GEAR TECHNOLOGY

SEPTEMBER/OCTOBER 2005

The Journal of Gear Manufacturing

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GEAR EXPO

- Gear Expo Product Showcase
- Map and Booth Listings

FEATURES

- Power Losses in Helical Gears
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SEPTEMBER/OCTOBER 2005 — *The Journal of Gear Manufacturing*

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GEAR TECHNOLOGY
The Journal of Gear Manufacturing

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BPA WORLDWIDE

VOL. 22, NO. 5

GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published bimonthly by Randall Publishing, Inc., 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL 60007, (847) 437-6604. Cover price $5.00 U.S. Periodical postage paid at Arlington Heights, IL, and at additional mailing office (USPS No. 749-290). Randall Publishing makes every effort to ensure that the processes described in GEAR TECHNOLOGY conform to sound engineering practice. Neither the authors nor the publisher can be held responsible for injuries sustained while following the procedures described. Postmaster: Send address changes to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL, 60007.

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Making It in America

U.S. Manufacturing is Alive, Well and Prospering

Most firms in the gear industry we've talked to over the past year are making more gears than ever, generating more sales, and filling up their schedule books into next year and beyond.

In fact, it's hard to talk with a gear manufacturer these days who isn't telling you about his new machine tools, robots or other new technology, as most are extremely proud of the investments they've made in their factories—expanded facilities, the newest machine tools, manufacturing cells, lean processes and automation.

I recently came across a newspaper article that confirmed many of the observations we at *Gear Technology* have made over the past year. The article, "If You Can Make It Here...," appeared in the Sunday, September 4 edition of *The New York Times*. It details how despite huge drops in the manufacturing employment rate, the United States surprisingly still accounts for nearly a quarter of the world's manufacturing output, measured as value added, a statistic that hasn't changed significantly since 1982.

The article also details the stories of some American manufacturers that have prospered with a focus on making goods at home. The first is Harley-Davidson Inc. Although Harley-Davidson buys some components from overseas—those no longer made domestically—the company still makes most of its parts, including all of the important ones, like engines and transmissions, right here in the United States. In fact, the domestic content of Harley-Davidson's motorcycles is higher today than it was 15 years ago.

The article attributes much of Harley-Davidson's success to creative arrangements with labor unions that have helped the company control costs. But a big part of the success is also due to increases in productivity through better technology.

The second company featured was Haas Automation. Now here's an American machine tool company that sells, builds and installs an astonishing 900 machine tools per *month* out of its factory in Oxnard, CA. While most of the world's machine tools are manufactured outside America, Haas is thriving by producing machine tools not only in the United States, but in California no less, a state not often mentioned as friendly to business or inexpensive.

According to the article, the Haas factory is highly automated. Machine tools run unattended overnight, and labor amounts to just 10% of production costs.

While the article points out some high-profile examples, there are plenty of American gear manufacturers also doing well.

Take the example of Delta Research of Livonia, MI, which we wrote about in the July/August issue. Not too long ago, Delta Research manufactured transmission prototypes. Now they've installed the latest machine tools in a fully automated gear manufacturing cell to produce transmission gears, with no human hand touching them from hobbing through grinding, roll testing, deburring, rustproofing and washing.

Also, we recently talked with Mark Garfien, president of U.S. Gear in Chicago. For him, the future of gear manufacturing is summed up in one word: automation.

Last issue, in an article by Tom Lang of Kapp Technologies, he observed the growing demand for automation in smaller, more agile shops. Automation clearly isn't just for the big companies anymore.

Over the past year, we've covered a number of advances in machine tools that help manufacturers get parts out faster. Machine tool manufacturers are eliminating wasted cycle time by adding spindles, incorporating multiple functions into single machines and integrating automation into the machine tools as standard packages. All this is on top of advances in technology, like dry hobbing, brought about by better machines, tool materials and coatings, all of which allow for production levels never seen before.

The companies incorporating these technologies are the ones doing well today. America's successful manufacturers aren't worried about Chinese companies taking away their business. Instead, they're wondering how they can take advantage of one of the world's fastest growing markets. According to the newspaper article, both Haas and Harley-Davidson are exporting to China, and both companies look at that market as one of huge potential growth.

The key to being successful at manufacturing in America is increasing your productivity.

If you'd like to know how it can be done—how to make more gears, better and faster—you'll never have a better opportunity than by visiting Gear Expo in Detroit, October 16–19. There, you can see all of the latest technology. You can also visit the booths of gear manufacturers who have made the investments described above. Gear Expo provides you the chance to go and talk to the experts in machine tools, cutting tools, heat treating, materials and coatings, the people who know and supply the technology that can help your company survive, compete and even prosper—no matter where you manufacture.

Michael Goldstein, Publisher & Editor-in-Chief

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Title: **Evaluation of the Scuffing Resistance of Isotropic Superfinished Precision Gears**
AGMA FTM Session III, October 18 - *Morning Session* • 8:00-12:00
Primary author: Lane Winkelmann , Co-authors: Paul Niskanen, Bruce Hansen

Title: **Repair of Helicopter Gears**
AGMA FTM Session III, October 18 - *Afternoon Session* • 1:00-5:00
Primary author: Dr. Suran Rao,
Co-authors: Dr. Gary Sroka, Doug McPherson

BOOTH 563

Corus' New Gear Steels Reduce Alloys Without Sacrificing Achievable Hardness

Corus Engineering Steels' formula for its new gear steels: Maintain achievable hardness while using fewer alloys, thereby cutting steel costs for gear manufacturers.

Located in Rotherham, England, Corus Engineering Steels (CES) announced the new gear steels in July 2004.

"Lean alloy steels have been developed as a direct response to rising alloy prices" says James Hunt, market & product development manager for CES. "Lean alloys are by definition less costly than the higher alloy steels they replace."

As an example, Hunt says a leaner alloy steel, like an MnCr, could be as much as £182 ($327) per ton cheaper than a higher alloy steel, like an NiCrMo. The savings comes from differences in the base prices and alloy surcharges of the two steels.

"The biggest difference comes when we take into account the alloy surcharges," he says. "The prices for molybdenum and nickel are extremely unstable and very much higher than chromium."

CES develops its gear steels mainly for the automotive industry, supplying them to BMW, Eaton, Ford, General Motors, Scania, Volvo and ZF. CES' new gear steels are, however, suitable for other industries, too.

"Examples might include gearboxes and geared motors for food processing, mechanical handling or marine applications," Hunt says.

"However, it must be recognized that there are limitations to the depth of hardness and core strength which can be achieved by a low alloy steel, hence larger diameter gears may not be suitable," he adds. "Each application should be considered on its specific requirements."

Despite these limitations, CES designed its new gear steels to behave like their higher alloy counterparts, so gear manufacturers who switch to leaner alloys don't sacrifice performance, such as the ability to achieve a particular hardness range.

"Corus steels are typically supplied to give hardenability response of +/– 4 HRc, with tighter ranges available by agreement," Hunt says. "This applies to all our steels, not just lean alloy grades."

Also, Corus' steelmaking helps promote predictable distortion.

"The precise control we are able to get over our steelmaking processes with full computer control, gives exceptional cast-to-cast consistency of chemical composition and hardenability," Hunt says. "Hence the predictability of all our steels ensures that the manufacturers need to make fewer adjustments to their processes from batch to batch."

Hunt, however, cautions that predictable distortion is a potentiality that depends on other processes: "The amount of distortion achieved has more to do with the gear manufacturing route than it does the composition of the steel."

Still, if the route provides reliable, repeatable results, gear manufacturers could ultimately save time and money through better control of distortion because they wouldn't have to machine their gears as much after heat treatment.

Hunt adds that the new gear steels are very clean, which promotes increased fatigue resistance and thereby increased durability.

"Fatigue and durability are influenced by surface hardness and steel cleanness," Hunt says. "The cleanness of all our steels ensures that harmful oxides and other inclusions are kept to a minimum, and it is this property that gives improvements in fatigue and durability."

"If you were to compare the fatigue performance of one of our lean alloy steels against a higher alloy steel of equivalent strength, also supplied by Corus," Hunt adds, "chances are there would be little difference as the cleanness will be the same."

He cites its MnCr steel as an example. That steel has a hardenability comparable with Corus' 1%NiCrMo steel. The MnCr was designed to replace the NiCrMo. CES also developed its lean alloy gear steels to help improve the noise, vibration and harshness (NVH) characteristics of resulting gearboxes. Hunt couldn't discuss details of specific tests on the NVH characteristics of gears made with CES' new steels. Even if he provided details, NVH characteristics vary as a result of many aspects of gear manufacturing. "Particular benefits seen in NVH will depend upon the consistency of the steels being substituted and the level of control achieved over the manufacturing process."

Despite depending on manufacturing processes, the new gear steels are Corus' effort to provide the same for less.

"Rising alloy addition surcharges and the continual focus on best value led to the development of lean alloys," Hunt says. "Expert product development ensured that reduced cost did not come at the expense of performance."

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E-mail: *enquiries.ces@corusgroup.com*
Internet: *www.corusengineeringsteels.com*

New Speed Drive for Wind Turbines from Voith

The WinDrive from Voith Turbo can convert a spectrum of input speeds into a constant output speed in the multi-megawatt range.

According to the company's press release, key components include a hydrodynamic torque converter and planetary gear designed as a superimposing gear. With this drive, low and variable rotor speeds are converted into high, constant speeds. Due to constant output speeds, synchronous generators can feed the produced electricity directly into the grid.

This eliminates the need for power electronics for frequency adaption. According to Voith, this drive concept can improve the weight and space requirements in the nacelles of wind turbines.

In addition, the WinDrive has an integrated torque converter because input and output are connected due to hydrodynamic transmission of power.

For more information:
Voith Turbo GmbH & Co. KG
Phone: +(49) 7951-32-683
Internet: *www.voithturbo.com*

Dow Corning Synthetic Oils to Improve Gearbox Life

Dow Corning Co. has introduced six new full synthetic Molykote® extreme pressure gear oils to provide improved wear protection plus greater thermal and oxidative stability when compared to conventional oils.

According to the company's press release, oils are designed to overcome severe operating conditions, such as temperature extremes and shock load

or vibration. The Molykote L-21XX Series PAO gear oils combine protection against scuffing and micropitting with water separation and corrosion protection. Additionally, the gear oils are formulated with carefully selected additives to inhibit rust and oxidation and suppress foam.

Available in ISO viscosities of 100 (L-2110), 150 (L-2115), 220 (L-2122), 320 (L-2132), 460 (L-2146) and 680 (L-2168), the oils meet or exceed the following standards: Flender, DIN 51 517 Part 3, ANSI/AGMA 9005, U.S. Steel 224, Cincinnati Milacron and David Brown SL.53.101.

For more information:
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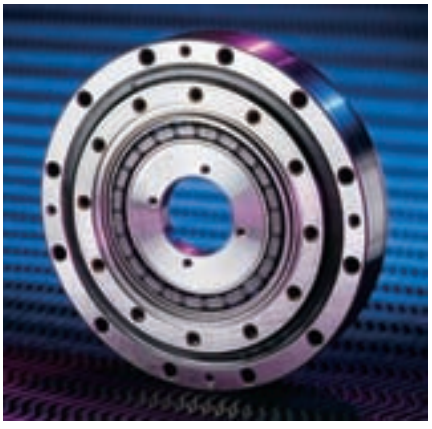
Oklahoma City, OK 73162
E-mail: ***industrial@dowcorning.com***
Internet: ***www.dowcorning.com***



New Ultra Flat Gearhead from HD Systems

The SHD Size 14 gearhead from HD Systems delivers zero backlash, 1.5 arc-min. positional accuracy, and +/– 5 arc seconds repeatability.

According to the company's press release, the gearhead combines high precision harmonic drive gearing and a high capacity cross roller bearing. The Size 14 gearhead has an outer diameter of 70 mm and is 17 mm in length. Rated torque is 33 in.-lbs. and comes in gear ratios of 50:1 and 100:1.

Suitable for OEM applications, the design allows the gearhead to be incorporated into the machine housing by providing the necessary motion control components, such as the harmonic drive gear and cross roller bearing. Support of a large output torque and moment load are possible.

For more information:
HD Systems
89 Cabot Ct.
Hauppauge, NY 11788
Phone: (800) 231-HDSI
E-mail: ***info@HDSI.net***
Internet: www.hdsi.net

Self-Calibrating Speed Sensor Detects Gear-Edge Motion

The ATS616LSG speed sensor from Allegro MicroSystems is designed as a solution for non-TPOS camshaft applications, providing improved electrostatic discharge and electromagnetic compatibility and eliminating the need for an external switching capacitor. The smaller gear tooth sensor package is integrated to maintain airgap range and high timing.

This sensor is a peak-detecting device using automatic gain control and an integrated capacitor to provide accurate gear-edge detection down to low operating speeds. Each sensor subassembly consists of a high temperature plastic shell that holds together a samarium-cobalt magnet, a pole piece and differential

open collector Hall IC that has been optimized to the magnetic circuit. The package can be used in conjunction with various gear shapes and sizes, according to the company's press release.

Gear sensing technology used for the sensor subassembly is Hall-effect-based. In addition, the sensor incorporates a dual element Hall IC that switches in response to differential magnetic signals created by ferrous targets. Processing circuitry contains an A/D converter that self-calibrates the device's internal gain to minimize the effect of airgap variations. A patented peak-detecting filter circuit eliminates magnet and system offsets and can discriminate relatively fast changes, such as those caused by tilt, gear wobble and eccentricities, yet provides stable operation to low rpm. The sensor is suitable for use in gathering speed, position and timing information using gear-tooth-based configurations.

For more information:
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Worcester, MA 01606
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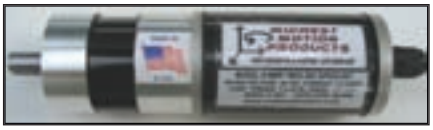
Internet: *www.allegromicro.com*



New Gearmotor from Midwest Motion

The new MMP-TM55-36V-007 DC gearmotor from Midwest Motion Products can accept any 36-volt DC source, including batteries.

According to the company's press release, this gearmotor measures 2.14" in diameter and 6.75" long and has a keyed output shaft of 12 mm in diameter by 25 mm long. Easy mounting is possible with four "face mount" M5 threaded holes, equally spaced on a 40 mm diameter bolt circle.

The motor is rated at 0.85 Nm continuous torque at 700 rpm and has a 40 Nm peak. Despite its size and weight, the motor requires 2.6 amperes at 36 volts DC to generate its full-load output torque.

Typical options include integral optical encoders, failsafe brakes, analog tachometers and planetary gearheads with ratios from 3:1–450:1 with standard or low backlash precision gearing.

For more information:
Midwest Motion Products
Phone: (952) 955-2626
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New Hob Sharpener from Star Cutter

The new PTG-1 from Star Cutter Co. is a hob sharpener featuring NUM axial CNC control with digital servo motors, NUMROTO plus software, integrated automatic part probing, grinding wheel probing, and on-board wheel dressing.

According to the company's press release, the PTG sharpens straight and spiral gash hobs, through-hole and shank type, up to 8" in diameter and 10" in length. The single-end, direct-drive grinding spindle has a 15 hp rating with maximum grinding wheel capabilities up to 8".

Additionally, the hob sharpener is equipped with a quick-change HSK wheel pack mounting, a planetary roller head stock with a glass scale feedback and glass scales on all linear axes. The PTG-1 can also perform quick-change HSK headstock tooling for hobs, including through-hole and shank type; shaper cutters; end-milling cutters; and drills. Coolant filtration, a delivery system and mist collection are featured as well.

Workpiece specifications for straight gash and spiral hobs with automatic part probing reach 8" by 10". Dish and helical shaper cutter specifications are 8" in diameter.

Visit Star Cutter at Gear Expo booth #370.

For more information:
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Marposs Unveils New Retoolable Manual Bench for Gear Inspection

Marposs Corp. will introduce its new M62 OBD (over ball diameter) retoolable manual bench measuring system for checking gear shafts at Gear Expo 2005.

According to the company's press release, the M62 OBD bench is designed to permit quick changeover to measure gears in a shop floor environment. With this bench, the part floats freely on a mechanical reference surface and can be checked in different longitudinal sections by means of simple mechanical adjustment.

In addition, the company will show its M62 DF double flank measuring system, which is designed for dynamic inspection of gears with automatic part rotation.

Gary Sicheneder, new market development manager at Marposs, says, "The M62 DF gage has one basic advantage, and that is the existence of an electronic bore gage that definitely does a better job in identifying the exact center of the gear bore. This is standard technology for Marposs. Most competitive DF [double flank] gages are supplied with expanding mandrels/arbors."

The company has added gimbal mechanics to evaluate the typical double-flank functional parameters (Fi", fi", Fr" Aa", Nick), lead angle, lead variation and taper.

Gear inspection systems will be displayed with different electronic units, including the E9066 industrial PC and an E4N electronic column, featuring the company's Quick SPC program with gear add-on software.

Also new is the M1 Star EBG electronic bore gage, with features like a 3–300 mm diameter measuring range, a frictionless and wear-free integrated transducer and IP67 protection against water and coolant.

"With a .5 micron guaranteed repeatability, we are confident this will be the best electronic bore gage available on the market," says Sicheneder.

Other Gear Expo products include the M1 Star MBG (mechanical bore gage) that was launched in July '04, a line of Go/No-Go gages for gear checking, the Quick Set modular bench gage for shaft type parts, new DigiCrown digital pencil probes and other manual gages and Mida spindle touch probes and laser tool setters.

Visit Marposs at Gear Expo booth #543.

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New Gear Oils from Shell Lubricants

The Shell Omala, Omala HD and Tivela S gear oils from Shell Lubricants that will be introduced at Gear Expo are specially designed to prolong gear life and reduce unscheduled downtime.

The Shell Omala is a mineral-based gear oil suitable for highly loaded gearboxes operating under normal temperatures. Additional features include improved water separation properties that are critical in wet environments such as mining and steel and paper mills.

The Omala HD is a high performance synthetic oil based on poly-alpha-olefin (PAO) and other specially selected additives. Improved thermal and oxidative stability minimize sludge formation, and increased viscosity helps increase oil life and reduce maintenance costs, while its anti-wear performance offers longer gearbox life in steel-on-steel applications. This oil aims to provide low friction properties, reduce power losses and lower operating temperatures.

The Shell Tivela S is a high performance synthetic gear fluid based on polyalkelene-glycol (PAG) technology and specially selected additives designed for use in enclosed, sealed-for-life and worm-helical gear units operating at higher temperatures. In addition, the gear oil has a low friction coefficient, which leads to reduced power losses and lower energy consumption when compared to mineral oils. This thermal and oxidative stability inhibits sludge formation and viscosity variations for longer service life and reduced maintenance requirements.

Visit Shell Lubricants at Gear Expo booth #703.

For more information:
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Fax: (713) 546-6675
Internet: *www.shellus.com*

Arrow Gear Introduces Curvic Couplings

Arrow Gear has introduced a line of curvic couplings for aircraft engines, helicopters and high precision commercial systems.

According to the company's press release, Arrow previously focused on producing high precision gearing products, including spiral bevels, spurs, helicals and complete gearboxes.

"Arrow has had the capability of cutting curvic couplings for the past several years," says Joe Arvin, president of Arrow Gear. "However, recent additions of specialized software and machine tools have allowed us to perform precision grinding on these products. We feel the investment of over a million dollars will translate into a significant volume of curvic coupling work for both new and existing customers," he says.

Arrow has a Gleason Phoenix 600 on order and scheduled for delivery in the first quarter of 2006. The new Phoenix was ordered with capabilities for curvic couplings as the main requirement and will also be used for production of ground tooth spiral bevel gears.

James J. Cervinka, co-founder of Arrow Gear and chairman of the board, says this commitment to reinvestment in the company is a long-standing Arrow Gear philosophy: "This is another example of our dedication to servicing our customers and investing in the latest manufacturing technologies, which keeps us at the forefront of the competitive market."

Visit Arrow Gear at Gear Expo booth #572.

For more information:
Arrow Gear Co.
2301 Curtiss St.
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Internet: *www.arrowgear.com*

New Single Stroke Honing Tooling from Sunnen

The new High Performance Single Stroke Honing (HPH) Tools from Sunnen Products Co. offer fast stock removal, consistent bore geometry and can economically optimize high production applications.

The plated-diamond HPH tools are manufactured with high strength steel as opposed to the typical cast iron. In addition, all tools incorporate an expandable, superabrasive sleeve mounted on a tapered mandrel for precision bore sizing and finishing. The tools are designed so that after the final pass, the bore is accurately sized, straight and round with the desired surface texture. Water- or oil-based coolants are both acceptable for use.

The honing tools perform best on bore diameters ranging from 6.325–25.057 mm. They are compatible with Sunnen's VSS Single Stroke System as well as models from other manufacturers.

The SV-10 Automatic Cylinder Hone will also be displayed at the company's booth. This machine has a patented, full-bore profile display illustrating a real-time visualization of the bore's geometry and includes a zoom feature. Operators can dwell the honing tool anywhere throughout the bore. A dwell taper can be moved to the position where the bore is visually

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ITW will also show its standard 3500 gear analyzer for process inspection applications at the expo.

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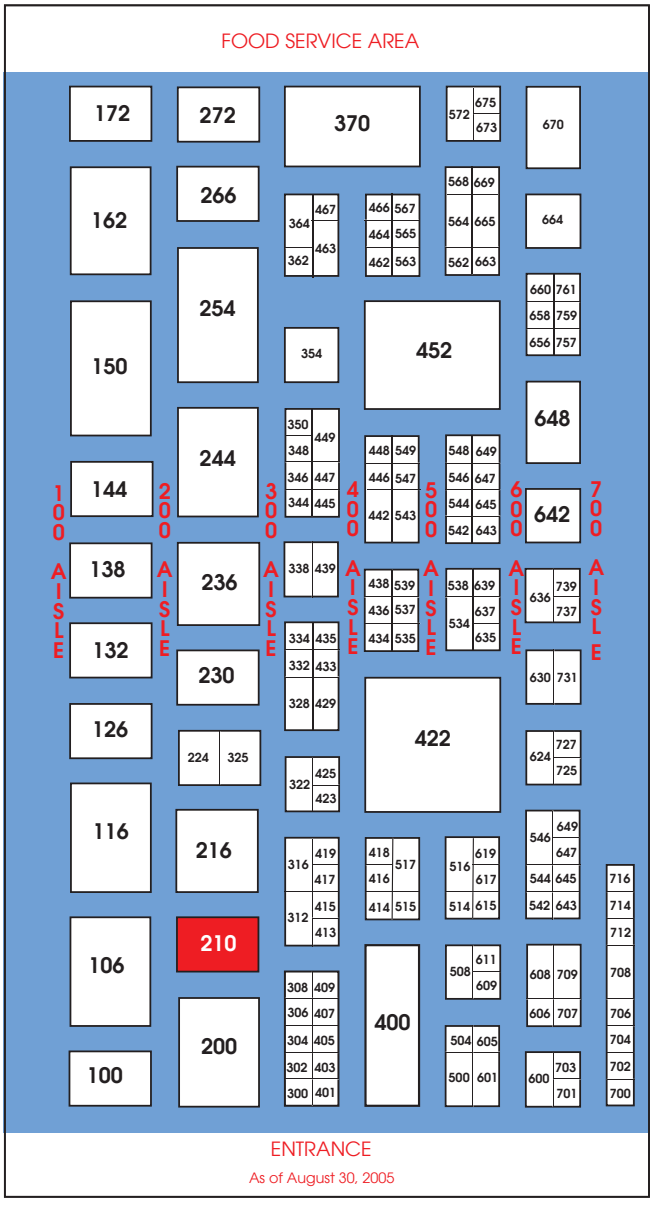

ENTRANCE

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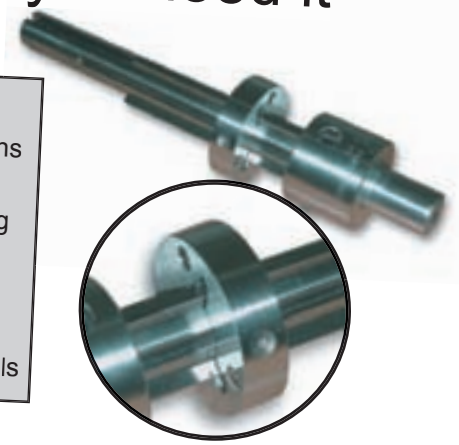

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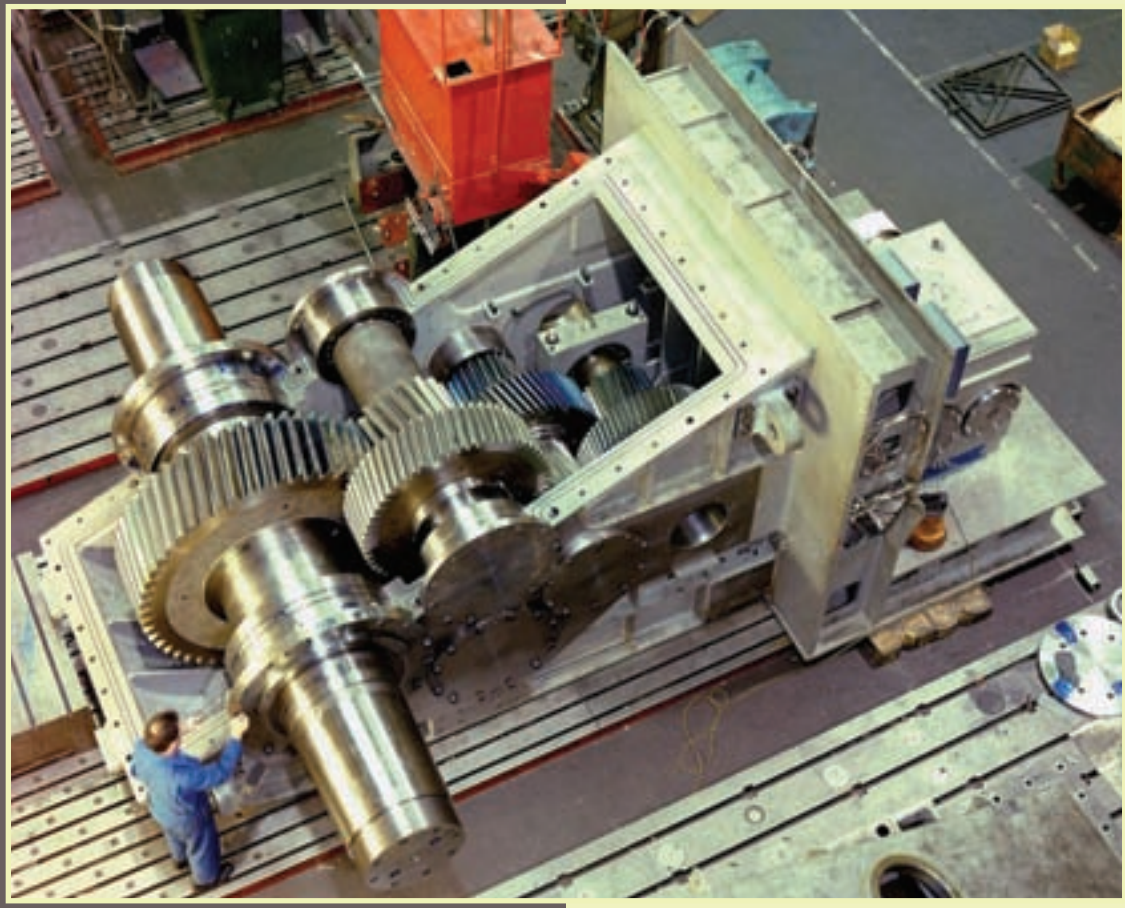

determining Power Losses in the helical gear mesh

Petra Heingartner *is a Textron Six Sigma black belt for fluid and power at Textron Fluid & Power, located in Huddersfield, England. She leads cross-functional teams to improve work processes throughout the business. Previously, she was a senior design engineer and a senior applications engineer of Textron's custom-made heavy industrial gear units.*

Dr. David Mba *is senior lecturer and postgraduate course director in the automotive, mechanical and structures engineering department at Cranfield University, located in Cranfield, England. He is director of the master's course "Design of Rotating Machines" and is responsible for its industrial collaborations and is director of the continued-professional-development course "Gear Design." He also leads the department's machine diagnostics group, which is involved with consulting and doctoral research. Mba specializes in machine diagnostics, rotor dynamics and machine design. Also, he's developing a research and experimental facility to predict nonload gear losses in worm gearboxes.*

A Case Study

Petra Heingartner and David Mba

Management Summary

Currently, legislation is in place in the United Kingdom to encourage a reduction in energy usage. As such, there is an increased demand for machinery with higher efficiencies, not only to reduce the operational costs of the machinery, but also to cut capital expenditure. The power losses associated with the gear mesh can be divided into speed- and load-dependent losses. This article reviews some of the mathematical models proposed for the individual components associated with these losses, such as windage, churning, sliding and rolling friction losses. A mathematical model is proposed to predict the power losses on helical gears highlighting the major contributor to losses in the gear mesh. Furthermore, the mathematical model is validated with a case study.

Introduction

To meet the increased demand for machinery with higher efficiencies, suppliers must design equipment that reduces the operational costs of the machinery and cuts capital expenditure.

In the past, gears have been considered as highly efficient in transmitting loads, but the requirements from the customer to achieve a minimum efficiency target and penalties for noncompliance are becoming more and more stringent. A reduction in the power loss of a gearbox will cut the running costs of the equipment as it becomes more efficient and also uses less lubricant to cool the gear teeth. This in turn will reduce the size of the auxiliary equipment, such as the lubrication pump, and will also lead to a reduction in the heat exchanger capacity. All this will contribute to an overall smaller footprint of the equipment, which saves space that can be at a premium in some applications.

The power losses consist of speed- and load-dependent losses. Speed-dependent losses can be divided into windage losses, churning losses, bearing churning losses and seal losses. The load-dependent losses are made up of sliding friction loss, rolling friction loss and bearing loss.

Speed-Dependent Losses

Windage losses. As gears rotate, lubricant is flung off the gear teeth in small oil droplets due to the centrifugal force acting on the lubricant. These lubricant droplets create a fine mist of oil that is suspended inside the gear housing/case. The effect of this oil mist is an increase in "windage frictional resistance" on the gears and hence an increase in the power consumption. In addition, the expulsion of the oily atmosphere from the tooth spaces as the gear teeth come into engagement creates turbulence within the gearbox and increases the power consumption. The combination of these factors, as well as the losses at the side faces of the gears, contribute to the total windage losses. Factors that influence the magnitude of the

Nomenclature	
A_g	Arrangement constant $A_g = 0.2$
C_1	Constant, 29.66
C_2	Constant, $9 \cdot 10^7$
D	Element diameter
F_R	Rolling force
F_S	Sliding force
L_c	Contact line length
P_{CL}	Churning power loss
P_R	Rolling power loss
P_S	Sliding power loss
P_{WL}	Windage power loss
R_f	Roughness factor
V_S	Sliding velocity
V_T	Rolling velocity
b	Face width
d_f	Root diameter
d_k	Outside diameter
f_g	Gear dip factor (Ratio of dipping depth to element outer diameter) $f_g = 1$ element fully submerged
h	Isothermal central film thickness
m_n	Normal module
n	Rotational speed
w	Load parameter
w_n	Normal gear contact load
β	Helix angle
λ	Gearbox space function
μ	Coefficient of friction
μ_0	Ambient viscosity at ambient temperature
ν	Kinematic viscosity at operating temperature
ϕ	Oil mixture function, $\phi = 1$ oil-free atmosphere
ϕ_t	Thermal reduction factor
Subscripts	
i	Element under consideration


windage loss include the rotational speed of the gear because power losses rise with an increase in peripheral velocity. Other factors are the tooth module, the amount of oil mist present inside the casing and the diameter of the gears.

A mathematical model to predict windage loss was proposed by Anderson et al. (Refs. 1–2). However, Anderson's windage loss equation accounted for neither the tooth module nor the helix angle. Townsend detailed a windage loss equation which included an oil mixture function ϕ and a gearbox space function λ and is presented here as Equation 1 (Ref. 3). The oil mixture function ϕ indicates the state/type of atmosphere inside the gear unit with $\phi = 1$ indicating an oil-free atmosphere. The gearbox space function λ is set at 1 for free space and reduces to a value of 0.5 for a closely fitting enclosure, i.e. the fitting of baffles or shrouds around the gear.

$$P_{WL} := n^{2.9}(0.16 \cdot d_f{}^{3.9} + d_f{}^{2.9} \cdot b^{0.75} \cdot m^{1.15})10^{-20} \cdot \phi \cdot \lambda \quad (1)$$

Oil Churning Loss. Townsend defined churning losses as the action of the gears moving the lubricant inside the gear case and referred in particular to the losses due to entrapment of the lubricant in the gear mesh, which is more applicable to spur gears than to helical gears (Ref. 3).

Factors influencing the oil churning loss are the viscosity of the oil, as this resists the motion of the gears; peripheral velocity; operating temperature; the tooth module; the helix angle; and the submerged depth of the gears. All rotating components that are in direct contact with the lubricant, i.e. dipped into the oil, contribute to the churning losses, and the deeper the components are submerged, the higher the losses. With larger helix angles, the power losses are lower as the gear teeth slice through the lubricant rather than displacing the lubricant along the whole gear face width. Other expressions for determining churning losses have been proposed (Ref. 4). The British Standard BS ISO/TR 14179 Part 1 details churning loss equations which had been modified for the effect of lubricant viscosity, element diameter, the gear dip factor and the arrangement constant (Ref. 5). These churning loss expressions were split into three different sections, which are detailed in Equations 2–4.

Churning Loss for Smooth Outside Diameters (i.e. shafts):

$$P_{CLi} := \frac{7.37 \cdot f_g \cdot v \cdot n^3 \cdot D^{4.7} \cdot L}{A_g \cdot 10^{26}} \quad (2)$$

Churning Loss for Smooth Sides of Discs (i.e. gear side faces, both faces)

$$P_{CLi} := \frac{1.474 \cdot f_g \cdot v \cdot n^3 \cdot D^{5.7}}{A_g \cdot 10^{26}} \quad (3)$$

Churning Loss for Tooth Surfaces:

$$P_{CLi} := \frac{7.37 \cdot f_{gi} \cdot v \cdot n_i^3 \cdot D_i{}^{4.7} \cdot b_i \cdot \left(\frac{R_f}{\sqrt{\tan(\beta)}} \right)}{A_g \cdot 10^{26}} \quad (4)$$

Load-Dependent Losses

Sliding friction loss. Principally, the instantaneous sliding friction loss is a function of the instantaneous sliding velocity and the friction force, which itself is a function of the instantaneous normal tooth load and the instantaneous coefficient of friction. The magnitude of sliding velocity depends on the position of contact along the contact path with a peak velocity at the start of the approach. The velocity reduces to 0 at the pitch point of the two mating gears and rises again to a peak value at the end of the recess. The effect of the sliding friction loss is an increase in power consumption, where the magnitude depends on the point of contact. It is influenced by the angular velocity of the gears, the ratio of the rolling velocities, the point of contact, the contact ratio and the lubricant properties.

Anderson et al. analyzed the sliding friction losses along the path of contact and postulated expressions for the instantaneous sliding velocity and instantaneous friction force, where the friction force was a function of the instantaneous coefficient of friction and gear load. The sliding friction loss is dependent on the position of contact during the engagement cycle (Refs. 1–2). While the model proposed was specific for spur gears, the authors have modified the expression for the instantaneous coefficient of friction to include helical gearing. This was accomplished by modifying the expression for instantaneous coefficient of friction to take into account the helical gear contact length.

The instantaneous sliding power loss is given as:

$$P_S(x) := 10^{-3} \cdot V_S(x) \cdot F_S(x) \quad (5)$$

where

$$F_S(x) := \mu(x) \cdot w(x) \quad (6)$$

The postulated expression proposed by Anderson et al. for the instantaneous coefficient of friction for spur gears is detailed in Equation 7 and was employed by the authors for this investigation (Refs. 1–2). The coefficient of friction is given as:

$$\mu(x) := 0.0127 \cdot \log\left(\frac{C_1 \cdot \frac{w(x)}{b}}{\mu_0 \cdot V_S(x) \cdot V_T(x)^2} \right) \quad (7)$$

The contact length for helical gears, L_c, has been detailed and was substituted by the authors into Equation 7 (Ref. 7). Therefore, the modified equation for the coefficient of friction for helical gears

is given as:

$$\mu(x) := 0.0127 \cdot \log\left(\frac{C_1 \cdot \frac{w_n(x)}{L_c}}{\mu_0 \cdot V_S(x) \cdot V_T(x)^2}\right) \quad (8)$$

The coefficient of friction used in this analysis is independent of the gear surface temperature, which—strictly speaking—is inaccurate. The expression in Equation 8 was substituted into Equation 6 to obtain the sliding power losses.

Rolling friction loss. The rolling friction loss is dependent on the instantaneous rolling velocity and the instantaneous lubricant film thickness. As the gear teeth come into mesh, an elastohydrodynamic lubricant film is developed between the teeth in contact. The action of the gear teeth during the engagement draws the lubricant into the contact zone. The parameters that influence the rolling friction loss are the lubricant film thickness, the angular velocity of the gears, the working pressure angle and the point of contact along its contact path. The lubricant properties influence the buildup of the lubricant film, its shear values and its thermal behavior. In addition, the gear material and the normal tooth load also influence the film thickness. In References 1–2, Anderson et al. postulated the instantaneous rolling friction force as:

$$F_R(x) := C_2 \cdot h(x) \cdot \phi_t(x) \cdot b \quad (9)$$

where the rolling power loss is given as:

$$P_R(x) := 10^{-3} \cdot V_T(x) \cdot F_R(x) \quad (10)$$

This expression for instantaneous rolling force includes a thermal reduction factor that accounts for the decrease in oil film thickness as the pitch-line velocity increases (Ref. 6). A relationship between thermal loading factor and reduction factor was presented by Anderson et al. and employed by the authors of this investigation (Ref. 1). The paper by Anderson et al. implied that prior to computing the thermal reduction factor, the thermal loading factor must be determined. To account for helical gears, the expression of Equation 12 was modified by the authors, as the contact line length in helical gears is not synonymous with the face width. The modified instantaneous rolling force is given as:

$$F_R(x) := C_2 \cdot h(x) \cdot \phi_t \cdot L_c \quad (11)$$

where L_c, has been defined as the contact length for helical gears. The expression was substituted into Equation 10 to obtain the sliding power losses.

Mathematical Model

The mathematical model employed for this investigation consisted of the following expressions:

1. The windage loss equation as employed by Townsend (Ref. 3).
2. Churning loss expressions as detailed in BS ISO/TR 14179, Part 1 (Ref. 5).
3. The rolling and sliding friction losses as postulated by Anderson et al. (Ref. 1).

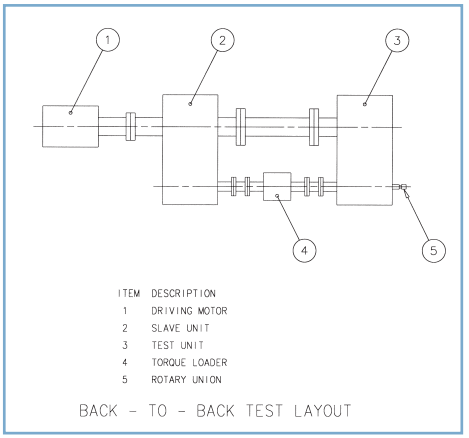

Figure 1—Schematic of the back-to-back arrangement for the high-speed gear unit.

Model Validation

The gearbox used to validate the model was a single-stage, double-helical, speed-increasing gear unit with oil film bearings of the circular type. The inlet oil temperature was 49°C, and the maximum bearing temperature did not exceed 87°C. The input shaft and the output shaft end were sealed by means of a shaft-mounted oil flinger and non-contacting baffle rings in the housing. A lubrication pump was driven via a set of reduction gears from the input shaft. The pump supplied the lubricant for the gear sprayers and the force-fed bearing. The gear unit was tested in a back-to-back arrangement (see Fig. 1). A torque loader was fitted between the output shaft coupling end of the slave unit and the output shaft tail end of the test unit. The torque loader used for the experiment employed a pressurized oil system. The oil was supplied via a rotary union (see Fig. 1). The torque loader consists of an inner rotor and outer rotor, which are supported in bearings. Oil is fed into the space between the two rotors, creating a torsional load in the test rig. The bearing losses in the torque loader were calculated separately beforehand and subtracted from the motor input power prior to calculating the gearbox efficiency.

The load conditions for the experiment included 25%, 50%, 75%, and 100% of full load (8.95 MWatts), at 100% input speed (1,460 rpm). Experimental torque readings were taken with a telemetric system from the low speed shaft via strain gauges. Appendix A details some gear data. It must be noted that the rig was run at full load and maximum speed (1,460 rpm) for a period of four hours.

Table 1—Experimental and Theoretical Results of Experiment I.				
Power input (kW)	8,952	6,714	4,476	2,238
% load	100%	75%	50%	25%
Experimental total loss (kW)	125.33	116.82	112.35	106.75
Experimental total loss (kW) for gear and windage only	55.08	50.15	50.16	49.42
Predicted total loss (kW)	57.41	53.3	49.86	47.42
Breakdown of Losses				
Sliding friction loss (kW)	10.88	6.672	3.05	0.34
Rolling friction loss (kW)	5.71	5.821	5.98	6.27
Churning loss (kW)	0	0	0	0
Windage loss (kW)	40.82	40.82	40.82	40.82

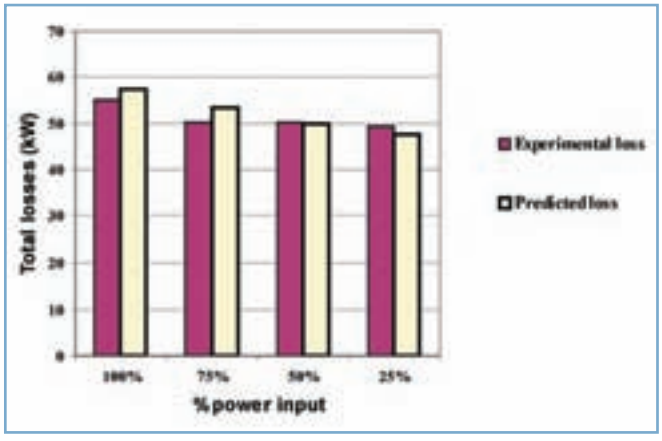

Figure 2—Comparison between experimental gear and windage losses and predicted power losses.

Prior to undertaking this test, the gear unit was visually inspected to ensure gear contact markings were satisfactory full face and full depth.

The mathematical model for the helical gears in this instance was accomplished by doubling the face width. The oil mix function was assumed to be $\phi = 3$ and the gearbox space function was taken as $\lambda = 1$ as the gear case walls were sufficiently far away from the gears to be considered as free space. No oil churning took place, as this was a spray-lubricated arrangement. The experimental results provided the efficiency for the complete gear unit; therefore, the bearing losses, seal losses and absorbed power for the lubrication pump had to be calculated from manufacturer's information (see Appendix B). The power loss calculations for the bearings assumed the maximum clearance condition. As non-contacting seals were employed, no power losses from the seals were assumed.

As the lubrication pump reduction gears were not separately sprayed, the oil mist present inside the gear case was assumed sufficient to provide lubrication. Again the same oil mix function and gearbox space function were assumed. The total predicted power losses were the sum of the losses from the lubrication pump and the gearbox. Results are detailed in Table 1. A comparison between predicted and experimentally determined power losses is shown in Figure 2. The model predicts a steady decline in power loss corresponding to a reduction in load.

It was observed that by increasing the power input at a fixed rotational speed, the windage losses remained the same and the rolling friction power losses decreased while the sliding loss increased.

Discussion and Conclusion

The mathematical model detailed in this paper has shown to be valid, providing an indication of the contribution of each element within a gearbox with helical gearing to the total power loss. The predictions and the experimental results show a good correlation, although the experimental results do not provide a breakdown of the various power losses. In the breakdown of losses in Experiment I, it can clearly be seen that the sliding friction losses are heavily load dependent, increasing with load. However, the rolling friction losses decreased slightly with an increased load, and this is due to a decrease in oil film thickness. As the speed was constant during the experiment, windage losses remained constant throughout the tests.

This investigation did not review or include new mathematical models for load-dependent bearing losses and speed-dependent bearing churning and seal losses.

Acknowledgments

The authors wish to express their gratitude to David Brown, Textron Power Transmission for allowing the publication of this investigation, undertaken as a degree project for a master's of science in Design of Rotating Machines, Cranfield University.

This paper was presented at the ASME/AGMA 2003 International Power Transmission and Gearing Conference, held Sept. 3–5, 2003, in Chicago, IL, and was published in *Proceedings of the 2003 ASME Design Engineering Technical Conferences & Computers and Information in Engineering Conference*. It is republished here with permission from ASME.

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Appendix A—Gear Data.				
			Units	
Pinion teeth number	z_1	115	–	
Wheel teeth number	z_2	21	–	
Center distance	a	609.6	mm	
Normal module	m	8	mm	
Normal pressure angle	α	20	deg.	
Helix angle	β	25	deg.	
Face width	b	285.75	mm	Double helical
Pinion shift coefficient	x_1	0.989	–	
Wheel shift coefficient	x_2	0.181	–	
Pinion, Young's modulus	E_1	207,000	N/mm^2	
Wheel, Young's modulus	E_2	207,000	N/mm^2	
Pinion, Poisson's ratio	ny_1	0.3		
Wheel, Poisson's ratio	ny_2	0.3		
Specific heat	C_1	544	J/kgK	
Specific heat	C_2	544	J/kgK	
Thermal conductivity	K_1	46	W/mK	
Thermal conductivity	K_2	46	W/mK	
Application				
Pinion speed	n_1	1,460	rpm	
Transmitted power	P	8,952	kW	
Lubricant				
Lubricant factor	–	1		for mineral oil
Viscosity	ny	46	mm^2/s	at 313 deg. K
	ny	21	mm^2/s	at 333 deg. K
Specific gravity	ρ	873	kg/m^3	
Dynamic viscosity	η	0.01833	Pa*s	
Viscosity-pressure coefficient	α	2.20E–08	m^2/N	
Thermal conductivity	K_f	1.25E–01	W/(m*K)	

Appendix A—Calculated Gear Data.			
Contact line length	L_c	mm	499.57
Operating diameter speed	v	mm/s	78,811
Gear contact tangential load	W_{pt}	N	113,588.59
Gear contact normal load	W_{pn}	N	133,374.62
Equivalent Young's modulus	E_{eq}	N/mm^2	227,472.53
Pinion torque	T_1	Nm	58,551.58

Appendix B—Bearing, Lubrication Pump and Seal Losses.				
Power input (kW)	8,952	6,714	4,476	2,238
% load	100%	75%	50%	25%
Experimental total loss (kW)	125.33	116.82	112.35	106.75
Power Loss in Each Bearing				
Pinion coupling bearing (kW)	22.9	21.37	19.37	17.28
Pinion tail bearing (kW)	22.9	21.37	19.37	17.28
Wheel coupling bearing (kW)	3.93	3.67	3.43	3.09
Wheel tail bearing (kW)	3.93	3.67	3.43	3.09
Lubrication Pump				
Absorbed power (kW)	16.59	16.59	16.59	16.59
Seal Losses at Each Shaft (kW)				
Pinion coupling (kW)	0	0	0	0
Wheel coupling (kW)	0	0	0	0
Total loss (kW)	70.25	66.67	62.19	57.33

Gear Finishing with a Nylon Lap

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Management Summary

The objective of this research is to develop a new lapping process that can efficiently make tooth flanks of hardened steel gears smooth as a mirror. The lapping is carried out using a nylon helical gear as a lap and using a simple mechanical device.

This paper first shows the lapping machine designed and manufactured by the authors and the procedure and principle of the lapping. As a result of the lapping tests, it becomes clear that the process can make tooth flanks of hardened steel gears smooth in a short time and that it is important to carry out the lapping using a nylon gear with a large helix angle and high teeth (teeth with addendums larger than their modules) under a condition of high rotating speed and light load in order to prevent increasing of tooth profile errors.

Moreover, the lapping process was varied to include microvibrations of the nylon gear. From the result, it is found that the addition of microvibrations enables this process to also reduce tooth profile errors of the work gears. The lapped gears' running noise and load carrying capacity are considered, too.

Introduction

It is known that the tooth flank roughness of a gear influences its performance in power transmission—that is, the load carrying capacity of gears is improved, and the running noise and vibration are reduced by making the tooth flanks smooth (Refs. 1–2).

Gear finishing methods such as shaving, hobbing with a carbide hob, grinding and honing are used in the gear industry. However, they generally require a highly accurate machine tool, advanced technical skills and a lot of time, or they require the hardness of the work gear be restricted. Furthermore, it is next to impossible to make tooth flanks smooth as a mirror by these methods.

We propose a new lapping process that can productively make tooth flanks of hardened gears smooth as a mirror. The lapping process is carried out using a nylon helical gear as a lap on a simple mechanical device.

This paper will first show the mechanism of the lapping machine designed and manufactured by the authors and the procedure and principle of the lapping. Next, the lapping tests were performed on highly accurate ground gears. The test results showed that: 1.) the process could improve tooth flank roughness of hardened gears in a short time and 2.) the lapping conditions to prevent increases in tooth profile errors were clarified. Therefore, we could get highly accurate gears with smooth tooth flanks, as shown in Figure 1, the lapped gear on page 38.

Also, by vibrating the nylon gear, the lapping process could improve the tooth profile and the tooth flank roughness of hobbed gears. Running noise and load carrying capacity of the lapped gears were investigated as well.

Lapping Machine and Procedure of Lapping Process

Figure 2 shows the lapping machine that was designed and manufactured by the authors. A work gear is meshed with the lap, a nylon helical gear, and they are rotated at high speed by the driving motor. Lapping load is applied by means of braking the driven spindle. Lapping fluid that contains abrasive grains is splashed between the tooth flanks of nylon and work gears as they rotate. The work gear is given a traverse motion at low speed along the axis in order to finish the whole flank.

The specifications of the work gear and the nylon gear are shown in Table 1. The work gears are spur gears with modules of 4, 25 teeth, and face widths of 10 mm. The work gears' tooth flanks were accurately ground after carburizing and hardening or induction hardening. The carburized gears and induction-hardened gears are made of chromium-molybdenum alloy steel JIS SCM415 and JIS SCM435, respectively. The hardness of the tooth flanks are 860 HV and 640 HV, respectively.

Also, their tooth profiles, tooth traces and tooth flank roughnesses are of similar accuracy. An example is shown in Figure 3. The tooth profile measuring position is at the center of the face width. The tooth trace was measured on the pitch circle. The roughness was measured along the tooth profile near the pitch circle. The tooth profile and tooth trace are highly accurate. The tooth flank roughness has a maximum peak-to-valley height (R_y) of 1.5–2 µm, which can translate to an average surface roughness (R_a) of 0.24–0.31 µm.

The test gears were ground using a Reishauer RZ300 E. This machine is a fairly common gear grinder in the United States and can achieve the above R_a. It may

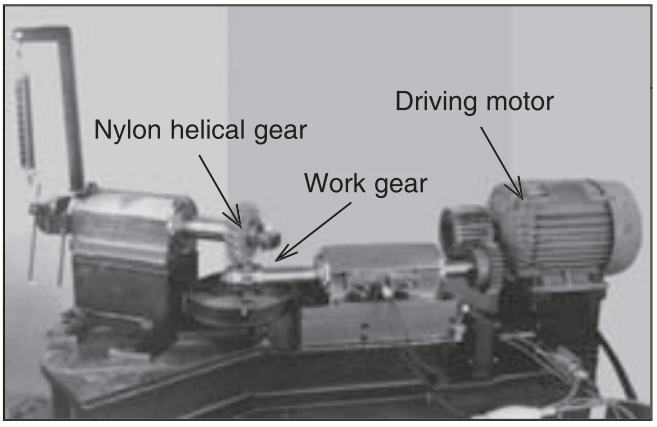


Figure 2—Lapping machine.

Table 1 — Specifications of Work and Nylon Gears.

	Work Gear	Nylon Gear
Module	4	4
Pressure Angle (deg.)	20	20
Helix Angle (deg.)	0	30 (R, L)
Number of Teeth	25	19
Face Width (mm)	10	34
Materials	SCM435 (640 HV) SCM415 (860 HV)	Nylon
Tooth Flank Condition	Ground	Hobbed

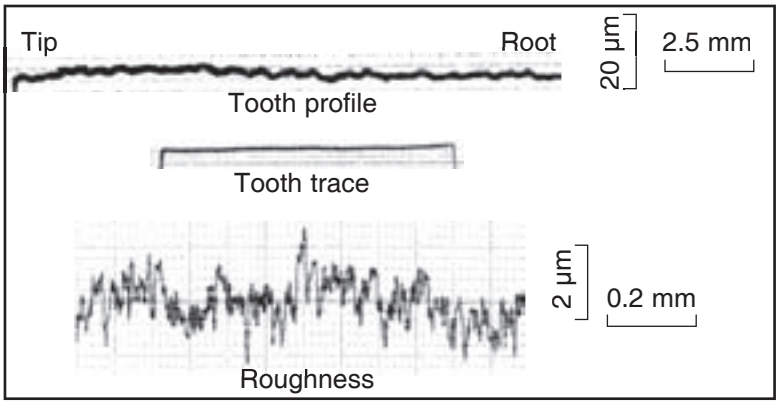


Figure 3—Tooth profile and roughness of ground gear.

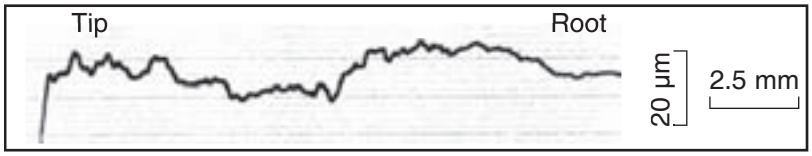
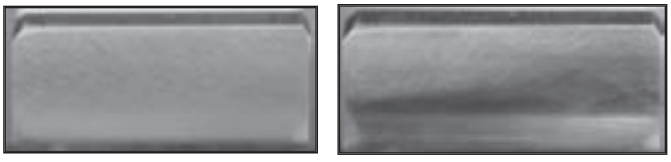


Figure 4—Tooth profile of nylon gear.

Table 2—Nylon Properties.		
Specific Gravity	Tensile Strength	Elastic Modulus
1.15–1.17	78–96 MPa	2.9–3.4 GPa

Table 3—Lapping Conditions.	
Abrasive Grain	WA1000, GC6000
Rotating Speed of Work Gear (rpm)	610; 1,210; 2,290; 3,420
Traverse Feed Rate (mm/min)	300
Stroke (mm)	11
Load (N)	3.8, 7.6, 15.2



a) Roughing (WA500) b) Finishing (WA2000)

Figure 5—Lapped tooth flanks.

Figure 6—Tooth flank of nylon gear.

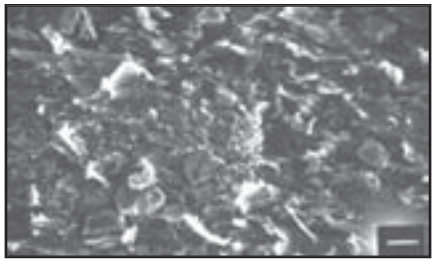


Figure 7—Abrasive grains embedded in nylon gear.

be difficult to achieve it in a normal production environment, though. An average roughness (R_a) of 0.4–0.5 µm may be more easily obtained in regular production with a threaded wheel grinding machine, like the Reishauer machine. However, this slightly rougher finish would not affect lapping time, cost, or other results presented in this paper.

The nylon helical gears were hobbed. Their tooth profiles were not highly accurate, as shown in Figure 4. The nylon gears are easily machined. The nylon's properties are shown in Table 2.

Table 3 shows the lapping conditions used in the experiments. This process consists of a roughing by coarse abrasive grains and a finishing by fine abrasive grains. A helical gear with right-hand teeth is used in the roughing, and a helical gear with left-hand teeth is used in the finishing. Since each direction of the lapping crosses the other as shown in Figure 5, the time for finishing becomes shortened. Lapping fluid for roughing consists of 200 cc of mineral oil and 5 g of alumina abrasive grains (WA1000). That for finishing is about 14% Carborundum abrasive grains (GC6000).

The rotating speeds of the work spindle are 610; 1,210; 2,290; and 3,420 rpm. The traverse speed and stroke of a work gear are fixed at 300 mm/min and 11 mm, respectively. Consequently, the lapping takes about four seconds per stroke. The lapping loads on the pitch cylinder of the work gear are 3.8, 7.6 and 15.2 N.

Principle of lapping process. Figure 6 shows an example of tooth flanks of a nylon gear after lapping six work gears with WA500. A section that became dark during the lapping can be seen on the surface. Figure 7 is a photograph of the darkened section enlarged by SEM. It was found that the surface became rough, and abrasive grains were embedded in it. Chips from lapping are shown in Figure 8. They look like chips from grinding. These figures show abrasive grains embedded in the surface of a nylon gear after they've worked between the tooth flanks of the work gear and nylon

gear. The nylon gear, in effect, acts like a grinding wheel.

Lapping Tests of High Accuracy Ground Gears

Increases in tooth profile error. Figure 9 shows the tooth profile, tooth trace and tooth flank roughness of a carburized gear lapped using WA1000 abrasive grains, a nylon gear with a helix angle of 30 degrees, a work spindle speed of 1,210 rpm, and a lapping load of 15.2 N. The roughness was reduced to an R_y of about 0.5 µm through seven strokes of lapping, about 30 seconds. However, a concavity is found near the pitch point on the tooth profile. It was supposed that a heavier load caused the concavity near the pitch cylinder during the engagement of one pair of teeth.

Therefore, we tried to lap a carburized gear with a lapping load of 3.8 N and a work spindle speed of 2,290 rpm. The roughness was reduced to an R_y of about 0.5 µm through two strokes of lapping, about eight seconds. The tooth profile is shown in Figure 10. A concavity does not occur near the pitch point.

Figure 11 shows the tooth profile of an induction-hardened gear lapped under the same conditions as Figure 10. The hardness of the tooth flanks is lower than that of the carburized ones. A concavity occurs on the dedendum flank near the root. The same concavity occurred on the carburized gear lapped under a load of 7.6 N.

Figure 12 shows the calculated sliding velocity at every degree of rotation on a tooth flank of a work gear meshing with a nylon gear under a rotating speed of 2,290 rpm. The helix angles of the nylon gear were 30 and 15 degrees. The radius of 50 mm indicates the tip of the tooth in the figure. With this figure, it is clear that on the dedendum flank near the root, the sliding velocity is higher and the interval between the dots is smaller—that is, the movement of the contacting point is also slower than at other positions.

Consequently, it was supposed that the removal amount near the root is larger so the concavity occurred there. With the smaller

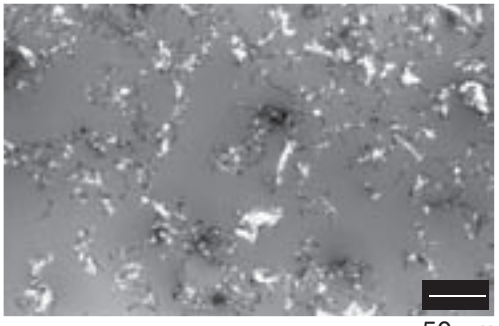


Figure 8—Chips from lapping.

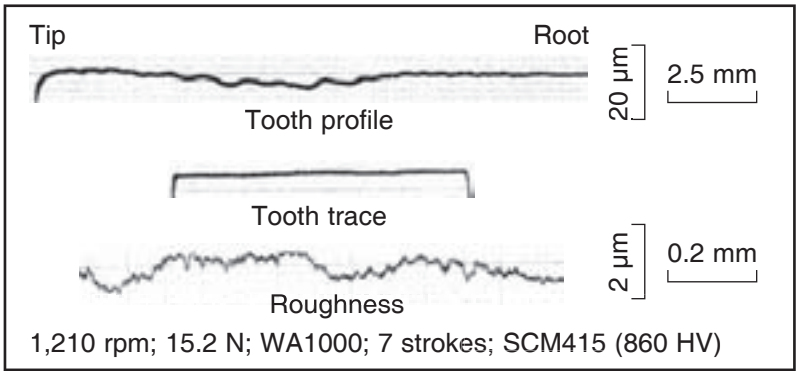


Figure 9—Tooth profile and roughness of lapped gear.

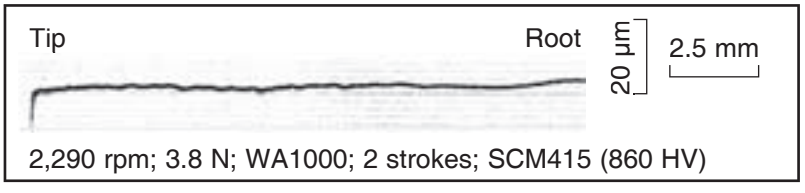


Figure 10—Tooth profile of lapped gear.

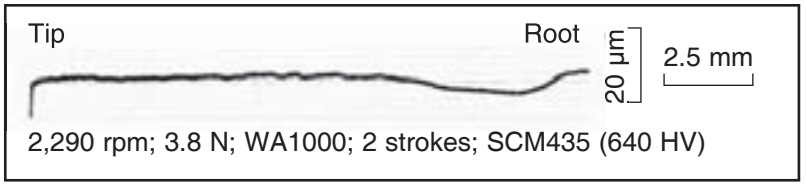


Figure 11—Concavity on dedendum flank.

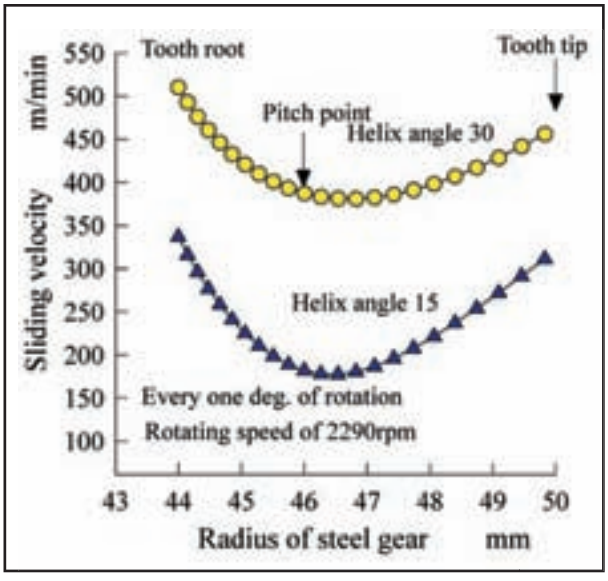


Figure 12—Sliding velocity on a tooth flank of a work gear.

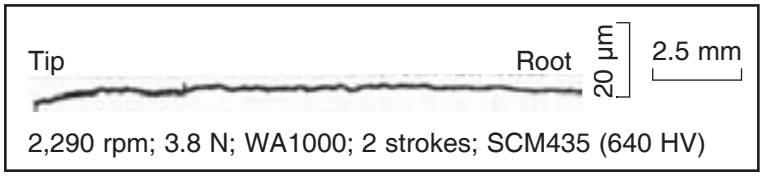


Figure 13—Effect of nylon gear with high teeth.

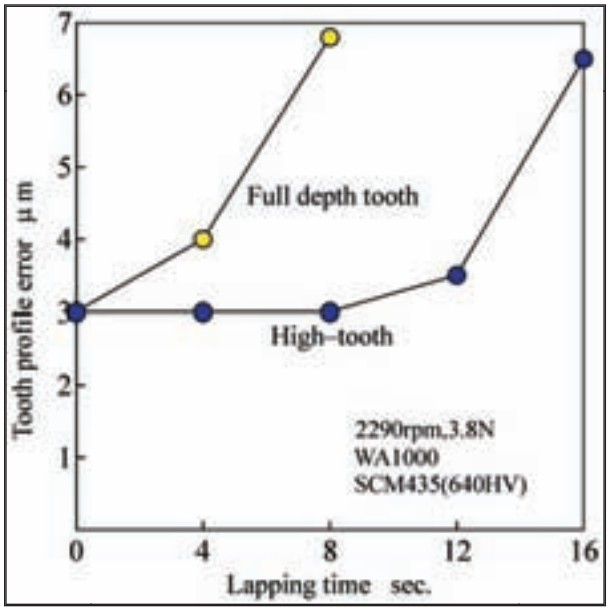


Figure 14—Effect of nylon gear with high teeth.

helix angle, the difference in sliding velocity along the tooth profile is larger. This resulted in the occurrence of a concavity. The same concavity also occurred on the carburized gear with a longer lapping time.

Prevention of the concavity near the root. In order to prevent the concavity near the root, we tried to lap an induction-hardened gear using a nylon gear with high teeth under the same conditions as Figure 11. The tooth flank roughness was reduced through two strokes of lapping. The tooth profile is shown in Figure 13. A concavity was not found.

Figure 14 shows changes of the profile error after every four seconds of lapping time, as compared with the case of using a nylon gear with full-depth teeth. When lapping with a nylon gear with full-depth teeth, profile error increases rapidly because of the occurrence of a concavity near the root. However, there is little change until three strokes of lapping (12 seconds) in the case of a nylon gear with high teeth. When a nylon gear with high teeth (see Fig. 15) is used, the contact ratio increases and three pair of teeth contact, so the load near the root shown in Figure 16 is decreased. This decrease means the concavity does not occur readily.

Figure 17 shows the tooth profile and the tooth flank roughness of a carburized gear after a roughing using a nylon gear with high teeth and a finishing using a nylon gear with full-depth teeth. The roughness is reduced in comparison with that before lapping, and no concavity is found on the tooth profile after four strokes of roughing. In the finishing stage, it took four strokes to remove roughing traces using GC6000 abrasive grains under a lapping load of 7.6 N. The roughness is reduced further through the finishing, but a small concavity is found near the root. Therefore, it is necessary to use a nylon gear with high teeth when finishing with fine grains.

Figure 18 shows changes of tooth flank roughness and profile error of carburized gears lapped using WA1000 abrasive grains under rotating speeds of 610–3,420 rpm.

It is clear in these figures that the roughness more rapidly decreases as the rotating speed increases, and the profile error increases at low rotating speeds because of long lapping time.

Lapping Process with Microvibrations Added to the Nylon Gear

We next tried to improve the tooth profile and tooth flank roughness of hobbed gears through this lapping process by adding microvibrations to the nylon gear.

The lapping tests were carried out using a new lapping machine and hobbed gears having the same dimensions as mentioned above. The amplitude and frequency of the microvibrations were fixed at 0.2 mm and 88 Hz, respectively. The lapping was carried out under a lapping load of 15.2 N. The rotating speed of the work spindle was 2,000 rpm. These work gears were finished with a carbide hob after they were hardened to 350 HB.

New lapping machine. The newly developed lapping machine is shown in Figure 19. It has basically the same mechanism as the one shown in Figure 2. Microvibrations can be added to the driven spindle along the axis by means of an eccentric cam on the new lapping machine. And the driving spindle is located above the driven spindle so that a work gear can easily be attached or removed.

Effect of microvibrating nylon gear. Figure 20 shows the comparison of the removal amounts based on the presence or absence of microvibrations during the lapping process. The removal amounts were measured through the weight of the work gears before and after the lapping process. It is clear from the figure that the addition of microvibrations increases the removal amount.

Figure 21 shows changes of tooth profile and tooth flank roughness via lapping with microvibrations. The gear was rough lapped using abrasive grains WA1000 and finish lapped using abrasive grains WA2000, respectively, for 16 seconds. It was found that the lapping with microvibrations improves the tooth profile as well as

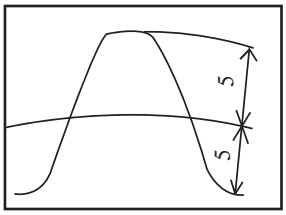


Figure 15—High tooth.

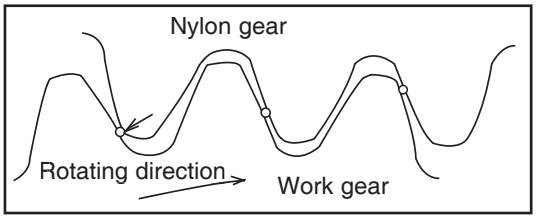


Figure 16—Contact of a work gear and a nylon gear with high teeth.

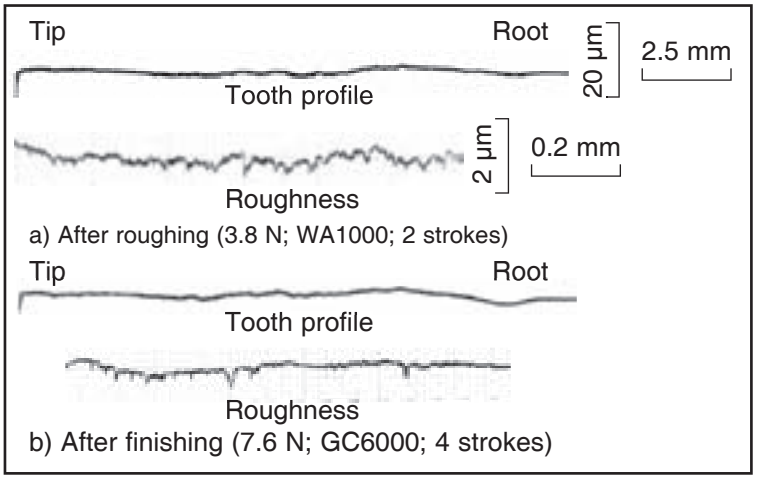
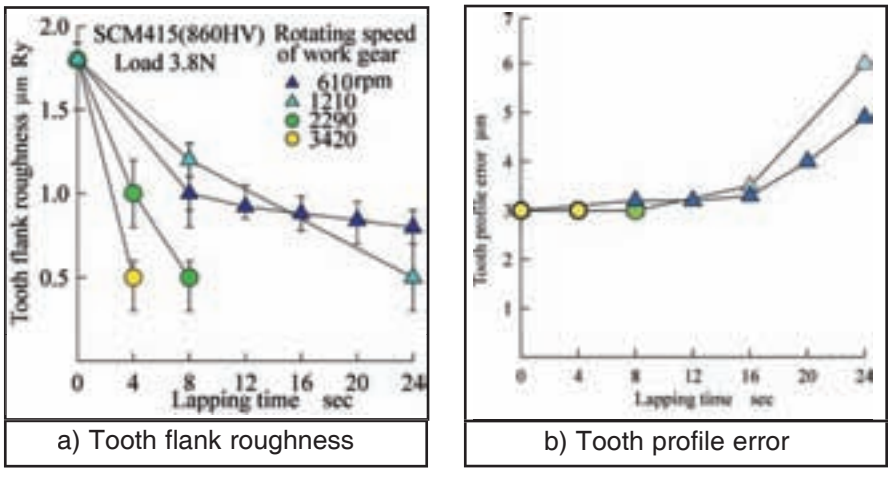


Figure 17—Effect of nylon gear with high teeth when made of SCM435 (860 HV) and rotating at a speed of 2,290 rpm.



a) Tooth flank roughness

b) Tooth profile error

Figure 18—Effect of rotating speed.

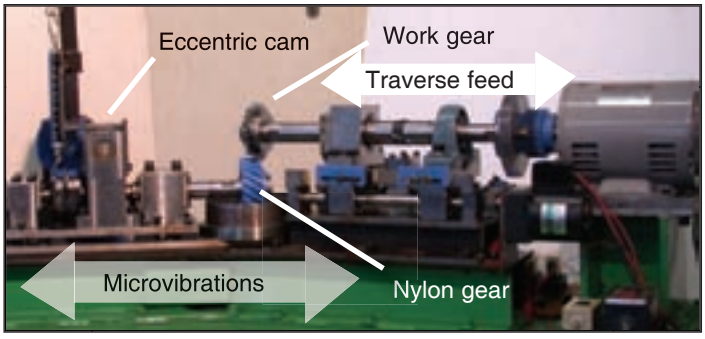


Figure 19—New lapping machine.

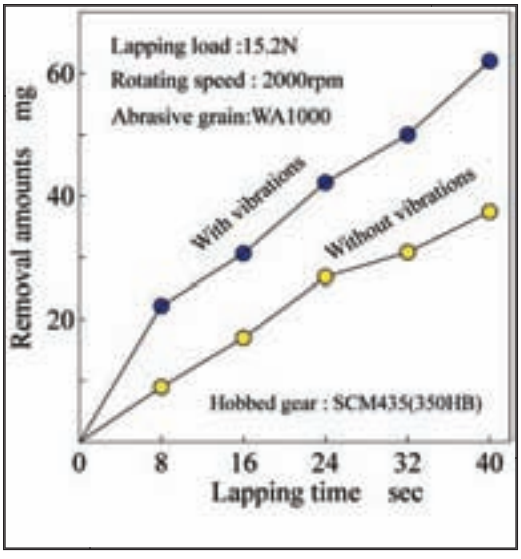


Figure 20—Effect of microvibrations on removal amounts.

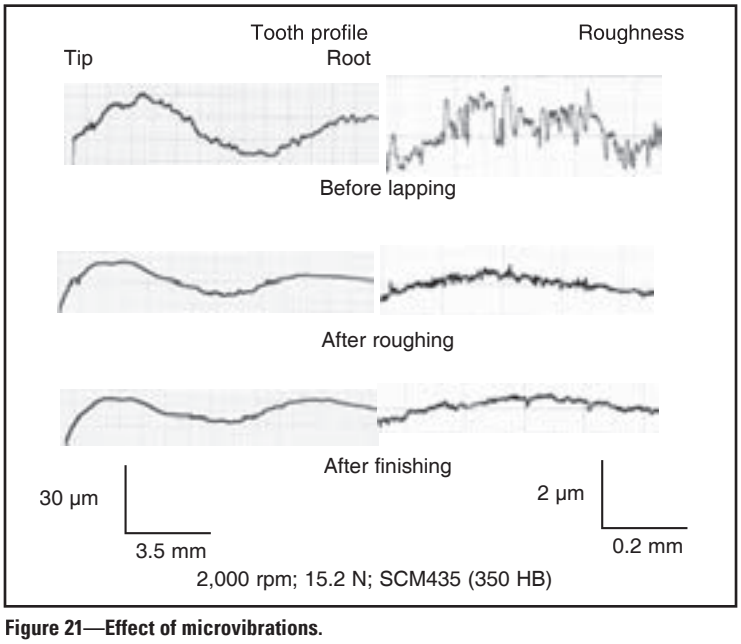


Figure 21—Effect of microvibrations.

the tooth flank roughness.

Effect of lapping. In addition, we investigated the running noise and durability of a pair of gears lapped under the aforementioned conditions. The numbers of teeth of the driving gear and the driven gear were 23 and 25, respectively. The running tests were carried out using a power-circulation-type gear testing machine with a driving gear rotation speed of 3,580 rpm. Applied pitch-line load was 1,188 N/cm, which gave a maximum Hertzian stress of 745 MPa.

Figure 22 shows the spectral distributions of the running noise for the lapped gears compared with that of a pair of hobbed gears. The sound pressure level of the lapped gears is lower at all frequencies. The difference is larger at higher frequencies.

Figure 23 shows changes in the pitting area ratio. The pitting on the lapped gears occurs later and increases more slowly than on hobbed gears.

Conclusions

We investigated the lapping process using a nylon lap and a simple mechanical device for finishing gear tooth flanks. Based on the results, the following conclusions are obtained:

1. In this process, abrasive grains embed in the surface of a nylon gear when they're injected between the tooth flanks of the work gear and nylon gear. The nylon gear, in effect, works like a grinding wheel.
2. The process can improve tooth flank roughness of hardened steel gears to an R_y of less than 0.5 µm in a short time despite using an inaccurate lap.
3. In lapping under a low rotating speed and a heavy load, a concavity is apt to occur near the pitch cylinder during the engagement of one pair of teeth. As the rotating speed increases, the removal amount near the root is greater, so the concavity occurs there.
4. It is important to complete lapping in a short time using a nylon gear with a large helix angle and high teeth under a high rotating speed and a light load in

order to prevent increases of tooth profile errors.

5. The addition of microvibrations to the process increases the removal amount and enables the process to improve tooth profile errors.
6. The running noise and the pitting rate of work gears are reduced through the lapping process.

This paper was presented at the ASME/AGMA 2003 International Power Transmission and Gearing Conference, held Sept. 3–5, 2003, in Chicago, IL, and was published in *Proceedings of the 2003 ASME Design Engineering Technical Conferences & Computers and Information in Engineering Conference*. It's republished here with permission from ASME.

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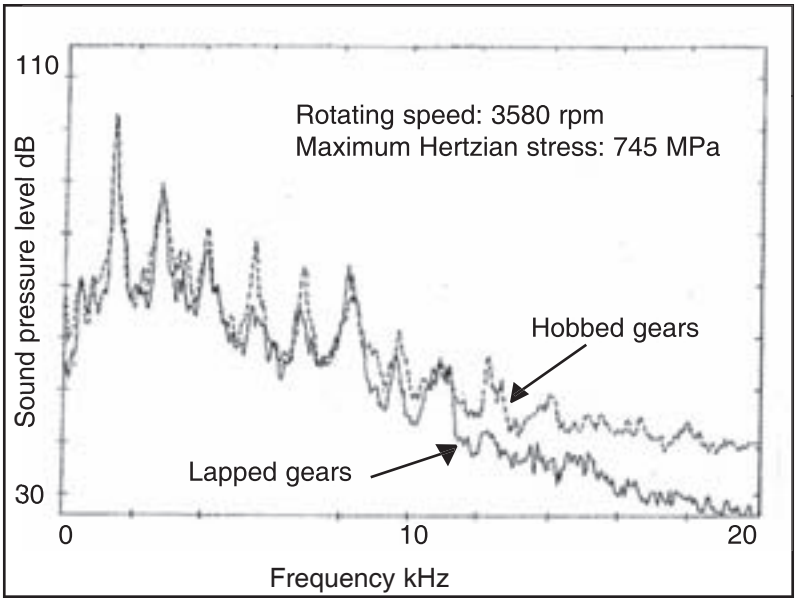


Figure 22—Effect of lapping on running noise.

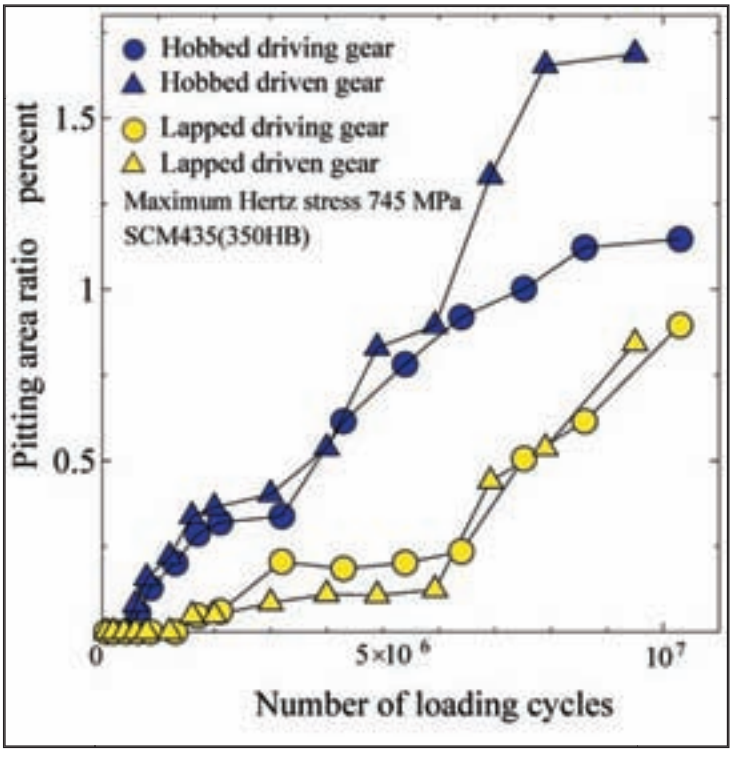
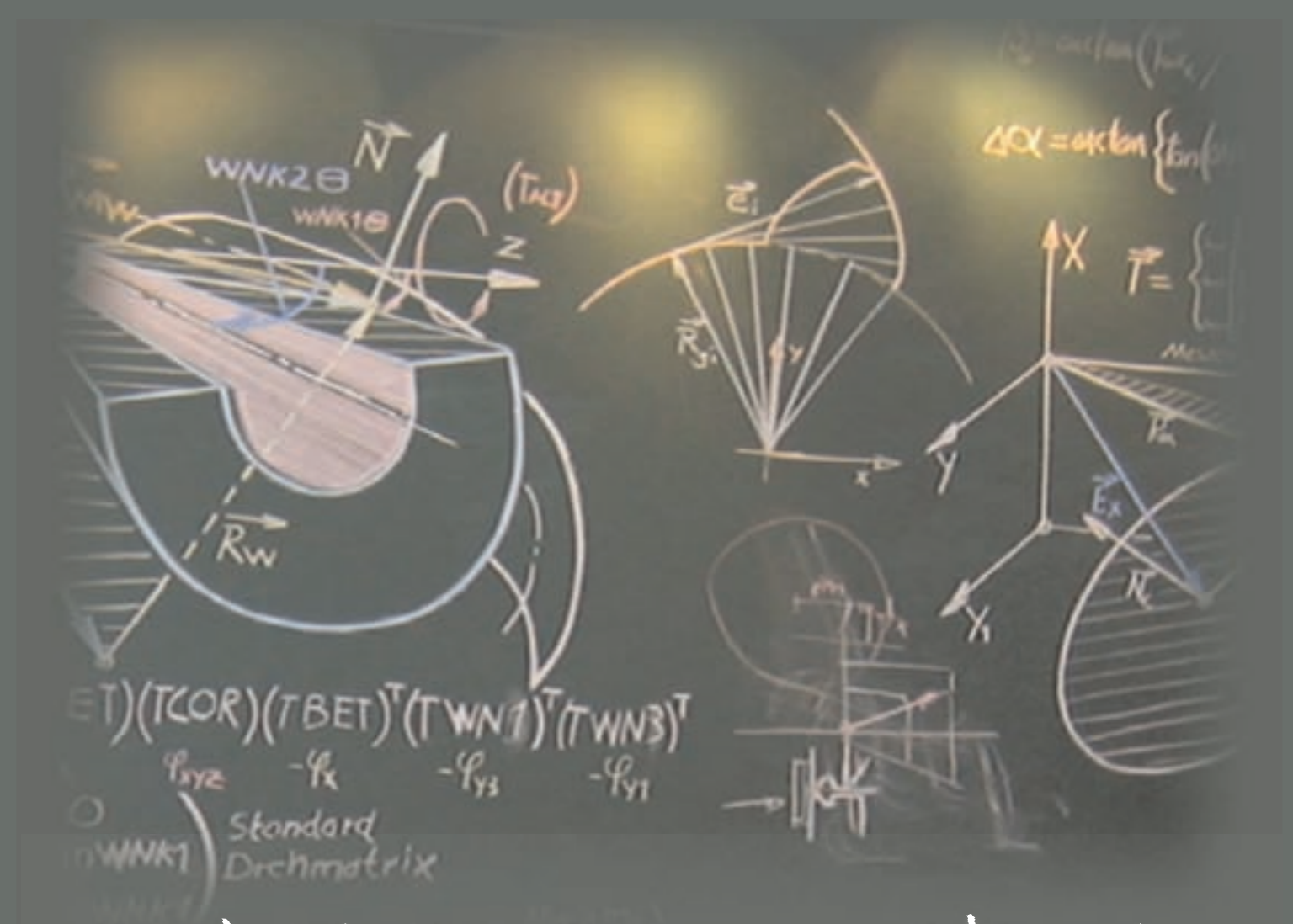


Figure 23—Effect of lapping on pitting.



What To Know About Bevel Gear Grinding

Dr. Hermann J. Stadtfeld *is managing director and co-founder of BGI GmbH & Co KG. Located in Eisenach, Germany, the company specializes in development and application of bevel gears for the automotive industry. Before BGI, Stadtfeld was a vice president and head of research & development for Gleason Corp. Effective Dec. 1, he will return to Gleason as its vice president–bevel gear technology. Previously, he's also served as director of engineering and research with Oerlikon GearTec AG in Zürich, Switzerland. His technical education began with an apprenticeship at ZF Friedrichshafen AG, a major European gear manufacturer. In addition, he received his doctorate in mechanical engineering from the Technical University of Aachen, Germany, researching bevel gears for his dissertation. Stadtfeld has spent more than 20 years developing software, hardware and other processes for gear design and optimization. He also trains engineering students at universities and gear engineers in companies around the world.*

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Guidelines are insurance against mistakes in the often detailed work of gear manufacturing. Gear engineers, after all, can't know all the steps for all the processes used in their factories, especially those used in the grinding of that most complicated type of gear: bevels.

And even when the steps are known, there are all the unwritten guidelines, the ones acquired through experience rather than from a design or manufacturing handbook. Those guidelines are numerous and require time to think about and finally see, whether the time is taken while standing on the shop floor or by going back to an office desk.

Gear engineers' work would be much easier if some of those major guidelines were written down, as they are here:

Several technological and geometrical factors are important to guarantee high-quality ground gears. The first factor is a smart strategy for semi-finishing gears. This strategy requires gear manufacturers to think about their processes in reverse order, making certain the gears they want to work on at the start of step four are the ones created by the end of step three. For example, uniform stock allowance on the flank surface is important, but only if the semi-finish cutting summary is derived from the finish grinding summary.

Also, sections of progressively increasing ease-off should not be ground without preparing them in the previous cutting operation. This applies particularly to universal-motion heel or toe sections as well as to second-order protuberance and flank relief.

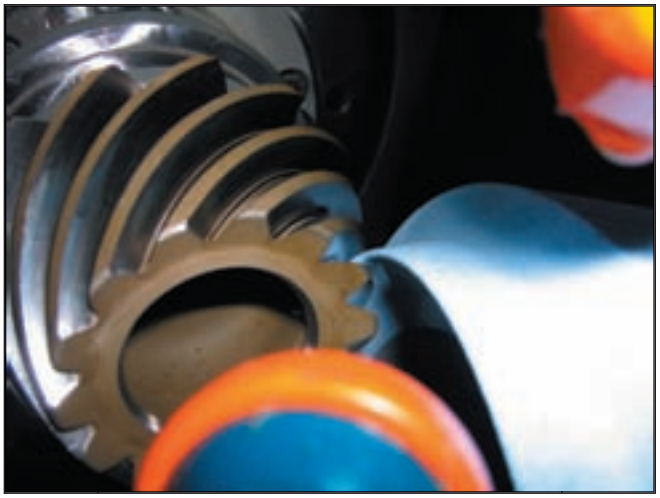
With heel or toe relief sections, a gear grinder sometimes has to remove 50% or more stock if the sections are not prepared properly during soft cutting. For example, a green gear may have a regular stock removal of 0.13 mm. Variation from heat treat distortion may add 0.07 mm in certain areas. Also, the hardened gear may require removal of an additional 0.10 mm of stock within the relief section. If the green gear isn't cut properly, a worst case could require the removal of 0.30 mm of stock in one grinding pass.

Possible results of such grinding include burn marks, new hardening zones or a reduction of surface hardness due to the reduced thickness of the case depth. The case depth of bevel gears in the module range of 3–6 mm is recommended to be between 0.8–1.2 mm after heat treatment. The worst-case scenario would reduce the case depth during grinding to 0.5 mm, perhaps less, reducing the surface and subsurface strength.

Also, grinding of root relief, the so-called blended Toprem®, leads to a critical condition on the grinding wheel because the small, sensitive tip of the grinding wheel might have to remove 10–30% more stock than the main profile section. Heavy material removal of the tip of the wheel causes a deterioration of the protuberance section and the edge radius after grinding only several slots. Subsequently, the remaining slots have lesser or no root relief and an unacceptable blend into the root-fillet radius. This effect cannot be cured by redressing after some number of slots.

An important part of the semi-finish strategy for modern bevel gear grinding is the root fillet area that is not ground. The optimal protuberance of the cutting blades relieves the transition between flank and root by a value equal to the stock allowance on the active flanks. The cutting blades should have an edge radius 0.1 mm smaller than the edge radius of the final grinding wheel profile. They also should cut 0.1 mm below the theoretical grind depth.

The transition between the grinding profile and the unground root area should be optimized on the drive side of both members,



using a grinding wheel tip extension and a setover, to get a smooth blend of the ground to the unground root surface and clean up the root radius to the area of 30° tangent.

The distortions due to heat treatment cause an unequal cleanup along the face width and from slot to slot around the circumference. Also, the first- and second-order corrections, applied after coordinate measurement to achieve a correct flank geometry, influence the angle of the ground root line versus the semi-finish cut root line. This root-angle difference might result in a partially ground root bottom.

This result is no problem as long as the slots' average length of ground root bottom is no more than 30%. Variations in cleanup can result in as much as 70% of a single slot's root bottom being ground. Such a high percentage in a single slot is acceptable, though. In that case, a slot's root geometry variation would be as much as 100 microns. Variation up to this amount in a slot will not adversely affect the performance of the gear set. Still, the average cleanup percentage of all slots can't be more than 30%.

If a face-milling geometry shows a high transition line between root and flank that was generated by the profile generating process and not by a too-large point radius of the tool, then an interference of the top edge of the mating teeth can initiate surface damage and noise excitation. This interference zone can be relieved via a second-order protuberance, which is a radius that connects the grinding wheel main profile and the edge radius with a tangential blend.

A finish-ground bevel gear set should have an R_z equal to or less than 5 µm and an R_a equal to or less than 0.8 µm.

The ground surfaces of hypoid gears always carry the risk of scoring during the initial wear-in period. To eliminate this risk, the flank surfaces of at least the ring gear should be phosphated. The risk of scoring may also be eliminated during the first operating period through the use of synthetic hypoid oil. Nevertheless, the risk of scoring in operation is reduced due to the enhancement of the surface finish.

Keys to efficient grinding are the abrasive material and the abrasive bond. Recommended for bevel gear grinding are grinding wheels with an abrasive of sintered aluminum oxide with an 80 grit and a soft ceramic bond with an open pore structure.

Three or four coolant pipes are directed tangentially to the grinding wheel circumference with a coolant speed about equal to the grinding wheel surface speed to apply a layer of coolant to the

grinding profile surface just before its entrance into the grinding zone. Additional pipes are located behind the grinding zone and are directed opposite to the grinding speed to extinguish the sparks, which would burn into the wheel bond and get into the grinding zone.

In addition, the wheel's surface has to be cleaned continuously with a high-pressure coolant jet that's connected to an extra pump, supplying the coolant at a pressure of at least 300 psi. The high-pressure jet has to be located about opposite the grinding zone and has to shoot coolant perpendicular to the profile surface. Because of centrifugal force, the chips tend to clog up the inside profile more, so appropriate attention must be paid to the design and function of the high-pressure cleaning system. The surface speed to achieve good surface finish in connection with minimal grinding wheel wear is 20–24 m/sec, a rather low value for a grinding process.

In the automotive industry, bevel gear grinding requires one rotation only, but each slot may be ground in two passes—the so-called double roll. This roll consists of one pre-finishing pass, during which the grinding machine moves the grinding contact from toe to heel (uproll) and one fine-finishing pass, rolling from the heel back to the toe. While a part is being ground, dressing should be done after a rotation is finished, not during the rotation.

In contrast, some gears for high quality machine tools require two rotations, and aircraft parts are ground in four or more rotations.

In all cases, though, the rule is: After each rotation, the grinding wheel should be redressed.

However, skip indexing should be avoided. This technique was developed in order to distribute wear more uniformly around the work, to avoid ramp-shaped spacing errors. Rather than grinding successive slots, skip indexing skips a preset number of slots. Consequently, grinding requires several revolutions to finish all slots.

Subsequently, though, it was discovered that many acoustic phenomena were caused by skip indexing. Experience with the technique showed that the resulting many small ramps generate noise with amplitudes in the tooth mesh frequency, the gear rotational frequency and with a frequency that corresponds to the number of ramps per revolution. In some cases, there even appears to be an additional modulation. Thus, wear compensation is the best way to reduce ramp-shaped spacing errors and other wear patterns.

Dressing the grinding wheel requires a dress roller that is diamond-plated and shapes a grinding wheel's profile like in a round grinding operation. But the wheel must not be dressed with a dress roller using single-point dressing. This is the type of dressing in which the dressing wheel is locked during dressing, so the relative surface speed comes only from the rotation of the grinding wheel (speed ratio: 0). Consequently, single-point dressing creates a flat spot on the wheel. Dress rollers, however, are designed to dress around wheel circumferences.

The wheel also must not be dressed when the roller's surface speed is equal to the wheel's surface speed (speed ratio: 1.0). At that ratio, the roller crushes the abrasive grain out of the wheel bond.

The crushing breaks complete grains out of the bond for the most open-pore wheel surface structure possible. The grinding improves the wheel's ability to grind gears, though it also reduces the structure's pore size. However, the combination of crushing and grinding is optimal.

The grinding between dress roller and grinding wheel surface reduces pore size by changing the remaining crystal-shaped grains, specifically by flattening their sharp corners, the ones towards the surface. Consequently, very small flats are generated.

This flattening is beneficial because the grains' original shape is only good for roughing, not finishing. Their crystal shape can't accurately represent the grinding wheel's dimensions and profile shape. Consequently, the surface roughness of ground gears will be high and not very accurate.

The very small flats can, however, accurately represent the wheel's diameter and profile shape. The changed crystals mean surface roughness of ground gears will be low, coolant will be able to access the grinding zone and removed material will be able to move out of the grinding zone.

For high-productive grinding, a dresser speed ratio of 0.6–0.8 is recommended.

For the highest possible surface finish, the recommended speed ratio is between –0.6 and –0.8. In that ratio range, the roller only shapes the abrasive grains—it doesn't crush them—and results in a grinding wheel surface with minimum pores. Grinding wheels can be dressed for this type of grinding by directing the roller speed against the grinding wheel speed, which leads to high relative velocities.

Gear manufacturers can also remove the grinding wheel's wear pattern from the gear finish by entering into the grinding machine's controller the number of teeth for fast wear and the percentage of this wear from the entire amount of wheel wear.

Wheel wear occurs rapidly during the first couple of slots and then with near-linearity for the following slots. Modern grinding machine controls have wheel wear compensation features that allow total wear compensation for the entire part and a fast wear compensation for the first slots. After total wear is compensated, the spacing measurement will show whether the first slots require additional compensation and whether that amount should also be removed from the remaining slots.

After grinding a number of development gears, a manufacturer can obtain the numbers needed for eliminating the wheel wear pattern—specifically, the amount of fast wear on the wheel, the number of teeth for this fast wear and the entire amount of wheel wear.

When finishing Formate™ ring gears, they should be ground using Waguri grinding to make the process more stable and faster.

Normally, Formate gears are ground in a plunge-cut cycle. In form grinding, that means the grinding wheel has simultaneous contact with the whole flank surface of both flanks of a slot. Coolant can't reach the grinding zones, with the results being burning and fast grinding wheel contamination with metal particles.

The process can be made stable, however, through a different, additional rotation of the rotating grinding wheel spindle. This other rotation should be about 0.3 mm eccentric in the plane of rotation about the theoretical grinding wheel axis. The eccentric rotation's speed should be 200–500 rpm lower than the grinding wheel speed, with a typical value of 2,000 rpm and the same hand of rotation as the grinding wheel itself. This grinding technique is called Waguri, after its inventor.

Waguri grinding is extremely fast. It can achieve grinding times of one second per slot. A Waguri ground gear with 35 teeth is ground more quickly than a conventionally ground pinion with 13 teeth.

Lapping both finishes a gear and pinion and mates them together. Grinding doesn't. This seeming deficiency can be an advantage for gear manufacturers. When finished via lapping, the mated gear and pinion must be kept together. When finished by grinding, gears and pinions can still be dealt with individually. Their mating can be delayed until assembly.

Before assembly, a gear and pinion can then undergo roll testing to determine their axial pinion shim value. Afterward, they must be labeled with the value and kept together. They are, in effect, mated via roll testing.

By being able to delay mating, gear manufacturers that grind have less complicated logistics and more flexible manufacturing than those that lap.

Also, each gear set should undergo roll tests for three or five different axial pinion positions. These tests are recommended because they often reveal a best pinion axial assembly position.

Moreover, roll testing should be combined with a structure-borne noise evaluation. The noise test isn't as important, though, for ground gear sets in transaxle or power takeoff units because noise is not an extremely critical issue for them.

Lastly, don't be tempted to favor lapping over grinding for economic reasons. Admittedly, grinding is slower and more expensive than lapping. For example, grinding a pair of bevel gears requires two grinding machines and two minutes for each gear. Lapping also takes two minutes for each gear, but it requires only one lapping machine. Moreover, the grinding machine cost per ground set is about three times the lapping machine cost per lapped set.

However, rejected parts and customer complaints are 1% or less for typical grinding production compared with 3–7% for lapping, depending on the requirements for certain jobs. Rejected gear sets include the cost of all previous operations plus the material. Consequently, the cost related to this difference in reject rate often makes grinding the more economical process.

Lapping remains competitive, though. Many United States bevel gear manufacturers use it rather than grinding, having selected it when grinding was still developing into a reliable finishing process for higher-volume gear production. Today, a split even exists between U.S. and European bevel gear manufacturers over lapping versus grinding. The split is discussed on page 50.

The above guidelines have covered topics from grinding wheels' abrasive bond to the transition line between root and flank of face-milled gears, but this list isn't all-encompassing. A major reason so many unwritten guidelines are unwritten is because they're infinite in number and change with advances in materials, processes and equipment. Still, these guidelines should help gear engineers to skillfully grind their bevel gears.

A Split Happened on the Way to Reliable, Higher-Volume Gear Grinding

Dr. Hermann J. Stadtfeld

In the past 15–20 years, gear grinding has evolved into a stable, much-used process in industrial production, especially for making high-quality automotive transmission gears.

During this evolution, though, a split occurred between U.S. and European bevel gear manufacturers. Today, they live in one of two camps: the face-hobbing/lapping camp and the face-milling/grinding camp.

In the mid- to late 1980s, when gear grinding was introduced for industrial production, gear manufacturers already knew that gears ground to a very high AGMA quality level didn't have particularly low noise emission levels. The reason was the high spacing and flank form accuracy, which generated tooth meshing impacts during operation. These impacts occurred in angular intervals with very little variation.

The precise impact frequency created a so-called pure frequency tone that was emitted as part of the structure and air-borne noise. In the automotive industry, this pure tone noise could be easily distinguished from other sounds, such as noise from the wind, the engine and the tires. Moreover, the pure tone noise felt disturbing.

Gear manufacturers, however, were still attracted to grinding. Its beauty lay in its ability to achieve a low amount of variation in gear flank geometry within a production batch and in its ability to achieve such low amounts from batch to batch regardless of different levels of heat treat distortion resulting from steel variations.

For cylindrical gears, some manufacturers tried to reduce or eliminate the excitation from the tooth mesh impacts by grinding their gears, then honing at least one of the rolling members. But, this honing was difficult to justify economically.

Also, the manufacturers had another option: power honing. This highly economical finish honing process didn't require a previous grinding operation to take advantage of a gear's defined flank surface with minimum variation in production. The advantage of honing was the surface structure. It had a different orientation than the contact lines between pinion and gear, reducing the dominant mesh frequency peaks in the noise emission.

The problem with the pure tone frequency noise of strictly finish-ground gears was also reduced when flank modifications were developed. These modifications resulted in low motion error, even if the gears were displaced from their theoretical positions.

These developments helped spread the use of grinding for cylindrical gears, but not for bevel gears. There was still a five- to eight-year delay for the bevels.

In the early to mid-1990s, though, there were a number of advances in bevel gear manufacture. A breakthrough promised to spread the use of bevel gear grinding in Europe; the other advances led many bevel gear manufacturers in the United States in another direction.

The result was today's split between U.S. and European bevel gear manufacturers.

Until the early 1990s, most bevel gear grinding was done in the aircraft industry in single setups for convex and concave pinion flanks. But a breakthrough occurred in the completing cutting and grinding process for face-milled bevel gears.

The breakthrough resulted from a series of advances: improvements in the completing tooth design, especially in the tooth taper; a new generation of free-form machine tools with the higher stiffness and accuracy needed for a completing cutting and grinding process; new machining cycles that greatly reduced grinding time; and higher-order machine motions able to make flank form modifications that increased strength and reduced noise.

The breakthrough changed the economics of bevel gear grinding. The aircraft industry's single setups hadn't been affordable for automotive applications, but the new completing setups were. Bevel gear grinding became popular for mass-produced automotive gear sets.

Consequently, European bevel gear manufacturers turned away from lapping and toward grinding as a finishing process.

But grinding as the only and final hard finishing operation was a struggle for many years. It wasn't until the new millennium that all the elements of this technology worked as a stable process in an industrial production environment.

Also in the early to mid-'90s, though, many U.S. bevel gear manufacturers chose lapping over grinding. The reason: a number of innovations in face hobbing. The innovations convinced them that face hobbing had the most potential to be a stable, economical process for making bevel gears with high strength and low noise.

But face-hobbed gears couldn't be—and still can't be—ground with today's gear grinders. The flank lines' epicyclical shape makes grinding impossible. Lapping works efficiently, though. It also delivers better results when used on face-hobbed gears than it does on face-milled gears.

In the Far East, China, Japan and South Korea knew about the technological advances affecting grinding and lapping, but they change their bevel gear manufacturing processes only slowly. Most gear manufacturers in those Far East countries decided to retain the practice of five-cut face milling with lapping after heat treatment—the cutting-and-finishing combination that prevailed around the world before the 1980s.

Whether the U.S.-European split will continue is an open question. Just as advances led to the separation, other advances may lead one camp into the other or may lead both into a third camp. In all cases, though, a successful finishing process has to provide economical advantages and result in gears of high strength and low noise.

GEAR TECHNOLOGY
The Journal of Gear Manufacturing

Reader Profile:

Name:

Mark Garfien, President, U.S. Gear Corp., Chicago, IL

About your career in the gear industry:

"My dad started the company in 1962. I started working when I was 13, and I've been here ever since. I love making things."

On U.S. Gear's success:

"We serve many of the major OEMs in the transportation industry. We've also developed and market several of our own products. Over forty years of experience, great engineering and a company commitment to the best technology and equipment keeps us globally competitive. To be competitive you have to re-invent yourself."

Your technology and equipment?

"We have nearly 600 machines used in the production process. About 150 Gleasons, Mitsubishi CNC hobbers, a few dozen Mazak and Mori Seiki turning machines, as well as broaching, parts cleaning, chip spinning, and inspection equipment…to name a few. We also have a separate heat-treating division. Our budget for new equipment and technology averages several million a year."

In the future?

"More emphasis on robotics and automation, new cutting technologies and a continuing focus on new market opportunities."

Your thoughts on *Gear Technology* magazine:

"I've been reading *Gear Technology* over twenty years—since it started. It's the only magazine that I read every issue. And I frequently refer to back issues. I rely on it to get new ideas on making better gears, and it's one of my most important tools to find new equipment and suppliers. *Gear Technology* knows its readers and focuses on their needs."

Gear Technology. **The magazine gear industry buyers rely on.**

Measuring Pitch Diameter?

Robert E. Smith

Wait a minute, we don't measure pitch diameter. We're sometimes asked to measure it by customers, though, especially ones with older drawings or with gear orders outsourced by them. But measuring pitch diameter is a fallacy.

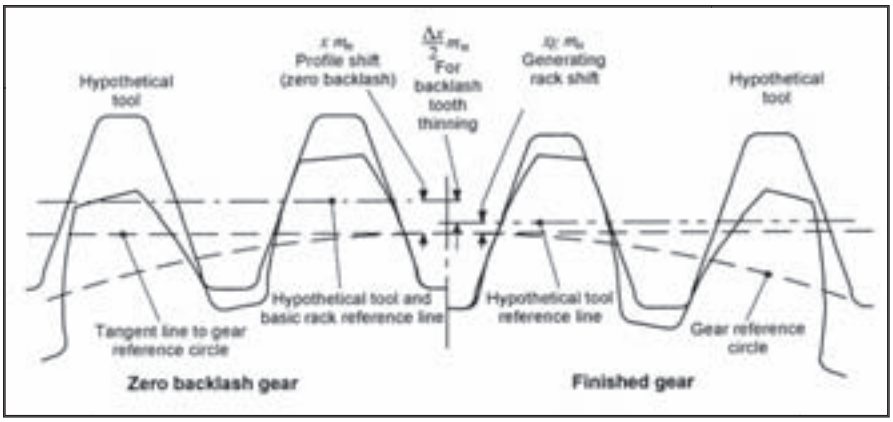


Figure 1—Hypothetical tool with zero backlash and finished gear. (Extracted from AGMA 913-A98 with AGMA permission.)

When a gear inspector takes a gage and uses its anvils or pins on diametrically opposed tooth spaces, what he's actually measuring is changes in tooth thickness.

Pitch diameter is the diameter of a pitch circle. The standard (reference) pitch circle is defined as: "The circle which intersects the involute at the point where the pressure angle is equal to the profile angle of the basic rack." The standard pitch diameter, D, is the diameter of this standard (reference) pitch circle and is obtained by $D = N/P_d$ or—in the metric system—by $D = Nm$, where N is the number of teeth, P_d is the standard transverse diametral pitch, and m is the module. See the American Gear Manufacturers Association standard ANSI/AGMA 1012-F90, *Gear Nomenclature Terms, Definitions, Symbols, and Abbreviations*.

Therefore, the standard pitch diameter is actually a reference dimension that doesn't change as long as the number of teeth and the diametral pitch or module don't change. **The standard pitch diameter shouldn't have a tolerance and isn't measured**.

When manufacturing a gear, the cutting or grinding tool may be fed in or out to make a size change. What this actually does is make the teeth thicker or thinner, at the standard pitch circle (gear reference circle). The standard pitch diameter, however, doesn't change. See Figure 1, which appears as Figure B.1 in AGMA 913-A98, *Method for Specifying the Geometry of Spur and Helical Gears*. The right and left halves of this drawing show the relative position of the cutting tool and the resulting change in tooth thickness. Note that the standard pitch circle (gear reference circle) doesn't change.

When measuring thickness change, it's common practice to measure the gear over pins or anvils. This is often called a measurement over wires (MOW). As the teeth get thinner, this measurement becomes smaller. Figure 2 shows an MOW operation. This illustration appears as Figure 6-1 in ANSI/AGMA 2002-B88, *Tooth Thickness Specification and Measurement*. Note that the caption says "Tooth thickness measurement over pins," not "Pitch diameter measurement." This is actually a diameter measurement (not pitch diameter) that has been calculated from the tooth thickness at the standard pitch diameter.

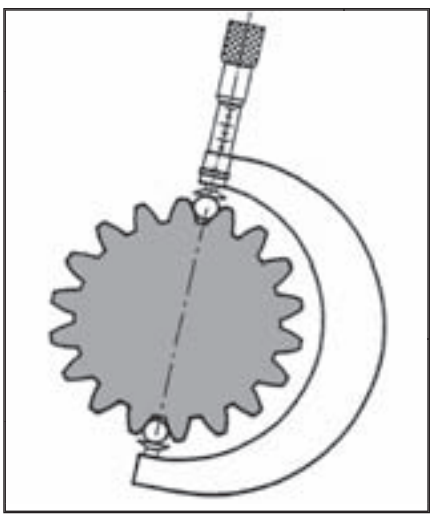


Figure 2—Tooth thickness measurement over pins. (Extracted from AGMA 2002-B88 with AGMA permission.)

It is highly recommended that specifiers of gear data and tolerances refer to the above AGMA standards and information sheets, as well as association documents on subjects such as gear accuracy. These are available from the AGMA, in Alexandria, VA, or through *www.agma.org*.

Robert E. Smith *is president of R.E. Smith & Co. Inc., a gear consultancy in Rochester, NY. A mechanical engineer, he has more than 55 years' experience in the gear industry. He chairs the AGMA Calibration Committee and is the ISO delegate for it and for the AGMA Inspection and Handbook Committee. Also, since 1991, he's been a* Gear Technology *technical editor.*

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UTS Teaches Plastic Gear Design & Manufacture

Plastic is different than metal. This obvious fact means plastic gears are different than metal gears. Yet there are gear engineers who don't learn the differences before sitting down at their workstations, starting their solid model programs and attempting to design plastic gears like they were metal ones.

Not surprisingly, the projects become problems.

"As consultants, we see a lot of those," says Philip Cooper, sales manager for Universal Technical Systems Inc. Located in Rockford, IL, UTS offers software and consulting services for plastic and metal gears.

It also offers a three-day course, "Plastic Gear Design & Manufacturing" to gear engineers who need to understand the different parameters used with plastic gears.

The class is concerned with solving practical problems involving plastic gears, such as high tooling and manufacturing costs, inadequate load carrying capacity, heat generation, performance variations (lot to lot), premature gear failure, and excessive noise.

The course covers the basics of plastic gearing, such as involute gear theory, important gear geometry factors, optimization of gear sets, mold design, materials characteristics, quality control and design methods for plastic spur, helical, crossed-axis, epicyclic and face gears.

It also covers minimum weight design strategies, gear size (including radial and axial shrinkage), geometry design, rating, producibility analysis, torsional analysis, lubrication, and profile analysis.

Students learn about a number of other advanced topics: robust design methods, application of non-standard proportions, accounting for extreme operating conditions, minimizing operational variation, predicting tooth deflection at mesh temperature, and applying proper tip relief.

They also learn about inspection and quality control issues. This includes a tour of Forest City Gear, a fine- to medium-pitch gear shop in nearby Roscoe, IL. Forest City makes plastic gears in low to moderate volumes. Moreover, as Cooper explains, Forest City has excellent analytical inspection facilities, and a number of companies send their plastic gears to it for inspection.

The course is taught by Jim Marsch, UTS' gear software product manager. Marsch has 35 years of experience in powertrain manufacturing, including 12 years at Allis-Chalmers, where he designed agricultural tractor powertrains, and 22 years at Harnischfeger Corp., where he designed powertrains for cranes.

Marsch's experience with plastic gears comes from his time with UTS, with consulting projects involving Xerox, Hewlett-Packard, Whirlpool, Maytag and other companies.

During the course, each student can discuss his particular gear problem with Marsch in one-on-one time on the class's last day or during previous evenings if necessary. Attendees interested in having one-on-one time should bring their prints and design problems.

The next class will be held Nov. 8–11 at the UTS operation in Rockford. The course is limited to 15 students, who share eight workstations. As long as space is available, people can register for the course as late as Nov. 1. The class costs $1,250 per person.

Besides lectures, the course offers hands-on exercises, including gear design problems for completing in class. The exercises are performed using Integrated Gear Software, UTS' suite of gear design software. Also, course discussion sometimes covers some point involving the use of TK Solver, the math modeling and programming application that is the calculation engine for Integrated Gear Software.

Consequently, UTS recommends students take its one-day TK Solver training course, given Nov. 7, the day before the plastic gears course starts. The class costs $295 per person.

The course covers various aspects of TK Solver, including solving equations, creating tables and plots, converting units, using built-in and user-defined functions, generating reports, and using TK Solver's library.

Students must reserve hotel rooms themselves, but UTS recommends these three hotels: Quality Suites of Rockford, Clock Tower Resort, and Candlewood Suites. Attendees making reservations at Candlewood can obtain a special room rate by mentioning UTS training.

Other details are available on the UTS website via a downloadable information packet.

For more information:
Philip Cooper
Universal Technical Systems Inc.
202 W. State Street, Suite 700
Rockford, IL 61101
Phone: (815) 963-2220
Fax: (815) 963–8884
E-mail: *sales@uts.com*

Measurement Training Comes To Your Facility


Ed Lawson

Whether you're involved in gear cutting, setup, engineering, management, metrology or quality control, keeping up with the changes in the world of standards is becoming more time-consuming every year. Gleason-M&M Precision hopes to demystify the pages of documentation from AGMA and ISO with its "Gear Measurement Course." The course, held periodically at the Gleason-M&M facility in Dayton, OH, is now being offered at customer locations as well. Usually, this option is more economical for companies with at least 3–4 participants.

Ed Lawson, director of metrology at Gleason-M&M, teaches most of the course. Lawson led the A2LA accreditation efforts for the company's gear lab. As chair of AGMA and ISO committees developing current generation gear accuracy and calibration standards, he developed the course after perceiving an industry-wide need for more detailed information.

"For decades, ISO and AGMA standards have been static. Then completely new documents have been written in the last few years. We're in a state of change at the moment, and everyone can stand to get up-to-speed," he says.

In addition to standards, course topics include:

- Gear Metrology: Terminology, Theory and Practice;
- Gear Accuracy Standards;
- Calibration & Uncertainty;
- Analysis of Involute & Helix Tests;
- Spline Metrology; and
- Gear Math.

Attendees take home a set of PowerPoint handouts, with the text conveying the primary message of a given slide. Lawson hopes the handouts become a book-like reference for the participants.

Content within these handouts varies according to the level of professional experience. The gear measurement course is designed in modules so that advanced topics can be presented to those directly involved with their gear labs. On the other hand, a two-hour executive overview format is popular with management personnel who don't have a need for the more detailed information.

Gleason-M&M tries to keep attendance at about 12 participants to encourage discussion. For companies who have larger numbers of employees interested in gear measurement, Gleason-M&M now offers off-site training at the customer's facility.

A session of the course is being held at Gleason-M&M on Sept. 19–21. Price per person is $495 per class. If all three clasess are taken at the same time, the cost is $1,465 per person. Customized training at Gleason-M&M is $4,895 per class, on site customer training is $6,400 per class. For information about the next session or to inquire about having the course presented at your location, contact Gleason-M&M directly.

For more information:
Gleason-M&M Precision Systems Corp.
300 Progress Rd.
Dayton, OH 45449
Phone: (937) 384-8977
E-mail: *info@mmprecision.com*
Internet: *www.mmprecision.com*

Gears2005 Focuses On Applications

The British Mechanical Power Transmission Association has sponsored gear seminars in the past, but Gears2005 Technical Awareness Seminar, slated for Nov. 17–18 at the Manchester Airport Moat House, promises to be more application-focused than in past years.

Dr. Harold Wright, technical secretariat for the event, recommended splitting the seminar into sessions specific to different industries. Specific focuses include transport (road, rail, and sea), automotive, design technology, process industries, environment and iron & steel production.

"We decided to move in a more application-oriented direction to make life exciting. We want people to leave saying, 'Wow, I didn't know I could sell to the Ministry of Defense.'"

The target audience of about 80 should consist of equipment users, sellers, manufacturers consultants and academics.

Day one is devoted to sessions on transport, design technology and process industries. The first session is chaired by Commander Tim Roberts of the Royal Navy, head of the

EVENTS

British Ministry of Defense Transmissions Group, and will cover developments in the leisure industry, public transport and military applications.

Each session includes a 20-minute PowerPoint presentation followed by discussion. The second session includes a presentation by Martin Halley, chief engineer for Xtrac Ltd., on the state of the automotive industry. BMW and other car designers and manufacturers will outline current trends in gearbox/drive conception and their applications to car performance.

The design technology session is chaired by Professor Tom Hyde, Hives professor of mechanical engineering and head of the school of mechanical, materials and manufacturing engineering at the University of Nottingham. New material development and post-millennium gear cutting tools will be discussed.

The final session of the first day analyzes process industries, recent legislation, modern bearing design and water treatment business opportunities and will be headed by Dr. Nigel Mainwaring, technical director at Russell Finex Ltd.

Day two includes focuses on the environment via sub-sea and wind turbines and is chaired by Bob Turner, supply chain manager at WindSupply. In-depth views on plant and equipment drive, transmission and lubrication needs will be presented in the iron & steel productoin session by Kevin Peacock, manager of engineering technology at Corus Northern Engineering Services.

The BMPTA offers six payment structures for attending Gears2005. The cost to attend all five sessions plus a dinner and overnight accommodations at the Manchester Airport Moat House in Manchester, U.K., is £360. For updated price information, visit the seminar's Internet site at *www.gears2005.org.uk.*

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October 10–13—Gear School 2005. Gleason Cutting Tools, Loves Park, IL. This course blends shop time and classroom study. Course content includes gear fundamentals, high speed steels and coatings, gear cutting and inspection. Participants tour the Gleason Cutting Tools plant and an offsite tour of a complete manufacturing facility is offered. $895. For more information, contact Gleason Cutting Tools by telephone at (815) 877-8900 or via the Internet at *www.gleason.com*.

October 25–27—Design2Part Show. Pheasant Run MegaCenter, St. Charles, IL. The largest contract manufacturing trade show includes exhibitors in the following categories: Machining, Cutting, Grinding, Finishing, Electronics, Assemblies, Components, Forming, Castings, Rubber, Plastics, Composites, Design, Prototyping, Testing, Fabrication, Tools, Dies, Molds, Patterns, and Fixtures. Registration is free, and the deadline is Oct. 18. For more information, contact Job Shop Shows on the Internet at *www.Design2PartShow.com*.

November 1–3—Mastering Shot Peening & Blast Cleaning. Renaissance Worthington Hotel, Dallas/Fort Worth, TX. Focuses on proper shot peening procedures, products and plotting a saturation curve, establishing machine parameters to ensure correct stress profiles and fatigue life, audit proceedings, prevention of machine malfunctions and more. $850 for one attendee, $750 each for two to three attendees, and $650 each for four or more attendees. For more information, contact *The Shot Peener* magazine by telephone at (800) 832-5656 or on the Internet at *www.shotpeener.com*.

November 5–11—ASME International Mechanical Engineering Congress and Expo. Walt Disney World Swan & Dolphin Hotel, Orlando, FL. Technical program includes engineering research presentations broken down into 600 sessions covering nearly 50 fields. Keynote speaker is Col. Mike Mullane, NASA shuttle astronaut and novelist who was recently inducted into the International Space Hall of Fame. Registration is accepted through Nov. 4 and ranges from $25–$700, depending on how much of the program a person registers for. One-day attendance ranges from $425–$625. For more information, visit ASME's website at *www.asmeconferences.org*.

November 13–16—FABTECH International & AWS Welding Show. McCormick Place, Chicago, IL. Designed for metal forming and fabricating personnel involved in cutting, lasers, fabrication/forming, stamping, tubes and pipes. For SME/FMA/AWS members, prices range from $160–$565, depending on the number of technical sessions attended. Non-member costs range from $180–$665. Registration is increased $25 after Oct. 21. For more information, contact the Society of Manufacturing Engineers on the Internet at *www.sme.org/fabtech* or call (800) 733-4763.

Stadtfeld Returning to Gleason

Hermann J. Stadtfeld

Hermann J. Stadtfeld will return to Gleason, effective Dec. 1, as vice president of bevel gear technology after a three-year absence.

According to the company's press release, Stadtfeld will be responsible for research and development related to advancing Gleason's bevel gear technology.

Stadtfeld was previously vice president of research and development for the company from 1994–2002. Prior to that, he was professor of machine and gear design at Rochester Institute of Technology and the Technical University Ilmenau, Germany. He has authored 15 books and more than 130 papers on bevel gearing.

Bourn & Koch Hires New VP

Hans Grass

Hans Grass was hired as vice president of engineering at Bourn & Koch.

According to the company's press release, Grass has more than 35 years' experience in gear machine tool technology. From 1972–1998, he worked for American Pfauter and served as vice president of engineering and manufacturing for 17 years. Most recently, he was president of Index Corp. and director of distributed products and special projects for Gleason's European affiliates.

Bourn & Koch also announced the breaking of ground on an expansion of its main facility in Rockford, IL. As an initial phase of the expansion, the company plans to add 40,000 square feet of warehouse space and 20,000 square feet of office space. Once the project is completed, Bourn & Koch's second Rockford location will be moved to its main facility on Kishwaukee Street.

New Location for Supreme Gear

Supreme Gear has moved from its Roseville, MI, location to a new facility. Please note the new billing/shipping address:

Supreme Gear Co.
17430 Malyn Blvd.
Fraser, MI 48026
Phone: (586) 294-7625
Fax: (586) 294-7648
Internet: *www.supremegear.com*

KISSsoft Opens U.S. Office

KISSsoft AG, the maker of gear and engineering software headquartered in Hombrechtikon, Switzerland, has opened a sales office in the Chicago area. The new office will be headed by Dan Kondritz, national sales manager for KISSsoft USA.

KISSsoft's products have been sold in Europe for more than 20 years, Kondritz says, and the company has more than 700 customers there. KISSsoft only recently began promoting its products in North America. The company already has dozens of U.S. customers, but there is plenty of room to grow, Kondritz adds: "It's an untapped market for us."

Potential users of KISSsoft's software include companies involved with the design of power transmission drives, especially those including gears, shafts, bearings, couplings and even mechanical details such as bolts.

Potential users of the software are in many industries, Kondritz says, including automotive, oil field, medical devices and plastic gearing. "Our customers range from the biggest companies of the world to independent, one-man consultant shops."

Opening the new office is already giving U.S. customers much better support and service, Kondritz says. "The customers feel that they can call up and get the kind of service they deserve." Kondritz has more than 20 years of experience selling and supporting technical software applications, including 14 years at gear software provider Universal Technical Systems of Rockford, IL.

Also, the U.S. presence will help customers stay up-to-date on the latest releases of the software, according to Dr. Ulrich Kissling, who originally wrote the software and who co-founded KISSsoft AG. "Our past history of twice-per-year software updates, along with our history of customer satisfaction, will impact the U.S. marketplace in a positive way," Kissling says.

For more information:
KISSsoft USA LLC
3719 N. Spring Grove Rd.
Johnsburg, IL 60050
Phone: (815) 363-8823
E-mail: *dan.kondritz@KISSsoft.com*
Internet: *www.KISSsoft.com*

Sunnen, RMC Form Strategic Alliance

Sunnen Products Co. and RMC Engine Rebuilding Equipment announced a strategic alliance in which Sunnen will market the full RMC line of engine building equipment, including RMC's CNC machining centers.

According to Sunnen's press release, the agreement has already gone into effect and RMC products are currently sold under the Sunnen/RMC brand. Sunnen's team of field engineers and service specialists now represent and support the RMC line. Products include CNC engine component machining centers, rod boring machines, line boring machines, cylindrical boring mills, surface mills, crankshaft grinding machines, belt resurfacers, piston pin presses and entry-level cylinder honing machines.

Mike Haughey, Sunnen's COO, says, "This alliance will help us achieve our goal of being the leading source of precision equipment for the automotive aftermarket and performance engine builders."

MMS Thermal Processing Orders Surface Combustion Technology

MMS Thermal Processing LLC has ordered Surface Combustion's VringCARB® technology and companion equipment for its vacuum processes in Davenport, IA.

According to the company's press release, MMS Thermal plans to build a new facility for the line, consisting of a multichamber system with a combination charge/vestibule/oil quench chamber with a quench oil heating system as well as two gas-fired vacuum carburizing chambers utilizing VringCARB vacuum carburizing technology, an electrically heated high temperature processing chamber and a 20-bar gas quench cell.

The installation will also include a variety of companion equipment, including multiple Uni-DRAW temper furnaces, a VacuDraw gas-fired vacuum temper furnace, a spray/dunk companion washer and an array of support equipment.

A complete metallurgical lab to document quality assurance and provide research assistance is available as well.

Broaching Machine Specialties Registered to ISO

Broaching Machine Specialties of Novi, MI, was awarded a certificate of registration for ISO 9001: 2000 conformance for the design, manufacture and servicing of new and remanufactured turnkey broaching systems.

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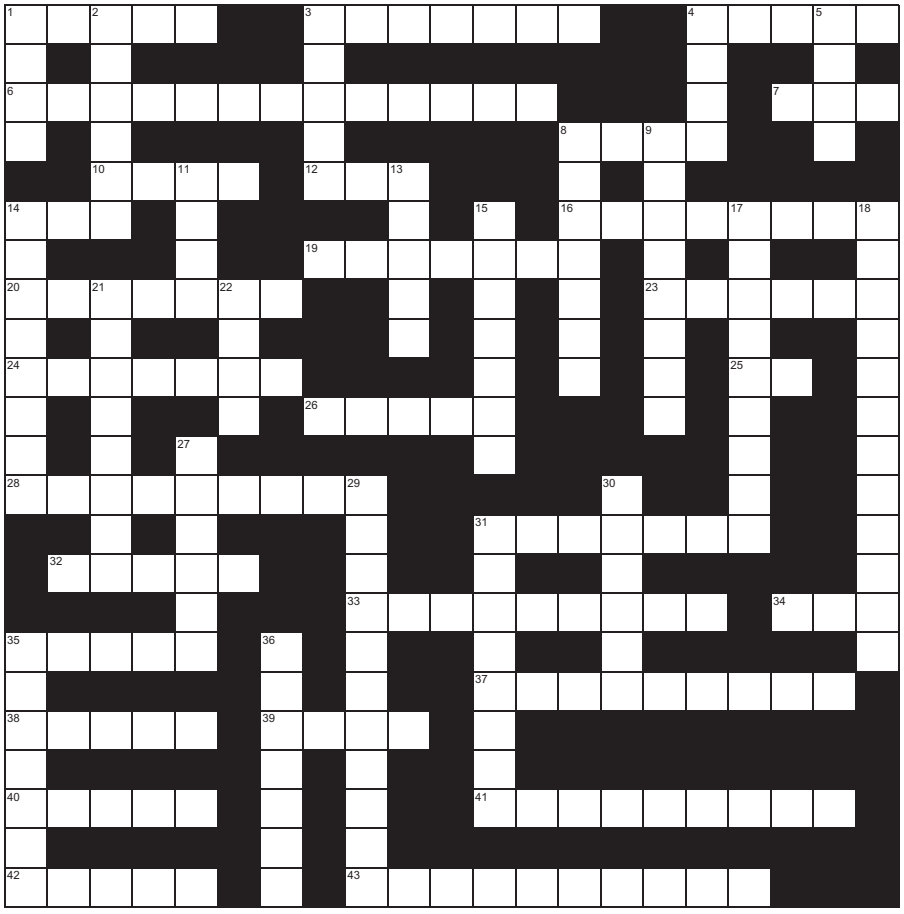

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19. _____-Wildhaber gears
20. Not Rockwell or Brinell
23. 3:1, 5:1, 10:1, etc.
24. Used to mate bevel gear pairs
25. Opposite of OD
26. A tooth modification, or a type of face gear
28. Planetary
31. A spur or helical generating process, not hobbing
32. Good plastic gear or stockings material
33. Metal cutting process; uses pullers
34. German standards-making body
35. Finish a gear with abrasive tool
37. Measurement, on a blueprint
38. Profile _____; causes noise
39. The center line about which a shaft rotates
40. A common gear material
41. How we measure pitch in America
42. The N in NVH
43. Creating an involute profile with a rack-shaped tool

DOWN

1. Sponsor of Gear Expo
2. Type of gage; either it fits, or it doesn't fit
3. Diametral
4. What happens to a gear that's poorly lubricated
5. The meeting of gear teeth in rotation
8. Finishing operation; uses a serrated cutter
9. When the involute form is cut away near the tooth root
11. Sponsor of the International Mechanical Engineering Congress & Exposition
13. Hypoid, spiral, Zerol are types of _____ gears
14. Euler's brainchild
15. Carbide rehobbing
17. Heat treating process involving ions
18. The state when gears are out of whack
21. Load-carrying _____
22. Bevel _____ gear; goes with pinion
27. Bevel gear with axis offset
29. Common method of achieving case depth
30. A toothed shaft
31. Norm
35. Inventor of the bevel gear planer
36. You need this to support your gear shaft

For the answers, visit: *www.geartechnology.com*

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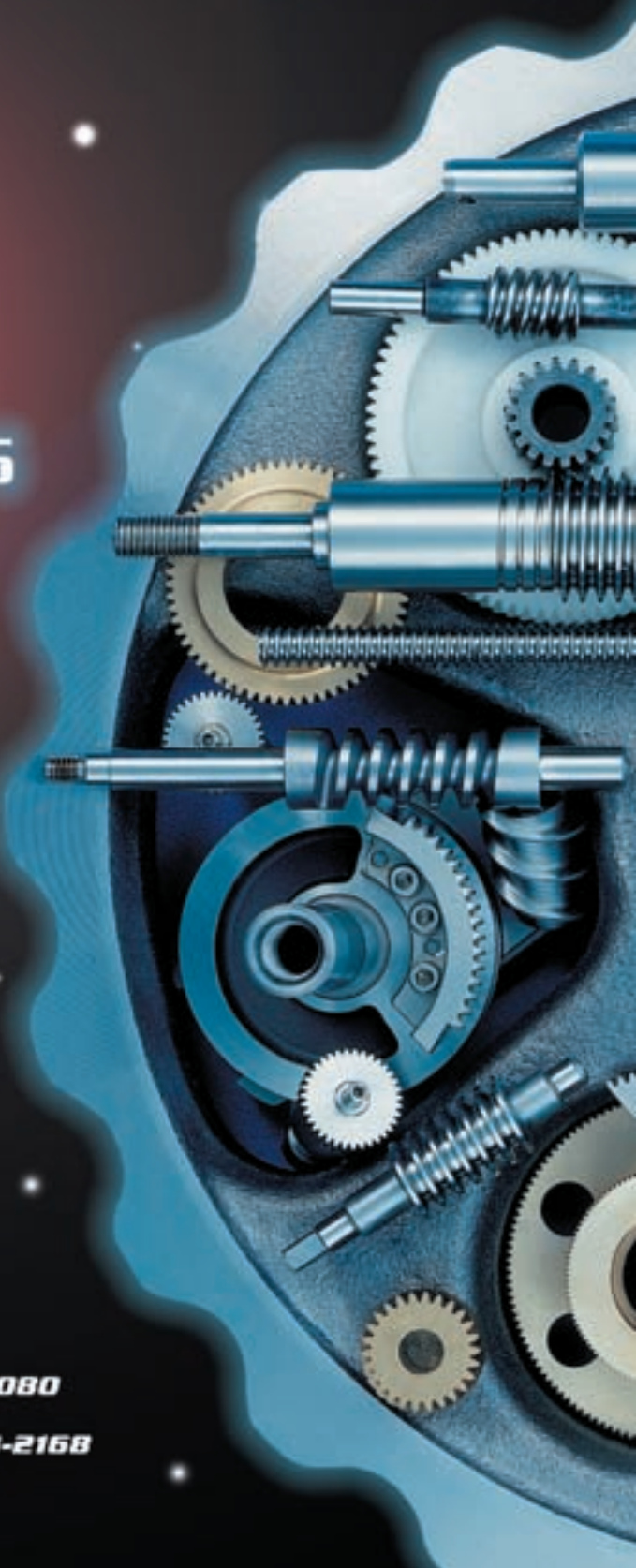

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▶ **Bourn & Koch** will demonstrate the 100H Series II CNC Hobbing Machine at the AGMA Gear Expo (Detroit, October 16-19, 2005). The 100H is a 6 axis hobbing machine featuring: NUM 1050H 6 Axes CNC control; full machine enclosure for wet or dry hobbing; pneumatic live center; automatic single – or double – cut hobbing on multiple gears or splines; automatic hob shifting; speed and feed change between cuts; crown hobbing, taper hobbing, radial or tangential feed worm gear cycles; and CNC hob swivel.

▶ **Bourn & Koch** will demonstrate multiple spline hobbing on a shaft with two splines with different numbers of teeth; with two hobs with different numbers of starts; and dry cutting in a single clamping.

◀ **Star Cutter** will demonstrate the Star Model PTG-1 at the AGMA Gear Expo (Detroit, October 16-19, 2005). The PTG-1 is a 5-axis grinding machine featuring: NUM Axium CNC control; NUMROTO plus software; integrated automatic part probing; automatic wheel probing; and on-board wheel dressing. The PTG-1 sharpens straight and spiral gash hobs, through-hole and shank type, up to 8" in diameter and 10" in length. The resharpening quality meets the AGMA “AAA” standards, with unsurpassed surface finish through the use of super abrasives.

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