GEAR TECHNOLOGY

MARCH/APRIL 2002
The Journal of Gear Manufacturing
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HEAT TREATING FOCUS

- Atmosphere vs. Vacuum Carburizing
- Case Depth and Load Capacity
- Trends in the Heat Treat Industry

ALSO IN THIS ISSUE:

- 3-D Finite Element Analysis of Composite Spur Gears

THE GEAR INDUSTRY'S INFORMATION SOURCE

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Gear Industry Barometers

A good sailor can predict when the weather is about to change. He uses simple tools to measure variables like air pressure, temperature and wind speed. Although those indicators can't perfectly forecast the weather, the sailor can get a good idea of what's going to happen by applying his experience, judgment and even his gut feelings.

While I'm not a sailor, I've been looking at signs of change, too. But the weather I'm watching is economic, not meteorological. We've been in a manufacturing recession for about 18 months; from June 2000 through November 2001, manufacturing production fell by 6.8%, the biggest drop-off since the 1981–82 recession. Given the current atmosphere, I'm sure most of you are also watching for signs of change.

While I've never been comfortable making predictions, I've made some observations that I think are telling about what may lie ahead. Some of the barometers I've been watching have been changing recently, and this may mean improved conditions for the gear industry are coming soon.

None of what I'm about to tell you is very scientific; just as the sailor's barometer is no Doppler radar, my gear industry barometers offer only hints about what's going to happen. Nevertheless, I'm hopeful enough about the signs to tell you what I've been seeing.

Recently, there's been plenty of news that the manufacturing sector is starting to turn around. For example, the Institute for Supply Management (formerly known as the National Association of Purchasing Managers) publishes an index that tracks economic expansion or contraction based on a survey of purchasing professionals. An ISM Index below 50 indicates contraction, while a measurement above 50 indicates expansion. In January, the index of manufacturing activity measured 49.9, up from 48.1 in December. Even though January was the 18th consecutive month of contraction, we're getting close. It's likely that by the time this issue goes to press, the index will be telling us that manufacturing is growing again or at least stopped shrinking.

Every day, I talk to gear manufacturers. Lately, more and more have been indicating that activity is beginning to pick up. Whereas most everything was dead quiet in the last quarter of 2001, now I'm hearing that there seems to be more activity, more quoting on work. Even if those quotes don't translate into jobs, the increased activity—when before there was none—is an indication that something is happening out there. Other manufacturers are telling me that their orders are beginning to pick up, and some are very busy.

I've also heard there's more activity in the machine tool field. Again, in most cases, that activity is increased quoting, but in some cases, it's actual sales. At the end of last year, almost nobody was buying machine tools. Now, at least, there seems to be interest in investing in equipment again.

Finally, changes in activity on our own websites, *powertransmission.com*™ and *The Gear Industry Home Page*™, also indicate that business may be improving.

Activity on *powertransmission.com*™ has picked up considerably over the last several months, both in the amount of traffic and in the number of requests for quotes that are sent to our advertisers. Over the past three months, we've averaged about 25,000 user sessions per month, after having leveled off at about 18,000 for most of 2001. More people than ever before are coming online and requesting information from and viewing the pages of manufacturers of gears, gear drives, motors, bearings and other power transmission components offered on that site.

The Gear Industry Home Page™ has also started showing signs of increased activity, especially in the quality and quantity of sales leads delivered to the advertisers there. People are requesting information about gear manufacturing equipment again. Page requests for *The Gear Industry Home Page*™ jumped to 96,000 in January, from an average of about 75,000 per month in October, November and December.

It would be easy to make predictions if all I had to do was tell you that we're going to have an economic recovery in the gear industry. We are. The hard part is telling you exactly when.

Nevertheless, there's no denying that we're beginning to see positive changes. Those signs may not reveal for sure that the storm is ending, but they bear watching. My experience, judgment and gut feeling point toward the change coming sooner rather than later.

What all this means is that I'm expecting the weather to improve. Although I'm not yet ready to venture out to sea, I've seen enough to know that I should start loading the cargo.

Michael Goldstein

Michael Goldstein,
Publisher and Editor-in-Chief

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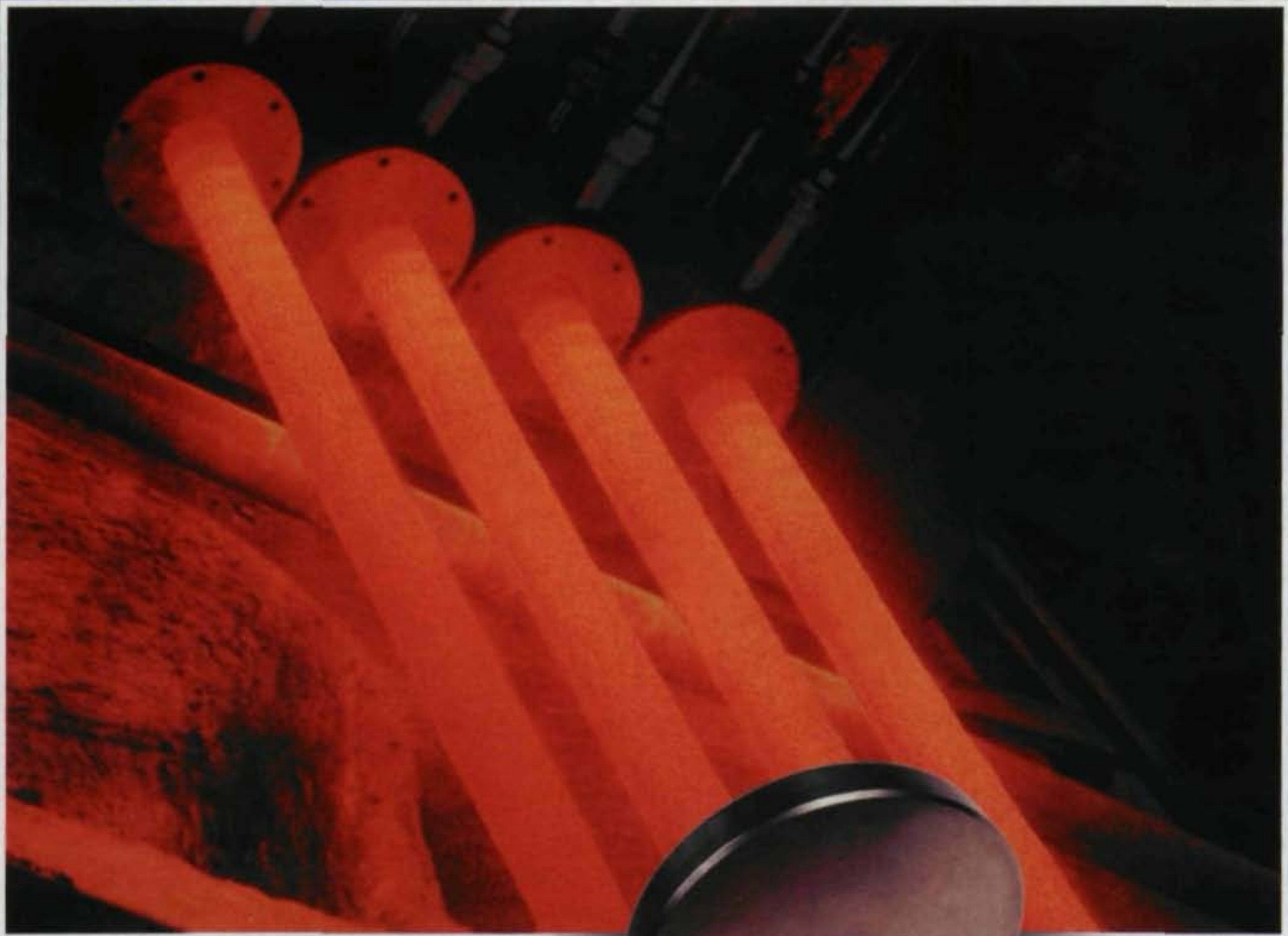

Dramatic Temperature Reduction Yields Longer Life

On a speedway, race cars hurtle through turns at speeds of 170 mph, their engines reaching temperatures of 180°F. To survive those extremes, their gears are often exposed to another extreme, treatment at –350°F.

The treatment, cryogenics, is used on gears to make them last longer, to increase their dimensional stability or to remove stress from them.

When cryogenics first became available for use on race car gears, the few companies that used the new technology kept it secret from the competition. Today, most companies require their race gears to undergo the treatment.

According to Rocky Beebe, general manager of One Cryo, a Puyallup, WA-based company that specializes in cryogenics, racing teams like Bob Panella Motorsports and others regularly send their gears in for treatment.

"Some of the racing teams want to treat the complete engine to make it tough enough to last 2–3 times longer. Others have trouble areas like the pinion and want us to concentrate on that part," he says.

The process works on nearly anything with moving parts. Gears, shafts, bearings and cases are treated for drivetrains and production machines. Among the gear-related applications for the cryogenic process are race cars, aerospace vehicles, tractors, turbochargers, even the brake rotors of ambulances.

In 1996, Lifestar Ambulance Co. in Springfield, IL, noted that its Ford E350 model rigs required a brake change every 9,000 miles. Van Prater, chief of operations for Lifestar, inspected the brakes

This machine lowers a gear's temperature by less than 1° per minute.

and noted the rotors were heat stressed and cracked, requiring a new set every third oil change. Lifestar consulted 300° Below Inc., a Decatur, IL-based company that specializes in low temperature cryogenic treatments designed to boost the performance and service life of critical components. Before its rotors were treated, Lifestar found that the brakes needed to be changed every 9,000 miles at $350 for each new set of rotors. After the treatment program, the rotors required changing every 55,000 miles.

"We were paying less than $200 to have a set of rotors treated and then inspected the brakes about every 6,000 miles. We had great luck, and the treatment ended up paying for itself," says Prater.

In short, the process involves lowering temperatures to –300°F to effect significant molecular changes, says 300° Below CEO Pete Paulin.

"There are mainly three benefits to undergoing the cryogenics process. First, it ensures a martensitic conversion from any retained austenite. Also, it relieves stress and promotes stabilization through thermomechanical compression and expansion. Finally, in ferrous materials, carbide precipitation leads to uniform refined structures enabling parts to withstand wear," Paulin says.

After parts undergo the procedure, the company claims they normally improve in performance and have reduced wear and breakage, sometimes expanding the life cycle by as much as 250–400%, according to literature from 300° Below. Treated tooling ranges from a handful of small drill bits to stamping dies that weigh upwards of 10,000 lbs. Cutting tools, such as hobs, can also be treated via the cryogenics process.

Whatever the end use, molecules are organized by closing the grain structure during the process. The material becomes more abrasion-resistant, but the hardness does not change, says Bob Reed, motor sports division manager at 300° Below.

"If we changed the surface to make it harder, then the product would become more brittle," explains Reed. Additionally, cryogenics is meant to affect the entire product rather than just the surface.

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

Treatment results vary from product to product, and some materials display resistance to cryogenics. Generally, higher carbon, higher alloy tool steels, martensitic stainless steels, some cast steels and gray iron have shown positive results. Mild steel and materials other than metal are among the components that are not recommended for cryogenic treatment.

Al Swiglo, a staff engineer with the Illinois Institute of Technology, investigated cryogenics for the U.S. Army. He was researching the manufacture of gears for carburized 9310 aerospace material and reviewed literature that analyzed breakage. Only one study provided inferior results for bending fatigue, but that test was done at different hardness levels, so it may be invalid.

"What we found was that for rolling/sliding contact fatigue, you could get 5% more loading capacity or 50% more life. Also, you could get [a] 50°F increase in the tempering temperature," Swiglo says. "Ordinarily, anytime you exceed the tempering temperature, or the temperature at which the part is heat treated, the hardness and life are decreased. Now, any time that part might get warm, we're raising the temperature that that could happen."

Through cryogenics, products change on the inside, so it's impossible to see on the outside that the material has been changed. Only a microexamination at a high magnification would indicate if the parts were treated.

"If a company has good records to know how much use they would get for an untreated product, then they would see that the life of the treated product is much enhanced," Reed says.

The cryogenics process takes approximately 72 hours at 300° Below. Reed estimates the company's deep cryogenics machine lowers the temperature by less than 1° per minute, but he says parts could break if the machine quickened the process.

Treatment costs vary according to weight. For example, the cryogenics operation on a V8 automotive engine runs about $562. As a rule, the greater the weight, the lower the cost per pound.

Finances aside, by exposing gears to these sub-zero temperatures, manufacturers are making it possible for the gears to achieve faster speeds on the race course and longer lives for military aviation and ambulances.

Fast Induction Hardening—without Preheating?

Take a 4" internal gear with a 1" face width, place it in a circular induction coil, heat for 0.6 seconds—without preheating, and what do you get?

According to Mike Hammond, you get a gear with a uniform case depth from tooth tips to roots.

Hammond is president of Electroheat Technologies L.L.C. Located in Auburn Hills, MI, the company has a system that transmits medium and high frequencies at the same time to induction harden gears and other complex shapes.

Dual frequency induction hardening isn't a new process to the gear industry. Companies, like Contour Hardening Inc. of Indianapolis, IN, have offered that process to the industry for some time.

Simultaneous dual frequency induction hardening is a new process, though.

Electroheat's system can achieve such hardening via an IGBT, an insulated gate bipolar transistor, which is part of a power circuit design that allows the coil's power supply unit to combine and filter medium and high frequencies.

On a shaft, a spur gear pattern is induction hardened by a medium frequency and a high frequency transmitted at the same time via an induction hardening system from Electroheat Technologies L.L.C.

Hammond describes the transistor as rugged and able to handle high frequencies. Previously, transistors in power output circuit devices for high frequencies weren't resilient enough. They were susceptible to premature failure—"especially so in heat treating applications," Hammond says.

With an IGBT, Electroheat's system can create and combine a medium frequency in the range of 10–25 kHz and a high frequency in the range of 150–500 kHz.

According to Hammond, the power supply unit filters the combined frequencies so the high frequency's feedback doesn't damage the medium frequency.

"This has always been the obstacle to overcome," he says.

Transmitting both frequencies at the same time, Electroheat's system heats tooth tip and root at the same time.

"We don't overheat either area," Hammond says. "This allows us to use very short heat times with rapid quenching."

The system's quenching time is 2–5 seconds, depending on the gear. According to Hammond, some gears cool by mass quenching because the hardening process uses rapid, shallow heating, which lessens distortion in gears.

According to Hammond, the system creates greater tooth bending fatigue strength than conventional heat treat processes, contributing to a longer tooth life.

The system can be used to induction harden spur gears, worm gears, internal gears and helical gears. Hammond adds that the system was tested and didn't need to preheat spur gears, internal gears and helical gears. Tests will be done on worm gears, but he says he doesn't expect the system to need to preheat those either.

Also, the system can emit a single frequency, either medium or high. So, it can treat a range of part configurations and variations and can be used for other applications, like tempering after hardening.

Hammond says the system is suited for in-house and commercial heat treating operations. He adds that the first system was sold to a commercial heat treating company in Europe.

Assuming the power supply unit can generate up to the 450 kW range, the whole system—the power supply unit, part-handling equipment and quenching/cooling machine—takes up 140 square feet.

Hammond says the system's drawback is probably cost. A typical system can cost $250,000–$650,000, depending on the complexity of the part-handling system and process-monitoring equipment.

Hammond adds: People may think the power supply unit is expensive—if they think of it as one power supply. If they think of it as two, then the unit doesn't look so pricey.

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3-D Finite Element Analysis of Long-Fiber Reinforced Composite Spur Gears

Çağdaş Alagöz, M.A. Sahir Arikan, Ö. Gündüz Bilir and Levend Parnas

This article was originally presented at the 2000 ASME Design Engineering Technical Conferences, held in Baltimore, MD, in September 2000. Reprinted with *permission.*

This article describes a method and a computer program that were developed for 3-D finite element analysis of long-fiber reinforced composite spur gears, in which long fibers are arranged along tooth profiles. For such a structure, the gear is composed of two regions; namely the long-fiber reinforced and the chopped-fiber reinforced regions.

Introduction

Compared with monolithic materials, composites have unique advantages, such as high strength, high stiffness, long fatigue life, low density, and adaptability to the function of the structure. Additional improvements can be realized in the corrosion resistance, wear resistance, appearance, temperature-dependent behavior, thermal stability, thermal insulation, thermal conductivity and acoustic insulation.

Plastic gears have excellent properties, such as self-lubrication, high chemical resistance, low noise and high impact resistance, but they are inferior to steel gears in their load capacities. For this reason, there are few examples of plastic gears being used for power transmission. If plastic gears reinforced with high-strength fibers, such as glass or carbon fibers, are realized, they will become useful plastic gears of high strength, as well as having the above-mentioned properties.

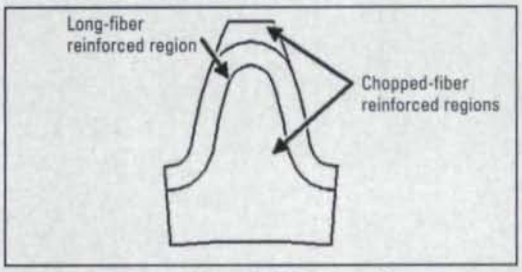

Figure 1—Long-fiber reinforced tooth.

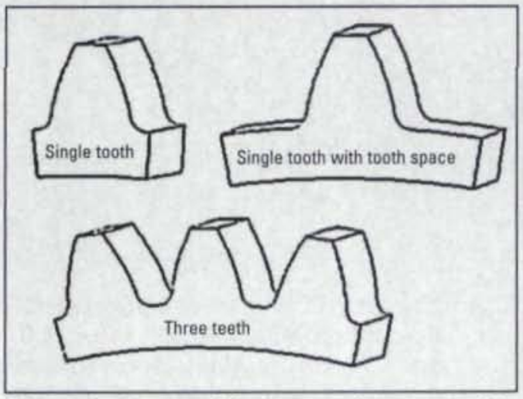

Figure 2—Gear models that can be generated by the developed program.

Although it is relatively easy to manufacture composite gears reinforced with chopped or particulate fibers by using the method used for isotropic gears, manufacturing a composite gear filled with long fibers is technically difficult. There are studies both on composite gears manufactured from chopped- or particulate-fiber reinforced materials (Refs. 11 and 9), and on long-fiber reinforced composite gears (Ref. 8). Almost no attempt has been made to investigate the possible stress and deformation variations for gears that could have been made of an orthotropic material.

In this study, composite spur gears, in which long fibers are arranged along tooth profiles, are analyzed in 3-D by using the finite element analysis method (Ref. 2). This type of composite spur gear was manufactured by Shiratori et al. (Ref. 8). A casting method was adopted to manufacture the gears. As shown in Figure 1, for such a structure, the gear is composed of two regions, which are the long-fiber reinforced and the chopped-fiber reinforced regions. It is important to note that there is almost no approach to analyze this type of composite gear because of different fiber orientations at different sections of the long-fiber reinforced region of the tooth.

Since elements have different orientations and material properties, for the finite element analysis of this type of composite gear with complex geometries, material properties of each element constituting the mesh should be defined separately. Forming the model geometry, subdividing it into elements, finding the appropriate mesh density and preparing the input data for a finite element program becomes a complex and time-consuming task, which is impossible to perform manually. All of the above-mentioned tasks are performed by the pre-processing module of the developed program. Main inputs for this module of the program are information on basic gear geometry, gear drive data, material properties and long-fiber reinforcement geometry. Finite element meshes are automatically generated, and mesh information with other required data is written to a file in the input-file format of ABAQUS®. Stresses are read from the output file of ABAQUS® by the post-processing module, and color-coded drawings for various stresses and failure indexes are displayed. For the long-fiber reinforced region, failure indexes are calculated by using the tensor polynomial failure criterion used by Herakovich (Ref. 5).

Finite Element Modeling

Finite element modeling mainly consists of gear model selection, mesh generation, boundary condition input, material definition and load definition. The first step is the selection of the gear model among the ones given in Figure 2. Next, the tooth profile is divided into segments, and the mesh is automatically generated for long-fiber reinforced and chopped-fiber reinforced regions. First, a 2-D mesh is generated. Then, the 3-D mesh is formed by considering the face width of the gear and the number of divisions specified in this direction. During mesh generation, thicknesses and locations of fiber arrangements and mesh density can be controlled. Then, boundary conditions, material definition and load definition are entered. Finally, mesh information and other required data are written to a file in the input-file format of ABAQUS®.

Gear Models. In the literature, various models are used for finite element analysis of gears. These include single-tooth models, models with one full tooth and two partial teeth at both sides, models with one full tooth and the tooth spaces on both sides, and three-tooth models. The most suitable model is determined by comparing the stresses found by using different models. Since all the models were developed and used for isotropic materials, the same comparisons should be made for composite materials. The program by the authors is capable of generating meshes by using the gear models given in Figure 2.

Generation of the Outside Profile for the Model. As shown in Figure 3, the outside profile of the model is generated by using seven guide points. Point 1 is on the tooth centerline; point 2 is at the tooth tip. The segment between points 2 and 3 is the involute tooth profile; the segment

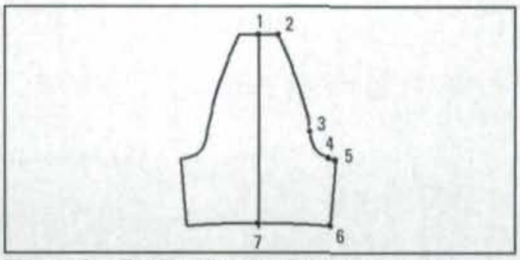

Figure 3—Guide points on the outside profile.

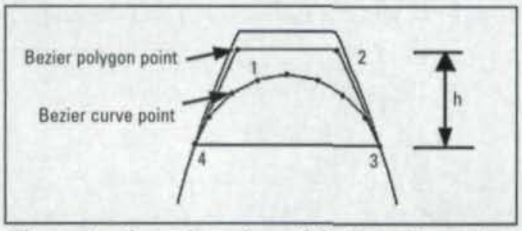

Figure 4—Upper boundary of the long-fiber reinforced region formed by using a bezier curve.

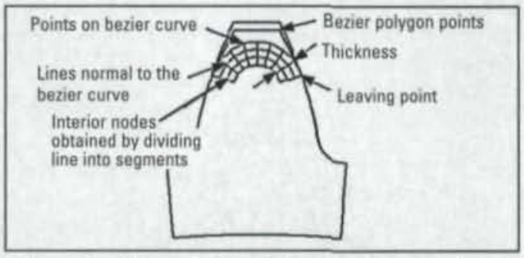

Figure 5—Generated mesh for the upper section of the long-fiber reinforced region.

bounded by points 3 and 4 is the trochoid tooth fillet profile. Point 5 is on tooth space centerline. The rim thickness of the gear is the distance between points 5 and 6. Finally, point 7 is again on the tooth centerline. Calculations on gear geometry and tooth profiles are made using the methods and equations of Arikan (Refs. 3–4).

Mesh Generation for the Long-Fiber Reinforced Region. In the long-fiber reinforced region, fibers are inserted into the tooth along the tooth profile. Bezier curves are used to form the geometry of the fibers. The shape of a Bezier curve can be controlled by making use of a defining polygon. As shown in Figure 4, four points are sufficient to form the polygon and the Bezier curve.

During mesh generation, lines normal to the Bezier curve and passing through the Bezier points are drawn. Line lengths are made equal to the fiber thickness. Interior nodes are obtained by dividing the lines into segments, as shown in Figure 5.

As seen in Figure 5, above the leaving point, fibers are not parallel to the tooth profile, but have a curved shape determined by the Bezier curve geometry. Below the leaving point, fibers become parallel to the tooth profile. Interior nodes below the leaving point can be generated by moving into the tooth in a direction normal to

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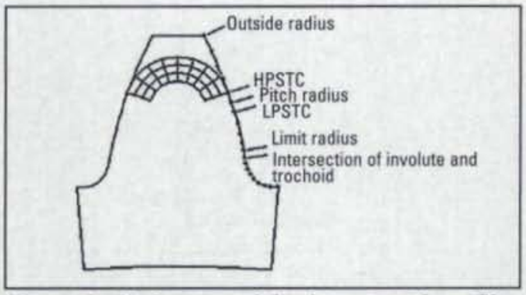

Figure 6—Nodes at special points on tooth profile.

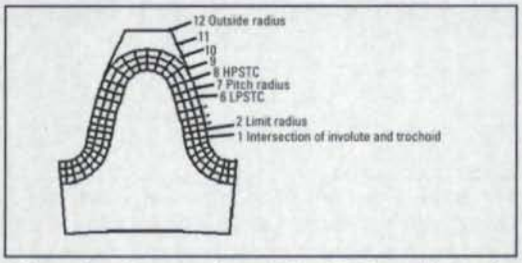

Figure 7—Generated mesh for the long-fiber reinforced region.

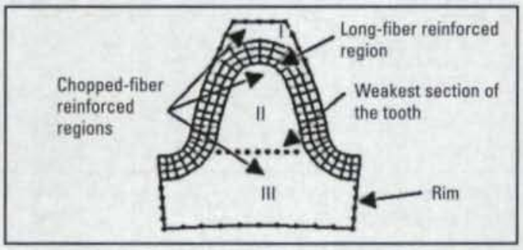

Figure 8—Sub-regions of the chopped-fiber reinforced region.

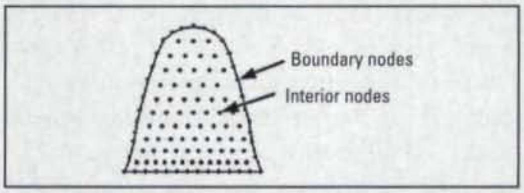

Figure 9—Boundary and interior nodes of sub-region II.

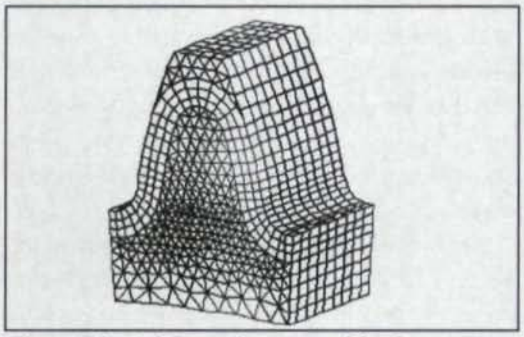

Figure 10—A 3-D mesh generated by the program.

the tooth profile. For this purpose, the first step is allocation of the nodes on the tooth profile. Since a finite element model can only be loaded at the nodes, first, nodes are allocated at special points for which solutions are desired; then intermediate nodes are formed between the special nodes. These special points are: highest point of contact (HPC), corresponding to outside radius; highest point of single tooth contact (HPSTC); pitch point, corresponding to pitch radius; lowest point of single tooth contact (LPSTC); lowest point of contact (LPC), corresponding to limit radius; and the point corresponding to the radius at which the involute tooth profile joins the trochoid fillet profile. Nodes allocated at the special points are given in Figure 6. Figure 7 shows the generated mesh for the long-fiber reinforced region.

Mesh Generation for the Chopped-Fiber Reinforced Region. A two-dimensional automatic triangular mesh generation algorithm (Refs. 6 and 12) is used for generation of the mesh for the chopped-fiber reinforced region. In order to be able to control the mesh density and have larger densities at critical areas, the region is separated into three sub-regions, as shown in Figure 8.

Mesh generation is performed in four steps: allocation of the nodes at the boundaries, generation of the interior nodes, formation of triangular elements, and smoothing of the mesh after triangulation.

Nodes at the boundaries are allocated by using the specified number of divisions for the sub-regions. For a sub-region, the boundary is represented by a disjoint union of a simple closed loop of straight-line segments. Then, the interior nodes are allocated by using the specified number of divisions. Figure 9 shows the boundary and interior nodes allocated in sub-region II.

Next, all of the nodes are connected, and triangular elements are formed in such a way that there are no overlapping elements and the entire region is covered. The triangulation scheme is designed to produce elements as near to equilateral triangles as the system of nodal points permits (Ref. 6). After triangulation, in order to have more equal element sizes, the smoothing process follows. Usually, the process converges after two cycles. Finally, elements in the tooth width direction are generated by considering the face width of the gear and the number of divisions specified in this direction. A 3-D mesh generated by the program can be seen in Figure 10. Sample 2-D meshes generated for different fiber arrangements and with different mesh resolutions are given in Figure 11.

Boundary Conditions, Material Definition and Load Definition. Two different boundary conditions can be used for the analysis. For solid gears, as shown in Figure 12a, all of the nodes at the sides and bottom of the model are fixed. For thin-rim gears, as given in Figure 12b, only the nodes at the sides are fixed. When material properties are considered, as mentioned before, two regions with different properties exist within the gear. These regions are the long-fiber reinforced region, in which unidirectional fibers are present, and the chopped-fiber reinforced region, in which there are randomly oriented, discontinuous fibers. If the fiber orientation in a composite material is truly random in a three-dimensional sense, the composite exhibits three-dimensional isotropy. Therefore, the material forming the chopped-fiber reinforced region can be considered as a material with a single elastic modulus and Poisson's ratio. In the long-fiber reinforced region, as shown in Figure 13, each element is a unidirectional, fiber-reinforced composite and has different material properties with respect to the global coordinate system because of its orientation. On the other hand, elements have the same elastic properties in the local coordinate system given in Figure 14. The local coordinate system is obtained by rotating the global coordinate system about axis 3 by the required angle. As seen in Figure 15, all of the elements in a slice have the same orientation and consequently the same elastic properties with respect to the global coordinate system. Thus, all elements in a slice can be reduced to an element set. Elastic properties of element sets are defined in their local coordinate systems.

For loading, distributed loads in the tooth width direction are resolved as shown in Figure 16. Thus, a uniform displacement distribution is obtained in the face-width direction (Ref. 3). Loads are applied to the nodes in a direction normal to the tooth profile.

Post-Processing of Results

For failure analysis, stresses are read from the output file of ABAQUS®, and color-coded drawings for various stresses and failure indexes are displayed. Figure 17 shows a sample output screen for failure index. Numerical values of the stresses and the failure index for an element can be displayed by bringing the cursor on the element and clicking the mouse button. In the figure, the black-colored element is the one for which the numerical results are displayed. At this step, it is also possible to replace the previously input material properties with new ones, and see new failure index values.

A tensor polynomial failure criterion (Ref. 5) is used to calculate the failure index for the long-fiber reinforced region. According to this criterion, failure occurs when,

$$\begin{aligned} f(\sigma_{ij}) &= F_1\sigma_1 + F_2\sigma_2 + F_3\sigma_3 \\ &+ F_{11}\sigma_1^2 + F_{22}\sigma_2^2 + F_{33}\sigma_3^2 + F_{44}\sigma_4^2 + F_{55}\sigma_5^2 + F_{66}\sigma_6^2 \\ &+ 2F_{12}\sigma_1\sigma_2 + 2F_{13}\sigma_1\sigma_3 + 2F_{23}\sigma_2\sigma_3 \geq 1 \qquad (1) \end{aligned}$$

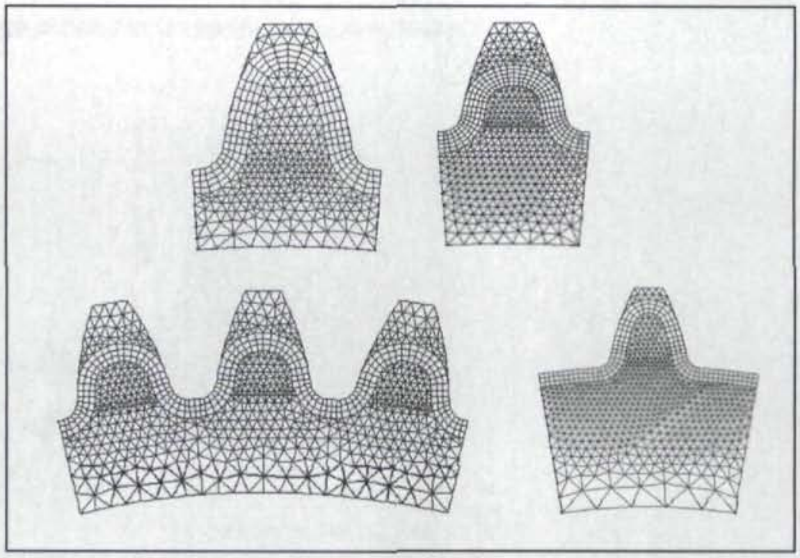

Figure 11—Sample 2-D meshes generated by the program.

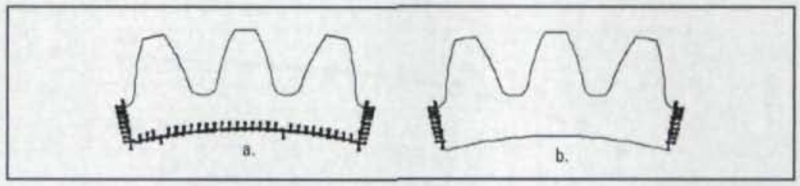

Figure 12—Boundary conditions.

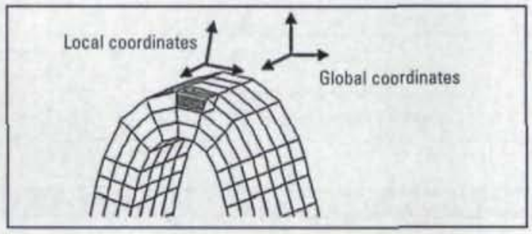

Figure 13—Fiber reinforcement in the long-fiber reinforced region.

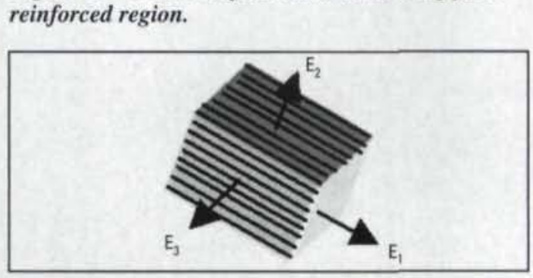

Figure 14—Material directions in the long-fiber reinforced region.

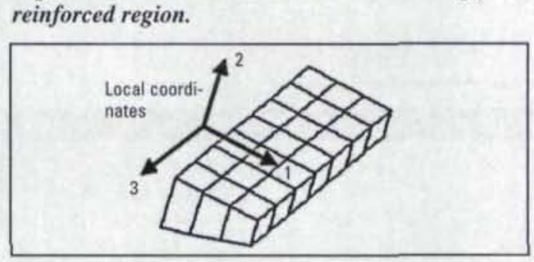

Figure 15—An element set formed by a slice.

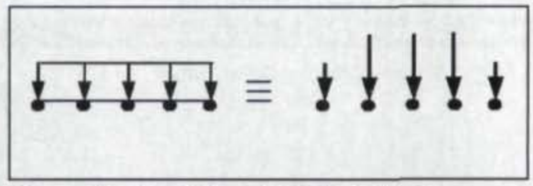

Figure 16—Resolution of distributed loads.

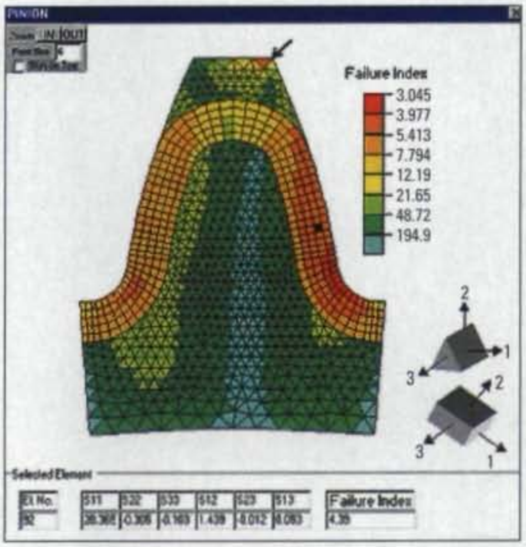

Figure 17—Sample output screen.

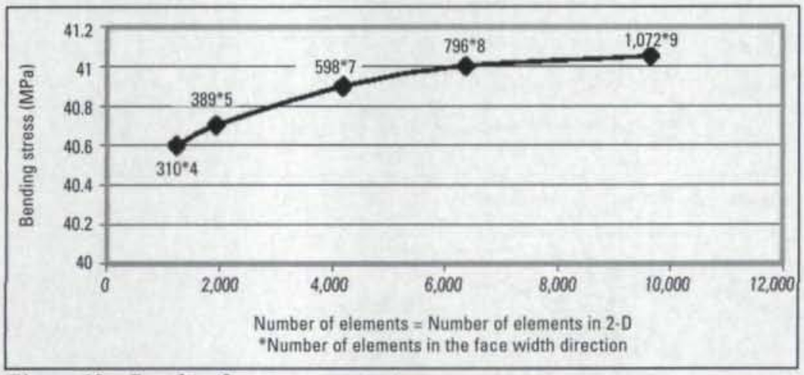

Figure 18—Results of convergence tests.

Table 1—Isotropic Spur Gear Data.

Basic Gear Geometry	
Pressure Angle	20°
Module, m	4.5 mm
Addendum	1.000 x m
Dedendum	1.250 x m
Generating Tool Tooth Tip Fillet Radius	0.300 x m
Gear	
Number of Teeth	72
Face Width	6.25 mm
Rim Thickness	2 x m
Material	
Elastic Modulus	210 GPa
Poisson's Ratio	0.3
Loading	
Load at Tooth Tip	500 N

Table 2—Bending Stresses at Tooth Root.

Tooth Model	Maximum Bending Stress (MPa)	
	Tensile	Compressive
Single Tooth	40.9	−48.2
Single Tooth with Tooth Space	39.7	−47.3
Three Teeth	39.8	−47.9

Table 3—Maximum Bending Stress at Tooth Root for Isotropic Gear.

Source	Maximum Bending Stress (MPa)
Developed Program	6.171
AGMA Equations	6.212
Oda et al. (FEM)	6.290
Oda et al. (Experimental)	6.236

where,

$$F_1 = \frac{1}{X_T} + \frac{1}{X_C}; \quad F_{11} = \frac{-1}{X_T X_C} \tag{2}$$

$$F_2 = \frac{1}{Y_T} + \frac{1}{Y_C}; \quad F_{22} = \frac{-1}{Y_T Y_C} \tag{3}$$

$$F_3 = \frac{1}{Z_T} + \frac{1}{Z_C}; \quad F_{33} = \frac{-1}{Z_T Z_C} \tag{4}$$

$$F_{44} = \frac{1}{Q^2}, \; F_{55} = \frac{1}{R^2}, \; F_{66} = \frac{1}{S^2} \tag{5}$$

In the above equations, X_T, X_C, Y_T, Y_C, Z_T and Z_C indicate tensile and compressive strengths in X, Y and Z directions; and Q, R and S indicate shear strengths in XY, XZ and YZ planes, respectively. Tsai and Hahn (Ref. 10) have proposed the following equations for interaction coefficients F_{12}, F_{13} and F_{23}.

$$F_{12} = -\frac{(F_{11} F_{22})^{1/2}}{2} \tag{6}$$

$$F_{13} = -\frac{(F_{11} F_{33})^{1/2}}{2} \tag{7}$$

$$F_{23} = -\frac{(F_{22} F_{33})^{1/2}}{2} \tag{8}$$

Failure index is defined as the inverse of $f(\sigma_{ij})$ given in Equation 1. Thus, a failure index less than one indicates failure, and a large failure index means a large margin of safety.

Verification of the Model

The model is verified by specifying all of the elements of the mesh as isotropic elements and comparing the results with previous results given for isotropic materials. Before the solution, convergence tests are performed, and a proper tooth model is determined. The spur gear with properties given in Table 1 is used for the tests. For the solution, all of the nodes at the sides and bottom of the model are fixed. Results of the covergence tests are given in Figure 18. By making use of the figure, it is decided to use about 6,000 elements (800 elements in 2-D and 8 divisions in the face width direction).

Bending stresses at the tooth root found by using different tooth models are given in Table 2. By making use of the table, it is decided to use the single-tooth model for the stress analysis, since more complex models give stresses that are only a few percent different than the ones given by the single-tooth model.

After deciding on the tooth model and the number of elements, tooth root stresses found for isotropic gears by using the model are compared to the results found by using AGMA equations (Ref. 1) and results given by Oda et al. (Ref. 7). Stresses given in Table 3 are again for the gear whose properties are shown in Table 1. In order to be able to compare the stresses with the results given by Oda et al. (Ref. 7), instead of 500 N, a tip load of 1 kg_f /mm is applied to the tooth. It is found that stresses calculated using the model are in good agreement with other results.

Sample Results

Sample results are obtained for the composite spur gear whose properties are given in Table 4. In the long-fiber reinforced and chopped-fiber reinforced regions, glass/epoxy is used. Material properties of the long-fiber reinforced, unidirectional composite material are shown in Table 5, and material properties of the chopped-fiber reinforced composite are shown in Table 6 (Ref. 8).

As shown in Figure 19, from a failure point of view, there are three critical sections in a composite gear tooth. In section 1, the critical stress component is the axial tensile stress in the direction parallel to the fibers. During a failure in this section, fibers are broken. In sections 2 and 3, the critical stress component is the transverse tensile stress in the direction normal to the fibers. During failures in these sections, fiber separation is observed. Arrows at the ends of the lines marking the critical sections indicate the failure direction.

Results obtained by using the developed program for glass/epoxy and carbon/epoxy gears, together with the results given by Shiratori, et al. (Ref. 8) are shown in Figures 20 and 21, respectively. For failure analysis of isotropic materials, the maximum stress should be considered; for unidirectional materials, the stress along the fiber direction should be considered. Thus, for the chopped-fiber reinforced regions (inside of the tooth), the ratio of the stress (in the direction parallel with the tooth centerline) to the strength is calculated at the critical (weakest) section of the tooth. For the long-fiber reinforced region (the region next to the outside surface of the tooth), the ratio of the stress along the fiber direction to the strength in the same direction is calculated. As seen in the figures, results of the program agree well with the results given by Shiratori, et al. (Ref. 8).

Effects of reinforcing thickness of the long-fiber reinforced region and the insertion depth of the fibers on strengths of glass/epoxy gears are shown in Figures 22 and 23, respectively. During interpretation of the results, failure is always observed in section 2, where the critical stress component is the transverse tensile stress in the direction normal to the fibers. As seen in Figure 22, the failure index (and the strength) increases with increasing reinforcing thickness until the thickness becomes 1.5 mm. Beyond 1.5 mm, reinforcing thickness has no positive effect on gear strength.

Figure 22 shows the effect of insertion depth (h_f) of the fibers on gear strength. Increased insertion depth has a small positive effect on gear strength.

In order to see the effects of different materials and different fiber volume ratios for long-fiber

Table 4—Composite Spur Gear Data.

Basic Gear Geometry	
Pressure Angle	20°
Module, m	5 mm
Addendum	1.000 x m
Dedendum	1.250 x m
Generating Tool Tooth Tip Fillet Radius	0.300 x m
Gear	
Number of Teeth	30
Face Width	6.25 mm
Rim Thickness	2 x m
Material	
Elastic Modulus	210 GPa
Poisson's Ratio	0.3
Loading	
Loading at Tooth Tip	1 kg_f/mm

Table 5—Material Properties for the Long-Fiber Reinforced Region (Unidirectional Composite).

		Glass/Epoxy	Carbon/Epoxy
Axial Modulus, E_1	GPa	15.2	31.7
Transverse Modulus, E_2	GPa	6.53	5.63
Shear Modulus, G_{12}	GPa	1.71	1.7
Poisson's Ratio, υ_{12}		0.13	0.06
Poisson's Ratio, υ_{23}		0.08	0.03
Fiber Volume Ratio, V_f		0.3	0.3
Axial Tensile Strength	MPa	103	289
Axial Compressive Strength	MPa	261	326
Transverse Tensile Strength	MPa	28.2	11.0
Transverse Compressive Strength	MPa	588	334
Transverse Shear Strength	MPa	41.4	37.6

Table 6—Material Properties for the Chopped-Fiber Reinforced Region (Isotropic Composite).

		Glass/Epoxy	Carbon/Epoxy
Elastic Modulus, E_1	GPa	9.2	17
Poisson's Ratio, υ_{12}		0.31	0.33
Fiber Volume Ratio, V_f		0.2	0.2
Tensile Strength	MPa	36	48
Compressive Strength	MPa	116	116
Shear Strength	MPa	98	98
Transverse Shear Strength	MPa	41.4	37.6

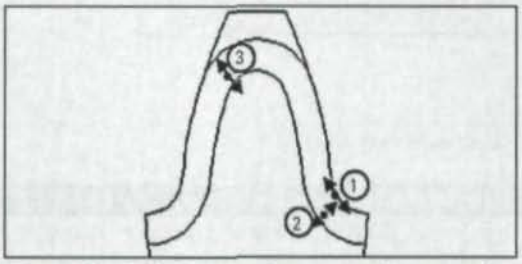

Figure 19—Critical sections in a composite gear tooth.

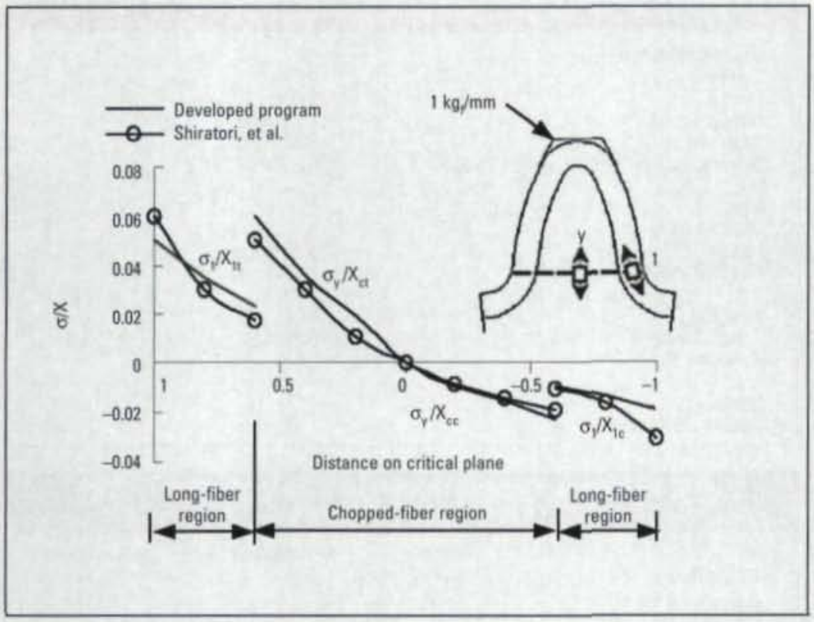

Figure 20—Stress Ratios for Glass/Epoxy Gear.

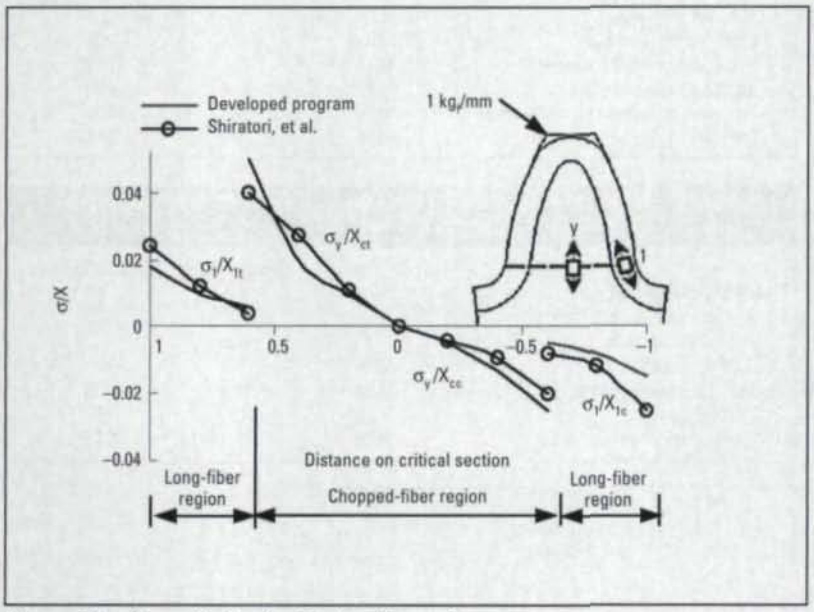

Figure 21—Stress Ratios for Carbon/Epoxy Gear.

Table 7—Material Properties for the Fibers.

		S-Glass	Carbon	Kevlar	Boron
Longitudinal Elastic Modulus	MPa	85,600	235,000	135,000	385,000
Transverse Elastic Modulus	MPa	85,600	16,550	5,200	385,000
Longitudinal Shear Modulus	MPa	5,170	14,400	2,890	166,500
Longitudinal Poisson's Ratio		0.22	0.2	0.35	0.21
Tensile Strength	MPa	4,200	2,599	3,445	3,900
Compressive Strength	MPa	520	2,500	551	4,550
Shear Strength	MPa	350	400	275	700

Table 8—Material Properties for the Matrix (Epoxy).

Elastic Modulus	MPa	4,200
Shear Modulus	MPa	1,340
Poisson's Ratio		0.38
Tensile Strength	MPa	52
Compressive Strength	MPa	130
Shear Strength	MPa	85

reinforced and chopped-fiber reinforced regions on gear strength, stress analyses are made for s-glass/epoxy, carbon/epoxy, kevlar/epoxy and boron/epoxy gears. For chopped-fiber reinforced regions, fiber volume ratios (V_{cf}) of 0.1, 0.3, 0.5 and 0.7 are used; for the long-fiber reinforced region, fiber volume ratios (V_{lf}) of 0.1, 0.3 and 0.5 are used. The same fiber and matrix materials are used for long-fiber reinforced and chopped-fiber reinforced regions.

Material properties for the fibers and the matrix are given in Tables 7 and 8, respectively. As a measure of the strength, calculated minimum failure index values together with the critical sections at which they are observed are given for s-glass/epoxy, carbon/epoxy, kevlar/epoxy and boron/epoxy gears in Tables 9–12.

Fiber volume ratio combinations, which result in the weakest and the strongest gears, are shown in Tables 13 and 14, respectively. As seen in the tables, for different materials, different fiber volume ratios should be used in order to obtain the strongest possible gear.

Discussion and Conclusion

A method and a computer program have been developed for 3-D finite element analysis of long-fiber reinforced composite spur gears, in which long fibers are arranged along tooth profiles. The developed program is verified by performing stress analysis on isotropic gears. Results of the program are compared to the results given for composite gears by Shiratori, et al. (Ref. 8), and good agreement is observed. Although the program is capable of using three different tooth models—namely single tooth, one full tooth with tooth spaces at both sides and three-tooth models—it is found that results given by the single-tooth model are satisfactory for stress analysis purposes.

For glass/epoxy gears, effects of reinforcing thickness of the long-fiber reinforced region and insertion depth of the fibers on gear strength are investigated. It is found that strength increases with increasing reinforcing thickness until a critical thickness is reached. Beyond the critical thickness, reinforcing thickness has no positive effect on gear strength. On the other hand, increased insertion depth has a small positive effect on gear strength.

Finally, in order to see the effects of different materials and different fiber volume ratios for long-fiber reinforced and chopped-fiber reinforced regions on gear strength, stress analyses are made for s-glass/epoxy, carbon/epoxy, kevlar/epoxy and boron/epoxy gears. It is found that the reinforcement with kevlar/epoxy and carbon/epoxy is effective in improving the tooth strength, with kevlar/epoxy gears being the strongest. For the gears made of boron/epoxy, the critical section is mainly section 3, where the critical stress component is the transverse tensile stress.

A general trend observed is: Increasing the fiber volume ratios results in fiber failures in the direction normal to the fibers, i.e. fiber separation.

Acknowledgments

The authors would like to thank the CAD/CAM/Robotics Center of Middle East Technical University for providing computer facilities. ○

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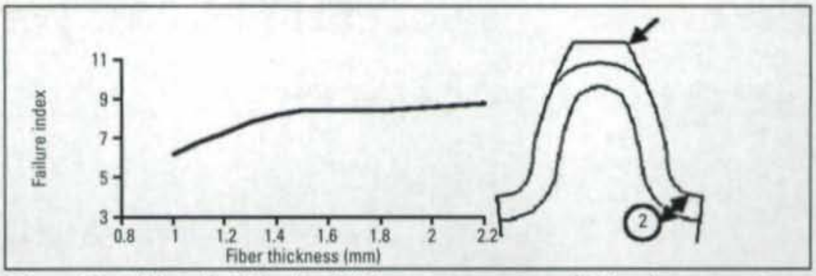


Figure 22—Effect of reinforcing thickness on gear strength (glass/epoxy gear).

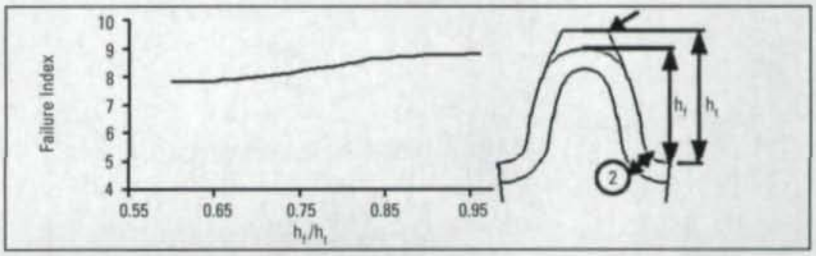


Figure 23—Effect of insertion depth on gear strength (glass/epoxy gear).

Table 9—Minimum Failure Indexes and Critical Sections for S-Glass/Epoxy Gear.

		V_{cf} 0.1	0.3	0.5	0.7
V_{ff}	0.1	3.90, 1*	4.58, 1	4.85, 1	4.95, 1
	0.3	4.63, 1	4.89, 1	4.68, 1	4.19, 1
	0.5	3.40, 2	3.99, 3	3.80, 3	3.27, 3

*(Minimum Failure Index: 3.90, Critical Section: 1)

Table 10—Minimum Failure Indexes and Critical Sections for Carbon/Epoxy Gear.

		V_{cf} 0.1	0.3	0.5	0.7
V_{ff}	0.1	5.99, 1	5.79, 2	5.85, 2	5.93, 2
	0.3	6.53, 1	6.10, 2	4.71, 2	4.73, 2
	0.5	5.97, 2	4.63, 3	4.55, 3	4.47, 3

Table 11—Minimum Failure Indexes and Critical Sections for Kevlar/Epoxy Gear.

		V_{cf} 0.1	0.3	0.5	0.7
V_{ff}	0.1	5.20, 1	5.99, 1	5.98, 3	5.83, 3
	0.3	6.31, 1	6.26, 1	5.99, 3	5.85, 3
	0.5	7.52, 1	6.44, 3	6.11, 3	6.02, 3

Table 12—Minimum Failure Indexes and Critical Sections for Boron/Epoxy Gear.

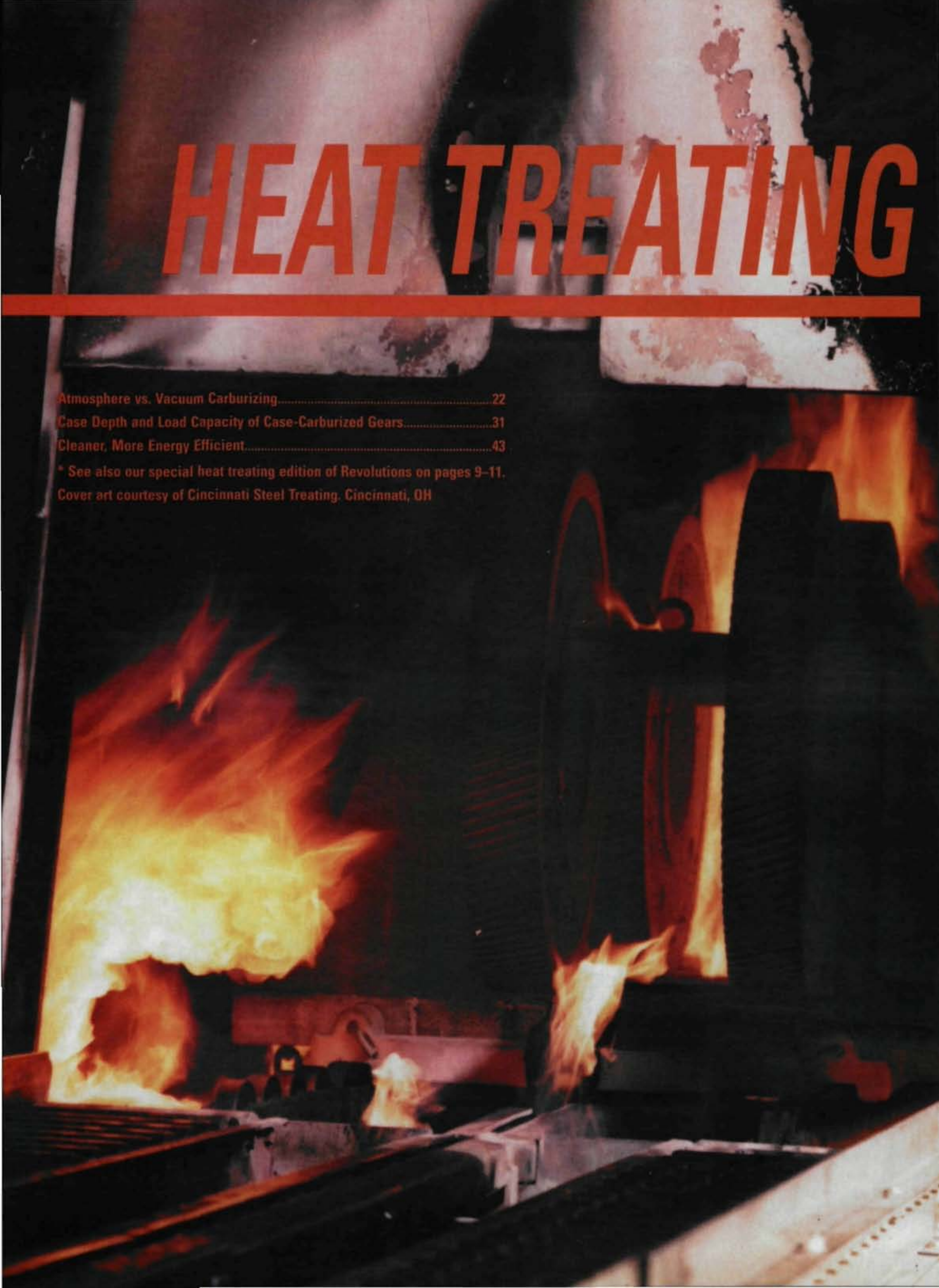
		V_{cf} 0.1	0.3	0.5	0.7
V_{ff}	0.1	5.10, 3	4.90, 3	4.40, 3	4.10, 3
	0.3	4.32, 3	3.94, 3	3.50, 3	3.94, 3
	0.5	3.80, 3	3.62, 3	3.20, 3	3.10, 3

Table 13—Fiber Volume Ratio Combinations which Result in the Weakest Gears.

	S-Glass/Epoxy	Carbon/Epoxy	Kevlar/Epoxy	Boron/Epoxy
V_{ff}	0.5	0.5	0.1	0.5
V_{cf}	0.7	0.7	0.1	0.7
Failure Index	3.27	4.47	5.20	3.10
Critical Section	3	3	1	3

Table 14—Fiber Volume Ratio Combinations which Result in the Strongest Gears.

	S-Glass/Epoxy	Carbon/Epoxy	Kevlar/Epoxy	Boron/Epoxy
V_{ff}	0.1	0.3	0.5	0.1
V_{cf}	0.7	0.1	0.1	0.3
Failure Index	4.95	6.53	7.52	4.90
Critical Section	1	1	1	3



HEAT TREATING

* See also our special heat treating edition of Revolutions on pages 9–11.

Cover art courtesy of Cincinnati Steel Treating, Cincinnati, OH

Atmosphere vs. Vacuum Carburizing

Gerald D. Lindell, Daniel H. Herring, David J. Breuer and Beth S. Matlock

Residual stress and microhardness data are used to compare atmosphere and vacuum carburizing of a low-alloy gear steel, and the effects of high-pressure gas quenching and post-heat treat grinding and shot peening are investigated.

Fig. 1 — These Twin Disc transmissions incorporate 14 to 16 large gears like the one shown above at right. The six-pitch helical gear is for one of the three 205 mm (8 in.) clutch pack assemblies in a TD-61-1175 transmission (right). The SAE 8620RH gear is carburized to 0.81 mm (0.032 in.) min. finished effective case depth at 60 to 64 HRC surface hardness. The TD-61-1175 transmission is used in airport crash-fire vehicles. It's rated at 400 kW (540 hp) at 2,300 max. rpm. The 2600 series transmission, above, is used in Versatile 1150 series tractors. It's rated at 520 kW (700 hp) at 2,200 max. rpm. Photos courtesy Twin Disc Inc.

In recent years, improvements in the reliability of the vacuum carburizing process have allowed its benefits to be realized in high-volume, critical component manufacturing operations. The result: parts with enhanced hardness and mechanical properties.

The purpose of the study described in this article was to investigate whether vacuum carburizing could be used to improve fatigue life. Fatigue is a major cause of gear failure, where the primary failure modes are gear tooth root bending and tooth pitting.

AISI 8620 Steel Studied

Twin Disc Inc. is a world-class supplier of heavy-duty transmissions and related equipment for off-road vehicles (Fig. 1).

Gears are an integral part of these assemblies, and they are carburized to produce a hard, fatigue-resistant case supported by a lower strength, ductile core.

The manufacturing of high-quality transmission gearing involves careful consideration of a number of critical factors, including component design, material, heat treatment, and the influence of subsequent manufacturing operations such as shot peening and grinding.

The methods used to compare the vacuum and atmosphere carburizing processes in this study were X-ray diffraction and microhardness testing.

Coupons of AISI 8620 low-alloy steel were heat treated using the different carburizing methods and subjected to identical post-heat treat grinding and shot peening operations.

X-ray diffraction was selected because it can be used to measure residual stresses. Residual stresses are additive with applied stress, which makes their level an important factor in fatigue-critical components such as gears. Residual compressive stresses are desirable because they oppose the applied, repetitive, and undesirable tensile stresses that cause fatigue failure. For gears, the areas of most concern are the flanks, which are subjected to contact loads that could cause pitting fatigue, and the roots, which experience tensile bending fatigue loads.

The greater the magnitude and depth of residual compressive stress, the greater the ability to improve fatigue properties. To enhance resistance to fatigue crack initiation, it is particularly important to have a higher compressive stress level at the outer surface. Also note that a deeper layer of compressive stress provides resistance to fatigue crack growth for a longer time than a shallower layer.

Carburizing process. Carburizing of a metal surface is a function of both the rate of carbon absorption into the steel and the diffusion of carbon away from the surface and into the interior of the part. Once a high concentration of carbon has developed at the surface, during what is commonly called the "boost stage," the process normally introduces a "diffuse stage," where solid-state diffusion occurs over time. This step results in a change in the carbon concentration gradient between the carbon-rich surface and the steel's interior. The result is a reduction of carbon concentration at the surface of the part accompanied by an increase in the depth of carbon absorption.

The carburization process also induces desirable residual compressive stresses through the case hardened layer. This stress state results from the delayed transformation and volume expansion of the carbon-enriched surface of the steel.

Atmosphere Carburizing

In atmosphere carburizing, parts are heated to austenitizing temperature in a "neutral" or "carrier" gas atmosphere that contains approximately 40% hydrogen (H_2), 40% nitrogen (N_2), and 20% carbon monoxide (CO). Small percentages of carbon dioxide (CO_2, up to 1.5%), water vapor (H_2O, up to 1%), and methane (CH_4, up to 0.5%), along with trace amounts of oxygen (O_2), also are present. The carburizing process also requires the addition of a hydrocarbon enriching gas, usually natural gas.

Of the 180 chemical equations that describe the reactions occurring during atmosphere carburizing, one of the most important is the water-gas reaction:

$$CO + H_2O = CO_2 + H_2 \qquad (1)$$

Control of the atmosphere carburizing process is done by looking at the CO/CO_2 and H_2O/H_2 ratios of this equation using instruments such as dew point analyzers, infrared analyzers, and oxygen (carbon) probes.

In atmospheres containing CO and H_2, carbon transfer is dominated by the CO adsorption (ad) and the oxygen desorption reactions:

$$CO \rightarrow CO_{ad} \rightarrow [C] + O_{ad} \qquad (2)$$

$$O_{ad} + H_2 \rightarrow H_2O \qquad (3)$$

These two reactions yield an alternate form of the water-gas reaction:

$$CO + H_2 = [C] + H_2O \qquad (4)$$

Thus, the transfer of carbon in atmospheres containing CO and H_2 is connected with a transfer of oxygen, giving rise to an oxidation effect in steel containing oxide-forming alloying elements such as silicon, chromium and manganese. This phenomenon is known as internal or intergranular oxidation of steel.

Figure 2 shows hardness profiles for an atmosphere-carburized and oil-quenched AISI 8620 steel gear.

Atmosphere carburizing to a depth of 0.36 mm (0.014 in.) produced a hardness of 58 HRC at both the gear tooth pitch line and root. From this depth, the hardness values quickly diverge. The effective case depth (at 50 HRC) is 0.76 mm (0.030 in.) in the root and 1.33 mm (0.0525 in.) at the pitch diameter. These values are typical of the vast majority of carburized gears currently in service.

For resistance to bending fatigue, it is desirable to achieve a deeper case in the root. This produces a deeper level of high-hardness, high-strength material with the benefit of residual compressive stress.

Vacuum Carburizing

Vacuum carburizing or low-pressure carburizing, by comparison, does not use a "carrier gas" atmosphere, but instead uses vacuum pumps to remove the atmosphere from the chamber before the process begins. For carburizing to take place in a vacuum furnace, all that's needed is a small, controlled addition of a hydrocarbon gas.

Unlike atmosphere carburizing, the breakdown of hydrocarbons in vacuum carburizing is via nonequilibrium reactions. This means that the carbon content at the surface of the steel is very rapidly raised to the saturation level of carbon in austenite. By repeating the boost and diffuse steps, any desired carbon profile and case depth can be achieved.

Today, vacuum carburizing is best performed using low-pressure techniques under 20 torr (25 mbar), and typically at tempera-

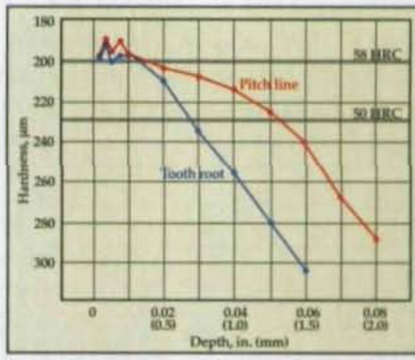


Fig. 2 — Microhardness profiles at pitch line and tooth root for an atmosphere-carburized and oil-quenched AISI 8620 gear. For resistance to bending fatigue, it is desirable to achieve a deeper case in the root.

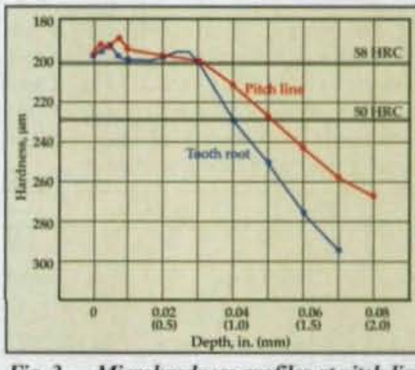


Fig. 3 — Microhardness profiles at pitch line and tooth root for a vacuum-carburized and oil-quenched AISI 8620 gear. The overall case depth of maximum hardness is deeper than that of the atmosphere-carburized part in Fig. 2.

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tures between 790 and 1,040°C (1,475 and 1,900°F). Hydrocarbon gases currently being used for vacuum carburizing are acetylene (C_2H_2), propane (C_3H_8), and, to a lesser degree, ethylene (C_2H_4). Methane (CH_4) is not used because it is nearly nonreactive at these low pressures, unless the temperature is at or above 1,040°C (1,900°F).

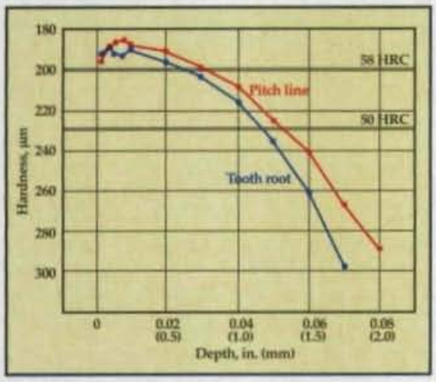

Fig. 4 — Microhardness profiles at pitch line and tooth root for a vacuum carburized and high-pressure gas quenched AISI 8620 steel gear. Use of gas quenching instead of oil quenching (Fig. 3) results in a more uniform case depth between pitch line and root.

Carbon is delivered to the steel surface in vacuum carburizing via reactions such as these:

$$C_2H_2 \rightarrow 2C + H_2 \qquad (5)$$

$$C_3H_8 \rightarrow CH_4 + C_2H_4 \rightarrow C + 2CH_4 \qquad (6)$$

$$C_2H_4 \rightarrow C + CH_4 \qquad (7)$$

In the past, propane has been the primary hydrocarbon gas used for vacuum carburizing; however, propane dissociation occurs before the gas comes in contact with the surface of the steel, thus producing free carbon or soot. This uncontrolled soot formation results in poor carbon transfer to the part and loss of up-time productivity due to the need for additional heat treat equipment maintenance.

Development work done in the past few years has demonstrated that acetylene is a good performing gas for vacuum carburizing. This is because the chemistry of acetylene (Eq. 5) is vastly different from that of propane or ethylene (Eq. 6 and 7). Dissociation of acetylene delivers two carbon atoms to the one produced by dissociation of either propane or ethylene and avoids formation of nonreactive methane.

Control of the vacuum carburizing process is on a time basis. Carbon transfer rates are a function of temperature, gas pressure, and gas flow rate. Simulation programs have been written to determine the boost and diffuse times of the cycle.

Figure 3 shows hardness profiles for a vacuum-carburized and oil-quenched AISI 8620 steel gear.

The overall case depth of maximum hardness for the vacuum carburized part is noticeably deeper than that of the atmosphere carburized part in Fig. 2. The vacuum carburized case depth of approximately 0.81 mm (0.032 in.) at 58 HRC is more than double that obtained with atmosphere carburizing, while the effective case depths (depth at 50 HRC) are similar. Also note the much greater consistency in root and pitch line hardnesses through a depth of 0.81 mm (0.032 in.) for vacuum carburizing vs. atmosphere carburizing (Fig. 3 vs. Fig. 2).

The hardness profiles shown in Fig. 4 are for an AISI 8620 steel gear that has been vacuum carburized and then high-pressure gas quenched in 20-bar nitrogen.

A comparison of Figures 3 and 4 shows that use of high-pressure gas quenching instead of oil quenching in vacuum carburizing results in a more uniform case depth between gear pitch line and root. The absence of a vapor layer in gas quenching results in a more uniform cooling rate along the gear tooth and root profile.

Test Procedure Outlined

The following procedure was used to properly evaluate the effect of different heat treatments and post-heat treatment processes on residual stress in coupons of AISI 8620 low-alloy gear steel.

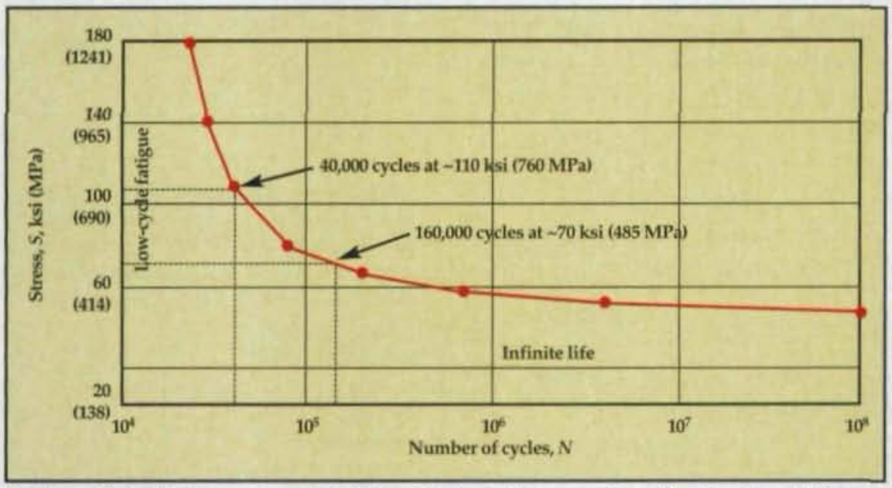

Fig. 5 — Typical S-N curve, or plot of (tensile) stress, S, vs. number of load cycles, N. The primary purpose of shot peening gears is to enhance their fatigue life by inducing a high residual compressive stress at the surface of the tooth roots. Shot peening is most effective for parts subject to high-cycle fatigue loading (>10^4 to 10^5 cycles).

Table 1 — Test coupon manufacturing processes

Coupon	ID	Process
1	EX2470	Vacuum carburize (VC)
2	EX2470-1	Vacuum carburize and shot peen (VC&SP)
3	EX2470-2	Atmosphere carburize (AC)
4	EX2470-3	Atmosphere carburize and shot peen (AC&SP)
5	EX2470-4	Vacuum carburize and dual shot peen (VC&DSP)

Table 2 — Test parameters for atmosphere and vacuum carburized coupons

Parameter	Atmosphere	Vacuum
Temperature, °C (°F)	940 (1,725)	940 (1,725)
Boost time, min	300	32
Diffusion time, min	120	314
Hardening temperature, °C (°F)	845 (1,550)	845 (1,550)
Quenching method	Oil at 60°C (140°F)	Nitrogen gas at 20 bar
Tempering temperature, °C (°F)	175 (350)	175 (350)
Tempering time, h	2	2

• Five coupons from the same heat lot of AISI 8620 were cut to size: 76 X 19 X 13 mm, ± 0.05 mm (3.00 X 0.75 X 0.505 in., ± 0.002 in.). The coupons were stamped, and a separate manufacturing process was defined for each (Table 1).

• Coupons were sent out for heat treatment—vacuum or atmosphere carburizing—according to the parameters in Table 2. The required surface hardness was 59 to 61 HRC. Vacuum carburized coupons were nitrogen gas quenched, while atmosphere carburized coupons were oil quenched.

• Heat treated coupons were ground to 12.7 mm ± 0.013 mm (0.5000 in. ± 0.0005 in.), removing no more than 0.15 mm (0.006 in.) from the nonstamped side where X-ray diffraction was to take place.

• Three of the five coupons were sent out for shot peening.

• All five coupons were sent out for X-ray diffraction on the nonstamped side.

Shot Peening's Benefits

The primary purpose of shot peening gears is to enhance their fatigue life by inducing a high residual compressive stress at the surface of the tooth roots. Shot peening is most effective for parts subject to high-cycle fatigue loading.

A basic explanation is provided by the graph in Fig. 5, a typical *S-N* curve. It plots (tensile) stress, *S*, vs. the number of load cycles, *N*. It is important to note that the vertical scale is linear, whereas the horizontal scale is logarithmic. This means that as tensile stress is reduced, fatigue life improves exponentially. A reduction of stress from 760 MPa (110 ksi) to 485 MPa (70 ksi) results in an improvement in fatigue life from 40,000 cycles to 160,000 cycles (400%). Additional reductions in tensile stress result in significantly more fatigue enhancement. At 415 MPa (60 ksi), for example, the anticipated fatigue life is approximately 400,000 cycles.

The residual compressive stresses produced by shot peening counteract applied tensile stresses. The compressive stresses are induced by impacts of small, spherical media (shot). The impact of each individual shot stretches the surface enough to yield it in tension. Because the surface cannot fully restore itself due to the mechanical yielding that has taken place, it is left in a permanent compressed state.

Shot peening results in a residual compressive stress at the surface—where most fatigue cracks initiate—that is approximately 55 to 60% of the material's ultimate tensile strength. For carburized gears, the surface compression is typically 1,170 to 1,725 MPa (170 to 250 ksi), which results in a significant improvement in fatigue properties.

Grinding Often Overlooked

The grinding process is applied to components so often and in so many forms (automatic, manual, with and without coolant) that it is often overlooked from a residual stress standpoint. However, its influence should not be discounted, especially when dealing with fatigue-critical parts.

During grinding, residual tensile stress may be created from generation of excessive, localized heat. The localized surface area being ground heats from friction and attempts to expand, but can't because it is surrounded by cooler, stronger metal. If the temperature generated from grinding is high enough, however, the metal yields in compression due to the resistance to its expansion and reduced mechanical properties at elevated temperature. Upon cooling, the yielded material attempts to contract. The surrounding material resists this contraction, thus creating residual tensile stress. Because heat is the major cause of residual tensile stress from grinding, the importance of coolant for controlling these stresses is paramount.

X-Ray Diffraction Measures Stresses

X-ray diffraction was used to measure the residual stresses at surface and subsurface locations. The technique measures strain by measuring changes in atomic distances. It is a direct, self-calibrating method that measures tensile, compressive, and neutral strains equally well. Strains are converted to stresses by multiplying by elastic constants appropriate for the alloy and atomic planes measured.

For this study, chromium Kα radiation was used to diffract the (211) planes at approximately 156° 2θ. The area measured was nominally 4 mm (0.16") in diameter. Since only a few atomic layers are measured, the technique is considered a surface analysis technique. The subsurface measurements

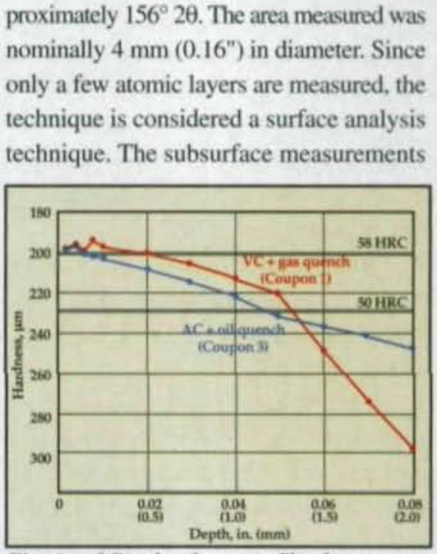

Fig. 6 — Microhardness profiles for vacuum carburized and gas quenched (Coupon 1) and atmosphere carburized and oil quenched (Coupon 3) AISI 8620 steel coupons. A major advantage of vacuum carburizing is a deeper case of high hardness.

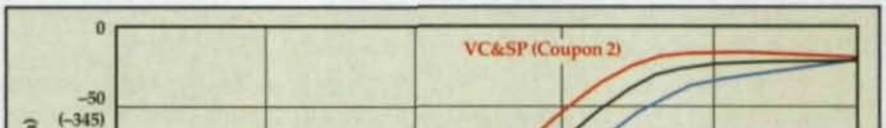

Fig. 7 — Residual stress distributions in the three carburized, ground, and shot peened coupons (Coupons 2, 4, and 5) of AISI 8620. Each boasts a solid layer of compression that implies excellent resistance to initiation and growth of fatigue cracks. Note that the residual stress curve for dual-peened Coupon 5 is ~0.025 to 0.05 mm (0.001 to 0.002 in.) deeper than those for the single-shot-peened coupons.

Table 3 — Comparison of atmosphere and vacuum carburizing results

Property	Atmosphere	Vacuum
Depth to 58 HRC, mm (in.)	0.20 (0.008)	0.58 (0.023)
Surface hardness before grinding, HRC	59	60
Surface hardness after removal of 0.1 mm (0.004 in.) stock by grinding, HRC	58	62

were made by electrochemically removing small amounts of material. These subsurface measurements were subsequently corrected for stress gradient and layer removal effects using standard analytical calculations.

Comparing the Processes

Hardness profiles for vacuum carburized (Coupon 1) and atmosphere carburized (Coupon 3) AISI 8620 steel coupons are compared in Fig. 6. A major advantage of vacuum carburizing over atmosphere carburizing is a deeper case of high hardness. Hardness values for the two carburizing processes are given in Table 3.

Effect of peening. Residual stress distributions in the three carburized, ground, and shot peened coupons—Coupons 2, 4, and 5—are plotted in Fig. 7.

From a fatigue standpoint, the solid layer of compression demonstrated for all three coupons implies excellent resistance to initiation and growth of fatigue cracks. The tensile stress required for a fatigue crack to develop must first overcome compressive stresses that are approximately 1,035 MPa (150 ksi) at the surface and approximately 1,515 MPa (220 ksi) at 0.05 mm (0.002 in.) below the surface. A tensile stress of 1,035 MPa (150 ksi) will produce a net stress of 0 MPa (0 ksi) at the surface when added to the residual compressive stress.

Coupons 2 and 4 were shot peened at an Almen intensity of 14 to 16. (Almen intensity is a measure of the energy of the shot stream.) The steel shot had a hardness of 55 to 62 HRC and a nominal diameter of 0.58 mm (0.023 in.).

The residual stress curves in Fig. 7 have shapes typical of shot peened material. All three have a similar maximum compressive stress of approximately 1,515 MPa (220 ksi). This value is approximately 55 to 60% of the steel's ultimate tensile strength at the surface. Because all three coupons were hardened to 59–62 HRC, they also had similar tensile strengths (at the surface).

The depth of the compressive stress layer is a function of the Almen intensity. It can be increased by increasing shot size and/or velocity. The depth is the location where the residual stress vs. depth curves would cross the neutral axis (into tension) if the positively sloped lines were extended. A greater depth of compression is desired because this layer is what resists fatigue crack growth. Coupons 2 and 4 were shot peened to the same intensity, so that the depth of their compressive stress layers is also essentially the same at approximately 0.18 to 0.20 mm (0.007 to 0.008").

Dual peening. The trade-off to increasing shot peening intensity is that there is additional cold work and material displacement at the point of shot impact. This generally results in less compression right at the surface (depth = 0) and a more aggressive surface finish.

Dual peening is performed to make up for the reduced compression resulting from high-intensity peening. The technique consists of shot peening the same surface twice—peening at a higher intensity is followed by peening at a lower intensity, usually with smaller media. The second peening reduces the degree of cold work at the surface, improving the surface finish, which, in turn, makes the surface more compressed.

Coupon 5 was dual peened. The process specified MI-230H shot at 18 to 20 Almen followed by MI-110H shot at 8 to 10 Almen. The residual stress curve for this coupon (Fig. 7) is approximately 0.025 to 0.05 mm (0.001 to 0.002") deeper than the curves for the coupons single shot peened at 14 to 16 Almen using MI-230H shot.

This would be expected for a carburized gear steel. The surface stress of Coupon 5 (at depth = 0) is the same as that of the other two shot peened coupons. What most likely occurred is that it was less compressed after the first peening step. When the second was performed, the surface became even more compressed, to the approximately 930 MPa (135 ksi) level shown in Fig. 7. Therefore, Coupon 5 would be expected to have the best fatigue performance of the three because it has the most compressive stress throughout its depth. This is particularly evident between 0.08 and 0.20 mm (0.003 and 0.008") below the surface. At 0.10 mm (0.004") below the surface, for example, there is still 1,380 MPa (200 ksi) of compression for Coupon 5, compared with 1,170 MPa (170 ksi) for Coupon 4 and 1,000 MPa (145 ksi) for Coupon 2.

Effect of grinding. Testing of coupons that had been ground gave unexpected results that required further investigation. All coupons were ground at the same time. Grinding was performed using a wheel that had coolant flow. The operator was instructed to remove no more than 0.025 mm (0.001") of stock per pass, for a total of 0.10 mm (0.004") of material removed from each coupon. This appeared to be acceptable grinding practice, and little thought was given to the technique prior to testing.

X-ray diffraction measurements indicated that tensile stresses existed on the surface of the vacuum carburized coupon (Coupon 1) as high as 255 MPa (37 ksi) at 0.013 mm (0.0005") below the surface. At a depth of approximately 0.10 mm (0.004"), the values crossed the neutral axis into compression. These results were immediately questioned. However, retesting at several locations using X-ray diffraction verified that the original values were correct.

The explanation lies in the fact that additional heat was generated when grinding vacuum carburized Coupon 1. The coupon was 1 HRC point harder at the surface and 4 HRC points higher after 0.10 mm (0.004") of stock removal. These values are higher than those for the atmosphere carburized specimen (Coupon 3). Additional heat from an increase in friction resulted in the generation of residual tensile stresses on the vacuum carburized coupon.

This is an excellent example of why it is important to carefully evaluate the amount of heat generated when grinding fatigue-critical parts. It also demonstrates that X-ray diffraction is an excellent tool for determining the residual stress state of components before they enter service.

Test Results Reviewed

This study compared atmosphere and

vacuum carburizing of AISI 8620 gear steel and evaluated the influence of the subsequent manufacturing operations of shot peening and grinding. The primary goal of the study was to determine which carburizing process was more suitable for heavy-duty transmission gears. Gears are subject to both sliding and rolling contact stresses on their flanks in addition to bending stresses in tooth roots. To meet these demanding performance criteria, the steel gears ideally would be hardened for strength and contact properties and have residual compressive surface stresses for bending fatigue resistance.

We concluded that vacuum carburizing is superior to atmosphere carburizing for heat treating heavy-duty transmission gears, and enjoys the following advantages:

• *Higher hardness:* In the coupons tested with X-ray diffraction, the surface, before grinding, was 1 HRC point higher (60 vs. 59 HRC); the subsurface, after 0.10 mm (0.004") stock removal, was 4 HRC points higher (62 vs. 58 HRC).

• *Greater depth of high hardness, ≥ 58 HRC:* In the coupons tested with X-ray diffraction, 58 HRC depths were 0.58 mm (0.023") for vacuum carburizing and 0.20 mm (0.008") for atmosphere carburizing.

• *Greater depth of high hardness, ≥ 58 HRC, at the pitch line and root of actual gears:* 58 HRC depths were 0.81 mm (0.032") for vacuum carburizing and 0.38 mm (0.015") for atmosphere carburizing.

• *Deeper effective case in tooth root of actual gears:* Vacuum, 1.0 mm (0.040"); atmosphere, 0.699 mm (0.0275").

• *Higher surface residual compression (determined by X-ray diffraction of coupons without shot peening or grinding):* Vacuum, 135 MPa (19.6 ksi); atmosphere, 98 MPa (14.2 ksi).

• *Improved consistency between the case layer at the pitch line of the gear flank and gear roots (actual gears):* Vacuum, 0.28 mm (0.011") variation; atmosphere, 0.648 mm (0.0255") variation.

Shot peening. The vacuum carburized and atmosphere carburized surfaces responded equally to shot peening:

• Maximum compressive stress: approximately1,515 MPa (220 ksi).

• Compressive layer depth: approximately 0.18 to 0.20 mm (0.007 to 0.008").

Dual shot peening at first a higher and then a lower intensity resulted in a greater depth of compression by approximately 0.025 to 0.05 mm (0.001 to 0.002"). The surface stress of the dual-peened coupon was very similar, at approximately 930 MPa (135 ksi), to that of the conventionally shot peened coupons. The higher-intensity first peen would have produced a less compressed surface, but the second, lower-intensity peen would have restored compressive stress to the approximately 930 MPa (135 ksi) level. The dual-peened coupon should have significantly better high-cycle fatigue properties than the single-peened coupons.

Fatigue. In terms of fatigue performance, the additional 34.5 MPa (5 ksi) of compression measured for the vacuum carburized coupon (not shot peened or ground) should yield significant increases in gear life under high-cycle fatigue loading, compared with that for the atmosphere carburized coupon.

The study also served as an excellent reminder of the importance of understanding how all manufacturing processes may affect residual stresses and, consequently, fatigue performance. Actual gears must now be tested to ensure that changes to the manufacturing process—involving material, part geometry, heat treatment, shot peening, and/or grinding —will have the same effects in production as those observed in this study.

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This article is based on "An Analytical Comparison of Atmosphere and Vacuum Carburizing Using Residual Stress and Microhardness Measurements," a paper presented by the authors at the 21st ASM Heat Treating Society Conference, held Nov. 5–8, 2001, in Indianapolis, IN.

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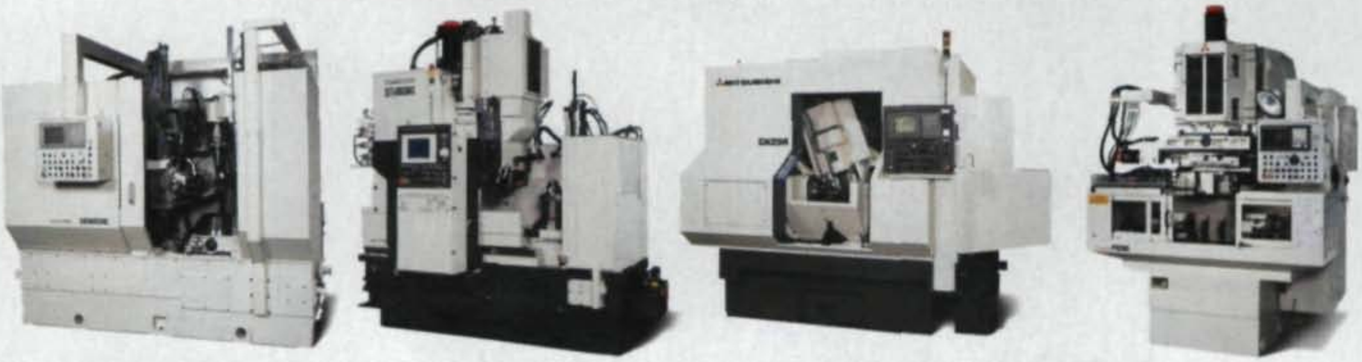

Case Depth and Load Capacity of Case-Carburized Gears

Thomas Tobie, Peter Oster and Bernd-Robert Höhn

Introduction

Compared to non-heat-treated components, case-carburized gears are characterized by a modified strength profile in the case-hardened layer. The design of case-carburized gears is based on defined allowable stress numbers. These allowable stress numbers are valid only for a defined "optimum" case depth. Adequate heat treatment and optimum case depth guarantee maximum strength of tooth flank and tooth root. Variable case depths can lead to a decrease in load capacity. For some applications, including large gears with small modules, maximum load capacity for the tooth flank often cannot be used. Therefore, the optimum case depth is not required. A smaller case depth can meet the load capacity requirement for the actual application without reaching the maximum load capacity and can thereby decrease distortion by hardening and reduce the need for grinding.

For case-carburized gears with adequate case depth, it is generally accepted that pitting cracks are initiated at the surface, where topography of surface and lubricating conditions are important parameters. However, on crack propagation, the stress field of the subsurface region also has an important influence. Furthermore, under special conditions, cracks also can initiate below the surface. The variable stress gradient over depth requires a corresponding gradient of strength. So, to determine an adequate case depth that will ensure pitting resistance, it is essential to know the stress field induced by loading of the tooth flank at the surface, as well as over depth below the surface for all points on the line of contact.

Based on theoretical work and experimental test results, it is planned to introduce an addition to the standardized gear rating according to ISO/DIN (Ref. 2), in which the influence of case depth on load capacity is taken into consideration.

As a gear's tooth flank and tooth root cannot be loaded independently from each other, it can be shown that the simple empirical method—case depth proportional to module—takes the basic principles of the rolling/sliding contact for tooth flank and of a bending beam for tooth root into good consideration for a wide range of standard gears.

The Loaded Tooth Flank—Some Basic Principles of Contact Stresses

Base model of line contact. In standardized rating of gears according to ISO/DIN, computation of pitting resistance is based on the nominal value of Hertzian pressure at the pitch point and, especially for helical gears, on the average length of line of contact. For the calculation of the Hertzian pressure, p_H, the meshing of two gear teeth can be represented by an analog model of the meshing of two cylinders under normal load. Important parameters are the radii of curvature for pinion and gear along the length of path of contact and the values of load and pitch line velocity. Characteristic data of material and lubricant have to be introduced. For helical gears, normal unit load is related to minimum total length of line of contact at the appropriate diameter. Figure 1 gives some basic data for two special test gear pairs. The profile of both gear pairs in transverse plane is equal. For the given gear size, a transmission ratio of approximately one leads to a maximum relative radius of curvature in the pitch point C. Both gear pairs have the same face width. The overlap ratio of the helical gear pair is 1.0 for minimizing excitation. As far as the standardized contact stress number σ_{H0}

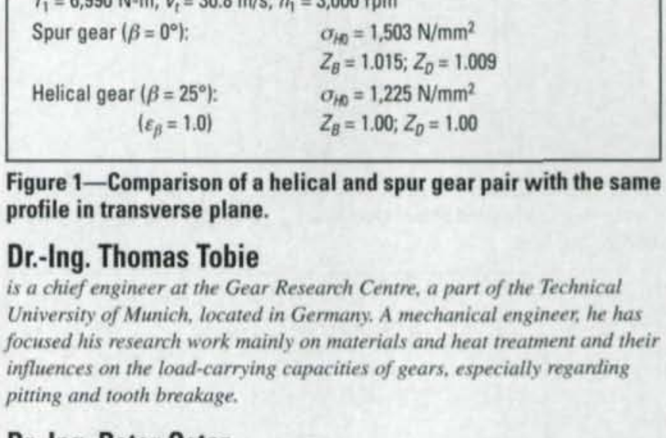

Figure 1—Comparison of a helical and spur gear pair with the same profile in transverse plane.

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Prof. Dr.-Ing. Bernd-Robert Höhn

is head of the Gear Research Centre. Under his leadership, the centre's main research efforts include examination of load-carrying capacity of gear drives, design of gear geometry and testing of gears.

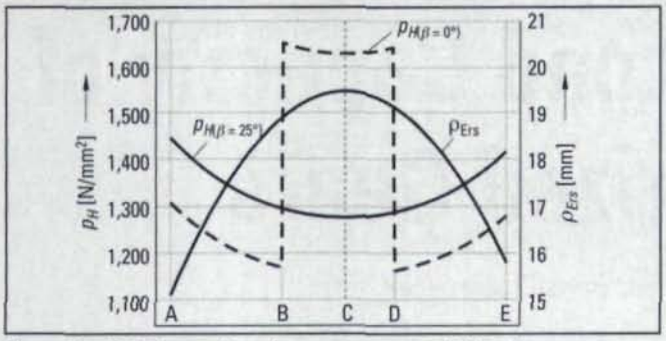

Figure 2—Relative radius of curvature ρ_{Ers} and Hertzian pressure p_H over the line of contact.

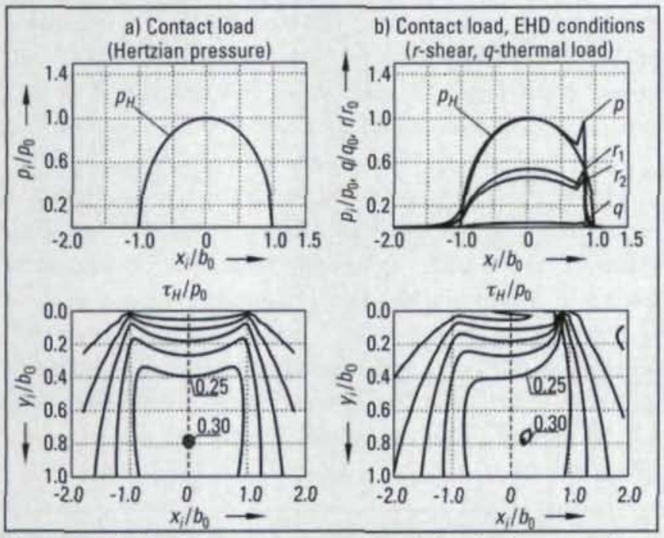

Figure 3—Field of shear stress over depth for the lowest point of single tooth contact of spur pinion regarding elliptical Hertzian pressure (a), EHD pressure (b).

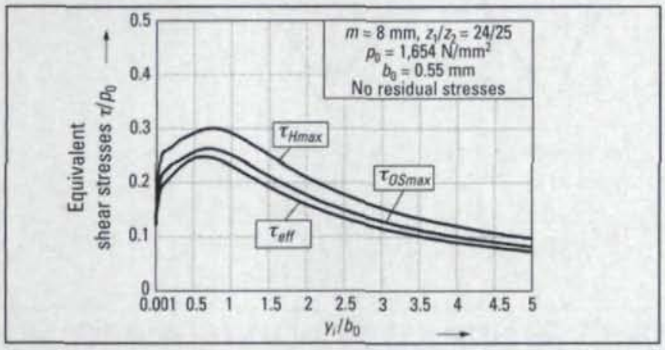

Figure 4—Equivalent shear stresses τ_{Hmax}, τ_{OSmax} and τ_{eff} over depth below surface.

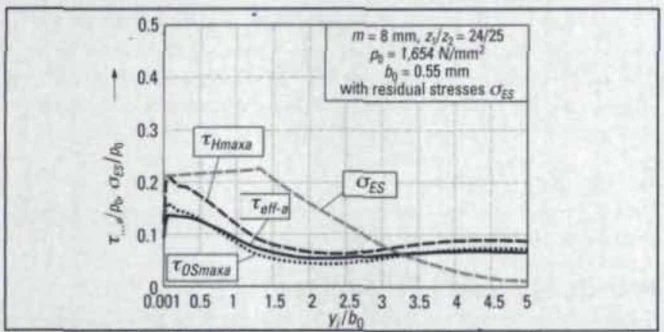

Figure 5—Equivalent shear stress over depth regarding superimposing residual stresses.

according to DIN 3990 (Ref. 2) is concerned, Figure 1 shows that the helical gear is obviously advantageous compared to the geometrically equivalent spur gear. While the helical gear has a constant total length of line of contact, the contact length of the spur gear is significantly characterized by the change of single- to double-tooth contact. Simplifying load distribution, Figure 2 shows the distribution of Hertzian pressure across the path of contact. A is the starting point of mesh at the dedendum flank of the pinion. In the area of small relative radius of curvature ρ_{Ers} and high sliding velocity, the helical gear is loaded with higher Hertzian pressure than the spur gear. The maximum value of Hertzian pressure appears for the spur gear at the lowest point of single tooth contact (point B). With the corresponding local value of load and relative radius of curvature of each contact point, the stress field for line contact can be calculated according to the rules of contact mechanics.

Contact load and contact stresses. For the gear designer, the maximum of the shear stress τ_H or orthogonal shear stress τ_{OS} are two well-known values. Figure 3a shows, for the lowest point of single tooth contact of the spur pinion, the stress field τ_H/p_0 over (material) depth regarding the indicated simple Hertzian pressure. The graphical representation is dimensionless with: p_0—maximum Hertzian pressure in contact point, y—distance below surface of contact, x—coordinate in contact band or coordinate of time, b_0—semi width of Hertzian contact band. During teeth meshing, the elliptically distributed Hertzian pressure moves along the length of path of contact. Thus, the load on each single element in the gear volume varies with time, and the direction of the shear stresses changes as load passes through one contact point (rotation by 180° for one mesh).

Therefore, the x-coordinate can be regarded as the time-axis, and consequently, Figure 3 illustrates the variable stress field above time for the chosen contact point. Instead of the dry contact model, actual gears are exposed to a tribological rolling/sliding contact with local friction and varying temperature. Therefore, the normal load induces a tangential component of load as well as a thermal source at the surface. The distribution of tangential load can be assumed as proportional to the distribution of normal pressure, if the average coefficient of friction μ_m is simplified and assumed to be constant.

For the distribution and value of thermally induced stress, the speed conditions of meshing teeth are the most important influence parameters. The lubricant in the tooth contact also affects the pressure distribution. Assuming elastohydrodynamic conditions, the distribution of pressure, film thickness and film condition across the width of the Hertzian contact band can be computed. The distribution of pressure under EHD conditions is primarily influenced by local lubricant viscosity. Important parameters that have an influence on the viscosity at the contact point are value of load, oil temperature and the type of lubricant. Figure 3b shows the local overall loading for EHD contact and the resulting field of shear stress τ_H for the given contact point of the spur gear. It is obvious that friction and an increase in temperature modify the stress field (for the given gear geometry and load) at and in the near surface-area in a significant way. Additionally, the EHD conditions increase the stress concentration in this area. On the other hand, it is evident that stresses in a depth $y_i > 0.5 \cdot b_0$ are not modified and so values of τ_H are equivalent to the well-known values of loading with pure Hertzian pressure (maximum value of $\tau_H = 0.3 \cdot p_H$ in a depth $y_i = 0.78 \cdot b_0$). The consideration of frictional shear moves the maximum stress in the direction of the end of the Hertzian contact band.

An equivalent stress criterion required for a loaded tooth flank has to take into consideration the multiaxial and variable-with-time stress state of a loaded tooth flank, where the maximum normal and shear stresses occur in different depths below the surface and out of phase.

Equivalent shear stresses. Maximum shear stress criterion (τ_H) and orthogonal shear stress criterion (τ_{OS}), which is proportional to the von Mises equivalent stress, are two basic equivalent stress criteria well known by the gear designer. Figure 4 illustrates for the given loading according to Figure 3b, the values of maximum shear stresses τ_{Hmax} and τ_{OSmax} and the value of the effective shear stress τ_{eff} over depth below surface. By comparing different criteria, it can be noted that τ_{Hmax} as well as τ_{OSmax} are defined as vectors, taking into consideration only the maximum shear stress in a specified plane at a defined point in time. The direction of these stress vectors is variable with time. According to the shear stress intensity method, τ_{eff} is defined as root-mean-square value of all maximum shear stress values $\tau_{y,\varphi}$ in each plane (y,φ) of analysis. Figure 4 points out that for the given gear geometry and load, the profile of the three equivalent stress values is similar, with a maximum value occurring below the surface and the depth and stress value being in the same order of magnitude.

Due to additional residual stresses, the stress distribution shown can be modified quite significantly. Residual stresses are induced by heat treatment and the grinding process. Assuming a biaxial stress state, the normal component of residual stresses is negligible at least in a near-surface region. Therefore, the normal component of residual stresses will be assumed as zero for the following computations.

When superimposing residual stresses, it has to be taken into consideration that residual stresses are more or less constant over time, while stresses induced by external load are variable over time and therefore dynamic components. For the following computations, a variation of the residual stress state due to the running of gears is not considered. The different time profile of load-induced stresses and residual stresses can be considered, if the quasi-static residual stress component—for example, $\tau_{eff\text{-}ES}$—is handled as a kind of mean stress. Then the equivalent stresses (indicated by the subscript a, as in $\tau_{eff\text{-}a}$) represent a threshold value (2 • amplitude), for example: $\tau_{eff\text{-}a} = \tau_{eff} - \tau_{eff\text{-}ES}$, where τ_{eff} is the calculated shear stress intensity induced by external load and residual stresses and $\tau_{eff\text{-}ES}$ is the equivalent

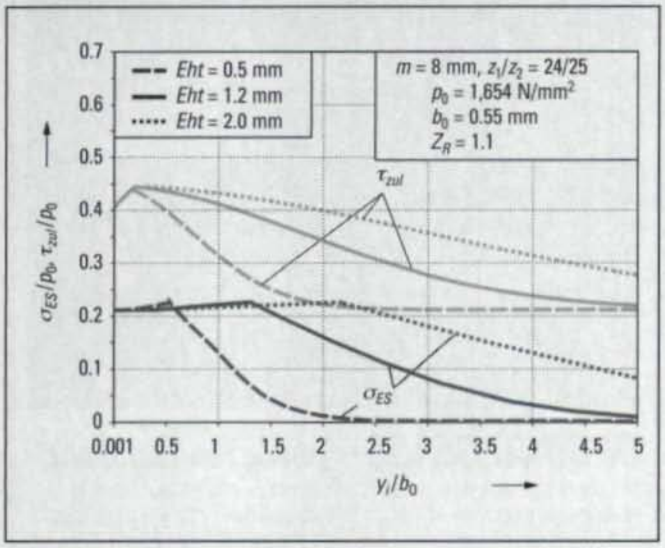

Figure 6—Hardness curve and residual stress profile for different case-depth values.

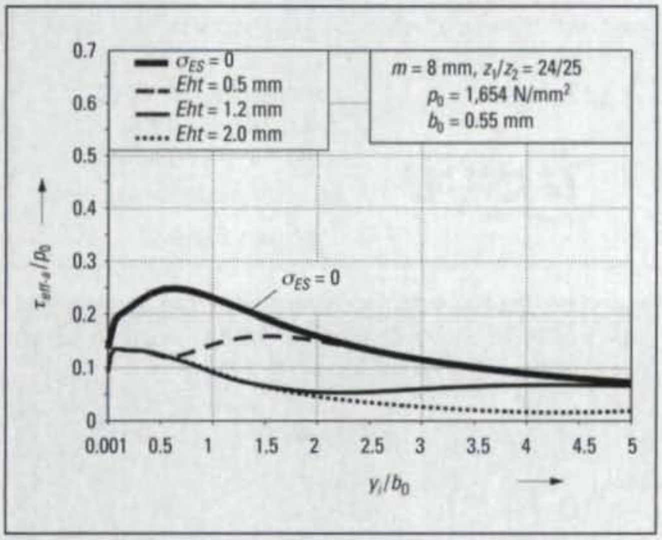

Figure 7—Profiles of $\tau_{eff\text{-}a}$ without residual stresses and with regard to different residual stresses according to Figure 6.

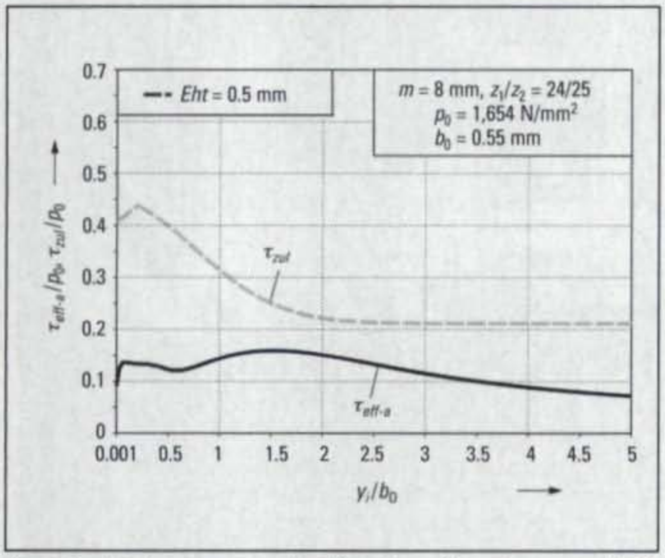

Figure 8—Hardness curve and profile of $\tau_{eff\text{-}a}$ for a given case depth of 0.5 mm.

shear stress induced by residual stresses only.

For the given example, Figure 5 illustrates the gradient of the equivalent shear stress values over depth considering the shown residual stress state (axial and tangential components of ρ_{ES}). Value and distribution of the given residual stresses agree with measurements in gears of this size and adequate case depth. It is obvious that, with consideration of residual stresses, the equivalent stress levels decrease, in the example in a region $y_i < 4 \cdot b_0$, compared to the stress state without regarding residual stresses. For $y_i > 4 \cdot b_0$, the value of the residual stresses is small in this case, so the influence on load-induced stresses decreases. The different equivalent shear stress values are modified in different ways, but one common effect for all criteria is that the maximum value is now to be found at or very near the surface. Furthermore, it can be seen that after a minimum, the stress values increase again for a depth of $y_i > 2.5 \cdot b_0$. According to Reference 4, the average value of equivalent shear stress in a near-surface area ($y_i < 0.1 \cdot b_0$) is defined as "local near-surface stress," $\tau_{eff\text{-}a0}$. For the given loading, the values of $\tau_{eff\text{-}a0}$ are reduced by the residual stresses. A decrease of the residual stress value due to running of the gears results in an increase of $\tau_{eff\text{-}a0}$.

Shear stress/strength gradient for different case depths. Modifying case depth varies the distance below surface where residual stresses influence the stress state. According to Reference 6, a residual stress profile over depth can be calculated according to a standardized hardness curve. The equations are based on test results. For example, Figure 6 illustrates the influence of variable case depths (*Eht*) on the gradient of strength (τ_{zul}) and residual stress profile (σ_{ES}) over depth. It is assumed by simplification that the gradient of strength is equivalent to the gradient of hardness (HV) normal to the surface ($\tau_{zul} = c \cdot HV$, c = 1.0). Values of surface hardness and core hardness are constant. At and in the near-surface region, the material strength is reduced due to notch effects from surface roughness. Figure 7 shows examples of the gradient of $\tau_{eff\text{-}a}$ over depth regarding the different residual stress values for different case depths. It can be seen that the stress gradient is modified over different distances below the surface. Especially for a small value of case depth, a second peak value below the surface occurs. At a greater distance below the surface, an increase in case depth leads to a decrease in stress; while in the near-surface area, the stress state is not decreased additionally. Figures 6 and 7 point out that the thickness of the case-carburized layer and thus the case depth modify the strength profile as well as the stress profile over depth significantly.

The variable stress gradient over depth requires a corresponding gradient of strength.

By comparing local stress and local strength over the whole profile, locations of critical stress/strength ratio can be found. In Figure 8, stress and strength profiles are illustrated for the given example with a case depth of 0.5 mm. Note that this case depth is smaller than recommended values in standards. In this case, it can be seen that a section of critical stress/strength ratio, with respect to minimum safety, is found below the surface at the boundary of the case-core transition. In the case-hardened layer as well as at the surface, the stress/strength ratio is not critical. Figure 9 shows a comparison of stress and strength profile when changing the case depth to 1.2 mm. It is illustrated that stress and strength profile correspond in a much better way over the depth. Maximum stress now occurs near the surface. At a greater distance below the surface, the stress/strength ratio increases, but note that the absolute stress level in these regions is small. However, localized material defects—for example, inclusions—can lead to an increase in stress and initiate a crack below the surface.

Effects of different case-depth values. It was demonstrated that, with consideration of residual stresses, the stress gradient as well as the strength profile over depth are modified with the case-depth value. The depth below the surface of the maximum stress/strength ratio (minimum safety) depends on the correlation of stress and strength profile. Adequate case depth leads to a peak stress value and a critical stress/strength ratio at or just near the surface so that pitting will be initiated in these regions, especially if the special conditions at the surface—for example, notch effects due to surface roughness and decrease of residual stresses due to running of gears—are taken into consideration. These influences modify surface stresses and surface strength. Note that in the examples shown, the different influences on the surface stresses were not taken into special consideration. Smaller values of case depth (or unfavorable residual stresses) can lead to moving the peak value of stress/strength ratio a greater distance below the surface. Thus, gear damage initiated below the surface, especially in regions of critical local stress/strength ratio, may occur. A decrease in load capacity can be imagined. Localized material defects in critically stressed areas increase the risk of damage. On the other hand, lower gear loading can result in lower required case depth.

Thus, for minimizing the risk of tooth flank damage, especially in critical applications, not only conditions at the surface should be regarded. Also, stress and strength profile over depth can be important parameters and should be analyzed by computing a local safety (stress/strength ratio) over depth below the contact. Note that the value of the coefficient c for calculating the material strength—$\tau_{Dul} = c \cdot HV$ with $c = 1.0$—is assumed for simplification and is not based on test results.

The demonstrated theoretical investigations are computed with an EDV-based program system called ROSLCOR, developed, owned and installed by the Technical University of Munich's Gear Research Institute (FZG). Basic principles are summarized in Reference 8.

Application of Test Results in Addition to ISO/DIN Standard

The demonstrated theoretical investigations are in good agreement with some test results run at the FZG for evaluation of the influence of case depth on load capacity. According to Reference 5, an "optimum" case depth guarantees maximum load capacity of tooth flank according to DIN 3990. Smaller values of case depth lead to a decrease in pitting resistance. For the tooth flank, optimum case depth was found as a function of relative radii of curvature. Test results are mainly based on investigations with small gears. Some few test results with larger gears

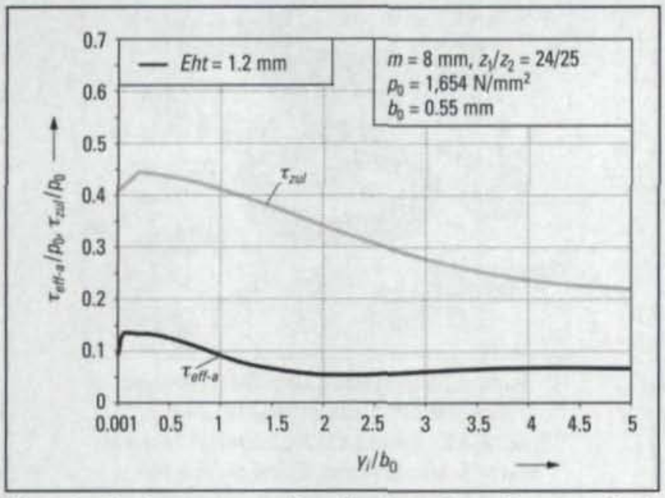

Figure 9—Hardness curve and profile of $\tau_{eff\text{-}a}$ for a given case depth of 1.2 mm.

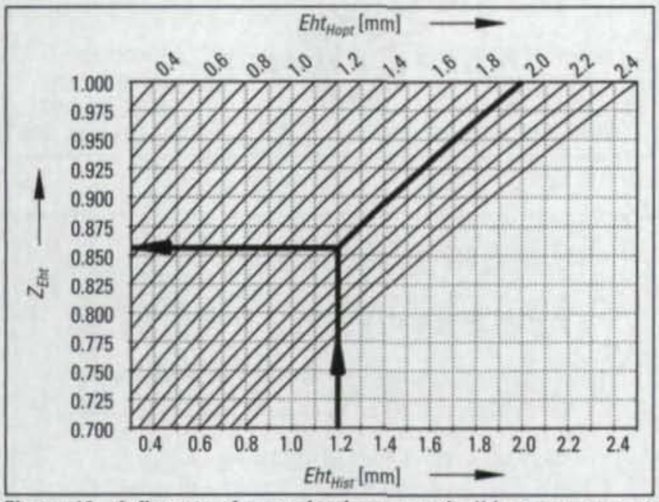

Figure 10—Influence of case depth on permissible contact stress for long life.

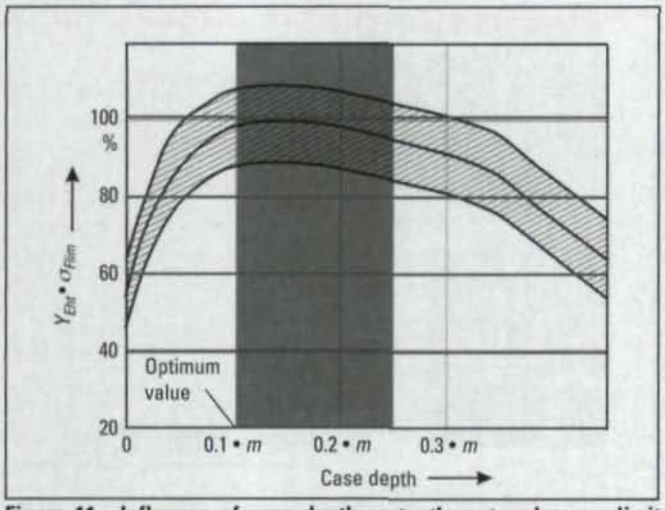

Figure 11—Influence of case depth on tooth root endurance limit (Ref. 7).

(a = 200 mm) according to Reference 3 confirm these results.

For the tooth root endurance strength, there was also an optimum case depth established, which depends on the module of the gear. Investigations show that smaller or greater values than the optimum case depth decrease the tooth root endurance strength.

The German guideline (Ref. 2), which is based on long practical experience, recommends a case depth of 0.15 • normal module, defined as the depth below surface at which the Vickers hardness has dropped to 550 HV, applicable for standard gear sizes. Using this method, the actual load on the gear has not been given special consideration. It is obvious and was demonstrated by theoretical investigations that lightly loaded gears tolerate less case depth than recommended by this rule.

Influence of case depth on contact load capacity. In standardized rating according to ISO/DIN, the allowable stress numbers σ_{Hlim} and σ_{Flim} are valid for a defined optimum case depth. Therefore, an addition to the standard, which is based on the reported test results, is planned to take the influence of variable case depth on load capacity into consideration. For evaluation of the influence of case depth on the permissible endurance contact stress, an influence factor Z_{Eht} (Eq. 1) is defined:

$$Z_{Eht} = \sigma_H \cdot S_H / (\sigma_{Hlim} \cdot Z_W \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_X) \quad (1)$$

where: σ_H is the actual contact stress number, N/mm²,
S_H is the required safety factor,
σ_{Hlim} is the allowable stress number (for optimum case depth), N/mm²,
Z.. are influence factors according to DIN 3990 (Ref. 2).

Within the range $Eht_{Hist} \geq Eht_{Hopt}$: $Z_{Eht} = 1.0$,
Within the range $Eht_{Hist} < Eht_{Hopt}$: Z_{Eht} according to Figure 10 or Equation 2:

$$Z_{Eht} = \sqrt{1 - (170 - 25 \cdot Eht_{Hopt}) \cdot (Eht_{Hopt} - Eht_{Hist})/360} \quad (2)$$

where:
Eht_{Hist} is the actual effective case depth (measured at the reference circle),

Eht_{Hopt} is the calculated optimum effective case depth at the reference circle according to Reference 2.

Z_{Eht} is applicable in a range of $0.7 \leq Z_{Eht} \leq 1.0$.

For example:
$Eht_{Hopt} = 2.0$ mm, $Eht_{Hist} = 1.2$ mm
$\rightarrow Z_{Eht} = 0.856$.

Influence of case depth on bending load capacity. For evaluation of the influence of case depth on the tooth root endurance strength, an influence factor Y_{Eht} (Eq. 3) is defined:

$$Y_{Eht} = \sigma_F \cdot S_F/(\sigma_{Flim} \cdot Y_{ST} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X) \qquad (3)$$

where: σ_F is the actual bending stress number, N/mm^2,
S_F is the required safety factor,
σ_{Flim} is the allowable stress number (for optimum case depth), N/mm^2,
$Y_{..}$ are influence factors according to DIN 3990 (Ref. 2).

Y_{Eht} can be calculated:

• within the range $Eht_{Fist} = 0.025...0.1 \cdot m_n$:

$$Y_{Eht} = 0.5 + \left(\frac{Eht_{Fist}}{m_n}\right)^{\left(0.4 - \frac{Eht_{Fist}}{m_n}\right)}$$

• within the range $Eht_{Fist} = 0.1...0.2 \cdot m_n$:

$$Y_{Eht} = 1.0$$

• within the range $Eht_{Fist} = 0.2...0.35 \cdot m_n$:

$$Y_{Eht} = 1 - 0.8 \cdot \left(\frac{Eht_{Fist}}{m_n} - 0.2\right)$$

• where Eht_{Fist} is the actual (measured) effective case depth at the root fillet normal to the 30° tangent,
Eht_{Fopt} is the calculated optimum effective case depth at the root fillet normal to the 30° tangent according to Figure 11.

For practical application, it can be assumed that:

$Eht_{Fist} \approx 0.75 \cdot Eht_{Hist}$.

The influence of case depth on the tooth root endurance limit is also shown in Figure 11. The proposed influence factors can be used in two ways: either for calculation of required case depth as a function of geometry and load or for calculation of the safety factors (load capacity of the gear) for a given case depth not equal to optimum case depth.

Verification of Different Approaches for Determining Case Depth

A gear's tooth flank and tooth root cannot be loaded independently from each other. Thus, an adequate case depth has to consider the basic principles of rolling/sliding contact for tooth flank as well as of a bending beam for tooth root.

In the following, it is demonstrated that the simple empirical method, case depth proportional to module, recommended by the German guideline, takes the different conditions of tooth flank and tooth root into good consideration for a wide range of standard gears. Furthermore, the results are in good agreement with other methods.

Calculation of case depth according to AGMA Standard. The AGMA standard for gear rating (Ref. 1) recommends a minimum case depth Eht_{min} based on the depth of maximum shear from contact loading (Eq. 4).

$$Eht_{min} = \frac{\sigma_H \cdot d_{w1} \cdot \sin\alpha_{wt}}{U_H \cdot \cos\beta_b} \cdot \frac{z_2}{z_1 + z_2} = 2.2 \cdot s_{\tau H}$$
$$= \frac{2 \cdot \rho_c \cdot \sigma_H}{U_H}\ ;\ u = \frac{z_2}{z_1} \qquad (4)$$

Eht_{min} depends on the actual load on the tooth flank and the geometry of the gear. U_H is a hardening process factor (U_H = constant = 66,000 N/mm^2 for grades MQ and ME carburized and hardened). Transforming the given formula, Eht_{min} is proportional to the depth below surface $s_{\tau H}$ where maximum shear stress τ_{Hmax} occurs, with respect to relative radii of curvature and applied load.

Using the fundamental rating formulas of DIN 3990 (Eqs. 5 and 6) (Ref. 2), some transformations lead to Equation 7.

$$\sigma_H = \sigma_{H0} \cdot \sqrt{K_A K_v K_{H\alpha} K_{H\beta}} \le \sigma_{HP} = \frac{\sigma_{Hlim}}{S_{Hmin}} \cdot Z_W Z_L Z_R Z_X Z_v$$
$$\sigma_{H0} = Z_H Z_E Z_B Z_\varepsilon Z_\beta \cdot \sqrt{\frac{F_t}{d_1 b}\frac{u+1}{u}} = Z_H Z_E Z_B Z_\varepsilon Z_\beta \cdot \sqrt{\frac{F_t}{b}\frac{\cos\beta}{z_1 m_n}\frac{u+1}{u}}$$
$$\Rightarrow \sigma_{H0} = Z_H Z_E Z_B Z_\varepsilon Z_\beta \cdot \sqrt{\frac{F_t}{b}\frac{\cos\beta}{z_1 m_n}\frac{u+1}{u}} = \frac{\sigma_{Hlim}}{S_H} \cdot \frac{Z_W Z_L Z_R Z_X Z_v}{\sqrt{K_A K_v K_{H\alpha} K_{H\beta}}} \qquad (5)$$

$$\sigma_F = \sigma_{F0} \cdot K_A K_v K_{F\alpha} K_{F\beta} \le \sigma_{FP} = \frac{2\sigma_{Flim}}{S_{Fmin}} \cdot Y_X Y_{\delta relT} Y_{RrelT}$$
$$\Rightarrow \sigma_{F0} = \frac{F_t}{b m_n} \cdot Y_{FS} Y_\varepsilon Y_\beta = \frac{2\sigma_{Flim}}{S_F} \cdot \frac{Y_X Y_{\delta relT} Y_{RrelT}}{K_A K_v K_{F\alpha} K_{F\beta}} \qquad (6)$$

$\frac{\sigma_{F0}}{\sigma_{H0}}$ = function of z_1 (z_1 – number of pinion teeth):

$$z_1 = 2 \cdot \frac{S_H^2}{S_F} \cdot \frac{\sigma_{Flim}}{\sigma_{Hlim}^2} \cdot \frac{u+1}{u} \cdot \cos\beta \cdot \frac{(Z_H Z_E Z_B Z_\varepsilon Z_\beta)^2}{(Z_W Z_L Z_R Z_X Z_v)^2} \cdot \frac{Y_X Y_{\delta relT} Y_{RrelT}}{Y_{FS} Y_\varepsilon Y_\beta} \cdot \frac{K_A K_v K_{H\alpha} K_{H\beta}}{K_A K_v K_{F\alpha} K_{F\beta}} \qquad (7)$$

Application of Equations 4 and 7 results in Equation 8, in which Eht_{min} is proportional to various influence factors, to the actual safety factors, and to the gear module:

$$Eht_{min} = \frac{2}{U_H} \cdot \frac{(Z_H Z_E Z_B Z_\varepsilon Z_\beta)^2}{(Z_W Z_L Z_R Z_X Z_v)} \cdot \frac{Y_X Y_{\delta relT} Y_{RrelT}}{Y_{FS} Y_\varepsilon Y_\beta} \cdot \frac{K_A K_v K_{H\beta} K_{H\alpha}}{K_A K_v K_{F\alpha} K_{F\beta}} \cdot \frac{\sin\alpha_{wt}}{\cos\beta_b} \cdot \frac{\cos\alpha_t}{\cos\alpha_{wt}} \cdot \frac{\sigma_{Flim}}{\sigma_{Hlim}} \cdot \frac{S_H}{S_F} \cdot m_n \qquad (8)$$

The given equations are based on mathematically exact transformations and therefore can be used for all types of involute spur and helical gears.

Assuming some simplifications and reasonable values for the influencing factors, applicable for standard gear sizes and standard gear applications, Equation 8 can be written as:

$$Eht_{min} \approx 0.16 \cdot \frac{S_H}{S_F} \cdot m_n \quad (9)$$

Comparison of the different approaches for determining case depth. From Equation 9, it is obvious that for standard gear sizes, the different methods for calculating case depth according to DIN 3990 ($Eht_{Hopt} = 0.15 \cdot m_n$) and AGMA lead to very close results, if optimum load capacity is required (safety factors near 1).

For lightly loaded gears, Equation 9 tolerates less case depth ($S_F \sim \sqrt{S_H}$, $S_F > S_H > 1$!).

According to the German guideline (Ref. 2), the actual load has not been given special consideration, so optimum case depth for optimum load capacity is always obtained. Introduction of the defined influencing factors Z_{Eht} and Y_{Eht} into the DIN standard takes actual stress conditions into consideration. This means that, for lightly loaded gears, the required case depth can be reduced compared to the recommended optimum case depth. Thus, for a wide range of standard gears, the two methods are applicable, especially if the actual safety factors S_H and S_F (actual load) are used for calculation. Especially for large gears, the maximum allowable stress numbers are often not used due to safety aspects.

Note that for some special gears, calculated values of case depth will differ because Eht_{min} depends on the gear geometry in question and on the real contact stress number σ_H. For critical gearing, detailed studies should be made according to the section "The Loaded Tooth Flank—Some Basic Principles of Contact Stresses."

Conclusion

The allowable stress numbers in standardized rating of gears are valid for normal (optimum) case depth. It is known that especially small case depth values can reduce contact and bending load capacity. In theoretical investigations, it was shown that the loading of tooth flank by Hertzian pressure and tribological parameters induces a stress field in the material, which is variable over depth and can be calculated according to the rules of contact mechanics.

Application of different equivalent shear stress criteria shows that residual stresses can modify the stress state significantly. It was demonstrated that variation of case depth influences the stress gradient as well as the strength gradient over depth. By analyzing the local stress/strength ratio over the tooth profile and depth, a local safety factor can be defined. The depth below surface where the maximum stress/strength ratio occurs depends on the relation of the stress and the strength profile. Adequate case depth leads to a peak value of the stress/strength ratio at or just near the surface. Smaller values of case depth can lead to a relocation of the maximum value of stress/strength ratio at a greater distance below the surface. That relocation may lead to gear damage that initiates below surface. Lower gear loading can result in lower required case depth.

An addition to the ISO/DIN standard is proposed in which the influence of different case depths on load capacity can be taken into consideration.

It was shown that the simple empirical method—case depth proportional to module—recommended by the German guideline (Ref. 2), takes the different conditions of tooth flank and tooth root into good consideration for a wide range of standard gears. Using the proposed influence factors Z_{Eht} and Y_{Eht}, the results for calculated case depth according to the empirical method are in good agreement with other methods (Ref. 1) for calculation of required case depth with consideration of actual load. ○

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March 19–23—2002 International Exposition for Power Transmission. Las Vegas Convention Center, Las Vegas, NV. A technical conference for professionals working with power transmission systems, components, and controls: hydraulic, pneumatic, mechanical and control industries. Cost ranges from $35 to $975, depending on registration and courses selected. For more information, call the International Fluid Power Association at (800) 867-6060 or (414) 298-4141 or visit *www.ifpe.com*.

March 20–22—Advanced Gear Design & Theory. Center for Continuing Engineering Education, University of Wisconsin-Milwaukee, Milwaukee, WI. The seminar is a continuation of Fundamentals of Gear Design and is aimed at the designer, user and beginning gear technologist. Specific lectures will cover manufacturing, heat treatment, drawing data requirements, lubrication types and the basics of load capacity rating, among other topics. $1,095. For more information, contact Richard Albers, program director, by telephone at (414) 227-3125 or by e-mail at *rgalbers@uwm.edu*.

April 15–18—Basic Gear Fundamentals Course. Gleason Cutting Tools Facility, Loves Park, IL. This four-day program is designed for individuals seeking a basic understanding of gear geometry, nomenclature, manufacturing and inspection. $895. This course will be repeated June 24–27 and at other times throughout the year. For more information, call Gleason Cutting Tools Corp. at (815) 877-8900 or visit *www.gleason.com*.

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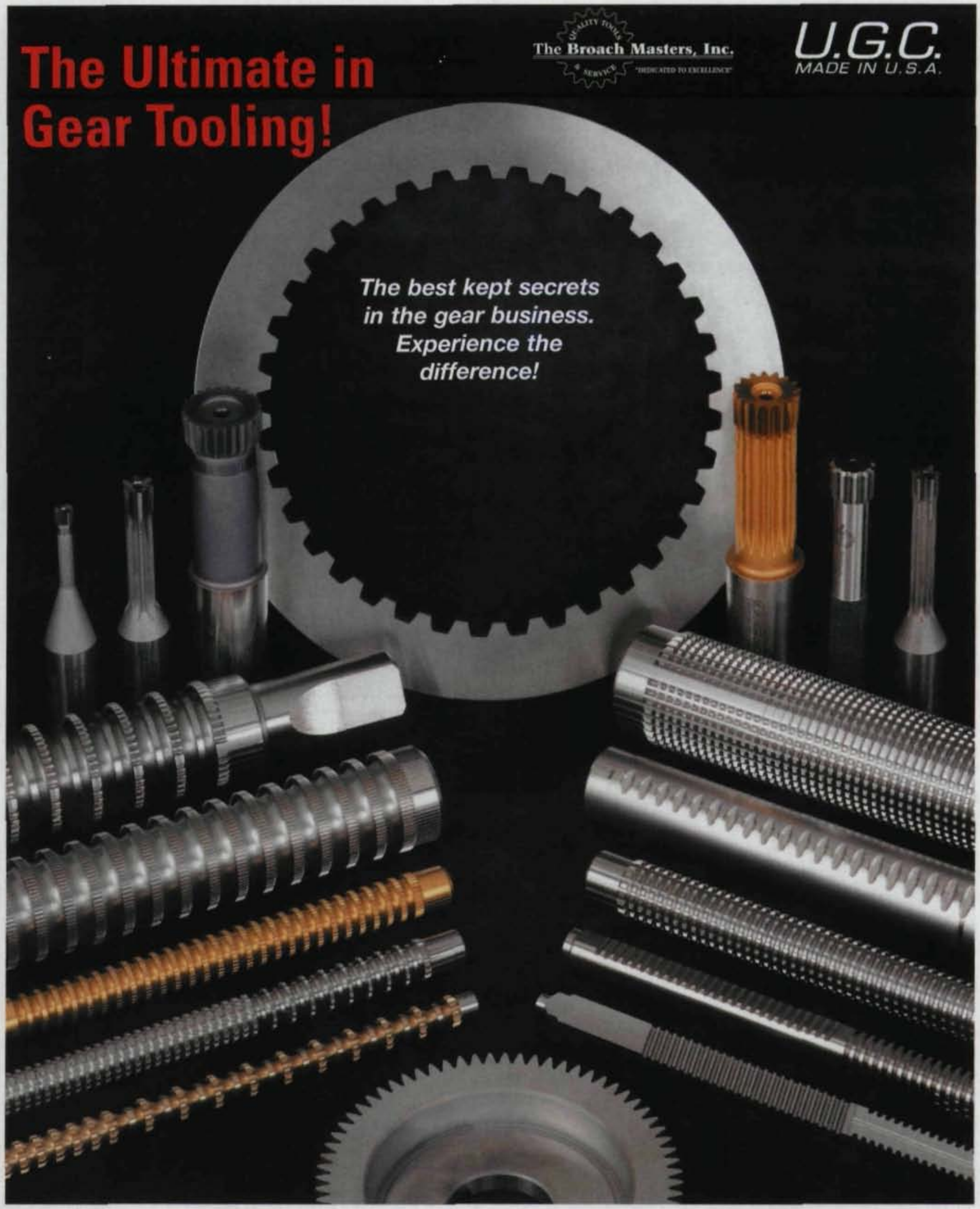
Correction

Regarding the article, "Failure Mechanisms in Plastic Gears," by Yong Kang Chen, Nick Wright, Chris J. Hooke and Stephen N. Kukureka, published in the January/February 2002 issue of *Gear Technology*, we would like to make the following correction:

The short-glass-fiber material was mistakenly presented as RFL4036. It should have read RF1006HS.

We apologize for any inconvenience.

—The Editors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Cleaner, More Energy Efficient: Trends in the Heat Treat Industry

Joseph L. Hazelton

An advancing technology and higher energy costs appear to be leading heat-treating companies in the gear industry toward cleaner, more energy-efficient processes. These processes may offer some relief to heat treaters through cooler factories and some relief to their companies through reduced energy usage.

Also, the heat treat industry has long-term goals for making heat treating cleaner and more energy efficient.

An Advancing Technology

An advancing technology among the gear industry's heat treaters is vacuum carburizing and hardening, according to Bob Cvetichan, manager of the heat treat department at Horsburgh & Scott Co. of Cleveland, OH.

Cvetichan says he sees an industry-wide trend toward vacuum carburizing and hardening. In his opinion, the process is becoming more popular because it's cleaner than gas carburizing and hardening, as well as conventional through-hardening.

In Cincinnati, OH, the president of Cincinnati Steel Treating Co. agrees.

"It's cleaner," says Jerry Wolf. "Vacuum furnaces are not hot. They're cold on the outside."

Their coolness could make those furnaces easier to install in a manufacturing plant because the heat treat operation wouldn't need to be a separate department in the plant.

Cvetichan adds that the cleaner vacuum process would make for cleaner workplaces, which could raise heat treaters' morale.

According to Wolf, vacuum carburizing and hardening appears to have advantages over conventional carburizing and hardening.

He says there's evidence that vacuum carburizing and hardening deepens the optimum carbon level in the case, increasing the gear's load-carrying capacity. He adds that the process—with high-pressure gas quenching—appears to reduce distortion in gears and makes remaining distortion more uniform. Also, there's no intergranular oxidation on the gear's surface.

Wolf cautions that the process may not be readily adaptable for use in job shops, but it can be used in high-volume production where there are large loads of similar gears.

Vacuum carburizing and hardening has been around for a long time. But Wolf says it had "a black eye" for a while because it was expensive to install and maintain, and it created soot, which got on everything—including gears being treated. Also, gas quenching—which is currently used with vacuum carburizing and hardening—didn't quench parts fast enough, thereby requiring the use of oil quenching.

Wolf says atmosphere carburizing and hardening of gears is "here to stay" as a heat treat process. But, he adds that vacuum carburizing and hardening will start nibbling away at traditional carburizing and hardening.

According to Wolf, people would look at vacuum carburizing and hardening more seriously today than they would've one or two years ago. Still, he says converting to the vacuum process will be slow because of high capital costs.

More Energy Efficient

Many heat treaters in the gear industry are trying to make their operations more energy efficient. Their efforts are in part a response to high energy costs they faced in 2000 and 2001.

Photo courtesy The Gleason Works

Merit Gear Corp. of Antigo, WI, installed recuperative burners in two of its three heat treat furnaces to make them more energy efficient. A recuperative burner uses an integrated inner and outer tube system to burn gas more efficiently.

"It's like having a high-efficiency furnace in your house," says Don Clemins, manufacturing manager of Merit Gear.

Heat treaters appear to benefit from the burners, too. Clemins explains the burners create less exhaust gas, so the furnaces give off less heat, making the factories not as warm during the summer.

Merit Gear installed those burners because its energy costs in 2001 went up 20% from its usual costs. The company absorbed those higher costs.

The Gleason Works of Rochester, NY, tried to reduce its energy usage by shutting down one of its three rotary hearth

Reduced Ammonia in Factories: A Trend Toward Ion Nitriding

Besides a trend toward vacuum carburizing and hardening, Bob Cvetichan of Horsburgh & Scott Co. also sees a trend toward ion nitriding, another type of vacuum process. Like vacuum carburizing and hardening, ion nitriding can create a better workplace for heat treaters.

Cvetichan describes ion nitriding as offering easier control over microstructures than gas nitriding, so resulting gears have desired microstructures. He adds that ion nitriding offers better repeatability than gas nitriding.

Ion nitriding mainly uses nitrogen and hydrogen as process gases—not ammonia, so there would be less ammonia smell in factories.

"A lot of heat treaters who do nitriding—gas nitriding," Cvetichan says, "you can walk into their plants and know they do nitriding."

Because ammonia isn't used, heat treaters don't need to measure its dissociation in ion nitriding.

Cvetichan, however, doesn't see the trend toward ion nitriding as a trend away from gas nitriding. He explains that in his opinion, ion nitriding provides better quality control, but gas nitriding can be cost effective.

furnaces, but the company had to fire it up later because of production demand.

Although a machine tool company, Gleason Works heat treats materials for its gear-manufacturing equipment.

In its heat treat operation, Gleason Works' natural gas costs had increased 39%. Dino Giordano, supervisor of heat treatment and electroplating, says those costs decreased significantly by the end of 2001 but have fluctuated since then.

The company looked for other cost savings by installing better thermal controls on its atmosphere furnaces, as well as installing thermal modules to speed the heat-up and cool-down processes, getting rid of some outdated heat treat equipment, and switching from generators that use natural gas to a liquid nitrogen/methanol mixture. With the nitrogen maintaining the furnaces' atmospheres, the furnaces can be started quicker for on-line use.

Horsburgh & Scott is also tweaking its equipment to save energy. Cvetichan says segments have been ramped up, holding times have been reduced and cycle lengths have been changed. Also, the company is installing better seals on its furnace lids to reduce heat loss.

Horsburgh & Scott saw its natural gas costs go up more than 100% in the first quarter of 2001.

Cvetichan remembers that Horsburgh & Scott's energy contract expired at a very bad time—when energy costs were out of this world. But, the company had to have the natural gas, so it had to negotiate a new contract—no matter the costs.

Since then, Horsburgh & Scott's natural gas costs have gone down 67%. But, the company got caught in another contract. It had to negotiate a new contract before natural gas returned to its previous prices.

Wolf says Cincinnati Steel Treating is always looking for better burners and insulation, but efforts to improve energy efficiency are incremental. For example, the company upgraded its radiant tube burners to recuperative units several years ago. He adds that there's no new super-efficient heat-treating equipment on the market.

Cincinnati Steel Treating saw its natural gas costs jump 60% in August 2000, when its contract expired and it had to negotiate a new one.

"We had escaped the big bullet," Wolf says. During the 1999–2000 winter, energy prices were higher.

"We didn't get hurt as bad as some heat treaters did," Wolf says.

Still, the company raised its prices. Wolf says the raises were the company's first in four years, that the higher energy costs were the straw that broke the camel's back.

Cincinnati Steel Treating negotiated a new contract, locking in energy prices that looked good—at the time. Energy prices later dropped and are now lower than Cincinnati Steel Treating's locked-in prices.

Wolf says that's the gamble that you take in today's volatile energy market.

Industry Goals

Many heat treaters took steps to reduce their energy usage in the short term. Their industry has goals for dealing with energy usage in the long term. The goals also deal with environmental impact.

In 1999, a research and development plan was issued by the ASM Heat Treating Society, an affiliate organization of ASM International. The plan was supposed to be a starting point for implementing the heat treat industry's "Vision 2020" goals.

Based on industry needs, "Vision 2020" was a view of the ideal future, with the following goals: use 80% less energy, improve insulation, eliminate emissions, reduce production costs by 75%, increase furnace life tenfold, reduce furnace prices by 50%, get rid of distortion—thereby maximizing uniformity—in heat-treated parts, get a 25% annual rate of return on investments in capital equipment, and create 10-year partnerships with customers.

In early 2001, the heat treat industry was facing higher energy costs. In February 2001, Roger J. Fabian, the heat treat society's president, wrote to the industry that higher natural gas and electricity costs again stressed the importance to the industry of achieving the "Vision 2020" goals.

To work toward those goals, the society was picked by the U.S. Energy Department to coordinate development and communication of the industry's research and development plan. To manage the research and development programs and projects, a number of industrial companies created an independent organization, the Center for Heat Treating Excellence, located in Worcester, MA.

Fabian wrote: "Only through the commitment of the entire heat treating community can we keep our industry competitive in an unpredictable economy." ○

FOR MORE INFORMATION

- **ASM International/ASM Heat Treating Society**
  9639 Kinsman Road
  Materials Park, OH 44073-0002
  Telephone: (440) 338-5151
  Fax: (440) 338-4634

- **Center for Heat Treating Excellence**
  Worcester Polytechnic Institute
  100 Institute Road
  Worcester, MA 01609-2280
  Telephone: (508) 831-5992
  Fax: (508) 831-5993

- **Cincinnati Steel Treating Co.**
  5701 Mariemont Ave.
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  Telephone: (513) 271-3173
  Fax: (513) 271-3510
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  5114 Hamilton Ave.
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Gleason, Kashifuji Announce Hob Sharpening Alliance

Gleason Corp. and Kashifuji Works Ltd. of Kyoto, Japan, formed an agreement that provides Gleason with exclusive rights to sell and service Kashifuji hob sharpening machines in various areas worldwide.

The agreement is the latest initative in the strategic alliance between Gleason and Kashifuji that began in March 2000 with the joint development and marketing of certain cylindrical gear production machines in Asia.

Products are manufactured by Kashifuji and sold through OGA Corp. of Tsukishima, Japan. Additionally, Gleason represents and sells Kashifuji's existing cylindrical gear products through Gleason's current sales and service network.

GM, DaimlerChrysler End Partnership In New Venture Gear

DaimlerChrysler and General Motors Corp. will end their joint powertrain venture, New Venture Gear.

DaimlerChrysler will buy GM's share of the business, started in 1990, and thereby become the sole owner. However, GM will continue to produce gears, transmissions and differentials as an independent business out of its current Muncie, IN, facility.

After the restructuring, New Venture Gear's main operations will include its plants in Syracuse, NY, and Roitzch, Germany. Management, engineering and development functions at its Troy, MI, headquarters will be restructured.

Philadelphia Gear Corp. Appoints New President, CEO

Carl D. Rapp

Carl D. Rapp was hired as president and CEO for Philadelphia Gear Corp.

His responsibilities include leading the company's efforts to align with its end users and develop new product offerings. Among Rapp's priorities is the optimization of the inspect and repair regional service and manufacturing centers.

Rapp has held several senior level positions in sales and general management, most recently as senior vice president of sales and marketing for Strategic Distribution Inc.

Bourn & Koch Buys Controlling Interest in Roto-Technology

Rockford, IL-based Bourn & Koch Machine Tool Co. purchased controlling interest in Roto-Technology of Dayton, OH. Under the new ownership, the Dayton facility will continue to manufacture, remanufacture/retrofit and service Roto-Technology gear inspection systems and rotary tables.

According to Roto-Technology's press release, the combined companies will use the Bourn & Koch distribution and quality system while eliminating duplicate efforts.

AGMA Announces Partnership with Gear Consulting Group

The American Gear Manufacturers Association announced a partnership with the Gear Consulting Group of Richland, MI. The two organizations will collaborate on the Training School for Gear Manufacturing, with the GCG facilitating regional and in-plant versions of the AGMA program.

The training school focuses on the relationships between manufacturing methods, inspection procedures and the underlying involute geometry and nomenclature.

Overton Gear Has New President, CEO

Lou Ertel was hired as president and CEO of Overton Gear and Tool Corp., located in Addison, IL.

Previously, he served as vice president of operations for Foote-Jones/Illinois Gear, a division of Regal Beloit Corp. Prior to that, Ertel worked in a variety of positions, including plant manager, during his 24 years at Philadelphia Gear Corp.

British Gear Association Names Chief Executive

Tom Lynch was named chief executive of the British Gear Association. Lynch has worked in the engineering industry for 36 years in various positions. Most recently, he served as commercial and technical director of an aluminum die casting foundry.

Lynch, a fellow of the Institute of Mechanical Engineers, said, "Although this is an industry which is not as buoyant as it has been in the past, there are—nonetheless—opportunities and the BGA has a team which intends to continue its support of current members and increase membership on an ongoing basis."

SU America Hires Vice President of Sales for Illinois, Indiana

SU America Inc. has hired Fred H. Schomaker as vice president of regional sales, with responsibility for Illinois and Indiana.

Schomaker came to SU America from Overton Gear and Tool Corp. of Addison, IL, where he served as vice president of sales and marketing.

Previously, he worked for Motch & Merryweather Machinery Co., American Pfauter Ltd. Partnership and Gleason Corp.

Schomaker served American Pfauter as a regional sales manager specializing in automotive markets. When Gleason bought American Pfauter in 1997, Schomaker continued to work in the Michigan territory.

According to SU America's press release, Schomaker has extensive knowledge of gear hobbing, shaping, shaving, grinding, deburring, chamfering and hard finishing.

Also, Schomaker has bachelor's degrees in mechanical and electrical engineering.

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New Gearhead Line from Mijno Precision Gearing

The MRP gearhead line from Mijno Precision Gearing is a group of all-planetary, low-backlash, servo-grade gearheads intended for applications that do not involve rapid reversing or acceleration.

The MRP is available in one or two stages with ratios from 100 to 1. Five sizes are offered. The product is shaped as a round-body gearhead with tapped holes in the face of the output shaft end. At the motor input end, there is a square flange for assembly to servomotors. A clamping-style input pinion is self-locating, enabling assembly to servomotors.

For more information, contact the North American office in Park Ridge, IL, by phone at (847) 698-9041 or visit *www.mijno.com.*

New Gear Grinding Machine from Reishauer

Reishauer's new RZ 400 gear grinding machine utilizes gearless planetary drives, acoustic sensing for alignment of dressing diamonds and low-noise shifting for surface finish improvement.

Applications for this machine include small and large batch production.

According to Reishauer's press release, the idle times are minimized. Flexibility in generating different gear tooth geometry, as well as applying diverse grinding cycles and profiling processes, is guaranteed.

For more information, contact Reishauer Corp. in Elgin, IL, by phone at (847) 888-3828 or on the Internet at *www.reishauer.com.*

Screw Jacks From Nook Industries

Nook Industries' Actionjac metric ball screw jacks are available in 0.5-ton to 20-ton capacities, in ball screw or machine screw designs.

The Actionjac metric ball screw jacks feature low-profile, internal return ball nuts, while machine screw jacks are fitted with bronze, trapezoidal nuts. Actionjac metric jacks are available in upright or inverted configuration with translating or rotating lifting screw and optional top plate or clevis mounting. IEC motor and motor mounting options are also available.

For more information, contact Nook Industries of Cleveland, OH, by phone at (216) 271-7900 or on the Internet at *www.nookindustries.com.*

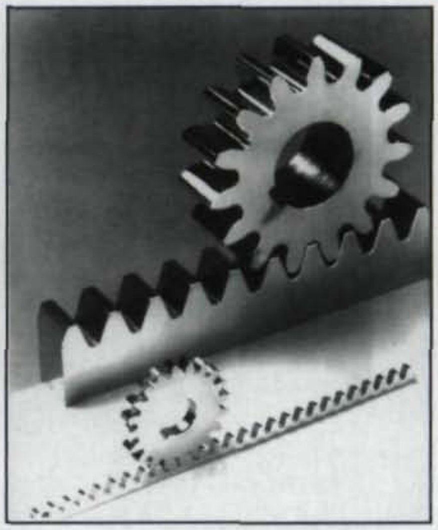
New Gearing Components from HD Systems

The CSD series is a line of ultraflat gearing component sets with a 50% reduced axial length and is designed for robotics, aerospace and factory automation applications.

The series uses an "S" tooth profile. Additionally, the construction of the component set allows the surrounding enclosure to be made more compact for size and weight savings.

The CSD size 20 has a rated torque of 248 in.-lbs. and a maximum torque of 673 in.-lbs. and is available in gear ratios of 50:1, 100:1 and 160:1.

For more information, contact HD Systems of Hauppauge, NY, by phone at (631) 231-6630 or visit *www.HDSI.net* on the Internet.



New Gear Rack from Emerson Power Transmission

Emerson Power Transmission expanded the Browning line of gear racks to include gears with 25° pressure angles.

According to Emerson's press release, other new capabilities include 1–64 DP, 0.5–28 module, a helical gear rack, a round rack and thickness and face width to 17.75". Products are developed to meet the requirements of the packaging, construction, machine tools, power plants, material handling, agricultural equipment and accessibility equipment industries.

For more information, contact Emerson Power Transmission of Maysville, KY, by phone at (606) 564-2087 or on the Internet at *www.emerson-ept.com.*

New Gear Reducer from Harmonic Drive Technologies

The Rotary Vector (RV) C hollow shaft gear reducer from Harmonic Drive Technologies offers high ratio gear reduction and a through-hole ranging from 31 mm (1.2") to 138 mm (5.4"). The measurements are suitable for vacuum lines, wiring harnesses, concentric shafting and cooling lines to be run through the center.

According to Harmonic Drive's press release, the rated torque range is 867–27,754 lbs. with a peak torque range of 4,337–138,768 lbs. Total lost motion from all sources, including backlash, spring rate and hysteresis, is one arc-minute. The reducer includes built-in output bearings that support large thrust and overhung loads.

Applications for the product include rotary tables, wrist axes, welding positioners and other precise rotary positioning uses.

For more information, contact Harmonic Drive Technologies of Peabody, MA, by phone at (978) 532-1800, by e-mail at *info@harmonic-drive.com* or on the Internet at *www.harmonic-drive.com.*

New Grinding Machines from United Grinding Technologies

Combining centerless grinding and between-center grinding into one setup, the Kronos L Dual from United Grinding Technologies Inc. was specially developed for workpieces such as gearshafts, crankshafts, camshafts, or shafts for electric motors.

According to United's press release, this machine reduces machining time by approximately 45%. A workhead and tailstock are installed between the grinding and regulating wheels and beside the fixed workrest blade.

For more information, contact United Grinding Technologies Inc. of Miamisburg, OH, by phone at (937) 859-1975, by e-mail at *ugt@grinding.com* or on the Internet at *www.grinding.com*

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UNDERCOVER GEARS

TOP SECRET

CODE NAME: Ginger
MISSION: Design, prototype and test a transmission for a new device. The transmission must be compact and efficient. It should have almost no backlash, and it must be able to operate in both forward and reverse. Most importantly, the transmission must be quiet. In fact, it shouldn't sound like a transmission at all. It should blend in with the environment and sound like music or the wind. This mission, should you choose to accept it, is top secret. Not even your employees can know what you're working on...

Sounds like something out of a spy novel, right? Well, not exactly. This "mission" was a real assignment, accepted by Axicon Technologies Inc. of Pittsburgh. It was given to Axicon by a startup company called Segway L.L.C. of Manchester, NH, whose Segway personal transportation device was unveiled in December after years of speculation, rumors and—towards the end—considerable hype.

Segway L.L.C. was founded by eccentric and renowned inventor Dean Kamen. For some time, there was a buzz about what he'd been working on, a project code-named "Ginger," which was also sometimes referred to as "IT."

Now, the secret is out. The Segway human transporter is designed to carry a single passenger at speeds up to 12.5 mph in a pedestrian environment. It was unveiled in a media blitz on *Good Morning America* and has been featured on *The Tonight Show* with Jay Leno. Highly engineered, it's equipped with all sorts of mechanical and electronic gadgets to ensure balance, smoothness of ride and safety.

Thanks to Axicon Technologies, the transmission is also highly engineered. For example, Segway's engineers were very particular about the sounds made by the gearbox. They wanted not only a quiet gearbox, but also one whose sound was consistent with the rest of the device.

Riding the Segway is supposed to be a "light, efficient and magical" experience, says J. Douglas Field, Segway's vice president of product development and chief engineer. "We wanted noise to be low in level and high in quality."

Working with gear noise is one of Axicon's specialties, says Brian Ahlborn, vice president of sales and marketing.

In this case, Ahlborn says, Axicon was instructed to make the gears "sound like the wind." To do that, the gears were carefully "tuned" to achieve just the right sound.

"It's pure music theory," says Field. He explains that by controlling the number of teeth in each mesh of the two-stage transmission, it makes sounds that are exactly two octaves apart. By tuning the transmission in that way, the engineers hoped to eliminate any dissonance and make the transmission sound pleasant to the ear.

Axicon also held jury tests, during which people listening to the transmission rated its sound quality. The transmission was continually improved based on those ratings.

According to Ahlborn, Axicon's strengths include a mix of proprietary technologies and engineering approaches, as well as a corporate culture that emphasizes creativity and technology. "We're pretty unique in how we do some of those things," he says, describing the corporate culture as "Silicon Valley meets the Rust Belt."

Field says it also helped that Axicon could act quickly and knew what companies to contact for manufacturing. According to Ahlborn, Axicon delivered working, sound-tested prototypes in five months. Axicon did some gear manufacturing in-house, but Schafer Gear of South Bend, IN, did most of the gear manufacturing as Axicon's gear supplier for the Segway project.

Now that the secret is out, Segway and Axicon are getting ready for mass production. Segway has built a 77,000 square foot factory in Manchester, NH. The facility will be capable of producing 40,000 Segways per month. Although the consumer version is not scheduled to go on sale until late in 2002, the Addendum team will be among the first in line to try them out, if for no other reason than to hear the gears. O

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