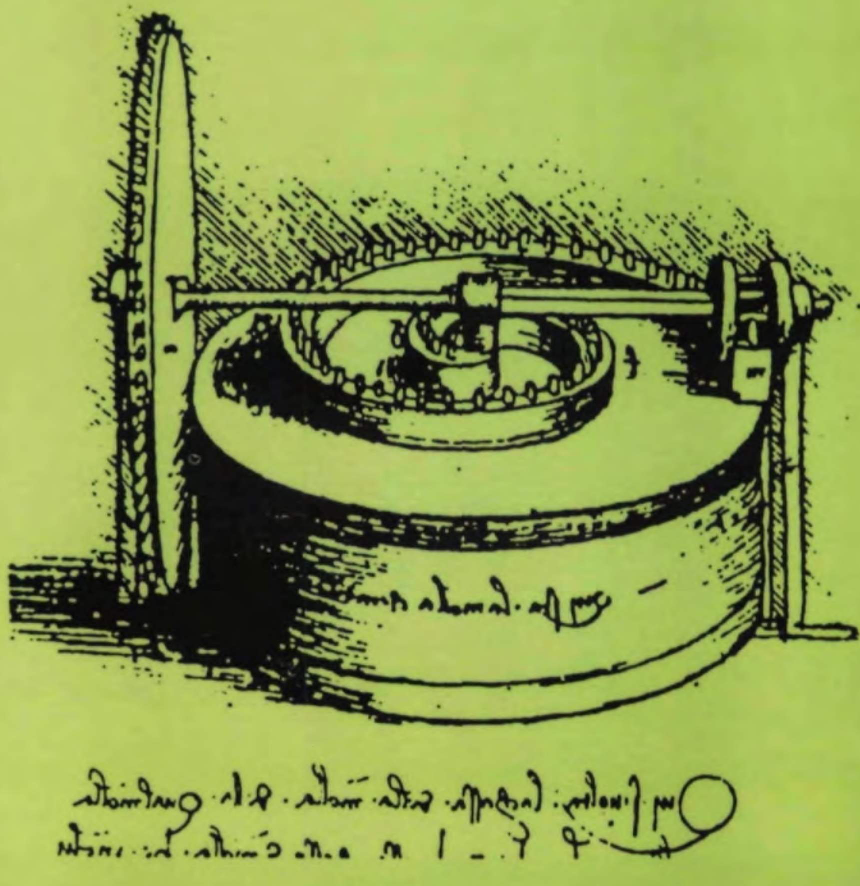
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

MAY/JUNE 1985



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COVER

The Advanced Technology of
LEONARDO DA VINCI
1452-1519

Leonardo persistently worked on designs to improve gearing and bearings in order to reduce frictional resistance. It appears that he was one of the first people to suggest that pivots could be nested in bearings made of semiprecious stones and even diamonds. Furthermore, he used his great genius to improve the construction of gear wheels.

A portion of his early designs dealt with ensuring the precision and reliability of timepieces.

The cover design is one of four in which he was searching for a better way to equalize the force of the clock spring. In this design, he suggests a volute gear on the spring barrel which would mesh with a sliding pinion guided by the edge of the volute. There are problems with this design as the straight guide on the side of the pinion would interfere on the inside curve of the volute. This interference would increase as the center of the volute were approached. It seems apparent that this design represents a tentative concept rather than a functioning mechanism.

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The Journal of Gear Manufacturing

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DEPARTMENTS

May/June 1985 — Vol. 2, No. 3

SUBSCRIPTION INFORMATION: GEAR TECHNOLOGY is distributed free of charge to **qualified** individuals and firms in the gear manufacturing industry. To continue receiving your subscription, complete and mail the application card found elsewhere in this magazine. Subscription rates for nonqualified individuals and firms are: $30.00 in the United States, $50.00 for foreign countries. Back issues are $7.00 in the United States, $15.00 for foreign countries. Subscription requests to RANDALL PUBLISHING CO., INC., P.O. BOX 1426 Elk Grove, IL 60007, (312) 437-6604.

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NOTES FROM THE EDITOR'S DESK

Photo by Jennifer Short

This issue, our sixth, marks the 1st Anniversary of GEAR TECHNOLOGY, The Journal of Gear Manufacturing. During the past year, I have met many of our readers at technical conferences, in their plants, or when they have visited our affiliate, CADILLAC MACHINERY CO. The enthusiastic response and encouragement I have received has been very gratifying to my staff and me. I would like to especially thank our advertisers, without whose support this magazine, the gear industry's magazine, would not be possible.

Special recognition and thanks to my very able Associate Publisher, Peg Short, who is responsible not only for turning manuscripts into error free articles, but also for the overall design and production excellence throughout the magazine. Also deserving special thanks is my brother, Richard, who spent countless nights working on the thousands of subscription cards you've sent back. And without the seven day-a-week effort of Debbie Donigian, we would not have our new mail list software installed, debugged and properly converted.

This next year, GEAR TECHNOLOGY will continue to bring you various articles presented at technical conferences and symposiums, and new articles about discoveries, techniques, theories and applications — to help you produce a better gear at a lower cost. It is our hope that this information will help the United States maintain a viable and competitive gear industry.

Thank You

Michael Goldstein
Editor/Publisher

INDUSTRY FORUM

"INDUSTRY FORUM" provides an opportunity for readers to discuss problems and questions facing our industry.

Please address your questions and answers to: INDUSTRY FORUM, GEAR TECHNOLOGY, P.O. BOX 1426, Elk Grove Village, IL 60007.

Letters submitted to this column become the property of GEAR TECHNOLOGY. Names will be withheld upon request; however, no anonymous letters will be published. Opinions expressed by contributors are not necessarily those of the editor or publishing staff.

Dear Editor,

This letter is in response to your article asking the readers where their interests lie. The division of Rockwell International where I work has engineering departments in Cicero, Illinois and Preston, England, each having its own gear standards department. This has caused a few differences because we use AGMA standards plus information published in the U.S.A. while Preston uses DIN, BS standards plus information published in Europe.

Our division builds newspaper printing presses which, because of geometry, use many non-standard gears. To my knowledge, we have always used the Zahorski method (Van Keuren) to determine the tooth thickness and measurement over wires. I have heard of a few instances when the gear set was placed on centers and we did not get the proper backlash. Some of these problems have occurred when checking gears with high helix angles or with low numbers of teeth. I realize that this method has been in existence since May 8, 1951, but I still wonder if there is a discrepancy in the geometry or equations.

My Preston counterpart is advocating the European belief of addendum modification coefficients which modify every gear to some extent. What happens to interchangeability when all gears are modified? Can addendum modification be used with more than one mate? Have the Europeans found a key to avoid problems the Zahorski method has (if there are problems) or have they created different problems?

Is there anything in AGMA or elsewhere (ANSI) describing tooth proportions for non-standard gears and measurement over wire calculations? What is the preferred method in the U.S.A. for determining tooth thickness and measurement over wires?

Thank you for your assistance.

Edward Ubert
Rockwell
International

Reference:

Precision Measuring Tools, Handbook #37, by Van Keuren Co. page 138-139.
DIN 3960 German Standard
PD 6457 British Standard
VSM 15525 Swiss Standard
PD436 British Standard

Gear Tooth Scoring Design Considerations for Spur and Helical Gearing

by
Peter Lynwander
Breeze-Eastern Corp., Union, N.J.

Introduction

High speed gearing, operating with low viscosity lubricants, is prone to a failure mode called scoring. In contrast to the classic failure modes, pitting and breakage, which generally take time to develop, scoring occurs early in the operation of a gear set and can be the limiting factor in the gear's power capability.

Scoring is a form of surface damage on the tooth flank, which occurs when overheating causes the lubricant film to become unstable allowing metal to metal contact. Local welding is initiated and the welded junctions are torn apart by the relative motion of the gear teeth resulting in radial score marks. Figs. 1, 2 and 3 illustrate degrees of the scoring phenomenon. Light scoring, which does not progress, may be acceptable, but heavier scoring can destroy the tooth profile and lead to pitting and breakage. Also, scoring can result in noisy operation and the release of metallic paricles into the lubrication system.

The American Gear Manufacturers Association Aerospace Gearing Committee has been investigating the scoring phenomen for many years. AGMA Information Sheet 217.01, "Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears," October, 1965, is a widely used method for predicting scoring based on Professor Blok's flash temperature index concept.[1] The Scoring Information Sheet is currently being updated and this paper is a review of the changes and additions being made.

In addition to the flash temperature approach, the minimum film thickness concept will be covered in the new publication. In the minimum film thickness approach, an elastohydrodynamic film thickness is calculated and compared to the asperity height to determine whether the lubricant film is adequate.

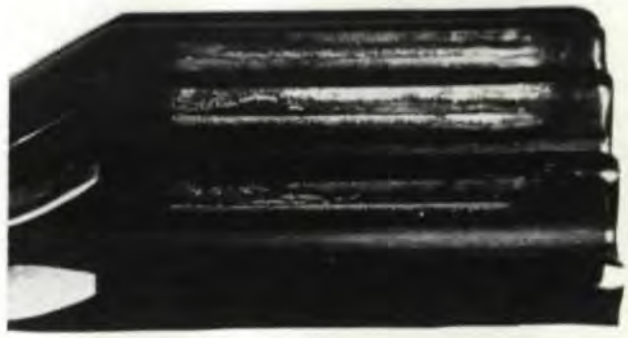

Fig. 2—Moderate Scoring

Flash Temperature Index

The critical total temperature hypothesis (flash temperature index) appears to be the most reliable method of analysis presently used to predict scoring. It states that scoring will

Fig. 1—Light Scoring

AUTHOR:

MR. PETER LYNWANDER *is the Director of Engineering at Breeze-Eastern Corporation, Union, New Jersey. His experience encompasses 25 years of design, analysis and development of mechanical systems, and includes establishing computer programs for the analysis and optimization of gear configurations. Mr. Lynwander received his B.S. and M.S. degrees from the University at Bridgeport, Connecticut, and he is a registered Professional Engineer. He has been active on several AGMA committees. Mr. Lynwander has authored numerous publications concerning gearing, seals and clutches including Gear Drive Systems published by Marcel Dekker Inc. This paper was written while Mr. Lynwander was employed by American Lohmann Corp., Hillside, N.J.*

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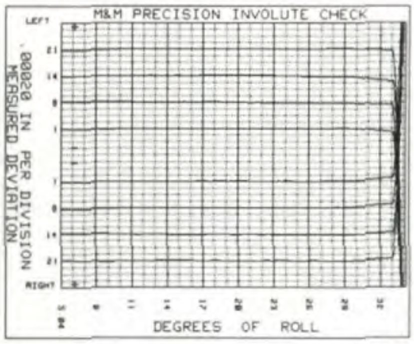

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Fig. 3 – Typical Heavy Scoring

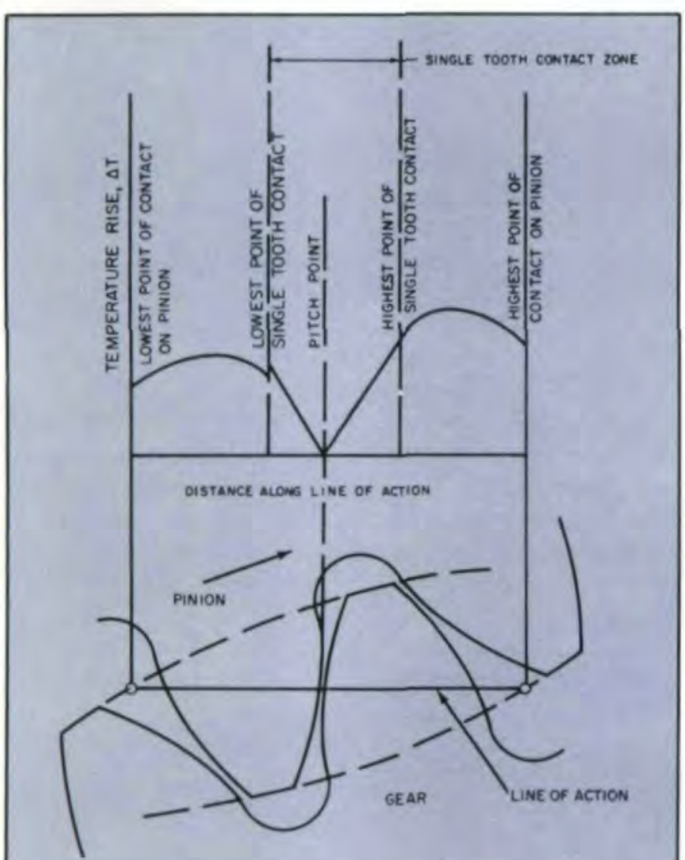

Fig. 4

occur when a critical total temperature, which is characteristic of the particular combination of lubricant and gear material, is reached.

$$T_f = T_b + \Delta T$$

where T_f = Flash temperature index, °F

T_b = Gear blank temperature, °F

ΔT = Maximum rise of instantaneous surface temperature in the tooth mesh above the gear blank's surface temperature.

The gear blank temperature is difficult to estimate. It may be significantly higher than the bulk oil temperature.[2] The heat transfer capability of the gear must be considered in attempting to estimate this parameter.

Often the blank temperature is approximated as the average of the oil temperature entering and leaving the gearbox.

One form of the fundamental flash temperature index formula is:[3]

$$T_f = T_b + \frac{C_f\, f\, W\,(V_1 - V_2)}{(\sqrt{V_1} + \sqrt{V_2})\sqrt{B_c/2}}$$

where C_f = Material constant for conductivity, density and specific heat

f = Friction coefficient

V_1 = Rolling velocity of pinion at point of contact, fps

V_2 = Rolling velocity of gear at point of contact, fps

B_c = Width of band of contact

W = Specific loading, normal load divided by face width, lbs./in.

For steel on steel gears, taking C_f as 0.0528, f as a constant 0.06 and adding a term taking surface finish into account[2] the following equation results.[4]

$$T_f = T_b + \left[\frac{W_{te}}{F_e}\right]^{3/4}\left[\frac{50}{50 - S}\right] Z_t\,(n_p)^{1/2}$$

where: W_{te} = Effective tangential load, lbs.

F_e = Effective face width (use minimum contact length for helical gears), in.

S = Surface finish (after running in), RMS

n_p = Pinion RPM

Z_t = Scoring geometry factor

$$Z_t = \frac{.0175\left[\sqrt{e_p} - \sqrt{\dfrac{N_p\, e_g}{N_g}}\right]}{\left[\cos \phi_t\right]^{3/4}\left[\dfrac{e_p \bullet e_g}{e_p + e_g}\right]^{1/4}}$$

Note: Use absolute value of Z_t

where e_p = Pinion radius of curvature, in. } See Appendix I
e_g = Gear radius of curvature, in. } See Appendix I
N_p = Number of pinion teeth (smaller member)
N_g = Number of gear teeth (larger member)
ϕ_t = Pressure angle, transverse operating

The $\frac{50}{50 - s}$ term was developed by Kelly[2] in an experimental program using gears with surface finish in the range of 20-32 RMS. For gears with surface finish rougher than this range, if the computed value exceeds 3, a factor of 3 should be used. For gears with surface finishes finer than 20, the resulting computed factor may be conservative.

The term W_{te} must be adjusted to allow for the sharing of load by more than one pair of teeth. The following analysis, which modifies the tooth load depending on the position of the gear mesh along the line of action, was developed by Dudley using spur gears of standard proportions. If a more accurate prediction of tooth load sharing is available to the reader it would be appropriate to use that analysis.

$$W_{te} = KW_t$$

where W_t = Tangential tooth load, lbs.

1. Unmodified tooth profiles

$$K = 1/3 - 1/3\left[\frac{\theta + \theta_{LD}}{\theta_L + \theta_{LD}}\right]$$

$$\theta_{LD} \leq \theta < \theta_L$$

$$K = 1.0$$

$$\theta_L \leq \theta \leq \theta_H$$

$$K = 1/3 + 1/3\left[\frac{\theta_O + \theta}{\theta_O + \theta_H}\right]$$

$$\theta_H < \theta \leq \theta_O$$

2. Modified tooth profiles

A. Pinion driving

$$K = \frac{6}{7}\left[\frac{\theta - \theta_{LD}}{\theta_L - \theta_{LD}}\right]$$

$$\theta_{LD} \leq \theta < \theta_L$$

$$K = 1.0$$

$$\theta_L \leq \theta \leq \theta_H$$

$$K = \frac{1}{7} + \frac{6}{7}\left[\frac{\theta_O - \theta}{\theta_O - \theta_H}\right]$$

$$\theta_H < \theta \leq \theta_O$$

B. Pinion driven by gear:

$$K = \frac{1}{7} + \frac{6}{7}\left[\frac{\theta - \theta_{LD}}{\theta_L - \theta_{LD}}\right]$$

$$\theta_{LD} \leq \theta < \theta_L$$

$$K = 1.0$$

$$\theta < \theta < \theta_H$$

$$K = \frac{6}{7}\left[\frac{\theta_O - \theta}{\theta_O - \theta_H}\right]$$

$$\theta_H < \theta \leq \theta_O$$

where θ = Any pinion roll angle (see appendix), radians

θ_{LD} = Roll angle at the pinion limit (form) diameter, radians

θ_L = Roll angle at the lowest point of single tooth contact on the pinion, radians

θ_H = Roll angle at the highest point of single tooth contact on the pinion, radians

θ_o = Roll angle at the pinion outside diameter, radians

A modified tooth profile would be one which has tip and/or flank relief rather than a true involute form.

The flash temperature index should be calculated at five specific points on the line of action and at several additional points of contact. The five specific points are:
1. Outside diameter of pinion
2. Highest point of single tooth contact
3. Pitch point (flash temperature rise will be zero since there is no sliding)
4. Lowest point of single tooth contact
5. Form (contact diameter) of pinion.

Fig. 4 is a typical plot of flash temperature rise along the line of action.

The most convenient way to generate a plot, such as shown in Fig. 4, is by the use of a computer program. By stepping through successive roll angles, the Flash Temperature Index can be calculated at many points. From Fig. 4 it can be seen that there will be two peaks, one during the arc of approach (pinion form diameter to pitch diameter) and one during the arc of recess (pinion pitch diameter to outside diameter). In order to achieve the minimum flash temperature index, the flash temperature rise in the arc of approach should be equal to the rise in the arc of recess. An optimum tooth design can be achieved by the use of long and short addendums. The computer program, starting with standard addendums (1/Diametral Pitch), automatically varies the pinion and gear addendums in defined increments until the optimum flash temperature is obtained. With the resulting long and short addendum designs of this nature, standard tooth thicknesses are no longer applicable. If standard tooth

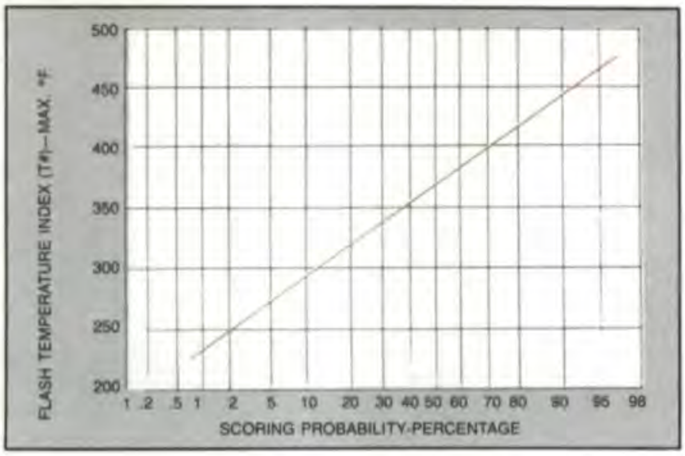

Fig. 5

thicknesses were utilized, an unbalance of bending stresses between pinion and gear would result. To optimize the bending stresses, the program enters a second iteration procedure which varies tooth thickness until bending stresses are balanced.

Fig. 5 presents the results of an aerospace industry survey[4] correlating scoring to flash temperature index. From the data used in the study, it was assumed that a gear set with a calculated index of 276°F, or less, represents a low scoring risk. A calculated index ranging from 277°F to 338°F represents a medium scoring risk where scoring may or may not occur, and a calculated index of 339°F and higher represents a high scoring risk. The data presented in Fig. 5 reflects cases using SAE 9310 steel operating with MIL-L-7808 or MIL-L-23699 synthetic oil. The viscosity of these oils is approximately 4-6 centistokes at 200°F and 18-30 centistokes at 100°F. Mineral oils such as light turbine oils, used in high speed industrial applications, are more viscous and, therefore, can tolerate a higher flash temperature index.

The equation for the flash temperature index assumes a constant coefficient of friction of 0.06. If it is desired to calculate the coefficient of friction at each point on the line of action, the following equation can be used.[5]

$$f = 0.0127 \log_{10} \left[\frac{3.17 \times 10^8}{\mu_o \, V_s \, V_t^2 / W} \right]$$

where f = Coefficient of friction

μ_o = Absolute viscosity, in centipoises

V_s = Sliding velocity, i p s } see appendix

V_t = Sum velocity, i p s

W = Specific loading, lbs./in.

The equation breaks down at the pitch point where the sliding velocity is 0.0 and the friction coefficient goes to infinity.

Using a variable coefficient of friction, the flash temperature index formula becomes:

$$T_f = T_b + f \left[\frac{W_{te}}{F_e} \right]^{3/4} \left[\frac{50}{50 - S} \right] \left[Z_t \, (n_p)^{1/2} \right]$$

$$Z_t = \frac{0.2917 \left[\sqrt{e_p} - \sqrt{\frac{N_p \, e_g}{N_g}} \right]}{\cos 0_t^{\;3/4} \left[\frac{e_p \bullet e_g}{e_p + e_g} \right]^{1/4}}$$

Scoring Criterion Number

A simplified form of the flash temperature index is presented in the *Gear Handbook*, by Darle Dudley published by McGraw-Hill, 1962. A scoring criterion number is defined.

$$\text{Scoring criterion number} = \left(\frac{W_t}{F_e} \right)^{3/4} \bullet \frac{(n_p)^{1/2}}{(P_d)^{1/4}}$$

where: W_t = tangential driving load, pounds

F_e = contacting face width inches

n_p = pinion RPM

P_d = diametral pitch

Table 1 gives scoring criterion numbers for various oils at various gear blank temperatures.

If the scoring criterion number is above the values shown in the table, a possibility exists that scoring will be encountered. The gear blank temperature can be taken as the average of the oil in and oil out temperatures.

Minimum Film Thickness Criterion

Scoring is a phenomenon that will occur when gears are operating in the boundary lubrication regime[6] rather than with a hydrodynamic or elastohydrodynamic oil film separating the gear teeth. The film thickness can be calculated[7,8,9] and compared to the combined surface roughness of the contacting elements to determine if metal to metal contact is likely to occur. A criterion used to determine the possibility of surface distress is the ratio of film thickness to composite surface roughness:

$$\lambda = h_{min} / \delta$$

$$\delta = \sqrt{\delta_p^2 + \delta_g^2}$$

where: λ = Film parameter

h_{min} = Minimum oil film thickness, in.

δ_p = Pinion average roughness, RMS

δg = Gear average roughness, RMS

The "Partial Elastohydrodynamic" or "mixed" lubrication regime occurs if the film parameter, λ is between approximately 1 and 4. At higher values, full hydrodynamic lubrication is established and asperity contact is negligible. Below a λ of 1.0 there is a risk of surface distress.

Table 1

Critical Scoring Criterion Numbers

Blank Temp., °F	100°	150°	200°	250°	300°
Kind of Oil	**Critical Scoring Index Numbers**				
AGMA 1	9000	6000	3000		
AGMA 3	11000	8000	5000	2000	
AGMA 5	13000	10000	7000	4000	
AGMA 7	15000	12000	9000	6000	
AGMA 8A	17000	14000	11000	8000	
Grade 1065 Mil-O-6082B	15000	12000	9000	6000	
Grade 1010 Mil-O-6081B	12000	9000	6000	2000	
Synthetic *(Turbo 35)*	*17000*	*14000*	*11000*	*8000*	*5000*
Synthetic Mil-L-7808D	15000	12000	9000	6000	3000

$$\text{Scoring number} = \left(\frac{W_t}{F_e}\right)^{3/4} \bullet \frac{(n_p)\ 1/2}{(P_d)\ 1/4}$$

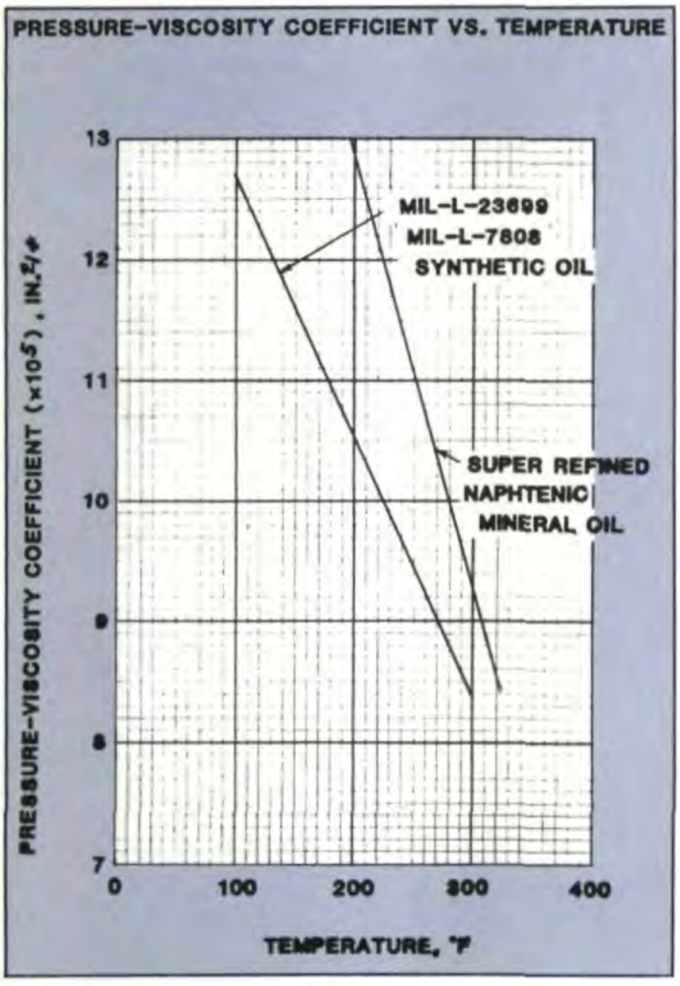

Fig. 6

The minimum elastohydrodynamic film thickness is calculated as follows: (Ref. 7 and 8):

$$H = \frac{2.65(G)^{.54}\ (U)^{.70}}{(W)^{.13}}$$

$$H = \frac{h_o}{R} \text{ (Film thickness parameter)}$$

h_o = Minimum film thickness, in.

R = Equivalent radius, in.

$$R = \frac{e_p \bullet e_g}{e_p \pm e_g}$$

$$\begin{pmatrix} + \text{ external} \\ - \text{ internal} \end{pmatrix}$$

e_p = pinion radius of curvature, inches

e_g = gear radius of curvature, inches

G = $\partial E'$ (Materials parameter)

∂ = pressure viscosity coefficient, in^2/lb (Fig. 6)

$$\frac{1}{E'} = \frac{1}{2}\left[\frac{1-\partial_1^2}{E_1} + \frac{1-\partial_2^2}{E_2}\right]$$

∂_1 = pinion Poisson's ratio

∂_2 = gear Poisson's ratio

E_1 = pinion Young's modulus

E_2 = gear Young's modulus

W = $W'/E.R$ (Load parameter)

W' = Specific loading, lbs/in.

$$U = \frac{\frac{1}{2}(v_1 + v_2)\ \mu_o}{E'\ R}$$

v_1 = $w_p e_p$ = rolling velocity of pinion at point of contact, in/sec.

v_2 = $w_g e_g$ = rolling velocity of gear at point of contact, in/sec.

w_p = pinion angular velocity, rad/sec.

w_g = gear angular velocity, rad/sec.

μ_o = absolute viscosity, Reyns (lb sec/in^2)

μ_o = $\varrho Z_k / 6.9\ (10)^6$

ϱ = specific gravity (Fig. 7)

Z_k = kinematic viscosity, centistokes (Fig. 8)

Closure

An extensive survey of aerospace power gears operating with synthetic lubricants at high temperature revealed that calculated oil films were in the order of 0.000010 to 0.000020 inches. With surface roughness in the order of 20 RMS it can be seen that these gears are operating with λ less than 1.0; are in the boundary lubrication regime and, therefore, prone to scoring problems.

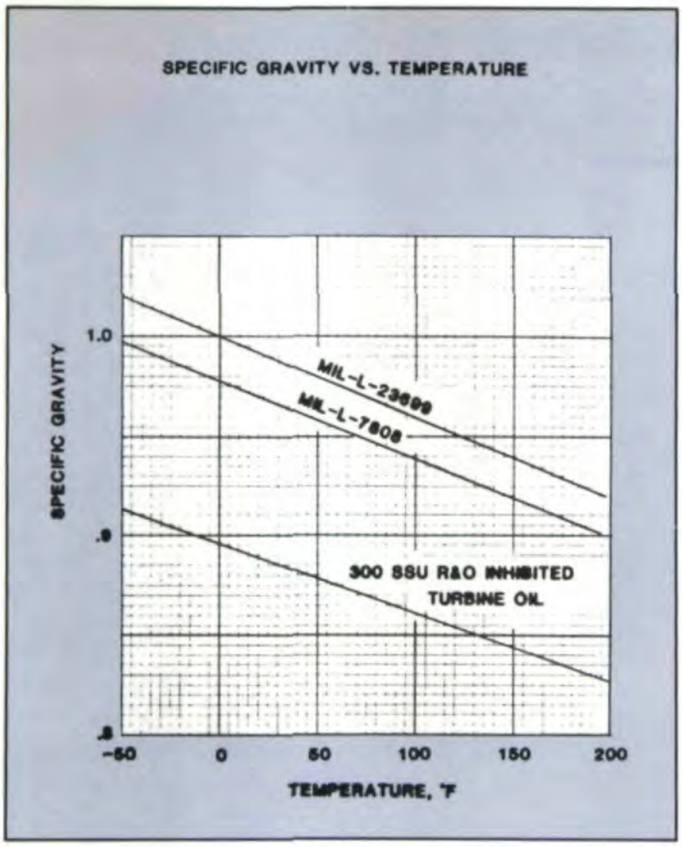

Fig. 7

Table 2

Scoring Analysis of a High-Speed Gearset

Design Parameters

Pinion teeth: 25	Gear teeth: 85
Pinion speed: 12,223 rpm	Horsepower: 281
Face width: 1.0 in.	Helix angle: 0
Diametral pitch: 10	Pressure angle: 25°
Lubricant type: MIL-L-23699 oil	Gear blank temperature: 160°F
Oil viscosity: 1.22 (10^{-8}) lb-sec/in.^{2}	Pressure viscosity coefficient: 11.4 (10^{-5}) in.^{2}/lb
Gear material: SAE 9310 steel	Surface finish: 20 μin. rms

Roll Angle, θ (deg)	Friction Coefficient, f	Flash Temperature Rise, ΔT(°F)*	Flash Temperature Rise, ΔT(°F)†	EHD Film Thickness, h_{min} (μin.)	Film Parameter, λ
15.32	0.0	0	0	0	0.0
16.41	0.014	0	40	22	0.77
17.50	0.018	18	58	21	0.73
18.60	0.021	23	68	20	0.72
19.69	0.023	28	71	20	0.71
20.78	0.025	30	70	20	0.71
21.87	0.027	30	64	20	0.72
22.96	0.030	30	[illegible]	20	0.71
24.05	0.032	22	42	21	0.73
25.14	0.035	13	25	21	0.75
26.23	0.041	5	7	22	0.76
27.32	0.040	7	9	22	0.78
28.41	0.034	13	25	23	0.80
29.50	0.031	22	41	23	0.81
30.59	0.028	25	52	24	0.84
31.68	0.026	25	59	25	0.88
32.77	0.024	25	62	26	0.91
33.86	0.021	22	61	27	0.96
34.95	0.019	18	57	29	1.01
36.04	0.016	13	48	30	1.08
37.13	0.012	7	33	34	1.19

*f is variable
†f held constant at 0.06

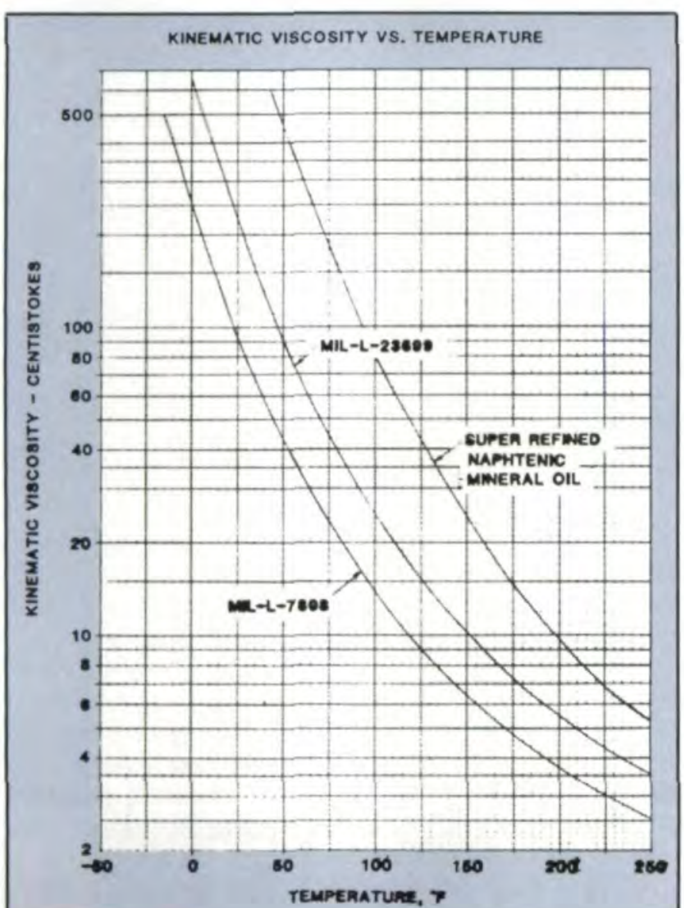

Fig. 8

(continued on page 48)

Advantages of Involute Splines as Compared to Straight Sided Splines

by
Tifco Spline, Inc.
Wixom, MI

Since the design of involute splines and their manufacture requires considerable knowledge, not only of the basic properties of the involute profile, but also of various other elements which affect the spline fit, and the sometimes complex principles underlying manufacturing and checking equipment, the question is frequently raised as to why the involute profile is given preference in designing splines over the seemingly simpler straight sided tooth profile.

As a matter of fact, the first spline coupling was a straight sided spline with one tooth: a shaft keyed to a hole. The simple expedient of increasing the number of teeth, in order to increase the load carrying ability of this machine element, led to the development of straight sided spline fittings.

More exacting requirements on torque transmitting splines compelled designers to look for ways to satisfy the following conditions:

1) To increase the contact surface without decreasing the minor diameter of the shaft or increasing the major diameter of the splined hole.
2) To insure full surface contact on the tooth flanks, independent of the unavoidable clearance between the teeth. Although the necessity of a clearance between spline fittings is not disputed, it may be pointed out that such clearance will not be negligible if the parts are heat treated after machining. The distortions caused by the heat treating process must be compensated for by sufficient initial clearance to permit assembly of mating splines.
3) *To design a tooth profile which conforms* to the stress distribution along the depth of the tooth, i.e. which is heavier at the root than at the crest.
4) To provide a tooth profile which can be easily produced by means of fast cutting mass production machine tools (hobbing machines, shaper cutters), not a profile which only appears simple in a cross sectional view.

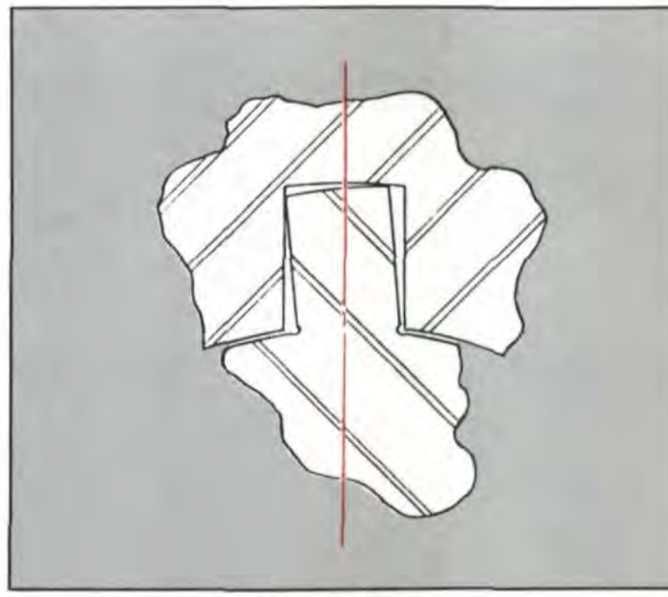

Fig. 1 – Line contact.

Conditions #1 and #2 are especially important when intermittent torque loads on the shaft cause intermittent length reductions and, as a consequence, longitudinal sliding movements in the spline fitting.

The following discussion of these four requirements demonstrates clearly why the involute profile has been selected, out of all other possibilities, as the best spline tooth form which meets practical requirements.

Condition #1

A profile which is not straight radial should be acceptable, insofar as Condition #1 is concerned, calling for increased contact surfaces without a change in *diameters*.

Condition #2

Fig. 1 shows that straight sided splines, with the sides of the shaft tooth parallel to the center line cannot have surface contact on the side of the tooth due to the clearance between the mating parts: the shaft rotates against the splined hole when under load and the points farthest from the center travel faster than points near the minor diameter. Therefore, the tip of the tooth on the shaft reaches the side of the hole splines before any other point of the tooth side can contact.

The problems can only be overcome by making the amount of clearance proportional to the distance from the center. In fig. 2-A, this condition is satisfied since the clearances C_1 and D_2 are in proportion to the diameters, D_1 and D_2. Such splines, however, with radial tooth sides are, by no means, the solution of the problem, due to the weak cross section of the shaft tooth at its root and the impractical shape of the splined hole.

Fig. 2-B shows how this condition could be satisfied with straight sided serrations. The clearance C_1 and C_2, *subtending the same angle are in proportion* to the diameters D_1 and D_2 at which they are measured. When the spline tooth rotates under load against the side of the female spline tooth, it matches this side along its whole active depth. Consequently, after it is rotated back into the position shown in fig. 2-B the angles 1 and 2, which represent the

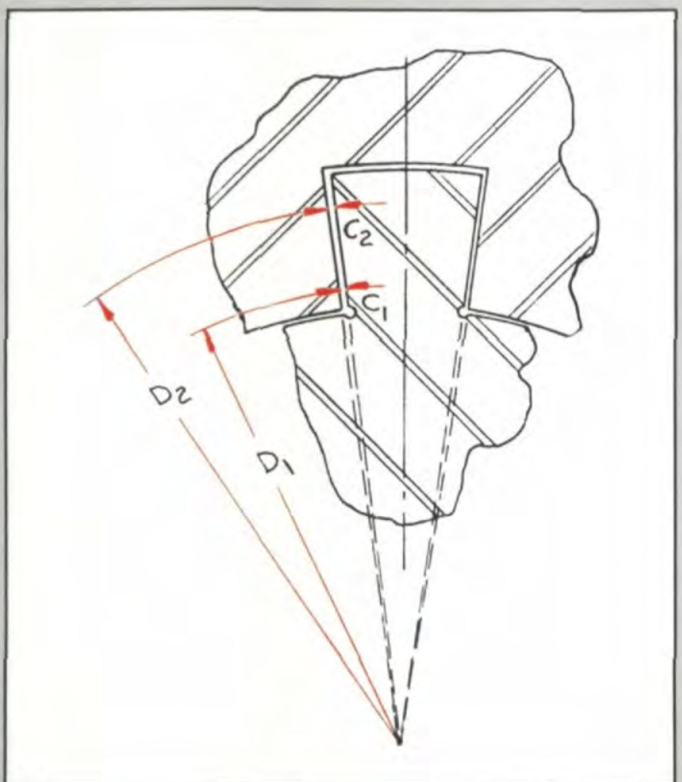

Fig. 2A

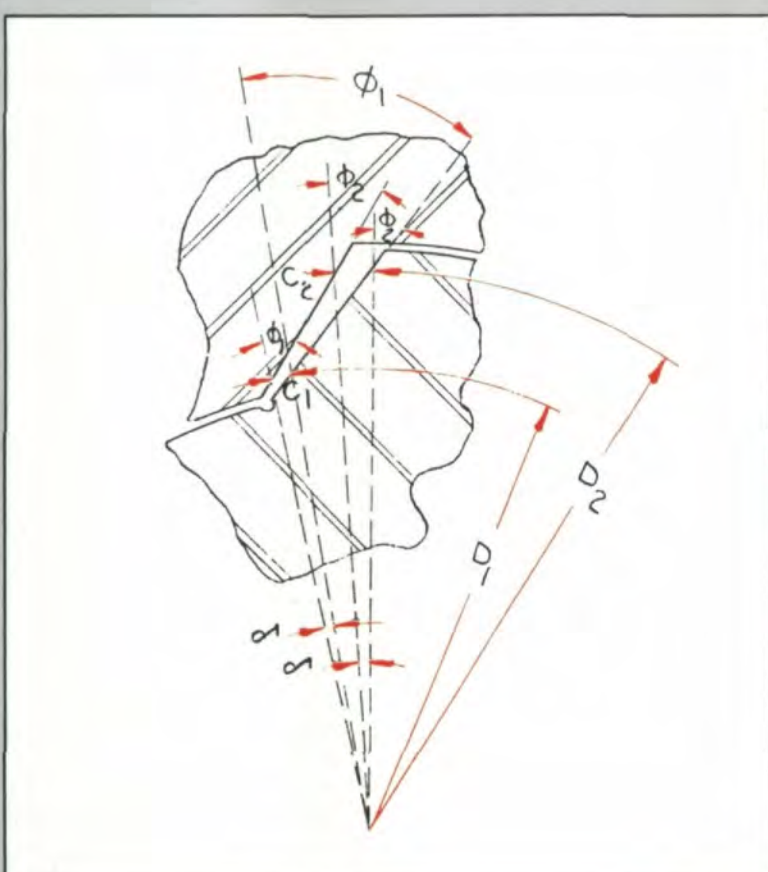

Fig. 2B

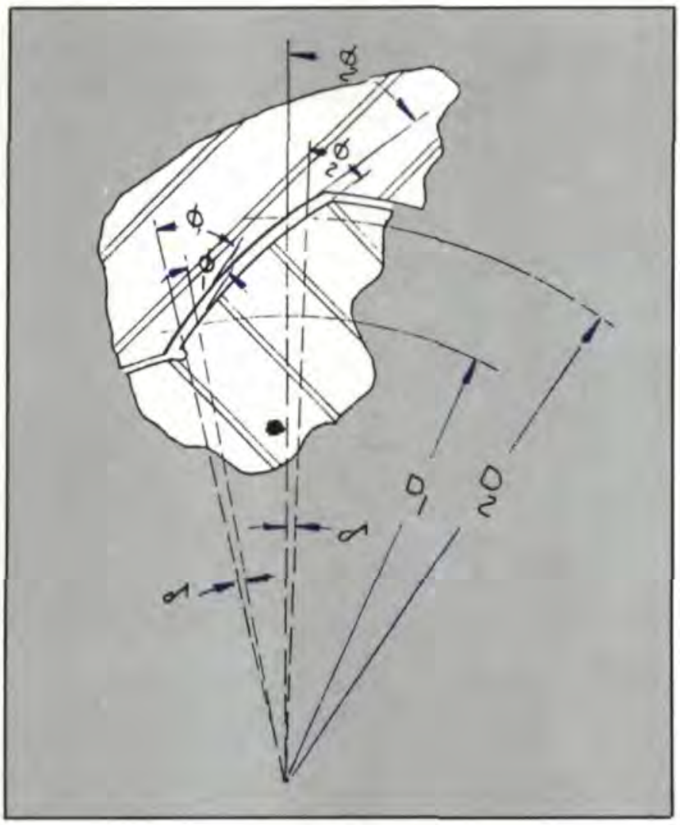

Fig. 2C

angles between the profile and a radial line through the intersection of the profile with reference diameter, are equal on the female and the male spline tooth. These angles are commonly called the pressure angle. *This discussion shows that the requirement of surface contact demands equal pressure angles at the same reference diameter for both splines.* The application of this principle, on the general case of splined teeth with curved sides, is shown in fig. 2-C. The pressure angles 1 and 2 are the same on both mating spline profiles. The advantages of curved tooth profiles will be discussed under Condition #4.

Condition #3

This condition will be satisfied by selecting a sufficiently large pressure angle, for instance 30°. An additional advantage of the large pressure angle appears in connection with case hardened splines and relatively fine pitches, where a small pressure angle is sometimes the cause of breakage, because no soft core is left at the root of the tooth. The larger pressure angle assures that such a core exists where it is needed.

Condition #4

If straight sided teeth could easily be produced on fast cutting machine tools, the development of the previous paragraphs would already present the solution of the problem. However, a straight sided spline can only be produced with straight sided cutters by milling, which requires time consuming indexing operations.

In view of the fact that the gear industry developed the machine element which conforms, not only to the first three conditions (presenting even a further improvement with regard to condition #1), but also to this last condition #4, the ideal of producing splines with the same shape as gear teeth appears as the next logical step. It is granted that splines of the serration type, as discussed above, could also be produced on fast cutting hobbing machines or on shaper cutters. The scales, however, are definitely turned in favor of involute splines, if the tool profiles are required for the two alternatives as compared.

It is possible to design a hob or shaper tool profile which will cut straight sided splines of any pressure angle. The development of this tooth profile requires either cumbersome calculations or time consuming layouts; but even after all of this work is completed, such tooth profiles present difficulties in manufacture and inspection. In addition to this, it would only be correct for cutting one gear of the pitch, pressure angle, *and the number of teeth for which it has been developed.* It could not be used interchangeably to cut any tooth form of the same pitch and pressure angle, regardless of the spline size. Besides these prohibitive disadvantages, the inherent inaccuracy of a tooth form on the cutter, which could not be generated by simple means, would cause corresponding inaccuracies of the tooth form on the part, which would defeat the purpose of the constant pressure angle, as outlined in condition #2. These difficulties restrict the general use of straight sided splines with constant pressure angle in the mass production of spline fittings for transmission of high loads. They are overcome by the application of involute profiles which can be easily generated by means of straight sided tools, by hobs with practically straight sided profiles, or by shaper cutters which themselves are produced by generation. It is true that this solution creates new difficulties in connection with producing small splined holes, since it is easier to produce a broach for straight sided splines than for involute splines. The involute spline broach, however, can, in many cases, be hobbed and, when grinding is necessary, is furnished with very satisfactory results by various tool manufacturers.

It should be borne in mind that the problem of providing a satisfactory involute profile, and at the same time the necessary back off (relief) angle, sometimes presents difficulties, but it is preferable to transfer manufacturing difficulties from the production line to the tool shop. Consequently, the involute splines have replaced straight sided splines in recent years to a very large extent, and they will continue to maintain a dominating position in the field of spline fittings.

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CIRCLE A-9 ON READER REPLY CARD

The Design and Manufacture of Machined Plastic Gears

Part 1

Design Parameters of Plastic Gears

by
John H. Chen and Frank M. Juarbe
The Polymer Corporation
Reading, Pennsylvania 19603

ABSTRACT

The use of Plastic Gearing is increasing steadily in new products. This is due in part to the availability of recent design data. Fatigue stress of plastic gears as a function of diametral pitch, pressure angle, pitch line velocity, lubrication and life cycles are described based on test information. Design procedures for plastic gears are presented.

Introduction

The use of machined plastic gears is increasing in industrial power transmission applications. Cast or extruded gear materials of 50 mm to over 2 meter (2" – 84") in diameter are machined to desired dimensions and gear tooth forms. Nylons and acetals are the most widely used thermoplastic gear materials. They offer resiliency, resistance to wear and corrosion, noise reduction, vibration suppression, lightweight and minimum maintenance. This increased use is due in part to the availability of new data in design parameters, manufacturing and installation techniques to optimize utilization of the materials and to prevent premature failures.

Some test data on plastic gears are the results of earlier publications.[1-4] Plastic gears are sensitive to frictional heat and hysteresis loss, which causes softening, expansion and deflection of the gear teeth. Tooth interference, caused by deflection and creep, can lead to different types of damage due to the viscoelastic nature of plastic gears.[5] For better heat dispersion and to reduce contact stress, plastic gears are usually mated with metal gears. To measure plastic gear temperature in operation, a technique utilizing an infrared radiometer was developed[6] and friction heat and hysteresis loss were investigated.[6, 7] Although plastic/steel contacts show rather low friction[8] and some plastic gears can be operated without lubrication, the lubrication also has a considerable effect on the performance of the plastic gears.[9] The tooth deflection of plastic gears and low Hertizan stress of plastic/steel contacts results in generally higher gear contact ratio and smooth rotation of plastic and steel gear mates. A model to assess gear contact ratios of plastic/steel gear mates was proposed.[10] A larger backlash allowance required for plastic/steel gear mates was suggested.[5] Factors which influence performance of plastic gears are, to an extent, clarified by the works mentioned above and others. However, design data available to meet power transmission of plastic gears is still insufficient for most applications. The purpose of this paper is to fill part of this gap by presenting some pertinent design parameters based on experiments, which were conducted at conditions close to typical field applications. Design parameters as functions of diametrical pitch, tooth form, lubrication, pitch line velocity, stress levels and life cycles using cast type 6 nylon and other plastic gears are presented.

AUTHORS:

DR. JOHN H. CHEN *is Project Leader for The Polymer Corp. where he has been employed since 1968. During this time he has been involved in various aspects of product development and application testing of thermoplastics. Dr. Chen received his BSME from Taiwan College of Engineering and his Dr. -Ing. degree from the Technical University of Aachen, West Germany. He is the author of over 30 technical papers in the field of plastic engineering, and holds patents in manufacturing methods of nylon parts. Additionally he is a member of the American Society of Mechanical Engineers, American Society of Testing Materials, Society of Plastics Engineers, and a committee member of the American Gear Manufacturers Association on "Design Guide for Plastics Gearing."*

MR. FRANK M. JUARBE *has been with Polymer Corp. for the past ten years as Product Manager where he has been involved in the development of Cut Tooth Plastic Gearing and Sprockets. Prior to his work at Polymer, he has over twenty years experience in related engineering fields. Mr. Juarbe is a member of the American Gear Manufacturers Association and the American Society of Mechanical Engineers.*

Experiments

Test Gears

Industrial experience indicates that anionically polymerized cast type 6 nylon is one of the highest strength unreinforced thermoplastic gear materials available. The typical test material contains molybdenum disulphide for improved lubricity and wear resistance. Impact modified Nylon 6/6 and ultra high molecular weight polyethylene (UHMWPE) are known to be very tough and resistant to fatigue and notch sensitivity. The latter starting resin has a molecular weight of 2-5 million by solution viscosity test. Polyacetal (melt viscosity 3360 poise), which features strength and low moisture absorption was also selected for this test. Physical properties of these four thermoplastic test gear materials are summarized in Table 1.

Maag Gear Inspection Center Model SP-42. Full CNC setup, operation and analysis of involute, profile, helix, pitch and runout in one setup. From 0" to 16.5" diameter gears including bevels and worms.

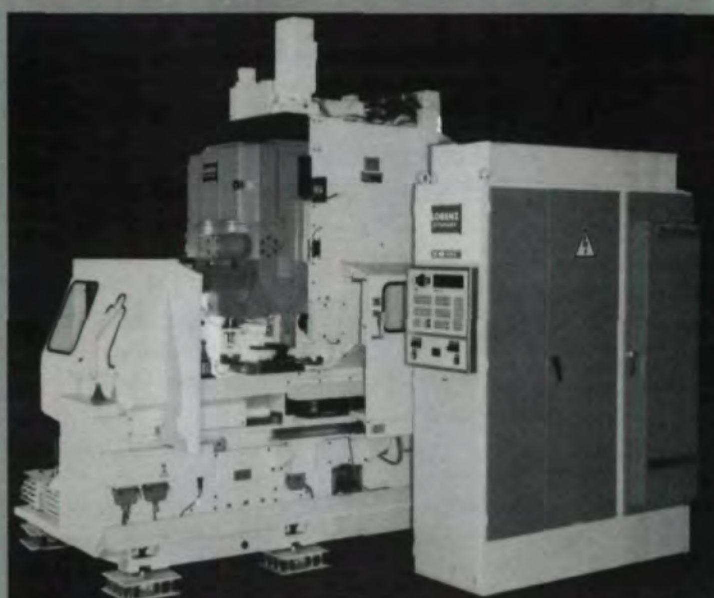

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TABLE 1. PHYSICAL PROPERTIES OF TEST GEAR MATERIALS

Property	Test Method ASTM	Unit	Cast Nylon 6 With MoS_2	Nylon 6/6[3] Impact Modified	UHMWPE	Polyacetal
Tensile Strength	D638	Mpa (psi)	84 (12,000)	52 (7,500)	33 (4,800)	73 (10,400)
Elongation	D638	%	15	40	320	12-75
Tensile Modulus	D638	Mpa (psi)	2,800 (400,000)	1,760 (255,000)	630 (90,000)	3,290 (470,000)
Hardness, Rockwell	D785	R	116	112	64	120
Deformation Under Load 14MPa (2000 psi), 50°C	D621	%	.75	—	6-8 (6 hrs.)	0.7
Deflection Temperature 1.8 MPa (264 psi)	D648	°C	156	71	—	117
Melting Point	D789	°C	215-227	250-260	130	165-170
Tensile Impact	D1822	kJ/m^2 (ft. lbs./in.^{2})	221 (105)	588 (280)	2,100 (1,000)	74 (35)
Specific Gravity	D792	—	1.16	1.09	0.94	1.42
Water Absorption						
24 Hours	D570[2]	%	0.9	1.2	.01	0.25
Saturation	D570	%	6.0	6.7	—	0.9

Note: (1) All values shown are based on as molded dry specimens
(2) Specimens 1/8" Thick, 2" Diameter (3) Published data of supplier

Test gears of 5, 8, 10, and 16 DP (5.1, 3.2, 2.5 and 1.6 modules) involute spur gears with 12.7 mm (½") face width were used. Tooth proportion was made according to AGMA standard 201.02 applying coarse-pitch, 20° full-depth involute form, except one series of gears with 14½° pressure angle. All test gears were cut with class A ground thread hobs. The test gears were "driven" gears. The "driver" steel pinion was hardened to Rockwell C44 and ground AGMA 390.03 Quality Number 13.

Test Equipment and Procedure

A four-square gear bench was utilized for fatigue testing of spur gears. The equipment consists of a variable speed electric motor, four-square mechanism, a torque-meter, a forced lubricating system with oil temperature control, a revolution counter, an electronic gear failure detection system and a digital panel instrument indicating torque or rotational speed. An overview of the test equipment is shown in Fig. 1.

Extreme pressure (EP) grade gear oil was heated to 120° ±2°F (49° ±1°C), while it circulated in a closed loop. After it reached this temperature, the oil was injected under pressure onto the departing side of the two meshing gears. The machine was started without load to run at the lowest speed for 10 minutes. Torque was then applied on the nylon gear,

Fig. 1—Overview of gear test equipment.

in the direction that it acts as a driven gear, before the machine speed was increased to test conditions. During the run, especially in the beginning, the torque had to be adjusted due to the creep of the plastic gear material. The adjustments were made, so that the torque was maintained within ±2% of its normal value throughout the test. The machine stopped automatically when the gear failed or when the load exceeded preset limits of high or low torque. The life of the gear could then be read from the revolution counter.

Test variables used are gear materials, diametral pitch, pressure angle, pitch line velocity and type of lubrication. Test parameters are listed in Table 2. At least four stress levels for each test parameter were applied.

The lubrication systems evaluated were continuous oil jet lubrication, initial greasing and unlubricated (dry). For the initial greasing test, a thin layer of calcium complex type petroleum base grease was brushed on the flanks of test thermoplastic gear and mating steel pinion before installation. For the unlubricated gear test, the flank surfaces were carefully wiped clean with acetone and then assembled on the test machine.

Test Results and Discussion

Root bending stress $\delta\beta$ vs. life cycles of the tested gears was calculated from the Lewis equation:

$$\delta\beta = \frac{F \bullet P}{f \bullet y} \qquad \text{Eq. (1)}$$

The form factor for thermoplastic gears, Table 3, was determined on the basis that the worst load conditions occur near the pitch point (1), due to multiple pairs of teeth sharing the load. The high contact ratio of thermoplastic gears can be observed by high speed photographic technique. The Lewis equation assumes that the gear contact ration is 1. This assumption presents a very practical way to assess fatigue strength of thermoplatic gears and serves design purposes, although it contains the factor of undetermined gear contact ratio.

Earlier test results indicate that thermoplastic gears, when properly designed and installed, fail eventually by root bending fatigue fracture.[5] Evaluation of fatigue life of thermoplastic gears for this paper was done by root stress using Equation (1).

Gear Fatigue Strength

S-N curves of 4 different diametral pitches of cast nylon 6 (filled with MoS_2) gears at pitch line velocity of 10.2 m/s (2000 fpm) are illustrated in Fig. 2. Test points of these oil jet lubricated gears, for each of the four diametral pitches, can be connected approximately as a straight line on a semi-logarithmic scale. The finest gears tested (16 P) could with-

TABLE 2. TEST PARAMETERS

Gear Materials	Diametral Pitch P	Pressure Angle, Degree	Gear Dp	Gear Teeth	Gear Dp	Gear Teeth	Pitch Line Velocity ft/min	Lubrication
Cast Nylon with MoS_2	5	20	3.6"	18	3.6"	18	2000	oil
	8	20 14½	4.5"	36	2.5"	20	2000	oil
	10	20	4.5"	45	2.5"	25	2000	oil & dry
	16	20	4.5"	72	2.5"	40	2000	oil
	10	20	4.5"	45	2.5"	25	680 1200 2000 4000	grease
Nylon 6/6, Impact Modified	10	20	4.5"	45	2.5"	25	2000	oil
UHMWPE	10	20	4.5"	45	2.5"	25	2000	oil
Polyacetal	10	20	4.5"	45	2.5"	25	2000	oil

Note: 1" = 25.4 mm 1000 ft./min. = 5.08 m/s

TABLE 3
TOOTH FORM FACTOR y

No. of Teeth	Pressure Angle 14½°	20° Depth	20° Stub
18	–	.521	.603
24	.509	.572	.664
28	.535	.597	.688
36	.559	.640	.721
45	.579	.681	.744
72	.611	.731	.788
150	.625	.779	.830
300	.650	.801	.855
Rack	.660	.823	.881

stand the highest root bending strength and vice versa. This can be explained by a higher contact ratio, i.e. more teeth to share the load, on fine pitch nylon gears.

For comparison, the endurance strength of 8 P, 14½° pressure angle gears were tested and the results charted with those of 8 P, 20° pressure angle gears (Fig. 3). The relationship between torque (load) and life cycle is linear and the two lines are approximately parallel. Gears with 20° PA are ca. 15% higher in load carrying capacity. Fatigue tangential force F at a life cycle is proportional to tooth form factor y as can be seen from Equation (1).

Root stresses of MoS_2 filled cast nylon 6 gears at 10 million cycles, from Fig. 2, are 6170, 4650, 3830 and 3180 psi for 16, 10, 8 and 5 pitches respectively. The root stress of other diametral pitches within the range tested can be illustrated as a curve (Fig. 4). Utilizing this curve, the fatigue life bending stress at 10 million cycles of other pitches can be obtained by interpolating the test points. Computer aided regression analysis show the root stress of other pitches can be expressed approximately as follows:

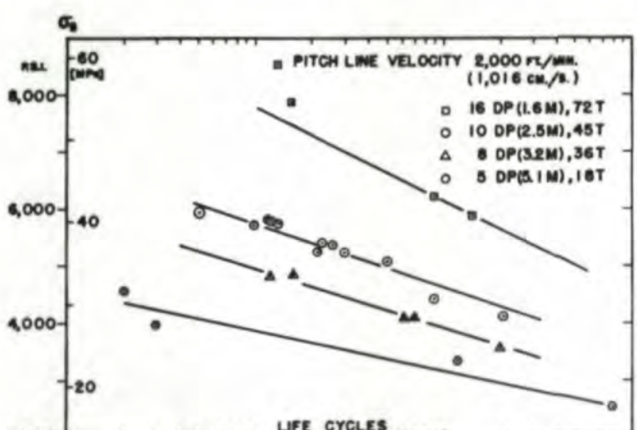

Fig. 2 – Bending fatigue stress of molybdenum disulphide filled cast nylon 6 gears at 2000 fpm (10.2 m/s). Standard 20° full-depth involute gears of different diametral pitches, oil jet lubricated (5).

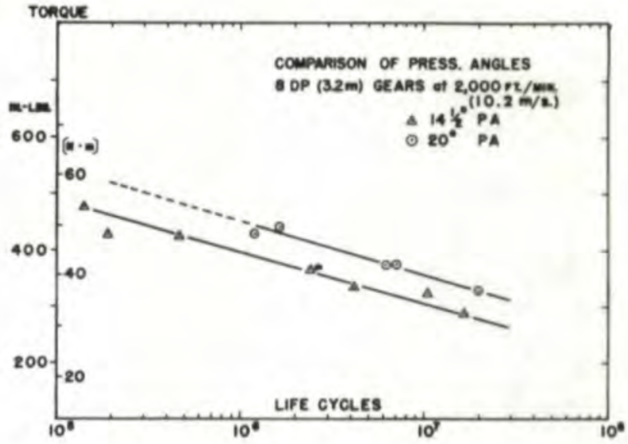

Fig. 3 – Torque vs. life cycle curves of gears with 14-1/2° and 20° pressure angles, oil jet lubricated (5).

$$\delta\beta = 28{,}992 - 25{,}637\,(\ln P) + 9{,}578.47\,(\ln P)^2 - 1{,}228.29\,(\ln P)^3 \qquad \text{Eq. (2)}$$

where $\ln P$ = natural logarithm of diametral pitch

Gears of 10 P, 45 teeth and ½" face width machined from Nylon 6/6 (impact modified), UHMWPE and polyacetal were

(continued on page 26)

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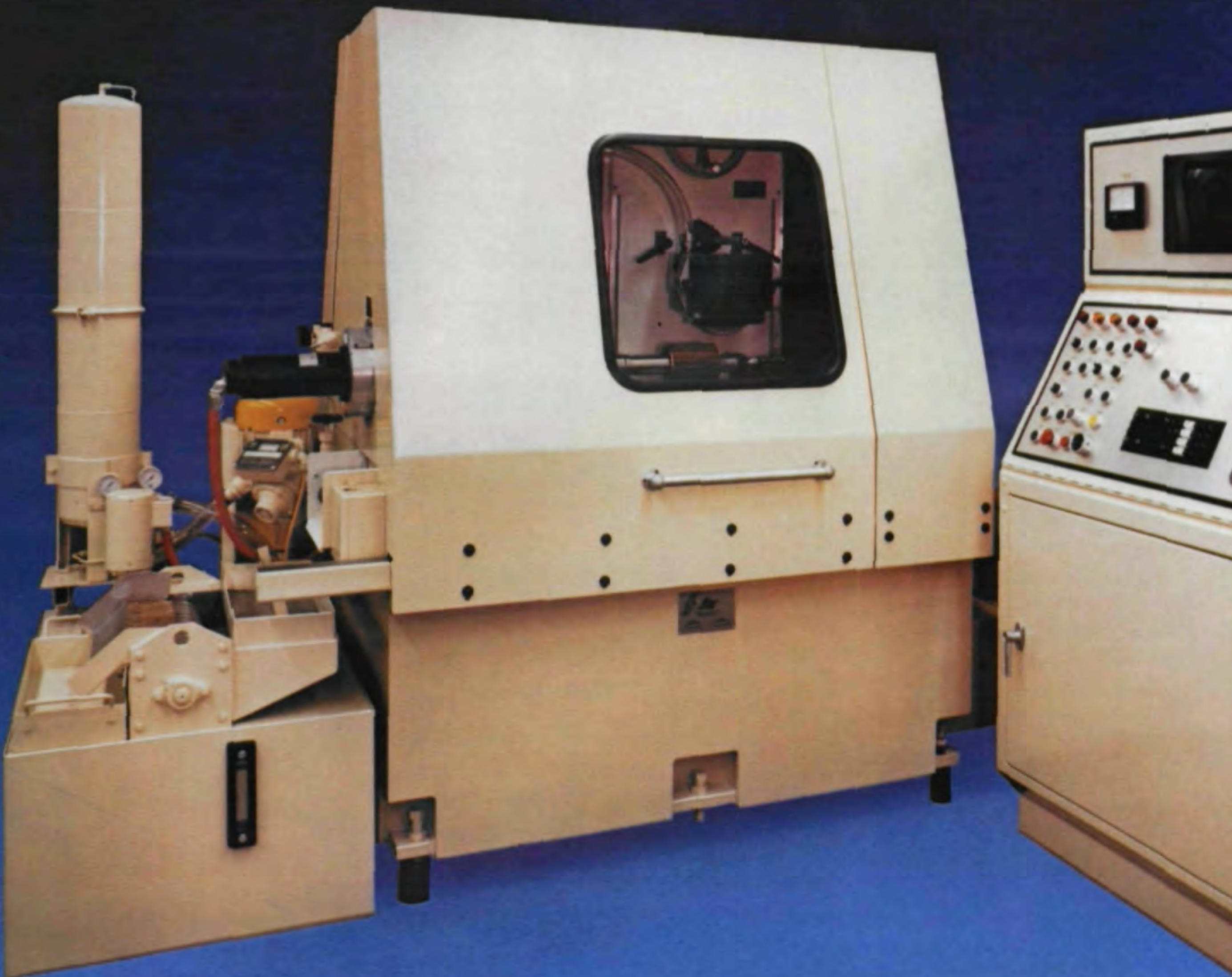

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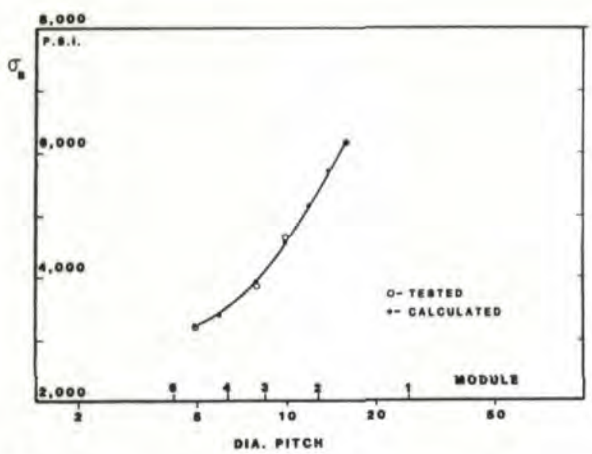

Fig. 4 – Bending fatigue stress of molybdenum disulphide filled cast nylon 6 gears vs. diametral pitch and module, oil jet lubricated.

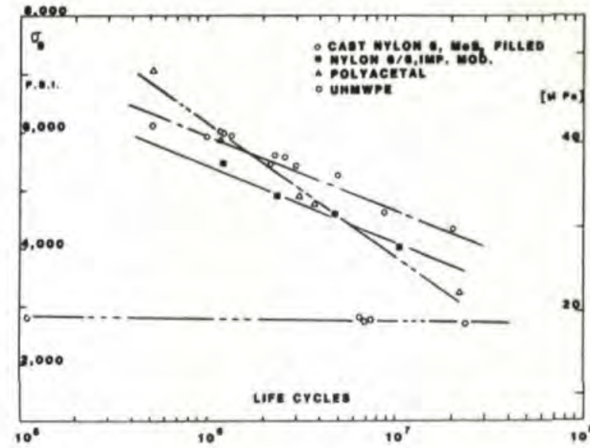

Fig. 5 – S-N curves of some thermoplastic gear materials, oil jet lubricated.

run at 2000 ft./min. for the endurance test and the results including those of cast Nylon 6 are illustrated in Fig. 5. The results of regression analysis of S-N curves are shown in Table 4. Root stresses at life cycles of 10×10^6 are 4650, 4084, 3856 and 2730 psi for cast Nylon 6, impact modified Nylon 6/6, polyacetal, and UHMWPE, respectively. A very moderate decline in the slope of S-N curve of UHMWPE indicates the material is extremely fatigue resistant, while the steep decline of the polyacetal curve indicates sensitivity of the material to fatigue. All coefficients of correlation for different thermoplastic gear materials are close to ± 1.0, which illustrates good approximation of S-N curves to straight lines. The small deviations are partially due to the high contact ratios of the gears. The slope of the S-N lines varies considerably with material.

Effects of Lubrication and Life Cycles

Fatigue root bending stress and life cycles of oil jet lubricated, initial greasing only and unlubricated (dry) cast nylon 6 gears are depicted in Fig. 6. The S-N curves form straight lines in the range of long cycle fatigue failures.[11] The results of regression analysis are given in Table 4. Test points of oil lubricated and initially greased gears form approximate parallel lines and they show small deviations from each of the straight lines, while a much larger scattering of the points resulted with dry gears. Under dry conditions, the slope of the S-N line toward longer life cycles is steeper than those of two other lubricated types.

As indicated in Table 4, the root stresses of initially greased and unlubricated cast nylon 6 at life cycles of 10 million are

TABLE 4.
S-N CURVE REGRESSION ANALYSIS OF SOME THERMOPLASTIC GEARS

Gear Materials	Lubrication	Root Stress, psi at Life Cycles of 10×10^6	Slope S-N Line psi	Coefficient of Correlation
Cast Nylon 6	Oil jet	4650 (100%)	−1240	−.9723
Nylon 6/6,	Oil jet	4084 (87.8%)	−1490	−.9973
Polyacetal	Oil jet	3856 (82.9%)	−2324	−.9949
UHMWPE	Oil jet	2730 (58.7%)	−80	−.9978
Cast Nylon 6	Initial greasing	3380 (72.7%)	−1063	−.9929
Cast Nylon 6	Unlubricated	1810 (38.9%)	−1659	−.9334

TABLE 5.
LIFE FACTOR K_1 OF MoS_2 FILLED CAST NYLON 6 SPUR GEARS EXTERNALLY LUBRICATED

	Diametral pitch or module			
No. of Cycles	16 P 1.6 m	10 P 2.5 m	8 P 3.2 m	5 P 5.1 m
1 million	1.26	1.24	1.30	1.22
10 million	1.0	1.0	1.0	1.0
30 million	0.87	0.88	0.89	0.89

72.7% and 38.9% of that of oil jet lubricated gears. These values can be used as lubrication factors.

Life factor K_1 of greased and oil lubricated cast nylon 6 spur gears at 1, 10 and 30 million cycles are shown in Table 5.

Effects of Pitch Line Velocity

10P cast nylon gears were examined for fatigue strength at pitch line velocities of 680, 1200, 2000 and 4000 fpm (3.5, 6.1, 10.2 and 20.3 m/s) under initially greased conditions. Test points of each velocity can be connected as straight lines (Fig. 7). S-N lines of 680, 2000 and 4000 fpm are close to parallel. Root fatigue stress of 1200 fpm at 2 to 4 million cycles are higher than those run at 680 fpm. This may be related to properties peculiar to the test machine, such as natural resonance. At 10 million life cycles and over, where the gears are rated, the root stress appears to correlate with other tested pitch line velocity.

Relative root stress at 20 million cycles compare gear strength run at different pitch line velocities. Root stresses illustrated in Figs. 2 (except as noted), 4 and 5 are obtained at pitch line velocity of 2000 fpm. Correction factors for pitch line velocities are listed below:

Pitch Line Velocity, fpm	680	1200	2000	4000
Root Bending Stress, psi	3856	3442	3030	2851
Velocity Correction Factor	1.27	1.14	1.00	0.94
Calculated Factor	1.27	1.11	1.00	0.92

The velocity correction factors can be calculated approximately by

$$K_v = 394/(200 + v) + 0.825 \qquad \text{Eq. (3)}$$

where v = Pitch line velocity, fpm
$= .262 \times D_p\, n$

Thermoplastics Gear Design – A Summary

The test results of fatigue bending stress described are used to obtain safe tangential force, safe torque and horsepower capacity of thermoplastic gears under various operational conditions. Conversely, knowing the required torque or horsepower to be transmitted, a gear design under specified conditions can be made.

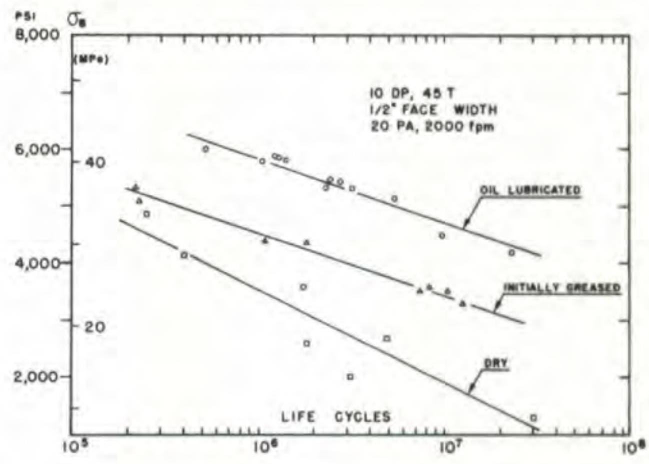

Fig. 6 – S-N curves of filled cast nylon 6 gears effected by types of lubrication (9).

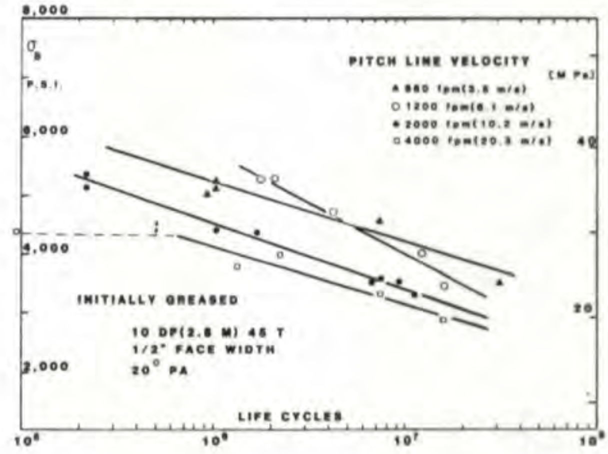

Fig. 7 – S-N curves of filled cast nylon 6 gears using different pitch line velocity as parameter.

Fatigue bending stresses under different test parameters are illustrated in Figs. 2, 4, 5, 6 and 7 and Table 4 using Eq. (1).

1. $\delta\beta$ – Fatigue Bending Stress

$$\delta\beta = \frac{F\,P}{f\,y}$$

2. F_t – Safe Tangential Force

$$F_t = \frac{S_{at}}{P} f\, y\, L_u\, K_v\, K_1 \qquad \text{Eq. (4)}$$

3. T – Safe Torque

$$T = \frac{D_p}{2} F_t \qquad \text{Eq. (5)}$$

4. HP – Horsepower Capacity

$$HP = \frac{F_t v}{33{,}000} \text{ or } \frac{T_n}{63{,}000} \qquad \text{Eq. (6)}$$

In obtaining the fatigue bending stress and velocity factor of gears, interpolation of the results using Fig. 5 and Eq. (3)

are permissible. However, extrapolation of the results to untested pitch as well as velocity ranges can be very misleading.

To determine the allowable bending stress S_{at} in Eq. (4), 75% of fatigue root bending stress is used for externally lubricated cast nylon 6 and impact modified Nylon 6/6. Tooth form factor y, in Table 3, is used for pressure angle or tooth form variables. For initial greased unlubricated cast nylon 6 (filled with molybdenum disulphide) gears lubrication factors L_u of 72.7% and 28.9%, respectively, are to be applied. Life factors in Table 5 apply to grease and oil jet lubricated cast nylon 6 spur gears.

NOMENCLATURE

D_p = pitch diameter
F = tangential force at pitch circle
f = gear face width
F_t = safe tangential force
HP = horsepower cavity
K_1 = life factor
K_v = velocity factor
L_u = lubrication factor
m = module
n = gear rotational speed, rpm
P = diametral pitch
S-N = root stress vs. life cycle
S_{at} = allowable root bending stress
T = safe torque
v = pitch line velocity
y = tooth form factor for load applied near pitch circle
δ_B = root fatigue bending stress

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This paper was previously presented at the 1984 technical meeting of the American Society of Mechanical Engineers.

Gear Inspection and Chart Interpretation

by
Robert H. Moderow
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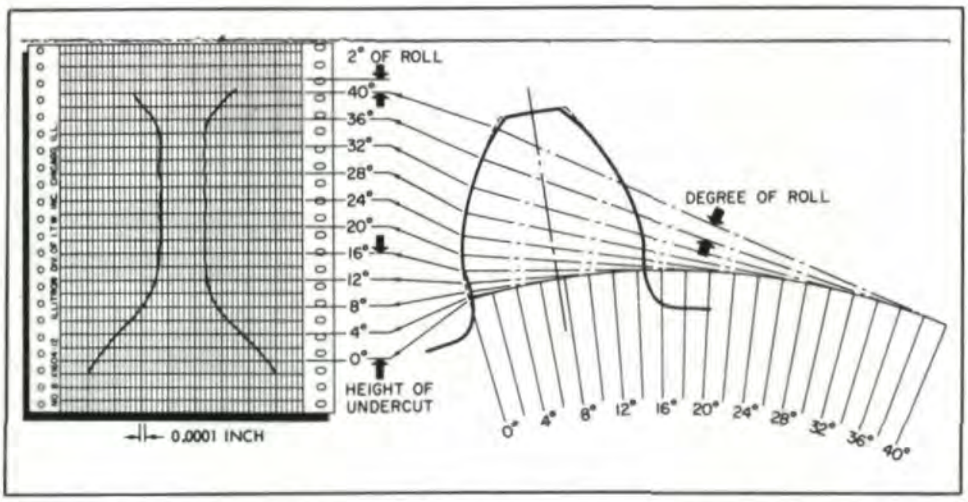

Fig. 1 – Involute charts were made on a variable involute measuring instrument and charted

Much information has been written on gear inspection, analytical, functional, semiautomatic and automatic. In most cases, the charts, (if you are lucky enough to have recording equipment) have been explained. If there is an error in the gear, however, there is little information to tell you what the causes of the errors might be, or to offer some idea on how to correct them. This article attempts to offer some rudiments of "what to do next".

Figs. 1-3 show typical involute and lead charts, as well as the methods used to check runout and spacing on all the Analytical Gear Checks shown in this presentation. These charts can be used as a guide to test results of a normal, error-free gear.

A typical functional rolling chart is shown in Fig. 4. Runout and wobble can be readily found from this type of check, but involute, spacing and lead errors are difficult to find. The tooth-to-tooth composite variation is *not* the part's spacing errors, but is a composite of lead, involute and spacing errors, as well as nicks, etc. Wobble or face runout will show up as a single or double runout pattern. However, sometimes the pattern of the error in the functional check can lead you to the cause of the error. Fig. 5 could be of some help and is self-explanatory.

Runout

Fig. 6 shows all the analytical checks (lead, involute, spacing and runout) on a given gear, when the gear is mounted and inspected on the same center as it was machined. This particular gear is of very good quality, AGMA Class 12 or 13. Note that the gear was checked in three different positions, and all the checks are basically alike. In order to demonstrate the chart readings indicating runout, the gear was then mounted with runout and checked again. The results are shown in Fig. 7. All three checking positions gave basically the same checks, but the runout shows up, as it should, and this runout affects the tooth spacing and the involute. The lead is not affected, and would not be, unless the gear had lead and involute modifications, or was helical. (Note – runout does affect the gear's involute, but this error shows up more on small numbers of teeth. Also, the involute varies from plus to minus, depending on the tooth checked.)

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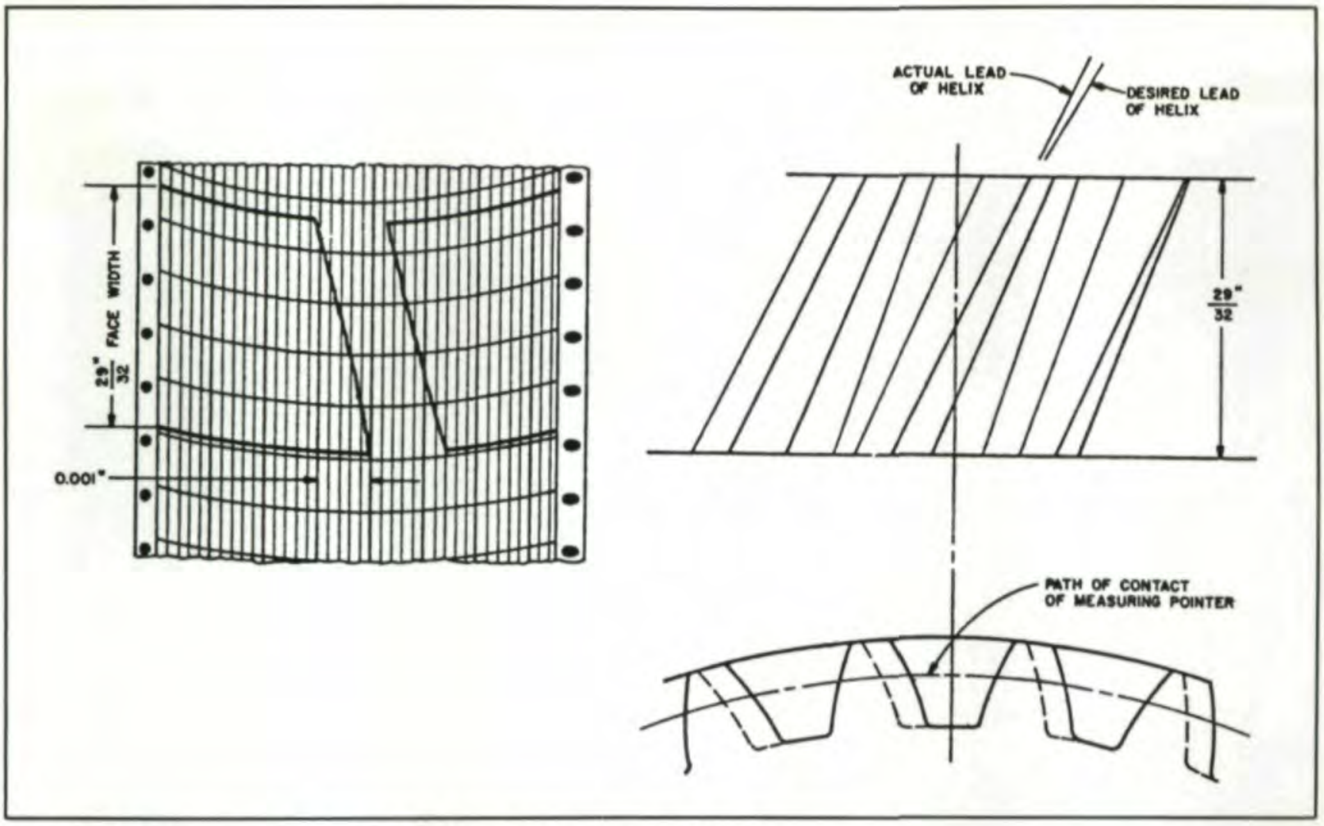

Fig. 2 – Lead checks were made on a helical lead measuring instrument and charted.

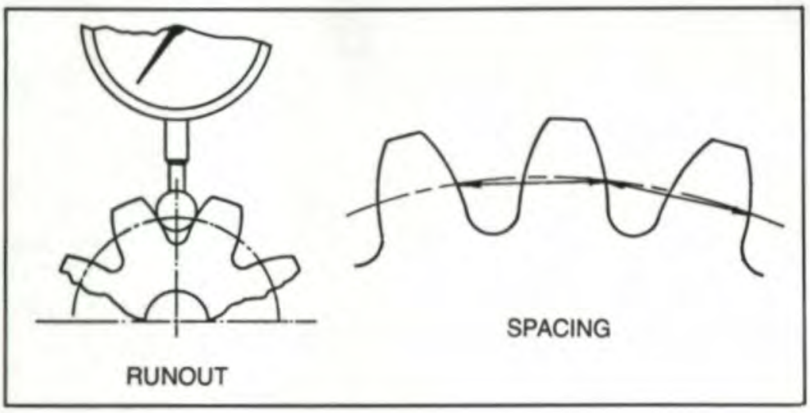

Fig. 3 – (Left center) Runout was checked using a ball and an indicator and charted. Tooth spacing was checked using a fixed finger and an indicating finger and charted.

Fig. 4A – (Below) Typical chart obtained when using a "Gear Roller".

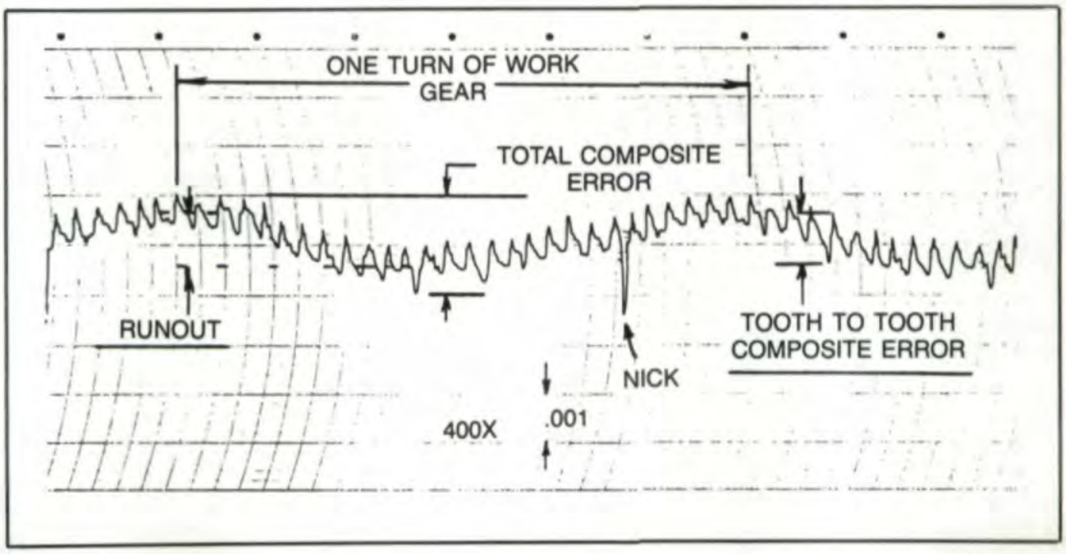

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Fig. 4B – Gear Rolling instrument

Wobble (Face Runout)

Fig. 8 shows the gear mounted with wobble (face runout) and inspected again at the three positions. This time, none of three checking positions gave charts which were alike. This depends upon the location of the pivot point of the wobble. Notice that a gear with wobble shows errors in all analytical checks, but they are of different amounts depending upon where the check is made relative to the wobble pivot point. Again, variation in the involute shows both plus and minus, but shows up more on small numbers of teeth, as in this test gear. The lead check shows variation (plus and minus depending on the tooth checked), and it is very typical of a gear with a wobble condition.

The charts which show variation (plus and minus, depending on the tooth checked) give you a very good indication of just why a gear should be checked by testing four teeth 90° apart. If this were not done, a bad gear could easily be passed as good by Inspection.

Studying these charts may help you to determine where to look when errors occur. For example, if a gear has runout, perhaps the work or inspection arbor is undersize, the blank hole is oversize, etc. If the gear has wobble, the mounting (machining or inspection), faces may not be square, work spacers not parallel (springing arbor), or the live center is running out. If you are stacking gears when cutting, and some gears check better than others, it may be that the good ones are at the center of pivot in a wobble condition. Also when stacking gears, if the blanks do not have parallel sides, one stack of gears may be entirely different from another. Subsequently, each load must be checked.

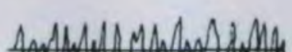
Fig. 5 – Machine should be checked for accuracy at least twice a year by hobbing a gear with exactly the same number of teeth as the worm wheel on the machine. This should be done with a single thread AA hob very carefully spacing and runout and the resulting errors will indicate the area of the machine which should be repaired.

Uniform Error Tooth to Tooth

Hob not accurate enough.
Hob not running true.
(Try shifting hob ⅓ to ½ pitch).

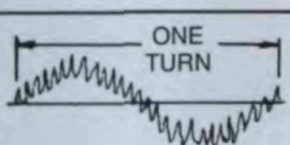
Machine

Last output gears driving hob are worn or running out. Hob spindle end thrust bearings are loose.



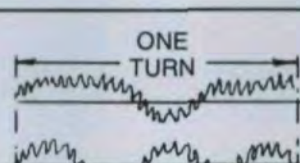
Non-Uniform Error Tooth to Tooth

Worm wheel on machine worn. Change gear or gears have excessive error.



Uniform P.D. Runout Error

Gear blank running out or has wobble. Worm wheel worn or running out. (Check arbor for deflection when arbor nut is tightened).



Non-Uniform P.D. Runout Error

Worn worm wheel or change gears.

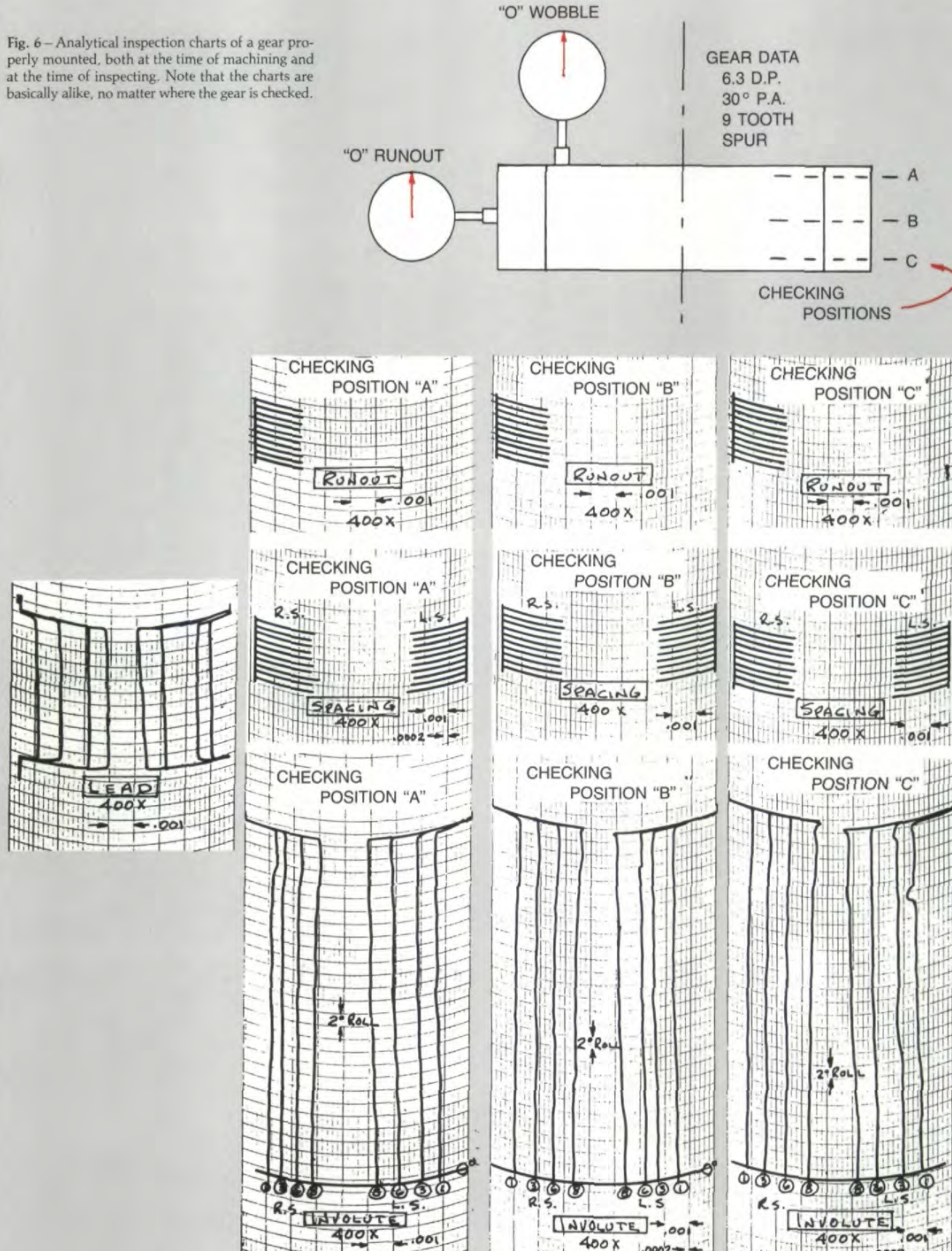


Fig. 6 – Analytical inspection charts of a gear properly mounted, both at the time of machining and at the time of inspecting. Note that the charts are basically alike, no matter where the gear is checked.

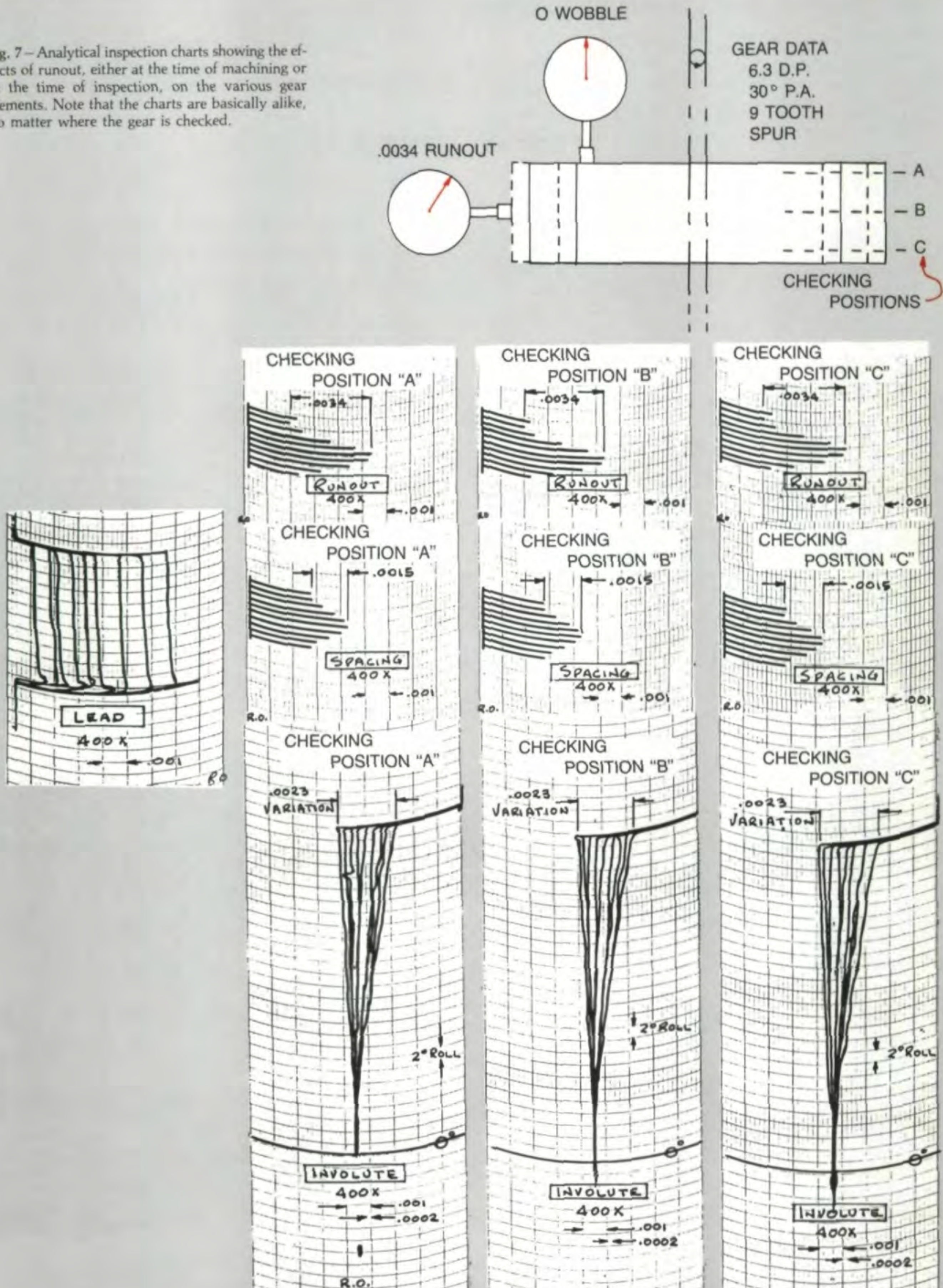
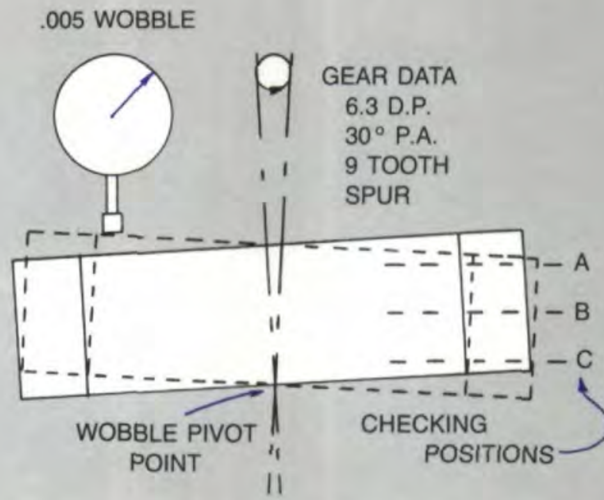
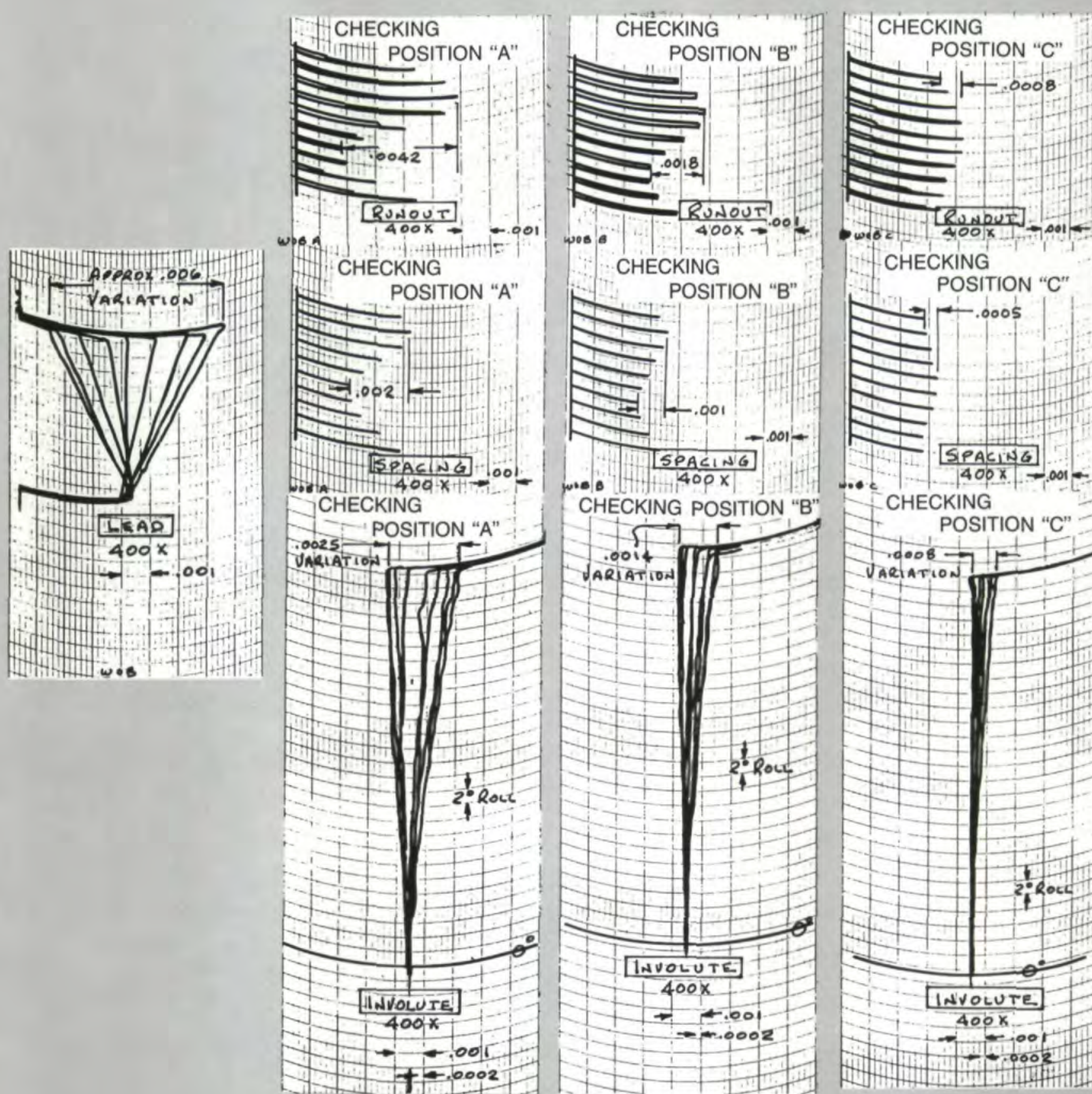


Fig. 7 – Analytical inspection charts showing the effects of runout, either at the time of machining or at the time of inspection, on the various gear elements. Note that the charts are basically alike, no matter where the gear is checked.

Fig. 8 – Analytical inspection charts, showing the effects of wobble, either at the time of machining or the time of inspection on the various gear elements. Note that the charts are different, depending on the position of the check, relative to the pivot point of the wobble.





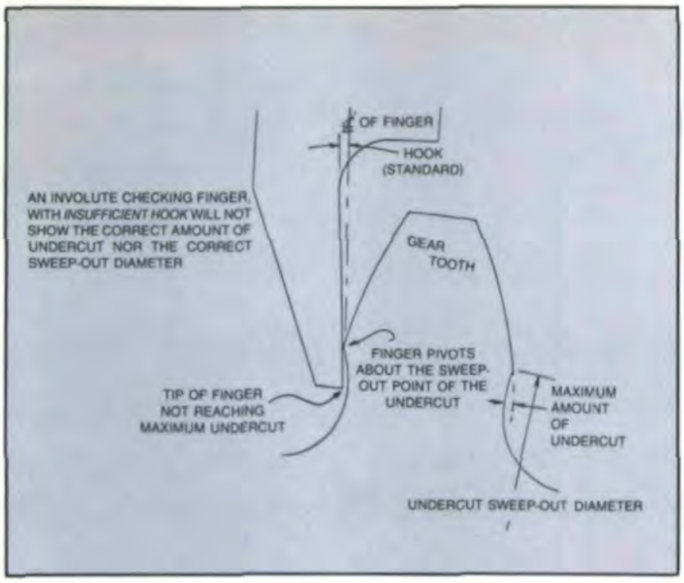


Fig. 9A

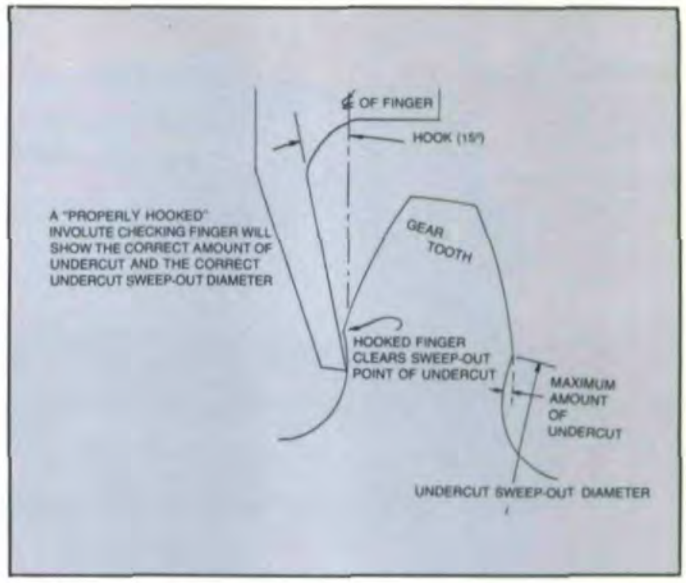


Fig. 9B

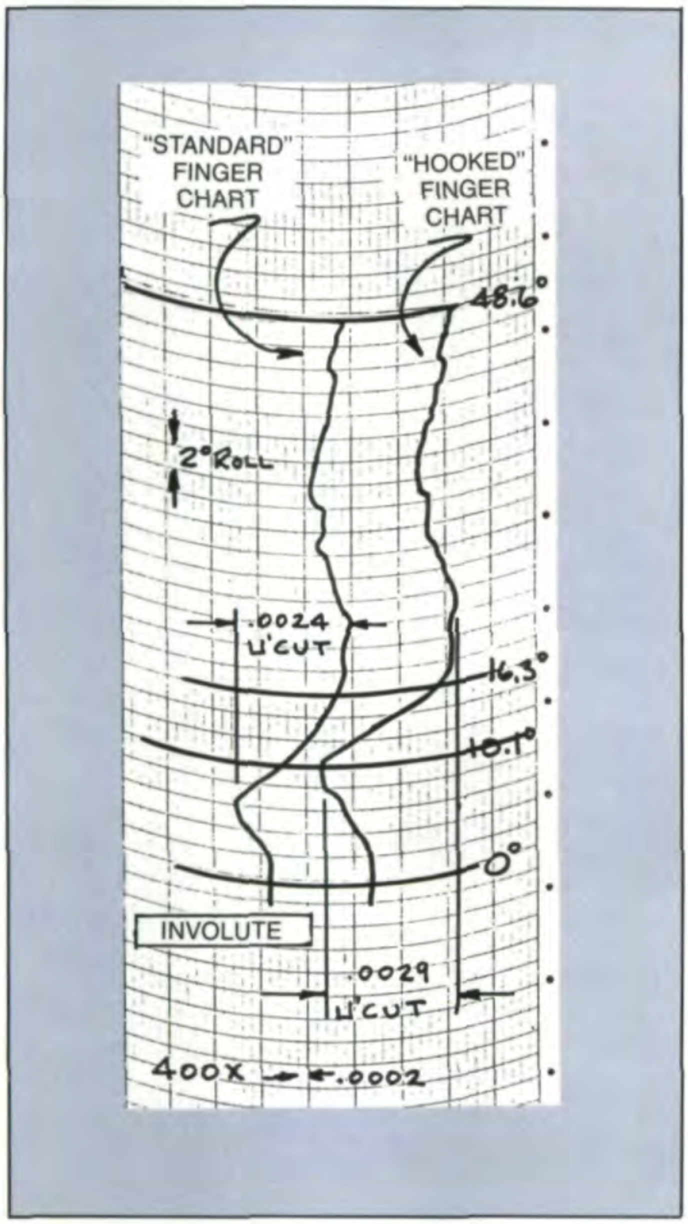


Fig. 9C

Checking Undercut Gears

Fig. 9 illustrates a very common problem in checking the involute on gears that have undercut, such as pre-shaved, pre-rolled, pre-grind. In all of the previous gear types, the position and the amount of undercut is very important. Not only should the undercut be positioned properly in the flank of the gear tooth, to assure true involute to the last point of contact of the mating gear, but the amount must also be correct to avoid the possibility of shaving or grinding nicks or steps in the fillet area. You are getting an incorrect picture of the gear tooth's undercut, if the involute checking finger is not "hooked" sufficiently; the pictures and the resultant charts are self-explanatory. Fig. 9B shows a 15° hooked finger, would probably be necessary only for gears with a large amount of undercut, such as pre-grind. In most cases, a 10° hook would be sufficient.

CORRECTION

John C. Leming, V.P. Engineering Arrow Gear Co., Downers Grove, IL was not listed as co-author of "GEAR GRINDING TECHNIQUES PARALLEL AXES GEARS", which appeared in the March/April 1985 issue of GEAR TECHNOLOGY. Illustrations for that article were provided by the courtesy of: American Pfauter Company, Kapp and Company, U.S. Representatives, Elk Grove Village, Illinois; Barber Colman Company, Okamoto Company, U.S. Representatives; BHS Hofler, Clinton, New Jersey, Maag Gear-Wheel Company Ltd., Zurich, Switzerland; National Broach & Machine Division, Clemens, Michigan; Reishauer Corporation, Elgin, Illinois.

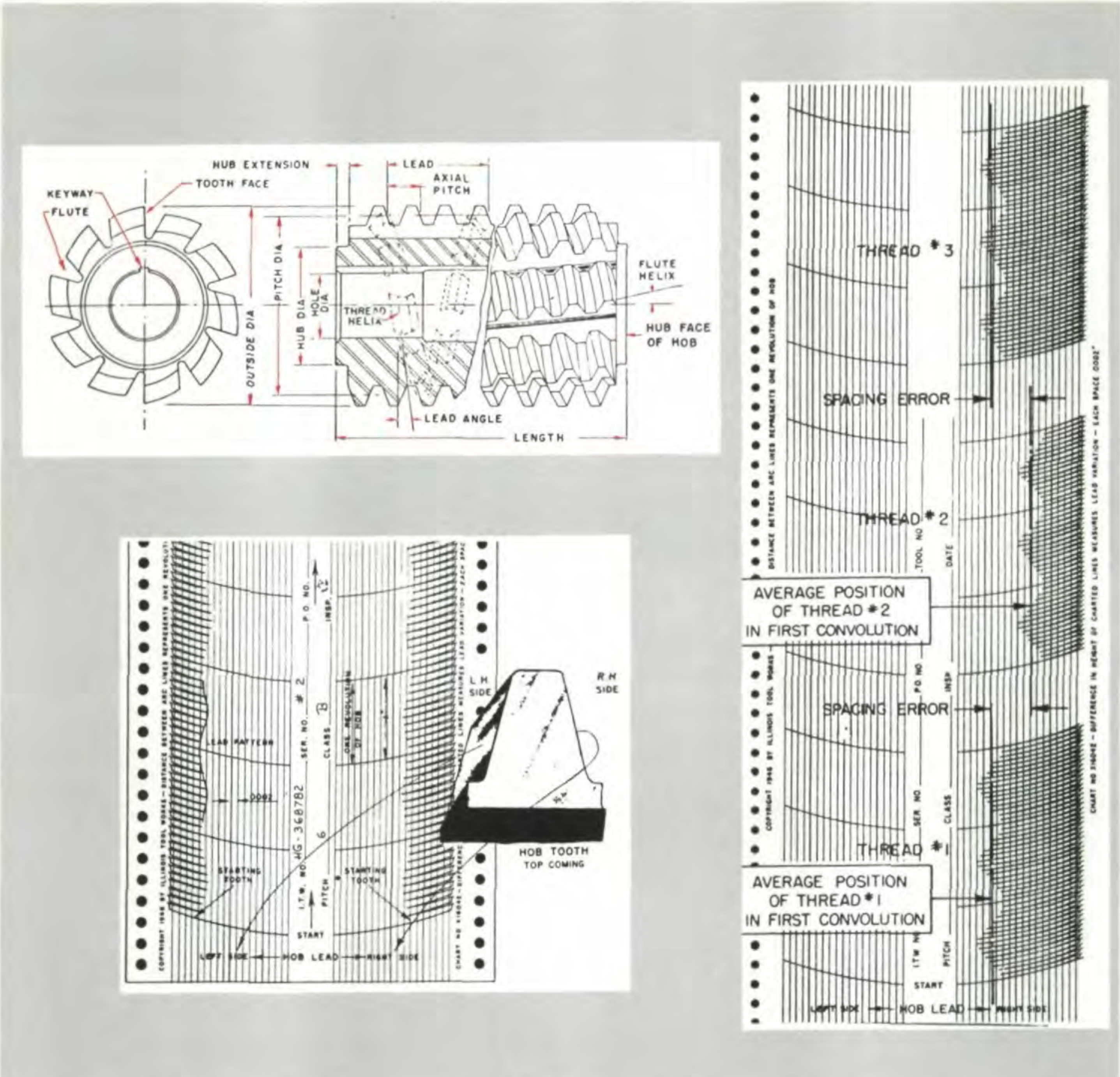


Fig. 10

Hob Errors

The lead chart of a hob is an indication of the accuracy of the helical path of a hob's teeth. A single thread hob can affect only the profile of the gear tooth. If the hob has a good lead, the gear's profile can be good. If the lead is bad, the profile will most likely be bad. Fig. 10 shows a hob lead chart.

The charts shown in Fig. 11 illustrate the problems which result when mounting a hob incorrectly. Number 1 is a very good hob mounted properly and the resulting gear involute produced. Numbers 2-4 show the same hob mounted incorrectly, with the resulting effect on the hob's lead chart (checked as mounted) and on the involute form produced on the gear.

As a hob cuts a gear, the top of the gear's involute form is finish cut at a different point in time than the bottom. Because of this, when hobbing with heavy feed rates, the entire involute form will tend to drift. (Fig. 12) The resulting involute checks may vary on the same tooth across the face of the gear, depending upon just where the involute checking finger is positioned. This is not to say there is an error here, but just to explain why two involute checks may vary when checking the same tooth.

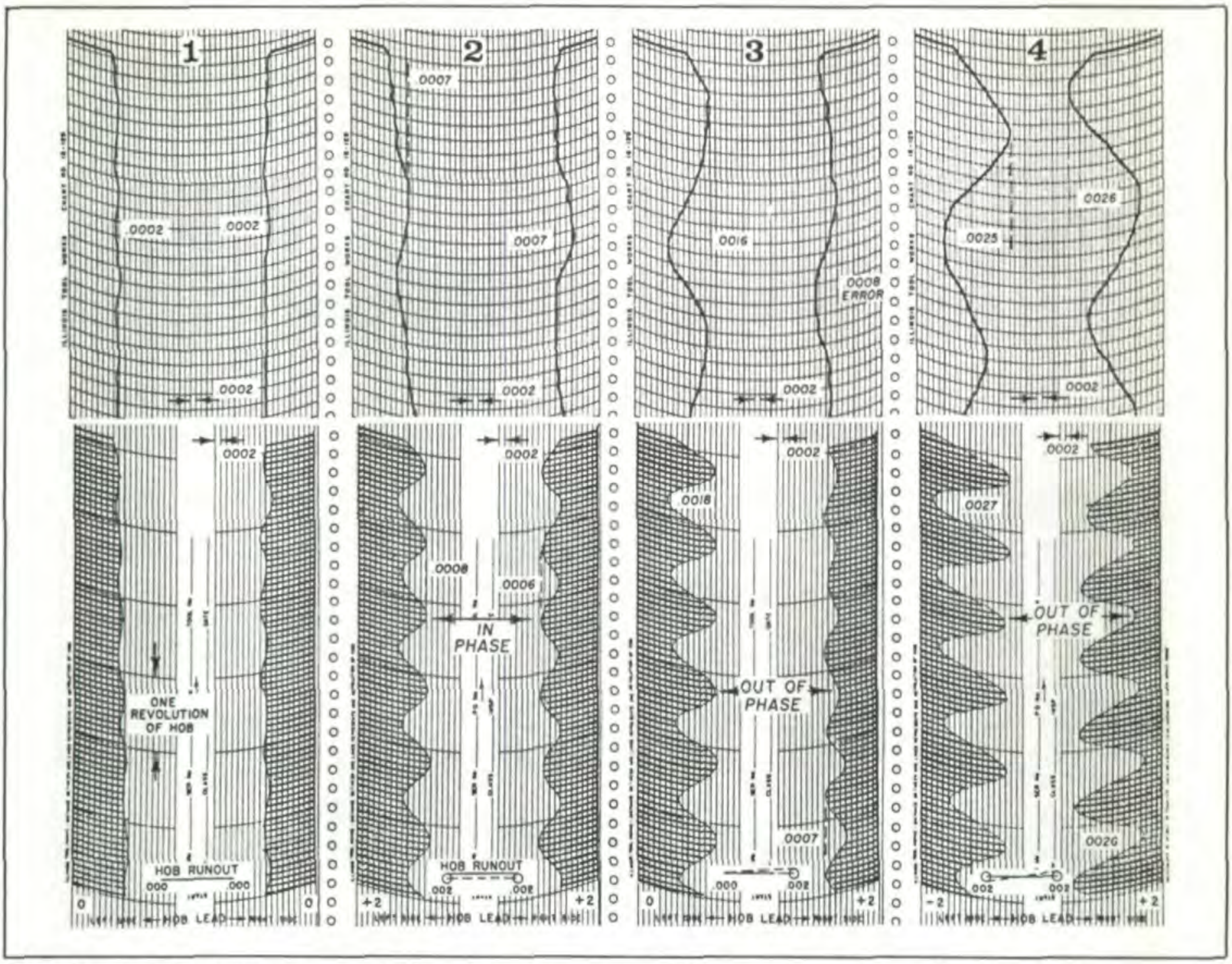


Fig. 11 – These charts show the types of weave produced by various conditions of runout and the amount of error each type produces in a gear tooth.

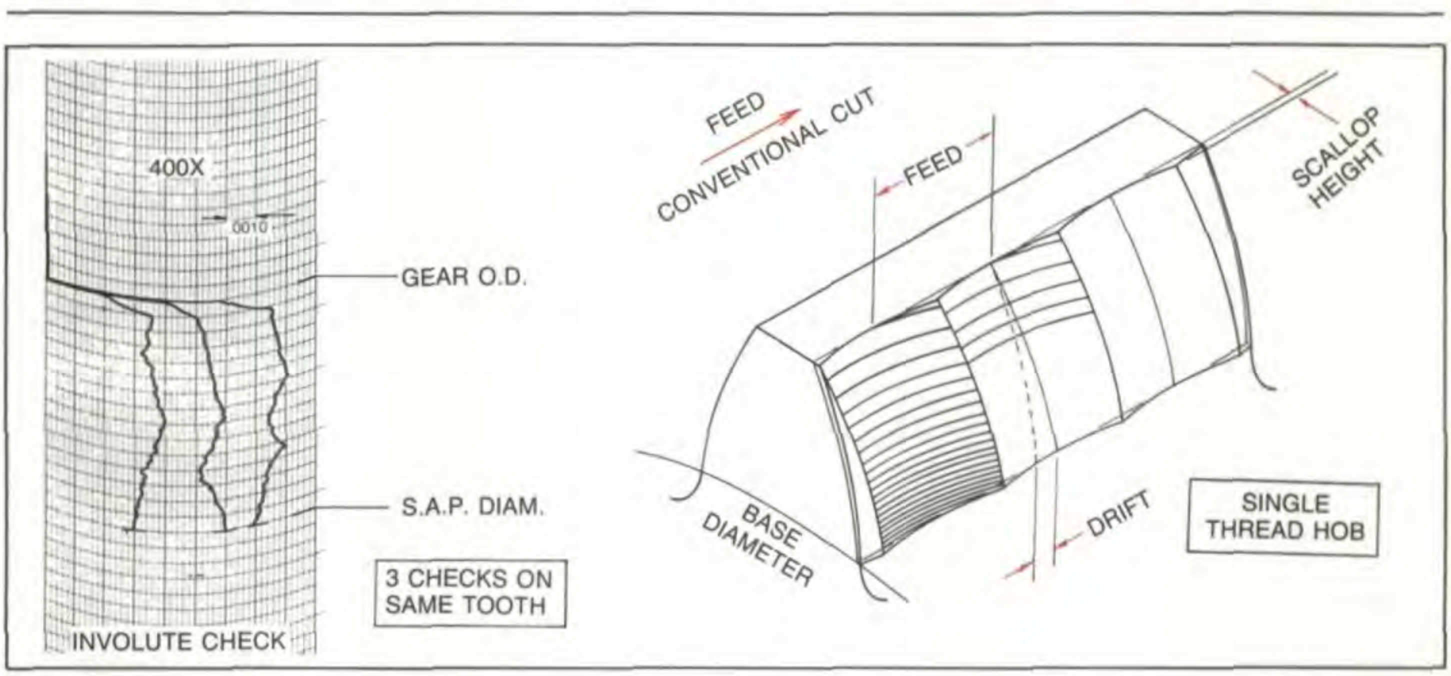


Fig. 12 – Involute checks on the same tooth vary if the gear is hobbed with a heavy feed rate, the involute drifts and one check may high point of feed scallop at pitch line, and another check may have high point at another place along the involute.

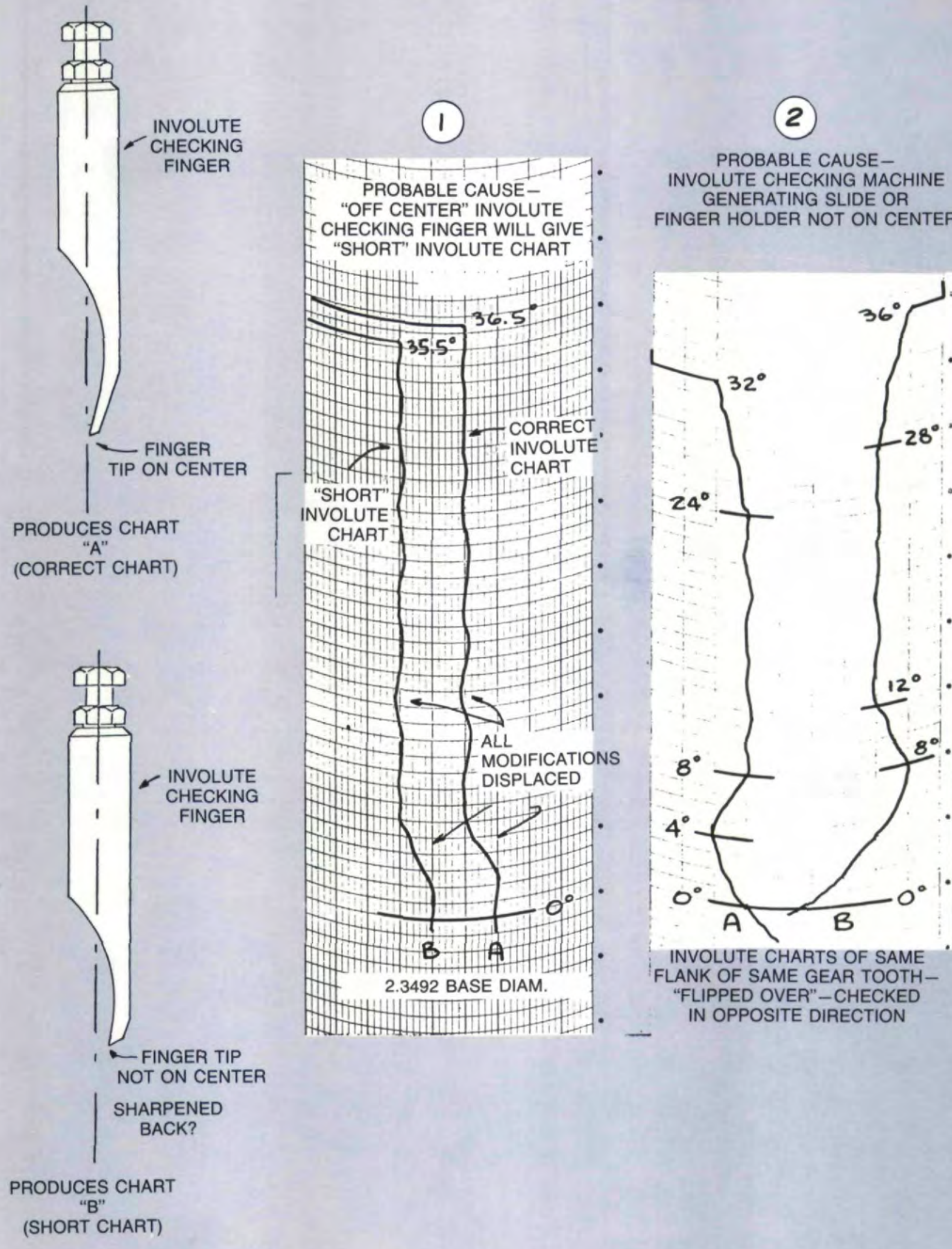

INVOLUTE CHECKING FINGER
FINGER TIP ON CENTER
PRODUCES CHART "A" (CORRECT CHART)
INVOLUTE CHECKING FINGER
FINGER TIP NOT ON CENTER
SHARPENED BACK?
PRODUCES CHART "B" (SHORT CHART)
1
PROBABLE CAUSE— "OFF CENTER" INVOLUTE CHECKING FINGER WILL GIVE "SHORT" INVOLUTE CHART
36.5°
35.5°
CORRECT INVOLUTE CHART
"SHORT" INVOLUTE CHART
ALL MODIFICATIONS DISPLACED
0°
B
A
2.3492 BASE DIAM.
2
PROBABLE CAUSE— INVOLUTE CHECKING MACHINE GENERATING SLIDE OR FINGER HOLDER NOT ON CENTER
36°
32°
28°
24°
12°
8°
8°
4°
0°
0°
A
B
INVOLUTE CHARTS OF SAME FLANK OF SAME GEAR TOOTH— "FLIPPED OVER"—CHECKED IN OPPOSITE DIRECTION

Fig. 13

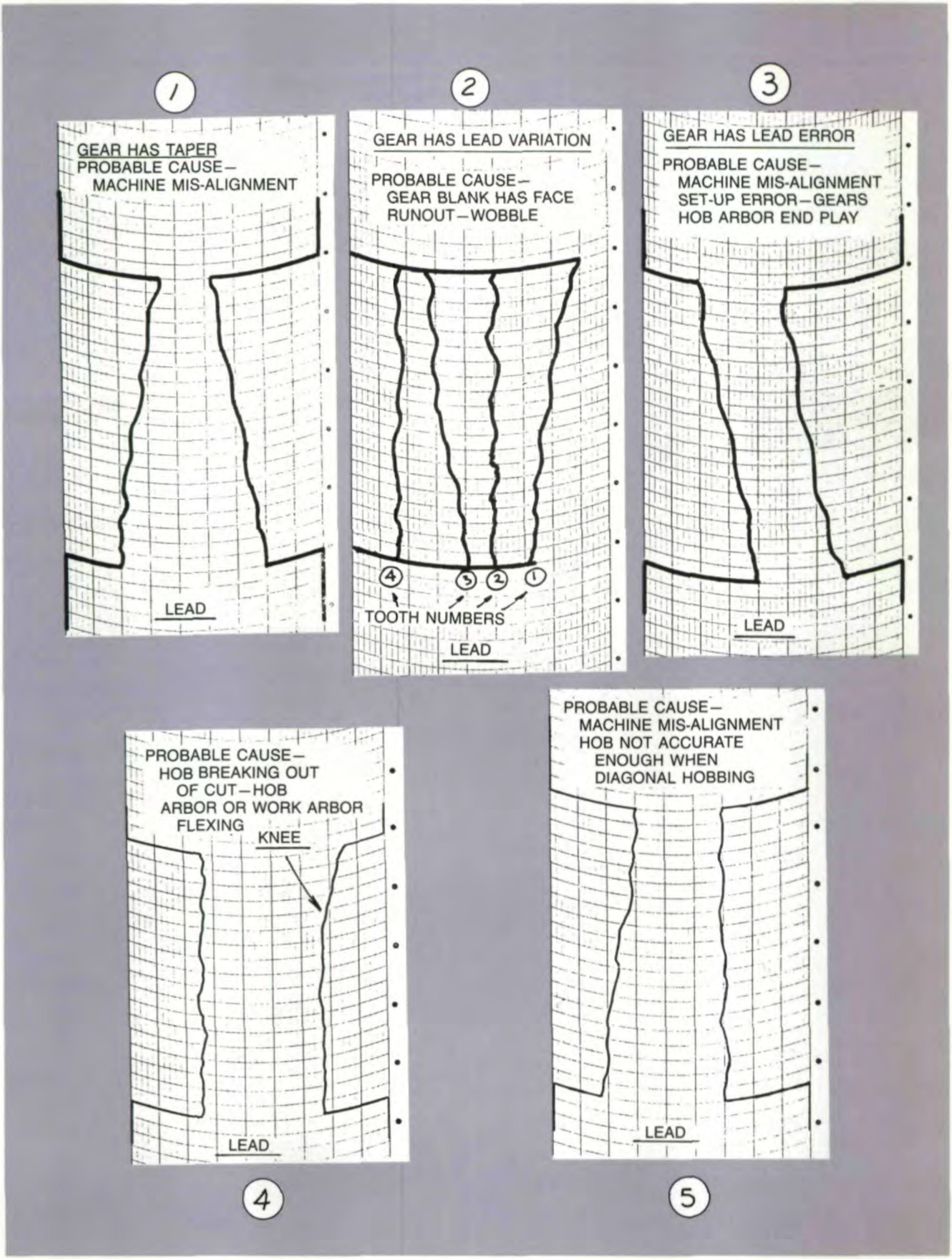

1
GEAR HAS TAPER
PROBABLE CAUSE–
MACHINE MIS-ALIGNMENT
LEAD
2
GEAR HAS LEAD VARIATION
PROBABLE CAUSE–
GEAR BLANK HAS FACE
RUNOUT–WOBBLE
4
3
2
1
TOOTH NUMBERS
LEAD
3
GEAR HAS LEAD ERROR
PROBABLE CAUSE–
MACHINE MIS-ALIGNMENT
SET-UP ERROR–GEARS
HOB ARBOR END PLAY
LEAD
PROBABLE CAUSE–
HOB BREAKING OUT
OF CUT–HOB
ARBOR OR WORK ARBOR
FLEXING
KNEE
LEAD
4
PROBABLE CAUSE–
MACHINE MIS-ALIGNMENT
HOB NOT ACCURATE
ENOUGH WHEN
DIAGONAL HOBBING
LEAD
5

Fig. 14

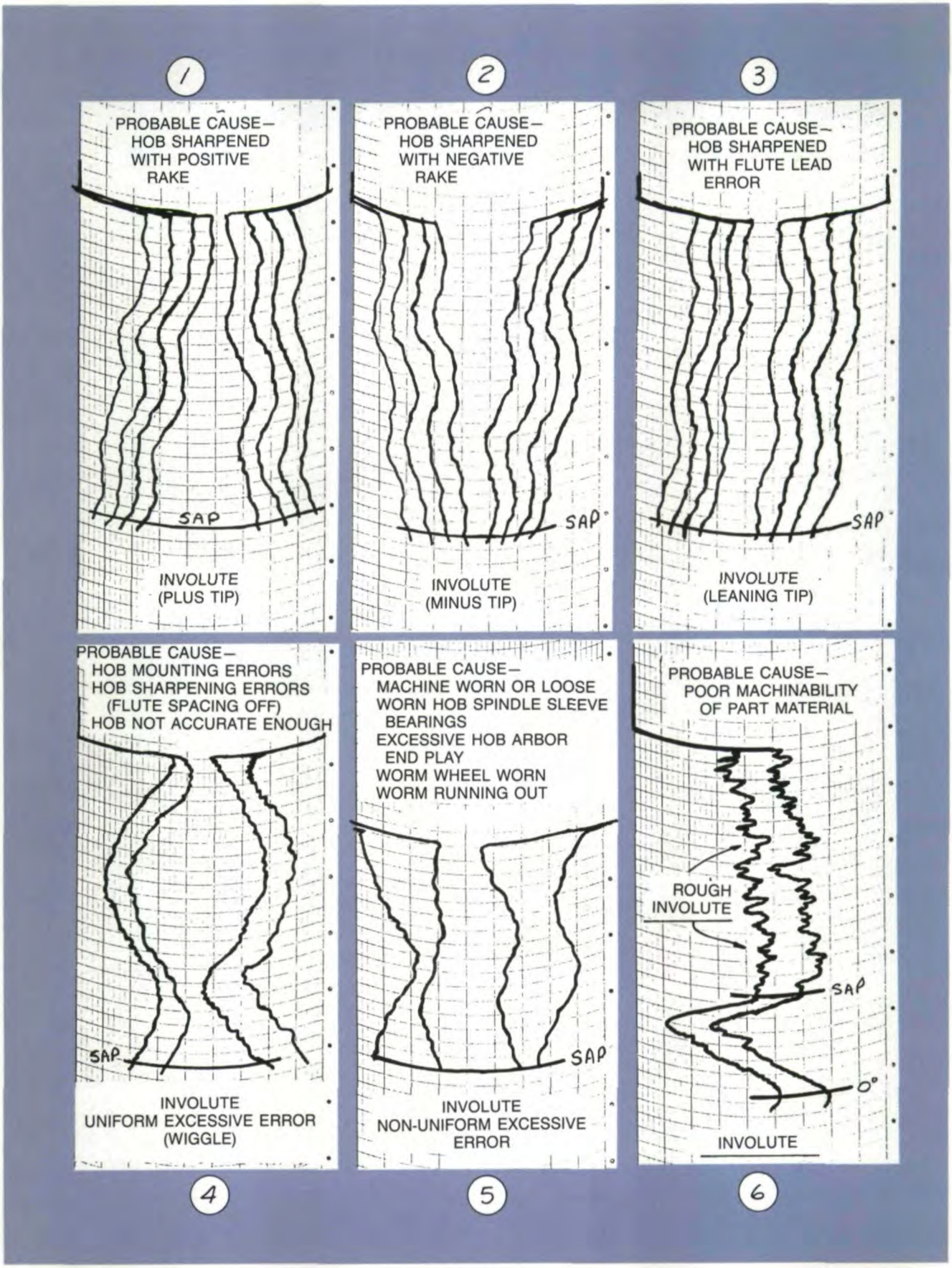
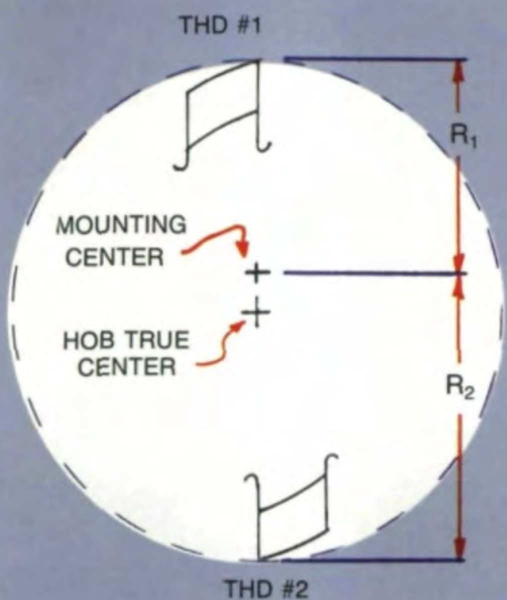

1
PROBABLE CAUSE–
HOB SHARPENED
WITH POSITIVE
RAKE
SAP
INVOLUTE
(PLUS TIP)
2
PROBABLE CAUSE–
HOB SHARPENED
WITH NEGATIVE
RAKE
SAP
INVOLUTE
(MINUS TIP)
3
PROBABLE CAUSE–
HOB SHARPENED
WITH FLUTE LEAD
ERROR
SAP
INVOLUTE
(LEANING TIP)
PROBABLE CAUSE–
HOB MOUNTING ERRORS
HOB SHARPENING ERRORS
(FLUTE SPACING OFF)
HOB NOT ACCURATE ENOUGH
SAP
INVOLUTE
UNIFORM EXCESSIVE ERROR
(WIGGLE)
4
PROBABLE CAUSE–
MACHINE WORN OR LOOSE
WORN HOB SPINDLE SLEEVE
BEARINGS
EXCESSIVE HOB ARBOR
END PLAY
WORM WHEEL WORN
WORM RUNNING OUT
SAP
INVOLUTE
NON-UNIFORM EXCESSIVE
ERROR
5
PROBABLE CAUSE–
POOR MACHINABILITY
OF PART MATERIAL
ROUGH
INVOLUTE
SAP
0°
INVOLUTE
6

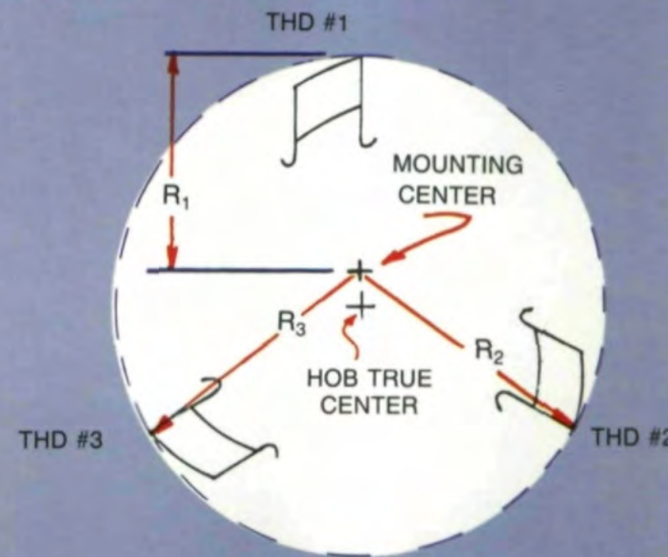
Fig. 15

HOB MOUNTING ERRORS
MULTIPLE THREAD



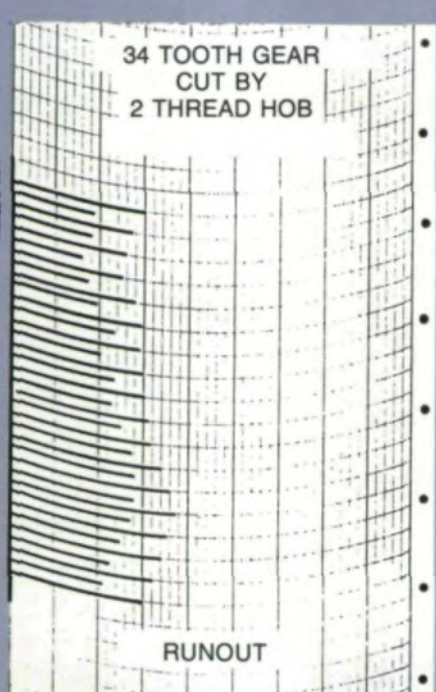
2 THREAD HOB

R_2 IS LARGER THAN R_1, AND WHEN CUTTING EVEN NUMBERS OF TEETH WILL CUT WIDER AND DEEPER TOOTH SPACES



3 THREAD HOB

R_2 AND R_3 ARE LARGER THAN R_1 AND WHEN CUTTING A NUMBER OF TEETH EVENLY DIVISIBLE BY 3, WILL CUT WIDER AND DEEPER TOOTH SPACES



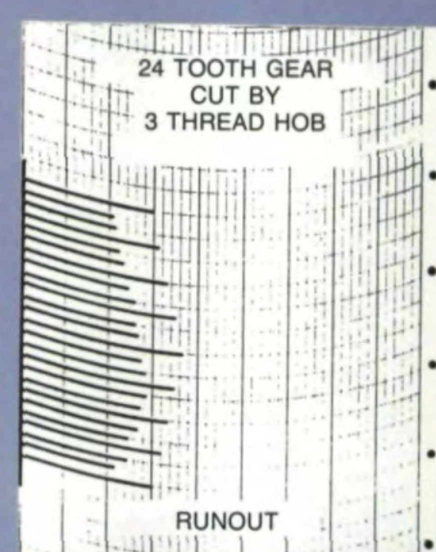
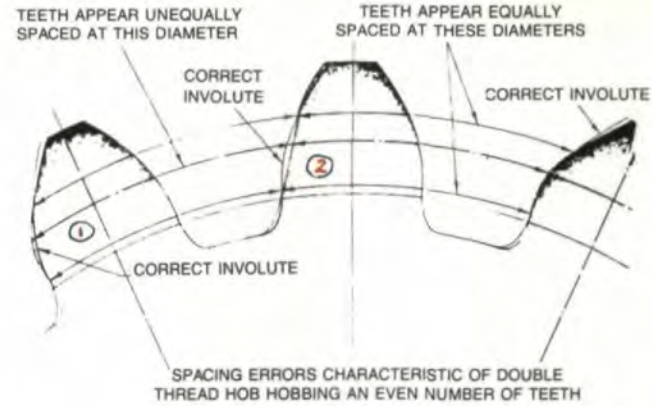
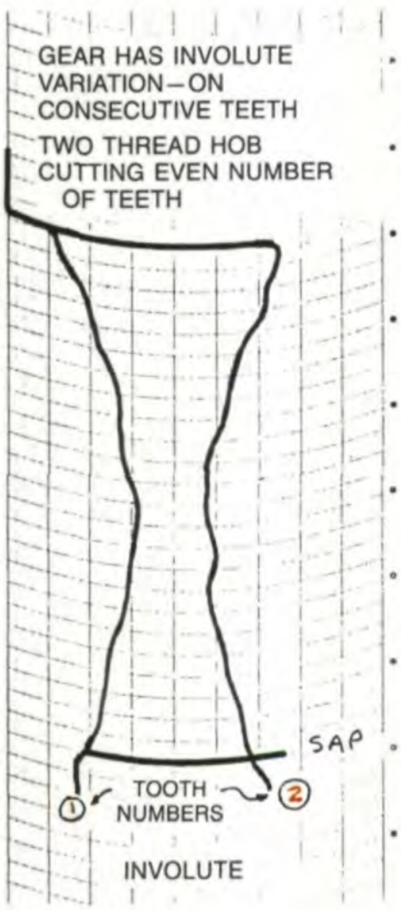
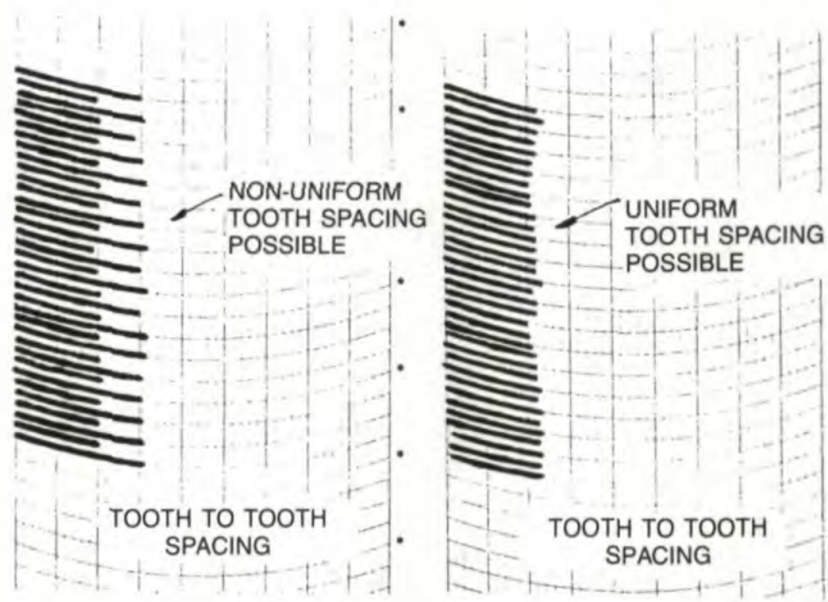


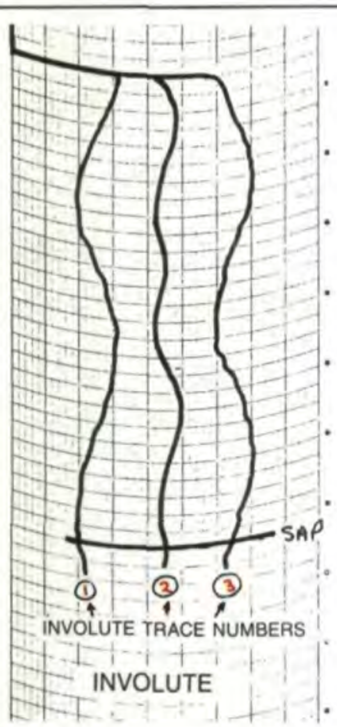
Fig. 16

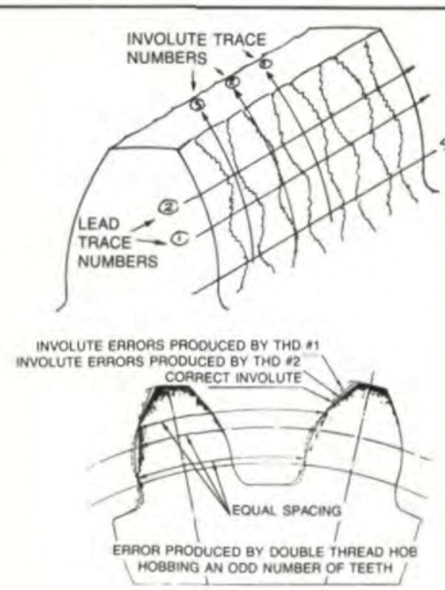
Fig. 17 – Possible cause:
1) Hob hubs running out – out of phase (high with low)
2) Hob has weave error in lead, out of phase, from thread to thread











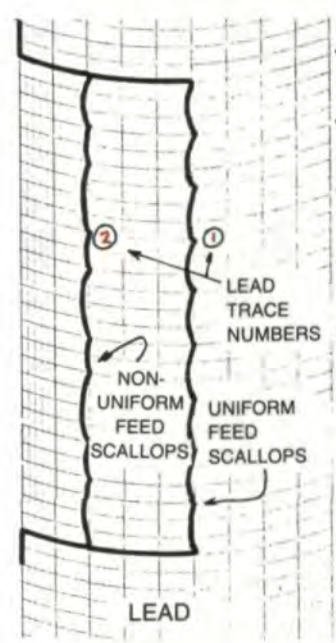


Fig. 18 – Possible cause:
1) Hob hub(s) running out
2) Hob has weave error in lead out of phase, from thread to thread

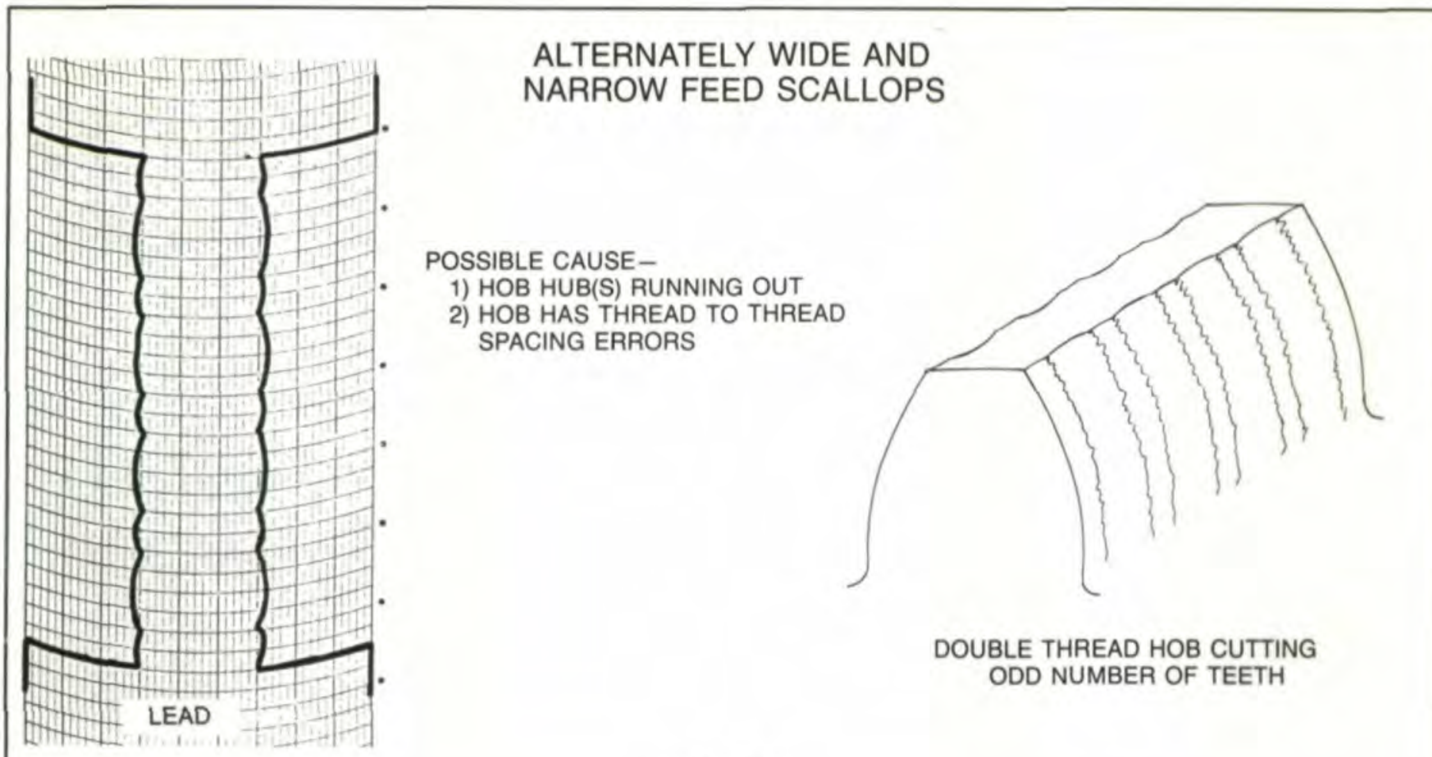

ALTERNATELY WIDE AND
NARROW FEED SCALLOPS
POSSIBLE CAUSE–
1) HOB HUB(S) RUNNING OUT
2) HOB HAS THREAD TO THREAD
SPACING ERRORS
DOUBLE THREAD HOB CUTTING
ODD NUMBER OF TEETH
LEAD

Fig. 19

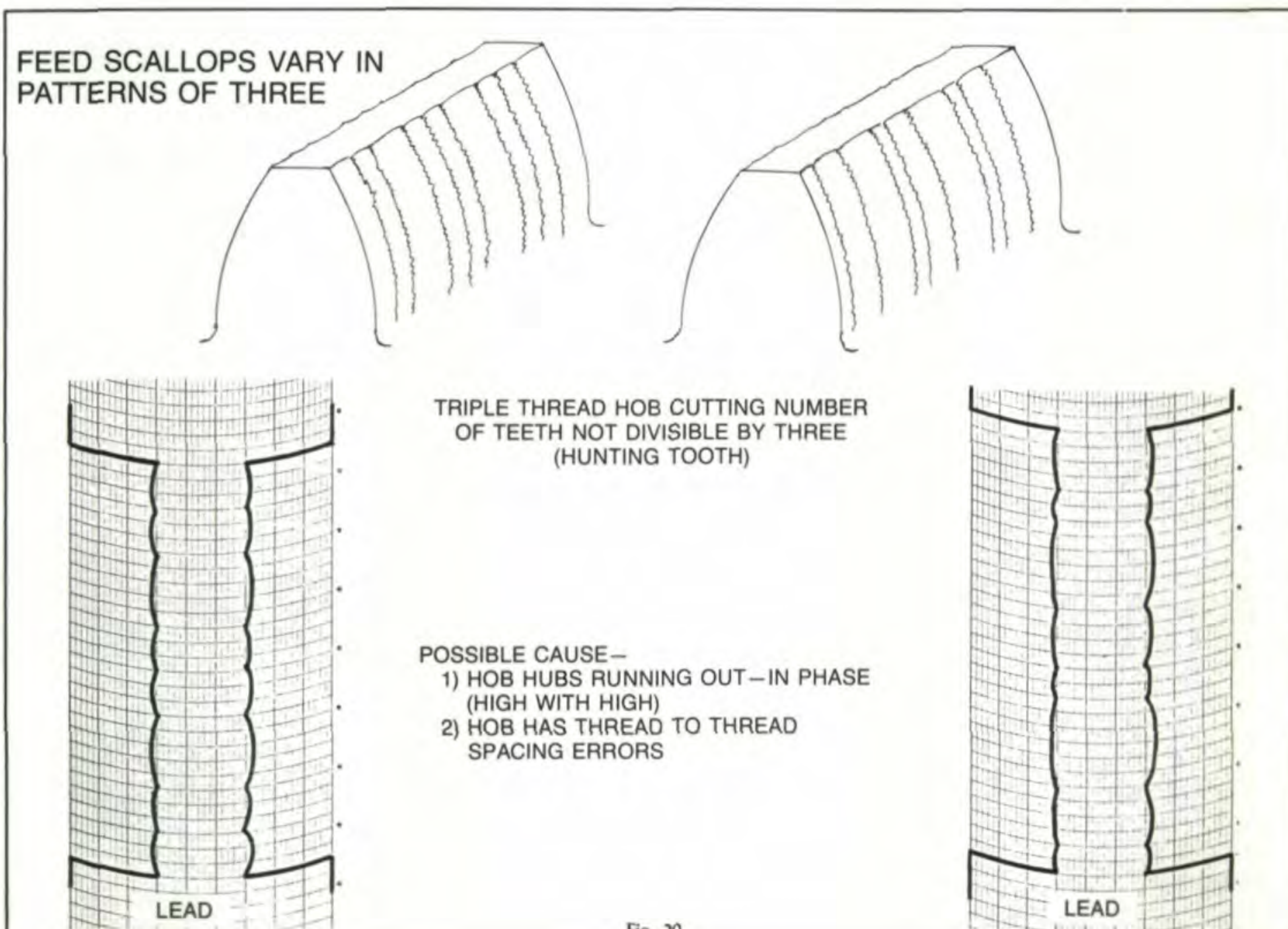

FEED SCALLOPS VARY IN
PATTERNS OF THREE
TRIPLE THREAD HOB CUTTING NUMBER
OF TEETH NOT DIVISIBLE BY THREE
(HUNTING TOOTH)
POSSIBLE CAUSE–
1) HOB HUBS RUNNING OUT–IN PHASE
(HIGH WITH HIGH)
2) HOB HAS THREAD TO THREAD
SPACING ERRORS
LEAD
LEAD

Fig. 20

Inspection Equipment Errors

Fig. 13 shows two involute charts which have errors resulting from inaccurate inspection equipment. Chart #1 (Fig. 13) shows actual charts of a gear having a 2.3492 base diameter. Chart "A" was made with a new involute finger which had its tip on center. Chart "B" was made with a hooked finger which was "sharpened back" and the *finger tip* was then *off center* by .022. This made a "short chart", being off one degree of roll. (Note that all modifications will also be displaced one degree of roll.) Chart #2 (Fig. 13) shows two charts again, "A" and "B", which are of the same flank of the same gear tooth, but turned end for end in the involute machine. When the charts do not come out identical as to the position of their modification relative to the degrees of roll, usually the *machine* is *not centered* properly. Most involute checking instruments are furnished with a centering gauge to check out this problem.

Lead Errors

Five lead charts with various errors have been grouped in Fig. 14. When dealing with lead errors, it is important to remember lead errors seldom, if ever, come from the cutting tool. They almost always come from machine misalignments, looseness, lack of rigidity, gear blank accuracy and mounting, etc. Chart #4, showing a lead error only on one side of the gear tooth near one end, usually occurs as the hob is breaking out of (or into) the cut. This is a result of machine or arbor deflections. This can be corrected by climb cutting, if the gear was cut by conventional cutting and vice versa. A more rigid setup of machine (such as larger diameter hob arbor) or lighter cuts may be in order. Chart #5 (Fig. 14) is a difficult problem, and is usually caused by machine misalignment, although, the hob could be at fault, if you are diagonal or oblique hobbing.

Involute Errors

Fig. 15 shows involute charts with various errors, all of which are self-explanatory. It is important to note, when using a single thread hob, if the involute error is uniform (essentially the same on all teeth), the error may be coming from the hob (accuracy, mounting, sharpening, etc.), but if the error is non-uniform, it is more likely to be coming from the machine.

Multithread Hobbing

Multithread hobbing has its own peculiar problems, somewhat similar to single thread hobbing, and yet quite different. Since more and more companies are going to multithread hobbing, it is important to cover some of these problems.

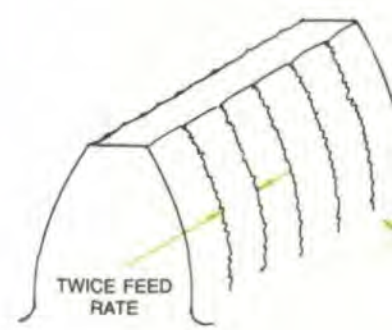
A drawing of a gear hob and the lead charts from both a single thread hob and a three-thread hob were shown in Fig. 10. The three-thread hob lead chart shows a thread-to-thread spacing error between thread #2 and thread #3. It is generally shown that if a gear with a number of teeth evenly divisible by the number of hob threads (3) were cut by this hob, the gear cut would have a spacing error approximately equal to the error in the hob. However, assuming a 3-thread hob had little or no spacing errors, because just a section of the hob teeth (one axial pitch or so) is finish cutting the gear, hob mounting would be very important. If not correctly mounted, it would have thick and thin teeth, spacing errors and runouts, which show up in patterns equal to the number of threads in the hob. Fig. 16 shows the condition and the result on the gear. In each case shown, the effective radius to thread #1 (R1) is shorter than the effective radius to the other thread(s), and R1 will cut shallower tooth spaces than the other thread(s). On the examples shown, the dimension over wires on the gears would be considerably different, from one tooth space to the next.

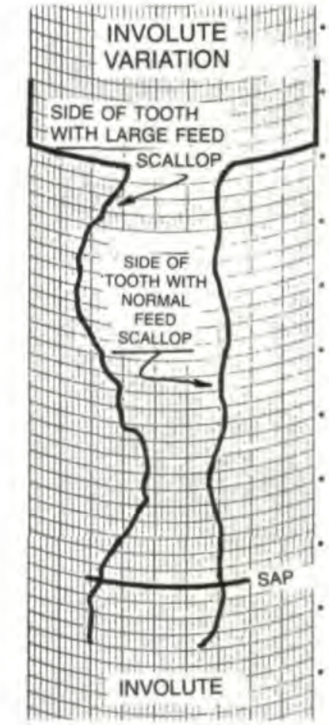
Again, when cutting a number of teeth in a gear which is divisible by the number of hob threads (non-hunting tooth), each tooth is cut by one given thread on the hob. Therefore, each thread will produce a given involute, and involutes may vary from tooth to tooth. This is shown in Fig. 17. In this case, the gear's tooth spacing may or may not vary, depending upon the position of the spacing check along the involute.

The usual "rule of thumb" in multithread hobbing is to cut a number of teeth not divisible by the number of threads (hunting tooth). This is definitely advisable, as many of the errors may tend to "cancel out", as each thread of the hob eventually passes through all of the gear's tooth spaces. This would work effectively if the hob was not fed through the part. However, with today's high feed rates, much of this "canceling effect" does not occur, and the errors are produced in sections along the gear's face width, in each individual feed scallop. Many times, feed scallops can tell a story.

Fig. 18 illustrates a feed scallop pattern which varies in width along the involute. Since each feed scallop is produced by a

FEED SCALLOPS NOT SAME ON BOTH SIDES ON TEETH

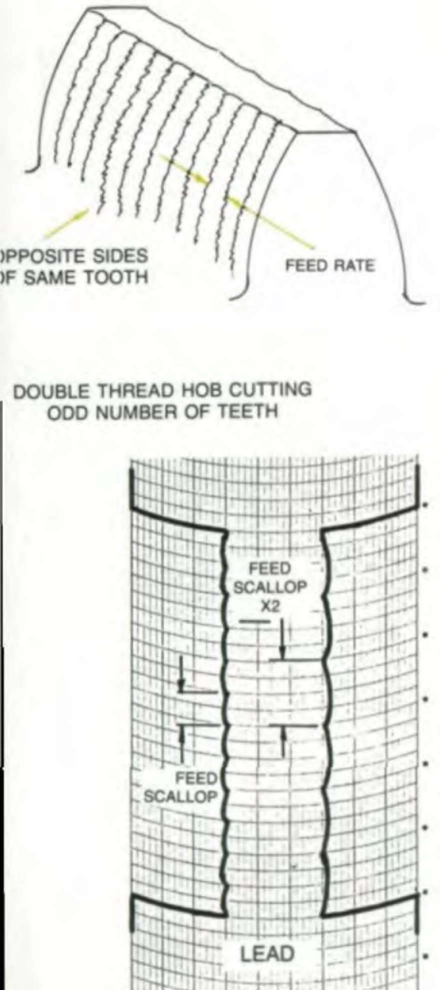




different hob thread, the involutes can vary by as many threads as there are in the hob and more. At the junction of two adjacent feed scallops, there is an "overlapping" area where the "canceling effect" does occur and another involute variation results. This is shown by the involute checks (Fig. 18). Trace #1 and #3 result from different hob threads producing that area (scallop) of the gear. Involute trace #2 is a combination of #1 and #3.

Figs. 19 & 20 show variations of feed scallops, in patterns relative to the number of threads in the hob.

Examples of feed scallops not being the same on both sides of the gear tooth or space are shown in Fig. 21. The "hunting tooth ratio rule" was followed, but the hob was mounted with runout on one end, or had severe thread spacing on one flank. If the hob is running out on one end, the side of the gear tooth with the twice normal size feed scallop will also have severe involute error. The other side of the gear tooth will have a much better involute.



GEAR TOOTH SCORING . . .
(continued from page 14)

APPENDIX 1

Gearset Geometry for Calculating Flash Temperature Index

Parameters for Calculating Friction Coefficient and Scoring Geometry Factor

$\theta = \tan \phi$

$e_p = r_{bp} \tan \phi$

$e_g = C \sin \phi_t - e_p$

$V_s = \left| \omega_p e_p - \omega_g e_g \right|$

$V_t = \omega_p e_p + \omega_g e_g$

Table 2 shows the results of a computer analysis of a high speed gear set with standard addendums. The flash temperature index is the maximum flash temperature rise, 71°F, plus the gear bulk temperature 160°F. The index of 231°F presents a low scoring risk (Fig. 5) which could be slightly reduced by optimizing tooth proportions. The calculated coefficient of friction is significantly lower than the assumed valued of 0.06 with a corresponding lower flash temperature rise. The calculated minimum film thickness is 0.000020 with a λ term of .71 indicating operation in the boundary lubrication regime.

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