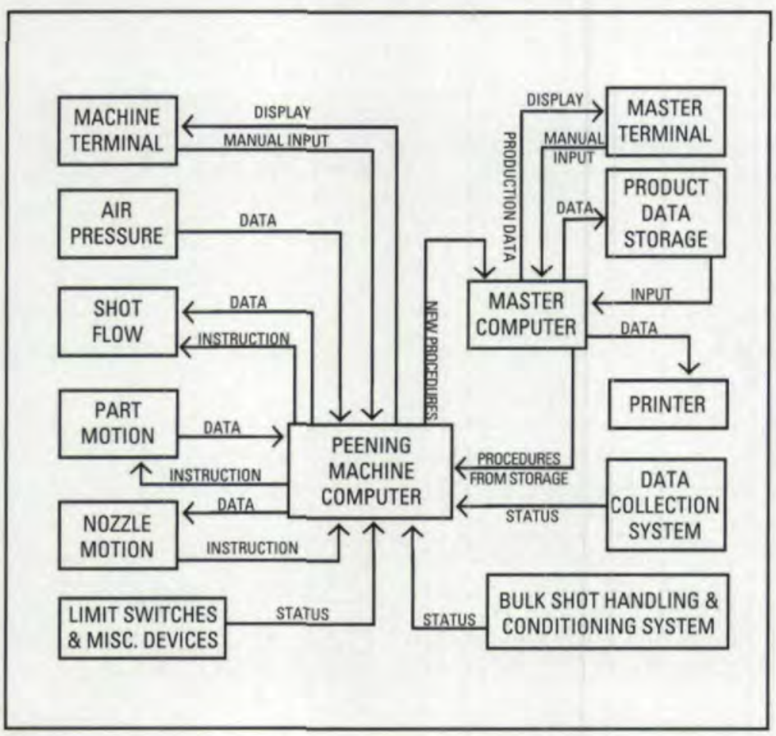

Fig. 8 - Software path flow diagram.

- Application
- Geometric configuration of part
- Material hardness and heat treatment
- Material
- Surface finish requirements before and after shot peening
- Optional peening methods and additional considerations, which might include the use of:
  - Strain peening
  - Dual intensity peening
  - Plating and salvage methods
  - Contour correction (forming) peening
  - Increasing wear due to work hardening
  - Porosity (closure in powdered metal parts and castings)
  - Salvage/Grinding - before and after
  - Stress corrosion cracking.

APPLICATIONS - The primary consideration in shot peening gears is to determine if the process is to be used to: A) increase bending fatigue strength of gear teeth; B) increase surface fatigue life; or C) change the texture to either break up continuous machining marks or to aid in lubrication of the gear face.

Numerous variables enter into how the gear's ultimate fatigue strength will be determined. Fig. 9 shows the variety of possibilities. We will specifically address the effect that the residual compressive stress has on the fatigue strength, along with the effect on hardness and microstructure.

As noted by Dudley and Seabrook,[9] shot peening is beneficial, and the fillets at the gear root should be peened. The authors show no hesitation in recommending the practice of shot peening carburized and hardened teeth, despite the high hardness and brittleness. Typically, 20-30% additional load-carrying capability is anticipated if root fillets are peened. Similar results were noted on through-hardened and induction-hardened gearing. Gears are probably the second-most commonly peened item in this country, so further discussion on this point is not necessary. (See Refs. 10-12).

It should also be determined if surface pitting found at the pitch line is the primary fatigue concern, or whether fatigue at the root because of the tooth flexure is primary. Tests by NASA on the effect of surface fatigue life of carburized and hardened spur gears[13] exhibited a 60% increase in life of the gears when shot peening was used to combat this phenomenon.

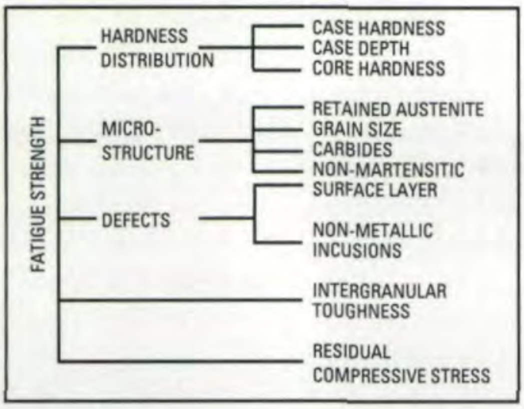

Fig. 9 - Metallurgical factors affecting the fatigue strength of gears.

The question of which fatigue problem is of primary concern is important when making a proper shot size selection. This will be considered further in the following section on geometry.

A third consideration is whether the peening will be used not primarily to introduce beneficial residual compressive stresses, but rather to improve surface finish. The texture produced by the peened surface consists of homogeneous, overlapping dimples which can be used to eliminate stress risers produced by various machining processes, such as hobbing. Typically, this operation is performed in the "green state" of the gear just prior to heat treatment. Proper shot selection is dependent on how disrupted the surface will be. At a given intensity, a larger size shot will produce a smoother finish than small shot.

An additional consideration is whether the dimpling will be used to aid in gear lubrication. This, however, is rarely the primary consideration. A variation of the use of different sizes of shot to produce a texture is to carburize, then slow cool to a hardness higher than the "green state," follow with a texturized shot peening, and then fully harden. Any compressive stresses produced prior to heat treatment will be dissipated in either texturizing case. Shot peening after heat treatment will be required to produce a surface with compressive stresses if either of the two fatigue conditions also need to be considered.

GEOMETRIC CONFIGURATION OF A PART - After the reason for the use of the shot peening has been determined, the next step is to determine the shot size based on the geometry of the part. The general rule used per Military Specification MIL 13165-B states that the maximum shot diameter "d," as shown in Fig. 10, must be equal to no more than 1/2 R (the radius to be peened). For example in Fig. 10a, it is obvious that the shot is too large and will not provide full coverage in the fillet radius.

After determination of the geometry into which the shot will move, the intensity of the shot must be determined. The general guideline is that the depth of compression cannot be greater than 10% of the thickness of the part. Fig. 11 provides an example of a range of thicknesses of steel that can be peened at a given intensity. *The chart also illustrates the range of intensities that can be used for any given thickness.* For example, at a 4A intensity, steel thicknesses from .018" to .15" could be peened. The same graph indicates that a steel part with a cross-section of .150" could be peened with an intensity as low as 4A and as high as 14A. Optimum selection of the correct intensity is a function of the size of the shot to be used, coupled with the hardness of the target material. Curves which provide examples for depth of compression should be used, remembering that the depth of compression cannot exceed 10% of the thickness of a part per peened side, or a total of no more than 20% of the cross section of a component.

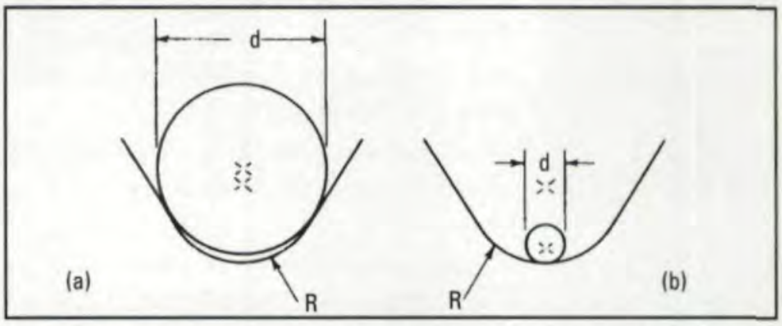

Fig. 10 - Shot diameter d and groove radius R. A) Shot size too large, d = 2.2 R; B) Maximum shot size permitted by Ref. 9, d = 1/2 R.[14]

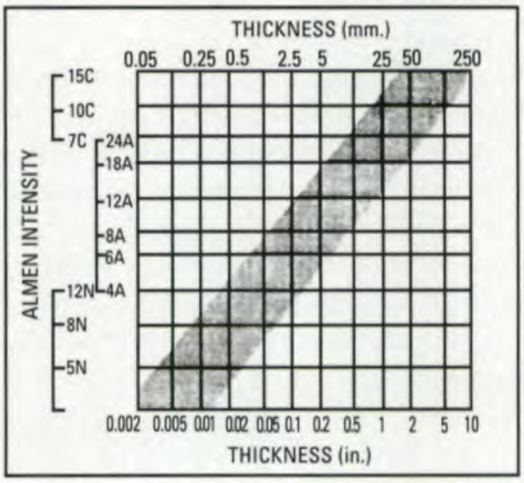

Fig. 11 - Peening intensities commonly used on steel parts of different thickness.[14]

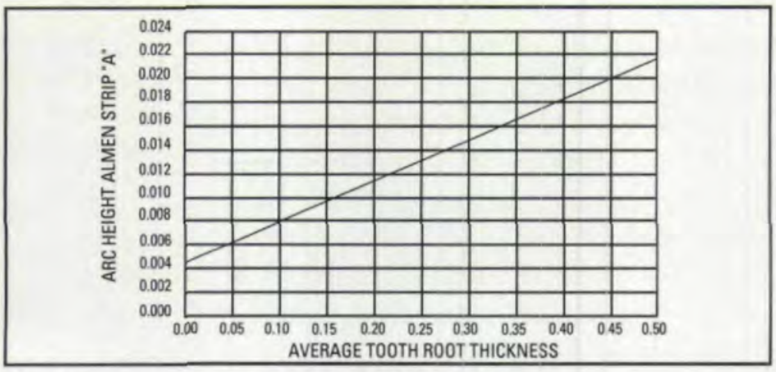

Fig. 12 - Average tooth root thickness.[15]

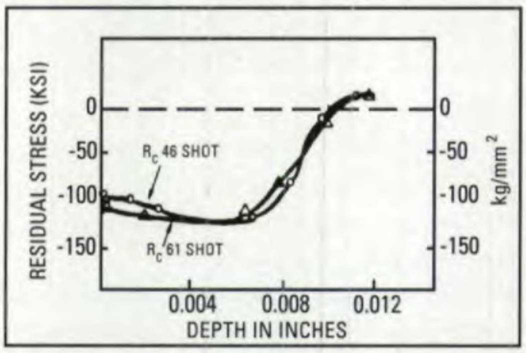

Fig. 13 - Peening 1045 steel at R_c 48.[18]

Fig. 12 shows Almen intensity as a function of average tooth root thickness. Note that it does not take into account various hardnesses of gear teeth, which could have an effect on selecting an optimum peening intensity, but it could be used as a starting guideline for carburized and hardened gears.

Realizing some of the difficulties in selecting an optimum shot intensity for a given gear type and material, AGMA has provided a typical shot size and intensity for shot peening based on diametral pitch as a guide.[16] Henry Fuchs' paper, "Optimum Peening Intensities,"[17] explains in depth a method for selecting optimum peening intensities.

In general, once a shot size is selected based on the geometry of the gear, the maximum intensity should be used. This maximum intensity should provide a depth of compression not exceeding 10% of the cross section of the gear at any point where the shot stream comes into contact with the thinnest cross section of that gear. To do this, proper charts showing depths of compression generated as a function of material type and hardness should be reviewed. (See Fig. 3 or Ref. 5).

MATERIAL HARDNESS AND HEAT TREATMENT METHODS - After the application, shot size, and intensity have been determined based on the part geometry, the next step is to determine if the intensity selected is correct to meet the depth of compression based on the material hardness. As shown in Fig. 2, the higher the ultimate tensile stress, the higher the magnitude of compressive stress. The 50-60% relationship of the compressive stress to the ultimate tensile stress is maintained as long as the shot hardness is equal to or greater than the surface hardness of the gear.

Fig. 13 clearly shows that when the target material hardness closely approximates the shot hardness, no difference occurs in the magnitude of the compressive stress or the depth of compression. However, when the target material hardness is greater than the shot hardness, a significant decrease in the residual compressive stress magnitude (46R_c shot curve at a maximum compressive stress of 100 KSI versus 61R_c shot, providing in excess of 200 KSI) results, as well as a decrease in the depth of compression. (See Fig. 14.) This was confirmed when tests were performed peening high-strength steel using not only 65 HRC shot, but also ceramic shot and 46 HRC cast steel. In Fig. 15, average fatigue life was higher for both ceramic and hard shot than for 46 HRC shot.

The benefits of using hard shot on high hardness gear materials was further demonstrated in a paper by Miwa et al.[8] Further support for the use of high-hardness shot for high-hardness materials even over increasing the intensity of shot peening is provided in Ref. 20. As the hardness of a material increases, so does the ultimate tensile strength of that material. However, as the hardness increases, a noticeable decrease in the fatigue strength in some materials may result because of an increase in notch sensitivity and brittleness, as shown in Fig. 16. For those steel specimens shown at a hardness above 42R_c that have not been shot peened,

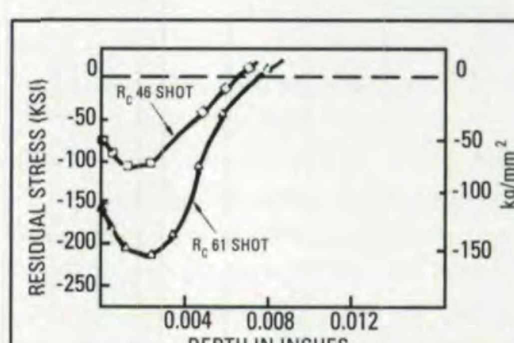

Fig. 14 - Peening 1045 steel at R_c 62 with 330 shot.

fatigue strength decreases as the ultimate tensile strength increases. By changing to peening with hard shot and peening the high-strength steel, not only will a higher ultimate tensile strength result, but the fatigue strength of the material also will be increased.

An additional consideration is whether decarburization may occur in heating the steel. Decarburization is the loss of carbon at the surface of a ferrous material, and it can result in the loss of fatigue strength of high-strength steel. Fig. 17 exhibits the capability of shot peening to restore almost all of the fatigue strength. If decarburization is suspected, incorporating shot peening into a part design can insure component integrity. Essentially, the hardness of the material must be considered to determine the depth of the compressive stress, whether hard shot is to be used, and whether decarburization will be a factor.

MATERIAL CONSIDERATION - The fourth major consideration is to determine if the media and intensity chosen to this point will have any adverse or additional desirable effects on the target material. Representative curves of shot peened material of a similar nature are helpful in determining the depth of compressive stress,[5] but the following factors also must be considered:

Will the peening media selected contaminate the target material? For instance, the use of cast steel shot on an austenitic stainless steel may require chemical passivating or other mechanical cleaning methods. Would it be preferable to use other peening media, such as stainless steel, ceramic, or glass? Is work hardening possible and/or desirable? For instance, austempered ductile iron (ADI) not only responds well to shot peening by increasing fatigue strength, but also has the added benefit of work hardening. Fatigue strength increases for ADI at various peening intensities are shown in Fig. 18. Refs. 23-26 support not only the fatigue benefits for ADI, but also the improved wear characteristics caused by desirable work hardening. Other materials, such as high manganese content steel and austenitic stainless steels, will also readily work harden.

Does the material have a tendency towards different microstructures, such as retained austenite? If the retained austenite is excessive, significantly reduced compressive stress magnitudes will be noted unless hard shot is utilized.[8]

Will the material respond favorably to the shot size selected, or should an alternate shot size be

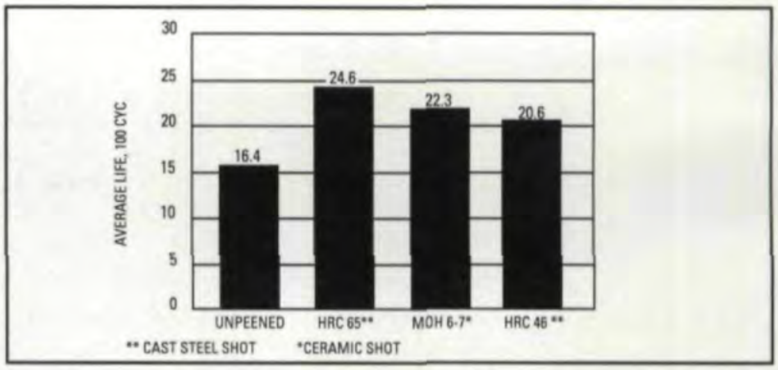

Fig. 15 - Shot peening high strength steel R_c 55. Fatigue life vs. shot hardness.[19]

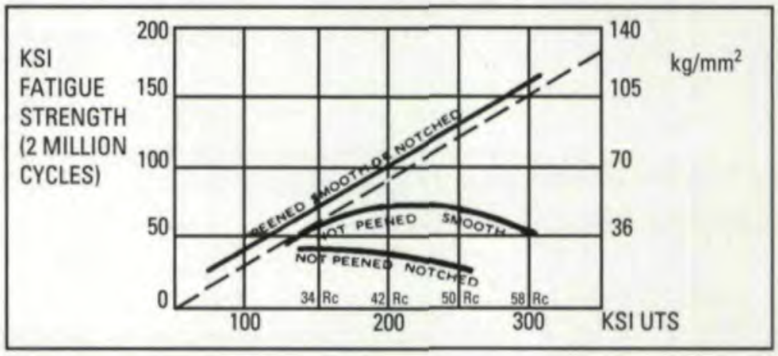

Fig. 16 - Comparison of peened and unpeened fatigue limits for smooth and notched specimens as a functions of ultimate tensile strength of steel.[6, 21]

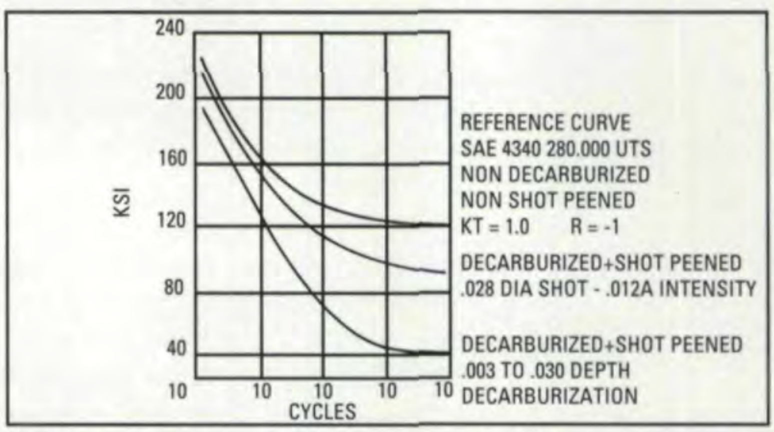

Fig. 17 - Effect of shot peening on decarburization.[22]

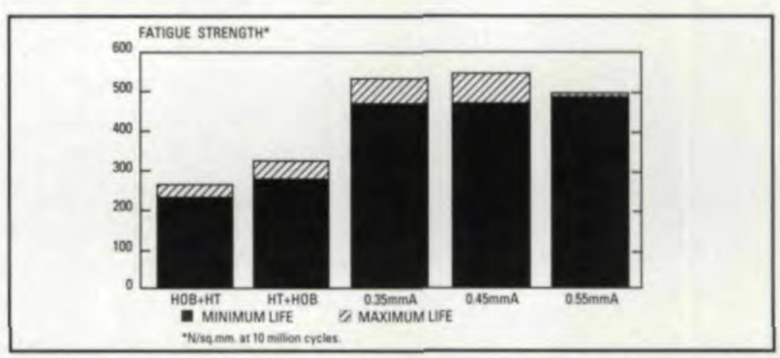

Fig. 18 - The effect of processing variables on the bending-fatigue strength of austenitic-bainitic (K9805) gear teeth: Measured after 107 loading cycles. Heat treatment after hobbing. Hobbing after heat treatment. Shot peening 0.35mm Almen A. Shot peening 0.45mm Almen A. Shot peening 0.55mm Almen A.[23]

selected? For instance, aluminum alloys will respond better at a given intensity to a larger shot size at a low velocity than to small shot at a higher velocity. Though both conditions could produce the same intensity, the larger shot size is more desirable if geometric constraints allow it.

Is the gearing of a powder metallurgy? Special considerations must be made here; however, if handled properly, fatigue life improvement due to increased hardness and residual compressive stresses is possible.[27]

SURFACE FINISH REQUIREMENTS BEFORE AND AFTER SHOT PEENING - Additional consideration should be given to the desired surface finish before and after shot peening. First, note that a shot peened surface's overall dimension will increase slightly because new measurements will be taken at the tops of peaks produced by the dimpling action. This growth is dependent upon hardness of target material as well as the shot size and intensity used, but typical growth rarely exceeds .0005" per side. If this change in size will be detrimental from a standpoint of fit, samples of the material should be peened experimentally before working with actual parts. All typical drawing dimensions should reflect dimensions prior to peening.

As a general guide, original surface finishes above 125 RMS can be improved by peening, whereas surfaces below 125 RMS will typically be increased in surface roughness, depending on material type, material hardness, shot size, and intensity used. Samples should be provided to confirm desired results.

Selective peening can be performed so that seal and bearing surfaces, along with other critical surface finishes, can be protected. These should be noted accurately on all drawings. In addition, any cleanliness requirements should be shown to properly protect these areas during peening.

If a surface finish in a peened area is required which will be finer than that produced by shot peening, certain machining processes may be performed after the peening. Cool processes, such as lapping and honing, are allowable, as they do not generate much heat and will not dissipate compressive stresses; however, material removal is limited to no more than 10% of the depth of compression. Additional material removal will adversely affect all benefits of the peening. ■

Part II of this article will run in our next issue. It will discuss Optional Shot Peening Methods and Specifying Shot Peening.

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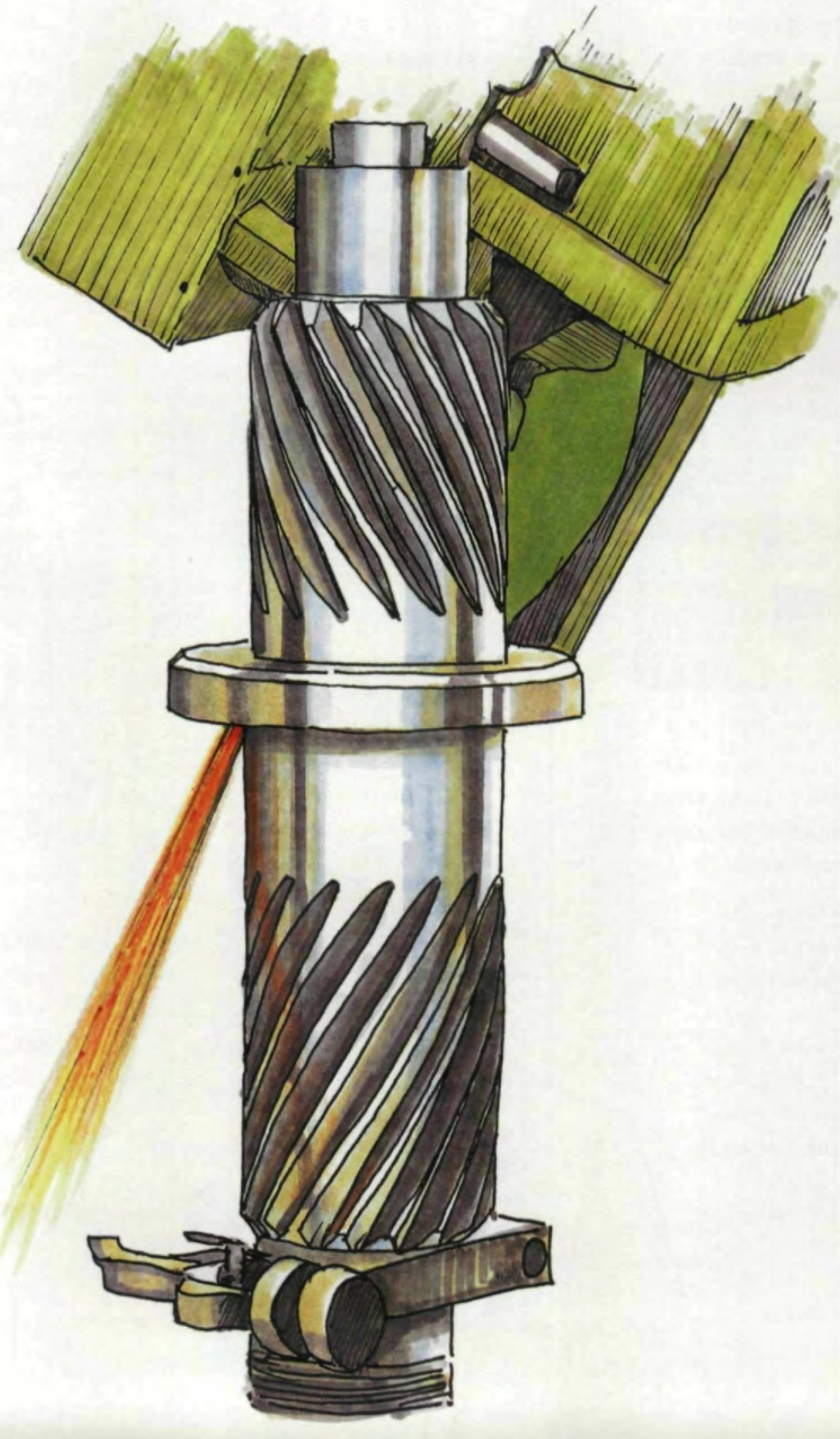

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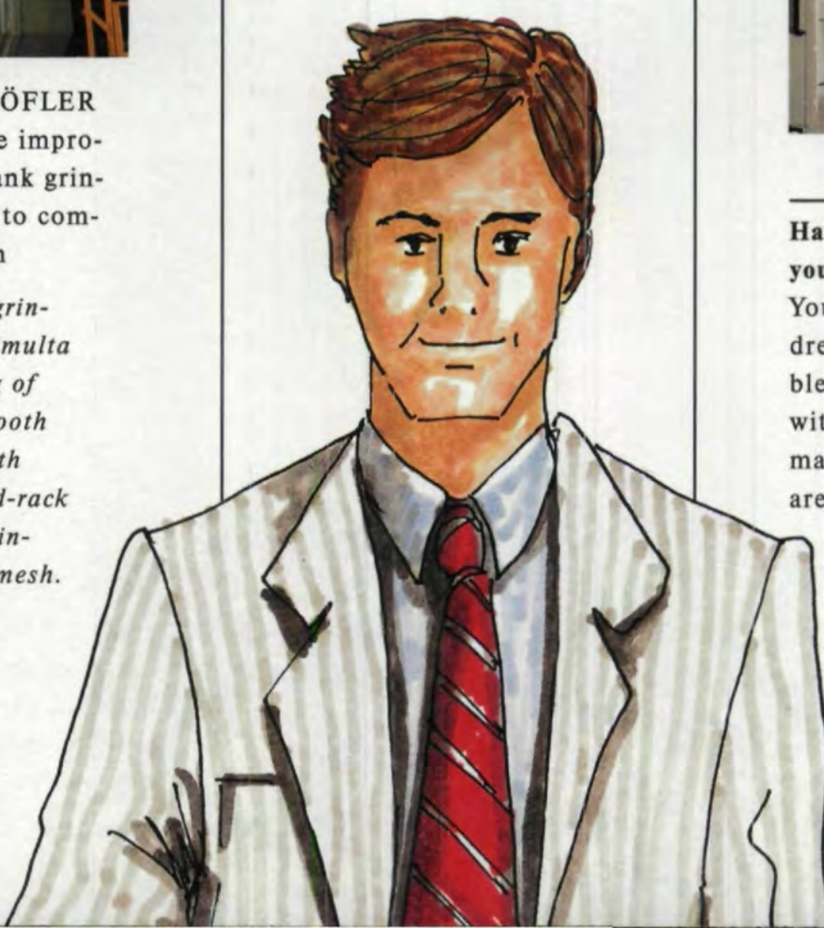

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Recent Developments in Gear Metrology

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Summary: Metrology is a vital component of gear manufacturing. Recent changes in this area, due in large part to the advent of computers, are highlighted in this article by comparison with more traditional methods.

Introduction

Various deviations from the true form of circular gears limit their ability to transmit uniform angular velocity at their designed speed and power. These deviations or errors are introduced during the production of the gear elements. The ability to measure these variations enables their magnitude to be determined and controlled during manufacture.

The various possible errors occurring in circular gearing are:

- Tooth Thickness (Backlash)
- Run-out Errors
- Pitch Errors
- Tooth Form Errors
  - profile
  - alignment
  - surface finish.

The types of circular gears in common use which can exhibit some or all of the above errors are involute spur and helical gears and bevel gears, both straight helical and spiral bevel.

This article will concentrate on involute gearing; however, some comments on bevel gears will be made.

The ability to measure the errors present in the various cutting tools, such as hobs, shaping cutters, and shaving cutters, as well as the gears themselves, is also useful.

Traditional/Classical Instruments

In order to highlight the advantages of the latest developments in gear metrology, a brief summary of the traditional/classical instruments used to measure the various gear errors will be presented first. The reader is referred to the various texts and papers listed in the bibliography for more detailed information.

Tooth Thickness. Three main methods have been prevalent in the measurement of tooth thickness of involute gears.

The first method involves the use of an instrument called a gear tooth caliper (Fig. 1). It consists of a depth gauge, which is used to set a known distance down from the outside diameter of the gear, and a caliper to measure the tooth

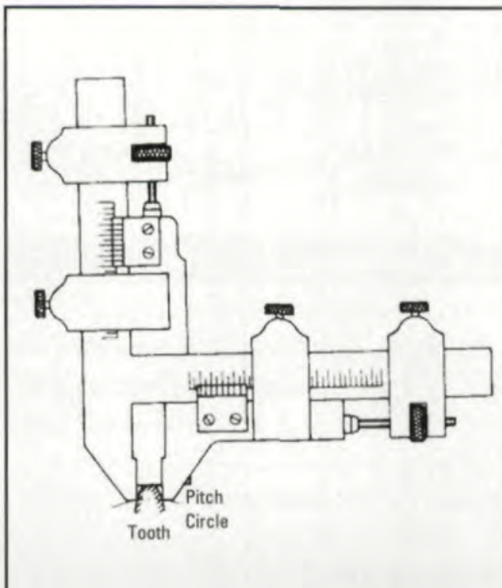

Fig. 1 - Gear tooth caliper

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thickness at this depth. Calipers are available for modules 1 mm to 50 mm.

This tool has the following disadvantages:

a) It relies on an easily worn, very sharp edge at the end of the caliper jaws to accurately measure the tooth thickness.

b) It relies on the outside diameter of the gear for its datum.

c) It relies on operator skill.

d) It is limited to about .001"- .002" accuracy.

This method's advantage is that it is inexpensive.

The second method uses the flange micrometer and is called the base tangent method. This instrument takes span measurements across a number of teeth (Fig. 2), which are then related to the actual tooth thickness by the involute geometry of the gear. Micrometers can measure modules from 0.5 mm to 11 mm.

The main disadvantages of this method are:

a) Pitch errors can influence the result.

b) Profile errors can influence the result.

c) Tip or root relief may make a valid measurement impossible.

d) It relies on operator skill.

The main advantages are:

a) It is more accurate than the gear tooth caliper.

b) It is not influenced by variations in the outside diameter of the gear.

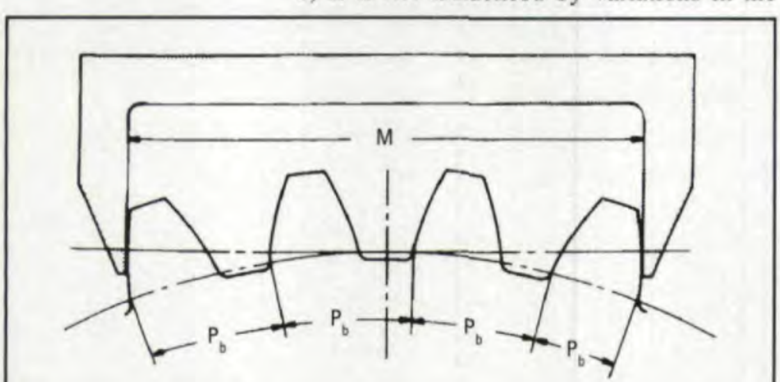

Fig. 2 - Span measurement of tooth thickness

The third method involves "measurement over rollers", where two rollers are placed on opposite sides of the gear, and the distance across these rollers is measured (Fig. 3). This value can be related to tooth thickness by the known geometry of involute gearing.

This method has the following disadvantages:

a) It is limited to gears small enough to be spanned by available micrometers and the available sizes of rollers.

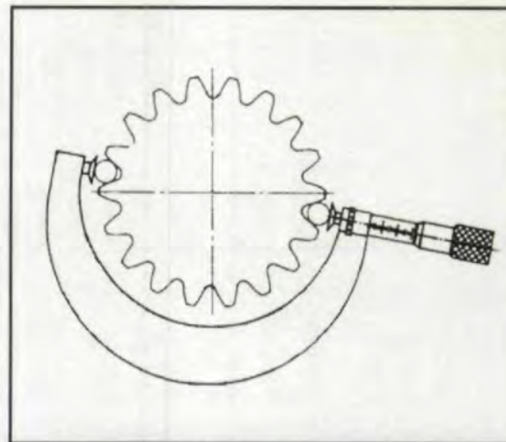

Fig. 3 - Tooth thickness measurement over rollers

b) The measurements are affected by errors in tooth spacing and profile.

The method's advantage is that it is not influenced by variation in the outside diameter of the gear.

All of the above methods are manual and require skill and dexterity. As such, they are not suitable for the measurement of all the teeth on gears with large numbers of teeth or on gears made in large quantities. The methods are most suitable for spot checks in small batch production situations.

Other methods available are:

1) Measurement of center distance at tight mesh. (See section on Dual Flank Rolling.)

2) Measurement of backlash at operating center distance. This method requires the ability to accurately assemble the gears at their nominal center distance with axes parallel.

Run-Out Errors. Radial run-out errors are defined as "the total range of reading of a fixed indicator, with the contact point applied to a surface rotated without axial movement, about a fixed axis, and measured perpendicular to the axis of rotation." The eccentricity is then also available, as it is half this radial run-out.

This type of error traditionally has been measured by inserting a ball (or cylinder) into each tooth space, and measuring the radial variation in distance from the center of rotation to the ball (See Fig. 4.).

Radial run-out measurements made in this way also reflect variations in tooth thickness. When radial run-out readings are plotted, the influence of each factor can be seen. Run-out errors are seen as a smooth, sinusoidal variation, while tooth thickness variations are seen as

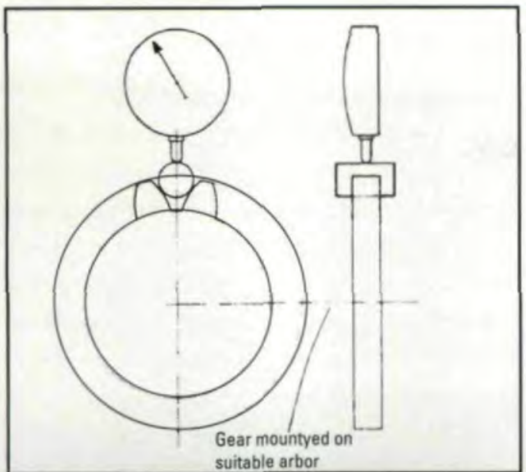

Fig. 4 - Measurement of radial run-out

fluctuations about a smooth curve drawn through all readings.

There are also automatic testers available which measure this error on a continuously rotating gear and plot the resulting error graph. These testers are available for module ranges of 1 mm to 40 mm and 0.1 mm to 20 mm respectively.

The double-flank rolling test can also be used, since in this test the run-out error is evident as a sinusoidal shaped variation with a period equal to one revolution of the gear. (See section on Dual Flank Rolling Test.)

Pitch Errors. The three types of pitch errors on spur and helical gears are pitch variation (deviation), fp; tooth-to-tooth pitch error, fu; and cumulative pitch error, Fp.

The classic method of measuring the cumulative pitch error is the index method, in which the gear is mounted on an angular dividing head (Fig. 5). The gear is then rotated through the nominal angular pitch, and deviations of each tooth relative to the first are noted and plotted.

The next more common pitch error measured is the tooth-to-tooth pitch error. This error is the change in pitch between successive pairs of teeth. As such, any instrument which can measure the change in pitch from one pair of teeth to the next can measure this error. A hand-held instrument that can do this is the Maag pitch measuring instrument type TIC. These instruments were available for normal modules ranges of approximately 2 mm to 10 mm. However, being hand-held and operated, this becomes a very tedious test on gears with more than a limited number of teeth. Automatic versions of this tester have been available for at least 20 years, and these testers measure the tooth-to-tooth pitch errors on continuously rotating gears

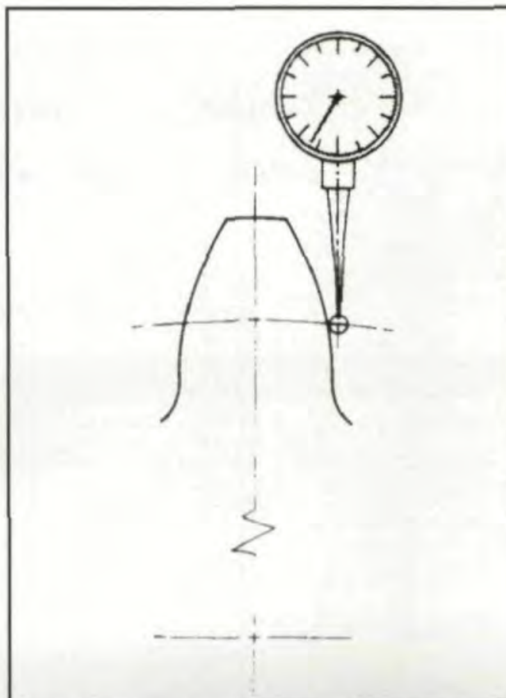
Fig. 5 - The index method of finding pitch error

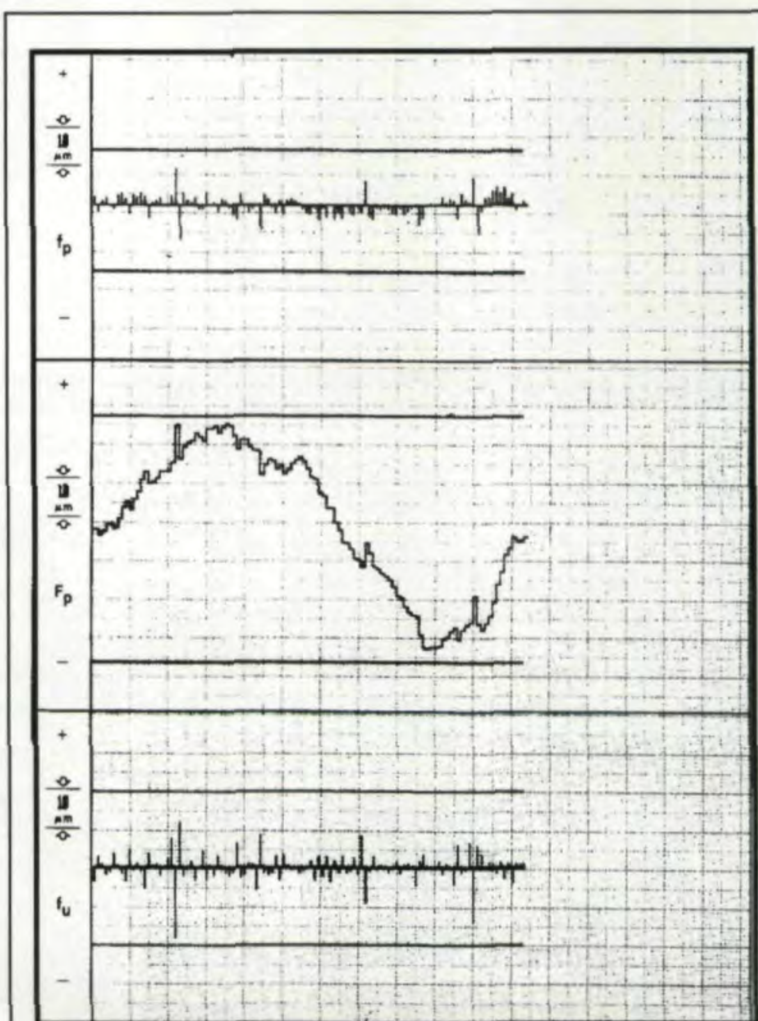

Fig. 6 - Printout of pitch errors

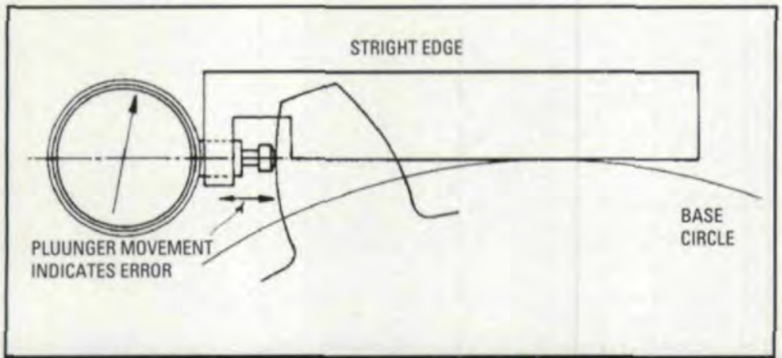

Fig. 7 - Measuring profile errors using base circles

Fig. 8 - Lead and profile measuring machine

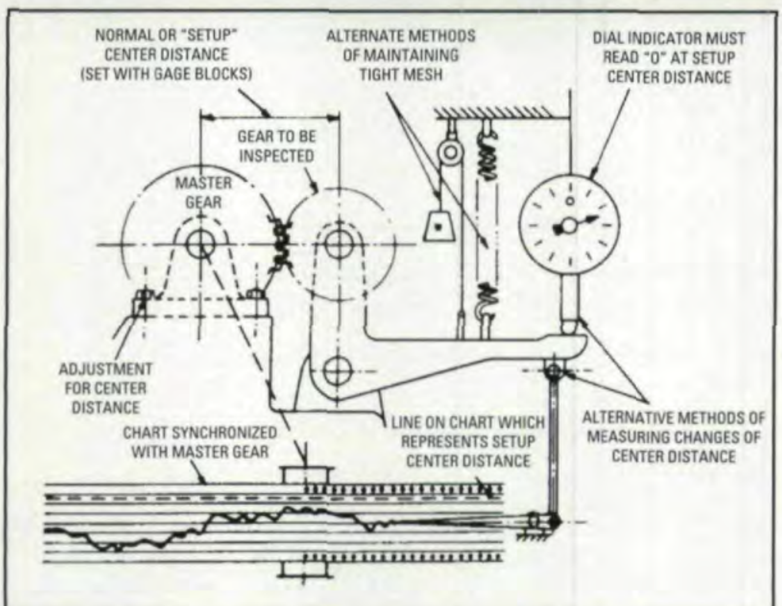

Fig. 9 - Dual flank rolling test

by inserting two probes to measure the tooth-to-tooth pitch error for that pair, and then retracting the probes until the next pair comes into position.

Pitch variation errors require measurement of the absolute pitch between two teeth and comparison with the nominal pitch. The Maag pitch measuring instrument mentioned above is capable of doing this test as well as the tooth-to-tooth test. It must, however, be set to the nominal pitch using gauge blocks or a setting master.

All of the above errors can be related mathematically (Ref. 8). This fact, coupled with recent advances in computer technology, has allowed the automatic pitch testers described here to measure the tooth-to-tooth pitch error, calculate the other two pitch errors, and plot all three for a complete picture of the pitch errors of the gear under test. (See Fig. 6).

The tester must understand that pitch errors and run-out errors are inseparable when present. If run-out errors are either introduced or removed during the test, the pitch errors measured will not give a true representation of the gear's actual performance in its final location. As the hand-held pitch measuring instruments do not refer their measurements to an axis of rotation, the resulting pitch errors are not affected by measurement-process-induced run-out errors. However, run-out errors present during manufacture will be evident as pitch errors on the gear. Pitch testers, in which the test piece is mounted and rotated about an axis, will be affected by run-out errors introduced or removed in setting up the test.

Profile Errors. Actual measurement, as distinct from contact bearing patterns of profile error, has until recently been limited to involute gears. Bevel gears, especially of spiral form, have not been measurable from first principles.

Because of the definition of the involute, profile checking machines have been available for many years. The early testers relied on the use of base circle discs for each gear to be checked (Fig. 7). Later machines overcame this limitation by the use of various mechanisms to either bridge the gap between a limited number of base circle discs or to eliminate them completely (Fig. 8). These testers all generate the true involute on a probe which is in contact with the gear tooth under test. Deviations in involute profile are detected by the probe and recorded for later analysis.

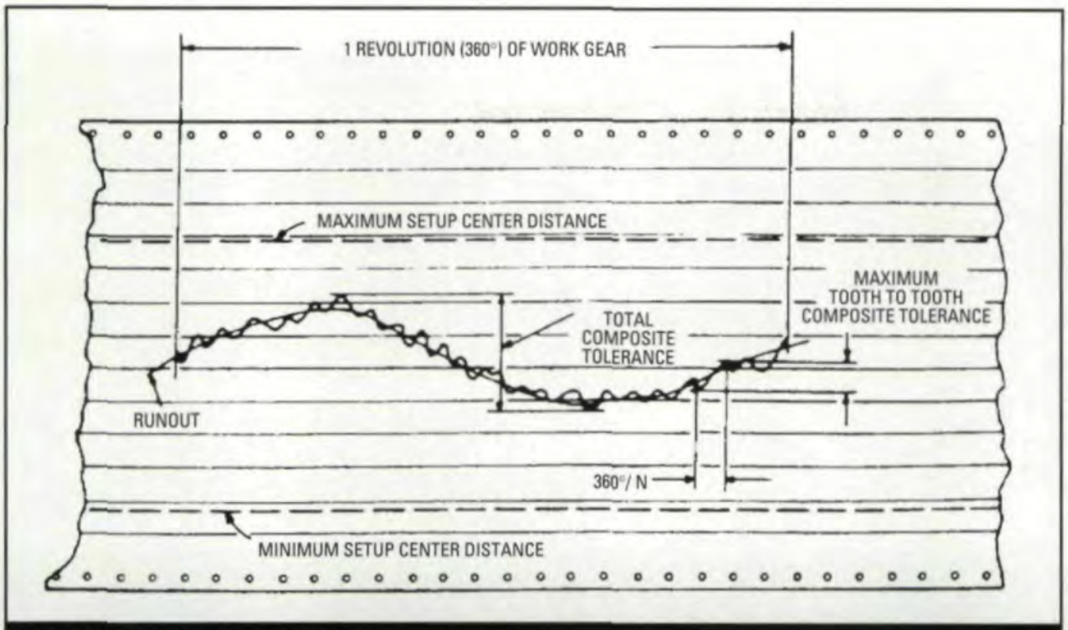

Fig. 10 - Dual flank rolling test results

Advances in computer technology have enabled the recording of the involute errors in digital form. This in turn allows computer analysis, which eliminates the weakest link in the measurement process: the human interpretation of the error curve. The use of mathematically rigorous techniques for this analysis can give an unbiased and consistent interpretation of the results.

The accuracy of these machines can be determined by well-understood tests on their geometry and construction. Because the software does not control the accuracy of generation of the involute on the probe tip, it is only necessary to test the software against known masters to verify its accuracy.

Alignment Errors. Alignment errors are often referred to as lead and/or helix errors. These errors have also been measured for many years on involute testers, modified with the necessary mechanisms to generate the nominal lead motion on the probe. The probe then detects variations in the same way as for the involute errors. Error analysis has been enhanced by the use of computer technology in exactly the same way as for involute errors.

Surface Finish. This feature has been measured by the use of surface finish testers. In some cases, these testers have been fitted to involute/lead testing machines to allow easier testing of gear tooth surface finish.

Meshing Tests. Meshing tests are functional tests to determine the actual performances of a gear or gear pair. These tests fall into the sections on dual and single flank rolling tests.

Dual Flank Rolling Test. In this test, the gears are brought into tight mesh so that both flanks are in contact. (Fig. 9) One gear, usually a master gear, is mounted on a fixed shaft. The other gear is mounted on a shaft constrained to move in a radial direction. As the gears are rotated, the variations in center distance are measured and often recorded.

This dual flank rolling error or composite error is then a combination of all the primary errors already discussed above. A plot of this error normally has two major components (Fig.10). The low frequency component is most commonly associated with run-out errors, while the high frequency component is associated with pitch errors. Because a composite error is generated, this test is only useful as a go/no go test. It is not good for identifying individual errors, and subsequently, rectifying a particular process.

Single Flank Rolling Test. In this test, the gear pair are positioned at their nominal center distance. The relative angular motions of the two gears are then measured. For a perfect gear pair, the motion of the output gear would be uniform relative to that of the input gear. Any errors would be reflected as a non-uniformity in angular motion on the output gear.

Because the gears are mounted at their nominal center distance, only one flank will be in contact. Also one gear needs to be driven while the other is braked. This test, therefore, comes

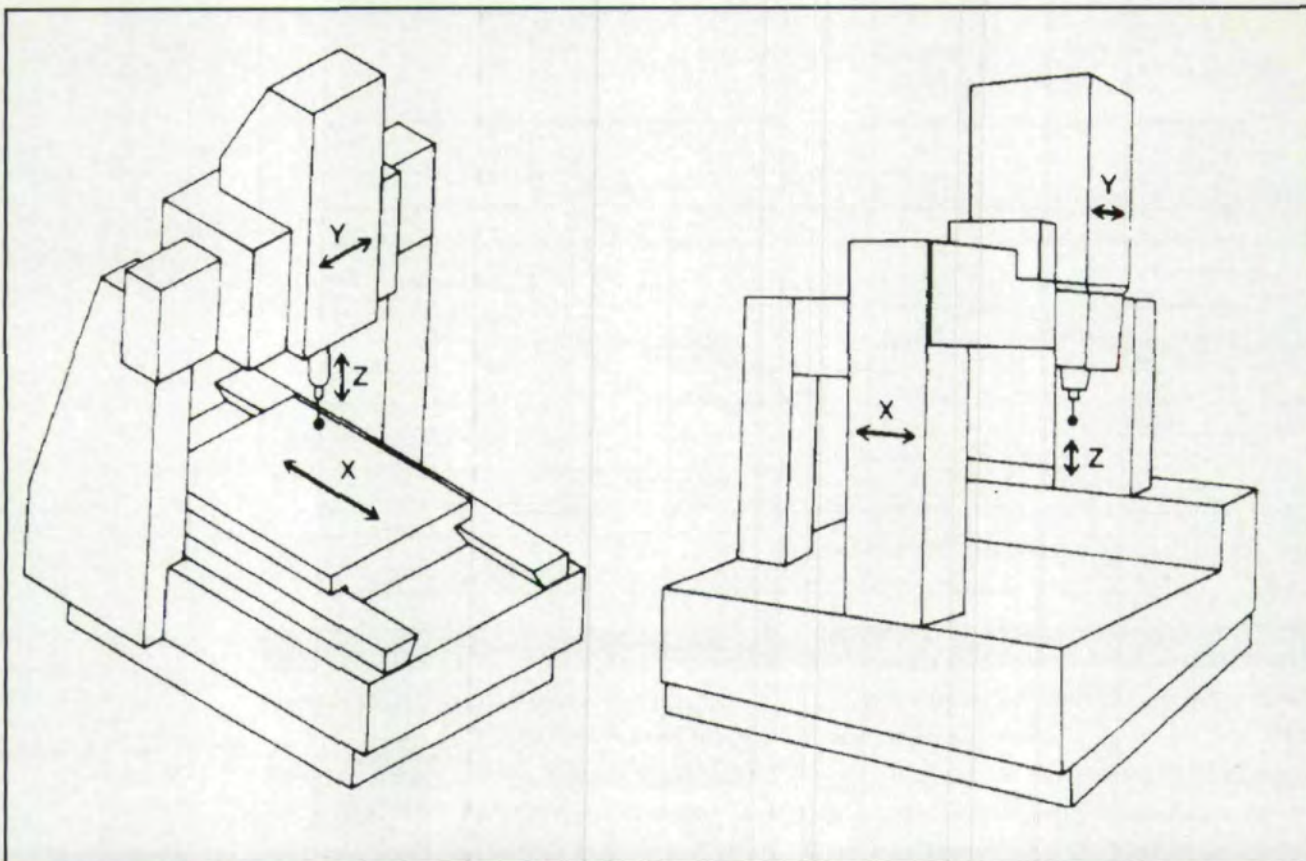

Fig. 11 - General purpose three-dimensional coordinate measuring machine

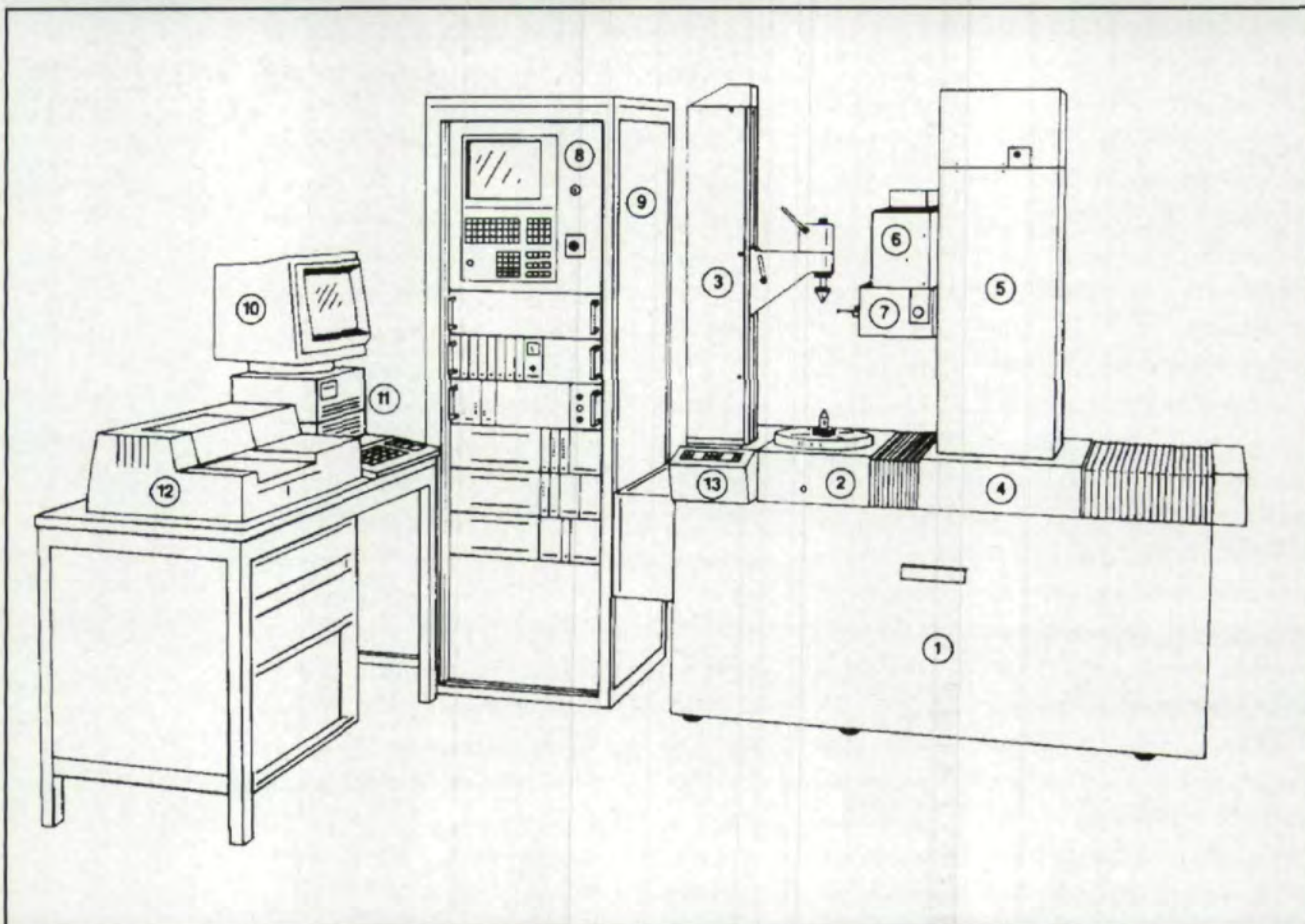

Fig. 12 - Gear measuring machine

very close to the actual operating conditions of the gear under test.

In the past, the difficulties involved in comparing the relative rotary motions of two rotating gears has meant that this test, although theoretically desirable, was physically impractical.

New Developments

The major changes in gear metrology have been brought about by advances in electronic/computer technology. The availability of large amounts of computing power in ever cheaper and smaller packages has spawned significant changes in all fields of technology.

The basic aims and principles of gear metrology have not changed. However, the methods by which these can be achieved have changed dramatically. These changes will be presented in the remainder of this article.

CNC 3-D Measuring Machines. The last 5 to 10 years have seen very rapid developments in the versatility of the computer numerically controlled, three-dimensional co-ordinate measuring machine (CNC 3-D CMM). Computers are becoming faster, more accurate, and have larger amounts of affordable memory in the form of volatile, random access memory, (RAM) and permanent (hard) discs. Therefore, the software on which all 3-D CMM's rely has become more and more sophisticated.

In this light, there has emerged two groups of CNC 3-D CMM's. The first is the general purpose machine (Fig. 11), which is primarily designed for the measurement of components that are not necessarily circular in nature. These machines are able to measure gears, but at a cost in speed and efficiency of measurement. This is because these machines have heavy and cumbersome arrangements for moving the measuring probes to allow for many variations in workpiece shape and orientation.

The second class of CMM's has been designed specifically for the measurement of solids of revolution. This allows the designer to tailor the machine to measure this class of workpiece much more quickly and efficiently than the universal type of CMM (Fig. 12). Machines are available to cater to gear diameters from 5 mm up to 2.6 m. They can also be used to measure general types of workpieces, but due to their specific design, this is slower and more difficult than for the general purpose type of CMM. Both of these testers can have 4 axes; three orthogonal and one rotational.

Therefore, a general rule can be stated: if mostly non-gear type components are to be measured with only occasional inspection of gears, a universal type of 3-D CMM is applicable. The reverse holds true if the major amount of work for the 3-D CMM is gear inspection. Also, due to the small tolerances generally present in gear specifications, only the more accurate versions of general purpose 3-D CMM's should be considered for use as gear measuring machines.

The new 3-D gear measuring machines (GMMs) are, by their very nature, computer controlled. They also rely on CNC for controlling the motion of the measuring probe. This type of control allows each machine to measure a number of parameters on the workpiece. This is in contrast to the previous generation of machines, which by their mechanical nature, were only able to measure one or, at most, two parameters. Now it is possible to place a gear on one of these new 3-D GMM's and instruct the tester to measure all of the primary parameters for that gear without further human intervention. This results in major time savings for any given level of thoroughness of gear measurement. The automation of the test sequences is even more useful when coupled to automatic loading facilities in large batch evaluation.

All measurements are processed in digital form, and this immediately allows the storage of these results for comparison with later batches. These statistical process control (SPC) abilities are a major advance, especially for manufacturers of large quantities of gears.

Also, some of these new 3-D CMM's and GMMs can measure other forms of gearing, with the addition of the appropriate software packages. The most notable advance has been the ability to measure spiral bevel gears in such a way that information relation to the actual corrections to the production machine settings are available. This can only lead to better bevel gears with, one hopes, more interchangebility than was available in the past. The ability to measure the condition of the various gear cutting tools (shaping, shaving, and hobs), again via the addition of software modules, also increases the usefulness of the GMM to the owner.

For similar expenditure, the purchaser of the

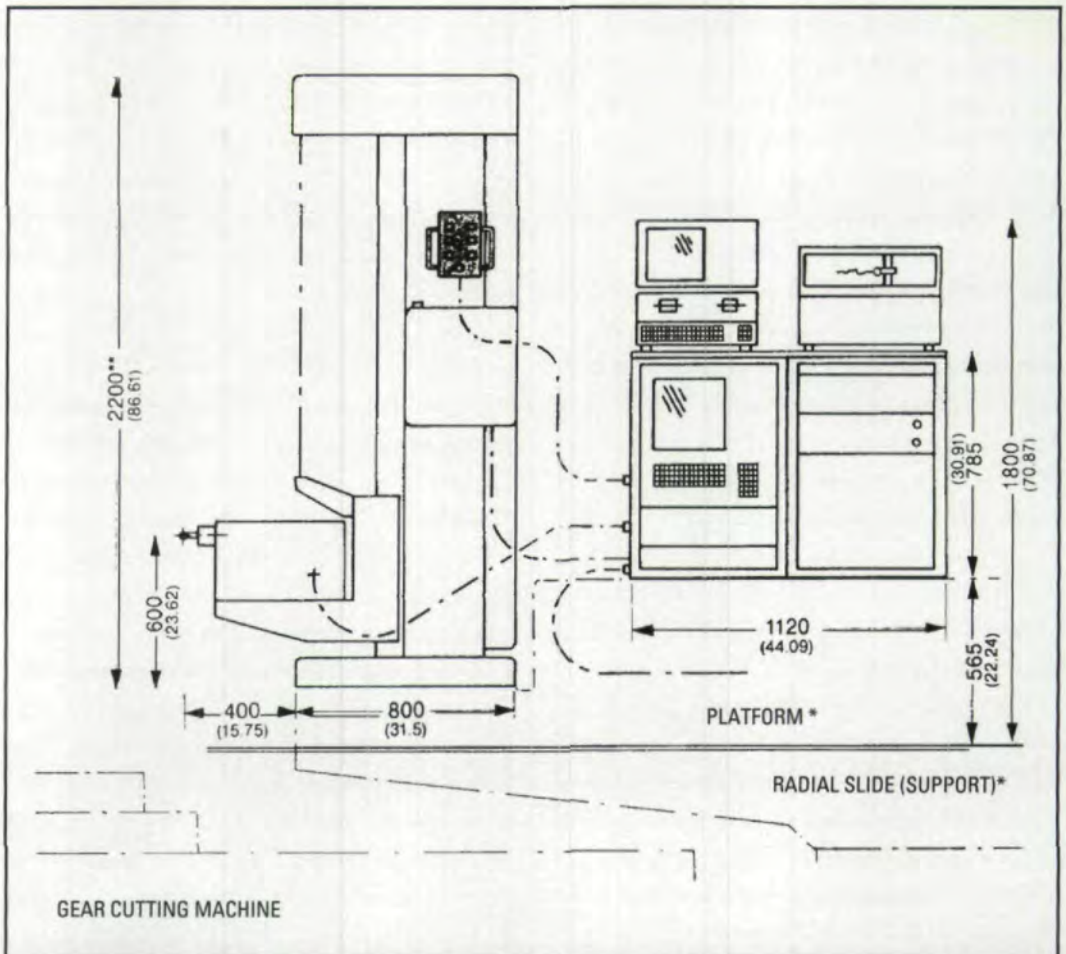

Fig. 13 - Transportable gear measuring machine

modern CNC 3-D GMM, obtains a machine with vastly increased performance and flexibility over the previous mechanical machines.

The verification of the accuracy of these machines requires the measurement of not only the mechanical accuracy of the machines, but also the accuracy of the CNC software control which actually generates the theoretical profile of the gear under test. This is in contrast to the generation-type machines described in the sections on profile and alignment errors, which generate these profiles mechanically. On the new machines, interactions between the CNC control software and the computer evaluation software needs to be understood and thoroughly checked.

Transportable CNC Lead/Involute Testers. Another development closely associated with the development of the fixed CNC controlled 3-D GMM described in the previous section is the transportable CNC lead/involute tester.

The largest fixed 3-D CNC GMM can accept gears up to 2.6 m in diameter. In the heavy engineering field many gears are larger than this maximum, and the measurement of the lead and involute profiles on these gears has previously not been possible.

The transportable gear measuring machine consists of a 3-axis CNC gear measuring machine which is designed to be fitted to the gear production machine tool (Fig. 13). This effectively allows the measurement of any gear diameter capable of being produced with modules in the range of 2 mm to 40 mm.

The transportable gear measuring machine can also measure gears on a large surface plate if necessary. The development of this tester has finally lifted the limit placed on gear metrology by the largest available dedicated testers.

The comments on verification of fixed 3-D GMMs apply equally as well to these testers.

Pitch/Run-Out Testers. As discussed in the section on pitch errors, automatic pitch and run-out testers have been available for at least 20 years. These testers were able to automatically measure the pitch errors on both tooth flanks as well as the run-out error present on a gear. However, this process involved three physical setups and passes around the gear.

The latest development in this area involves the ability of an automatic pitch tester to measure the pitch errors on both flanks, along with the run-out error in one pass of the gear. This results in a time saving on the order of 3, which becomes especially significant on gears with large numbers of teeth.

Rolling Testers. Advances in rotational and vibration transducers as well as in digital signal processing have resulted in improvements in this type of testing. The dual flank rolling test wave form can now be digitally analyzed to separate out some of the component errors, such as the detection and location of nicks, run-out, and tooth-thickness average.

The single flank rolling test, previously not effective because of signal processing limitations, has now become viable. Testers are now available which can detect nicks and the likelihood of gear whine, as well as measure the errors in angular velocity of the transmitted motion.

The angular velocity errors are further analyzed in the frequency domain to separate out the component frequencies contributing to the overall transmission errors. Further analysis to associate angular velocity variations with the number of teeth present allows each individual error component to be analyzed as a function of rotational speed. This gives powerful knowledge to the gear engineer, who can then use this information to eliminate the causes of gear noise and vibration.

Conclusions

Although the aims of gear metrology have not changed, the equipment and methods now available to the gear metrologist have given him/her very powerful tools with which to measure gears.

The new gear metrology machines can measure many, if not all, the parameters of a gear much more quickly and with more confidence than ever before. Measurements previously considered too difficult for routine application can new be performed easily.

Computers have played a major part in making these advances possible. They have made the control of CNC 3-D GMMs and CMMs possible, as well as the collection and storage of large amounts of data. Their unbiased assessment of this data, together with the clear presentation of the results, has given much faster and wider access to the results of gear metrology. This access has enabled much more effective action to be taken to minimize the errors found.

However, these advances should be tempered with caution, as the new machines are essentially only as good as the software driving them and the people operating them. It is very easy to have blind faith in impressively presented results, but a wary eye must be kept out for hidden sources of error. ■

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(Continued from page 7)

We know marketing works for other products and companies. When we hear a name like Mercedes Benz, Craftsman, or Tiffany's, we know we're dealing with the top of the line. For example, even if we don't know anything about fine jewelry, the name Tiffany reassures us that we're dealing with not just a reputable business, but one with a long-standing history of quality and customer service.

Name recognition like that doesn't come easily. A company has to have the products and services to back it up. Mercedes Benz does build some of the best cars in the world; Craftsman tools bought 50 years ago through the catalog are still working just fine, and if one breaks, you can still return it to Sears for a new one, no questions asked.

PUBLISHER'S PAGE

If you're not committed to developing a favorable image for your company, who will be?

But once you have earned such recognition, it's worth its weight in gold. Joe's Jewelry may be selling the same diamonds as Tiffany's - crystallized carbon is crystallized carbon, after all - but Tiffany's has a reputation. Buyers get more than just another ring in that pale blue Tiffany's box. They get a sense of confidence and well-being about the purchase on which it is hard to put a dollar value. Such a reputation is the result of years of careful planning and performance based on the fine-tuned combination of disciplines called marketing.

I don't think marketing is a cure-all for what ails the gear industry - or any other business, for that matter. Any image has to have something substantial to back it up. The basics of business are still building the best product you know how to build at the most competitive price.

But your company's reputation and image are important too. These are assets every bit as valuable as your factories, your machines, your employees, and your expertise, and you should work as hard at making the most of them as you do all the others. Perhaps some of our disappointing sales figures are the result of thinking like Joe at the jewelry store. We too have let the other guy develop the reputation of being THE place to go to get the product.

In today's competitive climate, we have to use every asset at our disposal to survive, much less get ahead. Maybe now we need to overcome some of our aversion to the "soft" science of marketing and learn to capitalize on our names and our images.

Teaching your customers to think of your product as the equivalent of Craftsman or Mercedes Benz won't solve all your problems. Living up to a reputation like that is also a challenge. But it is one of the many challenges you have to meet if you're going to survive in today's business climate.

Michael Goldstein -

Michael Goldstein
Publisher/Editor-in-Chief

Just 2 of the *many* reasons to choose Mitsubishi Gear Cutting Machine Tools

#1 A small shop with big results

"We bought our Mitsubishi GB15 Gear Hobber based on the additional jobs it would allow us to handle. For example, we were recently able to take on a job for a transmission company -- a 10 key spline, 3-1/2" in diameter, 7- to 8-inches in length. The lead tolerance was 0.001" along the length of the spline. We've been meeting the specs and the deadlines--*no* problem.

"But in addition to the *new* jobs, it's also outperforming our conventional hobber at least 2-to-1! We get better turn-around and better overall quality. The solid construction and 5-axis controls make it one of the best machine investments we've got. Ever since we installed the new machine, we've been getting a steady stream of new work, and everybody's happy."

--Plant Manager at a small precision gear producer

#2 A large company with no room for error

"We manufacture high precision electric motors and require tolerances of AGMA 8 and AGMA 10. We started working with the Mitsubishi GB10 Gear Hobber based on our earlier successes with their grinders. (We've just purchased our third and have had *zero* downtime attributable to the machine tool.)

"And the CNC is so easy to use we're doing virtually all of our programming right on the floor.

"But what we found most impressive about the Gear Hobber is the amazing ease of set-up and savings in changeover time. These days we can change from spur to helical gear in *1/4 the time* it used to take. That's saving us 75% in time, right up front. We can do very different gears, very fast, with exceptional quality."

--Manufacturing Engineer at a leading manufacturer of electric motors

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CIRCLE A-33 on READER REPLY CARD

Gear Inspection Chart Evaluation; Specifying Unusual Worm Gear Sets

Robert E. Smith
William L. Janninck

Question: When evaluating charts from a gear inspection machine, it is sometimes found that the full length of the profile traces vary, and that sometimes they are less than the length of active profile (above start of active profile-SAP) by up to 20%. This condition could be caused by a concentricity error between tooth grinding and shaping, or by unequal stock removal when grinding. (See Fig. 1.) Is it possible that some of the variation is coming from the inspection machine? How can variation from the inspection machine be reduced?

When looking at Fig. 1, the minus stock shows near the start of active profile (SAP), sometimes above and sometimes below the line. This is an undercut of the teeth. When producing gears with a small number of teeth, this can be a natural result of the trochoid generated by the cutting tool. However, the gear we are discussing had 31 teeth, so the undercut must be from intentional protuberance built into the pregrind shaping tool. Theoretically, if the cutter were designed correctly, this would clean up to a nice blend in the grinding operation. From a practical standpoint, this probably will never happen. The amount of protuberance would have to be just right, stock division evenly balanced, and the tool would only work for cutting a given tooth number. Also if there were runout of the blank between the cutting and grinding operation, the height of the undercut would be different in each tooth. Some would clean up and others would not.

If everything were theoretically correct, but the operator didn't divide the stock equally between sides, the undercut would show on one set of flanks only (e.g., the right flanks). That is not the case in Fig. 1. Some teeth look good on both sides, and some look bad on both sides. Therefore, I must assume that the operator did divide stock the best he could, and that runout did exist some place in the process between cutting and grinding of the teeth. Also, given the fact that the tips all line up on the chart, I don't believe that the inspection machine is at fault in this case.

However, there are some problems with the data furnished by the gear maker, and there are some things that can happen on the inspection machine to make the data faulty. First, the gear maker supplied charts without any substantiating scale factors or reference marks on it. The buyer of the gear put the dotted SAP line on by counting down from the tips of the teeth. This would only be correct if the gear OD were correct. The chart maker should mark at least one reference point for

SHOP FLOOR

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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is the principal in R.E. Smith & Co. Inc., gear consultants in Rochester, NY. He has over 50 years' experience in gearing methods, manufacture, and research.

William L. Janninck
is a consultant for ITW-Illinois Tools, a division of Illinois Tool Works, Inc. He has nearly 40 years' experience in gear engineering and manufacturing and is the author of numerous articles on gearing subjects.

degrees of roll or base tangent length, such as pitch diameter, SAP, etc. The chart maker should also put scale factors on a chart for profile deviations (X) and degrees of roll or base tangent length (Y). A chart that is unmarked is worthless to anyone the next day.

The inspection machine and recorder calibration could also be off as far as base tangent length is concerned. Another potential source of error in base tangent length (or degree of roll) evaluation would be a worn stylus. A worn stylus will give errors in the starting point of non-involute parts of a gear profile, such as tip modification and undercut. The best way to qualify a machine for this problem is to use a non-involute master as well as an involute master for calibration. Such masters are called pin masters or flank masters. They will quickly show a problem with stylus wear and placement relative to the master or theoretical base disk and tangent.

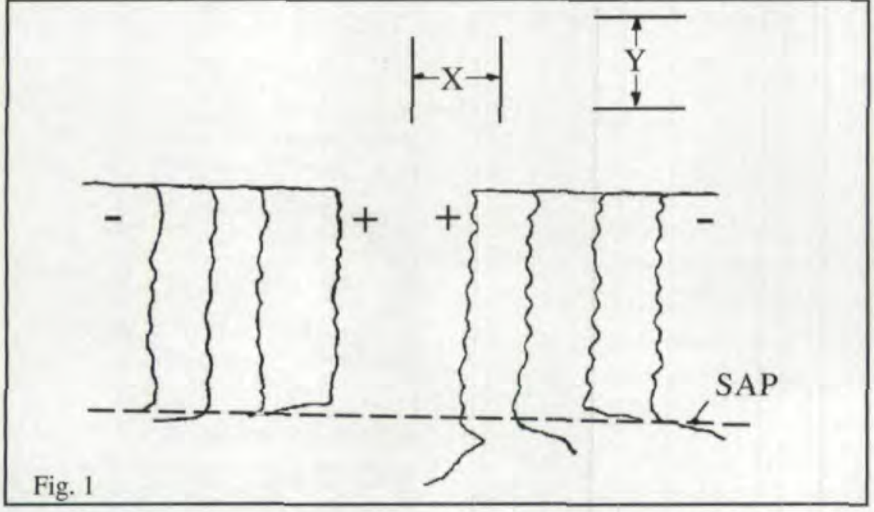


Fig. 1

Question: How should this undercut be evaluated in relation to performance and life?

Performance, in my mind, is related to noise or vibration (transmission error). If the undercut is severe and uses up 20% of the active profile as the questioner says, there could be loss of contact ratio and, therefore, loss of smooth motion. For the amount shown in Fig. 1, it should make only an insignificant difference.

Life of the gear set relates to surface durability and bending strength. Undercut will cause a reduction of bending strength. Whether this is serious or not depends on your design and application. If you have not experienced any breakages, it may not be a serious problem. If your design is borderline for strength, then it would be of concern.

To address questions to Mr. Robert E. Smith, circle Reader Service No. 78.

Question: Many of the gear sets manufactured by gear shops are of very low quantity, even as little as only one set. These are used frequently as replacement gearing for equipment breakdown and repairs. Along with the usual spur, helical, and worm gear sets that are needed, there is also a demand for some of the rather unusual designs, such as dual lead worm gear sets, double enveloping worm gear sets, and barrelled coupling parts. How does one layout and specify these, and can they be cut on conventional gear shop equipment?

Dual Lead Worm Gear Sets

Dual lead worm gear sets are somewhat special, since the worm has a different lead for the right and left flank. A conventional worm has the same lead on both flanks. The name dual lead is preferred, but we have seen part prints labelled as "tapered lead worm", yet the configuration is identical. The outside diameters and root diameters are cylindrical, and the worm thread thickness increases progressively from one end to the other. The purpose of these worm gear sets is to control or minimize backlash by axially advancing the worm relative to the wheel. The increasing thickness of the thread displaces the lash. Subsequent wear, which gradually increases lash, can periodically be adjusted out. These sets are typically used in indexing applications, including the main indexing drive in gear hobbing machines.

One method for specifying this type of set is to first establish the dimensions for a normal worm and gear pair, all dimensions being set according to the standard procedure and proportions. The basic lead established for the worm then becomes the helix that passes through the middle of the worm thread, and each flank lead is then made slightly longer and shorter than the basic by an equal amount. This is determined to suit the amount of thinning required from the thin to the thick end of the worm. Holding the worm pitch diameter and the adjusted lead for each flank, the right side and left side lead angles are determined. Each flank has separate dimensions, and the worm can be produced by thread milling, thread grinding, or thread chasing using two separate setups and passes, one for each flank, verifying thread thickness from end to end to complete the worm. All the operations use conventional cutting equipment.

The mating worm gear is also different in specification from flank to flank, with each being conjugate to its mating worm flank. If desired, a special dual lead worm gear hob can be used, and the gear is cut by the usual infeeding process. Each flank of the hob, which simulates the dual lead worm, generates each mating flank. As is usual in worm and gear cutting, the set should be run together on a tester or fixture or tried in a gear box to assure a proper match. The gear and worm must be properly oriented, and the worm axially located for proper fitting.

If one is making only a set or two, they can also be fly-cut using a tangential-feed-equipped hobber. Again the machine must be geared separately for each flank, making a separate pass for each one. Final fitting must again be checked on a tester or fixture.

Double Enveloping Worm Gear Sets

Double enveloping worm gears, that is, those sets where the worm has an hourglass form or throat, as well as a throat on the mating worm gear, are not readily duplicated using conventional gear cutting machines. Their primary purpose is increased load capacity or compactness. They can by layed out for specifications by duplicating the geometric shape of the parts to be replaced, maintaining center distance and part diameters and holding the number of

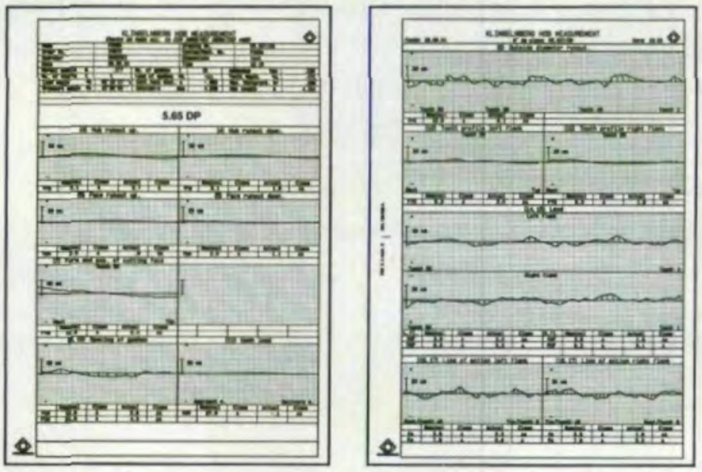
starts and teeth.

Hourglass worms can be machined on a regular hobber by putting a cutter or bladed cutter assembly on the usual work gear location on the hobber and mounting the worm blank in the hob arbor location. The cutter is then fed radially into the worm blank producing an hourglass form. The cutter may be of solid construction in high-speed steel with straight-sided teeth spaced around, or may be made of an assembly of tool points locked in a supporting body. Proper blade location is essential whether cutting single or multiple start worms. Grinding an hourglass worm does require special machines and cannot be done on conventional hobbing machinery.

Cutting the gear is another matter, and even if an hourglass shaped hob were available, some additional machine motions are essential. After the hob is fed to proper depth, the gear blank must be rotationally advanced

ADVERTISER INDEX

	Page Number	Reader Service Number
American Metal Treating Co.	46	A-25
American Pfauter, L.P.	Cover 2	A-1
Axicon Gear Company	46	A-22
BHS-Hofler Maschinebau GMBH	23, 24, 25, 26	
Bourn & Koch Machine Tool Co.	43	A-10
California Bevel Gears, Inc.	47	A-30
Ciateq A.C.	41	A-14
Forest City Gear	44-45	A-9, A- 12
GMI-Mutschler	42	A-15
High Noon	5	A-7
Intercontinental Industries Incorporated	9	A-8
James Engineering	38	A-17
Klingelnberg Gear Technology, Inc.	2	A-4
Koepfer America, Inc.	8	A-3
M & M Precision Systems	6	A-11
Manufactured Gear & Gage	46	A-24
Mitsubishi Heavy Industries	cover 4, 37	A-41,33
Niagara Gear	47	A-29
Normac	48, cover 3	A-19, A-20
Packer Engineering	46	A-26
Pfauter Maag Cutting Tools, L.P.	1	A-2
Pro-Gear Company, Inc.	46	A-27
Profile Engineering, Inc.	46	A-23
Software Engineering Service	47	A-28
Starcut Sales, Inc.	4-5	A-5, A-6
WMW Machinery, Inc.	38	A-18



and retarded for a trimming cut on each flank. If this operation is neglected, the extra benefits and increased capacity of the double enveloping set may be completely negated. All of my personal efforts to cut the mating worm gear by conventional hobbing, even on single start sets, has been unsuccessful.

It may be a better choice to go to the original manufacturer of the gears and procure a proper replacement. They have the necessary special worm and gear cutting and grinding machines, the hob manufacturing capability, and the technology to produce these sets.

Under certain circumstances, a temporary or interim fix may be to replace the double enveloping set with a conventional cylindrical worm set.

Barrelled or Crowned Coupling Splines

Barrelled or crowned coupling splines also can be dimensioned by copying the damaged parts, but cannot be produced

on a standard, conventional hobber without making some substantial modifications. It is necessary to make the hob change center distance with the part as it is fed along the part axis. This can be done by using a cam and follower mechanism or, if using a numerically controlled machine, by programming a rise and fall of the hob on the machine, such as is done in crown hobbing gear parts. We have seen some of these parts made first by hobbing the spline and then clearing away each end by form milling, leaving a small narrow tooth land. Functionally short term results were good.

If at all possible, it is best to try to procure these parts from the original source, where they have the special tooling and machinery to produce the parts properly. ■

To address questions to Mr. William L. Janninck, please circle Reader Service No. 79.

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NOVEMBER 24-26

JSME International Conference on Motion and Power Transmissions. Hiroshima, Japan. This conference covers all aspects of motion and power transmission systems theory and applications. For more information contact: Prof. Aizoh Kubo, Dept. of Precision Mechanics, Kyoto University, Kyoto, 606 Japan, FAX: (Japan) 75-771-7286. TELEX: 5423115 ENG KU J.

DECEMBER 4-6

University of Wisconsin-Milwaukee. Three-day seminar on "Fundamentals of Gear Design." For engineers who design or specify gears or gear drives, maintenance and process engineers, systems designers, and purchasing agents. Contact Richard G. Albers, Center for Continuing Engineering Education, U W-M, (414) 227-3125.

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.

APRIL 5-7, 1992

AGMA's 20th Gear Manufacturing Symposium. Indianapolis, IN. Seminars on numerous aspects of gear manufacturing and design. For more information, contact AGMA Headquarters, 1500 King St., Suite 201, Alexandria, VA 22314. Phone: (703) 684-0211 or FAX: (703) 684-0242.

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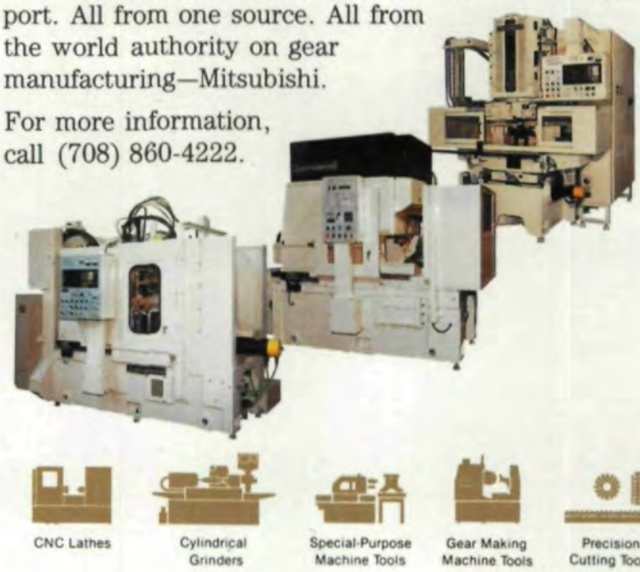

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