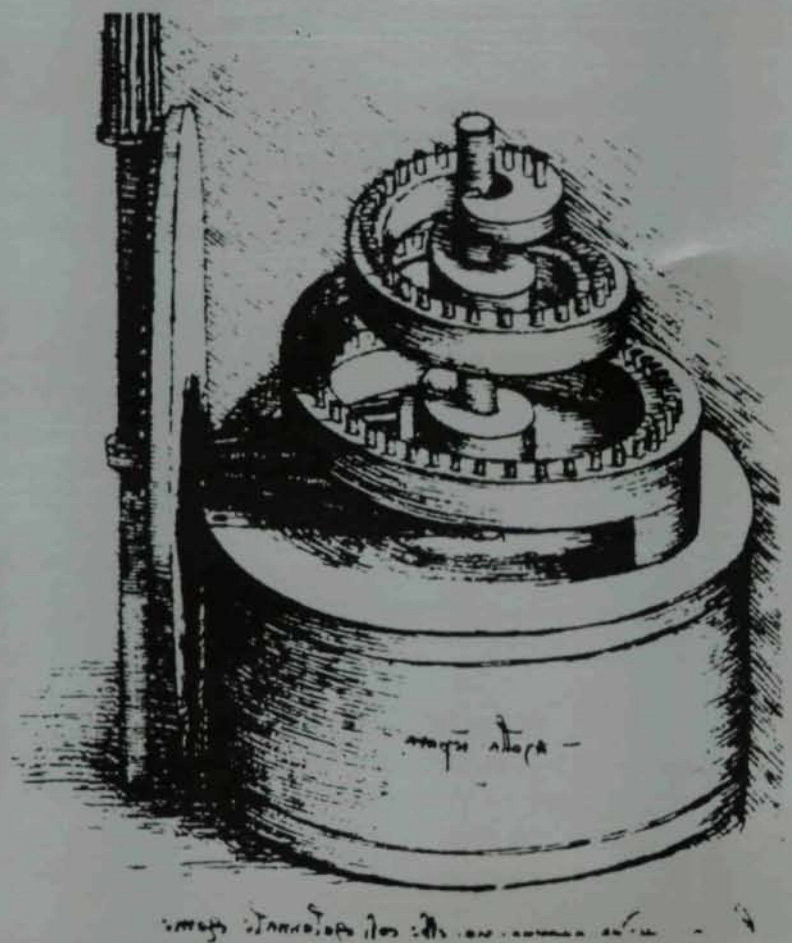
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

JANUARY/FEBRUARY 1985



Gear Tooth Stress as a Function of Tooth Contact

Computer Aided Design of Forging and Extrusion Dies

Effect of Lubricant Traction on Wormgear Efficiency

High Power Transmission with Case-Hardened Gears

Definitions of Gear Elements

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.

Dear Editor:

Your new journal is a most attractive, informative publication. I'm certain that you and the rest of the GEAR TECHNOLOGY publication staff labored long and hard to produce such an excellent, professionally-rendered product.

Your journal is also unique; there is none other like it in print. It appears that you have set out to serve an important, previously neglected area of technology. This action is commendable, and will be a definite service to gear designers, manufacturers, and users.

Please accept my congratulations on this new endeavor.

Ken G. Merkel, Ph.D., Editor
The Journal of Engineering Technology

The first two issues of your very fine journal deal extensively with basics. I have found the ordinary gear-man other than an engineer does not comprehend what is simple to the engineer.

The "Gear Ratio" article (second issue) was particularly interesting; I have made a program for that, but had thought it was impossible for the computer to find the factors. Now I know how to achieve the "impossible". Thank you for that. Best of success with your ambitious endeavor.

Robert P. Ellenberger, Engr.
Federal Gear, Inc.

In our country, the lack of reference sources and results-of-research information make keeping up to date in our job an extremely difficult enterprise. This publication will fill a sorely needed gap.

Alexander H. Danon G.
General Manager
Pedro Meusnier
Juarez, Mexico

Your Oct./Nov. issue article on Austempered Ductile Iron by Dale Breen was reviewed with great interest. We are currently involved in a test program with one of our customers on gears made from this material. The content of the article was quite helpful. Keep up the good work.

B. A. Schaler
Allison Gas Turbine Operations

During a recent business trip to our Rockford, Illinois headquarters to discuss Gear Manufacturing and Design Problems, I had an opportunity to scan your magazine and was very much impressed.

Here in our Denver, Colorado facility, we are the primary manufacturing source for most of the individual parts that are used in the Aerospace products that Sundstrand supplies to customers worldwide. These products include electrical generating units, commonly known as Constand Speed Drives, auxillary power units, wing actuators, rudder speed brakes for Space shuttle, and the list goes on and on. All of these products use gears, and almost all of our gears are precision ground. Suffice it to say, we in Denver, are very much involved in Gears.

Peter F. Palko, Section Mgr.
Sundstrand Aviation Operations

Your magazine GEAR TECHNOLOGY is a timely asset to me. We are just starting to hob our own gears in house. We have little actual gear cutting experience and any information we can get our hands on is a great help.

Keep up the fine articles especially the "Back to Basics" Series.

Alan R. Ayotee
Process Engineer

I would like to compliment you on your Journal. There has been a need for a publication of this kind for a long time. I have passed the number on to my colleague concerned with the CAD of Gears, and I am sure he will greatly appreciate the publication.

Professor S. A. Tobias
University of Birmingham, England

As a Manager of Special Projects leading company development in certain areas including non standard gearing, I have found your Gear Technology Journal Vol. 1 No. 3 to be an excellent reference source.

I would like to receive it regularly.

Also, if possible, I would like to receive Vol. 1, No. 1 and Vol. 1, No. 2.

Valentin G. Barba
Special Projects Manager
Plessey Dynamics Corp.

Editors note: This letter is typical of many that we have been receiving. Because of the extra time and expense involved in sending individual back issues, we must advise that there is a charge for back issues, $7.00 for the United States, $15.00 for Foreign Subscribers. There is also only a limited number of these issues available. If you wish to purchase any back issue, mail your check and indicate the issue you are requesting to:
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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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FEATURES

DEPARTMENTS

COVER
The Advanced Technology of LEONARDO DA VINCI
1452-1519

The machine-minded 20th century looks at Leonardo's work in the area of applied mechanics with great admiration. Although all his designs were not always practical, they show great ingenuity, and their presentation demonstrates Leonardo's marvelous draftsmanship applied to technical subjects.

The cover sketch demonstrates a clock spring equalizer. Leonardo lived in a time when there was a great interest in improving the mechanisms of clocks. This is perhaps because of the great size of public clocks. It is apparent from *Codex Madrid I* that Leonardo was searching for spring equalizing devices that would be superior to the fusee. The fusee had many disadvantages. One was that the gut cord that moved it would stretch and even snap. So Leonardo began designing possible conical equalizing devices geared to the mainspring for a more consistent transmission. The cover drawing is one of four similar attempts. This design appears to be operable with the exception that the teeth on the volute are of a constant pitch. However, they would need to vary with the varying pitch between one end of the pinion and the other.

January/February 1985 — Vol. 2, No. 1

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MANUSCRIPTS: We are requesting technical papers of every sort from manufactured 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. Second class postage applied for at Chicago, IL.

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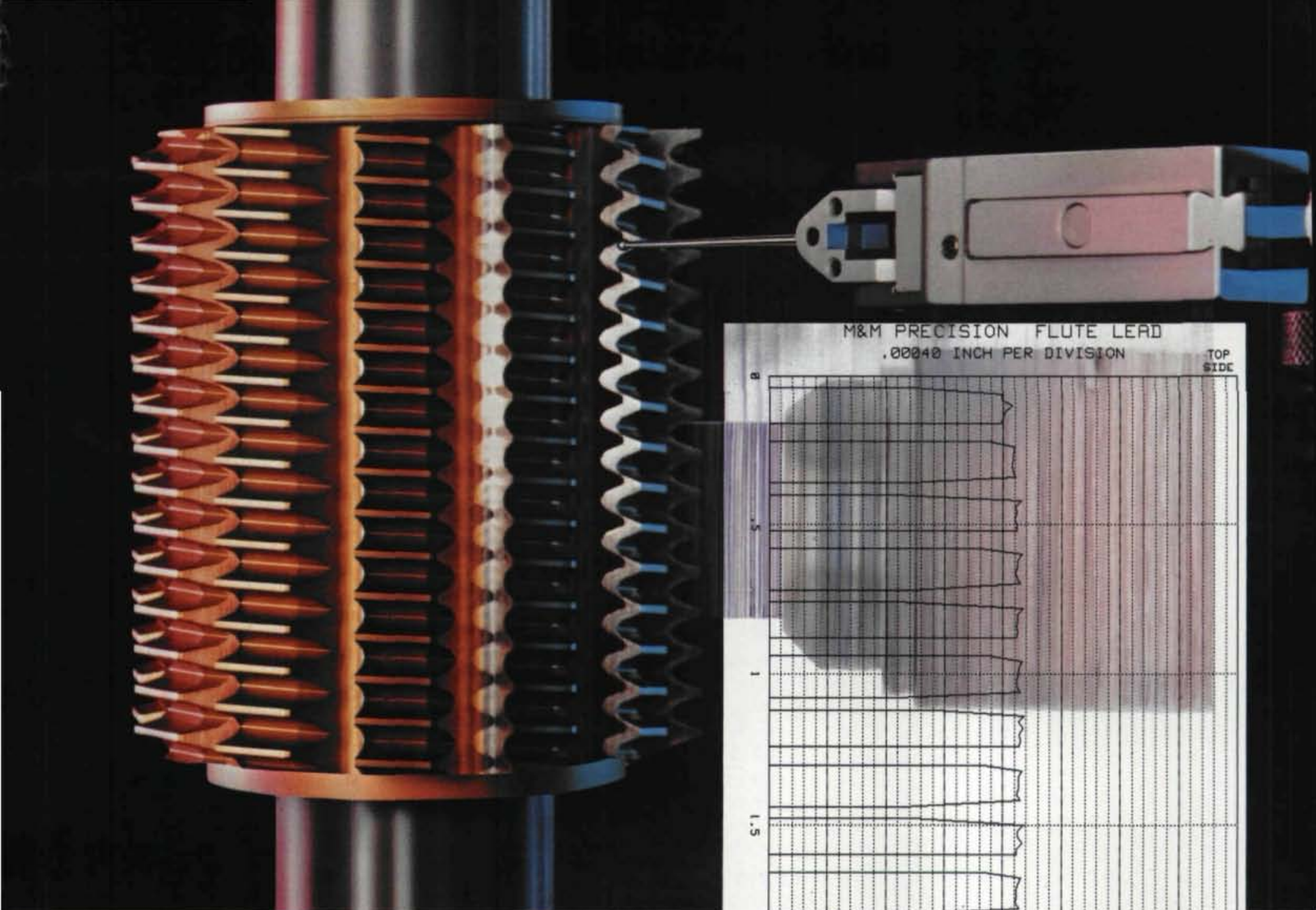

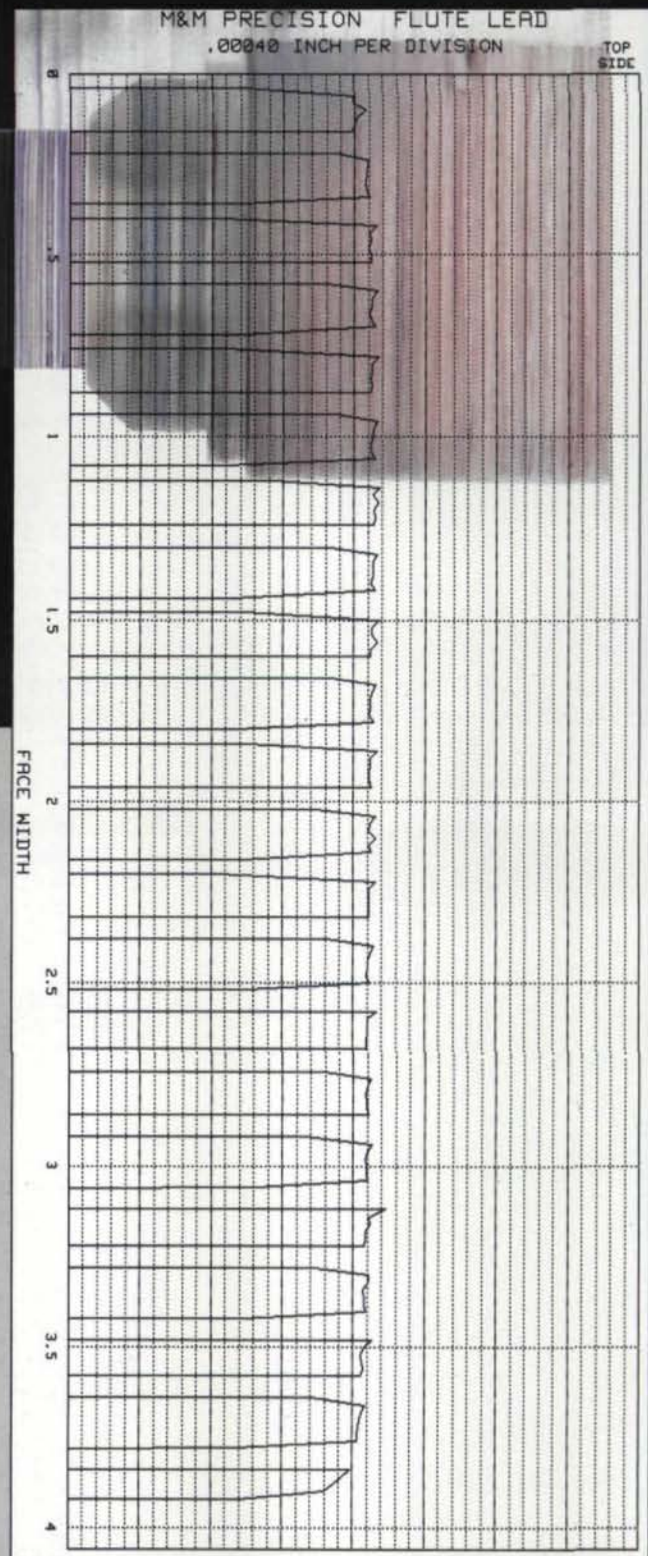

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NOTES FROM THE EDITOR'S DESK

WHAT'S IN IT FOR YOU?

In my travels over the past several months, it has been very gratifying to have so many readers come up to thank me, and to tell me of those articles which were beneficial to them. I recently attended three technical conferences held by ASME, AGMA and SME. It was there that I had the opportunity to meet with readers and get some feedback regarding which articles they liked, which ones helped them the most and what information they would like to see in the future. Since less than 5% of our industry attended those conferences, that did not really reflect what our readership in general would like to see in the future. We need to know what you have found helpful, timely or interesting. We spend an enormous amount of time and effort seeking the information that will best help you design and build a better, lower cost and higher quality gear product. We always welcome suggestions or copies of articles that you may have read or written which you think would be of interest to the gear manufacturing industry. Any time you see a good article or paper that is written from an educational or teaching standpoint, send it to us. We would be interested in reading it.

On several occasions at these conferences, attendees have commented that the articles they have read were something that they had seen before, that had possibly been presented at another conference. Since 95% of our readership have not had access to these seminars, they have not been exposed to some of the best technical writing available in the gear field. As there were approximately 160 gear papers presented at these three technical conferences, it is our hope to be able to present to those unable to attend, the best and most interesting papers from those conferences and other sources.

COMPLIMENTARY SUBSCRIPTIONS

There seems to be some confusion relating to subscriptions to Gear Technology. If you have filled out the complimentary subscription request and it has been approved, you need do nothing more. If we have not received your Free subscription card, your copy of the magazine will have a complimentary card attached that should be signed and sent back to us. We urge you to please fill this form out immediately and return it to our office so as not to interrupt your Gear Technology subscription. If there are additional people within your company that you feel would benefit by receiving Gear Technology, please recommend them by letter or card, and a subscription form will be mailed to them.

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Michael Goldstein
Editor/Publisher

GUEST EDITORIAL

GEAR RESEARCH, THE STATE OF THE ART

Gear research seems to be thriving. Between September 10th and October 17th, 120 papers about gears were presented, at three conferences in Milwaukee, Boston, and Washington, to a total audience of about 400. The authors were from nine countries. Slightly more than half of the papers were prepared by authors who live outside the US and Canada.

Why is it then, that gear designers, manufacturers, and standards writers seem uncertain about the performance of their products? Why do prototypes undergo extensive testing before release for volume manufacture? Why did the total audience at the three conferences, with duplicate attendees and authors eliminated, probably not exceed 200?

Let's look at the papers which attracted this audience. They might be divided into four groups:

- Many papers could have been subtitled "What I Did With My Computer". More than one third of them would fit into this group.
- Other papers might have been named "Here is a Product or Process I'd Like You To Buy". About one fifth of them fit into this group.
- Some covered new ways to do old tricks: calculating the AGMA geometry factors, laying out tooth forms and constructing ratio tables.

- *The rest of the papers, which were the most valuable to me, and I think most engineers, covered a variety of subjects, including interesting historical mechanisms, new methods of analysis with test results for verification, investigation of the fundamental mechanisms of failure and real test results on real gears.*

Few, if any, of the research projects reported on the subjects which are of the greatest concern in gear design:

- Transverse load distribution, particularly for gears with *large initial misalignments or large elastic deflection* under load.
- Internal dynamic factor, the effect of transmission error on tooth load. We sorely need an analysis method which gives good results for most gears using the normal quality measurements as input, without the use of a mainframe computer for the analysis.
- External dynamic factor or application factor. We used to call it service factor. It relates the performance of gears under steady state laboratory conditions to their performance under the fluctuating loads they usually see in the field. What is it about some loads that makes them so hard on drives?
- What material quality factors make the most difference in the allowable stress numbers for gears? What are reasonable limits on material and process for each level of allowable stress?

These little understood areas of gear design account for most of the failures in industrial gear drives. Our present

(Continued on page 47)

MR. DON McVITTIE, *author of the Guest Editorial, joined The Gear Works—Seattle, Inc. as Executive Vice President in 1969. The Gear Works specializes in the unusual, particularly small lot production of coarse pitch, ground gears, planetary drives and custom designed reducers and increasers. He is also the President of Gear Engineers, Inc. He has been an active participant in the American Gear Manufacturers Association since 1972. His main interest has been the Technical Division, where he has served on many committees, including the Gear Rating Committee, the Manufacturing Committee and the Metric Resource and Advisory Committee. He is chairman of section 1 of the Technical Division, comprising nomenclature, metrication, lubrication, and metallurgy, and is a member of the Technical Division Executive Committee. He is the current President of AGMA. Mr. McVittie is a Licensed Professional Engineer in Washington, and a member of ASME, SNAME, and SAE.*

Analyzing Gear Tooth Stress as a Function of Tooth Contact Pattern Shape and Position

by
Lowell Wilcox
The Gleason Works
Rochester, New York

Abstract

The development of a new gear strength computer program based upon the finite element method, provides a better way to calculate stresses in bevel and hypoid gear teeth. The program incorporates tooth surface geometry and axle deflection data to establish a direct relationship between fillet bending stress, subsurface shear stress, and applied gear torque. Using existing software links to other gear analysis programs allows the gear engineer to evaluate the strength performance of existing and new gear designs as a function of tooth contact pattern shape, position and axle deflection characteristics. This approach provides a better understanding of how gears react under load to subtle changes in the appearance of the no load tooth contact pattern.

Introduction

The fatigue life of bevel and hypoid gear designs has long been known to be a function of the length, width and position of the "no load" tooth contact pattern. For example, careful positioning of the tooth contact pattern relative to the gear member can produce dramatic increases in bending fatigue life. Fig. 1 shows a two-fold increase in bending fatigue life obtained by positioning the tooth contact pattern toward the toe of the gear tooth rather than at a central position.

Based on the results shown in Fig. 1, a gear designer might be tempted to conclude that a central toe position of tooth contact pattern should always be used to obtain maximum fatigue life. However, the relationship between gear fatigue life and the position of tooth contact pattern is complicated by additional considerations that must be made. For example, what position of tooth contact pattern produces the best sound qualities? What combination of the adjustability of the gear design and the stiffness of the axle housing permit the optimum position of tooth contact pattern to be obtained, and what will the mode of fatigue failure be? Clearly, it would be desirable if the relationship between fatigue life and tooth contact pattern parameters could be determined by analytical methods rather than relying on experimental data only.

In September of 1981 the author(1)[1] presented a paper outlining a new method of gear stress analysis based on the finite element method used in conjunction with the method of Tooth Contact Analysis TCA. This method of analysis incorporates adjustability and axle deflection data along with finite element modeling of the gear and pinion members to calculate root fillet and surface stresses as the gear and pinion deflect under load. In the sections of this paper that follow, the new method of stress analysis will be used to analyze the relationship between length, width and position of tooth contact pattern and the stresses that result in fatigue failure. The examples given will demonstrate that stress levels in gear teeth can be related quantitatively to the parameters describing tooth contact, thereby, improving the gear designer's ability to predict the fatigue performance of bevel and hypoid gears.

Stress Analysis Model

The stress model used to analyze fillet and surface stresses in gear teeth is based on the combination of three well known

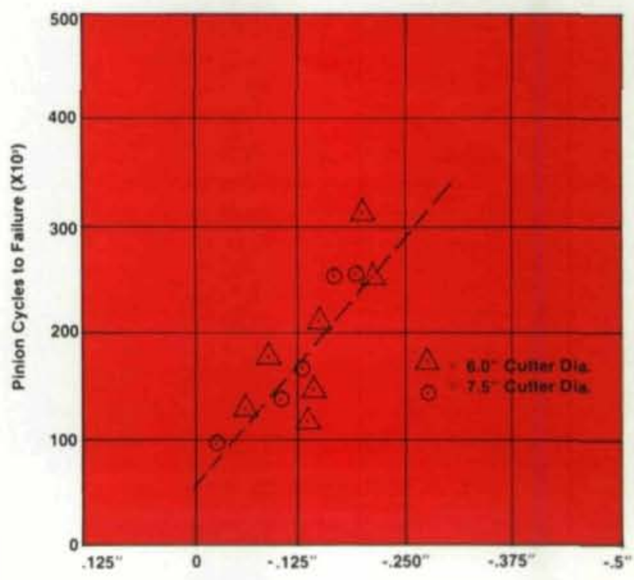

Fig. 1—Bending Fatigue Live Vs. Position of No Load Center of Contact (Σ) Relative to Gear Member

AUTHOR:

LOWELL WILCOX *is Senior Staff Research Engineer in charge of the Controls Research and Computer Aided Design Group at The Gleason Works. Dr. Wilcox earned his BSME, MSME and Ph.D degrees from the University of Rochester. Since 1970 he has been active in the development of finite element based software for use in analyzing stresses in bevel gear teeth. Presently the author is conducting research that will extend the capabilities of the finite element based software to include a wider range of stress induced gear failures than can be handled by present theories.*

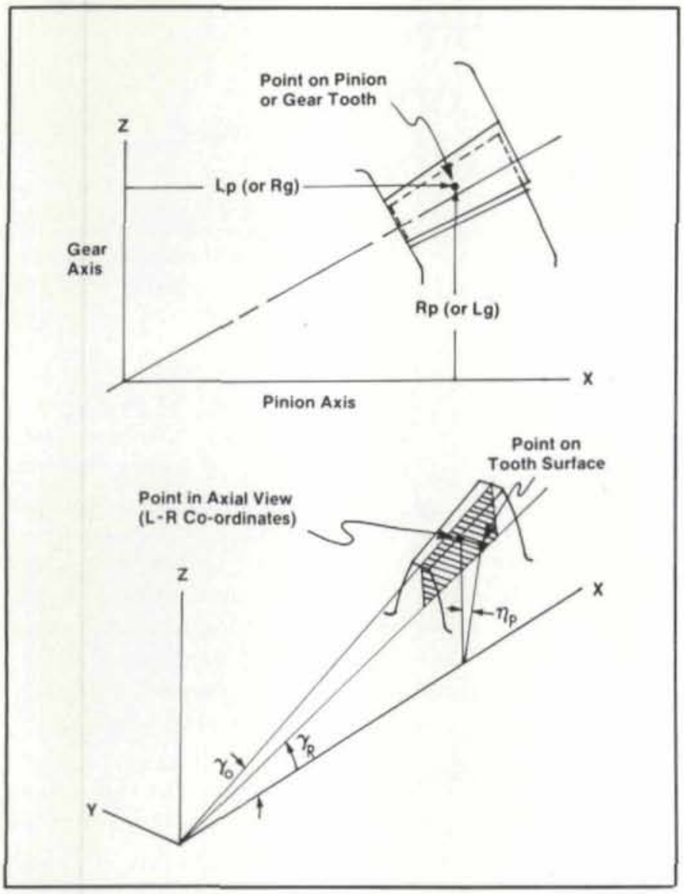

Fig. 2 – Method of Generating Points on Tooth Surface based on Co-ordinates in Axial Section

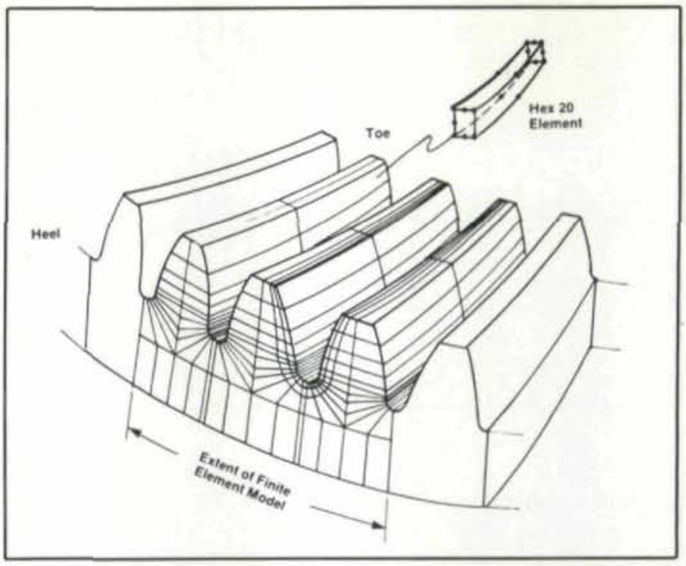

Fig. 3 – Five tooth segment of gear showing relative location of HEX 20 Finite Element Model

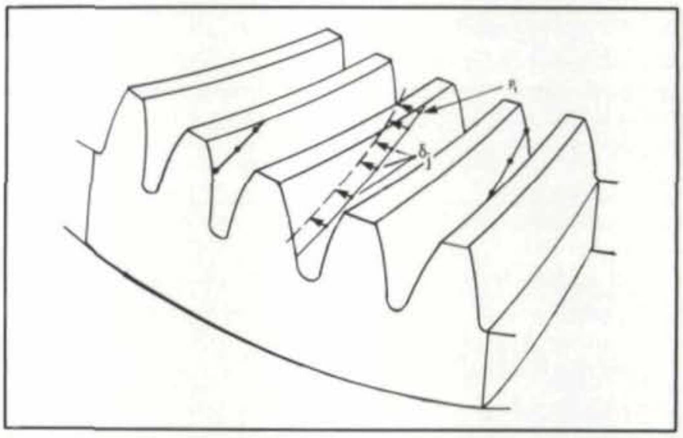

Fig. 4 – Location of instant lines of contact on gear tooth model

analytical approaches; TCA, the finite element method and the flexibility matrix method. A detailed discussion of how these separate approaches are blended to form a stress model suitable for use in gearing is found in references (1) and (2). What follows is a brief overview of the essential features of the stress model.

TCA is used to define the geometry of the gear and pinion tooth forms and to define the lines of contact that exist between gear and pinion teeth as they rotate through mesh. Fig. 2 shows, for the pinion member, how TCA is used to define the co-ordinate η_p based on given values of the axial co-ordinates L_p and R_p. The axial plane co-ordiantes L_p and R_p can readily be determined from ordinary algebraic equations using blank dimensions as input. Once L_p and R_p are specified at a point on the gear surface, TCA is used to calculate (by computer interation techniques) the third dimension η_p. In other words, TCA is used as a "black box" to develop a point by point description of the tooth surface. The resulting point by point description of the tooth surface is easily converted into a three dimensional finite element model. Fig. 3 shows a typical finite element model of the gear member.

1. Numbers in parentheses refer to similarly numbered references in bibliography at end of paper.
2. TCA is an accepted method of vector and matrix operations used to describe the Form and Kinematics of gear teeth.

Fig. 4 shows three lines of contact at a particular point of gear and pinion rotation as defined by TCA. The lines of contact represent the theorectical location of the possible contact points that can occur as load is applied to the teeth. Each of the three lines shown in Fig. 4 is discretized into a series of modes that can be analyzed using the flexibility matrix method.

The fundamental principal of the flexibility matrix approach (3) is that the tooth stiffness, interface load and tooth surface deflection can be related by an equation of the form

$$C_{ij}P_j = \delta_i \tag{1}$$

C_{ij} is the combined gear/pinion/axle compliance matrix and represents the load-deflection characteristics of each node along the lines of contact. By specifying the nodal deflection δ_i in a manner consistent with normal gearing constraints the interface load distribution P_j, Fig. 5, can be directly obtained from equation 1. Once the interface loads P_j, are determined, the finite element method can be used to relate fillet and surface stresses to the interface loads P_j.

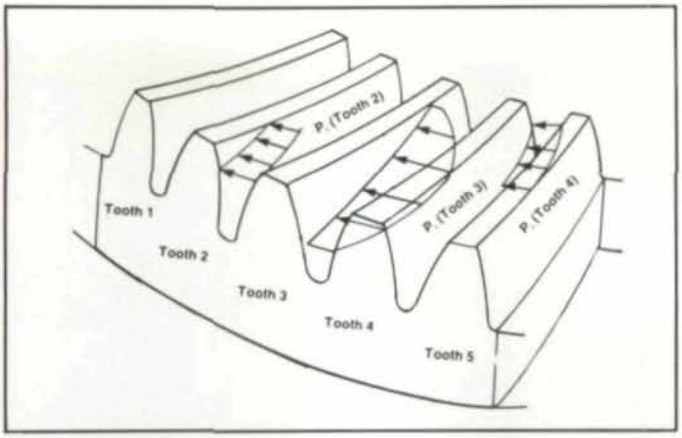

Fig. 5 – Calculated load pressure distribution for all contact lines at given position of rotation

The process discussed above provides a complete description of the gear and pinion stresses at a particular point of gear and pinion rotation as shown by the instant lines in Fig. 4. *In a straight forward manner, any point of rotation can* be specified thus giving a complete picture of the variation in stress as gear and pinion rotate through mesh. Furthermore, the maximum stresses, both bending and surface, are calculated directly as a function of the TCA input parameters, and more importantly, as functions of the length, width and position of the tooth contact pattern relative to the gear member. Additionally, as axle housing deflections enter into the compliance matrix C_{ij} (Equation 1), the stresses also reflect the influence of the axle housing compliance.

In summary, it should be mentioned that all of the above steps required to calculate gear stresses have been integrated into a series of computer programs that are essentially driven by the TCA program. In other words, the logic and decisions necessary to perform the above steps are automatically carried out by the computer and are, therefore, transparent to the gear designer. It is possible, therefore, to generate a complete stress picture corresponding to any developed TCA.

Selection of Gear Design

Table I lists the parameters defining the gear designs analyzed by the finite element stress program. Designs A, B and C are typical of a low ratio, double reduction truck axle. The nominal pressure angle and spiral angle for each design are 22.5° and 35° respectively and the diametral pitch equals 3.161.

Designs B and C are variations of Design A in that the cutter radius has been decreased from 4.5 inches to 3.75 and 3.0 inches respectively. Designs A, B and C represent three different designs from the point of view of their adjustability. Other parameters such as point widths, addendums, dedendums, etc., were calculated according to the Gleason spiral bevel dimension sheet program.

The axle deflections were measured in a Gleason T6R-II Tester. Fig. 6 shows the magnitude and direction of ΔE, ΔP, ΔG and £ corresponding to a pinion input of 36,500 lb-in. It should be noted that the direction of the ΔG component of deflection is opposite to that normally shown in both Gleason and AGMA literature. This is because the finite element gear strength program holds the gear member fixed and *applies all housing related deflections to the pinion member.* Therefore, ΔG becomes the pinion motion relative to the gear member, or in keeping with the original definition of ΔG, becomes minus the gear deflection.

Definition of Tooth Contact Pattern

In order to systematically study the influence of tooth contact pattern parameters on gear and pinion fatigue stress, three different combinations of length, width and position were analyzed. The first combination isolated the influence of tooth contact pattern position, Σ, by holding the no load contact pattern length equal to 0.5F, where F is the gear tooth facewidth and holding width equal to $0.5h_t$, where h_t is the gear tooth whole depth. Three separate positions of tooth contact pattern (Σ) were used in the analysis.

Fig. 7 illustrates these positions relative to the gear number. A non-dimensional coordiante system was selected such that the position of the center of the contact pattern, Σ, is zero

TABLE I
GEAR DESIGN PARAMETERS USED IN STRESS ANALYSIS OF SPIRAL BEVEL TRUCK AXLE

	DESIGN A		DESIGN B		DESIGN C	
	Pinion	Gear	Pinion	Gear	Pinion	Gear
No. Teeth	17	29	17	29	17	29
Diametral Pitch	——	3.161	——	3.161	——	3.161
Face Width	1.5"	1.5"	1.5"	1.5"	1.5"	1.5"
Pressure Angle	22.5°	——	22.5°	——	22.5°	——
Addendum	0.347"	0.188"	0.331"	0.178"	0.307"	0.164"
Dedendum	0.246"	0.404"	0.236"	0.388"	0.221"	0.365"
Cutter Radius	4.496"	4.500"	3.743"	3.750"	2.999"	3.000"
Cutter Edge Radius	0.055"	0.095"	0.055"	0.095"	0.055"	0.095"
Pitch Angle	30.379°	59.621°	30.379°	59.621°	30.379°	59.621°
Mean Spiral Angle	35.0°	35.0°	35.0°	35.0°	35.0°	35.0°

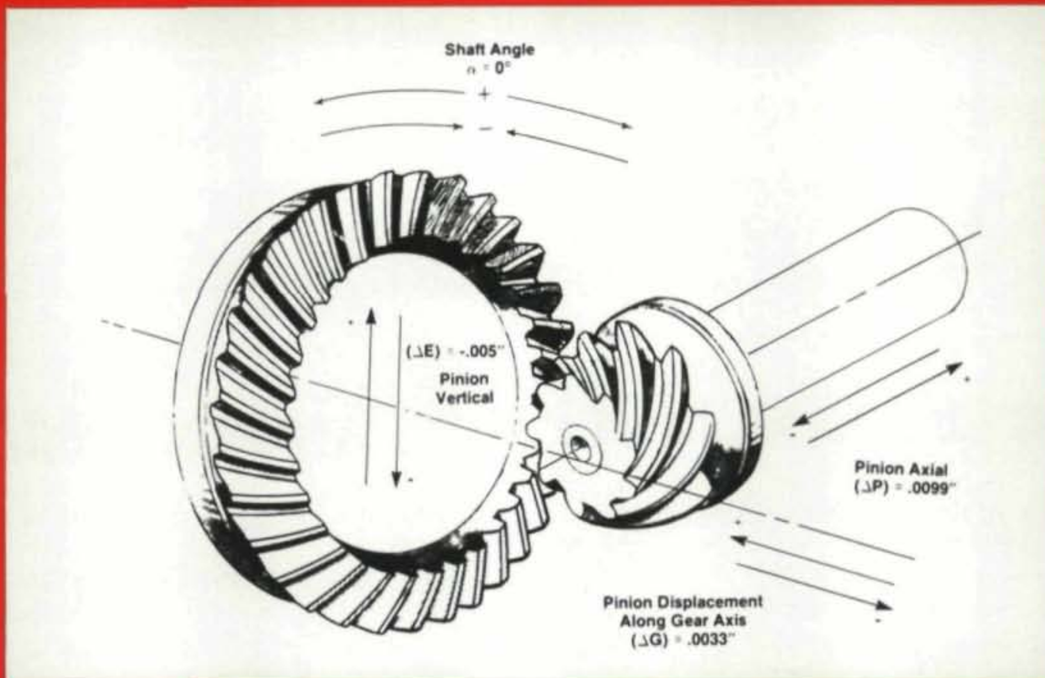

Fig. 6 – Displacement of pinion member relative to gear member at 36,500 lb.-in. gear torque

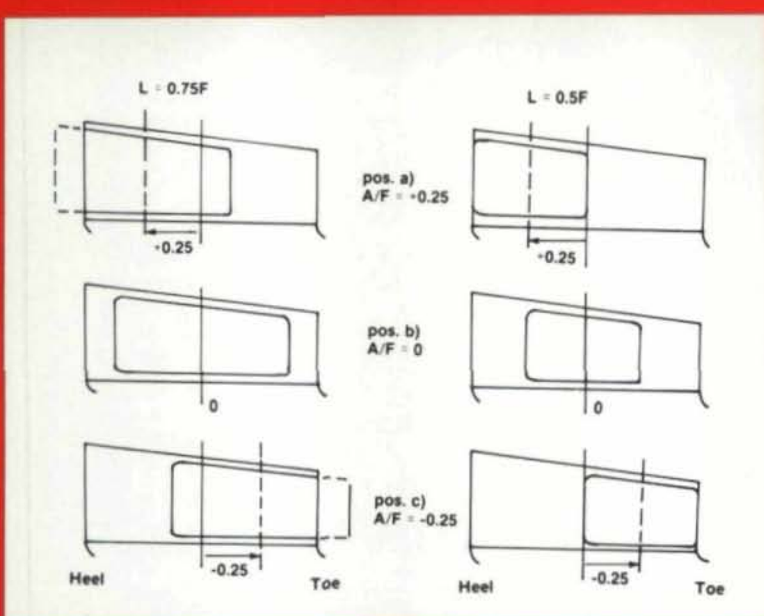

Fig. 7 – 0.5F and 0.75F contact patterns at each of the assumed positions

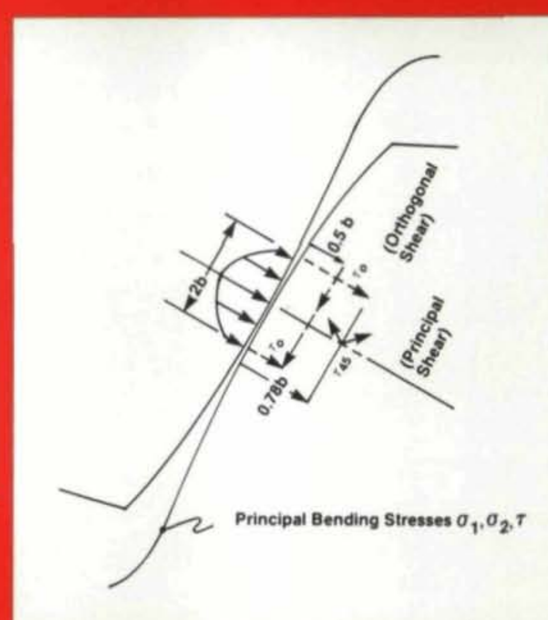

Fig. 8 – Components and locations of pinion stress analysis (2b = theoretical critical fatigue stresses for gear and width of loaded teeth contact line)

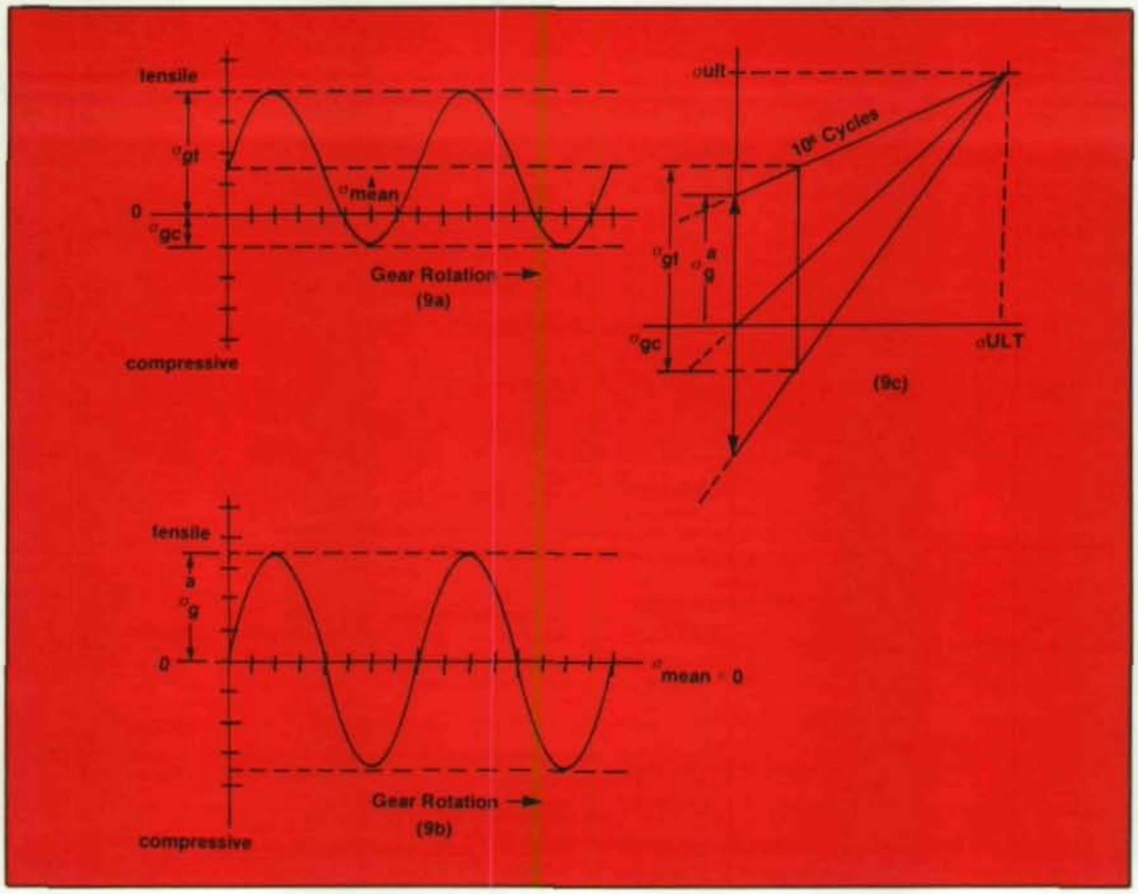

Fig. 9 – Equivalent alternating bending fatigue stress derived from tensile and compressive stress components using a modified Goodman diagram

at the center of the gear face width, -0.25 halfway toward the toe end and 0.25 halfway toward the heel end. The tooth contact patterns were positioned at three different locations along the face of the gear, corresponding to Σ = .025, 0, and -0.250. *The extreme values of Σ = .025, -0.25 corres*pond to contacts that just touch the ends of the gear tooth (in the case of 0.5F length) to slightly overlapping the ends of the tooth (in the case of 0.75F length).

The second combination of tooth contact pattern parameters varied Σ in the wame way as shown in Fig. 7, holding the width equal to 0.5 h_t, while length was set equal to 0.5F and 0.75F.

The third combination of tooth contact pattern parameters again repeated the Σ variation of Fig. 7, but held length equal to 0.5F, while width was set equal to $0.5h_t$ or $0.75h_t$.

Presentation and Discussion of Stress Analysis Results

Generally, three distinct types of stress pattern occur in gear teeth in response to laods applied to the gear teeth; bending stresses in the fillet regions, contact stresses on the surface of the tooth profiles (including friction) and subsurface shear stresses in the case region. The results of the stress analysis in this paper pertain only to the bending stresses in the fillet and subsurface shear stresses in the case region of the teeth.

Fig. 8 illustrates the locations and components of the bending and subsurface shear stresses. The bending stresses in the fillet region are principal stresses that lie on the surface of the fillets. As the gear and pinion rotate through mesh, the principal stresses range from maximum tensile values at a particular point to minimum compressive values. In order to *simplify the handling of both tensile and compressive stresses,* a modified Goodman diagram is employed to reduce the tensile and compressive stresses to an equivalent alternating stress whose mean value is equal to zero. Fig. 9a shows a typical tensile/compressive stress pattern, for either gear or pinion, calculated from the finite element gear strength program. The maximum tensile stress as the gear and pinion rotate through mesh is σ_{gt}, while the minimum absolute value of compressive stress is σ_{gc}. The mean stress is $\sigma_{mean} = (\sigma_{gt} - \sigma_{gc})/2$. Fig. 9b shows the alternating stress pattern equivalent to that shown in Fig. 9a, but with zero mean stress. The stress pattern shown in Fig. 9b can be determined from the modified Goodman diagram shown in Fig. 9c or alternatively from the following formula.

$$\sigma^a_g = \frac{\sigma_{ult}(\sigma_{gt} + \sigma_{gc})}{2\sigma_{ult} - (\sigma_{gt} - \sigma_{gc})} \qquad (2)$$

Equation 2 is derived from Fig. 9c with σ_{ult}, the ultimate stress of the case hardened gear structure, equal to 330,000 PSI. By resorting to an equivalent alternating stress a single stress life curve can be used to compare different gear designs.

In the case of subsurface shear (4, 5, 6) there are actually two such components as is shown in Fig. 8. The deepest penetrating shear is the τ_{45} shear. The τ_{45} shear results in fatigue cracks that usually occur at the case-core interface such as in heavy spalling. The other component of subsurface shear is the reversing orthogonal component of shear which occurs at 64% of the depth of the τ_{45} shear. The orthogonal component of shear, τ_o, is reversing in that as the contact zone moves across the tooth profile a point beneath the surface of the tooth, sees both a plus and minus value of τ_o. The τ_{45} shear, on the other hand, does not change sign as it lies along the centerline of tooth contact. It will be assumed in the ensuing discussions that although the magnitude of the τ_o component is less than the τ_{45} component the τ_o component is in fact the critical component for most gear failures because the fully reversed stress amplitude of the τ_o component results in a worse state of stress than does the single direction τ_{45} amplitude.

Influence of Position of Tooth Contact Pattern and Cutter Radius on Fatigue Stress

Fig. 10 shows the influences of tooth contact pattern position and cutter radius on bending stress and subsurface shear stress. Gear bending stress, σ_g^a, and pinion bending stress, σ_g^a, are shown on the left hand side of Fig. 10, while orthogonal shear stress, τ_o, is shown on the right hand side. The abscissa (Σ) of Fig. 10 represents the "no load" position of the center of contact along the gear face while the ordinate represents equivalent alternating stress measured in KPSI. Non-dimensional contact pattern lengths and widths are each set equal to 0.5 for the stress curves shown in Fig. 10.

The influences of tooth contact pattern position and cutter radius on fatigue stress can be generalized as follows. As the center of the tooth contact pattern is positioned toward the toe of the gear tooth, both gear bending and subsurface shear stress increase. At the same time, pinion bending stress decreases. As cutter radius is decreased from 9.0 inches to 6.0 inches, gear bending and subsurface shear increase while pinion bending stress decrease.

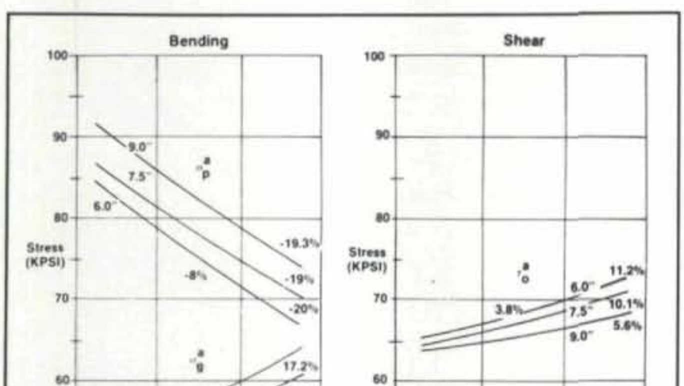

Fig. 10 – Variation of σ_g^a, σ_p^a and τ_o^a with tooth contact pattern position Σ (L = 0.5F, W = 0.5 h_t)

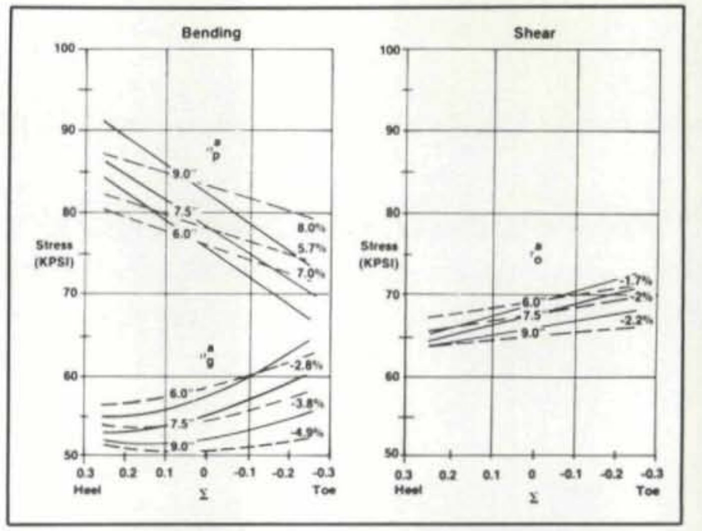

Fig. 11 – Variation of σ_g^a, σ_p^a and τ_o^a with tooth contact pattern position
Σ (———L = 0.5F)
(-----------L = 0.75F)

The percentage figures shown with each of the curves in Fig. 10 indicate the magnitudes of change found in the stress components for each case. The gear bending stress increases from between 6.9 and 17.2 percent as the contact pattern is positioned towards the toe. Subsurface shear increases from between 5.6 and 11.2 percent while the pinion bending stress decreases from between −19.0 and −20 percent.

The corresponding changes due to the influence of cutter radius are smaller. At the center position of tooth contact position ($\Sigma = 0$), the gear stress increase is 10.9% as cutter radius decreases from 9.0 inches to 6.0 inches. The subsurface shear stress increases very slightly, by 3.8% while pinion stress decreases by −8.0 percent.

Summarizing, both tooth contact pattern position and cutter radius have significant influences on fatigue stress. Based on the author's experience with fatigue tests conducted on the designs shown in Table I, the results above tend to support the observation that building the contact pattern center toe and using smaller cutter radii (better adjustability) does result in improved fatigue life.

Influence of Tooth Contact Pattern Length on Fatigue Stress

Fig. 11 shows the influence of the length of the contact pattern on fatigue stress as the contact pattern length is increased from 0.5F to 0.75F. The solid lines correspond to a 0.5F length while the dashed lines correpsond to a 0.75F length.

The primary influence of increasing contact pattern length is to rotate the stress position curves about the $\Sigma = 0$ value. The basic trends of stress versus Σ and cutter radius remain the same as outlined by the rotation of the stress curves.

When the detrimental effects of lengthening the tooth contact pattern are examined, it is apparent that the pinion bending stresses in Fig. 11 are increased from between 5.7 and 8.0 percent when the tooth contact pattern is positioned toward the toe of the gear tooth. The gear stress on the other hand decreases from between −2.8 and −4.9 percent. At

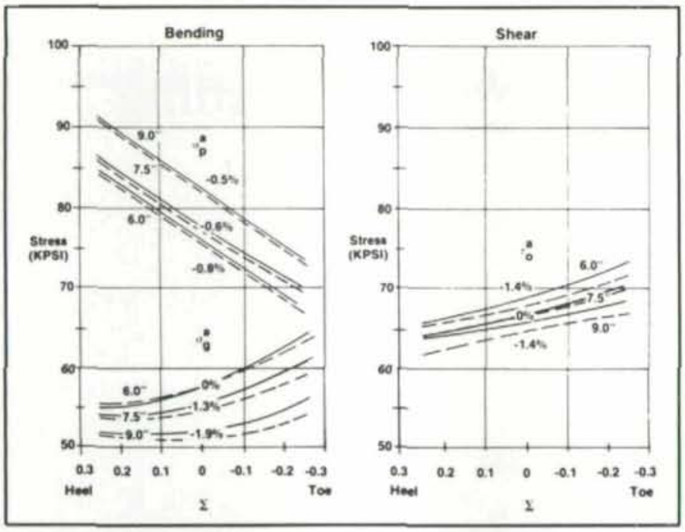

Fig. 12 – Variation of $\sigma^a_{g'}$, σ^a_p and τ^a_o with tooth contact pattern position
Σ (———— L = 0.5F h_t)
(------------ L = 0.75F h_t)

the same time, subsurface shear stresses decrease from between −1.7 and −2.2 percent. If the position of tooth contact pattern is moved in the opposite direction, toward the heel of the gear tooth, the trends given above are reversed by about the same magnitudes.

Summarizing, when Fig. 10 and 11 are compared, it is seen that the importance of contact pattern length changes on fatigue strength are approximately one half those experienced with position change and approximately the same as those attributed to cutter radius changes.

Influence of Contact Pattern Width on Fatigue Stress

Fig. 12 shows the influence of contact pattern width on fatigue stress when the contact pattern width is changed from $0.5h_t$ to $0.75h_t$, where h_t is the gear tooth mean whole depth. The dashed lines represent the $0.75h_t$ width while the solid lines represent the $0.5h_t$ width.

It is apparent from Fig. 12 that the change in contact pattern width from $0.5h_t$ to $0.75h_t$ has almost no effect at all on bending stress and only a very small effect on shear stress. The pinion bending stress decreases by less than one percent at the $\Sigma = 0$ position of tooth contact pattern. Gear bending stress decreases by −1.9 percent or less and shear stress decreases by −1.4 percent or less.

Summarizing, the width of the contact pattern has the least influence of all on bending and shear stresses. Changes in fatigue life due to changes in bending and shear stress of less than two percent would be difficult to detect experimentally.

Comments on Experimental Validity of Stress Data

In the preceding sections of this paper, analytical stress data were presented relating changes in fatigue stress to changes in the tooth contact pattern parameter length, width, Σ and to cutter radius. At present there is very little experimental data collected relative to two of these parameters; Σ and cutter radius.[7] Furthermore, a portion of the data concerning R_c was obtained from simulated gear tooth shapes rather than from actual bevel gears[8]. The influence of the parameters length and width have not been experimentally verified.

Although the results presented in Section 5 are encouraging, it must be stated that these results are largely theoretical predictions at this point in time. Hopefully, more experimental data will be available in the future to improve confidence in the results of Section 5.

Conclusions and Future Plans

The main objective of this paper has been to show that fatigue stress can be quantitatively related to tooth contact pattern parameters and to cutter radius. To that end, results have been presented showing the percentage changes in bending stress and subsurface shear stress as contact pattern parameters and cutter radii were changed in a systematic way.

The results presented in Figs. 10 thru 12 show that fatigue stress does change in a predictable manner as tooth contact patter parameters are varied for each of the three tooth designs employing different cutter radii. Generally, the changes in fatigue stress vary from 10 to 20 percent in the case of position and cutter radius variations down to only a few percent in the case of tooth contact pattern width.

(Continued on page 23)

Computer Aided Design (CAD) of Forging and Extrusion Dies for the Production of Gears by Forming

by
David J. Kuhlmann
P. S. Raghupathi
(Battelle's Columbus Division)

Gary L. Horvat
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Donald Ostberg
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Material losses and long production times are two areas of conventional spur and helical gear manufacturing in which improvements can be made. Metalforming processes have been considered for manufacturing spur and helical gears, but these are costly due to the development times necessary for each new part design. Through a project funded by the U.S. Army Tank – Automotive Command, Battelle's Columbus Division has developed a technique for designing spur and helical gear forging and extrusion dies using computer aided techniques.

Gear Forming Methodology

Gear manufacturing processes are highly specialized due to the complex geometry and high accuracy requirements of the gear teeth. Precision forming methods for gears offer considerable advantages including the reduction of material and energy losses during finish machining. However, to establish precision forming as an economical production technique requires the capability to design and manufacture dies with precise and reproducible dimensions with long life and at an acceptable cost.

The traditional method of forging and extrusion die design and manufacture is based on experience and trial and error. A preliminary die is made and a few parts are formed. Measurements are taken of the finished part and the die is adjusted accordingly. A second series of trials is conducted, and so on, until the final die geometry is obtained. Such a development program is required for every new design which makes the precision forming process economically less attractive, especially when complex and precise geometries are involved, as with spur and helical gears. Therefore, methods need to be developed to apply advanced computer aided design and manufacturing (CAD / CAM) technologies (finite element, metal forming and heat transfer analyses) to gear forming die design and manufacture. This approach benefits from the capabilities of the computer in computation time and information storage and allows the die designer to try various changes in the die design and the forming conditions, without trying out each new change on the shop floor.

CAD / CAM Applied to Forging and Extrusion

In recent years, CAD / CAM techniques have been applied

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to various forging processes. The experience gained in all of these applications implies a certain overall methodology for CAD/CAM of dies for precision and/or near-net shape forming. This computerized approach is also applicable to precision cold and hot forming of spur and helical gears, as seen in Fig. 1. The procedure uses as input: (a) the process variables and (b) the part (gear) geometry. The former consist of:

(1) data on billet material under forming conditions (billet and die temperatures, rate and amount of deformation),
(2) the friction coefficient to quantify the friction shear stress at the material and die interface, and
(c) forming conditions, such as temperatures, deformation rates, suggested number of forming operations.

Using the process variables and the part geometry, a preliminary design of the finish forming die can be made. Next, stresses necessary to finish form the part and temperatures in the material and the dies are calculated. The elastic die deflections due to temperatures and stresses can be estimated and used to predict the small corrections necessary on the finish die geometry. Knowledge of the forming stresses also allows the prediction of forming load and energy. The estimation of die geometry corrections is necessary for obtaining close tolerance formed parts and for machining the finish dies to exact dimensions. The corrected finish die geometry is used to estimate the necessary volume, and the volume distribution in the billet or the preform. Ideally, a simulation of the metal flow should be conducted for each die design. This is a computerized prediction of metal flow at each instant during forming. This simulation is mathematically quite complex and can only be performed at this time for relatively simple parts. In more complex applications, die design can be determined by computerized use of experience-based formulas.

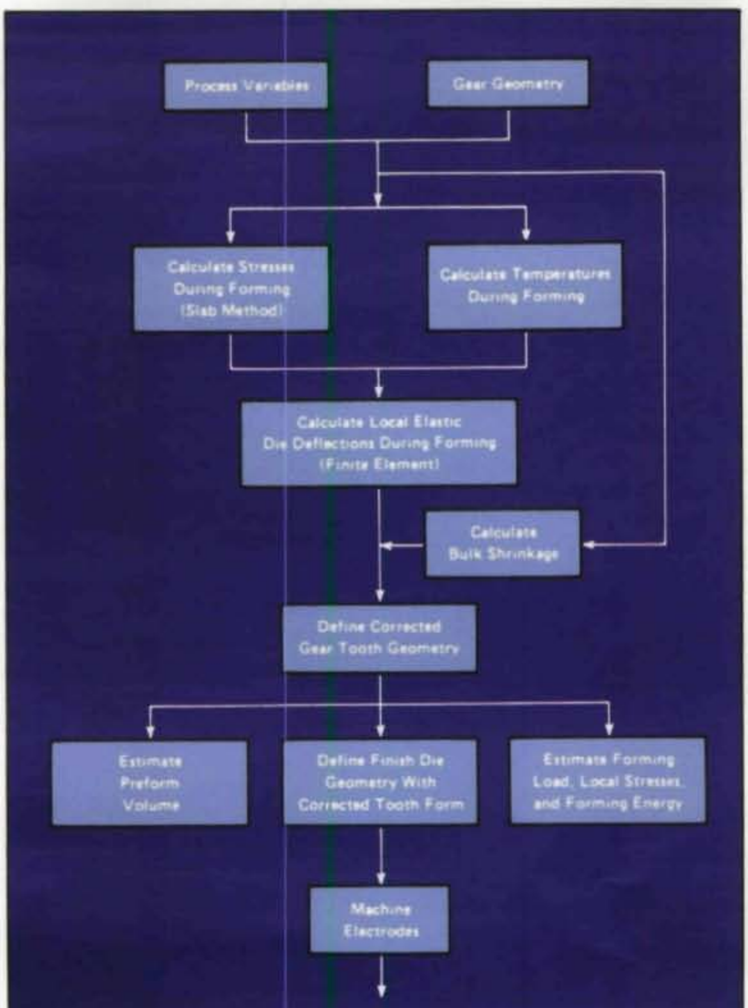

Fig. 1 – Descriptive Computer Aided Design Procedure for Finish Forging Dies

Two Phase Approach

The present study is still in progress and is being conducted in two phases as follows:

- Phase I
  Computer Aided Design (CAD) of forming dies.
- Phase II
  Computer Aided Manufacturing (CAM) of the forming dies and demonstration of the effectiveness of CAD/CAM by forming (forging and/or extrusion) a set of spur gears and a set of helical gears.

The Phase I work and the Phase II spur gear extrusion trials have been completed. A simplified flow diagram for the computer aided design and manufacturing of forging and extrusion dies for spur and helical gears is shown in Fig. 2. Using the overall outlines of Figs. 1 and 2, the die design effort was dividied into four tasks:

1. Definition of gear and gear tooth geometries.
2. Prediction of forming load, pressure and stresses.
3. Estimation of tool deflections, shrinkage and corrections.
4. Development of an interactive, graphics based computer program for performing Tasks 1 through 3.

Generating the Gear Tooth Geometry

To define the tooth geometry, certain gear and/or cutting tool parameters must be specified. Some additional data pertaining to the mating gear may also be required in certain instances. All the data required for the computations can be obtained from a "summary sheet" developed by gear designers (Fig. 3) and also the geometry of the cutting tool (Fig. 4). With this data, standard gear equations are used to calculate the X and Y coordinates of the points describing the gear tooth profile.

The basic geometry of a spur gear tooth is seen in Fig. 5, with the following major definitions (1):

- addendum –
  the radial distance between the top land and the pitch circle
- backlash –
  the amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the pitch circles
- circular pitch –
  the distance, measured on the pitch circle, from a point on one tooth to a corresponding point on an adjacent tooth
- clearance –
  the amount by which the dedendum of a gear exceeds the addendum of its mating gear
- dedendum –
  the radial distance from the bottom land to the pitch circle
- diametral pitch –
  number of teeth on the gear per inch of pitch diameter

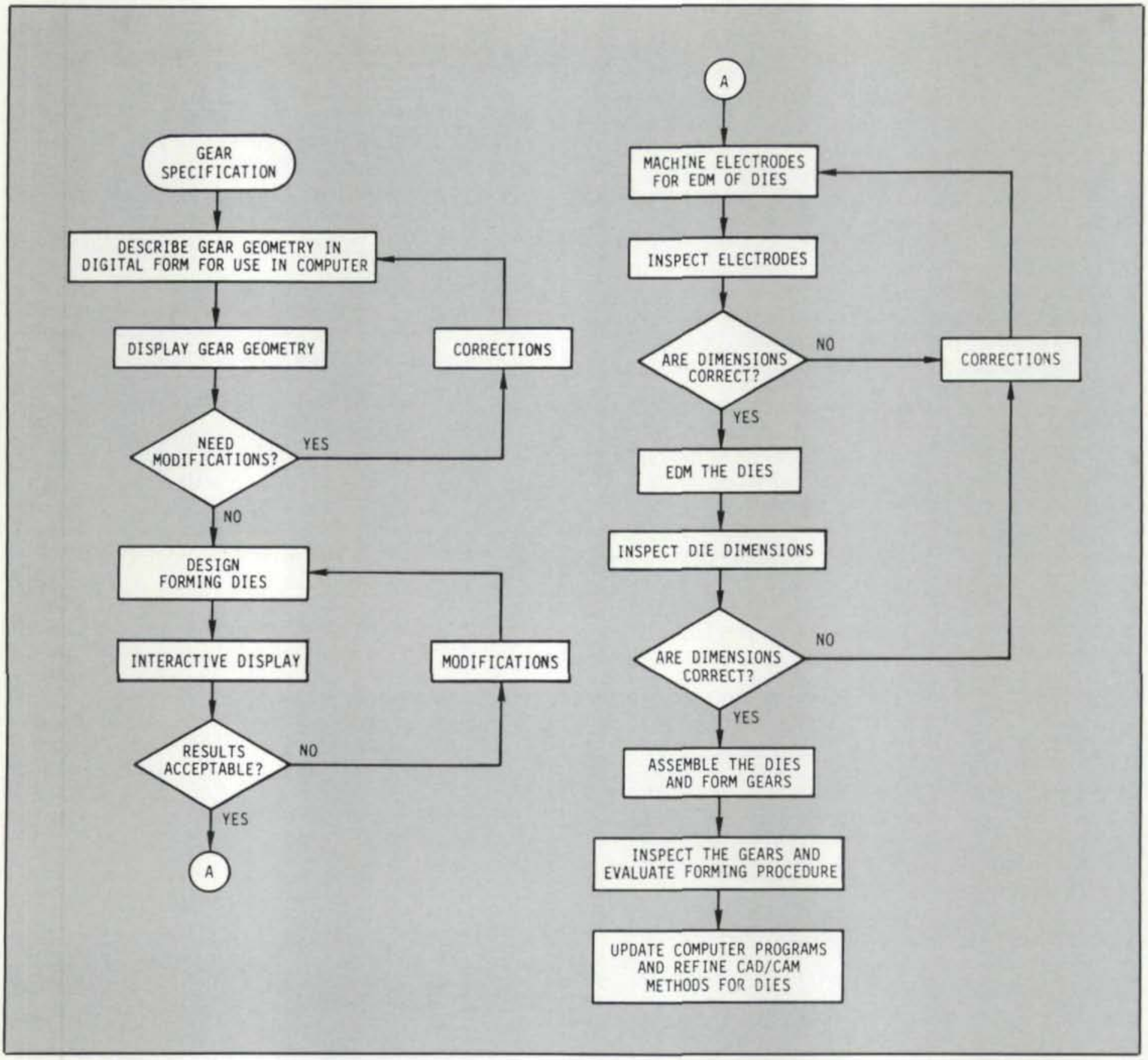

Fig. 2 – General Procedure for the Design of Gear Forming Dies

• pitch diameter – diameter of the theoretical pitch circle which is tangent to the corresponding pitch circle on a mating gear

The majority of the gears produced by conventional cutting methods are either hobbed or cut using a shaper cutter (2). In this study, for defining the tooth geometry, standard equations were used to simulate the gear cutting process (3-7). These equations are included into a computer program, called GEARDI, as discussed later.

Forming Load Prediction

To determine the elastic deflection of the forming dies, stresses acting on the die during the forming processes must be known. This stress analysis is necessary to obtain not only the elastic deflection, but also to calculate the forming pressure and load.

Extrusion

The extrusion process is seen schematically in Fig. 6. The punch load required to extrude a spur or helical gear is determined by estimating the following forces:

- the ideal deformation force,
- the force due to internal shear at the die entrance and exit,
- the friction force along the die walls and the punch.

Using the slab method of analysis, the equations for the punch force were determined and programmed.

DRAWING INFORMATION FOR: A DRIVEN COUNTER SHAFT		ENGLISH (INCH)	METRIC (MM)
IDENTIFICATION NUMBER		81.0220	81.0220
SET NUMBER		810.0123	810.0123
NUMBER OF TEETH		32.	32.
NORMAL DIAMETRAL PITCH (MODULE)		10.0000	2.540
NORMAL PRESSURE ANGLE		19.0000	19.000
HELIX ANGLE		31.5739	31.574
HAND OF HELIX		LEFT	
LEAD		19.2000	487.680
TRANSVERSE DIAMETRAL PITCH (MODULE)		8.5197	2.981
TRANSVERSE PRESSURE ANGLE		22.0064	22.006
MAXIMUM OUTER DIAMETER		4.079	103.60
MINIMUM OUTER DIAMETER		4.069	103.35
MAXIMUM TIP CHAMFER DIAMETER		4.049	102.84
MINIMUM TIP CHAMFER DIAMETER		4.039	102.59
THEORETICAL PITCH DIAMETER		3.7560	95.402
MINIMUM ROOT DIAMETER		3.511	89.18
BALL/PIN DIAMETER FOR (MOP)		0.2160	5.486
MAX. MEAS. OVER PINS (MOP)		4.1716	105.958
MIN. MEAS. OVER PINS (MOP)		4.1681	105.870
MIN. CALIPER MEAS. (4) TEETH		1.1056	28.082
MAX. CALIPER MEAS. (4) TEETH		1.1071	28.121
MEAN FACE WIDTH		0.875	22.23
MIN. NORM TOPLAND (MAX. O.D. W/O CHAM)		0.029	0.74
MIN. THEO. NORM. CIRC. TOOTH THICKNESS		0.1626	4.130
TOOTH THICKNESS @ HALF OF WHOLE DEPTH		0.1764	4.481
CASE DEPTH		.023/.033	0.59/0.83
MAX. PITCH DIAMETER RUNOUT (TIR)		0.0025	0.063

		ROLL ANG.	RADIUS	RADIUS
051 DATA	@ MAX. OUTER RADIUS	34.95	2.0395	51.803
	@ MAX. END OF ACTIVE PROFILE	33.99	2.0245	51.422
	@ MAX. HIGH CONTACT POINT	30.30	1.9697	50.030
	@ OPER. PITCH POINT	25.02	1.9000	48.260
	@ MIN. LOW CONTACT POINT	22.42	1.8697	47.491
	@ MIN. START OF ACTIVE PROFILE	17.45	1.8202	46.233
	@ START OF INVOLUTE CHECK	16.72	1.8138	46.070
	@ BASE RADIUS	0.00	1.7412	44.226
061 DATA	MAX. LEAD VARIATION		0.0004	0.010
	MAX. LEAD RANGE		0.0008	0.020
	CROWNING IN MIDDLE 80% OF TOOTH		.00000/.00050	.000/.012
071 DATA	MAX. RUNOUT (T.I.R.)		0.0025	0.063
	MAX. TOOTH TO TOOTH COMPOSITE VAR.		0.0008	0.020
	MAX. TOTAL COMPOSITE VARIATION		0.0032	0.081
	MAX. PITCH VARIATION		0.0004	0.010
	MAX. PITCH RANGE.		0.0029	0.073

Fig. 3—Example of A Typical Gear Manufacturer Summary Sheet Defining Gear Geometry

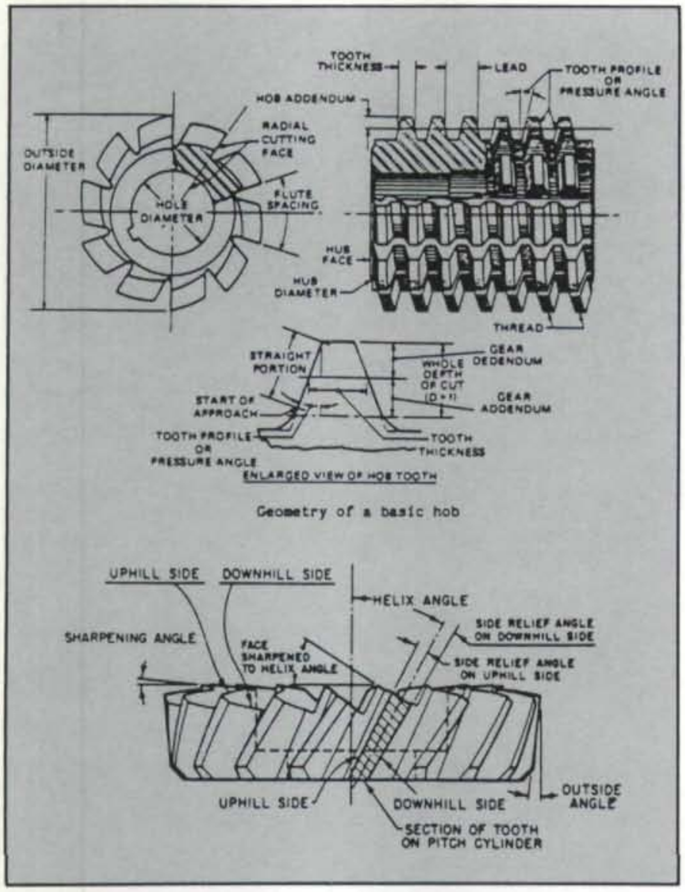

Fig. 4 – Geometry of A Hob and A Shaper Cutter

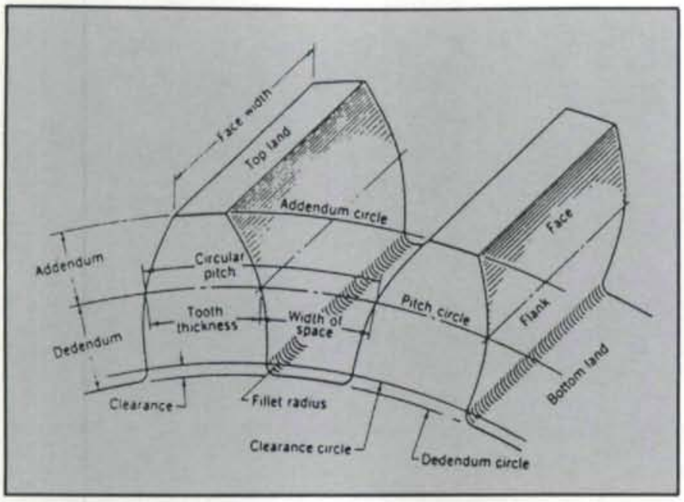

Fig. 5 – Gear Terminology

Forging

A typical tool setup for forging of gears is shown schematically in Fig. 7. The punch force necessary to fill the tooth cavity by radial metal flow was also calculated using the slab method of analysis and empirical equations. Additionally, the forging load was estimated using a Finite Element Method (FEM) based program in order to verify the calculations made by the slab method and empirical equations. The results of the FEM analysis correlated closely with the empirical analysis.

Estimation of Die Corrections

The geometry of the forming die is different from that of the formed gear because,

- The die insert is normally shrink fitted into the die holder causing a contraction of the die surfaces.
- In warm/hot forming, the dies may be heated prior to forming and further heated by the hot billet during forming. This causes the die insert to expand.
- During forming, due to forming stresses, the die surface deforms elastically.
- After forming, the gear shrinks during cooling from forming temperature to room temperature.

To obtain the desirable accuracy in the formed gears, each of the geometrical variations listed above was estimated and the die geometry corrected accordingly. Referring to Fig. 8, the original pitch radius is represented by R. Shrink fitting of the die causes the gear profile to shrink, hence the die must be increased by a corresponding amount, S. Similarly, increased die temperatures and forging pressures cause the die to expand. These two factors are compensated by the amounts H and E. Finally, a warm/hot formed gear will shrink during cooling; therefore, the die must be enlarged by the amount C. The results of the die correction analyses were used to alter the coordinates of the gear tooth profile to achieve the appropriate die geometry.

Cutting the Die

A common method of die manufacture is called electrical discharge machining (EDM). The process uses an electrode, usually made of graphite or brass, which is the negative of

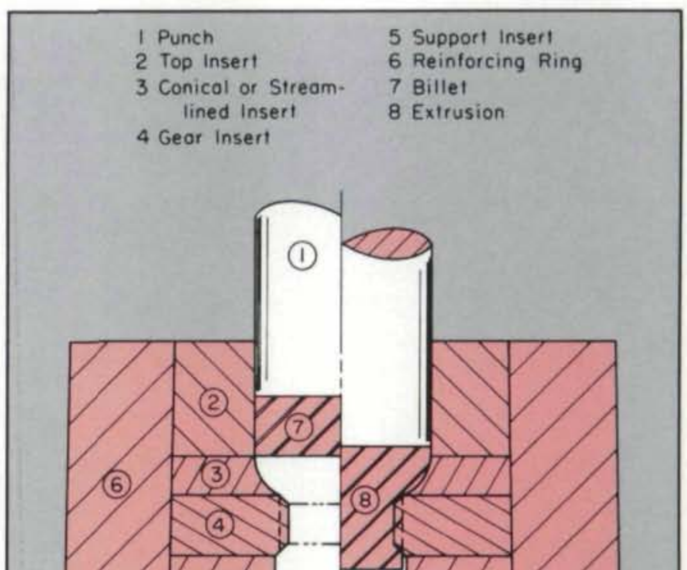

Fig. 6 – Schematic of the Extrusion Process

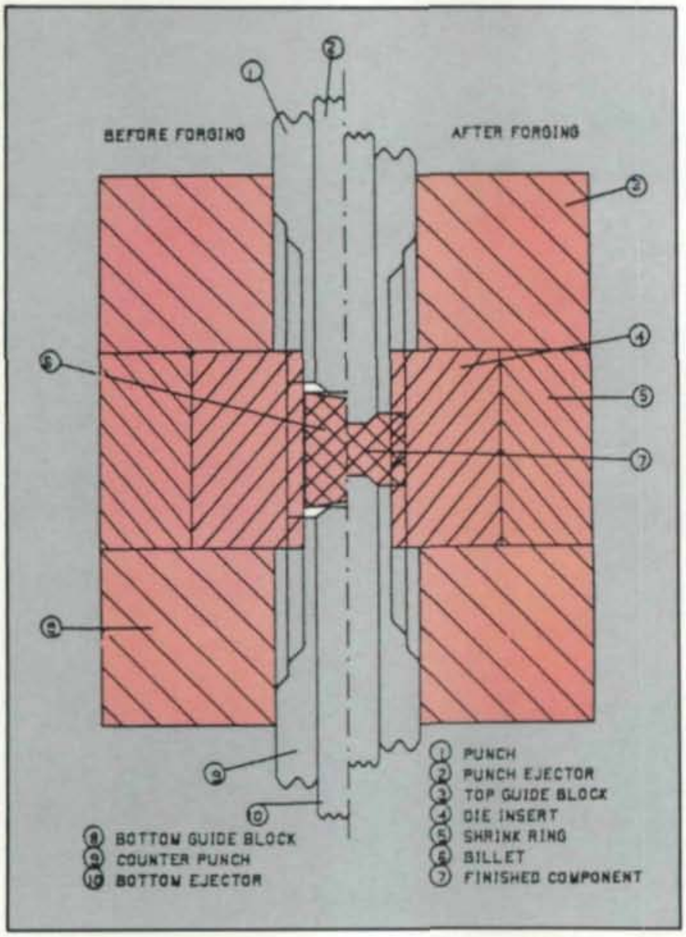

Fig. 7 – Typical Tool Setup for Forging Spur or Helical Gears

the die geometry. The electrode is brought close to, but not in contact with, the die material. An electrical current is allowed to arc across the gap which "burns" away the die material. Another form of this method of manufacture is called wire EDM. Current is passed through a straight wire that moves in two dimensions, burning the die geometry as it goes. Helical gear dies must be made by using a solid electrode but spur gear dies can be cut using either a solid electrode or a wire EDM process. In either case, a corrected set of gear tooth profile coordinates is needed. This new set of coordinates is computed by applying a correction factor to the radius vector from the center of the gear to each point on the gear tooth profile. The correction factor is a function of the values for S, H, E, and C as shown in Fig. 8.

As previously mentioned, the geometry of the die is different from the final gear geometry. When cutting the gear die using an electrode, it may be desirable to manufacture the electrode using a hob or shaper cutter specifically designed to cut the electrode geometry. The computer program, "GEARDI", allows the user to design this new cutting tool.

Computer Program "GEARDI"

Using the equations developed in Phase I, a graphics oriented computer program called GEARDI was developed. The main functions of GEARDI are:

- define the exact tooth form of a spur or helical gear,
- compute the forming load required to produce the current gear design,
- compute the coordinates of the corrected gear geometry necessary for machining the EDM electrodes by taking into account the change in the die geometry due to temperature differentials, load stresses and shrink fitting, and,
- determine the specifications of a tool which can cut the altered tooth geometry on a conventional hobbing or shaper cutter machine.

This program enables the user to design spur and helical gears, predict tooling loads and pressures, estimate metal flow for forming the gear, and define the geometry required to manufacture the tooling using conventional or wire EDM. Several examples of gears, currently being forged in industry, were tested in the GEARDI computer program. The predicted forging loads were within 10 percent of the actual loads measured during production runs.

GEARDI is an interactive, graphics-oriented program which runs on Digital Equipment Corporation VAX 11/780, 11/750, and PDP-11/44 computers. It is a menu driven program that allows the user to select various options from a pre-defined list. Fig. 9 is a simplified flow diagram of the program depicting the various menu options available to the user. One convenient feature, the "COMPARISON DISPLAY" option, allows the user to superimpose two gear profile drawings on the computer and determine the maximum difference between the two profiles. Fig. 10 shows the superposition of an original spur gear tooth profile and the corrected geometry which was determined by the program for a specific set of forming conditions.

The GEARDI program has powerful application possibilities, not only in the area of metalforming, but also in the area of gear and gear train design, with its ability to

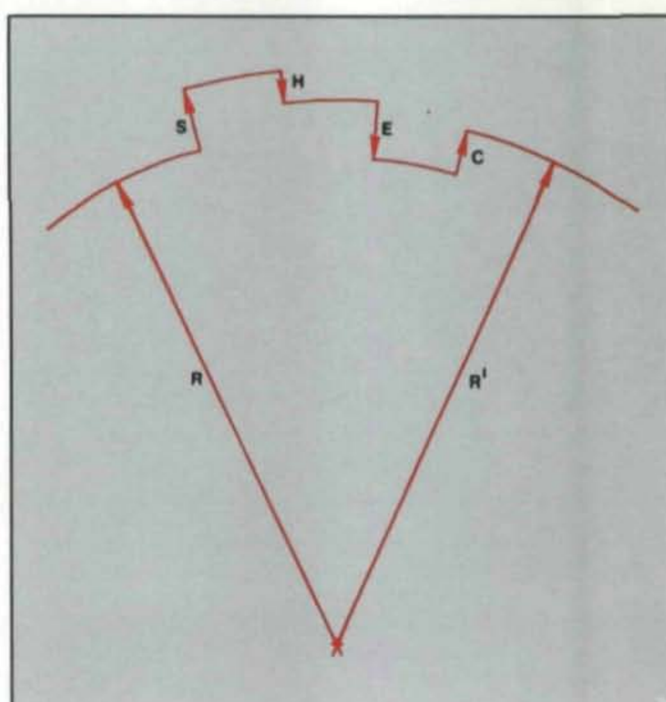

Fig. 8 – Correction to Gear Geometry (Symbols Explained In Text)

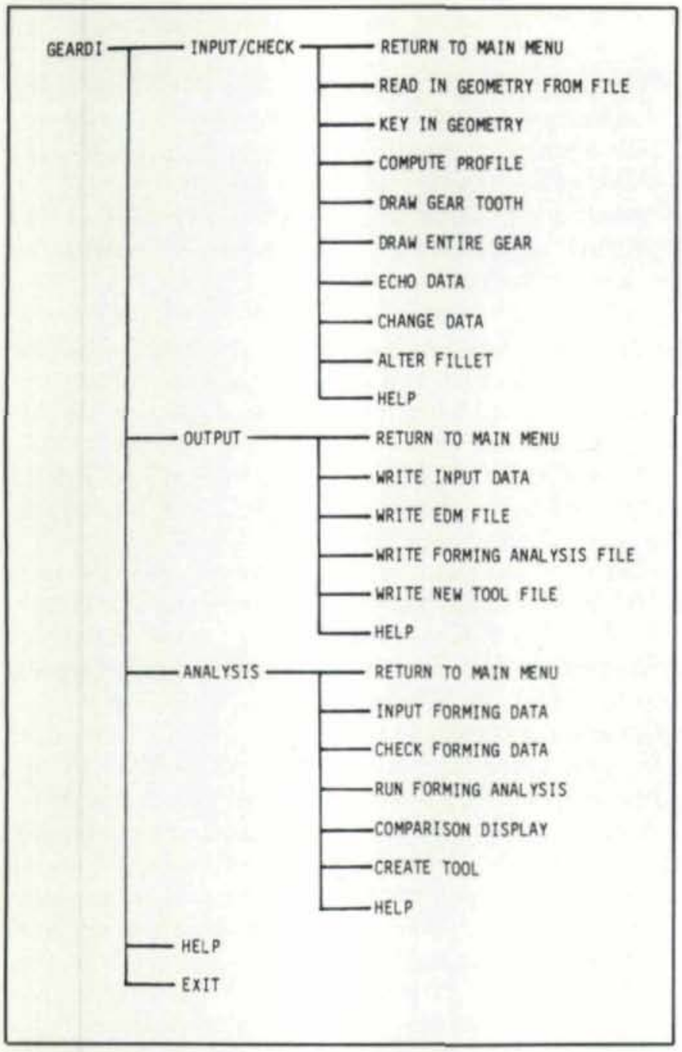

Fig. 9 – Flow Diagram for the Computer Program GEARDI

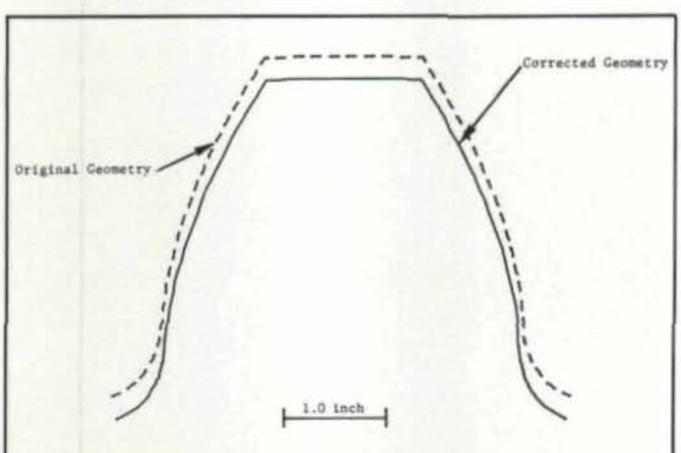

Fig. 10 – Computer Program Display of the Original Gear Tooth and the Corrected Gear Tooth Geometry

design hobs and shaper cutters and to modify the fillet from a trochoidal shape to a circular shape.

Spur Gear Extrusion Trials

Fig. 11 shows a picture of the tool setup for the spur gear extrusion trials, conducted at Battelle's Columbus Division using a 700-ton hydraulic press. The gears were extruded using a "push-through" concept. Each gear is first partially formed and left in the die while the punch is retracted. A second billet is placed on top of the partially formed gear and the press is cycled again. During this cycle, the partially

Fig. 11 – Tool Setup for Spur Gear Extrusion Trials

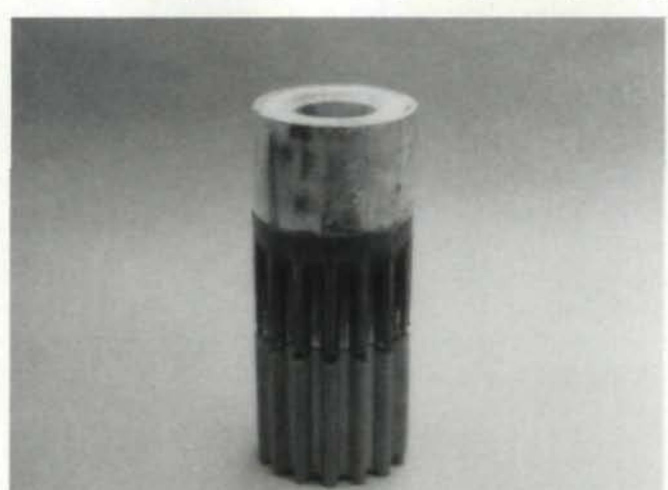

Fig. 12 – Sequence of Parts For Extruding Spur Gears. Billet is on Top, Partially Formed Gear is in the Middle, and Complete Extruded Gear is on the Bottom.

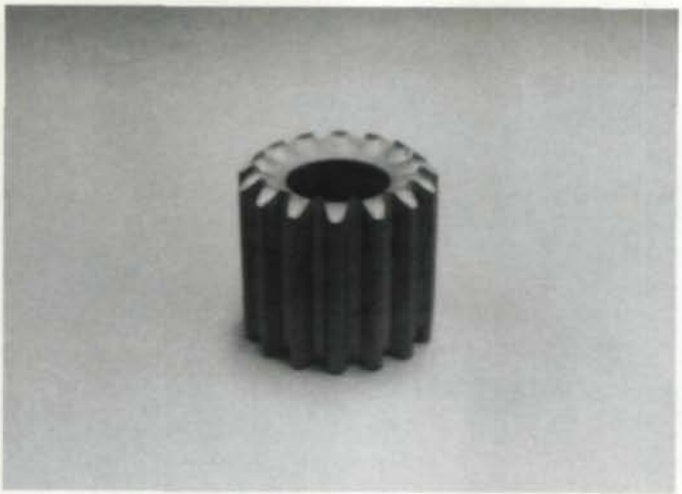

Fig. 13—Extruded and Shot-Blasted Spur Gear

formed gear is finish formed and pushed through the die, dropping out the bottom of the die. Fig. 12 shows the sequence of parts in the tooling. Once formed, the teeth on the gear are not machined further. A fixture which holds the gear on the pitch line of the teeth is used to finish machine the inside and ends of the gear. The spur gear formed in these trials was designed to be an AGMA quality class 8 gear. Measurements taken on the extruded gears indicated a gear of between AGMA quality 7 and 8. An extruded gear which has been shot-blasted is shown in Fig. 13.

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This paper is based on work conducted by a team of engineers. The authors gratefully acknowledge the contribution of Dr. T. Altan, Battelle Columbus Division, and Messrs. A. Sabroff and J. R. Douglas, from Eaton Corporation.

E-2 ON READER REPLY CARD

ANALYZING GEAR TOOTH STRESS . . .

(continued from page 15)

Having established that stress levels vary in a predictable and quantitive way, work has begun on correlating the stress data obtained from the finite element stress program to sources of experimental data. Two parallel programs are now underway to provide such correlation. The first program will analyze several hundred fatigue test data points from full scale axle tests on a four square fatigue tester. The purpose of this program is to establish a reliable S-N curve for each of the modes of fatigue failure; e.g., bending fatigue and subsurface shear. The second program will involve fatigue data obtained from simulated gear tooth specimens using a closed loop hydraulic tester. The test data obtained from the simulated gear tooth specimens will be used to augment the data obtained from the full scale axle tests thus providing a relationship between S-N curves for various materials and heat treatments to the S-N curve obtained from full scale testing. The successful completion of this final step should result in establishing the finite element gear strength program as a powerful gear analyzing program for the design of bevel and hypoid gears.

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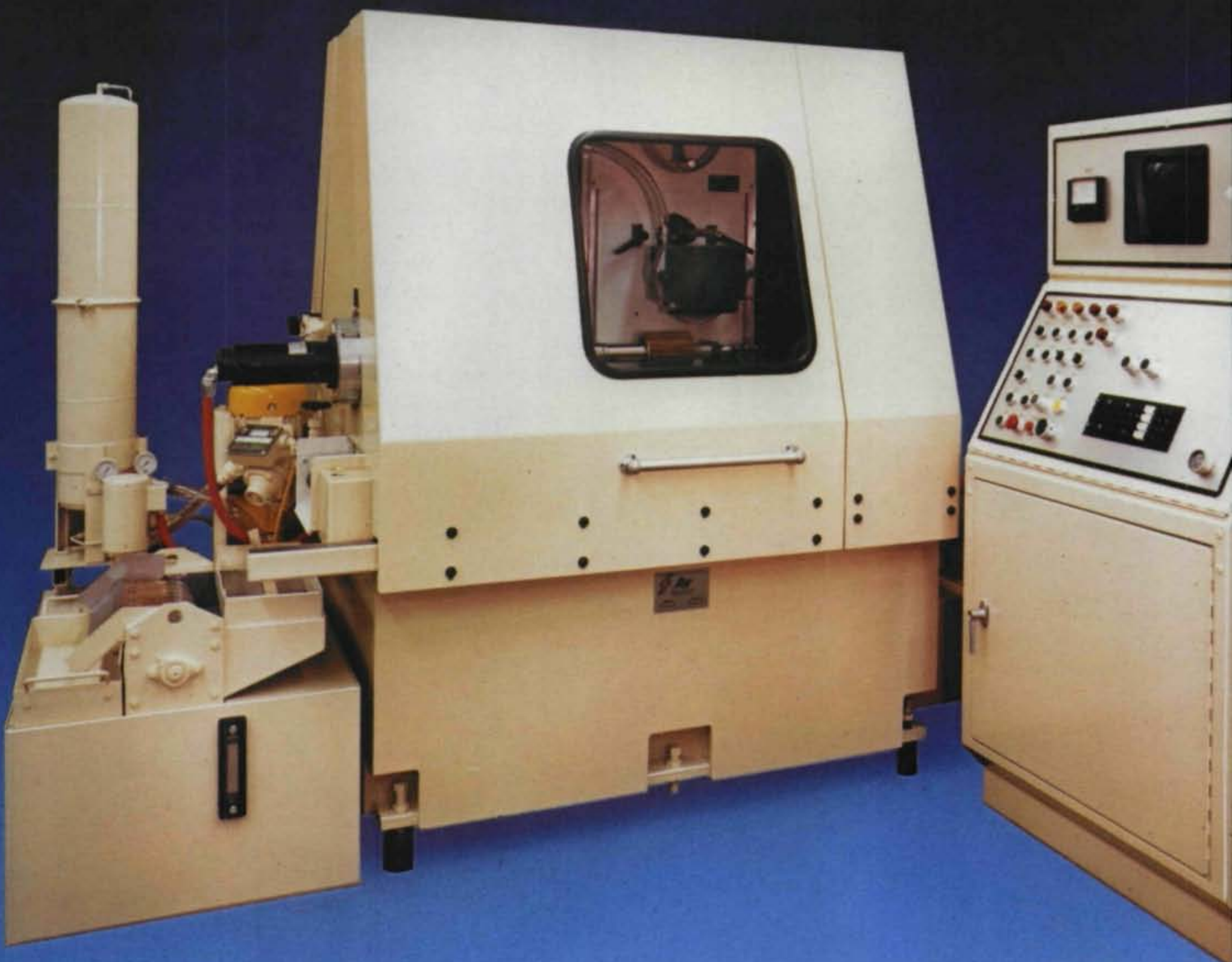

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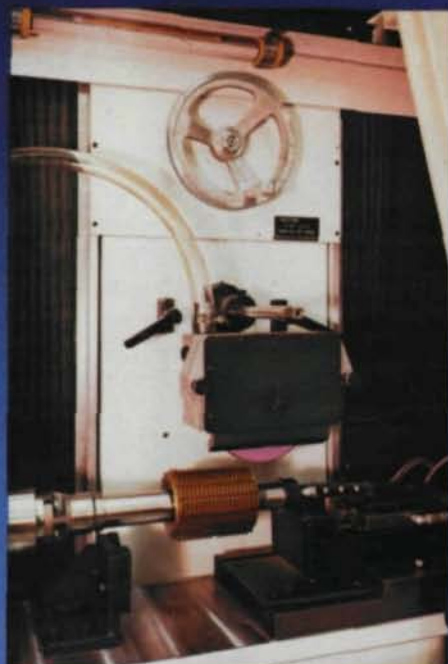

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CIRCLE A-7 ON READER REPLY CARD

The Effect of Lubricant Traction On Wormgear Efficiency

by
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Mobil Research & Development Corporation

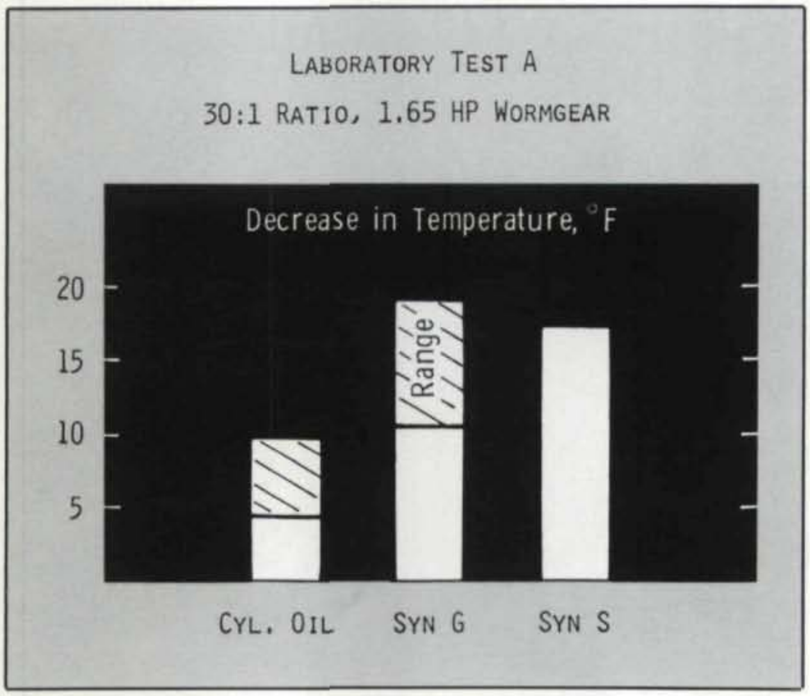

Fig. 1 – Improvement Over Conventional S/P Mineral Based Oil

Abstract

The effect of various lubricant factors on wormgear efficiency has been evaluated using a variety of gear types and conditions. In particular, the significant efficiency improvements afforded by certain types of synthetic lubricants have been investigated to determine the cause of these improvements. This paper describes broad wormgear testing, both in the laboratory and in service, and describes the extent to which efficiency can be affected by changes in the lubricant; the effects of viscosity, viscosity index improvers and, finally, synthetic lubricants are discussed. The work concludes that lubricant tractional properties can play a significant role in determining gear efficiency characteristics.

Introduction

Over the past ten years, the trends to hotter-running industrial gears has provided the authors' company with the opportunity to evaluate synthetic industrial oils developed with the superior thermal/oxidative performance. In several instances, it was noted that not only was oil life improved by using synthetic products, but that the equipment ran cooler than with the mineral-based products originally used. This behavior has been reported.[1] The simplest explanation for cooler-running is a reduction in horsepower loss; i.e. an improvement in power transmission efficiency.

These findings, in part, resulted in a significant effort to evaluate industrial gear efficiency characteristics of different lubricants, including the synthetic products.

Laboratory Evaluations of Wormgear Efficiency

Results of wormgear tests available in the authors' laboratories in Europe and the U.S. appear in Figs. 1 through 4.

Equilibrium operating temperature data obtained with Wormgear A operated at 1500 rpm and 100% nominal load are shown in Fig. 1. Both synthetics show significant temperature decreases. A conventional steam cylinder oil (AGMA 7 Comp.) is also shown for comparison against the reference, which is a conventional AGMA 7 EP mineral-based oil. The viscosity grades (see Table 1) for Syn S and Syn G were chosen from their respective product families to be similar to that of the reference oil at the approximate 100°C operating temperature of the tests.

Data in Fig. 2 compares both efficiency and temperature rise differentials using a second manufacturer's gearbox, Wormgear B. Similar temperature decreases were found for the synthetic products, but the steam cylinder oil ran slightly hotter than the reference conventional S/P mineral oil. These results are reflected in the efficiency data for these oils which show 2-3% benefit for the synthetics and a directional worsening for the cylinder oil. Efficiency measurements in this test were made by means of strain gauges on the input and output of the gearbox which was run at 1760 rpm input and reached about 100°C at equilibrium. Loading was achieved hydraulically as part of a four-square test arrangement.

Fig. 3 data shows benefits in efficiency for both synthetic products and the steam cylinder oil versus the reference mineral EP oil using Wormgear C (30:1 reduction ratio). This test is run at 1500 rpm and at 96 and 117% thermal load capacity. Fig. 4 shows similar data for Wormgear D, a 50:1 ratio wormgear operated at 1680 rpm input at 100 and 200% Class 1 mechanical rating. This latter test is run with the oil sump thermostatted at 95°C. Both tests C and D measure efficiency via input and output shaft torque strain gauges with dynamometer loading.

The data of Figs. 1 through 4 clearly show the correlation of improved efficiency and temperature control for lubricants Syn S and Syn G compared with the mineral sulfur/phosphorus oil reference. Steam cylinder oil, also, shows a general benefit for both measurements, but with significantly smaller degree of improvement.

These data are summarized in Table 2. Also shown are results of gear manufacturer tests with Syn S compared with reference mineral oils of either the compounded steam cylinder type or the AGMA EP type. The degree of efficiency benefit recorded in these latter tests depended on the operating conditions of the tests, and Table 2 lists the average benefits recorded. The gear manufacturer tests also indicated benefits in temperature control for Syn S.

The gear efficiency benefits for Syn S in tests B through G show a wide range of numerical results. These data are rationalized in Fig. 5 which shows a good correlation of efficiency benefit as a function of increasing gear reduction ratio. Based on the measured and catalog efficiencies for these gears (using conventional oil) the rule-of-thumb effect of Syn S is to reduce the gearbox *inefficiency* by 20-25%.

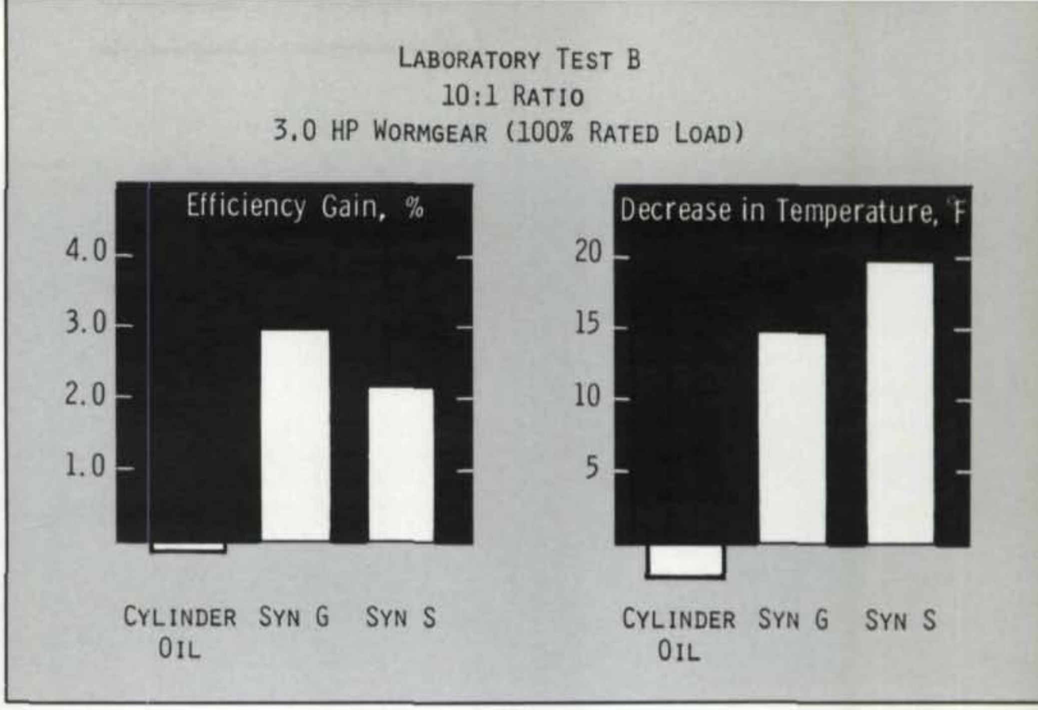

Fig. 2—Improvement Over Conventional S/P Mineral Based Oil

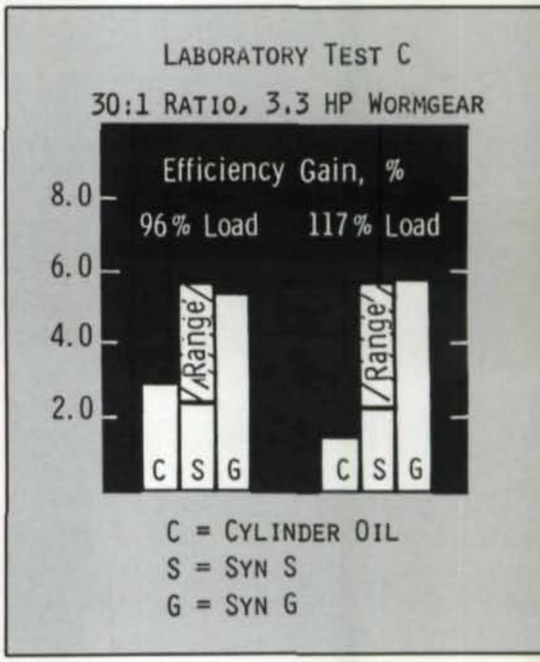

Fig. 3—Improvement Over Conventional S/P Mineral Based Oil

Practical Applications for Improved Wormgear Efficiency

Benefits other than the cooler running characteristics initially found for these synthetic products are apparent:

One gear manufacturer, Hub City Division of Safeguard Power Transmission Co., has applied the benefits of Syn S to increase wormgear horsepower ratings. Catalog thermal input horsepower ratings have been increased 10 to 15% when the recommended synthetic oil (a Syn S rebrand) is used. This represents a new design application for this product which already is used in sealed-for-life wormgear units based on its extreme operating temperature capability.

A second application for increased transmission efficiency is to reduce equipment power requirements, thereby, reducing utility costs. Field measurements of power draw of various industrial equipment in customer plants has now been demonstrated for Syn S and is currently being evaluated for product Syn G.

(continued on next page)

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Syn S field results in Table 2 show measured average electrical power requirements for both worm and steel gear industrial transmissions. Efficiency measurements in field applications can be more difficult than in the laboratory because operating conditions are not as precisely controlled. A statistical approach is often necessary. This was the case in the second steel mill test in which the combined power requirements of the motor drives of a series of five wormgears were measured by means of a recording wattmeter. Two basic operations were evaluated in this corrugated steel pipe drive: a regular drive cycle and a cut cycle. Data for the two cycles are shown as normal distributions in Figs. 6 and 7.

The Syn S results show lower mean (50%) power draw for both cycles: the non-overlap of the 90% confidence limits for each distribution comparison indicates a greater than 99% confidence that the benefits measured were real and not the result of chance. The average benefits measured for the two cycles were 5.8 and 5.9%.

A different approach was taken in the power consumption test at the food processing company shown in Table 2. In this case, two food aerator units were operating in parallel. In this test the mineral oil in the gear drive of only one of the units was replaced with synthetic product and the other was maintained as a control. Fig. 8 demonstrates the relatively constant current draw for blower 2 (oil unchanged) compared with the second unit which had a measurable (6%) reduction when the mineral oil was replaced by Syn S.

The efficiency/power benefits for Syn S in laboratory, gear builder and customer evaluations are summarized in Fig. 9

Correlation of Wormgear Efficiency with Lubricant Traction

The mechanism by which synthetic lubricant products, Syn S and Syn G, improve wormgear efficiency has been investigated in the authors' laboratories. The wide acceptance of steam cylinder lubricants which employ relatively large percentages of fatty additives has been attributed to their ability to produce low friction films on the surface of the gear teeth. The present work generally supports the low friction/improved efficiency characteristics of the steam cylinder oils, and it was the surface friction mechanism which was first investigated as a possible explanation for the greater benefits shown by the synthetic lubricants.

In one comparison using Wormgear D, Syn S additives blended in the reference oil mineral base versus the mineral reference oil additives in the synthesized hydrocarbon base oil, clearly demonstrated the efficiency benefits to be retained not by the Syn S additives, but with the synthetic basestock. This indicated the efficiency benefits to be associated with the nature of the synthetic basestock, and not to result from a change in the frictional properties at the gear surfaces.

Operating viscosity was also considered as a possible explanation even

TABLE 1

TEST LUBRICANT INSPECTION PROPERTIES

	TEST LUBRICANT		
	S/P Mineral Reference	Syn S	Syn G
Base Oil Type	Mineral	Synthesized Hydrocarbon	Polyglycol
Viscosity			
AGMA Visc Grade	7	"6.5"	5
cSt @ 40°C	437	383	225
cSt @ 95°C	35	46	35
cSt @ 100°	30.2	39.0	29.7
Viscosity Index	95	145	164
Pour Point, °F	+20	−40	−15
Flash Point, °F	450	500	440

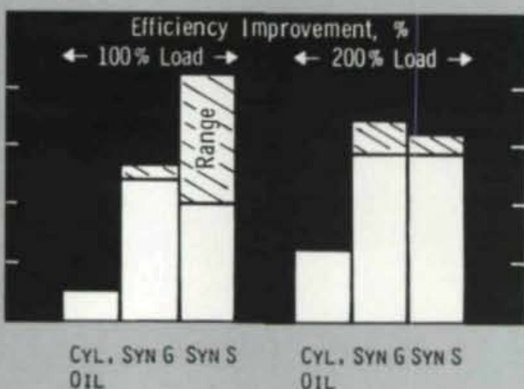

Fig. 4 – Improvement Over Conventional S/P Mineral Based Oil

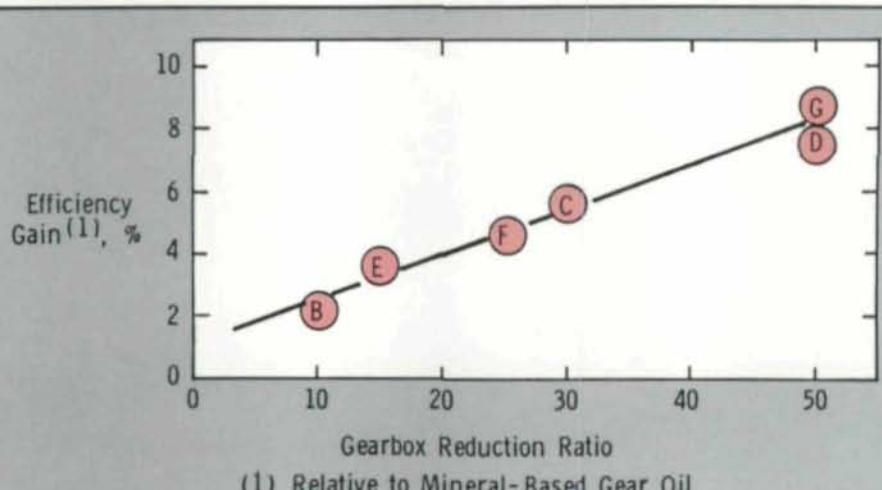

Fig. 5 – Energy Efficient Gear Lubricants Benefits For Syn S

though, as discussed earlier, the tests were designed to exclude this possibility. Nevertheless, ancillary experiments with Wormgear D evaluating the effect of changing the viscosity of the reference oil were undertaken and appear in Table 3. Compared with AGMA 7 EP reference oil, no significant difference is seen in either increasing or decreasing viscosity. This insensitivity to viscosity has also been verified in work with Syn S viscosity variations.

Table 3 also shows the results obtained with viscosity index-improved products. In this work, oil formulations were prepared with the normal reference oil functional additives, using AGMA 5 viscosity base oil, but with VI-improvers, X and Y, added to bring the final viscosity to the AGMA 7 level. In this way a quasi multigrade industrial oil was produced, but in neither case was a significant efficiency benefit obtained, with VI improper Y giving a negative effect. An explanation for this may be the poor shear stability found for Y in viscosity measurements under high shear conditions; X was found to be highly shear stable.

Finally, fluid tractional properties were considered as a possible explanation of efficiency effects. Traction here is defined as unit fluid friction in the high-pressure mesh of the gear, and is distinct from both viscous churning losses and gear surface metal/metal frictional effects.

The tractional properties of test fluids were measured using a roller disc machine. This equipment employs a pair of 3.265" diameter cylindrical test rings mounted on hydrostatic bearings which can be loaded against each other by means of a hydraulic piston. Each ring can be driven independently in either direction by induction drive units with electronic feedback control. The traction force between the discs is measured by strain gauges mounted in each disc drive mechanism. The absence of disc surface frictional effects is ensured by polishing disc surfaces to better than 2 microinch finish. Elastohydrodynamic calculations verified high specific film thickness and absence of metal contact during the tests.

Operating conditions for the roller disc machine traction measurements are shown in Table 4. Materials and surface loading were chosen to simulate the conditions of Wormgear Test D operating at 100% Class I load.

TABLE 2

EFFICIENCY BENEFITS MEASURED FOR SYNTHETIC INDUSTRIAL GEAR OILS COMPARED WITH CONVENTIONAL MINERAL-BASED PRODUCTS

	Gear Manufacturer	Reduction Ratio	Operating Horsepower	Average Efficiency Benefit, %	
				Syn S	Syn G
Authors' Laboratory Tests					
A – Mobil Oil Francaise	Lechner	30:1	1.65	*	*
B – Mobil R&D Corp.	Boston	10:1	3.0	2.2	3.0
C – Mobil Oil Francaise	Durand	30:1	3.2/3.9	5.6	4.0
D – Mobil R&D Corp.	Cone Drive	50:1	3.0/6.0	7.7	7.1
Gear Manufacturers' Evaluations					
E – Hub City Division Safeguard Power Transmission Co.	Hub City	15:1	1.55/2.0	3.8	–
F – Ex-Cell-O Corp., Power Transmission Division	Cone Drive	25:1	6.5/8.1	4.4	–
G – Winsmith Division of UMC Industries, Inc.	Winsmith	50:1	0.5-1.0	8.8	–
Equipment Users' Evaluations					
Steel Mill	B&K	15:1	122-142	6	–
Steel Mill	IMW Industries	39:1	75	5.8	–
Food Processing Company	Philadelphia Gear	3:1(a)	–	6	–
Textile Company	–	–(b)	~50	2	–

*Temperature Rise Measurements only; differential compared with reference mineral oils: Syn S, −17°F; Syn G, −15°F.

(a) Helical gear.
(b) Spur gear/chain drive combination.

TABLE 3

VISCOSITY EFFECTS ON WORMGEAR EFFICIENCY (TEST D)

AGMA Viscosity Grade	Percent Efficiency Change* Relative to Reference
8 EP	+0.1
7 EP	Ref
6 EP	−0.5
5 EP	−0.4
5/7 EP (V.I. Improver X)	+0.7
5/7 EP (V.I. Improver Y)	−3.3

*Test repeatability ~1.0%

TABLE 5

ROLLER DISC MACHINE TRACTION COEFFICIENT RESULTS

Lubricant	Traction Coefficient(1) @ 720 fpm	Estimated Gear(2) Friction, f
Syn S	0.012	0.015
Syn G	0.013	0.015
EP Mineral Oil	0.018	0.021
Syn T	–	0.030

(1) Slide/Roll Ratio $= \dfrac{U_A - U_B}{1/2\,(U_A + U_B)} = 2$ ("Pure" sliding)

(2) Wormgear Test D results using equation:

$$\text{Efficiency} = \frac{1 - f \tan\lambda}{1 + f \cot\lambda}$$

where f = friction coeff.
λ = lead angle (Reference 2)

TABLE 4

ROLLER DISC MACHINE OPERATING CONDITIONS

Disc Material:	A – AISI 4150 Resulfurized Steel, ~54 Rc (3/4" face width)
	B – SAE 65 Bronze, 80 Brinell Hardness (1" face width)
Disc Speeds, U:	A – 400-1600 rpm
	B – ± 120-1400 rpm
Bulk Oil Temperature:	150°F
Disc Load	400 lb
Hertzian Surface Stress:	48,600 psi

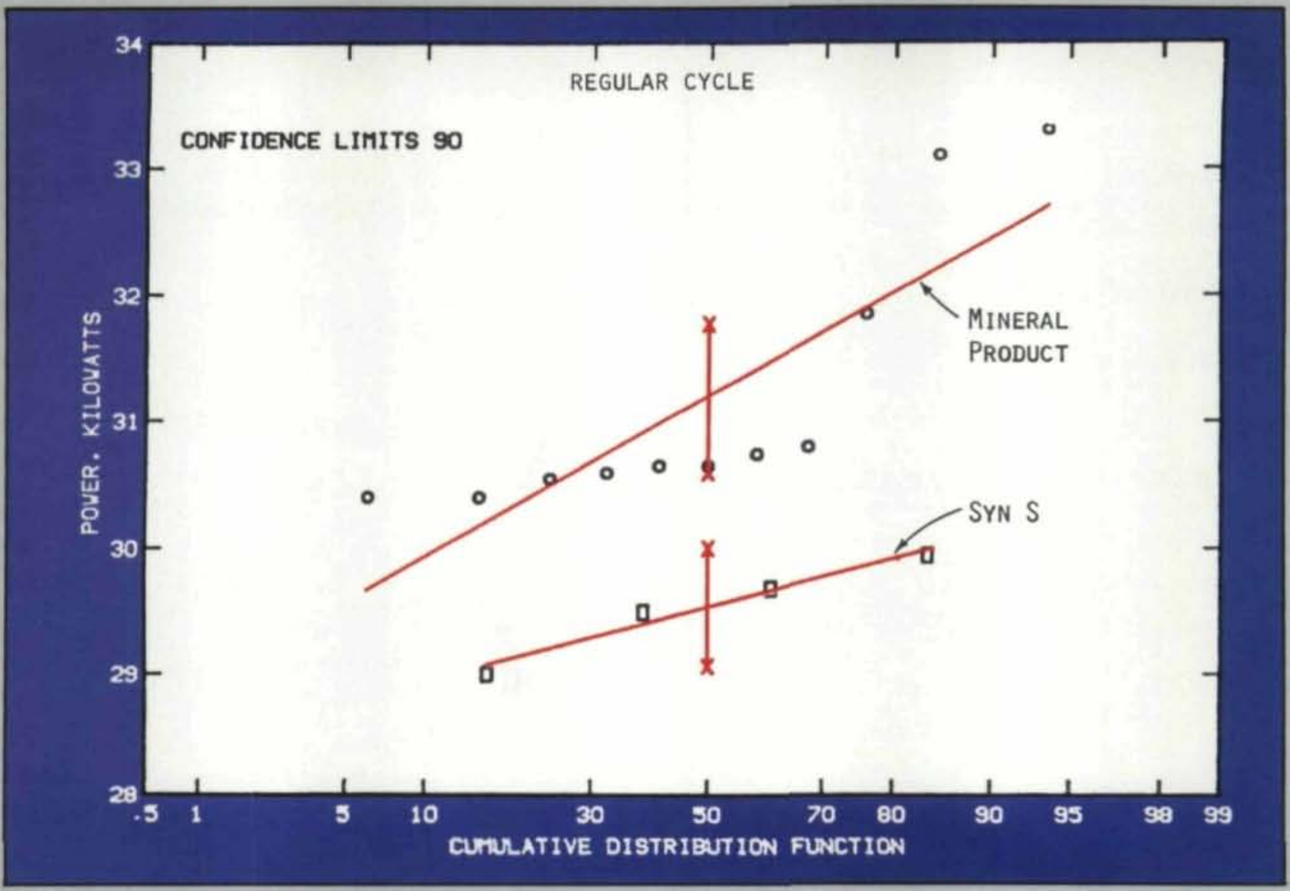

Fig. 6 – Corrugated Steel Tube Drive Test

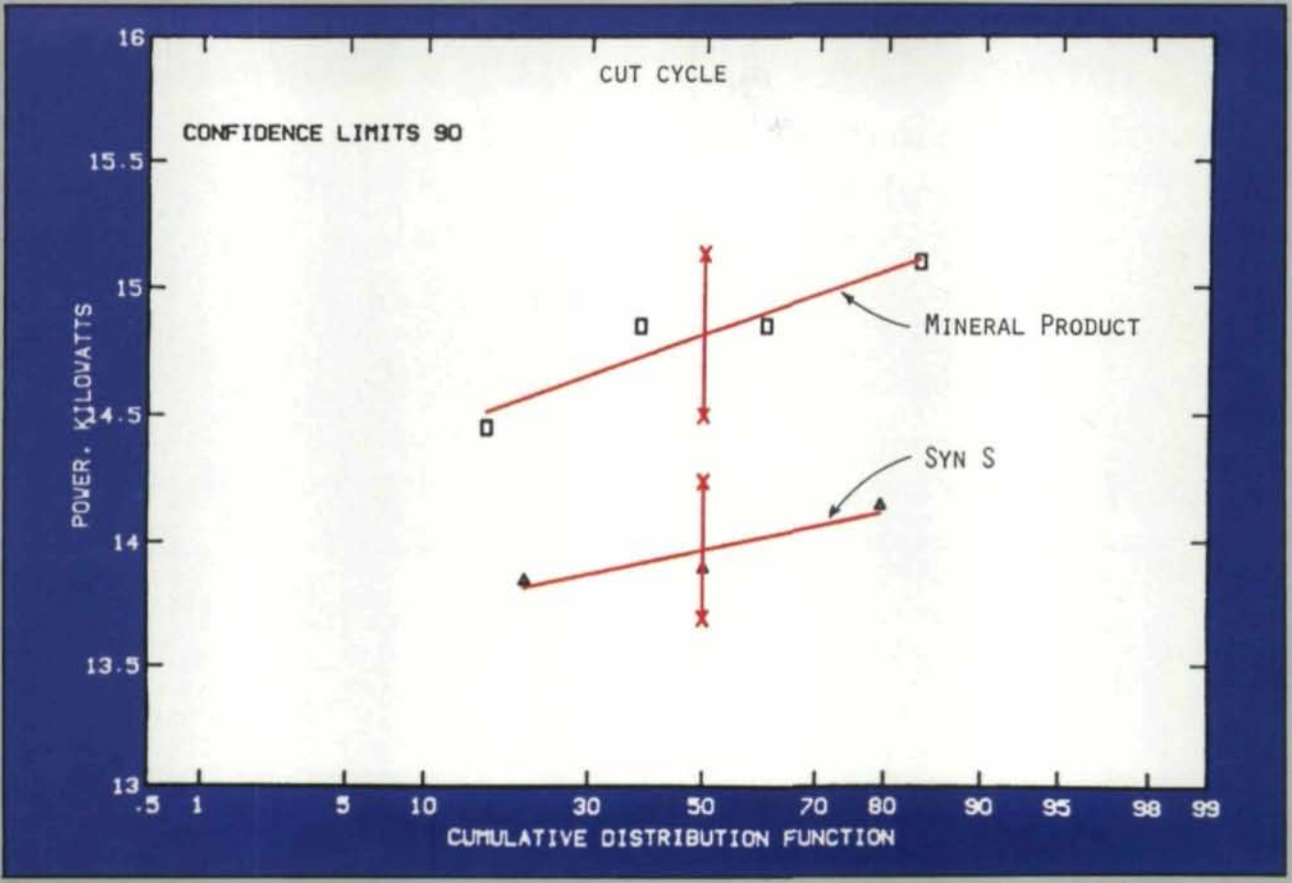

Fig. 7 – Corrugated Steel Tube Drive Test

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Based on the disc speeds chosen, results are calculated in terms of traction coefficient as a function of slide-to-roll ratio and relative sliding speed. Data for the condition of pure sliding at a speed of 720 fpm are shown in Table 5, indicating a significant reduction in traction coefficient for the synthetic oils. For comparison, an estimate of gear friction based on a theoretical relationship[2] for wormgear efficiency is also shown in Table 5. Despite the simplicity of the relationship, good correlation is obtained between the traction and friction data as a function of lubricant type. Note also that the traction co-efficients for the synthetic products are reduced about 30% compared with mineral oil – a reasonable agreement with the gear inefficiency reduction discussed previously. The calculated gear friction results are somewhat higher, which may be explained in terms of the other loss mechanisms outside the gear mesh; e.g., churning, bearing and seal losses.

This strong correlation was recently verified by evaluating a commercial synthetic traction fluid in Wormgear Test D. The manufacturer's information indicated the traction coefficient of this fluid, designated Syn T, to be about 50% greater than that of a mineral oil. Based on this, a significant detriment to efficiency in the wormgear was predicted. Due to the low viscosity of Syn T, the reference mineral oil viscosity was reduced to an AGMA 2 EP oil, and the tests were run at 125°F where both oils have viscosities equal to that of the usual reference oil at 195°F, the normal temperature of the test.

This work showed Syn T approximately 10% less efficient than the reference mineral oil at 100% loading. At 200% loading, the efficiency with Syn T was so poor that the cooling water to the gearbox was unable to maintain the operating temperature at 125°F and the test was terminated to avoid damage to the gears. These results strongly support the influence of fluid traction in determining wormgear efficiency.

Work is now underway to measure the traction coefficient of Syn T in the roller disc machine for inclusion with Table 5 results. Substitution of the gear efficiency result for Syn T in the gear friction equation yields a value approximately 50% greater than that for mineral oil – in good agreement with the manufacturer's reported finding for traction coefficient.

An important aspect of the work with Syn T is that not all synthetic lubricants possess low tractional characteristics. It is most likely that lubricant molecular structure is the key to tractional properties and that structures can be synthesized to give either high or low traction depending on the needs of the application.

Conclusions and Further Work

1. The cooler running characteristics of two synthetic industrial lubricants in gear applications have been correlated with wormgear efficiency.
2. Gear efficiency improvements have been shown to result in lower power requirements in industrial applications.
3. The improved efficiency afforded worm gears by these synthetic oils has been used to increase wormgear thermal horsepower ratings.

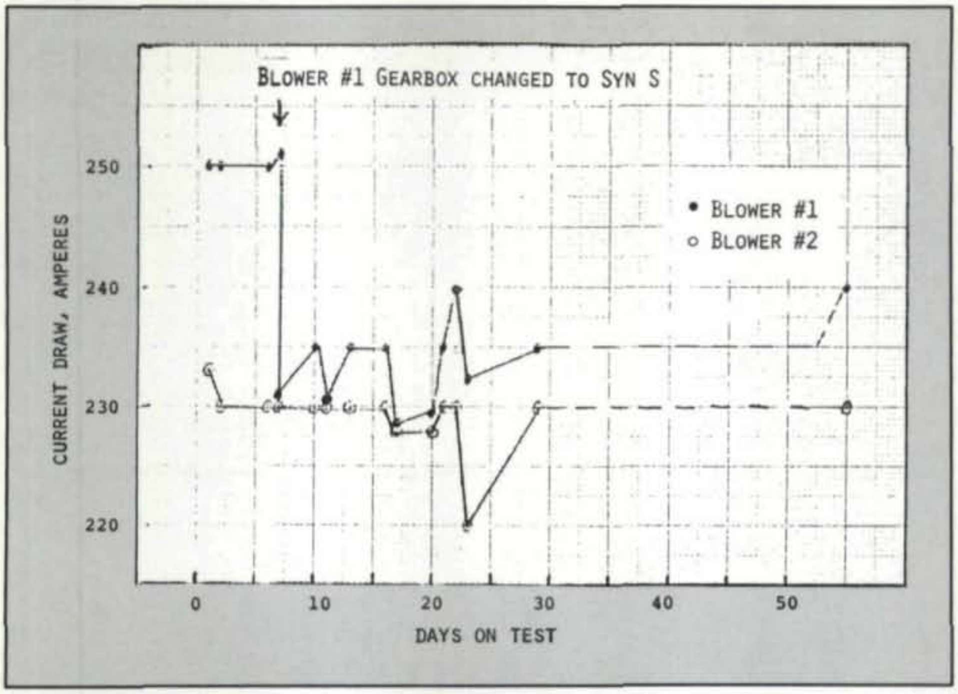
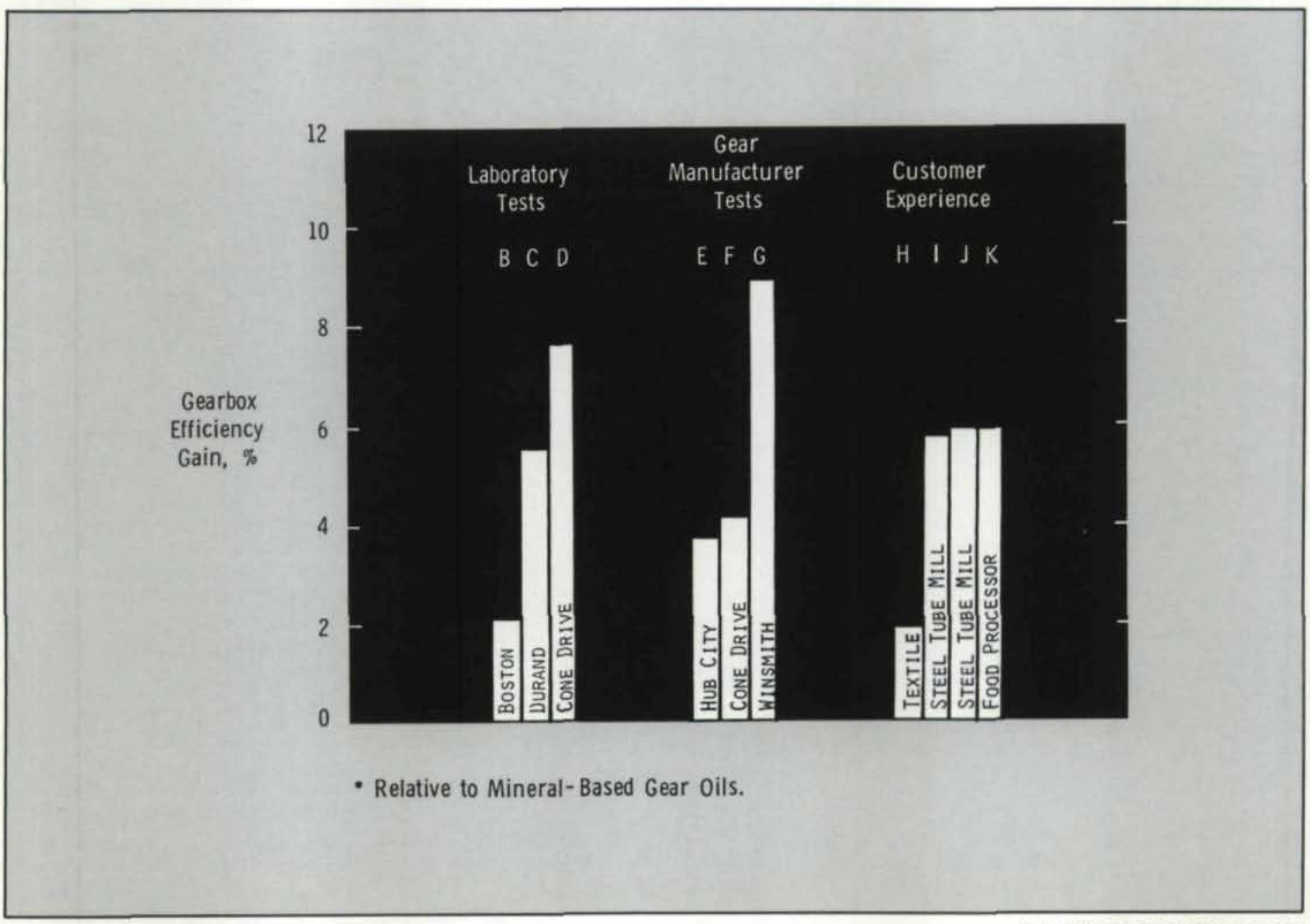


Fig. 8—Food Processor Power Consumption Test

Fig. 9—Average Efficiency Benefits* Demonstrated for Syn S



4. Lubricant tractional properties have been shown to be a significant factor in determining wormgear efficiency.
5. Future work will focus on the efficiency characteristics, particularly with respect to synthetic lubricants, of non-worm industrial gearing and will evaluate lubricant tractional properties at the higher pressures operating in steel gearing.

The authors gratefully acknowledge the permission granted by the gear manufacturers to publish their results.

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This paper was published previously by The American Gear Manufacturers Association 1981, #P254.33, and by Organi Di Transmissione (Italy) 1982.

High Power Transmission with Case-hardened Gears and Internal Power Branching

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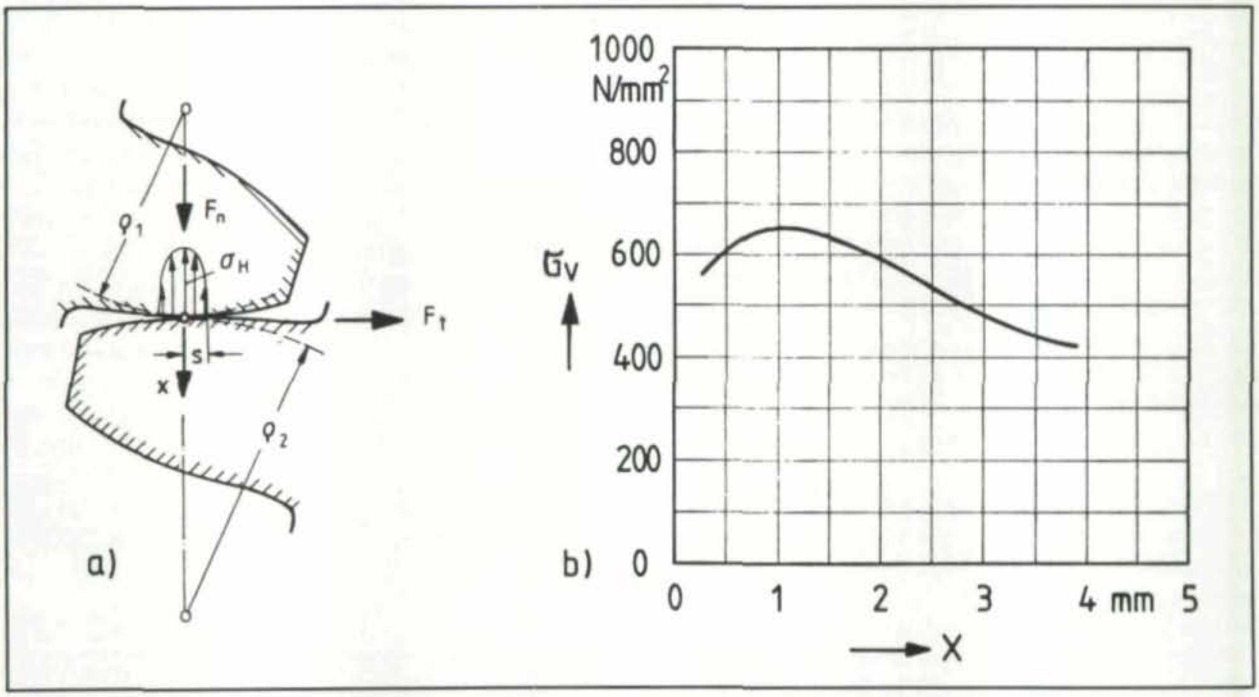

Fig. 1 a) Contact of two tooth flanks. $\rho 1$, $\rho 2$ = radii of curvature at the contact point of unloaded flanks;
Fn = tooth normal force; 2s = flattening width under load;
σH = maximum Hertzian contact stress at the tooth flank surface.
b) Dependence of equivalent stress σv on the depth x, module = 25 mm.

Introduction

In the field of large power transmission gear units for heavy machine industry, the following two development trends have been highly influential: use of case hardened gears and a branching of the power flow through two or more ways. The maximum possible torque of large gear units is limited by the machine capacity since the gear cutting machines can only manufacture gear wheels up to a maximum diameter. The highest possible output torque of gear units with case hardened gears and power branching is about ten times higher than conventional gear units with through hardened gears and without power branching.

The advantages of case hardened gears lie in their higher tooth flank load carrying capacity. The variation in the sub-surface strength matches ideally the sub-surface stress distribution. By optimising the gear geometry to balance the flank capacity to the root strength, the torque carrying capacity of case hardened gears can be four times higher than that of through hardened gears having the same diameter.

Power branching leads to a further increase in the torque carrying capacity. Such gear units have one input and one output shaft. Within the gear unit, the power at the gear of the input shaft is branched out and flows together at the out-

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put shaft. To achieve equal power distribution in each branch special design features are required.

Firstly, the advantages of case hardened gears are shown. Then, the dependence of output torque on the gear unit size and weight is demonstrated. Also the efficiency of different power branching gear stages and simple gear stages are compared with one another. Lastly, the design of three large gear units, as given below, are presented:

- Rolling Mill gear unit with two way power distribution
- Tube Mill central gear unit with three and six way power distribution
- Planetary gear unit for ball mill drive with three way power distribution.

Gear Materials

The load capacity of gear teeth increases with a decrease in diametral pitch, i.e. decrease in number of teeth. The tooth flank load capacity is the decisive parameter for the dimensioning of a gear pair that has a minimum number of pinion teeth without undercutting.

The loading of a tooth flank along the common line of contact is calculated as the pressure load between two cylinders with contact lines having the same lengths and radii of curvature as the unloaded tooth flanks. The load on the tooth flanks is obtained approximately from the maximum Hertzian contact stress assuming that the material properties are the same.

The Hertzian contact stress alone, does not determine the load configuration. In the contact zone an elasto-hydrodynamic oil film pressure is developed, dependant on the rolling velocity, which varies from that of the Hertzian surface stress distribution. The tooth flanks slide and roll on one another and as a result a frictional force in the tangential direction is created. In addition, there exist residual stresses on and below the tooth flank surface. However, the maximum effective Hertzian contact stress has proved to be a useful theoretical criterium.

Stresses below the surface of tooth flanks, with due consideration to the above mentioned influences, can be computed. [1,2]. They can be treated as an equivalent stress according to maximum distortion energy theory. Fig. 1 shows the relation between calculated equivalent stress at the inner single contact point of pinion and the depth "x" from the tooth surface. The maximum equivalent stress $\sigma_{vmax} \approx 0.56\ \sigma_H$ lies approximately at a depth $x \approx 0.68$ s whereby σ_H is the maximum Hertzian contact stress and 2 s the flattening width.

A sub-surface fatigue strength distribution that has a form closely matching that of the stress distribution below the surface, is obtained with gears of case hardening steel of low carbon content. By carburizing the tooth surface and hardening, a hardened case of martensite is obtained while the core of the tooth remains soft and ductile. Correct heat treatment and a proper depth of hardness increases the root fatigue strength. Finally, grinding the tooth flanks gives a good tooth quality. There will be no strength reducing notch effect when grinding the tooth root, if the teeth of the gears are protuberance-hobbed before carburizing and hardening.[3]

Fig. 2 shows the Vickers hardness curve and the related fatigue strength σ_{sch} for repeated load cycles below the tooth surface of a gear manufactured from 17 CrNiMo 6 steel. Due to the hardened casing the equivalent stress at any depth "x" is less than the fatigue strength for repeated (non-alternating) loads i.e. $\sigma_v < \sigma_{sch}$.

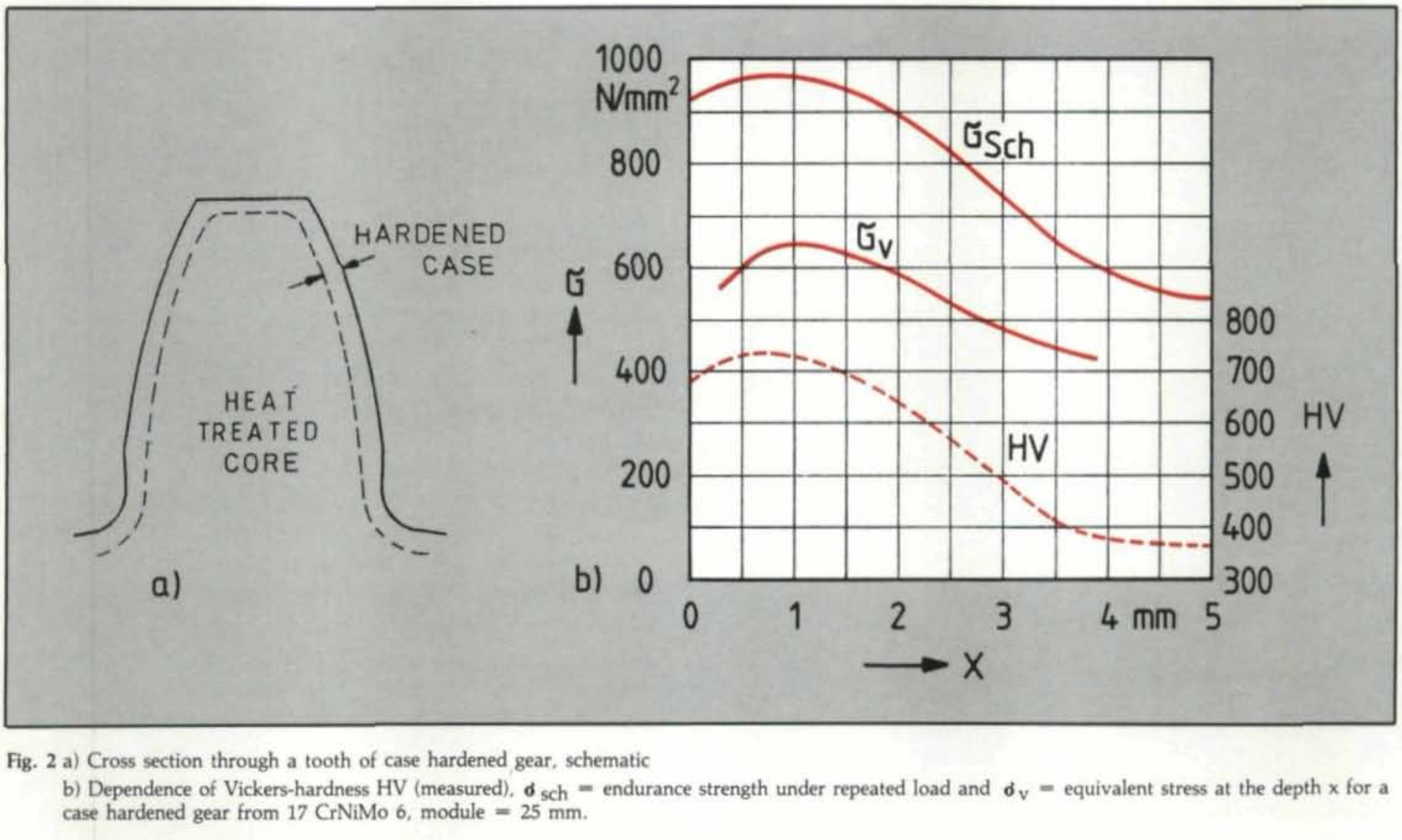

Fig. 2 a) Cross section through a tooth of case hardened gear, schematic

b) Dependence of Vickers-hardness HV (measured), σ_{sch} = endurance strength under repeated load and σ_v = equivalent stress at the depth x for a case hardened gear from 17 CrNiMo 6, module = 25 mm.

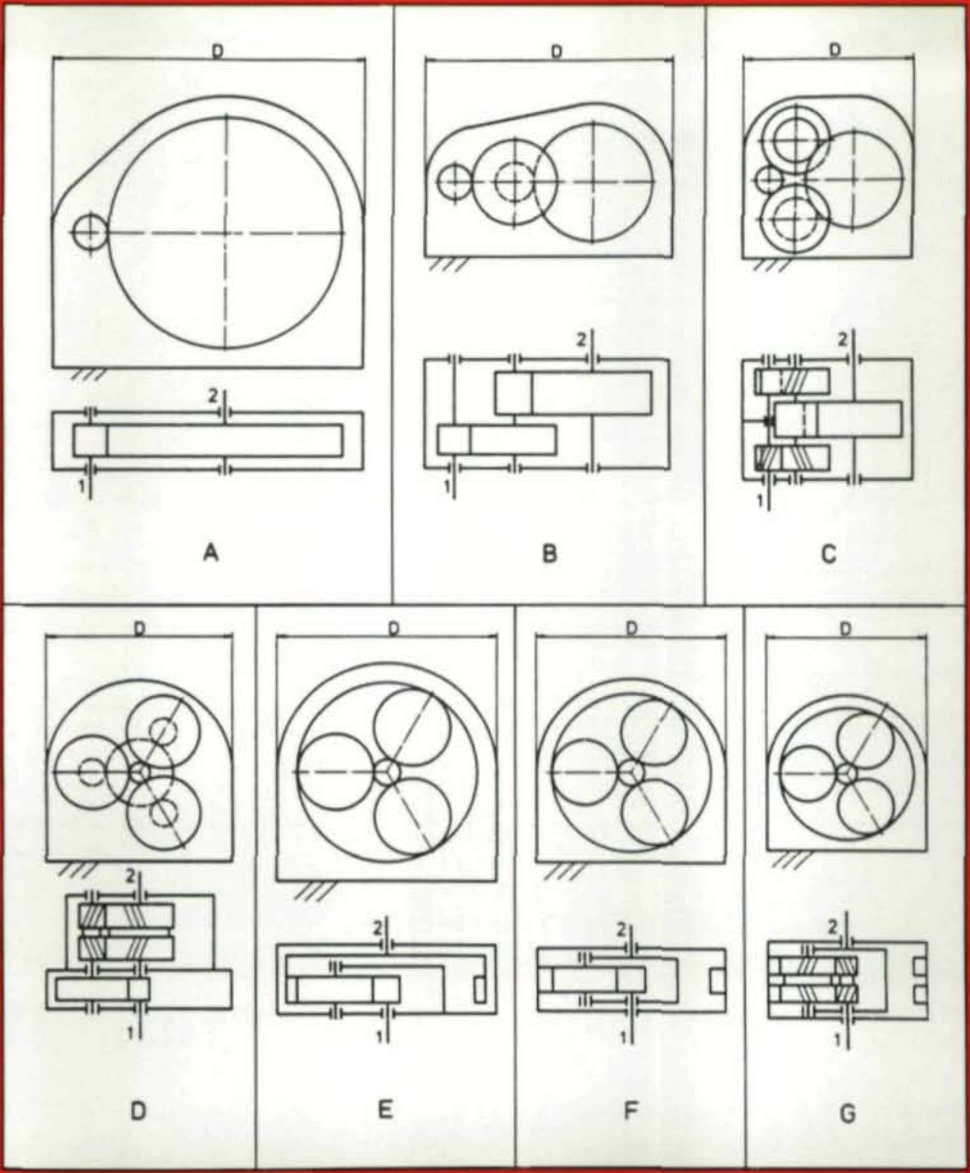

Fig. 3 – Schematic diagrams and size comparison of gear units with same shaft torques and transmission ratio $i = 7$ with and without power branching

Gear units with case hardened and ground gears from case-hardening steel have the following advantages over gear units with heat treated steel and manufactured by cutting tools:

- more compact in size and lighter in weight thus reducing manufacturing costs
- higher wear-resistance and a lower susceptibility to shock
- higher operating reliability through higher and more balanced capacity of the tooth root and the tooth flank
- lower rolling velocities and a lower tooth meshing frequency
- lower internal dynamic additional forces and a reduced noise level due to better tooth quality
- increased efficiency at part and full load operating conditions.

Case hardening and grinding of tooth flanks to increase the tooth flank capacity of gears has proved its usefulness, and has been successfully applied for several decades on small gears in Automobile Industries and for the last years on large gears for industrial gear units. Computation according to DIN 3990 or ISO/DP 3663, based on the experimental and theoretical investigations of is a safe method, pre-calculation of the capacity of case hardened gears.

Size, Weight and Efficiency

Fig. 3 shows schematically, gear units with and without power branching. The diameter ratio of the gears corresponds to an overall transmission ratio of 7. Shafts 1 and 2 are respectively the high and low speed shafts. The gear units A, B and C have parallel shafts and gear units D, E, F and G coaxial shafts.

Gear units A and B, respectively, are a single stage and a two stage unit. Neither has power branching.

Gear units C, D, E, F and G all have two stages and power branching. The gears on the intermediate shaft of the gear units C and D have different diameters, however, the intermediate gears on one intermediate shaft of E, F and G have been reduced to a single gear, hence, the latter are treated as single stage units.

Gear unit C has two way power branching. Equal power

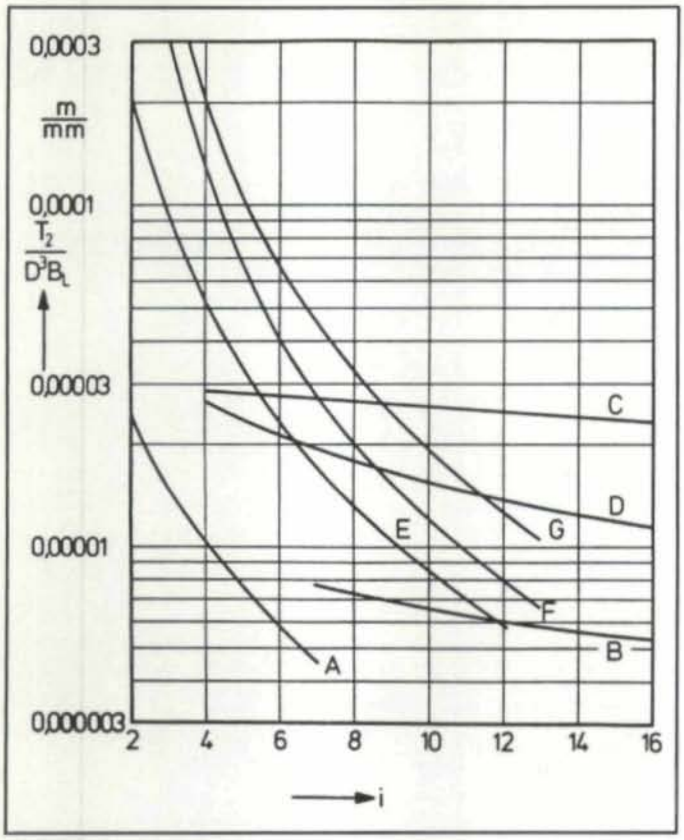

Fig. 4 – Dependence of relative (to size) torque of gear units according to Fig. 3 on the transmission ratio i. T_2 = torque of shaft 2 in Nm; D = design length or diameter in m; B_L = load value in N/mm^2

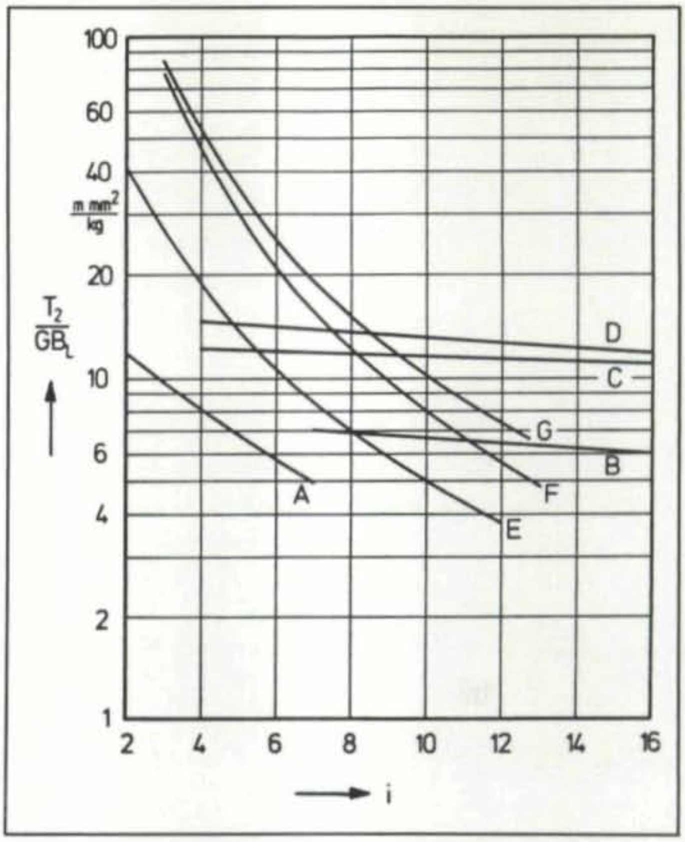

Fig. 5 – Dependence of relative (to weight) torque of gear units according to Fig. 3 on the transmission ratio i. T_2 = torque of shaft 2 in Nm; G = gear unit weight in kg; B_L = load value in N/mm^2

distribution is achieved by the herringbone gears of the high speed stage and the axial positioning of shaft 1.

In the case of gear unit D, the power of the high speed stage is equally distributed to three intermediate gears by the radial positioning of the small central gear on shaft 1. The power in the low speed stage is equally distributed through the three herringbone gears by the axial positioning of the three intermediate shafts.

To achieve equal power distribution on the three intermediate gears in the case of E, F and G, the small central gear on the shaft 1 of large units should be radially positionable.

The large outer gear is an internal gear and is connected to shaft 2, in the case of E, and to the gear unit casing in the case of F and G respectively. Gear units F and G both have the planet carrier and shaft 2 forming one unit.

The intermediate gears rotate about the central axis as planets. Herringbone gears and axial positioning of the intermediate gears give an equal power distribution through six branches in the case of gear unit G.

Power branching influences the size and weight of a gear unit. Figs. 4 and 5 show these relationships as a function of the total transmission ratio

$$i = \frac{n_1}{n_2}\ . \qquad (1)$$

n_1 and n_2 are the rotational speeds of shafts 1 and 2 respectively.

Figs. 4 and 5 have plotted T_2/D^3B_L and T_2/GB_L against the transmission ratio i whereby T_2 = torque of low speed shaft 2, D = gear unit size, G = gear unit weight and B_L = load value. B_L is calculated from the following equation.[4]

$$B_L = \frac{Fu}{d \bullet b} \qquad (2)$$

with F_u = tangenital force, d = diameter of pinion and b = facewidth.

Equation (3) gives the allowable load value $B_L{}^*$ approximately.

$$B_{L^*} = \frac{f_W}{K_A} B_o \qquad (3)$$

K_A is the application factor and f_W is the load factor. With repeated load on the gears $f_W = 1$ and with alternating load $f_W = 0.7$.

Gear units E, F and G have intermediate gears with an alternating load. For gear units dimensioned for an infinite fatigue life the value of B_o is approximately 4 . . . 5 N/mm^2 for case hardened gears and 1 . . . 1.3 N/mm^2 for through hardened gears. Gear size D and gear weight G can be approximately determined for known values of torque T_2 with the help of Figs. 4 and 5 and equations (2) and (3).

The gear weight G includes the weight of solid gears, shafts and case. The gear case forms shown schematically in Fig. 3, wall thicknesses = 0.02 D and the density of steel are the basis for the weight calculation.

Gear units G and F have the largest relative torques for low transmission ratios and, therefore, are preferred. A transmission ratio limit of $i \approx 8$ is obtained with $f_W = 0.7$ for gear units G and F.

The relative torque of units G and F can be favourable even for higher transmission ratios when two gear units are connected in series. For example, two F type gear units having transmission ratios $i_1 = i_2 = 4$ and are connected in series. The total transmission ratio is $i = i_1 \bullet i_2 = 16$. The total weight of these two units is about 1.8 times less than a single D type gear unit with $i = 16$.

Fig. 6 shows the relation between the efficiency n of gear units according to Fig. 3, and the transmission ratio i according to equation (1). Only gear tooth meshing losses are considered since this loss is significantly larger than all the other losses for gear units under full loading.

For the one stage gear unit, A the efficiency corresponds to the efficiency n_z of one gear meshing, i.e.

$$n = n_Z . \tag{4}$$

But in the case of B, C, D and E the total efficiency is the product of the slow and high speed stage efficiencies, i.e.

$$n = n_{Z1}\, n_{Z2}. \tag{5}$$

Gear units F and G transmit part of the power directly without any loss. Here, the efficiency with shaft 1 as the input shaft is

$$n = \frac{1}{i} \{1 + (i - 1)\, n_{Z1}\, n_{Z2}\}. \tag{6}$$

The tooth meshing efficiency n_Z of a gear pair is obtained from the following equation

$$n_Z = 1 - f_Z\, \mu_Z. \tag{7}$$

The geometrical factor f_Z is calculated[5] from equation (8) with the numbers of teeth z_1 and z_2 respectively of pinion 1 and gear 2

$$f_Z = \pi \left(\frac{1}{z_1} + \frac{1}{z_2} \right) \quad (E_1 + E_2) . \tag{8}$$

z_2 is negative for internal gears. The values for E1 and E2 are given by partial transverse contact ratios ϵ_1 and ϵ_2 of pinion 1 and gear 2 respectively.

$$E_{1,2} = 0.5 - \epsilon_{1,2} + \epsilon^2_{1,2} \quad \text{for } 0 \leqq \epsilon_{1,2} \leqq 1 \tag{9}$$

$$E_{1,2} = \epsilon_{1,2} - 0.5 \quad \text{for } \epsilon_{1,2} > 1 \tag{10}$$

$$E_{1,2} = 0.5 - \epsilon_{1,2} \quad \text{for } \epsilon_{1,2} < 0 \tag{11}$$

The equations (9) to (11) derived for spur gearing in[6] are also approximately valid for helical gearing.

It is assumed that the tooth flank coefficient of friction $\mu_Z = 0.06$ for all the tooth meashings of the gear units in Fig. 3. Internal gears enhance the formation of a good elastohydrodynamic oil film due to the complementary profiles of the meshing flanks.

However, the sliding properties of internal gearings are poorer due to higher tooth flank roughness and, thus, the assumption of the same coefficient of friction for external and internal gearing approximately holds good. The gearing geometry for Fig. 6 is according to DIN 3960 for gears having no addendum modification and number of teeth of the pinion $z_1 = 17$ for all the gear meshings.

Gear unit A with a single stage has the best efficiency, see Fig. 6. Since there are two gear meshings in gear units B, C, D, E, F and G, the efficiency curves lie below that of case A. In the case of gear units E, F and G, the internal gear provides a favourable geometrical factor f_Z, thus, a better efficiency compared to gear units B, C and D which have only external gearings. The power component transmitted without loss for gear units F and G gives a further improvement in efficiency.

Planetary gear units F and G, in view of their size, weight and efficiency, are the most favourable choice. Considering the high manufacturing cost to obtain a good tooth surface finish on internal gears, planetary gear units have a reduced advantage over gear units with power branching and external gears only.

Design of Large Gear Units

Three large gear units with power branching, which have proved to be useful in practice, are dealt with here. The rolling mill drive shown in Fig. 7 has a two way power branch similar to gear unit C of Fig. 3. It has four stages to attain

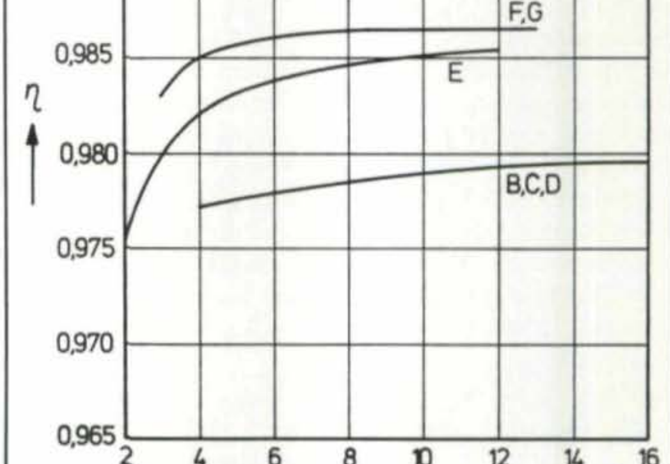

Fig. 6 – Dependence of efficiency η on the transmission ratio i for gear units according to Fig. 3.

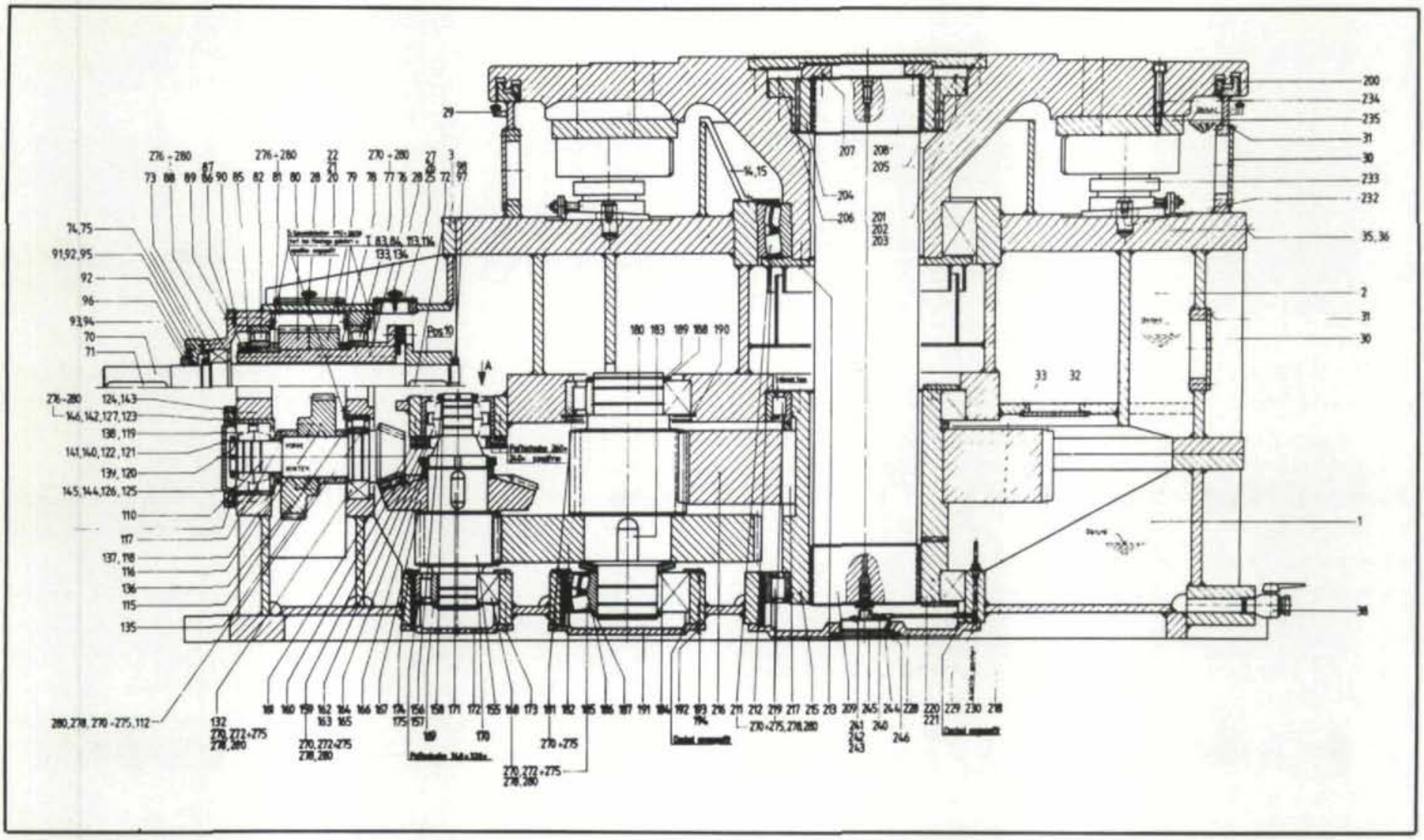

Fig. 7—Rolling mill gear unit with two way power branching (see scheme in Fig. 8). Power $P = 1350$ kW; shaft speeds $n_1 = 980$ min^{-1}, $n_2 = 24.5$ min^{-1}. Total weight including flange disk $G = 82$ t. (Flender-Kienast-Conception)

the high transmission ratio, whereby, the second stage is a bevel gear stage as the output shaft is in the vertical position. Fig. 8 shows the gearing system.

The input shaft 1 is connected to an axially free hollow shaft 2 via an axially moveable coupling made of steel lamellas. Herringbone gear 4, on the hollow shaft, branches the power equally due to its axial positioning. Bevel gears 5 and 6, with cyclo-palloid-spiral teeth and longitudinally crowned flanks, are case hardened and are cut after heat treatment "High Power Gear-teeth" (HPG). The power is transmitted onward to the output shaft via the helical gears 7, 8, 9 and 10. All the helical gears are case hardened, and the tooth flanks are ground. A stub shaft, with coupling teeth at the shaft end, connects the output gear with the flange disk, shown in Fig. 7.

Hydrostatically lubricated axial sliding bearings with tilting segments carry the large forces from the milling operation and the weight of the output shaft, flange disk and mill pan (not shown) and transmits the same to a strongly designed gear case. Each sliding bearing segment has a temperature feeler gauge to monitor bearing load. No inspection during operation is necessary due to an adequate oil supply.

Fig. 9 shows the central gear unit of a tube mill. The gear unit has, in the first stage, no power branching and has two input shafts. One of the two input shafts serves as an auxiliary drive, and is driven at a greatly reduced speed via an overrunning clutch and auxiliary gear unit (not shown). When the main input shaft is driving at a higher speed, the auxiliary drive will be disengaged due to the overrunning clutch. Both the main stages of the gear unit in Fig. 9 distribute the power similar to gear unit D in Fig. 3. The first main stage with spur gearing branches the power three ways. Equal power branching is achieved by the radial positioning of a central pinion on a shaft designed for elastic deflection. The second main stage, with the herringbone gear, branches out the power six times. The eccentric intermediate shaft has axial freedom ensuring their equal loading.

The large intermediate gears belonging to the first main stage are mounted onto the intermediate shaft by oil hydraulic shrink fitting. Thus, an exact alignment of the gearing of the

Fig. 8—Gearing scheme of rolling mill gear drive shown in Fig. 7. 1 = input shaft; 2 = hollow shaft; 3 = steel lamella disk coupling; 4 = herringbone gear; 5, 6 = bevel gear pair; 7, 8, 9 = spur gears; 10 = gear on output shaft

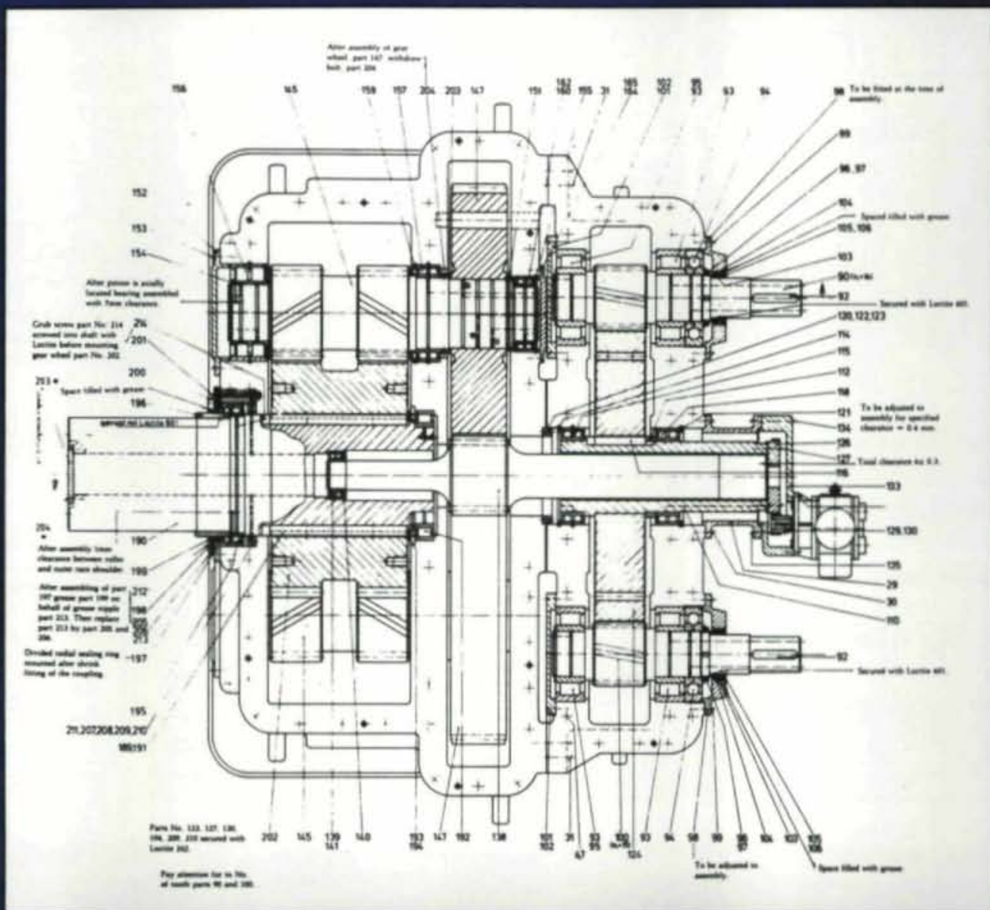

Fig. 9 – Tube mill central gear unit with two power branching stages. Transmission power $P = 4500$ kW; shaft speeds $n_1 = 490$ min^{-1}; $n_2 = 12.6$ min^{-1}; gear unit weight $G = 100$ t

two main stages during assembly is possible requiring only a small radial displacement of the central pinion of the first main stage. Measurements have shown that the radial displacement is not more than 0.2 mm and the variation of torque distribution is 4% maximum.[7]

All the gears of the gear unit, shown in Fig. 9, are case hardened and ground. It is worth noting that the resultant radial forces on the output shaft are theoretically non-existant, thus, the output shaft has relatively small roller bearings.

A two-stage planetary gear unit for a ball mill drive is shown in Fig. 10. Each stage has spur gearing and three way power branching and corresponds to the gear unit F of Fig. 3 with the internal gear fixed to the gear case.

A uniform force distribution on the three planet gears is achieved by the free pivoting configuration of the central pinion of each of the two planet stages. The sun gear centers itself in such a way that the three radial meshing forces of the planets are equal, thus, ensuring an equal transmission of the tangential forces.

Due to the equilibrium conditions, the three meshing points of the planet gears with the annulus also have equal tangential and centering forces. The sun gear which is centered by the planet wheel teeth has radial freedom due to the backlash of the teeth and the double jointed clutch in each stage. The ring gears of both the stages are of quenched and tempered steel. They are hobbed or cut and stress relieved after the final machining process.

Summary

Gear units, with case hardened gears and internal power branching, are relatively small in size and weight. This type is particularly suited for large gear units to transmit high powers. Case hardened and ground gears, due to their higher flank wear resistance and strength, give a higher load carrying capacity than through hardened gears. The strength characteristic below the tooth flank surface of a case hardened gear matches well with the stress characteristic.

(continued on page 48)

DEFINITIONS OF GEAR ELEMENTS

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As a means of identification, the various elements of gears and gear teeth have been given certain names which serve to classify and explain them. While some of the terms used are more or less self-explanatory, others are not. There is also some confusion in connection with the application of certain terms. The American Gear Manufacturers Association have adopted standard terms and definitions. This list, as it applies exclusively to involute gearing, is here arranged in alphabetical order. Most of these terms are also graphically presented in Figs. 1 and 2. The numbers in parentheses indicate the location of the identifying illustrations.

Addendum – is the height by which a tooth projects beyond the pitch circle or pitch line; also, the radial distance between the pitch circle and the addendum circle. (1).

Addendum, Chordal – is the height from the top of the tooth to the chord subtendin the circular thickness arc. (14).

Addendum, Normal Chordal – is the chordal addendum in the plane normal to the helix, or the tooth curve at the center of the tooth. (31).

Angle, Axial Pressure – is the pressure angle in the axial plane of a helical tooth. (30).

Angle, Base Helix – is the helix angle on the base cylinder of involute helical teeth. (32).

Angle, Base Lead – is the lead angle on the base cylinder. (33).

Angle, Helix – is the angle between any helix and an element of its cylinder. In helical gears and worms, it is at the pitch diameter, unless otherwise specified. (28).

Angle, Lead – is the angle between any helix and a plane of rotation. It is the complement of the helix angle, and is used for convenience on worms. It is understood to be at the pitch diameter unless otherwise specified. (28).

Angle, Normal Pressure – is the pressure angle in the normal plane of a helical tooth. (30).

Angle, Outside Helix – is the helix angle on the outside cylinder. (32).

Angle, Outside Lead – is the lead angle on the outside cylinder. (33).

Angle, Pitch Helix – is the helix angle on the pitch cylinder. (32).

Angle, Pitch Lead – is the lead angle on the pitch cylinder. (33).

Angle, Pressure – is the angle between a tooth profile and the radial line at its pitch point, or the angle between the line of action and the line tangent to the pitch circle. *Standard* Pressure Angles are established in connection with *standard* gear-tooth proportions. A given pair of involute profiles will transmit smooth motion at the same velocity ratio even when the center distance is changed. Changes in center distance, however, in gear design and manufacturing operations are accompanied by changes in pitch diameter, pitch and pressure angle. Different values of pitch diameter and pressure angle, therefore, may occur in the same gear under different conditions. Usually in gear design, and unless otherwise specified, the pressure angle is the *standard* pressure angle at the *standard* pitch diameter, and is standard for the cutter used to generate the teeth. The *Operating* Pressure Angle is determined by the center distance at which a pair of gears operates. The *Generating* Pressure Angle is the angle at the pitch diameter in effect when the gear is generated. Other pressure angles may be considered in gear calculations. In gear-cutting tools, the pressure angle indicates the direction of the cutting edge as referred to some particular direction. In helical gears, the pressure angle, may be specified in the transverse, normal or axial planes. For spur gears in which only one direction of cross section needs to be considered, the general term Pressure Angle may be used without qualification. (5).

Angle, Transverse Pressure – is the pressure angle in the transverse plane, or plane of rotation. (5).

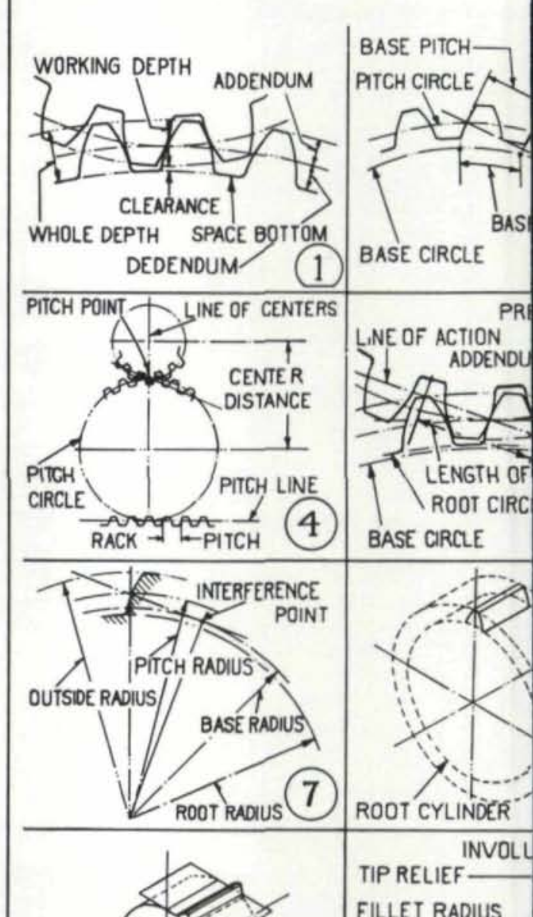

Fig. 1—Diagram Identifying Various Elem

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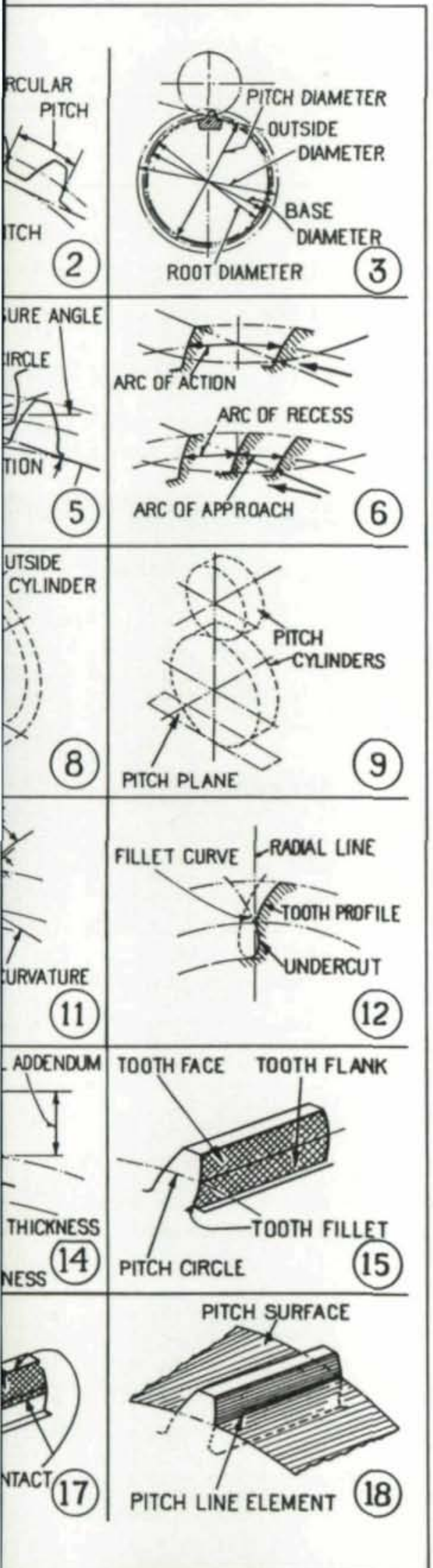

ts of Involute Spur and Helical Gears.

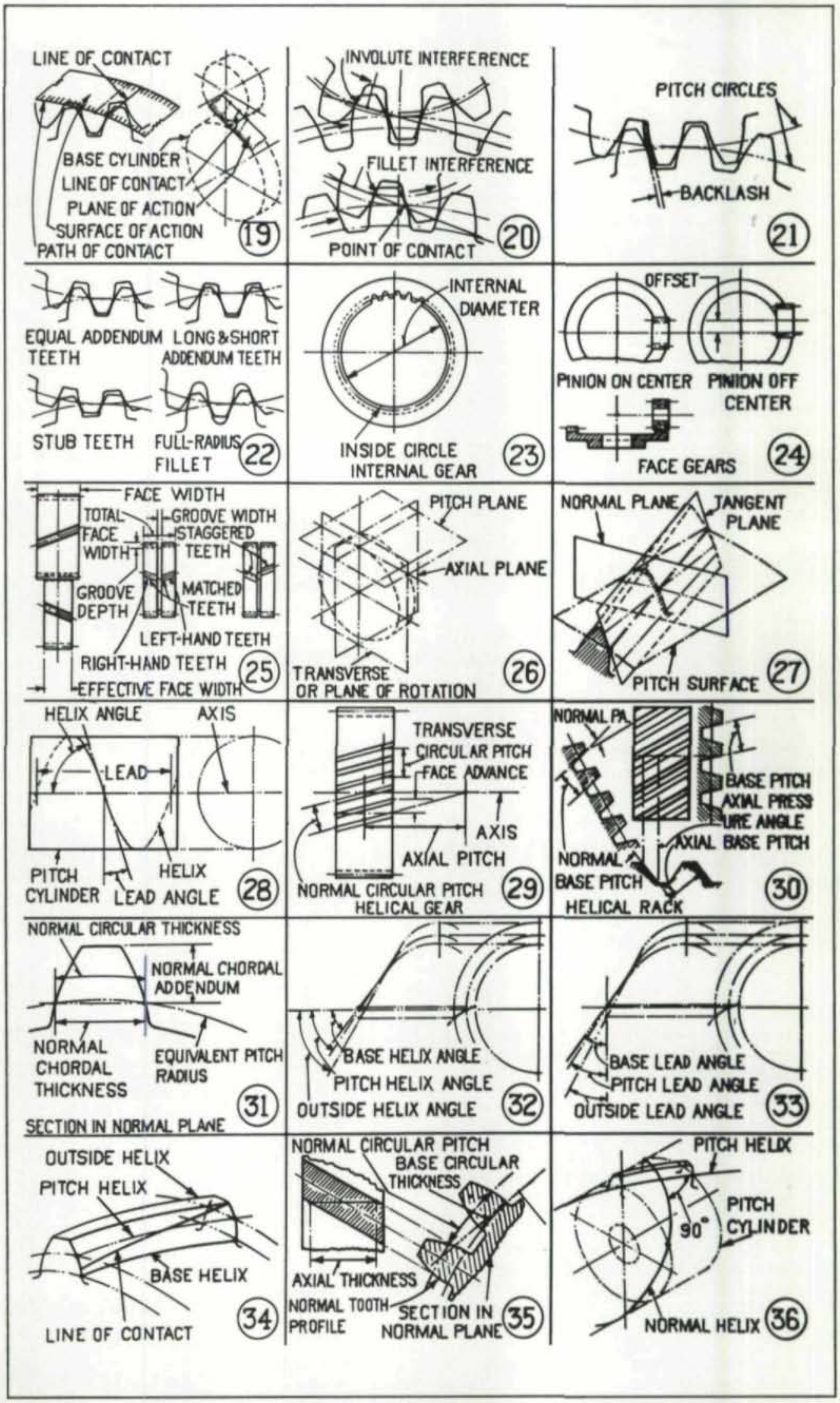

Fig. 2—Diagam Identifying Various Elements of Involute Spur and Helical Gears.

Arc of Action – is the arc of the pitch circle through which a tooth profile moves from the beginning to the end of contact with a mating profile. (6).

Arc of Approach – is the arc of the pitch circle through which a tooth profile moves from its beginning of contact until the point of contact arrives at the pitch point. (6).

Arc of Recess – is the arc of the pitch circle through which a tooth profile moves from contact at the pitch point until contact ends. (6).

Backlash – is 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 variously in the transverse, normal, or axial planes, and either in the pitch circles, or on the line of action. Such measurements should be corrected to corresponding values on transverse pitch circles for general comparisons. (21).

Bottom Land – is the surface at the bottom of the tooth space adjoining the fillet. (13).

Center Distance – is the distance between parallel axes of spur gears, and parallel axes helical gears, or the crossed axes of crossed helical gears and worm gears. Also it is the distance between the pitch circles. (4).

Circle, Addendum – is the circle which coincide with the tops of the teeth in a cross section. (5).

Circle, Base – is the circle from which involute tooth profiles are derived. (2).

Circle, Pitch – is the curve of intersection of a pitch surface of revolution and a plane of rotation. According to theory it is the imaginary circle that rolls without slipping with a pitch circle of a mating gear. (2).

Circle, Root – is the circle that is tangent to the bottoms of the tooth spaces in a cross section. (5).

Clearance – is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear. (1).

Contact, Point – is any point at which two tooth profiles touch each other. (20).

Contact, Zone – is that portion of the line of action bounded by the "natural" interference points.

Cylinder, Base – is the cylinder which corresponds to the base circle, and is the cylinder from which involute tooth surfaces, either straight or helical are derived. (10).

Cylinder, Inside – is the surface that coincides the tops of the teeth of an internal cylindrical gear. (23).

Cylinder, Outside – is the surface that coincides the tops of the teeth of an external cylindrical gear. (8).

Cylinder, Pitch – is the imaginary cylinder in a gear that rolls without slipping on a pitch cylinder or pitch plane of another gear or a rack. (9).

Cylinder, Root – is the imaginary cylinder tangent to the bottoms of the tooth spaces in a cylindrical gear. (8).

Dedendum – is the depth of a tooth space below the pitch circle or pitch line; also the radial distance between the pitch circle and the root circle. (1).

Depth, Whole – is the total depth of a tooth space, equal to addendum plus dedendum; also equal to working depth plus clearance. (1).

Depth, Working – is the depth of engagement of two mating gears – that is, the sum of their addendums. (1).

Diameter, Base – is the diameter of the base cylinder of an involute gear. (3).

Diameter, Internal – is the diameter of the addendum circle of an internal gear. (23).

Diameter, Outside – is the diameter of the addendum (outside) circle. (3).

Diameter, Pitch – is the diameter of the pitch circle. In parallel-shaft gears, the pitch diameter can be determined directly from the center distance and the number of teeth by proportionality. *Operating* Pitch Diameter is the pitch diameter at which the gears operate. *Generating* Pitch Diameter is the pitch diameter at which the gear is generated. (3).

Diameter, Root – is the diameter of the root circle. (3).

Face Advance – is the distance on a pitch circle of a helical gear tooth, or pitch line of a helical rack tooth through which a tooth moves from the position at which contact begins at one end of the tooth curve to the position when contact ceases at the other end. (29).

Face Width – is the length of the teeth in an axial plane. (25).

Face Width, Effective – is the portion that may actually come in contact with mating teeth, as occasionally one member of a pair of gears may have a greater face width than the other. (25).

Face Width, Total – is the actual dimension of a gear blank that exceeds the effective face width, or as in double helical (herringbone) gears where the total face width includes any distance separating right- and left-hand helices. (25).

Fillet Curve – is the concave portion of the tooth profile where it joins the bottom of the tooth space. (12).

Fillet Radius – is the radius of the fillet curve at the base of the gear tooth; this radius is an approximate radius of curvature. (11).

Fillet, Full Radius – is the arc of a circle at the bottom of a tooth space, the center of which arc is on the center line of the tooth space. (22).

Groove Depth – is the depth of the clearance groove between helices in double helical (herringbone) gears. (25).

Groove Width – is the width of the clearance groove between helices in double helical (herringbone) gears. (25).

Helix, Base – of a helical involute gear or worm is the intersection of the tooth surface with its base cylinder. (34).

Helix, Normal – is a helix on the pitch cylinder normal to the pitch helix. (36).

Helix, Outside – of an involute helical gear or worm is the intersection of a tooth surface and the outside cylinder. (34).

Helix, Pitch – is the curve of intersection of a tooth surface and its pitch cylinder in a helical gear. (34).

Interference – is contact between mating teeth at some other point than along the line of action. (20).

Interference, Fillet – is contact of mating teeth at some other point than on the line of action *inside* the zone of contact. (20).

Interference, Involute — is contact of mating teeth at some other point than on the line of action outside the zone of contact. (20).

Lead — is the axial advance of a helix for one complete turn, as in the threads of cylindrical worms and teeth of helical gears. (28).

Length of Action — is the distance on an involute line of action through which the point of contact moves during the action of the tooth profiles. (5).

Line of Action — is the path of contact in involute gears. It is a straight line passing through the pitch point and tangent to the base circles. (5).

Line of Centers — is the line which connects the centers of the pitch circles of two engaging gears, it is also the common perpendicular of the axes in crossed helical gears. When one of the gears is a rack, the line of centers is perpendicular to its pitch line. (4).

Line of Contact — is the line or curve along which two tooth surfaces are tangent to each other. (19).

Module (inches) — is the ratio of the pitch diameter in inches to the number of teeth. It is the reciprocal of the diametral pitch.

Module (millimeters) — is the ratio of the pitch diameter in millimeters to the number of teeth.

Offset — is the perpendicular distance between the axes of offset face gears. (24).

Pitch — is the distance between similar, equally-spaced tooth surfaces, in a given direction along a given line or curve. The single word "Pitch" without qualification has been used to designate circular pitch, axial pitch and diametral pitch, but such confusing usage should be avoided. (4).

Pitch Line — corresponds in the cross section of a rack to the pitch circle in the cross section of a gear. (4).

Pitch Line Element — is a line curved or straight formed by the intersection of the pitch surface and the tooth surface. (18).

Pitch Point — is the point of tangency of two pitch circles (or a pitch circle and a pitch line) and is on the line of centers. The pitch point of a tooth profile is at its intersection with the pitch circle. (4).

Pitch, Axial — is the circular pitch in the axial plane and in the pitch surface between corresponding sides of adjacent teeth in helical gears. The term axial pitch is preferred to the term linear pitch. (29).

Pitch, Axial Base — is the base pitch of helical involute tooth surfaces in an axial plane. (30).

Pitch, Base — in an involute gear is the pitch on the base circle or along the line of action. Corresponding sides of involute gear teeth are parallel curves, and the base pitch is the constant and fundamental distance between them along a common normal in a plane of rotation. (2).

Pitch, Circular — is the distance along the pitch circle or pitch line between corresponding profiles of adjacent teeth. (2).

Pitch, Diametral — is the ratio of the number of teeth to the number of inches in the pitch diameter. There is a fixed relationship between diametral pitch and circular pitch.

Pitch, Normal Base — in an involute helical gear is the base pitch in the normal plane. It is the normal distance between parallel helical involute surfaces on the line of action in the normal plane, or is the length of arc on the normal base helix. It is a constant distance in any helical involute gear. (30).

Pitch, Normal Circular — is the circular pitch in the normal plane, and also the length of the arc along the normal helix between helical teeth or threads. (35).

Pitch, Normal Diametral — is the diametral pitch as calculated in the normal plane.

Pitch, Transverse Circular — is the circular pitch in the transverse plane, or plane of rotation. (29.)

Plane of Action — is the surface of action in involute parallel-axis gears with either straight or helical teeth. It is tangent to the base cylinders. (19.)

Plane of rotation — is any plane perpendicular to a gear axis. (26).

Plane, Axial — of a pair of gears is the plane that contains the two axes. In a single gear, an axial plane may be any plane containing the axis and a given point. (26.)

Reference Planes — are: pitch plane, axial plane, and transverse plane, all intersecting at a point and mutually perpendicular. (26), (27).

Plane, Normal — is normal to the tooth surface at a point of contact, and perpendicular to the pitch plane. (27).

Plane, Pitch — of a pair of gears is the plane perpendicular to the axial plane and tangent to the pitch surfaces. A pitch plane in an individual gear may be any plane tangent to its pitch surface. The pitch plane of a rack or crown gear is the pitch surface. (26).

Plane, Tangent — is tangent to the tooth surfaces at a point or line of contact. (27).

Plane, Transverse — is perpendicular to the axial plane and to the pitch plane. In gears with parallel axes, the transverse plane and plane of rotation coincide. (26).

Radius, Base — is the radius of the base circle of involute profiles. (7).

Radius, Equivalent Pitch — is the radius of the pitch circle in a cross section of gear teeth in any plane other than the plane of rotation. It is properly the radius of curvature of the pitch surface in the given cross section. An example is the normal section of helical teeth. (31).

Radius, Outside — is the radius of the addendum circle of an external gear. (7).

Radius, Pitch — is the radius of the pitch circle. (7).

Radius, Curvature or Profile — is the radius of curvature of a tooth profile, usually at the pitch point or a point of contact. (11).

Radius, Root — is the radius of the root circle. (7).

Ratio, Axial Contact — is the ratio of the face width to the axial pitch in helical gears.

Ratio, Contact — is the ratio of the arc of action to the circular pitch, and sometimes is thought of as an average number of teeth in contact. For involute gears, the contact ratio is obtained most directly as the ratio of the length of action to the base pitch.

Ratio, Face Contact — is the ratio of the face advance to the circular pitch, usually having the same value as axial contact ratio.

Ratio, Gear — is the ratio of the larger to the smaller number of teeth in a pair of gears.

Ratio, Normal Contact — is the contact ratio in the normal section.

Ratio, Total Contact — is the sum of the transverse contact ratio and axial contact ratio which may be thought of as the average total number of teeth in contact in parallel helical gears.

Ratio, Transverse — is the contact ratio in the transverse plane, or plane of rotation.

Space Bottom — is a line joining two fillets of adjacent tooth profiles in the same plane. (1).

Surface of Action — is the imaginary surface made up of all positions of the lines of contact of a given tooth surface. (19).

Surfaces, Pitch — are the imaginary planes or cylinders that roll together without slipping. For a constant velocity ratio, the pitch cylinders are circular. Sometimes, however, the velocity ratio may be variable, in which case different forms of pitch surfaces may occur, as for instance elliptical surfaces. (18), (27).

Teeth, Equal-Addendum — are those in which two engaging gears have the same addendums. (22).

Teeth, Equivalent Number of — is the number of teeth contained in the whole circumference of a pitch circle corresponding to an equivalent pitch radius.

Teeth, Full-Depth — are those in which the working depth equals: $\frac{2.000 \text{ inch}}{\text{diametral pitch}}$ (22).

Teeth, Involute — of spur gears, helical gears, and worms, are those in which the active portion of the profile in the transverse plane, is the involute of a circle.

Teeth, Left-Hand — are helical gear teeth or worm threads in which the teeth twist counterclockwise as they recede from the observer looking along the axis. (25).

Teeth, Long and Short Addendum — are those in which the addendums of two engaging gears are unequal. (22).

Teeth, Matched — are double helical (herringbone) gear teeth, the pitch line elements of which intersect, or would intersect at the center of the groove if prolonged. (25).

Teeth, Number of (or Threads) — is the number of teeth contained in the whole circumference of the pitch circle.

Teeth, Right-Hand — are helical gear teeth or worm threads in which the teeth twist clockwise as they recede from the observer looking along the axis. (25).

Teeth, Staggered — are double helical (herringbone) gear teeth, the pitch line elements of which do not intersect or would not intersect at the center of the groove, if prolonged. (25).

Teeth, Stub — are those in which the working depth is less than: $\frac{2.000 \text{ inch}}{\text{diametral pitch}}$ (22).

Tip Relief — is an arbitrary modification of a tooth profile whereby a small amount of material is removed near the tip of the gar tooth. (11).

Tooth Bearing — is that portion of the tooth surface which actually comes in contact with a mating tooth surface. (17).

Tooth Chamfer — is the beveled edge between the end of a tooth and the tooth surface to break the sharp edge. (17).

Tooth Face — is the surface between the pitch line element and the top of the tooth. (13) and (15).

Tooth Fillet — is the curved line joining the tooth flank and the bottom of the tooth space. (15).

Tooth Flank — is the surface between the pitch line element and the bottom land—it includes the fillet. (13) and (15).

Tooth, Normal Profile — is the outline formed by the intersection of a tooth surface and a plane perpendicular to the pitch line element. (35).

Tooth Profile — is one side of a tooth in cross section. Usually a profile is the curve of intersection of a tooth surface and a plane or surface normal to the pitch surface, such as the transverse, normal or axial plane. (12).

Tooth Surface — forms the side of a gear tooth. (16).

Tooth Top — is a line joining the outer ends of two adjacent tooth profiles in the same plane. In internal gearing it applies to the inner ends of the teeth. (13).

Tooth, Axial Thickness — in helical gears is the tooth thickness in the axial cross section at the pitch line. (35).

Tooth, Base Circular Thickness — in involute teeth is the length of arc on the base circle between the two involute curves forming the profiles of a tooth. (14).

Tooth, Circular Thickness — is the length of arc between two sides of a gear tooth on the pitch circle, unless otherwise specified. (14).

Tooth, Chordal Thickness — is the length of the chord subtending a circular-thickness arc. (14).

Tooth, Normal Chordal Thickness — is the chordal thickness in the plane normal to the pitch helix, or the tooth curve at the center of the tooth (31).

Tooth, Normal Circular Thickness — is the circular thickness in the normal plane. In helical gears, it is an arc of the normal helix. (31).

Tooth, Transverse Circular Thickness — is the circular thickness in the tranverse plane, or plane of rotation. See Circular Thickness. (14).

Top Land — is the surface of the top of the tooth. (13).

Undercut — is a condition in generated gear teeth when any part of the fillet curve lies inside of a line drawn tangent to the working profile at its lowest point. Undercut may be deliberately introduced to facilitate finishing operations. (12).

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E-5 ON READER REPLY CARD

Guest Editorial
(Continued from page 7)

design methods don't predict all failures, so we tend to be conservative in gear design or in instituting gear standards to cover all of the unknowns. Certainly we over design and over specify most of our products to cover those cases where our analyses or our materials fall down.

None of these problems will be easy to solve, but isn't it time that we redirected our research if we are going to solve them? A good research job requires real experimental data and data acquisition is expensive. No single company, nor single institution, private or public, has the funds today. It's tragic that the funds we do have are frittered away on peripheral projects, without attacking the areas of real need.

Kettering's famous quotation bears another repetition. "Secrecy in industrial research keeps out more good information than it keeps in."

Most gear manufacturers, and many large users, have data in their own fields of experience, which if analyzed and correlated with that of others, would provide a broader basis than we now have. The initiative must come from the industry, where the data, the judgement and the need all reside. We must find a way for those companies which have this data to share it to their own advantage.

It's time for the users of gears, gear manufacturers, and gear specifiers to show some leadership in gear research. The direction must come from those who will use the results, but there must be direct feedback of field experience which is pertinent to the research. When did you last talk to a mechanical engineering professor, a mechanical engineering graduate student, or an engineering class about gear research? How can they know the need if we don't tell them?

It's time to share our information with the universities and research institutions, and to make a united effort to fund the work required to improve our solutions to these problems.

Don McVittie
President, AGMA

NOTE: The opinions expressed herein are mine, not those of AGMA or any other organization.

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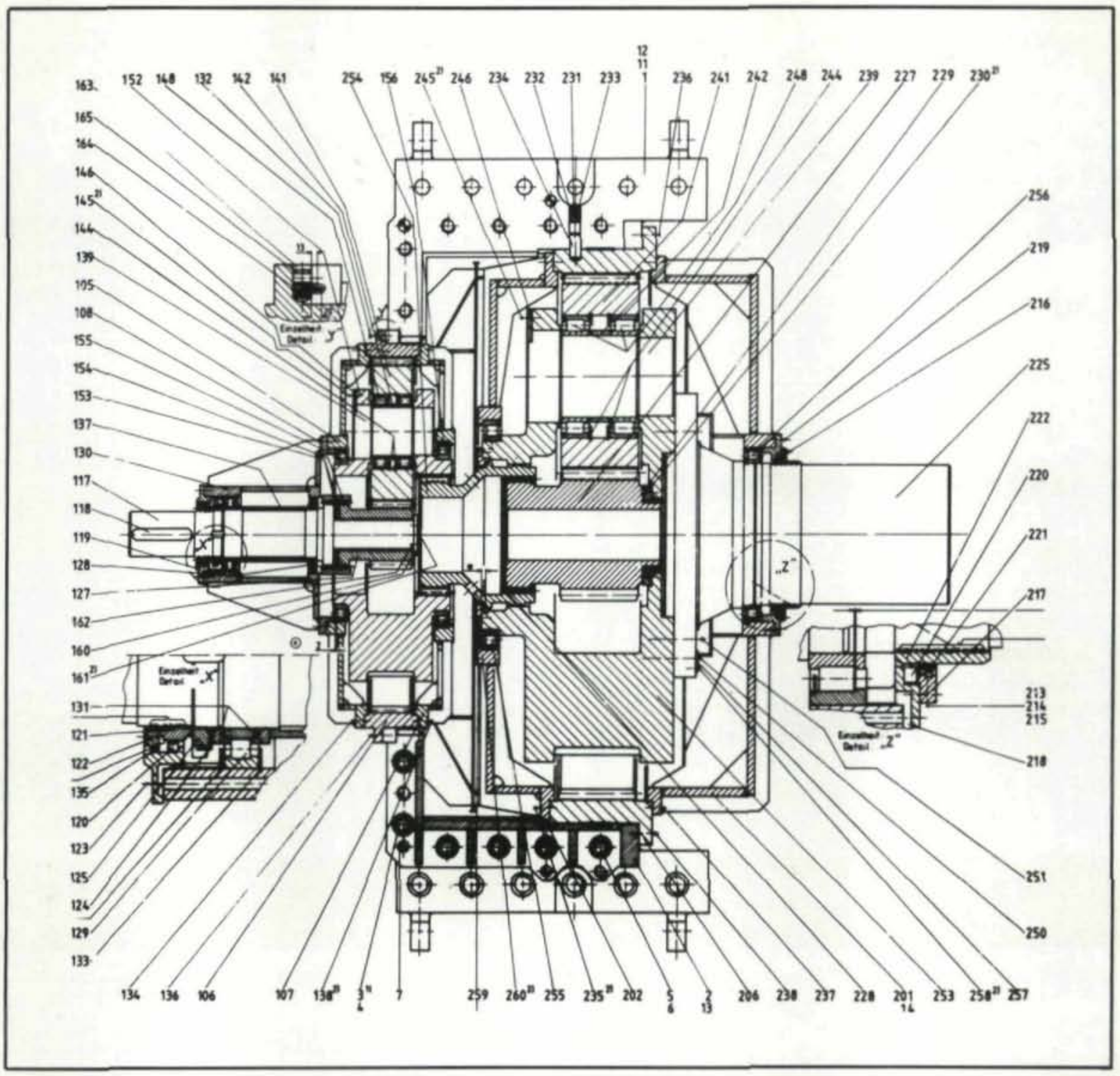

Fig. 10—Two stage planetary gear unit for ball mill drive with three way power branching per stage. Transmission power P = 4050 kW; shaft speeds n_1 = 485 min^{-1}; n_2 = 13.8 min^{-1}; gear unit weight G = 72 t

High Power Transmissions
(Continued from page 41)

A comparison of size and weight relative to output torque shows that among the power branching gear units planetary gear units are favourable and better especially for smaller transmission ratios of single planet stages. The relative torque decreases with increasing transmission ratio. For transmission ratios above i = 8 the power branching gear units having external gearing only are preferred. Their relative torque decreases at a lower rate with increasing transmission ratio. However, two or more planetary gears, coupled one after another, may still be more advantageous for higher transmission ratios above i = 15.

The efficiency of planetary gear units is comparatively better than that of other power branching gear units. However, for higher transmission ratios, one must couple more planetary gear units one after another and this advantage may be lost.

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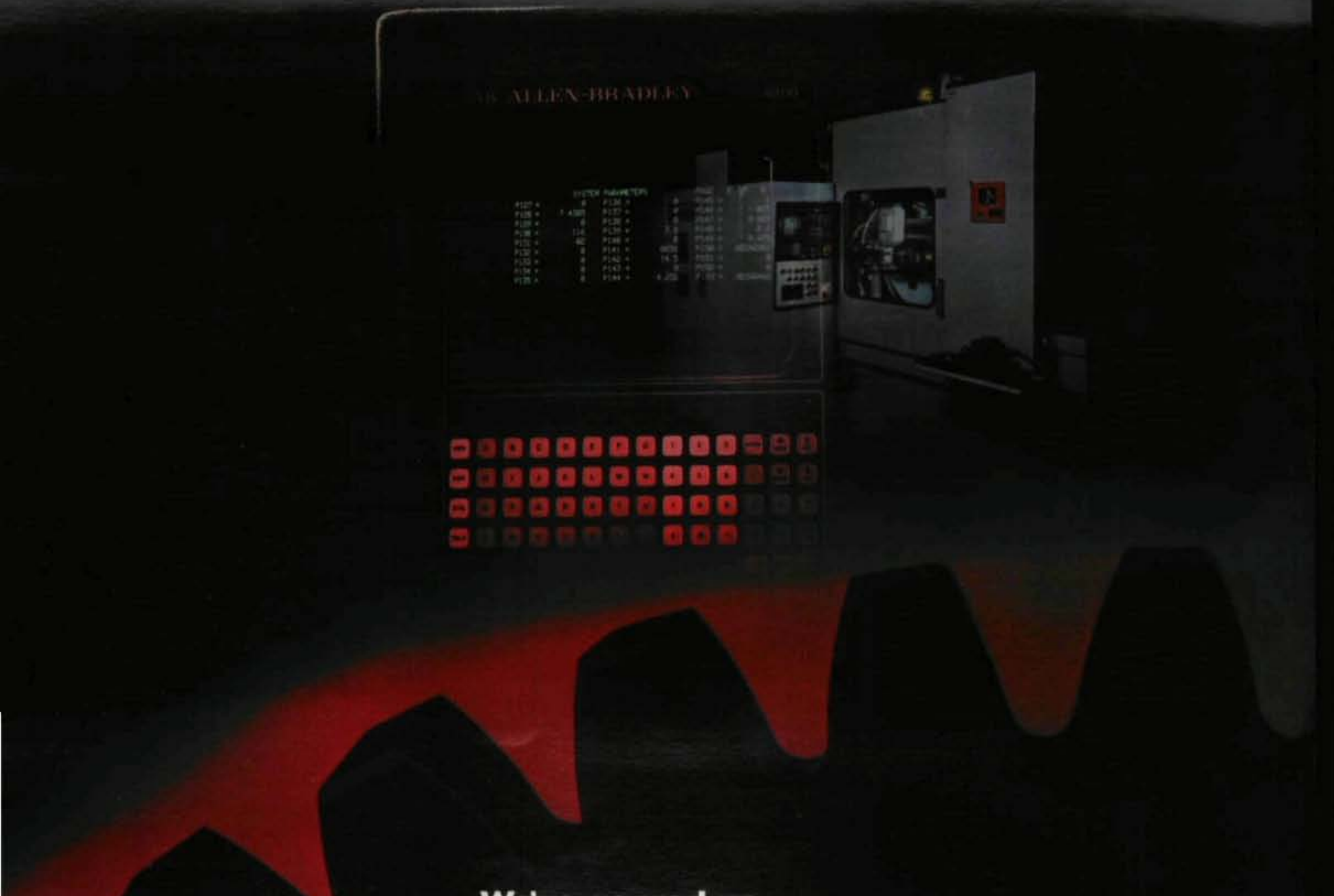
This article was reprinted from AGMA paper no. 429.05, Fall Technical Conference 1982.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