GEAR TECHNOLOGY

March/April 2011

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The Journal of Gear Manufacturing

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- Induction Hardening Technology
- New Grinding, Finishing—and Profit—Advancements
- New Turbine Gearbox Standard—An Update

Technical Articles

- Face Gears
- Gear Measuring Machine by "NDG Method" for Gears Ranging from Miniature to Super-Large
- Optimal Modifications of Gear Tooth Surfaces

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GEARTECHNOLOGY

March/April 2011

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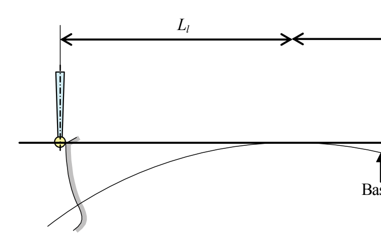

Figure 4—Amount of probe movement by TDG-M

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GEAR TECHNOLOGY

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BPA WORLDWIDE

REACHING OUT

At *Gear Technology*, we've long considered ourselves digital pioneers. We were among the first to launch our websites, beginning with *www.geartechnology.com* in 1996 and *www.powertransmission.com* in 1997. We were also early adopters of the electronic magazine format, launching *E-GT* in 2003.

But until now, we've held off from entering one particular digital frontier: e-mail newsletters. Although e-mail newsletters have been around for a long time, we've resisted the urge to launch any of our own—at least until we could get it right. We didn't want to inundate our readers with canned news items, articles only tangentially related to the gear industry, or rehashed information they've already seen. If we were going to publish newsletters, we wanted the issues to have a purpose, be focused and be valuable to our readers.

I'm pleased to say that with the launch of the *Gear Technology* and *Power Transmission Engineering* e-mail newsletters in January, I think we achieved those goals. And if our readers' reactions—as judged by the e-mail open rates and click-through rates—are any indication, many of you agree.

Our goal with the newsletters was to expand the depth and breadth of information about our industry that we're delivering to our readers. So one of the first things we did was line up a selection of topics—one for each month—that would be important to our readers. Many of these are topics we don't always have the opportunity or the space to cover as well as we'd like in the printed magazines.

For 2011, the *Gear Technology* topics are hobbing, shaving, shot peening, deburring, grinding, bevel gear manufacturing, services, heat treating, non-gear machinery, honing/lapping/polishing, gages and measuring tools, and used machinery.

For *PTE* the topics are motion controllers, custom gear manufacturing, linear motion, sensors, hydraulics and pneumatics, enclosed gear drives, maintenance tools, seals, gearmotors, lubrication, belt and chain drives, and servomotors.

With each issue of the e-newsletters, we aim to give you at least one exclusive article that you can't get anywhere else. For example, *Gear Technology*'s January e-newsletter included a feature article on gear hobbing, including coverage of the latest in equipment and technology from the major machine tool manufacturers. February's edition included an in-depth interview with executives at Sicmat, who discussed the latest innovations in gear shaving.

One of the benefits of a digital newsletter is the format makes it easy to embed links to videos or other dynamic offerings that just aren't possible with our print edition. For example, the January newsletter article on hobbing included links to videos of many of the machines and processes in action.

Each issue also includes links to articles in our archives that are related to the focus topic, and these are among the most popular links followed so far. It's gratifying to know that many of these articles are just as important and useful today as they were when they were first written. We've made an effort to choose the best articles and we've also included a number of "Back to Basics" articles—for which we've had numerous requests—because they are so useful for newer engineers, or those new to gearing.

The *Gear Technology* e-mail newsletter is being successfully delivered to about 10,000 recipients per issue. The *PTE* e-mail newsletter is being successfully delivered to about 8,500. Thank you to all of you who have helped us keep your e-mail addresses up to date.

If you haven't received one of these newsletters yet, you can read the previous issues online. Just visit *www.geartechnology.com/newsletter* or *www.powertransmission.com/newsletter*.

While you're at the site, why not also renew your subscription to *Gear Technology*'s printed version. It only takes a minute or two, and subscribing gives you access to our archive of online back issues. If you're not sure whether you've signed up recently or not, check the front cover of your magazine. If there's a subscription form attached, it's time to renew.

Also, if you think you have a story to tell, or if one of the upcoming focus topics is of interest to you, you can become either an editorial contributor or an advertiser. Just send an e-mail to *publisher@geartechnology.com*.

Finally, we'd love to know what you think about the e-newsletters. Judging by the traffic and activity we've seen so far, it's clear that the e-newsletters have been well received by many readers. But every opinion is important to us, so if there's some other kind of information you'd like us to add or change, please let us know. As always, the suggestion box is open at *publisher@geartechnology.com*.

Michael Goldstein,
Publisher & Editor-in-Chief

Standards Development: Enclosed Drives

Todd Praneis, Cotta Transmission Co.

Who doesn't like to see their name in print, or words that they've written be included in part of an international document? It's an amazing feeling to be part of a group that helps shape worldwide activities.

What am I talking about? Well, it's membership and activities in a committee in the AGMA Technical Division of course. I originally became interested in finding out what exactly happens in one of these committees, and now I am chairman on one committee and contributing to two others!

I currently hold the chairman position of the Enclosed Drives committee. We have responsibility for AGMA standards 6001-E08 "Design & Selection of Components for Enclosed Drives" and its metric counterpart 6101-E08; 6013-A06 "Standard for Industrial Enclosed Gear Drives" and its metric counterpart 6113-A06; as well as information sheet 14179-1 "Gear Reducers – Thermal Capacity Based on ISO/TR 14179-1." Our committee also provides a technical advisory group (TAG) response to ISO working group 10 activities. This basically provides the U.S. position on any proposed standards or documents that ISO is revising or creating.

AGMA 6001/6101 deals with design of the major non-geared, load-bearing components in a geared drive: shafting, keys, fasteners, bearings and housings. AGMA 6001/6101 went through a major edit for its "D" revision released in 1997. Since then, the committee was tasked to review the content and make improvements and modernization where possible. A lot of work was put into making the shaft deflection example more "programming friendly" and to continue the great work done for the D97 release. Another significant amount of time was spent on refining the key and fastener sections to align more with other international standards.

AGMA 6013/6113 is a "new" standard in that it combines and supersedes two other standards: AGMA 6009-A00 "Standard for Gearmotors, Shaft Mounted and Screw Conveyor Drives" and AGMA 6010-F97 "Standard for Spur, Helical, and Herringbone and Bevel Enclosed Drives." AGMA 6013/6113 covers topics dealing with both gear rating specifics from AGMA 2001-D04 and general enclosed gear drive component design and configurations such as service factors, preferred ratios, shaft diameters, standard shaft configurations, lubricant selection and others. This was an admirable effort by all involved to not only combine standards and resolve any inconsistencies, but to incorporate an entirely new lubrication section based on AGMA 9005-D94. During the creation of the new standard, the thermal rating section of AGMA 6010-F97 was removed and incorporated into the AGMA information sheet 14179-1. As with most standard revision, efforts were made to incorporate the latest information from other AGMA standards. This required an update to the rating formulas based on AGMA 2001-D04 and also added some updates to the stress cycle factor.

AGMA/ISO 14179-1 was an extract from AGMA 6010-F97 with an update based on the referenced ISO standard. We felt that the subject matter was deserving of its own information sheet and shouldn't be tucked away inside of another standard. At the time of creating the information sheet, we were able to incorporate the great work done by ISO and fellow AGMA member companies within the document. Information sheets are different from standards in that they may be focused on one aspect of an enclosed gear drive, or are on a narrow scope of application and therefore are not suitable for an entire standard.

So you can see that the enclosed drives committee is involved in a lot of different aspects of a geared system from gear design and rating to component design, lubrication and all the other bits and pieces that a designer must consider for the total geared package. We rely on the experience and involvement of everyone on the committee to contribute not only their time and company's expertise, but their willingness to help create what you read in the standard. If you've heard of the old adage that "you don't truly know a subject until you teach it," you can definitely apply that thinking to "…until you have to write a clause in a standard." There's no better way to understand a subject than to have to create the illustrative example. I've found it a great way to refresh and reaffirm my design practices, and it is a comfortable feeling knowing that we may help a gearbox purchaser to better communicate with his vendor, or to help guide a gearbox manufacturer in the proper design methodology for his product. Contributing on a standard also allows a company to voice its opinion on the subject, which will benefit not only that company, but the gearing community in general. Member companies can come to a consensus on the best practice for a particular subject, put it in writing, and publish the subject for all to use.

Committee 6b, the enclosed drives committee, is currently working on further enhancing 6001-E08 "Design & Selection of Components for Enclosed Drives" by expanding on recommendations for housing and other static or interface design considerations. Equally as important as rotating elements, the static components must be able to maintain gear position, carry shaft loading, be able to be assembled and allow a user access for various monitoring and service functions. We have also begun discussions on creating a new document that will cover geared units specifically for crane service. We are at the very beginning stages of this discussion and would welcome any input from the crane community on the viability and subject matter.

So what's it like to be on an AGMA Technical Division committee? In a word: great! I've met some really great people involved in the gearing industry, and am continually impressed with the dedication of the AGMA staff. No matter what your level of experience, there is always a home in a Technical Division committee. Find something that you know a lot about, something you know a little about or something you'd like to know more about and get involved! Who knows, you may end up being chairman…

Carl Zeiss CMM

GUIDES ANDREW TOOL WITH COMPLEX MARS ROVER PROJECT

Andrew Tool is utilizing the Zeiss ACCURA with VAST technology for the Mars rover Curiosity (courtesy of Carl Zeiss).

When Andrew Tool got the call to quote on a MSL (Mars Science Laboratory) project, they were determined to succeed by taking advantage of their expertise. At Andrew Tool, CMMs have been an integral part of their manufacturing processes for years, but they had never faced a project with such intricate measurements, tight tolerances, heat treatments and a very short time frame requirement. Carl Zeiss had proven to be a great resource at Superior Tool (Andrew's sister company).

"Our PRISMO CMM at Superior Tool is the backbone of our quality department, and my goal was to replicate and build on this capability at Andrew," says Bruce Hanson, president and CEO. This goal and initial project discussions with the MSL top-tier contractor pointed Andrew Tool toward a Zeiss CMM as the best option.

Andrew Tool is a unique machine shop with more than 30 years of experience handling five-axis milling, EDM, grinding and more. Their success is dependent upon fostering good relationships between the tool designer or engineer and the machinist.

"They need each other, and our employees understand that it's a team effort that links us to our customers," Hanson says. The majority of Andrew Tool's customers are found in the aerospace, defense, medical and micro electronic industries, all having parts that often require extremely tight tolerances.

This new, complex project from MSL required Andrew Tool to manufacture actuators (gearboxes) for the next Mars rover Curiosity. This new rover will weigh more than 10,000 lbs., five times the weight of the current rovers, and carry more than ten times the weight in scientific instruments compared to the current Spirit or Opportunity rovers.

Therefore, the propulsion system's power and torque will be more robust, and the unit's wheels considerably larger than previous designs. NASA engineers believe these changes will help prevent the problem that Spirit is encountering now: it's stuck in a sand pile. There were many parameters that Andrew Tool had to adhere to in order to help NASA make their new actuator design a success. Many of the parts had very deep pockets (almost a 20:1 ratio), and small radii added to the challenge, along with extremely tight tolerances, many of which are tied to different gear pitch diameters. The parts are very labor intensive with thousands of points of data measured on individual parts. The VascoMax material used for the actuator parts changes size slightly during heat treatment and, as a result, many of the part features were machined and inspected to process dimensions that allowed for this size change if the feature was not going to be final finished post heat treat. Additionally, position tolerances of .0002", geometric control of .00008" and size control within .0001", even on relatively large (5" range) dimensions added to the challenge. All of these factors, coupled with a demanding 18-month timeline and AS9100 certification requirements, made it critical that Andrew Tool bolster its CMM capabilities for precision and speed.

Andrew Tool decided to purchase the Zeiss ACCURA with the VAST XT gold active scanning sensor. The Zeiss ACCURA was an affordable solution with the range they needed while the VAST technology and automatic stylus rack system increased flexibility and productivity when determining size, form and position. The VAST XT gold is suitable for the complex and heavy stylus configurations required in measuring MSL's actuators.

Of course with any new machine being added to the production process, there's always a learning curve. The first surprise they had was seeing how seemingly benign parameter setting changes could dramatically affect measurement results. The discipline of always verifying critical measurements with another method was crucial to the learning curve. Brent Helgeson, metrology applications manager at Concept Machine, was very responsive in efficiently diagnosing the correlation problems, enabling Andrew to quickly get up to speed with their new CMM. Additionally, probing strategy for some of the features and proper alignment sequences to achieve correct results were developed and refined to achieve consistency.

"The Zeiss ACCURA helped us orchestrate the project by providing timely, accurate and understandable in-process reporting. It can't be overstated how critical good inspection is for process development and setup," says Bryant Broderick, quality control engineer at Andrew Tool. The project included 14 different part numbers with quantities ranging from four to 12 plus setup parts. Throughout all of the different processes including milling, gear cutting, heat treatment and stabilization, each part was successfully measured up to 60 times by the Zeiss CMM to ensure accuracy every step of the way. The temperature compensation feature was especially helpful because they were able to check parts right off the machine tool and relay the results to the toolmaker instead of having to wait for the part temperature to stabilize before measurement. With a coefficient of expansion of five and one half millionths of an inch per degree, a few degrees could mean the

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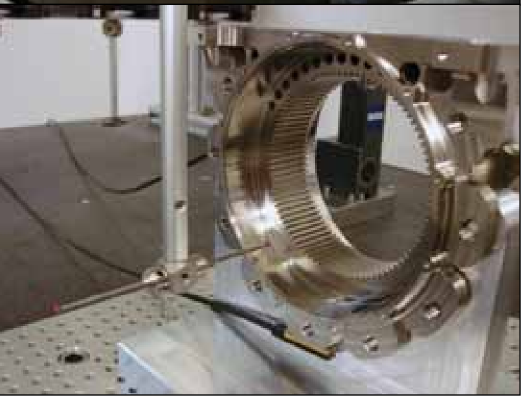

difference between pass or fail. "We were amazed by how effortless the ACCURA made this whole process."

The robust reporting capability in *CALYPSO* is a great asset for the AS9100 certification process required by NASA. This certification involved a lot of time on the program management side of things ensuring there was documentation for all of the critical paths of each part throughout the whole manufacturing process. All of this information was contained in a spreadsheet with all of the serialized parts to clarify where each part was at any point. The *CALYPSO* software easily documented the information electronically each time the part was measured.

"Verbal information of part details means nothing; documented data is everything," states Don Felix, director of sales and marketing. "We couldn't have been successful without Carl Zeiss. The benefits Zeiss brings have been enormous, and we see positive results every time we use the CMM." Ongoing training continues to drive Andrew toward the goal of using the ACCURA to its full capacity.

Andrew Tool even acquired a new customer after another NASA supplier saw several of the MSL actuator parts. They were so impressed with the precision and complexity of the parts that they were instantly sold on Andrew Tool's capabilities. This supplier also required AS9100 certification and fortunately, Andrew Tool was well on their way to attaining this certification and ready to take on this new opportunity with enhanced confidence in data from their Zeiss CMM.

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Carl Zeiss Industrial Metrology
6250 Sycamore Lane North
Maple Grove, MN 55369
Phone: (800) 327-9735
www.zeiss.com

Zoller and Ingersoll

PARTNER FOR MEASURING HOB CUTTERS

With growing markets in aerospace and energy technologies, measuring hob cutters used in gear cutting is becoming an essential requirement for workpieces and machine tools. Zoller, a provider of solutions for tool pre-setters, measuring and inspection machines and tool management software, has developed a new partnership with Ingersoll/Germany for shop floor checking of hob cutters by a combined hardware and software approach. "Ingersoll's experience enabled the Zoller software engineers to optimize the measuring process for hobs," says Robert Auer, export sales manager at Zoller. "Due to this partnership Zoller now offers a measuring program for hob cutters which ensures an accurate and fast measurement combined with unrivaled user friendliness."

Its base is a Zoller Venturion 600 or 800 with a CNC driven tilting optic carrier with *Pilot 3.0* software featuring photo-realistic user dialog. Programming boils down to nothing but entering the nominal parameters as indicated in the corresponding graphic. According to these nominal values, the measuring program is automatically generated by the *Pilot 3.0* software. Positioning and determining individual

continued

measuring points on the cutting edges is carried out automatically and—more importantly—independent from the operator. Even superimposed or distorted edge images as they occur with the helical pitch pose no problem as Zoller Venturion optic carriers are automatically and precisely rotated by numeric control if necessary.

Another advantage of this measuring approach is scanning the entire effective outline of all edges: This is of eminent importance for hob cutters with carbide inserts as the contour in the work piece is produced by an entire range of different inserts. Only by scanning the contours of individual inserts and merging them into a total contour can errors be avoided, as they occur in the transmission zone, insert tolerances, imprecise mounting or excessive tolerances concerning insert positions.

When combining a Zoller Venturion with hob cutters by Ingersoll, users get a hands-on system to test tools quickly and with micron precision before using them on the machining equipment. As opposed to most tactile-based measuring machines in this area, measuring results do not vary according to which operator happens to be on duty. Zoller systems are tailored to the demands as posed by hob cutters and offer additional value as they may be used for measuring and presetting all types of other tools as well: reamers, mills and milling heads. The result of this partnership between Ingersoll and Zoller shows how users can profit when industries join forces for their mutual benefit.

"In today's market it is important to not look just at individual components but at processes instead," Auer says. "The combined hardware and software approach ensures an integrated, transparent manufacturing process from tool manufacturing to quality control, machining, process control and regrinding."

For more information:

Ingersoll Cutting Tools
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Rockford, IL 661108
Phone: (815) 387-6600
www.ingersoll-imc.com

Zoller Inc.
3753 Plaza Drive
Ann Arbor, MI 48108
Phone: (734) 332-4851
www.zoller-usa.com

Honing System

PROVIDES CLOSED-LOOP CONTROL OF TOOL SIZE

Sunnen's SV-1005 series vertical CNC honing system with integrated air gaging system provides closed-loop control of tool size, along with downloadable SPC data. Matched with Sunnen's super abrasive MMT tool, the air-gage-equipped machine can automatically control hole size to accuracies of 0.25 µm (0.00001") without operator intervention, working in a bore size range of 3–65 mm (0.120–2.56") diameter. It is suitable for automated, high-Cpk production of small engines, hydraulic valves/bodies, fuel injectors, gears, compressor parts, turbocharger housings and gun barrels in medium and high volumes.

According to Sunnen, by combining the new air gaging system with the machine's patent-pending tool-feed control, the SV-1005 eliminates the need for an experienced honing operator to tweak the process. The new air gaging system controls bore diameter and geometry by taking post-process measurements of the parts while they are still fixtured on the machine's rotary table. Feedback from the air gage provides the highest possible accuracy for tool-feed control.

The SV-1005 matches an ultra-precise tool feed system with CNC control to allow any CNC-experienced machinist to master honing quickly. Setup is simple with a three-axis hand wheel for fine tuning the tool feed and the servo-controlled stroke system and rotary-table. The control includes several features that put honing expertise at the fingertips of novices, such as a switchable autocorrect feature for bore

continued

shape. Using measurements from the air-gage system, it allows the operator to select from a library of "problem" bore images (taper, barrel, etc.) to match to the part on the machine. The servo-controlled stroke system ensures a consistent crosshatch pattern and can dwell in any part of the bore, end-to-end, selectively removing stock for ultra-precise straightness and roundness.

The vertical design of the SV-1005 conserves shop floor space, requiring just 2,400 x 2,300 x 2,700 mm WxDxH (95 x 91 x 107 in.). A cast polymer base and cast-iron column provide enhance vibration damping for precision honing, while removable stainless steel side doors facilitate integration with automatic part loading systems. The high-torque, belt-driven spindle is rated at 7.5 kW (10 hp) to cover a wide range of sizing and finishing work.

For more information:
Sunnen Products Company
7910 Manchester Avenue
St. Louis, MO 63143
Phone: (314) 781-2100
www.sunnen.com

Forest City Gear

PURCHASES TAKISAWA LATHE

Forest City Gear has purchased a Takisawa TT-200G, a fully-automated turning center with twin-spindle, twin-turret and twin-CNC operation, for its in-house blanking department. By the acquisition of this machine, according to a company spokesman, the production in the blanking department has radically improved, as the machine combines full automation with twin-sided, simultaneous machining.

With a 16-pallet capacity, this Takisawa 8" chuck-type machine boasts a feed rate of 8 m/min and features a standard spindle and turret plus a second C-axis spindle and turret with milling function. In addition, a bar loader, workpiece stacker, turnover unit, chip conveyor, air blower, tabulating counter and other equipment are onboard for fully automatic mode operation of the machine.

As a strictly custom gearmaker, Forest City Gear made the decision recently to develop an in-house blanking department, thereby improving its turnaround time on most jobs, according to company president, Wendy Young. "We were reliant on a number of outside suppliers and, while our volume overall is quite substantial, we were often slow to receive some small, project-specific blanks for production. Many of our jobs are short-run, highly specialized precision gears, and that means we place a premium on being very efficient in our time-to-first-prototype. The Takisawa is already making a big impact on our blanking operation here."

Tommy Kalt, who runs the blanking department at Forest City Gear, concurs. "We're achieving a 27-second cycle of continuous turning, and the fully automatic mode means a big boost in production for our department. Because we do so many jobs that require relatively few blanks, our overall speed was hampered, due to excessive downtime for set-ups. That situation is diminished to a great degree with the Takisawa machine. Having this capability allows us to reduce our lot sizes on high-volume blanket orders and increases our ability to prototype."

For more information:
Forest City Gear
11715 Main Street
Roscoe, IL 61073
Phone: (866) 623-2168
www.forestcitygear.com

Big Kaiser 318 Series–

A COMPLETE TOOLING SYSTEM

Big Kaiser has introduced the new, large diameter Kaiser 318 Series—a complete tooling system supporting a variety of aluminum and steel components optimized for twin cutter rough boring, precision finish boring and precision OD turning operations.

continued

The new series features a standard boring range of 7.87–13.39" diameter for ISO40/HSK-A63 tapers, and 7.87–24.41" diameter for ISO50/HSK-A100 and larger tapers. Special components allow the system to extend up to 118" diameter. The mounting flanges feature Kaiser's new CKN modular connection with a three-screw interface, developed for high torque transmission with lightweight tools. The system supports spindle speeds of up to 6,600 SFM thanks to safe and secure "pinned to fit" lightweight aluminum mounting components. The high strength aluminum components are hard coated to

protect against wear and corrosion, and all assemblies deliver high pressure coolant through the tools to cutting edges.

"The 318 series features aluminum extension slides that allow for diameter and length setting adjustments without a tool pre-setter," says Jack Burley, Big Kaiser vice president of sales and engineering. "The simplicity of this system virtually eliminates operator error during assembly and promotes safety during operation."

For more information:
Big Kaiser
2600 Huntington Blvd.
Hoffman Estates, IL 60192
Phone: (847) 228-7660
www.bigkaiser.com

Luren

RELEASES THE LFG-8040 VERTICAL GRINDING MACHINE

Luren Precision Co., Ltd. recently released a vertical type CNC gear profile grinding machine. The LFG-8040, which can form grind spur and helical gears up to 800 mm (31.7") in outside diameter and maximum module of M20 (1.27 DP), incorporates an eight-axis CNC controller with close-loop servo control system, direct drive motor and personal computer. A Windows-based user-oriented smart interface developed by Luren is used to operate the LFG-8040. The operator can easily key in gear data, modify gear profile and specify the grinding steps. No NC programming is needed because the grinding program will be generated automatically. Involute profile can be checked easily as desired. K-chart points can be set and adjusted freely by rolling arc, rolling angle or radius on the base circle. Lead modification can be carried out easily as well. Operators can also set up the grinding procedure based on their own data or experience. On-board gear measurement, automatic stock dividing and rotary dressing are also included in the standard machine.

For more information:
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Schaumburg, IL 60173
Phone: (847) 598-3555
www.luren.com.tw

Milwaukee Machine Works

PURCHASES HEXAGON CMM

Hexagon Metrology, Inc. recently announced that precision component manufacturer Milwaukee Machine Works has purchased a Leitz PMM-G ultra-accurate coordinate measuring machine (CMM). The Leitz measuring machine has a supersized measuring capacity of three meters wide by four meters long, and two and a half meters high, tailor made for the large scale machined components that are a Milwaukee Machine Works specialty. The PMM-G is a gantry configuration machine that allows large scale parts up to 30,000 pounds to be easily moved inside for precision measurement. It also boasts the highest accuracy specifications for its size, enabling ultra

continued

precision measurements to be made for case, cover and housing parts for the wind, off-highway and mining industries which comprise MMW's typical customer base.

"This new machine is a huge enhancement to our capabilities," said Mike Manna, general manager of Milwaukee Machine Works. "We are known for tackling the toughest, highest precision, turning and machining jobs in the business—jobs that other shops just don't have the capability or expertise to handle. Now we will have a measuring machine with the size, capacity and accuracy to handle the very largest parts we are capable of making—single parts that are the size of small cars."Milwaukee Machine Works has always been a good customer of Hexagon Metrology, and represents the best of what American manufacturing is capable of," said Jack Rosignal, vice president of sales for Hexagon Metrology, Inc. "We are pleased that the first PMM-G Coordinate Measuring Machine to be installed in America is going to a company like Milwaukee Machine Works, that has the commitment to precision manufacturing and the highest quality standards."

The new Leitz PMM-G is scheduled to come on line in Milwaukee Machine Works' ISO 9001:2008-certified facility by the second quarter of 2011.

For more information:

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www.hexagonmetrology.com

Milwaukee Machine Works
404 South 116th Street
Milwaukee, WI 53214
Phone: (414) 476-3285
www.milwaukeemachineworks.com

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Induction Heat Treating

Gains Ground through Advances in Technology

William R. Stott, Managing Editor

This cutaway shot shows the heat treat and quenching pattern of a spur gear induction hardened with the TSH technology (courtesy of ESR Engineering Corp., the North American distributor of TSH steels).

In recent years, there has been significant interest in expanding the use of induction hardening in gear manufacturing operations. One of the reasons is that induction hardening is easily incorporated into a manufacturing cell. Parts don't have to be taken out of the production flow and sent to a separate heat treating department. Another reason is that induction hardening is often considered a "green" alternative.

"Induction continues to have advantages that include precise heating of just the are requiring heat treat, thus minimizing the energy applied to the part," says George Welch, manager of heat treat products for Ajax-Tocco Magnethermic. "In addition, induction does not consume energy when idle. Combine this with reduced energy for auxiliary equipment for quenching, etc., energy usage and costs are generally much less. Other advantages include fast processing time, fewer parts in process, cell manufacturing, less distortion and individual part traceability."

In the past, the problem was that induction hardening was limited in its potential applications. Only certain gear shapes and configurations were good induction hardening candidates. If you wanted to case harden a spiral bevel automotive pinion, for example, carburizing has long been the practical choice.

Also, induction hardening has been possible only with certain steels.

continued

Tooth-by-tooth induction hardening is often used for larger gears. Modern machines are equipped with sophisticated controls and software, allowing precise control over the hardness pattern and tooth-by-tooth documentation for quality control (courtesy Ajax Tocco Magnethermic).

Induction Heating a large Caterpillar gear for hardening (courtesy of Caterpillar, Inc. and Ajax Tocco Magnethermic Corp).

Typically this has meant both plain and alloy medium-carbon steels with 0.4 percent to 0.6 percent carbon content, according to Dr. Valery Rudnev, group director of science and technology for Inductoheat.

Furthermore, induction heat treating machines have often been designed for specific gear parts or a limited range of parts. Each part often requires customized tooling, and a power supply (or power supplies) that provide the necessary power and frequency to achieve the desired heating pattern. These systems have limited flexibility.

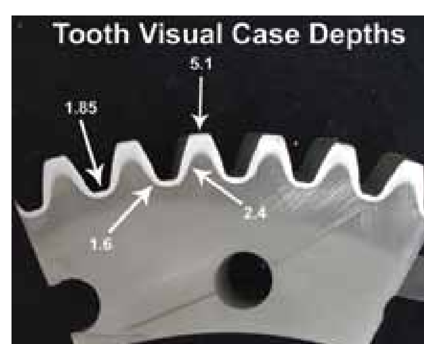
But over the past several years, many of the limits to induction hardening have shrunk, thanks to recent advances in technology, materials and processing techniques.

Power Supplies

The heart of an induction hardening system is the coil, but its brains are the power supply and controls. In recent years, these are the areas where induction hardening technology has improved the most.

"The power supplies have continued to advance and provide new solutions for heat treating parts including gears," says Welch. "Higher power and greater frequency flexibility allow for high power densities at the ideal heating frequency. This aids in processing a greater range of gear sizes and pitches. In addition, advancements in controls allow for better tracking of the entire heat treat process."

Inductoheat's Rudnev agrees that power supplies have been an area of significant advancement. In fact,



New steels and processing techniques allow case hardening of gears by induction at a much lower cost and with less energy consumption than previous technologies (courtesy ESR Engineering Corp., North American distributor of TSH steels).

TSH technology allows difficult components, such as spiral bevel gears and pinions, to be heat treated via induction. Most of these gears are currently carburized (courtesy ESR Engineering Corp., North American distributor of TSH steels).

Rudnev says, Inductoheat has recently introduced power supplies that significantly increase the flexibility of induction hardening systems by providing the unique capability of controlling not just an inverter's output power, but also its frequency, in a manner similar to the control of CNC machines.

Whereas previous generations of power supplies had limited flexibility, Rudnev says, the current generation can produce a high-end frequency up to eight times as great as the low-end frequency. What this allows, Rudnev says, is expanding the range of gears to be induction hardened using the same power supply and/or tempering of a part in the same setup as hardening.

"The majority of gears are tempered in ovens," Rudnev says, "but we were able to develop a new power supply that allows induction hardening and tempering at optimal frequencies."

For example, with the new power supply, you can "dial" a frequency of 40 kHz to induction harden a fine-tooth gear. Then you can dial it down to 6 kHz to temper it at low frequency.

"Inductoheat's Statitron-IFP power supply technology provides a wide range of frequencies," Rudnev says.

The same capability also allows the same machine to induction harden different pitch gears.

The use of simultaneous dual-frequency induction hardening is a proven way to effectively control distortion when induction hardening small and medium size gears using encircling inductors. In many cases, simultaneous dual-frequency technology allows keeping distortion after induction hardening within the range of 80–100 microns, Rudnev says.

Increasing the frequency range of power supplies has also been on the mind of designers at Ajax-Tocco Magnethermic. "Induction systems can be designed to heat treat a range of part sizes and families with tooling changes," says Welch. "Power systems can be provided with wide frequency ranges and load matching capability to process a variety of case depth requirements."

In addition, Welch says, larger power supplies have enabled induction hardening of larger gears.

Controls

Of course, like all machine tools, induction hardening machines have benefited from advances in control technology and software.

"Large gears are generally hard-

continued

Correction:

In the January/February issue of Gear Technology, *a photo caption incorrectly identified a Mazak machine as a Depo machine (Bottom photo, page 30) in the article, "The Merits of Multifunctional Machining."*

Gear Technology regrets the error.
—The Editors

Clarification:

A statement appears on page 77 of the Jan/Feb gear schools article—"Now, More than Ever"—that may have been misleading to readers. In the second complete paragraph, a quote attributed to Geoff Ashcroft of the Gear Consulting Group (GCG) includes the word "certification." To be clear, the AGMA board has made a conscious decision not to have a certification program at this time, due to certain liability issues. The course Geoff Ashcroft instructs is indeed licensed to GCG by AGMA, but participants receive a certificate of completion, *which of course is not AGMA certification.*

Gear Technology regrets any resultant confusion.
—The Editors

ened with the tooth-by-tooth method," Welch says. "This process was developed about 60 years ago. The newest machine controls include recipes that precisely locate the gear tooth and provide motions to scan harden OD, ID, spur and helical gears."

Furthermore, the controls and software provide important quality control data.

"These systems collect the process information available for each tooth for traceability of each gear heat treated," Welch says.

Materials

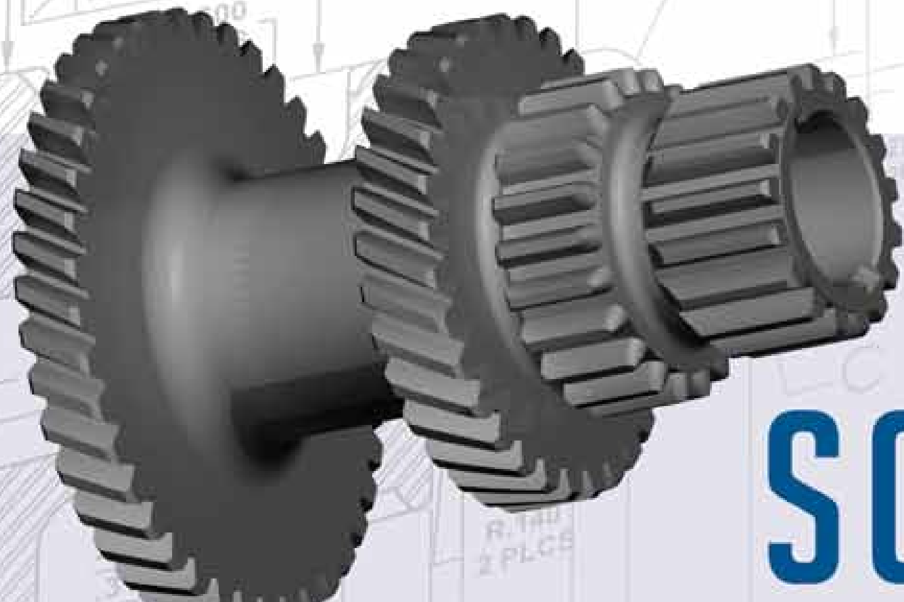
Induction hardening is generally limited to certain types of steels. Unlike carburizing, during which carbon is infused into the part, induction hardening requires higher-carbon steels to begin with. Typical induction hardening steels include AISI 4140, 4340, 1045, 4150, 1552 and 5150, Rudnev says.

But new advances in materials have opened up new applications to induction hardening. One example is the range of TSH steels available exclusively from ERS Engineering Corp. ERS has worked with its European partners to develop the steel metallurgy and process that will allow induction hardening to replace carburizing for a wide range of parts, including parts previously thought to be impossible to induction harden successfully.

Inductoheat has been chosen by ERS to be their exclusive manufacturer of induction heat treating systems for TSH technology in the North American market.

The latest TSH steels are low-alloy carbon steels, which are characterized by reduced grain growth during heating into the hardening temperature range. They can be substituted for more expensive standard steels typically used for conventional induction hardening or carburizing.

TSH steels have significantly less alloying elements such as manganese, molybdenum, chromium and nickel, making them less expensive than conventional carbon steels, Rudnev says. "Their chemical composition is somewhere between micro-alloy steels and plain carbon steels, providing fine-grain martensite with extremely high compressive stresses at the tooth surface."



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With TSH technology, parts are induction heated at low frequency and rapidly quenched, resulting in a contour induction hardening pattern (courtesy ESR Engineering Corp., North American distributor of TSH steels).

The combination of the new steels with the right technique has even opened up the application of induction hardening to such difficult parts as spiral bevel automotive pinions.

"In the past, all of those parts used to be carburized," Rudnev says. "Due to a combination of shape complexity and gear sizes, it was practically impossible to induction harden those parts. Now it is possible to get those beautiful hardness patterns on complex-shaped gear parts by through-heating those parts using low frequency inverters."

With TSH technology, after through heating, the parts are rapidly quenched. The hardened depth is mainly determined by the steel's chemical composition and initial microstructure.

Also, because the parts are through-heated using a low frequency (1–25 kHz, depending on the application), the cost of capital equipment is also low, Rudnev says.

The TSH technology is extremely promising, due to the obvious cost savings and it applicability to the "lean and green" approach, Rudnev says. "I do believe that this technology has a very bright future in the North American Market."

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Grinding, Finishing and Software Upgrades Abound

Matthew Jaster, Associate Editor

Machine tool companies are expanding capabilities to better accommodate the changing face of manufacturing. Customers want smaller-sized equipment to take up less valuable floor space, multifunctional machines that can handle a variety of operations and easy set-up changes that offer simplified operation and maintenance. Recent grinding and finishing innovations include Mitsubishi's new ZE40A and ZE60A grinding machines, Reishauer's grinding wheel technology, Liebherr's LCS 1200, Gleason's Opti-Grind technology and the multifunctional capabilities of Samputensili's Invento series. These companies are proving that R&D is essential as they actively seek out more accurate drives and guidance and control systems in an effort to achieve better gear tolerances.

Mitsubishi. With the release of the new ZE40A/ZE60A series and future upgrades to the ZI20A internal grinding machine, Mitsubishi Heavy Industries America, Inc. is keeping busy in the gear market. Ian Shearing, vice president of sales at Mitsubishi, recently discussed the advantages of the company's newest grinding technology.

"High speed, smooth rotation without vibration, resulting in higher gear accuracy and lower gear noise," Shearing says. "Approximately 50 percent of the floor space for any grinder is occupied by the coolant tank and temperature control systems. Mitsubishi has concentrated on this area to reduce the floor space required by the system while at the same time maintaining its working efficiency."

The ZE40A/ZE60A series is aimed at job shops that produce a wide variety of gears in small lots (courtesy of Mitsubishi).

The ZE40A/ZE60A can accommodate both generating grinding and profile grinding and also enables modification of biased flank shapes, i.e., tooth surface torsion. Mitsubishi's Machine Tool Division will explore demand for the new machine, primarily targeting job shops that produce a wide variety of gears in small lots.

"Owing to the development of special operator software, the equipment is easy to use," Shearing says. "By simply answering easy gear-related questions, the machine's CNC control is able to write the process program automatically."

The ZE40A, a nine-axis, fully automated numerical control (NC) machine, was developed with features such as high-precision machining for a broad variety of workpieces and easy setup changes. In order to accommodate small-lot production of various gear types, the machine offers easy and simple operation and maintenance and compact size for reduced installation space.

The new machine has achieved smooth revolution of the main (grinding) spindle and table spindle through adoption of a direct-drive system employing built-in motors to elimi-

continued

nate vibration from reducers. In the dressing unit, which functions to maintain the shape and sharpness of the grinding device, torsion angle adjustment is automated by NC. The installation space required for the ZE40A, 3,750 mm in width and 4,000 mm in depth, marks a 15 percent reduction, even when compared with the ZE15 and 24A models for smaller workpiece machining, achieved by slashing the size of the coolant device and rearranging the hydraulic and pneumatic devices.

While the maximum external gear diameter of workpieces is 400 mm for high-efficiency machining by the ZE40A, the machine can actually accommodate up to 600 mm diameter workpieces when machining requirements are stepped down, thanks to the wide space secured around its workpiece table. The ZE40A is able to machine gears with between 0.5 and 8 modules. Users can set the desired gear shape, in either generating grinding or profile grinding mode, by inputting NC data. In the case of helical gear machining, it is possible to modify bias generated during crowning.

With the addition of the ZE40A universal-purpose gear grinding machine to its product portfolio, Mitsubishi will now further boost marketing activities in a quest to expand users of its gear cutting machines.

As far as grinding technology moving forward, Shearing says that Mitsubishi's ZI20A internal gear grinding machine will be enhanced with external gear grinding for cluster gears and shaft-type pinion gears. "This development will enhance and expand the capabilities of the machine, thus opening it up to a wider customer base."

Reishauer. Since the introduction of the current generation of Reishauer machines in 2001, the continuous generating grinding process has become the dominant gear finishing process among the available alternatives for new transmission projects worldwide. This success is mainly due to the fact that the final quality and process reliability, along with decreasing production costs per piece, can be achieved with the continuous generation process.

"Production costs are not only dependent on the machine but as well on the required tooling," says Michael Engesser, CMO at Reishauer. "Reishauer has decided to take advantage of its extensive knowledge of this technology to develop new tools to further reduce the costs per piece."

Grinding wheels from Reishauer have demonstrated extended life and productivity, according to Michael Engesser, CMO (courtesy of Reishauer).

Reishauer now takes the next step with the introduction of a new grinding wheel that is specifically designed to meet the high requirements of continuous generating grinding technology.

Introduced during the 2010 AMB exhibition in Stuttgart, Germany, the new grinding wheels have been successfully launched into production for various customers worldwide. Based on several applications, the grinding wheels have demonstrated extended life and increased productivity given the tough environment of industrial use. These new wheels are available in a variety of specifications including increased demands in surface finish, hardness of the workpiece material or tool life.

In addition, Reishauer grinding wheels can also be used for discontinuous profile grinding on the latest generation of Reishauer gear grinding machines. Reishauer has invested in a new production site in the state of Luzern, Switzerland for the production of the grinding wheels. "By optimizing production facilities to produce only grinding wheels for Reishauer gear grinding machines, production costs can be kept at a competitive level, despite the technological benefits for the customers. Currently, production capacity is ramped up to meet the increasing demand for this unique new product of Reishauer AG," Engesser says.

Liebherr. The LCS 1200 from Liebherr combines both generating grinding and profile grinding into one machine. It is able to handle workpieces up to a diameter of 1,200 millimeters.

"Gears up to module 12 can be economically processed with the generating grinding method," says Dr. Alois Mundt, managing director at Liebherr-Verzahntechnik GmbH. "The machine has the capacity to produce noise-minimized gears in a two-flank grinding process exactly to a specified twist design."

With profile grinding, gears up to module 22 or a profile height up to 50 millimeters can be ground. Tools with either electroplated CBN or dressable tools with a corundum, sinter corundum or CBN basis can be used for both grinding technologies. The high productivity machine can be supported by machine integrated automation, also designed and built by Liebherr in Kempten, Germany. For the generating grinding method, the maximum outside diameter of 320 millimeters on a grinding worm and the minimum usable diameter (which depend on the gear data), built in combination with the tool length of 230 millimeters, offer the longest tool life in this machine type class on the market.

The technical data of Liebherr's LCS 1200 includes a maximum workpiece diameter of 1,200 mm; workpiece speed of 250 rpm; maximum module (generating grinding) of 12 mm; maximum module/profile height (profile grinding) of 22/50 mm; maximum axial travel of 1,000 mm; maximum tool speed of 12,000 rpm; total weight of machine of 28,000 kg.

Gleason. Gleason's latest generation of profile grinding machines are equipped to perform Opti-Grind, a process that improves productivity and quality levels for fine finishing cylindrical gears as large as six meters in diameter, and up to module 16. The Opti-Grind process enables end users to achieve both "optimum" productivity, and "optimum" surface finishes, by simultaneously using multiple dressable grinding wheels for profile grinding rather than just the single grinding wheel that is typically used. The process offers end-users a number of variations to choose from to meet their specific applications.

Opti-Grind's multiple-wheel configuration offers advantages for gear producers serving the wind power and other industries requiring optimum surface finishes. Where a single dressable wheel must be designed for compromise in order to perform both roughing and finishing, the multiple wheel configuration of Opti-Grind makes it possible to utilize wheels designed for maximum roughing productivity, up to 40 percent faster than a single dressable wheel. Then a finishing wheel is designed to produce the desired tooth modifications including grinding of the root without burning, and to deliver surface finishes up to four times finer than what would be possible conventionally.

In addition to the new multiple wheel design, the Opti-Grind process relies on the use of the latest Siemens 840D CNC and Gleason Windows-based *Intelligent Dialogue* software to greatly simplify setup and operation.

Samputensili. Samputensili developed the HG 1200 Invento for production of large gears with a diameter up to 2.5 m and module 35.0. It is distributed in the United States, Mexico and Canada by Star-SU.

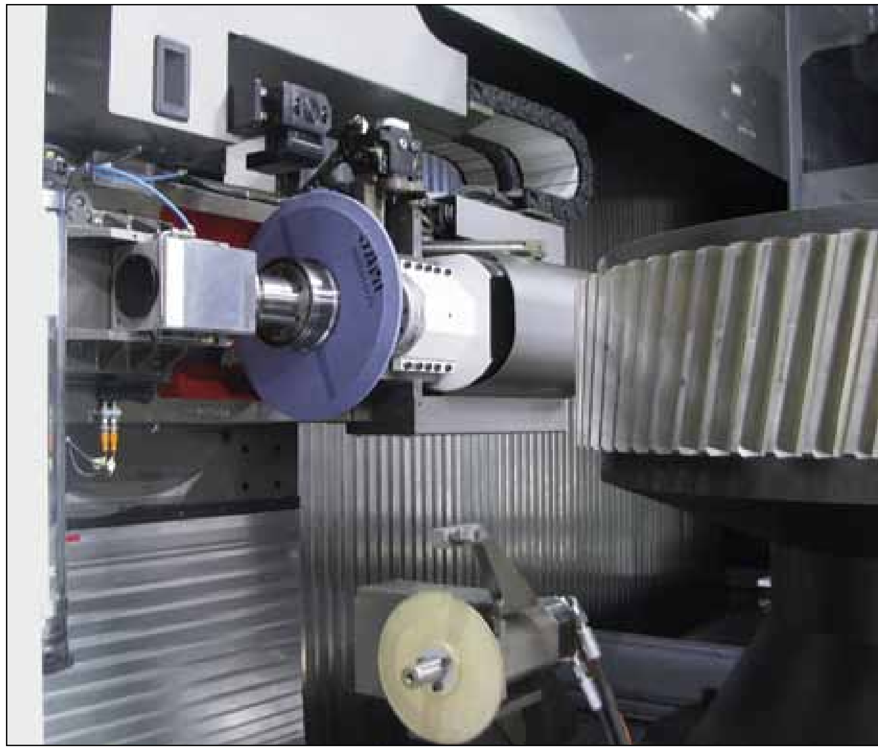
While some gear manufacturing machines are limited by the basic difference between the processes of hobbing and grinding, the HG Series overcomes this problem, thanks to its functioning principle: during the hobbing process the hob head is blocked to unload the cutting forces to the stable tool column while the table slide actuates the feed travel. In this way the strong hobbing forces can be mitigated without compromising the necessary grinding quality in the long run.

When grinding, the process is reversed. The tool head is unlocked and moves against the blocked work table. In this way the reference axes during the centering and grinding process maintain the right position and guarantee the required highest precision. In the same way, the optional

continued

Gleason's Opti-Grind process debuted in 2010 and enables users optimum productivity and surface finishes (courtesy of Gleason).

The LCS 1200 combines generating and profile grinding in one machine (courtesy of Liebherr).



The HG Invento series executes various processes including grinding with optimal precision (courtesy of Star-SU).

back columns are fixed laterally to the work table so as not to block either the access to the working area or the optimized flow of loading and unloading workpieces.

In order to execute the various processes with optimal power transmission and precision, two tool heads are available with three different spindle inserts. The stiff hob spindle of the HG Series is designed for the toughest cutting conditions. The 100 kW drive runs all state-of-the-art roughing tools—especially inserted blade roughing and finishing cutters in different combinations. Additionally, it is possible to apply a turn-milling spindle in the hob head so as to realize a wide variety of turning functions. The use of the turn-milling process is especially recommended for very small lots and prototypes, including the production of bevel gears. The grinding spindle has been designed for a large variety of different tool combinations, including single-profile grinding wheels and multi-rib wheels, as well as for generating grinding worms. Thanks to a combination of the various processes—i.e., hobbing and grinding—production time can be cut by 25 percent, provided that this is allowed by the module range of the application.

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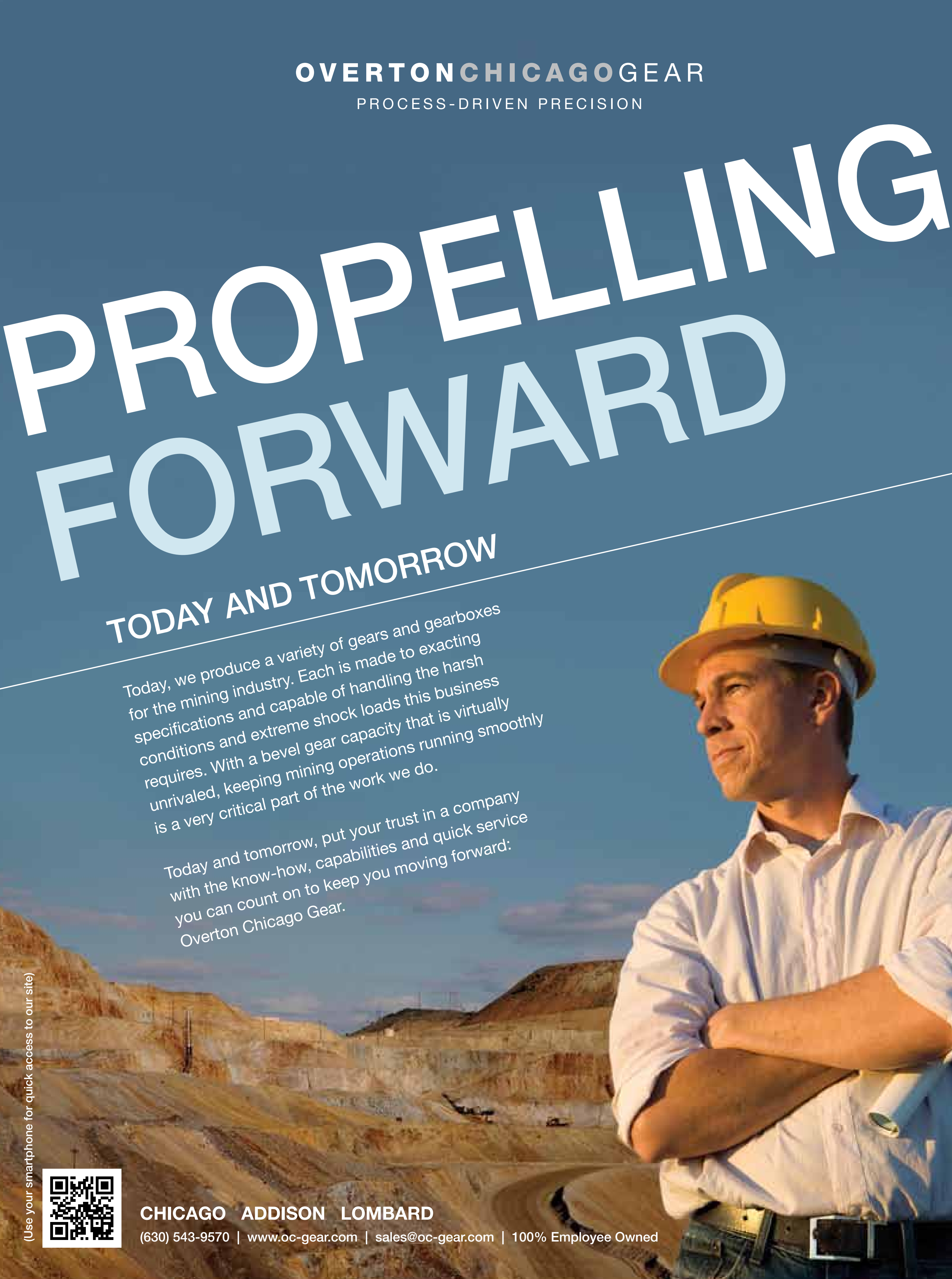
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Long-Awaited Int'l Standard

FOR WIND TURBINE GEARBOXES INCHING CLOSER TO COMPLETION

Jack McGuinn, Senior Editor

It's hard to imagine a world without manufacturing standards, although in fact the concept is relatively new when one thinks about it. Some history: The record states that international standardization began in the electro-technical field with the creation in 1906 of the International Electrotechnical Commission (IEC), and was further advanced in 1926 by the creation of the now-defunct International Federation of the National Standardizing Associations (ISA), whose emphasis was mechanical engineering. It was in a post-war meeting in 1946 that delegates from 25 countries created a new international organization that would "facilitate the international coordination and unification of industrial standards," i.e.—the ISO.

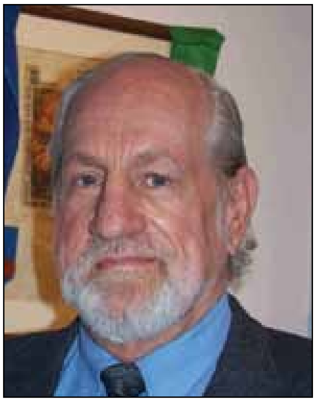
AGMA entered the picture in 1916 as an evolving gear industry was striving to meet the auto industry-driven demand for gears with quiet operation, particularly for timing gears. AGMA began when the R.D. Nuttall Company brought together several gear manufacturers to discuss developing such standards. From Day One, AGMA standards development has been predominantly market-driven, beginning in 1919 with the first rating standard, and the first gear quality standard was established in the late 1930s.

Today, international standards serve as the manufacturing rule of law, if you will, regarding guidelines or definitions of characteristics to ensure that materials, products, processes and services are fit for their purpose.

All of which brings us to the issue at hand—the status of publication of ISO/IEC 61400-4, otherwise known as a new standard for wind turbine gearboxes.

Faithful *Gear Technology* readers may recall that our July 2009 issue contained an update of the deliberations provided by Bill Bradley (*Ed.'s Note: Bradley, a Gear Technology technical editor, is also a consultant within the gear industry from Longmont, CO with over 45 years' experience. As a member of the American Gear Manufacturers Association, he is active on a number of standards committees, including—Wind Turbine, Helical Gear Rating and Gear Accuracy. As an AGMA VP, he was responsible for facilitating national and international standards development until he retired in 2007*).

Now, almost two years later, there is an ISO/IEC wind turbine gearbox standard out for draft international standard ballot (ballot closes 2011-05-17), with an AGMA meeting in Denver (March 16–17) convened to decide the U.S. position on its wording.



Bill Bradley

With that, we contacted Bradley again, along with two additional authorities on the subject, to see where things stand on its ratification and to determine what, if any, effects the standard will actually have on gear industry players.

Where we're at, according to Bradley: "The activity on the ISO/IEC wind turbine project 61400-4 since July 2009 includes ballot of a committee draft (CD) between October 2009 and January 2010, which resulted in over 900 comments (about 50 percent were deemed editorial). Two meetings of the Joint Working Group (JWG) for resolution of the comments were held in Copenhagen (March 3–5, 2010) and Hangzhou, China (April 26–28, 2010). In these meetings, over 300 substantive (non-editorial) changes to the document were agreed to, and the resulting document had at least 30 major changes in the technical requirements. Contrary to normal ISO procedures, it was agreed to send the resulting document for draft international standard (DIS) ballot, rather than another CD. The DIS ballot opened December 17, 2010 and will close on May 17, 2011. There is an AGMA meeting scheduled for March 16–17 to decide the U.S. position on this document for which about 100 U.S. committee members submitted

continued

N.K. "Chinn" Chinnusamy

comments to be reviewed.

"In my more than 30 years' experience in ISO standards development, this project has been the most difficult; i.e., to maintain a reasonable schedule; for the ISO/IEC JWG members to develop a consensus document; and for the chairman and document editors to maintain neutral positions. This is probably due to vested interests of individuals and companies who are involved in the JWG."

Just keeping the acronyms straight seems like a big enough task, much less passing the thing.

The next, and obvious question for Bradley—When will the development of this standard come to fruition? Melville probably wrapped up "Moby Dick" in quicker fashion than this years-in-the-making standard, with work on it beginning in the 1990s.

"It is very difficult to predict the time to completion of this document," he says. "Based on past performance, where twice the usual time has been taken by the JWG to accomplish its tasks, I think it may take one to two years to be published, if it is to be a standard. After the May ballot closure, there should be a JWG meeting scheduled to resolve comments. After the resolution meeting(s), there is normally a 3–4 month edit period, which has taken twice that amount in the past. If there are any substantive changes, another three-month re-ballot is required. Then another resolution and edit period is required."

Clear on that?

Aside from actual passage of the standard, it is useful to know—once it becomes operational—what effect it will have in the real gear world. After all, standards are created resulting from, according to ISO, consensus agreements reached between all economic players in that industrial sector—suppliers, users and often governments. They agree on specifications and criteria to be applied consistently in the choice and classification of materials, the manufacture of products and the provision of services. The aim is to facilitate trade, exchange and technology transfer through:

- Enhanced product quality and reliability at a reasonable price
- Improved health, safety and environmental protection, and reduction of waste
- Greater compatibility and interoperability of goods and services
- Simplification for improved usability
- Reduction in the number of models, and thus reduction in costs
- Increased distribution efficiency and ease of maintenance

As for this standard's particular effect, consider profitability, for example. Says N.K. "Chinn" Chinnusamy, president of Roscoe, IL-based Excel Gear, "I doubt that it will provide a more level playing field. The AGMA standard is a good reference, but everything is not set in stone and many factors are left to the experience of designers."

"It's hard to say," says Chuck Schultz, chief engineer for Beyta Gear Service and a *Gear Technology* technical editor. "Better gearboxes will reduce warranty expenses, and that certainly goes to the bottom line."

Says Bradley, "An international gearbox rating standard is not worth having unless it provides minimum requirements for a defined operating environment and life. The requirements should result in 'a more level playing field' for its users. A company's profitability could be positively affected if it typically has provided more than the minimum requirements. A company's profitability could be negatively affected if it typically has provided less than the minimum requirements."

And then the question arises whether the new standard might be sufficiently stringent and unforgiving to the point that smaller players may be left on the sidelines.

"Smaller job shops, to the extent they compete today mostly as sub-contractors, will not be precluded from competing by the standard, provided they meet the minimum requirements," says Bradley. "To my knowledge, many can meet these requirements."

Exel's Chinn agrees. "No, I do not think so. There are already AGMA guidelines for wind turbine gearbox design and manufacture."

Beyta's Schultz has a more cautionary take. "The barriers to market entry are equipment, knowledge and test stands," he says. "The testing requirements will certainly make it harder for smaller players. Putting over a million dollars into non-chip-producing assets is a very difficult business decision for most companies."

Chuck Schultz

And what about the seemingly never ending upgrades in grinding, gashing, cutting tools and machinery? Is anticipation of the new standard driving the innovation?

"I think that is an overstatement. Standards don't normally drive advancements," Bradley flatly states.

At Excel, "Yes," says Chinn, in agreement with Bradley. "Gear gashing is mainly for wind turbine gear manufacturing and gear gashing is very cost-effective for large internal and external gears."

But, says, Schultz, "The equipment builders are constantly improving their products for all users. The wind turbine gearbox designer is just as likely to exploit those improvements as the automotive or industrial gearbox designer. We would not see some of the intricate modifications if the machines were not capable of producing it, however.

"In my personal opinion, you cannot 'modify' your way out of a poor overall design. Modifications optimize for one

continued

load condition and sub-optimize for all other load conditions. If you don't know what your operating loads are, you can modify yourself into more trouble."

As for the question, which comes first—market demand or the standard?—all responders were in agreement.

"A market demand normally exists before efforts to standardize," says Bradley. "Standards try to catch up to the demands."

"I think market-driven demand comes first," says Chinn.

Schultz agrees as well, but with a slight distinction. "The standards reflect market demand."

But one wonders, given the lengthy deliberation process, is there a possibility that innovation could soon make the standard somewhat "old school"?

For starters, "Siemens has developed new generators that may not require gearboxes for small wind turbines," says Chinn.

"There is renewed talk and efforts to eliminate the gearbox in wind turbines," says Bradley. "However, efficiencies and cost still seem to make a gearbox more desirable for the large megawatt wind turbines. What might blunt the impact is that by the time (the standard) is finished the industry will already be practicing its requirements. Also, in the end, its requirements may not be much different than today's standard requirements or practice."

Adds Schultz, "The standard will continue to evolve and incorporate new methodology, especially in the heat treat area."

Schultz, as the following makes clear, is fully behind the standard's implementation.

"More effort is needed to understand the loads on the turbines," he says. "The gearboxes continue to underperform and all the torque load testing doesn't seem to be resolving the problem. We know there are deflections of the housing and chassis that affect gear and bearing alignment, but it is very difficult to know which of these loads to test with and how to model them.

"Anyone who claims to know what loads and accelerations the gearboxes will actually see in service is less than truthful. Control systems and condition monitoring are in need of much improvement. Predictive maintenance will be an active area of development."

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Tribology Aspects in Angular Transmission Systems

Part V: Face Gears

Dr. Hermann Stadtfeld

(This article is part five of an eight-part series on the tribology aspects of angular gear drives. Each article will be presented first and exclusively by Gear Technology*, but the entire series will be included in Dr. Stadtfeld's upcoming book on the subject, which is scheduled for release in 2011.)*

Design

If two axes are positioned in space and the task is to transmit motion and torque between them using some kind of gears, then the following cases are commonly known:

- Axes are parallel → Cylindrical Gears (line contact)
- Axes intersect under an angle → Bevel Gears (line contact)
- Axes intersect under an angle → Face Gears (line contact)
- Axes cross under an angle → Crossed Helical Gears (point contact)
- Axes cross under an angle (mostly 90°) → Worm Gear Drives (line contact)
- Axes cross under any angle → Hypoid Gears (line contact)

The axes of face gears in the best sense of the word intersect under an angle of 90°. The face gear geometry is based on a cylindrical pinion (spur or helical) that meshes with a flat-face or crown gear. This naturally requires a shaft angle of 90°. The face gear is derived from the cylindrical pinion, using the principle of conjugate surface generation. Also, shaft angles smaller or larger than 90° can be realized using the same principle, and the face gear becomes a "beveled" gear. Short

continued

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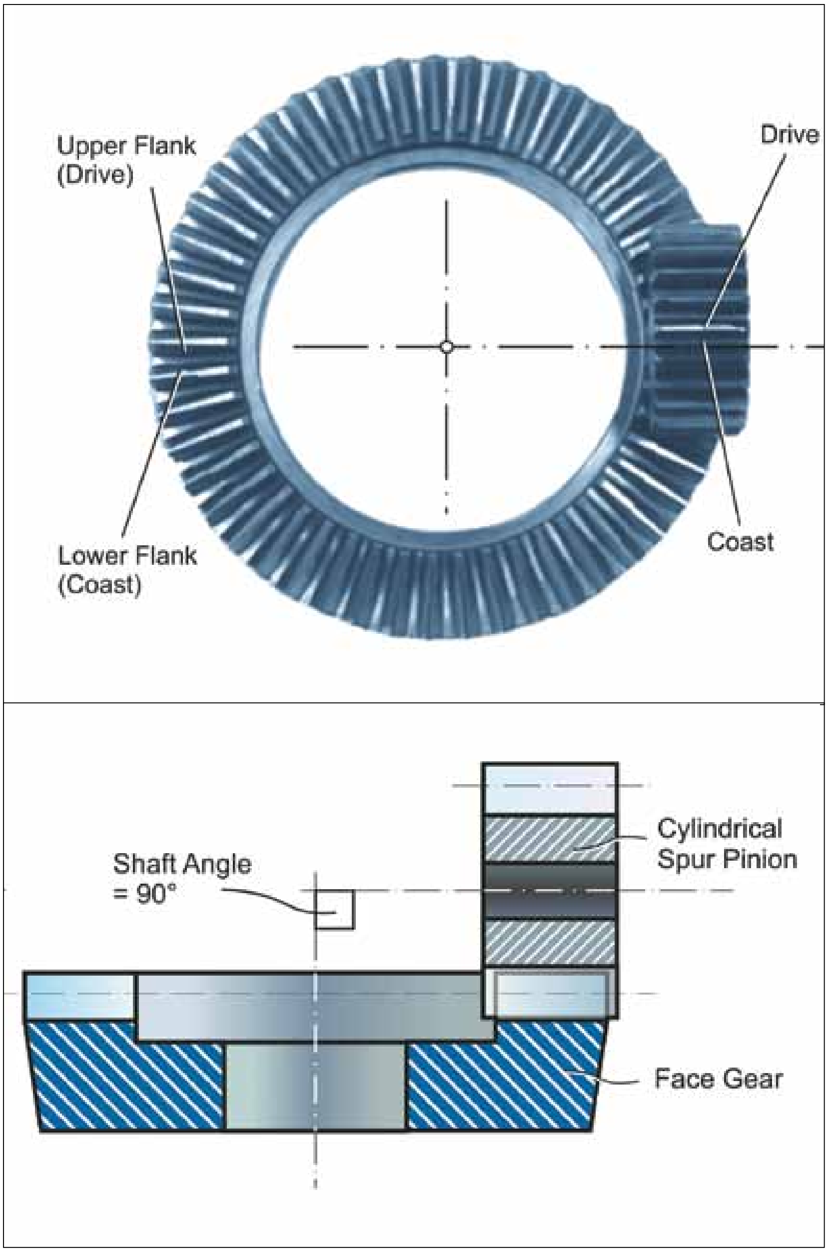

Figure 1—Face gear geometry.

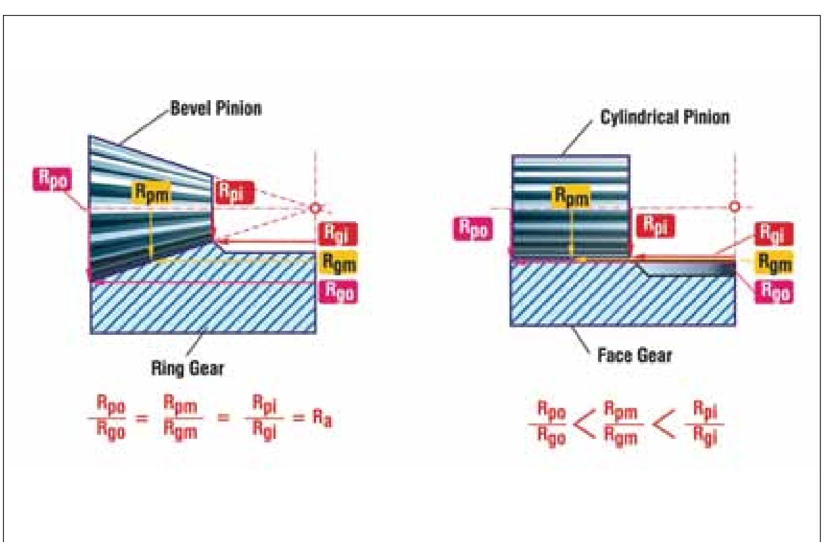

Figure 2—Radii discrepancy in face gears vs. bevel gears.

of a separate name, modified gears in mesh with cylindrical pinions in a non-parallel axis arrangement are called face gears. In beveloid gears it is also possible to use a cylindrical pinion as one member and a beveloid as the second member (manufactured with cylindrical gear cutting and grinding technology). The beveloid in this case is the special case of a face gear with similar flank geometry features.

The pitch surfaces are a cylinder and a cone which are calculated with the following formula:

$$\gamma_1 = 0°$$
$$\Sigma = \gamma_1 + \gamma_2$$
$$\gamma_2 = \Sigma - \gamma_1$$

in case of: $\Sigma = 90°$, $\gamma_2 = 90°$

where:
γ_1 Pinion pitch angle
Σ Shaft angle
γ_2 Gear pitch angle

Face gears are designed and manufactured with parallel tooth depth. The problem is the changing circumference of the face gear member between outer and inner diameter. If the gearing law is applied in order to establish conjugate face gear flank surfaces, it results in heel-pressure angles of up to 45° and toe-pressure angles of down to 0° (in case of 20° pinion-pressure angle). In some area towards the toe—where the pressure angle falls below the limit pressure angle (at about 7°)—the gearing law no longer applies and undercut occurs that extends to the top of the face gear flanks, which is true of a face width commonly used in spiral bevel gears. The solution is to use smaller face width. The profile function of the face gear is not a function found in the family of involutes (octoids).

Figure 1 shows an illustration of a face gear set and a cross-sectional drawing. Face gears with 0° spiral angle have no preferred driving direction. Because of the orientation of the flanks during manufacture, the designations "upper" and "lower" flank are used. Per definition, the calculation programs treat the spur pinion like a left-hand member and the mating face gear like a right-hand member. Consequently there is a drive-side and a coast-side designation for proper definition rather than for implying better suitability of torque and motion transmission.

Analysis

The attempt to apply the gearing law in order to generate a precise mating flank to a given cylindrical pinion flank is possible—but not without obstacles. The straight bevel gear set in the left-side graphic in Figure 2 has identical ratios between the pinion and gear radii along the operating pitch line. This is different for the face gear set to the right in Figure 2 that shows the changing ratio between the pinion and gear radii along the pinion pitch line.

The laws of physics establish the operating pitch line—which is the locus with no sliding and pure rolling—while the correct ratio is transmitted. The operating pitch line in face gear drives is shown in Figure 3. It is identical to the pitch line in an analogue bevel gear set. The inclination between the pinion pitch line and the operating pitch line in Figure 3 leads to a pressure angle distortion. The nominal pressure angle matches only at the crossing point of the operating pitch with the pinion pitch line (in one single point). It increases towards the heel, where it causes pointed topland and reduces towards toe, where it reaches the limit-pressure angle and results in undercut. The face width of face gears is therefore limited in both directions.

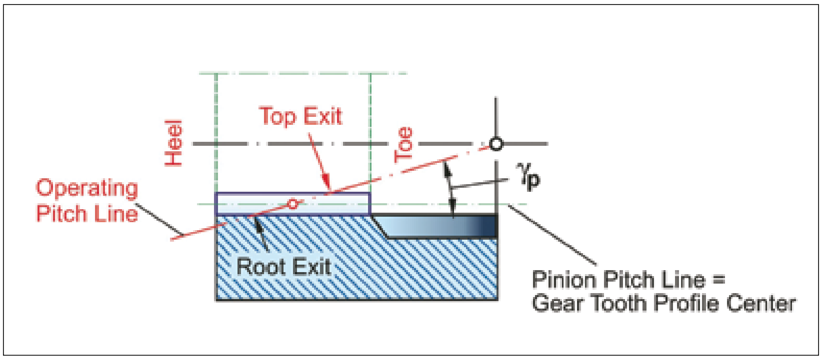

Figure 3—Pinion pitch line and operating pitch line.

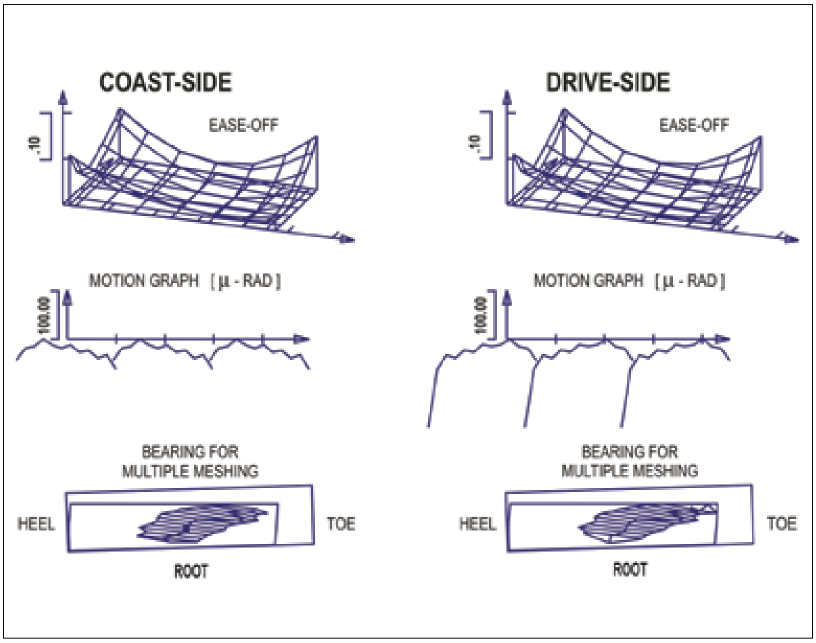

Figure 4—Tooth contact analysis of a face gear set.

The precise conjugate face gear flank surface will result in line contact between the two mating flanks (rolling without any load). In the case of a torque transmission, the contact lines become contact zones (stripes) with a surface stress distribution that shows peak values at the two ends of each observed contact line and where the contact line is limited by the outer end of the tooth (heel) and the undercut zone (toe). In order to prevent this edge contact, a crowning along the face width of the teeth (length crowning) and in profile direction (profile crowning) is introduced into the gear flanks. A theoretical tooth contact analysis (TCA) previous to gear manufacturing can be performed in order to observe the effect of the crowning in connection with the basic characteristics of the particular gear set. This also allows the possibility to return to the basic dimensions in order to optimize them if the analysis reveals any deficiencies. Figure 4 shows the result of a TCA of a typical face gear set.

The two columns in Figure 4 represent the analysis results of the two mating flank combinations (*see also "General Explanation of Theoretical Bevel Gear Analysis"*); however, for face gears the designation "drive" and "coast" are strictly a definition rather than a recommendation. The top graphics show the ease-off topographies. The surface above the presentation grid shows the consolidation of the pinion and gear crowning. The ease-offs in Figure 4 have a combination of length and profile crowning, thus establishing a clearance along the boundary of the teeth.

Below each ease-off the motion transmission graphs of the particular mating flank pair are shown. The graphs show the angular variation of the driven gear in the case of a pinion that rotates with a constant angular velocity. The graphs are drawn for the rotation and mesh of three consecutive pairs of teeth. While the ease-off requires a sufficient amount of crowning in order to prevent edge contact and allow for load-affected deflections, the crowning in turn causes proportional amounts of angular motion variation of about 50 micro radians.

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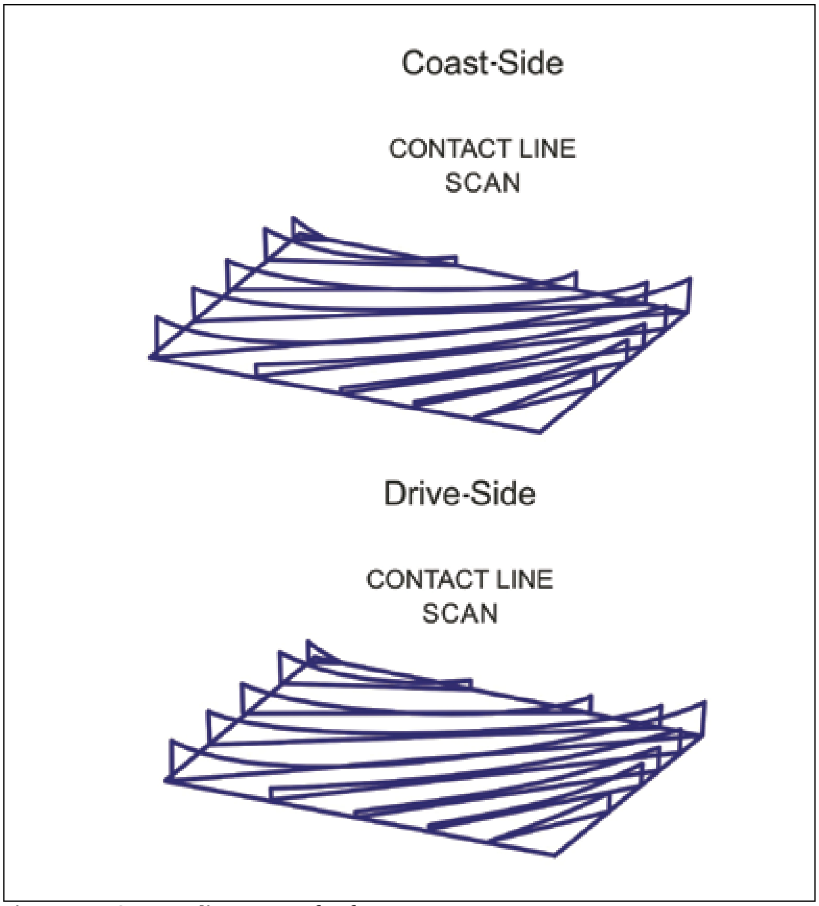

Figure 5—Contact line scan of a face gear set.

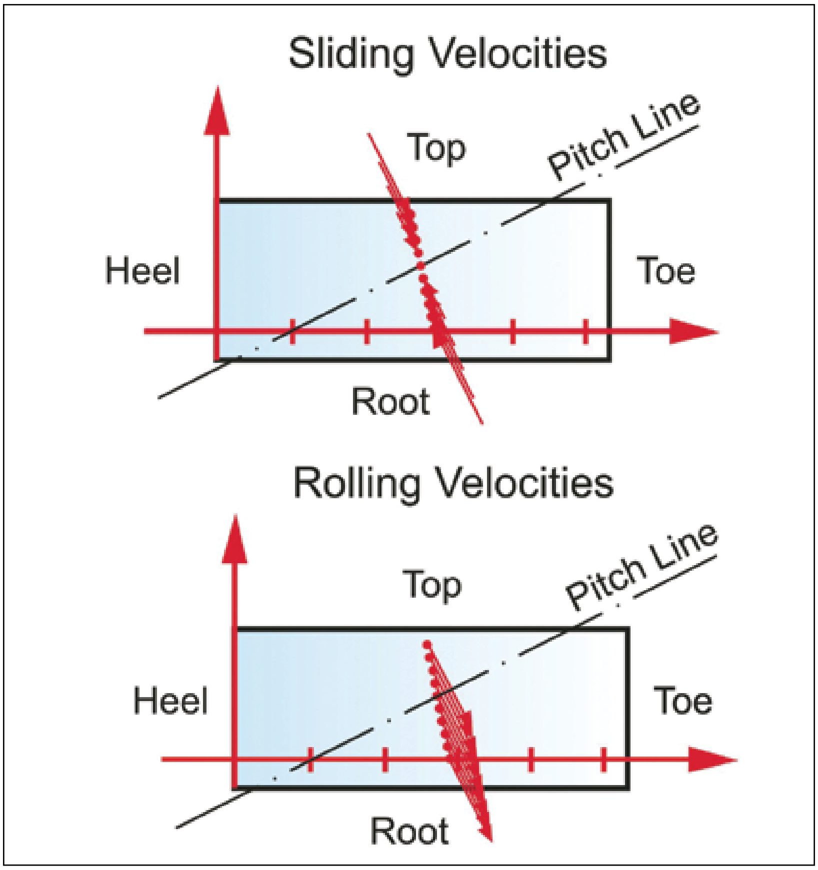

Figure 6—Rolling and sliding velocities of a face gear set along the path of contact.

At the bottom of Figure 4, the tooth contact pattern is plotted inside of the gear tooth projection. These contact patterns are calculated for zero-load and a virtual-marking-compound film of 6 μm thickness. This basically duplicates the tooth contact, one could observe, rolling the real version of the analyzed gear set under light load on a roll tester while the gear member is coated with a thin layer of marking compound. The contact lines show a changing inclination between toe and heel—each of them points to the shaft intersection point. The path-of- contact has a dominating profile orientation; it crossed the central contact line under about 90°.

The crowning reflected in the ease-off results in a located-contact zone inside of the boundaries of the gear tooth. A smaller tooth contact area generally results from large magnitudes of ease-off and motion-graph magnitudes, and vice versa.

Figure 5 shows nine discrete, potential contact lines with their crowning amount along their length (contact-line scan). The orientation of the contact lines is identical to the contact-line orientation in Figure 4. If the gear set operates in drive direction, then the contact zone (instant-contact line) moves from the top heel area to the root toe area.

The graph in Figure 6 illustrates the rolling and sliding velocity vectors. Each vector is projected to the tangential plane at the point of origin of the vector. The velocity vectors are drawn inside the gear tooth boundaries (axial projection of one ring gear tooth). Figure 6 shows at the top the sliding velocity vectors with their tips grouped along the path of contact points. The rolling velocity vectors are drawn in the lower graphic in Figure 6 with their origin grouped along the path of contact. Contrary to spiral bevel and hypoid gears, the directions of both, sliding and rolling velocities, are oriented perpendicular to the pitch line. The rolling velocities in all points are directed to the root while the sliding velocities point to the root, above the pitch line, and to the top, below the pitch line. At the pitch line the rolling velocity is zero, just like in the case of straight bevel gears.

Face gears have properties very similar to straight bevel gears, but for the exception that the rectangular tooth area is under an angle to the pitch line (Fig. 3). Sliding and rolling velocity vectors are pointing perpendicular to the pitch line direction (Fig. 6), which will shift the contact lines in Figure 5 strictly perpendicular to the pitch line. This means that the crowning of the contact lines has no significant influence on the lubrication case (*see "General Explanation of Theoretical Bevel Gear Analysis"*), but only the relative profile curvature perpendicular to the pitch line will define the lubrication case and the hydrodynamic condition. This will lead to lubrication case 3 above the pitch line, and case 2 below the pitch line. In contrast to straight bevel gears, the sliding is high in the toe and heel region caused by the inclined pitch line and the increased distance between the pitch line and the observed flank region. As a result, the efficiency is lower.

Manufacturing

The soft manufacturing processes of face gears are hobbing, shaping or generating with a disk cutter. The hard-finishing processes are continuous-threaded-wheel grinding, single-index generating grinding and skive-shaping.

One example of a cutting and grinding process of face gears is shown in Figure 7. The blades of the disk-shaped cutter (top, Fig. 7) represent with their cutting edges a plane perpendicular to the axis of rotation. A generating motion with a non-constant ratio of roll around the virtual axis of the mating pinion forms the face gear flank profile. The cutter represents a pinion flank meshing with the face gear like the mating pinion. This is a single-indexing process that requires an upper and lower cutting arrangement in order to cut both flanks (*see also "Straight Bevel Gears"*). Despite the single-indexing and single-side cutting, the process is rather fast because of the possibility of high-speed dry-cutting and the fact that the cutter disk only plunges and rolls in one position (without traversing along the face width). The described process leaves an arc in the root along the face width.

The following grinding process for hard-finishing—if required for a particular application (Fig. 7, bottom)—uses a CBN-coated grinding disk that essentially duplicates the cutter-blade silhouette. An advantage of such a related process combination is the fact that soft and hard geometry in flank and root area match perfectly.

Application

Most face gears used in power transmissions are manufactured from carburized steel and undergo a case hardening to a surface

continued

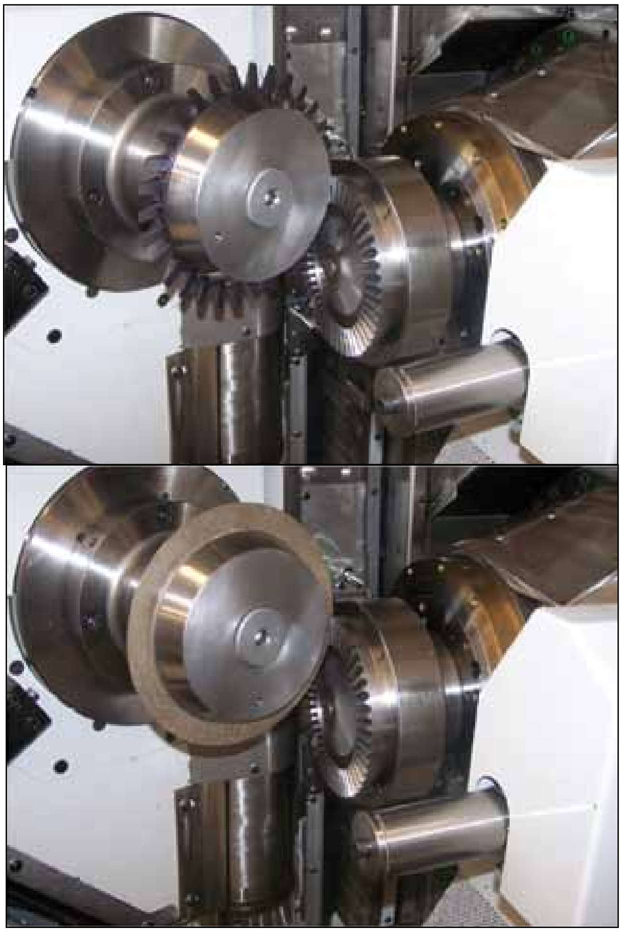

Figure 7—Face gear cutting (top) and grinding (bottom) in free form machine.

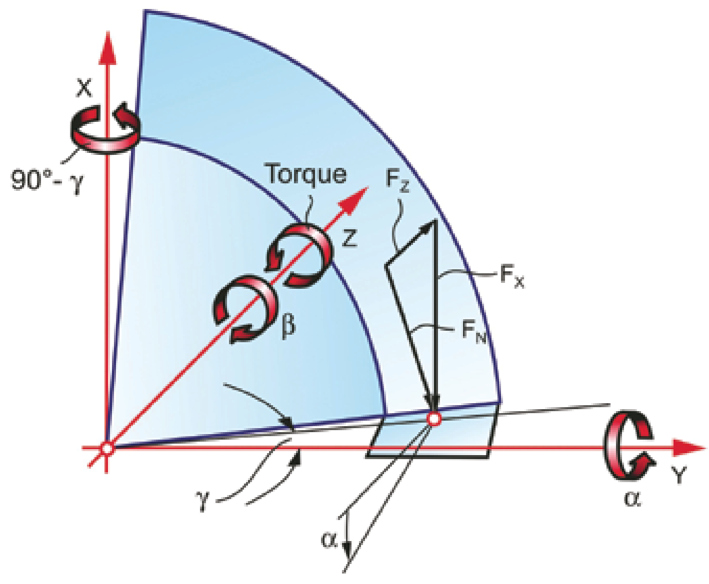

Figure 8—Force diagram for calculation of bearing loads.

hardness of 60 Rockwell C (HRC) and a core hardness of 36 HRC. Because of the higher pinion revolutions, it is advisable to give the pinion a higher hardness than the ring gear (e.g., pinion 62 HRC, gear 59 HRC).

Regarding surface durability, face gears are very similar to straight bevel gears and spur gears. At the pitch line, the sliding velocity is zero and the rolling velocity under certain loads cannot maintain a surface-separating lubrication film. The result is pitting along the pitch line that can destroy the tooth surfaces and even lead to tooth flank fracture. However, it is possible that the pitting can stabilize if the damage-causing condition is not constant in the duty cycle.

In a face gear set, the pinion is a standard spur or helical pinion. Its axial position will not change the backlash between pinion and gear and it is not critical to the gear set's performance. This reduces the requirements for bearings and assembly accuracy, which is welcome in many applications.

The spur pinion of a face gear set has no axial forces. All axial forces a face gear is subjected to are caused by the pressure angle, which is similar to straight bevel ring gears with a simmilar pitch angle γ. The axial force components—due to the spiral and pitch angles—are, for both members—zero. The zero spiral angle minimizes the face contact ratio of face gears to zero, but results in the maximal tooth root thickness.

The bearing forces of face gears can be calculated by applying a normal force vector at the position of the mean point (see "General Explanation of Theoretical Bevel Gear Analysis"). The force vector normal to the transmitting flank is separated in its X and Z components (Fig. 8).

The relationship in Figure 8 leads to the following formulas, which can be used to calculate bearing force components in a Cartesian coordinate system and assign them to the bearing load calculation in a CAD system:

General

$$F_x = -T / (A_m \bullet \sin\gamma)$$

$$F_y = -T \bullet (\cos\gamma \bullet \sin\alpha / (A_m \bullet \sin\gamma \bullet \cos\alpha)$$

$$F_z = T \bullet (\sin\gamma \bullet \sin\alpha) / (A_m \bullet \sin\gamma \bullet \cos\alpha)$$

Cylindrical pinion

$$F_x = T / A_m$$

$$F_y = T \bullet \tan\alpha / A_m$$

$$F_z = 0$$

Face gear in case of ∑ = 90°

$$F_x = -T / A_m$$

$$F_y = 0$$

$$F_z = -T \bullet \tan\alpha / A_m$$

where:

T	Torque of observed member
A_m	Mean cone distance
Σ	Shaft angle
γ	Pitch angle
α	Pressure angle
F_x, F_y, F_z	Bearing load force components

The bearing force calculation formulas are based on the assumption that one pair of teeth transmits the torque, with one normal-force vector in the mean point of the flank pair. The results are good approximations that reflect the real bearing loads for multiple tooth meshing within an acceptable tolerance. A precise calculation is, for example, possible with the Gleason face gear design and analysis software.

Face gears can operate with regular transmission oil or, in the case of low RPM's, with a grease filling. In case of circumferential speeds above 10m/min, a sump lubrication with regular transmission oil is recommended. The oil level has to cover the facewidth of those teeth lowest in the sump. More oil causes foaming, cavitations and unnecessary energy loss. There is no requirement for any lubrication additives at normal speeds. However, in the case of circumferential speeds above 500 m/min, the higher sliding velocities—compared to straight bevel and spiral bevel gears—might require a high-pressure oil with additives. Because the two flank types in a face gear—upper and lower—are mirror images of each other, there is no preferred operating direction. This is advantageous for many industrial applications.

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(Next issue: "Beveloid and Hypoid Gears")

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Gear Measuring Machine by "NDG Method" for Gears Ranging from Miniature to Super-Large

Masatoshi Yuzaki

"This is an interesting concept and should be of interest to your readers."
—Robert E. Smith

Robert E. Smith *is president of R.E. Smith & Co. Inc., a gear consultancy in Rochester, NY. A mechanical engineer, he has more than 60 years' experience in the gear industry. He is chairman of the AGMA Calibration Committee and was AGMA's ISO delegate for that panel as well as for the AGMA Inspection and Handbook Committee. Since 1991, he has volunteered his services as a* Gear Technology *technical editor. As Bob was the technical reviewer of this article, we believe his comments regarding this paper's relevance will be of interest to readers. (The Editors)*

"This is an interesting concept and should be of interest to your readers. As the author points out, it has several advantages over the conventional TDG method. When reducing the X axis movement while checking large-diameter gears, the potential errors of probe positioning are reduced. Also, the instrument can be smaller in that direction. There are definite advantages to checking small-diameter internal gears, also. While the conventional instruments go "back to basics" and measure normal to the tooth surface at the base circle tangent, (the author's) NDG method has the probe moving in a direction that is not normal to the surface. He therefore has to make a correction to all measurements involving the cosine of the transverse pressure angle. However, that is not a problem. Gear measurements on a CMM-type instrument have to be corrected in a similar way to the surface normals. When we established our National Gear Metrology laboratory at Oak Ridge, we had them measure artifacts by the first principles (TDG) method in order to compare to CMM-type measurements that required algorithms for probe corrections. We were satisfied that the results were good and the differences were insignificant. All artifacts today are measured by the CMM instruments. In this case, the author did compare measurements by both TDG and NDG systems, on the same gear, and showed the results to be the same. It was done on a gear of approximately 13-inch diameter, but I would expect good results on much larger gears also. In fact, the error, or uncertainty of measurement, should be (even) less on larger gears."

Management Summary

A study was conducted on the development of a CNC gear measuring machine for measuring involute tooth profile by a new measurement method. Involute tooth profile measurement has been done, until now, by almost always using two-axis control in which the probe moves only in the X-axis direction synchronously with the gear rotation angle (θ). In contrast, the newly developed measurement method uses three-axis control in which the probe moves along the line of action under control in two orthogonal, axial directions (along the X and Y axes) synchronously with the gear rotation angle (θ).

This new method enables high-accuracy measurement because the small X-direction movement of the probe reduces the guaranteed accuracy range and minimizes movement of the probing head gravity center. As probe movement in the X-direction is unaffected by gear outside diameter, the advantage of the new method over earlier ones is particularly relevant to the measurement of super-large gears. While conventional measurement methods must use multiple probes to avoid probe-tooth interference in the measurement of inner gears, the new method uses fewer probes in inner gear measurement and eliminates the need for an automatic tool changer (ATC). In the case of a small-diameter inner gear (outside circle diameter of 10 mm or less), measurement of tooth profile, helix and pitch deviation can be completed with a one-time setting. A CNC gear measuring system is developed using this new measurement method that provides numerous advantages over conventional measuring systems.

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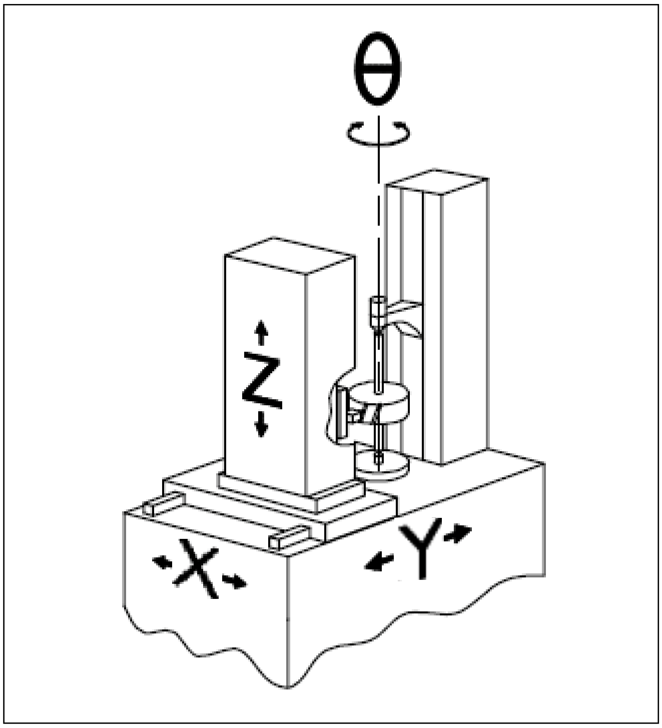

Figure 1—Coordinate system of gear measuring machine.

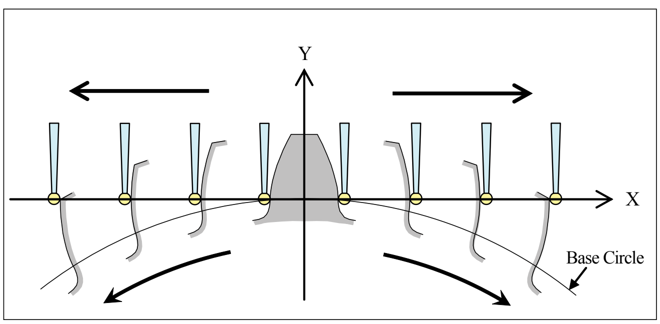

Figure 2—Typical tooth form measurement.

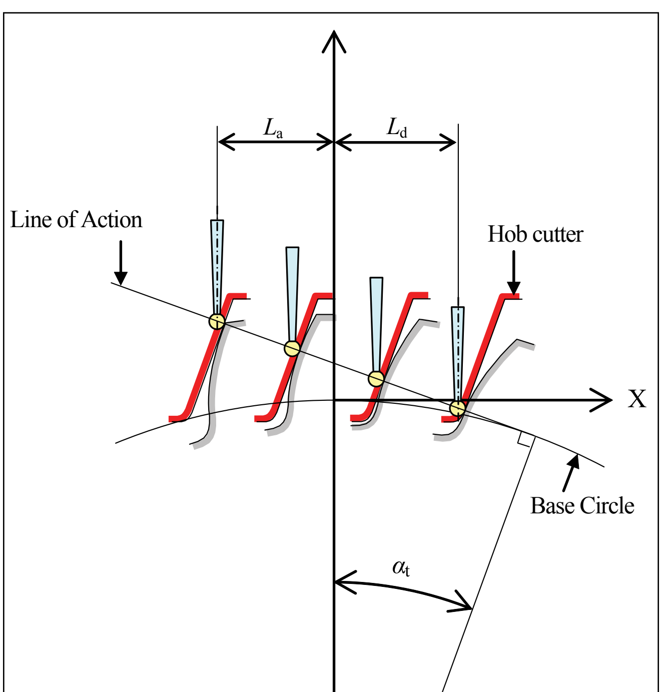

Figure 3—Tooth form measurement by NDG Method.

Introduction

Almost without exception, conventional dedicated gear measuring machines measure involute tooth profile by the two-axis control method: the probe moves only along the X axis, orthogonal to the axis of rotation synchronously with the gear rotation angle (θ) (Figs. 1–2). As it is based on the principle of involute tooth profile generation, this conventional method is quite simple and easy to understand.

However, the working positions in the hobbing machine, gear shaper, gear grinder and other tooling are almost all near the center of the gear, and these working positions are very different from the conventional tooth profile measurement positions. In the hobbing machine, the tooth cutting position of the hob cutter moves along the gear line of action while rotating synchronously with the gear blank (Fig. 3).

The author long questioned the reason for the significant difference between the working positions during gear machining and the measurement positions during gear measurement (tangent to the base circle). This current study grew out of an intuition that it should work to control the probe movement at the gear cutting position.

The author will use the term "tangential direction generate method" (TDG Method) for the ordinary measurement in which the probe moves only in the direction tangential to the base circle (X-axis direction orthogonal to the axis of rotation), and the term "normal direction generate method" (NDG Method [patent applied]), for the new measurement method in which the probe moves along the lines of action (X and Y axes orthogonal to the axis of rotation).

Normal Direction Generate Method (NDG Method)

Amount of probe movement. As seen in Figure 3, the NDG Method measures tooth profile as the probe is moved along the line of action in the same way as the hob cutter during tooth cutting. The probe moves in the direction of touching the base circle, and the principle of involute tooth profile generation is exactly the same as in the TDG Method.

Measurement of a standard gear without any profile shift (whose height of the involute tooth profile portion is one module on both the addendum side and the dedendum side) by the NDG Method gives $L_a = L_d = m/\tan\alpha_t$ (mm), where L_a and L_d are the amounts of probe X-axis direction movement from the Y axis passing through the gear center on the addendum side and the dedendum side, respectively; m is the module (mm), and αt is the transverse pressure angle. When α_t = 20°, $L_a = L_d \approx 2.7\,m$. Measurement is the same for the right and left tooth faces. Thus the amount of probe movement in the NDG Method is independent of the number of gear teeth and diameter of the reference circle.

On the contrary, the amount of X-axis direction probe movement in the TDG Method is approximately proportional to the reference circle diameter (Fig. 4) and given by:

$$L_l = L_r = 0.5z \cdot m\sqrt{(1+2/z)^2 - \cos^2\alpha_t}$$

A graph representing $(L_l + L_r)$ when m = 10 mm is shown in Figure 5. The amount of X-axis direction probe movement increases with the diameter of the reference circle. A machine capable of measuring a super-large gear like that in Figure 6 using the TDG method is therefore very difficult to build.

Measurement error. While probe movement is solely in the X-axis direction during tooth profile measurement by the TDG Method, it occurs in both the X-axis and Y-axis directions of the measuring machine in the NDG Method. The profile measurement error caused by probe movement error during measurement therefore differs between the TDG Method and the NDG Method. The nature of this difference was investigated.

In Figure 7, the probe being positioned at A indicates tooth profile measurement by the TDG Method, and the probe being positioned at B indicates tooth profile measurement by the NDG Method.

Let us assume that maximum tooth profile measurement error in the TDG Method, designated $F_{\alpha\text{-}TDG}$, can be expressed by:

$$F_{\alpha-TDG} = |e_X| + |e_L| + |\Delta\theta| r_b \qquad (1)$$

where:

$|e_X|$ is the maximum X-axis direction error of the probe arising independently of its X axis position,

$|e_L|$ is the maximum X-axis direction error of the probe arising due to large movement of the probe,

$|\Delta^\theta|$ is the maximum gear rotation angle error, and:

r_b is the radius of the gear base circle.

Let us further assume that maximum tooth profile measurement error in the NDG Method, designated $F_{\alpha\text{-}NDG}$, can be expressed by:

$$F_{\alpha-NDG} = |e_X|\cos\alpha_t + |e_Y|\sin\alpha_t + |\Delta\theta| r_b \qquad (2)$$

where:

$|e_Y|$ is the maximum Y-axis direction error of the probe.

As the amount of probe movement is small in the NDG Method, $|e_L|$ is assumed to be negligibly small.

The maximum tooth profile measurement errors by the NDG Method and TDG Method are compared by taking the difference between Equation 1 and Equation 2 to obtain Equation 3:

$$F_{\alpha-NDG} - F_{\alpha-TDG} = |e_X|(\cos\alpha_t - 1) + |e_Y|\sin\alpha_t - |e_L| \qquad (3)$$

From the fact that $|e_Y|$ is about the same size as $|e_X|$, the following expression holds:

$$F_{\alpha-NDG} - F_{\alpha-TDG} \approx |e_X|(\sin\alpha_t + \cos\alpha_t - 1) - |e_L| \qquad (4)$$

when:
$\alpha_t = 20°$,

$$F_{\alpha-NDG} - F_{\alpha-TDG} \approx 0.28|e_X| - |e_L| \qquad (5)$$

This means that the tooth profile measurement error of the TDG Method is greater than that of the NDG Method when $|e_L|$ exceeds $0.28\,|e_X|$.

To give a specific example: at $|e_X|$ of 0.2 µm, the TDG Method becomes greater in measurement error than the NDG

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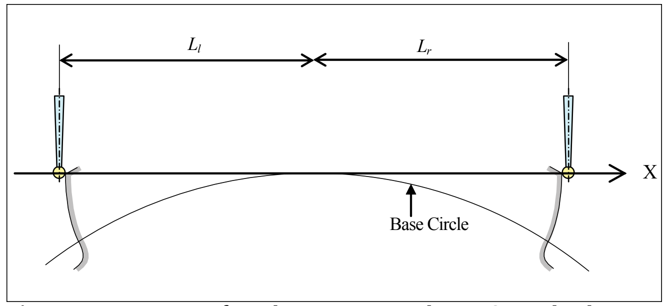

Figure 4—Amount of probe movement by TDG Method.

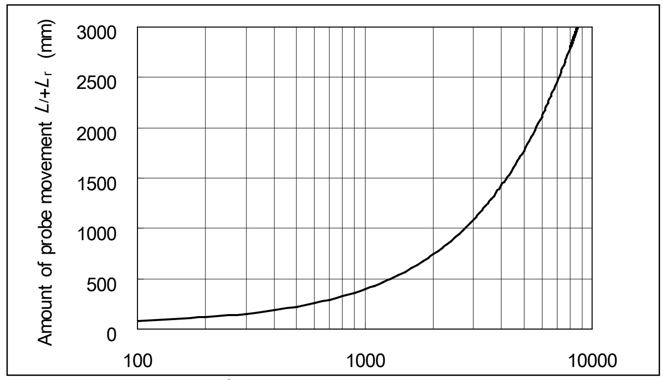

Figure 5—Amount of probe movement in X-axis direction by TDG Method.

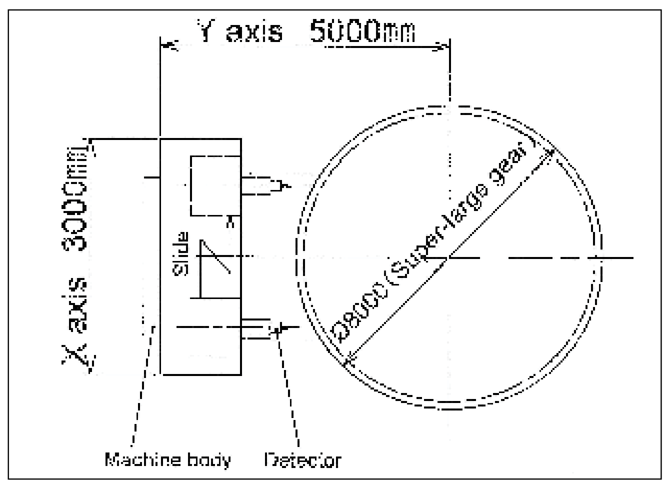

Figure 6—TDG Method measuring machine for a super-big gear.

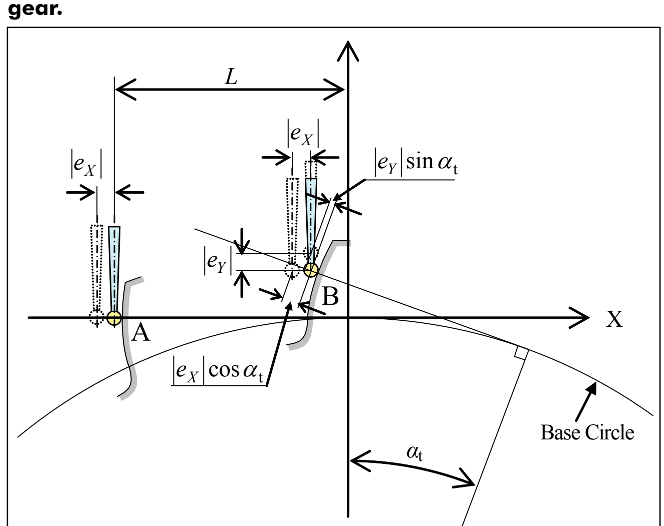

Figure 7—Measurement error by TDG Method (A) and NDG Method (B).

Table 1—Specifications of the Developed NDG Measuring Machine			
Machine	Test Mode	profile, lead, pitch for spur/helical gear, internal gear with auto alignment system	
	Size	4,300 x 1,200 x 3,400	mm
	Weight	6,000	kg
	Measurement accuracy	0.1	µm
Gear	Module	1.0 to 32	mm
	Outer diameter (max)	2,000	mm
	Face width (max)	1,500	mm
	Helix angle (max)	±65	deg
	Shaft length	150 to 2,000	mm
	Weight (max)	10,000	kg

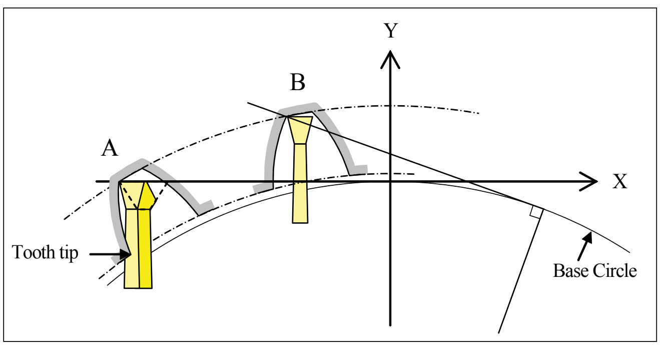

Figure 8—Measurement of inner gear by TDG-Method (A) and NDG-Method (B).

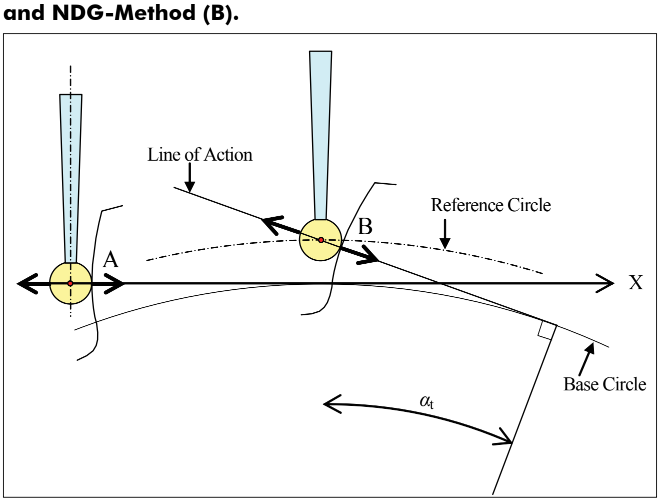

Figure 9—Measurement by sphere tip probe.

Method when $|e_L|$ becomes 0.056 mm or larger. In a machine for measuring super-large gears, it is extremely difficult to achieve a probe movement error of less than 0.056 mm. The NDG Method is therefore better than the TDG Method for measuring super-large gears.

Measurement of inner gear. When an inner gear is measured by the TGD Method, it becomes impossible to measure tooth profile, helix and pitch deviation with a one-time setting because interference arises between the tooth face and probe, thereby causing frequent interruptions.

In Figure 8, the case where the probe is positioned at A indicates measurement of inner gear tooth profile by the TDG Method. When a probe of small tip diameter is used, interference occurs between the tooth tip and the probe stem. If an attempt is made to avoid interference between the tooth tip and probe stem by enlarging the probe tip diameter, interference will then occur between the probe tip and the opposite tooth face. Although this problem can be overcome by removing one side of the probe tip, it would require the probe orientation to be reversed laterally when measuring the opposite tooth face.

On the other hand—as shown in Figure 8B—measurement by the NDG Method does not experience interference between the tooth tip and probe, no matter how much probe tip diameter is reduced. The left and right tooth faces can therefore be measured with a single, small probe. The efficacy of the NDG Method is therefore particularly evident in the measurement of small-diameter, inner gears (φ 10 mm or less).

Probe tip position. It should also be noted regarding measurement by the NDG Method that it differs from that of the TDG Method not only in the direction of probe movement, but also—depending on the probe type—in the initial probe position.

Figure 9 shows an example in which the probe has a spherical tip. In the TDG Method—indicated by A in the drawing—the center of the tip sphere is positioned on the X axis, where the measurement is performed. In the NDG Method—indicated by B in the drawing—the center of the probe tip sphere is positioned at the intercept of the reference circle and the Y axis, and measurement is performed along the line of action at pressure angle α_t.

Figure 10 shows an example in which the probe has a chisel type tip. In the TDG Method—indicated by A in the drawing—the tip of the chisel is positioned on the X axis, where the measurement is performed. In the NDG Method—indicated by B in the drawing—the axis of the probe stem is aligned with the Y axis and the chisel tip must be moved from the intercept of the reference circle and the Y axis toward the gear center by $(d_p/2)\tan\alpha_t$, where d_p is the tip diameter of the chisel type probe..

NDG Measuring Machine

A measuring machine utilizing the newly developed NDG Method is shown in Figure 11. The specifications of the developed measuring machine are provided in Table 1. The X axis direction width of the measuring machine can be slimmed down considerably, compared with one adopting the conventional system.

In the measuring machine using the NDG Method, the probe is controlled in orthogonal, two-axis (X and Y) direction at a given angle to move along the line of action. It should therefore be noted that the tooth profile error output of the probe—which has sensitivity in the X axis direction—is the cosine (cos) of the transverse pressure angle. In other words, the tooth profile error must be multiplied by the displacement output ($1/\cos\alpha_t$). And, as mentioned previously, the probe must be initially positioned so that the measurement point falls on the line of action.

Measurement Result

The dimensions of a gear for test measurement are shown in Table 2.

NDG Method measurement is performed near the gear center, analogous with the working positions in a gear manufacturing machine. This makes measurement possible in a much shorter time than by the TDG Method. A comparison of measurement times using the same developed measuring machine showed that the NDG Method achieved a time reduction of 35% for profile measurement, compared with the TDG Method.

Figure 12 and Table 3 show the results when tooth profile measurement is conducted by the NDG Method and TDG Method in the same developed measuring machine. The results show that there is no substantial difference in measure-

continued

Table 2—Gear Dimensions for Test Measurement

Module	5	mm
Number of teeth	60	
Outer Diameter	329.5	mm
Pressure Angle	20	deg
Helix Angle	20	deg
Gear Width	50	mm

Table 3—Profile Deviations by TDG-Method and NDG-Method

Unit: µm

	Left Flank			Right Flank		
	F_α	$f_{f\alpha}$	$f_{H\alpha}$	F_α	$f_{f\alpha}$	$f_{H\alpha}$
TDG	3.8	1.7	-3.4	2.4	0.7	-2.4
NDG	3.8	1.7	-3.4	2.4	0.7	-2.4

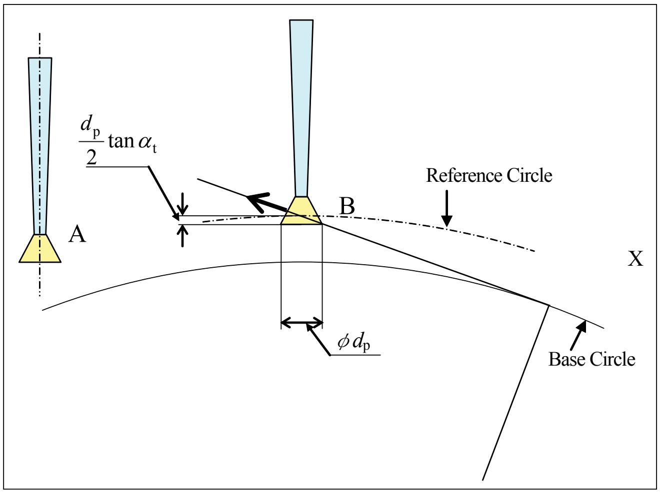

Figure 10—Measurement by chisel tip probe.

Figure 11—Developed NDG measuring machine.

ment accuracy between the NDG Method and TDG Method.

Conclusion

The following are some typical problems that arise when tooth profile is measured by the conventional TDG Method, in which the probe moves only in the X axis direction:

- High-accuracy measurement cannot be anticipated in a measuring machine for large gears because the large movement of the probing head expands the guaranteed accuracy range and increases movement of the probing head gravity center.
- The X axis direction movement of the probe is proportional to the gear reference circle diameter, making large-gear measurement time consuming.
- When an inner gear is measured, it is often impossible to measure tooth profile, helix and pitch deviation with a one-time setting because interference arises between the tooth face and probe, thus necessitating frequent interruptions.
- Numerous probes matched to different gear sizes are necessary. An automatic tool changer is therefore more often required, and probe calibration work increases.
- Measurement of a small-module inner gear (minimum outside circle diameter of 10 mm or less) is difficult.

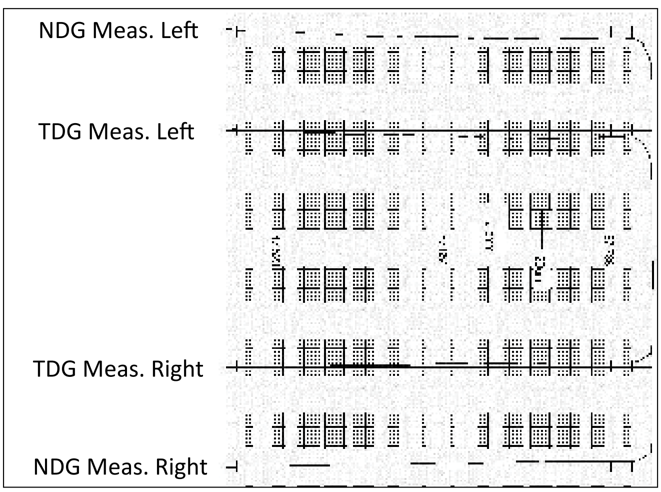

Figure 12—Profile measurement results by TDG Method and NDG Method.

- High-accuracy tooth profile management is hard to achieve in the measurement of dies and molds used for plastics, sintered metals, forgings and the like.

In contrast, advantages such as the following are obtained when tooth profile measurement is performed by the NDG Method, in which the probe moves along the line of action under control in two orthogonal, axial directions (along the X and Y axes):

- High-accuracy measurement is possible, even in a measuring machine for large gears, because the small X axis direction movement of the probe reduces the guaranteed accuracy range and minimizes movement of the probing head gravity center.
- As measurement is conducted near the gear center, even a large gear with a large reference circle diameter can be measured in a relatively short time.
- In inner gear measurement, the fact that no interference arises between the gear and the probe reduces the number of probes required, eliminates the need for an automatic tool changer, and minimizes probe calibration work.
- Measurement of tooth profile, helix and pitch deviation can be completed with a one-time setting, even in the case of a small-diameter inner gear (outside circle diameter ϕ of 10 mm or less).

A program enabling NDG Method measurement can be incorporated into an existing CNC gear measuring machine.

When an NDG Method measuring machine was actually built, it was found that the X-axis direction width of the machine could be made slimmer than a conventional one. In addition, the tooth profile measurement results were found to be substantially no different in accuracy from those by the conventional system. The NDG Method was incorporated into a conventional measuring machine and was confirmed to be a tooth profile measuring method applicable to gears ranging widely in size—from super-large to miniature.

(*In closing, the author wishes to express heartfelt gratitude to the team members who built the measuring machine.*)

Masatoshi Yuzaki *is president of Tokyo Technical Instruments, a company he founded in 1972 as a manufacturer of gear measuring machines and instruments. Yuzaki's continuing commitment to the development of innovative products that contribute to gear quality improvement worldwide is evidenced by his company's many and diverse offerings matched to ever-changing global measurement needs. Applications have been filed for international patent protection of the NDG Method measuring machine. Yuzaki is a member of the Measurement Committee of the Japan Gear Manufacturers Association (JGMA) and a member of the Japan Society of Mechanical Engineers (JSME).*

Optimal Modifications of Gear Tooth Surfaces

Vilmos Simon

(Proceedings of the 3rd International Conference on Power Transmission 2009. October 1–2, 2009, Chalkidiki, Greece.)

Management Summary

In this paper a new method for the introduction of optimal modifications into gear tooth surfaces—based on the optimal corrections of the profile and diameter of the head cutter, and optimal variation of machine tool settings for pinion and gear finishing—is presented. The goal of these tooth modifications is the achievement of a more favorable load distribution and reduced transmission error. The method is applied to face milled and face hobbed hypoid gears.

Corresponding computer programs are developed. By using these programs, the optimal head cutter geometry and the optimal machine tool settings for pinion and gear tooth processing are determined. The influence of tooth errors and misalignments of the mating members on load distribution and transmission error is investigated. The obtained results show that the influence of tooth errors and misalignments on gear performances is significantly reduced by the introduction of optimal tooth modifications.

Introduction

Conjugated spur, helical, spiral bevel, hypoid and worm gears are theoretically in line contact. In order to decrease the sensitivity of the gear pair to errors in tooth surfaces and to the mutual position of the mating members, carefully chosen modifications are usually introduced into the teeth of one or both members. Over the years, much research has been directed towards the optimization of design and manufacture of different types of gears. Some of the related references are for:

- Spur and helical gears (Refs. 1–3)
- Spiral bevel and hypoid gears (Refs. 4–27)
- Cylindrical worm gears (Refs. 28–35)
- Double-enveloping worm gears (Refs. 36–40)

As a result of these modifications, the gear pair becomes "mismatched," and point contact of the meshed tooth surfaces appears instead of line contact. In practice, these modifications are usually introduced by applying the appropriate machine tool setting for pinion and gear manufacture, or by the optimization of the tool geometry.

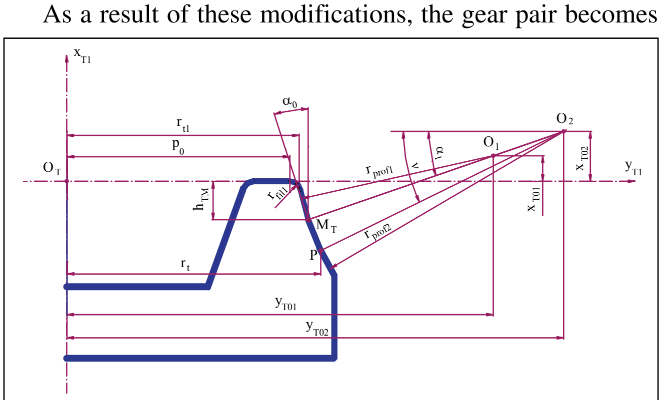

Figure 1—Bi-circular tool profile for pinion tooth finishing.

The method for the determination of optimal tooth modifications—based on minimized tooth contact pressure and transmission error—is presented in the example of face milled and face hobbed hypoid gears.

Theoretical Background

The major differences between the face milling and face hobbing process are:

- In face hobbing, a timed continuous indexing is provided, while in face-milling, the indexing is intermittently provided after cutting each tooth side or slot.
- The lengthwise tooth curve of face milled hypoid gears is a circular arc with a curvature radius equal to the cutter radius, while the lengthwise tooth curve of face hobbed gears is an epicycloid that is kinematically generated by the indexing motion.
- Face hobbing gear design uses the uniform tooth depth system while face milling gear design uses the tapered tooth depth system; i.e.—the pinion is the driving member. The convex side of the gear tooth and the mating concave side of the pinion tooth are the drive sides.

The Generation of Tooth Surfaces in Face Milled Hypoid Gears

A Gleason-type hypoid gear pair—with generated pinion and non-generated gear—is treated. The optimal tooth modifications are introduced into the pinion tooth surface by using a cutter with bi-circular profile (Fig. 1), optimal tool diameter and appropriate machine tool settings.

The tool surface, with bi-circular profile and for the generation of the driving side of pinion teeth, is in the coordinate system Kr_1 (attached to tool T_1), defined by the following equation (and Fig. 1):

continued

Nomenclature

c, c_i	=	Sliding base setting, mm
d	=	Basic machine center to back increment setting for gear finishing, mm
e	=	Basic radial setting, mm
f, f_i	=	Basic machine center setting, mm
g, g_i	=	Basic offset setting, mm
h	=	Basic horizontal setting for gear finishing, mm
h_d	=	Distance from tilt center to machine plane, mm
h_{TM}	=	Position of cutter radii's connection point, mm
i_c	=	Indexing ratio
i_g, i_{gi}	=	Velocity ratios in the kinematic scheme of the machine tool for the generation of pinion and gear tooth surfaces
N_b	=	Number of head cutter's blade groups
N_c	=	Number of teeth of the imaginary generating crown gear
N_w	=	Number of teeth of the workpiece
N_1, N_2	=	Number of pinion and gear teeth
p_{max}	=	Maximum tooth contact pressure, N/cm^2
r_{profi}	=	Radius of circular tool profile, mm
r_{t1}	=	Pinion finishing cutter radius, mm
r_{t2}	=	Gear finishing cutter radius, mm
s	=	Tooth thickness, mm
u, t	=	Tool surface variables, mm
v	=	Basic vertical setting for gear finishing, mm
x_{epi}, z_{epi}	=	Coordinates of the center of the circular blade profile, mm
x_{T0i}, y_{T0i}	=	Coordinates of the center of circular arc tool profile, mm
z_{eM}	=	Position of blade profile radii's connection point, mm
α	=	Profile angle of the straight-lined cutting edge, deg
α_1	=	Pinion finishing tool profile angle, deg
α_2	=	Gear finishing tool profile angle, deg
β	=	Basic tilt angle setting for pinion finishing, deg
β_F	=	Load distribution factor
$\Delta\phi_2$	=	Angular displacement of the driven gear member, arc second
$\Delta\phi_{2s}$	=	Angular displacement of the driven gear member due to edge contact, arc second
$\Delta\phi_2^{(d)}$	=	Dynamic angular displacement of the driven gear member, arc second
$\Delta\phi_2^{(k)}$	=	Kinematic angular displacement of the driven gear member, arc second
δ	=	Basic swivel angle setting for pinion finishing, deg
ε_h	=	Angular misalignment of pinion axis, deg
ϕ_1, ϕ_2	=	Rotation angles of the pinion and gear, respectively, in mesh, deg
ϕ_{10}, ϕ_{20}	=	Initial rotation angles of the pinion and gear, respectively, in mesh, deg
γ_i, γ_0	=	Initial setting angle of head cutter axis, deg
γ_1	=	Machine root setting angle for pinion finishing, deg
γ_2	=	Machine root angle setting for gear finishing, deg
η_i	=	Initial setting angle of head cutter, deg
ν	=	Parameter of circular cutting edge, deg
κ	=	Tilt angle of cutter spindle with respect to cradle rotation angle, deg
μ	=	Swivel angle of cutter tilt, deg
$\omega^{(c)}$	=	Angular velocity of the imaginary generating crown gear, deg
$\omega^{(r)}$	=	Angular velocity of the tool in pinion teeth generation, sec^{-1}
$\omega^{(T)}$	=	Angular velocity of the head cutter, sec^{-1}
$\omega^{(w)}$	=	Angular velocity of the workpiece, sec^{-1}
θ	=	Tool surface variable, mm
ρ_c	=	Radius of the rolling circle of the imaginary generating crown gear, mm
ρ_t	=	Radius of the rolling circle of the head cutter, mm
ξ	=	Tool surface variable, mm
ξ_i	=	Offset angle of cutter blade, deg
ψ	=	Cradle rotation angle, deg
ψ_i	=	Rotation angles of the pinion and the gear in the generation of tooth flanks, deg
ψ_c	=	Rotation angle of the imaginary generating crown gear in its generation, deg
ψ_{ci}	=	Rotation angle of the imaginary generating crown gear in pinion gear tooth generation, deg
ψ_t	=	Rotation angle of head cutter, deg

$$\vec{r}_{T1}^{(T_1)}(\nu,\theta)=\begin{bmatrix} x_{T0i}-r_{profi}\cdot\sin\nu \\ \left(y_{T0i}-r_{profi}\cdot\cos\nu\right)\cdot\cos\theta \\ \left(y_{T0i}-r_{profi}\cdot\cos\nu\right)\cdot\sin\theta \\ 1 \end{bmatrix} \quad (1)$$

where: $i=1, 2$—for pinion and gear, respectively.

In the case of straight-sided profile (Fig. 2), the equation of tool surface is as follows:

$$\vec{r}_{T_1}^{(T_1)}(u,\theta)=\begin{bmatrix} -u \\ (r_{t1}+u\cdot tg\alpha_1)\cdot\cos\theta \\ (r_{t1}+u\cdot tg\alpha_1)\cdot\sin\theta \\ 1 \end{bmatrix} \quad (2)$$

The generated tooth surface of the pinion (Fig. 2) is defined by the system of equations:

$$\begin{gathered}\vec{r}_1^{(1)}=\boldsymbol{M}_{p4}\cdot\boldsymbol{M}_{p3}\cdot\boldsymbol{M}_{p2}\cdot\boldsymbol{M}_{p1}\cdot\vec{r}_{T1}^{(T1)} \\ \vec{v}_0^{(T1,1)}\cdot\vec{e}_0^{(T1)}=0\end{gathered} \quad (3)$$

where: $\vec{v}_0^{(T1,1)}$ is the relative velocity vector of the tool T_1 to the pinion, and $\vec{e}_0^{(T1)}$ is the unit normal vector of the tool surface.

On the basis of Figure 2 and Equations 1 and 2, for the relative velocity vector and for the unit normal vector of the tool surface, it follows that:

$$\vec{v}_0^{(T1,1)}=\omega^{(T)}\cdot\begin{bmatrix} i_g\cdot\left(z_0^{(T1)}+g\right)\cdot\cos\gamma_1 \\ z_0^{(T1)}-i_g\cdot\left(z_0^{(T1)}+g\right)\cdot\sin\gamma_1 \\ i_g\cdot\left[y_0^{(T1)}\cdot\sin\gamma_1-\left(x_0^{(T1)}-c\right)\cdot\cos\gamma_1\right]-y_0^{(T1)} \end{bmatrix} \quad (4)$$

$$\vec{e}_0^{(T1)}=\boldsymbol{M}_{p2}\cdot\boldsymbol{M}_{p1}\cdot\vec{e}_{T1}^{(T1)}=\boldsymbol{M}_{p2}\cdot\boldsymbol{M}_{p1}\begin{bmatrix} \sin q \\ \cos q\cdot\cos\theta \\ \cos q\cdot\sin\theta \\ 0 \end{bmatrix} \quad (5)$$

where: $q=\alpha_1$ is in the case of straight-sided tool profile; $q=\nu$ in the case of bi-circular profile and:

$$\vec{r}_0^{(T1)}=\boldsymbol{M}_{p2}\cdot\boldsymbol{M}_{p1}\cdot\vec{r}_{T1}^{(T1)} \quad (6)$$

Matrices $\boldsymbol{M}_{p1}-\boldsymbol{M}_{p4}$ provide the coordinate transformations from the coordinate system K_{T1} (attached to the tool) into the system K_1 (attached to the pinion). The coordinate transformations are defined by the following equations (Fig. 2):

$$\vec{r}_{T0}=M_{p1}\cdot\vec{r}_{T1}=\begin{bmatrix} \cos\beta & -\sin\beta & 0 & 0 \\ \cos\delta\cdot\sin\beta & \cos\delta\cdot\cos\beta & -\sin\delta & 0 \\ \sin\delta\cdot\sin\beta & \sin\delta\cdot\cos\beta & \cos\delta & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}\cdot\vec{r}_{T1} \quad (7)$$

$$\vec{r}_0=M_{p2}\cdot\vec{r}_{T0}=\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \sin\psi & -\cos\psi & e\cdot\cos\psi \\ 0 & \cos\psi & \sin\psi & -e\cdot\sin\psi \\ 0 & 0 & 0 & 1 \end{bmatrix}\cdot\vec{r}_{T0} \quad (8)$$

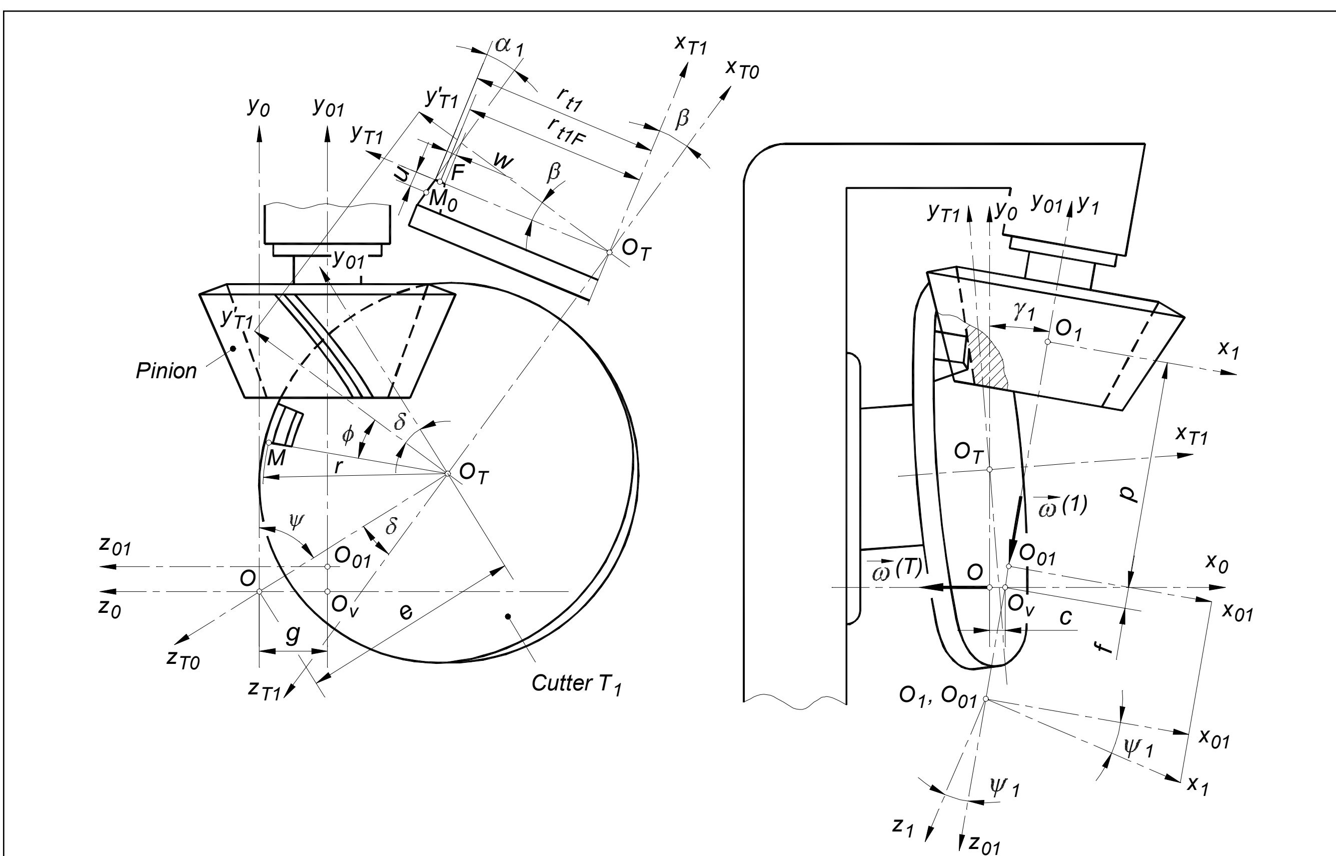

Figure 2—Machine tool setting for pinion tooth finishing.

$$\vec{r}_{01} = M_{p3} \cdot \vec{r}_0 = \begin{bmatrix} \cos\gamma_1 & -\sin\gamma_1 & 0 & -c\cdot\cos\gamma_1 \\ \sin\gamma_1 & \cos\gamma_1 & 0 & -f-c\cdot\sin\gamma_1 \\ 0 & 0 & 1 & g \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_0 \quad (9)$$

$$\vec{r}_1 = M_{p4} \cdot \vec{r}_{01} = \begin{bmatrix} \cos\psi_1 & 0 & \sin\psi_1 & 0 \\ 0 & 1 & 0 & -p \\ -\sin\psi_1 & 0 & \cos\psi_1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_{01} \quad (10)$$

The surface of the tool used for gear tooth processing is in the coordinate system K_{T2} (attached to tool T_2), defined by the equation (Fig. 3.):

$$\vec{r}_{T2}^{\,(T2)}(t,\xi) = \begin{bmatrix} -t \\ (r_{t2} + t\cdot tg\alpha_2)\cdot\cos\xi \\ (r_{t2} + t\cdot tg\alpha_2)\cdot\sin\xi \\ 1 \end{bmatrix} \quad (11)$$

The position vector of the formed gear tooth surface points is obtained by simple coordinate transformation of vector (Eq. 11) from system K_{T2} into the system K_2 (attached to the gear), as follows:

$$\vec{r}_2^{\,(2)} = \boldsymbol{M}_{g1} \cdot \vec{r}_{T2}^{\,(T2)} \quad (12)$$

The matrix of this coordinate transformation is given by the expression (Fig. 3):

$$M_{g1} = \begin{bmatrix} \sin\gamma_2 & -\cos\gamma_2 & 0 & d-h\cdot\cos\gamma_2 \\ \cos\gamma_2 & \sin\gamma_2 & 0 & h\cdot\sin\gamma_2 \\ 0 & 0 & 1 & -v \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (13)$$

The Generation of Tooth Surfaces in Hobbed Hypoid Gears

In this paper the face hobbing process is based on the generalized concept of hypoid gear generation in which the mating pinion and gear can be considered respectively, generated by the imaginary generating crown gear shown in Figure 4. This imaginary generating gear is a virtual gear whose teeth are formed by the traces of the cutting edges of the head cutter blades (Fig. 5), although its tooth number is not necessarily an integer. Rather, it can be considered as a special case of a hypoid gear with a 90° pitch angle.

In the face hobbing process, two independent motions—timed continuous indexing and generating motion—are superimposed. The indexing motion between the tool and the imaginary generating gear forms the tooth surface of the generating gear with the extended epicycloid lengthwise tooth curve (Fig. 4). The indexing relationship can be expressed as:

$$\frac{\omega^{(t)}}{\omega^{(c)}} = \frac{N_c}{N_b} = i_c \quad (14)$$

where: $\omega^{(t)}$ and $\omega^{(c)}$ denote the angular velocities of the head cutter and the imaginary generating gear N_b and N_c, the number of the head cutter's blade groups, and the number of teeth of the imaginary generating gear, respectively.

The second related motion is the rolling motion of the generating gear and the rotation of the workpiece-generated gear (Fig. 4). It can be represented as:

$$\frac{\omega^{(w)}}{\omega^{(c)}} = \frac{N_c}{N_w} = i_g \quad (15)$$

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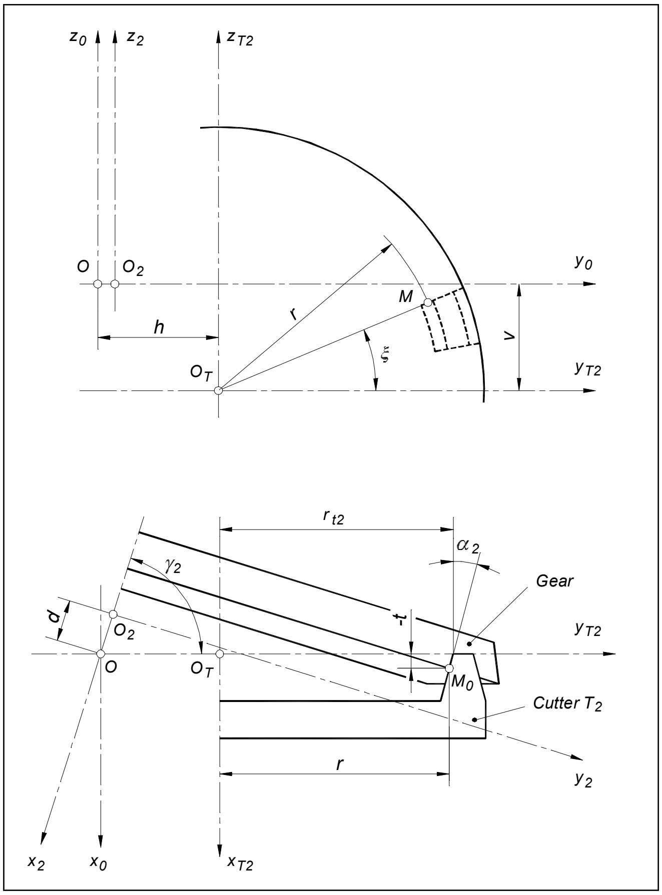

Figure 3—Machine tool setting for gear tooth finishing.

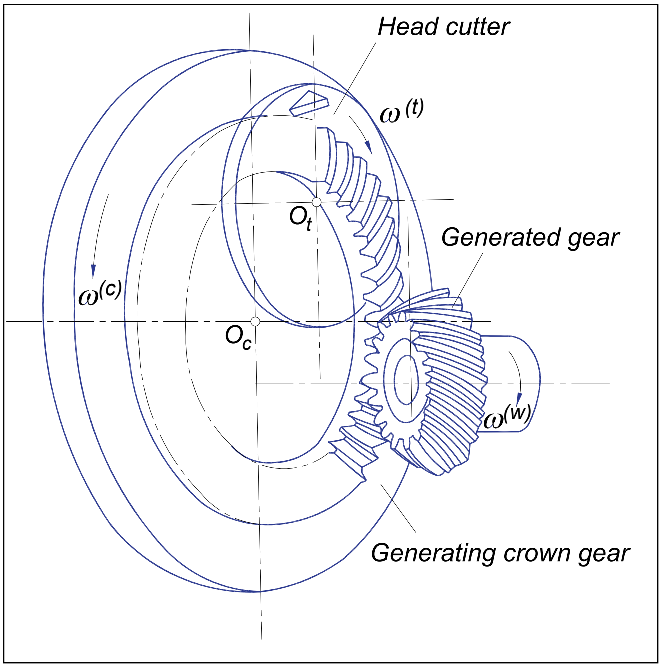

Figure 4—Concept of hypoid gear hobbing.

where: $\omega^{(w)}$ and N_w denote the angular velocity of the workpiece and the tooth number of the workpiece, respectively; and i_g is the ratio of roll in the generation of pinion and gear tooth surfaces.

Figure 5 shows the relative position between the head cutter and the imaginary generating crown gear. The relative position for two face hobbing methods for the implementation of the longitudinal tooth crowning of tooth flanks is presented:

- In the first system, two separate rotation centers with non-tilting cutter head cutter is applied to achieve lengthwise crowning.
- In the second method, cutter tilt is used to modify the curvatures of tooth surfaces.

In the generation process of the imaginary crown gear, the head cutter is rolling with its roll circle of radius ρ_t on the base circle of the imaginary crown gear of radius ρ_c without slippage. This rolling is realized by the rotations of the head cutter with angular velocity $\omega^{(t)}$ and of the crown gear with angular velocity $\omega^{(c)}$ around their axes in different directions (Fig. 5). The radii of rolling circles of the head cutter and of the generating crown gear are determined, respectively, by:

$$\rho_t = \frac{N_b}{N_b + N_c} \cdot e \qquad (16)$$

$$\rho_c = \frac{N_c}{N_b + N_c} \cdot e \qquad (17)$$

where: e is the radial machine setting.

In the system with non-tilting head cutter, the axes of the imaginary generating crown gear and of the head cutter are parallel. In the hypoid gear pair, where the convex gear flank rolls with the concave pinion flank, these are the "drive sides." The other pair of flanks is the "coast side." In Figure 5, the inside blade generates the convex gear tooth flank and the outer blade generates the concave pinion tooth flank. The lengthwise curvature of the concave surface is modified by increasing the radius of the outer head cutter. In addition, the eccentricity Δ and orientation angle $\Delta\psi$ of the outer rotation center O_{t0} to inner rotation center O_{ti} are used to control tooth thickness and contact position. The inside and outer blades are rigidly connected to the rolling circles with centers O_{ti} and O_{t0}, respectively. The effective cutting direction of the blades in the head cutter is not perpendicular to the cutter radius vector. The blades are moved in the head cutter tangentially to an offset position to accommodate the correct orientation with respect to the cutting motion vector. Therefore, the normal to the loci of blades at the middle point of tooth profiles, M_0, has to pass through instantaneous centers of rotation O_i and O_o simultaneously. The profile of blades in the plane (y_e, z_e) of coordinate system $K_e(x_e$, y_e and $z_e)$ may be a straight-lined or a circular arc in order to introduce tooth profile modification (Fig. 6).

Equation of the straight lined profile (based on Fig. 6a):

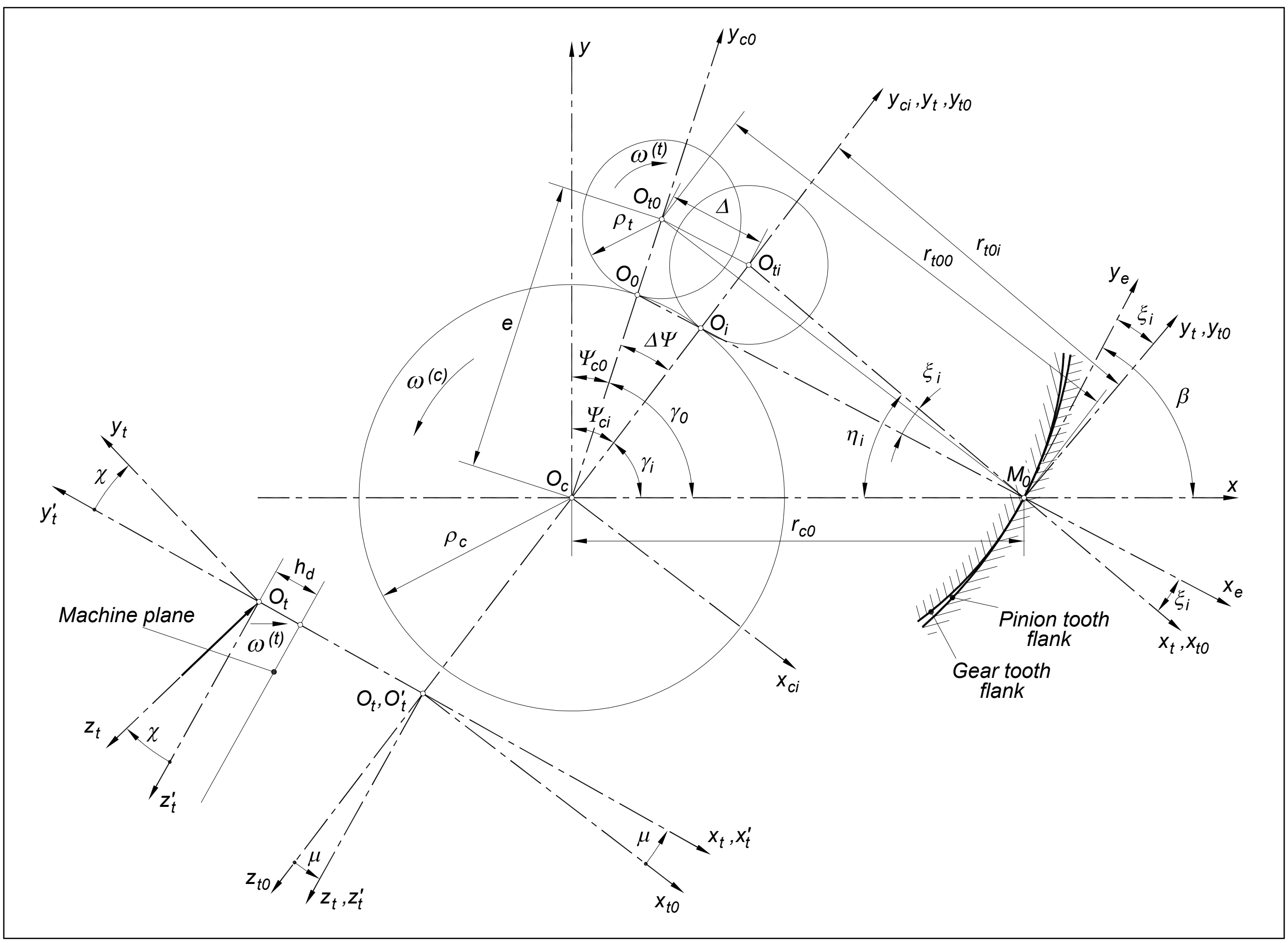

Figure 5—Relative position between the head cutter and the imaginary generating crown gear.

$$\vec{r}_e^{(e)}(u) = \begin{bmatrix} sign \cdot \left(\frac{s}{2} + u \cdot tg\alpha\right) \\ 0 \\ u \\ 1 \end{bmatrix} \tag{18}$$

Equation of the circular arc profile (based on Fig. 6b):

$$\vec{r}_e^{(e)}(\nu) = \begin{bmatrix} sign \cdot \left(x_{epi} + r_{profi} \cdot cos\,\nu\right) \\ 0 \\ z_{epi} - r_{profi} \cdot sin\,\nu \\ 1 \end{bmatrix} \tag{19}$$

where: *sign* = 1 for the convex tooth flank of the pinion and concave tooth flank of the gear; and *sign* = –1 for the concave tooth flank of the pinion and convex tooth flank of the gear member.

The tooth surface of the imaginary generating crown gear produced by coordinate transformation from coordinate system $K_e(x_e, y_e$ and $z_e)$—rigidly connected to the head cutter—to coordinate system $K_c(x_c, y_c$ and $z_c)$—connected to the imaginary generating crown gear—is represented by the following matrix equation (Figs. 5 and 7):

$$\vec{r}_c = M_{c4} \cdot M_{c3} \cdot M_{c2} \cdot M_{c1} \cdot \vec{r}_e = M_{ec} \cdot \vec{r}_e \tag{20}$$

The coordinate transformations between the main coordinate systems $K_e(x_e, y_e$ and $z_e)$, $K_c(x_c, y_c$ and $z_c)$ and the auxiliary coordinate systems (Figs. 5 and 7), are performed as follows:

$$\vec{r}_t = M_{c1} \cdot \vec{r}_e^{(e)} = \begin{bmatrix} cos\,\xi_i & -sin\,\xi_i & 0 & r_{t0} \\ sin\,\xi_i & cos\,\xi_i & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_e^{(e)} \tag{21}$$

$$\vec{r}_{t0} = M_{c2} \cdot \vec{r}_t = \begin{bmatrix} cos\,\mu & -sin\,\mu \cdot sin\,\kappa & sin\,\mu \cdot cos\,\kappa & 0 \\ 0 & cos\,\kappa & sin\,\kappa & h_d \\ -sin\,\mu & -cos\,\mu \cdot sin\,\kappa & cos\,\mu \cdot cos\,\kappa & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_t \tag{22}$$

$$\vec{r} = M_{c3} \cdot \vec{r}_{t0} = \begin{bmatrix} cos(\eta_i + \psi_t) & sin(\eta_i + \psi_t) & 0 & e \cdot cos\,\gamma_i \\ -sin(\eta_i + \psi_t) & cos(\eta_i + \psi_t) & 0 & e \cdot sin\,\gamma_i \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_{t0} \tag{23}$$

$$\vec{r}_c = M_{c4} \cdot \vec{r} = \begin{bmatrix} cos\,\psi_c & sin\,\psi_c & 0 & 0 \\ -sin\,\psi_c & cos\,\psi_c & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r} \tag{24}$$

In the case of non-tilting head cutter:

$$\vec{r}_{t0} = M_{c2} \cdot \vec{r}_t = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & h_d \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_t \tag{25}$$

Due to the indexing motion: $\psi_t = i_c \cdot (\psi_c - \psi_{c0})$; where: $\psi_c = -(90^0 - \gamma_1)$, or $\psi_{c0} = -(90^0 - \gamma_0)$.

To obtain the pinion/gear tooth surface in the generating process, the work gear is rolled with the imaginary crown gear. Figure 8 describes the coordinate systems between the imaginary generating crown gear and the work gears:

- Coordinate system $K_c(x_c, y_c$ and $z_c)$ is rigidly connected to the generating crown gear;
- Coordinate systems $K_1(x_1, y_1$ and $z_1)$ and $K_2(x_2, y_2$ and $z_2)$ are rigidly connected to the pinion and the gear, respectively.

continued

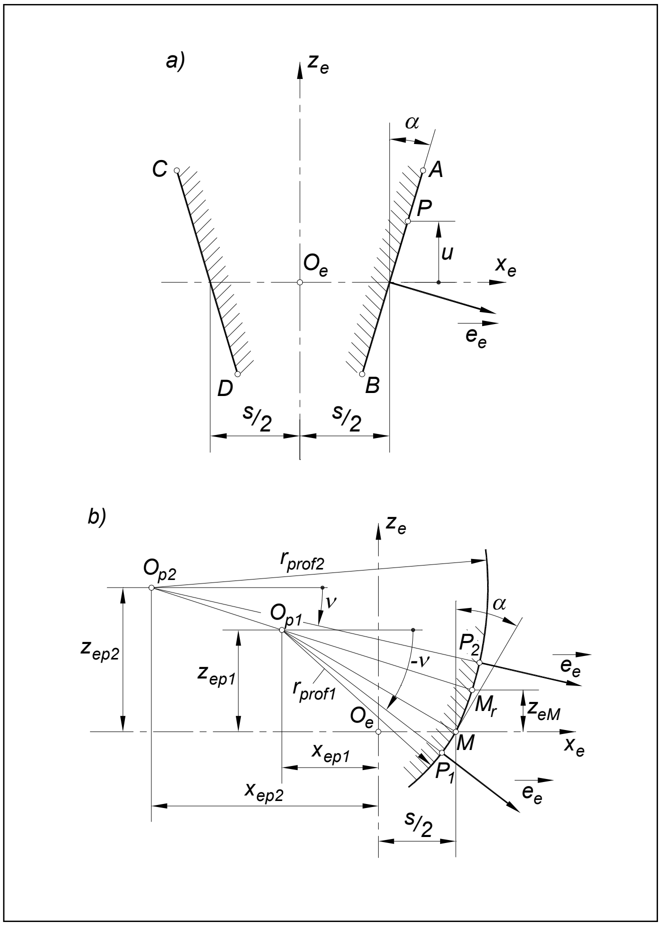

Figure 6—Blade profiles.

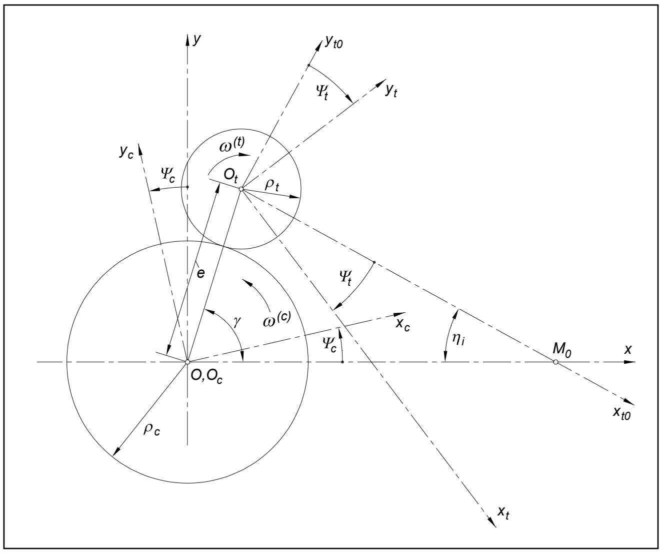

Figure 7—Rolling of head cutter and imaginary generating crown gear.

The tooth surfaces of the pinion and gear are defined by the following system of equations:

$$\vec{r}_i^{(i)} = M_{i3} \cdot M_{i2} \cdot M_{i1} \cdot \vec{r}_c^{(i)} = M_{i3} \cdot M_{i2} \cdot M_{i1} \cdot M_{ec} \cdot \vec{r}_e^{(i)} \quad (26a)$$

$$\vec{v}_{c0}^{(c,i)} \cdot \vec{e}_{c0}^{(i)} = 0 \quad (26b)$$

The coordinate transformations between the main coordinate systems $K_c\ (x_c, y_c, z_c)$, $K_1(x_1, y_1, z_1)$, $K2\ (x_2, y_2, z_2)$ and the auxiliary coordinate systems (Fig. 8), are performed as follows:

$$\vec{r}_{c0}^{(i)} = M_{i1} \cdot \vec{r}_c^{(i)} = \begin{bmatrix} \cos\psi_{ci} & -\sin\psi_{ci} & 0 & 0 \\ \sin\psi_{ci} & \cos\psi_{ci} & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_c^{(i)} \quad (27)$$

$$\vec{r}_{i0}^{(i)} = M_{i2} \cdot \vec{r}_{c0}^{(i)} = \begin{bmatrix} -\sin\gamma_i & 0 & \cos\gamma_i & -c_i \cdot \cos\gamma_i \\ \cos\gamma_i & 0 & \sin\gamma_i & -c_i \cdot \sin\gamma_i - f_i \\ 0 & 1 & 0 & g_i \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_{c0}^{(i)} \quad (28)$$

$$\vec{r}_i^{(i)} = M_{i3} \cdot \vec{r}_{i0}^{(i)} = \begin{bmatrix} \cos\psi_i & 0 & \sin\psi_i & 0 \\ 0 & 1 & 0 & 0 \\ -\sin\psi_i & 0 & \cos\psi_i & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \cdot \vec{r}_{i0}^{(i)} \quad (29)$$

$$\vec{v}_{c0}^{(c,i)} = \omega^{(c)} \cdot \begin{bmatrix} y_{c0}^{(i)} - i_{gi} \cdot \left(y_{c0}^{(i)} + g_i\right) \cdot \sin\gamma_i \\ -x_{c0}^{(i)} + i_{gi} \cdot \left[x_{c0}^{(i)} \cdot \sin\gamma_i - \left(z_{c0}^{(i)} - c_i\right) \cdot \cos\gamma_i\right] \\ i_{gi} \cdot \left(y_{c0}^{(i)} + g_i\right) \cdot \cos\gamma_i \end{bmatrix} \quad (30)$$

where: the velocity ratio in the kinematic scheme of the machine tool for the generation of pinion and gear tooth surfaces is:

$$i_{gi} = \frac{\omega^{(i)}}{\omega^{(c)}} = \frac{N_c}{N_i} \quad (31)$$

The normal surface vector of the imaginary generating crown gear for pinion and gear tooth-surface generation is:

$$\vec{e}_{c0}^{(i)} = M_{i1} \cdot \vec{e}_c^{(i)} = M_{i1} \cdot M_{ec} \cdot \vec{e}_e^{(i)} \quad (32)$$

Loaded Tooth Contact Analysis

In order to determine maximum tooth contact pressure and transmission error, the new method of load distribution calculation is applied (Ref. 22).

The load distribution calculation is based on the conditions that the total angular position errors of the gear teeth being instantaneously in contact under load must be the same, and along the contact line (contact area) of every tooth pair instantaneously in contact, the composite displacements of tooth-surface points—as the sums of tooth deformations, tooth surface separations, misalignments, and composite tooth error— should correspond to the angular position of the gear member. Therefore, in all the points of the instantaneous contact lines, the following displacement compatibility equation should be satisfied:

$$\Delta\phi_2 = \Delta\phi_2^{(d)} + \Delta\phi_2^{(k)} = \frac{\Delta y_n}{r_D} \cdot \frac{\left|\left(\vec{r} \times \vec{a}_0\right) \cdot \vec{e}\right|}{\left|\vec{r}\right|} + \Delta\phi_2^{(k)} \quad (33)$$

where: Δy_n is the composite displacement of contacting surfaces in the direction of the unit tooth surface normal $\vec{e}$, $\vec{r}$ is the position vector of the contact point, r_D is the distance of the contact point to the gear axis, and $\vec{a}_o$ is the unit vector of the gear axis.

The composite displacement of the contacting surfaces in contact point D, in the direction of the tooth-surface normal, can be expressed as:

$$\Delta y_n = w(z_D) + s(z_D) + e_n(z_D) \quad (34)$$

where: z_D is the coordinate of point D along the contact line, $w(z_D)$ is the total deflection in point D, $s(z_D)$ is the relative geometrical separation of tooth-surfaces in point D, and $e_n(z_D)$ is the composite error in point D, which is the sum of manufacturing and alignment errors of pinion and gear.

The total deflection in point D is defined by the following equation:

$$w(z_D) = \int_{L_{it}} K_d(z_D, z_F) \cdot p(z_F) \cdot dz + K_c(z_D) \cdot p(z_D) \quad (35)$$

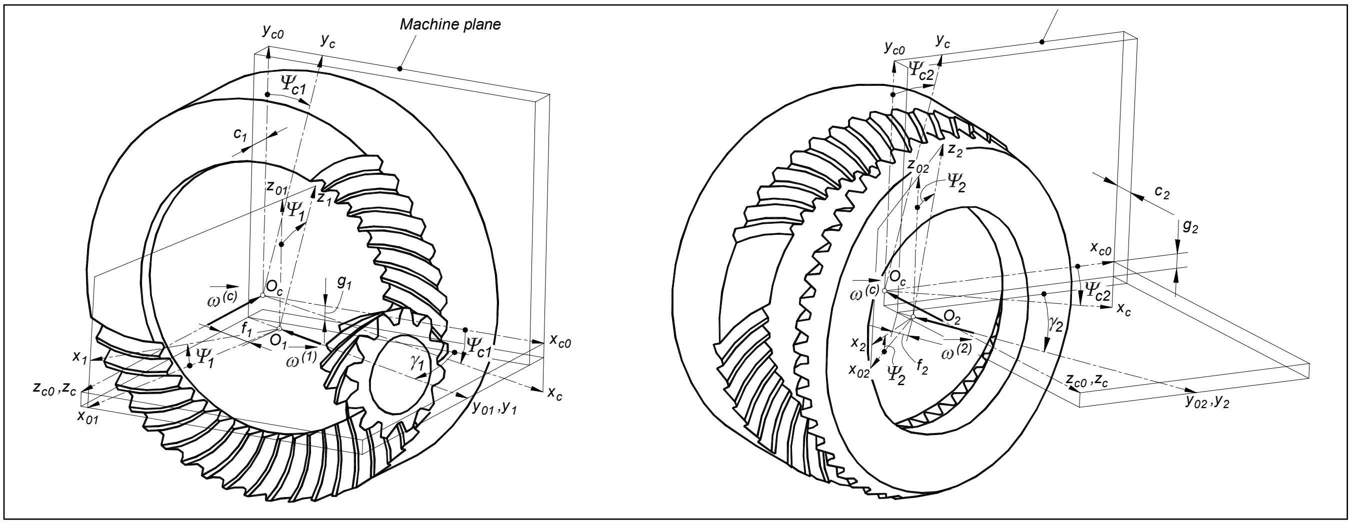

Figure 8—Generation of pinion and gear tooth surfaces.

where: L_{it} is the geometrical length of the line of contact on tooth pair i_t, $K_d(z_D, z_F)$ is the influence factor of tooth load acting in tooth-surface point F on total composite deflection of pinion and gear teeth in contact point D. K_d includes the bending and shearing deflections of pinion and gear teeth, pinion and gear body bending and torsion, and deflections of supporting shafts. A finite element computer program is developed for the calculation of bending and shearing deflections in the pinion and in the gear. K_c (z_D) is the influence factor for the contact approach between contacting pinion and gear teeth—i.e., the composite contact deformation in point D under load acting in the same point, and $p(z_F)$, $p(z_D)$ are the tooth loads acting in positions F and D, respectively.

As the contact points are at different distances from the pinion/gear axis, the transmitted torque is defined by the equation:

$$T = \sum_{i_t=1}^{i_t=N_t} \int_{L_{it}} r_F \cdot \left| \vec{p}(z_F) \cdot \vec{t}_{0F} \right| \cdot dz \qquad (36)$$

where: r_F is the distance of the loaded point F to the gear axis, $\vec{t}_{0F}$ is the tangent unit vector to the circle of radius r_F, passing through the loaded point F in the transverse plane of the gear, and N_t is the number of gear tooth pairs instantaneously in contact.

The load distribution on each line of contact can be calculated by solving the nonlinear system of Equations 33–36. An approximate and iterative technique is used to attain the solution. The contact lines are discretized into a suitable number of small segments, and the tooth contact pressure, acting along a segment, is approximated by a concentrated load, ΔF, acting in the midpoint of the segment. The actual load distribution, defined by the values of load ΔF, is obtained by using the "successive-over-relaxation method." In every iteration cycle, a search for the points of the "potential" contact lines that could be in instantaneous contact is performed. For these points, the following condition should be satisfied:

$$\Delta y_{n(i_t, i_z)} \leq \frac{\Delta\phi_2 - \Delta\phi_{2(i_t)}^{(k)}}{\left(\dfrac{\left| (\vec{r} \times \vec{a}_0) \cdot \vec{e} \right|}{r_D \cdot |\vec{r}|} \right)_{(i_t, i_z)}} \qquad (37)$$

The details of the method for load distribution calculation in hypoid gears are described in Reference 22.

Transmission Error

Total transmission error consists of kinematical transmission error due to a mismatch of the gear pair, eventual tooth errors, misalignment of the meshing members and transmission error caused by tooth deflection.

It is assumed that the pinion is the driving member and is rotating at a constant velocity. As the result of the gears mismatch, a varying, angular velocity ratio of the gear pair—and an angular displacement of the gear member from the theoretically exact position based on the ratio of the number of teeth—occur. This angular displacement of the gear can be expressed as:

$$\Delta\phi_2^{(k)} = \phi_2 - \phi_{20} - N_1 \cdot (\phi_1 - \phi_{10}) / N_2 + \Delta\phi_{2s} \qquad (38)$$

where:

- ϕ_{10} and ϕ_{20} are the initial angular positions of the pinion and the gear.
- ϕ_2 is the instantaneous angular position of the gear for a particular angular position of the pinion ϕ_1.
- N_1 and N_2 are the numbers of pinion and gear teeth, respectively.
- $\Delta\phi_{2s}$ is the angular displacement of the gear due to edge contact misalignment of the mating members when a "negative" separation occurs on a tooth pair different from the tooth pair for which the angular position is calculated.

The angular displacement of the gear, $\Delta\phi_2^{(d)}$, caused by the variation of the compliance of contacting pinion and gear teeth rolling through mesh, is obtained as one of the results of the load distribution calculation. Therefore, the total angular position error of the gear is defined by the equation:

$$\Delta\phi_2 = \Delta\phi_2^{(k)} + \Delta\phi_2^{(d)} \qquad (39)$$

Computed Results

The computer program, based on the theoretical background presented, has been applied for the investigation of the combined influence of machine tool settings; head cutter's profile and diameter for pinion finishing; misalignments of the mating members on load distribution; maximum tooth contact pressure (p_{max}); and angular displacement of the driven gear member from the theoretically exact position based on the ratio of the number of teeth ($\Delta\phi_{2max}$). The calculation was carried out for the hypoid gear pair of design data given in Table 1.

The influence of machine settings; sliding base setting (c); basic machine center (f); basic offset setting (g); swivel angle (δ) (Fig. 2) on maximal tooth contact pressure (p_{max}); load distribution factor (β_F); and transmission error ($\Delta\phi_2$) is shown in Figs. 9–12. Factors $k_{p\,max}$, $k_{\beta F}$ and $k_{\Delta\phi 2}$ represent the ratios of maximum tooth contact pressures, load distribution factors and transmission errors obtained by applying arbitrarily chosen values of machine tool setting parameters for pinion tooth generation and obtained by applying the basic values of these

continued

Table 1—Pinion and gear design data

	Pinion	Gear
Number of teeth	11	41
Module, mm	4.41402	
Running offset, mm	25.4	
Outside diameter, mm	77.585	181.859
Face width, mm	31.911	27.762
Crown to crossing point, mm	86.998	30.671
Front crown to crossing point, mm	57.684	
Mean radius, mm	27.495	77.274
Mean spiral angle, deg	50.2597	32.3007
Pitch angle, deg	18.5400	70.5799
Face angle of blank, deg	23.2733	71.5859
Root angle, deg	17.5722	65.6684
Pitch apex beyond crossing point, mm		-0.023
Face apex beyond crossingpoint, mm	3.193	-0.398
Root apex beyond crossing point, mm	5.735	0.767
Mean addendum, mm		1.083
Mean dedendum, mm		6.295
Mean working depth, mm		6.372
Minimal normal topland width, mm		1.930

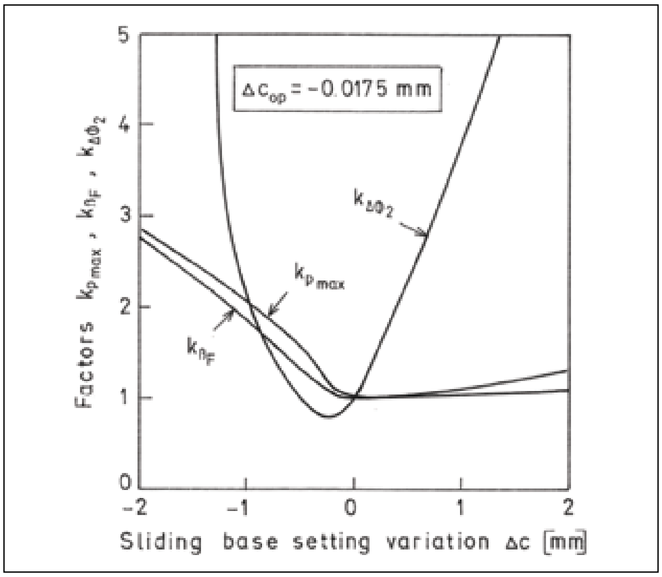

Figure 9—Influence of sliding-base-setting variation on load distribution and transmission error.

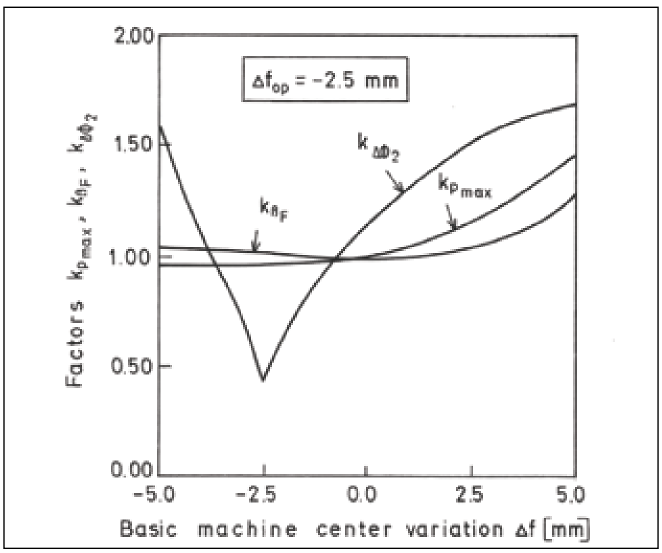

Figure 10—Influence of basic machine-center variation on load distribution and transmission error.

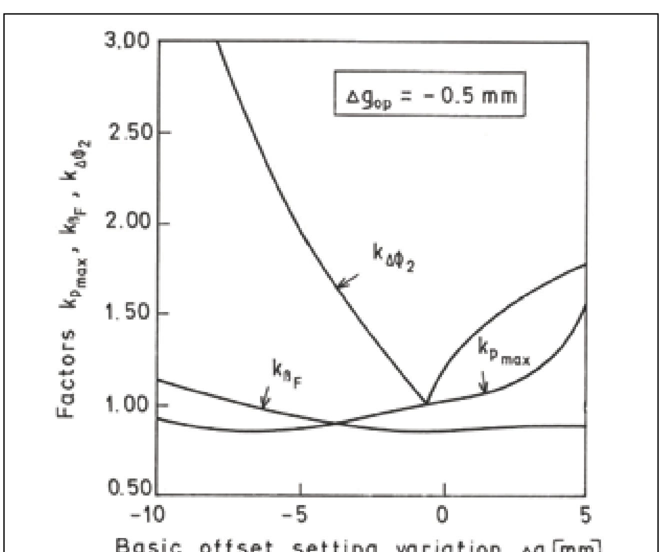

Figure 11—Influence of basic offset-setting variation on load distribution and transmission error.

parameters calculated by the Gleason or Litvin, et al., method (Refs. 41–42).

On the basis of obtained results, optimization of the machine tool setting was performed. By applying the optimized machine tool setting, the maximum tooth contact pressure was reduced by 5.8% and the angular position error of the driven gear by 65.4% when the pinion is finished by the machine tool setting determined by the commonly used methods.

The investigations have shown that a much better tooth contact pattern and less transmission error are obtained by finishing the pinion teeth with a cutter profile consisting of two circular arcs with radii of r_{prof1} = 300 mm, r_{prof2} = –1,150 mm (it means that the tool profile segment of radius r_{prof2} is convex), and when the cutter radius is corrected with $\Delta r_{t1} = -0.18$ mm. By applying these optimal values of cutter parameters, the maximum tooth contact pressure is reduced by 16% and the angular position error of the driven gear by 179%.

The influence of the angular misalignment of the pinion's shaft ε_v and the cutter's parameters on maximum tooth contact pressure was investigated. The computer simulation of loaded tooth contact analysis has shown that the influence of angular pinion shaft misalignment on the optimal value of profile radius r_{prof1} is negligible, but its effect on optimal profile radius r_{prof2} is considerable (Fig. 13). It can be noted that the optimal values of profile radius $r_{prof\,2}$ are moved towards its higher values by the increase of angle ε_h (Fig. 13). The character of curves in Figure 14 indicates that angular shaft misalignment has little effect on optimal cutter radius correction.

Conclusions

A method is presented for the optimization of machine tool setting parameters and cutter for pinion finishing of face milled and face hobbed hypoid gear pairs. The aim of this optimization is to improve load distribution by reducing maximum tooth contact pressure and transmission error. On the basis of the obtained results the following conclusions are made:

- By applying the optimized machine tool setting, the maximum tooth contact pressure is reduced by 5.8%, the load distribution factor by 5.9%, and the angular position error of the driven gear by 65.4% when the pinion is finished by the machine tool setting determined by the commonly used methods.
- By applying the optimal cutter profile and diameter for pinion finishing, the maximum tooth contact pressure is reduced by 16% and the angular position error of the driven gear by 179%, when the hypoid gear pair is manufactured by cutter parameters determined by the commonly used methods. Also, by applying the optimal cutter parameters, the influ ence of shaft misalignments on maximum tooth contact pressure is reduced.

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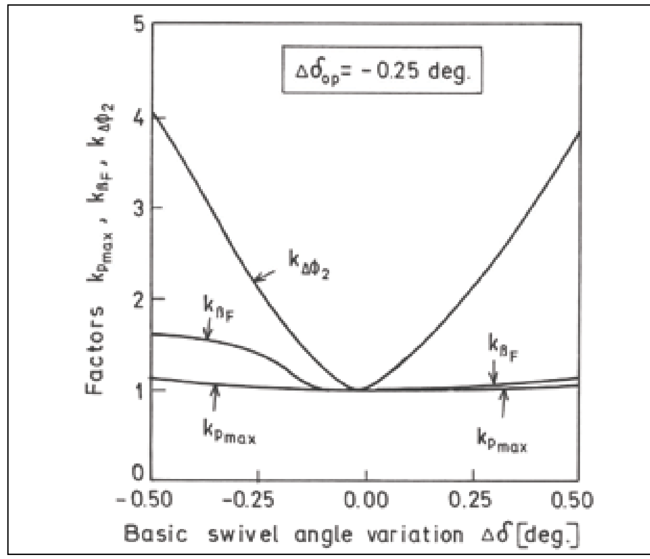

Figure 12—Influence of basic swivel-angle variation on load distribution and transmission error.

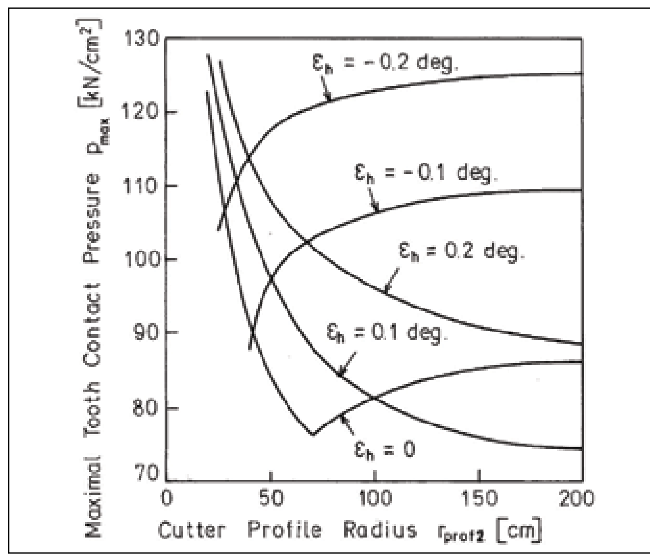

Figure 13—Influence of cutter-profile radius $r_{prof\ 2}$ and angular-shaft misalignment in the horizontal plane on maximal tooth contact pressure.

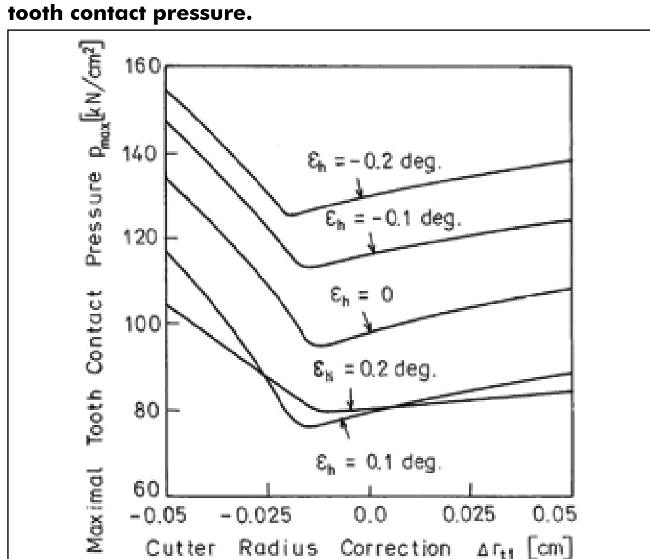

Figure 14—Influence of cutter radius correction and angular-shaft misalignment in the horizontal plane on maximal tooth contact pressure.

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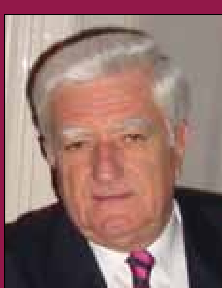

Vilmos Simon *received his B.S.E. in 1963 from the University of Belgrade; mechanical engineering (engineer specialist) and technical doctorate degrees (machine design) in 1968 from the University of Budapest; and a doctor of science degree (machine design) in 1996 from the Hungarian Academy of Science. His work, research and faculty experience include stints at the Jugoalat tool factory, Novi Sad; University of Novi Sad/Faculty of Technology; Szent István University of Godollo; Budapest University of Technology and Economics; Rochester Institute of Technology; Gleason Works; University of Pennsylvania; Philadelphia Gear; Zhengzhou Research Institute; and Technical University of Budapest. Simon's many areas of expertise include—but are not limited to—geometry, kinematics, dynamics, manufacture and elastohydrodynamic analysis of lubrication; finite element development; creation of more than 30 computer programs in support of the above, and more. Simon has headed up a number of research projects and is the author of more than 50 papers and 80 conference presentations.*

CLARIFICATION

A technical paper by Alexander Kapelevich—Measurement of Directly Designed Gears with Symmetric and Asymmetric Teeth—*in the Jan/Feb issue requires clarification. Revisions of Figures 7 and 8, as well as Equations 21–24 are as follows:*

Revised Figures 7–8:

Span Measurement

Span measurement is the measurement of the distance across several teeth, along a line tangent to the base cylinder (Ref. 4). This kind of inspection is used for gears with external teeth. It is also applied only for gears with symmetric teeth, because it is impossible to have a common tangent line to two concentric base cylinders of asymmetric tooth flanks.

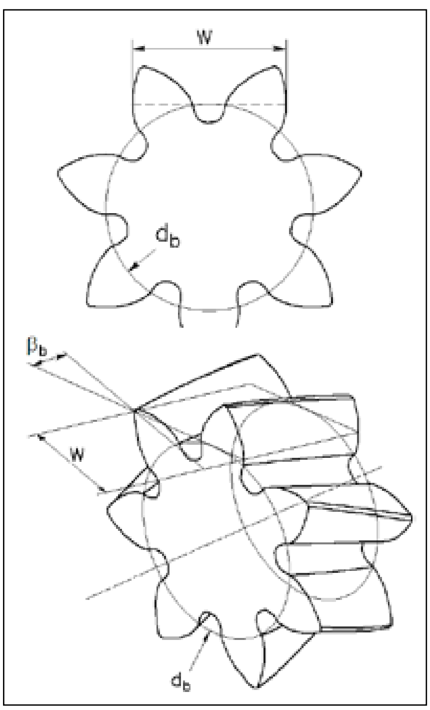

Figure 7 (Revised)—Span measurement; a–for spur gear; b–for helical gear

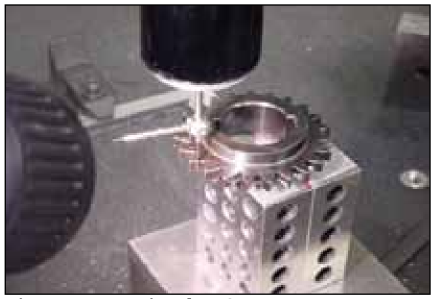

Figure 8 (Revised)—CMM measurement of asymmetric gear.

Revised Equations:

For external helical gears with odd number of teeth, it is more practical to inspect the radial measurement over one pin from the gear center line. In this case the gear with the arbor should be placed between two centers or on two V-blocks. The radial measurement over one pin is defined as:

$$M = \frac{d_p + D}{2} \tag{21}$$

Span measurement over n_w teeth (Fig. 7) is:

$$W = (S_b + (n_w - 1) \text{ x } p_b) \text{ x } \cos \beta_b \tag{22}$$

Where: S_b is the tooth thickness at the base diameter

$$S_b = S \text{ x } \cos \alpha + d_b \text{ x } \cos \beta_b \tag{23}$$

p_b is the circular pitch at the base diameter

$$p_b = \frac{\pi \times d_b}{n} \tag{24}$$

April 4–8—Hannover Messe. Hannover Fairgrounds, Hannover Germany. The world's leading trade show for industrial technology returns in 2011 with 13 leading trade fairs including Industrial Automation; Motion, Drive & Automation; Energy; Power Plant Technology; Wind; MobiliTec; Digital Factory; ComVac; Industrial Supply; CoilTechnica; Surface Technology; MicroNanoTec and Research & Technology. Under the banner "Smart Efficiency," exhibiting companies will be presenting solutions aimed at cutting costs, optimizing process efficiency and conserving resources at every stage of the production chain. Special events include Metropolitan Solutions, the Job and Career Market, TectoYou, WoMenPower and the Hermes Award 2011. France is the official partner country for this year's exhibition. For more information, visit *www.hannovermesse.de*.

April 5–7—AeroDef Manufacturing. Anaheim Convention Center, Anaheim, California. AeroDef Manufacturing is designed to meet the manufacturing challenges in the entire aerospace and defense manufacturing segment. Organized by the Society of Manufacturing Engineers (SME), AeroDef Manufacturing offers a comprehensive exposition and technical conference that includes demonstrations of innovative, enabling technologies and advanced, integrated systems for manufacturing in the military and commercial aerospace, defense and space industries. Educational sessions will have a heavy emphasis on what the industry needs to know to move forward in advancing U.S. aerospace and defense manufacturing. For more information, visit *www.aerodefevent.com*.

May 6–8—Gears, Motors and Controls Expo. Bombay Exhibition Centre, Mumbai, India. GMC 2011 is a showcase of gears, motors, controls and allied products that builds on the success of the earlier editions held in Chennai & Mumbai. It will be held in conjunction with the Pumps, Valves & Compressors Expo 2011. The three-day event will be promoted extensively across India and the region, and visitors will comprise key decision makers from nearly every industry segment. With customer satisfaction at the heart of the trade show's strategy, GMC 2011 hopes to build on previous efforts and deliver maximum rewards to the participants. Bonfiglioli and Elecon are industry partners for the 2011 event. For more information, visit *www.gmconline.in*.

May 10–12—Detailed Gear Design–Beyond Simple Service Factors. Harrah's Las Vegas. Gear engineers, gear designers, application engineers and people who are responsible for interpreting gear designs are recommended to attend this three-day course. The majority of the course material is presented through qualitative descriptions, practical examples, illustrations and demonstrations, which require basic mathematical and engineering skills. However, some familiarity with gear design and application will enhance overall understanding of the material. AGMA members: $1,395 first registrant, $1,195 additional member registrant from same company. Non-members: $1,895 first registrant, $1,695 additional registrant from same company. For more information, contact Jan Alfieri at *alfieri@agma.org*.

May 16–20—Basic Training for Gear Manufacturing. Richard J. Daley College, Chicago. This AGMA training course covers gearing and nomenclature, principles of inspection, gear manufacturing methods, hobbing and shaping. The course is intended for those with at least six months of experience in setup or machine operation. Classroom sessions are paired with hands-on experience setting up machines for high efficiency and inspecting gears. For more information, contact Jan Alfieri at *alfieri@agma.org*.

May 18–21—PowderMet 2011. Marriot Marquis, San Francisco. PowderMet offers exhibitors an opportunity to meet with customers during the course of the three-day conference, both in the hall and during conference meals and special events. Highlights include an extensive technical program, social events and the 6th annual golf tournament. This conference attracts worldwide delegates in the powder metallurgy, powder injection molding and particulate materials industries. This year the International Tungsten & Refractory Metals conference will be hosted by MPIF and co-located at the Marriot Marquis at the same time. For more information, contact Jessica Schade Tamasi at *jtamasi@mpif.org* or call (609) 452-7700 Ext 103.

American Broach

PLANS NEW R&D CENTER

American Broach and Machine Company will establish a new research and development center in the Ann Arbor region. The company recently received a Michigan Economic Growth Authority (MEGA) tax credit totaling $572,782 to help encourage the expansion. "American Broach is focused on quality processes and investing in manufacturing capability in order to make the world's highest quality internal spline broach cutting tools for gears and spline parts," said Ken Nemec, president, American Broach. "American Broach would like to design, develop, and build newly innovated broaching machines to add value to manufacturing while growing jobs in Michigan. The company's new research and development center would give American Broach an edge in a tough industry, and help rebuild Michigan's manufacturing base."

American Broach, a designer and builder of machine tools for the auto and defense sectors, plans to invest $25 million in a new research and development center in Ypsilanti. The company expects the project to create up to 43 new jobs. The MEGA award helped convince the company to invest in Michigan over competing sites in Canada and China. "The Ann Arbor region is a destination for advanced manufacturing," said Skip Simms, interim president and CEO. "Access to talent as well as existing facilities and equipment help advanced manufacturing businesses hit the ground running and grow quickly in the Ann Arbor region." The City of Ypsilanti is considering tax abatements in support of the project.

Gleason

EXPANDS ADVANCED COATINGS FACILITY

Gleason Corporation has recently announced a new investment to expand the Advanced Coatings Facility at its Gleason Cutting Tools Corporation (GCTC) facility located in Loves Park, Illinois. The expansion includes an enlargement of the coating department that will accommodate an additional Balzers Coating System and an additional automated computer controlled cleaning line. The additional coating unit and clean line complements the substantial coatings capability that already exists at Gleason. "The additional capacity this investment will bring will help further reduce our lead times to meet the increasing demand for GCTC's Contract Sharpening and Recoating Service business, as well as coating requirements on our new cutting tool products," says Roger Hackman, vice president of technology and quality. Gleason Cutting Tools Corporation can provide Titanium Nitride (TiNite), Titanium Carbonitride (CarboNite), Titanium Aluminum Nitride (AlNite) and Aluminum Chromium Nitride (AlCroNite) coatings, applied to high speed steel and carbide substrate cutting tool materials.

Solar

OFFERS WHITE PAPER ON PID PARAMETERS

Solar Atmospheres, Inc. has written a technical white paper on understanding and proper usage of PID parameters as they relate to controlling vacuum furnace temperature. This tutorial will assist furnace engineers and operators to better understand and properly use the functions of a PID Controller and thereby obtain improved temperature control, particularly in program ramping and minimizing temperature overshoot, required for AMS 2750D. This paper describes how to both manually and automatically set PID values for various temperatures of operation. It highlights how to establish better results especially in the lower temperature ranges of operation (300–1500 degrees F) and also how to better understand instruments that have the auto-tuning feature. The paper can be viewed at *http://www.solaratm.com/technical-paper-downloads*.

PMA

ELECTS NEW LEADERS

The Precision Metalforming Association (PMA) recently announced the election of Augusto Gil (pictured right) to the position of Metal Spinning Division vice chair and Benjamin Barnett to Next Generation Leaders Division vice chair. Gil, general manager for Hialeah Metal Spinning, Hialeah, Florida, has been active within the Metal Spinning Division for nearly a decade and previously served as a member of the group's program committee. In his new role as vice chair, he will work to increase division participation and membership by preparing valuable content for presentations and group discussions. Gil will serve alongside Metal Spinning Division chair/director John McGeever, president of Charles Schillinger Co. McGeever acted as vice chair in 2009 before being elected chair in 2010-2011. Gil will


Augusto Gil


John McGeever

continued


Benjamin Barnett

serve one year as vice chair and then will become chair/director in 2012-2013. Barnett, vice president of Principal Manufacturing Corp., looks to increase participation of younger people within PMA's Next Generation Leaders Division through his responsibilities as vice chair of the group. He has been participating in division activities for the past several years and serves as the membership chair for the local PMA Chicago District. Barnett joins chair/director Rick Bachman, vice president of logistics for Feintool North America, with leadership positions in the division. Bachman served as vice chair in 2009-2010 before being named chair/director for 2011–2012. Barnett will act as vice chair of the division in 2011–2012 before becoming chair/director in 2013–2014.


Rick Bachman

Manufacturing Technology Consumption

UP 85 PERCENT IN 2010

December U.S. manufacturing technology consumption totaled $446.76 million, according to The Association for Manufacturing Technology (AMT) and the American Machine Tool Distributors' Association (AMTDA) This total, as reported by companies participating in the United States Manufacturing Technology Consumption (USMTC) program, was up 40.9 percent from November and up 104.8 percent when compared with the total of $218.16 million reported for

December 2009. With a year-to-date total of $3,236.00 million, 2010 is up 85.3 percent compared with 2009.

These numbers and all data in this report are based on the totals of actual data reported by companies participating in the USMTC program.

"For the first time in USMTC history, we experienced four months of consecutive growth following IMTS, ending the year on a solid upswing," said Douglas K. Woods, president of AMT. "2010 orders closed strong, up 85 percent over 2009, and December's orders were 40.9 percent higher than the previous month. With backlogs firming and quotation levels accelerating, we are very optimistic that the industry will see strong results in 2011."

The USMTC report, jointly compiled by the two trade associations representing the production and distribution of manufacturing technology, provides regional and national U.S. consumption data of domestic and imported machine tools and related equipment. Analysis of manufacturing technology consumption provides a reliable leading economic indicator as manufacturing industries invest in capital metalworking equipment to increase capacity and improve productivity. For more information on this report, visit *www.amtonline.org*.

Madeira

APPOINTED VICE PRESIDENT OF HEAT TREATING

Inductoheat, Inc., located in Madison Heights, Michigan, has appointed Robert Madeira as the vice president of heat treating. Madeira has worked in the induction heat treat industry for more than 26 years, most recently as the director of sales for Fluxtrol, Inc. He has a B.S. from Michigan State and a M.A. from University of Detroit. As the vice president of heat treating, Madeira is responsible for successfully managing customer's needs, lead-

Robert Madeira

continued

ership of Inductoheat's business strategies and spearheading the North American sales efforts. This position reports directly to the president. "With his extensive background in the heat treating industry, we are excited to see what new business opportunities Rob will bring to Inductoheat," said Douglas Brown president/COO of Inductoheat Inc.

Gary Doyon, president/CEO of the Inductotherm Group adds, "We are thrilled to have Rob join us in our continued effort to add great teammates with experience and enthusiasm. We will have Rob help push our efforts to grow both domestically as well as internationally for our fa

Interactive Manufacturing Experience

CHARTS NEW COURSE FOR U.S. MANUFACTURING

From some of the United States' largest industrial technology innovators to the heads of small machine shops and providers of workholding, drilling and grinding tools, manufacturers from across the country will gather this fall at an unprecedented industry summit to chart a new course for the domestic manufacturing industry. Kennametal Inc., a global market leader in delivering productivity in demanding environments through innovative custom and standard wear-resistant solutions, has become the newest experience partner to sign on for the event. Taking place Sept. 12-14, 2011, in Las Vegas, the Interactive Manufacturing Experience (imX) will bring together industry leaders from companies large and small to recognize that only through education and collaboration can the nation's manufacturers thrive in today's global marketplace. "Although we're still seven months out from the event, early response from manufacturers coast to coast is an indicator of imX's vital importance to this industry," said Steve Prahalis, experience manager, imX.

Dozens of exhibit partners already have committed to joining the conversation and to demonstrating their own manufacturing innovations at imX, among them: Big Kaiser, Blum LMT, Briney Tooling Systems, Carl Zeiss IMT, Caron Engineering, Castrol Industrial, CGTech, Chiron America Inc, Cimcool Industrial Products, Data Zen Engineering, Delcam, Esprit/DP Technology, Exsys Tool, GTR mbH, Heimatec GmbH, Hermle Machine Co., Hurco Companies, Index Corp, Jauch Schmider GmbH, Iscar Metals Inc, Koma Precision, Kuka Robotics, Losma, Marubeni Citizen-Cincom, MasterCAM, MD Tooling, Midaco Corp, Mitsui Seiki, Pat Mooney Inc, REGO-FIX, RobbJack Corp/Crystallume, Royal Products, Schunk, Seco Tools, SNK America, StarragHeckert, Stratasys, Unisig, United Grinding Tech, Von Ruden Mfg, WTO Inc and Zoller. For more information, visit *http://imxevent.com*.

Manufacturing Summer Camps

ENTICE AMERICA'S YOUTH

Nuts, Bolts and Thingamajigs (NBT) and the National Association for Community College Entrepreneurship (NACCE) have partnered together to launch a unique summer camp program that combines elements of manufacturing and entrepreneurship with how things are made and how businesses develop. The schools have hosted a 2010 camp as part of a pilot program that will eventually develop into a national program with as many as 300 locations across the United States. Campers design and build a product experiencing the start to finish satisfaction of creating something they can show off with pride. Throughout the process, they learn how to do CAD design and operate various kinds of manufacturing machinery under the close supervision of expert manufactur-

continued

ing trainers. They will also tour local manufacturing facilities learning what kinds of jobs exist, what skills and training are required and how those businesses developed. They will have the opportunity to hear directly from local manufacturing company owners how they started their businesses, applying basic entrepreneurship principles to understand how a single product idea becomes a business. Whether a young person aspires to work in manufacturing or medicine or law or any other field, having a basic understanding of how things are made and how businesses develop will make them more appreciative of the world around them and the "tools" they will use in their adult life. Each camp is unique in its own way and addresses local needs. All are aimed at ensuring the future of manufacturing by changing the image of manufacturing for today's youth. For more information on these camps or to make a contribution in honor of one of the camps, call 888-394-4362 or visit *www.nutsandboltsfoundation.org*.

Nevada Heat Treating

Purchases BeaverMatic Furnace

Nevada Heat Treating, Inc., a progressive commercial heat treating company, has purchased a BeaverMatic Internal Quench Furnace (IQF) and a BeaverMatic Temper Furnace for enhanced acceleration of product development cycles in growth industries.

Based on consistently high performance results of existing BeaverMatic equipment installed in Nevada's plant operations, vice president, Patrick McKenna, purchased a new

BeaverMatic IQF Furnace with a work area of 30 inches wide by 48 inches long by 26 inches high for hardening 1,500 pounds of small-to-medium size parts.

A trade secret of BeaverMatic is its "RAM" transfer system, a pull-push design, for reliably controlling a load from different zones. Unique to this furnace design is a third RAM which pushes loads from the hot zone to the quench elevator. This third RAM is located between the two existing RAMS.

As an environmentally-conscious heat treater, McKenna chooses eco-friendly products and methods to minimize their environmental impact. Therefore, when purchasing this internal quench furnace which utilizes 1,300 gallons of oil for heating and cooling, it was critical to incorporate a quench oil centrifugal filtration system.

For precision control of temperature and furnace operations, Super Systems Inc. of Cincinnati, Ohio is being used for software and process control. The 9205 controller will provide all the necessary temperature and atmosphere control. The 9205 is specially designed for complex atmosphere furnace applications using sophisticated recipe programming, Ethernet communications, built in data logging and expandable I/O to meet the most complex requirements. Complementing the process control is the SuperDATA SCADA system which will provide Nevada Heat Treat with quick access to real-time and historical data throughout the plant.

Ipsen

OPENS SECOND OFFICE IN JAPAN

Ipsen has expanded its existing coverage in Japan by opening a new sales office in Tokyo, Japan. The new office will focus on sales and service in the Kanto region. This expansion will assist Ipsen in continuing to provide superior customer service and support backed by Ipsen's knowledge and expertise. At the same time, Ipsen is pleased to announce the newest member of its sales team heading up the Tokyo office, Stephan Gagne. With more than 12 years experience in the Japanese market, Gagne has extensive knowledge of production, aerospace, tooling and the automotive industries. Gagne will focus his attention and support on the growing demand for new equipment with an emphasis on aftermarket sales and service. Ipsen, Inc. designs and manufactures thermal processing systems for a wide variety of markets including aerospace, medical, energy, chemical and automotive.

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6) What is your primary job function responsibility? (Check one)

- ❑ Corporate Executives (B)
- ❑ Manufacturing Production Management (C)
- ❑ Manufacturing Production Department (D)
- ❑ Manufacturing Engineering Management (E)
- ❑ Manufacturing Engineering Department (F)
- ❑ Product Design, R&D Management (H)
- ❑ Product Design/ R&D Personnel (I)
- ❑ Purchasing (L)
- ❑ Marketing & Sales, Communications (M)
- ❑ Quality Control Management (P)
- ❑ Quality Control Department (Q)
- ❑ Other (N) ____________________

7) How is THIS LOCATION involved in the gear industry?
(Check all that apply)

- ☐ WE MAKE GEARS (or Splines, Sprockets, Worms, etc.) (20)
- ☐ WE BUY GEARS (or Splines, Sprockets, Worms, etc.) (22)
- ☐ WE SELL NEW MACHINES, TOOLING OR SUPPLIES TO GEAR MANUFACTURERS (24)
- ☐ WE provide SERVICES to gear manufacturers (25) (please describe) ____________________
- ☐ WE distribute gears or gear products (including agents and sales reps. (26)
- ☐ WE are a USED MACHINE TOOL dealer (30)
- ☐ Other (please describe) ____________________ (32)

8) Which of the following products and services do you personally specify, recommend or purchase? (Check all that apply)

Machine Tools

- ☐ Gear Hobbing Machines (50)
- ☐ Gear Shaping Machines (51)
- ☐ Gear Shaving Machines (52)
- ☐ Gear Honing Machines (53)
- ☐ Gear Grinding Machines (54)
- ☐ Gear Inspection Equipment (55)
- ☐ Bevel Gear Machines (56)
- ☐ Gear/Spline Roll-Forming Equipment (57)
- ☐ Broaching Machines (58)
- ☐ Heat Treat Equipment (59)
- ☐ Deburring Equipment (60)
- ☐ Non-Gear Machine Tools Turning, Milling, etc.) (61)

Tooling & Supplies

- ☐ Functional Gages (62)
- ☐ Workholding (63)
- ☐ Toolholding (64)
- ☐ Cutting Tools (65)
- ☐ Grinding Wheels (66)
- ☐ Gear Blanks (67)
- ☐ Lubricants/Cutting Fluids (77)

Service & Software

- ☐ Heat Treat Services (69)
- ☐ Gear Consulting (70)
- ☐ Tool Coating (71)
- ☐ Tool Sharpening (72)
- ☐ Gear Design Software (73)
- ☐ Gear Manufacturing Software (74)

Power Transmission Components

- ☐ Gears (75)
- ☐ Gear Drives (76)
- ☐ Bearings (78)
- ☐ Motors (79)

9) What is the principal product manufactured or service performed at **THIS LOCATION?**

____

10) How many employees are at THIS LOCATION (Check one)

☐ 1-19 ☐ 20-49 ☐ 50-99 ☐ 100-499 ☐ 500+

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The Broach Masters/Universal Gear Co.	13	(530) 885-1939	info@broachmasters.com	www.broachmasters.com
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The Gear Works—Seattle, Inc.	87	(206) 762-3333	sales@thegearworks.com	www.thegearworks.com
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Metal Finishing News	83	(212) 633-3100	metalfinishing@elsevier.com	www.metalfinishing.com
Midwest Gear & Tool Inc.	80	(586) 779-1300	midwestgear@sbcglobal.net	
Mitsubishi	8	(248) 669-0614		www.mitsubishigearcenter.com
Mitutoyo	19	(888) 648-8869		www.mitutoyo.com
Niagara Gear	4	(800) 447-2392	info@niagaragear.com	www.niagaragear.com
Normac Inc.	22	(828) 209-9000	info@normac.com	www.normac.com
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Sandvik Coromant	33, 73, 86	(800) 726-3845		www.coromant.sandvik.com/us
Schafer Gear Works Inc.	28, 29	(574) 234-4116		www.schafergear.com
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The Gear Ring and Other Creative Anomalies

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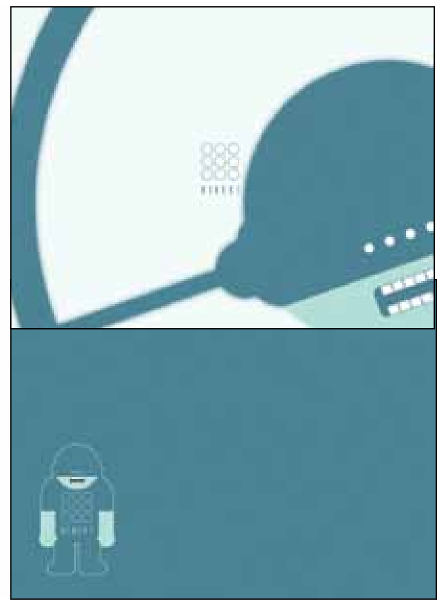

Created by conceptual designer Glen Liberman, the Gear Ring is ideal for those that like to fidget and play with their own jewelry (courtesy of Kinekt Design).

Glen Liberman's interest in micro-precision parts led him to design and sell products that fuse a modern aesthetic with physical interaction. Liberman launched his company Kinekt Design in the New York area in 2010. The design firm creates everything from toys to T-shirts as well as jewelry and paper crafts. It just so happens that his most popular product to date has a certain familiar theme—gears.

"I had been purchasing some Diesel jewelry and really liked the spring-loaded push-button clasps they use on some of their necklace designs. I also like the concept of moveable products that offer some play value in their aesthetic and/or form."

With these concepts in mind, Liberman came up with the Gear Ring, a piece of jewelry fabricated in 316L stainless steel that features gears that turn in unison when the outer rims are spun.

"The idea for the Gear Ring stems from a few different influences and inspirations. I noticed people tend to fidget and play with their jewelry or clothing items in an attempt to alleviate anxiety," Liberman says. "Secondly, I had been looking into micro-precision parts and saw the need for a product that could implement these types of small components. It came together quite nicely."

The Gear Ring has been Kinekt Design's biggest seller thanks to interest from a wide range of individuals. "We've had many different demographics of people that it's been a bit difficult to pinpoint a particular group. However, those who are interested in transportation and vehicle design have seemed to take a particular liking to the product," Liberman says.

An overwhelming positive response from those involved in automotive, engineering and even steampunk has prompted the design firm to create more gear-themed products in the future—maybe.

"It's possible that Kinekt Design is currently working on another gear-inspired jewelry piece that will be interactive, but most of it is top secret. We urge those interested to check back in on the website now and again for new product updates," Liberman adds.

Until that time, Kinekt Design has more than enough creativity and design appeal on its website to intrigue visitors including original MP3 tracks composed by Liberman and Ben Mullins.

One of its original electronic MP3 tracks entitled, *Builder*, was licensed by Toyota for a national campaign. This catchy, minimal yet melodic composition highlights the fun and playful style found throughout Kinekt Design's products. Other original tracks include *Beeptweek, Ghosts/Drifting* and *MosquitoMaps*. For fans of minimalism and 8-bit graphics, Liberman has collaborated with animator/VJ Motomichi Nakamura, to offer Twitter icons, I-phone wallpaper and desktop wallpaper (pictured above right)..

With innovative gear jewelry, electronic dance tracks and do-it-yourself paper computer models (check this interesting item out online at *www.kinektdesign.com*) it's obvious those involved with Kinekt Design don't like to get pigeon-holed into a particular area of interest. Liberman considers himself a self-taught conceptual designer who enjoys creating things in various media including electronics, mechanics, graphic design, paper craft and sound design with more to come in the future.

It's unusual and clever concepts like the Gear Ring that inspires the entire design firm.

"Originally, we came up with the idea for the Gear Ring back in March of 2009, but it took more than a year to bring it to the market due to the mechanical complexities. With our interests in both machining and micro-precision parts, it just made sense on many levels."