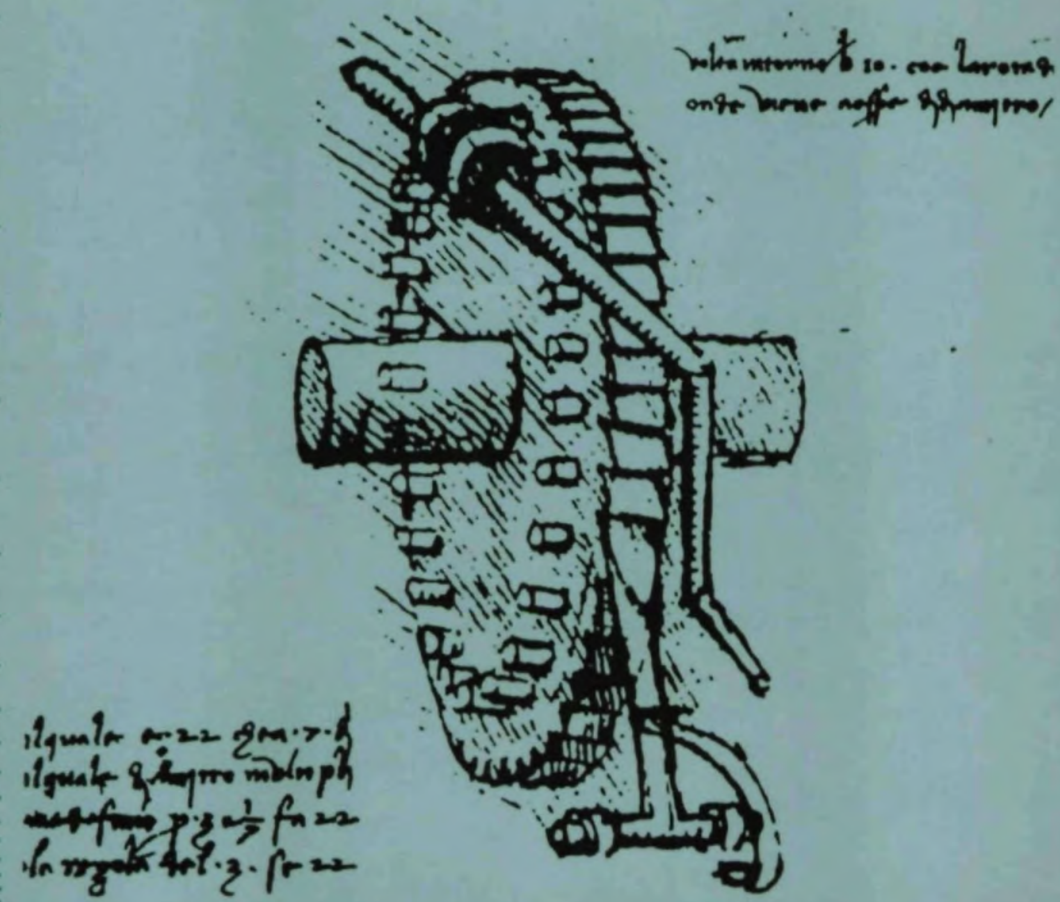
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

MARCH/APRIL 1986



A Computer Solution for the Dynamic-Load, Lubricant Film Thickness, and Surface Temperatures in Spiral-Bevel Gears

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COVER

The cover sketch is one of many of *Leonardo's sketches of gears. In this* version, he illustrates a typical type of worm gear. He noted the following problem, "When you make a screw that engages only a single tooth on the wheel, it will be necessary to add a pawl in order to avoid the reversal of the wheel's motion should that tooth break. Such a screw could cause great damage and destruction."

GEAR TECHNOLOGY
The Journal of Gear Manufacturing

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DEPARTMENTS

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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.

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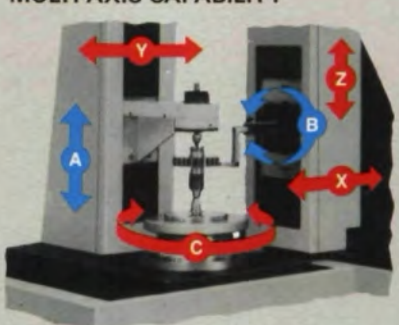

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NOTES FROM THE EDITOR'S DESK

Sitting down to write my comments for this issue, one event filled my thoughts—the transformation and uninhibited euphoria that overcame Chicago, and the whole Midwest, by the Bears reaching and winning the Superbowl.

Chicago, like many cities in the United States, has been battered for the past few years by layoffs, consolidations, plant closings and the resulting unemployment. Life has been hard and times have been rough. Normally at this time of the year, we settle down for the remainder of the cold, dark winter, and utter the motto of all Chicago's sports team fans: "Wait'l next year." But this year, things were different. At last, this was the "next year" we were waiting for—the year of the Superbowl, and the Superbowl Shuffle. Chicago was ecstatic. Commodity traders, usually under strict dress code, dressed in outrageous Bear costumes. Sir George Solti and the Chicago Symphony Orchestra broke out in "Bear down, Chicago Bears" during one of their concerts and all over the city there was evidence of Bear-mania.

I was a very young man at the championship game in Chicago in 1963, and after a 23 year wait, it was thrilling to be going to my first Superbowl game. The bus trip in New Orleans started the excitement. As we approached the Superdome, we were surrounded by people crying out for extra tickets. The crowds were thick and hawkers were everywhere in a sea of orange and blue. Frankly, it was hard to tell that another team was there. It appeared that this was The Bears' Superbowl. The anticipation and tension in the crowd was enormous, and Superdome itself was an absolute wonder. Everyone seemed to stay at that high pitched state until early in the third quarter. By then, the fans had begun to suspect that the game itself might be anti-climatic to an unbelievable season. Still, triumph was sweet as evidenced by 500,000 Bear fans who lined the streets the next day in a wind chill of 40° below, to welcome their heros home with a tickertape parade.

Few sports teams reflect the character of a city, if not a nation the way the Bears do. They are hard working and tough, looking within themselves for the strength to persevere and prevail, but knowing that having fun has its place. They have a deep faith that if you work hard and have a sharp vision of your goal, you will ultimately succeed—and succeed they did!

Our industry has also been saying "Wait'l next year" for around five years. Maybe, just maybe, 1986 . . .

Michael Goldstein
Editor/Publisher

VIEWPOINT

Dear Editor:

I received a letter from Mr. G. W. Richmond, Sullivan Machinery Company, N.H., in which in addition to correcting mistyping, he made several suggestions concerning my article "General Equations for Gear Cutting Tool Calculations". (Gear Technology, Nov/Dec 1985)

I have found his recommendations commendable and would like to share them with the readers interested in the article:

1. The term "$-r_w/i$" in the second equation of (2), page 27, should not have been used. It is just enough to subtract the generating radius r_w *after* differentiation to bring the ordinate Y_c for the hob profile to the pitch line.
2. The expression for r_A, page 23, can be also received directly from Fig. 2.
3. The "$-$" sign before Y_c on Fig. 2 might be confusing. It is better to disregard it. (as the "$-$" sign before 0 in the second equation on page 22).

Please express my personal gratitude to Mr. G. W. Richmond for these suggestions.

Ilya Bass
Bourn & Koch
Machine Tool Co.

Editors Note: We regret that some typographical errors appeared in the Bass' article. Formula corrections should appear as follows.

The first formula on page 21 is:

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

Formula (2) is:

$$y_c = \frac{r_w}{i} \times \cos(\psi_P \times i) + \frac{r_A \times \sin\sigma_P}{\cos\alpha} \times \sin(\alpha + \psi_P \times i) - \frac{r_w}{i} \qquad (2)$$

The 2nd line of formula (3) is:

$$y_c = r_A \times \cos\sigma_P - r_w$$

The last line of the last formula on page 23 is:

$$\varphi_A = \frac{180^\circ}{30} - (\mu_A - \beta) = 3.860106^\circ$$

(continued on page 23)

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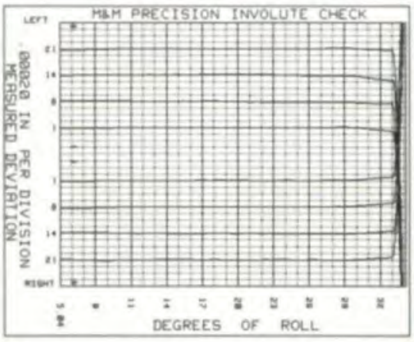

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A Computer Solution for the Dynamic Load, Lubricant Film Thickness, and Surface Temperatures in Spiral-Bevel Gears

H. C. Chao
Solar Turbines, Inc.
M. Baxter
Consultant Gear Technology
H. S. Cheng
Northwestern University

Spiral-bevel gears, found in many machine tools, automobile rear-axle drives, and helicopter transmissions, are important elements for transmitting power. However, the basic mechanisms which govern the major failure modes of spiral gears are still not fully understood. Because of the complicated geometry of spiral-bevel gears, the analyses are considerably more difficult than those conducted earlier for spur and helical gears. In military applications, such as the transmissions used in V/STOL aircraft, gears are often designed under conditions very close to the failure limits for maximum power density. A thorough understanding of spiral-bevel gears under critical operations is urgently needed to prevent premature failure.

Gear failures usually fall into two categories, structural failures, which include flexure fatigue, tooth breakage, case crushing, and lubrication-related failures, which include wear, surface pitting, and scuffing. Among these types of failure modes, lubrication-related failures are much more difficult to predict since the basic mechanisms are still not fully understood. Current methods used for predicting gear pitting and scuffing are mainly empirical and are not completely reliable. Recent failure tests of gears and rollers strongly suggest that surface pitting as well as scuffing are critically influenced by lubricant film thickness and surface temperature in the gear teeth contact. To develop improved methods for failure prediciton, it is important to develop accurate tools to determine the film thickness and surface temperature.

In this article a computer method is first described for determining the dynamic load between spiral-bevel pinion and gear teeth contact along the path of contact. The dynamic load analysis is necessary because it governs both the surface temperature and film thickness. Computer methods for determining the surface temperature and film thickness are then presented along with some results obtained for a pair of typical spiral-bevel gears.

Symbols

$\{D\}$	displacement column vector for pinion or gear in x, y, z coordinates, m or rad (ft or rad)
$\{D'\}$	displacement column vector for pinion or gear in x', y', z' *coordinates, m or rad (ft or rad)*
$[DC_{ji}]$	elastic compliance matrix, m/N (ft/lb)
$[DK_j]$	bearing stiffness matrix, N/m (lb/ft)
$\mathbf{F}_c$	teeth contact force, N (lb)
$\mathbf{F}_r$	bearing force, N (lb)
$I_{x'}, I_{y'}, I_{z'}$	polar moment of inertia about x', y', z' axes, kg•m^2 (slug•ft^2)

AUTHORS:

DR. HERBERT S. CHENG *is a professor at Northwestern University. His research interests are in the areas of tribology and failure in mechanical components. He has published about 65 technical articles and chapters in reference journals and books. He received his BS from the University of Michigan, his MS from the Illinois Institute of Technology and his Phd from the University of Pennsylvania. He is the recipient of a number of prestigious honors including Pi Tau Sigma, Tau Beta Pi, Sigma Xi and in 1971 the American Society of Mechanical Engineers Lubrication best paper award. Currently Dr. Cheng is Vice President of The Gear Research Institute and an advisory board member. Additionally he served as the ASME chairman for Research Committee on Lubrication and has served on a number of other committees. He is on the Editorial Advisory Board of Tribos published by BHRA of England.*

DR. CHARLES CHAO *is a Senior Engineer at Solar Turbines, Inc. in San Diego. He has been involved in the design and application of high speed gearing in the aerospace and oil industries. Dr. Chao received his Phd in Mechanical Engineering at Northwestern University. He is a member of ASME and chairman of ASME Gear Lubrication and Sub-Committee.*

MR. MERIWEATHER BAXTER *is the author of approximately 16 papers on gears and cams and has seven patents which are international. A graduate of Yale University with a BA in Mechanical Engineering, Mr. Baxter worked for Gleason Works for forty years. While at Gleason, he held the titles of Chief Engineer of Research and Design, and Director of Engineering. Mr. Baxter retired from Gleason in 1976. Since that time, he has been a consultant for a number of companies including Gleason, Batelle Columbus Laboratories, Lawrence Livermore, Beoring Vertol and Mobile Research and Development. His areas of expertise are shapes of tooth surfaces and contact.*

$\mathbf{i}', \mathbf{j}', \mathbf{k}'$	unit vectors along x', y', z' axes
m	mass of pinion or gear shaft, kg (slug)
$\mathbf{r}_c$	position vector of the teeth contact point with respect to the mass center, m (ft)
$\mathbf{r}_r$	position vector of the bearing supports with respect to the mass center, m (ft)
T_{in}, T_{out}	average input torque, average output torque, N•m (ft•lb)
X, Y, Z	fixed coordinates with origin at the intersection of two shafts
x, y, z	fixed coordinates with origin at the mass center of the pinion or gear
x', y', z'	moving coordinates along the principal axes of inertia of pinion or gear
x'_c, y'_c, z'_c	displacements of mass center of pinion or gear, m (ft)
$\theta_x', \theta_y', \theta_z'$	angular displacements of pinion or gear, rad
ω	angular velocity of pinion or gear shaft, rad/sec

Subscripts:

g	gear shaft
p	pinion shaft

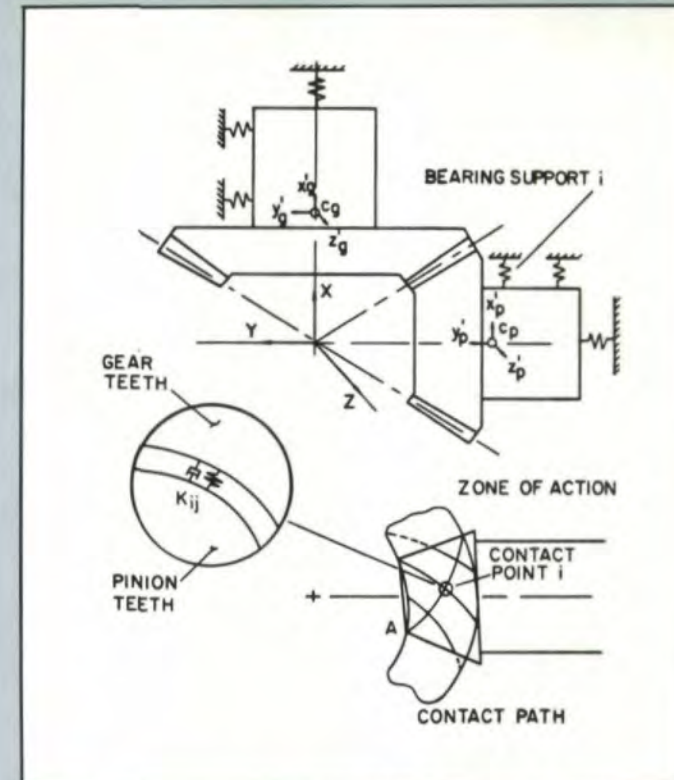


Fig. 1 – Geometry and mathematical model of spiral-bevel gears.

Dynamic Load Equations of Motion

Figure 1 shows the model used for deriving the equations of motion to simulate the steady-state, periodic motion of both pinion and gear as well as the tooth load during a typical cycle during which a pair of teeth traverse through the zone of action from point A to point C. In developing the equations of motion, the pinion and gear are assumed to be rigid bodies each having 6 degrees of freedom. The supporting radial and thrust bearings are assumed to be flexible with known spring stiffnesses. At the contact of each mesh, the teeth are assumed to be connected by a linear spring, which is oriented normal to the contact point and has a stiffness to be determined separately by a finite-element model.

Based on Newtonian mechanics, it is shown (ref. 1) that the equations governing the pinion motion can be expressed in a moving coordinate axes x'_p, y'_p, z'_p, which are instantaneous principal axes of inertia of the pinion. However, the pinion is not fixed in the axes x'_p, y'_p, z'_p, but rotates about the y'_p axis with a nominal angular velocity ω_p. The two vectorial equations of motion for the pinion are

$$\sum_{i=1}^{M} \mathbf{F}_{rpi} + \sum_{i=1}^{N} \mathbf{F}_{cpi} = m_p(\ddot{x}'_{cp}\mathbf{i}'_p + \ddot{y}'_{cp}\mathbf{j}'_p + \ddot{z}'_{cp}\mathbf{k}'_p) \qquad (1)$$

$$\sum_{i=1}^{M} \mathbf{r}_{rpi} \times \mathbf{F}_{rpi} + \sum_{i=1}^{N} \mathbf{r}_{cpi} \times \mathbf{F}_{cpi} + \mathbf{T}_{in} = (I_{x'p}\ddot{\theta}_{x'p} - I_{y'p}\omega_p\dot{\theta}_{z'p})\mathbf{i}'_p + I_{y'p}\ddot{\theta}_{y'p}\mathbf{j}'_p + (I_{z'p}\ddot{\theta}_{z'p} + I_{y'p}\omega_p\dot{\theta}_{x'p})\mathbf{k}'_p \qquad (2)$$

where M is the total number of bearing forces acting on the pinion shaft and N is the total number of contact forces.

Similarly, the equations of motion for the gear can be expressed in the coordinate axes $x'_{g'}$, $y'_{g'}$, $z'_{g'}$ and the gear rotates about the $x'_{g'}$ axis with a nominal angular velocity, w_g. The two vectorial equations for the gear appear as

$$\sum_{i=1}^{M} \mathbf{F}_{rgi} + \sum_{i=1}^{N} \mathbf{F}_{cgi} = m_g(\ddot{x}'_{cg}\mathbf{i}'_g + \ddot{y}'_{cg}\mathbf{j}'_g + \ddot{z}'_{cg}\mathbf{k}'_g) \tag{3}$$

$$\sum_{i=1}^{M} \mathbf{r}_{rgi} \times \mathbf{F}_{rgi} + \sum_{i=1}^{N} \mathbf{r}_{cgi} \times \mathbf{F}_{cgi} + \mathbf{T}_{\text{out}} = I_{x'g}\dot{\theta}_{x'g}\mathbf{i}'_g \tag{4}$$

$$+ I_{y'g}\ddot{\theta}_{y'g} + I_{x'g}\omega_g\dot{\theta}_{z'g})\mathbf{j}'_g + (I_{z'g}\ddot{\theta}_{z'g} + I_{x'g}\omega_g\dot{\theta}_{y'g})\mathbf{k}'_g$$

In these four equations of motion, $\mathbf{F}_{rpi}$ and $\mathbf{F}_{rqi}$ are the bearing reaction forces for the pinion and gear. These can be expressed directly as the product of the stiffness and the displacement vectors in matrix form as

$$\{F_{rj}\}_g = -[DK_j]_g\{D\}_g$$

$$\{F_{rj}\}_p = -[DK_j]_p\{D\}_p \tag{5}$$

The tooth contact forces $\mathbf{F}_{cpi}$ and $\mathbf{F}_{cqi}$ at the contact point are equal and opposite forces, and $\mathbf{F}_{cqi}$ in matrix form can be expressed in terms of the combined teeth elastic compliance matrix $[DC_{ij}]$ and the displacement vectors $\{D\}_p$ and $\{D\}_g$ as

$$\{F_{ci}\}_g = -([DC_{ji}]_g + [DC_{ji}]_p)^{-I}([DG_i]_g\{D\}_g + [DG_i]_p\{D\}_p) \tag{6}$$

Substituting equations (5) and (6) into equations (1) to (4), one obtains a set of 12 equations which can be put in the following matrix form:

$$[m]\begin{Bmatrix} \{\ddot{D}'\}_g \\ \\ \{\ddot{D}'\}_p \end{Bmatrix} + [C]\begin{Bmatrix} \{\dot{D}'\}_g \\ \\ \{\dot{D}'\}_p \end{Bmatrix} + [K]\begin{Bmatrix} \{D'\}_g \\ \\ \{D'\}_p \end{Bmatrix} = \{R\} \tag{7}$$

The details of matrix $[m]$, $[c]$, $[k]$, and $\{R\}$ can be found in reference 2.

It was also found that the rotational equations governing the perturbed gear rotation $\theta'_{x'g}$ and the pertubed pinion rotation $\theta'_{y'p}$ can be combined into one equation to solve for the relative angles θ'_{yp} and $\theta'_{x'g}$. Thus, the reduced system becomes a set of 11 equations which are solved by Runge-Kutta procedure for the displacements of pinion and gear mass center and their small angular rotations.

In implementing these equations, the information needed includes

(1) The tooth contact position as a function of the relative rigid body displacements of the two shafts
(2) The direction of normal vector at the contact point
(3) The combined stiffness of the teeth at the contact point

The teeth contact position and the direction of the contacting normal vector are obtained from a computer code (ref. 3). Because of the geometric complexity of spiral-bevel gears, calculations of combined teeth stiffness are not as simple as those shown in reference 4 for spur gears. For this study, a large-scale finite-element program is used to calculate the deformation due to a unit load at various contact points in the zone of action for a given set of spiral-bevel gears.

Tooth Deflection

For most gears the contact ratio is greater than one, and the load is, in general, not equally shared among the pairs of teeth in contact because the system is a statically indeterminate case. Therefore, one must consider the tooth deflection under the load for each pair in order to determine the load sharing characteristics among the pairs.

Because of the complexity of the spiral-bevel gear geometry, there is no simplified method currently available to calculate the tooth deflection. In order to investigate the system response, shaft deformation must also be included in the calculation of tooth deflection. Therefore, it is more difficult to calculate the tooth deflection by some simple equations. A numerical solution using finite-element method is used to overcome these difficulties.

Some of the recent work (refs. 5 and 6) has proven that the finite-element method yields better results in determining tooth deflection. However, most of this work dealt with two-dimensional problems and did not include the whole gear body. In the present work the spiral-bevel geometry necessitates the use of a three-dimensional, finite-element code. Figure 2 shows a typical eight-node, solid-element grid pattern for a typical spiral-bevel gear and a pinion with three adjacent teeth attached to the gear wheel and shaft. Figure 3 shows a central tooth and its attached ring element of gear. Figure 4 shows parts of a gear shaft and gear wheel. Figure 5 shows whole ring elements with three adjacent teeth of pinion. Figure 6 shows the elements of pinion shaft. Only the central tooth is loaded to calculate the deflection. There are 941 nodes, 562 elements for the gear model and 1029 nodes, 584 elements for the pinion model. Using these grids, one can readily compute the deflection δ under a load P applied at any grid point on the tooth surface. For this analysis the MARC-CDC program was used; the boundaries are considered to be fixed for all the points connected to the thrust bearing to eliminate rigid body displacement; and the boundary nodes connected to the radial bearing were allowed to move in the direction of the rotational axis.

The stiffness at grid point i is defined as:

$$KS_{gi} = \frac{P}{\delta_{gi}}$$

$$KS_{pi} = \frac{P}{\delta_{pi}}$$

The stiffness of a point other than a grid point on the tooth surface can be calculated by the interpolation method. The combined stiffness at the contact point is found to be

$$KS = \frac{KS_g \cdot KS_p}{KS_g + KS_p}$$

Results of Dynamic Load

A series of solutions were obtained to simulate the dynamic response of a set of spiral-bevel gears currently being tested at NASA Lewis. The data for this gearset and the lubricant data are listed in Table I. Effects studied include the running speed, shaft misalignment, and system damping. These results are presented in this section. The dynamic response is expressed by a dynamic load factor defined as the ratio of the maximum dynamic load along the contact path to the average static load. This factor is plotted as a function of speed with different damping ratios and contact ratios.

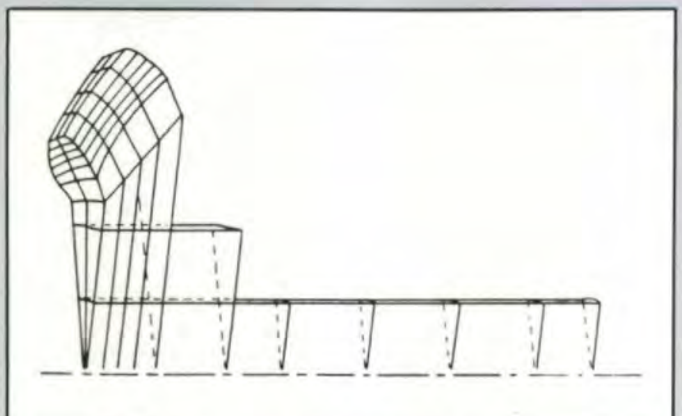


Fig. 2 – Typical section of finite elements of gear segment.

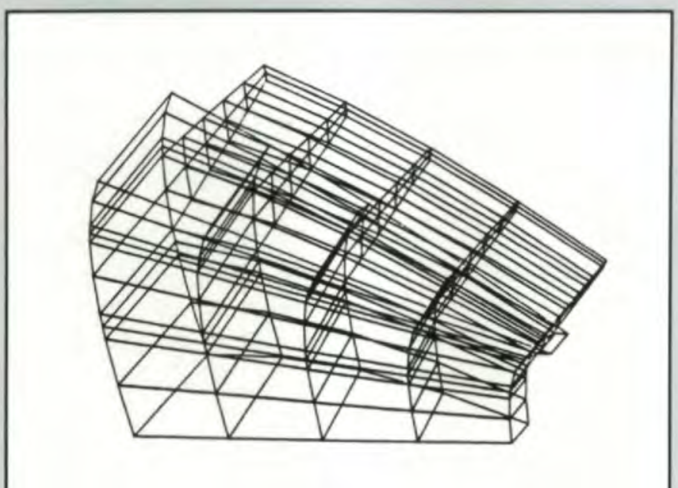


Fig. 3 – Center tooth elements and attached ring elements of gear.

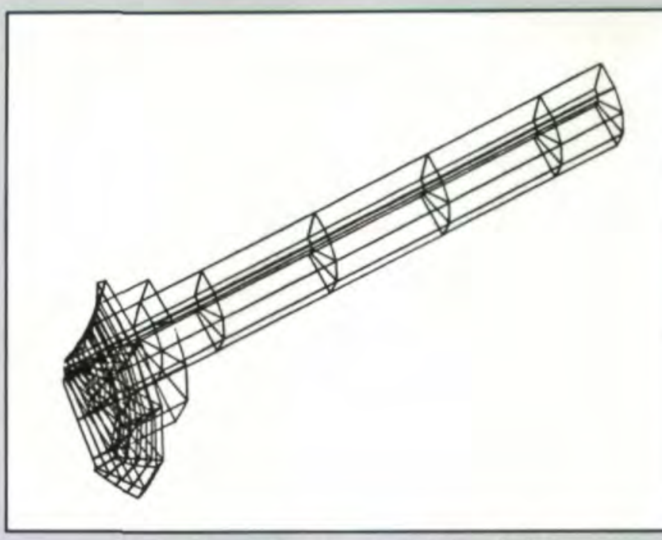


Fig. 4 – Elements for segment of shaft and wheel.

TABLE I. – GEAR AND LUBRICANT DATA

Gear data:	
Number of teeth:	
Gear	36
Pinion	12
Pitch angle:	
Gear	71°34′
Pinion	18°26′
Shaft angle, deg	90
Spiral angle, deg	35
Diametral pitch	5.14
Standard operating conditions:	
Gear rpm	5000
Pinion rpm	15 000
Load at pitch point, N (lb)	11 800 (2660)
Ambient temperature, °C(°F)	37.8 (100)
Geometry dimensions (see fig. 25), m (in.):	
DGG	0.1658 (6.527)
ROG	0.07620 (3.0)
RIG	0.04336 (1.707)
RZG	0.1964 (7.733)
DGP	0.2515 (9.901)
ROP	0.07620 (3.0)
RIP	0.09311 (3.6656)
RZP	0.1987 (7.824)
Gear material data:	
Material	steel
Density, g/cm^3 (lb/in^3)	7.81 (0.282)
Thermal conductivity at 311 K (100°F), W/mK (Btu/sec*in*°F)	46.7 (0.000625)
Young's modulus, GPa (psi)	207 (30 000 000)
Poisson ratio	0.3
Surface convectivity, $W/m^2 \cdot K$ ($Btu/sec.in^2$ °F):	
Oil jet	397 (0.000135)
Oil/air mist	19.8 (0.00000765)
Air	3.97 (0.00000135)
Lubricant data:	
Material	superrefined, naphthenic, mineral oil
Dynamic viscosity at 311 K (100°F), cp ($lb\ sec/in^2$)	64.7 (0.00000938)
Density at 311 K (100°F), g/cm^3 (lb/in^3)	0.61 (0.022)
Thermal conductivity at 311 K (100°F), W/mK (Btu/sec in °F)	0.125 (0.00000168)
Pressure viscosity coefficient, α, m^2/MN ($in^2 lb$)	0.023 (0.00016)
Temperature-viscosity coefficient, β, K (°R)	3890 (7000)
Viscosity-pressure temperature relation	$\mu = \mu_o \exp\{\alpha p + \beta[(1/T) - (1/T_o)]\}$

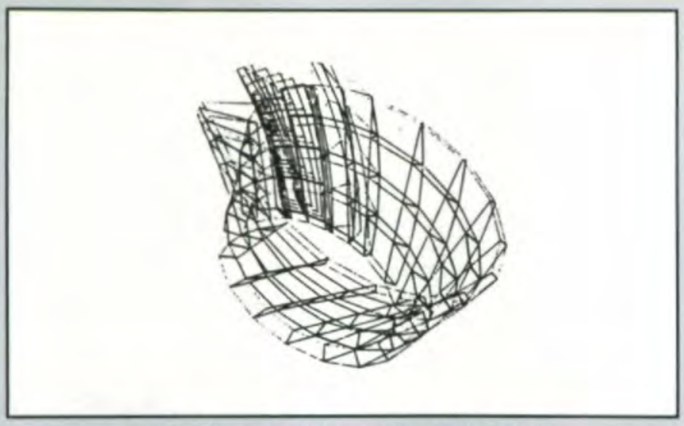


Fig. 5 – Elements of three pinion teeth and rim.

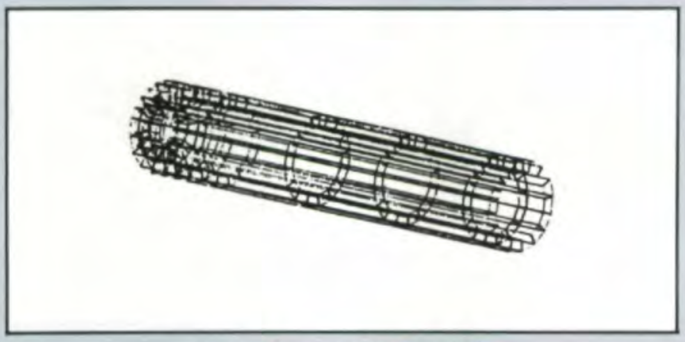


Fig. 6 – Elements of pinion shaft.

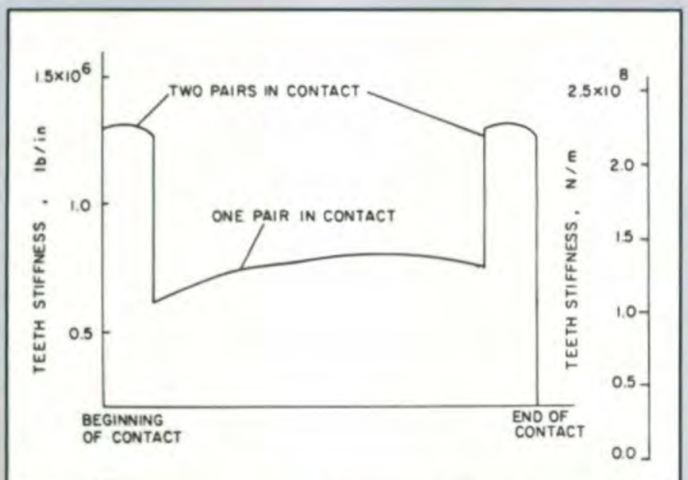


Fig. 7 – Stiffness variation along contact path.

Dynamic Load Variation

For constant input torque the load on the contact point of the two meshing teeth along the path of contact is not constant; this load variation is mainly caused by the following factors:

(1) The variation of stiffness along the contact path
(2) The transition from single pair of contacts to double and from double to single
(3) The effective radius not constant along the contact path.

Figure 7 shows the variation of stiffness for the transition of contacts.

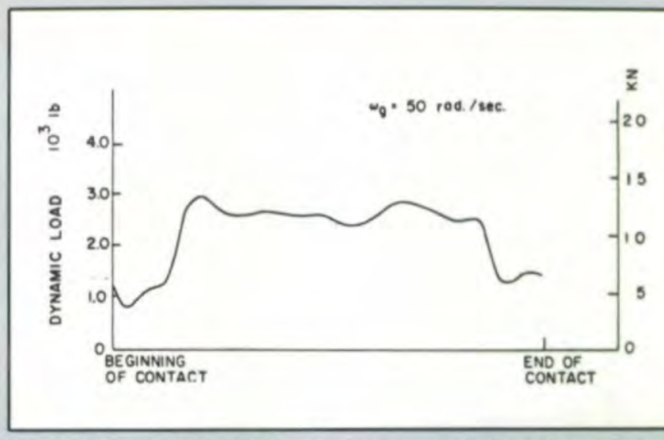


Fig. 8 – Dynamic load variation along contact path. Central contact position, ω_g = 50 rad/sec.

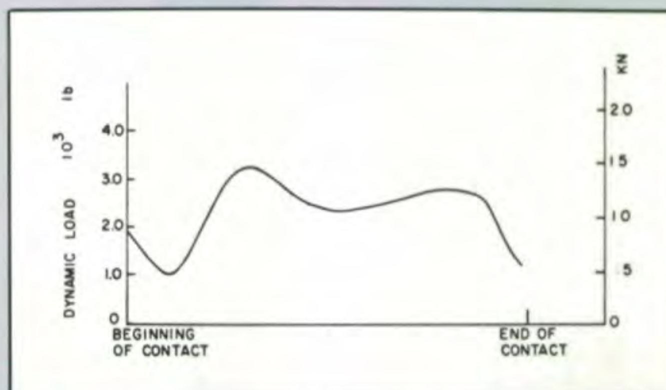


Fig. 9 – Dynamic load variation along contact path. Central contact position, ω_g = 150 rad/sec.

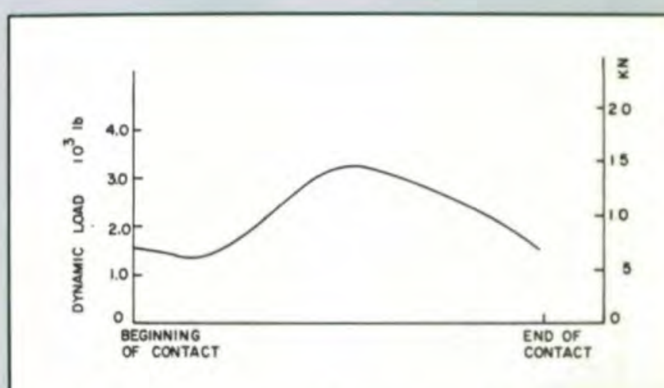


Fig. 10 – Dynamic load variation along contact path. Central contact position, ω_g = 300 rad/sec.

The main excitation to the gear system comes from the periodical change in teeth stiffness due to the alternating engagement of single and double pairs of teeth. The frequency of this excitation force expressed as a meshing frequency depends on the operating speed. Therefore, it dominates the resulting mode of vibration. Figures 8 to 11 show dynamic load variation at four different speeds in the case of central contact; that is, the contact path located centrally between the toe and heel of the tooth.

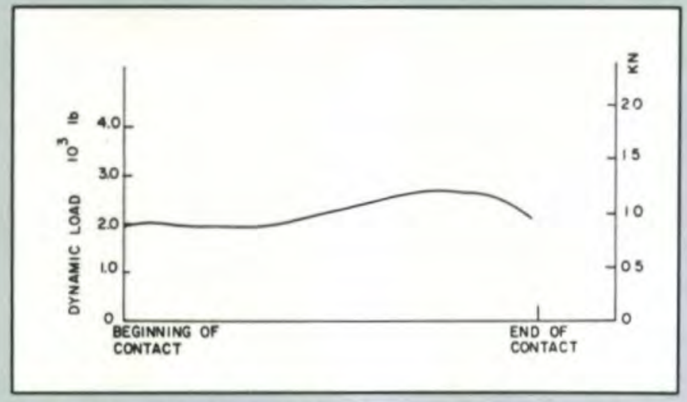


Fig. 11 – Dynamic load variation along contact path. Central contact position, ω_g = 523 rad/sec.

Since there are 11 degrees of freedom in the system, 11 resonating frequencies of the system should exist. In the low-speed region where the excitation frequency from the change of stiffness is much lower than all resonating frequencies, the dynamic load response along the path of contact is somewhat like static load superimposed by an oscillatory load due to the system's resonating frequency.

When the speeds are near the resonance region (Figure 8), the dynamic load response becomes very severe (Figures 9 and 10). The maximum dynamic load is much higher than the static load, which is the case when overloading occurs. Sometimes the oscillation of dynamic load will make meshing teeth separate when the load becomes negative and thus will generate noise and surface fatigue.

As the speed increases beyond the zone of resonating frequencies, the dynamic load becomes smoother along the contact path, and the value is less than the static load (if the contact ratio is greater than one). The variation of dynamic load at this region is out of phase with the change of the teeth stiffness (Figure 11).

Effect of Shaft Misalignment

When the assembly errors are introduced in the system, the contact bearing will shift to either end of the tooth surface.[7] Figure 12 shows the typical paths of central contact, toe contact, and heel contact. Usually the central contact is desired because it can tolerate more possible running position errors and avoid edge contact. The dynamic load

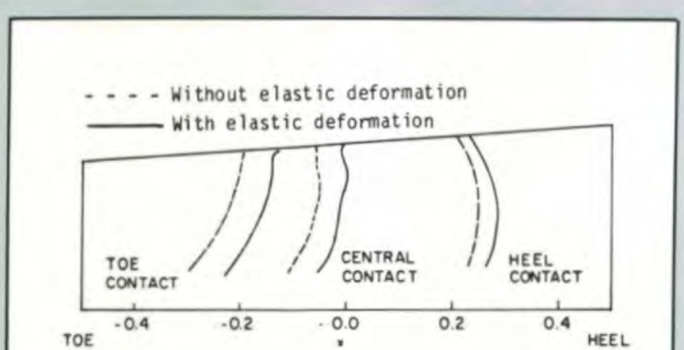


Fig. 12 – Typical contact path for central, toe, and heel contact positions.

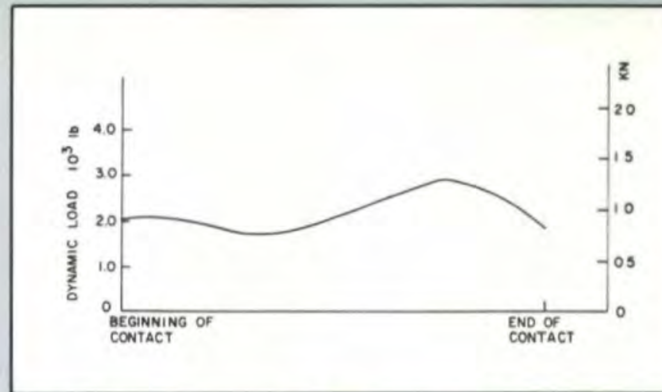


Fig. 13 – Dynamic load variation along contact path. Toe contact position, ω_g = 523 rad/sec.

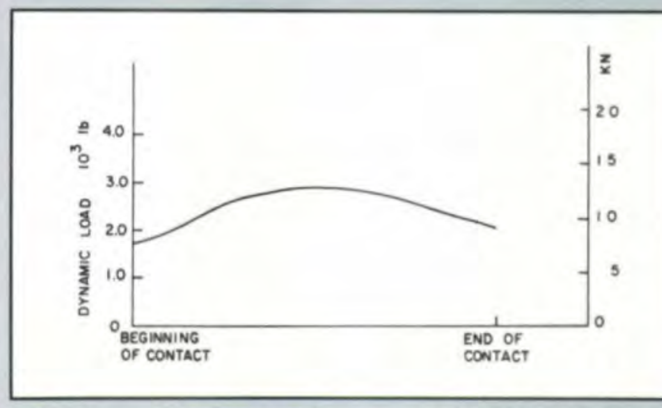


Fig. 14 – Dynamic load variation along contact path. Heel contact position, ω_g = 523 rad/sec.

response of toe and heel contact is shown in Figures 13 and 14. The change of the contact bearing from center to either edge will also change the contact ratio of the system because the tooth surface is not a perfect involute along the profile direction and is mismatched along the lengthwise direction. In the current example the contact ratio for the toe contact is 1.26, the central contact, 1.16, and the heel contact, 1.0. In this case, if the contact bearing is moved farther toward the heel region, there would be no tooth contacts between the time when the previous tooth finishes the contact and the current tooth goes into the contact zone (discontinuity in tooth mesh). This situation would cause very large impact force which would generate noise and severe damages to the tooth surface. The effect of the tooth contact ratio on dynamic response is shown later.

Contact Path Variation Due to Dynamic Response

In addition to showing the contact paths due to the assembly errors in the system in Figure 12, the real contact path, not only due to the assembly errors but also to the running position errors induced by the dynamic responses, is plotted in the same figure. When this real contact path is compared with that caused by the assembly errors and running position errors induced by the average static elastic deformations, the deviation is found to be surprisingly small. One

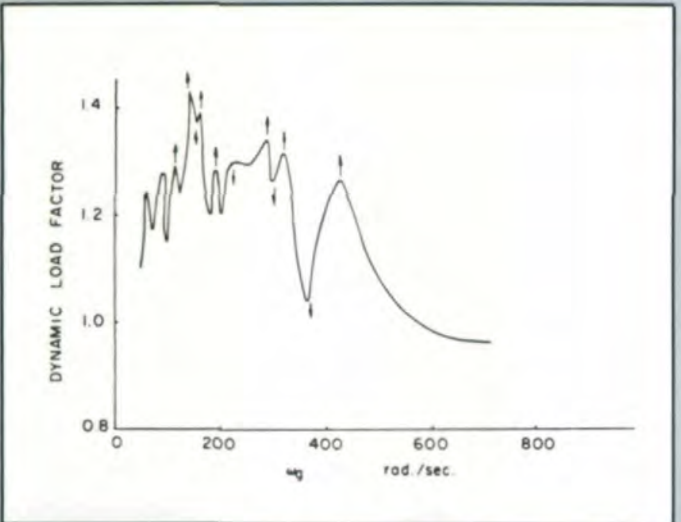


Fig. 15 – Dynamic load factor versus gear speed. All damping coefficients, 2627 N•s/in (15 lb•s/in); contact ratio, 1.16.

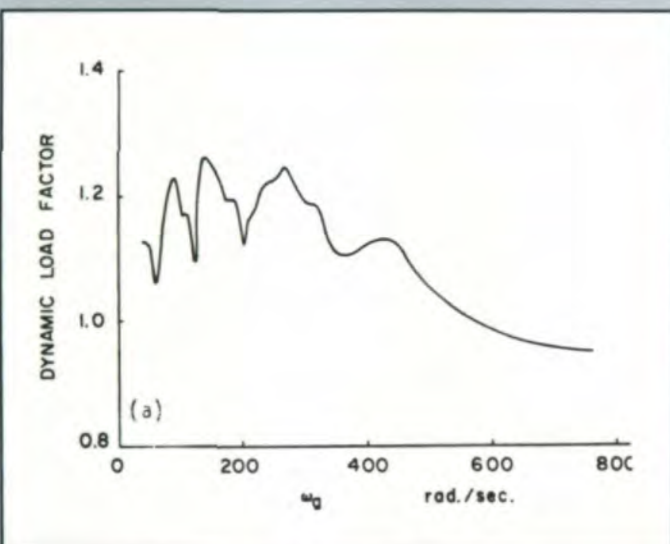
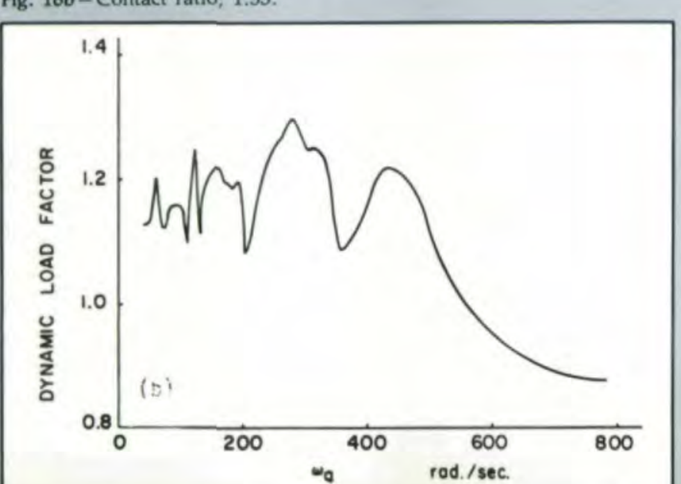


Fig. 16a – Dynamic load factor versus gear speed. All damping coefficients, 4378 N•s/in (25 lb•s/in). Contact ratio, 1.16.

Fig. 16b – Contact ratio, 1.33.



explanation of this small difference might be that the displacements changes due to the dynamic oscillation are small and that they do not produce a large change in contact path compared with those caused by the static displacements only. The closeness between these two contact paths suggests *that one can use average static elastic deformation to calculate* contact path, which can be used directly to solve for the dynamic load and lubrication problems without having to solve the dynamic load and contact path simultaneously using an iterative technique. The elimination of this iterative procedure greatly reduces the computation time.

Effect of Speed

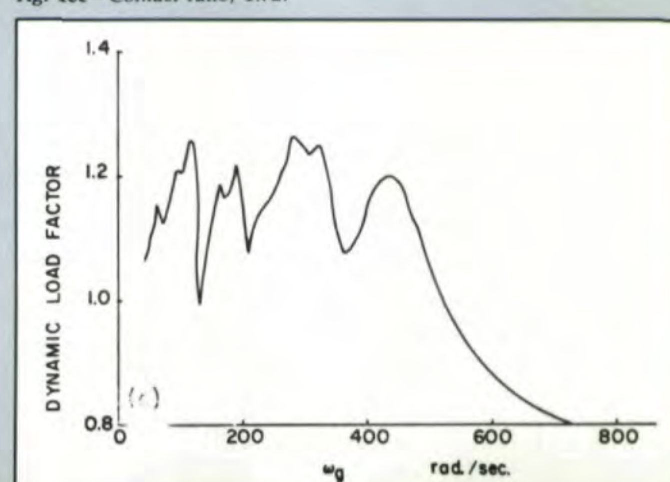
Once the physical conditions of a gearset are determined, the dynamic response depends on the operating speed. For the system of 1 degree of freedom, such as spur gears, the maximum dynamic load occurs when the meshing frequency, which depends on the operating speed, is near the system natural frequency. Some peaks of dynamic load are caused by the varied meshing stiffness along the contact path, and they appear at meshing frequencies lower than the system natural frequency. The dynamic load factor, defined as the ratio of maximum dynamic load to the average static load, is plotted against the gear speed to illustrate the effect of speed in Figure 15. Since there are 11 degrees of freedom in the spiral-bevel gear system, more peaks of dynamic load are expected.

The highest dynamic load appears to occur near the natural frequencies that correspond to the mode associated with a larger displacement in the motion along the line of action. The frequency marked ↑ in Figure 15 shows the system natural frequency causing a larger displacement in the motion along the line of action, and the one marked ↓ shows the system natural frequency with a small displacement in that motion. It is clearly shown that the dynamic load factor at the frequency marked ↑ has a peak response and that the response at the natural frequency marked ↓ is not necessarily a peak.

Effect of Contact Ratio

The contact ratio is defined as the ratio of the duration

Fig. 16c – Contact ratio, 1.72.



(continued on page 26)

CNC Gear Shaping

Dr. G. Sulzer
Liebherr Machine Tools
Germany

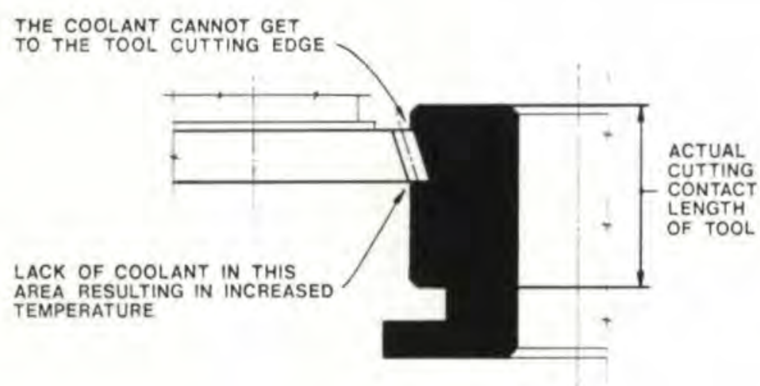


Fig. 1

Introduction

New Developments in Tools and Machines

Two major processes are used for cutting gears, hobbing and shaping. Because of it's universal application and it's high performance, hobbing is generally preferred. A hobbing machine is universal enough to produce worms, worm gears, spur and helical gears. By using Multistart hobs, the productivity of the machine can be increased considerably.

Normally shaping is only used when hobbing is impossible due to the form of the parts to be cut, i.e. internal gears and cluster gears with shoulders. Multistart cutters cannot be used. Nevertheless, the technological disadvantages can be reduced by advanced machine design and better cutter materials.

Tools

Sintered and tin coated high speed steels are generally established as cutter materials. Sintered high speed steel is particularly suited to shaping because of its durability when cutting at high temperatures, which arises more in shaping than in hobbing. Depending on the width of gear, a longer cut occurs and the access for coolant is limited during cutting.(Fig. 1 and Fig. 2)

The additional tin coating improves wear resistance on the tooth flanks. The time of contact between hob and workpiece is generally shorter than in shaping, and the access for coolant is much better.

To a limited extent, carbide cutters are also used but in the field of machining hardened gears only. Similar to skive hobbing, cutting speeds are increased by about 30% compared with soft machining.

The life of a tin coated cutter in it's original condition and after multiple resharpening, has been studied under practical conditions paying particular attention to filtration of the coolants. (Fig. 3) Using a filtered coolant, a life of up to 25 hours can be achieved between cutter regrinds. (Fig. 4) It is obvious that filtration of coolants is an important factor. Another factor is the condition of the tin coating itself. New cutters may vary a lot, but after several sharpenings, consistant life is obtained. At least 20 resharpenings per cutter are achieved and calculated cost should be based on the resharpened cutter rather than on a new tool.

Increasing the productivity of the shaping operation is one aim, another is increasing the flexibility of this method of machining. By using suitable tool shapes, shaping can be universal as well. Some applications are as follows:

Shaping a square using a pinion type cutter: other shapes can be obtained with the use of specially designed cutters. Shaping is then in competition with spark erosion and broaching as an operation.

Simultaneous shaping of three identical gears on one part reduces cutting time by one third.

The same principle obviously applies to two gears per part. (Fig. 5) When coarser pitches are shaped, the two cutters are displaced by half a pitch, thus giving more equal cutting forces. Using multiple cutters means that the tools have to be resharpened as pairs in order to give identical diameters.

Shaping Machines (Fig. 6)

Conventional shaping machines are driven by mechanical gear trains. A main motor drives the reciprocating motion of the cutter spindle, and a gear train driven from the stroke motion synchronizes a cam mechanism for relieving the cutter on its return stroke. The rotary motion of cutter and workpiece can be driven by a separate motor, but for the correct synchronization individual index gears have to be mounted.

Radial motion is produced by hydro or electromechanical drives and table positions are controlled by tripdogs and microswitches. Setting stroke position and stroke length is carried out manually and the same applies to the lateral offset

AUTHOR:

DR. G. SULZER *is Manager of Research & Development at LIEBHERR in West Germany. He earned his doctorate at the University of Aachen in Aachen, Germany where he has worked in the role of Manager Research Development for twelve years. He has been instrumental in the development of the Liebherr CNC machines.*

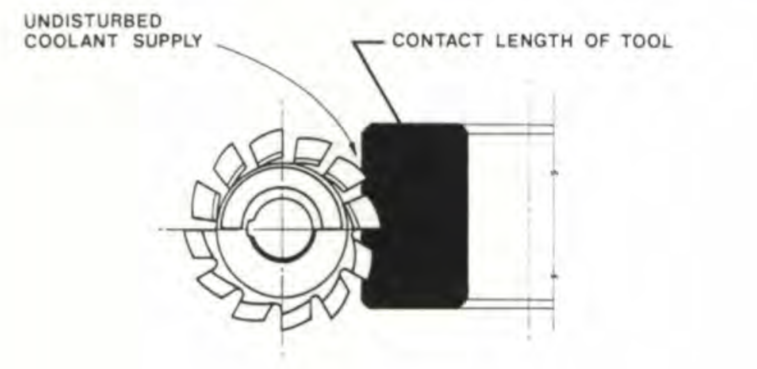


Fig. 2

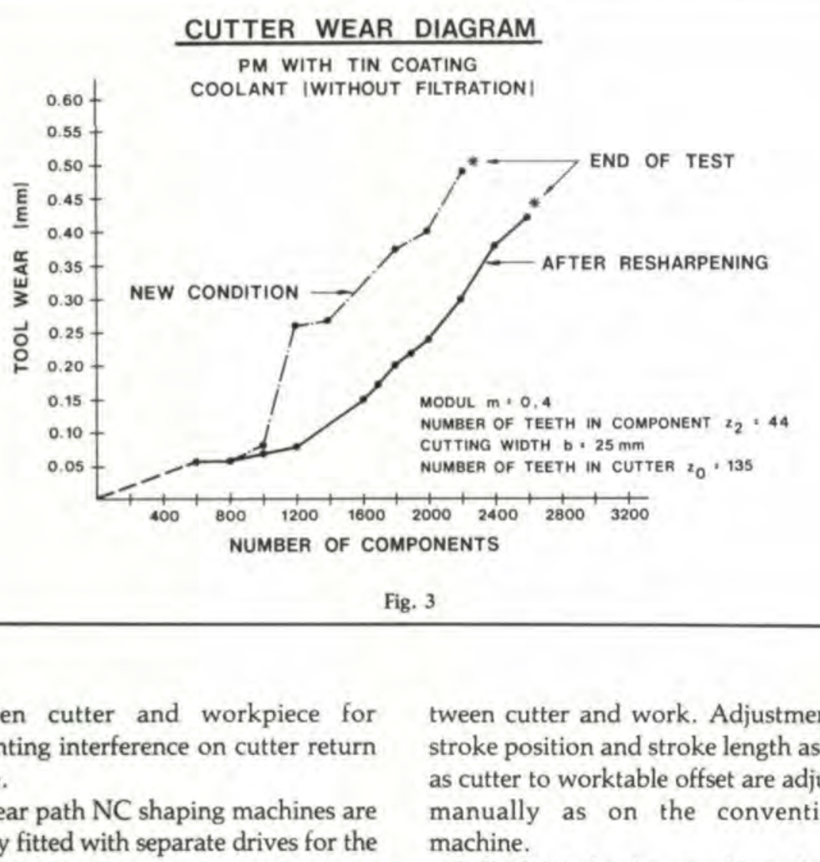


Fig. 3

between cutter and workpiece for preventing interference on cutter return stroke.

Linear path NC shaping machines are usually fitted with separate drives for the following:

Stroke motion
Rotation of tool and workpiece
Radial motion (X axis)
Stroke position (Z axis)

With this type of machine, it is still necessary to use index change gears in order to achieve the synchronization between cutter and work. Adjustment of stroke position and stroke length as well as cutter to worktable offset are adjusted manually as on the conventional machine.

Full CNC shaping machines feature separate drives for the following: (Fig. 7)

Stroke motion with depth center positioning (S axis)
Rotation of tool (D axis)
Rotation of workpiece (C axis)
Radial motion (X axis)
Stroke position (Z axis)
Stroke length (V axis)
Offset cutter stroke workpiece (Y axis)
Relief angle (taper) (B axis)

Through using NC on all axes, a CNC shaping machine gives the following advantages:

S Axis, stroke motion with dead center positioning

Cutter speed (number of strokes per minute) is programmable for any combination of infeeds, roughing, finishing or dwell operation. Therefore shortest cutting times are achieved without overloading the cutter. The dead center positioning feature assures safety when cutting internal gears (clearance between the cutter and workpiece during radial motions) and enhances stroke positioning in the set up mode.

D/C-axis, rotation of tool and workpiece

In normal applications, the axes D and C are performing a synchronizing motion between cutter and workpiece and controlling the generating feed programs. Mounting of index change gears is obsolete, feeds are programmable and dependent on the geometry and machinability of the parts to be cut. For balanced wear distribution on the flanks of the cutter, the direction of rotation can be changed automatically.

For special applications, each of the axes can be moved individually, such as for single index shaping of splines or keyways or angular positioning of the cutter to the workpiece. A combination of shaping by the generating method and subsequent single indexing (with a second cutter) in one operation is possible. Thus the range of applications is considerably extended.

X-axis, radial motion

Positional accuracy of the NC is approximately 0.005 mm. This means that safety clearances can be reduced to the outside diameter to be cut, and rapid traverses close to the outside diameter of the workpiece can be used. Tool offsets (difference in diameter) of the cutter to be used can be measured outside of the machine and an offset entered into the control while the machine is in operation. A combination of rotary and radial feeds allows any process such as pure radial infeed and extreme spiral infeeding to be

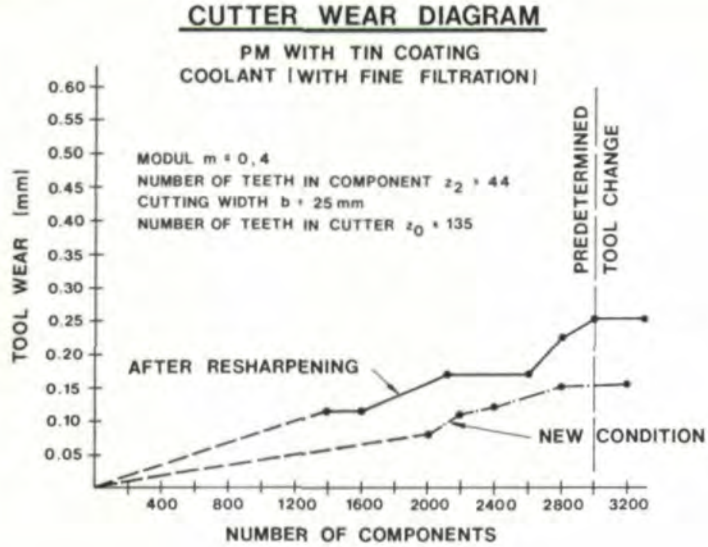


Fig. 4

Full CNC shaping machines feature separate drives for the following:

Stroke motion with depth centre positioning (S axis)
Rotation of tool (D axis)
Rotation of workpiece (C axis)
Radial motion (X axis)
Stroke length (V axis)
Offset cutter stroke workpiece (Y axis)
Relief angle (taper) (B axis)

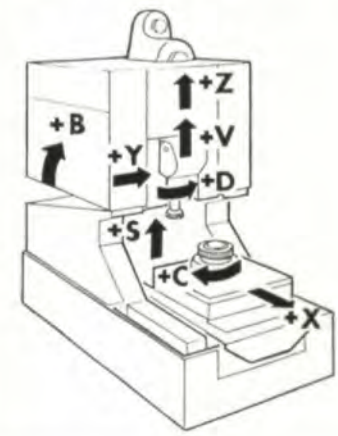


Fig. 7

Fig. 5

used. Optimum conditions for any combination of cutter and workpiece can be used.

Z-axis, stroke position

On conventional machines, it is necessary to set a stroke position after every tool change and sometimes during the machining cycle in the case of cluster gears (shaping two or more different gears in one operation), (Fig. 8 and Fig. 9) or with certain types of internal gears where the gear itself is lower than the upper face of the workpiece. Automatic stroke positioning by the NC saves a considerable amount of time and is more accurate and faster than manual setting.

Fig. 8

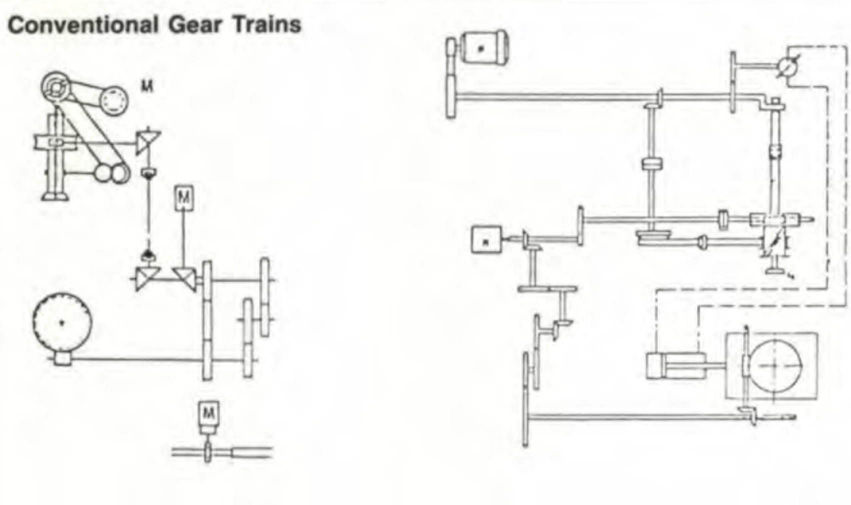


Fig. 6

V-axis, stroke length (Fig. 10)

Programming the stroke length reduces set up time. With advance control systems it is only necessary to enter the gear width pitch and helix angle; the stroke length itself is calculated and set by the control system. A major advantage of NC setting of the stroke length is with cluster gears with different face widths, (Fig. 11b) where for any gear, optimum stroke lengths can be applied. This is essential for economic production. (See Fig. 11a)

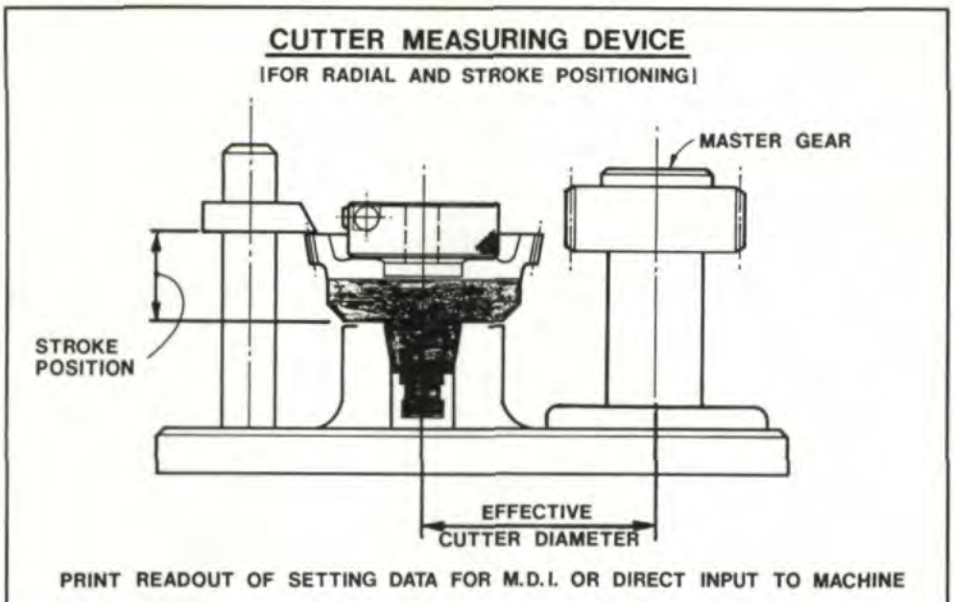


Fig. 9

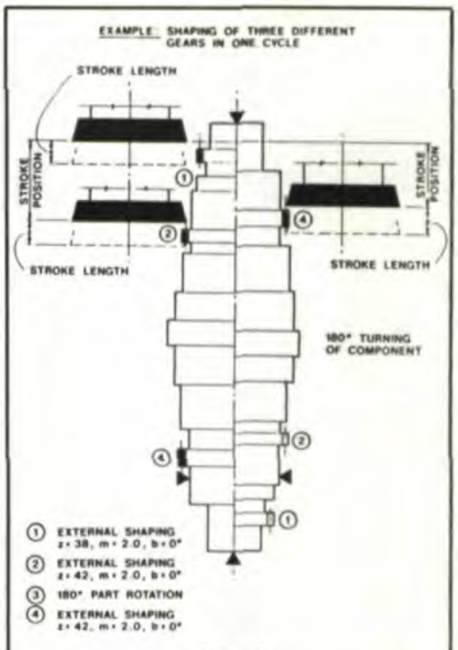


Fig. 11b

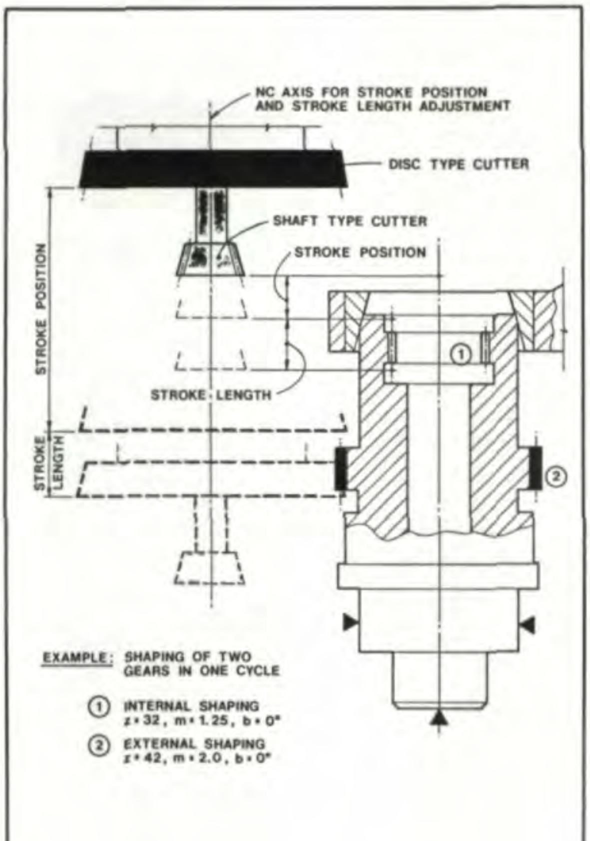


Fig. 10

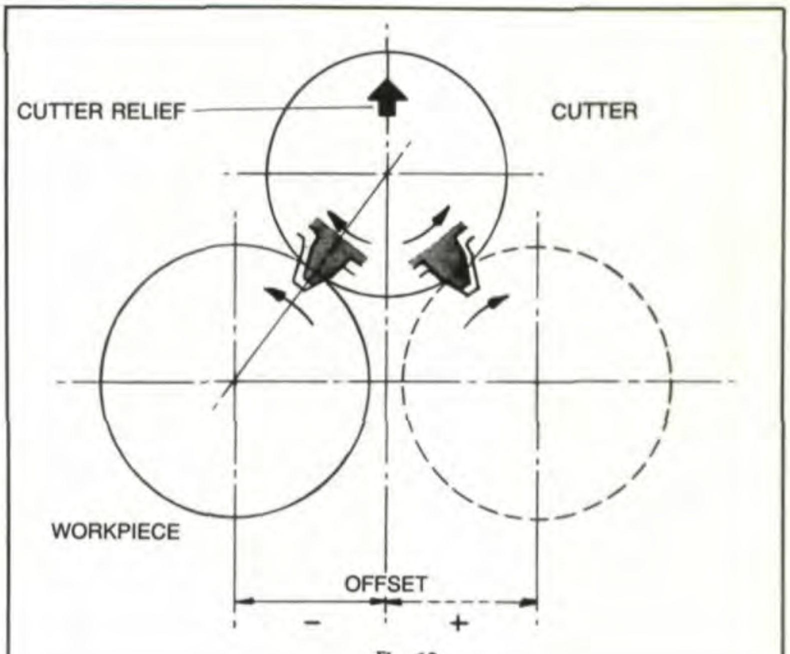


Fig. 12

EXAMPLE:

Gear 3: 42 teeth
stroke 20 mm

with CNC-stroke length:
speed: 50 mm/min
strokes: 800 per min
time: 1.2 min

Gear 2: 42 teeth
stroke 15 mm

speed: 50 mm/min
strokes: 1200 per min
time: 1.2 min

without CNC-stroke length:

speed: 50 m/min	speed: 37 m/min
strokes: 800 per min	strokes: 800 per min
time: 1.2 min	time: 1.2 min

Savings with CNC: 0.4 min per part = 33%

Fig. 11a

Y-axis, offset cutter/workpiece (Fig. 12)

This axis provides the adjustment necessary for optimum cutter relief in order to prevent interference between tool and workpiece on the return stroke. This setting normally changes with every tool-workpiece combination and in some cases with the resharpening of the cutter. Setting via NC means a reduction in setting time, optimization of cutting conditions, and better tool life by changing

Fig. 13

the direction of rotation, which means reverse offsets between cutter and workpiece.

B-axis, relief angle (taper) (Fig. 13)

Angles can be programmed for any taper and for minor corrections in order to correct heat treatment distortions. General advantages of a full CNC shaping machine in production:

- ease of operation
- high accuracy
- maximum repeatability
- short cycle times
- easy servicing
- extended applications

Ease of operation

Fixed programs (canned cycles) are used for repetitive operations. These programs are fed with basic parameters only, which can be read directly from the part drawing and the tool specification. NC programming is not necessary, as entering the parameters is by a question and answer dialogue between machine control and the operator, and feasibility checks are carried out.

For special applications, which deviate from the canned cycles, tailored CNC programs can be written either by the user or the machine tool manufacturer. These programs and their parameters can easily be adapted to similar components. All programs and parameters can be stored in the controlled memory and recalled on request. Programming, as well as I/O operations can be carried out while the machine is working. Thus downtime is minimized and the safety of programming is enhanced.

High Accuracy

The closed loop between cutter and workpiece drive prevents any of the distortions known with conventional shaping machines which have long kinematic trains. This postively affects both lead and pitch accuracy, and the old dropped tooth condition is virtually eliminated.

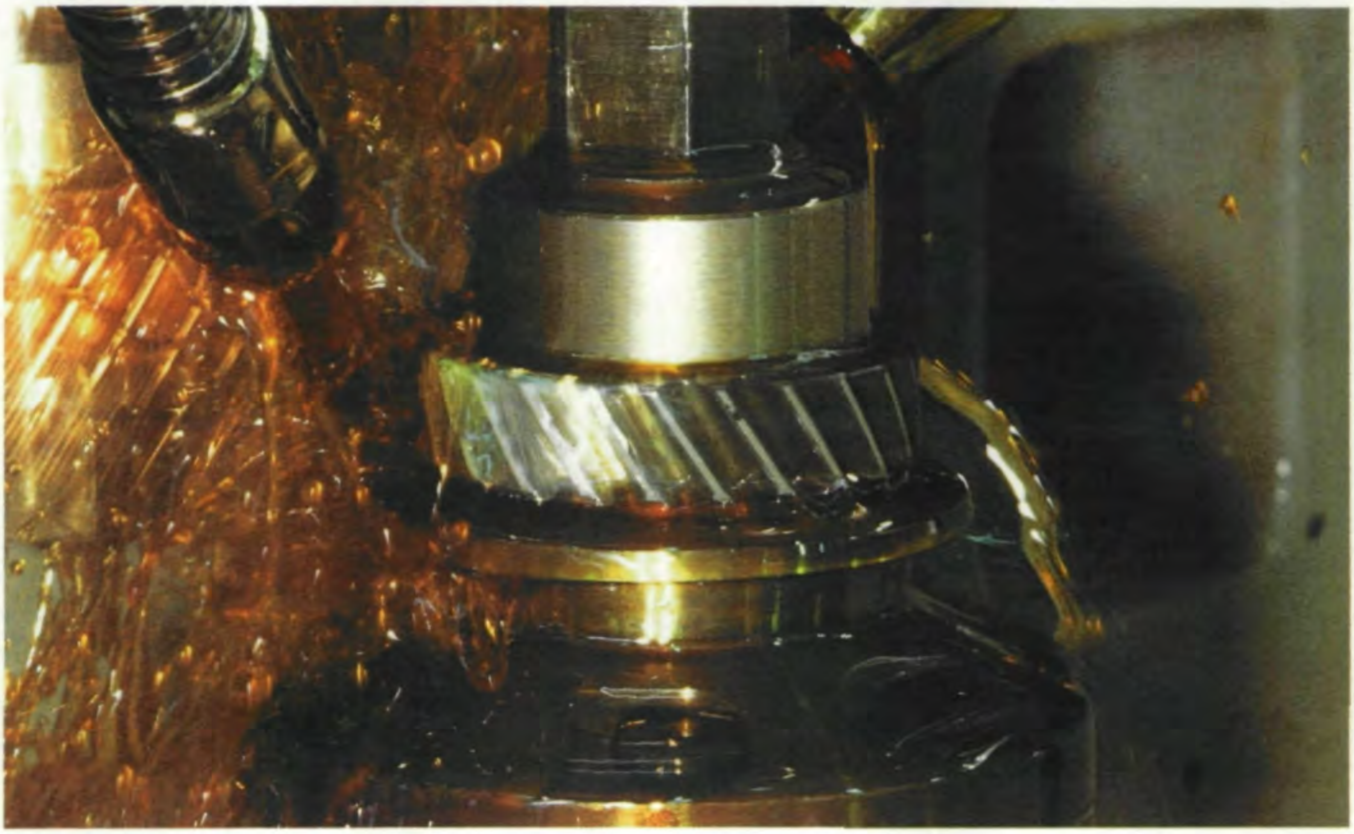
Maximum repeatability

Once optimum settings are established, they can be repeated identically on every batch of components. The closed loop control on the radial axis guarantees a constant size of all parts within one batch.

Short cycle times

The stored programs considerably reduce set up operations. The storage capacity (about 100 parts and their cut-

THE INNOVATIVE "SILENT SHAFT" GEAR SHAPER CUTS CYCLE TIME TO 1/2!



Mitsubishi has developed a new "silent shaft" mechanism that overcomes the vibrations encountered in high speed operations of gear shaping and lets you cut the cycle time in half. Two silent shafts rotating together with the crankshaft, cancels vibration on the SA25NC Gear Shaping Machine. This innovative mechanism permits speeds as high as 1,500 strokes per minute ... yet the vibration is one third of conventional machines.
Another feature of the SA25NC is the high rotary feed. The combination of the two features enables you to cut gears faster maintaining high accuracy and gives longer tool life to the cutter.
Simplified programming, reduced setup times and many, many more features to save you money. For more exciting information, contact our sales office in Bensenville, Illinois now!

Main Specifications

Max. dia. of work	
External gear	10″ (4″ cutter)
Internal gear	4.72″ + cutter dia.
Max. D.P.	4DP
Max. work width	2.36″
Cutter stroke	300 ~ 1500str/min
Rotary feed (4″ cutter)	0.0004″/str (at 300 str/min) 0.00008″/str (at 1500 str/min) 0.11 ~ 177.16 ipm
Table dia.	13″
Main motor power	7.5hp
Machine weight	11,000 lb.

CNC GEAR SHAPER

MITSUBISHI
HEAVY INDUSTRIES, LTD.
5-1, Marunouchi 2-chome, Chiyoda-ku, Tokyo, Japan
Cable Address: HISHIJU TOKYO

Mitsubishi Heavy Industries America, Inc.
873 Supreme Drive, Bensenville, IL 60106 Phone; (312) 860-4220

Mitsubishi International Corporation
873 Supreme Drive, Bensenville, IL 60106 Phone: (312) 860-4222

CIRCLE A-8 ON READER REPLY CARD

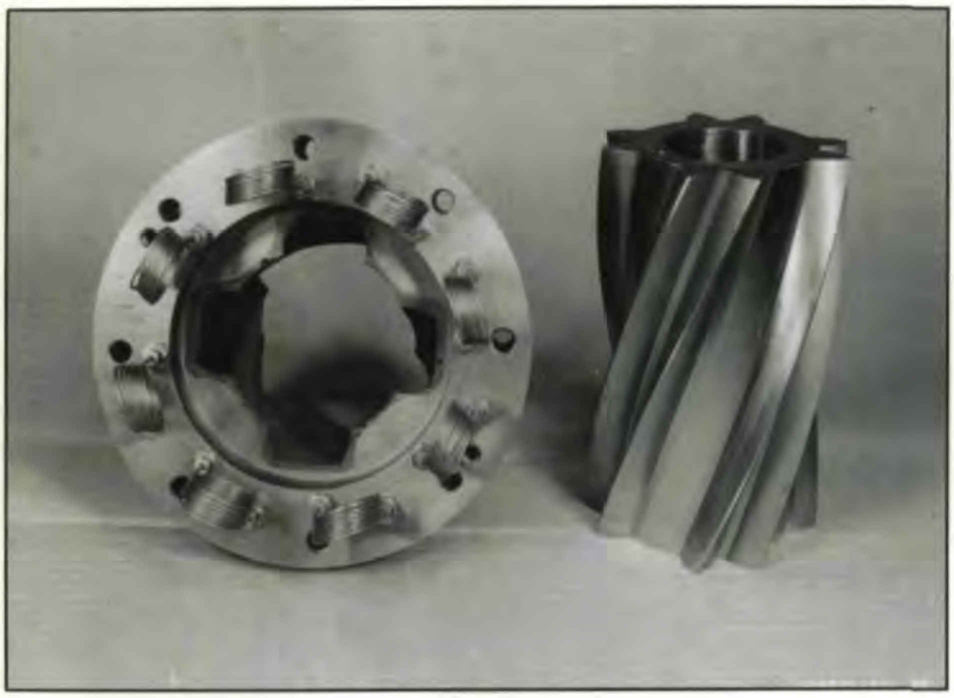


Fig. 14

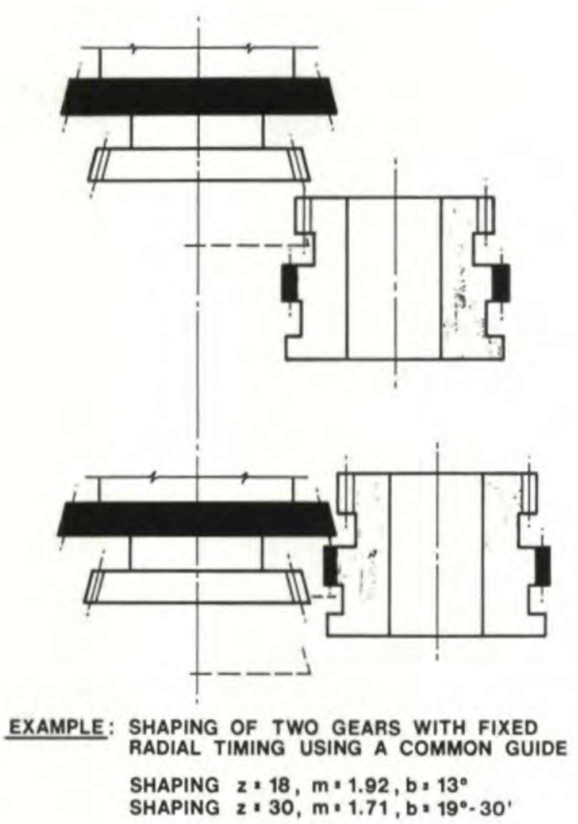


Fig. 15

ters) can be increased on request. Non productive times can be further reduced by using quick change systems for cutters and fixtures. Parallel programming increases uptime, and automation systems for part loading and unloading reduces idle time. Combining more than one operation in one loading of the workpiece also increases uptime and reduces idle time.

Cutting performance can be optimized by advanced programming and additionally by means of 3 override switches which control speed, radial feed and generating feed individually. If the number of strokes is changed, the generating feed per stroke remains unchanged so there is no risk of overloading the cutter.

Easy servicing by a machine diagnostic covering 100 potential faults gives plain language messages on the CRT display panel. According to the type or error, the system distinguishes between warnings and dangerous faults, and reacts accordingly. In the case of a fault which can cause damage, the machine is cut off automatically by the control system. Remote diagnosis through a modem and the normal telephone lines can be made by the machine tool manufacturer. Clear and quick diagnostics are the essential factor. Repairs can be easily made by the exchange of faulty control boards, encoders etc.

Extended applications

Full CNC shaping machines feature extended flexibility. Slots, keyways and other forms can be machined in almost any shape. Forms which cannot be shaped or hobbed can be manufactured by the single indexing method.

With the features offered by the CNC control, spare gears with the following specifications can be manufactured automatically.

Differing pitches
Differing numbers of teeth
Differing axial positions
Differing gear widths

The only limiting factor is still the lead of the cutter guide. (Fig. 14) This guide has to be changed for left hand and right hand helical gears and for spur gears; a combined operation of differing hand leads is not possible. This limitation does not apply to hobbing machines, as the NC controlled hob head swivel and the

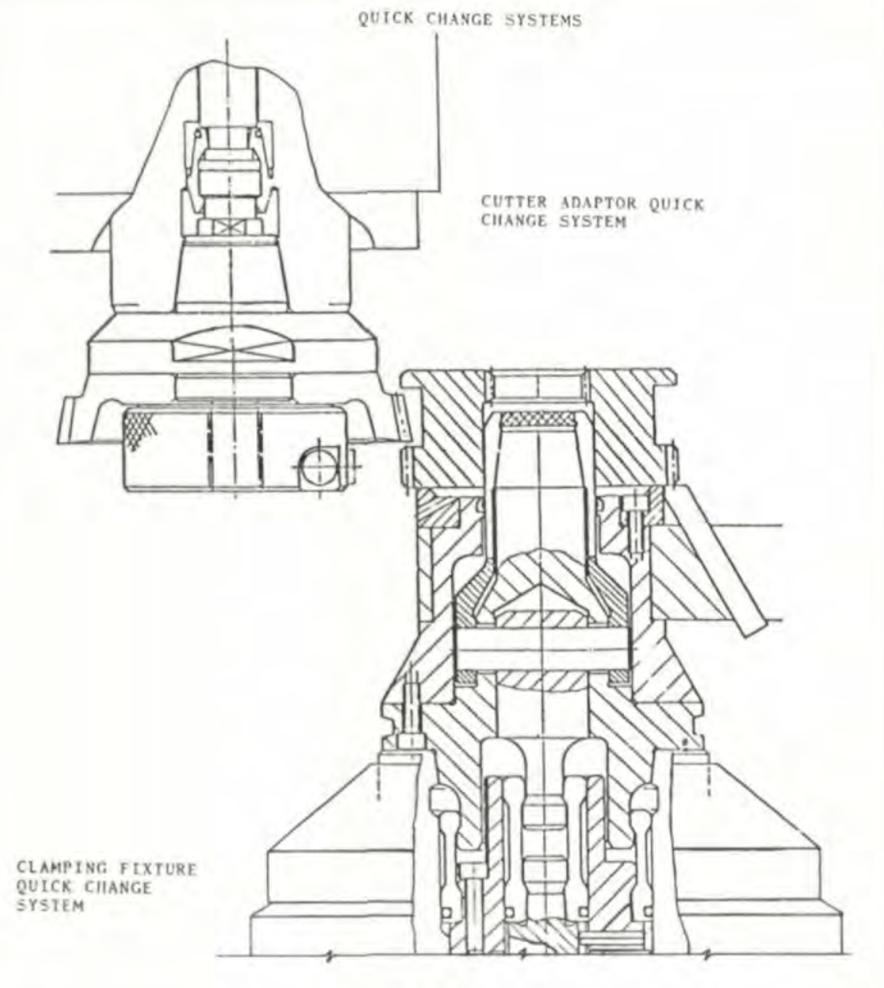


Fig. 16

1) fixture quick change system
2) tool quick change system
3) parallel programming
4) Y-axis required
5) V-axis required
6) Z-axis required
7) B-axis required

NC differential can cope with any helix angle.

It is possible to shape different helix angles of the same hand with one guide. (Fig. 15) A large range of helix angles can be cut by using differing numbers of teeth on the cutters (different diameter). Currently, the possibilities of modern CNC shaping machines are not being fully exploited by gear designers.

Conclusions

Some of the technical possibilities of CNC gear shaping have been described. The decision about installing a full CNC shaping machine is however, based on economic factors. Determining factors are savings in set-up cycle times. The following table shows a comparison between the set-up times for a conventional and a full CNC machine: (Fig. 16)

Operation	Time (min) conv	full CNC
Fixture change	20	5 1) (Fig. 19)
Cutter change	5	1 2)
Change index gears	10	-
Select program feeds speeds	1	- 3)
Set cutter/workpiece offset	8	- 4)
Set limit switches radial	3	-
Set cutting depth	1	-
Set stroke length	5	- 5)
Set stroke position	5	- 6)
Set relief angle (taper)	8	- 7)
Correct cutting depth after 1 gear		
	71 min	7 min

This paper was presented at the SME "Gear Processing & Manufacturing Clinic," Nov. 1985.

E-3 ON READER REPLY CARD

COMPUTER SOLUTION . . .

(continued from page 30)

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Work on this article was done under NASA Lewis contract NSG-3143.

E-1 ON READER REPLY CARD

VIEWPOINT

(continued from page 6)

I would like to point out an error in the November/December 1985, Gear Technology article "Finding Gear Teeth Ratios" which may be causing undue stress to some of your readers.

Equation number 4 on page 26 which is shown as:

$$Y_n = 1 - A_n Y_n - 1 + Y_n - 2$$

Should Be

$$Y_n = Y_n - 2 - A_n Y_n - 1$$

I found the article interesting and plan to use the program as a computerized method of selecting change gears for setting up hobbing machines.

Patrick J. Radle
Doerr Electric Corp.

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CIRCLE A-5 ON READER REPLY CARD

COMPUTER SOLUTIONS . . .

(continued from page 15)

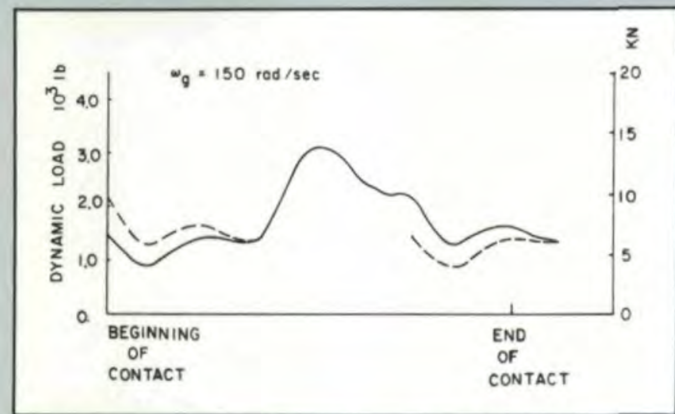


Fig. 17 – Dynamic load variation along contact path. Damping coefficient, 4378 N•s/in (25 lb•s/in); contact ratio, 1.72.

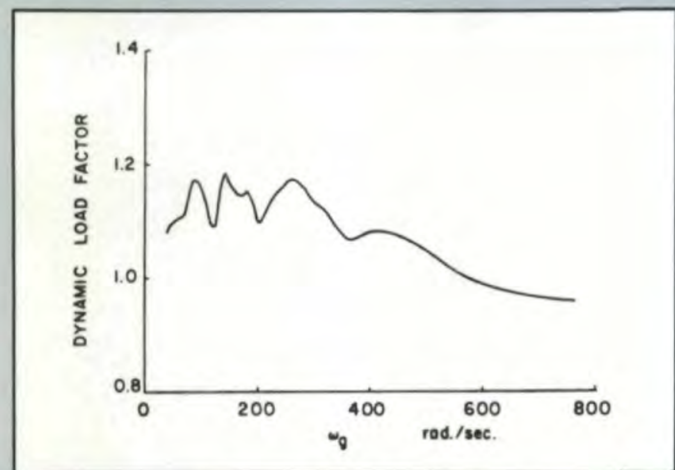


Fig. 18 – Dynamic load factor versus gear speed. All damping coefficients, 6129 N•s/in (35 lb•s/in); contact ratio, 1.16.

for one tooth going through the whole contact zone to the duration of a periodic meshing cycle. It is believed that the load sharing characteristics caused by more than one tooth in contact will reduce the static load. The dynamic load factor due to the effect of contact ratio is shown in Figure 16. It can be seen that the maximum dynamic load factor does not change much. However, the effect of contact ratio is significant in high-speed region, where the load is spread out averagely between meshing teeth path. A typical dynamic load variation with a high contact ratio along the contact path is shown in Figure 17.

Effect of Damping

Since the damping forces are usually not known in the gear system, three arbitrary values are chosen for the damping coefficients: 2627, 4378, and 6129 N•sec/m (15, 25, and 35 lb•sec/m). These values are selected to give a range of nondimensional damping ratios corresponding to those commonly used in spur gears (0.1 to 0.2). The nondimensional damping ratios that correspond to the above three damping coefficients are 0.087, 0.14, and 0.203. The dynamic response for these damping cases can be observed from Figures 15, 16(a), and 18. It is expected that the larger the damping force, the smaller the dynamic load factors will be in the resonance region. The large damping force will also level off the peak of dynamic load factor in the subresonance region, and there is no effect to the dynamic load factor due to damping force in the superresonance region.

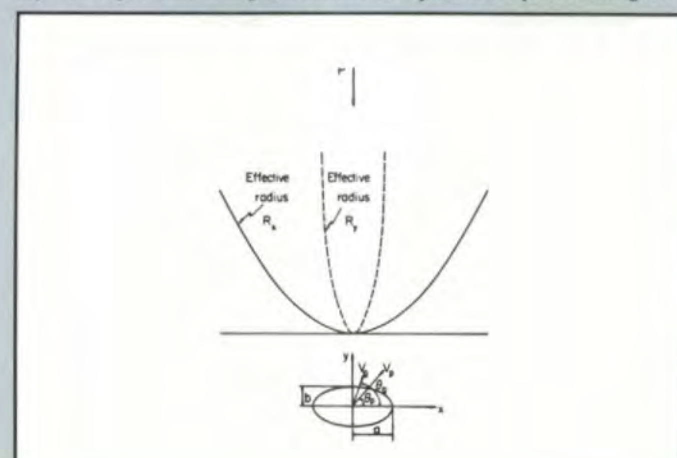
Lubrication of Spiral-Bevel Gears

It is well accepted that the two major modes of gear failure, surface pitting and scuffing, are most strongly related to lubrication at the contact. Considerable gains in pitting life can be realized if the ratio of the lubricant film thickness to the surface roughness is increased. The knowledge of film thickness is believed to be essential for developing new analytical models for the prediction of surface durability for spur and helical, as well as spiral-bevel, gears. The variation of film thickness along the path of contact is mainly controlled by the local inlet lubricant viscosity, the local entrainment velocity, and the local contacting load. The local inlet lubricant viscosity in turn, depends on the bulk surface temperature at the inlet of the contact. Since the bulk surface temperature is directly controlled by the sliding heat generated by the sliding tractive force which is, in turn, affected by the film thickness, the temperature and film thickness are mutually dependent and are solved simultaneously. The approach used in solving the simultaneous film thickness and bulk surface temperature is very similar to that used by Wang and Cheng[7] with the exception that the three-dimensional, spiral gear geometry necessitates the use of the point contact EHL theory for the film thickness and a three-dimensional, finite-element temperature code for the temperature influence coefficients, which are required for calculating the bulk surface temperature.

Lubricant Film Thickness

In determining the lubricant film thickness, the quasi-steady-state model is used, and the transient squeeze-film effect, which was included in a previous work for spur gears, is neglected in this film analysis for spiral-bevel gears. The neglecting of this squeeze-film effect is justified on the basis

Fig. 19 – Equivalent EHL point contact for spiral-bevel pinion and gear.



that it was shown by Wang and Cheng in spur gears to be a secondary effect.

The contact of a spiral-bevel gear and pinion set can be seen in Figure 19, in which there is effectively a flat plane contact with a body, which is described by the difference of the neighboring surfaces between gear and pinion at contact point. This curved body has effective radii R_x and R_y along the principal axes, x and y, respectively. Under a load P, the surface near the flat plane will deform to an elliptical shape with simimajor axis a and semiminor axis b. The velocities of the pinion and gear at the contact point are V_p and V_g. The ellipticity parameter is defined as a/b. The minimum film thickness in the contact zone, following Hamrock and Dowson[8], can be related to Dowson-Higginson's line contact solution[9] by the equation

$$H_{\min} = H_{\min,l}\,(1.0 - 1.6\,e^{-0.62\,k}) \tag{10}$$

$$k = \frac{a}{b}$$

$$H_{\min} = \frac{h_{\min}}{R_x}$$

$$R_x = \frac{R_{x1}R_{x2}}{R_{x1}+R_{x2}}$$

$$u_{px} = V_p \cos\theta_p$$

$$u_{py} = V_p \sin\theta_p$$

$$u_{gx} = V_g \cos\theta_g$$

$$u_{gy} = V_g \sin\theta_g$$

$$u_x = \frac{U_{px}+U_{gx}}{2}$$

$$u_y = \frac{Y_{py}+U_{gy}}{2}$$

$$V = \sqrt{U_x^2 + U_y^2}$$

where

- $H_{min,l}$ dimensionless film thickness of Dowson-Higginson solution
- V_p, V_g velocity of pinion or gear tangent to contact plane
- θ_p, θ_g angle between pinion or gear velocity and y axis

The dimensional $h_{min,l}$ can be expressed as

$$h_{\min,l} = 1.6\,\alpha^{0.6}\,(\eta_o V)^{0.7}\,E'^{\,0.03}\,\frac{R_x^{0.43}}{W^{0.13}} \tag{11}$$

where

- α pressure viscosity coefficient
- η_o viscosity taken at bulk surface temperature
- W effective line contact load per unit length
- E' $[(1-\nu_1^2)/2E_1 + (1-\nu_2^2)/2E_2]^{-1}$

It is important to note that η_o, W, k, and V are variables along the contact path. The V depends on the gear kinematics; a and b depend on the gear geometry; and η_o depends strongly on the local bulk surface temperature which is, in turn, influenced by the local film thickness through the frictional heating. Thus, the film thickness and the bulk surface temperature are interdependent and are solved as a coupled system.

Bulk Temperature

Before the gear system starts to operate, all elements are in ambient temperature. Then the temperature builds up as gears are running, due to the frictional heat generation. After a sufficient period of running, the gears reach a steady-state temperature, that is, the heat flux flowing into the body equals that flowing out of the body. At each revolution the tooth is subject to the same heating condition. Since the time period of each contact point in the contact zone is only a very small fraction of the period of revolution, the local temperature jump (flash temperature) is completely damped out before it enters the contact zone at the next revolution. An average heat input over one revolution will be used to calculate the temperature rise of the body at the steady state.

The heat input is due to the heat generated at the instantaneous contact ellipse, and the amount depends on the load and the shear force of the lubricant. The heat flux flowing out of the body is due to the heat convection to surrounding air and lubricant. The relative importance of the heat-transfer coefficient at different surface areas was discussed by Patir and Cheng and Townsend and Akin[10 & 11] in spur gear systems. They also revealed the significant effect of lubrication method on temperature distribution. In this study, the oil-jet impingement depth is assumed to cover the whole area of contact side, which can be obtained by using a properly placed pressurized oil jet. The heat-transfer coefficients at other various areas are estimated to calculate the bulk temperature.

A three-dimensional, finite-element program is used to calculate the temperature coefficient. The mesh of the system includes gear shaft, gear body, and contact tooth with one adjacent tooth in both sides. The eight node element, which is used for the elastic deflection, is also used here for the bulk temperature. However, the boundary conditions are different. In the temperature analysis all surfaces are subject to heat convection with different heat-transfer coefficients.

The heat-transfer coefficient h_j is assigned to the contacting tooth face which is oil-jet cooled. The top land, bottom land, and another side of the tooth surface, which are not cooled by the oil jet, have a heat-transfer coefficient h_t for air or air/oil mist. Since only three teeth are made in the model, there is a surface region A (Figure 20) that covers the surfaces where the teeth are taken off and the bottom land which is in between these teeth. The heat-transfer coefficient at this region A is given value h_j, which is the same as that of the coefficient for the surface cooled by the oil jet. The reason is that, because there is an oil-jet-cooled surface on each tooth, most of the heat will flow out of the tooth from this surface ($h_j >> h_t$). All the other convective surfaces of the gear system are given a coefficient h_s (Figure 20). The theoretically estimated values of h_s and h_j can be found in references 12 and 13. However, the estimated values of h_s, h_j, and h_t, based on the experimental results[11], are used in this study.

There are 30 nodes created in the contacting surface. A unit heat flux is applied to the grid node i. The temperature distribution in this surface due to the heat flux is T_{ji}, which is the temperature at the grid node j due to the heat flux at node i. By the interpolation method, the temperature at the contact point m, due to the unit heat flux at the contact point n (T^c_{mn}) can be obtained in terms of T_{ji}. Once the contact path is located and the heat flux flowing into each body at each contact point is calculated, the bulk temperature at the contact point m can be found as

$$T_{m,B} = \sum T^c_{mn} Q_n \qquad n = 1, \ldots, k_{\max}$$

where k_{max} is the total number of contact points along the contact path.

The heat generation term Q_n is based on the recent traction models developed for EHL contacts by Johnson and Tevaarwerk[14], Bair and Winer[15], and Dyson[16]. All three models are incorporated as subroutines in the bulk temperature calculation. Because there is a lack of the rheological constants for gear oils in the Johnson and Tevaarwerk's model and Bair and Winer's model, the limiting shear stress formula developed by Dyson for mineral oils in general is used first to obtain some preliminary results for the bulk temperature.

Flash Temperature

During meshing each tooth face experiences a sudden temperatue increase (flash temperature) due to the frictional heat developed at the contact moving along the tooth face. This temperature rise is restricted in the instantaneous contact area and disappears very rapidly as soon as this instantaneous area of tooth face is out of contact. Usually, this

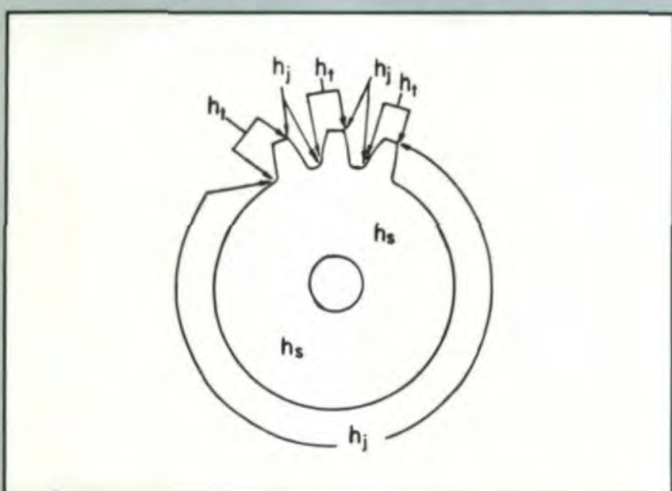


Fig. 20 – Convective heat transfer coefficients for a three-tooth model for a spiral-bevel gear or pinion.

temperature is very high and is a contributor to the gear scuffing failure.

The first successful prediction of flash temperature, introduced by Blok[17], is based on the heat-conduction analysis in a semi-infinite body with a uniformly distributed moving heat source. Jaeger[18] solved the problem of a moving source of heat with variable heat source and variable velocity. Archard introduced a simple harmonic mean to obtain the interface temperature. A refined solution including a local heat partition coefficient between a pair of disks was derived by Cameron, et al.[19]. More recently, Francis[20] made a further refinement in Blok's calculation by considering a variable heat flux in the contact.

Archard showed that when the Peclet number, vR/a, is greater than 10, the heat flow in the direction perpendicular to sliding may be neglected. The temperature distribution within a heat source of finite area can be determined by dividing the whole contact area into differential strips parallel to the sliding direction. And the temperature profile along each strip is the same as that of an infinitely long band source (in the perpendicular direction to sliding) of width equal to the strip length and has the same heat flux profile along the strip.

For the division of heat between the two contacting surfaces, an average heat partition factor is used throughout the entire contact region. The average heat partition factor is determined by a method suggested by Francis[20] for a thin film with heat generated at the midfilm. Once the average partition factor is known, the flash temperature within each strip is calculated by the method suggested by Cameron, et al.,[19] assuming a uniform heat source within the strip. Details of this procedure are documented.[2]

Results of Lubrication Performance

The same set of gears used for the dynamic load calculation are used here to demonstrate calculation of lubrication performance. Results were obtained for a range of operating conditions to determine the effect of speed, load, lubricant viscosity, and ambient temperature on the film thickness, bulk temperature, and flash temperature. A sample of results is presented here. More complete results can be found in Reference 2.

The sliding velocity decreases from the beginning of the contact path where the gear tip contacts the pinion root, until the contact point is near the pitch point where the sliding velocity becomes zero. Then the sliding velocity increases all the way to the end of the contact path where the pinion tip contacts the gear root. The current set of gears has the feature that the sliding velocity at the end of the contact path is larger than that at the beginning of the contact path; this fact creates a situation that more heat is generated at the end of the contact path.

A typical distribution of bulk temperature along the contact position is shown in Figure 21. The bulk temperature of the pinion is always larger than that of the gear because the pinion speed is three times faster than the gear speed and receives more heat per unit time than the gear does. Although the temperature coefficients are higher near the gear tip, the maximum bulk temperature of both gears occurs at the end

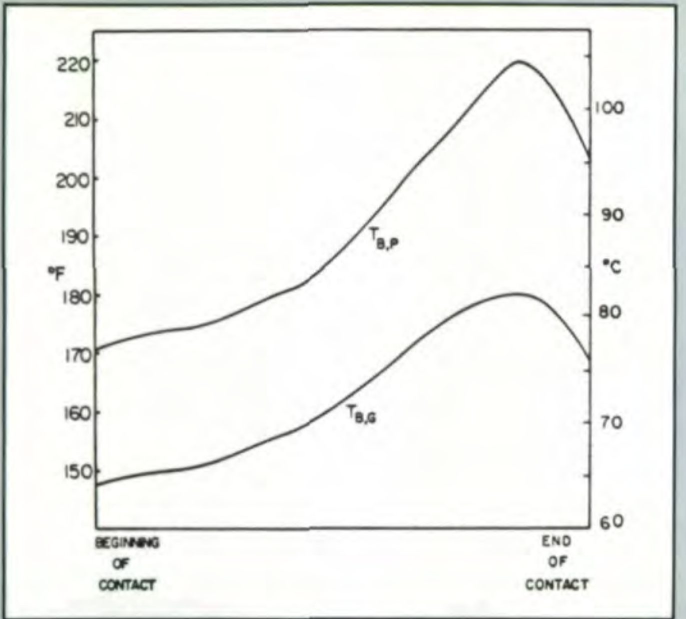


Fig. 21 – Typical bulk temperature distribution for pinion and gear. Contact position ω_g, 523 rad/sec.

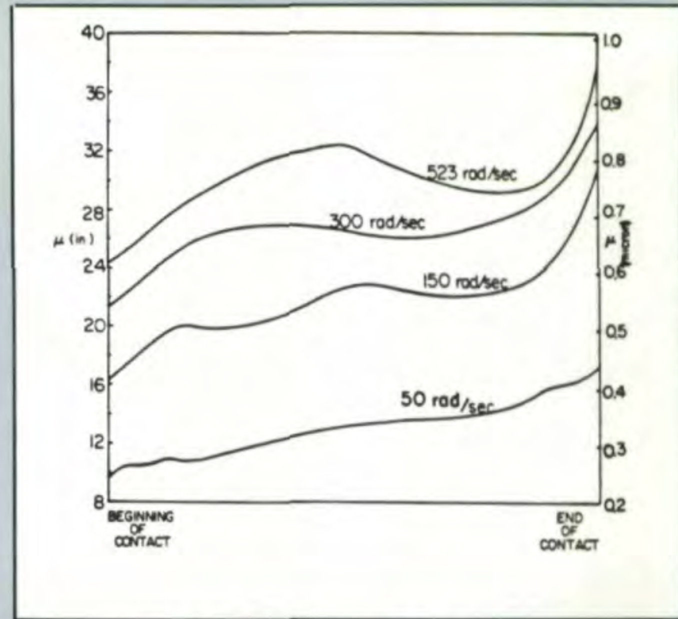


Fig. 23 – Film thickness distributions.

of the single-tooth contact point where the maximum heat is generated. A distribution of the total flash temperature for the same case is plotted in Figure 22. The minimum flash temperature occurs at the pitch point where the sliding velocity is zero. For this high-speed case the variation of dynamic load is less pronounced along the contact path. The rise of flash temperature on both sides of the pitch corresponds directly to the variation of sliding speed at the contact. The slight decrease at the end of the contact path is attributable to the decrease in dynamic load in this region.

Figure 23 shows the distributions of film thickness for four different gear speeds. No excessive variations are seen along the contacting path. A moderate peak is evident at the pitch point for the high-speed cases, and this is associated with the slight drop of bulk temperature at the pitch point. The steady rise of film thickness along the contact path is due to the in-

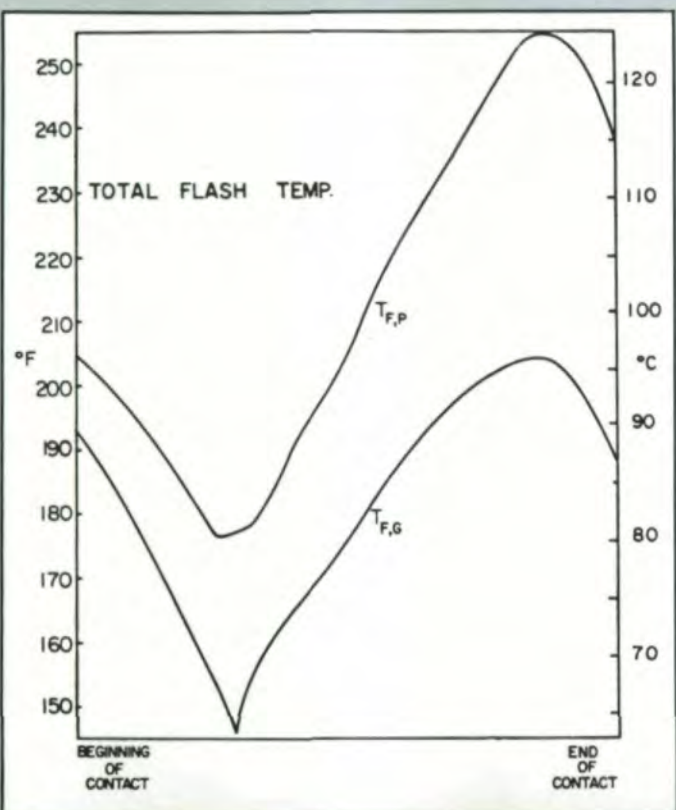


Fig. 22 – Typical total flash temperature distribution for pinion and gear. Contact position, ω_g, 523 rad/sec.

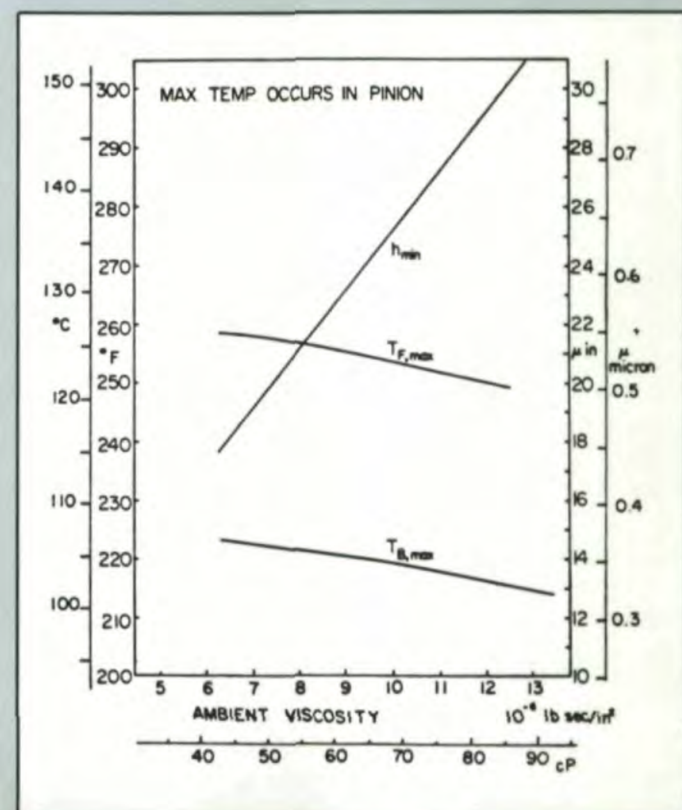


Fig. 24 – Effect of lubricant viscosity on lubrication performance.

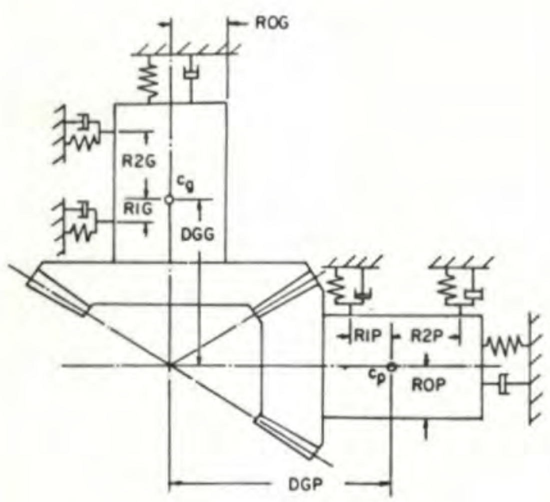


Fig. 25 – Labels for distances between pinion and gear, mass centers to bearing supports.

crease in the entrainment velocity. The final uptrend of film thickness near the end of contact is again due to the decrease in bulk temperature.

Finally, the effect of increase in ambient viscosity on the minimum film thickness, h_{min}, maximum bulk temperature as well as total flash temperature is demonstrated in Figure 24. As expected, an increase in viscosity would improve lubrication performance with a much thicker film thickness and a slight drop in both bulk and flash temperature.

Concluding Remarks

A computer solution to the dynamic load in a pair of spiral-bevel gearsets was developed by solving the equations of motion for the pinion and gear shaft. An existing finite-element code was used to calculate the combined stiffness of the contacting pinion and gear teeth as a function of contacting position in the zone of action. In addition to the dynamic load analysis, a computer solution was also developed to predict the bulk surface temperature, the flash temperature, and the film thickness along the contact path. An existing finite-element heat code was also used to calculate the temperature influence coefficients from which the bulk surface temperature is calculated. Both the lubricant film thickness and the sliding traction are calculated from the recent findings in EHL theories.

Results were obtained for a set of experimental spiral-bevel gear currently being tested at the NASA Lewis Research Center. The results of dynamic load tests show that there exist numerous peaks in the variation of dynamic load against the gear shaft speed. These fluctuations correspond reasonably well with the critical frequencies of the system. The envelope of the peaks suggests that the highest dynamic load occurs somewhere near the critical frequency corresponding to the rotational mode oscillations of the two gears.

Results of the film thickness show that its variation along the contact path is not large and that it is caused mainly by the increase in the entrainment velocity and the change in bulk surface temperature. The total flash temperature variation is controlled by the sliding velocity and has its maximum near the end of the contact path where the transition from double to single mesh occurs. Effects of operating variables on the minimum film thickness and maximum surface temperatures along the contact path can also be obtained readily with this program. Results for the effect of ambient viscosity show trends consistent with those anticipated from exiting EHL theories.

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(continued on page 23)

Controlling Tooth Loads In Helical Gears

Eliot K. Buckingham
Buckingham Associates Inc.
Gear Consultants
Springfield, Vt.

Helical gears can drive either nonparallel or parallel shafts. When these gears are used with nonparallel shafts, the contact is a point, and the design and manufacturing requirements are less critical than for gears driving parallel shafts. With parallel shafts, the theoretical contact is a line (at an angle to the gear axis) that sweeps across the face of the teeth. Most problems with parallel-shaft helical gears can be traced to the fact that this theoretical contact is difficult to produce.

As with any gear drive, the performance and capacity of helical gears depend on the nature and extent of the contact between mating gear teeth. With helical teeth, and a wide enough face, multitooth contact is possible, but the load will not be shared equally among the teeth. Factors that prevent equal load distribution include: 1. Form, spacing, and lead errors. 2. Elastic deformation of the teeth and blanks under load. 3. Thermal distortion of the gears, shafts, and housing.

Despite these problems, properly designed and manufactured helical gears make effective drives. Good gear operation is a function of the action between mating teeth (approach or recess) and how well the gear leads match. Gear action is determined by geometry and can be controlled by design. Lead accuracy, on the other hand, is restricted by the quality of the gear generating machinery and cannot be controlled directly. But lead accuracy can be controlled indirectly by properly specifying the lead and by ensuring the gears are produced on "exact-ratio" cutting machines.

Cutting Machine Limitations

Helical gears are generated by shaping or hobbing, and they are finished by shaving, grinding, or honing. Both generating mnethods have disadvantages in that they introduce forming errors that are difficult to correct with a finishing operation. Close monitoring while cutting is the only way to minimize these errors.

AUTHOR:

MR. ELIOT BUCKINGHAM *is President of Buckingham Associates, Inc., a gear consulting firm. He received his B.S. from the Massachusetts Institute of Technology and his M.S. from the University of New Mexico. During his career he has worked as an Engineer and Supervisor for several companies and he was also an instructor at the University of Mexico. He has been a gear consultant since 1960. Mr. Buckingham is the author of numerous articles on gearing. A member of ASME, he is also a Registered Professional Engineer in the State of Vemont.*

Shaping uses a pinion shaped cutter in conjunction with a helical guide to produce the helix. But, this technique leaves tool marks (perpendicular to the direction of sliding) that must be removed by a finishing operation. The only factor controlling lead accuracy is the accuracy of the guide.

Hobbing, on the other hand, uses an indexing drive train to produce the helix. Most helical gears are hobbed because new shaping guides do not have to be built for each new gear design. With this technique, each stage of manufacturing must be closely monitored to ensure quality gears. Errors in tooth form, spacing, or lead produced during rough-cutting may remain after final grinding. To assume that grinding corrects all hobbing errors is poor practice.

Gear generating equipment that may be adequate to hob other gear forms usually is not adequate for helical gears. The nature of the action in the other forms tends to correct errors introduced by the generating equipment. This correction results from cold-working of the contact surfaces. But, because of the cross-sliding action in helical gears, cold-working occurs to a much lesser degree, if at all.

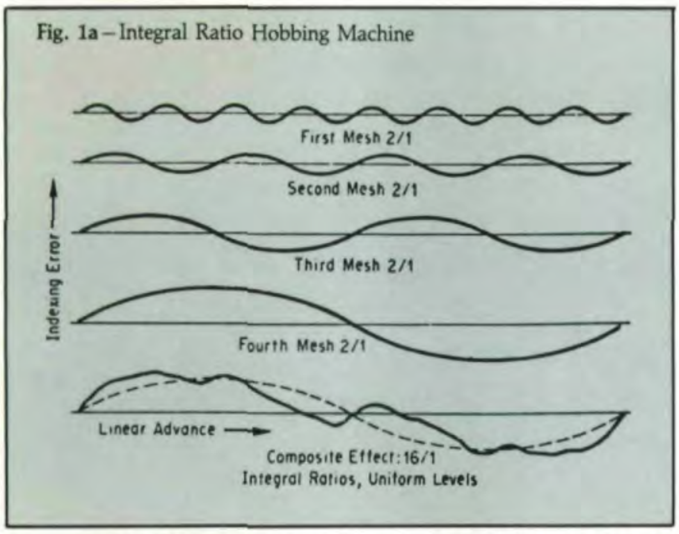
For example, a spur gear may be "free-wheeled" on a grinder with a threaded grinding wheel. (Free-wheeling means the drive train to work the spindle is disconnected, and the wheel drives the work.) If this method is handled properly, form and spacing errors on a spur gear can be reduced from one-half to one-third the errors produced when the work is driven through the indexing gear train. Unfortunately, free-wheeling does not work on helical gears.

Controlling Hobbing Machine Errors

In a hobbing machine, the positional or indexing error of the output shaft is a composite effect of the indexing errors of all the gear pairs in the machine drive. For example, Fig. 1a shows the composite effect of a hobbing index drive where each gear pair is an exact ratio. If the indexing error is the same for each pair, the total error has an amplitude greater than that of any of the individual errors, but less than the sum total of all the errors. The curves close at the end of each revolution of the slow-speed shaft, so the total indexing error is fixed and repeats for each revolution.

But, for the nonintegral (or hunting-tooth) ratio, Fig. 1b, the maximum indexing error is greater, and the curves do not close at the end of each revolution. Thus, the indexing error is constantly changing.

Hobbing machines with long gear trains or differentials are hunting-tooth drives. When these drives produce a helical gear, the indexing error causes "drunkenness" (nonuniformi-



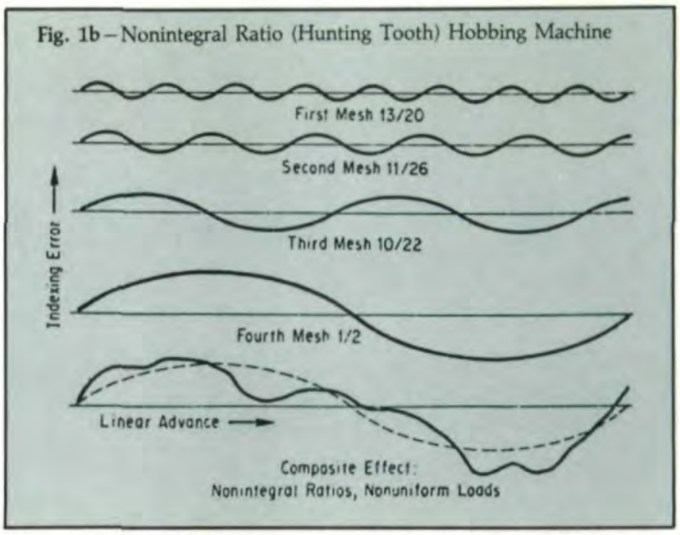


Fig. 1 – Hobbing machine indexing errors. With short-gear train, integral ratio drives, *a*, hobbing machines generate more uniform, better matched helical gears. Indexing error is uniform and repeats for every revolution of the slow-speed shaft. With nonuniform, hunting-tooth drives, *b*, indexing error varies for each revolution, producing less accurate gears.

ty) in the lead. For example, compare the lead profiles in Fig. 2. The first gear was produced with no effort to ensure a short gear train or integral ratios, and the second gear was produced on a machine using exact ratios. The second gear has a more uniform lead. Therefore, to minimize lead errors, use a cutting machine with an exact-ratio indexing drive.

Specify Lead

The lead, advance of the helix in one revolution, is one of the most important design parameters of a helical gear. For helical gears to contact properly, the leads must be exactly proportional to the number of teeth; that is, $L_1/L_2 = N_1/N_2$, where L_1 and L_2 = gear leads and N_1 and N_2 =

Table 1—Conventional Design vs. Exact-Lead Designs

Design Parameter	Conventional Design	Exact Leads Standard Form	Exact Leads Full Recess
Number of Driver Teeth	60	60	60
Number of Follower Teeth	120	120	120
Hob Pressure Angle (°)	*20*	*20*	*20*
Hob Diametral Pitch	5	5	5
Operating Center Distance (in.)	18.75	18.75	13.75
Face Width (in.)	3.54	3.54	3.54
Helix Angle (°)	16.26020470	16.5946787	16.5946787
Driver Lead	134.6396851	132.0	132.0
Follower Lead	269.2793703	264.0	264.0
Driver OD (in.)	12.90	12.90	13.30
Driver Pitch Diam (in.)	12.50	12.50	12.50
Follower OD (in.)	25.40	25.40	25.00
Follower Pitch Diam (in.)	25.00	25.00	25.00
Contact Ratio			
Plane of Rotation	1.715	1.610	1.452
Face Contact	1.577	1.609	1.609
Total	3.292	3.219	3.061

By specifying exact leads, helical gears can be produced more easily on conventional hobbing machines. Exact leads can be generated and measured more readily than leads calculated to eight places by a machinist. And a drive with exact leads has nearly the same helix angle and contact ratio as conventional designs.

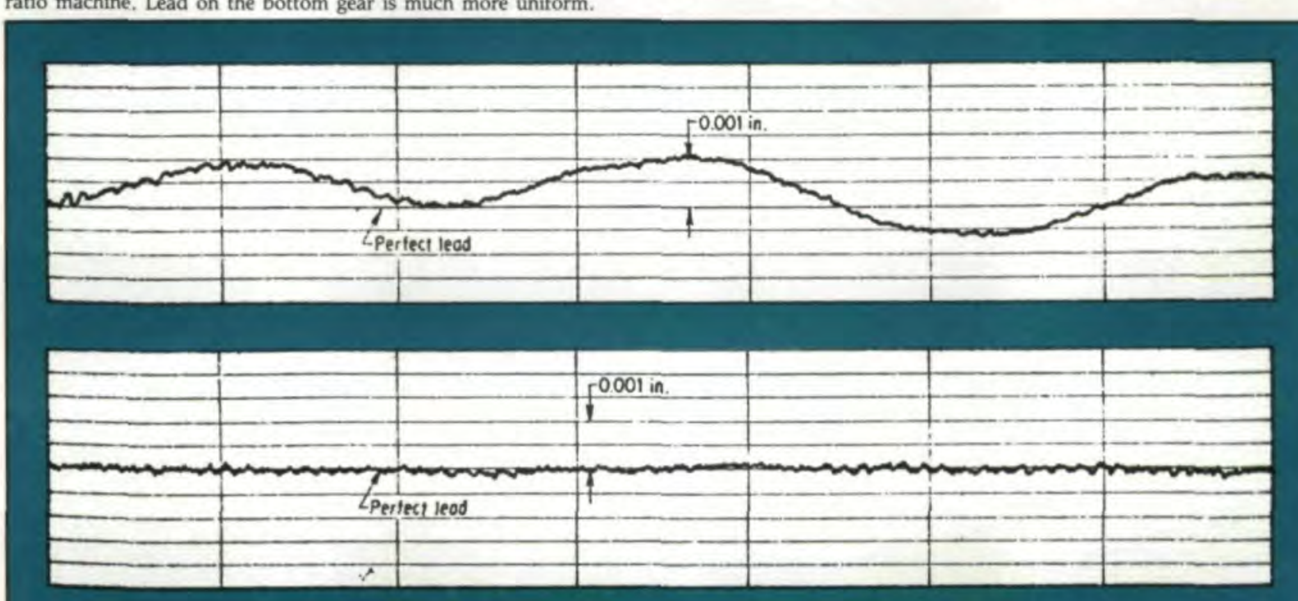
number of gear teeth. If this relationship is not maintained, then the two helixes will not be parallel, resulting in unequal contact across the face and high unit loading.

For shaped gears, the helix and lead are controlled by the mechanical guide. Provided the guide is accurate, this system produces leads that are proportional to the number of teeth. But most helical gears are hobbed, the helix angle is specified, and the lead often is not even listed on the print.

Helix angle cannot be measured by conventional equipment and cannot be set on the hobbing machine. The lead, however, can be measured easily and also can be set on the equipment. But, if the lead is not specified, the machinist must calculate it from $L = \pi N / P_{nc} \sin \psi$, where P_{nc} = hob diametral pitch and ψ_1 = helix angle. To ensure proper tooth contact, L should be accurate to eight places. But, unless L is a simple number (an integer, common decimal, or ratio of two integers less than 100), it cannot be manufactured or measured accurately to eight places. Therefore, leads should always be exact numbers.

Table 1 compares a set of helical gears cut by the conventional hobbing method (of simply specifying helix angle) with two gear sets designed to have exact leads. Note the small difference between the leads and helix angles. However, the

Fig. 2—Helical gear lead profiles. The top gear was hobbed on a nonintegral, hunting-tooth ratio machine, and the bottom gear on an integral-ratio machine. Lead on the bottom gear is much more uniform.



(continued on page 48)

Design and Selection of Hobs

Robert P. Phillips
Barber-Colman Co.
Specialty Tools Division
Rockford, IL

Introduction

The following is a general overview of some of the different factors that lead to the specific design and the selection of the correct tool for a given hobbing application. There will be three main sections investigated:

A. Basic review of the generating process as it relates to gear hobbing, including general nomenclature of hob terms, and specific tooth modifications.
B. Hob error as it relates to gear accuracy.
C. Optimization of hob design.

Hob Nomenclature

The terms in Fig. 1, along with additional terms and their definitions follow.

Nomenclature of Hob Elements and Other Terms Relating to Hobbing

adjacent flute spacing – The variation from the desired angle between adjacent tooth faces measured in the plane of rotation.

adjacent thread spacing – The difference in the average variations obtained by traversing along the desired helical path of one thread, indexing and traversing in a similar manner on an adjacent thread.

approach – See preferred term **tip relief modification.**

approach distance – The linear distance in the direction of feed between the point of initial hob contact and the point of full hob contact.

arbor collar – A hollow cylinder which fits an arbor, and is used to position the hob.

auxiliary leads – A feature employed on some hobs, especially worm gear hobs, wherein both sides of the hob thread have leads different from the nominal hob lead; one side longer, the other side shorter. This results in the tooth thickness being successively less toward the roughing end of the hob.

axial feed – The rate of change of hob position parallel to the workpiece axis usually specified in inches per revolution of the workpiece.

axial plane – A plane containing the axis of rotation.

axial pressure angle – See definition under **pressure angle.**

back-off – See preferred term **cam relief**, under **relief.**

cam – The radial drop of the form in the angular distance between adjacent tooth faces.

centering device – A ground locating pin used to center a tooth or space of the hob on the centerline of the workpiece.

chamfer – A beveled surface to eliminate an otherwise sharp corner.

climb hobbing – Rotation of a hob in the opposite direction to the feed of the hob relative to the workpiece at the point of contact.

clutch keyway – See term **keyway.**

common factor ratio – In multiple thread hobs, the condition wherein the **gear tooth-hob thread ratio** is not a whole number, but there is a common factor of the number of gear teeth and the number of hob threads.

conventional hobbing – Rotation of a hob in the same direction as the feed of the hob relative to the workpiece at the point of contact.

cutting face width – The axial length of the relieved portion of the hob.

cutting speed – The peripheral lineal speed resulting from rotation, usually expressed as surface feet per minute. [sfm]

depth of cut – The radial depth to which the hob is sunk into the workpiece.

drawbar – A rod which retains the arbor, adapter or hob shank in the spindle.

even ratio – In multiple thread hobs, the condition wherein the **gear tooth-hob thread ratio** is a whole number.

feed – The rate of change of hob position while cutting.

fillet – 1 – A curved line joining two lines to eliminate a sharp internal corner. – 2 – A curved surface joining two surfaces

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AUTHOR:

MR. ROBERT PHILLIPS is an Engineering Supervisor at the Barber-Colman Company where he supervises the design engineering of their hob product line. He holds a BS in Mechanical Engineering from the University of Wisconsin in Madison. He is a Certified Professional Engineer and has served as a SME Chapter Committeeman.

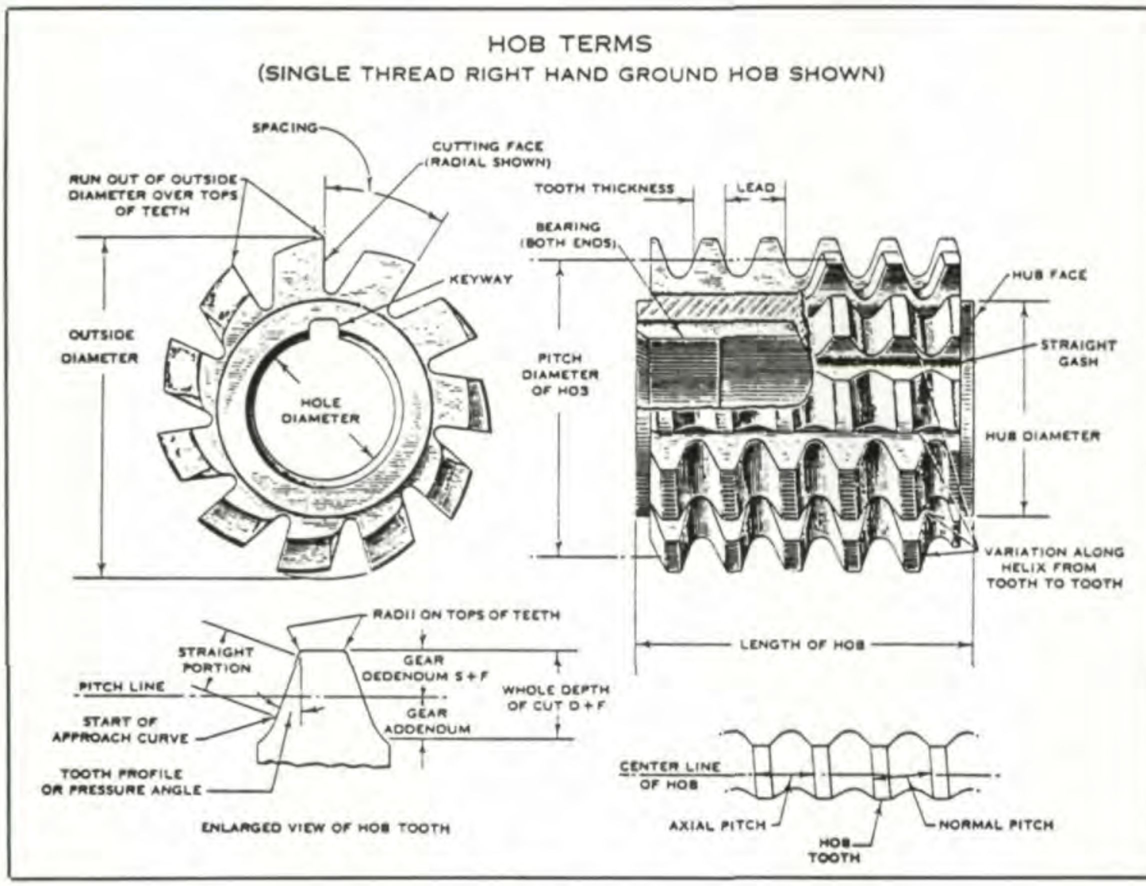


Fig. 1

to eliminate a sharp internal corner.

flute – A longitudinal groove either straight or helical that forms the tooth face of one row of hob teeth and the backs of the preceeding row. It also provides chip space.

flute helix angle – The angle which a helical tooth face makes with an axial plane, measured on the hob pitch cylinder.

flute lead – The axial advance of helical tooth face in one turn around the axis of a hob.

flute lead variation – The deviation of a hob tooth face from the desired helical surface.

former – See preferred term **sharpening guide.**

full top radius – Continuous radius tangent to top and side cutting edges.

gear tooth-hob thread ratio – The ratio of the number of teeth in the workpiece to the number of threads in the hob.

generated fillet – At the bottom of the hobbed form a fillet joining the root diameter with the desired generated form. This fillet is not a true radius.

generated fillet height – On the hobbed workpiece, the radial distance from the root diameter to the point where the generated fillet joins the desired generated form.

grinding cracks – Fractures in the hob caused by improper grinding techniques in sharpening.

high point – See preferred term **protuberance.**

hob addendum – Radial distance between the top of the hob tooth and the **pitch cylinder.** Do not confuse with gear addendum.

hob arbor – A device to mount in or on the spindle of a hobbing machine, which is designed to carry and drive an arbor-type hob.

hob dedendum – In **topping hobs,** the radial distance between the bottom of the hob tooth profile and the **pitch cylinder.** Do not confuse with gear dedendum.

hob length – Overall length of hob.

hob runout – The runout of hob when mounted in hobbing machine, measured radially on hub diameter, and axially on hub face.

hob shift – The axial movement of a hob along its axis to engage a different section with the workpiece.

hub – A qualifying surface at each end of an arbor type hob which is provided for checking diameter and face runout.

hub diameter runout – The total variation in radial distance of the hub periphery from the axis.

hub face – The side surface of the **hub.**

hub face runout – The total axial variation of the hub face from a true plane of rotation.

hunting ratio – See preferred term **prime ratio.**

infeed – The radial rate of change of hob position, relative to the workpiece axis, usually specified in inches per revolution of the workpiece.

key – A mechanical member through which the turning force is transmitted to the hob.

keyseat – The pocket, usually in the driving element, in which the key is retained.

keyway – A slot through which the turning force is transmitted to the hob. May be either a longitudinal slot through the hole or a transverse slot across the hub face. If the latter, it is called a **clutch keyway.**

lead – The axial advance of a thread for one complete turn, or convolution.

lead angle – The angle between any helix and a plane of rotation. In a hob, **lead angle** usually refers specifically to the angle of thread helix measured on the **pitch cylinder**.

lead variation – The axial deviation of the hob teeth from the correct thread lead.

leader – See preferred term **sharpening guide.**

linear pitch – See preferred term **axial pitch**, under **pitch.**

linear pressure angle – See preferred term **axial pressure angle,** under **pressure angle.**

lug – An extension of hob tooth profile above the nominal top cutting edge. Sometimes called **spurs** or **prongs.**

non-adjacent flute spacing – The variation from the desired angle between any two non-adjacent tooth faces measured in the plane of rotation.

normal circular pitch – See definition under **pitch.**

normal diametral pitch – See definition under **pitch.**

normal plane – A plane perpendicular to a pitch cylinder helix.

normal pressure angle – See definition under **pressure angle.**

number of threads – In multiple thread hobs, the number of parallel helical paths along which hob teeth are arranged, sometimes referred to as **number of starts.** Should not be confused with the term, **number of threads per inch,** which is commonly used in designating the axial pitch of screw threads.

offset – See preferred term **rake offset.**

outside diameter – The diameter of the cylinder which contains the tops of the cutting edges of the hob teeth.

outside diameter runout – The total variation in the radial distance from the axis to the tops of the hob teeth.

overtravel – The linear distance in the direction of feed of the hob beyond the last point of contact of the hob with the workpiece.

pilot end – On shank type hobs, a cylindrical or conical bearing surface opposite the driving end.

pin measurement – The measurement taken over pins of equal diameter placed in specified tooth spaces in the workpiece.

pitch – The distance between corresponding, equally spaced hob **thread** elements along a given line or curve. The use of the single word **pitch** without qualification may be confusing. Specific terms such as **normal diametral pitch, normal circular pitch,** or **axial pitch** are preferred.

Axial Pitch – The pitch parallel to the axis in an axial plane between corresponding elements of adjacent hob thread sections. The term *Axial Pitch* is preferred to the term *Linear Pitch.*

Circular Pitch – The distance along the pitch cylinder between corresponding elements of adjacent hob thread sections.

Linear Pitch – See preferred term *Axial Pitch.*

Normal Circular Pitch – The distance between corresponding elements on adjacent hob thread sections measured along a helix that is normal to the *Thread Helix* in the *Pitch Cylinder.*

Normal Diametral Pitch – π [3.1416] divided by the *Normal Circular Pitch.*

pitch circle – A transverse section of the hob **pitch cylinder.**

pitch cylinder – A reference cylinder in a hob from which design elements, such as lead, lead angle, profile, and tooth thickness are derived.

pitch diameter – The diameter of the **pitch cylinder.**

pitch point – The point at which a **tooth profile** intersects the **pitch cylinder.**

pressure angle – The angle between a tooth profile and a line perpendicular to the **pitch cylinder** at the **pitch point.** In hobs, the **pressure angle** is usually specified in the **normal plane** or in the **axial plane.**

Axial Pressure Angle – The *Pressure Angle* as measured in an *Axial Plane.* The term *Axial Pressure Angle* is preferred to the term *Linear Pressure Angle.*

Normal Pressure Angle – The *Pressure Angle* as measured in a *Normal Plane.*

prime ratio – In multiple thread hobs, the condition wherein the **gear tooth-hob thread ratio** is not a whole number and there is no common factor of the number of gear teeth and the number of hob threads.

protuberance – A modification near the top of the hob tooth which produces **undercut** at the bottom of the tooth of the workpiece.

rake – The angular relationship between the tooth face and a radial line intersecting the tooth face at the hob outside diameter measured in a plane perpendicular to the axis.

Negative Rake – The condition wherein the peripheral cutting edge lags the tooth face in rotation.

Positive Rake – The condition wherein the peripheral cutting edge leads the tooth face in rotation.

Zero Rake – The condition wherein the tooth face coincides with a radial line.

rake offset – The distance between the tooth face and a radial line parallel to the tooth face. Used for checking rake.

ramp – A modification at the bottom of the hob tooth which produces a chamfer at the top corners of the tooth of the workpiece.

relief – The result of the removal of tool material behind or adjacent to a cutting edge provide clearance and prevent rubbing [heel drag].

Cam Relief – The relief from the cutting edges to the back of the tooth produced by a cam actuated cutting tool or grinding wheel on a relieving [back-off] machine.

Side Relief – The relief provided at the sides of the teeth behind the cutting edges. The amount depends upon the radial cam, the axial cam, and the nature of the tooth profile.

scallops – The shallow depression on the generated form produced by hob action.

setting angle – The angle used for setting hob swivel to align the hob thread with the workpiece teeth.

shank – That projecting portion of a hob which locates and drives the hob in the machine spindle or adapter.

sharpening allowance – The amount by which the pitch diameter of a worm gear hob exceeds that of the worm, to allow for the reduction in diameter by sharpening.

sharpening guide – A cylindrical part with flutes, having the same lead as the hob flutes, used for guiding the hob along the correct lead when sharpening.

short lead – A feature employed on some hobs to obtain generated fillet or undercut conditions not obtainable with nominal lead.

side relief – See definition under **relief.**

stock allowance – The modification of the hob tooth to leave material on the workpiece tooth form for subsequent finishing.

tangential feed – The rate of change of hob position along its own axis, usually specified in inches per revolution of the workpiece.

thread – A helical ridge, generally of constant form or profile. In a hob, unlike a worm or screw, the thread is not continuous and exists only at the cutting edges of the hob teeth. Therefore, it is sometimes referred to as the **thread envelope.**

thread envelope – See preferred term **thread.**

thread helix – The helix of the hob thread in the **pitch cylinder.**

tip relief – A modification in which a small amount of material is removed from the basic profile near the tip of the gear tooth.

tip relief modification – A modification on the sides of the hob tooth near the bottom which produces a small amount of **tip relief.** Such modification is usually incorporated in **finishing hobs** except in the finer pitches.

tooth – A projection on a hob which carries a cutting edge.

tooth face – The tooth surface against which the chips impinge.

tooth profile – Outline or contour of hob tooth cutting edges.

tooth thickness – The actual width or thickness of the hob thread at the **pitch cylinder.** The use of the single term **tooth thickness** without qualification may be confusing. The specific terms **normal tooth thickness** and **axial tooth thickness** are preferred.

Axial Tooth Thickness – The tooth thickness as measured in an axial plane.

Normal Tooth Thickness – The tooth thickness as measured along a helix normal to the thread helix.

top radius – Radius of the arc joining the top and a side cutting edge of a hob tooth.

total indicator reading [tir] – See preferred term **total indicator variation.**

total indicator variation [tiv] – The difference between maximum and minimum indicator readings during a checking cycle.

undercut – The condition at the base of a hobbed workpiece form wherein additional material beyond the basic form is removed. Under certain conditions this may occur naturally, while in other cases it may be produced by intentional modification of the hob tooth.

wear land – A cylindrical or flat land worn on the relieved portion of the hob tooth behind the cutting edge.

Fig. 2

wobble – The motion of a hob when the radial runout varies along the hob length.

worm gear hob oversize – See preferred term **sharpening allowance.**

For varying specific reasons, it is possible to alter the straight sided rack form in order to achieve a modified generated gear tooth form.

The hob in Fig. 2 was designed to eliminate sharp corners at the tops of gear teeth. It can be made to produce a desired amount of chamfer or radius, but to do this the number of teeth in the gear must be known. Obviously, the form of such a modification produced by a given hob will vary with the number of teeth in the gear, just as the width of the top of the gear tooth varies.

The topping gear hob shown in Fig. 3 is used for cutting spur gears and helical gears. This hob finishes the tops of gear teeth, holding the outside diameter of the gear to a given dimension in relation to the pitch line and root diameter.

Several advantages accompany the use of topping hobs that in many cases result in material savings. For instance, the finish-hobbed gears may be chucked on the outside diameter in subsequent operations for hole finishing when necessary. Moreover, their use eliminates an accurate finish-turning operation on gear blanks before hobbing. Additionally, gears hobbed with topping hobs may be quickly inspected for pitch line thickness to ordinary commercial limits by measuring the outside diameter with a micrometer.

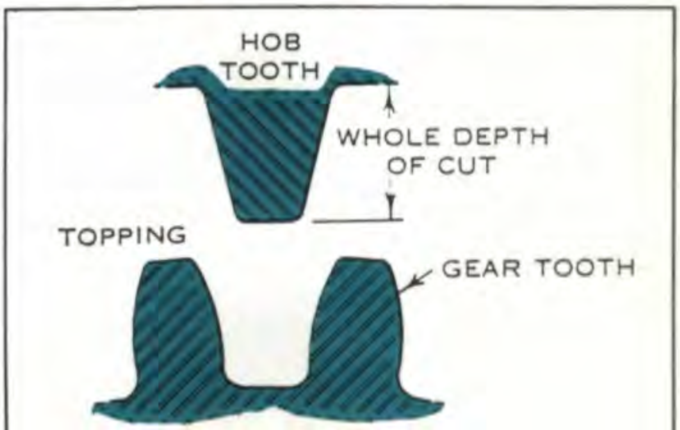


Fig. 3

Gears are frequently hobbed and then finished by shaving or grinding. Hobs for producing such gears are referred to as semi-finishing hobs. The hobbing of such gears has not been considered a roughing operation because of the accuracy required for leaving a minimum and uniform amount of finishing stock. (Fig. 4)

The protuberance type hob shown in Fig. 5 generates undercut at the bottom of the gear tooth to provide clearance for the shaving cutter and to prevent the formation of an abrupt change in profile with its resulting stress concentration. With small numbers of teeth, the tooth form cut with a hob without protuberance is often undercut enough, but a protuberance is required for larger numbers of teeth to eliminate contact between the tip of the shaving cutter and the fillet on the gear tooth.

Hob Error vs. Gear Accuracy

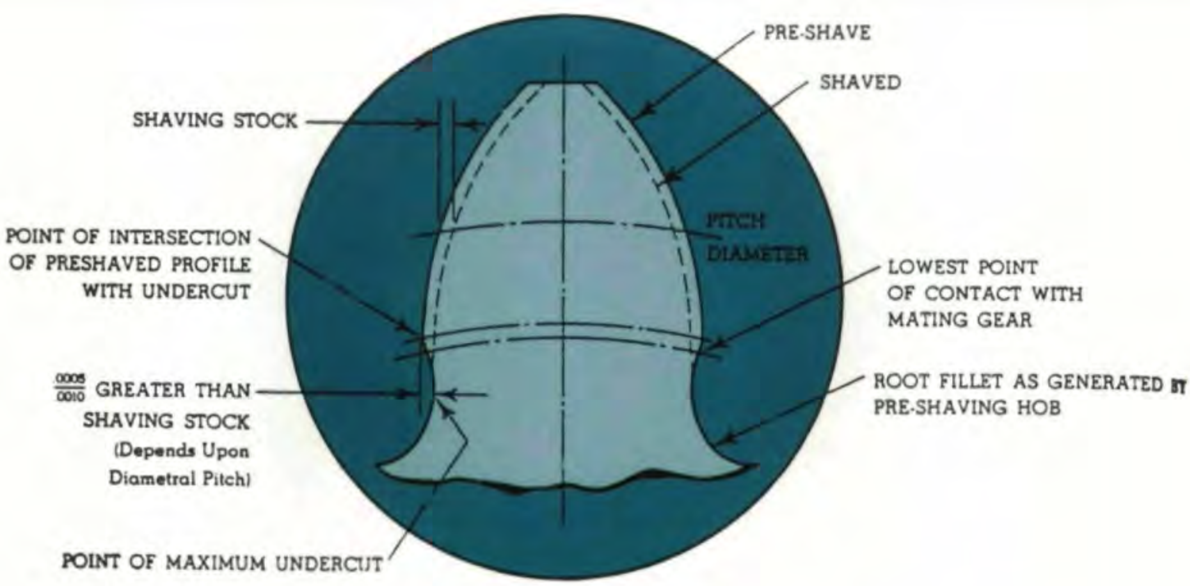
All hob manufacturers work to a given set of standard tolerances. A reprint of tolerances from the Metal Cutting Tool Institute Standards are as follows. It should be noted, these tolerances pertain to standard outside diameter sizes only. (Table of standard hob sizes – see pages 40-43) Tolerances for lead on oversized hobs can be increased proportionally to the diameter increase.

Tolerance Definitions

Hole

diameter – The basic diameter of the hole in the hob.

tolerance – The amount that a hole may be oversize from the basic diameter of the hole.



Comparison between hobbed and shaved profiles.

Fig. 4

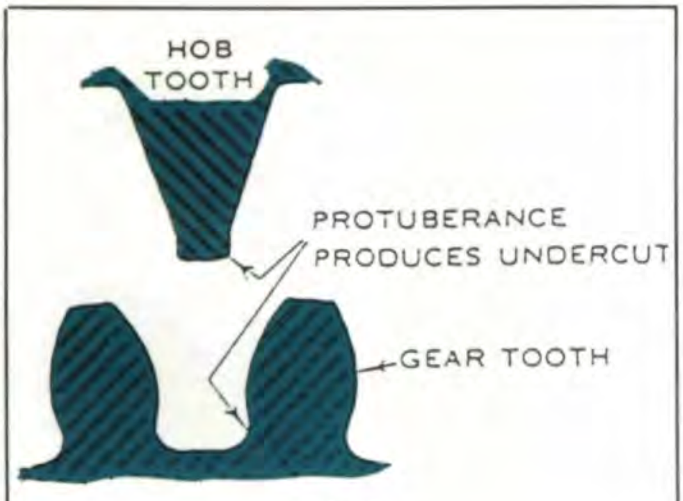


Fig. 5

bearing contact – The area of contact obtained using a plug gage that makes contact over the full length of the hob.

Runout

hub face – The total indicator variation on the end face in one revolution of the hob.

hub diameter – The total indicator variation on the hub diameter in one revolution of the hob.

outside diameter – The total indicator variation on the tops of the hob teeth in one revolution of the hob.

Sharpening

spacing between adjacent flutes – The total indicator variation obtained between two successive flutes when a hob is indexed.

spacing between non-adjacent flutes – The total indicator variation obtained between any two flutes when the hob is indexed through one complete revolution.

rake to cutting depth – The total indicator variation when traversing the tooth face from the top to the cutting depth.

flute lead – The total indicator variation when traversing the faces of all of the teeth in any one flute following the specified lead.

Lead Variation

tooth-to-tooth – The total indicator variation on successive teeth when traversing along the true helical path.

any one axial pitch – The total indicator variation in one complete revolution along the true helical path (360 degrees) on a single thread hob.

any three axial pitches – The total indicator variation in three revolutions along the true helical path (1080 degrees) on a single thread hob.

total – The total indicator variation on teeth when traversing along true helical path of all teeth in hob.

Tooth Profile

pressure angle or profile – The departure of the actual tooth profile from the correct tooth profile as denoted by total indicator variation or by magnified layout comparison.

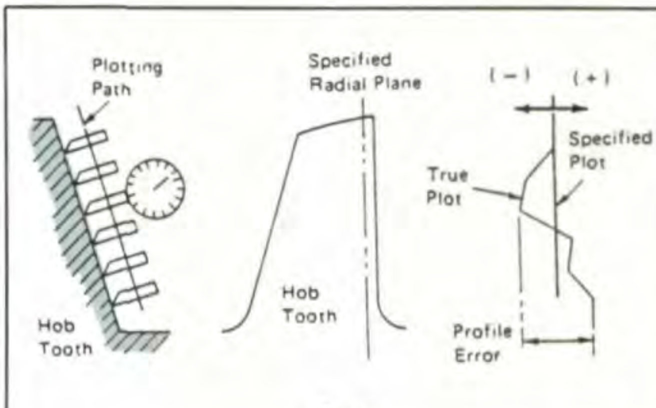


Fig. 6

Tolerances shown apply to straight side profiles in the axial or normal section.

Hobs with curved profiles are special and subject to individual consideration.

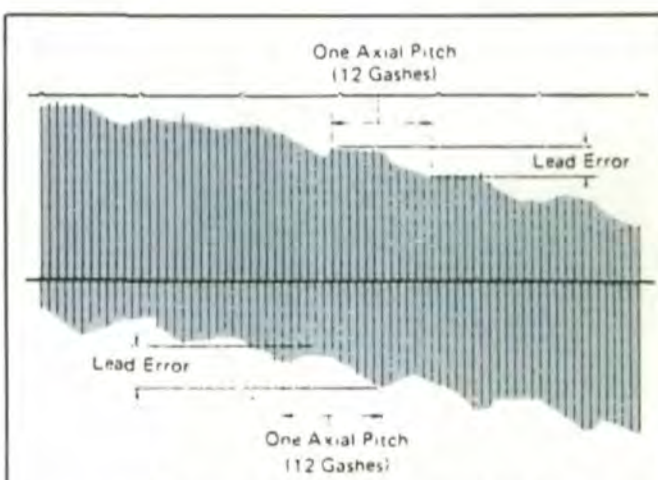
tooth thickness – The difference between the measured thickness and the specified thickness at the hob pitch cylinder.

start of tip relief modification – The tolerance permitted in locating the point on the hob tooth, plus or minus, at which a profile modification begins.

symmetry in start of tip relief modification – The radial tolerance for the start of the modification with reference to the start of the modification on the opposite hob tooth profile.

Hob profile error

The actual hob profile is allowed to vary from the specified hob profile entirely in the plus direction, entirely in the minus direction or split and divided in any ratio, provided the total deviation does not exceed the specified value. This maximum value can occur anywhere along the hob profile, and the variation of the profile on one side of the thread has no relationship to the variation on the other side of that same thread. (See Fig. 6)



Recording Lead Checker Chart for a 12 Gash, 1 Thread Hob Illustrating Lead Error in One Axial Pitch

Fig. 7

SINGLE-THREAD AND MULTITHREAD GEAR HOB TOLERANCES

(All readings in tenths of a thousandth of an inch)

DIAMETRAL PITCH		1 Thru 1.999	2 Thru 2.999	3 Thru 3.999	4 Thru 4.999	5 Thru 5.999	6 Thru 8.999	9 Thru 12.999	13 Thru 19.999	20 Thru 29.999	30 Thru 50.999	51 and Finer
RUNOUT (1-4 Thread)	**CLASS**											
	AA			2	2	2	1	1	1	1	1	1
Hub Face*	A	8	5	2	2	2	2	2	2	2	2	2
	B	10	8	4	4	3	3	2	2	2	2	
	C	10	8	4	4	3	3	2	2	2	2	2
	D	10	8	5	5	4	4	3	3	3	3	
	AA			2	2	2	1	1	1	1	1	1
Hub Diameter*	A	10	5	4	3	3	3	2	2	2	2	2
	B	12	8	6	5	4	4	3	2	2	2	
	C	12	8	6	5	4	4	3	2	2	2	2
	D	15	10	8	8	6	6	5	5	4	3	
	AA			5	4	3	3	3	3	2	2	2
Outside Diameter*	A	30	20	15	15	10	10	10	10	10	7	5
	B	10	30	25	20	15	15	15	10	10	7	
	C	50	45	40	25	20	17	17	12	12	10	8
	D	60	55	50	45	35	35	30	25	20	15	
LEAD VARIATION												
	AA			4	3	2	1.7	1.7	1.7	1.7	1.5	1.5
Tooth to Tooth*	A	7	5	4	3	2	2	2	2	2	2	2
1 Thread	B	10	8	6	4	3	3	3	3	3	2	
	C	15	12	8	6	5	4	4	4	4	3	3
	D	25	20	16	14	12	10	10	8	6	5	
	A	8	6	5	4	3	3	3	3	2	2	2
2 Thread	B	12	10	7	6	5	5	5	4	3	2	
	C	18	14	10	9	7	6	6	5	5	3	3
	D	27	22	18	16	14	12	11	9	8	6	
	A	9	7	6	4	4	4	3	3	3	2	2
3 Thread	B	14	12	8	7	6	6	5	5	4	3	
	C	21	16	12	10	8	7	6	5	5	4	3
	D	29	24	20	18	16	14	12	10	9	7	
	A	10	7	6	5	4	4	4	3	3	3	2
4 Thread	B	16	13	9	8	7	6	6	5	4	4	
	C	24	18	13	11	9	7	7	6	5	4	4
	D	31	26	22	20	18	16	13	11	10	8	
	AA			8	6	4	3	3	2	2	1.5	1.5
Any One Axial Pitch*	A	25	18	10	8	6	5	5	4	4	3	3
1 Thread	B	35	25	17	11	9	7	7	6	6	4	
	C	45	35	22	14	11	9	9	8	8	8	6
	D	60	50	40	30	25	20	20	18	16	14	
	A	25	20	10	8	6	5	5	4	4	3	3
2-4 Thread	B	35	30	17	12	10	8	8	7	7	4	
	C	45	35	22	18	15	12	12	10	10	8	6
	D	60	50	40	30	25	20	20	18	16	14	
	AA			12	9	6	5	5	4	4	3	3
Any Three Axial Pitches*	A	38	26	15	12	9	8	8	7	7	5	5
1 Thread	B	53	38	22	16	12	11	10	9	9	7	
	C	70	50	30	21	16	14	13	12	12	12	8
	D	120	100	80	60	50	40	35	25	20	16	

*Total indicator variation.
Class AA Ultra Precision Hobs are made single thread only.
Tolerances apply only to standard or recommended hob diameters.

(All readings in tenths of a thousandth of an inch)

DIAMETRAL PITCH		1 Thru 1.999	2 Thru 2.999	3 Thru 3.999	4 Thru 4.999	5 Thru 5.999	6 Thru 8.999	9 Thru 12.999	13 Thru 19.999	20 Thru 29.999	30 Thru 50.999	51 and Finer
LEAD VARIATION (con't.)	**CLASS**											
Any Three Axial Pitches*	A	38	30	15	12	9	8	8	7	7	5	5
2-4 Thread	B	53	38	22	20	15	12	12	10	10	7	
	C	70	50	30	28	20	18	16	14	14	12	8
	D	120	100	80	60	50	40	35	25	22	18	
Adjacent Thread to	A	11	9	8	7	6	5	4	3	3	3	3
Thread Spacing*	B	14	12	11	10	9	8	6	5	5	5	
2 Thread	C	20	17	15	13	11	10	9	8	7	6	5
	D	26	22	19	17	15	13	12	11	10	9	
3 Thread	A	13	11	10	8	7	6	5	4	4	4	3
	B	16	14	12	11	10	9	7	7	6	6	
	C	22	19	16	14	13	11	10	9	8	7	6
	D	28	24	20	18	16	15	13	12	11	10	
4 Thread	A	15	13	12	9	8	7	6	5	4	4	3
	B	18	16	14	12	11	10	8	7	7	6	
	C	24	21	18	15	14	12	11	10	9	8	7
	D	30	26	22	20	18	16	14	13	12	11	
TOOTH PROFILE												
Pressure Angle	AA			2	2	1.7	1.7	1.7	1.7	1.7	1.5	1.5
or Profile*	A	10	5	3	3	2	2	2	2	2	2	2
1 Thread	B	16	8	5	5	4	3	3	3	3	2	
	C	25	15	10	5	4	3	3	3	3	3	3
	D	80	55	30	18	12	8	8	6	5	4	
2 Thread	A	12	7	5	4	3	3	2	2	2	2	2
	B	18	10	7	5	5	4	3	3	3	2	
	C	27	16	11	7	5	4	3	3	3	3	3
	D	80	55	30	18	12	8	8	7	6	5	
3-4 Thread	A	15	8	5	4	3	3	3	2	2	2	2
	B	20	10	7	5	5	4	4	3	3	2	
	C	27	16	11	7	5	4	4	3	3	3	3
	D	80	55	30	18	12	8	8	7	6	5	
Start of Approach	AA			100	80	70	60	60	40	40	30	
(Plus or Minus)	A	200	180	160	140	120	100	80	60	40	30	
1 Thread	B	220	200	180	160	140	120	100	80	50	40	
	C	220	200	180	160	140	120	100	80	60	50	
	D	260	240	220	200	180	160	140	120	100	80	
2-4 Thread	A	200	180	160	140	120	100	80	60	50	40	
	B	220	200	180	160	140	120	100	80	60	50	
	C	220	200	180	160	140	120	100	80	60	50	
	D	260	240	220	200	180	160	140	120	100	80	
Symmetry of Approach*	AA			70	60	50	40	40	25	25	25	
1 Thread	A	150	130	120	100	90	80	60	50	35	25	
	B	180	150	130	120	100	90	80	70	45	35	
	C	180	150	130	120	100	90	80	70	55	45	
	D	200	180	160	140	120	110	100	90	80	60	
2-4 Thread	A	150	130	120	100	90	80	60	50	40	30	
	B	180	150	130	120	100	90	80	70	60	50	
	C	180	150	130	120	100	90	80	70	60	50	
	D	200	180	160	140	120	110	100	90	80	60	

*Total indicator variation.
Class AA Ultra Precision Hobs are made single thread only.
Tolerances apply only to standard or recommended hob diameters.

(All readings in tenths of a thousandth of an inch)

DIAMETRAL PITCH		1 Thru 1.999	2 Thru 2.999	3 Thru 3.999	4 Thru 4.999	5 Thru 5.999	6 Thru 8.999	9 Thru 12.999	13 Thru 19.999	20 Thru 29.999	30 Thru 50.999	51 and Finer
TOOTH PROFILE (con't.)	**CLASS**											
Tooth Thickness (Minus Only) 1-4 Thread	AA			15	15	10	10	10	10	10	5	5
	A	30	20	15	15	10	10	10	10	10	5	5
	B	30	20	15	15	10	10	10	10	10	5	
	C	35	25	20	20	15	15	15	15	15	10	10
	D	40	35	30	25	20	20	20	20	20	15	
SHARPENING (1-4 Thread)												
Spacing Between Adjacent Flutes*	AA			20	15	10	8	8	6	6	6	6
	A	40	30	25	20	15	10	10	10	10	10	10
	B	50	45	40	30	20	15	15	10	10	10	
	C	50	45	40	30	20	15	15	10	10	10	10
	D	60	60	50	50	30	25	25	20	17	17	
Spacing Between Non-Adjacent Flutes*	AA			40	35	25	15	15	15	15	15	15
	A	80	60	50	40	30	30	30	25	25	20	20
	B	100	90	80	60	50	50	50	40	35	30	
	C	100	90	80	60	50	50	50	40	35	30	30
	D	120	120	100	100	80	80	70	60	50	40	
Cutting Faces Radial To Cutting Depth*	AA			10	8	6	5	5	3	3	3	3
	A	30	15	10	8	6	5	5	3	3	3	3
	B	50	25	15	10	8	7	7	5	5	5	
	C	50	25	15	10	8	7	7	5	5	5	5
	D	100	75	50	40	30	20	20	15	15	10	

		FACE WIDTH 0 to 1"	1" to 2"	2" to 4"	4" to 7"	7" and up
Accuracy of Flutes, Straight And Helical*	AA	8	10	15	20	20
	A	10	15	25	30	50
	B	10	15	25	30	50
	C	10	15	25	30	50
	D	15	23	38	45	75

BORE (1-4 Thread)

		BORE DIAMETER 2.500"	2.000"	1.500"	1.250"	.750"	.500" & smaller
Diameter, Straight Bore (Plus Only)	AA				2	2	2
	A	8	8	5	2	2	2
	B	10	10	8	3	2	2
	C	10	10	8	3	2	2
	D	10	10	8	5	4	3

		ALL DIAMETERS	LENGTH
Percent of Bearing Contact, Straight Bore	AA		75
	A		75
	B		75
	C		60
	D		50

		ALL TAPERS	CIRCUMFERENCE	LENGTH
Percent of Bearing Contact, Taper Bore	AA		95	75
	A		90	60
	B		90	60
	C		90	60

*Total indicator variation.
Class AA Ultra Precision Hobs are made single thread only.
Tolerances apply only to standard or recommended hob diameters.

STANDARD HOB SPECIFICATIONS & TOLERANCES
RECOMMENDED HOB SIZES

Single Thread Coarse Pitch Hob Sizes
for Ground and Unground Hobs
(1-19.99 Normal Diametral Pitch)
For Spur and Helical Gears

Normal Diametral Pitch	Nominal Hole Diameter	Outside Diameter	Overall Length
1	2½"	10¾"	15"
1¼	2"	8¼"	12"
1½	2"	8"	10"
1¾	2"	7¼"	9"
2.00-2.24	1½"	5¼"	8"
2.25-2.49	1½"	5½"	7½"
2.50-2.74	1½"	5"	7"
2.75-2.99	1½"	5"	6"
3.00-3.49	1¼"	4½"	5"
3.50-3.99	1¼"	4¼"	4¾"
4.00-4.99	1¼"	4"	4"
5.00-6.99	1¼"	3½"	3½"
7.00-7.99	1¼"	3¼"	3¼"
8.00-11.99	1¼"	3"	3"
12.00-13.99	1¼"	2¾"	2¾"
	¾"	2"	2"
14.00-19.99	1¼"	2½"	2½"
	¾"	1⅞"	1⅞"

Single Thread Fine Pitch Hob Sizes
for Ground and Unground Hobs
(20 Normal Diametral Pitch and Finer)
For Spur and Helical Gears

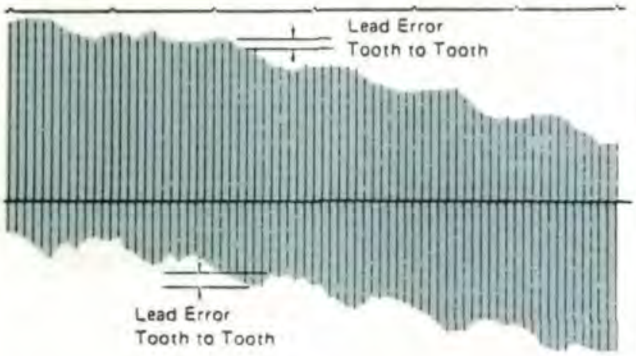
Normal Diametral Pitch	Nominal Hole Diameter	Outside Diameter	Overall Length
20-21.99	1¼"	2½"	2½"
	¾"	1⅞"	1⅞"
22-23.99	1¼"	2½"	2"
	¾"	1⅞"	1⅞"
24-29.99	1¼"	2½"	2"
	¾"	1⅞"	1⅞"
	½"	1¼"	1¼"
30-55.99	¾"	1⅞"	1½"
	½"	1⅛"	1⅛"
	.3937 (10 mm)	1⅛"	¾"
	.315 (8 mm)	⅞"	½"
56-85.99	¾"	1⅝"	1⅛"
	½"	1⅛"	1⅛"
	.3937 (10 mm)	1⅛"	¾"
	.315 (8 mm)	⅞"	½"
86-130.99	¾"	1½"	¾"
	½"	1⅛"	⅞"
	.3937 (10 mm)	1"	¾"
	.315 (8 mm)	¾"	½"
131-200	¾"	1⅜"	⅝"
	½"	1⅛"	½"
	.3937 (10 mm)	1"	½"
	.315 (8 mm)	¾"	½"

Multiple Thread Course Pitch Hob Sizes
for Ground and Unground Hobs
(2-19.99 Normal Diametral Pitch)
For Spur and Helical Gears

Normal Diametral Pitch	Number of Threads	Nominal Hole Diameter	Outside Diameter	Overall Length
2-2.99	2	1½"	6½"	8"
3-3.99	2	1½"	5½"	5½"
4-4.99	2 or 3	1½"	5½"	5½"
5-6.99	2, 3 or 4	1½"	5"	5"
7-7.99	2, 3 or 4	1¼"	4"	4"
8-8.99	2, 3 or 4	1¼"	3¾"	3¾"
9-11.99	2, 3 or 4	1¼"	3½"	3½"
12-13.99	2, 3 or 4	1¼"	3¼"	3¼"
14-15.99	2, 3 or 4	1¼"	3"	3"
16-19.99	2, 3 or 4	1¼"	2¾"	2¾"

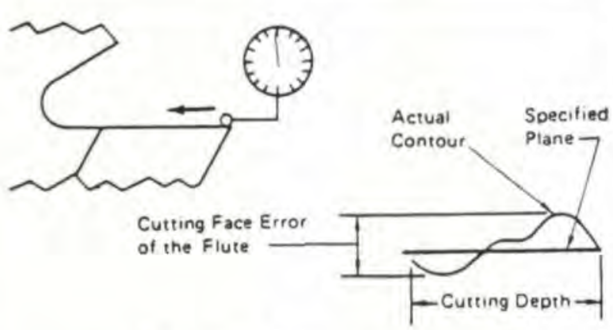
Multiple Thread Fine Pitch Hob Sizes
for Ground and Unground Hobs
(20 Normal Diametral Pitch and Finer)
For Spur and Helical Gears

Normal Diametral Pitch	Number of Threads	Nominal Hole Diameter	Outside Diameter	Overall Length
20-29.99	2, 3 or 4	1¼"	2½"	2½"
30-50	2, 3 or 4	¾"	1⅞"	1½"



Recording Lead Checker Chart for a 12 Gash, 1 Thread Hob Illustrating Lead Error Tooth to Tooth

Fig. 8



Method of Measuring the Cutting Face Error of the Hob Flute

Fig. 9

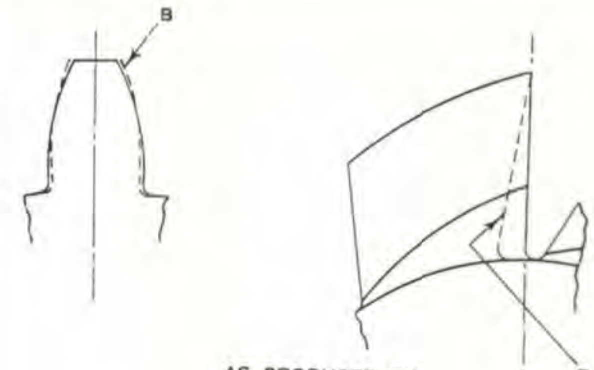


Fig. 10

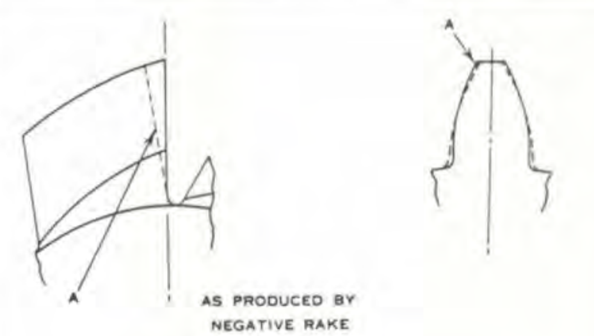


Fig. 11

Lead error — one axial pitch

Lead error in one axial pitch is the maximum deviation from the theroretical thread helix in any group of hob teeth equal to the number of hob teeth in one axial pitch. This number of hob teeth may be selected anywhere in the length of the hob and is equal to the number of hob gashes divided by the number of hob threads. (See Fig. 7)

Lead error — tooth-to-tooth

Tooth-to-tooth lead error is the maximum deviation between any two consecutive hob teeth from their relative position as measured at any point along the thread in the entire hob length. Fig. 8 illustrates this error as read from an automatic lead checker chart.

Cutting face error

The cutting faces of the hob flute are usually designed to lie in a given radial plane in the case of straight flute hobs and in a specific helicoidal surface for helical flute hobs. However, in straight flute hobs the plane can be radial or can be inclined in either direction from radial providing either positive or negative rake. When the cutting faces are designed to lie in a plane, the variation of the actual hob flute cutting faces from that plane is considered to be a flute cutting face error. (See Fig. 9)

Positive rake in sharpening increases depth and decreases pressure angle of the hob tooth. The resulting gear tooth is too heavy at the top and too thin at the bottom as shown in Fig. 10.

Negative rake in sharpening decreases the depth and increases the pressure angle of the hob tooth. This results in a cutting drag and makes the gear tooth lighter at the top and heavier at the bottom. (See Fig. 11)

Optimizing Tool Design

Multiple thread hobs

The two primary considerations in determining the number of threads on the hob are the production requirements and the accuracy and finish requirements. Increased production is the outstanding advantage of multithread hobs.

A single-thread hob rotates once for each tooth on the part, a double-thread hob rotates once for every two teeth on the part, a triple-thread hob rotates once for every three teeth, etc. Therefore, the workpiece rotates faster in relation to the hob speed, depending upon the number of threads on the hob.

The increase in the speed of indexing, however, does not usually result in a proportional gain in production. The diameter of a multithread hob is larger than a corresponding single-thread hob but does not increase in proportion to the number of threads. Therefore, the number of flutes does not increase in direct proportion to the number of threads.

Material

When considering the total manufacturing cost per metal cutting operation the price of the tool is, in fact, minor in percentage. It is not uncommon that the purchase cost of a tool will amount to only about 5% of the total cost per part. With this in mind, it follows that simply buying the cheapest tool is not an effective way of reducing cost.

If the purchase price of the tool is combined with the cost of resharpening the tool, we find that the total tool cost is approximately 15% of the metal cutting cost per part produced. The balance, or about 85%, is considered machining cost. This will be discussed in detail later. The percentage stated above are for gear cutting tools such as hobs and shaper cutters. Less expensive tools will have different ratios. Table 1 shows the relative cost of tools made of different steels as they are related to the base price of an M2 tool.

Table 1 takes into account both the increase due to material cost (premium steels because of manufacturing process and alloying elements generally are more expensive) and the additional machining cost encountered by the tool manufacturer.

The life of any particular tool is directly related to the following factors:

a. Crater wear
b. Corner/flank wear
c. Chipping

Crater wear is largely a function of red hardness and abrasion resistance of the tool steels being examined. The usual causes of cratering are cutting conditions (speed and feed) too high for the tool material.

Corner/flank wear depends on micro-chipping due to lack of toughness or abrasive wear due to lack of hardness. Under normal conditions, it is flank wear that puts the limit on the life of the tool.

Chipping occurs when the tensile strength of particular tool steel is exceeded. This type of wear is due to brittle tool materials subjected to excessive mechanical loads.

TABLE 1

M2	M2HC	M3	M4	M35
100	100	125	2125	140
M42	REX 20	REX 25	REX 76	T1
150	—	170	200	150
T15	ASP23	ASP30	ASP60	
180	110	125	160	

% Relative Price (To M2 Material)

TABLE 2

M2	M2HC	M3	M4	M35
100	100	110	125	120
M42	REX 20	REX 25	REX 76	T1
120	120	130	130	100
T15	ASP23	ASP30	ASP60	
130	110	120	135	

% Relative Life (To M2 Material)

TABLE 3

M2	M2HC	M3	M4	M35
100	100	110	115	110
M42	REX 20	REX 25	REX 76	T1
125	125	125	125	100
T15	ASP23	ASP30	ASP60	
125	110	110	135	

% Relative Feed (To M2 Material)

TABLE 4

M2	M2HC	M3	M4	M35
100	100	100	100	120
M42	REX 20	REX 25	REX 76	T1
125	125	140	200	100
T15	ASP23	ASP30	ASP60	
200	100	120	160	

% Relative Speed (To M2 Material)

As pointed out earlier, there is a definite trade-off between wear resistance/red hardness (flank wear and cratering) vs. toughness (chipping and flank wear). It is due to this trade-off that in most instances, the cutting conditions will change in relation to the tool material being used.

Table 2 lists some relative extended life values for different tool steels as they are related to an M2 tool. Example: You can expect an estimated 25% more parts per tool if you compare M4 to M2.

In the tool cost section it was stated that approximately 85% of the total manufacturing cost is due to machining cost. This machining cost is directly related to feeds and speeds in any given application. To effectively decrease machining cost, it is required that the volume of stock removal per unit

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CIRCLE A-23 ON READER REPLY CARD

DESIGN AND SELECTION OF HOBS . . .
(continued from page 45)

time be increased by increasing the feed and/or speed. Maximum values in this area are sought with regard to:

a. Tolerances specified.
b. Surface finished required.
c. Stability and design of machine.

Some typical figures for relative feeds and speeds are shown in Table 3 and 4. As with the other comparison tables, all the materials are shown in relation to an M2 tool. It should be pointed out that these figures are only approximations. Actual results may vary according to how aggressively the original M2 tool is being applied.

Titanium Nitride Coatings

To develop the proper tool design for a specific application, yet another variable must be given consideration – titanium nitride coatings.

In an effort to improve tool life and increase the productivity of the gear cutting process, extensive research and development has taken place in the past few years to successfully apply titanium nitride coatings to high speed steel tools.

The intent of this section is to suggest the importance of proper selection of high speed steels used in combination with titanium nitride coatings.

There are two modes of failure directly related to the substrate material used with titanium nitride coated tools, cratering, and lack of adherence.

After a coated tool (hob, shaper cutter) is sharpened, the titanium nitride layer is removed from the cutting face of the tooth. This exposes the substrate material and thus, the cratering resistance is only as good as the base metal. If the primary mode of failure on an uncoated tool is cratering, simply coating the tool with titanium nitride is not the solution. Another point to keep in mind is that although cratering was not a problem with the uncoated tool, with increased feeds and speeds and more pieces per sharpening being cut, it is likely that cratering may become a problem. Whether cratering was a problem with the uncoated tool or developed after titanium nitride coating, it would be suggested to try a higher alloy steel (one high in abrasion resistance). If cratering continues to be a problem after trying different tool steels, it may be necessary to revise (slow down) the operating conditions.

Occasionally, the wear pattern of coated cutting tools can be flaking close to the cutting edge. Consequently, the tool

(continued on page 48)

TECHNICAL CALENDAR

March 23-25 **14th Annual AGMA Gear Manufacturing Symposium**
Holiday Inn, Airport
Indianapolis, Indiana

AGMS's 1986 Manufacturing Symposium will offer an open forum with industry experts and papers on topics of interest to everyone involved in gear manufacturing. The focus of this year's Symposium will he Hard Finishing, Heat Treatment, Process Control and Basic Gear Technology. As with past symposiums, the papers presented will provide the latest information on each of the subjects. Attendess will have the opportunity to ask questions of the speakers following each presentation.

For further information call: Polly MacKay, Meetings Coordinator, American Gear Manufacturers Association — (703) 684-0211.

April 7 **Deburring & Surface Refinement by Mass Finishing Methods**

Contact Anna Guy at SME, (313) 271-1500, ext. 370.

April 8-9 **Applying Modern Buff, Brush & Polish Techniques**

Contact Dianne Leverton at SME, (313) 271-1500, ext. 394.

April 10-11 **Nontraditional Deburring & Final Finish Machining Methods**

Contact Dianne Leverton at SME, (313) 271-1500, Ext. 394.

CONTROLLING TOOTH LOADS . . .
(continued from page 33)

last two designs are much easier to set up and measure. Also, $L_1/L_2 = N_1/N_2$ for the exact designs, but not for the conventional design.

If L is chosen to be $L = A/P_{nc}$, where A = any integer, then equating this expression for L with the previous equation for L and solving for sin ψ_1 yields sin $\psi_1 = \pi/A$. Table 2 lists ψ_1 for various values of A. If a helical gear pair is to be redsigned to use exact leads, then a value of A can be chosen from Table 2 to give approximately the same helix angle as the original design. (Refer to Table 1 and compare $\psi_1 = 16.26020470°$ for the original design to $\psi_1 = 16.26020470°$ for the improved designs.)

To accommodate the same center distance, one or both of the gears may be enlarged or reduced slightly. If, for some reason, the helix angle must be closer than those listed in Table 2, a decimal value for A can be used (9.1 or 9.3 for example). This approach is still preferable to trying to make L accurate to eight places.

Approach vs. Recess

Helical gears are best used in single pairs only. When the operating conditions are such that one gear is always the driver and the other always the follower, all recess action should be specified. This design places the pitch line of the driver at the bottom of the working tooth depth and the pitch line of the follower at the outside diameter. The result is low noise and friction, improved lubrication characteristics, and increased surface endurance. If the drive is used in an application where either gear is the driver, then the pitch line should be at the center of the tooth working depth.

E-2 ON READER REPLY CARD

DESIGN AND SELECTION OF HOBS . . .
(continued from page 47)

life is very much dependent on the adherence between the substrate and the titanium nitride layer.

Measuring the ability to adhere is a difficult problem. The most common method is the "scratch test". A small radius diamond is scratched across the surface of the titanium nitride coated sample. The load on the diamond is successively increased until flaking occurs. The load at which flaking occurs is referred to as the critical load. This critical load, however, is also dependent on hardness of the substrate material, cleaning process, and the method of titanium nitride application. It is not possible to rate present high speed steels according to adherence capability due to the measuring difficulty described above. It can be said, however, that generally the same tool life relation between the different high speed steels also exists after the titanium nitride coating, but at a higher level.

Once the titanium nitride begins to flake or abrade away, the wear resistance of a coated tool depends to a great extent on the substrate material. For this reason along with previous comments on crater resistance, it is suggested that the best results (tool life) of coated tools have been obtained using the high alloy powdered metal tools steels.

E-4 ON READER REPLY CARD

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