GEAR TECHNOLOGY

MARCH/APRIL 2003

The Journal of Gear Manufacturing

www.geartechnology.com

HEAT TREATING

- Process & Material Selection for High Performance Gears
- Wear Resistance of Plasma & Pulse Plasma Nitrided Gears

ALSO IN THIS ISSUE

- Contact Pattern of Worm Gears
- Design Robustness

THE GEAR INDUSTRY'S INFORMATION SOURCE

GEAR TECHNOLOGY

MARCH/APRIL 2003 *The Journal of Gear Manufacturing*

HEAT TREATING ISSUE

26

FEATURES

DEPARTMENTS

42

Cover photo courtesy of The Cincinnati Steel Treating Co.

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

EDITORIAL

Publisher & Editor-in-Chief
Michael Goldstein

Managing Editor William R. Stott

Associate Editor Joseph L. Hazelton

Assistant Editor Robin Wright

Editorial Consultant Paul R. Goldstein

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

ART

Art Director Jean Bartz
Graphic Designer Brian A. Blevins

ADVERTISING

Advertising Manager Patricia Flam

INTERNET

Web Developer Dan MacKenzie

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

***powertransmission.com*™ Sales**
Michael A. Gianfrancesco

RANDALL PUBLISHING STAFF

President Michael Goldstein
Vice President Richard Goldstein
Accountant Michael Ellis
Accounting Dianne Borg
Office Coordinator Gladyce J. Cooper

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

BPA INTERNATIONAL

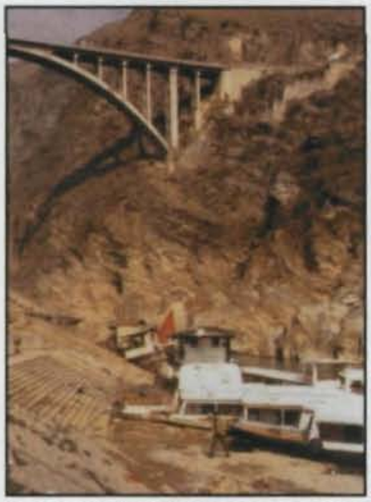
VOL. 20, NO. 2

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 (USPS No. 749-290). 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.


• Improve hobbing speed
• Lower costs
• Raise quality
LC 80
LIEBHERR
Liebherr hobbing technology does it all. As the pioneer of high-speed dry machining for gear cutting, Liebherr has compressed cycle times and improved quality with innovative machine features. Our CNC hobbing machines cover the full spectrum of gears up to a diameter of 4500 mm. Over 1,000 installations of this machine concept testify to its acceptance. Couple all this with over 50 years of experience and support, and your choice for gear hobbing has to be Liebherr.
SIGMA POOL
The Gearing Partnership of Liebherr, Klingelnberg and Oerlikon
Liebherr Gear Technology Co.
1465 Woodland Drive
Saline, MI 48176-1259
734.429.7225 • Fax 734.429.2294
info@lgt.liebherr.com
LIEBHERR
Gear Hobbing • Shaping • Grinding • Measuring

Chinese Butterflies

Have you ever heard the story about the butterfly that flaps its wings in China and causes a hurricane in another part of the world? I've heard many variations of that story, but each illustrates the idea that even the tiniest change can produce enormous effects in the future.



The Dragon Gates Bridge crosses a gorge approximately 125 meters above the water of the Yangtze River. In June 2003, the water will be raised to about 10 meters above where the top of the bridge now stands.

After recently visiting China, I can assure you that the butterfly is flapping—furiously. The changes going on in China are having an enormous impact on the world. Those of us in manufacturing should consider those changes and how they could affect us and our companies down the line.

Most of us have known for some time that the butterfly is flapping in China. But it wasn't until I went there and saw some of the activity myself that I realized the butterfly weighs about 500 pounds.

For example, China has building projects that dwarf anything ever done. One of the best examples is the Three Gorges Project on the world's third longest river, the Yangtze.

Through this project, the Chinese government is building what will be the world's largest hydroelectric dam. The concrete structure will tower 610 feet (186 meters) above the Yangtze. The 1.3 mile-wide dam will raise the water level by 175 meters, creating a reservoir from Yichang City in Sandouping to Chongqing—a distance of 400 miles (more than 600 km).

The Three Gorges Project began in 1993 and is supposed to be completed by 2009. This year, in June, they will be raising the level of the water behind the dam by 135 meters.

When complete, the Three Gorges Dam will contain 27.15 million cubic meters of concrete, more than twice as much concrete as the Itaipu Dam on the Parana River between Brazil and Paraguay. That dam used 12.8 million cubic meters and is currently the world's largest hydroelectric dam. The Grand Coulee Dam, America's largest, used 9.2 million cubic meters.

The Three Gorges Dam will use 26 turbine generators to produce up to 18,200 megawatts of electricity. That's more electricity than a dozen nuclear power plants.

When the river is fully dammed, 13 cities, 140 towns and more than 1,300 villages will be submerged. Consequently, nearly 2 million people and thousands of houses, apartments, businesses, factories, farms, ports and wharfs are being relocated from the banks of the river to the tops of the gorges.

Also, the dam is supposed to improve flood control and navigability of the river for ocean-going vessels, allowing more cargo to go into and out of central China.

There is plenty of other construction going on throughout China, too. Office buildings are going up like mushrooms in Beijing and Shanghai. Those in Beijing are very utilitarian, but the new buildings being constructed in Shanghai are gorgeous examples of distinctive modern architecture.

What I saw of the characteristics of the Chinese people leads me to believe that their level of activity is going to continue for some time. The Chinese people I met and observed seemed to be ambitious, cheerful, optimistic, hard working and skilled. In addition, more and more of the Chinese workforce is becoming highly educated, and recent economic reforms have enabled the spirit of entrepeneurship to flourish.

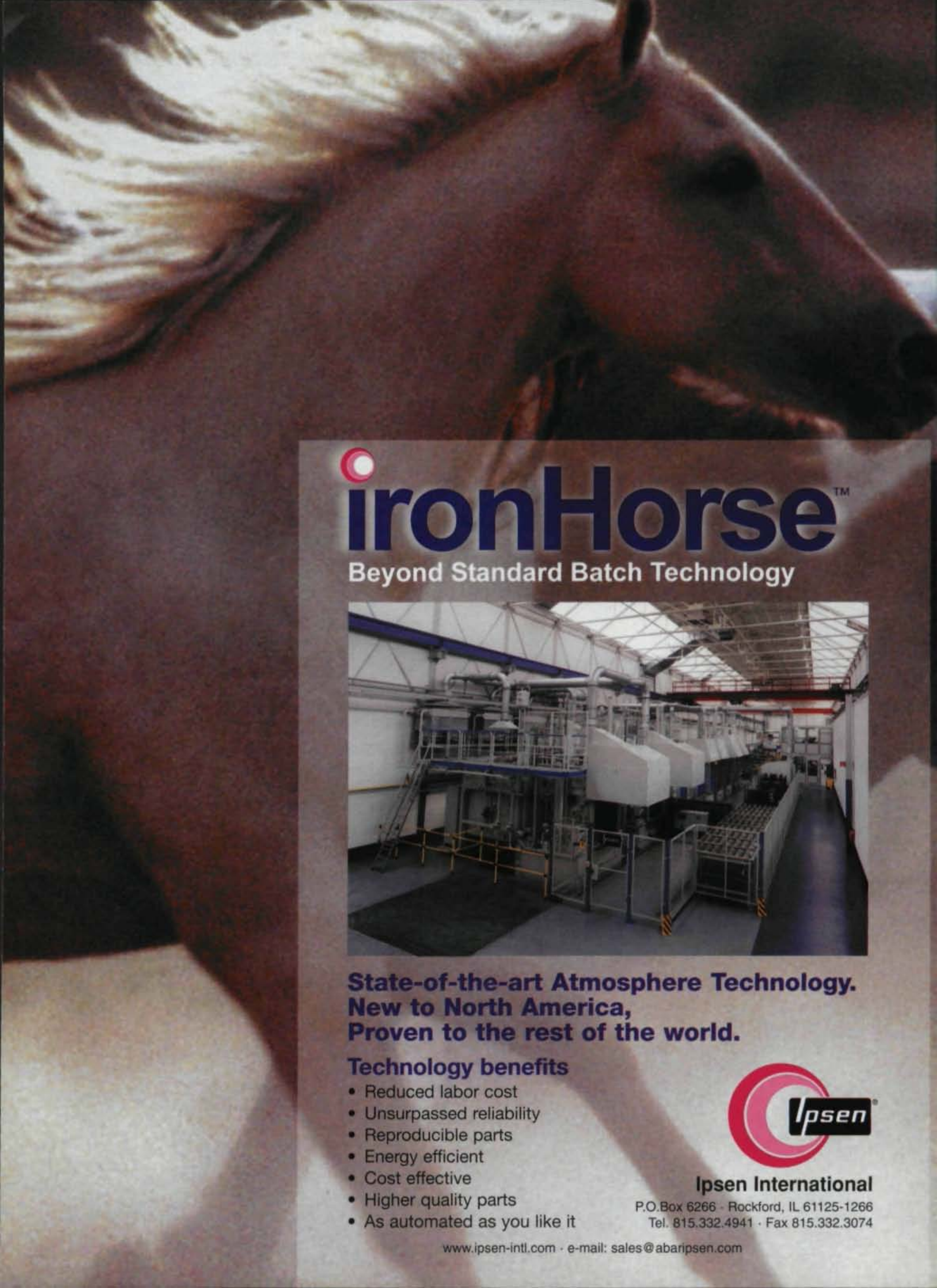
So what does this mean to those of us who are involved in manufacturing in the rest of the world? It's hard to tell what will happen when a butterfly flaps its wings.

I know a number of gear manufacturers who have formed relationships with one or more Chinese companies. Some are buying their blanks from overseas. Some are importing rough gears from China and are finish-grinding them here. Others are producing gears in China. Today's manufacturing world is becoming one without borders. Things that happen in one area of the world can have a huge impact in other areas. Those people who are best able to think globally today may be best able to compete globally tomorrow.

In the end, it may turn out that the butterfly is not in China after all. Perhaps it's in another region of the world altogether. Then again, perhaps it is *you* who are the butterfly, and it's the small changes you make today—including paying attention to the global economy—that will have a big impact on *your* future.

Michael Goldstein

Michael Goldstein, Publisher & Editor-in-Chief


ironHorse™
Beyond Standard Batch Technology
State-of-the-art Atmosphere Technology.
New to North America,
Proven to the rest of the world.
Technology benefits
• Reduced labor cost
• Unsurpassed reliability
• Reproducible parts
• Energy efficient
• Cost effective
• Higher quality parts
• As automated as you like it
Ipsen
Ipsen International
P.O.Box 6266 · Rockford, IL 61125-1266
Tel. 815.332.4941 · Fax 815.332.3074
www.ipsen-intl.com · e-mail: sales@abaripsen.com

Gear Technology Launches Electronic Version

E-GT, the electronic version of *Gear Technology*, launched with our January/February 2003 issue. Feedback from our readers around the world has been tremendous.

Within 24 hours of notifying our e-subscribers that *E-GT* was available, nearly 500 of them had come to our website to retrieve the files. Many of those who have used *E-GT* have written to offer their congratulations and comments.

"I have just received your electronic version of *Gear Technology* and wish to express my satisfaction for having all the information available on my desktop. Keep up the good work," wrote the chief operating officer of a major gear and drive manufacturer in Italy.

E-GT is an exact duplicate of the printed magazine, except that it has been prepared in PDF format. *E-GT* includes all of the same technical articles and departments found in the printed version. It includes all of the charts, graphs, figures and tables. It even includes all of the advertisements.

"I have now read the January/February 2003 issue online," wrote a gear specialist at the transmission division of a major American automobile manufacturer. "The quality of the electronic issue is excellent. All the text and figures can be easily read, even better than in the paper edition (the zoom feature is quite helpful). The uniquely electronic nature of *E-GT* is a real plus since I can search far more powerfully."

E-GT reaches nearly 1,500 subscribers in more than 50 countries—and the list grows daily. Many of those subscribers wouldn't otherwise be able to receive our content, especially those who are outside the United States.

"*E-GT* sets us apart from other trade magazines," says publisher and president Michael Goldstein. "Through this format, we're able to deliver our information and advertising anywhere in the world. We're using *E-GT* to expand our international readership."

E-GT is available for free to qualified readers—anywhere in the world—and subscribers don't have to wait for it to be delivered through the mail.

"Thanks for mailing me the electronic version of *Gear Technology*," wrote the general engineering manager of a major industrial gear drive manufacturer in India. "I had no problems in downloading the same."

If you'd like to join our growing list of *E-GT* subscribers, visit our website at *www.geartechnology.com* and follow the links under the *E-GT* logo, or fill out the card that's bound inside this magazine.

Ipsen Offers a New Variation of Automated Heat Treat System

A set of rails, a row of process chambers and a mobile vacuum loader make up the latest heat treat system from Ipsen International Inc.

The system's purpose: Allow companies to run different heat treat jobs at the same time and move their parts (read: gears) for quenching while keeping each gear load at its ideal temperature.

Ready for sale in October, the system was designed to make a company's heat treat operation as flexible as possible.

Called mult-i-cell, the system can include process chambers that vacuum carburize, vacuum harden or carbonitride. It also can include low temperature chambers that can nitride or ferritic nitrocarburize.

New from Ipsen International Inc., mult-i-cell includes a mobile vacuum loader (far left) that moves on rails in front of process and quench chambers.

Mult-i-cell can heat treat various types of loose gears. It can also heat treat transmission and drive shaft gears, as well as axles.

In a heat treat operation, the chambers are arranged in a row and the rails are laid down in front of them. Riding the rails, the 2.5-ton vacuum loader moves in front of the chambers, transferring gears from process chambers to high pressure gas quench chambers or oil quench chambers.

The vacuum loader is the unique part of the system, says Thomas Wingens. A materials engineer and metallurgist, Wingens is vacuum product manager at Ipsen International's operation in Rockford, IL.

The loader transports processed loads under heat and vacuum, so a gear load can be transferred from process chamber to quench chamber without exposure to a factory's cooler air.

Also, as Wingens explains, the loader can be changed to any temperature in minutes. So, it can carry a load of gears at one temperature, then carry another load at a different temperature.

The loader's temperature can range up to 2,000°F. Lowering the temperature over a wide range, like from 1,700°F to fully cooled down, takes about six minutes and requires using the system's cooling station.

For cooling more than 500°F, the loader is moved in front of the station, which consists of blowers. The loader's door is opened, and the station's blowers force cooler air into it.

Lowering the temperature over a narrower range, like from 1,700°F to 1,600°F, happens via natural heat loss. Increasing over such a narrow range can be done in two minutes, Wingens says.

He adds that processing temperatures usually vary within a 50–100°F range.

Mult-i-cell is the latest among a number of heat treat systems designed to process gears under vacuum and transfer them for quenching without exposing them to atmosphere.

Other systems for transferring gears without exposure to cooler air come from ALD Thermal Treatment Inc. of Blythewood, SC, and ECM U.S.A. Inc. of Kenosha, WI.

Like Ipsen's mult-i-cell, ALD's ModulTherm consists of a row of process chambers. But, ALD uses its rails to mobilize the system's quench chamber.

The quench chamber contains the loader and moves in front of the process chambers. Therefore, ModulTherm transfers gear loads directly from process chamber to quench chamber.

Introduced about a year ago, ALD's system has process chambers for plasma carburizing, chambers for vacuum carburizing and chambers for vacuum hardening. Its quench chamber can gas quench gears. ALD has sold one system.

Available since 1996, ECM's system consists of two rows of chambers. The rows face each other and are connected to a vacuum tunnel via ports on its two long "sides." Inside the large tunnel, a shuttle car moves gear loads between process chambers, a quench chamber, and a load/unload chamber.

Called ICBP, ECM's system can include process chambers that carbonitride, vacuum harden and vacuum carburize. ECM has sold more than 55 systems, in the horizontal configuration described and in a vertical configuration.

All three systems can be expanded with additional chambers. ECM's tunnel usually has a maximum of eight ports: six for process chambers, one for a quench chamber and one for a load/unload chamber. Still, tunnels can be created with 10 or 12 ports.

Ipsen's system can include eight process chambers with one loader. It can also include up to 10 process chambers with one quench chamber. (With nine or 10 process chambers, mult-i-cell would have two loaders on the same track.)

With 11 or more process chambers, a company would need to set up a second system. But, Wingens says the Ipsen system might be kept from its maximum size by other constraints, like the amount of floor space available for a row of chambers.

ALD's standard system can have up to six carburizing chambers, but the company can attach more process chambers.

The number of process chambers depends on the production volume per time unit and on the case depths that a company needs in its gears. As Wingens explains, if the gears need shallow case depths, mult-i-cell can consist of six process chambers, for example. If they need deeper case depths, then the system would need more process chambers, like eight, because of longer carburizing times.

Ipsen's base system—a process chamber, a quench chamber and a vacuum loader—starts at $1.2 million. Additional process chambers cost $300,000 apiece.

The system can handle gear loads that are 24" wide, 36" deep, 25" high, and weigh 1,100 pounds.

All three systems are fully automated and include software that schedules workloads for maximum efficiency. The more varied the heat treat processes and the tighter the time constraints, the more helpful the software for scheduling jobs.

Also, according to their representatives, all three systems have uptimes of 90% or more.

Since Ipsen finished testing mult-i-cell in October, it has sold the system to a commercial heat treater in Germany and a captive gear job shop in China.

Wingens says the system was designed for captive and commercial heat treaters serving the automotive and aerospace industries.

Still, he predicts that automotive companies will process the greatest volume of parts with the system and that their greatest demand will be for carburizing gears.

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- Rate this column
- Request more information
- Contact the companies mentioned
- Make a suggestion

Or call (847) 437-6604 to talk to one of our editors!

Determination and Optimization of the Contact Pattern of Worm Gears

Bernd-Robert Höhn, Karl Steingröver and Michael Lutz

Summary

The load capacity of worm gears is mainly influenced by the size and the position of the contact pattern. A new method was developed that allows for the determination and optimization of the idle and load contact patterns in the design stage. By this method, the contours of worm and worm wheel are simulated point by point, taking into account the boundary conditions of the manufacturing process.

The idle contact pattern can be derived from these contours by pairing them together in such a way that the assembly deviations define the position of worm and wheel. The load contact pattern can be determined from the idle contact pattern by adding the deflections of the teeth and gear bodies and the elastic deformations of the bearings and the housing.

This procedure can also be used for automated optimization of the contact pattern, so optimum machine settings can be found without a trial manufacturing. Comparisons of these theoretical contact patterns with real contact patterns of gears in practice showed a good correlation.

Introduction

The load capacity of worm gear drives is mainly influenced by the size and the position of the contact pattern. The actual load capacity calculations according to DIN 3996 (Ref. 1) or ISO CD 14521 (Ref. 2) assume contact patterns that are well positioned and cover nearly the whole flank of the wheel.

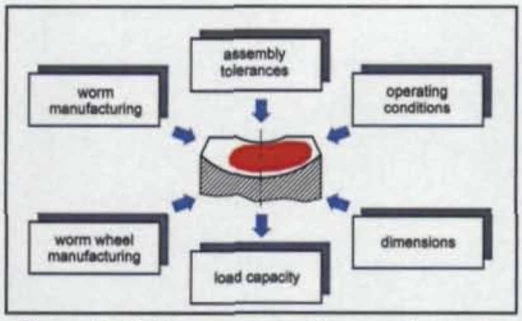


Figure 1—Influence parameters on the contact pattern of worm gears.

Size and position of the contact pattern depend on many parameters, like manufacturing type, accuracy, geometry of the housing, kind of bearings and operating conditions (see Fig. 1).

The estimation of the influence of these parameters on the contact pattern, and therefore on the load capacity, requires great experience.

Normally, the idle contact pattern is checked after assembly by painting some teeth of the wheel with contact paint. After several revolutions, the abrasion of the contact paint is used to evaluate the idle contact pattern. Although this procedure is simple, it is time consuming. Furthermore, experience is needed if the contact pattern has to be adjusted. Other disadvantages of this method are that the load contact pattern and local specific overloads cannot be detected. To avoid this old-fashioned procedure, an analytical method was developed that allows for the determination of the idle and load contact patterns in the design stage.

These investigations were carried out at the Gear Research Centre (FZG) at the Technical University of Munich, Germany, and were supported by the Gear Research Organization (FVA) of Frankfurt, Germany, through research project 252 (Ref. 3).

Idle Contact Pattern

By this new method, the contours of the worm and the wheel are calculated point by point by taking into account the boundary conditions and deviations of the manufacturing process.

The points of the worm in the axial sections and the wheel in the corresponding sections are described by simulating the final manufacturing process (grinding wheel or hob). Then, these two contours are brought into contact in a way so the center distance and the assembly deviations define the position of each contour. If this is done for several mesh positions, the idle contact pattern is then the summation of the smallest distances between the contours of worm and wheel at each mesh position.

Calculation of the contour of the worm. The contour of different types of worm flanks can be described by simulating the final manufacturing process, which is usually done by grinding or cutting. In accord with DIN 3975, the worm's flank contours that are ground include a concave profile in the axial section, an involute profile in the transverse section and a convex profile in the axial section. The flank contours that are cut include a straight profile in the axial section and a straight profile in the normal section (Ref. 4). The grinding disk, for example, can be characterized by the diameter d_0, the pressure angle α_0 and the profile (see Figure 2).

The contour of the grinding disk can be either described by analytical equations (Ref. 5) or approximated by a series of discrete points. Here the approximation by discrete points is used.

The discrete points of the worm contour can be achieved from the points of the grinding disk by simulating the manufacturing process in a way so the grinding disk has to be rotated around the worm axis in several steps and simultaneously has to be moved in the direction of the worm axis to achieve the lead. An example is shown in Figure 3.

The advantage of this procedure over the analytical method is that the real geometry—which deviates from the ideal geometry—can be taken into account. These deviations are grinding with a modified center distance, modified pressure angle or modified lead. Furthermore, modifications like crowning can be added to this model.

Calculation of the contour of the worm wheel. The basic idea for calculating the contour of worm wheel flanks is the same as shown for the worm. Here the final manufacturing process is usually done by cutting. The hob can be characterized by the diameter d_0, the pressure angle α_0 and the profile.

The hob's contour can also be approximated by a series of discrete points. The discrete points of the wheel contour can be achieved by simulating the manufacturing process as shown in Figure 4. Here the hob has to be rotated around the wheel axis in several steps and simultaneously moved in the feed direction. An example for such a wheel tooth contour is shown in Figure 5.

Because the contour of the wheel is not based on empirical equations, the influences of modifications on the hob, like increased hob diameter, center distance modification and lead modification, can be taken into account.

Simulation of the assembly of the worm gear. During the assembly of worm and wheel in the

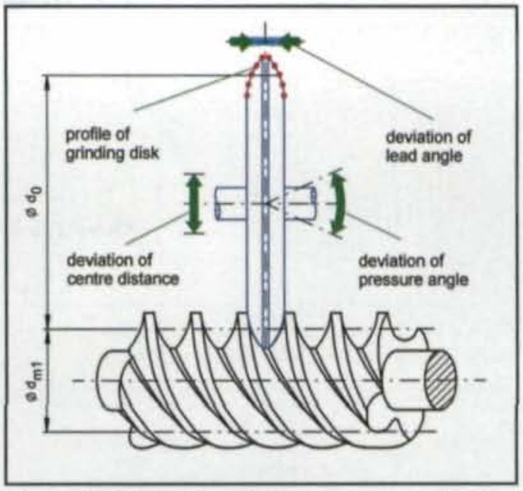


Figure 2—Derivation of the worm contour by simulating the grinding process.

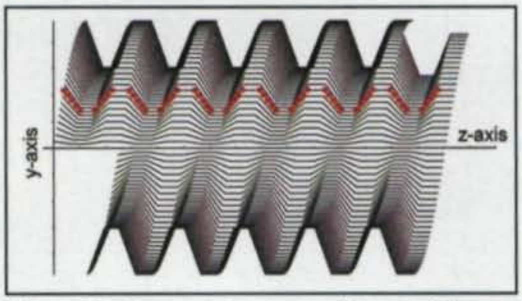


Figure 3—Contour of a worm with a straight profile in the axial section and with two teeth (individual points only shown for one plane).

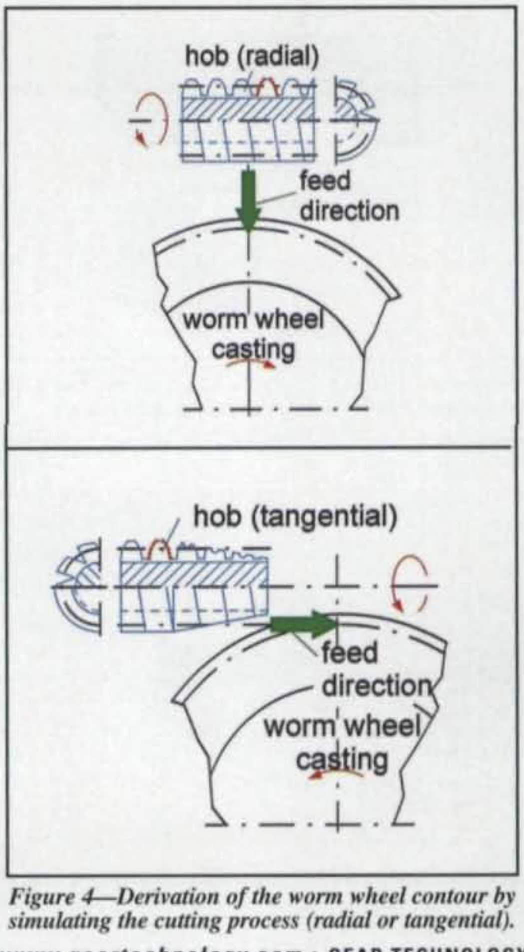


Figure 4—Derivation of the worm wheel contour by simulating the cutting process (radial or tangential).

Prof. Dr.-Ing. Bernd-Robert Höhn

is head of the Gear Research Centre, a part of the Technical University of Munich, located in Germany. The centre's main activities include theoretical and experimental investigations of spur, helical, bevel and worm gears and development of software for calculating geometry and load capacity of gears.

Dr.-Ing. Karl Steingröver

is a chief engineer at the Gear Research Centre and is responsible for its worm gear and computer programming groups.

Dr.-Ing. Michael Lutz

is a distributor of gear and chain drives via his company, Ingenieurservice Lutz GmbH of Altdorf, Germany. Previously, he worked in the Gear Research Centre's worm gear group and carried out the investigations described in this paper.

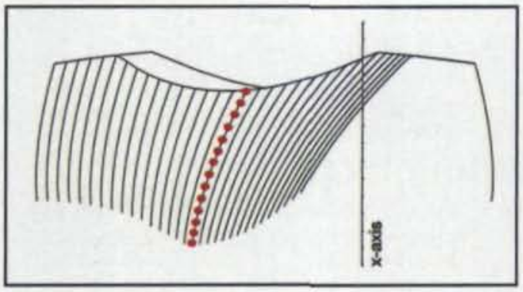


Figure 5—Contour of one tooth of a worm wheel (individual points only shown in one plane).

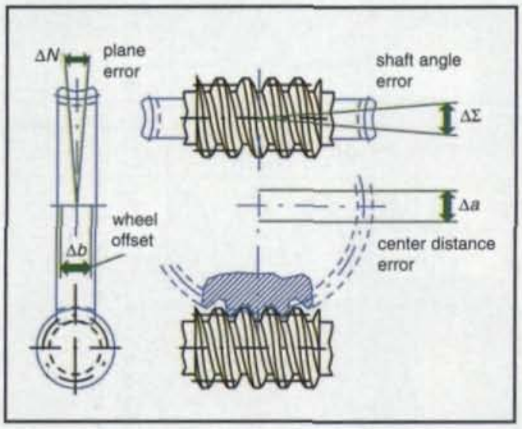


Figure 6—Deviations from the ideal mounting position of worm and wheel in the housing.

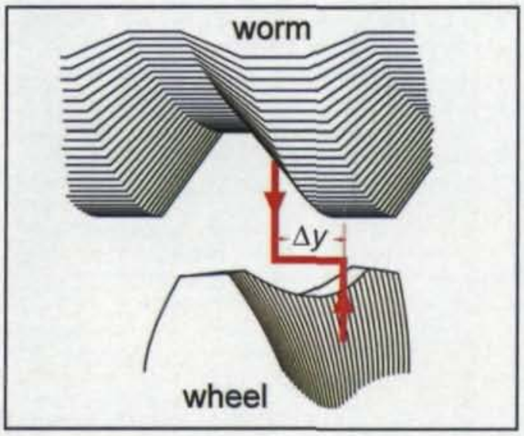


Figure 7—Pairing of the contours of the teeth of worm and wheel (individual points not shown).

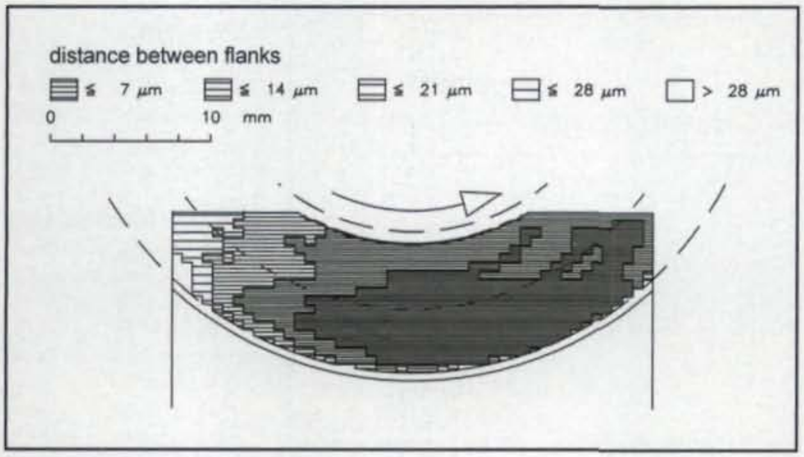


Figure 8—Example for an idle contact pattern.

housing, deviations from their ideal position can occur. These deviations, of course, also have an influence on the size and the position of the contact pattern. The most important deviations are (see Fig. 6):

- wheel offset Δb,
- deviation of the center distance Δa,
- shaft angle error $\Delta\Sigma$, and
- plane error ΔN.

If the contour from the discrete points of the worm and wheel teeth are known, the assembly process can then be simulated by sliding the worm and wheel together, taking into account the aforementioned boundary conditions.

Now the distances between the different points of worm and wheel can be calculated. Usually for one pair of points there is a minimum distance Δy_0 (see Figure 7). From all distances Δy_i, Δy_0 has to be substracted. Distances, which fall into a specified small range, can now be viewed as contact points for this position of worm and wheel. These contact points form one contact line. This procedure has to be repeated for a series of positions of worm and wheel. The individual contact lines then form the idle contact pattern. An example is shown in Figure 8.

Load Contact Pattern and Specific Load

The idle contact pattern is not identical to the load contact pattern. Under load, the contact pattern is further influenced by the worm deflection, the tooth deflection, deflection of the gear bodies and the elastic deformations of the bearings and the housing. For the determination of the load contact pattern and the specific load, the method of influence numbers is used. In this model, all influences that do not depend on the load are described by a rigid body, while all load-dependent influences are described by a spring system (see Fig. 9). The application of a tooth load to this model leads to deformations at the different stiffnesses and therefore to different specific loads at the specific points along the flank. The load contact pattern is the summation of all points where the load is > 0.

For load-free conditions and ideal geometries, the pads (spring system) and the wedges (rigid body) are in contact over the whole length (see Fig. 9a). This is the theoretical contact line. For real geometries with no load, there is a contact only in one point (see Fig. 9b). In this position, the pad is still undeformed. These are the boundary conditions of the idle contact pattern. Under load, the pads will be deformed (see Fig. 9c). The total deformation can be described by a system of equations. For the determination of the discrete deformations at each contact point and the determination of the single loads at these points, the following equations have to be solved:

Summation of the single loads:

$$F_{bn} = \Sigma F_i = E \cdot F \qquad (1)$$

with: F_{bn} = total load, F_i = single load in a spring element, E = unit vector, and F = vector with all single loads.

Components of a single deformation:

$$\delta_i^* = \delta_{ges} - \delta_i \qquad (2)$$

with: δ_i^* = single deformation in one spring element, δ_{ges} = total deformation of the pad, and δ_i = distance between spring element and pad with no load.

System of load-deformation equations:

$$\delta^* = q \cdot F \qquad (3)$$

with: δ^* = vector with single deformations and q = matrix with all elastic components.

These equations lead to Equation 4:

$$F_{bn} = E \cdot q^{-1} \cdot [\delta_{ges} - \delta_i] \qquad (4)$$

This is a scalar equation, where F_{bn} is known from the torque. The determination of the matrix q with all elastic components is made by calculating all influences separately according to Equation 5:

$$q = q^{SW} + q^{SZ} + q^{RW} + q^{RZ} + q^{WL} + q^{GH} \qquad (5)$$

where the single matrices with elastic components are (see Fig. 10):

- q^{SW} for the influence of worm shaft deflection,
- q^{SZ} for the influence of the deflection of a thread of the worm,
- q^{RW} for the influence of the wheel shaft deflection,
- q^{RZ} for the influence of the deflection of a tooth of the wheel,
- q^{WL} for the influence of the deformation of the bearings, and
- q^{GH} for the influence of the deformation of the housing.

The determination of the different matrices can be done by using the method of influence numbers. This is shown as an example for the deformation of a tooth of the wheel (see Fig. 11).

To determine the influence numbers for the tooth deformation, the theory of thin slices is used. For each contact point, one slice is used. Depending on the tooth shape and the point where the load acts, the deviation of the tooth can be cal-

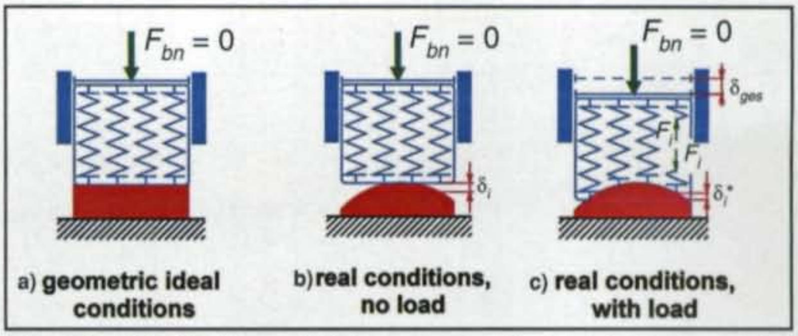


Figure 9—Model of the mesh between worm and worm wheel: a) geometric ideal conditions; b) real conditions, no load; c) real conditions, with load.

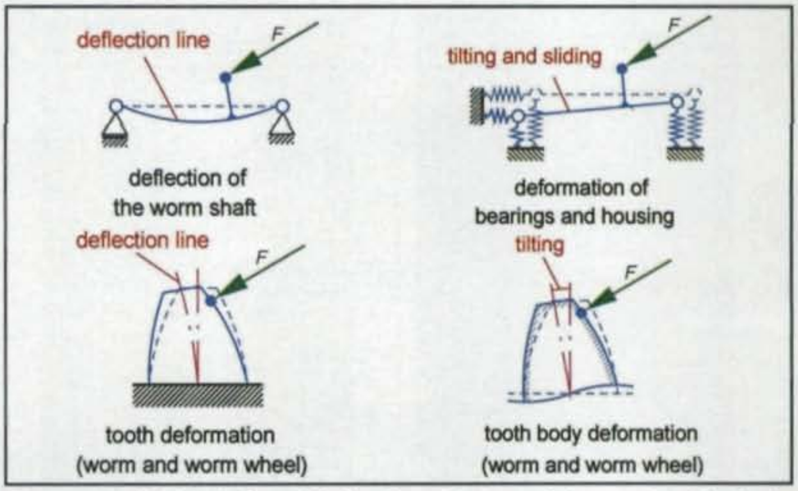


Figure 10—Mechanical models for determination of the matrix with elastic components.

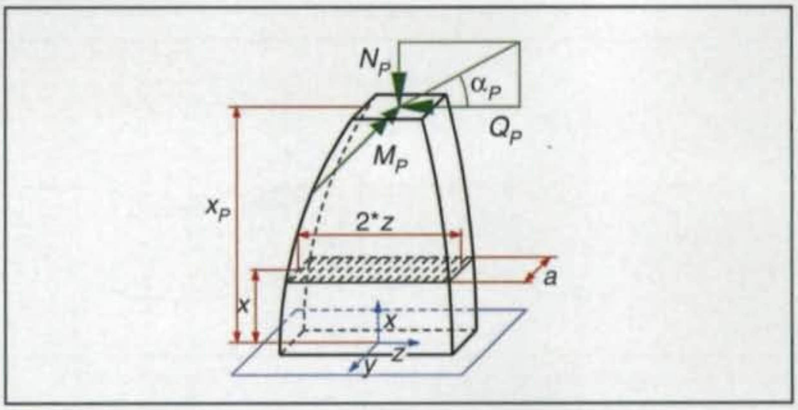


Figure 11—Slice model for the influence of the deformation of a tooth of a wheel; thickness a = constant.

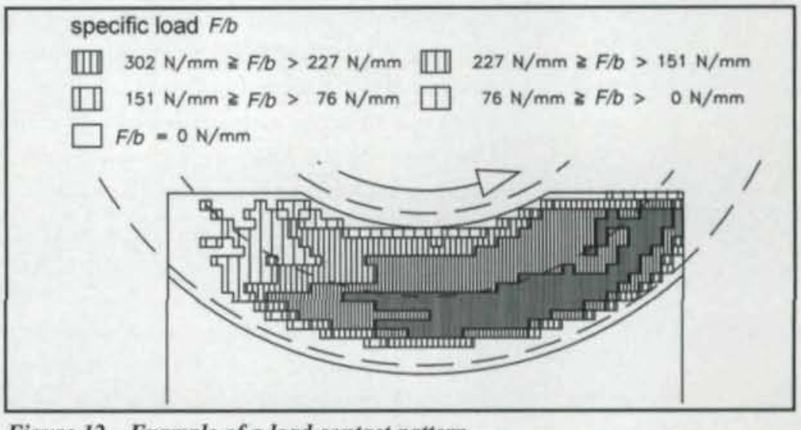


Figure 12—Example of a load contact pattern

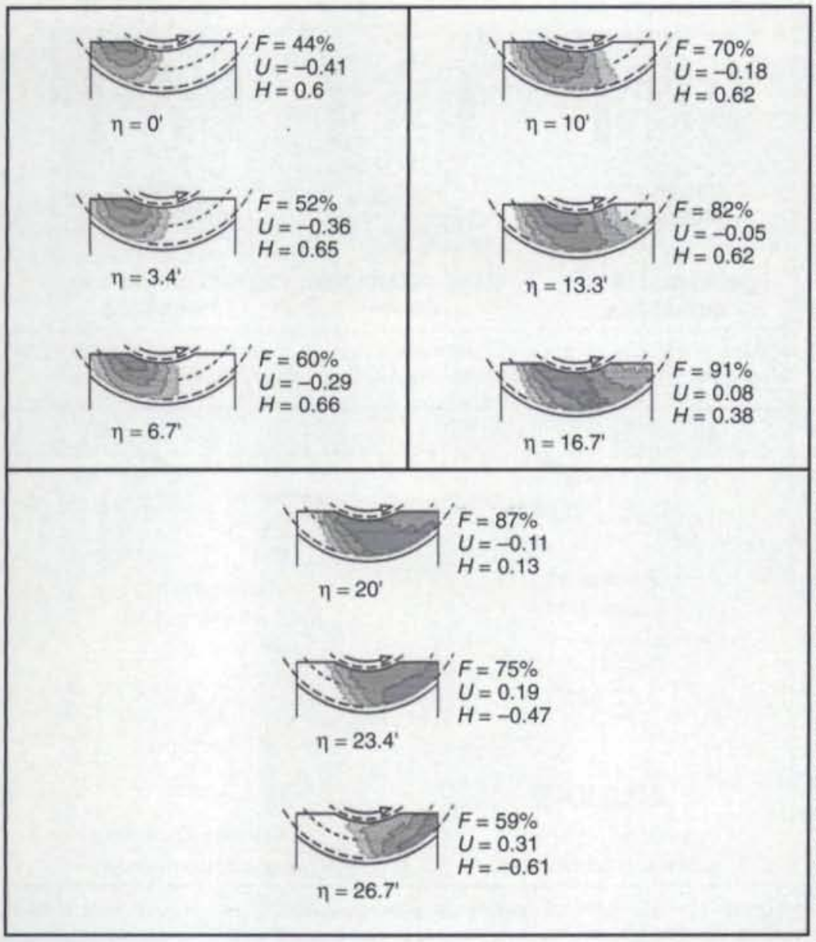


Figure 13—Series of contact patterns depending on the traverse angle of the hob.

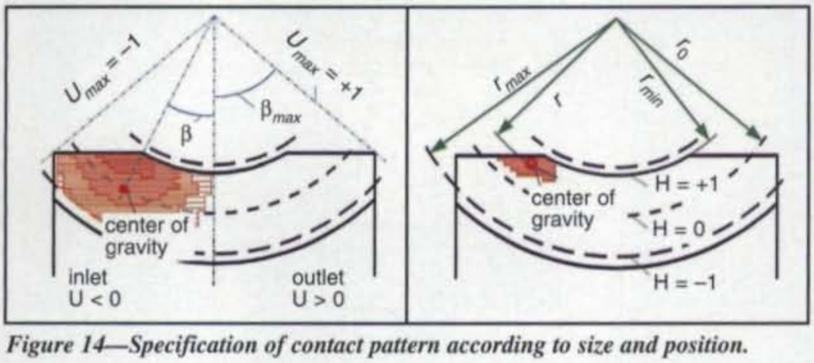


Figure 14—Specification of contact pattern according to size and position.

culated according the theory of Weber and Banaschek (Ref. 6). The load P is divided into its components: Q_P, N_P and M_P. These lead to a normal load N, a transverse load Q and a bending moment M in the section at x.

To determine the deformation in the direction of the load, the deformation energy is set equal to the integral over the elastic stress energy. It is:

$$\frac{1}{2} P w_z = \frac{1}{2}\int_0^{x_p} \frac{M^2}{EI}\,\mathrm{d}x + \frac{1}{2}\int_0^{x_p} \frac{Q^2}{GA_s}\,\mathrm{d}x + \frac{1}{2}\int_0^{x_p} \frac{N^2}{EA}\,\mathrm{d}x \tag{6}$$

with: P = load, w_z = deformation in direction of the load, x_p = length of the slice, α_p = load direction angle, Q = transverse load at x, N = normal load at x, M = bending moment at x, E = Young's modulus, G = shear modulus, I = mass moment of inertia at x, A = cross section at x, and A_S = shear area at x.

This equation can be solved in the direction of the deformation w_z, which can be viewed as one point in the matrix q^{RZ}.

The result of this procedure is a load contact pattern as shown in Figure 12. Contrary to the idle contact pattern, where the different levels characterize the minimum distances between worm and wheel, the different levels of the load contact pattern characterize the specific loads along the contact lines.

Optimization of the Contact Pattern

The procedures described in the previous sections lead to contact patterns that correspond very well with measured contact patterns of gears in practice. Normally, the measured contact patterns do not have the optimum size and position. Therefore, an optimization must be done, for which great experience is necessary. A contact pattern, calculated by using the procedures described in the previous sections, can be optimized by variation of the different manufacturing and assembly parameters. This makes sense only if there is knowledge of how the different parameters influence the size and position of the contact pattern. In the following, a procedure is described stating how this selection can be done automatically.

First, the parameters—which can be varied for achieving a better contact pattern—have to be selected. Then, a series of calculations has to be made by varying these parameters in several small steps. This leads to a series of contact patterns. An example of such a series is shown in Figure 13.

From this series, the optimum contact pattern (and the corresponding manufacturing and assembly parameters) can be found by classifying the size and position of the contact patterns using the characterizing parameters F, U and H:

The parameter F is the calculated area of the contact pattern as a percentage of the theoretical possible contact area as shown in the left part of Figure 14.

The parameter U characterizes the position of the contact pattern in the circumferential direction of the worm. For the calculation of U, the center of gravity of the area used for the determination of F, is used (see left part of Fig. 14). It can be determined using Equation 7:

$$U = \pm \beta/\beta_{max} \quad (7)$$

Per definition, values for U between −1 and +1 are possible. If the center of gravity is in the inlet area, U will have a negative value; $U = 0$ if the center of gravity is in the middle of the flank and a position in the outlet area has a positive value.

The parameter H describes the position of the contact pattern in the direction of the tooth height. For this, the center of gravity of that area of the contact pattern is used where the smallest distances were calculated in the case of the idle contact pattern or the highest specific loads were calculated in the case of the load contact pattern. (see right part of Fig. 14). The restriction on this small area was done because if the whole area is used, contact patterns with peaks at the tip or the root would be classified as well-adjusted contact patterns. H can be determined as follows:

$$H = (r - r_{min})/(r_0 - r_{min}) \text{ if } r < r_0$$
$$H = (r - r_{min})/(r_0 - r_{max}) \text{ if } r \geq r_0 \quad (8)$$

Also, the parameter H can reach values between −1 and +1, where a positive value represents a position of the contact pattern in the direction of the tip of the tooth and a negative value represents a position in the direction of the root.

The result of this classification is shown by an example in Figure 15. A typical position of a contact pattern is in the tooth height direction in the middle and in the circumferential direction tending slightly into the outlet area. This means H should be in the area of 0 and U should be in the area of 0.1. In the example, this corresponds to a traverse angle $\eta \approx 20'$, which leads to a size of the contact pattern (parameter F) of 87%.

Conclusion

These procedures were verified by a comparison of calculated idle contact patterns with real contact patterns of worm gears in practice. An example from this comparison is shown in Figure 16. From this figure, it can be seen that the calculated contact pattern correlates very well with the measured contact pattern. Therefore, these procedures are suitable for the determination and optimization of the contact pattern of worm gears. ○

This paper was previously presented at the International Conference on Gears, held March 13–15, 2002, in Munich, Germany. Also, the paper was previously published by VDI Verlag in the conference's proceedings, in VDI report 1665.

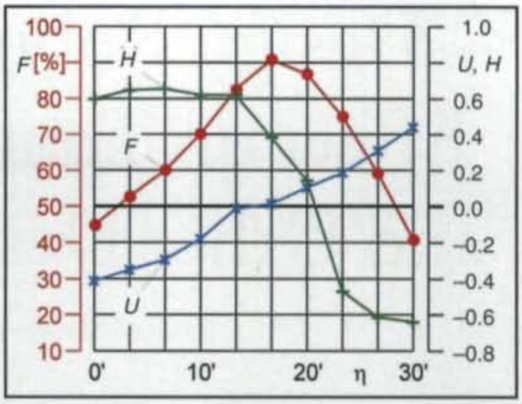


Figure 15—Results of the classification of the calculated contact patterns (example).

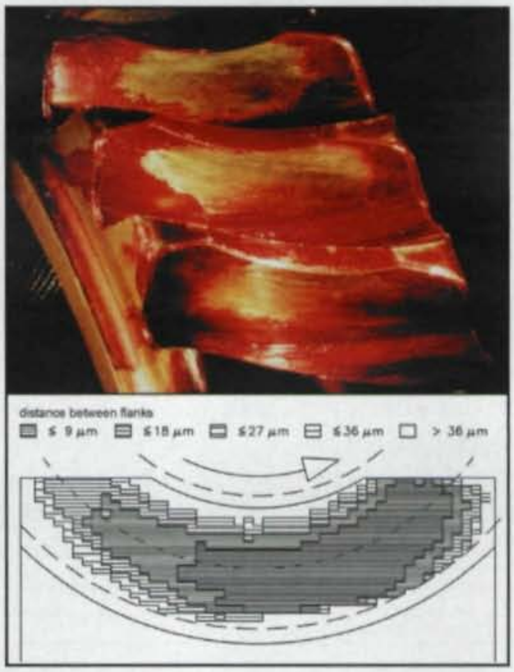


Figure 16—Comparison of a real contact pattern with the calculated contact pattern.

References

1. Deutsches Institut für Normung e.V. "DIN 3996—Tragfähigkeitsberechnung von Zylinder-Schneckengetrieben mit Achsenwinkel $\Sigma = 90°$," Beuth Verlag, Berlin, Germany, 1998.
2. International Organization of Standardization, "ISO CD 14521—Load Capacity Calculation of Worm Gears," 2001.
3. Lutz, M. "Berechnungsprogramm zur Ermittlung des Tragbildes zwischen Schnecke und Schneckenrad," FVA-Forschungsvorhaben No. 252, *Forschungsvereinigung Antriebstechnik e.V.*, Issue 500, Frankfurt, Germany, 1996.
4. Deutsches Institut für Normung e.V., "DIN 3975—Begriffe und Bestimmungsgrößen für Zylinderschneckengetriebe mit sich rechtwinklig kreuzenden Achsen," Beuth Verlag, Berlin, Germany, 2001.
5. Niemann, G. and H. Winter. *Maschinenelemente Band III*, Springer Verlag, Berlin, Germany, 1986.
6. Weber, C. and K. Banaschek. *Formänderung und Profilrücknahme bei gerad- und schrägverzahnten Rädern*, Vieweg-Verlag, Schriftenreihe Antriebstechnik 11, Braunschweig, Germany, 1955.

Design Robustness and its Effect on Transmission Error and Other Design Parameters

Donald R. Houser and Jonny Harianto

Abstract

Transmission errors, axial shuttling forces and friction result in bearing forces that serve as the major excitations of gear noise. This paper will use these factors as well as gear stresses and tribological factors to assist in obtaining optimal gear designs. The design basis comes from an actual application in which two different gear pairs were tested. One of the pairs was exceptionally noisy and the other exceptionally quiet, with the latter being insensitive to manufacturing variation.

Introduction

The reduction of gear noise has been a longstanding goal for automotive engineers who are seeking to improve the noise-vibration-and-harshness (NVH) performance of vehicles. Methods of reducing gear noise include attempting to reduce excitations at the mesh by minimizing dynamic forces due to transmission error or by reducing force transmissibility from the mesh to noise radiation surfaces. This paper will focus on obtaining gear designs that minimize these excitations and then evaluate the sensitivity of some of these designs to manufacturing variability.

Table 1—Gear Geometry for Coarse-Pitch Gear Pair.

Case	Coarse-Pitch	
Type of gears	Pinion	Gear
Number of teeth	27	32
Module	4.0	
Pressure angle (degrees)	20.0	
Helix angle (degrees)	14.0	
Active face width (mm)	43.0	
Center distance (mm)	123.68	
Outside diameter (mm)	121.65	142.29
Root diameter (mm)	101.94	123.41
Profile contact ratio	1.52	
Face contact ratio	0.83	
Total contact ratio	2.35	

Table 2—Gear Geometry for Fine-Pitch Gear Pair.

Case	Fine-Pitch	
Type of gears	Pinion	Gear
Number of teeth	30	35
Module	3.63	
Pressure angle (degrees)	20.0	
Helix angle (degrees)	16.0	
Active face width (mm)	43.0	
Center distance (mm)	123.68	
Outside diameter (mm)	122.10	143.13
Root diameter (mm)	102.21	122.94
Profile contact ratio	1.83	
Face contact ratio	1.04	
Total contact ratio	2.87	

The paper will first focus on the analysis of two gear designs, one that was exceptionally noisy and one that was exceptionally quiet. The transmission manufacturer attempted many modifications to the profile of the first design, given in Table 1. No matter what manufacturing variations were applied, though, the gear sets were very noisy. A second gear design, given in Table 2, with finer module was designed. When installed in the application, this design proved to be very insensitive to manufacturing errors. Literally all of the gears of this design had an acceptable noise characteristic. The designers were indeed fortunate to have come up with this improved design, but the greater issue was to learn what was different in the two designs so that this knowledge could be used to achieve future designs that are less sensitive to manufacturing errors. This paper will also present procedures that allow the designer to determine the robustness (sensitivity to manufacturing errors) of a given design.

Gear Noise Excitation Prediction

Transmission error, which results from both gear tooth deflections and manufacturing errors, has long been felt to be the main exciter of gear noise (Refs. 1–3). Providing tip and root relief to the profiles of the gear teeth usually compensates for the transmission error component caused by tooth deflection. However, there are often occasions when low transmission error gears are still unacceptably noisy. These occurrences have resulted in a rethinking of the total gear noise excitations by considering two additional force excitations, one due to the once-per-mesh-cycle *axial shuttling of the centroid of the gear tooth* force and the second due to time-varying friction forces (Refs. 4–5). Although the forces due to these three excitations must really be added as vectors, in this paper we will algebraically add the first harmonic of each of the individual forces as follows:

Sum of Forces = $SF + TEF + FF$

where,

SF	= Shuttling force
TEF	= Transmission error force
FF	= Friction force

The evaluation of each of these forces will use an analytical approach that predicts the load distribution along the lines of contact of the gear teeth (Refs. 6–7). This procedure accounts for tooth and shaft deflections, tooth profile shape, and mounting and misalignment errors of the gears.

Optimal Profile and Lead Modifications

An obvious goal in gear design is to come up with "optimal" profile and lead modifications that will minimize gear noise excitation yet still provide adequate load distribution and root and contact stresses. A problem with designing "optimal" modifications is that they are only truly optimal at one load. In many applications, the loads at which noise is a problem are only a small fraction of the peak load that the gear pair is designed for. For the gear design studied here, we chose to optimize the profiles and leads at 564 N-m of torque at the input shaft. This load is about one-half of the rated load of the gear set. It was hoped that the modifications would still be good at loads less than 564 N-m. A check of the excitation values was made at 40% of this load, at 226 N-m. In this case, we varied both the profile modification and the lead shape in order to minimize transmission error.

In order to come up with an optimum modification, we first assumed that the shape of the modification would be parabolic and then simultaneously varied the starting roll angle of the parabola and the parabola's amplitude until we minimized the transmission error. Figure 1 shows the results of running 400 simulations for the fine-pitch gear pair. The optimal modification starts at the center of the tooth and has a pinion and gear tip modification of 19.05 microns (750 μin.). The figure also shows that the transmission error changes less for increases in modification than for lesser values, hence telling the designer that it would be better to skew the tolerance to the positive side of the design value.

We next performed a similar variation in the lead direction and found it best to provide parabolic crowning only near the edges of the gear teeth. After some iteration of the lead and profile,

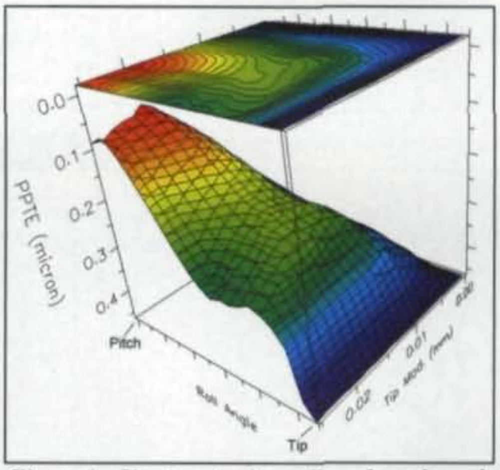


Figure 1—Varying starting roll angle and parabolic tip modification.

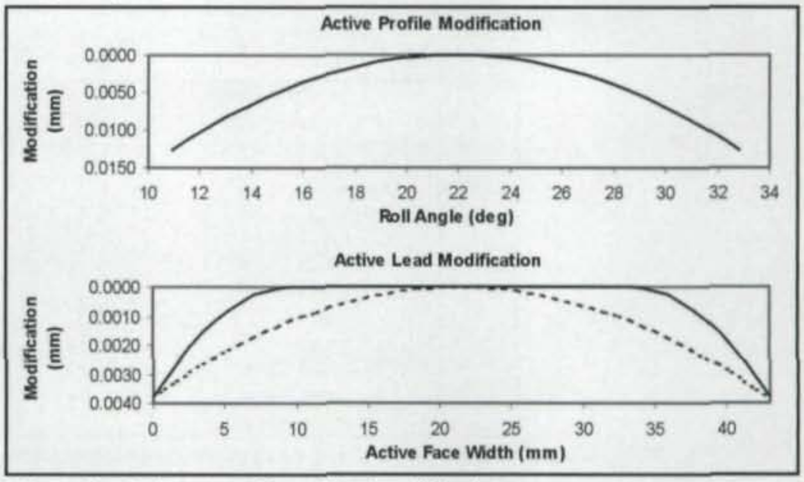


Figure 2—Optimum profile and lead modifications.

the modifications of Figure 2 were established as the optimum, and the resulting peak-to-peak transmission error (PPTE) for this modification was 0.043 microns (1.71 μin.). Although not shown, the optimal profile modification for the coarse-pitch gear pair was similar to that of the fine-pitch gear pair, but we found that a straight lead (no lead modification) gave the lowest transmission error. It should be pointed out that each of these modifications does not consider the effects of misalignment.

However, in order to simplify profile modeling, subsequent design analyses will use a circular profile modification of 12.7 microns and a circular lead modification of 3.8 microns. This lead modification is shown as the dashed line on the lead chart in Figure 2.

Force and Transmission Error Results

Figures 3 and 4 show the effect of load on predicted transmission error for the following three cases of both the coarse- and fine-pitch gear sets, respectively:

- Perfect involute teeth (Involute),
- Optimally modified teeth (Optimum), and

Dr. Donald R. Houser

is director of the Gear Dynamics and Gear Noise Research Laboratory (Gear Lab), located at The Ohio State University in Columbus, OH. Gear Lab is an industrially sponsored research consortium with 30 participating companies. Houser also is a professor in the university's mechanical engineering department. He teaches and researches in the areas of gear design, gear manufacture, mechanical measurements, automatic controls, system dynamics and machinery vibrations.

Jonny Harianto

is a research engineer in The Ohio State University's mechanical engineering department. He holds a master's degree in mechanical engineering and specializes in gearing systems. Also, he develops and improves gear analysis software for Gear Lab.

• Teeth with the optimal profile and a circular lead modification (Design).

For each set of gears, the optimal profile has its lowest transmission error near the design torque of 564 N-m, and transmission errors are lower than the perfect involute's from about one-half of the design torque up to the maximum torque. The transmission errors for the optimized fine-pitch gear pair are much lower than the values predicted for the coarse-pitch gear pair. The circular lead modification provides slightly higher values than the edge modification, but not so much as to change the strategy of using it in all gear designs.

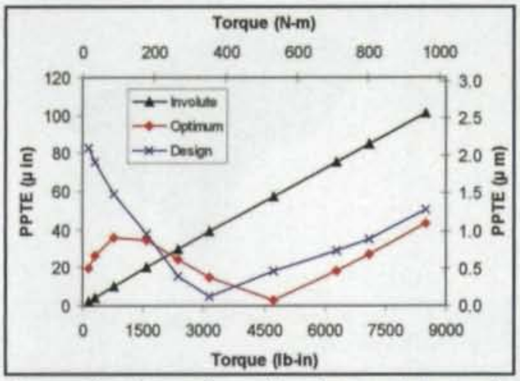


Figure. 3—Comparison of peak-to-peak transmission error for coarse-pitch gear pair.

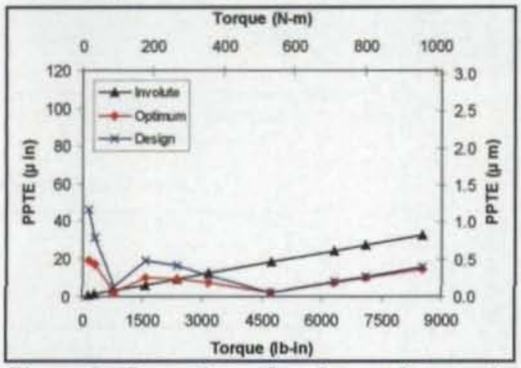


Figure 4—Comparison of peak-to-peak transmission error for fine-pitch gear pair.

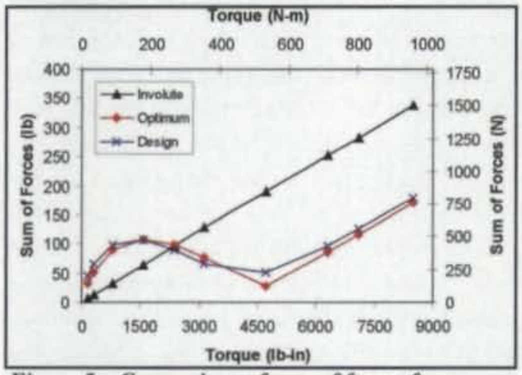


Figure 5—Comparison of sum of forces for coarse-pitch gear pair.

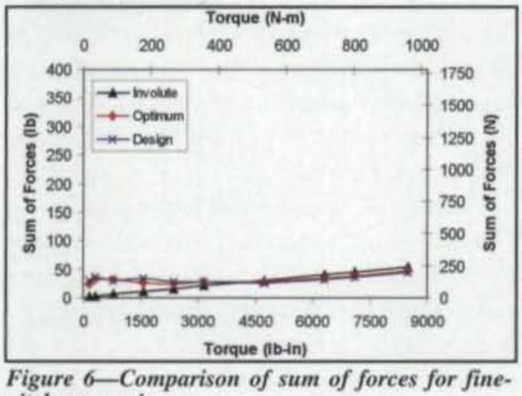


Figure 6—Comparison of sum of forces for fine-pitch gear pair.

Figures 5 and 6 show similar plots for the predicted sum of forces. For each of the cases, the forces predicted for the fine-pitch gear pair are much less than for the coarse-pitch gear pair. This would indicate that the mathematical model's results correlate well with the experimental noise results of the gear manufacturer.

Optimal Designs

In an effort to further improve the design, a variation of the procedure from Houser, et al. (Ref. 8) was used to survey a large number of designs in an effort to obtain designs that have both low transmission error and low sum of forces, but also have favorable stresses, efficiency, lube film thickness and flash temperature. In addition, we wished to check out the best of these designs for their sensitivity to manufacturing errors.

The first step in the procedure was to select ranges of variables to be studied and then run a huge number of design cases within the selected design space. In this instance, a two-stage evaluation was performed where the first range of variables was developed around the original designs (see Table 3). Center distance and face width were kept at the original values for each design. Approximately 20,000 designs were evaluated, with the main conclusion being that the next set of designs should have higher tooth numbers and the possibility of longer tooth profiles. The variables of the second iteration that were evaluated in much more detail are given in Table 4. In this case, close to 100,000 designs were evaluated.

A plotting routine has been developed that allows the user to first plot any design or output variable versus any other variable. Currently, we have 40 variables available for plotting. Figure 7 shows peak-to-peak transmission error plotted vs. sum of forces results for about 22,000 designs. Although the two variables do appear to be related to one another, the low transmission error cases are not at the lowest sum of forces and vice versa. This implies that there is no one best design based on these two parameters, and compromises must be made in selecting the "best" design. Figure 8 shows the same plot with a portion of the data set being selected as favorable designs. In this case, the limits were 0.1 microns for transmission error and 67 N for sum of forces.

In subsequent plots, the 386 selected designs out of a total of 22,903 will be highlighted. Of the 22,903 designs, roughly 12% of the designs have transmission errors less than 0.1 microns while another 11% have sum of forces less than the cut-off value of 67 N. It is interesting that only 1.7% of the designs simultaneously satisfy both criteria.

One might ask the question: Which design variables profoundly affect either transmission error or sum of forces? In the appropriate literature, many authors have advocated using integer face contact ratios to minimize transmission error. Although this tends to be true for gears with perfect involutes, Figure 9 shows only a slight effect from face contact ratio, with the few really low transmission error designs occurring at face contact ratios between 1.05 and 1.20. In general, we have found that once profile and lead modifications have been applied to gear teeth, the face contact ratio plays only a secondary role in minimizing transmission error.

However, when we check the effect of total contact ratio on transmission error (shown in Fig. 10), we see that minimum transmission error values occur for contact ratios near 2.9. Apparently, none of these low transmission error designs also have low sum of forces, since no points in the low transmission error region are highlighted. Total contact ratio also has a pronounced effect on sum of forces, but the region of lowest sum of forces has shifted slightly up to face contact ratios between 2.9 to 3.2, as shown in Figure 11.

Our selection process did capture some of the very best sum-of-forces designs. Had we wanted to capture more of the best transmission error designs, we would have had to change our selection criteria by increasing the level of the sum of forces used in the selection process. From Figure 12, we see that there is also a region for minimum sum of forces around the profile contact ratio of 1.65.

One should be cautioned on two fronts before making global conclusions regarding this information:

1.) Even at the contact ratios that give minimum values, there are still many more designs that give unacceptable values, so simply selecting a total contact ratio near 2.9 or 3.0 does not guarantee low transmission error or sum of forces.

2.) The actual values of contact ratio that minimize the noise excitations may change when we select new design variable ranges, so each of these "optimum" ranges may be unique to a given design specification.

Table 3—First Set of Design Parameters.

	Levels	Range of Variables
Gear ratio		1.078–1.192
Pinion teeth		25–32
Pressure angle (degrees)	3	18–22
Helix angle (degrees)	5	12–20
Center distance ratio	2	0.968–1.036
Hob length x module	2	2.35–2.55
Tip relief (mm)	1	0.0127

Table 4—Second Set of Design Parameters.

	Levels	Range of Variables
Gear ratio		1.078–1.192
Pinion teeth		29–33
Pressure angle (degrees)	3	18–22
Helix angle (degrees)	4	16–22
Center distance ratio	5	0.968–1.036
Hob length x module	3	2.35–2.55
Tip relief (mm)	3	0.0114–0.0140

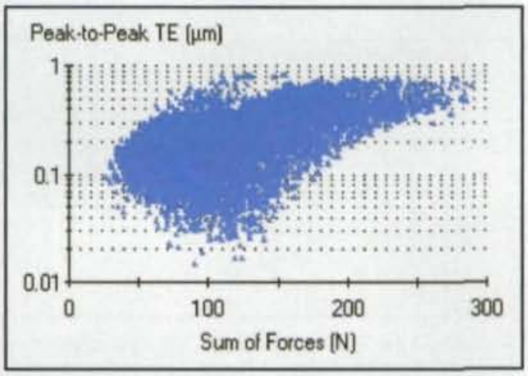


Figure 7—Transmission error vs. sum of force results for second design parameters.

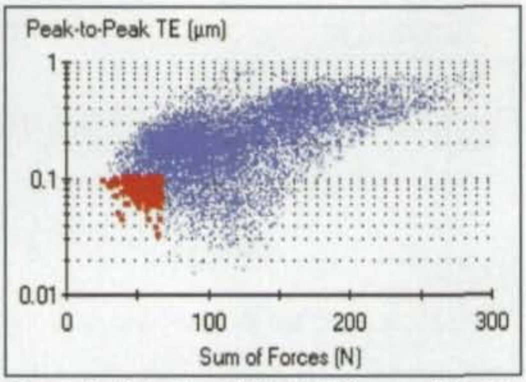


Figure 8—Selection of "Best" 386 runs of the second design parameters.

Figure 13 shows that the contact stresses seem to be highly dependent on the profile contact ratio, again with a best range appearing in the 1.5–1.8 contact ratio range. The stresses are relatively low because we did our design evaluation at roughly 50% of design torque, but we feel that there would be few changes in the trends if we were to re-evaluate the design at the design torque. Also of interest is the fact that most of our selected designs have relatively low contact stresses.

This is not the case for root stresses that are shown in Figure 14, where we see that pinion bending stresses of the selected designs are pretty much in the middle region of the stress range. However, if minimum stress is an important design criterion, it could have been used in the selection process, but some compromise regarding transmission error and other factors would have had to be made in order to obtain truly low stresses.

Finally, the last response variable we shall look at is flash temperature, which is shown plotted vs. profile contact ratio in Figure 15. Again our selected designs seem to have relatively low flash temperatures. The lowest flash temperatures tend to occur for gears with the lowest profile contact ratio, but the highest flash temperatures also occurred for these contact ratios. The range of flash temperatures of the designs tends to become narrower as profile contact ratio increases.

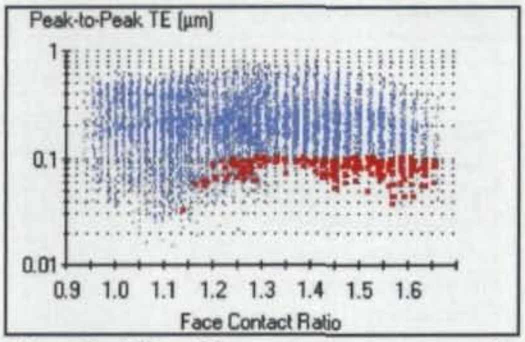


Figure 9—Effect of face contact ratio on transmission error for the second design parameters.

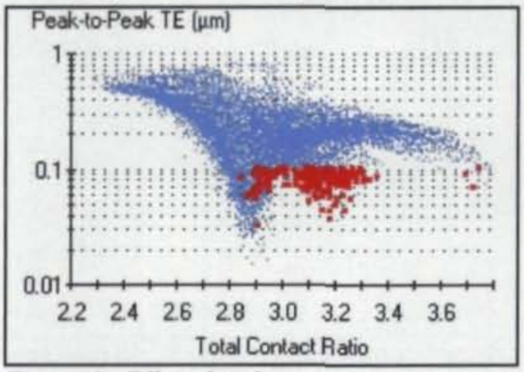


Figure 10—Effect of total contact ratio on transmission error for the second set of design parameters.

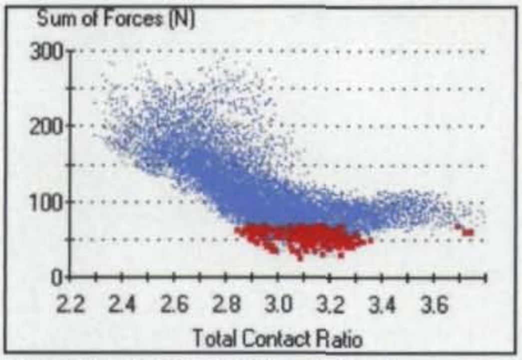


Figure 11—Effect of total contact ratio on sum of forces for the second set of design parameters.

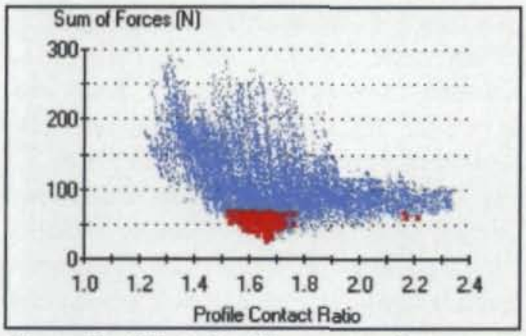


Figure 12—Effect of profile contact ratio on sum of forces for the second set of design parameters.

Manufacturing Robustness

The manufacturer's experimental evidence indicated that the coarse-pitch gear pair tended to be noisy for all manufactured tooth profiles and the fine-pitch gear pair tended to be insensitive to manufacturing variation. Consequently, we set out to perform simulations that emulated the effects of manufacturing variation for both the manufactured geometries and the "best" of the selected geometries of the previous figures. Now, however, we expanded our base of designs by repeating the runs of the previous figures and by expanding our range of tip relief amplitudes to three levels.

A special analysis procedure was developed where the following errors were deemed to be simple representations of manufacturing errors that might happen in practice:

- Profile slope error (often called pressure angle error),
- Profile curvature error (crown-type error),
- Lead slope error (also incorporates misalignment effects), and
- Lead curvature error (lead crown error).

The load distribution simulation program has a module that allows one to input the standard deviation for each of the errors. Then the errors are randomly sampled from a normal distribution. The user either supplies standard deviations or may enter the AGMA values (Ref. 9). We feel this procedure is very representative of gears that are randomly selected from production. In each case, 50 computer simulations were made using randomly selected profiles and leads with each of the four manufacturing variations having a standard deviation of 2.5 microns.

Figure 16 shows a plot of the transmission error robustness results, and Figure 17 shows a similar plot for sum of forces for the 30/35 tooth pair of Table 2 using circular profile and circular lead. General conclusions from Figures 16 and 17 are that the error band tends to be narrowest near the torque used to optimize the transmission error. Sometimes using manufacturing deviations actually results in a slightly better design because the new types of profile modifications might stumble onto a more optimum type of modification shape. For instance, providing a slight amount of pressure angle error might improve one of the response variables, such as transmission error or contact stress. Similar plots were made for stresses, flash temperature, efficiency and film thickness in order to determine the effects of manufacturing variability on the many response variables of a given design.

Table 5 shows a summary of the robustness analysis for 10 different designs as follows:

a) The two manufacturer's designs given in Tables 1 and 2, respectively;

b) The lowest transmission error design and the lowest sum of forces design;

c) Two similar designs that were proclaimed "best," based not only on noise excitation, but also including flash temperature, efficiency, lube film thickness and root and contact stresses;

d) The best high contact ratio design; and

e) Three other "good" designs.

There is a lot of data in the table, so only the highlights will be discussed. The first 12 rows present general design information. CD Enlargement indicates whether the design is operating on standard centers (1.0), enlarged centers (> 1.0) or contracted centers (< 1.0). Tool height indicates the length of the rack used to create the involute tooth. Full radius cutters are used in evaluating the root stresses.

TE at 564 N-m indicates the transmission error at the design torque. Note that the "new" design has much lower transmission error than the original design, but that most of the additional designs have similar or lower transmission errors. The robust average is the mean value of transmission error for the 50 robustness runs when evaluated at the design load. It is interesting to note that the "new" design is the best of the group in this regard. The robust maximum is the worst case value coming from the 50 runs. Again the "new" design seems exceptional in this regard.

The next three rows show similar results for the sum of forces. However, now some of the other designs exhibit much better characteristics than the "new" design.

The next three rows are the maximum stress values recorded for the worst of the 50 test cases. Here, the "best" designs have very low contact stresses and the high contact ratio set has extremely high contact stress for its worst case errors.

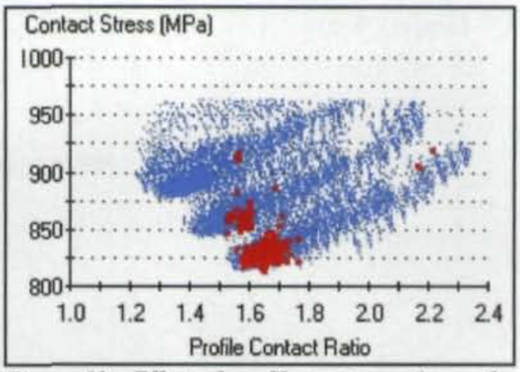


Figure 13—Effect of profile contact ratio on the contact stresses for the second set of design parameters.

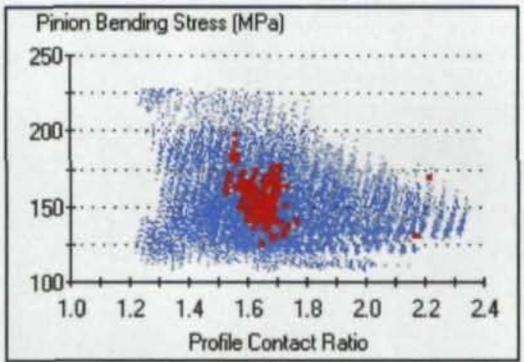


Figure 14—Effect of profile contact ratio on the pinion bending stresses for the second set of design parameters.

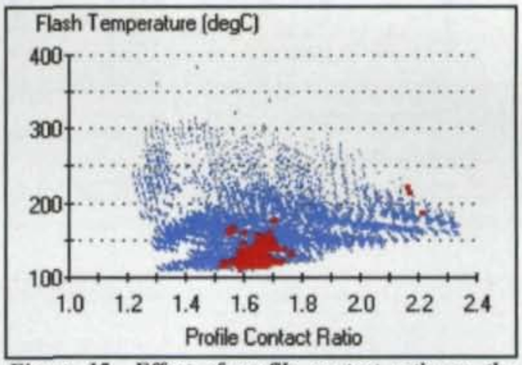


Figure 15—Effect of profile contact ratio on the flash temperature for the second set of design parameters.

The next three rows are all factors related to sliding velocity, namely: flash temperature, film thickness and percent of energy loss. Each is an average value of the 50 test cases, but data is also available for "maximum" values. One of the reasons the two "best" designs were selected is that they are equivalent to most other designs in terms of noise excitation and stresses, but have very good levels of these three values. The high contact ratio set does not have acceptable levels of these variables.

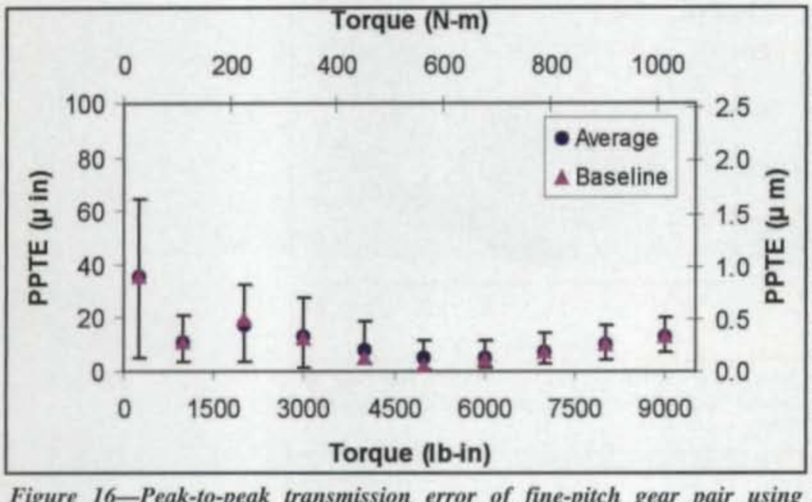


Figure 16—Peak-to-peak transmission error of fine-pitch gear pair using robustness analysis to circular profile and circular lead.

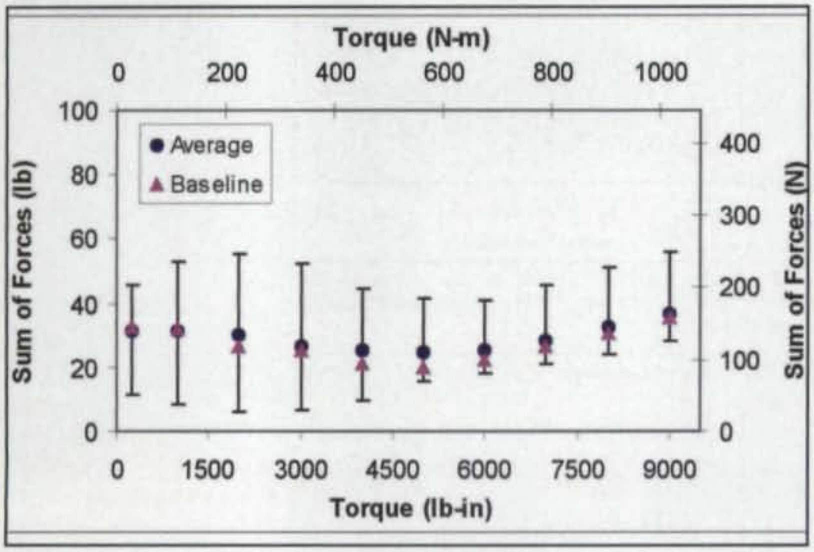


Figure 17—Sum of forces of fine-pitch gear pair using robustness analysis to circular profile and circular lead.

Finally, the last two rows show the average values of transmission error and sum of forces at a lower load of 226 N-m, which is more typical of the noisy application load for these gears. Here, the high contact ratio set seems to shine, having by far the lowest mean values of transmission error and sum of forces. It is interesting that our "best" designs are not so good in these features and the "new" design is quite good.

Summary

In this paper, we have developed a procedure that allows the incorporation of manufacturing variability into the gear design process. Although our main focus has been on gear noise excitations, the procedure also allows one to determine the effects of manufacturing variability on other design responses, such as root and contact stresses and various scoring indices. Two examples were used to demonstrate the procedure: one gear set that has been known to be noisy and a second that was known to be quiet. In addition, a procedure has been presented for evaluating numerous different gear geometries for the same application. The predictions show that many designs with far differing geometries can provide "good" designs. How one weighs the many factors used to assess a design will dictate which of the many good designs might be selected.

Acknowledgments

The authors would like to thank the sponsors of the Gear Dynamics and Gear Noise Research Laboratory (Gear Lab) at The Ohio State University for their support of the development

Table 5—Summary of Robustness Analysis of 10 Different Designs.

		Original	"New" Design	Low TE	Low Sum of Forces	"Best" Design	Similar Design	High Contact Ratio	"Good" Design	"Good" Design	"Good" Design
	Identifier	27/30	30/35	43017	61685	418355	618355	620425	511385	74429	52515
	Pinion Teeth	27	30	29	29	33	33	33	32	32	32
	Gear Teeth	32	35	34	33	37	37	39	35	36	33
	Pressure Angle (degrees)	20	20.7	20	18	20	20	18	18	20	20
	Helix Angle (degrees)	14	16	18	22	22	22	22	18	16	22
	Module (mm)	4.00	3.63	3.79	3.60	3.18	3.18	3.26	3.42	3.42	3.69
	Profile Contact Ratio	1.52	1.83	1.77	1.66	1.62	1.62	2.19	1.74	1.81	1.63
	Face Contact Ratio	0.83	1.04	1.12	1.42	1.61	1.61	1.57	1.24	1.08	1.39
	Total Contact Ratio	2.35	2.87	2.89	3.08	3.23	3.23	3.76	2.98	2.89	3.02
	CD Enlargement	1.017	1.008	0.987	1.028	1.028	1.030	0.976	1.027	1.001	1.001
	Tool Height	2.41	2.76	2.55	2.75	2.75	2.75	2.75	2.75	2.75	2.75
	Tip Relief (microns)	12.70	12.70	12.70	11.43	12.70	11.43	11.43	12.70	12.70	12.70
564 N-m	PPTE (μm)	0.603	0.054	0.015	0.124	0.044	0.066	0.020	0.060	0.028	0.092
	TE Rob., Avg. (μm)	0.67	0.13	0.15	0.18	0.14	0.16	0.13	0.21	0.13	0.22
	TE Rob., Max. (μm)	1.26	0.29	0.40	0.45	0.35	0.38	0.40	0.57	0.32	0.60
	Sum of Forces (N)	263.3	88.8	90.1	26.1	38.9	32.9	53.4	39.0	77.9	33.2
	Sum of Forces Rob., Avg. (N)	299.9	108.6	121.9	59.0	73.9	68.8	74.9	75.5	102.7	71.6
	Sum of Forces Rob., Max. (N)	637.4	185.6	207.8	168.5	172.9	169.7	158.5	171.6	175.8	181.0
	Max. Contact Stress (MPa)	967	1,014	1,109	964	883	873	1,504	902	908	926
	Max. Pinion Stress (MPa)	203	189	172	180	180	178	224	164	174	182
	Max. Gear Stress (MPa)	210	175	183	227	189	212	221	172	175	182
	Avg. Film Thickness (μm)	0.26	0.24	0.17	0.22	0.37	0.36	0.09	0.32	0.26	0.29
	Avg. Flash Temperature (°C)	151	159	176	176	128	133	223	132	146	151
	Avg. Percent Loss	0.82	0.84	0.79	0.88	0.71	0.71	0.79	0.88	0.80	0.75
226 N-m	TE Rob., Avg. (μm)	0.52	0.44	0.50	0.48	0.49	0.41	0.15	0.59	0.47	0.66
	Sum of Forces Rob., Avg. (N)	330.2	133.2	130.9	120.2	163.1	146.2	59.3	129.2	121.3	135.7

of the modeling procedures used for the gear analyses presented in this paper. ○

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

References

1. Niemann, G. and J. Baethge. "Errors in Action, Tooth Flexibility, and Noise of Spur Gearing," *VDI Journal*, Vol. 112, No. 4, 1970, pp. 205–214.
2. Kubo, A., T. Nonaka, N. Kato, S. Shogo, and T. Ohmori. "Representative Form Accuracy of Gear Tooth Flanks on the Prediction of Vibration and Noise of Power Transmission," AGMA Paper 92FTM9, 1992.
3. Houser, D. "Gear Noise," Chapter 14 in *Dudley's Gear Handbook*, ed. D. Townsend, McGraw-Hill, New York, NY, 1992.
4. Borner, J. and D. Houser. "Influence of Friction and Bending Moments as Gear Noise Excitations," SAE Paper 961816, *Proceedings of the International Off-Highway and Powerplant Congress and Exposition*, Indianapolis, IN, August 26–28, 1996.
5. Hochmann, D. and D. Houser. "Friction Forces as a Dynamic Excitation Source in Involute Spur and Helical Gearing," ASME Paper DETC2000/PTG-14429, *Proceedings of the ASME Design Engineering Technical Conferences*, Baltimore, MD, September 10–13, 2000.
6. Conry, T. and A. Seireg. "A Mathematical Programming Technique for the Evaluation of Load Distribution and Optimal Modifications for Gear Systems," *Journal of Engineering For Industry, Transactions*, ASME, Vol. 95, No. 4, November 1973, pp. 1115-1123.
7. Houser, D. "LDP Program Manual," The Ohio State University, August 1993.
8. Houser, D., J. Harianto, B. Chandrasekaran, J. Josephson, and N. Iyer. "A Multi-Variable Approach to Determine the "Best" Gear Design," ASME Paper DETC2000/PTG-14362, *Proceedings of the ASME Design Engineering Technical Conferences*, Baltimore, MD, September 10–13, 2000.
9. American Gear Manufacturers Association, *AGMA Gear Handbook*, American Gear Manufacturers Association, Alexandria, VA, March 1980.

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- **Rate this article**
- **Request more information**
- **Contact the authors or organization mentioned**
- **Make a suggestion**

Or call (847) 437-6604 to talk to one of our editors!

Heat Treat Process and Material Selection for High Performance Gears

Gerald J. Wolf

Figure 1—An etched section of a gear that has been case hardened by progressive induction hardening.

Figure 2—A load of large gears are transferred into a furnace for carburizing.

Figure 3—A load of large pinions are transferred into a furnace for carburizing.

Figure 4—Carburized gears are transferred from the heating chamber into the quench section for hardening.

Introduction

The selection of the heat treat process and the congruent material required for high performance gears can become very involved.

A wide variety of options are available, and many issues must be considered. The application requirements must take into consideration the service conditions in which the gears will operate, the speeds and power to be transmitted, and the design life. This data will in turn define the size and shape of the gears and, along with the load carrying capacity required and the volume to be produced, allow material selection in conjunction with the choice of heat treatment to achieve the required mechanical properties.

The improved mechanical properties achieved by case hardening dictate that virtually all high performance gears are processed in this manner. Still, we will include through hardening for core treating in our discussion because it is required to achieve the desired properties when induction hardening and nitriding.

Through Hardening, General Requirements for Core Treating

Core treating prior to induction hardening and nitriding is used to obtain both the desired microstructure for subsequent processing and the required mechanical properties in the finished part. In most cases, core treating is accomplished by hardening and tempering the parts prior to gear cutting. A hardness range of 28–32 HRC (269–302 HBN) is commonly specified in order to obtain good mechanical properties without severely sacrificing machinability.

However, higher core hardnesses are used where increased strength levels are necessary. The material selection and processing sequence must take into consideration the mass of the part and the hardenability of the material, as these influence the depth of hardening in the core treating process. Heavier sectioned parts and coarser pitches require higher hardenability materials or premachining for proper core treating response. Good hardening response is also needed when core treating prior to nitriding because the tempering temperature must be at least 50°F higher than the nitriding temperature, or typically at least 1,050°F, to insure dimensional stability during nitriding.

Case Hardening Processes for High Performance Gears

High performance gears generally require case hardening to produce the required mechanical properties. Contour and tooth-to-tooth progressive induction hardening, carburizing and hardening, and nitriding are therefore preferred while processes such as spin flame hardening and conventional single-shot induction hardening are not capable of producing the required properties.

Each of the above processes is unique and has its own set of pluses and minuses relative to each other. Contour and tooth-to-tooth induction hardening involve heating and hardening only a small portion of the gear, so size change and distortion are relatively small, especially when compared with those produced during hardening of a carburized part. Contour induction hardening is suitable only for small- to medium-sized gears in high volume applications.

In contrast, tooth-to-tooth induction hardening is cost effective only for low volume applications. Also, not all parts

Figure 5—A large carburized pinion is transferred to the oil quench tank for hardening.

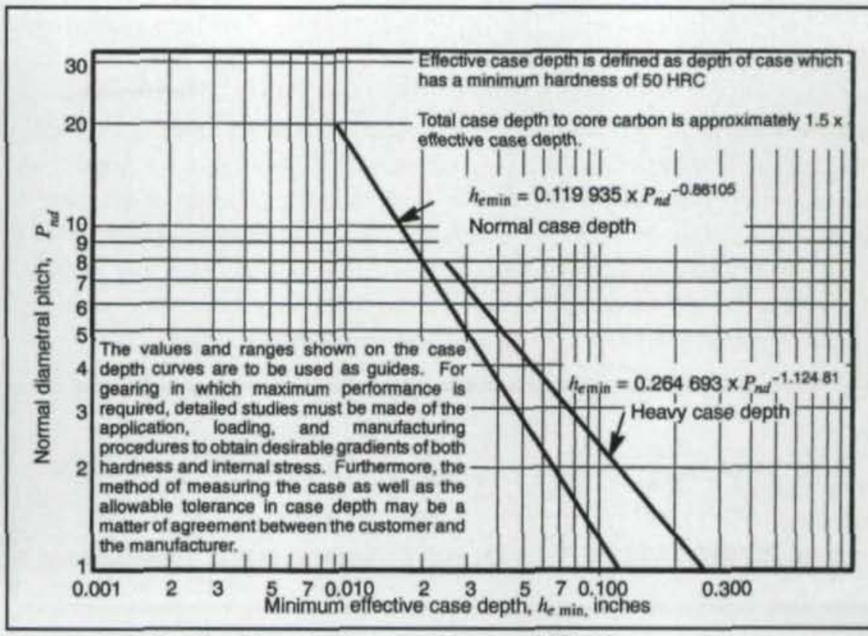


Figure 6—Carburized Case Depth Requirements. (Courtesy of AGMA.)

can be hardened by either type of induction hardening due to their size and shape, and neither type of induction hardened gears has quite as high a load carrying capability as carburized and hardened gears.

For some applications, nitriding is the heat treatment of choice because very good mechanical properties are achieved. Also, it produces distortion and size change so small and predictable that generally hard finishing is not required. Nitriding, however, involves long cycle times and is therefore very costly when trying to obtain deeper case depths required for coarser pitches.

But where the best performance is required, carburizing and hardening is usually relied upon because it produces excellent mechanical properties and is suitable for low and high volume applications. Size change and distortion can be serious problems, but they can be dealt with.

Tooth-to-Tooth Progressive Induction Hardening

Tooth-to-tooth induction hardening is often used for low to medium volume requirements when the size or design of the part allows this process to be used. In tooth-to-tooth induction hardening, an inductor with a profile matching the shape of the tooth space is traversed through each tooth space. Controlling the power input and scanning speed progressively hardens the active profile and root area of each adjacent tooth face to the desired depth. Since the size of the air gap between the inductor and the tooth surface is critical, a custom inductor is often required for each gear design. For fine- to medium-pitch gears, this process is generally performed with the part submerged in a quenchant.

As each tooth space must be individually scanned, the floor-to-floor time for hardening a gear in this manner is relatively long when compared to contour hardening where all the teeth are hardened at one time. Offsetting this is the versatility and size range capability of the tooth-to-tooth process. Gears from 10–1 DP are routinely hardened in this manner. To determine the case depth and pattern produced, it is necessary to cross-section several teeth from a hardened gear in contrast to being able to check a test piece when carburizing or nitriding (see Fig. 1).

A core treated, medium-carbon alloy steel, such as 4142, 4150 or 4340, is generally specified for most applications with the choice dictated by the diametral pitch and size of the gear. The larger the part and coarser the pitch, the higher the hardenability required for good hardening response. In higher volume applications, lower cost grades, such as 15B41, are utilized where the material's hardenability is matched to the specific parts requirements and large enough quantities are used so that availability is not an issue.

Contour Induction Hardening

For higher volume applications of small- to medium-sized gears in the 8–4 DP range, contour induction hardening is a viable option and can compete with carburized gearing. Contour induction hardening is similar in appearance to the classic single shot, spin induction process where the gear is rotated in an induction coil to heat the teeth and then quenched in place. In most cases, however, this through hardens rather than case hardens the teeth. The contour induction hardening process has been developed to produce a heat pattern that

Gerald J. Wolf

is a gear/heat treat consultant who specializes in gear failure analysis and materials. A metallurgical engineer, Wolf was previously president of The Cincinnati Steel Treating Co. of Cincinnati, OH. He has more than 30 years of experience in the heat treating industry. He spent the last 25 of those years at Cincinnati Steel Treating, specializing in heat treating gears.

follows the profile of the teeth and produces a true case hardened tooth. This is generally accomplished through the use of high intensity, multistage heating where the teeth are first preheated using a low frequency, and then the tooth surfaces are rapidly heated with a burst of high power, high frequency energy.

The nature of contour induction hardening requires that the tooling, process parameters and equipment be tailored to each application. Multiple, high-powered power supplies with very sophisticated control systems are necessary. This makes the process capital-intensive and seldom practical for lower volume applications, but justifiable for higher volumes. Another advantage it has over all other gear hardening processes is that it lends itself to integration into manufacturing cells, especially when combined with induction tempering.

Carburizing and Hardening

Of the case hardening processes, carburizing and hardening is the most widely used process today for heat treating high performance gears. It is adaptable to both high and low volume requirements as well as to both large and small parts (see Figs. 2 and 3). And, with the proper choice of materials and processing techniques, it will produce the highest allowable contact and bending stress ratings obtainable.

To achieve the desired results, it is important that the material selection be in accord with the nature and size of the gear as well as its service requirements. All carburizing grades are low carbon alloy steels, but coarser pitch, heavier sections and more severe service necessitate using higher hardenability, higher alloy, and more expensive materials. Common grades in increasing hardenability are: 4118, 8620, 8822, 4320, 4820, 9310 and 17CrNiMo6. These materials are also produced in different quality levels to comply with the different AGMA grade requirements, depending on service requirements.

It is recommended that, prior to machining, the lower alloy grades be normalized and the higher alloy grades be normalized and tempered or normalized, quenched and tempered to obtain improved machinability as well as more reproducible size change and distortion results during subsequent carburizing and hardening.

From a manufacturing standpoint, the

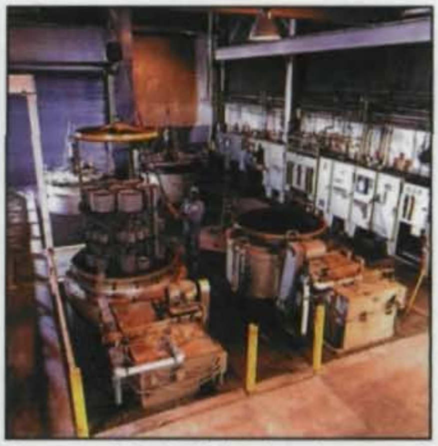


Figure 7—A load of diesel engine gears is removed from the furnace after nitriding.

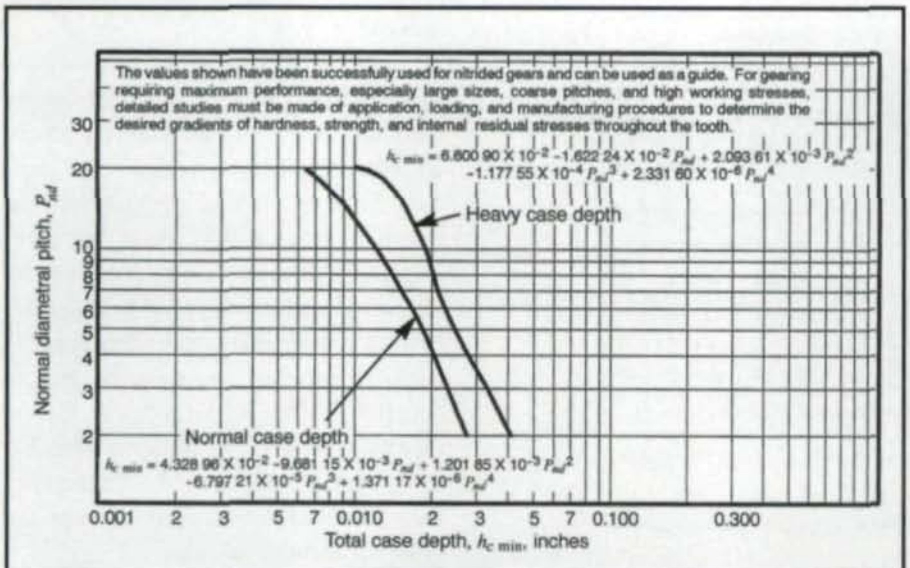


Figure 8—Nitrided Case Depth Requirements. (Courtesy of AGMA.)

most serious problem with carburized gearing is the size change and distortion that occurs during hardening. This is very design-dependent. Gears that have nonsymmetrical cross sections or are heavily relieved for mass reduction will present more problems than relatively solid symmetrical parts.

While proper fixturing must be employed during carburizing to minimize creep-induced distortion, part orientation during hardening is most critical in order to obtain a uniform quench. Quenching in hot (250–350°F) oil, marquenching and high-pressure gas quenching can often be employed to reduce the size change and distortion but, since they are slower quenches, it is often necessary to use a higher hardenability grade than necessary with conventional oil quenching (see Figs. 4 and 5).

When the design of the gear is such that excessive distortion is produced during free quenching, techniques such as plug and/or press quenching should be used. For size and shape control of thin-walled gears and internal splines, plug quenching is recommended. However, where the movement of the whole gear must be controlled, press quenching is necessary. This requires the availability of a suitably sized quench press and generally custom tooling for holding the part during the quenching process. Since both of these processes require handling each part individually, they often significantly increase processing costs.

The depth of the carburized case must be in accord with the diametral pitch and service requirements of the gear. Coarser pitches and more severe applications require deeper cases. Figure 6 shows the AGMA chart for determining the effective case depth to be used for a given diametral pitch. The effective case depth is typically defined as the depth at which the hardness has decreased to 50 HRC and is measured on a representative test sample or a sectioned part.

Nitriding

Nitriding is another case hardening process that has been gaining popularity for use on high performance gearing due to advances that have been made in both gas and ion nitriding. Nitriding produces a hard surface with excellent wear properties, improved corrosion resistance and very good high-cycle fatigue properties. This is a rather unique process in that it is performed at a relatively low temperature, in the 950–1,050°F range, and quenching is not involved. Since the base material does not go through any structural change, size change and distortion are very low, and rarely is any finishing necessary after nitriding (see Fig. 7). The processing time required, however, is relatively long, especially for coarser diametral pitches where deeper case depths are

Table 1—Allowable Design Stress for Steel Gears.[1] *								
Heat Treatment	Material	Minimum Surface Hardness	Contact Stress (KSI)			Bending Stress[5] (KSI)		
			GR1	GR2	GR3	GR1	GR2	GR3
Through Hardened	4140/4340	32 HRC	127	140		36	48	
		43 HRC	160	175		43	58	
Induction Hardened[2]	1045/4340	54 HRC	175	195		45	55	
Carburized and Hardened	8620/9310	58 HRC[3]	180	225	275	55	65	75
Nitrided[5]	4140/4340	84.5 HR15N	155	168	180	37	50	
	Nitralloy 135M	90 HR15N	170	183	195	38	50	
	31CrMoV9[4]	90 HR15N	176	196	216	41	52	60[5]

[1] Table compiles data from various tables and graphs in ANSI/AGMA 2001-C95.
[2] Tooth-to-tooth and contour induction hardening, with ANSI/AGMA type A pattern.
[3] Minimum surface hardness of 55 HRC is acceptable for GR1; all other grades require a *minimum of 58 HRC.*
[4] Basically the same as 2.5%Cr without Al.
[5] The seven bending stress values for nitrided materials are based on a core hardness of 300 HB.

needed.

Three different types of medium-carbon alloy steels are used for nitrided gears, depending on the service requirements. In all cases, core treating is employed to produce both the desired strength and the structure needed for good nitriding response. When the service requirements are not too severe, 4140 or 4340 is typically specified. For more demanding applications, it is necessary to use a grade, such as Nitralloy 135M or 31CrMoV9, that is especially designed for nitriding. Figure 8 shows the AGMA chart for determining the total nitrided case depth required for different diametral pitches.

Summary

The purpose of this paper was to give a general overview of the different heat treat processes and materials used in high performance gearing and thus help guide the designer and process engineer in making the best selection for a given job. Table 1 outlines values of contact and bending allowable stresses for each of the aforementioned hardening processes. For full data and details, see ANSI/AGMA 2001-C95. ○

This article was given as a presentation at the Heat Treating & Hardening of Gears 2002 Technical Program, in Nashville, TN, on January 29, 2002. That program was organized by the Society of Manufacturing Engineers.

***Table 1 is a revision of material presented in ANSI/AGMA 2001-C95. The original material is printed with permission of the copyright holder, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314. Statements presented are those of the author and may not represent the position of the American Gear Manufacturers Association.**

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- **Rate this article**
- **Request more information**
- **Contact the author or organization mentioned**
- **Make a suggestion**

Or call (847) 437-6604 to talk to one of our editors!

Corrections

In *Gear Technology*'s article titled "Direct Gear Design for Spur and Helical Involute Gears" (September/October 2002), references 5, 6 and 8 were printed incorrectly. The correct references are as follows:

5. Kleiss, R.E., A.L. Kapelevich, and N.J. Kleiss. *New Opportunities with Molded Gears*, AGMA Fall Technical Meeting, Detroit, October 3–5, 2001, (01FTM9).

6. Vulgakov, E.B., and A.L. Kapelevich. "Expanding the Range of Involute Helical Gearing," *Vestnik Mashinostroeniya*, 1982, Issue 3, pp. 12–14 (in Russian). Translated to English, *Soviet Engineering Research*, Vol. 2, Issue 3, 1982, pp. 8–9.

8. Litvin, F.L., Q. Lian, and A.L. Kapelevich. "Asymmetric modified gear drives: reduction of noise, localization of contact, simulation of meshing and stress analysis," *Computer Methods in Applied Mechanics and Engineering*, 2000, Issue 188, pp. 363–390.

We would also like to make the following corrections regarding *Gear Technology*'s January/February 2003 issue:

1.) In the article "Spiral Bevel Gear Development: Eliminating Trial and Error with Computer Technology," Joseph L. Arvin's author biography says he started in the gear industry as a machinist at "Indiana Gear of Plymouth, IN, in 1959." The biography should say "Indiana Gear of Indianapolis, IN, in 1959."

2.) In the article "A Man and His Mania for Antique Machines," the Sloan and Chace Mfg. Co. was formed in 1872 and Sloan's name was A.K. Sloan, not Charles T. Sloan.

We apologize for any inconvenience.

—The Editors

TECHNICAL CALENDAR

March 19–20—Tolerance Stacks Using Geometric Dimensioning & Tolerancing, Level 2. Holiday Inn, Chicago City Center, Chicago, IL. Focuses on detailed procedures for performing tolerance stacks on parts and assemblies, including both coordinate and geometric dimensions and tolerances. This course is suitable for engineers and designers familiar with the principles, concepts and practices contained within Y14.5 and who are looking for a comprehensive process for getting GD&T into a tolerance stack. $845 for SME members, $945 for non-members. People who register before March 5 receive the special rate of $795 for members, $895 for non-members. For more information, contact SME by telephone at (312) 240-8252 or on the Internet at *www.sme.org/training.*

April 1–4—2003 Metal Gear School. Universal Technical Systems Inc., Rockford, IL. The class, conducted by an instructor with 25 years' experience, begins with basic gear theory and proceeds with topics such as minimum weight design strategies, gear size, geometry design, torsional analysis, gear noise, lubrication and profile. $995. For more information, contact UTS by telephone at (815) 963-2220 or by e-mail at *sales@uts.com.*

April 7–12—Hannover World Fair for Industrial Technology. Hannover Fairgrounds, Hannover, Germany. This is a series of trade shows taking place at the same venue that comprise the world's largest concentration of industrial technology. The eight separate trade shows include factory automation; microtechnology; motion, drive and automation; energy; compressed air and vacuum technology; factory equipment and tools; subcontracting; and research and technology. $46 for a season ticket, $22 for a daily ticket. For more information, contact Hannover Messe by telephone at (609) 987-1202 or on the Internet at *www.hannovermesse.de.*

May 21–23—Fundamentals of Parallel Axis Gear Manufacturing. Pheasant Run Resort, St. Charles, IL. This seminar, with a limited class size of 35, focuses on gear nomenclature, automation systems, basic gear mathematics, hobbing, skiving, fixturing and blanks, as well as other topics. $750 includes a workbook, materials and a demonstration of gear manufacturing equipment. For more information, contact Koepfer America L.L.C. by telephone at (847) 931-4121 or on the Internet at *www.koepferamerica.com.*

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- **Rate this article**
- **Request more information**
- **Contact the organizations mentioned**
- **Make a suggestion**

Or call (847) 437-6604 to talk to one of our editors!

Wear Resistance of Plasma and Pulse Plasma Nitrided Gears

Bojan Podgornik and Jožef Vižintin

Summary

In this study, wear behavior of plasma and pulse plasma nitrided gears, made from 42CrMo4 steel, was evaluated under a lubricated sliding and pitting wear regime. The nitrided samples were fully characterized before and after wear testing using metallographic, microhardness and surface examination techniques. Sliding wear tests were carried out on a pin-on-disc machine in which hardened ball-bearing steel discs were mated to nitrided pins. Pitting wear tests were performed using the standard FZG machine with C-type gears. (FZG refers to the Gear Research Centre, a part of the Institute for Machine Elements at the Technical University of Munich, Germany.)

Also, economic evaluation and comparison of hardening and plasma nitriding processes were carried out. The test results indicate that sliding wear resistance and, even more so, pitting wear resistance of 42CrMo4 steel gears can be greatly improved by means of plasma and pulse plasma nitriding.

Introduction

Traditionally, gear manufacturers employ techniques such as carburizing, flame hardening, induction hardening, and gas nitriding to increase the strength of gearing components. While carburizing is the most common and effective surface hardening method used to improve load-carrying capacity of gears, the technique has shown substantial difficulties in the production of large gears (Ref. 1). The quench hardening from a high austenitizing temperature often results in unpredictable levels of tooth deflection, helix angle change, and overall distortion (Ref. 2).

During conventional gas and bath nitriding, a multiphase compound layer is formed on the nitrided surface (Ref. 3). This layer contains high internal stresses in the transitional regions between the various lattice structures, which make the layer brittle and can cause it to spall off in service. Such layers are clearly undesirable and hence have to be removed from load-carrying surfaces before the gears can be used in service (Ref. 4).

One of a new generation of heat treatment processes that has been employed to improve the performance characteristics of gears is plasma nitriding (Refs. 5 and 6). Plasma nitriding, an environmentally clean method of nitriding, permits a fully automated and controlled nitrogen-diffusion process, which makes it possible to perform nitriding without a compound layer formation (Ref. 7). Furthermore, the low temperatures used in plasma nitriding and the absence of a quenching requirement assure

Table 1—Details of the Heat Treatment Processes.

Process	Gas mixture	Temp. (°C)	Time (hrs.)	Pulse (sec.)
Through Hardening	oil quenched	870/180	2/1	—
Plasma Nitriding	99.4% H_2, 0.6% N_2	540	17	—
Pulse Plasma Nitriding	99.4% H_2, 0.6% N_2	540	17	0.48/0.02

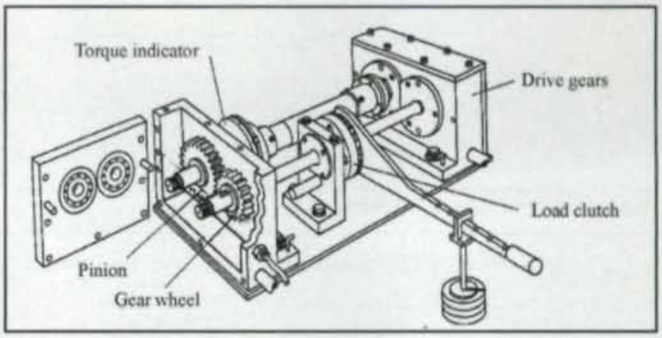


Figure 1—FZG gear test rig.

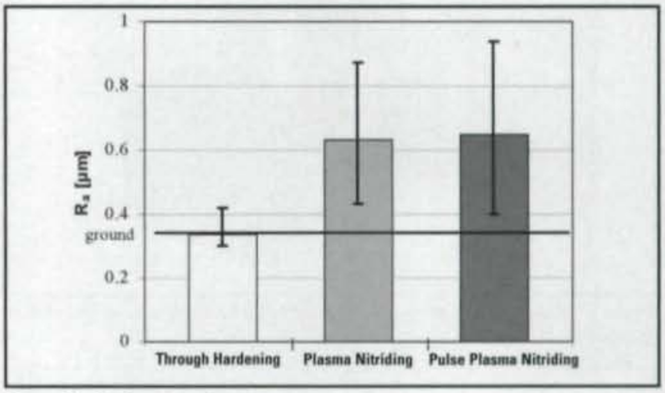


Figure 2—Surface roughness.

Dr. Bojan Podgornik

is an assistant professor in the mechanical engineering faculty at the University of Ljubljana, Slovenia. He also works in the university's Centre for Tribology and Technical Diagnostics. Specializing in surface engineering and hard coatings, Podgornik has researched the mechanical and tribological properties of hardened and plasma nitrided steels.

Dr. Jožef Vižintin

is a professor in the University of Ljubljana's mechanical engineering faculty and is head of the Centre for Tribology and Technical Diagnostics. He has researched the surface engineering and tribology of different materials. For his doctorate, Vižintin studied power losses in gears.

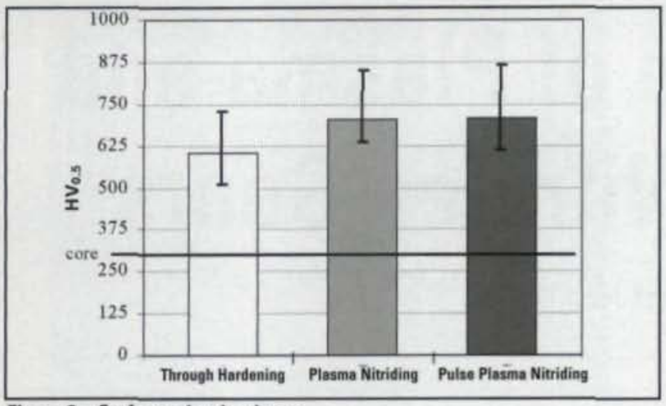


Figure 3—Surface microhardness.

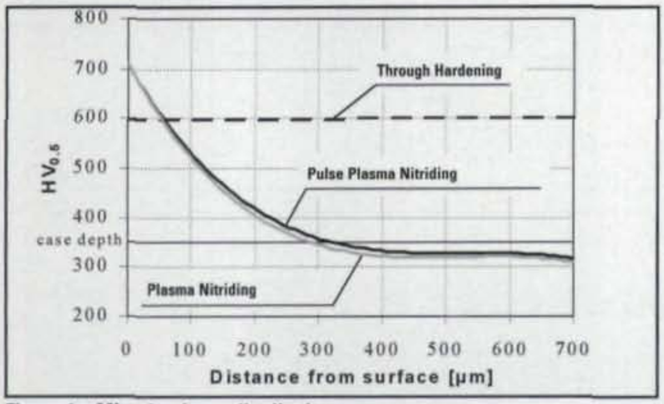


Figure 4—Microhardness distribution.

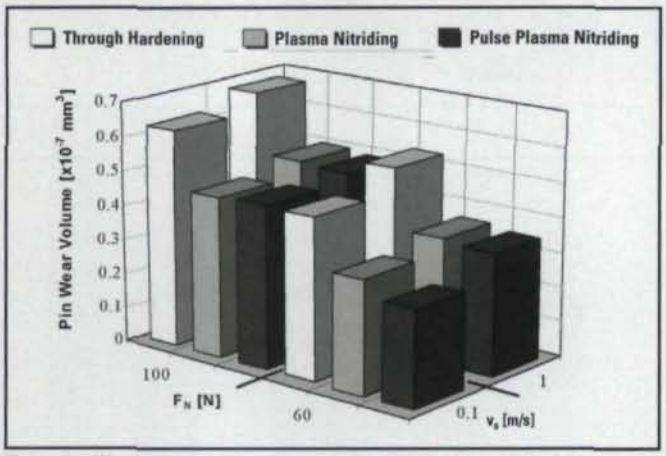


Figure 5—Wear volume of heat treated pins as a function of load and sliding speed.

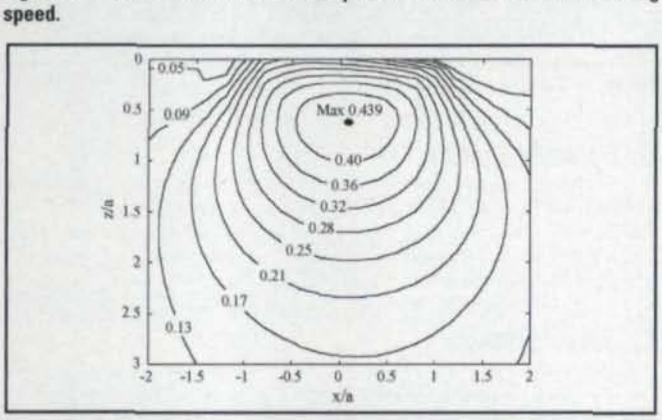


Figure 6—Principal shear stress distribution (τ/p) of tested gears (μ = 0.1, T = 239.3 N-m, contact radius a = 0.20 mm, contact pressure p = 1,125 N/mm²).

minimal distortion and dimensional variations (Ref. 8).

Therefore, by using plasma nitriding, the need for a subsequent grinding operation can be reduced or even eliminated. More advanced pulse plasma technology, employing pulse duration and duty cycle control, allows the use of the smallest amount of plasma power for the process to prevent overheating and to ensure uniform temperature distribution. Furthermore, almost every type of steel can be nitrided using a pulsed plasma technology (Ref. 9).

Experimental

The material used in the present investigation was commercial structural steel for hardening and nitriding, 42CrMo4 (0.5%C, 1.0%Cr, 0.2%Mo). After the specimens (pins and gears) were machined from hardened and tempered bars (300 $HV_{0.5}$), they were ground ($R_a \approx 0.4$ µm) and degreased before plasma nitriding.

The specimens were nitrided in a commercial, sensor-controlled plasma nitriding furnace. Plasma nitriding was carried out with precise control of all process parameters, under plasma and pulse plasma mode, to form a nitrided case with a nitriding depth of ≈0.3 mm and surface structure without a compound layer (Table 1), which was realized using low nitrogen content in the plasma (Ref. 10). For comparison purposes, one group of specimens was also hardened (oil quenched and tempered at 180°C), ground to a surface roughness of ≈0.35 µm.

Stylus profilometry was used to analyze surface roughness and topography of thermochemically treated specimens. Surface hardness and subsurface hardness distribution were measured using a Vickers microhardness tester at a 50 g indentation load.

Sliding wear resistance of heat treated flat-ended pins (ϕ 2 mm) was determined on a pin-on-disc machine. The pin was loaded against a rotating ball-bearing steel disc, hardened to 700 $HV_{0.5}$ and ground to an average roughness value of ≈0.4 µm. Lubricated sliding wear tests (non-aditivated ISI VG68 oil) were carried out at room temperature (≈20°C) and relative humidity of about 50%, sliding speeds of 0.1 m/s and 1 m/s and normal loads of 60 N and 100 N. Wear tests were stopped after 2,000 m of sliding.

Pitting wear tests were performed using the standard FZG machine (Fig. 1) with the C-type gears. A detailed description of the test rig is given by Winter and Michaelis (Ref. 11). The same heat treatment process was used for the pinion and the gear wheel. Heat treated gears were lubricated by formulated gear oil ISI VG68. Pitting wear tests were carried out as a two-step process at an oil temperature of 90°C. After a running-in sequence (2 hrs. at 94.1 N-m—stage 5), the test was run at 239.3 N-m torque (stage 8) until pitting failure occurred. The failure criterion was the occurrence of a pitted area on one pinion tooth greater than 4% of the tooth's area.

Results—Surface Properties

After nitriding, both the average roughness value R_a and the maximum peak-to-valley height R_{max} increased when compared with the original ground surface. The average roughness of the original surface changed from 0.35 μm to approximately 0.65 μm, measured for plasma and pulse plasma nitrided specimens. Pulse plasma nitriding, however, was found to cause larger scattering of the results, as shown in Figure 2.

Characteristic surface microhardness of heat treated 42CrMo4 steel is shown in Figure 3. Compared to hardening with the highest obtainable surface hardness of ≈600 $HV_{0.5}$ and a constant hardness over hardened zone (≈1 mm), plasma nitriding increased the surface hardness of investigated steel to approximately 700 $HV_{0.5}$. For both plasma nitriding techniques, the hardness decreased gradually with increasing distance from the surface (Fig. 4). Comparison of plasma and pulse plasma nitrided 42CrMo4 steel showed larger scattering of the surface hardness values in the case of pulse plasma nitrided specimens (Fig. 3). Furthermore, pulse plasma nitriding was found to produce almost the same nitriding depth (0.3 mm) compared with plasma nitriding, as shown in Figure 4.

Wear Properties—Pin-on-Disc Test

From the pin-on-disc tests, it was found that, under all testing conditions, the coefficient of friction was largely independent of differences among heat treatments used in this study. In all cases, the coefficient of friction was in the range of 0.15. Figure 5 shows wear volume of heat treated pins as a function of heat treatment, test load and sliding speed after 2,000 m of lubricated sliding. Depending on the testing conditions, plasma and pulse plasma nitriding were found to improve sliding wear resistance of 42CrMo4 steel by 30–40% as compared to hardening (Fig. 5), which is in agreement with earlier results (Refs. 12–14). However, comparison of plasma and pulse plasma nitriding showed no noticeable difference in wear of plasma and pulse plasma nitrided pins, as shown in Figure 5.

FZG Test

Contact stresses of tested gears (pitch point C) were calculated using Hertzian equations (Ref. 15), taking into account coefficient of friction, measured on pin-on-disc machine (v_s = 1 m/s, F_N = 60 N), and load conditions and geometry of testing gears. As shown in Figure 6, maximum principal shear stresses of tested gears are located ≈0.12 mm below the surface, which for nitrided gears is inside the nitrided zone. However, in the case of highly loaded gears, longer nitriding times should be used in order to obtain larger nitriding depths and adequate strength of the material (Refs. 5 and 16).

The results of the pitting wear tests are shown in Figure 7. Plasma nitrided and pulse plasma nitrided 42CrMo4 steel gears showed greatly improved pitting wear resistance as compared to hardened steel gears. In the case of hardened gears, pitting failure occurred after ≈2 million cycles. Nitriding increased pitting wear resistance of 42CrMo4 steel by a factor of 5. The increase can be attributed to the combination of a high surface hardness, a fine surface microstructure and a tough core obtained by nitriding. As in the case of sliding wear tests, plasma and pulse plasma nitriding give very similar pitting wear resistance to the investigated steel, as shown in Figure 7.

Economic Evaluation

In order to complete the comparison of hardened and plasma nitrided gears, calculation of production and replacement costs was carried out on a medium-sized gearbox (gear outer diameter of ≈100 mm and tooth width of ≈20 mm), as an example. This calculation was carried out with the assumption that, due to pitting failure, hardened gears have to be replaced at least once during use of an industrial gearbox. Furthermore, analysis of maintenance in Slovenian companies revealed that, in the case of medium-sized gearboxes, replacement of a gear takes approximately six hours.

Figure 8 shows treatment costs per unit for induction hardened, through hardened and plasma nitrided gears as a function of number of treated gears. In all cases, treatment costs decrease with increasing numbers of gears to be treated, as expected. However, by increasing the number of gears (> 100), plasma nitriding became more profitable compared with hardening (Fig. 8). Another very important cost-saving advantage of plasma nitriding is the fact that gears of different sizes and shapes can be nitrided at the same time.

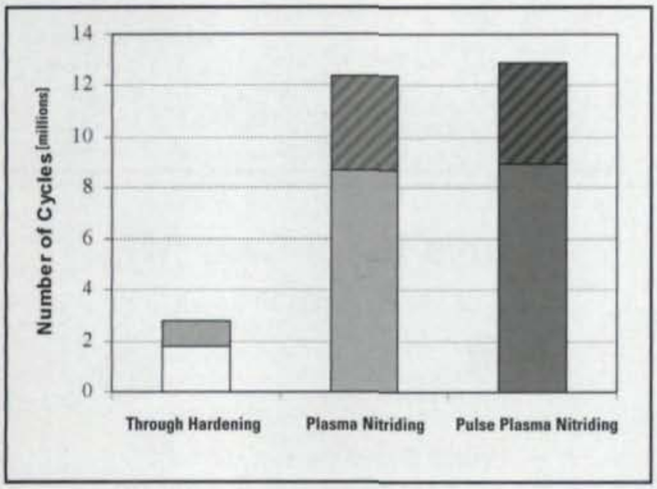


Figure 7—Pitting wear resistance of heat treated gears. (The top segments of the bars represent scattering of the results.)

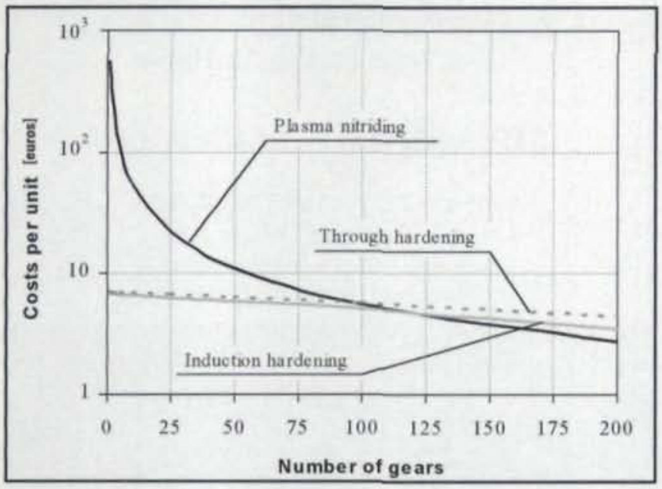


Figure 8—Heat treatment costs per unit as a function of number of treated gears.

By taking into account the results of pin-on-disc and FZG tests, as well as production and replacement costs, the following evaluations can be made:

1.) In the case of special, small-volume gears (< 10), changing from hardened to plasma nitrided gears may not be profitable. However, in the case of specialized gears, production of gears represents the main part of overall costs. Therefore, prevention against gear failure represents the main criterion for heat treatment selection.

2.) In the case of larger volumes (> 100), changing from hardened to plasma nitrided gears represents a cost reduction of up to 50%. For highly loaded gears, this reduction can be even higher, since plasma nitriding gives up to five times better pitting wear resistance compared with through-hardened gears (Fig. 7).

Conclusions

The combination of a high surface hardness, a fine surface microstructure and a tough core, obtained with plasma nitriding, leads to favorable tribological properties in nitrided steel. Pin-on-disc and FZG wear test results show that, compared with hardening, plasma and pulse plasma nitriding can greatly improve sliding wear resistance and—even more so—pitting wear resistance of 42CrMo4 steel gears.

Depending on the production quantity, both types of plasma nitriding can be rather expensive. However, changing from hardened to plasma nitrided gears reduces the probability of gear failure, and therefore leads to a reduction of overall costs connected with the gear replacement and cessation of production. ○

This paper was presented at the International Conference on Gears, held March 13–15, 2002, in Munich, Germany, and was published by VDI Verlag in the conference's proceedings, in VDI report 1665. Also, the paper was published in *Surface Engineering*, Vol. 17, No. 4 (2001), under the title "Sliding and Pitting Wear Resistance of Plasma and Pulse Plasma Nitrided Steel."

References

1. Albertin, L., R.L. Frohlich, H. Winter, B.-R. Höhn and K. Michaelis. "Load carrying capacity of nitrided steel gears" (94FTM4), American Gear Manufacturers Association Fall Technical Meeting, St. Louis, MO, October 1994.

2. Fronius, S. *Maschinenelemente*, VEB Verlag Technik, Berlin, 1971.

3. Karamis, M.B. "Tribological behaviour of plasma nitrided 722M24 material under dry sliding conditions," *Wear*, Vol. 147, No. 2, 1991, pp. 385–399.

4. Arques, J.L., and J.M. Prado. "The dry wear resistance of a carbonitrided steel," *Wear*, Vol. 100, 1985, pp. 321–332.

5. Roelandt, A., J. Elwart and W. Rembges. "Plasma nitriding of gear wheels in mass production," *Surface Engineering*, Vol. 1, No. 3, 1985, pp. 187–191.

6. Bell, T. "Towards designer surfaces," *Industrial Lubrication and Tribology*, Vol. 44, No. 1, 1992, pp. 3–11.

7. Lampe, T., S. Eisenberg and G. Lauden. "Compound layer formation during plasma nitriding and plasma nitrocarburising," *Surface Engineering*, Vol. 9, No. 1, 1993, pp. 69–76.

8. Karamis, M.B. "Friction and wear behaviour of plasma-nitrided layers on 3% Cr-Mo steel," *Thin Solid Films*, Vol. 203, No. 1, 1991, pp. 49–60.

9. Huchel, U., J. Crummenauer, S. Stramke and S. Dressler. "Pulsed plasma nitriding and combined processes," *Proceedings of the 5th World Seminar on Heat Treatment and Surface Engineering* (IFHT'95), Isfahan, Iran, 1995, pp. 329–335.

10. Sun, Y., and T. Bell. "Modeling of the plasma nitriding of low alloy steels," Developments in the nitriding of iron and titanium based alloys, Royal Society London, London, U.K., 1995.

11. Winter, H., and K. Michaelis. "FZG Gear Test Rig—Description and test possibilities," CEC 2nd International Symposium of Performance Evaluation of Automotive Fuels and Lubricants, Wolfsburg, Germany, June 1985.

12. Podgornik, B., J. Vizintin and V. Leskovsek. "Dry sliding wear characteristics of induction hardened, conventional plasma nitrided and pulse plasma nitrided AISI 4140 steel," *Proceedings of the COST 516 Tribology Symposium*, Espoo, Finland, May 1998, pp. 369–379.

13. Podgornik, B., J. Vizintin and V. Leskovsek. "Wear properties of induction hardened, conventional plasma nitrided and pulse plasma nitrided AISI 4140 steel in dry sliding conditions," *Wear*, Vol. 232, No. 2, 1999, pp. 231–242.

14. Podgornik, B., J. Vizintin and V. Leskovsek. "Sliding wear characteristics of induction hardened, conventional plasma nitrided and pulse plasma nitrided structural steels," *Proceedings of the 2nd COST 516 Tribology Symposium*, Antwerp, Belgium, May 1999, pp. 171–179.

15. Johnson, K.L. *Contact Mechanics*, Cambridge University Press, Cambridge, U.K., 1985.

16. Karamis, M.B. "Experimental study of the abrasive wear behaviour of plasma nitrided gearing steel," *Wear*, Vol. 161, 1993, pp. 199–206.

Norma B.V. ASS AG Buy Crown Gear

The manufacturing sector of Crown Gear B.V. was acquired by ASS AG and Norma B.V. on Jan. 27.

According to Norma's press release, ASS AG of Düdingen, Switzerland, has taken over the sales and engineering activities, while Norma B.V. is the coordinating partner for manufacturing, production and quality assurance. For the time being, Crown Gear's production facilities will continue to be located in Hengelo, The Netherlands, to finish outstanding supply duties. Soon all manufacturing activity will be moved to the Norma production facilities, which are also in Hengelo.

The Brunner-Antriebstechnik Co. will represent ASS AG in south Germany and take over customer support.

Crown Gear, inventor of the patented *Cylkro* angular face gears, declared bankruptcy Dec. 18.

Gleason and Kashifuji Add Hob Sharpening Products to Alliance Agreement

Gleason Corp. of Rochester, NY, and Kashifuji Works Ltd. of Kyoto, Japan, announced an agreement to develop, promote, sell and service hob sharpening equipment worldwide.

Under the terms of the agreement, Gleason has exclusive rights to the sales and service of Kashifuji hob sharpening machines throughout much of the world. Gleason's rights are non-exclusive in Japan, the Republic of China (Taiwan) and the Republic of Korea.

Both companies cooperated in the development of the KG 252L CNC Hob Sharpener, which grinds flutes on straight or helical hobs up to 10" in diameter and length.

This machine is the latest initative in the alliance between Gleason and Kashifuji, which began in March 2000.

Welduction Merges With Inductoheat

Welduction Corp., a manufacturer of induction equipment, merged operations with Inductoheat and relocated. Both companies are now located in the same facility in Madison Heights, MI.

Inductoheat has product lines that range from general purpose machines to custom automated heat treating systems, mass heating, wire heating, bonding, lost core technology and brazing machines.

Both companies are owned by Inductotherm Industries.

The merger was complete Jan. 1.

AGMA *Standards Available* on Internet, CD-ROM

A CD-ROM of all technical standards will be mailed to members of AGMA. Additionally, members can download updates through a new agreement with the American National Standards Institute (ANSI).

When AGMA publishes a new set of standards, members will be notified via e-mail that they can access the information for free on the ANSI website by typing in a user name and password.

Non-members can still view the standards, but they must order them through the AGMA website.

Under the terms of the agreement, members are eligible for discounts on other standards available from ANSI's Internet site.

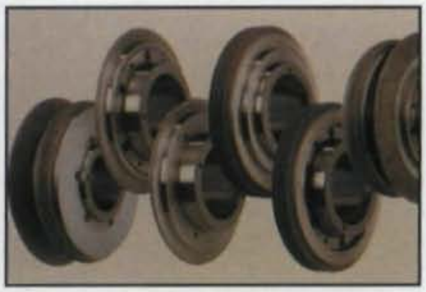
According to AGMA's newsletter, engineers and designers at member companies and divisions of multinational companies have unlimited access to AGMA standards electronically.

Bodycote Installs New Vacuum Furnace

Bodycote North America installed a new Abar Ipsen HR 46 vacuum furnace at its Santa Ana, CA, facility.

According to the company's press release, the furnace should improve vacuum capabilities and turnaround time. The product will target the tool and die, stainless steel and aerospace industries.

Working dimensions of the furnace are 36" x 48" x 33". Load sizes in excess of 2,000 lbs. can be processed. The all-metal hot zone will operate at temperatures up to 2,200°F with ultimate vacuum of 1×10^{-5}.



Reishauer Expands Capabilities for Diamond Dressing Tools

Reishauer Corp. of Elgin, IL, completed a new facility for manufacturing rotary diamond dressing tools for gear and non-gear forms.

According to the company's press release, the expansion enables Reishauer to be a single source supplier of quality diamond dressing tools in addition to its hard-gear finishing machines.

The new series of dressing tools includes SPA, SPA-S, RP-1SW and RP-2SW as well as DSA dressing units in single taper discs and full form rolls in single and multiple start configurations. The RZF/RZP diamond masters are measured in positive and reverse electroplated versions.

Alfe Heat Treat Names Vice President

Ernie Lackner was promoted to vice president of Alfe Corporate Group, the parent of Alfe Heat Treating.

Among his new responsibilities will be overseeing the three Alfe divisions in Ohio, Michigan and New York.

Previously, Lackner was general manager of the Wadsworth, OH, division. He has been with Alfe since 1995.

Alfe, headquartered in Fort Wayne, IN, serves automotive, industrial and aerospace customers with facilities for ferrous and nonferrous heat treating.

Rockwell Automation Acquires CNC Technology

Rockwell Automation acquired some of the intellectual property associated with Power Automation's software-based CNC for $5 million.

This intellectual property will be used to provide CNC motion solutions focused on power train applications. These solutions will be delivered through Rockwell's global manufacturing solutions business to expand its presence in the global power train automotive industry.

Rockwell Automation of Milwaukee, WI, is a global provider of industrial automation power, control and information solutions for manufacturers.

Power Automation is located in Pleidelsheim, Germany.

Timken to Receive Payment from U.S. Treasury

The Timken Co. will be receiving $54 million from the U.S. Treasury under the U.S. Continued Dumping and Subsidy Offset Act (CDSOA).

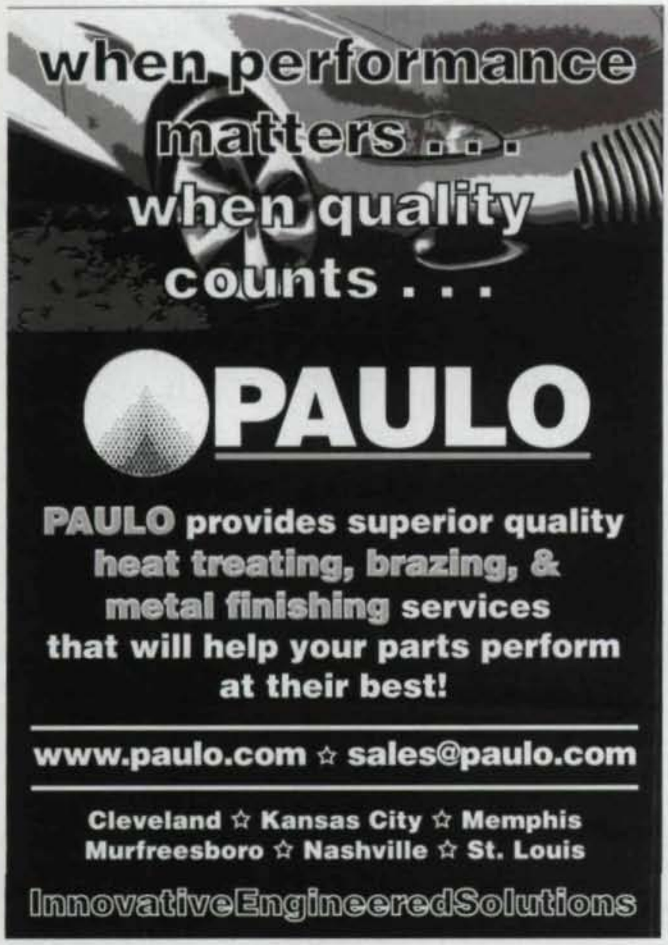
The CDSOA provides for distribution of money collected by the U.S. Customer Service from antidumping cases to qualified domestic producers that have continued to invest in their technology, equipment and people.

Timken received a payment from the U.S. Treasury in 2002 as well.

The Timken Co. of Canton, OH, is an international manufacturer of bearings, alloy and specialty steels and components.

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- **Rate this column**
- **Request more information**
- **Contact companies mentioned**
- **Make a suggestion**

Or call (847) 437-6604 to talk to one of our editors!



Fundamentals of Gears and Gear Manufacturing

SME recently released a 22-minute video, "Fundamentals of Gears and Gear Manufacturing," which introduces beginners to the gear business.

The narrator starts with an overview of manufacturing methods and finishing techniques, then moves to a description of the functions of gears: reversing rotational direction, altering the angular orientation of rotary motion, converting rotary to linear motion and vice versa, changing speed and changing power transmission ratios.

The narrator defines the key aspects of a gear, including base and pitch circle, line of center, pitch point, line of action, pressure angle, outside and root circles, addendum, dedendum, tooth thickness, circular pitch, face width, tooth face and tooth flank.

As soon as viewers learn the specifics of gears and their measurements, images of gear teeth pop up on the screen. The narrator elaborates on these images by noting the difference between internal and external teeth as well as the varieties of axis configurations.

Once a gear's features are defined, the video goes into greater detail about gear manufacturing processes. In this section, the narrator focuses on gear generating and gear form cutting. The gear generating process is broken down into hobbing and shaping.

Several minutes are devoted to hobbing, describing its benefits and drawbacks. For instance, hobbing is limited to producing external gear teeth on spur and helical gears.

The video also explains how shaping produces gears by rotating the workpieces in conjunction with a reciprocating cutting tool. The video demonstrates the use of shaper cutters that are pinion shaped or multi-tooth rack-shaped or are single-point cutting tools.

As for the other processes, the film touches on broaching and milling. Broaching is recognized as the fastest method of machining gears. Furthermore, pot broaching for external teeth is shown. Two variations on the milling process are given, standard and gashing on heavy-duty machines.

The video also comes with a free study guide and review quiz.

"Fundamentals of Gears and Gear Manufacturing" is available to SME members for $229 and to non-members for $255. To order, contact SME's customer service center by telephone at (800) 733-4769 or by fax at (312) 240-8252.

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- Rate this column
- Request more information
- Contact the organization mentioned
- Make a suggestion

Or call (847) 437-6604 to talk to one of our editors!

New Gear Roller from M&M Precision

The GRS-2 double flank gear roller from M&M Precision Systems Corp. is designed for composite roll testing or for use in a comprehensive analysis package.

According to the company's press release, the roller provides PC-based analysis software as well as a rugged construction for use on a shop floor.

A new manual system with plain diameter fixtures and a dial or digital indicator is offered as an alternative for those people seeking a simpler solution in roll testing.

For more information, contact M&M Precision of Dayton, OH, by telephone at (937) 859-8273 or by e-mail at *info@mmprecision.com*.

New Bevel Gear Grinding Machine from Gleason

The Phoenix II 275G gear grinding machine from Gleason utilizes a monolithic column design to accommodate gears with pitch diameters up to 275 mm and has a machine footprint of 2.3 x 3.4 meters.

Also included in the design is a self-contained electrical enclosure and hydraulic unit.

According to the company's press release, the grinding wheel spindle allows fast grinding cycles, high accuracies, maximum efficiency and low grinding costs. The automatic stock dividing system further assures accurate and rapid grinding cycles.

The product is available with either Siemens 840D or Fanuc 160I Model B CNC controls.

For more information, contact Gleason Corp. of Rochester, NY, by telephone at (585) 473-1000 or on the Internet at *www.gleason.com*.

New Gearhead from Mijno

The Type BDB gearhead from Mijno Precision Gearing substitutes a rotating flange for an output shaft, opening the door for creative design solutions.

Designed for powering the elbows of robotic arms and rotating large gears or capstans, the gearhead is offered in diameters from 64–285 mm (2.5–11.25").

According to Mijno, this gearhead eliminates the shaft, thereby omitting its torsional deflection and leaving the designer with only the gear teeth backlash to consider. The maximum backlash offered is 5, 3 or 1 arc-min.

The company added that the gearheads' internal design concept maximizes transmittable torque in a short length. The product family also features hypertorque versions in selected ratios.

For more information, contact Mijno Precision Gearing of Park Ridge, IL, by telephone at (847) 698-9041 or on the Internet at *www.mijno.com.*

New Speed Reducer Sizes from Cone Drive

The Series J Shaft Mounted Sala Speed Reducer Range from Cone Drive can now satisfy torque requirements from 450–57,800 N-m.

All gear teeth are case hardened and ground, and gear data is selected for minimum noise and vibration. Fixturing for the torque arm is integrated in the gear case, which is prepared for the mounting of vibration sensors.

Ratios from 5:1, 15:1, 20:1 and 25:1 are offered. In addition, the cast iron gear case is split for simple assembly and overhauling, and supplied ready for mounting of the V-belt cover.

For more information, contact Cone Drive of Traverse City, MI, by e-mail at *cnelson@hydeburnett.co.uk* or on the Internet at *www.textronpt.com.*

New Electric Rotary Hearth Oven from Grieve

The No. 815 is a 500°F electric rotary hearth oven that is used for preheating gears at a customer's facility.

According to the company's press release, the hearth is driven by a 1/4 hp motor through a gear reducer with a torque limiting device. The hearth indexes one position each time the loading door is opened and closed.

In addition, two 2,000 CFM, 2 hp recirculating blowers provide a vertical downward airflow over the workload.

For more information, contact The

Grieve Corp. of Round Lake, IL, by telephone at (847) 546-8225 or on the Internet at *www.grievecorp.com*.

New Gearing Component from HD Systems

The Quantum Series from HD Systems is a high torque capacity harmonic drive gearing component, delivering 50% more torque than the company's other gearing components.

The size 14 component maintains zero backlash, 1 arc-minute positional accuracy and +/–5 arc-second repeatability, according to the company's press release.

The series has an outer diameter of 50 mm and a maximum torque of 60 in.-lbs. Additionally, it is available in gear ratios of 50, 80, and 100:1.

For more information, contact HD Systems Inc. of Hauppauge, NY, by telephone at (800) 231-HDSI or by e-mail at *info@HDSI.net*.

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- Rate this column
- Request more information
- Contact companies mentioned
- Make a suggestion

Or call (847) 437-6604 to talk to one of our editors!

ADVERTISER INDEX

USE RAPID READER RESPONSE to contact any advertiser in this issue

www.geartechnology.com/rrr.htm

Rapid Reader Response: It's FAST! It's EASY! It's ONLY ONLINE!

Advertiser	Page	Phone	E-mail	Internet
AGMA	31	(703) 684-0211	gearexpo@agma.org	www.gearexpo.com
Ajax Tocco Magnethermic Corp.	3	(800) 342-1577	mmorrison@ajaxtocco.com	www.ajaxtocco.com
American Metal Treating Co.	46	(216) 431-4492	bruce@americanmetaltreating.com	www.americanmetaltreating.com
Applied Process Inc.	4	(734) 464-2030	info@appliedprocess.com	www.appliedprocess.com
Arrow Gear Co.	44	(630) 969-7640	bevels@arrowgear.com	www.arrowgear.com
A/W Systems Co.	10	(248) 544-3852		
B&R Machine and Gear Corp.	35	(800) 238-0651	inquirygt@brgear.com	www.brgear.com
Barit International Corp.	44	(847) 272-8128	people@barit.com	www.barit.com
Bourn & Koch Machine Tool Co.	47, OBC	(815) 965-4013	bournkoch@worldnet.att.net	www.bourn-koch.com
Cincinnati Steel Treating Co.	35	(513) 271-3173	cst@steeltreating.com	www.steeltreating.com
Dr. Kaiser (S.L. Munson)	28	(800)775-1390	info@slmunson.com	
Dragon Precision Tools Co. Ltd. (Greg Allen Co.)	30	(440) 331-0038	gallenco@msn.com	www.gallenco.com www.dragon.co.kr
Fässler Corp.	25, 37	(414) 769-0072	usa@faessler-ag.ch	www.faessler-ag.ch
Formax Precision Gear Inc.	47	(586) 323-9067		
The Gear Industry Home Page™	40	(847) 437-6604	people@geartechnology.com	www.geartechnology.com
Gear Motions Inc.	47	(315) 488-0100		www.gearmotions.com
The Gear Works-Seattle Inc.	46	(206) 762-3333	sales@thegearworks.com	www.thegearworks.com
Gleason Corp.	IFC, 47	(585) 473-1000	dmelton@gleason.com	www.gleason.com
Gleason Cutting Tools Corp.	1, 46, 47	(815) 877-9800	dmelton@gleason.com	www.gleason.com
Indiana Tool / Indiana Gear	11	(574) 936-2112		www.itamco.com
Inductoheat	6	(800) 624-6297	sales@inductoheat.com	www.inductoheat.com
Insco Corp.	37	(978) 448-6368	sales@inscocorp.com	www.inscocorp.com
Interstate Tool Corp.	47	(216) 671-1077		www.interstatetoolcor.com
Ipsen International	8	(815) 332- 4941	sales@abaripsen.com	www.ipsen-intl.com
ITW Heartland	32	(320) 762-8782	itwgears@rea-alp.com	www.itwgears.com
Koro Sharpening Service	47	(763) 425-5247	steve@koroind.com	www.koroind.com
Kreiter Geartech	47	(713) 237-9793	kreiter@kreiter-geartech.com	www.kreiter-geartech.com
LeCount Inc.	39	(800) 642-6713, (802) 296-2200	sales@lecount.com	www.lecount.com
Liebherr Gear Technology Co.	5	(734) 429-7225	info@lgt.liebherr.com	www.liebherr.com
Midwest Gear & Tool Inc.	32	(586) 754-8923	rosscr@attglobal.net	
Midwest Gear Corp.	46	(330) 425-4419	sales@mwgear.com	www.mwgear.com
Nachi Machining Technology Co.	43	(586) 263-0100	sales@nachimtc.com	www.nachimtc.com
Niagara Gear Corp.	46	(800) 447-2392, (716) 874-3131	info@niagaragear.com	www.niagaragear.com
Paulo Products Co.	40	(866) PAULO4U	sales@paulo.com	www.paulo.com
Perry Technology Corp.	IBC	(860) 738-2525	sales@perrygear.com	www.perrygear.com
powertransmission.com™	41	(847) 437-6604	ptsales@powertransmission.com	www.powertransmission.com
Process Gear/Bengal Industries	29	(847) 671-1631	jhertl@processgear.com	www.processgear.com
The Purdy Corp.	42	(860) 649-0000	sales@purdytransmissions.com	www.purdytransmissions.com
Russell, Holbrook & Henderson Inc.	39	(201) 796-5445	sales@tru-volute.com	www.tru-volute.com
Star-SU Inc.	46, OBC	(847) 649-1450	sales@star-su.com	www.star-su.com

CLASSIFIEDS

SERVICE

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.

HELP WANTED

SERVICE

WEBSITES

Gear Museum Road Trip

What's the perfect vacation destination for a gear aficionado? Aspen? Too trendy. Miami? Too humid. For a true machinery enthusiast, the perfect vacation is a gear museum road trip.

Like Caribbean island-hoppers, gear road trippers can send postcards from their many destinations.

First stop—Dearborn, MI, site of the Henry Ford Museum and Greenfield Village. The 12-acre museum is described as "a haven for mechanical engineering nuts." The museum is home to steam locomotive repair facilities as well as Thomas Edison's Menlo Park labs (where the light bulb was invented) and Ford's workshops. The museum also contains old cars, steam engines and restored 1880s factories.

Manufacturing machinery is held in the museum's "Made in America" exhibit. Most of the equipment falls into the category of production manufacturing. Among other things, the exhibit houses the manufacturing equipment for the Model T cars. Other gear equipment is still in storage at the museum, like a gear cutting machine from the 1890s and various Gleason and Bilgram gear cutting machines. Gears exist within the museum in grandfather clocks, steam engines and various other locations within the museum. In addition, curators boast of a generator that stands two stories high.

The Midwest is home to plenty of other gear-related destinations. Among the more famous is Chicago's Museum of Science and Industry. A permanent fixture at the museum is the "Animated Industrial Gears" exhibit, which showcases gears and motion devices from Borg-Warner Corp.'s exhibit at the Century of Progress International Exposition in 1933. One feature is Cartwright's straight line mechanism, which has two gears and two connecting arms of equal size, as well as historic motion devices.

After that quick stop in the Windy City, start caravaning eastward to New England. It's a long drive to Windsor, VT, but well worth it. Once there, postpone the skiing and try the American Precision Museum.

There, you'll find engineering masterpieces, like two gear measuring machines for involutes developed by Fellows Corp. (the company founded by gear pioneer Edwin R. Fellows, inventor of the gear shaper machine). You'll also find gear milling machines that date as far back as 1836 and one from Ezra Gould's 1858 collection.

According to collections technician John Alexander, most of the visitors at the American Precision Museum are retired machinists and engineering professionals. A real crowd pleaser is the demonstration of indexing on a benchtop miller. The process involves milling gears on a 1/8" piece of brass with the museum's name and Windsor, VT, stamped on it. Museum attendees can take the makeshift gears home as souvenirs of the day.

Don't spend all your money on souvenirs in Vermont, though. The gift shop at The Smithsonian Institution in Washington, D.C., probably has lots of memorabilia available for purchase. Although it's more well-known for the First Lady exhibit or the Hope diamond exhibit, the Smithsonian also houses a wing called "Engines of Change: The American Industrial Revolution, 1790–1860." In those halls, you can find wheels, steam powered machines, and replicas from machine shops. A special highlight of this stop is seeing gears constructed of five species of wood.

These antique gears at The Smithsonian Institution's "Engines of Change" exhibit are constructed with five species of wood.

For those people who want the complete gear-themed road trip and are willing to spend some lira, the Leonardo Museum located inside a medieval castle in Vinci, Italy, is the ultimate culmination to this trip. At the museum's opening in 1953, the first exhibit was dedicated to models of machines inspired by da Vinci's designs. Today, the museum is home to more than 50 models, many of which include various types of gears.

How to cross the Atlantic during a road trip is up to you. But, you're a gear pro; you know how to solve difficult problems. O

Tell Us What You Think . . .
Visit *www.geartechnology.com* to
- **Rate this column**
- **Request more information**
- **Contact the organizations mentioned**
- **Make a suggestion**

Or call (847) 437-6604 to talk to one of our editors!