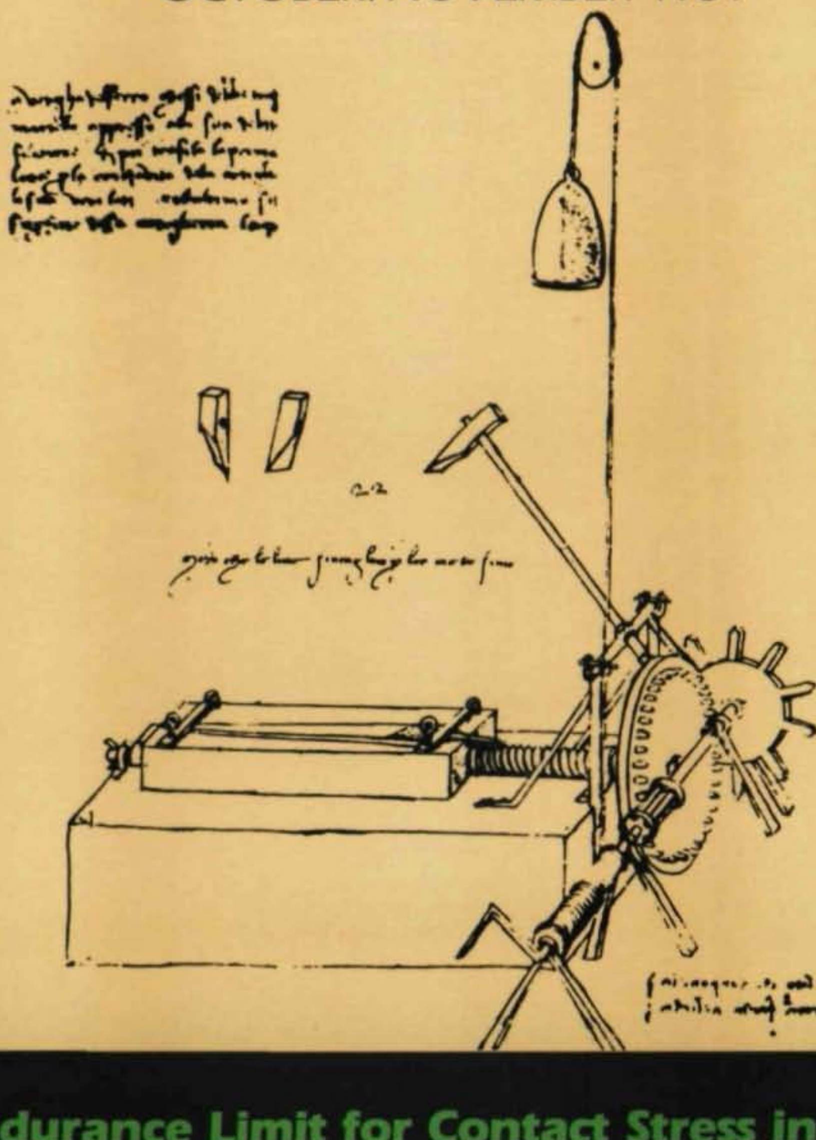
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

OCTOBER/NOVEMBER 1984



Endurance Limit for Contact Stress in Gears

Cone Drive Double Enveloping Worm Gearing

Gear Generating Using Rack Cutters

Scoring Load Capacity of Gears Lubricated with EP-Oils

Austempered Ductile Iron

Design of Involute Gear Teeth

NOTES FROM THE EDITOR'S DESK

History comes around full circle. It is interesting to talk to gear manufacturers who service the defense, aerospace, automotive and computer industries and find that their sales, production and backlogs reflect excellent and, in some cases, record breaking business. Yet companies primarily dependent upon industrial and commercial gearing, while doing slightly better, are still having difficulties.

It was not too many years ago that the currently booming industries were flat on their back. Like the industrial and commercial gear manufacturers today, those companies pared down their operations, cut their overhead, sold off or scrapped old machinery, and in other ways consolidated to become leaner and more efficient. They did this out of necessity — out of a sense of survival. When business finally started to come their way, they were ready. They have been, and are continuing to purchase the finest, latest technology available in order to produce the best product at the lowest possible cost.

We can always count on things to change, and so can those in industrial and commercial gearing. Unfortunately, the past and present do not give us a good perspective of our future. It is a human trait to assume that the future will continue to be like the present. When energy in this country was booming, almost everyone assumed that the cost of oil would continue to rise without interruption. Loans and projections were being made on the basis of fifty-sixty dollar oil. The subsequent bankruptcy of the Penn Square Bank, and the close escape of Continental Illinois National Bank, are good examples of the folly of this assumption. Two years ago, economists were projecting that inflation would continue to rise far into the future. Yet today, we live with inflation at its lowest point in twenty years.

There is a lesson in this, as history comes around full circle. While industrial and commercial gearing is now somewhat in the doldrums, and many in this field cannot see anything on the horizon to spur their business, this too will change. The recovery in many areas of manufacturing has been very slow. However, in the long run, this will be for the best because this business growth is being built on a solid foundation. The recovery is gradually reaching the industrial and commercial gear manufacturers; and although they are not yet ready to back their growing optimism with hard dollars, there is a general belief that business will be much better one year from now. How each company conducts itself as it comes out of the recession will influence the likelihood of its survival and prosperity in the 1980's.

While things are not always good, we can take heart that we can always count on things to change. Will you be ready to reap the rewards when history comes around full circle?

Michael Goldstein

Michael Goldstein
Editor/Publisher

LOVE AMERICAN STYLE.

Hard Skived Fully Conjugate Gear Generation

The Klingelnberg AMK-400 spiral bevel gear generator offers **ground gear quality at lapped gear prices.** The system assures a gear design that results in higher power transmission, smaller gear diameter and quieter gear operation.

The *unique parallel* depth tooth design, characteristic of spiral bevel gears by the Klingelnberg system, *provides excellent axial* adjustability without sacrificing load carrying capacity.

Easy to set up and without the conventional tooth bearing development, the AMK-400 guarantees maximum productivity. The BZN* coated carbide blades produce a superior finish and uniform tooth quality, up to AGMA Class 13. The AMK-400 will skive gears up to 20 *inches in diameter, from 2 to 25 DP.*

For further information, contact:
Klingelnberg Corporation,
15200 Foltz Industrial Parkway,
Strongsville, OH 44136.
Phone: (216) 572-2100.

Geared to Progress

KLINGELNBERG

*BZN is a trademark of General Electric Co.

CIRCLE A-2 ON READER REPLY CARD

EDITOR/PUBLISHER
Michael Goldstein

ASSISTANT PUBLISHER
Peg Short

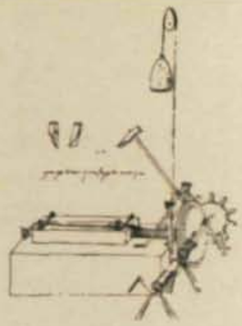
BUSINESS MANAGER
Deborah A. Donigian

ART CONSULTANT
Marsha Feder

PUBLISHING CONSULTANT
Ray Freedman

EDITORIAL ASSISTANTS
Deborah Martin
Michelle Short

COVER DESIGN:
Kathy Mitter



COVER

The Advanced Technology of
LEONARDO DA VINCI 1452 - 1519

Almost five centuries have passed since Leonardo sketched his thoughts and designs for machines. His purpose, like design engineers of today, was to find ways to have work done more quickly and easily, while maintaining standards of uniformity and precision. He seemed intent upon discovering universal laws by experimenting with basic machine elements and assigning quantitative values wherever he could. He analyzed every possible type of bearings, linkages, gears and other modes of mechanical transmission. His drawings seem to indicate, however, that he continued to review his designs looking for ways to combine steps making it simpler, faster, more useful.

The cover sketch, a file maker, is one of Leonardo's earliest machine designs. It clearly shows his intuitive understanding of the concept of modern automation. It was designed so that once it was started, it required no further human intervention. Once the crank in the foreground was turned, a coordinated set of motions began. The rope would unwind around the crankshaft, causing a lug at the far end of the crankshaft to strike a lug attached to the pivotal axle, which held a long hammer handle. The prongs of the sprocket wheel would strike the lugs of the hammer axle, causing the hammer to rise. When the hammer fell, it would strike the blank file and leave a series of furrows on its surface. This design provided for increased productivity of the files, and also reduced the chance of human error thus providing for greater uniformity.

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

CONTENTS — PAGE NO.

FEATURES

DEPARTMENTS

October/November 1984 — Vol. 1, No. 3

SUBSCRIPTION INFORMATION: GEAR TECHNOLOGY is distributed free of charge to qualified individuals and firms in the gear manufacturing industry. To continue receiving your subscription, complete and mail the application card found elsewhere in this magazine. Paid subscription rates are $5.00 per issue + $1.75 for postage & handling or $30.00 per year. Subscription rates outside the United States available upon request. Subscription requests to RANDALL PUBLISHING CO., INC., P.O. BOX 1426 Elk Grove, IL 60007.

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.

GEAR TECHNOLOGY (ISSN 0743-6858) is published bimonthly as a reference source of technological developments in the gear industry. It welcomes independent opinion, but the opinions expressed by authors and contributors to GEAR TECHNOLOGY are not necessarily those of the editor or publishing staff, and we assume no responsibility for the validity of claims in connection with items printed.

POSTMASTER: Send address corrections to GEAR TECHNOLOGY, P.O. Box 1426, Elk Grove, IL 60007.

TECHNICAL CALENDAR

AGMA Calendar

The following American Gear Manufacturers Association meetings are scheduled. For further information, please call AGMA (703) 525-1600.

Oct. 14-17 1984	Fall Technical Meeting* *L'Enfant Plaza Hotel, Washington, DC*
Oct. 31-Nov. 1 1984	Flexible Couplings Committee (6.4) Green-Tree Marriott, Pittsburgh, PA
Nov. 8-9 1984	Bevel Gearing Committee (4a) AGMS Headquarters, Arlington, VA
Nov. 9 1984	Epicyclic Enclosed Drive Committee Capital Airport Hotel, Atlanta, GA
Nov. 29-30 1984	Marine Gearing Committee (6.3a) The Peabody Hotel, Memphis, TN

*Fall Technical Meeting Highlights

"Analytical and Experimental Tooth Strength Analysis of Spur Planet Gears With Integral Bearings" by Ray Drago and B.R. Uppaluri of Boeing.

Panel Presentation-AGMA Inspection Handbooks (Millimeter and Inch) Messrs. Paul Dean (MTI), William Bradley (Phila Gear), Irving Laskin (Polaroid), Robert E. Smith (Gleason)

"What Single Flank Measurement Can Do For You" – Robert E. Smith (Gleason)

"Involute Spline Size Inspection" – John C. Nielson (Illitron)

A Discussion of the doctorate dissertation by Dr. Manfred Hirt (Renk) "Stresses in Spur Gear Teeth and Their Strength as Influenced by Fillet Radius" as translated by John Maddock.

ASME Calendar

The following is a calendar of events for The American Society of Mechanical Engineers. For further information, contact the ASME office (212) 705-7722.

Oct. 7-9 1984	Diesel & Gas Engine Power Technical *Conference, Hilton Harvest House, Boulder,* CO (212) 705-7054.
Oct. 7-10 1984	Design Automation, Mechanisms, and Medical Devices & Sporting Equipment Conferences, Hyatt Regency, Cambridge, MA (212) 705-7788.
Oct. 10-12 1984	Power Transmission & Gearing and Production Engineering Conferences, Hyatt Regency, Cambridge, MA (212) 705-7788.
Oct. 17-20 1984	Management Executives Conference—East. Innisbrook Resort, Tarpon Springs, FL (212) 705-7788.
Oct. 22-24 1984	*Joint Lubrication Conference, San Diego* Hilton, San Diego, CA (212) 705-7793
Oct. 23-26 1984	Textile Industries Division & Manufacturing Technical Group Joint Conference, Sheraton Central, Greenville, SC (212) 705-7788.
Oct. 28-31 1984	Industrial Power Conference, Philadelphia Marriott, Philadelphia, PA (212) 705-7795.
Dec. 9-14 1984	WINTER ANNUAL MEETING, New Orleans Hilton, New Orleans, LA (212) 705-7053.

SME Calendar

The following events offered by the Society of Manufacturing Engineers will be of special interest to the Gearing Industry. For further information, contact SME Public Relations, P.O. Box 930, Dearborn, MI 48121, (313) 271-1500.

Nov. 13-15 1984	Gear Processing & Mfging. Clinic* Netherland Plaza Hotel, Cincinnati, OH
Jan. 22-24 1985	HOUSTEX Tool & Mfg. Conference & Expo. Albert Thomas Convention Center, Houston, TX
Mar. 18-21 1985	WESTEC '85 Metal & Tool Expo. & Conference Los Angeles Convention Center, Los Angeles, CA (Co-sponsored by SME & ASM)

*GEAR PROCESSING & MANUFACTURING CLINIC

J. Richard Newman, retired coordinator of Gear Tools for National Broach and Machine, Division of Lear Siegler will serve as clinic leader. Mr. Newman has over 40 years experience in gear tool design and application.

Presentations at this clinic will benefit everyone responsible for gear operations, regardless of background. Time will be allocated daily for discussion of specific gear concerns with clinic staff. Attendees are encouraged to bring problem parts and/or drawings for individual attention.

VIEWPOINT

Letters for this column should be addressed to Letters to the Editor, GEAR TECHNOLOGY, P.O. Box 1426, Elk Grove Village, IL 60007. Letters submitted to this column become the property of GEAR TECHNOLOGY. Names will be withheld upon request; however, no anonymous letters will be published. Opinions expressed by contributors are not necessarily those of the editor or publishing staff.

SINGLE FLANKTESTING

Dear Editor:

It was very interesting to see Robert Smith's article on single flank testing of gears. This method of testing is widely used in Britain because it is very fast and allows checking in a fraction of the time that conventional methods take; this has been found to be a tremendous commercial advantage for checking supplies from subcontractors and also allows 100% checking.

Although mainly used here for parallel shaft gears, single flank (or Transmission Error) checking is very useful for matching bevel gears rapidly and for finding the best orientation of worms to their wheels when very high precision drive is required.

In certain cases where misalignment of helical gears occurs, single flank checking has given very useful information on the variation of alignment with load (Ref. 1). Generally it is not possible to carry out advanced vibration development work on high precision gears without using single flank checks. This is because Transmission Error may need holding to less than 5 microns (2/10ths mil) at once per tooth although at least 4 separate tolerances are involved in controlling alignment.

Extension of Transmission Error testing to full load conditions, unlike the inspection stage, has occurred on several rigs already and promises to become a routine part of gear production control giving very much quieter gears.

Yours faithfully,

J D Smith
Cambridge University Engineering Dept.

1. Gears and their Vibration, J.D. Smith, Marcel Dekker, 1983. Ch. 9.

Endurance Limit for Contact Stress in Gears

by
G. C. Mudd and J. M. France
David Brown Gear Co.

Synopsis

With the publishing of various ISO draft standards relating to gear rating procedures, there has been much discussion in technical papers concerning the various load modification factors. One of the most basic of parameters affecting the rating of gears, namely the endurance limit for either contact or bending stress, has not, however, attracted a great deal of attention. In view of the fact that ISO and other modern gear ratings attempt to assess the real stresses experienced by the gear teeth, it is important that the material allowable stresses are equally understood. This is particularly so when material properties are varying, as in a surface hardened layer.

This paper reports on work done examining variations in surface hardened gears and the interaction of surface, bending, and residual stresses in a field of varying hardness.

Introduction

The traditional stress analysis approach to determining the performance of a load carrying member is to evaluate the stress cycle and, by means of a Goodman diagram, determine a factor of safety for the material in question. There is no fundamental reason why a gear tooth should not be evaluated in the same way. However, for surface hardened gears, there are complications, because of the following reasons:

a) The stress cycle is a combination of contact (Hertzian) and bending stresses.
b) Residual stresses are present in the surface of the gear tooth.
c) The resistance of the material to fatigue, which is a function of hardness, varies with depth below the surface.

The Stress Cycle

As illustrated in Fig. 1, the stress cycle, experienced by a point on the flank of a gear tooth, consists of a compressive Hertzian stress followed by a tensile bending stress.

AUTHORS:

DR. GEORGE MUDD *is Director of Engineering at David Brown Gear Industries, Huddersfield, England. He has a B. Sc, C. Eng., and MI. Mechanical Engineering degrees. He began his career in 1951 in Design, moving in 1953 to Research and Development with particular emphasis on the fundamental nature of gear tooth failure in all applications.*

MR. J. M. FRANCE *joined David Brown Gear Industries in 1974 after spending eight years in the airframe industry. Mr. Brown holds a B. Sc. and has worked in various sections of the Engineering Division including gear development testing. Currently he is supervisor of a specialist section dealing with gear technology.*

Cycle of Stress at a Point on Tooth Flank

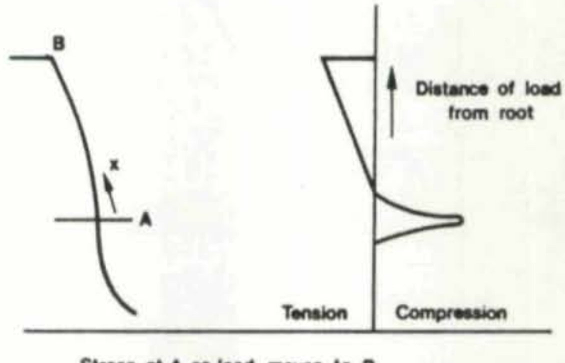


Fig. 1

Hertzian stresses are well known, and are dependant upon the relative radius of curvature of the gears at the point of contact, and vary with depth below the surface and distance from the point of application of the load. Bending stresses are traditionally associated with the tooth root bending calculations, but also have an effect on the stress cycle at the pitch line as the load moves towards the tip. In the tooth fillet, of course, the stress cycle consists of the bending stress alone, experienced once per revolution.

Residual Stress

Residual stresses may be present in all materials, but may be regarded as negligible in (through) hardened materials. In surface hardened steels, however, residual stress is induced, because of volume changes as austenite transforms to martensite during the quenching operation. As shown typically in Fig. 2, heat treatment procedures are chosen to produce a compressive residual stress at the surface, which is balanced by tensile residual stresses near the case/core interface.

The total stress state resulting from the combination of Hertzian, bending and residual stresses is, therefore, complex varying in pattern at every point from the tooth tip, to the root fillet, and from the surface to sub-surface.

Hardness Gradients

For each point, the total stress must be compared with the fatigue resistance of the material at that point, which, for a surface hardened gear, because of the hardness variation, also changes with distance from the surface (Fig. 3).

Criterion of Failure and Endurance Ratio

Over a period of years, disc tests have been carried out at D.B.G.I. as a means of comparing material performance, using a variety of materials (refs. 1 to 3). One of the limitations of tests conducted with similar sized discs has been, that

Typical Residual Stresses

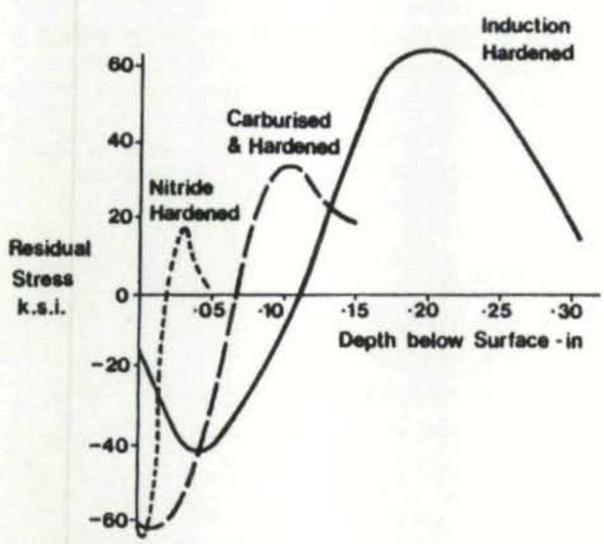


Fig. 2

Typical Hardness Gradients

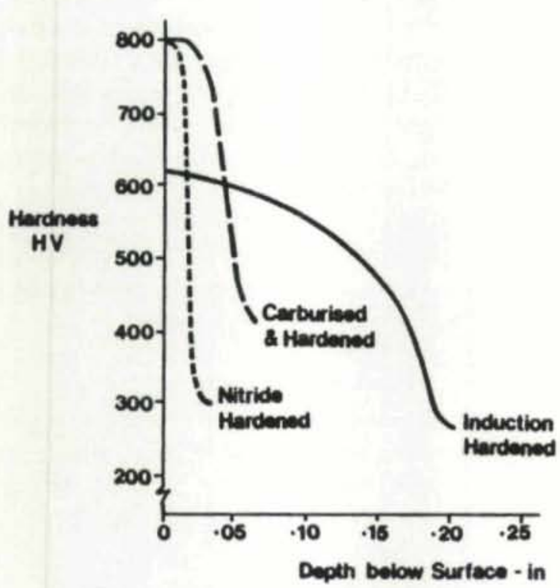
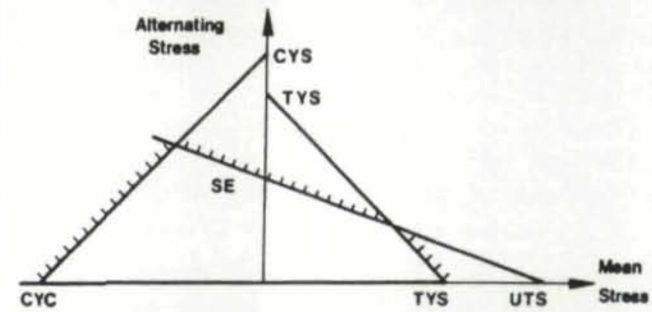


Fig. 3

for surface hardened materials, the results could not reflect the effect of radius of curvature on the performance; but these results were of great value in providing the information for carrying out an evaluation of the effectiveness of several alternative failure criteria. By comparing the predictions with the disc tests results, a judgement was made as to the criterion which most closely predicted the actual results. The combination of criterion of failure and endurance ratio, which gave the best correlation between theory and test results, was a Direct Stress criterion using an endurance ratio of 0.5. The Goodman diagram used in the evaluation is shown in Fig. 4.

Goodman Diagram used in Analysis



Indicates boundary of permissible stress

Nomenclature:-

CYS Compressive yield strength TYS Tensile yield strength

SE Endurance limit (0·5 UTS) UTS Ultimate tensile strength

Fig. 4

Use of HTZ to Evaluate Endurance Limit for Contact Stress

A computer program, HTZ, has been written to analyze the total stress field experienced by a gear tooth and compare this with the material properties at points on and below the surface of the tooth flank and fillet. A description of program HTZ, with a typical output, is shown in Appendix 1.

Using the criterion of failure selected above, the program has been used to analyze a large selection of hypothetical and real gears. The following hardening processes were considered:

1. Through hardened
2. Carburised and hardened
3. Nitride hardened
4. Induction hardened

The effect of varying the core hardness (within limits typical for each process) and of varying the casedepths has also been investigated.

Previous work (Ref. 8) has indicated that the limiting contact stress for gears is a function of the dimensionless parameter – relative radius of curvature/module (ζ_e/m_n). This parameter has been used in the presentation of the theoretical results and good correlation was again found. This is shown in Fig. 5 and Fig. 6.

In the case of surface hardened gears, a further complication is introduced by the possible variation in casedepth. The actual casedepth used on a gear may be chosen for geometrical, or economic reasons, or as a limitation of the process

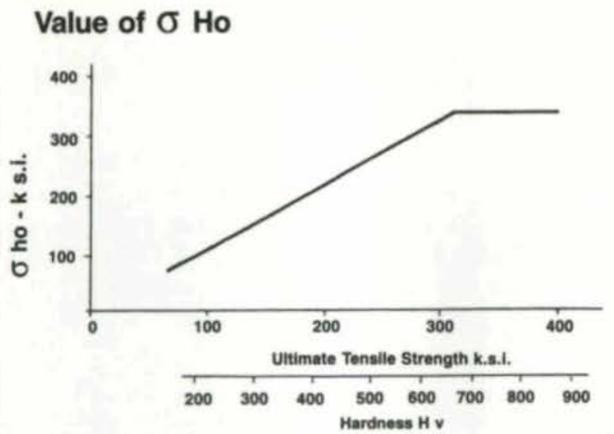


Fig. 5

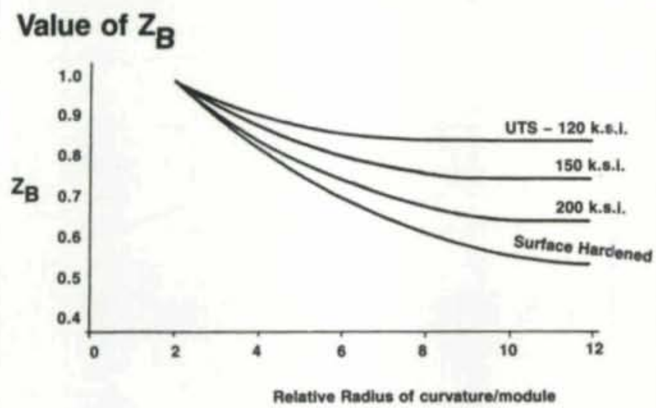


Fig. 6

used. The gear examples were, therefore, computed with different values of effective casedepth and the data generated used to define:

a) A 'limiting depth' (i.e. the effective casedepth beyond which a further increase in casedepth does not produce further increase in failure load), and
b) The reduction in failure load resulting from effective casedepths less than the limiting depth. This applies in particular to large module nitrided gears where the limiting casedepth cannot normally be achieved.

The limiting depths were found to be:

Steel	Limiting Casedepth
655M13 carburised and hardened	0.16 x normal module
722M24 nitride hardened	0.20 x normal module
817M40 induction hardened	0.32 x normal module

Effective casedepth is defined as the depth at which the hardness falls below 500 HV. The value of 0.16 x normal module for carburised and hardened steel was found to be remarkably close to the historical values, which have proved to work well in practice.

The reduction in the endurance limit for contact stress, with reduction in effective casedepth, is shown graphically in Fig. 7 against the parameter actual casedepth/limiting casedepth. The same graph can be used for each of the three hardening processes considered.

The endurance limit for surface stress is then calculated from the product of the values from the three graphs in Fig. 5, Fig. 6 and Fig. 7, i.e.

$$\sigma_{Hoeff}\ (S_{AC} \text{ in AGMA}) = \sigma_{Ho} \bullet Z_B \bullet Z_C$$

when σ_{Hoeff} = endurance limit for surface stress

σ_{Hoeff} = endurance limit for surface stress of a disc

Z_B = disc/gear correllation factor

Z_c = casedepth factor

Experimental Results

Disc Results

Disc results are presented as Hertzian stresses at the failure load calculated from:

$$\sigma_{HD} = \sqrt{\frac{F}{b\pi} \bullet \frac{E}{1-\nu^2} \left(\frac{1}{r_1} + \frac{1}{r_2}\right)}$$

when F = force between discs (failure load)

b = width of disc

E = Young's modulus

ν = Poisson's ratio

r_1, r_2 = radii of curvature of roller and disc respectively

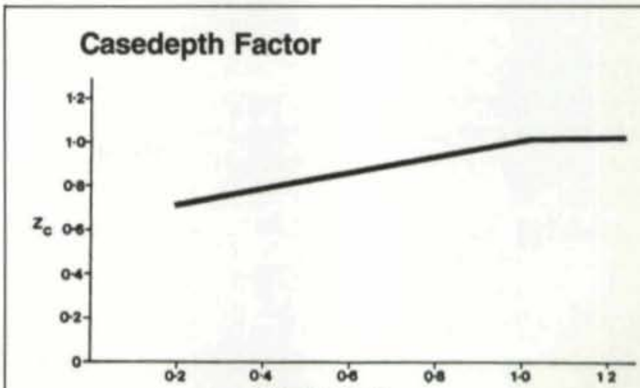


Fig. 7

Experimental and HTZ Results for Discs

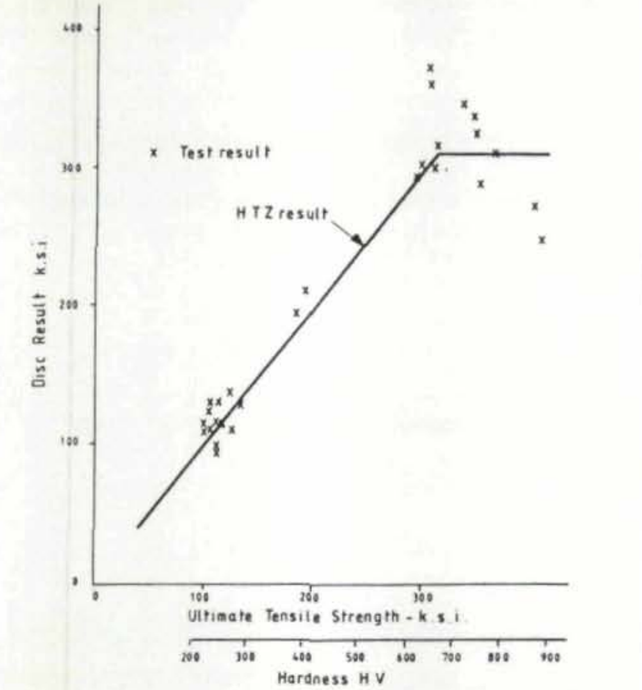


Fig. 8

The results are shown graphically in Fig. 8 with the theoretical result from HTZ for comparison.

Gear Circulator Tests

In parallel, with the theoretical analysis, a number of gear circulator tests have been performed. The first tests were to investigate the effect of the manufacturing process on tooth root bending stress. For example, grinding after heat treatment tends to reduce the compressive residual stress at the surface; shot-peening tends to produce compressive residual stress at the surface. These results have been used for comparison with the theoretical bending results from HTZ.

Before the test result can be used to calculate a bending stress, however, it is necessary to know the ratio of the peak bending moment experienced by the tooth throughout the tooth cycle, to the nominal bending moment calculated using the AGMA inscribed parabola. This was achieved using a computer program CLODA. A description of program CLODA, with a typical output, appears in Appendix 2. The misalignment was assessed from no-load contact markings and other deviations from true involute were measured. Dynamic and surface finish effects were accounted for by factors from a gear rating standard internal to David Brown Gear Industries Ltd. This gives values of dynamic factor and roughness factors similar to ISO. Step loading was accounted for by the use of Miner's Rule.

Test Details

The tests were carried out on an 8 inch centers gear circulator. Torque was locked into the system by means of an adjustable coupling, and measured using strain gauges and telemetry equipment. Gears were 7 P_n with a 37 tooth pinion and 75 tooth wheel. Two facewidths were tested, namely 1 inch and 0.5 inch.

Test Results

Test results are, in most cases, 99% confidence levels based on five tests on identical gears. The raw results have been modified for non-uniform load and moment distribution by the factors calculated by computer program CLODA. Results are tabulated in Fig. 9 in which the theoretic result from program HTZ is also given for comparison.

Batch No.	Batch Size	Face-width	Failure Type	Test Failure Power-HP	Theoretical Power Capacity from HTZ-HP
1	5	1	Pinion Bending	647	648
2	5	1	Pinion Bending	700	648
3	4	0.5	Pinion Bending	365	324
4	2	0.5	Wheel Bending	359	330
5	1	1	Pinion Pitting	1047	965

Fig. 9

Gear Circulator Test Results

Note: The test results have been adjusted for the effect of load distribution (CLODA values), dynamic, lubricant and roughness effects. Tests to produce surface failure (pitting) are still proceeding, but early results have been used for comparison with the theory. Again, it is necessary to modify the load before calculating a Hertzian stress, in this case, by the load distribution factor as evaluated by CLODA.

Discussion of Results

The table of results (Fig. 9) shows that good agreement exists between the experimental and theoretical results for both contact and bending stress failures. For contact stress, the disc test results (Fig. 8) provided further confirmation of the theory.

The results of the effect of reduced casedepth (Fig. 7), it is believed, have an importance beyond the field of gear rating only. This graph can be used as a production optimisation and when carburised pinions are meshing, for instance, with large through-hardened wheels. In such cases, the limiting casedepth will not usually be required and considerable cost saving can be achieved.

References

1. Chesters, W. T. "Study of the Surface Behaviour of Gear Materials with Specimens of Simple Form," Paper 21 Proceedings of the International Conference on Gearing 1958.

2. Chesters, W. T. "The Effect of Material Combination on Resistance to Surface Fatigue," Paper 8 Proceedings of Institute of Mechanical Engineers, Symposium on Fatigue in Rolling Contact 1963.
3. Chaney, J. M. "Tabulation of Surface Fatigue Results," Unpublished work by David Brown Gear Industries Limited.
4. Rhodes, A. "Theoretical Fatigue Failure Criteria Based on Correlation with Disc Results," Internal D.B.G.I. Report.
5. Jaramillo, T.J. "Deflections and Moments Due to a Concentrated Load on a Cantelever Plate of Infinite Length," ASME Paper No. 49-APM-15.
6. Wellauer, E. J. & Seireg, A. "Bending Strength of Gear Teeth by Cantelever Plate Theory," ASME Paper No. 59-A.50.
7. Various David Brown Gear Industries Ltd. internal reports.
8. Harrison, W. H. & Mudd, G. C. "Gear Load Capacity Spur and Helical Gears, Proposal for a New Worldwide Gear Rating Procedure," Paper to BGMA meeting May 1973.

Appendix 1 - Program HTZ

The program 'HTZ' analyses the complete stress history of a disc, or a gear tooth, down the flank and in the root fillet at varying depths below the surface in the following way:

A matrix is set up consisting of ten points down the flank and six points round the root fillet, each at fifteen depths below the tooth surface.

Considering a particular point in this matrix a nominal load intensity is applied at the tooth tip and a finite element analysis performed to establish the bending stress at that point.

The same nominal load intensity is then applied at points down the flank, and using classical Hertzian theory, the contact stress at the point under consideration is calculated and combined with the bending stress previously calculated. Residual stresses, either from heat treatment processes or from finishing processes, are added to this stress pattern. Using a Goodman diagram and the stress cycle established above, the load intensity and position (for Hertzian stress) is iterated until a reserve factor of unity on fatigue life occurs at that point.

This is repeated for each point in the matrix, such that a matrix of failure load intensities can be constructed. In a surface hardened gear where the hardness varies with depth, each point in the matrix has an identifiable hardness and, therefore, a different Goodman diagram.

The failure load intensities are then converted to reserve factors by dividing throughout by their minimum (i.e. a reserve factor of unity will occur at one point in the matrix, all other values being greater or equal to unity).

The minimum reserve factor at the pitch line is then used to calculate the basic endurance limit for contact stress (SAC in AGMA) from classical Hertzian theory.

In the tooth fillet, where there is no contact, the tooth root bending stresses, including stress concentration, are used in the Goodman diagram. Again residual stresses and the hardness gradients are taken into account. In this case, the load is taken as acting at the tooth rip and by iterative procedure is used until a reserve factor of unity is achieved. Using the AGMA inscribed parabola procedure an equivalent permissible bending stress (SAT in AGMA) can be evaluated.

Input to the program consists of:

a) gear geometry

b) residual stress pattern

c) material hardness gradient (if surface hardened) or ultimate tensile strength (if through-hardened)

Appendix 2 - Program CLODA and Its Use in Interpretation of Gear Tests

Description of Program

Program CLODA evaluates the distribution of load and the resultant bending moment at the critical section across a meshing spur or helical gear pair. The load distribution is calculated using the contributory flexibility of the gears, the transmitted torque and the deviation of the mesh from true involute and from true alignment. The bending moment distribution evaluated from the load distribution using our integration of Jaramillo's cantelever plate theory (ref. 5) modified by the moment-image method of Wellauer and Seireg (ref. 6).

Tooth deflections are based on a three dimensional finite element model (ref. 7) of a helical gear tooth which enabled the end effects to be evaluated. For speed of operation it was found possible to construct a modification file based on these three dimensional results which modifies the two dimensional model of each gear tooth as it is considered.

Input to the program consists of:

a) gear details

b) shaft sectional details

c) operating torque

d) alignment and pitch errors

e) profile and helix modifications

f) imposed deflections (due to other loads on the same shaft)

The analysis is performed for one mesh at ten positions through the engagement cycle. An option is available to graph the load intensity, moment intensity and Hertzian stress at the position at which the maximum occurs. Typical graphs are shown in Fig. 10.

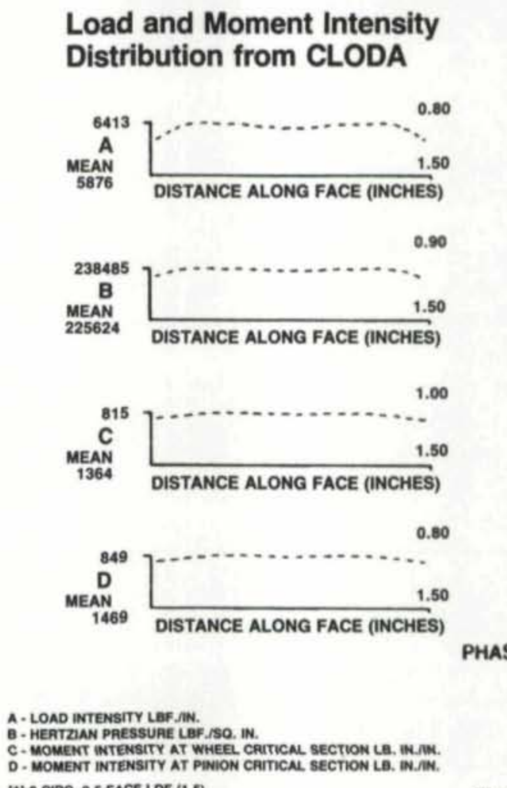


Fig. 10

This article was reprinted from the American Gear Manufacturers Association, Technical Paper P229.29.

Cone Drive Double Enveloping Worm Gearing Design & Manufacturing

W. G. Loveless, Chief Engineer
Ex-Cell-O Corporation
Cone Drive Operations
Traverse City, Michigan 49684

History of The Evolution of Double Enveloping Worm Gearing

Worm gearing is of great antiquity, going back about 2100 years to Archemides, who is generally acknowledged as its inventor. Archemides' concept used an Archemidial spiral to rotate a toothed wheel. Development of the worm gearing principle progressed along conventional lines until about 500 years ago when Leonardo DaVinci evolved the double enveloping gear concept. Worm gearing today is basically divided into three classes or types as follows: (Fig. 1)

1. Those having neither element throated.
2. Those having one (1) element throated (generally the gear) (cylindrical worm gearing).
3. Those having both elements throated (double enveloping worm gearing).

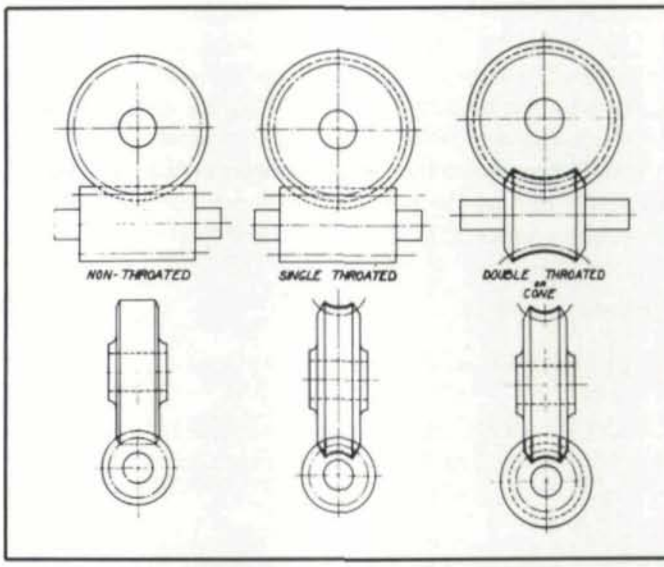


Fig. 1—Diagrams of three classes of worm gearing.

Early worm gearing was made of wood, or wood and metal. Some ancient gears made of stone have been discovered in Sweden where they were used for grinding grain. Most of the early gearing was of the non-throated design. The precise origin of the single element throated gear cannot accurately be established, although Hughes and Phillips were making single element throated gearing in this country as early as 1873.

Historical records indicate that approximately 200 years ago in York, England the famous clockmaker, Henry Hindley, made the first throated worm design. It was used in a dividing machine which he is also credited with originating. The gear was approximately 13" in diameter and had 360 teeth. The teeth were about 1/16" thick at the pitch line, and the helix angle was about 1°. John Smeatson, a contemporary, said in part, "The threads of this screw were not formed upon a cylindrical surface, but upon a solid, whose sides were terminated by arches of circles — the screw and wheel, being ground together as an optic glass to its tool, produced that degree of smoothness in its motion that I observed, and lastly, that the wheel was cut from the dividing plate". It can be determined, from the above, that the worm was throated, but it is not clear whether the wheel was also throated. (Fig. 2)

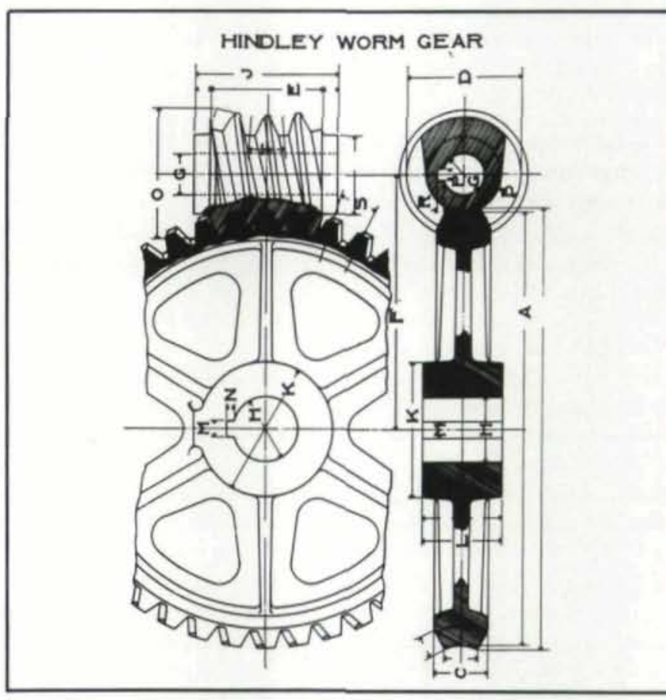


Fig. 2—Typical Hindley worm gear design.

There is no indication of further progress in throated worms

AUTHOR:

MR. W. G. LOVELESS *worked at Cone Drive Ex-Cell-O Corp. for 34 years. During this time he served as a design engineer, Assistant Chief Engineer, Chief Engineer and Director of Engineering. He attended General Motors Institute and received a BA in Mechanical Engineering. Mr. Loveless has authored and presented papers on double enveloping worm gearing to the American Gear Manufacturers Association, the National Conference on Power Transmission and co-authored a paper presented to the American Society of Mechanical Engineers. In 1984, he retired from Cone Drive and is presently acting as a consultant.*

until 1878 when Stephen A. Morse became interested in a patent on a machine for cutting them. About 1883 the U.S. Government became interested in the Hindley type gearing, and began using it where heavy shock loads were encountered and where an absence of backlash was desirable.

In general, the Hindley double-enveloping worm gearing utilized in this country incorporated an enveloping worm having straight sides in the axis of the worm. Also, the worm was generally bronze and the gear steel. It early became apparent that as a result of the constantly varying diameters and helix angles of the Hindley double enveloping worm, throughout its length, that it would be necessary to utilize a fluted worm as a hob in the production of the throated gear element. The form of the worm (and the fluted hob) prevented the use of tangential hob feed because the larger diameter of the hob at its ends (compared to the center diameter of that hob) would effectively destroy the gear form and gear teeth diameter in the hobbing process. This left the radical feed method as the most viable method for producing Hindley double enveloping worm gearing. In this method, the hob and gear were fed toward one another in a radial direction while geared to the proper time relation for the ratio involved.

For the same reason that the completed elements will not operate satisfactorily on other than the designed center distance, with Hindley designs it was equally impossible to hob a *true* form by radially feeding the hob and gear blank together. This radial feed method resulted in a "destroyed action", whereby, the ends of the hob, because of their larger diameter and rotational arc, removed excessive stock from the flanks of the gear teeth in the infeed process. (Fig. 3)

To minimize this problem, Hindley hobs were made very short, so as to reduce the cutting arc of the hob ends. The negative effect of this was to reduce the effective teeth in contact, since the worm could not be longer than the hob without creating worm/gear interference. After hobbing, Hindley double enveloping worms and gears were extensively lapped in an attempt to broaden the worm gear contact. Such lapping sometimes exceeded 48 hours — using sand and water.

In the early 1920's Mr. Samuel I. Cone of Portsmouth, Virginia, manufactured at the Norfolk Navy Yard, a double throated or double enveloping type of worm geaing, which presently carries his name, i.e., Cone Drive double enveloping

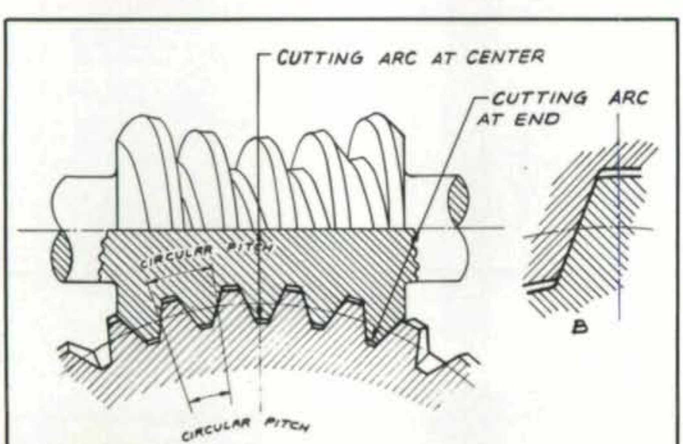


Fig. 3—Cutting arc resulting in "destroyed action".

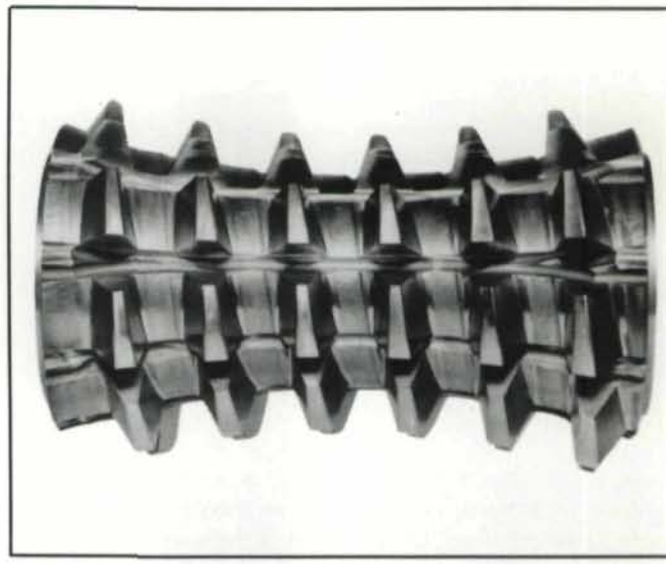


Fig. 4—Typical double enveloping gear hob.

worm gearing. Mr. Cone had developed and patented a rational method of generating the elements of double enveloping gearing which permitted the cutting of both elements, without interference, when operated at center distance. The Cone double enveloping principle utilized a hob made by gashing a worm (Fig. 4) but having thread flanks thinner in cross section than the worm from which it was evolved. When such a hob was radially fed into a gear blank to proper center distance, the gear was merely roughed out, in that its tooth form was oversize to the extent that the hob was made undersize. The "destroyed action" was still there. However, the hob and gear were now on center distance and there was stock available for truing up the gear tooth form. Radial feed here was not the answer because the hob and gear were *already* on center distance. Tangential feed would not work because it would alter the pressure angle and tooth form of the gear tooth. In the Cone principle, rotational feed is used, whereby the relative rotational position of the hob and the gear blank are changed. This creates

Fig. 5—Typical double enveloping worm cutter.

a cutting action to bring the geartooth to size, remove the "destroyed action", and produces a true tooth form. This rotational feed is not unlike taking up backlash in the gearset.

The double enveloping worm is produced in a manner similar to the double enveloping gear. A cutter representing the midplane section of a gear is used. (Fig. 5)

The teeth in the cutter, insofar as mating form is concerned, are identical with the teeth in the gear. (While this is basically true — certain liberties are taken in both hob and cutter design to provide better cutting action, chip clearance, uniform cutting pressures, etc.). Here also, the cutter teeth are undersize, but true in form to permit recutting of the thread flanks once correct center distance has been reached. Rotational side feed is again used in this finishing operation.

While the above hobbing procedure seems simple, there are many factors adding to the complexities of achieving economical manufacture. Hob and cutter heat treat distortion, true form backoff, generating and hobbing large gearsets up to 50" center distance, indexing and non-indexing ratios, special hobbing and generating machines with rotational side feed features, individual hobs and cutters for each center distance and ratio were only a few of the problems which had to be solved by Cone Drive engineers to effectively amd economically manufacture double enveloping gearsets. Double enveloping worm gearing is manufactured in this country primarily by Cone Drive, Franke Gear, Western Gear and Vard.

Design of Double Enveloping Worm Gearing

The design of Cone double enveloping worm geaing is based on a different tooth form concept. Instead of involute or other curved tooth forms, most double enveloping gears have straight sided forms on both the gear teeth and worm threads with this form tangent to a common base circle. (Fig. 6)

As with cylindrical worm gearing, experience over the years has established the practical design proportions for helix angles, pressure angles, number of teeth in gear, gear widths, relationship of worm P.D. to worm root diameter, tooth thickness, backlash, etc.. Most of these are outlined in AGMA specification 342.02, which contains design formulas as well as tables of recommended proportions. While the average engineer can use these formulas to design his own gearsets, double enveloping manufacturers prefer to establish these designs themselves, in most cases to specifically suit the application. Since a hob and cutter or other suitable tools must be made for each center distance and ratio, it is desirable that a gearset design be of the proper proportions the first time through, because changes to the design proportions cannot be readily accomplished without changing the tooling.

The straight sided form, as well as the side feeding operation in manufacturing, enable double enveloping worm gears to have variable tooth thickness. This gives considerable latitude in design. Normally we hold to a 55%-45% ratio with the gear 55% of the circular pitch and the worm 45% of the circular pitch. This gives a much more balanced design since the worm, which is made from steel, is the stronger member (120,000 PSI yield) and the gear, made from bronze, is the weaker member (25,000 PSI yield). By making the gear tooth thickness greater than the worm thread thickness, the two members are more nearly equal in relative strength. Obviously, if we can normally

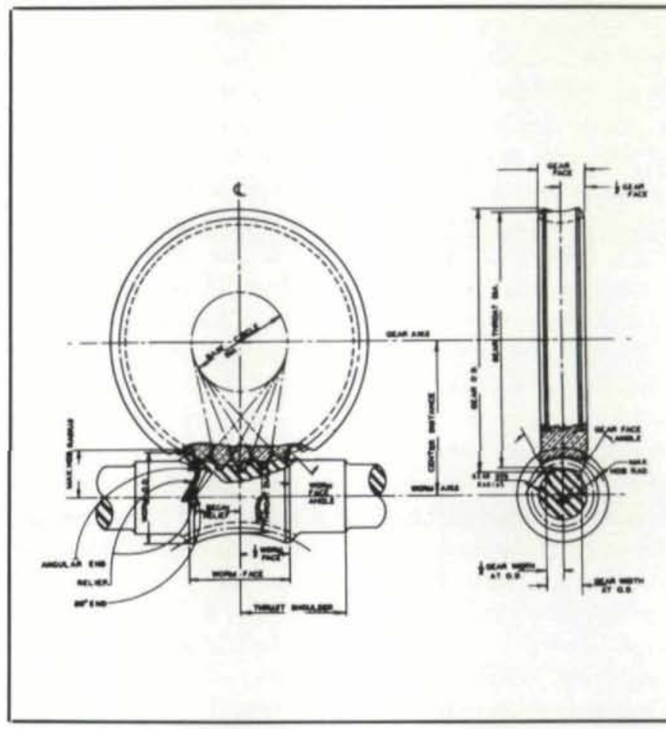


Fig. 6—Cone drive double enveloping worm gear design.

(*Continued on the next page*)

obtain a 45/55 tooth thickness relationship, then specific applications of an unusual nature which would benefit from a 60/40 or other worm/gear thickness relationship can be provided.

Another advantage to the side feeding feature of Cone double enveloping worm and gear manufacture is selective backlash. Since we can produce worm and/or gear tooth thickness exactly to suit, we can readily create close backlash gearsets. We have produced numerous designs and manufactured hundreds of gearsets with backlash in the .000 to .0002 range on design center distance. Obviously, it is extra work and more expensive to make such close backlash designs — but it can be and is done.

Gear material for double enveloping worm gearing follows the normal worm gearing practice of using SAE 65 or 65N bronzes-statically chill cast or centrifugally cast. Forged bronzes and some of the maganese aluminum bronzes are sometimes used for gear materials. They are primarily noted for their strength and not their bearing characteristics. Aluminum bronzes are often used in place of regular tin or nickel/tin bronzes where additional strength is required, not, however, without a certain penalty in bearing characteristics. Most aluminum bronzes must be restricted in rubbing velocity to work effectively in worm gear applications. Whereas we use tin bronzes up to 2000 ft/minute rubbing speed with splash lubrication, the aluminum bronzes are restricted to 600-800 ft/minute.

Gear bronzes are a unique material and one should not assume that just any tin-bronze alloy will suffice. It takes a precise blend of tin and copper in the right proportions and cast in a precise manner to create the proper dendritic formation and the correct amounts of the alpha/delta phase so necessary to make a *good* bearing bronze for worm gearing. Much of the secret in obtaining an effective gear bronze results in the structural formation of the material with hard load carrying phases of a high tin concentration finely dispersed throughout a matrix of bronze which tends to cushion the tin particles. (Fig. 7) (Fig. 8)

This provides an effective bearing surface to carry the load. The copper phase should be sufficiently ductile so that it will yield and flow with that load. This "flow-ability" assists in creating the broad area of contact so prevalent in double enveloping worm gearing and also compensates for minor errors in manufacturing and assembly.

Most gear bronzes have physical characteristics of 45,000-50,000 PSI tensile, 22,000-25,000 PSI yield, 10%-12% elongation and hardness ranges of 85-120 BHN. For a comprehensive listing of gear bronzes and their physical characteristics, reference should be made to AGMA 240.01.

Fig. 7—Dendritic structure in alpha matrix tin bronze centrifugally cast-50x.

Fig. 8—Delta particles in dendritic alpha matrix tin bronze centrifugally cast-50x.

Manufacturing Methods For Cone Drive Double Enveloping Worm Gearing

Manufacturing of double enveloping worm gearing beyond normal preparatory stages involves basically:

1. Hobbing
2. Generating
3. Matching and lapping
4. Assembly

Hobbing and generating is done on the same machine. On gearsets up through 18.000" center distance, there are specially designed hobber/generators which mount the work piece and the cutting tool in exact position and location with respect to each other. (Fig. 9)

Center distance is set by dial indicators, side and end positions are set by the use of gage blocks. The radial feed mechanism and rotational feed mechanism are geared into the machine so that the hobber/generator is in effect semi-automatic in operation. Gearsets larger than 18.000" center distance up

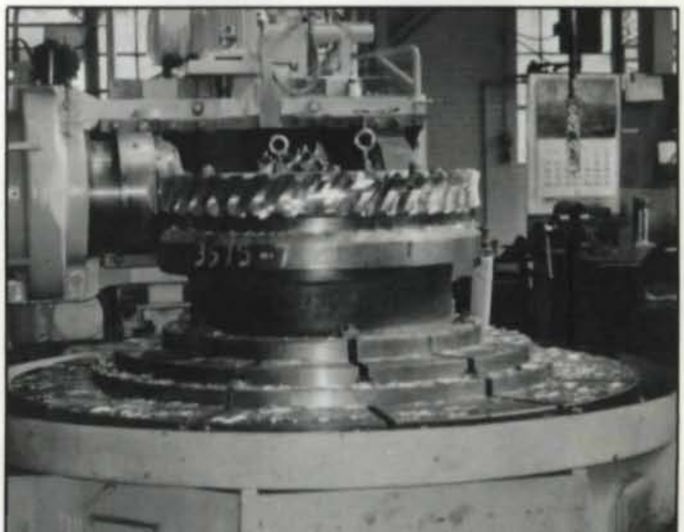


Fig. 9—Double enveloping worm-gear-hobber-generator.

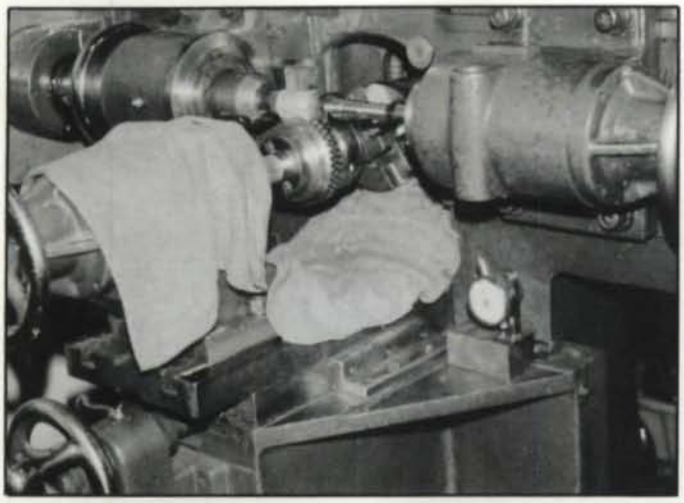


Fig. 10—Matching lapping machine.

to 52.000" center distance are manufactured on commercial hobbers which have been considerably altered to suit double enveloping gearing and have rotational feed boxes attached to the machine system.

Hobs and cutters are manufactured at our own facility where material, heat treat, back off, grinding, spacing, lead, pressure angle, etc., can be controlled within exacting limits.

Cone double enveloping worms are not ground. This precludes certain material selections and hardness. Cone double enveloping worms are made from 4150 resulphurized steel heat treated to Rc 35-38 or nitrided with a 87-15N case (28-30 Rc core). Generating steel this hard has an obvious effect on cutter life. Hob and cutter life controls the processing after hobbing and generating. Tooth form will change as tools dull. As a result, within the usable life of a sharpening cycle, some tooth forms will be minutely different than others.

To provide the customer with a uniform and consistent quality gearset, most double enveloping worms and gears are matched on center distance, end position, and side position for a quality check. (Fig. 10)

While this is not a mandatory procedure, in that many customers purchase worms and gears separately and use them at random, it is, nevertheless, the ideal procedure to achieve the high performance capabilities of the double enveloping product. When a worm and gear do not produce contact patterns up to standards, the worm and gear are lightly lapped to produce the desired contact. Such a lapping procedure is very minimal—no more than 1 - 2in-lbs. load and 3 - 8 gear revolutions in each direction of rotation. Total lapping time is generally less than one (1) minute.

Assembly of double enveloping worm-gears, to *ideal* contact, necessitates that the set be on true gear side position, worm end position, and center distance. We recommend, for *maximum initial performance* of a double enveloping worm gearset, that center distance, gear side position and worm end position tolerances be in accordance with the following table.

Center Distance	Tolerance
Up to 6"	±.001
6" to 12"	±.002
Over 12"	±.003

Comments have been made about the additional effort involved in controlling worm end position. If you provide control of center distance and gear side position, and both cylindrical and double enveloping worm gearing requires that you do so, then controlling worm end position is only one more step, and certainly the technique is readily available. We are consistently holding worm end position along with center distance and side position while producing 2000, and more, speed reducers per month at our facility. The effort in providing worm end position is minimal when compared to the benefits of the greatly increased load capacity which will be realized with double enveloping worm gearing. In addition, we have found that double enveloping worms and gears — off slightly on center distance, worm end position, or gear side position (or any combination) rapidly seat against each other during the break-in and are soon providing full contact and design capacity. The straight sided conjugate form, parallel to a common base diameter, enables the double enveloping design of worm gearing to regenerate to an ideal matching relationship.

Worm gear efficiency is worthy of discussion. Testing has established that center distance for center distance and ratio for ratio cylindrical worm gearing and double enveloping gearing will have the same basic efficiency values. Since Cone Drive manufactures not only worm gearing but helical and herringbone gearing as well, we feel that we are also somewhat qualified to compare worm gear efficiencies with helical efficiencies. That worm gear efficiency is less than that of helical gearing will not be disputed. However, the overall variations between these two (2) types of gear efficiencies are not as great as generally assumed. For years it has been commonly stated that helical and herringbone gearing have efficiency losses of 1%-2% at mesh. This is a reasonably valid value. However, what most people fail to consider is the fact that helical and herringbone gearing at 1%-2% mesh inefficiency must be installed in gearboxes where bearing losses, oil seal drag and churning losses within the bearbox add to the mesh loss. When these losses are added to mesh loss, then the overall gearbox efficiency is considerably less.

We consider that a single reduction helical or herringbone gearbox will have approximately 95% overall efficiency, a double reduction will have 92%-93%, a triple reduction will have 89%-90% and a quadruple reduction will have approximately 86%-87%. It should also be recognized that the higher ratio helical and herringbone gearboxes require multiple gearsets and each one of these gearsets must be suitably mounted in its own bearing mounting arrangement. The multiplicity of bearings and gears compound the bearing and churning losses within the gearbox.

A worm gearbox within the normal ratio range of 5:1-100:1 generally accomplishes the ratio change using a single gearset. The churning losses will be higher on a 5:1 design but the churning losses on the 100:1 design will be substantially reduced because of the very low rotational speed of the gear. We consistently find 5:1 ratio worm gearboxes running at 95% efficiency, which compares on a par with helical boxes of this ratio. With a 20:1 ratio reducer, the helical box will have either two

(Continued on page 45)

YOU CAN'T AFFORD TO MISS THIS CHANCE

to receive your 1 year's FREE SUBSCRIPTION to GEAR TECHNOLOGY, The Journal of Gear Manufacturing

Just complete and mail the response card

ALL INFORMATION MUST BE FILLED OUT TO QUALIFY FOR FREE SUBSCRIPTION

☐ YES! SEND GEAR TECHNOLOGY ***FREE!***

PRIMARY BUSINESS ACTIVITY AT THIS LOCATION (Circle One)

A. Manufacturer with Gear Department
B. Gear Manufacturer—Job Shop—Contract Work
C. Gear Machine Tool Manufacturer
D. Machine Tool Distributor/Agent
E. School/Research Organization
F. Government Agency
G. Industry Association
H. Advertising Agency
I. Other: Please specify ______________________

____

Does your company have a specialty, and if so, what is it? ______________________

____

____

Does your company have an unusual machine? Yes ☐ No ☐

Does your company do sub contract work? Yes ☐ No ☐

How many employees at this location?

☐ 1-20 ☐ 21-50 ☐ 51-100 ☐ 101-500 ☐ 500 and over

Your job function (circle all that apply)

A. President/Owner/General Manager
B. Manager __________ Dept. (specify)
C. Design Engineer
D. Process Engineer
E. Sales
F. Consultant
G. Educator
H. Other: Please specify ______________________

Editorial scores: Help us to serve you better by rating feature articles in this issue. Refer to numbers at the end of each article then mark this card accordingly.

	E-1	E-2	E-3	E-4	E-5	E-6	E-7	E-8	E-9	E-10
Excellent	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐
Good	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐
Fair	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐
Poor	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐

In which of the following ways are you personally involved with gear machinery?

A. Use them
B. Recommend them
C. Establish specifications
D. Approve purchase
E. Acquire them
F. Other involvement __________

____

G. No involvement

Are there any other individuals at this location that may qualify for a complimentary subscription to Gear Technology?

Name __________ Title __________

Name __________ Title __________

Name __________ Title __________

Please complete or attach business card.

Name __________

Company __________

Div. of __________

Co. Address __________

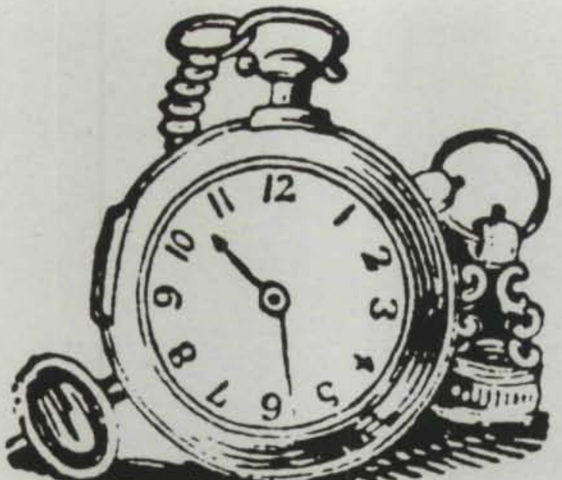
City __________ State __________ Zip __________ Phone __________

Your Signature __________ ***Your Title*** __________

PLEASE SIGN HERE TO CONTINUE (OR BEGIN) RECEIVING GEAR TECHNOLOGY FREE.

Advertisements Circle number(s) for additional information

A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9	A-10
A-11	A-12	A-13	A-14	A-15	A-16	A-17	A-18	A-19	A-20
A-21	A-22	A-23	A-24	A-25	A-26	A-27	A-28	A-29	A-30



BE UP TO THE MINUTE WITH GEAR TECHNOLOGY

FREE ONE YEAR SUBSCRIPTION

(Value $27.00)

And all *you* have to do is *complete the response card to qualify!*

READ about solving specific problems, explanations of new technology, discussions of new techniques, AND stay up to the minute with the Gear Industry.

ALL INFORMATION MUST BE FILLED OUT TO QUALIFY FOR FREE SUBSCRIPTION

NO POSTAGE NECESSARY IF MAILED IN THE UNITED STATES

BUSINESS REPLY MAIL

FIRST CLASS PERMIT NO. 944 ELK GROVE VILLAGE, IL 60007

POSTAGE WILL BE PAID BY ADDRESSEE

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

P.O. Box 1426
Elk Grove, Illinois 60007

TEAR HERE AND MAIL — NO POSTAGE REQUIRED

Gear Generating Using Rack Cutters

by
P.G. Miller
J. Parkinson & Son
England

Universal machines capable of cutting both spur and helical gears were developed in 1910, followed later by machines capable of cutting double helical gears with continuous teeth. Following the initial success, the machines were further developed both in England and France under the name *Sunderland*, and later in Switzerland under the name *Maag*.

This article explains the fundamental advantages which have made rack-shaped cutting machines successful, particularly in the production of large gears.

Basic Principles

A basic rack is essentially the starting point of the geometry of all conjugate gear forms, and from this basic rack, a series of gears may be graphically developed. If the rack has straight-sided teeth, the gears developed from it will have involute form — now the most common form of gear tooth. The development of an involute gear is illustrated in Fig. 1.

Such a straight sided basic rack may be used to develop both gears with standard proportions and also, within limits, gears with larger or smaller diameters than standard. Advantage is frequently taken of this fact to achieve stronger teeth by increasing the outside diameter, *ie* lengthening the addendum of the teeth. As can be seen in the example illustrated in Fig. 2, although both pairs of gears mesh correctly together, the 12-tooth pinion shown to the right of the diagram exhibits a much stronger tooth form as a result of 'addendum modification'. If the generating process is used to produce gears developed from the basic rack, the foregoing principles are fully exploited and the same cutter may be used to cut all numbers of teeth.

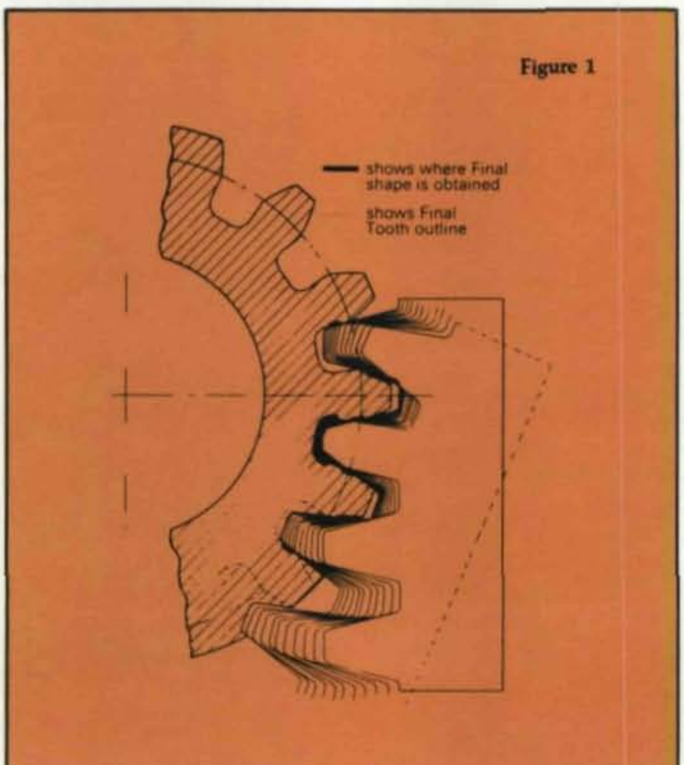


Figure 1

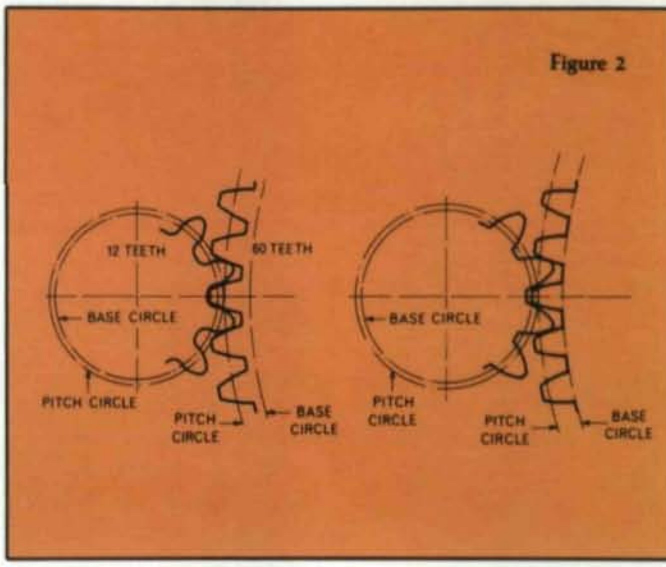


Figure 2

To generate accurate conjugate gears it is necessary to use cutters which have the same characteristics as the basic rack. A rack-shaped cutter is thus the logical first choice; therefore, the gears are produced virtually by the basic rack itself, simply and directly, without the use of any intermediary.

Practical Advantages

Important as the theoretical justification is, the main considerations for the use of rack-shaped cutters are practical ones. There is a considerable advantage with regard to cost due to the relative ease of achieving accuracy in manufacture. Some hobs are five times as expensive, and some pinion-type cutters twice as expensive. Additionally, rack-shaped cutters, with modified or special tooth profiles, may be manufactured with little difficulty. Maintenance costs also are much lower than for other types of cutter, as sharpening is not difficult, and does not require expensive equipment as a conventional surface grinding machine may be used.

There is no alteration to the tooth form when rack-shaped cutters are re-ground. By using support plates mounted behind the cutter, cutters can by repeatedly sharpened until they are only 5 mm ($\frac{3}{16}$ in.) thick, giving almost complete utilization.

Due to the very rigid mounting of the rack-shaped cutter, high rates of metal removal are possible. Even the largest pitches may be generated, whereas the use of 'standard' pinion-type cutters is largely restricted to gears of limited pitch. Lastly, compact integral spur or helical gearing which cannot be made by using a hob, are easily manufactured using a rack-shaped cutter.

Automatic Indexing

During any gear generating process the movements of cutter and gear must be synchronised. The cutter moves tangentially to the gear, *ie* a rolling motion, and as it is mounted on a reciprocating slide it is continually planing the gear and generating teeth and spaces.

On first examination, it would appear that a cutter would always have to be equivalent in length to the gear circumference. Such a long cutter would not be practicable, however, except when cutting small pinions. Therefore, the cutters index automatically after the rolling motion has operated for a distance of one or more pitches; then the rolling motion commences once more. The cutting action of the rack-shaped cutter is similar to other machining operations, as both roughing and finishing cuts are required, the number of cuts depending on the specification of the material, the amount to be removed and the surface finish required.

Wide Range of Gear Cutting

Gear generating machines can be used for the economic production of a wide range of gears. When cutting helical gears, *there is infinite choice of helix angle, from zero up to the maximum* capacity of the machine. No special equipment need be used; the cutter slide is simply set to the correct helix angle, either right hand or left hand as required.

When cutting coarse pitch spur gears a double-acting cutter box is a most important aid to efficient production. The boxes are designed to achieve heavy metal removal on coarse pitches, *ie* over 12 Mod (2 DP). In the double-acting process two cutters are mounted back to back, one cutting the sides of the tooth on the forward stroke and the other the root of the tooth on the return stroke. Thus, there is virtually continuous cutting, and as cutting loads are more evenly distributed high metal removal rates are possible.

Special profiles and a wide range of other types of gears and components may be easily produced, including integral cluster gears, pinions with long shaft extension, chain wheels, ratchet wheels, fluted rolls and splined shafts.

For cutting double helical gears with either a narrow gap or with continuous teeth, special purpose cutter slides designed with two slides inclined at fixed helilx angles (usually 30°) have been developed (Fig. 3). Matched pairs of cutters with inclined teeth are utilized and arranged to cut each hand of helix alternately. As both helices are finished simultaneously, matching is easily achieved and rotational errors between the two halves are eliminated.

The facility to produce double helical gears without a gap enables gears of maximum strength and load-carrying capacity to be designed within a given width. Continuous teeth have proved to give greater resistance to shock loads, hence, both strength and smoothness of operation are achieved.

Purpose-Built Machines

On some machines, the gear is mounted with its axis horizontal so that it is cut as it will run (see Fig. 4). This layout allows both large wheels and small pinions to be cut on the same machine, without sacrificing the accuracy of the large gears; it permits double-acting cutting to be carried out and lends itself to the cutting of double helical gears. Swarf automatically falls away from the cutting area into the base, where it can easily be removed. As the axis of the work spindle is horizontal, it is easy to accomodate gears and pinions with long shaft extensions.

The course of the cutter across the whole face of the gear is determined accurately by the cutter slide guides; this accuracy of helix is maintained without adjustment over a long period. The involute form is more readily generated using a cutter with straight sides. Therefore, a given surface accuracy may be achieved, even when using high feed rates.

Rugged construction of the machines, with cutters of virtually 'built-in' design, permits high metal removal rates which may be further improved where it is possible to use a double-acting cutter box.

Fig. 3 — below

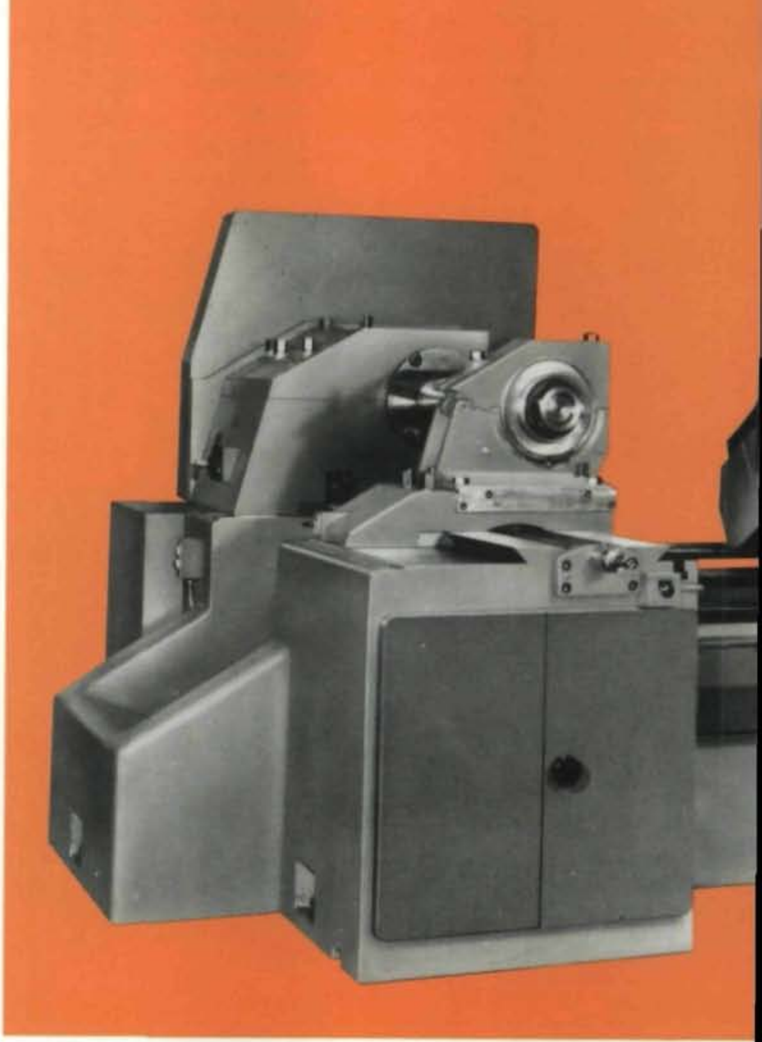


Fig. 4 — (Right) Illustrates a G250 cutting a large spur gear of 76" diameter 9" face width and 1 DP pitch, for a crane in a steelworks.

Recent Developments

Following the introduction of a new range of roughing cutters the cutting times on coarse pitch spur gears, *ie* over 16 Mod (1½ DP), have been reduced considerably. The principle of the new process is to reduce the cutting stresses normally set up during roughing, by using cutters of thin form and low pressure angle in conjunction with serial cutters of similar form. When the tooth has been fully 'roughed out', cutting is completed by the use of standard cutters.

Using normal methods of cutting, a spur gear of 25 Mod (1 DP), 3650 mm (12 ft.) *d* and 610 mm (2 ft.) face width, previously took seven cuts, *ie* a floor-to-floor time of 90 h. This has now been reduced to 64 h, a saving of 35 percent.

Current developments include new designs of both roughing and finishing cutters for continuous tooth double helical gears. These promise improvements in both production rate and surface finish.

Scoring Load Capacity of Gears Lubricated With EP-Oils

by
H. Winter and K. Michaelis
Technical University of Munich

Abstract

The Integral Temperature Method for the evaluation of the scoring load capacity of gears is described. All necessary equations for the practical application are presented. The limit scoring temperature for any oil can be obtained from a gear scoring test. For the FZG-Test A/8.3/90 acc. DIN 51 354 and the Ryder Gear Test acc. FTM STD Nr. 791, graphs for the direct evaluation of the scoring temperature as a function of oil viscosity and test scoring load are given.

The method is compared with the Total Contact Temperature Criterion acc. Blok (1)—the alternate procedure to the Integral Temperature Method as standardized in ISO DP 6336 part IV—and the Scoring Index Method acc. Dudley (2). Comparative calculations for practical gears with and without scoring damages showed good correlation with experience for the Integral Temperature Criterion.

Introduction

In different fields of application, the load carrying capacity of gears is limited by scoring damage.

In highly loaded, case carburized turbine gears, the normally used mineral oils with rust and oxidation inhibitors do not always give sufficient scoring protection. On the other hand, the necessary EP-additives adversely affect the anti-oxidation, anti-foam, etc. properties, so that the life of the oil may be reduced.

In the case of carburized marine gears with diesel engine drives, motor oils are frequently also used for the gears. These oils do not always provide sufficient scoring load capacity.

Also, in some types of locomotive drives, the same lubricant is used for the hydraulic torque converter and the gears. For high efficiency of the hydraulics, low viscosity oils have to be used. Because of then reduced film thickness between the gear flanks, EP-additives have to compensate for viscosity.

In these cases, a reliable scoring load calculation could help to define the necessity of EP-additives and their percentage.

The type of damage occuring in the range of medium to high speed gears is the so called "warm" scoring (Fig. 1), which is covered by this paper. "Cold" scoring, which can be observed in the area of low speed, low quality, through hardened gears of low hardness, has to be handled with some different method.

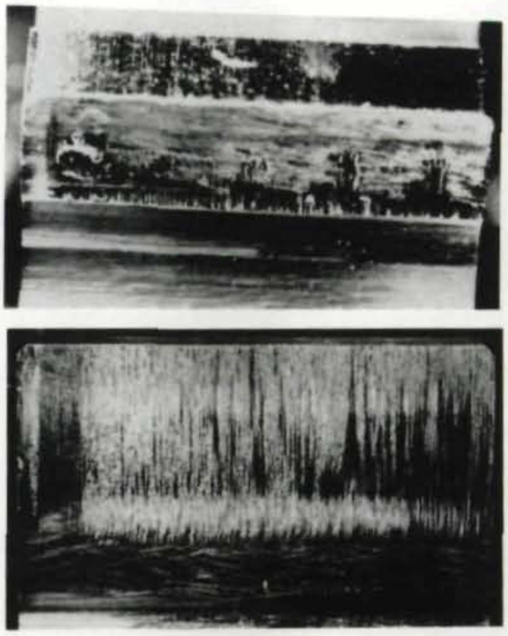


Fig. 1—Scoring Damage of Tooth Flank

Integral Temperature Method

Principle

Derived from hundreds of tests with different gear oils, different gear geometries, materials, operating speeds, temperatures, etc. in back-to-back gear test rigs of center distances a = 91.5; 140 and 200 mm, a mean surface temperature on the engaging flanks has been established as a governing criterion of the scoring damage.

For an assumed load distribution along the path of contact as shown in Fig. 2, the flash temperature distribution acc. Blok (1) can be calculated. The sum of the

AUTHORS:

DR. KLAUS MICHAELIS *studied machine engineering at the Technical University of Munich. Since 1970, he has worked at the Laboratory of Gear Research and Gear Design (FZG) at this university. In 1977 he became Chief Engineer. He specializes in lubrication, scoring, and wear of gears.*

PROF.DR.-ING. HANS WINTER *has studied as an associate of Prof.Dr.-Ing.h.c. Gustav Niemann. He received his Doctoral degree at the Technical University of Munich. In 1956, he began his work in the German gear industry: Zahnradfabrik, Friedrichshafen (Calculation, research, manufacturing), Demag, Duisburg (Research, development, design, selling). Since 1969 he has been the head of the Laboratory of Gear Research and Gear Design (FZG) at the Technical University of Munich.*

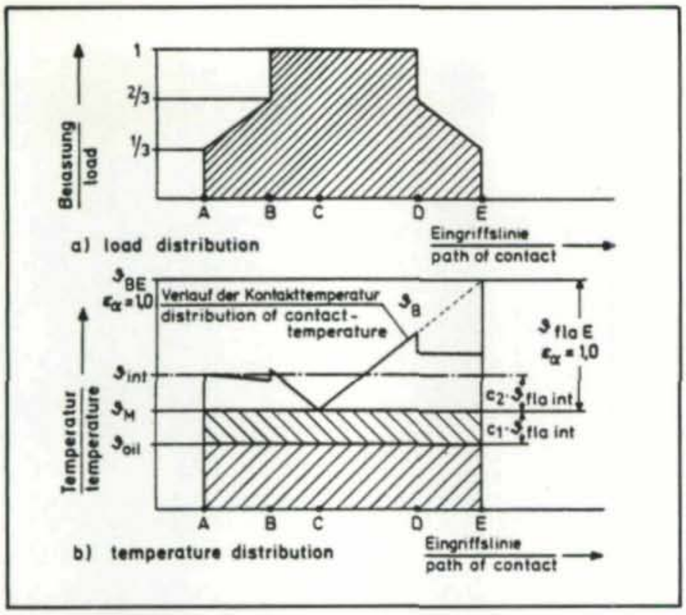


Fig. 2—Load and Temperature Distribution acc. Blok (schematic)

mean flash temperature (multiplied by a weight factor) and the gear bulk temperature is defined as the integral temperature. The weight factor accounts for the certainly differing influences of the real bulk temperature and the mathematically established mean flash temperature on the scoring damage (3).

The integral temperature of a practical gear must not exceed a critical value which is independent of the operating conditions and constant for a given material-lubricant combination. This limiting value, the scoring temperature, can be calculated according to the same set of equations introducing the parameters of any gear scoring test of the oil under consideration. If both, the actual gear and the test gear, differ in material or heat treatment, empirical correction factors have to be introduced.

Integral Temperature Rating

The integral temperature is calculated in the transverse section of the gear pair.

*) $$\vartheta_{int} = \vartheta_M + C_2 \cdot \vartheta_{fla\,int} \qquad (1)$$

*) For symbols and units see table 1.

**) For gears with internal power distribution (e.g. planetary gears) a non uniform load distribution has to be considered. In these cases $F_t\ K_A$ has to be replaced by $F_t\ K_A\ K_\gamma$.

***) The evaluation of a load quotient can be approximated by

$$S_{SL} = \frac{W_{t\,max}}{W_{t\,eff}} \approx \frac{\vartheta_{S\,int} - \vartheta_{oil}}{\vartheta_{int} - \vartheta_{oil}}$$

The weight factor, as described above, has been determined from test results $C_2 = 1.5$.

The mean flash temperature, $\vartheta_{fla\,int}$, can be approximated by the determination of the flash temperature at the tip of the pinion, $\vartheta_{fla\,E}$, for a contact ratio, $\epsilon_\alpha = 1.0$ (no load sharing) and the contact ratio factor X_ϵ (see Fig. 2).

$$\vartheta_{fla\,int} = \vartheta_{fla\,E} \quad X_\epsilon \qquad (2)$$

The nominal flash temperature, $\vartheta_{fla\,E}$, at the pinion tip is calculated acc. Blok (1)

**

$$\vartheta_{fla\,E} = \mu_B \cdot X_M \cdot X_{BE} \cdot X_{\alpha\beta} \frac{\left(\frac{F_t}{b} \cdot K_A \cdot K_{B\beta} \cdot K_{B\alpha l} \cdot K_{B\gamma}\right)^{3/4} \cdot v^{1/2}}{|a|^{1/4} \cdot X_Q \cdot X_{Ca}} \qquad (3)$$

The scoring temperature is evaluated using the same equations for the conditions of a gear oil test

$$\vartheta_{S\,int} = \vartheta_{MT} + C_2 \cdot X_{W\,rel\,T} \cdot \vartheta_{fla\,int\,T} \qquad (4)$$

The safety factor against scoring damage is defined as a temperature quotient

$$S_S = \vartheta_{S\,int} / \vartheta_{int} \qquad *** \qquad (5)$$

From recalculation of practical gears, safety factors, less than unity, refer to a high risk of scoring, while safety factors over 2.0 indicate a low scoring risk. Gears with calculated safety factors between 1.0 and 2.0 are of a borderline type. They can be operated without scoring damage when a good load distribution across the face width, smoothed, run-in surfaces, etc. are obtained. In cases where, e.g., new manufactured flanks without a run-in process are operated under nominal load, scoring can occur.

Influence Factors

The coefficient of friction, μ_B, is calculated as a mean value along the path of contact. It can be approximated by introducing the parameters of the pitch point

$$\mu_B = 0.045 \left[\frac{(F_t/b) \cdot K_A \cdot K_{B\beta} \cdot K_{B\alpha}}{\cos\alpha_{wt} \cdot v_{\Sigma C} \cdot \rho_{Cn}}\right]^{0.2} \cdot \eta_\theta^{-0.05} \cdot X_R$$

$$= \mu_m \cdot (K_{B\beta} \cdot K_{B\alpha})^{0.2} \qquad \text{with } \mu_m \leq 0.2 \qquad (6)$$

$F_t/b = 150$ N/mm is introduced for $F_t/b \leq 150$ N/mm.

Eq. (6) for the evaluation of the coefficient of friction has only been introduced in the DIN standard, not yet in the ISO document. Recent investigations showed a good correlation of μ_m with practical experience and measurements of gear power loss and efficiency (Fig. 3) so that Eq. (6) can also be used for the determination of absolute frictional losses in gears (4).

The overload factors K_A, $K_{B\beta}$ and $K_{B\alpha}$ can be determined acc. ISO DP 6336 Part I for surface durability $K_{H\beta}$ and $K_{H\alpha}$.

The rolling speed on the pitch circle is

$$v_{\Sigma C} = 2 \cdot v \cdot \sin\alpha_{wt} \tag{7}$$

For the speed range v below, 1 m/s and above 50 m/s the evaluation of μ_B becomes uncertain and is no longer based on experimental data. In this range μ_B is assumed to be constant, with $v = 1.0$ m/s for $v < 1.0$ m/s and $v = 50$ m/s for $v > 50$ m/s to be introduced into Eq. (7). The radius of curvature in the normal section is

$$\rho_{Cn} = 0.5 \frac{\tan\alpha_{wt}}{\cos\beta_b} d_{b1} \frac{u}{u + 1} \tag{8}$$

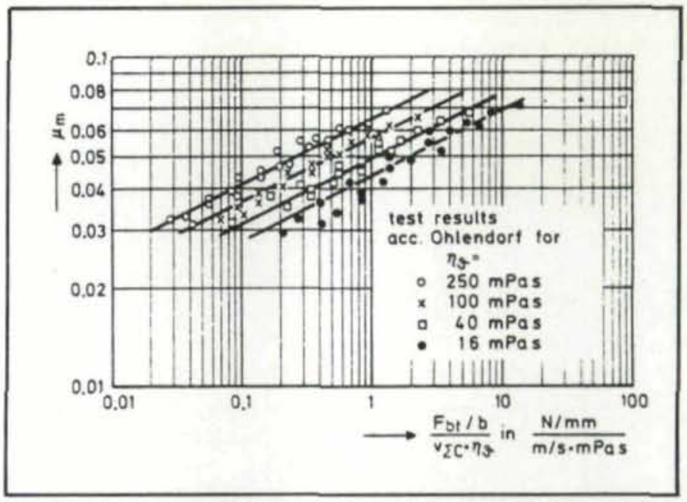


Fig. 3—Comparison of Calculated Coefficient of Friction and Test Results

Table 1: Symbols, Terms, and Units

a	centre distance	mm
b	facewidth	mm
c_γ	mesh stiffness	N/(μm mm)
c'	single stiffness	N/(μm mm)
C_a	amount of tip relief	μm
$C_{1,2}$	constants	–
d	reference diameter	mm
d_b	base diameter	mm
d_{Na}	effective tip diameter	mm
E	Young's modulus	N/mm²
F_t	tangential force, reference circle	N
F_{bt}	tangential force, base circle	N
K_A	application factor	–
$K_{B\alpha}$	transverse load distribution factor	–
$K_{B\beta}$	logitudinal load distribution factor	–
$K_{B\gamma}$	helical load distribution factor	–
K_γ	load distribution factor for more than one mesh	–
m	module	mm
R_a	arithmetic average roughness (CLA)	μm
S_B	safety factor, flash temperature	–
S_S	safety factor, integral temperature	–
T_1	pinion torque	N m
u	gear ratio $f_1/f_2 \geq 1$	–
v	linear speed at reference circle	m/s
v_Σ	rolling speed	m/s
w_t	specific load including overload	N/mm
X_{BE}	geometrical factor, pinion tip	–
X_{Ca}	tip relief factor	–
X_M	thermal flash factor	$K\,N^{-3/4} s^{1/2} m^{-1/2} mm$
X_Q	rotation factor	–
X_R	roughness factor	–
X_S	lubrication factor	–
X_W	welding factor	–
$X_{\alpha\beta}$	angle factor	–
X_ε	contact ratio factor	–
z	number of teeth	–

α	pressure angle	°
β_b	base helix angle	°
ϵ_1	addendum contact ratio of pinion	–
ϵ_2	addendum contact ratio of wheel	–
ϵ_α	transverse contact ratio	–
ϵ_γ	total contact ratio	–
ϑ_B	instantaneous contact temperature	°C
ϑ_{fla}	flash temperature	K
$\vartheta_{fla\,E}$	flash temperature, pinion tip	K
$\vartheta_{fla\,int}$	mean flash temperature	K
ϑ_{int}	integral temperature	°C
ϑ_M	bulk temperature	°C
ϑ_{oil}	oil temperature	°C
$\vartheta_{S\,int}$	scoring temperature	°C
η_ϑ	dynamic oil vicosity at ϑ_{oil}	mPa s
μ_B	coefficient of friction, scoring	–
μ_m	mean coefficient of friction	–
ν	Poisson's ratio	–
ν_{40}	kinematic viscosity at 40 °C	mm²/s
ρ	radius of curvature	mm

Suffixes

b	base circle
C	pitch point
eff	effective values
E	pinion tip
max	maximum
n	normal section
t	transverse section
T	test gear
w	working
1	pinion
2	wheel

The roughness factor accounts for surface roughness

$$X_R = 3.8\,(R_a/d_1)^{0.25} \tag{9}$$

$$\text{with } R_a = 0.5 \cdot (R_{a1} + R_{a2}) \tag{10}$$

In Eq. (10) the CLA-values of the new manufactured flank have to be introduced. An amount of normal run-in is included in Eq. (9).

The thermal flash factor X_M depends on the elastic and thermal properties of the gear materials. For gears made out of steel, mean values of conductivity λ_M = 50 N/(s.K); density ρ_M = 7.85 kg/dm^3, specific heat capacity c_M = 485 N m/(kg.K); E = 206,000 N/mm^2, and ν = 0.3 can be introduced

$$X_M = 50\ \mathrm{K} \cdot \mathrm{N}^{-3/4} \cdot \mathrm{s}^{1/2} \cdot \mathrm{m}^{-1/2} \cdot \mathrm{mm} \tag{11}$$

For non steel materials for pinion and/or gear see ISO DP 6336, Part IV.

The geometrical factor X_{BE} takes account for the Hertzian stress and the contact time at the pinion tip E.

$$X_{BE} - 0.5\sqrt{\frac{|Z_2|}{Z_2}(u+1)}\,\frac{\sqrt{\rho_{E1}} - \sqrt{\rho_{EZ}/u}}{(\rho_{E1} \cdot |\rho_{EZ}|)^{1/4}} \tag{12}$$

$$\text{with } \rho_{E1} = 0.5 \cdot \sqrt{d_{Na1}^2 - d_{b1}^2} \tag{13}$$

$$\text{and } \rho_{EZ} = a \cdot \sin\alpha_{wt} - \rho_{E1} \tag{14}$$

in the transverse section. Eqs. (12, 13, 14) are valid for internal and external cylindrical gears.

The angle factor $X_{\alpha\beta}$ accounts for the recalculation of the acting normal load to the circumferential load at the pitch cylinder.

$$X_{\alpha\beta} = 1.22\,\frac{\sin^{1/4}\alpha_{wt} \cdot \cos^{1/4}\beta_b}{\cos^{1/4}\alpha_t \cdot \cos^{1/2}\alpha_{wt}} \tag{15}$$

For approximate calculations and a pressure angle $\alpha = 20°$, $X_{\alpha\beta}$ can be set unity.

The helical load distribution factor, $K_{B\gamma}$, accounts for the empirical decrease of scoring load capacity for increasing total contact ratio.

$$K_{B\gamma} = 1.0 \quad \text{for} \quad \varepsilon_\gamma \leq 2.0$$

$$K_{B\gamma} = 1 + 0.2 \cdot \sqrt{(\varepsilon_\gamma - 2)(5 - \varepsilon_\gamma)} \ \text{for } 2 < \varepsilon_\gamma < 3.5 \tag{16}$$

$$K_{B\gamma} = 1.3 \quad \text{for} \quad \varepsilon_\gamma \geq 3.5$$

The rotation factor, X_Q, considers the effect of a simultaneous load impact and high sliding at the beginning of the mesh. For gears with normal addendum modification

$$X_Q = 1.0 \text{ for } 1/1.5 < \varepsilon_1/\varepsilon_2 < 1.5 \tag{17a}$$

$$\text{with } \varepsilon_{1,2} = \frac{|z_{1,2}|}{2\pi} \cdot \sqrt{\left(\frac{d_{Na1,2}}{d_{b1,2}}\right)^2 - 1} - \tan\alpha_{wt} \tag{18}$$

In the case where the approach path of contact of the driving partner exceeds 1.5 times the recess path, X_Q is set 0.6.

$$\begin{aligned} &X_Q = 0.6 \text{ for driving pinion and } \epsilon_2 \geq 1.5\epsilon_1 \\ &X_Q = 0.6 \text{ for driving wheel and } \epsilon_1 \geq 1.5\epsilon_2 \end{aligned} \tag{17b}$$

In all other cases X_Q = 1.0.

The tip relief factor X_{Ca} accounts for the benefit of a profile modification in the area of high sliding (Fig. 4) acc. Lechner(5). Tip relief is only effective up to the amount where it compensates tooth deflection under load

$$X_{Ca} = 1 + 1.55 \cdot 10^{-2} \cdot \epsilon_{max}^4 \cdot C_a \tag{19}$$

with ϵ_{max} as the maximum value of ϵ_1 or ϵ_2 acc. Eq. (18)

$$\varepsilon_{max} = \max \begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \end{Bmatrix} \tag{20}$$

The effective tip relief $C_{a\,eff}$ can be approximated by

$$\begin{aligned} &C_{aeff} = F_{bt} \cdot K_A/(b \cdot c') \text{ for spur gears} \\ &C_{aeff} = F_{bt} \cdot K_A/(b \cdot c_\gamma) \text{ for helical gears} \end{aligned} \tag{21}$$

with the stiffness values c' resp. c_γ acc. to ISO DP 6336 part I.

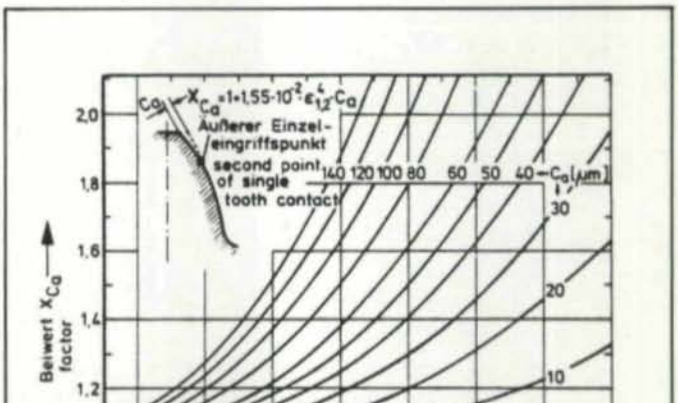


Fig. 4—Influence of Tip Relief

A profile modification increases scoring load capacity only when it is applied to the area of highest risk. In gears with normal addendum modification, the approach path with the load impact of the ingoing mesh is more dangerous. For extreme addendum modification, a tip relief has to be applied to the recess path.

For driving pinion:

$$C_a = \min \begin{Bmatrix} C_{a1} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_1 > 1.5 \cdot \varepsilon_2$$
$$C_a = \min \begin{Bmatrix} C_{a2} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_1 \leq 1.5 \cdot \varepsilon_2 \qquad (22a)$$

For driving gear:

$$C_a = \min \begin{Bmatrix} C_{a2} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_2 > 1.5 \cdot \varepsilon_1$$
$$C_a = \min \begin{Bmatrix} C_{a1} \\ C_{aeff} \end{Bmatrix} \quad \text{for } \varepsilon_2 < 1.5 \cdot \varepsilon_1 \qquad (22b)$$

The contact ratio factor, X_ε, recalculates a mean flash temperature along the path of contact from the maximum temperature, $\vartheta_{fla\ E}$, at the pinion tip for $\epsilon_\alpha = 1.0$. The equations are valid for a load distribution acc. (Fig. 5) and an approximately linear increase of the flash temperature towards the tooth tip and tooth root (Fig. 2).

For $\varepsilon_\alpha < 1.0$:

$$X_\varepsilon = \frac{1}{2\,\varepsilon_\alpha \cdot \varepsilon_1} (\varepsilon_1^2 + \varepsilon_2^2) \qquad (23a)$$

For $1 \leq \varepsilon_\alpha < 2.0$: (23b)

ε_1 and $\varepsilon_2 < 1.0$

$$X_\varepsilon = \frac{1}{2\,\varepsilon_\alpha \cdot \varepsilon_1} [0.7(\varepsilon_1^2 + \varepsilon_2^2) - 0.22 \cdot \varepsilon_\alpha + 0.52 - 0.6\varepsilon_1 \cdot \varepsilon_2]$$

ε_1 or $\varepsilon_2 \geq 1.0$

$$X_\varepsilon = \frac{1}{2\varepsilon_\alpha \cdot \varepsilon_1} [(0.18\varepsilon_{1,2})^2 + (0.7\varepsilon_{2,1})^2 + 0.82\varepsilon_{1,2} - 0.52\varepsilon_{2,1} - 0.3\varepsilon_1\varepsilon_2]$$

with the first index for $\epsilon_1 \geq 1.0$ and the second index for $\epsilon_2 \geq 1.0$.

(23c)

For $2.0 \leq \varepsilon_\alpha < 3.0$ and ε_1 and ε_2 less than 2.0

$$X_\varepsilon = \frac{1}{2\,\varepsilon_\alpha \cdot \varepsilon_1} [(0.44\varepsilon_{1,2})^2 + (0.59\varepsilon_{2,1})^2 + 0.3\varepsilon_{1,2} - 0.3\varepsilon_{2,1} - 0.15\varepsilon_1 \cdot \varepsilon_2]$$

with the first index for $\epsilon_1 > \epsilon_2$ and the second index for $\epsilon_2 > \epsilon_1$.

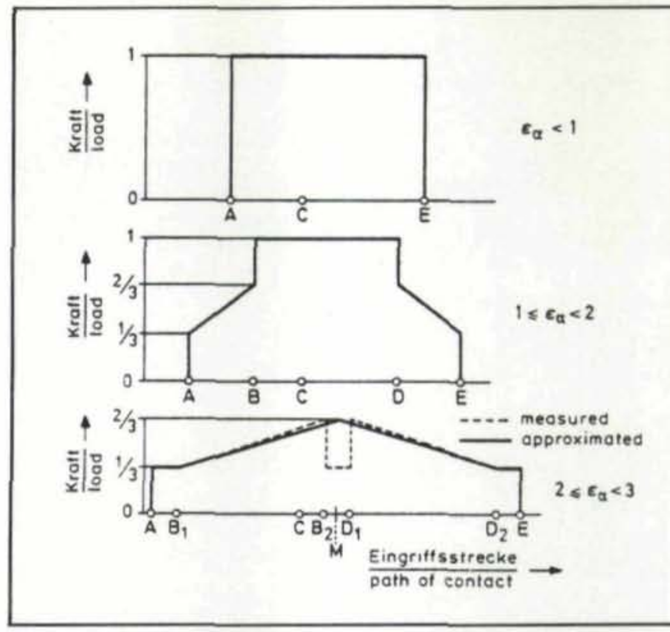


Fig. 5—Approximated Load Distribution as a Function of Contact Ratio

The gear bulk temperature, ϑ_M, is the temperature of the tooth surface before the mesh. It can be measured or calculated according to thermal network theory(6) or finite element methods.

An approximation is given by

$$\vartheta_M = (\vartheta_{oil} + C_1 \vartheta_{fla\,int}) \cdot X_S \qquad (24)$$

where $C_1 = 0.7$ has been determined as a mean value from test results. For gears with more than one engagement on their circumference, higher bulk temperatures than calculated may occur.

The lubrication factor X_S accounts for the better heat transfer in splash lubricated gears compared with jet lubricated. From experience it can be assumed

$X_S = 1.0$ for splash lubrication and

$X_S = 1.2$ for jet lubrication (25)

The choice of the lubrication system, of course, has to be made due to other considerations, e.g., pitch line velocity.

$X_W = 0.45$	for austenitic steel (stainless steel)
$X_W = 0.85$	for steel with content of austenite more than average
$X_W = 1.00$	for steel with normal content of austenite
$X_W = 1.15$	for steel with content of austenite less than average
$X_W = 1.50$	for bath and gas nitrided steel
$X_W = 1.50$	for copper plated steel
$X_W = 1.25$	for phosphated steel
$X_W = 1.00$	for all other cases (e.g. through hardened steel)

Table 2—Estimation of Material Factor X_W

Scoring Temperature Evaluation

The scoring temperature, $\vartheta_{S\,int}$, can be determined according to the same set of equations (2) through (25) introducing the actual parameters of a gear oil test run with the oil under consideration. For differences between the materials or heat treatments of the test and actual gears, a relative correction factor has to be introduced.

$$\vartheta_{S\,int} = \vartheta_{MT} + C_2 \cdot X_{W\,rel\,T} \cdot \vartheta_{fla\,int\,T} \tag{26}$$

$$\text{with } X_{W\,rel\,T} = X_W \,/\, X_{WT} \tag{27}$$

Empirical data on the influence of the material resp. heat treatment are summarized in the welding factor X_W acc. table 2.

From our experience, only scoring tests on test gears can be correlated with the scoring performance in practical gears. Comparative tests with different gear oils, as well as milk and beer, have been made by Vogelpohl(7) and Wirtz(8). Different test principles are shown in Fig. 6. From the results as shown in Fig. 7, it is evident that frequently used test methods as Four Ball Test and Timken Test, do not correlate with the scoring properties in gears. Therefore, only data from oil tests on gears can be introduced into the evaluation of the scoring temperature.

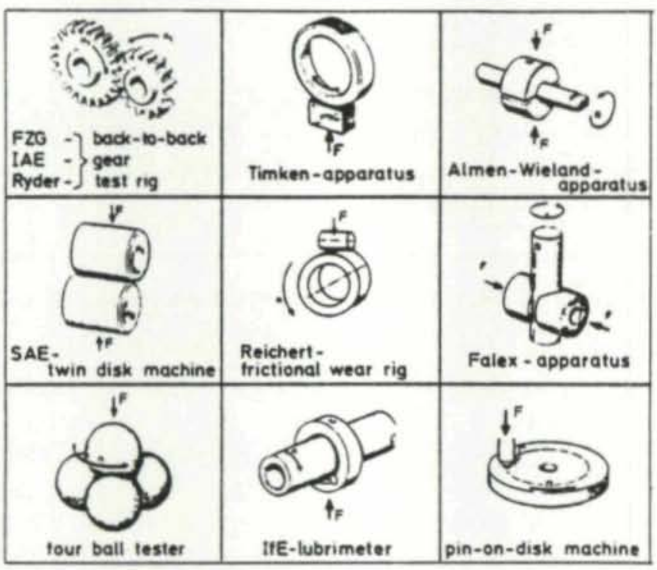


Fig. 6—Gear Oil Test Machines, Principles

An often used method is the FZG-Test A/8.3/90 as standardized in DIN 51 354 (see also AGMA 250.04). From the pinion scoring torque, T_{1T}, or the damage load stage, ϑ_{MT} and $\vartheta_{fla\,int\,T}$, can be taken from Fig. 8 for introduction into Eq. (26).

For computer calculations, the curves can be approximated by

$$\vartheta_{MT} = 80 + 0.23 \cdot T_{1T} \tag{28}$$

$$\vartheta_{fla\,int\,T} = 0.2 \cdot T_{1T} \cdot \left(\frac{100}{\nu_{40}}\right)^{0.02} \tag{29}$$

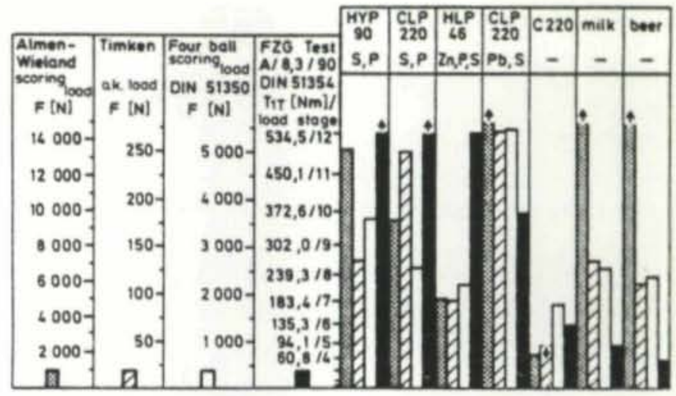


Fig. 7—Evaluation of Scoring Load in Different Test Rigs

The welding factor $X_{WT} = 1.0$ for the FZG-Test.

Starting from a Ryder Gear Test acc. FTM STD Nr. 791, and introducing the constant parameters of gear geometry and of operating conditions to the Eqs. (2) through (25), Fig. 9 is obtained.

The curves can be approximated by

$$\vartheta_{MT} = 90 + 0.0125\,(F_{bt}/b)_T \tag{30}$$

$$\vartheta_{fla\,int\,T} = 0.015(F_{bt}/b)_T \cdot \left(\frac{100}{\nu_{40}}\right)^{0.03} \tag{31}$$

with the Ryder scoring load $(F_{bt}/b)_T$ to be introduced in Eqs. (30, 31) in ppi and the welding factor $X_{WT} = 1.0$.

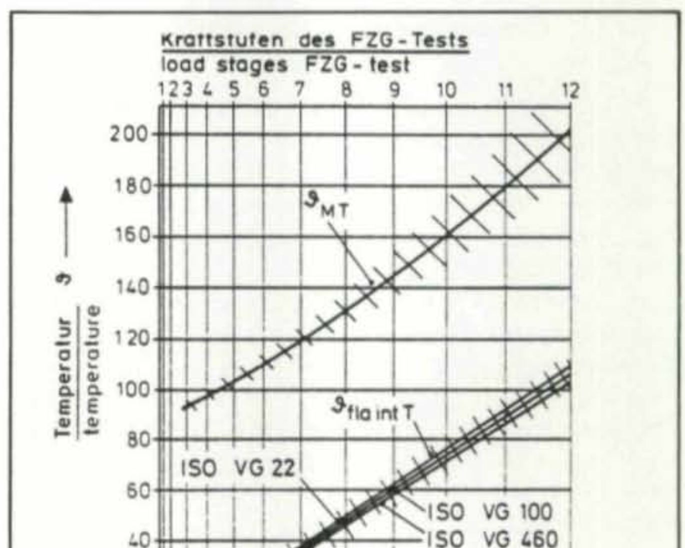


Fig. 8—Scoring Temperature $\vartheta_{S\,int}$ for FZG-Test A/8.3/90

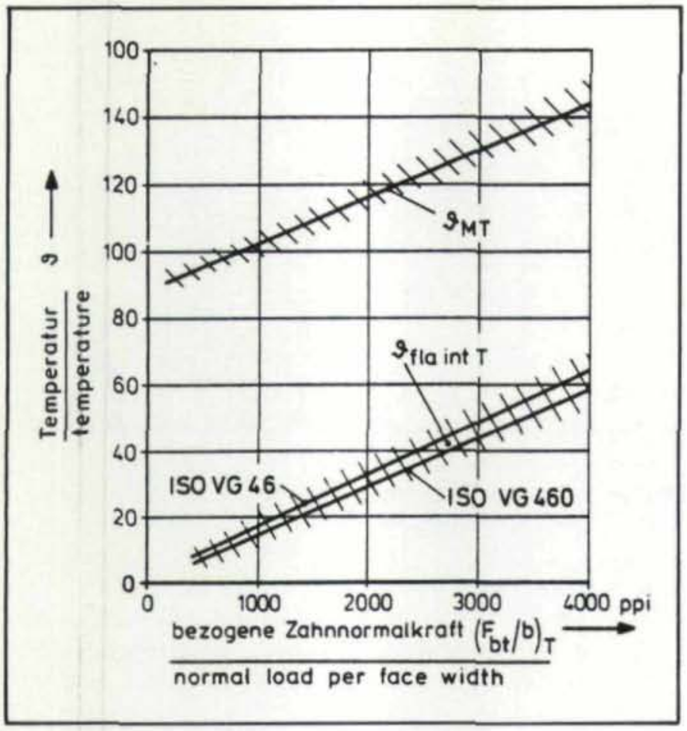


Fig. 9—Scoring Temperature $\vartheta_{S\ int}$ for Ryder-Gear-Test

Thus, test results of different test methods can be used as basic "strength" values. One of the major differences between FZG-Test and Ryder Gear Test is the pitch line velocity.

For high speed application, Ryder results obtained at v = 46 m/s and for low to medium speed application, FZG results at 8.3 m/s are somewhat closer to practical gear conditions and would be preferred, if available.

Comparison with Other Methods

General

An often used method for the evaluation of the risk of scoring damage is the Total Contact Temperature Criterion acc. Blok(1). The method predicts scoring when a maximum, local, instantaneous contact temperature, $\vartheta_{B\ max}$, exceeds a critical value, ϑ_{crit}. The contact temperature distribution along the flank is given by the sum of the constant bulk temperature and the local flash temperature (Fig. 10). The critical value is only dependent on the oil-material combination and independent of geometry and operating conditions. It can be expressed as a function of oil viscosity (Fig. 11). The total contact temperature method is also standardized in ISO DP 6336, and should be applied in parallel whenever possible. After some time of practical experience with both methods, it should be decided which one can be dropped.

The Scoring Index Method acc. Dudley(2) is derived from the Total Contact Temperature Criterion. It uses only the flash temperature part in a simplified way. Therefore, our objections against the Total Contact Temperature Criterion are also valid for the Scoring Index Method, at least to the same degree. Table 3 compares the field of application of the Total Contact

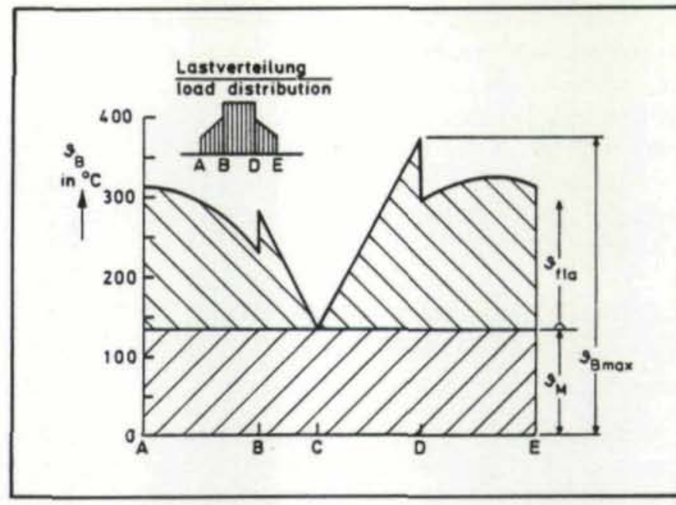


Fig. 10—Temperature Distribution along the Path of Contact acc. Blok

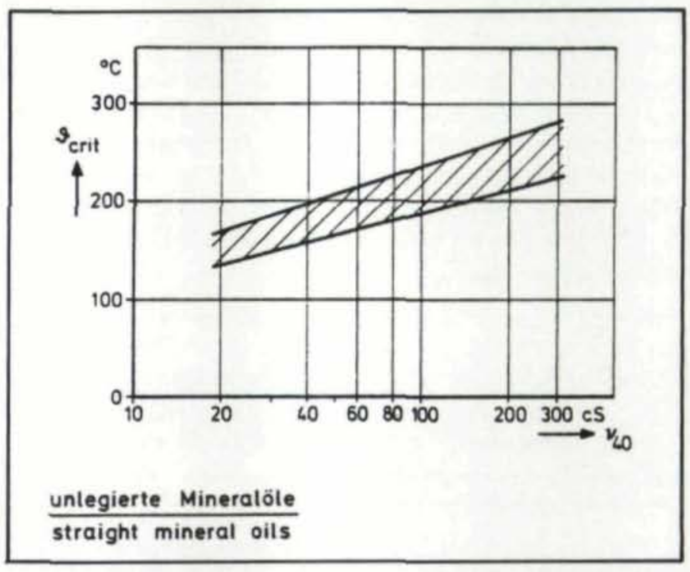


Fig. 11—Critical Contact Temperature for Flash Temperature Method

Temperature Criterion to that of the Integral Temperature Method.

In addition to the difficulties in the evaluation of local and instantaneous parameters of load—think of dynamic load distribution along the path of contact (Fig. 12)—coefficient of friction, radius of curvature under load, etc. Quite a few test results indicate that a single

	TOTAL CONTACT TEMPERATURE (SCORING INDEX)	INTEGRAL TEMPERATURE
Criterion	maximum, local, instantaneous contact temperature (simplified flash temperature)	mean, weighted flank temperature
Field of Application	straight mineral oils	straight, mild and EP mineral oils, synthetic oils
Critical Value	dependent on viscosity	from gear scoring test (e.g. FZG or Ryder test)

Table 3—Comparison of Total Contact Temperature and Integral Temperature Criterion

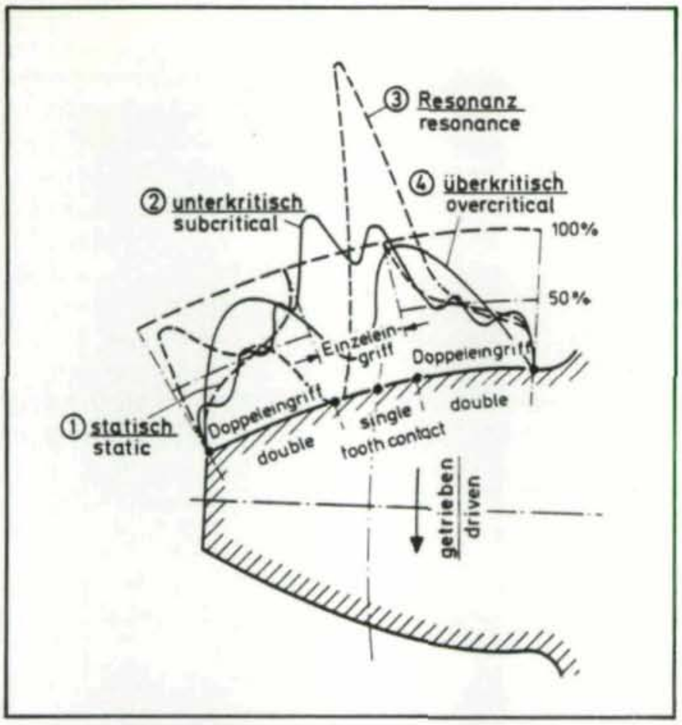


Fig. 12—Dynamic Load Distribution along the Tooth Flank acc. Rettig

temperature flash is not sufficient for a scoring catastrophy. Fig. 13 shows a tooth flank with incipient scoring of nearly the same severity, within an area of calculated contact temperatures between 320°C and 700°C. Deeper and more severe scoring and seizure would have been expected in the area of the tooth tip. This indicates the validity of a mean surface temperature as a critical energy level more than a temperature flash.

Another problem arises when tip relief is applied to gears with their critical temperature in the second point of single tooth contact (Fig. 14). In these cases, the calculated maximum contact temperature is not influenced by the tip relief while a strong increase in scoring load capacity can be observed in the test(9).

A series of tests of Ishikawa(9) were evaluated with

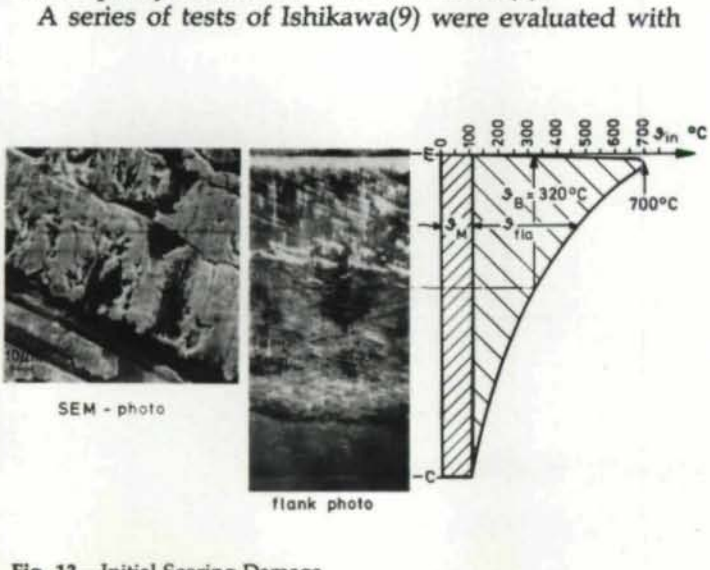


Fig. 13—Initial Scoring Damage

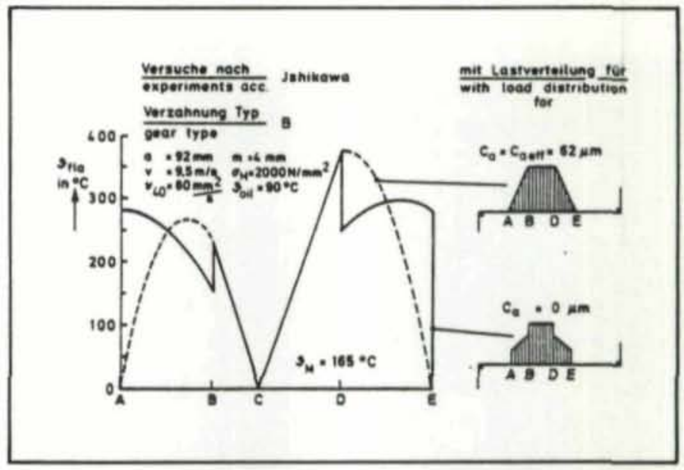


Fig. 14—Influence of Tip Relief on Flash Temperature

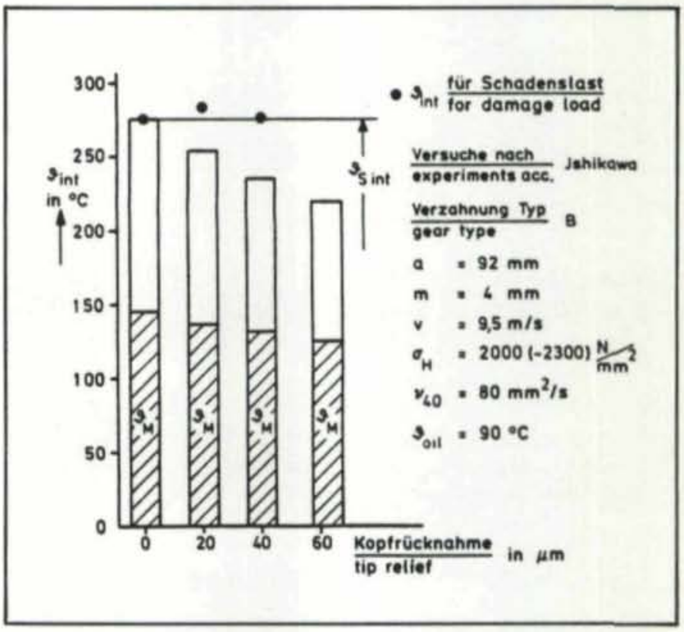


Fig. 15—Influence of Tip Relief on Integral Temperature

the Integral Temperature Method. They showed both steadily decreasing bulk and integral temperature, with increasing tip relief at constant load and a constant scoring temperature introducing the measured scoring loads (Fig. 15).

Examples

The validity of the Integral Temperature Method has been checked, with test results on different back-to-back test rigs, with center distances a = 91.5, 140 and 200 mm, with different gear geometries, different oils—straight mineral oils, compounded and EP-oils, synthetic oils of different viscosities—and different pitch line velocities up to v = 50 m/s. Fig. 16 shows the results of the calculations. For best correlation, the calculated safety factor for scoring conditions should be unity. The scattering is between about 1.0 and 1.4, which indicates a good correlation between test results and calculations, having in mind that the overload factors for the calculations have been set unity. For realistic overload factors, the calculated safety factor would somewhat decrease.

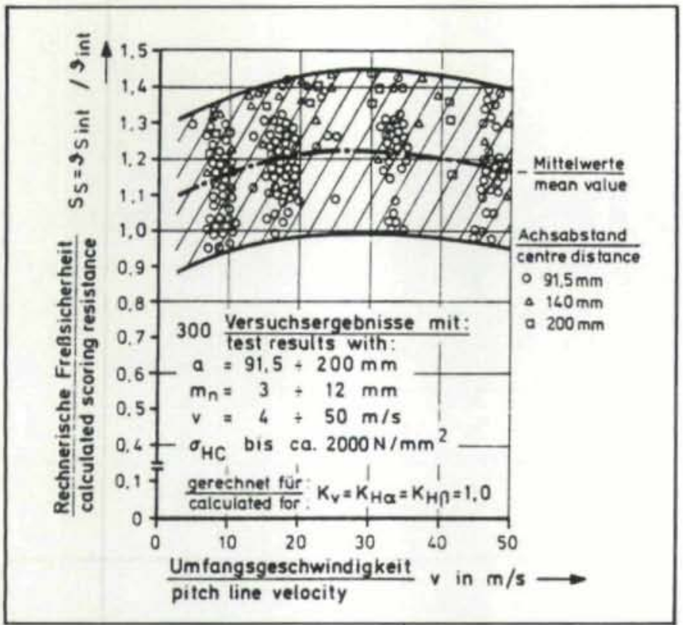


Fig. 16—Calculated Scoring Resistance for Test Results acc. Integral Temperature Method

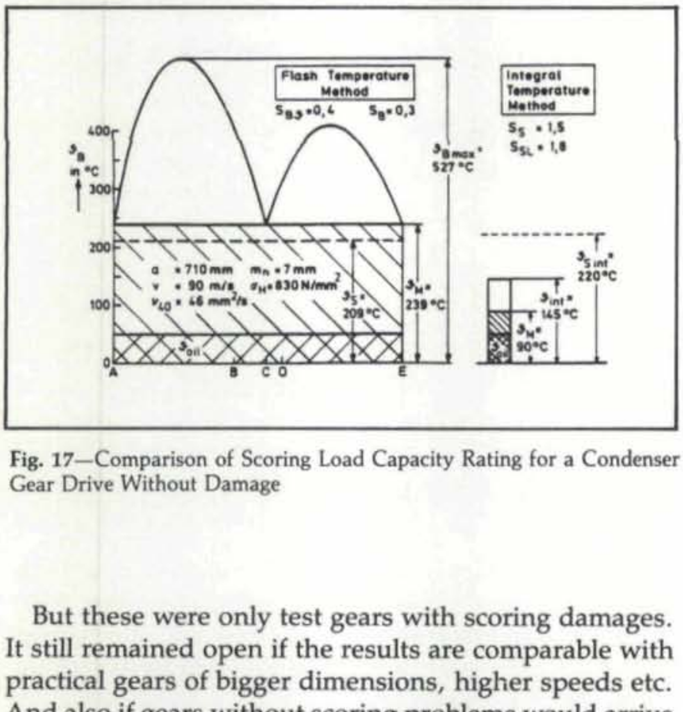


Fig. 17—Comparison of Scoring Load Capacity Rating for a Condenser Gear Drive Without Damage

But these were only test gears with scoring damages. It still remained open if the results are comparable with practical gears of bigger dimensions, higher speeds etc. And also if gears without scoring problems would arrive at calculated safety factors significantly higher than 1.0. Imagine that it is fairly easy to arrive at a value of 1.0, only extract often enough the square root of any figure and you will arrive at unity.

Therefore, we collected data from all kinds of practical gears with and without scoring damages. An example is shown in Fig. 17 for a condenser gear drive without scoring damages in service. The Total Contact Temperature Method calculates a safety factor of 0.4, the Integral Temperature Method of 1.5. A change of the unrealistic bulk temperature value, $\vartheta_M = 239$ °C of the Total Contact Temperature Method to $\vartheta_M = 90$ °C of the Integral Temperature, doesn't make it any better. The safety factor, $S_B = 0.5$, remains still far below 1.0, indicating a high scoring risk.

Similar experiences resulted when calculated safety factors of a variety of typical gears, out of more than one hundred examples were compared, with their scoring behavior in service. For the possibility of a comparison of Total Temperature resp. Scoring Index Criteria, we chose mainly gears which were lubricated with non EP-oils. The range of the operating conditions is shown in Fig. 18, and the results in Fig. 19. In cases where only the result of the Integral Temperature Method is shown, the other two criteria where not applicable because of the EP-character of the lubricant used. It is evident that the best correlation between calculated safety factors and practical experience is achieved with the Integral Temperature Method in a wide range of application.

From these recalculations, the different fields of scoring risk—high, borderline, low—as defined in *Intergal Temeprature Rating,* were established.

Achsabstand Centre distance	a	=	92	÷	2800	mm
Modul Module	m_n	=	3	÷	50	mm
Umfangsgeschwindigkeit Pitch line velocity	v	=	3	÷	120	m/s
Spezifisches Gleiten Specific sliding	v_{gmax}/v	=	0.1	÷	0.5	-
Übersetzung Gear ratio	u	=	1	÷	7	—
Hertzsche Pressung am Wälzkreis Hertzian stress at pitch circle	σ_{HC}	=	300	÷	2200	N/mm²
Unlegierte, mild und hochlegierte Mineralöle mit Viskosität straight mineral oils, mild and heavy E.P. - oils with viscosity	ν_{40}	=	20	÷	400	mPa.s.

Fig. 18—Range of Significant Gear Paramaters for Actual Gears in Fig. 19

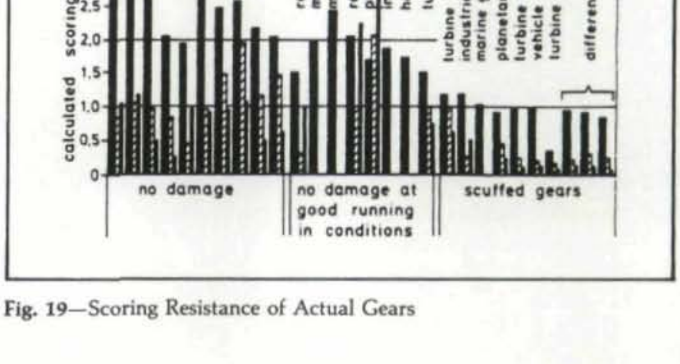


Fig. 19—Scoring Resistance of Actual Gears

(Continued on page 45)

Austempered Ductile Iron:
Technology Base Required for an Emerging Technology

By Dale Breen
Gear Research Institute

This paper addresses Austempered Ductile Iron (ADI) as an emerging technology and defines its challenge by describing the state-of-the-art of incumbent materials. The writing is more philosophical in nature than technical and is presented to establish a perspective. Incumbent materials are those materials which are solidly entrenched as favorites for given applications. Materials resulting from new emerging technologies must be more attractive than incumbents and other potential competitive materials. Materials technology is dynamic; incumbent materials are always in jeopardy from potential replacements. Improvements in processing, performance and cost are in perpetual demand. They have one strong advantage, however, and that is that they are proven and there is an existing data base with which practicing engineers are familiar. Our litigation prone society causes the engineering community to be very conservative. I recently saw the following as a title for an article in a popular trade magazine: "Pass the Aspirin, We're Changing Materials". That statement gives a reasonably accurate description of the environment in which new materials find themselves.

For numerous reasons, though, alternate materials technology continues to be an attractive place to put development dollars. Materials technology can have a very beneficial impact on the competitive position of a firm. It can impact on engineering performance, product development times, productivity, and costs including gratis and capital goods costs.

Why are we so high on Austempered Ductile Iron technology? Because of important favorable attributes, refer to Table 1, it has the potential of giving birth to a new series of engineering materials which will challenge existing ductile iron and cast steel applications, and, more importantly, critical applications which have been dominated by forged steel as well. Fig. 1[1] is presented with two purposes in mind: one, to show the tonnage of the domestic alloy steel market and, two, to show the variations in generic alloy steel applications along a time line.

TABLE I

- GOOD ENGINEERING PERFORMANCE
  (i.e., fatigue & toughness)
- ATTRACTIVE COSTS
  (cost improvements of 30% plus
  reported in some cases)
- 10% LIGHTER WEIGHT THAN STEEL
- IMPROVED NOISE AND VIBRATION DAMPENING
- IMPROVED WEAR AND SCUFFING RESISTANCE
- INCREASED FLEXIBILITY IN DESIGNING
  FOR OPTIMUM SHAPE

The latest tonnage figure available to the writer was 10,000,000 tons which was for 1979. I'm sure there was a decrease in the recent recession years, but certainly this is a potentially lucrative market. The total current shipments of ductile iron are only 2.0 million. ADI is the most promising cast material for use in critical machine elements to emerge for some time. Although it won't replace all steel applications, I believe that a decade from now when we look back, we'll all be surprised at the inroads ADI has made if the technology is adequately nourished.

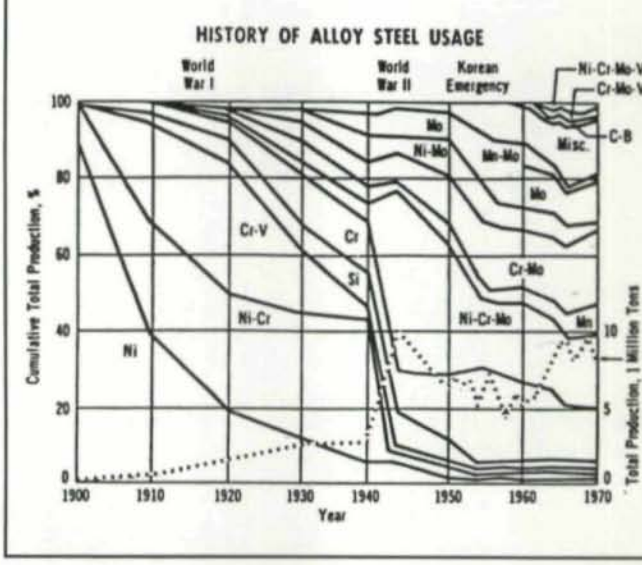


Fig. 1—Through the years since 1900, the pattern of alloy steel usage has changed greatly. The dotted line indicates alloy steel production.

AUTHOR:

MR. DALE BREEN *is Secretary and Director of the Gear Research Institute. His experience includes research and research management, materials engineering and failure analysis, manufacturing metallurgy and teaching. Formerly, he managed the corporate metalurgical research function and laboratories of International Harvester Company where for over ten years, he was Chairman of the Corporate Gear Committee. He has a BS in Mechanical Engineering from Bradley University, a MS in Metallurgy from the University of Michigan and MBA from the University of Chicago. Mr. Breen is the author of numerous papers in the areas of gears, metallurgy and fatigue and is co-author of a book,* Hardenability of Steel. *He is active in numerous professional societes and has been named to Who's Who in Engineering and Who's Who in Technology.*

Its proponents can't, however, become over-confident as forged steel technology is not lying dormant. The literature is beginning to reveal significant new developments such as warm precision forging, superplastic steels and injection molded powdered metals. So ADI must beat out the other competitors as well as incumbent materials. ADI, though, has many additional unexplored possibilities such as near net shape castings finished by rolling or grinding, induction austempering, and reduction in numbers of parts by combining into a single casting and so on.

DESIGN TECHNOLOGY

The existence of a mature technology base was mentioned earlier as a plus for the incumbent materials. These demonstrate both some potential applications and some of the failure type which designers and metallurgists work together to prevent. The prime design criteria is fatigue. In these figures, axial, bending, torsional, contact plus traction, contact plus environmental assist and thermal stress fatigue are demonstrated. Some applications may require unusual toughness and wear properties (lubricated and unlubricated).

Designers of machine elements have considerable information available to them concerning the performance characteristics of heat treated and carburized type steels. In the following discussion, the intent is to present a "thumbnail" sketch of some important performance data available concerning steel. In specialized applications, such as gears and bearings, special performance information is required. This will be discussed subsequently. First, as was stated previously, usually machine elements are designed to a fatigue performance criteria. Two important tools are the Allowable Stress Range (ASR) Diagram and the Stress-Number of Cycles to Failure (SN) Diagram.[3] Fig. 2 shows the former and Fig. 3 the latter. They are related as shown in Figs. 3a and 3b. The ASR diagram, actually a strength diagram, gives a variety of useful information. It can thoroughly describe the fatigue capacity of a material under a multiplicity of load types and manners (contact fatigue excepted). Using fatigue ratios for ADI taken from Fig. 4[4], a

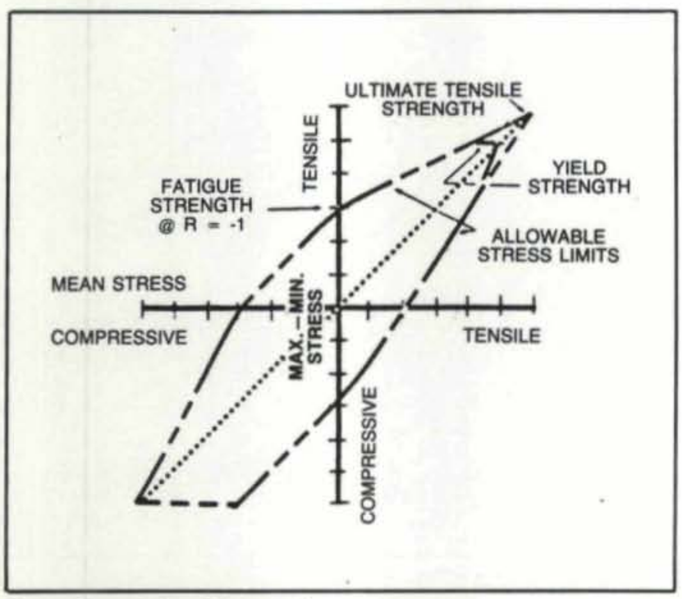


Fig. 2—Typical allowable stress diagram construction.

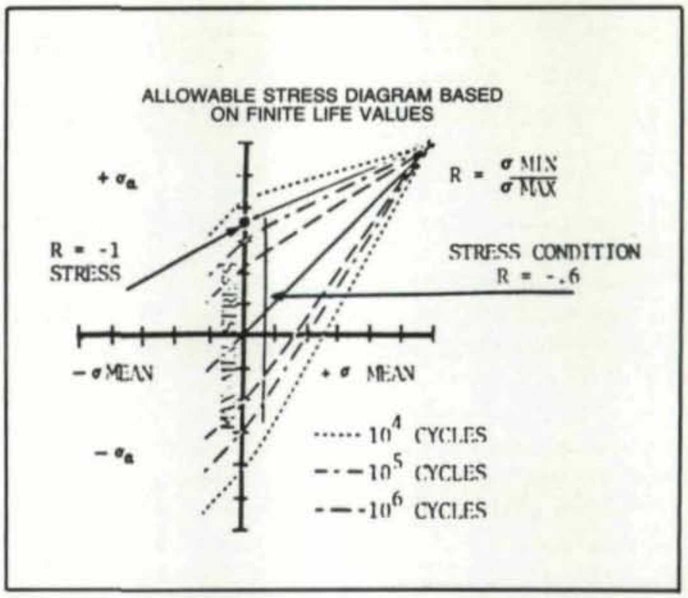


Fig. 3a—Finite Life ASR Diagram
Shows R=−1 equivalent stress for R=0.6 loading.

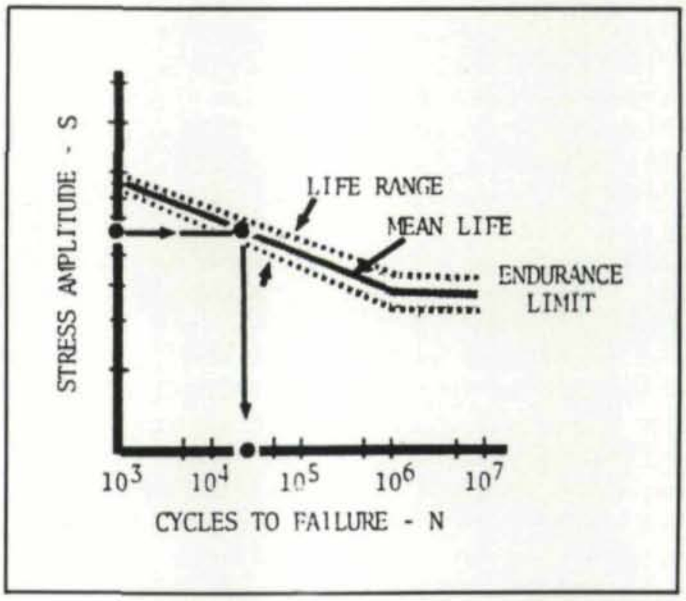


Fig. 3b—S-N Diagram
Shows life prediction for R=0.6 loading using R=−1 equivalent stress.

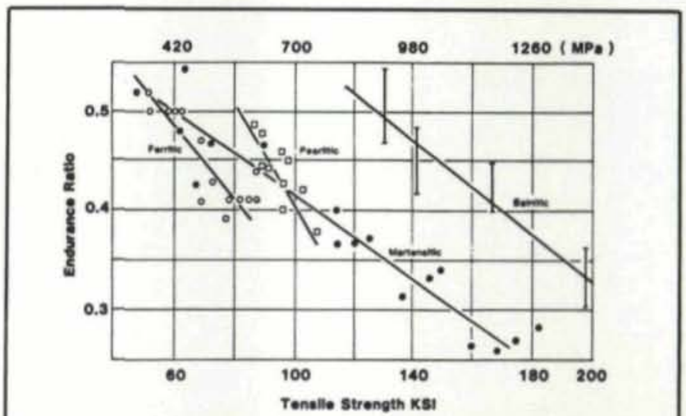
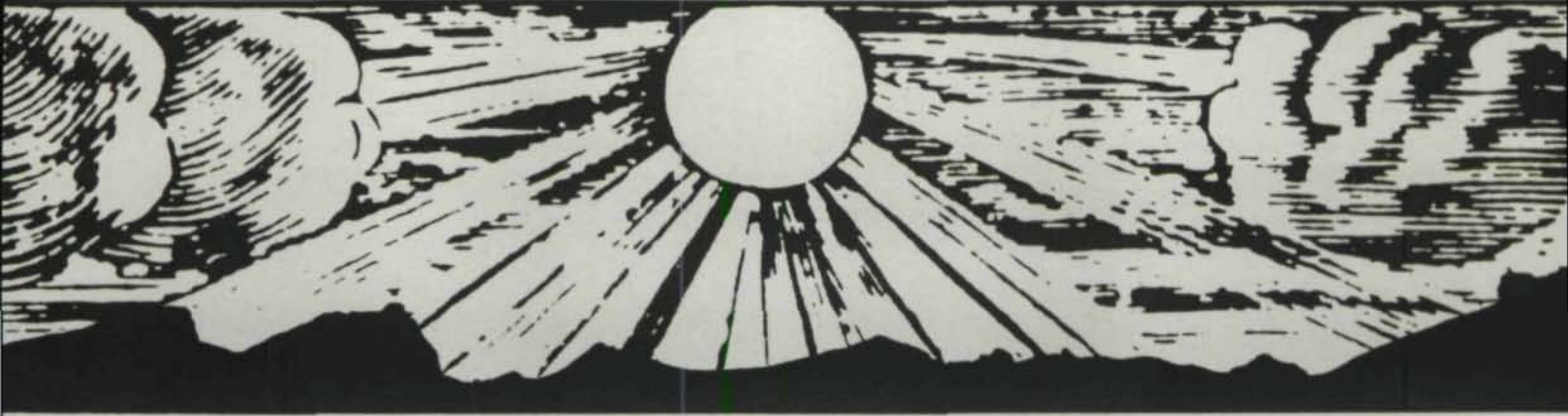


Fig. 4—The endurance ratios for nodular iron as influenced by its tensile strength and matrix microstructure. Endurance ratio for the bainitic irons defined at 2×10^6 cycles; for all other irons the ratio was calculated at 10^7 cycles.



EXPAND YOUR HORIZONS WITH GEAR TECHNOLOGY

Receive 1 Full Years Subscription

FREE

(Value $27.00)

by completing the attached card

MAIL TODAY
and increase your knowledge of developments in the gear industry

ALL INFORMATION MUST BE FILLED OUT TO QUALIFY FOR FREE SUBSCRIPTION

☐ **YES!** SEND GEAR TECHNOLOGY ***FREE!***

PRIMARY BUSINESS ACTIVITY AT THIS LOCATION (Circle One)

A. Manufacturer with Gear Department
B. Gear Manufacturer—Job Shop—Contract Work
C. Gear Machine Tool Manufacturer
D. Machine Tool Distributor/Agent
E. School/Research Organization
F. Government Agency
G. Industry Association
H. Advertising Agency
I. Other: Please specify ______________

Does your company have a specialty, and if so, what is it? ______________

Does your company have an unusual machine? Yes ☐ No ☐

Does your company do sub contract work? Yes ☐ No ☐

How many employees at this location?

☐ 1-20 ☐ 21-50 ☐ 51-100 ☐ 101-500 ☐ 500 and over

Your job function (circle all that apply)
A. President/Owner/General Manager
B. Manager ________ Dept. (specify)
C. Design Engineer
D. Process Engineer
E. Sales
F. Consultant
G. Educator
H. Other: Please specify ______________

In which of the following ways are you personally involved with gear machinery?

A. Use them
B. Recommend them
C. Establish specifications
D. Approve purchase
E. Acquire them
F. Other involvement ______________
G. No involvement

Are there any other individuals at this location that may qualify for a complimentary subscription to Gear Technology?

Name ______________ Title ______________

Name ______________ Title ______________

Name ______________ Title ______________

Please complete or attach business card.

Name ______________

Company ______________

Div. of ______________

Co. Address ______________

City ______ State ______ Zip ______ Phone ______

Your Signature ______________ **Your Title** ______________

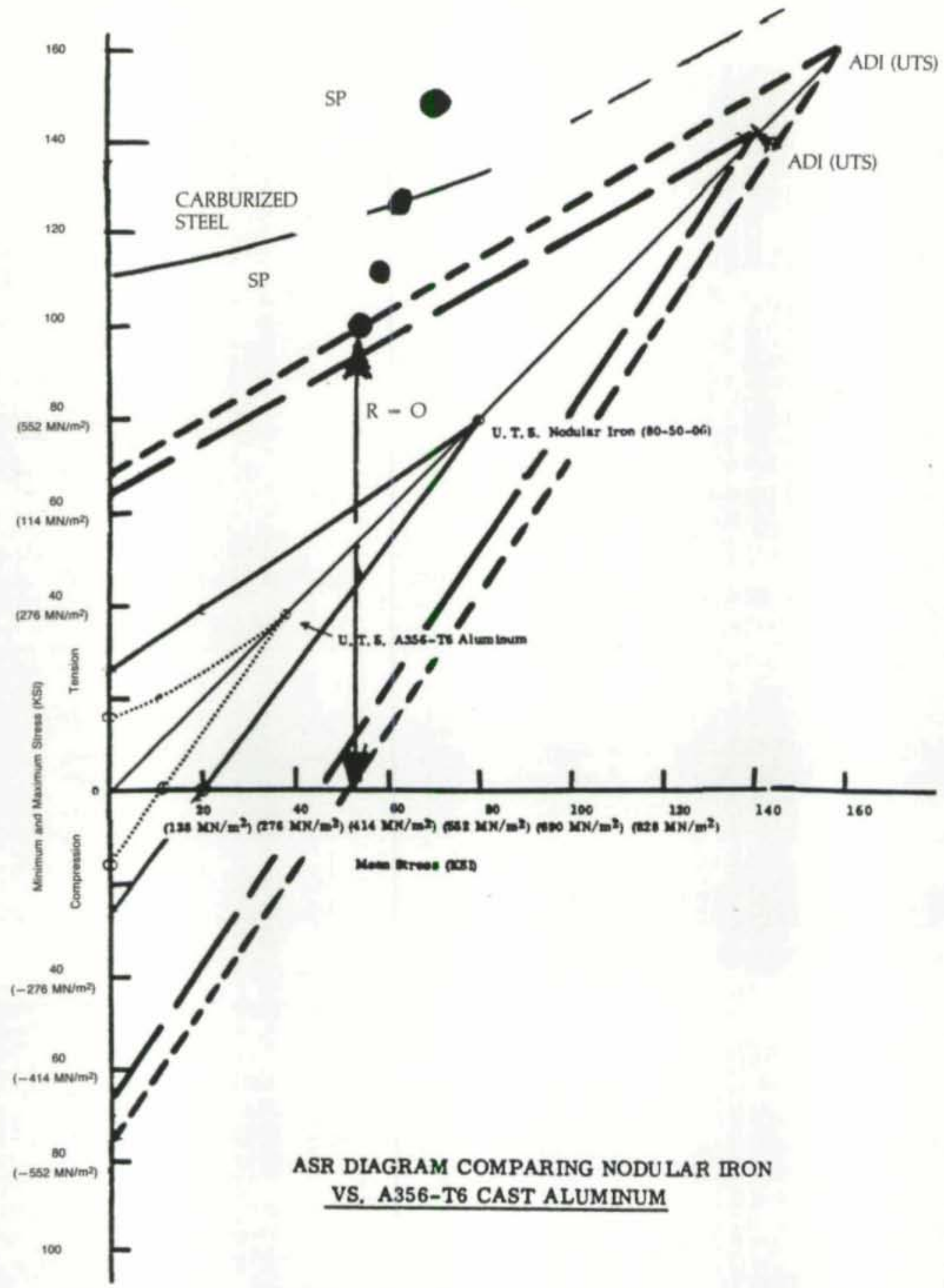
PLEASE SIGN HERE TO CONTINUE (OR BEGIN) RECEIVING GEAR TECHNOLOGY FREE.

Editorial scores: Help us to serve you better by rating feature articles in this issue. Refer to numbers at the end of each article then mark this card accordingly.

	E-1	E-2	E-3	E-4	E-5	E-6	E-7	E-8	E-9	E-10
Excellent	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐
Good	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐
Fair	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐
Poor	☐	☐	☐	☐	☐	☐	☐	☐	☐	☐

Advertisements Circle number(s) for additional information

A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9	A-10
A-11	A-12	A-13	A-14	A-15	A-16	A-17	A-18	A-19	A-20
A-21	A-22	A-23	A-24	A-25	A-26	A-27	A-28	A-29	A-30



ASR DIAGRAM COMPARING NODULAR IRON VS. A356-T6 CAST ALUMINUM

Fig. 5

comparison of some ASR information is shown in Fig. 5. This is not quite accurate, since the comparison is at $2x10^6$ cycles for the ADI and $1x10^6$ cycles for all others. Since the reverse bending (R=-1) endurance limit for the 356-T6 and the 80-50-06 NI is for one million cycles, the 10 million cycle envelope would be even smaller. Obviously, ADI at 140 ksi and 160 ksi tensile strength has some attractive load carrying capabilities. To put it in perspective, however, I have shown some dots for undirectional bending, R=O type loading, so one can consider applications such as gear tooth bending which is close to R=O. The dot for ADI/160 ksi occurs at 100 ksi (R=0). Shot peening, assuming 25 percent increase in R=-1 limit, translates to the next dot up, so the R=0 load limit in the shot peened condition would be about 110 ksi. The upper two dots are for carburized steel. The dot corresponding to an R=0 strength of about 125 ksi is for the unpeened condition, whereas the one at 150 ksi is for the peened condition. These are all constructed diagrams, thus do not represent actual data. Real data is urgently needed. One can develop a perspective though and conjecture about possibilities. Shot peening or rolling to improve bending fatigue properties is apparently going to be a necessity in many applications. This is one of the challenges. We need to know quantitatively what to expect and the best procedures for processing. The data shown in Fig. 6 may be conservative, as it

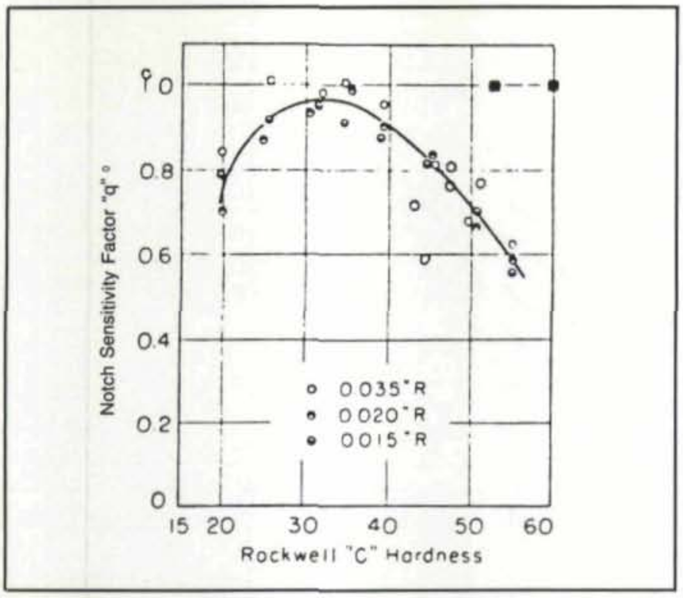


Fig. 6—Variation of notch sensitvity factor "q" with hardness.

has been assumed that peening will increase the EL by only about 25 percent. Previous data reported[5] has indicated improvements by as much as 65 percent, possibly because of the favorable work hardening of austenite.

Contact fatigue properties look very promising, but the literature is quite terse in this respect. Enough is known, however, to recognize the potential; but here is another challenge: we need data! ADI is not a single material, but consists of many possible grades with wide ranges of possible properties.

Fatigue notch sensitivity factors (q) and fatigue notch reduction factors (K_f) are also needed. q is related to K_f (fatigue reduction factor) and K_t (theoretical stress concentration factor) as follows:

$$q = \frac{K_f - 1}{K_t - 1}$$

The relationship of q and strength/hardness is shown in Fig. 6[6] for heat treated steel. This type of information on ADI is not yet available, but is badly needed.

It is not within the scope of this writing to discuss toughness as a design criteria, but it is certainly important, especially in applications where significant yielding can be expected such as roll-over protective structures.

GEAR DATA REQUIREMENTS

Machine elements such as gears are normally designed to bending fatigue and contact (pitting) fatigue criteria. Figs. 7 and 8* show the design allowables for carburized steel as shown in AGMA standards. Comparable data, in which the designer can have confidence, is required for ADI.

* "Extracted from AGMA Standard Design Guide for Vehicle Spur and Helical Gears, AGMA 170.01, with the permission of the publisher, the American Gear Manufacturers Association, 1330 Massachusetts Avenue, NW, Washington, D.C. 20005."

In regard to toughness, carburized steel has K_{Ic} values in the high carbon case region varying from 15-22 ksi in. There are

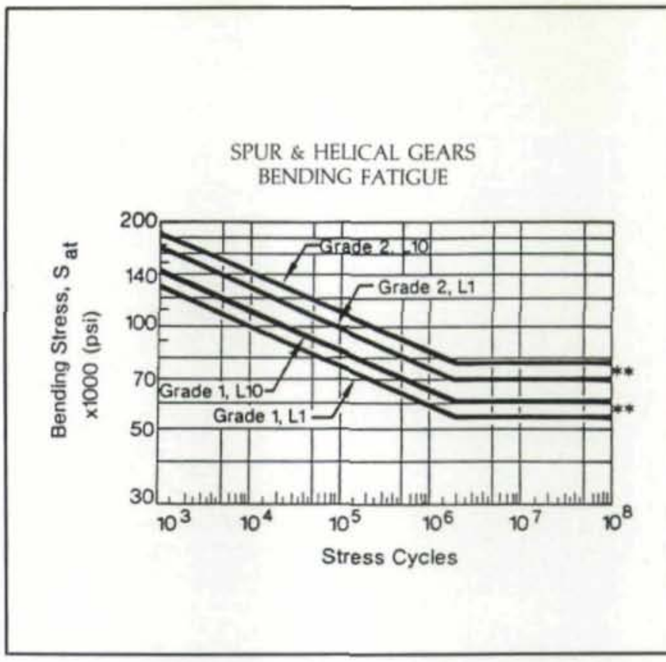


Fig. 7—Alloy steels case carburized to Rc 58-63 case hardness, Rc 30-42 core hardness.

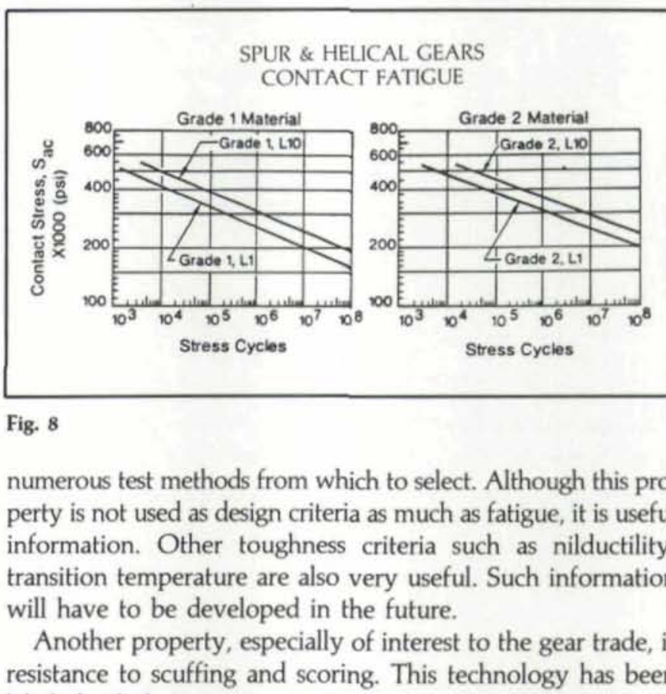


Fig. 8

numerous test methods from which to select. Although this property is not used as design criteria as much as fatigue, it is useful information. Other toughness criteria such as nilductility-transition temperature are also very useful. Such information will have to be developed in the future.

Another property, especially of interest to the gear trade, is resistance to scuffing and scoring. This technology has been labeled tribology.

Scuffing and scoring properties of materials per se have not been systematically categorized. A recent publication by Terauchi[7] is a good review of the subject. The technology might best be understood by looking at a few excerpts from his paper. The safety factor against the danger of scoring can be defined by:

$$Sf = \frac{Oa}{O}$$

where O is the instantaneous surface temperature at the point of contact on the active profile and Oa is the allowable maximum temperature to prevent scoring. The following equation is given for O:

$$O = Oo + 157 \times 10^{-4}\, u\, K_R\, E_A$$

where Oo is the surface temperature just prior to mesh and

TABLE II

INFLUENTIAL FACTORS ON SCORING RESISTANCE OF GEARS

Influential factors on 0	*Operating conditions* Dynamic load, Over load, Speed variation, Impact load, etc.	*Geometric variable of gears* Module, Pressure angle, Face width, Helix angle, Amount of addendum modification. Profile modification, etc.
		Accuracy of gears Tooth profile error, Spacing error, Pitch error, Lead error, Surface roughness, etc.
		Basic amount of gear operation Nominal peripheral velocity, Nominal carrying load
Influential factors on 0_a	*Characteristics of working tooth faces* Metallurgical structure, Hardness, Property against wear, Thermal property, etc.	
	Characteristics of lubricants Viscosity, Kind and composition of lubricants, Additives (Thermal property, Adsorption and reactivity to tooth material), etc.	
	Lubricating methods Splash lubrication and/or jet lubrication, Oil volume, Oil supplying rate, Oil supplying position, etc.	

u is the coefficient of friction. E_n contains factors relating to loading, material thermal coefficients, tooth load sharing and relative velocities. K_r is related to surface roughness after "run-in". Table II shows the factors influencing the scoring resistance of gears. Fig. 9 shows the calculated temperature rise along the involute of a gear active profile. Note that at the pitch line, where there is no relative sliding, there is no significant temperature rise. Fig. 10 shows the relationship between hardness and critical scoring temperature for several steels.

The technological aspects of lubricated contact are fairly well understood, and there is basic agreement on the elastohydrodynamic-critical temperature concept when nonreactive lubricants are involved. The literature is extensive. [8] [9] [10] [11] [12] Most, however, deal with the lubricants, surface finish, dynamics or coatings, not the metallurgy of the contacting materials. Also, much of the work relates to systems with zero or low slide/roll ratios, such as bearings. The contacting surfaces of gears are subject to rather high slide/roll ratios.

Traditionally non-reactive oils have been simpler to study than reactive oils.[13] Scoring is dependent on a number of independent variables. At some speeds where elastohydrodynamic lubrication prevails, performance is determined by lubricant properties as influenced by load and speed. The coefficient of friction is basically independent of surface roughness or the frictional properties of the contacting surfaces. Performance is not dependent on the boundary lubrication properties of the lubricant. When appreciable asperity interaction occurs, the situation changes. Surface roughness and the metallurgy of the contacting surfaces become important and the lubricant reactivity plays a role. The interactions become complex. Wear and temperature effects make a contribution and instability occurs. It will be interesting to see where ADI fits in the scheme of things in a quatitative way. The presence of free graphite and strong, plastic austenite may enhance this property significantly.

The other possible plus for ADI is its potential to improve noise and vibration damping capability. Quatitative information is needed to help assess this potential benefit.

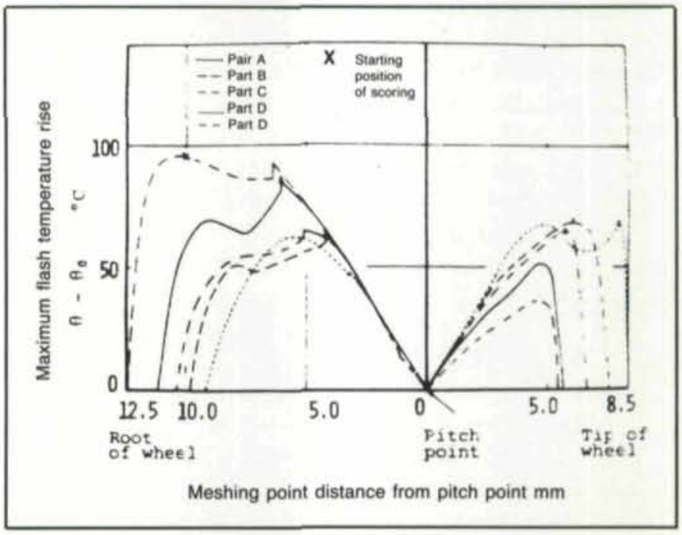


Fig. 9—Calculated variation in surface temperature rise along the line of action (load sharing included).

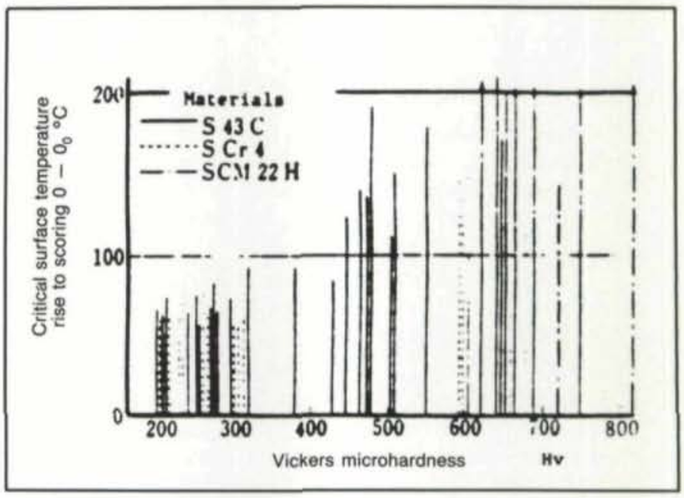


Fig. 10.—Relation between hardness and critical surface temperature rise to scoring of disks.

Concluding Statement

This has been a terse scan of some of the technologies that are important to the design of machine elements. It is intended to help define some of the needs related to ADI, so that engineers are in a better position to take advantage of it, in both new designs, and as a substitute material in existing designs. The challenge then has many aspects, but two important ones are: on target generation of necessary data and timely and competent use of it by the user community. Processing needs such as influence of alloy on hardenability, machinability data and quality assurance procedures, although very important, have not been discussed.

As a concluding thought, I'd like to reflect on a quote from Harry McQuaid, who said "I was once told that the ideal design is one that is just good enough; that anything better than good enough was wasting someones money; and that anything not good enough means you wouldn't have a job very long."

That is a very succinct and on target statement. It has two technical elements: one, components must perform and two, their costs must be optimized. The implications of this are profound. This implies that we must have comprehensive technical knowledge concerning what the performance criteria are in terms of loads, their frequency and variations, the states of stress they induce, environmental conditions, and so on. In addition, we must quantitatively understand properties of materials of construction in terms of fatigue and fracture resistance, and the influence of the myriad environments components they are exposed to, and thirdly, we must have comprehensive knowledge concerning producing the complex shapes required. This includes costs and sensitivity to capital equipment requirements. This is a call for dedication and competency. There are numerous instances where these elements have been displaced by lack of appreciation for technology and corner cutting by U.S. industry. The third element of the statement has to do with job security. McQuaid's prediction of job loss has occurred on a massive scale. Certainly a large percent of U.S. component tonnage is now produced abroad.

ADI is on a threshold. We must learn all we can about it as quickly as possible, to help it gain its proper position as an engineering material, and thus, enjoy its benefits. We need to be careful, however, and find successful applications so as not to jeopardize ADI by developing a track record of failures.

References

1. "The Sorby Centennial Symposium on the History of Metallurgy", Cyril Stanley Smith, ed. Gordon and Breach Science Publishers, NY, 1963, p. 475, and D. H. Breen, G. H. Walter, and J. T. Sponzilli, "Computer-Based System Selects Optimum Cost Steels", ASM Metal Progress, November, 1973.
2. D. H. BREEN AND E. M. WENE, "Fatigue in Machines and Structures Ground Vehicles", in *Fatigue and Microstruture*, pp. 60-63, ASM, Metals Park, Ohio, 1979.
3. IBID, 2.
4. V. K. SHARMA AND D. VUKOVICH, "Some Observations on the Contact Fatigue Properties of Austempered Ductile Iron". Paper given at 1st International Conference on Austempered Ductile Iron, ASM, April 2-4, 1984, Chicago, Illinois.
5. JOHANSSON, et al, "AusteniticBainitic Ductile Irons". AFS Transactions, Vol. 85, 1977, p. 117.
6. M. F. GARWOOD, H. H. ZURBURG, AND M. ERICKSON, "Correlation of Laboratory Tests and Service Performance," in *Interpretation of Tests and Correlation with Service*. ASM, Metals Park, Ohio, 1951.
7. Y. TERAUCHI, "Scoring of Spur Gear Teeth," Lubrication Engineering, Vol. 40, No. 1, January, 1984.
8. H. BLOK, "Lubrication as a Gear Design Factor," Proceedings of International Conference on Gearing, 1958, pp. 144-158, Inst. Mech. Engnrs., London, England.
9. H. BLOK, "Hydrodynamic Effects in Rolling With Slippage," Proc. Symp. Rolling Contact Phenomena, edited J.E. Bidwell, pp. 186, Elsevier, Amsterdam, Holland.
10. P.M. KU and B.B. BAKER, "The Effect of Lubrication on Gear Tooth Scuffing," ASCE Transactions, Vol. 2, No. 2, October, 1959.
11. R.S. FEIN, "Operating Procedure Effects on Critical Temperature," ASCE Transactions, Vol. 10, No. 4, October, 1967.
12. ASME Handbook, Wear.
13. B.W. KELLY and A.J. LEMANSKI, "Lubrication of Involute Gearing," Proc. IME, Vol. 182, 1967.
14. *Three Keys to Satisfaction*, a publication of the Climax Molybdenum Co., Ann Arbor, Michigan.

This article was previously published for the ASM Austempered Ductile Iron Conference Proceedings, April, 1984.

CIRCLE E-5 ON READER REPLY CARD

CORRECTION: Two formulas were shown incorrectly in the Back to Basics article, Vol. 1, No. 1, page 41. They should be corrected as follows:

$$\text{arc}\,\theta = \text{invo}\,\phi = \tan\phi - \text{arc}\,\phi$$

and

$$r = \frac{rb}{\cos\phi}$$

Design Of Involute Gear Teeth

by
Fellows Corporation

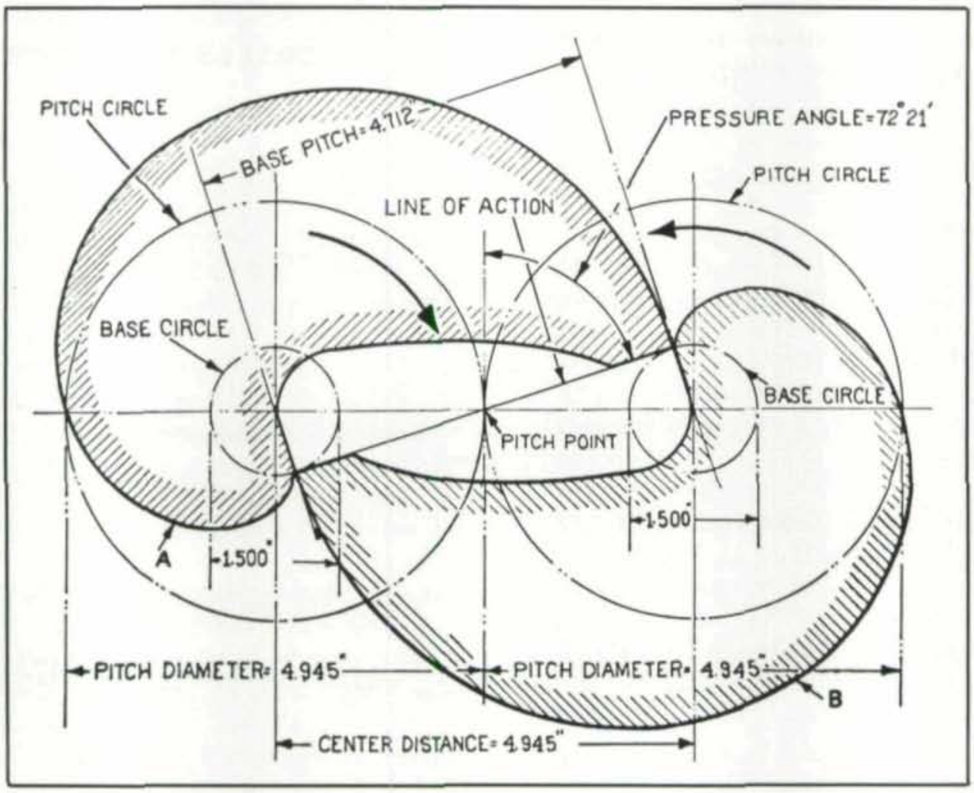


Fig. 1—Involutes applied to two one-tooth gears, indicating that the involute has its origin at the base circle, but is not limited in length.

In designing involute gear teeth, it is essential that the fundamental properties of the involute curve be clearly understood. A review of "the Fundamental Laws of the Involute Curve" found in last issue will help in this respect. It has previously been shown that the involute curve has its origin at the base circle. Its length, however, may be anything from zero at the origin or starting point on to infinity. The problem, therefore, in designing gear teeth, is to select that portion of the involute, which will best meet all requirements.

Involute Curve Not Limited in Length

Fig. 1 shows a pair of one-tooth involute gears in theoretically perfect action. The application of the involute curve, as here presented, offers an interesting study. Although of little or no practical value as driving members, the extremities at which involute action may take place are here made plain and the nature of the involute curve made clearer.

It will be noted in Fig. 1 that each of the two involutes constitutes an unsymmetrical tooth. If *B* acts as a driver and rotates in the direction indicated by the arrow, *A* will rotate in the opposite direction. Contact will take place along the line of action which, in this particular case, is the circumference of the base circle and also the base pitch of the involute.

The circular pitch of these two involutes developed from a base circle of only 1½ inches is 15.537 inches. The diametral pitch is 0.2022. It is of passing interest that the pressure angle must always be 72 degrees, 21 minutes, when a single involute effects complete rotation of a single engaging involute. The pressure angle of 72 degrees, 21 minutes is, of course, excessive, and these engaging involutes are obviously incapable of transmitting any but the lightest loads.

Factors to be Considered in Gear-Tooth Design

It has previously been shown that the transmission of smooth

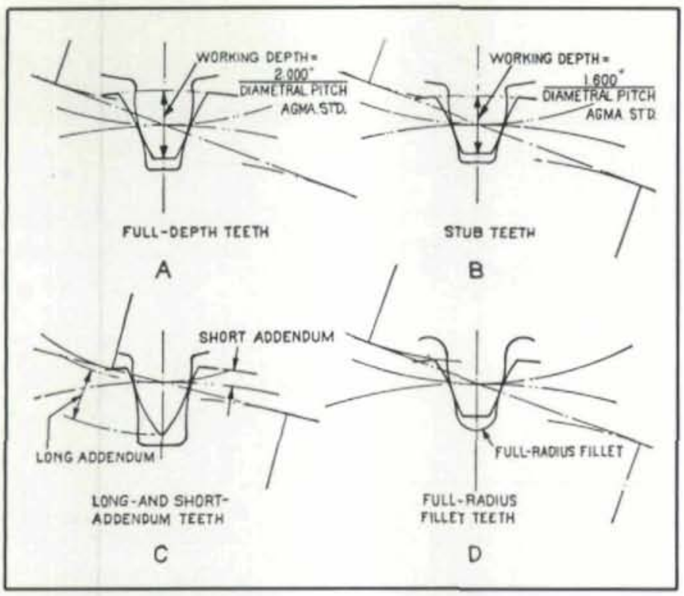


Fig. 2—Diagram illustrating various forms of involute gear teeth.

positive power requires that there must be a number of engaging teeth on driver and driven members. Also, that the following requirements must be satisfactorily met.

1. That there is no involute interference.
2. That there is no fillet interference.
3. That there is ample overlap of tooth action.
4. That a suitable pressure angle has been selected.
5. That excessive slippage is avoided.

Most of these requirements were briefly explained in the last issue, and we will now deal with their relationship to gear tooth design.

Gear-Tooth Shapes

There are in use today several forms of gear teeth; such as: 14½° full-depth teeth, 20° full-depth teeth; and 20° stub teeth. Full-depth teeth, as shown at *A* in Fig. 2, have a working depth equal to:

$$\frac{2.000 \text{ inch}}{{}^{*}\text{Diametral Pitch}}$$

There are also two stub-tooth forms, both having a pressure angle of 20 degrees. The so-called "Fellows" stub-tooth system, originated in 1906, is a combination of two diametral pitches. For example, 6/8 pitch; in which the numerator of the fraction controls the number of teeth, circular pitch and pitch diameter; and the denominator of the fraction controls the working depth and the clearance. The American Gear Manufacturers Association has adopted a stub-tooth standard, see *B*, Fig. 2, in which the working depth is a constant proportion throughout the entire range and is equal to:

$$\frac{1.600 \text{ inch}}{{}^{\dagger}\text{Diametral Pitch}}$$

*Refer to Tables III to VI, inclusive.

†Refer to Table VII.

In those cases where a pinion having a relatively small number of teeth must operate with a gear having three or more times as many teeth, a condition known as interference is sometimes encountered. Several solutions to this problem have already been given. One method is to use long-and short-addendum teeth as indicated at *C* in Fig. 2. This form of tooth cannot be standardized as the lengths of the addendums on pinion and gear are governed by the ratio of the number of teeth in mesh, and the pressure angle used. For average applications, the maximum enlargement and reduction of the outside diameters of pinion and gear, respectively, seldom exceeds 10% of the standard outside diameters.

Still another form of tooth is indicated at *D* in Fig. 2. This is known as the "full-radius fillet" form. This form of tooth is used quite extensively for aircraft gears and pump gears. The advantages are that it provides not only a stronger tooth, but also one which is not so liable to have fracture cracks develop at the root of the tooth as a result of heat treatment.

Of course, other pressure angles, in addition to the standard 14½ and 20 degree tooth shapes are used, but, a simple change in pressure angle, cannot in reality be considered as a different form of tooth. There are, however, other forms of teeth, but these are chiefly to meet special requirements, as will be discussed later.

Pressure Angle Depends on Portion of Involute Used

In designing gear teeth a primary consideration is to select that portion of the involute for the teeth which will best meet requirements. In Fig. 3 a series of parallel involutes, *A*, *B*, *C* and *D*, have been developed from the same base circle, and on these involutes, teeth of the same diametral pitch have been constructed. These teeth, as shown, have 14½, 20, 25 and 30 degree operating pressure angles. It will be noted that as the pressure angle is increased, a different section of the involute is employed for that portion of the tooth above the base circle. It is also interesting to note the shift in the location of the pitch circles relative to the base circle as the operating pressure angles are increased. This would, of course, necessitate an increase in the center distance when mating with another gear.

In the case of the 14½ degree pressure angle tooth, there is an undercut of the flank of the tooth which almost reaches to the pitch circle. This undercut condition naturally reduces

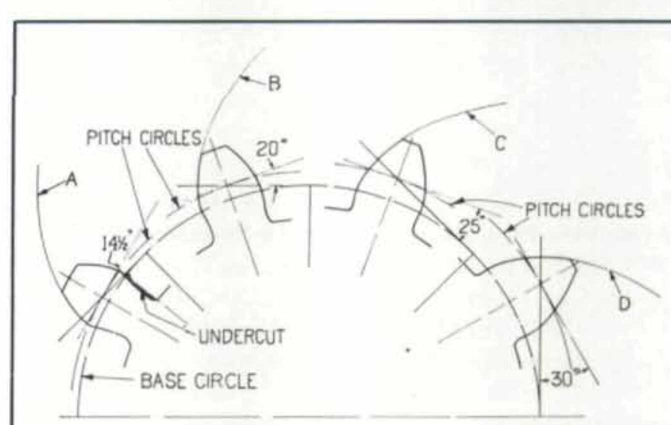


Fig. 3—Diagram illustrating that pressure angle is governed by portion of involute used for gear tooth.

the effective length of the line of contact with a mating tooth, and might affect the tooth action. The amount of undercut would depend, also, on the number of teeth in the gear, as well as, the pressure angle. In all of these cases, the teeth were developed from a base circle for a 12-tooth, 1 diametral pitch gear.

Relation of Pressure Angle to Interchangeability

The previous remarks regarding slippage, interference, continuous action, etc., should be sufficient to make plain the necessity for a careful study of these factors in the design of involute gears to obtain the best possible results under specific conditions. In connection with the design of gears for interchangeable application, the pressure angle selected is of great importance. Together with the length of the addendums, it determines the possible range of involute action between mating gears. For full-depth teeth of standard proportions, the addendum is made equal to the reciprocal of the diametral pitch. For example, the addendum of an 8-pitch gear is 1/8 inch.

It is also common practice in the design of involute gearing for interchangeable application to keep the number of teeth in the pinion as large as possible, and preferably not go below 12 teeth in the pinion. The other extreme is the rack.

An interchangeable system of gearing to meet these requirements without tooth interference, and at the same time provide a suitable length of contact, is diagrammatically presented in Fig. 4. At *A*, a 12-tooth pinion of 1 diametral pitch is shown in engagement with a rack tooth of 1-inch addendum, and at *B*, two 12-tooth pinions are shown in engagement.

Obviously, if the top of the rack tooth extends beyond the interference point, it will interfere with the flank of the pinion tooth. This interference point can, therefore, be used to establish the base circle of the mating pinion tooth. The base radius can be determined by the following formula:

$$R_1 = \sqrt{R(R - a)}$$

In which:

R = Pitch radius of pinion

R_1 = Base radius of pinion

a = Addendums of rack and pinion teeth

Assuming that the pitch radius of the pinion is 6 inches, and the addendums are 1-inch, then:

$$R_1 = \sqrt{6(6 - 1)} = \sqrt{6 \times 5} = 5.477 \text{ inches}$$

The pressure angle p can be found by the following formula:

$$p = \frac{R_1}{R} = \frac{5.477}{6} = 0.91283 = \text{cosine of } 24^\circ\, 6'$$

The diagrams in Fig. 5 show a 1 diametral pitch 12-tooth pinion and rack, and two 12-tooth pinions in engagement. In one case, the addendums are 1-inch, and in the other, 0.800 inch.

In the case of the 12-tooth pinion and rack with 1-inch addendums, and 24° 6′ pressure angle, contact starts at the interference point. The length of contact exceeds the base pitch by 1.491 inches. This gives 1.491 ÷ 2.868 or approximately 51% overlap of action. In the case of the two 12-tooth pinions, which represent the low point of the range, tooth contact, as shown at *B* remains well inside the interference points. Here, the length of contact exceeds the base pitch by 0.950 inch. This

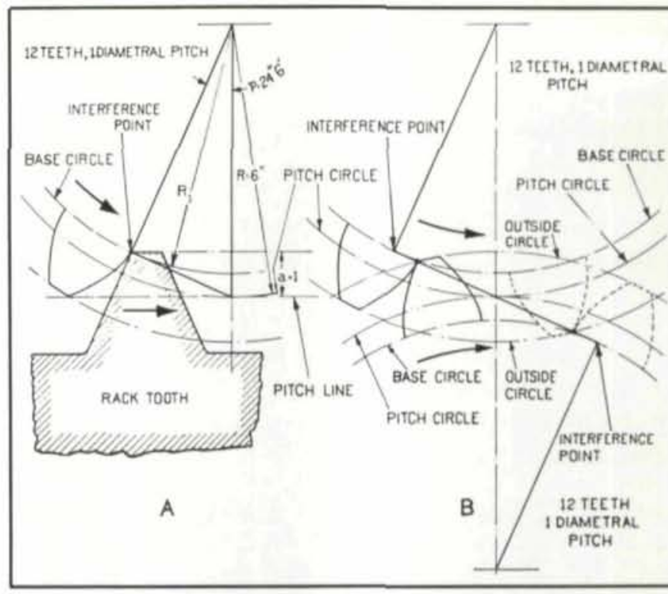


Fig. 4—Diagram illustrating that a pressure angle of 24° 6′ meets all requirements from a 12-tooth pinion to a rack.

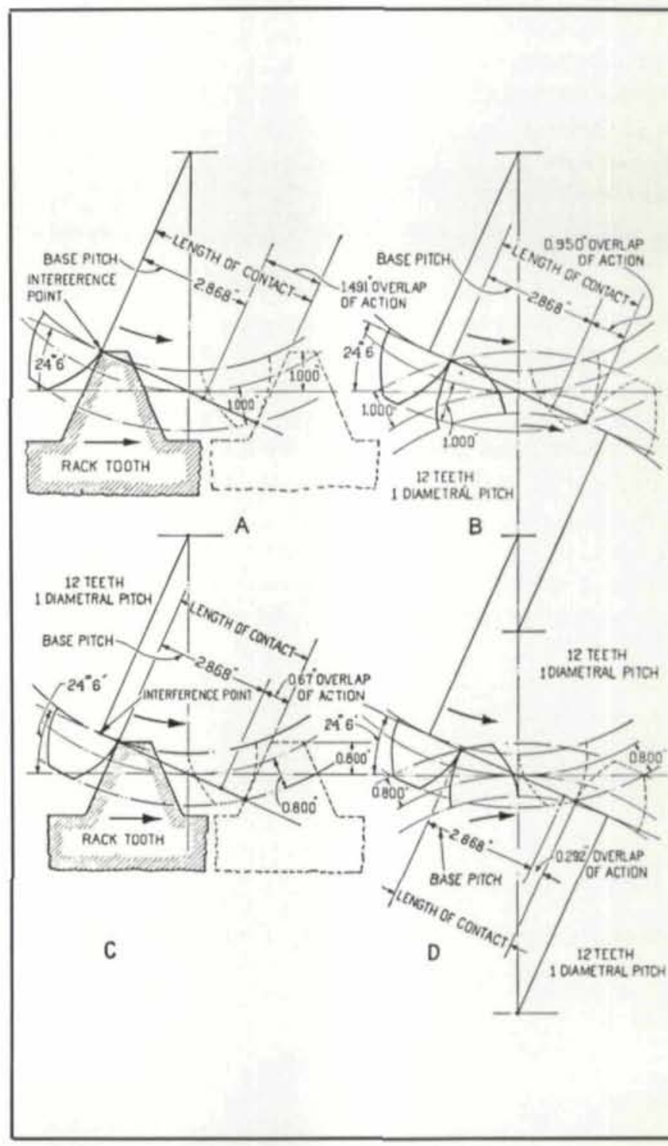


Fig. 5—Diagram illustrating gear and rack teeth having full-depth and stub-tooth forms of 24° 6′ pressure angle.

gives 0.950 ÷ 2.868, or approximately 33% overlap of action.

At *C* and *D*, Fig. 5, the addendums of the rack and pinions, respectively, have been shortened to 0.800 inch. In other words, the teeth are of stub-tooth form. At *C*, it will be noticed that contact between rack and pinion tooth does not start at the interference point, as was the case at *A*, but further along the line of action. At *C* the length of contact exceeds the base pitch by 0.670 inch. This gives 0.670 ÷ 2.868, or approximately 23% overlap of action. At *D*, contact of the two 12-tooth pinions is well inside the interference points, and hence the line of contact is shorter than in all the previous cases. There is only 0.292 inch difference between the base pitch and the length of contact. Hence, the overlap of action is only 0.292 ÷ 2.868, or approximately 10%. It will be seen from these diagrams that a pressure angle of 24° 6′ avoids interference and at the same time provides sufficient overlap of tooth action for the entire range of 12 teeth to a rack.

Relation of Pressure Angle to Interference

It has previously been explained that interference occurs when the involute of one tooth extends beyond the point of tangency of the line of action and the base circle. In Fig. 6, a 12-tooth pinion of 20 degrees pressure angle, full-depth tooth, is shown in engagement with a 40-tooth gear. It will be noted that the 40-tooth gear contacts the flank of the pinion tooth well in advance of the zone of contact. Therefore these gears will not operate satisfactorily, because the gear tooth "hooks" into the flank of the pinion tooth. The positions of the pinion and gear teeth, where involute interference commences, are indicated by the dotted outlines.

Involute interference between gear teeth can be determined graphically, as shown in Fig. 7, or by means of a simple calculation. In Fig. 7, $R_{(1)}$ represents the maximum permissible outside radius of the gear to avoid involute interference; R represents the base radius of the pinion; C is the center distance, and a the pressure angle.

Example: Assume that it is necessary to determine if involute interference will be present between a 10-pitch pinion of 12 teeth and a 30-tooth gear, the teeth to be 14½ degree pressure angle and of full-depth. The pitch diameter of the pinion is 1.20 inches, and the base diameter is 1.20 X cos 14½ degrees, or

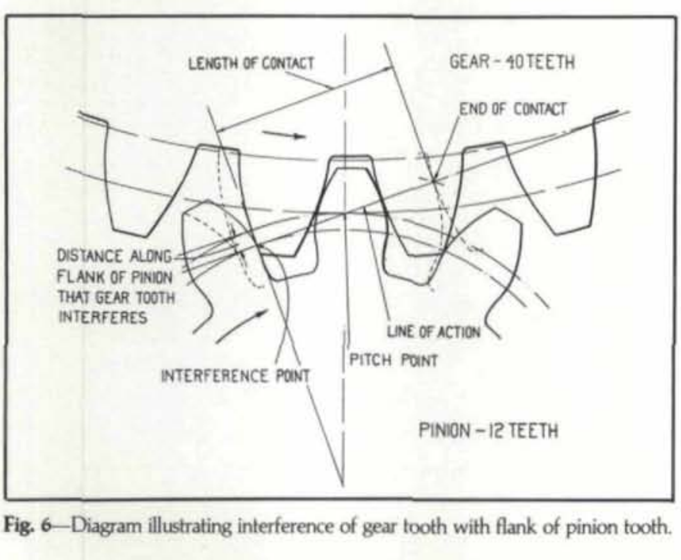


Fig. 6—Diagram illustrating interference of gear tooth with flank of pinion tooth.

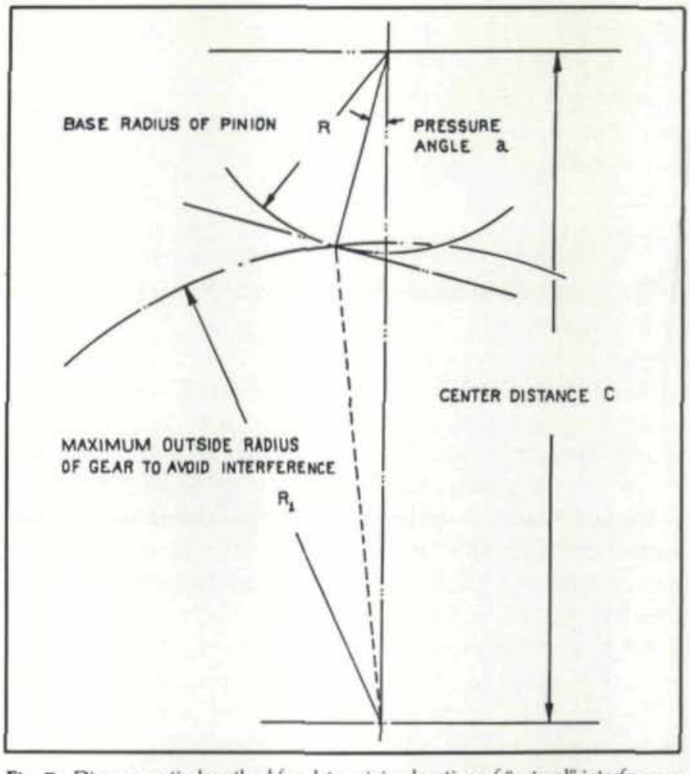


Fig. 7—Diagrammatical method for determining location of "natural" interference point.

1.1618 inches. The base radius of the pinion is therefore 1.1618 ÷ 2, or 0.5809 inch. The center distance *C* is 2.10 inches. The maximum permissible outside radius of the gear to avoid involute interference is found as follows:

$$R_1 = \sqrt{C^2 + R^2 - 2\,(CR)\cos a}$$

$$R_1 = \sqrt{2.10^2 + 0.5809^2 - 2(2.10 \times 0.5809)\cos 14\tfrac{1}{2}^\circ}$$

$$R_1 = 1.544 \text{ inch, approximately}$$

The standard outside radius of the gear is 1.60 inch which is greater than R_1 by 0.056 inch, indicating that the outside radius of the gear would have to be reduced 0.056 inch to avoid involute interference; or other methods, previously explained in last issue, would have to adopted.

Involute Interference between Gear and Rack Teeth

The previous example presented a method for determining involute interference between two gears. Fig. 8 illustrates a method for determining the smallest permissible number of teeth in a gear that will operate with a rack without involute interference. There are three controlling factors: diametral pitch, length of addendums and pressure angle.

When the pressure angle, diametral pitch, and addendums are known, the pitch radius and the smallest permissible number of teeth in the gear at which involute interference commences are determined as follows: Assume that it is necessary to find the smallest number of teeth in a gear of 10-diametral pitch, 0.100-inch addendum, and 14½° pressure angle.

Referring to Fig. 8, distance $X = A \times \cot. 14\tfrac{1}{2}^\circ = 0.38667$ inch. Distance $Y = X \times \cot. 14\tfrac{1}{2}^\circ = 1.4951$ inches. Then, the pitch radius $= Y + A$, or 1.4951″ +

0.1000" = 1.5951 inches, and the pitch diameter = 1.5951" × 2 = 3.190 inches. The number of teeth = pitch diameter × diametral pitch, or 3.190" × 10 = 31.9, or 32 teeth.

The minimum number of teeth for 20° full-length is 18, and for 20° stub teeth, with 8/10 addendum, is 14.

Relation of Pressure Angle, Addendum and Pitch to Length of Contact

There is a limit to the amount that the tooth can be modified, the pressure angle increased, or the teeth shortened, if continuous action is to result. As shown in Fig. 9, the length of contact L must be greater than the base pitch B to avoid lack of continuous action. With this particular tooth ratio, pitch, pressure angle and tooth length, the theoretical length of the line of action extends from points e to f. The actual usable length of the line of action—or line of contact—is determined by the outside radii of both gear and pinion. If interference were present, however, this would not be the case. The starting point of action is at point h where the outside radius of the pinion cuts the line of action, and could extend to point e without interference. In this case, however, the other limit of contact is at point k, where the outside radius of the gear cuts the line of action.

In general practice, it is considered that for the best action, the length of the line of contact should be at least 1¼ times the base pitch (the base pitch is the circular pitch transferred to the base circle). The length of the line of contact can be determined graphically as shown in Fig. 9, or it can be calculated.

In those cases where a small pressure angle and long addendums are used, and especially in conjunction with a small number of teeth in the pinion and a high ratio, the length of the line of contact is not controlled by the length of the addendums of both gear and pinion due to interference. If for instance, the pinion has such a small number of teeth that the distance hf (f, representing the interference point) is less than the distance hk, interference would be present to reduce the effective length of the line of contact.

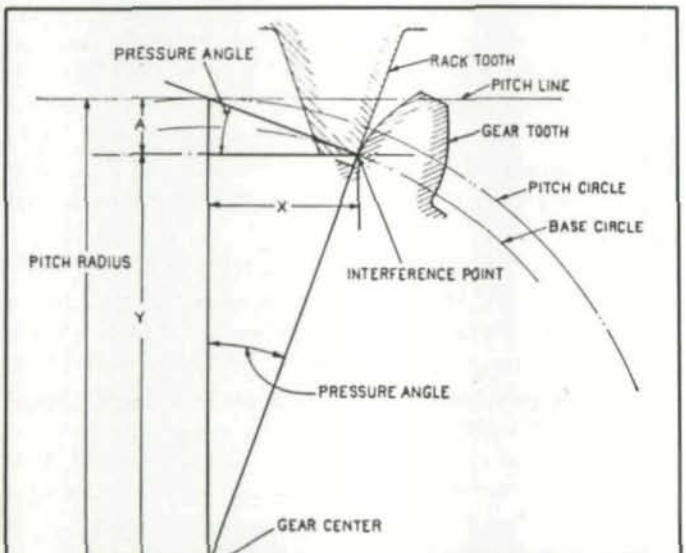


Fig. 8—Diagram illustrating method for determining permissible minimum number of teeth in a gear that will operate with a rack tooth without involute interference.

The length of the line of contact, ignoring the presence of interference, which should be determined separately, as previously explained, can be found by the following formula: (For notation see Fig. 9).

$$L = \sqrt{r_1^2 - r^2} + \sqrt{R_1^2 - R^2} - \sqrt{C^2 - (R + r)^2}$$

In which:

L = Active length of line of contact
r = Base radius of pinion
r_1 = Outside radius of pinion
R = Base radius of gear
R_1 = Outside radius of gear
C = Center distance

Example: Assume that it is necessary to determine the length of the line of contact L of a 10-pitch gear and pinion, the pinion having 15 teeth and the gear 30 teeth, the tooth form being 20 degrees full-depth.

Dimensions	*Pinion*	*Gear*
Pitch radii	0.750"	1.500"
Outside radii, r_1 and R_1	0.850"	1.600"
Base radii, r and R	0.7048"	1.4095"

Center distance, 2.250". Then:

$$L = \sqrt{0.850^2 - 0.7048^2} + \sqrt{1.600^2 - 1.4095^2} - \sqrt{2.250^2 - (.7048 + 1.4095)^2}$$

$$L = \sqrt{0.2258} + \sqrt{0.5733} - \sqrt{.5921}$$

$$L = 0.4752 + 0.7572 - 0.7695$$

$$L = 0.4629"$$

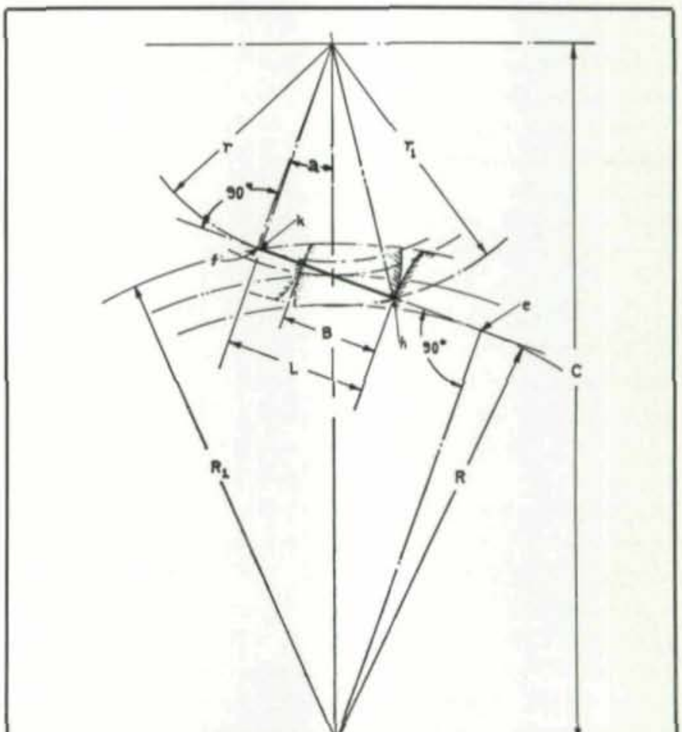


Fig. 9—Diagram illustrating mathematical method for determining actual length of tooth contact.

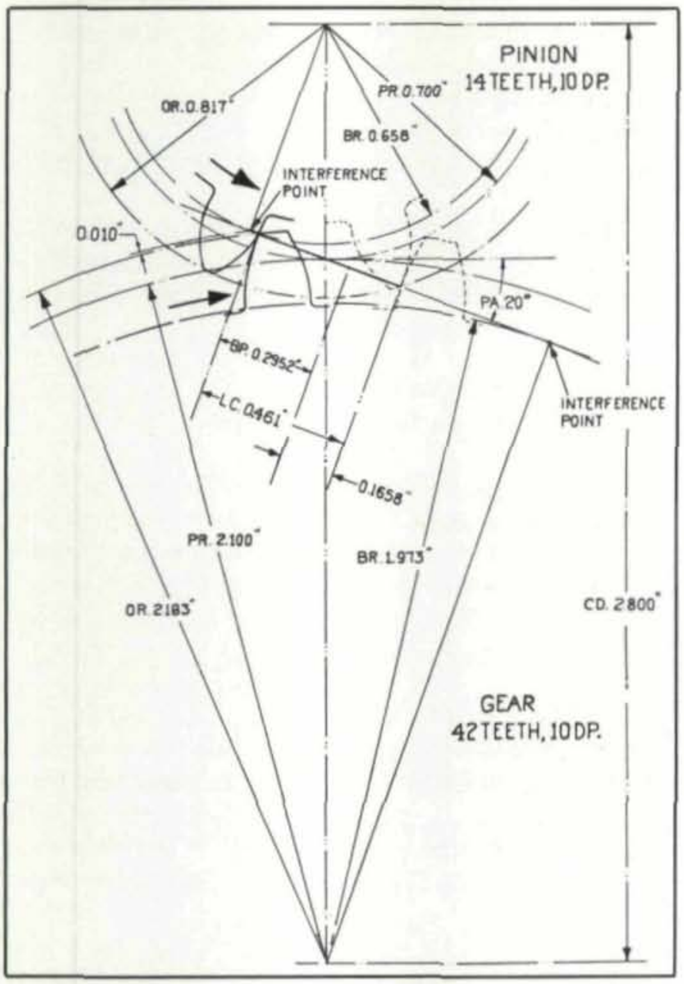


Fig. 10—Diagram illustrating method of designing gears for maximum efficiency by using long-and short-addendum teeth.

The contact ratio is then equal to the length of the line of contact *L* divided by the base pitch, and the base pitch is equal to the circular pitch times the cosine of the pressure angle. The circular pitch is 0.3142", and the base pitch is 0.3142" × cosine 20° = 0.2952". Then the contact ratio equals 0.4629" ÷ 0.2952 = 1.57 approximately, which is greater than the theoretical minimum required for the best action.

Designing Gears for Maximum Efficiency

Mention has already been made of the relationship of tooth ratio, tooth length, pressure angle, etc., to interference, undercut, length of line of contact, etc. When so-called standard tooth proportions, pressure angles, and interchangeability, are neither necessary nor desirable, it is possible to so proportion the addendums of gear and pinion, respectively, and select a pressure angle which will provide the best possible operating conditions to meet the requirements.

The first step, after the tooth ratio has been decided upon, is to select a diametral pitch, which, with standard tooth proportions, will provide the tooth strength necessary to carry the assumed load. The tooth ratio and pitch, of course, will establish the center distance and pitch diameters. The tooth ratio will have a bearing on the pressure angle selected, as a starting point in determining the proportions of the teeth.

As a concrete example, assume that the ratio is 3 to 1, and that 14 and 42 teeth of 10-diametral pitch have been selected. The calculated pitch diameters would then be: 1.400 inches for the pinion, and 4.200 inches for the gear. The center distance would then be:

$$\frac{4.200 + 1.400}{2}, \text{ or } 2.800 \text{ inches.}$$

To proceed, we lay out a diagram, on an enlarged scale, as shown in Fig. 10, space off the center distance, and assume a pressure angle of 20 degrees. This establishes the line of action and the interference points. In cases of an unequal ratio, it is always the larger of the two gears that is liable to cause interference. Hence, the outside circle of the gear should not extend beyond the interference point on the pinion tooth.

Assuming that the pinion is the driver, we can now proceed to lay out the teeth, and in order to be on the safe side, and avoid possible interference, we draw a circle representing the outside circle of the gear 0.010 inch inside the interference point. If the outside diameter of the gear, thus determined, is less than the standard diameter for a 10 pitch 42-tooth gear, then the outside diameter of the 14-tooth pinion would be enlarged a similar amount. The next step is to decide whether standard or special cutters will be used. Assume in this case that it is decided to use standard cutters, 20 degree pressure angle, full-depth teeth. The whole depth of a 10-pitch gear is 0.2250 inch. This distance for gear and pinion, respectively, is laid out on the center line, and circles drawn representing the root circle of the gear, and outside and root circles of the pinion. Where the outside circle of the gear cuts the line of action is one extremity of the line of contact, and where the outside circle of the pinion cuts the line of action is the other extremity.

We can now measure (or calculate), the actual length of contact, and by comparing this with the base pitch, can determine the overlap of action. If this meets the requirements, the problem is solved. It will be noticed in Fig. 10 that the normal pitch is 0.2952 inch, and the length of the line of contact is 0.461 inch. The overlap of action then equals (.461 ÷ .2952)—1, or 56%, approximately. The use of long- and short-addendums for pinion and gear, respectively, have avoided involute interference, and provided a sufficient overlap of tooth action.

Referring to Fig. 10, it will be seen that the outside diameter of the pinion has been increased from 1.600 inch to 1.634 inch, an increase of 0.034 inch. The outside diameter of the gear has been reduced from 4.400 to 4.366 inches, a decrease of 0.034 inch, the same amount as the pinion. In effect, long- and short-addendums for pinion and gear, respectively, have solved our problem.

If on the other hand, the ratio had been such that a sufficient overlap of action could not be obtained and interference avoided, other pressure angles could be used until the desired results had been obtained. It also might be necessary to use special cutters—this would especially be the case if it was desired to balance the teeth in gear and pinion—, respectively, for strength. In most cases, the pinion teeth would be weaker than the gear teeth; therefore, the thickness of the teeth on the gear would be reduced, and the thickness of the pinion teeth increased.

Pitch Diameter and Its Relation to Center Distance

The pitch circles of a pair of gears are the imaginary circles on which the gear teeth "roll" without slippage. These circles are tangent to each other at the pitch point. The radii of the pitch circles of a pair of gears are determined by dividing the center distance into the same proportion as the numbers of teeth in the two mating members. Thus, for a given center distance and tooth ratio, the pitch circle diameters are fixed. In some cases, in order to indicate backlash, the pitch diameters are dimensioned a slight amount undersize. This procedure is incorrect. Backlash is obtained by decreasing the thickness of the teeth, and should be indicated by a chordal tooth thickness dimension. Backlash in a pair of gears can be determined by the thickness of a feeler, which can be placed between the teeth, or by a change in center distance, see Fig. 11.

When gears are cut by the generating method, backlash between the teeth can be obtained either by using a cutter with teeth thicker than standard, or by feeding the cutter in to a sufficient depth to reduce the thickness of the gear teeth the necessary amount.

Backlash between Gear Teeth

Theoretically speaking, gear teeth should run together without appreciable backlash. From a practical standpoint, however, this is impossible due to the following reasons: 1. Perfection in cutting and mounting is an impossible achievement because of manufacturing tolerances, which should be as wide as possible to reduce costs. 2. Space between the teeth must be provided to aid lubrication. 3. Temperature changes due to speed and other causes affect sizes of gears and spacing of shafts on which gears are mounted. In view of these conditions, it is necessary to provide a certain amount of freedom between the teeth, so that they will not bind when operating together.

The term "backlash" can be defined as the amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the pitch circles, as actually indicated by measuring devices. Backlash may be determined in the plane of rotation or normal plane, and along the line of action.

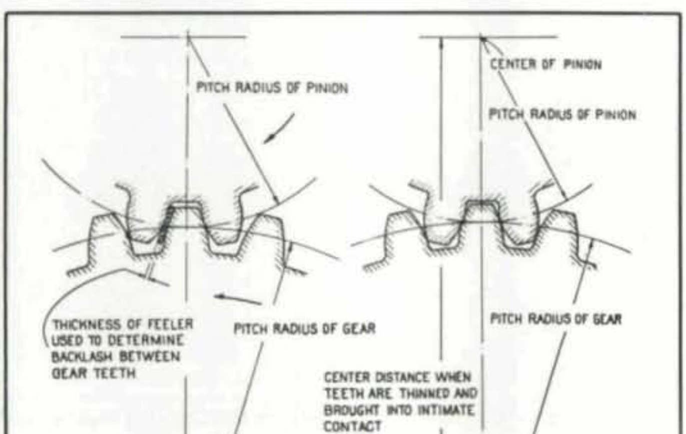


Fig. 11—Diagram illustrating two methods of determining backlash between gear teeth.

Methods for Determining Amount of Backlash

Several methods are used for indicating and checking backlash. The common method, particularly with spur gears, is to place the mating gears on pins located at the correct center distance, and then measure the backlash by the use of a feeler gage inserted between the teeth, as shown at *A* in Fig. 11.

Another method is to place the gears on pins and bring the teeth into intimate contact, and then determine the difference between the standard and measured center distance, as shown at *B*, Fig. 11. This last check does not indicate backlash directly, but the amount of backlash can be determined by the following formula:

$$B = 2 \tan a \times d$$

In which:

- B = Backlash in inches
- a = Pressure angle
- d = Difference between standard and measured center distances

A third method, shown diagrammatically in Fig. 12, is to use a dial indicator in connection with a fixture for holding the gears on studs. One of the gears should be fixed so that it cannot rotate on the stud. The gears are set at standard center distance. The "free" gear is rotated so that its profile (away from the indicator plunger) is in intimate contact with the other gear. The indicator plunger is then set in contact with the profile of one tooth, and the needle set at zero. The "free" gear is then rotated to bring the opposite sides of the teeth in contact, and the reading on the dial indicator noted. This reading indicates the relative "rotary" movement of the "free" gear, and, hence, the backlash or freedom that exists between the teeth.

In checking helical gears, the backlash is measured in the normal plane, instead of in the plane of rotation, as is the case with spur gears. The method just described can be applied satisfactorily to both spur and helical gears. In providing for backlash, it is customary, when a small pinion is to operate with a larger gear, to reduce the thickness of the teeth on the larger gear to provide the necessary backlash, leaving the pinion teeth of standard tooth thickness.

Effect of Center Distance Change and Pressure Angle on Amount of Backlash

The chart in Fig. 13 is presented to illustrate how a change

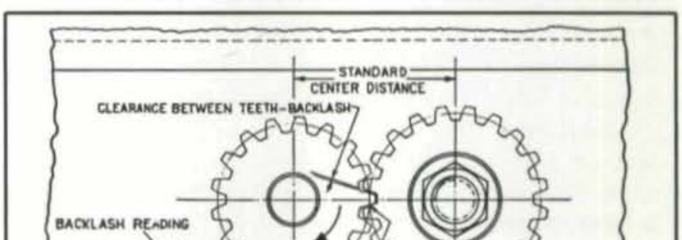


Fig. 12—Diagram illustrating method of measuring backlash with dial indicator—gears held on pins at standard center distance.

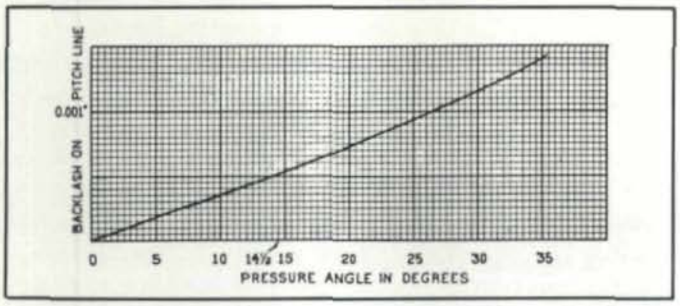


Fig. 13—*Chart illustrating how pressure angle affects change in backlash for each 0.001 inch change in center distance.*

in pressure angle affects the amount of backlash between gear teeth when the center distance is increased. As shown on this chart for each 0.001 inch increase in center distance, the backlash between the teeth increases as the pressure angle is increased. This increase in backlash for the various pressure angles listed for each 0.001 inch change in center distance is as follows:

Pressure Angle in Degrees	*Increase in Backlash in Inches*
5	0.00017
10	0.00035
14½	0.00052
15	0.00054
20	0.00073
25	0.00094
30	0.00115
35	0.0014

For example, the difference in the amount of backlash for each 0.001 inch change in the center distance between 14½ and 20 degrees pressure angles is: 0.00073 — 0.00052 = 0.00021 inch more backlash for 20 than 14½ degrees pressure angle.

Other Factors Affecting Amount of Backlash between Mating Gear Teeth

In addition to changes in center distance, several other factors affect the actual amount of backlash between mating gear teeth such as: tooth spacing errors, runout, and errors in lead of helical gears. Obviously, the amount should be sufficient to permit the gears to rotate freely when cut to prescribed manufacturing tolerance. Table I lists minimum, maximum, and average backlash for spur and helical gears.

It should be understood that gears, which are operated at high speeds, require more backlash than gears operating at slower speeds. The values in Table I should prove satisfactory for gears operating at speeds up to 1500 surface feet per minute. For gears operating at speeds in excess of 1500 surface feet per minute, the values should be increased slightly over those listed in the maximum column. Gears which are to operate at higher speeds should also be more accurately cut and mounted than slower operating gears.

In precision gears, where close tolerances on backlash are demanded, several methods are employed. One method is to use selective assembly; the other is to mount the gears so that the center distance can be adjusted, such as by the use of eccentric bushings.

TABLE I

BACKLASH BETWEEN TEETH OF MATING SPUR AND HELICAL GEARS

DIAMETRAL PITCH	BACKLASH IN INCHES		
	MINIMUM	AVERAGE	MAXIMUM
1	0.025	0.0325	0.040
1½	0.018	0.0225	0.027
2	0.014	0.0170	0.020
2½	0.011	0.0135	0.016
3	0.009	0.0115	0.014
4	0.007	0.0090	0.011
5	0.006	0.0075	0.009
6	0.005	0.0065	0.008
7	0.004	0.0055	0.007
8 and 9	0.004	0.0050	0.006
10 to 13	0.003	0.0040	0.005
14 to 19	0.003	0.0040	0.005
20 to 40	0.002	0.0030	0.004
41 to 60	0.0015	0.0022	0.003
61 to 120	0.0010	0.0015	0.002
121 and Finer	0.0005	0.0007	0.001

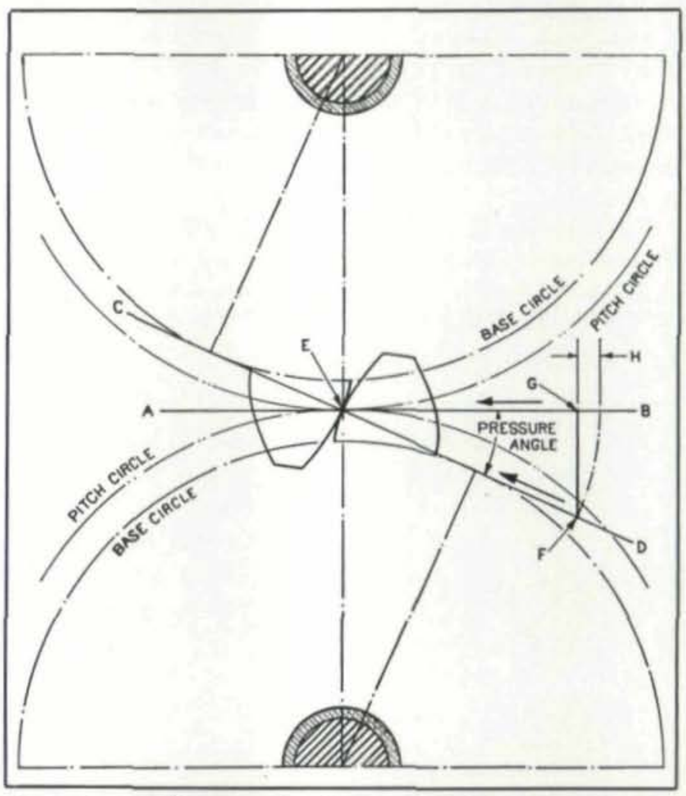


Fig. 14—Diagram illustrating effect of an increase in the pressure angle on the load transmitted to the supporting bearings.

Relation of Pressure Angle to Load on Supporting Bearings

An increase in the pressure angle does not have a marked effect on the resultant load on the supporting bearings as is

(Continued on page 45)

Double Enveloping Worm Gears . . .
(Continued from Page 16)

(2) reductions at approximately 92%-93% overall efficiency or three (3) reductions at about 89%-90% efficiency. The worm gearbox with a 20:1 ratio will have about 85%-87% efficiency. A 30:1 ratio helical reducer will generally require three (3) meshes with approximately 89%-90% efficiency. The 30:1 wormgear speed reducer will have an efficiency of approximately 83%-84%. You can see the helical box *is* more efficient, but certainly not to the degree often claimed.

There are other inherent advantages in worm gearing which must be considered in evaluating the application and the type of gearing intended for that application. Double enveloping worm gearing will take a momentary overload of 300%, whereas helical gearboxes are only designed for 200%, momentary overload. Helical gearboxes restrict motor starting capacity to 200%, whereas double enveloping worm gearboxes permit 300%. Generally speaking, worm gearboxes are smaller in overall size and weight, and in terms of horsepower capacity, generally less expensive. In addition, with compactness of the double enveloping wormgear principle, double enveloping gearboxes are more compact and weigh less, horsepower for horsepower, than cylindrical gear reducers.

This paper was published for the National Conference on Power Transmissions 1979 and reprinted in "Technical Aspects of Double Enveloping Worm Gears, a Cone Drive Publication.

Design of the Involute . . .
(Continued from page 44)

generally supposed. In other words, bearing pressures are not greatly affected by an increase in the pressure within the usual limits. This condition is graphically presented in Fig. 14. To construct this diagram, draw a line *A B* at right angles to the line of centers and tangent to both pitch circles. Then draw a line *C D* tangent to the base circles and passing through the pitch point *E;* this line representing the pressure angle. Now drop a perpendicular at any point *G* on line *A B*, passing through line *C D* at point *F*. With *E* as a center and *E F* as a radius scribe an arc. Increases in the load on the supporting bearings due to changes in pressure angle can be determined graphically by noting the changes in distance *H*, as the pressure angle changes. It is apparent that the load-increase is the ratio of lengths *E G* to *E F*, and is, therefore, proportional to the secant of the pressure angle.

The second column in Table II gives the secants of various pressure angles listed in the first column, and ranging from 14½ up to and including 30 degrees.

The last column lists in terms of percentage, the increase in the load as compared with 14½ degrees. It will be noticed that an increase in the pressure angle from 14½ to 20 degrees, results in an increased load on the supporting bearings of only 3 percent.

(Continued on the next page)

Scoring Load Capacity . . .
(Continued from page 30)

Conclusion

A new method for scoring load capacity rating, based on the calculation of a mean, weighted flank temperature, the integral temperature, has been described. The limiting temperatures necessary, for the definition of a scoring safety factor, can be obtained from any available gear oil test. The method is valid for all types of oils as straight mineral, mild and EP-oils, as well as, synthetic oils where gear scoring tests are available. The method was checked with more than 300 scoring tests on test rigs and more than 100 practical gears with and without scoring damages. A good correlation was found for the Integral Temperature Criterion, and it was obviously superior to the Total Temperature Method, as well as, to the Scoring Index Method.

The method has been modified for bevel and hypoid gears(10) and even in this field of application a good correlation between calculated scoring factors and field experience was achieved.

References

1. Blok, H.: Theoretical Study of Temperature Rise at Surface of Actual Contact under Oiliness Lubricating Conditions. Proc. Gen. Disc. Lubric. I. Mech. Eng., London, Bd.2 (1937) S. 225-235.
2. Dudley, D.W.: Gear Handbook, McGraw-Hill, New York, 1962.
3. Winter, H.; Michaelis, K.: Fresstragfähigkeit von Stirnradgetrieben. Antriebstechnik 14 (1975) S. 405-409 und 461-465.
4. Winter, H.; Michaelis, K.: Investigations on the Thermal Balance of Gear Drives. In: Proceedings of the Fifth World Congress on Theory of Machines and Mechanisms, Vol. I, July 8-13, 1979, Montreal, S. 354-358.
5. Lechner, G.: Die Fress-Grenzlast bei Stirnrädern aus Stahl. Diss. TH-München 1966.
6. Blok, H.: The Postulate about the Constancy of Scoring Temperature. Interdiscipl. Approach to the Lubrication of Concentrated Contacts (ed. P.M.Ku), NASA SP-237, 1970.
7. Vogelpohl, G.: Verschleiss in Maschinen und die Möglichkeiten seiner Verminderung mit Hilfe der auf Ölprüfgeräten gewonnenen Ergebnisse. Erdöl und Kohle 14 (1961) S.824-829.
8. Wirtz, H.: Schmierstoffe und anwendungsbezogene Schmierstoffprüfung. Vortrag TA Wuppertal, Mai 1980.
9. Ishikawa, J.; Hayashi, K.; Yokoyama, M.: Measurement of Surface Temperature on Gear Teeth by Dynamic Thermocouple Method. Bull. Tokyo Inst. Techn. 112 (1972) S. 107-121.
10. Winter, H.; Michaelis, K.: Berechnung der Fresstragfähigkeit von Hypoidgetrieben. Antriebstechnik 21 (1982) S. 382-387.

TABLE II

RELATION BETWEEN PRESSURE ANGLE AND LOAD ON BEARINGS

PRESSURE ANGLE DEGREES	SECANT	COMPARISON OF INCREASED LOAD ON SUPPORTING BEARINGS IN TERMS OF PERCENTAGE
14½	1.0329	
17	1.0457	1½%
20	1.0642	3%
22½	1.0824	4.8%
24-6	1.0955	6%
27½	1.1274	9.1%
30	1.1547	11.3%

TABLE III

GEAR TOOTH PARTS

Standards in **Bold-Face** Type. For Full-Length Tooth Generated Gears

Dedendum = 1.250 ÷ D. P. Up to and Including 19 D. P.

Diametral Pitch (D P)	Circular Pitch (C P) (Inches)	Module (M)	DIMENSIONS IN INCHES Circular Thickness (CTh.)	Addendum (A)	Dedendum (D)	Whole Tooth Depth (W D)	Double Tooth Depth (D D)
3.1416	1	8.0851	0.5000	0.3183	0.3979	0.7162	1.4324
3.1750	0.9895	8	0.4947	0.3150	0.3937	0.7087	1.4174
3.2774	0.9586	7¾	0.4793	0.3052	0.3814	0.6866	1.3732
3.3510	15/16	7.5798	0.4687	0.2984	0.3730	0.6714	1.3428
3.3867	0.9276	7½	0.4638	0.2953	0.3691	0.6644	1.3288
3.5034	0.8967	7¼	0.4484	0.2854	0.3568	0.6422	1.2844
3.5904	⅞	7.0744	0.4375	0.2785	0.3482	0.6267	1.2534
3.6286	0.8658	7	0.4329	0.2756	0.3445	0.6201	1.2402
3.7630	0.8349	6¾	0.4174	0.2657	0.3322	0.5979	1.1958
3.8666	13/16	6.5691	0.4062	0.2586	0.3233	0.5819	1.1638
3.9077	0.8040	6½	0.4020	0.2559	0.3199	0.5758	1.1516
4	0.7854	6.3500	0.3927	0.2500	0.3125	0.5625	1.1250
4.0640	0.7730	6¼	0.3865	0.2460	0.3076	0.5536	1.1072
4.1888	¾	6.0638	0.3750	0.2387	0.2984	0.5371	1.0742
4.2333	0.7421	6	0.3711	0.2362	0.2952	0.5314	1.0628
4.4174	0.7112	5¾	0.3556	0.2264	0.2830	0.5094	1.0188
4.5696	11/16	5.5585	0.3437	0.2188	0.2735	0.4923	0.9846
4.6182	0.6803	5½	0.3401	0.2165	0.2707	0.4872	0.9744
4.7124	⅔	5.3900	0.3333	0.2122	0.2653	0.4775	0.9550
4.8381	0.6493	5¼	0.3247	0.2067	0.2584	0.4651	0.9302
5	0.6283	5.0800	0.3142	0.2000	0.2500	0.4500	0.9000
5.0266	⅝	5.0531	0.3125	0.1989	0.2487	0.4476	0.8952
5.0800	0.6184	5	0.3092	0.1968	0.2461	0.4429	0.8858
5.3474	0.5875	4¾	0.2938	0.1870	0.2338	0.4208	0.8416
5.5851	9/16	4.5478	0.2812	0.1790	0.2238	0.4028	0.8058
5.6444	0.5566	4½	0.2783	0.1772	0.2215	0.3987	0.7974
5.9765	0.5257	4¼	0.2628	0.1673	0.2091	0.3764	0.7528
6	0.5236	4.2333	0.2618	0.1667	0.2083	0.3750	0.7500
6.2832	½	4.0425	0.2500	0.1591	0.1989	0.3580	0.7160
6.3500	0.4947	4	0.2473	0.1575	0.1968	0.3543	0.7086
6.7733	0.4638	3¾	0.2319	0.1476	0.1845	0.3321	0.6642
7	0.4488	3.6286	0.2244	0.1429	0.1786	0.3215	0.6430
7.1808	7/16	3.5372	0.2187	0.1393	0.1741	0.3134	0.6268
7.2571	0.4329	3½	0.2164	0.1378	0.1722	0.3100	0.6200
7.8154	0.4020	3¼	0.2010	0.1279	0.1599	0.2878	0.5756
8	0.3927	3.1750	0.1964	0.1250	0.1563	0.2813	0.5626
8.3776	⅜	3.0319	0.1875	0.1194	0.1492	0.2686	0.5372
8.4667	0.3711	3	0.1855	0.1181	0.1477	0.2658	0.5316
9	0.3491	2.8222	0.1745	0.1111	0.1389	0.2500	0.5000
9.2364	0.3401	2¾	0.1701	0.1082	0.1353	0.2435	0.4870
9.4248	⅓	2.6950	0.1667	0.1061	0.1326	0.2387	0.4774
10	0.3142	2.5400	0.1571	0.1000	0.1250	0.2250	0.4500
10.0531	5/16	2.5266	0.1562	0.0995	0.1244	0.2239	0.4478
10.1600	0.3092	2½	0.1546	0.0984	0.1230	0.2214	0.4428
11	0.2856	2.3091	0.1428	0.0909	0.1137	0.2046	0.4092
11.2889	0.2783	2¼	0.1391	0.0886	0.1107	0.1993	0.3986
12	0.2618	2.1167	0.1309	0.0833	0.1042	0.1875	0.3750
12.5664	¼	2.0213	0.1250	0.0796	0.0995	0.1791	0.3582

TABLE IV

GEAR TOOTH PARTS

Standards in **Bold-Face** Type. For Full-Length Tooth Generated Gears

Dedendum = 1.250 ÷ D. P. Up to and Including 19 D. P.

Dedendum = 1.200 ÷ D. P. + 0.002" for 20 D. P. and Finer

Diametral Pitch (D P)	Circular Pitch (C P) (Inches)	Module (M)	DIMENSIONS IN INCHES Circular Thickness (CTh.)	Addendum (A)	Dedendum (D)	Whole Tooth Depth (W D)	Double Tooth Depth (D D)
12.7000	0.2474	2	0.1237	0.0787	0.0984	0.1771	0.3542
13	0.2417	1.9538	0.1208	0.0769	0.0962	0.1731	0.3462
14	0.2244	1.8143	0.1122	0.0714	0.0893	0.1607	0.3214
14.5143	0.2164	1¾	0.1082	0.0689	0.0861	0.1550	0.3100
15	0.2094	1.6933	0.1047	0.0667	0.0833	0.1500	0.3000
16	0.1964	1.5875	0.0982	0.0625	0.0781	0.1406	0.2812
16.7552	3/16	1.5159	0.0938	0.0597	0.0746	0.1343	0.2686
16.9333	0.1855	1½	0.0928	0.0591	0.0738	0.1329	0.2658
17	0.1848	1.4941	0.0924	0.0588	0.0735	0.1323	0.2646
18	0.1745	1.4111	0.0873	0.0556	0.0694	0.1250	0.2500
19	0.1653	1.3368	0.0827	0.0526	0.0658	0.1184	0.2368
20	0.1571	1.2700	0.0785	0.0500	0.0620	0.1120	0.2240
20.3200	0.1546	1¼	0.0773	0.0492	0.0611	0.1103	0.2206
21	0.1496	1.2095	0.0748	0.0476	0.0591	0.1067	0.2134
22	0.1428	1.1545	0.0714	0.0455	0.0565	0.1020	0.2040
23	0.1366	1.1043	0.0683	0.0435	0.0542	0.0977	0.1954
24	0.1309	1.0583	0.0655	0.0417	0.0520	0.0937	0.1874
25	0.1257	1.0160	0.0628	0.0400	0.0500	0.0900	0.1800
25.1328	⅛	1.0106	0.0625	0.0398	0.0497	0.0895	0.1790
25.4000	0.1237	1	0.0618	0.0394	0.0492	0.0886	0.1772
26	0.1208	0.9769	0.0604	0.0385	0.0482	0.0867	0.1734
27	0.1164	0.9407	0.0582	0.0370	0.0464	0.0834	0.1668
28	0.1122	0.9071	0.0561	0.0357	0.0449	0.0806	0.1612
29	0.1083	0.8759	0.0542	0.0345	0.0434	0.0779	0.1558
30	0.1047	0.8467	0.05236	0.0333	0.0420	0.0753	0.1506
31	0.1013	0.8194	0.05067	0.0323	0.0407	0.0730	0.1460
32	0.0982	0.7938	0.04909	0.0313	0.0395	0.0708	0.1416
33	0.0952	0.7697	0.04760	0.0303	0.0384	0.0687	0.1364
33.8667	0.0928	¾	0.04638	0.0295	0.0374	0.0669	0.1338
34	0.0924	0.7471	0.04620	0.0294	0.0373	0.0667	0.1334
35	0.0898	0.7257	0.04488	0.0286	0.0363	0.0649	0.1298
36	0.0873	0.7056	0.04363	0.0278	0.0353	0.0631	0.1262
37	0.0849	0.6865	0.04245	0.0270	0.0344	0.0614	0.1228
38	0.0827	0.6684	0.04134	0.0263	0.0336	0.0599	0.1198
39	0.0806	0.6513	0.04028	0.0256	0.0328	0.0584	0.1168
40	0.0785	0.6350	0.03927	0.0250	0.0320	0.0570	0.1140
41	0.0766	0.6195	0.03831	0.0244	0.0313	0.0557	0.1114
42	0.0748	0.6048	0.03740	0.0238	0.0306	0.0544	0.1088
43	0.0731	0.5907	0.03653	0.0233	0.0299	0.0532	0.1064
44	0.0714	0.5773	0.03570	0.0227	0.0293	0.0520	0.1040
45	0.0698	0.5644	0.03491	0.0222	0.0287	0.0509	0.1018
46	0.0683	0.5522	0.03415	0.0217	0.0281	0.0498	0.0996
47	0.0668	0.5404	0.03342	0.0213	0.0275	0.0488	0.0976
48	0.0655	0.5292	0.03272	0.0208	0.0270	0.0478	0.0956
49	0.0641	0.5184	0.03205	0.0204	0.0265	0.0469	0.0938
50	0.0628	0.5080	0.03141	0.0200	0.0260	0.0460	0.0920
50.2656	1/16	0.5053	0.03130	0.0199	0.0258	0.0457	0.0914
50.8000	0.0618	½	0.03092	0.0197	0.0256	0.0453	0.0906

TABLE V

GEAR TOOTH PARTS

Full-Length Teeth Fine-Pitch Gears

Dedendum = 1.2000 ÷ D. P. + 0.002"

Whole Depth = 2.2000 ÷ D. P. + 0.002"

Diametral Pitch (D P)	DIMENSIONS IN INCHES Circular Thickness (CTh.)	Addendum (A)	Dedendum (D)	Whole Tooth Depth (W D)	Double Tooth Depth (D D)
51	0.03080	0.0196	0.0255	0.0451	0.0902
52	0.03021	0.0192	0.0251	0.0443	0.0886
53	0.02964	0.0189	0.0246	0.0435	0.0870
54	0.02909	0.0185	0.0242	0.0427	0.0854
55	0.02856	0.0182	0.0238	0.0420	0.0840
56	0.02805	0.0179	0.0234	0.0413	0.0826
57	0.02756	0.0175	0.0231	0.0406	0.0812
58	0.02708	0.0172	0.0227	0.0399	0.0798
59	0.02662	0.0169	0.0223	0.0393	0.0786
60	0.02618	0.0167	0.0220	0.0387	0.0774
61	0.02575	0.0164	0.0217	0.0381	0.0762
62	0.02534	0.0161	0.0214	0.0375	0.0750
63	0.02493	0.0159	0.0210	0.0369	0.0738
64	0.02454	0.0156	0.0208	0.0364	0.0728
65	0.02417	0.0154	0.0205	0.0358	0.0716
66	0.02380	0.0152	0.0202	0.0353	0.0706
67	0.02344	0.0149	0.0199	0.0348	0.0696
68	0.02310	0.0147	0.0196	0.0344	0.0688
69	0.02277	0.0145	0.0194	0.0339	0.0678
70	0.02244	0.0143	0.0191	0.0334	0.0668
72	0.02182	0.0139	0.0187	0.0326	0.0652
74	0.02123	0.0135	0.0182	0.0317	0.0634
76	0.02067	0.0132	0.0178	0.0309	0.0618
78	0.02014	0.0128	0.0174	0.0302	0.0604
80	0.01964	0.0125	0.0170	0.0295	0.0590
82	0.01916	0.0122	0.0166	0.0288	0.0576
84	0.01870	0.0119	0.0163	0.0282	0.0564
86	0.01827	0.0116	0.0160	0.0276	0.0552
88	0.01785	0.0114	0.0156	0.0270	0.0540
90	0.01745	0.0111	0.0153	0.0264	0.0528
92	0.01707	0.0109	0.0150	0.0259	0.0518
94	0.01671	0.0106	0.0148	0.0254	0.0508
96	0.01636	0.0104	0.0145	0.0249	0.0498
98	0.01603	0.0102	0.0142	0.0244	0.0488
100	0.01571	0.0100	0.0140	0.0240	0.0480
102	0.01540	0.0098	0.0138	0.0236	0.0472
104	0.01510	0.0096	0.0135	0.0232	0.0464

(Continued on page 48)

CLASSIFIED

Rates: $75 per column inch (minimum 1 inch) Classified Display-$100 per inch (minimum 3") Type will be set to advertiser's layout or *Gear Technology* will set type at no extra charge.
Word Count: 35 characters per line, 7 lines per inch.

Payment: Full payment must accompany classified ads. Mail copy to *Gear Technology*, P.O. Box 1426, Elk Grove Village, IL 60007.
Agency Commission: No agency commission on regular or display classifieds.

Closing date: Ads must be received by the 25th of the month, two months prior to publication. (e.g. August 25th for the October issue).
Acceptance: Publisher reserves the right to accept or reject classified advertisements at his discretion.

OPEN TIME

GENERATED GROUND GEARS
ANY PITCH, ANGLE, HELIX
MAX 4DP, 80" DIA
SPUR, HEL, FACE, INT, CONICAL

INVINCIBLE GEAR CO.
11970 MAYFIELD DEPT GT
LIVONIA, MI 48150
(313) 421-4620

We have a very modern CNC machine shop with capability of cutting spur and helical gears up to 96". Excellent Quality Assurance Department to oversee machining, gear cutting and assembly. We have been providing production machinery since 1946.

GEO CHURCHILL, INC.

3701 Johnson
Safari Industrial Park
P.O. Box 518
Chanute, Kansas 66721

Contact: Bernard Dick
316-431-0500

HELP WANTED

GEAR ENGINEER

Position open for applicants with extensive experience in air craft gear processing. Must have complete knowledge of gear manufacturing. Applicants with gearbox design preferred. Must be ambitious and have a desire to advance.

ACR INDUSTRIES, INC.

29200 Calahan
Roseville, MI 48066

(313) 773-0450

SUBCONTRACT WORK

SPIRAL BEVEL GEARS

Protype or production up to 34"
Breakdown work — fast turnaround
Complete machining, heat treat, lapping and testing

CALL US!

B&R MACHINING
PO Box 536
Sharon, TN 38255
1-800-238-0651

Gleason Spiral Bevel Gears Specialist any quantity 12" to 100" diameter

AMARILLO GEAR
PO Box 1789
Amarillo, Texas 79105
(806) 622-1273

There's still time . . .
closing date for a classified ad in the Dec/Jan Issue is **October 25th.**

TELL OUR ADVERTISERS YOU SAW IT IN GEAR TECHNOLOGY

A FULL SERVICE COMPANY SUPPLYING THE AEROSPACE AND COMMERCIAL INDUSTRIES WITH QUALITY PRODUCTS.

VINCO® B-1 RADIUS/ANGLE WHEEL DRESSER
VINCO® SPLINEDICATORS® AND WEAR GAGES
VINCO® GEAR MEASURING INSTRUMENTS
GEAR COMPONENTS AND PARTS
OPERATIONAL GEAR GRINDING
SPLINE GAGES AND ARBORS
MASTER GEARS

TIFCO SPLINE, INC.
VINCO® PRODUCTS
29905 ANTHONY DRIVE WIXOM, MICHIGAN 48096
(313) 398-6260

SINCE 1963

TABLE VI

GEAR TOOTH PARTS

Full-Length Teeth Fine-Pitch Gears
Dedendum = 1.2000 ÷ D. P. + 0.002"
Whole Depth = 2.2000 ÷ D. P. + 0.002"

Diametral Pitch (D P)	DIMENSIONS IN INCHES Circular Thickness (CTh.)	Addendum (A)	Dedendum (D)	Whole Tooth Depth (W D)	Double Tooth Depth (D D)
106	0.01482	0.0094	0.0133	0.0228	0.0456
108	0.01454	0.0093	0.0131	0.0224	0.0448
110	0.01428	0.0091	0.0129	0.0220	0.0440
112	0.01402	0.0089	0.0127	0.0216	0.0432
114	0.01378	0.0088	0.0125	0.0213	0.0426
116	0.01354	0.0086	0.0123	0.0210	0.0420
118	0.01331	0.0085	0.0122	0.0206	0.0412
120	0.01309	0.0083	0.0120	0.0203	0.0406
122	0.01288	0.0082	0.0118	0.0200	0.0400
124	0.01267	0.0081	0.0117	0.0197	0.0394
126	0.01247	0.0079	0.0115	0.0195	0.0390
128	0.01227	0.0078	0.0114	0.0192	0.0384
130	0.01208	0.0077	0.0112	0.0189	0.0378
132	0.01190	0.0076	0.0111	0.0187	0.0374
134	0.01172	0.0075	0.0110	0.0184	0.0368
136	0.01155	0.0074	0.0108	0.0182	0.0364
138	0.01138	0.0072	0.0107	0.0179	0.0358
140	0.01122	0.0071	0.0106	0.0177	0.0354
142	0.01106	0.0070	0.0105	0.0175	0.0350
144	0.01091	0.0069	0.0103	0.0173	0.0346
146	0.01076	0.0068	0.0102	0.0171	0.0342
148	0.01061	0.0068	0.0101	0.0169	0.0338
150	0.01047	0.0067	0.0100	0.0167	0.0334
152	0.01033	0.0066	0.0099	0.0165	0.0330
154	0.01020	0.0065	0.0098	0.0163	0.0326
156	0.01007	0.0064	0.0097	0.0161	0.0322
158	0.00994	0.0063	0.0096	0.0159	0.0318
160	0.00982	0.0063	0.0095	0.0158	0.0316
170	0.00924	0.0059	0.0091	0.0149	0.0298
180	0.00873	0.0056	0.0087	0.0142	0.0284
190	0.00827	0.0053	0.0083	0.0136	0.0272
200	0.00785	0.0050	0.0080	0.0130	0.0260
210	0.00748	0.0048	0.0077	0.0125	0.0250
220	0.00714	0.0045	0.0075	0.0120	0.0240
230	0.00683	0.0043	0.0072	0.0116	0.0232
240	0.00655	0.0042	0.0070	0.0112	0.0224
250	0.00628	0.0040	0.0068	0.0108	0.0216

TABLE VII

GEAR TOOTH PARTS

ASA Standard Stub-Tooth Gears
Addendum = .8 ÷ D. P.
Dedendum = *1 ÷ D. P.
Whole Depth = *1.8 ÷ D. P.

Diametral Pitch (D P)	DIMENSIONS IN INCHES Circular Thickness (CTh.)	Addendum (A)	Dedendum (D)	Whole Tooth Depth (W D)	Double Tooth Depth (D D)
3	0.5236	0.2667	0.3333	0.6000	1.2000
3½	0.4488	0.2286	0.2857	0.5143	1.0286
4	0.3927	0.2000	0.2500	0.4500	0.9000
5	0.3142	0.1600	0.2000	0.3600	0.7200
6	0.2618	0.1333	0.1667	0.3000	0.6000
7	0.2244	0.1143	0.1428	0.2571	0.5142
8	0.1963	0.1000	0.1250	0.2250	0.4500
9	0.1745	0.0889	0.1111	0.2000	0.4000
10	0.1571	0.0800	0.1000	0.1800	0.3600
11	0.1428	0.0727	0.0909	0.1636	0.3272
12	0.1309	0.0667	0.0833	0.1500	0.3000
13	0.1208	0.0615	0.0769	0.1384	0.2768
14	0.1122	0.0571	0.0714	0.1285	0.2570
15	0.1047	0.0533	0.0667	0.1200	0.2400
16	0.0982	0.0500	0.0625	0.1125	0.2250
17	0.0924	0.0471	0.0588	0.1059	0.2118
18	0.0873	0.0444	0.0556	0.1000	0.2000
19	0.0827	0.0421	0.0526	0.0947	0.1894
20	0.0785	0.0400	0.0500	0.0900	0.1800
22	0.0714	0.0364	0.0454	0.0818	0.1636
24	0.0654	0.0333	0.0417	0.0750	0.1500
26	0.0604	0.0308	0.0384	0.0692	0.1384
28	0.0561	0.0286	0.0357	0.0643	0.1286
30	0.0524	0.0267	0.0333	0.0600	0.1200
32	0.0491	0.0250	0.0312	0.0562	0.1124
34	0.0462	0.0235	0.0294	0.0529	0.1058
36	0.0436	0.0222	0.0278	0.0500	0.1000
38	0.0413	0.0211	0.0263	0.0474	0.0948
40	0.0393	0.0200	0.0250	0.0450	0.0900

. . . AND FROM THE INDUSTRY

Items for this column should be sent to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, P.O. Box 1426, Elk Grove Village, IL 60007 or phone (312) 437-6604. We welcome news announcements concerning new employees, retirement, or other announcements of interest to the gear industry. Announcements must be received by the 25th of the month, two months prior to the date of the next publication. Items received after this date will be held for the following publication.

M & M Precision & Reishauer Marketing Agreement

M & M Precision Systems have recently completed a marketing agreement with **Reishauer Corp.** The Reishauer staff have completed training at the M & M Precision Systems plant in Dayton, Ohio, and they are available immediately to assist the M & M network.

American Pfauter Adds New Distributor

American Pfauter Ltd., manufacturer of gear production and measuring systems, has expanded its distributor organization by naming Machine Tool Systems, Inc., a subsidiary of the Robert E. Morris Co. of Hartford, Connecticut. Under the management of President David Yaged and V.P. of Sales Dick Long, the new distributorship will serve Alabama, Georgia, Florida, Virgina, North and South Carolina from its offices in Charlotte and Greenville.

Gleason Opens Detroit Office

The Detroit Sales Office for **Gleason** is now open at: 370 Franklin Center, 29100 Northwestern Highway, Southfield, MI 48034. Geoffrey Ashcroft, Regional Sales Manager for this office can be reached by calling (313) 353-5205.

Acme-Cleveland Purchases Two British Firms

Spline Gauges Ltd. and its affiliate, **Piccadilly Precision Engineering Ltd.** of Tamworth England have been purchased by **Acme-Cleveland.** These companies make gauges and other products used to control the accuracy in the production of high precision gears. They will continue to be managed by Keneth Foster, who is the current managing director.

Cadillac Machinery Sales Program

Cadillac Machinery, Elk Grove, Illinois announces a new sales program to allow gear manufacturers to convert their under utilized machinery into the capital dollars needed to upgrade to the latest technology that new machinery has to offer. Call Richard Goldstein (312) 437-6600 for details.

New Fellows FS180 Gear Shaper Combines CNC with Hydromechanical Stroking System.

Pitch Diameters to 180 mm (7"); Part Quality to DIN Class 6 (AGMA Class 11)

Fellows new FS180 CNC HYDROSTROKE® Gear Shaper combines computer numerical control with the proven HYDROSTROKE stroking system. It's a completely new design for the demanding, rapid production levels of gears up to 180 mm (7") pitch diameter with precision to DIN Class 6 (AGMA Class 11).

The compact FS180 meets industry requirements for extremely high production of smaller gear sizes such as those in current automotive transmission designs.

Productivity Advantages

The FS180 Gear Shaper promises dramatic productivity gains with these features: CNC control, a new quick-return stroking system, uniform controlled cutting force, high stroking rates up to 1700 SPM, machine function monitoring sensors, and variable infeed rates. Variable infeed rates during infeed reduce total cycle time, particularly on small gears or when multipass cutting.

The machine is also designed to take advantage of coated cutting tools, such as TiN. Production can be increased without sacrificing tool life.

CNC Controls the FS180 Machine Functions

The modular CNC system—designed specifically for gear shaper control—utilizes the latest microcomputer technology, interface hardware, and state-of-the-art modular software, and is "operator-friendly"—making it easy to learn and operate.

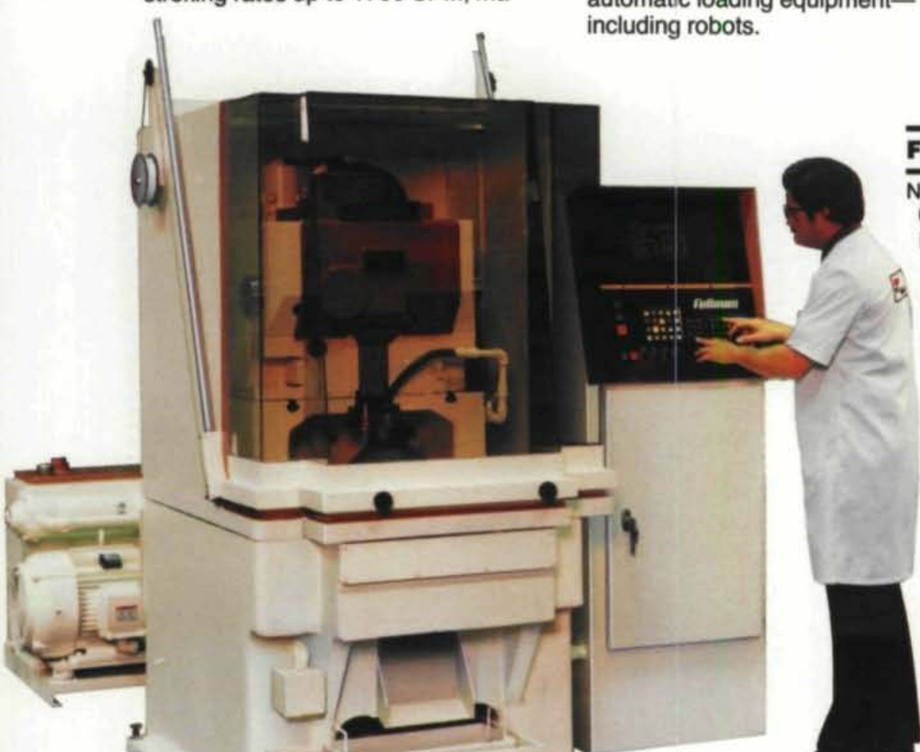
Built-in sensors monitor and control machine functions with feedback to the input/output data monitoring console. Critical sensors and control logic are backed up with a redundant system to ensure uninterrupted operation in the unlikely event of malfunction. The CNC control can be readily interfaced with automatic loading equipment—including robots.

Should a malfunction occur, the CNC multilevel monitoring and diagnostic system shows the operator—

on the CRT screen—where the problem is, and how to remedy it . . . even shuts the machine down if the malfunction warrants.

HYDROSTROKE System at the Heart of the Machine

While CNC optimizes the FS180's performance, Fellows patented HYDROSTROKE system gives it the extra competitive edge: the entire cutting force is concentric to the cutter; stroking speeds to 1700 per minute—infinitely variable; proportional quick-return ratios permit higher stroking speeds without loss of tool life; and variable feed rates produce shorter cycle times.



FS180 Machine Data	Metric	Inch
Nominal Pitch Diameter (Internal or External)	180 mm	7"
Maximum Diametral Pitch	4 MOD	6.3
Maximum Face Width	32 mm	1.250"
Maximum Helix Angle	45°	45°
Maximum Stroking Speed In Actual Production	1700 SPM	1700 SPM

Write for Complete Information

For complete information on the new Fellows FS180 CNC HYDROSTROKE Gear Shaper, write Fellows Corporation, Emhart Machinery Group, Box 851, Springfield, VT 05156-0851 or call (802) 886-8333.

Fellows
MACHINERY GROUP

EMHART

CIRCLE A-6 ON READER REPLY CARD