June/July 2012

GEAR TECHNOLOGY®

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The Journal of Gear Manufacturing

QUALITY & INSPECTION

RENEWABLES: A WIND AND WATER MIX

Technical Articles

- Ask the Expert
- Asymmetric Gears: A Parameter Selection Approach
- Flank Load-Carrying Capacity of Internal Gears
- AGMA's Accuracy Classification System for Double-Flank Composite Measurements

Addendum

- da Vinci—The Smartest Guy in the Room

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IMTS 2012

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PTG-1

GS 400

Affordable hob sharpening and in-house tool maintenance

Star's PTG-1 sharpens both straight and spiral gash hob designs up to 8" OD x 10" OAL. Additionally, it sharpens disk, shank and helical type shaper cutters and a wide range of round tools, making it a versatile tool room machine.

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Designed to grind shaving cutters and master gears, the GS 400 sets new standards for precision, reliability and ease of use. An integrated measuring unit automatically checks the quality of the first tooth ground without unclamping the workpiece.

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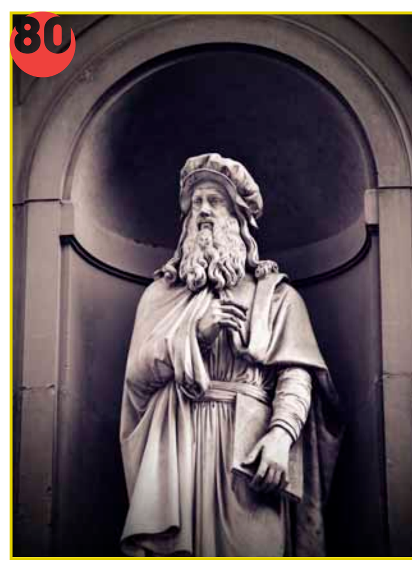

FEATURES

TECHNICAL ARTICLES

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GEAR TECHNOLOGY

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GEAR TECHNOLOGY

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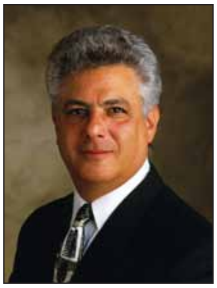


SUSTAINING EXPERTISE in the GEAR INDUSTRY

One of the most talked-about subjects in manufacturing is sustainability. Motivated individuals are looking for ways to make their organizations more efficient, less wasteful and more environmentally responsible. Many manufacturing managers are looking at their energy consumption, water use and physical waste for ways they can reduce, reuse and recycle. The idea is to minimize waste and conserve vital resources.

Those who fully commit to the lean and green approach are almost always rewarded with more than the satisfaction of corporate responsibility. Their organizations become better, stronger, faster—and most importantly, more profitable.

But there's one vital resource that's often overlooked. And although it has little to do with the green movement, it's one that's crucial for the continued success of any organization. The resource I'm referring to is expertise.

Unfortunately, expertise is also a resource that's hard to sustain. Because it's tied to the individuals who've earned it through experience and practical application, it tends to go away when they go away.

Employees are mobile. They leave, and they take their expertise with them. Or they retire—and this is an important factor for the gear industry, which has more than a little gray hair.

Every time I go to an industry event, I am alarmed at the aging of the gear industry's "go-to" guys. While there are plenty of younger engineers working in the gear industry, some of the best and most experienced minds in gearing are either approaching retirement or already retired. Most of the experts I've encountered are extremely grateful for their careers and their livelihoods, and they're eager to give back by sharing what they've learned with the next generation.

So if you have one of these experts in your organization, it behooves you to ensure that his or her knowledge and expertise is transferred to the next generation—before it's too late. You probably have plans in place to conserve other resources at your plant. But what are you doing to conserve knowledge? Do you have a mentoring program in place? Do you have a formal training program for younger engineers? Are your experts' best practices being codified into your company's written procedures? What are you doing to protect what may in fact be your most valuable resource?

This magazine was founded as a teaching journal. Our mission has always been to help the gear industry maintain a body of knowledge, so I like to think that all along, we've been doing our part as "The Gear Industry's Information Source." But over the past several issues, we've expanded that focus with the introduction of our "Ask the Expert" column. We take reader-submitted questions and put them before one or more experts. It's our way of sustaining the knowledge base.

The column has generated enormous interest and extremely positive feedback. Not surprisingly, many of you are dealing with challenging gear-related problems every day. You have questions about gear design, manufacturing, inspection, heat treating and more. Maybe there isn't an appropriate expert at your own company or within your circle of peers. Fortunately, our role as publisher puts us in a position to make acquaintances with a wide array of experts, starting with our roster of technical editors and extending to many the world's most gifted and experienced gear engineers and researchers among our advertisers and other companies in the industry.

So if you have a question, we invite you to submit it. You can do so by following the "Ask the Expert" link on our home page (*www.geartechnology.com*). We'll track down the experts who can best answer it, and we'll help preserve their knowledge for the industry by presenting the most useful and relevant answers in our pages. And if you're one of those lucky few who have spent a career in the gear industry and want to give something back, we invite you to join our panel of experts. There's already a list of questions posted online that need your answers.

Whether you help us by contributing questions or answers, participating in the "Ask the Expert" column is a great way to help us ensure that the gear industry's body of knowledge continues to grow.

Michael Goldstein,

Publisher & Editor-in-Chief

Dear Editors,

I received the May issue of *Gear Technology* today and have been fascinated reading your "Ask the Expert" feature.

Assuming you haven't already heard from someone else, I wanted to check in on this subject.

The engineer at WEG (Walmir Fernandes Navarro) seems to be questioning the discrepancy between what is known about efficiency vs. ratio of hypoid and other gears and what gear motor manufacturers claim in their marketing for gear motors containing hypoid gearing.

I'm not an engineer, but a lot of good information and theory was presented without addressing the obvious answer that explains the discrepancy puzzling Mr. Navarro.

(George) Lian and (Ted) Krenzer touched on it by noting the Sumitomo gear motors use hypoid in the 'input' stage.

It's pretty simple that the majority of the reduction without loss in efficiency comes from using spur gears in subsequent stages. They're just using a low-ratio hypoid (10:1 or less) to turn the corner.

I have attached two Sumitomo cutaway images that show the hypoid in the first stage followed by spur gearing.

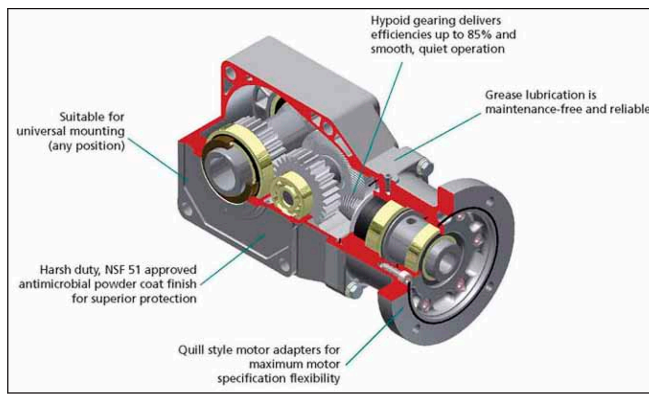
It's surprising nobody mentioned (New Jersey-based) Brother International (*www.brother-usa.com*) gear motors (made by Nissei and marketed here by Brother), because they use basically the same idea.

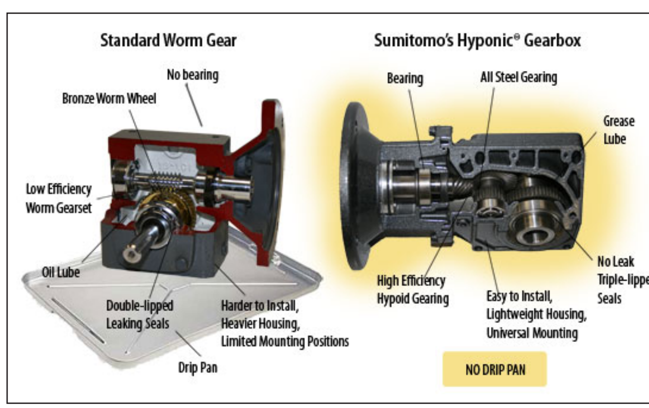
With the cost of hypoid gearing and complexities of different offsets, it's much easier to standardize on one gear set to turn the corner and use spur or helical to maintain high efficiency for overall high reduction in the gear motor.

Thanks for indulging me.

Best regards,

John Morehead, *vice president, business development*
Dunkermotoren USA Inc.





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Tomorrow's Gear Inspection Systems: Arriving Just in Time

The gear inspection systems of the future will be very different from those found in many of today's quality labs. Part setup and programming, regardless of part type or complexity, will be largely automated and require minimal operator experience. The operating system will be native Windows-based so that gear inspection data in dozens of languages can flow seamlessly into the user's network, and every major industrial standard—from AGMA to GOST—can be easily supported. Operators will work in a much more ergonomic environment than they're used to, in close proximity to the work area or using a remote hand-held pendant that puts Internet connectivity right at their fingertips, and supports voice notes, work messages, even video telephony for record-keeping and remote diagnostics. The system itself will be much more robust, and built for the rigors and temperature swings of the shop floor. Most importantly, these inspection systems will inspect *all* types of gears, gear cutting tools, and even prismatic parts at speeds anywhere from 20 to 45 percent faster than existing systems.

Those are just some of the reasons forward-thinking gear producers like Eaton have begun using Gleason's new GMS system.

With installation of the next-generation Gleason 350GMS Analytical Gear Inspection System, Eaton Kings Mountain's gear lab is able to meet its high-volume gear and gear cutting tool inspection requirements: 1,500 parts per week, three shifts a day.

Wanted: An Inspection 'Workhorse' at Eaton.

For almost a century, Eaton has been supplying the trucking industry with products designed to improve vehicle performance and power, and increase profitability. Today, Eaton transmissions are used by many of the world's leading OEM vehicle manufacturers. At Eaton's Kings Mountain, North Carolina facility, meeting the company's growing demand for high-quality transmission gears that run smoother, quieter and more reliably has placed a heavy load squarely on the shoulders of the gear inspection lab, according to Eaton gear engineer Angela Hastings.

"We had been relying on two older M&M Precision Systems (now Gleason Metrology Systems) machines and two older competitor machines to inspect an average of 1,500 parts per week," Hastings says. "The lab inspects parts three shifts per day and most weekends, performing inspections on internal teeth and external teeth at various states of manufacturing; any delays in the lab would cause unacceptable production losses. We also routinely inspect shave cutters and periodically, shaper cutters—the accuracy of these inspections helps keep our production running smoothly."

The search for a new inspection solution to augment these older machines began in 2011 when Hastings and other Eaton engineers and a gear lab technician visited several suppliers of gear inspection equipment, and evaluated their products. "Along with the normal characteristics of cost, delivery and cycle time, we also considered the operating system, ease of programming (both inspection and alignment), training requirements, technical assistance and repair parts supply," recalls Hastings. "The Gleason GMS system emerged as the leader."

GMS Improves on Gleason's GMM Series

At IMTS 2010, Gleason Metrology Systems introduced the first of its new line of GMS model inspection machines. Today the line is comprised of seven different models, with capacities from 0–3,000 mm gear diameter. According to Gleason Metrology Systems' sales manager, Dennis Traynor, the GMS series has evolved from the company's GMM series and incorporates a host of design improvements and enhancements that make the GMS machines faster, easier to operate and more reliable.

"The GMS incorporates everything that was unique to the GMM series, like *GAMA*, Gleason's native Windows-based operating system, but takes it to a new level, with *GAMA 2.0*," says Traynor.

Like the original *GAMA*, *GAMA 2.0* easily communicates inspection results across a customer's plant-wide network, a process much more difficult for competitors who don't operate with Windows, Traynor says. But *GAMA*

2.0 also helps reduce cycle times. GMS series machines offer a 20–45 percent reduction in cycle times as compared to the GMM systems, Traynor says, and they offer a 10–25 percent reduction in cycle times versus competitive systems. These improvements result from *GAMA 2.0*'s faster calculation speeds, coupled with axis movement optimizations that have been made in the GMS.

Traynor also points out that the *GAMA 2.0* applications suite makes programming a remarkably simple task for almost any operator, regardless of part type or inspection requirement.

"When you open the *GAMA* suite, whether you go to cylindrical gear, spiral bevel gear, hob, shaper cutter, shaver cutter—any package for gear or gear tool inspection—the graphical user interface (GUI) is virtually identical, so the operator is as comfortable programming a hob as he would be a cylindrical gear," says Traynor. "You don't have to remember a special routine or rely solely on an operator's experience. Most importantly, we're able to support all major industrial standards—AGMA, DIN, JIS, ISO, GOST—and some 20 different languages, thus accommodating the needs of almost any user globally. The user can additionally choose a variety of chart output styles and configurations for analysis, and save documentation in many formats, including .jpg, .bmp, .gif or PDF."

Hastings agrees. "The ease of programming for part inspections and alignment functions is simple enough for even our new gear lab technicians to learn quickly," she says. "The ability to quickly change the part programs and parameters makes this our favorite machine to run prototypes or perform special inspections to troubleshoot production issues."

Support and Control Right at Your Fingertips

If and when one of Eaton's gear lab technicians need support, whether creating a part program, or managing an inspection, or troubleshooting a problem, there are only two things a Gleason engineer sitting hundreds of miles away can't help with or solve in real time: (1) physically setting up the part, and (2) pushing the start button. This might be one of the GMS system's most important new capabilities, says Traynor. "Through a simple Ethernet connection or a secure 'Team Viewer' web browser, we can view the screen, share part prints, even create a part program. We've also introduced a video telephony and voice mail messaging capability through a new remote pendant control, enabling the user to capture video, describe a particular programming issue and transmit it over the web to our support team."

The remote pendant is also particularly useful during setup, because it enables the operator to answer program prompts from anywhere in the work area and is equipped with twin thumb-controlled joysticks for controlling the speed and positioning of each individual axis.

The main operator work station is situated in close proximity to the work area. A control pendant with twin variable speed joysticks mounted close to the work zone also gives the operator excellent manual control, with axes-selectable operation, feed rate override selection, drive ON, reference point and E-STOP.

Better Sensors, Surface Roughness and Barkhausen.

GMS productivity and versatility are greatly enhanced through use of the Renishaw SP80H 3-D scanning probe, available in various probe configurations and stylus sizes, and with an automatic probe change system for every model. The Renishaw probe makes it possible for the GMS to deliver a faster, more accurate measurement capability for even the most complex gear tooth profiles, including crowning, hollow and taper. In addition, the SP80H is kinematically coupled to the drive system, thus helping minimize the potential for a costly collision and damage to the probe, part or machine.

GMS also has the capability to perform surface roughness testing for cylindrical gears (spiral bevel gear surface roughness testing is in development), with special probes pre-configured for the automatic probe changer.

Barkhausen inspection—a 'non-destructive' measurement of surface hardness and residual and compressive stresses—is also a GMS capability.

Built for Reliability

The GMS systems easily meet VDI/VDE Class 1 specifications, with 2 micron system accuracy. Exceptional accuracy, repeatability and reliability can be attributed to a number of unique design features. For example, all axes are made from highly stable, robust Meehanite cast iron, as compared to more common cast irons or weldment designs. The GMS systems also use linear drives for improved speed and positioning accuracy on all axes. All models use a solid-granite base as well, providing greater stability as compared to designs using cast-iron weldments or partial-granite bases.

Even the controls cabinet on the GMS systems has been re-designed and relocated for better access, safety and reliability, with a simpler design, fewer failure points and reduced noise.

"We knew from our experience with our M&M Precision Systems machines that the service and parts side of the business is excellent," concludes Eaton's Hastings. "We know from these calibrations that the machines are built to keep running accurately through many inspections without problems or the need for adjustments. They are pure workhorses."

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Seco to Offer Diverse Technologies at IMTS

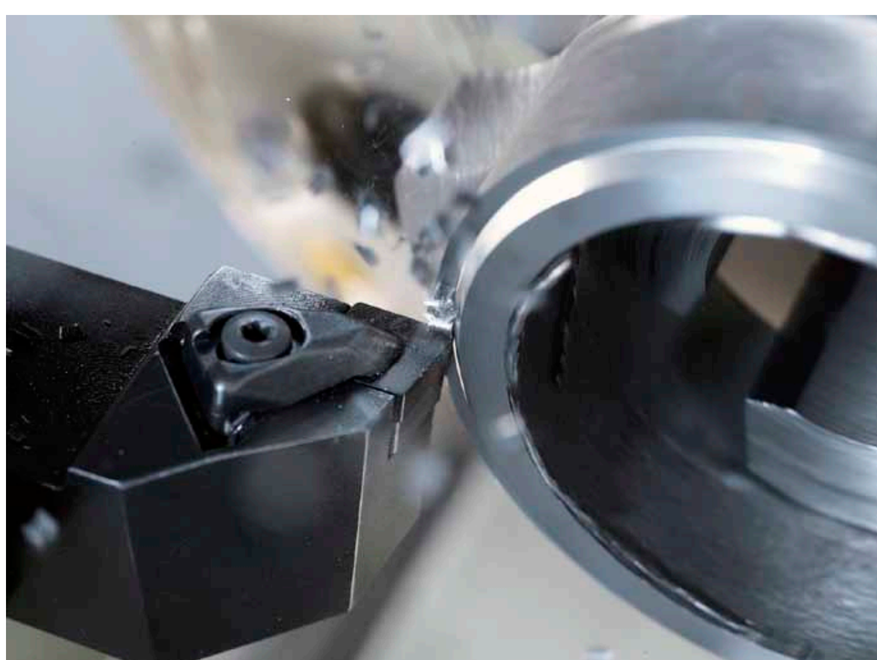
Seco Tools will meet the increased productivity needs of various manufacturers, from aerospace to windpower to automotive to medical, by showcasing a broad range of innovative metal cutting solutions and diverse technologies for turning, milling and holemaking applications in booth W-1564 at IMTS in Chicago.

Products on display will include Duratomic TK insert grades for cast iron turning, Duratomic DP3000 heat-resistant inserts for Perfomax Indexable Drills, the Square 6-04 shoulder milling cutter, the Turbo 10 cutting tool, the Double Octomill-05 face milling cutter for smaller machines and a new generation of disc milling cutters for large slot width.

The TK1001 and TK2001 insert grades, featuring Duratomic coating process technology, offer high performance turning, allow for faster cutting speeds and increased tool life, and effectively reduce the number of grades needed in cast iron applications. TK1001 effectively tackles gray cast iron from finishing up to semi-stable conditions and nodular cast iron under stable conditions as well as hardened steels in the 40–45 Rc range. The TK2001 steps in for the successful machining of nodular cast iron and gray cast iron in tougher applications, all the way to heavy interrupted cuts. The TK grades are available in 211 new geometries and chip grooves, ensuring applicability across a broad range of operations.

The Duratomic DP3000 heat-resistant inserts for Perfomax Indexable Drills are suitable for very high feeds and speeds. This versatile insert grade provides for excellent wear resistance and edge toughness, and is suitable for most materials and applications. In combination with a strong drill body, DP3000 offers high productivity, long tool life and excellent application security. With the Duratomic process, as featured on TK1001, TK2001 and DP3000 product, aluminum and oxygen are manipulated at the atomic level to create insert coatings with unmatched toughness and abrasion resistance.

The Square 6-04 is the smaller version of Seco's Square 6 square shoulder milling cutters. It includes tool diameters down to 0.75" (20 mm). This addition allows the benefits of Square 6 to be applied to an even greater range of applications. Square 6-04 is well suited for small and medium milling machines. It incorporates trigonal inserts with six cutting edges and when combined with the close pitch cutter, Square 6-04 can result in improved cost-efficiency and increased productivity. The 90-degree setting angle ensures a true 90-degree square in one



operation, saving valuable production time. Different pitches are available to maximize machine usage.

Seco will showcase another advancement in square shoulder milling with its Turbo 10 cutting tool, suitable for most roughing, semi-finishing and finishing operations. The cutter offers improved tool life and precision by optimizing cutting properties that reduce heat generation and cutting forces. Turbo 10 tackles slotting, contouring, helical interpolation and ramping applications with cutting diameters that range from 0.625" (16 mm) to 4" (100 mm) and a maximum cutting depth of 0.354" (9 mm). The Turbo 10 employs a coated, pre-hardened cutter body with two different pitch configurations for meeting tight tolerances and providing high reliability. Mounting types for the Turbo 10 include Cylindrical, Weldon, Arbor and Combimaster. With strong, highly positive geometries, optimized edges, a wide range of industry-leading carbide grades and two different corner radii, the inserts used in conjunction with the Turbo 10 offer maximum levels of adaptability, accuracy and reliability. Furthermore, inserts can be set to a true 90-degree cutting angle to create clean 90-degree walls and eliminate secondary operations. The Double Octomill - 05 brings strong cost and performance benefits to smaller machines. This highly efficient and economical face milling cutter is available in diameters from 1" (25 mm) to 5" (125 mm) and provides optimal performance in applications requiring a maximum of 0.118" (3 mm) depth of cut. The insert pockets of the Double Octomill - 05 incorporate a strong center lock screw and hardened HSS insert locating pins, simplifying the mounting of inserts and ensuring maximum stability during operation. The pockets also feature an 8-degree negative angle to allow the use of double sided inserts, while the inserts themselves are positive, creating a positive cutting rake to minimize power consumption. Seco offers four insert geometries for the Double Octomill - 05, allowing the tool to be successfully applied across various materials.

Seco will expand the disc milling range with the introduction of a new generation of disc cutters for large slot width (1" and 25 mm) fixed pocket cutters. This new generation of disc milling cutters has new features that allow application in all industry segments and include free cutting geometries, wiper flats, a broad range of corner radii, fixed pocket and adjustable versions, and optimized chip flow.

For more information:

Seco Tools Inc.
2805 Bellingham Drive
Troy, MI 48085
Phone: (248) 528-5200
www.secotools.com

Exsys Tool

OFFERS TOOLING SYSTEM INNOVATIONS AT IMTS

Exsys Tool, Inc. will display a variety of tooling system innovations for CNC turning centers in booth W-1664 at IMTS, taking place Sept. 10–15 at the McCormick Place in Chicago. Products to be highlighted include the company's Preci-Flex modular toolholder system and specialized tooling such as a gear hobbing system as well as Double Square Shank and Quad Square Shank static toolholders. The company's Preci-Flex toolholder system is a fast, accurate and cost-effective solution for lathe tooling changeovers. It is the first system of its kind to have a single base holder and multiple tooling adapters that utilize the ER collet pocket. The Preci-Flex's compact design assures maximum torque transmission and rigidity, resulting in increased machining accuracy and improved productivity. IMTS attendees will find Exsys manufactures specialized tooling systems for use with various machine tool models and brands. One such system is a special compact gear hobber system that allows shops to generate splines, spur or helical gears in one operation. As an adjustable toolholding system, it eliminates having to rough gears on one machine, and then transfer them to another for gear hobbing. Built for heavy machining loads, the gear hobber system delivers 45 N-m of torque and speeds of up to 3,000 rpm for hobs or slotting saws up to 2.48" (63 mm) in diameter. Designed to increase tool turret capacity, the Exsys Double Square Shank and Quad Square Shank static specialized toolholders allow for having two or four inserts, as opposed to just one, in a single tool turret station. With multiple insert capability, different types of inserts can be located in the same station, saving time and money.

For more information:

Exsys Tool, Inc.
11654 Corporate Lake Blvd
San Antonio, FL 33576
Phone: (800) 397-9748
www.exsys-tool.com

Bourn and Koch Hobbing Machine

OFFERS SINGLE SETUP OPTIONS

The Bourn and Koch 100 H horizontal hobbing machine can hob splines and geared shafts up to 100 mm in diameter. Mount tools in combinations and cut different gearings on one workpiece or mill keyways and slots in one tool setup without reclamping the workpiece. Since the chip conveyor is located directly under the tool spindle, chips are evacuated immediately from the machine to avoid any thermal distortions. In addition, the 100 H can optionally be ordered with automation for machining larger lots. Star SU plans to exhibit an extended version of the 100 H with a NUM Flexium 68 CNC control at IMTS 2012 in Chicago in September. This extended version can accommodate a workpiece up to 915 mm (36") long and 126 mm (5") in diameter.

For more information:

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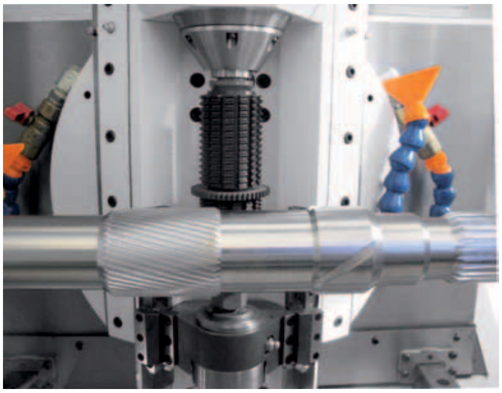
Phone: (847) 649-1450
www.star-su.com



Microprecision

OFFERS NEW GEAR PRODUCTION MACHINES

Hemel Hempstead, U.K.-based Microprecision, a specialist in the field of precision component manufacturing, is now exclusively distributing a high precision and cost-effective range of CNC gear-production machine tools from South Korean company S&T Dynamics. The range of gear production machines, which includes hobbers, shapers, shavers, deburrers and associated accessories, offers quality levels normally seen on top-of-the-range machines, but without the price

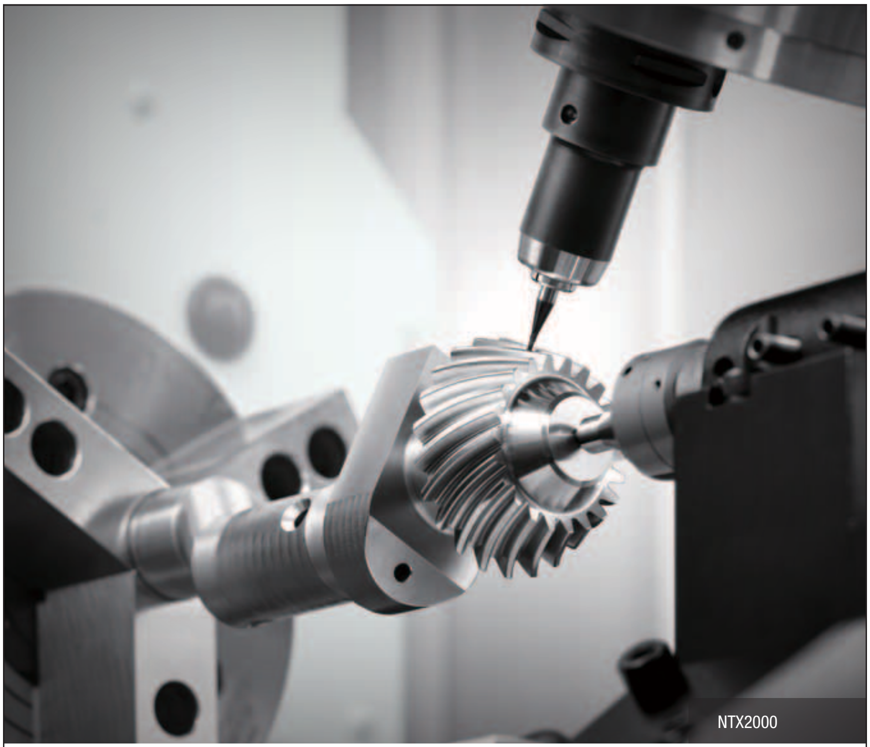
premium. The company's faith in the capabilities of the range is reflected in the fact that it uses two brand new S&T machines in its own 20,000 sq. ft. precision-manufacturing facility; which it uses not only for its own advanced production, but also for customer demonstrations.

Before making the decision to act as a distributor, Microprecision made sure the machines could meet the demands of the aerospace industry—with extensive inspection and testing of machined components before they were shipped. Once the machines and the manufactured components passed Microprecision's exhaustive quality control standards the decision to distribute the machines was given the green light.

The S&T Gear Manufacturing range comprises: GHO-200, -350 & -500 CNC vertical gear hobbing machines, each with corresponding bed capacity. This range of machines offers high-precision capabilities thanks to low surface pressure and high cutting resistance. With a Global User Data (GUD) resource, separate programming is unnecessary and gear production is easy—all that is required is the cutter specifications, the workpiece specifications and the cutting conditions.

There are two GHO-200 machines installed at Microprecision in constant use and available for inspection and machining trials. Having the two machines in operation in the U.K. by the vendor means that any prospective purchasers can draw upon firsthand experience on set-up and operating procedures, plus the capacity to subcontract work and train operators on the machines before new machines are delivered onsite, aiding changeovers and ensuring any purchaser is up and running quickly and smoothly, with the ability to ramp up production progressively.

GSP Series CNC Shaping Machines are also included in the range; these three-axis shaping machines offer high-precision shaping thanks to high-



strength columns and counter columns. They also offer reduced installation space thanks to a front-loading cabinet and the GUD environment for easy programming.

GSV Series CNC Shaving machines are next in the range; the 4.9 kW spindle motors and three-axis motion combine to offer power, easy operation and high precision. They also come complete with a magnetic chip separator to aid chip processing. The GSV range also uses the GUD environment for easier programming.

TCR Chamfering (Deburring) machines offer high strength, high-precision cutting spindles. An inverter also gives them the capability to offer a variable speed transmission for optimal control of the work spindle and work slide. Convenience is improved with the simple arm-type control panel that allows the operator to adjust its position.

TCG Chamfering machines offer the same high strength, high-precision spindle, inverter control, and arm-type control panel as the TCR range. In addition, no separate change gear is needed thanks to the deployment of a settable split plate.

(*Editors' Note: S&T Dynamics' gear machines are sold in the United States by Toolink Engineering,* www.toolink-eng.com).

For more information:

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www.microprecision.co.uk

Jenoptik Wavemore System

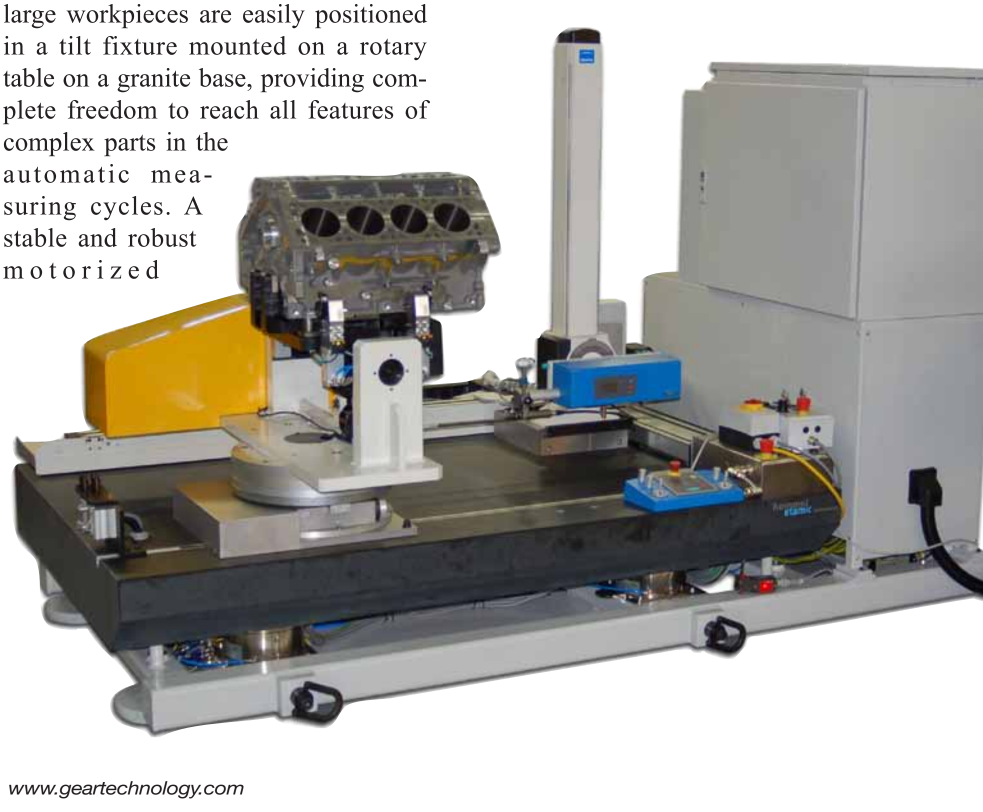
OFFERS HIGH ACCURACY MEASUREMENT

The new Jenoptik Wavemove automated surface roughness and contour measuring system includes up to seven CNC axes for complete high-accuracy measurement and evaluation of complex shaft or prismatic parts such as automotive crankshaft, cylinder heads, engine blocks, transmission housings, and more on the production floor, close to manufacturing processes. Detailed, exportable analyses of the measuring results provide exact statements about the quality of the production process, based on many special parameters from the automotive industry. The automated system eliminates reproducibility concerns and allows the user to track part quality through the entire manufacturing process.

In the ergonomic measuring station, large workpieces are easily positioned in a tilt fixture mounted on a rotary table on a granite base, providing complete freedom to reach all features of complex parts in the automatic measuring cycles. A stable and robust motorized measuring column automatically positions pick-ups. The system automatically verifies the proper orientation of each part prior to measurement routine.

Roughness and contour measurement is easily accomplished through the intelligent arrangement of probing systems for each type of measurement. The drive bar positions the roughness pick-up even in hard-to-reach measuring positions. The contour probe is easy to change and can be operated parallel to the roughness pick-up if necessary. The Wavemove provides: uniform user interface for roughness and contour evaluations, calculation of all common profile, roughness and waviness parameters (more than 90), evaluation of geometric characteristics such as distances, angles and radii and traverse lengths of 120 mm, or 200 mm, for roughness and contour measurement.

Measurement routines are selectable for specific features, as necessary. The *Turbo Wave* software for roughness and contour measuring provides an icon-based, user-friendly, uniform interface for complex CNC measurement runs and automatic evaluations. Custom screen and print forms are easily designed, allowing efficient creation



of measuring programs, and extensive evaluation of measuring results. *Turbo Wave* software for contour and roughness measuring includes interactive control of the measuring station, individual measuring programs with automatic measurement runs, extensive profile analysis functions for profile and material ratio curves and Fourier analysis and more than 90 roughness and waviness parameters according to ISO 4287 and other ISO and national standards (ASME, DIN, JIS, Motif etc.) plus evaluation of workpiece characteristics including radii, distances and angles.

The Wavemove system can also evaluate 'twist' on ground shafts, according to MBN 31007-07. Twist structures at sealing surfaces occur during grinding and impair the sealing function between the shaft and the sealing ring. The most important properties of these surface structures are measured three-dimensionally and the parameters relevant to the tightness determined.

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Saint-Gobain Abrasives

INTRODUCES NORTON PARADIGM DIAMOND AND CBN WHEELS

Saint-Gobain Abrasives has recently introduced Norton Paradigm Diamond and CBN Wheels. Paradigm wheels feature a new proprietary, patent-pending bond delivering high grinding performance on carbide and high-speed steel round tool fluting, resulting in fast cycle times and low cost per parts. "The new patent pending bond on Paradigm Diamond and CBN wheels enables high performance one-pass flute grinding for highly efficient round tool manufacturing operations," says Matt Simmers, product manager at Norton.

For maximum productivity, new Norton Paradigm wheels are online and offline truable. Wheels are wear/load resistant for grinding on 6 to 12 percent cobalt, and offer better control over core growth. A high grain retention and uniform structure provides a high G-ratio (ratio of material removal rate versus wheel wear) up to 2.5x longer wheel life and a 30 percent higher material removal rate than other superabrasive wheels. Paradigm Diamond and CBN Wheels also offer low specific cutting energy, which enables faster grinding with a lower power draw and less burn. All Paradigm Diamond and CBN Wheels are custom-manufactured to precisely meet end-user requirements. Wheels are available for Anca, Makino, Rollomatic, Star, Walter and other grinding systems. Diamond wheels are available for tungsten carbide and CBN wheels are offered for high-speed steel applications.

For more information:

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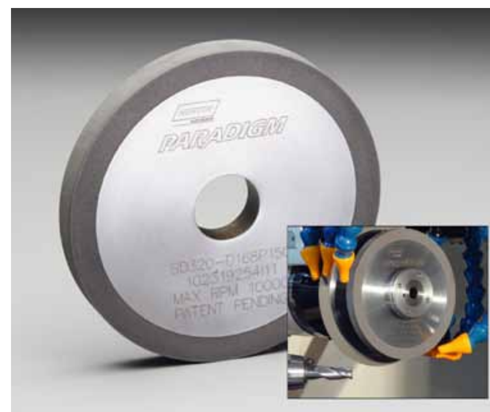

American Wera

EXPANDS OPERATIONS

American Wera represents various German metalworking machine builders, including Profilator, Pittler, Praewema, Diskus, WMZ and MAE. These machines are sold for gear and spline production, as well as bar, pipe and tube straightening plus wheelset pressing. The company's target markets include automotive, off-highway, oil and gas, rail and other heavy equipment manufacturing. Caterpillar recently selected Pittler as manufacturing partner for a new cylinder liner project. The Peoria, Illinois location will use inverted spindle turning centers with 16 spindles for high production. Linamar Mexico added a second

scudding machine from Profilator at its Nuevo Laredo plant for use on a Getrag Ford project. American Wera is expanding its facility in Ann Arbor, Michigan with additional staff to be added soon; full application engineering assistance, sales, service and training offered for customers and field sales representatives. The company will be exhibiting at IMTS in Chicago (Booth N6260) this September.

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Portable CMM

DEBUTS AT HEXAGON 2012

Hexagon Metrology recently announced the North American debut of its new Romer Absolute Arm series. The new series was launched at the Hexagon 2012 conference that took place June 4–7 in Las Vegas. The upgraded design innovations increase the accuracy of the portable coordinate measuring machine (PCMM) by up to 23 percent compared to previous versions. With point repeatability values from 0.016 mm, the Romer Absolute Arm is the most accurate portable measuring arm produced by Hexagon Metrology. Other features include SmartLock technology, which securely locks the arm in its rest position using a simple switch at the base. SmartLock enables users to lock the arm in any intermediate position to ease inspections in physically limited areas. The new version also contains an easy-to-access battery pack to minimize downtime required for battery changes. Absolute encoders recognize the position of the arm at all times, effectively eliminating the need for complex homing procedures. Automatic probe recognition allows operators to change probes within seconds without the need for recalibration. Its optional integrated

scanning solution is factory calibrated and certified with the arm and scanner as a complete system.

The Romer Absolute Arm is available in seven lengths, from 1.5 m to 4.5 m. "The Absolute Arm was already the most accurate arm on the market, but with advancements in design, the new version is even more accurate than before. This pushes the envelope of what is possible with portable measurement technology," states Eric Hollenbeck, product manager for portable technology at Hexagon Metrology. "The most dramatic improvements will be seen on the larger models including our 4 m and 4.5 m arms—the largest in the industry." All new orders will be shipped from the state-of-the-art facility in Oceanside, California with the revised specifications.

For more information:

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Mitutoyo

OFFERS SOLAR ABS DIGIMATIC INDICATOR

The new ID-S Solar ABS Digimatic Indicator available from Mitutoyo America Corporation combines, for the first time in an indicator, the convenience of ABS origin memory with "always ready" solar power to provide highest levels of availability.

The new Mitutoyo ID-S Solar ABS Digimatic Indicator's solar power source can function effectively at light levels as low as 40 lux illumination—significantly below values typically found in even low-light work conditions. In addition the ID-S Solar ABS Digimatic Indicator incorporates a super capacitor to maintain a power reserve of approximately 3.5 hours for instances when even lower light levels are encountered.

The ID-S Solar ABS Digimatic Indicator features Mitutoyo's proprietary ABS (Absolute) measuring system. When the indicator is turned on, ABS automatically restores the most recent origin position. This eliminates the need to re-set the origin at power-on, which can be especially helpful in multipoint measurement—saving time and improving repeatability.

Data-hold and data-output functions are included in the new ID-S Solar ABS Digimatic Indicators enhancing the operator's ability to manage measurement results. Data functions include a lock to prevent mis-operation. The indicator's measuring range is 0.5" (12.7 mm) while its resolution is selectable at 0.0005"/0.01 mm or 0.00005"/0.001 mm.

Additionally, the Mitutoyo ID-S Solar ABS Digimatic Indicator supports output to measurement data applications such as MeasurLink, Mitutoyo's proprietary statistical-processing and process-control program which performs statistical analysis and provides real-time display of measure-

ment results for SPC applications. The program can also be linked to a higher-level network environment for enterprise-wide functionality.

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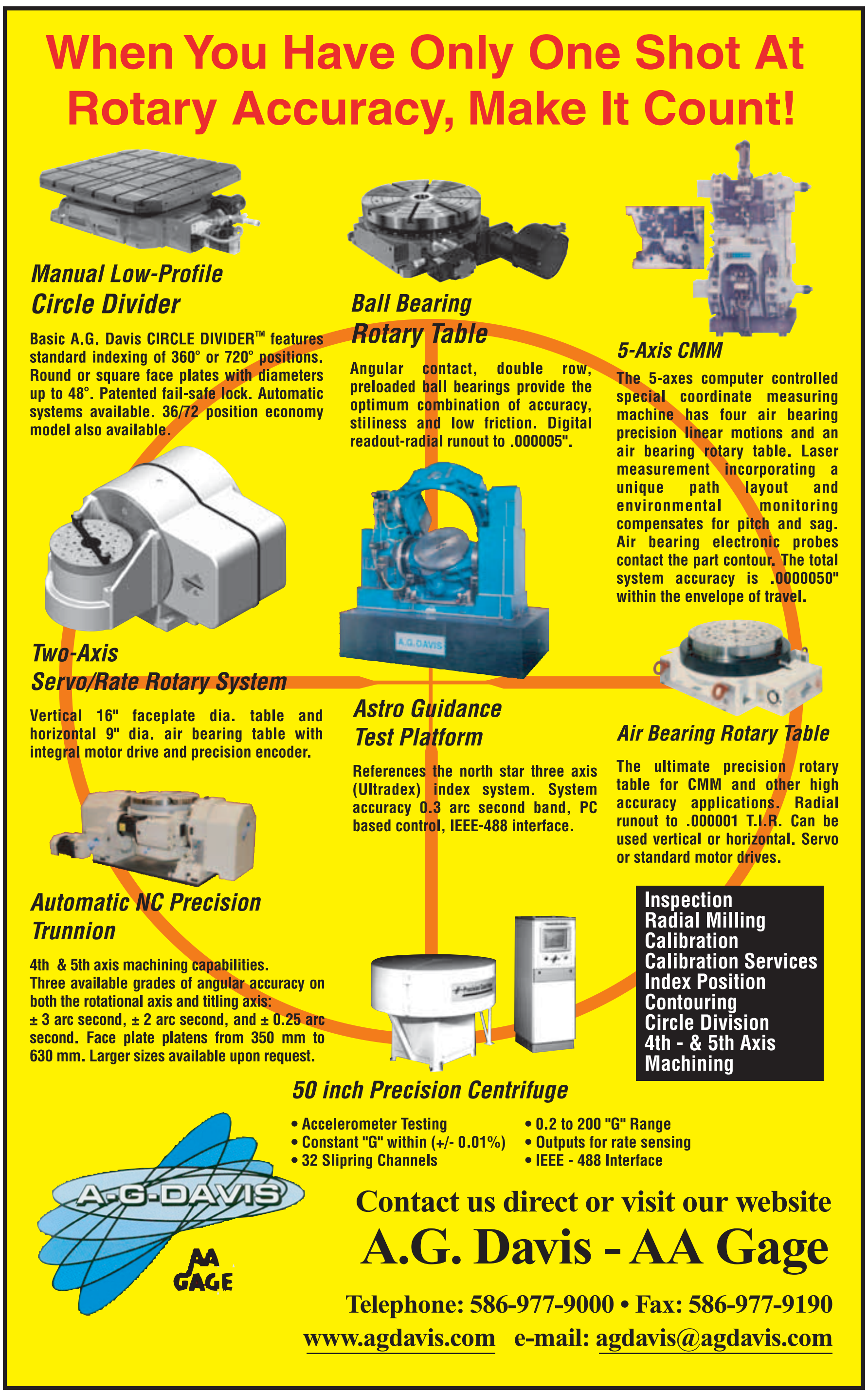
GH Induction Atmospheres

DEVELOPS INDUCTION HARDENING MACHINE



GH Induction Atmospheres (GH IA), part of the worldwide GH Group, has built an induction heating machine that hardens, tempers, and quenches automotive parts on a rotating turntable. A global supplier of quality products for automotive and other industrial markets will use GH IA's new machine to harden automotive ball studs with increased speed, economy, and consistency. The development of this machine marks the first comprehensive technology transfer between GH IA and its new parent, GH Group. GH Group completed the basic system design at its engineering center in Valencia Spain; final design and manufacturing were completed at GH IA's Rochester, New York facility. The new induction hardening machine can process up to 240 parts per hour. "This new machine fulfills the promise of our exciting partnership with GH Group," said Dale Wilcox, GH IA's CTO. "With our combined engineering expertise and GH IA's American manufacturing and service capabilities, this machine is just the first of many innovative induction heating solutions we'll be developing for our customers."

For more information:

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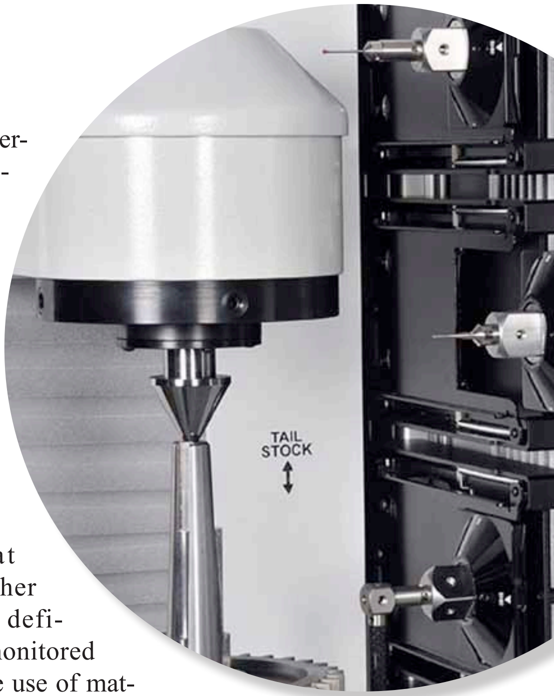
GEAR QUALITY INSPECTION:
How Good Is YoUrs?



How well you conduct your inspections can be the difference-maker for securing high-value contracts from your customers.

And as with most other segments of the gear industry, inspection continues striving to attain "exact science" status. With that thought in mind, following is a look at the state of gear inspection and what rigorous inspection practices deliver—*quality*.

A Brief Introduction

"Today's inspection process must include many methods to determine the overall functionality of a part or assembly," says Kim Gradolf, regional sales manager for Comtorgage Corp. "Simple go/no-go-type gages can be effectively used to verify functional aspects of a part, but other characteristics must be physically measured with dedicated, variable gaging at the machine to insure dimensional acceptability. A computer-based analytical check of a part using a CMM or dedicated machine—such as a gear checker—will usually confirm that the manufacturing process is maintained and the part will function properly in the assembly.

"Visual inspection using optical comparators, cameras, borescopes, etc., may reveal imperfections or flaws that might go unnoticed, and may be unacceptable.

"The high level of manufacturing that is capable today requires many different methods of inspection which must be utilized as a complete process for proper verification of part operation."

And what type of gear poses the greatest challenges regarding its manufacture and inspection?

"Every manufacturer will have issues, given a broad enough customer base," says Dennis Traynor, sales manager for Gleason Metrology Systems Corporation in Rochester, New York. "All gears have challenges—whether fine pitch; large wind energy size-gears with very tight tolerances (and which can deform easily with material handling); or spiral bevel gears with extremely tight tolerances, densely populated flank form grid sizes and supporting multiple manufacturing designs—all are tough applications with no easy answers.

"Although there are many sources of failure/rejection, I would say the quality of the gear blank drives a lot of the quality outcome," Traynor continues. "For instance, if the center bore location is out, the gear is destined to encounter runout (a potential for noise) and probably tooth thickness issues as well—just one example. I believe the bigger issue today is modified profiles and understanding how those profile changes contribute to gear strength/weakness, noise transmission, contact pattern changes and shifts—overall gearbox performance. Low-speed devices, for instance, typically do not have noise issues; however with the move to electric vehicles and gear motors replacing combustion engines—and running at significantly higher speed—noise is definitely a highly monitored issue. Through the use of mating gear contact pattern analysis, these transmission errors can be evaluated."

Step One: Check the Process

One of the most important steps in ensuring part quality is verifying the reliability of the measurement process itself, as in, for example, methods used for testing/verifying the manufacturing process's ability to meet the dimensional tolerance, especially if meeting that requirement is critical to part performance.

"The process is a gage repeatability and reproduceability study," says Louis Todd, president of California-based QC Solutions. "The (GR&R) is the amount of measurement variation introduced by a measurement system, which consists of the measuring instrument itself and the individuals using the instrument."

"A gage R&R study is a critical step in manufacturing Six Sigma projects," Todd says, "and it quantifies three things:

"Repeatability—variation from the measurement instrument

"Reproducibility—variation from the individuals using the instrument

"Overall gage R&R, which is the combined effect of (1) and (2) above

Another way to ensure that quality is being maintained is to use multiple types of checks and different machines and/or gages so that you're not reliant on just one system."

"Whether it is a gear or a spline we are manufacturing, we check the parts analytically for the involute profile, lead, adjacent tooth index variation and accumulated index variation with runout," says Fred Young, owner and operator of Forest City Gear in Roscoe, Illinois. "This is supplemented with measurement over or between wires to determine the tooth thickness or space width. The rule of thumb is to have inspection equipment capable of measuring to 10 percent of the individual tolerance allowed. Further, it is important to incorporate any variables in the manufacturing processes such as production on different machines by different operators or tooling and fixturing. Optimal in this case is to run the job on a variety of machines with different operators and tooling to determine all variables. You have to assume some givens, such as same material, hardness and blank qualities."

"Large gears are usually the most challenging (for inspection)."

*— **Dennis Traynor,** sales manager*
Gleason Metrology Systems Corporation

Photo courtesy of Gleason

Tighter Tolerances Cost More

With the ability of today's equipment to measure more accurately, there is sometimes an inclination to "enhance" an already tight tolerance without regard to the costs to achieve the new tolerance or its effect—or not—on the part performance? We're talking "tolerance creep."

"We always attempt to cut parts as accurately as possible to keep the CPK high and to distinguish our capabilities from those of our peers," says Young. "At the same time, many customers are not terribly sophisticated in gear/spline design and may assign tolerancing that is way too loose or, conversely, that tightens things down excessively, driving the price out of sight."

QC's Todd agrees. "With today's ability to design complex shapes and tighter tolerances in CAD software without the understanding of the ability to manufacture and measure to features, every time a tighter tolerance is applied, the cost to produce and inspect goes up."

"In many cases, just because the equipment or process is capable of holding very close tolerances, it is not always necessary or wise to reduce a tolerance just because it can be done," says Gradolf. "Reducing the tolerance of certain characteristics may result in a need to adjust tolerances of related dimensions of the part."

Gradolf explains: "For example, on an internal spline there is a relationship between the minor diameter, major diameter and the measurement between pins. Dimensional tolerances should be relative to each other. If not, this may cause poor fit, premature wear, poor lubrication and premature failure—just as too 'loose' of a tolerance. And, as mentioned, the additional costs involved to achieve a closer tolerance may not be justifiable. This would include more costly methods of inspection and more sophisticated measuring equipment."

The Importance of Setup

"With most of today's automation in high production manufacturing plants, precautions are in place to eliminate common sources (and) forces (of) contamination/workpiece misalignment," Traynor says. "Still—if the setup is misaligned, there is little hope of a quality gear being produced. On the inspection machine, pictures, graphics, audio and text are used to eliminate the chance for human error."

In addition, he adds, "Both inspection and production need to use quality workholding devices to support the best possible outcome," says Traynor.

On-Board Inspection

CNC onboard charting is another method used for inspection—but there are tradeoffs.

"It is popular to utilize charting equipment on newer gear grinding equipment, but generally, more credence and acceptance is given to checking the gear off the equipment that produced it," Young says. "For example, you usually have a well-defined interval of calibration for gear inspection equipment. It is much harder to keep current with the inspection aboard the cutting equipment. Also, the independent equipment can verify to datum surfaces such as bearing journals or bearing surfaces perpendicular to bores. We also encounter a whole host of parts that require alignments between different sets of gear teeth of some feature external to a tooth or space. Most often, the onboard inspection feature is designed to measure the involute and/or lead without having to remove a part to an independent checker due to size, weight or convenience.

"Perhaps an even more important consideration is to know what the charting you get means and whether the evaluation makes sense. There are a lot of gear companies who are willing to do outside inspection, and this may be a better option. If one does choose to go the in-house inspection route I would strongly recommend you send your inspector(s) to a qualified gear school to learn about gear measuring instruments. There is considerable variation in the experience/expertise of the setup/trainers for this type of gear inspection."

CMMs

And what of the pros and cons in using general purpose CMMs for gear inspection?"Using CMMs for gear inspection is excellent for those who only have a few gears or splines, while the vast majority of their work is positional relationship and sizing for non-gear work," Young says. "The software to check gears can be expensive, as can developing the appropriate probe to use. Is the CMM 3-axis or 4-axis? Are there standard 'artifacts' to verify the accuracy of involute, lead and spacing?

"The multi-sensing CMM systems today provide more data than in the past," Todd says. "Full 3-D scans of the actual part, compared to the CAD nominal or a scanned gear that is put into service and then compared to its actual manufactured specifications, are useful. Visual images can be stored for later recall. Surface finish is still a measurement that is not available on a multi-sensing system."

Gleason's Traynor also points out that "CMM's are typically much slower and much less efficient than dedicated gear inspection machines (GMMs). Typically, this is not their forte; they are best designed for prismatic part inspections." (*Ed.'s Note: See sidebar "Multifunctional Measurement" on page 33.*)

The Latest Technology

"I would hope we all recognize that charters that operate by printing lines on graph paper, with .0002" per-box requiring you to count the number of boxes traversed on the graph paper, is not nearly as informative as utilization of new analytical equipment that gives you a printout of all the individual parameters—sometimes with average values or graphics defining modified involutes (K-charts) or leads," Young says. "One gains much more information to control noise, longevity and wear."

"With the shift to global manufacturing it is imperative that inspection equipment be upgraded to compete with world-class manufacturers who would love to snag some of your customers. I find great reluctance on the part of many to upgrade their inspection capabilities, as they are not convinced it helps get parts out the back door. We use two different brands of analytical checkers so we can measure them against each other. You also made a good point (earlier) about the accuracy of the old plotters, which can be further defined by the cutoffs of the filters in the software of the newer, computerized gear measuring instruments."

100% Inspection

In today's quality-is-king manufacturing environment, is 100 percent inspection becoming the norm?

"This is industry-independent," Traynor says. "Automotive transmissions use composite methods (double-flank rolling) to support identification of defects. They don't tell you specifically what is in error, other than that some problem exists and that elemental analysis should be reverted to for proper diagnostics. Therefore only a smaller sampling of this type of gear passes over an analytical device, usually for process development, improvement and periodic sampling to assure maintained quality levels.

"Whereas with wind energy, every gear is inspected on an 'independent-of-process' inspection machine due to the quality level requirements of the gears, the performance levels the geartrain must support in longevity and durability, and of

"With today's ability to design complex shapes and tighter tolerances... every time a tighter tolerance is applied, the cost to produce and inspect goes up."

— ***Louis Todd,*** *president, QC-Solutions* power transmission. Due to the stringent requirements of all gearboxes in use today, the demand for quality is at an all-time high; whether it's in mining, wind, chemical processing, marine, construction or power transmission, expected service life of these powertrains must meet design requirements and be absolutely reliable. Also, if there are geometric deviations, with the cost of the workpiece blanks being high, identification of the deviations is required."

Big Gears

Very large-diameter gear applications are proliferating, including the gears for wind turbines that Traynor alludes to. As it happens, they are also a BIG pain in the inspection.

"Large gears are usually the most challenging," Traynor allows. "Requirements for workholding, material handling and safety are also issues. The workholding needs to be efficient in order to support workpiece alignment and fine movement. Experience shows us that there is usually no universal device" for pain-free big-gear inspection.

"Large gears—especially internal ring (yaw) gears tend to 'potato-chip' or deform from being moved on 3- or 4-point slings, as well as on conventional spider-arm tooling for workholding. Typically, if the gear is mounted on less than four (slings)—preferably six supporting positions for leveling—it will have deformation quickly—by sagging—thus affecting measurement accuracy. This is to say nothing of trying to center this type of a ring; we have devised tooling to minimize distortion to the ring body."

"Our inspection machines—especially the larger-section 1,500 mm capacity and up—are 5-axis; all 4-linear axes can position to sub-micron levels, and the rotary axis is down to sub-arc second capability, (which is) extremely important to measuring gears of up to three-meter diameter that have tolerances higher than most Swiss watches. The geometry, timing and location tolerances on these parts are extremely critical when the end goal is a 25-year service life of a gearbox some 200 feet in the air. No one wants to bear those service costs, so the final OK has got to be absolute on quality."

"It is popular to utilize charting equipment on newer gear grinding equipment, but generally, more credence and acceptance is given to checking the gear off the equipment that produced it."

– ***Fred Young,*** *president*
Forest City Gear

What's Next

"(The) introduction of other devices to be supported on the inspection device—for instance, surface roughness analysis, Barkhausen noise detection, crack detection—are some of the more frequent test requests," Traynor says. "Non-contact inspection for speed is always on our radar, (but) industrially available sensors do not support current system configurations and require manipulation of the gear for line-of-site measurement."

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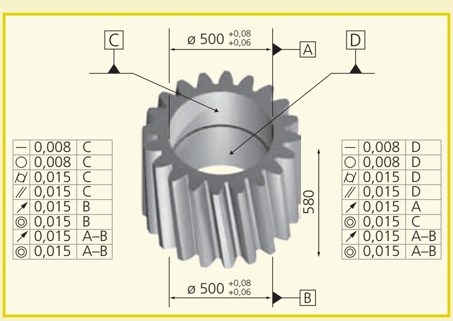
MULTIFUNCTIONAL MEASUREMENT

Klingelnberg's P 150 W tackles demanding wind turbine gear measurement with advanced gear inspection, form measurement and 3-D CMM features

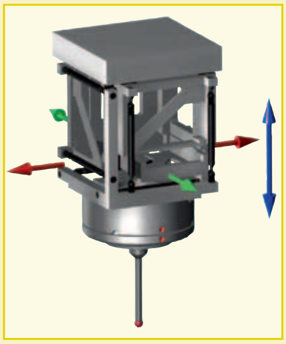
Manufacturers of wind turbine gearboxes continually strive for higher power density in smaller spaces. This means that gear components often have to do double-duty, with internal bores and faces often acting as load-bearing surfaces. Instead of using separate bearings to support the rotating motion, complex bearing geometries are being ground directly into the geared parts. Klingelnberg's P 150 W, based on the familiar P-series machines, was designed with these demanding components in mind.

A typical planet gear from a wind turbine transmission is shown in Figure 1. It includes tolerances for dimension, form and position, in addition to the normal gear tooth measurement requirements. By combining in one machine the ability to measure both gear and bearing features—such as roundness, concentricity and parallelism—the P 150 W allows for significant reduction in measurement cycle times.



The P 150 W also includes these key features:

- ***Vertically aligned boom***. Unlike the familiar P series of machines, the P 150 W uses a vertically aligned boom that enables measurement of internal gears and bearing surfaces with short probe rods.
- ***Unique parallel 3-D kinematics.*** Klingelnberg's 3-D measurement system combines the 3 axes of movement into one nested package; i.e., the mass moved is identical in all directions, ensuring the machine's applicability for both gear inspection and high-resolution form measurement tasks. This is a patented system that Klingelnberg says allows for more precise movement than a traditional CMM's kinematics.



- ***High-precision rotational bearing.*** Klingelnberg's rotational bearings have a static strength of 20 tons and a concentricity precision of less than 0.5 micrometers. This allows for highly precise circular measurements—a benefit both for measuring gear teeth as well as precision, internal bearing surfaces.

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Growing Pains

Renewables Hang on in Turbulent Energy Market

Matthew Jaster, Associate Editor

The turbines are still spinning. They're spinning on large wind farms in the Great Plains, offshore in the Atlantic and even underwater where strong tidal currents offer new energy solutions. These turbines spin regularly while politicians and policy makers—tied up in discussions on tax incentives, economic recovery and a lot of finger pointing—sit idle. Much like the auto and aerospace industries of years past, renewable energy is coping with its own set of growing pains. Analysts still feel confident that clean energy will play a significant role in the future of manufacturing—it's just not going to play the role envisioned four to five years ago.

Wind's Altered Forecast

BTM Consult, an international wind market research firm, reported that wind installations reached a record 41.7 gigawatts last year. And though the consulting firm says the market will continue to grow, it has reduced its cumulative forecast for the five years through 2016 by 14 percent to almost 270,000 megawatts, citing the European credit crisis as denting confidence in the industry (fiscal irresponsibility has altered many European countries' long-term plans for wind energy). Here in the United States, the issue of a Production Tax Credit (PTC) extension remains a priority as wind supporters look to eliminate the yo-yo policies that have plagued the industry in the past.

The Windpower 2012 conference and exhibition that took place in Atlanta from June 3–6 boasted 900-plus exhibitors in an industry town square that spanned nearly five football fields. The theme of the AWEA event was "Manufacturing the Future Today." Show floor chatter included everything from the PTC extension to cutting costs in offshore wind. Everyone agreed that more work needs to be done to meet some of the lofty goals the renewable energy industry has set for itself.

"Today, we have a difficult market environment in the wind industry," says Kerstin Eckert, Siemens AG, Energy Sector. "The main challenges are price pressure, increasing competition and uncertainties regarding the future of subsidy schemes. Still, the wind industry's long-term market perspective is positive."

The SeaGen power plant is located in a strait in the natural harbor Strangford Lough. The turbine is attached to a pile structure swept by the tidal currents, which is anchored to the seabed at a depth of approximately 30 meters. The turbine's rotor blades are driven by the water currents (courtesy of Siemens).

Matt Whitby, global communication and media relations at Vestas, echoes a similar sentiment. “Vestas has communicated that the wind energy market is no longer a growth market as it was just a couple of years ago and as a result we are restructuring our business around that. The financial and economic crisis has added substantial pressure on a number of heavily indebted European countries which are facing demands for a tight fiscal economic policy. Although only very few subsidy schemes for wind power represent public expenditure, short-term considerations may have an adverse impact on the expansion of renewable energy, including wind.”

These restructuring initiatives, however, do not suggest companies are no longer pursuing new wind technologies and products. In fact, a slow growth period typically provides more time to concentrate on future business, which is exactly what companies like Siemens, Vestas and GE Energy seem to be focusing on.

Vestas has recently developed the Gridstreamer technology for its latest 2 MW and V112-3.0 MW wind turbines. This technology allows for full power conversion to meet increasingly tough electrical grid requirements. Vestas is partnering in developing and testing a floating offshore turbine platform, researching the potential of so-called stealth turbines that would not interfere with nearby aviation radar.

Vestas is also developing the V164-7.0 MW turbine for the offshore market. “This will be a game changer for offshore wind energy and will be developed with a focus on the North Sea where Vestas sees the largest opportunities for offshore wind,” Whitby says.

GE Energy has also expanded its wind energy portfolio with a production-based operations and maintenance (O&M) agreement unveiled at Windpower 2012. Historically, wind O&M contracts have provided an availability guarantee measured from time-based availability of wind turbines. GE has moved from time-based

Vestas is focused on cutting down costs in 2012 and beyond for its wind energy products (courtesy of Vestas).

availability to production-based ability with a goal of maximizing production for its customers. Production-based O&M reduces upfront costs, better balances the risk between the owner and service provider and ties directly into GE's annual energy production enhancement initiative.

"By focusing on production, we are adding value for our customers. We are better aligning our goals with theirs and better sharing the risk between GE and our customer. Production-based O&M brings us another step closer to running the turbines like we own them," said Andy Holt, general manager of wind services for GE Energy, in a recent press release.

At Siemens, the focal point is research and development and offshore wind opportunities.

"Innovation is one of the cornerstones of our strategy, and research and development is a high priority for our business," Eckert says. "In 2011, Siemens decided to invest 150 million euros in the further development of its wind business. Two new R&D Centers are currently being set up in the Danish towns of Brande and Aalborg. They will be opened soon."

Siemens launched its new 6 MW direct-drive wind turbine in November 2011. "The new SWT-6.0 is available with rotor diameters of 120 and 154 meters and is designed for the most challenging offshore sites," Eckert says. "The Siemens direct-drive design features 50 percent fewer parts than comparable geared wind turbines. This unique combination of robustness and low weight significantly reduces infrastructure, installation and service costs, and boosts lifetime energy output and profitability."

Siemens' traditional stronghold is the offshore business in wind, with key markets in the U.K., Germany and Scandinavia. Onshore, the company is particularly strong in North America and Europe. "Siemens and Shanghai Electric intend to set up two new joint ventures to form a strategic alliance with the Chinese wind power market.

The Predictable Tidal Turbine

One key area of potential growth in renewable energy is using tidal power as a source for electricity production. "Experts anticipate double-digit growth rates for the ocean power business and they see the (theoretical) global tidal power generation potential as large as 800 TWh/year," Eckert says.

Marine Current Turbines (MCT), which is now part of Siemens, is a producer of horizontal axis marine turbines. Since November 2008, the company has successfully generated more than 3 GWh of clean hydro power at its test site, SeaGen. Located in Strangford Lough in Northern Ireland, SeaGen is currently the world's most powerful marine current turbine. "The twin axial flow rotors of SeaGen are mounted on a single vertical structure and are driven by the flow of the tides," Eckert says. "The system looks, from a technology point of view, like an underwater wind turbine. Water has an energy density of more than 800 times that of wind. Twin rotors rotate with the movement of tidal flow and the blades can be pitched through 270 degrees to work optimally in both tidal directions."

This energy source is advantageous thanks to its predictability. "Being driven by the gravitational pull of the moon and sun, tidal cycles and flow speed are known in advance, making the power output predictable," Eckert says. "The power output of the systems could be calculated for centuries ahead."

Siemens invested in MTC to help commercialize the technology, and the company expects ocean power will be an attractive renewable technology. The first tidal turbine started operation in 2003 in the Bristol Channel, called SeaFlow (this pilot plant had a capacity of 300 kW). Five years later, MCT installed SeaGen.

"In two years the installation of two arrays is planned with four to five tidal turbines each," Eckert says. "Siemens is expecting that in 2020 tidal turbines with a combined capacity between 1 to 2 gigawatts could be installed. The price of electricity generated with these tidal turbines is similar to power offshore wind plants today."

Another project featuring several marine current turbines is at the planning stage: The eight-megawatt Kyle Rhea facility is to be built in a strait between the Scottish mainland and the Isle of Skye. Coastal regions with strong tidal currents such as those in the U.K., Canada, France and East Asia have major potentials for the utilization of this technology.

"The marine environment is extremely challenging," Eckert says. "It is still complex to execute these projects and harness the power of fast flowing currents in the open sea. The need for reliable, predictable low-carbon power gives a new impetus to technological development in this field.

What's Next?

Despite widespread economic turmoil, a recent report from the Renewable Energy Association (REA) believes now is the time to invest in renewable sources, according to Leonie Greene, REA head of external affairs.

"Renewable energy is now a very big business internationally. From householders looking to turn their homes into micro power stations, right through to the biggest investment banks in the world, millions of people are investing in renewable energy. Failure to wake up to the vast opportunities and get fully behind the domestic industry will leave us lagging even further behind our international competitors."

So what needs to be done to make renewable energy competitive with other segments?

In 2012, with the cost challenges faced by the whole wind industry, Vestas is focused on taking the cost out of producing wind turbines, as well as on technologies which will earn a profit for Vestas in 2012.

"Our overall objective is to meet our customers' main requirements: lowering the cost of generating energy from the wind and doing so predictably and reliably," Whitby says. "Vestas is 100 percent in the wind energy business. Regarding other key segments, Vestas is working with customers not traditionally engaged in the energy industry, such as large corporations, that may be looking to expand their business or lower their carbon footprint.


The global potential for power generation using tidal power plants is estimated at 800 terawatt-hours (TWh) per year. That is equivalent to approximately one-quarter more than Germany's entire power demand and equals three to four percent of power consumption worldwide (courtesy of Siemens).

Cost is also the emphasis at Siemens.

"We need to make wind competitive with traditional energy sources, because price pressure is growing and wind power cannot be dependent on subsidies forever," Eckert says. "The industry will need to invest massively in innovation and industrialization. But these investments will only be realized if companies have a stable, predictable and profitable project pipeline."

Got a Gear Question?

Ask the Expert!

Welcome back to *Gear Technology*'s Ask the Expert—a regular feature intended to help designers, specifiers, quality assurance and inspection personnel in addressing some of the more complex, troublesome gearing challenges that never cease to materialize—whether on the drafting table or the shop floor. Simply e-mail your question—along with your name, job title and company name (if you wish to remain anonymous, no problem)—to: *jmcguinn@geartechnology.com;* or, you can submit your question by visiting *geartechnology.com*.

QUESTION #1

Runout and Helix Accuracy

Regarding inspection: Why does DIN 3962 specify F beta tolerances, taking into consideration only the gear face width? Should not consideration also be given to the influence of gear diameter due to the runout?

Durval Baroca, gear process specialist
ZF do Brasil

Dear Durval,

All standards for accuracy of gears, such as DIN 3962, ISO 1328 and AGMA 2015, provide a series of classes, or grades, that assign different levels of tolerances for aspects of gear quality. These relate to analytical parameters such as involute, helix (or lead), pitch variation, accumulated pitch variation and runout.

These different levels of tolerance have been developed over time, by experience gained with different manufacturing methods such as cutting, shaving and grinding. Function in final applications has also been considered.

You mention *F* beta, which is the symbol for helix. You are indeed correct in recognizing that runout (F_r) has an influence on helix accuracy. However, it doesn't matter what causes the error; the only concern is that it doesn't exceed the tolerance—regardless of cause. The design engineer presumably determined which level of accuracy would meet his requirements for transmission error or functional life, and picked an appropriate accuracy grade.

When looking at the analytical gear inspection charts, the shape of the traces for helix—as well as the amplitude—can be very informative as a diagnostic tool. Typically, four teeth—approximately 90° apart—are inspected on both sides.

If all of the traces have a slope error and are parallel, your problem is from the cutting or grinding machine set-up.

Table 3. Test Gear Quality (without eccentricity)
(Variations in .0001 inch)

	Left flank	Right flank
Runout Variation	0.8	
Profile Variation between 10 and 40° roll angle	-0.3 to -1.1	-0.2 to -1.0
Total Index Variation	1.5	1.0
Maximum Pitch Variation	0.7	0.5
Tooth Alignment (Lead) Variation over 3.5 inch face width	-1.8 to -3.3	-3.8 to -4.3

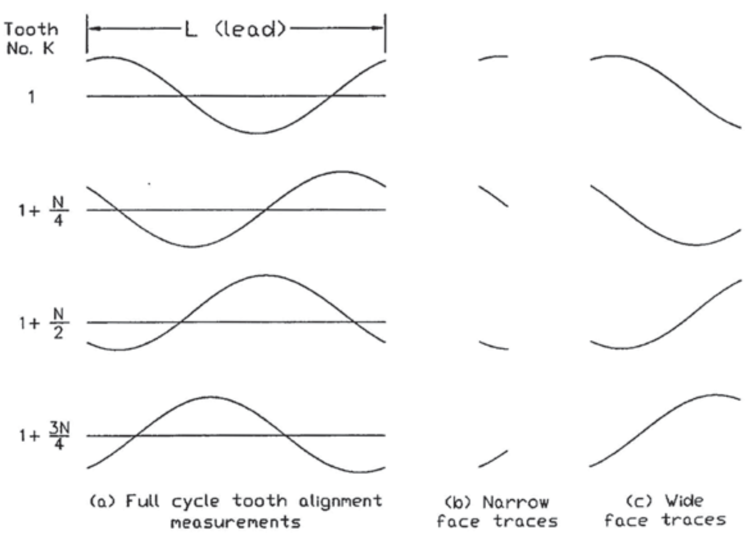


Figure 13: Tooth Alignment Measurements

If they all vary in slope direction from one trace to another, the problem is probably from runout.

An AGMA technical paper (93FTM6), "Effect of Radial Runout on Element Measurements," by Irvin Laskin, Robert Smith and Edward Lawson, explains this in detail (available at *www.agma.org*). To illustrate the effects, a gear was made of master gear quality ("Table 3"). It also was made with a 3½-inch-long face width in order to show the resulting sinusoidal-type helix traces when .0034" run-out was introduced in the inspection. Continuing with the same referenced paper, "Figure 13" shows the theoretical result and "Figure 14" shows the actual, measured results. "Figure 13" also shows what the tooth alignment would look like if it had a narrow face width; only short pieces of the sinusoidal wave would show, looking like typical helix error—but at different angles for each tooth.

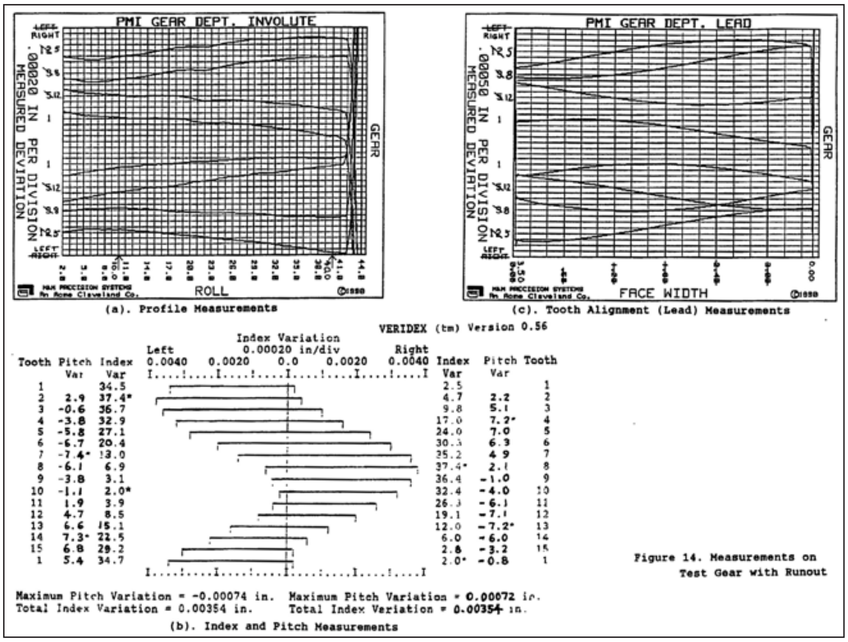


Figure 14. Measurements on Test Gear with Runout

This gear, as checked with runout of 88.9 micrometers, would meet a DIN Class 11. The tolerance for helix for this DIN Class–11 gear would be 100 micrometers; the measured maximum *F* beta equals 73.6 micrometers.

In conclusion, it only matters whether the traces fall within the allowable limit for any prescribed grade or class. Looking at the characteristic of the traces can be useful when determining what to fix in bringing the gear within tolerance.

The above remarks also apply to involute traces that are either parallel or have varying slopes.

Best regards,

Robert E. Smith

Robert Smith is a *Gear Technology* technical editor and gear industry consultant (*gearman@resmith.com; www.resmithcoinc.com*), and is an active member of AGMA's Gear Accuracy and Calibration committees.

QUESTION #2

Maximum Number of Teeth in a Shaper Cutter

How can I determine the maximum number of teeth in a shaper cutter to avoid trimming or rub on up-stroke when cutting an internal gear?

Marv Snider, senior manufacturing engineer
Allison Transmission

Dear Marv,

The easy answer is to direct you to Table 12 in my book, *An Introduction to Gear Design* (available for free download at my website, *www.beytagear.com*). This chart (reprinted here) was developed using the method described in professor Earle Buckingham's landmark book, *Manual for Gear Design*, Section 2—"Spur and Internal Gears" (The Industrial Press, copyright 1935; copyright renewed 1962; reprinted 1981). On pages 40–43, Buckingham describes those places where interference or trimming can occur and presents mathematical methods for determining whether or not they occur for a given cutter and gear combination. Unfortunately, many manufacturing engineers are not familiar with Buckingham's book, or are unable to obtain a copy of it. Check online book sellers for a used copy.

Guidance may also be available from your cutter supplier or machine builder. I developed Table 12 *(page 46)* to avoid having to perform the calculations frequently and to aid in the design of parts that are manufacturing-friendly. It helps avoid tooth numbers that require special cutters.

Best regards,

Charles D. Schultz, PE

Charles Schultz is a *Gear Technology* technical editor and gear industry consultant (*chuck@beytagear.com*; *www.beytagear.com*), and is an active member of AGMA's Helical Gear Rating, Epicyclic Enclosed Drives and Wind Turbine Gear committees.

continued

Table 12
Shaper Cutter Teeth vs. Minimum Internal Gear Teeth to Avoid Interference

# teeth (Nc) in cutter	# teeth in internal gear (Ng) 14.5° FD	20°FD	20°Stub	25°FD	30° FF	30° FR
3	15	11	11	10	9	9
4	18	13	13	12	12	10
5	20	15	15	14	13	11
6	22	17	16	15	14	12
7	24	19	17	17	16	13
8	26	21	18	19	17	14
9	28	23	20	21	18	15
10	30	24	22	22	19	16
11	32	26	23	23	20	17
12	34	27	24	24	21	18
13	36	29	25	25	22	19
14	38	31	26	26	23	20
15	39	33	27	27	24	21
16	41	34	28	28	25	22
18	44	36	30	30	27	24
20	47	38	32	32	29	26
21	49	39	33	33	30	27
24	54	42	36	36	33	30
25	55	43	37	37	34	31
27	58	45	39	39	36	33
28	59	46	40	40	37	34
30	62	48	42	42	39	36
over 30	Nc +32	Nc + 18	Nc + 12	Nc + 12	Nc + 9	Nc + 6

Note: Small shank cutters (Nc<10) do not produce a true involute form.

CLARIFICATION

In the May *Gear Technology* Ask the Expert feature, we failed to mention that Robert Wasilewski, design engineering manager at Arrow Gear Company and one of our valued Experts, also serves as current chairman of the AGMA Bevel Gearing Committee. *Gear Technology* regrets the omission.

Asymmetric Gears: Parameter Selection Approach

Dr. Alexander L. Kapelevich

(An earlier version of this paper was first presented at the International Conference on Power Transmissions (ICPT 2011), Xi'an, China)

Introduction

In many gear transmissions, a tooth load on one flank is significantly higher and is applied for longer periods of time than for the opposite one; an asymmetric tooth shape reflects this functional difference.

There are publications addressing gears with asymmetric teeth (Refs. 1–2) where the tooth geometry is defined by the pre-selected asymmetric generating (tooling) gear rack parameters. A similar approach is commonly used in traditional gear design of conventional gears with symmetric teeth. With asymmetric gears, the standard symmetric tooling gear rack is modified by altering the pressure angle of one of its flanks. However, such a simplified approach to asymmetric gear design greatly limits opportunities to maximize performance for a wide variety of possible applications for these gears.

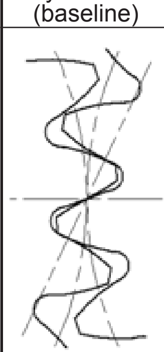
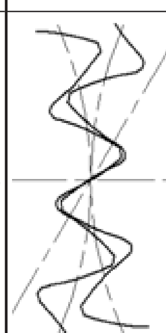
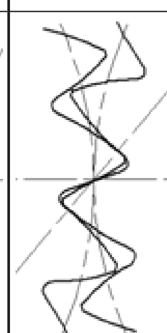
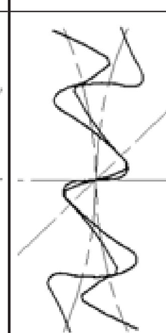
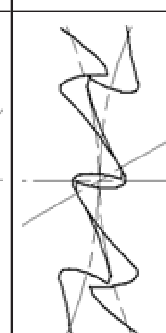
The alternative Direct Gear Design method is not bound by the preselected basic rack parameters and provides asymmetric gear tooth geometry optimized for specific gear drive applications (Refs. 3–4).

This paper describes an approach that rationalizes the degree of asymmetry (or asymmetry factor *K*) selection to meet a variety of operating conditions and requirements for custom gear drives.

Torque Transmission Conditions

Table 1 presents different torque transmission examples of spur gear pairs with identical, 24-tooth mating gears to illustrate bi-directional and unidirectional drive applications.

Table 1—Torque transmission examples of spur gear pairs with identical, 24-tooth mating gears

Case #	1	2	3	4	5
Load transmission	Bidirectional		Mostly unidirectional	Unidirectional	
Loaded flanks	both	both	drive, coast with lower load	drive, coast with no load	drive flank only
Tooth profile	Symmetric (baseline)	Symmetric	Asymmetric	Asymmetric	Asymmetric
Gear mesh					
Pressure angle, °	25	32	40/24*	46/10*	60/-**
Asymmetry coefficient	1.0	1.0	1.19	1.42*	-**
Contact ratio	1.35	1.2	1.2/1.44*	1.2/1.0	1.2/-**
Hertz contact stress, %	100	92	88/102*	86/150*	94/-**
Bearing load, %	100	107	118/99*	130/92*	181/-**
Specific sliding velocity, %	100	94	75/108*	68/97*	49/-**

* for drive/coast tooth flanks;
** coast flank mesh does not exist.

Examples 1 and 2. The gear teeth are symmetric and their surface durability is identical for both tooth flanks. Example 1 presents the traditionally designed, 25º pressure angle gears with full radius fillet. This example is considered as a baseline and its Hertzian contact stress, bearing load and specific sliding velocity are assumed as 100% for comparison with other gear examples. This type of gear profile is used in the aerospace industry because it provides better bending strength and flank surface endurance in comparison with the standard, 20º pressure angle gears typical for commercial applications. Example 2 uses high 32º pressure angle symmetric gears, optimized by the Direct Gear Design method. Their Hertzian contact stress is about 8% lower and the specific sliding velocity is about 6% lower than for the baseline gear pair. This should provide better flank tooth surface pitting or scoring resistance. However, the bearing load is 7% higher.

Example 3. These asymmetric gears are for mostly unidirectional load transmission with a 40º pressure angle driving tooth flanks providing 12% contact stress and 25% sliding velocity reduction. At the same time, the contact stress and sliding velocity of the coast flanks are close to these parameters of the baseline gears and should provide the tooth surface load capacity similar to the baseline gears. These types of gears may find applications for drives with one main load transmission direction, but they should be capable of carrying a lighter load for shorter periods of time in the opposite direction.

Example 4. These asymmetric gears have a 46º drive pressure angle that allows a reduction in contact stress by 14% and sliding velocity by 32%. A disadvantage of such gear teeth is a very high, i.e., + 30%, bearing load. These types of gears are only for unidirectional load transmission. Their 10º coast pressure angle flanks have insignificant load capacity. These types of gears may find applications for drives with only one load transmission direction that may occasionally have no-load coast flank tooth contact, as in the example of a tooth bouncing in high-speed transmissions.

Example 5. Asymmetric gears have only driving tooth flanks with the extreme 60º pressure angle and with no involute coast tooth flanks at all. As a result the bearing load is significant.

Asymmetry Factor Selection

Gear Pair

The gear asymmetry factor K is:

$$\mathrm{K} = \frac{\cos \alpha_{wc}}{\cos \alpha_{wd}} \tag{1}$$

where:

α_{wd} = drive pressure angle;

α_{wc} = coast pressure angle

There are many applications—e.g., Example 3—where a gear pair transmits load in both load directions, but with significantly different magnitude and duration (Fig. 1). In this example the gear asymmetry factor K can be defined by equalizing the potential accumulated tooth surface damage that depends on operating contact stress and the number of the tooth flank load cycles. In other words, the contact stress safety factor S_H should be the same for both the drive and coast tooth flanks. This condition can be presented as:

$$S_H = \frac{\sigma_{HPd}}{\sigma_{Hd}} = \frac{\sigma_{HPc}}{\sigma_{Hc}} \tag{2}$$

where:

σ_{Hd} and σ_{Hc} = operating contact stresses for the drive and coast tooth flanks

σ_{HPd} and σ_{HPc} = permissible contact stresses for the drive and coast tooth flanks that depend on the number of load cycles

Then from Equation 2:

$$\frac{\sigma_{Hd}}{\sigma_{Hc}} = \frac{\sigma_{HPd}}{\sigma_{HPc}} \tag{3}$$

The contact stress at the pitch point (Ref. 5) is:

$$\sigma_H = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_{w1} b_w} \frac{u \pm 1}{u}} \tag{4}$$

where:

$$\mathrm{Z_H} = \sqrt{\frac{2\cos(\beta_b)\cos(\alpha_{wt})}{\cos(\alpha_t)\sin(\alpha_{wt})}} = \text{zone factor that for the directly designed spur gears is} \quad Z_H = \frac{2}{\sqrt{\sin(\alpha_w)}} \tag{5}$$

Z_E = Elasticity factor that takes into account gear material properties (modulus of elasticity and Poisson's ratio)

Z_ε = Contact ratio factor; its conservative value for spur gears is $Z_\varepsilon = 1.0$

Z_β = Helix factor, for spur gears $Z_\beta = 1.0$

F_t = Nominal tangent load, that at the pitch diameter d_{w1} is

$$F_t = \frac{2T_1}{d_{w1}}$$

T_1 = Pinion torque

b_w = Contact face width

sign "+" for external gearing; sign "-" for internal gearing.

For the directly designed spur gears, the contact stress at the pitch point can be presented as

$$\sigma_H = Z_E \frac{2}{d_{w1}} \sqrt{\frac{2T_1}{b_w \sin(2\alpha_w)} \frac{u \pm 1}{u}} \tag{6}$$

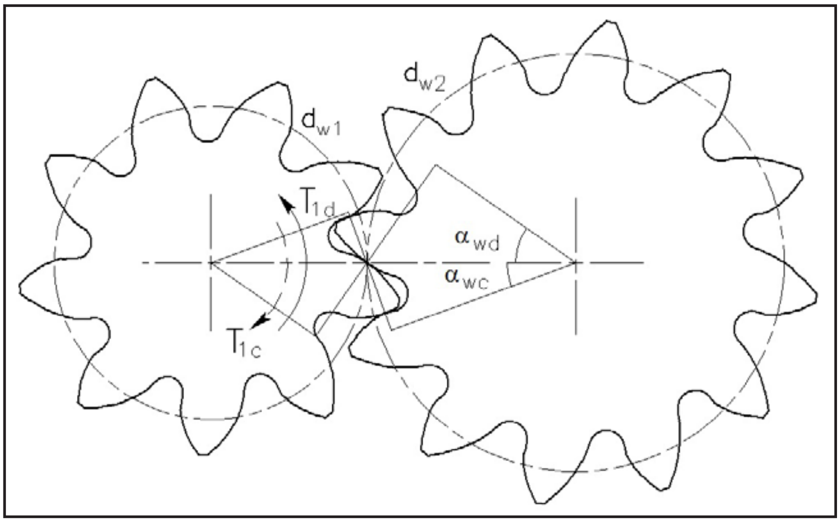


Figure 1—Asymmetric gear pair.

Now this equation can be presented for the drive and coast flank contact, and be used for Equation 3:

$$\frac{\sin(2\alpha_{wc})}{\sin(2\alpha_{wd})} = A \tag{7}$$

where:

$$A = \frac{T_{1c}}{T_{1d}}\left(\frac{\sigma_{HPd}}{\sigma_{HPc}}\right)^2 \tag{8}$$

= coefficient that reflects a difference in the applied load and number of cycles for the drive and coast tooth flanks

T_{1d} and T_{1c} = pinion torque applied to the drive and coast tooth flanks

According to Reference 5, "The permissible stress at limited service life or the safety factor in the limited life stress range is determined using life factor Z_{NT}." This allows substitution of the permissible contact stresses in Equation 8 for the life factors:

$$A = \frac{T_{1c}}{T_{1d}}\left(\frac{Z_{NTd}}{Z_{NTc}}\right)^2 \tag{9}$$

When the coefficient A is defined and the drive pressure angle selected, the coast pressure angle and asymmetry coefficient are calculated by Equations 7 and 1, accordingly.

If the gear tooth is equally loaded in both main and reversed rotation directions, both the coefficient A and the asymmetry factor K are equal to 1.0 and the gear teeth are symmetric.

Example 1: The drive pinion torque T_{1d} is two times greater than the coast pinion torque T_{1c}. The drive tooth flank has 10^9 load cycles and the coast tooth flank has 10^6 load cycles during gear drive life. From the *S-N* curve (Ref. 5) for steel gears, an approximate ratio of the life factors $Z_{NTd}/Z_{NTc}=0.85$. The coefficient $A=0.85^2/2=0.36$. Assuming the drive pressure angle is $\alpha_{wd}=36°$, the coast pressure angle from Equation 7 is $\alpha_{wc}=10°$ and the asymmetry factor from Equation 1 is $K=1.22$.

In many unidirectional gear drives such as, for example, where propulsion system transmissions may seem irreversible, the coast tooth flanks are loaded due to the system inertia during the drive system deceleration or the tooth bouncing in the high-RPM drives. This can be significant and should be taken into consideration while defining the asymmetry factor K.

If the gear drive is completely irreversible and the coast tooth flanks never transmit any load (Examples 4 and 5), the asymmetry factor is defined exclusively by the drive flank geometry. In this example an increase in the drive flank pressure could be limited by a minimum selected contact ratio and by separating load applied to the bearings. Application of a very high drive flank pressure angle results in reduced coast flank pressure angle and, possibly, an involute profile undercut near the tooth root. Another limitation of the asymmetry factor of the irreversible gear drive is a growing, compressive bending stress at the coast flank root. As is typical for conventional symmetric gears, compressive bending stress does not present a problem because its allowable limit is significantly higher than that of tensile bending stress. However, for asymmetric gears it may become an issue—especially for gears with thin rims.

Gears in Chain Arrangement

In the chain gears, the idler gear transmits the same load by both tooth flanks. While this arrangement may seem unsuitable for asymmetric gear application, in many examples the idler's mating gears have a significantly different number of teeth (Fig. 2). This allows application of asymmetric gears to equalize the contact stress and to achieve maximum load capacity.

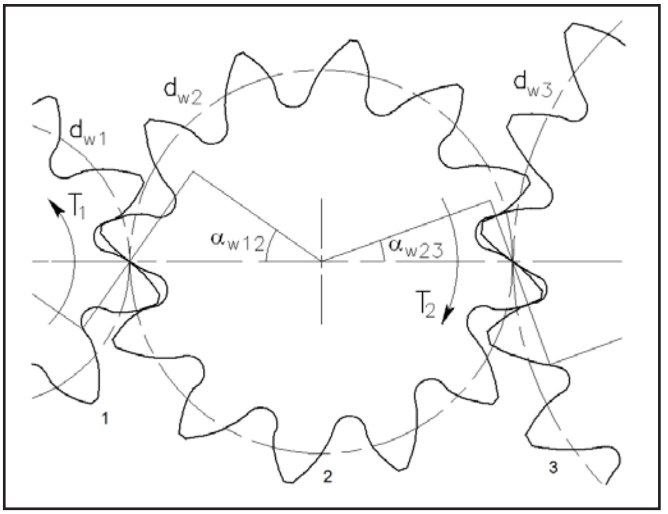


Figure 2—Chain gear arrangement: 1 = input pinion; 2 = idler gear; 3 = output gear.

Equation 6 is used to define the pitch point contact stress in the pinion/idler gear mesh:

$$\sigma_{H12} = Z_E \frac{2}{d_{w1}}\sqrt{\frac{2T_1}{b_{w12}\sin(2\alpha_{w12})}\frac{u_{12}+1}{u_{12}}} \tag{10}$$

and in the idler/output gear mesh:

$$\sigma_{H23} = Z_E \frac{2}{d_{w2}}\sqrt{\frac{2T_2}{b_{w23}\sin(2\alpha_{w23})}\frac{u_{23}+1}{u_{23}}} \tag{11}$$

or, ignoring gear mesh losses:

$$\sigma_{H23} = Z_E \frac{2}{u_{12}d_{w1}}\sqrt{\frac{2u_{12}2T_1}{b_{w23}\sin(2\alpha_{w23})}\frac{u_{23}+1}{u_{23}}} \tag{12}$$

where the subscript indexes "12" and "23" are for the pinion/idler gear and the idler/output gear meshes, accordingly.

If all three chain gears are made from the same material, a condition $\sigma_{H12}=\sigma_{H23}$ describes the equal potential for accumulated tooth surface damage of the idler gear flanks. Using Equations 11 and 12, this condition can be presented as:

$$\frac{\sin(2\alpha_{23})}{\sin(2\alpha_{12})} = \frac{b_{w12}}{b_{w23}}\frac{u_{23}+1}{u_{23}(u_{12}+1)} \tag{13}$$

where:

$u_{12}=n_2/n_1$ and $u_{23}=n_3/n_2$ are the gear ratios

b_{w12} and b_{w23} are the contact face widths in the pinion/idler gear and the idler/output gear meshes, accordingly;

n_1, n_2 and n_3 are number of teeth of the input pinion, idler gear and output gear

When these parameters are known and the drive pressure angle in the pinion/idler gear mesh is selected, the coast pressure angle in this mesh and asymmetry coefficient are calculated by Equations 13 and 1, accordingly.

Example 2: The pinion number of teeth is $n_1 = 9$; the idler gear number of teeth is $n_2 = 12$; the output gear number of teeth is $n_3 = 20$; the contact face width ratio is $b_{w12}/\ b_{w23} = 1.2$; and the pinion/idler gear pressure angle is $\alpha_{w12} = 35^{\circ}$. Then the idler/output pressure angle from Equation 13 is $\alpha_{w23} = 25.3^{\circ}$ and the asymmetry factor K from Equation 1 is 1.104.

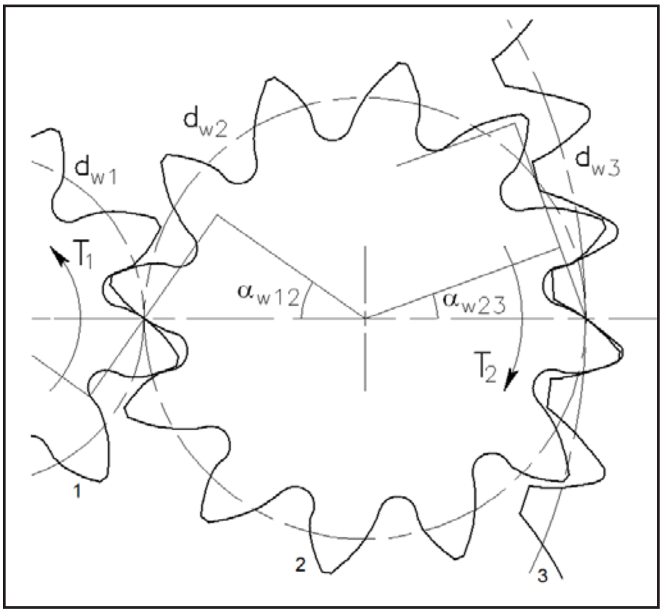


Figure 3—Planetary gear arrangement: 1 = sun gear; 2 = planet gear; 3 = ring gear.

Planetary Arrangement

A similar contact stress equalization technique can also be applied for the planetary gear arrangement (Fig. 3) because the planet gear is considered as the idler gear, engaged with the sun gear and ring gear.

In this example, considering that the planet and ring gears are in the internal mesh, Equation 13 looks like:

$$\frac{\sin(2\alpha_{w23})}{\sin(2\alpha_{w12})} = \frac{b_{w12}}{b_{w23}} \frac{u_{23} - 1}{u_{23}\,(u_{12} + 1)} \tag{14}$$

where:

$u_{12} = n_2/n_1$ and $u_{23} = n_3/n_2$ are the gear ratios

b_{w12} and b_{w23} are the contact face widths in the sun/planet gear and the planet/ring gear meshes, accordingly

n_1, n_2, and n_3 are number of teeth of the sun, planet and ring gears

When these parameters are known and the drive pressure angle in the sun/planet gear mesh is selected, the coast pressure angle in this mesh and asymmetry coefficient are calculated by Equations 14 and 1, accordingly.

Example 3: The sun gear number of teeth is $n_1 = 9$; the planet gear number of teeth is $n_2 = 12$; the output gear number of teeth is $n_3 = 33$; the contact face width ratio is $b_{w12}/\ b_{w23} = 1.8$; and the pinion/idler gear pressure angle is $\alpha_{w12} = 40^{\circ}$.

Then the planet/ring pressure angle from Equation 14 is $\alpha_{w23} = 14.5^{\circ}$ and the asymmetry factor K from Equation 1 is 1.264.

Summary

Selection of the gear tooth asymmetry factor K should be considered, depending on the gear drive application.

For an asymmetric gear pair that has different load application conditions in opposite directions, selection of the asymmetry factor K is based on equalizing of potential accumulated tooth surface damage in both load transmission directions.

For unidirectional chain and planetary gear arrangements, selection of the asymmetry factor K is based on equalizing of potential accumulated tooth surface damage in both flanks of idler (or planet) gear.

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Towards an Improved AGMA Accuracy Classification System on Double-Flank Composite Measurements

E. Reiter

Management Summary

AGMA introduced ANSI/AGMA 2015–2–A06—Accuracy Classification System: Radial System for Cylindrical Gear*s,* in 2006 as the first major rewrite of the double-flank accuracy standard in over 18 years.

This document explains concerns related to the use of ANSI/AGMA 2015–2–A06 as an accuracy classification system and recommends a revised system that can be of more service to the gearing industry.

Introduction

In 2001 AGMA introduced the first of two documents: ANSI/AGMA 2015–1–A01: Accuracy Classification System—Tangential Measurements for Cylindrical Gears, followed in 2006 by the release of ANSI/AGMA 2015–2–A06: Accuracy Classification System—Radial System for Cylindrical Gears. Both of these documents, when combined, officially replace the still widely used standard, AGMA 2000–A88: *Gear Classification and Inspection Handbook*.

Although ANSI/AGMA 2015–2–A06 was adopted by the general membership, it is likely that most members—even to this day—do not fully understand how this document differs from AGMA 2000–A88 in its application to the double-flank measurement of a gear. In the author's viewpoint, some of the improvements that ANSI/AGMA 2015–2–A06 was intended to provide, in fact resulted in just the opposite; i.e., more uncertainty in terms of product quality than what existed previously. It is recommended that ANSI/AGMA 2015–2–A06 be revised to reflect the concerns expressed in this document.

Concerns with ANSI/AGMA 2015–2–A06 and How to Resolve Them

Removal of the long-term component in the calculation of tooth-to-tooth deviations. ANSI/AGMA 2015–2–A06 in its definition of the tooth-to-tooth radial composite deviation, f_{id}, recommends that "The long-term component sinusoidal effect of eccentricity *should* be removed from the waveform before determining the tooth-to-tooth deviation value," whereas the AGMA 2000–A88 document relies on raw data.

This long-term component, its definition and its implications are the crux of the concerns relating to ANSI/AGMA 2015–2–A06. AGMA used techniques from single-flank testing in relation to data filtering and long-form component removal and applied it to double-flank testing on the assumption that its use would be a better predictor of noise quality. This correlation between tooth-to-tooth radial composite deviation measured in this manner and noise has never been proven; rather, as will be demonstrated here, it will cause problems that cast doubt upon any unproven usefulness to predict noise better than AGMA 2000–A88.

The explanation of the measurement and application of the long-term component is detailed in the AGMA Information Sheet 915–2–A05, i.e., "Inspection Practices, Part 2: Cylindrical Gears/Radial Measurements." In using a double-flank tester, AGMA 915–2–A05 recommends filtering of the data either by analog or digital electronic means.

Essentially, the technique is to take all of the collected data and then, by applying a fast Fourier analysis, the data is separated into different orders. All of the orders combined result in the total data collected. Of particular interest in this document is the first-order data that is defined as the long-form component. The first-order data consists of a single, sinusoidal waveform that is calculated based on all of the original data, and is representative of the radial run-out F_r of the gear. AGMA 915–2–A05 recommends that the first-order data be removed from the original measurements for the purposes of reporting the tooth-to-tooth radial composite deviation. An example of the resulting charts is shown in Figure 1, which was extracted from AGMA 915–2–A05. If one mathematically subtracts the sinusoidal waveform of the middle graph Figure 1 from the original data in the top graph, we obtain the filtered result shown in the bottom graph in Figure 1.

AGMA 915–2–A05 recommends this data filtering technique in order to segregate the superimposition of the involute variations from the run-out variations, as is true of the gear manufacturing process where correction of these issues is done individually as well. An example of such an effect is shown in Figure 2 where, in the unfiltered raw data, the tooth-to-tooth variation is exaggerated along the slope of the run-out curve that has the greatest slope, while in the filtered result, a smaller tooth-to-tooth variation is shown.

Some unintentional flaws exist in the approach used in ANSI/AGMA 2015–2–A06. The most significant issue is that in practical use, the maximum tooth-to-tooth deviation rarely occurs exactly at the position of the greatest slope of the run-out curve (Fig. 2, top "Unfiltered"). ANSI/AGMA 2015–2–A06 incorrectly assumes that if the worst tooth-to-tooth occurs in this position, the filtered tooth-to-tooth deviation magnitude will be dramatically reduced, compared to the unfiltered deviation. As a result, the magnitude of the tooth-to-tooth deviation tolerances in relation to the total composite tolerances is greatly reduced in this standard, as compared to AGMA 2000–A88.

In fact, the position of the worst tooth-to-tooth deviation on any given part is independent of the positioning of the sine wave of the run-out curve; as a result, the tooth-to-tooth deviations with filtering are typically not dramatically better (or worse) than the unfiltered results. It is even possible that the magnitude of the filtered tooth-to-tooth variation is larger than the unfiltered variation if the worst unfiltered tooth-to-tooth variation occurs in proximity to the peak or valley of the sine wave of the run-out curve as shown in Figure 3a and Figure 3b, where the unfiltered result was 9 microns and the filtered result was 11 microns. Of particular interest in looking at Figure 3a and Figure 3b is that the position of the worst tooth-to-tooth deviation—as indicated by the vertical

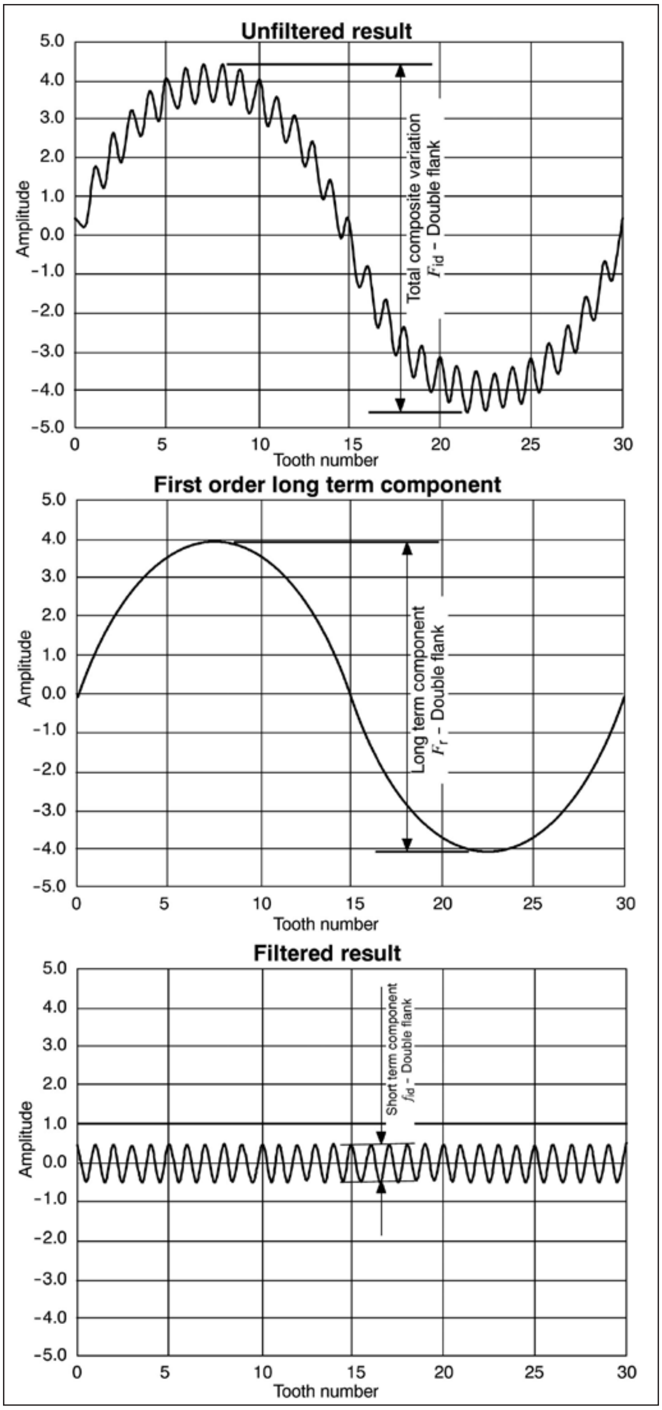

Figure 1—Double-flank, tight-mesh center distance data taken for a 30-tooth gear (extracted from AGMA 915–2–A05).

hashed boundary—is shifted showing two different positions by the two approaches. This is not uncommon; depending on the nature of the gear, the worst tooth-to-tooth deviations can be in completely different parts of the gear when analyzed by the two different methods.

The difference in the result between the filtered and unfiltered tooth-to-tooth deviations are typically only a few microns. In fact, out of hundreds of different gears measured over a range in module of 0.5–3.0, and with tooth counts between 10 and 120, all exhibited similar differences in readings of less than approximately ±0.003 mm. This leads one to question the value of an elaborate filtering technique that requires computerized equipment if the results are not dramatically different than the AGMA 2000–A88 approach. If the results are so similar, how can this actually be a better predictor of noise, as was a stated goal in ANSI/AGMA 2015–2–A06?

There has been no published information to date about how such a technique is a better predictor of noise.

It is recommended that any subsequent change to ANSI/AGMA 2015–2–A06 include a return to the AGMA 2000–A88 approach to measuring tooth-to-tooth deviation, and that the revised standard clearly specifies that filtering of data by electronic or mathematical means is not allowed in determining whether a part meets AGMA accuracy class requirements.

Gears with significant higher-order effects superimposed on the long-form component. Another shortcoming of the ANSI/AGMA 2015–2–A06 approach is that not all gears exhibit well-behaved, tight-mesh center-distance plots where eccentricity is the major effect. Although one can mathematically calculate the first-order sine wave that exists in the data, it is not always the only predominant effect in all gearing.

Consider, for example, a plastic gear that is injection-molded with three gates. It is common to see the effect of a third-order wave from the gates superimposed on the first-order effect of the eccentricity. Yet when this happens, since only the first order effect is removed in the calculation of the tooth-to-tooth deviation, the effect may be an increase in the tooth-to-tooth deviation value reported using ANSI/AGMA 2015–2–A06, as opposed to AGMA 2000–A88—even if the maximum tooth-to-tooth deviation occurs at the greatest slope of the run-out curve.

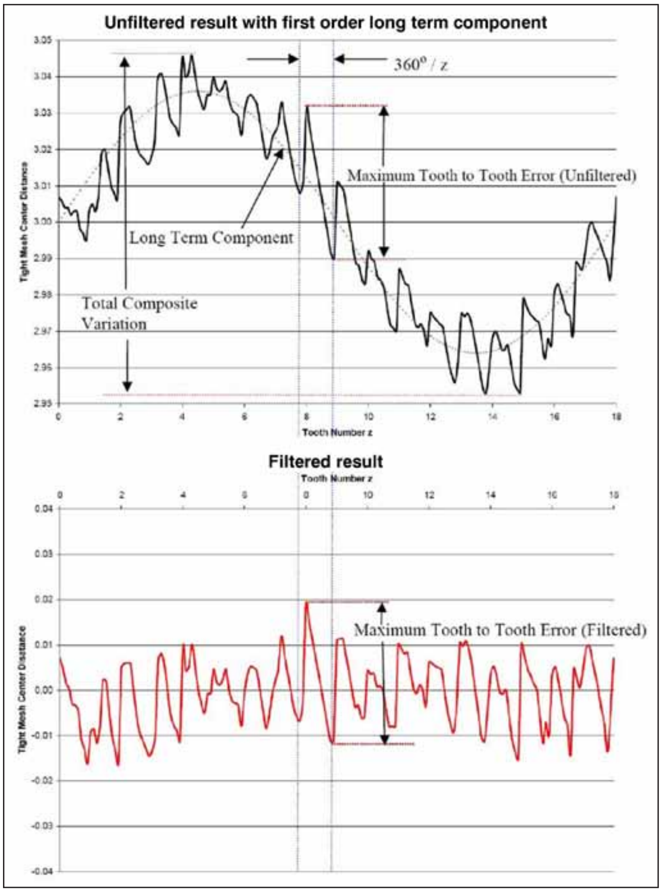

Figure 2—Tight-mesh center-distance showing the highest unfiltered tooth-to-tooth variation along the greatest slope of the runout curve (adapted from ANSI/AGMA 2015–2–A06).

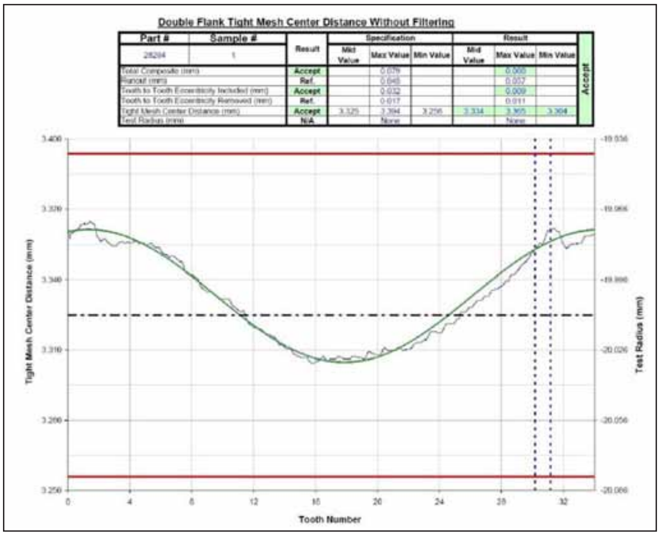

Figure 3a—Unfiltered test showing worst tooth-to-tooth deviation near the peak of the runout curve.

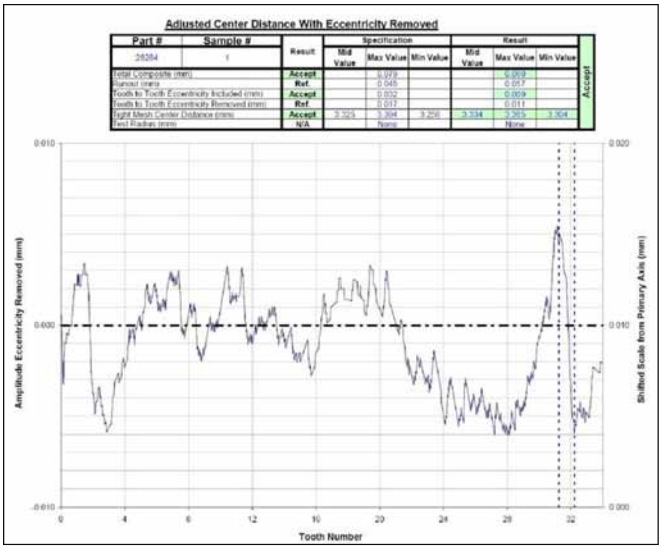

Figure 3b—Filtered test result of Figure 3a with shifted worst tooth-to-tooth position and increased magnitude (displayed with increased vertical axis scale).

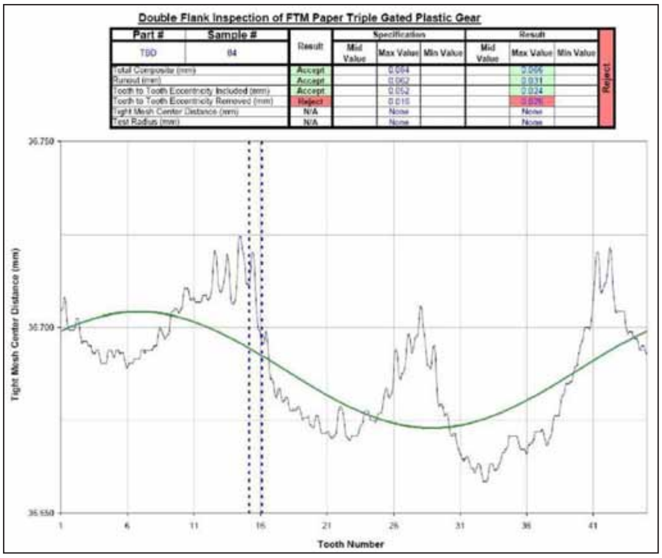

Figure 4a—Unfiltered double-flank test of a triple-gated plastic gear.

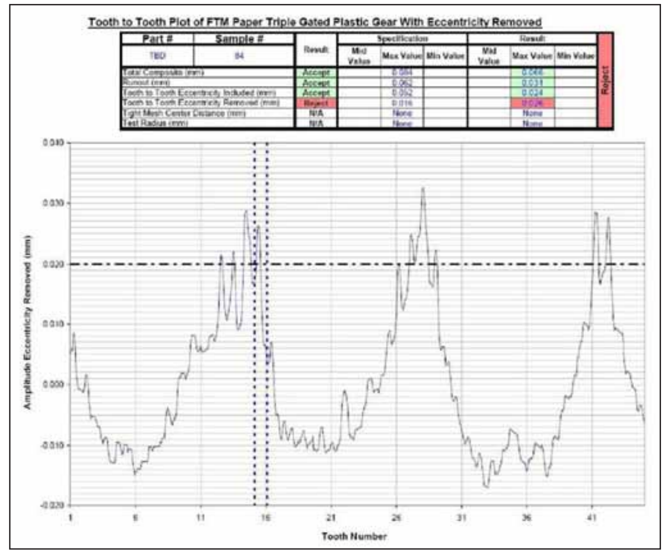

Figure 4b—Filtered double-flank test of a triple-gated plastic gear.

This situation is shown in Figure 4a and Figure 4b for a 45-tooth, 30%-glass-filled nylon gear. In the figure, the peaks in the tight mesh center distance are actually the weld lines between the gates. The three valleys in the tight-mesh center-distance plot are the positions of the gates. Superimposed on the plot is the once-per-revolution run-out curve that generally follows the shape of the tight-mesh center-distance curve. However, one can also clearly see that a higher-order wave is also an overriding effect on the data. The writers of ANSI/AGMA 2015–2–A06 recognized that higher-order waves can be problematic in this filtering technique, but did not have a clear, standardized way to deal with these higher orders, so this issue was ignored in the ANSI/AGMA 2015–2–A06 standard. In Figure 4a and Figure 4b it can be seen that the unfiltered tooth-to-tooth deviation on this part was 0.024 mm, while the filtered result was 0.026 mm. The complexity of filtering does not really provide any significant benefit to the measurement result on this part that, as mentioned, is quite typical amongst all gears measured.

This type of higher-order effect is not just evident in plastic gears with multiple gates. Similar higher-order issues may exist in hobbed and shaved gears; steel gears; heat-treated gears where distortions may be affected by grain direction; ring gears with thin rims; fine blanked gears with irregular shapes on the same part; powder metal gears with lightening holes; or steel gears with lightening holes, etc.

Reduced tooth-to-tooth tolerances compared to total composite tolerance in each accuracy class. The filtered tooth-to-tooth variation was expected by the writers of the standard to be significantly smaller than the unfiltered, tooth-to-tooth variation. Hence, the tooth-to-tooth tolerances of ANSI/AGMA 2015–2–A06—in any accuracy class—are considerably smaller as a percentage to total composite tolerance as compared to AGMA 2000–A88. Table 1 shows examples of how these tolerances are smaller in ANSI/AGMA 2015–2–A06 at a fixed level of 18.52% of the total composite tolerance, compared to AGMA 2000 at 35–60% of the total composite tolerance.

As previously explained, the difference in results between the filtered and unfiltered tooth-to-tooth deviations are only a few microns—not nearly the difference as the tolerances that Table1 would suggest. Most users of ANSI/AGMA 2015–2–A06 are coming to the realization that this shift in tolerances has much larger implications than what was originally anticipated. Under the AGMA 2000–A88 approach,

Table 1—Comparison examples for quality class Q8 to accuracy class C9 and the magnitude of the tooth-to-tooth tolerance (B) compared to the total composite tolerance (A)

	Tolerances per AGMA 2000-A88, Q8			Tolerances per ANSI/AGMA 2015-2-A06, C9		
	Total composite tolerance, mm, A	Tooth-to-tooth tolerance, mm, B	B/A, %	Total composite tolerance, mm, A	Tooth-to-tooth tolerance, mm, B	B/A, %
0.5 module, 10 tooth spur	0.045	0.031	68.9%	0.083	0.015	18.52%
0.5 module, 40 tooth spur	0.047	0.025	53.2%	0.085	0.016	
0.5 module, 60 tooth spur	0.050	0.025	50.0%	0.086	0.016	
1.5 module, 10 tooth spur	0.074	0.041	55.4%	0.086	0.016	
1.5 module, 40 tooth spur	0.077	0.031	40.3%	0.089	0.016	
1.5 module, 60 tooth spur	0.085	0.032	37.6%	0.094	0.017	
2.0 module, 10 tooth spur	0.086	0.043	50.0%	0.087	0.016	
2.0 module, 40 tooth spur	0.089	0.033	37.1%	0.091	0.017	
2.0 module, 60 tooth spur	0.099	0.035	35.4%	0.098	0.018	

most gear designers usually select a quality class level based on the magnitude of the total composite deviation, since it is more difficult to control relative to a tooth-to-tooth deviation in any quality class. Now with tooth-to-tooth tolerances being so small relative to the total composite tolerances, a supplier usually cannot achieve both the tooth-to-tooth requirements and the total composite requirements in the same accuracy class. It is even likely that they may be two accuracy classes apart.

An even greater concern is if the accuracy class is only specified based on the more difficult tooth-to-tooth specification, resulting in an overly generous total composite specification. For example, under ANSI/AGMA 2015–2–A06, a 1.0 module, 20-tooth spur gear using a C11 accuracy class results in a typical tooth-to-tooth tolerance of 0.032 mm, which may be a reasonable tolerance for such a gear. If C11 is also specified for the total composite deviation tolerance, the tolerance would be a whopping 0.171 mm. Most likely, a C9 specification would need to be made for the total composite tolerance at 0.086 mm, which would be more reasonable related to the 0.032 mm tooth-to-tooth tolerance.

However, the majority of users of AGMA accuracy grades do not even realize that a different accuracy grade can be specified for tooth-to-tooth—as opposed to total composite—tolerances. Practically, however, specifying different classes is self-defeating since one can ask the question of why an accuracy classification system is needed at all under those circumstances, as opposed to explicitly stating the tolerances.

It is recommended that a new accuracy class methodology be adopted based on the suggestions outlined below to correct for the condition that exists today in ANSI/AGMA 2015–2–A06.

Step factor in tolerances between accuracy classes. ANSI/AGMA 2015–2–A06 is similar to most other gear classification standards in the step factor between two consecutive accuracy classes. Values of the next higher or lower class are determined by multiplying or dividing the value from the previous class by $\sqrt{2}$ or 1.414. Hence there is a 41.4% step factor between classes, a rather large step factor by today's technological standards. The result is that most of the gearing specified by double-flank inspection falls in relatively few of the 11 available accuracy classes, with most falling between C7 and C10.

When creating a new accuracy standard, it is recommended that the historic use of a $\sqrt{2}$ step factor be reconsidered for a smaller one, thus allowing for the practical use of more accuracy classes. A suggestion would be to consider a step factor of 1.2; i.e., a 20% step factor between classes.

The relationship between the number of teeth, module and tolerances. ANSI/AGMA 2015–2–A06, in comparison to AGMA 2000–A88, is an improvement in reducing the sensitivity of the total composite tolerance value to the number of teeth, and even the module of a given gear. Practical experience shows that module and number of teeth really do not have any influence on the manufacturing capability in gearing. It is suggested that in a future revision to the accuracy standard, the sensitivity can be completely eliminated, which would greatly simplify the standard and be of more practical use as well.

The tooth-to-tooth tolerances, on the other hand, do have some sensitivity to the number of teeth—although not the module. In the extreme case, consider a double-flank test on a one-start worm: its tooth-to-tooth deviation will be identical to its total composite deviation. A two-start worm will most likely have a tooth-to-tooth deviation that is a bit smaller than the total composite deviation. As the number of teeth increases, the tooth-to-tooth deviation tends to drop until some practical limit is reached.

It is recommended that a new accuracy tolerance calculation method be adopted that makes the total composite tolerance a constant value based on each class, and the tooth-to-tooth tolerance a fraction of the total composite tolerance based on the number of teeth in the gear. The suggested calculation method is further outlined below.

The gear accuracy standard should include worms, sector gears, bevel gears and racks. ANSI/AGMA 2015–2–A06 limits the number of teeth to be between 3 and 1,000, and the helix angle to be less than or equal to 45°.

This means that cylindrical worms fall outside the scope of this document. Also, racks are excluded in the sense that the equivalent tooth count on a rack would be infinity. In addition, there is no specific mention in the scope about how to apply the standard to sector gears—or if it even applies.

Worms could be included by creating tooth-to-tooth tolerances that are based on the number of teeth and excluding the helix angle limitation.

Sector gears can be accommodated by fractionally adjusting the tolerances based on the number of actively used teeth in the sector in comparison to the number of teeth in the full circle.

Bevel gears should follow the same tolerance scheme as cylindrical gears.

Racks and gears with more than 200 teeth can be accommodated by using the same tolerances as gears with 200 teeth.

A further explanation of these issues follows.

Using the fast Fourier transform to calculate the long-form component. ANSI/AGMA 2015–2–A06 suggests that in order to calculate the long-form component, a fast Fourier transform be used. This is not a practical solution for the calculation of the run-out curve. Fast Fourier transforms are a shortcut calculation method to a full Fourier transform. This shortcut method was developed in the day of slide rules and very slow computers because the number of calculations involved in a fast Fourier transform is significantly less than what is needed in a full Fourier transform. The problem with fast Fourier transforms is that the data set needs to have an exactly predefined number of data points that go into the calculation. The allowable number of data points is exactly 2^n, where "n" is any whole number. Hence you must have 2 or 4, 8, 16, 32, 64, 128, etc., data points to enter into the calculation. This is a huge chore to do electronically, as it introduces a secondary data filtering issue.

A practical number of data points that can be taken in a double-flank test is about 1,000 points. According to the 2^n

theory, one would need exactly 512 data points, or possibly 1,024 data points. If the apparatus can precisely measure only 512 or 1,024 data points—for whatever reason—which is usually the case due to computer timing issues—then one needs to either first discard data points or add data points by some other mathematical filtering algorithm. By discarding the wrong points one could alter the tooth-to-tooth results, creating more inaccuracy. Given our fast computers of today, the solution is to use a full Fourier transform using the full data set, and one does not need to deal with the problems of data selection at all.

In any future standard, since it is not recommended that the long-form component be used for the determination of the quality of a gear, the use of a full Fourier transform may seem to be a moot point. However, it is very useful if a designer wishes to establish an "approximate" run-out specification or if the measurement of "approximate" run-out is a benefit for process control.

Exclusion of manual gaging with ANSI/AGMA 2015–2–A06. Due to the sophisticated calculating methods required for tooth-to-tooth results, the proper implementation of ANSI/AGMA 2015–2–A06 makes all non-computerized (manual) double-flank testers incapable of providing a measurement result. These types of testers—even today—remain the most frequently used testers in the industry. The standard should not ignore what is happening in industry and the use of these testers. Any future standards must include the provision to use indicator-type testers.

Test pressure. Currently, AGMA 915–3–A05 makes recommendations for test pressure based on the gear's module and face width, along with an adjustment if the gear is made of plastic instead of metal. A new standard should include the definition of a double-flank test pressure as, in some cases, test pressure may influence results. However, the information sheet should provide more detail on how to establish an appropriate test pressure based on the specific tester's natural response, and based on the geometry, material and structure of the gear being tested. A simple table—as currently exists in AGMA 915–2–A05—is not sensitive to all of these issues.

Numbering for accuracy grades can be confusing. One of the biggest sources of confusion in specifying the accuracy class is when only the numerical grade is used, without reference to whether it is a Q designation—as covered by AGMA 2000–A88—or a C designation—as covered by ANSI/AGMA 2015–2–A06. ANSI/AGMA 2015–2–A06 uses an accuracy class designation from C4 to C12, with C4 being the most accurate grade, where AGMA 2000–A88 uses classes Q3 to Q15, with Q15 being the most accurate grade. The reverse in the numbering brings the AGMA classes more in line with other standards. Those who specify gears using AGMA accuracy grades need to take greater care when specifying just a class number on their drawings; simply stating AGMA 8 would be quite misleading, as it would not be clear if it is a Q8 or a C8—two entirely different things. The overlap in numbering is unfortunate; any future system should consider an entirely different set of numbers to avoid further confusion.

Proposal for a Revised Radial Gear Accuracy Classification System

Based on the discussion above, the following recommendations are made for a revised AGMA radial gear accuracy classification system:

- Return to the AGMA 2000–A88 definition of the tooth-to-tooth deviation as being the difference in the tight-mesh center-distance within a single-tooth zone without adjustment for the long-form component. This simplifies the issue and allows for use of either electronic or manual gages.
- Include in the scope the use of the system for cylindrical worms, worm gears, sector gears, racks and bevel gears. For racks, use 200 teeth in the calculation of tooth-to-tooth tolerances. For gears with more than 200 teeth, default to 200 teeth for calculation purposes. For sector gears, in the calculation of total composite and tooth-to-tooth tolerance, adjust the full-circle value by the fraction of the number of teeth in the sector compared to the number of teeth full-circle.
- Create accuracy grades R20–R30 (R for radial) that are based on a constant value for total composite tolerance, with an associated tooth-to-tooth value that varies with the number of teeth. Adjust the tooth-to-tooth tolerance to be more practical in relation to the total composite tolerance. The class value 20–30 is used to avoid numerical duplication with any other system.
- Reduce the step size between classes to 20%.
- Recommend a way to calculate "approximate run-out" from the double-flank charts in case people can benefit from this information for part development purposes. This can be based on using a full Fourier transform to calculate the long-form component only in cases where an optional run-out requirement is explicitly stated on the drawing—not necessarily for everyday use, but as a way to assist in calculation of other gear geometry.
- Recommend a method of determining the ideal test pressure based on the natural response of the tester and the test piece characteristics, using the least pressure stable reading approach. The test pressure needs to be documented and held consistent for all measurements of the same part number.

The formulas for the calculation of the tolerances are recommended as follows:

For the radial total composite tolerance:

$$F_{idT}=\frac{1.2^{(R-26)}}{10},\text{mm} \tag{1}$$

where
R is the accuracy class

For the radial tooth-to-tooth tolerance:

$$F_{idT}=\frac{1.2^{(R-26-\text{Log}_{26}z)}}{10},\text{mm} \tag{2}$$

where
Z is the number of teeth in the full circle gear

For the "approximate" run-out tolerance:

$$F_{rT}=\frac{(0.741)\,1.2^{(R-26)}}{10},\text{mm} \tag{3}$$

Table 2—Proposed radial accuracy tolerances for class R26—table form

Class R	F_{idT}, mm	F_{rT}, mm	f_{idT}, mm											
			Z=1	Z=2	Z=3	Z=4	Z=10	Z=16	Z=24	Z=30	Z=40	Z=50	Z=100	Z=200
20	0.033	0.025	0.033	0.029	0.027	0.025	0.021	0.019	0.018	0.017	0.016	0.015	0.013	0.005
21	0.040	0.030	0.040	0.035	0.032	0.030	0.025	0.023	0.021	0.020	0.019	0.018	0.016	0.006
22	0.048	0.036	0.048	0.042	0.039	0.037	0.030	0.028	0.026	0.025	0.023	0.022	0.019	0.007
23	0.058	0.043	0.058	0.050	0.047	0.044	0.037	0.033	0.031	0.029	0.028	0.027	0.023	0.008
24	0.069	0.051	0.069	0.060	0.056	0.053	0.044	0.040	0.037	0.035	0.033	0.032	0.028	0.010
25	0.083	0.062	0.083	0.073	0.067	0.063	0.053	0.048	0.044	0.042	0.040	0.038	0.033	0.011
26	0.100	0.074	0.100	0.087	0.080	0.076	0.063	0.058	0.053	0.051	0.048	0.046	0.040	0.014
27	0.120	0.089	0.120	0.105	0.096	0.091	0.076	0.069	0.064	0.061	0.058	0.055	0.048	0.016
28	0.144	0.107	0.144	0.125	0.116	0.109	0.091	0.083	0.077	0.073	0.069	0.066	0.059	0.020
29	0.173	0.128	0.173	0.151	0.139	0.131	0.109	0.100	0.092	0.088	0.083	0.079	0.069	0.024
30	0.207	0.154	0.207	0.181	0.167	0.157	0.131	0.119	0.110	0.105	0.100	0.095	0.083	0.028

These formulas result in the tolerances shown in Figure 5 for class R26 and in Table 2 for gears with selected tooth counts.

For more-accurate classes, divide the R26 value by a 1.2 step factor for each class below R26 between R20 and R25. For less-accurate classes, multiply the R26 value by a 1.2 step factor for each class above R26 between R27 and R30. The tooth-to-tooth curve has steps in it due to a rounding to the nearest micron.

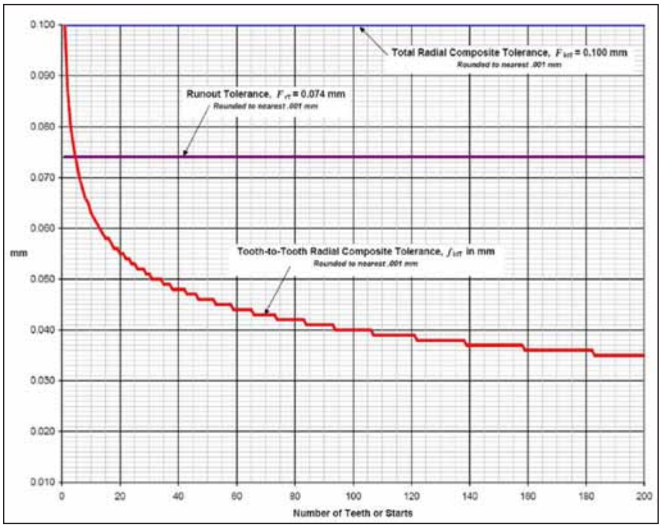

Figure 5—Proposed radial accuracy tolerances for class R26—graphical form.

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4. ANSI/AGMA 2015–1–A01. *Accuracy Classification System: Tangential Measurements for Cylindrical Gears*.
5. ANSI/AGMA 2015–2–A06. *Accuracy Classification System: Radial Measurements for Cylindrical Gears*.

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FZG Rig-Based Testing of Flank Load-Carrying Capacity Internal Gears

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Management Summary

Micropitting, pitting and wear are typical gear failure modes that can occur on the flanks of slowly operated and highly stressed internal gears. However, the calculation methods for the flank load-carrying capacity have mainly been established on the basis of experimental investigations of external gears.

The target of the research project was to verify the application of these calculation models to internal gears. Therefore two identical back-to-back test rigs for internal gears have been designed, constructed and successfully used for gear running tests. These gear test rigs are especially designed for low and medium circumferential speeds and allow the testing of the flank load-carrying capacity of spur and helical internal gears for different pairings of materials at realistic stresses. The three planet gears of the test rig are arranged uniformly across the circumference. Experimental and theoretical investigations regarding the load distribution across the face width, the contact pattern and the load-sharing between the three planet gears have been conducted.

Furthermore, substantial theoretical investigations of the characteristics of internal gears were performed. Therefore, internal and external spur gears were compared regarding their geometrical and kinematical differences, as well as their impact on the flank load. Based on the results of these theoretical investigations, an extensive test program of load stage tests and speed stage tests of internal gears of different material, different finishing of the flanks and different operating conditions has been performed. The main focus of this test program was on the fatigue failures—micropitting and wear—at low circumferential speeds.

This paper describes the design and functionality of the newly developed test rigs for internal gears and shows basic results of the theoretical studies. It furthermore presents basic examples of experimental test results.

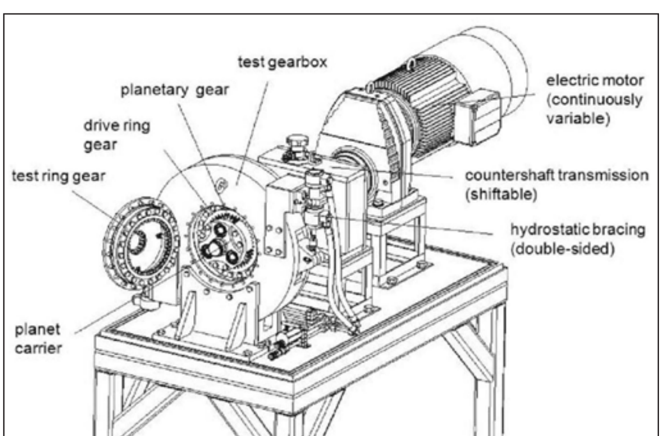

Figure 1—FZG back-to-back test rig for internal gears (schematic representation).

Introduction

Transmissions for wind turbines and low-speed industrial gear units often feature a slowly operated planetary gear stage. The flanks of these gears can be at risk for micropitting, pitting and wear due to the low circumferential speeds. The increase in power density—especially in transmissions for wind turbine gearboxes—enhances the risk of flank damages.

However, the calculation methods for the flank load-carrying capacity have mainly been established on the basis of experimental investigations of external gears. These calculation methods consider the possible advantages of internal gears regarding geometrical, kinematical and tribological conditions, but in only a limited form. As such, findings of systematic investigations on the flank load-carrying capacity of internal gears are practically nonexistent.

The scope of this research project was to enlarge the state of knowledge on the flank load-carrying capacity of internal gears and to verify the application of the calculation models to internal gears. Therefore substantial theoretical investigations of the characteristics of internal gears were performed and an extensive program of gear running tests of internal gears was conducted.

FZG Back-To-Back Test Rig for Internal Gears

In order to carry out the gear running tests on internal gears, two identical, back-to-back test rigs for internal gears were designed and constructed (Fig. 1). These gear test rigs are especially designed for low and medium circumferential speeds and allow the testing of the flank load-carrying capacity of spur and helical internal gears for different pairings of materials at realistic stresses. The essential component of this back-to-back test rig is the test gearbox, in which both test gear and drive gear are situated. The test gear is a planetary gear whose planet carrier is driven by a continuously variable electric motor via a shiftable, countershaft transmission. The three-stepped planet gears of the test rig are arranged uniformly across the circumference and are engaged with the test ring gear as well as the drive ring gear.

By means of the sectional representation of the test gearbox (Fig. 2), the design and functioning of the test rig is explained. The hydrostatic bracing device shown in Figure 1 twists the bracing ring (Fig. 2/2) relative to the two housing parts (Fig. 2/1a and 1b). Spring elements (Fig. 2/3) are used to apply a static torque to the drive ring gear (Fig. 2/4). The spring elements (Fig. 2/3) are mounted uniformly across the circumference and transmit the peripheral forces. They also provide a larger bracing displacement and thereby allow a defined adjustment of the bracing torque. The drive ring gear (Fig. 2/4) can adjust itself freely in radial direction whereby a uniform load-sharing between the three planets (Fig. 2/5) is promoted. The bracing torque is transmitted to the test ring gear (Fig. 2/6) by the planetary gears (Fig. 2/5). The stages of the gearing (Fig. 2/4 and 5; Fig. 2/5 and 6) exhibit the same gear ratio. The symmetrical design of the planetary gears (Fig. 2/5) allows testing of the rear flanks by simply reversing the planets without changing the orientation of bracing. The planets—as well as the planet shafts—are of rigid design so as to reduce irregular load distribution across the face width in the contact between planets (Fig. 2/5) and test ring gear (Fig. 2/6) caused by shaft deflection. The power loss of the test rig is induced via the two-piece planet carrier (Fig. 2/8).

In order to ensure a uniform distribution of the induced torque to the three planetary gears, the drive ring gear (Fig. 2/4) is mounted radially free in the test gearbox. The spring elements (Fig. 2/3) substantially support only peripheral forces. The axial force occurring in helical gears can be supported by thrust washers (Fig. 2/9) mounted in the housing.

Despite the rigidity of the planet gears (Fig. 2/5) and the planet shafts (Fig. 2/7), a slight deflection of these components has to be taken into account. In the same way, a slight torsion of the planet carrier (Fig. 2/8), due to the tilting torque, must also be considered. For this reason an optimization with FEM analysis has been performed.

The torque and rotational speed of the continuously variable electric motor are monitored. Variations in torque and/

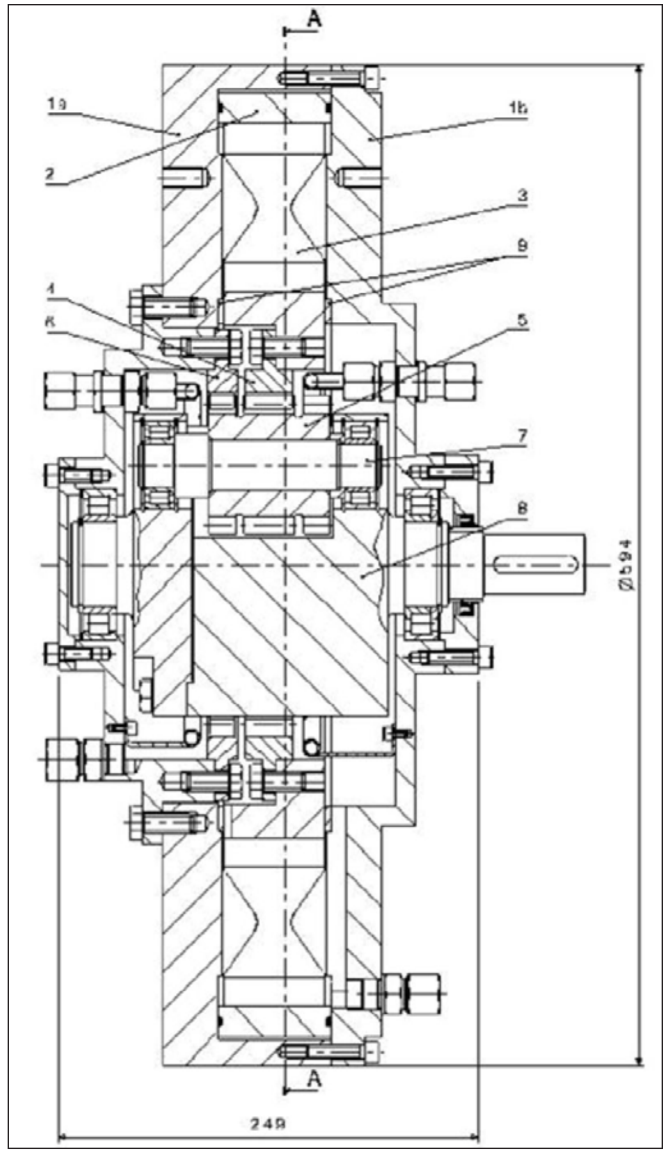

Figure 2—Sectional representation of the test gearbox.

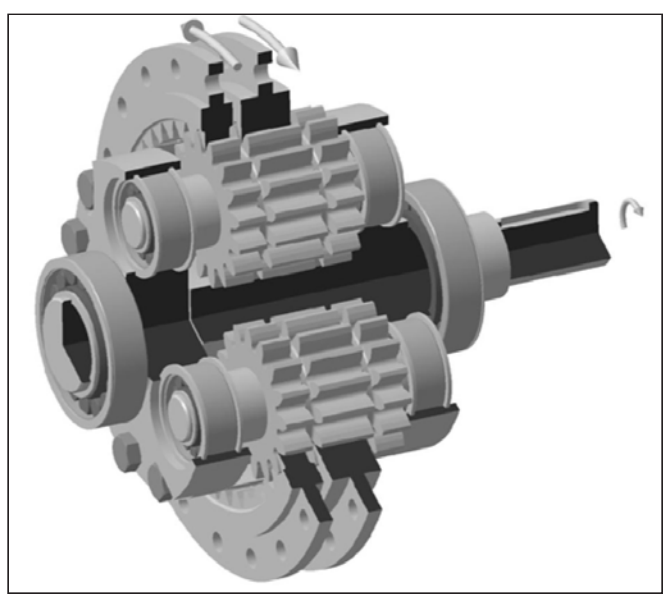

Figure 3—Schematic representation of the assembly of the test gear unit.

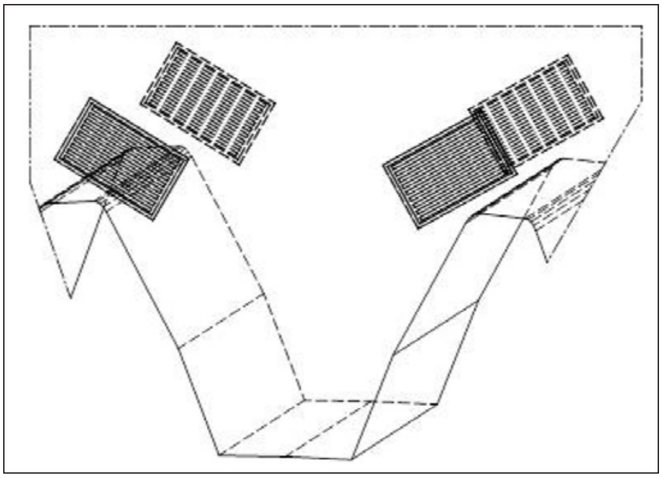

Figure 4—Application of the strain gauges (schematic representation).

or rotational speed can serve as switch-off criteria. The countershaft transmission (Fig. 1) enables the required low rotational speeds.

The oil feed, with a constant oil temperature, is implemented with oil spray lubrication by an external oil unit. Due to the low rotational speeds, the test and drive gears are wetted around full perimeter by means of annularly arranged spray nozzles.

The two housing parts (Fig. 2/1a and 1b) are screwed together with the grounding to enable application of the bracing torque. The bracing ring (Fig. 2/2) is twisted relative to the housing (Fig. 2/1a and 1b) by two hydraulic cylinders. A strain gage monitoring system was installed to maintain the required constant static bracing torque while operating. The strain gages are applied to multiple spring elements, arranged uniformly across the circumference.

Table 1 shows the specifications of the FZG back-to-back test rig for internal gears. The range of the circumferential speed from $v_t = 0.05$–10 m/s allows for the evaluation of micropitting and wear resistance at low speeds, as well as for performing test runs under test conditions comparable to standardized test methods, such as the pitting test ($v_t = 8.3$ m/s). The dimensioning of the maximum bracing torque was based on the calculative pitting resistance of the pairing of a case-hardened planet with case-hardened ring gear, according to ISO 6336 (Refs. 3 and 4).

The deflection of the planet shafts due to the radial forces resulting from the bracing torque T_{rg} (Fig. 3), as well as the deformation of planet wheel and planet shaft due to the tilting torque resulting from the double ring gear design, can affect the load distribution across the face width in an unfavorable way. This problem especially occurs in transmissions with stepped planets. An increasing misalignment of the planets results from an increasing bracing torque. A transverse load factor of $K_{H\beta} = 1.08$ was calculated for the test gear at a bracing torque of $T_{rg} = 1{,}500$ N-m, which corresponds to a pitting safety factor of $S_H \approx 1.0$ for the pairing of a case-hardened planet and a through-hardened ring gear. Considering the twisting of the planet carrier (FEM calculation), a transverse load factor of $K_{H\beta} = 1.09$ was calculated. However, for a maximum bracing torque of $T_{rg} = 6{,}000$ N-m, an even better transverse load factor of $K_{H\beta} = 1.06$ was calculated (considering the twisting of the planet carrier, also $K_{H\beta} = 1.06$), despite a larger misalignment of the planets. This is due to a more favorable ratio of the maximum-force-per-tooth-width to the average force-per-tooth-width (Eq. 1).

$$K_{\mathrm{H}\beta} = \frac{(F/b)_{\mathrm{max}}}{F_{\mathrm{m}}/b} \tag{1}$$

Strain gages have been applied close to the fillet on both sides of the test ring gear in order to analyze the load-sharing between the three planetary gears (Fig. 4). The strain gages have been aligned with the 60° tangents of the fillet because, according to ISO 6336 (Refs. 3 and 4), this is the location of the maximum bending stress for internal gears. The four strain gages are wired in a Wheatstone bridge for temperature compensation. The resulting signal gives information about tooth root bending stress of the test ring gear caused by the three planets. Ideally, the maximum amplitudes of the three signals (planets 1–3) are identical for a constant load. Figure 5 shows the results of the measurements on the load-sharing for different loads at a circumferential speed

Table 1—Specifications of the FZG back-to-back test rig for internal gears

Parameter		Unit	Value
rotational speed of the carrier	n_{carrier}	min^{-1}	5 - 1000
circumferential speed	v_{t}	m/s	0.05 - 10
rated motor torque (power loss)	T_{motor}	Nm	200
maximum bracing torque (test ring gear	$T_{\mathrm{rg,max}}$	Nm	6000

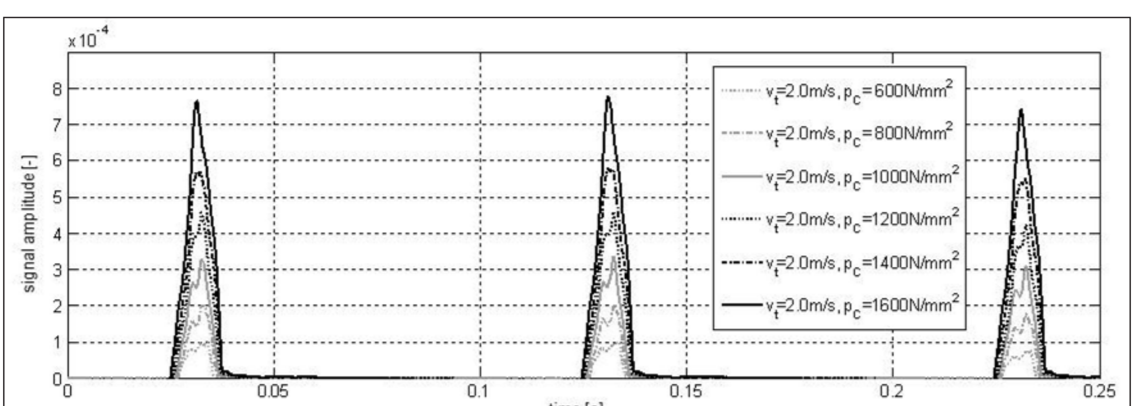

Figure 5—Measurement of the load sharing between the three planets at $v_t = 2.0$ m/s for different loads.

of v_t=2.0 m/s. The three curves of one load show the signal behavior of the strain gauge measurement during one revolution of the planet carrier, which is proportional to the maximum tooth root bending stress. In this case, the second tooth contact caused a slightly higher bending stress than the two other tooth contacts. However, the measurement indicates good load-sharing behavior between the three planets.

In order to evaluate the contact pattern of the test gears in the FZG back-to-back test rig for internal gears, a contact pattern check with contact pattern paint was performed at a medium load (p_C=400 N/mm^2). The contact pattern paint was removed with relative consistency across the face width of all teeth of the three planets and the ring gear. Therefore, the good load distribution across the face width—as well as the consistent load sharing between the three planets—could be confirmed.

Theoretical Investigations on the Characteristics of Internal Gears

The theoretical investigations were performed by comparing an external gear to an internal gear. The external gear for these theoretical studies is of the standard FZG test gear type C, which is also used in multiple standard gear running tests like the FZG–FVA pitting test, the FZG–FVA micropitting test and the FZG–DGMK wear test. The pinion of the external gear and the planet of the internal gear have the same tooth geometry. Table 2 summarizes the main gear data of both gears. The internal gear was also used for the gear running tests within the experimental investigations.

For the external gear, the beginning of contact (point *A*) is in the area of the dedendum flank of the pinion. In contrast, the beginning of contact (point *A*) for the internal gear was laid in the area of the dedendum flank of the ring gear. These drive directions are consistent with the most common applications in practice. For the external gear this is the configuration with a driving pinion, while internal gears—especially in the application range of wind turbines—are often operated with a reverse-drive direction (driving ring gear).

It should first be noted that external gears are a pairing of convex/convex flanks, whereas internal gears are a pairing of convex/concave flanks. Because of these geometrical conditions, the pinion of the external gear shows an increasing radius of curvature (ρ_1) along the transverse path of contact (AE), while the radius of curvature of the wheel (ρ_2) is decreasing (Fig. 6). This leads to an equivalent radius of curvature (ρ_{eq}) that can be regarded as being nearly constant in comparison to the internal gear. The absolute values of

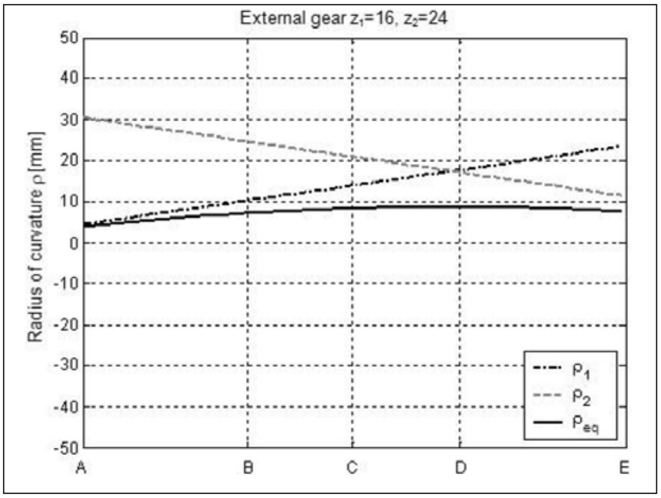

Figure 6—Radius of curvature of the external gear (beginning of contact: dedendum flank of the pinion).

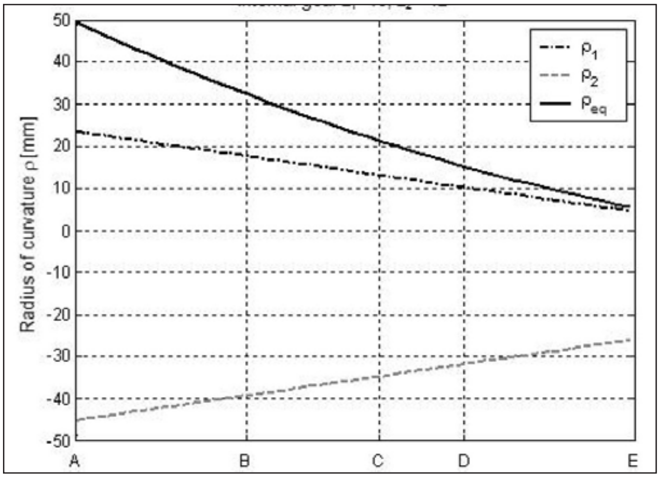

Figure 7—Radius of curvature of the internal gear (beginning of contact: dedendum flank of the ring gear).

Table 2—Gear data of external and internal gear for the theoretical studies

Parameter	Unit	External gear		Internal gear	
		Pinion	Wheel	Planet	Ring gear
normal module m_n	mm	4.5			
number of teeth z		16	24	16	-42
face width b	mm	14.0			
pressure angle α_n	°	20.0			
helix angle β	°	0			
addendum modification coefficient x	—	0.1817	0.1715	0.1817	-0.2962
tip circle diameter d_a	mm	82.5	118.4	82.5	-185.0
center distance a	mm	91.5		-59.0	

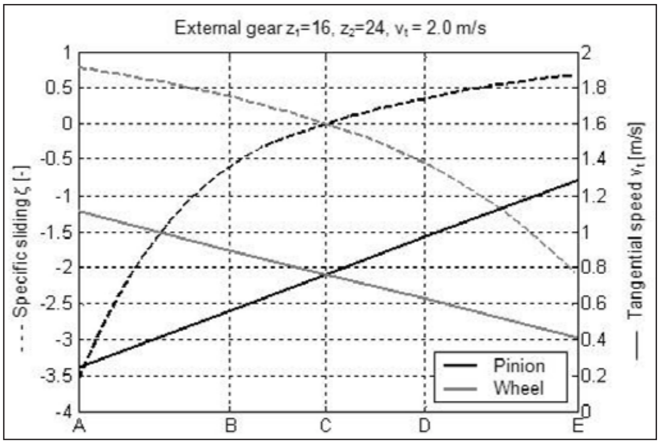

Figure 8—Specific sliding and tangential speed of the external gear.

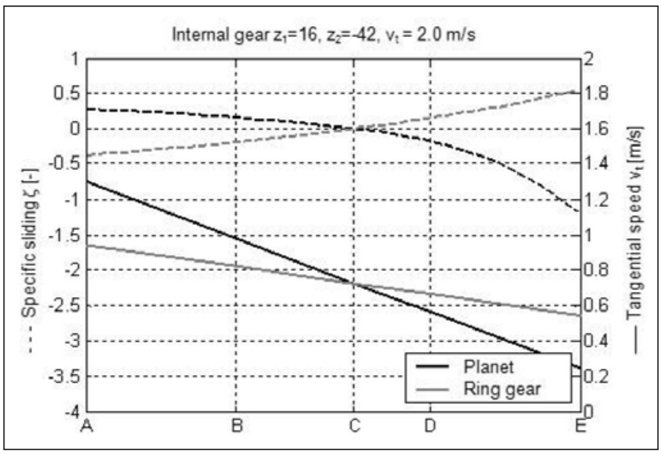

Figure 9—Specific sliding and tangential speed of the internal gear.

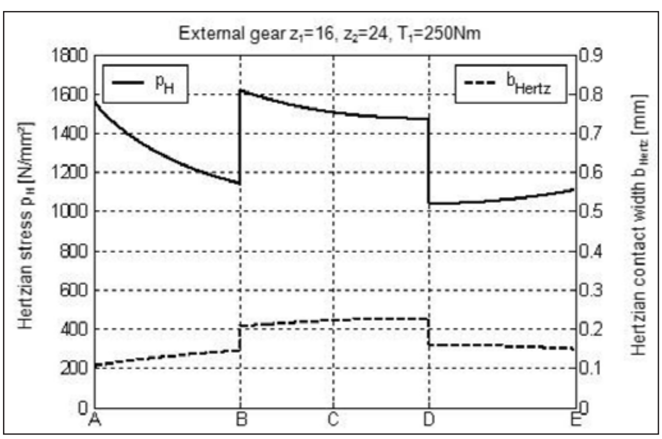

Figure 10—Hertzian pressure and Hertzian contact width of the external gear.

the radiuses of curvature of planet (ρ_1) and ring gear (ρ_2) of the internal gear, both starting at high values, are decreasing along the transverse path of contact (Fig. 7). This leads to a high equivalent radius of curvature (ρ_{eq}) in the beginning of contact, which continuously decreases over AE. The equivalent radius of curvature in point A of the external gear is comparable to the equivalent radius of curvature in the equivalent point E of the internal gear.

Figures 8 and 9 show that negative, specific sliding occurs in the area of the dedendum flank of pinion and wheel of the external gear, as well as in the planet and ring gear of the internal gear. Compared to the external gear, the absolute values of the specific sliding for the internal gear are significantly lower. However, for the internal gear, the negative, specific sliding of the planet is more distinct than the negative, specific sliding of the ring gear.

The mean sum of velocity for both gears can be calculated by adding the tangential speeds of the two engaged gearwheels. Figure 8 shows that the tangential speed of the driving gear of the external gear increases over the transverse path of contact (AE), while the tangential speed of the driven gear decreases equally; therefore the mean sum of velocity stays approximately constant. In contrast, the tangential speeds of both gearwheels of the internal gear (Fig. 9) start at their maximum values in the beginning of contact, and both decrease over AE—which leads to a decreasing sum of velocity. This results in an improvement of the hydrodynamic conditions in the beginning of contact which, in combination with the favorable equivalent radius of curvature, suggests that the area of the dedendum flank of the ring gear is exposed to a smaller risk of failure than the equivalent area of the wheel of the external gear. The tangential speeds shown in Figures 8 and 9 relate to a circumferential speed of $v_t = 2$ m/s.

Figures 10 and 11 show the Hertzian contact pressure p_H and the Hertzian contact width b_H over the respective transverse path of contact for an identical torque ($T_1 = 250$ N-m) on the pinion of the external gear and the planet of the internal gear. This load corresponds to the common fatigue strength of a case-hardened type C gear. The Hertzian pressure in the area of the dedendum flank of the ring gear of the internal gear is, compared to the wheel of the external gear, significantly lower. As a result, favorable conditions exist with regard to the flank load-carrying capacity this area. The area of the dedendum flank of the planet—at risk of micropitting due to negative specific sliding—is subjected to higher loads than the respective area of the ring gear due to the smaller equivalent radius of curvature.

Figures 12 and 13 show the calculated flash temperatures along the transverse path of contact—according to Blok (Ref. 10)—for the given torque and a surface roughness of $R_a = 0.3$ µm. The flash temperature serves as a measure of the contact temperature in the tooth contact and is presented for the reference lubricant FVA 3+4% Anglamol 99, and a range of circumferential speed from $v_t = 2.0 - 0.05$ m/s in both diagrams. It shows that the flash temperatures for the internal gear are significantly lower than for the external gear over the complete path of contact—with exception of the pitch point C. These lower contact temperatures, in essence, imply a tribological advantage over the external gear regarding risk of scuffing—especially at higher circumferential speeds.

The relative thickness of the lubricating film l, considering the flash temperature, for external and internal gear, is shown in Figures 14 and 15. The calculated values of the thickness of the lubricating film also relate to the reference lubricant FVA 3 + 4 % Anglamol 99 (ISO VG 100) at an injection temperature of 60°C. The more favorable hydrodynamic conditions at the beginning of contact of the internal gear are also apparent in the run of the thickness of the lubricating film. Yet, at the end of the transverse path of contact (point E), the internal gear shows low thicknesses of the lubricating film comparable to the conditions in the beginning of contact (point A) of the external gear.

If one examines both external and internal gear with regard to the threat of micropitting, the ring gear of the internal gear has the advantage that the Hertzian contact pressure in the area of negative, specific sliding is low due to the high values of the radius of curvature. This suggests, assuming equal material and heat treatment of the gears, that the area of the dedendum flank of the ring gear is exposed to significantly lower risk of micropitting than the equivalent area of the wheel of the external gear. Negative, specific sliding also occurs in the area of the dedendum flank of the planet. In contrast to the area of the dedendum flank of the ring gear, this area is exposed to the maximum value of the Hertzian pressure (Fig. 11) and the minimum value of the thickness of the lubricating film (Fig. 15). As a result, it is assumed that on internal gears micropitting predominantly occurs in the area of the dedendum flank of the planet.

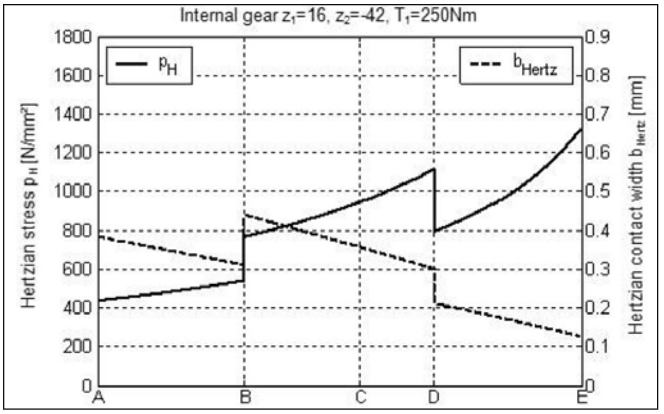

Figure 11—Hertzian pressure and Hertzian contact width of the internal gear.

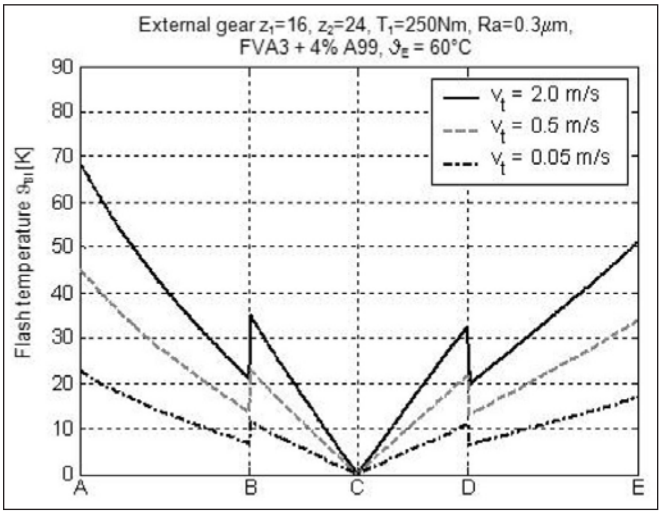

Figure 12—Flash temperature as a measurement of the contact temperature of the external gear.

Experimental Test Results

Within an extended research project (Refs. 8 and 9), an extensive test program has been conducted in the FZG back-to-back test rig for internal gears. The main focus of this test program was the gear failure modes micropitting and wear; it therefore covered numerous load stage and speed stage tests on internal gears of different material, different finishing of the flanks and different operating conditions. The reference operating conditions for the experimental tests are summarized in Table 3.

The load stage tests have all been performed at a constant circumferential speed of $v_t = 2.0\,\text{m/s}$ (Table 4). According to the standard micropitting test (Ref. 2), the duration of the load stages was set to 16 h/stage. The Hertzian pressure is gradually raised between two stages. Thereby, the load stage test allows comparison of information regarding the load impact on the pitting, micropitting and wear behavior of the analyzed variants. For the material pairing of the case-hardened planet and through-hardened ring gear, load stages 1–6 were performed. For the material pairing of the case-hardened planet and case-hardened ring gear, the load was consecutively increased from load stage 6 to load stage 11.

In the speed stage test, in contrast to the load stage test, the circumferential speed is varied while the load is held constant. The load for the speed stage tests was chosen in such a way that pitting was unlikely to occur (load stage 4 for the

Table 3—Reference operating conditions for the experimental test on internal gears

Parameter	Unit	Value
lubrication type		oil spray lubrication
injection volume	l/min	2
injection temperature	°C	60
reference lubricant		FVA 3 + 4% Anglamol 99
reference driving direction		driving ring gear

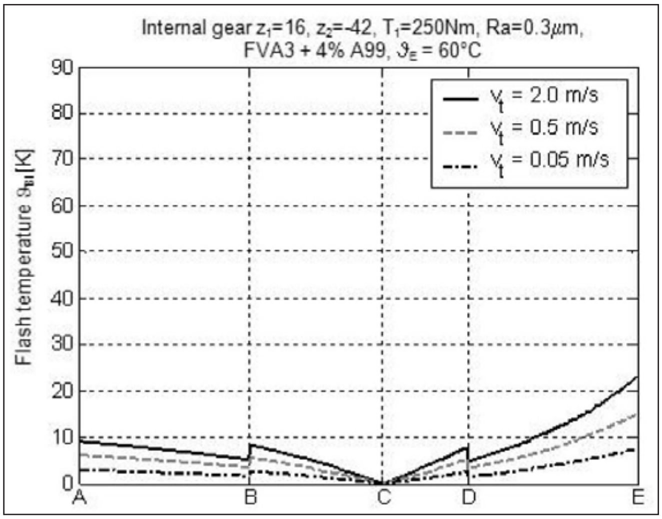

Figure 13—Flash temperature as a measurement of the contact temperature of the internal gear.

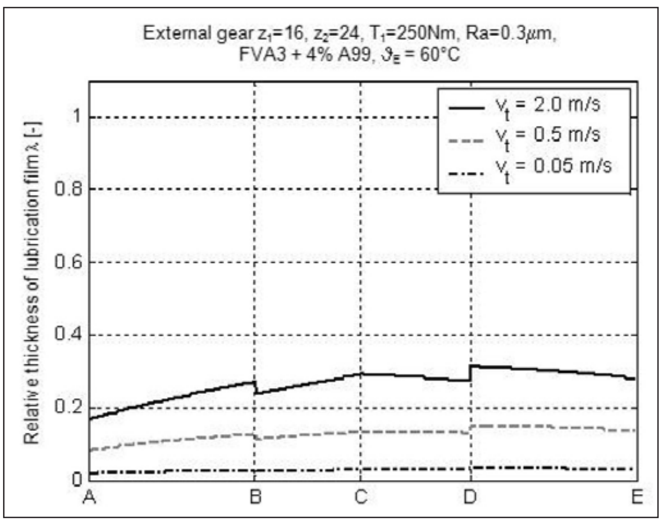

Figure 14—Relative thickness of the lubricating film for the external gear.

pairing of case hardened/through hardened; load stage 11 for the pairing of case-hardened/case-hardened). The duration of the speed stages was also set to 16 h/stage. Because of the number of planets and a gear ratio of $z_2/z_1=2.625$, the load-cycles-per-unit-of-time are reasonably comparable.

The extensive program of gear running tests showed that the calculation of the micropitting resistance according to FVA 259/I (Refs. 1 and 7), as well as the calculation of the wear resistance according to FVA10 (Refs. 5 and 6), can be directly applied to internal gears. Furthermore the test results basically support calculation of the pitting resistance according to ISO 6336 (Refs. 3 and 4). The following sections briefly summarize a few results of the experimental investigations.

Within the performed research project, the pitting resistance was not examined systematically. However, fundamental statements for the pairing of the case-hardened planet (18CrNiMo7-6) with a through-hardened ring gear (42CrMo4) can be made based on the results of the load stage tests. The first, smaller pitting areas for this material pairing were monitored after load stage 5 (p_C=700 N/mm²). The load stage test was stopped after load stage 6 (p_C=800 N/mm²) because progressive pitting was observable. Following the load stage test, an endurance test at the same load (p_C=800 N/mm²) was performed, since no failure criteria regarding the pitting area had yet been developed. Figure 16 shows one tooth flank of the through-hardened ring gear after the endurance test (3 million load cycles on the ring gear). The conversion of the Hertzian pressure p_C, according to ISO 6336 (Refs. 3 and 4), resulted in a contact stress of σ_H=750 N/mm², which is slightly higher than the common, permissible contact stress for this material (σ_{Hlim}=700 N/mm²; 42CrMo4V). Therefore it is possible to classify the result of the endurance test according to the current state of knowledge. In addition, Figure 16 shows that pitting on ring gears typically occurs in the area of singular contact and negative, specific sliding (points B and C). In contrast to external gears, the maximum stress in the area of the dedendum flank of the ring gear occurs directly at the pitch point C. It was observed that pitting particularly progressed in the area of the pitch point. As a result, ISO 6336 consequentially determines the contact stress σ_H at the pitch point.

Table 4—Load stages for the internal gear (load stage test)

Load stage	Hertzian pressure, N/mm²	Duration per stage, h	Torque, Nm	
			T_i (planet)	T_2 (ring gear)
1	300	16	25.2	198.3
2	400		44.8	352.8
3	500		70.0	551.1
4	600		100.8	793.8
5	700		137.2	1080.3
6	800		179.2	1411.2
7	950		255.4	2010.9
8	1100		338.8	2667.9
9	1250		437.5	3445.2
10	1400		548.8	4321.8
11	1550		672.6	5297.1

The micropitting and wear resistance of the paired case-hardened planet and case-hardened ring gear (18CrNiMo7-6) were examined in numerous speed stage tests. The speed stage tests for this material pairing were performed at a Hertzian contact pressure of $p_C = 1{,}550$ N/mm^2. With decreasing circumferential speed in the speed stage tests, a change of failure modes was monitored. The results showed an overlapping of failure modes micropitting and wear. Figure 17 shows pictures of one tooth flank of a case-hardened planet after the different stages of the speed stage test. The case-hardened planet already displayed initial micropitting in the area of the dedendum flank after the first stage, with a circumferential speed of $v_t = 2.0$ m/s. This micropitting area moderately increased until speed stage 5 ($v_t = 0.25$ m/s). During speed stage 6, enhanced wear-related changes of the tooth surface were monitored; the micropitting area was removed due to abrasive wear.

The corresponding ring gears did not show any damage or noteworthy changes of the surface structure in speed stages 1–5 ($v_t \geq 0.25$ m/s). However, for lower circumferential speeds ($v_t < 0.25$ m/s), significant wear-related damage of the tooth flanks was monitored (Fig. 18).

These results show that, in the respective range of circumferential speeds, only the planets are exposed to risk of micropitting. Given that micropitting typically occurs in the area of negative, specific sliding, these test results are in agreement with the results of theoretical studies that have shown that the area of the dedendum flank of the planet is subject to negative, specific sliding and high Hertzian pressure, while the dedendum flank of the ring gear—also subject to negative, specific sliding—is exposed to considerably lower values of Hertzian pressure. In contrast, the tooth flanks of both planet and ring gear can show wear-related damage for low circumferential speeds. The involute profiles at the end of the speed stage test (Figs. 19 and 20) show that the changes of the profile of the planet due to micropitting and abrasive wear are more distinct than the changes of the profile of the ring gear. For both planet and ring gear, the

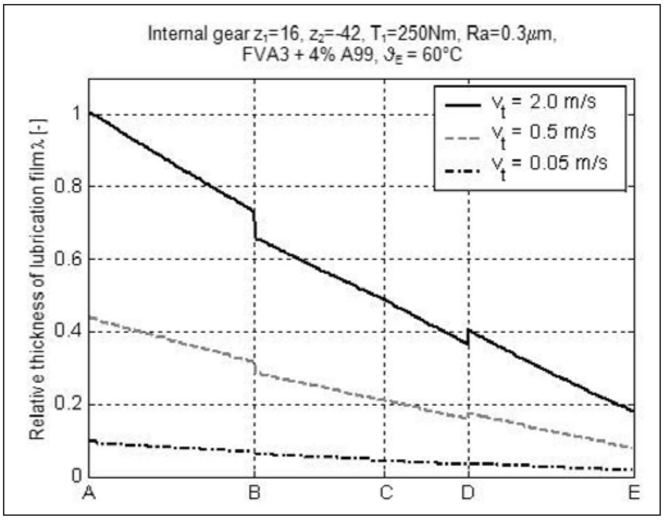

Figure 15—Relative thickness of the lubricating film for the internal gear.

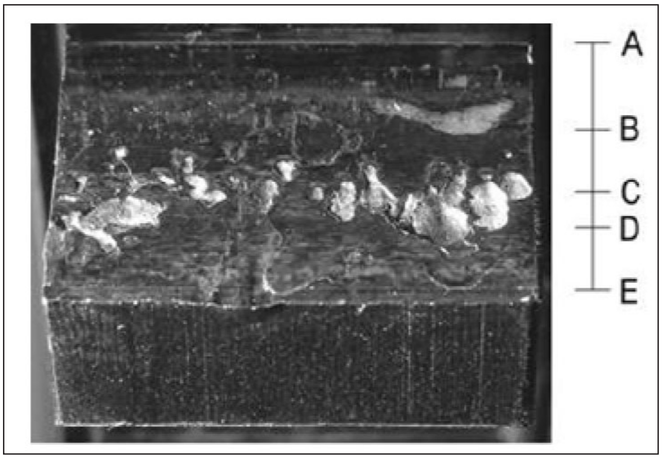

Figure 16—Pitting on the tooth flank of a through-hardened ring gear (p_C = 800 N/mm^2).

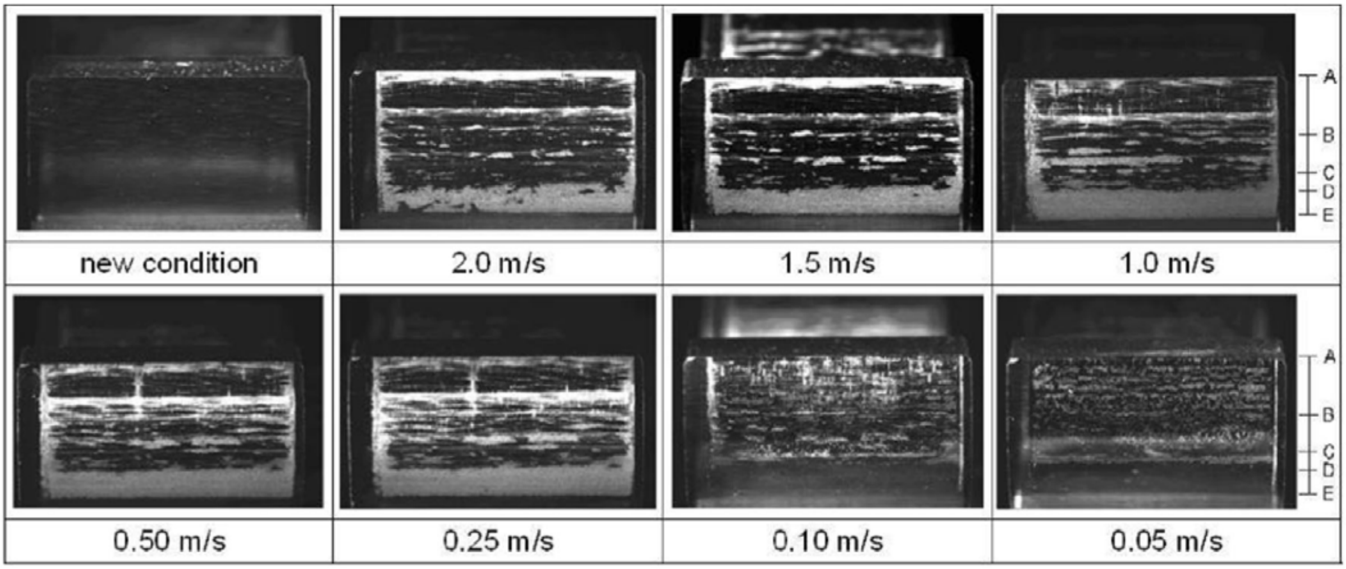

Figure 17—Exemplary pictures of one tooth flank of a case hardened planet in new condition and after every stage of the speed stage test; R_a = 0.3 µm.

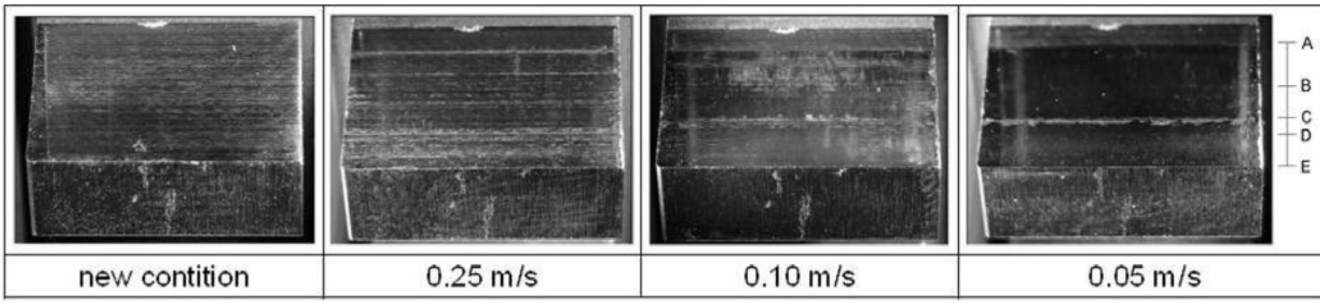

Figure 18—Exemplary pictures of one tooth flank of a case-hardened ring gear in new condition and after the relevant stages of the speed stage test (vt ≤ 0.25 m/s); Ra = 0.3 μm.

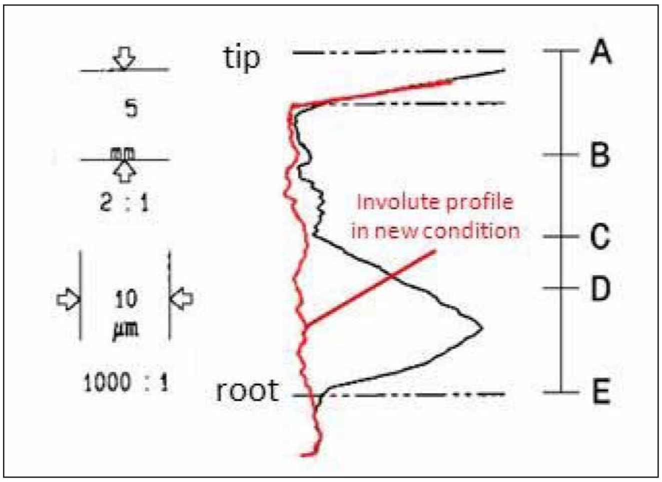

Figure 19—Involute profile of the tooth of a planet in new condition and after the end of the speed stage test.

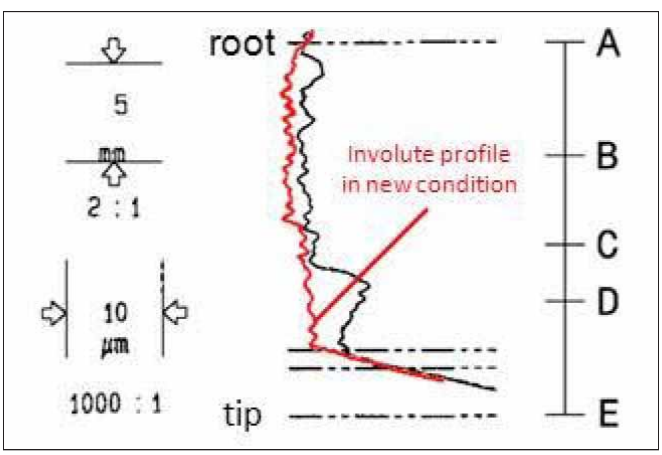

Figure 20—Involute profile of the tooth of a ring gear in new condition and after the end of the speed stage test.

location of maximum damage is in the area of maximum contact stress due to minimal, equivalent radius of curvature. Thus, the planet shows maximum changes of the involute profile in the area of the dedendum flank, while the ring gear shows maximum damage in the area of the addendum flank.

Summary

Pitting, micropitting and wear are typical gear failure modes that can occur on the flanks of slowly operated and highly stressed internal gears. In order to verify the calculation methods for the flank load-carrying capacity, substantial theoretical studies, as well as an extensive program of gear running tests, have been conducted. Therefore, two identical back-to-back test rigs for internal gears have been designed, constructed and successfully used for experimental tests.

Theoretical studies have shown that the area of the dedendum flank ring gear—which is at risk of micropitting and pitting due to negative, specific sliding—benefits from a high equivalent radius of curvature and low values of the specific sliding. This results in relatively low contact stress and relatively thick lubricating film. In contrast, the area of the dedendum flank of the planet is exposed to considerably higher values of the specific sliding and Hertzian pressure, while the relative thickness of the lubricating film is significantly lower. With that stipulation—assuming comparable material and heat treatment—pitting and micropitting will almost exclusively occur on the flanks of the planet of the internal gear.

The experimental tests have shown that calculation of the micropitting (Refs. 1 and 7) and wear resistance (Refs. 5 and 6) can be applied to internal gears. The procedure of the calculation of the pitting resistance (Refs. 3 and 4) was also supported by test results. Furthermore, the gear running tests have shown that an existing micropitting area, which had been generated at medium circumferential speeds (v_t = 0.25 m/s), can be overlapped and removed by wear-related damage of the flank at low circumferential speeds (v_t < 0.25 m/s).

Further results of the experimental investigations are given in References 8 and 9.

Acknowledgement. This research project was sponsored by the working group of Industrieller Research Associations Registered Association (AiF) with funds from the Federal Ministry for Economics (BMWi) and the Research Association for Propulsion Technology (FVA).

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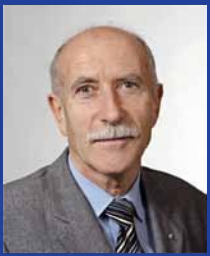

Dr.-Ing. Bernd-Robert Höhn *studied mechanical engineering at the Technical University Darmstadt (1965-1970) and served as an assistant lecturer (1970-1973) at the Institute for Machine Elements and Gears at the Technical University Darmstadt prior to becoming an assistant professor at the university (1973-1979); in 1978, he received his PhD (Dr. Ing.) in mechanical engineering. In early April, 1979 Höhn worked as a technical designer in the department for gear development of the Audi, and by 1982 was head of the department for gear research and design for the automaker. In 1986 Audi named Höhn department head for both gear research and testing of automotive transmissions, until his departure in 1989 to become head of both the Institute of Machine Elements at the Technical University and of the Gear Research Centre (FZG). Since 2011, he has served as director emeritus of the Institute. Höhn has also served as vice president for VDI for research and development, as well as leader of working groups 6 and 15 for ISO TC 60—calculation of gears.*

Prof. Dr.-Ing. Karsten Stahl *became head of the Institute for Machine Elements and the Gear Research Centre (FZG) at the Technical University of Munich in 2011. As the successor to Dr. Höhn, Stahl directs more than 60 employees at the institute, which focuses on examination and testing of machine elements, such as gears, bearings, synchronizations and couplings.*

Dr.-Ing. Thomas Tobie *is a department leader at the Institute for Machine Elements – Gear Research Centre (FZG). His research specialties are in the fields of heat treating, gear material and gear load capacity regarding tooth root bending fatigue, pitting and micropitting.*

Juergen Schudy *is a research associate at the Institute for Machine Elements – Gear Research Centre (FZG).*

Dipl.-Ing. Bernd Zornek *is a research associate at the Institute for Machine Elements – Gear Research Centre (FZG).*

Eitel

NAMED WORLDWIDE SALES DIRECTOR AT HÖFLER

Left to right: Bernd Lotter, Ralf-Georg Eitel and Thorsten Haug participate in the Sandvik/Höfler "Get Into Gear" event at the Sandvik Coromant Productivity Center in Schaumburg, Illinois.

Ralf-Georg Eitel, president and CEO of Höfler America Corp., has been named sales director, worldwide for Höfler Maschinenbau GmbH. Eitel has played a vital role in developing business in the North American market along with team members Bernd Lotter, vice president, and Thorsten Haug, assistant service manager. To date, there are 300+ Höfler machines running in the United States). Eitel's new position as worldwide sales director took effect July 1st. He is also a member of the board of directors in Germany.

Höfler America Corp. is a sales/service/spare parts provider located in Pittstown, New Jersey. In order to further improve service and sales support in the North American market, Höfler Maschinenbau and Sandvik Coromant joined forces at the Sandvik Coromant Productivity Center in Schaumburg, Illinois just outside of Chicago. A Höfler HF 1000 gear hobbing machine equipped with the latest indexable insert tooling solutions is onsite for training, demonstrations and joint activities with North American customers.

The two companies recently presented a one-day technology fair, "Get Into Gear," that offered customers and clients hob milling demonstrations, innovations in gear machining and cost savings analysis. Eitel, along with presenters like Kenneth Accavallo, industry and applications specialist at Sandvik and Jack Lynch, product manager at Sandvik, led a variety of discussions throughout the day on the advantages of Sandvik and Höfler products and services. Attendees were able to view machine tool demonstrations of Sandvik disc cutters and indexable hobs on Höfler and Mazak machines. In addition, AGMA President Joe Franklin presented an overview of gear industry trends and the advantages of being an AGMA member. With several top gear manufacturing personnel in attendance, the event was a remarkable success and gave Sandvik and Höfler an opportunity to show off their latest gizmos and gadgets prior to IMTS in September.

"This is an exciting opportunity to see new markets and bring my experience from North America to the global gear industry," Eitel says. "Since the beginning of our activities in North America, the importance of this market grows every year for Höfler. The sales in 2011 increased compared to 2009 and 2010. I expect equal sales results for the North American market in 2012. The first four months of the year have reflected this."

EMCO

ANNOUNCES NEW PRESIDENT

Philipp Hauser

EMCO Maier Corporation, Novi, Michigan, part of the EMCO Group, Hallein, Austria, has named Philipp Hauser its new president, according to Dr. Stefan Hansch, EMCO Group CEO. Hauser will report to Dr. Hansch. Previously, Hauser was sales manager for MS Precision Components, Fowlerville, Michigan, and for 15 years was with Wenzler, a German machine tool builder, eventually as managing director at Wenzler USA. Hauser said one of his chief goals is to double U.S. sales of EMCO turning/milling machines and turnkey machining solutions in five years across a range of industries, including mechanical components, aerospace, medical, hydraulic, and more. "We have a good core group of competent employees and dealers, and we will be growing the group to properly support our customers with high quality machine tools and a high level of applications engineering and service," Hauser said.

Arvin

HONORED AT ILLINOIS INSTITUTE OF TECHNOLOGY

Joseph L. Arvin

On Friday, April 27, IIT's Industrial Technology and Management (INTM) Program honored Joseph L. Arvin as its 2012 "Outstanding Leader in Industry." Arvin is the president of Arrow Gear Company, a $42 million precision gear manufacturer in Downers Grove, Illinois, specializing in spiral bevel, spur and helical gears and custom gearboxes for the aerospace industry and commercial applications. He is a senior member of the Society of Manufacturing Engineers (SME) and past board member of the American Gear Manufacturers Association (AGMA). He holds multiple patents for innovative machining processes and has traveled throughout the world to assess the industrial capacities of foreign countries. Arvin is founding member and current president of Citizens for American Manufacturing (CAM) and an active advocate for the reform and rebuilding of the U.S. industrial base through manufacturing. Arvin was chosen as Outstanding Leader in Industry for his significant professional achievements and outstanding dedication and service to the field of manufacturing.

Mitutoyo

CELEBRATES NEW SOLUTION CENTER

Mitutoyo America Corporation recently announced the opening of the first M3 Solution Center in the south central United States region, located in Birmingham, Alabama. This new, 4,800 sq. ft. showroom is conveniently located so customers can schedule appointments for product demonstrations and assistance with metrology solutions and application challenges.

"Our goal is to provide timely metrology solutions to our customers, in a region that is home to a booming automotive industry. The benefit of opening this new M3 Solution Center was the accessibility in offering an experienced metrology specialist to our customers that could provide up-to-date and knowledgeable metrology information for any situation they may encounter," says Mike Dukehart, regional sales manager, South Central region. The grand opening celebration was held on Tuesday, May 22 and Wednesday, May 23. The festivities included a tour of the facility, refreshments, raffles and live demonstrations.

NEWS

Seco/Warwick

ANNOUNCES CHANGES TO AFTERMARKET TEAM


Dan Alabran

Dan Alabran, formerly the team leader for Aftermarket Services and Short Cycle Manufacturing Projects, will take over as the vacuum furnace team leader. John Hughes will replace Alabran as the aftermarket team leader. Alabran joined Seco/Warwick in 1989 as a project engineer for aluminum process furnace applications. During his tenure with the aluminum group, Alabran demonstrated a proficiency in upgrading, refurbishing and improving efficiencies in existing equipment in addition to planning and engineering new equipment facilities and was promoted to manager of Aftermarket Services and Short Cycle Manufacturing projects in addition to his project engineering duties. He holds a degree in electrical engineering from Kent State University. Hughes has an associate degree in engineering from the Shenango Valley campus of Penn State University. He started at Seco/Warwick on the aftermarket parts team in 2006. Hughes had been in the tool and die trade designing automated assembly equipment for most of his career prior to joining Seco/Warwick.


John Hughes

Star SU

APPOINTS NEW DIRECTOR OF MARKETING

Mark Parillo has been appointed director of marketing at Star SU, LLC. In his new position, Parillo will be responsible for the development, implementation, and management of marketing tools and strategies to further strengthen Star SU's market position and brand recognition. "Mark brings a wealth of experience from the machine tool and tooling industries, which will allow us to expand our presence in the marketplace," said David Goodfellow, president of Star SU, LLC. Parillo, 42, brings nearly two decades of automotive and industrial marketing experience to his new position. Most recently he served as marketing manager for MAG

IAS, LLC, automotive division in Sterling Heights, Michigan. He holds a master's degree in management (MSM) at Walsh College of Accountancy and Business Administration, Troy, Michigan, and bachelor of business administration degree (BSBA) from Aquinas College, Grand Rapids, Michigan. "

Mark Parillo

EMAG

RECEIVES AGREEMENT WITH OAKLAND COMMUNITY COLLEGE

EMAG recently announced its receipt of a five-year, $200,000 agreement with Oakland Community College (OCC), through the Michigan New Jobs Training (MNJT) program, for the training of 21 new employees in manufacturing technology, CNC machine tools, mechanical maintenance, electrical and robotics disciplines. Peter Loetzner, EMAG CEO, accepted the award from Dr. Timothy Meyer, chancellor of Oakland Community College, and J. Brooks Patterson, Oakland County executive, during the Oakland County Economic Outlook Luncheon. Introducing Loetzner, Meyer detailed the collaboration between OCC and EMAG, noting how the study of mechatronics would raise the skill level of both engineers and the plant workforce to higher levels of technical competence and multi-functional abilities. During his remarks, Loetzner recounted his own experience as a student in this now rapidly growing field of mechatronics, where mechanical engineering melds with electrical and electronic engineering to help students better understand the inter-relationship of components on a machine. Mechatronic engineers and field technicians are now highly valued individuals in many industries, Loetzner noted. He added that EMAG will have at least 20 openings in engineering throughout the next few years at his company, the result of increased sales and the EMAG

Left to right: J. Brooks Patterson, Oakland County executive and Peter Loetzner, EMAG CEO, recently announced a partnership between EMAG and Oakland Community College.

NEWS

commitment to serving the North American market from its expanding headquarters in the Detroit area. He further commented how such programs have the dual advantages of growing the manufacturing base in the area, as well as raising the skill level of the employees, which in turn attracts more businesses to Oakland County.

ANCA Group

OPENS ANCA MOTION HEADQUARTERS

The ANCA Group has formally opened their new ANCA Motion headquarters in Melbourne's Eastern suburbs. The plaque was unveiled by Greg Combet, the minister for Industry and Innovation and Pat Boland, the chairman of ANCA. ANCA Motion, a division of the ANCA Group, manufactures innovative motion control systems. Their computer controlled systems form part of ANCA's market leading tool and cutter grinders manufactured at a 14,000 m^2 site located in the same industrial park.

The ANCA Group CEO, Grant Anderson, told the attendees how ANCA was launched in 1974 by Boland and Pat McCluskey, who today remain the owners of the ANCA Group. "The ANCA Group is a truly global company employing approximately 800 people worldwide," said Anderson. As an example of ANCA's commitment to innovation, Anderson advised that "the ANCA Group spends around nine percent of sales on research and development each year."

Boland congratulated David Fisher, ANCA Motion general manager, and his team, on the opening of their new facilities in Australia. 70 full-time staff members work in the new building, designing and manufacturing CNC solutions, servo drives, motors and associated equipment. Combet listed the ANCA Group's key export markets as including Germany, Japan, China and the United States, stating that "these are not easy markets to market machine tools into." ANCA exports virtually its entire production with exports totalling $800 million to date. ANCA continues to make considerable investments in state-of-the-art factory equipment and R&D. The new ANCA Motion building represents the commitment of the ANCA Group to growth and continuous improvement. For more information, visit *www.anca.com*.

Left to right: Greg Combet, the minister for industry and innovation and Pat Boland, the chairman of ANCA, celebrate the opening of ANCA Motion headquarters in Melbourne, Australia.

CALENDAR

August 6-9—CAR Management Briefing Seminar. Grand Traverse Resort and Spa, Traverse City, Michigan. The Center for Automotive Research (CAR) presents its traditional summer gathering for the automotive industry. This year's scheduled sessions will focus on global manufacturing strategies, tooling technology, money matters, surviving the skills shortage and many more. The briefing seminar is a networking opportunity for manufacturers and suppliers, purchasing and marketing executives, energy representatives, financial analysts, government and education representatives, information managers, labor leadership, media members and plant managers and superintendents to share thoughts on the changing automotive industry. Speakers include Joseph Bakaj, vice president, powertrain engineering, Ford Motor Company, Jay Baron, president and CEO, CAR and Erik Berkman, president, Honda R&D Americas Inc. For more information, visit *www.cargroup.org*.

September 10-15—IMTS 2012. McCormick Place, Chicago. The 29th edition of the manufacturing technology show boasts more than 1,100 exhibiting companies that will occupy 1.1 million net square feet of exhibit space. The show attracts 82,000 buyers and sellers from more than 116 countries. Leading manufacturers will display their equipment in pavilions including Metal Cutting, Tooling and Workholding Systems, Metal Forming and Fabricating/Laser Processes, Gear Generation, Industrial Automation and many more. The IMTS 2012 Conference brings the industry together, under one roof and at one time, to discuss revered technologies, business development and optimization, plus workforce efficiency and productivity. Special emphasis will be placed on maintaining focus on short- and long-term goals during a tough economic environment. For more information, visit *www.imts.com*.

September 11-13—International Conference on Manufacturing Research 2012. Aston University. For over two decades it has been the main manufacturing research conference organized in the U.K., successfully bringing researchers, academics and industrialists together to share their knowledge and experiences. Initiated as a National Conference by the Consortium of U.K. University Manufacturing Engineering (COMEH) it became an International Conference in 2003. COMEH is an independent body established in 1978. Its main aim is to promote manufacturing engineering education, training and research. Keynote speakers for the event include Hamid Mughal, executive vice president, manufacturing, Rolls Royce Plc.; professor Sir Mike Gregory, Institute for Manufacturing, Cambridge University; John Ladbrook, European simulation specialist, Ford Motor Company and Professor Jay Lee, director of the Center for Intelligent Maintenance Systems, University of Cincinnati. For more information, visit *www1.aston.ac.uk/icmr2012*.

September 18-19—Human Error Prevention Seminar. Fogelman Executive Conference Center, Memphis, Tennessee. The principles and practices of human error prevention are universally applicable regardless of the type of industrial, commercial or governmental enterprise, and regardless of the type of function performed within the enterprise. This seminar is truly unique and up to date with the latest developments in human error prevention. Ben Marguglio's new taxonomy of human error causal factors and his human error-related models demonstrate his leadership in this subject. Examples and case studies amply reinforce the human error prevention principles and practices. This seminar covers: classifications of human error; quality and safety culture and the quality- and safety-conscious work environment; leadership responsibilities; the total quality and safety function and much more. For more information, contact Ben Marguglio at (845) 265-0123 or e-mail *ben@hightechnologyseminars.com*.

September 20-21—Root Cause Analysis Seminar. Fogelman Executive Conference Center, Memphis, Tennessee. This seminar covers all of the elements of a problem/condition reporting, root cause analysis and corrective action system with emphasis on the following root cause analysis techniques: Failure Mode and Effects Analysis for hardware problems and Hazard-Barrier-Effects Analysis for management and technical process problems. This seminar will also cover a modified Hazard-Barrier-Effects Analysis technique that allows the root cause analysis resource expenditure to be proportional to the significance of the problem, while still enabling the analyst to identify human performance root causes. Persons who are responsible for identifying and reporting off-normal conditions, evaluating the conditions and their effects, identifying causal factors, recommending various types of corrective actions, tracking the implementation of corrective actions, and managing the overall system should consider attending this seminar. For more information, contact Ben Marguglio at (845) 265-0123 or e-mail ben@hightechnologyseminars.com.

October 15-19—AME Chicago 2012. Sheraton Chicago Hotel and Towers. The Association for Manufacturing Excellence (AME) has a long track record for finding and convincing some of the best manufacturing practitioners from around the world to share their lean practice experiences. More than 60 leading presenters will be on hand to discuss customer focus, process sustainment, continuous improvement, material flow and other lean practices and strategies. Manufacturing tours highlighting some of the best lean and six sigma operations in and around the Chicago area include Caterpillar, Bimba Manufacturing, Whiting Corporation, S&C Electric Company and Winzeler Gear. Workshop topics include maintenance management, lean behaviors, training within industry, lean business simulation and lean tools for the office. Six keynote speakers will be featured at the conference including Mike Abrashoff and Jason Jennings. For registration information, visit *www.ameconference.org*.

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____
City:____________________
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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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Mitutoyo	21	(888) 648-8869		www.mitutoyo.com
Niagara Gear	4	(800) 447-2392	info@niagaragear.com	www.niagaragear.com
Ontario Drive & Gear	30	(888) 662-2840		www.odg.com
Overton Chicago Gear	7	(630) 543-9570	sales@oc-gear.com	www.oc-gear.com
Presrite Corp.	24	(216) 441-5990		www.presrite.com
Process Equipment	45	(800) 998-4191		www.gearinspection.com
Proto Manufacturing	37	(519) 737-6330		www.protoxrd.com
Quality Transmission Components	72	(516) 437-6700	QTCsupport@qtcgears.com	www.qtcgears.com
Riverside Spline & Gear	23	(810) 765-8302	valerief@splineandgear.com	www.splineandgear.com
Sandvik Coromant	34	(800) 726-3845		www.coromant.sandvik.com/us
Schafer Gear Works	47	(574) 234-4116		www.schafergear.com
Schnyder S.A.	71	(630) 595-7333	hanikcorp@aol.com	www.hanikcorp.com
Sicmat	31	(847) 649-1450	sales@star-su.com	www.star-su.com
Sigma Pool	5	(734) 429-7225	info.lgt@liebherr.com	www.sigma-pool.com
South Bend Gear LLC	25	(574) 234-4116		www.southbendgear.com
Star SU LLC	IFC–1, 31, 79	(847) 649-1450	sales@star-su.com	www.star-su.com
Stock Drive Products/Sterling Instrument	71	(516) 328-3300	SDP-SIsupport@sdp-si.com	www.sdp-si.com/e-store
Stresstech Oy	42	(412) 784-8400	info@ASTresstech.com	www.astresstech.com
Tokyo Technical Instruments USA	6	(201) 634-1700		www.tti-geartec.jp
Toolink Engineering Inc.	59	(303) 776-6212		www.toolink-eng.com
Wenzel America Ltd.	22	(248) 295-4300		www.wenzelamerica.com

GEAR MANUFACTURING

All Hail Leonardo

If you were at all like many college freshmen, you may recall your required (at many schools) Art History 101 lecture class—over-crowded, often slotted for an eight or nine a.m. start, and suitably dark for art slides viewing. The Old Masters never had a chance. Who among us can deny on occasion catching up on some much-needed shut-eye while "the old perfesser" droned on about various artistic styles, eras, etc?

Which is perhaps why a good many people—"educated" or otherwise—are not aware that Leonardo da Vinci—arguably the poster child for the Renaissance Man—was of, how to put it—undetermined parentage. And yet, has there ever been clearer evidence that the old saw—It's not where you start, it's where you finish that matters—holds meaning?

What we do know is that he was born in 1452 to one Ser Piero—a wealthy Italian—and "Catrina"—a woman of means by no means—i.e., a peasant, last name unknown.

To address the no-last-name issue, the boy became known as "da Vinci," or, from Vinci. Some of the back story remains murky, but it is documented that the boy lived with his paternal grandparents and uncle in Vinci. It has also been reported that, while Leonardo showed no early signs of genius—he did receive an education—in Latin, geometry and math—but who tutored him, and where, remains a mystery.

This Addendum is not a biography. We won't regale you with da Vinci factoids regarding his "other career." Yes, he created The Last Supper, and yes, he created the Mona Lisa. (Come to think of it, those achievements alone make for a pretty good legacy.) Rather, it is an expression of wonder and marvel that a child born of such sketchy circumstances could and would go on to become one of the charter members of the Smartest Men in the Room club.

Best of all, he wasn't just a Renaissance man; he was—as most reading this already know— one of our own—a *gear man*.

Consider these da Vinci-concepted, "first-iteration" inventions—all still used today in some capacity:

The retractable landing gear. The mechanism could be allowed to drop under its own weight, or retracted by the simple pull of a string.

The ball bearing. First attributed to the Romans, da Vinci's design "perfected" its functionality in reducing components' friction for another of his marvels—the helicopter. The first bearing patent was not granted until 1791.

The automobile. Spring-driven, the "vehicle" was wound up in order to propel it. (Perhaps a precursor to today's early "electric" cars?) Pegs (pinions) installed in matching holes directed the wheels of the car to turn intermittently—controlled "under the hood" by what one da Vinci site describes as "programmable, complex gearing and cog assemblies." Based on the spring diameters and instructions of da Vinci's original design, some have written that the machine "could move for up to forty meters before needing to be recoiled."

Continuously variable transmission (CVT). Quoting from *howstuffworks.com,* "Some say you can't teach an old dog new tricks but the continuously variable transmission, which Leonardo da Vinci conceptualized more than 500 years ago and is now replacing planetary automatic transmissions in some automobiles, is one old dog that has definitely learned a few new tricks. Today, several car manufacturers—including General Motors, Audi, Honda and Nissan—are designing their drivetrains around CVTs."

The robot. Credited as "the first robot in history." Ever notice all those da Vinci references—"vitruvian man," for example—used by surgical robotics companies?

Rack-and-pinion gear system. Although da Vinci didn't invent this type of gearing, he certainly made ingenious use of it in many of his devices. As common today as thin-crust pizza.

So next time you hop in your car, land safely at LAX or have your gallbladder removed by a robot, reflect for a moment and give the man his props. (Sources: *museoscienza.org*; *mostredileonardo.com;* and *howstuffworks.com.*)

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