GEAR TECHNOLOGY

JANUARY/FEBRUARY 2002
The Journal of Gear Manufacturing
www.geartechnology.com

ALTERNATIVE GEAR MANUFACTURING

- Plastic
- Powdered Metal
- Forging

ALSO IN THIS ISSUE:

• Book Review • Moving Tooth Load and Crack Propagation • Addendum's Gear Word Search

THE GEAR INDUSTRY'S INFORMATION SOURCE


STAR EXPRESS HOBS...
Any HOB. Any Time.
Your tool of choice for material, coating, features, size and delivery options.
Fast: 3-week delivery on express hobs for involute gear manufacturing.
Flexible: Built to your requirements for size, gash, tooth form and material.
Gashes: Number of gashes will be determined by the tooth form data, unless otherwise specified. Tooth Forms: All forms, including topping, protuberance, modified pressure angles and non-symmetrical tooth forms can be supplied under the Express program.
Material: PM-M4 is our standard steel for hobs. For those tougher applications, optional materials are in stock. Coating: TiN, Ti (C,N) and (Ti,Al)N Gold Star Coatings are also available with only a slightly longer lead time.
Star: The Number One Choice for Products and Service.
Starcut Sales, Inc,. Subsidiary of Star Cutter Company.
23461 Industrial Park Drive. Farmington Hills, Michigan 48335-2855
Phone 248.474.8200 Fax 248.474.9518
www. starcutter. com
Since 1927
Star®
STAR CUTTER COMPANY
THE LATEST INNOVATION IN HOBBING.
SUPERIOR QUALITY, SERVICE AND DELIVERY.
AGMA
ISO 9001 REGISTERED
SIGMA POOL

CIRCLE 100

GEAR TECHNOLOGY

JANUARY/FEBRUARY 2002

The Journal of Gear Manufacturing

FEATURES

FOCUS ON ALTERNATIVE GEAR MANUFACTURING

26

DEPARTMENTS

a	t	h	g	e	a	r	s	p
h	e	g	o	y	u	b	m	i
y	a	f	a	b	e	d	e	n
t	r	i	n	f	b	i	n	v
i	r	i	p	p	l	i	n	g
c	g	n	i	n	o	h	n	e
i	f	g	h	h	u	j	n	g
r	a	c	k	e	u	g	i	t
t	b	y	n	l	j	h	h	g
n	b	i	d	i	d	x	d	d

48

Cover art courtesy of Metal Powder Industries Federation, Princeton, N.J.

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

EDITORIAL

Publisher & Editor-in-Chief
Michael Goldstein

Managing Editor William R. Stott

Associate Editor Joseph L. Hazelton

Editorial Consultant Paul R. Goldstein

Technical Editors
Robert Errichello
Don McVittie
Robert E. Smith
Dan Thurman

ART

Art Director Jean Bartz

ADVERTISING

Advertising Manager Patricia Flam
Advertising Coordinator Gladyce J. Cooper

CIRCULATION

Circulation Coordinator Dina Krauss

INTERNET

Web Developer Dan MacKenzie

***Gear Industry Home Page*™ Sales**
Patricia Flam
David Tomalis

RANDALL PUBLISHING STAFF

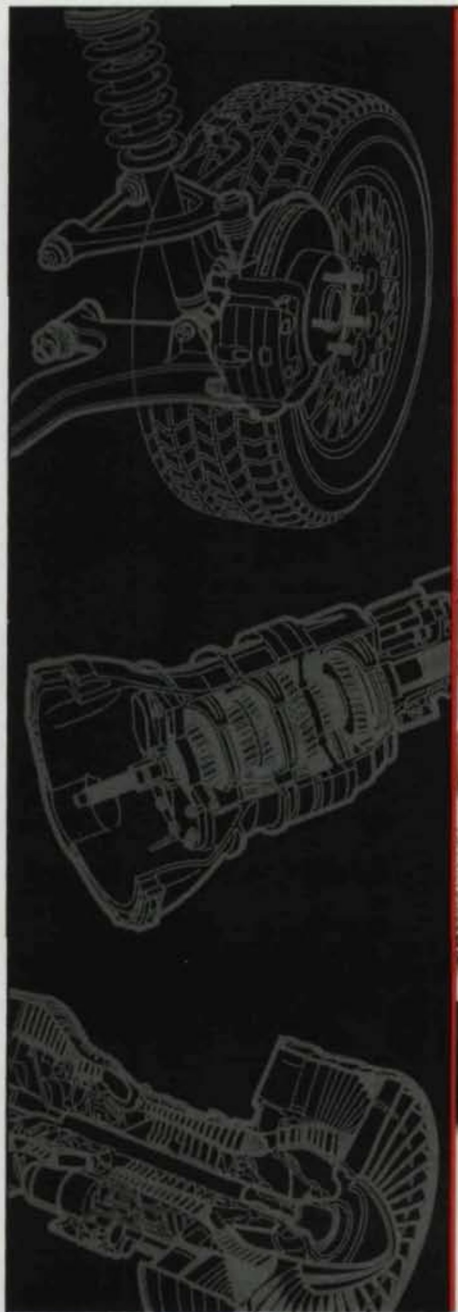
President Michael Goldstein
Vice President Richard Goldstein
Controller Michael Grafman
Accounting Dianne Johnson

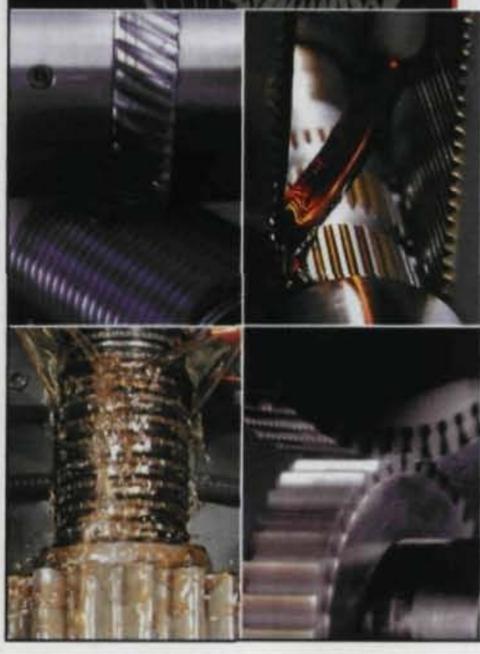
Phone: 847-437-6604
E-mail: ***people@geartechnology.com***
Web: ***www.geartechnology.com***
www.powertransmission.com

BPA INTERNATIONAL

VOL. 19, NO. 1

GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published bimonthly by Randall Publishing, Inc., 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL 60007, (847) 437-6604. Cover price $5.00 U.S. Periodical postage paid at Arlington Heights, IL, and at additional mailing office. Randall Publishing makes every effort to ensure that the processes described in GEAR TECHNOLOGY conform to sound engineering practice. Neither the authors nor the publisher can be held responsible for injuries sustained while following the procedures described. Postmaster: Send address changes to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL, 60007.





Cyber-Seminars: A Virtual Success

You hardly have to leave your office anymore, because the whole world is being piped right to your desktop. I know, because I recently attended my first seminar by Internet.

The seminar, sponsored by the American Gear Manufacturers Association, was called "The Economy and the Gear Market: What Comes Next?" It was presented by Dr. Mike Bradley, professor of economics at George Washington University in Washington, D.C. Dr. Bradley presented slides over the Internet while he spoke to participants via telephone.

Dr. Bradley's presentation, as always, was both interesting and informative. He's the economics professor everyone wishes he had in college. Not only is Dr. Bradley well informed about the gear industry, but also, he's able to explain economics in simple, easy-to-understand language.

Regrettably, there weren't a lot of positive things for Dr. Bradley to say in the middle of a manufacturing recession. But the presentation gave us a better idea of what's been happening and what to expect in the coming months.

Of course, the question everyone wants answered—when will the manufacturing recession be over?—can't be predicted with any certainty. However, history has provided some measure of assurance in this area. According to Dr. Bradley, the average recession since World War II has lasted around 11 months. Although some have lasted considerably longer, we're already more than a year into this manufacturing recession, so hopefully, we don't have to wait much longer for recovery.

While I wasn't terribly encouraged by Dr. Bradley's near-term expectations, I was certainly encouraged by the format and technology of the seminar itself. Dr. Bradley's slides appeared on our computer screen, and he controlled them from his location, flipping through them just as he would in a lecture hall. He was even able to highlight and animate portions of the presentation as he spoke.

Also, participants were able to ask questions, either by typing them into their browsers or, at certain points during the presentation, by speaking them over the phone lines. The phoned-in questions could be heard by all the other participants, just as if they were sitting at a traditional conference.

As a whole, I was very impressed at how the technology provided an experience nearly duplicating that of a live seminar.

This is not to say that the format does not have its drawbacks. It's not the same as being there in person. Much of the interaction that takes place in person can't happen in a virtual setting. Often, at conferences, it's the communication that takes place in the hallways or the hotel lobby that proves the most valuable. The value of that face-to-face contact can't be replaced altogether.

However, one of the main advantages of the Internet teleconference is its ability to expand the potential audience. According to Joe Franklin, AGMA's executive director, 42 company locations participated in the event. The average number of individuals at each location was 4.9, resulting in more than 200 total attendees. At a typical AGMA marketing committee meeting, somewhere between 50 and 75 people normally attend, Franklin says.

At most technical seminars, just one or two people from any company are able to attend. But with a virtual seminar, there's no airfare or accommodations to worry about, so the price of attending goes way down. Plus, the only time lost at the office is the time for the seminar itself.

Increasing the availability of technical information was one of the founding principles of this magazine. About 18 years ago, when I founded *Gear Technology*, many gear-related technical papers were being presented at conferences around the world, but most of the gear manufacturing community never saw the papers. The fact that a much larger audience is interested in that information is one of the reasons for this magazine's success.

The virtual meeting technology has similar potential. We'll soon see this format used for more comprehensive events, such as AGMA's many technical committee meetings or any number of other seminars and presentations held by other organizations.

According to Franklin, the AGMA plans to make good use of the technology over the next year. I believe that anything that gives us the ability to increase the spread of information—especially information that's of value to our industry—is commendable.

So I congratulate the AGMA for their efforts in this endeavor, and I look forward to future uses of this and similar technologies.

Michael Goldstein

Michael Goldstein, Publisher and Editor-in-Chief

The Gear Analysis Handbook by James L. Taylor Vibration Consultants Inc.

Reviewed by Robert E. Smith

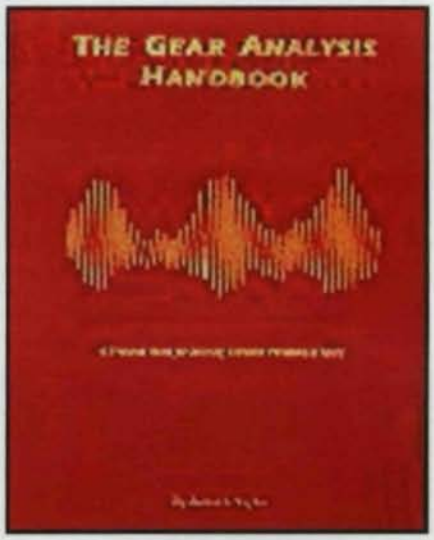
The author has written this book primarily from the viewpoint of analyzing vibrations on heavy industrial and mill gearing that may have been in service for a prolonged time. The purpose is to diagnose problems, especially the source or cause of failure. However, the principles and analysis techniques can be used for all types and sizes of gears, as well as for gear noise analysis.

Gears are of complex geometry, and there are many possible sources of problems, including the mounting or assembly (housing). The author has done a very thorough job of describing the techniques of analyzing vibration signals, both in the time domain as well as the frequency domain (FFT). He has stressed the importance of looking at time-based signals. Many times, engineers look at only the frequency spectrum and will miss some very important data or the fact that something about the gears has distorted the spectrum or made the data useless, such as nicks and burrs or overloaded signals.

There are many bits of information in a gear vibration or noise spectrum besides just mesh frequency. The author has been very thorough in describing the causes of "unusual" peaks in a spectrum (other than mesh frequency and harmonics of mesh). These are such things as 1/2 and 1-1/2 harmonics of mesh, which occur quite often.

Some other unusual peaks are "ghost harmonics." In the gear trade, these are peaks caused by "undulations," which are a unique form of waviness on the gear teeth, caused by kinematic errors in the gear train of the machine that produced the gear teeth. The culprit is usually the final drive gear, mounted directly on the workspindle. There has to be an integer number of waves around the product gear, which is equal to the number of teeth on the workspindle drive gear. Ghost harmonics are the type of thing that submarines have looked for in the analysis of sonar data. They were used to identify ships and the machines that produced the marine drive gears on the ships. Current AGMA and ISO standards on gear tooth surface finish and texture describe this ghost harmonics phenomenon.

Chapter 10 is a good place for any technician or engineer to start when thinking about making vibration or noise measurements of gears or a gearbox. The rest of the book goes a long way to guide the person in methods and techniques of getting the most information out of the analysis. ⭘



The Gear Analysis Handbook, ***ISBN 0-9640517-1-0, was published in 2000 by Vibration Consultants Inc. The 256-page book costs $109.95 plus shipping. It can be ordered through Vibration Consultants Inc., by calling (813) 839-2826, by sending e-mail messages to*** **info@vibcons.com** ***or by visiting*** **www.vibcons.com.**

Circle 304

Tell Us What You Think . . .
If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618.

ADVERTISER INDEX

For more information about a product or service advertised in this issue of *Gear Technology*, circle the appropriate number on the Reader Response Card and put the card in the mail.

TRY OUR RAPID READER RESPONSE SYSTEM!
Go to *www.geartechnology.com/rrr.htm* to request additional information from any advertiser in this issue. Your request will be sent to the advertiser within 24 hours for super-fast turnaround!

ADVERTISER	READER SERVICE NUMBER	PAGE NUMBER
Aero Gear Inc.	127	43
American Metal Treating Co.	181	47
Applied Process Inc.	131	46
Arrow Gear Co.	166	4
Bourn & Koch Machine Tool Co.	156	40
Dr. Kaiser (S. L. Munson)	116	41
Dragon Precision Tools Co. Ltd. (Greg Allen Co.)	149	12
Dura-Bar	158	5
Enplas Inc.	152	13
Fässler Corp.	119	9
The Gear Industry Home Page™		22
Gear Technology Center, Div. of Mitsubishi International Corp.	175	24–25
Gleason Corp.	162	46
Gleason Cutting Tools Corp.	105, 147, 160	1, 47, 46
The Gleason Works	110	IFC
Insco Corp.	144	39
Interstate Tool Corp.	151	47
ITW Heartland	124	39

ADVERTISER	READER SERVICE NUMBER	PAGE NUMBER
KISSsoft	136	46
Koepfer America L.L.C.	137	47
Koro Sharpening Service	163	47
Kreiter Geartech	141	47
LeCount Inc.	164	4
Midwest Gear & Tool Inc.	168	40
Midwest Gear Corp.	154	47
Nachi Machining Technology Co.	114	6
Okamoto Machinery	153	47
Perry Technology Corp.	134	IBC
powertransmission.com™	171	8
Process Equipment Co.	122	45
The Purdy Corp.	170	34
Randall Publishing Inc.	328	36
Raycar Gear & Machine Co.	180	46
Star Cutter Co.	100, 142	2, 47
SU America Inc.	107	BC
TSA Transmecanica (Greg Allen Co.)	172	43
United Tool Supply	123	10
University of Wisconsin—Milwaukee	130	33

Gordon New Wins Gear Clock

Gordon New, managing director of operations for Ronson Gears of Australia, was the grand prize winner in *Gear Technology*'s drawing held in October at Gear Expo 2001 in Detroit. New won a one-of-a-kind gear clock sculpture, custom made for the event.

The drawing was part of *Gear Technology*'s booth promotion at Gear Expo. The emphasis of the booth was "Marketing in the Gear Industry." Individuals had the opportunity to consult with our staff about the advertising and promotional opportunities available to companies in the gear industry or companies wanting to reach the gear industry.

Visitors to the booth entered the contest by dropping their business cards in a box. The drawing was held Tuesday, October 9.

In addition to the custom-made original clock, *The Gear Industry Home Page*™ and *powertransmission.com*™ held drawings for smaller worm-and-wheel clocks. The winners of those drawings were:

- John R. Arbisi, Ingersoll Contract Manufacturing Co.;
- Gerard J. Connell, Cloyes Gear & Products Inc.;
- Jeff Coursey, Nachi Machining Technology Co.;
- Alexander J. Gunow, Midwest Thermal-Vac;
- George T. Shturtz, Metal Powder Products Co.; and
- Paul Wandler, L&H Welding & Machine Co.

Congratulations to all of the winners, and thank you to all who came to Gear Expo and visited us. Those who are interested in a marketing consultation, but who didn't have the chance to come to the show, can call us at (847) 437-6604.

Circle 300

Gordon New of Ronson Gears (left) and *Gear Technology* publisher Michael Goldstein hold the gear clock that New won in the *Gear Technology* drawing at Gear Expo 2001.

Grinding by Broaching

The parts: internal gears with small diameters and heat-treat distortion. The problem: how to grind the distortion from the teeth? A solution: grind by broaching.

Fässler AG has a modified broaching process that provides the surface roughness of ground gears where grinding isn't possible because of space problems, like in internal gears with small diameters.

The process uses a diamond-coated short broach and multiple, up-and-down strokes to remove heat-treat distortion from broached, hardened workpieces.

Located in Dübendorf, Switzerland, Fässler has offered the process since the mid-1990s. While not new, the diamond-coated short broach appears unique as a finishing tool for internal gears. According to Martin Gerber, a Fässler salesman, only his company makes such a broach. The broach operates in the company's HS-100 hard broaching machine.

Bülent Yesilalp, Fässler's sales director, describes the short broach process as simple and reliable, and as a low cost solution for high volume production.

In this process, the internal gear lies on the broaching machine's deposit table and is held in place by a hold-down bar. The gear isn't rigidly clamped, so the broach can move it according to the gear's center.

With a mounting flange, the short broach has a centering zone to position the gear with the profile and check the allowable runout, a tapered zone to remove stock, and a ground cylindrical zone to calibrate, or flatten, the profile.

In a normal broaching machine, driven by a hydraulic cylinder, the broach is pulled down through a gear blank in one stroke.

In the HS-100, the broach is pushed up and pulled down through an internal gear's opening. When the broach exits up or down, chips are rinsed from between the broach and gear by the deposit table's ring nozzle. Designed to finish gears, the broach can remove only 25–40 microns of stock.

Also, the short broach process avoids a problem with the long broach process:

Welcome to* Revolutions*, the column that brings you the latest, most up-to-date and easy-to-read information about the people and technology of the gear industry.* Revolutions *welcomes your submissions. Please send them to* Gear Technology*, P.O. Box 1426, Elk Grove Village, IL 60009, fax (847) 437-6618 or e-mail* people@geartechnology.com. *If you'd like more information about any of the articles that appear, please circle the appropriate number on the Reader Response Card.

elastic deformation.

Fässler has produced diamond-coated long broaches for more than 20 years. Such long broaches could be made for finishing internal gears. But, the broach's single, long stroke expands its workpieces, which later shrink.

The short broach process uses multiple, short strokes, so elastic deformation in a gear from one stroke can shrink and be removed during the next stroke.

"We have no expansion in the workpiece," Gerber says. "That was the reason to change to this short broach process."

Thus, the short broach grinds heat-treat distortion from internal gears, giving them their proper profile, within their tolerance range. The broach provides such accuracy whether it's new or old.

Fässler Corp.'s multiple-stroke short broaches, like this one, can finish-broach internal gears with small diameters and heat-treat distortion. A gear's major diameter can be as small as 20 mm.

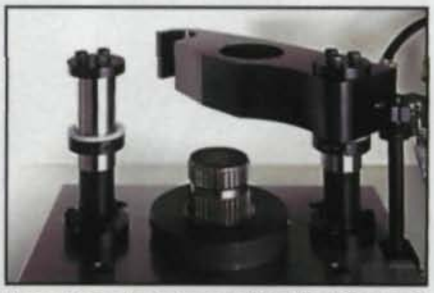
When new, the broach's dimensions are their largest, so its finished gears will have dimensions at the lower, smaller end of their tolerance range.

As the broach is used, its diamond grit will wear away. The broach's dimensions will become smaller, so its gear's dimensions will move toward the upper, larger end of their tolerance range.

When the broach is smallest in size, and needs to be replated, gear dimensions will be at the upper end of their tolerance range. The change in the broach's size is the tolerance range of its internal-gear workpiece.

The workpiece's dimensions can range from 30–250 mm for its outside diameter, 20–80 mm for its internal-gear diameter, 3–100 mm for its height and 3–55 mm for its gear-profile height.

Each diamond-coated short broach has a lifetime of 200–300 meters of broaching length and a cycle time of



A short broach sticks out of the HS–100 deposit table and ring nozzle, below the machine tool's hold-down bar. The broach uses multiple up-and-down strokes to provide internal gears with the surface roughness of ground gears.

The short broach centers a gear with its upper set of teeth, then removes stock and flattens the gear's profile with the lower set of teeth.

20–40 seconds to finish an internal gear.

Assuming a gear height of 20 mm, each broach can finish-broach the flank and major diameter of 10,000–15,000 gears. Also, each gear would have a cycle time of 30 seconds and cost about 30 cents to finish. The 30 cents per gear includes the cost of replating the broach during its lifetime. Each new broach costs $6,000–$15,000, depending on its size.

The broach's lifetime is defined by its stroke length: The longer the stroke, the greater the stress on the broach's diamond coating and the shorter the broach's lifetime. That lifetime also can be shortened by work parameters set by the machine operator.

At its smallest size, the broach must have its coating removed, then be replated and reground with a new coating of metallically attached industrial-diamond grit. The broach can be replated three times before it must be discarded. Replating can be done by companies other than Fässler.

Besides removing heat-treat distortion, the broach removes an internal gear's helix, pitch and taper errors. Gerber and Yesilalp add that the broach increases the gear's contact ratio with lower peak stresses and lengthens the gear's lifetime.

According to Yesilalp, the short broach process can simplify assembly of gears and shafts. He explains that the process creates gears with correct dimensions—gears don't even need to be measured—so gears and shafts don't have to be built in pairs.

He adds that the process can reduce heat treatment costs for some applications by eliminating over-pinion heat treatment.

And, Fässler's short broach process isn't limited to internal gear shapes. The process also can finish-broach single and multiple keyways, polygons and other spline profiles.

Circle 301

Tell Us What You Think . . .

If you found this column of interest and/or useful, please **circle 302.**
If you did not care for this column, **circle 303.**

Consideration of Moving Tooth Load in Gear Crack Propagation Predictions

David G. Lewicki, Lisa E. Spievak, Paul A. Wawrzynek, Anthony R. Ingraffea and Robert F. Handschuh

This paper was presented at the 8th International Power Transmission and Gearing Conference, Baltimore, MD, September 2000. It was later published in the *Journal of Mechanical Design*, March 2001, by the American Society of Mechanical Engineers.

David G. Lewicki

is a senior aerospace engineer with the U.S. Army Research Laboratory at NASA's Glenn Research Center in Cleveland, OH. He has been involved in gear crack propagation research, as well as transmission life and reliability predictions and gear dynamics predictions. He also has worked on low-noise, high-strength spiral bevel gears; face gears for helicopter drive systems; lubricants; and diagnostics. He has written or co-written more than 70 technical articles in the field of drive systems.

Lisa E. Spievak

is a structural engineer in the design and analysis group of ATA Engineering Inc., located in San Diego, CA. She is an expert in applying computer techniques to design, analyze and test highly stressed structures and in interpreting fracture and fatigue requirements for those structures. As a graduate student, she performed research with the Cornell Fracture Group in which she simulated three-dimensional fatigue crack growth in spiral bevel gears.

Introduction

Effective gear designs balance strength, durability, reliability, size, weight, and cost. Even effective designs, however, can have the possibility of gear cracks due to fatigue. In addition, truly robust designs consider not only crack initiation, but also crack propagation trajectories. As an example, crack trajectories that propagate through the gear tooth are the preferred mode of failure compared to propagation through the gear rim. Rim failures will lead to catastrophic events and should be avoided. Analysis tools that predict crack propagation paths can be a valuable aid to the designer to prevent such catastrophic failures.

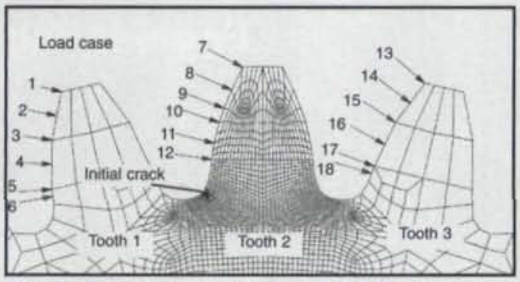


Figure 1—Location of load cases for finite element mesh.

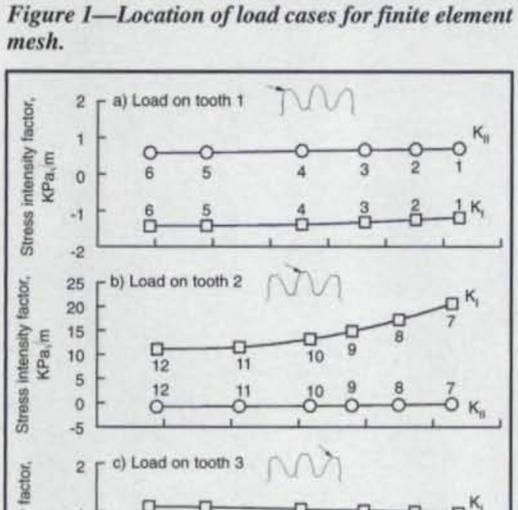


Figure 2—Mode I and mode II stress intensity factors for a unit load and an initial crack of 0.26 mm.

Pertaining to crack analysis, linear elastic fracture mechanics applied to gear teeth has become increasingly popular. The stress intensity factors are the key parameters to estimate the characteristics of a crack. Analytical methods using weight-function techniques to estimate gear tooth stress intensity factors have been developed (Refs. 1 and 17). Numerical techniques, such as the boundary element method and finite element method, have also been studied (Refs. 12 and 21). Based on stress intensity factors, fatigue crack growth and gear life predictions have been investigated (Refs. 2, 3, 5 and 9). In addition, gear crack trajectory predictions have been addressed in a few studies (Refs. 6, 7, 13, 14 and 19).

From publications on gear crack trajectory predictions, the analytical methods have been numerical (finite element method or boundary element method) while solving a static stress problem. In actual gear applications, however, the load moves along the tooth, changing in both magnitude and position. No work has been done investigating the effect of this moving load on crack trajectories.

The objective of the current work is to study the effect of moving gear tooth load on crack propagation predictions. Two-dimensional analysis of an involute spur gear using the finite element method is discussed. Also, three-dimensional analysis of a spiral-bevel pinion gear using the boundary element method is discussed. A quasi-static numerical simulation method is presented in which the gear tooth engagement is broken down into multiple load steps, with each step analyzed separately. Methods to analyze the steps are discussed, and predicted crack shapes are compared to experimental results.

Two-Dimensional Analysis

Gear Modeling. The two-dimensional analysis was performed using the FRANC (FRacture ANalysis Code) computer program developed by

Wawrzynek (Ref. 23). The program is a general-purpose finite element code for the static analysis of two-dimensional cracked structures. The program uses principles of linear elastic fracture mechanics and is capable of analyzing plane strain, plane stress, or axi-symmetric problems. A unique feature of the program is the ability to model cracks and crack propagation in a structure. A rosette of quarter-point, six-node, triangular elements is used around the crack tip to model the inverse square-root stress singularity. Mode I and mode II stress intensity factors, K_I and K_{II} respectively, can be calculated using a variety of methods. (As a refresher, mode I loading refers to loads applied normal to the crack plane and tends to open the crack. Mode II refers to in-plane shear loading.) The stress intensity factors quantify the state of stress in the region near the crack tip. In the program, the stress intensity factors can be used to predict the crack propagation trajectory angles, again using a variety of methods. In addition, the program has a unique re-meshing scheme to allow automated processing of the crack simulation.

A spur gear from a fatigue test apparatus was modeled to demonstrate the two-dimensional analysis. The modeled gear had 28 teeth, a 20° pressure angle, a module of 3.175 mm (diametral pitch of 8/in.), and a face width of 6.35 mm (0.25 in.). The gear had a backup ratio (defined as the rim thickness divided by the tooth height) of 3.3. The complete gear was modeled using mostly 8-node, plane stress, quadrilateral finite elements. For improved accuracy, the mesh was refined on one of the teeth in which a crack was inserted. The total model had 2,353 elements and 7,295 nodes. Four hub nodes at the gear inner diameter were fixed to ground for boundary conditions. The material used was steel.

Tooth Loading Scheme. To determine the effect of gear tooth moving load on crack propagation, the analysis was broken down into 18 separate load cases (Fig. 1). An initial crack of 0.26 mm (0.010 in.) in length was placed in the fillet of tooth 2, normal to the surface, at the location of the maximum tensile stress (uncracked condition). Six load cases were analyzed separately with the load on the tooth ahead of the cracked tooth, six on the cracked tooth, and six on the tooth after the cracked tooth. The calculated stress intensity factors for unit loads at each of the load positions are shown in Figure 2. These stress intensity factors were calculated using the J-integral technique (Ref. 20). Loads on tooth 2 (cracked tooth) produced tension at the crack tip.

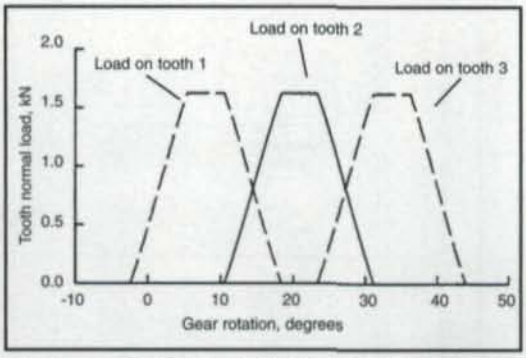


Figure 3—DANST computer program output of static gear tooth load, 68 N-m driver torque.

K_I increased as the load moved toward the tooth tip (load cases 12 to 7, Fig. 2b) due to the increased load lever arm. Loads on tooth 3 also produced tension at the crack tip, but at an order of magnitude less than those produced from the loads on tooth 2 (Fig. 2c). Loads on tooth 1 gave compression to the crack tip as shown by the negative K_I values (Fig. 2a).

Next, the actual load magnitudes on the gear tooth were considered as it went through the mesh. The computer program DANST (Dynamic ANalysis of Spur gear Transmission, Ref. 15) was used for the analysis. This program is based on a four-degree-of-freedom, torsional, lumped mass model of a gear transmission. The model includes driving and driven gears, connecting shafts, a motor, and a load. The equations of motion for this model were derived from basic gear geometry, elementary vibration principles, and time-varying tooth stiffnesses. For simplicity, the static gear tooth loads of the solution were determined (Fig. 3). These loads were determined from well-established gear tooth stiffness principles and static equilibrium. The loads are shown as a function of gear rotation for a driver torque of 68 N-m (599 in.-lb.). Tooth 2 began contact at a gear rotation of 10°. As the gear rotation increased, the load on tooth 2 gradually increased. Tooth 1 and tooth 2 shared the load for a rotation from 10° to 18°. From 18° to 23°, tooth 2 carried the complete load. At 23°, tooth 2 is considered at its highest point of single tooth contact (HPSTC).

The stress intensity factors as a function of gear rotation were then determined by multiplying the stress intensity factors determined from the units' loads (Fig. 2) by the actual tooth loads (Fig. 3) and applying superposition since linear elastic fracture mechanics was used. The results are shown in Figure 4. As expected, the mode I stress intensity factor (Fig. 4a) was mostly influenced by the load on tooth 2. Note that the largest value of K_I occurred at the HPSTC. Also note that

Paul A. Wawrzynek

is a senior research associate in the Computational Materials Institute at Cornell University's Theory Center and in the Cornell Fracture Group. A civil engineer, he has focused his research primarily on developing software for simulating crack growth in a range of engineering structures and materials. Also, he manages Fracture Analysis Consultants Inc., which provides crack growth analysis and software development services for different industries.

Anthony R. Ingraffea

is a professor at the School of Civil and Environmental Engineering at Cornell University, located in Ithaca, NY. A civil engineer, Ingraffea focuses his research on computer simulation and physical testing of complex fracturing processes. He and his students have performed research in using interactive computer graphics in computational fracture mechanics. With his students, he has written more than 180 papers on complex fracturing processes and computational fracture mechanics.

Robert F. Handschuh

is an aerospace engineer with the U.S. Army Research Laboratory at NASA's Glenn Research Center. His research has concentrated on power transmission; his research in gearing has focused on experimental and analytical studies of spiral bevel, face, and high-speed gearing. He has written or co-written more than 65 reports in the fields of seal, gear and numerical methods.

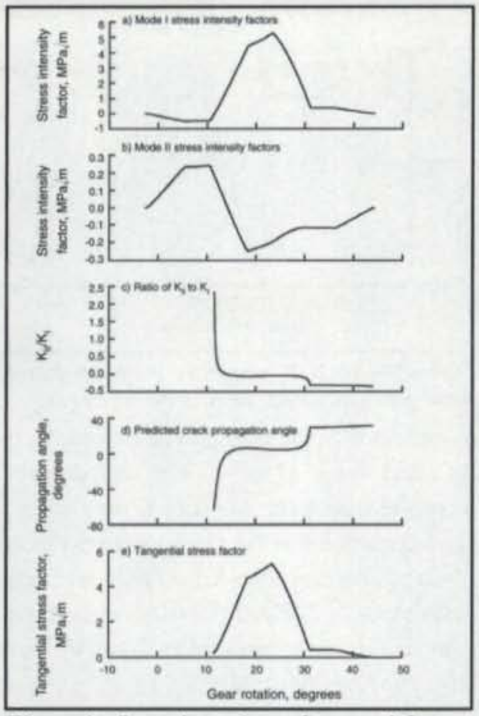


Figure 4—Stress intensity and tangential stress factors as a function of gear rotation, 68 N-m driver torque, 0.26 mm initial crack size.

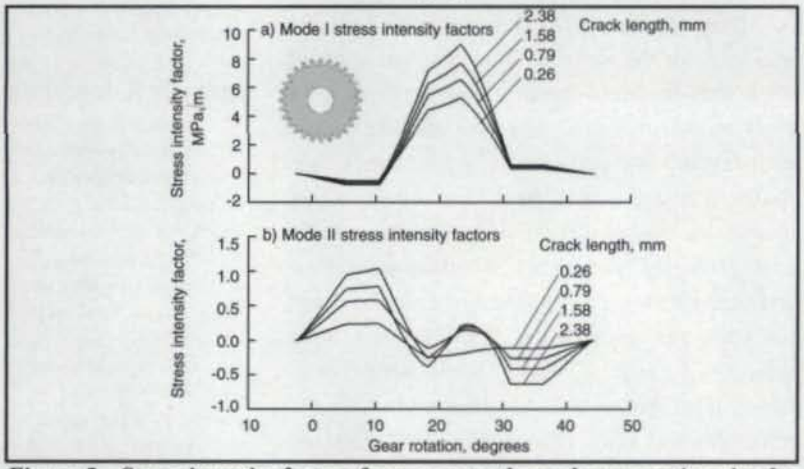


Figure 5—Stress intensity factors from gear tooth crack propagation simulation, backup ratio = 3.3.

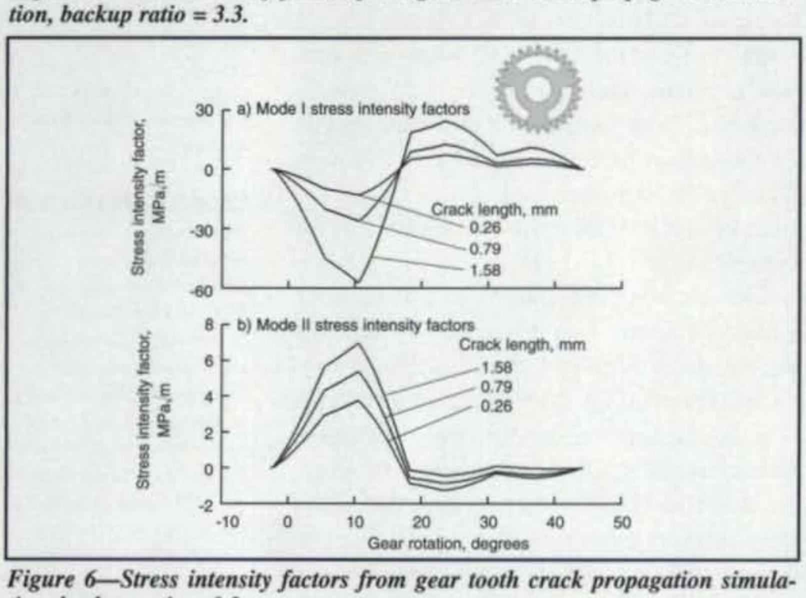


Figure 6—Stress intensity factors from gear tooth crack propagation simulation, backup ratio = 0.2.

the magnitude of K_I (Fig. 4a) was much larger than that of K_{II} (Fig. 4b). This implied that K_I was the driving force in the crack propagation. K_{II}, however, affected the crack propagation angle, as will be shown in the next section.

Crack Propagation Simulation. From Williams (Ref. 24), the tangential stress near a crack tip, $\sigma_{\theta\theta}$, is given by

$$\sigma_{\theta\theta}=\frac{1}{\sqrt{2\pi r}}\left(K_I\cos^3\frac{\theta}{2}-3K_{II}\sin\frac{\theta}{2}\cos^2\frac{\theta}{2}\right) \quad (1)$$

where r and θ are polar coordinates with the origin at the crack tip. Erdogan and Sih (Ref. 8) postulated that crack extension starts at the crack tip and grows in the direction of the greatest tangential stress. The direction of the greatest tangential stress is determined by taking the derivative of Equation 1 with respect to θ, setting the expression equal to zero, and solving for θ. Performing the math, this predicted crack propagation angle, θ_m, is given by

$$\theta_m = 2\tan^{-1}\left[\frac{\frac{K_I}{K_{II}}\pm\sqrt{\left(\frac{K_I}{K_{II}}\right)^2+8}}{4}\right] \quad (2)$$

From Equation 2, the predicted crack propagation angle is a function of the ratio of K_I to K_{II}. Erdogan and Sih (Ref. 8) used brittle plexiglass plates under static loading to validate their proposed theorems (i.e., the ratio of K_I to K_{II} was constant). For the gear problem in the current study, however, the ratio of K_I to K_{II} was not constant during gear rotation. This is shown in Figure 4c (actually plotted as the ratio K_{II} to K_I for clarity). In addition, Figure 4d gives the calculated θ_m from Equation 2 as a function of gear rotation.

In order to simulate gear crack propagation, a modification to the Erdogan and Sih theory was postulated in the current study. This modified theory states that the crack extension starts at the crack tip and grows in the direction of the greatest tangential stress as seen during engagement of the gear teeth. The procedure to calculate the crack direction is as follows:

1) K_I and K_{II} are determined as a function of gear rotation (Figs. 4a and 4b, as described in the previous section),

2) the ratio of K_I to K_{II} as a function of gear rotation is determined (Fig. 4c),

3) θ_m (using Eq. 2) as a function of gear rotation is determined (Fig. 4d),

4) $\sigma_{\theta\theta}$ (using Eq. 1) as a function of gear rotation is determined (Fig. 4e),

5) the predicted crack direction is the value of θ_m

for which $\sigma_{\theta\theta}$ is greatest during gear rotation.

For the gear example given, the tangential stress factor (defined as $\sigma_{\theta\theta}\sqrt{2\pi r}$) is plotted as a function of gear rotation in Figure 4e. This plot looks very similar to the mode I stress intensity factor plot (Fig. 4a) since K_I was much larger than K_{II} (see Eq. 1). The tangential stress was largest at the HPSTC (gear rotation of 23°) and the predicted crack propagation angle at this gear rotation was $\theta_m = 4.3°$.

Using this propagation angle, the crack was extended by 0.26 mm (0.010 in.), re-meshed, re-analyzed, and a new propagation angle was calculated using the method described above. This procedure was repeated a number of times to produce a total crack length of 2.38 mm (0.094 in.). The 0.26-mm crack extension length was based on prior experience in order to produce a smooth crack path. Figure 5 shows the stress intensity factors versus gear rotation for a number of crack lengths. Note that the mode I stress intensity factors looked similar but with increased magnitude as the crack length increased. In all cases, the selected crack propagation angles occurred when the tooth load was placed at the HPSTC. Figure 6 shows a similar analysis but with a model of a thin-rimmed gear. Here, the gear was modeled based on the previous design, but with slots incorporated in the rim to simulate a thin-rimmed gear. The backup ratio for this model was 0.2. As seen, the magnitudes of the mode I stress intensity factors during tension (gear rotations 18° to 45°) were larger than that of the 3.3 backup ratio gear. Also, there was a significant increase in the compressive K_I (gear rotation less than 18°) due to the increased compliance of the thin rim gear.

Comparison to Experiments. Figure 7 shows the results of the analysis compared to experimental tests in a gear fatigue apparatus. The original model (backup ratio of 3.3), as described before, was compared along with models of backup ratio of 1.0 and 0.3. These later two models were created using slots in the gear blank, as previously described. The experiments were first reported by Lewicki and Ballarini (Ref. 13). Here, notches were fabricated in the tooth fillet region to initiate tooth cracking of test gears of various rim thicknesses. The gears were run at 10,000 rpm and at a variety of increasing loads until tooth or rim fracture occurred. As seen from the figure, good correlation of the predicted crack trajectories to experimental results was achieved. For backup ratios of 3.3 and 1.0, tooth fractures occurred. For the backup ratio of 0.3, rim fracture occurred.

As a final note, the analysis indicated that the maximum tangential stress at the crack tip always occurred when the tooth load was positioned at the HPSTC. Thus, for two-dimensional analysis, crack simulation based on calculated stress intensity factors and mixed mode crack angle prediction techniques can use a simple static analysis in which the tooth load is located at the HPSTC. This was based on a modification to the Erdogan and Sih crack extension theory and the fact that the mode I stress intensity factor was much larger than the mode II factor.

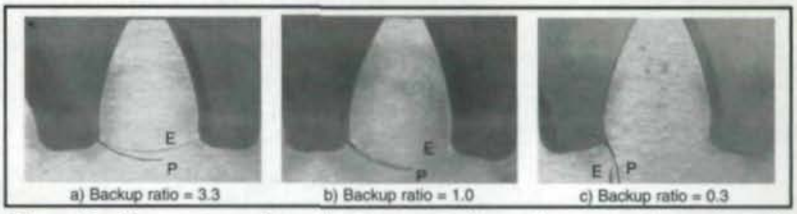


Figure 7—Comparison of predicted gear tooth crack propagation paths with experimental results (P = predicted, E = experiments).

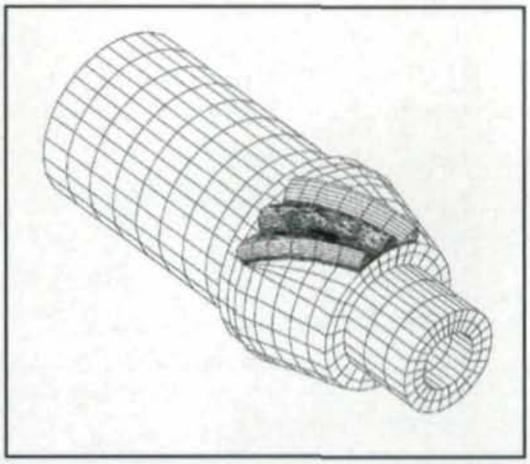


Figure 8—Boundary element model of OH-58 spiral-bevel pinion.

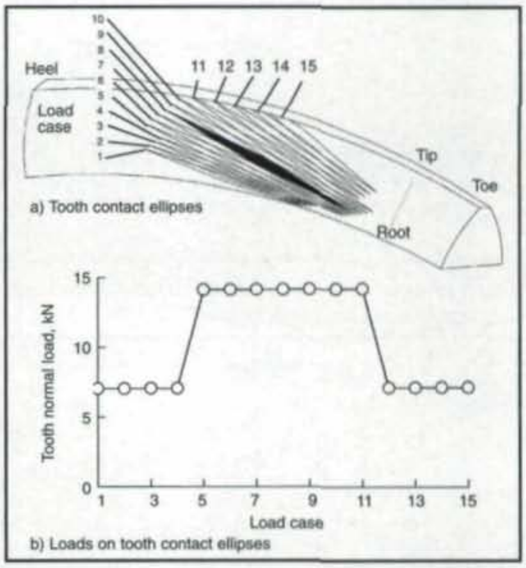


Figure 9—Location of tooth contact ellipses and magnitude of load on OH-58 spiral-bevel pinion tooth.

Three-Dimensional Analysis

Gear Modeling. The three-dimensional analy-

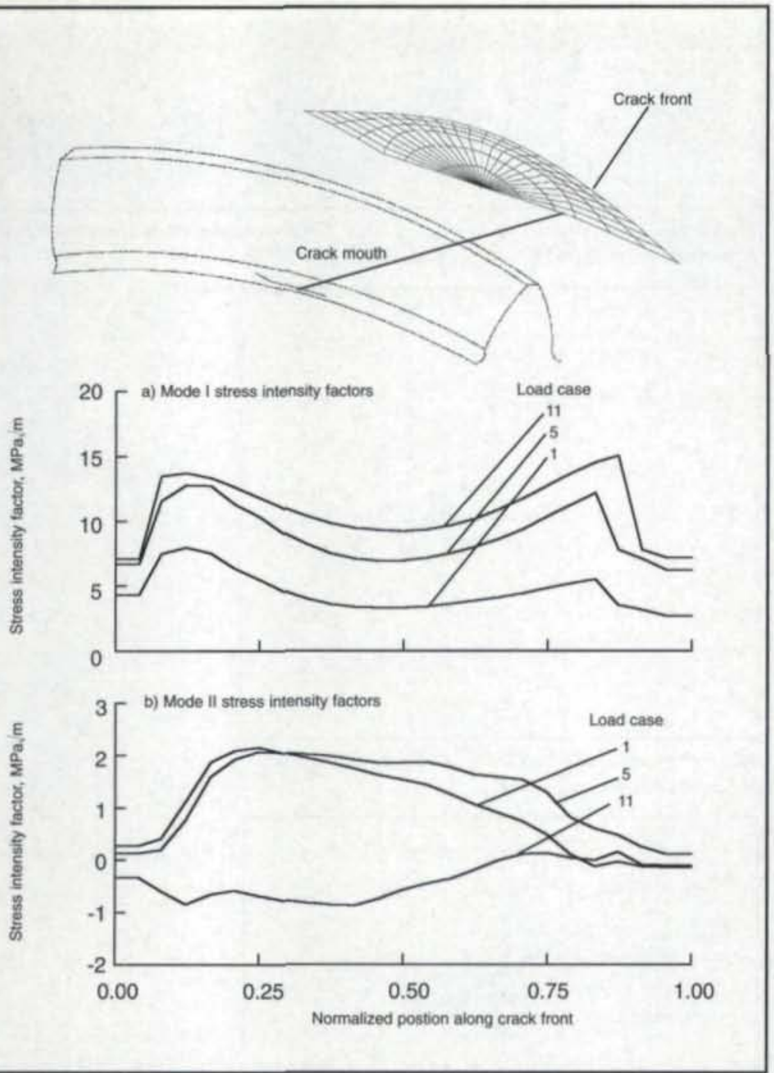


Figure 10—Stress intensity factors from three-dimensional OH-58 pinion tooth crack propagation simulation; step 1, crack area = 5.96 mm².

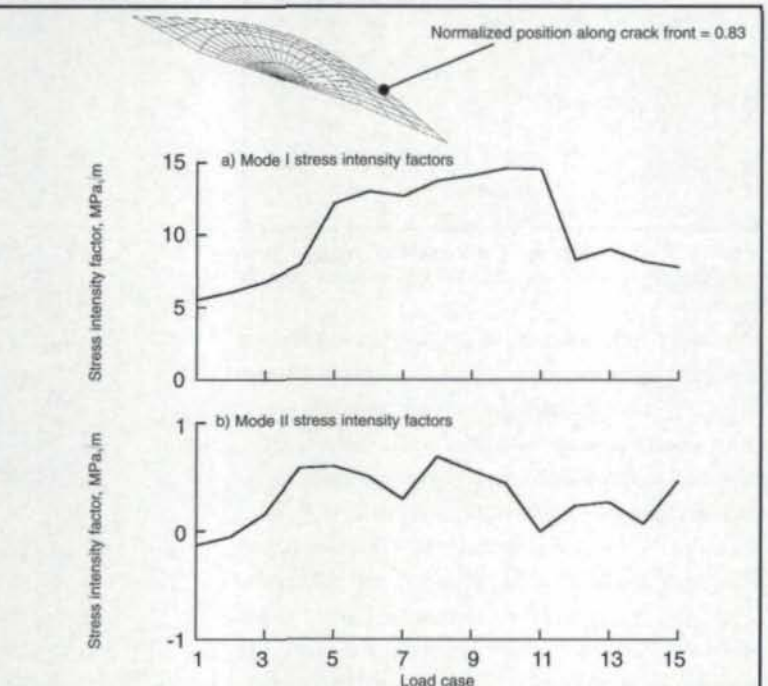


Figure 11—Stress intensity factors from three-dimensional OH-58 pinion tooth crack propagation simulation; step 1, normalized position along crack front = 0.83.

sis was performed using the FRANC3D (FRacture ANalysis Code for 3 Dimensions) computer program developed by Wawrzynek (Ref. 23). This program uses boundary element modeling and principles of linear elastic fracture mechanics to analyze cracked structures. The geometry of three-dimensional structures with non-planar, arbitrary shaped cracks can be modeled. The modeling of a three-dimensional cracked structure is performed through a series of programs. Structure geometry grid point data are imported to a solid modeler program. Here, appropriate curves and faces (or patches) are created from the grid data, as well as a closed-loop surface geometry model. This surface model is then imported to the FRANC3D program for boundary element model preparation. The user can then mesh the geometry model using 3- or 6-node triangular surface elements, or 4- or 8-node quadrilateral elements. Boundary conditions (applied tractions and prescribed displacements) are applied on the model geometry over faces, edges, or points. Initial cracks, such as elliptical or penny shaped, can be inserted in the structure. After complete formulation, the model is shipped to a boundary element equation solver program. Once the displacement and traction unknowns are solved, the results are exported back to the FRANC3D program for post-processing. Fracture analysis, such as stress intensity factor calculations, can then be performed.

The spiral-bevel pinion of the OH-58C helicopter main rotor transmission was modeled to demonstrate the three-dimensional analysis. The pinion had 19 teeth, a 20° pressure angle, a 30° mean spiral angle, a module of 3.66 mm (diametral pitch of 6.94/in.), and a face width of 32.51 mm (1.28 in.). For OH-58 operation, the pinion mates with a 71-tooth spiral-bevel gear, operates at 6,060 rpm, and has a design torque of 350 N-m (3,099 in.-lbs.).

The boundary element model of the OH-58 pinion developed by Spievak (Ref. 22) was used for the study. Three teeth, the rim cone, and the bearing support shafts were modeled (Fig. 8). The tooth surface and fillet coordinates were determined from the methods developed by Handschuh and Litvin (Ref. 11) and Litvin and Zhang (Ref. 16). The mesh of the three teeth was refined for improved accuracy. A half-ellipse initial crack with major and minor diameters of 3.175 mm and 2.540 mm, respectively (0.125 in. and 0.100 in.), was placed in the fillet of the middle tooth normal to the surface. The crack was centered along the face width and centered along the fillet. The complete gear model had a total of about 2,600 linear elements (both triangular and quadrilateral) and about 2,240 nodes. For boundary conditions, the end nodes of the larger-diameter shaft were fixed and the nodes on the outer diameter of the smaller-diameter shaft were constrained in the radial directions. Again, the material was steel.

Tooth Contact Analysis and Loading Scheme. Due to the geometrical complexities and three-dimensional action, numerical methods are required to determine the contact loads and positions on spiral-bevel teeth since no closed-form solution exists. The method of Litvin and Zhang (Ref. 16) was used to determine the mean contact points on the spiral-bevel pinion tooth. The method modeled tooth generation and tooth contact simulation of the pinion and gear. With the mean contact points taken as the centers, contact ellipses were determined using Hertzian theory (Ref. 10). Figure 9 shows the estimated contact ellipses on the spiral-bevel pinion tooth. Fifteen separate ellipses (load cases) were determined, starting from the root of the pinion and moving toward the tooth tip and toe. Load cases 1–4 and 12–15 were double tooth contact regions while load cases 5–11 were single tooth contact regions. Note that load case 11 corresponds to the load at the HPSTC. For each load case using the boundary element method, tractions were applied normal to the surface, to the appropriate ellipse with the magnitude equal to the tooth normal force divided by the ellipse area.

Crack Propagation Simulation. The procedure for the three-dimensional crack propagation simulation of the OH-58 spiral-bevel pinion was as follows. For each of the load cases of Figure 9, the mode I and mode II stress intensity factors were determined at 25 points along the crack front (note that for three-dimensions, there is a crack front, not just a crack tip as in two-dimensions). The extended crack directions at each of these 25 points were determined using the modified Erdogan and Sih crack extension theory as described in the two-dimensional analysis. That is, as the cracked spiral-bevel pinion tooth was engaged in the mesh, the crack extension started at each point along the crack front and grew in the direction of the greatest tangential stress at those points during mesh. The amount of crack extension at each point along the crack front was determined based on the Paris crack growth relationship (Ref. 18) where

$$a_i = a_{max}\left(\frac{K_{I,i}}{K_{I,max}}\right)^n \qquad (3)$$

Table I—Results of multiple load case crack simulation analysis.

Step	Crack area (mm²)	Crack front point(s)	Load case for largest $\sigma_{\theta\theta}$
0	3.12	1	8
		2–25	11
1	5.96	1	9
		2,4–7, 21, 23–25	10
		3,8–20, 22	11
2	10.35	1	10
		2–25	11
3	13.35	7, 9, 20	8
		5, 6, 10–15, 21, 26, 27	9
		1–4, 8, 16–19, 22–25	11

where a_i was the amount of extension of the i_{th} point along the crack front, $K_{I,i}$ was the mode I stress intensity factor of the i_{th} point along the crack front corresponding to the load case which gave the largest tangential stress for that front point, $K_{I,max}$ was the value of the largest $K_{I,i}$ along the crack front, a_{max} was the maximum defined crack extension along the crack front, and n was the Paris material exponent. From experience, the maximum extension size, a_{max}, was set to 1.27 mm (0.050 in.). The Paris exponent, n, was set to 2.954 based on material tests for AISI 9310 steel by Au and Ke (Ref. 4). A third-order polynomial was then used to smooth the extended crack front. The new crack geometry was then re-meshed. After re-meshing, the model was rerun and solved for stress intensity factors and crack propagation directions. The above procedure was repeated a number of times to simulate crack growth in the gear tooth.

Table I gives results from the first four steps during this process. Note that step 0 corresponds to the initial half-ellipse crack. For steps 0 and 2, the largest tangential stress occurred at the HPSTC (load case 11) for the majority of the points along the crack front. For steps 1 and 3, the largest tangential stress occurred at load cases 8, 9, 10, or 11.

As previously stated, the mode I and mode II stress intensity factors were determined at 25 points along the crack front. This was true for steps 0 through 2. For step 3, however, the mode I and mode II stress intensity factors were determined at 27 points along the crack front. This was due to the way the FRANC3D program extended the crack surface of the third step. For steps 0 through 2, the crack front was a member of one continuous geometry face (FRANC3D defines a geometry face as a 3- or 4-sided surface.) For

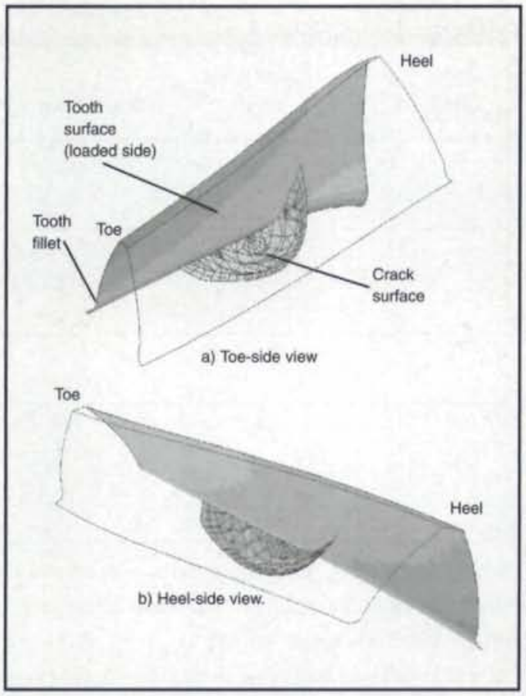


Figure 12—OH-58 spiral-bevel pinion tooth crack propagation simulation after seven steps.

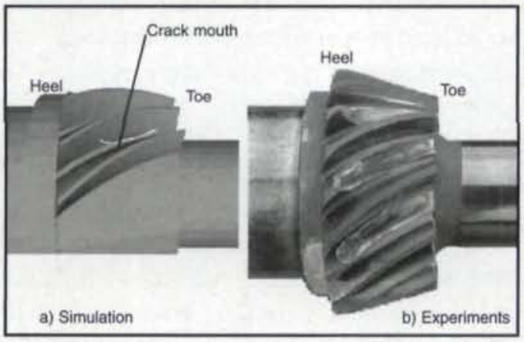


Figure 13—Comparison of OH-58 spiral-bevel pinion tooth crack propagation simulation to experiments.

step 3, the crack front was a member of three adjacent geometry faces, thus producing 27 points along the crack front.

Figure 10 shows the stress intensity factor distribution along the crack front for step 1 (crack area of 5.96 mm^2 (0.009 $in.^2$)). Similar to the spur gear analyses, K_I was larger as the load moved from the root to the tip due to the larger load lever arm. Other than absolute magnitude, the K_I distributions along the crack front looked similar for the various load cases. Figure 11 depicts the stress intensity factors plotted against load case (at a point along the front, biased toward the toe, normalized position along the crack front of 0.83) This figure shows the simulated distribution as the pinion engages in mesh with the gear. Note again that the ratio of K_{II} to K_I was not constant during engagement.

Figure 12 shows exploded views of the pinion crack simulation after seven steps. It should be noted that the loading was placed only at the HPSTC for the last three steps. This was due to modeling difficulties encountered using the multi-load analysis. It was felt that this simplification did not significantly affect the results due to the smoothing curve-fit used. In addition, the tangential stress near the crack tip was either largest, or near its largest value, when the load was placed at the HPSTC.

Comparison to Experiments. Figure 13 shows the results of the analysis compared to experimental tests. The experimental tests were performed in an actual helicopter transmission test facility. As was done with the gear fatigue tests described before, notches were fabricated in the fillet of the OH-58 pinion teeth to promote fatigue cracking. The pinion was run at full speed and with a variety of increasing loads until failure occurred. Shown in the figure are three teeth that fractured from the pinion during the tests (Fig. 13b). Although the notches were slightly different in size, the fractured teeth had basically the same shape.

A side view of the crack propagation simulation is shown in Figure 13a for comparison to the photograph of the tested pinion in Figure 13b. From the simulation, the crack immediately tapered up toward the tooth tip at the heel end. This trend matched that seen from the tests. At the toe end, the simulation showed the crack progressing in a relatively straight path. This also matched the trend from the tests. Toward the latter stages of the simulation, however, the crack tended to taper toward the tooth tip at the toe end. This did not match the tests. One problem encountered in the simulation during the later steps was that the crack at the heel end of the tooth became close to the actual contact ellipses. It was felt that the crack-contact interaction may have influenced the trajectory predictions to cause the discrepancy.

Spievak (Ref. 22) reported on another method to account for the non-uniform K_{II} to K_I ratio during pinion tooth engagement. This method considered contributions from all load cases in the crack angle prediction scheme and presented a method to accumulate the load effects. From these studies, reported crack propagation simulation of an OH-58 pinion also predicted the erroneous taper toward the tooth tip at the toe end. Again, the crack-contact interaction may have influenced the trajectory predictions to cause the discrepancy. Spievak also reported on a simulation using only the load at the HPSTC. The crack trajectories from that simulation were similar to the trajectories in the current study. It should be noted that the proposed method in the current study to account for moving tooth loads for the three-dimensional analysis was extremely cumbersome. It is therefore felt that the analysis using only the load at the HPSTC appeared accurate as long as the crack did not approach the contact region on the tooth.

Conclusions

A study to determine the effect of moving gear tooth load on crack propagation predictions was performed. Two-dimensional analysis of an involute spur gear using the finite element method was investigated. Also, three-dimensional analysis of a spiral-bevel pinion gear using the boundary element method was discussed. The following conclusions were derived:

1) A modified theory for predicting gear crack propagation paths based on the criteria of Erdogan and Sih was validated. This theory stated that as a cracked gear tooth was engaged in mesh, the crack extension started at the crack tip and grew in the direction of the greatest tangential stress during mesh.

2) For two-dimensional analysis, crack simulation based on calculated stress intensity factors and mixed mode crack angle prediction techniques can use a simple static analysis in which the tooth load is located at the highest point of single tooth contact.

3) For three-dimensional analysis, crack simulation can also use a simple static analysis in which the tooth load is located at the highest point of single tooth contact as long as the crack does not approach the contact region on the tooth.

References

1. Abersek, B. and J. Flasker, "Stress Intensity Factor for Cracked Gear Tooth," *Theoretical and Applied Fracture Mechanics,* 1994, Vol. 20, No. 2, pp. 99–104.
2. Abersek, B. and J. Flasker, "Experimental Analysis of Propagation of Fatigue Crack on Gears," *Experimental Mechanics,* 1998, Vol. 38, No. 3, pp. 226–230.
3. Arikin, M.A., A.I. Tarhan, and O.S. Yahoj, "Life Estimate of a Spur Gear with a Tooth Cracked at Fillet Region," Proceedings of the ASME Design Engineering Technical Conference, 1998, Atlanta, GA.
4. Au, J.J. and J.S. Ke, "Correlation Between Fatigue Crack Growth Rate and Fatigue Striation Spacing in AISI 9310 (AMS 6265) Steel," *Fractography and Material Science, ASTM STP 733,* 1981, pp. 202–221.
5. Blarasin, A., M. Guagliano and L. Vergani, "Fatigue Crack Growth Predictions in Specimens Similar to Spur Gear Teeth," *Fatigue & Fracture of Engineering Materials & Structures,* 1997, Vol. 20, No. 8, pp. 1171–1182.
6. Ciavarella, M. and G. Demelio, "Numerical Methods for the Optimization of Specific Sliding, Stress Concentration, and Fatigue Life of Gears," *International Journal of Fatigue,* 1999, Vol. 21, No. 5, pp. 465–474.
7. Curtin, T.J., R.A. Adey, J.M.W. Baynham and P. Marais, "Fatigue Crack Growth Simulation for Complex Three-Dimensional Geometry and Loading," Proceedings from the 2nd Joint NASA/FAA/DOD Conference on Aging Aircraft, 1998, Williamsburg, VA.
8. Erdogan, F. and G.C. Sih, "On the Crack Extension in Plates Under Plane Loading and Transverse Shear," *Journal of Basic Engineering,* 1963, Vol. 85, pp. 519–527.
9. Glodez, S., S. Pehan, and J. Flasker, "Experimental Results of the Fatigue Crack Growth in a Gear Tooth Root," *International Journal of Fatigue,* 1998, Vol. 20, No. 9, pp. 669–675.
10. Handschuh, R.F. and T.P. Kicher, "A Method for Thermal Analysis of Spiral Bevel Gears," *Journal of Mechanical Design,* 1996, Vol. 118, No. 4, pp. 580–585.
11. Handschuh, R.F. and F.L. Litvin, "A Method for Determining Spiral-Bevel Gear Tooth Geometry for Finite Element Analysis," NASA TP-3096, AVSCOM TR-91-C-020, 1991.
12. Inoue, K. and M. Kato, "Crack Growth Resistance Due to Shot Peening in Carburized Gears," Presented at the 30th AIAA/ASME/SAE/ASEE Joint Propulsion Conference, 1994, Indianapolis, IN.
13. Lewicki, D.G. and R. Ballarini, "Effect of Rim Thickness on Gear Crack Propagation Path," *Journal of Mechanical Design,* 1997, Vol. 119, No. 1, pp. 88–95.
14. Lewicki, D.G., A.D. Sane, R.J. Drago, and P.A. Wawrzynek, "Three-Dimensional Gear Crack Propagation Studies," Proceedings of the 4th World Congress on Gearing and Power Transmission, Paris, France, 1999, Vol. 3, pp. 2311–2324.
15. Lin, H.H., R.L. Huston and J.J. Coy, "On Dynamic Loads in Parallel Shaft Transmissions: Part I - Modeling and Analysis," *Journal of Mechanisms, Transmissions, and Automation in Design,* 1988, Vol. 110, No. 2, pp. 221–225.
16. Litvin, F.L. and Y. Zhang, "Local Synthesis and Tooth Contact Analysis of Face-Milled Spiral Bevel Gears," NASA Contractor Report 4342, AVSCOM Technical Report 90-C-028, 1991.
17. Nicoletto, G. "Approximate Stress Intensity Factors for Cracked Gear Teeth," *Engineering Fracture Mechanics,* 1993, Vol. 44, No. 2, pp. 231–242.
18. Paris, P.C. and F. Erdogan, "A Critical Analysis of Crack Propagation Laws," *Journal of Basic Engineering,* 1963, Vol. 85, pp. 528–534.
19. Pehan, S., T.K. Hellen, J. Flasker and S. Glodez, "Numerical Methods for Determining Stress Intensity Factors vs. Crack Depth in Gear Tooth Roots," *International Journal of Fatigue,* 1997, Vol. 19, No. 10, pp. 677–685.
20. Rice, J.R. "A Path Independent Integral and the Approximate Analysis of Strain Concentration by Notches and Cracks," *Journal of Applied Mechanics,* 1968, Vol. 35, pp. 379–386.
21. Sfakiotakis, V.G., D.E. Katsareas and N.K. Anifantis, "Boundary Element Analysis of Gear Teeth Fracture," *Engineering Analysis with Boundary Elements,* 1997, Vol. 20, No. 2, pp. 169–175.
22. Spievak, L.E. "Simulating Fatigue Crack Growth in Spiral Bevel Gears," Masters of Science Thesis, Cornell University, 1999.
23. Wawrzynek, P.A. "Discrete Modeling of Crack Propagation: Theoretical Aspects and Implementa-tion Issues in Two and Three Dimensions," Ph.D. Dissertation, Cornell University, 1991.
24. Williams, M.L. "On the Stress Distribution at the Base of a Stationary Crack," *Journal of Applied Mechanics,* 1957, Vol. 24, No. 1, pp. 109–114.

Tell Us What You Think . . .
If you found this article of interest and/or useful, please **circle 307.**
If you did not care for this article, **circle 308.**

ALTERNATIVE GEAR MANUFACTURING

Mitsubishi narrowed the focus.

Certificate Number FM 57549

MITSUBISHI HEAVY INDUSTRIES, LTD.
MACHINE TOOL DIVISION

ISO 9001 | QS 9000 TE

Net-Shape Forged Gears—The State of the Art

Trevor A. Dean and Zhongmin Hu

This paper was previously presented at the International Conference on Mechanical Transmissions, Chongqing, China, April 5–9, 2001.

Introduction

Traditionally, high-quality gears are cut to shape from forged blanks. Great accuracy can be obtained through shaving and grinding of tooth forms, enhancing the power capacity, life and quietness of geared power transmissions. In the 1950s, a process was developed for forging gears with teeth that requires little or no metal to be removed to achieve final geometry. The initial process development was undertaken in Germany for the manufacture of bevel gears for automobile differentials and was stimulated by the lack of available gear cutting equipment at that time. Later attention has turned to the forging of spur and helical gears, which are more difficult to form due to the radial disposition of their teeth compared with bevel gears. The main driver of these developments, in common with most component manufacturing, is cost. Forming gears rather than cutting them results in increased yield from raw material and also can increase productivity. Forging gears is therefore of greater advantage for large batch quantities, such as required by the automotive industry.

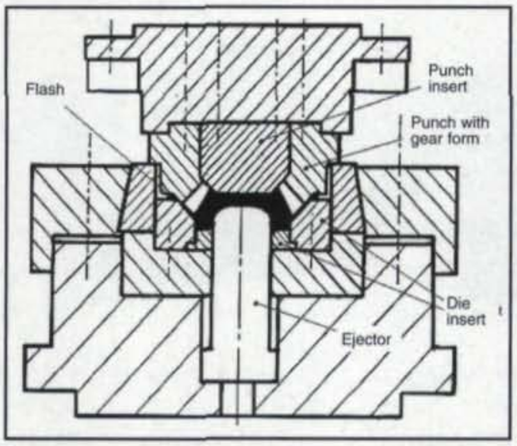


Figure 1—Flash bevel gear die.

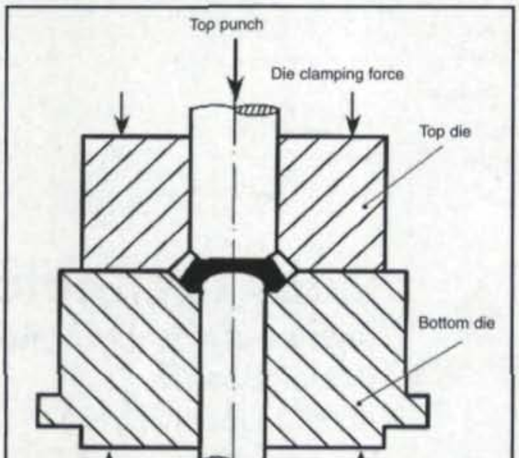


Figure 2—Flashless bevel gear die.

Cold forging (forging with workpieces at room temperature) results in parts with the highest accuracies. Differential bevel gears can be forged cold to finished geometry (net-shape). However, it is normally cheaper to forge them with a small amount of excess metal (near net-shape) and use a simple machining operation on their back faces to bring them to finished size. No machining of teeth is necessary. Depending on overall geometry, some spur and helical gears can be cold extruded with a net-shape tooth form. But, gears that have large diameter-to-width ratios—typical of those used in gearboxes and other power transmitting systems—must be forged in completely closed cavity tools using preheated workpieces. Thus, such gears are at best near net-shape, and considerable efforts are being undertaken to devise a second cold forming operation that will improve their tooth accuracy to net-shape standard. A cutting operation subsequent to forging results in an uneconomic processing route.

It has been shown that forged gears have higher strengths than cut ones, and this offers the opportunity for using them at higher power density ratings. This is attractive where weight is a penalty, such as in automobiles.

Bevel Gears

A commercial process for forging bevel gears using hot workpieces for automobile differential gears was available by the early 1960s (Ref. 1). The accuracy of the tooth form of the as-forged gears was sufficient for the automobiles of that period, but the design of the forging tool resulted in flash being formed (Fig. 1). Also, post-forge machining was required on the back face and the bore of the components. Continual developments of the process have resulted in tool sets with completely enclosed die cavities (Fig. 2). Using these cavities, it is possible to forge net-shape bevel gears. But, most often the bore is finish formed in subsequent operations, not in the forging tool. It may be said with little qualification that the technology for forging radial-and spiral-toothed bevel gears is virtually developed to its ultimate stage of commercial refinement.

Spur and Helical Gears

Extruded gear forms. Essentially two types of forming processes may be used to form these gears. If the aspect ratio (width/diameter) is large, they can be formed by extrusion. Typical extruded part types are the starter motor pinion and the helical shaft gear shown in Figure 3. Depending on the composition of the workpiece steel, these parts may be extruded at room temperature (cold formed). This results in high accuracy, and the tooth forms usually do not have to be finish machined. A gear of lower aspect ratio, which has been cold extruded, is shown in Figure 4. A considerable amount of metal has to be machined from the end faces, which have been distorted during extrusion. The loss of metal in machining these faces will be a consideration in judging the economic viability of extrusion.

Obviously, the distortion arising in extrusion of gears of even lower aspect ratios would render the process uneconomical because the amount of metal to be removed would be too high a proportion of the total. For this reason, such gears are forged in cavities in the manner of bevels. However, due to the fact that the teeth of spur and helical gears radiate normally to the axis of symmetry, they are more difficult to forge than bevel gears.

Forging machines & tooling. Forged gear technology is directed to high-volume production, and the forging machine most suitable for this is a mechanical press. However, virtually any forging machine with controllable stroke, load or energy, having accurate guidance, can be used if economic considerations allow.

Several forms of tooling designs are usable for gear forging, and the best choice depends on the geometry of the particular gear to be forged. A simple design that has been used to undertake early experiments at the University of Birmingham, in England, is shown in Figure 5 (Ref. 2). Essentially, it consists of a die insert with a female form of the gear teeth to be forged in its bore. A gear-shaped ejector, which can slide along the gear teeth, closes the bottom of the die cavity. The periphery of the punch, which is attached to the slide of the forging machine, is gear shaped so that the punch can slide in the cavity and close its upper end. The load cell shown is used for experimental purposes only and is not likely to be found in commercial situations. A gear is forged by placing a cylindrical billet on the ejector in the cavity and squeezing it sideways into the teeth of the insert under the force of the downward moving punch. When the punch has moved upwards, the forging is removed from the cavity by forcing the ejector upwards. This design can be used only for spur gears, as the necessary rotation of the punch to enable it to mate with a helical die insert is not practicable. An alternative tool design that is suitable for a wide range of spur and helical gear shapes is shown in Figure 6. The important features of this design that differ from the previous one are as follows:

The die insert is supported on light springs and can move vertically, guided by an external cylinder. The punch does not enter the die insert but contacts it on its top face. Thus, as the punch moves downwards, it closes the top end of the die cavity and pushes the insert downwards.

The diameter of the ejector is the same as the

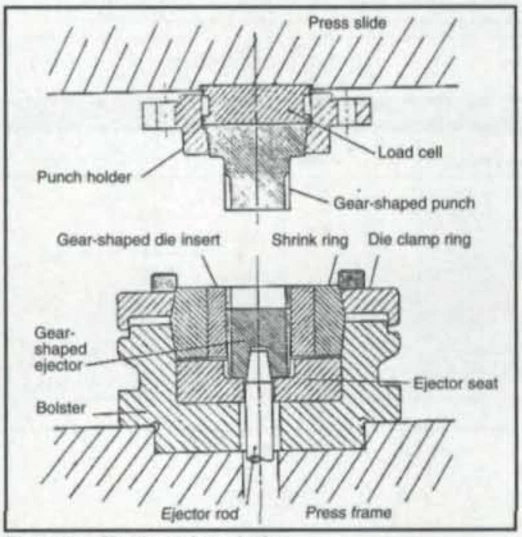


Figure 5—Simple tool set design.

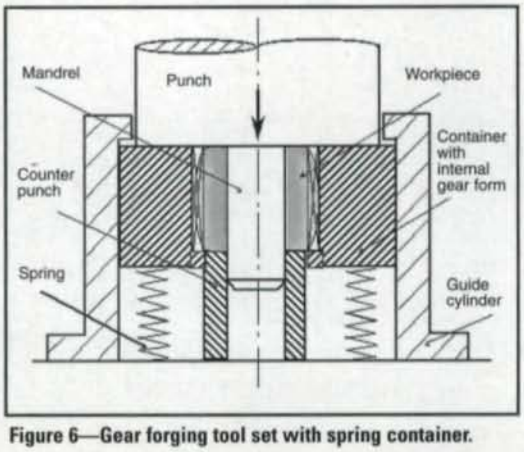


Figure 6—Gear forging tool set with spring container.

Figure 3—Extrusion-forged gear forms.

Figure 4—Through-extruded spur gear.

Dr. Trevor A. Dean

is emeritus professor of manufacturing engineering at the University of Birmingham, located in England. He also is leader of the metal forming group in the university's school of manufacturing and mechanical engineering. The group works on aspects of precision-forming engineering components. A mechanical engineer, Dean has worked on the forming of near-net and net-shape gears for more than 20 years and has published more than 140 papers on the subject.

Dr. Zhongmin Hu

is a research scientist in the University of Birmingham's School of Manufacturing and Mechanical Engineering, in the school's metal forming group. A mechanical engineer, he works on the net-shape forging of gears, aerofoil blades and other components. His main research work is in finite element analysis, friction and lubrication, and surface topography of forging products.

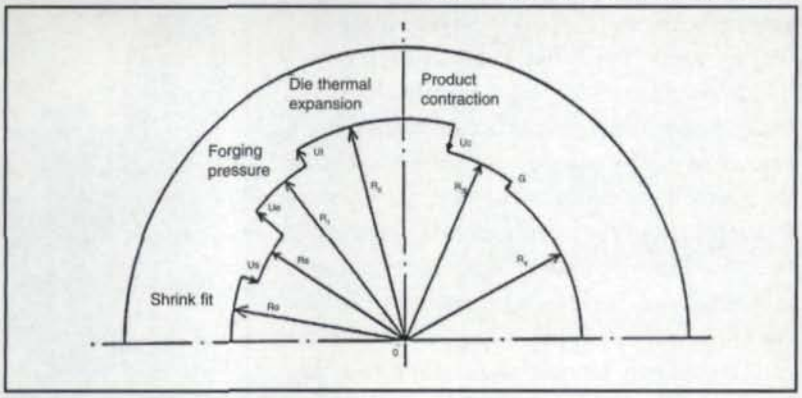


Figure 7—Correction factor in dies.

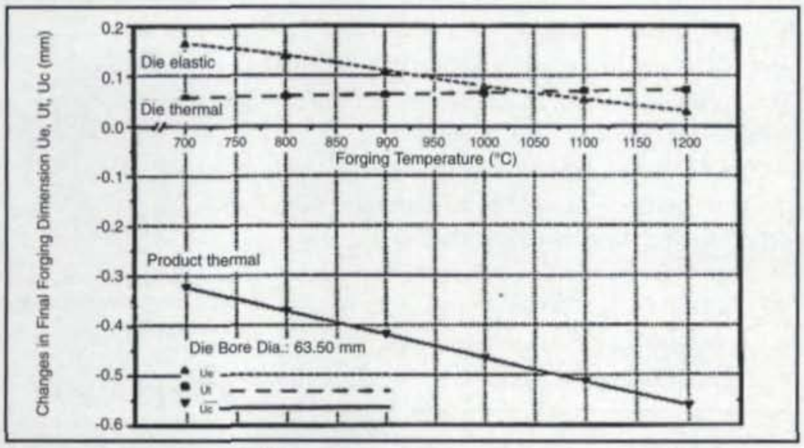


Figure 8—Temperature variations.

Table 1—Typical Accuracy of Forged Parts.				
Comparison Item	**Hot Forging**	**Warm Forging**		**Cold Forging**
Temperature Range	1,000 ~ 1,250°C	Over Ac_1	Below Ac_1	Room Temp.
Decarbonized Layer (mm)	0.3 ~ 0.4	0.10 ~ 0.25	0.1	0
Roughness (Ra)	> 100 µm	> 50 µm	> 20 µm	> 10 µm
Draft	< 7°	< 1°	< 1°	= 0°
Accuracy (µm)	±0.5 ~ ±1.0	±0.05 ~ ±0.2	±0.05 ~ ±0.15	±0.005 ~ ±0.1
Thickness (mm)	±0.50 ~ ±1.5	±0.20 ~ ±0.40	±0.10 ~ ±0.25	±0.10 ~ ±0.20
Eccentricity (mm)	0.5 ~ 1.5	0.10 ~ 0.70	0.10 ~ 0.40	0.05 ~ 0.25

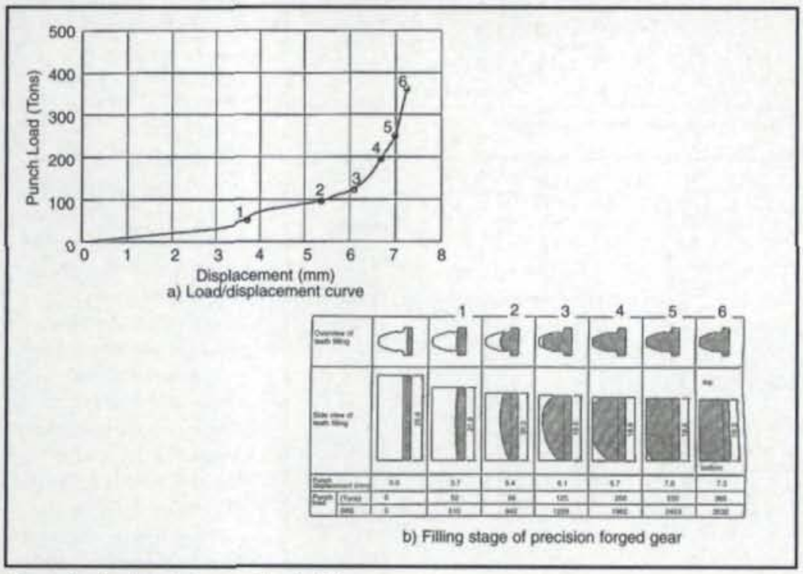


Figure 9—Load variation and tooth filling.

root diameter of the gear teeth, allowing it to be a simple cylindrical shape.

A mandrel is mounted on the punch, enabling hollow workpieces to be used. Thus, only a small amount of metal has to be removed to finish the bores.

There are eighteen different practical configurations of the four elements of the tool set—punch, insert, ejector and mandrel—which may be used on a press with one moving slide (Ref. 3). Each has advantages and disadvantages, and the best design depends on the overall geometry of the gear to be produced.

Factors affecting accuracy. The dimensional accuracy of a forging is affected by tooling and the process by basically three factors shown diagrammatically in Figure 7:

Most die cavities are made by EDM, and to compensate for spark gap and/or wire thickness, an allowance, G, on nominal dimensions is made.

Elastic expansion of the die, U_e, is caused by forging pressures.

Thermal expansion of the die, U_t, occurs as it is preheated to reduce thermal shock when forging is being undertaken at elevated temperature.

Post-forging thermal contraction, U_c, of a forging made at elevated temperature occurs after it is removed from the die.

The relative magnitudes of die thermal and die elastic effects can be seen by referring to Figure 8, which shows values obtained for a steel forging of nominal diameter 63.5 mm, forged in steel dies, and which are qualitatively applicable to all sizes of forging. It can be seen that as the temperature of the forging is increased, the elastic expansion of the die decreases. This is because the strength of the workpiece metal reduces as temperature is increased and stresses on the die wall are reduced. Also, it may be seen that the higher the forging temperature, the greater is the increase in forged dimension due to the thermal expansion of the die. This is due to the greater amount of heat transferred from hot workpiece to die at higher forging temperatures. The greatest absolute effect, and also the effect that varies most with change in forging temperature, is the thermal contraction of the forging. From this figure, it can be deduced that dimensional consistency can be achieved only if workpiece temperature and forging stresses are closely controlled. As forging stresses are related to billet size, temperature and tool lubrication, the whole production process—from incoming raw material to release of forging from the die—must be executed with utmost control if dimensional consistency is to be achieved. In an ideal situation, if thermal distortion could be predicted and controlled, preheating billets alone would not affect accuracy. However, practical limitations on temperature control and the sensitivity of dimensions to temperature leads to the situation that accuracy decreases as forging temperature increases. Thus, unheated (cold) forging is the technology that enables the greatest accuracies and the most consistent dimensions to be achieved in forging production. Accuracies typical of cold-, hot- and warm-forged components are given in Table 1.

To reduce elastic distortion of the die, loads and stresses must be kept as low as possible. Figure 9 shows the load associated with a given level of tooth filling during a forging operation. A noticeable increase in load arises when the workpiece reaches the roots of the teeth in the die cavity (Point 1). When the metal reaches the tips of the teeth in the die cavity, the load increase with ram displacement is very rapid and increases dramatically as the corners are filled. As the corners of gear teeth are usually chamfered, it is possible not to forge them fully. In the case of the example shown, a load reduction of about 50% could be achieved. Mathematical treatment of the distortions arising in tooling described above enables computer-based predictive programs to be developed so that corrections to cavity geometries may be introduced during manufacture so that the teeth of forged gears may be close to the specified shape. Figure 10 (Ref. 4) shows the variation in forged tooth profile that arises with changing workpiece temperature as predicted in theory and obtained experimentally, for a 13-tooth gear with 5.08 module and 20° pressure angle. Figure 10a shows that at room temperature, theoretical forged and die tooth profiles are closely matched above the base circle, and the forged base circle corresponds closely to that of the die. The tooth forged at 1,000°C (Fig. 10b) is smaller than that of the die, as is the base circle diameter. The differences between theoretical and experimental profiles between base and root circles is because the computer program was not arranged to allow for undercutting of the teeth that was machined into the die.

Forged gears. Net-shape processing routes for both spur and helical gears are under considerable investigation by a number of institutions. Obviously, whether or not a gear form is net-shape depends on the quality standard specified by the customer. But, as it appears that forged gears are likely to be commercially viable when made in large batch quantities, the standards being aimed for are those of automotive manufacturers. For use in gearboxes, ISO standard grade 5 is

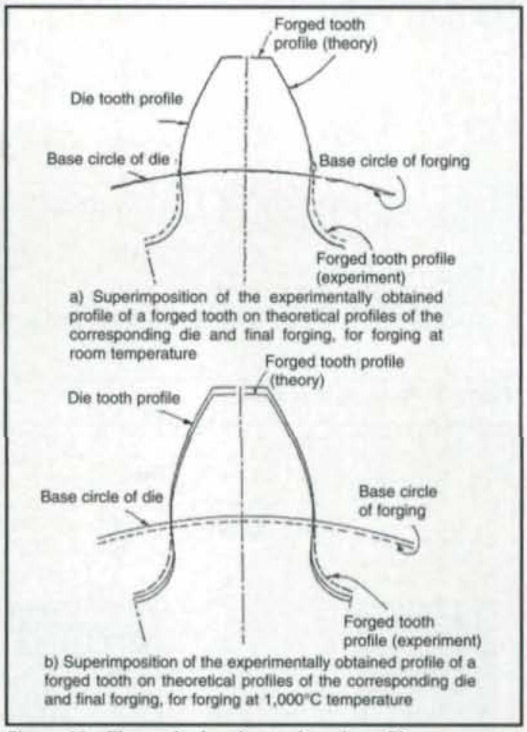


Figure 10—Theoretical and actual tooth profile.

Figure 11—Three gears being investigated.

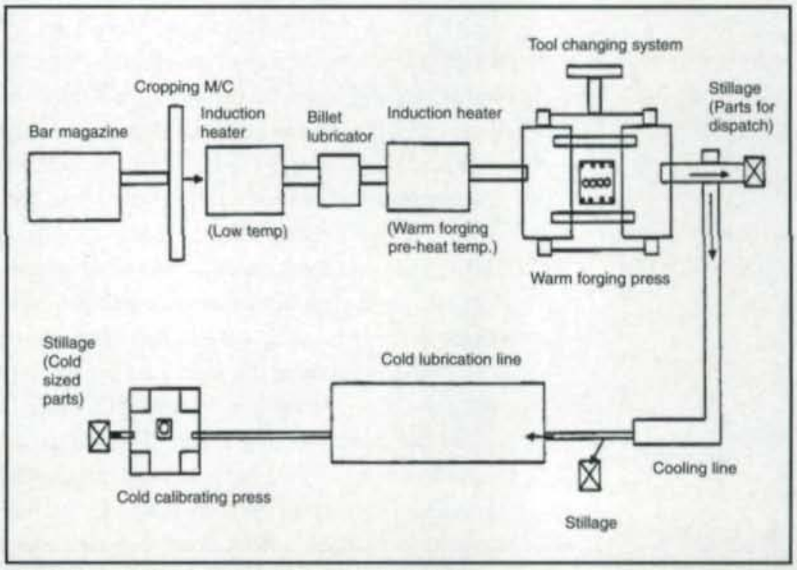


Figure 12—Equipment layout for warm/cold forging.

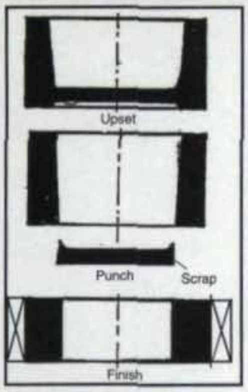


Figure 13—Forging stages of a gear.

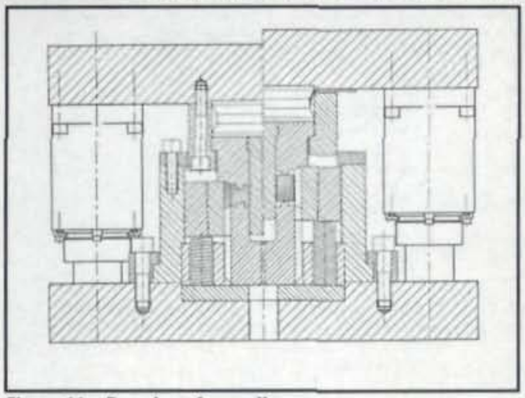


Figure 14—Drawing of gear die.

Figure 15—Photo of gear forging die.

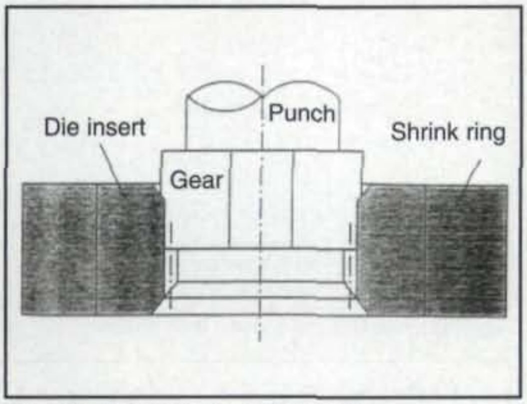


Figure 16—Schematic of gear ironing die.

Table 2—Effect of Ironing on Gears.

Feature	Finishing Die		Induction Hardened			
			As forged		Ironed	
	Left Hand	Right Hand	LH	RH	LH	RH
Pressure Angle (°)	0.12	0.10	0.73	0.58	0.19	0.16
Involute (μm)	15	13	57	46	27	22
Tooth Trace (μm)	8	5	105	94	93	66
Max. Cum Pitch (μm)	38	25	180	163	113	101
Adj. Pitch (μm)	12	9	73	28	21	16
Tooth Thickness (μm)	31	12	468	285	55	46
Runout (μm)	32		96		62	

required, except for the reverse idler gear, which may be ISO standard grade 10. Currently, investigations are underway at the University of Birmingham with the aim of developing commercial processing routes for the three gears shown in Figure 11. It is envisioned that a shop floor set up similar to that shown in Figure 12 will be utilized. Billets are sawed or sheared from rolled bars with circular cross-section, weighed, heated to about 100° C and coated in a water-based graphite lubricant. They are then heated in a second induction heater to a preheat temperature appropriate to the size, shape and alloy of the gear. That temperature will normally be in the region of 900° C, which is within the warm forging range. The billets are then forged on the first press in three operations: upsetting (squeezing them in a cylindrical die cavity), to produce a prescribed diameter and a central web; piercing of the web to produce a central hole; and finish forging in a die with appropriate peripheral tooth form. The operations are shown diagrammatically in Figure 13. The resultant forging is a near net-shape gear and is oversized by between 0.1 mm and 0.2 mm on all surfaces. The gears are cooled, cleaned and coated with a lubricant suitable for cold forging. They are then passed through an ironing die to bring them to specified dimensions. It is the cold-finishing operation that is still the subject of intensive research activity.

A drawing of the tool set used at the University of Birmingham is shown in Figure 14. Figure 15 is a photograph of the tool set mounted on a crank press. A schematic of the ironing die is given in Figure 16. Some preliminary results from a gear-ironing operation are given in Table 2. The improvements brought about by ironing are obvious, but the quality of the ironed gear is less than ISO grade 5. One of the reasons for this is that the quality of the ironing die was not high enough.

Concluding Remarks

The technology for net-shape forging of spur and helical gears is now well established.

The major remaining task is to develop a forming technique by which teeth of high accuracy may be produced with good productivity and at acceptable costs.

Acknowledgments

The experimental work reported in this paper was undertaken with support of the Engineering and Physical Science Research Council of the United Kingdom.

References

1. Lacy, K. "Hot Forged Gears to Fine Tolerances," *Metalworking Production*, 1967.
2. Sadeghi, M.H. and T.A. Dean, "Analysis of Dimensional Accuracy of Precision Forged Axisymmetric Components," *Proceedings of Institution of Mechanical Engineering*, Part C, 1991, 205.
3. Tuncer, C. and T.A. Dean, "Die Design Alternatives for Precision Forging in Completely-Close Die Cavities," *Proceedings of the 25th International Machine Tool Design and Research Conference*, 1985, 479–483.
4. Dean, T.A. "Precision Forging of Spur & Helical gear Forms," *Proceedings of Drives & Controls Conference*, British Gear Association Annual Congress, 1997, 66–72.
5. Abdel-Rahman, A.R.O. and T.A. Dean, "The Quality of Hot Forged Spur Gear Forms, Part 1: Mechanical & Metallurgical Properties," *International Journal of Machine Tools & Manufacture, Design, Research and Application*, 1981, Vol. 21, No. 2, 109–127.

Tell Us What You Think . . .
If you found this article of interest and/or useful, please **circle 311.**
If you did not care for this article, **circle 312.**

Failure Mechanisms in Plastic Gears

Yong Kang Chen, Nick Wright, Chris J. Hooke and Stephen N. Kukureka

Introduction

Plastics as gear materials represent an interesting development for gearing because they offer high strength-to-weight ratios, ease of manufacture and excellent tribological properties (Refs. 1–7). In particular, there is a sound prospect that plastic gears can be applied for power transmission of up to 10 kW (Ref. 6).

Typical plastics, such polyamide 66, more commonly known as nylon 66, have long been known as suitable materials for gearing. It was reported that polyamide gears experienced fatigue failure before significant wear was observed (Refs. 6–7), but crack initiation and propagation mechanisms were unclear. The failure mechanisms of polyamide 66 (PA66) gears when run in like pairs are still not clear (Refs. 6–15) and, as a result, PA66 users have to substantially underrate their designs for gears.

In a previous study (Ref. 15), a rolling-sliding test rig for other types of plastic gears has proved capable of measuring both friction and wear continuously during tests. This has enabled considerable detail to be obtained about the wear processes in polymer gears. The tests are thus a versatile way of studying wear mechanisms in order to contribute to accurate life prediction for gears.

This present work was thus initiated to study wear and friction mechanisms of PA66 and its composites. It was hoped that this investigation would enable the failure mechanisms of PA66 gears to be interpreted.

Experimental Apparatus and Procedure

The twin-disc wear testing machine used in our previous work (Ref. 15) was again employed in this investigation. With this machine, measurements of both the frictional force and wear between two discs in contact can be made continuously so that both the wear process and friction behavior during tests can be monitored. Experiments were carried out either at different slip ratios under a given normal load or at different loads with a fixed slip ratio.

Slip ratio is defined here as the ratio between sliding and rolling velocities. If the tangential velocities on both contact surfaces are v_1 and v_2 respectively, then the sliding velocity is (v_1-v_2) and the rolling velocity is $(v_1+v_2)/2$.

With this testing method, the typical loading and sliding conditions of plastic gears can be simulated (Ref. 15). Gears were tested using a back-to-back test configuration (Refs. 7–8, 13–14 and 16). The materials used were PA66 (R1000) and short-glass-fiber reinforced PA66 (RFL4036) (Ref. 17). The proportion of glass fiber added was 30% by weight. To ensure proper contact along the face width of the discs, all of the specimens were prepared by machining 20 µm from the molded surface and then polishing to a surface roughness of around 5 µm.

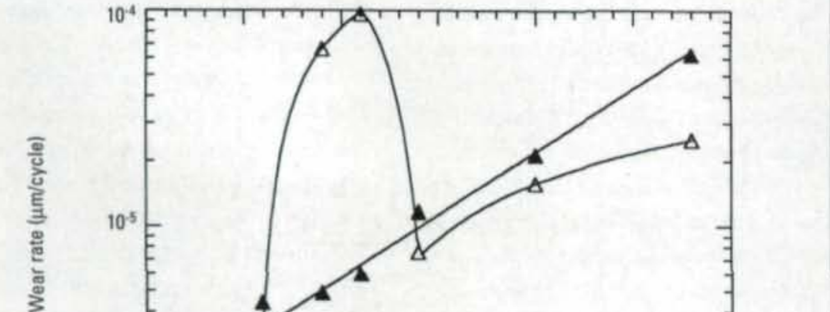


Figure 1—Wear rate versus slip ratio for PA66 at 200 N and 1,000 rpm and for 30% by weight short-glass-fiber reinforced PA66 at 300 N and 1,000 rpm.

Dr. Yong Kang Chen
is a senior lecturer in the aerospace, civil and mechanical engineering department at the Univeristy of Hertfordshire, located in Hatfield, England. Chen's research interests include tribological characterization of polymers and polymer composites and fatigue assessment of composites.

Dr. Nick Wright
is a materials development engineer with Cosworth Racing Ltd., located in Northampton, England. He is responsible for developing materials and processes for building racing engines used in Formula 1, CART and World Rally Car. He holds a doctorate in the tribology of polymer matrix composites for gearing applications.

Dr. Chris J. Hooke
is a research scientist in applied mechanics in the University of Birmingham's Engineering School, located in Edgbaston, England. He has published more than 150 papers on tribology. He has researched the wear mechanisms of polymers in unlubricated concentrated contacts, such as gears, and their relationship to surface temperature and stress, sliding speed and material composition.

Dr. Stephen N. Kukureka
is a senior lecturer in engineering at the University of Birmingham's Institute of Metallurgy and Materials Science, located in Birmingham, England. He holds a doctorate in materials science and has written more than 40 papers on polymers and composites.

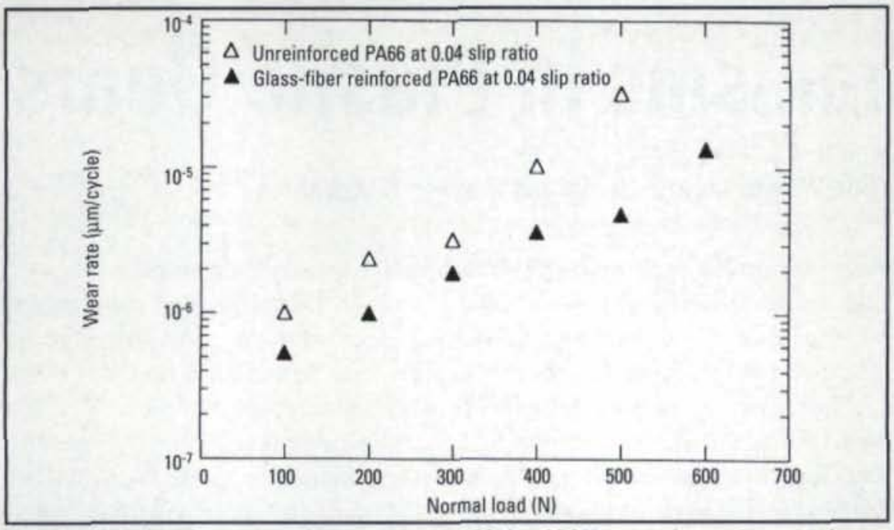


Figure 2—Variation of wear rate with normal load for PA66 and PA66 composite running at 1,000 rpm.

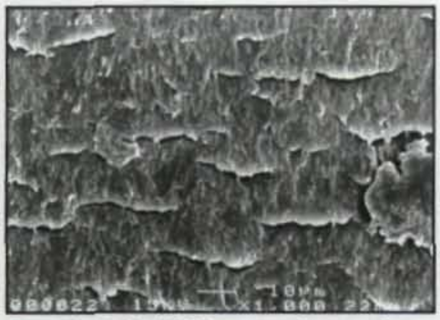


Figure 3—Typical PA66 worn surface running at 0.06 slip ratio, 32 MPa and 1,000 rpm; direction of friction force from bottom to top.

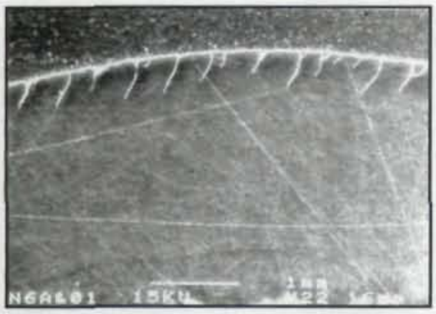


Figure 4—Transverse cross section through the disc parallel to direction of friction force running at 0.14 slip ratio, 32 MPa and 1,000 rpm; direction of friction force from left to right.

Figure 5—Geometry and dimension of typical debris to be produced on the PA66 worn surface running at a transition slip ratio of 0.11, 32 MPa and 1,000 rpm; direction of friction force from top to bottom.

Both the discs were 30 mm in diameter and 10 mm in face width.

Before testing, the samples were cleaned with methanol. They were then run at the test conditions for an extended period to remove the machining asperities and any subsurface layer affected by the manufacturing process. After running them, they were dried at 70°C for 15 hours to remove any absorbed water that might affect the measurement of wear and then weighed. Finally, they were left under atmospheric conditions for about two weeks to allow the water content to return to equilibrium conditions.

After this preliminary treatment, the specimens were remounted in the test rig in an identical position to that under which they had been run-in. Tests were run for running speed of 1,000 rpm and at a range of loads and slip ratios. The slip ratios used were between 0 and 0.28. Tests were performed under dry, unlubricated conditions at ambient temperature (22 ± 1)°C until failure or for up to 10^7 contact cycles.

At the end of the tests, the discs were again dried at the same temperature and for the same time period as they were before they were tested. Then, the discs were weighed to measure the weight loss of each disc. With this drying procedure, the measurement of wear by weighing is accurate to about $\pm 10^{-5}$g. Finally, the worn surfaces were observed in detail by using a JEOL JSM6300 scanning electron microscope.

Experimental Results

Wear rate. Wear rate is defined here as the average depth of material removed from each disc per rolling cycle (Refs. 15 and 18–19) and was calculated by measuring the weight loss of the specimens.

Figure 1 shows how the wear rate of unreinforced PA66 varies with slip ratio for a fixed normal force of 200 N and a constant running speed of 1,000 rpm. It can be seen that slip ratio has a significant effect on wear rate. The wear rate rises slightly with an increase of slip ratio when the slip ratio is less than 0.09, at which point discolored material appears on the contact surfaces during the tests. The wear rate starts to increase sharply from 2.0×10^{-6} μm/cycle to 7.0×10^{-5} μm/cycle as the slip ratio increases from 0.06 to 0.09 and the wear rate reaches its highest value of 10^{-4} μm/cycle at a slip ratio of 0.11.

The unique characteristic property of this material is that a further increase in slip ratio from 0.11 results in a dramatic decrease in wear rate. When the slip ratio increases to 0.14, the wear rate decreases rapidly from 10^{-4} μm/cycle to 8.0×10^{-6} μm/cycle and discolored material returns to the contact surfaces. The difference between the two wear rates is more than tenfold while that between the two slip ratios is approximately 27%. After this considerable decrease, the wear rate increases slowly to 1.0×10^{-5} μm/cycle as slip ratio increases to 0.21. It is suggested that the slip ratio of 0.11 is a critical one corresponding with maximum wear rate.

Figure 2 shows the effect of normal load on the wear rate at a given slip ratio and running speed (at 1,000 rpm with a slip ratio of 0.04). It can be seen that the wear rate varies in the range from 2.0×10^{-6} μm/cycle to 4.0×10^{-6} μm/cycle. When the load is between 300 N and 500 N, the wear rate starts to increase significantly from 2.0×10^{-6} μm/cycle to 3.0×10^{-5} μm/cycle.

The effect of short-glass fibers. Reinforcement with short-glass fibers has significant effects on wear and friction. As shown in Figure 1, both wear and friction are dominated by the ability or inability of a thin layer of self-lubricating film to be formed continuously and to be retained on the surfaces in contact (Ref. 18). Similar to the case for PA66 at low slip ratios, when the film was found on the contact surface, wear debris could hardly be observed and friction coefficient was less than 0.1. Unlike PA66, once the film was disrupted, the friction coefficient of short-glass-fiber reinforced PA66 varied between 0.25 and 0.3 while that of unreinforced PA66 was in the range of 0.42 to 0.72.

Figure 2 shows the effect of normal load on the wear rate of 30% short-glass-fiber reinforced PA66 composites at a fixed slip ratio of 0.04 and a constant running speed of 1,000 rpm. Here, the wear rate increases uniformly on a logarithmic scale from 10^{-7} μm/cycle to 1.35×10^{-5} μm/cycle as the normal load increases from 100 N to 500 N.

Figure 1 shows the wear rate of 30% short-glass-fiber reinforced PA66 composites as a function of slip ratio for a fixed normal force of 300 N and constant running speed of 1,000 rpm. The self-lubricating film on the contact surfaces exists during all of Figure 1's wear-rate measurements. It can be seen that the wear rate increases nearly logarithmically from 10^{-6} μm/cycle to 10^{-5} μm/cycle as the slip ratio increases from 0.04 to 0.21. The wear rate is of the same order of magnitude as that of unreinforced PA66, apart from the unique peak in the wear rate.

Discussion

The wear mechanisms of discs can be used to explain wear behavior of gear teeth. Figure 3 shows the early stage of a general surface damage on a disc after running for 5.8×10^6 cycles when the slip ratio was 0.06 and less than its critical value of 0.11. It can be seen that the length of these cracks varies from a couple of microns to tens of microns. These cracks eventually propagated across the whole width of the disc and were perpendicular to the direction of the friction force that moves from the bottom to the top.

Figure 4 shows a section through the disc parallel to the direction of the friction force when the slip ratio was 0.14, which is greater than its critical value of 0.11. It can be seen that the typical width between two fully developed cracks is 200–400 μm and that the distance between two sub-cracks is less than 100 μm. This can be compared with this material's typical Hertzian contact width of 400–980 μm (Ref. 19). It can be seen that, as shown in Figure 4, the main cracks initiated in the radial direction, perpendicular to the friction force. Then, they propagated in a direction at an angle to the friction force instead of perpendicular to the friction force. The depth to which the main cracks propagated varied from 150 μm to 450 μm.

Between these main cracks, there

were some sub-cracks. The main cracks propagated up to 450 μm, and the small cracks propagated to about a quarter of the depth of the main cracks. These sub-cracks eventually joined the main cracks. As shown in Figure 4, a sub-crack initiated from the contact surface and propagated in an arc towards the adjacent main crack. Since the sub-crack is wider at the contact surface than beneath it, it is suggested that the small crack was initiated from the contact surface rather than from beneath it (Ref. 19). Some cracks propagated to join their neighbor, and as a result, the material between the two cracks was fractured, severe spalling occurred and debris was formed. It was noted that the crack propagated and fractured gradually rather than suddenly. This compares well with the wear observed on a PA66 gear tooth surface, as shown in Figure 8.

Figure 5 shows a general view when the slip ratio reached its critical value, that is 0.11. The worn surface shows unique characteristics but is not discolored. A large quantity of roll-like debris is attached to the surface and tends to accumulate on the surface. It can be seen that roll-like debris consists of the middle part of the roll and two tails on both sides. The length of roll-like debris varies from tens of microns to a couple of hundred microns. The diameter of the middle part is about 1.0–10.0 μm. A tail has a very small diameter (about 0.1 μm) and is very long (about 100 μm). Most tails were broken and separated from their body during the friction process.

It was observed that before roll-like debris on the worn surface was formed, the surface suffered a very deep shear deformation and surface material moved in the direction of friction. This deformed material was gradually rolled in the direction of friction. As a result, the body of a piece of roll-like debris was formed, as shown in Figure 5. It was noted that there was, overall, a great deal of debris collected during the test and that it appeared very thin and long. Since the slip ratio along gear tooth profile in contact varies and covers the slip ratio range of 0–0.28, it is suggested that failure mechanisms of PA66 gears are severe wear due to a critical slip ratio and tooth fracture due to macrotransverse cracks on the contact surface.

The reinforcement of PA66 with short-glass fiber also has an effect on transverse cracks, which occur when PA66 is in non-conformal rolling-sliding contact. Although the wear mechanisms are very complicated, as shown in Figures 6–7, no transverse cracks on the worn surfaces were observed under a variety of test conditions.

Because of the film on the surface, as shown in Figure 7, the friction coefficient can be below 0.1, and as a result, the shear stress at the interface between the films should be low. This low shear stress may play an important role in the self-lubricating property of the composite. Also, because of the low friction coefficient, the maximum shear stress is not at the contact surface but on the subsurface, and there is little chance for a transverse crack to initiate on the contact surface (Ref. 18).

Figure 6 shows a surface in the severe wear stage obtained from a disc run at 1,000 rpm, with a 300-N load and a 0.28 slip ratio. It can be seen that the fibers on the surface were highly aligned and approximately parallel to the friction force. Figure 6 also indicates that both long and short broken fibers can move on the surface, causing alignment from an initially random distribution. It is observed that the aligned broken fibers do not remain on the surface indefinitely but are expelled out as debris. It appears that, in the rapid wear stage, wear is due to the removal of unworn, but fragmented fibers rather than to their gradual abrasion. This also compares well with the wear observed at the composite gear tooth surface, as shown in Figure 9.

These test results clearly explain why PA66 gear teeth fracture near the pitch line area with little debris, where the sliding ratio between gear teeth in mesh is very small (Refs. 6–8 and 13). From the above study of non-conformal, unlubricated rolling-sliding contact, one of the dominating factors in surface failure at low and high slip ratios is transverse crack propagation on the surfaces in contact. Also, there was not much debris before cracks fully developed and spalled. Since the slip ratio on the gear teeth in contact varies and is around 0–0.21 near the pitch line (Refs. 6–8 and 13), the behavior of crack propagation on a gear tooth surface should be similar to that on disc surfaces. In other words, these cracks on a gear tooth propagate not only across the gear face width but also into the subsurface of the gear tooth.

After a certain number of cycles, the cracks will propagate down to a depth of 0.5 mm near the dedendum of a gear tooth—in the region near initial contact on a driving gear tooth—where a high slip ratio is expected and near the pitch line area on the gear tooth where a low slip ratio occurs. For a gear of module 2 mm, a reduction of 0.5 mm of the tooth thickness near the dedendum is considerably significant. Stress concentrations at the tip of cracks are severe. Therefore, the bending stress on the dedendum of the tooth could be much higher than the tooth was designed for. Near the pitch line area on a gear tooth, corresponding to lower slip ratios, bending stresses are high since this is the position where only a pair of gear teeth are in contact. Crack propagation at lower slip ratios will also cause severe stress concentrations at the tip of cracks if the tooth is subjected to a reasonably higher bending moment. Under high bending stresses, it is suggested that the severe stress concentration results in the tooth fracture near the pitch line area on the gear tooth. Therefore, it is suggested that tooth fracture in PA66 gears is due to initiation and propagation of transverse cracks rather than creep as shown in Figure 8.

Conclusions

The failure mechanisms of polymeric (PA66 and PA66 composite) gears have been investigated by testing plastic against plastic in counter-conformal, unlubricated, rolling-sliding contact over a wide range of slip ratios, loads and running speeds. Comparisons between tests on discs at varying slip ratios and the results of gear tests under comparable conditions have been very favorable.

The wear and friction behavior of PA66 was dominated mainly by three major features: a critical slip ratio under a fixed load and running speed, macrotransverse cracks and a layer of film on the contact surface. These results corresponded closely to the failure phenomena of PA66 gears. It is suggested that the transverse cracks caused the plastic gear teeth to fracture, even near the pitch line. The macrotransverse cracks in the gear teeth on the contact surfaces are a serious disadvantage of PA66 gears.

To remedy this, the effect of reinforcements of short-glass fiber on the wear and friction behavior has been studied. Both the wear and friction properties of

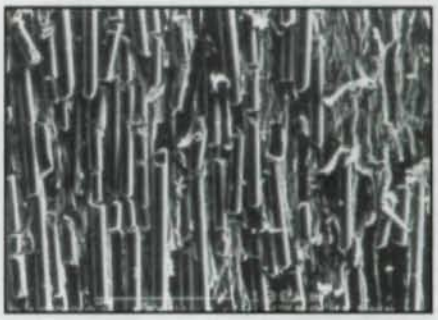


Figure 6—Aligned broken fibers on the composite surface after disruption of the surface layer (0.28 slip ratio, 63 MPa and 1,000 rpm); direction of friction force from bottom to top.

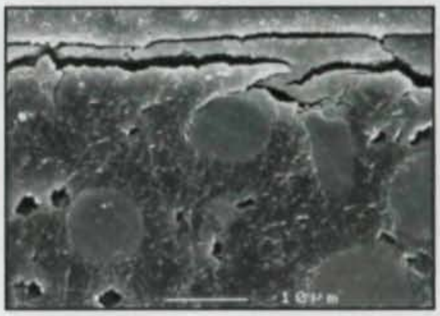


Figure 7—Transverse cross section of typical surface film on the contact surface of short-glass-fiber reinforced PA66 running at 1,000 rpm, 0.04 slip ratio and 63 MPa; direction of friction force from left to right.

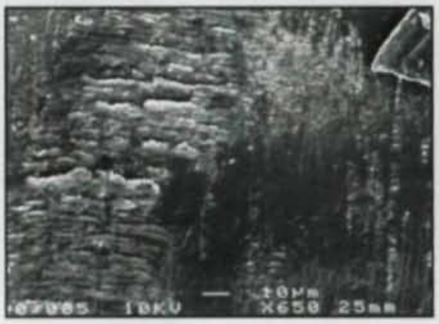


Figure 8—Typical surface topography of a PA66 gear running at 1,500 rpm and maximum contact stress of 39 MPa after 2.25 x 10^6 cycles.

Figure 9—Typical surface feature on the contact surface of short-glass-fiber reinforced PA66 driving gear running at 1,500 rpm and 66 MPa after 2.25 x 10^6 cycles. (Pitch line not in photograph.)

unreinforced PA66 were improved considerably by the reinforcement of 30% by weight short-glass fiber. This reinforcement prevented both the initiation and propagation of transverse cracks on the contact surfaces that occurred in the unreinforced material. Also, it decreased both the wear rate and the friction coefficient substantially. A thin film on the contact surfaces was observed and played a dominant role in the "self-lubricating" behavior of the composite and in suppressing the transverse cracks. These results offer the prospect of enhanced applicability of polyamide 66 in gears.

Acknowledgments

We would like to thank Brian Duke of Davall Gear Co. and Davall Moulded Gears Ltd. for the supply of several components of the test rig and for many useful discussions.

References

1. British Standard BS 6168, *Specification for non-metallic spur gears*, London, 1987.
2. *Polypenco Gear Design*, Polypenco Corp., U.S.A., 1985.
3. Drago, R.J. *Fundamentals of Gear Design*, Butterworths, Boston, MA, 1988.
4. Yamaguchi, Y. *Tribology of Plastics Materials*, Elsevier, New York, 1990.
5. Guicheaar, P.J., Bernard S. Levy and N.M. Parikh, *Gear Manufacture and Performance*, American Society for Metals, Ohio, U.S.A., 1974.
6. Crippa, G. and P. Davoli, "Fatigue resistance of polyamide 6 gears," *Kunststoffe*, 1991, 81, 147.
7. Mao, K., C.J. Hooke and D. Walton, "The wear behaviour of polymer composite gears," *Synthetic Lubrication*, 1995, 12 (4), 337.
8. Breeds, A.R., S.N. Kukureka, K. Mao, D. Walton and C.J. Hooke, *Wear*, 1993, 166, 85.
9. Clerico, M. "A study of the friction and wear of nylon against metals," *Wear*, 1969, 13, 183.
10. Clerico, M. "Sliding wear mechanisms of polymers," *Proceedings of the International Conference on the Fundamentals of Tribology*, N.P. Suh and N. Saka, Eds., MIT Press, MA, U.S.A., 1978, 769.
11. Clerico, M. and V. Patierno, *Wear*, 1979, 53, 279.
12. Clerico, M. *Wear*, 1980, 64, 259.
13. Hooke, C.J., K. Mao, D. Walton, A. Breeds and S.N. Kukureka, "Measurement and prediction of the surface temperature in polymeric gears and its relationship to gear wear," *Transactions of the ASME, Journal of Tribology*, 1993, Vol. 115, p. 119.
14. Hooke, C. J., K. Mao, D. Walton, A. Breeds and S.N. Kukureka, "Temperature prediction in polymer gears," *3rd World Congress on Gearing and Power Transmission*, Paris, 1992.
15. Kukureka, S.N., Y.K. Chen, C.J. Hooke and P. Liao, *Wear*, 1995, 185, 1.
16. Wright, N. *Polymer Matrix Composites for Power Transmission Elements*, M.Phil. dissertation, University of Birmingham, Birmingham, United Kingdom, 1996.
17. LNP Engineering Plastics Inc., in *A Guide to LNP's Internal Lubricated Thermoplastics*, U.S.A., 1994.
18. Chen, Y.K., S.N. Kukureka, and C. J. Hooke, *Journal of Materials Science*, 1996, 31, 5643.
19. Chen, Y.K. *Tribology of polymers in unlubricated rolling and sliding contact*, Ph.D. dissertation, University of Birmingham, Birmingham, United Kingdom, 1996.

This paper was previously presented at the 4th World Congress on Gearing and Power Transmission in Paris, March 1999.

Tell Us What You Think . . .

If you found this article of interest and/or useful, please **circle 314.**

If you did not care for this article, **circle 315.**

If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618 or send e-mail messages to *people@geartechnology.com.*

Technology Tidbits

William R. Stott

New Technique for Forging Crowned Helical Gears

Createch Co. Ltd., a forging die manufacturer from Shizuoka, Japan, has developed a net-shape cold-forging process for forming helical gears and splines with crowned teeth.

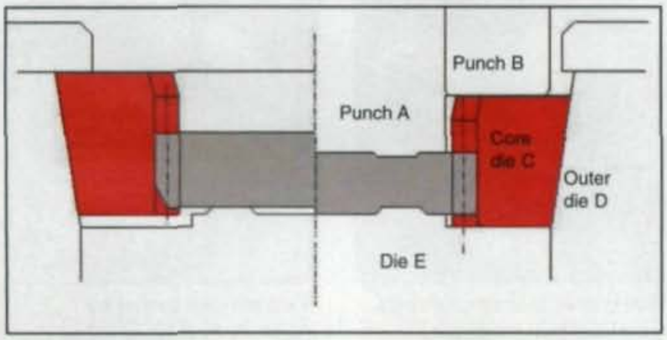
The process is being tested by Japanese automotive manufacturers to replace conventional forging and machining processes. Potential advantages include improvements in gear noise, pitting resistance and fatigue strength, says Hitoshi Ishida, Createch's president and founder. Also, there is the possibility of eliminating processes such as tooth shaving or tooth grinding.

The process is capable of forming gear teeth to JIS 2–3 quality for spacing error, tooth profile error, lead error with crowning, and runout, according to Ishida. That is similar to the quality level obtained by gears that have been cut and shaved or burnished, Ishida says.

Createch has created what Ishida refers to as a "dialog with molds," meaning that the interaction between the dies and the forged materials is so well understood and controlled, it's as if the mold is speaking with the material, Ishida says. This "dialog" allows a homogeneous distribution of inner stresses in the finished part, he adds.

According to Ishida, the secret to creating crowned teeth is radial force. With a conventional forging die, the material is forced into the mold with axial force. Essentially, the work material is pressed through a helical die from one end. Createch uses a conventional forging process to create rough helical gears without crowning. However, because crowned teeth are thicker in the middle than at the ends, they cannot be formed by such a process, nor could they be easily removed from the die.

To finish the gears, the Createch die sets include a tapered die case and a core die. The rough gear is placed inside the core die.



When the core die is forced into the tapered die case, the core die deforms elastically, shrinking inward as it's pushed through the die case. That inward force forms the crowning on the gear teeth. When the core die is removed from the die case, it returns to its original form, expanding away from the part and enabling the crowned teeth to be removed easily from the mold.

Another unique aspect of the Createch die sets is that they are equipped with a mechanism to rotate the core die during the crown forging process. The friction caused between the core die and the workpiece helps create a homogeneous stress distribution and symmetrical crowning of the tooth flanks, Ishida says. It also allows for continuous metallic fiber flows at the gear root, providing high fatigue strength and pitting resistance.

The Createch die sets are made for conventional forging presses, including hydraulic, knuckle, link motion and mechanical presses commonly used for forging operations. They are most applicable for high-precision gears, such as gears for planetary reducers and in some automotive applications, says Ishida.

Circle 316

Japanese Companies Develop High-Strength Plastic Gears

Shin-Kobe Electric Machinery Co. Ltd. of Tokyo has developed plastic secondary balance shaft gears that greatly reduce the noise in four-cylinder automobile engines, according to the company.

Plastic gears have been used in automobile engines for many years—Shin-Kobe developed its first phenol resin balance shaft gear in 1955—but this is the first instance they have been used for the secondary gear, which rotates twice as fast as the primary balance shaft gear. Until now, steel gears were required for that application, according to Akikazu Tazawa, manager of overseas marketing and planning for Shin-Kobe.

The gears were developed in cooperation with Toyota Motor Corp. and are being used in the four-cylinder engines of the Estima, Harrier, Kluger, Ipusam and Camry models of Toyota vehicles, according to Tazawa. The company expects total sales of the gears to reach 3 billion yen (roughly $24 million) in 2002.

The new gears are made of KOBE LITE® KM-9000, an aramid-reinforced fiber and polyaminoamide resin, resulting in increased durability and heat resistance when compared to conventional plastic gears, such as phenolic resin with glass fibers. The gears also significantly reduce noise when compared with metal gears previously used. At 2,000 rpm, the noise can be reduced by at least 15 decibels, Tazawa says.

Shin-Kobe uses a special spun yarn to improve the adhesion of the aramid fiber, which normally doesn't have a good affinity with resin, Tazawa says. The fibers are arranged so that they radiate from the center of the gear, giving the gear added strength. Aramid also is less damaging to the mating gear teeth than glass or carbon fibers would be.

In addition, the plastic gears allowed designers to eliminate backlash shims, which were required with metal gears previously used. This saves assembly labor and complexity, Tazawa says.

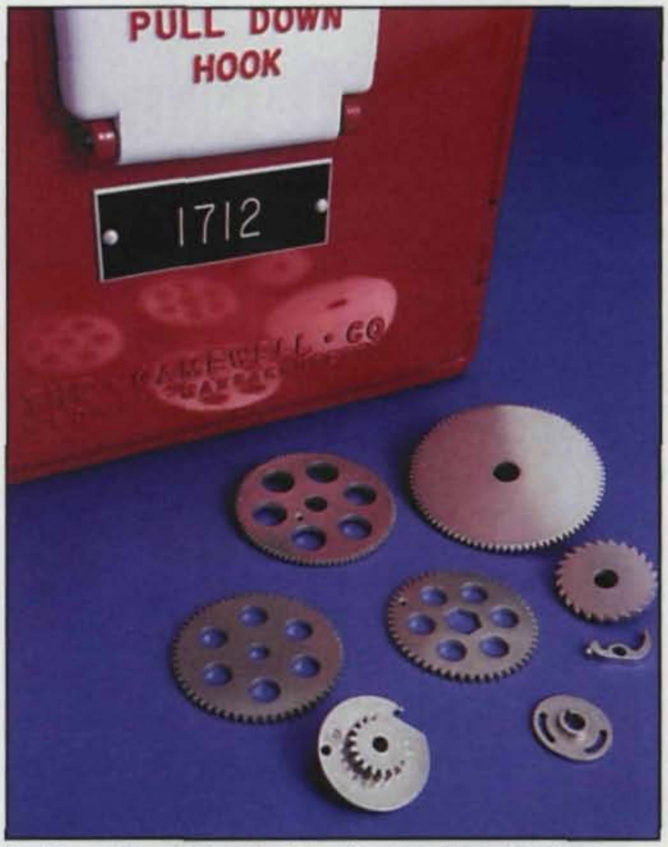
Circle 317

Gear Parts Win in P/M Competition

Each year, the Metal Powder Industries Federation (MPIF) holds a design competition to highlight the best uses of powder metal technology. The 2001 awards included many gears and gear-related items among the winners.

The Ferrous Grand Prize went to Stackpole Ltd. of Stratford, Ontario, Canada, for an intricate planetary gear carrier assembly made for GM Powertrain.

Ferrous Grand Prize: Stackpole Ltd.



Nonferrous Grand Prize: Precision Powdered Metal Parts.

The carrier assembly replaces a cast iron carrier in GM's PT 4L60E heavy-duty transmission, which is used in 800 series vehicles, including the Sierra, Silverado, Suburban, Escalade and Corvette.

The assembly consists of a low-alloy steel spider and copper-steel clutch hub, which are sinter-brazed together into one unit. The multifunctional assembly combines the planetary carrier and clutch hub functions, a first for Stackpole. The new assembly reduces machining, reduces the part's weight and increases durability without secondary heat treatment of the splines.

The Nonferrous Grand Prize also went to companies working with gears. Eight nickel-silver parts, including five gears, won that prize. The parts were made by Precision Powdered Metal Parts of Pomona, CA, for a three-fold fire alarm box manufactured by Gamewell Worldwide of Ashland, MA.

Previously, these parts were manufactured by stamping and machining, but the manufacturer was able to reduce costs by 50% after switching to powder metal manufacturing.

Bevel gear indexing ratchets by Allied Sinterings Inc.

Transmission carrier by Keystone Powdered Metal Co.

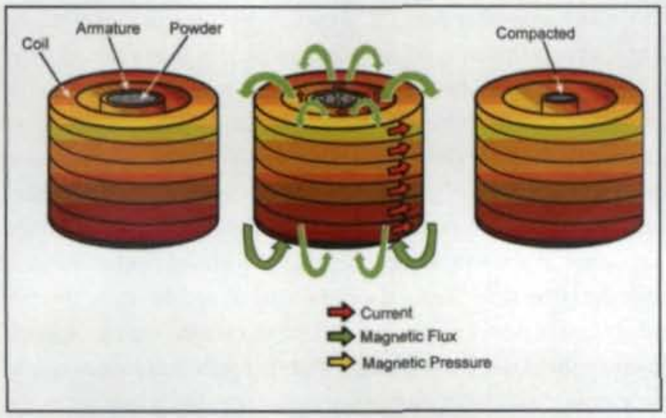
Other gear-related parts that won awards were an automotive manual transmission synchronizer ring made by Sinterstahl GmbH of Fussen, Germany; a counter-shaft transmission hub with external involute spline teeth made by Caterpillar Inc.'s Advanced Compacting Technology Group of Rockwood, TN; a transmission carrier made by Keystone Powdered Metal Co. of St. Marys, PA; a bevel gear/indexing ratchet for a surgical stapler, made by Allied Sinterings Inc. of Danbury, CT; and a ratchet gear assembly for a cordless drill, made by Jenn Feng Industrial of Taiwan, R.O.C.

The annual design competition has awards for different categories of parts. The categories are injection-molded products, ferrous (iron, steel or iron-based), nonferrous (copper, copper-based, bronze, brass, nickel-silver or aluminum), advanced particulate materials, stainless steel and other high alloys (less than 50% iron). The organization also presents an overseas award, which is open to all materials.

Parts are reviewed by a panel of judges appointed by the MPIF's industry development board, and the criteria for awards are design configuration, engineering properties and promotional value.

The competition is open to MPIF member companies, and detailed rules can be obtained from MPIF by sending an e-mail message to *info@mpif.org*.

Circle 318



Magnetic Compaction of Powdered Metal

IAP Research Inc. of Dayton, OH, has a process that uses magnetic forces to form highly dense parts from powdered metal materials. Those parts may soon include automotive transmission ring gears.

IAP has been working with a major U.S. automotive manufacturer to adapt the process, called dynamic magnetic compaction (DMC), to produce the ring gears. Development testing has shown that the gears can be manufactured to AGMA 9 quality, with material density of 7.6 g/cm^3 in the gear tooth area, and material properties approaching those of machined parts, says Ed Knoth, senior research engineer for IAP. Although the parts meet design parameters, they need extensive laboratory testing to determine long-term fatigue life and other properties, Knoth says.

A container and die for making internal ring gears by magnetic compaction of powdered metal.

According to Knoth, DMC also is capable of manufacturing other parts with ring-type geometries and radially symmetric features. The process is targeted for parts requiring high density and material properties traditionally associated with wrought metals. Knoth is confident that the process is a viable alternative for many manufacturers. "I think it's ripe for production," he says, adding that the company is quoting jobs to use the process.

With the DMC process, the powdered metal is loaded into an electrically conductive container. For ring gears, the center of the container is a core tool with the negative pattern of the gear teeth to be formed. Outside the core tool is a ring made of electrically conductive material. End caps close off the faces of the container. When the container is placed within an induction coil, the magnetic field forces the outer ring to accelerate at the rate of several hundred meters per second toward the core tool, compacting the powder in between. The ring also decelerates quickly, causing a high pressure spike in the powder, which gives the material its high density, Knoth says.

The process is capable of pressing steel powder to 96% of full density, according to IAP literature.

Circle 319

Tell Us What You Think . . .

If you found this article of interest and/or useful, please **circle 309.**

If you did not care for this article, **circle 310.**

If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618 or send e-mail messages to *people@geartechnology.com.*

UQM Technologies To Exit Gear Contract Manufacturing Business

UQM Technologies Inc. announced it will exit the gear contract manufacturing business because of weak demand for gears in the agricultural and over-the-road truck markets, which the company has served.

In its press release, UQM said it plans to seek buyers for its gear business or—if no buyers are found—sell its gear manufacturing assets and stop operations. UQM's gear operations had annual revenue of $1.6 million for the fiscal year that ended March 31, 2001, and a net loss of $1.1 million.

The revenue represents 6 percent of annual consolidated revenue. The loss represents 45 percent of consolidated operating losses before items.

"Our gearing operation has not met our expectations due to continuing weakness in the industry generally, and specifically in the markets we serve," said William G. Rankin, UQM's president and CEO. "The outlook for recovery is not promising as more and more gear purchasers source their gearing requirements overseas.

"Continued weak demand and the resulting poor financial performance of this operation has been a continuing drag on consolidated financial performance for the last couple of years, and poor industry fundamentals and the resulting grim prospects for near-term recovery have led us to the decision to shed this non-core operation."

Rankin added UQM is receiving increasing demand for its motor, generator and power electronic products.

Also, Rankin said UQM will continue to produce and supply gears to customers and will consider gear orders from new customers, until it sells the business or stops operations. He explained that if the business were sold, remaining orders would be completed by the new owner.

Lapointe Buys Oswald Forst

Lapointe International Corp. bought Oswald Forst GmbH & Co. KG of Solingen, Germany; Forst Broach Ltd. of Leicester, England; and Cardinal Broach Plc. of Biggleswade and Ratby, England.

Forst provides helical broaching machines. Also, Forst designs and manufactures other broaching machines, tools and broach sharpeners for auto industry producers and suppliers. Cardinal Broach Plc. designs and manufactures broaching machines and tooling for the automotive and turbine industries.

In its press release, Lapointe said Forst's helical broaching machines would give Lapointe an important new product line to offer to its customers.

Lapointe International Corp. of Worcester, MA, designs and manufactures machines and tooling for industries that produce precision forms, such as helical gears, turbine disks and rifled gun barrels. The industries include automotive, jet engine and land-turbine manufacturing. O

Gleason Offers New Machine For Bevel Gear Roll Testing

Gleason Corp. is offering a new machine for roll testing spiral and hypoid bevel gear sets.

In its press release, Gleason said its new 600 HTT Turbo Tester performs faster CNC-controlled roll testing of those gear sets.

The tester can accommodate workpieces with 600-mm diameters. Besides basic pattern checking, the 600 HTT tests for single-flank transmission error and structure-borne noise for soft and hard parts.

According to Gleason, the roll tester has a small-footprint, ergonomic, easy-to-use design.

For more information, visit Gleason Corp.'s website at *www.gleason.com.*

Circle 320

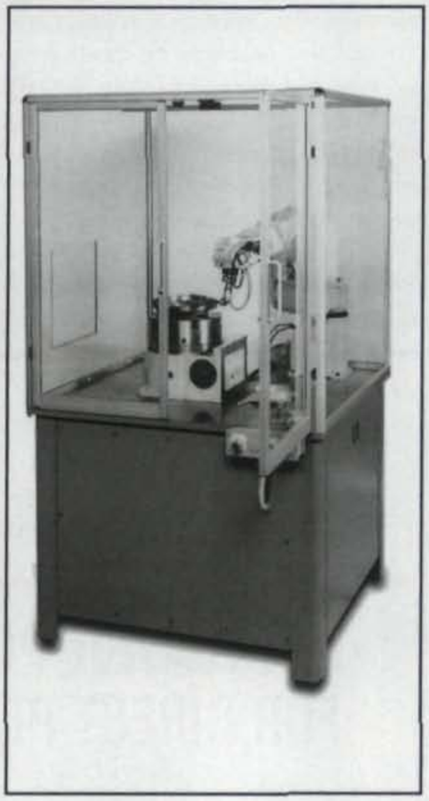
Textron Has New Helicoidal Gear Geometry in Gearmotors, Reducers

Textron Power Transmission has a new helicoidal gear geometry for its Series B Conex™ helicoidal right angle gearmotors and reducers.

According to Textron, the new geometry, called Conex™ inside, provides high capacity and efficiency. The gearmotors and reducers have power capabilities up to 20 hp, with a maximum torque of 5,000 lb.-in. They also have gear ratios up to 60:1 in one stage.

For more information, visit the company's website at *www.textronpt.com.*

Circle 321



Samputensili Automates Bevel Gear Deburring, Chamfering

Samputensili has a new machine, the S 450/750 DBC Gear Debur Cell, that automates deburring and chamfering of bevel gears.

The cell uses a FANUC 6-axis servo-driven robot and a programmable controller. The cell can be used to do top and bottom tooth and root gear deburring and chamfering geometry.

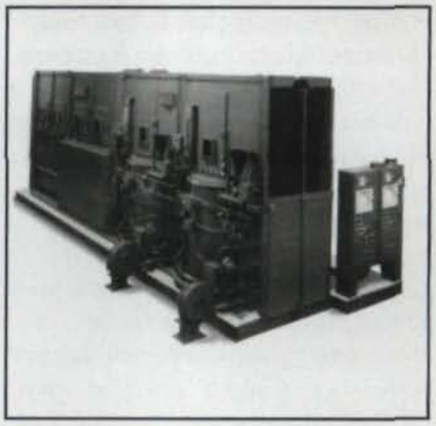
In its press release, SU America Inc. said the S 450/750 reduces the time and cost normally associated with most deburring operations. It also said the cell provides more precise, consistent part-to-part accuracy.

According to SU America, the cell eliminates the need to manually debur and chamfer gears.

The cell is available in two sizes: 2" through-hole with 18" diameter and 4" through-hole with 30" diameter.

In North America, for more information, contact SU America through Meritage Inc. of Rockford, IL, by telephone at (815) 484-9250 or by fax at (815) 484-9254.

Circle 322



Kolene Adapts Ferritic Nitrocarburizing for Better Corrosion Resistance

Kolene Corp. has developed a variation of its Nu-Tride ferritic nitrocarburizing process to improve parts' corrosion resistance and give them a black, low RMS finish. The variation, called the QPQ process, is used to finish automotive gears and other parts, such as hydraulic and pneumatic parts.

According to Kolene's press release, QPQ processing follows Nu-Tride processing and further improves parts' corrosion resistance.

For more information, contact Kolene Corp. of Detroit, MI, by telephone at (313) 273-9220 or visit its website at *www.kolene.com.*

Circle 323

***Gear Technology* welcomes your new product announcements for gears, gear drives and products for designing, manufacturing and testing gears.**

Send your new product releases to:
Gear Technology
1401 Lunt Avenue
Elk Grove Village, IL 60007
Fax: 847-437-6618
E-mail: *people@geartechnology.com*

Tell Us What You Think . . .

If you found this column of interest and/or useful, please **circle 324.**

If you did not care for this column, **circle 325.**

If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618 or send e-mail messages to *people@geartechnology.com.*

Jan. 29—Heat Treating & Hardening of Gears. Embassy Suites Nashville-Airport, Nashville, TN. Program teaches basic concepts of heat treatment and hardening of gears, and alternative methods. Intended for people involved in heat treating, quality control, material development or industrial operations. $445 for SME members, $495 for non-members. For more information, contact Lynn Albertson, senior program developer, by telephone at (800) 733-4763 or by e-mail message at *albelyn@sme.org*.

Jan. 30–31—Basic Gear Manufacturing & Design. Embassy Suites Nashville-Airport, Nashville, TN. Program provides basic concepts of gear design and manufacturing processes. Program can be used by gear designers and inspectors, project engineers and process control engineers and technicians. $745 for SME members, $895 for non-members. For more information, contact Lynn Albertson, senior program developer, at (800) 733- 4763 or by e-mail at *albelyn@sme.org*.

Feb. 4–7—Falk School Gear & Coupling Workshop. Falk's Renew Business Unit, New Berlin, WI. Provides "hands-on" training to familiarize working maintenance mechanics with field-practical, factory-approved installation, alignment, maintenance and failure analysis procedures. Specific attention to preventative maintenance and early-warning diagnostic procedures for all equipment. $1,495. Course presented throughout 2002. Other upcoming dates are Feb. 18–21, March 4–7 and March 11–14. For more information, contact Falk Corp. by e-mail at *bstefl@falkcorp.com*.

Feb. 18–21—Gleason Corp. Gear Fundamentals Course. Gleason Cutting Tools Corp., Loves Park, IL. Four-day program for people who are new to gear-making and want basic understanding of gear geometry, nomenclature, manufacturing and inspection. $895. Course presented throughout 2002. Other dates of course include April 15–18 and June 24–27. For more information, call (815) 877-8900 or visit *www.gleason.com*.

Feb. 20–23—Fluidtrans Compomac. Portello Halls, Milan Fairgrounds, Milan, Italy. International exposition featuring fluid power and mechanical motion control component manufacturers, including hydraulics, pneumatics, electronics, gears, actuators and motors. Admission is free. Contact the show organizer, Fiere & Mostre S.r.l. of Milan, Italy, by telephone at (39) 02-409493-1, by fax at (39) 02-409493-68 or by e-mail at *promo.ftc@fieremostre.it*.

Tell Us What You Think . . .
If you found this column of interest and/or useful, please **circle 330.**

If you did not care for this column, **circle 331.**

If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618 or send e-mail messages to *people@geartechnology.com*.

Returning the Personal Touch to Business

Dear Editor:

Your recent editorial, "Out of the Cave," which appeared in the March/April 2001 issue, was one of the most timely and meaningful messages that has appeared in a trade journal in a long time. Today, as you stated, the personal touch in business has been replaced by fax machine, e-mail, impersonal direct mail, cell phones, Internet and/or telemarketing. The face-to-face sales contact of my generation has been replaced by these so-called more efficient and less time-consuming ways of conducting business.

Salespeople today are taught that time is money, to qualify an account and to call ahead to make an appointment. It is much easier to say no to a salesperson over the phone than it is when he is in your lobby. I recall passing a small tool shop and questioning the salesman I was with why we weren't stopping to call on the shop. He said he had called them, and they were not buying any new equipment. We stopped and called on the owner and learned that they were considering expanding, as they had just received a large contract from an automotive supplier. I doubt that we would have received this information over the phone.

People still buy from people. Personal service and interacting with management, manufacturing and shop people on a face-to-face basis is still the most effective way of doing business. I wish I had every order I lost because of personal reasons…to someone who was closer to the customer than I was.

Very few salespeople today make cold calls. In my day, every salesman was required to make cold calls. There is a place for telemarketing, and it can produce results, but there is still a place today for cold calls. They can be very productive.

I also have a problem with cell phones and how they are used. I was with a salesman who called a manufacturing engineer to ask for an appointment. He then called him to tell him we were leaving the office. He called again to tell him we were tied up in traffic, and again to tell him we were in the area. He had interrupted the manufacturing engineer four times—twice when the engineer was in the shop—and when we arrived, we received less than a warm welcome. We haven't learned how to use the toys we have in the correct manner.

Trade shows provide a neutral ground for personal interacting with potential buyers. However, unfortunately, salespeople on the floor are taught to qualify the individual. They will ask if you specify equipment, how soon you plan to purchase a piece of equipment, etc., and if the answers are not positive, you are not likely to receive much time from the salesperson. I recall receiving a large order from a manufacturing engineer. When he gave me the order, he said, "You probably don't remember me, but I was that student with whom you took time to show your machines at IMTS many years ago." He had graduated from engineering school and was now in a position to buy capital equipment.

You are so right that today we focus on projects, meetings and quotas, rather than the people who help us meet sales goals and objectives. We are driven by the bottom line, and we forget that service and personal contact are still the best and oldest methods of dealing with our customers. I had a standing rule that all proposals and quotations were to be hand delivered. This allowed us to look into the eyes of the manufacturing engineer or purchasing agent and obtain an immediate reaction to the quote. It is impossible to get any feedback from a proposal that has been mailed, faxed or e-mailed without any personal contact.

The unfortunate thing is that the cost of having a salesperson on the road continues to increase. In addition, the congested airways and highways reduce the number of calls and contacts that a salesperson can make in a day. The salesman is taught that time is money, so he only calls on the paying customers…the 20% that account for 80% of his business. One solution taken by some machine tool companies is to set up regional technical sales and service centers. This is a way of getting their customers to come to them for service and technical help. It allows them to maintain personal relationships.

I recall telling one of my salesmen to call on a particular client every Monday morning at the same time. He objected, saying that they never bought anything from him. However, he followed my instructions. On one occasion, there was a serious problem on the factory floor the morning he made his weekly call. He was able to help them out, and in turn, they became one of his best customers. Lucky, maybe, but it was the personal calls that made the difference, not the computer, fax machine, phone calls, direct mail or telemarketing.

I am sorry for the lateness in responding to your editorial. As you have probably guessed, I am a retired machine tool person, formerly vice president of sales for Waterbury Farrel Division of Textron (Jones & Lamson turning machines, Cleveland hobbing machines, Waterbury Farrel presses, Thompson grinders, J&L metrology and J&L grinders).

Regards,
Albert B. Albrecht

Tell Us What You Think . . .
If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618.

CLASSIFIEDS

WEBSITES

GLEASON CUTTING TOOLS CORPORATION

Gleason Cutting Tools Corporation is the leading gear cutting tool manufacturer in North America. We offer a wide variety of form-relieving milling cutters, hobs, shaper cutters, shaving cutters and CBN-plated form grinding wheels as well as our engineering, coating, heat treating, metallurgical, tool sharpening and reconditioning services.

www.gleason.com

CIRCLE 160

GLEASON CORPORATION

Gleason is the world's most comprehensive resource for gear manufacturing technology. We offer a comprehensive gear cutting machine program for every gear hobbing, milling and grinding application for cylindrical and bevel gears. To learn more about our products and services or to find key contact information, see our Web site.

www.gleason.com

CIRCLE 162

RAYCAR GEAR & MACHINE CO.

Raycar is a privately owned company that manufactures many types of gears. We have hobbing, shaping, shaving, tooth chamfering and gear grinding capabilities. All drilling, milling, tapping and I.D. grinding is done in-house, enabling us to make gears more efficiently and economically and to maintain our scheduled delivery dates.

www.raycargear.com

CIRCLE 180

WEBSITES

APPLIED PROCESS

A leader in the heat treating industry for more than three decades, Applied Process and its affiliates use austempering technologies to produce a stronger, tougher structure than those produced with conventional heat treatments. Austempering is an isothermal heat treatment applied to ferrous materials to solve a wide range of engineering challenges.

For more information, visit our website at *www.appliedprocess.com.*

CIRCLE 131

SOFTWARE

CALCULATION PROGRAM FOR GEARBOXES

KISSsoft is a comprehensive suite of calculation programs designed to enhance speed and accuracy when designing common machine elements (Gears, Shafts, ...).

KISSsoft automates the design through optimization routines. The calculation methods are according to internationally recognized standards (AGMA, ISO, DIN, et al.).

To request more information on this product, please contact:

M. Schroder, 156 Carlton Drive, Mooresville, NC 28117; (704) 528-1509; or info@kiss-soft.com.

CIRCLE 136

HELP WANTED

SALES/ENGINEER

PARKER INDUSTRIES (Long Island, New York) seeks sales-oriented Engineer with some knowledge of gear cutting tools and related products. Good technical and communication skills a must.

Reply in confidence with relevant information to:

George A. Parker
Parker Industries Incorporated
P.O. Box 465
Bohemia, New York 11716

PARKER

Rates—Line Classified: 1" minimum, $325. Additional lines $45 per line (8 lines per inch). Display Classified: 3" minimum: **1X**—$735, **3X**—$685 per insertion, **6X**—$645 per insertion. Additional per inch: **1X**—$255, **3X**—$235 per insertion, **6X**—$225 per insertion. **Color Classified:** Add $150 per insertion for color. *Gear Technology* will set type to advertiser's layout or design a classified ad at no extra charge. **Payment:** Full payment must accompany classified ads. Send check drawn in U.S. funds on a U.S. bank or Visa/MasterCard/American Express number and expiration date to *Gear Technology*, P.O. Box 1426, Elk Grove Village, IL 60009. **Agency Commission**: No agency commission on classified ads. **Materials Deadline:** Ads must be received by the 15th of the month, two months prior to publication. **Acceptance:** Publisher reserves the right to accept or reject advertisements at his discretion.

HOB SHARPENING SERVICE

Star Cutter Co.

Since 1927

STAR CUTTER COMPANY
ISO-9001 CERTIFIED

• THIN FILM COATINGS

West Branch Industries
Subsidiary of Star Cutter Co.
2083 W. M-55, West Branch, MI 48661
1-888-Resharp • 1-888-737-4277
Phone: (989) 345-2865 • FAX: (989) 345-5660

CIRCLE 142

Induction Hardening

Specialists in Tooth by Tooth Contour Hardening of Internal
Spur, helical and bevel gears

Our gear hardening equipment includes 5 NATCO submerged process machines and 5 AJAX CNC-controlled gear scanning machines. Tooth by tooth gear hardening from .5DP-10DP, up to 15 tons, 200" diameter.

American Metal Treating Company
Cleveland, Ohio
Phone: (216) 431-4492 • Fax: (216) 431-1508
Web: *www.americanmetaltreating.com*
E-mail: *mark@americanmetaltreating.com*

Breakdown Service Available

CIRCLE 181

LOOK!

We are the source for quality gears. From design to manufacture . . . to delivery, count on Okamoto Machinery for your demanding precision gear and related power transmission products.

Look to us for the finest Precision Gears . . .

- Spiral Bevel
- Worm
- Spur & Helical
- Splines

Custom design service
High production capability
Unmatched quality, service & support

Call today!
Okamoto Machinery
847-520-7700 ext. 625
www.okamoto-kouki.co.jp

CIRCLE 153

INTERSTATE TOOL CORP. CLEVELAND, OHIO

CUSTOM ENGINEERED & MANUFACTURED CUTTING TOOLS
ESTABLISHED 1960

FORM RELIEVED & PROFILE GROUND MILLING CUTTERS
GEAR SHAPER & SHAVING CUTTERS
ALL CLASSES OF HOBS
HSS. SOLID CARBIDE & CARBIDE TIPPED

WHETHER YOU NEED NEW TOOLS, MODIFICATIONS, RESHARPENING, REPAIRS OR M&M INSPECTIONS,
CONTACT US FOR A QUOTE TODAY!

www.interstatetoolcor.com
Tel: 216-671-1077 • Fax: 216-671-5431

CIRCLE 151

HOB SHARPENING
(763) 425-5247

★ *THIN FILM COATINGS* ★

HSS & Carbide up to 5" Dia.
Straight Gash,
Sharpened & Inspected
Per AGMA STANDARDS
Quick Turnaround

KORO SHARPENING SERVICE
9530 - 85TH AVENUE NO.
MAPLE GROVE, MN 55369

CIRCLE 163

GEAR TOOTH GRINDING SERVICES

Spur - Helical - Double Helical

Capacity up to 60.5" O.D., 1 D.P., 29" Stroke. All ground gears certified up to AGMA Class 14+ on Zeiss-Höfler 1602 CMM. Inventory of grinders includes Höfler 800, Höfler 1253 Supra, Höfler 1500, two Höfler Nova CNC 1000s (Fully CNC with on-board CMM checker), and Höfler Rapid 1000 Form Grinder.

Kreiter Geartech
2530 Garrow St., Houston, TX 77003
Phone: 713-237-9793 Fax: 713-237-1209
Contact: Mr. Willie Whittington
Visit our Website at
www.kreiter-geartech.com

CIRCLE 141

GEAR TOOTH GRINDING

Spur • Helical
• Herringbone (with groove)
Capacity up to 63" O.D., 1 D.P., 16" face

AGMA Certification Inspection
Delivery to Meet Your Requirements

Midwest Gear Corp.
2182 E. Aurora Rd.
Twinsburg, OH 44087
Phone 330-425-4419
Fax 330-425-8600

Direct your inquiries to
Ron Humphrey, General Manager
ronh@mwgear.com

CIRCLE 154

- BROACH SHARPENING
- HOB SHARPENING
- SHAVING CUTTER GRINDING
- THIN FILM COATING
- CUSTOM HEAT TREAT SERVICE

PICK UP AND DELIVERY IN MANY AREAS

Gleason Cutting Tools
CORPORATION
1351 Windsor Road
Loves Park, IL 61111 USA
Phone: 815-877-8900
Fax: 815-877-0264
Website: www.gleason.com
E-Mail: gctc@gleason.com

CIRCLE 147

HOB SHARPENING

- State-of-the-art CNC sharpening and inspection machines
- Wet grinding with free-cutting CBN or diamond wheels for "Burn free" sharpening of carbide or high speed steel
- Optional recoating and stripping
- Rush service available

Koepfer America, LLC
J K S
635 Schneider Drive
South Elgin, IL 60177
Phone 847-931-4121
Fax: 847-931-4192
sales@koepferamerica.com

CIRCLE 137

gerrGeargeer rdwerdWorddSearchsearsh

Find the gear-related words listed on the right in the puzzle below. The words may be horizontal, vertical or diagonal and may be written forward or backward.

s	e	f	g	e	n	i	l	p	s	p	i	r	a	l	e
a	t	h	g	e	a	r	s	p	e	x	e	v	e	e	x
h	e	g	o	y	u	b	m	i	k	m	y	d	c	f	t
y	a	f	a	b	e	d	e	n	d	u	t	v	e	l	w
t	r	i	n	f	b	i	n	v	o	n	i	n	o	l	u
i	r	i	p	p	l	i	n	g	e	y	c	s	k	e	w
c	g	n	i	n	o	h	n	e	n	l	i	d	e	c	e
i	f	g	h	h	u	j	n	g	n	a	r	p	i	c	a
r	a	c	k	e	u	g	i	t	a	f	t	i	p	n	r
t	b	y	n	l	j	h	h	g	f	g	n	g	g	i	c
n	b	i	d	i	d	x	d	d	n	r	e	r	u	f	u
e	w	h	j	c	h	g	e	i	u	e	c	r	s	h	n
c	r	t	t	a	n	g	d	p	r	r	n	e	e	e	d
c	o	o	p	l	h	n	s	t	a	n	o	e	t	r	e
e	b	o	n	i	i	j	d	h	e	u	c	c	u	r	r
t	b	t	h	r	n	k	h	n	b	v	m	u	l	i	c
n	a	n	g	d	u	i	i	f	w	u	i	l	o	n	u
e	c	e	f	v	g	t	o	h	d	m	k	o	v	g	t
c	k	c	p	i	t	t	i	n	g	a	c	v	n	b	t
c	l	w	o	b	b	l	e	n	b	v	u	n	i	o	p
e	a	o	n	y	n	d	n	b	n	a	p	i	s	n	f
c	s	r	i	t	e	m	u	d	n	e	d	d	a	e	i
c	h	m	o	d	j	u	o	d	i	o	p	y	h	i	n

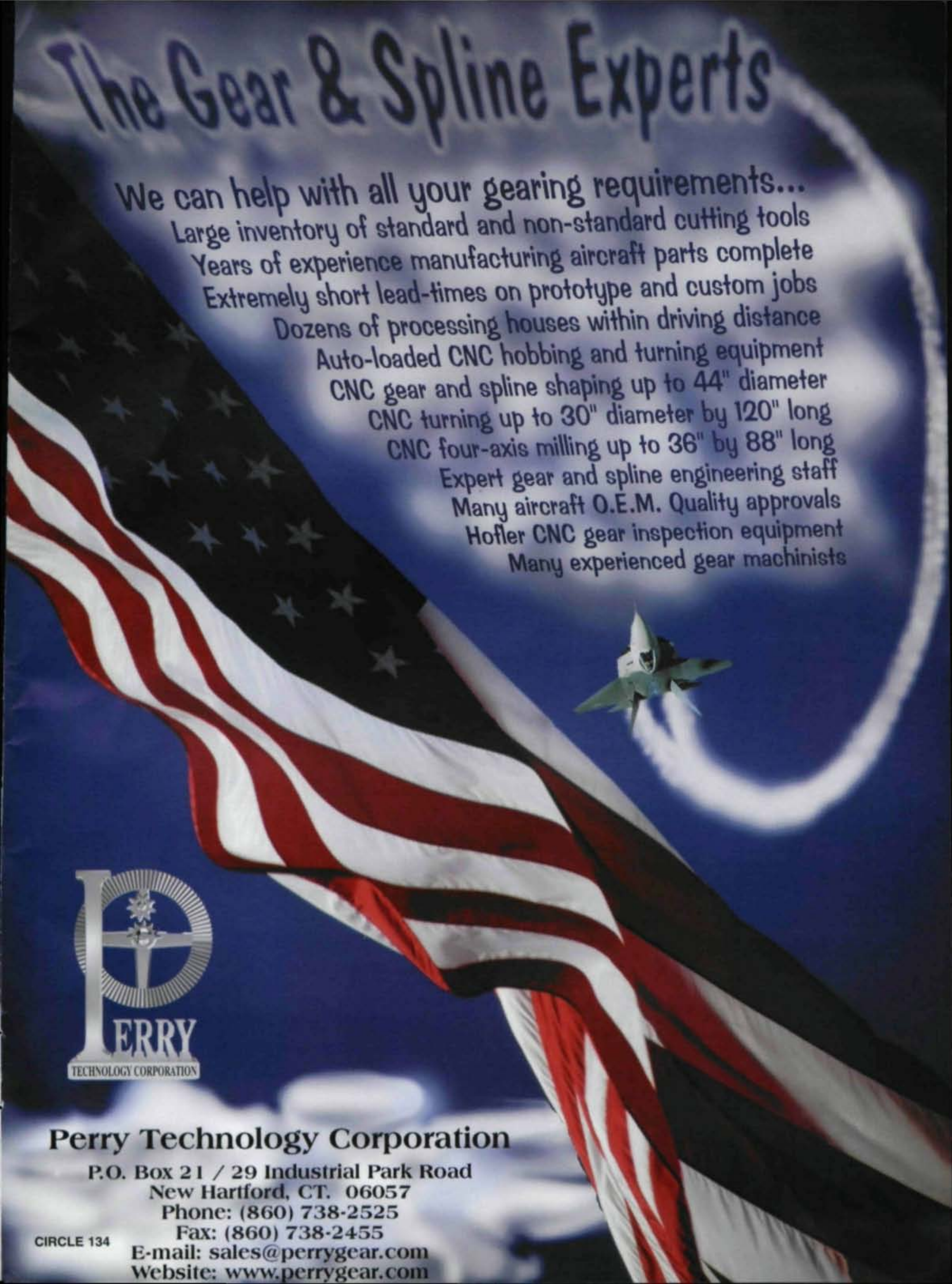
addendum*
apex
backlash
bevel
concentricity
dedendum
eccentricity
fatigue
gears
grinding

honing
helical
herringbone
hobbing
hypoid
involute
pinion
pitting
rack
rippling

skew
spiral
spline
spur
tip
tooth
undercut
wear
worm
wobble

***A very excellent gear word.**

Tell Us What You Think . . .
If you found this article of interest and/or useful, please **circle 340.**
If you did not care for this article, **circle 341.**


The Gear & Spline Experts
We can help with all your gearing requirements...
Large inventory of standard and non-standard cutting tools
Years of experience manufacturing aircraft parts complete
Extremely short lead-times on prototype and custom jobs
Dozens of processing houses within driving distance
Auto-loaded CNC hobbing and turning equipment
CNC gear and spline shaping up to 44" diameter
CNC turning up to 30" diameter by 120" long
CNC four-axis milling up to 36" by 88" long
Expert gear and spline engineering staff
Many aircraft O.E.M. Quality approvals
Hofler CNC gear inspection equipment
Many experienced gear machinists
PERRY
TECHNOLOGY CORPORATION
Perry Technology Corporation
P.O. Box 21 / 29 Industrial Park Road
New Hartford, CT. 06057
Phone: (860) 738-2525
Fax: (860) 738-2455
E-mail: sales@perrygear.com
Website: www.perrygear.com
CIRCLE 134