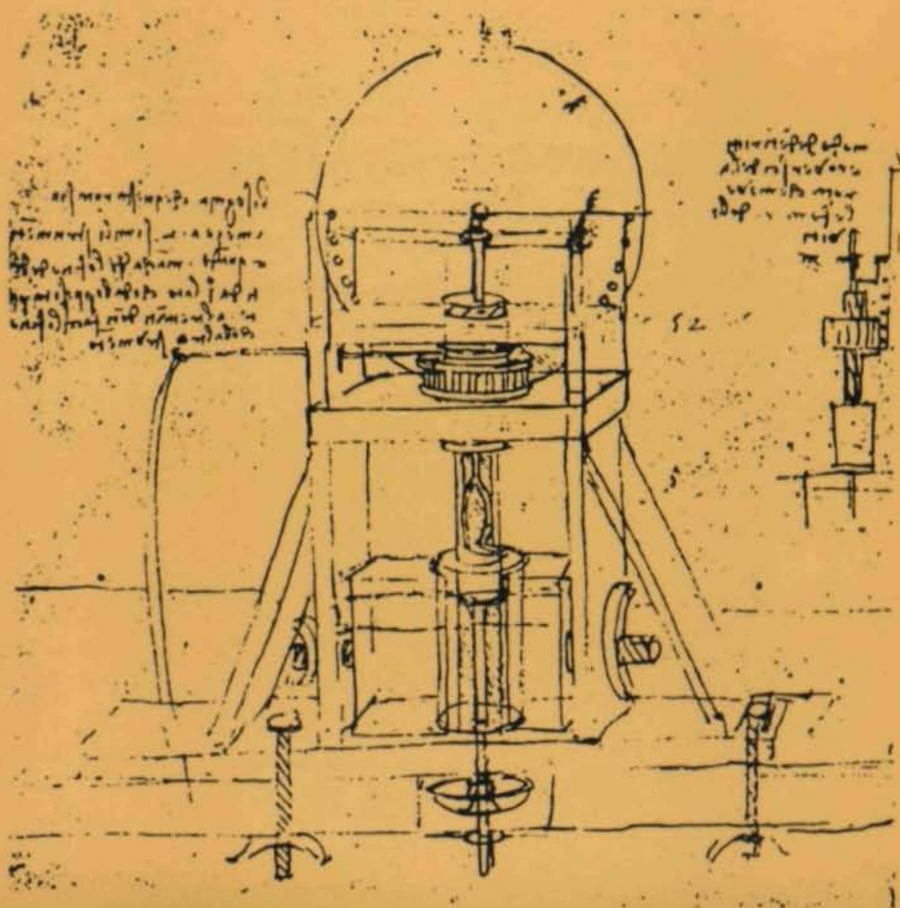
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

NOVEMBER/DECEMBER 1985



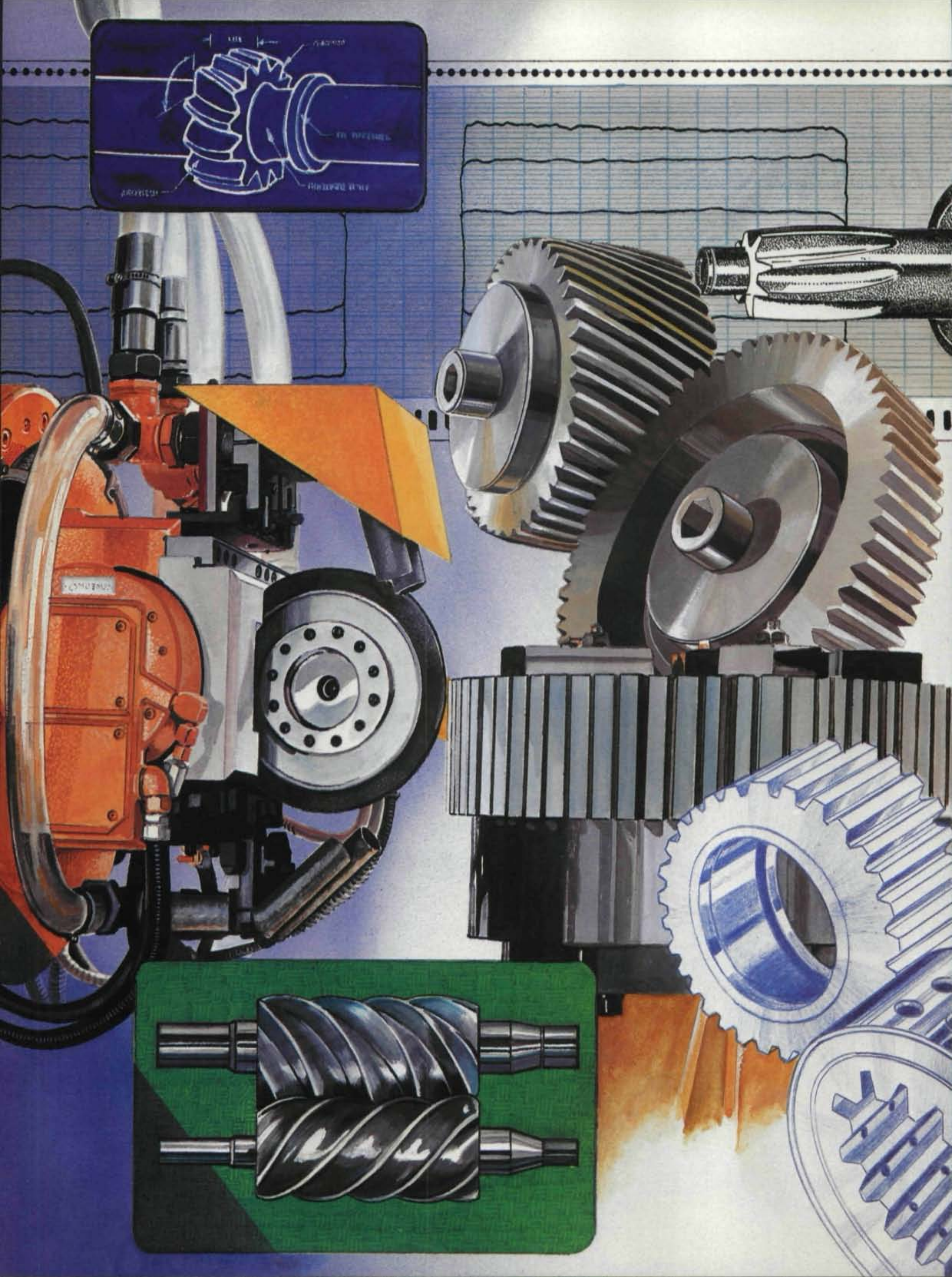
Calculation of Slow Speed Wear of Lubricated Gears

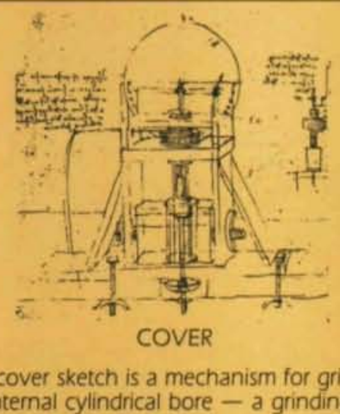
Equations for Gear Cutting Tool Calculations

Finding Gear Teeth Ratios

Fundamentals of CBN Gear Finish Grinding

CNC Gear Shaping Machines





COVER

The cover sketch is a mechanism for grinding an internal cylindrical bore — a grinding and lapping device. The sketch shows a rough cylinder clamped by two hollow vise jaws. These are held firm by adjusting wing nuts. The reamer extends partially into the bore. To hold the polishing or lapping compound, grooves are cut into the outer surface of the reamer. The reamer rises through an ingenious mechanism consisting of a spur gear, having an internal thread of high pitch. A bolt is fit into the thread and has a matching outer pitch and a rectangular inner bore. The gear is confined by two horizontal plates. In this way, it can revolve, but it can not move vertically. As the gear turns, the threaded bolt moves up and down. The horizontal gear is engaged by a semicircle of gear teeth on the circular plate. There is a spring rod at the extreme left of the sketch. From this rod a string passes through the stud and winds onto a collar extending above the gear. At this point, the cylinder bore is ready to lapped. As the large disc turns, the teeth engage the gear. The revolution causes the bolt to enter the bore. After a half-revolution, the gear becomes disengaged. The spring takes over and the string unwinds. This causes the gear motion to reverse and the bolt raises. Then the cycle is repeated.

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MANUSCRIPTS: We are requesting technical papers of every sort from manufacturers of gear making machinery and related equipment, universities, and engineers. Articles should be of an educational and training nature with general appeal to anyone having anything to do with the purchase of materials or machinery, or the design, manufacture, testing or processing of gears. Subjects sought are solutions to specific problems, explanations of new: technology, techniques, designs, processes, and alternative manufacturing methods. These can range from the "How to . . ." of gear cutting **(BACK TO BASICS)** to the most advanced technology. All manuscripts submitted will be carefully considered. However, the Publisher assumes no responsibility for the safety or return of manuscripts. Manuscripts must be accompanied by a self-addressed, self-stamped envelope, and be sent to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, P.O. Box 1426, Elk Grove, IL 60007, (312) 437-6604.

EDITORIAL

Photo by Jennifer Short

Three things have happened in the last few weeks, that lead me to believe the worst is over — not that great times are ahead, but that things will get better.

The strength of the dollar has probably been the single most important reason that our industries are simultaneously facing an onslaught of imported goods and finding themselves unable to export. On September 22, the finance ministers of the five leading industrial nations agreed to a plan to lower the value of the dollar. This is the first "official" action along these lines.

The next day, President Reagan, who is as staunch a "free-trader" as can be found, advocated a "fair-trade" program. For the first time, the President has turned his attention to the plight of manufacturers and farmers, if only to hold off the multitude of protections bills working their way through Congress.

Several weeks ago, an economist at the Federal Reserve requested data from the Machinery Dealers National Association (of which I am an Executive Committee member) to help determine the rates of deterioration and obsolescence of machine tools. This government agency is obviously concerned about the declining efficiency of old equipment and the increased productivity of technological improvements, a more hopeful approach than that of the Treasury or the Administration who play with the Investment Tax Credit and depreciation periods for political or budget balancing purposes.

Why are these 3 events important? Because it indicates that various branches of our government will no longer treat the problems facing American industry with neglect. As the dollar slowly drops, the President takes a more assertive stance on trade, and the government starts to research ways to make America's industry more competitive, things will stop getting worse. When the effect of these and other changes take hold, I think most everyone will agree, that things will begin to get better.

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 DISCUSSION, 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:

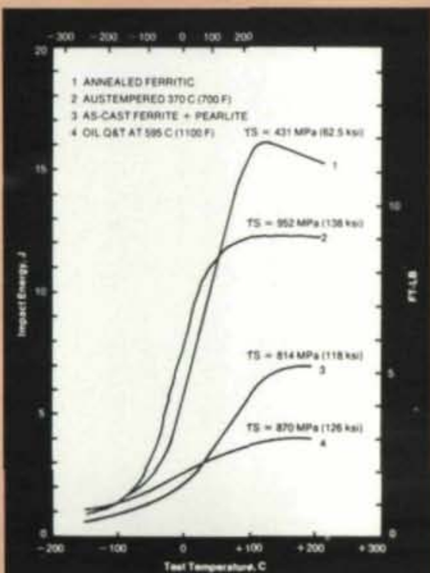
One of the current research activities here at California State University at Fullerton is systematization of existing knowledge of design of planetary gear trains. Efforts are made to avail the designers with useful computerized approaches which would aid assessments of existing or proposed planetary gear transmissions. Simultaneously, we are also examining the historical development of the planetary gear trains and their utilization in various industries.

Since the information on this subject appears rather scarce, any help and direction would be truly appreciated and gratefully acknowledged.

Patrick Safarian
California State University

Editor's Note: While we were able to provide Mr. Safarian with a few sources of information, we felt our reader's might have additional sources to offer. Please respond to the magazine and we will forward the information to Mr. Safarian.

Are you missing out? If there are back issues that you would like to receive, there are limited back issues available at a cost of $7.00 per issues in the USA, $15.00 per issue outside of the USA. Below is listed a complete index for all issues. (Foreign payments must be in US funds, drawn on a U.S. Bank) Checks must accompany requests.



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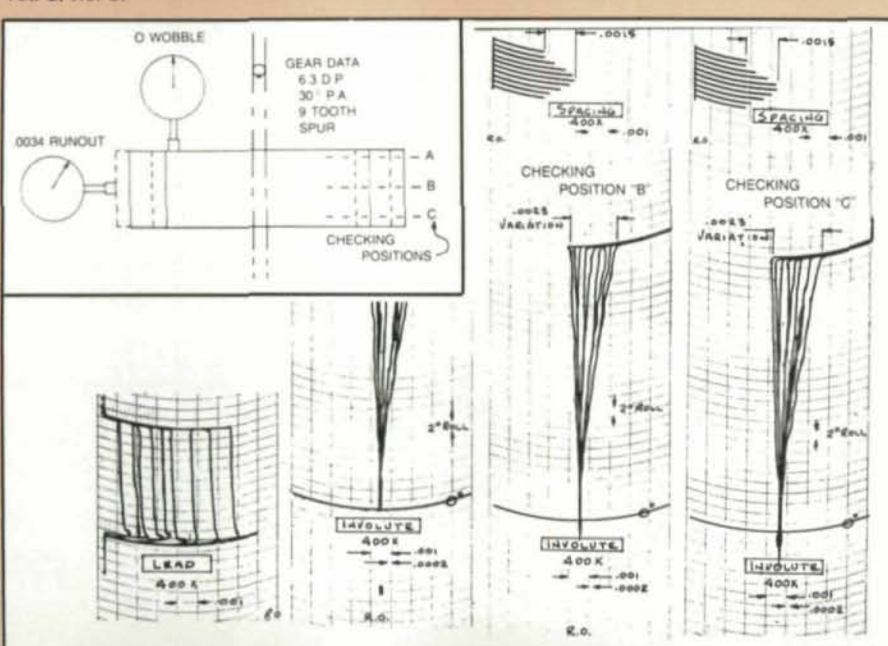

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Calculation of Slow Speed Wear of Lubricated Gears

by
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West Germany

Summary

On gear drives running with pitch line velocities below 0.5 m/s so called slow speed wear is often observed. To solve some problems, extensive laboratory test work was started 10 years ago. A total of circ. 300,000 h running time on FZG back-to-back test rigs have been run in this speed range. The test results showed a correlation between calculated EHL film thickness and a wear coefficient. Based on these experimental results and some wear data of different size industrial gear drives, a method for calculating slow speed wear was derived and is presented in this paper.

To calculate the service life of slow speed gears, the limits of wear for different modes of failure are discussed. Design guides to increase slow speed wear load capacity are presented.

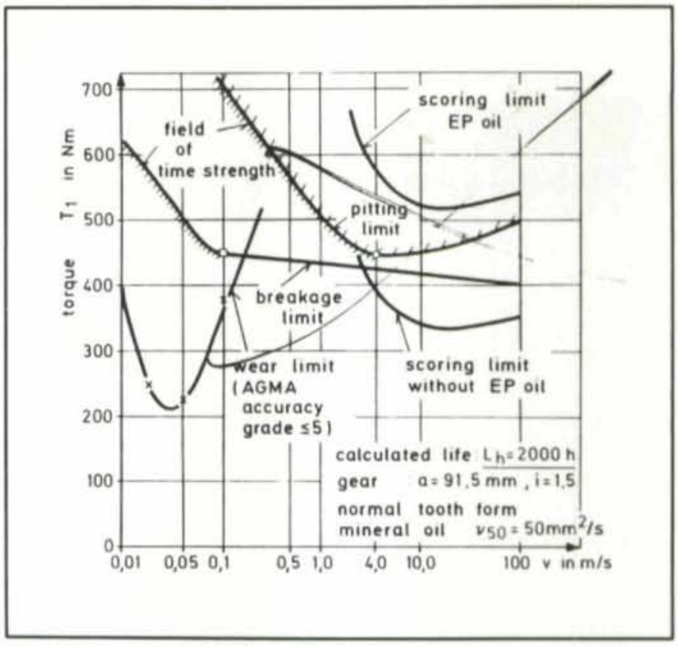

Fig. 1 – Failure limits of a casehardened gear pair. (example)

Introduction

As an example, Fig. 1 shows the limits of the load capacity of case hardened gears to resist pitting, scoring (with and without EP-additives) and breakage. At pitch line velocities below 0.5 m/s, the oil film may become so thin that it is disrupted by asperities or approaches a condition in which boundary lubrication occurs. Slow speed wear results, and probably limits the service life. Gears working in this speed range can be found in open running gear drives of tube mills, in final gearings of infinite variable drives, antenna drives and, as a very new application, in gear drive mechanisms of solar energy reflectors (Heliostats). Slow speed wear can cause gear failure according to change in involute geometry or breakage of the worn off tooth.

Early observations of this kind of failure are described in 19th century papers.[1] Attempts to relate this failure to load – lubrication conditions were made during the last twenty years.

From industrial gear case studies, Dudley[2] derived an approximate film thickness at the pitch line below 0.2 μm as an inadequate oil film with probable wear. He found that slow speed gears with thin lubricants may wear as much as 1/4 in. in 10 million cycles of operation.

Experimental studies of Landen,[3] using a roller test machine, showed relations between wear rate and calculated oil film thickness. At specific conditions, he produced rippled surfaces on the rollers and gave some explanations for this phenomenon.

By application of EHL Oil Film Theory to Industrial Gear Drives Wellauer and Holloway[4] found that gear tooth surface distress is related to the specific film thickness λ. Here λ is the ratio of calculated oil film thickness to the magnitude of the composite surface texture. As another observation they divert:

When λ is less than 0.7 boundary lubrication prevails and lubricant, surface physical and chemical interactions, loads and temperatures are said to have a strong effect on distress modes and rates. To predict wear performance, the authors[4] published a wear formula which is based on a relationship between a wear coefficient k and the specific film thickness λ, see also Seireg.[5]

Experimental

It was the main object of the experiments, to evaluate transferable wear coefficients from steady state region of slow speed wear. Therefore, long time tests (normally more than 500 h) were run on the FZG - gear rig test machine. (For data of the gears see Table 1).

The range of the pitch line velocities investigated was from $v = 0.007$ up to $v = 1$ m/s. To find correlations of gear failure and experience at higher velocities, tests at $v = 2.76$ m/s were also performed. The gears were lubricated by an oil jet with about 1 1/min into the tooth contact. Special tests (for example with grease) were run with splash lubrication. After a first running, in periods of 10,000 revolutions of the pinion and later every 150 h running time, the gears were dismounted and the weight loss was determined by a balance (accuracy $\pm$ 1 mg). The change of the involute geometry was measured with an involute gear test machine.

To solve the complex tribologic problem the following factors of influence were investigated.

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DR.-ING. HANS-JUERGEN PLEWE *has studied as an associate of Prof. Dr.-Ing. H. Winter. He received his Doctoral degree at the Technical University of Munich. In 1980, he began his work with MAN Munich as development engineer.*

- Influence of the pitch line velocity on quantitative wear behavior for two material combinations (casehardened/casehardened and through hardened/through hardened) with three mineral lubricants of different viscosity. (Table 2 and 3)
- Influence of the kind of lubricant (mineral, synthetic, grease . . .) and material combination for a critical pitch line velocity which results in a relatively large amount of wear within a time period.
- For two material combinations and some lubricants, we investigated the influence of
  - load
  - surface finishing
  - profile correction and
  - tooth geometry and module

Influence of Pitch Line Velocity on Weight Loss

Fig. 2 shows the weight loss of the gears (pinion + gear) for 3 mineral oils of different viscosity. The upper graph shows the cumulative wear of the gears after a transmitted power of 50 kWh. For decreasing speed, the lubrication regime goes bad and the cumulative amount of wear increases. Increasing viscosity results in decreasing the amount of wear.

The cumulative wear after 150 h running time is demonstated in the lower graph. For each viscosity, a speed with maximum wear exists. This speed we call " critical speed" because this speed will give the minimum working life. As a result of the decreasing number of rotations, further decreasing speed leads to less weight loss. This behavior is important for designing a slow speed gear for a defined working service.

Influence of Lubricant — Viscosity

Fig. 3 shows typical curves of cumulative wear as a func-

Fig. 2 – Wear for constant transmitted power (above) and constant running time (below) dependent on the pitch line velocity for oils of different viscosity.

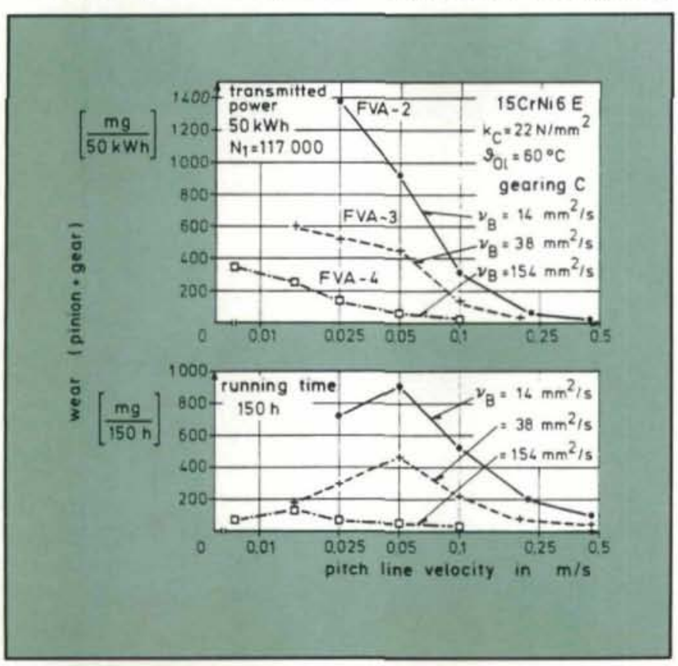


Table 1: Data of test gears

	tooth form C[x]	range of test data
center distance	91,5 mm	constant
gear ratio z_1/z_2	16/24	32/48 ··· 12/18
module	4,5 mm	2,25 ··· 6 mm
addendum modification	$x_1=x_2=0,181$	$x_1=+0,86$; $x_2=-0,5$
active facewidth	20 mm	14 ··· 30 mm
working pressure angle	20°	constant
wear-relevant specific sliding ζ_w	0,74	0,46 ··· 0,79
surface finishing	ground CLA<0.5μm	hobbed CLA<2μm shaved CLA<0.5μm

(x) Mainly used for the tests

Table 2: Data of material and heat treatment

15 CrNi 6 E 16 MnCr 5 20 MnCr 5	case carburized hardness: 700-750 HV 10
42 CrMo 4	flame hardened hardness: 650-700 HV 10
42 CrMo 4 GS 42 CrMo 4	through hardened hardness: 300-330 HB
 31 CrMo V9 (gas) Ck 45 (tufftrided)	nitrided hardness: 650-750 HV 10 hardness: 350 HV 2
42 CrMo 4 GGG-70 GGG-90	borided hardness circ. 1700 HV 0,015 sphäroidal graphite iron pearlitic 280-290 HB austenitic-bainitic, 360-375 HB

Table 3: Data of lubricants (oil sump temperature 60°)

Fluids: Straight mineral oils	kinematic viscosity ν_{40} in mm²/s	dyn. viscosity η_{60} in mPas
Diesel fuel	2,5	1,3
FVA-1	16,7	7,1
FVA-2	30	11,9
FVA-3	98,5	33,0
FVA-4	500	135,5
Synthetic fluids:		
PIV – Varifluid	28	11
Santotrac 70	90	38
BP – Energol GRS 450 EP	235	107
Flow grease: Different base oil viscosities: ν_{40}=68, 120 and 340 mm²/s NLGI 00, Na – soap.		

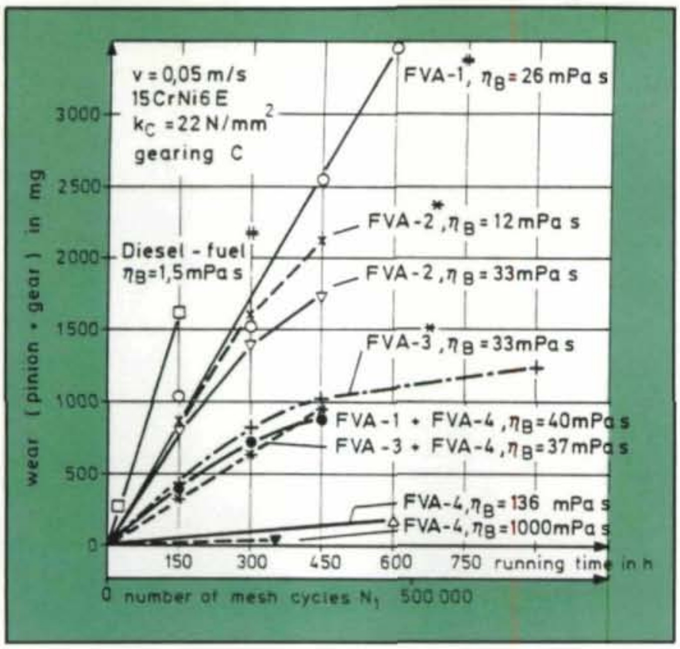


Fig. 3—Slow speed wear. Influence of mineral oil viscocity.

tion of the running time for constant pitch line velocity. Lubrication with mineral oils of increasing viscosity (for decreasing temperature as well as for increasing base oil viscosity) results in decreasing the weight loss of the gears. Tests with very thin oils (Diesel Fuel, FVA 1) at very slow speeds showed fretting corrosion due to changings of the wear mechanisms.

It can be seen that qualitatively there is the same influence of the two parameters pitch line velocity and oil viscosity on the weight loss. A quantitative analysis shows nearly the same amounts of wear for a constant result of speed and working viscosity. This also corresponds to a constant calculated EHL-film thickness acc. Dowson:[6]

$$h_{min} = 2.65 \cdot (\alpha \cdot E')^{0.54} \cdot \left(\frac{\eta_O \cdot v \cdot \sin \alpha_w}{E' \cdot \varrho_c}\right)^{0.7} \cdot \left(\frac{2 \cdot T \cdot 10^9}{d_b \cdot b \cdot E' \cdot \varrho_c}\right)^{-0.13} \qquad (1)$$

Although there are some further important factors in gear tooth surface performance (surface texture, coefficient of friction), we found only the film thickness as a constant characteristic quantity to calculate slow speed wear. Moreover, we tested the effectiveness of EP-additives to reduce wear. The results were divergent, it seems that no general diction is possible. For example, 4% of a phosphor – sulfur – EP – package in FVA – 3 oil showed increasing wear rates. Additional tests were performed with grease lubrication. The wear behavior cannot be described only by the base oil viscosity. For slow speeds, the additional soap grid results in an increase of "effective viscosity" or decreasing wear rates.[7]

Influence of Load for Different Materials

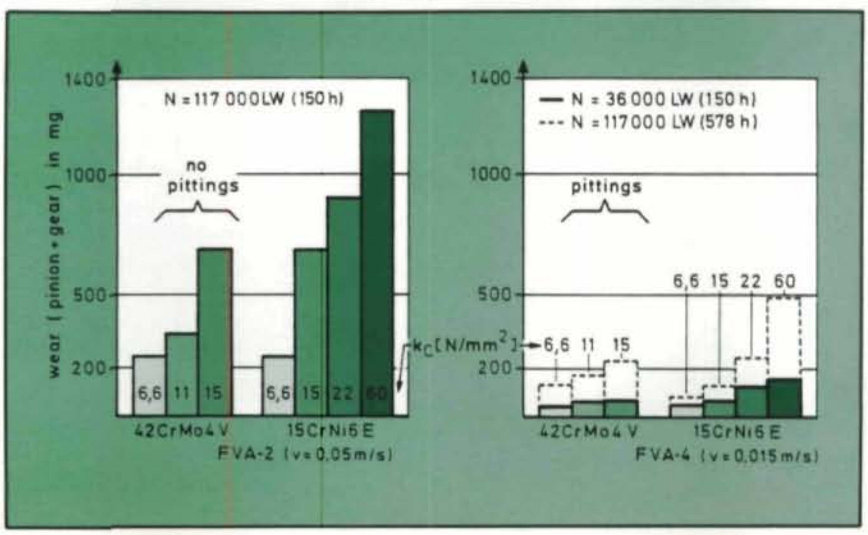
For the most usual materials in gear practice, casehardened and through hardened gears, the influence of load was tested, applying 2 mineral oils of different viscosity at their critical speeds. Fig. 4 shows the results for different test parameters. The left side graph shows the effect of load – variation for lubrication with a mineral oil of low viscosity at a speed of $v = 0.05$ m/s. Up to mean loads of $k_C = 15$ N/mm², both materials show about equal amounts of wear. Therefore, at low Hertzian pressures there is no real advantage in wear reduction for casehardened gears in relation to through hardened gears of 42 CrMo 4 steel.

Application of lubricants of higher viscosity at the critical speed of $v = 0.015$ m/s results in much smaller amounts of wear (right side graph). Here, at all loads, it is an advantage in wear reduction to take casehardened gears instead of through hardened gears. Under such conditions, casehardened gears are superior (in wear load capacity) compared to through hardened gears.

Effect of Surface Finish

A common way to evaluate the danger of wear failure is to compare the calculated film thickness with a characteristic roughness height. As indicated in the introduction, experts use different methods for choosing an appropriate roughness height measure.

Fig. 4—Influence of load on slow speed wear for two gear materials and lubricants.



To investigate these surface effects, gears of different surface finish were run. After a minimum running time of circ. 500 h, hardly any difference in wear behavior was observed between hobbed and ground through hardened gears. The worn in surface texture was dependent on the working conditions.

Therefore, we decided not to consider the surface conditions of manufacturing as an important influence factor for a coefficient of wear.

Change of Tooth Profile According to Slow Speed Wear

Slow speed wear alters the involute tooth profile. Therefore, we decided to take the weight loss to measure the amount of wear in a test run. The alteration of tooth profile is dependent on the amount of wear and the material–hardness–combination of the gear pair.

Fig. 5 shows the measured profile form along the path of contact for different amounts of wear. The upper line demonstrates the alterations for casehardened pinions and gears having the same hardness. In the beginning of the wear process, most of the tooth form alteration occurs near the starting point of contact of the driving pinion, and near the end point of contact of the driven gear. After the tooth profile has arrived at an AGMA accuracy grade of about 5, further slow speed wear proceeds nearly equidistantly to this tooth profile.[8]

The lower line of Fig. 5 shows the alteration of a tooth profile in the case of a large difference between the surface hardness of the pinion and gear. The figures demonstrate that wear mainly occurs on the through hardened gear with lower surface hardness.

Wear Calculation Procedures

A method to calculate the slow speed wear of gears was published by Wellauer/Holloway.[4] They based their

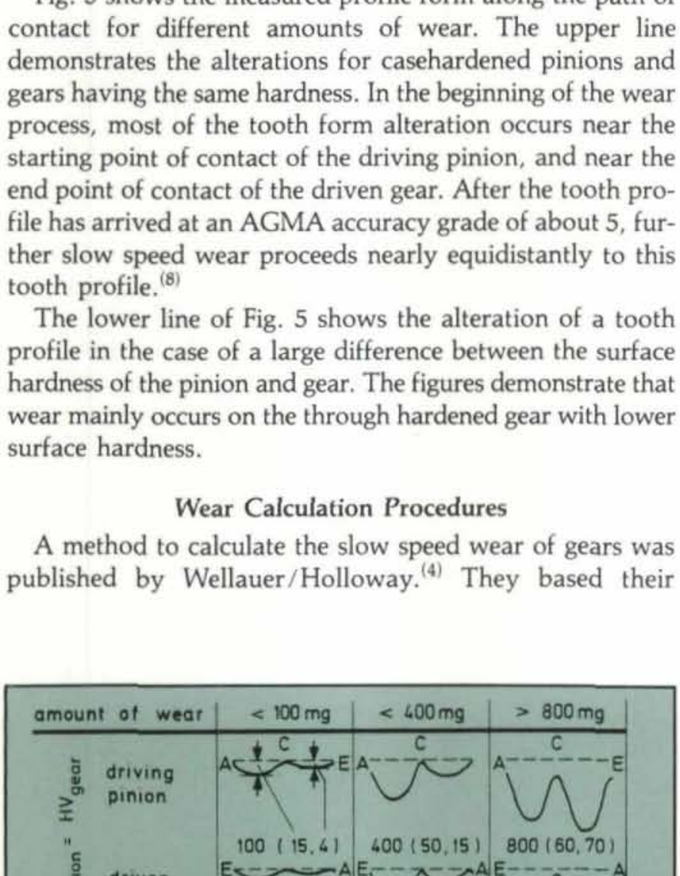


Fig. 5 – Alteration of tooth profile along the path of contact for different amounts of wear and two material combinations (casehardened/casehardened and casehardened/through hardened gears).

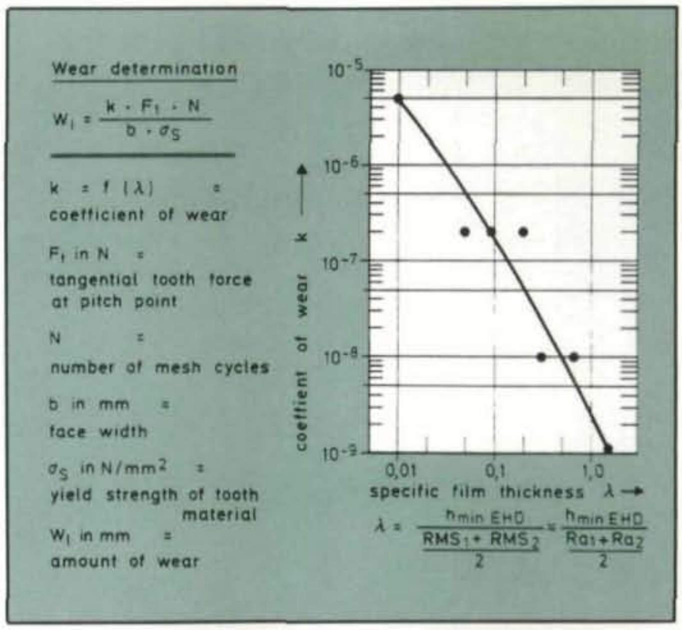


Fig. 6 – Calculation of wear according to Wellauer/Holloway /4/ and Seireg /5/ (The medium line was added by the authors).

calculation method on a very common wear equation:

$$W_v = K \cdot 1 \cdot A_a \tag{2}$$

Where K is the coefficient of wear, 1 the sliding distance and A_a the real Hertzian contact area.

Fig. 6 shows the wear equation according to Wellauer/Holloway and Seireg.[5] They published this gear wear formula for the region $\lambda < 0.7$. The dependence of the wear coefficient k from the specific film thickness λ is based on a theoretical model of Seireg. To compare these results with the test-results reported here, the medium line in Fig. 6 was added by the authors.

Evaluation of the Wear Coefficient

Equation 1 describes a linear wear process. Examples of typical experimental slow speed wear curves are given in Fig. 2. The linear running in wear, is followed by the regime of steady state wear. The wear coefficient describes the wear behavior in the steady state regime. The test showed that the running in process lasts, approximately, up to 500 h running time, independent of the working conditions.

Fig. 7 demonstrates the way we evaluated the wear coefficient from experimental wear curves. The left graph shows wear curves at three different working conditions. They represent the influence of the film thickness.

By searching for the best fitting approximation formula, we found an exponential equation of the form:

$$W_m = a' \cdot N^{b'} \tag{3}$$

where N is the total number of tooth contacts. The coefficient a' and the exponent b' are dependent on the material and working conditions. Linear regression of the test results and equation (3) showed correlation factors $\geqslant 0.98$.

As wear coefficient, we defined the gradient of the best

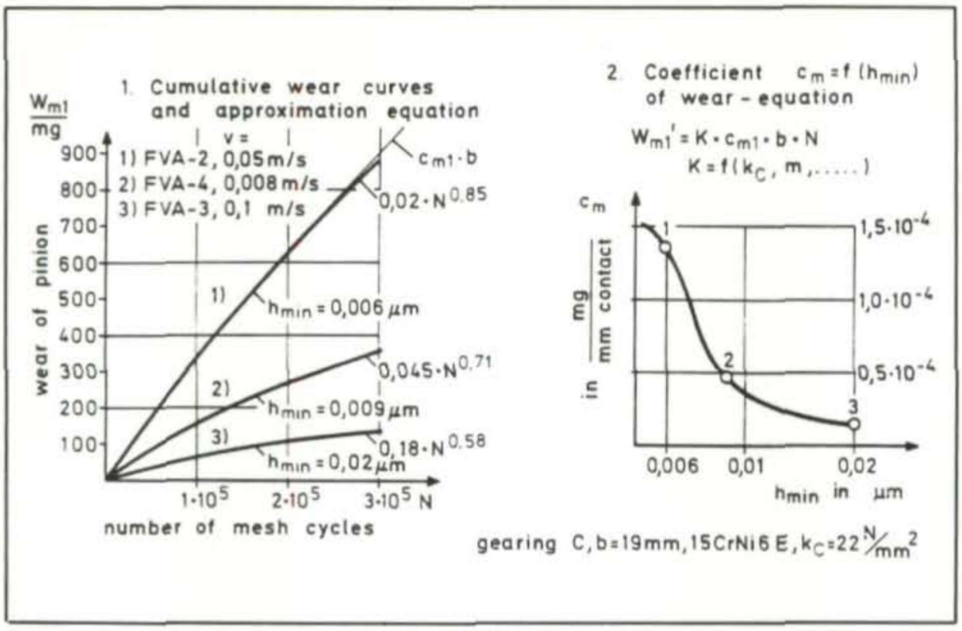


Fig. 7 – Evaluation of wear coefficient c_m from wear curve. (Example for tooth form C according table 1).

fitting exponential equation at 500 h running time.

The right graph in Fig. 7 demonstrates this mass – wear – coefficient c_m over the EHL-film thickness.

$$c_m = \frac{dW_m/dN}{b}$$

where b is the active facewidth.

Tooth form alterations and wear failure modes are best described by *linear* amounts of wear. The mean coefficient of linear wear c_l can be derived from the measured coefficient of *mass* wear c_m. In Fig. 8, this linear wear coefficient of the test gears c_{lt} is plotted against the EHL minimum film thickness. Moreover, the influence of different material combinations, loads and lubrication conditions is shown.

Proposed Wear Calculation Method

Sliding Distance and Influence of Size.

Several different wear formulae have been published by now. One common characteristic of all these methods is the use of the sliding distance to calculate the amount of wear.

Fig. 9 demonstrates how to calculate the sliding distance for disc and gear contacts. At the two-disc-contact, the sliding distance for one point on a disc is given by the contact time in the contact area and the sliding speed. (For example for point 1 on disc 1):

$$I_{P1} = \frac{2 \cdot b_H}{v_1} \cdot (v_1 - v_2) \cdot N_1$$

Herein $(v_1 - v_2)/v_1$ is the (constant) slip related to disc 1. The Hertzian contact length 2 b_H is also constant for the whole circumference.

Looking at the tooth contact, there are variable slip and

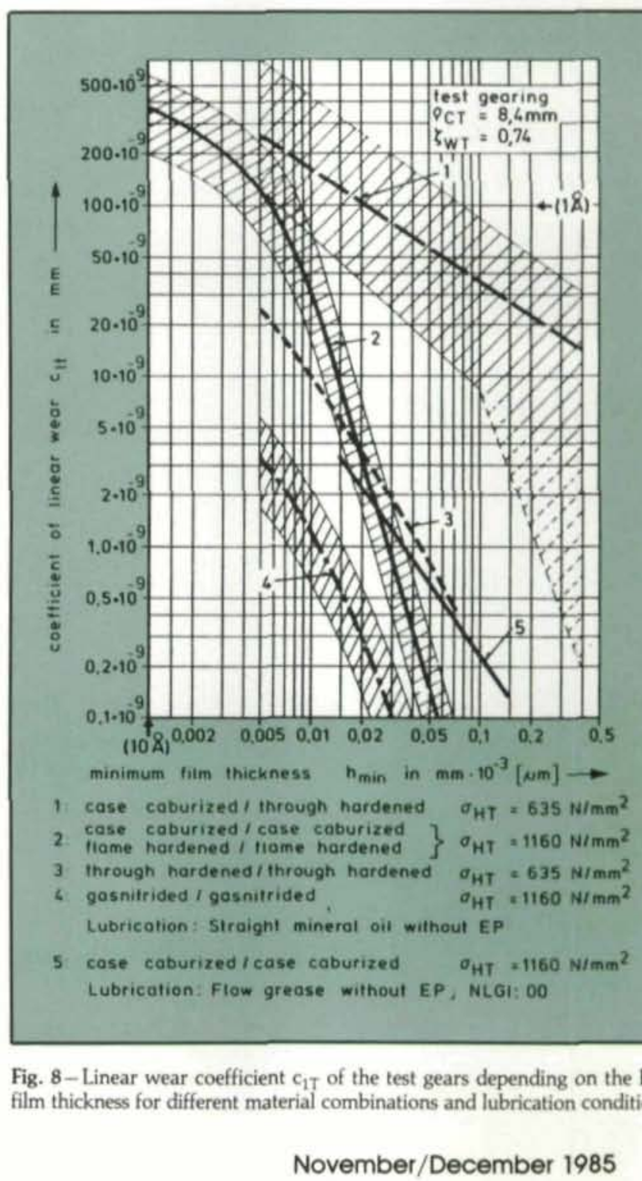


Fig. 8 – Linear wear coefficient c_{lT} of the test gears depending on the EHL film thickness for different material combinations and lubrication conditions.

contact conditions. As an equivalent to the slip, there is the variable specific sliding ζ along the line of action. Because of the alteration of the contact-radius, there is also a small variation of the Hertzian contact length.

As described in Fig. 5, only during the running in process is the alteration of tooth profile nearly corresponding to the slope of the specific sliding. During a steady state process, however, no additional alteration of the tooth profile is observed. Therefore, from tests with gears of different module, profile correction and gear ratio, we derived a wear relevant specific sliding, which considers the sliding conditions in the dedendum and addendum path of contact according to equation 6:

$$\zeta w = \frac{\zeta E1 \cdot e_1 + \zeta A2 \cdot e_2}{e_1 + e_2} \tag{6}$$

The relations for internal and external gears are shown in Fig. 10. As a representative Hertzian contact length, we determined the values at the pitch point ($2b_{HC}$). Relating to the test gears (Index T), the following relationship for gears of different size at the same pressure can be given:

$$b_H/b_{HT} \sim \varrho_c/\varrho_{CT} \tag{7}$$

This relation represents a tribological size effect. To check this influence by experiment with large test gears would be too expensive. Considering information about slow speed wear behavior of large industrial gears, at somewhat different working conditions, showed results which are in good agreement with equation 7.

Quantitative Load Influence

Some experimental results of load influence were shown in Fig. 4. Considering more tests with different material combinations, we found the following relationship:

$$\frac{c_l}{c_{lT}} = \left(\frac{k_C}{k_{CT}}\right)^{0.7} = \left(\frac{\varphi_H}{\varphi_{HT}}\right)^{1.4} \tag{8}$$

When exceeding the endurance limit for Hertzian stress, the exponent in equation 8 seems to increase.

Rule for Calculation of Slow Speed Wear for Mineral Oil or Flow Grease Without EP Additives.

1. To evaluate the danger of slow speed wear distress, one first calculates the EHL-film thickness according to equation 9.

$$h_{min} = 2.65 \cdot (\alpha \cdot E')^{0.54} \cdot \left(\frac{\eta_O \cdot v \cdot \sin \alpha_w}{E' \cdot \varrho_c}\right)^{0.7} \cdot \left(\frac{2 \cdot T \cdot 10^9}{d_b \cdot b \cdot E' \cdot \varrho_c}\right)^{-0.13} \tag{9}$$

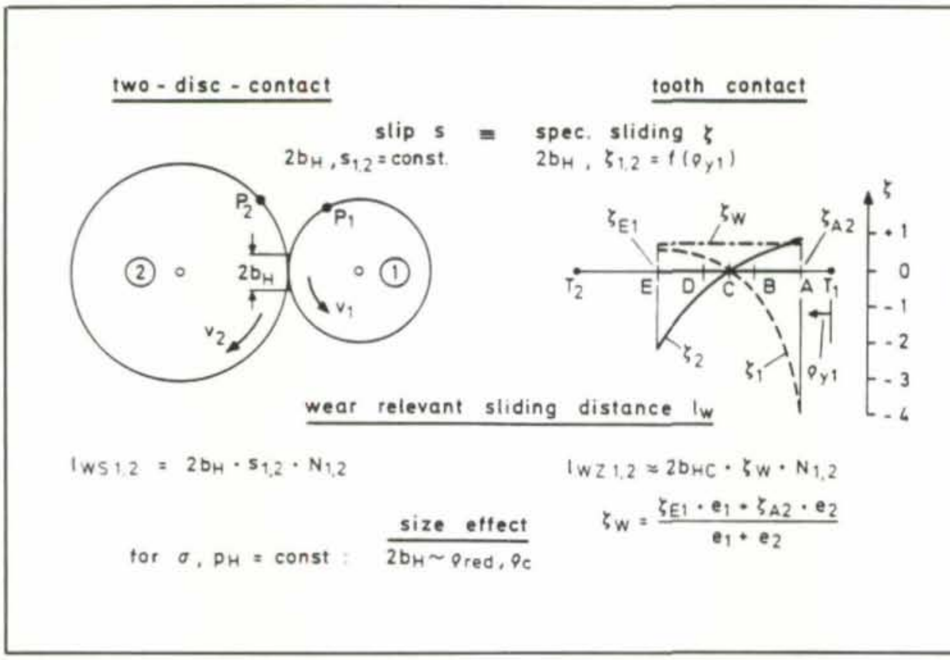


Fig. 9 – Calculation of sliding distance for disc-contact and tooth contact.

Normally only the oil jet or sump temperature is to be considered for viscosity because only small friction heating occurs at slow speeds.

Danger of distress must be expected in $h_{min} \leqslant 0.4\ \mu m$ for gearings with large hardness differences (for example casehardened mating with through hardened gears). In the case of the pinion and the gear having the same surface hardness, the critical figure is in the order of 0.05 μm.

2. Determination of wear coefficient c_{IT}

If no experience from actual gears is available, Fig. 5 can be used to evaluate a wear coefficient c_{1T}, which depends on the calculated EHL-film thickness. This refers to mineral oils without EP-Additives.

For flow grease lubrication, the viscosity of the base oil at working temperature can be taken for a first approach.

3. Calculation of the linear amount of wear W_e:

$$W_e = c_{IT} \cdot \left(\frac{\varphi_H}{\varphi_{HT}}\right)^{1.4} \cdot \left(\frac{\varrho_C}{\varrho_{CT}}\right) \cdot \left(\frac{\zeta_W}{\zeta_{WT}}\right) \cdot N \qquad (10)$$

Herein φ_{HT}, ϱ_{CT} and ζ_{WT} are the test gear data on which c_{1T} is based.

4. Calculated life according wear failure:

$$L_{hW} = \frac{W_{lzul}}{c_{IT}\left(\frac{\varphi_H}{\varphi_{HT}}\right)^{1.4} \cdot \left(\frac{\varrho_C}{\varrho_{CT}}\right) \cdot \left(\frac{\zeta_W}{\zeta_{WT}}\right) \cdot n \cdot 60} \qquad (11)$$

Guides to estimate the allowable amount of linear wear $W_{1\ zul}$ can be taken from Fig. 11.

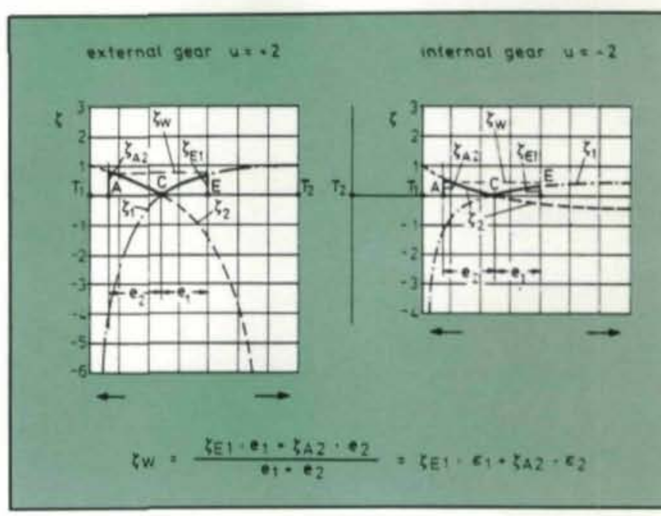


Fig. 10 – Wear relevant specific sliding ζ_W.

Modes of failure and allowable amounts of linear wear: Fig. 11 shows the failure tooth profile forms and the allowable depth of wear for different tooth geometry and heat treatment.

1. Worn out tooth flanks:

For precision gears, the profile form error, resulting from wear, causes reduced accuracy in torsional transmission. Consequently, increased dynamic loads must be expected at high speeds in the case of gears running at different speeds.

2. Wear of surface hardness layer:

If the surface hardness layer is worn-out, danger of pitting and plastic deformation increases.

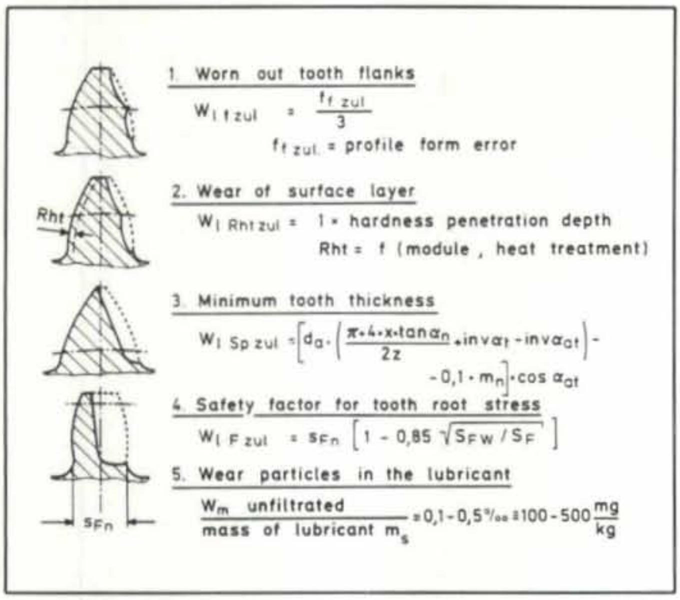


Fig. 11 – Wear failure modes and allowable linear amount of wear. (S_F, S_{FW} = safety factors; new, worn out).

3. Minimum tooth thickness at the tip cylinder:

   A figure of $s_a = 0.1\ m_n$ perhaps can be taken as a limit.

4. Reduction of the safety factor for tooth root stress:

   The factor 0.85 (instead of 1.0) was found from the test results of the authors.

5. Wear particles in the lubricant:

   Abrasive particles in the lubricant result in increasing rates of wear. So, only a certain percentage is acceptable.

Conclusions

Extensive long time tests on FZG back to back test rigs in the speed range below 1 m/s showed the influence of the main parameters on slow speed wear.

For straight mineral oils the influence of pitch line velocity and viscosity at working conditions can be described by the EHL-film thickness. We found the film thickness as a parameter to evaluate the danger of wear distress. It can also be used to determine a wear coefficient to calculate the amount of wear. A danger of wear distress is given at film thicknesses below 0.05 μm. The material combination casehardened/through hardened gears showed critical wear conditions for a film thickness up to 0.4 μm.

Tests with different additives in the mineral oil showed divergent results. Their effect cannot generally be predicted. A calculation based on the EHL-film thickness is not possible. The test with grease lubricated gears showed that the additional soap grid results in an increase of "effective viscosity" or decreasing wear rates.[7]

Increasing loads below the endurance limit for Hertzian stress results in degressive increase of wear. Loads above this limit can also result in pitting.

Calculation Example

Planetary gears are often used in high loaded, slow speed drives. The typical material combination (casehardened planetary gears/through hardened or normalized internal gear) of planetary gears gives a high probability of slow speed wear failure. As an example, the danger of wear failure and the expected life due to wear in a planetary gear drive is calculated by the described method.

Description of the Gear:

The power of the driving sun pinion (casehardened) is branched into three planetary gears (casehardened) which are in contact with the fixed internal gear (through hardened).

Gear Data:

Number of teeth

Sun pinion	$z_S = 17$	module = 6mm (circ. 4 DP)
Planet gear	$z_P = 49$	pressure angle $\alpha = \alpha_w = 20°$
Internal gear	$z_I = 115$	active facewidth = 60 mm
Transmitted power	72 kW,	driving speed $n_S = 100$ min.$^{-1}$

Lubricant: Mineral oil without EP additives, viscosity at working temperature = 33 m Pa s, pressure-viscosity coefficient $\alpha = 17.6\ 10^{-9}\ m^2/N$.

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Nomenclature

a	mm	center distance
b	mm	active facewidth
b_H	mm	half of Hertzian contact length
c_1	mm/U	coefficient of linear-wear
c_m	mg/mmU	coefficient of mass-wear
d_a	mm	tip diameter
d_b	mm	base diameter
d_w	mm	pitch diameter
e	mm	length of addendum path of contact
h_{min}	mm	minimum film thickness (acc EHL)
k_C	N/mm^2	stress acc. Stribeck at pitch point
l	mm	sliding distance
m	mm	module
m_S	kg	total mass of lubricant
n	min^{-1}	pinion speed
p_C	N/mm^2	stress acc. Hertz at pitch point
s_{Fn}	mm	tooth thickness in the critical section at the tooth root
v	m/s	pitch line velocity
x	–	addendum modification coefficient
z	–	number of teeth
A	mm^2	active tooth contact area
E	N/m^2	effective elastic modulus
L_{hW}	h	working life acc. wear
N	–	number of mesh cycles
S_F	–	safety factor for tooth root stress
S_{FW}	–	safety factor for tooth root stress (of worn out gear)
T	Nm	torque
W_1	mm	amount of linear-wear
W_m	mg	amount of mass-wear
α	m^2/N	lubricant pressure – viscosity coefficient
α	°	pressure angle
α_a	°	pressure angle at tip diameter
α_w	°	working pressure angle
ζ	–	specific sliding
ζ_W	–	wear-relevant specific sliding
ϵ_a	–	transverse contact ratio
η_0	mPas	lubricant viscosity at amibient conditions
ϱ	mg/mm^3	density
ϱ_C	mm	relative radius of curvature at pitch point
ϱ_H	N/mm^2	surface stress at the pitch point acc. ISO

The index T characterises the actual working parameters at test conditions.

From this data we derive the following results:

	sun gear/plant	planet/internal gear
Rolling speed	0.465 m/s	0.465 m/s
Surface stress acc. Hertz φ_H	1504 N/mm^2	581 N/mm^2
Relative radius of curvature at pitch point ϱ_C	12.95 mm	87.5 mm
EHL-minimum film thickness h_{min}	0.053 μm	0.12 μm
Coefficient of linear wear of test gears (from Fig. 8) c_{IT}	$0.1 \cdot 10^{-9}$mm/ *contact*	$35 \cdot 10^{-9}$mm/ *contact*
Wear relevent specific sliding ζ_W	0.762	0.183
Real number of mesh cycles /min (3 planets)	$n_S = 261.36\ min^{-1}$	$N_I = 38.64\ min^{-1}$
Linear wear after 1000 h of service life W_I	0.004 mm	0.185 mm
Allowable amount of wear	~1.2 mm (case depth)	~4 mm$^{(1)}$ (minimum thick-nness at tip cylinder)
Service life due to wear L_{hW}	335 000 h	22 000 h

(1) Another allowable amount of wear should be considered: The percentage of wear particles in the lubricant. For the above example the mass wear of the internal gear results in an amount of $W_m \approx 120$ g after 1000 h.

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This paper was previously presented at the Fall Technical Conference of AGMA, P. 219.16, 1982

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TECHNICAL CALENDAR

SME's annual "Gear Processing and Manufacturing" clinic and tabletop exhibits will be held at The Dearborn Inn in Dearborn, Michigan November 19-21, 1985. Current technologies in the gear industry will be covered. J. Richard Newman, formerly of National Broach and Machine Division of Lear Siegler, and Carl S. Eckberg of Bourn & Koch Machine Tool Co., are co-chairing this three-day clinic. An evening of vendor tabletop exhibits will accompany the daily technical presentations. For further information on the clinic or the exhibits, contact Dianne Leverton at SME, 313/271-1500, extension 394.

March 3-5 — 2nd World Conference on Gearing
Institute de l'Engrenage et des Transmissions
Paris, France

The Paris Congress has brought together various groups of technicians, practitioners, buyers, production, lubrication and control experts (150 speakers from all over the world) to present the results of their research on gearing techniques. To obtain further information contact: Maurice Allard, Director, Institute de l'Engrenage et des Transmissions, 162 Boulevard Malesherbes, 75017, Paris, France. Telephone: 43.80.04.09

March 17-19 — International Conference on Austempered Ductile Iron, Ann Arbor, Michigan

For further information contact: Dale Breen, Gear Research Institute, P.O. Box 353, Naperville, IL 60566. Telephone: (312) 355-4200.

Fig. 3 – SINGLE FLANK TESTING OF GEARS by Robert Smith, Gleason Works, Vol. 1, No. 1.

GEAR TECHNOLOGY

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General Equations for Gear Cutting Tool Calculations

by
I. Bass
Barber-Colman
Rockford, IL

The proper design or selection of gear cutting tools requires thorough and detailed attention from the tool designer. In addition to experience, intuition and practical knowledge, a good understanding of profile calculations is very important.

The main purpose of this article is to acquaint readers with a method of cutter profile calculations for both involute and noninvolute forms. The formulas given below are applicable to gear cutter racks, shaper cutters and hobs with thread angles of less than 4°. By a slight rearrangement, they can be used for finding the part profile when the cutter profile is known.

Basic Principles

The following is the basis for the development of the general equations:

1. Fundamental law of gear teeth conjugated action.
2. Geometry of generating action.
3. Cutter rack or hob are considered as a shaper cutter with infinite number of teeth.

According to the fundamental law mentioned above, a tangent to a cutter profile at a given point is simultaneously the tangent to the part profile at the moment when the generation of this point takes place. As a result (Fig. 1a), the point A on the cutter profile will generate the corresponding point on the part profile irrespectively of its form. Thus, any complicated profile can be defined by a family of tangents or, as a final step, by a set of points. Therefore, any profile can be analyzed by applying the equations developed for one point being taken separately.

From this standpoint, three coordinates, radius r_A, angles φ_A and μ_A will provide us with the necessary information to describe the position of point A (Fig. 1b). The μ_A locates the tangent to the part profile at the current point. The general expression for μ_A is:

$$\tan \mu = \frac{r}{\frac{dr}{d\varphi}}$$

Gear cutter profile equations

AUTHOR:

For 25 years, **ILYA BASS** *has been involved in practical and research work relating to manufacturing systems, and specifically to cutting tools. He has authorized two books and numerous articles on gear cutting tools. Currently, Bass is a Programmer/Software Engineer at Bourne & Koch Machine Tool Co. in Rockford, IL. Prior to this he was a Programmer Engineer with Barber-Colman.*

The meshing of gear shaper cutter and part is shown on Fig. 2. Axis Y_P coincides with the axis of a part space; whereas, axis Y_c coincides with the axis of a cutter tooth. Let this position be considered as a zero-position. During meshing rotation from the zero-position to the position of generation at point A, axes Y_P and Y_c will rotate by the angles ψ_P and ψ_C, respectively. The following relationship is true:

$$\psi_c = \psi_p \times \frac{n}{N} = \psi_P \times \frac{r_w}{R_w} = \psi_P = i,$$

where

n, N – numbers of teeth in part and gear cutter,

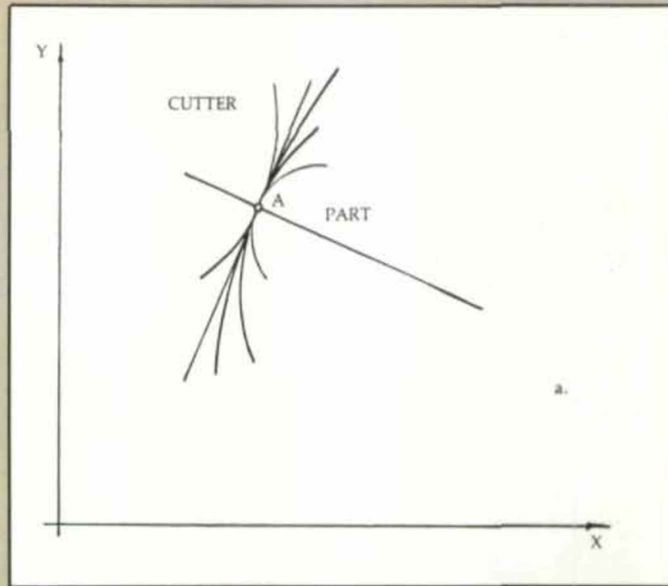
r_w, R_w – generating radii of the part and gear cutter. Note that at the instant of generation, according to the fundamental law, a normal to the part profile at a given point is also the normal to the cutter profile, and passes through the pitch point O. This geometric condition is used in deriving the following set of equations which determine the x_c, y_c – coordinates of the shaper cutter:

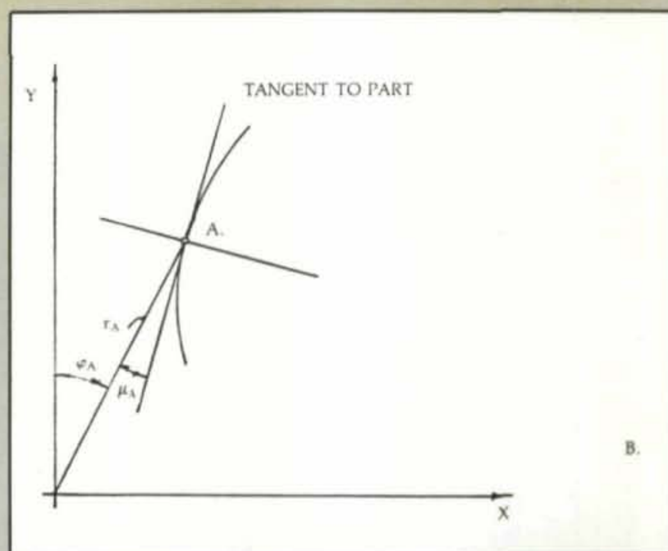
$$
\begin{aligned}
u_A &= r_A \times \cos \mu_A \\
\cos \alpha &= \frac{u_A}{r_w} \\
\sigma_P &= \alpha - \mu_A \\
\psi_P &= \sigma_P - \varphi_A \qquad (1)\\
x &= r_A \times \sin \sigma_P \\
\psi_c &= \psi_P \times i \\
x_c &= -R_w \times \sin \psi_c + \frac{x}{\cos \alpha} \times \cos(\alpha + \psi_c) \\
y_c &= R_w \times \cos \psi_c + \frac{x}{\cos \alpha} \times \sin(\alpha + \psi_c)
\end{aligned}
$$

where

α – pressure angle at a given point,
σ_P – angle from a zero-position to generating position.

The equations for the rack or hob profile can be derived from[1] considering $N \rightarrow \infty$. By making substitutions for R_w, ψ_c, x we get:





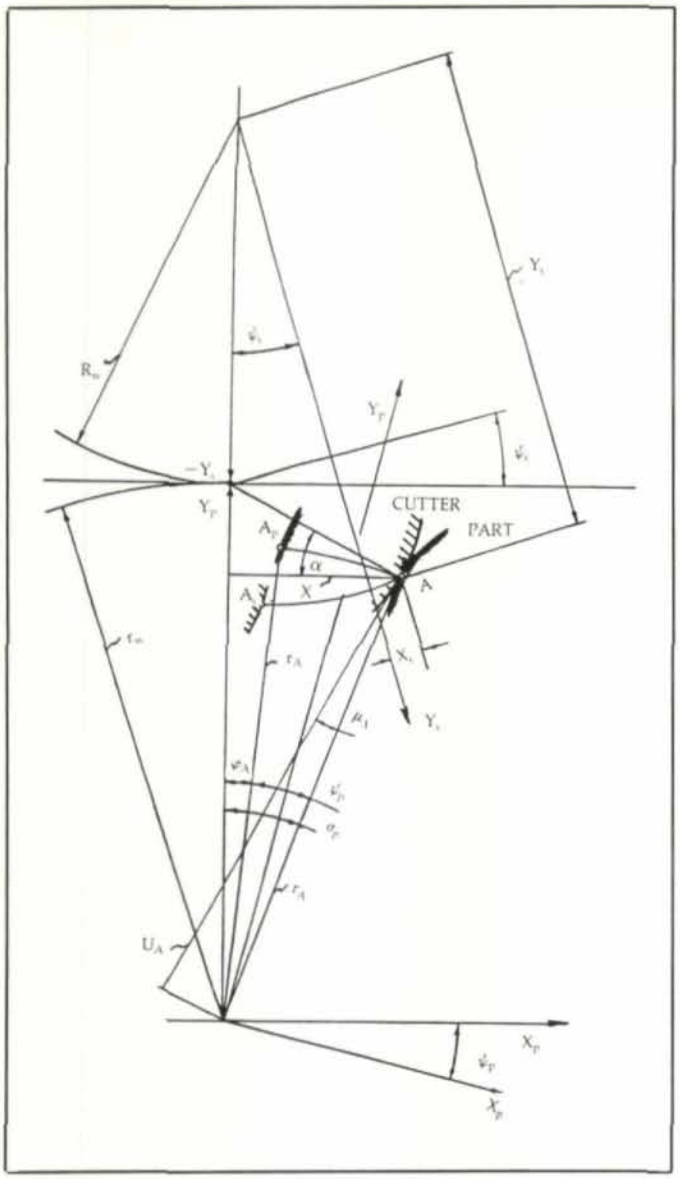
$$
\begin{aligned}
x_c &= -\frac{r_w}{i} \times \sin(\psi_P \times i) + \frac{r_A \times \sin \sigma_P}{\cos \alpha} \times \cos(\alpha + \psi_P \times i) \\
& \qquad (2) \\
y_c &= \frac{\mu_w}{i} \times \cos(\psi_P \times i) + \frac{r_A \times \sin \sigma_P}{\cos \alpha} \times \sin(\alpha + \psi_P \times i) - \frac{r_w}{i}
\end{aligned}
$$

The generating radius R_w was added to bring the y_c coordinate to the pitch line.

Since for the hobbing process $i \rightarrow 0$, expressions[2] can be partially simplified as:

$$x_c = -\frac{r_w}{i} \times \sin(\psi_P \times i) + r_A \times \sin \sigma_P$$

$$y_c = \frac{r_w}{i} \times \cos(\psi_P \times i) + r_A \times \sin \sigma_P \times \tan \alpha - \frac{r_w}{i}$$

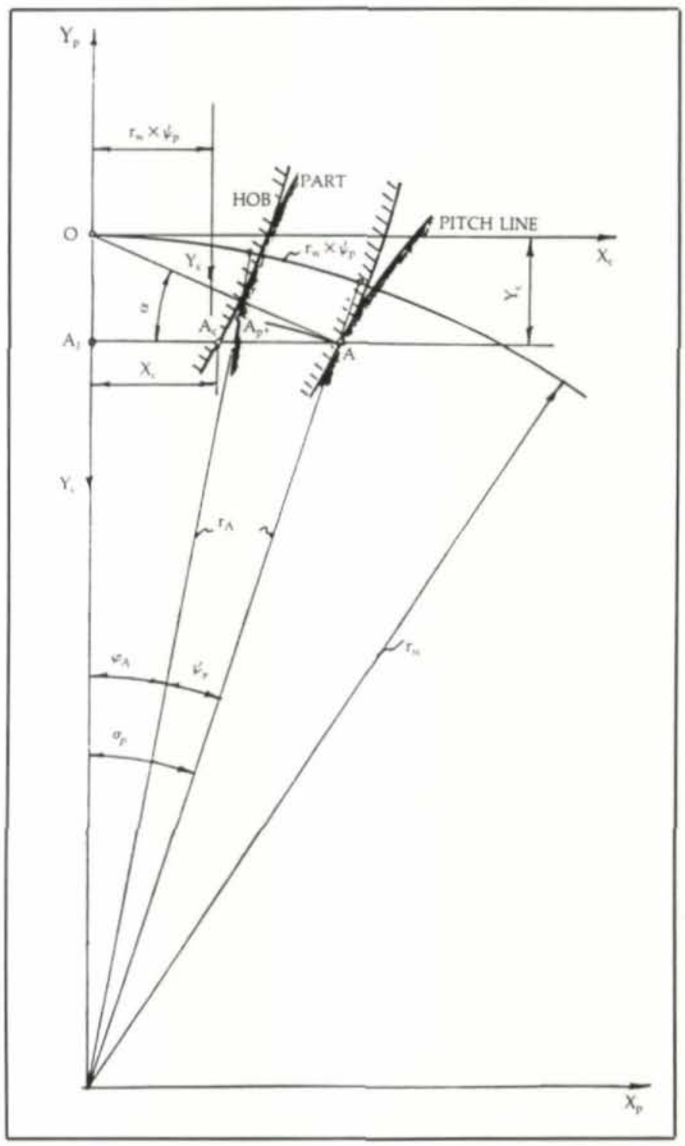


Referring to Fig. 3 it is interesting to note that equations[1] and [3] give the same results, with sufficient accuracy, using $R_w = 10^6$ in[1].

Reverse Calculations

Reverse calculations – when cutter profile is known – are very useful in many cases. Being calculated discretly, the cutter profile has to be approximated by certain curves (or by straight lines). As a result, the actual part profile will deviate from the theoretical one. The same kind of calculations have to be made when the tool designer has to decide whether a cutter "on hand" can be used to cut a part with a slightly different profile. Reverse calculations are also of importance for protuberance and lug design, when analyzing fillets, tip reliefs, and so on.

For gear shaping process, the equations[1] can be readily used. It is sufficient to consider R_c, φ_c, μ_c instead of r_A, φ_A, μ_A to receive mating part coordinates. For hobbing,



To evaluate these expressions as $i \to 0$ we apply the theorem of limits (L'Hospital's Rule):

$$\lim_{i \to 0}\left\{-\frac{r_w \times \sin(\psi_P \times i)}{i}\right\} = \frac{\lim_{i \to 0}[r_w \times \sin(\psi_P \times i)]'}{\lim_{i \to 0}(i)'} = -r_w\,\psi_P$$

$$\lim_{i \to 0}\left\{\frac{r_w \times \cos(\psi_P \times i)}{i}\right\} = \frac{\lim_{i \to 0}[r_w \times \cos(\psi_P \times i)]'}{\lim_{i \to 0}(i)'} = -0$$

$$\lim_{i \to 0}\left\{\frac{r_w}{i}\right\} = \frac{r_w}{\lim_{i \to 0}(i)'} = r_w$$

Substituting these expressions into[2] we obtain:

$$\begin{aligned} x_c &= r_A \times \sin\sigma_P - r_w \times \psi_P \\ y_c &= r_A \times \cos\alpha_P - r_w \end{aligned} \tag{3}$$

equations[3] must be rearranged as follows:

$$r_A = \frac{y_c + r_w}{\cos \sigma_P}$$

$$\psi_P = \frac{r_A \times \sin \sigma_P - x_c}{r_w}$$

$$\varphi_A = \sigma_P - \psi_P$$

The formula for σ_P is obvious from geometric conditions shown in Fig. 3:

$$\tan \sigma_P = \frac{AA_1}{r_w + y_c} = \frac{y_c}{\tan \alpha\ (r_w + y_c)},$$

where α is a slope of the tangent to hob profile.

The coordinates of the path, traced by any point of shaper cutter during generating action, can be found from the same equations[1]. Assuming x_C and y_C are constant we get:

$$\tan (\alpha + \psi_C) = \frac{y_c - R_w \times \cos \psi_c}{x_c + R_w \times \sin \psi_c}$$

Assigning different values to ψ_c we compute α, x, r_A, σ_P, ψ_P and φ_A. The r_A and φ_A coordinates will describe the location of the point A_c in the X_P-Y_P coordinate system at any moment with respect to angle ψ_c. We recommend the following formula for r_A:

$$r_A = \sqrt{\frac{x^2}{\cos^2 \alpha} - 2 \times x \times r_w \times \tan \alpha + r_w^2}$$

The formula was obtained from the second, third and fourth expressions of[1].

The same approach for hobbing gives:

$$\tan \sigma_P = \frac{x_c + r_w \times \psi_P}{y_c + r_w}$$

Conclusion

1. The method of gear cutting tool calculations discussed in this article are applicable for hobs and gear shaper cutters with involute and noninvolute forms. For precise hobs, the equations should be used for rack calculations followed by three-dimensional calculations of the hob cutting edges.
2. The general equations are based on the same, comparatively simple geometric approach and give a good understanding of the generating process. They can be successfully used for analitical solutions of a number of problems in gear cutting design.
3. The method allows one to build simplified and reliable computer programs.

The example below represents a spline profile with the following dimensions:

$$\beta = 14.5°$$
$$a = 0.77316$$
$$n = 30$$
$$N = 22$$
$$r_o = 2.7395$$
$$r_r = 2.6645$$
$$r_w = 2.6845$$
$$R_w = 1.968633$$

Calculations given in the table were made for $r_A = 2.7$.

$$\mu_A = \arcsin \frac{a}{r_A} = 16.639894°$$

$$\psi_A = \frac{180°}{30} - (\mu_A - \beta) = 3.860106°$$

u	2.586933	1st formula (1)
α	15.494625°	2nd formula (1)
σ_P	− 1.145269°	3rd formula (1)
ψ_P	− 0.087360 rad	4th formula (1)
x	− 0.053966	5th formula (1)
ψ_C	− 6.825512°	6th formula (1) not applicable for hob
x_{HOB}	0.180552	1st formula (3)
y_{HOB}	− 0.000539	2nd formula (3)*
x_{SC}	0.178603	7th formula (1)
y_{SC}	1.946240	8th formula (1)

*the hob profile coordinate is taken from the pitch line

Editors Note: Special thanks to Dennis Gimpert, American Pfauter Ltd., for his technical editing assistance.

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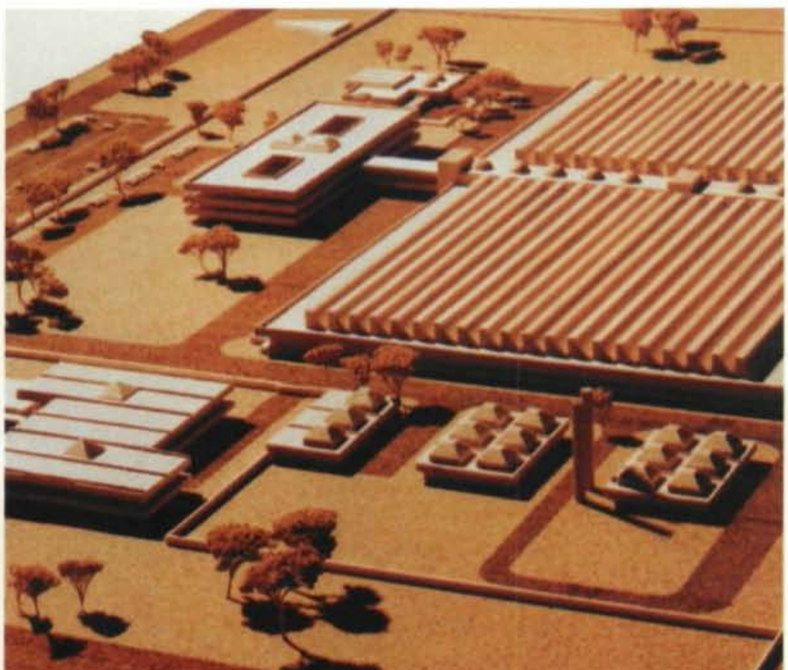

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Finding Gear Teeth Ratios

by
William C. Orthwein
Department of Mechanical Engineering
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Carbondale, Illinois

When designing gears, the engineer is often faced with the problem of selecting the number of teeth in each gear, so that the gear train will provide a given speed ratio. This article will describe a microcomputer program that determines such combinations of gear teeth quickly, easily, and accurately.

The program is founded upon the theorem that a denumerable infinity of rational numbers can be found between any two real numbers whose difference is greater than zero. Practically, if the acceptable error for the desired gear ratio is of the order of several hundred times the minimum numerical values that the computer can process without truncation errors, it is usually possible to find multiple combinations of gear teeth which provide that ratio to within required accuracy.

Program Description

Input to the method are the desired ratio R and the acceptable error E.[1] The method produces two integers P and Q, where P/Q differs from R by an amount equal to or less than E. The search for P and Q is begun by defining quantities a_1, y_1, p_1, and q_1 as

$$a_1 = \mathrm{INT}(R) \qquad y_1 = \mathrm{FRC}(R)$$
$$p_1 = a_1 \qquad q_1 = 1 \tag{1}$$

Quantities a_2, y_2 p_2, and q_2 and qa_2 are then defined according to

$$a_2 = \mathrm{INT}\left(\frac{1}{y_1}\right) \qquad y_2 = 1 - a_2 y_1$$
$$p_2 = a_1 a_2 + 1 \qquad q_2 = a_2 \tag{2}$$

If $y_2 = 0$ the process stops because $P = p_2$ *and* $Q = q_2$. If $y_2 \neq$o the process continues until

$$E \geqq \left| R - \frac{p_N}{q_N} \right| \tag{3}$$

where p_N and q_N represent the values of p_n and $q_n$which satisfy (3) for the smallest n. In (3), p_n and q_n are determined by the relations

$$a_n = \mathrm{INT}\left(\frac{y_{n-2}}{y_{n-1}}\right) \qquad y_n = 1 - a_n y_{n-1} + y_{n-2}$$
$$p_n = a_n p_{n-1} + p_{n-2} \qquad q_n = a_n q_{n-1} + q_{n-2} \tag{4}$$

for $n \geqq 3$. The computational procedure using these relations is described in the flowchart in Fig. 1.

Fig. 2 is a flowchart of the main program, Gearratio. It

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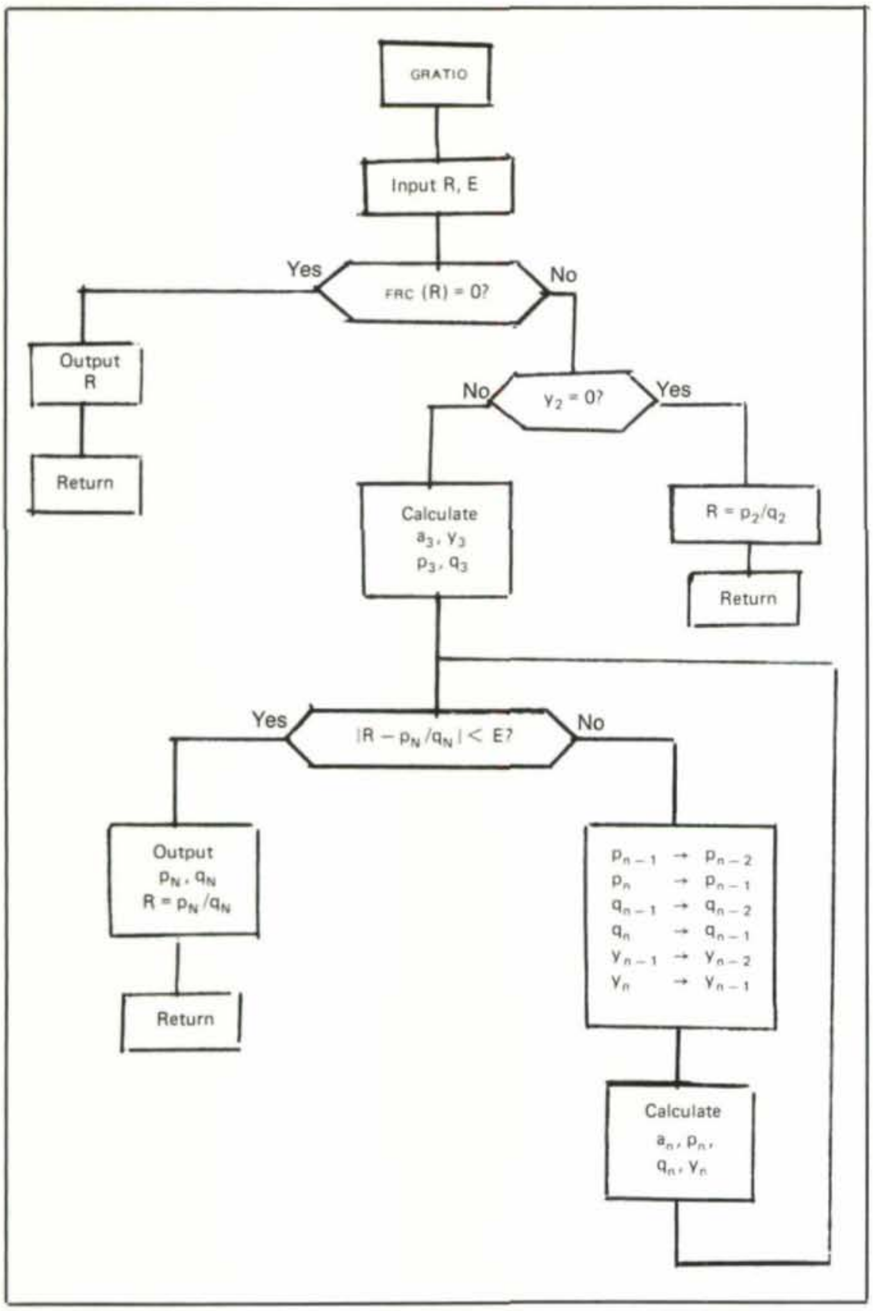

Fig. 1 – Flowchart for the subprogram GRATIO.

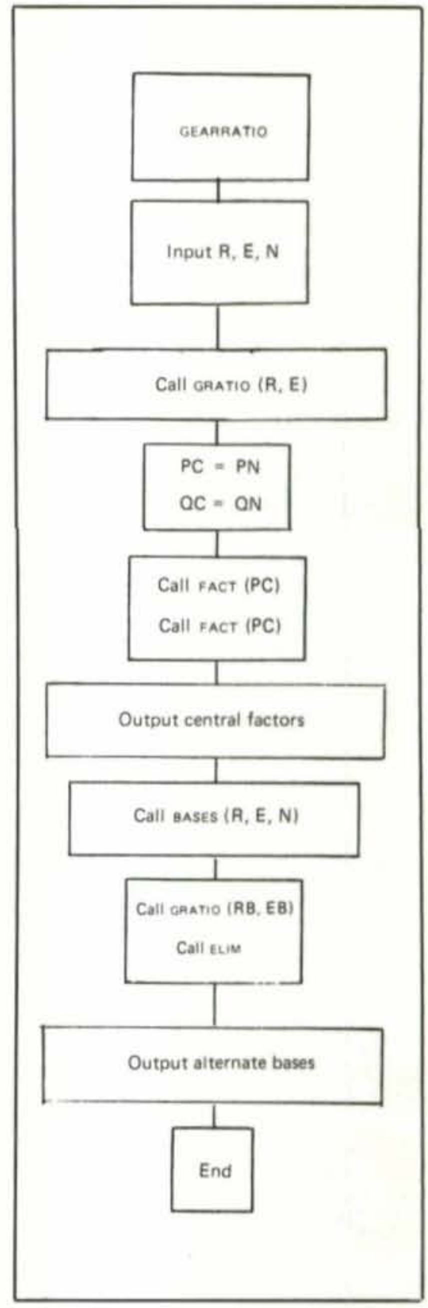

Fig. 2 – Flowchart for the main program Gearratio.

Nomenclature

a_n	value of parameter a at the nth step
E	acceptable error in approximating the desired gear ratio
FRC(R)	fractional part of R: FRC(R) = R − INT(R)
INT(R)	largest integer in R
P	numerator of a rational number (fraction) approximating R
p_n	value of P at the nth step
p_N	final value of p_n
Q	denominator of a rational number approximating R
q_n	value of Q at the nth step
q_N	final value of q_n
R	desired gear ratio
R_i	additional gear ratios which differ from R by less than E
y_n	value of y at the nth step

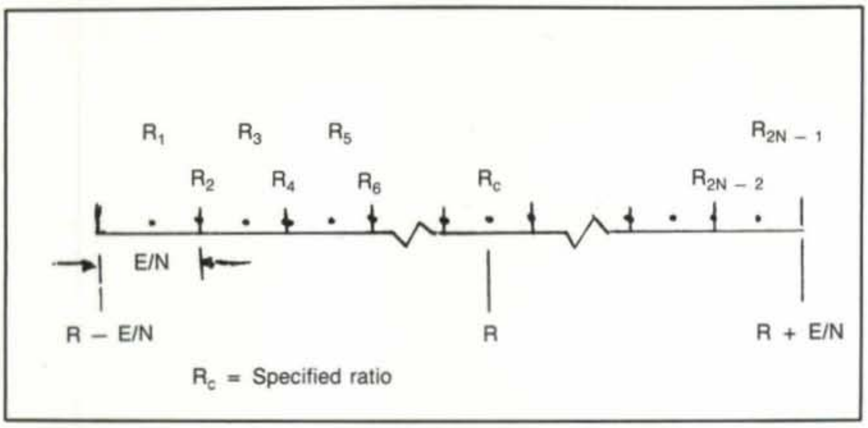

Fig. 3 – Locations of R_i and R in the interval E. N subdivisions of length E/N are shown.

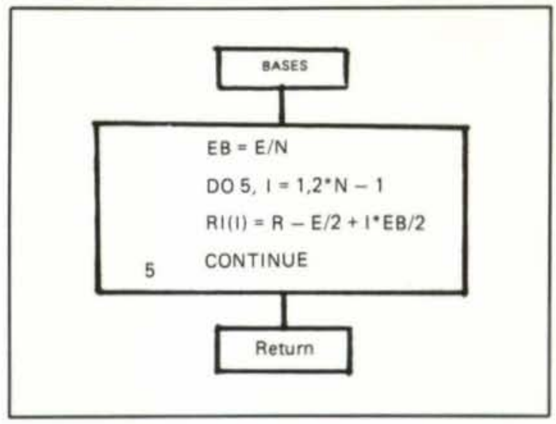

Fig. 4 – Flowchart for subroutine BASES.

uses the acceptable error E to define an interval $R - E/2$ to $R + E/2$. Other ratios R_i can be selected using error limits that are reduced in proportion to the number of subdivisions of E (Fig. 3). Division of E into N equal subdivisions each of length E/N is performed by the four-line subprogram labeled BASES (Fig. 4) This subroutine then locates $2N - 1$ additional base ratios R_i by the relation

$$R_i = R - \frac{E}{2} + (2i - 1)\frac{E}{2N} \qquad (5)$$

Each base ratio is associated with a permissible error of magnitude E/N, to ensure that all calculated values fall within the originally specified interval. Values P and Q for each additional base ratio are calculated after the original base ratio R is used to find the central P and Q values to within permissible error E.

Different P and Q values can be found by a variety of methods. For example, the permissible error E can be allowed to grow as the base ratios move closer to the center of the range. Or, one can use an uneven distribution of base ratios over the interval and locate the base ratio at other than the center of the subintervals of length E/N. In the following examples, the results are generated using only base ratios at a set of equally spaced points associated with an error of $E/100$. This division produces 198 additional base ratios (199 minus the central base ratio already calculated), including duplicates. Elimination of duplicates is achieved by the subroutine ELIM (Fig. 5).

After all duplicates are eliminated the remaining values of P and Q are decomposed into their prime factors by the subroutine FACT (Fig. 6). Output of the factors is controlled by a write statement in this subroutine.

Example 1

In this example problem we will find the numbers of teeth in each gear in a gear train which is to have an overall speed ratio of 2.94643 ± 0.0001. There should be at least 18 and at most 200 teeth on each gear.

Because P and Q generally do not contain an equal number of factors, it may be necessary to append unit factors to each. The factors in P and Q are then arranged in ascending or descending order and grouped in pairs. Using the gear train shown schematically in Fig. 7, P and Q can be written in the form

$$\frac{N_1}{N_2}\ \frac{N_3}{N_4}\ \frac{N_5}{N_6}$$

Comparison with gear train shows that (6) implies a gear train of six gears in which N_1 and N_2, N_3 and N_4, and N_5 and N_6 are in mesh. Gear N_1 is on shaft 1, gears N_2 and N_3 are on shaft 2, gears N_4 and N_5 are on shaft 3, and gear N_6 is on shaft 4.

After each P/Q ratio is arranged as in (6), any pair which contains a factor less than 18 is multiplied by the smallest integer required to make that factor equal to or greater than 18. When this is completed, it may be necessary to rearrange the P and Q factors to have the modified factors again be in ascending or descending order. Had the ratio in this example been unusually large, it is possible that this multiplication could cause the other factor in a P/Q pair to be larger

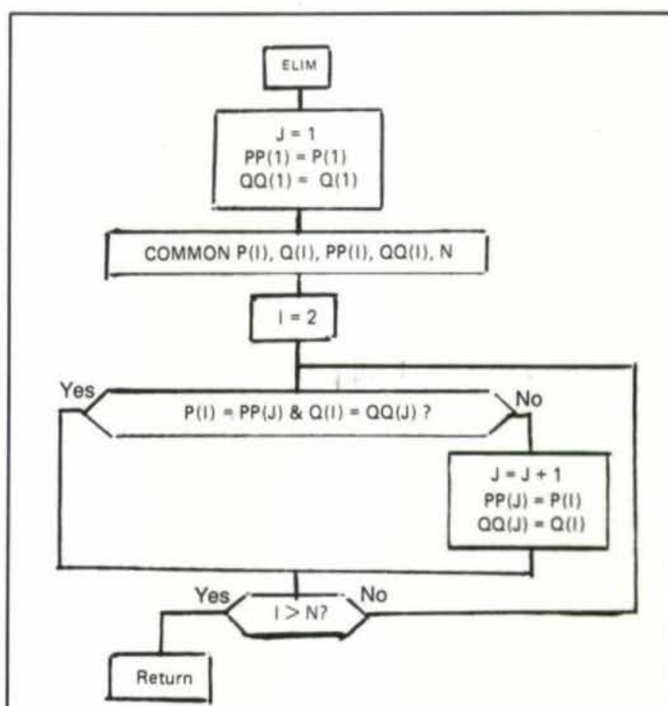

Fig. 5 – Flowchart for subprogram ELIM.

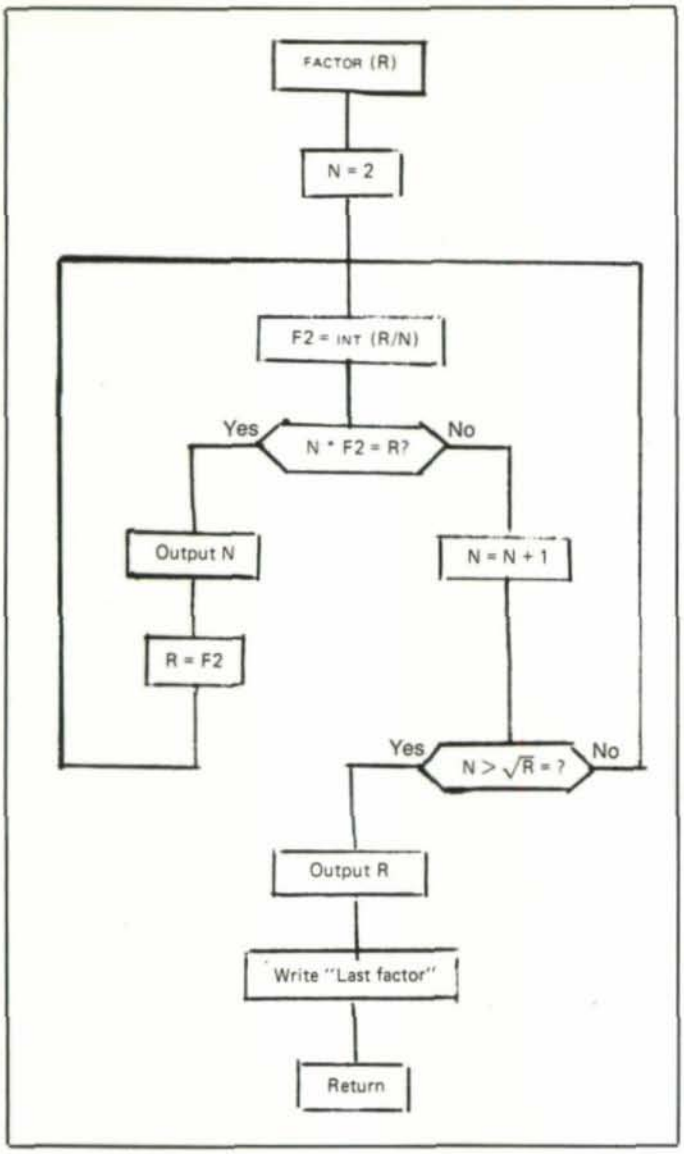

Fig. 6 – Flowchart for subprogram FACT.

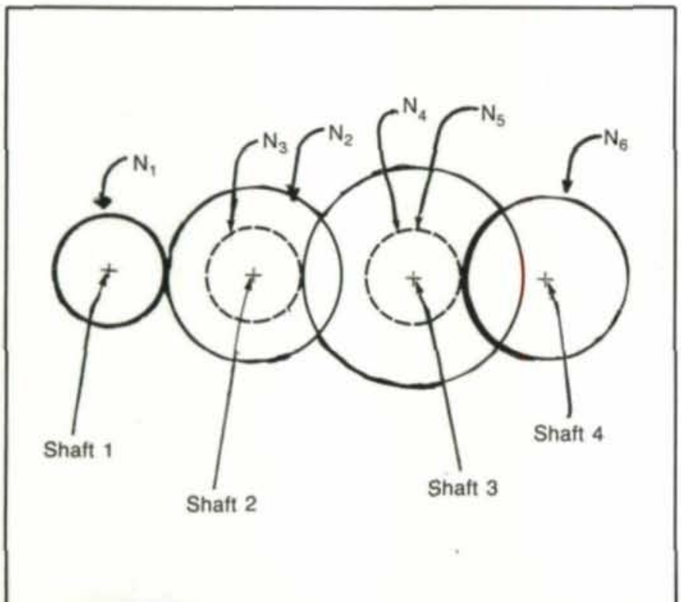

Fig. 7 – Schematic of gear train.

Table I

Tooth Combinations to Provide a Gear Ratio Of 2.94643 ± 0.0001 (18 ≤ N ≤ 200)

Ratio Number	Program Output Ratio	Gear Tooth Numbers	Error (x 10^6)
1	$\frac{3(5)11}{(2^3)(7)}$	$\frac{45}{24}\ \frac{33}{21}$	1
2	$\frac{2^4(79)}{3(11)13}$	$\frac{32}{22}\ \frac{79}{39}$	43
3	$\frac{29(127)}{2(5^4)}$	$\frac{29}{25}\ \frac{127}{50}$	30
4	$\frac{2(7^2)23}{(3^2)5(17)}$	$\frac{28}{20}\ \frac{28}{18}\ \frac{46}{34}$	25
5	$\frac{2(31)47}{23(43)}$	$\frac{47}{23}\ \frac{62}{43}$	19
6	$\frac{(2^6)\ 61}{(5^2)\ 53}$	$\frac{64}{53}\ \frac{61}{25}$	15
7	$\frac{3(5)11}{23(7)}$	$\frac{33}{23}\ \frac{20}{28}$	14
8	$\frac{137(163)}{11(13)53}$	$\frac{163}{143}\ \frac{137}{53}$	1
9	$\frac{(2^2)19(131)}{31(109)}$	$\frac{76}{31}\ \frac{131}{109}$	4
10	$\frac{(2^2)7(167)}{3\ (23^2)}$	$\frac{28}{23}\ \frac{167}{69}$	10
11	$\frac{2(41)53}{(5^2)\ 59}$	$\frac{53}{25}\ \frac{82}{59}$	11
12	$\frac{37(113)}{3(11)43}$	$\frac{37}{33}\ \frac{113}{43}$	11
13	$\frac{2(19)97}{(3^2)\ 139}$	$\frac{194}{139}\ \frac{38}{18}$	13
14	$\frac{2(17)89}{13(79)}$	$\frac{68}{26}\ \frac{89}{79}$	16
15	$\frac{2(13^2)7}{11(73)}$	$\frac{28}{22}\ \frac{169}{73}$	21
16	$\frac{31(71)}{(3^2)\ 83}$	$\frac{62}{18}\ \frac{71}{83}$	22
17	$\frac{17(191)}{2(19)29}$	$\frac{34}{38}\ \frac{191}{58}$	31
18	$\frac{(3^4)\ 53}{31(47)}$	$\frac{81}{47}\ \frac{53}{31}$	35
19	$\frac{7(173)}{3(137)}$	$\frac{42}{18}\ \frac{173}{137}$	42
20	$\frac{(2^2)\ 3(17^2)}{11(107)}$	$\frac{24}{22}\ \frac{289}{107}$	44

(continued on page 48)

Technological Fundamentals of CBN Bevel Gear Finish Grinding

by
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INTRODUCTION

The bevel gear grinding process, with conventional wheels, has been limited to applications where the highest level of quality is required. Grinding with conventional wheels has not been used in high production applications, because of the long cycle times and resultant high manufacturing costs. A further hindrance to the wider application of bevel gear grinding has been the lack of an understanding of the fundamental principles involved in the process. Rather than basing it on applied engineering knowledge, gear grinding success has been dependent upon the experience of skilled grinding machine operators. This can make it difficult to get consistent results from day to day.

Dr. Kegg[1] has noted that the industry trend is toward higher part quality, indicating increased use of grinding, but with a diminishing supply of skilled machine operators.

In addition to these diverging tendencies, a new grinding wheel abrasive (CBN) has been introduced. Much information has been published highlighting the benefits of CBN grinding, such as reduced grinding time and overall costs, maintaining consistent geometrical tolerances and improved metallurgical integrity. Despite these promised results, CBN grinding has been slow to be implemented in high volume production applications because the additional technology also has not been well understood by many potential users.

During the past few years a significant amount of fundamental grinding research has been conducted by several researchers [2-17]. This article will show how the fundamental research has been applied to CBN bevel gear grinding. The understanding and implementation of applied grinding technology has made it possible to change the grinding machine from the mysterious black box of the past to a system having predictable and consistently repeatable results.

The combination of a new grinding tool material (CBN), new applied grinding technology, and new bevel gear grinding machines has made it possible to drastically reduce the grinding time and cost, while at the same time improving quality and consistency, making bevel gear grinding technically and economically feasible for mass production.[18]

FUNDAMENTALS OF CBN GRINDING OF HARDENED MATERIALS

CBN Physical Propeties

Elevated Temperature Hardness

Fig. 1 shows the elevated temperature hardness of the four major abrasives[12].

The temperatures encountered by the cutting points of the abrasive grains are well above room temperature, thus the elevated temperature properties are of greater significance than the room temperature properties. It can be seen that the hardness of diamond decreases at a much faster rate than the other three abrasives with increasing temperature. In addition, diamond wears rapidly by graphitization or oxidation in the presence of iron at high temperatures and is, therefore, not usually successful in grinding ferrous materials. CBN on the other hand maintains its hardness advantage over silicon carbide and aluminum oxide at all temperatures up to 1830°F (1000°C) and is chemically inert in the grinding of ferrous materials.

Grain Shape and Its Influence on the Rate of Wear Flat Development

Attritious wear, the slow gradual development of a wear

AUTHORS:

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MR. K. V. KUMAR *is Research Project Engineer at The Gleason Works, and is actively involved in hard finishing process development for bevel gears. He received his MS from Carnegie Mellon University and PhD from Arizona State University in Mechanical Engineering. He is an associate member of ASME.*

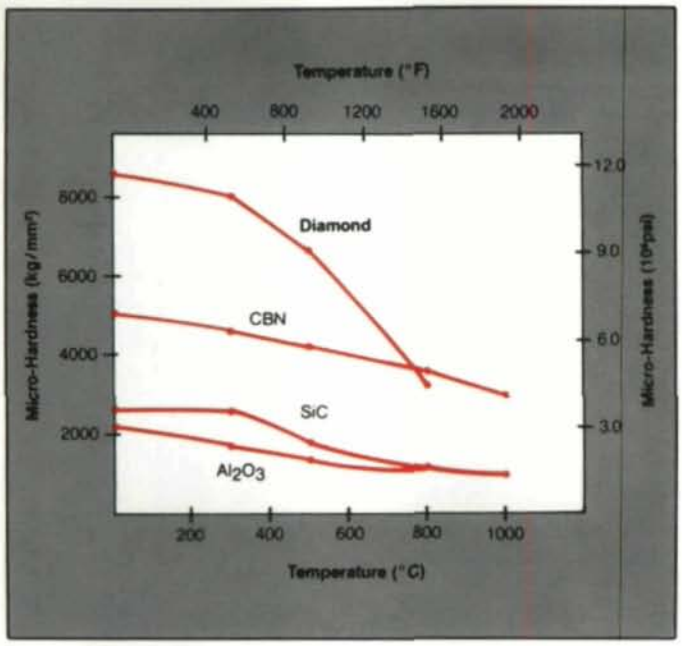

Fig. 1 – Micro-Hardness of CBN in terms of the temperature in comparison to other abrasive grains.[12]

flat, limits the useful operation of a grinding wheel during finish grinding.

Fig. 2 shows photographs of an aluminum oxide and a CBN crystal. The CBN crystal has a well-defined structure, while the aluminum oxide grain does not. Aluminum oxide crystals on the average have a spherical form[19] while the CBN crystals have a block form shown in Fig. 2. It can be seen that for a given amount of crystal wear the length of the wear flat will be significantly greater for the aluminum oxide. A comparison of wear flat areas shows an even greater difference.

The size of the wear flat area has a direct effect on burning. Malkin[20] has shown that when a wear flat area of 4% of the wheel surface has developed, burning is encountered with conventional wheels. The cubic structure of CBN will allow for greater radial wear before the same size wear flat is developed.

Anisotropic Crystal Strength

Like diamond, CBN has anisotropic strength properties. This can result in a self-sharpening action under certain

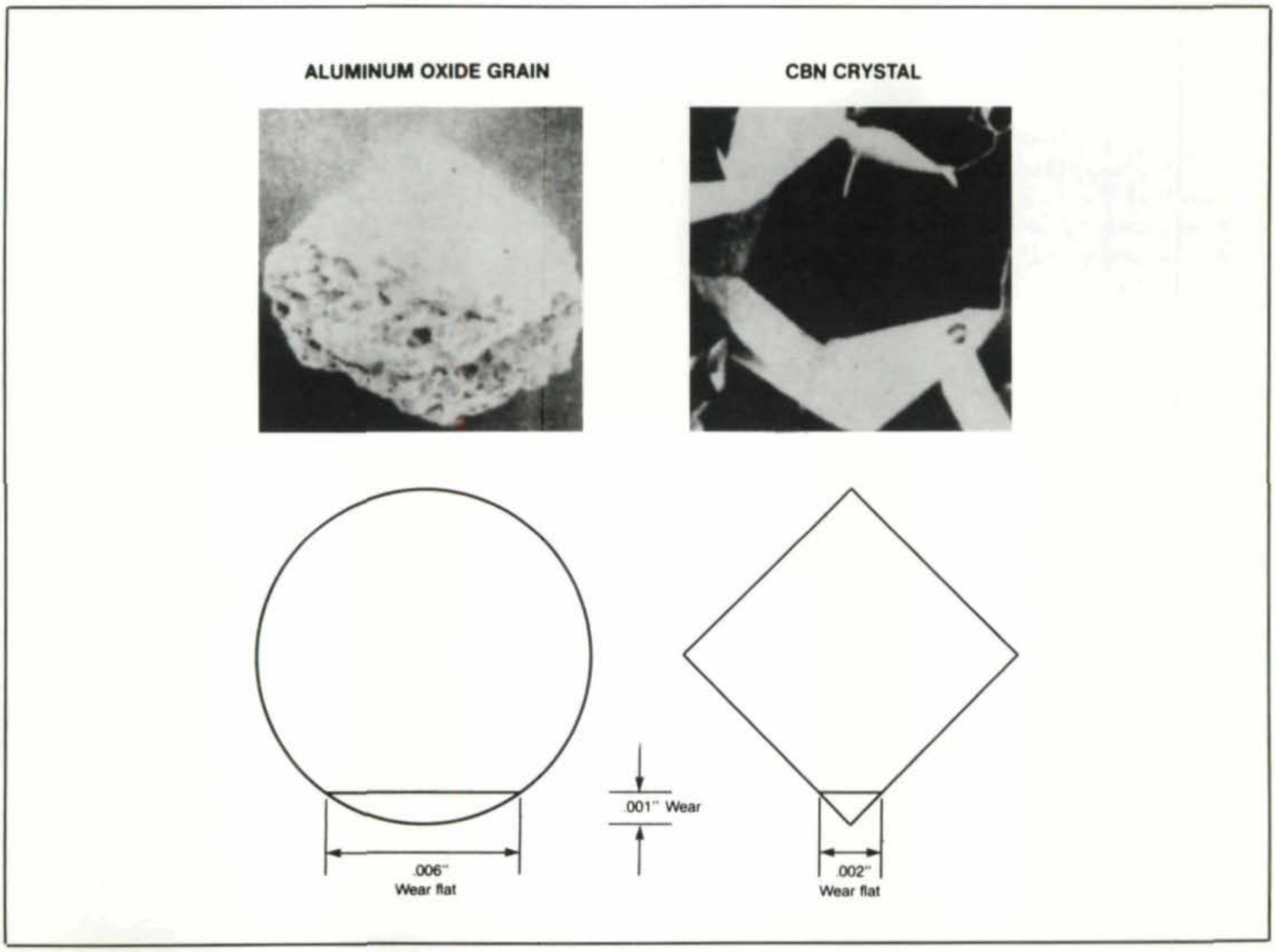

Fig. 2 – Effect of Grain Shape on the Rate of Wear Flat Development.

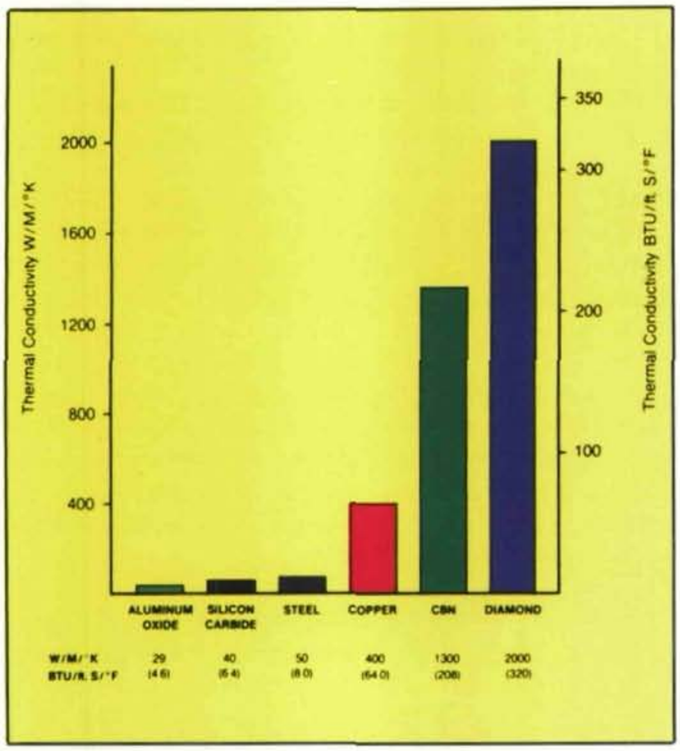

Fig. 3 – Comparison of Thermal Conductivity at Room Temperature.

conditions,[14] providing a new sharp cutting edge without crystal pullout.

Thermal Conductivity

Fig. 3 shows the thermal conductivity of the four major abrasives, and of steel and copper.[21,22] The significance of this property will be shown in the next section on thermal input to the workpiece.

Grinding Characteristics of CBN vs. Conventional Abrasives

Low Thermal Input With Increased Force and Power Requirement

In CBN grinding the forces and power are higher than with conventional wheels. CBN grinding machines thus require greater spindle stiffness and power. Salje[2] has shown that at the same metal removal rate a CBN wheel required 1.25 to 2.6 times the power of an aluminum oxide wheel. This has been rather puzzling since CBN is a harder and sharper abrasive. The reason for this difference in power requirement between the two abrasives can be explained by a simple analysis of the heat flow between wheel and work in the two cases.

Fig. 4 is a schematic of the surface grinding operation. It is assumed for simplicity that all of the power consumed in grinding is dissipated as heat between wheel and work only.

$$q = q_s + q_w \tag{1}$$

where q = total heat flux in grinding
q_s = heat flux to wheel
q_w = heat flux to work

Denoting k_s, and k_w as the thermal conductivities for wheel and work, A as the wheel-work contact area and t_s, t_w as thicknesses for wheel and work across which there is an equal temperature drop, then

$$q_s \frac{t_s}{k_s A} = q_w \frac{t_w}{k_w A} \tag{2}$$

The quantity within each bracket is the thermal resistance of wheel and work respectively. Since the thickness layer from the point of view of thermal damage to work and wheel is of similar magnitude, equation[2] can be simplified as

$$\frac{q_s}{k_s} = \frac{q_w}{k_w} \tag{3}$$

From equations (1) and (3)

$$q_s = \frac{q}{1 + (k_w/k_s)} ; \quad q_w = \frac{q}{1 + (k_s/k_w)}$$

Using the room temperature thermal conductivity values from Fig. 4, in the above relationships, for the case of an aluminum oxide wheel it is seen that 63% of the total heat generated goes into the work and 37% into the wheel. In the case of a CBN wheel it is seen that only 4% of the total heat generated goes into the work while 96% ends up in the wheel. In actual practice, thermal conductivity varies with temperature for both the abrasive and the work material. Nevertheless the amount of heat input to the work will be much less with CBN than with conventional abrasives. Therefore, chips are formed at a lower temperature with CBN. Since the forces required to work a material at lower temperatures are higher, CBN grinding is accompanied by higher forces and power requirements.

It should not be misconstrued from the above analysis that thermal damage to work is not possible with CBN wheels. Thermal damage can be produced with CBN, but when the

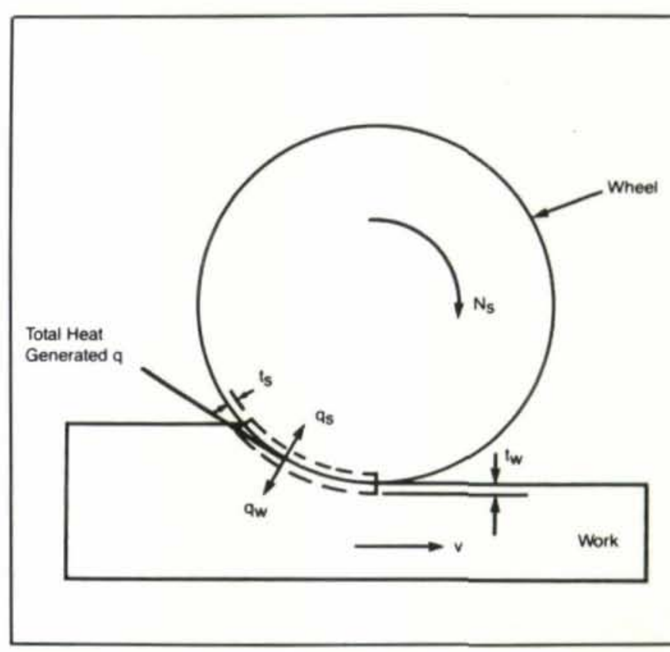

Fig. 4 – Heat Distribution between wheel and worm in surface grinding

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wheel is properly prepared this will happen at a much higher metal removal rate and grinding spindle power as compared to conventional wheels. On the other hand, the high heat input to the CBN wheel requires proper coolant application to increase the longevity of the wheel.

As a result of the lower heat input to the workpiece with CBN grinding wheels and a reduction in the tendency to burn, CBN wheels are normally used at higher metal removal rates than conventional wheels. This further increases the grinding forces and power. Thus, it is not unusual for the forces encountered in CBN grinding to be 4 to 10 times greater than with an aluminum oxide wheel.

Residual stresses and fatigue strength

The functional behavior of a ground component is substantially determined by the material physical properties, as well as, the residual stresses near the surface.(11,23) Residual stresses will have an effect on both the static strength and the dynamic strength (fatigue strength).

The initiation and growth of cracks can be accelerated or retarded by residual stresses. Tensile residual stresses will increase the possibility of crack initiation and growth, while compressive stresses will retard them.

The near surface residual stresses as a result of grinding can be produced by one or a combination of the following means:

thermal: heat generated during grinding;

mechanical: plastic deformation;

chemical: reactions with machining fluids and absorption of elements in the machined surface.(24)

Thermally induced residual stresses are tensile. The highly localized temperatures at the surface cause the work material to yield in compression as shown in Fig. 5. After the heat source has passed and the material cools, the plastically compressed surface material is left in tension. Snoeys(25) has found that increased maximum grinding temperatures cause higher peak residual tensile stresses. Bellows(26) has found that less abusive grinding conditions decrease the tensile residual stresses.

Fig. 5—Mechanism for Thermal-Plastic Induced Residual Stresses

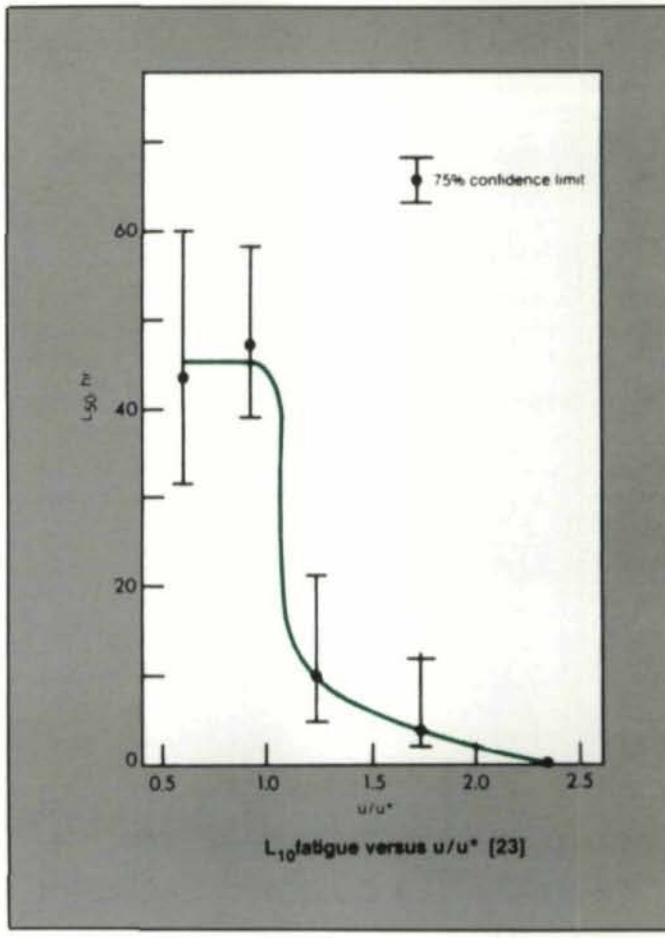


Fig. 6

The effect of various levels of thermal input on the fatigue life of bearing rings has recently been investigated.(23) In Fig. 6 it can be seen that the L10 fatigue life decreased dramatically with an increase in U/U* (the ratio of the measured specific energy to the specific energy at burning). This ratio is directly proportional to the grinding power and the grinding temperature; and points out that higher thermal input to the work surface will greatly reduce the fatigue life.

When the grinding temperature is sufficiently low, residual stresses are caused primarily by plastic deformation. This mechanical cold working of the surface material results in compressive residual stresses in a manner similar to shot peening. Also, at lower temperatures chemical reactions are less likely, or at least will often proceed at a slower rate.

Many investigators have compared the residual stresses produced by CBN and conventional wheels and have shown that CBN wheels tend to produce compressive residual stresses while aluminum oxide wheels tend to produce tensile residual stresses.(4,5,13,15,27,28,29) This difference can be explained by the previously discussed differences in thermal conductivity of the two abrasives. This proposition is further supported by the grinding tests of Ratterman(5) in which the residual stresses produced by CBN, diamond, and conventional wheels were compared. The test results showed that diamond wheel grinding produced residual compressive stresses which were larger (more compressive) than that from

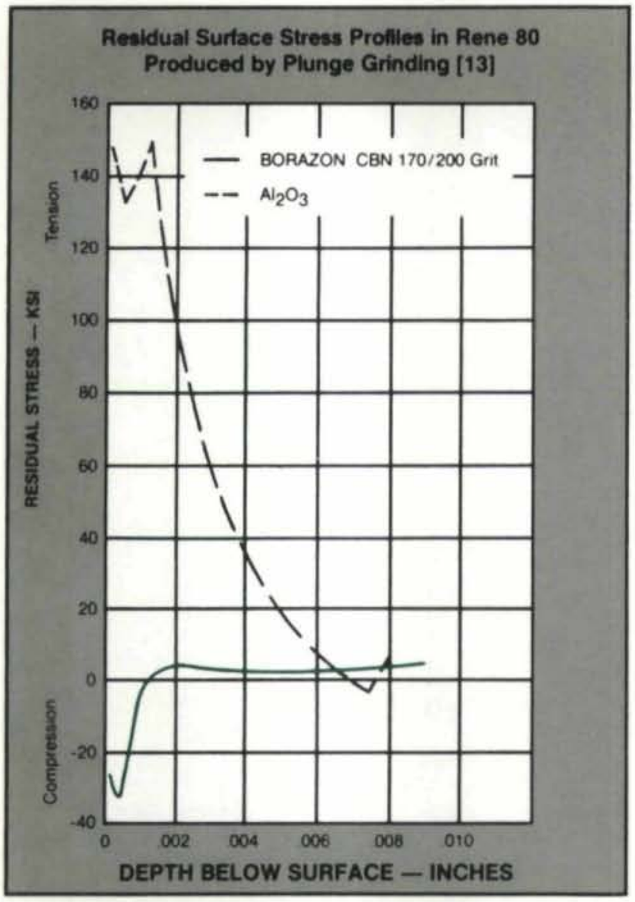
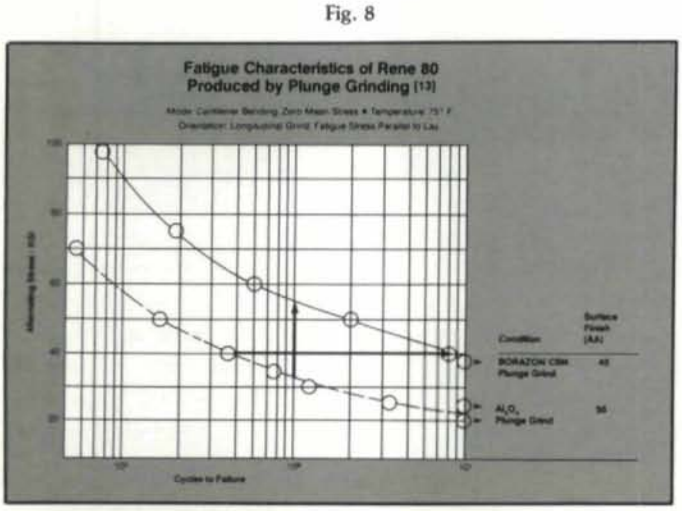


Fig. 7

CBN wheel grinding. From Fig. 3 it can be seen that diamond has a higher thermal conductivity than CBN. Therefore, in grinding with diamond the percentage of heat going into the workpiece is less than with CBN, causing greater residual compressive stresses.

Metcut Research Associates compared the residual stress and fatigue life of parts ground with CBN and aluminum oxide wheels.[13] The measured residual stresses are shown in Fig. 7 and the resulting S-N curve is shown in Fig. 8. These tests show that the bending fatigue life was increased by 27 times at an alternating stress of 40,000 lbs/in^2 or that the load carrying capacity for a life of one million cycles could be increased by 70%. Although the material in this test was not hardened steel, it clearly shows the effect of decreased thermal input.

Fig. 8



It is clear that CBN has a tendency to cause beneficial compressive residual stresses, resulting in increased fatigue life, because CBN grinds at lower temperatures. Any factor that reduces the grinding temperature will also reduce the tendency for wheel loading, chemical reactions, and wheel dulling, as well as burning and tensile residual stresses.

Furthermore, the presence of grinding burns results in metallurgical changes such as tempering (softening resulting in lower strength) and/or the formation of untempered martensite (a very hard and brittle material). These defective materials, at the point of highest stress for gear tooth bending, have a lower strength than the materials of proper hardness.

Fundamentals of CBN Grinding of Bevel Gears

Wheel Workpiece Contact Conditions

In the bevel gear grinding process the tooth profile shape is produced by the relative rolling motion that takes place between the gear or pinion and the grinding wheel. The action is as though the gear or pinion being ground were rolling with an imaginary motion generating gear of which the grinding wheel represents one tooth (see Fig. 9).

Fig. 10 is a cross section through the grinding wheel profile, showing how the wheel and the gear move together in a timed relationship to generate the tooth profile on each side of the tooth slot. At position 1 the wheel first contacts the gear tooth at the top of the outside wheel profile. As the generation continues to position 2, the wheel rolls down the profile to the pitch line. Grinding with the outside diameter of the wheel is almost completed at position 3. When both tooth profiles have been completely generated at position 4, the wheel is withdrawn from the tooth slot and returned to

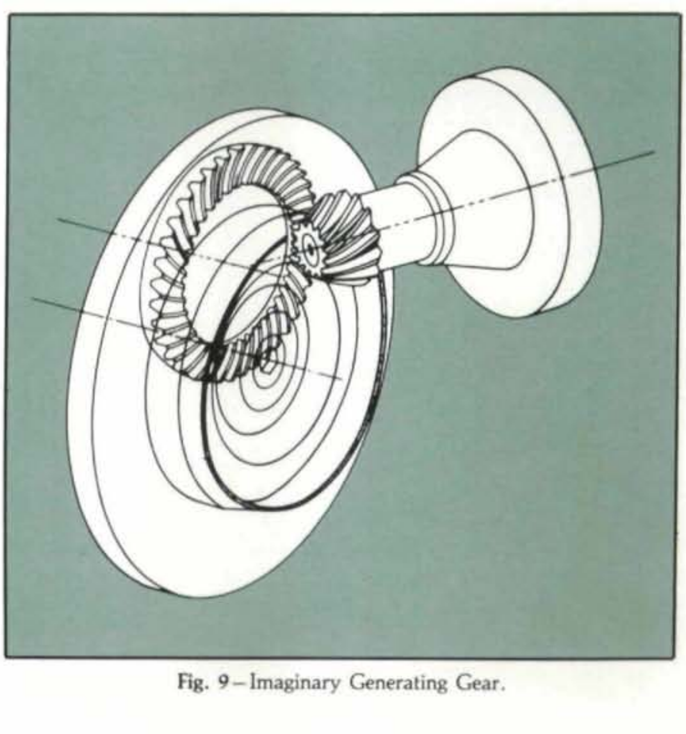


Fig. 9 – Imaginary Generating Gear.

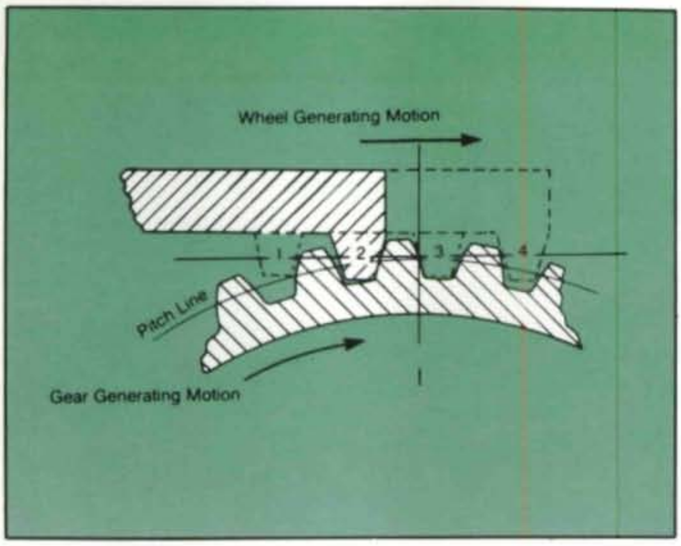


Fig. 10 – Generating Roll.

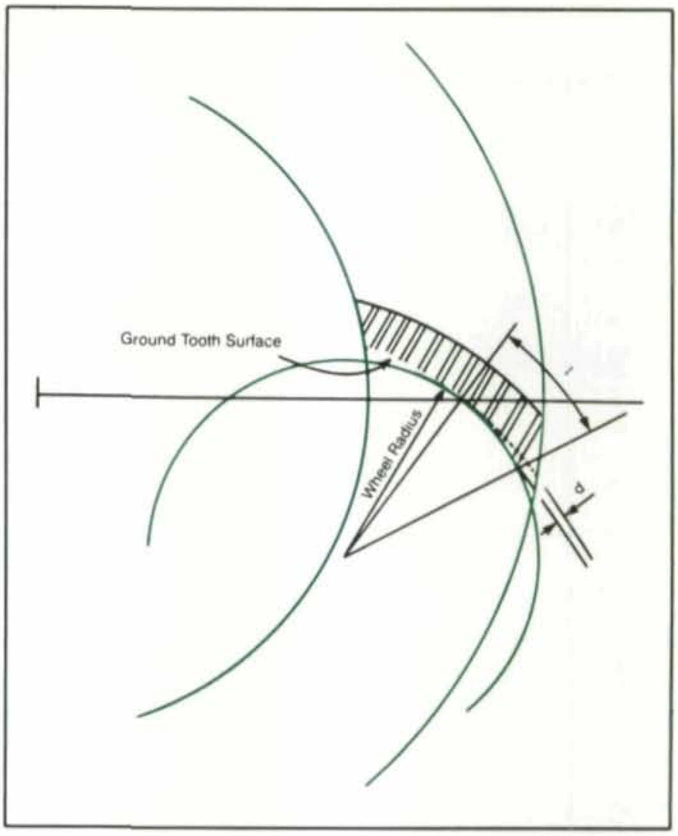


Fig. 12 – Length of Wheel – Work Contact.

position 1, while the gear is indexed to grind the next tooth slot.

Fig. 11 shows a three-dimensional representation of how the wheel/work, contact area starts at position 1 and proceeds to positions 2 and 3. The abrasive grains cut in a direction parallel to the root line.

A lengthwise section of the tooth in position 2 (Fig. 12) shows how the gear tooth wraps around the wheel in a way quite similar to internal grinding. The contact length will be very long as shown in Fig. 13. It can be seen that when the depth of grind is .001" the contact length can be two to five times longer than when surface grinding at the same depth of grind.

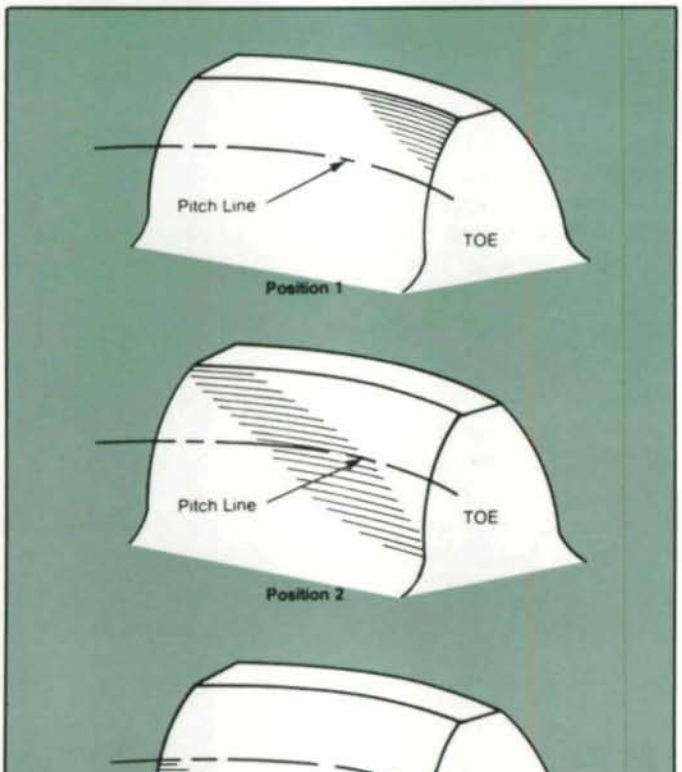


Fig. 11 – Area of Contact of Gear Tooth Surface with Conical Wheel Surface.

When finish gear grinding is done in a single pass at .005" depth of grind, an equivalent length of contact in surface grinding would not be encountered until a depth of grind of .080" was used. This shows that bevel gear grinding contact length can be as long as that encountered in creep feed grinding.

Thermal Aspects

From Fig. 12 it can be seen that all along the contact length,[1] the distance (d) to what will soon be the ground tooth surface is very small. Thus, the heat generated in the contact zone is easily conducted to the gear tooth surface. Furthermore, due to the fact that the feed rate is limited by machine dynamics (the acceleration and deceleration of the generating system), the amount of time that the long contact length heat source is over a given point on the gear tooth is relatively long, allowing for a greater temperature rise of the work material.

The required grinding power is also increased because of the long contact length and the resulting increased amount of rubbing encountered.

These factors indicate some of the reasons why bevel gear grinding with conventional wheels has been sensitive to thermal damage and one reason why the depth of grind has been limited to .001" or less, which results in low metal removal

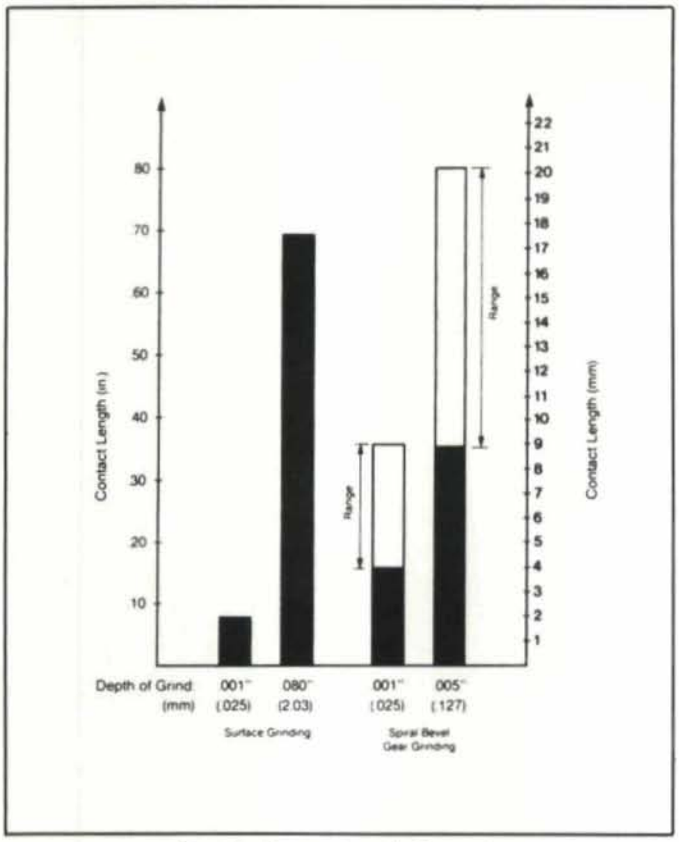


Fig. 13 – Contact Length Comparison.

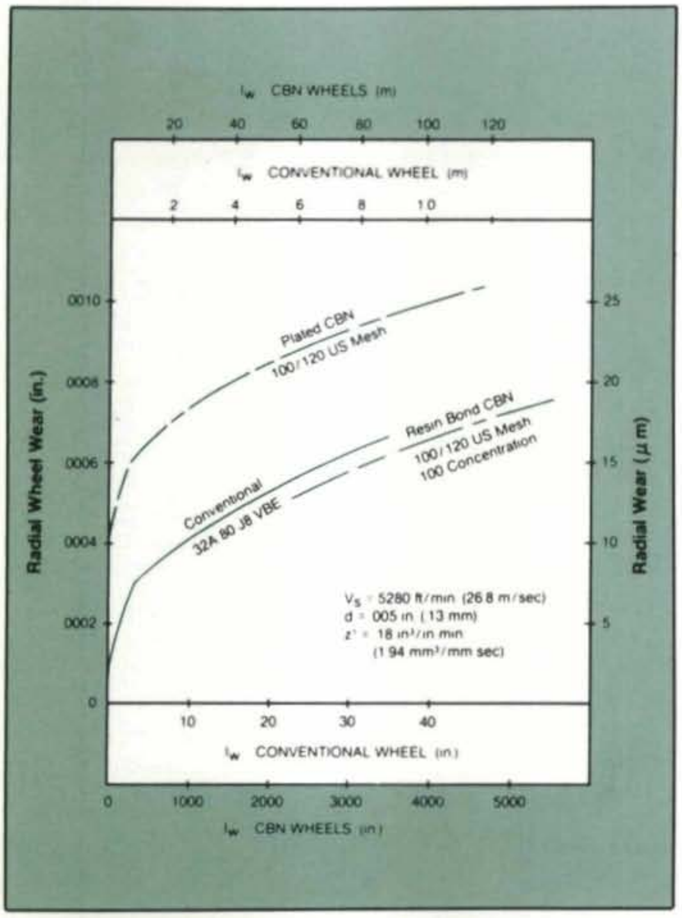


Fig. 14 – Laboratory Wheel Wear Rate Comparison.

rates and long grinding cycle times (typically 40-250 sec/tooth). The long cycle times are due to the necessity of multiple grinding passes and several wheel dressings per gear ground.

It has been found that the thermal input to the gear tooth surface can be so substantially reduced, due to the enhanced properties of CBN, discussed in the section on "Fundamentals of CBN Grinding of Hardened Materials," that all the finishing stock can be removed in a single pass without thermal damage, at a cycle time as short as 4.0 seconds per tooth. The result of the cool cutting action of CBN is also indicated by the substantially improved fatigue life of CBN ground gears as discussed later in the section on "Results of CBN Gear Grinding."

Wheel Wear Laboratory Results

The wheel wear as a function of the length of the gear tooth slot ground is shown in Fig. 14. The wear of a vitrified aluminum oxide wheel is compared to a plated and a resin bond CBN wheel. The slot length scale for the CBN wheels are 100 times the conventional wheel scale. All three wheels showed a rapid initial wear, followed by a slow steady wear rate. It can be seen that the slope of the steady wear rate for each of the wheels is about the same, which means that the CBN wheels wore at a rate 1% of the conventional wheel wear rate.

A borderline burning condition was encountered with the conventional wheel after grinding only 35" of slot length (1 pinion). After grinding 125 to 150, times the slot length with the CBN wheels, the thermal input to the pinion tooth surface was still far from the burning limit. Thus, the CBN wheels were still capable of grinding more parts.

As a result of such encouraging laboratory results and the desire to determine how this process might work in a production environment, it was decided to conduct further wheel life tests in a gear manufacturing plant.

Production Results

Grinding trials in a production environment showed that more than 90,000 inches of slot length (1.42 miles, 2050 gears) could be ground with a single plated CBN wheel, with the wheel still capable of grinding more parts. This was 20 times the amount of material ground in the laboratory tests.

This length of wheel life would represent three weeks of single shift production at 100% efficiency.

Resulting Implications

Such very long wheel life has the following important implications:

a. excellent tooth spacing can be achieved as a result of the small wheel wear per gear (in the micro inch per gear range).[18]

b. excellent consistency can be achieved from gear to gear despite variations in stock left for finishing.

c. the possibility of unmanned manufacturing with automatic loading and automatic periodic coordinate part inspection.[30]

d. a dresser is not required on the grinding machine, mak-

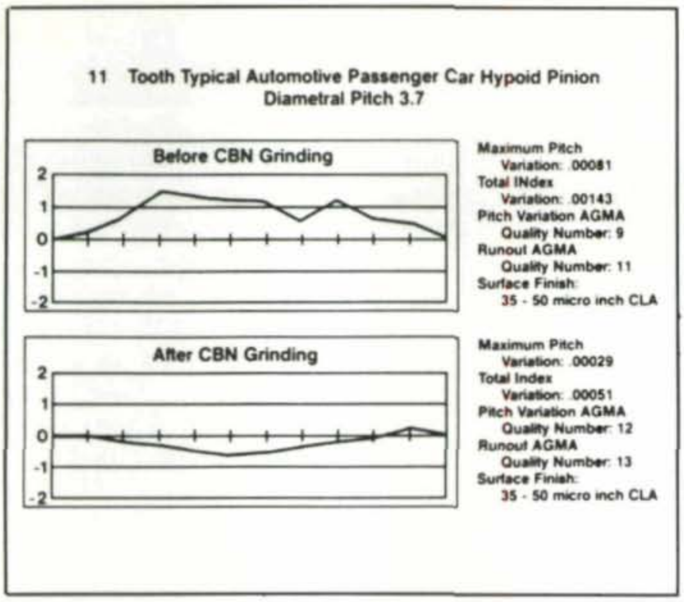


Fig. 15—Index Variation Measurements Before and After CBN Grinding.

ing the machine less complex and allowing greater machine utilization.

Results of CBN Gear Grinding

Tooth Spacing Quality

Fig. 15 shows that an automotive pinion with an AGMA quality number 9 was improved to quality number 12 by single pass CBN grinding. This shows the corrective nature of this process.

Fatigue Life

Bending fatigue life improvements of 17 times have been achieved as compared to cut hardened and lapped gear sets, the most common method used in the manufacture of land application bevel gears.[18] Current surface durability tests have shown a life improvement of 2 to 5 times for CBN ground gear sets. The benefits resulting from CBN grinding are achieved by the removal of harmful effects, such as dimensional inaccuracies of tooth position and geometry, excessive surface roughness and unwanted surface microstructural features.

Cost

Many times it has been thought that the use of CBN grinding wheels necessitated high tool cost. This has not been found to be the case with CBN finish grinding of bevel gears. Even when the production cycle time for CBN finish grinding was faster than soft finishing, the CBN wheel cost per gear was essentially the same as the cutter cost for soft finishing. By changing the gear manufacturing methods to take advantage of CBN hard finishing, one manufacturer found that the production cost per gear set was reduced by thirty percent.

Summary

This paper has attempted not only to show the possible benefits which may be derived from grinding with CBN, but also to explain the physical reasons for the observed phenomena. This knowledge permits taking advantage of the characteristics of CBN to simultaneously improve productivity and workpiece quality.

The combination of this new grinding tool material (CBN), the applied grinding technology, and new bevel gear grinding machines designed for the necessary increased requirements have now made this new process both technically and economically practical for the mass production of bevel gears.

This process appears to have great potential for becoming the dominant finishing method in the near future because it combines the benefits of better quality control and lower cost.

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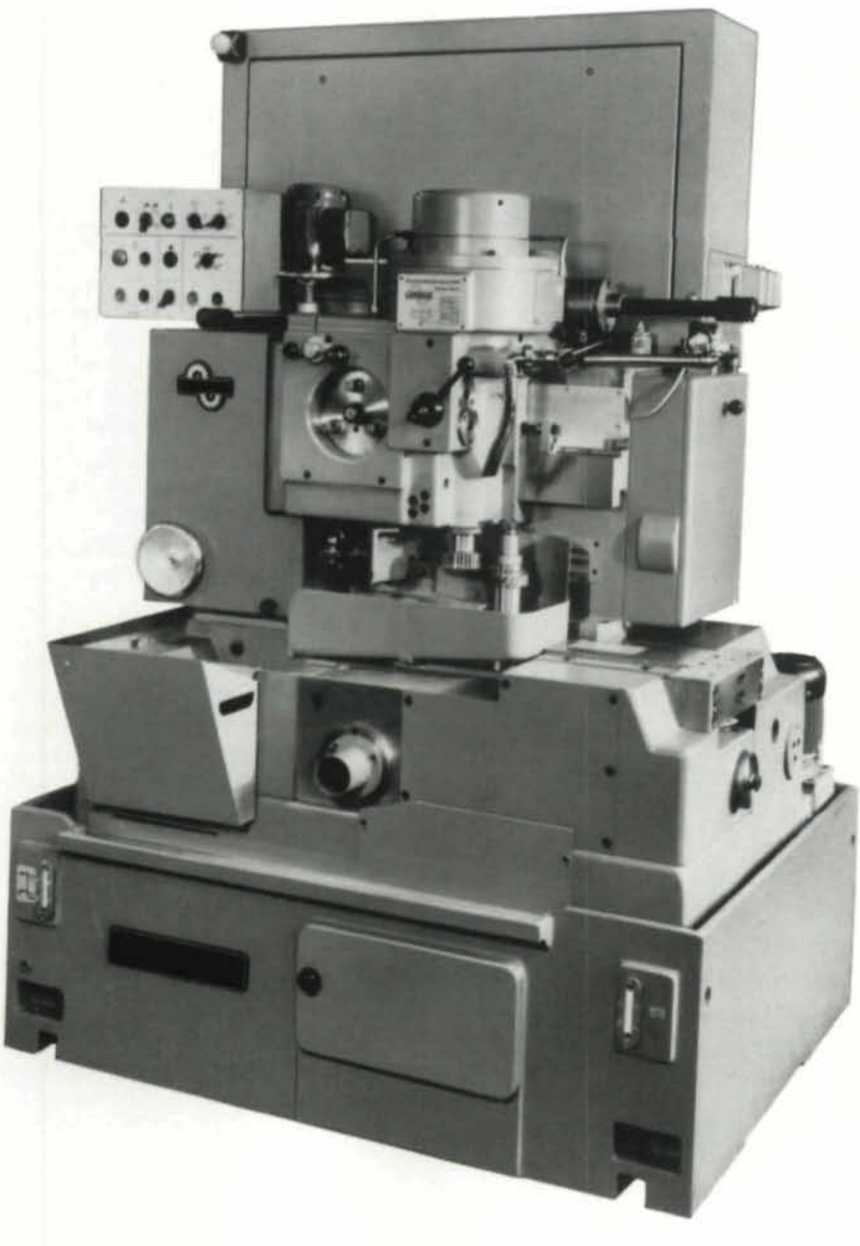
BACK TO BASICS...

Gear Shaping Machines CNC Development

by
John M. Lange
Miller Associates, Inc.



Up until approximately 1968-69, pinion cutter-type gear shaping machines had changed very little since their conception in the early 1900's. They were bridge-type cutter head machines, with a table relieving system to clear the cutter from the workpiece on the return, nonproductive stroke of the cutter spindle, see Fig. 1. The "modern shapers," introduced in 1968-69, went to a cutter spindle relieving action compared to the table relieving movement on the older style machines. Furthermore, the cutter spindle (and its moving housing) were mounted into a robust column, see Fig. 2.

Modern machines are at least two times heavier than old style machines of equal diameter. They are also two to three times more productive than the old style machines. This increase in productivity is directly attributed to the following:

- rigidity in the machine because of cutter spindle relief stroking drive train. This is a much smaller and constant mass to move, as compared to the larger mass of the table on the old style machine. That mass also varied depending on the size and weight of the gear being cut and the fixture.
- stroking rates in the range of 1,000 to 2,000 strokes per minute made possible by a cutter spindle relieving mechanism and hydrostatically mounted cutter spindle bearing and guides, Fig. 3.

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- larger cutter spindle diameters with proportionally increased horsepower of the main drive motor. Example: 20" maximum diameter capacity modern machine, with a 3.93" diameter cutter spindle and 20 horsepower, stroke drive motor; old style machine, cutter spindle diameter 3.34" and 5.7 horsepower motor driving the entire machine, i.e. cutter spindle stroking, rotary and radial feed change gears, see Fig. 4. Note: Maximum DP rating on this size machine went from 5 DP for the old style machine to 3 DP for the modern machine.
- overall weight of the machine increased by a factor of two to three times. Example: 6" maximum diameter capacity modern machine, 12,500 lbs.; old style machine, 4,900 lbs. This extra weight helps to absorb the higher cutting forces and reduces vibration.

While the first generation modern gear shaping machines were substantially more productive, they were still limited in flexibility as were the old style machines. For example:

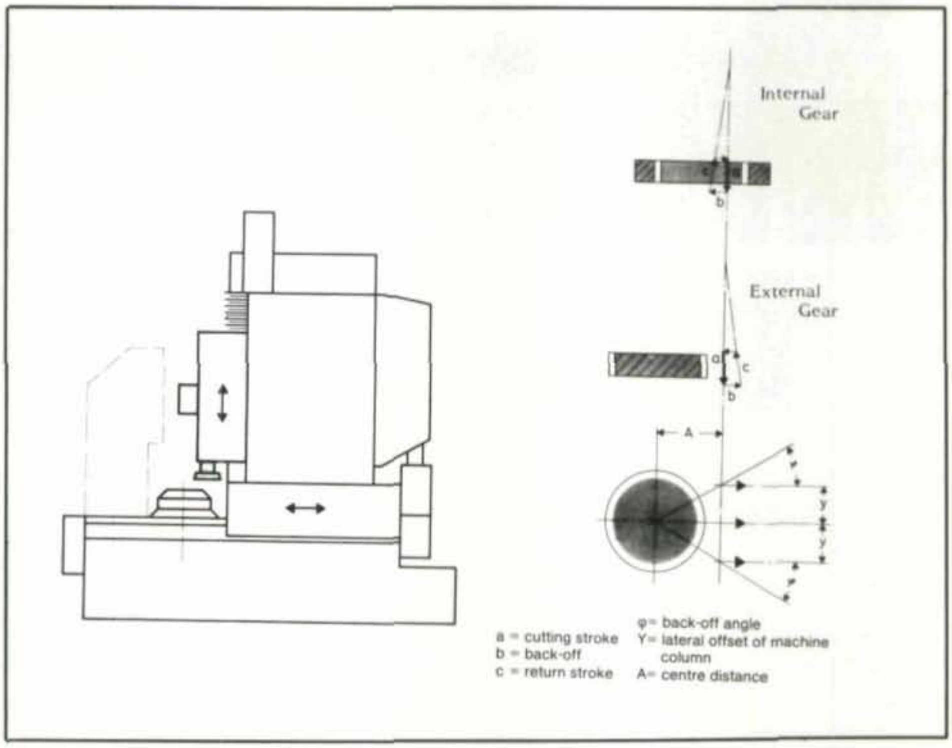

Fig. 2—"Modern" Gear Shapers — Cutter Spindle Back-Off System

1. A gear shaping machine with a two inch stroke has a very limited vertical height position (distance above the work table) in which that two inch stroke can occur, i.e. normally only three inches or less. This deficiency results in the need to supply the machine with a riser block (spacer mounted between the bed of the machine and column) to elevate the maximum stroke height to the same level as the tallest part to be cut. Consequently, shorter parts must be raised up in the special fixtures to this predetermined height. Obviously, riser blocks and built-up fixtures reduce the desired rigidity of the machine, and in turn, accuracy of the cut part and tool life. The cost for fixturing elements increases proportionately.
2. Quite frequently, shapers are used for cutting one gear in a cluster of gears, because one or two elements in the cluster must be shaped, i.e. cutter runout clearance is restricted. In addition to the shaped gear in the cluster, it would have been advantageous to shape another cluster in the same setup. However, because of

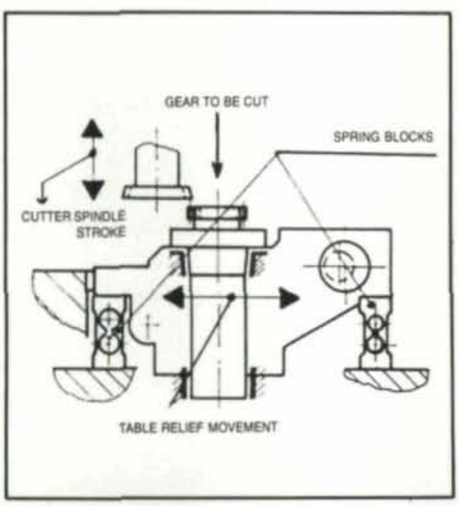

Fig. 1—"Old Style" Gear Shapers — Work Table Back-Off System.

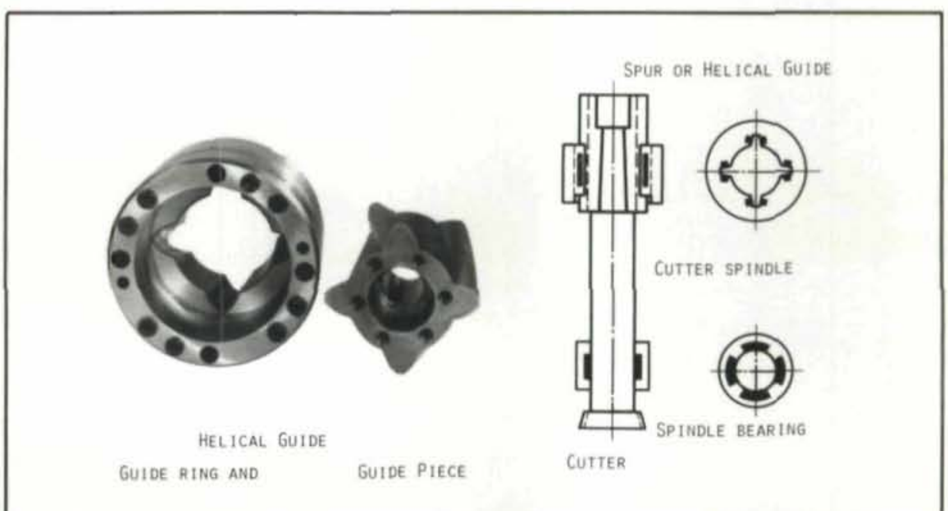

Fig. 3—Hydrostatic lubrication of cutter spindle guide and lower bearing

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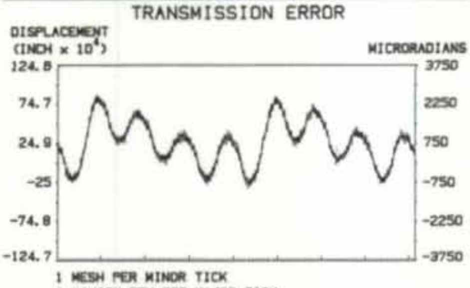
When it comes to pinpointing the causes of gear vibration, noise, and tooth damage, it's no secret that single flank measurement gives you the kind of comprehensive data you simply can't get with other forms of measurement. Reliable information about



transmission error is what you need to predict noise levels of a gear set in operation, as well as to reduce process errors and the costly scrap and rework that go with it.

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	TOLERANCE	MEAS. ERROR	% OF TOLER
TOTAL TRANSMISSION ERROR	.0125	.0024	*
MAX TOOTH-TOOTH TRANS. ERROR	.003	.0004	*
AVG TOOTH-TOOTH TRANS. ERROR	.003	.0002	*
NUMBER OF BURRS OVER TOL	.005	0	*
MAX EFFECTIVE PROFILE VAR	.0015	.0001	*
AVG EFFECTIVE PROFILE VAR	.0005	.0001	**
COMB. ACCUMULATED PITCH VAR	.0015	.0029	REJECT
COMB. MAXIMUM PITCH VAR	.0034	.0005	*
COMB. MAXIMUM SPACING VAR	.0025	.0002	*

(RESULTS IN in)

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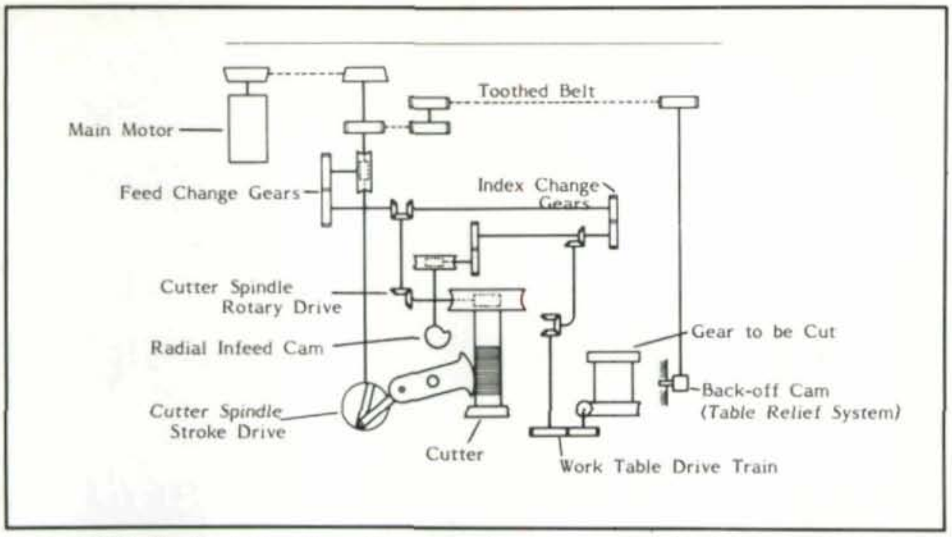


Fig. 4 – "Old Style" table relief machines with numerous drive trains necessitated by having a single drive motor.

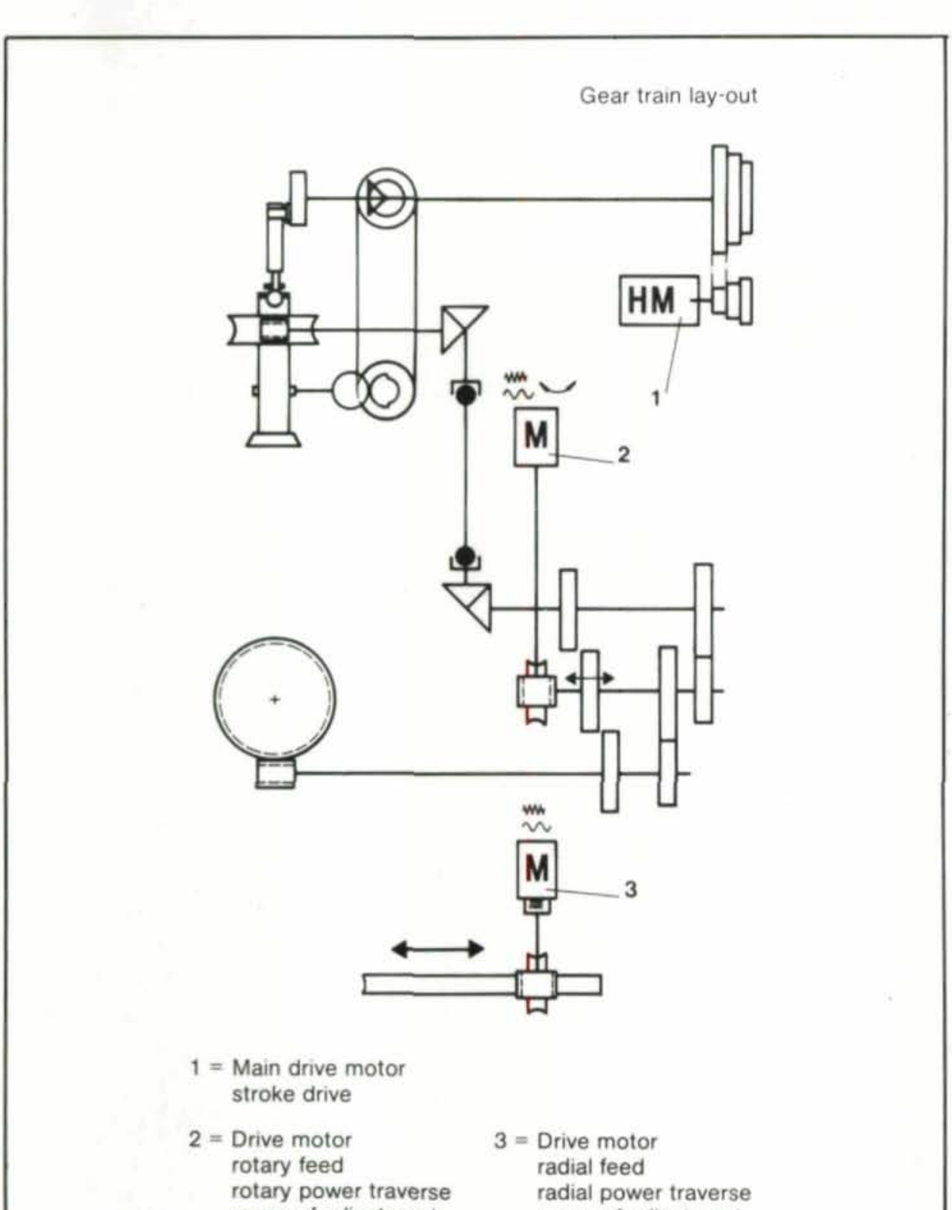


Fig. 5 – Second generation modern gear shaper with independent drive trains.

the different number of teeth, the required index ratios, and because of the fixed index change drive trains, this was not possible with the older style and first generation spindle relief machines. Also, the location of the second gear on the shaft made it difficult to reach, even with stacked cutter, i.e. two cutters mounted to the cutter spindle.

3. In the 1960's, the cutting tool was not the limitng factor in the cutting process. Light weight machines with numerous gear drive trains, see Fig.4, slow stroking rates and general lack of rigidity, made the machine the limiting factor; not the tool. The modern first generation spindle relief shapers in the early 1970's, using conventional M-2 tool steels, found the tool the limiting factor; not the machine. In the late 1970's and early 1980's, with the advent of powdered metal ASP 30 and 60, titanium-nitrided coated cutters, we found, in many cases, the machine to be the limiting factor once again. New infeed techniques had to be developed to realize the full potential of these new tools.

The second generation of the spindle relief machines added independent wide speed range, rotary and radial feed DC servo drives, Fig. 5. The third generation machines incorporated an automatic cutter clamping feature and a CNC controller which eliminated index change gear drive trains. These machine advances dealt with the previously mentioned limitations throught the following design features, see Fig. 6.

A. The machine is fitted with a vertical adjustable cutter head slide (Z axis) which allows vertical positioning of the entire head. In most cases, riser blocks and specially elevated work fixture are not necessary. For example, this 2.35" stroke length machine has an axial displacement of the cutter head slide (stroking position) of 5.9". Using NC techniques, the positioning of the cutter head slide is accurate to within a tolerance of .0008".

B. The C and D axes, which required rotary movement (index ratio) of the cutter spindle and rotary movement of the work table respectively, are controlled by CNC technique with rotary encoders. There are no index change gears in the machine. The resolution of the rotary encoders is 3.6 arc seconds. This design feature makes it possible to cut two or more gears in a cluster, having different index ratios, in a single setup. Depending on the gear data of the cluster gear, it might be necessary to use stack mounted cutters. The lead of both cutters must be the same. A CNC guide has not yet been developed, but experimental work is being done in this area. Fig. 7 shows two external gear clusters being shaped in a single setup. CNC shapers are also perfectly capable of cutting components having both internal gears (or splines) and external gears in a single setup.

C. Frequently, cluster gears have a timing requirement between a tooth on a gear in the cluster in relation to another tooth on a second gear in the cluster. The use of a CNC control system makes it possible to meet these demanding requirements. Fig. 8 illustrates such an alignment requirement. This automotive transmission component requires a tooth alignment accuracy between the two gears of .0008". The part has been cut on a CNC shaper, as illustrated in Fig. 1, achieving an alignment accuracy of .0004". This accuracy will be maintained in a production environment.

D. Down and up shaping of a component is made possible with CNC. The part configuration illustrated by Fig. 9 dictates that both the upper and the lower gear be shaped. To do this part in a single setup, down and up shaping is required. The center section of the component has an outer diameter larger than the root diameter of the two gears. There is also an alignment requirement of the teeth of the upper gear to the lower gear. That alignment can be easily obtained, because the part is cut in a single setup using keyed cutters. The relation of the cutter spindle backoff and cutting stroke direction is controlled by the CNC unit. When cutting the upper gear, the cutter relief occurs on the upward, nonproductive stroke. In the case of the

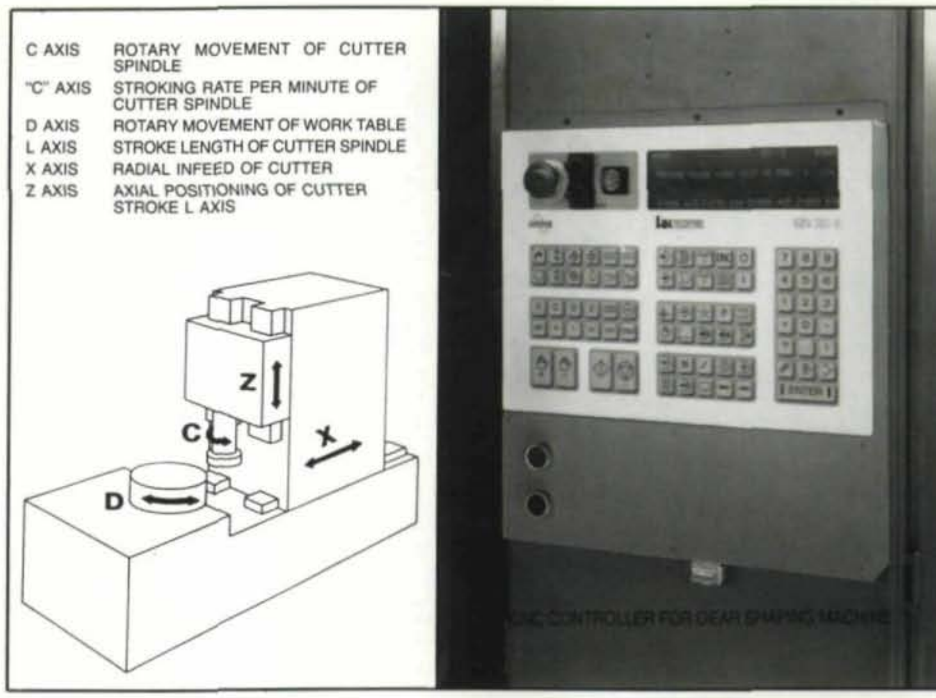


Fig. 6 – 6 Axes CNC gear shaping machine.

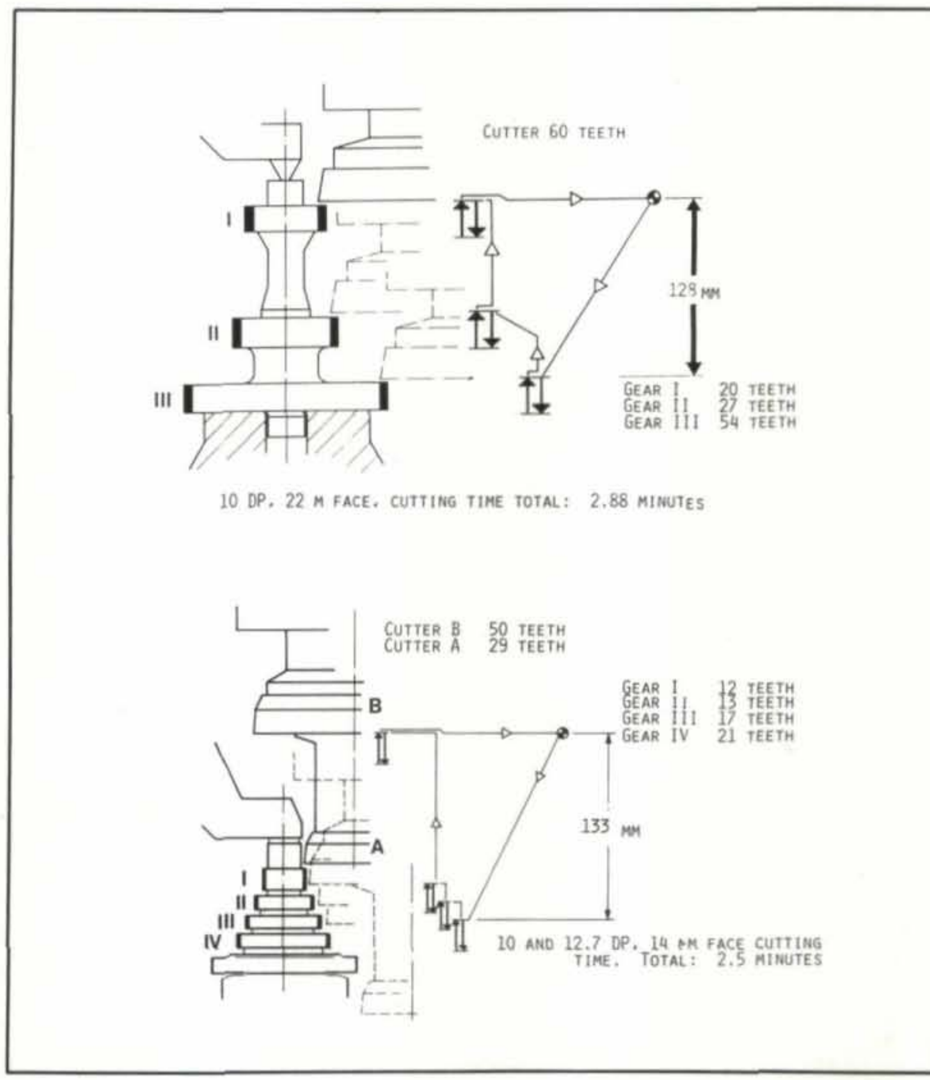


Fig. 7 – Spur cluster gear cutting in a single set-up.

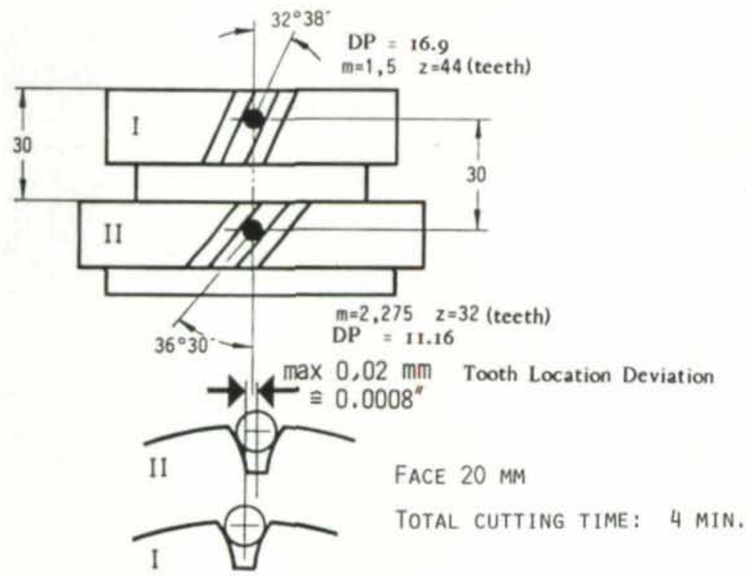
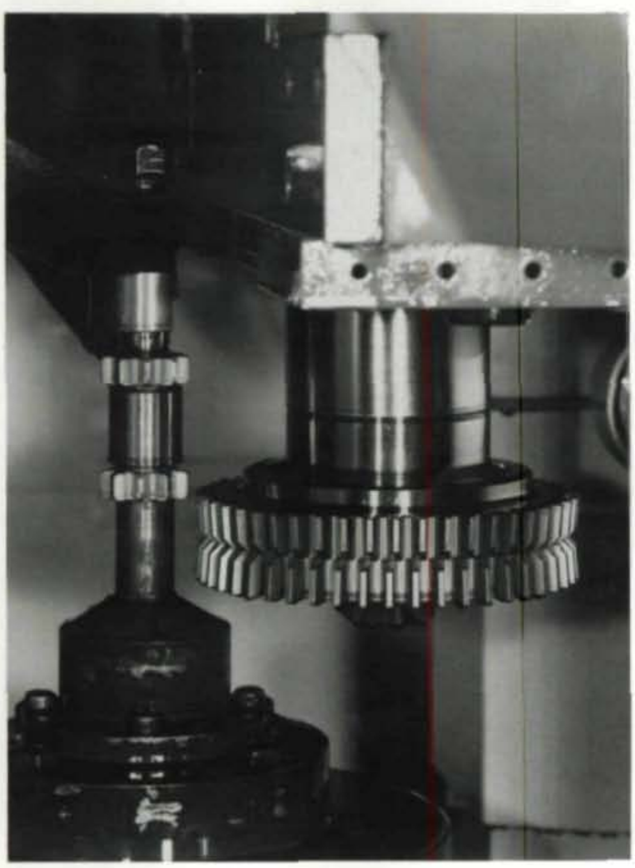


Fig. 8—Helical cluster gear cutting with a tooth location requirement.



8/10 DP, 1.2" pitch dia., .4" face, 9 teeth cutters
38 teeth, 6" dia. — total floor-to-floor cutting time
1.05 minutes

Fig. 9—Down and up shaping of a cluster gear in a single set-up.

lower gear, the cutter relief action occurs in the downward stroke.

E. A total CNC gear shaping machine has fully independent, short and rigid drive trains, i.e. radial feed, rotary feed and cutter spindle stroking (for kinematic drawing, see Fig. 10.) Independent drive trains with wide speed range DC servo motors, i.e. 1-3,000 RPM, permit a new cutting technique called CCP (Controlled Cutting Process). The conventional cutting technique produces a chip with a considerable difference in thickness from the leading to trailing flanks of the cutter, see Fig. 11. This type of chip formation leads to a "burning back" of the leading flanks, especially the leading flank tooth tip of the cutter. That cutter edge deterioration occurs long before the trailing flanks and tips show signs of wear. The CCP cutting process produces a chip with a uniform thickness and wear land from the leading to the trailing flanks of the cutter. The CCP technique uses extremely high rotary feeds and matching radial feeds to suit the workpiece and cutter geometry. This more uniform chip thickness formation increases cutter life by as much as 100% and reduces cutting time in the range of 50% to 100%. In addition, better gear quality is achieved with a superior surface finish. In quite a few cases, subsequent finishing operations such as shaving, can be eliminated. (See Table I)

F. The CNC controller allows for a programmable, constantly reducing radial infeed. Higher radial infeeds can be tolerated when the cutter first enters the workplace, i.e. the first cut. Chip loading is slight and increases progressively as the cutter works its way into the depth. As the chip load increases, the radial infeed is progressively reduced. The factors considered when calculating the reduction in radial infeed are diametral pitch, pressure angle, diameter of the workpiece, diameter of the cutter and whether the part is an internal or external gear.

G. Machine down time, whether it be in a transferline situation, a flexible

TABLE I

Two production examples of gear shaping to preshave conditions:

Workpiece			**Machine Setting**	**Conventional**		**CCP**	
				▽	▽▽	▽	▽▽
No. of Teeth:		46					
Normal module	4.2 DP	6 mm	Strokes per Minute ______	180	2 × 450	180	500
Helix Angle:		0°	Cutting Speed ______ m/min	33	2 × 82	33	91
Face Width:	1.96"	50 mm	Rotary Feed ___ mm/stroke	0.75	0.75/0.5	4.710	3.770
Material:		CK 45	Radial Feed ___ mm/stroke	0.05	0.05/0.01	0.006	0.005
Tensile strength:		190–210 Brinell	**Cutter life**	**approx. 15 parts**		**approx. 36 parts**	

Cutter life increased by more than 100 %

Workpiece			**Machine Setting**	**Conventional**		**CCP**	
				▽	▽▽	▽	▽▽
No. of Teeth:		46					
Normal module	4.2 DP	6 mm	Strokes per Minute ______	180	2 × 450	180	500
Helix Angle:		0°	Cutting Speed ______ m/min	33	2 × 82	33	91
Face Width:	1.96"	50 mm	Rotary Feed ___ mm/stroke	0.5	0.5/0.5	4.710	3.770
Material:		CK 45	Radial Feed ___ mm/stroke	0.03	0.03/0.005	0.006	0.005
Tensile strength:		190–210 Brinell	**Cutting Time ____ minutes**	**20.0**		**14.2**	

Approx. 30 % increase in output.

machining cell or a job shop, must be avoided. Semiautomatic and fully automatic cutter change techniques have been developed to help reduce idle items. Fig. 12A and 12B show how a shaper cutter, mounted to an adapter, is automatically clamped concentric to the cutter spindle. Concentricity is held to .0002" or better. This reduces cutter mounting radial runout error and the part's accumulated pitch errors as the result of that cutter runout. Cutter change, because of a new setup or tool wear, can be accomplished in a matter of seconds, Fig. 13A and 13B. The CNC controls are programmed prior to the cutter change, so that down time is kept to a minimum. An axial moving cutter head slide or cutter spindle (Z axis) is needed to automatically compensate for the new stroke height position. The infeed control (X axis) is needed to automatically compensate for cutter diameter change and the necessary infeed depth change. Cutting tools must be accurately measured after sharpening to obtain new X and Z axes tool off-set positions. A special measuring instrument is used to make these measurements. Such a measurement of the tool and electronic positioning of the X axis (infeed) slide to an accuracy of +/− .000040", assures the "holding of size," i.e. pin dimensions. It is not necessary to verify size by making an overpin dimension check after each tool change.

The addition of a CNC gauging unit at a station of an automatic part loading and unloading system enables the CNC shaper to function fully independent of operator attention. This in-process gauging and compensation unit carries out the following measurements and dictates required machine setting changes:

- measures gear runout error
- average center distance error
- a trend calculation – SPC (statistical process control for, let's say, 10 parts) is established from the measured values of the average center distance. A new X axis radial position is calculated from the analyzed data, and the CNC shaper control directs and sets the new X axis position. If tolerances are still exceeded, then a tool change can be

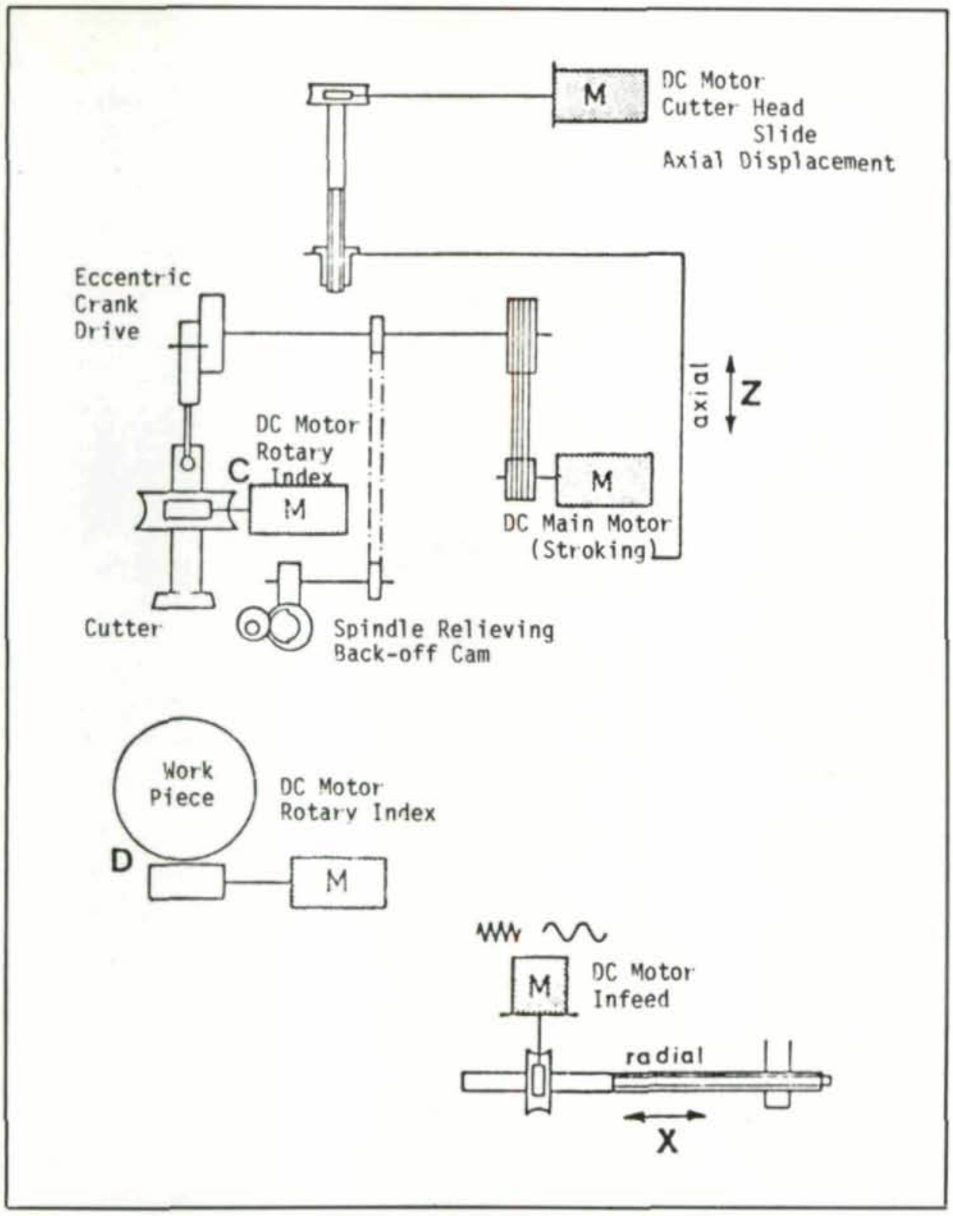


Fig. 10 – Kinematic drawing of gear drive train of a CNC gear shaper.

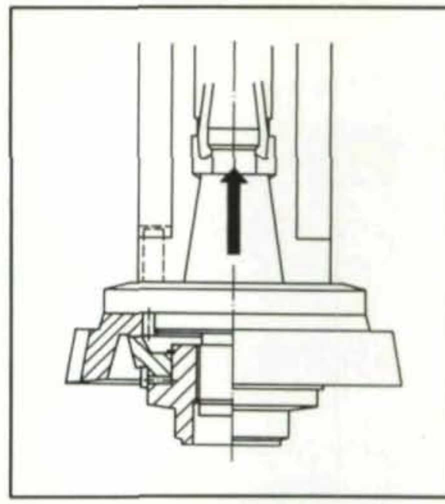


Fig. 12A – Automatic clamping of cutter and adapter to the cutter spindle.

Fig. 12B – Cutter adapter with ball cage for concentric location of cutter to cutter spindle.

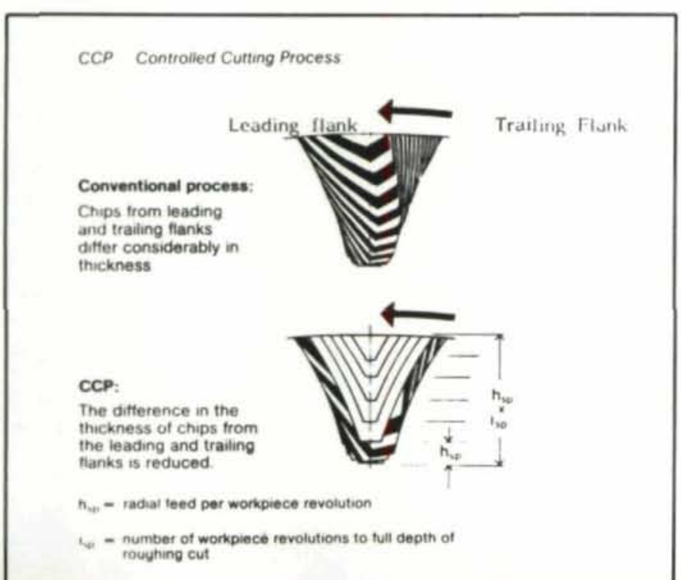


Fig. 11 – New Infeed Technique – CCP Controlled Cutting Process.

called for. If, after the tool change, the prescribed tolerance is still exceeded, the CNC controller can send out a fault signal requesting operator/setup man assistance.

Measurements are made by means of a ball-type probe plunged into a tooth gap. Initial calibration of the measuring head is done with a calibration master gear. Measuring time per tooth is about one second.

It is fashionable in manufacturing discussions today to use the abbreviation FMC (flexible manufacturing cells) and FMS (flexible machine systems). Those abbreviations are also commonly used with such comments as just-in-time manufacturing (inventory), zero setup time and autoloading of parts, fixtures

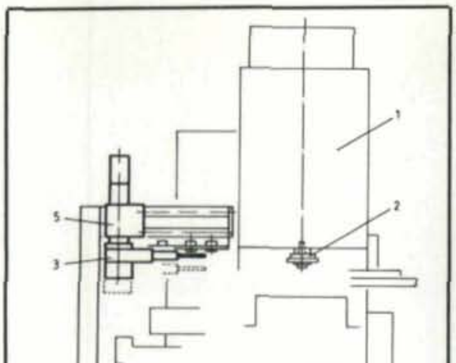
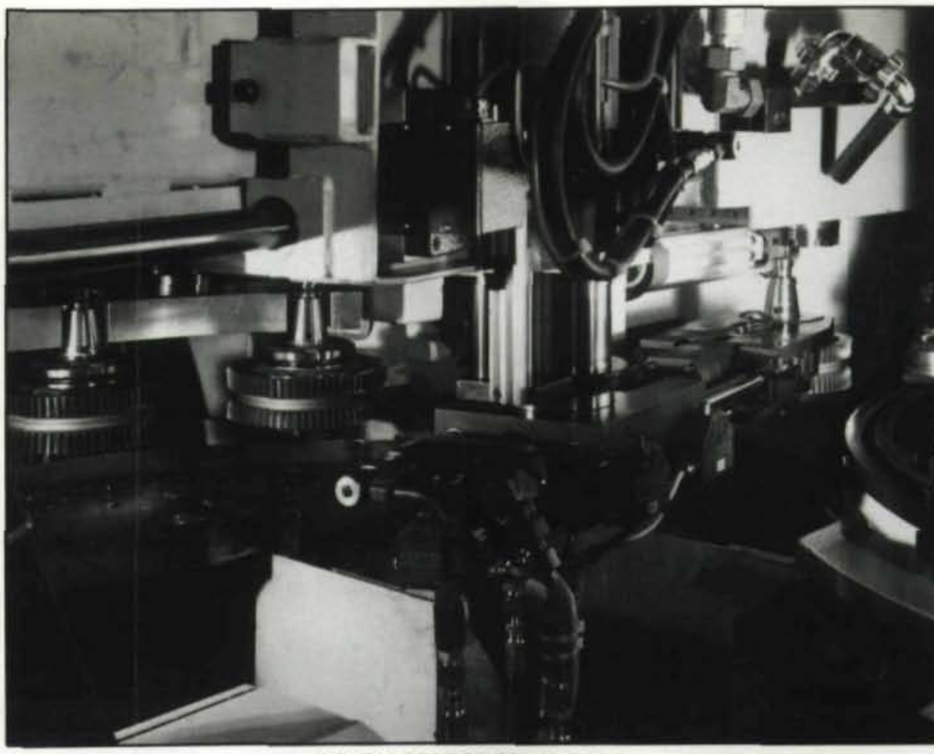


Fig. 13A & B – Automatic cutter changing unit with storage for 3 cutters.

and tools. The conventional first and second generation modern gear shaping machines are, most likely, not suitable for an FMS or FMC system. Many manufacturers say the state-of-the-art gear shaping machine is not ready for installation in an FMS or FMC manufacturing technique. Most certainly, the third generation spindle relief CNC gear shapers can, indeed, fulfill all design parameters needed for installation in an FMS and FMC applications, namely because of their special features, i.e.:

- zero setup time – automatic machine setup
- tool offset compensation
- fully automatic fixture change
- fully automatic tool change
- gear cutting flexibility – multiple gear cuttings per setup, i.e. cluster gears
- integration of a CNC post process gauging unit to the CNC control of the shaper



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(continued from page 37)

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SUBCONTRACT WORK

(continued from page 29)

Table II

Tooth Combinations to Provide a Gear Ratio Of 5.17220 ± 0.0001 (18 ⩽ N ⩽ 200)

Ratio Number	Program Output Ratio	Tooth Combinations	Error (x 10^6)
1	$\frac{(11)(71)}{151}$	$\frac{71}{18}\ \frac{198}{151}$	15
2	$\frac{(7)(41)(47)}{(2^4)(163)}$	$\frac{47}{18}\ \frac{63}{24}\ \frac{123}{163}$	37
3	$\frac{29(173)}{2(5)97}$	$\frac{58}{20}\ \frac{173}{97}$	35
4	$\frac{5(7)103}{17(41)}$	$\frac{70}{34}\ \frac{103}{41}$	34
5	$\frac{3(17)43}{(2^3)(53)}$	$\frac{43}{18}\ \frac{51}{24}\ \frac{54}{53}$	30
6	$\frac{(2^2)(3^2)5(131)}{47(97)}$	$\frac{180}{97}\ \frac{131}{47}$	13
7	$\frac{(2^7)(3)37}{41(67)}$	$\frac{96}{41}\ \frac{148}{67}$	12
8	$\frac{3(4)7(113)}{2(31)37}$	$\frac{105}{37}\ \frac{113}{62}$	12
9	$\frac{(2^3)(5)199}{(3^4)(19)}$	$\frac{40}{19}\ \frac{199}{81}$	10
10	$\frac{41(137)}{2(3)181}$	$\frac{123}{18}\ \frac{137}{181}$	9
11	$\frac{(2^2)(3)13(31)}{5(11)17}$	$\frac{48}{20}\ \frac{26}{22}\ \frac{62}{34}$	8
12	$\frac{2(17)197}{5(7)37}$	$\frac{36}{18}\ \frac{34}{20}\ \frac{197}{37}$	0
13	$\frac{61(97)}{(2^3)(11)(13)}$	$\frac{61}{26}\ \frac{97}{44}$	3
14	$\frac{6(17)53}{16(37)}$	$\frac{106}{26}\ \frac{85}{67}$	16
15	$\frac{2(41)89}{17(83)}$	$\frac{82}{34}\ \frac{178}{83}$	18
16	$\frac{(7^2)(19)}{(2^2)(3^2)5}$	$\frac{76}{36}\ \frac{49}{20}$	21
17	$\frac{(2^2)(5)11(43)}{31(59)}$	$\frac{55}{31}\ \frac{172}{59}$	25
18	$\frac{2(29)131}{13(113)}$	$\frac{116}{26}\ \frac{131}{113}$	26
19	$\frac{2(13)149}{7(107)}$	$\frac{36}{18}\ \frac{39}{21}\ \frac{149}{107}$	30
20	$\frac{(2^6)(3^2)11}{(5^2)(7^2)}$	$\frac{64}{25}\ \frac{99}{49}$	44
21	$\frac{23(47)}{11(19)}$	$\frac{46}{19}\ \frac{47}{22}$	49

than 200. This P/Q would then have to be treated by the same method.

Twenty-one tooth ratios are left after duplicate gears and those with more than 200 teeth are eliminated. These are displayed in the second column of Table I in the form returned to the main program by FACT. Column three is the rewritten form of these tooth ratios after multiplication factors have been applied, so that no gear has fewer than 18 teeth and after the P/Q factors have been rearranged into ascending or descending order. According to the form described by (6), it follows that gear ratio 1 in Table I represents a shorter gear train. It has only three shafts. Gear 1 has 45 teeth and is on shaft 1, the input shaft. It is in mesh with gear 2, with 24 teeth, mounted on shaft 2. Shaft 2 also holds gear 3, which has 33 teeth and is in mesh with gear 4. Gear 4 has 21 teeth and is mounted on shaft 3, the output shaft.

Example 2

We will repeat the selection process used in Example 1 for the ratio 5.17220 ± 0.0001. E will be divided into 100 and 113 intervals and the same criteria for the acceptable number of teeth on each gear will be used.

The results for 100 subintervals are displayed in Table II. Division into 113 subintervals provided only two more ratios, as shown in Table III. As these two examples demonstrate, the number of ratios obtained depends on both the input ratio and on the number of subdivisions of the permissible error E.

Note that the maximum error in all three tables is 49×10^{-6}.

Table III

Additional Tooth Combinations Found for R = 5.17220 Using N = 113 (18 ⩽ N ⩽ 200)

Ratio Number	Program Output Ratio	Tooth Combinations	Error (x 10^6)
1	$\frac{(2^5)5(61)}{3(17)37}$	$\frac{64}{34}\ \frac{30}{18}\ \frac{61}{37}$	31
2	$\frac{19(2^3)\ 67}{11(179)}$	$\frac{76}{22}\ \frac{36}{18}\ \frac{134}{179}$	31

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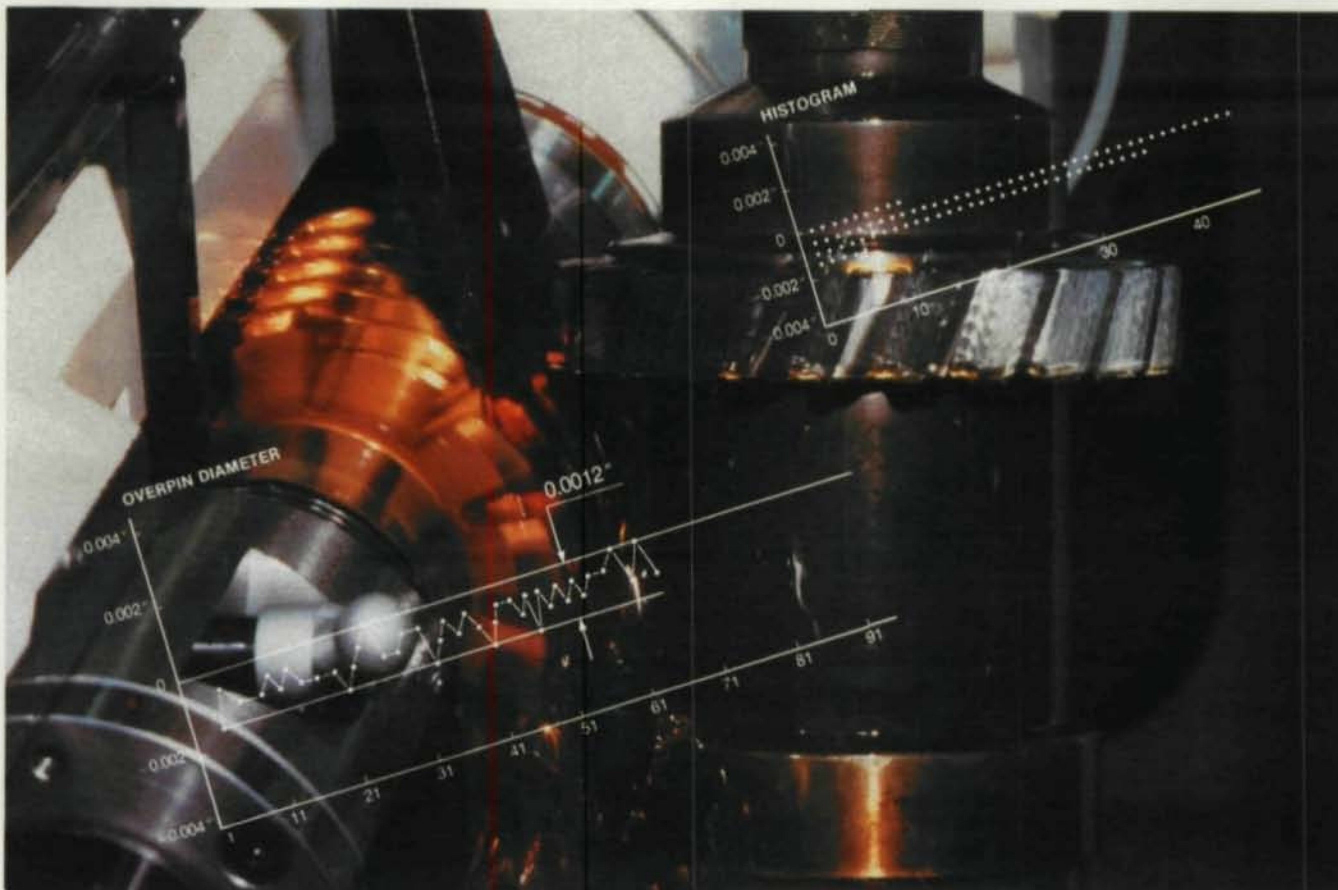

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