June/July 2011

GEAR TECHNOLOGY

www.geartechnology.com

The Journal of Gear Manufacturing

Feature Articles

- Large Gear Inspection
- Turbine Energy Efficiency
- Gear Hobbing Technology Update

Technical Articles

- Helicopter Main Drive Possibilities
- High-Power Spiral Bevel Gears
- Hypoid Gears

Plus

- Addendum: Remember the Corvair?

THE GEAR INDUSTRY'S INFORMATION SOURCE


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Applying more than 50 years of automotive-related machine tool building experience Sicmat developed a honing machine that is inexpensive and easy to maintain.
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• workpiece/tool synchronization
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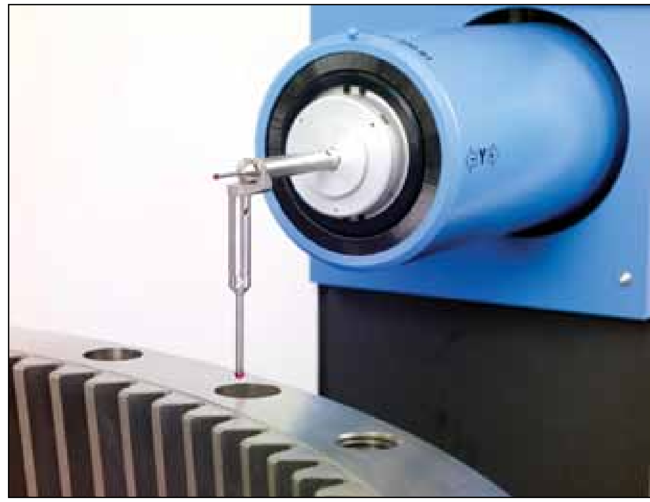

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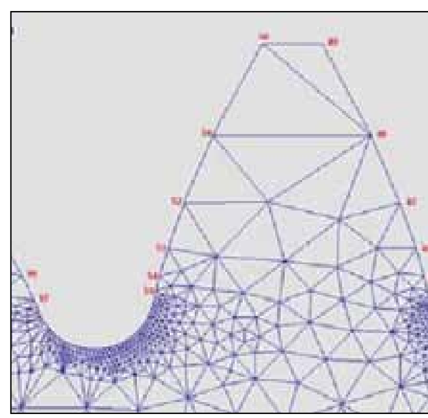

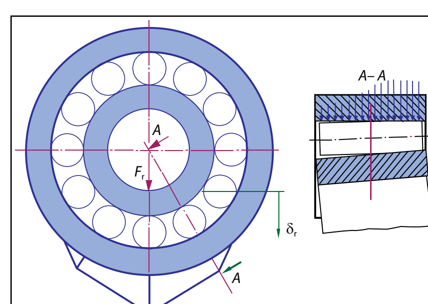

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Forest City Gear works with customers to create the highest quality, precision gears on the market. We pride ourselves on innovative approaches to new and old ideas of mechanical motion. Our application engineers have often found ways to improve the performance, the wear life and the accuracy of gears, simply by taking a *"rule the roost"* attitude about problem-solving for our worldwide customers.

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We know *"you need to crack some eggs to make the omelet"* and we incorporate that philosophy into our business, every day. Better gears are the result. Plus, you'll find the experience of dealing with our Forest City Gear employees is never a *"shell game."*

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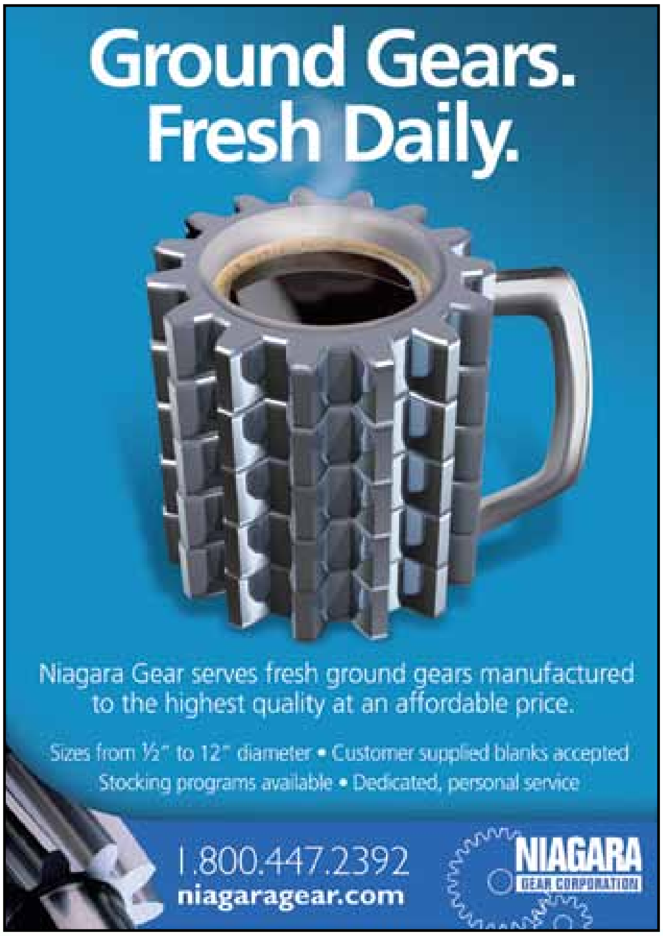

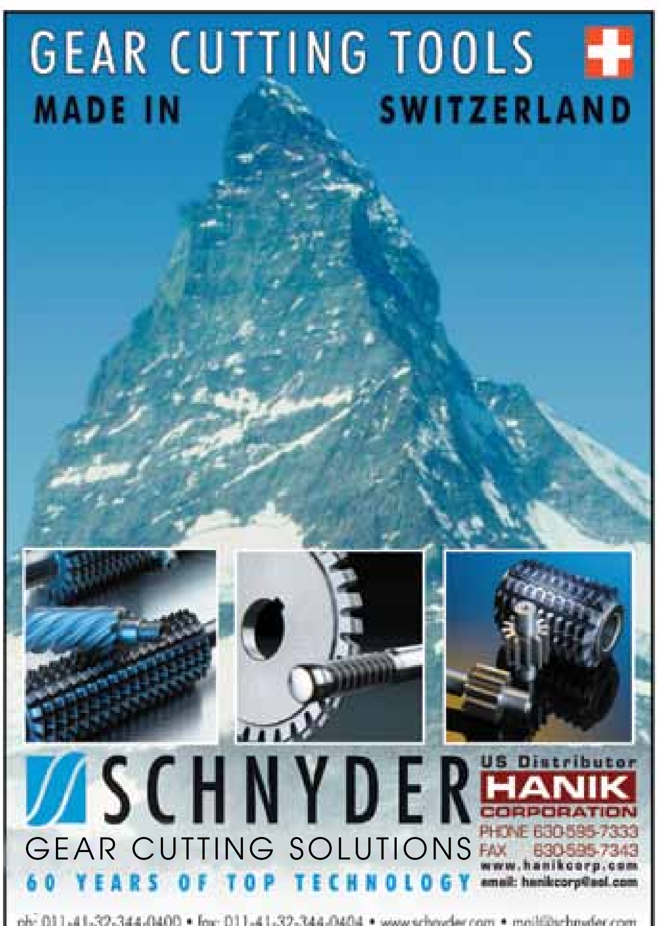

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GEAR TECHNOLOGY

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NILES
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"Today's customers demand high quality custom gearing at competitive prices and lead times. We recognize the Niles ZP40 as the machine which gives us maximum flexibility while maintaining the highest level of quality and efficiency possible. Our ability to grind and inspect large internal and external gears has definitely given us the ability to enter new markets such as Wind Energy, Marine, Rail, and Defence. The machine is well built, user friendly, consistent and reliable. Niles offers very prompt service and support. We recommend Niles Machines to any gear shop.

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Havlik Gear Inc. provides our customers start to finish service on large and small gearing projects. We supply complete products from customer drawings and specs, or offer design services if required. We can also take contract gear cutting or machining only as required by the customer. We are flexible enough to handle a wide range of projects from large Mill Gears supplied complete, to gear cutting only on small customer supplied blanks" —Havlik Gear

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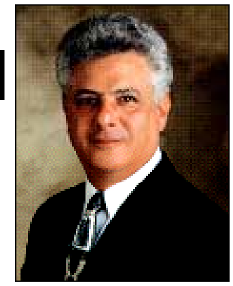

Are *You* Having Your Best Year Ever?

More than a few gear manufacturers have told me recently that they're having the **best year** in the history of their company. At the recent AGMA annual convention, the mood was especially upbeat. Orders are up, capacity is booked, and employment levels are at or near all-time highs.

Many of the suppliers to our industry are also doing well. The major machine tool and cutting tool manufacturers are also reporting a lot of quoting and sales activity.

Our industry is benefiting from a confluence of events, not the least of which is a manufacturing resurgence that's driving the current economic recovery.

Also, there's the weak dollar, which has helped American manufacturers compete on the worldwide market.

But perhaps most importantly, there was a lot of pent-up demand resulting from the recent recession. Companies were so frightened to spend any cash, they drew down their inventories to bare-minimum levels. Consumables in plants were running on fumes.

Two major bankruptcies in the American auto industry didn't help either. Instead of investing in new projects and setting up new assembly lines, the automobile companies were forced to make do with what they had, often consuming and even cannibalizing existing inventory just to stay alive.

Now, with sales propping up the automotive companies to some semblance of health, those projects have begun moving again. The result has been more orders to auto industry suppliers, including suppliers of gears as well as gear machines and cutting tools.

A number of gear manufacturers I've talked to recently—including representatives of Arrow Gear, Forest City Gear, Overton Chicago Gear and Schafer Gear, to name a few—all have shared their optimism about the current economic climate for gear manufacturers.

Of course, these are all very well respected, very well run companies that focus a lot of energy on improving processes, upgrading equipment and hiring and keeping the best talent.

One thing that these companies have all done well is differentiate themselves in the marketplace. You know that these companies are the cream of the crop, in some cases because you compete against them. But each company has also done an excellent job of communicating their strengths and unique capabilities to the marketplace. Their advertising message is part of that process.

I'm well aware that the success enjoyed by these companies hasn't been enjoyed by everyone. There are probably quite a few gear manufacturing companies that are struggling out there. There are probably some who aren't enjoying the current manufacturing market transformation.

One has to wonder if they've put the same kind of effort into improving their technology, training—and yes, marketing—that these others have done. The companies mentioned above have all had a steady marketing effort over many years—through good times and bad. Each of them has had a consistent, clever, engaging print advertising campaign. They are also active with Internet marketing, including search engine optimization, e-mail newsletters, online videos and the like. It's no wonder that when the good times rolled around, everybody knew they were the go-to guys.

Maybe your segment of the market hasn't yet come around. But when it does, whose name will be rolling off potential buyers' tongues? Will it be yours, or someone else's?

Michael Goldstein,
Publisher & Editor-in-Chief

AGMA VOICES

Gear Expo: Changing with the Times

"Welcome to Gear Expo '87...as you can see by looking around you, the need to create a show exclusively for the gear industry was a real one...we felt that the industry deserved its own marketplace to demonstrate its new technology under one roof."

One of the creators of Gear Expo and Chairman of the AGMA's Product Division in 1987, Joe Arvin (President, Arrow Gear, then and now) wrote this introduction to the show program for the 1987 Gear Expo. This is relevant today because for the first time since that beginning, Gear Expo will return to Cincinnati, Ohio for our 13th biennial trade show.

The test of time clearly shows that Arvin and the other planners were accurate in their assessment of the need for an industry-specific event for our industry.

Sixty-eight exhibitors in the 1987 show used just over 14,000 net square feet and fit comfortably in the North Hall of the convention hall. The only education event was a two-day Fall Technical Meeting, and the exhibits were open only 12 hours.

One of the reasons AGMA has been successful over our 95-year history is that the association's agenda, programs and activities reflect the voice of our members. The board of directors, advisory committees and councils and the staff vigilantly review programs and vet them with members for needed changes, updates or cancellation. The organization today is much changed from what it was in 1987, a decade ago, even a year ago.

In the 2005 show, we introduced the very popular Solutions Center which allows exhibitors to give focused presentations on new technology and products; then, invite interested listeners back to their booths for more discussion. The Solutions Center is on the exhibit floor for the convenience of participants.

We have added more education and training courses during Gear Expo and keynote presentations in the Solutions Center. We have encouraged other engineering and technology-oriented associations and groups to build more valuable education programs for attendees.

To offer more value for attendees and more attendees for the exhibitors, Gear Expo 2011 will be co-located again with the Heat Treat Show owned by the American Society of Materials.

AGMA is a focused trade show, the only event of its kind for the gear industry. The show brings together the equipment builders, gear manufacturers and an interested and quality audience. For many exhibitors, Gear Expo is their marketing program.

For attendees it is a very efficient way to see the industry innovations and the best in technology. Attendees can see a variety of suppliers in a day or two. Equally, those who need to buy gears and related products find high-quality suppliers at the show. Gear Expo is a major networking event for the gearing industry.

Attendees in 2009 came from 38 states and 30 countries; truly, Gear Expo is the place to be for anyone in this industry. Four of every five attendees, 82 percent, make the final decision or directly influence the final decision on purchases.

Finally, the schedule of education events at Gear Expo adds value for many attendees. In addition to the heat-treating programs from ASM, AGMA will offer:

• ***AGMA's Fall Technical Meeting***—This year's Fall Technical Meeting will feature more than 25 presentations from leading experts in the industry during an expanded schedule for 2011.

• ***How to Organize and Manage a Failure Investigation***—Robert Errichello will present proven techniques for organizing and managing a failure investigation that will maximize your chances of identifying the failure mode and the root cause of the failure, and help you recommend repairs or improvements to prevent future failures.

• ***Training School for Gear Manufacturing***—Dramatically improve your knowledge and productivity through this classroom-only version of AGMA's popular "Basic Course" of the Training School for Gear Manufacturing. This course, in three, half-day sessions, will give in-depth training in standard nomenclature, gear involute geometry, inspection procedures and much more.

• ***Why Bearings Fail***—Understanding the causes around bearing failures is critical to avoiding down time and improving performance. This course will be a basic primer for addressing the most common causes of bearing failures in gearboxes and related equipment.

• ***The Solutions Center is back***—offering free, short presentations on the show floor from exhibitors plus keynote presentations from leaders in the industry.

In 1987, we took one small hall of the Cincinnati Convention Center; this year, we are taking the entire Convention Center with an expected 200 exhibitors in over 40,000 net square feet of exhibit space. We will be back in Cincinnati from October 30 (for the Fall Technical Meeting) and November1–3 for the Gear Expo.

I'll see you there!

Joe T. Franklin, Jr., President
American Gear Manufacturers Association

Profiling Grinding Technology

CORVUS CPP 1100 BOASTS SHORTER SET-UP TIME, MORE MACHINE INTEGRATION

The Corvus CPP 1100 was created to meet the demand for large or coarse pitch hob manufacturing (courtesy of Schneeberger).

In the mid 1980s, Schneeberger played a leading role in the development and construction of the first CNC-controlled profile grinders. Today, the company remains on the forefront of grinding technology with its latest entry, the Corvus CPP 1100 mm profile grinding machine. Created to meet the demand for large or coarse-pitch hob manufacturing, the Corvus CPP 1100 is designed to accept up to 20" diameter coarse-pitch hobs with maximum cutting depth. Form grinding is achieved by profile dressing the wheel to the correct configuration and grinding the axial and radial relief with the profiled wheel. This process is better known as pencil grinding, where the grinding wheel has the form of a large pencil to allow the cutting edge to be relieved with the slanted spindle head. The powerful direct drive grinding spindle offers 30 hp (45 hp at 150 percent duty cycle) to meet the requirement of the large contact area the formed grinding wheel has to grind. "In the past, the customers have used old CAM-driven machines and were grinding dry," says Rolf Herrmann, general manager at Schneeberger. "The new CNC machine has a much shorter set-up time, grinds more efficiently using coolant and, with the wheel changer, offers all operations necessary to grind the tool in one set-up including roughing, pre-grinding and finish grinding. Also, dressing the finishing wheel to the required form is part of the integrated manufacturing process."

Onboard dressing, tool support, hydraulic tailstock, probing and more useful application solutions can be offered or are standard with the machine. To utilize full production capability and have the ability to rough grind and finish grind as well as maintain a useful wheel diameter, the machine is equipped with an automatic eight-station wheel loading system. The new wheel is loaded—including the correct coolant nozzle manifold—for best grinding and coolant conditions. For rough stock removal, the CBN-plated grinding wheels can be employed while finishing is done with vitrified wheels. The Schneeberger software *Quinto5* rounds out the latest in hob manufacturing capability. "The software offers a very graphical programming environment geared to the hob grinding application for the Corvus C500 face grinder, Gemini GHP hob profiler or the Gemini DMR hob sharpener," Herrmann says. "All machines use Fanuc 310 control technology with linear motor drive systems for the axis drives."

Schneeberger offers a range of machines for the cutting tool industry including the Corvus C500 face grinder for coarse-pitch hobs up to 70 hp, max wheel and tool size up to 20" including integrated dressing, O.D. grinding spindle, gap eliminator for dressing and fully-automated balancing of the wheel. The Gemini GHP is specifically designed for profiling

continued

smaller hobs as well as spiral hobs with extreme helix. Other applications Schneeberger machines are known for include bevel gear cutters, stick blade grinding, sharpening or manufacturing with a large geometry database. Schneeberger offers other small and large machines for spline grinding, manufacturing of any cutting tools as well as tool room machines for re-sharpening. In the near future, the company will be offering CNC capability for bevel gear tooling. Also, stick blades of all forms can be ground and offered to the market.

Schneeberger's catalog includes CNC grinders Corvus, Gemini, Sirius, Norma and the new Aries5 with capacities of up to 26 kW. The Schneeberger concept conforms to a movable or fixed column construction, depending on the machine design and application—the grinding heads with double spindle are rotatable up to 370 degrees. All CNC machines can be equipped with options for measuring and inspection or with loading robots. The Aries line is an alternative to the five-axis technique for regrinding. A simple control takes care of the helical interpolation whilst manual settings are assisted by an ergonomic mechanical design. Large end Machines come with strokes of up to 3,000 mm for round, spiral and flat broaches, cutter heads, splined shafts. The company also offers CNC machines for the production of exacting ground parts outside the tool domain.

"One special feature Schneeberger offers is in-process quality control with automatic compensation inside the machine, a feature using a camera set-up checking profile accuracy and adjusting dressing cycles to compensate for thermal shifts or wheel wear," Herrmann says. Thanks to this new machine technology, Schneeberger has dramatically improved set-up times, accuracies and surface finish for their customers as well as the manufacturing time for a hob. "We have gotten excellent response on the machine's design, rigidity, capability and ease of operation."

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Gleason

INTRODUCES ROLL TESTING AND SINGLE-FLANK TESTING OF BEVEL GEARS

Now roll testing and single-flank testing of bevel, angular bevel and parallel-axis gears with diameters up to 2,500 mm can be completed much faster with the introduction of the Gleason 1000T, 1600T, 2000T and 2500T Universal Gear Testers. The new series of Gleason six-axis fully CNC Universal Gear Testers can perform the full range of pattern checking and single-flank (SFT) motion transmission error testing on any bevel gear set with the widest possible range of angles. Unlike competitive products, the Gleason Universal Testers are equipped with a CNC-controlled pivoting gearhead that can swing from 0 to 180 degrees, thus enabling it to precisely mesh pinion and ring gears of varying angles and types, and even cylindrical gears—both parallel axis and off-angle.

In addition, the new design also makes day-to-day operations such as setup, manual or full automatic operation and "jogging" much simpler and more intuitive—despite the added complexity of testing an angular gear set. The Gleason testers enable the operator to relate to the machine through virtual gear set axes that are easier to understand and work with, rather than typical machine axes that can be difficult to translate into angular motions. Siemens 840D or Fanuc 31i CNC controller and intuitive Windows-based Gleason inspection software not only support this unique virtual capability, but also make more routine functions like manual roll checks, cycle set-up and V & H testing that much easier for the operator.

The Gleason Universal Testers also are designed and built for rugged, dependable operation and the day-to-day rigors of the production floor. They feature, for example, a particularly rigid and thermally stable granite machine base frame, and an advanced light curtain to protect the integrity of the work zone. They also combine an extremely compact size with a particularly large and accessible test area. Load/unload is manual, so the open test area and long axis travels make it easier for the operator to quickly load and unload gears of any size and geometry.

First of the new series, a 1000T, is already in production at Sauter Bachman, a Netstal, Switzerland based leader in the production of high quality gears and gearboxes for aircraft, automotive, railway and textile industry applications. The 1000T, 1600T, 2000T and 2500T (for workpiece diameters ranging from 1,000 mm, 1,600 mm, 2,000 mm and 2,500 mm respectively) help complete the Gleason family of highly popular testers that also includes the 360T, a similar universal tester for gears up to 420 mm in diameter, and the 600HTT Turbo Tester, for bevel gears up to 600 mm in diameter.

For more information:

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Rochester, NY 14692-2970
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Fax: (585) 461-4348
www.gleason.com

MAG

INTRODUCES VTC WITH GEAR CUTTING CAPABILITY

The first technology advancement for large vertical turning centers (VTCs) resulting from MAG's recent acquisition of Modul, a gear manufacturing equipment specialist in Germany, introduces integrated gear cutting capability on a VTC 2500H for workpieces up to 2,700 mm diameter.

The new multi-axis machine combines gear hobbing and milling capabilities with the VTC's wide range of turning, milling, drilling, threading and contouring capabilities. It can turn blanks and subsequently cut gear teeth, all in one setup. The machine's ability to execute rough turning of the outside diameter and gear cutting on the inside diameter in one setup results in higher quality gears. It also eliminates a separate gear cutting machine and reduces labor, part transfers, setups, work-in-process and plant footprint/overhead.

Gear production capabilities include hobbing, form milling (of external and internal gears), turn-milling of gear teeth with carbide end mills, and chamfering/deburring. This system is suitable for the part making requirements of the wind and power generation industries, mining equipment, marine and other heavy machinery. The hobbing module is fully integrated into the VTC to allow turning, milling and hobbing or form milling of internal and external geared slewing rings. Two different heads are available with 36 or 46 kW power. They can hob modules up to 24.0 mm or form-mill teeth up to module 28.0 mm on parts with a maximum face width of 600 mm.

The functionality of the gear manufacturing software is fully integrated into the turning center's control with one common HMI, so programming gear cutting operations is simple and intuitive. Features include workshop-oriented dialog for input of all geometric and technical parameters for the workpiece and hob.

In addition to gear cutting, MAG's VTCs can perform standard and hard turning, as well as live-spindle machining, contouring and part probing operations on multiple sides and the full diameter of a part. Customer-driven features include use of standard modular tooling, such as KM 80 and Coromant Capto solutions, as well as automatic tool changers; green design with minimal hydraulics; minimal or flat-floor foundation requirements for many models; and innovative chip management and removal that minimizes operator intervention. A full machine enclosure meets CE specifications and ensures a dry floor environment.

MAG offers horizontal turning centers, inverted-spindle VTCs and vertical twin-table turning centers equipped for integrated gear processing, along with a full range of gear cutting tools, including solid hobs and milling cutters, heavy-duty hobs, inserted-blade hobs and milling cutters, as well as carbide end mills. In addition, MAG produces a line of dedicated gear production machines and related technology at its facility in Chemnitz, Germany.

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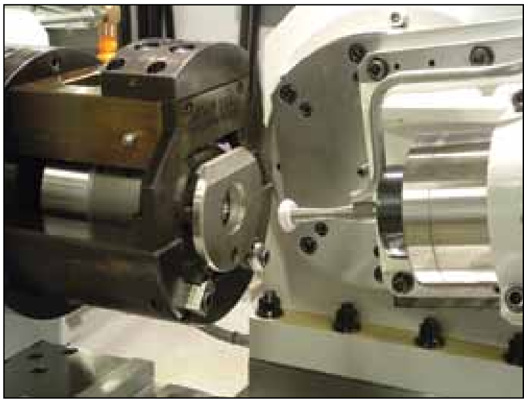
3940 Olympic Blvd. Ste 500
Erlanger, KY 41018
Phone: (859) 534-4600
www.mag-ias.com

Drake Manufacturing

ENJOYS EXPORT SURGE WITH THREAD GRINDER

Drake Manufacturing Services Co. is enjoying a surge in exports to Chinese manufacturers, with orders in hand for more than 25 machines. This latest machine is a GS:TI-LM 200, a four-axis internal thread grinder with a 200 mm maximum grinding length and 350 mm swing over the table. It was tooled and programmed to grind recirculating ball nuts used in the power steering systems of heavy-duty commercial trucks. It is the second machine delivered to this customer, and more than doubles their production capacity. The steering nut weighs over nine kg and is manually loaded into a custom workholding fixture. Robot loading is also available. Cycle time for the internal thread grinding process is approximately two-minutes. The GS:TI-LM 200 is equipped with linear motors for maintenance-free operation. No ball screws and 0.05



micron scales mean accurate leads and helical path. A CNC rotary diamond dressing unit mounted on the workhead can generate wheel forms either by contouring or plunge dressing with a form roll. The wheel spindle is equipped with a power helix that can grind lead angles of +/– 12 degrees. The machine also has a safe shutdown feature that prevents crashes in the event of a plant power loss. Each Drake machine is delivered with *Part Smart* programs. The operator needs only to input part specific variables into the control to run parts. No customer programming is required and the customer can change from one part to another in minutes.

For more information:
Drake Manufacturing
4371 North Leavitt Road
Warren, OH 44485
Phone: (330) 847-7291
www.drakemfg.com

Contour Automatic Series

ENABLES HIGH-SPEED, IN-LINE INSPECTION

The new opticline CA614 non-contact shaft measuring system is designed for fast, easy, economical and robust submicron measurement of all contour features on complex shaft-type parts up to 600 mm in length, 140 mm diameter. The stroke is driven by a motorized tailstock, freely positionable over the full holding range, permitting easy, fast, and precise workpiece loading and holding. The value of the Contour Automatic series has been proven in automotive and aerospace applications, including carrier shafts in aircraft engines, drive parts in helicopters, and various rotor parts. In these instances, the system inspects length, geometry (angles, radii, etc.), form (roundness, run-out) and diameter in rapid cycles. Ease of operation and programming, plus the system's unique flexibility to measure any num-

continued

ber of different part variations with little or no changeover, delivers maximum return on investment to manufacturers.

Depending on the measurements required, standard measuring functions can be quickly combined into one optimized measuring run. Examples of combined dynamic functions include: size, roundness, run-out, minimum size, maximum size, etc. and can be computed based on a single contour record. The measurement system monitors itself for negative ambient influences, providing automatic active temperature compensation before every measuring run. Contour Automatic hardware and software are optimized for use in production environments to provide maximum reliability over long periods of time. The optical sensor operates in an environment that protects it from part wear and collision, while delivering short cycle times, accuracies into sub-microns, and computed measurement results within a few seconds. Once a new part is loaded into the system, a simple scan is performed that within seconds commits the part to memory. User-friendly software then allows the user to intuitively create new measuring programs within minutes based on the preliminary scan.

The creation of part measuring programs, data display, and further processing of the measurement results takes place through an easy to understand graphical user interface (GUI). The user interface also provides the capability to customize inspection reports for satisfying plant and customer demands. In operation, the part is loaded into position then automatically scanned optoelectronically utilizing the silhouette principle. The workpiece contour can be evaluated quickly and precisely utilizing the high resolution associated with each measured datum (statically or dynamically). A flexible, pneumatic driving fixture provides defined workpiece driving and easy workpiece handling.

For more information:
Hommel-Etamic America Corp.
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Phone: (248) 853-5888
Fax: (248) 853-1505
www.jenoptik.com

Techniks Boring Tools

Maximize Productivity

Pinzbohr High Precision Modular Boring Tools from Techniks offer accuracy, rigidity and repeatability. All Pinzbohr boring tools are built with the quality, reliability and ease-of-use features machinists need to maximize productivity. Innovative features include the AccuSET dial for easy insert adjustments in increments of .0001". The SpeedSET feature enables synchronized setting of roughing heads without having to adjust individual slides, or having slides slip out of position during adjustment. The SafetySET feature prevents the operator from accidentally damaging the lock screw by over-tightening. The offset-center tapered lock screw coupling system provides maximum tightening force between the tool and the toolholder, while still allowing heads to be changed in spindle. Minimum boring size is .315". Maximum boring size is 19.685”. Also available are the popular BohrSTAR kits, sized in ranges from .2362" up to 8.2677".

continued

For more information:
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9930 East 56th Street
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End Mills

IMPROVE CHAMFERING PERFORMANCE

A new family of Iscar end mills extend the benefits of tangential Sumomill inserts to chamfering, countersinking and face milling. Featuring the proprietary Sumo Tec coating treatment, Iscar's Sumomill creates a fundamentally stronger cutting platform, which leads to longer edge life and elimination of fracture failures at higher cutting rates, even in unfavorable conditions. Sumomill tangential end mills, all four-flute in 12 mm diameters, come in 30, 45 and 60 degree geometries. In face milling applications they can run at very high table feeds due to their small diameter and high flute density. The proprietary Sumo Tec surface treatment produces a much smoother, more lubricious surface on the insert, with fewer thermal stresses and stress factors than would be found in an untreated plain PVD or CVD coating. This leads to cooler cutting (heat is a key enemy of machining inserts), less friction and much reduced risk of insert rupture. On average, SumoTec coated tools have proven to literally double throughput and reliably extend insert life by 40 percent versus identical but untreated inserts, according to the company.

For more information:
Iscar Metals Inc.
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Phone: (817) 258-3200
www.iscar.com

BIG Kaiser

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BIG Kaiser's 319 SW series is an exceptionally rigid, new style, twin cutter rough boring system. These corrosion-resistant boring heads offer a unique feature—by simply reversing the insert holders, users can switch between balanced and stepped cutting. The Kaiser 319 SW series will replace the older 314 RW series. By not introducing additional components or having to manually adjust insert height for different roughing methods, the 319 series reduces setup time and eliminates the need for a pre-setter due to fixed tool heights and a diameter scale on the tool. "These tools are fully compatible with our existing KAB and CKN modular systems and offer much larger variety, versatility and capability to our rough boring program." says Jack Burley, BIG Kaiser vice president of sales and engineering. "Although the current 314 series will be phased out, our customers will maintain the same roughing methods from one head using the 319 program, but now with greater ease of setup and higher performance."

The tools, with a diameter range .787"–8.000", utilize standard carbide inserts (CC, SC, and SP) and provide coolant-through direct to the cutting edge to improve chip evacuation and lengthen tool life.

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NUM Gear Shaping Software

ACCELERATES WIND TURBINE GEAR PRODUCTION

Advanced gear shaping software and motion control hardware from NUM has been used to bring an old

manually-controlled gear shaper machine up to modern day control standards. The five-axis Stanko machine is now controlled by a NUM Axium Power CNC system, using electronic gearbox techniques to synchronize the rotary cutter, gear blank and stroking axes. It also incorporates an innovative programmable replacement for the stroking axis, which reduces product changeover time from hours to minutes—significantly improving productivity. The gear shaper was bought by DePe Gear Company, which specializes in the design, manufacture and refurbishment of gears and gearboxes for a diverse range of industrial and commercial applications, including the steel processing, rail, mining, quarrying and aeronautical industries. The company operates a considerable number of gear cutting, shaping and grinding machines at its Stoke-on-Trent manufacturing facility and is no stranger to NUM as three of its current gear cutting machines are equipped with NUM Axium CNC systems and *NUMgear* software.

In this particular case, DePe Gear Company purchased the Russian-built Stanko gear shaper initially for manufacturing large internal gears for the wind turbine industry, and subsequently commissioned machine tool engineering company Euro CNC to carry out the necessary refurbishment work. Euro CNC specializes in retrofitting, rebuilding and upgrading machines. This often involves equipping manual machines with partial or full CNC systems, and the company consequently maintains a close working relationship with NUM, providing it with access to the latest CNC technology, control software, digital drives and motors. In recent years, Euro CNC has built up considerable knowledge of machine tools for gear production, and nowadays handles a wide variety of gear hobbing and shaping machines.

Euro CNC quickly ascertained that although the machine was fully mechanically serviceable, it would benefit from being equipped with new motors and drives, including high performance digital units for all axes, together with a CNC system for operational flexibility and a customized HMI to replace outmoded mechanical switchgear.

Traditionally, gear shaping machines employ a complex cam-driven 'nodding' axis arrangement to move the cutting tool up and down the gear blank as it is cut, the stroke of which needs to be synchronized to the rotation of the tool and the blank. This approach suffers from numerous disadvantages: it can involve up to three axes of movement, each subject to error, and is extremely difficult and time-consuming to set up, which does not sit well with the fast and flexible changeover requirements of modern manufacturing. Euro CNC consequently decided to develop an entirely new form of stroking axis, based on a fully programmable linear actuator. The end position, length and speed of the stroke can be freely changed under software control.

In addition to the stroking axis, the gear shaper has three rotary axes—to rotate the cutting tool and the gear blank—and to retract the cutting tool on the up stroke together with a linear positioner based on a motor and ball screw, which drives the gear blank to the cutting tool. All of these axes are controlled by NUMDrive C servo drives and NUM brushless motors.

Euro CNC chose to use a NUM Axium Power CNC system to control all five machine axes, networked to a

continued

NUM industrial PC and a large touch-sensitive screen. The software includes NUM's powerful *NUMgear* package, but in this instance it is used mainly to provide the electronic gearbox functions for synchronizing the cutting tool rotation, gear blank rotation and linear stroking axes. The HMI for the gear shaper machine is primarily created by a special version of NUM's PC *ProCam* software, which was jointly developed by Euro CNC and NUM's USA facility specifically for this type of application. The software combines a highly intuitive graphical user interface using common gear shaping terminology with a 'conversational' style of programming, enabling operators who are not familiar with CNC-based machines to become proficient very quickly. The refurbished Stanko gear shaper was recently installed at DePe Gear Company's Stoke-on-Trent facility and aside from a few minor initial issues has performed flawlessly.

Nigel Parker, Technical Director of DePe Gear Company, points out that, "We are using the gear shaper for a variety of internally cut gears, including spur gears for wind turbine generator gearboxes and a variety of splined gears. Although it is too early to provide quantified data, we are definitely seeing a reduction in setup and operating times. Like our other CNC machines, the most significant benefit comes from the sheer versatility of this all-digital approach, which enables us to switch freely from manufacturing one type of gear to another under software control. Machine operators no longer need to laboriously count the number of teeth being cut, but simply push the appropriate button on the menu, which helps maximize throughput."

According to Tim Clarke, Director of Euro CNC, "We have worked with NUM for about five years now, and have found their CNC products to be extremely reliable. We also benefit from excellent technical support from their U.K. facility, and have recently experienced a similar level of backing from NUM USA. So far, we have installed *PC ProCam* on some 25 machines—mostly gear hobbers rather than gear shapers—and have been delighted with the positive feedback from customers."

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An Emphasis on Accuracy

MEETING THE MANY CHALLENGES OF LARGE GEAR INSPECTION

Matthew Jaster, Associate Editor

The large contracts in gear manufacturing are vital to a company's bottom line. Ask any gear manufacturer involved in the mining, tunnel boring, wind or heavy equipment industries

Options such as roughness measurement and overheating detection during the grinding process offer additional practical test procedures on Klingelnberg inspection machines (courtesy of Klingelnberg).

and they'll agree. In order to promote longevity in this market segment, these large gears cannot fail. While big gear manufacturing hints at larger financial opportunities, it also means that the quality/inspection has to be unsurpassed. Companies like Wenzel, Gleason, Klingelnberg and Hexagon Metrology are consistently improving their quality/inspection capabilities with this in mind.

Inspection Machines to Fit Your Needs

Is the equipment reliable? How does the service and support measure up? What new features have been implemented that will interest potential customers? These are all valid questions in regards to inspection equipment. Reliability, productivity and precision are all critical components to Wenzel's product offerings. "In an industry such as gear metrology there is no room in the market for a product whose aim isn't to be the most accurate system," says Elliott Mills, gear product manager at Wenzel America. "A Wenzel machine is a robust and reliable system due to its granite construction and air bearing design; this is a wear-free system. Our large capacity hydrostatic rotary tables are also wear-free. We eliminate the fail modes as opposed to managing them."

Cycle times for Wenzel machines are not an issue, according to Mills, as the company strives to achieve its productivity outside of the inspection cycle. "We look to eliminate changeover time between cycles using innovative part fixturing systems and offline programming. A machine should not sit idle while there are parts awaiting inspection."

This year, the company has a new machine being delivered to a customer. The WGT-4000 machine is capable of measuring four-meter gears with two meters between centers and a two-meter measureable face width. "We will be delivering a WGT-4000 to a very well-known German manufacturer of grinding equipment, "Mills says ,"Additionally, we have some exciting software developments that will be on display at the EMO and AGMA shows later this year."

Before releasing inspection equipment, all Klingelnberg gear testing machines undergo extensive internal acceptance testing. "This ensures a high degree of uniformity among measurement results across all machine models," says Gunther Mikoleizig, head of product management gear inspection machinery at Klingelnberg. "This is all the more important for companies that intend to use a combination of gears produced internationally. Highly durable machine components and state-of-the-art drive and measurement systems in all modules provide the basis for such results."

While the capability of testing toothed gear flanks has been available for some time, the option for checking overheating during grinding is now available on Klingelnberg machines. "For large gears, this obviates the need for additional time-consuming test operations on other devices," Mikoleizig says. "Another improvement is the new probe calibration program Stylus Manager that allows probe elements to be selected individually and calibration procedures to be performed with the workpiece mounted."

Gleason Metrology Systems (GMS) machines start off with the traditional foundation for the majority of inspection machines, a granite base. "But not just any granite base," says Dennis Traynor, sales manager at GMS. "The material supplied by GMS has a very small, very dense, crystalline structure to avoid humidity absorption. To this base we add a large-diameter, high-workpiece-weight-capacity direct drive rotary table. Finally, we add our X-Y-Z compound axes constructed of Meehanite cast iron, all driven by linear motors for smooth, reliable, trouble free operation. All these mechanical subsystems are then controlled by *GAMA* (*Gleason Automated Measurement and Analysis*), an object-oriented (native Windows-based) suite of gear and gear tools applications software.

GMS is continually improving its product offering, and this year marks the introduction of the new GMS series five-axis machines that will debut at EMO in Hanover, Germany this September. "GMS' five-axis machines traditionally support the large gear sector, in particular wind energy applications. We will have on display a 2000GMS model which will display all the features we have previously mentioned as well as some unique work-holding strategies," Traynor adds.

Updating Inspection Capabilities

Klingelnberg began offering testing machines for large gearing back in the 1980s. Since then, the company has continued to update its inspection capabilities based on market demands. Software is one area that has seen continuous improvements. "The current software modules are designed to Microsoft Windows standards offering a familiar interface, and the use of graphical symbols makes the interface essentially self-explanatory and shortens the learning curve for new staff," Mikoleizig says. "By making simple, specific changes to an existing measuring procedure, new procedures can be created quickly and easily. The user interface can easily be switched to another language setting and even another character set—such as Chinese."

Klingelnberg's software department provides comprehensive, continuous service to its customers. "Alongside their theoretical gearing knowledge and participation in working groups for the development of new standards, our experts also update the software to keep pace with continuous advances in hardware, including the latest operating systems," Mikoleizig says.

Being a modular program, Gleason's *GAMA* software utilizes preconfigured tolerance tables for industry's most popular industrial standards. *GAMA* also supports most of the world's leading gear OEMs with custom analysis modules particular to their proprietary gear analysis standards. "We have incorporated many modern-day features like digital images embedded in the software to confirm proper part and orientation, voice mail messages embedded in the program for operator-to-operator or manager-to-operator communications to convey the latest updates in part geometry or fixture changes, etc. Finally we have incorporated a webcam into the remote operator pendant to allow for still image capture, or live video streaming of the inspection process, convenient for communicating issues to engineer-

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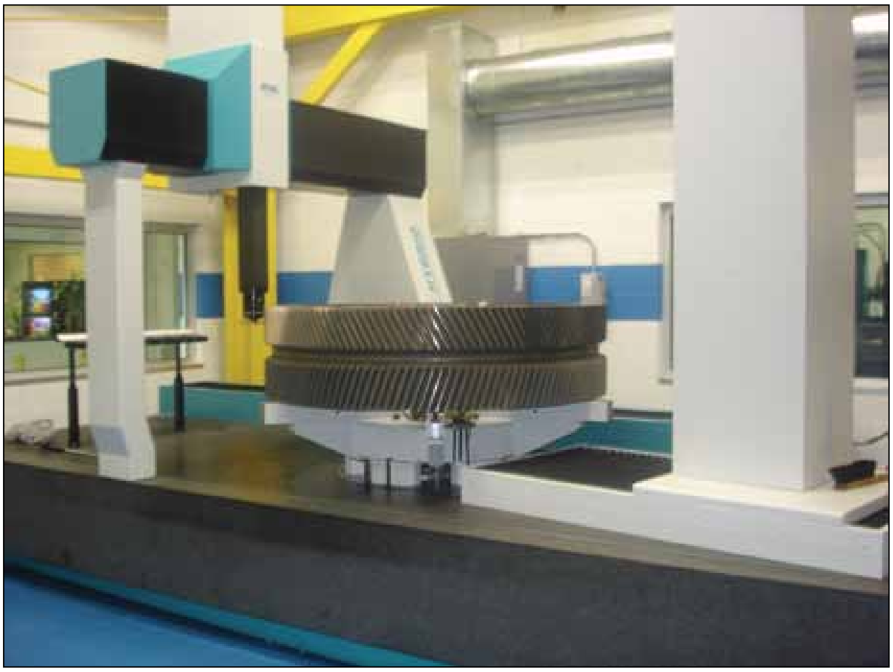
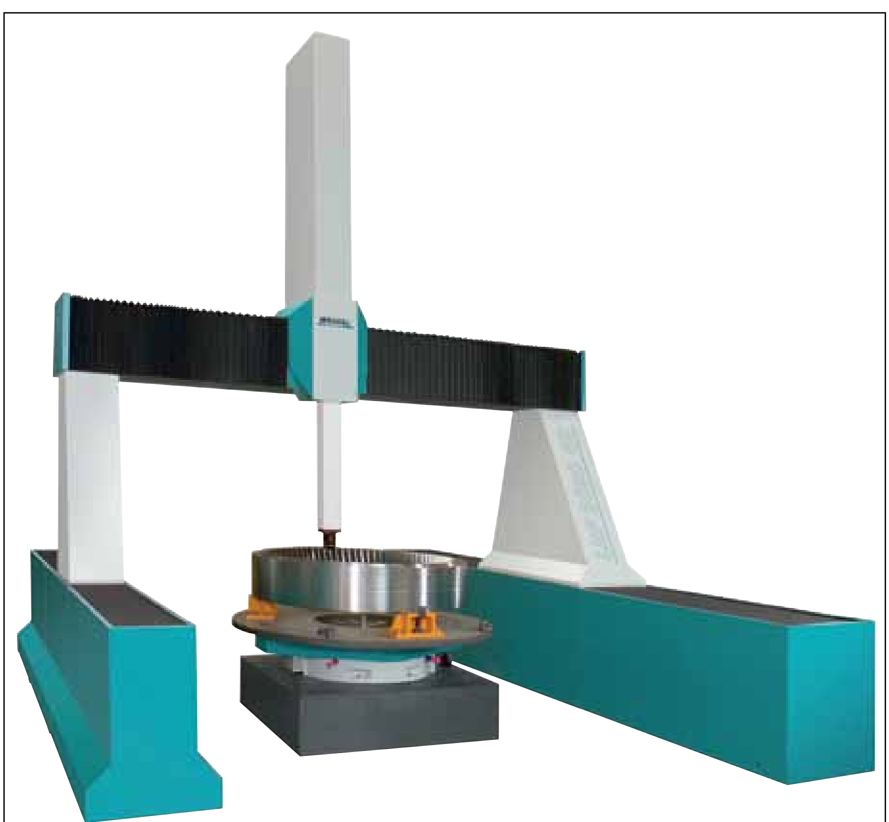
ing, customers, or other quality staff," Traynor says.

In addition, *GAMA* utilizes a very intuitive operator interface which contains both text and graphical information about the part and its setup, and has a logically menu driven structure to support a feature-rich part programming routine. "We also offer features like "Center Part" that allow the machine to probe the workpiece radially and axially at its journals, calculate the eccentricity of the rotary axis, and then allow the software to guide the operator to easily true the part. Part data can be recalled at any time and is organized in a variety of ways: by part number, by part type, by customer, etc. *GAMA* also stores part inspection output data in either pdf file formats or SVG (Scalable Vector Graphics), an HTML style output, small in file size and easily emailed and viewed with no special software," Traynor says.



Reliability, productivity and precision are all critical components to Wenzel's product offerings (courtesy of Wenzel).



Wenzel has two distant product lines for large gear inspection, traditional gear inspection machines and CMM gear inspection machines (courtesy of Wenzel).

"Any internet browser will open the file and display the measured results. Additionally, actual raw data can be recalled and re-analyzed under different standards or parameters without having to re-inspect the gear physically. *GAMA* also supports the world's languages. Either the interface or just the output charting can be easily converted to over 25 different languages used worldwide, supporting today's global sourcing strategies."

Large Gear Inspection: A Cheat Sheet

While many companies struggled during the recent economic crisis, Wenzel America's staff actually grew. "We added additional applications and service support through both strategic hiring and cross training many of our service and applications engineers to best match our capacity to our demand. We have numerous customers who have different Wenzel products and it allows us to be flexible with staffing and to better cover a large geographic area," Mills says.

Wenzel can handle measurements up to five meters in diameter with what are standard products, not tech specifications or a design concept.

In the large gear market, Klingelnberg offers gear testing machines for large workpieces in the range of 1,800 mm, 2,800 mm, and 3,800 mm in diameter, with machines for 4,000 mm and 6,000 mm coming soon. The permissible workpiece weights are in the range of 8,000 kg, 15,000 kg, 20,000 kg and 40,000 kg. Workpieces can be up to 3,000 mm long with a measuring range of up to 2,000 mm in one fixture.

GMS offers four different machines that fall into the large gear sector. They are 1,000 mm, 1,500 mm, 2,000 mm and 3,000 mm capacity systems, with the latter three being five-axis systems. The model numbers (1000GMS, 1500GMS, 200GMS and 3000GMS) denote the maximum diameter work piece. "All systems are available with extended Z axis or tail stock travels to accommodate taller workpiece or tooling build up geometries. Machines of this size have a myriad of fixturing and tooling options, which are typically identified during our presales interview process. All of our machines also have the capability of prismatic geometry inspection which is very typical with today's wind energy components," Traynor says.

The PMM-G, a custom configured product built by Hexagon Metrology GmbH, is suitable for the extremely large gears used in the windpower industry, for power generation, and power transfer products. The system is not a single-purpose inspection station just for gears, but a fully capable coordinate measuring machine that can be used to inspect many other kinds of large machined assemblies, such as gearboxes and engine blocks. "The PMM-G represents the ultimate in Leitz large-scale gear inspection," said Pete Edge, product manager, Leitz products. "Unlike other gear inspection products, it does not require a rotary table, which makes part loading and unloading much easier. It also allows a maximum part weight of 15 metric tons."

The PMM-G is available in 55 standard measuring sizes ranging from 3,000 mm x 2,000 mm x 1,200 mm to 7,000 mm x 4,000 mm x 3,000 mm. The maximum part load is 15 metric tons. Gear types inspected include cylindrical gears (spur, helical, double helical, internal and external spline, internal and external clutch, gear segments and gear racks) plus straight, spiral and hypoid bevel and crown gears. Gears can be evaluated to all major standards including DIN, ISO, AGMA, ANSI, JIS, CNOMO and CAT.

Additional Measuring Tasks

For a number of years, Klingelnberg machines have also been qualified for additional measuring tasks beyond gear measurement. "The ability to carry out dimension, form and position measurements on drive components provides the conditions for nearly complete measurement," Mikoleizig says. "Options such as roughness measurement and overheating detection during the grinding process offer additional practical test procedures. The expansion of the model series for workpieces up to 6,000 mm in diameter and weights of up to 40,000 kg will enable a qualitative assessment of additional drive components in the future."

Milwaukee Machine Works recently purchased a Leitz PMM-G ultra-accurate CMM from Hexagon Metrology that is tailor made for large-scale machined components. "The primary capability for our application was size in conjunction with accuracy," says Mike Manna, general manager at Milwaukee Machine Works. While the company is machining most of the exterior parts for the larger geared products, it does not inspect gears at this time. "We expect to define

continued

whether there will be a market for gear inspection in this area," Manna says. "This will drive the decision on purchasing the gear inspection software for the machine."

The challenges with inspecting large gears include standard issues associated with handling any large parts made to precision tolerances. Manna adds, "This requires a clear understanding of the equipment that will be used to move the parts and address these issues ahead of time. Once presented for inspection, the appropriate probes and appropriate temperature control are critical to assure correct measurement."

Complete inspection of gears as large as one meter in diameter now can be performed up to 25 percent faster with the Gleason 1000GMS Analytical Gear Inspection System (courtesy of Gleason).


Extensive internal acceptance testing is performed before machines hit the market (courtesy of Klingelnberg).

As a rule Hexagon is a cooperative, professional supplier who understands the urgency of repairing a piece of manufacturing equipment that goes down. "They will do what it takes to support us. They also have a comprehensive training program and users group to further our knowledge and understanding of the inner workings of the equipment we use," Manna says.

With the Leitz probe head and *QUINDOS* software, Leitz measuring machines from Hexagon Metrology are increasingly used to measure all kinds of complex toothed gears. Compared to conventional gear testers, these machines can inspect all kinds of toothed wheel work and cutting tools without the need for a rotary table. This translates into higher system accuracy, above all with heavy wheels, simple mounting without guides and centering devices and lower capital and maintenance costs.

Wenzel has two distant product lines for large gear inspection, traditional gear inspection machines and CMM gear inspection machines. "Of course, the two meet with our hybrid machine which is essentially a combination of both. Wenzel offers dedicated gear inspection machines up to five meters as a standard product. Our LHGear machines are traditional CMM frames coupled with a 4th rotary axis and are the perfect solution for customers who have prismatic inspection needs in addition to large gears."

Mills notes that the company boasts a design team that takes on special projects where customers have inspection challenges that are outside of the capability of standard products offered in the industry.

Inspection Wish List

So where do these inspection professionals see this market segment heading in the future? "We are constantly looking at many technologies to offer more beneficial features and options to our customers," Traynor says. "We continuously look at new and innovative ways to inspect the finite geometry of gears. We also evaluate sensor technologies, controls technologies and mechanical subsystems to improve accuracy and uptime requirements.

"Given Gleason's strong after sales strategies and the robust features of *GAMA*, we are capable of supporting our systems virtually anywhere in the world. We have several systems installed in the far reaches of the gear manufacturing community and these machines are fully operational and fully functioning each and every day. In addition to our online strategy, Gleason has local field service engineers located in virtually all of the world's established markets and many of the emerging markets."

Mikoleizig adds, "Modern gear testing machines are equipped with temperature-neutral measurement systems and use appropriate sensors to monitor the temperature of individual modules. All guides and measurement systems are equipped with covers. The materials used in the measuring machines are suitable for use in a production setting. The use of measurement technology on the production floor is aided by the overall improvement of conditions in modern production facilities."

"I think we will continue to see traditional gear inspection machines in labs and on floors in the future," Mills says. "Demands for longer life of gears with continue to tighten tolerances and requirements for elemental gear inspection will continue to grow, parts that weren't required to be inspected five years ago are today. Parts that were not feasible to inspect in the past can be inspected with innovative thinking, these barriers will continue to be eroded."

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With the Leitz probe head and *QUINDOS* software, Hexagon Metrology can offer gear measuring capabilities (courtesy of Hexagon Metrology).

Wind Turbines: Clean Energy, but Energy-Efficient?

Jack McGuinn, Senior Editor

Green Energy (stock photo by Dimitri).

It has often been reported that when the legendary—but true-to-life—bank robber and certified colorful character Willie Sutton was asked why he concentrated on robbing banks, his simple, reasonable reply was, "Because that's where the money is." But that's not quite accurate. What he did say—in an autobiographical sketch—was "Go where the money is. And go there often."

So it is no surprise that the manufacturing sector—the products they make and how they make them—is a prime target for efforts—both governmental and private—to enhance energy efficiency, reduce consumption and maintain or exceed current productivity levels.

But beyond the big-picture state of energy consumption in America, we talked energy efficiency with some major players in the lubricants industry—but with a focus on their products' impact regarding energy efficiency of gears and gearboxes in wind turbines—a much narrower universe than gear drives in general. We also gathered some insights from an expert in the gear/wind turbine sector relative to gears and gearboxes (*see sidebar*).

A question that immediately comes to mind is, what, in general, is happening with lubricants and their role in energy efficiency in a wind turbine gearbox application—and the gears that go in them?

"Most gearboxes in wind turbines today are designed around a single planetary and two parallel-shaft helical gears," explains Felix Guerzoni, product application specialist and team leader for Shell Global Solutions U.S. Inc. "The process of gears meshing and un-meshing results in a combination of rolling and sliding contacts. With spur and helical gears, much of the contact is rolling contact and as a result the efficiencies are quite high. For worm gears, where the majority of the contact is sliding contact, these tend to be much less efficient. It is for this reason that PAG-based (poly-alkylene glycol) gear oils with lower coefficients of friction relative to PAO (poly-alphaolefin) are favored for worm gear lubrication. The efficiency losses in a spur or helical gear

Table 1-Types of Gears.

Types	Gears	Position of the shafts	Tooth Flank Contact	Gear Components	Type of Movement	Sliding Percentage (%)
	Spur Gears	Parallel	Line	Cylinders	Rolling and Sliding	10 - 30
	Bevel Gears	Intersecting	Line	Cones	Rolling and Sliding	20 - 40
	Crossed Helical Gears	Crossing	Point	Cylinders	Increased Sliding	60 - 70
	Hypoid Gears	Crossing	Line	Cones	Increased Sliding	60 - 70
	Worm Gears	Crossing	Line	Cylindrical and Globoid Element	Mainly Sliding	70 - 100

(Courtesy Klüber)

Wind Turbine Gearing: Moving Towards Energy Efficiency?

Following is a Q & A with Charles D. Schultz—PE and chief engineer with Beyta Gear Service (*www.beytagear.com*) and Gear Technology technical editor.

What are the most common issues regarding gear/gearbox energy efficiency in wind turbine gearing? Efficiency improvements will require improved accuracy and better surface finishes on both gears and bearings. Extending super-finishing technology to larger gears will be challenging.

What is the gear industry doing to address those issues? Huge investments will be needed to keep advancing the technology. Some companies are not in a position to make those investments and will fall behind. Whoever has the latest equipment will have an advantage.

Does it all begin with the design engineer or is that an over-simplification? There are big advances possible even on existing designs. (But) good design makes the task easier.

What renders worm gears useless for wind turbines? By their nature worms have more sliding action and therefore lower efficiency. They will still have their place due to simplicity and high-ratio-per-reduction-stage, but there is tremendous room for improved efficiency in worms simply because they have so much farther to go. (Even) low-tech helicals are perhaps 97 percent-efficient while worms might be 60 to 70 percent-efficient. Improving from 97 to 98.5 will be possible, but getting that last 1.5 percent will be challenging. Getting a 5 percent improvement in a worm set will become commonplace once the very talented engineers get on the task.

What improvements have been made to date in energy-efficient wind turbine gears? Better bearing and gear accuracy.

Which processes—heat treat, lubrication, grinding, etc.—need improvement most? Or are already being improved for energy-efficient turbine gearing? It depends on how you define energy efficiency. If you consider 'cradle-to-grave' energy costs you have to look at things like heat treat. If a furnace cycle could be reduced from 60 hours to 48 it wouldn't show up in operating efficiency but it certainly reduces the product's carbon footprint.

Do you see direct-drive replacing gearbox-driven power for wind turbines at some point? I'm not as optimistic as many direct-drive proponents. The geopolitical situation on rare earth magnets is not favorable and the cost-per-pound of the material far exceeds that of the materials it displaces.

From an historical perspective, what machinery has gotten bigger, heavier and slower as it 'advanced'? Auto motors and electric motors were much bigger in their early days than they are now. That huge armature does act as a great flywheel to smooth out the load spikes that occur with every blade rotation, but as knowledge improves on reducing those peak loads the gearboxes will become better able to survive.

What other factors affect gearing/energy efficiency in wind turbines? Every aspect of the machines will need continuous improvement. Blades, towers and controls have a long way to go. Energy conservation re-fits for existing facilities and equipment will become commonplace.

General Properties of Different Base Oils				
Properties	**Mineral Oil**	**Polyalphaelefines**	**Polyglycol**	**Ester***
Viscosity-temperature Behavior	0	+	++	+
Aging Resistance	0	+	++	+
Low-temperature characteristics	---	++	+	+
Wear protection	0	+	++	+
Friction coefficient	0	+	++	++
Neutrality towards sealing materials and paints	++	++/	/+	

++ = very good 0 = satisfactory
+ = good --- = poor

The Properties of esters depend on the specific type of ester and may differ strongly (courtesy Küber).

can be as low as one to five– percent per stage, while in a worm gear they can be up to 30 percent. Planetary gears minimize efficiency losses to just one percent per stage."

Put another way, "Efficiency is loss-per-mesh, says Chuck Schultz, licensed chief engineer for Beyta Gear Service and a member of this magazine's technical editors staff. "More meshes, more loss; oil churning and windage (A force created on an object by friction when there is relative movement between air and the object.) also contribute. Details of helical and spur gear design can make some improvement but it still comes down to rolling and sliding. Worms have more sliding and lower efficiency as a result."

With all that rolling and sliding to deal with, a chart provided by Klüber Lubrication North America L.P. helps break things down in some detail as to various gear characteristics.

"As the sliding percentage increases, the efficiency of the gearbox decreases," says Dennis A. Lauer, P.E. and Klüber vice president of engineering. "A gear unit performs rolling and sliding movements on the power transmitting flanks of the meshing teeth. The load on the tooth flanks is a function of the tooth geometry and the forces generated by the sliding movement. In gears mainly performing a rolling movement (spur and helical gears, for example), the load on the tooth flanks is generally lower than in gears mainly performing a sliding movement (such as worm and hypoid gears). The higher the sliding percentage and wear load on the tooth flanks, the higher the requirements a lubricant has to meet."

An obvious second question—at least when speaking with a lubricant expert—is what exactly can lubrication do to increase energy efficiency in wind turbines?

"Due to past reliability issues, it is extremely important that a lubricant is properly designed, says Lubrizol's Michelle Graf, product manager/hydraulic and industrial gears. "A balanced formulation will enhance the overall performance of the gearbox by providing optimum lubrication while also protecting all other components of the gearbox. Lowering friction helps to reduce energy consumption while optimizing oxidation resistance, and may allow oil drain interval extension."

"Research is ongoing to identify ways of designing gears and lubricant combinations which can increase the energy efficiency of wind turbine gears," says Guerzoni. "A major area of focus lies in bearing selection."

Klüber's Lauer points out that "Originally, mineral oils were used in wind turbine gears but we know that the synthetic gear oils can reduce friction over mineral oils, so many windmill operators are beginning to take advantage of this. Of course, if you can reduce the friction in a gearbox, the efficiency will increase. "But," he adds, "an increasing number of wind farms are using PAO synthetic oil. We have also introduced poly-glycol and rapidly biodegradable ester synthetic oils for their additional advantages over PAOs."

"The majority of wind turbine lubricants are usually ISO 320 viscosity grades," says Graf. "Although both mineral-based and synthetic formulations are utilized, synthetic lubricants tend to be more widely used due to their enhanced oxidation performance, wide temperature operating range and the potential for drain (replenishment) interval extension."

Shell's Guerzoni makes the point that, "As wind turbine technology has evolved, so too have the range of gear oils required to lubricate them effectively and reliably. The trend in the marketplace today and the gear oil types for most of the leading wind turbine OEMs are synthetic PAO-type ISO 320 gear oils. Some operators still specify the use of mineral oil, ISO 320-type gear oils; however this is not as widespread as previously. Synthetic biodegradable ester-type gear oils may be recommended on specific projects, but are not widely used in general.

So, non-synthetic or synthetic lubrication—what are the plusses and minuses? See Table 2 for a comparison.

And while synthetic certainly appears to be the lubricant of choice, there are caveats. One has to do his homework before making a choice.

"Not all synthetic gear oils perform the same," Guerzoni advises. "It is still critical to formulate a product having a balance of the appropriate base fluids and additives to provide synergistic benefits from the product in use. Synthetic PAO fluids (ISO VG 320) are now the most commonly used within the wind turbine industry. Some consider the up-front cost of the synthetic gear oils prohibitive; however, on a life-cycle cost basis and the fact they offer extended lifetimes relative to mineral oils, the payback on the return of investment is rapid.

"Well-formulated products offer benefits in the area of robustness to contamination from water, excellent rust and corrosion inhibition, very good low-temperature fluidity and low-foaming tendency. Synthetic PAO-type gear oils can also lead to reduction in operating temperatures relative to mineral oil-based gear oils of equivalent viscosity grade. This further helps extend oil life and equipment reliability."

Yet with all of the above understood, the question remains—Which synthetic lubricant is best for wind turbine gearing? It depends; it's not like you can just draw up an equation.

Says Guerzoni, "Given the remote location of many wind farms, the desire for improved reliability and the challenges involved in conducting a gear oil change (in a wind) tower—not only logistics but crane rental as well—the selection of gear oils which can extend oil life, improve reliability and reduce the costs associated with oil change-outs are critical. Gear oils must be able to provide not only extended service periods (with demands today for four years and upwards), but protect gears from common failure modes such as micropitting and scuffing wear. They must be robust to contamination from salt-laden water preventing rust and corrosion to gears and bearings. They must operate without the formation of deposits which can plug filters and impact efficiency. Based on the lowest and highest ambient operating temperatures, consideration must be given to the low temperature fluidity—i.e., how easily the product will flow under cold temperature start-ups. Compatibility with seal materials and paints used in the gearboxes is also important. The fact that industry specifications for wind turbine gear oils today include such a significant number of separate laboratory and rig tests before they are even subjected to a field trial is testament to the desire to select products which will improve reliability for the wind farm operator."

Or, as Lauer puts it, "First we need to select the proper viscosity and additive package to protect the gears and the bearings. We want to provide a chemistry that will achieve the longest possible life in the gearbox—like polyglycol—and strongly recommend oil condition monitoring to maximize the change intervals."

Photo (courtesy Lubrizol).

"Consult with the manufacturer of the turbine who may or may not refer the end user to the gearbox manufacturer for their recommendation on proper fluid selection," says Lubrizol's Graf. "Because of the gearbox reliability issues in the past, wind turbine and gearbox manufacturers tend to be con-

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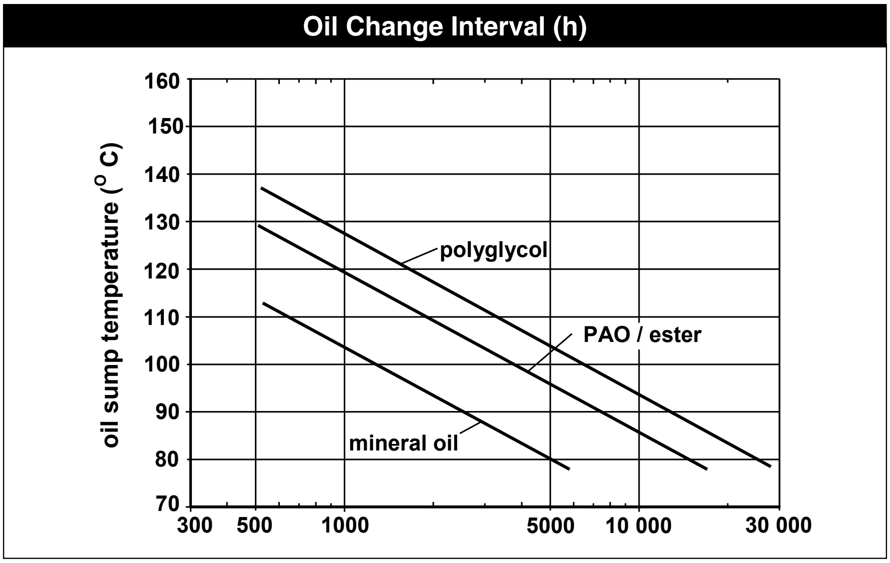

Polyglycol provides the best aging resistance, providing the longest oil change intervals (Courtesy Klüber).

servative relative to establishing lubricant requirements and require extensive laboratory and field performance validation. A fluid for wind turbine gearbox lubrication must be a balanced formula which provides protection for all gearbox components across a wide range of operating conditions (see above)."

Bearings, you'll notice, have been mentioned several times thus far as an element of lubricant-gear efficiency. And it's no wonder, given the role they play in turbines. And the fact that bearings are perhaps the most high-maintenance-demanding parts in the mix. Bearing failure is an accepted complication in wind turbine maintenance and repair, at least until such time that someone comes up with the "silver bullet bearing."

"In most gearbox designs the gear oil also lubricates the bearings within the gearbox, Graf explains. "Demonstrating bearing protection is integral to the performance of a gearbox lubricant. Gearbox manufacturers require lubricant approvals from their bearing suppliers to ensure suitable bearing protection."

"As gearbox bearing failures continue to undermine the reliability of wind turbines, much focus is now being placed on gear oil cleanliness and the role of contamination," says Guerzoni. "Studies on bearing life from other sectors have identified the negative role of contaminants including particulate and water on bearing life. This is now being translated to tighter cleanliness controls on wind turbine gear oils and the use of finer filtration. Current industry specifications require that gear oil be filtered to very low contaminant levels. ISO 81400 and the draft IEC 61400 (standards), for example, recommend a gear oil cleanliness of oil-added-to-gearbox at any location -/14/11 and < - /17/14 in service as measured by ISO 4406." (*ISO 4406:99 is the internationally recognized standard by which the number and size of solid particulate in 1ml of oil is quantified. The numbers represent the number of particles in a given particle size range against a standard scale of <4 micron, <6 micron, <14 micron—i.e., the smaller the number the cleaner the fluid.*)

Lauer believes that "The gear oil should always be tested on a rolling bearing test rig such as the FAG FE 8 rig to assure that the gear oil not only meets the demands of the gears but also protects the bearings. Oil contamination—particulate as well as moisture—is not good for the gears. But the bearings are even more sensitive."

Those companies involved in wind turbine manufacture are well aware that off-shore wind installations—already popular in Europe—are beginning to take hold here in the United States. One wonders how lubrication requirements might change in that scenario.

"As stated above in selecting the best lubrication, it is necessary to maximize the life of the oil in the gearbox," says Lauer. "A chemistry that is more resistant to moisture contamination would also be beneficial. Here again poly-glycol appears to be the best chemistry."

"There will continue to be interest in extending equipment maintenance intervals, including lubrication," says Graf. "This need certainly applies to turbines that are located in difficult-to-reach areas which increase the cost of maintenance significantly. For this reason some of the most recent designs for off-shore turbines are featuring direct-drive technology which eliminates the gearbox altogether."

"For offshore wind turbines," says Guerzoni, "there is increasing requirement on reliability, but also resistance to rust and corrosion. Wind turbine gear oils must stand up to the demanding SKF Emcor test when exposed to synthetic sea water with passing results. Long oil life is again critical."

When talking about energy efficiency in lubricants and gearing for wind turbines, the all-encompassing "green" is a word on most players' lips. Just how green are today's lubricants?

Ironically, given that they are used in a "clean energy production" application such as wind turbines—not much.

"In general there seems to be an increasing desire for fluids that can lubricate turbine components operating in environmentally sensitive areas," Graf states. "However the lubrication challenges for wind turbine applications make this extremely challenging. There are wind turbine lubricants commercially available that can minimize environmental impact, but there has been limited market acceptance due to their price premium."

"The term 'green' may be interpreted in many different ways," Shell's Guerzoni explains. "At this point in time very few projects call on the use of biodegradable gear oils, hydraulic fluids or greases. While such products do exist—offering a more environmentally considerate option for the operator—there are currently no regulations or industry requirements for more

Photo (courtesy Lubrizol).

widespread utilization of biodegradable gear or hydraulic oils for the wind turbine sector. As off-shore wind turbines move more towards direct-drive instead of geared units, the potential need for biodegradable gear oils reduces further."

Lauer is even more frank in his assessment. "If by 'green' you mean rapid biodegradability and sustainability, then I am afraid the current state of the art is not very 'green,' Lauer says, but adds, "Klüber has a rapidly biodegradable gear oil that has been used in one location for almost ten years, but we seldom get requests for this technology. 'Green' gear oils are available, but until it becomes a priority of the end user, I am afraid its use will not expand."

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Gear Hobbing Technology Update

Q&A WITH LIEBHERR'S DR. ALOIS MUNDT

William R. Stott, Managing Editor

Dr. Alois Mundt

Recently our editors had the opportunity to interview Dr. Alois Mundt, general manager of Liebherr-Verzahntechnik GmbH, regarding the state of the art in gear hobbing technology. We talked about the latest advances in machine tools and cutting tools, as well as how Liebherr is keeping up with the changes in technology.

What are the most significant recent advances in gear machine tool technology related to gear hobbing?

Dr. Alois Mundt: Right now, the technology is facing various trends: Other than the further development of direct drives and modern controls we see significant progress in automation, process integration and process development.

What Liebherr machine models feature these advances?

Dr. Alois Mundt: As for the drive technology, all of our machines offer various options. Regarding table drives for example, we are offering both worm gear drives and direct torque drives. The customers can choose the best fitting solution for their requirements, based on our advice. This benefit also applies to a number of other components.

continued

Automation is vital to increase efficiency of the production process. For our machines, we realize automation concepts up to a part size of 2,000 millimeters. In addition, we provide the integration of our technology into the customer‘s processes on individual demand.

When were these machines introduced?

Dr. Alois Mundt: Liebherr introduced these machines on the market in the late 1990s—starting with the models for smaller parts. Over the years we have been constantly improving the technology of every model and expanding the range of sizes that we offer.

How will gear hobbing machines continue to improve over the next 5–10 years?

Dr. Alois Mundt: The basic prospects in this area are somewhat similar to those in other industries. Gear hobbing machines will focus on even more functionality, higher complexity, user-friendly operation, energy savings and low maintenance.

What are the most significant recent advances in gear cutting tool technology (i.e., hob coatings and substrate materials)?

Dr. Alois Mundt: Recently developed coatings play a major role in the progress of gear cutting tool technology. By providing longer tool life, for example, they make for more efficient processes altogether. Also, the breakthrough of ICI technology has been a crucial advance, i.e. for coarse pitch and large gears.

How will gear cutting tools continue to improve over the next 5–10 years?

Dr. Alois Mundt: The processes will be even more specific. Due to this development the design of the cutting tools, among other things, will be more detailed and dedicated.

What are the current trends in gear hobbing technology?

Dr. Alois Mundt: Chamfering and deburring has become more important in the context of small gears. Also, manufacturers focus more on the pre grinding quality and higher process stability. Other trends for large gears include the growing significance of ICI hobs; increasing component sizes applicable for automation; and high performance gears. The design features in the gear geometry are getting more sophisticated. For example, smaller pressure angles, higher helix and higher cutting depths. This takes away some manufacturing improvement back to lower chip removal rate.

What methods are your customers using to increase gear hobbing throughput?

Dr. Alois Mundt: Generally speaking, there is no silver bullet. We jointly develop the most efficient solution for the needs of each individual situation in discussion with our experts.

What methods SHOULD they be using but aren't yet using?

Dr. Alois Mundt: There are a number of untapped or underutilized potentials in the industry. This is particularly true for the training of employees—an area that has been broadly neglected by many customers in the past. Additional fields in this context are: the application of modern tools, cutting-edge technology, modern powerful machines, fast automation and a modern and rigid fixture design.

How are gear hobbing operations being combined with other operations to reduce overall cycle time?

Dr. Alois Mundt: A key point in this context are add-on processes that run simultaneously to maintenance.

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Analysis and Testing of Gears with Asymmetric Involute Tooth Form and Optimized Fillet Form for Potential Application in Helicopter Main Drives

F.W. Brown, S.R. Davidson, D.B. Hanes, D.J. Weires and A. Kapelevich

Management Summary

Gears with an asymmetric involute gear tooth form were analyzed to determine their bending and contact stresses relative to symmetric involute gear tooth designs, which are representative of helicopter main-drive gears. Asymmetric- and baseline (symmetric)-toothed gear test specimens were designed, fabricated and tested to experimentally determine their single-tooth bending fatigue strength and scuffing resistance. Also, gears with an analytically optimized root fillet form were tested to determine their single-tooth bending fatigue characteristics relative to baseline specimens with a circular root fillet form. Test results demonstrated higher bending fatigue strength for both the asymmetric tooth form and optimized fillet form compared to baseline designs. Scuffing resistance was significantly increased for the asymmetric tooth form when compared to a conventional, symmetric involute tooth design.

Introduction

The objective of the work described here is to begin the process of evaluating the potential benefits of asymmetric involute gear teeth and optimized root fillet geometry for helicopter main transmission applications. This involves not only quantifying performance improvements achieved by these concepts, but also evaluating the practicality of manufacturing gears with asymmetric teeth and optimized root fillet geometry for aerospace applications. (*Authors' note: This work was performed under the sponsorship of the Center for Rotorcraft Innovation (CRI). The authors are grateful to CRI for the opportunity to investigate these technologies.*)

In helicopter main-drive applications, minimizing gear weight while maintaining the necessary balance of tooth bending strength, pitting resistance and scuffing resistance is given high priority during design of the gears. In many helicopter applications, gears are required to transmit high, continuous-torque loads in one direction, but are lowly loaded in the opposite direction. Traditionally, spur gear designs in helicopter gearboxes utilize conventional (symmetric) involute teeth that provide the same torque capability in both the drive- and coast-loading directions. An overall weight reduction may be realized by using gears with higher capability than conventional gears in the primary drive direction, even if some capacity is sacrificed in the secondary coast direction.

The design intent of asymmetric gear teeth is to improve performance of the primary drive profiles at the expense of performance of the opposite-coast profiles. In many cases the coast profiles are more lightly loaded and only for a relatively short duration. Asymmetric tooth profiles make it possible to simultaneously increase the contact ratio and operating pressure angle in the primary-drive direction beyond the conventional gears' limits. The main advantage of asymmetric gears is contact stress reduction on the drive flanks—resulting in reduced gear weight and higher torque density.

Many traditional helicopter spur gear designs utilize circular root fillet geometries that are form-ground along with the gear flanks. Gear specimens were analyzed, designed and manufactured to compare the single-tooth bending fatigue strength of gears with optimized root fillets with gear specimens having circular root fillets.

Areas addressed in this article include analysis, design, manufacture and testing of gear test specimens with asymmetric teeth. Conventional symmetric tooth specimens were also produced and tested to provide a baseline for comparison. The design and manufacture of the gear specimens are representative of helicopter main-drive gears. Testing included single-tooth bending fatigue and scuffing tests of both the asymmetric and baseline gears. Additionally, symmetric-toothed gear specimens with optimized root fillet geometry were analyzed, designed, manufactured and tested in single-tooth bending fatigue and compared to conventional specimens with circular root fillet geometry as a baseline.

Background

Typically, helicopter main-drive gears are required to transmit high, continuous torque loads in their primary or drive direction. Torque loads in the opposite (secondary or coast) direction are lower in magnitude and of shorter duration than torques in the primary direction. In simple planetary gear arrangements—often used in helicopter gearboxes—the planet gears are required to transmit load on both sides of their teeth due to contact with the sun gear at one mesh and contact with the internal ring gear at the other mesh. In such cases the size of the planet gear teeth is usually dictated by the requirements of the sun/planet mesh and stresses are lower at the planet/internal ring gear mesh due to the more-conforming contact. Traditionally, spur gear designs in helicopter gearboxes are symmetric involute teeth that provide essentially the same torque capability in both the drive- and coast-loading directions. There may be an overall weight benefit from using gears with higher capability in the primary drive direction—despite a degree of capacity loss in the secondary direction.

Gears with asymmetric teeth have existed for many years. Cambridge University Professor Robert Willis (*Ed.'s note: Willis's* Principles of Mechanism *was published in 1841 and became a standard text for engineering students.*) wrote about buttress gears in 1838 (Ref. 1). Since then, many articles on the subject of asymmetric gears have been published. However, there are very few practical applications for such gears. One of them was an application of the asymmetric teeth in the planetary gearbox in a turboprop engine (Ref. 2).

The work described herein presents the design, manufacture and testing of:

- Asymmetric involute gears with circular root fillet geometry
- Symmetric gears with optimized root fillet geometry

The asymmetric tooth geometry and optimized fillet geometry were developed by the "direct gear design method" (Ref. 3). Unlike traditional gear design, this method does not use a pre-selected basic or generating gear rack to increase the gear tooth profile; rather, it defines the gear tooth formed by two involutes of two different base circles (in the case of asymmetric teeth) with the arc distance between them and tooth tip circle to avoid the sharp-pointed tooth tip. If these base circles are identical, the gear has symmetric teeth; the fillet between teeth is not in contact with the mating gear teeth. However, this portion of the tooth profile is critical because this is the area of the maximum bending stress concentration. The fillet profile is designed independently and a subject of optimization providing minimum bending stress concentration and sufficient clearance with the mating gear tooth tip in mesh (Ref. 4).

Test Specimen Design and Analysis

Test specimen gears designed for this program are representative of helicopter main-drive gears in diametral pitch, pressure angle, material and processing. Standardized, conventional-toothed designs have been developed for bending fatigue and scuffing test rigs that Boeing Rotorcraft uses for

continued

Table 1—Single tooth bending fatigue test gears

Parameters	Symmetric teeth with circular fillets (baseline)	Symmetric teeth with optimized fillets	Asymmetric teeth with circular fillets
Number of teeth	32	32	32
Diametral pitch	5.3333	5.3333	5.3333
Pressure angle	25°	25°	35° Drive 15° Coast

gear research. The standardized test specimen designs were modified to incorporate the asymmetric tooth configuration and another for the optimized fillet configuration. Specimens of each type were manufactured using aerospace production techniques and requirements. A manufacturing approach was developed with the goal of reducing material and processing variability.

The single-tooth bending fatigue test gears are 32-tooth gears with groups of four teeth removed per quadrant to allow for assembly into the single-tooth bending fatigue (STBF) test fixture. Table 1 summarizes the basic design parameters for the single-tooth bending fatigue test gears, and also shows these design parameters for the baseline specimen design.

The optimized fillet gear is similar to the baseline design, with the circular fillet replaced by the optimized root fillet. A comparison is shown in Figure 1.

Similarly, the scuffing test gears are within the design experience range of typical main-transmission helicopter power gears. The test gear design is of similar size to a first-stage

Table 2—Scuffing test gears

Parameters	Symmetric teeth with circular fillets (baseline)	Asymmetric teeth with circular fillets
Number of teeth	30	30
Diametral pitch	5.3333	5.3333
Pressure angle	25°	35° Drive 18° Coast

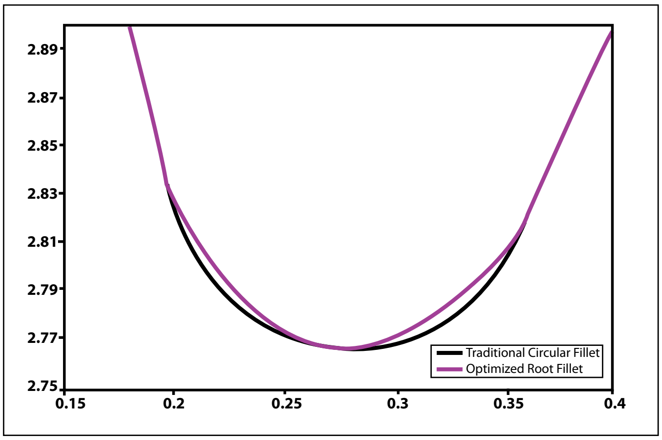

Figure 1—Coordinate plot of circular fillet and optimized fillet design geometries.

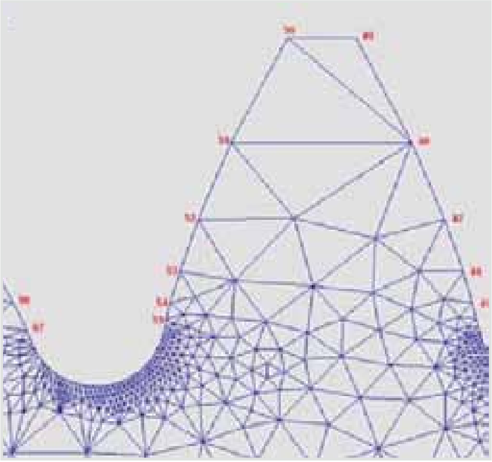

Figure 2—FEA mesh STBF baseline symmetric tooth.

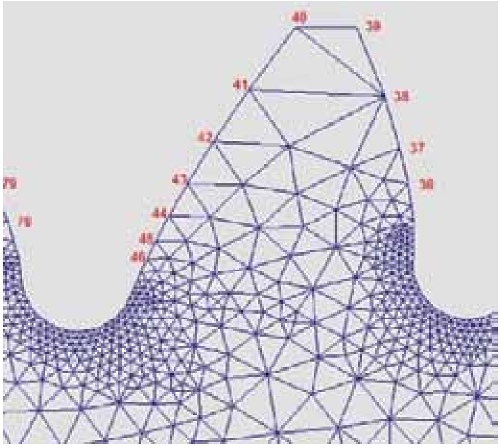

Figure 3—FEA mesh STBF asymmetric tooth specimen.

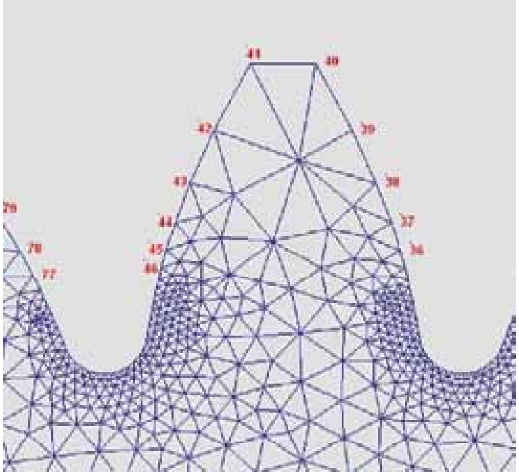

Figure 4—FEA mesh baseline symmetric scuffing gear specimen.

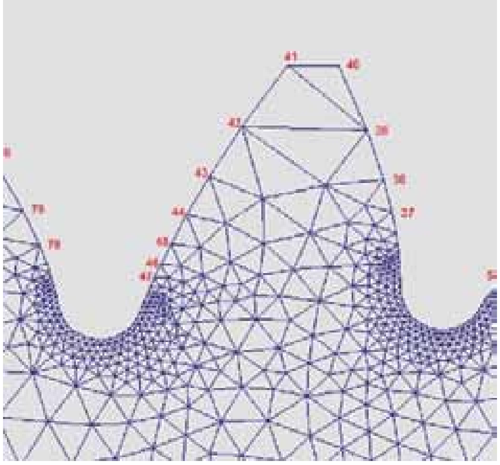

Figure 5—FEA mesh asymmetric tooth scuffing gear specimen.

planetary sun/planet mesh that can be found in a medium-to-large-size helicopter (Table 2).

Test Specimen Analysis

The test specimen gear designs were analyzed to predict their bending and contact stresses, and compared to stresses predicted for the baseline test specimens.

Asymmetric tooth geometry*: *single-tooth bending specimens. Single-tooth bending fatigue specimens were designed employing asymmetric involute teeth. For comparison, conventional, symmetric involute gears were designed and tested. Both asymmetric-toothed and conventional baseline specimens employ ground, circular root fillets.

Table 3—Comparison of data for baseline symmetric and asymmetric toothed STBF gear speciments

Parameters*	Baseline Symmetric toothed specimen	Asymmetric toothed specimen
Number of teeth	32	32
Diametral pitch	5.3333	5.3333
Drive pressure angle, deg	25	35
Coast pressure angle, deg	25	25
Pitch diameter, P_d, in	6.0000	6.0000
Drive base diameter in	5.4378	4.9149
Coast base diameter, in	5.4378	5.7956
Outside diameter, in	6.4000	6.3864
Root diameter, in	5.571	5.558
Drive TIF diameter, in	5.6939	5.6581
Coast TIF diameter, in	5.6939	5.8810
Circular tooth thickness, in	0.2905-0.2885	0.2905-0.2885
Fillet radius, in	0.074 min circular	0.078 min circular
Face width, in	0.375	0.375
Torque, in-lb	5,000	5,000
Load application radius, in	3.06	3.06
Calculated maximum bending stress, psi	57,887	54,703

NOTE: * Length dimensions in inches, angles in degrees.

The asymmetric gear tooth form for the STBF test specimens is nominally based on the standard STBF gear specimen. This enables the asymmetric-toothed specimen to fit the existing test fixture with only minor modifications for tooth load angle, and provides a direct comparison between asymmetric and conventional gears of the same diameter and face width. Finite element analysis (FEA) meshes of the baseline symmetric-toothed gear specimen and the asymmetric-toothed specimen are shown in Figures 2–3.

continued

Table 4—Comparison of data for baseline symmetric and asymmetric toothed scuffing test specimen gears

Parameters*	Baseline Symmetric toothed specimen	Asymmetric toothed specimen
Number of teeth	30	30
Diametral pitch	5.0000	5.0000
Driveside pressure angle, deg	25	35
Coastside pressure angle, deg	25	18
Pitch diameter, P_d, in	6.0000	6.0000
Drive base diameter in	5.4378	4.9149
Coast base diameter, in	5.4378	5.70636
Outside diameter, in	6.400 max	6.4034 max
Root diameter, in	5.459 max	5.510 max
Drive TIF diamter, in	5.6864	5.6415
Coast TIF diamter, in	5.6864	5.7607
Circular tooth thickness, at (P_d), in	0.3106-0.3086	0.3106-0.3086
Fillet radius, in	0.059 min	0.081 min
Face width, in	0.50	0.50
Contact ratio	1.417	1.25
Torque, in-lb	6,000	6,000
Calculated maximum contact stress, psi	193,180	174,100

NOTE: * Length dimensions in inches, angles in degrees.

The gear parameters and calculated bending stresses for the STBF test gears are presented in the Table 3.

***Asymmetric tooth geometry*: *scuffing test specimens*.** FEA meshes of the baseline (symmetric) scuffing gear tooth and the asymmetric scuffing gear tooth are shown in Figures 4–5, respectively.

The gear parameters and comparison results are presented in Table 4.

***Optimized root fillet geometry: single-tooth bending specimens*.** Single-tooth bending fatigue specimens were designed employing traditional, symmetric involute teeth. This specimen tooth geometry fits the single-tooth bending fatigue test rig at Boeing. Baseline specimens employ a ground, circular root fillet representative of gears in the aircraft application. The optimized fillet specimens share the same tooth geometry as the baseline—except for the form of the root fillet. The form of the optimized root fillet profile was determined analytically. The fillet optimization technique is based on two-dimensional finite element analysis (FEA) employing a random search method (Ref. 4). The FEA meshes of the circular fillet gear tooth and the gear tooth with the optimized fillet are shown in Figure 6; notice that optimization of root fillet results in a non-constant radius that is slightly "pinched"—as opposed to the traditional, circular fillet. Optimized fillet geometry is defined as a series of coordinate points. These points are plotted (Fig. 7) and can be compared to the coordinates of the circular root fillets (Fig. 8).

The gear parameters and comparison of results are presented in Table 5.

***Test specimen manufacturing*.** The asymmetric gear specimens, optimized root fillet gear specimens and baseline circular fillet test gears were fabricated by Aero Gear in South Windsor, Connecticut. The specimens were fabricated from aerospace-quality (AGMA Grade 3) 9310 steel, with all pertinent records and certifications retained. All specimens were low-pressure carburized and high-pressure gas quenched. Low-pressure carburizing and high-pressure gas quench heat treating processes were performed at Solar Atmospheres of Souderton, Pennsylvania. The material for all specimens was from the same heat lot, and the heat treat processes, grind stock removal and shot peening processes for all specimens were identical. All gears were surface temper etch-inspected and magnetic particle-inspected after the completion of machining.

All specimens produced for this project were ground using conventional gear tooth form grinding equipment, including

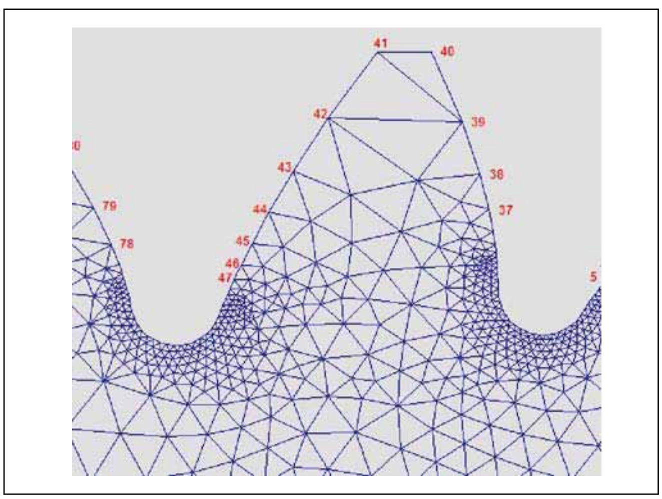

Figure 6—FEA mesh STBF symmetric tooth circular fillet.

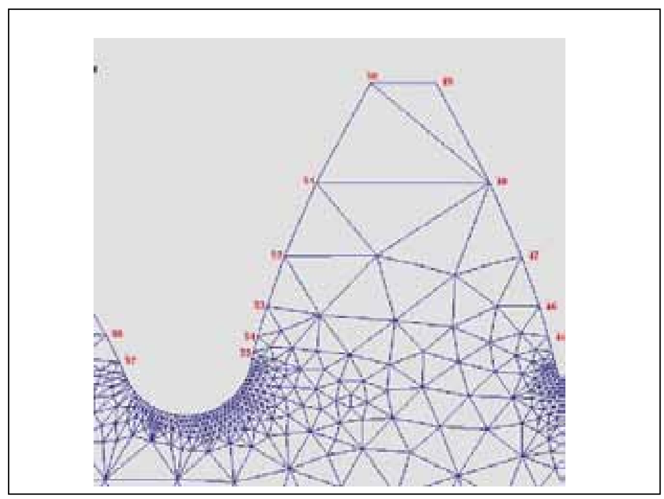

Figure 7—FEA mesh STBF gear symmetric tooth optimized fillet geometry.

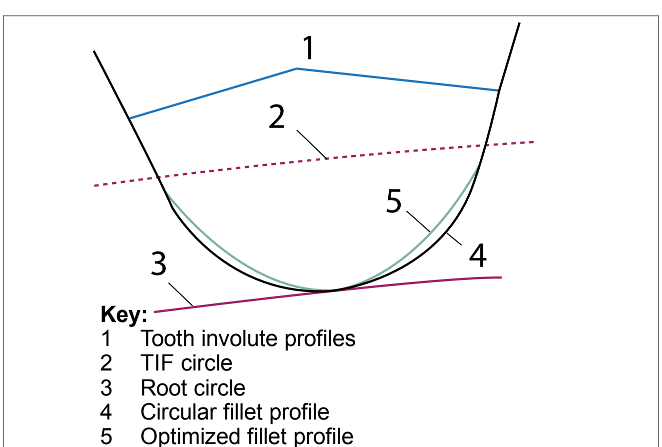

Figure 8—Comparison of circular fillet and optimized fillet geometries. Key: 1 = Tooth involute profiles; 2 = TIF circle; 3 = Root circle; 4 = Circular fillet profile; 5 = Optimized fillet profile.

Figure 9—Gear form grinding set-up with CBN grinding wheel.

the asymmetric tooth specimens and specimens with optimized root fillet geometry. The form grinding process is often used to grind conventional symmetric gear teeth with circular fillets in helicopter main-drives. CBN form grinding wheels were produced from data shown on the engineering drawings for both the asymmetric gear teeth and optimized fillet geometry. An example of the CBN gear grinding set-up is shown in Figure 9.

Inspections of the gear teeth were carried out using conventional CMM gear checking equipment and software. The optimized root fillet geometry was checked using conventional CMM inspection equipment that generated plots showing measured fillet coordinates relative to upper- and lower-tolerance limits that had been established prior to manufacturing. An example of a plot is shown in Figure 10.

Test arrangement and procedure. Single-tooth bending fatigue tests were performed at Boeing-Philadelphia on non-rotating, single-tooth bending fatigue test fixtures (Fig. 11). These fixtures are loaded by Baldwin-Lima Hamilton IV-20 universal fatigue machines through a series of alignment fixtures and in-line load cells. These fatigue machines are capable of 18,000 lb.—i.e., 10,000-lb. steady load and 8,000-lb. alternating load).

For the STBF testing of the subject gears, pulsating fatigue load is applied to the tooth through the load link and flexure arrangement (Fig. 11). The test gear teeth were cycled

continued

Table 5—Comparison circular fillet and optimized fillet STBF gear specimens

Parameters *	Baseline Symmetric toothed specimen	Asymmetric toothed specimen
Number of teeth	32	32
Diametral pitch	5.3333	5.3333
Pitch diameter, P_d, in	6.0000	6.0000
Outside diameter, in	6.3975	6.3975
Root diameter, in	5.561	5.561
TIF diameter, in		
Circular tooth thickness, in	0.2895	0.2895
Fillet radius, in.	0.086 (circular fillet)	Optimized fillet profile
Face width, in	0.375	0.375
Torque, in-lb	5,000	5,000
Load application radius, in	3.06	3.06
Calculated maximum bending stress, psi	57,887	48,387 (-16.4%)
Fillet curvature radious at max. bending stress point, in	0.086	0.317

***NOTE:** Length dimensions in inches, angles in degrees.

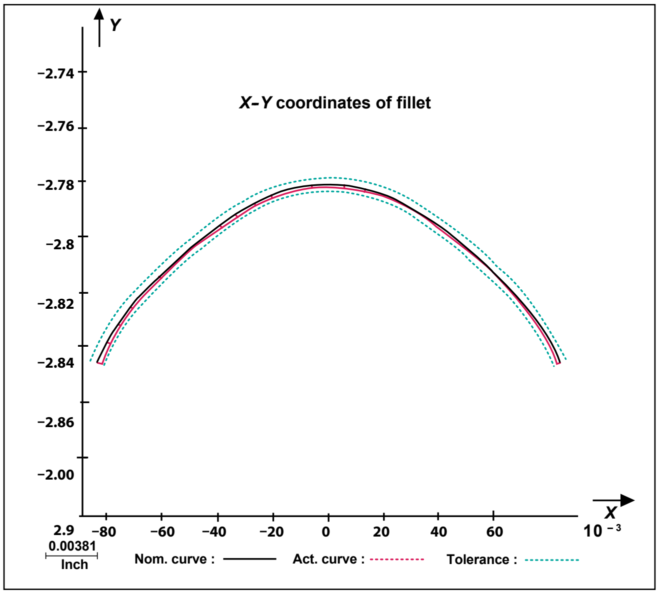

Figure 10—Inspection chart for optimized fillet grinding.

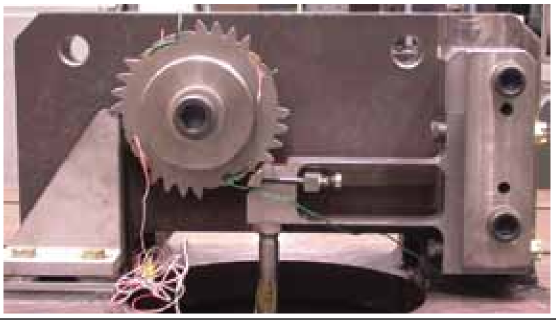

Figure 11—STBF test fixture with asymmetric gear installed.

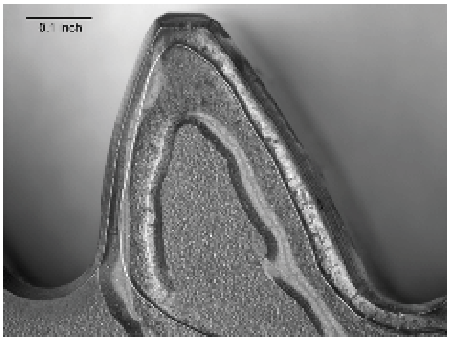

Figure 12—Asymmetric STBF test tooth with crack-wire installed.

at approximately 1,200 cycles per minute. Prior to the start of testing, alignment of the fixture was verified with a strain-gaged baseline specimen. The specimen was instrumented with three strain gages across the face width, and was used to align the fixture as well as to correlate load applied to stress in the fillet of the tooth. For fatigue testing, each tested tooth is instrumented with a crack-wire (Fig. 12; *Ed.'s note: A crack-wire is a sensor used for monitoring cracks and crack growth in supporting structure elements. In case of cracks caused by overload, the crack-wire breaks. Source:* CHOSEN Consortium 2008-2011).

Upon failure of the crack-wire—and due to the presence of a fatigue crack—the test machine is triggered to shut down. The crack-wire is placed so that a 0.050-inch crack length is detected.

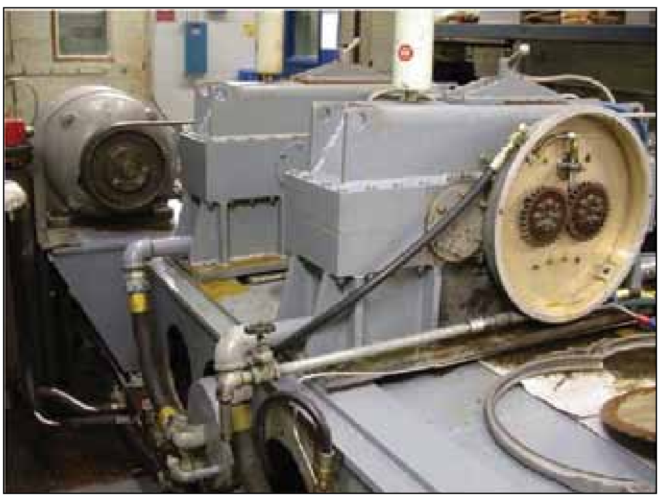

Figure 13—Scuffing test rig with cover removed and test specimen gears installed.

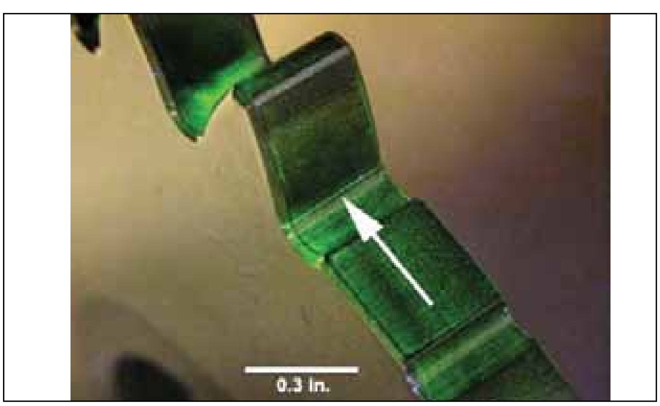

Figure 14—Cracked STBF test gear tooth showing MPI crack indication.

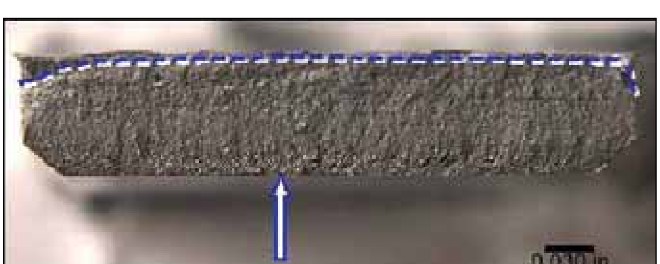

Figure 15—Fractograph of STBF test tooth. The blue dashed line represents the extent of fatigue propagation, and the arrow indicates the fracture origin.

Magnetic particle inspection is used to confirm the presence of a crack. Each tooth specimen was run continuously until failure or run-out; for this project, run-out was defined as $1x10^7$ cycles.

Scuffing tests of asymmetric gear specimens and baseline specimens were conducted on a gear research test stand at Boeing Philadelphia; the test stand is a split/coupling torque design. The test gears are outboard of the main housing and can be quickly inspected or changed by removal of a simple cover (Fig. 13).

A separate lubrication system serves the test specimen chamber, which is isolated from the test stand drive lubrication system. The lubricant supply to the test gears can be heated or cooled to supply lubricant at a constant temperature to the test gears. The test gears were subjected to a series of 15-minute-increment loaded runs. At the end of each run, a visual evaluation of the test gear teeth was conducted. If the condition of the gears did not meet the criteria for scuffing failure, the next-higher incremental load was applied. This procedure was continued until a scuffing failure was observed. For purposes of this test program, a scuffing failure was deemed to be 25% of the available tooth contact surface exhibiting visible evidence of radial scratch marks—characteristic of scuffing—on a minimum of 10 teeth.

Test Results

At the conclusion of the single-tooth bending fatigue tests, all crack locations were verified, both visually and using magnetic particle inspection (MPI) (Fig. 14); cracks were also opened to determine the origins and confirm the validity of the results (Fig. 15).

Fatigue results of the single-tooth bending fatigue tests are presented in Figure 16. The asymmetric tooth and the optimized root fillet tooth are compared with the baseline specimens tested in this project.

(Curves for the optimized root fillet data and the asymmetric data were assumed to be parallel to the baseline curve.)

Typical scuffing failures are shown in Figures 17–18, revealing the vertical scratches indicative of a scuffing failure associated with the breakdown of the separating lubricant between the gears.

Figure 19 shows the scuffing results for baseline and asymmetric gears. The 35°-pressure-angle asymmetric gears showed an improvement of approximately 25% in mean scuffing load (torque) compared to the baseline symmetric tooth specimens. Mean-3 sigma levels are also shown, based on a population of eight baseline data points and six asymmetric data points.

Discussion of Results

The STBF test results (Fig. 16) indicate the asymmetric tooth gear design mean endurance limit was significantly higher—on the order of 16% higher—than the mean endurance limit of the baseline symmetric tooth design. It should be pointed out that there are relatively few data points—four failure points and one run-out (included as a failure point in the data analysis)—for the asymmetric tooth specimens. Nonetheless, the results of this testing indicate that asymmetric teeth offer an improvement in bending fatigue strength, although additional testing would serve to refine the magnitude of the improvement. It is interesting to note that the FE analysis of the asymmetric tooth STBF design predicted a 5.5% reduction in maximum bending stress compared to the baseline symmetric design.

The STBF results for the optimized fillet geometry design showed an improvement in mean gear tooth bending fatigue strength exceeding 10%, based on limited testing—i.e., six failure points. The data points for these tests display more variation (scatter) than either the baseline data or the asymmetric tooth data. As of this writing a confirmed cause of the variation has not been ascertained. Post-test evaluation of the test specimens and observations of the fracture surfaces did not indicate any anomalies that could explain the variation, such as variations in optimized fillet form/dimensions or specimen metallurgy. One theory is that the test fixture was damaged while testing at the higher load levels. Additional testing is required to fully understand the cause of the observed variation in the test data. The FEA of the optimized fillet design indicated a (calculated) reduction in maximum bending stress of 16.4%, compared to the baseline circular fillet design.

While not tested in this project, the combination of asymmetric teeth and optimized fillet geometry—in the same gear design—may offer improvements in tooth bending fatigue strength greater than either of the concepts taken individually. The decision was made early in this project to test each concept separately. The reasoning was that if one concept or the other proved to be impractical from a manufacturing standpoint, data of value would still be attained for the other concept. Since both concepts appear viable from a manufacturing standpoint, their combination in one gear design is worth investigating further. Indeed, this is the philosophy of the "direct gear design method" (Ref. 3).

The scuffing test results (Fig.19) indicated an improvement in mean scuffing load (torque) to failure of 25% for the asymmetric tooth gear specimens, compared to the baseline symmetric tooth specimens. The improvement in calculated Mean-3 Sigma scuffing performance is even greater; although, based on limited testing—eight baseline points and six asymmetric tooth data points—this is a very significant improvement in scuffing resistance due to asymmetric gear tooth geometry. One must bear in mind that this improvement is in the primary-drive direction of the asymmetric teeth. The opposite (coast) direction scuffing performance of the asymmetric teeth was not tested in this project. This data indicates a significant improvement in scuffing resistance that can be utilized to advantage in high-speed, scuffing-critical gear applications.

Conclusions

- The project successfully fabricated aerospace-quality, optimized-fillet-radius gears and asymmetry gears. Conventional gear cutting, grinding and inspection equipment were used to manufacture all gear specimens; large-scale manufacturing development was not required. These gears were compliant with the engineering requirements based on aerospace practice without rework or regrinding.
- Asymmetric STBF gears demonstrated a 16% im-

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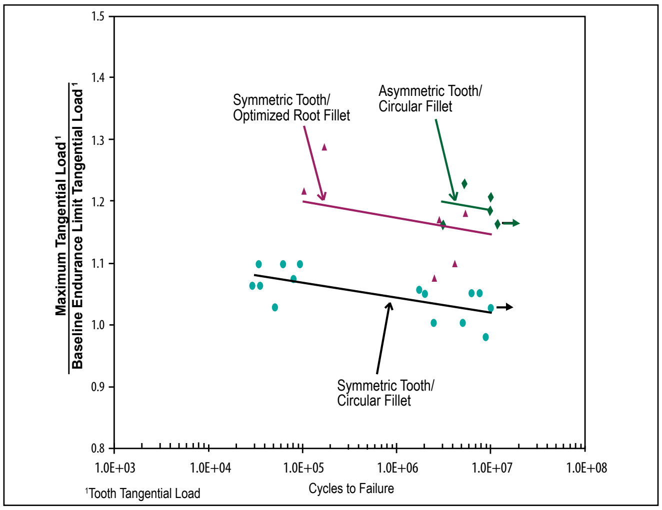

Figure 16—STBF data for asymmetric gears and optimized root fillet gears, along with baseline symmetric tooth/circular fillet test data. Figures show the vertical scratches indicative of a scuffing failure associated with the breakdown of the separating lubricant between the gears.

Figure 17—Scuffing failure of baseline test gear.

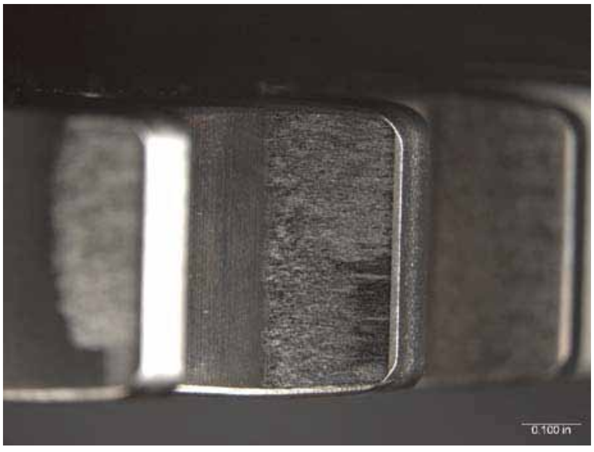

Figure 18—Close-up view of a representative scuffed tooth.

provement in mean single-tooth bending fatigue load capacity compared to baseline symmetric tooth STBF gears. Additional testing/data points would be beneficial to further support these results.

- Based upon the limited data set, the mean endurance limit of optimized fillet gears is estimated to be 12% above the baseline circular fillet gears. Due to the limited number of data points and variation (scatter) in the results, a statistical analysis would define a much lower improvement. Additional testing/data points would be beneficial to further understand these results.
- The asymmetric gear tooth form demonstrated superior scoring performance when compared to conventional symmetric gears. The mean value for a limited data set showed an improvement of approximately 25%.

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Recommendations

- Asymmetric gears demonstrated significant performance benefits in STBF and scoring tests. Additional testing of this gear tooth form, including fabrication of a strain-gaged specimen to obtain measured stresses, is needed.
- Optimized fillet radius gears demonstrated improved mean bending fatigue strength. Boeing recommends additional fatigue testing of this gear tooth form, including fabrication of a strain-gaged specimen to obtain measured stress data.
- Design, fabrication and testing of asymmetric tooth pitting fatigue test specimens to verify pitting resistance.
- Design, fabricate and test specimens incorporating both asymmetric teeth with optimized fillets—including fabrication of a strain-gaged specimen to obtain measured stress data.

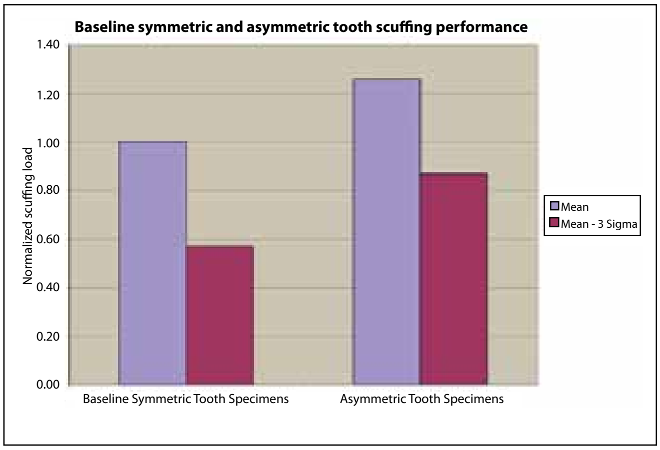

Figure 19—Results of baseline symmetric and asymmetric gear scuffing tests.

References

1. *Engineer and Machinist's Assistant*, 2nd Edition, 1838, Pg. 142.

2. Novikov, A.S., A.G. Paikin, V.L. Dorofeyev, V.M. Ananiev and A.L. Kapelevich. "Application of Gears with Asymmetric Teeth in a Turboprop Engine Gearbox," *10th International ASME Power Transmission and Gearing Conference*, September 4–7, 2007, Las Vegas, Nevada, (DETC2007–34823).

3. **Kapelevich, A.L. and R.E. Kleiss, "Direct Gear Design for Spur and Helical Gears," Gear Technology, September/October 2002, 29–35**.

4. Kapelevich, A.L. and Y.V. Shekhtman. "Tooth Fillet Profile Optimization for Gears with Symmetric and Asymmetric Teeth," *AGMA Fall Technical Meeting*, San Antonio, Texas, October 12–14, 2008, (08FTM06).

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Drive Line Analysis for Tooth Contact Optimization of High-Power Spiral Bevel Gears

J. Rontu, G. Szanti and E. Mäsä

Management Summary

It is common practice in high-power gear design to apply reliefs to tooth flanks; they are meant to prevent stress concentration near the tooth edges. Gears with crowning have point contact without load, and when load is applied, instantaneous contact turns from point into a Hertzian contact ellipse. The contact area grows and changes location as load increases. To prevent edge contact, the gear designer has to choose suitable reliefs considering contact indentations as well as relative displacements of gear members.

In the majority of spiral bevel gears, spherical crowning is used. The contact pattern is set to the center of the active tooth flank and the extent of the crowning is determined by experience. Feedback from service, as well as from full-torque bench tests of complete gear drives, has shown that this conventional design practice leads to loaded contact patterns, which are rarely optimal in location and extent. Oversized reliefs lead to small contact area, increased stresses and noise, whereas undersized reliefs result in an overly sensitive tooth contact.

Today it is possible to use calculative methods to predict the relative displacements of gears under operating load and conditions. Displacements and deformations originating from shafts, bearings and housing are considered. Shafts are modeled based on beam theory. Bearings are modeled as 5-degree-of-freedom supports with non-linear stiffness in all directions. Housing deformations are determined by FEM analysis and taken into account as translations and rotations of bearing outer rings. The effect of temperature differences, bearing preload and clearances are also incorporated.

With the help of loaded tooth contact analysis (LTCA), it is possible to compensate for these displacements and determine a special initial contact position that will lead to well-centered, full-torque contact utilizing a reasonably large portion of the available tooth flank area. At the same time, crowning can be scaled to the minimum necessary amount. This systematic approach leads to minimum tooth stressing, lower noise excitation, increased reliability and/or power density as compared to conventional contact design method.

Introduction

In a majority of spiral bevel gears produced, the tooth contact is initially placed at the center of the tooth flank during manufacturing. Sufficient crowning is applied to prevent the contact from reaching tooth edges under load. However, the use of large crowning also has a downside of increasing contact stresses because the area in contact at a particular moment is reduced. With constantly growing demands for higher power density and lower noise generation, there are pressures for decreased crowning. In a larger sense, there is often a great need to optimize tooth flank topography for a certain application. This requires accurate knowledge about the behavior of tooth contact under load. Regardless of the optimization goal, the change of relative position of bevel gears under load is an important factor.

When loads and temperature differences are applied to a gear drive, the relative position of pinion and wheel changes due to deformations and displacements related to bearings, shafts and housing. This causes changes in the tooth contact—the significance of which is dependent on magnitude and mutual relations of the displacements as well as characteristics of the tooth geometry. One of the main concerns is the spreading and movement of contact pattern that, ideally, should be located at the center of the tooth flank under load and cover as much of the flank area as possible. If the behavior of the contact pattern is known, pre-compensation can be applied in the finish machining of tooth flanks to ensure good running properties under load. This usually means that the tooth flank topography is modified so that the initial contact pattern (without load) is moved from the center of the flank by a certain amount.

Traditionally, the knowledge of tooth contact behavior has been attained through practical experience, but that requires time-consuming and expensive prototype testing. An alternative approach is one based on computer simulation, by which significant cost savings are possible. In recent years, tooth contact optimization based on loaded tooth contact analysis (LTCA) has been applied with good success in numerous customer projects involving marine, industrial and automotive bevel gear applications. Using LTCA, the mesh of spiral bevel gears is simulated using 3-D tooth geometry, taking into account the actual, relative position of gears under load. This paper describes a computational process used to determine how the relative position of bevel gears changes when load and temperature differences are applied on a gear drive. The process is a combination of different calculation methods and is hereafter referred to as drive line analysis (DLA). In addition to the methods usually used in DLA, some alternative approaches are also mentioned to provide a more general overview of applicable methods.

Relative Position of Bevel Gears

In nominal position, the pitch cone apexes of bevel pinion and wheel (if not a hypoid gear pair) coincide. Deviation from this position (location + orientation) can be fully defined by four displacement values, hereafter referred to as relative displacements. As shown in Figure 1, they consist of deviation of shaft angle (*S*); offset (*E*); pinion axial location (*P*); and wheel axial location (*G*).

Driveline Analysis

To fully understand tooth contact behavior in a certain application, the chain of events from assembly (tooth contact adjustment) to operating conditions (loads and temperature differences applied) must be traced. To accurately determine displacement of bevel gears, a detailed analysis of the whole drive line—consisting of shafts, bearings and housing—is required. The "core" of analysis is comprised of separate calculation models for pinion shaft and wheel shaft, hereafter referred to as shaft-calculation models. These models are used to simulate deformations and displacements of shafts and bearings. Commercial software with ranging levels of capabilities is available for this purpose. Beam theory is practically always used to calculate shaft deflection, but there are significant differences in the way bearings are modeled. At the simplest level, bearings are considered as radially stiff "hinges" that do not represent reality very well. In the more advanced software, such as used in DLA described in this paper, bearing internal geometry and stiffness nonlinearity are considered in order to accurately model real behavior.

Bearing Stiffness

Gears are usually supported by a gear unit housing with rolling bearings. Bearing stiffness varies significantly, depending on the type of rolling element (Fig. 2) affecting the displacement behavior of the shaft-bearing system. Another significant factor is the internal alignment capability (Fig. 3).

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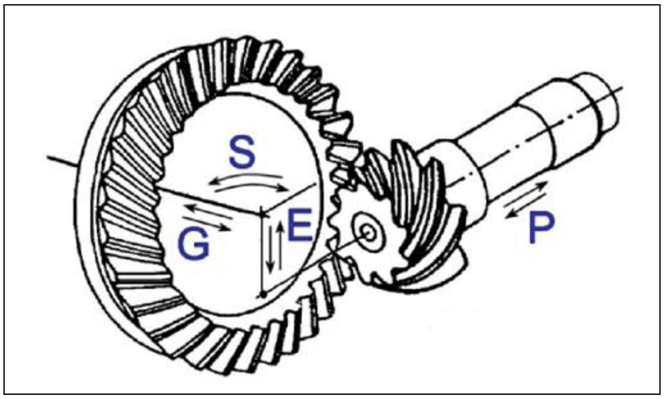

Figure 1—Relative displacements of bevel gears.

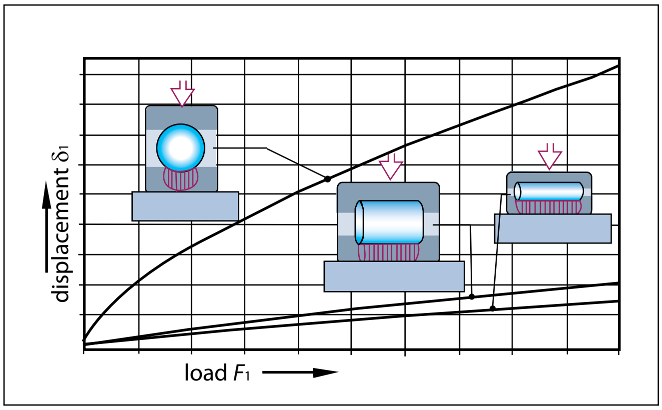

Figure 2—Radial stiffness of different bearing types (Ref. 2).

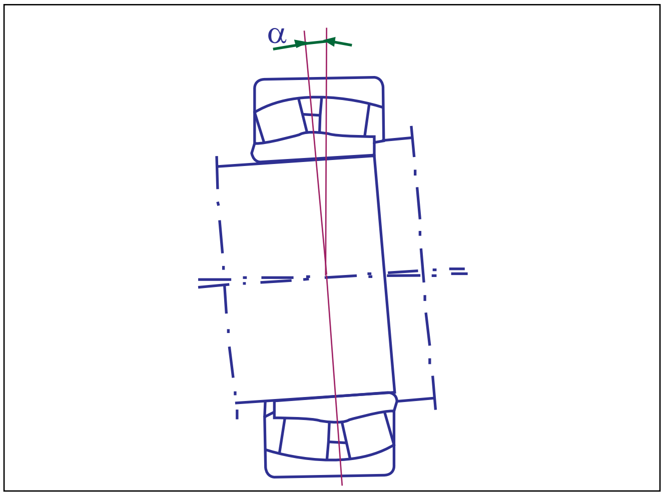

Figure 3—Bearing internal alignment capability.

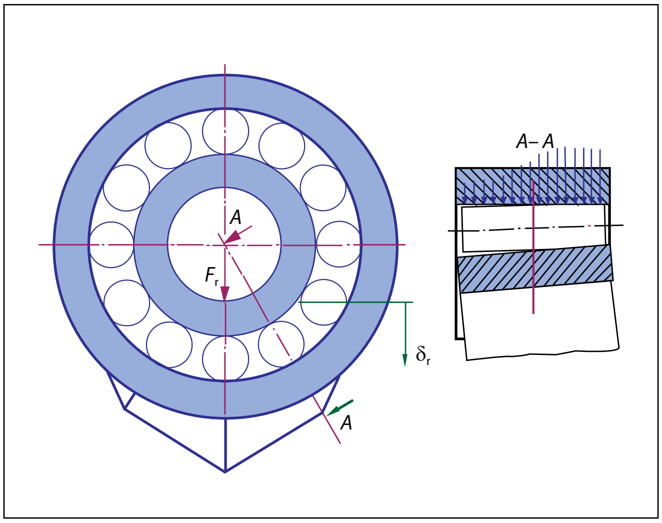

Figure 4—Deformation of individual element contact.

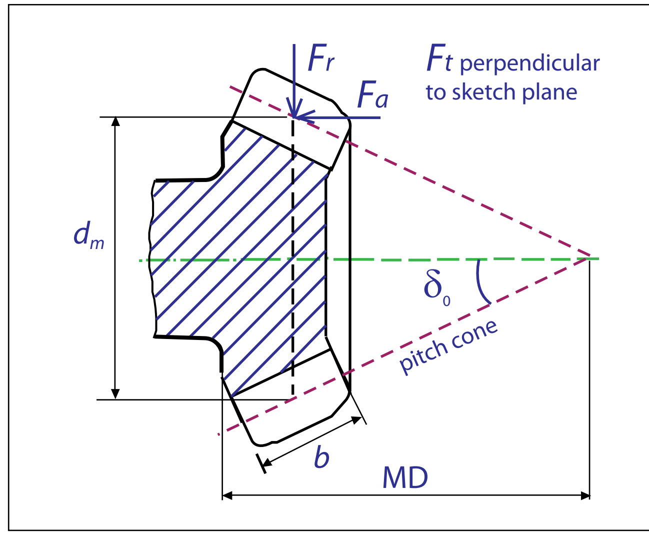

Figure 5—Application point of tooth forces.

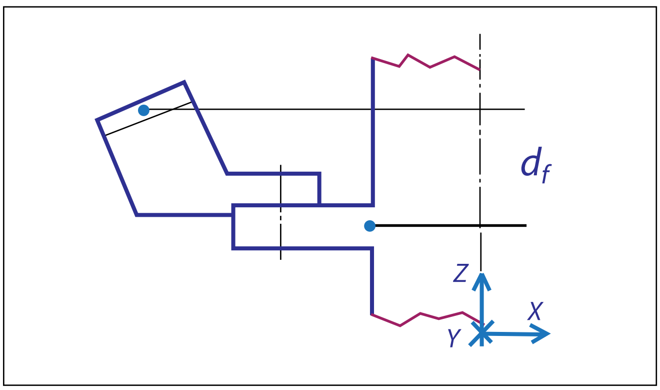

Figure 6—Application point used in shaft calculation.

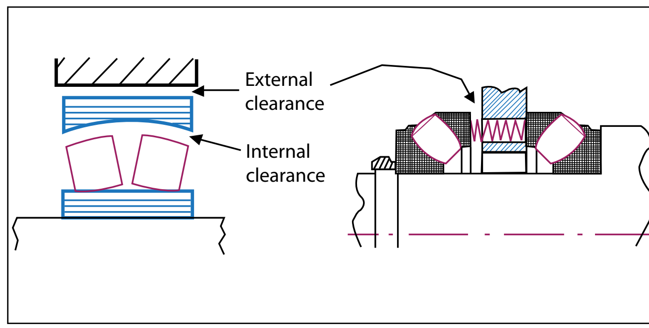

Figure 7—Examples of internal and external bearing clearances.

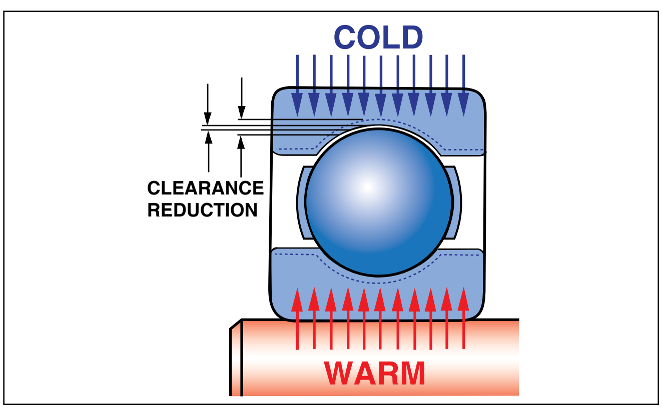

Figure 8—Clearance reduction due to temperature difference (Ref. 3).

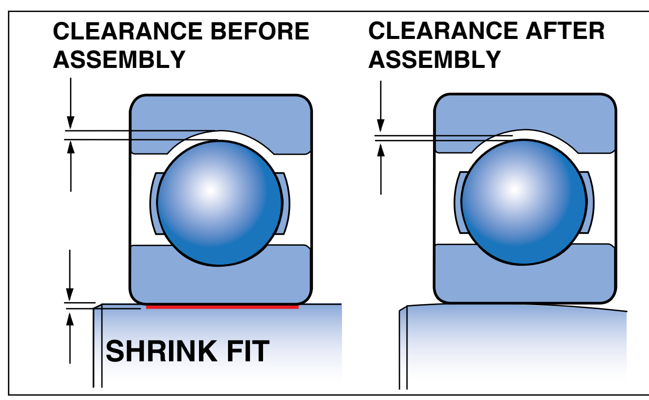

Figure 9—Clearance reduction due to shrink fit (Ref. 3).

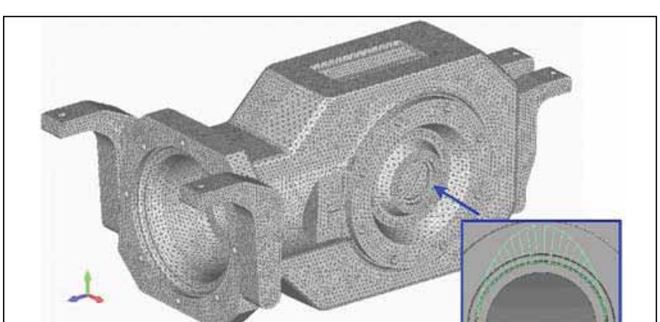

Figure 10—Example of radial bearing load on FE model of housing.

In shaft calculation models used in DLA, bearings are modeled as supports with five degrees of freedom—two radial, two tilt and one axial direction (Fig. 11). The "missing sixth degree of freedom" is the bearing rotation, which is typically of no interest.

Nonlinear bearing stiffness in every direction is modeled starting from deformations of individual contacts between rolling elements and raceways (Fig. 4), also taking into account the internal clearance and operational contact angle. With this modeling method it is possible to accurately predict the distribution of loads and, subsequently, the displacements. Although nonlinear bearing stiffness leads to iterative calculation, the calculation times are minimal due to the analytical theories applied. A more detailed description of the modeling theory can be found in DIN ISO 281, Supplement 4 (Ref. 1).

Loads

Tooth forces are considered as point loads acting on the center of the tooth flank at mean pitch diameter d_m (Fig. 5). Tooth force components F_t, F_r and F_a are calculated based on mean spiral angle β_m, normal pressure angle α_n and pitch cone angle δ. Especially for the wheel, the axial location of the acting point of the tooth forces does not always represent the axial location where the forces are actually conveyed to the shaft. An example of such a situation is presented in Figure 6. In shaft calculation models this is taken into account by transferring the tooth forces axially by a distance of d_f and correspondingly adding two bending moments $My = F_r\, d_f$ and $Mx = F_t\, d_f$. The same principle is used to correctly model bearing reactions in cases with bearings with a non-zero pressure angle (e.g., taper roller bearings).

In addition to tooth forces there are usually external forces that also need to be included, such as propeller thrust force in marine thrusters. They are applied to their appropriate location on the shaft using the same principles as with the tooth forces. The weight of components is seldom important from deformations' point of view, but might instead be significant for other reasons discussed later in this paper.

Bearing Clearances and Pre-Load

Depending on the arrangement, bearing clearance and pretension can have significant influence on gear displacements. In shaft calculation models, values in operating conditions are used, which often differ significantly from assembly values due to temperature differences. Clearances can be divided into *internal* and *external* clearances (Fig. 7).

Internal radial clearances in operating conditions are calculated based on clearance class (e.g., CN, C3, etc.), shrink-fits of bearing rings and temperature difference between inner and outer ring (Figs. 8–9). Both internal and external radial clearances cause shaft displacement, but internal clearance also affects bearing stiffness. Therefore, precise, external radial clearance should be modeled as movement of the outer ring, not as increased internal clearance. However, the significance of this matter is minor. In axial direction, internal and external clearances are basically the same thing. Axial pre-load/external axial clearance in operating conditions is calculated based on initial setting (assembly), temperature difference between shaft and housing, distance of bearings and bearing pressure angle.

External radial clearances are used when bearings need to be free in the axial direction. External axial clearances are sometimes applied to bearings in *O* or *X* arrangement to prevent excessive preloading due to temperature differences. Ideally, these clearances should be reduced to very small values in operating conditions, in which case their influence on displacements would be negligible. However, because temperature differences are usually not exactly known in the design phase (with clearances chosen preferably "on the safe side," i.e.—too large rather than too small) and gear drives are often loaded in different operating temperatures, consideration of external clearances is also a part of DLA.

Deformation of Gear Housing

Deformation of gear housing is considered through FE analysis, performed with commercial software. Bearing reactions obtained from preliminary shaft calculation models are used as loadings for the FE model. Loads are applied to the radial and axial support surfaces of bearings as pressure distributions with resultant forces corresponding to the bearing reactions (Fig. 10). External loads are applied if such exist.

After the FE model is solved, translation and rotation values of bearing bores are extracted from the displacement results. With displacements as the main result, a relatively coarse FE mesh (compared to, for example, stress analysis) is sufficient. In shaft calculation models, bearing bore displacements are described with the same five degrees of freedom as bearing stiffness (Fig. 11)—i.e., three translational and two rotational displacements are used to describe movement of one bearing bore. These values can be extracted from the FE node displacements in different ways. One way is to choose representative nodes with 90° spacing from the support surfaces and calculate the five displacement values from them. A more sophisticated method is to place a node in the middle of the bearing bore and connect it to the cylindrical surface by beam elements with very small axial stiffness and ball joint-type connections at the ends. In this way the displacement of the center node directly represents the sought after values. Similar results can also be achieved by fitting an un-deformed cylinder to the displacement field using a best-fit procedure. All of the mentioned methods have been used successfully as a part of DLA.

After the shaft calculation models are re-run using the bearing bore displacements, changes in bearing reactions are checked. If considerable change is observed, the FE model is

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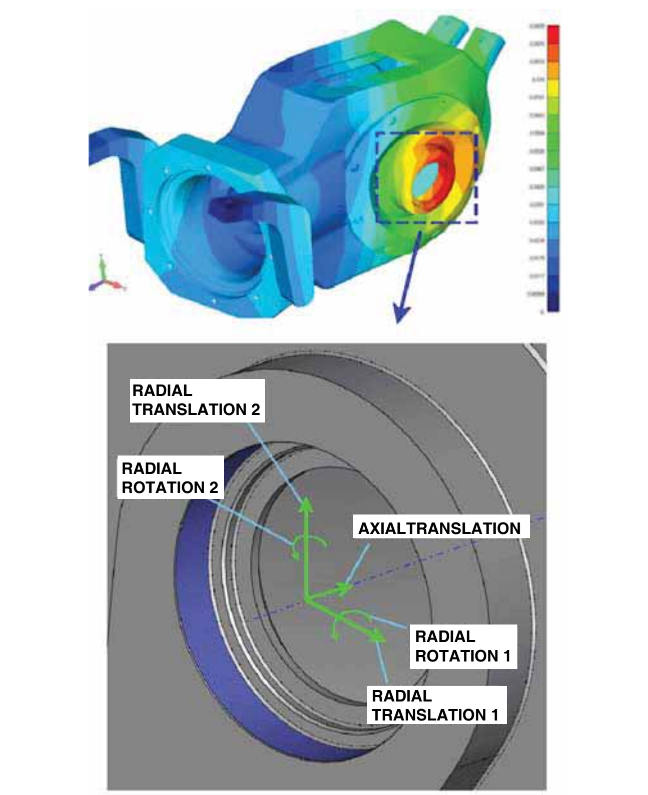


Figure 11—Determination of bearing displacement from FE results.

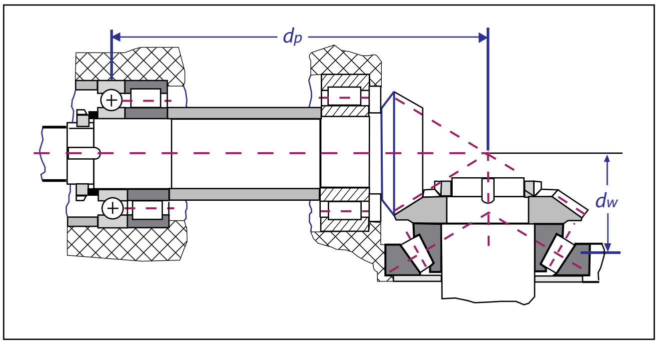


Figure 12—Effective distance for temperature difference between housing and shafts.

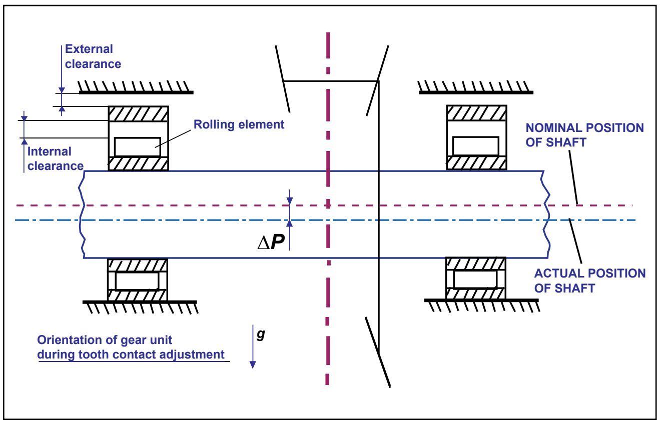
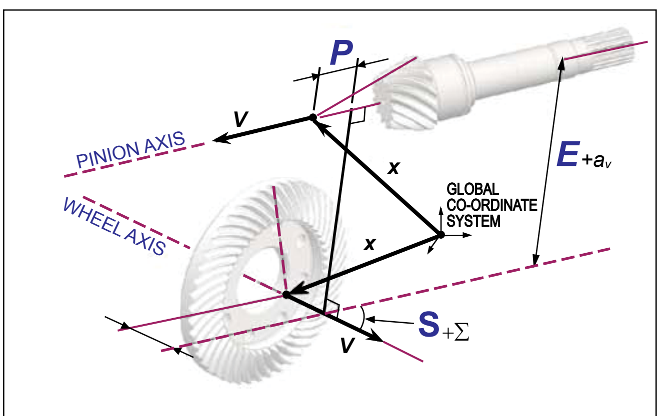


Figure 13—Example of additional displacement (P) due to gear unit orientation during tooth contact adjustment.



$$\mathbf{S} = a\cos\left(\frac{\bar{v}_1 \bullet \bar{v}_2}{\|\bar{v}_1\| \bullet \|\bar{v}_2\|}\right) \cdot \frac{180}{\pi} - \Sigma \qquad \mathbf{E} = \frac{|(\bar{x}_1 - \bar{x}_2)\ (\bar{v}_1 \mathrm{x} \bar{v}_2)|}{\|\bar{v}_1 \mathrm{x}\ \bar{v}_2\|} - a_v$$

$$\frac{\|(\bar{x}_p - \bar{x}_2) \mathrm{x}\ \bar{v}_2\|}{\|\bar{v}_2\|} = \mathrm{E} \ \text{,where } \bar{x}_p = \bar{x}_2 + \mathbf{P} \cdot \bar{v}_1$$

$$\frac{\|(\bar{x}_p - \bar{x}_2) \mathrm{x}\ \bar{v}_2\|}{\bar{v}_1} = \mathrm{E} \ \text{,where } \bar{x}_p = \bar{x}_2 + \mathbf{G} \cdot \bar{v}_2$$

Figure 14—Arbitrary position of gear member described with vectors.

no longer valid and the process is repeated. Typically, only one iteration is required.

Effect of Temperature Differences to Axial Location of Bevel Gears

In addition to bearing clearances and pretension, temperature differences also affect the axial location of bevel gears. The significance is strongly dependent on the material of the housing and the distance between the bevel gear centerlines and axial bearing location (Fig. 12).

Effect of Gear Drive Orientation during Tooth Contact Adjustment

Usually during the assembly of gear drives no thermal differences exist. Therefore, bearing clearances (internal and external) have significantly larger values compared to operating conditions. In shaft calculation models it is assumed that without load, shafts are initially in their nominal position—i.e., "hovering" in the middle of bearing clearances in the radial direction. However, during tooth contact adjustment, the position of shafts might differ significantly from this assumption due to the clearances and weight of components. This is especially true in cases where enlarged clearances are used due to expected high-temperature differences. The matter is illustrated with an example in Figure 13 where the horizontal shaft is displaced from its nominal position due to gravity and internal/external bearing clearances. This corresponds to an additional change in relative displacement *P*. Generally, also *E*, *G* and *S* displacements can be affected depending on the bearing arrangement and orientation of the gear unit. In addition to clearances in radial direction, axial clearances can also cause additional displacements. For example, in situations with spring-loaded axial clearance (Fig. 7), the clearance during assembly might lie on the opposite side as compared to operating conditions. If significant displacements from the nominal position of shafts are to be expected during tooth contact adjustment, they are taken into account in the calculations. This requires knowledge of the orientation of the gear drive during tooth contact adjustment.

In addition to clearances, bearing deformation can also contribute to the gravity-induced additional displacements. One example is a taper roller bearing (small pressure angle) without preload that has a small axial stiffness that can lead to axial movement of the shaft if it has been subjected to large axial force from heavyweight components.

Calculation of Relative Displacements of Gear Members

After the relative displacements have occurred, pinion and wheel are considered to be in arbitrary position in three dimensional space, which can be described by location and direction vectors x_1, x_2, v_1 and v_2 (Fig. 14). These vectors are extracted from the results of shaft calculation models—i.e., deflections and inclinations of the neutral axis at the locations of the bevel gear teeth (Fig. 15). Relative displacements *E*, *P*, *G* and *S* are then calculated from the vectors using basic vector algebra.

When determining the vectors, care has to be taken so that displacements from correct axial location on the shaft are used. As Figure 6 demonstrates, the axial location of tooth forces does not always coincide with the location that determines displacements of the bevel gear. The significance of this is magnified in cases where the shaft inclination changes rapidly near the location of the bevel gear (Fig. 16).

Application of Analysis Results

Relative displacements are used in tooth geometry optimization—which is based on LTCA. In most cases, the goals of optimization are related to noise, power density, robustness and efficiency. Regardless of the goal, knowledge of the relative position of gears under load is a valuable piece of information. For instance, in optimization for stresses the goal is to distribute load evenly on the tooth and to utilize as much of the tooth flank area as possible. When the relative displacements become known, a centralized location of tooth contact under load can be assured. Therefore the portion of crowning that was previously intended to prevent edge contact—due to unknown movement of tooth contact—can be reduced.

The simplest case of optimization is the one with constant load and operating conditions. The situation becomes more complex when multiple load levels and different temperature conditions have to be considered. In such cases drive line analysis is repeated several times, with different input. The resulting, optimal tooth geometry might be a compromise between several different load cases. Figure 17 is an example of such a situation in an automotive application. With a 50% load, the contact approaches the toe; and with a 100% load, the heel. In this case the crowning and initial contact pattern location were chosen so that a satisfactory compromise between different loaded conditions was obtained.

In reality the actual relative position of gears under load will vary in a certain range—even in cases with single load. To some extent, the tooth geometry should be designed so that it can withstand these variations; the essential sources of variations include:

- Variation in tooth finish machining
- Variation in tooth contact adjustment during assembly
- Variation of housing machining and assembly clear ances between components
- Variation of temperature and loading conditions

The first two of the mentioned sources of variations are strongly operator-specific, especially when the correctness of tooth contact is judged by applying a marking color on the tooth flanks and observing the contact pattern after rotation of the gears. In tooth finish machining, variation is also affected by repeatability related to machine kinematics and tool settings. In cases where gears are assembled to certain mounting distances without checking the contact pattern, variation is determined by a tolerance stack-up of related components.

Variation related to manufacturing deviations and assembly clearances usually increases with the number of mounting surfaces between bevel gears and housing (Fig. 18). These deviations are distinguished from the gear unit's orientation-dependent deviations discussed earlier by the fact that they are "locked" during assembly and will not change after the bolts are tightened. In shaft calculation models it is assumed that relative positions (say, perpendicularity and concentricity) of all mounting surfaces are free from deviations. Furthermore, the clearances between housing components (such as bearing carriers and main housing) are not considered. Indeed, all components are assumed to be situated in the middle of their

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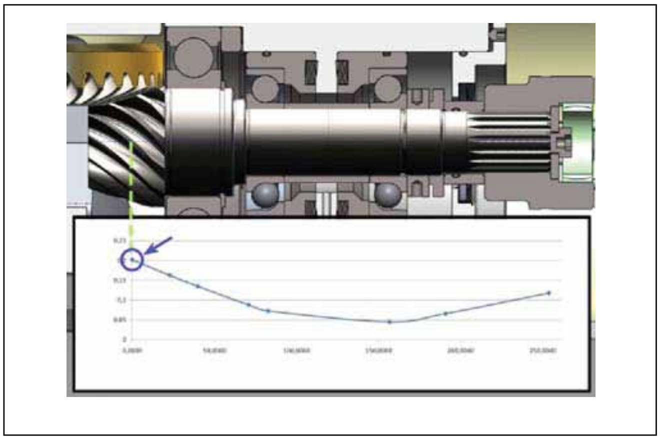


Figure 15—Location on neutral axis where shaft displacements are taken.

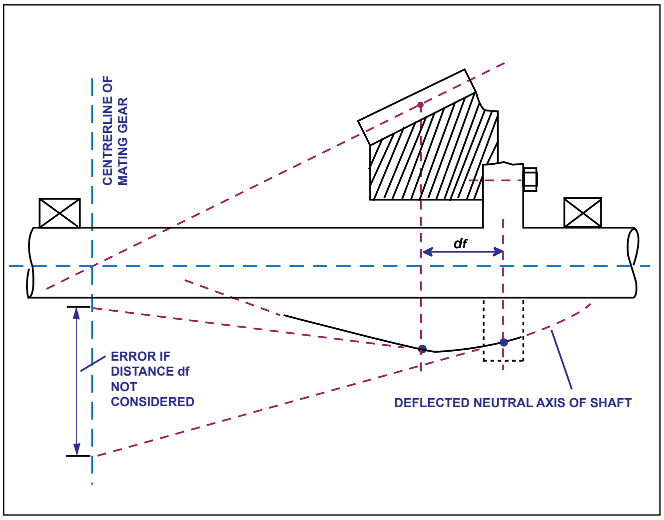


Figure 16—Shaft inclination changes.

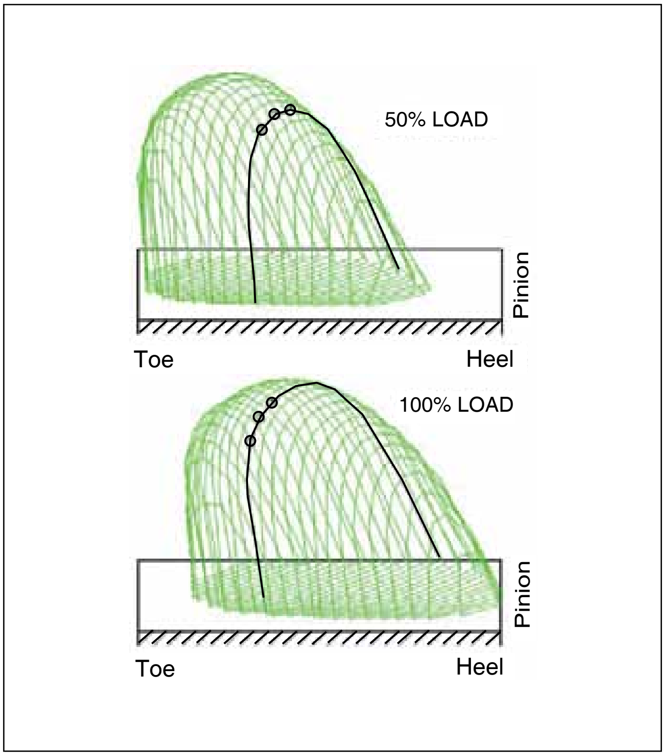


Figure 17—Tooth contact optimization in different load cases.

assembly clearances.

For each application, probable ranges of the abovementioned variations are determined and tooth contact's sensitivity to them is analyzed. The results of this kind of sensitivity study can either be used to specify appropriate tolerances for the mentioned variations or to modify the tooth geometry to accommodate for known tolerances.

Validation of Analysis Methods

To demonstrate the validity of the described analysis procedures, two actual example cases are presented. The demonstration is done by comparing actual, documented loaded contact patterns to ones determined by LTCA using the calculated, relative displacements. LTCA is performed using *Becal* software (Ref. 4). In *Becal* the simulation of mesh under load is based on a combination of analytical and numerical methods that have been calibrated against experimental data as well as numerical reference calculations (Ref. 5).

In both example cases the relative displacements of gears were determined by the analysis procedures described in this paper and used in LTCA. Actual topography of tooth flanks was measured with a coordinate measuring machine and used in LTCA. The calculation methods of *Becal* require that the measured topography deviations are approximated by a second-order surface. Accuracy of the approximation declines when the deviations form a complex surface. In all of the examples, the deviations were such that the approximation was able to represent reality with reasonable accuracy. To improve the accuracy, areas of the tooth flanks that were concluded to be free of contact were excluded from the surface fitting procedure.

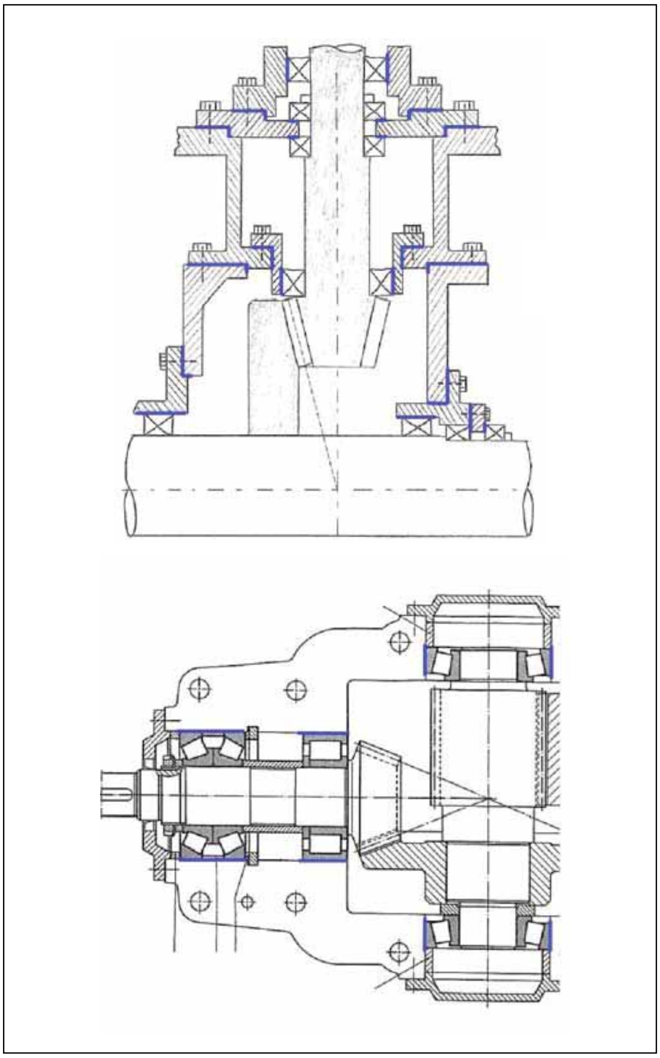


Figure 18—Examples of constructions with large and small number of mounting surfaces (marked with blue).

Example 1: ***Full-torque test of a marine thruster upper gear unit***. Actual loaded contact pattern was documented in a full-torque test under quasi-static conditions (slow roll). Temperature differences were nonexistent and therefore omitted from analysis. Basic construction of the gear unit and results of DLA are presented in Figure 19. Comparison of actual and calculated contact patterns showed good agreement (Fig. 20). Note that the contact patterns calculated by Becal are presented in radial projection, but all dimensions are given along tooth arc.

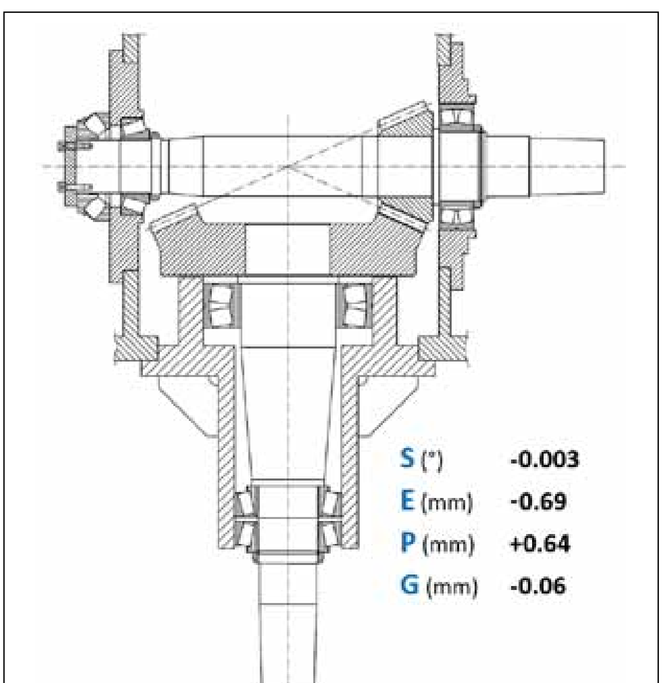


Figure 19 (Example 1)—Basic drive line construction and results of drive line analysis.

The results of DLA were also verified by directly measuring shaft displacements during full-torque testing of another gear drive of the same type. Figure 21 shows that the there was good agreement between calculation and reality. The effect of housing deformation on measurements (translation and rotation of the surface to which the dial indicator was attached) was taken into account to enable valid comparison.

Example 2: ***Endurance test of bevel gears of an industrial gear unit***. In this case actual loaded contact patterns were documented during an endurance test of an industrial gear unit (Fig. 22). Comparison to calculated results showed good agreement (Fig. 23). It should be noted that in this case the tooth contact pattern was not yet optimized.

Without consideration of relative displacements, the calculated contact pattern for the drive side would have looked

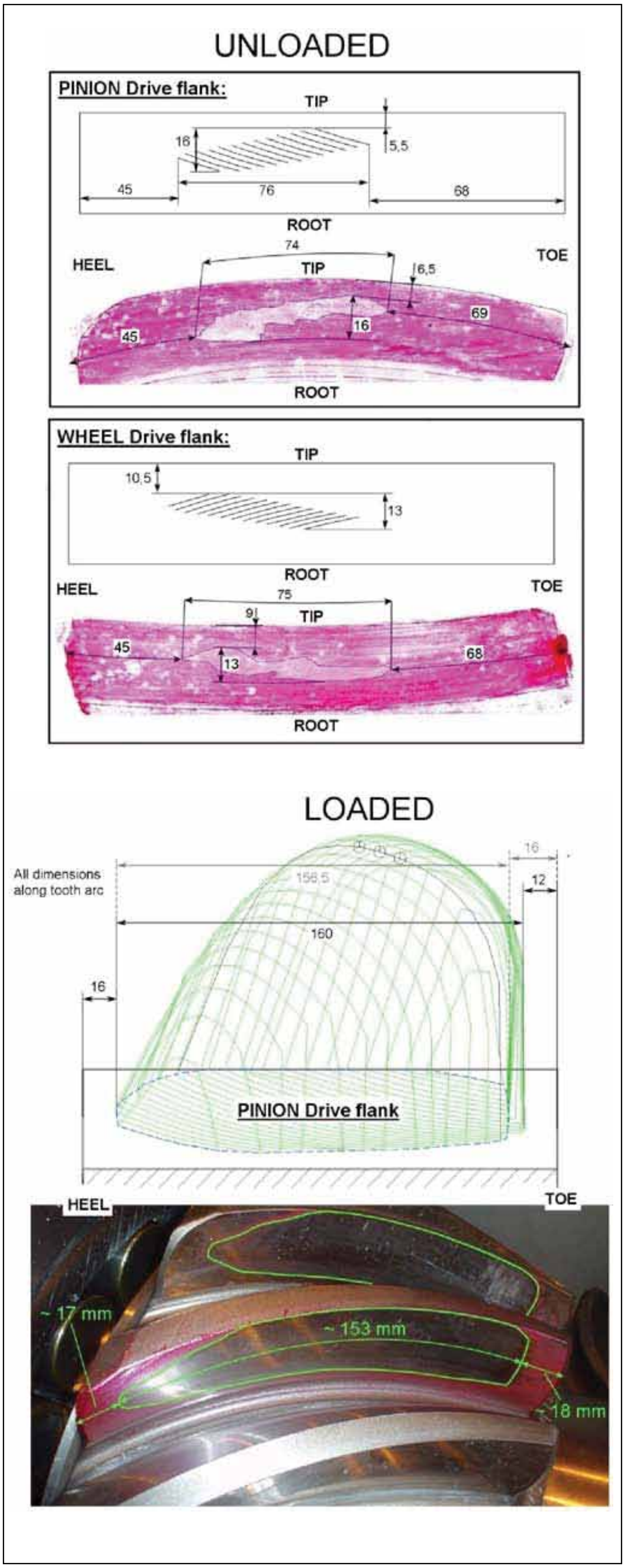


Figure 20 (Example 1)—Comparison of actual and calculated contact patterns.

Figure 21—Comparison of measured and calculated shaft displacements.

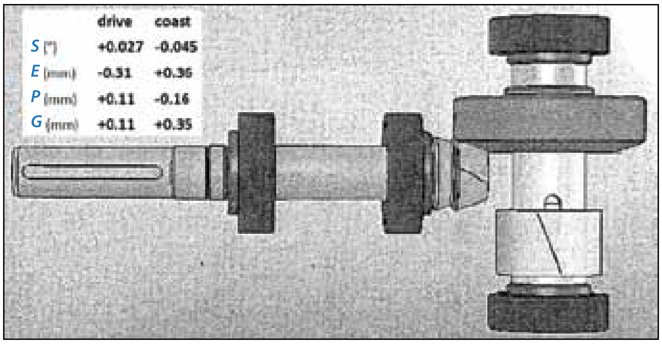


Figure 22 (Example 2)—Basic drive line construction and results of drive line analysis.

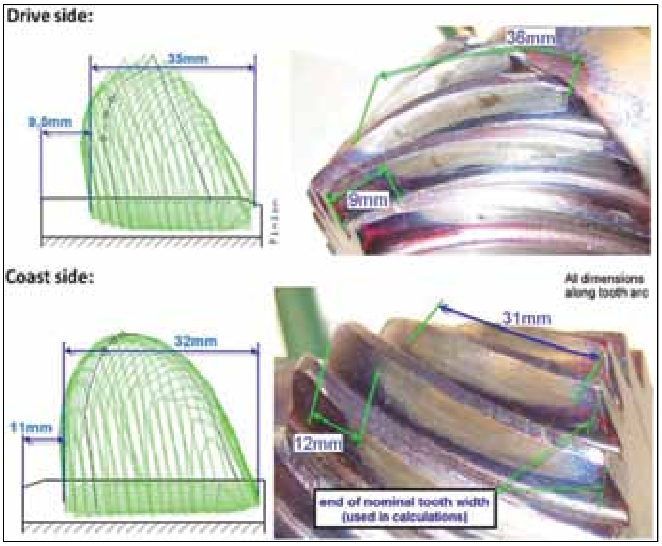


Figure 23 (Example 2)—Comparison of actual and calculated contact patterns under load.

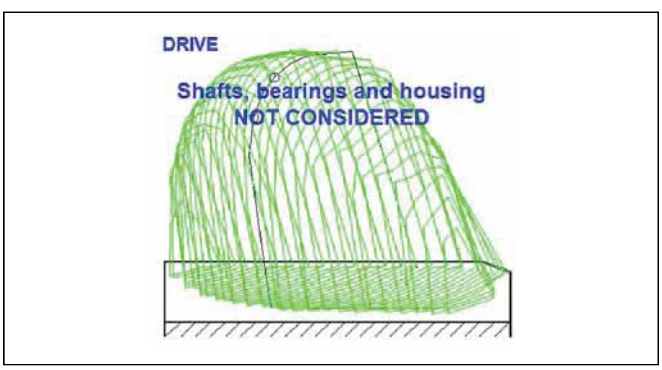


Figure 24—Calculated contact pattern without consideration of relative displacements.

as presented in Figure 24. This illustrates the significance of drive line analysis in this particular case.

Conclusions

In this paper computational analysis procedures for determining relative displacements of spiral bevel gears under load have been presented. The method has been verified by comparison with actual test data.

Significance of the presented factors (e.g., bearing clearances) greatly varies by application. The described analysis method is used to analyze gear drive constructions of different designs. Therefore the goal has been to make the analysis process generally applicable, containing as many of the influencing factors as possible and regardless of their significance. It is the gear designer's responsibility to assess which factors are relevant, but it has also been seen that many factors with little effect can add up to a significant one.

Outlook: Subjects for Future Study

1. ***Effect of dynamic loading***. The following questions are yet to be answered: Is the quasi-static approach presented in this paper sufficient to represent dynamic situations? How does the relative position of gears vary during vibration? How should application factor K_A and dynamic factor K_v be dealt with in drive line analysis?

2. ***Variation of tooth forces***. Currently the tooth forces are considered as components F_t, F_r and F_a that are calculated at d_m using β_m, α_n and δ. Friction is not considered and the tooth forces are assumed to act on the same point at all times. In reality the resultant of tooth forces is comprised of multiple pressure distributions acting on different tooth flanks. Location, direction and magnitude of the resultant vary during mesh; the effect of this variation to gear displacements should be assessed.

3. ***Comparison to FEM-based LTCA***. Thus far the validation based on contact patterns has been limited by the accuracy of the *Becal* program. *Becal's* semi-analytical calculation approach results in short calculation times—but also limits accuracy. Another limitation is the second-order surface approximation method used to model actual tooth flank topography. By using more accurate methods—such as a nonlinear FE model—more accurate data could possibly be obtained. In near future, an in-house-developed FE code (Ref. 6) will be used for this task.

Acknowledgements

Great gratitude is expressed to the customers with whose permission the pictures of the two example cases were presented in this paper.

References

1. DIN ISO 281Supplement Four. Rolling Bearings/Dynamic Load Ratings and Rating Life: Methods for Calculation of the Modified Reference Rating Life for Universally Loaded Rolling Bearings, Berlin, 2003, Deutsches Institut für Normung.
2. FAG Bearing Catalog, December, 2008.
3. SKF Bearing Catalog, January 2004.
4. Linke, H. et al. "The Development of the Program *BECAL*–an Efficient Tool for Calculating the Stress of Spiral Bevel Gears," *International Conference on Mechanical Transmissions*, April 5–9, 2001, Chongqing, China.
5. Baumann, V. and J. Thomas. "Grundlagen zur Ermittlung der Zahnflanken and Zahnfussbeanspruchung Bogenverzahnter Kegelräder auf der Basis Experimentell Gestützter Näherungsbeziehungen," Frankfurt: Forschungs–Vereinigung Antriebstechnik, 1995, Abschlussbericht (FVA–Heft 429).
6. Szanti, G. "Method for Designing Silent Running Spiral Bevel Gears Using Loaded Tooth Contact Analysis," *JSME International Conference on Motion and Power Transmissions*, May13–15, 2009, Matshushima Isles Resort, Japan.

Tribology Aspects in Angular Transmission Systems

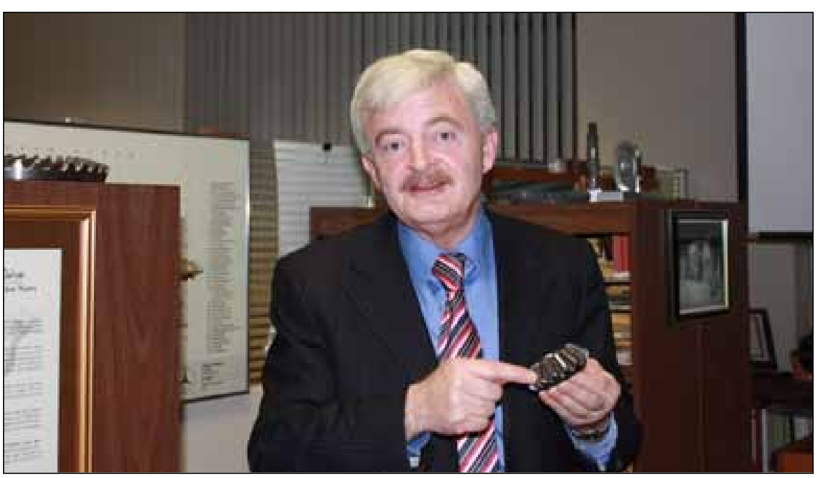
Part VII: Hypoid Gears

Dr. Hermann Stadtfeld

(This article is part seven of an eight-part series on the tribology aspects of angular gear drives. Each article (chapter) is a Gear Technology *exclusive. The entire series will appear in book form in Dr. Stadtfeld's upcoming work on the subject, scheduled for release in late 2011.* Gear Technology *(Randall Publications) is proud and most grateful to Dr. Stadtfeld and the Gleason Corporation for choosing this magazine as the platform for presenting this very informative and most-relevant series of "teach-ins" to its readers.)*

Design

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



Dr. Hermann Stadtfeld *received a bachelor's degree in 1978 and in 1982 a master's degree in mechanical engineering at the Technical University in Aachen, Germany. He then worked as a scientist at the Machine Tool Laboratory of the Technical University of Aachen. In 1987, he received his Ph.D. and accepted the position as head of engineering and R&D of the Bevel Gear Machine Tool Division of Oerlikon Buehrle AG in Zurich, Switzerland. In 1992, Dr. Stadtfeld accepted a position as visiting professor at the Rochester Institute of Technology. From 1994 until 2002, he worked for The Gleason Works in Rochester, New York—first as director of R&D and then as vice president of R&D. After an absence from Gleason between 2002 to 2005, when Dr. Stadtfeld established a gear research company in Germany and taught gear technology as a professor at the University of Ilmenau, he returned to the Gleason Corporation, where he holds today the position of vice president-bevel gear technology and R&D. Dr. Stadtfeld has published more than 200 technical papers and eight books on bevel gear technology. He holds more than 40 international patents on gear design and gear process, as well as tools and machines.*

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

To establish line contact between the pitches in hypoid gears, the kinematically correct pitch surfaces have to be determined based on the axoids. In cylindrical and bevel gears, the axoids are identical to the pitch surfaces and their diameter or cone angle can be calculated simply by using the knowledge about number of teeth and module or ratio and shaft angle. In hypoid gears, a rather complex approach is required to find the location of the teeth—even before any information about flank form can be considered.

The offset in hypoid gears introduces a screw motion along a screw axis H_0-H_0 (Fig. 1). The screw axis has a certain angular orienta-

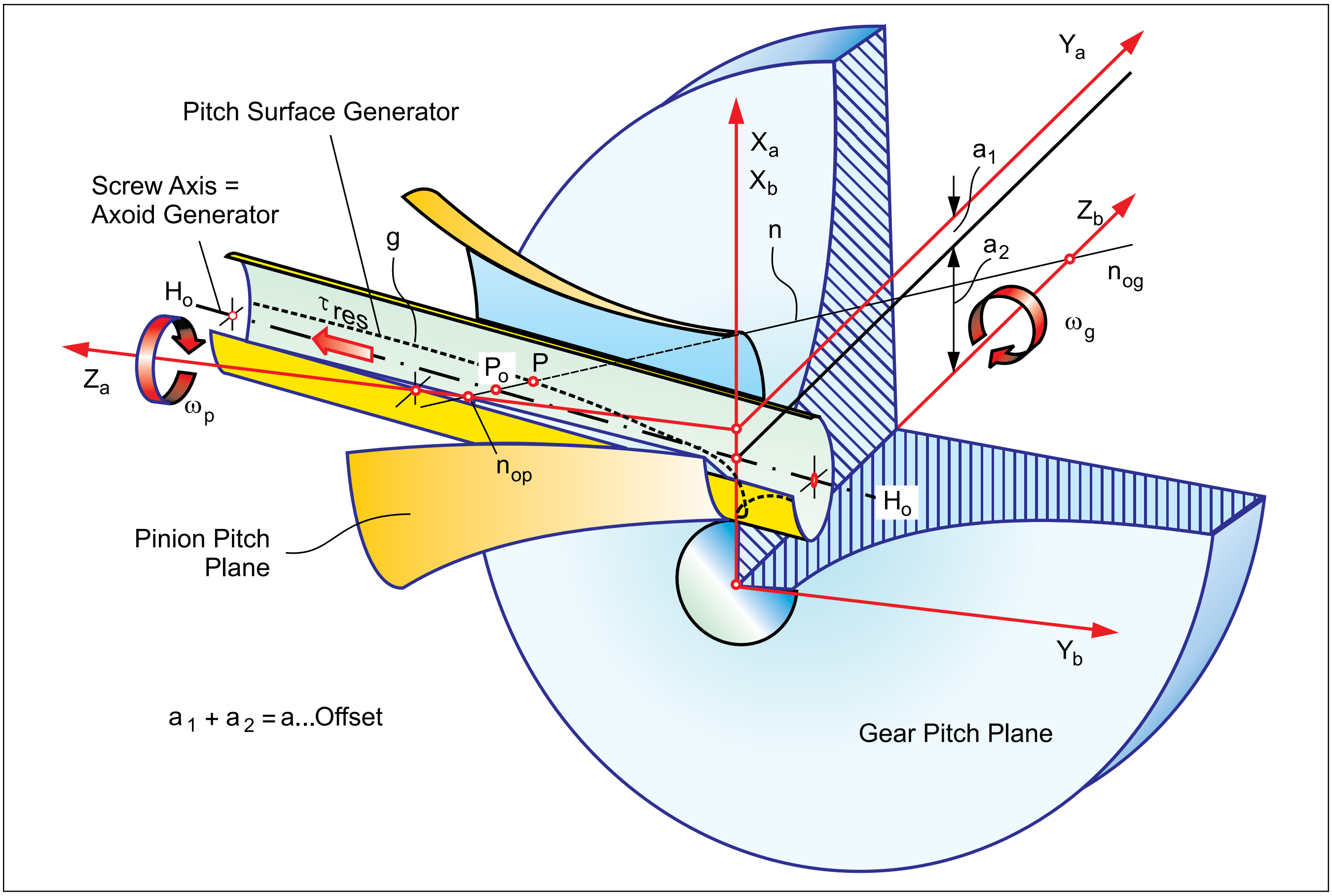


Figure 1—Axoids and pitch surfaces in gears with screw motion.

tion and vertical position, caused by the split of the offset a into a_1 and a_2. The screw axis is the generator of the pinion axoid when rotating around pinion axis Z_a. It is also the generator of the gear axoid when rotating around Z_b. The axoids are equidistant surfaces to the pitch surfaces. A connecting line between the two axes of rotation—Z_a and Z_b (line np-ng)—intersects with the screw axis H_0-H_0 in point P_0 and is perpendicular to the common pinion and gear pitch surface point P. If the line n_p is moved along screw axis H_0-H_0, (and point P_0 with it) the pitch surface generating line becomes evident as a curve g on the surface of a cylinder in order to establish a pitch surface that is equidistant to the axoids. In all real-world applications the pitch surfaces are defined as cones that are smoothed onto the hyperboloids in the calculation point P (Fig. 1). The screw motion results in a length-sliding that is present in any flank surface point and is superimposed to the profile-sliding known from straight bevel and spiral bevel gears (Refs.1–2).

The axes of hypoid gears, in most cases, cross under an angle of 90°. This so-called shaft angle can be larger or smaller than 90°, where shaft angles above 90° can lead to internal ring gears that are often limited in their manufacturability due to cutter interference. But the axes of hypoid gears do not intersect and the smallest distance between them is called the hypoid offset. The shaft angle is defined in a plane perpendicular to the offset direction (Fig. 2, right).

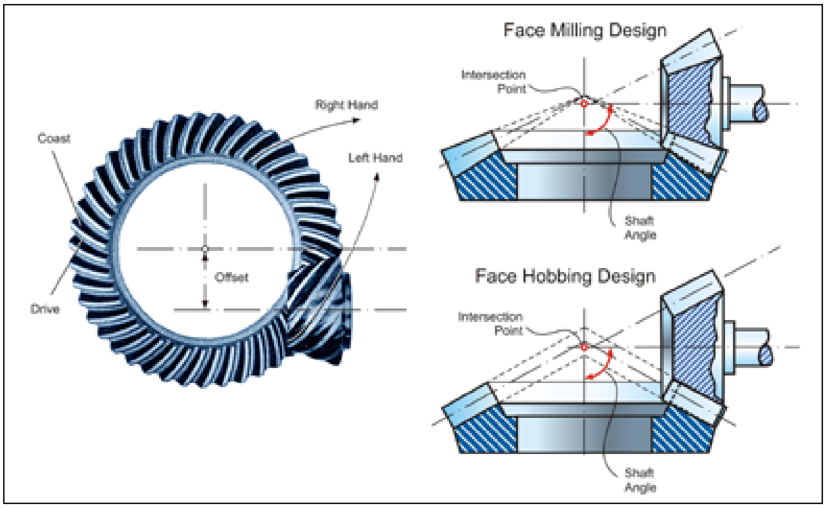


Figure 2—Hypoid gear geometry.

Hypoid gears have a parallel-depth profile along the face width if they are manufactured in the continuous face hobbing process or a

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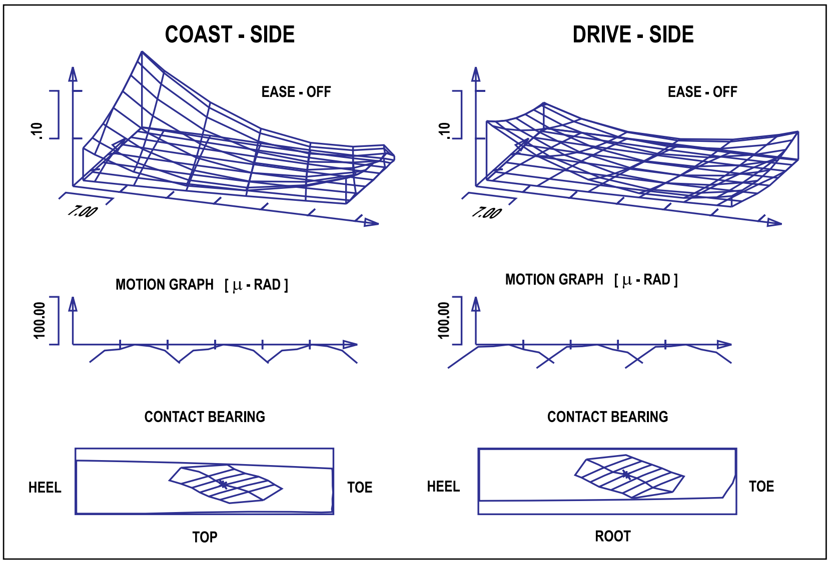

Figure 3—Tooth contact analysis (TCA) of a hypoid gear set.

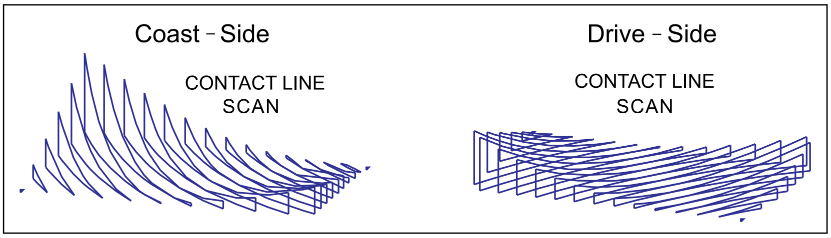

Figure 4—Contact line scan of a hypoid gear set.

tapered-depth profile along the face width if the manufacturing is done using the single-indexing face milling process.

Hypoid gear teeth follow in face width direction a curve on the conical gear and pinion body that lies under an angle to a cone element (spiral angle). The tooth lead function in face width direction—if unrolled into a plane—is an epicycloid or a circle, depending on the manufacturing method.

The photo of a hypoid gear set in Figure 2 explains the definition of right-hand and left-hand spiral direction and indicates the coast- and drive-side gear flanks. The cross-sectional drawings to the right in Figure 2 illustrate the blank design for face milling on top (tapered depth teeth) and face hobbing design at the bottom (parallel depth teeth).

Analysis

Since the mentioned distortions in tapered-depth tooth systems are detected by comparison to conjugate mating flanks, it is possible to define potential contact lines that would apply in case the distortions are removed or in case of load-affected deflections that allow for a contact spread. In order to allow for deflections of tooth surfaces, shafts, bearings and gear box housing without unwanted edge contact, a crowning in face width and profile direction is applied. A theoretical tooth contact analysis (TCA) previous to the gear manufacturing can be performed in order to observe the effect of the crowning in connection with the basic characteristic of the particular gear set. This also allows the possibility of returning to the basic dimensions in order to optimize them if the analysis results show any deficiencies. Figure 3 shows the result of a TCA of a typical hypoid gear set.

The two columns in Figure 3 represent the analysis results of the two mating flank combinations (*see also "General Explanation of Theoretical Bevel Gear Analysis"*). The use of the drive-side as main load transmission direction is for hypoid gears a rather binding rule. Transmission of torque and speed and the additional length sliding forces lead on the coast-side to a pinion deflection towards the ring gear, which reduces the backlash in extreme cases to zero. This situation occurs already under moderate load and interrupts any lubrication that results in surface damages and may be followed by tooth fracture.

The top graphics show the ease-off topographies. The surface above the presentation grid shows the consolidation of the pinion and gear crowning. The ease-offs in Figure 3 have a combination of length crowning, profile crowning and flank twist, and result in a clearance along the boundary of the teeth being established.

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

At the bottom of Figure 3, the tooth contact pattern is plotted inside of the gear tooth projection. These contact patterns are calculated for zero load and a virtual marking-compound film of 6 µm thickness. This basically duplicates the tooth contact one could observe rolling the real version of the analyzed gear set under light load on a roll tester, while the gear member is coated with a marking compound layer. The contact lines lie under an angle to the face width direction, depending basically on the spiral angle. The path of contact connects the beginning and end of meshing. Its orientation is nearly perpendicular to the contact lines.

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

Figure 4 shows 20 discrete, potential contact lines, with their individual crowning amounts along their length (contact-line scan). The gap geometry in contact-line direction can be influenced by a change in ease-off topography, and optimized regarding the gap kinematic cases (*see also "General Explanation of Theoretical Bevel Gear Analysis*," Fig. 8). The gap geometry perpendicular to the contact line direction (not exactly the same as the path of contact direction) does not significantly depend on the ease-off topography but is mainly dominated by the geometry of the mating tooth profiles. Typical for hypoid gears is the lubrication gap change from contact line to contact line. Effects like those discussed in cases 5 and 6 are likely to be applicable in hypoid gears and can also be controlled to some extent in hypoid ease-off developments.

Figure 5 shows the sliding- and rolling-velocity vectors of a typical hypoid gear set for each path-of-contact point for the 20 discussed roll positions; each vector is projected to the tangential plane at the point-of-origin of the vector. The velocity vectors are drawn inside the gear tooth projection plane. The points of origin of both rolling- and sliding-velocity vectors are grouped along the path-of-contact, which is found as the connection of the minima of the individual lines in the contact-line scan graphic (Fig. 4). The velocity vectors can be separated in a component in contact-line direction and a component perpendicular to that in order to investigate the hydrodynamic lubrication properties by applying the information from the contact-line scan (curvature and curvature change) and the tooth surface curvatures perpendicular to the contact-line direction (*see also "General Explanation of Theoretical Bevel Gear Analysis," Figure 8, cases 1–6*).

In the case of the discussed hypoid gear set, the sliding-velocity vectors are length-oriented because of the high screw motion

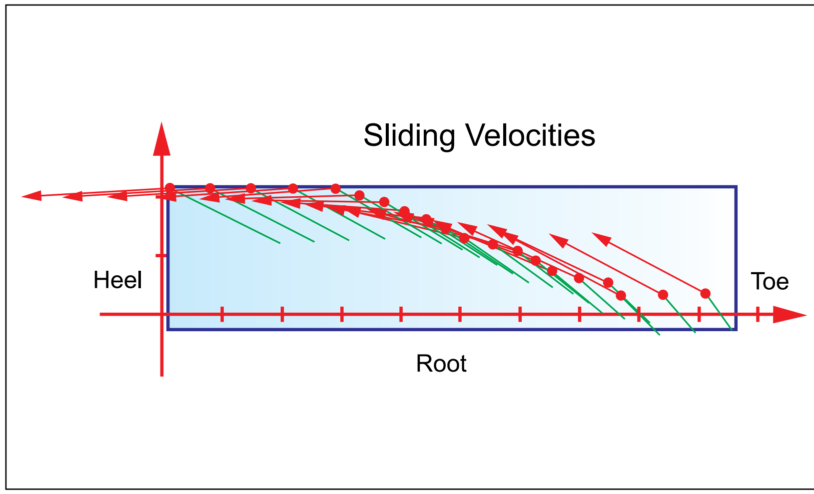

Figure 5—Rolling and sliding velocities of a hypoid gear set along the path of contact.

component. In the top area (top, left) the sliding vectors point to the left and slightly to the bottom (pinion gear drive on the drive-side). Moving along the path-of-contact from top to bottom (left to right, in Figure 5), the sliding velocity reduces its profile component and reaches a purely length-oriented magnitude at the pitch line. Below the pitch line the sliding velocity develops a positive profile component. The maximal magnitudes of the sliding velocities are dependent on the location of the pitch line in the profile direction at one of the extreme ends of the path of contact (top heel or root toe). The top-oriented pitch line in this example leads to the largest sliding velocities in the root area. The rolling vectors point down and to the right and have basically all the same direction. The small change in orientation is a result of the spiral angle that changes along the face width. The shrinking magnitude of the rolling velocity (moving from heel-top to toe-bottom) is caused by the decreasing circumferential speed towards the inner diameter.

It therefore becomes evident that a complex gap and velocity evaluation in a variety of discrete points, and considering the two principal curvature directions, is important in hypoid gears in order to achieve reliable results regarding lubrication mechanics.

Manufacturing

Hypoid gears are manufactured in a continuous-indexing face hobbing process or in a single- indexing face milling process. In the face milling process the blades are oriented around a circle and pass through one slot (while they plunge or generate the flanks of that particular slot), as illustrated in Figure

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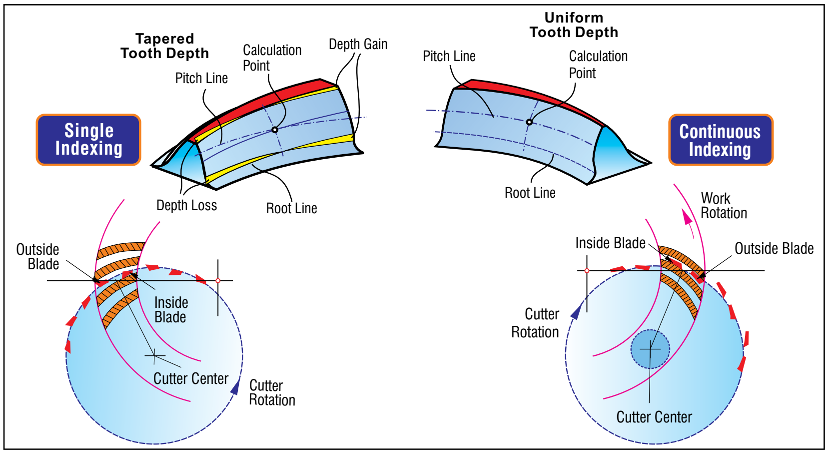

Figure 6—Left: Face milling; Right: Face hobbing.

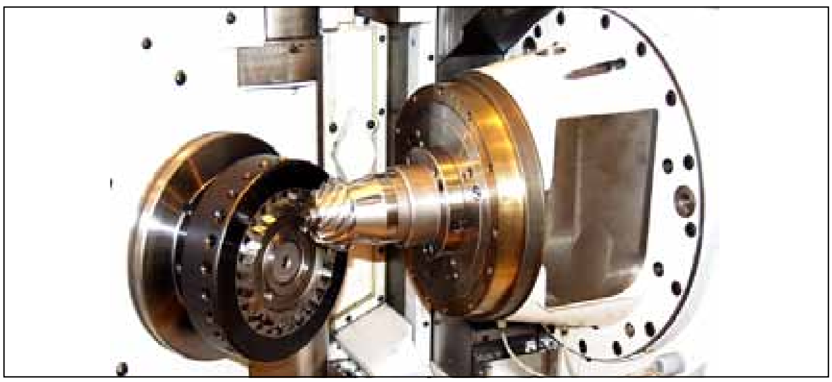

Figure 7—Hypoid pinion cutting with face hobbing cutter (continuous process).

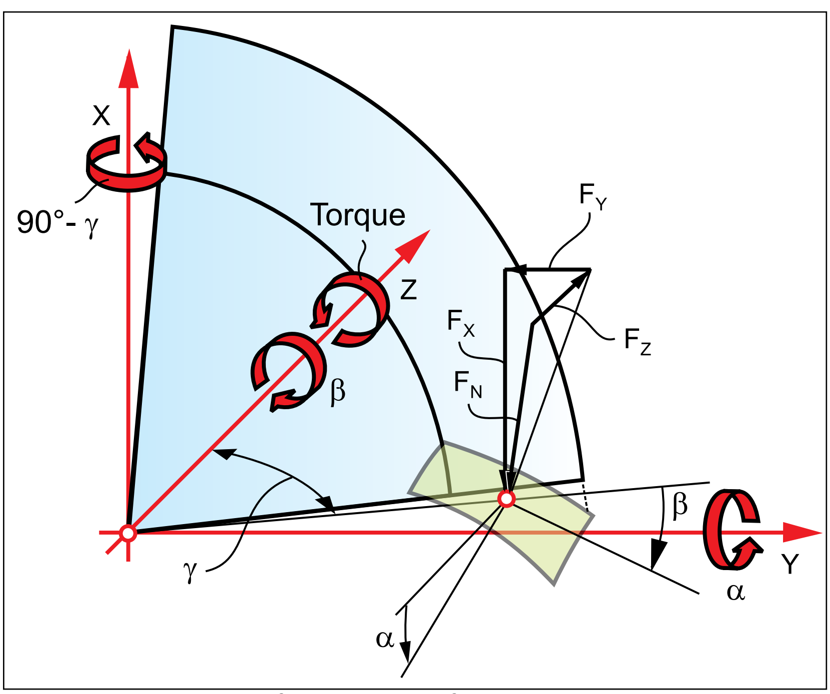

Figure 8—Force diagram for calculation of bearing loads.

6, left. The work is not performing any indexing rotation. At the blade tip and in equidistant planes (normal to the cutter head axis) the slot width produced has a constant width between toe and heel. In order to achieve a proportionally changing slot width (and tooth thickness), the root line of face milled bevel gears is inclined versus the pitch line (Fig. 6, left). This modification must be implemented in both members—which is the reason the face angle requires the same modification as the root angle of the mating member (Ref. 1).

In face hobbing (Fig. 6, right) there exists a group of mostly one inside and one outside blade passing through one slot, while the work rotates with:

$\omega_{Work} = \omega_{Cutter}$ • (Number of Cutter Blade Groups)/(Number of Work Teeth)

Due to the relative motion, the following blade group passes through the next slot. The blades in one group are positioned along a spiral, where the sum of the blade groups is oriented around a circle equidistant to the cutter head center. With the described kinematic, the flank lines of the outer and inner flank are epicycloids that divide slot width and tooth thickness in equal fractions of the circumference at any point along the face width. The result is a "natural" slot width taper proportionate to the distance from the pitch apex (Ref. 3). A root angle modification is not required—or useful—because of the already-existing perfect fit of mating teeth and slots.

Figure 7 shows a view into the work chamber of a free-form bevel and hypoid gear cutting machine during high-speed dry cutting of a spiral bevel pinion. The face cutter head has coated carbide stick blades arranged in blade groups for a continuous face hobbing process.

Hard finishing after heat treatment of face milled hypoid gears is generally done by grinding. The grinding wheel resembles the cutter head geometry, while the grinding machine uses the same set-up geometry and kinematic as the cutting machine for the previous soft machining. Hard finishing of face hobbed hypoid gears is generally done by lapping. Pinion and gear are rolled under light torque while a lapping compound of a silicone carbide oil mixture is present between the flanks. Lapping embeds abrasive grain in the flank surfaces that might lead to problems such as wear, temperature and lowered efficiency. The lapping process is better suited for hypoid gears than for all other types of bevel gears due to the significant sliding velocities of hypoids in the face width direction. A key benefit of the good lapping results of hypoids is that this length sliding is not interrupted in the profile direction. The lapping results regarding tooth bearing and motion error—i.e., noise emission—are usually very good.

Application

Most hypoid gears are manufactured from carburized steel and undergo case hardening to a surface hardness of 60 Rockwell C (HRC) and a core hardness of 36 HRC. Because of the higher pinion revolutions, it is advisable to give the pinion a higher hardness than the ring gear; e.g., pinion 62 HRC, gear 59 HRC. This will also reduce the affinity between the pinion and gear flank surfaces and therefore reduce the risk of scoring. An example of the appearance of a scored hypoid ring gear surface is documented in Figure 9.

Regarding surface durability, hypoid gears present a high requirement on surface finish and lubrication because of the sliding velocities. An advantage is the fact that the sliding velocities are not zero at the pitch line, which will maintain a surface-separating lubrication film. However, the sliding in root and top areas is dependent upon an extremely high offset that may lead to scoring (Fig. 8) that can destroy the tooth surfaces and even lead to tooth flank fracture. The correct high-pressure hypoid oil is mandatory for hypoid gears.

The six advantages of hypoid gear sets are:

1. *Welcome design freedom*, such as lowering the center of gravity of vehicles
2. *Good hydrodynamic conditions* in connection with correct hypoid oil
3. *Enhanced efficiency* with small offsets, compared to spiral bevel gears
4. *Pinion diameter increase* provides lower root bending stress
5. *Increase in face-contact ratio* due to pinion spiral angle
6. *Dampening effect* due to high sliding velocity (noise reduction)

Hypoid gears have axial forces that can be calculated by applying a normal force vector at the position of the mean point at each member (see "*General Explanation of Theoretical Bevel Gear Analysis*"). The force vector normal to the transmitting flank is separated into its *X*, *Y* and *Z* components, from which the force components in those directions are calculated (Fig. 8).

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

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

$$F_y = -T \cdot (\sin\gamma \cdot \sin\beta \cdot \cos\alpha + \cos\gamma \cdot \sin\alpha) / (A_m \cdot \sin\gamma \cdot \cos\beta \cdot \cos\alpha)$$

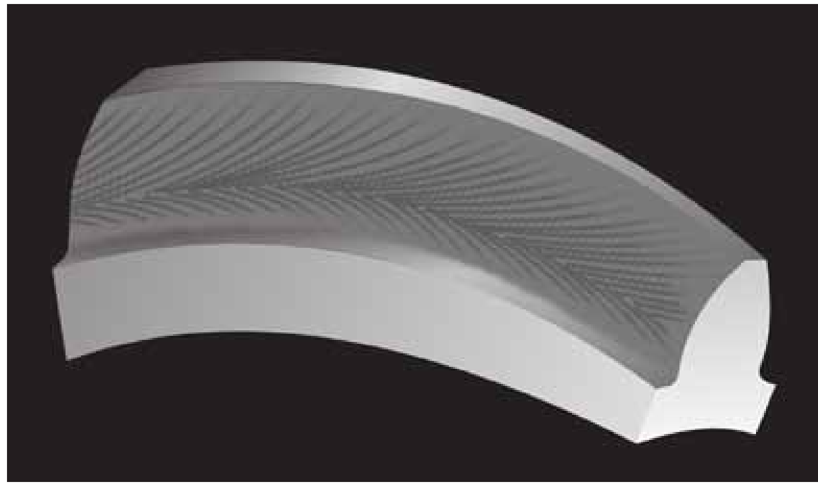

Figure 9—Appearance of a scored hypoid ring gear surface.

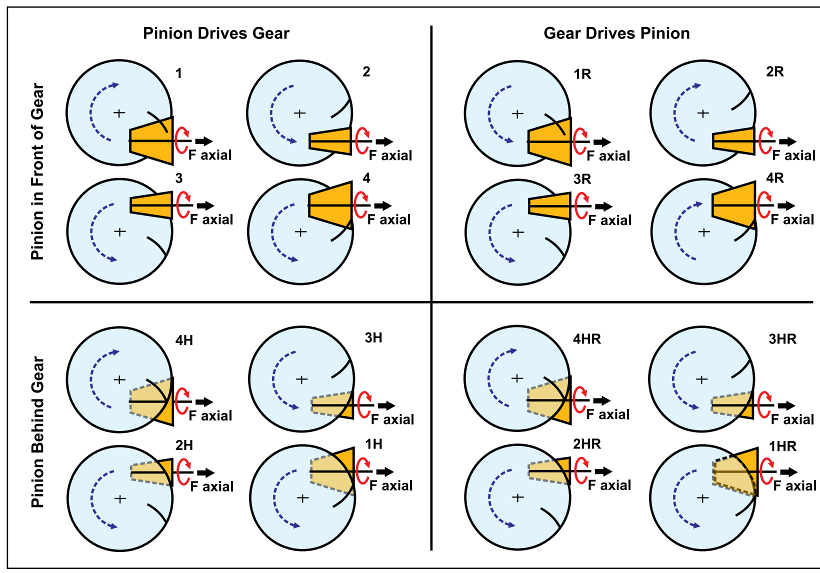

Figure 10—The 16 cases of hypoid offset.

$$F_z = -T \cdot (\cos\gamma \cdot \sin\beta \cdot \cos\alpha - \sin\gamma \cdot \sin\alpha) / (A_m \cdot \sin\gamma \cdot \cos\beta \cdot \cos\alpha)$$

where:

T	Torque of observed member
A_m	Mean cone distance
γ	Pitch angle
β	Spiral angle (in hypoids for pinion)
α	Pressure angle
F_x, F_y, F_z	Bearing load force components

To achieve correct results one must use the pinion spiral angle for the hypoid pinion and the gear spiral angle for the hypoid gear. Between pinion and gear spiral angle in hypoids is the following relationship:

$$\beta_{pinion} = \beta_{gear} + \arctan(a/A_m)$$

continued

where:

a = shaft offset

The offset a is positive for cases 1 and 4; negative for cases 2 and 3 (Fig. 10). The pinion spiral angle is positive in all left columns (Fig. 10) and negative in the right columns (gear spiral angle has the opposite sign) (Fig. 10). The bearing force calculation formulas are based on the assumption that one pair of teeth transmits the torque with one normal force vector in the mean point of the flank pair. The results are good approximations reflecting the real bearing loads for multiple tooth meshing within an acceptable tolerance. Precise calculation can be attained with Gleason bevel and hypoid gear software.

The introduction of spiral angles lead to a face contact ratio—which in turn reduces the tooth root thickness. The tooth thickness counts squared in a simplified root bending stress calculation using a deflection beam analogy; i.e.:

- The thickness reduces by: cos (spiral angle).
- The face contact ratio increases simplified by: tan (spiral angle).

Due to the offset in the tooth mesh position, the pinion spiral angle must be considered in the observations above. Note that the formulas applied to a numerical example will always show an advantage of the spiral angle in root bending strength. Note, too, that crowning of real hypoid gears will always cause one pair of teeth to transmit a disproportionately higher share of the load, while the one or two additionally involved tooth pairs will only share a small percentage of the load.

Hypoid pinions have an advantage in that if the offset is chosen it increases the pinion spiral angle. Together with the spiral angle, the pinion diameter increases. Figure 10 summarizes the 16 different hypoid cases. The graphic's left-side column is for a driving pinion, the right-side column for a driving gear. In the upper (#2—Pinion Drives Gear) section (Fig. 10) the pinion is in front of the ring gear; in the lower section (Fig. 10), the pinion is behind the ring gear. The torque transmission in all cases utilizes the drive-side. The vector F_{axial} points in the opposite direction in the case of coast-side torque transmission (which will expand the scheme in Figure 10 to 32 cases in total). Cases 1 and 4 (and sub-cases R, H and HR) in Figure 10 are the hypoid cases with a positive offset that increase the pinion diameter to:

$$d_{0\,hypoid} = d_{0\,spiral} \bullet (1/\cos\beta_{pinion} - 1/\cos\beta_{gear})$$

Finite element calculations can be particularly useful in connection with hypoid gears in finding the optimal spiral angle for maximal root strength.

Hypoid gears require—even with low RPMs—a high-pressure oil with additives or special synthetic hypoid oils. A sump lubrication is recommended. The oil level has to cover the face width of the teeth lowest in the sump. Excessive oil causes foaming, cavitations and unnecessary energy loss. The preferred operating direction of hypoid gears is the drive-side, where the convex gear flank and the concave pinion flank mesh. Note well that this is not only a recommendation—it is a binding rule. In the drive direction (Fig. 9) the forces between the two mating members bend the pinion sideways and axially away from the gear generating the most backlash. Coast-side operation reduces the backlash—in extreme cases—to zero, thus interrupting any lubricant flank separation and leading to immediate surface damage and often followed by tooth fracture. (*Next issue and final series installment: Super Reduction Hypoid Gears*.)

(*Note to readers: The release date for the German-language version of Dr. Stadtfeldt's book—*Gleason Kegelradtechnologie*—is September, 2011; published by Expert, Esslingen, Germany; Pages: 500; Price: Euro 51.40; ISBN: 978-3-8169-2983-3.The English-language version—*Gleason Bevel Gear Technology*—will be released approximately one year later, September, 2012.*)

References

1. Stadtfeld, H.J. "Handbook of Bevel and Hypoid Gears," Rochester Institute of Engineering, Rochester, New York, March 1993.

2. Wildhaber, E. and A.J. Steward. "Design, Production and Application of a Hypoid Rear-Axle Gear," *The Journal of the Society of Automotive Engineers*, June, 1926.

3. Hotchkiss, R.G. "The Theory of Modern Bevel Gear Manufacturing," *AGMA Gear Manufacturing Symposium*, Cincinnati, Ohio, April, 1989.

PM Design Winners

ANNOUNCED AT POWDERMET2011


Design innovation, superior engineering properties, high end-market visibility and sustainability distinguish the winners of the 2011 Design Excellence awards, the annual powder metallurgy (PM) design competition sponsored by the Metal Powder Industries Federation. The PM process demonstrates unique value propositions over competing forming processes such as investment casting, die casting, deep drawing, machining, stamping and fine blanking. PM's design performance is underscored in automotive engines, emergency breathing units, audio devices, security locks, endoscopic surgical devices, hand tools, recreational products, industrial refrigeration equipment and pumps, railway brake systems and rifle sights.

Presented during the 2011 International Conference on Powder Metallurgy and Particulate (PowderMet2011), GKN Sinter Metals, LLC, located in Auburn Hills, Michigan, earned the grand prize in the automotive transmission category for a carrier and one-way rocker clutch assembly made for Ford Motor Company. Used in the new Ford Super Duty TorqShift six-speed automatic transmission, the hybrid assembly contains five PM steel parts weighing a total of 17 pounds.

The sinter-brazed subassembly consists of four multilevel PM parts, of which three parts (cage, spider and carrier plate) are made to a density of 6.8 g/cm3. The rocker plate is sinter-hardened during the sinter-brazing phase and has a density of 7.0 g/cm3. The assembly also has a double-pressed and double-sintered cam plate made to 7.3 g/cm3 density with an ultimate tensile strength of 170,000 psi and a mean tempered hardness exceeding 40 HRC. To form the parts and maintain precision tolerances, innovative tooling was developed and used in conjunction with unconventional press motions. Ford subjected the assembly to stringent durability testing—ultimate torsional torque loading at a minimum of 8,000 foot-pounds and fatigue testing at a minimum of 299,000 cycles at 1,730 foot-pounds. The application provided an estimated 20 percent cost savings over competitive processes, and represents a new era in the scope and size of PM parts. "Designed for PM," the assembly uses fewer components compared to alternate designs and manufacturing methods, reducing the mass of the assembly, thus improving fuel efficiency without sacrificing function or performance.

GKN won another grand prize in the automotive chassis category for a differential bearing adjuster made for its customer American Axle and Manufacturing and used in the GMT 900 rear differential on GM Tahoe and Yukon models. The diffusion-alloyed PM steel part preloads the bearing and is locked in place through the side holes. Formed to a density of 6.8 g/cm3, it has a 155,000 psi transverse rupture strength, 79,000 psi tensile strength, yield strength of 63,000 psi, and 90 HRB apparent hardness. A special die and dual-upper-punch design form the cross-holes during compaction. Selecting PM saved the customer $320,000 annually by replacing a casting that required extensive machining.

A porous bronze filter made by Capstan California, located in Carson, for Chase Filters and Components received the grand prize in the aerospace/military category. Gravity sintered, the filter is used in an ignition-resistant, fault-tolerant oxygen cryopac filter for medical and emergency breathing systems.

Parmatech Corporation, located in Petaluma, California, earned the grand prize in the hand tools/recreation category for a large-game 420 stainless steel hunting arrowhead, the 300Xtreem broadhead, made for Optek Precision Tooling Ltd. Fabricated by the metal injection molding (MIM) process, the thin blade features a critical straightness necessary for accuracy and the external thread for attachment to the arrow shaft is molded in the design.

A PM copper–steel outer hub exit spindle used in electronic door locks won the grand prize in the hardware/appliances category. Made by ASCO Sintering Company, located in Commerce, California, for Ingersoll Rand Security Technologies (Schlage), the part connects a standard lock and an exit device. When activated through a code or electronic card reader, the spindle rotates to the standard lock assembly.

FloMet LLC, Deland, Fla., earned the grand prize in the medical/dental category for a housing cup and lid used in an audio device with magnetic shielding capabilities. This application is the first of its kind in the high-power audio device sector. The anti-magnetic MIM material with high nickel content provides electromagnetic interference, or EMI shielding, preventing interference from other electronic signal sources.

A PM diffusion-alloyed steel rotor made by Lovejoy Sintered Solutions, LLC, in Downers Grove, Illinois won the award of distinction in the industrial motors, controls and hydraulics category. The rotor operates in an industrial rotary gear pump for handling fluids like waste water or chemicals. Fabricated to a density of 6.95 g/cm3, the rotor has a tensile strength of 105,000 psi, yield strength of 55,000, and 89 HRB hardness.

Burgess-Norton Mfg. Co., Geneva, Illinois, earned the award of distinction in the hardware/appliances category for a PM steel crimp retainer operating in a valve assembly that regulates gas flow in a high-performance compressor for commercial refrigerators.

A 17-PH stainless steel distal channel retainer formed via the MIM process by Kinetics Climax Inc., located in Wilsonville, Oregon, received the award of distinction in the medical/dental market category. The complex, multi-level part is the main distal-side component of an articulation joint in an articulating mechanical stapler/cutter used in endoscopic surgery.

A bronze filter plate made by Capstan California, Carson, California., for Knorr-Bremse GmbH in Austria won an award of distinction in the off-highway category. Made via the gravity sintering process, the net-shape part is used in the braking system of European commuter trains. A highly innovative graphite mold design incorporates the 8.2 mm cross-hole, which eliminates a machining operation for drilling the hole.

Cloyes Gear and Products, Inc., located in Paris, Arkansas, won the award of distinction in the automotive engine market category for an intake sprocket gear and an exhaust gear used in a coupling assembly operating in 2.0 and 2.2 liter diesel engines made by General Motors Korea. The gears receive torque from the timing chain, which drives the intake camshaft and transmits torque to the exhaust camshaft.

A rear sight used on sporting and military rifles such as the AR-15, M-4, and M-16 models, received the award of distinction in the aerospace/military market category. Made by Megamet Solid Metals, Inc., located in Earth City, Missouri, for its customer, Yankee Hill Machine Co., Inc, the nickel steel MIM part features very close tolerances and a complex geometry requiring an elaborate tool design. The sight allows the shooter to target objects at ranges up to 200 yards by using the larger aperture, and to target objects at longer ranges by flipping the sight down and using the smaller aperture.

Indo-US MIM Tec (P) Ltd., Bangalore, India, won an award of distinction in the hand tools/recreation market category for a 17-4PH stainless steel hammer used in a Leatherman Tool Group multi-purpose military utility tool (MUT) designed for military and law enforcement personnel, and civilians. Made by the MIM process, the intricate complex part performs five of the MUT's 27 functions.

It is formed to a density of 7.5 g/cm3 and has an ultimate tensile strength of 175,000 psi, yield strength of 158,000 psi, heat-treated 35–40 HRC hardness, and a minimum six percent elongation. Secondary operations include threading two tapped holes, age hardening and glass-bead blasting, and an optional blackening treatment performed by Leatherman.

Webster-Hoff Corporation, located in Glendale Heights, Illinois, and customer ACCO Brand Inc. received the second award of distinction in the hand tools/recreation market category for a PM sinter-hardened steel cam and bushing used in a manual paper hole punching machine. The cam transfers power to the cutters and the bushing supports the shaft. Both parts are formed to a density of 6.7 g/cm3 and have an ultimate tensile strength of 120,000 psi and 27 HRC apparent hardness.

For more information on PowderMet2011, visit *www.mpif.org*.

July 22–24—2011 Solid State Power Supply Technical Seminar. Madison Heights, Michigan. This class is scheduled to examine Inductoheat's Statipower SP12 Power Supply. The principal objective of this seminar is to have Inductoheat certified technicians provide end-users with detailed presentations and hands-on experience for the operation, maintenance and troubleshooting of Inductoheat solid state power supplies. These seminars are held in the company's training center and will cover basics of induction, special maintenance training techniques, power supply orientation, load matching, troubleshooting, how to check for common failed components, safety procedures, process control and monitoring and quality assurance. Inductoheat will cover diagnostic techniques and component replacement procedures so your personnel can efficiently troubleshoot and perform preventive maintenance on specific induction equipment. Cost: $525 per attendee includes: Three days of classes, training materials, catered lunch and dinner Thursday evening. For more information, visit *www.inductoheat.com.*

July 18–20—Basic PM Short Course. Penn Stater Conference Center Hotel, State College, PA. This three-day course is designed for users of powder metal (PM) parts and those looking for an introduction. Topics include the history of PM, why it's a viable method of producing metal parts, designing PM, MPIF standards, special tests for powders, metal injection molding, secondary operations, how sintering develops the functional properties and more. Attendees receive more than $400 worth of publication material. The course is designed for engineers, tool designers, product designers, metallurgists, supervisors, purchasing agents, technicians, managers and quality control personnel. For more information, visit *www.mpif.org*.

September 12–15—The Eleventh International Conference on Shot Peening (ICSP-11). South Bend, Indiana. ICSP-11 is a triennial conference and exhibition of the International Scientific Committee for Shot Peening. The commercial benefits of applying mechanical surface treatments are increasingly recognized, particularly in the automotive and aerospace industries. ICSP-11 will be an important international meeting for discussing the science, technology and applications of mechanical surface treatments. It will offer a unique forum, enabling scientists and engineers to deepen and update their knowledge of all aspects of mechanical surface treatments. The conference will cover a range of surface treatment topics based on technological aspects, process procedures, changes in the surface state, process simulation, service properties and fields of application. For more information, visit *www.shotpeening.org/ICSP-11*.

September 15–17—AWEA Small and Community Wind Conference. Des Moines, Iowa. More than 100 exhibitors will have the latest wind technology to show consumers, renewable energy professionals and installers how to best capitalize on wind technology for homes, farms and ranches, businesses and rural electric cooperatives. Community wind development has proven itself to municipalities, schools, universities and other groups willing to band together to produce their own energy. Designed with direct input from AWEA members who are shaping these important wind markets, this conference offers two tracks focusing on all the facets of the small and community wind industries. Hear from wind experts, investors, and stakeholders from across the nation on how wind can create a cleaner energy future, strengthen regional and national economies and lead to a sound, profitable energy market. For more information, visit *www.awea.org*.

October 3–7—Basic Training for Gear Manufacturing. Richard J. Daley College, Chicago. This AGMA training course covers gearing and nomenclature, principles of inspection, gear manufacturing methods, hobbing and shaping. The course is intended for those with at least six months of experience in setup or machine operation. Classroom sessions are paired with hands-on experience setting up machines for high efficiency and inspecting gears. For more information, contact Jenny Blackford at *blackford@agma.org* or (703) 684-0211.

Congressman Roskam

VISITS OVERTON CHICAGO GEAR

Overton Chicago Gear (OCG) had the pleasure to greet their Illinois representative recently. Congressman Peter Roskam, 6th District of Illinois (serving Addison and Lombard, IL), visited Overton Chicago Gear on April 27, 2011. Roskam is a member of the Ways and Means Committee and is known as an advocate of ESOP companies. Congressman Roskam was interested to hear about the strength of the industrial manufacturing sector in his district. "It was gratifying to have Congressman Roskam come to see OCG on our silver anniversary as an ESOP company," said Overton Chicago Gear president/CEO Lou Ertel. OCG shared with him their continued success in the manufacturing of gears and gearboxes, even with the challenges of foreign competition and challenging economic conditions. All at OCG thoroughly enjoyed the time Roskam and his staff spent at Overton and they look forward to having him back in the future.

Timken

TO ACQUIRE PHILADELPHIA GEAR

The Timken Company, located in Canton, Ohio, has announced plans to purchase Philadelphia Gear Corp. for $200 million. Based in King of Prussia, Pennsylvania, Philadelphia Gear provides gear drives and components to the industrial and military sectors. Timken makes bearings, assemblies and alloy steels for auto producers and other manufacturers. The addition of Philadelphia Gear to Timken's Process Industries segment significantly expands the range of industrial services capabilities for both companies to offer their customers. The acquisition advances Timken's strategy to offer comprehensive services and solutions to end-users that enhance the performance and productivity of their mission-critical mechanical applications.

"Philadelphia Gear is an excellent fit with Timken," said Chris Coughlin, president of Timken's Process Industries segment. "Like Timken, it is a leader in industrial services providing highly engineered solutions and replacement components for mechanical power transmission equipment. It will strengthen our presence precisely in the areas we've targeted, with excellent prospects for profitable growth via extension to our customer base around the world."

Timken plans to combine Philadelphia Gear, which employs approximately 220, with its Industrial Services business to leverage the best capabilities of each organization across the global markets. Coughlin noted that Timken plans to expand the business with the existing management team. "The synergies in this combination are all about growth," he said. Timken expects Philadelphia Gear to be accretive to earnings in its first full year and to generate income exceeding the cost of capital by 2014. The company plans to complete the acquisition through its Timken Gears and Services Inc. subsidiary by the third quarter of 2011, pending certain government and regulatory approvals.

Star SU

ADDS ENGINEERING AND SALES STAFF

Grady Knight has joined Star SU as a project engineer

continued

for the machine tool division. He will work closely on large projects with vice president of sales and engineering Mark Ritchie and will be based in the Hoffman Estates, Illinois office. Knight earned both of his degrees, Bachelor of Science in Manufacturing Engineering Technology and Masters of Science in Industrial Management, from Northern Illinois University in DeKalb, Illinois. He holds a graduate certificate in strategic management as well as a graduate teaching certificate. He was awarded the Elgin Community College Trustee Academic Scholarship, the Grainger Tools for Tomorrow Scholarship and the 2011 Outstanding Graduate Student in Engineering Technology at NIU.

Grady Knight

Jeffory Fadler has joined the sales staff of Star SU as a regional sales manager for cutting tool sales in western Michigan. He will take over the territory formerly covered by Craig Weirich, who retired in June. Fadler earned his Bachelor of Science degree in Mathematics at Michigan State University. He has extensive cutting tool sales and management experience serving the automotive market.

Jeff Fadler

Ryan Moore returns to Star SU as a regional sales manager for cutting tool sales in eastern Michigan and Ontario. Moore's wide-ranging sales experience includes serving accounts in the automotive, machine and power tool, mining, construction and aerospace industries. He will take over the cutting tool sales territory formerly covered by John Simpson, who plans to concentrate his efforts on machine tool sales in Michigan and Ontario. Moore earned his Bachelor of Science degree in Industrial Technology at the Purdue University School of Industrial Technology in Indiana.

Ryan Moore

Forest City Gear

APPOINTS SALES REPS

Forest City Gear recently announced the appointment of new sales representatives and additional territories for several existing reps, effective immediately. President Wendy Young, in making this announcement, remarked, "Our growth in the last several years has been substantial and we determined it was necessary to expand our field sales force. After considerable effort and an exhaustive search, we are very pleased with the new team." Young also noted these additions were particularly challenging, as the highly customized nature of Forest City Gear's business base requires the sales force to have a keen knowledge of gear manufacturing, application engineering assistance and customer service through an often lengthy buy cycle. The following new sales representatives are now handling Forest City Gear: Joseph Kemple for Ohio, Kentucky and Indiana, south of Route 30; Stephen Peterson for Iowa, Wisconsin, Minnesota; Curt Nicholson for Utah, Arizona, New Mexico, Colorado and southern California; Dennis Young for northern Illinois. The following existing reps have been reassigned to cover these states: Jim Hagen for Washington, Oregon and Idaho; Stu Glasby for Pennsylvania, upper New York and southern New Jersey; Bob Rundle for northern New Jersey, lower New York, Rhode Island, Connecticut and Massachusetts.


Joseph Kemple


Stephen Peterson


Curt Nicholson


Dennis Young

NEWS

Solar Atmospheres

APPOINTS ENGINEER AND MARKETING MANAGER


Trevor Jones

Trevor Jones has been promoted to principal engineer at Solar Atmospheres, Inc. Jones started with Solar as a part-time employee in 1998 and continued working as a summer intern over the next six years during college. He received a dual Associate's Degree in Mechanical and Electrical Engineering from Penn State University. Jones began full time employment at Solar upon graduation in 2004. He is currently on the Executive Committee for the ASM Philadelphia Liberty Bell Chapter as Secretary and will become vice-chairperson for the 2011–2012 chapter year. Jones' responsibilities include involvement in all R&D projects from inception to production. He has two R&D technicians that report to him. Don Jordan, vice president of R&D and Corporate Metallurgist states: "Trevor is a true authority on vacuum heat treatment and vacuum furnace technology, understanding the scientific principles of the field and applying them to practical ends both mechanically and electrically. Trevor is admired by all who work with him for his professionalism and genuine gentleman-like character. He is an eager learner and I am confident to say that one would think that Trevor also has an associate's degree in heat treat metallurgy."


Laura Edwards

Laura Edwards, recently joined Solar Atmospheres as marketing manager. Her focus is on providing strategic directions for all of Solar's marketing activities, such as corporate branding and communications; advertising; website development; tradeshows; public relations and social media. Edwards has more than 20 years of experience in business-to-business marketing. In addition to freelance consulting, she has worked with a variety of marketing agencies and business. "Laura brings a wealth of knowledge in the marketing field to this position and has a great amount of experience. We are excited to see what Laura will do to help Solar Atmospheres," states Roger Jones, corporate president.

Edwards serves as vice president of publicity for the Women's Referral Network of Montgomery County and co-teaches a leadership program in conjunction with Pearl S. Buck International. She is a double graduate of Penn State University with a B.S. in Marketing and a Masters in Business Administration.

MSI

NAMES NEW VICE PRESIDENT FOR ILLINOIS MAZAK BUSINESS UNIT

Machinery Systems Inc. (MSI) recently announced the appointment of Eric Hilliard to the position of vice president of the company's Illinois Mazak business unit. In this position, Hilliard will oversee the region's Mazak sales, application engineering, customer service and manufacturing software. Having joined MSI in 1996, Hilliard has worked as a sales engineer in three geographic territories. Prior to that, he held multiple other positions in the industry, including working as an electrical and mechanical technician. "Over the past fifteen years, Eric has demonstrated a tremendous commitment to our mission of serving American manufacturers in the Midwest," said Ron Mager, president and CEO of MSI. "As the economy continues to rebound, we are seeing more and more manufacturers looking to maximize their future potential by investing in and integrating new Mazak technology. Eric will play a vital role in helping these customers optimize their operations."

Eric Hilliard

continued

Rexnord Corporation

FILES FOR IPO

Rexnord Corporation, formerly known as Rexnord Holdings, Inc. ("Rexnord"), the ultimate parent of RBS Global, Inc. and Rexnord LLC, recently announced it has filed a registration statement on Form S-1 with the U.S. Securities and Exchange Commission in connection with the proposed initial public offering of its common stock. The offering of common stock will be made only by means of a prospectus. When available, a copy of the preliminary prospectus relating to this offering may be obtained from: Rexnord Corporation, 4701 West Greenfield Avenue, Milwaukee, Wisconsin, 53214 or from such underwriters as Rexnord will engage in the future. A registration statement relating to these securities has been filed with the Securities and Exchange Commission but has not yet become effective. These securities may not be sold, nor may offers to buy be accepted, prior to the time the registration statement becomes effective.

Manufacturing Technology Consumption

CLIMBS IN 2011

April U.S. manufacturing technology consumption totaled $396.92 million according to the Association for Manufacturing Technology (AMT) and the American Machine Tool Distributors' Association (AMTDA). This total, as reported by companies participating in the USMTC program, was down 21.0 percent from March but up 74.9 percent when compared with the total of $226.99 million reported for April 2010. With a year-to-date total of $1,595.98 million, 2011 is up 105.3 percent compared with 2010.

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

"It is very encouraging to see year-to-date orders more than double last year's pace particularly with the price of oil, unrest in the Middle East, and the disasters in Japan," said Douglas K. Woods, President of AMT. "Despite April's numbers being slightly lower than March, recent levels of outstanding order activity are now approaching pre-recession levels which is a positive long-term indicator for our industry." For more information, visit *www.amtonline.org*.

Gleason

ANNOUNCES CHANGES TO EUROPEAN SALES TEAM

Gleason Corporation recently announced changes to their European Sales organization. Ralf Nierlein has been appointed vice president, sales - Europe, with responsibility for sales activities for all Gleason products in Germany, Austria and the Commonwealth of Independent States (CIS). Nierlein has been with the company for nearly 20 years in a variety of leadership positions in the European market.

Ralf Nierlein

Michael Vranic has been appointed vice president, sales – Europe, with responsibility for sales activities for all Gleason products in the remaining European markets, including France, Italy, Spain, Sweden, the United Kingdom and other countries. Vranic has been with the company for 15 years in various positions at Gleason-Pfauter in Ludwigsburg, Germany, most recently as Sales Director.

Michael Vranic

Udo Stolz, Gleason vice president worldwide sales and marketing, said "Mr. Nierlein and Mr. Vranic are seasoned professionals who will be significant contributors to achieving our mission as The Total Gear Solutions Provider. Their number one priority will be ever-higher customer satisfaction, which will drive growth for our company in this important region."

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☐ No thank you.

2) I want to receive the ☐ PRINT version ☐ *DIGITAL version ☐ *BOTH print and digital

(*Please include your e-mail address for DIGITAL or BOTH versions.) ______________________
(Please print e-mail address)

3) Name:______________________ (print) Job Title/Position: ______________________

4) Signature ______________________ Date: ______________________

5) Please tell us about your company:

Company Name:______________________
Company Address:______________________
____
City:______________________
State/Province:______________________
ZIP/Postal Code:______________________
Telephone:______________Fax:______________
E-mail:______________________
Co. Internet Address:______________________

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

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

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

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

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

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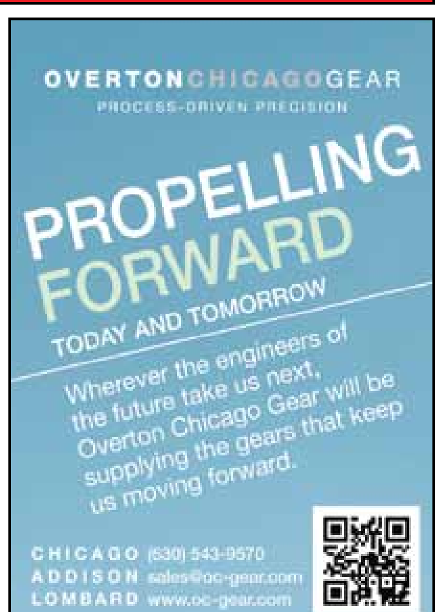

SERVICE

THE CHEVY CORVAIR:

RELIC OF AN ERA WHEN QUALITY WAS AN AFTERTHOUGHT

The 1960 Chevrolet Corvair four-door sedan (courtesy Stephen Foskett (Wikipedia user: sfoskett) and the Bay State Antique Automobile Club show).

Photo of Chevrolet Corvair 164 Turbo rear-mounted engine—and spare tire compartment!—taken at Bay State Antique Automobile Club's July 10, 2005 show at the Endicott Estate in Dedham, MA (courtesy of User: Sfoskett/ WikimediaCommons).

The recent and continuing comeback of the U.S. auto industry has been something to behold. Sales are booming—relative to the sagging economy—quality is king and dealerships are reopening. Couple that with Toyota's recall nightmares and the future looks pretty rosy.

All of which—for no reason at all, really—might remind some people of a car that shall forever live in infamy and ignominy—the Chevrolet Corvair. The car debuted to great acclaim in 1960 and was unceremoniously dropped from the GM line after the 1969 model. Yes, 1960 to 1969—a time of auto industry-conceived "planned obsolescence," prematurely rusting Bodies by Fisher and on and on.

But all of that was just warming up in the bullpen compared to 1965, when Ralph Nader's groundbreaking consumer advocate tome *Unsafe at Any Speed* was published. The book had a profound effect on the U.S. auto industry and consumer education—seat belts were just one improvement. And guess which car Nader chose as a prime example of poor quality and unsafe engineering—yep—the Corvair.

Nader's book documented problems associated with the Corvair's steering, tire pressures and stability on the road. This resulted in many accidents that, Nader said, could have been avoided with a better-designed car. Former—and legendary—industry executives John DeLorean and Lee Iacocca agreed. (Prior to the car's launch in 1960, two Corvairs were tested at the Riverside International Raceway in Riverside, California for 24 hours. One car rolled over. But hey—the other completed the drive consuming only one quart of oil.)

So it was no surprise that the Corvair is introduced in *Unsafe at Any Speed's* first chapter: "The Sporty Corvair—The One-Car Accident." That just might be because the 1960–1963 models had a faulty swing-axle suspension design that was prone to buckle under certain conditions—like stopping, for example.

But enough aspersion casting. Fact is, the Corvair was a very popular ride in its day, soon morphing into models like the 900 Series Monza—"the poor man's Porsche." Did you know that the Corvair was the only American-made, mass-produced passenger car to feature a rear-mounted air-cooled engine? That it was *Motor Trend* magazine's Car of the Year for 1960?

But as with many cars—in those days at least—the engine was the attention-grabber.

The Corvair engine was an aluminum, air-cooled 140 cubic inch (2.3 L) "flat-six" that eventually evolved to 145 and then to 164 cubic inches. The engine produced 80 hp (60 kW) and, for extra oomph, offered in 1965 a turbocharged 180 hp (134 kW) Corsa engine option.

Depending on the model, the engine sported dual or quad carburetors—unusual for the time. Or as a Corvair fan site puts it: "The carbs sit right on the engine. The carbs make the engine appear complicated, one on each side. Mechanical linkages connect to the other carb on the other side of the engine. Testing showed that using either a two- or four-barrel carb (one carb with two or four barrels) perched in the center of the engine was problematic in maintenance and during cold and sub-freezing temps. Before changes were made, the carbs tended to take much longer to heat up in cold weather, or freeze to a light film that caused fuel problems."

A change in the 1966 model was a more robust four-speed synchromesh transmission using the standard Saginaw gear set with 3.11:1 first gear ratio used by other GM six-cylinder vehicles. But sales began to decline as a result of Nader's book, although that may also be attributed to the debut of the new Ford Mustang and "inside-auto industry" rumors of the upcoming Panther—apparently the code name for the imminent arrival of the Camaro.

Accounts at the time regarding the Corvair's demise ranged from "sadness and regret" that such a "fine car" could not survive to anger at Chevrolet's decision to continue making the car for as long as they did.

One final note—an ironic epitaph of sorts: Today, Corvair engines are used in many small aircraft because of their durability.