GEAR TECHNOLOGY

JANUARY/FEBRUARY 2005

The Journal of Gear Manufacturing

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- New Approach to Computerized Design of Spur & Helical Gears
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Expanded Coverage of Products, Industry News and Events

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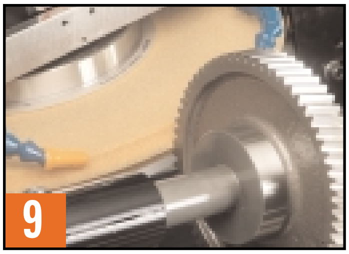

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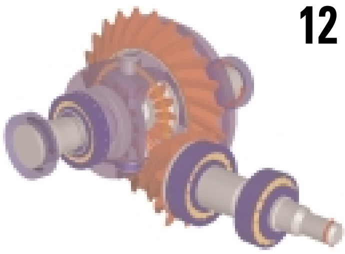

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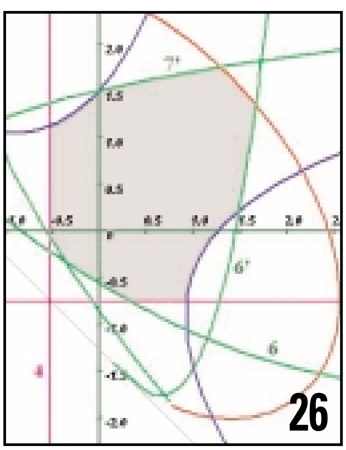

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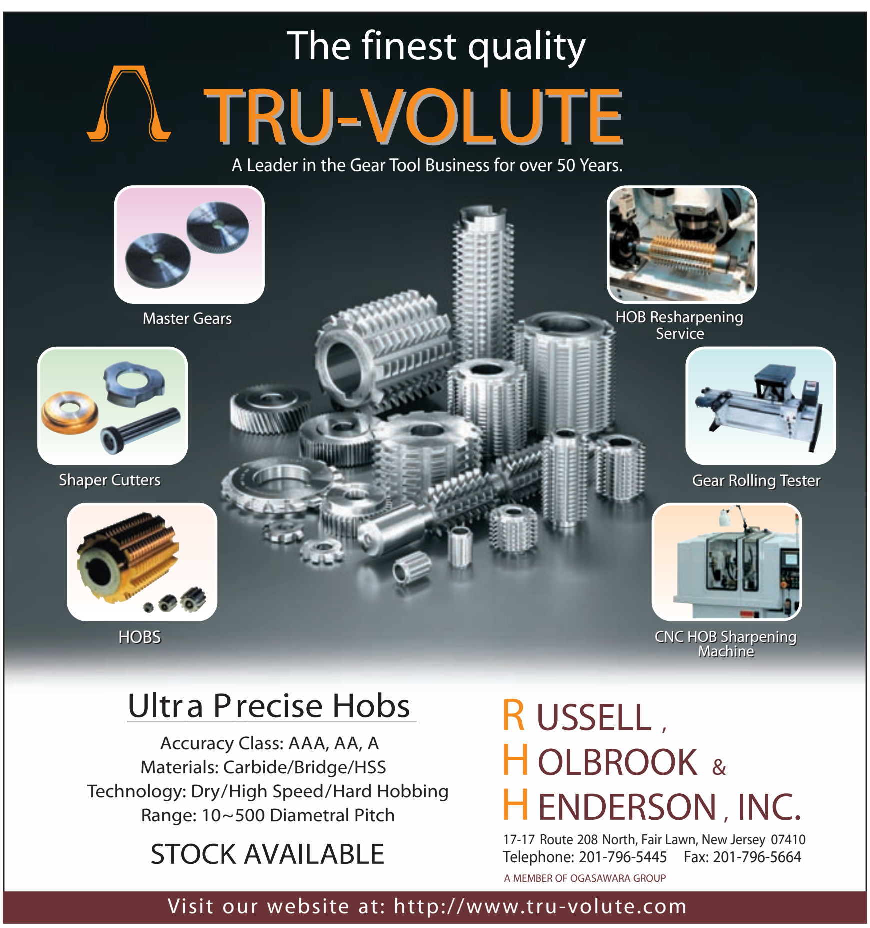

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The Journal of Gear Manufacturing

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The Perfect Complement

For more than 20 years, *Gear Technology* has been the world's leading resource for technical information about gear design, manufacturing, processing and use. We've always considered *Gear Technology* to be an educational journal, helping gear industry professionals learn how to be more productive and efficient, as well as how to design and manufacture better gears or judge the quality of gears they're purchasing.

That mission continues to be our focus today. We remain firmly committed to bringing you the highest quality, most relevant technical information available. *Gear Technology* will continue to be your source for information about all phases of gear manufacturing.

Over the years, we've continually adjusted *Gear Technology* to best suit the needs of our readers. With this issue, you'll see some of the latest changes. In particular, you'll notice that we've removed advertising from the pages on which technical articles appear. We've always prided ourselves on the independence and integrity of our editorial material. It's not our job to give you advertorials. Instead, it's our job to provide you with the most thorough, balanced information available, so that you can make informed decisions yourself.

In *Gear Technology*, you'll never find a company's advertisement next to an article that features that company. Other magazines do this, but then, as a reader, you never quite know where the advertisement leaves off and the article begins. We want you to understand that our articles are not for sale, that you can continue to trust the content of *Gear Technology*, as you have for more than 20 years.

To accommodate all of our advertisers, we have expanded our news sections. We consider this a win-win situation. Not only does it keep the more commercial material separate from the technical articles, but it also gives us the opportunity to provide more in-depth information on the news items that are of most interest, including the products of our advertisers and other suppliers in the industry.

We recognize that in today's lean environment, most of you are busier than you've ever been, but having information on the latest products and services available in the marketplace is of extreme importance for you to remain competitive and successful.

We also recognize that we can't possibly cover everything you need in *Gear Technology* alone. That's why, beginning in February, we are launching a companion publication, called *Gear Product News*.

Gear Product News, the perfect complement to Gear Technology, will begin publication in February 2005.

Gear Product News will be the industry's No. 1 sourcebook for information about the products and services that are available to help solve gear-related challenges. Six times per year—in *Gear Technology*'s off months—it will approach the gear industry from a less technical angle. *Gear Product News* will be aimed at corporate executives, purchasing managers and decision-making manufacturing personnel. Also, we promise it will be light and easy to read!

Suppliers to the industry will be able to use *Gear Product News* to reach a different part of the marketplace—those interested in the gear industry, but not necessarily its technical aspects—at an affordable price.

Gear Product News will be free to qualified readers, just like *Gear Technology*. Subscribers in the United States will receive a printed copy by mail. Subscribers outside the United States will be able to receive a free electronic copy. Please log on to our new website, *www.gearproductnews.com*, to sign up for your free subscription today. While you're there, you'll have a chance to download a mock-up sample version of the magazine, and you can also learn more about advertising in *Gear Product News*.

Between our expanded news sections in *Gear Technology* and the addition of *Gear Product News*, we're more committed than ever to helping companies in the gear industry publicize their news. For example, if you've introduced a new product or service for the gear industry, we want to hear about it. If you've recently hired new key personnel, we want to hear about it. If you've expanded your capacity or are offering new services, we want to hear about it. If you've recently gained ISO or other accreditation, we want to hear about it. If you think you have an interesting story to tell, we want to hear about it. Send your ideas to *editors@geartechnology.com* or *editors@gearproductnews.com*. We look forward to your submissions.

As always, we'd like your feedback about the changes we are making, both with *Gear Technology* and our new publication, *Gear Product News*. Give us a call at 1-847-437-6604 to let us know what you think.

Michael Goldstein

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Holroyd Launches New Gear Grinder

New machine promises DIN2 accuracy and unique features at a low cost

Holroyd, long known for its expertise in creating machine tools for manufacturing rotors and other helical components, has now applied that expertise to the manufacture of gear grinding machines with the October 2004 launch of the GTG2, a machine tool specifically designed for small- to medium-batch production of high precision helical gears.

"Our design aim was to produce the lowest cost, highest accuracy grinding machine for helical and worm gears in the world," says Dr. Tony Bannan, Holroyd's engineering director, who led the design and manufacturing team that developed the machine.

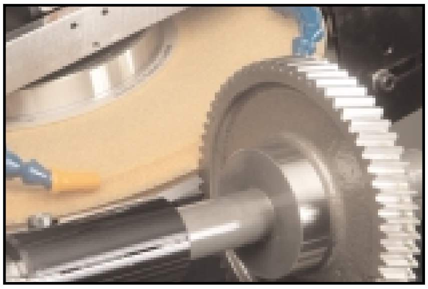
The GTG2 has all the features of a modern gear grinding machine—including on-board inspection and grinding wheel dressing. But it also comes with some uncommon features that stem from Holroyd's experience with helical components *other* than gears.

"We've approached the development of gear grinding from a rather different angle, which we think has led to many strengths," says Dr. Chris Holmes, Holroyd's R&D director.

One of the GTG2's uncommon features is its approach to topological modifications of the gear teeth. The Holroyd method, say its developers, solves the problem of flank twist on helical gears. Many older models of form gear grinding machines use a method called "bob crowning," which varies the depth of the grinding wheel in order to achieve tooth modifications.

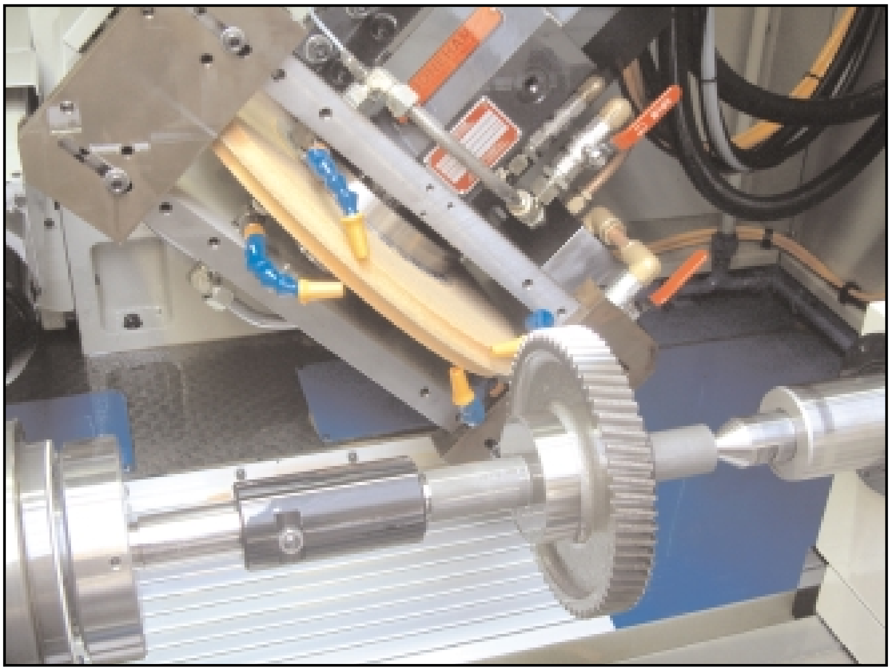
"When you do bob crowning, you get a distortion of the gear," says Alan Stephen, special projects manager for Holroyd. "This is unavoidable, because of the way you're generating the shape. It's correct in the middle, but not near the ends."

Helical twist causes the profile of the tooth to vary across the face. In many applications, this "error" is not of concern, but in high precision and low noise applications, it affects gear performance by concentrating loads on particular areas of the teeth during meshing. "You may actually want the gear to have this kind of relief," Stephen says, "but the problem is, you can't control it. It depends on the wheel diameter, and you get what you get. Our object was to make an improvement on this."



Holroyd's improvement was to apply a different kind of motion to the grinding pass. Instead of the workhead depth, the GTG2 varies the workhead's swivel angle in order to achieve crowning. This method is patent-pending.

"What we do is, as we're grinding through the gear, the grinding head will



Holroyd's New GTG2 Gear Grinder promises DIN2 accuracy and unique features at a low cost

actually noticeably alter its angle of swivel as it passes through the gear," says Paul Hannah, Holroyd's VP of global sales and marketing. But varying the swivel angle alone isn't enough to solve the problem entirely, Hannah says.

The GTG2 also comes with a built-in simulation package, which allows the machine to simulate the actual cut, assess any inaccuracies, and make adjustments to the grinding wheel. "We then correct the wheel," Hannah says, "making the wheel wrong, if you want, to make the gear right."

The mechanical motion, the simulation package and on-board inspection combine to create a closed-loop feedback system, which enables this machine to provide the highest levels of accuracies, Hannah says. "After we've ground the first one, we measure it, feed back and correct all over again. So instead of something that becomes an approximation of a target profile, we're actually creating the target profile that the designer has wanted. That's something which, I believe, is quite innovative in the field of gear manufacturing."

The on-board inspection is accomplished through the use of a Renishaw CMM probe. "This Renishaw scanning probe that we use on our machine is a full 3-D scanning probe," says Holmes. "It's not a touch-trigger probe. It's the sort of scanning probe that you would have on a full, high-accuracy CMM. The deflection in each of the three directions is measured. It can measure deflections of two microns."

The development of the technology associated with the GTG2 has resulted in Holroyd receiving the British Queen's Award for Enterprise 2004, in the category of innovation. "We are delighted that the awards committee has recognized the ingenuity of our designers, their hard work and effort," says Hannah.

Some of the machine's development was carried out at the Design Unit at the University of Newcastle upon Tyne. "We looked for what we felt was the strongest gear development center in Europe," Hannah says. "We found it very useful that one of the top gear development centers in Europe is the Design Unit at the University of Newcastle, so we forged strong links with Professor Dieter Hofmann of the Newcastle gear design center. We worked with him and his students—we put some of our guys working with them—and they also worked with us at our facility, sharing a lot of knowledge that developed the first machine. The first one went into Newcastle. Then we continued adding on after that."

The first GTG2 machine for production has been sold to Micro Precision Ltd., a Hemel Hempstead, U.K., manufacturer of precision components for the aerospace, medical and Formula 1 industries. "We're still going through a learning curve with the machine," says Barry Cave, director at Micro Precision, "but we are extremely satisfied with it. In our brief experience with it, it is holding very close tolerances."

Holroyd developed its own software and controls for the GTG2.

In some cases, under optimum conditions, those tolerances are being held to DIN 1 levels for some measurements, Cave says. He also notes that "on some components, our set-up times have been reduced from three days to three hours. In addition, we now also have the flexibility to quickly change gear forms to meet our customers' requests."

The staff of Micro Precision had plenty of experience with the GTG2 before making their purchase, says Hannah. "We actually took them into the Newcastle design center. They started to produce their gears on our machine at Newcastle before they actually bought the machine."

Micro Precision also conducted machine trials at Holroyd before making their purchase, Cave says.

According to Hannah, allowing the customer to try the machine out is normal operating procedure for Holroyd. "We recognize that we have a high profile in the rotor industry, but in the gear industry, whatever our history was, we're the new kid on the block now. So we're willing to come to people like this and say, 'What do you need from us? We'll prove it to you. Don't take the risk. We'll happily put our money where our mouth is. I'm not talking about a quick demonstration. Let's make a batch of gears, let's make 10 batches of gears. You can, in fact, send your own operator to work the machine in our plant or some other place to make the gears on it. Then you can make your decision.'"

To make that decision easier, Hannah says, the price of the GTG2 will compare very well with competitive machines. "We designed this machine the way we design all machines now, looking at a design for cost. And, in doing that, we set ourselves aggressive cost targets. We're well within those cost targets, and we have a product which is very aggressively priced in the gear industry."

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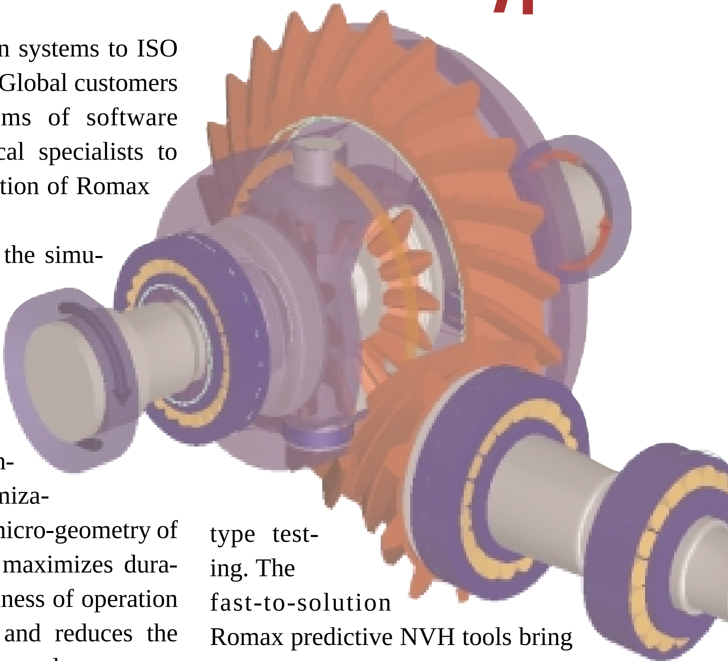
Robust Transmission Design through Automated Optimization of Virtual Prototypes



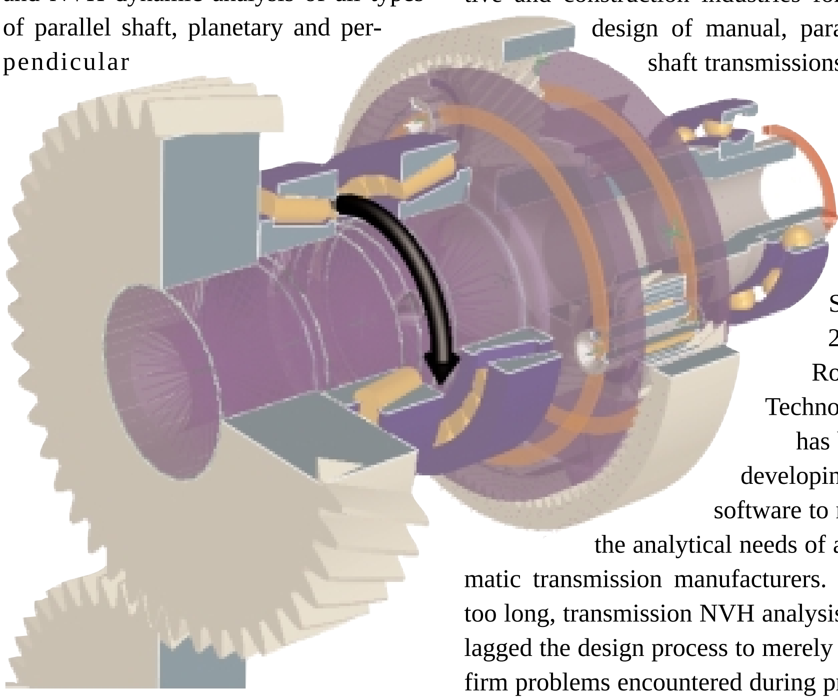
Romax Technology, the U.K.-based supplier of virtual transmission simulation software and technical consultancy services, is automating the design iteration optimization process to allow companies to be faster to market with the highest quality, most robust geared products.

"RomaxDesigner has the closed-loop analytical techniques required for the addition of optimization algorithms to enable the software to search the entire design space for the optimal 'right first time' solution," says Dr. Peter Poon, managing director. "Romax can run hundreds of simulated tests, reducing the need for costly prototype hardware and lengthy, expensive development test programs."

Romax offers a suite of fully integrated software modules for the durability and NVH dynamic analysis of all types of parallel shaft, planetary and perpendicular axle power transmission systems to ISO and AGMA standards. Global customers are supported by teams of software developers and technical specialists to smooth the implementation of Romax into organizations.

According to Poon, the simulation models accurately predict all the system deflections under all loading conditions that have an influence on gear mesh misalignment. Automated optimization of the macro- and micro-geometry of spur and helical gears maximizes durability, improves smoothness of operation to reduce gear noise, and reduces the effect of manufacturing tolerances on transmission error.

The software is used in the automotive and construction industries for the design of manual, parallel-shaft transmissions.

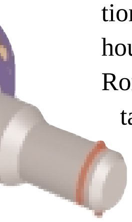


Since 2001, Romax Technology has been developing its software to meet the analytical needs of automatic transmission manufacturers. "For too long, transmission NVH analysis has lagged the design process to merely confirm problems encountered during prototype testing. The fast-to-solution Romax predictive NVH tools bring design and analysis into phase, allowing analytical engineers to have an influence on the design before hardware is made," says Simon Roberts, business development manager for the North American market. "The extension of the NVH analysis to improve the torque density and reduce gear whine in planetary transmissions has opened a whole new market for us."

The Romax Approach

Traditional finite element (FE) models of transmissions require the combination of measured test results with analytical techniques to understand system dynamics. The models are large, time consuming to build and modify, and require "fudge factors" to correlate the predicted results with measured data before modifications can be made to improve system NVH characteristics.

RomaxDesigner analysis models include the lateral, axial and torsional degrees of freedom to accurately predict the dynamic behavior of the internal components at the design stage without the need for measured data, Roberts says. The dynamic characteristics can then be used to identify any potential gear whine problems. Modifications to the gear microgeometry and component dynamic characteristics, to minimize transmission error excitation and reduce system responses, can be implemented to optimize transmission NVH levels far in advance of hardware manufacture and test.



The development of such complex virtual simulation software requires close interaction between software developers and experienced transmission design and analysis engineers, Roberts says. In addition to developing the software, this in-house engineering capability also allows Romax to offer comprehensive consultancy services from "find and fix" projects eliminating durability and NVH issues to full turnkey transmission design projects.

"As a company, Romax Technology is committed to enabling transmission design and manufacturing companies to achieve 'right first time' designs," says Poon, "eliminating the risk of encountering problems during development testing, reducing the need for verification testing, shortening the time to market, and minimizing the piece cost of the final product."

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"RomaxDesigner has the closed-loop analytical techniques required for the addition of optimization algorithms to enable the software to search the entire design space for the optimal 'right first time' solution," says Dr. Peter Poon, Romax Technology's managing director.

PRODUCT NEWS

A Planetary System that Increases Power Density

More power at your fingertips—Turnkey Design Services is manufacturing a planetary gear system to increase power density.

A gear buyer has a motor, an actuator and a new requirement for the gears between them: More power in the same space. Or he may face the other common requirement: The same power in less space. Whichever is the case, he needs a planetary gear set that will do the trick in his low horsepower application.

Also, whichever is the case, he may find the needed gear set in a design conceived by the National Aeronautics and Space Administration and developed by it and Turnkey Design Services LLC, a small engineering firm.

As an example of more power in the same space, TDS compares its planetary set to a similar, conventional set. Both have gearhead diameters of 23 mm. The TDS set has a reduction ratio of 222:1, the other a ratio of 246:1. The TDS set, however, can sustain a maximum torque of 1.2 Nm to the other set's 0.7—that's a 71% increase in maximum continuous torque.

Moreover, the TDS set does its work in two stages; the other set uses four. Not surprisingly, the TDS set weighs less than its counterpart, 35 grams compared to 100.

The greater power and fewer parts—and the consequent reduced weight—result from the planetary set's design. The heart of that design is the elimination of a separate bearing and the incorporating of a cylindrical roller on the sun gear and each planetary gear. These rollers provide the same function as a bearing would. They also allow the TDS set to achieve greater load sharing between its gears, thereby increasing load carrying capacity, says president & engineering manager Rob Kennedy.

He adds that the gear-bearing product can achieve several gear ratios in the same space by swapping out several gears. These ratios include 3.8:1, 28:1, 43:1 and 1,298:1. TDS is already in the midst of proving this ability to a Chicago manufacturer of electric gear motors.

TDS is manufacturing three prototype sets, each with the same diametral and axial envelopes, 3" and 2.6" respectively, but each with a different ratio, 38:1, 55:1 and 346:1. Based on TDS's detailed drawings, these ratios can be achieved by replacing the first stage gears (sun, planets and ring) without replacing the second stage ones, Kennedy says.

TDS is looking to use the planetary set for speed reducing in conjunction with electric and hydraulic motors that drive actuators, pumps, compressors and gearboxes.

Kennedy says TDS intends to market the gear-bearing product to the automotive and aerospace industries and describes several possible applications in both industries. The planetary set could be used to open and close sliding doors on vans, to open and close lift gates on SUVs and to move power seats back and forth in automobiles. In aerospace, the assembly could be used to open and close bay doors of cargo and fighter aircraft and to extend and retract aircraft wing flaps.

NASA conceived and initially developed the gear-bearing product and later licensed TDS to look into its commercial uses and to manufacture it. TDS itself is a custom-product design firm that specializes in hydraulic and pneumatic products and applications and in turbomachinery.

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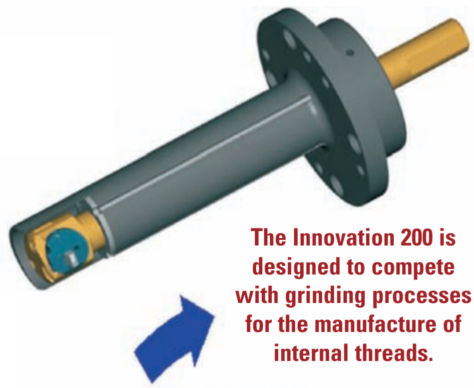
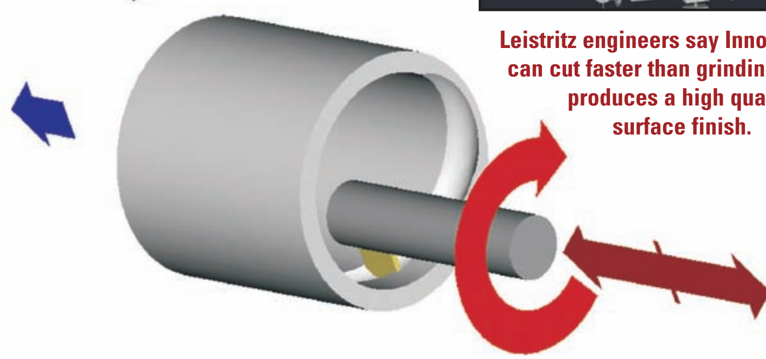
New Internal Whirling Process from Leistritz



The Innovation 200 is designed to compete with grinding processes for the manufacture of internal threads.

Leistritz engineers say Innovation 200 can cut faster than grinding but still produces a high quality surface finish.



Leistritz Corp., of Allendale, NJ, has developed the Innovation 200 internal whirling process for machining difficult internal threads. Performed without coolants, internal whirling is compatible with either soft material or steel as hard as 65 HRC using carbide tooling.

Engineers at Lesitritz saw the need for a process that did not involve the high spindle speeds associated with grinding, which can require speeds in excess of 30,000 rpm. In addition to this, the complications of distributing the coolant to the exact interface between the workpiece and the grinding wheel add more difficulty to the grinding of internal threads. Finally, the grinding wheel must be dressed with a compensated contour to generate an acceptable thread profile.

"Innovation 200 does not require a coolant," says Ralph Wehmann, sales manager at Leistritz. "Number two, we don't need such a high spindle speed."

This machine is an inverted vertical machine with the chuck on top. The "z" axis longitudinal travel moves the chuck and its rotary "c" axis up and down. The head remains stationary.

One of the key advantages of this design is that it isolates vibration caused by the oscillating masses directly into the massive machine bed. The bed is very dense with different vibration absorption characteristics. Hydrostatic guideways for all axes guarantee high precision and accuracies.

Integrating automatic loading is easier because the chuck remains above the work, and the combined "x" and "z" axes have expanded travel areas. Therefore, the chuck can act as its own gripper system and go to a familiar location to pick up a blank workpiece.

Sample parts made by internal whirling.

Probes can be used to obtain the true position of timing holes, so the thread can be precisely timed for the start point. It is also possible to hard turn the bore prior to whirling.

A 12,000 rpm milling spindle is available for auxiliary operations. As the tool rotates at cutting speeds of approximately 600 sfm, it also oscillates axially. The amount of oscillation is calculated from the thread parameters and tooling circle data. Higher lead threads require larger oscillation amounts, and the oscillation continually maintains the cutting insert normal to the helix angle. According to the company's press release, a completed thread is achieved in a single whirling process.

"Really, the most unique thing is that as it turns, it cuts in and at the angle," Wehmann says, "When it's not cutting, it oscillates back so that each revolution it moves up and down. On a half turn, it oscillates in and the other half oscillates out within a single revolution. The oscillation amount is programmable."

As far as the quality of the finished product, Wehman says it's as good as grinding for every dimension, and it can cut faster with high-quality surface finish.

Wehmann estimates that the Innovation 200 is comparable price-wise to grinding, with price tags at about $600,000–$700,000 for the complete machine. However, with threading, there is no danger of additional costs for coolant or filtration systems, he says.

Innovation 200 is designed for ball screws, acme nuts, rope or other threads and power generation. Berger Precision, a ball screw company in Germany, has already tried this concept to produce an automotive component with a 6 lead thread used in a light truck to control automatic vehicle leveling.

"That machine was one of the first delivered in August. It's been running with no problems since then," Wehmann says.

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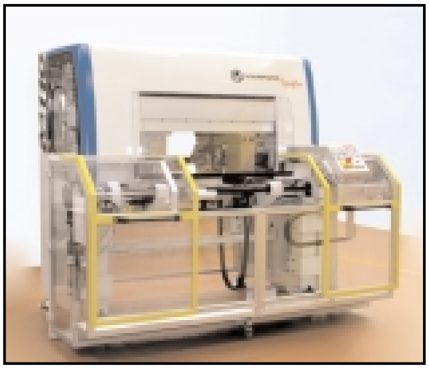
New Measuring Machine from Marposs



The M110 Optoflex from Marposs is designed for the dimensional inspection of shafts and shaft-like parts. Utilizing optoelectric technology, this machine can inspect a number of parts while performing different types of measurements.

Designed for use in a shop floor environment, the machine can be integrated into a product line with the possibility of feedback to the machine tool. It can also be used as an audit station or for post-process inspection applications.

According to the company's press release, the machine offers a high degree of flexibility by using shadow cast opto-electronic solutions. As opposed to the traditional contact and air-to-electric solutions, this method allows measurements to be performed without touching the part.

A single light blade produces measurements and an LVDT probe can be incorporated for axial TIR measurements. Performance checks include static and dynamic diameters, stroke and index, straightness and profile analysis.

For more information, contact Marposs Corp. of Auburn Hills, MI, by telephone at (248) 370-0404 or on the Internet at *www.marposs.com.*

New Grinder from United Grinding

The new S33 universal cylindrical grinder from United Grinding can grind longer shaft-type parts due to its 1,000 mm capacity between centers.

According to the company's press release, the machine was designed specifically for the North American market.

Other features include digital direct drive, V- and flat guideways coated with abrasion-proof Granitan S200 and three-phase servomotors with 40 mm diameter prestressed precision ballscrews that power the X- and Z-axes.

In addition, this machine features Fanuc 21i digital control and axis drive components. The control package has Studer grinding cycles for diameters, shoulders (left and right), tapers (negative and positive) and contours.

For more information, contact United Grinding Technologies of Miamisburg, OH, by telephone at (937) 847-1222 or on the Internet at *www.grinding.com*.

New Measuring Machine from Mahr Federal

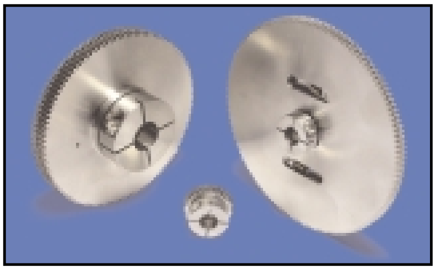
The MarVision from Mahr Federal is designed to provide a mid-sized solution for high precision applications that require multiple modes of inspection.

According to the company's press release, this machine is fully automatable and incorporates a CCD camera, TTL laser and mechanical touch-probe sensor to provide precision measurement in a design best suited for use near production line and work cell environments.

The software is available in an open-architecture style that supports use of the GUI for operators. Other features include a measuring table and crossed axes equipped with precise roller bearings and precision, backlash-free, DC servo drives.

For more information, contact Mahr Federal of Providence, RI, by telephone at (800) 333-4243 or on the Internet at *www.mahr.com.*

New Instrumentation Gears from Precision Alliance



A new line of specialty and instrumentation gears from Precision Alliance includes spur, segment, anti-backlash synchro, beveled, worm and customized gears.

According to the company's press release, all are rated at AGMA Class 14 and beyond. Available in a variety of standard materials including bronze, aluminum and stainless steel, these gears come with or without hubs.

For more information, contact The Precision Alliance of Fort Mill, SC, by telephone at (803) 396-5544 or on the Internet at *www.theprecisionalliance.com*.

New Geared Motors from Danfoss Bauer

The BG Series of three-phase geared motors from Danfoss Bauer is engineered to drive a variety of types of machinery and equipment.

The motors have a power capability of 0.075–50 hp with output speeds of 0.3 –540 rpm and a torque range of 177–163,000 lbf.-in. Available as a standard two-stage design, the motors include stator winding of high quality insulated enameled copper wire, an aluminum die-cast cage rotor and gear wheels that are constructed of high tensile and die-cast steel.

According to the company's press

New Coated Insert from Seco-Carboloy

The TP3000 from Seco Carboloy is a new generation of coated inserts for steel turning to be used on machining applications such as heavy roughing, interrupted cuts, heavy feed rates and workpiece materials with uneven machinability between different material batches.

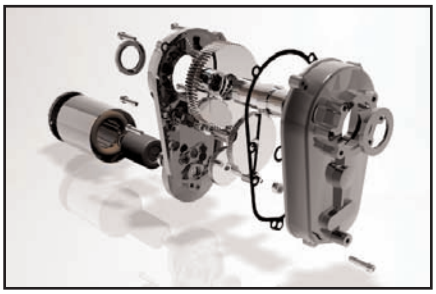
The carbide incorporates a hard coating composed of an alpha alumina material with a fine grain structure. According to the company's press release, the coating's smoothness prevents the built-up edge that can occur in machining stainless materials.

For more information, contact Seco-Carboloy of Warren, MI, by telephone at (586) 497-5000 or by e-mail at *carboloy@carboloy.com*.

release, the gearbox housing is completely enclosed for operation in harsh conditions and prevents lubricant loss and dirt build-up.

For more information, contact Danfoss Bauer of Somerset, NJ, by telephone at (732) 469-8770 or on the Internet at *www.danfoss-bauer.com*.

New Bison Gearmotor



The 562 Series hollow-shaft gearmotor from Bison Gear has an increased center distance and greater clearance for wider faced gear shafts.

The 562 offers an optional fourth stage of gear reduction and provides ratios up to 2,200:1 in the same envelope. According to the company's press release, this allows for a smaller input motor without sacrificing output torque.

Featuring hollow shaft options with multiple mounting configurations, this gearmotor is offered with a variety of AC and DC input motors, including three-phase and inverter duty.

For more information, contact Bison Gear of St. Charles, IL, by telephone at (630) 377-4327 or on the Internet at *www.bisongear.com.*

New DC Gear Motor from Midwest Motion Products

The new MMP.PDSF80–12V.018.1 geared motor from Midwest Motion Products can accept VDC power sources, including batteries.

According to the company's press release, the reversible geared motor measures 2.8" in diameter by 5.6" in length with an 80 mm square gearbox. The output shaft is 10 mm in diameter and includes a square shaft key and keyway.

The geared motor has a shaft output speed of 160 rpm and is rated for 26 in.–lbs. of continuous torque at 3.9 amps of current. Popular options with this servomotor include integral optical encoders, fail-safe brakes, analog tachometers and optional ratios from 3:1 to 180:1.

For more information, contact Midwest Motion Products of Watertown, MN, by telephone at (952) 955-2626 or on the Internet at *www.midwestmotion.com.*

New Compact Gearbox from ZF Industrial

The CG gearboxes from ZF Industrial are best suited for use in robotic or other automation applications, like electronics or other OEM manufacturing industries.

According to the company's press release, these gearboxes feature a backlash absorbing system that includes a tapered gear tooth profile optimized for the highest levels of precision. Backlash rates are less than 0.5 arc-min.

In addition, the gearboxes include a large output flange that is supported by oversized cylindrical roller bearings that can produce an assembly with an output torque of up to 2,500 N-m. Acceleration torques of up to 2,500 N-m can be achieved due to an interlocking power transmission design that reduces the gearbox's length and provides efficiency levels up to 94%.

Available in five models with ratios from i = 20 to i = 103, the units can accept speeds up to 5,300 rpm and maintain sealing levels up to IP65.

For more information, contact ZF Industrial of Nottingham, U.K., by e-mail at *rob.pearson@zf-group.co.uk*.

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Good Basic Design or Sophisticated Flank Optimizations? Each at the Right Time

Dr. Hermann J. Stadtfeld

is a co-founder of BGI GmbH & Co KG. Located in Eisenach, Germany, the company specializes in development and application of bevel gears for the automotive industry. Previously, Stadtfeld served as a vice president and head of research & development for The Gleason Works in Rochester, NY, and as director of engineering and research with Oerlikon GearTec AG in Zürich, Switzerland. He started his technical education with an apprenticeship at the ZF Friedrichshafen AG, a major European gear manufacturer, and received his doctorate in mechanical engineering from the Technical University of Aachen, Germany, researching bevel gears for his dissertation. Stadtfeld has spent more than 20 years developing software, hardware and other processes for gear design and optimization. Besides his BGI work, he trains engineering students at universities and gear engineers in companies around the world.

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More strength, less noise. Those are two major demands on gears, including bevel and hypoid gears. Traditionally, gear engineers have met the first demand by changing a gear set's basic parameters—tooth height, pressure angle, spiral angle, etc.—and met the second by making flank form modifications, also known as ease-off.

With today's computer technology, though, many engineers are modifying flank topography and discovering their gear sets are both stronger *and* quieter. Unfortunately, this coincidence is tricking some engineers into believing that ease-off itself adds strength to gear sets. In fact, the flank form modifications are only allowing the sets to make greater use of the strength that was possible in their basic designs.

Most strength optimizations require changing a gear set's basic parameters. Most minor reductions in gear noise are made by optimizing just flank topography. In some cases, though, there are gear sets that don't use all their possible strength because they need sophisticated ease-offs but lack them.

The challenge for gear engineers is to know when to change which, basic parameters or flank topography, to increase a gear set's strength. To know that, engineers need to understand both basic parameters and flank form modifications. The parameters are discussed in the side article, "Influences of Major Basic Parameters."

Changing Basic Parameters Not Always Possible

A gear set's basic parameters offer gear engineers many ways to optimize the set's performance, but changing them may not be an option in many cases. Gear engineers often must optimize for strength or noise in existing gear sets that have been in service for many years, but that need improvement because their gearbox requirements have changed. One example is a prime mover in which the gear set must transmit more power—but still fit in the same space inside the existing housing. Another example is a gear unit in which noise must be reduced.

Generally, a strength optimization requires a new or improved basic design. The exception is a gear set with an optimal basic design but with a conventional ease-off, one that allows for improvement. In that case, a gear engineer will significantly strengthen the set via sophisticated flank modifications. Such ease-off may be the engineer's only option if the space for the set can't be increased, as in the prime mover example.

In the other example, gear engineers are usually told not to change the noisy gear set's basic design. The reasons are time and money. If basic parameters are changed, the gear set will require a strength requalification. But requalifying an automotive axle drive unit, for example, may cost $40,000 and take six months.

Fortunately, sophisticated optimizations of flank form have a neutral or positive influence on a gear set's strength. It's widely accepted in the industry that a gear set with flank modifications doesn't need its strength requalified as long as the modifications didn't change the basic parameters. So, whenever possible, gear engineers would be well advised to follow this rule: "Keep the existing, proven basic design and adjust only the contact topography."

Flank Form Modifications

Flank form modifications are deviations from the correct, or conjugate, flank form. If a ring gear were used in a virtual process to generate a pinion, the result would be a pinion that has—at all times—line contact with one or more gear tooth flanks and rolls perfectly with no motion transmission error. This would be a conjugate tooth system.

But this system isn't possible in the real world. As gear engineers know, gear and pinion interaction is affected by manufacturing tolerances and load-dependent deflections of the gears, bearings and gear-

Influences of Major Basic Parameters

A gear set's basic parameters establish the potential of its properties, including strength and noise. There are many major basic parameters, and they have a variety of effects on the operating performance of a gear set. Gear engineers need to understand these effects, especially when optimizing gear sets.

A tall tooth is more elastic than a standard tooth. The tall one's thickness is less than the standard one's, but it isn't reduced by much. The tall tooth's height is limited to the minimum point width of the cutting blade. But the tall tooth has advantages over the standard one: higher effective contact ratio and lower impact intensity at the tooth entrance. In other words, there are strength and noise advantages without significant disadvantages.

A high spiral angle reduces normal tooth thickness and increases the theoretical contact ratio. The reduced thickness has a second order influence on root bending stress, reducing a gear's load carrying capacity. The higher contact ratio has a less than proportional influence on load sharing, but the teeth will mesh more smoothly and have reduced tooth impact excitations.

Reducing pressure angle increases the root and top widths of teeth. Given this effect, a gear engineer can increase tooth depth because blade point width and pointed tops won't reach their limits until much later.

A fine-pitch gear will have a higher contact ratio and proportionally shorter and thinner teeth than a coarse-pitch gear of the same circumference. These effects result from the fine-pitch gear having more teeth on its circumference. The fine-pitch gear will also make much less noise, but its load carrying capacity will be significantly lower.

A smaller face width means a low transverse contact ratio and a short contact pattern. It also means there will be a continuous risk of edge contact even when deflections under load are small. But a wider face width creates problems in manufacturing. Chips are too long, and heat treat distortion is significant. Moreover, a wider face width doesn't increase strength to the expected extent. Among bevels and hypoids, face width has an optimal value that is 33% of the ring gear's mean cone distance.

A hypoid offset lowers the center of gravity for rear-wheel-driven vehicles, but it creates relative sliding in the tooth lead direction, increased pinion diameter and a higher pinion spiral angle. The offset also leads to better hydrodynamic lubrication film, good dampening properties, higher overall contact ratio

Continued on page 23.

box. Line contact becomes edge contact along the teeth boundaries, and motion transmission takes on a saw-tooth profile. Moreover, bevel gears—which include hypoid gears—require relief in the profile and lead directions from their conjugate flank forms. This relief starts in a tooth's center and provides circular relief towards the tooth ends, top and root.

Today, the ease-off of a typical bevel gear set consists of a circular relief in the lead direction (length crowning), a circular relief in the profile direction (profile crowning) and a circular relief in the path-of-contact direction, diagonally across the flanks (flank twist). Figure 1 shows calculated contact analysis of these three basic corrections in three columns, with each ease-off on top, its tooth contact in the middle and its resulting motion graph on the bottom.

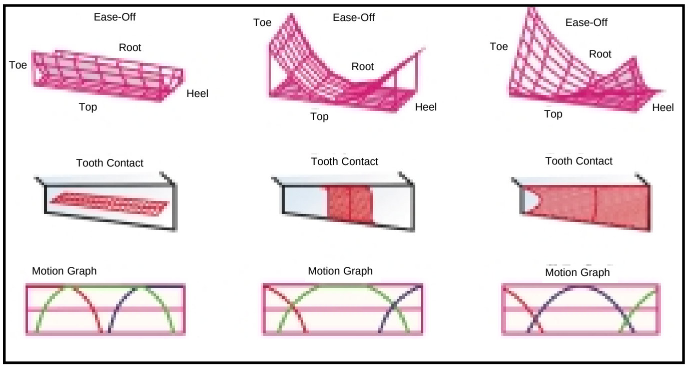

Figure 1— Profile crowning (left column), length crowning (center column), and flank twist (right column).

The latest computer developments also make it possible to apply higher order flank modifications along the path of contact. These modifications are created during the generating of pinions and gears via a machine's axes, through a combination of additional roll-position-dependent movements. All these modifications are plotted together over the tooth projection plane, just like the result of a coordinate measurement. This ease-off represents all deviations from the conjugate tooth system, regardless of whether the modifications were done in the pinion, the ring gear or both.

What Can Ease-Off Do?

Flank form modifications can be very powerful tools for noise and strength optimizations, especially since the modifications are available for the leading machine tool brands. Ease-off can increase strength if, for example, a gear set has a contact ratio that's lower—even much lower—than the ratio possible given the set's existing basic parameters. With the right modifications, the contact ratio can be increased, so the gear set makes greater, more efficient use of its existing basic properties. As gear engineers know, when contact ratio is increased, strength is increased.

Using higher order modifications, gear engineers may even be able to define different flank sections, allowing them to distinguish between the changing requirements of the contact entrance and exit areas as well as the mean contact section. Figure 2 shows a motion graph that can result from higher order flank form modifications.

Also, higher order modifications can improve load sharing between two or three consecutive pairs of teeth, decreasing surface compression stress and root bending stress, possibly by 25% or more.

A gear set's basic geometry defines the modified contact ratio as a result of profile and transverse contact ratio, but load sharing between consecutive pairs of teeth depends on the load and ease-off between meshing flanks. If the ease-off is parabola shaped (of second order) in the profile and lead directions, then the pair of teeth with contact areas close to the center transmits 60–90% of the load. Conversely, this ease-off decreases the load transmitted by the pair of teeth with contact towards the entrance area and the pair of teeth with contact towards the exit. Those pairs transmit only 10–40% of the load.

Good Basic Design or Sophisticated Flank Optimizations? Each at the Right Time

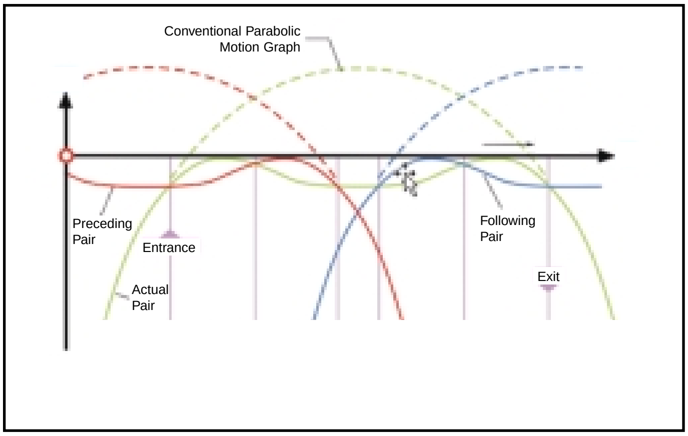

Figure 2— Sophisticated flank optimization with higher order motion graph.

Moreover, gear engineers can use a higher order ease-off to reduce the load transmitted by the pair of teeth with center contact. Such an ease-off can reduce it to 40%. This reduction in turn increases up to 60% the load transmitted by the pair with entrance contact and the pair with exit contact.

This effect has no drawback. The load distribution between consecutive teeth is more equal, so the load change in pairs of teeth during mesh is smoother and less abrupt, as shown in Figure 2. This is a win-win situation that is just now being used to its full extent in the gear industry.

In both cases—parabola shaped ease-off and higher order ease-off—the calculated contact ratio remains constant, but only the higher order ease-off uses the contact ratio more intelligently.

Gear engineers should be careful, though, if a noisy gear set has previously been optimized using higher order ease-off, such as Gleason UMC or Klingelnberg Modified Crowning, and the set's strength was based on the unconventional, optimized design. Focusing too much on the noise, an engineer might try to reduce the noise by superimposing additional higher order modifications on the existing ease-off. The new modifications might partly reverse the existing ease-off's positive effect, sacrificing the design's strength.

The engineer, however, can check on whether his noise optimizations sacrificed strength by using finite-element calculation systems dedicated to high-accuracy calculating of gears. These systems can be used to verify a gear set's before and after situations. However, if the calculation shows an increase in critical stress, the engineer would have to change the basic design to keep the stress below the required limits and to enhance the transmission quality.

Conclusions

Gear engineers should be aware that basic gear design parameters create the foundation for a gear set's properties and that flank form modifications can help the set fully realize its properties. A bad ease-off, even a conventional one, will keep a gear set from achieving the potential offered by its properties. A good ease-off will allow it to achieve its potential by taking optimal advantage of those properties. By optimally combining basic design and flank modifications, gear engineers can create gear sets that are able to transmit twice as much torque as other sets of the same size and still be quieter than those other sets.

Generally, a strength optimization requires a new basic design. The exception is a basic design that is found to be optimal but the ease-off is conventional and leaves room for improvement; here a sophisticated flank modification will make a big difference and might be the only possibility if the space of the gear set can't be increased.

A noise optimization generally requires a flank modification only. The exception here is if the gear set already employs higher order flank modifications. In that case, the noise reduction may not be easily possible. If an improvement still seems reasonable, a finite element calculation of the before and after situations is necessary. If the calculation shows critical stress is increased, the gear engineer will need to change the basic design to keep the stress below its required limits and to reduce noise through better transmission quality.

Gear engineers should see the possibility of developing a new basic design as a great chance, not as a burden. They should always start with the previous design because they may be able to make many small improvements to better adjust the gear set to its changed requirements. Also, they should never make the mistake of thinking that the effort of thoroughly developing the basic parameters can be reduced to making only higher order flank form modifications. They must always remember the rule: "The basic design parameters set the direction for the gear set's properties, sophisticated flank form modifications will only give us better access for realizing the properties' potential."

Continued from page 21.

and increased pinion strength.

Whenever possible, gear engineers should design bevel gear sets to include hypoid offsets—that is, they should design hypoid gear sets. A hypoid gear pair is preferable to a spiral bevel gear pair because the hypoid offset provides strength and noise advantages that are significant and should not be underestimated. However, the hypoid set requires high pressure hypoid oil, so it will have scoring resistance. Also, if the offset is too high, the sliding velocity causes increased operating temperature, additional energy loss and the risk of scoring. The optimal hypoid offset lies between 10% and 20% of the ring gear diameter. The coolest running bevel gears are hypoids with an optimal offset: Their efficiency and strength are unbeatable compared with spiral bevel gear sets.

A small cutter radius increases a gear set's contact ratio. It does so because, compared with a conventional or large cutter radius, a small one increases spiral angle on the heel and decreases the "natural" loaded contact movement towards the heel. So gear engineers should choose a cutter diameter that allows the mean contact point to move from light load to maximal load using about 30–50% of the face width, while the contact area increases to cover the entire flank area without hard edge contact. However, an engineer's choice of cutter diameter depends on displacement between pinion and gear under load in a gearbox housing.

A face-hobbed tooth surface, when lapped, will present an optimal condition for smooth operation. This surface depends on face hobbing because the machining flats and contacting lines between pinion and gear cross each other under an angle that provides pockets for the lapping compound. In face milling, the flats and lines are parallel, so they don't create pockets. When a face-hobbed gear set is rolled in a lapping machine, the compound will fill the pockets so only the peaks of the machining flats have contact at the start of lapping. This contact leads to initial material removal that is rapid but uses low torque only.

As lapping progresses, this rapid removal reaches its natural end when 90% of the cutting structure is removed. For face hobbing, like for face milling, a gear engineer should pick a cutter radius that optimally controls loaded contact movement.

Gear engineers should also keep in mind: If the cutter has a high number of starts, the tooth lead function will approximate the shape of an involute. This approximation results in a gear set with additional insensitivity to deflection. In particular, it protects the teeth from edge contact in high load situations.

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New Approach to

By Veniamin I. Goldfarb and Alexander A. Tkachev

Computerized Design of Spur and Helical Gears

Management Summary

In gear design, choosing shift coefficients x_1 and x_2 for a pinion and gearwheel may be crucial to providing the necessary quality for spur and helical gears. This choice can be made by means of a so-called dynamic blocking contour (DBC), which is described in this paper. The contour contains important information about a number of quality parameters for a gear design and can be drawn with help from a "Contour" CAD system when choosing a gear's initial parameters.

When applying a DBC, the designer can predict the gear's quality at the earliest stage of its design, i.e. when choosing the values x_1 and x_2. This ability allows a designer to increase considerably the productivity and quality of the gear design process, eliminating complex iterative procedures for finding optimal solutions.

Introduction

One effective way to enhance gear design is to allow users to forecast the quality of a gear at the initial design stage—that is, when choosing the basic, initial geometric parameters. Such opportunity decreases considerably the range of variable parameters and raises the design quality and efficiency.

This idea is implemented in a new approach to the design of spur and helical gears. The approach is based on the application of a special type of geometrical objects named blocking contours (Refs. 1–4). These BCs are used to choose rational values for addendum modification coefficients (profile shift coefficients) of a pinion and gearwheel, x_1 and x_2, respectively. The influence of shift coefficients on the calculation of a gear's tooth geometry and kinematic and strength parameters is widely known. Given this influence, many properties of a gear can be estimated at the stage of x_1 and x_2 selection by means of blocking contours—that is, at the initial design stage. Thus, the process of choosing optimal gear parameters is simplified considerably.

Based on the BC concept, computer-aided design of spur and helical gears has been developed to achieve the stated design principles. In order to better illustrate the design, CAD shows the meshing processes of the gear pair and shows meshing element generation by a rack-type cutting tool. Release of the developed system allows users to master the methodology and possibilities of gear design using the BC concept to evaluate the selection of shift coefficients to obtain specified gear properties.

Dr. Veniamin I. Goldfarb
is a professor in the production mechanical engineering department of Izhevsk State Technical University, located in Izhevsk, Russia. He's also the department's head and is director of the university's Institute of Mechanics. Goldfarb specializes in the theory and practice of gear design, investigation and production. In addition, he's chair of the gearing technical committee of the International Federation for the Theory of Machines and Mechanisms.

Dr. Alexander A. Tkachev
is an assistant professor in the university's production mechanical engineering department, but he's mainly responsible for programming gear design software, especially for designing spur and helical gears. Tkachev is also executive engineer of Izhevsk Mechanical Plant, a company that produces power tools and other products. As executive engineer, he is involved in developing CAD systems for gears.

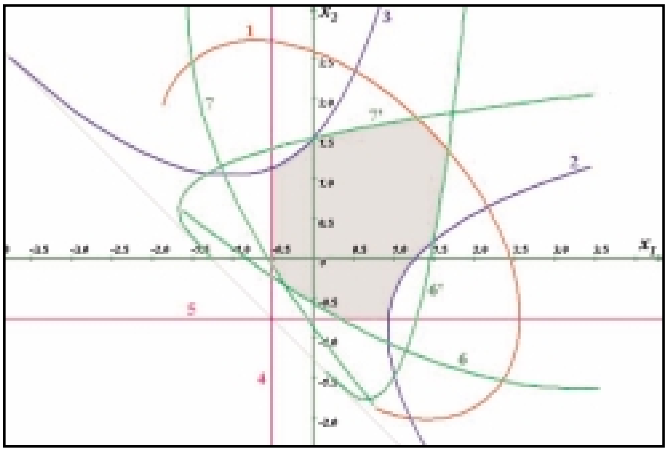


Figure 1—View of a blocking contour.

Essence of the Blocking Contours Method

A set of lines is plotted on the coordinate plane x_1, x_2, the lines corresponding to certain values of basic factors which limit the valid process of meshing. These values can include maximum allowable ones. The factors are:

1.) the absence of undercut when cutting pinion and gearwheel teeth,

2.) the absence of interference in operating meshing,

3.) a transverse contact ratio ε_α—including its allowable value, for example $\varepsilon_\alpha = 1$, and

4.) tooth thickness at the addendum circle s_a and its minimum allowable value, for example $s_a = 0$, corresponding to sharp or pointed gear teeth.

When intersecting with each other, these lines form a closed area on the coordinate plane. The point (x_1, x_2) for the pinion (x_1) and wheel (x_2) must be within this area, otherwise one or more of the gear operation criteria stated above will not be true. Figure 1 shows limiting lines in terms of undercut (straight lines 4 for a pinion and 5 for a wheel), interference (two lines for a pinion, 6 and 6', and two for a wheel, 7 and 7'), transverse contact ratio $\varepsilon_\alpha = 1$ (line 1) and the sharp teeth of the pinion ($s_{a1} = 0$, line 2) and wheel ($s_{a2} = 0$, line 3). The area defined by these lines (the figure's shaded area) is the domain of coefficients x_1 and x_2 which meet the above criteria. This shaded area is bounded by parts of the abovementioned limiting lines 1–7'. These parts make up a closed line. This line is called a blocking contour (BC).

To determine the limiting lines, the following mathematical equations should be used.

1.) Lines for the undercut are straight lines parallel to the coordinate axes:

$$x_i = h_l^* - h_a^* - \frac{z_i \sin^2\alpha_t}{2\cos\beta} \qquad (1)$$

where $i = 1$ for a pinion, $i = 2$ for a gearwheel;
h_l^* is a boundary height factor equal to the ratio of the height of the rectilinear section of an initial contour ("producing, or basic rack") to the module m; h_a^* is an addendum factor, one of the main basic rack parameters (in most cases $h_l^* = 2h_a^*$, the standard value $h_a^* = 1$);
z_i is the tooth number (z_1 is for a pinion, z_2 is for a gearwheel);
β is the helix angle ($\beta = 0°$ for spur gears);

$$\alpha_t = \arctan\left(\frac{\tan\alpha}{\cos\beta}\right)$$

is the pitch profile angle in lateral cross-section; here α is the pressure angle, another important basic rack parameter.

2.) Lines of interference.

For a pinion:

$$a_w \sin\alpha_{tw} - 0.5 d_{b2} \tan\alpha_{a2} = 0.5 d_1 \sin\alpha_t - \frac{h_l^* - h_a^* - x_1}{\sin\alpha_t} m \qquad (2)$$

where
a_w is the gear center distance;
α_{tw} is the gear pressure angle

$$\mathrm{inv}\alpha_{tw} = 2\frac{x_1 + x_2}{z_1 + z_2}\tan\alpha + \mathrm{inv}\alpha_t \qquad (3)$$

where "inv" is the designation of an involute function of an angle ($\mathrm{inv}\alpha = \tan\alpha - \alpha$), sometimes also called an involute angle; a_w and α_{tw} are interconnected with a known relation

$$a_w = a\frac{\cos\alpha_t}{\cos\alpha_{tw}}$$

where $$a = \frac{(z_1 + z_2)m}{2\cos\beta}$$

is the pitch center distance;
d_{b2} is the gearwheel base diameter;
α_{a2} is the tooth profile angle of a gearwheel at the point of tooth addendum circumference

$$\cos\alpha_{a2} = \frac{d_{b2}}{d_{a2}}$$

where d_{a2} is the gearwheel tip diameter (diameter of the addendum circle); and
formulae for d_{bi} and d_{ai} are well known.

With these formulae, one can obtain the expressions for $\cos\alpha_{a1}$ and $\cos\alpha_{a2}$, which include shift coefficients x_1 and x_2:

$$\cos\alpha_{a1,2} = \frac{z_{1,2}\cos\alpha_t}{2(h_a{*} - x_{2,1})\cos\beta + (z_1 + z_2)\dfrac{\cos\alpha_t}{\cos\alpha_{tw}} - z_{2,1}} \quad (4)$$

d_1 is a pinion reference diameter ($d_1 = mz_1$).

In the equation for a gearwheel, indices 1 and 2 interchange their places.

Using the known relations for a_w, d_{b2} and d_1 in equation 2, we can obtain the following correlation for the pinion:

$$(z_1 + z_2)\cos\alpha_t\tan\alpha_{tw} - z_2\cos\alpha_t\tan\alpha_{a2} = z_1\sin\alpha_t - 2\frac{\cos\beta}{\sin\alpha_t}(h_l{*} - h_a{*} - x_1) \quad (5)$$

where α_{tw} and α_{a2} depend upon x_1 and x_2 according to equations 3 and 4.

3.) Line of transverse contact ratio ε_α

$$\frac{1}{2\pi}[z_1\tan\alpha_{a1} + z_2\tan\alpha_{a2} - (z_1 + z_2)\tan\alpha_{tw}] = \varepsilon_\alpha \quad (6)$$

where ε_α is a given value of transverse contact ratio, e.g. $\varepsilon_\alpha = 1$.

4.) Lines of tooth thickness at the addendum circle s_{ai} (i = 1 or 2, as before):

$$s_{ai} = d_{ai}\left(\frac{\pi}{2z_i} + \frac{2x_i\tan\alpha}{z_i} + \mathrm{inv}\alpha_t - \mathrm{inv}\alpha_{ai}\right) = k_{si}m \quad (7)$$

The value k_{si} determines the tooth thickness at the addendum circle proportional to the module. For the minimum allowable value $k_{si} = 0$, the expression defines the line which is the boundary limited by sharp or pointed teeth.

Using the known expression for d_{a1}, we can obtain the following formula for the pinion:

$$\left[2(h_a{*} - x_2) + \frac{z_1 + z_2}{\cos\beta}\frac{\cos\alpha_t}{\cos\alpha_{tw}} - \frac{z_2}{\cos\beta}\right] \bullet \left[\frac{\pi}{2z_1} + \frac{2x_1\tan\alpha}{z_1} + \mathrm{inv}\alpha_t - \mathrm{inv}\alpha_{a1}\right] = k_{s1} \quad (8)$$

which, for a given k_{s1}, is the equation of the line s_{a1} in the plane x_1, x_2.

The equation of the line s_{a2} for the gearwheel is almost the same, indices 1 and 2 interchanging their places.

Equations 1–8 have been used for calculation and construction of BC limiting lines in the coordinate plane x_1, x_2. For more fast and reliable calculation of the lines, special algorithms were developed and used in a "Contour" CAD system (see the "Example" section below) and proved to be correct and effective. This system is a gear CAD system created by the authors based on the BC concept.

The lines ε_α, s_{a1} and s_{a2}, which are among the mentioned BC curves, depend on initial parameters such as tooth number z_1 and z_2 for a pinion and gearwheel respectively, helix angle β and basic rack parameters—in particular, pressure angle α and addendum factor $h_a{*}$. They also depend on values of parameters for which these lines are constructed. When designing a gear, the values of these latter parameters are taken either from standards or from practice. For example, the minimum allowable value of ε_α is chosen as 1.0, 1.1 or 1.2 in accordance with different standards and the type of gear being designed (spur or helical); the minimum allowable value of $s_{a1,2} = 0.25m$ or $0.4m$. In the practice of design and research, however, it is often necessary to operate with non-standard values for these parameters. Moreover, in order to increase the flexibility of the design process, one may need to simultaneously construct several lines of one type with different values for the determining parameter. This was the reason for the development of the dynamic blocking contour (DBC) concept.

Let's consider the case when an engineer designs a helical gear and wants the transverse contact ratio to be within the range $1.0 \le \varepsilon_\alpha \le 1.4$. Then the BC configuration containing two transverse contact ratio lines $\varepsilon_\alpha = 1.0$ and $\varepsilon_\alpha = 1.4$ (represented by lines 1 and 1' in Fig. 2) is required. If the engineer takes shift coefficients [point (x_1, x_2)] within the area between these curves (the figure's shaded area), the value of ε_α is guaranteed to be within the given range.

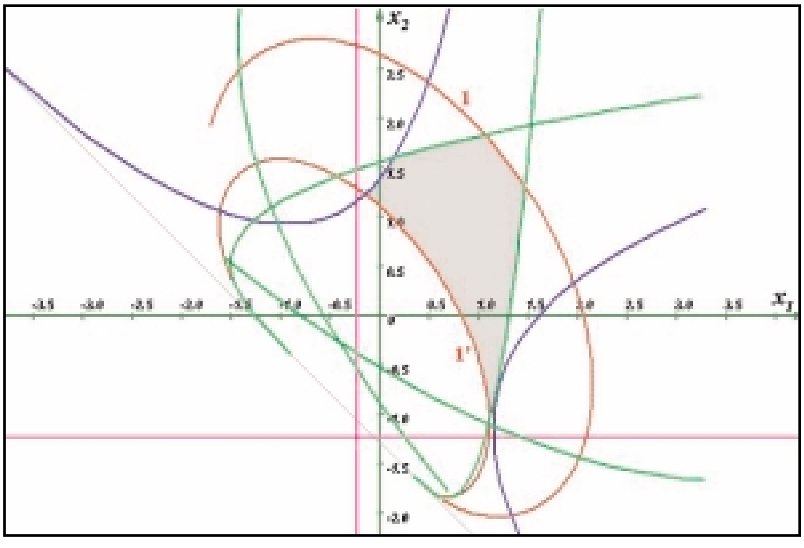


Figure 2—Blocking contour with $\varepsilon_\alpha 1 = 1.0$ (line 1) and $\varepsilon_\alpha 1 = 1.4$ (line 1').

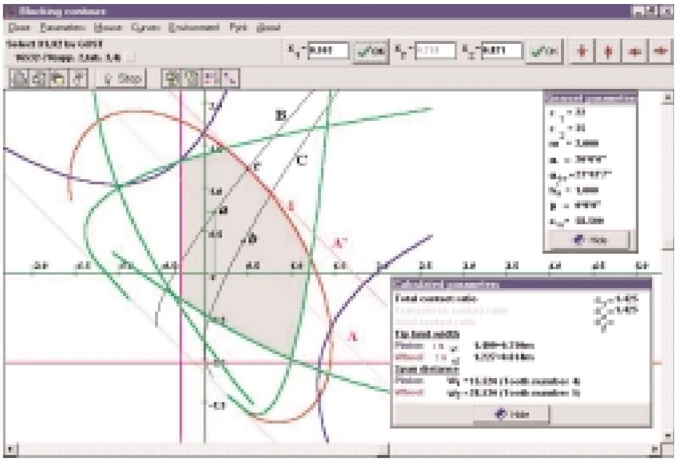


Figure 3—Description of features of BC lines.

Some Features of Blocking Contour Lines

A blocking contour (BC) has a property which becomes very important for implementing the abovementioned concept of gear design. The property is: In order to calculate and construct a BC, it is necessary to have certain minimum information, which is usually known at the first design stage:

1.) tooth number of pinion z_1 and gearwheel z_2;
2.) initial contour (producing rack) parameters—in particular, profile angle α, tooth addendum coefficient h_a* and radial clearance coefficient c*; and
3.) helix angle β (for helical gears).

In the simplest case, when designing a spur gear (β = 0°) with "standard" initial contour (α = 20°, h_a* = 1, c* = 0.25), it is necessary to know only values z_1 and z_2. Note here that the geometric configuration of a BC does not depend on gearing module m.

In order to apply a BC more effectively, "additional" lines within the coordinate plane x_1 and x_2 are used along with the abovementioned lines generating the blocking contour. These lines, like the lines forming the BC, are the geometrical representation of several properties of a gear under design. Let's consider some of these lines (Fig. 3).

I. Straight line with the equation $x_1 + x_2 = x_\Sigma$ (line A in Fig. 3)

Here x_Σ is the so-called coefficient of shifts sum. This line is important in the practical operation of a blocking contour. When the value of the shifts sum coefficient is fixed (x_Σ = constant), we obtain constant values of pressure angle α_{tw} and center distance a_w according to formulas

$$\mathrm{inv}\alpha_{tw} = 2\frac{x_\Sigma}{z_1 + z_2}\tan\alpha + \mathrm{inv}\alpha_t \quad \text{and}$$

$$a_w = a\frac{\cos\alpha_t}{\cos\alpha_{tw}}$$

which appeared earlier.

From a geometrical point of view, it means that:

1.) the position of the line $x_1 + x_2 = x_\Sigma$ in the coordinate plane x_1, x_2 is defined by the value of one of three unambiguously interconnected parameters x_Σ, α_{tw} or a_w;
2.) pressure angle α_{tw} and center distance a_w remain constant when displacing the point (x_1, x_2) along the stationary straight line $x_1 + x_2 = x_\Sigma$ (that is along the line with the fixed x_Σ = constant). In practical terms, this is important because in actual gear design, center distance a_w is often predetermined and can't be changed.

II. Line with the equation $\alpha_{a1} = \alpha_{a2}$ (line B in Fig. 3)

Here α_{a1} and α_{a2} are tooth profile angles of a pinion and a gearwheel, respectively, at the points of tooth addendum circumference. As research showed, this line has two following interconnected properties:

1.) for a given center distance a_w = constant, the maximum transverse contact ratio ε_α is achieved at the point (x_1, x_2) of intersection of the given line with line A ($x_1 + x_2 = x_\Sigma$) compared with other points (x_1, x_2) of line A. Since the coefficient ε_α is known to be one of the main parameters defining smoothness of gear operation, the line under consideration may be called "the line of maximum smoothness";
2.) for a given transverse contact ratio ε_α, the maximum possible value of the shifts sum coefficient x_Σ (and therefore the maximum values of a_w and α_{tw}) is achieved along line B, at the point of its intersection with the line ε_α (line 1 in Fig. 3). As the value x_Σ increases, the values of curvature radii for the pinion and gearwheel tooth involute profiles also increase, leading to a decrease of contact stresses and, therefore, an increase of gear contact strength. This opportunity, however, can be practically implemented only

when the center distance a_w is variable within a wide range, which happens very seldom. That is why this line is alternately called "the line of maximum contact strength" or, at least, "the line of *increased* contact strength." The line may also be called "the line of minimum contact stresses."

III. Line of equal specific sliding (line C in Fig. 3)

This line has the equation (Ref. 5)

$$u\left(\frac{\tan\alpha_{tw}}{\tan\alpha_{a2}}-1\right)=\frac{\tan\alpha_{tw}}{\tan\alpha_{a1}}-1 \qquad (3)$$

where $$u=\frac{z_2}{z_1}$$

is the gear ratio. If the shift coefficients x_1 and x_2 are chosen so the point (x_1, x_2) could belong to this line (or, at least, to its vicinity), the gear operating conditions will be favorable for reducing the risk of scuffing and abrasive wear.

Moreover, it is emphasized in Reference 1 that such choice of x_1, x_2 provides more beneficial values of tooth shape coefficient Y_F from a bending strength point of view.

Therefore, it is possible to choose the values of x_1, x_2 at the initial stage of gear design by means of a BC and to choose lines A, B and C described above to solve the following tasks:

- the given value of center distance a_w (choice of the position of line A in the plane x_1, x_2 is provided);
- maximum smoothness of gear operation (point of intersection of lines A and B) is achieved for a given a_w, or minimum risk of scuffing and abrasive wear is achieved by equalizing of specific sliding (point of intersection of lines A and C);
- minimum contact stress is achieved for the chosen overlap ratio ε_α at the point of intersection of line B with line 1 corresponding to the given value ε_α. The user immediately obtains the displayed values of x_Σ, a_w and ε_α corresponding to the chosen point (x_1, x_2) where contact stress is noted to be minimum.

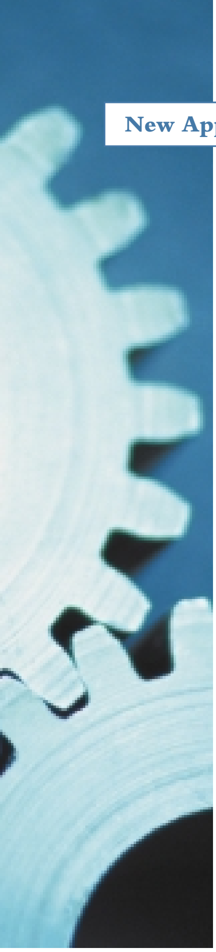
Thus, the concept of gear property evaluation and selection of the values x_1, x_2 at the initial design stage is provided by means of blocking contours with the lines A, B and C described above. Later in the gear design stage, the corresponding properties can be proved by calculations. Here the designer avoids performing time-consuming iterative procedures when choosing optimal shift coefficients of a pinion and gearwheel.

Example

Let's consider the blocking contours of an external spur gear (Fig. 3) presented to the user of a "Contour" CAD system based on the described concept. A spur gear with the following initial parameters is considered: z_1 = 22, z_2 = 35, m = 2 mm; α = 20° and h_a* = 1. (This data is presented in a small panel "Gearset parameters" near the figure's upper right corner.) Center distance a_w is assumed to be predetermined a_w = 58.5 mm. The area limited by the blocking contours is shaded in the figure. Line A corresponds to the initial center distance a_w = 58.5 mm. The shift coefficients at point *a*, where lines A and B intersect, are equal to x_1 = 0.103, x_2 = 0.718 and x_Σ = 0.821. The maximum allowable transverse contact ratio ε_α = 1.425 for the given a_w is achieved at this point. (This ratio is shown in the panel "Calculated parameters" in the figure's lower right corner.)

If the point (x_1, x_2) is displaced along line A to the right and downwards to position b—that is, to the intersection with line C—then x_1 = 0.445 and x_2 = 0.376. For these coefficients, the advantage of the line of equal specific sliding will be obtained (see above). It is clear that choosing the position of the point (x_1, x_2) on line A between points a and b corresponds to some compromise between maximum transverse contact ratio and equal specific sliding. This choice is also interesting in practice.

If the gear design allows the center distance a_w to be increased from a geometrical point of view, the increase corresponds to the displacement of line A to line A', where it is tangent to the line ε_α = 1.2. This line (the curved line 1 in Fig. 3) limits the blocking contour from above and from the right. Note here that the minimum recommended value of the transverse contact ratio for spur gears according to Russian standards for their geometry calculation is ε_α = 1.2. Maximum values of x_Σ, a_w and α_{tw} are reached here within the blocking contour. Reference 5 shows that the contact of line A (a_w = constant) and ε_α = constant (line 1 in Fig. 3) always takes place at the intersection of lines 1 and B independent of the value ε_α. This is the point *c*, where the shift coefficients are x_1 = 0.497 and x_2 = 1.232, and the values of shifts sum coefficient, center distance and pressure angle are x_Σ = 1.729, a_w = 59.960 mm and α_{tw} = 26°42'33", respectively.



The abovementioned "Contour" CAD system also provides some additional possibilities for visualization of:

- tooth profiles of meshing pinion and gearwheel in any desired scale with many auxiliary elements, such as a line of action with marked out sections of single- and double-pair engagement, active sections of tooth profiles, arcs of relevant circumferences;
- gears in operation, i.e. rotating gearwheel and pinion, with all explaining geometric engagement attributes available;
- process of tooth cutting by means of a rack tool. Change of tool parameters and visualization of consequent changing geometry of a tooth being formed is available.

The mentioned additional capabilities, which are not always necessary for gear design in practice, may be useful for a designer and especially for educational purposes due to their possibilities of visualization.

Prospects for further development of the described approach are associated with implementation of some more auxiliary lines into the DBC structure. These lines include equal bending strength of a pinion and gearwheel, as well as a number of others. Prospects are also associated with application of the developed methodology of DBC construction for other types of gears: internal cylindrical, straight bevel, Novikov and others.

Conclusion

This paper gives a new approach to computer-aided design of spur and helical gears, based on implementation of dynamic blocking contours for choosing pinion and gearwheel shift coefficients in order to easily obtain some predetermined properties of a gear without preliminary calculations and complex optimizing procedures. The developed approach allows prediction of gear properties at the first stage of design, at the stage of initial data assignment.

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Tooth Flank Corrections of Wide Face Width Helical Gears that Account for Shaft Deflections

Shan Chang, Dr. Donald R. Houser, Jonny Harianto

Management Summary

The tooth flank correction of power transmission helical gears with wide face width is studied using a finite element based shaft deflection analysis program in conjunction with a numerical load distribution analysis procedure. The load distributions along the line of action, the elastic deflections and transmission errors of gear pairs are obtained by solving the equations of compatibility of displacement and equilibrium of forces.

This paper discusses the influence of tooth flank corrections (tip relief, root relief, load modification, end relief and their combinations) on gear stresses and transmission errors due to shaft deflections. The technique used in the paper has the capability of modeling all significant geometric and elastic contributions due to tooth contact of the pair being analyzed as well as other gears mounted on the same shafts. The results show that it is possible to optimize at the design stage the gear micro-geometry for minimum stresses and transmission errrors without changing the gear macro-geometry.

Nomenclature

W_{1k}	total elastic deformation of point k on the first body
W_{2k}	total elastic deformation of point k on the second body
e_k	initial separation between the first and second body at point k
$R\theta$	rigid body approach
F_k	force acting on point k
r_b	distance of point k from the center line or base radius
T	applied torque
[S]	N x N matrix of influence coefficients or compliances
{F}	N x 1 vector of forces
{i}	N x 1 vector of ones
[I]	N x N identity matrix
{Y}	N x 1 vector of slack variables
{e}	N x 1 vector of initial separations

Introduction

Since 1938, when Walker (Ref. 1) first pointed out the importance of tooth flank corrections for spur gears, much work has been done to improve the load-carrying capacity and reduce the noise of power transmission gears. Terauchi (Ref. 2) obtained the tooth modifications (tip relief) based on the deflection results found using the combined approach of 3-D elastic theory and a mapping function. The tooth proportions that mesh on two teeth pairs can be corrected according to loaded deformation. Lee and Lin (Refs. 3 & 4) studied the influence of tooth flank modification and loading conditions on the dynamic tooth load and stress for high contact ratio spur gears. Maruyama (Ref. 5) achieved reduced transmission errors for automobile gears by optimum tooth crowning. Sigg (Ref. 6) provided a set of rules for profile and lead modifications required to provide smooth load distribution to correct for shaft bending and torsional deflections. Conry and Seireg (Ref. 7) provided an optimum design procedure for tooth flank corrections of helical gears. They solved non-linear contact equations of loaded gear pairs by a mathematical programming technique and set load distribution along the contact lines as the objective function. Other researchers (Refs. 8–9) also investigated optimum tooth modifications for spur and helical gears in considering the combination of various kinds of tooth modification types and parameters.

The key problem for gear tooth flank modification is how to get the precise deflections, including the loaded tooth elastic deformations and shaft deflections, and how to get load distribution along the contact lines. This paper's authors have citations to about 200 additional papers on the load distribution of gears, with some of them (Refs. 10–18) giving approaches to evaluate tooth load distribution of wide face width spur and helical gears.

Although it is recognized that shaft deflections and the loading of another gear on the same shaft have significant influences on tooth load distribution, few papers have been published on this topic.

Gopinath (Ref. 19) has developed the finite element-based shaft deflection analysis procedure used in this paper. His method was extended by Merugu (Ref. 20) to predict natural frequencies and mode shapes as well as dynamic response due to transmission error or external excitations of complex geared systems having several shafts. This shaft analysis procedure is used as a preprocessor to obtain influence coefficients and deflections due to outside loadings. These coefficients and deflections are then used in a load distribution solving routine similar to that developed by Conry and Seireg (Ref. 7). This routine predicts load distribution, stresses and transmission error of a gear pair.

This paper demonstrates these procedures for optimizing tooth flank corrections to reduce transmission error, root stresses and contact stresses of wide face width helical gears. The importance of considering shaft deflections and the deflections due to secondary gear loading will be demonstrated. The results in this paper show that very significant improvements can be achieved by careful tooth flank corrections to minimize transmission error and stresses under load.

Calculation Procedure

The load distribution is assumed to be a function of elasticity of the gear system and errors or tooth flank corrections on the gear pair. Below is a list of factors that the load distribution procedure considers in its calculation.

Elastic deformation:

- Bending and shear deflections of contacting teeth
- Base rotation and base translation of contacting teeth
- Local contact deflection
- Bending deflection of gear bodies and supporting shafts
- Flexibility of bearings and housings
- Torsional deflections of gear bodies
- Buttressing effects at tooth ends

Errors or modifications (initial separations):

- Shaft misalignment and shaft runout
- Involute errors or modifications
- Lead errors or modifications
- Tooth spacing errors

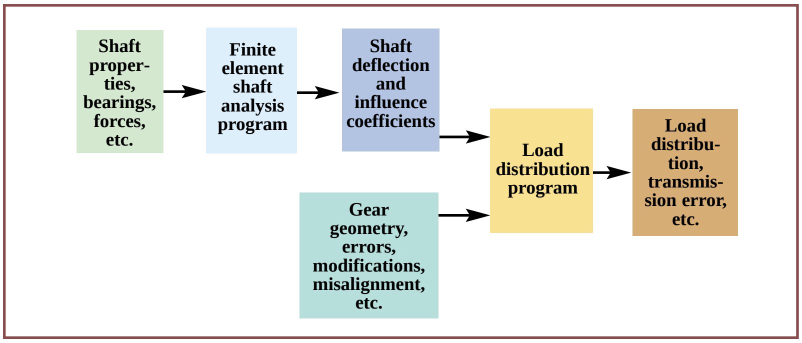

Figure 1—The relationship of LDP and SHAFT module.

The analysis procedure is incorporated into a program called the Load Distribution Program (LDP), which can compute the bending and torsion deflections of the gear bodies and supporting shafts, assuming the shafts are simple cylinders supported by two bearings. For more complex shafts, the shaft analysis module (SHAFT) uses a finite element procedure for the calculation of influence coefficients and misalignment. In the SHAFT module, shafts are modeled as beam elements with six degrees of freedom at each node and gears are modeled in a manner similar to shafts with diameter equal to pitch circle diameter. Bearings are modeled using a 6x6 bearing stiffness matrix. Matrix values may be obtained from bearing manufacturers or may be computed using a procedure developed by Lim and Singh (Ref. 21).

The effect of a pinion-gear pair is modeled as a set of forces and displacements for computing the displacements across the face width of another gear mounted on the same shaft. Only one shaft is analyzed at a time. The schematic of the SHAFT module and its interaction with LDP is shown in Figure 1. Although not shown in this paper, the SHAFT program also may perform a forced vibration analysis of multi-shafted transmissions such that natural frequencies, mode shapes and dynamic motions and forces are predicted for the entire system.

Solution of Compatibility and Equilibrium

The relatively complex problem of determining the load distribution between mating gear teeth and the elastic deflection of gear pairs can be solved by setting up compatible equations for displacement and equilibrium of forces for a sufficient number of discrete points representing the contact region along the contact lines. Load distribution is obtained with the method that is based on the work of Conry and Seireg (Ref. 7) for elastic bodies in contact. A simplex type of algorithm is

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Tooth Flank Corrections of Wide Face Width Helical Gears that Account for Shaft Deflections

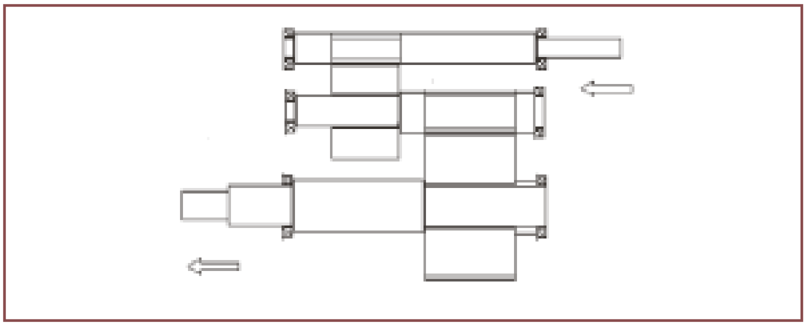

Figure 2—Shaft arrangement of typical two-stage gearbox.

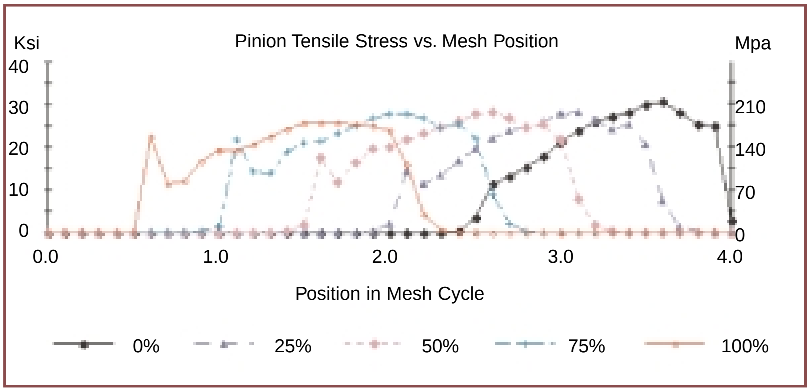

Figure 3—Pinion root stress without considering shaft deflections.

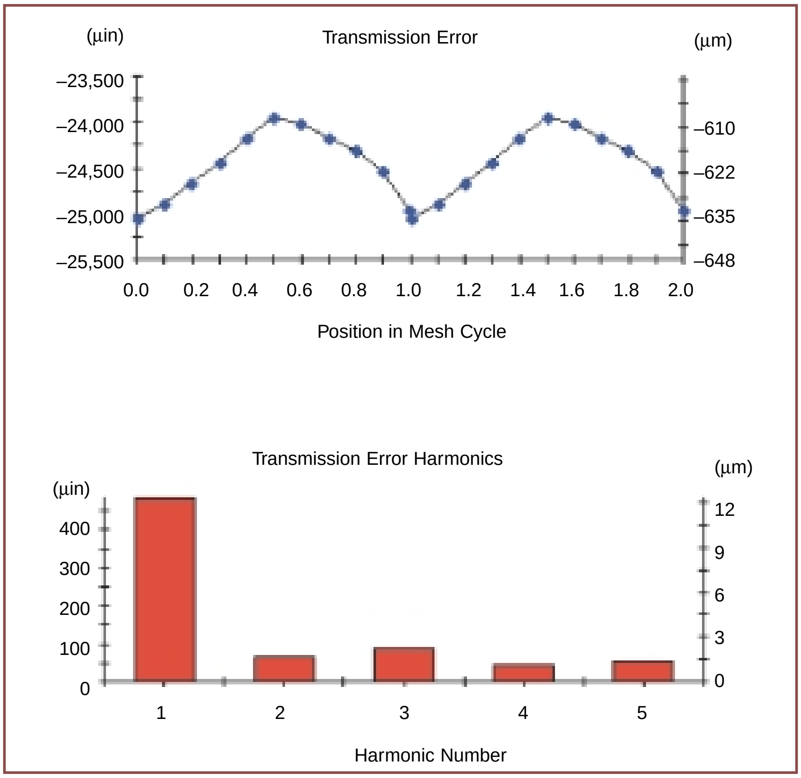

Figure 4—Transmission error without tooth flank modification.

used to compute the load distribution and the rigid body rotation based on the compliance of each point in the contact zone, the applied load, and the initial separations under no load, including errors or tooth flank modifications. The criteria used for the formulation of the solution are:

1.) Compatibility—This specifies the condition under which two points may come into contact.

2.) Equilibrium—This states that the sum of the torques acting on the system is zero.

According to the first criterion, the contributions due to the initial separations and the elastic deformation must exceed the rigid body motion along the line of action, for any discrete point *k* in the contact zone. Mathematically, this means:

$$W_{1k} + W_{2k} + e_k \geq R\theta \qquad (1)$$

The second criterion may be expressed as:

$$\sum_k (F_k \bullet r_b) + T = 0 \qquad (2)$$

In Equation 1, the inequality may be converted into an equation using a slack variable Y(k) and rewritten as:

$$W_{1k} + W_{2k} + \mathrm{e}_k - R\theta - \mathrm{Y}_k = 0 \qquad (3)$$

For contact between the bodies, Y(k) = 0 and consequently F(k) ≥ 0. If, however, Y(k) > 0, then the two bodies are not in contact at the discrete point *k* and F(k) = 0. Introducing the compliance coefficient matrix, the problem involves the estimation of the values of F(k), Y(k) and θ using Equations 4 and 5.

$$-[\mathrm{S}][\mathrm{F}] + R\theta[\mathrm{I}] + [\mathrm{I}][\mathrm{Y}] - \{e\} = 0 \qquad (4)$$

$$\{\mathrm{I}\}^T\{\mathrm{F}\}r_b + T = 0 \qquad (5)$$

Equation 4 expresses the conditions for compatibility of displacement at N discrete points and represents N equations in N+1 unknowns. The N+1th equation is provided by the equilibrium condition expressed by Equation 5. Equations 4 and 5 are solved using a modified simplex algorithm (Ref. 7). To obtain the tooth load distribution along the contact lines (Refs. 22–23), there are two places where shaft information is used. First, the LDP procedure needs the shaft deflections (due to the effects of another gear mounted on the same shaft, if any) as one of the components of the initial separation. Secondly, the shaft compliance is added to the compliances for tooth

Table 1—Gear Details.	First Stage		Second Stage	
	Pinion	Gear	Pinion	Gear
Number of teeth	15	63	17	69
Normal module, mm	4.5		6	
Normal pressure angle, deg.	23		23	
Helix angle, deg.	11		9	
Center distance, mm	182.88		266.7	
Outside diameter, mm	82.3	301.5	120.9	436.4
Root diameter, mm	60.45	279.6	92.176	407.4
Face width, mm	152.4	152.4	241.3	241.3
Torque Nm	3,114.4		13,080.5	
Bearing radial stiffness, x10^9 N/m	1.28, 0.86	1.7		1.75, 2.27
Bearing thrust stiffness, x10^9 N/m	0.16, 0.16	0.14		0.28, 0.28
Bearing span, mm	530	485		500
Shaft diameter, mm	82	105		150
Distance from face width center to bearing span center, mm	127	127	90	90

bending and shear, base motion, and Hertzian contact. In this case, the shaft influence coefficients supplied by the SHAFT program include the effects of bearing and housing deflections, as well as shaft torsion and bending. The analytical contact analysis can then be carried out for a specified micro-geometry at a number of positions of mesh to determine load distribution, transmission error, contact stresses and root stresses at given torques and misalignments. The specified micro-geometry can be tip relief, root relief, profile crowning, end relief, face crowning, lead modification, bias modification or topographic correction of the flanks.

Analysis Results

As an example, the load distribution procedure using the SHAFT module is used to analyze a typical two-stage gear reduction gearbox. The pinion of the second stage is mounted on the same shaft with the first stage gear, as shown in Figure 2. The gear details are given in Table 1.

Figure 3 shows the predicted root stresses (done with a hybrid finite element approach) at five positions along the face width of the pinion. Stress traces look quite similar, but the peak stress seems to be higher for the right side of the tooth pair.

Influence of Shaft Deflections on Tooth Contact

To emphasize the importance of making accurate assessments of the misalignment caused by the gear mounted on the same shaft in the wide face width helical gear pair, elastic mesh analysis

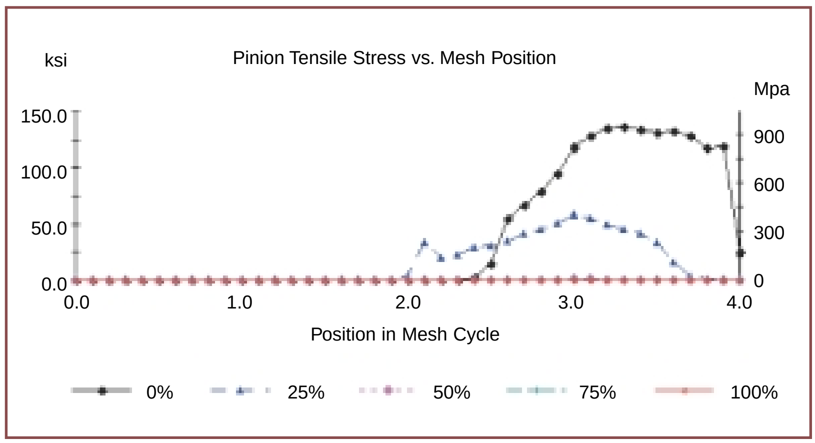

Figure 5—Effect of second-stage pinion deflections on the pinion root stresses along the face width (without tooth modification).

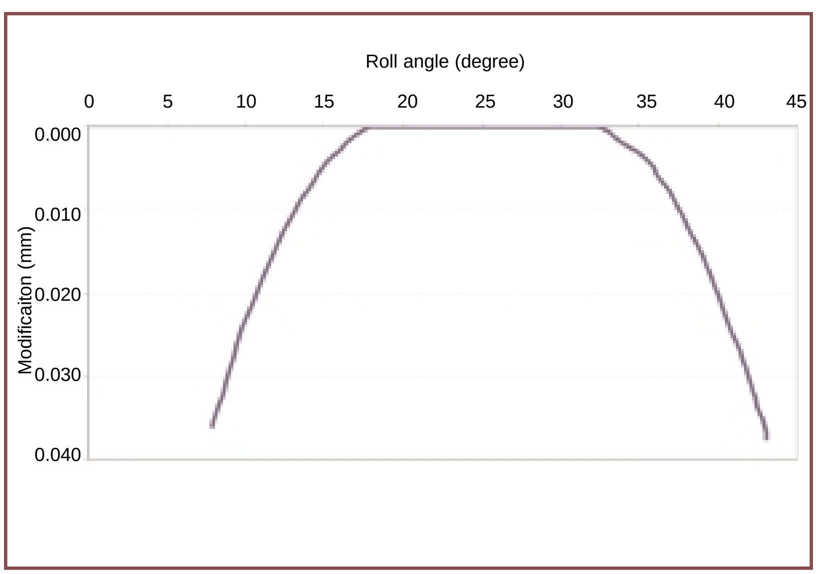

Figure 6—Profile modification for the first stage pinion.

Tooth Flank Corrections of Wide Face Width Helical Gears that Account for Shaft Deflections

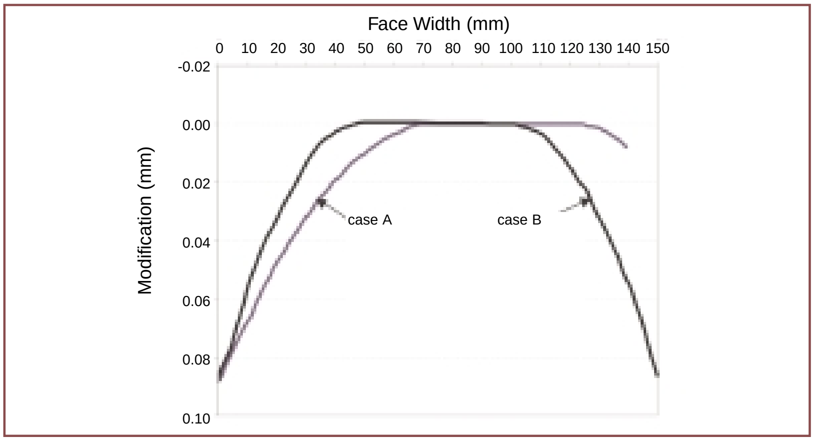

Figure 7—Lead modification for cases A and B.

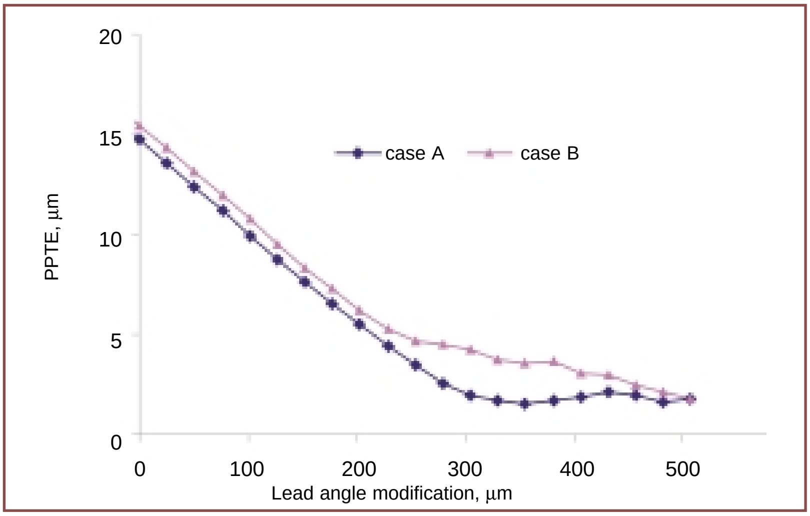

Figure 8—Effect of lead angle modification on peak-to-peak transmission error for two cases of lead crowning.

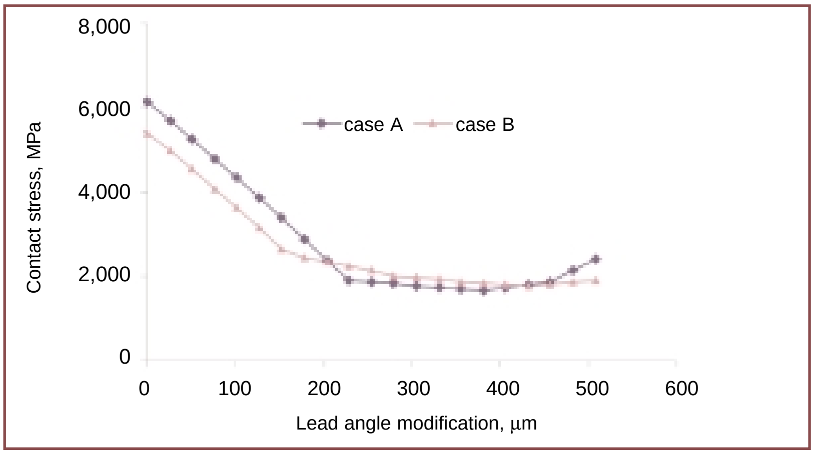

Figure 9—Effect of lead angle modification on peak contact stress for two cases of lead crowning.

has been carried out for the input gear pair. The misalignment along this gear pair's face width can result in an uneven load distribution across the tooth face width.

Using the data in Table 1 to calculate transmission error, load distribution, contact stresses and root stresses of the input gear pair without any tooth flank correction, it is noted that:

- Without tooth flank correction, contact stress is shifted to one side of the tooth and corner contact peak exists.
- Only about 48% of gear face width carries load.
- The maximum contact stress increases 3.6 times and the maximum pinion root stress increases 4.5 times compared with that of the stresses prior to considering shaft deflection due to loading of the second stage pinion (Fig. 5).
- Although not shown, the peak-to-peak transmission error increased significantly due to the misalignment.

Tooth Flank Corrections

The tooth flank corrections are carried out only on the first stage pinion. Tip relief and tooth relief are used to reduce the high contact stresses that occur at the tooth corners (the entering and exiting regions). Tip relief is 38 µm and the starting modification point is at a roll angle 31.6°. Root relief is 38 µm and its starting modification point is at a roll angle of 19°. The profile modification curve is parabolic, as shown in Figure 6.

To discuss the validity of end relief (lead crown) and lead angle modification on wide face width gears, calculated examples are given in this paper. End relief and lead angle modifications are considered according to the various conditions shown in Table 2. The values used in cases A and C–E are based on a procedure recommended by Sigg (Ref. 6). The two types of end relief are shown in Figure 7. The asymmetry of the Sigg form is meant to compensate for the large torsional effect of the small diameter pinion of the first gear pair. All of the calculations used in the simulations in cases A–E include tooth tip relief and root relief.

The peak-to-peak transmission error (PPTE), contact stresses, pinion root stresses and load distribution factor are shown in Figures 8–11 respectively, for cases in which base-line lead crowning was included and the lead slope was varied in order to compensate for shaft deflections. In these cases, all shaft deflections are considered. It is interesting to compare the performance of gear pairs that have different end reliefs. Figures 8–11 clearly show that both cases A and B achieve improved results. The asymmetric end relief can obtain slightly lower values of PPTE and pinion root stress, while contact stress and load distribution factor are about the same for the two types of end relief.

Based on the results in Figures 8–11, it is observed that lead angle modifications between 220 µm and 435 µm provide the best values of PPTE, stresses and load distribution factor. Three lead angle modifications are chosen as listed in Table 2 to optimize the end relief. The results are shown in Figures 12–14.

Figure 12 shows that both lead angle modification and end relief are important for obtaining the lowest PPTE for wide face helical gears. Comparing Figure 13 with Figure 15, the variation of contact stress with tooth flank correction parameters is very similar to that of load distribution factor. Figure 14 shows that when lead modification is much less, it is difficult to lower the root stress by using end relief. From the point of view of improving the load carrying capacity of wide-faced helical gears, lead angle modification is dominant. However, without the combination of tip relief, root relief and end relief, it is very difficult to obtain the favorable load distribution and lower contact stress as shown in Figures 12–14.

When end relief at the torque input end is 51 µm, lead angle modification is 432 µm, and PPTE gets the lowest value of 0.73 µm. On the other hand, the lowest load distribution factor of 1.15 is achieved with different tooth flank correction parameters, that is, with a 63.5 µm end relief at the torque input end and a 355 µm lead angle modification. With the tooth flank corrections that give the lowest PPTE, the TE and pinion root stress distributions of the wide face width gear pair are calculated and shown in Figures 15–16, respectively.

Comparing Figures 3–4 with Figures 15–16, it is observed that:

- The peak-to-peak transmission error is reduced significantly from 28.5 µm to 0.73 µm. The PPTE of this gear pair is very close to that of a modified set in which shaft deflections are not considered.
- Load carrying area increases up to 99% of face width.
- Without tooth flank correction, the contact stress at the corner region is more than 8,300 MPa, also the maximum contact stress posi-

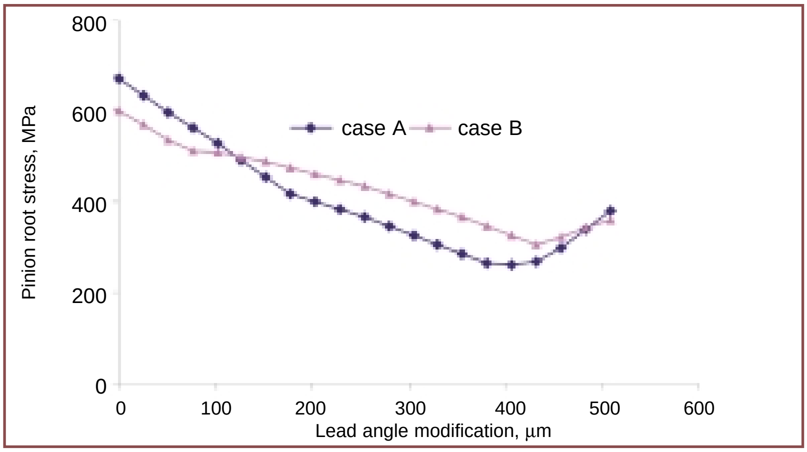

Figure 10—Effect of lead angle modification on pinion root stress for the two cases of lead crowning.

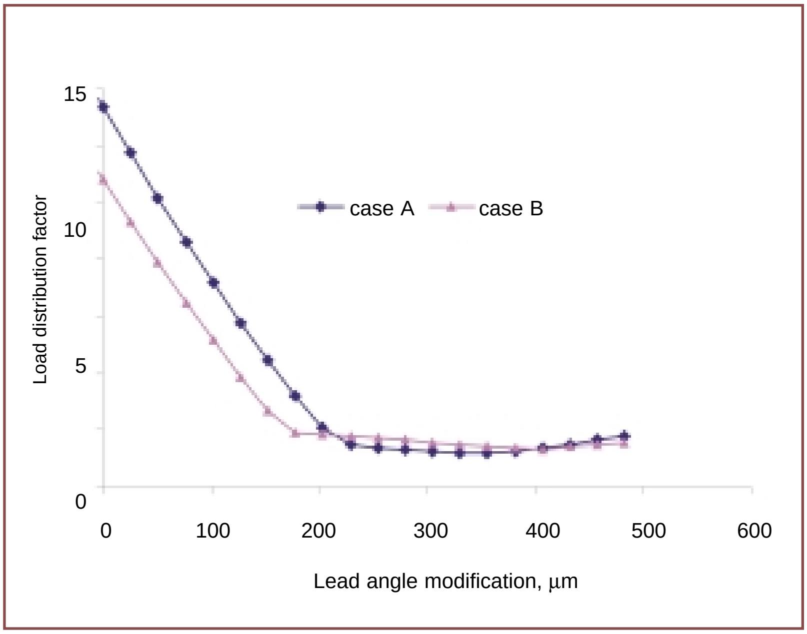

Figure 11—Effect of lead angle modification on load distribution factor for two cases of lead crowning.

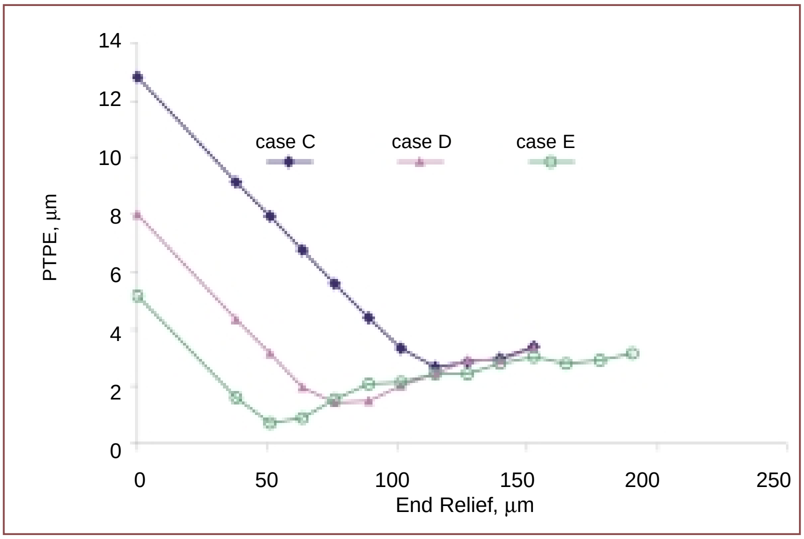

Figure 12—Effect of amplitude of end relief (Sigg's Method) on peak-to-peak transmission error for three different lead angle corrections.

Tooth Flank Corrections of Wide Face Width Helical Gears that Account for Shaft Deflections

Table 2—End Relief and Lead Angle Modification.

	End Relief	Lead Angle Modification
Case A	Relief: 90 µm at torque input end, 25 µm at free end Length: 76.2 mm at torque input end, 25.4 mm at free end Curve: Parabolic curve	Variable
Case B	Relief: 90 µm at torque input end, 90 µm at free end, Length: 51 mm at torque input end, 51 mm at free end Curve: Parabolic curve	Variable
Case C	Relief: Variable The relief at torque input end is three times that of free end. Length: 76.2 mm at torque input end, 25.4 mm at free end Curve: Parabolic curve	228 µm
Case D	Relief: Variable The relief at torque input end is three times that of free end. Length: 76.2 mm at torque input end, 25.4 mm at free end Curve: Parabolic curve	355 µm
Case E	Relief: Variable The relief at torque input end is three times that of free end Length: 76.2 mm at torque input end, 25.4 mm at free end Curve: Parabolic curve	432µm

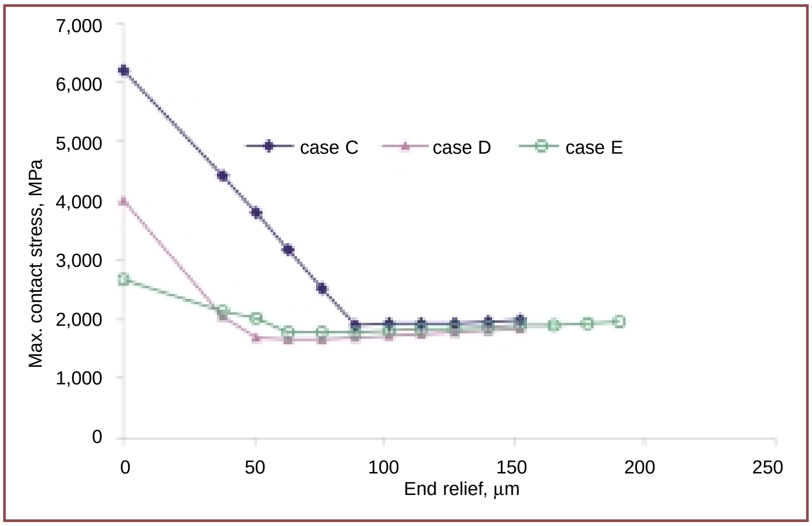

Figure 13—Effect of amplitude of end relief (Sigg's Method) on peak contact stress for three different lead angle corrections.

tion. After tooth flank modification, the contact stress at this position reduces to 490 MPa, and the contact point that has the maximum contact stress (2,018 MPa) shifts to the center region in the tooth face. A bit larger lead correction could further reduce this maximum stress and relocate it closer to the center of the tooth.

- Maximum root stress reduces greatly from 961 MPa to 313 MPa, and the root stress distribution along the face width improves. Again, a further shifting of the lead correction could further reduce the peak stress value.

Conclusions

The use of an elastic mesh model that is coupled with a detailed finite element model of the supporting shafting has been shown to be a useful tool for improving the geometric design of gearing. From the results of the above examples, the following conclusions are made:

1.) Shaft deflections and misalignments caused by another gear mounted on the same shaft have a significant effect on load carrying capacity and performance for wide face gear pairs.

2.) The optimum tooth flank correction for helical gears with wide face widths can achieve a good balance between contact stress, root stress, peak-to-peak transmission error and load distribution factor.

The technique described can be applied to spur and helical gears with wide face widths and to gear systems with complex shaft arrangements. The method is capable of computing the influence of another gear mounted on the same shaft. Properly applied, it can achieve significant reduction in transmission error and root bending stress and contact stress by optimizing the gear micro-geometry at the design stage without changing the gear macro-geometry.

Acknowledgments

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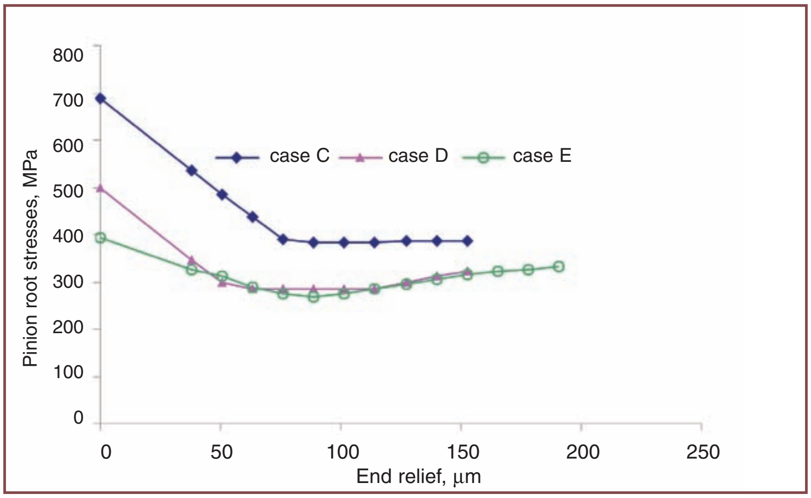


Figure 14—Effect of amplitude of end relief (Sigg's Method) on peak pinion root stress for three different lead angle corrections.

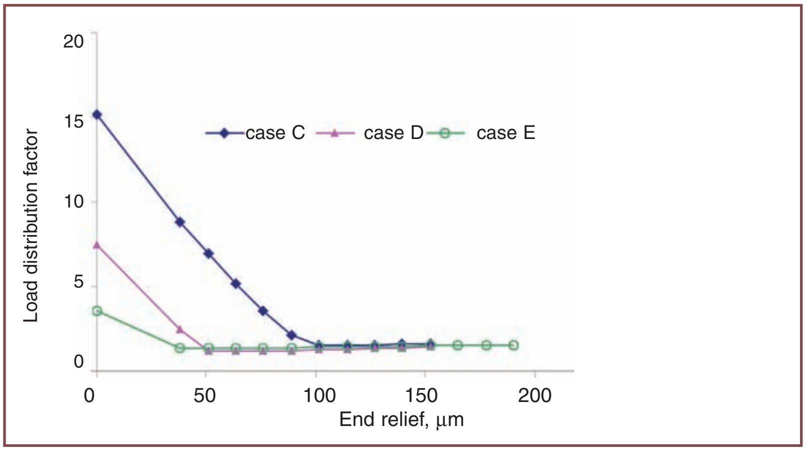


Figure 15—Effect of amplitude of end relief (Sigg's Method) on load distribution factor for three different lead angle corrections.

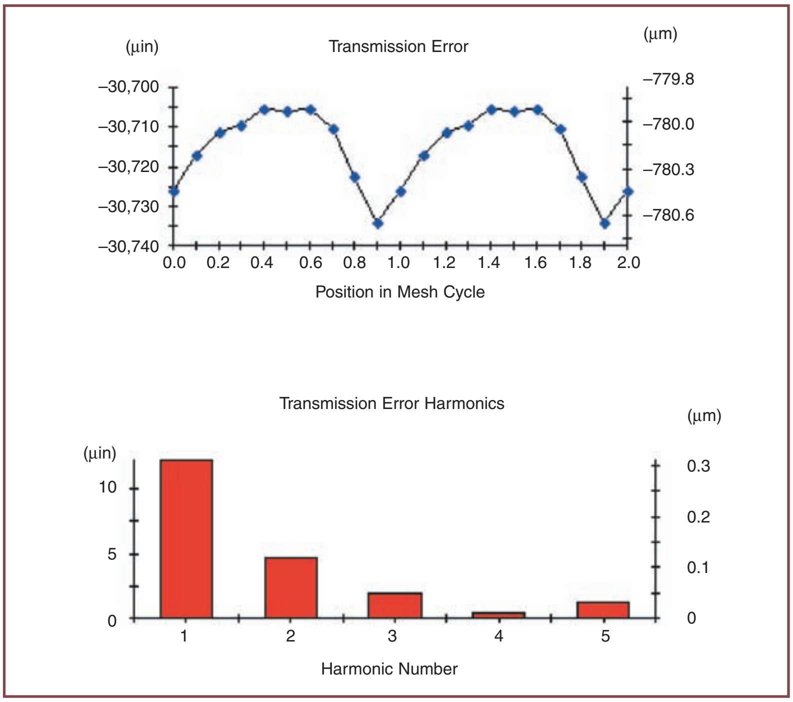
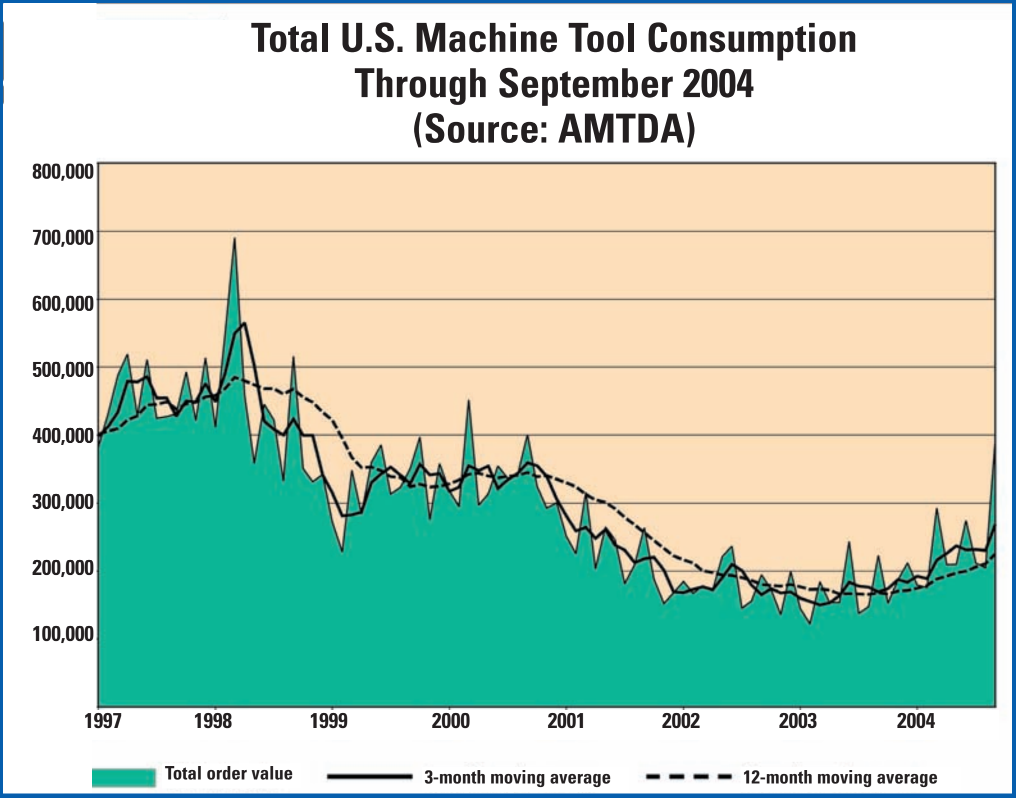


Figure 16—Transmission error after tooth flank correction.

Machine Tool Consumption at the Highest Level Since 2000



"Manufacturing in the U.S. continues to show signs of a strong comeback."
–Ralph J. Nappi, AMTDA president.

The American Machine Tool Distributor's Association (AMTDA) released a report stating that September's machine tool consumption was up 89.8% from August.

According to the association's press release, this year-to-date total of $2,146.95 million is up 42.2% from 2003's total.

"Manufacturing in the U.S. continues to show signs of a strong comeback," says Ralph J. Nappi, in an AMTDA press release. "Machine tool orders in September hit their highest levels in four years with all the regions in the country showing growth. The re-election of President Bush will further solidify the comfort of manufacturing with continued capital investments."

This data comes from a joint report by the AMTDA and the Association for Manufacturing Technology (AMT).

When broken down by region, the Midwest showed the greatest improvement and surpassed August consumption by 157%.

Gary Schiffres, membership director for the AMTDA, credited the upswing to a number of factors.

"The combination of IMTS, pent-up buying, and a need for new equipment account for this news. Also, it's the end of the year and many companies have to get machinery off the floor by Dec. 31 for tax incentive purposes," he says.

As far as the outlook for 2005, Schiffres predicts an initial drop, followed by an upswing.

"Personally, I think there will be a dip in the first quarter because of all the activity at the end of 2004. However, big turnkey products and systems will go regardless, and I think we'll see an increase of about 15.3% for 2005," he says.

Ikona and Magna Advanced Technologies Partner for Gear Tooth Profile

Ikona Gear International and Magna Advanced Technologies have jointly introduced a patented gear technology that utilizes a newly designed, patented tooth profile for use in planetary systems.

Together, Dr. John R. Colbourne, author of *Geometry of Involute Gears*, and Laith Nosh, president of Ikona Gear, developed a non-involute system that can be made smaller and lighter than conventional planetary gears. In applications with ratios from 8:1 to 5,000:1, the Ikona gear has fewer moving parts and the highest single-stage reduction ratio with zero backlash, says Nosh.

"Ikona's design is a technological breakthrough—it's a revolution," says Colbourne.

"Pinion teeth have convex profiles that are derived mathematically from a path of contact shaped as in the Archimedean Spiral. The internal gear teeth are conjugate to the pinion teeth. The Ikona gear pair has high contact ratio, no tip interference and minimal backlash," Colbourne says.

Magna Advanced Technologies is working to develop the commercial applications for the Ikona gear. Since April 2003, Magna has been conducting extensive prototyping and testing in its Canadian and Austrian facilities. Magna pinpointed five automotive markets most suitable for the Ikona gear: steering systems, braking systems, closure systems, seating systems, and microactuators.

Servo gearing seemed to be a fit for the Ikona gear system because, according to its developers, this profile is currently the only form that is completely conjugate with many teeth in contact, sharing the load. Other types of servo gearing employ the orbital gear method to achieve their ratios and stiffness, but often struggle with load shar-

ing and conjugating issues, Nosh says.

Aside from servo gearing, the Ikona gear is suitable for many industries. Its most recent utilization is in conjunction with AirCast Inc., which is marketing the Mayo Clinic Armbrace with the plastic Ikona Gear in the elbow joint.

"The Ikona Gear was the only technology that could enable the arm brace to be mechanically operated to provide patients a self-exercise option in post-operative care situations," says Barrie Freeke, chief engineer at Ikona. "The elbow gear provides a three-fold torque increase over the original AirCast."

The use of plastic gearing can lead to a reduction in noise, vibration and harshness in the final product, and the load sharing characteristic of the Ikona system enables it to carry higher loads, making plastic a viable option in applications that were previously thought impossible.

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Internet: *www.ikonagear.com*

Star SU Reaches Sales Agreement with Hainbuch Welge

Star SU LLC and Hainbuch Welge Corp. of Milwaukee, WI, have reached an agreement for the sales and distribution of Hainbuch workholding tools that are used in gear manufacturing operations.

According to a joint press release, Star SU will market, sell and distribute Hainbuch Welge's chucking, feed finger systems, and expanding mandrels worldwide. The Hainbuch product brochures are now downloadable as PDF files from the Star SU website.

Hainbuch Welge is an 80% holding of Hainbuch GmbH of Marbach, Germany.

Honda Plans Expansion of Transmission Facilities in Ohio, Georgia

Honda announced a $100 million plan for construction of a plant in Tallapoosa, GA, and a plan for the expansion of its existing Ohio plant that will also total $100 million.

According to the company's press release, the Ohio expansion will create 100 new jobs to expand Honda Transmission Mfg. of America and will result in the transfer of production of high precision gears from Japan to Ohio. As the new Georgia plant begins operations, Honda Transmission Mfg. will begin shifting transmission production responsibility there from the Alabama plant. The Georgia plant, located 40 miles west of Atlanta, will also be a hub for gear production.

Honda currently has an annual production capacity of 1 million automatic transmissions in the U.S.

Genstar Acquires Colfax Power Transmission Group

Colfax Power Transmission Group was recently acquired by Genstar Capital L.P., a private equity firm. In a related move, Colfax PT has merged with Kilian Manufacturing Co., another Genstar portfolio company, to form an entity called Altra Industrial Motion Inc.

According to Genstar's press release, the combined purchase price for the two companies was approximately $200 million before taxes. Michael Hurt, formerly president of TB Woods, has been appointed as CEO of Altra Industrial Motion.

Existing management personnel of both companies will take leadership positions.

Colfax Power Transmission Group is headquartered in Quincy, MA, and encompasses Boston Gear, Warner Electric Industrial Products, Nuttall Gear and Delroyd Worm Gear, Wichita Clutch, Industrial Clutch, Marland Clutch, Formsprag, Stieber, Warner Electric Europe and Ameridrives Couplings.

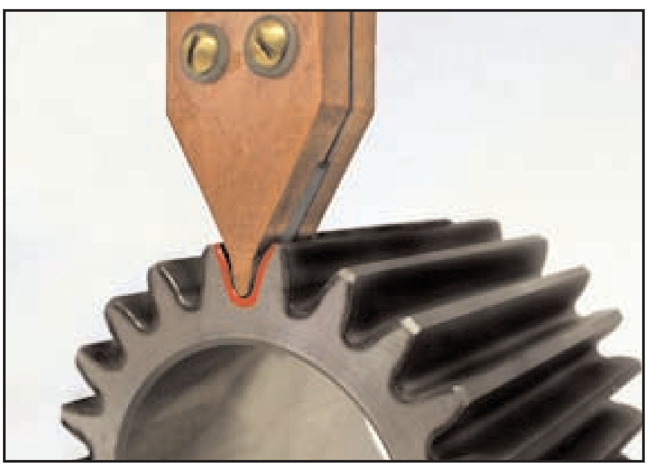
mG miniGears Officially Opens in China

mG miniGears in Suzhuo, China, is now operating at full capacity.

According to the company's press release, the Chinese operation is solely devoted to mass supplying of powder metal and machined steel parts to power tool customers.

By early 2005, the company plans to increase production volume and begin to assemble products for lawn and garden equipment.

New Gear Hardening Capability for Penna Flame



Penna Flame Industries of Zelienople, PA, a surface hardening specialist, now has the capability for induction hardening and single-tooth contour gear hardening.

Boston Gear Wins Military Contract

Boston Gear's facility in Charlotte, NC, is under contract to supply gearboxes for the U.S. military operations in Iraq and throughout the Middle East.

The RA 1022 gearbox line will be mounted on Humvees deployed in Iraq and in other locations to help the war effort. According to the company's press release, Boston Gear was awarded the contract by O'Gara-Hess & Eisenhardt of Fairfield, OH, which was hired to equip 350+ Humvees a month for the war.

Jim Miller, operations manager for Boston Gear, is in charge of the gearbox project.

"We're excited and proud to do our part to improve the safety of U.S. military personnel serving overseas," he says.

Manufacturing Association Elects New Board of Directors

The Association for Manufacturing Technology has its elected new officers at its 2004 conference in Bonita Springs, FL.

David J. Burns of Gleason Corp. was elected chairman, Douglas K. Woods will be the new vice chairman, Bradley L. Lawton of Star Cutter was elected second vice chairman and treasurer. J. Patrick Erwin was elected secretary.


David J. Burns


Bradley L. Lawton

In addition, Doug Currie was elected to a three-year term as a board member, and Richard Bodine Jr., Robert T. Smith and Lawton were all re-elected.

Rossi Motoriduttori and Habasit Form Partnership

Rossi Motoriduttori, a manufacturer of geared motors and gear drives located in Modena, Italy, has formed a cooperative partnership with Habasit, a Reinach, Switzerland-based belting company.

According to the company's press release, the board of directors will include various members of the Greco, Habegger and Volpi families.

David de Palma will be general manager of the North American facility in West Chicago.

Philadelphia Gear Hires Canadian Rep

Gaetan Perron was hired as the sales representative for established Philadelphia Gear customers in Canada and for prospects in eastern Ontario, Quebec and the Maritimes.

Perron's specific responsibilities include selling in gas-fired and hydroelectric power plants, the pipeline industry, petrochemical sector, hydrocarbon processing plants and wind power generation industries in areas from Cobourg, Ontario, through St. John's, Newfoundland.

Most recently, Perron worked as a sales representative for Omega Machine Shop, developing customers in western Quebec and eastern Ontario. He has also held field rep. positions with PYG Equipment and Castle Metals.

Vermont Machine Tools Achieves Retrofit Certification

Vermont Machine Tools announced its certification in GE Fanuc's Five Star Machine Tool Retrofit program.

The certification guarantees a retrofit provider's technical and project management capabilities, matches retrofit projects with qualified best-in-class providers and helps to ensure that machine tool users can maximize the productivity of existing equipment retrofits.

Vermont Machine Tool delivers CNC retrofits and remanufactured machine tools.

Niagara Gear Adds Gleason TWG to its Line

Niagara Gear has purchased the Gleason 245TWG high speed threaded wheel grinder.

According to its press release, the company hopes this new machine will reduce process-related deviations and subsequent heat treat distortion with a hard finishing solution.

The 245TWG enables hard finishing using metal removal rates. It can also improve tooth profiles by eliminating undulated form error by utilizing the latest drive technology.

In addition, this equipment incorporates form grinding methods and components into a water-cooled base. Thermal compensation via sensors throughout the process facilitates accuracy and parts consistency.

Niagara Gear also recently acquired the Reishauer RZ300E precision gear grinder, which can grind gear segments, gears with intermittent teeth and pump gears with few teeth.

New Facility, Markets for Kleiss Gears

Kleiss Gears has traded in its 3,000-square-foot facility in Centerville, MN, for a 14,000-square-foot building in Grantsburg, WI.

According to president Rod Kleiss, the plan is to add both staff and new capabilities for gear molding.

The company plans to develop more automated production capabilities as well. Currently, the company designs and prototypes gears with the aim of eventually getting into gear production work.

"We're also playing with some pretty remarkable materials," Kleiss says. "We're using PEEK (polyetherether ketone), which can replace lighter duty metal gears. You can never really replace metal when it's fully recognized, but you can modify gear shapes. We're continuing to evolve in the field of plastic gearing, too."

Hamilton Sundstrand Awarded Contract for Trent 1000 Engine

The Rolls-Royce Co. selected Hamilton Sundstrand to design and manufacture the gearbox system for the Trent 1000 engine that is being developed by The Boeing Co.

Hamilton Sundstrand's press release estimated that, over the lifetime of the engine program, the work is potentially worth more than $350 million to the company.

The environmental control system, electric power generation and start system, remote power distribution system, auxiliary power unit, primary power distribution system, nitrogen generation system, emergency power and electric pump subsystems are all among the components Hamilton Sundstrand is supplying.

This is the second time in 2004 that Hamilton Sundstrand and Boeing will work together on 7E7-related projects. The first time, Boeing selected Hamilton Sundstrand for eight aircraft systems on the 7E7 Dreamliner.

Promotions at American Axle & Manufacturing

American Axle & Manufacturing of Detroit, MI, has made numerous appointments at its executive level.

Yogendra N. Rahangdale

Yogendra N. Rahangdale was named chairman of the operating committee. According to the company's press release, he will supervise vice presidents of the driveline and metal products division, the program management and launch divisions, quality assurance and customer satisfaction initatives, materials management and logistics, as well as the directors of industrial engineering and productivity, and capital and capacity planning.

John Sofia

Thomas O. Delanoy was appointed to the position of vice president of quality assurance and management. Previously, he was the company's executive director of material and production control.

John Sofia was named vice president of quality assurance and customer satisfaction.

Charles B. Comeau was appointed general manager of U.S. operations at the driveline division.

Other promotions include: Kenneth Koncilja to the position of executive director of internal audits, Gerald Costner to production control and materials management, Michael Dorah to director of global supply base management, Dr. Robert Karban Jr. to director of advanced quality planning, Donald L. Joseph as plant manager of Detroit Gear & Axle, and Lawrence Oliver as plant manager at Three Rivers Driveline.

Getrag Announces Plans for Slovakian Plant

The Getrag corporate group affiliates Getrag Getriebe- und Zahnradfabrik Hermann Hagenmeyer GmbH & Cie KG, headquartered in Untergruppenbach, Germany, and Getrag Ford Transmissions GmbH of Cologne, Germany, will establish a joint manufacturing site in the Kosice region in eastern Slovakia.

Getrag Ford will build a transmission plant at this site. Getrag GmbH & Cie KG will relocate its motorbike facilities to this location.

According to the company's press release, overall investment will total $300 million and employ 750 people.

GE Energy Announces $1.3 Billion in Wind Turbine Projects

GE Energy announced that it has secured contracts and order projections for wind turbine projects in 2004–2005 that are valued at more than $1.3 billion.

In the company's press release, GE Energy's wind energy division CEO Steve Zwolinski says, "The federal government's recent extension of the renewable energy production tax credit has, again, provided a tremendous impetus for the wind industry in the U.S."

The tax credit provides $0.018 per kilowatt hour credit for electricity produced during the first 20 years of a wind project's operation and will remain in place through December 2005.

GE's new contracts include projects throughout the U.S. that utilize the company's 1.5-megawatt wind turbines.

In addition to the U.S. projects, GE Energy announced in October its contract to supply 660 wind turbines for eight projects in Quebec.

Regal Beloit Acquires GE Motor and Capacitator Operations

Regal Beloit Corp. announced a definitive agreement with GE Consumer and Industrial for its HVAC/refrigeration motor and capacitor operations.

According to the company's press release, the $379 million transaction includes a full line of motors for HVAC refrigeration operations and high intensity lighting operations.

HVAC operations are currently based in Fort Wayne, IN, and are included in the sale as are motor manufacturing operations in Springfield, MI, Reynosa, Mexico, and Faridibad, India; a capacitor manufacturing facility in Juaraxc, Mexico; and a call center in India.

As a result of this transaction, Regal Beloit expects an additional $520 million in sales.

INDUSTRY NEWS

Headquartered in Beloit, WI, Regal Beloit manufactures mechanical and electrical motion control and power generation products.

Danfoss CEO Wins National Leadership Awards


Jørgen M. Clausen

Jørgen M. Clausen was named "Leader of the Year" by the Danish Association of Managers and Executives at Denmark's largest management conference.

The presentation was attended by more than 800 private and public leaders throughout Denmark.

In October, Clausen was awarded the Order of Merit for the German Federal Republic by the ambassador of the German Federal Republic in Denmark.

Mahr Partners with OKM Jena

Mahr Goettingen has recently acquired 56% of Optische Koordinatemesstechnik Jena (OKM).

According to Mahr's press release, this partnership will increase their market share for 1-D length measurement machines by 25%. Both companies manufacture the 1-D length measurement machine. New capacities will include horizontal length measurements and optical measurement of tools.

OKM Jena is a manufacturer of the ULM optical coordinate measurement machine, which can supplement Mahr's multisensor program.

Holger Hage, Petra Bogdanski and Karl Joachim-Bode will jointly manage the new operation.

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Boeing Metallurgist Calls for Vacuum Carburizing Standard

Gear manufacturers and heat treaters: If you can vacuum carburize, you may be missing out on business opportunities because you lack an industry specification for this heat treatment. There are companies that design gears suitable for vacuum carburizing, but they may not use the process on their parts because they don't have a way to assure reliable quality.

An industry specification would provide a way. It would establish procedures for proving companies' vacuum carburizing processes are capable of carburizing gears uniformly, so customers could rely on the quality of those gears. Such confidence would promote wider use of the heat treatment.

This is the situation that was suggested Nov. 4 by The Boeing Co.'s Dale Weires in his keynote address, "Potential Applications of Low Pressure Vacuum Carburizing for the Rotorcraft Industry." Weires, a Boeing associate technical fellow for metallurgy, was speaking to suppliers and users of vacuum carburizing equipment at "Practical Uses and Applications of Vacuum Carburizing," held in Chicago, IL. The conference included more than 70 attendees.

In his speech, Weires described the

"We need to look at some standardization of our methods," said Dale Weires, a metallurgist for The Boeing Co., about vacuum carburizing.

opportunity presented by Boeing's Chinook helicopter. The CH-47D is the primary product, the biggest volume product, in Weires' division, Boeing Philadelphia. The Chinook drive system includes 70+ major parts that are atmosphere carburized, but are suitable for vacuum carburizing. The parts include 10 bevel gears and 19 spur gears.

"Gears are an integral part of this drive system," Weires said.

Moreover, Boeing knows it could benefit from vacuum carburizing. The aerospace company evaluated the process in the '90s, Weires said, and learned it could reduce cycle times for heat treating the Chinook drive parts.

The times weren't sufficient reason to start vacuum carburizing the parts. Boeing could rely on the quality of its atmosphere carburized parts, Weires later said, and switching to vacuum carburizing wouldn't have reduced the company's costs.

Weires' example pointed up a main problem that vacuum carburizing faces: It can be difficult to displace a successful, standardized process when the alternative is not similarly standardized. In the past, the lack of an industry specification has often restricted the use of vacuum carburizing to captive or specialized heat treat shops, Weires said, so:

- Each shop's furnaces were unique and depended on their heat treaters' knowledge of vacuum carburizing.
- The companies viewed their process knowledge as proprietary, so they limited their sharing of it.
- The number of knowledgeable vacuum carburizers was very limited.
- The availability of vacuum carburizing was limited.

Given these circumstances, it's easy to imagine that companies would view consistent quality as a hit-or-miss proposition. Weires, however, offered a way to spread the use of vacuum carburizing: "We need to look at some standardization in our methods."

In his opinion, standardization could be accomplished through a new industry specification. Weires acknowledged that AMS standard 2759/7 addressed carburizing, but it didn't distinguish between its two types: atmosphere and vacuum (low pressure).

"I see nothing out there that I can push a gear manufacturer to use," he said.

According to Weires, the specification would need to include:

- a method for qualifying vacuum carburizing furnaces,
- definitions for the elements of a process "recipe,"
- definitions of process changes and significant repairs, and
- more frequent quality control checks.

The specification, Weires said, would give companies an important ability. They would be able to tell suppliers of their drive parts: Use this specification.

To create and publish a specification, though, companies would have to share what they know about vacuum carburizing. Sharing could include publication of information in trade journals, Weires said. The information would have to document vacuum carburizing's benefits in process cost and quality and its reduced maintenance costs, he added.

Weires also remarked that publishing information would help vacuum carburizing become more widely accepted and used: "Published data spurs interest."

The conference was sponsored by Heat Treating Society, an ASM International affiliate, and was held at the Illinois Institute of Technology, home to the Thermal Processing Technology Center, a consortium that includes nine member companies, such as Dana Corp.

Other speakers were:

- Frederick Otto, president of Midwest Thermal Vac;
- Dan McCurdy, director of North American business development for Bodycote Thermal Processing;
- Dennis Beauchesne, general manager of ECM-USA Inc.;
- William R. Jones, CEO of Solar Atmospheres Inc.;
- Gerald D. Lindell, corporate engineering metals specialist with Twin Disc Inc.;
- Steve Ellison, president of North American Cronite Inc.;
- Tony Wu, president of ALD Thermal Treatment Inc.;
- Thomas Wingens, vacuum product manager of Ipsen International Inc.;
- Janusz Kowalewski, HPQ group leader of SECO/Warwick Corp.; and
- Ralph P. Poor, director of standard heat treat products with Surface Combustion Inc.

TECHNICAL CALENDAR

February 2–3—Smart Solutions for Metal Cutting Conference.
Dorint Sofitel Quellenhof, Aachen, Germany. Held in both German and English, this conference will focus on the most recent developments in powder metallurgy. International manufacturers of HSS/HSS-PM and cutting tools will provide their views on the latest technology. 495 euros includes conference documents, snacks, lunches and an evening festivity. For more information, contact WZL-RWTH Aachen by e-mail at *k.marso@wzl.rwth-aachen.de* or on the Internet at *www.aachen-tourist.de*.

February 7–10—Gear School 2005.
Gleason Cutting Tools facility, Rockford, IL. This fundamentals course covers cutting, gear inspection, gear geometry, nomenclature and inspection. $895 includes handbook and materials, lunches and a group dinner. For more information, contact Gleason Cutting Tools at (815) 877-8900 or on the Internet at *www.gleason.com*.

February 22–24—Expo Manufactura.
Cintermaz Expo Center, Monterrey, Mexico. Mexico's largest machine tool and metalworking exposition. Sponsored by AMT, registration is free. For more information, visit the show's website at *www.expomanufactura.com.mx*.

March 7–10—National Manufacturing Week.
McCormick Place, Chicago, IL. Broken into ten segments: Aluminum, CleanTech, Design Engineering, Enterprise IT, EnviroTech, Industrial Automation, Legal Issues, Management & Manufacturing, Micro Systems, and Plant Engineering and Facilities Management. Registration before Feb. 4 ranges from $75–$595, depending on the level of conference participation. Registration after Feb. 4 is listed from $95–$695. For more information, visit *www.nationalmanufacturingweek.com*.

EVENTS

March 14–16—AGMA Gear School for Gear Manufacturing Technology.
Star SU LLC, Hoffman Estates, IL. Presented by the Gear Consulting Group, this three-day course concentrates on a logical approach to troubleshooting the gear manufacturing process, concentrating on hobbing, shaping and inspection procedures and the relationships between the process, inspection results and the underlying gear geometry. Suitable for all grades of personnel from machine operators to engineering and management staff. For more information, contact the Gear Consulting Group by mail at P.O. Box 647, Richland, MI, 49083, by telephone at (269) 623-4993 or by e-mail at *gearconsulting@aol.com.*

March 17–19—Schleifring Grinding School.
ExpoCenter, Thun, Switzerland. Technology presentations involving 25 grinding machines will be offered as well as lectures that will be translated into four languages. For more information, contact United Grinding Technologies by telephone at (937) 859-1975.

SAVE THE DATE

September 14–16—3rd International Conference on Gears. Munich, Germany. Co-sponsored by the AGMA, ASSIOT, CMES, FVA, JSME, KIVI-NIRIA, UNITRAM and VDMA.

CALL FOR PAPERS

2005 AGMA Fall Technical Meeting— Abstracts due Jan. 7 at AGMA headquarters on the following topics—Noise/vibrations; heat treatment (including distortion control); tribology; load distribution; calculations; lubrication; dynamics; effect of finish, grinding or profile modification on surface durability; applications; micropitting or failure analysis case studies. Send a page-long abstract to Amy Lane by e-mail at *lane@agma.org*.

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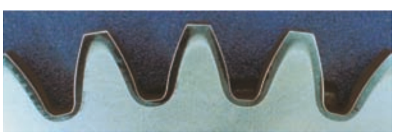

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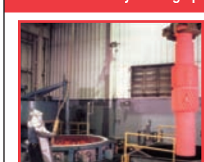

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RUNNING GEAR

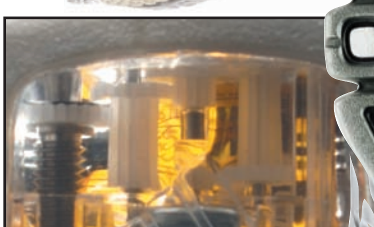
The Addendum team has seen the future of running, and it's geared. adidas-Salomon AG has come out with a running shoe that has gears in it, a whole train of them.

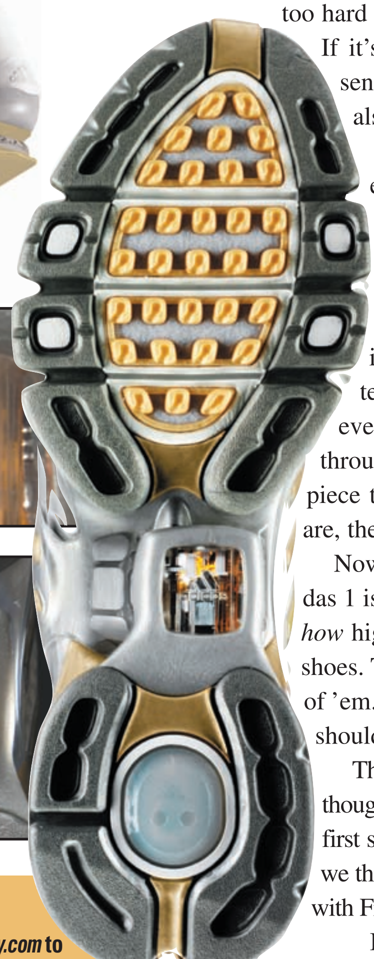
Unfortunately, they aren't gears that you shift into. If they were, the Addendum team would be ecstatic. Imagine: lacing up your sneakers, putting them in third gear, and going for a good, long jog—just carried along by your mechanized feet.

Geared though they are, adidas 1s aren't Fiats for your feet. But these sneaks are geeked, and the Addendum team found out how from adidas.

The geeking starts in the heel of an adidas 1. Each heel has a specially designed plastic cushion with a sensor on the top and a magnet on the bottom. During each footfall, the cushion is compressed, and the sensor measures the distance between itself and the magnet based on changes in magnetic flux density. So far, the electrical engineering: fairly cool.

The geeking continues in the mid-sole. The distance measures are sent to a tiny computer. It figures whether cushioning is too hard or too soft. If it's too hard, the heel cushion needs softening. If it's too soft, it needs hardening. In either case, the computer sends a signal to the shoe's motor. This computer engineering: also fairly cool.





Things get cooler, though, when we get to the mechanical engineering. Also in the mid-sole is a lead screw connected to a cable that runs through the cushion. If the screw is extended, the cushion is hardened. If the screw is contracted, the cushion becomes softer.

And, at last, there's the *really* cool mechanical engineering. Between the motor and lead screw, at the heart of the system, is a gear train of white, plastic spur gears. The gears are even on display. Turning over an adidas 1, we can see the spurs through a small window in the sole. It's like looking at a masterpiece through a rubberized picture frame. As cool as adidas shoes are, they're even cooler with gears in 'em.

Now, with all this machinery, no one should be surprised the adidas 1 is a high-end sneak. Still the Addendum team was surprised at *how* high-end. We've paid as much as $130 for a pair of basketball shoes. That was nothing. $130 would buy *an* adidas 1, but not a pair of 'em. These sneaks will retail for $250 a pair. Only serious runners should even try on these shoes.

The Addendum team is holding out for the ultimate running shoe, though. The shoe may not come for many years, but the adidas 1 is a first step in the right direction. It has gears and has to be turned on, so we think it's only a matter of time before the shoe company comes out with Fiats for the feet, fully mechanized running shoes.

Imagine: shoes that run for us, taking the effort out of exercise. We can hardly wait, sitting here on our couch. And once we get those shoes, we'll get doctors to get rid of the sweat thing. We're not keen on that, either.

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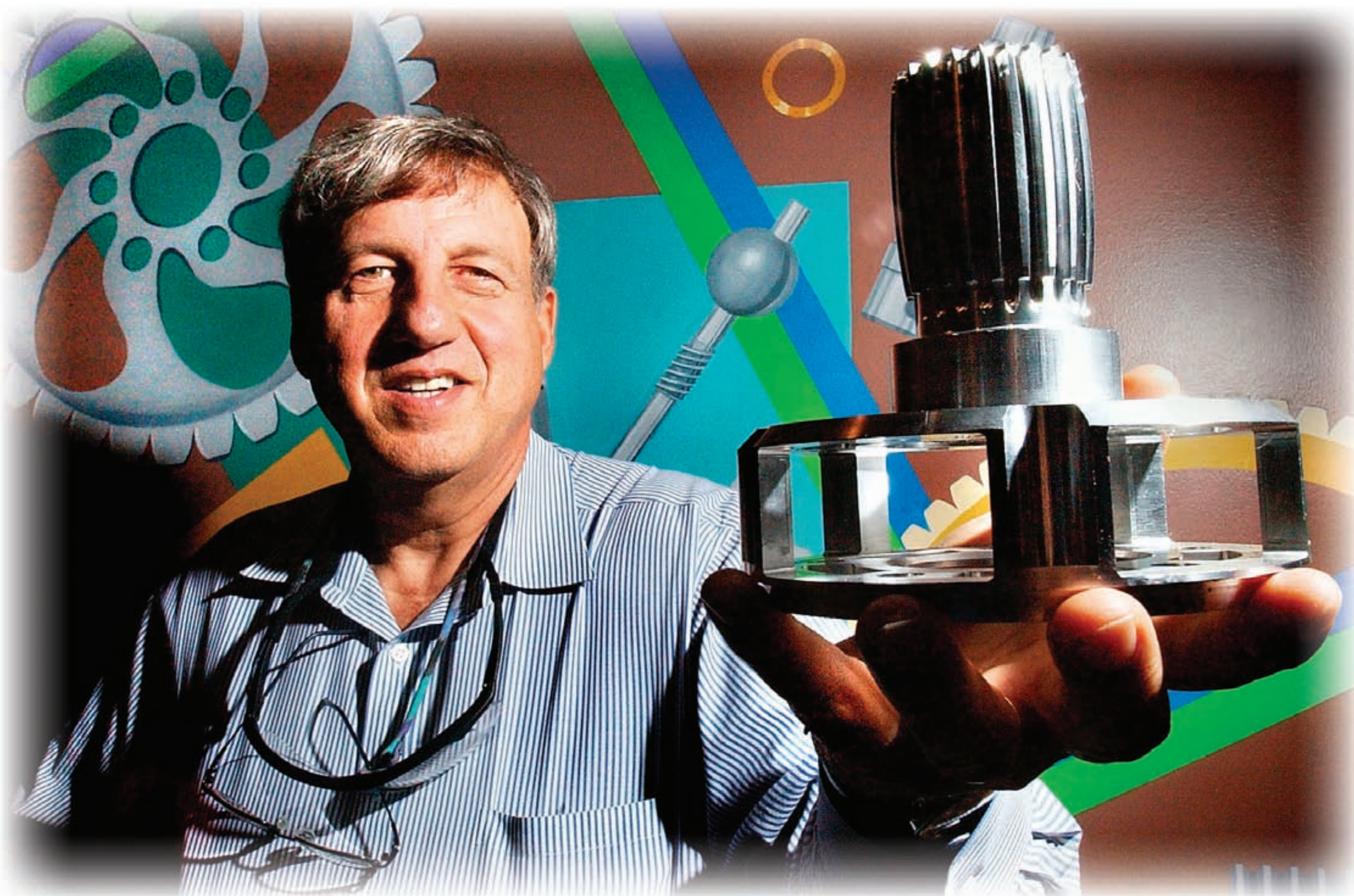

Gear Used in Mars Rover

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