September/October 2007

GEAR TECHNOLOGY

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The Journal of Gear Manufacturing

Gear Expo 2007
- Booth Listings & Map
- Sneak Preview
- Solutions Center

Technical Articles
- Advances in Bevel Gear Blades
- Plastic Tooth Bending
- High-Speed Gears for Extreme Applications

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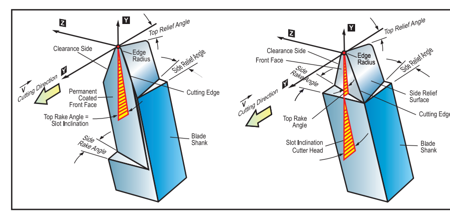
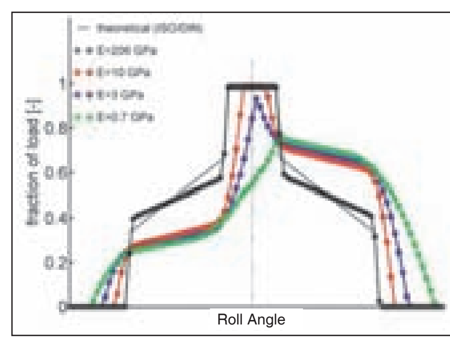
C O N T E N T S

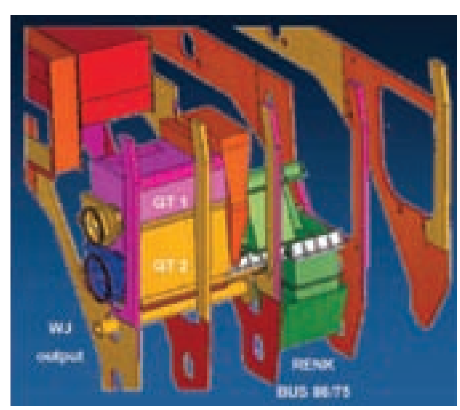
FEATURES

Gear Expo 2007. **See pages 32–49 for full coverage.**

TECHNICAL ARTICLES







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BPA WORLDWIDE

Do I Have to Go to Gear Expo?

A lot of gear manufacturers are having sweet dreams these days…

Ahh, business is great! The people and machines on my plant floor are all working hard. Customers keep placing orders. We've just installed new machines to expand capacity, and we've got more on order. Cash flow has never been better…

But dreams have a funny way of turning, don't they?

I can't stop running. Even though I'm exhausted, I can't stop, not even for a second. Everyone needs a vacation, and I feel like I'm last in line! Sure, business is great, but every time I look over my shoulder, there are new orders, new opportunities and more work to be done. Sure, cash flow is great, but I've been borrowing left and right to pay for new equipment. I've got employees to hire—and train! Where is the finish line?

Many of you have told me lately how busy you are, and that Gear Expo is the last thing you want to think about right now. Maybe you aren't quite to the point of waking up in cold sweats about it, but a number of you feel like you can't spare an hour to walk the show floor—let alone two days. To quite a few that I've talked to, Gear Expo seems like just one more task to add to an already long list.

But it'll be alright. Just take a deep breath and try to relax.

I strongly believe that it is important for you to attend the show, but maybe it will be all right if you don't go. There, I said it.

If you feel like you're too busy for Gear Expo, fine. You've probably earned the right to stay home. If you've been there, done that and feel like skipping this one, it probably won't be the end of the world. But just because you're not going doesn't mean somebody else from your company shouldn't.

SEND SOMEONE ELSE.

Instead of looking at the Show solely as a potential machine tool buying experience, remember that it's also a unique educational experience.

As I've said in the past—and as AGMA has started to echo in their advertising—I believe that Gear Expo offers the largest gathering of gear manufacturing machinery and gear manufacturing experience under one roof anywhere in the world. On the floor you are able to meet sales engineers, service engineers, and often times, the designers and builders of the machines and tools themselves. You can also meet the very top executives of all of the manufacturers, who have insight into where the gear machines and gear tooling industry is going in the foreseeable future. If there's a crystal ball for the gear industry, it's there on the floor of Gear Expo.

Rather than bringing the regular crew that has been coming every two years, maybe it's time to introduce some of your newer, younger engineers and operators to the show—not only to start developing relationships with the exhibitors at the 175 booths, but also to avail your company of the 18 presentations at AGMA's Fall Technical Meeting; AGMA's Gear Manufacturing Basic Course Training School Presentations; the three SME gear seminars held concurrently with the show; and the presentations being held in Gear Expo's Solutions Center (see page 46 for details).

There are plenty of educational opportunities to be had at the show. Surely someone at your company could benefit from this type of gear knowledge and experience. And if they benefit, your company benefits. So even if you can't attend, send someone to Gear Expo. Some time in the future, you can't or won't be able to go anyway. If you're already among those who've seen their fair share of Gear Expos, maybe now is the time to think about passing the reins onto someone else. The opportunity only comes once every two years, and this type of knowledge and experience is cumulative.

I know that it could be very easy to blow off the show, but if you see value in having this collection of knowledge available to you, all in one place, all to be seen and discussed in four days' time, then you need to make the commitment to have someone from your company attend this Show. You can't be two places at once, but your company can.

Maybe doing so will put your mind at ease.

Maybe doing so will be good for your company's future.

Maybe doing so will keep those nightmares at bay.

Michael Goldstein

Michael Goldstein,
Publisher & Editor-in-Chief

P.S. See our coverage of Gear Expo, beginning on page 32. You'll find a show map, list of exhibitors and interviews with some of the companies who will be there. The same information is also available online at www.geartechnology.com/gearexpo. And if you do come to the show, please visit us at booth #210 and enjoy a complimentary cup of espresso or coffee.

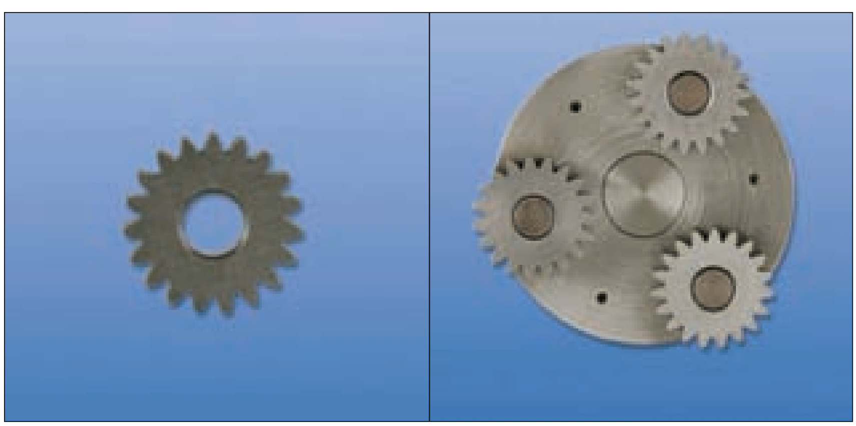
Determining the Knowledge Base Uncovers a Company's Core Competencies

John Walter, president and CEO, Precipart Corporation
Abby Dress, associate professor, Long Island University

The gear market globally exceeds $45 billion today and is growing. Most of the business is dedicated to automotive needs, according to a report funded by the Gear Research Institute (Energetics 1). So, what happens if a company is based in the United States and does not manufacture automotive gearing? How can a company grow its business or plan for growth when its niche area only accesses the smaller part of the pie?

Add to the mix the fact that some gear orders are headed to producers with lower labor costs in China, India and Eastern Europe. Then there is the increasing competition from other technologies that substitute for gears and ultimately reduce the dollars available in certain gear sectors in which a company operates. In particular, electric motors and hydraulic systems are appropriating certain gear applications and potential revenue. What markets then should be considered a best fit with a company's production capabilities is mostly likely the best question to ask.

All of these factors shape the environment in which gear companies operate today. Poor or incorrect decisions for a company could spell disaster for their futures. However, it is often difficult to measure a company's decision-making abilities for long-term growth. Many chief executive officers simply look at various inputs and outputs



Is your company selling a $2 gear when it could be selling a $500 assembly?

with the resulting revenue streams or lack of them. This, in turn, leads to identification of cost and profit centers. But, really this is a short-term view. Though this strategy often provides valuable information, it really does not take into consideration market trends. A perfect case in point: 75 percent of the gear work is targeted to automotive contracts. Should a company address this demand? Or, better yet, how should it compete to increase its share of the remaining 25 percent?

Obviously, gear industry managers are engaged in constant evaluation to determine whether their product applications have been achieved through engineering and production methodologies to meet the expectations of their customers. These areas now are calculated and reviewed regularly for appropriate performance. Of course, the next question is: If customers are satisfied, why should a company look any further, except to meet that need? To be so customer-focused without the perspective of the marketplace, however, could put a company in jeopardy. A case in point is IBM. It missed changes occurring in the early days of the personal computer market due to its customer-focused philosophy and clone policies, which allowed other companies to be created to develop these products and capture its business.

Sometimes, however, more sophisticated analysis is needed for the business as a whole. The "make or buy" choice to meet customer expectations is just one way to discover whether to manufacture parts in-house or outsource. The decision either way could be an

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avenue for the company to reduce costs or increase revenues and profits. It may be an incentive to increase capacity and add customers in new markets at the same time. A decision also may expand the market base—increasing the value of a company's existing share without increasing the actual share percentage. This means as the market grows, the company grows, too. This is not a very proactive approach, but many businesses take this growth route.

Perhaps, however, the most important consideration is the determination of what the company does best. What are the skills of employees or what is the knowledge base that the company possesses?

Management and marketing guru Peter F. Drucker believes that it may seem simple to define what specific knowledge a business owns and at which it excels. However, a knowledge analysis takes practice. In fact, few questions force management into an objective look at itself as does the question: What is our specific core knowledge? (Drucker 117). This really is the critical decision-making determinant for longer-term growth, for it determines a company's core competencies.

Knowledge is not static. It changes. It can become enhanced and also can become obsolete. A product line related to a specific set of information also can disappear as new technologies take over. Remember the rotary phone? A better example, for instance, is Boeing's 787 fuselage. Now it is manufactured as a composite barrel wound on a frame as opposed to a riveted aluminum skin.

There are different options today in the gear industry to take advantage of molded and sintered gears, like those using net surface technologies. Likewise new materials, such as high-strength complex steels and synthetics, permit the use of smaller gears in certain applications. Key to success is the identification of knowledge throughout the organization, from

Manufacturing and assembling a complete gearbox, such as this 3-stage planetary unit, may be more profitable and allow a gear manufacturer to take better advantage of its core competencies.

engineering design to purchasing and production. Such identification helps a company optimize its resource utilization, capitalize on new programs, and stay tuned up for any type of market challenges.

But, one of the first assessments of a company's knowledge base is related to revenue. Is the company giving away some expertise in order to manufacture a gear, gear assembly or gear system? Maybe the company has an engineering department that reviews customer drawings for manufacturability and then recommends changes. While this counseling helps facilitate the company's own production process and keep costs in line, this review is a step that adds value to the customer.

As such, a value-added activity should be associated and tracked with a fee—whether or not a company actually charges the customer for the service. After all, a company expends resources when it conveys built-in knowledge—first to evaluate a drawing, and second to suggest alterations—whether or not this involves a redesign or different materials to increase the program's manufacturability and the customer's requirements. Knowledge about this resource permits it to be targeted for a separate revenue stream down the road.

Knowledge and revenue are tied in other ways, too. A company may devote, for example, 20 percent of its resources to a program that yields only five percent of its total revenue for the year. Is this a wise decision for deployment of resources?

Maybe the decision to take on a highly specialized program enables the company to prove itself in a new product or new market. An articulated arm prosthetic project may position the company as a leader in gearing excellence and design. Reputation helps capture work and perhaps higher pricing. The program, moreover, may serve potentially as a prototype to acquire new business and engage in a higher level of design and manufacturing. It also allows the company to increase its skill base and expand its core competencies at the same time.

Another way a company leverages its knowledge and revenue is through licensing. This may involve two different kinds, however. One involves designs for production like a company that manufactures a fuel pump—a mature product. It can license its production elsewhere. This permits the company to earn income from its proprietary knowledge without tying

continued

up capacity, while it concentrates on developing technologies and acquiring new programs.

At the same time, a review of the company's products may reveal that a group of them can be standardized for sales licensing agreements. A look at outside sales channels may relieve some pressure on an internal resource as well. It is important to assess what an outside source knows about the product's parameters and application. The knowledge base continues to play a critical role in decision-making.

Some situations also force a company to explore size limitations or added capability. A company may consider taking on more downstream work, such as assembly work, because it has usable space, capable individuals, and engineering know-how. The $2 gear that is purchased by a customer now can be supplied in an assembly that generates $500 of income and contributes more profit to the company. Another manufacturer, for example, may make gears up to 6" in diameter. It then receives a requirement from a customer for gears that are 8" in diameter. Is this worth exploring?

Here is where a company's knowledge base drives awareness of core competencies onto the playing field. Should the company leverage its assets and look at these requests as opportunities? Knowing what makes up the company's assets helps. These are its core competencies, which really can be defined as its people's expertise, equipment capability, workflow process, production quality, facility layout, and available space, etc. How these assets are utilized for cost, timing, capacity, and quality affect the decision to take on new or different work.

Maybe individuals with specialized knowledge should be promoted and supported with other staff or with added equipment. These can involve added expenses for the business, which likewise can increase revenues over time. If the company has licensed production partners, this may work, too. Moreover, strategic alliances in which companies collaborate with other manufacturers can broaden the reach of both entities for specialized capability, larger programs, and global applications. These typically are fiduciary relationships, which also can be informal.

Determination of a company's knowledge base is key to successful decision-making for longer-term growth. It provides an accurate picture of what its people know and what the company

Ed Gallow, supervisor of the assembly department of Precipart Corp.

does best. Periodic evaluation, maybe annually, is important. To take advantage of market globalization, capture new business, and compete with lower-cost foreign producers, U.S. gear companies must take stock of their expertise as they widen their horizons. Gears remain "the most efficient, cost-effective means of power transmission available in many applications, a fundamental advantage that will continue to bolster gear use in many sectors of the economy" (Energetics 10). The company that has a profound understanding of itself and of the segments it serves is one that can best address its own future, as well as the future of its markets.

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Haas

Simple Innovation #18

Direct Gear Drive

True Geared-Head

The Haas EC-630 is a powerful 50-taper HMC with a true geared-head. Our unique design couples the motor directly to the spindle through a Haas-built, high-precision gearbox. This smooth-running system is very efficient, and offers tremendous thermal stability The two-speed gearbox provides 450 ft-lb of torque for heavy material removal, and speeds to 6,000 rpm for finish cuts.

To simplify service, the EC-630 spindle, gearbox and motor are assembled as a single modular unit that is easily removed and installed, if necessary.

Koepfer America

LAUNCHES LARGE-SCALE CNC HOB SHARPENING MACHINE

Koepfer America developed the new KFS250 CNC hob sharpening machine, which is based on its KFS100 series.

The KFS250 series has a high-speed, direct-drive grinding spindle, and the ability to sharpen to AGMA "AAA" quality.

According to the company's press release, the KFS250 series' most distinguishing characteristic is its large grinding capacity: 10" maximum hob diameter and 12" maximum hob length. Plus, the KFS250 provides a maximum work-spindle-to-tailstock distance of 25". The five-axis CNC control is managed by a compact GE Fanuc panel with a touch screen color display and conversational programming. The control panel swivels flat into the body of the machine when not in use.

KFS250 series is suitable for carbide and high-speed steel tools using vitrified, CBN or diamond abrasive. The series also features pre-loaded linear guide ways, AC servo motors with pre-loaded ball screws for linear axes, integrated crane for loading and unloading larger hobs, high-volume coolant flow for wet grinding, and a centrifuge filtration system that is integrated into the machine bed for a single, consolidated footprint and shop floor cleanliness.

According to Koepfer's press release, the clarity of the filtration system is rated at 2 to 5 microns. The KFS250 series sharpens both straight-gash and spiral-gash hobs.

Koepfer America will be displaying a KFS100 hob sharpener, a Koepfer Model 300 hobbing machine, Monnier + Zahner & Lambert-Wahli fine pitch hobbing and worm-milling machines, and Wenzel GearTec gear inspection machines at Booth #122 during Gear Expo 2007.

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Morgan Advanced Ceramics

INTRODUCES NEW COATINGS FOR THE RACING INDUSTRY

The Diamonex Products division of Morgan Advanced Ceramics (MAC) introduced its Diamonex® Diamond-like Coatings (DLC) for the performance racing industry.

The DLC coating is a thin-surface coating that provides high hardness, improved chemical resistance and low friction. Best suited for applications in the high-performance racing industry, the DLC coatings increase wear life and reduce surface friction of internal engine components, including intake valves, wrist pins, spring retainers and cylinder sleeves. The DLC coatings also offer the same attributes when camshafts are coated. According to the company's press release, the DLC coating provides 40% less friction than chrome coating and 10% greater surface hardness than tin, leading to less friction-oriented power loss and more wear protection.

In addition to the aforementioned improvements, the coatings also reduce engine sludge build-up, allowing the engine to maintain balance for more horsepower.

Other Diamonex DLC racing industry applications include coating shock shafts and fork tubes for motocross and motorcycle racing. This helps reduce friction, which minimizes scoring on critical surfaces of these components.

Using patented plasma chemical vapor deposition (CVD) technologies, Diamonex Products creates a coating with chemical and physical properties comparable to diamond, but without long-range crystalline order.

Diamonex DLC coatings are applied to the product at near-ambient temperature and can maintain tight dimensional tolerances. A high degree of vapor ionization is used in Diamonex Products' DLC processes.

For more information:

Morgan Advanced Ceramics
Phone: (800) 433-0638
E-mail:
ussales@morganadvancedceramics.com
Internet:
www.morganadvancedceramics.com

Sunnen's SV-310 Bore Sizing System

DESIGNED FOR SMALL ENGINES, COMPRESSORS

Sunnen's new SV-310 vertical CNC honing system combines a choice of tooling options, large work envelope, and 762 mm (30") stroke for bore sizing of small and medium-sized gas/diesel engine blocks, large gears, air compressors, aerospace parts, refrigeration compressors and similar small or large multi-bore parts.

A work envelope of 1,016 mm (40") x 1,778 mm (70") and weight capacity up to 909 kg (2,000 lbs.) provide versatility for processing a wide range of parts. An optional tool-guide assembly and variety of bore-diameter gaging systems combine with a servo-controlled X-axis travel of 1,143 mm (45") to allow automated honing of multi-bore blocks with high precision.

Straightness sensing capability provides a display of the bore profile during the cycle. Combined with a patented, servo stroking system, it allows the SV-310 to auto-dwell in any part of the bore to correct straightness automatically in a short cycle time. The

continued

SV-310 can be used to hone parts with inside diameters from 19–200 mm (0.75–8.00"), depending on the tool options selected.

Designed for automated processing of mid- to high-part volumes, the SV-310 features full-height access doors and an enclosure preconfigured for use with robotic part loading systems, while the CNC control includes a built-in automation interface. A 20-amp power supply and multiple E-stop contacts add versatility and allow automation control. Setup is simplified with a three-axis hand wheel for fine-tuning X-axis position, vertical stroke and tool feed. The Windows®-based, color touch-screen control features multi-language capability, pull-down menus, unlimited capacity for job set-ups, and programmable custom tools.

In addition to optional in-process air gaging, a variety of optional post-process air-gage systems can integrate with the control for SPC data collection, as well as automatic compensation of size, taper and straightness. The control allows easier programming of multiple bore positions.

The SV-310 is built on a malleable iron base for rigidity and vibration damping. The machine's servo-controlled, straight-line stroke motion is driven by a ballscrew at rates of 1–160 strokes per minute, while a powerful 5.5 kW (7.5 hp) spindle provides ample torque for fast metal removal with tools outfitted with metal-bond cubic boron nitride and diamond abrasives or standard aluminum oxide and silicon carbide stones. Oversized, lubed-for-life guide ways for horizontal and vertical axes provide smoother operation.

An optional T-slot table 800 x 1,545 x 45.5 mm can be mounted on the casting base, along with an elevated fixture frame for holding all types of fixtures.

The SV-310 can be equipped with a wide range of Sunnen tooling, such as the new PH precision multi-stone tool. The PH tool can hold as many as

continued

12 stones for optimum straightness and roundness on bores 76.2–190.5 mm (3.00–7.5"), while delivering efficiency in mid- and high-volume production. It is available in single or two-stage configurations, along with optional, integral air-gaging for in-process monitoring.

For more information:
Sunnen Products Co.
Phone: (800) 325-3670
E-mail: *sales@sunnen.com*
Internet: *www.sunnen.com*

Trico Corp.'s Dispensers

LUBRICATE CRITICAL PARTS DURING MACHINE OPERATION

Trico's new Streamliner grease dispensers are an alternative to manual lubrication. These units are designed for applications where it is essential to have a precise amount of grease delivered. Machine and plant downtime due to critical component failure is also reduced.

Units provide variable feed rates lasting from two weeks to 24 months. Trico's Streamliner grease dispensers feed up to eight lubrication points. They also provide precise metering of lubricant in remote areas up to 30 feet. They are equipped with electromechanical drives to adapt to various grease applications.

According to the company's press release, the Streamliner grease dispensers have high operating pressures to flush contaminants out and for multipoint lubrication. The economical reservoir capacities, up to 500 cc, provide for fewer replacements and less disposal. A UL explosion-proof unit is available as well.

For more information:
Trico Corp.
Phone: (800) 558-7008
E-mail: *lkiepert@tricocorp.com*
Internet: *www.tricocorp.com*

Emag's New VMCs

APPLY SIMULTANEOUS FIVE-AXIS MACHINING

EMAG's BA S03 vertical machining center is designed as a monobloc enclosed on three sides and topped by a gantry and combines a minimal footprint, extreme rigidity and protection of guideways and drive components against the ingress of chip particles and coolant.

Its degree of rigidity is increased by the box-in-box design of the three-

axis unit and features ball screw drives in the linear and direct drives in rotary axes. The torque drives guarantee backlash-free operation and a gantry drive in the Y-axis ensures parallel movement of the slide, even where cutting forces exert off-center pressure. The direct drives for the swivel axes, in conjunction with direct measuring systems, offer a positioning accuracy of +/-5 angular seconds. This ensures the positioning accuracy Tp achieved in the machining area of the BA S03 is 0.01 mm to VDI DGQ 3441. The acceleration of up to 10 m/s2 linear axis speeds of max 75 m/min and rotary axis speeds up to 50 rpm ensure that idle times are kept to a minimum.

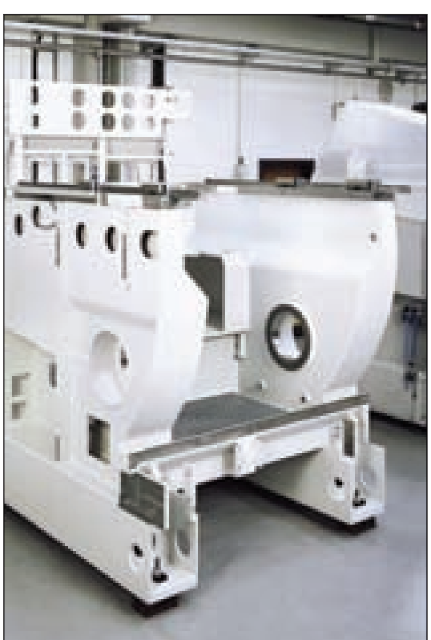
The company has been offering manufacturing solutions utilizing twin-spindle, five-axis machining processes since 2004.

"Our twin-spindle solution gives us an advantage over the single-spindle process, especially when it comes to manufacturing of impellers, which calls

for relatively few tools. All it usually involves is between three and five tool changes. With the SW chuck, the workpieces are quickly loaded and unloaded. In fact, the twin-spindle design of the BAS03 allows us to double production times," says Erik Pfeiffer, manager of technological development at Schwäbische Werzeugmaschinen GmbH.

According to the company's press release, the SW is applicable for small batch production. Where parts are milled from solid stock, the raw material (aluminum or titanium) is held in two concentrically clamping universal chucks. Emag says this makes resetting quicker and easier as all component variants can be clamped in the same chuck.

To arrive at a cycle time that is as accurate as possible and generate and optimize the NC program in the early stages of the project, SW employs the 3-D machine simulation system eM RealNC for its five-axis machining operations. All NC axes are parameterized (speed, acceleration, jerk rate, V-constant, etc.) for the purpose. In the same way, movements of the chain type tool magazine are simulated to be able to assess the time taken for tool changes as accurately as possible. Canned cycles and subroutines are also displayed. Users find security in studying possible collision factors and reducing idle times. With the simulation, Emag aims for realistic cycle times that provide a basis for optimal capacity planning before a chip has fallen.



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Internet: *www.sw-machines.de*

Alpha Gear Drives

INTRODUCES NEW GEARHEAD

The inline basic precision alphira product from Alpha Gear is packaged in a lightweight aluminum case and offers power-dense and maintenance-free operation.

According to the company's press release, the alphira's mounting dimensions make it suitable for low-precision applications.

Technical characteristics include four sizes (40, 60, 80, 115), a weight of 2.53 kg, efficiency greater than 95%, low noise level, simplified motor mounting, service life of more than 20,000 hours and lifelong lubrication and sealing.

For more information:
Alpha Gear Drives
E-mail: *info@alphagear.com*
Internet: *www.alphagear.com*

Meister Abrasives

DEVELOPS EXTERNAL PRODUCT LINE

Meister Abrasives USA recently developed a line of external abrasive products.

Bruce Northrup, general manager of Meister Abrasives USA, says users report cycles up to 20% faster and/or cost per part reductions up to 30%.

continued


Bilek Krämer Jäger

Meister's external wheels are customized for each application and feature top-shelf raw material and state-of-the-art bonding technologies, including its newest high-performance bond HPB. Furthermore, wheel specifications can be tuned to provide consistent high-volume manufacturing or the versatility that allows smaller shops to use a single wheel on a number of different applications.

Applications include O.D. cylindrical, creep feed, surface, double disk, wafer backgrinding, as well as fine grinding and lapping. The most active markets for these products include the medical, aerospace, semiconductor, optical, tool and die, and automotive industries.

For more information:
Meister Abrasives USA, Inc.
201 Circuit Dr.
North Kingstown, RI 02852
Phone: (401) 294-2530
Internet: *www.meister-abrasives.com*

ONLINE

Product News is updated online. For the latest product information, go to *www.geartechnology.com*

Pico Chemical's New Solvent

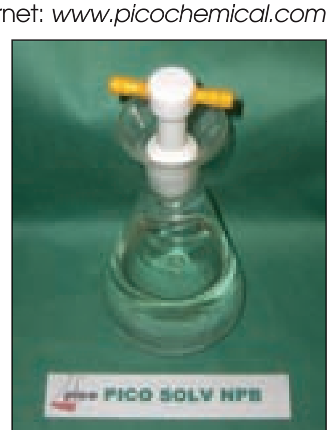
REPLACES HAZARDOUS MATERIALS FOR GEAR CLEANING

Pico Chemical Corp released its Pico Solv NPB as a new-technology, chlorine-free solvent cleaner that is manufactured specifically to replace hazardous and chemical solvents that clean gears.

The new cleaner removes cutting, machining, lubricating and stamping oils as well as greases and protective waxes. It evaporates quickly, has no flash point, is chemically stable and leaves no films or residues.

According to the company's press release, metalworking plants, metal service centers, fabricating and finishing facilities and rebuild shops are target consumers. Typical applications include oil removal from screw machined parts, dirty electrical components, oxygen service cleaning and coil-to-coil metal cleaning.

For more information:
Pico Chemical
E-mail: *jmanfreda@picochemical.com*
Internet: *www.picochemical.com*



Emuge

INTRODUCES NEW THREAD MILLS

Emuge Corp. introduced a new series of miniature solid carbide thread mills that are optimally designed for easier machining of difficult materials such as stainless steels, titanium, K-Monel, Hastelloy, and Inconel.

The line is suitable for a variety of applications, including those in the medical, aerospace, defense, and computer industries.

According to the company's press release, the new thread mills eliminate the possibility and consequences of tap breakage. Thread mills are available in both one-flute and three-flute versions, with the latter being suitable for high-volume output needs. The new thread mills remove manual operations by eliminating the need for hand tapping during full bottom threading applications. Bottom threading can be performed to within 1 pitch. The miniature thread mills function with externally supplied coolant and eliminate the need for thread cutting oil, so parts and coolant reservoirs are not contaminated.

The new miniature solid carbide thread mills include an easily controlled pitch diameter and offer precise thread depth controls, up to 2xD, so that one tool can be used for both through and blind holes. Overall length of the thread mill is 1 5/8" (41 mm) with a shank diameter of 1/8"(3.175 mm). The line includes thread sizes of #0-80, #2-56, #4-40, #5-40, #6-32 and #8-32 and cutting diameters from 0.045–0.124" (1.143–3.15 mm). Thread length ranges from 0.125– 0.328" (3.175–8.33 mm).

For more information:
Emuge Corp.
1800 Century Dr.
West Boyslton, MA 01583
Phone: (800) 323-3013
E-mail: *emuge@emuge.com*
Internet: *www.emuge.com*

Toyoda Machinery

INTRODUCES NEW TURNING CENTER

Toyoda Machinery will introduce its newest machine line at the Canadian Machine Tool Show from October 15–18 in Booth 3600 with its new Canadian distributor Elliott Matsuura.

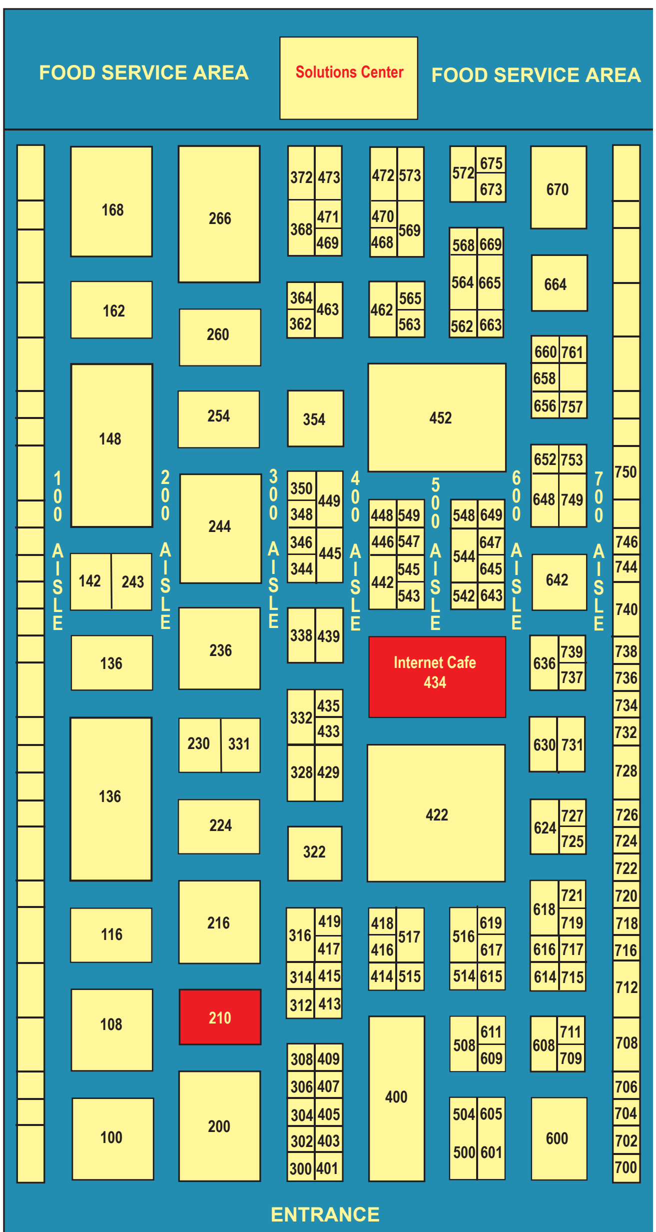
The inverted turning centers have a traveling inverted spindle that allows for direct part pick up and delivers improvement over other chip management designs. A mono-block polymerized concrete-filled, steel fabrication base delivers vibration damping.

The horizontal SX Series has a 40 horsepower, 6,000 rpm spindle (with a 15,000 rpm spindle option). The spindle has double the driving force on the Y-axis with a ballscrew and motor on each side. A column helps support the spindle's cutting power to maintain accuracy in hard ferrous materials and maximize tool life. The new machines are available with 550 mm or 630 mm pallet sizes, and the table load capacity is 2,860 lbs.

For more information:
Toyoda Machinery Co.
Phone: (847) 253-0340
Internet: *www.toyodausa.com*

The Unofficial Guide to Gear Expo 2007

October 7–10, 2007
Cobo Conference/Exhibition Center
Detroit, MI



ALPHABETICAL LIST OF EXHIBITORS

See page 33 for these listings arranged by booth number. All listings current as of August 8, 2007. See *www.gearexpo.com* or *www.geartechnology.com* for updates.

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ADF Systems Ltd.	463
Advanced Nitriding Solutions LLC	717
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American Bearing Manufacturers Association	711
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American Stress Technologies	433
American Wind Energy Association	704
Ancon Gear & Instrument Corp.	614
Ann Arbor Machine Gear Technologies	562
Applied Process Inc.	548
Arrow Gear Co.	472
Ash Gear & Supply Corp.	645
B&R Machine & Gear Corp.	508
Barit International Corp.	542
Bevel Gears India Pvt. Ltd.	656
Bodycote Thermal Processing Inc.	568
Broach Masters/Universal Gear Co.	544
Broaching Machine Specialties Co.	435
Cadillac Machinery Co. Inc.	210

Canton Drop Forge	721
Capstan Atlantic	448
Carraro Argentina	564
Chamfermatic Inc.	469
Cincinnati Gearing Systems	230
Clifford-Jacobs Forging Co.	515
Colonial Tool Group Inc.	600
Comtorgage Corp.	405
Cornell Forge	615
Creative Automation Inc.	407
Custom Gear & Machine Inc.	401
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Delta Research Corp.	675
DixiTech CNC	648
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Dragon Cutting Tools	142
Drake Manufacturing Inc.	468
DSM Engineering Plastics	547
Dupont	514
Eaton Steel Bar Co.	749
ECM-USA Inc.	417
Eldec Induction USA	445
Eltro Services Inc.	446
Emuge Corp.	500
Engineered Abrasives	429
Engineered Tools Corp.	517
Euro-Tech Corp.	331
Extrude Hone Corp.	715
Fairfield Manufacturing Co. Inc.	516
Fässler Corp.	100
Felss LLC	565
Finish Up Inc.	746
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Gear & Broach Inc.	338
Gear Solutions magazine	332
Gear Technology, The Journal of Gear Manufacturing	210
Gearspect USA	609
Geiger Machinen-Vertrieb GmbH	300
Gibbs Machinery Co.	728
Gleason Corp.	422
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Great Taiwan Gear Ltd.	618
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Harbin Tool Works	716
Hexagon Metrology	761
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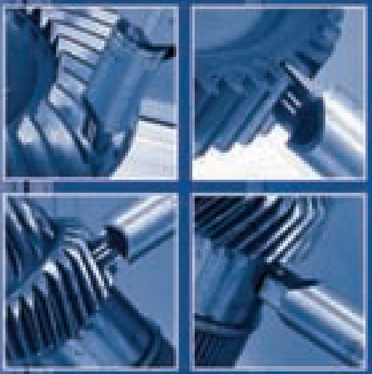

Gear Expo 2007—You'll Be There, Correct?

"Eighty percent of success is showing up"
—Woody Allen

Jack McGuinn, senior editor

Woody Allen was talking about success in show business when he uttered those words, but you can visit any number of corporate meeting rooms and sales centers having nothing to do with the entertainment arts and find that quote emblazoned on the wall.

It's no stretch to believe the same can be said in support of exhibiting at Gear Expo 2007. Moaning and carping about exhibiting at trade shows have existed since perhaps the Columbian Exposition, but here we are in 2007 and another Gear Expo is upon us. And one premise remains etched in stone—you cannot win if you do not play. So it is that—as we go to press—there will be at least 159 companies setting up shop and selling their wares and services to attendees from around the globe.

Gear Technolgy was able to grab a few minutes with a small but representative sampling of this year's exhibiting companies. Call it what you will—a necessary evil, an invaluable sales and marketing opportunity, or simply a way to help support the worldwide gear industry—the following companies will be there with bells on and a whole lot more.

Company: Gleason Corp.
Booth: 422
Contact: Al Finegan, director of marketing
What's New: The 210-H Genesis hobber will make its first show appearance; capabilities-upgraded Sigma 175GMM CNC gear analysis machine; a new line of cutting tools for large cylindrical gear applications.
Why Gear Expo?: "The market in the U.S. has not been particularly good, and there hasn't been a whole lot of investment going on in this country and it's disturbing. So we're looking forward to it."

Company: KAPP Technologies
Booth: 244
Contact: Bill Miller, vice president sales
What's New: The KX300P with a significant upgrade; i.e., flexible dressing of worm gears allowing for quicker setups, job runs and reduced lead time.

Why Gear Expo?: "It has become more and more a good venue for a lot of customers to see and touch the latest equipment. You have to be there. I always equate it to the family reunion where you have to go or they talk about you."

Company: Koepfer America

Booth: 122

Contact: Dennis Gimpert, Koepfer America president; Chris Otte, marketing

What's New: Redesigned Wahli 100 CNC machine for extremely fine-pitch applications (instrumentation, etc.); its enhanced velocity now provides spindle control even at very high speeds. Also, not in the booth, but a formal announcement will be made of the KFS250 hob-sharpening machine. Koepfer will share its booth with Emag and its VL 3 CNC vertical turning machine.

Why Gear Expo?: *Gimpert*: "We have great success at the shows. I think some people complain because they don't have equipment (at the show) and nobody wants to come into their booth and just stand around and drink a Coke. It is expensive, but if you do it intelligently, I think there's a return on it." *Otte*: "I think it's important for us to be there so that attendees can see our machine. They may have heard about or saw it in an ad, but now they can see it actually being demoed."

Company: Great Lakes Gear Technologies, Inc.

Booth: 100

Contact: Ray Mackowsky, Great Lakes president

What's New: One-meter Höfler HS 900 hobber weighing in at approximately 50,000 lbs. Popular in Europe, it is part of the official introduction of Höfler hobbers to the U.S. market. Also on hand will be a sampling of the Escofier line of gear and spline roll-forming quipment. (The Great Lakes booth will also have a Fässler presence.)

Why Gear Expo?: "It's always been a good venue for us. All our principals—Hofler, Fässler—have always been exhibitors—through good times and bad—with leading-edge technology. I'm excited."

Company: DuPont North America

Booth: 514

Contact: Rob Johnson, DuPont North America gear team leader

What's New: Nothing specific regarding new materials. Booth personnel will be available to answer questions regarding existing resins, etc., as well as to work with customers with application-specific questions.

Why Gear Expo?: "We have a history of being there for many years; it helps develop new leads and provides us with a market presence. It also helps us find out what's on the minds of customers in terms of what they'd like to see."

Company: Capstan Atlantic

Booth: 448

Contact: Rich Slattery, vice president engineering; Mike Smith, vice president sales and marketing.

What's New: Solutions Center presentation of four case studies by Slattery regarding Capstan's net shape capabilities and the conversion from wrought steel parts to parts made using powdered metallurgy. Featured are Capstan's patented HD4 and HD4R processes relative to high-dense, high-performance precision gearing applications. The presentations will address high-performance transfer case sprockets, crankshaft gears for lawn and garden applications; custom tooth shape for enhanced noise reduction; and transmission gears.

Why Gear Expo?: *Slattery*: "I'm there as a customer; I go to see presentations to see how technical information is presented about gears. I like to walk the show to make sure I'm up to speed on the latest gear processing equipment. *Smith*: "We participate for the exposure and because it allows us to present new technologies to a larger audience, as well as technical papers and workshops. It also allows us to support the industry as well."

Company: Forest City Gear

Booth: 116

Contact: Everett Hawkins, vice president of manufacturing

What's New: The company will use the show as a platform to tout its lean efforts over the last year or so, and how that has greatly improved both quality and thru-put in their operation. FCG will also share with attendees the number of latest technology machine tools they have acquired recently, demonstrating their dedication to continuous improvement.

Why Gear Expo?: "We look at the show as an opportunity to stress our strengths to customers, as well as our success over the last three years."

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E-mail: *sales@koepferamerica.com*
Website: *www.koepferamerica.com*

Solutions Center Educates Gear Expo Attendees

In 2005, Gear Expo debuted its first Solutions Center, a forum that features short exhibitor presentations on gear-related topics. This year, AGMA says the Solutions Center will return with a slightly different format.

"Based on the success of the Solutions Center in 2005, we again gave exhibitors the chance to present short presentations at the Solutions Center. This year will also feature two keynote speakers—Monday will have Dr. Mike Bradley's 'Gear Market Report,' which was successful in 2005. On Tuesday, the keynote speaker will be Casey Selecman, an automotive industry expert from CSM Worldwide, presenting 'The Outlook for the Automotive Industry,'" says Kurt Medert, vice president at AGMA.

As of August 20, AGMA has scheduled 28 free Solutions Center presentations. All presentations will be held in the rear of the exhibition hall and are open to any show visitors.

Sunday, October 7

2:00 p.m. Gearspect. *DO3PC User-friendly Gear Measurement in the Workshop* Gearspect (Prague, Czech Republic) presents DO3PC, a gear measurement instrument designed for use in the workshop environment. European gearbox and automotive manufacturers requiring fast decisions and SPC about quality of splines and spur gear parameters such as gear profile, helix, pitch and deviations according to AGMA, DIN or JIS, are using DO3PC directly in the workshop. Although classified as a VDI class II instrument, measurement values using this strategy are compatible with the results obtained in the measurement room with class I gear measuring machines.

2:30 p.m. Fässler Corporation. *New Developments in Gear Honing.* The new developments in gear honing are focused on the use of ceramic bonded honing stones. The extremely high stock removal rates of this new honing stone result in very short process times. In addition the new machine concept keeps idle time low.

3:00 p.m. Ticona Engineering Polymers. *Producing Precision Polymer Gears*. Staff and operating procedures are often the deciding factor in the production of precision engineering polymer gears. Learn how gear molders can adapt their molding environment and alter how they function so the gears they make are as uniform as possible from shot to shot and cavity to cavity. Get the most out of your molding machine, tooling and engineering polymer.

3:30 p.m. Kleiss Gears Inc. *The Importance of Measurement in Producing Precision Repeatability in Gearing*. Many people can produce plastic gears. Only a very few companies can produce an engineered polymer gear with dimensional control of less than .001". Kleiss' Douglas Felsenthal will discuss the advantage of CNC measurement in assuring that custom shapes are met. Knowing the interaction and relationship between tool cavity and final part are crucial in the repeatability process.

Monday, October 8

9:00 a.m. Omax Corporation. *Gear Making Applications for Precision Abrasive Water Jets.* Discussion of making gears with precision abrasive water jets.

9:30 a.m. Applied Process, Inc. *Austempered Ductile Iron (ADI) for Gears & Powertrain Components*. Properties and applications of ADI gears and powertrain will be discussed along with specific case studies. A review of AGMA 939-A07 Austempered Ductile Iron (ADI) for Gears Information Sheet will also be included.

10:00 a.m. Gleason Corporation. *How Are You Dealing With Bias Error?* This presentation will define bias error, explain why it occurs, how to prevent it, and how to induce it when desired.

11:00 a.m. KEYNOTE PRESENTATION: *Dr. Mike Bradley Gear Market Report*.

1:00 p.m. Metal Improvement Company. *Shot Peening of Gears and Shafts.* The fatigue properties of power transmission components can be improved by the application of residual compressive stress from shot peening.

1:30 p.m. KissSoft USA LLC. *Accurate Bearing Calculation Considering the Load on the Single Ball/Rolls.* A new ISO 16281 Standard (or DIN ISO 281-4) describes the calculation method to check the load and pressure distribution on the individual balls or rolls of a bearing. Examples will be shown that illustrate the difference between the classical calculation method and the more accurate method included in the *KISSsoft* calculation program based on ISO 281-4.

2:00 p.m. Reishauer Corporation. *The Manufacturing and Application of Diamond and CBN Tools.* This presentation will describe the entire production process and include topics such as electroplating, superabrasive grain size selection, blank manufacturing, and final validation procedures. Optional design for diamond dressing tools based on customer specific applications will also be discussed.

2:30 p.m. Comtorgage Corporation. *Hand-Held Gaging for Gears & Splines.* Overview of various gage designs offered by Comtorgage for measuring: between pin dimensions, over pin dimensions, major diameter, minor diameter, outside diameter, bore and keyway widths.

3:00 p.m. Thermotech Company. *Introduction to Direct Gear Design®.* Direct Gear Design® is an alternative approach to traditional design and analysis of involute gears. It separates gear geometry definition from the generating rack and tool selection to achieve higher gear transmission performance for a particular product and application. Direct Gear Design allows for significant increase in gear load capacity, efficiency and reliability; reduced size, weight, noise, and vibration of gear transmissions.

3:30 p.m. Midwest Thermal-Vac, Inc. *Latest Advancements in Vacuum Carburizing with High Pressure Gas Quenching.* Higher pressures with different gas mixtures are allowing this new technology to perform even better than first expected.

4:00 p.m. Felss LLC. *Spline Forming with Frequency Modulation.* This unique and patented manufacturing process reduces the forming forces by up to 50%. In comparison with other forming processes, frequency modulation spline forming provides economical and technical advantages. The presentation will develop them and explain the process.

Tuesday, October 9

9:00 a.m. ORT Italia. *Advancements in Spline Rolling Technology.* Latest technical innovations in cylindrical progressive spline rolling technology.

9:30 a.m. Eltro Services, Inc. *Plasma Nitriding of Gears.* Overview of patented pulsed plasma surface enhancement technologies to plasma nitride, nitrocarburize and oxy-nitride. All types of ferrous-based gears. Homogeneous case and properties and tip and root per gear tooth. No dimension or core hardness change.

10:00 a.m. Capstan Atlantic. *Net Shape Processing of High-Performance Gears.* A series of case studies focusing on net shape processing alternatives for gear manufacturing. Alternative systems for 8620 carburized and 4140 induction heat treated gears will be evaluated.

10:30 a.m. KissSoft USA LLC. *A Customer Perspective on Gear Software: Develop In-House or Purchase*? Charles Schultz, vice president of engineering, Brad Foote Gear Works, Inc., will present a customer's perspective highlighting the pitfalls of developing and maintaining custom gear design software versus adopting commercially available software solutions from KissSoft. Actual software examples of both solutions will be incorporated.

11:00 a.m. KEYNOTE PRESENTATION: *The Outlook for the Automotive Industry.* Casey Selecman, manager, North America Power Train forecasts, of CSM Worldwide

1:00 p.m. Gleason Corporation. *Measurement Traceability for Gears.* The increasing influence of ISO standards on business and industry has resulted in revised definitions for many fundamental terms. One such revision applies to the term "traceability," which now involves estimation of measurement uncertainty. This presentation will clarify the new definition and outline an approach to its proper implementation.

1:30 p.m. S.L. Munson/Dr. Kaiser. *Diamond Dressers for Gear Industry.* Discussing different types of dressing products used in the gear industry.

2:00 p.m. ECM USA, Inc. *Fatigue Strength Property Improvement on Gearbox Components by Using Low-Pressure Carburizing, Low-Pressure Carbonitrding and Gas Quenching.* ECM will present a comparison study on several processes: Conventional carbonitriding and carburizing with oil quenching, low-pressure/carburizing and carbonitriding with gas quenching. A complete metallurgical analysis and mechanical properties results will be discussed. ECM will show how the optimization of carbon enrichment and the mastering of the gas quenching can improve fatigue strength.

2:30 p.m. Chamfermatic. *Gear Deburring*: Latest tooling and machinery in gear deburring.

3:00 p.m. Seco/Warwick. *Increasing Production Time in Vacuum Carburizing Furnaces.* Pit and Rotary Hearth Furnace systems are effective furnace systems for both heat treating and handling gears. Atmosphere furnaces are very forgiving and produce good metallurgical results when heated by either natural gas or resistance heating elements. Each system provides unique benefits in material handling and plant production work flow.

3:30 p.m. Frenco Euro-Tech Corporation. *Technical Spline Seminars and the Holy Grail: One Does Exist.* You can search a lifetime for the holy grail and never find it. If you search for a technical spline seminar you will find only one. Norbert Weiss from Frenco will explain how manufacturers and suppliers can be on the same page, settle gaging and design disputes, educate operators on spline gaging and discuss all technical aspects of splines and standards, all in a simple one- or two-day seminar.

4:00 p.m. Eldec Induction USA. *Simultaneous Dual Frequency Induction Gear Hardening.* A presentation describing the evolution of utilizing simultaneous dual frequency induction hardening techniques, and how it applies well to gear hardening requirements.

2007 GEAR EXPO Show Sto

The New Freedoms: Three- & Four-Face Ground Bevel Gear Cutting Blades

Dr. Hermann J. Stadtfeld

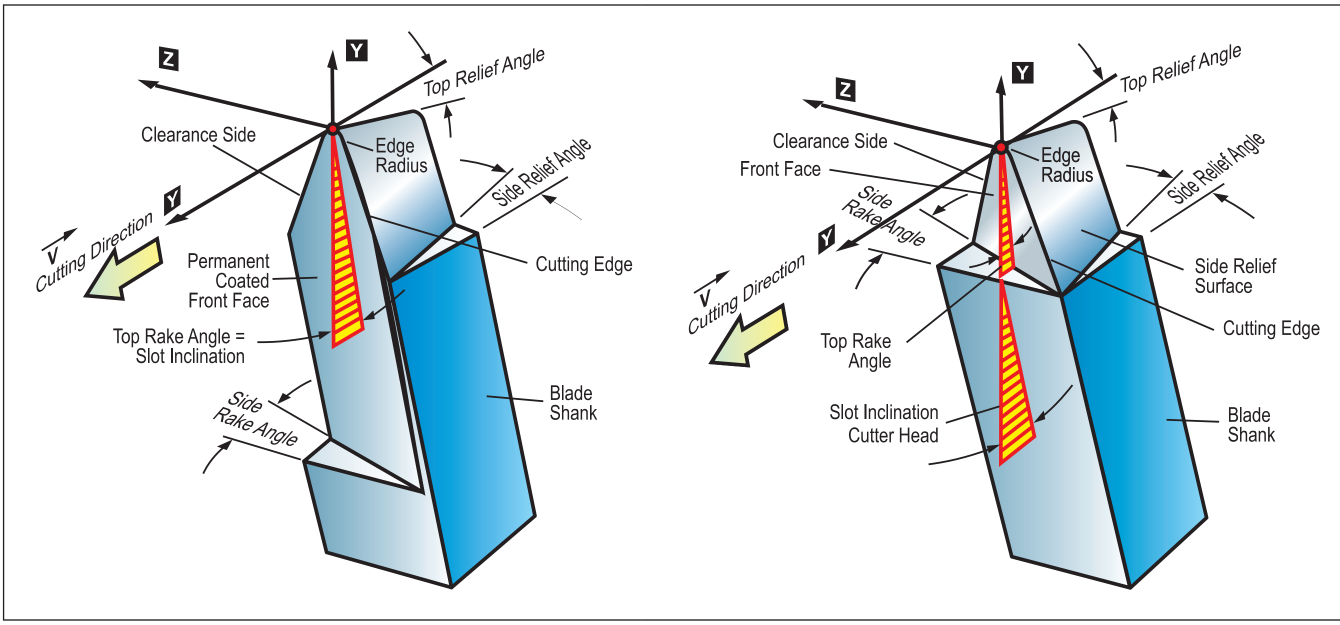

Figure 1—Blade angles and blade nomenclature: left, two-face; right, three-face-ground.

Management Summary

In the 1970s, three-face-sharpened HSS blades were used effectively in bevel and hypoid gear cutting, providing flexibility in blade geometry. When TiN coatings became available, the three-face sharpened blades gave way to two-face sharpened blades with the front face permanently coated. Because of the cost of coating and the turnaround time required, all-around coatings of three-face-sharpened baldes were not economically justifiable—even into the late 1990s when TiAIN coatings and carbide stick blades became the norm.

Today, because of reduced cost of coatings and quicker turnaround times, the idea of all-around coating on three-face-sharpened blades is again economically viable, allowing manufacturers greater freedoms in cutting blade parameters, including three-face-sharpened and even four-face-sharpened blades.

Introduction

Three-face sharpening was used in older face hobbing systems in order to provide the ultimate flexibility in blade geometry, but also to have the freedom to compensate for the discrepancy between the fixed-blade offset in the cutter head and the kinematic offset required for the cutting of individual gear sets (Ref. 1). This system was very successful at the time where uncoated powder metal high-speed steel (HSS) was used and advanced coatings were not available.

When TiN coating, applied to the front face of the blade, showed a significant increase of tool life and surface finish, the two-face sharpened blades with permanent-coated front face became the state-of-the-art cutting tool of choice. For face milling, this did not require a certain cutter head design. For face hobbing, comprehensive studies that included all possible gear set designs were required to find a cutter design offset which was less than 2° different from the extreme values of the kinematic offset within the expected variety of jobs. Only this could guarantee an adequate side rake angle for all real gear designs because of the fixed-blade front face. Other technological angles such as side-and top-relief angle could still be chosen freely since those surfaces were ground during the sharpening operation.

The high cost of coating and the turnaround time between the coating facility and the gear manufacturer did not justify an all-around coating in connection with a three-face sharpening—not even after 1997, when high-speed cutting with TiAlN-coated carbide blades began its success story (Ref. 2). The PowerCutting process used blades with permanent-coated front face, which were two-face-sharpened at the side relief and clearance side surface.

Already in the late 1990s, some manufacturers tried the advantages of all-around coating; however, this was mostly experimental and no advantage was taken of the possibilities the freedoms of flexible front-face design could provide.

Today, modern coatings are readily available at nearby coating facilities. The cost of re-coating, as well as the turn-around time, has become more reasonable. This led to the desire of many manufacturers to take advantage of both the flexible design of all technological blade surfaces and the protection of the cutting edge from the front face to the side relief surface. Figure 1 left, shows a blade with permanent front face and to the right, a three-face-ground blade.

All-around coating could be questionable or even impossible if all three functional blade surfaces of the cutting blade are not ground before re-coating. The grinding provides a mechanical removal of all coating on the functional blade surfaces. A chemical stripping of the coating is required if two-face-ground blades should be re-coated. The chemical stripping can cause cobalt leaching and can only be repeated about six times before a degradation of the carbide grit is noticeable. In view of this conflict, the mechanical stripping by three-face sharpening in combination with the additional freedoms in the blade parameters seems an ideal combination which is beneficial for all gear sets cut in larger quantities.

Traditional Blade Angle Definition

The precise values of the blade angles are today only known relative to the blade shank and relative to the cutter heads where the stick blades are assembled. Figure 2 shows blades assembled in a cutter head under a certain slot inclination. The left blade in Figure 2 has a positive hook angle. This means the cutting edge, observed from the side, perpendicular to the visible side of the blade shank, is inclined counterclockwise. The blade in the center of Figure 2 has a neutral or zero hook angle, where the graphic at the right side in Figure 2 shows a blade with a negative hook angle.

The definition of the side rake angle is analogous to the hook angle definition. Figure 3 shows a graphic with a positive side rake angle in the top portion, a neutral side rake angle in the center and a negative side rake angle at the bottom. The observer looks perpendicular to the face of a cutter head and measures the side rake angle in a plane parallel to the cutter front face (Ref. 3).

The traditional angle designation is a static or geometric definition, which does not consider the relative cutting velocity nor the direction of chip flow. The relative cutting velocity between work and blade and the cutting plane, which has the relative cutting velocity vector as normal vector and includes the calculation point of a blade as surface point, is introduced in Figure 4. The kinematic or effective blade angles can be calculated and expressed in relationship to the cutting plane. Gleason three-face blade grinding summaries include additionally to the traditional angles the effective blade angles.

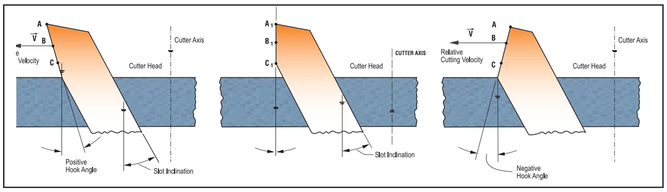

Figure 2—Positive hook angle (left), neutral hook angle (center), negative hook angle (right).

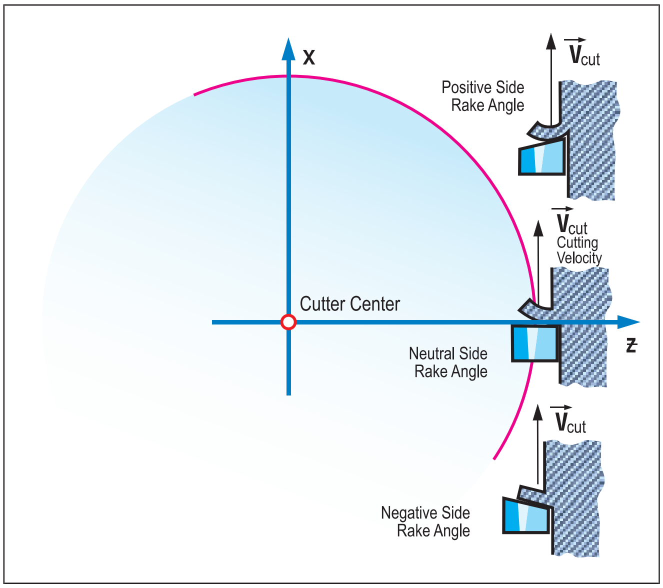

Figure 3—Side rake angles: positive at top, neutral in center, negative at bottom.

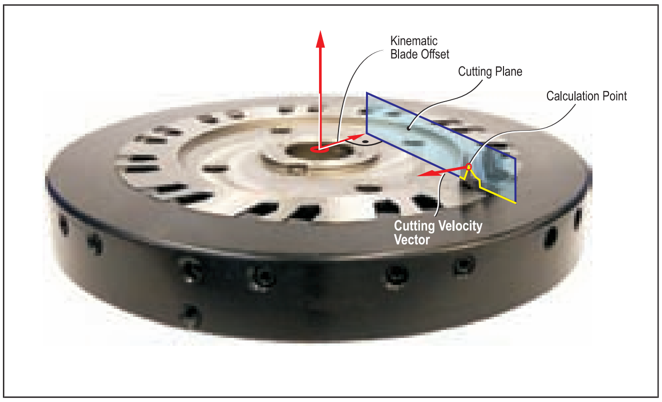

Figure 4—Kinematic blade angle definition using cutting plane.

Blade Geometry Features

In face milling—but more so in face hobbing—the kinematic blade geometry differs from the aforementioned static blade geometry and in most cases is not shown in summaries. In the past, it was not possible to optimize or even change the kinematic angles, sometimes also referred to as dynamic blade angles.

Extensive studies of the cutting action in connection with three-face blade designs have led to important discoveries and are supported in new software tools for the design of

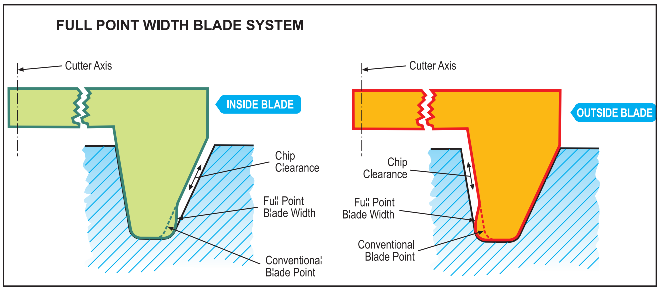

Figure 5—F-point blades use the entire root width.

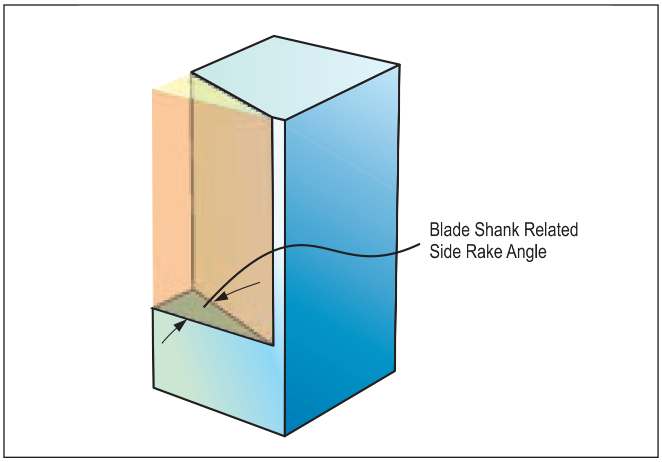

Figure 6—Traditional definition of side rake angle.

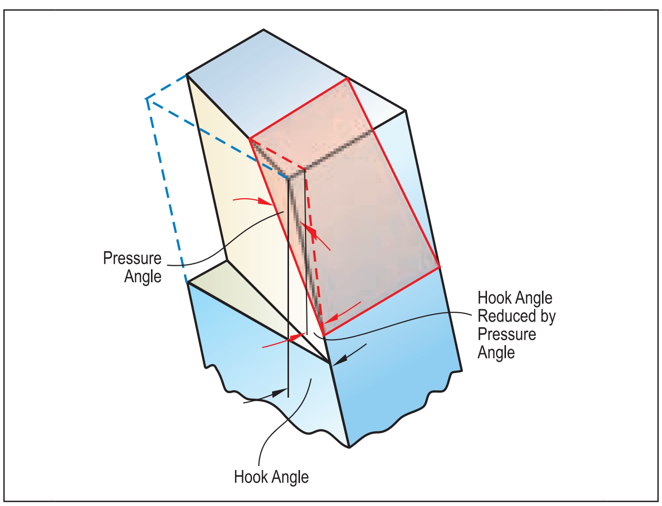

Figure 7—Hook angle reduction due to side rake and pressure angle.

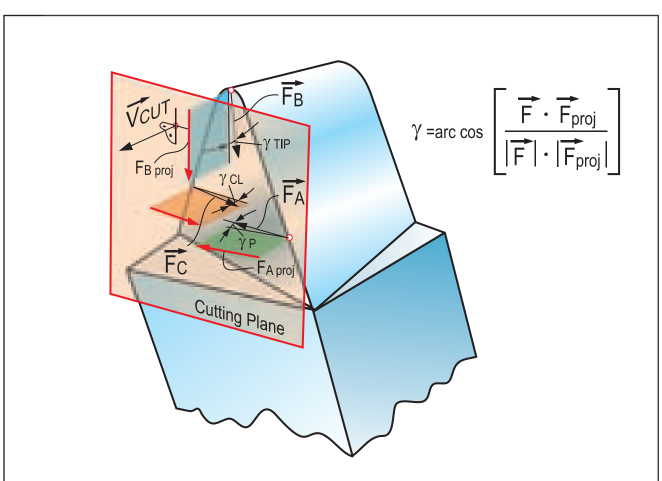

Figure 8—Calculation of effective normal rake angles.

optimal performing tools.

At first, existing tool life data of two-face-ground blades were used to analyze issues such as why inside blades had a shorter tool life than outside blades; under which condition the largest blade pressure angle wear occurred; and in which cases the tip or the edge between tip and cutting edge showed a fast degradation. In order to understand the cutting mechanism better, several additional terms were added to the three-face blade grinding summaries.

The duplication of two-face ground blades with three-face mathematics allowed analysis of a large amount of older data and led to a better understanding of the wear mechanism, which allowed the establishment of a set of recommendations for good blade design.

Next to the blade geometry input parameter, the new summaries show the following effective kinematic values of:

- Side rake angle
- Cutting edge normal rake angle
- Top rake angle
- Cutting edge hook angle
- Wedge angle of cutting edge
- Effective blade point
- Effective cutter point radius
- Effective side relief angle of cutting edge
- Effective side relief angle of clearance side
- Effective pressure angle

Other geometry features for the optimal use of three-face geometry are the full use of the root fillet width and the proper relief from the clearance side edge radius into the clearance blade edge. The recommendation is to back away from the flank surface on the clearance side to avoid the cutting action with a negative rake angle γ.

Blades designed with maximal blade point that avoids negative angles for clearance side cutting (as mentioned above) are called F-Point design. The advantage of F-Point blades is a smooth root fillet without steps and fins and a roughing action of the clearance side edge radius in order to reduce the wear of the edge radius of the following blade.

Blade Sharpness

The blade sharpness is a new expression, which defines the effect of the rake angle to the chip forming and chip removal process. For pre-raked blades with permanent-coated front face, a side rake angle of 20º was used for high-speed-steel blades used in face milling and 12º was used in face hobbing. This side rake angle was defined relative to the blade shank as the angle between the actual front face and the front surface of a virtual block blade (Figure 6).

The side rake angle of a pre-raked blade and the blade shank inclination due to the slot tilt angle in the cutter head define the general sharpness of a blade. For two-face-ground blades, this is an adequate strategy, which allows the use of standard blade blanks as well as standard cutter heads across the board for all bevel gear cutting jobs.

In the transition from HSS wet cutting to carbide dry cutting, short of deeper experiences, the 12º side rake angle of TRI-AC® face hobbing was chosen for the pre-raked carbide blades. The experiences that have been gathered in the past several years have shown that, besides the fact that the 12º blade side rake angle was too large, a general definition of technological blade angles according to "one size fits all" does not result in optimal tool life and best cutting performance.

The slot inclination in the carbide, high-speed cutter heads was chosen so that the cutting edge inclination vs. the cutting plane would always have positive values, and in borderline cases be permitted to be zero. The interaction of pressure angle and side rake angle will always reduce the angle given by the slot tilt angle value as shown in Figure 7.

The aforementioned considerations lead to slot tilt angles for Pentac® Cutter Heads of 7.42º for face milling cutters and 4.42º for face hobbing cutters.

Process research of The Gleason Works has discovered that only the effective sharpness reflects the chip forming mechanics correctly and explains the phenomena of tool life differences between cutting jobs that seem similar, yet deliver dramatically different tool lives.

If the total chip width along the cutting edge and blade tip is observed in infinitesimally small increments, three fundamentally different areas can be identified. A, B and C in Figure 8 show those areas. Chip flow vectors, which lie in the plane of the blade front face, are displayed in Figure 8 and are perpendicular to the cutting edge tangent at the specific points.

Figure 9 shows how the chip flow vectors such as FA are calculated as cross products of the front face normal vector VF and the local cutting edge tangent vectors like TA. The angle between the chip flow vector and the cutting plane shown in Figure 8 is the true rake angle relevant to the cutting action.

Chip Wrinkle Dynamics

There are three fundamentally different chip flow and wrinkle mechanisms depending on one of the following blade types (Ref. 4):

- Side blades, IB/OB-Blades
- Full profile blades
- Dual cut, double side and bottom blades

Side blades produce an L-shaped chip. As the chip is formed by the shearing action and flows away from the cutting edge, a plastic deformation wrinkles the chip (Figure 10, center). In case of a positive cutting edge hook angle, chips are less wrinkled and flow without obvious disturbances. The side chip part is pulled away from the top, which reduces the load on the edge radius and, in turn, reduces edge radius wear, but also eliminates catastrophic failure like the so called "banana peel" as it is shown in the photograph in Figure 11.

Highly positive cutting edge hook angles can lead to a separation of top and side chip or cracks between them. This is a positive sign of a desired load reduction in the major wear zone of the blades.

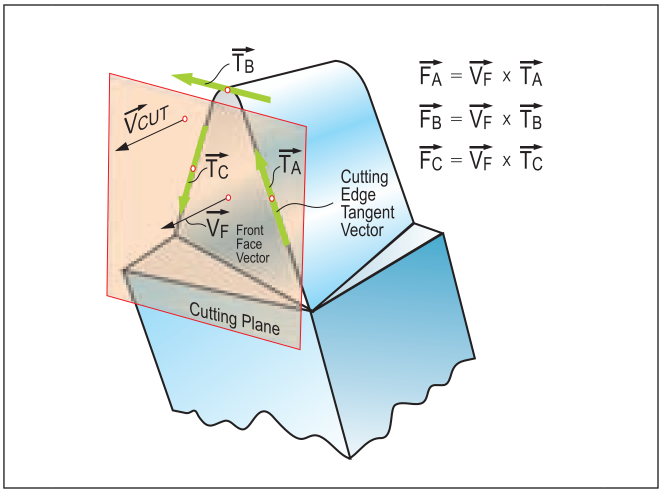

Figure 9—Cutting edge tangent vectors, cutting plane and cutting velocity.

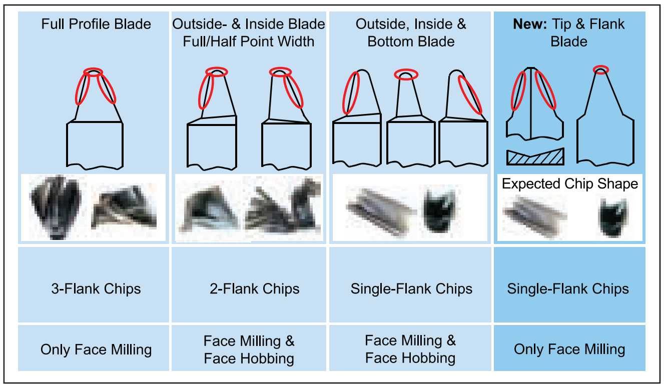

Figure 10—Left side: full profile chips; center: side blade chips; right: dual cut chips.

Figure 11—Large blade top chipping called "banana peel."

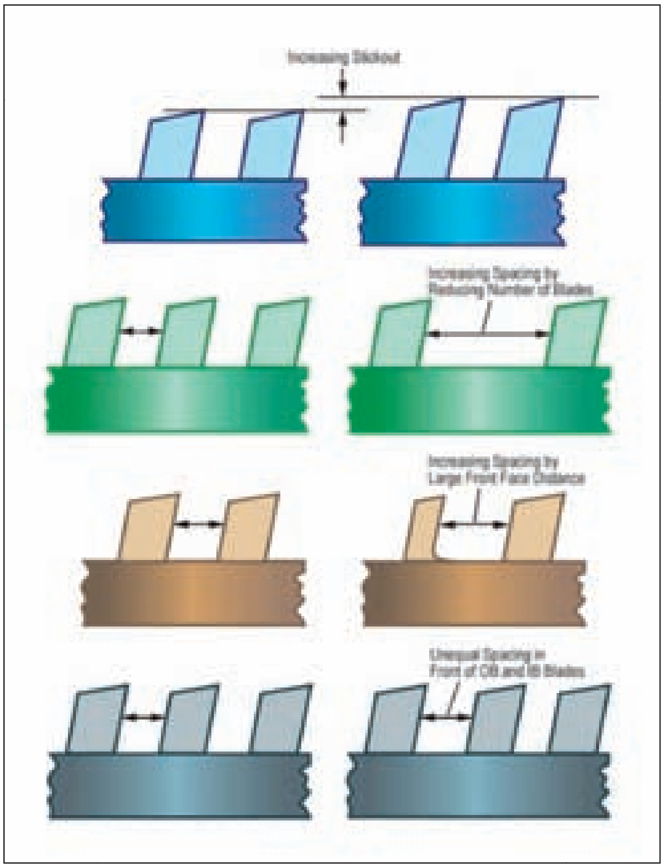

Figure 12—Different possibilities to increase the chip gap volume.

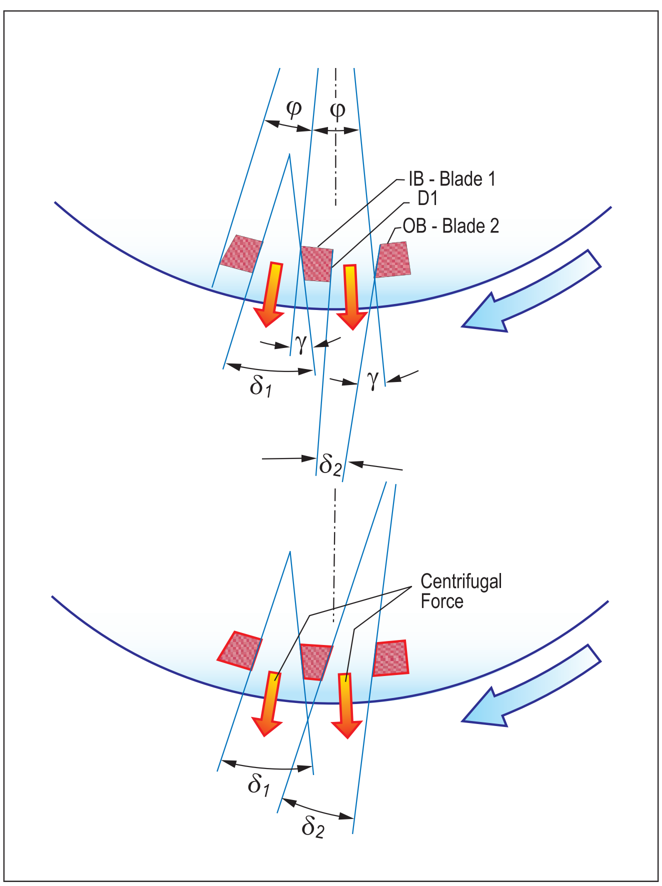

Figure 13—Regular chip gap (top), optimized chip flow by four-face grinding (bottom).

Full-profile blades are traditionally three-face ground. It is recommended to give the IB side 0.5º positive normal rake angle (which will result in a slightly negative value for the normal OB rake angle). Figure 10 shows typical full-profile blade chips. The top rake angle of full-profile blades should be maximized because this will change; e.g., a zero degree normal rake angle of the cutting edge to a positive value. The top rake angle is limited due to the cutter slot tilt angle. Figure 1 shows that the maximal possible top rake angle is equal to the slot inclination angle, which for three-face-ground blades needs to be at least 0.5 degrees smaller than the slot inclination angle because of grindability.

Dual cut blades allow the choice of independent rake angles between side and bottom blades. The side chips roll up without any wrinkling, independent from the hook angle (Figure 10). The bottom blades require the maximal possible top rake angle, similar to the full-profile blade system. Full-profile blades and dual cut blades would benefit from chip flutes, chip groves, chip breaker, etc., designed in order to provide both blade sides with a sharp cutting edge with a positive normal rake angle.

Chip Flow

Highly aggressive dry cutting cycles lead to large chips, which are often highly wrinkled. There is a potential that those chips will not always leave the gap between a blade and the blade in front of it. It has been observed that this phenomenon occurs only in front of outside blades. The chip gaps in front of outside blades are mostly tapered, getting smaller towards the outside (Figure 13).

The centrifugal force in high-speed cutting is up to 100 times the gravitation force g. If the first chip is caught in a chip gap, the following chips from the following cutter revolutions are not able to leave the gap toward the outside of their openings (keystone effect), but the accumulated chips compress to a hard pack, which fills out the entire gap. Chip packing leads to catastrophic failure, blade breakage or in the best case to a cycle abort due to an extensive power draw.

The change of the physical gap size by larger blade stick out, reduced number of blades, increased blade front distance or unequal IB/OB blade spacing lowers the blade stiffness or requires a special cutter head design, as shown in Figure 12. However, it has not been proven that the mentioned changes of the gap size eliminate chip packing, but negative side effects like chatter marks and consequently lower tool life have been recorded during cutting investigations.

Gleason discovered that the size of a chip gap is of less importance than the ability of the chips to flow toward the outside and leave the gap. Chip packing can be avoided if the gap is formed in order to enhance this flow by a gap opening in front of all outside blades. The flow of chips can be analyzed using the basic principles of fluid mechanics. A chip gap formed like the chamber between two blades of a turbo compressor will provide enhanced chip flow. In order to achieve such an effect, the back side of inside blades have been modified with a fourth ground surface (see bottom part of Figure 13).

The fourth blade surface is defined by three parameters, the back face distance DBF, the back side rake angle ε, and the back hook angle η. Figure 14 is a view onto the back side of an inside blade, which shows the three parameters for a sufficient definition of the orientation and location of the fourth blade surface. The blade in Figure 14 shows a "positive" value for the back hook angle η, which provides an additional gap opening away from the cutter head body.

The approach of a fourth face grinding of inside blades is a proprietary, patent pending technology of The Gleason Works utilized with Pentac® cutting tools.

Edge Treatment

The cutting edges of three-face-ground blades are treated differently than two-face-ground blades. The edge-rounding radius before all-around coating is recommended to be below one micron versus five to ten microns for the edge rounding of two-face-ground blades. In reality, a one-micron edge rounding is a rather academic value. The treatment before all coatings is a vapor blasting which is done as part of the cleaning process, prior to coating. Basically, the vapor blasting eliminates surface particles with low adhesion to the carbide grit. The cratering along the cutting edge is cleaned from loose particles during the vapor blasting where the following vapor deposition coating develops a thicker coat around all corners, which has the effect of a certain rounding and smoothing of the cutting edge roughness.

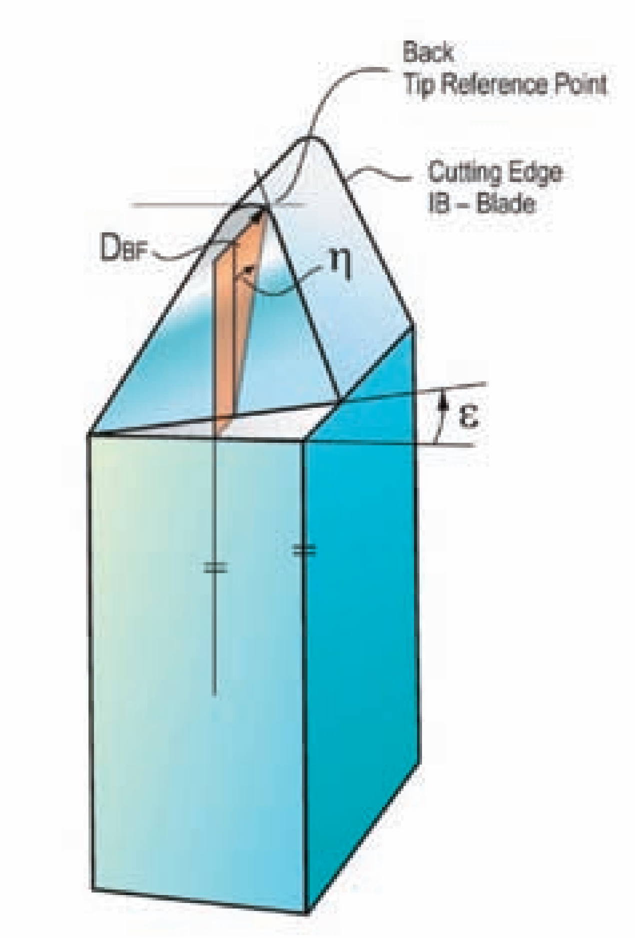

Figure 14—Parameters for back face definition.

		Blade Tilt Angle	Cutting Edge Hook Angle	Effective Side Rake Angle	Effective Top Rake Angle	Normal Rake Angle - Cutting Edge	Normal Top Rake Angle	Normal Rake Angle - Clearance	Wedge Angle - Cutting Edge	Wedge Angle - Top	Wedge Angle - Clearance Side
Car/Truck	OB	4.42°	1°	3°	4.4°	5°	3°	-4.5°	72°	63°	81°
Face Hobbing	IB	4.42°	1°	3°	4.4°	5°	3°	-4.5°	79°	63°	81°
Car/Truck	OB	7.42°	3°	5°	4.5°	5°	5°	-4.5°	82°	69°	79°
Face Milling	IB	7.42°	3°	5°	4.5°	5°	4°	-4.5°	76°	68°	81°

Figure 15—Three-face parameter table.

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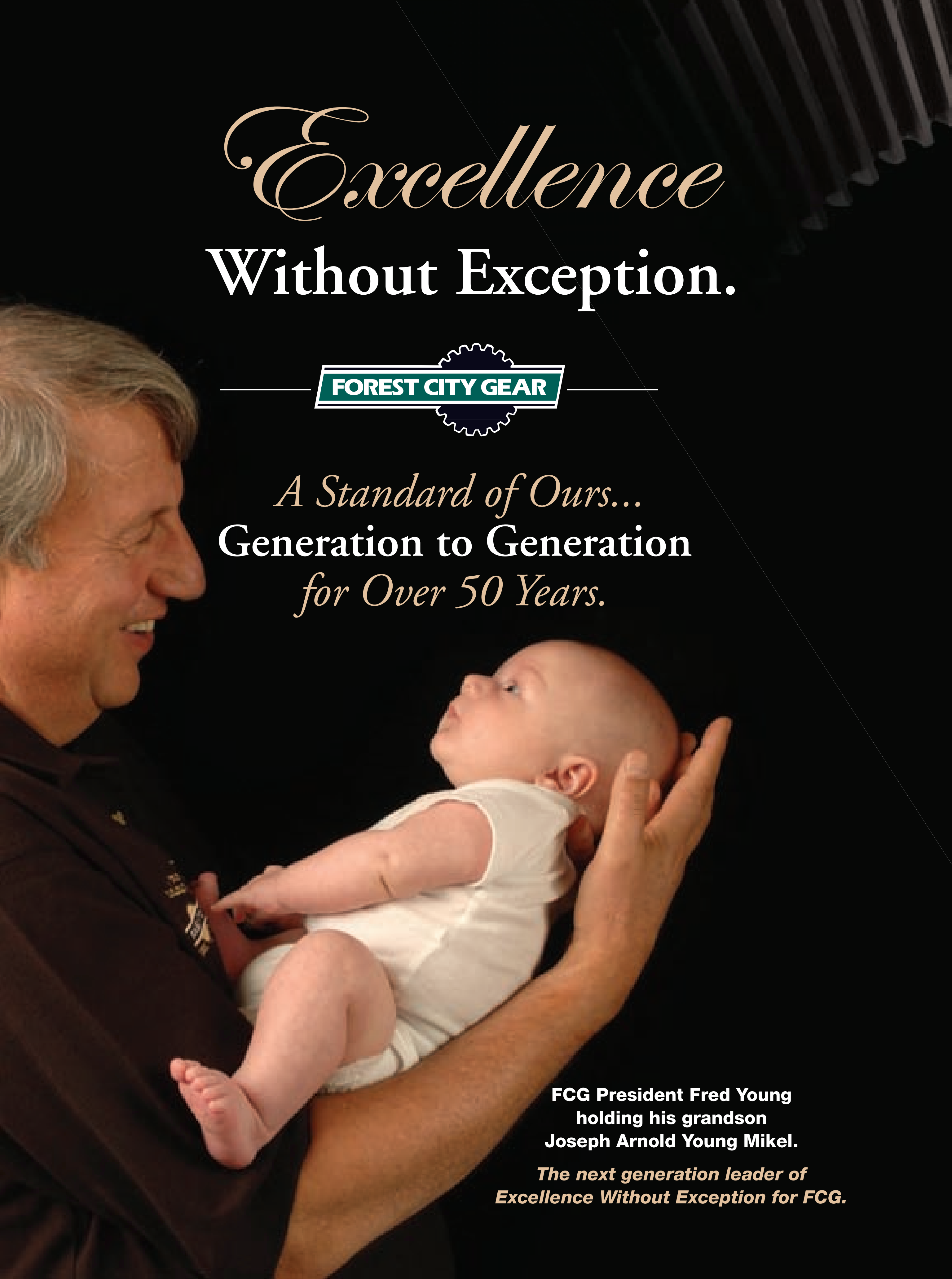
Hermann J. Stadtfeld *is vice president of bevel gear technology at Gleason Corp. He received a doctoral degree from Technical University of Aachen (WZL) in 1987 and taught bevel gear design at Rochester Institute of Technology and at Technical University Ilmenau, Germany. From 1991-2002, Stedtfeld was vice pesident of research and development at Gleason. He has authored more than 15 books and 130 papers on bevel gearing and has received more than 30 patents related to gearing.*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Tooth-Bending Effects in Plastic Spur Gears

Influence on load sharing, stresses and wear, studied by FEA

Dr. Ir H.G. H. van Melick

Management Summary

This paper describes the investigation of a steel-and-plastic gear transmission, using both numerical (FE) and analytical methods. The aim was to study the influence of the stiffness of the gear material on the bending of the gear teeth, and the consequences on contact path, load sharing, stresses and kinematics. As has been shown in previous literature, the tooth bending of plastic gear teeth results in an increase in the contact path length and in a considerable change in load sharing. The root stresses, in principle independent of the modulus, become quite different for plastic gears due to changes in load sharing. Also, on the contact stresses—which depend on the modulus—the preliminary and prolonged contact path have a significant influence, inducing very high stress peaks. These peaks, due to contact of the sharp tooth tips, result in pressure-velocity (PV) values, which are up to a factor of seven times higher than calculated using conventional theory.

Furthermore, the FEA results show that the kinematics of plastic gears change dramatically. In the extended part of the contact path, the tooth tip makes a reciprocating movement on the root of the mating gear. Our hypothesis now is that this high PV, reciprocating movement is the governing mechanism of wear in plastic gears. Experimental results in literature seem to substantiate this hypothesis.

Harold van Melick *is a senior research scientist at the Material Science Centre of DSM. He is also a member of the Global Segment Team Gears of DSM Engineering Plastics, a team leading the development of Stanyl in gear applications. He joined DSM after studying mechanical engineering at Eindhoven University of Technology, and receiving his PhD in 2002 on The Deformation and Failure of Polymer Glasses.*

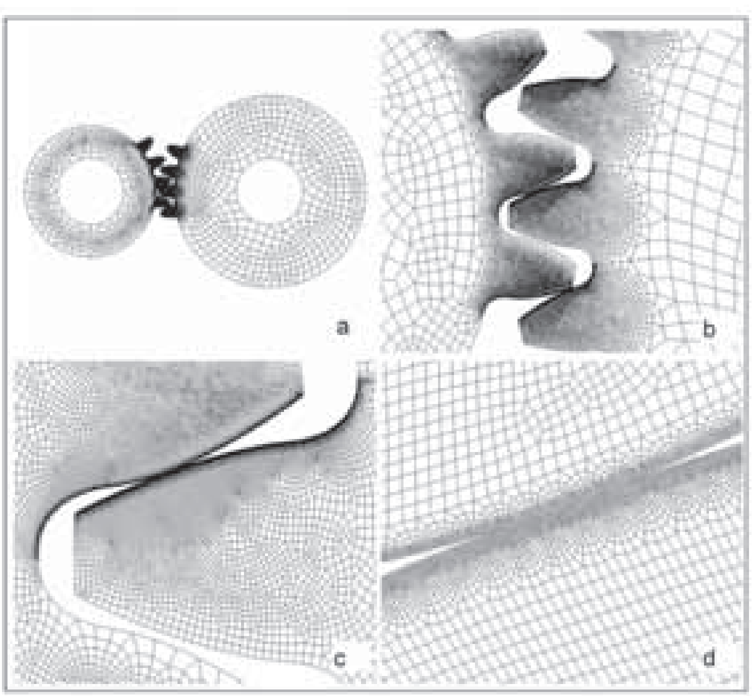

Figure 1—Details of the FE mesh. An overview of the complete model is shown in figure a, while zooming in on the refined mesh in the contact area from figures b to d.

Introduction

Plastic gears are now commonly used in the more demanding automotive applications, such as electronic power steering, electronic throttle control and starter motors. However, the fundamental knowledge of plastic gear design and engineering does not seem to have kept pace with the number of gear applications.

Much research was done in the 1970s and 1980s, especially in Germany, on predominantly polyacetal (POM) and polyamide (PA) gears (Refs. 1–8). These experimental investigations focused on measuring the load capabilities for various gears under varying conditions. These studies typically yielded curves for the normalized force versus the number of cycles to failure. In this approach, the module of the gear and other geometry factors are still important variables. Relatively little attention was paid to stress analysis and wear mechanisms and how these changed under varying conditions. The standardized calculations of ISO 6336 and DIN 3990 were used to calculate indicative stresses, but it was understood that a lot of discrepancies existed between "metal" theories and "plastic" practice. In the 1980s, a specific plastic gear standard was developed by VDI (VDI 2545), and although it was withdrawn again in 1996 for unknown reasons (Ref. 9), this standard is still commonly used.

In the 1980s, Yelle and Burns (Ref. 10) conducted substantial research to develop a more fundamental base to plastic gearing. Their approach succeeded in accounting for the tooth bending of plastic gears and in calculating real contact ratios for this type of transmission. With the ongoing development of finite element software packages and improved algorithms—accompanied by sufficient computing power in the 1990s—it became possible to solve complex contact problems like meshing (plastic) gears (Refs. 11–14). In the group of Walton at the University of Birmingham, the experimental research on plastic gears was combined with numerical work (Refs. 11 and 15–17). They confirmed the analytical findings of Yelle and Burns by FEA and showed that load sharing changes dramatically for plastic gears. Kapelevich and co-workers used FEA to modify tooth shapes and to optimize the tooth geometry specific for plastic gears (Ref. 14).

However, although substantial research has been conducted in the field of plastic gears, this does not seem to have resulted in generally accepted design rules. Today's plastic gear designs still seem to be based on empirics, experience and comparative calculations based on metal gear standards. The essential differences between metal and plastic gears—and how they affect kinematics, stresses and wear issues—are still rather undefined.

In building knowledge in the field of plastic gearing, many fundamental questions were encountered. For instance, what are the differences in kinematics between plastic gears and metal gears? To what extent are the standards developed in the 1970s for metal gears applicable for plastics?

To find answers to these questions, a program to compare the results of semi-analytical methods (e.g., ISO standards and *KissSoft*) and finite element analysis was begun, in addition to an extensive experimental program on plastic gears. In our approach, these calculations were used to study previously observed phenomena, like changes in contact path length and load sharing, but also to investigate stresses and kinematics. Finally, a hypothesis was formulated on the governing mechanism of wear in plastic gears.

Table 1—Details of gear geometries.

Parameter	Symbol	Pinion	Gear	Unit
Module	m	2	2	mm
Center Distance	a	53	53	mm
Pressure Angle	α	20°	20	°
Width	b	12	12	mm
Number of Teeth	z	22	31	-
Profile Shift	x	0	0	Mod
Addendum Height	haP	1.25	1.25	Mod
Dedendum Height	hfP	1.0	1.0	Mod
Root Radius Profile	ρfP	0.38	0.38	Mod

Table 2—Relevant material properties required for FEA.

Material	Young's Modulus [GPa]	Poisson's Ratio [-]
Steel	206	0.3
Stanyl GF30	10	0.4
Stanyl UF 23°C	3	0.4
Stanyl UF 140°C	0.7	0.45

Methods

This paper describes the results of Finite Element Analyses (FEA) performed in order to study the effect of modulus and tooth bending on the kinematics and stresses in a gear pair. The FEA were done using a commercial FE software package, *MSC.MARC*. The gears were modeled as two discs with four teeth each, under plane strain conditions (2-D analysis). Around 80,000 first-order quads were used, with a refined mesh near the contact surfaces, in order to capture accurately the contact stresses. Detailed pictures of the finite element mesh of the gears are shown in Figure 1.

Linear elastic deformation behavior was assumed for all materials. Although the unfilled materials do show significantly inelastic behavior, this is expected to have only a minor influence.

The calculations were done using the ISO 6336 standard (similar to DIN 3990) and *KissSoft* machine design program (semi-analytical).

In this study the focus was on a gear transmission, consisting of a steel pinion and gear, which was made of various materials. Both gears have a module of two, a width of 12 mm, and no profile shifts are applied. The tooth profile is standard according to DIN 867, with a pressure angle of 20°. Further details are given in Table 1.

In the FEA, different material properties were assigned to the gears. A steel gear was chosen, with the same properties as the pinion, a glass-fiber (GF)-reinforced grade of Stanyl (PA-46, 30 wt % glass fiber), and an unfilled Stanyl, tested at either room temperature or at 140°C. The relevant elastic properties are given in Table 2.

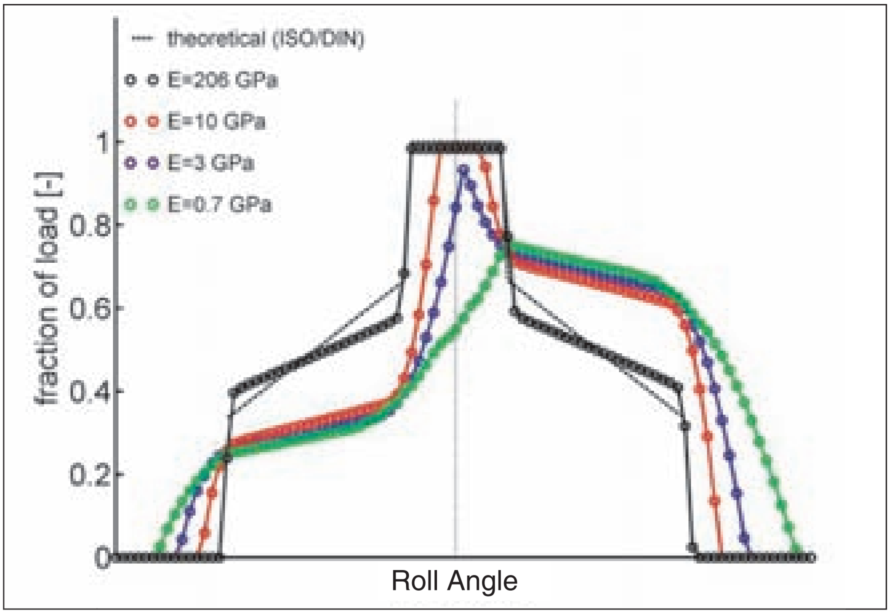

Figure 2—Load sharing between teeth of a steel pinion and a steel gear (black circles), a glass fiber (GF)-reinforced plastic gear (red circles), an unfilled plastic gear (blue circles) and an unfilled gear at elevated temperature (green circles). The dashed black line represents the theory (ISO 6336) and the dotted black line indicates the pitch point.

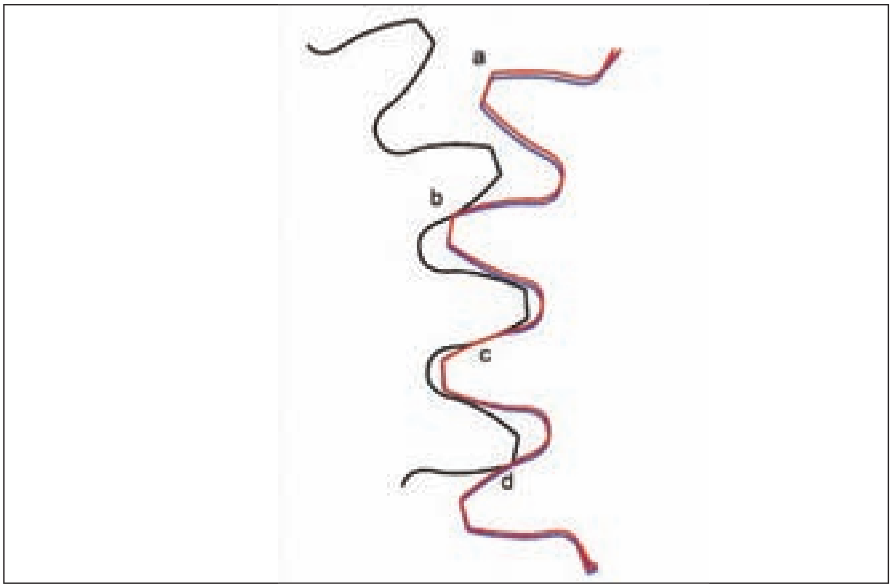

Figure 3—Tooth bending due to the combination of high load/low stiffness. The black lines represent the pinion, the blue line a steel gear, and the red line a plastic gear.

Results

Load Sharing. During meshing of a gear pair, the total load is shared between different teeth. In standard spur gear transmissions, a contact ratio between one and two is found, meaning that for a certain part of the meshing cycle, a single tooth carries the load; while for the remaining time of the meshing cycle, two teeth share the load. Following the classical theory—for instance the ISO 6336 standard (specific for metal gears)—the load is shared according to the 1/3-2/3-3/3 rule, meaning that the load fraction at the initial moment of contact equals 1/3, increases to 2/3 at the moment the preceding tooth is leaving the contact, and is carrying the full load during the period of single-tooth contact around the pitch point. After the pitch point, the load fraction follows the same scheme, but in reverse order. In Figure 2, which shows the load fraction as a function of the roll angle (rotational angle of the pinion), this is represented by the dashed (theoretical) line.

Using FEA, the load sharing can also be studied. Calculations and other studies in literature (Refs. 11 and 12) show that during meshing of two steel gears, load sharing is closer to a 2/5-3/5-5/5 rule, represented in Figure 2 by the black circles (E=206 GPa).

When a steel pinion meshes with a plastic gear, the load sharing changes dramatically. Figure 2 shows the load sharing for combinations of steel and various plastics, and the most striking change is that the load sharing becomes skewed when a steel pinion meshes with a plastic gear, resulting in a load share of approximately 1/3 in the first part of the meshing cycle and a load share of 2/3 in the last part.

Another significant effect is that the contact path increases, resulting in a preliminary contact earlier in the meshing cycle and a prolonged contact at the end, reducing the period of single-tooth contact. This has already been reported in literature in analytical (Ref. 10) and numerical studies (Refs. 11 and 12). For a glass fiber-reinforced plastic gear (E=10 GPa), represented by the red circles in Figure 2, the single-tooth contact period is approximately halved, compared to a steel-on-steel combination. For an unfilled plastic gear (E=3 GPa) and an unfilled plastic gear at elevated temperature (E=0.7 GPa), respectively, represented by the blue and green circles, single-tooth contact no longer occurs during meshing. With decreasing modulus, the maximum load share decreases to a plateau value of approximately 2/3. The origin of these effects lies in the fact that considerable tooth bending takes place, as is shown in Figure 3.

The black steel pinion meshes with the blue steel gear, as expected, while considerable tooth bending for the red plastic gear is observed. At position 'a,' the red gear has a significant lag compared to the blue gear. At position 'b,' a lag can still be seen in that the red gear comes into contact with the black pinion before the theoretical point of initial contact. For position 'c,' the teeth of the red and blue gear are virtually at the same position. However, this tooth is the most deformed, making the rest of the gear and teeth lag. At position 'd,' the load on the tooth reduces and the tooth bends back.

Tooth bending is not solely determined by the modulus of the gear, but also by a combination of stiffness (Young's modulus) and loading. As the ratio of Young's modulus and yield stress (i.e., loading limit) for polymers is roughly an order of magnitude lower than for steel, tooth bending plays a much more prominent role in plastic gearing.

Contact Path. Preliminary and prolonged contact takes place during meshing of plastic gears, resulting in an extended contact path. In theory, the contact path length is defined as the length along the contact line from the initial point of contact until the last point of contact divided by the base circle pitch.

As can be seen in Figure 4, the theoretical line of contact—typical for involute gearing—is well captured by two meshing steel gears (black circles). A contact ratio, ε_α, of 1.634, derived by FEA, also agrees well with theory (1.63). For a GF-reinforced plastic (red circles), the contact ratio increases to 1.84, and for an unfilled plastic, a contact ratio of two is just exceeded, which corresponds well with the fact that single-tooth contact was not seen in Figure 2. For an unfilled plastic gear at elevated temperature, the contact ratio goes to 2.37.

As can be seen in Figure 4, for plastic gears the contact path increases at the beginning and end. However, the extension of the contact path does not coincide with the straight contact line, but instead bends off. In fact, the extension of the contact path lies exactly on two circles—the tip circles of both gears.

Stresses. Table 3 shows the values for the root (bending) stress and the (Hertzian) contact stresses as calculated following ISO 6336 (or DIN 3990), VDI 2545, using *KissSoft* and *MSC.MARC* (maximum around pitch point). The values for steel gears calculated by ISO/DIN, *KissSoft* and FEA agree very well. The VDI standard gives much lower values for the root stress because the stress correction factor, which is taken into account by ISO/DIN and *KissSoft*, is ignored. For the contact stresses, the agreement between the standards is excellent, given that they are using virtually the same equations.

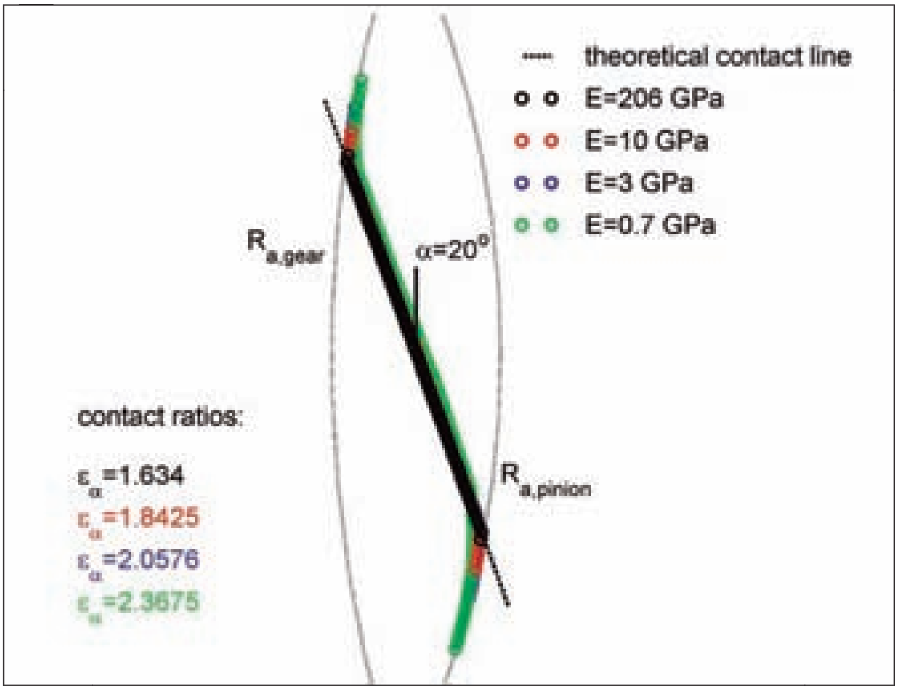

Figure 4—Contact path during meshing of a steel pinion with a gear of various materials.

Table 3—Maximum occurring root stresses and Hertzian contact stresses for a steel pinion meshing with a gear of various materials (Young's moduli).

Stress	Modulus	ISO 6336	VDI 2545	KissSoft	MSC.MARC
Root Stress	E=206 GPa	74.8	42	77.4	73.4
	E=10 GPa	74.8	42	77.4	70.7
	E=3 GPa	74.8	42	77.4	65.1
	E=0.7 GPa	74.8	42	77.4	51.7
Contact Stress	E=206 GPa	609	609	609	800
	E=10 GPa	193	193	193	230
	E=3 GPa	107	107	107	120
	E=0.7 GPa	54	54	54	50

Furthermore, Table 3 shows that both the root stresses and the Hertzian contact stresses decrease with decreasing Young's modulus of the mating gear. For the Hertzian contact stresses, this perfectly agrees with theory, as the contact stresses are dependent on the contact stiffness. For the root stresses, this is less evident, as the stress level in theory only depends on geometry and load. Nevertheless a decrease is seen for plastic gears, which is the subject of further investigation in the next sections.

Root Stresses. By using FEA, the root stresses of a single tooth can be monitored during meshing. Figure 5 shows the root stress as a function of the roll angle for a steel pinion meshing with gears of various materials. Similar regions can be recognized, as

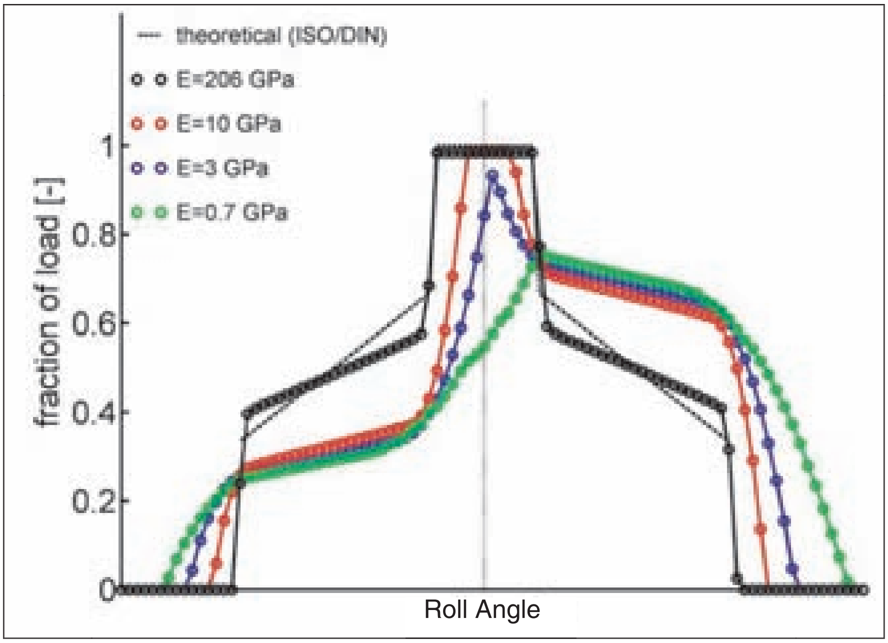

Figure 5—Root stress as a function of the roll angle for a steel pinion meshing with gears of various materials (moduli).

were seen in the load-sharing graph of Figure 2.

For a steel pinion and gear, the root stress is around 45 MPa in the first part of the meshing cycle. At the first moment of single-tooth contact, a jump to 74 MPa is seen. With increasing roll angle, the stress level lowers again as the contact point moves down along the tooth flank of the gear, reducing the lever arm. At the end of single-tooth contact, a large decrease is seen—to some 35 MPa—reducing even further towards the end of the meshing cycle.

As the load fraction carried by a single tooth determines the level of the root stress, the fact that the load sharing becomes skewed for plastic gears is expressed in the root stresses. The root stresses are considerably lower in the first stage of the meshing cycle compared to steel gears, and are considerably higher at the last part of the meshing cycle. Depending on the contact ratio and thus modulus, a period of single-tooth contact is achieved during which the root stresses become high. As can be seen in Figure 5, the level of root stresses reached in the single-tooth contact period follows the same trend as those found in steel gears. However, due to the changes in load sharing, the time span of single-tooth contact becomes shorter, and the high stress level of the initial moment of contact for a steel gear is not reached. For an unfilled plastic gear at elevated temperature, the peak stresses around the pitch have completely disappeared, and the maximum values tend to go to a value of some 2/3 of the theoretical maximum value. It is expected that a further decrease of the modulus will not result in a further decrease of the root stress level.

An implication of this lowered root stress (compared to standardized calculations) is that the gears can be loaded to higher torques than expected based on the comparison of the theoretical root stress and yield stress of the material. This does not mean that the standardized approach is fundamentally wrong, but rather that it should be modified by incorporating the altered load sharing due to tooth bending.

Contact Stresses. The contact stresses occurring during meshing of two gear teeth can be estimated using Hertzian theory. The contact stresses can be expressed as a function of the load, contact stiffness, reduced radius of the contact and the width of the contact, according to:

$$\sigma_{H_z} = \sqrt{\frac{FE_r}{2\pi b R_r}} \qquad (1)$$

where F is the force (load), E_r reduced modulus or contact stiffness (see Equation 2), b is the width of the gear and R_r is the reduced radius of the contact (see Equation 3).

$$E_r = \frac{1}{2\left(\dfrac{1-\upsilon_g^2}{E_g} + \dfrac{1-\upsilon_p^2}{E_p}\right)} \qquad (2)$$

where E_g and R_p are the Young's modulus, υ_g and υ_p are the Poisson's ratios of the gear and pinion, respectively.

$$R_r = \frac{1}{\dfrac{1}{R_g} + \dfrac{1}{R_p}} \qquad (3)$$

where R_g and R_p are the local radii of the tooth flanks of the gear and pinion, respectively.

The FEA results of the contact stresses as a function of the roll angle are represented in Figure 6. The contact stresses for a steel pinion and gear, represented by black circles, have a maximum in the region of single-tooth contact (around 800 MPa). This agrees with the values that were calculated semi-analytically. Furthermore, the local radii of curvature and the load fraction determined the course of the contact stress, as a function of roll angle.

According to Equations 1 and 2, the maximum contact stress—occurring around the pitch point—will reduce markedly with decreasing modulus. For a GF-reinforced plastic gear, the maximum contact stress around the pitch point reduces from 800 MPa to 250 MPa. For an unfilled plastic gear, this reduces to 120 MPa, and at elevated temperature to 50 MPa. The values found using FEA correlate very well with the values using standardized calculations or *KissSoft* (Table 4).

However, as can be clearly seen in Figure 6, very high peaks appear at the beginning and end of the meshing cycle. Due to the tooth bending and the increased contact path, preliminary and prolonged contact occurs for plastic gears. At preliminary contact, the tip of the gear meshes with the root of the pinion, while at the prolonged contact, the tip of the steel pinion meshes with the root of the gear. This implies that it is not the involute parts of the flanks that are meshing, but that the tip of the tooth, which is determined by tip radius (tip radius equals 0.1 mm in the calculations), is in contact with the other tooth flank. For a GF-reinforced plastic gear (red circles), the contact of the sharp tip radius of the steel pinion and the tooth root of the gear (end of meshing cycle) results in a contact stress of 940 MPa, which exceeds the values found for the steel pinion and gear. For an unfilled plastic gear, the value of the contact stress reaches 500 MPa at room temperature and 220 MPa at elevated temperature.

Using Equation 1 and taking the load sharing into account, the contact stress at the prolonged contact can be estimated using Equation 4.

$$\sigma_{H_z} = \sqrt{\frac{\alpha_{ls} F E_r}{2 b R_{tip}}} \qquad (4)$$

where α_{ls} is the load sharing factor and R_{tip} is the tip radius. Figure 2 shows that the load share carried by a single tooth at the moment the maximum contact stress is observed is around 50%, and hence α_{ls} is chosen to be equal to 0.5. The maximum values found by FEA and those obtained using Equation 4 compare very well (Table 4, Column 3). This equation can be used to estimate contact stresses, based on geometry and loading, without going through the elaborate process of FEA.

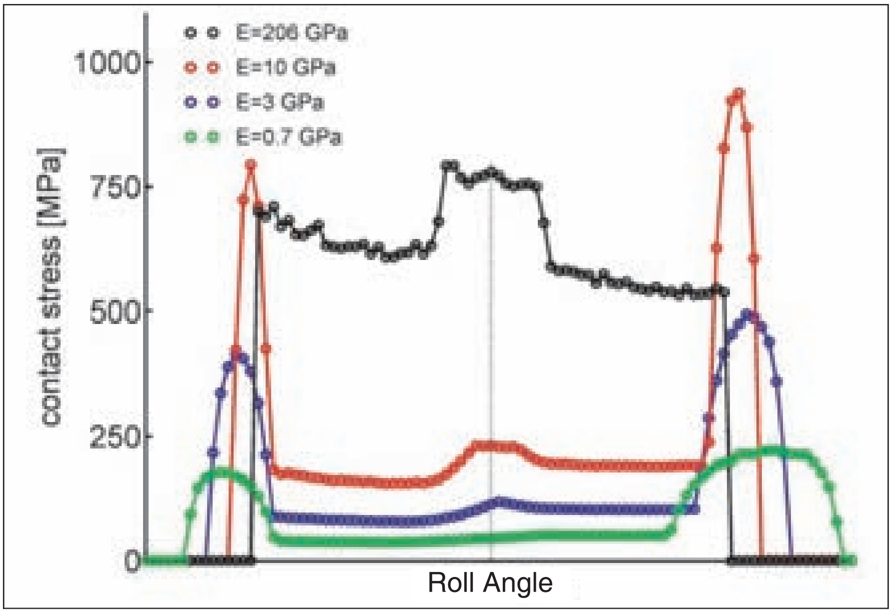

Figure 6—(Hertzian) contact stress as a function of the roll angle for a steel pinion meshing with a gear of various materials.

Table 4—Contact stresses determined by various methods.

Modulus	KissSoft/Analytical	FEA, Maximum
GPa	10^5 MPa.mm/s	
206	8.7	8.7
10	2.1	11.8
3	1.1	6.6
0.7	0.55	3.5

Pressure-Velocity Values. The large difference in theoretical contact stress and maximum contact stress due to the extended contact path has significant implications. In the conventional approach, the maximum contact stress occurs around the pitch point, at which the sliding velocity is low or even zero. The maximum pressure-velocity (PV) value, crucial in wear predictions, was found at the beginning and end of the meshing cycle. However, since the maximum value now is found at the beginning and end of the meshing cycle, combined with a larger sliding velocity (up to 20%), PV values are found which are much higher than calculated based on conventional theory.

Figure 7 shows the PVvalues for a steel pinion meshing with a gear of various materials. For a GF-reinforced gear, the PV value is expected to drop by a factor of 4; instead, a value is found which exceeds the PV value of a steel-on-steel combination by 30%. For the unfilled materials, the PV values are about a factor of seven times higher than those calculated by conventional theory. The PV values calculated by conventional theory and FEA are summarized in Table 5.

It must be noted that the calculations are performed using a tip-rounding radius of 0.1 mm. If a larger tip radius is used, contact stresses will drop according to Equation 3. In a steel-plastic combination, the peak in contact stress and PV value at the beginning of the meshing cycle (left side of picture, negative roll angles) is less important than the peaks at the end. At the beginning of the meshing cycle, the tip of the plastic gear is in contact with the root of the steel gear. After running in, the tip of the plastic gear will be worn in such a way that its contact radius has become larger and the contact stresses have reduced. Apart from a lowered contact ratio, this will have no significant negative side effects.

Implications of Tooth Bending: Wear. It has been shown that due to tooth bending, many secondary effects occur in plastic gearing. Apart from an increasing contact and lowering of the root stresses, the effects on the kinematics and contact stresses are negative. As shown, the PV values can

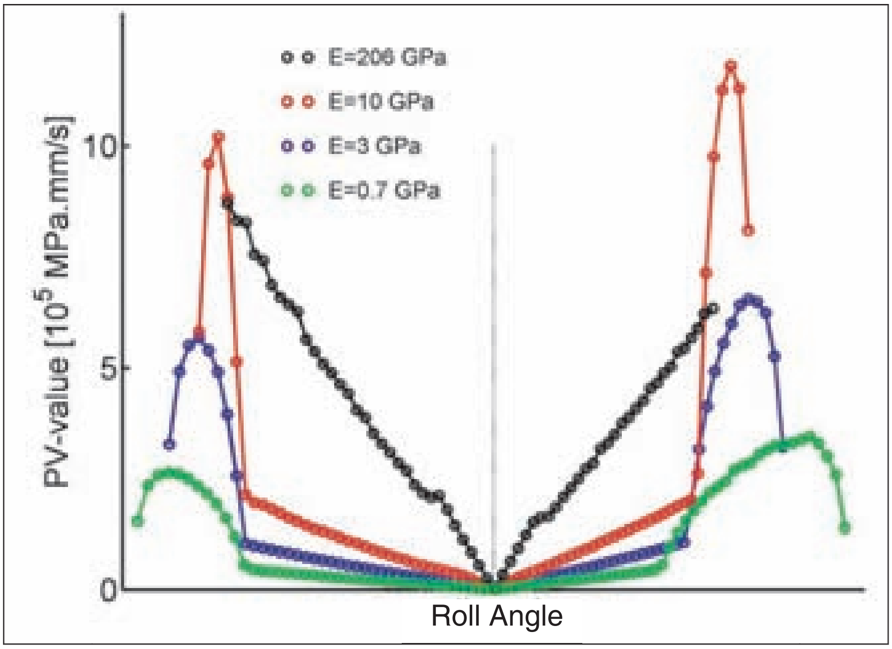

Figure 7—PV values found for a steel pinion meshing with gears of various materials.

Table 5—PV values determined analytically and by FEA.

Modulus	KissSoft/analytical	FEA, Around Pitch Point	FEA, Maximum	Analytical Maximum Equation 4
GPa	MPa			
206	686	800	N/A	N/A
10	217	230	940	983
3	121	120	495	548
0.7	60	50	220	273

become much higher due to the extended contact path, which puts the bad correlation between tribological tests (e.g., pin-on-disk measurements) and experimental wear data on gears into a completely different context.

A closer look at Figure 4 reveals that the extension of the contact path is perpendicular to the contact line and the rest of the contact path. Combined with the bending of the teeth, the question arises whether the involute flanks are still meshing properly. Therefore the kinematics of the meshing teeth were investigated in more detail. Figure 8 shows four subsequent images of the gear flank on four stages during the meshing cycle.

Figure 8a shows that, at the initial contact stage, the tip of the gear flank is in contact with the root of the pinion. Although not visible in this picture, the tip (and thus tip rounding) remain in contact for quite awhile, meaning that the rolling component of the velocity is very low during this stage. Therefore this stage can be regarded as if a plastic pin (or cylinder) is sliding on a steel surface.

With increasing roll angle, the contact point slides along the flank, as is expected based on theory. The involute flanks mesh properly and the contact point moves with fluid motion from the tip to the root of the gear (Fig. 8b).

Figure 8c visualizes the last moment of the meshing cycle when the contact point has moved along the contact line and reached the root of the gear. The contact point then enters the stage of prolonged contact.

Figure 8d reveals what is actually happening in the extension of the contact path—the contact point is moving back along the flank towards the pitch point. Hence in the prolonged—and also preliminary—contact, a reciprocating movement takes place at the root of the gear with a high sliding velocity, and very high contact pressure (and high PV).

In the case of a steel pinion meshing with a plastic gear, it is as if a sharp, cylindrical, steel tip (radius of curvature equals the tip-rounding radius) is digging into the root of the gear. Depending on the length of the contact path (and thus tooth bending and Young's modulus), the end of contact approaches the pitch point.

Our hypothesis now is that this high-PV, reciprocating movement at the beginning and end of the contact is the governing mechanism of wear in plastic gears.

To substantiate this hypothesis, the results of experimental studies were consulted. The first results investigated are experimental studies conducted by Walton and Weale at the University of Birmingham, in which they performed experiments on polyacetal (POM) gears to investigate their wear behavior (Ref. 17). Although our simulations are done on steel-plastic combinations, the same phenomena will occur in plastic gear pairs.

Figure 9 shows a SEM-image of the wear scar on a polyacetal-driven gear, run at 1,000 rpm and 8 Nm, after 105 cycles. Four areas of interest can be recognized—(a) the tooth tip, (b) the addendum, (c) the pitch point, and (d) the dedendum.

At the tooth tip, indicated by ‘a’ in Figure 9, high wear is observed. This can be explained by the fact that the tip of the gear is coming into preliminary contact with the root of the pinion, experiencing a high PV (Fig. 7; i.e., negative roll angles, left-hand side).

The addendum shows a much smoother wear pattern, indicated by ‘b’ in Figure 9. In light of the simulations, in which the involute tooth flanks of the addendum are meshing properly (although still at a considerable PV), this is also in agreement.

In the region indicated by ‘d,’ the most severe wear occurs, and the surface shows surface pits and scoops. This is the region in which the high PV, reciprocating movement takes place, and indeed the most severe wear is expected in this area.

The last striking feature is the formation of a ridge near the pitch point. This can be explained by the fact that, due to the reciprocating movement, the tip of the pinion returns at a high PV towards the pitch point. The high contact pressure digs out and deforms material, and pushes it toward the pitch point. At the end of the meshing cycle the pressure reduces, leaving the deformed material on a ridge. The severe effects seen in Figure 9 are the result of many repeated cycles.

If the mechanism described above is the governing mechanism in the wear of plastic gears, it would imply that the wear scars on the pinion (driver gear) and driven gear must be different. The reason for this is that, for the driven gear, the contact point moves from the tip to the root and then moves back towards the pitch point, as shown in Figure 8. For the drive gear, the opposite is true; the tip of the gear contacts the pinion near the pitch point and moves towards the root of the pinion, pushing the material down. The direction of the movement changes, and the contact point moves along the contact

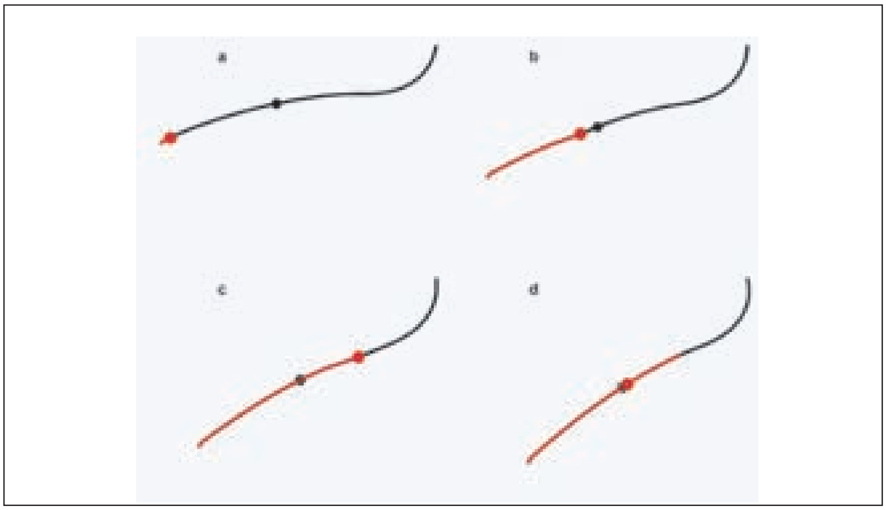

Figure 8—Kinematics of two meshing tooth flanks (only gear flank is drawn). The solid black line represents the tooth flank of the gear, the black dot represents the pitch point, the red dot represents the moving contact point and the solid red line represents the part of the flank that has already been in contact.

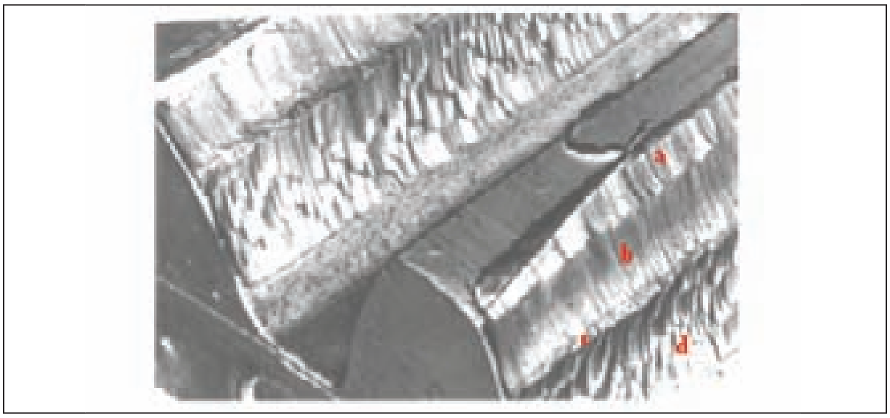

Figure 9—Scanning electron micrograph of a polyacetal-driven gear, run at 1,000 rpm and 8 Nm after 105 cycles (x16) (Ref. 3). Reproduced with permission from Breeds, et al. (Ref. 17).

line from the root to the tip of the pinion. The experimental study on polyacetal gears (Ref. 17) shows that the wear patterns at the dedendum of the pinion (drive gear) are very different and that a valley or depression without ridges is seen near the pitch point.

Breeds, et al., (Ref. 17) explained the observed differences by the fact that the surface stresses resulting from the friction forces are compressive for the driven gear, while the driver gear experiences tensile stresses. The differences in wear of the addendum and dedendum were explained by the fact that the directions of the rolling and sliding velocities are different above and below the pitch point. Thus the rolling component of the tip of the gears in the preliminary and prolonged contact is only of minor influence.

Further support for our hypothesis is found in an experimental study of

White, et al. (Ref. 18), in which they investigated the effect of tip relief on wear in plastic spur gears. Their general observation is that tip relief has very beneficial effects on the wear performance of plastic gears and that doubling the amount of required tip relief (based on calculations) gives the lowest wear rates. They state further that the shape of tip relief is important, as it determines the level of contact stress. These observations complement our simulations, as tip relief reduces the effects of preliminary and prolonged contact. Moreover, the level of contact stress can also be considerably lowered. For example, applying a tip rounding radius of 0.3 mm instead of 0.1 mm reduces the PV value by 30%.

Conclusions

It has been shown that the load sharing of a steel-plastic gear pair changes dramatically compared to the conventional theory of steel gears. Not only does the length of the contact path increase, so also does the shape of load sharing change due to significant tooth bending. For steel-to-steel and plastic-to-plastic gear pairs, the load sharing is symmetrical around the pitch, while for a steel-plastic gear pair it becomes skewed, and the teeth face the most severe loading in the last part of the meshing cycle (roughly 2/3 of the load). It was also shown that the extension of the contact path lies perpendicular to the theoretical contact line of involute gearing.

The changes in load sharing also change the stresses. Although the root (bending) stresses are only dependent on load and geometry, due to the changes in load sharing, the bending stresses decrease for plastic gears. Here a plateau value of roughly 2/3 of the theoretical root stress is predicted. With decreasing modulus the contact stresses around the pitch point become lower. But the extended contact path tip of one gear comes into contact with the other gear, resulting in very high contact stress peaks. As a result the PV value can be up to seven times higher than predicted in theory.

In the preliminary and prolonged contact, the involute tooth flanks do not mesh properly, but the tooth tips make a reciprocating movement on the root of the other tooth. We state that this high-PV, reciprocating movement of the tooth tip is the governing mechanism of wear in plastic gears. Experimental observations in the literature substantiate our hypothesis.

Acknowledgments

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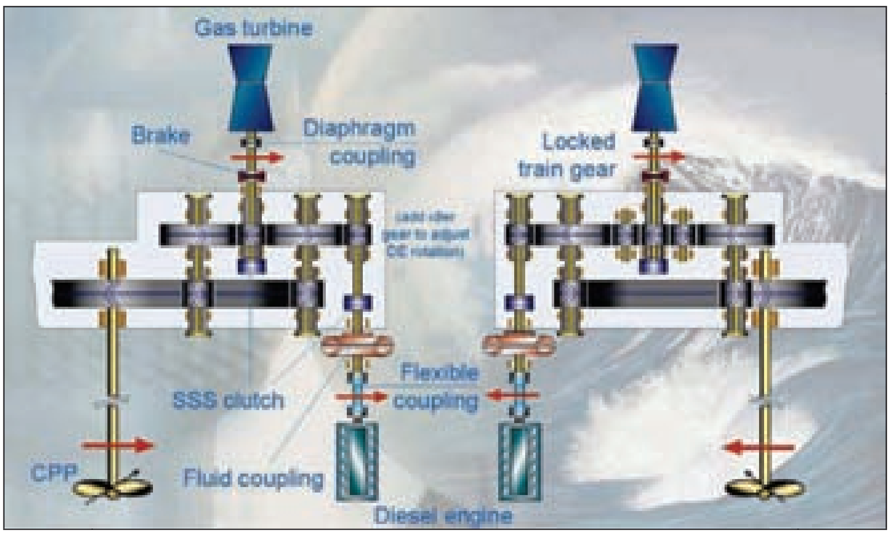
High Speed Gears for Extreme Applications in Industrial and Marine Fields

By Dr. Ing. Toni Weiss, Dr. Ing. Franz Hoppe



Management Summary

Today, advanced gear systems in both the marine and industrial fields are exposed to increasing demands. No matter whether stationary high-speed gears up to 140 MW, or COGAG and CODAG power transmission marine gears are addressed, a supplier of heavy-duty reduction gears has to provide—at the utmost interface—flexibility and the optimized technique for any kind of installation in close co-operation with power plant suppliers, or Navies and shipyards.

Optimization of load carrying capacity and lowest noise performance require ultimate refinement of toothed gears and bearings in continuous development of technology enhancement. In this respect, it is extremely important to select the appropriate macro geometry parameters and tooth correction values, supported by continuously adapted calculation methods. Experimental investigations as carried out by highly respected institutes lead to results transferred directly into the gear design.

Moreover, as theory needs to be transferred to real operable gears with the required accuracy applicable, heat treatment processing and grinding tools are to be continuously adjusted to the latest stage of technology.

Above all, a gear is not just a mechanical transmission, but is developed to a system fulfilling multiple demands, such as clutch integration, selectable output speeds, and controls of highest electronic standards. This paper shows the basics for high-speed gear design and a selection of numerous applications in detailed design and operational needs.

Introduction

Tooth corrections for high-power gears have to be evaluated and manufactured to compensate for all influences which disturb even-load transmission and even-load distribution during operation.

Load distribution in circumferential direction. Today's gear manufacturing technology for high-precision, high-power gears typically provides deviations in a range where an impact on load distribution in circumferential direction can merely be neglected, as remaining pitch deviations are much less than tooth deflections under load.

This, of course, also is an inevitable precondition for adequate tooth and tip relief and in turn even optimization of how tooth and tip relief are affected. The major difficulty of such an optimization is to choose the load for which the amount of correction is selected. The calculation of the necessary amount is relatively easy, as gear mesh stiffness values and calculation procedures are widely available.

When extreme loads are applied to gears designed for normal operation, there is a certain risk of hard contact, e.g., in the dedendum area near the tooth root where tooth contact with the mating gear tip starts. Similarly the influence of tooth and tip relief on gear scuffing has to be observed. A "soft start" of the gear coming in mesh of course also influences gear noise, which is discussed later.

Load Distribution Across Face Width

Elastic deflections. Based on the assumption that manufacturing deviations on even relatively large gears can be minimized to very high quality, especially for any actual gear pair in mesh, elastic deviations will be the predominant influence factors on load distribution, which have to be taken into consideration. Again different load conditions will result in different deflections.

From a load carrying point of view, in most cases the lowest maximum stress is obtained if the deflection at maximum load is considered for tooth geometry corrections. It may be advisable, however, to study these influences for different loads.

As the number of influence factors—such as bending and torsion of pinion and wheel, bearing and housing deflection—will increase, the more accurate a calculation must be. It becomes relatively difficult to evaluate these influences in their common effect, and it is widely accepted that only proven computer programs like RIKOR in its latest edition are a suitable tool for the designer to find the best values for tooth correction.

In particular, helical gears are very sensitive when the tooth ends come into mesh. The effect of typical end relief, which has been state-of-the-art for a long time, can be analyzed in more detail by such a computer program and it is even possible to manufacture adequate three-dimensional corrections.

In any gear calculation procedure, according to valid standards, the load distribution factors are described in more or less detail. In reality, however, the influences are not simply meant to be superposed, but rather observed at any location of the gear mesh field and at various loads. The result can be shown and analyzed in a three-dimensional graph.

A very simple example for a helical gear and a certain load is shown in Figure 1. Whereas this distribution is easy to understand and most likely reflects the conventional expectations, Figure 2 shows a helical gear with a very uneven load distribution and its effect. It can clearly be seen that due to the high local load, the correct tip relief cannot avoid high local stresses, even higher than represented by the load distribution factor.

Non Load-Related Deviations

Deviations affecting load distribution due to manufacturing, assembly and alignment are typically measured and to a large extent eliminated for high-precision, high-power gears. They are not discussed further here.

High-power gears, however, are normally running at high circumferential speeds. To minimize inertia or

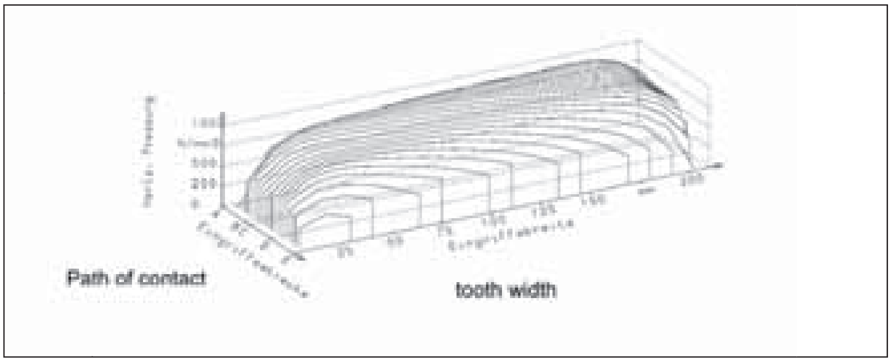


Figure 1—Hertzian pressure distribution for a helical gear with optimized tip and root relief as well as end relief.

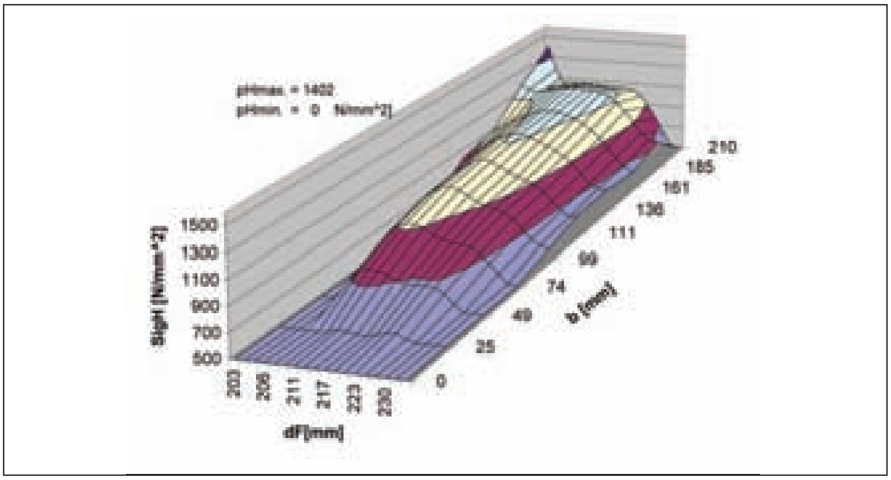


Figure 2—Hertzian pressure distribution for a helical gear with optimized tip and root relief as well as end relief, but assuming a very uneven load distribution ($K_{H\beta}$ > 2) for study purposes.

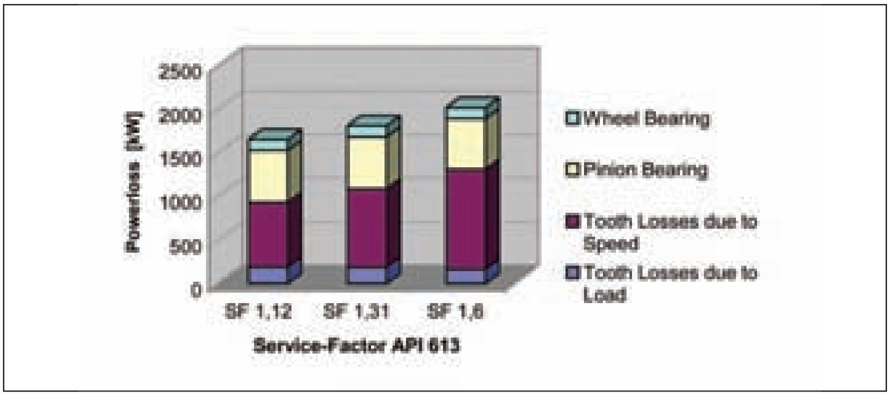


Figure 3—Power loss for a high-power gear calculated for different theoretical safety requirements, i.e., different API Service Factors.

overall gear weight, it is sometimes necessary to reduce the mass of the gear body. Thus the design will be similar to a gear rim and a web. In this case, centrifugal forces can deform the gear rim in addition to the variation in stiffness.

Another effect is gear bulk temperature, which obviously is of great importance for large gear sizes and high circumferential speeds, generating high power loss, because there is always a certain temperature gradient. Therefore due to the negative effect of high temperature on scuffing risk, it is important to reduce gear size to a reasonable minimum. This is not reducing the actual gear safety when all influence factors are sufficiently considered. This must be taken into consideration when choosing appropriate safety factors for comparatively simple calculation procedures. Figure 3 gives an example of gear size and power loss, based on different API service factors.

The higher the temperature and the size, the greater the effect on the gear diameter and length of pinion and wheel will have to be considered. Such effects of temperature on the load distribution can be minimized if the typical temperature distribution is known by a modification of the lead correction for a certain steady state condition or by minimizing the temperature distribution itself by optimized cooling. It goes without saying that this influence is reduced when the gear is optimized when running in vacuum-like conditions as provided with etaX technology, as shown in Figure 4.

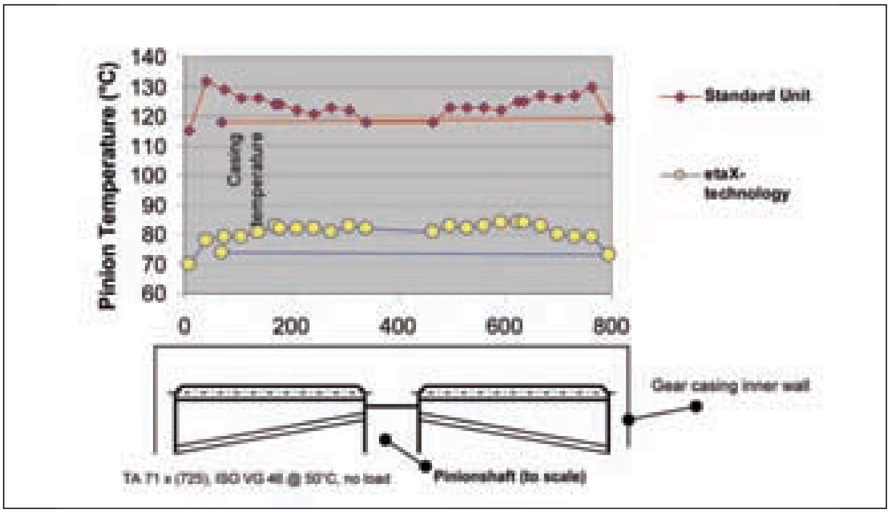


Figure 4—Oil temperature distribution of a 100 MW class gear measured after the mesh for an already optimized standard unit and with etaX technology.

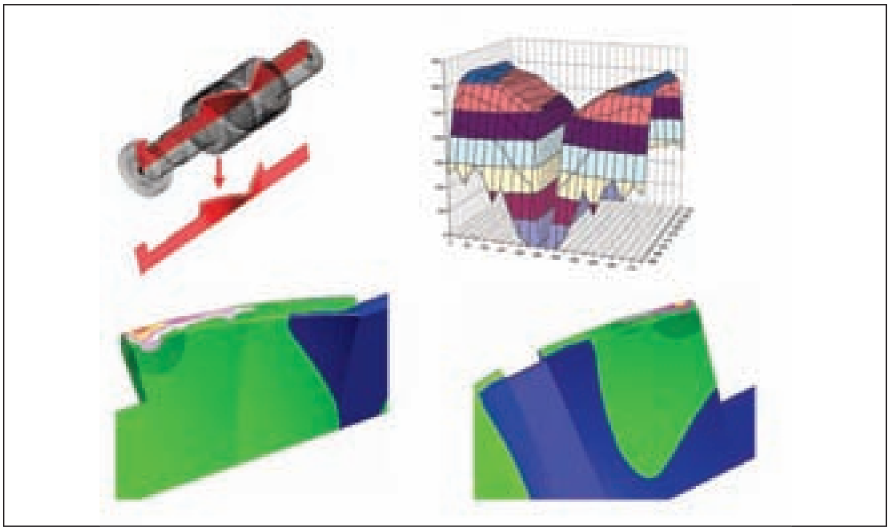


Figure 5—Temperature distribution of pinion and wheel, calculated with an FEM model and its effect on contact stress distribution (Ref. 7).

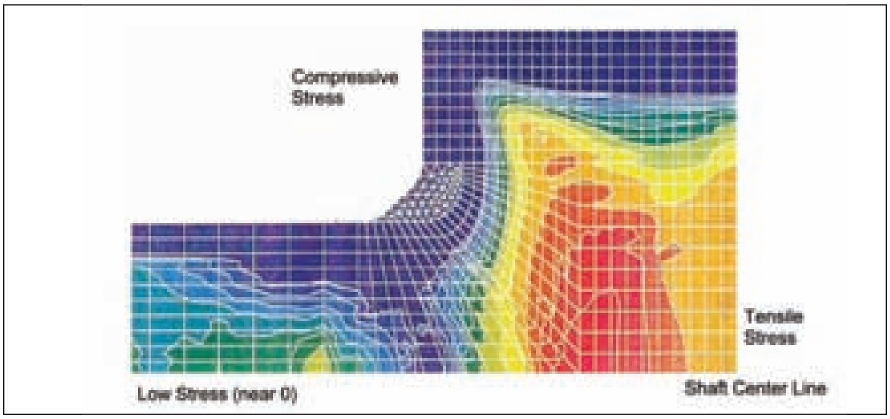


Figure 6—Calculated residual stress distribution for a (half) double helical model gear.

The influence of such an effect can only be calculated using an extensive FEM calculation which takes into consideration all heating and cooling sources and heat dissipation. Several boundary conditions have to be estimated, but when different measured temperature distributions can be realized with good correlation, it can be assumed that the calculation provides realistic results. Figure 5 shows an FEM model of a large double-helical pinion and the temperature distribution with the highest value at the outer end of the helix as well as the corresponding temperature distribution for the mating bull gear. The theoretical influence on load distribution of this temperature distribution then is calculated as an example neglecting all other influences like elastic deformation, etc. It can be seen that there may be significant influence.

As high-power gears for 100 MW and more must have hardened gears to limit the size, the gears and their integrated shaft still have enormous dimensions, especially for case hardening, with the quenching process of a very large solid part. The material for such parts has to be carefully selected and the process of manufacturing the gear blank needs to fulfill restrictive quality requirements. This is especially true because of unavoidable residual stress in the gear body after quenching. The result of such an evaluation is shown in Figure 6, which can be taken as a principle distribution, whereas details are still questionable as to whether boundary conditions and some material behavior have an important effect, but are only known to a limited extent. Stress-relieving heat treatment after hardening is limited in order to not negatively influence the tooth hardness. Various calculations and measurements have proven such residual stress distribution. Stress relief in operation after a long period of time—due to a high number of stress cycles (e.g., start ups)—has been experienced to a certain extent.

This stress relief obviously depends on its original value and on the gear dimensions. It will therefore be different for pinion and gear. In any case, it results in some permanent deformation of the gear body. It therefore is essential to evaluate the expected influence of this deformation on the tooth load distribution and to find an optimum compromise of its effect on gear life.

Optimized Geometry of Low-Noise Gears

Source of Noise. A main power transmission gear is subjected to various external influences, such as reaction loads from adjoined external couplings, foundation distortion, or dynamic mass forces caused by heavy sea states, and, not lastly, heat expansion due to the power loss generated by gear teeth and bearings.

In light of all these impacts and with respect to low noise signature, the tooth design has to be specifically observed, as pinions and gears represent "the heart" of a gearbox. The decision on the basic type of gear teeth is important, where principally spur gears, single-helical or double-helical gears are available.

Figure 7 shows the principle coherence between tooth mesh noise excitation and overlap ratio, ε_β. With spur gears, ε_β equals zero, with low single helices, ε_β values up to 2 are achievable. High helices are in a practical sense realized only with double-helical gears, achieving $\varepsilon_\beta > 3$. Apart from significant noise reduction at increased $\varepsilon\beta$, excitation appears minimal with integer value of overlap ratio. The fundamental results as depicted in Figure 7 are still considered as state-of-the-art and have been confirmed throughout the past 25 years with numerous research programs, supported by experience with countless applications in service.

Involute gears theoretically mesh without periodical angular deviation in rotation and without dynamic excitation when the gear teeth get into mesh. However, due to manufacturing tolerances, misalignment and elastic deformations under load, this theoretical optimum is not achieved in reality without use of appropriate design criteria.

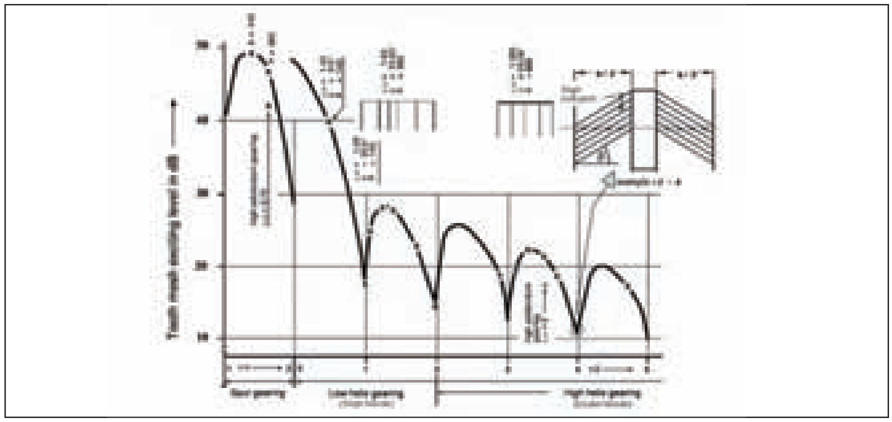


Figure 7—Noise in dB generated in tooth mesh, dependent on basic layout and transverse contact ratio $\varepsilon\beta$. Note minima achieved with integer ε_β values.

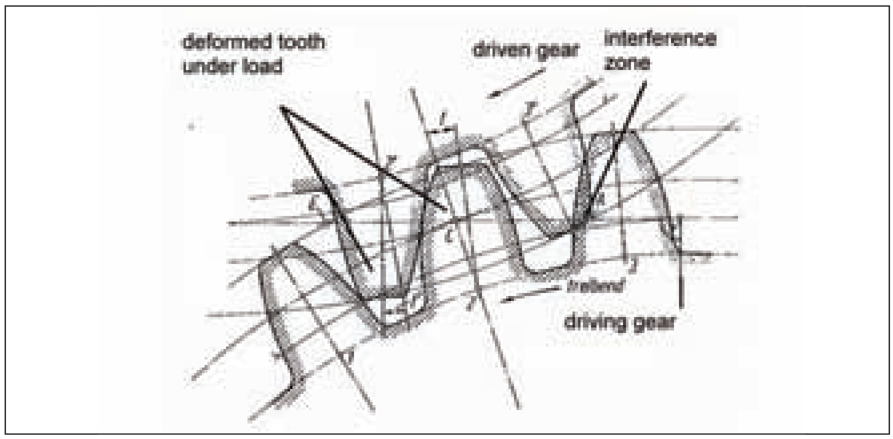


Figure 8—Tooth deflection under load and subsequent interface zone in following gear mesh (Ref. 6).

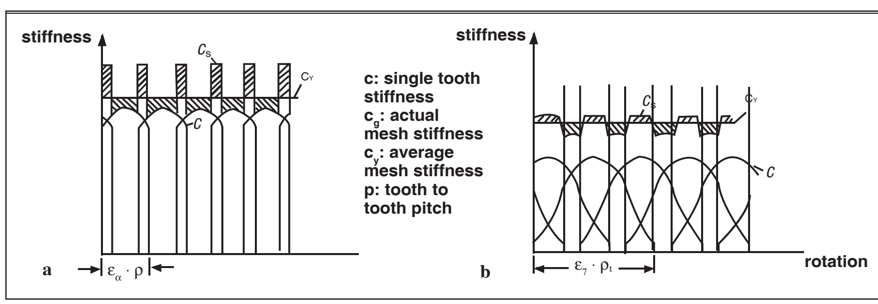


Figure 9—Mesh stiffness: a) Spur gear mesh; b) Helical gear mesh (Ref. 6).

Hence, these influences are to be compensated by adequate tooth flank corrections generated on high-precision grinding tools. Above all, the macro geometry still is the decisive criterion on noise excitation, as shown here.

The geometric ideal position is changed under load due to the Hertzian deflection on gear flanks, bending of the teeth, and elastic deflection of gear bulks and shafts. Even when considering deviation-free tooth flanks—as almost achievable with modern manufacturing methods—interference between the gear teeth in mesh occurs due to elastic deformation, causing periodical noise excitation as shown in Figure 8. This interference can be compensated by appropriate flank modification, achievable normally for one load level and one average stiffness value. Thus, gears requiring low noise emission in a wide power range, e.g. naval gears, need a minimized amplitude of the total mesh stiffness, c_γ.

Figure 9 shows the course of mesh stiffness over rotational angle for spur

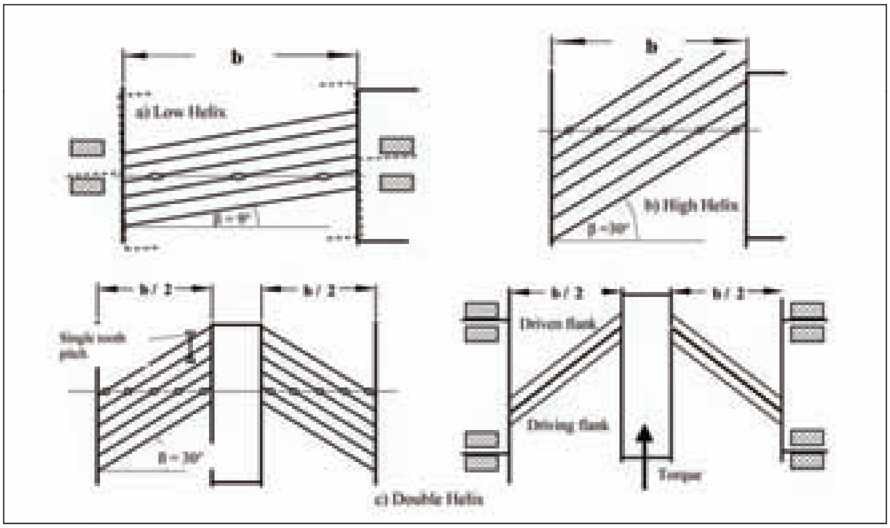


Figure 10a—Low helix, maximum overlap ratio ε_β =3; Dotted: position in bearings under load (overscaled). Figure 10b—High helix (Ref. 5). Figure10c—Double helix at overlap ratio ε_β = 5, here sole radial displacement in bearings at no shaft tilting, thus equal tooth contact over entire face width at all load conditions.

Table 1—Gear teeth base (macro) geometry of investigated gears, gear layouts, Figure 10, center distance 400 mm, ratio 3:1.			
Tooth Design	*a) Low Helix*	*c) Double Helix*	*b) High Helix*
Module m_n mm	6	4.5	6
Profile angle α	15°	15°	17.5°
Helix angle β	14.3°	31.7°	28.7°
Transverse contact ratio ε_α	2.0	2.0	2.0
Overlap ratio ε_β	3.0	5.0	5.0

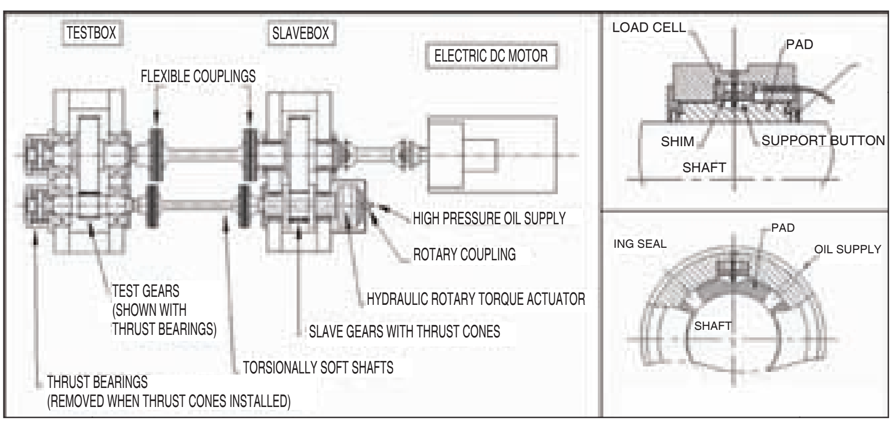


Figure 11—Test rig arrangement (left) and bearing load cell installation (right) of marine noise test rig (Ref. 5).

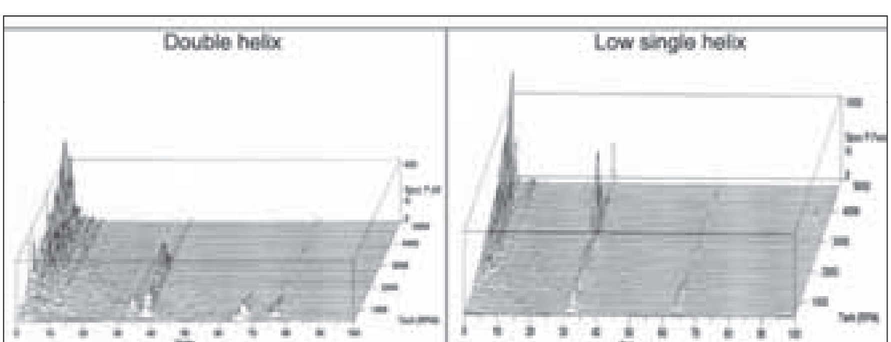


Figure 12—Typical waterfall plots showing significant frequencies in order and peaks with low helical and double helical gears. Gear mesh frequency at 38th order with double helical gears at 32nd order with single helical gears.

and helical gears. The mesh stiffness is the sum of the individual single tooth stiffness values for the teeth in contact shown on the bottom side of the diagram. Obviously, with an increasing number of teeth being in mesh at the same time, the amplitude of the periodical change in mesh stiffness decreases. The number of teeth in contact is determined by the transverse contact ratio, ε_α, (number of teeth in mesh along the path of contact; 1–2 for most gear applications) and the overlap ratio, ε_β (number of teeth in mesh along the face width; zero for spur gears, up to eight for double helical gears). The decisive total contact ratio, ε_γ, is the sum of ε_α and ε_β. Integer figures for ε_β further improve the mesh quality (see Figure 7).

Selection of Basic Geometry for Noise Attenuation. Coming to more distinct views on these principles, spur gears are not suitable for high-speed gears, according to their zero helix angle. As can be seen with Fig. 9a, the tooth load transferred from one tooth pair in contact to the next is followed with a sudden load jump, causing periodical force impacts and, subsequently, increased excited noise. Any attempt to decrease these impacts by profile correction grinding would just end up in some marginal excitation-reduced levels, with the remaining disadvantage of lack of contact ratio and high sensitivity for misalignment and concurring tooth edge overload.

Single helical gears, as shown in Figures 10a and 10b, comprise some benefits against spur gears. Normally, low helices apply in practice, where tooth load impacts are diminished by helix-angle-influenced transverse axial travel of the tooth mesh—a significant benefit against spur gears. But, with single helical gears, major disadvantageous aspects are to be considered:

- Limited helix angle $\beta < 10°$ is not to exceed axial loads, and still need of separate thrust bearings;
- Low transverse contact ratios, thus limited minimization of noise performance;
- Axial forces cause tilting shafts

in bearings, thus unpredictable non-contact areas and uneven load distribution across tooth face width;
- Axial forces cause bending of casing structures, to be compensated with added up structural weight.

In spite of these aspects, single helical gears could apply if measures such as reinforced casing structures, carefully selected tooth flank corrections, and an increased demand to foundation rigidity are realized. The high helix according to Figure 10b, would definitely show a much better noise significance compared to low helices, due to the high overlap ratio, but generates inadmissible high-axial loads not compensable in realistic views with moderately sized thrust bearings and reasonably designed casing structures (Ref. 5).

Finally, double helical gears, (Figure 10c), include the consequent resolution of the aspects above. They combine insensitivity regarding external forces to casing structures, followed by a high degree of load pattern consistency throughout the full power range—due to their symmetric design—with low noise performance for maximum achievable helix angles at optimized macro geometry of the gear teeth.

In summary, the following aspects support double helical gears:

- Maximum total contact ratio, ε_γ, for the benefit of smooth tooth engagement and lowest noise performance, as shown in Figure 7;
- Radial symmetric tooth forces, no axial impact to bearings, due to self centering effect, as shown in Figure 10c;
- Even contact pattern throughout all loading conditions;
- No axial loads to casing structures generated, thus most light weight designs achievable, by avoidance of excessive casing structural reinforcements other than ultimately required;
- Tooth corrections by grinding respecting just bending and torsional deflection of pinions, at no tilting load or casing deflection unpredictable impacts.

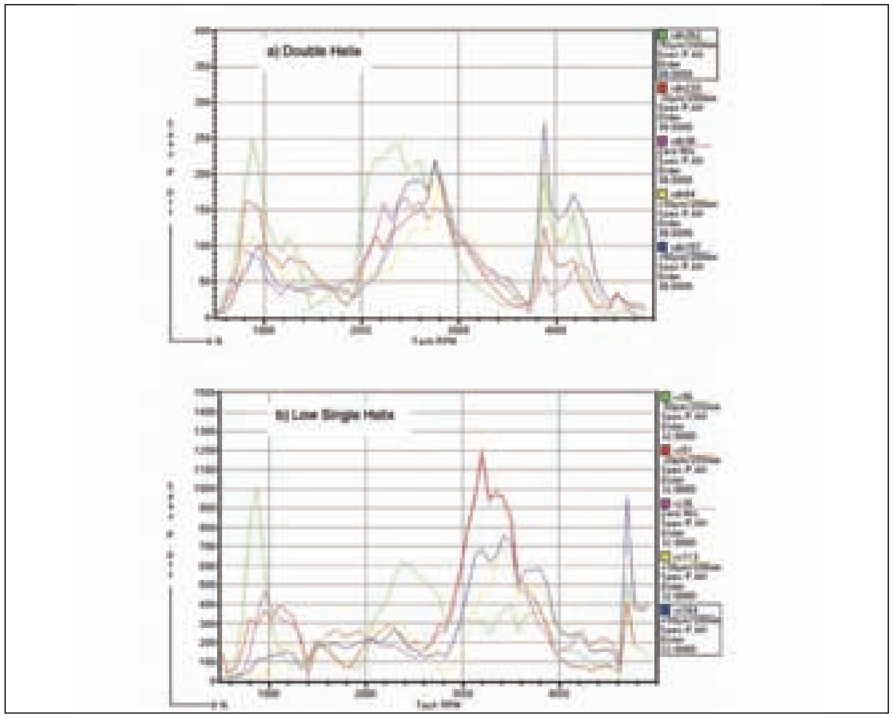


Figure 13—Sensitivity of double helicals (a) and single helicals (b) to variation of misalignment between pinion and gear, single curves as orthogonal slice through waterfall at gear mesh frequency.

Research on Gear Noise. For fundamental investigation purposes, a research program on dynamic bearing forces with different macro tooth geometries, manufactured to ISO quality level 2, was performed at the Design Unit Institute, Newcastle University, UK, supported with funding of the British Ministry of Defense (MoD). In effect, the tests were related to a comparison between the three basic gear designs as per Figure 10. The parameter combinations a) and c) as given with Table 1 were tested on a 8 MW back-to- back test rig and compared to previous results obtained with version b) under the same conditions (Ref. 5).

The test setup and measurement principles are given with Figure 11. In a back-to-back test rig, the slave gear and the testing gear are connected to each other by flexible couplings, the electric motor provides just enough power to overcome the losses of both gear boxes. The operating torque is introduced to gear circle by the hydraulic torque actuator variable up to 15,000 Nm. The test gearbox is supported by servo-pneumatic soft mounts for foundation isolation.

The dynamic excitation generated in the tooth mesh is measured as dynamic forces directly in the journal bearings of pinion and wheel of the testing gear. The bearing assemblies consist of four tilting pads with piezo-electric load cells mounted against the bearing shell, capable of discriminating dynamic forces down to 0.1 N per 100 kN. A special dynamometer at each bearing combines the load cell signals at high resolution and effective frequency response to waterfall results as shown with Figure 12.

Various parameters were subjected to investigation, such as torques up to 15,000 Nm, pinion speeds up to 5,000 rpm, and misalignments between pinion and gear ranging from –100–100 µm per 200 mm face width in distinct steps. For good comparison between the various gear designs, especially related to marine gearing applications, all tests were performed following the propeller law T ~ n 3.

Results of Noise Investigation. In comparison of two typical waterfall plots shown with Figure 12, some interesting basic results are being

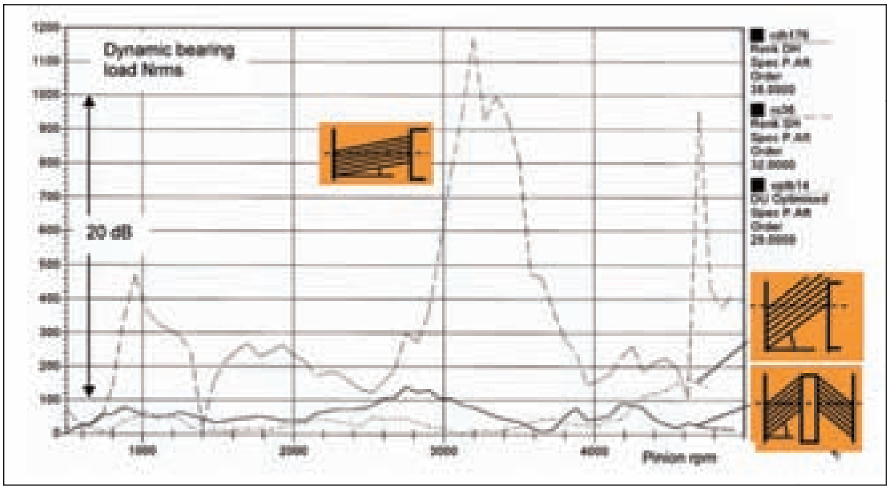


Figure 14—Representative comparison of dynamic bearing loads v. pinion speed (and related propeller law torque) with three gear designs, as detailed in Table 1.

revealed. Double helicals have significantly lower rotational and mesh frequency excitation than single helicals, combined with a broader band behavior. These fundamental results were observed throughout all parameter variations, as obtained with more than 200 single measurements taken.

Figure 13a and b give an impression on the sensitivity of double helicals and single helicals depending on various adjusted misalignment values between pinion and wheel. Basically, with all measurements, a single curve represents an orthogonal slice at gear mesh frequency orders. Evidently, double helicals show lower excitations by a factor of four compared to single helicals, and a fairly low insensitivity to misalignment. This effect is interpreted as the consequence of the self-centering effect in the gear mesh.

As a conclusion, high helices and double helices show a similar performance regarding their dynamic behavior, whereas low helices—even at equally high accuracies—present higher dynamic forces due to their limited macro geometry and structural impacts as discussed above. As an overall representative result, Figure 14 shows single helicals generating up to 20 dB higher structural noise compared to double helicals.

Conclusion

The stat-of-the-art high-speed gear includes a sophisticated gear design respecting any kind of external impacts to operational conditions. Heat expansion and low noise signature requirements are to be compensated by the appropriate selection of macro tooth geometry combined with refined lead and profile tooth corrections. Double helicals show the best performance throughout numerous applications as can be seen with various installations in industrial fields or aboard vessels of any kind. Moreover, if specific solutions are required, such as complex gear trains with multi-functional clutch arrangements, the gear layout is to be flexibly adjusted to environmental conditions, without leave of design principles.

__Dr. Toni Weiss__ studied mechanical engineering and design at Munich Technical University and earned his doctorate while working as a scientific assistant at FZG's gear research center in 1983. He matriculated to Allianz Centre of Technology in Ismaning, Germany where he worked as a specialist for machine elements. Weiss was hired as head of design and development department for industrial gears at RENK AG of Augsburg, Germany in 1985 and became general manager of the industrial gear division in 1991.

__Dr.-Ing Franz Hoppe__ graduated from the Technical University of Munich and was subsequently appointed assistant teacher responsible for experimental and theoretical investigations in propulsion system technology. After earning his PhD, he held several positions in engineering management and propulsion systems sales. Currently, Hoppe is general manager at RENK AG and responsible for engineering and sales of marine gears.

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Chicago's Movable Bridges—

A Perfect Balancing Act of Gear-Driven Splendor

Jack McGuinn, Senior Editor

Every spring and fall, a flotilla of pleasure craft rely on Chicago's string of downtown movable bridges for access to and from Lake Michigan. (Photo by David M. Solzman)

Chicago has been known as many things over the years—"Hog Butcher to the World," "The City That Works," "The Windy City" and "The City of Big Shoulders" among them.

Although perhaps lesser known, add "City of Bridges" to the list.

But we're not talking about traditional, stationary bridges. Rather, we're talking trunnion bascule bridges—or draw bridges—that span the Chicago River along many of the Loop's downtown intersections.

Although Chicago's first trunnion bascule bridge is the Cortland Street Bridge, built in 1902, the crown jewel of the city's movable bridge system is the Michigan Avenue Bridge at the intersection of Wacker Drive and Michigan Avenue. Begun in 1917 and completed in 1920, the bridge was conceived by renowned city planner and architect Daniel Burnham—he of "make no little plans" fame—and designed by transplanted Englishman and Chicago Plan colleague Edward H. Bennett. Burnham envisioned the Michigan Avenue Bridge in part as a means of encouraging more development of Michigan Avenue north of the river, which had been neglected due to a lack of speedy access from its more populated area to the south. Reminiscent of the meeting of the eastern and western railroads in the previous century, the bridge dedication was celebrated by cannon fire, marching bands, blaring fog horns from the ships below and thousands of Chicagoans.

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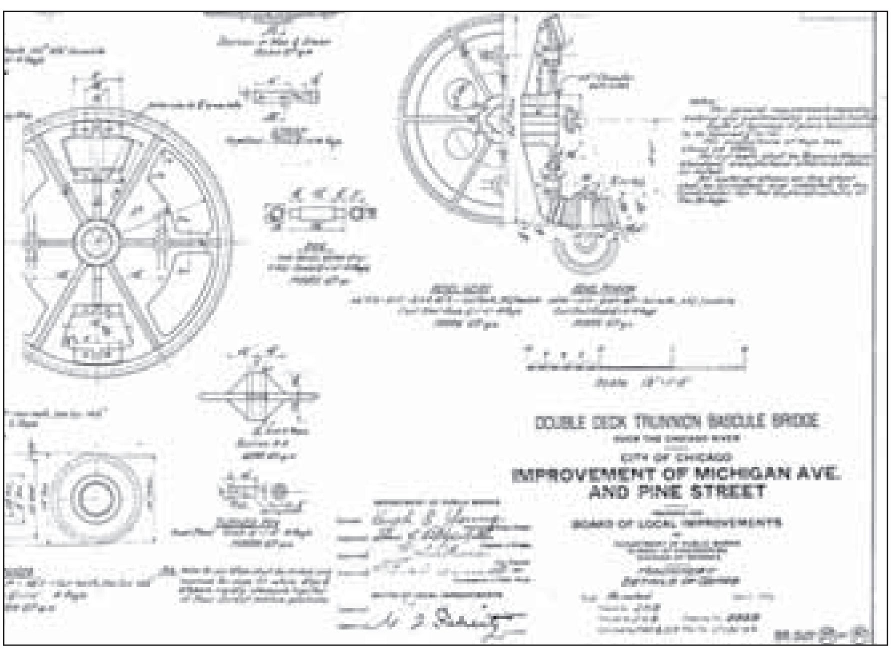
But before the first (Cortland Street) bridge was built, the city's public works department went to work to realize Burnham's vision.

"The department of public works commissioned a study in 1898 to look at the suitability of different types of bridge structures to implement on a large scale in the city," says Dan Burke, assistant chief engineer bridges for Chicago's department of transportation. "And they studied and looked at several different existing technology models from around the world, which led to a new take on bridge concepts. It was a combination and further cultivation of existing technologies for the time." And so was born what is now internationally known and duplicated as the "Chicago Style" bascule bridge.

The city now boasts 35 operational movable bridges—the most of any city on the planet. Burke estimates that the bridges along the Chicago River are drawn up and down approximately 200 times a year. (At the busy 24/7 Calumet Harbor on the southeastern edge of the city, however, he says the number is 5,000 to 10,000.) The busiest times on the river are in the spring and fall, at the beginning and end of Chicago's boating season on Lake Michigan just east of the river. Indeed, the migration of pleasure craft out to Lake Michigan in the spring and their return in the fall for winter dry docking is a Chicago tradition.

Cortland Street Bridge—the first movable bridge built in Chicago (1902) and a national historic civil engineering landmark. (Photo courtesy of City of Chicago)



Detail of a 1918 schematic of the Michigan Avenue Bridge specifying "minor changes and additions" to the original design. (Drawing courtesy of City of Chicago.)

The bridge design is an engi-neering—and gearing—rarity in that it is driven by an intricate and yet almost primitive system employing gears, counter weights, and electric motors. Not unlike today, (electric) energy was costly in those days, and so the bridges were designed to run on relatively small energy output, to be complemented by the gear and counterweight functions. As the opening is initiated, the two sections, or leaves, of the bridge open at the center and remain so until lowered to the original position. The two rising leaves are actually a portion of Michigan Avenue, which is why it is known as a double-leaf, double-deck trunnion bascule; most other bridges in the system have only one leaf. The opening and closing time of the bridge is about eight minutes.

To appreciate how the bridge works, simply look at a playground seesaw. Weight being about equal, notice how each child on either end acts as a counterweight and you'll have an idea of how a movable bridge works. The steel and roadway (leaf) construction are of equal weight with its concrete-and-steel counterweight, thus providing a delicate yet precise balance, one to the other.

As mentioned, relatively little power is required to operate a bridge of this type. The Michigan Avenue bridge is powered by four 75–100 horsepower DC motors, two for each leaf or street portion. In what today is still considered an engineering marvel, those four motors drive the trunnion—or axle—which in turn activates a gear set of increasing size to power the bridge. The last in the series—the bull gear—has the largest

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teeth of the set and actually rotates the rack and in turn powers the raising and lowering of the bridge sections. The result, says Burke, "is a simple concept executed on a very large scale."

Specifically, he says, "(The gears) are open-gearing gear trains with a combination of spur and interlocking pinion gears. The trunnion is the axis of rotation on the bridge."

For those looking for an up-close view of the Michigan Avenue Bridge, a designated Chicago landmark, there now exists the recently opened McCormick Tribune Bridgehouse & Chicago River Museum. The Beaux Arts-style bridgehouse—replete with four bas relief sculptures depicting key historic events in the city's history—is located at the northwest corner of Michigan Avenue and Wacker Drive. The museum is open from June 1 to September 30th, 10 a.m. to 5 p.m.; closed Tuesday and Wednesday. Admission is $3, and children under five are admitted free. Groups of 20 and up welcome @ $2 per person.

For More Information:

McCormick Tribune Bridgehouse &
Chicago River Museum
376 N. Michigan Avenue
Chicago, IL 60610
Phone: (312) 977-0227
E-mail: *info@bridgehousemuseum.org*
Website:
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A look at the gear set that powers the Cortland Street Bridge. (Photo courtesy City of Chicago.)

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TOS SVITAVY WEILER HOLOUBKOV S.R.O. INTOS CKD BLANSKO

Full Factory Parts & Service

Machine Tool Leaders
DESCEND UPON HANNOVER

Billed as "the metalworking world all in one place," this year's EMO will take place September 17–22 at the Hannover Fairgrounds in Hannover, Germany.

In addition to its usual array of cutting, splitting, milling and metal forming tools, manufacturing systems, precision tools, automation components, computer technology, industrial electronics and accessories, show organizers hope to top its 2005 attendance level of 160,751.

For Samputensili marketing manager Patrizia Fiaccadori, it's the quality of the EMO visitors that makes the show the most important event for the company all year. "The EMO show in Hannover is undoubtably the most important trade fair event in Europe in the gear manufacturing sector. In recent years, the number of visitors has decreased since companies no longer send large delegations to the fair. Rather, in our experience, the trend is now for one or two top level decision makers to attend with the aim of conducting important technical meetings and sales negotiations with suppliers during the fair itself," she says.

In addition to the trade show, EMO has several educational offerings to enhance the show experience.

The Intelligent Lightweight Design Symposium provides an overview of the current development status for modern, lightweight design concepts and highlights examples from various sectors of manufacturing technology. The symposium takes place September 18, from 10 a.m.–4:30 p.m. in Rooms 3A and 3B of the convention center. Registration is 195 euros, and 125 euros for exhibitors.

Results of the NEXT Generation Production Systems, a four-year project exploring the complete value-chain for

the European production machinery sector, will be discussed on September 21 from 9 a.m.–12:30 p.m. The free lecture is broken down into five objective tracks, three technical tracks and one education and dissemination track including;

- Track 1—The Green Machines
- Track 2—User-centric autonomous machine tool
- Track 3—Manufacturing breakthroughs
- Track 4—New Business Concepts
- Track 5—New Training and Dissemination Methods.
- NEXT Generation Production Systems
- Youth Show
- Symposium on Growth Market in Central and Eastern Europe

Corporate management is also invited to attend the International Business Symposium on Growth Markets in Central and Eastern Europe. For 195 euros (125 euros from exhibiting companies), decision makers in production technology can learn the latest updates on the economic situation of various markets involved, the conditions involved for establishing sales, services and manufacturing operations, tips on creating the right networks, political and legal idiosyncrasies, financing operations in eastern Europe as well as case studies and practical examples. The symposium is September 19 from 8:30 a.m.–6:10 p.m. in Room 1A of the convention center.

Lastly, the VDW and the DMG Training Academy are presenting a special youth show for students on how to start training for a manufacturing career, and will display machine tools and robots in Hall 25 Booth C03.

EMO occurs bi-annually, twice consecutively in Hannover, Germany and once in Milan, Italy. EMO 2009 will be in Milan. For the 2007 show, day tickets are 27 euros, season tickets are 48 euros and student tickets are 12 euros and all can be ordered online. For more information on EMO 2007, visit the show's website at *www.emo-hannover.de*

September 10–12—Romax Technology USA Users Conference. Romax facility, Detroit, MI. Presentations include various transmission engineering topics such as bearings, gearing, NVH, advanced transmission concepts (AMT/DCT/CVT/Hybrid), design of equipment and optimization, process integration and process improvement. For more information, email *events@romaxtech.com* or visit the company's website at *www.romaxtech.com*.

September 10–14—AGMA Training School for Gear Manufacturing. Richard J. Daley College, Chicago, IL. The basic course offers classroom and hands-on training in gearing and nomenclature; principles of inspection; gear manufacturing methods; and hobbing and shaping. Although the course is designed primarily for newer employees with at least six months' experience in setup or machine operation, it can also benefit quality control managers, sales reps, management and executives. This course is taught by Dwight Smith, president of Cole Mfg. Systems, and Peter Grossi. $850 for AGMA members, $950 for non-members. *Note: A classroom version of this course will be offered at AGMA's Gear Expo in Detroit that will be taught by Mike Tennutti of Gleason Cutting Tools. For more information, contact the American Gear Manufacturers Association by telephone at (703) 684-0211 or on the Internet at *www.agma.org*.

September 12–14—Basic Gear Noise Short Course. Department of Mechanical Engineering, Ohio State University, Columbus, OH. Offered for gear designers and noise specialists who encounter gear noise and transmission design problems. Attendees will learn how to design gears to minimize the major excitations of gear noise: transmission error, dynamic friction and shuttling forces. Fundamentals of gear noise generation and gear noise measurement will be covered along with topics on gear rattle, transmission dynamics and housing acoustics. Course includes extensive demonstrations of specialized gear analysis software in addition to the demonstrations of Ohio State gear test rigs. An interactive workshop session invites attendees to discuss specific gear and transmission noise concerns. $1,550. For more information, contact GearLab by telephone at (614) 292-5860 or by e-mail at *harianto.1@osu.edu*.

September 14—Korean Technology Transfer Seminar. Seoul Venture Town, Korea. This seminar focuses on planetary gearbox analysis. Other topics include gearbox design and analysis, NVH, gears, bearings, shafts and housings. For more information, contact Romax Technology by email at *SeungwonJ@romaxtech.com* or on the Internet at *www.romaxtech.com*.

September 17–18—Advanced Gear Noise Short Course. Department of Mechanical Engineering at Ohio State University, Columbus, OH. This advanced session is an extension of the Basic Gear Noise Short Course. This advanced course will be taught through lectures on selected topics coupled with a series of hands-on workshops. Based upon their interests, the attendees may select from the following topics:

- Analytical and computer modeling (prediction of gear whine excitations, general system dynamics, bearing/casing dynamics, gear rattle models).
- Experimental and computational approaches (modal analysis of casings, acoustic radiation, advanced signal processing, sound quality analysis, transmission error measurement).

$1,050. For more information, contact the GearLab by telephone at (614) 688-3952 or by e-mail at *harianto.1@osu.edu*.

September 17–19—ASME Heat Treating Society Conference and Expo. Cobo Center, Detroit, MI. The conference occurs from September 17–19 and the expo on September 18–19. The technical program is broken into sessions on applied energy, brazing, emerging technology, equipment innovation, global issues, processes and applications and vacuum technology. Prices range from $25–$780. For more information, contact the ASME Heat Treating Society by telephone at (301) 694-5243 or online at www.asme.org.

September 17–20—Basics of Bevel and Hypoid Gears. Zurich, Switzerland. This course is limited to 10 participants and conducted in German. Topics include gear cutting systems, gear geometry, offset and hand of spiral, ease-off, load related deflections, bias conditions, influence of cutter diameter, example gear design calculations, and gear measurement and closed loop. 2,000 euros per person. For more information, contact Klingelnberg AG online at *www.klingelnberg.com*.

September 17–22—EMO Hannover. Hannover Fairgrounds, Hannover, Germany. Featuring products and services for metalworking technology, with special emphasis on machine tools, manufacturing systems, precision tools, automated systems, computer technology, industrial electronics and accessories. Sponsored by the VDW (German Machine Tool Builders Association). See our coverage on p. 84. For more information, visit the show's website at *www.emo-hannover.de*.

September 25–27—Assembly Tech Expo. Donald E. Stephens Convention Center, Rosemont, IL. Co-located with Quality Expo, PlasTec Midwest, Electronics Assembly Show and National Manufacturing Week. Registration is free if you pre-register online or bring registration materials to the show floor. Otherwise, registration is $55 at the door. For more information, visit the show's website at *www.atexpo.com*.

September 24–27—KIMoS Basics. Zurich, Switzerland. This course is limited to eight participants. Topics include:

- Repetition of basic theory and spiral bevel gears
- KIMoS modules
- Dimensioning
- Ease-off optimization
- Re-engineering
- Data stroking and data transfer via e-mail
- Practice on a master gear set.

  2,000 euros per person.

For more information, contact Klingelnberg AG online at *www.klingelnberg.com.*

October 2–4—South-Tec 2007 Conference and Expo. Charlotte Convention Center, Charlotte, NC. SOUTH-TEC 2007 also features SME Motorsports Charlotte exhibits, which are geared toward motorsports manufacturers in need of technologies and processes. This event also includes a full technical conference. Visit the event's website closer to the date of the show for prices and registration information. Online registration (valued at $25) is free and open throughout the show. For more information, visit the show's website at *www.sme.org/southtec.*

October 7–10—Gear Expo 2007. Cobo Center, Detroit, MI. "The Worldwide Gear Industry Event." This year's show promises 200 exhibitors with more than 60 gear manufacturers worldwide. AGMA, SME and the American Bearing Manufacturing Association will conduct gear-related seminars and the Gear Expo Solutions Center will host free presentations from more than 20 exhibiting companies. For more information, visit *www.gearexpo.com* or contact the AGMA by telephone at (703) 684-0242. See our full coverage on pages 32–49.

October 15–17—Gear Manufacturing Technology Course. Perry Technology Co., New Hartford, CT. Instructors Geoff Ashcroft and Ron Greene aim to teach participants gear theory as well as practical aspects of manufacturing and troubleshooting techniques. The course will cover gear inspection, gear manufacturing, gear hobbing, gear shaping, hobs, gear shaping tools, production estimating, hard finishing and gear shaving. $750 includes tuition, materials, an AGMA reference manual and certificate of completion. For more information, contact the Gear Consulting Group by telephone at (269) 623-4993.

October 22–26—17th Annual Shot Peening and Blast Cleaning Workshop. Fiesta Inn & Resort, Tempe, AZ. This is the world's largest workshop specializing in shot peening and blast cleaning and offers more than 49 topics. FAA-recognized shot peening certification exams are offered for Level 1, 2 and 3. A flapper peening exam is also offered. For more information, contact *The Shot Peener* magazine by telephone at (800) 832-5653 or on the Internet at *www.shotpeener.com.*

October 31–November 1—Wind Power Shanghai. *Sheraton Grand Tai Ping Yang Hotel, Shanghai, China.* Turbine manufacturers and component suppliers can attend lectures by senior international executives on growth potential in the wind energy market from Oct. 31–Nov. 1. A trade exhibition will follow at the Shanghai International Exposition Center from Nov. 1–Nov. 3 and will include wind turbines, rotor blades, gearboxes, electricity generators and controlling and breaking systems. For more information, visit the show's website at *www.windpowershanghai.com.*

October 10–13—PTC Asia 2007. Shanghai New International Expo Center, Shanghai, China. Exhibition for power transmission and control, fluid power (hydraulics and pneumatics), mechanical and electrical transmission, compressed air technology, internal combustion engines and gas turbines. Organizers expect more than 1,300 exhibitors and approximately 58,000 visitors. For more information, visit *www.ptc-asia.com.*

October 18–20—Power Transmission Distributors Association Annual Summit. Marriott Desert Springs Resort and Spa, Desert Springs, CA. Main networking and educational event for the power transmission industry. Workshops scheduled include topics such as driving growth and shareholder value and understanding the demographics of the evolving workforce. Registration is available to PTDA members online at *www.ptda.org.*

October 22–24—International Joint Tribological Conference. Marriott Mission Valley, San Diego, CA. Focuses on research in the multidisciplinary field of tribology. Sponsored by The Society of Tribologists and Lubrication Engineers (STLE) and the Tribology Division of ASME International. Papers will be presented in all fundamental and applied fields of tribology, and will be in the form of two- to three-page (maximum) Extended Abstracts. Topical areas include: Tribomaterials and Tribology Fundamentals; Lubricants and Additives; Elastohydrodynamic Lubrication; Hydrodynamic Lubrication and Fluid Film Bearings; Rolling Element Bearings; Engine Tribology; Machine Components Tribology; Contact Mechanics; Magnetic Storage Tribology; Manufacturing/Metalworking Tribology; Nanotribology; Engineered Surfaces; Biotribology; and Emerging Technologies. For more information, contact the Society of Tribologists and Lubrication Engineers at www.stle.org or visit the official conference website at *http://www.asmeconferences.org/ijtc07/.*

GREAT LAKES GEAR TECHNOLOGIES

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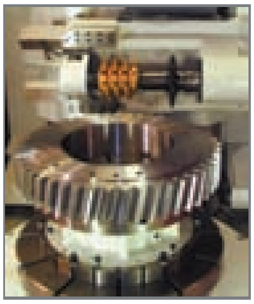
Höfler HF Gear Hobber/Gashers featuring exclusive Hi-accuracy torque motor worktable drives, heavy duty horsepower direct drive spindles, and more in sizes from 1 to 3 meters.



Höfler HF 900 – booth # 200

Höfler Rapid 900 – booth # 100

Höfler Rapid Gear Grinders featuring our exclusive Hi-accuracy torque motor worktable drives, onboard inspection, internal grinding arms, and more, up to 6 meters.

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Fässler Gear Honing Machines -

Why Grind when you can ***"MegaHone"***.

Visiting the Great Lakes Gear Technologies booth you will see one of the largest machines ever to be displayed at the Gear Expo - the Höfler HF 900 gear hobber - so prepare to be "very impressed" ! And in the adjoining Höfler booth will be the Rapid 900 gear grinder.

Also exhibiting for high volume gear and spline manufacturers we will feature Fässler and Escofier equipment.

Fässler **gear honing for fast and efficient hard gear finishing for durable and reduced noise geartrains. Diamond spline broaching machines and the well regarded Fässler threaded wheel dresser and tools.**

Escofier **gear and spline rolling equipment for the most efficient and accurate alternatives to gear shaving or for powder metal gear finishing. Utilizing circular die spline rolling for unmatched flexibility, tool life and size control.**

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GREAT LAKES GEAR TECHNOLOGIES

Carraro Group

FINALIZES TAKEOVER OF MG MINIGEARS

Gear World SpA, a newly established company, put the finishing touches on its 100% takeover of mG holding SpA, the controlling company of mG miniGears.

According to a company press release, Gear World's pro-forma sales are expected to top 220 million euros by the end of 2007.

Gear World is part of a strategy aimed at granting independent growth to the group's activities in the international gear industry. Carraro's businesses already in the field include Siap in Italy, Turbo Gears in India and the Gear Division of Carraro Argentina, all of which will be integrated with miniGears S.p.A.

In addition to the production facility and head offices of Padua, Italy, miniGears has offices in Virginia Beach, VA and Suzhou, China.

"The task of Gear World will also be the development of new high-tech components with high gear-mechanism content, such as phase transformers for automotive use, which we expect will contribute significantly to the group's growth in the components sector," says Carlo Borsari, managing director of the Carraro Group.

Riverside Spline and Gear

ACQUIRES HÖFLER GRINDER

Riverside Spline and Gear announced the acquisition of a Höfler Rapid 900 gear grinder, due to arrive at its 9,000 square foot gear grinding center in November.

According to the company's press release, the new machine offers AGMA class 14+ capability and on-board gear inspection.

Additional features for external gears include gear diameters from 50–1,000 mm, helix angles ranging from 45–90°, a module of 1–25 mm that is extendable to 34 mm, a grinding wheel depth of 60 mm that is extendable to 90 mm and table load of 3,000 kg.

For internal gears, the Höfler offers a workpiece diameter of 250–900mm, a face width of 280 mm, grinding wheel diameter of 200–110 mm and grinding wheel width of 30 mm.

Riverside Spline and Gear also offers CMM gear analysis, magnetic particle inspection and nital etch inspection.

McNeil Industries

RECOGNIZED AS A TOP FIVE PRIVATE GROWTH COMPANY

By posting a second-quarter growth rate of 30% in total revenues, McNeil Industries was recognized as a "top five" private growth company by the Private Company Index (PCI).

Used by financial market and accounting professionals, PCI is an index that benchmarks the performance of more than 500 small- to mid-sized U.S. private companies in diverse market sectors, whose annual revenues range from $1–$80 million annually.

McNeil Industries' ranking was based on average month-on-month growth for the second quarter of 2007, relative to the previous year results. According to company CEO Randall J. McNeil, the company's growth was driven by significant quarterly sales growth in industrial MAXAM bearings, stamping-related guide systems components, precision machined products, and its acquisition of OLS, Inc., a Cleveland-based developer of automated deburring equipment.

McNeil Industries, based in Painesville, OH delivers MAXAM® plain, spherical, and cylindrical bearings that offer extended service life, even under harsh load, moisture, heat, and contamination conditions. McNeil Guide Systems' pins, bushings, bearing cages and spare parts are used by the precision stamping industry, while deburring systems automate secondary operations.

Romax

OPENS NEW OFFICES IN MICHIGAN AND JAPAN

Romax Technology announced the opening of its new offices in Tokyo and Troy, MI.

continued

In addition, the company hired Hayao Yazawa as representative director and Ryo Hiratsuka as Japanese business manager of the Romax Technology Japan KK subsidiary.

According to the company's press release, Yazawa has more than 40 years' automotive experience and Hiratsuka has extensive experience in the Japanese business community.

The U.S. office will be led by Paul Lefief, Chris Baker and Kurt Sheridan.

Romax Technology recently opened field offices in Japan, South Korea and India.

Lufkin Industries

ELECTS NEW PRESIDENT

Lufkin Industries announced the promotion of John (Jay) F. Glick to the office of president of the company.


John F. Glick

Glick, who has been a senior officer with the company since 1994, will take over for former president, Douglas V. Smith, who served in that position since January 1993.

Glick was previously the vice president/general manager of both the oilfield division and the power transmission division at Lufkin Industries. Prior to that, he also worked at Cameron Iron Works in England, managing first one and then two plants.

Former president Smith will continue to serve the company as chief executive officer and chairman of the board.

According to a Lufkin press release, the company authorized the use of $30 million in August to repurchase shares of Lufkin's stock. It had 15.1 million shares outstanding as of June 30.

Lufkin Industries sells and services oilfield pumping units, foundry castings, power transmission products and highway trailers.

AGMA Foundation

HIRES NEW DIRECTOR

AGMA hired Sandra Detwiler as its new executive director of the AGMA Foundation.

Detwiler has 10 years of association-foundation experience, most recently with the Society of Interventional Radiology Foundation in Fairfax, VA.

As executive director, Detwiler's responsibilities include managing the foundation's affairs, including the annual campaign, fundraising events at AGMA's annual meeting, promotions in association publications, and grant projects to advance gear science, standards and education.

Five New Companies

JOIN AGMA

The American Gear Manufacturers Association added five new member companies this month.

New member companies include:

- Bluffton Motor Works LLC
- Broaching Machine Specialties
- DSM Engineering Plastics
- MLS-Technologies
- Red Rover Ltd.

Inductoheat

SHIPS FIVE FREQUENCY SYSTEMS

Inductoheat Inc. shipped five IROSS induction line frequency systems to a supplier of engine crankshaft vibration dampers.

Units are designed with automated walking beams for induction bonding of vibration dampers. The systems produce up to 125 pieces per hour depending on part weight. According to the company's press release, the first units purchased in 1993 are still in production.

Brad Foote Gear

SOLD TO TOWER TECH

According to a report issued by the American Wind Energy Association, Tower Tech Holdings Inc. acquired Brad Foote Gear Works Inc., a producer of gearing systems for the wind energy, oil and gas and energy-related industries.

Under the terms of the agreement, Brad Foote CEO Cameron Drecoll, majority shareholder in the gear manufacturing company, will assume chief executive responsibilities of the combined company and Tower Tech president Raymond Brickner will maintain his current role as head of the Tower business.

Brad Foote manufactures gear systems at two locations in Cicero, IL and one in Pittsburgh, PA. Shareholders will receive an aggregate of 16 million shares of Tower Tech common stock and $64 million in cash.

Wheelabrator

HIRES NEW SERVICE MANAGER

Juan Jose Limón recently joined Wheelabrator Group as sales/service engineer in Mexico.


Juan Jose Limón

In this position, Limón will provide technical assistance and account management to Wheelabrator customers in northern Mexico. Additionally, he will promote the sale of Wheelabrator's aftermarket products, the Equipment Modernization Program (EMP), Original Equipment Manufacturer schools and in-house training.

According to the company's press release, Limón, an electronic engineer, previously worked at Nippon Crane as an electrical service specialist.

Mitsubishi Heavy Industries

COMPLETES GEAR CUTTING TOOL PRODUCTION PLANT IN INDIA

Mitsubishi Heavy Industries Ltd. completed construction of a new plant at Mitsubishi Heavy Industries India Precision Tools, Ltd., an Indian subsidiary dedicated to manufacturing and sales of gear cutting tools.

The $10 million new plant was built in response to demand from the Indian automotive industry, which has been growing 20% per year. According to the company's press release, production is expected to double.

The new plant, with 37,500 square feet of floor space, was built next to an existing plant. To double the production capacity, more than 80 machines were installed in the new plant, including the latest numerical control machines. The number of employees was increased from 400 to 590.

Automotive output in India exceeded two million units in 2006 and is expected to reach four million in 2010.

MHI-OPT, an overseas production base of MHI's machine tool division, is headquartered in southern India's Tamil Nadu state, at Ranipet near Chennai. The company manufactures precision gear cutting tools including hob cutters, gear shaping cutters, gear shaving cutters and broach cutters.

Moventas

INVESTS IN WIND TURBINE GEAR PRODUCTION, HIRES NEW EXECUTIVES

Moventas invested approximately 115 million euros in a new wind turbine gear factory in central Finland. Construction will begin in fall 2007 and the building is slated for completion in 2008. Overall production capacity will be taken on-stream progressively throughout 2009.

According to the company's press release, the new factory will double Moventas' wind turbine gear production capacity and follows a 100 million euro investment that was announced in February.

The current investment creates about 100 new jobs. An additional 200 new production employees will be required for the new factory.

Since the 22,500 m^2 facility will be constructed new, the factory can be optimized for megawatt-scale wind turbines. On a global basis, the wind power market faces a 25% yearly growth.

Moventas recently reported nearly 100 million euros of wind turbine gear orders. The company has signed agreements worth 97 million euros with REpower, WinWinD and Dewind.

The power range of the gear units varies from 660 kW to 3.0 MW. Gear units are planetary-helical, weighing 4.5–20 tons with the exception of the planetary gear solution developed in cooperation with WinWinD. The units will be delivered to turbine manufacturers in Germany, India, Finland and the United States

The company also announced the appointment of Henry Böhling as vice president of research and development and quality, Kari Reini as vice president of the Karkkila factory and Olavi Rahkonen as vice president of expansion projects.

Böhling will be responsible for global industrial gears R&D support and coordination as well as managing industrial gears R&D in Finland.

Reini will be responsible for operational management of the Karkkila factory.

Rahkonen will lead the new wind gear factory project, effective September 19.

Process Equipment

ANNOUNCES PARTNERSHIPS

Process Equipment Co. announced a distributorship agreement with Arthur Klick Co. for distribution in Europe of the company's gear inspection equipment

The two companies will be exhibiting jointly at EMO Hannover in Hall 6, Booth F26.

Process Equipment will also partner with Star SU for sales of gear inspection equipment. Star SU and Process Equipment will occupy the same booth at Gear Expo, where they will exhibit the ND300.

NEWS

Ford

ANNOUNCES SALE OF AUTOMOTIVE COMPONENTS HOLDINGS' PTU BUSINESS

Ford Motor Co. and Linamar Corp. announced the signing of definitive agreements for the sale of the Automotive Components Holdings' Power Transfer Unit (PTU) business and its Concerca I Plant in Neuvo Laredo, Mexico.

The ACH Converca I Plant is among the leading manufacturers of power transfer units in the North American auto industry. As part of the deal, assets of the plant will be transferred to Linamar. The plant also produces propshafts, stabilizer bars and steering gears. According to Ford's press release, stabilizer bar production will end later this year and propshaft and steering gear production will continue at Converca.

"This sale is another demonstration of our commitment to achieve the material cost goals in our Way Forward strategy," says Mark Fields, president of the Americas and Ford executive vice president. "This is critical as we work toward our goal of profitability in North America by 2009."

Ikona

RECEIVES $150,000 ORDER, SIGNS DISTRIBUTION AGREEMENT

Ikona Gear International received a $150,000 order from Cummins API S.A. for its new 1100 HDC hydraulic power take-off product line for the oil and gas industry.

Cummins API S.A. is the South American distributor for a designer and manufacturer of diesel engines ranging from 55–3,500 horsepower and a company that designs and manufactures rotary positive displacement pumps.

According to Ikona's press release, its 1100 HDC hydraulic power take-off is used to drive a mud pump as a means of transmitting power to it from a diesel engine. The 1100 HDC provides a high-speed clutch disengagement and an all-steel torsional coupling. Additional features include overhaul intervals of 50,000 hours and electrical controls for integration with diesel engine controls.

Ikona also announced a distribution agreement with AmeriMex Motor and Control Inc.

continued


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Under the terms of the agreement, AmeriMex will be the exclusive distributor of Ikona's oil and machinery product line, including its two-speed mud pump drives and drawworks in Texas, Oklahoma and Louisiana. AmeriMex offers new and remanufactured electric motors, generators and motor controls.

Bill Toelke, vice president of sales at AmeriMex, says "Ikona's products, which were designed and manufactured specifically for the oil and gas drilling rigs, weigh less, present a smaller footprint and are more energy-efficient than competitive products. The innovative design of these products has made them relatively maintenance-free—another factor that reduces the overall cost of the drilling process."

Northstar Aerospace

RECEIVES CONTRACT FOR NEW DRIVE SYSTEM DEVELOPMENT

Northstar Aerospace was awarded a $4.5 million development contract from The Boeing Co. for its Enhanced Rotor Rotorcraft Drive System (ERDS), a joint program with The Boeing Co. and U.S. Army Aviation Applied Technology Directorate.

The ERDS program focuses on developing new drive system technologies to improve the performance of the U.S. Army's current and future force fleet of rotorcraft, including the AH-64D Apache Longbow and CH-47 Chinook.

Northstar will provide Boeing with design support, fabrication and validation testing of critical drive system technologies such as face gearing to improve power-to-weight ratio and reduce noise.

The two companies are also collaborating on development of a transmission using face gear technology for the U.S. Army's Block III AH-64D Apache Longbow helicopter modernization program. According to the company's press release, the Block III will be the first aerospace application of the transmission technology. The transmission uses face gears in a split-torque design facilitating more horsepower without the penalty of added weight.

Acme Manufacturing

OPENS SHANGHAI OFFICE

Acme Manufacturing Co. announced the opening of an office in Shanghai, China and the hiring of Ling Yun to manage the new office.

According to the company's press release, Yun has more than 20 years' experience in the metalworking and machine tool industry.

The new facility will be located at:
Acme Manufacturing Co.
1F Bldg. #36, 458 North Fu Te Rd.
Shanghai Waigaoqiao, FTZ 200131 China

ANCA

APPOINTS EUROPEAN GENERAL MANAGER

ANCA appointed Markus Berger as general manager responsible for European operations.

According to the company's press release, Berger was recruited from Schunk in Lauffen, Germany, where he was global sales and marketing manager for workholding systems.

Berger will be based in Mannheim, Germany.

Star SU and Sicmat Sign

DISTRIBUTION AGREEMENT

Sicmat has partnered with Star SU for the North American distribution of the Sicmat line of gear shaving machines.

Located in Torino, Italy, Sicmat is a producer of gear shaving machines for the automotive industry with a customer list that includes Caterpillar, DaimlerChrysler, Fiat, General Motors, John Deere, Peugeot-Citroen, Piaggio, Renault, Tata, VCST and Visteon.

Magna International

ANNOUNCES NEW EXECS

Magna International Inc. announced that Greg Deveson has assumed the position of president of Magna Powertrain North America.

I am very pleased to have Mr. Deveson join Magna Powertrain," says Dr. Herbert Demel, president of Magna Powertrain. "I have known Mr. Deveson for a long time, particularly from our previous cooperation in the Saab 9(3) program at Magna Steyr, as well as from the Fiat - GM Powertrain Joint Venture. He brings vast experience in the powertrain business, especially in the area of operations, to our management team." agna Powertrain, an operating group of Magna, supplies drivetrain components and systems, and transmission and engine technologies. Demel became president of Magna Powertrain in June. Previously, Demel had been CEO and president of Magna Steyr from 2002 to 2003 and CEO of Fiat Auto in Turin since 2003.

Renk AG

ANNOUNCES EXEC CHANGES

Dipl.-Ing. Florian Hofbauer joined the executive board of Renk AG (a MAN Group company) in September. Hofbauer previously worked for various MAN Group companies, most

continued

recently on the management board of Schwäbische Hüttenwerke GmbH, Hofbauer has since December 2005 headed the strategy & structure department of MAN AG, located in Munich.

The company also announced that executive board spokesman Prof. Dr. Manfred Hirt will retire at the end of 2007. Hirt has been on the board of RENK AG since 1990, and he was appointed spokesman in 1995.

BorgWarner

ANNOUNCES OVERSEAS INVESTMENTS

BorgWarner has announced $120 million in investments for a technical center in China, a new production campus for the Drivetrain Group in Mexico and a production facility for turbochargers to be located in Poland. The spending for these projects will occur over the next four years.

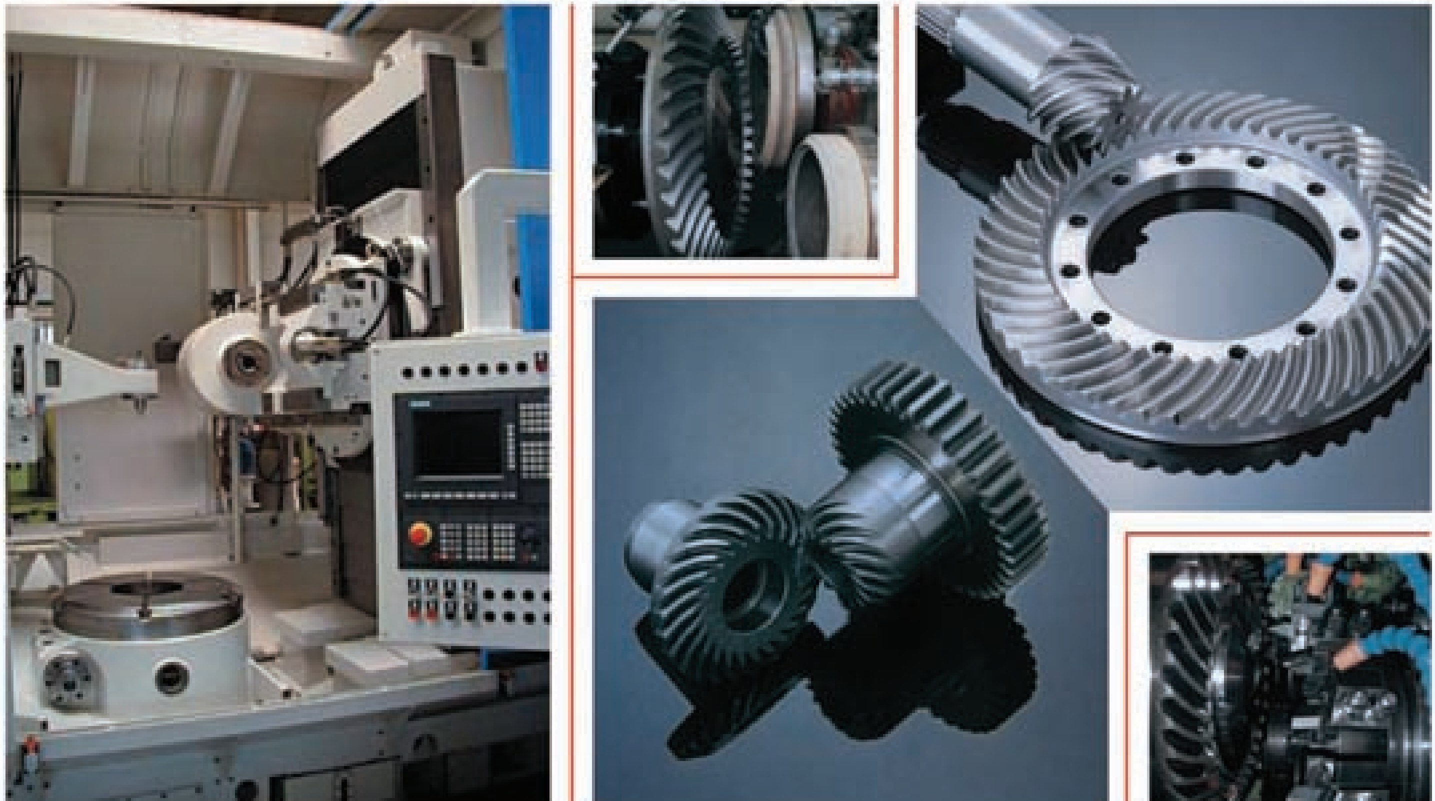
"This is an exciting time for BorgWarner, as our growth around the world is propelled by the need for vehicle fuel efficiency and improved air quality," said Tim Manganello, BorgWarner chairman and CEO. "These growth projects are evidence of the momentum we are gaining as our innovative engine and drivetrain technologies like turbochargers and dual-clutch transmission modules lead the market in delivering benefits for drivers and the environment."

According to a press release, the China technical center will support BorgWarner's accelerating growth in Asia. The proposed four-story, 286,000 square foot (26,000 square meters) China facility will be part of BorgWarner's network of global technical centers. The center will house engine and drivetrain product research and development, application engineering and administration. The company expects to invest $35 million in the project, which is anticipated to house over 300 people by 2012.

The new drivetrain campus in Mexico will support the company's first North American program for dual-clutch transmission modules as well as other transmission and all-wheel drive business. The plant is expected to produce some 680,000 dual-clutch transmission modules annually at full production volumes.

The BorgWarner Drivetrain Group is expected to build the 260,000 square foot (24,000 square meter) facility in Saltillo, Mexico with an initial investment of $67 million. The plant is expected to employ several hundred people at full ramp-up.

The new, $18 million facility in Rzeszow, Poland is expected to have capacity to make 500,000 turbochargers per year.



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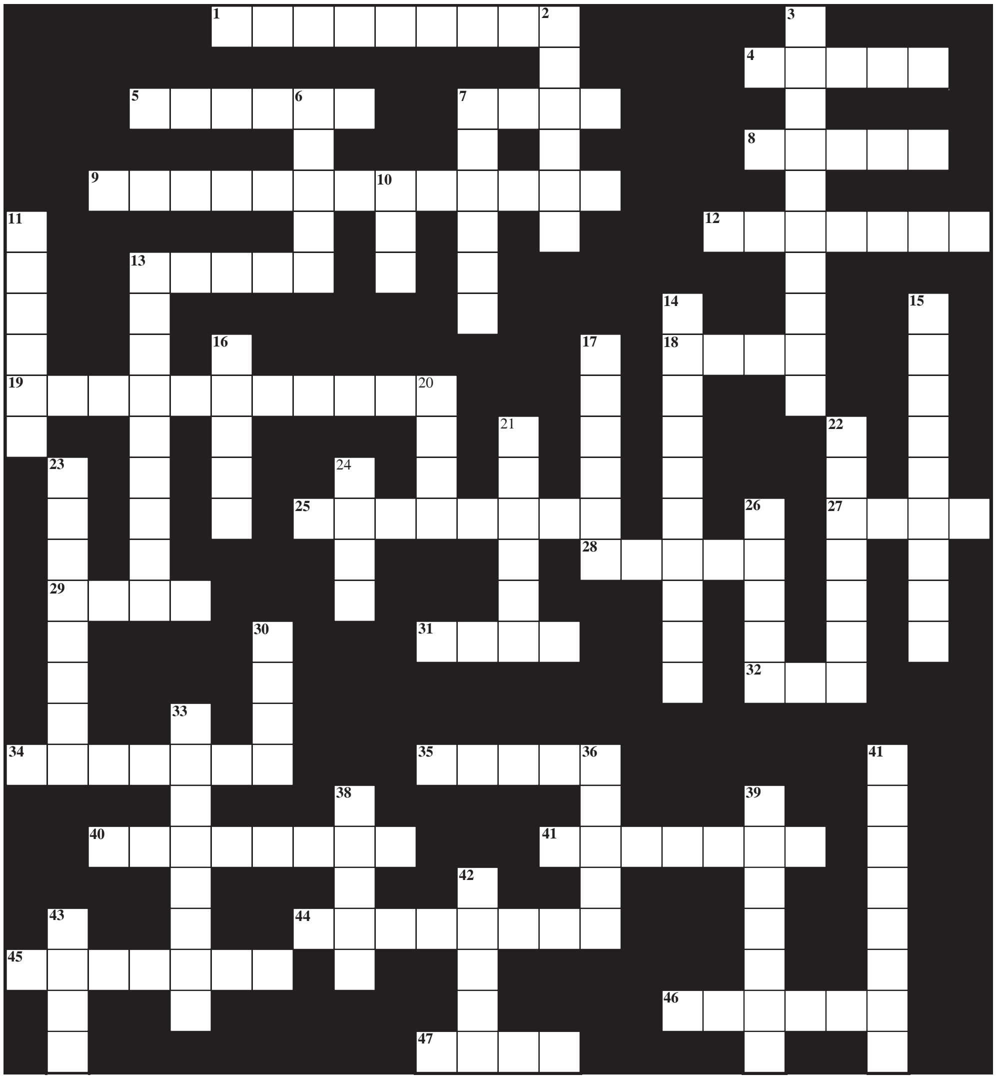

For answers go to *www.geartechnology.com/crossword.htm*

Across

1. The E in SME
4. Beta to a gear designer (_____ angle)
5. Cutting tool for internal splines
7. Measurement over ______
8. What gears are mounted to
9. Alpha, to a gear designer
12. The C in LTCA
13. Gears off the shelf
18. Bearing association at Gear Expo
19. These drive gear sales
25. TiN and TiAlN, for example
27. How every engineer wants to see himself
28. The E in TE; causes noise
29. When gears work together
31. The H in HSS
32. Four-square test _____
34. The P in SPC
35. Profile ______; gear modification
40. Wind _____
41. City of Gear Expo 2007
44. What you get from your customer if there's too much of it
45. Indication that gears should be put to bed
46. The F in FEA
47. The G in AGMA

Down

2. The S in LPSTC
3. The 'M' in ASME
6. Charlie's favorite workholding device
7. The driver
10. The electronic version of Gear Technology
11. How to "lock" a gear to its shaft
13. Don't mesh with these gear cousins
14. What a tool sharpener has to face
15. The D in EDM
16. The 'P' in HPSTC
17. A type of lube that John Travolta and Olivia Newton-John might like
20. ____ peening
21. Processes for finishing gears or holes
22. Face hobbing or face _______
23. Twice the radius
24. Hall for Gear Expo 2007
26. Toolholder for trees?
30. X, Y and Z
33. What you need to support your shafts
36. These give gears their bite
37. Location of EMO 2007
38. Powder ______; sintered gears
39. This kind of dressing is not for salads
42. Bevel gear stick ______
43. Torture device for gear lovers?