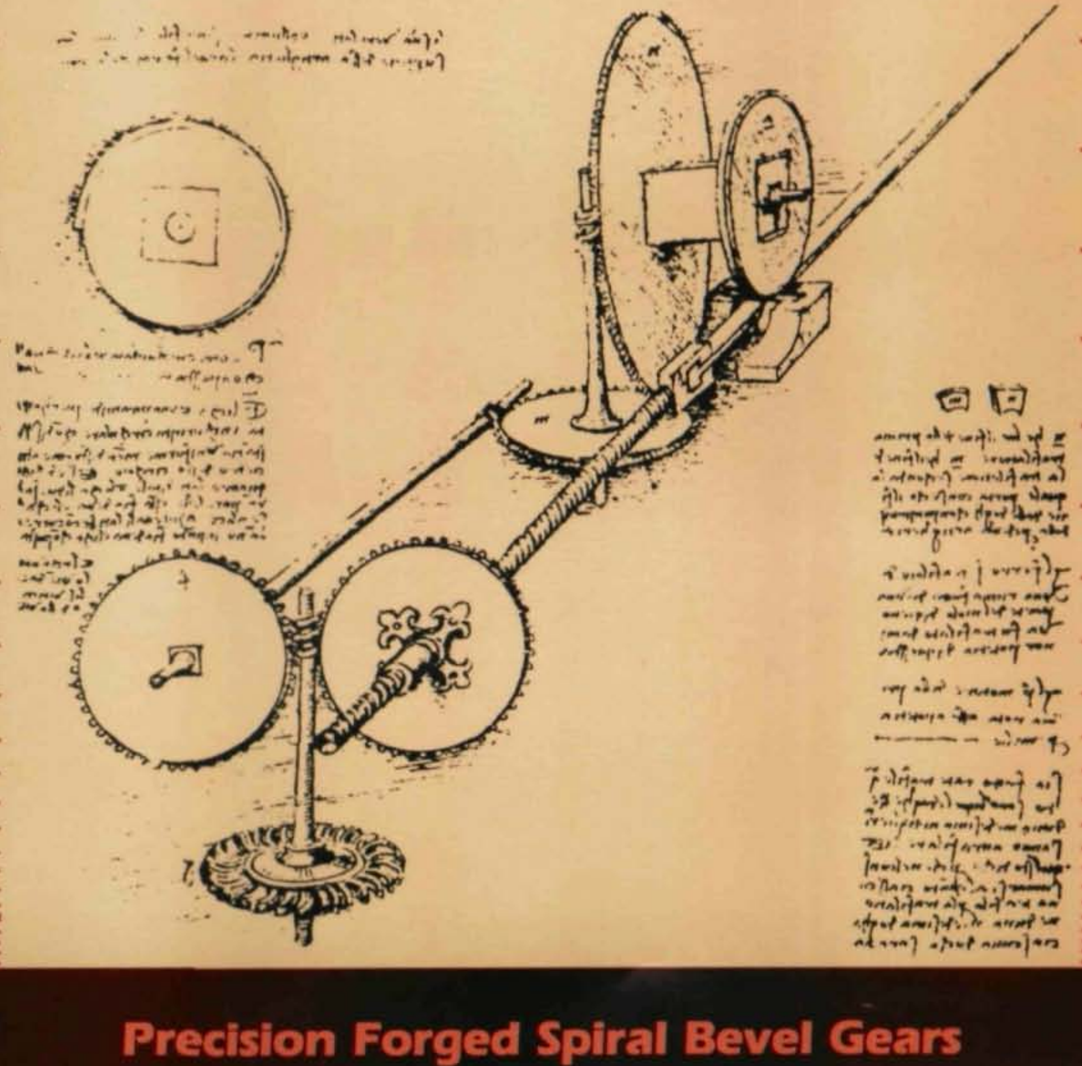
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

AUGUST/SEPTEMBER 1984



Precision Forged Spiral Bevel Gears

Correction of Damaging Resonances in Gear Drives

Maximum Surface Temperature of the Thermoplastic Gear

Economics of CNC Gear Gashing

Determination of Gear Ratios

Application of the Involute

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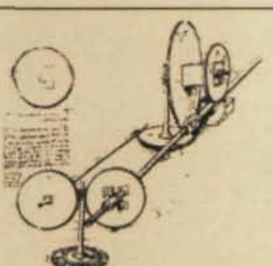

COVER:
The advanced technology of
LEONARDO DA VINCI
1452-1519

It has been said that Leonardo Da Vinci was one of the most extraordinary geniuses who has ever lived. In a series of notebooks, he recorded descriptions, drawings, and diagrams of "inventions" which were centuries before their time. Leonardo comprehensively studied each problem and derived the operation of machines from mathematical theory.

The cover sketch is a design for a water-powered mechanism for forming the stave or segment of a gun-barrel. The barrel would be built of several of these segments. These segments would be banded together, heavier at the breech section than at the muzzle. All segments were to be smooth and even, with a uniform taper in order to obtain a straight gas-proofbore. Leonardo also designed a machine that would draw and roll the bars from one power source, a reaction water turbine. Through two sets of worm drives, the sheet iron was simultaneously made to advance while being rolled.

CONTENTS — PAGE NO.

FEATURES

DEPARTMENTS

August/September 1984 — Vol. 1, No. 2

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

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

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

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

AGMA Calendar

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

Sept. 5-6, 1984	Metallurgy and Materials Committee (1b) Hyatt Chicago O'Hare, Chicago, IL
Sept. 10-11, 1984	Handbook and Inspection Committee (3c) AGMA Headquarters, Arlington, VA
Sept. 11, 1984	*Product Division Executive Committee Westpark Hotel, Arlington, VA
Sept. 12, 1984	*Technical Division Executive Committee AGMA Headquarters, Arlington, VA
Sept. 13-14, 1984	Helical, Herringbone, and/or Spiral Bevel Enclosed Drives and Standard Units Committee (6.3a) AGMA Headquarters, Arlington, VA
Sept. 17-18, 1984	Gear Rating Committee (5c) Brown Palace, Denver, CO
Sept. 19, 1984	*Policy & Practice Guides Standing Committee Brown Palace, Denver, CO
Sept. 19, 1984	Component Design Committee (5b) 7 Continents Hotel, Chicago, IL
Sept. 20-21, 1984	Mill Gearing Committee (2c) AGMA Headquarters, Arlington, VA
Sept. 25-26, 1984	Helical and Herringbone Drives Shaft Mounted Units Committee (6.3b) AGMA Headquarters, Arlington, VA
Sept. 27, 1984	*Flexible Couplings Council Westin Hotel, Copley St., Boston, MA
Sept. 27, 1984	*Statistical Subcommittee, Marketing Council Westin Hotel, Copley St., Boston, MA
Sept. 28, 1984	*Marketing Council Westin Hotel, Copley St., Boston, MA
Oct. 14-17, 1984	Fall Technical Meeting L'Enfant Plaza Hotel, Washington, D.C.
Nov. 8-9 1984	Bevel Gearing Committee (4a) AGMA Headquarters, Arlington, VA

*Company Member employees only or Specified members of committees.

. . . AND FROM THE INDUSTRY

HERMAN RICCIO, CHICAGO GEAR WORKS PRESIDENT TO RETIRE

Herman Riccio, long-time President of **Chicago Gear Works,** the company he lead to national prominence in the commercial gear industry, will retire in November, 1984 after 55 years of service. He began working at Chicago Gear Works in 1929, at the age of 15, and never worked for another company.

In a recent interview, Riccio reiterated his basic business philosophy, "Try giving the customers the best service you can. Give them quality. You can't stand still." Riccio proved that himself by retiring once before, in 1982, only to return six months later.

Mr. Riccio began his work at Chicago Gear Works as janitor and errand boy, working for 26 ¢/hour. He was nearly fired when he requested a one-cent raise! However, by 1937, he had moved to the office as assistant to the Vice-President for Manufacturing and Engineering. Fourteen years later, he assumed those responsibilities himself as Executive Vice-President for Manufacturing; and in 1967, Riccio and two partners purchased Chicago Gear Works.

Although Mr. Riccio states that he is "retiring," he will continue working 1-2 days per week and will be an active member of the management committee.

GLEASON OPENS MI SALES OFFICE

The Gleason Works announces the formation of a regional sales office in the greater Detroit area. **Geoffrey Ashcroft** has been named Regional Sales Manager and will be coordinating the sales and service of Gleason Equipment in the Michigan area. Ashcroft, who was associated with the company's overseas dealer network in the 1960's, has a wide range of experience with bevel and parallel axis machinery and technology.

This announcement closely follows Gleason's recent press release inaugurating a new domestic marketing philosophy with the signing of the first domestic dealership with State Machinery Co., Inc. of Indianapolis, Indiana.

James E. Cronkwright, Vice President of Marketing, in commenting on today's announcement, has noted the importance of quick customer response in today's competitive economy.

PETERSON JOINS AMERICAN PFAUTER

Steve Peterson has joined **American Pfauter Limited,** Elk Grove Village, IL as Regional Sales Manager. Steve brings over 20 years of experience in the machine tool industry support equipment field. Steve will be responsible for technical sales and customer support of Pfauter gear cutting, Hoefler gear inspection, Kapp grinding, and Sykes shaping applications.

NOTES FROM THE EDITOR'S DESK

THANK YOU! . . . the response to our first issue has been extremely exciting for us. Our advertisers have told us GEAR TECHNOLOGY is being talked about wherever they go. *Thank you* for the wonderful and enthusiastic reception.

Our response to the subscription card in the last issue has far exceeded our expectations. We have been receiving response cards at an average of 100 per day. As we are in the process of changing computer systems, and implementing new mailing list software, we will be a little delayed adding new subscribers. Thus, if you recommended someone to receive "The Journal" it is likely they won't be added to the subscription list until the third issue.

There are a few ways in which you can help us do a better job. If you have a "Free Subscription" insert in your book, it means you probably didn't fill out the subscription card in the last issue, it got lost, or you forgot to fill in your company name. Even if this issue is addressed to you, please fill it out COMPLETELY and SIGN it. This is needed for post office regulations. The quicker we meet the postal regulations by getting a SIGNED card from EVERYONE the sooner we can concentrate all our efforts on the contents. If only an editorial reply card can be found in your magazine, thank you, we have already received your signed subscription card; but don't forget to tell us what you thought of the articles. Send us your comments on any subject. (We might include it in "Viewpoint.") We invite you to tell us what you want to see in future issues.

We have been in contact with prospective researchers and authors all over the world and have a continuing interest in receiving new articles. Have you developed a new technique/process, or solved some unusual problem that would be of interest to the gear manufacturing community? Tell us about it so we can consider publishing it. Articles submitted should be written from an educational and/or training viewpoint. Everything we publish, however, may not be new. We will be publishing papers that may have been presented at technical conferences, or published in small circulation or foreign publications, but if we decide they deserve a wider audience, they will be published.

Finally, our objective is not only to be informative but also to help you get more business. Virtually everyone having anything to do with the gear business receives "The Journal." Wouldn't an ad in GEAR TECHNOLOGY be a natural way to tell your potential customers something about your company, and to expand your customer base? The classifieds will find that engineer, that job for your special or unique machine, etc. . . . Please, just keep us in mind.

Michael Goldstein
Editor/Publisher

GUEST EDITORIAL

A SECOND RATE SOCIETY — NEVER!

Forrest D. Brummett, SME's 1984-85 International President.

What was once recognized as the unique genius of America is now slipping away from us and, in many areas, is now seen as a "second rate" capability. Unless action is taken now, this country is in real danger of being unable to regain its supremacy in technological development and economic vigor. First, all Americans must understand the serious implications of the problem; and second, we must dedicate ourselves to national and local *actions* that will ensure a greater scientific and technological literacy in America.

To help all of us understand the problem, let me cite just a few facts:

- Many of our nation's high schools do not offer the necessary math and science courses to qualify their graduates for consideration by our accredited Engineering institutions.
- Educational institutions throughout the country report an ever increasing number of students cannot read and comprehend, and are not prepared to select careers in science and technology. Remedial courses in both Reading and Mathematics are the order of the day. In addition, these same institutions report that there is a woeful lack of people qualified to teach courses in modern Math, Engineering, and Natural Science. Many advanced courses in high schools, colleges, and universities languish through lack of qualified teachers.
- There is very little "real world" career guidance at the secondary school level in the area of Engineering, Technology, and the Free Enterprise System. Consequently, when students get into college, many choose these careers without adequate preparation as they become aware of the potential opportunities in these fields.

The end result to this dilemma is that industry ends up with a product they cannot utilize without major training and redirection. This points up what, I believe, is one of the most serious problems facing American industry today – the technical illiteracy of its engineers, managers, and executives. Few of them, due to lack of adequate training in the academia, have the skills to solve "real world" problems and implement current and new technology required to compete in the marketplace today or in the future. Business decisions require weighing variables, such as sales and costs, product quality and productivity, price and market share, profit and taxes. Many of our managers aren't well equipped to reason and think in those terms. Is it any wonder that they are drawn to legal or financial solutions rather than technical or human ones?

New and innovative technological concepts have brought about a clear awareness and recognition of the need for the secondary schools and universities to produce an abundant supply of "hands on," "real world," "applications oriented," individuals that more adequately meet the needs of today's industries. This means updating and improving the technology, engineering, and science curriculum; and providing the most modern Lab facilities and teaching methods available.

In addition, the faculty must be given the opportunity for updating their skills as new technology emerges, and as current technology changes. Herein lies the need for interaction between the academia, industry, and the government.

Since we cannot predict, with any precision, what kind of jobs people will hold over their 35-40 year working life, it is best to provide a good general education with an ability to adapt to changing jobs and careers. If work requirements change abruptly and quickly, the educational system must

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MR. FOREST BRUMMETT, *author of the "Guest Editorial," is Chief Engineer at Detroit Diesel Allison Division of General Motors Corporation — Indianapolis Operations. He joined Detroit Diesel in May of 1949 and has held responsible positions in Production, Tool & Die Making, Tool & Die Design, Plant Engineering, Methods Engineering, CAD/CAM Systems and Capacity Management. Active in the Society of Manufacturing Engineers since 1962 holding office at Chapter, Region, and International levels. He served on the S.M.E. Technical Council as Chairman of the Assembly Division and became chairman of the Technical Council for (2) terms in 1975 through 1977. Currently International President of S.M.E. and has been on the Board of Directors for the past 12 years. In addition, Mr. Brummett is among those listed as Who's Who in Engineering in America. He is a Certified Manufacturing Engineer in the field of General Manufacturing Engineering. Mr. Brummett attended Purdue University and received his Bachelors Degree in Mechanical Engineering.*

respond faster and more efficiently to training needs. This will require better ties with industry and should not exclude the possibility of more industry based activities.

Like it or not, we are in an international marketplace that impinges on our nation much more than they used to, and basic institutions must change to account for that.

The trends are now reflecting the loss of our technological edge, e.g.: We have allowed our piece of the world trade to drop from 17% to 12% from 1965 to 1983 while the volume of international trade increased from 150 billion to 1.9 trillion. For example, we have lost our share of the world market in consumer electronics from 35% to 9% in the last 15 years.

These are depressing facts and it will take an enormous amount of concerted *action* by all Americans to overcome this terrible handicap, which we are now recognizing and which will commit us to *mediocrity* and *second rate*, if not addressed. Technology impacts all our lives !!

Citizens need to change their attitudes on manufacturing technology to become *economically more responsible*. This also begins in the educational system. A long range view is needed to emphasize technical innovation, increased productivity, and quality improvements. *We must learn how to produce more than we consume!*

The time is here – we are an industrial nation in deep trouble. We can no longer afford the luxury of confrontation. Traditional adversary relationships must be tempered by the necessity of industrial growth and survival. This effort must begin in our educational systems, where specific skills and knowledge are taught, utilizing modern teaching methods, computers, and facilities to produce a product that meets the *real needs* of industry today and in the future.

We cannot be bound by tradition; we must initiate change — flexibility in curriculum adjustment to meet current needs as well as the future, based on manufacturing technology trends. *Education is everyone's responsibility.*

Forrest Brummett

Forrest D. Brummett

SME, International President

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:

I read with great interest your first issue of GEAR TECHNOLOGY. I find it a refreshing move based on these economic times.

I would be most interested in communicating with you on one of our major concerns—foreign competition. I believe this will greatly affect the industry more than any technological change now or in the foreseeable future. Foreign competition, for all the reasons we have heard, is making startling inroads into the available market of the American gear manufacturers. In some cases, based on the company's size, overhead expenses and services rendered, I can certainly understand how these companies can be competitive. This situation even occurs in some of our smaller gear companies when they are compared price-wise to some of our larger companies within this country.

But, I can tell you that we've also seen some highly questionable pricing situations which can only occur because of some unfair competitive position or a willingness to undermine our market until it's weakened to the point where they can raise their prices or run the competition out of business. Some of the unfair competition has to include our tax base structure compared to theirs, based on our cost of defense, our personal income taxes, our unemployment taxes, our benefits programs, etc.

If as a nation we're sincerely interested in maintaining our standard of living and offering all these opportunities, then our industries must be strong. For our industries to stay strong, they must stay productive. To do this, we must be on a fair competitive basis with those people that compete against us.

When I speak of unfair practices, I am not just vocalizing "sour grapes" since we are a strong advocate of free trade. However, we are finding in these economic times it is preposterous to be bidding on a federally funded work program (which our tax dollars are paying for) against foreign competition. In other words, we are paying corporate income, personal income, and unemployment insurance taxes, and those funds are being used to pay our overseas competitors to build products for our unemployed people. This situation seems incomprehensible to us.

I'm afraid that the political issues in the gear industry and this issue alone may be more important and more interesting than the technical side of our industry.

Again, I wish you good luck in this new venture.

Sincerely,
MILWAUKEE GEAR COMPANY

Harold Trusky
President

Precision Forged Spiral Bevel Gears

CAD/CAM Technique Makes It Practical

by
Aly Badawy
P. S. Raghupathi
(Batelle's Columbus Laboratories)

Gary Horvat
(Eaton Corporation)
Don Ostberg
(U.S. Army Tank Automotive Command)

Close tolerance forging of U.S. Army spiral bevel gears, requiring only a single finish machining operation (or none), now is feasible in production with the help of a newly developed Computer Aided Design and Manufacturing (CAD/CAM) technique. This method of manufacture offers many advantages because it reduces material losses and machining costs while increasing the fatigue life of the gears by 30 percent.

A recent U.S. Army Tank-Automotive Command project, conducted by Battelle's Columbus Laboratories, successfully developed the methodology of CAD/CAM procedures for manufacturing dies (via EDM) for forging spiral bevel gears. Further, it demonstrated that precision forging of spiral bevel gears is a practical production technique. Although no detailed economic evaluation was made in this study, it is expected that precision forging offers an attractive alternative to the costly gear cutting operations for producing spiral bevel gears.

CAD and CAM Integrated

In industrial practice, attempts are continuously made to introduce improved manufacturing methods to reduce production and life cycle costs. Close tolerance forging of spiral bevel gears—requiring only a single or no finish machining

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AUTHORS:

DR. ALY BADAWY *is Principal Research Scientist, Engineering and Manufacturing Technology Department for Battelle's Columbus Laboratories. His experience covers research and development in Marine Engineering and Naval Architecture, Applications of Optimization Theory in Design, Computer Aided Design and Interactive Computer Graphics, and Geometrical Modeling of Sculptured and Complex Surfaces. Dr. Badawy has authored or co-authored 15 technical papers related to Computer Aided Design and Computer Graphics. He holds a B. Sc. from Alexandria University, and a M. Eng. and a Ph. D. from McMaster University, Canada. Additionally, he is a member of the American Society of Mechanical Engineers, Computer Aided Design Committee.*

DR. P. S. RAGHUPATHI'S *experience is in the area of cold extrusion, closed die forging, deep drawing, metal forming machine tools and computer aided design and manufacturing. He is currently the Associate Manager, Metalworking Section of Battelle's Columbus Laboratories. In addition to being the author/co-author of more than 20 publications, he is also a co-editor of a Metal Forming Handbook which is soon to be published. Dr. Raghupathi holds a B. E., University of Madras, India, M. E. from Indian Institute of Science, and a Dr. Ing. from the University of Stuttgart, W. Germany. As a member of the International Cold Forging Group based in Europe, he maintains close contact with European Universities, research laboratories and companies active in manufacturing technology.*

MR. GARY L. HORVAT *has been employed at Eaton Corporation since 1977. His work in Forging and Forging Development at various Eaton Divisions has given him a unique background in the precision forging of gears. Currently, Mr. Horvat is a Manufacturing Development Engineer. He attended Cleveland State University, graduating with a B. and M.S. Industrial Engineering. He is a member of American Society for Metals, Society of Manufacturing Engineering, CASA, Computer and Automated Systems Association of S.M.E.*

MR. DON OSTBERG *is currently a Materials Engineer for the United States Army Tank Automotive Command. He has been at TACOM since June 1977. During this time he has been involved in the manufacturing technology efforts. Mr. Ostberg studied at Cleveland State University and holds a BA in Chemical Engineering.*

operation—offers considerable advantage over machining, because this method of manufacture (1) reduces material losses and machining costs and (2) increases the fatigue life of gears up to 30 percent.

A few companies around the world are able to produce spiral bevel gears by precision forging. However, the development of the process for each new gear design requires considerable trial and error. Thus, application of computer techniques to the design and manufacture (CAD/CAM) of the gear forging dies represents an attractive alternative. Therefore, in this program, methods were developed to apply existing advanced computer aided design and manufacturing (CAD/CAM) technology (finite element, metal forming, and heat transfer analysis) to gear forging die design and manufacture. Gear forging dies were designed and manufactured according to the data supplied by the output of the CAD procedure; thus, the CAD and CAM processes were integrated. The results of the CAD/CAM techniques were evaluated for a given spiral bevel gear/pinion set by designing and manufacturing the forging dies via CAD/CAM.

In recent years, CAD/CAM techniques have been applied to die design and manufacture for forging rib-web type aircraft structural parts, track shoes for military vehicles, and precision turbine and compressor blades. The experience gained in all these applications indicates that a certain overall methodology is necessary for CAD/CAM of dies for precision and/or near net shape forging. This approach indicates that the necessary inputs to the CAD/CAM system are: geometric description of the forging, data on billet material under forging conditions (billet and die temperatures and rate and amount of deformation), friction coefficient, to quantify the friction shear stress at material and die interface, and forging conditions (i.e., temperatures, deformation rates, die lubricants, method of heating the billets, and suggested number of forging operations).

With these input data, a preliminary design of the finish forging die can be made. Next, stresses necessary to finish forge the part and temperatures in the forging and the dies are calculated. The temperature calculations take into account the heat generated due to deformation and friction, and the heat transfer during the contact between the hot forging and the cooler dies. Thus, 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. The estimation of die geometry corrections is necessary for obtaining close tolerance forgings and for machining the finish dies to the exact dimensions.

The overall procedure described above has been applied to CAD/CAM of spiral bevel gears as Phase I of this project.

The second part of the project (Phase II) involved Computer Aided Manufacturing (CAM) of the forging dies (from rough billet) and demonstration of the effectiveness of CAD/CAM by forging 20 spiral bevel gear sets. Phase III—Application of CAD/CAM techniques to actual production of bevel gears (spiral or straight)—has not yet started.

Five Tasks Carried Out

Five separate tasks were carried out under Phase I of the work: preform design, tool design, manufacturing of forging dies, forging trials, and finishing and dimensional checking of forged gears.

One of the most important aspects of the forging process is the proper design of preforming (or blocking) operations. The following features were considered in the design of the preform of the spiral bevel gear considered in this project.

Assure Defect Free Metal Flow and Adequate Die Filling. Adequate metal distribution is necessary in the blocker design to avoid forging defects, such as cold shuts and folds. The preform was designed as a solid ring (no teeth) with the outer dimensions as close as possible to the outer dimensions of the finished gear. This minimizes the amount of material to be moved during forging, and this in turn, enhances die filling.

Minimize the Material Lost in the Flash. In steel forgings, approximately half of the cost of forging consists of material

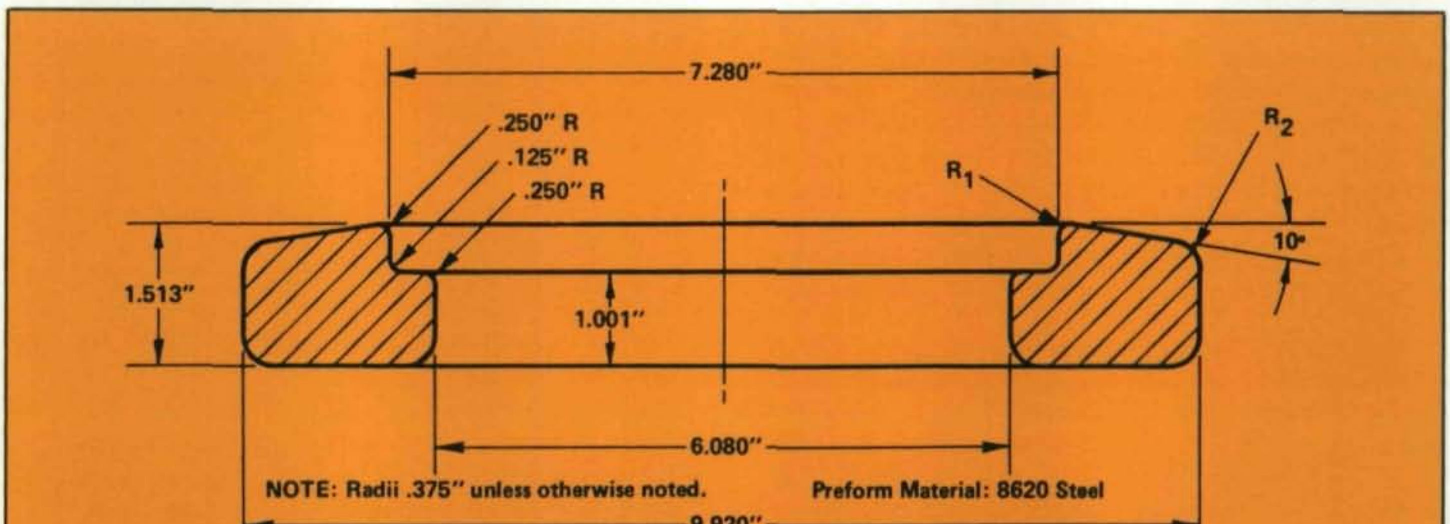

Figure 1

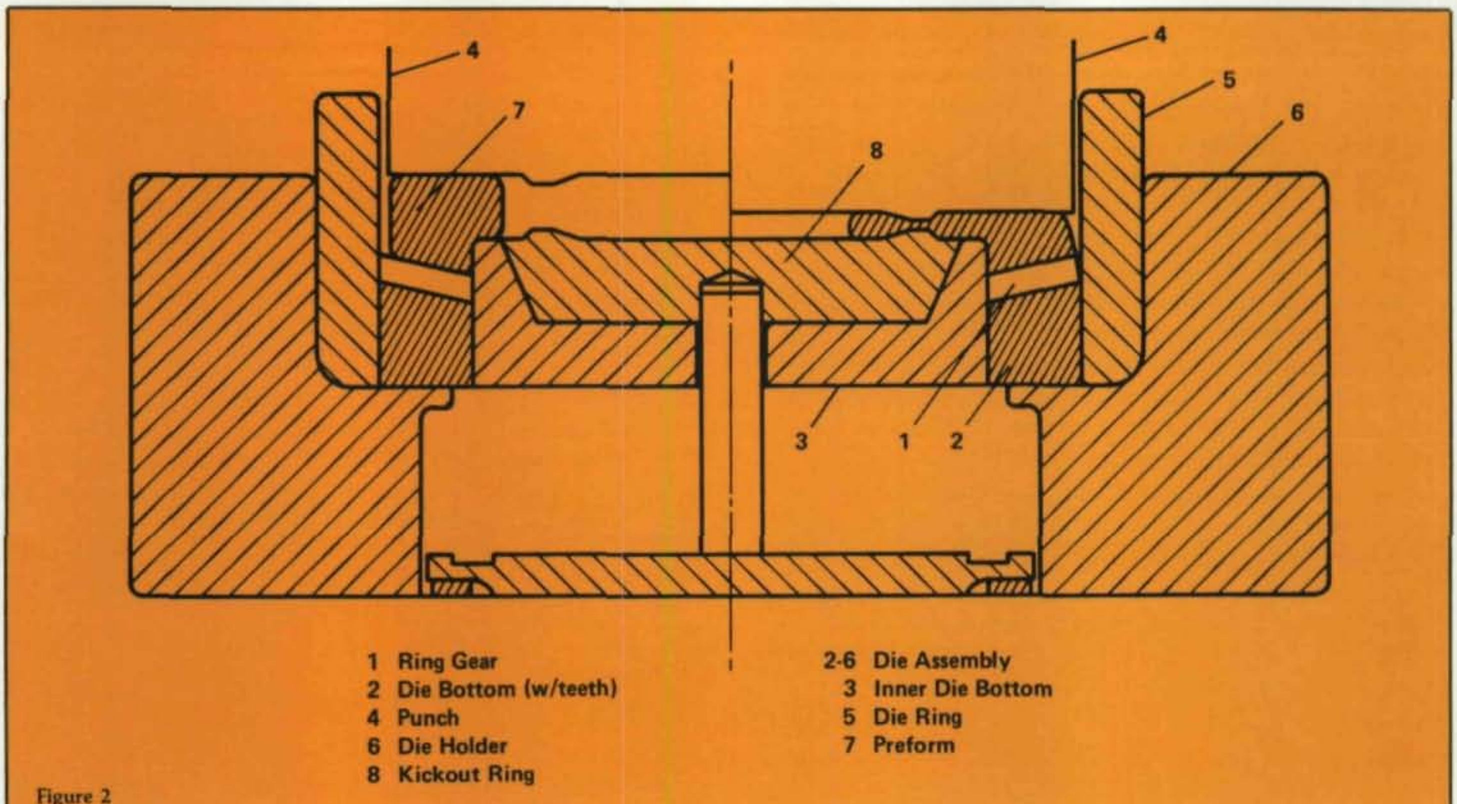

Figure 2

costs. On the average, 30 percent of the incoming forging stock is lost in the form of flash. Thus, approximately 15 percent of the forging costs are in the flash material of relatively little recoverable scrap value. The design of the blocker of the gear produced no flash. This was because the volume (or weight) of the preform was slightly larger than the volume of the finish gear, and that the proper material distribution throughout the preform volume was achieved.

Centering of Preform in Die. The preform was designed as a pancake with its center lying exactly on the center of the die. This was thought to insure even filling of the die cavity.

With the above considerations, an initial preform design was developed as shown in Fig. 1. Fig. 2 shows the preform positioned in the die. As discussed later, a subsequent preform was designed and used in the forging trials. The new design shown in Fig. 3 was wider, so that the metal would not have to move very far to fill the cavity. The size of the corner radii was also reduced to provide more material at the corners. The billet material, that was used for forging spiral bevel gears, was cut from bar stock. The billet was upset to form a pancake having the proper diameter. The pancake was subsequently machined to the dimensions specified for the preforms.

Tool Design

The forge tooling was designed by using the results of Phase I of this project. The die assembly is a two piece design. The die insert, with the teeth, is one piece with a die ring around the insert to form the outer diameter of the forging. At the center of the die is the die insert and the center kickout. The insert forms the inner diameter of the forging, while the kickout removes the gear from the die after forging. The kickout is designed to lift the part by pushing on the center flange of the gear. It is activated by a mechanical kickout mechanism of the forge press which raises during the upstroke of the press. The kickout is, also, designed to contact the preform in such a way as to minimize the amount of material that is moved across the face of the insert. Fig. 2 shows the kickout system.

The tooling assembly is shown in Fig. 4. Incorporated in the forging design is the straight sided outer diameter with the provision of flashing toward the inner diameter. The inner diameter allows a 3/8 inch flash thickness on each side to trap the material. Inside the flash land is a gutter for excessive material to flow.

The punch holder design is different from the die holder in that it has a solid punch without any kickout. The punch also has provisions for placing spacers, or shims, between the punch and punch holder to vary the forging and flash thickness. This is necessary because it was decided to conduct the forging trials on a press, which has a fixed bolster, with no wedge adjustment. The shims help to adjust the forging thickness, and the die fill, within certain limits. The components of the die tooling setup are shown in Fig. 4. The assembled tooling is shown in Figs. 5 and 6.

Manufacture of Forging Dies

Precision manufacture of forging dies plays an important role in the success of precision forging of spiral bevel gears. The use of sound die manufacturing methods is essential, if the required gear precision is to be achieved.

Based on the results of Phase I of this project, it was decided to use the hot work steel, H-11, as a die material for the near net gear forging trials, and the hot work steel, H-13, as a die material for the net gear forging trials. The billet material used in both trials was 8620 steel. The die blanks were heat treated and then machined prior to the EDM of tooth cavities. The EDM electrodes were machined on a conventional gear cutting machine, using machine settings supplied by the CAD Computer Program SPBEVL of Phase I.

The electrode geometry accommodated all the corrections (elastic deflection due to loading, temperature differentials, and bulk shrinkage), as described eariler. The gear impression on the die was obtained by EDM. The EDM operation was performed using six electrodes in sequence. Each subsequent electrode was burned deeper until the required depth was obtained. Important steps in the manufacture of the forging dies included preparation of the electrode, EDM burning of the die, and the final grinding of the die after EDM.

It is worth mentioning here that this task (Manufacture of Forging Dies) makes use of all the data supplied from the Phase I part of this project; hence, integration of the Computer Aided Design (CAD) with the Computer Aided Manufacturing (CAM) was achieved in the production of forged spiral bevel gears.

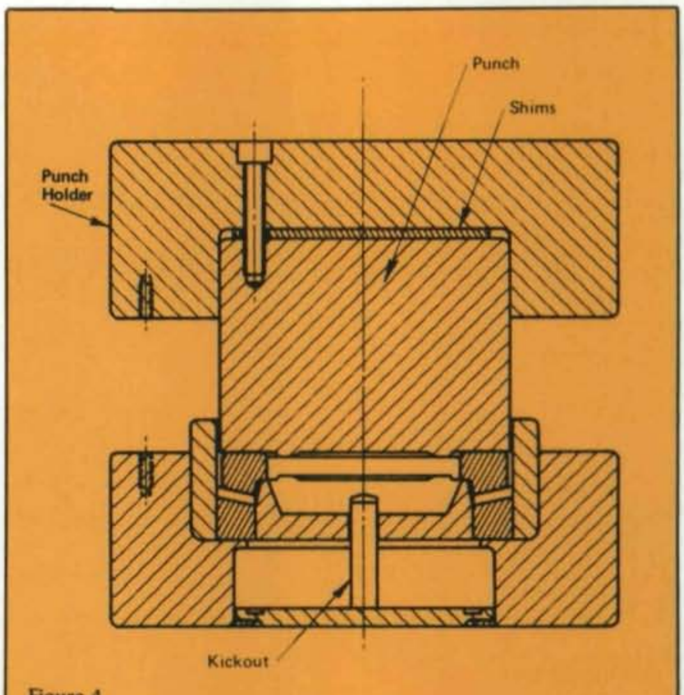

Figure 4

Forging Trials

After manufacturing the dies with the required precision, as described earlier, the gear forging trials were conducted at Eaton Corporation's Forging Division in Marion, Ohio. A 3,000 ton mechanical forging press manufactured by National Machinery Company, Tiffin, Ohio, was used to perform the trials. The press was selected based upon the anticipated forging load of about 2,500 tons and the space available for the tooling.

Three series of forging trials were conducted. During the first series, the technological details of the forging procedure such as heating, lubrication, part transfer, and cooling were established. During the second series of trials, 20 gears were produced with gear teeth forged to near net dimensions. These gears subsequently were machined with a single machining operation. In the third series of trials, 20 spiral bevel gears were forged with net teeth dimensions. Thus, the gear-pinion sets were obtained by machining only the back side of the forged gears, and by machining the matching pinions.

The forging loads were monitored using load transducers attached to the frame of the press. These strain gage devices sense the strain in the frame of the press during forging, and generate an electrical signal that is proportional to the load.

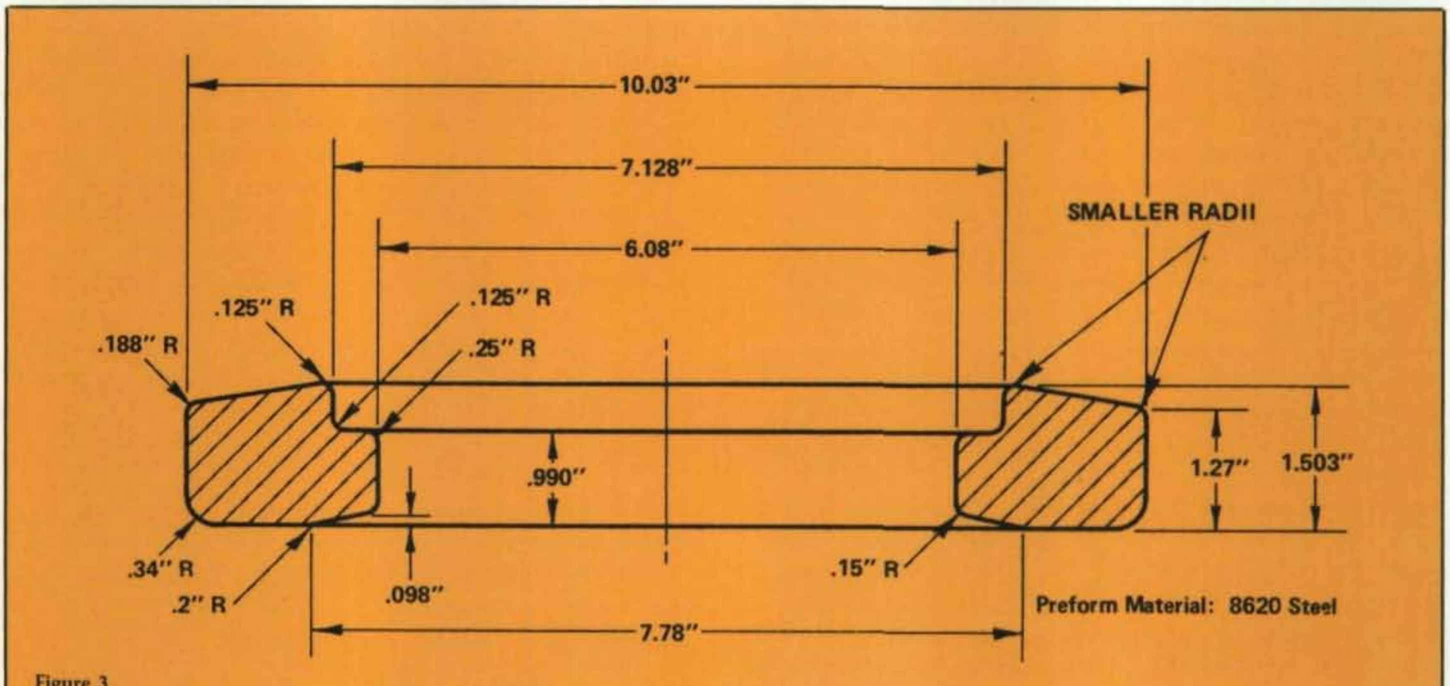

Figure 3

Figure 5

Once the transducer system has been calibrated, the electrical signal can be read directly as load on the digital readout device. The system used was a Model LG-Il, designed and built by Helms Instrument Company.

The die lubrication used during the forging trials consisted primarily of a water base graphite material sprayed with pressurized air. A hand wand was used to direct the lubricant onto the die. Several billets were coated with a graphite based coated material to reduce oxidation during heating, and improve lubrication during forging. However, no advantage was noted in surface finish, die fill, or forging load. The practice was discontinued after the initially coated billets were used.

After forging, the gears were placed, teeth down, in a sand-graphite mixture to reduce oxidation of the teeth during cooling. The back surfaces of the gears were still exposed to air, so that the cooling rate would not be excessively slow.

Results of Forging Trials

During the first set of trials, a gas furnace was used to heat the preforms. This resulted in excessive scale formation and poor surface finish of the forged gear teeth. The heating was done by induction in the subsequent trials. The outside diameter of the preforms was considerably smaller than the internal diameter of the die cavity. As a result, some preforms could not be centered accurately and the forged teeth configurations were not uniform.

The second set of trials was considered very successful for the following reasons. First, the forged gear was uniform. All teeth looked almost alike. This meant that the centering problem encountered during the first forging trials was eliminated. Second, the surface quality of the forged gear teeth was excellent. The induction heating of the preforms produced forgings with minimal scale. That meant that the scale problems, encountered during the first forgings trials, where a gas furnace was used to heat the billets, were eliminated.

However, two problems were encountered during the second series of forging trials: (1) There was incomplete filling at the toe and heel of the tooth. This problem was due mainly to the preform design. The radii at the outer and inner part of the ring are generous; consequently, there was not enough material at these parts to completely fill the die cavity. (2) Non-uniform temperature of the billet was noticed due to the change in colors in the inner part of the ring (cooler) and the outer part of the ring (hotter, i.e., the red color was brighter). This problem could be solved later by trying different heating cycles and times and lower induction frequency to obtain a uniform preform temperature.

The next forging trials (third set) were successful in producing gears with excellent surface quality and with superior die fill, as compared to near net forging trials. The new preform design used in this trial was the main reasons for the better fill in the toe and heel of the gear. As shown in Fig. 3, the preform has a smaller radii in the toe and heel of the gear, compared with the near net preform. This additional material in the toe and heel enhanced the filling of those parts.

Dimensional Checking of Forged Gears

The forged gears were checked for dimensional accuracy on the Zeiss machine. The Zeiss machine is a computer controlled coordinate measuring machine (manufactured by Zeiss Corporation in West Germany) which produces plots of the tooth form variation as compared to the tooth surface of the cut master gear, produced by conventional cutting on a Gleason generator.

(Continued on page 48)

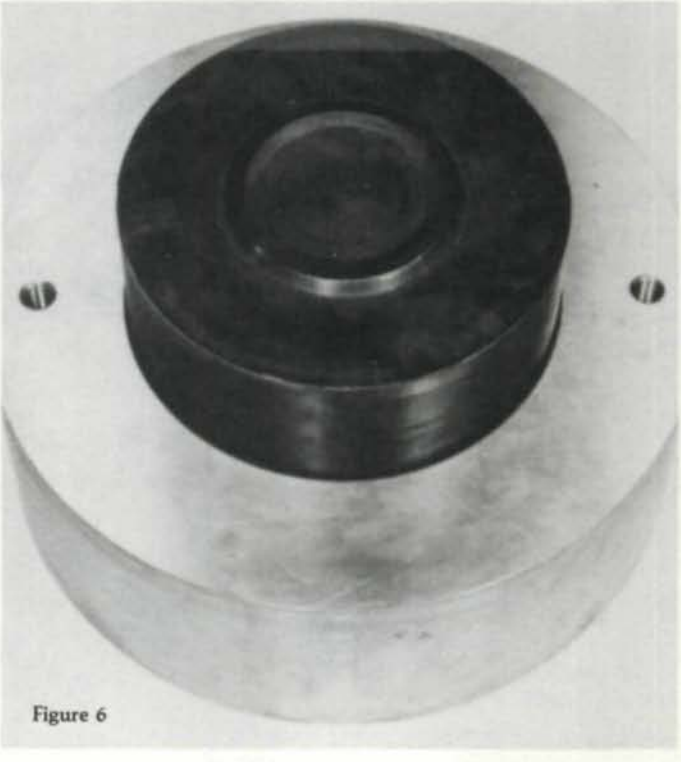

Figure 6

Identification and Correction of Damaging Resonances in Gear Drives

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Abstract

As a result of extensive research into the vibration characteristics of gear drives, a systematic approach has evolved, by which damaging resonances can be eliminated. The method combines finite element techniques with experimental signature and modal analyses. Implementation of the bulk of the method can be carried out early in the design stage.

A step-by-step description of the approach, as it was applied to an existing accessory drive, is given in the text. It is shown how premature bearing failures were eliminated by detuning the torsional oscillations of a gearshaft. A dramatic reduction in vibration levels was achieved as a result of detuning the problem gear.

The proposed approach can be extended to other types of rotating machines.

Introduction

During the endurance test of an Aircraft Mounted Accessory Drive, premature failure of the ball bearings on the starter shaft was discovered. The test, intended to last 8,000 hours without failure, was abruptly halted after only 900 hours of testing. The failed bearings were of the single row deep groove type with split riveted cages.

When the drive was disassembled and the failed bearings examined, the rivets were torn off and the cage was split open. Further analysis showed evidence of plastic deformation and pitting at the cage interface. Dimensional check of the bearing showed that load carrying elements were within tolerance and that no measurable defects existed. Also, no discoloration, such as due to lack of lubricant, was evident.

All results of the failure analysis were pointing towards vibration as being the cause of the cage failure. The fact that material was being upset on either side of the ball pockets, and the indications of pitting at the cage interface, could only have derived from ball repetitive hammering, thus inducing tensile stresses in the cage rivets. When sufficient number of loading cycles have been accumulated, and the fatigue limit of the rivet material has been reached, tensile failure of the rivets would ensue.

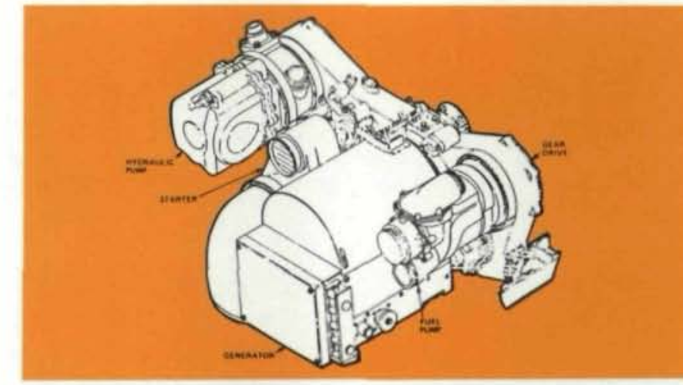

Fig. 1—Aircraft mounted accessory drive with accessories

The literature is full of examples where gear vibrations lead to more dramatic failures than the one presented here. Drago and Brown[1], for example, refer to a helicopter transmission gear that exploded during operation, because one of the resonant frequencies coincided with an excitation frequency. A number of case histories of gear-excited torsional vibrations are illustrated in reference[2]. In the latter paper, Rieger showed that torsional modes may be excited by low order harmonics of shaft rotation. The magnitude of excitation is directly related to gear machining errors as clearly analyzed by Mark[3]. He identified three types of transmission error: those due to tooth spacing errors, tooth-to-tooth random error other than tooth spacing, and tooth elastic deformations combined with mean profile deviations.

The purpose of this paper is to illustrate how the gear resonance problem was identified through the use of waterfall diagrams and finite element techniques. A systematic procedure is then proposed to eliminate damanging gear resonances from the operating range, early in the design stage.

Waterfall Diagram Survey

The accessory drive under consideration is shown schematically in Fig. 1. The outline of the gear case is represented by a dash-dot line. The various accessories consist of an hydraulic pump (HP), an air turbine starter (ATS), a variable speed constant frequency generator (VSCF), a fuel boost pump (FP), and two lube pumps (not shown). A frontal section of the drive shows the gear arrangement in Fig. 2.

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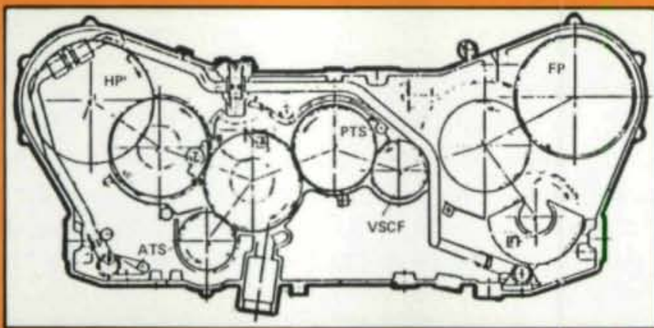


Fig. 2—(top left)
Frontal section
of gear drive

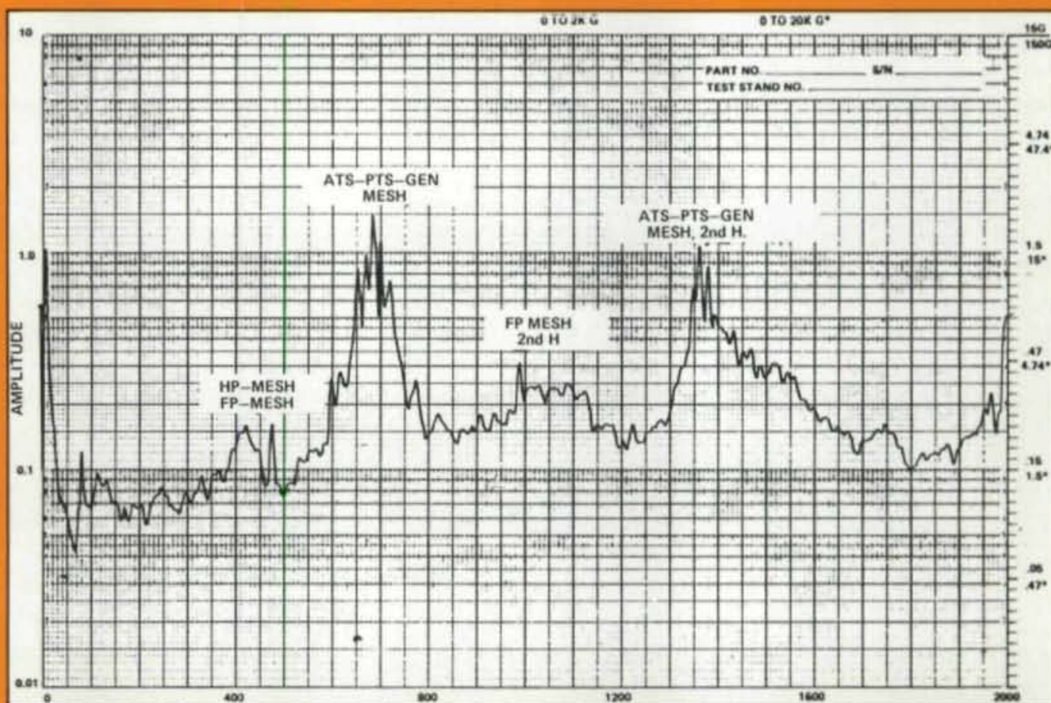


Fig. 3—(center)
Typical vibration signature
measured off gear case

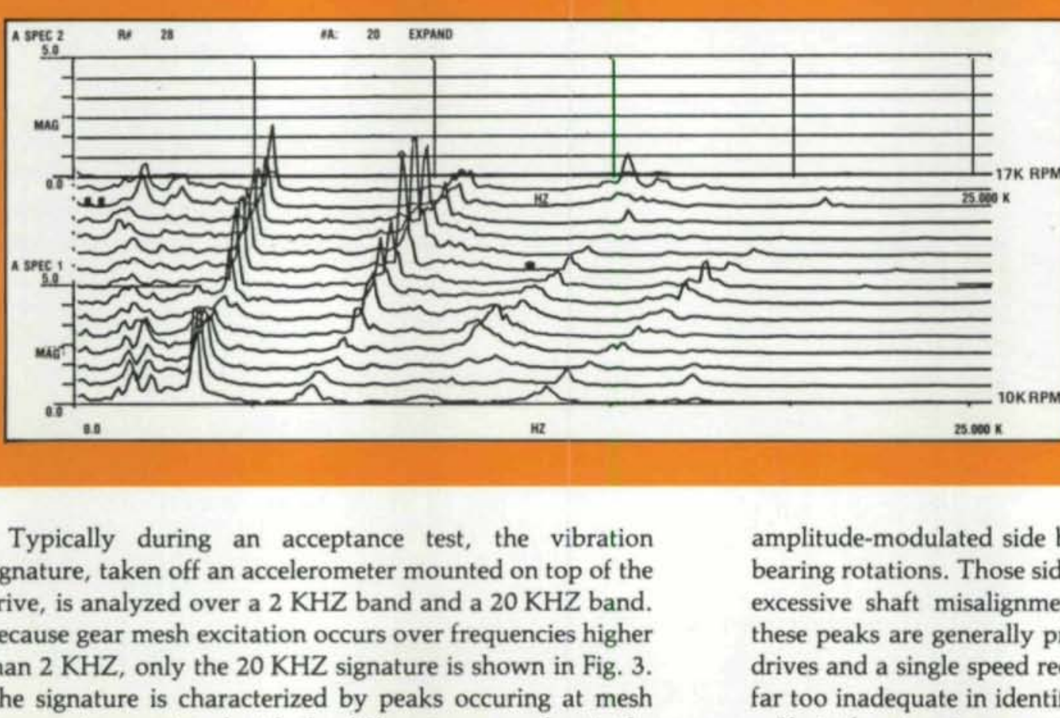


Fig. 4—
Waterfall
diagram of base-
line configuration

Typically during an acceptance test, the vibration signature, taken off an accelerometer mounted on top of the drive, is analyzed over a 2 KHZ band and a 20 KHZ band. Because gear mesh excitation occurs over frequencies higher than 2 KHZ, only the 20 KHZ signature is shown in Fig. 3. The signature is characterized by peaks occuring at mesh frequencies associated with the different gear meshes in the drive. Also indicated are the second harmonic as well as amplitude-modulated side bands associated with shaft and bearing rotations. Those side bands are usually indicative of excessive shaft misalignments and bearing defects[4 5]. All these peaks are generally present to some extent in all gear drives and a single speed record, such as shown in Fig. 3, is far too inadequate in identifying a vibration problem.

If similar signatures, over a given speed range, are collected at equal speed increments and then arranged in tandem as shown in Fig. 4, the so-called waterfall diagram is obtained. This waterfall diagram was generated using an autosequence program on the HP5423A Hewlett Packard Structural Dynamics Analyzer. Some features are now clearly apparent from Fig. 4. Peaks associated with gear impact propagate along straight lines defined by the mesh orders of the drive, i.e. their frequencies vary linearly with the speed of the corresponding gears. If a peak frequency is speed independent, then it must be either associated with a constant speed shaft or with a stationary resonance, such as that of the casing or the mounting structure. On the other hand, if a peak, excited by one of the gear meshes, attains a maximum at a speed within the test range, then the likelihood of a rotating element resonance exists. Such is the case with the gear mesh excitation of the ATS, VSCF, and PTS (Power Take-off Shaft).

An equally illustrative method of representing the data encompassed in a waterfall diagram is shown in Fig. 5. This plot is generated for the same drive using a Gen Rad dual channel analyzer. The fanning lines represent orders of excitation, the values of which are indicated on the right ordinate. The abscissa and the left ordinate represent the PTS speed in RPM and the vibration frequency in HZ respectively. The varying size square symbols shown in the figure are indicative of the vibration amplitude at the corresponding point.

It is interesting to note that, again a resonant point is detected at the same speed (14,000 RPM) and frequency (9.1 KHZ) as observed in Fig. 4. The order of excitation in this case is 39 which is the number of teeth of the PTS gear, as would be expected. The problem now is to determine which one of those three gears is the culprit. The PTS gear geometry, shown in Fig. 6, lends itself well to impulse hammer testing; while the other two, shown in Fig. 7, because of their compactness, are not suitable for impluse hammer testing. The former acts more like a plate in transverse vibrations, while the latter mostly behave as torsional members. An analytical finite element approach is, therefore, adopted for the ATS and VSCF gears.

Impulse Hammer Testing of PTS Gear

Use of a calibrated impulse hammer, in modal structural, analysis, has proven to be very effective in a great number of applications. The main reason is that the frequency spectrum of the time-varying impulse signal is nearly flat over a wide frequency range (up to 10 KHZ with a hard tip). Thus, all resonances of the structure within this frequency band can be excited.

The test procedure involves impacting the object with the hammer at many points, and measuring the motion at one critical point (or vice versa). These tests supply the stimulus and response information to compute the classical transfer functions. An example of such transfer functions is shown in Fig. 8.

The input and output signals are fed into a HP5423A dual channel FFT spectrum analyzer. An autosequence program is written to automate the conversion of the transfer function data into natural frequencies, modal damping, and mode shapes. A Hewlett Packard 9872 X-Y plotter is used to reproduce a plot of the animated mode shapes. Fig. 9 shows a plot of the first four modes. It is significant to note that none of the obtained modes coincided with the 9.1 KHZ observed in the waterfall diagram. It is, therefore, unlikely that the PTS gear is the source of the vibration problem.

Finite Element Modal Analysis of the ATS-Gear

The MSC/NASTRAN program is used to compute the natural frequencies and mode shapes of the ATS gear. Because gear and spline teeth contribute very little to the circumferential stiffness of the gearshaft, it is safe to assume

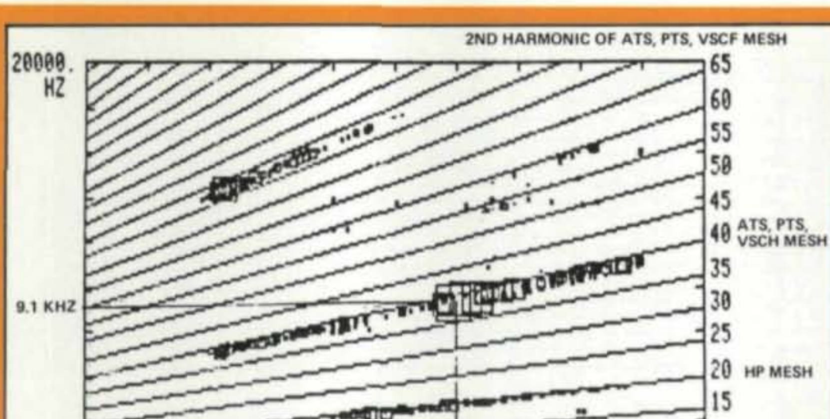
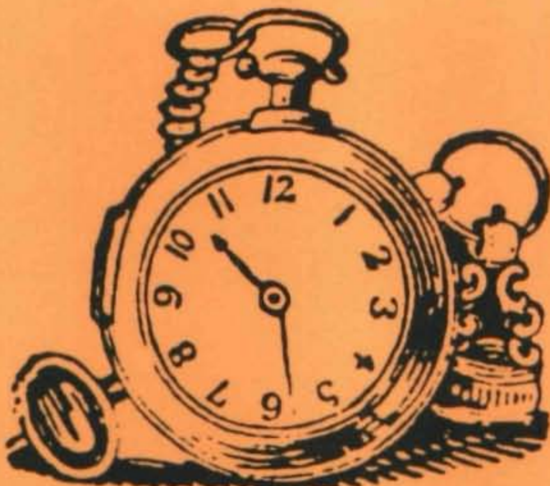


Fig. 5—(above) Excitation order diagram of baseline configuration

Fig. 6—(below) PTS Gearshaft



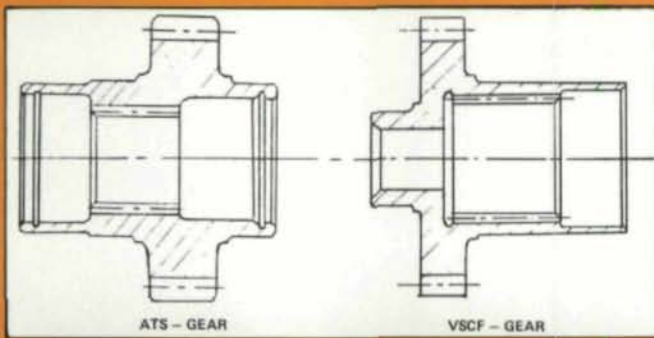


Fig. 7—(above left) ATS and VSCF Gearshafts

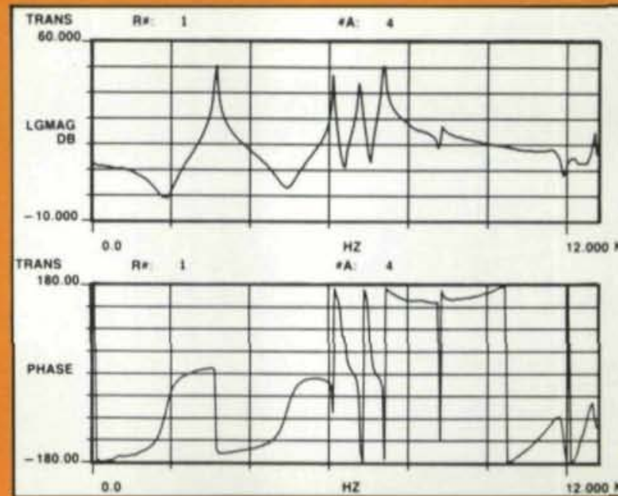
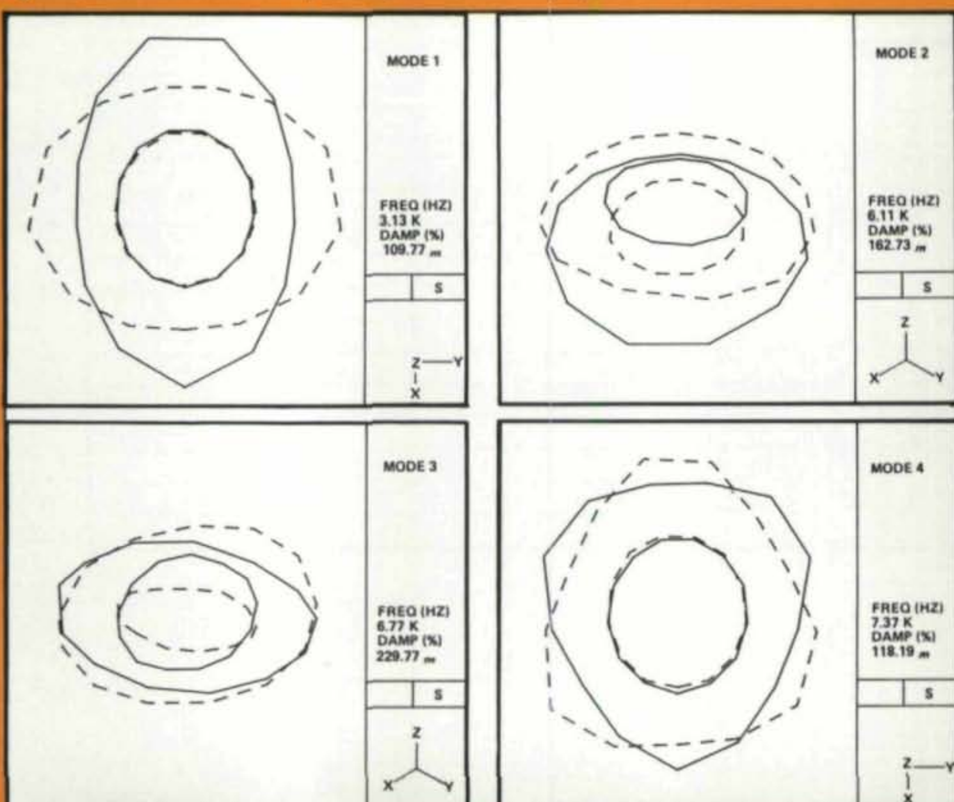


Fig. 8—(top right) Transfer function amplitude and phase for PTS—Gear

Fig. 9—(below center) Mode shapes, frequencies, and damping of PTS—Gear



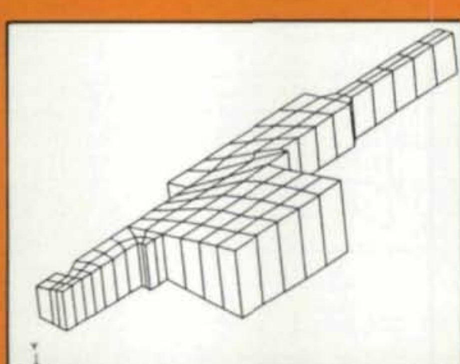


Fig. 10—Finite element model of ATS—Gear

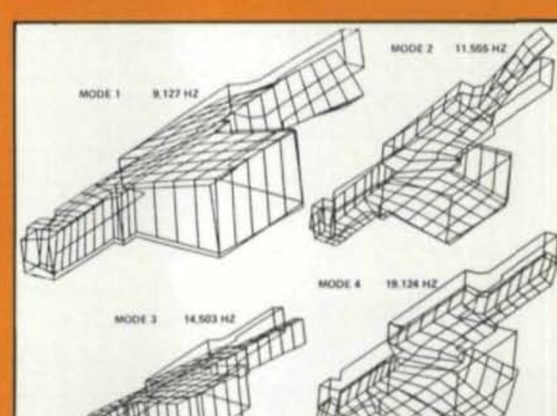
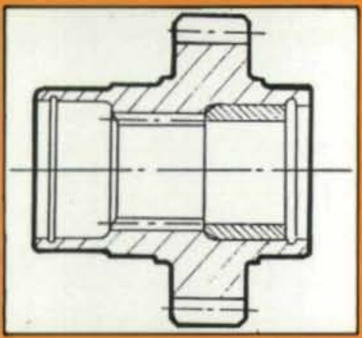


Fig. 11—(bottom right) Mode shapes and frequencies of baseline ATS—Gear

Fig. 13—(top right) Finite element model, mode shapes, and frequencies of modified ATS—Gear



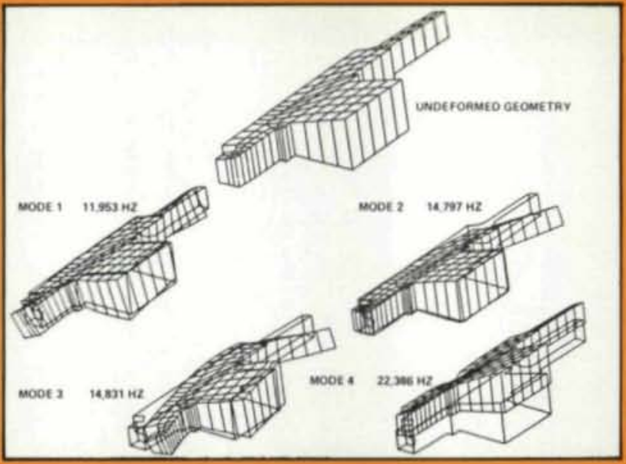


Fig. 12—(above left) Modified ATS Gear

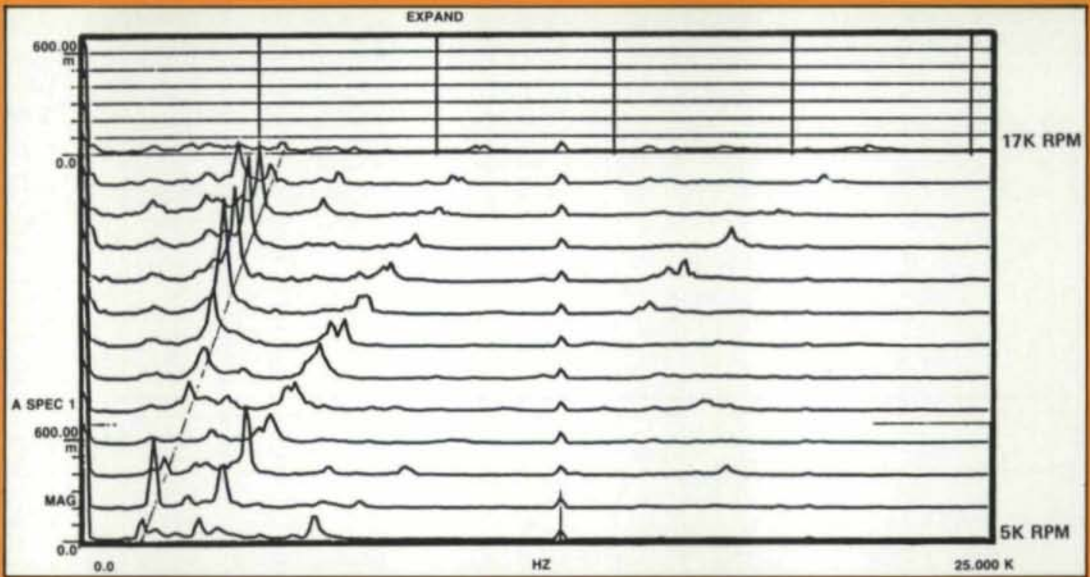


Fig. 14—(center) Waterfall diagram of drive with ATS—Gear removed

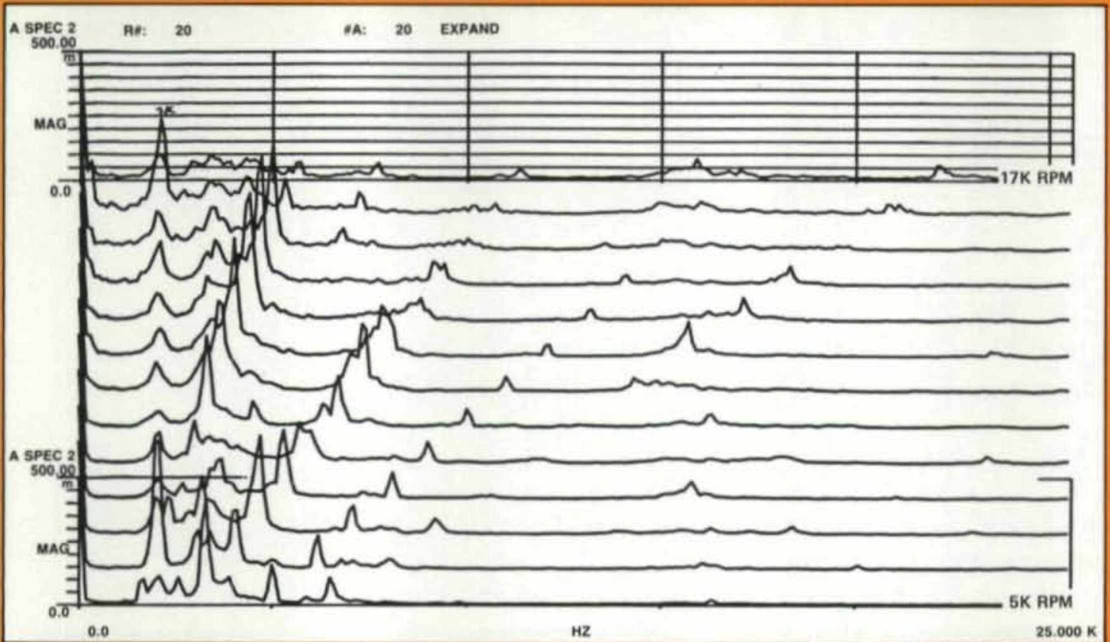


Fig. 15—(bottom left) Waterfall diagram of drive with modified ATS—Gear installed

the latter is an axisymmetric structure. The cyclic symmetry option in the program is, therefore, utilized and a plot of a 15° segment is reproduced in Fig. 10 with the hidden lines removed. The model is composed of 80 hexahedrons over 230 grid points. The model was constrained from rotation about the axis of rotation at one end of the spline. The analysis results in the plotted mode shapes and associated frequencies shown in Fig. 11. It is interesting to note that the fundamental frequency of torsion for the ATS gearshaft is 9.1 KHZ which coincides with the resonance observed in the waterfall diagram of Fig. 4.

Similar analysis of the VSCF gear showed that none of its modes coincided with the indicated resonance. Since the ATS gear has proven to be the culprit, a structural modification of the same is necessary. The object of the modification is to detune the subject gear, moving its first torsional frequency beyond the operating range (higher than 10.93 KHZ). A number of modified configurations were analyzed before a successful fix was reached. The recommended configuration entailed pressing a sleeve into the drive end of the gearshaft as shown in Fig. 12. The finite element model of the new gear and resulting mode shapes and frequencies are shown in Fig. 13. The NASTRAN program predicts the first torsional of the modified gear to be 11.95 KHZ, sufficiently above the operating range.

Verfication Testing of the Modified Configuration

To confirm the results of the above analysis, a waterfall diagram is generated for the subject drive with the ATS gear removed. Fig. 14 shows such a diagram. Indeed, as theoretically conjectured, the ATS gear must have been the resonating element in the drive because the amplitude along the corresponding excitiation order dies down with speed when the ATS gear is no longer in the gear train.

The modified gear was subsequently installed and another waterfall diagram developed as shown in Fig. 15. This clearly shows that the 9.1 KHZ resonance has now disappeared. The diagrams of Figs. 14 and 15 are almost identical proving the effectiveness of the detuning process.

Conclusions

The above investigation clearly illustrates the importance of performing a detailed modal analysis of the gear elements in a high speed drive. As can be seen, the alternatives may be very costly. A systematic procedure to avoid the damaging consequences of gear resonance may be outlined as follows:

1. Identify excitation mechanisms in the drive.
2. In the design stage, evaluate natural frequencies and mode shapes of individual gears using finite element techniques.
3. Eliminate torsional modes from operating range.
4. For helical and bevel gears, minimize axial vibrations by detuning and/or damping treatments.
5. Test gear blanks to verify analytical results.
6. Introduce additional fixes if necessary.
7. Generate waterfall diagrams of assembled drive.
8. Identify remaining resonant problems, if any, with other elements of the drive such as housings and accessories.
9. Correct remaining problems accordingly.
10. Generate new waterfall diagrams after all fixes have been instituted.

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Maximum Surface Temperature of the Thermoplastic Gear in a Non-Lubricated Plastic/Steel Gear Pair

by
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Abstract

One of the major problems of plastic gear design is the knowledge of their running temperature. Of special interest is the bulk temperature of the tooth to predict the fatigue life, and the peak temperature on the surface of the tooth to avert surface failure. This paper presents the results of an experimental method that uses an infrared radiometer to measure the temperature variation along the profile of a plastic gear tooth in operation. Measurements are made on 5.08, 3.17, 2.54, 2.12 mm module hob cut gears made from nylon 6-6, acetal and UHMWPE (Ultra High Molecular Weight Polyethylene). All the tests are made on a four square testing rig with thermoplastic/steel gear pairs where the plastic gear is the driver. Maximum temperature prediction curves obtained through statistical analysis of the results are presented and compared to data available from literature.

Introduction

It is common practice to rate thermoplastic gears by using the metal gear equation developed by Lewis[1] with correction factors to allow for differences in properties between metals and plastics. Although mechanical properties of plastics as a function of temperature are relatively well known,[2, 3, 4] they are of little help, if the temperature at which the gear operates, is not known. There is some information on how to calculate the bulk[5, 6] and the mean tooth surface temperatures.[6, 7, 8, 9] However, the maximum tooth surface temperature on the flank, which can be related to the flash temperature concept developed by Blok[10] for metal gears, is especially difficult to measure, and virtually no information is available in the literature concerning plastic gears.

This article presents the measurements made of the temperature variation along the tooth profile of a plastic gear in operation. The technique is similar to that used in a previous program to measure the mean tooth surface temperature.[8] For the present investigation, the plastic materials are unfilled and unreinforced polyamide 6-6 (nylon), polyoxymethylene homopolymer (acetal) and Ultra-High-Molecular-Weight-Polyethylene (U H.M.W.P.E.). Table 1 summarizes the geometry of the gears used. In all cases, the plastic gear acts as the pinion (driver) and is paired with a metal gear. The gears are run without lubrication on a four square testing rig.[7] A parametric equation, based on a large number of measurements, is proposed for each material investigated to predict the maximum temperature in the operating conditions described.

Analysis of Temperature Evolution Around Gear Contour

Neglecting heat flow through shafts and radiation from surroundings, heat generated in a gear pair containing at least one plastic gear has two sources: friction and viscoelastic hysteresis.

Fig. 1 represents, on a arbitrary scale, the ideal distribution of friction losses on a tooth segment, A-B-C-P-D-E, of a

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typical 24 tooth nylon gear in mesh with a steel gear. The calculations are done using the load sharing technique. [11] The conditions are pitch line velocity $V = 7.2$ m/sec, tangential load per unit face width $W_t = 70$ N/mm and no lubrication. A representative coefficient of friction of $\mu = 0.1$ is used for dry nylon on steel.[1]

Hysteresis losses are due to cyclic stressing of the viscoelastic material. Assuming a linear viscoelastic material, hysteresis losses per unit volume of material are, for unidirectional stressing:[12]

$$\frac{\tan(\delta)}{1 + \tan^2(\delta)} \quad \frac{\omega}{2} \quad \frac{\sigma_0^2}{E'} \tag{1}$$

where $\tan(\delta)$ is the loss factor; σ_0 is the maximum stress amplitude; E' is the storage modulus and ω the loading frequency.

Hysteresis losses in a volume of material to a depth of $p_b/8$ below the tooth surface (p_b is the base pitch) are also shown on the same scale in Fig. 1. Both types of stressing are represented in that figure, i.e. contact compression and bending. A value of $\tan(\delta) = 0.17$ is used for nylon at 50 percent relative humidity and 37°C.

Fig. 1 indicates that, theoretically, most of the heat generated in a plastic/steel gear pair comes from friction and contact compression hysteresis on the loaded flank. Bending hysteresis losses, shown on the non loaded profile, appear to be almost negligible. Since little heat is generated on the non loaded profile (region A to B in Fig. 1) and knowing that plastics are poor heat conductors, the temperature in this region of the tooth is expected to be relatively low. However, beyond point B, temperature may reach a peak in the vicinity of point C, decrease towards the operating pitch point P, increase again to another peak in the region of the lowest point of contact D to finally reach, at E, the same value as A.

In other words, assuming friction and hysteresis as being the only heat sources, the temperature distribution along a tooth profile may be expected to follow the pattern pictured

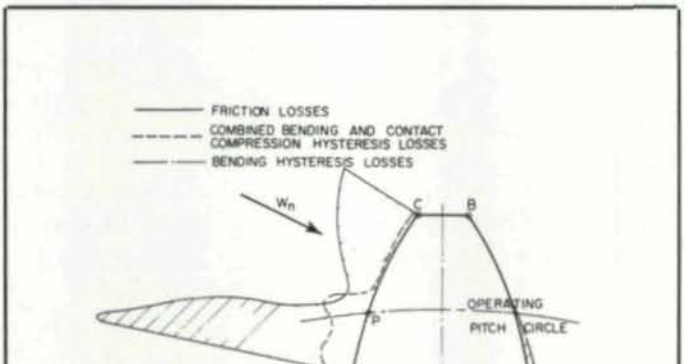


Fig. 1—Repartition of friction losses along gear profile.

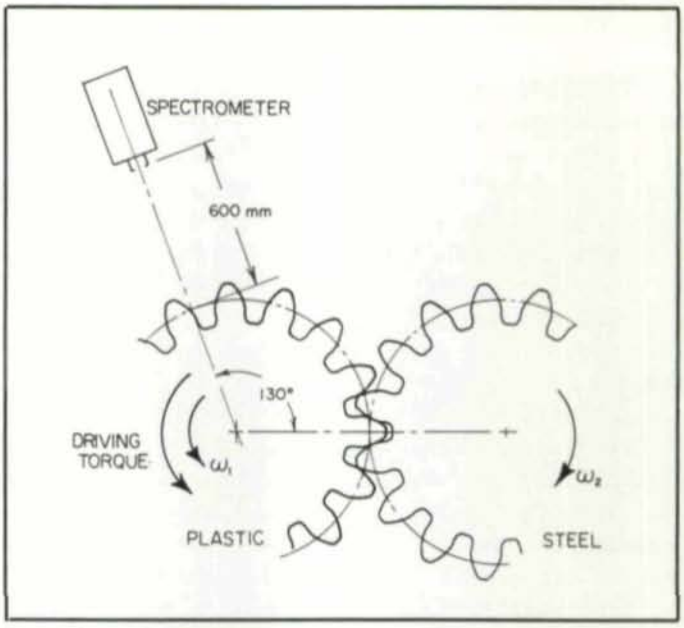


Fig. 2—Set-up for radiance measurements.

in Fig. 1. However, the peaks will be attenuated by conduction in the plastic gear tooth and the mating gear tooth, which is, in the present case, made of steel. Nevertheless, according to the theoretical evaluation of reference,[13] depending on load, module, speed, and mating gear material, temperature differences as much as 40°C may be recorded between region A-B and point C, 20°C between points C and P, and 50°C between P and D.

As Fig. 1 shows, the temperature gradients mentioned above occur over a relatively small surface on the tooth profile. From gear geometry and cinematic, it can be shown[13] that this surface decreases with the increasing number of teeth and speed. In the present investigation, the highest number of teeth in the plastic gear is 36 (2.116 mm module) and the highest speed is 9.2 m/sec. Noting that one complete cycle occurs between points B and D (Fig. 1), the frequency appears to be in the order of 3 KHz. This is the lowest frequency response characteristic the measuring instrument must have in order to detect the hottest points on the tooth profile.

Besides frequency response requirements, the measuring apparatus must be able to measure temperatures with adequate precision, have sufficient resolution to follow accurately temperature variations along the tooth profile, and sufficient depth of view to compensate for working distance variations due to the height of the teeth.

System Used for Measurements

Fig. 2 is a schematic representation of the measuring set-up. The measuring instrument is an infrared radiometer, which uses a nitrogen cooled indium antimonide detector, to provide a theoretical accuracy of .02°C on the calculated temperature, and can measure signals with a frequency up to 10 KHz.[14] This is well over the 3 KHz requirements stated in the preceding section. However, in the present application, such performances cannot be obtained mainly because of variations in the gear emissivity, which translate into an error on temperature, and the diameter of the measuring spot, which rounds off temperature peaks.

The experimental error on temperature is evaluated from the standard deviation of emissivity measurements made on new and worn gears of each material. From these measurements, it appears that the maximum error on temperature lies within limits set as ±1.5°C at 20°C and ±3C at 180°C. This error includes a variation of emissivity from gear to gear, which accounts for about 70 percent of the total, and a smaller error from tooth to tooth, which accounts for the rest. Therefore, for a given gear, it can be said that the maximum error at 180°C is in the order of ±1°C.

The size of the measuring spot is directly related to the field of view, which represents the solid angle that the target subtends, as viewed from the apparatus, and is defined by an unvariable field aperture built into the apparatus. Theoretically, the particular apparatus used has a conical field of view of 2.5 milliradians. This means that the target should be a circle of diameter equal to the base of a cone having an apex angle of 2.5 milliradians, and a height equal to the operating distance. This corresponds to a measuring spot of 1.5 mm at a working distance of 600 mm. Also, theoretically the detector output should be directly proportional to all of the radiation emitted from the surface of the circular target.

In practice, this does not appear to be exactly true; the actual shape of the target, field of view, and response curve of the detector, as given by the manufacturer,[14] are presented in Fig. 3. This figure shows that the actual field of view is an elliptical cone with minor and major axes of 3.0 and 5.0 milliradians respectively. For measurements reported in this paper, minor and major axes of the target are respectively aligned perpendicular and parallel to the axis of rotation of the gear.

If the response curve of the detector is approximated by a sinusoid, and the output signal is analyzed by Fourier decomposition, the original signal can, theoretically, be

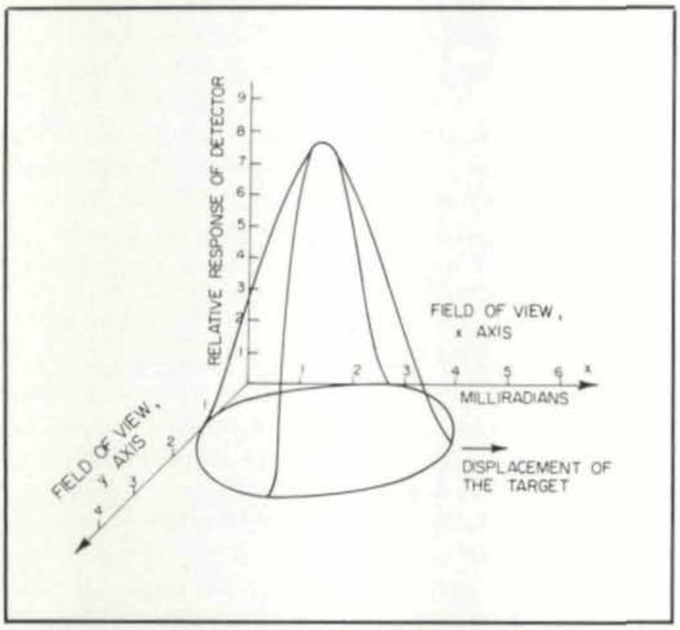


Fig. 3—Shape of the measuring spot and relative response of the detector.

reconstituted by a computer program[15] to simulate a punctual measuring spot. However, this method is limited by random noise present in the signal. As the measuring spot size is further reduced by the Fourier decomposition, the noise becomes more and more important in relation to the temperature signal. Practical considerations limit the lower bound of the simulated spot size to approximately .5 mm. This is a threefold increase in the original resolution of 1.5 mm.

The depth view is measured as about 10 mm, which is adequate for a 5.08 mm module gear. The system used answers the requirements outlined in section 2.

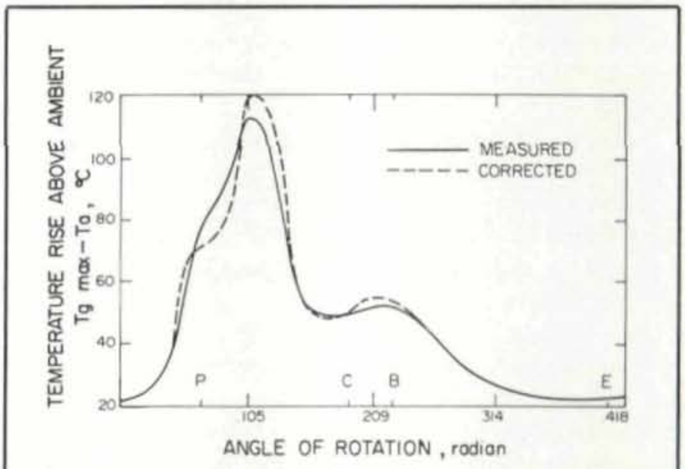


Fig. 4—Measured and corrected temperature signal for a used nylon. 5.08 mm module gear, Wt = 38 N/mm, V = 9.6 m/s.

For example, Fig. 4 presents the orginal and corrected temperature signals obtained with a used 5.08 mm module nylon gear. Although the variation of the maximum temperature is small, the shape of the signal is significantly altered, with peaks and throughs amplified, as can be expected with a smaller measuring spot. In this particular case, the correction is in the order of 8°C on the peak temperature (110°C). This is representative of the correction on temperatures of this order of magnitude.

Fig. 5 shows the corrected temperature signal for a 5.08 mm module UHMWPE gear. On this particular tooth, a reflective aluminum foil of low emissivity is installed on the top land of the tooth (C-D in Fig. 1) in order to precisely correlate the measured signal with specific points on the gear tooth surface. This explains the difference of the shape of the signals between Figs. 4 and 5.

Comparison of heat generation as calculated (Fig. 1) and temperature signals, as measured (Fig. 5) shows that the main temperature peak, which is predicted to be below the pitch circle by theoretical analysis, actually lies above the pitch circle for this particular signal. In fact, the shape of the temperature signal varies widely from one material to the other. In general, it is observed that the maximum temperature peak first appears below the pitch circle as predicted by the theory, when the gear is new; but, as the gear wears, two peaks of different amplitude, which could

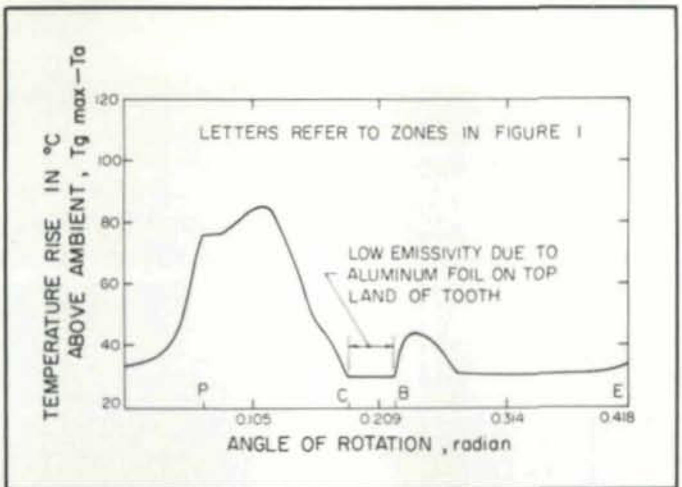


Fig. 5—Corrected temperature signal with aluminised land on top of tooth, 5.08 mm module UHMWPE, Wt = 28 N/mm, V = 9.6 m/s.

vary from tooth to tooth, start to develop gradually. This is attributed to the degradation of the profiles with wear and, the consequent departure from their exact shape with which the theoretical calculations are done.

Results and Discussion

Some two hundred thirty measurements have been done with the radiometer. These were made with several conditions of load and speed for each material and modules. The plastic-metal gear pairs were first run until the operating temperature stabilizes. As mentioned previously, the measurements are taken on the plastic gear (driver) 130 degrees after the operating pitch point. The experimental results are then corrected for spot diameter, and they are statistically analyzed by computer to derive a parametric expression to match the maximum surface temperature as a function of the operating parameters. Of several models

(Continued on page 26)

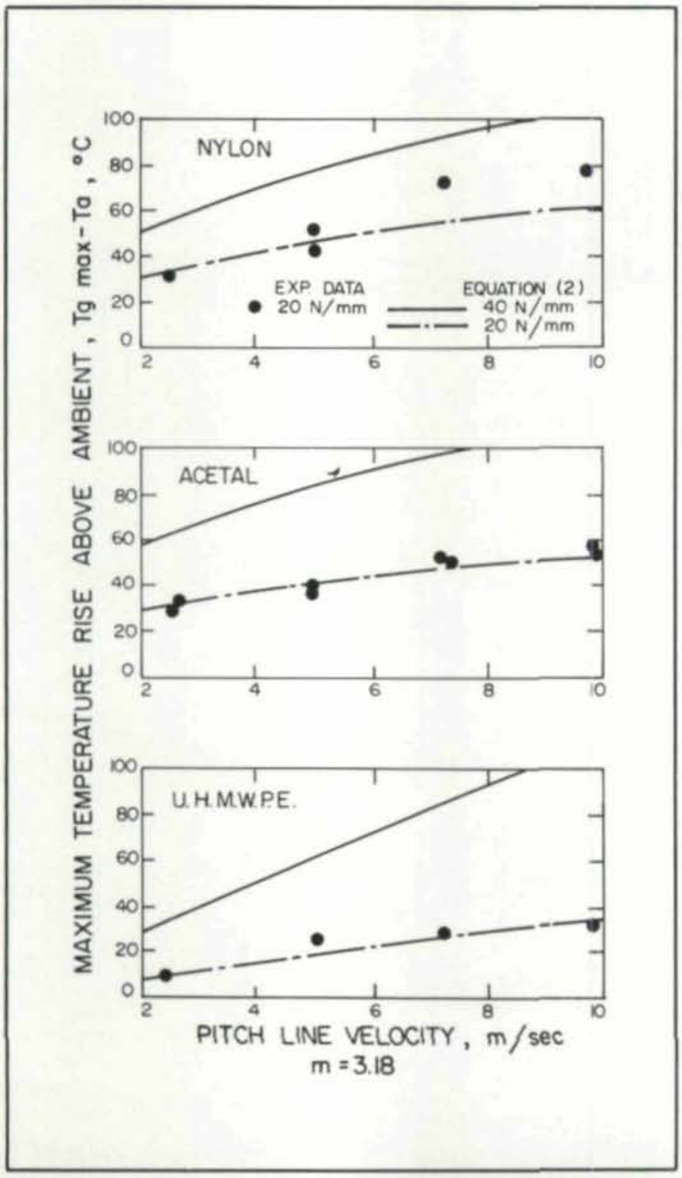


Fig. 6—Predicted maximum temperature rise above ambient ($tg_{max} - Ta$) as a function of pitch line velocity (V) with two different loads, three materials, all: 8 pitch (3.175 module)

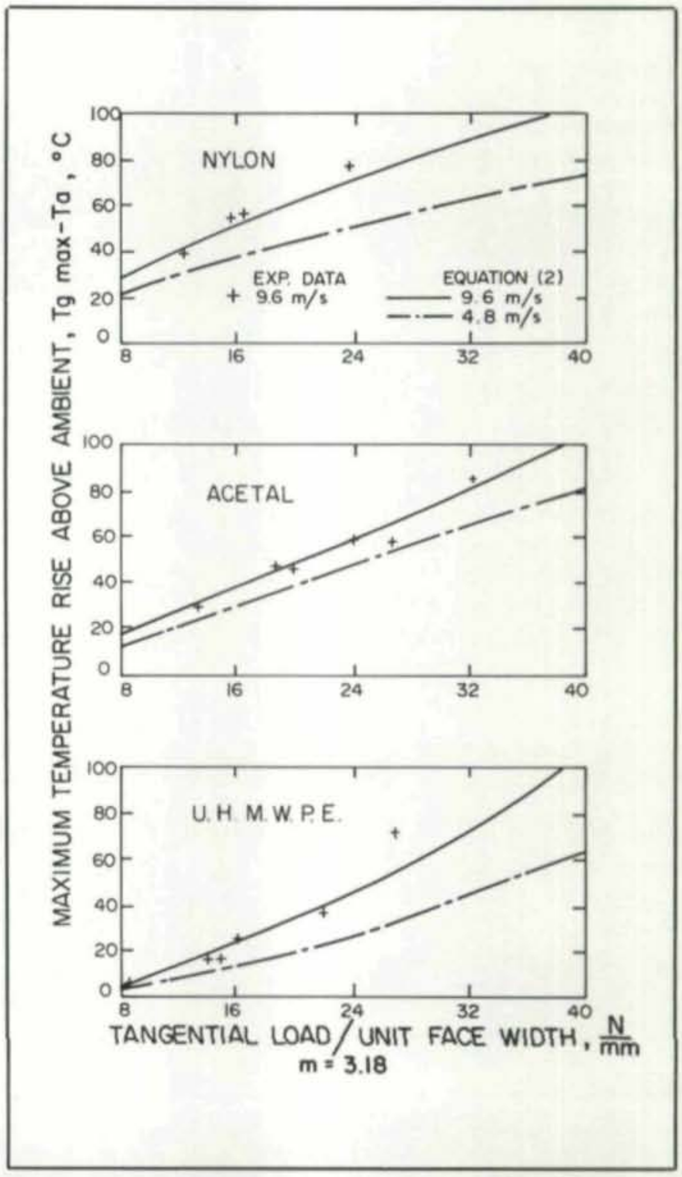


Fig. 7—Predicted Maximum temperature rise above ambient ($Tg_{max} - Ta$) as a function of the tangential load per unit face width (Wt) for two different speeds. Three materials, all: 8 pitch (3.175 module)

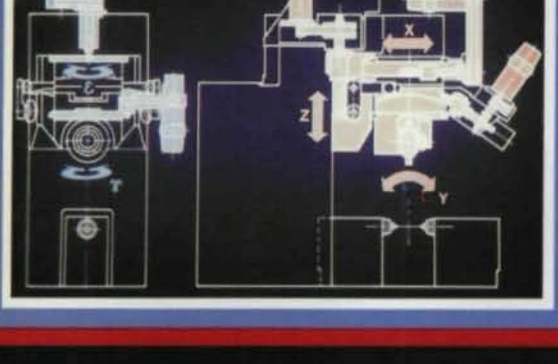
HURTH ZS 350 and ZS 550 CNC Gear Shaving Machines—All The Design And Engineering Aspects Necessary For High-Precision Gear Manufacturing

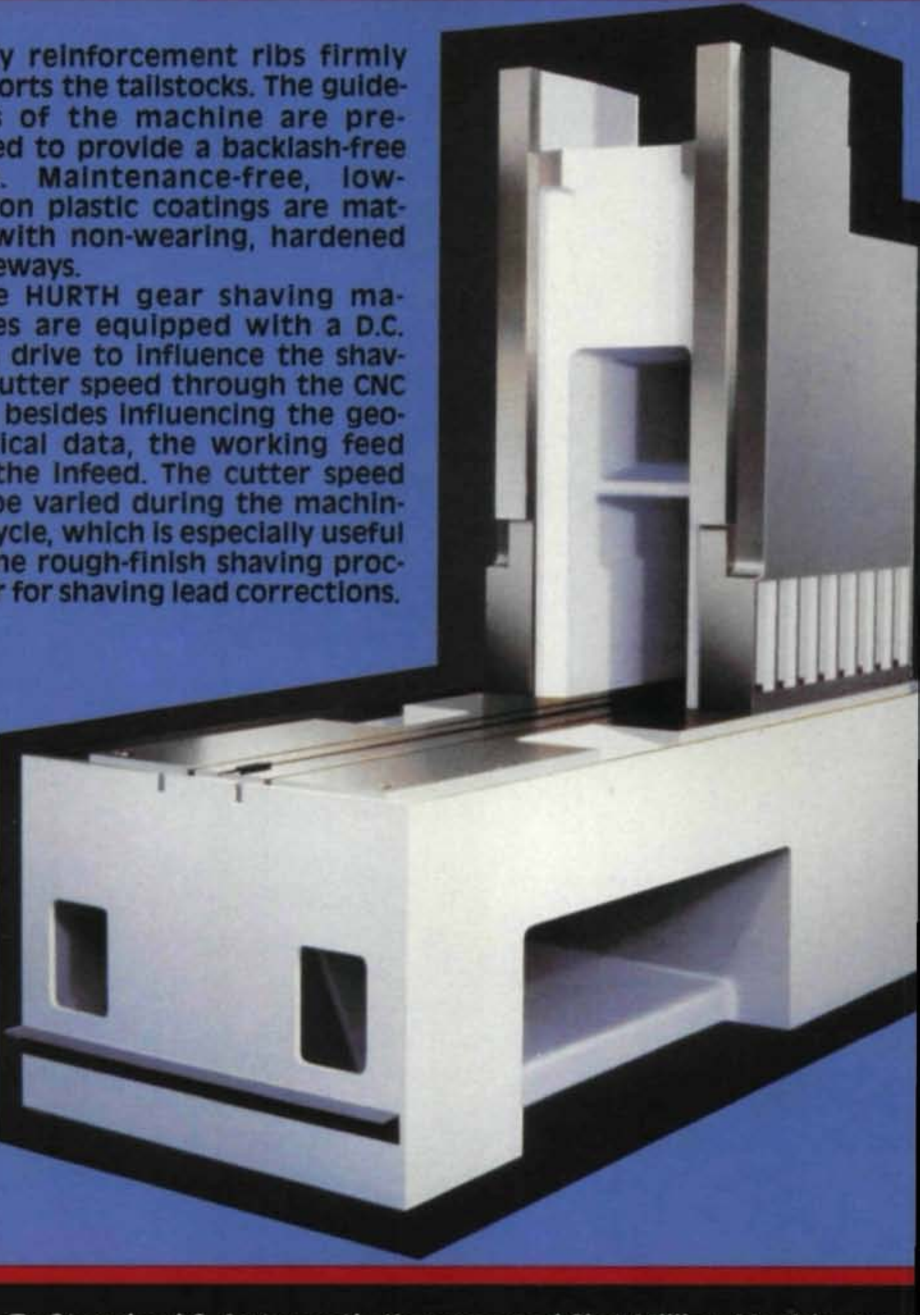
Gear shaving with CNC is a big step toward gear finishing with manufacturing flexibility. Whatever your gear finishing and manufacturing requirements, HURTH provides the means to solve the task with the ZS 350 and ZS 550.

In addition to short machining times, fast machine changeover is of extreme importance for the economical manufacturing of small and medium quantities. To achieve this goal, the CNC unit controls 5 axes of the HURTH ZS 350/550 with 3 working axes and 2 set-up axes. These control systems allow digital data input of the shaving feed, infeed, crown/taper shaving as well as the setting of the diagonal and crossed-axis angle.

The HURTH ZS 350 and ZS 550 gear shaving machines feature a machine bed with a fixed column. The box-type bed casting with its heavy reinforcement ribs firmly supports the tailstocks. The guideways of the machine are preloaded to provide a backlash-free slide. Maintenance-free, low-friction plastic coatings are matted with non-wearing, hardened guideways.

The HURTH gear shaving machines are equipped with a D.C. main drive to influence the shaving cutter speed through the CNC unit, besides influencing the geometrical data, the working feed and the infeed. The cutter speed can be varied during the machining cycle, which is especially useful for the rough-finish shaving process or for shaving lead corrections.





HURTH MACHINE TOOLS: *Standard & Automatic Keyway and Slot Milling Machines • A Hobbing Machines • CNC Gear Shaving Machines • Shaving Cutter Grinding Machines • Ge Hard Gear Finishing Machines •* **STAR TOOLS:** *Hobs • Form-Relief Cutters • Gun Drills • Gu ard and CNC Hob Sharpeners • Shaper Cutter Sharpeners • Standard Gun Drilling Machines*

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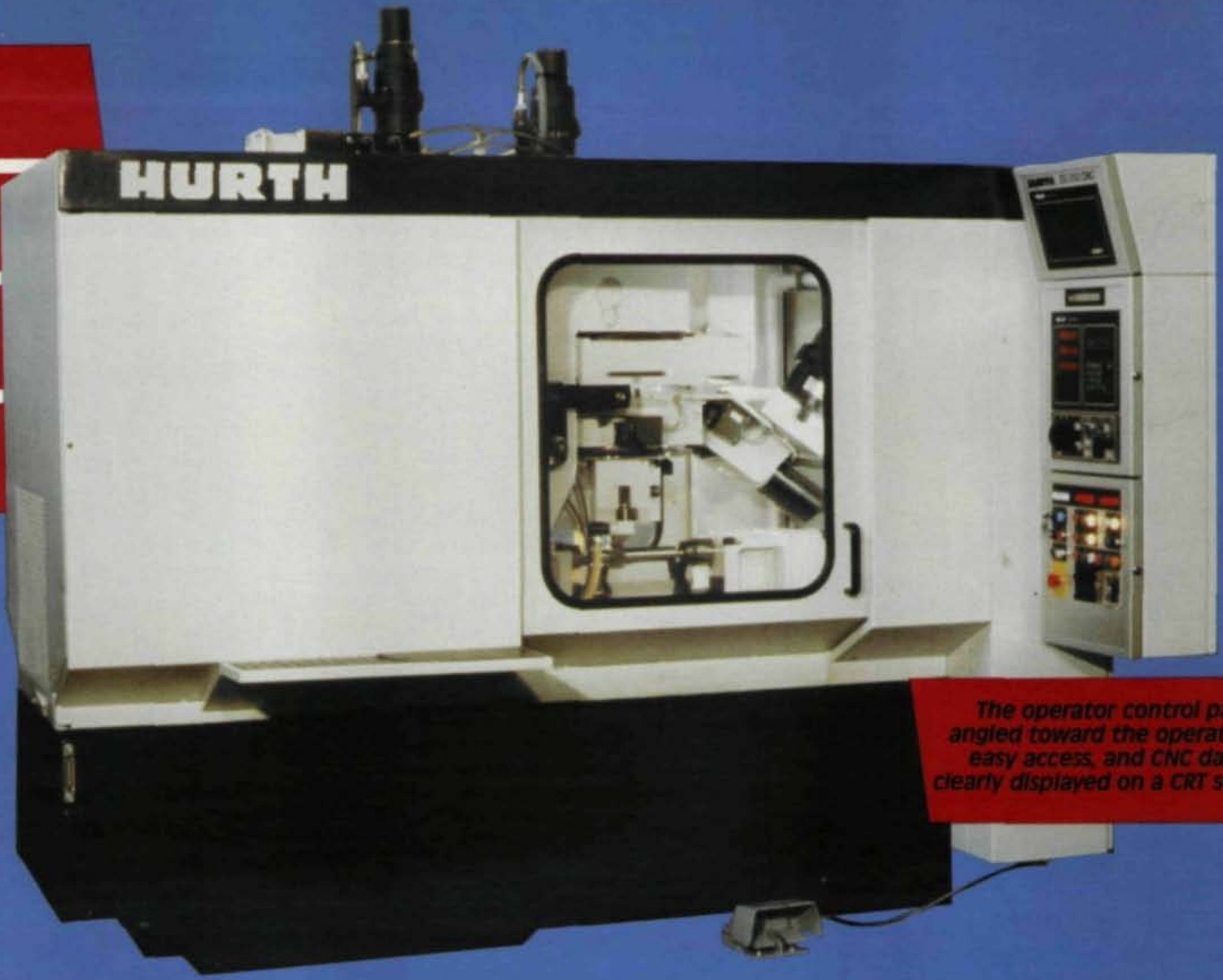

The operator control panel is angled toward the operator for easy access, and CNC data are clearly displayed on a CRT screen.

The operator control panel has conveniently grouped machine setting controls inclined toward the operator for easy access. The upper panel accommodates the controls for pre-selecting the mode of operation, the shaving cycle and machining parameters.

For all-around manufacturing flexibility and precision gear shaving, HURTH ZS 350 and ZS 550 machines perform.

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tried, the following gives the best fit:

$$(T_g \max - T_a) = b_0 W_t^{b_1} V^{b_2} m^{b_3} \qquad (2)$$

where T_g max T_a are expressed in °C and b_0, b_1, b_2, b_b are regression coefficients calculated for each material and given in table 2.

Using the values of regression coefficients from table 2 and equation (2), curves are drawn on Figs. 6, 7 and 8 for the three materials studied and typical operating parameters. Some of the experimental results, for which the operating parameters were similar to those chosen to draw the curves, are given for reference purposes.

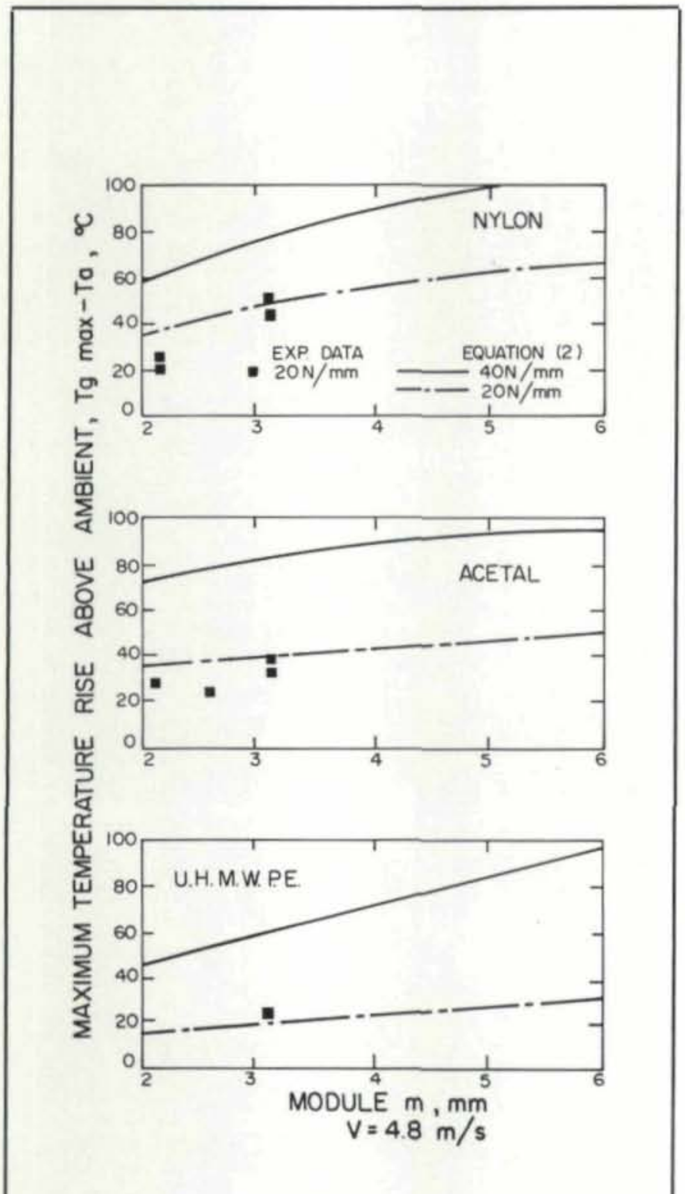

Fig. 8—Predicted Maximum temperature rise above ambient ($Tg_{max} - Ta$) as a function of the module m for two different loads. Three materials, all 4.8 m/s pitch line velocity.

As expected, the temperature increases with increasing velocity, load and module (coarser). However, their respective influence varies from one material to another and this reflects on values of the coefficients in table 2. While an exponent smaller than 1.0 produces a convex curve, Figs. 6 and 8, an exponent greater than one produces a concave curve. This is illustrated by comparing curves for UHMWPE to these for acetal or nylon in Fig. 7. The fact that UHMWPE behaves differently than the two other materials might be related to its peculiar properties. As a matter of fact, nylon and acetal are viscous liquids above their melting points, while UHMWPE is so high in molecular weight that it does not really melt and behaves more like a rubber at elevated temperatures. Since the maximum temperature generated by friction along the tooth profile can exceed the melting temperature of the material, a reduction in the friction coefficient for nylon and acetal can be expected, because of the viscous liquid generated which moderates the rate of increase in temperature. But no such liquid is generated for UHMWPE, and temperature rises more and more steeply with load with a corresponding increase in tooth flexibility and degradation of meshing qualities.

It must be understood that equation (2) is derived for a certain range of operating parameters which can be obtained from the various axis of Figs. 6 to 8. They are: pitch

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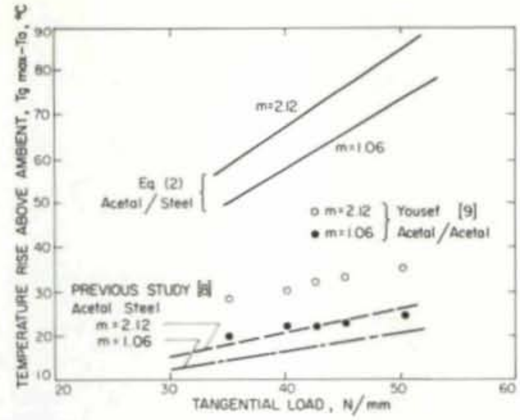

Fig. 9—Comparison of mean tooth surface temperature from ref. [8] and [9] with maximum tooth surface temperature obtained with eq (2).

line velocity from 2 to 10 m/sec, tangential load 8 to 40 N/mm, and modules 2.11, 2.54, 3.175 and 5.08 mm. Even though there is no reason to believe that the general behavior of the parameter would change outside these limits, caution must be takin in extending the use of equation (2).

Youssef et al.[9] *measured the mean tooth surface* temperature of running gears with an infrared microscope for an acetal/acetal contact. Their results, along with those of a previous study[8] for mean tooth surface temperature of an acetal/steel contact, are compared in Fig. 9, with the maximum tooth surface temperature, as calculated by equation (2) for an acetal/steel contact. Comparable values for mean tooth surface temperature are obtained from both references, even though one is for acetal/acetal, while the other one, which gives the lower values as expected, is for acetal/steel contact. As one can see, however, maximum tooth surface temperature can be considerably higher than mean tooth surface temperature.

Conclusion

The infrared radiometer permits to measure the temperature peaks on the profile of running gears. A considerable amount of data was collected and analyzed by computer to come up with an equation to predict the maximum tooth surface temperature as a function of the operating parameters.

The results show the influence of the material, as well as that of tangential load, speed, and gear module. By comparison with published data for mean tooth surface temperature, it is seen that the maximum temperature on the tooth surface might be considerably higher than the mean surface temperature, which is usually considered in gear analysis.

Nomenclature

b_0, b_1, b_2, b_3 regression coefficients to use in equation (2)
b gear face width, mm (inch)
d_p pitch diameter, mm (inch)
E' storage modulus of a viscoelastic material, MPa (lb/inch^2)
m module, mm
P pitch, (diametral)
p_b normal base pitch of a spur gear, mm (inch)
$\tan(\partial)$ hysteresis loss factor for a viscoelastic material
T_a ambient temperature, °C (°F)
T_{gm} mean temperature on the surface of the plastic tooth, °C (°F)
$T_{g\,max}$ maximum temperature on the surface of the plastic tooth, °C (°F)
V pitch line velocity, m/sec (ft/min)
W_t tangential load per unit of face width, N/mm (lb/inch)
$_tY$ Lewis tooth form factor determined with load applied near the middle of the tooth
Z_1, Z_2 number of teeth in pinion (driver) and driven gear respectively
α cutting pressure angle deg.
ϵ plastic gear material emissivity
μ coefficient of friction between contacting profiles
ω angular frequency of loading
σ_o maximum stress amplitude

(Continued on page 46)

Economics of CNC Gear Gashing Vs. Large D.P. Hobbing

by
Joseph W. Coniglio
Vice President, Engineering
Gould & Eberhardt Gear Machinery Co.

Gashing

Gear gashing is a gear machining process, very much like gear milling, utilizing the principle of cutting one or more tooth (or tooth space) at a time. The term "GASHING" today applies to the roughing, or roughing and finishing, of coarse diametral pitch gears and sprockets. Manufacturing these large coarse gears by conventional methods of rough and finish hobbing can lead to very long machining cycles and uneconomical machine utilization.

Gear gashing *is* gear milling, but the term "GASHING" is used to designate the application of using gear milling machines and tools designed with today's technology in mind and in effect (Fig. 1). Gear cutting tool development has taken a "back seat" to the advances in modern tooling technology. The advent of carbide and numerical controls has led to complete tooling systems for all other types of machine tools. The acceptance, use, and total dependence on carbide tools, has proven mandatory to meet the production demands of today. As a result of the "back seat" position, the gear cutting industry two years ago was far behind all other machine tools in all applications of machine design and tooling technology.

Gear Milling

Gear milling is still alive in both small and large shops. Most of them, however, are using 40 to 50 year-old underpowered mechanical machines with undersize tool arbors and high-speed cutters designed more than a generation ago. They are on our production floors nibbling away, creeping through material at a snail's pace and increasing manufacturing costs.

Today's millers are GASHERS, powerful milling machines with Numerically Controlled axes, oversize slide ways,

AUTHOR:

MR. JOSEPH CONIGLIO is Vice President of Engineering at Gould & Eberhardt Gear Machinery Corporation. He studied Mechanical Engineering at Union College in upstate New York and Newark College of Engineering. Since 1962, he has been involved in precision and automation of machine tools with emphasis on material removal and the application of numerical controls. His duties at G&E include responsibility for product design and development of their CNC gear "gashers" and CNC gear hobbers. His articles on gear "gashing" and milling have been included in the latest edition of the SME handbook and "Organi de Transmissione" published in Italy.

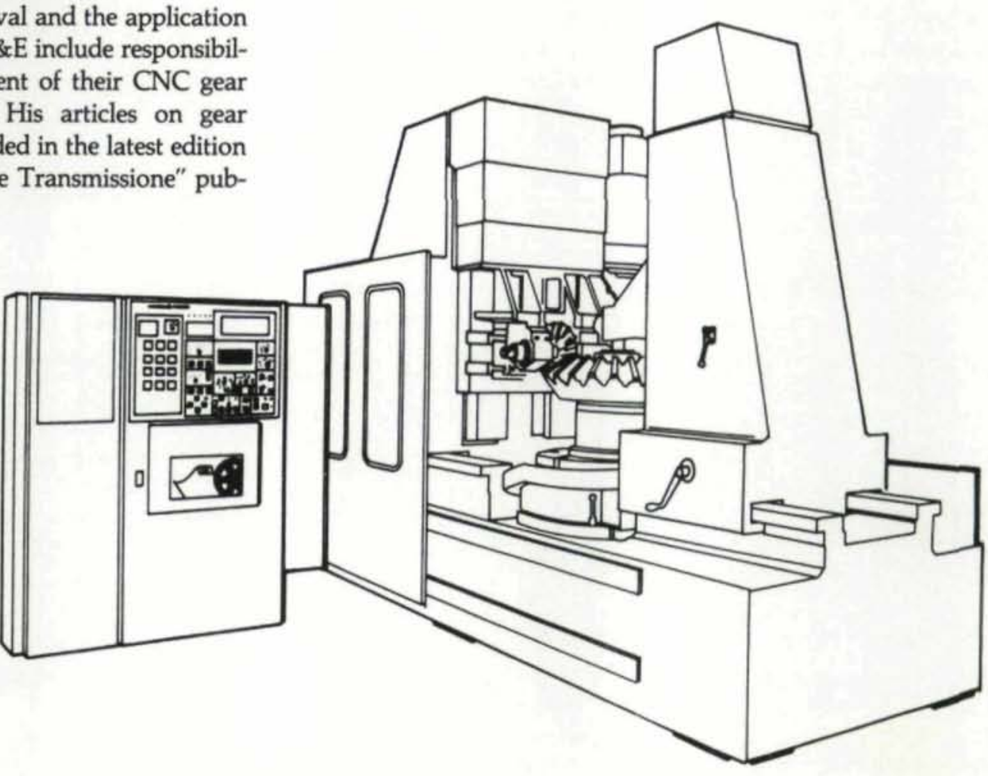

FIG. 1—The Gasher

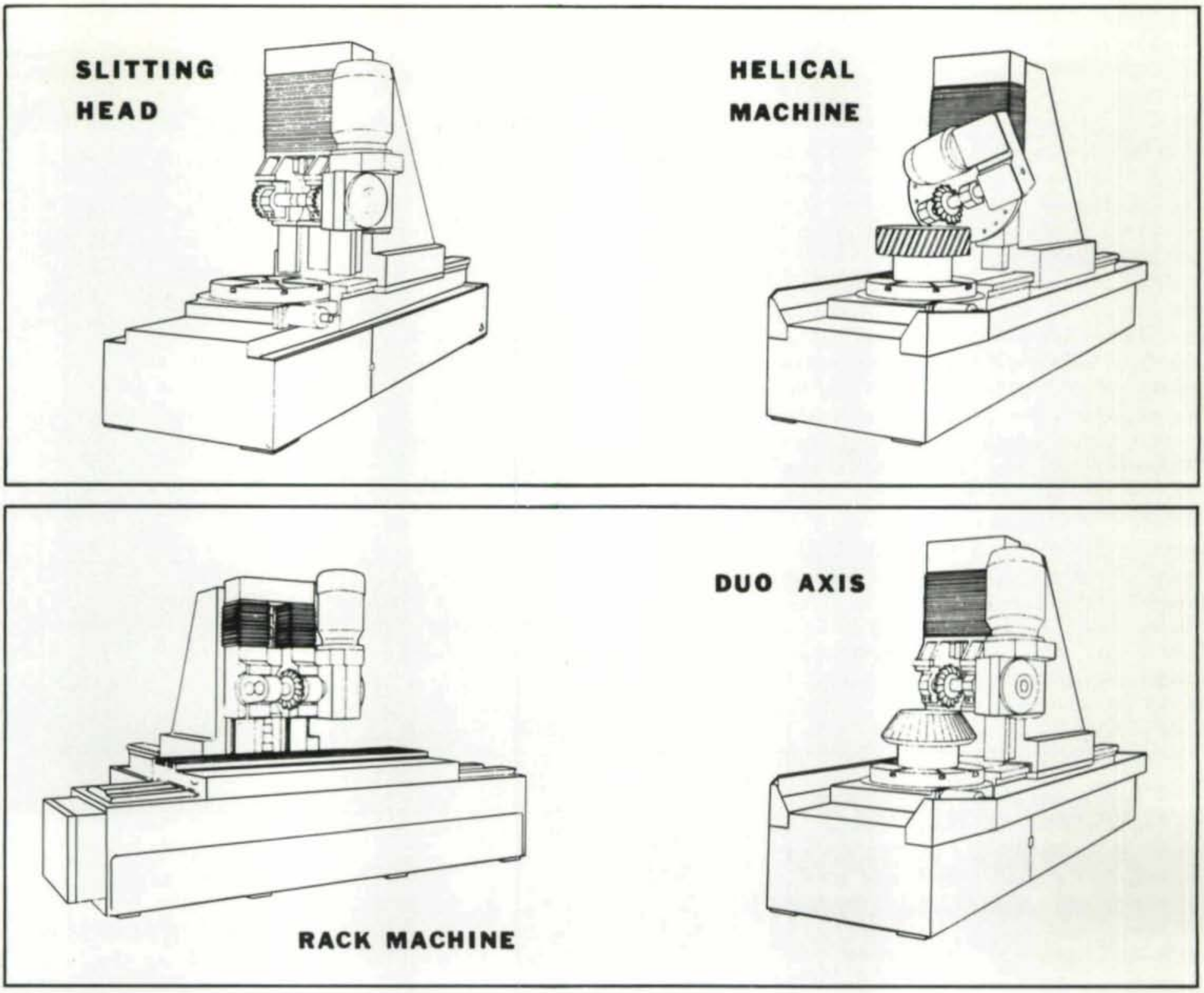

FIG. 2

50-100 HP D.C. spindle drives with carbide speed ranges, and super accurate indexing work tables.

Gashers are built in four different configurations: See Fig. 2.

1. Spur gear and roller chain sprocket machines
2. Spur and helical machines
3. Spur and straight bevel machines
4. Rack cutting machines

All the machines, with the exception of the rack cutter, can be tooled to become slitter/gashers. The differences in the various types of gashers are accommodated in the control or control software.

Basic Economics

The economics of using the new GASHERS fall into the following areas: lower cutter cost per workpiece by use of inserted carbide cutters, ability to cut harder material more efficiently, time savings, increased production on other machines used to finish the workpiece, if necessary, and increased cutter life on the finishing equipment. Predictable cutting times permit more accurate work flow through the gear shop with proper machine utilization. One gasher, set up for roughing, can feed many hobbers and other types of gear finishing equipment.

Cutting Tools

Any material that can be milled by conventional methods can be "Gashed." The use of carbide and inserted carbide cutters permit cutting material in the upper recommended range of material removal charts. Harder materials, of course, pose no problem for the carbide cutters. The carbide cutters cut dry, without coolant, permitting cleaner cutting conditions, no oil contamination of workpiece, and cost reduction in handling clean chips. In some instances, it is advisable to direct a jet of compressed air on the inserts, during cutting, to dislodge any hair-like chips that may cling to the cutting edges. These chips are generally workhardened and can cause hairline cracks in the cutting edges, and ultimately the failure of the cutting edge.

In large diametral pitch gear cutting, the process of roughing with inserted carbide cutters has proven to be the most efficient of Gasher machine utilization.

Rough Gashing The Involute Form

In Fig. 3, we see the involute form of a gear tooth. The straight sided carbide inserts are superimposed to illustrate the "roughed out shape." The inserted carbide cutter, that roughs the "V" shape form, is shown in Fig. 4. Its mounting and use are also shown.

This cutter is efficiently used in the 300-375 SFM range. The normal feed range is between 12 and 22 inches per minute. The positioning of the carbide inserts is such that, even at the higher feed rates, the inserts have a unique chip thinning effect, and although the chips are large, they are thin. In using a carbide cutter of this type, the gear can be either finish hobbed, shaped, or milled to final involute form, and can be efficiently completed in one pass by the finishing machine. The accuracy of index of the gasher worktable permits a minimum of finishing stock to be allowed. The resultant effect on the finishing tools, reported by the users of this system, is the ability to obtain, as much as, twelve times longer tool life. Completion of the rough gashed blank can also be done on the gasher by using a form ground involute milling cutter. Again, the accuracy of the work table permits finishing to size and form without sacrifice of tooth to tooth spacing. Unfortunately, almost all the finish form involute milling cutters currently available are of H.S.S. Some brazed carbide cutters are around, but require grinding equipment that most shops do not have.

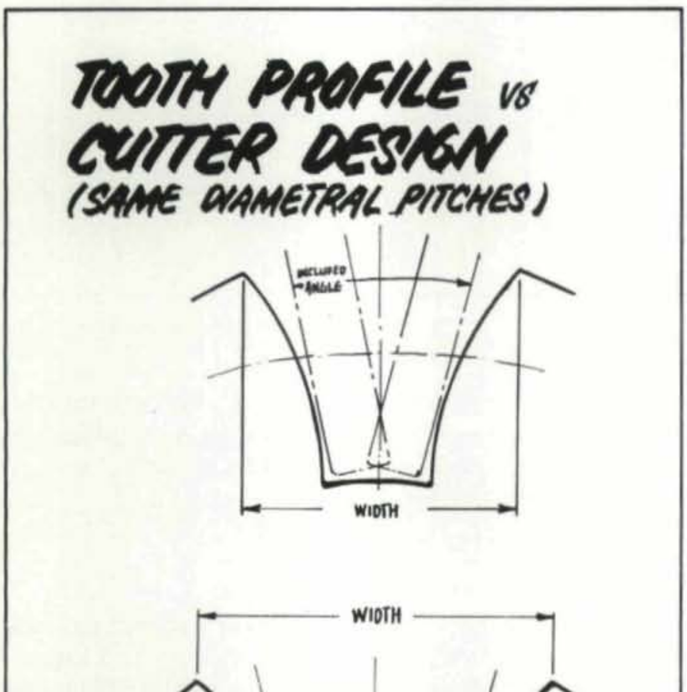

FIG. 3—(Courtesy: Ingersol Cutting Tool Co.)

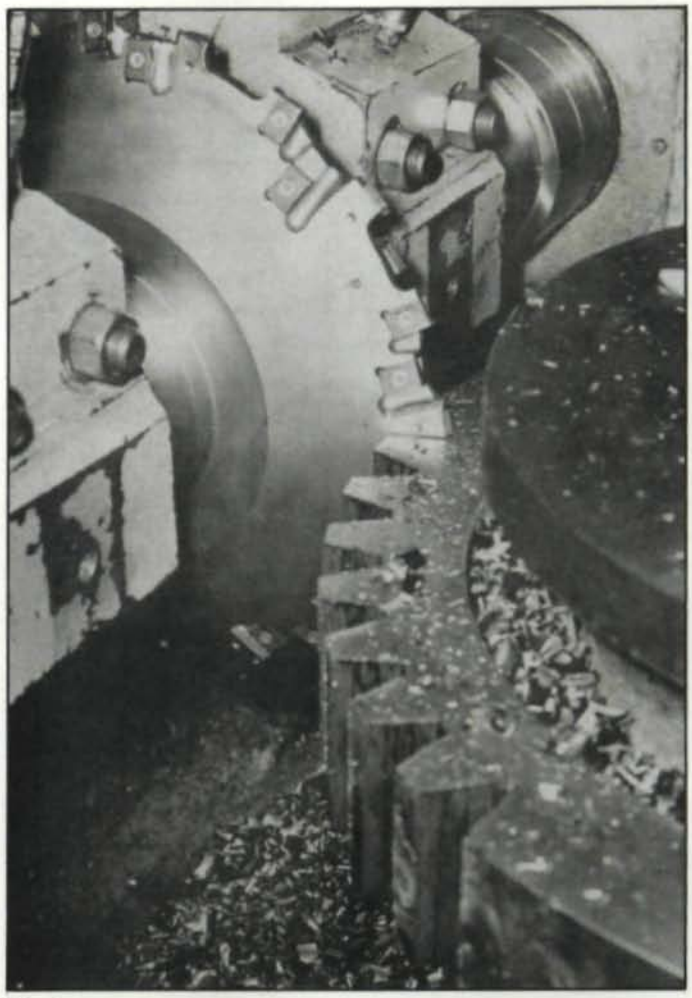

FIG. 4

Spur Gears

For example: in time savings only, for a mild steel gear 44 tooth, spur gear of 1 DP, and 5" face width; hobbed, two passes, 9" diameter hob, 100 SFM, .120 IPR roughing and .06 IPR finishing, the cutting time is a total of 229.8 minutes (3.83 hours) of which 153.2 minutes is the finish hobbing time.

If the gear is *rough gashed* at 400 SFM and 18 inches/minute feed, rate the over-all gear cutting time is reduced to 186.4 minutes — basically, a nominal savings of 40.4 minutes. Generally, we are not favored with mild steel as a gear material when cutting coarse pitch gears. Harder and tougher gear materials would, substantially, increase the rough hobbing time (Fig. 5). However, if we were to mill the gear by rough gashing at approximately 18 IPM feed rate; and then finish milled at a slightly reduced feed rate, say 10 inches per minute, we would now see an overall machining time of 86.6 minutes, a savings of 100 minutes per gear for rough gashing/finish hobbing.

Helical gears could be similarly processed, but at this time most roughed helical gears are finish hobbed or shaped rather

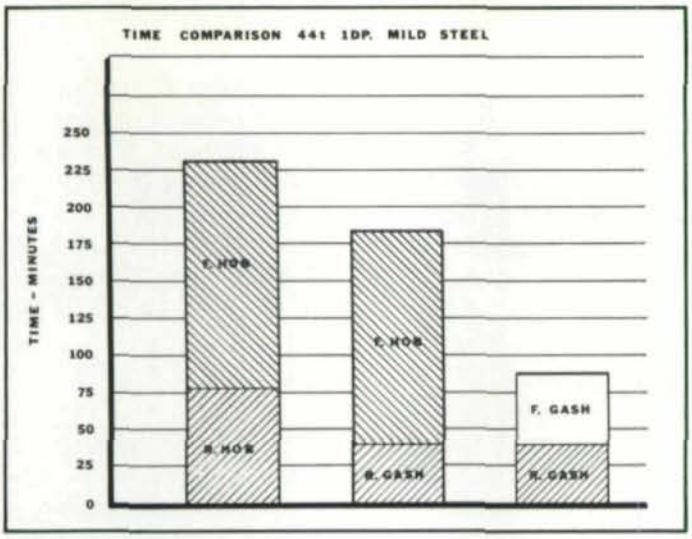

FIG. 5

than finish milled. Test data on helicals, at this time, is almost non-existent, because only a few gashers so equipped have been produced.

Bevel Gears

The gear gasher cutter shown in Fig. 4 is, also, used to rough gash straight bevel gears. The process used today is somewhat different from what most shops use, or have used, in the past. Older gear milling machines had indexing work tables that pivoted up, so the root angle could be generated by the vertical travel of the work slide. This created a very compromising situation where rigidity suffered and feeds and speeds disintegrated to nothingness. Today, through the use of controls capable of linear interpolation, the workpiece and table remain in the horizontal plane and the root angle is generated by the control.

The actual figures for previous methods of straight bevel roughing are not available, but data regarding rough gashing is available. For example: a 1 DP straight bevel gear, 25 teeth, material SAE 4340 forging, 340 BHN, gashed 1.8" depth of cut, 5.5 gear face, 14" diameter inserted carbide cutter at 90 RPM 14 IPM, took only 25 minutes cutting time. The machine set-up time was, also, greatly reduced from previous methods. Utilizing the gasher for roughing bevel gears leaves the finishing machine, the bevel generator, free to do what it was designed to do, produce the proper involute shape. The root angle/face angle effect of tapering the tooth encourages one more possibility in cutting straight bevels. The ability of the control to drive the cutting axis at variable feed rates permits the programmer to plunge into the workpiece at one feed, drive the root angle at the next feed (generally at 200-300% of plunge), and then exit the cut at a third rate (150-200% of second feed). The change in feed is accomplished by modern controllers without any hesitation in the motion of the cutter slide.

The programming ability of linear interpolation permits rapid moves that take the shortest path between two points. For lessening cutter shock when entering a workpiece, especially on bevel gears, we can program the plunge cut perpendicular to the root angle, thereby, reducing the tangent angle

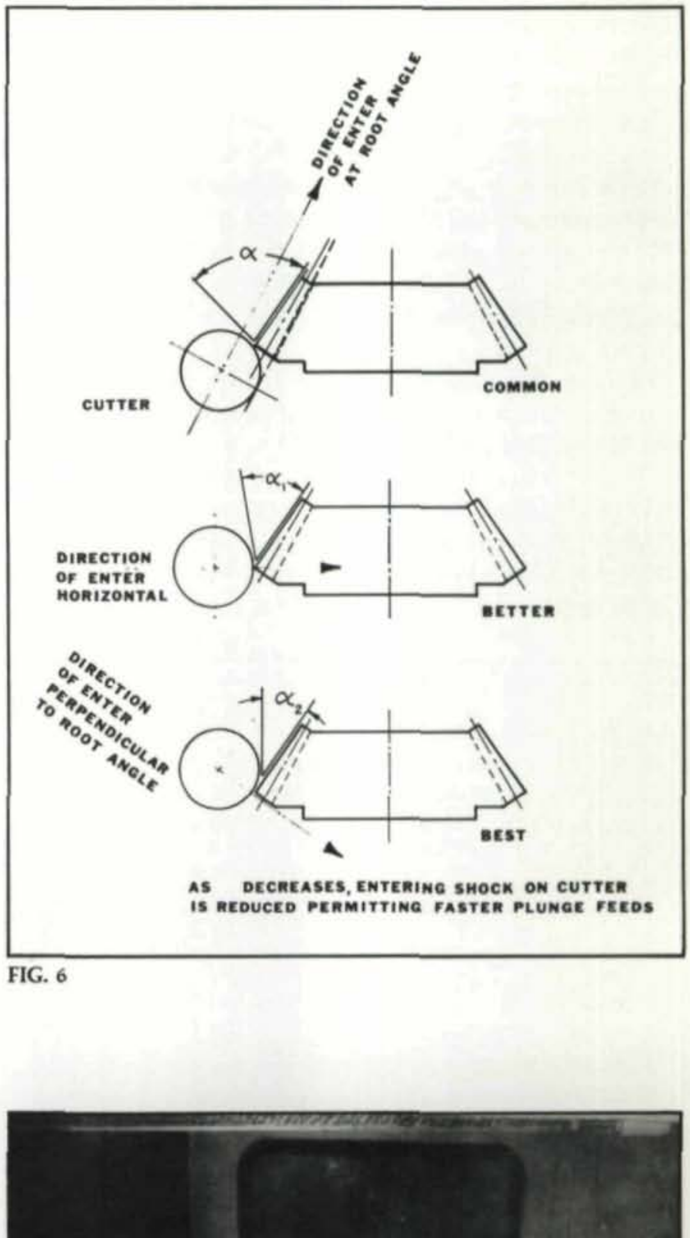

FIG. 6

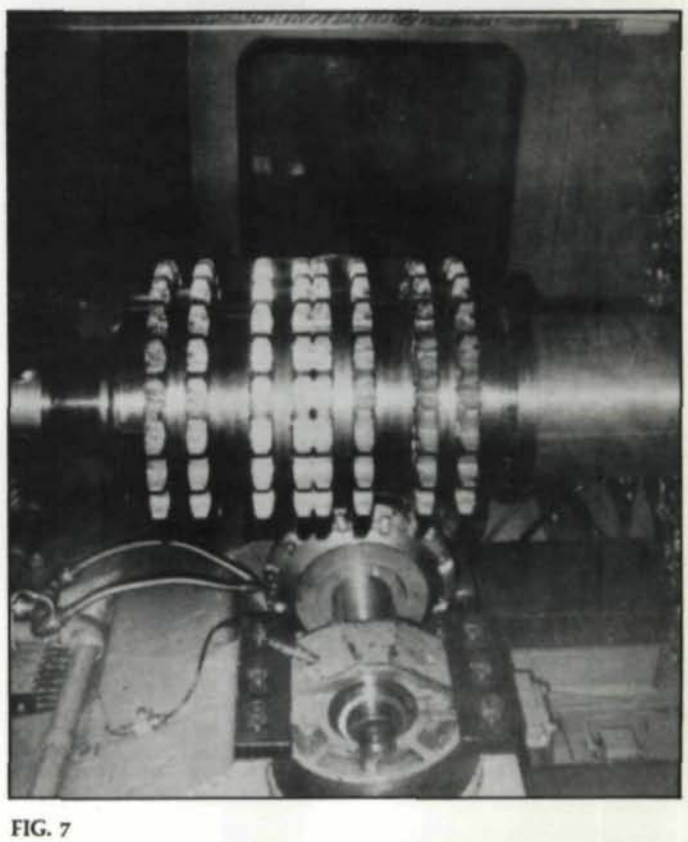
FIG. 7

between the workpiece and the top rake of the cutter teeth (Fig. 6). This gives longer cutter life by reducing the possibility of edge fractures or chipping of the cutter teeth.

Microprocessor equipped gashers and the linear interpolation of bevel gear angles generate very accurate angles. For example: the root angle tolerance on approximately an 8.5 inch face straight bevel would be about ± .002°.

The machine flexibility, created by the control and the non-pivoting work table, allows the methods people to switch readily from spur to bevel and back again without long set-up times.

We have seen excellent examples of using the machine to its maximum flexibility by gashing an integral bevel and spur gear in one program, one set up.

Sprockets

Almost everyone using gashers today produce sprockets. While not your ordinary bicycle type sprockets, the production of large sprockets, on older milling machines, because of the H.P. and tools available, has always been a slow process. In Fig. 7, we see the application of an inserted carbide sprocket milling cutter. The carbide inserts are placed and staggered to cut the full depth form as shown. During recent tests using a 75 HP gasher and a 1.5" circular pitch cutter with speeds of 375 surface feet per minute, full depth cuts were taken at 36 inches per minute. The finish was within acceptable limits of the end user. Thirty-six inches per minute is really fast, and the tests were performed without worry of excessive cutter wear. Interpolation of the results of the testing show that, dependent on the material and hardness, predicted rates of 22 to 24 inches per minute will give reasonable cutter economy.

The chips produced by this type of cutter differ greatly from the chips produced by the cutter in Fig. 4. These chips are short and almost flat. They have, however, substantial weight. Chips of this type are compact, easy to handle, and do not have the tendency to be carried over the top of the cutter and back into the cut.

```
SPROCKET EXAMPLE #1* (H.S.S. HOB vs. INSERTED CARBIDE CUTTER)

PART - SPROCKET
CIRCULAR PITCH - 1.75"
STRANDS        - 4
TEETH          - 25
MATERIAL       - AISI 240 BHN
PARTS/STACK    - 2
TOTAL TRAVEL   - 22.0"

Using the Formula:
```

$$\text{TOOL COST PER CUT} = \left[\frac{(\text{COST OF CUTTER}) + \text{REGRIND COST}}{\text{NO OF USEFUL REGRINDS}}\right] \div \text{PTS PER REGRIND}$$

```
H.S.S. HOB

PRICE          - $700.00
REGRIND LIFE   - 34
REGRIND COST   - $25.00
CUTS/GRIND     - 6
```

$$\left[\frac{(700)}{34} + 25.00\right] \div 6 = \$7.60$$

```
TOOL COST PER PART = $7.60

CARBIDE INSERTS

                  TYPE1*                                  TYPE 2**
PRICE          -  $440.00                                 $ 257.00
PRICE/INDEX -  $ 88.00 (5 edges average with               51.40
                         cross positioning)
CUTS/INDEX        18                                        18
TOOL COST/CUT     4.93 ($88.00 - 18)                       2.85 $51.40 - 18)

CUTTING TIME   - HOB/HOB                               GASH/GASH

ROUGHING  -       84 minutes @ .120 IPR                35.5 @ 18 IPM
FINISHING -      126 minutes @ .080 IPR                35.5 @ 18 IPM
                 210 minutes                           71.0 minutes

MACHINE TIME COST $45.00/hr
     HOB/HOB   = $157.00 + 7.60 + 7.60 = $172.20
     GASH/GASH = $ 53.00 + 4.93 + 4.93 = $ 62.86
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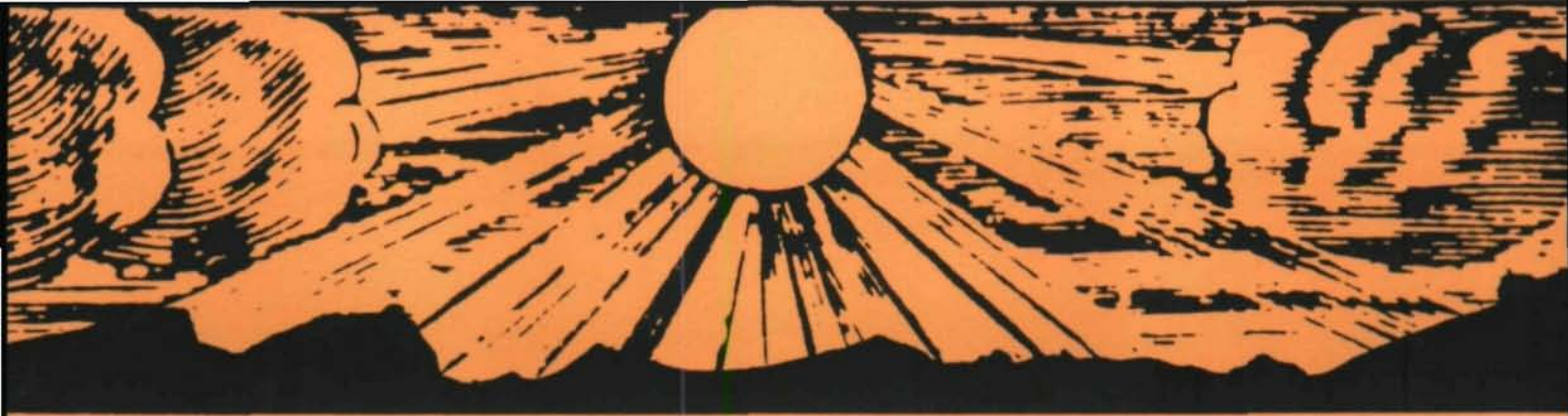

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B. Gear Manufacturer – Job Shop – Contract Work
C. Gear Machine Tool Manufacturer
D. Machine Tool Distributor/Agent
E. School/Research Organization
F. Government Agency
G. Industry Association
H. Advertising Agency
I. Other: Please specify ____________

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

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

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

How many employees at this location?

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

Your job function (circle all that apply)

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B. Manager ________ Dept. (specify)
C. Design Engineer
D. Process Engineer
E. Sales
F. Consultant
G. Educator
H. Other: Please specify ____________

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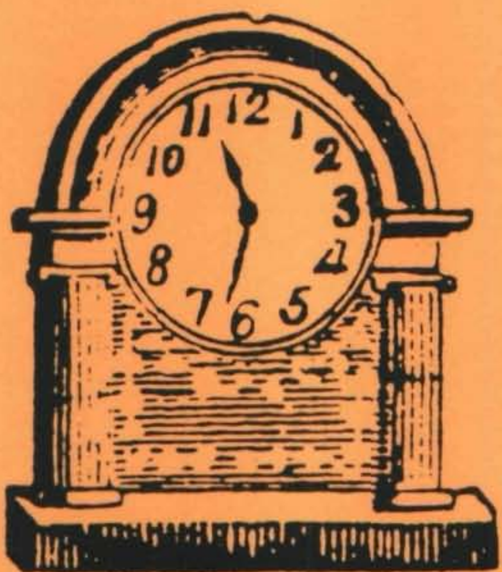

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In using the new inserted carbide sprocket cutters, we find that up to 2" C.P. can be cut easily in one pass. In the example above, the cost for one pass gashing would then be based on 35.5 minutes @ \$45.00/hour or \$26.62 + \$4.93 = \$31.55 or approximately 1/5 the cost of two-pass hobbing.

**Data by Ingersoll Cutting Tool Company, Rockford, IL.*
***Data by Greenleaf Corporation, Seagertown, PA.*

Fixturing

No machine today is worth its weight if the application of the machine is not 100%. The efficiency of any machining system will suffer if the cutting tools, tool arbors, fixtures and set-ups are not commensurate with the machine potential.

The fixtures used on the old millers are not adequate today. Manual clamping of fixtures is not adequate today. Today's fixtures must be capable of transmitting the cutting force into the work table, without introducing vibrations or harmonics. Full ring type fixtures, with solid support directly at the root diameter of the cut, are essential. Plate top clamps, held by the integral work holding cylinder, become the vibration dampener. The mass of the fixture, also, is a good vibration dampener.

For most bevel and pinion cutting, the work support arm adds the rigidity we lose by the smaller diameter support under the gear. Simple systems of a basic fixture, with substantial locating bushings, adapter rings, and top plates, have been designed and proven to be more than adequate and readily adaptable to many types of gear set-up.

Options

The usual options on gasher machines include: coolant systems for H.S.S. cutter applications, various sized cutter arbors to accommodate the old and new cutters, workholding cylinders and assistance in programming and fixturing.

A high volume chip conveyor is a necessity as is a cutter removal frame, which permits an entire cutter arbor assembly to be preset off the machine, and installed as a single unit for fast and easy cutter changeover.

The following charts are examples of inserted carbide costs, and production records of both carbide and high speed steel cutting of an assortment of gears and sprockets.

Synopsis

The last few years have become all important to the gear industry. The advances in tooling for milling gears have been forced by the manufacturers of large D.P. gears and the manufacturers of the gear milling machines. They responded with a well-managed research and development program. The people involved were interested enough not to stop at the first, second, or even third generation design. Each generation of cutter design brought us larger, stronger, smoother cutting tools. The book on feed rates was thrown out the window.

The application of microprocessor based numerical controls to gear milling has given us the ease and speed of set-up, the elimination of shafts and gears in drive lines. Axis resolution and table index accuracy are enabling better quality gears to be milled than ever before. All of these features make better gears faster at a lower cost. The microprocessor is taught how to make the *good* gears. The machine is strong and rigid to cut the *big* gears, and the tools match the machines and the technology. Gear Milling has caught up.

This article was reprinted from the American Gear Manufacturers Association Technical Paper 129.26.

CARBIDE SPROCKET MILLING
PERISHABLE COST ESTIMATE*

Test No.	201	202	203	204	205	206	207
Material	Cast Steel	Steel Plate	Fab	Cast Steel	Cast Steel	Cast Steel	Cast Steel
Circular Pitch	1.25	1.75	1.75	1.25	1.50	1.75	2.50
No. of Teeth	88	50	18	29	42	26	50
No. of Strands	3	1	1	3	6	2	3
Inserts/Set	\$325.00	\$384.00	\$384.00	\$324.00	\$384.00	\$384.00	\$480.00
Inserts/Edge	\$ 81.00	\$ 96.00	\$ 81.00	\$ 96.00	\$ 96.00	\$ 96.00	\$120.00
Parts/Edge	11	43	121	34	9	43	10
Insert Cost/Part	\$ 7.36	\$ 2.23	\$ .79	\$ 2.38	\$ 10.66	\$ 2.23	\$ 12.00
Insert Cost/Tooth	\$ .083	\$ .044	\$ .043	\$ .082	\$ .25	\$ .085	\$ .24
Insert Cost/Inch	\$ .041	\$ .047	\$ .046	\$ .04	\$ .049	\$ .047	\$ .059

**Data per Ingersol Cutting Tool Company*
Rockford, Illinois

(Continued on next page)

CARBIDE SPROCKET MILLING
PRODUCTION ESTIMATE*

Test No.	201	202	203	204	205	206	207
Material	Cast Steel	Steel Plate	Fab	Cast Steel "H"	Cast Steel "H"	Cast Steel "H"	Cast Steel "H"
Circular Pitch	1.25	1.75	1.75	1.25	1.50	1.75	2.50
No. of Teeth	88	50	18	29	42	26	50
No. of Strands	3	1	1	3	6	2	3
Part width O/A	3.5"	1.31"	4.87"	6.75"	12.0"	6.0"	7.5"
Parts/Stack	6	16	4	3	2	4	3
Total Travel	28.5"	28.5"	27.0"	26.75"	28.5"	28.5"	29.0"
Cutter Diameter	12"	12"	12"	12"	12"	12"	12"
Speed	350 SFM	350 SFM	350 SFM	300 SFM	300 SFM	300 SFM	300 SFM
Feed	18 IPM	18 IPM	18 IPM	15 IPM	15 IPM	15 IPM	15 IPM
Cut Direction	Climb	Climb	Climb	Climb	Climb	Climb	Climb
Total Linear	1061.28"	739.2"	66.52"				
Cut Time/Pass	1.58 min.	1.58 min.	1.5 min.	1.78 min.	1.9 min.	1.9 min.	1.93 min.
Idle/Pass	.12 min	.12 min.	.12 min.	.12 min.	.12 min.	.12 min.	.12 min.
Total Time/Pass	1.70 min.	1.70 min.	1.62 min.	1.90 min.	2.02 min.	2.02 min.	2.05 min.
Time/Stack	149.6 min.	85.0 min.	29.6 min.	55.1 min.	84.84 min.	52.52 min.	102.5 min.
Time/Piece	24.93 min.	5.31 min.	7.4 min.	18.4 min.	42.42 min.	13.13 min.	34.1 min.

**Data per Ingersol Cutting Tool Company
Rockford, Illinois*

TEST NO Unit Material	208 Gear SAE 1041	209 Sprocket CAST STEEL "H"	210 Sprocket Cast Steel "H"
Pitch	1 D.P.	2.0" C.P.	1.5" C.P.
No. of teeth	74	43	42
No. of strands	N/A	3	6
Part width O/A	5.0"	3.4"	3.7"
Parts/Stack	1	1	1
Total Travel	7.5"	7.7"	13.5"
Cutter Diameter	5.75"	9.0"	6.5"
Speed	72 SFM	80 SFM	68 SFM
Feed	10 IPM	5 IPM	8 IPM
Depth	.850	Full	Full
Cut Direction	Climb	Conv.	Conv.
Time/Stack	58 Min.	63 Min.	78 Min.
Time/Piece	58 Min.	63 Min.	78 Min.

Determination of Gear Ratios

by
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Introduction

Selection of the number of teeth for each gear in a gear train such that the output to input angular velocity ratio is a specified value is a problem considered by relatively few published works on gear design. McComb and Matson [1] have listed five methods, all of which involve cut-and-try procedures, Spotts [2] described a sixth cut-and-try technique, and Page [3] listed 14,000 gear ratios and the number of teeth on the gears involved.

The following sections show that several theorems from continued, or chain, fractions are applicable to the gear problem at hand, and suggest a direct means for determining the number of teeth on each gear. A numerical method is outlined and three examples of its use terminate the discussion.

Continued Fractions

Following Hardy and Wright [4], the continued fraction

$$a_0 + \cfrac{1}{a_1 + \cfrac{1}{a_2 + \cfrac{1}{\ddots + \cfrac{1}{a_N}}}} \tag{1}$$

will be denoted by $[a_0, a_1, \ldots, a_N]$ if the continued fraction has $N+1$ variables $a_0, a_1, \ldots, a_N$ and by $[a_0, a_1, \ldots]$ if it continues indefinitely. If $0 \leqq n \leqq N$ the partial fraction represented by $[a_0, a_1, \ldots, a_n]$ is termed the nth *convergent* to $[a_0, a_1, \ldots a_N]$. A continued fraction is said to be *simple* if a_1 through a_N inclusive are all positive and integral.

The fundemental theorem for gear applications is that *Theorem 1*:

If p_n and q_n are defined by

$$p_0 = a_0 q_0 = 1$$

$$p_1 = a_0 a_1 + 1 \quad q_1 = a_1 \tag{2}$$

$$p_n = a_n p_{n-1} + p_{n-2} \tag{3}$$

$$q_n' = a_n q_{n-1} + q_{n-2} \tag{4}$$

then

$$[a_0, a_1, \ldots, a_n] = \frac{p_n}{q_n}. \tag{5}$$

Substitution into (1) verifies that

$$[a_0] = a_0 \tag{6}$$

$$[a_0, a_1] = \frac{a_0 a_1 + 1}{a_1} \tag{7}$$

$$[a_0, a_1, a_2] = \frac{a_2 a_1 a_0 + a_2 + a_0}{a_2 a_1 + 1} \tag{8}$$

from which it follows, after algebraic manipulation, that

$$[a_0, a_1] = a_0 + \frac{1}{a_1} \tag{9}$$

and

$$[a_0, a_1, \ldots, a_{n-1}, a_n] = [a_0, a_1, \ldots, a_{n-2}, a_{n-1} + \frac{1}{a_n}] \tag{10}$$

Equations (6) and (7) verify the theorem for $n = 0$ and $n = 1$. Assuming it to be true for $n \leqq m$, for $m < N$, then

$$[a_0, a_1, \ldots, a_{m-1}, a_m] = \frac{p_m}{q_m} = \frac{a_m p_{m-1} + p_{m-2}}{a_m q_{m-1} + q_{m-2}} \tag{11}$$

where p_{m-1}, p_{m-2}, q_{m-1}, q_{m-2} all depend only upon a_0, $a_1, \ldots, a_{m-1}$. From (10) it follows that

$$[a_0, a_1, \ldots, a_{m-1}, a_m, a_{m+1}]$$

$$= \left[a_0, a_1, \ldots, a_{m-1}, a_m + \frac{1}{a_{m+1}}\right]$$

$$= \frac{\left(a_m + \frac{1}{a_{m+1}}\right) p_{m-1} + p_{m-2}}{\left(a_m + \frac{1}{a_{m+1}}\right) q_{m-1} + q_{m-2}}$$

$$= \frac{a_{m+1}(a_m p_{m-1} + p_{m-2}) + p_{m-1}}{a_{m+1}(a_m q_{m-1} + q_{m-2}) + q_{m-1}}$$

$$= \frac{a_{m+1} p_m + p_{m-1}}{a_{m+1} q_m + q_{m-1}} = \frac{p_{m+1}}{q_{m+1}}. \tag{12}$$

and the theorem is proved by induction. Replace m by $n-1$ in (12) to obtain relations (3) and (4).

Hardy and Wright [4] also proved the following theorems which are of importance to gear design.

Theorem 2:

The convergents to a simple continued fraction are in their lowest terms.

Theorem 3:

Any rational number can be represented by a finite simple continued fraction.

Theorem 4:

If $N > 1$, $n > 0$, then the difference

$$\frac{p_N}{q_N} - \frac{p_n}{q_n}$$

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decreases steadily in absolute value as n increases. Also

$$q_n \frac{p_N}{q_N} - p_n = \frac{(-1)^n \delta_n}{q_{n+1}}$$

where $0 < \delta_n < 1, \delta_{N-1} = 1$ for $1 \leqq n \leqq N-2$

and $$\left| \frac{p_N}{q_N} - \frac{p_n}{q_n} \right| \leqq \frac{1}{q_n q_{n-1}} \leqq \frac{1}{q_n^{\,2}}$$

for $n \leqq N-1$, with inequality in both places except when $n = N-1$.

Theorem 5:

If $a_0, a_1, a_2, \ldots$ is a sequence of integers for which $a_1, a_2, \ldots$ are all positive, then $[a_0, a_1, \ldots, a_n]$ tends to a limit as n goes to infinity.

Theorem 6:

Every irrational number can be expressed in just one way as an infinite simple continued fraction.

Theorem 7:

Theorem 4 holds (except for references to N) for infinite continued fractions, and

$$\left| \frac{p_N}{q_N} - \frac{p_n}{q_n} \right| < \frac{1}{q_n q_{n+1}} < \frac{1}{q_n^{\,2}}.$$

Together these mean that the numerator and denominator of the rational gear ratio have no factors in common and that if the gear ratio is irrational it is possible to approximate it to within any degree of accuracy by increasing the number of terms in the continued fraction. In particular, this enables a computer program to be written in which the user may specify, and realize, the accuracy of a gear train to approximate any desired speed ratio.

Numerical Method

Evaluation of the $a_0, a_1, \ldots$ terms is according to Euclid's algorithm, which yields

$$\begin{aligned} a_1 &= \mathrm{INT}(R) & y_1 &= \mathrm{FRC}(R) \\ p_1 &= a_1 & q_1 &= 1 \end{aligned} \tag{13}$$

$$\begin{aligned} a_2 &= \mathrm{INT}\left(\frac{1}{y_1}\right) & y_2 &= 1 - a_2 y_1 \\ p_2 &= a_1 a_2 + 1 & q_2 &= a_2 \end{aligned} \tag{14}$$

and for $n \geqq 3$,

$$\begin{aligned} a_n &= \mathrm{INT}\left(\frac{y_{n-2}}{y_{n-1}}\right) & y_n &= -a_n y_{n-1} + y_{n-2} \\ p_n &= a_n p_{n-1} + p_{n-2} & q_n &= a_n q_{n-1} + q_{n-2} \end{aligned} \tag{15}$$

where R is either the desired ratio of the angular velocity of the output shaft to that of the input shaft, or the inverse, INT (R) is the largest integer equal to or less than R, and FRC (R) $= R -$ INT (R).

Because of the variety of programming codes, the program will be presented as a flow chart. In it E is the maximum acceptable error, defined as

$$E = |R - \frac{p_N}{q_N}|$$

Input data to program GEAR RATIO is the desired speed ratio, R, and the magnitude of the acceptable error, E. Error E may be decreased by REVISE if the user wished to examine the effect of considering rational approximations for a sequence of decreasing E values.

Examples

Three examples will be considered: one where R is rational and p_N and q_N are factorable, one where both p_N and q_N are not factorable, and one where R is irrational.

Example 1. Find a gear train providing an angular velocity

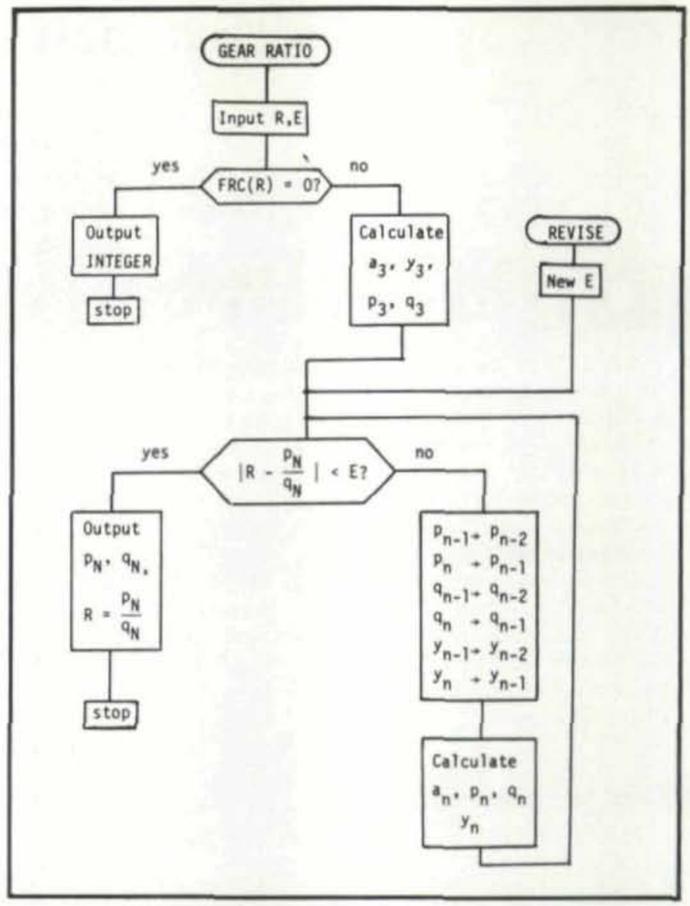

FIG. 1—Flowchart for GEAR RATIO program

ratio of 2.871 with an error not greater than 0.000001.

Computation time for a program implemented on a Hewlett-Packard HP-41CV is approximately 9.9 s to find that

$$\frac{p_N}{q_N} = \frac{2871}{1000}$$

is the only rational that will satisfy the accuracy requirement. Use of a factoring routine leads to the possible gear train

$$R = \frac{\omega_1}{\omega_2} = \frac{33}{25} \times \frac{87}{40} = 2.871000.$$

Thus, $N_1 = 25$, $N_2 = 33$, $N_3 = 40$, and $N_4 = 87$, where gears 2 and 3 are on the same shaft, gear 1 on the input shaft meshes with gear 2, and gear 4 on the output shaft meshes with gear 3.

Example 2. Find a gear train providing an angular velocity ratio of 2.68 with an error no more than 0.0001.

According to formulas (13), (14), and (15)

$$\begin{aligned} a_1 &= \mathrm{INT}(2.68) = 2 & a_2 &= \mathrm{INT}\left(\frac{1}{y_1}\right) = 1 \\ p_1 &= 2 & p_2 &= 3 \\ q_1 &= 1 & q_2 &= 1 \\ y_1 &= 0.68 & y_2 &= 1 - 0.68 = 0.32 \end{aligned}$$

$$\begin{aligned} a_3 &= \mathrm{INT}\left(\frac{0.68}{0.32}\right) = 2 & a_4 &= 8 \\ p_3 &= 8 & p_4 &= 67 \\ q_3 &= 3 & q_4 &= 25 \\ y_3 &= 0.04 & y_4 &= 0 \end{aligned}$$

so that $N_2 = p_4 = 67$, $N_1 = q_4 = 25$, and $R = 67/25 = 2.680000$.

(Continued to page 44)

Functions of Gearing and Application of the Involute to Gear Teeth

by
Fellows Corporation

When it is required to transmit rotary motion from one shaft to another at a definite ratio, with the shafts rotating in opposite directions, several means can be employed. One method is to use friction disks, as shown at *A*, Fig. 1.

Pulleys and a crossed belt can also be used, as shown at *B*, Fig. 1. If, however, uniform angular motion must be transmitted, neither the friction disks, nor the pulleys and crossed belt, will prove satisfactory.

Friction disks cannot be relied upon to transmit uniform angular motion, because they lack positive driving contact. As soon as sufficient load is imposed on the driven member, slippage between the disks occurs.

Pulleys and a crossed belt are more dependable than the friction disks due to the greater area of frictional contact, but here again, because of the lack of positive driving contact between the crossed belt and the pulleys, belt slippage occurs when sufficient load is applied to the driven pulley. Therefore, uniform angular motion cannot be maintained by pulleys and a crossed belt.

Gears, as shown at *C*, Fig. 1, offer the most practical and dependable means for transmitting uniform angular motion, but, the shape of the teeth has an important bearing on the smoothness of the motion transmitted.

Experience has proven that the involute provides the most satisfactory profile for spur and helical gear teeth, and fulfills the requirements for transmitting smooth uniform angular motion.

The Involute

The involute can be described as that curve traced by a point on a crossed belt as it moves from one pulley to the other without slippage. This can be demonstrated by attaching a template to the driven pulley, as shown at *A*, Fig. 2. Rotation of the driver pulley will cause the tracing point on the crossed belt to trace an involute on the template attached to the driven pulley.

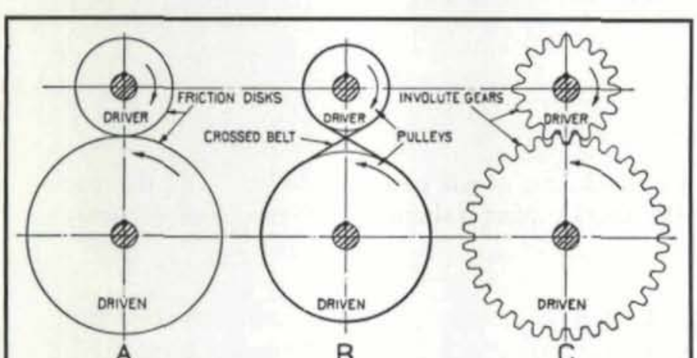

Fig. 1 — Diagram Illustrating Three Means for Transmitting Rotary Motion from One Shaft to Another.

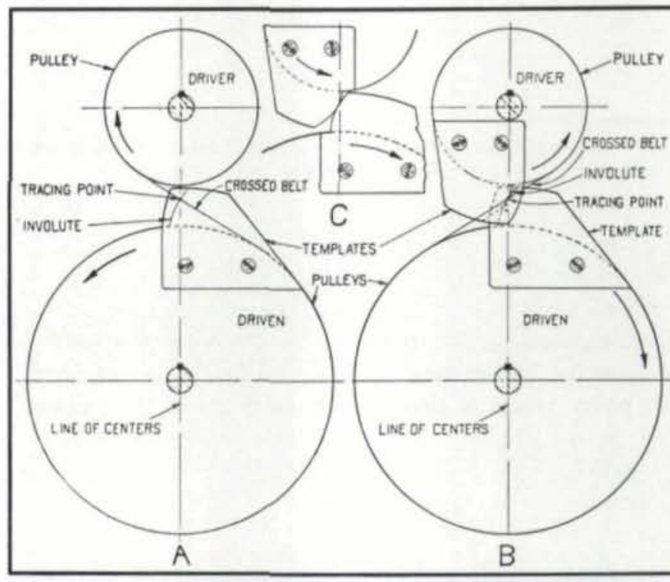

Fig. 2 — Diagram Illustrating That the Involute Is That Curve Traced by a Point on a Crossed Belt as It Moves from One Pulley to Another.

Similarly, a mating involute can be developed by attaching a template to the driver pulley, as shown at *B*, Fig. 2. A tracing point on the other side of the crossed belt will trace an involute on this template when the driver pulley is rotated in the opposite direction.

If these traced templates are cut to the traced curves, as shown at *C*, in Fig. 2, they produce mating involute profiles, which will transmit uniform angular motion, when the connecting belt is removed.

Since each of these profiles is only one side of a single involute gear tooth, they will only transmit motion through a slight arc of rotation. To transmit continuous uniform angular motion, succeeding equally-spaced and parallel profiles must be provided. This has been done, as shown at *A* in Fig. 3, where equally-spaced tracing points *a* have been located on the crossed belt.

If motion is to be transmitted in both directions, then similar profiles must be provided, as shown at *B* in Fig. 3, where tracing points *b* on the other side of the crossed belt have been added. We now have developed involute gear teeth to transmit uniform angular motion which replace the friction disks, pulleys and crossed belt.

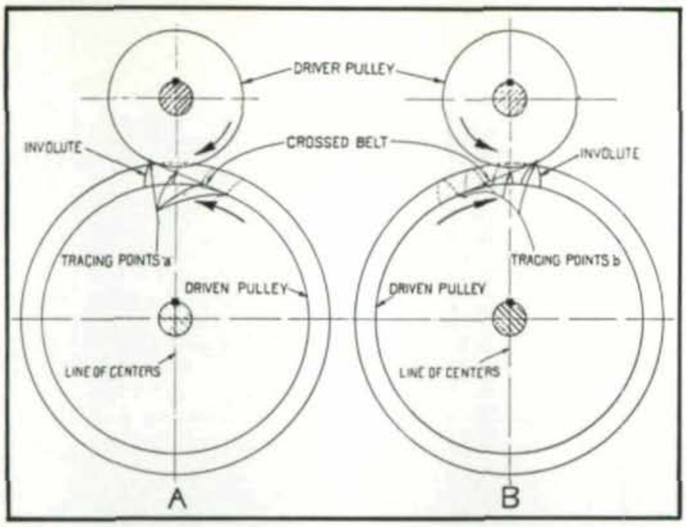

Fig. 3 — Diagram Showing How Equally-spaced Points on Crossed Belt Trace Successive Equally-spaced and Parallel Involute Profiles.

In Fig. 4, the gears, pulleys and crossed belt have been super-imposed on the friction disks. Here, it will be noticed that the friction disks are the pitch circles of the gears. The pulleys are the base circles, and the crossed belt represents the line of action. Where the belt crosses the line of centers is the pitch point, which is also the tangent point of the friction disks, or the common tangent. The angle between the crossed belt, or line of action, and the common tangent is called the pressure angle.

Changes in Center Distance Do Not Affect Velocity Ratio of Conjugate Tooth Action

A cardinal virtue of the involute is that it permits a change in center distance without affecting the velocity ratio, or conjugate tooth action. When the center distance is increased, as

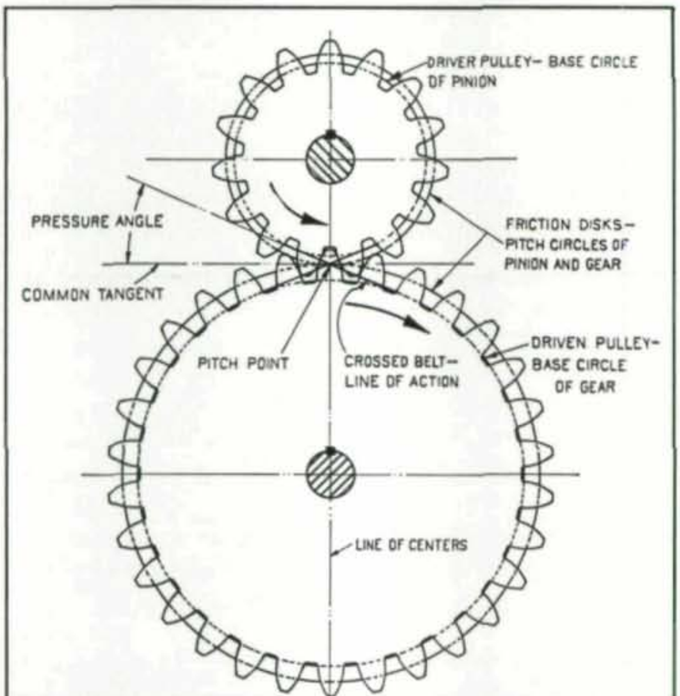

Fig. 4 — Diagram Showing Gears, Pulleys and Crossed Belt, Superimposed on Friction Disks, Illustrating Basic Gear Elements.

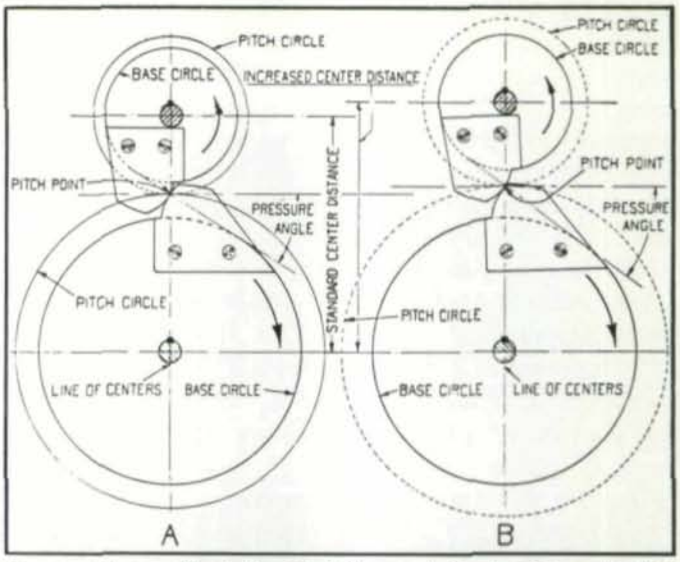

Fig. 5 — Diagram Illustrating That Changes in Center Distance Do Not Affect Velocity Ratio, or Conjugate Tooth Action.

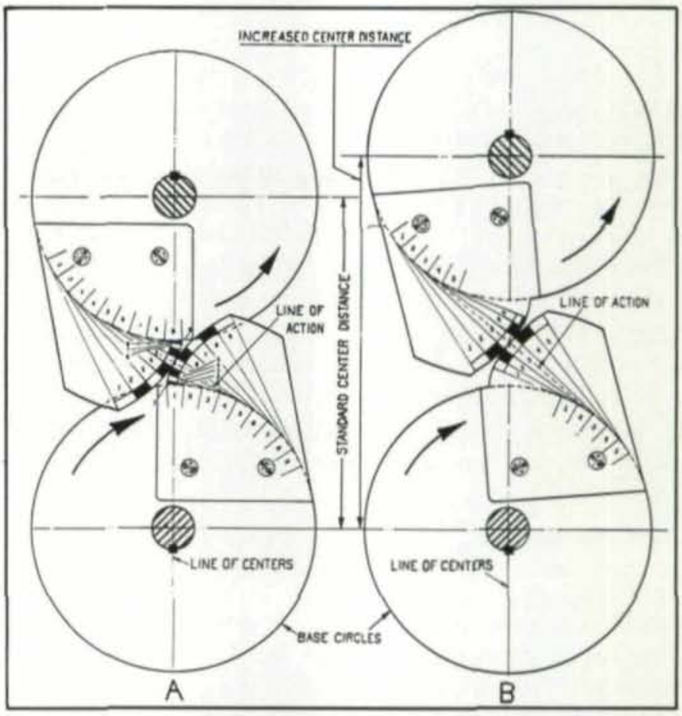

Fig. 6 — Diagram Illustrating Relative Slippage of Involute Gear Teeth.

shown at *B* in Fig. 5, the base circles remain the same but a new line of action, new pitch circles and a new pitch point are created. All of these elements, when located at the standard center distant at *A*, are shown in full lines; and the same elements in dotted lines, when the center distance is increased, as shown at *B*. It will also be noted that the operating pressure angle is increased when the center distance is increased; and, as will be subsequently shown, the length of the line of contact is shortened if the tooth lengths remain the same.

Gear Tooth Slippage

A study of the action of mating involute profiles shows that they slide upon each other for their entire contact. At *A* in Fig. 6, both base circles are divided into an equal number of spaces, and tangent lines are drawn from the division points on the base circles to the involute profiles.

It will be noted that while the divisions of the base circles are equal, the corresponding arcs on the involute profiles are unequal, increasing in length as they depart from the base circles. Corresponding divisions on the base circles and involute profiles are numbered for comparison and identification, and indicate those portions of the profiles that are in contact with each other during equal pitch line movements of the gears.

The relative lengths of the mating arcs on the profiles is a measure of the amount of slippage between the profiles. The greatest amount of slippage takes place when the end of one profile is in contact with the mating profile in the vicinity of the base circle. It will also be noted, that involute contact takes place on a line tangent to both base circles—*this is the line of action.*

When the center distance is increased, the relative rate of slippage is not so great as before, as shown at *B* in Fig. 6. The reason for this is that the contact between the involute profiles takes place farther away from the base circles.

The Involute Rack

The parent rack of the involute system of gearing has straight sides. In Fig. 7, the involute template representing one side of a gear tooth is shown contacting one side of a rack tooth and driving the latter with a uniform linear motion. Note that when the involute template has moved distance *Y* on the line of action, the rack tooth has moved distance *X* in a linear direction. If distance *Y* is the base pitch of the gear—the circular pitch transferred to the base circle—and distance *X* is the linear pitch of the rack, then equal angular rotations of the gear must transmit equal linear movements to the rack, because the circular pitch of the gear must equal the linear pitch of the mating rack. It should be noted that the face of the rack tooth must be at right angles to the line of action.

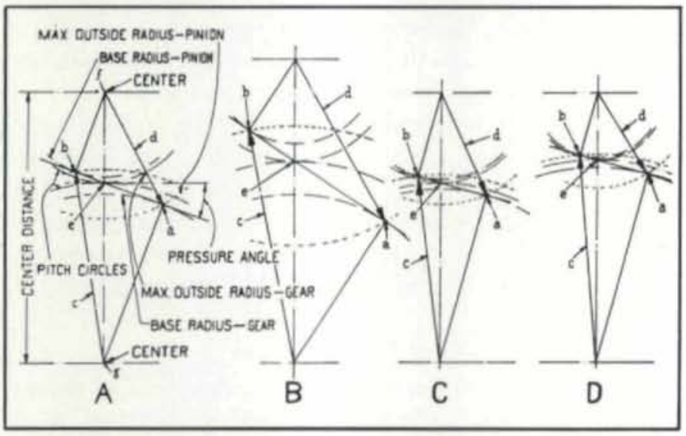

Fig. 7 — Diagram Illustrating That a Straight-sided Rack Tooth is the Proper Shape to Engage with an Involute Gear Tooth—Also That an Involute Contacting a Rack Transmits Uniform Linear Motion to the Rack.

Basic Principles of Involute Gearing

With properly designed and generated involute gears, smooth continuous action results, but improper design can defeat the obtainment of this objective. The problem of designing gears can be greatly simplified if we thoroughly understand the basic principles involved. There are three basic elements in the design of involute gearing, *viz.*: center distance, base circle diameter and tooth ratio.

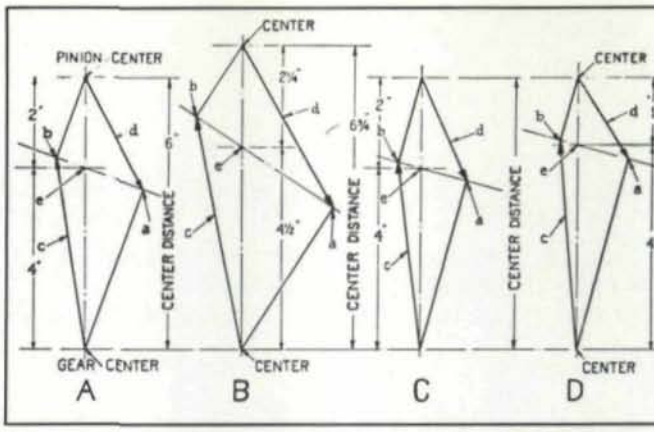

Fig. 8 — Diagram Illustrating Basic Elements of Involute Gearing.

These fundamental conditions or requirements can be clearly illustrated and explained by constructing simple diagrams, as shown at *A, B, C* and *D*, Fig. 8. To proceed: draw a center line, as shown at *A*. Space off on this line the required center distance. The ratio of the two gears is then spaced off, as represented by point *e*, which is the pitch point of the pair of gears. An angular line drawn through pitch point *e* represents the line of action.

Lines drawn from the centers of gear and pinion, and at right angles to the line of action, locate points *a* and *b* which represent the origins of the involutes, or the base radii of gear and pinion, respectively. Now draw lines *c* and *d* from both centers to points *a* and *b* on the line of action. Points *a* and *b*, in addition to representing the points of origin of the involutes, are also the interference points, as will be explained later. Radius *c* represents the maximum permissible outside radius of the gear, and radius *d*, the maximum permissible outside radius of the pinion.

Now notice what happens to the diagram at *A*, when the center distance is increased as shown at *B*, the base radii and ratio remaining the same as at *A*. First, the pitch point *e* is located at a different position. Locations of the interference points *a* and *b* on the line of action have changed, affecting the maximum permissible outside radii of gear and pinion, respectively. The pressure angle has also increased. This increase in the center distance, as has been previously ex-

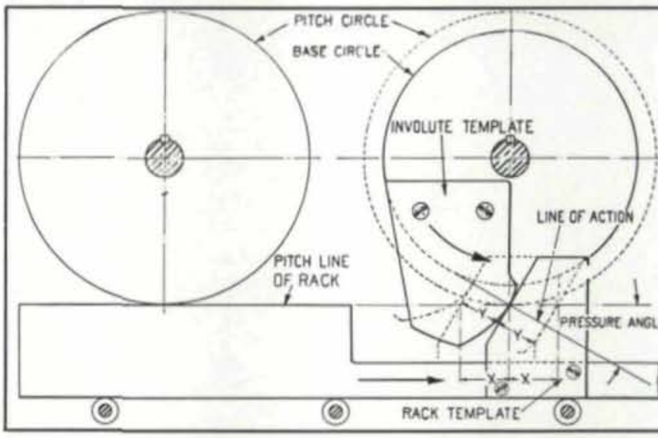

Fig. 9 — Similar Diagrams to Fig. 8, Except That Arcs Have Been Added to Indicate Pitch Circles, Base Circles, and Maximum Permissible Outside Radii of Gear and Pinion, Respectively.

plained, will not necessarily affect conjugate tooth action, but it does change the relative contact positions of the tooth profiles, as indicated by the tooth slippage diagrams in Fig. 6.

At *C*, in Fig. 8, the center distance and tooth ratio remain the same as at *A*, but the angle of inclination of the line of action has been reduced. Note that this affects the base radii of gear and pinion, and also the maximum permissible outside radii of gear and pinion.

At *D* in Fig. 8, the inclination of the line of action is the same as at *C*, but the ratio has been changed. This change affects the outside and base radii of gear and pinion, respectively.

At *A*, *B*, *C* and *D*, Fig. 9, diagrams similar to those in Fig. 8 have been constructed, with the addition of arcs passing through points, *a*, *b* and *e*. It will be noted that these arcs establish, in each case, the base circles, pitch circles and maximum outside circles of gear and pinion, respectively. A comparison of these diagrams clearly demonstrates the interrelationship of the three fundamental elements of gear design, *viz.*: center distance, base circles and tooth ratio.

It has been demonstrated that a change in the base circle center distance affects the operating pressure angle of two involute curves developed from these base circles. This is not the case when an involute gear tooth and rack are brought into contact. It has been demonstrated in connection with Fig. 7 that the line of action must lie at right angles to the rack tooth face. At *A* in Fig. 10 a gear and rack tooth are shown in mesh at distance *a*. At *B*, Fig. 10, this distance *a*, as indicated by distance *b*, is shown appreciably increased.

In order to keep the line of action at right angles to the rack face, the pitch point has moved a distance equal to *c*, and thus established a new position for the rack pitch line. It will also be noted that increasing the distance of the base circle center has not affected the base circle, or the pitch circle of the involute gear tooth, nor has it changed the operating pressure angle.

From the foregoing, we may make the following conclusions concerning the involute curve and its application to gears operating on parallel axes.

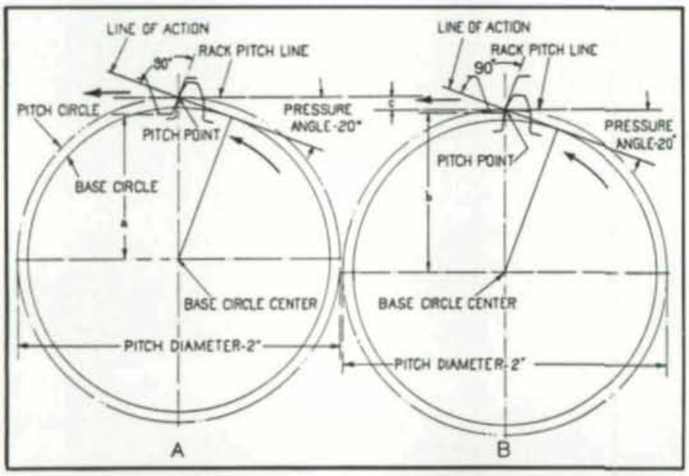

Fig. 10 — Diagram Illustrating That the Pressure Angle and Pitch Circle Diameter Do Not Change When a Gear Tooth Is Moved Closer to or Farther Away from a Rack.

Fundamental Laws of the Involute Curve

1. The involute is wholly determined by the diameter of its base circle.

2. An involute moving about its base circle center imparts rotative motion to a contacting involute in the exact ratio of the diameters of their respective base circles.

3. An involute has no pressure angle until brought into intimate contact with another involute or a rack.

4. The pressure angle is determined by the center distance and the base circle diameters.

5. The pressure angle once established is constant for a fixed center distance.

6. An involute has no pitch diameter until brought into intimate contact with another involute or a rack.

7. The pitch diameter of an involute contacting another involute is determined by the center distance and the ratio.

8. The pressure angle of an involute contacting a rack is unchanged when the base circle center is moved toward or away from the rack.

9. The pitch diameter of an involute contacting a rack is unchanged when the base circle center is moved toward or away from the rack.

10. The pitch line position of an engaging involute and rack is determined by the intersection of the line of action and a line passing through the base circle center and perpendicular to the direction of rack travel.

Gear Tooth Interference

The term gear tooth interference is defined as contact between mating teeth at some other point than along the line of action. Two types of interference are sometimes encountered. One is known as involute interference, and the other as fillet interference. Involute interference is the term used to indicate that contact between mating teeth takes palce at some other point than on the line of action *outside* the zone of contact. Fillet interference refers to contact at some other point than on the line of action *inside* the zone of contact. The zone of contact is that portion of the line of action bounded by the "natural" interference points, see Fig. 11.

Involute Interference

The condition known as involute interference is sometimes encountered when the mating gear teeth lock, or will not pass freely through mesh. Fig. 11 illustrates diagrammatically a pair of gears the outside circles of which are enlarged to the interference points. It will be noted that the tooth length on the smaller gear can be increased appreciably more than that on the larger gear without interference. Hence, it is the teeth on the larger gear which cause the trouble. If the ends of the teeth on the larger gear extend beyond the "natural" interference point, non-involute contact results, because the teeth on the gear contact the non-involute flank of the pinion teeth and thus prevent free rotation. At *A* in Fig. 12, it will be noted that the outside circle of the teeth on the larger gear extends considerably beyond the interference point, and hence interferes with the flanks of the teeth on the smaller gear. This condition has been exaggerated to illustrate the term known as "involute interference." If the flanks on the pinion teeth are radial, or extend inside of a radial line tan-

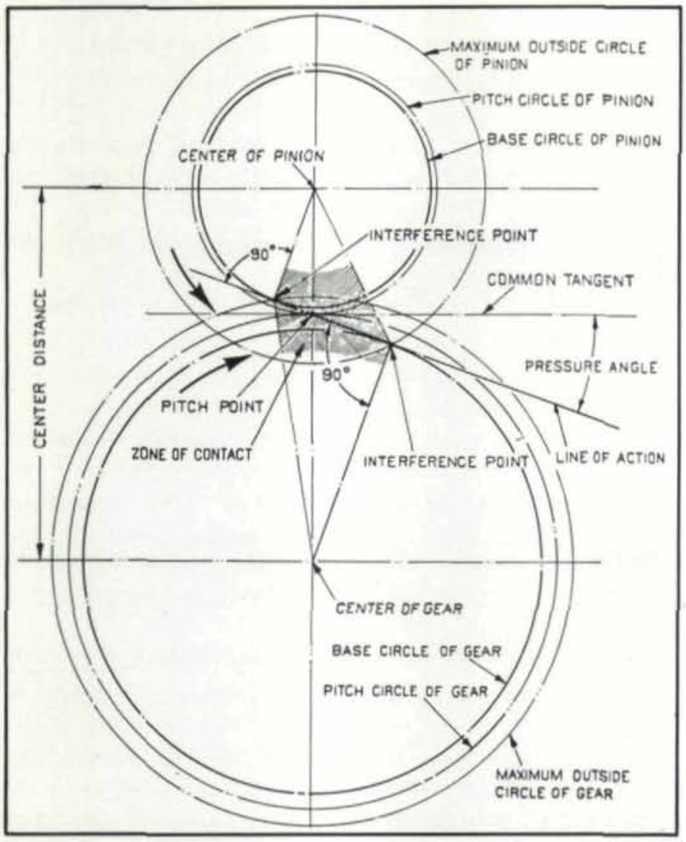

Fig. 11 — Diagram Presenting Principal Boundary Surfaces of Gear and Pinion, Showing Maximum Permissible Outside Diameters to Avoid Involute Interference.

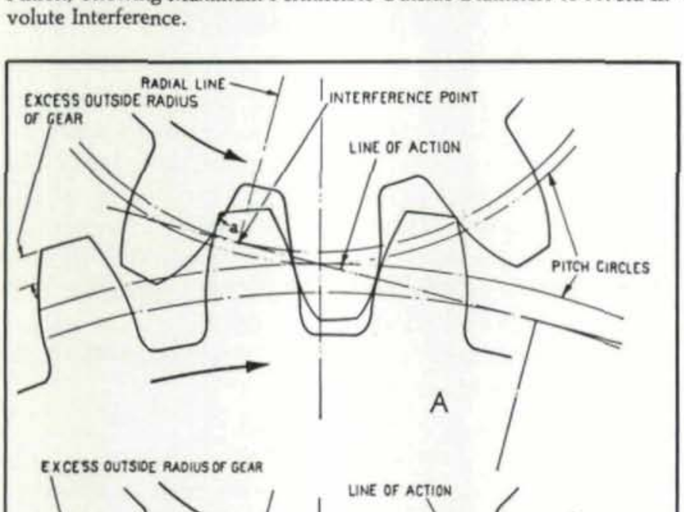

Fig. 12 — Diagram Illustrating Term Known as Involute Interference. Gear Tooth Contacts Non-Involute Portion of Pinion Tooth Outside the Zone of Contact.

gent to the involute at the base circle, the amount of interference would be greater than if the flanks were undercut. The amount of undercut is governed by the type of

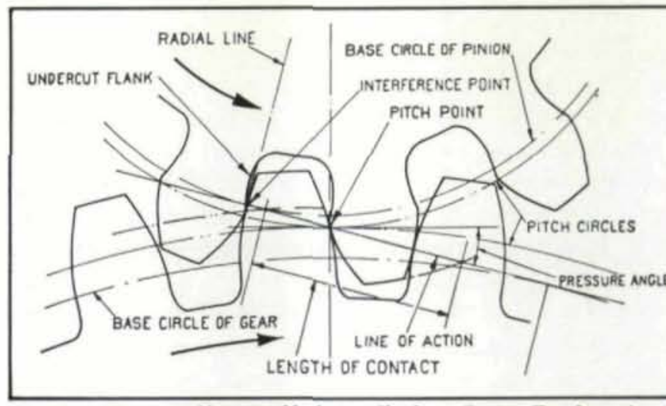

Fig. 13 — Diagram Showing Undercut Flank on Pinion Tooth to Avoid Involute Interference.

cutter used, the ratio between the number of teeth on the pinion and cutter, and the pressure angle. The cross-sectioned portion *a* of the tooth at *A*, Fig. 12, shows that the gear tooth overlaps the pinion tooth, and thus prevents free rotation.

At *B*, Fig. 12, the gears are rotated into a different position to illustrate two points of interference at *b* and *c*. It is evident that these gears cannot be rotated in either direction, or in other words they "lock," and free rotation is impossible.

Involute interference can be avoided in several different ways. One method is to design the cutter so that it removes the interfering portions of the pinion teeth, as shown in Fig. 13. This method is objectionable, because, as will be noted, it undercuts the flanks of the pinion teeth, reducing their strength. Furthermore, if the undercutting is excessive and extends above the base circle, it shortens the active involute portion of the teeth, reducing the length of the line of contact, and may result in lack of continuous action.

Another method is to use enlarged and reduced diameters on pinion and gear, respectively, or what is known as long- and short-addendum teeth. This particular method is recommended when the pinion has a comparatively small number of teeth and meshes with a gear having three or more times as many teeth; also when a small pressure angle is used, and standard center distance must be maintained. This method is illustrated diagrammatically in Fig. 14, where the outside diameter of the pinion is enlarged, and the outside diameter of the gear reduced the same amount. In Fig. 14, it will be noted that the original outside circle of the gear, indicated by

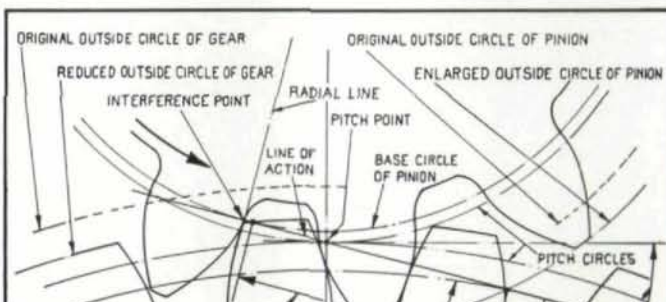

Fig. 14 — Diagram Illustrating Application of Long- and Short-Addendum Teeth to Avoid Involute Interference.

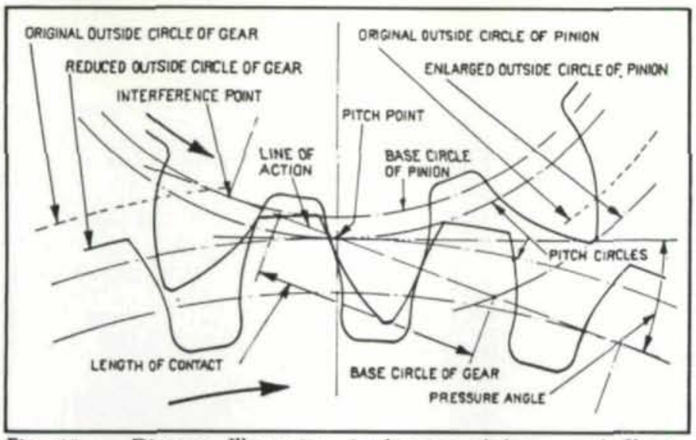

Fig. 15 — Diagram Illustrating Application of Long- and Short-Addendum, Coupled with Increased Pressure Angle to Avoid Involute Interference.

the dotted lines, extends beyond the interference point on the pinion teeth. By shortening the addendum on the gear tooth interference is avoided, and by lengthening the pinion addendum, length of contact is increased.

A comparison of the diagrams in Figs. 13 and 14 representing a pair of gears of the same tooth ratio, and designed to operate at standard center distance, shows that with the undercut condition in the flank of the pinion tooth, the length of contact is shorter than in Fig. 14, where long- and short-addendum teeth are used. In fact, the length of the line of contact is increased 30% in the case of the long-and short-addendum teeth.

Still another suggestion is to maintain the same reduction and enlargement of gear and pinion diameters, and increase the pressure angle. This condition is presented diagrammatically in Fig. 15. Here it will be noticed that the start and end of contact lie well inside the interference points, and that the length of contact is about 15% less than in Fig. 14. Therefore, an increase in the pressure angle coupled with long- and short-addendum teeth keeps contact on both gear and pinion teeth farther away from the interference points. When severe cases of interference are encountered, this method is recommended.

Fillet Interference

The condition in gear operation known as fillet interference is illustrated diagrammatically in Fig. 16. This occurs when

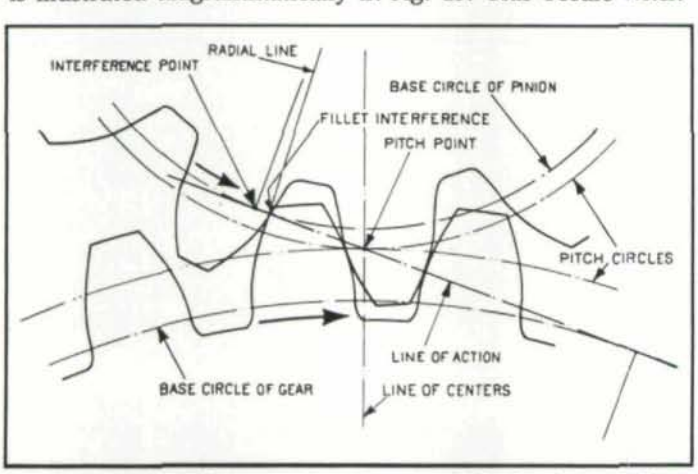

Fig. 16 — Diagram Illustrating Term Known as Fillet Interference. Gear Tooth Contacts Non-Involute Portion of Pinion Tooth Inside the Zone of Contact.

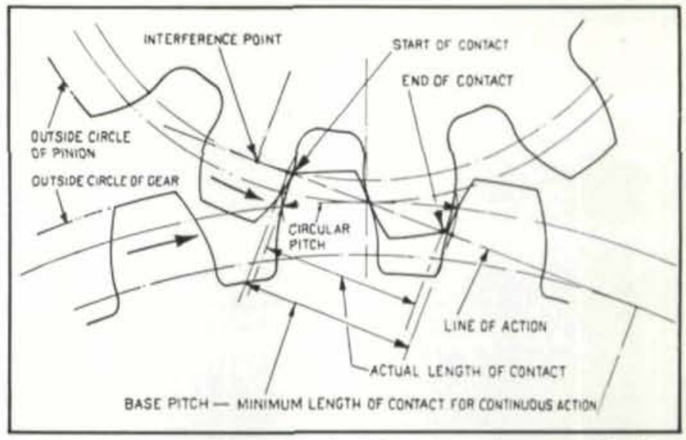

Fig. 17 — Diagram Illustrating Lack of Continuous Action as a Result of Shortened Teeth.

the flank of the pinion tooth below the base circle lies inside the radial line tangent to the involute at the base circle. This condition prevents free rotation of the teeth and the transmission of uniform angular velocity. There are several solutions that can be adopted to avoid fillet interference. One is to enlarge the pinion diameter and reduce the mating gear diameter a similar amount, or in other words use long and short addendum teeth as explained more fully in Chapter II. Another method is to design the cutter so that it will remove the interfering portion in the flank of the tooth. This method might result in shortening the length of the line of contact, especially if the undercut extends beyond the base circle towards the pitch circle.

Lack of Continuous Action Due to Shortened Teeth

This term refers to a condition in gear design, where the actual length of tooth contact is less than the base pitch. In other words, one pair of teeth go out of contact before a succeeding pair are in position to make contact, resulting in

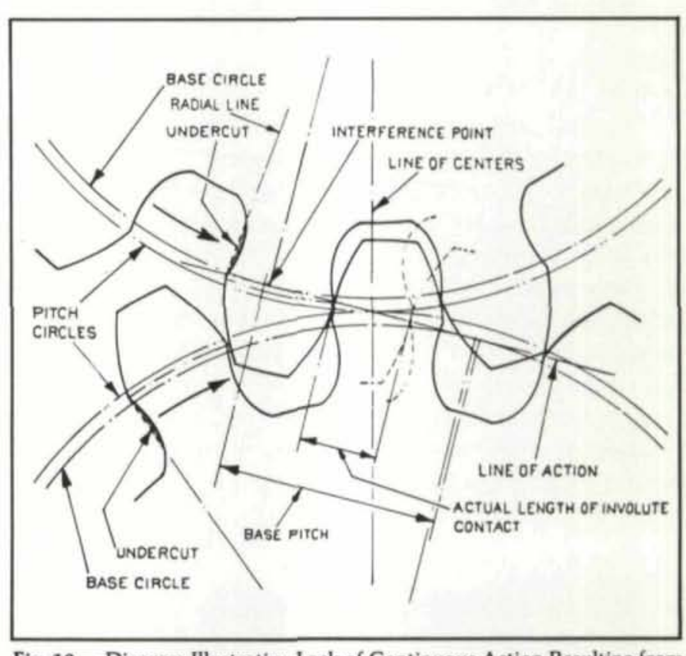

Fig. 18 — Diagram Illustrating Lack of Continuous Action Resulting from Undercutting of Flanks of Teeth. Also Due to Small Pressure Angle and Small Number of Teeth in Both Members.

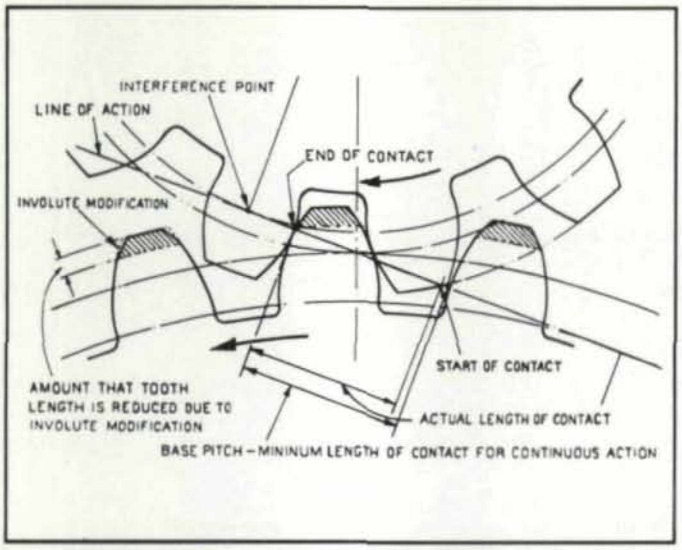

Fig. 19 — Diagram Illustrating Lack of Continuous Action Resulting from Involute Modification.

interrupted action. Fig. 17 shows diagrammatically a pair of gears in which the teeth are shortened to such an extent that they lack continuous action, and hence fail to transmit uniform angular motion. It will be noticed that the base pitch, which is the circular pitch transferred to the base circle is greater in length than the actual length of contact.

Lack of Continuous Action Due to Undercut

Fig. 18 shows another condition that can result in lack of continuous action. Here two 12-tooth pinions are shown in mesh, and it will be seen that the actual length of involute contact falls far short of meeting the requirements for continuous action. There are two reasons for this. One is the small number of teeth in the mating gears, and the other is the small pressure angle. The small pressure angle results in a severe undercutting of the flanks of the teeth, and the small number of teeth, of course, reduces the number of teeth in contact.

Lack of Continuous Action Due to Tooth Modification

Another condition that can cause lack of continuous action is involute modification. Fig. 19 illustrates a pair of gears, the tooth shape on one of these gears being modified near the tip. In this case, true involute contact does not start at the tip of the tooth on one gear, and hence, falls short of complete profile contact. Under light loading conditions, these gears might fail for continuous action, because of the shortened tooth profile above the pitch circle; but under heavy loading, the contact could extend over a greater length of the tooth profile. Gears are sometimes cut "off" pressure angle to compensate for tooth deflection under heavily loaded conditions; also to take care of tooth distortion due to heat treatment.

E-6 ON READER REPLY CARD

DETERMINATION OF GEAR RATIOS . . .

(Continued from page 36)

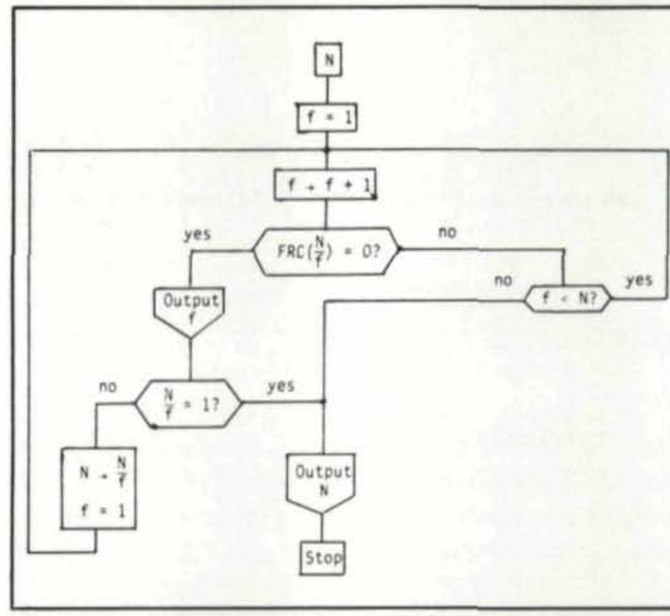

FIG. 2—Flowchart for FACTOR program

Example 3. Design a gear train to provide an angular velocity ratio of $\sqrt{5}$. Computer output is displayed in Table 1 along with the computation times using program GEAR RATIO followed by successive uses of REVISE.

Use of a factoring routine once again for $N_1 = 2889$ and $N_2 = 1292$ yields a four gear set with

$$R = \frac{27}{19} \times \frac{107}{68} = 2.2360681.$$

Thus, $N_1 = 19$, $N_2 = 27$, $N_3 = 68$, and $N_4 = 107$ teeth. Obtaining still greater accuracy is more difficult from a practical viewpoint because 13 and 421 are the only factors of 5473 while 47 and 1103 are the only factors of 51,841.

The flowchart for a typical factoring routine is shown in Fig. 2.

Table 1 Rational numbers approximating $\sqrt{5}$

E	p_N	q_N	Approx. computation* time (s)
1×10^{-2}	38	17	4
1×10^{-3}	38	17	2
1×10^{-4}	161	72	3
1×10^{-5}	682	305	4
1×10^{-6}	2889	1292	3
1×10^{-7}	12,238	5473	4
1×10^{-8}	12,238	5473	5
1×10^{-9}	115,920	51,841	5

* The first calculation used the GEAR RATIO program and the remaining calculations were performed by entering the program at REVISE. Direct calculation of the last ratio (115 920/51841) using GEAR RATIO required approximately 18 s, rather than the 30 required for the step-by-step calculations.

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E-5 ON READER REPLY CARD

MAXIMUM SURFACE TEMPERATURE ...

(Continued from page 27)

TABLE 1

GEAR GEOMETRY

	PINION (DRIVER) PLASTIC[(1)]	DRIVEN GEAR STEEL
module, m, mm (Pitch, P, diametral)	5, 8, 10, 12 (5.08, 3.175, 2.54, 2.116)	
Pitch diameter, d_p mm (inch)	3.0 (76.2)	3, 4.5[(2)] (76.2, 114.3)
Number of teeth $Z_{1,2}$	15, 24, 30, 36	15, 24, 30, 36, 45[(2)]
Pressure angle α, deg	20	
Face width, b mm (inch)	0.5 (12.7)	1.0 (25.4)
AGMA quality number	5 to 8	11

TABLE 2

VALUE OF COEFFICIENTS USED IN EQ (2)

MATERIAL	b_0	b_1	b_2	b_3
nylon 6-6	.2354	.755	.420	.502
acetal	5.556×10^{-2}	1.08	.354	.225
U.H.M.W.P.E.	1.985×10^{-4}	1.76	.831	.687

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CIRCLE E-3 ON READER REPLY CARD

This article was reprinted from the American Gear Manufacturers Association.

CORRECTION:

ROBERT E. SMITH's first name was listed incorrectly on the content page of the last issue. Mr. Smith was the author of "Single Flank Testing of Gears."

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Spiral Bevel Gears
(Continued from page 13)

Figure 7

Fig. 7 shows one of the forged gears being measured on the Zeiss machine. The plots show the relative deviation of the forged tooth profile as compared to a "master gear" tooth produced by conventional cutting (using a Gleason generator). Note that the relative error at the center of the profile is zero; i.e., the variations were measured relative to the center of the coast and drive surface of the master gear. The maximum variation was 0.003 inch (0.0762 mm). This difference can be compensated for easily in the cutting of the matching pinion.

Economics Attractive

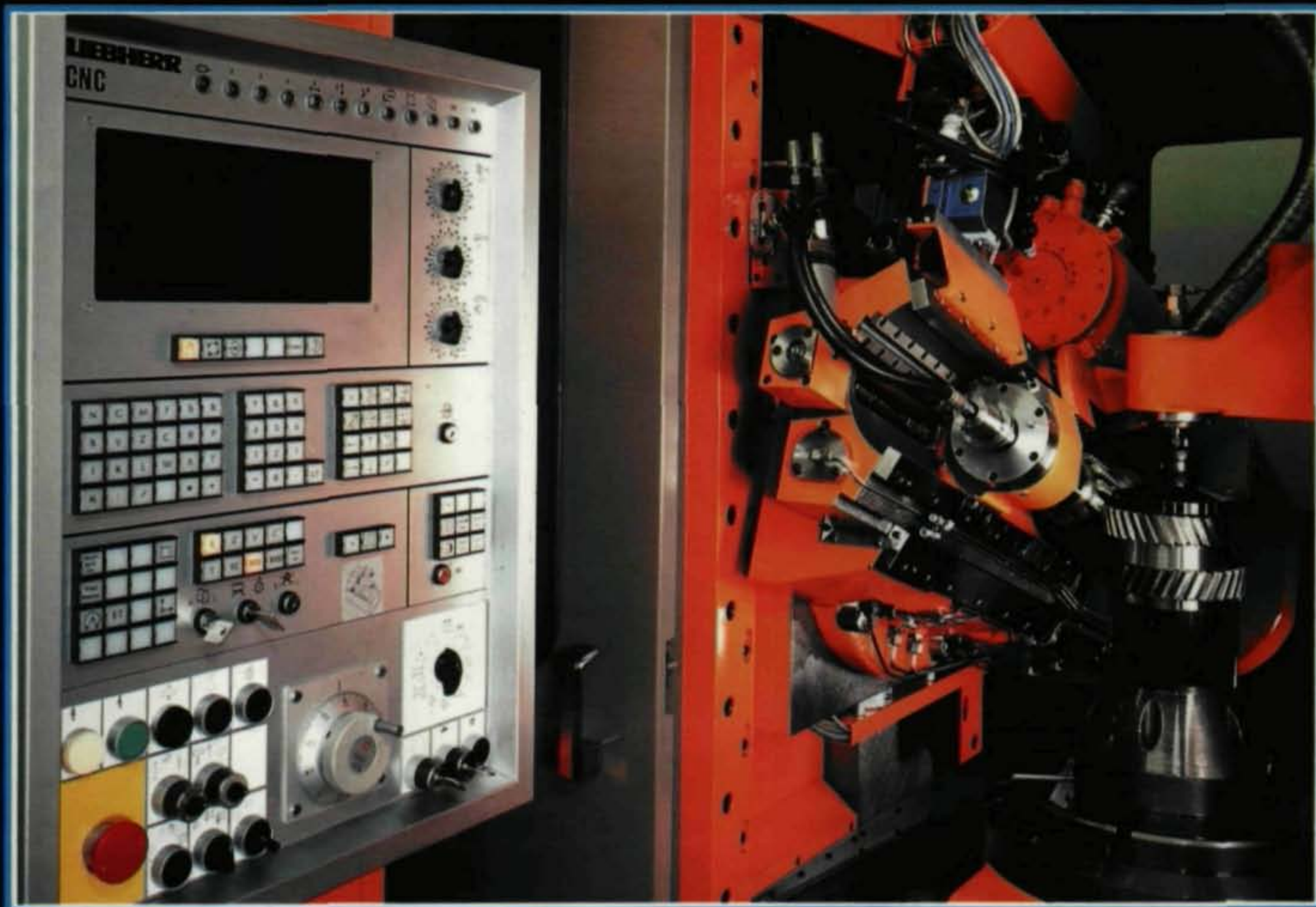
The main goal of this program is to demonstrate that the close tolerance forging process, combined with CAD/CAM, is an attractive and economical method for manufacturing spiral bevel gears. To achieve this goal, the use of advanced CAD/CAM techniques to design and manufacture the forging dies was necessary. The project was successful in that it:

- Developed the details of CAD/CAM for making the forging dies
- Forged gears with near net and net teeth surfaces
- Demonstrated the practicality and economics of precision forging spiral bevel gears.

It is expected that the techniques demonstrated by this project can be used for manufacturing spiral bevel gears (on a production basis) by forging. The matching pinions are to be manufactured conventionally by gear cutting. Thus, by eliminating the tooth cutting process for the gears, which represents the costliest operation in producing matching gear-pinion sets, considerable savings in manufacturing costs can be expected. In addition, existing data on forged bevel gears illustrate that forged gears are superior, in terms of fatigue life and load carrying capability, to cut gears. Consequently, a similar improvement in performance can also be expected from forged spiral bevel gears.

This article is based on work conducted by a team of engineers. The authors gratefully acknowledge the contribution of Dr. T. Altan, Battelle's Columbus Laboratories, and Mr. J. Chevalier, of U.S. Army Tank Command, Messrs. A. Sabroff and Mr. J. R. Douglas, from Eaton Corporation.

This article was reprinted from the U.S. Army's ManTech-Journal, Volume 8/ Number 4/1983.



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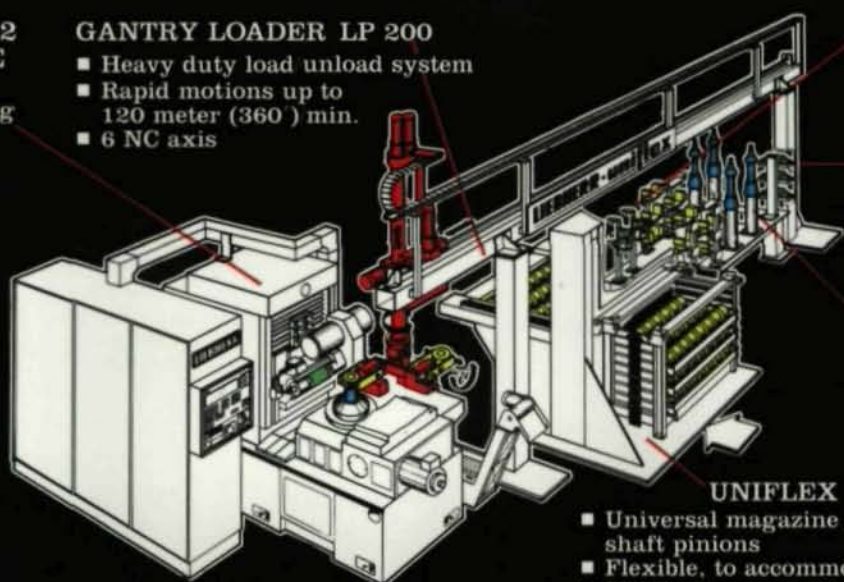

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