May 2011

GEAR TECHNOLOGY

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The Journal of Gear Manufacturing

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GEAR TECHNOLOGY

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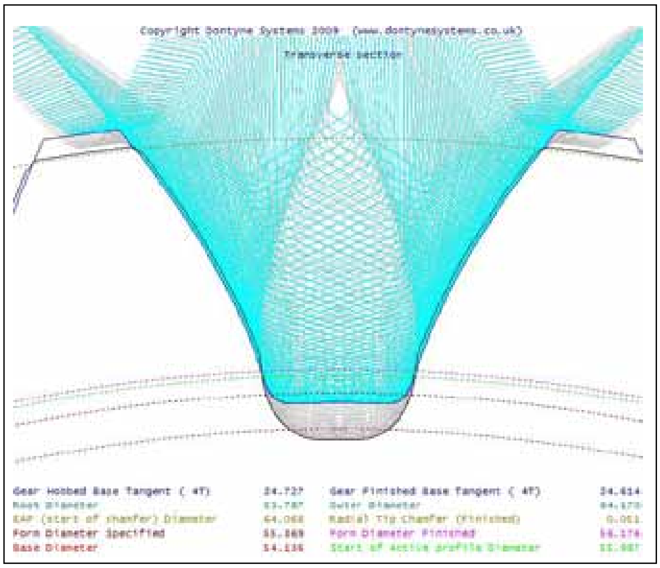

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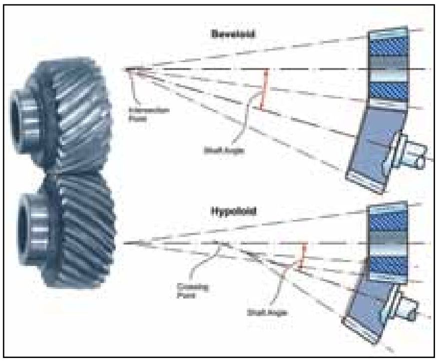

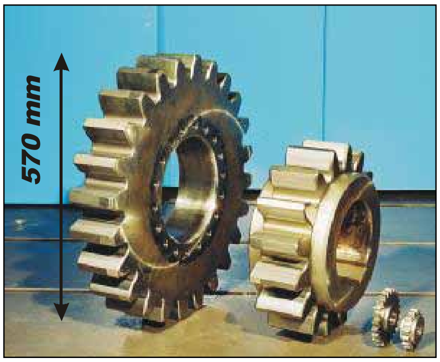

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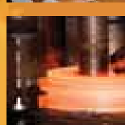

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BPA WORLDWIDE

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With their new ZE 1200, AGW is now able to grind gears ***in-house*** at their 34,000 sq. ft. facility near Atlanta, GA.

"We chose the NILES ZE 1200 not only because of the ease of use of the machine controller, but also because the ZE offers the ability to grind pinions and gears in the size range that was the most advantages for us. The ZE also allows us to inspect, kiss grind, and reclaim the proper gear profile and geometry on gears reused during rebuild, to provide a higher quality repair. This is another step to help us continue to provide cost effective, high-quality gears and gearbox repairs to our customers."

—Atlanta Gear Works Inc.

AGW

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Greenhouse of the Gear Industry

According to the calendar, it's been spring for more than a month now, but here in Chicago, it's been hard to tell. Sure, we've had a few warm, sunny days, but with snowfalls well into the middle of April, it seemed as though winter would never relax its grip. Now, it seems we've finally turned the corner. We're still waiting for the sun to come out, but at least the snow has turned to rain.

With the weather so uncertain, it's hard to get your garden started. First, you have to wait for the danger of frost to pass. Then you either have to brave the deluge or find a day when it's nice enough to plant. If you're not careful, you could miss the growing season altogether. That's why many gardeners like to get a jump-start on spring by planting seeds indoors before the weather breaks. Some even use a greenhouse, because new plants need the right conditions, the right nurturing and the right timing.

Young people are like plants in that way. They too, need nurturing—and we can't always wait for the timing to be right in order to plant the seeds.

In March, Forest City Gear received a visit from a young person interested in manufacturing. In this case, the visitor was 13-year-old Alexi Cluff, accompanied by her grandfather Brian Cluff, vice president of Star-SU. They visited Forest City Gear to learn about career options in manufacturing.

Alexi Cluff is one of 48 students enrolled in a program sponsored by a grant from the Motorola Foundation and conducted at the College of Engineering and Engineering Technology at Northern Illinois University. The program is a workshop in partnership with NIU-Enhanced Engineering Pathways, the Society of Women Engineers and the Girl Scouts of Northern Illinois. Girls in the program are mentored by women engineering professors and women engineers from a variety of industries, and they often take tours of manufacturing facilities to absorb "real world" experiences.

Forest City Gear owners Fred and Wendy Young went out of their way to give Alexi as much exposure as possible, including a tour of all departments of the company. They arranged a roundtable discussion with women from these departments, including application engineering, human resources, gear grinding, gear deburring, gear hobbing, order processing, estimating, expediting, procurement, materials inspection, quality validation, machine setup and company management.

Although Alexi Cluff obviously has an interest in engineering and manufacturing, I believe there has always existed a certain percentage of young people that are naturally curious about the way things work, and we're not doing enough to engage those young people and make them aware of the rewarding possibilities of working in a manufacturing environment.

If we can make available to these young people more opportunities to see what goes on in a manufacturing environment, we can give them an outlet for their natural curiosity that isn't being stimulated by their parents, teachers and counselors.

Those of us who work in manufacturing can probably agree that our country already has enough lawyers, bankers and financial wizards. In the past, those were viewed as prestigious and desirable career paths. Although they may seem less desirable following the most recent financial crisis, we're not doing enough to convince people that careers in manufacturing and engineering can be rewarding, both personally and financially.

So what are you doing to encourage young people in manufacturing careers? What student groups have you invited to tour your facility? Have you approached the local Boy Scout or Girl Scout troop? What about the local schools and youth groups?

Not only do you have the opportunity to encourage young people to consider a career in manufacturing, but also you can expose them and their adult leaders to the many possible jobs where they could satisfy their natural curiosity about how things work, as well as earn a good living. Just as importantly, our country needs to maintain these important skills.

With the manufacturing recovery underway, you're all probably very busy, and you might not think you have time for taking on extra projects. But if not now, then when? Nobody wants to take on extra projects when business is bad. We're all too busy trying to improve our business. But we can't afford to wait until the weather breaks. Each of your facilities is like a greenhouse for the gear industry—a place where you can plant the seeds of a manufacturing future, a place where you can nurture those seeds and help them grow. There are a lot of kids out there like Alexi Cluff who could benefit from such a place. What can you do to provide it?

Michael Goldstein,
Publisher & Editor-in-Chief

Developing Flexible Couplings Standards

Glenn C. Pokrandt
Principal Engineer (Retired), The Falk Corporation
Chairman, AGMA Flexible Coupling Committee

Flexible couplings? When or how did they become a type of gear drive or a component thereof and result in a committee within AGMA? Why would they even be a part of AGMA?

Well I don't know the exact answer, but many gear drives would not be turning without one of these flexible couplings on the input shaft, nor would the power be transmitted to the end application without one. So it makes some sense to be a part of AGMA. Many gear drive manufacturers also manufactured flexible couplings or used these products manufactured by others. There seemed to be a need for standardization of basic items such as bore-to-shaft fits. Creating such standards allowed for coupling manufacturers and power transmission distributors to stock many different couplings in the "rough stock bore" condition (reducing their inventory) and yet be able to provide consistent and interchangeable components.

My personal involvement with the committee goes back to 1984 when I attended my first AGMA Flexible Coupling Committee Meeting. I was readily accepted as a member and developed many new friends from our competitors. This was a new experience for me. This group did not act like competitors (although we are always aware of our company's competitiveness in the marketplace). Our goal was to try to understand our customers' technical and commercial needs and then standardize our products, where possible, to better fit those needs. Among our companies, there was a vast amount of technical expertise in many areas, bridging from general-purpose, off-the-shelf couplings stocked and sold by power transmission distributors, to the high performance type couplings used with gas turbines in the power generation or petrochemical industries.

I became chairman of the committee in the mid 1990s. My goal was to develop two new standards. One was a pure metric bore and keyway standard (similar to one already existing for the inch system) with the hopes that it would become an ISO standard, and a "basis for rating" standard. The metric bore and keyway standard was needed because the metric world had many standards (each company using its own combinations), but no standardization. The U.S. industry also had the problem of converting the metric bores and shaft sizes into the inch system and then trying to fit them into the inch standard. The "basis for rating" standard was needed if we were to truly have a service factor standard (which we had, but ultimately it was demoted to an information sheet as we had no standardized method for rating of flexible couplings).

The flexible coupling committee is currently composed of the following members and their associated companies: chairman Glenn Pokrandt (myself), Rexnord Industries, LLC; vice chairman Todd Schatzka, Rexnord Technical Services Corporation; Amy Lane, AGMA; Brian Greenlees, A-C Equipment Services Corporation; Doug Lyle, Ameridrives Couplings—Altra Industrial Motion Group; Tom Glasener, Emerson Industrial Automation; Chris Hatseras, KTR Corporation; Elliot Wilson, Lovejoy, Inc.; and Don Hindman, Rexnord Industries, LLC.

The committee is responsible for 10 current standards, two standards pending release, one information sheet and one new standard under development. Current standards and information sheets must be reviewed and re-affirmed every five years to confirm their applicability and that there are no updates or revisions required.

AGMA 9000-C90, "Flexible Couplings—Potential Unbalance Classification," is probably the most referenced flexible coupling standard specified by end users and system suppliers of both general purpose and high performance flexible couplings.

AGMA 9000-Dxx, "Flexible Couplings—Potential Unbalance Classification (Inch Edition)," is an update of AGMA 9000-C90. It is planned for release in 2011. This standard relates to flexible couplings' potential unbalance (and not the residual unbalance of rigid rotors). Flexible couplings have both runout and clearances between mating parts. They are normally shipped as a group of unassembled components, field assembled and sometimes disassembled and re-assembled. As you can see, there is a great difference when compared to the balancing of a high speed shaft with integral pinion or an intermediate shaft with the gear and pinion already mounted when balanced. These parts are not expected to be disassembled prior to being put in the gearbox and subsequent operation.

AGMA 9110-Axx, "Flexible Couplings—Potential Unbalance Classification (Metric Edition)," is a pure metric version of AGMA 9000-Dxx and will be able to be used by those preferring the metric system. It is also planned for release in 2011. The AGMA balance class required or calculated will be the same from either standard.

This standard recognizes the coupling weight and the speed at which it will be operating. The standard then suggests a "balance class" and a corresponding maximum amount of potential displacement of the principal inertia axis of the coupling relative to the axis of rotation (unbalance) permitted based on the system sensitivity. Ultimately, the purchaser of the coupling is responsible for the balance class specified and may override the standards recommendation based on his operating experience. The standard also provides a "tutorial" on how the coupling manufacturer will determine the potential displacement of the principal inertia axis of the coupling relative to the axis of rotation and if and how the coupling should be balanced to achieve the end results required. The annexes provide example calculations for both general purpose and high performance type requirements.

AGMA 9002-B04, "Bores and Keyways for Flexible Couplings (Inch Series)" (and its predecessors) combined several individual standards into one and has been used by coupling manufacturers for many years. It standardizes the recommended bore tolerances and bore to shaft fits based on the standard shaft sizes and tolerances used by both motor and enclosed drive gear manufacturers. The bore to shaft fit choices are clearance or interference (0.0005 inches interference per inch of shaft diameter). It also contains the recommended key/hub keyway dimensions and tolerances based on shaft diameter. The annexes contain information on the inspection of both straight and tapered bores and their keyways.

AGMA 9112-A04, "Bores and Keyways for Flexible Couplings (Metric Series)" is a pure metric version similar to AGMA 9002-B04. It does not convert the inch series to metric, but rather uses the metric standards employed by the metric world for their system of "standardized" shafting and tolerances (ISO R775:1969 which is no longer active, but was the basis for their shaft data being used today). The metric system is based on the "basic hole" system and results in a different system than those familiar with the domestic system (basic shaft system). Shaft diameters will be larger than indicated (for shaft diameters over 30 mm which use the k6 or m6 shaft tolerance). The k6 and m6 tolerances are plus to a greater plus tolerance over nominal. The recommended bore will typically uses an H7 tolerance and will be nominal size with a plus tolerance. The resulting bore to shaft fit will be a transitional fit (clearance to interference depending on tolerance combinations). This standard also contains tables of bore tolerances which will result in bore to shaft fits similar to those found in AGMA 9002-B04 (clearance and interference) which the domestic community is familiar with.

AGMA 9008-B00, "Flexible Couplings—Gear Type, Flange Dimensions, Inch Series," recognizes and specifies the standard flange diameters and mounting hole dimensions for both shrouded and exposed bolt gear couplings (nominal sizes 1½ thru 7). This standard was the result of the coupling industry meeting the domestic steel industry needs for the standardization of these dimensions for interchangeability/purchasing and stocking requirements that industry demanded. There is no metric equivalent as the domestic gear coupling (which historically finds worldwide usage) is only available as an inch product.

AGMA 9003-B08, "Flexible Couplings—Keyless Fits" and AGMA 9103-B08, "Flexible Couplings—Keyless Fits (Metric Edition)." The intent of these standards is to offer rotating equipment designers and users a standard for design practice and dimensions regarding keyless fits for flexible couplings. In general, the information in these standards is a consolidation of the most common practices and standards currently in use in the industry.

AGMA 9004-B09, "Flexible Couplings—Mass Elastic Properties and Other Characteristics" and AGMA 9104-A06, "Flexible Couplings—Mass Elastic Properties and Other Characteristics (Metric Edition)." These standards were developed through intensive study of existing practices, standards, text books and literature. The intent of these standards is to offer to rotating equipment designers, builders and users, a standard for design practice and methods of calculation of certain physical and mass elastic properties of flexible couplings.

AGMA 9009-D02, "Flexible Couplings—Nomenclature for Flexible Couplings" was developed to reduce the language barriers that arise between designers, manufacturers, and users when attempting to designate or describe various types of flexible couplings and their elements.

AGMA 9001-B97, "Flexible Couplings—Lubrication" covers the lubrication of gear couplings, chain couplings and metallic grid couplings and generally applies to other types of lubricated couplings.

Coupling lubrication requirements are unique. High speed couplings act as centrifuges and will displace the heavier components of the lubricant (typically not the oil component of a grease) radially outward into the working zone of the coupling. Conversely, low speed couplings do not disperse or resupply the lubricant into the working zone.

Information Sheet AGMA 922-A96, "Load Classification and Service Factors for Flexible Couplings" remains in waiting for the completion of the new standard "Basis for Ratings." When the "basis for rating" standard is completed, the committee will review and modify the data in this information sheet and release it as a standard for use by all coupling manufacturers that follow the rating requirements of AGMA 9006-Axx.

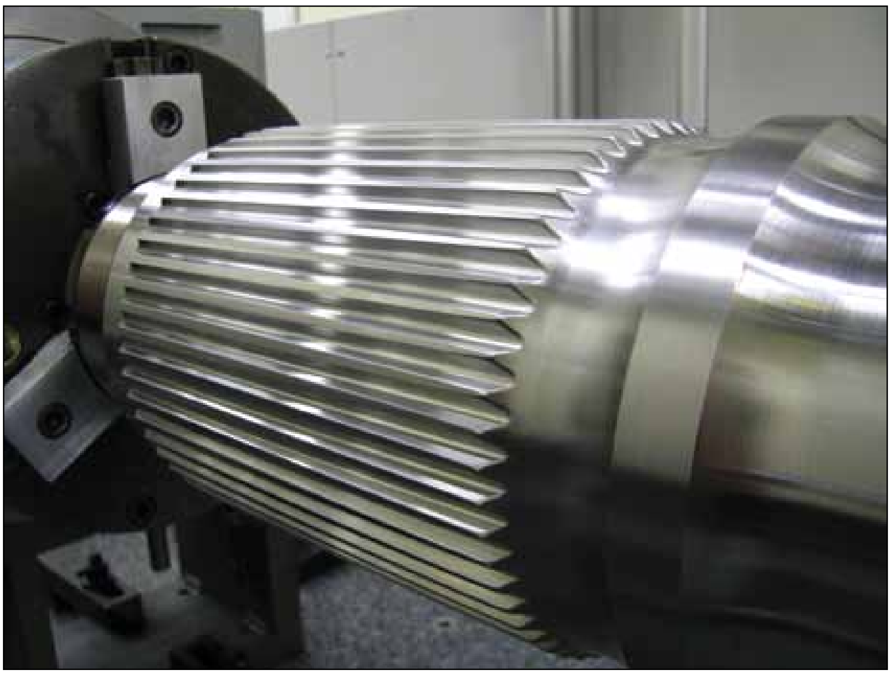
The new standard being worked on is **AGMA 9006-Axx, "Flexible Couplings—Basis for Ratings."** The committee started working on this standard in 1998. Progress has been impeded by the need for existing standard reaffirmation, the need to prepare metric equivalent standards for those written in the inch system (an AGMA requirement) and the need to expand AGMA 9000-C90 into AGMA 9000-Dxx and its metric equivalent, AGMA 9110-Axx. Now that these requirements have been completed, the committee will spend the next several meetings devoted full time to this new standard.

If after reading this article you feel you would like to contribute to our efforts, we welcome your input and assistance. We can learn from each other. We would especially like to involve additional elastomer type coupling manufacturers to participate. We generally meet twice a year (in the spring and fall) at cities to be determined at each meeting. Contact AGMA headquarters at (703)-684-0211 for the agenda and next meeting location.

Purchasers of couplings conforming to AGMA standards are assured of quality products made to uniformly accepted quality standards and design criteria.

Dearborn Precision

PUTS DUAL PURPOSE ZEISS CMM TO THE TASK



Gear measurements verified using *Gear Pro* software (courtesy of Carl Zeiss).

A 4.2 meter Accura with Vast XT gold sensor measures a turbine shaft with gears (courtesy of Carl Zeiss).

When the parts you manufacture pass through numerous processes such as deep hole drilling, machining, hobbing and grinding, a CMM (coordinate measuring machine) is essential when your customers require 100 percent in-process and final inspection.

Dearborn Precision Tubular Products, Inc., located in Fryeburg, Maine, has been involved in deep hole drilling, machining tubular components and precision tubing since 1947. Their work experience has led to the development of machinery and processes that are now applied to commercial nuclear, aircraft, land-based turbine parts, and tubular parts in the oil and gas exploration and nuclear industries. With a production facility featuring more than 100,000 square feet of climate controlled manufacturing space, Dearborn can easily produce either single prototypes or large production orders with a variety of parts.

In 2003, Dearborn purchased their first CMM, a Zeiss Spectrum, because of new turbine engine shaft work that required tighter tolerances. Today, this machine is handling the inspection of a large family of standard production parts. Prior to the Spectrum, they were using more traditional instruments such as hard and composite gages, and hand tools.

In 2007, Dearborn was approached to do a large aviation project involving the measurement of multiple parts including long turbine shafts with lengths of up to four meters and multiple gears on these shafts. The goal was to efficiently measure them in one run. This required a CMM large enough to measure the shaft lengths and one that would also allow them to easily change and use multiple sensors to measure the different part characteristics.

After comparing different systems, Dearborn decided to purchase a 4.2 meter Accura as the measurement solution for these long turbine shafts

continued

configured with the Vast XT gold sensor for its active control of long probes needed to measure deep into the long shafts. The Vast XT can be used with styli up to 800 millimeters long and up to 600 grams in weight. For software, the easy choice was Gear Pro, including involute, bevel, and worm, which allows them to do a full analytical inspection of the gears. Gear Pro will calculate the tolerances to multiple standards such as ISO 1328, AGMA 2000-A88, and DIN 2962. "This CMM is serving a dual purpose for us," says Andrew March, quality control engineer at Dearborn. "It can be used for traditional production run inspections and it also allows us to offer a gear checking service to our aviation customers. Most gear checkers are million dollar machines and they can't handle the length of these gear shafts." Dearborn can now measure about 10 of these long shaft parts in a day instead of what typically took a week without the Accura. They are verifying runouts of splines and gears that cannot be done with a gear checker. Typically, a gear checker is limited to about 1.5 meters of measuring height. With the Accura, they can measure gear shafts horizontally with 4.2 meters of measuring length, so runouts with data that distance apart can be measured in one run. This Accura is designed to handle long, horizontally measured gears versus the heavier 3,000-pound gears that are typically measured on three-axis rotary tables.

Dearborn specializes in deep hole drilling, so a typical part would start with drilling long parts followed by cylindrical grinding to hold the tight tolerances of + one ten-thousandth of an inch. There could be precision milling for added features, and gearing and splining would follow grinding. Roundness and runout tolerances are

Standard tubular products inspected on the Spectrum and Accura (courtesy of Carl Zeiss).

in the ten-thousandths of an inch. All parts go through in-process and final inspection before being shipped to the customer.

The Zeiss CMMs have helped Dearborn add customers, especially those requiring specific certifications. Their ISO 9001:2008 and AS9100B certifications depend on the process and measurement documentation that these measurement systems provide. It shows customers that Dearborn has control of their production processes.

Calypso has definitely helped Dearborn with their measurement requirements. They use the latest version, *Calypso 5.0*, especially for features like the enhanced graphical reports and have an SMA (the annual software maintenance agreement) to stay current with any new features and developments. "I've always been impressed with the improvements of *Calypso*," March says. "It allows us to provide more for our customers each year."

Overall, application support and scanning technology have been the most valuable aspects of owning a Carl Zeiss system for Dearborn. They are able to get answers to their questions quickly when working with Carl Zeiss applications engineers. And the scanning technology has allowed them to take a closer look at form measurement, which has made the CMM even more valuable. Without it they wouldn't be able to do analytical inspection of gears. The Vast XT provides them with sound raw data as well as proven algorithms for filtering data.

The two most beneficial results Dearborn has realized since purchasing the CMMs are reduced inspection time and reduced downtime for production machines. Inspection times have been reduced from four hours to half an hour per part with the Accura, while also eliminating operator influence. There is also significantly less downtime for production machines that are waiting for parts in inspection.

Looking into the future, Dearborn desires to further increase its business in the aerospace industry. Currently, they are at a point of expansion, adding square footage and employees. "Our customers demand 100 percent part inspection," March says. "Without these CMMs, we couldn't ship quality parts and be as successful as we are today."

For more information:
Carl Zeiss Industrial Metrology
6250 Sycamore Lane North
Maple Grove, MN 55447
Phone: (763) 559-0402
www.zeiss.com

Sunnen

INTRODUCES TWO-STAGE HONING EFFICIENCY

Sunnen's new HTG series tube hones are designed as oil field workhorses with high-volume throughput and increased part capacity. The standard HTG-10000 is capable of handling standard part lengths up to 30 feet (9.14 m) and weights up to 17,600 lbs. (8,000 kg). The HTG's ID range is 2 to 24 inches (50.8 to 609.6 mm), double that of previous generation machines, with an OD capacity of 26 inches (660.4 mm) standard, and up to 48 inches (1,219.2 mm) as an option. The machine is suitable for instrument piping and down hole equipment applications, such as pumps/motors and hangers, and large hydraulic cylinders like those found on offshore oil platform stabilizers. A new proportional load control hydraulic system enables maximum utilization of hydraulic power, delivering power where needed during operation for maximum efficiency. The system can deliver up to 40 hp (29.83 kW) to the spindle and up to 18 hp (13.42 kW) to the tool stroking system, for metal removal rates of 200+ in^3/hr (3,500 cm^3/hr) using superabrasives in steel as needed.

For increased speed and accuracy, the HTG uses a new precision hydraulic feed system that combines the brute force of hydraulics with the finesse of a sophisticated control. This includes servo position control of the feed system actuator, electronic pressure control, closed loop feedback,

and ability to operate both standard tooling as well as two-stage tooling. The machine features a 100-foot/min (30.48 m/min) maximum stroke velocity, 0–300 rpm spindle, and a tool feed force of up to 2,500 lbs. (11,200 N). The new servo system on the feed axis provides increased accuracy, as well as improved maintenance, service and setups with no tuning required.

Another upgrade is a next-generation two-stage hone head capable of roughing and finishing without changing abrasives. The two-stage hone head is available with conventional abrasives or superabrasives. The U.S.-made HTG is available in two standard model sizes—the HTG-4000 with 4-meter (13.12 ft) stroke length and the 10-meter (32.81 ft) stroke HTG-10000. Built-to-order machine sizes include 6-, 8-, 12-, 14- meter stroke lengths, and custom-length designs can be quoted.

To save floor space, the HTG's electrical panel is mounted with the hydraulic power unit. The power unit is designed for ease of maintenance with accessible components, an easy-to-clean tank, cleanable air filters on the oil heat exchanger, and oil temperature/level indicators on the machine and on the control. A new work base has an improved design sloped to the center of the machine for removal of honing "mud," and includes standard full-time coolant flush. This means operators spend less time shoveling out the base of the machine during high production runs. The HTG features a new easy-to-use Siemens PLC touch-screen control system with zero shutoff, feed pressure control, cross hatch angle calculator and a joystick to allow the operator to move the honing tool over the stroke length. The new control system features complex software to maximize process efficiency in the background while providing easy-to-use, intuitive touch-screen operation. The power of the hydraulic system is controlled with precision, enabling better control for tough parts such as blind bores and shoulders. A load meter enables detection of tight bore conditions and the bore profile is included on the display. Other features designed to increase productivity include feed system soft start, and integral load meter and recommended settings based on part dimensions and material. The standard chain vise fixture is taken to the next level with a new ease of movement on the machine rails for locating during setups, as well as improved ergonomics and adjustments.

For more information:

Sunnen
7910 Manchester Rd.
St. Louis, MO 63143
Phone: (314) 781-2100
www.sunnen.com

MAG

DEVELOPS PROCESS TO PRODUCE GIANT GEAR

MAG proved it knows how to use machine tools, as well as build them, by developing a process that used a horizontal boring mill (HBM) and specially designed tools to cut 588 teeth in a 19-meter (62.5 ft) diameter gear assembly weighing 60 tons (54,836 kg).

The two-piece gear assembly, made of ASTM A290 steel, consists of a 24-section track, which serves as the base, and a 12-section upper gear rack. The MAG team designed dedicated fixtures for each operation and special tooling for cutting and finishing the gear teeth on an HBM.

"We cut the gear teeth on an unconventional machine," said Mark Huhn, project manager at MAG Fond du Lac. "In most cases the tooth involute would be generated by the machine itself, but we used a tool with the involute built into the cutter, which was accomplished by grinding the tooth form into the cutter first."

The gear teeth were manufactured to American Gear Manufacturers Association (AGMA) Gear Quality No. 6, and the gear was assembled to a pitch diameter concentricity of .8 mm (.031 in). The track, comprised of A148M Grade 620-415 castings, required a special cutter to produce a 2.127 degree surface angle.

"The angle was circular interpo-

MAG cut 588 teeth in a 19-meter gear assembly using dedicated fixtures and special tooling (courtesy of MAG).

Cutting and finishing the gear teeth was accomplished on a horizontal boring mill (HBM).

lated onto the track surface and we designed a special cutter to cut the angle on one of our gantry-type machining centers," Huhn said.

For more information:
MAG IAS, LLC.
13900 Lakeside Circle
Sterling Heights, MI 48313
Phone: (586) 566-2400
www.mag-ias.com

Rustlick

RELEASES ULTRACUT PRO

Rustlick recently announced the release of a new premium, bioresistant water-soluble oil cutting fluid. Ultracut Pro delivers reliable performance in a wide range of machining and grinding applications. R&D chemist Steve Badger II explains, "Ultracut Pro uses the very latest lubrication and anti-microbial technologies for a coolant that is high-performance, long-lasting and extremely hard water stable." The dependable performance of Ultracut Pro makes it a first choice for manufacturers working for industries as diverse as aerospace, automotive, agriculture, energy, industrial components, job shops, plastics and transportation. Ultracut Pro delivers proven resistance to microbials, like bacteria and fungus, and is shown to last longer than competitive coolants without the need for costly additives. In independent lab studies, Ultracut Pro lasted longer and outperformed top competitors in bacterial and fungal resistance. This hard water stable formula is effective in a wide range of light, moderate and heavy-duty applications, including

continued

machining, cutting, grinding, milling, broaching, threading and turning. "Ultracut Pro's tight emulsion guarantees consistent, dependable lubrication and cooling for great tool life and surface finish with less carryoff," states Badger. Ultracut Pro contains no boron, phenol, nitrites, triazines, sulfur, copper or SARA 313 reportable chemistry and is easy to recycle or dispose of with conventional techniques and equipment. Ultracut Pro is available in five- or 55-gallon containers and is recommended for ferrous and nonferrous metals, aluminum, brass, bronze and copper. The nonchlorinated version, Ultracut Pro CF, is recommended for all metals, including titanium and high nickel alloys.

For more information:
ITW ROCOL North America
3624 West Lake Avenue
Glenview, IL 60026
Phone: (800) 657-5278
www.itwfpg.com

Northfield

RELEASES RUBBER GRIP CHUCK

Northfield Precision Instrument Corporation, a designer and manufacturer of precision workholding chucks, introduces their Automobile Rubber Grip chuck. This special chuck grips automotive "rubber coupling" drive shaft joints. The small collet in the center grips a 14 mm ID to an accuracy of less than 1/10,000 in. T.I.R. The three extended fingers pass thru three bolt holes, expand, and "face clamp" only. These fingers simulate "bolting" the flange in place the way it is used in the vehicles. Drive shafts are balanced from 3,500 rpm to over 6,000 rpm. Northfield Precision designs and manufactures accurate air chucks for any lathe, boring machine, grinder or VMC. Models include through-hole, high-speed and quick-change. Available in SAE or metric, in sizes from 3" (76 mm) to 18" (457 mm). Accuracies of 0.001" to 0.00001" (0.254 m) guaranteed. Custom work-holding chucks and jaws are available and free engineering assistance is offered.

For more information:

Northfield
Precision Instrument Corp.
4400 Austin Blvd.
Island Park, NY 11558
Phone: (800) 810-2482
www.northfield.com

Mitutoyo

OFFERS USB INPUT DEVICE

Mitutoyo America Corporation recently announced the availability of a new USB input device that streamlines the interfacing of Mitutoyo Digimatic hand measurement tools with PCs. The new USB Input Tool Direct: USB-ITN includes seven models—each model is dedicated to a specific type of cable plug/connector pin configuration. The new design negates the need for two cables, lowering overall costs by as much as 32 percent. When connected to a PC's USB port, the USB Input Tool is automatically recognized as an HID (Human Interface Device) keyboard device—a standard Windows driver. No special software is required. A USB keyboard signal converter translates Digimatic display values to keyboard signals. This enables the direct inputting of data into the cells of off-the-shelf spreadsheet software, such as Excel. Data can also be automatically entered into Notepad or similar programs.

continued

Data capture is much faster than manual entry. Additionally, reliability is increased because transcription errors are eliminated. Optional Mitutoyo *USB-ITPAK Measurement Data Collection* software further enhances the productivity of USB Input Tool Direct: USB-ITN by facilitating set-up. Excel input destinations (workbook, sheet, or cell), cell-fill direction (right or down), cell fill intervals, and other settings can be specified. Sequential, batch, or individual measurement methods can be selected. *USB-ITPAK* also enables mouse button, function key and foot-switch functions.

For more information:
Mitutoyo America Corporation
965 Corporate Blvd
Aurora, IL 60502
Phone: (888) 648-8869
www.mitutoyo.com

OSG

HIGHLIGHTS CUTTING TOOLS AT EASTEC

OSG Tap and Die, North American Headquarters of OSG Corporation, is a manufacturer of taps, end mills, drills and thread-making tools. At the 2011 Eastec Exposition—held May 17–19 in West Springfield, Massachusetts—OSG will be exhibiting an extensive line of high technology cutting tools, including the 3-flute coolant-through Mega Muscle carbide drill, the HY-PRO CARB 3-D and 5-D drills and the brand-new VGx and UVX end mill series. OSG's 3-flute coolant-through Mega Muscle drill is designed specifically for drilling at feed rates 1.5 to 2 times faster than a 2-flute drill. It may also be used at lower RPMs, which decreases the amount of wear and prolongs tool life. This design also leads to higher hole accuracy with less work hardening, which gives secondary operations such as tapping even more tool life. The brand-new HY-PRO CARB VGx Variable Geometry End Mill is an innovative milling tool with capabilities of chatter-free and stable consistent milling in mild to severe milling operations. Last but not least, the EXOCARB-Aero UVX End Mill is one of OSG's latest innovations on variable index and variable helix end mills, designed exclusively for exotic aircraft materials.

For more information:
OSG Tap and Die, Inc.
676 E. Fullerton Ave.
Glendale Heights, IL 60139
Phone: (630) 790-1400
www.osgtool.com

Duo Plus

OFFERS EFFICIENT GRIP SYSTEM

One of the most important keys to success in modern workpiece production is efficiency. Machines are supposed to perform more than just one task with fewest separate steps and tool changes possible. Dual high-pressure clamping sets like the

continued

Duo produced by Allmatic-Jakob Spannsysteme GmbH are suitable for such applications. With the Duo Plus 125, Allmatic unites the advantages of its predecessor model with some additional innovations. Duo Plus 125 has been equipped with Allmatic's grip system. That way, clamping unfinished parts as well as pre-machined workpieces of different geometries is possible without any ado. In earlier systems like Titan, T-Rex, and Centro, Allmatic has already incorporated its grip system which is a modular assembly kit with exchangeable jaws. Duo Plus is primarily meant as a mono block in large machines such as horizontal milling machines. However, the high-pressure clamping set can be mounted to a machine table, too. The fully enclosed maintenance-free high-pressure spindle facilitates chip removal and maintenance and guarantees reproducible clamping forces of up to 40 kN.

For more information:
Allmatic-Jakob Spannsysteme GmbH
Jägermühle 10
87647 Unterthingau
Phone: +(49) 8377 929 0
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ModulTherm® 2.0

Worm Gears—Higher Energy Efficiency and Less Strain on Resources

Dipl.-Ing. Hermann Siebert,
Head of Application Engineering,
Klüber Lubrication

Introduction

The concepts of sustainability and energy efficiency are gaining importance in power transmission engineering. More than ever, manufacturers are looking for ways to reduce raw material consumption and attain a lower energy consumption and hence a better CO_2 balance. This can be done by making machines more efficient, extending the lifetime of components as well as maintenance intervals. The added benefit of such measures is a reduction of operating costs.

Gear Lubricants

The increase of gear efficiency harbors a frequently overlooked potential for increasing the efficiency of a machine as a whole. A very direct and effective way of increasing power transmission efficiency—which goes along with excellent wear protection—is a changeover from mineral-oil-based to synthetic lubricants. Synthetic lubricants based on polyalphaolefin, ester or polyglycol oils, for example, have proven to reduce energy costs and in addition extend the service life of gears. The possible extent of efficiency increase, however, depends on the type of gears: while gears featuring a low percentage of sliding friction, such as spur or bevel gears, offer a relatively low potential, gears with a high percentage of sliding friction enable considerable improvements.

A particularly positive effect can be noted when worm gears are switched over to polyglycol oil: their efficiency has been increased by up to 35 percent. In addition, their lifetime can be extended tenfold. A conversion to synthetic oils offers an enormous potential for savings especially in facilities where a large number of gears are operated—for example in logistics centers, filling stations, breweries or airports. The example in the sidebar illustrates how several million euros can be saved at a large airport.

Tribological factors are decisive in attaining the maximum performance of a machine and its components. When choosing a lubricant for a gearbox or machine, therefore, design engineers should be aware of the characteristics of the various types of lubricants and know how to use them. While, as a rule, synthetic special lubricants tend to be more expensive than mineral oils in terms of the sales price, they pay off after a short time when taking into account efficiency, oil change intervals, oil consumption and the longer lifetime of lubricated components. With such lubricants, gear manufacturers offer their customers the added benefit of lower operating costs.

Tested and Proven

Applied to the effect a lubricant provides in worm gears, the aspects of sustainability and energy efficiency can be translated into the wear behavior and the efficiency of a gearbox. Reduced wear means longer service life of components, which has a consequential effect on the exploitation of resources as less raw material is required to make new components to replace the old ones. Higher gear efficiency has a direct effect on the amount of energy consumed.

On a worm gear test rig developed by Klüber Lubrication specifically for the purpose, the influence gear oils have on the wear and efficiency behavior in heavily loaded worm gears is examined under real-life conditions. Both the speed and the torque of the worm can be measured on this test rig. This is correlated to the worm wheel output torque to calculate the total efficiency of the gear unit. Wear on the worm wheel is measured by determining the weight loss and the abrasion of the tooth flanks occurring during operation. Various temperature values are also measured in the standard version of this test, namely oil sump temperature, mass temperature of the worm shaft, casing temperature and ambient temperature.

Minimizing Wear

A hint that polyglycol oils offer the best wear protection to worm gears is already included in DIN 3996 on the design of cylindrical worm gears. They can help to extend the lifetime of a worm gearbox significantly compared with a mineral oil. The examination of the wear behavior of various polyglycol gear oils made by Klüber Lubrication performed on Klüber's worm gear test rig shows that with these high-performance gear oils wear is even lower than what DIN 3996 stipulates that gear designers should assume for gears lubricated with polyglycol products.

Figure 1 shows that the use of a high-performance polyglycol oil made by Klüber Lubrication rather than a standard polyglycol oil enables an even more significant reduction of wear. Consequently, the worm wheel survives longer with the same load, or the output torque can be increased without dimensional changes. Additional benefits for machine operators are cost savings due to longer maintenance intervals, a lower risk of equipment failure and minimized downtime.

Maximum Efficiency

Maximum energy efficiency of a gear unit means that it produces the highest possible output power for a given input power. The energy lost in the process manifests itself in the form of heat, for example in bearings, O-ring seals or gear wheels. As gear efficiency increases, its temperature will go down. This has a number of positive effects: a decreasing temperature not only extends the oil life, but the service life of seals as well. This in turn reduces the risk of leakage. Another benefit is that fans or air conditioners in production facilities might be switched off, which is another contributor to lower energy costs and a better CO_2 balance (Fig. 2).

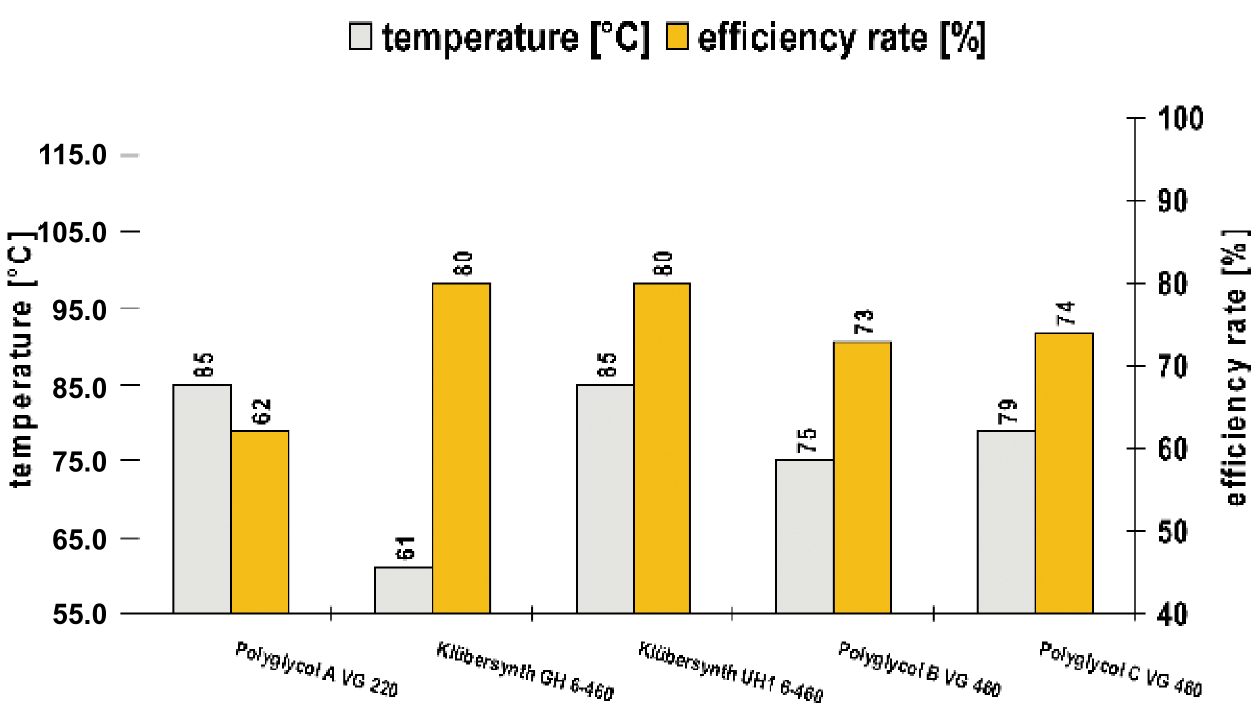
According to DIN 3996, the efficiency of gears in mesh is influenced by, among other factors, the oil's basic friction

continued

Test gearbox: Flender CUW 63
Center distance: 63 mm
Transmission ratio: i 1:39

Test conditions:
Speed n1: 350 rpm
Output torque: 200 Nm ISO VG 220, 300 Nm ISO VG 460
Runtime: 300 h

Fig. 1—Tests performed on Klüber Lubrication worm gear test rig: weight loss and wear rates measured.



Test gearbox: Flender CUW 63
Center distance: 63 mm
Transmission ratio: i 1:39

Test conditions:
Speed n1: 350 rpm
Output torque: 200 Nm ISO VG 220, 300 Nm ISO VG 460
Runtime: 300 h

Fig. 2—Tests performed on Klüber Lubrication worm gear test rig: temperatures and efficiency rates measured.

Airports Offer Enormous Potential for Savings

A large airport may utilize more than 20,000 gear units, for example in conveyor belts and escalators. Approximately 15,000 of them may be spur- and bevel gears with a mean power of 5 kW, and another 5,000 worm gears with a mean power of 15 kW. With some 4,000 operating hours a year and a utilization rate of 40 percent, total power consumption is at approximately 240 GWh. Replacing a mineral oil with a polyglycol special oil will increase the efficiency of all gears by roughly 5.25 percent on average. The power saved thus totals 12.6 GWh—this is the annual power consumption of approx. 3,000 private households. 12.6 GWh equals 12,600 MWh. Based on Germany's average mix of energy sources encompassing fossil fuels, nuclear power and renewables, that's equivalent to the emission of more than

8,500 tons of CO_2*. Based on an energy price of 9.5 cents per kWh, more than a million Euros can be saved this way.

* Source: CARMA (*www.carma.org*), 2008

continued

coefficient. Consequently, oils with a low friction coefficient offer potential for increasing gear efficiency. Similar to their wear characteristics, polyglycol oils show a lower friction coefficient than other base oils. Suitable additives can help to further improve the friction coefficient of a polyglycol. Diagram X shows a comparison of various polyglycol oils. The basic friction coefficients determined for Klübersynth GH 6 and Klübersynth UH1 6 are clearly below the figures to be assumed for polyglycol oils according to DIN 3996 (Fig. 3).

The described effects of higher efficiency make themselves strongly felt in the energy balance of facilities operating several hundred gearboxes. This is shown in detail in the example of a large airport (see sidebar).

Conclusion

The changeover from mineral-oil-based to synthetic gear oils is a simple and highly effective way of minimizing wear and improving energy efficiency. The extent of optimization possible depends on the individual gear type. Best results are obtained where polyglycol oils are used in worm gears. Additional potential for improvement is offered by polyglycol oils based on special formulations and containing specific additives. Such lubricants enable even longer gear and machine life as well as a lower energy consumption for a given output power.

The result is savings both in terms of financial resources and raw materials. Besides these savings, operators enjoy the benefit of a much better CO_2 balance in their operations.

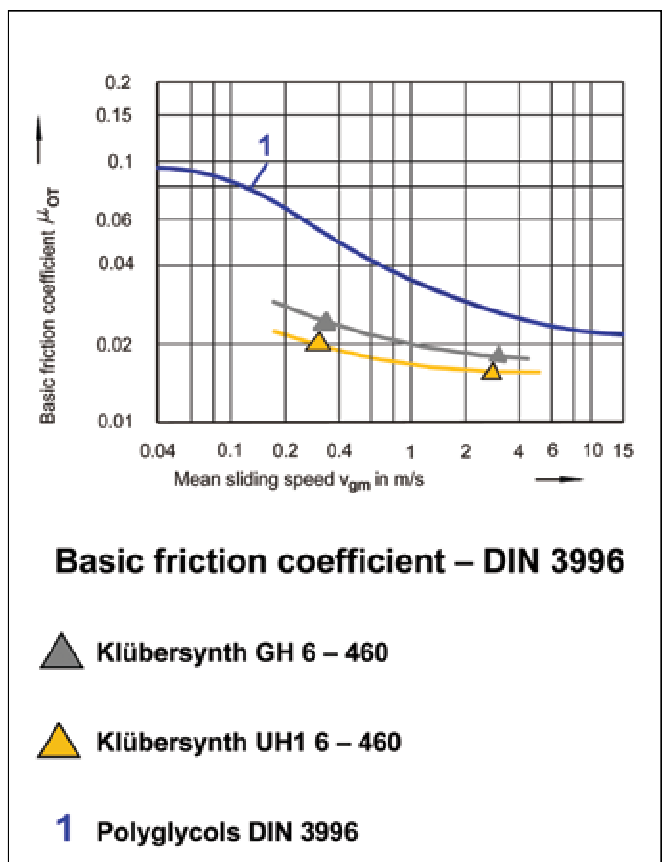


Fig. 3—A low basic friction constitutes a design advantage in worm gears.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The Global Gear Industry

INSIGHTS, PROJECTIONS, FACTS AND FIGURES

Jack McGuinn, Senior Editor

Gear science and manufacturing technology will play a key role in wind power technology's continued evolution and widespread acceptance. (Photos this page courtesy Siemens Corp.)

As reported in an April AP story, "A survey from the National Association for Business Economics finds that economists are hopeful that the broader economy is substantially improving, with rising employment reported for the fifth quarter in a row. The outlook for employment rose slightly, reaching a 12-year high," and "nearly all of the 72 economists surveyed—about 94 percent—now expect the economy to grow at least 2 percent in 2011."

U.S. firms cut 2.9 million domestic jobs while adding 2.4 million overseas, according to a recent *Wall Street Journal* report—"Big U.S. Firms Shift Hiring Abroad: Work Forces Shrink at Home." It goes on to state that in the last ten years, U.S. multinationals have cut 2.9 million American jobs, while hiring 2.4 million outside the United States.

The report notes that GE CEO Jeffrey Immelt explains the trends this way: "We've globalized around markets—not cheap labor. The era of globalization around cheap labor is over. Today we go to Brazil. We go to China. We go to India. Because that's where the customers are."

In another published report, Bank of America Merrill Lynch economist Ethan Harris said the combination of supply chain problems and higher energy prices are bound to slow manufacturing growth in the months to come. At particular risk are automakers and electronics firms.

Lastly, but significantly, manufacturing's declining share of the U.S. economy over recent decades has reduced the pool of skilled workers available to fill jobs.

continued

Mostly Positive Outlook

Brings Gear Players In Off The Ledge

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Global demand for gears and gear assemblies is forecast to climb 4.7 percent annually through 2013 to $169.5 billion.

Market gains will be driven by rising motor vehicle production, ongoing economic growth, increased manufacturing output and a shift in the product mix toward more expensive, energy-efficient units such as seven- and eight-speed automatic transmissions.

Strong demand in relatively small but fast-growing markets like wind and solar energy will also contribute to gear sales advances. Demand in developing parts of Asia, Eastern Europe, the Africa/Mideast region and Central and South America will outpace product sales in the United States, Western Europe and Japan.

Market gains in the developing world will be fueled by healthy economic growth, continuing industrialization efforts and climbing personal income levels, resulting in higher motor vehicle and other manufacturing output.

In addition, rising standards of living will help stimulate demand for motorcycles and other gear-containing durable goods, boosting both original equipment manufacturing and aftermarket gear sales. China and India will register some of the strongest market advances. China is expected to account for one-third of all additional gear demand through 2013 and will surpass Japan to become the second-largest national market behind the United States By 2018, total gear sales in China will exceed product demand in the United States. Market growth is also expected to be healthy in Indonesia, Thailand, Iran and Russia.

Although advances will be less robust than in developing countries, gear product demand in the United States and Western Europe will increase as well—spurred by renewed strength in motor vehicle output following a period of decline. Gear sales in Japan—even prior to the recent tragic events there—are predicted to slow noticeably, also negatively impacted by a drop in automotive industry production and continued sluggishness in capital equipment markets. Also prior to the disaster, Japan anticipated a more favorable outlook for machinery manufacturing to provide some impetus to growth, along with the large numbers of gear-containing equipment in use that would help support aftermarket gear demand. Only time will tell how that is resolved.

The global market for gears and gear assemblies is heavily reliant on the motor vehicle industry. In 2008, seven-tenths of all product sales were automotive related, with motor vehicle transmissions alone accounting for 45 percent of the entire gear market. Transmission demand will also grow at a faster rate than accessory, driveline, steering and other motor vehicle gear sales through 2013, bolstered by an upturn in industry output in the United States, where more costly automatic transmissions are used in the vast majority of vehicles.

Average-per-vehicle transmission demand will also increase as these products become more complex and as medium- and heavy-vehicle output outpaces that of light vehicles.

Machinery will remain the second largest gear market, but will post somewhat slower gains. However, suppliers will benefit from industrialization activity in China and other developing areas, fueling demand for gears used in construction and manufacturing machinery.

Demand for gears used in all other applications—which include everything from aircraft and home appliances to motorcycles and solar energy systems—will expand more quickly than either motor vehicle- or machinery-related product sales. Advances will be driven by growth in global economic activity and higher income levels, boosting demand for a number of gear-containing products.

Sales of individual gears will rise somewhat faster than demand for gear assemblies, spurred by generally healthy aftermarket sales conditions. Growing demand for high-value individual gears, such as large-diameter units utilized in heavy machinery and

SUMMARY TABLE/ WORLD GEAR DEMAND (million dollars)						
Item	2003	2008	2013	2018	08/03	13/08
Gear Demand	97,850	134,500	169,500	217,000	6.6	4.7
North America:	29,900	32,650	40,450	48,800	1.8	4.4
United States	23,690	24,800	30,100	36,100	0.9	3.9
Canada & Mexico	6,210	7,850	10,350	12,700	4.8	5.7
Western Europe	27,200	32,150	36,200	41,100	3.4	2.4
Asia/Pacific:	30,900	51,100	68,350	94,600	10.6	6.0
China	6,720	17,250	28,900	45,600	20.7	10.9
Japan	15,200	19,500	19,800	21,900	5.1	0.3
Other Asia/Pacific	8,980	14,350	19,650	27,100	9.8	6.5
Central & South	2,910	5,500	7,090	9,340	13.6	5.2
Eastern Europe	4,670	8,850	11,800	15,700	13.6	5.9
Africa/Mideast	2,270	4,250	5,610	7,460	13.4	5.7

(Source: The Freedonia Group, Inc.)

wind turbines, will also help bolster overall dollar gains.

(***The Freedonia Group** is an international business research company that publishes more than 100 industry research studies annually, providing an unbiased outlook and reliable assessment of industries relative to product and market forecasts, industry trends, threats and opportunities, competitive strategies and market share determinations. More than 90% of the industrial companies in the Fortune 500 use Freedonia research to help with their strategic planning.*)

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The Top 7 Gear and Gear-Assembly Manufacturers

1. General Motors (U.S.)
2. ZF Friedrichshafen (Germany)
3. Volkswagen (Germany)
4. Ford (U.S.)
5. Aisin Seiki (Japan)
6. Honda (Japan)
7. Toyota (Japan)

In 2008, five of the companies shown (chart) accounted for 31 percent of world gear demand. A significant slice of that pie is automotive-related (Source/Chart: The Freedonia Group.)

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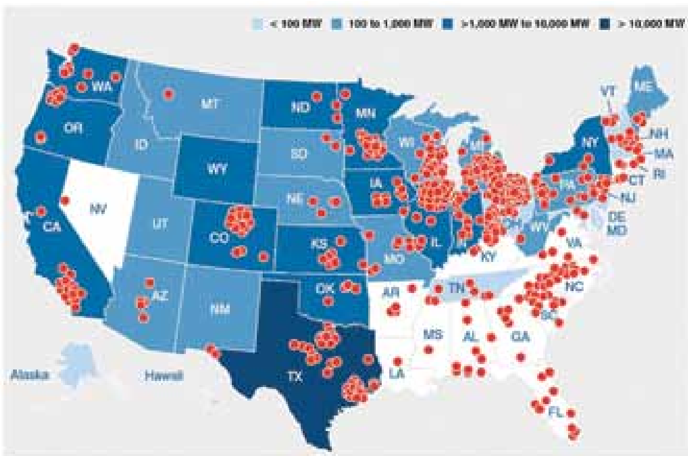
U.S. Wind Industry Makes Gains Despite Economy and No National Energy Policy



A recent AWEA U.S. Wind Industry Annual Market Report for 2010 underscores wind's affordability as a domestic generation source. America's wind power industry grew by 15 percent in 2010 and provided 26 percent of all new electric generating capacity in the United States. With the 5,116 MW added last year, U.S. wind installations now stand at 40,181 MW, enough to supply electricity for over 10 million American homes.

"The American wind industry is delivering, despite competing with energy sectors that have permanent government subsidies in place," says Denise Bode, CEO of the American Wind Energy Association (AWEA). "Wind is consistently performing," she goes on, "adding 35 percent of all new generating capacity since 2007. That's twice what coal and nuclear added combined."

Statistics from the April AWEA U.S. Wind Industry Annual Market Report, developed in conjunction with the AWEA Wind Power Finance and Investment Workshop in New York, reveal that wind continues to be an important player in the nation's energy sector, with lower costs competitive with other generation sources, and it's second in new generation capacity only to natural gas.

"It's simple—wind is affordable," says Elizabeth Salerno, director of data and analysis and chief economist for AWEA. "It's costing less than ever, and competing with other sources thanks to improved turbines built for better performance without a big price tag."

In addition to wind power's increased affordability, the 1603 investment tax credit program contributed to new project starts in 2010. On top of new construction starts, 2010 saw new manufacturing as well. A virtuous cycle was in play—manufacturers continued to respond to the demand and set up shop in the U.S. The industry brought 14 new manufacturing facilities online, consistent with 2009.

"Continued interest and investment by manufacturers in America demonstrates that the U.S. continues to be a global powerhouse for wind development—today and in the future," says Bode. "With these new investments, wind energy is now up to 20,000 manufacturing jobs across 42 states."

As for 2011, the U.S. wind market began with 5,600 MW under construction—more than twice the megawatts under construction at the start of 2010. The extension of the 1603 tax credit in December 2010 provided a signal to investors to continue growing wind in the U.S., as this strong performance indicates. "we remain on track to produce 20 percent of America's electricity by 2030 with wind, as laid out by the Department of Energy during the Bush administration," Bode continues. "We know wind is ready to deliver even more of our portfolio with clean, affordable, homegrown power."

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Fax: 202.383.2505
windmail@awea.org

Just-In-Time is Fine—
But Beware That Supply Chain!

There's a Sudden Breach in Your Supplier Source. Now What?

Global player TÜV Rheinland's Industrial Services division, a full-service inspection, testing and certification company providing both field and laboratory inspection services—including all NDT methods, QC/QA functions and material testing—has introduced a new emergency, rapid response service—the Containment Emergency Response Team (C.E.R.T.). This specialized team has been assembled to quickly minimize OEM production disruptions when out-of-specification products have entered the supply chain.

C.E.R.T. is comprised of certified inspectors who are readily deployable nationally and internationally to respond to suspected quality issues throughout the supply chain. Using non-destructive testing techniques, the team provides rapid on-location inspection, sorting and containment services to quickly minimize the likelihood of faulty products causing production shutdowns.

According to Mark Forbes, laboratory sales manager for TÜV Rheinland Industrial Services, "While adhering to JIT and lean manufacturing principles can lower production costs, the necessitated minimal inventory volumes can magnify the cost of faulty products entering the supply chain. Often,

continued

a product defect is not discovered until well after the part is already on trailers, in transit, on the customer's floor or in assembled products on vehicles. Our team can travel to where the suspect products are located to provide on-site inspection, sorting and containment services to minimize possible production disruptions."

Continues Forbes, "As the recent Japanese earthquake and tsunami have shown, it can only take one missing component to quickly bring an entire production line to a halt. Our C.E.R.T. service can be thought of as the paramedics for the industrial world, quickly getting to the scene and applying on-site services to keep the production line moving."

C.E.R.T. inspectors are ASNT-TC-1A qualified using procedures from A2LA 17025:2005 laboratory accreditations. They are qualified in both laboratory and field services. TÜV is a Tier 1 NDT service provider to General Motors and performs inspections on parts for almost every auto manufacturer. The company is accredited to test to the Pressure Equipment Directive and has inspectors that can test to EN473 (European NDT standards).

Both laboratory and field-based inspection services are available in multiple industries, including: chemical, petrochemical, power generation, food processing, pulp and paper mills and general manufacturing. In addition, services are available to government agencies, storage facilities, architectural/engineering companies and education facilities.

The company also provides inspection services using a variety of inspection techniques including visual, real-time X-ray, computed and film radiography and eddy current, as well as hardness, magnetic particle, ultrasonic and liquid penetrant testing. In addition to its on-location capabilities, TÜV maintains laboratories in Michigan, Pennsylvania and Alabama.

For more information:

Virginia Palmer-Skok
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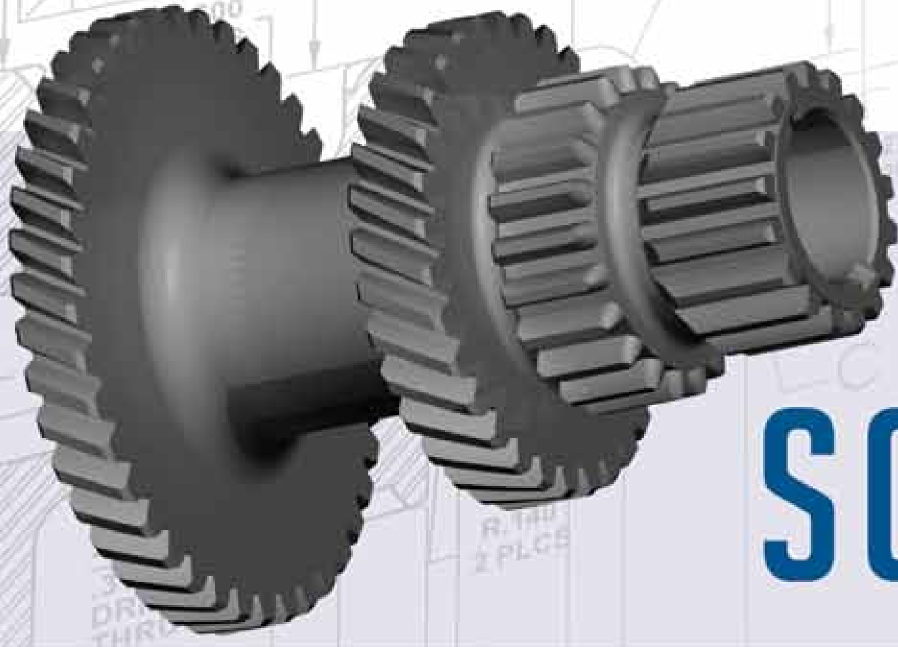

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Micropitting of Big Gearboxes: Influence of Flank Modification and Surface Roughness

Prof. Dr.-Ing. W. Predki, Dr.-Ing. K. Nazifi and Dr.-Ing. G. Lützig

(First presented at the VDI International Conference on Gears, October 2010, Technical University of Munich)

Management Summary

Micropitting—or grey staining—is a fatigue failure on tooth flanks that is mainly influenced by lubrication conditions that can compromise the dynamics and noise behavior of gearboxes. Furthermore, it can lead to larger cracks and, finally, to pitting. Present experiments on micropitting have been conducted, primarily on small-module gears. Test runs on spur gears with a module of 22 mm on a large-gear test bench (center distance 447 mm) and, for comparison with previous investigations, test runs on a standard gear test machine were performed.

The experiments concentrated on lubrication behavior impacted by changing additives, temperatures and circumference velocities on the one hand, and surface roughness and profile form modification on the other. A comparison of calculations according to FVA 259 (Ref. 1) and actual test results help to validate the method for large modules.

Introduction

Gears have long been integral parts of machinery, and vast knowledge of gearboxes and gears has been gained through research. The results of these projects have been used as a basis for a high number of design instructions and calculation methods. Yet a lot of these research projects are carried out on relatively small gears. The positive points of this approach are pretty obvious: The whole test bench gets smaller, easier to handle and, most importantly, more economical. Often, high revolution speeds can be achieved that lead to a high number of load cycles up to the range of infinite life in a substantially short time. But not all results achieved on small gearboxes can be scaled up to big applications. Therefore, tests on big gearboxes with teeth close to the size that are actually used in the application are necessary. Several producers of industrial gearboxes have test benches for big gearboxes that are used for regular, functional and noise tests on gearboxes out of the production line.

Nevertheless, these test benches cannot be used for comprehensive investigations due to cost and capacity reasons. Moreover, the utilization of gearboxes in daily use for systematic investigations with changing parameters like speed and torque is not possible, because they are tied to an economical operation management. As a result, research facilities at universities are challenged.

Big-Gearbox Test Bench at the Ruhr-Universität Bochum

The big-gearbox test bench at the Ruhr-Universität Bochum is designed for tests on big gearbox teeth. It is the biggest spur gear test bench worldwide at a research facility. The key parameters of the test bench are shown in Table 1.

The test bench is built up similarly to the standard back-to-back test bench according to DIN 51354 (Ref. 2), while the center distance is five times bigger. Figure 1 shows a principal sketch of the test bench arrangement. The wheel shaft of a spur gearbox is driven over a one-staged adjustment gearbox by an electrical motor with a variable speed. This spur gearbox is known as a "power return gearbox." A second spur gearbox contains the testing gear set. The tooth geometry of the power return gear set is equal to the test gear set, yet the face width of the test gear set is reduced. The shafts of both spur gearboxes are connected to each other via curved tooth couplings. The torque measurement system is applied to the connecting shaft of the big wheels. The back-to-back connection is closed through the shaft linking the pinions of the spur gearboxes by use of a hydraulic bracing unit. The bracing unit consists of a hydraulic turning motor that is applied hydraulic lubricant through a rotary feed-through. The change of load level and direction is controlled by a proportional and way valve. The valves are controlled via computer, allowing an exact adjustment of the applied torque in operation. Therefore, load collectives and the change of torque direction are possible.

The lubrication systems of both the test and power return gearbox work separately—each with its own lubrication tank—so that both gearboxes can be operated with different lubricants. The lubricant injection systems are equipped with temperature-regulated, cooling water valves and heat exchangers. Furthermore, extra cooling devices keep the room temperature at a constant level. The lubrication system of the test gearbox has additional immersion heaters that heat the lubricant to the desired test temperature before test start. The tank is filled with 600 liters of the test lubricant. From there the lubricant is pumped with a flow rate of 180 l/min to the injection nozzles within the gearbox housing. The lubricant is injected into the starting and ending mesh via flat fan nozzles. More injection nozzles are directed to the bearings for lubrication.

The gearbox housings are prepared to be equipped with axial bearings, so the ability exists to perform tests with spur as well as helical gears. Several measurement systems with torque, temperature and vibration sensors allow for documentation of the operating condition for further test evaluation. Furthermore, a facility of this size needs intensive monitoring and the ability to shut down the test bench automatically in case of damaged bearings or gears.

Aside from the described back-to-back test bench arrangement (Fig. 2, right), often a driver-break arrangement (Fig. 2, left) is used. Therefore, a second inverse connected test gearbox is used to increase the speed and decrease the torque at the break. This second arrangement is closer to the practical application of a common system with driver, gearbox and working machine. Yet this arrangement demands that driver and break are always working at the full power that is applied on the test gearbox. These high powers and the long durations of tests, though necessary for research studies, result in disproportionate effort in energy and heat removal. However, in the back-to-back arrangement the driver just needs to compensate the losses deriving from the bearings, seals and gears. Only the heat resulting from these losses needs to be removed by the cooling facilities.

Micropitting on Gear Teeth

As stated, micropitting is gear wear that is mainly influenced by the lubricant and by the lubrication conditions. The matte-grey appearance of the teeth is due to numerous small flakes and cavities. Micropitting often appears with flank loadings below the calculated pitting endurance strength according to ISO 6336 (Ref. 3). Figure 3, left shows micropit-

continued

Table 1—Test bench parameters

Parameter	Abbreviation	Value
Center distance	a	447 mm
Maximum cycling power	P_{cyc}	6 MW
Maximum brace torque	T_2	114 kNm
Power of electrical drive	P_{Drive}	240 kW
Module	m_n	22 mm
Tip circle diameter	d_{a1} / d_{a2}	403 mm/570 mm
Number of teeth	z_1 / z_2	16 / 24
Face width	b	100 mm
Profile modification	x_1 / x_2	0.18 / 0.17
Load appliance	back-to-back with controlled hydraulic bracing unit	

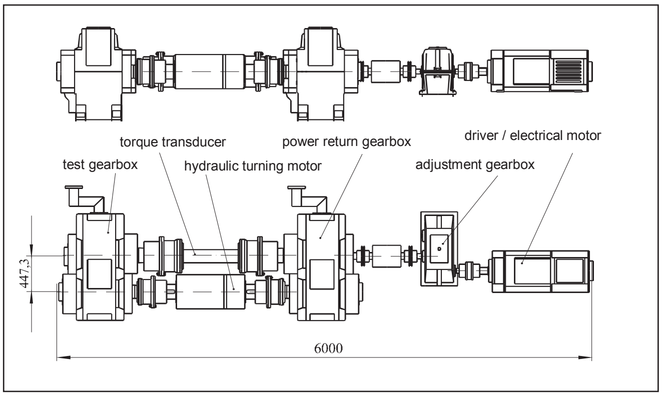

Figure 1—Big-gearbox test bench.

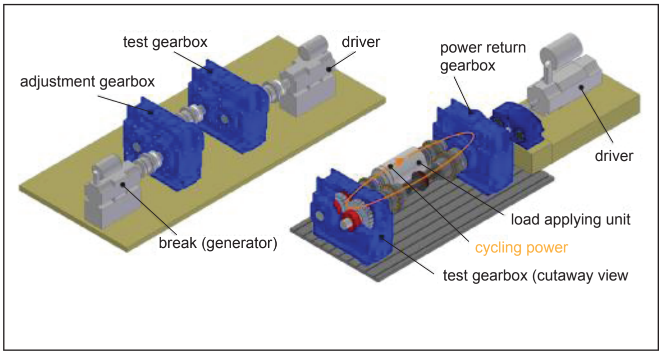

Figure 2—Test bench arrangement concepts: driver-break (left); and back-to-back (right, with a cut section through the test gearbox).

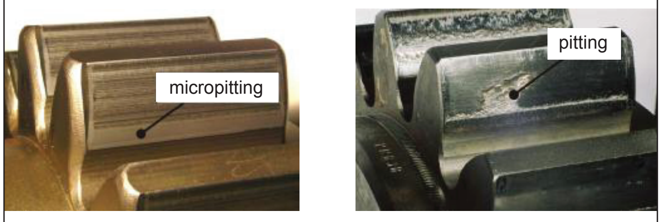

Figure 3—Gear flanks with micropitting (left) and pitting (right).

ting on a gear flank.

The occurrence of micropitting is accompanied by material removal—which leads to profile form deviations. These deviations, with the grey-stained area and weight loss, are conducive to an assessment of micropitting development. Micropitting can lead to deep cracks and eventually to larger spalling (Fig. 3, right). These profile form deviations can change gear dynamics and acoustic behavior.

A popular model of the development mechanisms of micropitting indentifies the surface roughness and the sliding speed as the most important influences on appearance. Via the sliding between the meshing teeth, roughness peaks on the surfaces are pushed aside and folded over, causing a flaked structure. Figure 4, right, shows a flaked surface through a scanning electron microscope. While these flakes are built, the material is plastic-deformed and weakened so that cracks can occur. The loading due to the Hertzian contact—as well as the shear forces caused by friction—make these cracks propagate in a relatively flat manner beneath the surface. A chunk of material drops off when two cracks meet; new flakes are built at the edges of the spalling on every further revolution. This progression can be continued, and the cracks can propagate and cause large pitting.

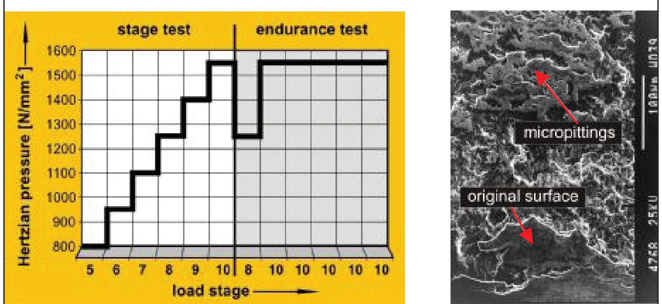

Figure 4—Standard micropitting test procedure (Ref. 4) (left); tooth surface through a scanning electron microscope (right).

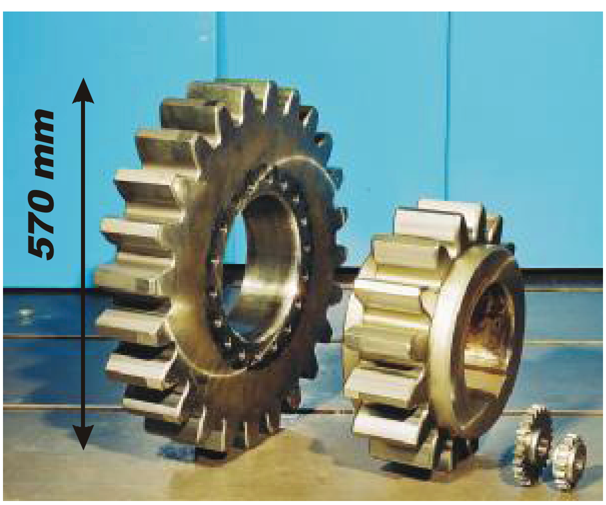

Figure 5—Test gear sets: big (left); and standard (right).

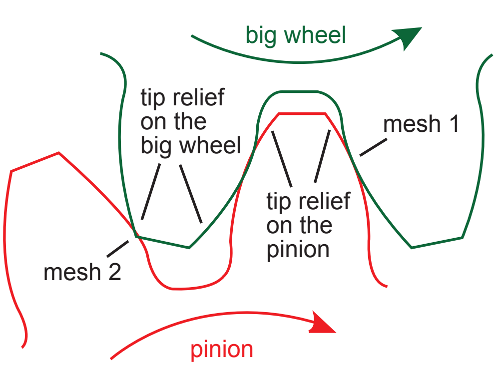

Figure 6—Engagement of gears, with/without tip relief.

Often it can be observed that micropitting stops propagating by itself. The most influencing parameters for the micropitting propagation are the lubricant and additives used, the lubrication temperature, the geometry of the teeth, the surface roughness and the applied load.

Standard Micropitting Test

The investigations on micropitting are done in a standardized test (Ref. 4). Since the lubricant has a significant influence on micropitting occurrence, this standard test is mainly used to prove a certain micropitting load-carrying capacity of lubricants. This test consists of a step and then an endurance component. A defined test gear set is increasingly loaded in six load stages. Each load stage consists of 2.1 million revolutions of the pinion. After each load stage, the pinion is dismounted and weighed, the profile form deviation is measured and the surface is visually inspected. The step test is followed by the endurance test with another six intervals of 10.5 million revolutions of the pinion on a high-load level. For the characterization of the load, the Hertzian pressure at the working diameter is determined. The test-achieved Hertzian pressure of 1,550 N/mm² is close to the material strength limit of the surface according to pitting. The standard test is performed with relatively small gears, with a module of 4.5 mm, a center distance of 91.5 mm and a gear ratio of 24/16. The damage-load stage is defined as the load stage where the limiting profile form deviation of the pinion is exceeded.

Micropitting Test Runs on a Big-Gearbox Test Bench

The test runs on the big-gearbox test bench follow the same scheme as the standard test (Fig. 4, left). The other test conditions and the gear set geometry are chosen according to the standard test, but this time scaled by the factor of five. Therefore, the comparability between results of the standard test bench and the big-gearbox test bench is shown. The Chair of Mechanical Components and Power Transmission employs several standard test benches in addition to the big-gearbox test bench for these kinds of comparisons. The key parameters of the used gears are listed in Table 1, and Figure 5 displays the size difference of the gears. The difference of size has its biggest impact on the testing time—i.e., due to the significant influence of the circumferential velocity on the micropitting behavior of gears, the velocity must remain constant over the different gear sizes. For that reason, the revolutions-per-minute are decreased by increasing gear diameters. A complete stage test on the big-gearbox test bench takes about 25 days, not including assembly time. Compared to the big gearbox, the same test takes only six days on the standard test bench.

Because of the amount of labor expended for assembling and disassembling, the measuring of the big gearbox's pinion after each load stage is not possible. The solution to this problem is found in a replica of the gear tooth flank. With a special-made device, the replica can be produced with good repeatability and can later be arranged on a gear measuring center for measure.

Test Scheme

As stated, the tests on the big gearbox are very time-consuming. Moreover, the costs for gear sets in the desired high quality are very high. Therefore, testing is necessarily limited. The main aim of the tests on the big gearbox is the spot-check comparison of gained knowledge on the standard gearbox with that of the tests carried out on big gear sets. Through this approach the ability exists to see which effects can be transferred linearly to big gear sets and where the size influences the micropitting occurrence.

Thus far, two micropitting research projects have been carried out on the big-gearbox test bench.

The first project (Ref. 5) focused on the following parameters:

- ***Lubricant***: Two mineral oils with different additive packages and one synthetic lubricant (polyglycol)
- ***Operation temperatur***e: Oil injection temperatures of 60° C and 90° C
- ***Circumferential velocity***: Standard velocity of 8.3 m/s according FVA 54 I-IV [4] and a reduced velocity of 4.2 m/s

The second project (Ref. 6) focused on two parameters:

- ***Surface roughness***: Arithmetic mean roughness between Ra = 0.5 µm and 1.0 µm
- ***Profile modifications***: Linear and parabolic tip relief between Ca = 50 µm and 170 µm

Profile modifications are directed deviations from the ideal involute of the teeth. The purpose of the modifications is to compensate the load-applied deformations of housings, shafts and gear teeth. But these modifications also influence load distribution between the teeth. The meshing of two spur gears is divided into phases by which two teeth pairs are in contact and phases where just one pair is meshing. The phases—with two pairs in the mesh of the addendum and the dedendum of the meshing flanks—are in contact.

Figure 6 shows a pinion driving the big wheel—two teeth form the Mesh 1 (right). In the shown position an additional tooth pair is entering the mesh (Mesh 2, left). The teeth addenda have a profile form modification—i.e., there is material loss at the tip of the teeth. The teeth in Mesh 1 now have to bend more until the teeth in Mesh two come into contact; thus Mesh 1 is carrying a heavier part of the load than Mesh 2. This effect can be seen in the run of the Hertzian pressure along the pinion flank (Fig. 7) for the tooth completely on the left side in Figure 6. The unmodified gear mesh shows a relatively equal load distribution between the two tooth pairs in the double engagement area. In return there is a relatively harsh step as the second tooth pair leaves the mesh and just one tooth pair takes over the complete load (single engagement area). The picture changes completely for the same gears with a tip relief of 170 µm; the load distribution changes, and there is a significant lower load at the beginning of the mesh in the dedendum of the pinion. The load increases continuously and the step between the single- and double-engagement area is much smaller.

Micropitting occurs mostly in the dedendum of the driving wheel. This area shows the most disadvantageous sliding conditions on the tooth flank. This is the area where the tip relief lowers the load, and big impact is to be expected. The same applies for lower surface roughness of the flanks, as the high-roughness peaks are the origin of cracks.

Figure 8 shows the result of a test run. The flank picture

continued

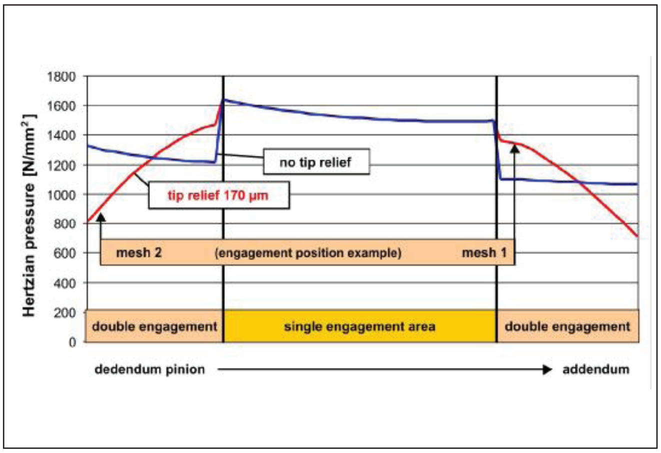

Figure 7—Run of the loading along the pinion flank for modified and unmodified gears.

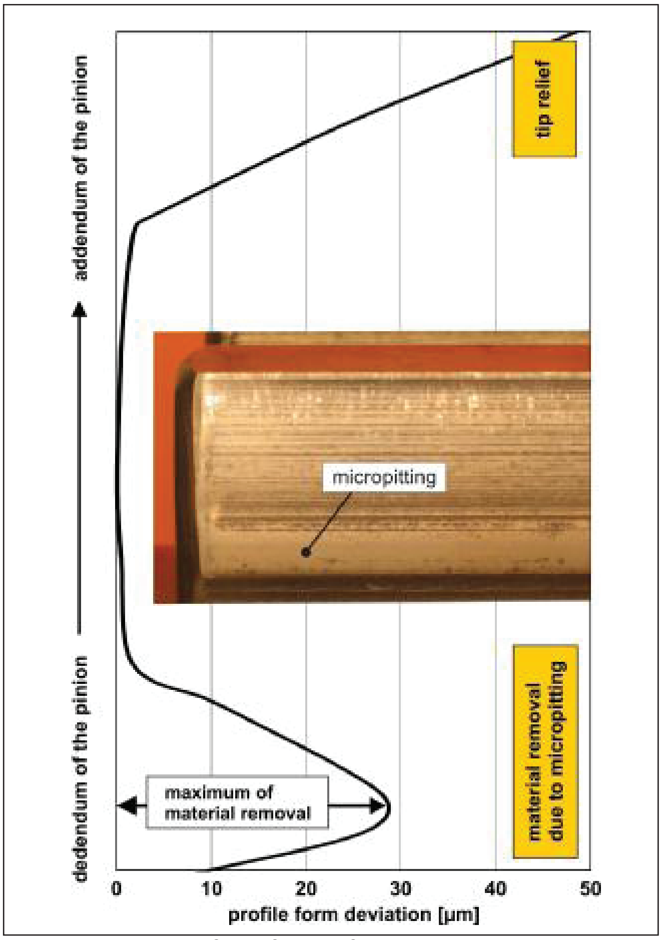

Figure 8—Picture of the flank after test; corresponding profile measurement.

shows micropitting on the dedendum of the pinion flank. The profile measurement diagram shows the deviation of the pinion surface to the ideal involute. On the tip area of the flank, the diagram in Figure 6 shows engagement of gears with/without tip relief, and intentional material removal of the tip relief. Starting at the maximum value of 50 µm, the relief goes linearly back to zero. The middle part of the flank shows almost no deviations. The dedendum area of the flank with micropitting shows a deviation to the unused flank. The microscopic spallings lead to a material removal and a deviation of the flank surface up to a depth of 30 µm. The comparison of these profile form deviations allows an objective relationship between the different test variants. These results are used to validate a calculation method that allows an estimation of the micropitting occurrence on the flanks of a gearbox already in draft stadium.

Test Results

In the first project, lubricants that have a higher micropitting capacity on the standard gearbox show a better micropitting resistance on the big gearbox as well. Yet this improved micropitting behavior does not lead to a lower material removal in the dedendum of the gears. In fact, such lubricants might even lead to a higher possibility of pitting occurrence due to the insufficient flank modifications. The lowered injection temperature from 90°C to 60°C causes higher profile form deviations, thus proving that the additives of the oil must fit to the temperature at and during gear operation. No influence of the circumferential velocity on the profile form deviation depth can be observed. The comparison between the big and the standard gearbox shows that the profile deviations on the big gearbox are deeper, but they don't increase constantly by the proportional factor of 5 between the two gearbox sizes.

In the second project, the systematic variation of the surface roughness proved its influence on the micropitting behavior for big gearboxes. Here high roughness leads to wider micropitting areas. And yet, with low profile modification amounts, the profile form deviations depth becomes independent of the arithmetic mean roughness. The rising of the profile modification amount leads to lower profile form deviations; only in very large modification amounts do we again see higher deviations. Most of the material removal migrates from the dedendum towards the pitch diameter, with rising profile modification amounts. In addition, very small radii were identified at the changing point of the modified flank to the unmodified flank. These radii lead to high pressures and, so it follows, to a higher material removal at these positions.

The comparison calculations with the method described in FVA 259 (Ref. 1) show that for the majority of test runs the calculation leads to larger depth than the experiments; in general, the calculation method is also valid for gears with a large module.

Conclusion

The projects helped to investigate micropitting on big gearboxes. The comparison between the two modules shows that the profile form deviations grow with the module size, yet that there is no constant factor. The lubricant behavior is comparable between the module sizes—i.e., a high micropitting carrying capacity on a small gearbox leads to small micropitting areas on big gearboxes. The influence of the surface roughness on the micropitting behavior is the same for the different module sizes—higher roughness leads to wider micropitting areas. The variation of tip relief forms and amounts shows that this has a huge influence on micropitting. After reaching the minimum of the profile form deviation, a further rising of the tip relief amount leads to deeper profile form deviations again, yet on a different position. The comparative calculations show that in general the calculation method is valid for large modules. Overall, the projects aid in the design of more reliable gearboxes with big modules in respect to micropitting behavior.

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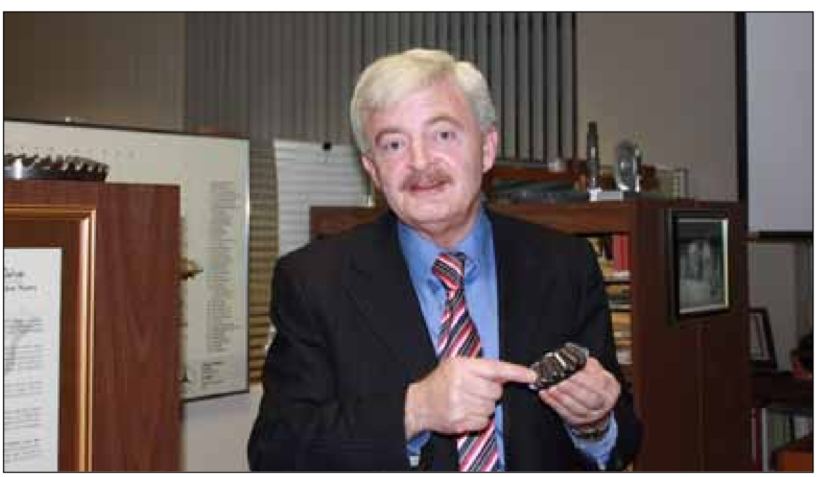
Tribology Aspects in Angular Transmission Systems

Part VI: Beveloid & Hypoloid Gears

Dr. Hermann Stadtfeld

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



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.*

Design

Beveloids are helical gears with non-parallel shafts, with shaft angles generally between 5° and 15°. If two axes are positioned in space and the task is to transmit motion and torque between them using some kind of gears, then the following cases are commonly known:

- Axes are parallel → Cylindrical Gears (line contact)
- **Axes include a small angle → Conical 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 → Hypoid Gears (line contact)**

For beveloid and hypoloid gears, the three bold-printed bullets above show possible gearing systems to be employed for their realization. The axes of beveloid gears always intersect—i.e., they have, at their crossing point, no offset between them (*see next issue's chapter on "Hypoid Gears"*). The pitch surfaces are cones that are calculated with the following formula:

$$z_1/z_2 = \sin\gamma_1/\sin\gamma_2$$
$$\Sigma = \gamma_1 + \gamma_2$$

Iterative solution of:

$$\gamma_1 = \arcsin(\sin[90° - \gamma_1]) \cdot z_1/z_2)$$
$$\gamma_2 = \Sigma - \gamma_1$$

Where:

z_1 Number of pinion teeth
z_2 Number of gear teeth
γ_1 Pinion pitch angle
Σ Shaft angle
γ_2 Gear pitch angle

Beveloids have a parallel depth profile along the face width and are manufactured using a modified cylindrical gear cutting and grinding process. The helical teeth are not wound around a cylinder but around a slim cone that crosses the pitch element under the helix angle. The tooth profile is generated with a continuous tool withdrawal along the face width that results in a variable profile shift and a distorted involute profile. Beveloids have line contact in each angular position if no crowning has been applied. The tooth lead function in face width direction—if unrolled into a plane—is a straight line. It is possible to apply the shaft angle difference to 0° to both or exclusively to one of the two beveloid members. The latter has one member that is a true cylindrical gear, while the second member presents the special case of a face gear (*see "Face Gears"*).

The close relationship between beveloids and spiral bevel gears—and the desire for the additional freedom of an axis offset—led to the development of *hypoloid* gears. The hypoloids are manufactured with face mill cutter heads in a single-index operation (face milling) on bevel and hypoid gear machines. The tooth lead function in face width direction—if unrolled into a plane—is a circle. The hypoloid system offers the possibility of an offset between the axes ("*Hypoid Gears*"). Hypoloids have—at zero offset—a better roll performance than beveloids, especially if deflections under load are present (Refs. 1–2).

Figure 1 shows a photograph of a hypoloid gear set and cross-sectional drawings of both—a hypoloid gear set and a beveloid gear set. For the beveloid example, the shaft angle was split onto both members. All following discussion and analysis results are based on a hypoloid gear set. However, they can be equally applied to beveloid gears because of the identical or similar results.

Analysis

In order to allow for deflections of tooth surfaces, shafts, bearings and gearbox housings without unwanted edge contact, a crowning in face width and profile direction is applied. A theoretical tooth contact analysis (TCA) previous to gear manufacturing can be performed in order to observe the effect of the crowning in connection with the basic characteristic of the particular gear set. This also affords the possibility of returning to the basic dimensions in order to optimize them, should analysis results reveal any deficiencies. Figure 2 shows the result of a TCA of a typical hypoloid gear set.

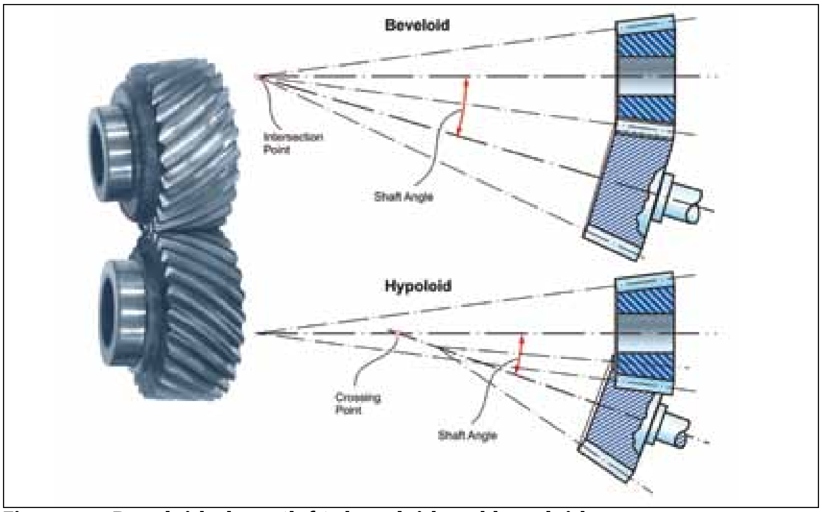


Figure 1—Beveloid photo (left), beveloid and hypoloid gear geometry.

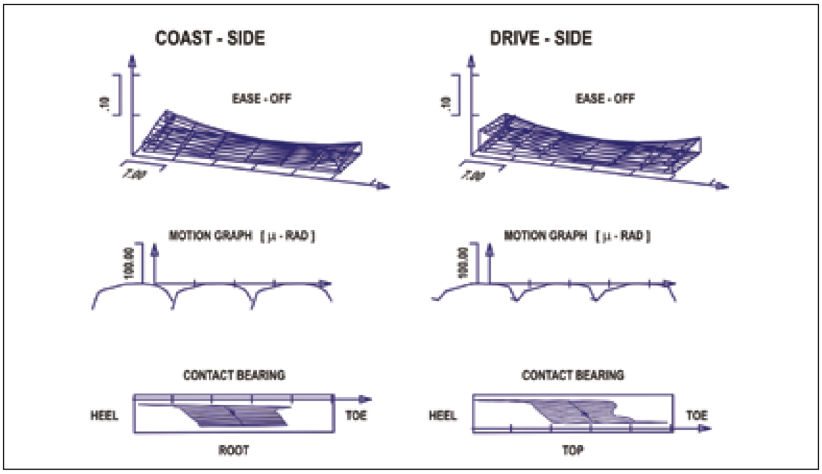


Figure 2—Tooth contact analysis of a beveloid gear set.

Figure 2 displays the analysis results of the two mating flank combinations ("*General Explanation of Theoretical Bevel Gear Analysis*"). 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 have a combination of length and profile crowning, as well as flank twist, to the extent that a clearance along the boundary of the teeth is established.

Below each ease-off, the motion transmission graphs of the particular mating flank pair are shown. The motion transmission graphs

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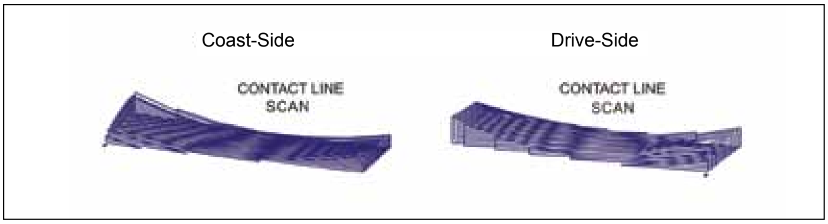

Figure 3—Pinion pitch line and operating pitch line.

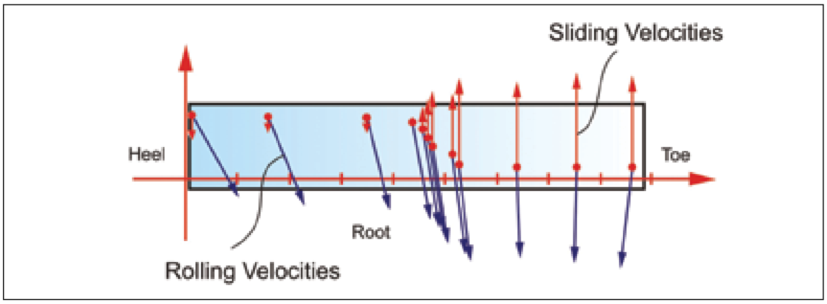

Figure 4—Rolling and sliding velocities of a beveloid gear set along the path of contact.

show the angular variation of the driven gear in the example 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 30 and 60 micro radians (coast and drive).

At the bottom of Figure 2, 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—i.e., rolling the real version of the analyzed gear set under light load on a roll tester, while the gear member is coated with a thin layer of marking compound. The contact lines are almost in line with the face width direction, depending on the small helix angle, which is, in hypoloids, between 5°–10°. 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 contact zone located 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 3 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 in gap kinematic cases ("*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; it is dominated by the geometry of the mating tooth profiles. With hypoloids, the change of the lubrication gap geometry from one contact line to the next is rather small.

Figure 4 shows the sliding- and rolling-velocity vectors of a typical hypoloid 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—i.e., 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 employing the information from the contact line scan (curvature and curvature change) and the tooth surface curvatures perpendicular to the contact line direction (*"General Explanation of Theoretical Bevel Gear Analysis,*" Fig. 8, cases 1–6).

In the discussed hypoloid gear set example, the sliding-velocity vectors are basically profile-oriented. In the top area (Fig. 4, left), the sliding vectors point down. If the rolling velocities, as well as the curvature of the contact line scan, are included in this observation, the reference to one of the lubrication cases in the chapter, "General Explanation of Theoretical Bevel Gear Analysis," seems to be impossible. In truth, the crowning is not contributing to the hydro-dynamic conditions, because the velocities are nearly perpendicular to the horizontal contact lines and the profile crowning presents only a minimal modification of the involute profile form. Here, the evaluation of the macro geometry (in "*General Explanation of Theoretical Bevel Gear Analysis,*" Fig. 5) will lead to the result that—as with the straight bevel gears—lubrication case 3 is applicable above the pitch line. Moving along the path of contact from top to bottom (Fig. 4, left to right), the sliding velocity reduces its magnitude and reaches a magnitude of zero at the pitch line. Below the pitch line, the sliding velocity develops, growing positive magnitudes to the bottom of the gear tooth. The maximal magnitude of the sliding velocities (top-versus-root) is a result of the distance from the pitch line. In this case, the distance between the lowest active flank line to the pitch line is larger than the distance from the pitch line to the top. The rolling velocity vectors are inclined to the pitch line under +15° to –5° and point down to the root. The change in orientation is reflective of the curved tooth and the resulting, changing spiral angle. If the interaction between the rolling and sliding velocities is observed below the pitch line, the unfavorable lubrication case 2 is applicable.

Manufacturing

Traditional beveloids are cut with a cylindrical hobbing process and then hard finished with, for example, a threaded worm grinding wheel in a continuous-indexing grinding process. Hypoloids, on the other hand, are manufactured in a face milling process—the blades, oriented around a circle on the face of the cutter head, pass through a slot while plunging or generating that spot's flanks (Fig. 5). There is no indexing rotation. At the blade tip and in equidistant planes (normal to the cutter head axis), the produced slot width now has a constant width between toe and heel. Because of the small change in circumference between toe and heel, based on the slim cones in beveloids and hypoloids, small, adverse spiral angle changes between convex and concave flanks are sufficient in achieving a proportionally changing slot width (and tooth thickness). As illustrated in Figures 1 and 5, beveloids and hypoloids use a uniform tooth depth design.

Hypoloid gears are soft-manufactured with a high-speed, dry cutting process in a free form bevel gear cutting machine (Fig. 6). Face cutter heads with coated, carbide stick blades arranged around a circle—as outside and inside blades—are used in a single-indexing face milling process (Ref. 3).

The hard finishing process for hypoloid gears after heat treatment is grinding with a cup-shaped wheel. The grinding wheel basically has the silhouette of the face milling cutter, and the grinding machine uses the same settings and motions as the cutting machine. However, the outside and inside dimensions of the grinding wheel are consistent with the finish geometry of the gears, while the cutting blades for soft cutting are dimensioned in

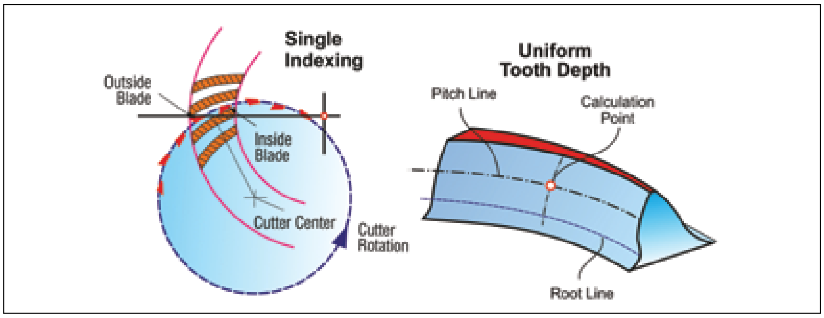

Figure 5—Face milling process combined with uniform (non-tapered) tooth depth.

order to leave a certain stock allowance for the hard finishing process.

Application

Most beveloid and hypoloid gears to be used in power transmission are manufactured with carburized steel and undergo a case hardening to a surface hardness of 60 Rockwell C (HRC) and a core hardness of 36 HRC. It is advisable to give the pinion a higher hardness than the gear (e.g., pinion–62 HRC; gear–59 HRC); this will reduce both the affinity between the pinion and gear surfaces as well as the risk of surface failure. The dominating surface failure mode for beveloids and hypoloids is pitting, beginning on the pitch line. This is virtually identical to cylindrical gears (Fig. 7).

An advantage of hypoloid gearing is the fact that the small sliding velocities in face width direction—which are superimposed with the profile sliding—prevent a zero sliding at the pitch line, thus maintaining a surface-separating lubrication film. The length-sliding in the root and top areas is, even in cases of extremely high offsets, moderately low for all hypoloid designs. This allows the benefit of the hypoloid sliding without risk of scoring. Beveloid and hypoloid gears require no special lubrication; regular transmission oil, suitable for cylindrical gears, is recommended.

The advantages of hypoloid gears vs. beveloid gears are:

- Offset is a welcome design freedom, in addition to a small shaft angle
- Good hydrodynamic conditions, even at pitch line
- Efficiency increase
- Lesser negative influence of housing and shaft deflections under load, due to curved tooth design

Beveloid and hypoloid gears have, in addition to the forces of cylindrical gears, small axial forces that can be calculated by apply-

continued

Figure 6—Hypoloid pinion cutting.

Figure 7—Pitch line pitting on a beveloid flank surface.

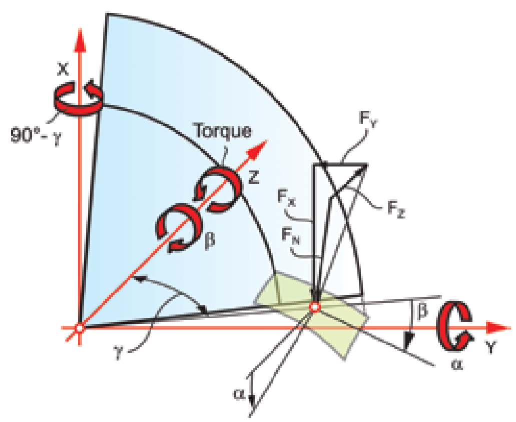

Figure 8—Force diagram for calculation of bearing loads.

ing a normal force vector at the mean point of each member ("*General Explanation of Theoretical Bevel Gear Analysis*"). The force vector normal to the transmitting flank is separated into in 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:

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

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

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

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 spiral angle for the hypoloid pinion. With beveloids, the spiral angles between pinion and gear are identical, and the pitch angle γ is the shaft angle, divided by two.

Offset *a* is positive for cases 1 and 4 and negative for cases 2 and 3 in "*General Explanation of Theoretical Bevel Gear Analysis.*" The pinion spiral angle is positive in all left columns and negative in the right columns (gear spiral angle has the opposite sign). 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 reveal truer bearing loads for multiple tooth meshing, within an acceptable tolerance. A precise calculation is possible with the Gleason bevel and hypoid gear software.

As with cylindrical gears, beveloid and hypoloid gears require regular transmission; a sump lubrication is recommended. The oil level has to cover the face width of those teeth lowest in the sump. Excessive oil causes foaming, cavitations and unnecessary energy loss. The preferred operating direction of hypoloid gears is the drive side, where the convex gear flank and the concave pinion flank mesh. Beveloids have no preferred operating direction, due to their straight flank lead line.

(*Next issue: Hypoid Gears*)

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Flank Load Carrying Capacity and Power Loss Reduction by Minimized Lubrication

B.-R. Höhn, K. Michaelis and H.-P. Otto

Management Summary

Gears are machine components that determine the capability and reliability of many technical products. Continuous demand for higher efficiency and reliability, increased load carrying capacity and endurance life, smaller size, lower noise and vibrations, prolonged service intervals, low environmental impact and low costs will remain the main driving forces in the development of gear drives in the future. The fatigue of contacting surfaces in gears is often the life-limiting factor in transmissions which should, ideally, operate reliably for 20 years or more.

Low oil levels—and, thus, reduced oil quantities—are sometimes used for the reduction of no-load losses in dip-lubricated automotive and industrial transmissions. Specifically, a higher efficiency can be obtained by reducing no-load power losses such as squeezing, splashing and ventilation losses. These losses can be reduced by lowering the oil volume, namely the oil level in dip-lubricated transmissions and the oil flow rate with oil injection lubrication. In these cases, the oil amount required for lubrication may be sufficient, but there may be a lack of cooling oil. This leads to high gear bulk temperatures resulting in thin separating films with higher friction and wear on the mating surfaces and, therefore, an increased risk of gear failures such as scuffing, pitting, micropitting and low-speed wear.

Therefore the objective of this study was to investigate the limits concerning possible reduction of lubricant quantity in gears that could be tolerated without detrimental effects on their load carrying capacity.

Introduction

The lubrication of gears has two major functions: Reducing friction and wear as well as dissipating heat. The power losses, especially the no-load losses, lessen with decreasing immersion depth using dip lubrication. The load-dependent gear power losses are nearly unaffected by minimized lubrication. However, the gear bulk temperatures rise dramatically by using minimized lubrication due to a lack of heat dissipation.

With minimized lubrication, the scuffing load carrying capacity decreased by up to 60%, compared to rich lubrication conditions. Therefore, the dominating influence of the bulk temperature is very clear. Starved lubrication leads to more frequent metal-to-metal contact and the generation of high, local flash temperatures must be considered. An additional factor for the scuffing load carrying capacity calculation in case of minimized lubrication conditions is proposed.

Concerning pitting damage, test runs showed that by lowering the oil level, the load cycles without pitting damage decreased by approximately 50% and up to 75% for minimized lubrication, compared to the results with rich lubrication conditions. The allowable contact stress is clearly reduced (up to 30%) by minimized lubrication. A reduced oil film thick-

continued

ness as a consequence of increased bulk temperatures results in more frequent metal-to-metal contacts causing a higher surface shear stress. In combination with a decreased material strength due to a possible tempering effect at high bulk temperatures, the failure risk of pitting damage is clearly increased. The common pitting load carrying capacity calculation algorithms—according to DIN/ISO—are only valid for moderate oil temperatures and rich lubrication conditions. For increased thermal conditions, the reduction of the pitting endurance level at increased gear bulk temperatures can be approximated with the method of Knauer (FZG TU München, 1988). An advanced calculation algorithm for pitting load carrying capacity calculation at high gear bulk temperatures (valid for high oil temperatures as well as for minimized lubrication) is proposed.

The micropitting risk was increased by low oil levels, especially at high loads and during the endurance test. The micropitting damage is caused by poor lubrication conditions that are characterized by a too-low, relative oil film thickness due to high bulk temperatures. Again, the actual bulk temperatures are of major significance for calculation of the micropitting load carrying capacity.

The wear rate of the gears is almost unaffected by the oil level. Only a slight increase of wear could be observed with minimized lubrication. This increase can be explained by the higher bulk temperature of the gears running under minimized lubrication conditions. The investigations showed that there exists a natural limitation for lowering the oil quantity in transmissions without detrimental influence on the load carrying capacity. Knowing these limitations enables the user to determine the possible potential benefits of reduced oil lubrication. The correct prediction of the actual gear bulk temperatures is of major importance in this context. A method for the estimation of the gear bulk temperature at reduced immersion depth under poor lubrication conditions is proposed.

Gears are machine components that determine the capability and reliability of many technical products. Continuous demand for higher efficiency and reliability, increased load carrying capacity and endurance life, smaller size, lower noise and vibrations, prolonged service intervals, low environmental impact and low costs will remain the main driving forces in the development of gear drives in the future. The fatigue of contacting surfaces in gears is often the life-limiting factor in transmissions which should, ideally, operate reliably for 20 years or more.

Low oil levels—and, thus, reduced oil quantities—are sometimes used for the reduction of no-load losses in dip-lubricated automotive and industrial transmissions. Specifically, a higher efficiency can be obtained by reducing no-load power losses such as squeezing, splashing and ventilation losses. These losses can be reduced by lowering the oil volume, namely the oil level in dip-lubricated transmissions and the oil flow rate with oil injection lubrication. In these cases, the oil amount required for lubrication may be sufficient, but there may be a lack of cooling oil. This leads to high gear bulk temperatures resulting in thin separating films with higher friction and wear on the mating surfaces and, therefore, an increased risk of gear failures such as scuffing, pitting, micropitting and low-speed wear.

Therefore the objective of this study was to investigate the limits concerning possible reduction of lubricant quantity in gears that could be tolerated without detrimental effects on

Table 1—Main geometrical data of test gears

Parameter	Symbol	Unit	C	A
Center distance	a	mm	91.5	91.5
Number of teeth pinion	z_1	--	16	16
Number of teeth gear	z_2	--	24	24
Normal module	m_n	mm	4.5	4.5
Normal pressure angle	a_n	°	20	20
Face width	b	mm	14	20
Profile shift factor pinion	x_1	--	0.182	0.864
Profile shift factor gear	x_2	--	0.172	–0.500
Material		--	16MnCr5	20MnCr5

Table 2—Properties of used lubricants

Name		RL133	FVA2 + 4% A99	FVA 3 + 6.5% A99	FVA 3 pure	M220 + 4% A99
Abbreviation		RL133	FVA2A	FVA3A	FVA3	M220A
ISO VG		100	32	100	100	200
Kinematic viscosity at 40 C (mm²/s)	ν_{40}	101.4	29.8	95	950	210
Kinematic viscosity at 100 C (mm²/s)	ν_{100}	13.45	5.2	10.7	10.7	19
Density (at 15 C) (kg/m³)	ρ_{15}	892	871	872	872	895

their load carrying capacity.

Test Apparatus

Test gears. Table 1 shows the main geometrical values of the two gear types used in this investigation.

For all investigations on the flank load carrying capacity under minimized—as opposed to rich—lubrication conditions, standard test gears (Table 1) were used. For the investigations on pitting and wear, gear type C-PT ("PT" stands for pitting) was used. For evaluation of the micropitting load carrying capacity, gear type C-GF ("GF" stands for Grauflecken, the German word for micropitting), was used (Ref. 4). The gear type C has a close to practical design with a well-balanced sliding speed distribution along the path of contact. For the conducted scuffing tests, gear type A (Ref. 3) was used. Gear type A has an uneven sliding speed distribution along the path of contact in order to the increase the scuffing risk—especially towards the tip of the pinion. The ground flanks of the test gears had a mean roughness of Ra = 0.2 µm–0.4 µm for gear types C-PT and A, respectively. Ra = 0.4 µm–0.6 µm (gear type C-GF). The higher roughness of gear type C-GF increases the risk for the gear-failure-mode micropitting. The mean surface roughness is the only difference between C-PT and C-GF.

Type of lubrication. For the investigations of power loss and bulk temperature for dip lubrication, seven different immersion depths at standstill were chosen (Fig. 1).

For the main scuffing, pitting, micropitting and wear tests, an immersion depth of one and three times the module of the gear was chosen.

An oil/air lubrication device (Fig. 2) was used to lubricate the gears with an oil-air mixture and the bearings with plain oil.

It is a lubrication system in which small, measured quantities of oil from the reservoir are introduced into an air/oil mixing device that is connected via a lubricant supply line to the gear set. The air velocity transports the oil along the interior walls of the lubricant line to the point of application. The continuous air stream to the gear set is only the transporting medium for very small oil quantities—only some milliliters per hour.

Oil/air lubrication is used for tests under severe, starved lubrication conditions where the gear mesh is lubricated in order to decrease friction and wear, but no heat dissipation by cooling oil is available due to the very low oil quantities.

Lubricants. For the tests at different immersion depths and corresponding oil quantities, the FVA reference oils FVA 3 pure, FVA2A and FVA3A (with a high EP-additive content in order to prevent scuffing during the efficiency and bulk temperature experiments), M220 (a mixture of FVA3A and FVA4A) and the CEC reference oil RL 133 were used (Table 2).

Test rig. For all durability tests (pitting, micropitting and wear) and the scuffing tests, a standard FZG back-to-back test rig (Ref. 5; Fig. 3) with a pitch-line velocity of 8.3 m/s was used. For the wear tests at 0.05 m/s, a speed reducer was installed between the motor and drive gear. For the test program at a pitch-line velocity of 30 m/s, one of the FZG test rigs was equipped with a variable speed, three-phase asynchronous

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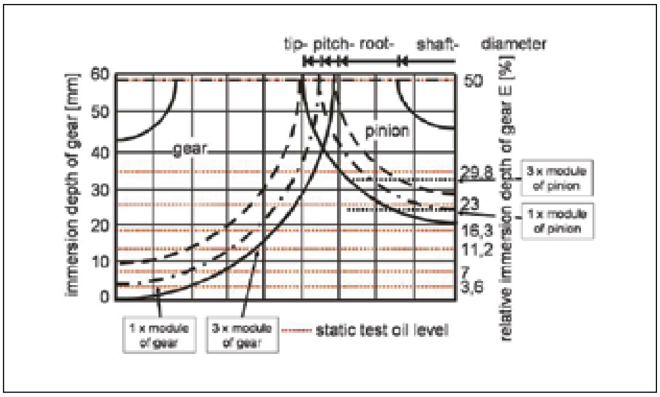

Figure 1—Oil levels at standstill for testing for dip lubrication.

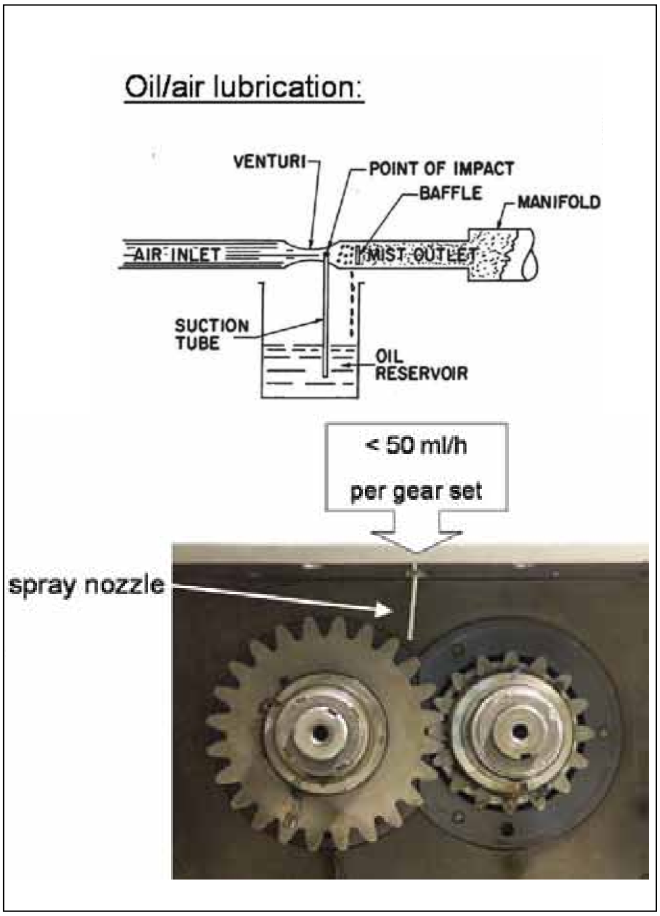

Figure 2—Oil/air lubrication device.

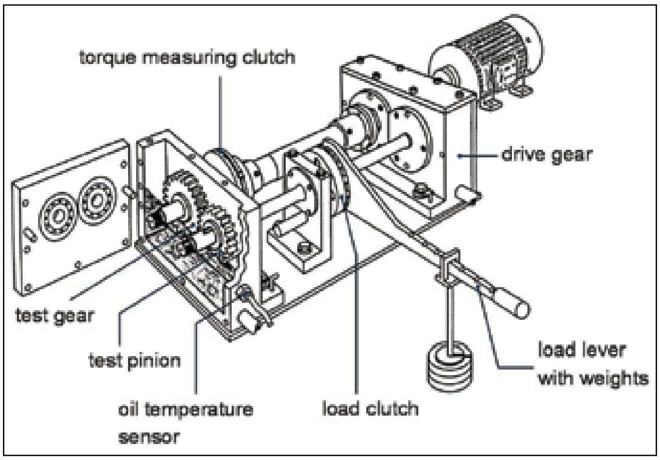

Figure 3—FZG back-to-back test rig.

motor and a speed increaser.

The main elements of the test rig are the two connected gearboxes and an engine behind the drive gear. The test gearbox has an easily removable front and top plate in order to enable a quick change of the test gears. The two spur gear units (test and drive gear set) are connected by two parallel shafts.

In order to put a specific load on the gear flanks, one of the shafts can be divided into two separate parts connected with the load clutch. With the load lever and different weights, a defined torsion can be generated. Table 3 shows the different load stages L_S in the FZG scuffing, pitting and micropitting test, the corresponding pinion torque T_1 and the Hertzian stress at the pitch point p_C.

The great advantage of the closed power loop is that the engine only has to provide the power loss of the two gearboxes.

Therefore the back-to-back test rig can also be used for efficiency tests by installing a torque- and-speed measuring device on the input shaft.

Too, one tooth of the pinion and one tooth of the gear can be equipped with a bulk temperature sensor in order to measure the actual gear bulk temperatures for different operational conditions.

Power Loss and Bulk Temperature with Minimized Lubrication

Load-dependent and no-load losses. Using either dip lubrication with high to very low oil levels, or oil/air lubrication at very low oil quantities, showed no significant change in the load-dependent losses (Fig. 4) and is in excellent agreement with the theoretical values calculated according to Ohlendorf (Ref. 10).

The friction regime is obviously not changed by using minimized lubrication; even small oil quantities (less than 30 ml per hour) are able to reduce friction in the gear mesh, quite the same as with abundant lubrication (oil level up to centerline of the gears).

Tests with oil/air lubrication—compared to tests with oil-moistened gears—showed the importance of a continuous lubrication. Otherwise, an immediate scuffing damage is the fatal consequence.

The no-load losses, which are dominated by the gear splashing and squeezing losses, decreased significantly with decreasing immersion depth (Fig. 5).

Using oil/air lubrication with oil quantities below 100 ml/h results in very low, no-load losses when compared to dip lubrication with high, medium and very low immersion depths. No splashing and almost no squeezing effect can be observed in this case.

Gear bulk temperatures. The bulk temperature of gears has a strong influence on the load carrying lubricant film. Increasing temperatures result in decreasing oil viscosity, which in turn produces thinner oil films under constant load and speed conditions.

According to Oster (Ref. 11), the bulk temperature ϑ_M of the gears can be calculated based on the idea that the generated heat in the tooth contact is dissipated by the tooth surface area to the surrounding oil, which works as a cooling fluid with the following empirical equation:

$$\vartheta_M = \vartheta_L + 7400 \left(\frac{P_{VZP}}{a\,b} \right)^{0.72} \frac{X_S}{1.2\,X_{CA}} \qquad (1)$$

Where:

ϑ_M	Gear bulk temperature, °C
ϑ_L	No-load oil temperature, °C
P_{VZP}	Load dependent gear losses, W
a	Center distance, mm
b	Tooth width, mm
X_S	Lubrication coefficient
X_{Ca}	Profile modification coefficient

Table 3—Load stage L_S, pinion torque T_1, and Hertzian stress at pitch point p_c

Load stage, L_S, (-)	FZG scuffing test		FZG pitting test		FZG micropitting test	
	Torque, T_1, (Nm)	Hertzian stress at pitch point, p_c, (N/mm²)	Torque, $T1$, (Nm)	Hertzian stress at pitch point, p_c, (N/mm²)	Torque, T_1, (Nm)	Hertzian stress at pitch point, p_c, (N/mm²)
1	3.3	146	3.3	172	3.3	172
2	13.7	295	13.7	346	13.7	346
3	35.3	474	35.3	565	28.8	510
4	60.8	621	60.8	741	46.6	649
5	94.1	773	94.1	922	70.0	795
6	135.3	927	135.3	1,105	98.9	945
7	183.4	1,080	183.4	1,287	132.5	1,094
8	239.3	1,232	239.3	1,470	171.6	1,245
9	302.0	1,386	302.0	1,651	215.6	1,395
10	372.6	1,538	372.6	1,834	265.1	1,547
11	450.1	1,691	450.1	2,016	319.3	1,698
12	534.5	1,841	534.5	2,197	378.3	1,848

The factor X_S is set to 1.0 in the case of dip lubrication, and to 1.2 in the case of oil-injection lubrication. The equation for the bulk temperature does not differentiate between pinion and gear nor does it take into account different immersion depths for dip lubrication or the rotational direction (Fig. 6).

From the experimental data, a simple equation is derived that allows calculation of a mean X_S factor for different immersion depths and rotational directions for gear types A and C in the FZG test rig:

$$0.3 \le X_S = 0.35\left(\frac{e}{d_a}\right)^{-D} \le 3.7 \qquad (2)$$

Where:

e	Immersion depth of gear, mm
d_a	Tip diameter of gear, mm
D	Parameter for rotational direction

With:

D = 0.75 for the standard (meaning long distance between oil sump and gear mesh) rotational direction, and D = 0.5 for the reversed rotational direction, meaning direct transportation of cooling oil to the gear mesh.

For oil/air lubrication, a constant X_S = 3.7 can be used that indicates starved lubrication without any heat dissipation by cooling oil, but only by convection and conduction to the surrounding metal components.

Several negative consequences arise from high bulk temperatures:

- Decreasing oil film thickness
- Shift from mixed lubrication conditions to boundary lubrication conditions
- Higher degree of metal–to–metal contacts
- Higher local surface shear stress
- Reduced material strength due to possible tempering

All these effects have a negative influence on wear, scuffing, micropitting and pitting. This amounts to an overall negative influence on the durability of the gears.

Flank Load Carrying Capacity with Minimized Lubrication

Scuffing investigations. Scuffing is an instantaneous form of damage caused by the occurrence of solid-phase welding between sliding surfaces in the area of high sliding speed. Gear scuffing is characterized by material transfer between the sliding tooth surfaces—it destroys the flanks of the gears in the direction of the involute and in the direction of the face width.

The scuffing tests were performed according to the standard test procedure described in ISO 14635 (Ref. 5).

The scuffing load carrying capacity with oil/air lubrication at the medium pitch-line velocity of v = 8.3 m/s is comparable to that obtained with minimized dip lubrication with one or three times module immersion depth of the gear. At the high pitch-line velocity, the decrease of the scuffing torque due to oil/air lubrication was even stronger (Fig. 7).

The calculation methods for the scuffing load carrying capacity published in DIN 3990 (Ref. 1) and ISO/TR 13989 (Ref. 7), the flash and integral temperature methods are based on the idea that if the local surface temperature (bulk temperature plus local flash temperature) exceeds a permissible

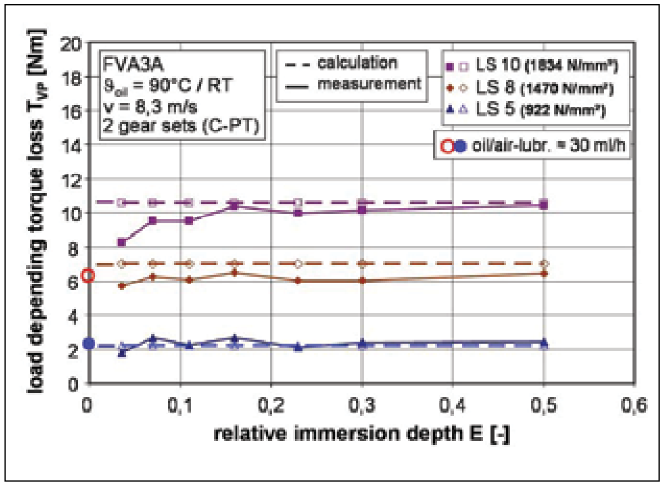

Figure 4—Load-dependent torque losses with minimized dip and oil/air lubrication (Ref. 12).

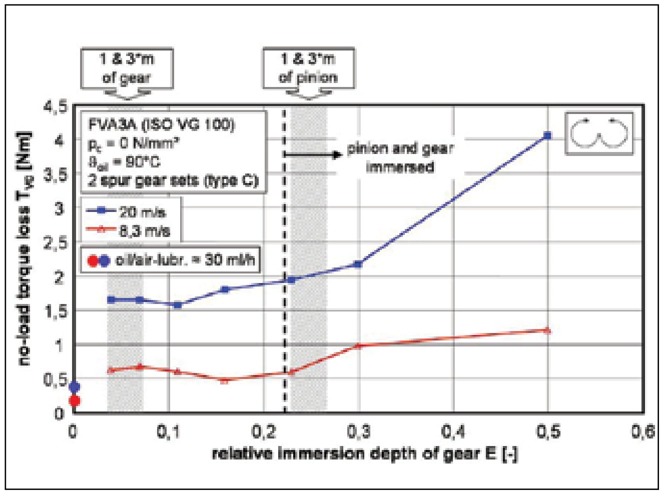

Figure 5—No-load torque losses with minimized dip and oil/air lubrication (Ref. 12).

contact temperature, scuffing occurs. In contrast to the flash temperature method, the integral temperature method averages the flash temperature and supplements empirical influence factors.

According to the integral temperature method, which was presented first by Michaelis (Ref. 9), scuffing damage occurs if a mean critical flank temperature, called integral temperature ϑ_{int} exceeds the allowable integral temperature ϑ_{intP}.

A lower allowable integral temperature with decreasing immersion depth is the main result of the conducted investigations for both pitch-line velocities and for both rotational directions (Fig. 8). A maximum decrease of the permissible integral temperature of approximately 25% to 30%—by using minimized lubrication—can be observed.

Despite the fact that calculations of the integral temperatures were based on the measured bulk temperatures, a clear decrease of the permissible integral temperature could still be observed. Therefore, the increase of the bulk temperatures due to an increasing lack of cooling oil with minimized lubrication is not the only reason for a decreased scuffing capacity—i.e., the lubricant supply to the gear mesh must also be taken into consideration.

Therefore, the integral temperature can be calculated with

continued

the modified equation:

$$\vartheta_{int} = \vartheta_M + C_2 \, C_{LS} \, \vartheta_{flaint} \qquad (3)$$

Where:

ϑ_{int}	Integral temperature, °C
ϑ_M	Actual bulk temperature, °C
C_2	Weighting factor (C_2 = 1.5 for spur gears)

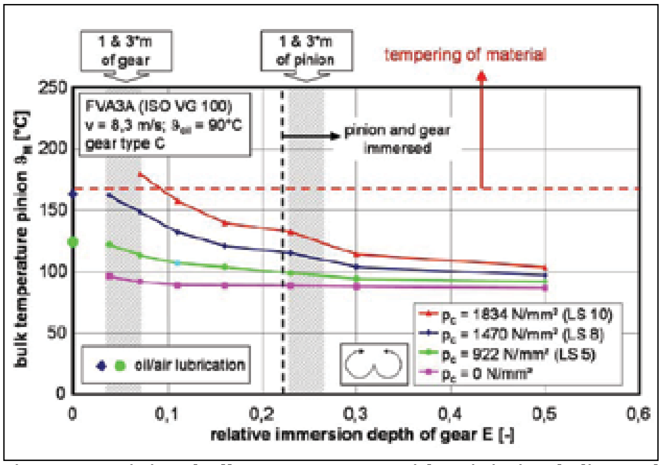

Figure 6—Pinion bulk temperature with minimized dip and oil/air lubrication (Ref. 12).

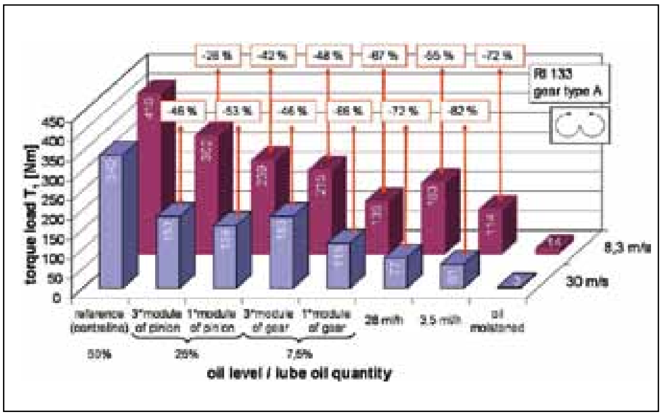

Figure 7—Scuffing load carrying capacity with minimized dip and oil/air lubrication (Ref. 12).

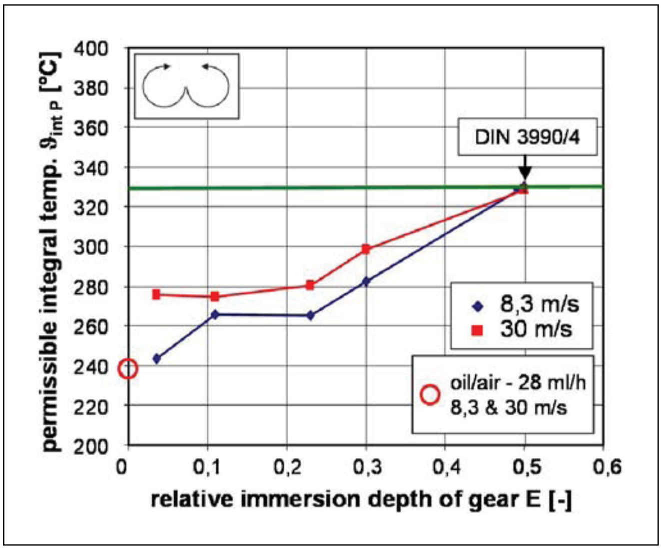

Figure 8—Permissible integral temperature with minimized dip and oil/air lubrication (Ref. 12).

C_{LS}	Lubricant supply factor
ϑ_{flaint}	Flash integral temperature, °C

The factor C_{LS}, which takes the lubricant supply to the gear mesh into account, can be roughly estimated (Table 4). Due to a partial lack of lubricant on the gear flank, the local flash temperature is probably increased due to a higher coefficient of friction.

For judging the scuffing performance of a candidate oil under full and minimized lubrication conditions, it should be tested on the FZG test rig close to the actual practical application (speed, oil level, oil supply to gear mesh, etc.). Scuffing tests close to the actual pitch-line velocity, with an oil level up to the centerline and a reduced oil level below three times the module of the gear, should be run. In this way the permissible integral temperature can be calculated from the test results.

In order to transfer the test results to practical applications, the actual gear bulk temperature has to be either calculated or, preferably, measured.

Pitting investigations. Pitting damage is a fatigue failure mode that occurs if the material strength is exceeded locally or across the whole width of the tooth by the imposed stress. Typically, pits are the result of surface or subsurface fatigue cracks caused by local metal-to-metal contact at the roughness peaks.

The pitting tests were performed according to the standard test procedure described in the FVA Information Sheet No. 2/IV (Ref. 14).

By lowering the oil level from the centerline to three times the module of the gear, the endurance limit decreases clearly by –8% (Fig. 9). In lowering the oil level to only one time the module of the gear, the endurance limit decreases compared to the one for centerline significantly—by –23% (Fig. 9).

The calculation of the surface durability concerning pitting damage for spur gears is based on the contact stress at the inner point of single pair contact B or the contact at the pitch point *C*, whichever is greater.

The allowable contact stress s_{HP} can be calculated knowing the endurance limit $s_{H\,lim}$ with Equation 4:

$$\sigma_{HP} = \frac{\sigma_{H\,lim} \, Z_{NT}}{S_{H\,min}} \, Z_L \, Z_v \, Z_R \, Z_W \, Z_X \qquad (4)$$

Where:

s_{HP}	Allowable contact stress, N/mm²
$s_{H\,lim}$	Endurance limit for contact stress N/mm²
Z_{NT}	Life factor for contact stress
S_{Hmin}	Required safety factor
Z_L	Lubricant factor
Z_v	Speed factor
Z_R	Surface roughness factor
Z_W	Material factor
Z_X	Size factor

The lubricant factor Z_L accounts for the influence of the lubricant viscosity, the speed factor Z_v accounts for the influence of the pitch-line velocity and the roughness factor Z_R accounts for the influence of the surface roughness on the surface endurance capacity.

For the tests with minimized lubrication at a pitch-line velocity of 8 m/s and 30 m/s, the values for the endurance limit s_{Hlim} (Fig. 10) can be read from the S-N curves. The endurance limit sHlim is a material factor that should be equal for all tests if the calculation algorithm of the standard takes all operational conditions of the conducted tests correctly into account.

However, testing of gear type C-PT with rich and poor lubrication conditions resulted in a significantly reduced endurance strength s_{Hlim} with minimized lubrication (Fig. 10). The lubricant factor Z_L, the roughness factor Z_R and the speed factor Z_v of DIN 3990 (Ref. 2) or ISO 6336 (Ref. 6) obviously do not fully take into account the actual tribological conditions in the tooth contact, especially the operating viscosity. Due to a lack of sufficient cooling oil, gear bulk temperatures rose dramatically. This results in a reduction of the material strength due to tempering effects and high surface shear stress due to low oil film thicknesses caused by low operating oil viscosities. Knauer (Ref. 8) investigated these effects with oil injection lubrication at medium to high oil temperatures. He derived from his experiments the empirical factors Z_ϑ, Z_λ and Z_μ.

The pitting tests with minimized dip and oil/air lubrication showed that these factors can also be used for poor lubrication conditions.

The factors Z_L, Z_v and Z_R of the present standard can therefore be replaced by Z_ϑ, Z_λ and Z_μ. Thus the correct endurance strength s_{Hlim} for gears running under increased operational temperatures can be calculated by taking into account the actual thermal conditions in the tooth contact. The allowable contact stress s_{HP} can be calculated correctly for the usual value for the endurance limit s_{Hlim} using Equation 5:

$$\sigma_{HP} = \frac{\sigma_{H\,lim}\, Z_{NT}}{S_{H\,min}}\, 1.1\, Z_\lambda\, Z_\mu\, Z_\vartheta\, Z_W\, Z_X \qquad (5)$$

In essence, the pitting load carrying capacity is strongly correlated to the actual thermal operating conditions. Knowing the actual gear bulk temperatures enables precise prediction of pitting lifetime and endurance limit.

Micropitting investigations. Micropitting damage is a fatigue-failure mode and occurs normally below the pitch line in the area of negative sliding and consists of microscopic disruptions.

Because of these disruptions, the flank starts to look grey in the affected area. With increasing running time, the whole flank can be damaged by micropitting.

The micropitting tests were performed according to the standard test procedure described in the FVA Information Sheet Nr. 54/I–IV (Ref. 4).

Micropitting tests with minimized lubrication were conducted at an oil temperature of 60°C. Considering the whole test, the profile deviation for different oil levels is almost the same during the first load stages (load stages 5 to 8) and starts to increase for the two low immersion depths during load stage 9 and 10 compared to the reference test, with an oil level up to the centerline (Fig. 11).

The test done with dip lubrication at low oil levels only showed a higher profile deviation in load stage 9 during the stepwise test compared to the test with high oil levels.

The test gears running with oil/air lubrication show a higher profile deviation already after the test run in load stage 6. Obviously the greater increase of the bulk temperature, which results in a lower oil film thickness, with the test gears running with oil/air lubrication results in a higher micropitting risk.

An expanded calculation method is presented by Schrade (Ref. 15) which evaluates a safety factor against the occurrence of micropitting. The evaluation is based on the assumption that micropitting can occur if the minimum relative oil film thickness GF at any point in the entire zone of action is lower than a corresponding critical value λ_{GFP}, which is the limiting permissible relative oil film thickness. A micropitting safety factor can be determined by comparing the minimum relative oil film thickness with the corresponding critical value that is derived from gear testing of the lubricant (e.g., FZG micropitting test; Fig. 12).

continued

Table 4—Lubricant supply factor	
Lubricant supply conditions	CLS
Good	1
Medium	1.3 - 1.6
Poor	1.8 - 2.2

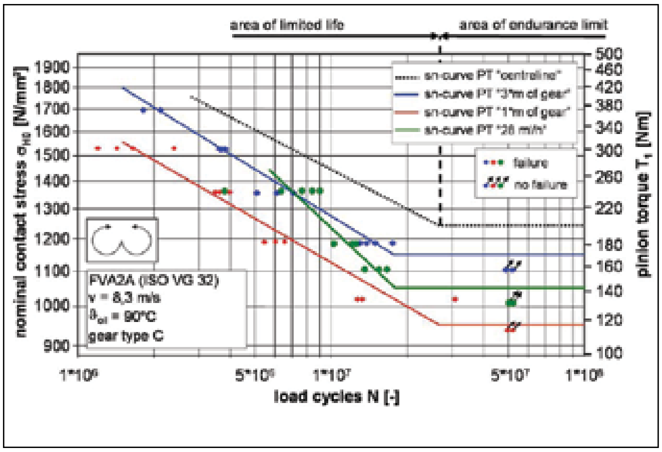

Figure 9—S-N curves for pitting damage using different immersion depth and oil/air lubrication at medium pitch-line velocity of 8.3 m/s (P_A = 50%).

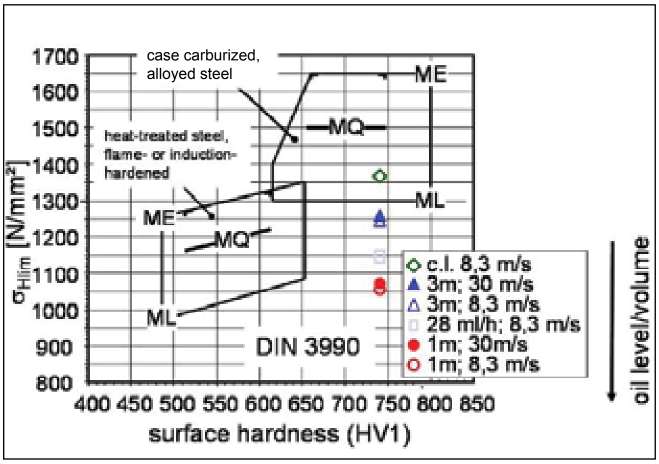

Figure 10—Endurance limit for contact stress σ_{Hlim} according to DIN 3990 or ISO 6336 (Ref. 6).

The micropitting safety factor is strongly related to the relative oil film thickness, which is dominated by the local contact temperature.

The micropitting tests with gear type C-GF showed an increasing risk of profile deviations when using minimized lubrication. As the calculation algorithm for prediction of the micropitting load carrying capacity is strongly determined by the actual lubrication conditions in the tooth contact (local flash temperatures, actual gear bulk temperatures, local Hertzian stress, local sliding speed and local oil film thickness), the test results with minimized lubrication are in good accordance with the state of art. Of major influence are the actual bulk temperatures, which have to be either calculated or measured for prediction of the micropitting risk of actual applications.

Slow-speed wear investigations. Wear is a continuous stock removal on the active gear flank. The course of wear failure is a continuous removal of material in every load cycle. The results are scratches over the whole flank. In contrast, scuffing failure is a sudden damage with a comparable scratch structure in the tooth height direction. Due to low sliding velocities in the area of the operating pitch point, scuffing failures are limited to the areas of the addendum and dedendum flank. With increasing running time a profile form deviation starting below and above the pitch line and finally on the whole flank can be observed.

The slow-speed wear tests were performed according to the standard test procedure described in the DGMK Information Sheet, Project No. 377 (Ref. 1). The wear tests with FVA3 with dip lubrication were conducted using the reference oil level (centerline) and three times and one times the module immersion depth of the gear. The wear slightly increases with decreasing immersion depth or by using minimized oil/air lubrication (Fig. 13).

No basic change can be observed concerning the low-speed wear failure with minimized oil/air lubrication. Due to the low pitch-line velocities and therefore low sliding speed between the contacting surfaces in the gear mesh, low losses are generated. Therefore the small amount of oil is sufficient for the necessary heat dissipation.

Wear is the result of sliding and pressure acting on the contacting surfaces of flanks. The amount of wear is determined by the relative oil film thickness l related to sliding speed, contact pressure, lubricants, surface structure as well as oil additives and oil contaminants.

According to Plewe (Ref. 13), a linear wear coefficient c_{lT} can be evaluated from the measured weight loss of the pinion running under different lubrication conditions by taking into account the number of load cycles during the test run. For evaluation of the slow-speed wear category the linear wear coefficient can be plotted over the minimum oil film thickness hmin at the pitch point (Fig. 14).

The slow-speed wear tests with gear type C showed nearly no effect of minimized lubrication conditions compared to rich lubrication conditions. Again the actual bulk temperatures, which increase due to a lack of sufficient cooling oil, have a dominating influence. Knowing the actual temperatures allows a precise calculation of the wear rate with starved lubrication conditions.

Conclusion

The investigations showed that there exists a natural limitation for lowering the oil quantity in transmissions without detrimental influence on the load carrying capacity.

In summary, minimized lubrication had a negative influence on the load carrying capacity of the standardized test gears.

Due to the increased bulk temperatures resulting from a lack of cooling oil with low immersion depths, the scuffing load carrying capacity was strongly decreased. With minimized lubrication, the scuffing load carrying capacity decreased by more than 60%, compared to rich lubrication conditions.

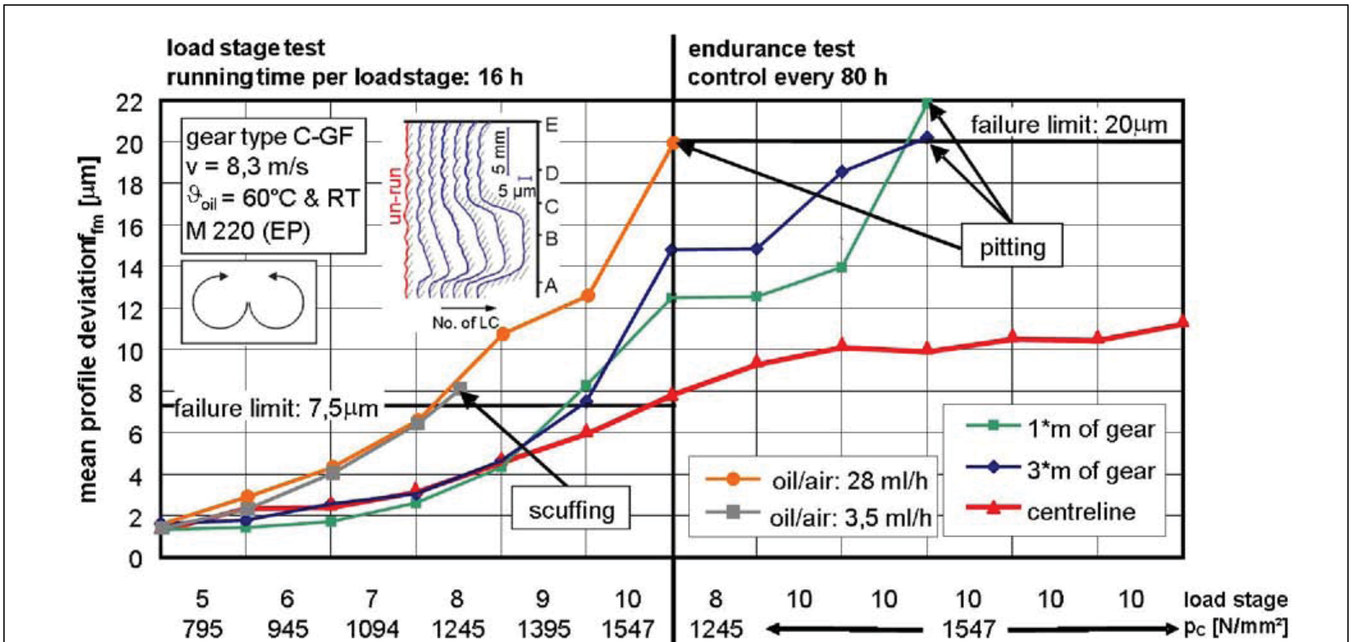

Figure 11—Mean profile deviation of pinion with minimized dip and oil/air lubrication (Ref. 12).

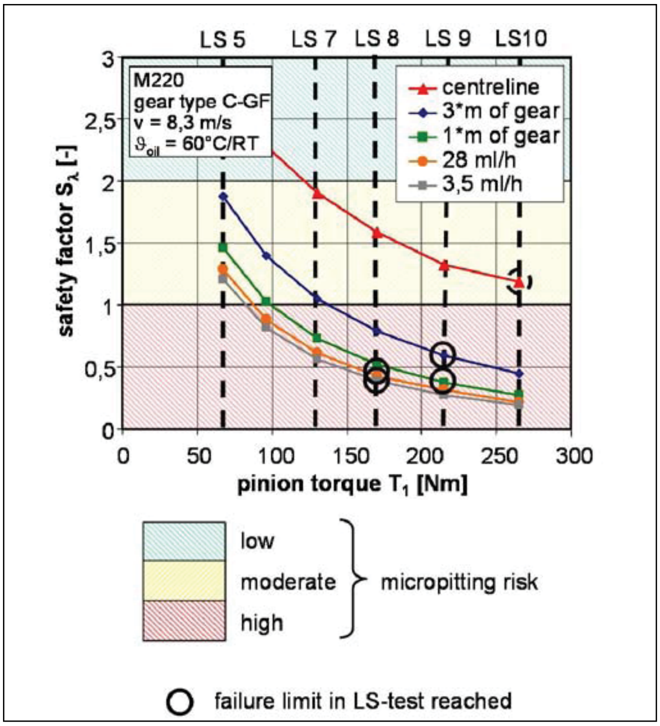

Figure 12—Calculated micropitting safety factor with minimized lubrication.

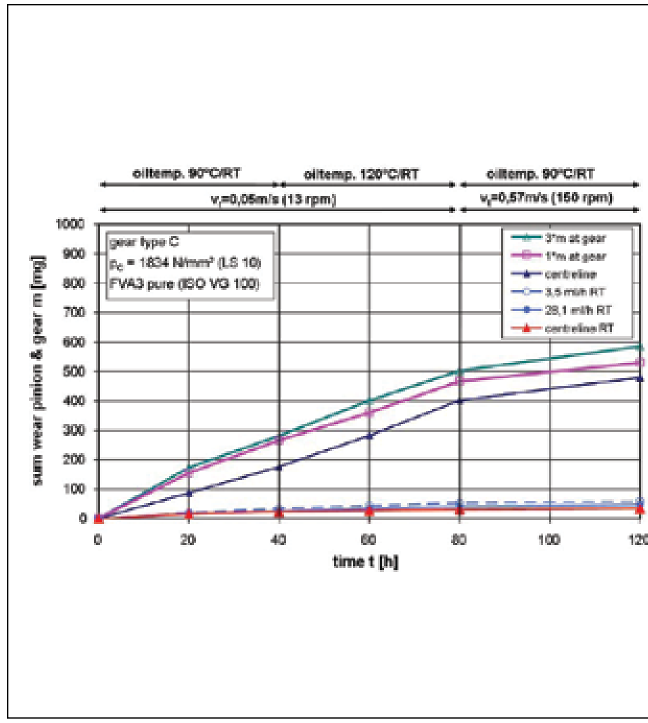

Figure 13—Wear of pinion with minimized dip and oil/air lubrication (Ref. 12).

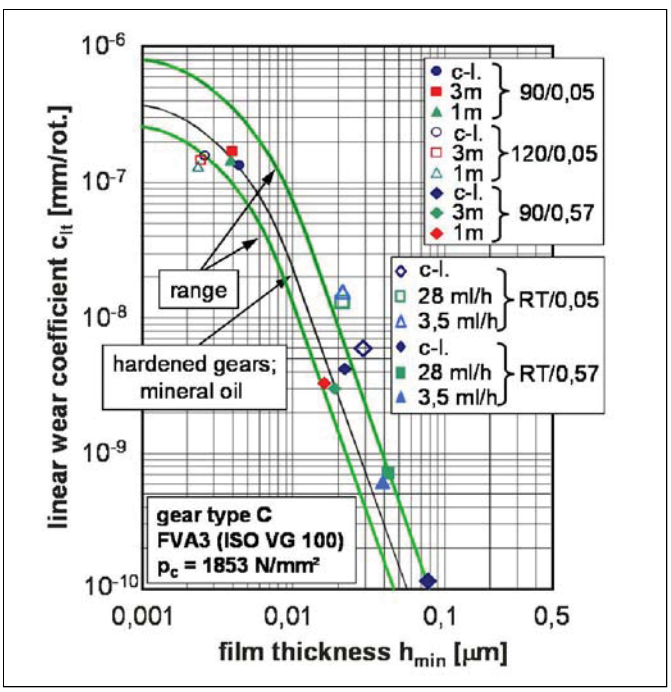

Figure 14—Plewe diagram with minimized dip and oil/air lubrication (Ref. 12).

Concerning pitting damage, test runs showed that by lowering the oil level the load cycles without pitting damage decreased by approximately 50% up to 75% for minimized lubrication, compared to the results with rich lubrication conditions. The allowable contact stress is clearly reduced (up to 30%) by minimized lubrication. High temperatures may result in a reduced material strength due to tempering, and a low oil viscosity results in a low oil film thickness. The common pitting load carrying capacity calculation algorithms according to DIN/ISO are only valid for moderate oil temperatures and rich lubrication conditions. An advanced calculation algorithm for pitting load carrying capacity calculation at high gear bulk temperatures (valid for high oil temperatures as well as for minimized lubrication) is therefore proposed.

The micropitting risk is clearly increased by insufficient cooling of the gears when operating at low oil levels and low oil quantities.

Slow-speed wear is almost unaffected by minimized lubrication in the tests described.

Nevertheless, minimized lubrication is often used nowadays due to the reduced power losses and therefore higher gear efficiency which results, e.g., in a reduced fuel consumption in automotive applications. Knowing the limitations for reducing the oil quantity in transmissions enables the design engineer to build small, lightweight and efficient powertrains in order to comply with the demanding environmental, technical and financial requirements.

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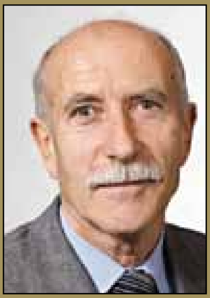
Bernd-Robert Höhn *studied mechanical engineering at the Technical University Darmstadt (1965–1970) and served as an assistant lecturer (1970–1973) at the Institute for Machine Elements and Gears at the Technical University Darmstadt prior to becoming an assistant professor at the university (1973–1979). In 1978, he received his Ph.D. (Dr. Ing.) in mechanical engineering. In 1979 worked as a technical designer in the department for gear development of Audi. By 1982, he was head of the department for gear research and design for the automaker. In 1986 Audi named Höhn department head for both gear research and testing of automotive transmissions, until his departure in 1989 to become head of both the Institute of Machine Elements at the Technical University of Munich and of the Gear Research Centre (FZG). Höhn has served since 2004 as vice president for VDI for research and development and since 1996 has led the working groups 6 and 15 for ISO TC 60—calculation of gears.*



Klaus Michaelis *studied mechanical engineering at the Technical University Munich, receiving his degree (Dipl.-Ing) in 1969. Until 1970 he was a mechanical engineer at Linde, Lohhof, Germany, before working as a research assistant at the Institute for Machine Elements and Gear Research Centre (FZG) at the Technical University Munich (1970–1977). Since 1997 he has served as chief engineer and research group supervisor at FZG for gear load carrying capacity and tribology and experimental hydrodynamics. Michaelis received his Ph.D. in mechanical engineering in 1987.*

Hans-Philipp Otto *studied mechanical engineering at the Technical University Munich, receiving his Dipl.-Ing. degree in 2002. From 2003–2009 he was a research assistant at the Institute for Machine Elements and Gear Research Centre (FZG) at the Technical University Munich and in 2009 received his Ph.D. in mechanical engineering. Since that time Otto has worked as chief engineer and research group supervisor at FZG for gear load carrying capacity and tribology and experimental hydrodynamics, and as head of the FZG-Augsburg branch office—the technical application center of the FZG institute.*

Desktop Gear Engineering

SOFTWARE TRENDS, TECHNOLOGIES AND MARKET FORECAST

Matthew Jaster, Associate Editor

Non-circular gear segments where corrective action was successfully applied by Dontyne Systems (courtesy of Dontyne).

Design activity in the gear industry—by most accounts—is picking up pace in 2011 especially in the transportation and wind segments, and the global machine industry is clawing its way back from the 2008–2010 debacle. Commercial software providers understand this and, in turn, know that offering substantial product updates, addressing customer requests and bolstering their technical support and training programs is essential to remain competitive.

Why go outside the company to evaluate the design components within, say, a gearbox? This is the question that commercial software providers like KISSsoft, Dontyne Systems, GWJ Technology and Romax Technology must answer when discussing their respective software services. Many argue that calculation procedures can be developed internally and don't bear the same investment risks found from outside vendors.

If a case is to be made for commercial gear software, it starts and ends with a fairly simple concept: permanent maintenance. Software has become a critical tool in gear manufacturing, and as technologies advance, these software tools need to be consistently modified and maintained. The increased challenges of cost pressure, global competition, reduced development times and product liability means that making the extra investment in the right software suite might be worth it in the long run.

"You must have the technical knowledge within the company to succeed. This involves not only software development, but also engineering services, expert assessments and software upgrade services," says Gunther Weser, manager at GWJ Technology GmbH in Braunschweig, Germany.

"In the past, gear engineers just had to design gears to be quiet and durable," adds Barry James, chief engineer at Romax Technology. "Now they have to be efficient as well. This additional target places additional burden on the design process."

The Experience Factor

For more than 25 years, KISSsoft AG has been providing a single niche product to gear customers. "*KISSsoft* is a practical tool that was always extended on some engineer's request," says Dr. Stefan Beermann, CEO at KISSsoft AG. "In the beginning it was Dr. Kissling who wrote the program for

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his own purposes. Now it is our customers that ask for specific extensions. Since we are working in the committees defining the respective standards, we also know—and sometimes influence—the way these standards are growing."

Familiarity with the material never hurts, as well. "A technical lead in software can only be established by actually doing the same work as your clients," James at Romax says. "Romax has done more than 60 NVH projects in the last 10 years, designed automotive transmissions that are made in volumes of over 1 million per annum and has completed dozens of wind turbine gearbox designs, up to 5 MW. We see an explosion in the amount of design activity in gearboxes because of the requirements for low carbon vehicles and wind energy."

At GWJ, the focus is on the detailed and practical implementation of the calculation methods. The company's *eAssistant* and *GearEngineer* software is continually being improved and its functionality is increased based on specific user requirements. "The *eAssistant* is available immediately and requires no installation or maintenance. There are no investment costs. It provides a valuable reduction in time and costs. Users appreciate the fast technical advice and support," Weser says.

Experience and engineering support is the reason many companies choose a software developer over in-house packages or a consulting firm.

"There has always been such software available, usually a consultant that also offers software, but you need a lot of manpower to keep software like *KISSsoft* on a current level," Beermann says. "If someone is willing to work around the clock, seven days a week, he can be very successful in a niche. For a larger package, a larger infrastructure is needed. Since prices were quite moderate in the last 20 years for this type of software, everything beyond Excel spreadsheets is hard to make profitable."

"In the past, OEMs created their own gear software suites, but it is impractical to maintain these packages, as the technical experts who write them inevitably retire or are promoted out of the role," James at Romax says. "In the end they have to be replaced by external packages. There is always a place for smaller companies to service low-tech clients or those companies whose budgets cannot stretch to the quality products, but overall they will remain on the peripheries of the industry."

Creating in-house software also leads to high development and maintenance costs, according to Weser. "The trend is towards using professional calculation software to meet increasing demands. However, smaller guys can compete if they combine both the knowledge of creating software with the calculation of mechanical elements."

Dontyne Systems in England keeps its software suite modular and costs down to allow smaller companies to justify the expenditure.

"We have become very adept at working with large companies to provide a hybrid solution that brings their own calculation methods in a modern operating system while allowing direct comparison with common international standards from ISO and AGMA," says Mike Fish, Dontyne Systems. "We are developing more and more building blocks within the software library to help create a customer-specific solution even faster. This has been applied not only to design and rating tools, but also provided links to system models and CAD packages."

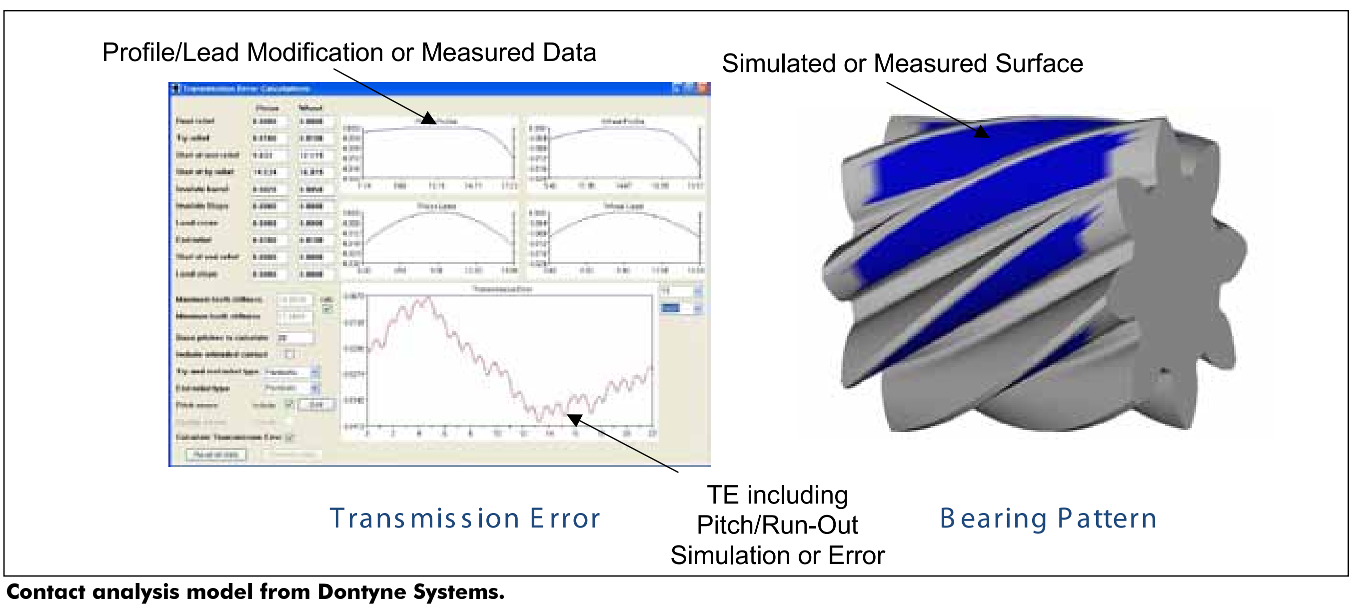
While the market has its fair share of gear-specific software development, making sure the software is being utilized correctly—and efficiently—is an ongoing dilemma thanks to a generational impasse.

The Copy-and-Paste Generation

As was the case in 2008, the quality of gear engineers that have the knowledge to use programs like *KISSsoft*, *RomaxDesigner/RomaxWind*, *GearProduction Suite* and *GearEngineer/eAssistant* isn't what it used to be. Beermann, in fact, believes the issue hasn't gotten much better.

"To be honest, the largest challenge in my opinion is still—and I'm afraid increasingly—the spreading lack of knowledge about gears amongst the engineers. We try to address this with the training we offer. However, since we are a relatively small company we can't do enough training to have a general impact."

"Most of the current students are



Contact analysis model from Dontyne Systems.

not willing to do things like their ancestors, being much more copy-paste oriented. And the universities do not provide the same education as in the past. In Asia there are tons of (engineering) students, some of which are really good," he adds.

GWJ Technology in Germany tries to address the skills issue for both young and old gear engineers by keeping them honest with workshops and training sessions. "Many students already use our web-based calculation software *eAssistant* in the classroom," Weser says. "After they get used to the software, they continue to use it as soon as they start a new job and are able to work efficiently and cost effectively."

No matter how good the gear software is, however, it can never replace the engineers themselves. "Training of young engineers and the retirement of old engineers remains a huge problem. Much of Romax's work involves training engineers on gear technology as well as using the software. This is often mixed in with a consultancy project and/or a bespoke gearbox design project," James says.

Fish at Dontyne Systems believes many gear companies are more aware of this problem today and are addressing it through a variety of initiatives.

"David Brown, for example, has founded a gear academy to teach gear design as a three-year course part time with heavy contribution from industry. An online course is currently being developed at the University of Huddersfield to ensure that there is access to wider and more in-depth knowledge of gear design for trainee engineers. Online learning gives improved access and flexibility both nationally and internationally. I am told from those organizing the course that there has been an amazing and refreshing response to the proposals."

Quality technical support is one area software developers hope will help alleviate some of these issues. Today, support goes far beyond a person on the phone in a remote country telling you which button to press.

"The skills shortage means that you need people in the same time zone and territory to support your engineering needs. To this extent we have engineering offices in the United States, (2), India (2), China (3), Japan (2) and Korea, in addition to the U.K. headquarters. Romax will be adding another in the United States this year and one in Germany," James says.

"To ensure excellent support, in 2010, we launched an online portal where customers can report and track their support issues. This has been very well received by our customers, and we are expecting to solve our 1,000th support ticket later this month. These include a large number of 'what is behind the software?' rather than just 'how do I use the software?' Again, this provides further evidence for the skills shortage and confirms the importance of a strong, technically capable support team," James says.

A variety of workshops are held throughout the year in Braunschweig, Germany for those using software from GWJ. "The user can gain a deeper understanding of basic skills, design strategies or optimizations of machine elements. To provide a good balance between theory and practice,

continued

This bevel gear set was reproduced using software from GWJ Technology (courtesy of GWJ).

the attendees will have the opportunity to work on their own workstations to complete exercises and to put theory into practice. Individual questions are allowed and welcomed during the workshop. If the customer is unable to join the workshops, GWJ also provides on-site workshops on an individualized basis. GWJ also offers interactive software training sessions over the Web," Weser says.

Dontyne Systems conducts a variety of web conferences in order to support customers and agents from all over the world. "We've given product demonstrations and training to personnel in several international locations simultaneously," Fish says. "Companies find the sessions useful for reducing development time and costs. However, we still find the presence of local agents a massive advantage and have formed relationships in several key countries. The feedback they receive and the knowledge of local conditions is an invaluable part of our complete software package."

Furthermore, Fish insists on actively participating in gear events to meet face to face with gear companies. "We had a very good response after our participation in the AGMA FTM and we'd like to participate in this year's AGMA Gear Expo. We think it's important to meet with companies (or do online demonstrations) as we often find they are under some misconceptions about our products."

Stick With What You Know

If permanent maintenance is the key to gear software development, the companies interviewed for this article are practicing what they preach. Each company has a slew of updates and innovations for their respective software suites (*Ed's note: See* Software Bits 2011 *for latest technologies on page 67*).

While the rest of the software world has turned its attention to mobile technology, gear software providers know that maintaining their classic commercial products will produce promising returns in 2011 and beyond.

Beermann at KISSsoft notes that it remains difficult to "wow" customers in gear design software for many reasons. "We only put a practical twist to what was seen five years ago. The IT industry is currently in the clouds, but it's not something which is really accepted for our type of software. Most technical improvements you see in computer technology do not have any impact on our business."

Dontyne considered developing a web application in 2009, but ultimately decided the demand wasn't there, at least not at that moment. "Despite the increase in mobile technology, we've had no inquires for a web-based version of our applications. Some clients may have security issues with the data transfer and from our own product development point of view it is another set of code to maintain."

Though mobile technology is increasing in other areas, gear design remains fairly basic. KISSsoft recently released a free iPhone app for the conversion of hardness values from one system to another; it's the only mobile application the company currently has. "Although the trend is going into the cloud, we think that for our type of software most users prefer the classical software type, installed on a client or server," Beermann says. "Of course, we are evaluating the different ways for an online version of *KISSsoft*, but nothing is going to happen in 2011."

For the most part, mobile application requests might be offered if the demand increases. "We are always looking at new ways of implementing our technology, and have consistently led the world in technical innovations over the last 15 years," James says.

"Currently, there are no fixed plans to support mobile applications but it can all change if there's enough customer demand," adds Weser. "GWJ is open-minded to new technologies and business ideas online because *eAssistant* already offers a web-based application since 2003."

Concentrating on their core products through the recession has prepared many developers for future gains in the market.

"We were going through the crisis quite well, adapting our structures and organization to more rough times," Beermann says. "This is now paying off, as the gear business is coming out of crisis. The only problem we are currently facing is that we have to restrict ourselves to our core business, because we are running out of staff if we try to do everything requested."

Fish at Dontyne says that the recession marked a cautionary period in software development. "There is no doubt the market was essentially frozen from 2008 well into 2010. We have seen a marked increase in activity since the end of summer 2010. The biggest challenge at the moment is not to over-extend despite the enthusiasm for our product and the rapid growth we've experienced so that if a recession hits again, we'll have the resources to ride it out. Diversification may be one way of doing that."

With its *eAssistant* software, GWJ sees great potential on an international level, due to an increased interest in "software as a service" (SaaS) and on-demand services to help increase productivity and efficiency. "Also, the user of *GearEngineer* software benefits from several new machining technologies and principles, particularly in conjunction with multi-axis machining centers," Weser says. "The software opens up completely new possibilities for the engineering and manufacturing of gears."

The real difference in the software today might just be the attention and focus on everything other than the gears themselves. "Gears cannot be considered on their own; they are part of the gearbox system that includes the bearings, shafts, housing etc.," James says. "If you consider the gear on its own and ignore the system, then the design will be sub-optimal."

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Software Bits 2011

In order to keep current customers—and gain new ones—software developers have to update their products regularly. Whether it's small intermediate updates or a major release once a year, gear designers have decidedly long "wish lists" for what they'd like to get from their software packages. Developers try to oblige with each new update.

Barry James says the current focus of the software package at Romax includes an emphasis on the design for efficiency, design for manufacturing and design for low noise. The company's latest release, *RomaxDesigner 12.8*, contains an improved module on the *System Design for Efficiency*. It contains efficiency prediction methods that have been correlated against test data and a methodology for designing gears for high efficiency, low noise and high durability. "In fact, given sufficient data, it delivers to the gear designer, in real time, an indication of the g (CO_2)/km saving that arises from changes to the gear design," James says.

Dr. Stefan Beermann, CEO at KISSsoft AG, says the launch of the *03/2011* package comes with a variety of new features and functions including a significant improvement of the 3-D models. "For the generation the *Parasolid* core is used. This opened the possibility to provide accurate spiral bevel gear, globoid worm gears and face gears with arbitrary angle between the axes and offset in addition to the other gear types (spur and helical gears, straight bevel gears). The models have sufficient accuracies for manufacturing and measuring purposes. The contact analysis for cylindrical gears was also extended," Beermann says.

03/2011 now takes the misalignment and deformation of the gear flanks due to shaft deformation into account. "The models of the two shafts can be read from files saved in our shaft calculation. For the optimization of the micro geometry a tool is integrated that varies the flank and profile modifications within the parameters defined by the user, performs a contact analysis for the respective variants and evaluates it with respect to transmission error, lubrication film (micropitting), wear and other criteria."

Additionally, the graphical representation now includes 3-D gear bodies with stresses and normal forces marked on the flanks and inputting the tilting and deformation of the rings in bearing calculation. "This is a typical request from the wind business, where the large slew rings on the tower are subject to significant deformations. The same applies to most of the classical gearboxes as well," Beermann says.

GWJ has developed CAD plugins to combine calculation and CAD. The *eAssistant* offers these powerful CAD plugins for different CAD systems (e.g, *SolidWorks*, *Solid Edge*, *Autodesk Inventor*). Other new areas include an emphasis on calculation modules and functions of the *eAssistant* software as well as new types of gears and tooth contact analysis (TCA) in a future *GearEngineer* software release.

Dontyne Systems recently released a products update for its *Gear Production Suite* in April 2011. This update incorporates many of the changes relevant to developments in standards such as micropitting and the IEC/ISO 61400-4- design requirements for wind turbine gearboxes. The company also added new variants to the machine center module for wire erosion and shaving—in addition to the existing hob/grind simulation. At the British Gear Association Technical Awareness Seminar in November of 2010, Dontyne's Mike Fish and David Palmer gave a presentation on "Modelling Production Techniques for Accurate Gears."

Ken Chase

RETIRES FROM BYU MECHANICAL ENGINEERING DEPARTMENT

Longtime AGMA member and AGMA FTM author Dr. Kenneth W. Chase has retired from the Mechanical Engineering Department at Brigham Young University. In his 42-year career, Dr. Chase taught mechanical engineering at BYU since 1968, including machine design, design for manufacture and structural analysis. An advocate of computer technology, he served as a consultant to industry on numerous projects involving engineering software applications. In 1984, he founded the Association for the Development of Computer-Aided Tolerancing Systems (ADCATS). The mission of the association is the development of CAD-based tolerancing software. Members of the association include Allied Signal Aerospace, Boeing, Cummins, FMC, Ford, Hewlett Packard, Hughes, IBM, Motorola, Sandia Labs, Sigmetrix and Texas Instruments.

A 1962 graduate of BYU's ME department, Dr. Chase went on to receive a Master's Degree from BYU in 1964 and a Ph.D. from the University of California Berkeley in 1972. Dr. Chase became an NSF Fellow in 1963 and was a member of Tau Beta Pi and Sigma Xi. He received the Fulton College Research Professorship in 1988 and the Outreach Award in 2002.

Dr. Chase was the faculty advisor for BYU's student chapter of ASME, and took student teams to competitions and conferences all over the world. As a professor, he has influenced thousands of students, both inside and outside of the classroom. Many students have commented on his dedication to helping them, even when it required time beyond Dr. Chase's regular office hours.

Schafer Gear

ANNOUNCES JOINT VENTURE WITH SOMASCHINI

Schafer Gear Works, Inc. and Somaschini S.p.A. of Italy, recently announced a joint venture to produce gears for engines for the heavy-duty truck market in North America. A new manufacturing facility will be located in South Bend under the name of South Bend Gear LLC. The plant will utilize the process technology developed by Somaschini S.p.A. in Italy with Schafer Gear managing the operations. As part of the joint venture, a new 50,000-square-foot plant is being built on the Schafer Gear campus on Nimtz Parkway in South Bend. Production is scheduled to begin in late 2011 at 50 percent capacity and will reach 100 percent production capacity by the end of 2012. The new plant will employ 12 people in phase one and an additional 13 people, for a total of 25 people, when fully operational. The total joint venture investment is $18 million, including the new manufacturing facility. "We are extremely excited about the opportunity this joint venture brings to Schafer and to South Bend," said Bipin Doshi, president of Schafer Gear. "Somaschini's long history of quality gear production and its proven technology complements our world-wide reputation for precision cut gears and precisely machined components."

Mazak

APPOINTS NEW SERVICE MANAGER

Mazak Corporation has appointed Steve Mackay to the position of service manager at Mazak Corporation Canada in Cambridge, Ontario. Mackay has an extensive background in the machine tool industry, with responsibilities at previous positions encompassing service, installation and project management.


Steve Mackay

"Steve brings over 20 years of experience and service expertise that is invaluable," said Ray Buxton, general manager at Mazak Canada and Technology Center. "His considerable knowledge of the industry and experience in machine installation and repair make him a key asset to our team."

Mackay's responsibilities include overseeing machine installations, all warranty and non-warranty machine repair and managing Mazak Canada service personnel and contractors. "Mazak is a true leader in our industry, and I'm very proud to be part of the team," Mackay said. "Everyone wants to be the best at what they do, and working with the best is a good way to make that happen."

Engis

PLANS ILLINOIS EXPANSION PROJECT

Engis Corporation, a leading provider of complete superabrasive finishing solutions for lapping, honing, polishing and grinding, recently announced expansion plans that will nearly double the size of its world headquarters in Wheeling, IL, a Chicago suburb. The company will expand its state-of-the-art

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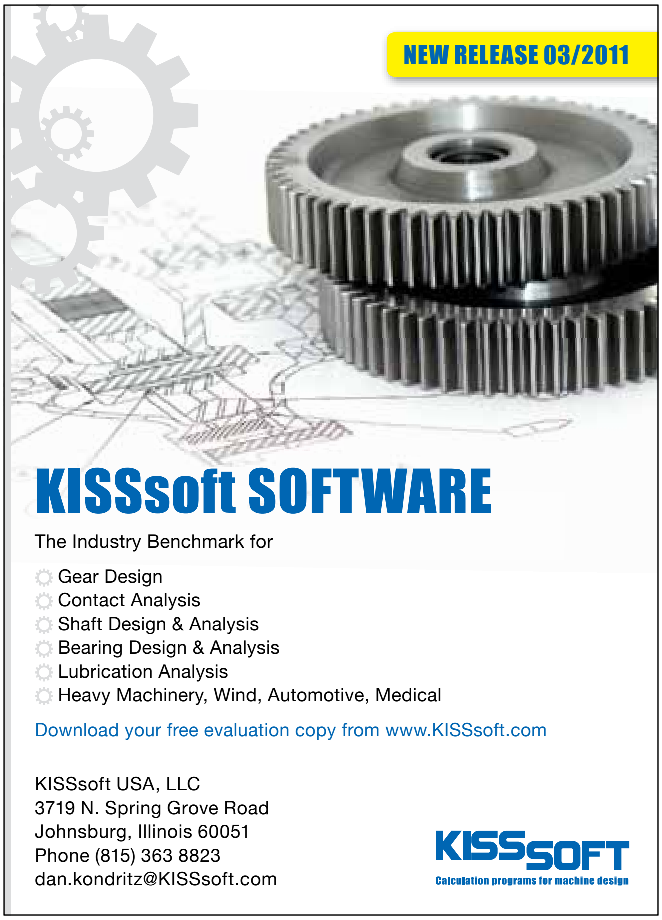

NEWS

manufacturing and warehouse facilities and create space for the new Engis Technology Center. The 54,000-sq.-ft., multi-million-dollar expansion will bring total space to 121,500 sq. ft. and enable the company to consolidate all engineering, process development labs, manufacturing, warehousing and administrative offices into a single facility. In conjunction with this project, Engis will add new manufacturing, process development and testing and measurement equipment. Construction will start in April and be complete by fall of 2011. "This expansion demonstrates our long-term commitment to customers as a leading-edge provider of superabrasive solutions," says Stephen Griffin, president of Engis Corporation. "For example, new automated manufacturing equipment will enable us to increase our ability to support global markets, while the Engis Technology Center will enhance our capability to develop turnkey manufacturing solutions. It also demonstrates our commitment to keep and generate future jobs in Wheeling and in the United States."

ABA-PGT

AWARDED ISO TS 16949 AUTOMOTIVE CERTIFICATION

ABA-PGT Inc., a producer of high-precision tooling and precision-molded plastic gears, has been awarded the ISO TS 16949 Automotive Certification. ABA-PGT currently provides a variety of plastic gear solutions for automotive applications

including dashboard instrumentation, lift gates, windshield wiper motors, door latch and seat adjustor mechanisms for Chrysler, Ford, General Motors, Honda, Nissan, Scania and Volkswagen vehicles. Attaining the ISO TS 16949 certification expands its opportunities in the global automotive industry. ISO is an International Organization for Standardization whose TS 16949 certification aligns American, German, French, Italian and Japanese automotive quality systems standards within the global automotive industry. "ABA-PGT has always provided products of consistently high quality and this ISO TS 16949 automotive certification supports our initiative to become a leader in the automotive plastic gear market," said Terry R. Holmes, vice president, sales and marketing.

Manufacturing Institute

RELEASES ROADMAP FOR EDUCATION REFORM

The Manufacturing Institute (the Institute), the non-profit affiliate of the National Association of Manufacturers (NAM), has released a comprehensive blueprint for education reform designed to develop the 21st century talent critical to U.S. manufacturing and global competitiveness. The Roadmap to Education Reform for Manufacturing lays out six principles

continued

for innovative reform, including moving to competency-based education; establishing and expanding industry-education partnerships; infusing technology in education; creating excitement for manufacturing careers; applying manufacturing principles like "lean" to reduce education costs; and, expanding successful youth development programs.

"These principles can and should be readily applied in current federal and state legislative and budget deliberations," said Emily DeRocco, president, The Manufacturing Institute. "Building an educated and skilled workforce is one of the most significant actions we can take to ensure U.S. leadership in manufacturing."

Research by the Manufacturing Institute has shown that innovation is the greatest driver of success for U.S. manufacturers and that a skilled and educated workforce is the single most critical element of innovation capacity. A skilled workforce is also the hardest asset to acquire; during the height of the last recession, 32 percent of manufacturers cited difficulty finding skilled workers.

The roadmap is a result of the December 1, 2010 National Manufacturing Talent Development Roundtable, hosted by NAM and the Institute, where manufacturing executives, education officials and thought leaders gathered to provide input on a national strategy to reform education in support of U.S. manufacturing. Participants reviewed and applied some of the foremost research and writings on education reform to design an integrated strategy that will enable the education system to develop a smart, safe, and sustainable 21st century manufacturing workforce.

"Manufacturers from across the country and in all sectors have engaged their energy, time, and resources to lead efforts in their communities and states to ensure a highly skilled and educated workforce," said DeRocco. "Manufacturers look to address deficits in the education system the same way they look to improve and expedite their supply chain. We have partnered with the disruptive innovators in education to develop strategies to address each critical choke point along the education continuum, ultimately to develop and advance the new workforce that will keep us competitive in the complex global economy."

AMB 2012

WANTS TO BUILD ON 2010 TRADE SHOW

More visitors, an increasingly international character, a top-quality accompanying program: The International Exhibition for Metal Working (AMB 2012) has set its sights high. When it opens (September 18-22, 2012 in Stuttgart), it wants to surpass the great success of 2010 where more than 1,300 exhibitors and 86,000-plus visitors came to the exhibition for metal working. “Registration has started and we are receiving requests for placements every day,” explains Sengül Altuntas, project manager for machine tools at AMB. The response to AMB 2010 was overwhelming, with regard to both exhibitors and visitors. There was talk of the best AMB of all time. “AMB has finally developed to become the leading European trade fair for the machine tool and precision tool industry in even years,” states Gunnar Mey, project manager for precision tools and peripherals.

The international character of the exhibitors was shown by the fact that 20 to 25 percent came from abroad, while the proportion of foreign visitors increased from seven to eleven percent. They came from 80 countries, a new record and an indication that AMB is also gaining in importance internationally. The visitors from Germany and abroad were primarily from the mechanical engineering sector, followed by the metal working and processing industry, automobile industry/vehicle construction, tool construction and mold making, as well as metal construction enterprises. At 38 percent, the group of visitors working in manufacture and production was by far the largest. But members of company management, developers, designers, planners and planning engineers were equally represented. They all found solutions at AMB for their manufacturing problems, and were able to gain a comprehensive overview of the status quo in metal-cutting processes and the associated precision tools. The main focus was on milling machines, lathes and automatic lathes, machining tools, chucking tools, CAD, CAM, CAE, software, grinding machines, workpiece and tool handling.

Fully booked halls are again expected at AMB 2012, with more than 1,300 exhibitors on an exhibition area of 105,000 square meters. Top experts from the promotional supporter associations for AMB—the Precision Tools Association in the VDMA (German Engineering Federation), the VDMA Software Association and the VDW (German Machine Tool Builders’ Association)—will ensure technical input. Furthermore, an immeasurable advantage for exhibitors and visitors is the location of Messe Stuttgart directly beside Stuttgart Airport and the A8 motorway, the main route between Munich and Frankfurt. For more information, visit *www.messe-stuttgart.de.*

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

May 11–13—Solid-State Power Supply Technical Seminar. Madison Heights, Michigan. This class is scheduled to examine Inductoheat's Statipower SP16 Power Supply. The principle objective of this seminar is to have 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. For more information, visit *www.inductoheat.com*.

May 11–13—FABTECH Mexico. Cintermex, Monterrey, Mexico. Co-sponsored by the Fabricators and Manufacturers Association, Int'l (FMA) and the Society of Manufacturing Engineers (SME), FABTECH Mexico will co-locate with the established AWS Weldmex and METALFORM Mexico trade shows. The event is anticipated to bring more than 300 exhibitors, 50,000 square feet of floor space and more than 8,000 manufacturing professionals from Mexico and Central America. Anyone involved in the evaluating and purchasing of new equipment, products or services in coil processing, robotics, laser and plasma cutting, saws and cut-off machines, plate and structural fabricating, shears, press brakes, tooling, punching, tube and pipe equipment and roll forming should consider attending. For more information, visit *www.fabtechmexico.com*

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

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

June 5–7—Manufacturing Velocity: SME Annual Conference. Bellevue, Washington. The SME Annual Conference provides insights and connections to accelerate you and your company by discovering the year's biggest trends and developments from nearly every technology area of manufacturing. Dive into the innovations that are changing the way—and the products—people manufacture. Find and discover solutions for today's critical manufacturing challenges. This event condenses just about everything SME has for manufacturing into an intensive, fast-paced, high-value couple of days. Your paid conference registration includes a one-year membership to SME. If you're already a member, your paid registration includes a one-year renewal at your next renewal date. Learn more about SME membership at *www.sme.org/join*.

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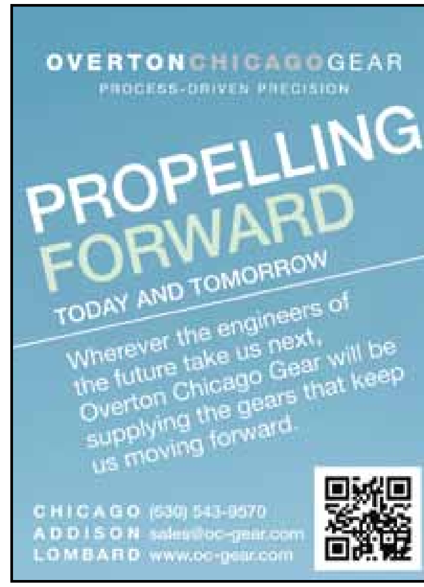
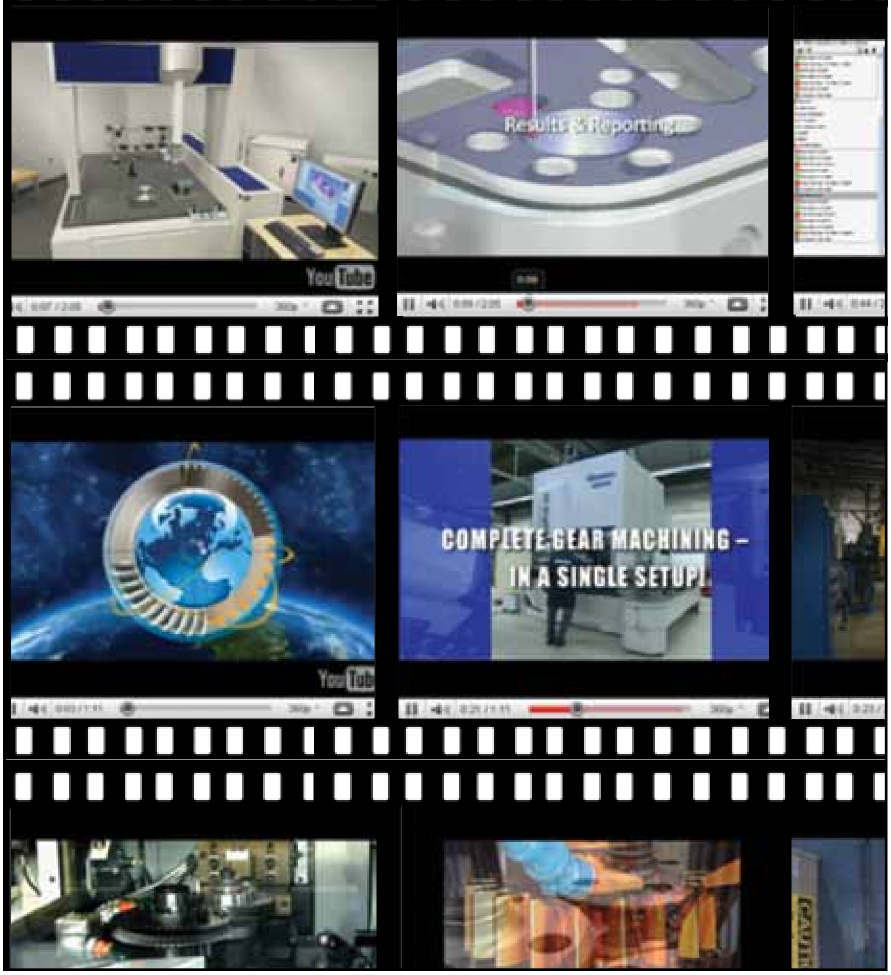
GEAR MANUFACTURING





Results & Reporting
COMPLETE GEAR MACHINING – IN A SINGLE SETUP!

Monty Cleeves, CEO
Pinnacle Engines

Gearheads Rejoice! The Internal Combustion Engine is Back

More specifically—an opposed-piston, internal combustion engine

While all the transportation/energy innovation-related news of late has been dominated by the ongoing development of the electric engine, an old-wine-into-new-bottles alternative is making some unheralded headway into the conversation.

Please join the Addendum crew in saying "welcome back" to a century-old technology and power source that had been languishing for many years—the internal combustion engine. More specifically—an opposed-piston internal combustion engine.

That's the news gleaned from an April *New York Times* article by Todd Woody in the newspaper's energy special section. Cited in the article were three entrepreneurial-driven companies—Pinnacle Engines in Silicon Valley, Detroit-area-based Eco Motors and San Diego's Achates Power. All are intent upon developing and marketing a redesigned old-style engine into one that they believe will provide significant upgrades in fuel economy and reduced greenhouse gas emissions—at lower cost.

"While the buzz is all about electrics, the people who will actually adopt electrics are not a majority of the market," says Pinnacle president and CEO Monty Cleeves, who founded the company in 2007 and has intentionally run the operation in an under-the-radar mode. "The impact we will have over the next 15 to 20 years will be much larger than the impact of the electrics."

The *Times* article points out that while GM and Ford, for example, always have and probably always will design and manufacture their own engines, the global demand for fuel efficiency—especially for commercial vehicles—along with "climate change concerns and the rise of China and India as automotive markets, have opened the door to start-ups like Pinnacle."

"Many automotive houses don't buy engines from outside, but in the truck market people do," says Rohini Chakravarthy, a partner at NEA, a venture capital firm in Menlo Park, California., that has invested in Pinnacle. "In Asia, there's tremendous demand, and you're not going up against the same level of incumbents."

In fact, Pinnacle has already signed a deal to license their engine technology to an undisclosed overseas scooter manufacturer for production in 2013. Eco Motors—with backing from Khosla Ventures and Bill Gates—has signed a development agreement with Navistar and a Chinese company—also unnamed—and Achates Power is currently pitching automakers, says David Johnson, Achates CEO, who added that he had also met with potential customers in China and India.

The design magic of the opposed-piston engine is that it eliminates the cylinder head—i.e., the combustion chamber for a conventional engine. Rather, two facing pistons and the space between them form the combustion chamber where fuel is ignited. The average weight of a car engine is 600 lbs.—and much heavier, of course, in commercial vehicles—with much of that weight attributed to the cylinder head. Discarding that component enables lower-cost, lighter engines and an overall lighter vehicle—either for personal or commercial operation. As with virtually any power-supplying component in use today, much of an engine's energy usage and loss are generated by heat. The nimbler opposed-piston design, however, has a more robust energy source to operate a vehicle.

However, making the opposed-piston engine vehicle-friendly has been the greatest challenge thus far, something that Cleeves has been grappling with since his days as a mechanical engineering student.

"I stayed 30 years in (the) semiconductors (industry) as my day job, but worked on cars in the garage as my passion in the evenings," says Cleeves.

That passion is now represented by an in-testing prototype—an iteration of a one-cylinder, four-stroke opposed-piston gasoline engine—designed, the *Times* article states, "to power scooters and three-wheel auto rickshaws in Asia."

Pinnacle executives claim that 500 hours of "independent" testing verified that the engine "was 30 percent more efficient than current scooter engines, while emitting fewer pollutants." Further, the company says that an automotive version of its "Cleeves Cycle Engine" would "increase the fuel economy of a Fiat 500 from 33 gallons to 59 miles a gallon—with "no performance loss."

For its diesel version, Eco Motors is touting an "up to a 50 percent improvement in efficiency for its two-stroke, diesel-opposed piston engine now in development for heavy-duty vehicles and equipment." (Today's conventional two-stroke engines emit more greenhouse gases and are commonly found in scooters, lawnmowers and outboard boat engines.)

The key, says Donald Runkle, EcoMotors' chief executive and a General Motors émigré, is a high-power output and an electrically controlled turbocharger that allows for a small, lightweight engine with emissions comparable to a four-stroke power plant.

At Achates Power, CEO David Johnson says his company's version of the two-stroke, opposed-piston diesel engine "would most likely be used in commercial vehicles, but it also could be installed in plug-in, electric-hybrid cars like the Chevrolet Volt."

"The Volt needs a better engine," says Johnson, stating that "1,600 hours of testing had shown Achates's engine was 15 percent more efficient than conventional diesel counterparts."

And Ron Hoge, a Pinnacle chief executive, says even more dramatic change is needed.

"It's the challenge of incremental thinking versus radical thinking," says Hoge, previously at engine maker Cummins. "If we're only going to make incremental improvements, we're not going to solve our problems in the world, so someone has to step forward."