GEAR TECHNOLOGY

MAY/JUNE 2006
The Journal of Gear Manufacturing
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HIGH PERFORMANCE

- Racing Gears
- Marine Gears

TECHNICAL ARTICLES

- Surface Roughness and Gear Pitting
- Dry Bevel Cutting

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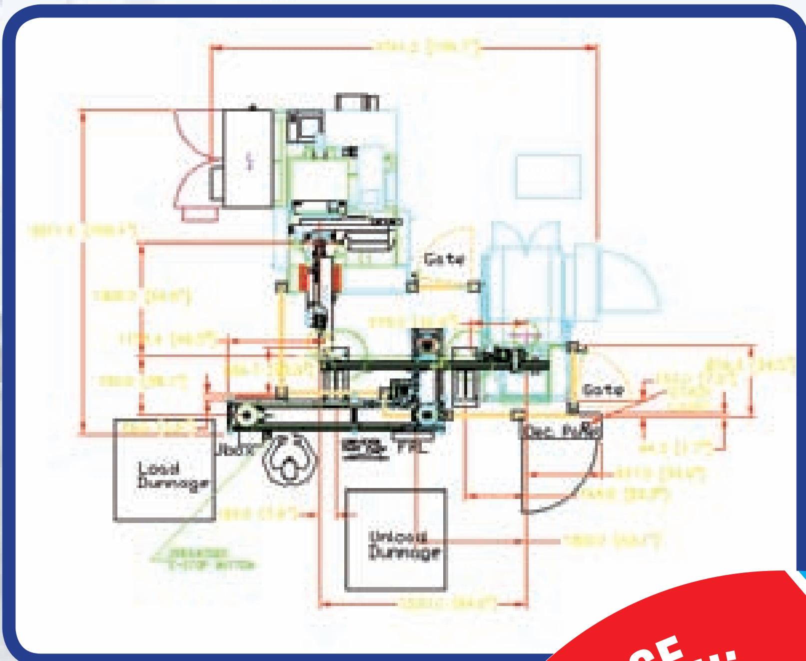

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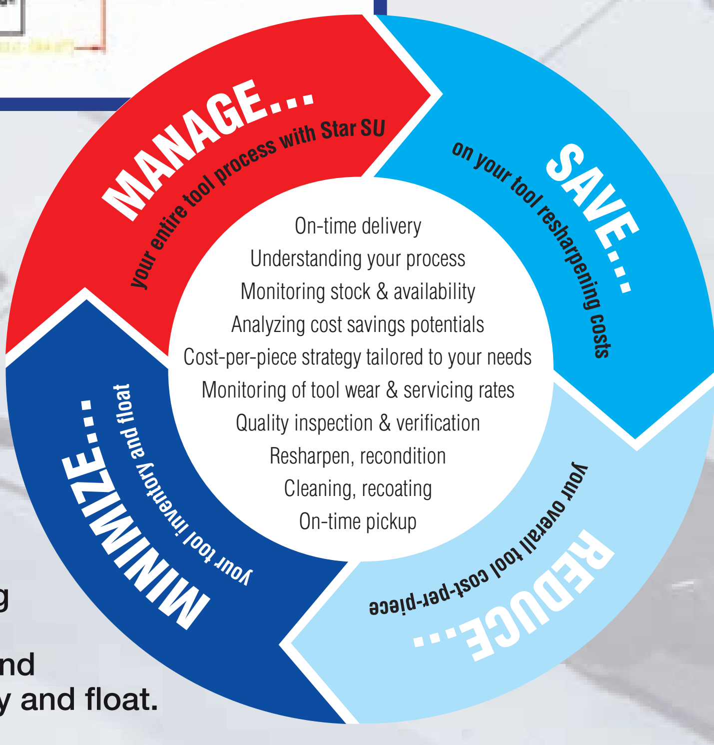

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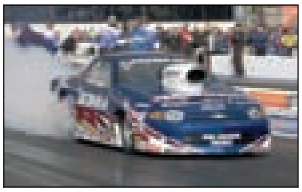
MAY/JUNE 2006 *The Journal of Gear Manufacturing*

HIGH PERFORMANCE GEARS



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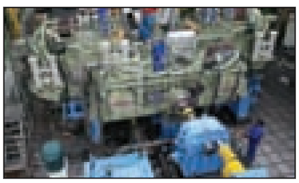

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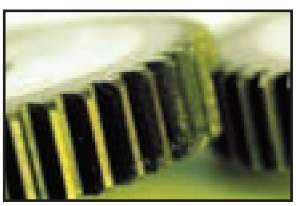
TECHNICAL ARTICLES



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DEPARTMENTS

9

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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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A Wellspring of Opinion

For more than 22 years, I've been dropping rocks down the well of the gear industry's public opinion. Most every issue, I drop another rock. Sometimes I think I hear a faint splash, but most times I just wait.

However, my editorial in the November/December 2005 issue ("Is Gear Expo Worth It?") splashed so hard and so loud, it seems like the water is ready to gush over the top. Admittedly, the rock was a bit larger than normal, and it caused a lot of reaction—from AGMA board members as well as members of the gear manufacturing community.

Having recently returned from the AGMA annual meeting, I can confirm that the level has, indeed, risen—substantially. The rock—my editorial—was just the catalyst in getting people thinking and talking about the role of both the AGMA and the Gear Expo. In the two days I was in Tucson for the annual meeting, members told me "you're asking the questions that need to be asked," "you hit the nail on the head," and "keep up the good work."

While it's nice to be congratulated for getting things rolling, what's needed is a continued open discussion. Last week I met with several members of the AGMA board, and I'm confident after that meeting that the association's leadership is asking the right questions, not just about Gear Expo, but also about the role of the association in the success of its members.

But the officers and board of the AGMA can't help you unless you're willing to help yourselves. What they need most is the input of the gear manufacturing community. In this forum, we've already discussed Gear Expo, so I'd like to turn the discussion toward the association itself. What I'd like to know is how well the AGMA is meeting your needs.

The AGMA states the following mission on its website: "To help members compete more effectively in today's global marketplace."

Also, in 2004, the AGMA announced a new vision and strategic objectives. They are:

1.) To continue AGMA's leadership role in the development of domestic and international technical standards.

2.) To help members compete/benefit in global growth.

3.) To stimulate interest in careers in gears and gear/coupling-related products.

4.) To provide for the long-term viability of the AGMA membership through leadership development.

5.) To communicate important industry information in the most effective/efficient manner to get the desired positive response.

6.) To provide value to the organization and to meet and grow revenue through membership growth and retention.

By introducing this subject, I'm not trying to pass judgment one way or another, nor am I trying to suggest that there are huge problems here. What I want is your opinions about the AGMA, its mission, its goals and how well the association is achieving them.

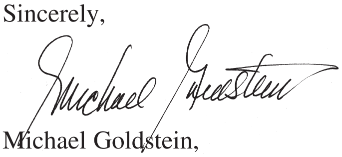
In today's competitive environment, all associations—not just the AGMA—are having to evaluate (or re-evaluate) the value they bring to members. The camaraderie and chance to meet your competition and suppliers are a given. But I believe all associations today need to have a greater impact on the success of their members.

What is the AGMA doing that helps ensure your success? What is it not doing that it should be? How can the AGMA better serve its members?

I'm sure that many of you have opinions about what you'd like to see the AGMA accomplish over the coming years. Don't keep those opinions to yourself. And if you believe the association is doing everything it possibly can for its members, I'd like to hear from you, too. If your company is not an AGMA member, I'd like to hear why it does not fit in with your vision of your company's success.

My job is to drop the stone in the well and see if anything splashes. Now it's your turn, so splash away! Send your replies to *publisher@geartechnology.com*.

Sincerely,



Michael Goldstein,
Publisher & Editor-in-Chief

Gleason's Genesis 130SV Gear Shaving Machine

The 130SV shaving machine from Gleason is the newest of the company's Genesis family of gear production equipment.

Introduced to the gear community in February, the 130SV is designed for the fine finishing of soft spur and helical gears with outside diameters of up to 130 mm. Gleason reports plans to ship its first order to a large German automotive manufacturer in May, and two more machines will ship to a Korean automotive manufacturer in July.

According to the company's press release, the 130SV's shaving head has been simplified by controlling the Y-axis with Gleason's spheric software to provide the linear motion for precise correction. Therefore, plunge shaving and diagonal shaving can be performed on the same machine.

"The biggest difference from the previous machines is that for doing corrections on the tooth flank, we don't use a so-called cradle anymore but added a tangential axis," says Johann Mall, engineering director at Gleason Hurth. "This allows us to use the same machine configuration for several technologies like hobbing, shaving and threaded wheel grinding. The biggest advantage to the customer is that he gets a fully capable machine to do all the shaving technologies, like plunge and diagonal shaving."

Mall says that although previous shaving machine models allowed plunge and diagonal shaving, the mechanics were different and more expensive. When purchasing the machine, a customer had to know its uses ahead of time. A diagonal shaving machine needed more axes than a plunge shaving machine.

A new mechanical, cam-driven double gripper loader is fully integrated into the machine and can perform the load/unload sequence in approximately four seconds. It can also accommodate disk or shaft-type gears and readily integrate with common parts handling systems, including palletized, gantry, blue steel and robot systems for maximum throughput.

"Although there are also loaders on other machines in the market which work near to this speed, they are all dedicated, inflexible and very special. The loader for the Genesis machine is universal," says Mall.

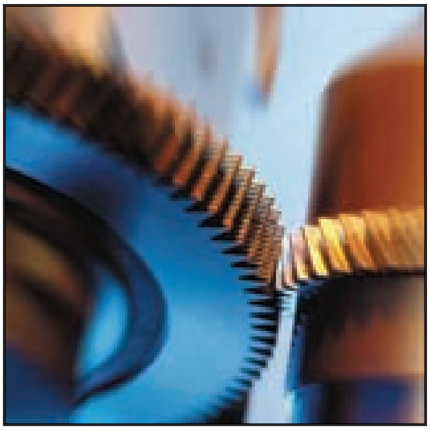
Special features include a footprint of seven square meters (73 square feet) including all hydraulics, lubrication, chip removal, coolant and pneumatic systems; an easy access service module to consolidate hydraulics, lubrication and pneumatics into one location; a single piece, mineral cast polymer composite base/frame; a new shaving head that operates without a cradle; a stock dividing system mounted on the shaving head; a magnetic chip filter/conveyor that can relocate to meet different cell/system floor space requirements; direct-drive spindles; advanced Siemens Sinumerik 840D controls with Gleason Spheric shaving technology software, Windows-based user friendly software and PC front end; optional on-board chamfering, deburring and burnishing capability; and a common design with the other Genesis machines. The fully self-contained machine can be moved as a single unit.

A proprietary Power Shaving option is available, enabling the 130SV to be equipped like the ZS series of Gleason shaving machines. With this option, both the work spindle and shaving cutter are driven such that the workpiece is automatically meshed on the fly with the continuously rotating cutter. In addition, the shaving cutter applies a torque on the workpiece during the shaving cycle.

Gleason estimates a 20% time savings and recommends this add-on for especially small parts, such as pinions.

Process data calculation software is offered as an option to be run on the machine controls. After feeding in the gear data, the program suggests appropriate shaving parameters. For its software development, Gleason employed its knowledge from the design of the Spheric Honing machines, which have similar kinematics to the 130SV. Therefore, the linear axis moves similarly to the swiveling cradle axis' movements.

The Genesis stock dividing system mounts to the shaving head and adds one



second to the loading time, eliminating the need for adjustment from part to part. During loading, the stock divider sensor swivels in position between the cutter and part, finds the tooth gap and reverts back to its parking position as cutter and part are brought into mesh. Movement is conducted by the NC axis, eliminating the need for manual operations. The CNC machine uses part data for automatic positioning.

Gear producers should find this machine very competitive, says Mall. As a vertical machine, it can integrate into interlinked production lines and does not require additional part carriers or swiveling units.

"There is no comparable machine concept on the market. Customer interest has been very high," Mall says. "The best proof for the success of this is to have three machines out in the market five months after quotation. We've appointed a group of application and design engineers to answer all our requests for quotes. Several orders are expected within the next two months."

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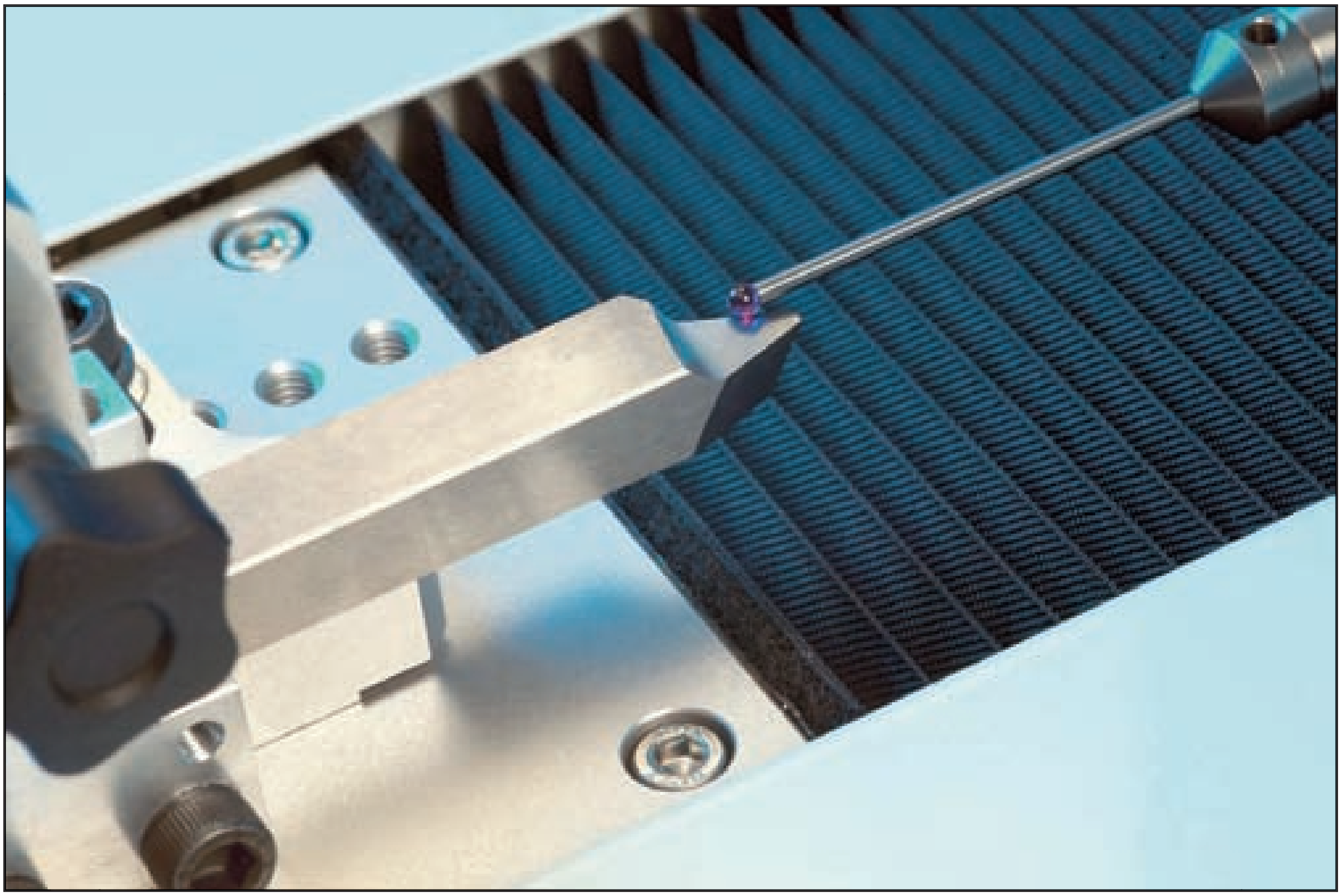
New Checker Scan-Measures Stick Blades with Ruby-Tipped Probes



Meant for use on the shop floor, the Oerlikon BC 10 scan-measures stick blades via ruby-tipped probes, collecting large amounts of information—like 200 data points on a blade's curved cutting edge—to ensure it properly cuts spiral bevel and hypoid gears.

This month, German automakers will receive the first three units of Klingelnberg's new automated blade checker designed for the shop floor. The Oerlikon BC 10 uses ruby-tipped probes to perform scanning measurement of stick blades, with direct measurement of a blade's shaft, rake face and relief face and indirect measurement of its edge for cutting spiral bevel and hypoid gears.

Introduced at EMO '05, the BC 10 measures the rake face, relief face, cutting edge, tip radius and shaft of stick blades with heights of 5–36 mm, widths of 9–36 mm and lengths of as much as 100 mm for use in cutter heads 2–16" in diameter.

The checker performs topographical, 3-D measurement of rake and relief faces to make certain they're in their proper places relative to the blade's shaft. The BC 10 also compares the faces' forms and positions relative to each other, uses its software to calculate the cutting edge's position and form, then compares the edge's actual form to its ideal form. Also, the checker can create a visual 3-D representation of the edge, with tolerances shown as double lines around the edge. If there are deviations, the BC 10 can use its software to calculate corrections for transfer to a stick blade grinding machine.

Roger Kirsch, head of Klingelnberg's division in Ettlingen, Germany, says measurement of both rake and relief faces results in more precise measurement of the cutting edge. "Measurement of a sharp edge is critical, especially when the stick blade is made of carbide," he adds. "The edge is very sharp."

In fact, the BC 10 doesn't directly measure the cutting edge because its sharpness could create wear on the probe. "We do not touch the edge," Kirsch says. "We calculate it. We make an interpolation."

The BC 10's use of scanning measurement allows it to collect a large amount of information in little time. For example, the checker can collect 200 data points on a curved edge of a blade 10 mm long—more if the blade is longer. "The measurement time is about 90 seconds," Kirsch says.

The scanning is done via a series of machine measuring motions the BC 10 creates in its measuring program from neutral data in its database.

Besides the faces and cutting edge, the BC 10 can measure the shaft for width, thickness, parallelism, straightness and angularity, checking for inaccuracies, like concaveness, twist and waviness. These measurements ensure a stick blade's shaft fits well in a cutter head's presized slot. Such slots usually have tolerances of only a few microns. "The stick has to be manufactured very precisely," Kirsch says.

Deviations can be displayed numerically with tolerances or on the BC 10 monitor in a 3-D format with selectable scale.

The BC 10 measures stick blades via three probes. Each probe's size is based on its ruby diameter. The diameters are 1.5 mm, 3 mm and 5 mm. According to Kirsch, the three diameters cover the size range of stick blades. The smallest 20% of blades can be measured with the 1.5-mm ruby, the largest 20% with the 5-mm ruby and the 60% in between with the 3-mm ruby.

The BC 10 can check stick blades for all Klingelnberg cutter head systems: Arcon®, FN®, FS®, FSS®, Spiron®, and Twin Blade by Klingelnberg®. It also can check them for Gleason Corp.'s RSR® system.

The BC 10 was designed for use on the shop floor, next to the blade grinder to minimize distance—and therefore time—needed to grind and check blades. "You don't need a special measuring area," Kirsch says.

Klingelnberg's Roger Kirsch says about inspection of blades: "You assure the stick blades will meet your quality requirement. If you don't measure them, you don't know it."

A stand-alone workcenter, the BC 10 consists of measuring machine, controller, personal computer with the Windows XP operating system, keyboard, mouse, display screen and printer. The computer includes a CD-DVD burner for data transfer. There's also an oil-proof drawer for storing accessories and calibration tools.

As for its use, the BC 10's setup time depends on a stick blade's size, but the checker was designed to keep that time short.

Kirsch estimates the time at about 60 seconds if a different probe is required. In that case, an operator would need perhaps 30 seconds to unscrew one probe, screw in its replacement and use a ceramic ball to calibrate the new probe to an error margin of less than 1 micron. The stick blade itself is held in its fixture by two pressure springs. The operator would then need maybe 30 more seconds to access the BC 10's database, choose the right theoretical data for comparison, select the blade features to be measured and start the measurement program.

Also, during calibration, the BC 10 automatically runs a wear program to show the probe's condition. Moreover, an operator only needs to roughly clean a blade before checking it. The BC 10's measuring force negates a blade's oiliness, Kirsch says.

Measurement data can be stored in the BC 10's computer, on compact disc or via a gear manufacturer's KIMoS network. This network is created using Klingelnberg software that connects gear-manufacturing machines for computerized, closed-loop production of gears.

Electronic storage of data can speed production and reduce the possibility of error by eliminating repeated manual entry of necessary contour data. Kirsch says a stick blade might have 45 different input values for its contour data, with each value having a minimum of five characters: one to the left of the decimal point, four to the right.

Manually entering the data: "It takes some time," Kirsch says.

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DSM's Stanyl Precison Gears Help Disabled Patients Drive Independently

Precision gears made out of Stanyl, a high-performance polyamide 46 (PA46) resin from DSM Engineering Plastics, help keeps the Joyster moving. The Joyster, demonstrated at Hannover Messe 2006, is a new joystick-like system designed to enable drivers with arm muscle disabilities to drive by themselves.

Developed at the Bern University for Applied Sciences HTI-Biel, the new product augments a car's conventional steering wheel with a pair of joysticks mounted on either side of the wheel. These are electronically coupled to the vehicle's steering mechanism.

The Stanyl gears were fabricated by Mikron Plastics Technologies and are utilized at the joystick and the vehicle's steering shaft.

"The gear sets in both the motor drives and joysticks are zero backlash," says Hans Wennekes, Stanyl's business development manager. "That's the only way the joystick can deliver absolute precision for encoding, and it enables the tightest possible steering control without wander. Technical collaboration between DSM, Mikron and HTI Biel—or, if you will, the material maker, the gear cutter and the design team—was the only way such a precise mechanism could have been developed."

The joystick gears must precisely translate small movements to programmable encoder circuitry. The movement required is small, and the touch must remain light. The steering shaft gears, on the other hand, must apply strong forces to the vehicle's steering system.

A critical aspect of the design is feedback to the driver. Small motors in the joystick mechanism provide resistance that is sent back to the driver through the joystick gears. This resistance signals the severity of the turn and transmits the road feel of bumps and surfaces to the driver. This enables the system to give drivers the same kinds of tactile information a driver would sense while using a conventional steering wheel. The degree of feedback can be programmed for a given driver's muscle capabilities.

For more information:
DSM Engineering Plastics – Americas
2267 W. Mill Rd.
Evansville, IN 47732-333
Phone: (812) 435-7539
Internet: ***www.dsm.com***

Alpha Gear's Gearmotor Offers Significant Reduction in Inertia

The TPM from alpha gear drives Inc. is a geared motor containing a rotary actuator based on an AC servo motor and high precision planetary gearing.

The rotary actuator is characterized by a high dynamic response. With an overall length of 186 mm, the TPM 50 builds up a maximum output torque of 500 Nm at a mass moment of inertia of 2.9 $kgcm^2$.

This high dynamic response is due to the 50% reduction in the actuator's mass moment of inertia, according to the company's press release. This is because the rotor inertia is reduced by the low-speed motor and mass inertia on the gear side is reduced by about 90% since the gear pinion is directly integrated into the motor shaft, eliminating the need for a clutch.

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RESCUE ONE

Faro's Power Gage Capable of CAD-to-Part Analysis

The PowerGage from Faro Technologies is a new measurement device that adds CAD-to-Part analysis capability to its Faro gage product line.

The gage runs exclusively on a version of Delcam's PowerInspect software. According to the company's press release, the software is used by a majority of the on-machine inspection market.

When equipped with both the PowerInspect's programmer and play-only modules, users can create inspection programs/routines for anyone in the facility, import all major industry CAD formats, perform surface inspection against master CAD files, receive on-screen instructions including images and video, access a full suite of geometric inspection tools, automatically optimize the viewing angle of parts measured and save all inspection data and run customized reports.

Instead of taking the part to a fixed CMM in a climate-controlled room, users can mount the PowerGage directly to where the part is being made. As the user traces the arm's tip over the part's entire surface, the system's laptop verifies all the 3-D measurements against the original CAD file to see if the part was made correctly. If it was not, the digital blueprint pinpoints where it needs to be corrected.

For more information:
Faro Technologies Inc.
125 Technology Park
Lake Mary, FL 32746
Phone: (800) 736-0234
Internet: ***www.faro.com***

Marposs' Gage Head Designed for Internal Grinders

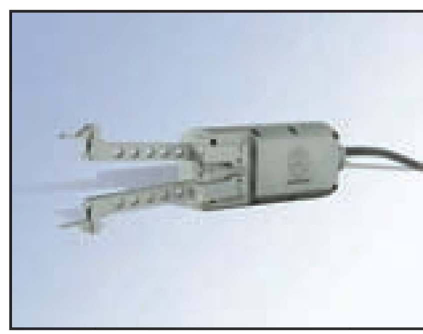
The Thruvar from Marposs is a through-the-spindle gage for in-process measurement application on internal grinding machines and was introduced at Eastec.

According to the company's press release, the gage reduces downtime and eliminates the special tools that are sometimes necessary. Using an automatic setup feature, it is possible to reset the gage without operator intervention for a size variation up to 25 mm. Similar gages that use a special tool for manual adjustment may require about five minutes of changeover.

The gage is mounted inside the grinder's workhead spindle, and it measures workpiece inner diameter as the part is ground. Signals are sent from the gage to the monitor, providing outputs to the grinder control based on real-time measurement of the workpiece size. Outputs are used to control the wheelslide infeed for precise size and finish consistency.

Access to the fingers and contact is obtained without removing the gage from the spindle. Finger retraction is pneumatically controlled, and the retraction distance is selectable in three increments: 92 mm, 6 mm and 10 mm.

One set of fingers and contacts covers a 25 mm diameter size range with no downtime for changeover.



The gage is available in two sizes—the Model 50 with a 50 mm (1.97") body diameter for measuring inner diameters from 50 mm (1.97") up to 170 mm (6.69") and Model 70 with a 70 mm (2.75") body diameter for measuring inner diameters from 75 mm (2.95") up to 220 mm (8.66").

Marposs says the gage head complements the Unimar line of in-process gages for OD measurements. Therefore, it is possible to cover the application range for grinding components, including width or face grinding, OD grinding and ID grinding using two gage head styles.

For more information:
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Alpha Gear's New Gearhead Follows Trends in High Reduction Torque and Ratios

The new SPK+ right-angle gearhead is designed for applications with high reduction ratios and torques.

This product combines features of the company's planetary gearhead and right-angle gearhead. The SPK+ is also lighter than conventional bevel planetary gearboxes, the company says.

The SPK+ achieves higher reduction ratios than other right-angle gearheads, according to the company's press release. In particular, handling tasks, which often necessitate a reduction ratio of i = 30 to i = 40, are best suited for this gearhead.

In the output stage, the gearhead assures improved positioning accuracy and synchronism due to the fatigue-resisting design of the alpha hypoid and planetary gearsets.

A smooth-running hypoid gearset on the input and optimized helical teeth in the SPK+ facilitate a lower noise level. On the output side, noise emissions have also been reduced.

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Kapp to Highlight Gear Center and Profile Grinder at IMTS

The Kapp Group plans to display its Kapp KX300P and Niles ZE400S machines at this year's IMTS show.

The KX300P utilizes either dressable or non-dressable worms and form wheels and features on-board inspection and integrated balancing. A vertically-oriented workpiece spindle allows for efficient integration of automatic loading systems.

Kapp also plans to showcase an integrated ring loader system for the machine as well. According to the company's press release, the ring loader reduces part-to-part exchange time over gantry or robot-type automation systems. With the optional integrated ring loader, the KX300P can achieve cycle-stop to cycle-start times of approximately four seconds.

The Niles ZE 400S features an additional 150 mm of stroke length that improves internal capabilities. The machine is designed for finishing internal and external spur and helical gears using either dressable or non-dressable form wheels. Other features include a cast-iron machine base for all machine components, torque table and common CNC dressing device with a high precision spindle for both internal and external applications. The machine will be displayed with an optional dressable internal attachment and features integrated inspection and GMG (grind-measure-grind) technology. Niles ZE grinders range from 400–1,200 mm capacity and have Siemens Sinumerik 840D controls.

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Send your product news to Robin Wright, Assistant Editor, *Gear Technology*, P.O. Box 1426, Elk Grove Village, IL 60007, USA Fax (847) 437-6618 or e-mail *robin@geartechnology.com*

CONSTANT INNOVATION

Keeps Suppliers to Race Car Manufacturers

EARNING TROPHIES

Robin Wright, Assistant Editor

When the fans start screaming at the Daytona 500, they're cheering for Jeff Gordon. Only the die-hard racing fan can appreciate the gearing and engineering that goes into each race car. The technical aspects, though, keep many gear companies working at a breakneck pace.

The benchmark for difficult gearing markets has always been aerospace, says Richard Wolfe, COO of EMCO Gears. Although manufacturing racing gears doesn't require the same kind of certifications as in aerospace applications, part performance can make or break a career when you're working for a race team. This market offers other unique challenges, Wolfe says, because users push gears to the limit and only have one opportunity to showcase maximum performance.

EMCO Gears supplies transmissions, gears, engine gears, rear axles, bell housings and steering racks to racers from the Indy Racing League, NASCAR Series, and the Rolex Grand American Road Racing Series, which includes the EMCO Gears Classic, sponsored by Wolfe's company. The Classic takes place at the Mid-Ohio Sports Car Course on June 23–25.

"It's the original old boys' network," he says. "Racing is interesting because if you make something that works, it's in [the race team's] interest to keep it a secret. If it doesn't work, everyone knows right away."

Trade secrets are a reality for every supplier and manufacturer in this industry, and Wolfe says the barriers for entry in the racing gears market are especially high.

"In this business, you need a reputation for quality. There aren't very many good ways to test a gear in racing applications other than putting it on the track, which is expensive to do. So your reputation must precede you," he explains.

Innovation is Key

One way to build that reputation is to innovate, Wolfe says.

An illustration of this occurred a couple of years ago when EMCO started noticing all of its competitor's rubber boots failing during the Rolex 24 Hour Race at Daytona. Rubber boots protect a car's C-V joints by keeping grease in and dirt and debris out. EMCO's new axle kits, released on the market in 2005, include a rubber boot constructed with the same materials used in the most demanding endurance racing in the world, and they also have improved geometry that's customized for the dynamic movement the boot sees, says Wolfe.

"We set out to engineer something with a part that runs cooler. Our rubber boot is specially designed for endurance racing, and we've manufactured enough for about 40 car sets in the Grand Am, International Racing League (IRL), and for cars in the GT class of the Grand American," he says.

The customer reaction has been positive, according to Wolfe. Orders are coming in at a brisk pace and, best of all, none have been returned to the company.

Experimenting with Coatings

Another area where innovation is important is in the use of new materials, processes and coatings.

Wolfe says the company favors diamond-like coatings, which are very hard surface coatings used mostly for anti-friction. EMCO has also added a new process, which chemically finishes the surface to remove peaks and help bring the parts' R_a down to 4 micro-inches. Shot peening uses balls to hit the surface of a gear and create craters. Wolfe says the craters are sometimes referred to as negative space, which ordinarily is not a problem because oil can collect in the craters to aid lubrication. Problems arise when material projects above the surface. This leaves open the possibility that the metal can puncture the oil film. EMCO's Superficial Chemical Obliterations for –RSK Enhancement (otherwise known as SCO-RE) smooths the part so that any imperfections exist below the surface only.

To avoid other problems with the finishing, gear manufacturers and race teams frequently send their gears to coating specialists.

Balzers Inc. usually applies one or two of 15 possible coatings to race car gears as well as various transmissions, cleans them and often applies pre- or post-treatments. One of the most popular pre-treatments is a proprietary enhancement of surface after mechanical or chemical microfinishing.

Torsten Doering, product manager at Balzers, says coating gears for better lubrication is essential to lower the coefficient of friction, increase efficiency and reduce wear, especially in racing applications.

"One of the most common modes of failure is fatigue. With coatings, it's possible to apply higher loads, or you can run gears for longer periods of time before fatigue failure will occur," he says. "A lower coefficient of friction and the run-in characteristics of our coatings improve gear performance. With dry conditions, the coefficient of friction of steel on steel is 0.7. Our coatings are consistently between 0.1 and 0.2. If there are lubricants in the mix, your coefficient is much lower."

Reduced oil in transmissions is another issue that coating engineers have to deal with, says Doering. In order to meet the weight requirements dictated by every racing series, it's commonplace to reduce the oil in transmissions. Since everything is moved through the oil, less oil means less resistance. Therefore, they need the coatings to provide enough lubrication.

Balzers' newest series aims to toughen their coatings. The BALINIT Star carbon coatings have a chromium nitride layer underneath to distribute load better into substrate material. PVD coatings are much harder than the bulk material and very thin; point loads and softer substrates pose a challenge to this system. BALINIT Star coatings have a reliable supporting layer ensuring that the superposed carbon layer can have its proper effect, even on somewhat softer substrates such as those in the 40HRC range. This is meant to increase the load bearing capacity, allow for higher impact load and increase the operating reliability of components.

Jerry Hemmingson, owner of Jerico Racing Transmissions, has first-hand experience with what works in coatings as well as other tricks of the trade in racing transmissions, assembly and software. The 62-year-old had been racing cars his entire life and, around 1970, noticed a trend—the transmissions always broke. He started making transmissions for himself and today supplies to racing teams worldwide, including Joe Gibbs, Ralph James and most of the Winston Cup teams.

Jerico makes two-, three-, four-, five- and six-speed transmissions for numerous drag races with a heavy emphasis on stock, super stock, and elimination competitions. Hemmingson says that cars have been known to hit more than 400 mph in land speed with his transmissions and that, at one point, all the NASCAR teams used Jerico transmissions.

Early Trials Led to Modular Transmissions

"A lot of people have copied what we do. When we first started, our shifting mechanism was extremely forgiving and shifted like butter; it was so easy. We were one of the first to use straight cut gears when most everyone else was using stock transmission," Hemmingson says. "We were also one of the first to make the whole transmission modular where it comes apart and you can change individual gear sets instead of being stuck with standard transmissions."

Although Jerico does not manufacture the castings of cases or perform heat treating processes in-house, it does a lot within its 50,000-square-foot facility in Concord, NC. Any sawing, machining of gear blanks, broaching, cutting of gear blanks, shaft grinding after heat treating and machining of cases and shifting mechanisms occurs at Jerico. Hemmingson says they're the only company to manufacture everything right from raw material.

There's a downside to this, he says, and that's the increasing cost of obtaining the raw material. Hemmingson estimates a 500% increase in prices for raw materials since 2000, especially since the company only uses premium, vacuum arc remelt steel.

On the flip side, Hemmingson considers his supply of high quality equipment to be one of the secrets of his success. Jerico boasts seven CNC mills as well as 50 assorted CNC lathes, grinders and automated gear machines. The Studer S30 grinder is the pride and joy of the shop floor, promising tolerances within 1/10,000", which makes the tolerances more precise than for standard automotive applications. The Studer software uses gaging and probe touch-off, simplifying the profiling of the straight shafts that are commonplace at Jerico.

"It's super accurate and reliable," Hemmingson says. "It's expensive, but I'd do anything to have two more."

The business of racing cars isn't easy to comprehend, Hemmingson acknowledges. "If you're outside looking in, it probably looks like a very small number of customers or clients. But the number of customers, like the numbers of fans

and enthusiasts in motorsports everywhere, continues to grow all over the world, and there's racing going on in places where you never think of when you think of racing, like South Africa, Finland, the Netherlands, England. This may reflect a wide variety of types of racing, but it's still racing and engineering, and the mechanics have to work. To be successful and stay ahead, you have to invest in the best technology and find the best people to run it."

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Photos are courtesy of Balzers Inc.

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Marine Gears

Special Aspects for

High Performance

Joseph L. Hazelton, Associate Editor

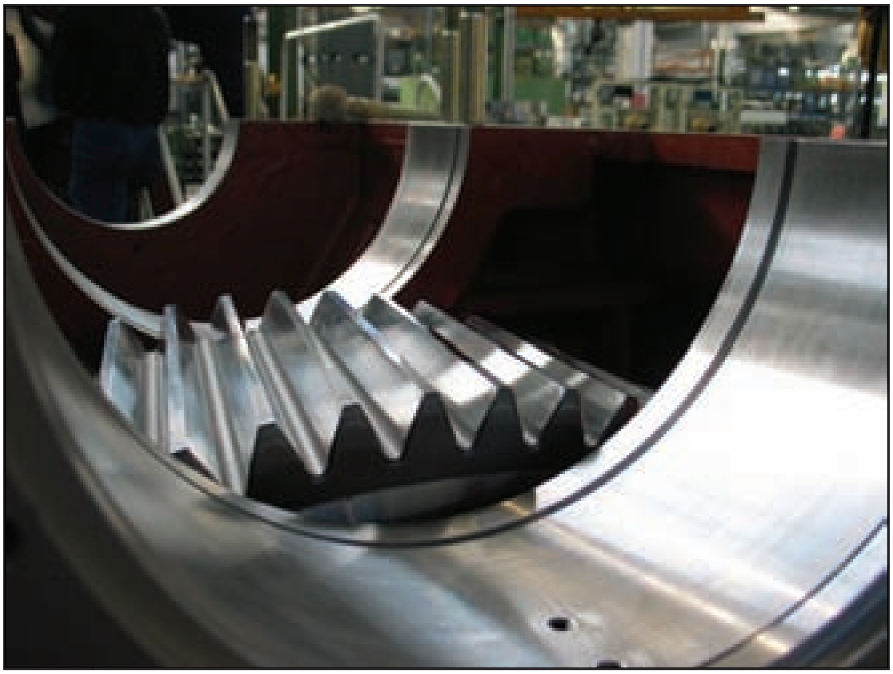

Photo courtesy of ZF Marine GmbH.

A gearbox that absorbs 30 percent of external forces, transmits power from two engines operating at different speeds, and uses gears that meet several design and specification standards at the same time.

Many gear manufacturers would wrinkle their brows at this unusual box. But what's odd to them would be ordinary to marine gear manufacturers. The above features are commonplace in high performance marine gears and their boxes, which are used in vessels as wide ranging as luxury yachts, fast ferry boats, frigates and workboats.

Given these features, marine gear manufacturers often have to create gearboxes that are much more complicated than their non-marine counterparts.

Connected to Multiple Engines

Whether a luxury yacht or a massive warship, many larger marine vessels have two or more engines and use certain ones based on whether they're in a small harbor or on the open sea. This situation, multiple sources of power, contrasts starkly with non-marine gearboxes.

"Normally, you have just one input for non-marine gears," says Franz Hoppe, general manager–marine for Renk AG. Hoppe estimates that more than 90 percent of non-marine gears have only one input.

In a larger marine vessel, the multiple inputs may be gas turbines, electric motors or diesel engines. Depending on the combination, the power sources often operate at different speeds, so they require gear trains with different reduction ratios.

The trains are usually combined inside a gearbox, with one train for the primary engine and one for the secondary. Further complicating the box's design is that both trains may be connected to one or more output shafts. Also, the gear trains can overlap; they can share gears.

Hoppe uses a modern naval frigate as an example to explain the need for multiple engines. A smaller warship, a frigate may have as many as three engines: a gas turbine able to transmit 20,000–30,000 kilowatts of power and two diesel engines, each one able to transmit 4,000–8,000 kilowatts.

The different engines are necessary because a ship operates in different environments and speed modes: loiter, cruise and fast. In a harbor, the frigate has to move slowly, carefully. Its speed engine, the gas turbine, isn't suitable, so it switches to its lower-power diesel engines.

To do that, the crew may also need to switch from one gear train to another. This switch involves activating and deactivating various clutches in the gear system, in a certain sequence, disengaging some gears and engaging others so the box achieves the right reduction ratio between the online engine and the propeller.

These gearboxes have resulted in an alphabet soup of marine gearbox types: CODAD, CODAG, CODOG and so on. For instance, CODAG refers to a combined gearbox able to transmit power from a ship's diesel engine and/or its gas turbine. The box can transmit power from both engines, driving two propeller shafts at the same time. CODOG, on the other hand, is the acronym for a combined gearbox able to transmit power from a ship's diesel engine *or* its gas turbine.

Marine gearboxes connected to two or more engines include commercial vessels, like ferries and other roll-on, roll-off ships. Also, high-speed yachts can feature two gas turbines with power output via waterjets, not propellers. Waterjets allow for much more condensed output of the engines' power.

"They're providing much higher speeds," Hoppe says. "With waterjets, you can reach up to 50–60 knots [about 60–70 mph]."

No Infinitely Stiff Foundation

Marine gear manufacturers also have to keep in mind that their gears and boxes will lack what are considered infinitely stiff foundations. The problem isn't the gears, their box, its immediate base, or the ship itself. The problem is the water.

It's not stiff. Even when calm and in a sheltered harbor, water isn't a road or a rigid, stationary platform.

Still, gear mesh mustn't be affected by even rough waters.

"Stiffness is most important to have reliable tooth contact between the gears," Hoppe says.

That reliable contact is no small feat given that a characteristic of high performance marine gears is their mechanical efficiency. At ZF Friedrichshafen AG, Gerald Rowe, R&D manager–marine transmissions, puts that efficiency at 99 percent tooth contact between pinion and bull gear.

To keep gears in mesh, their casings are strengthened with ribs or built with a double-wall design. "The casing has to be as stiff as possible, always," says André Thuswaldner, chief design engineer for Maag Gear AG. Thuswaldner is responsible for the initial design of his company's marine gears.

Overcoming water as foundation is necessary, though, for high performance marine gears to provide their high performance. But what constitutes high performance? What measurable characteristics define a high performance marine gear?

What is a High Performance Marine Gear?

The answer is more elusive than might be expected because physical, performance and other characteristics vary considerably. A high performance marine gear's size can range from 300 mm to 5 meters. Also, marine gears can transmit as little as 225 kilowatts of power and still be considered high performance.

The other end of the power range is more open. Tom Wampler, chief engineer–marine design for Twin Disc Inc., says high performance marine gears' maximum power can reach "astronomical numbers." Renk's Hoppe provides a maximum—50,000 kilowatts—but he adds the upper end can go as high as a customer needs: "We are not limited on the upper side."

High performance marine gears can also mean speed, not just power. The gears inside "fast craft," vessels capable of at least 25 knots (29 mph), are considered high performance. These ships include luxury yachts, patrol boats and high-speed ferries.

Accounting for power and speed, Thuswaldner defines a high performance marine gearbox as having a power/weight ratio of more than 1.5 kilowatts/kilogram or operating at a pitch-line velocity of at least 120 meters per second.

Another characteristic is high reliability. Hoppe defines that as: "Never having a gear tooth or bearing failure." At Maag, Christoph Blättler, general manager for sales–marine gear units, is number-specific. He says naval gears have to be designed for 99.95 percent reliability, with the overall gearbox having a reliability of 99.8 or 99.9 percent.

The need to avoid tooth failure is critical, especially with larger ships. If the gears are in motion, the vessel is likely under way. If a tooth fails then, the ship isn't in charge of its movement anymore, the water is.

The problem is serious when it occurs on the open sea, like in the middle of the Atlantic Ocean. In that case, the ship would need to be towed into port. But it's more serious if the vessel is maneuvering in harbor. A tooth failure then and the vessel could collide with nearby ships, damaging them and maybe killing people.

Given these consequences, marine gear manufacturers are very much concerned with avoiding gear failure, whether due to the gears themselves or

A Complex Gearbox—The combined marine gearbox here, a CODAG, can simultaneously transmit power from a diesel engine and a gas turbine or can transmit power from one or the other separately. Such complicated gearboxes are common among many larger marine vessels. Photo courtesy of Renk AG.

Designed for Space, Weight—Marine vessels often require more complicated gear arrangements, such as planetary gear systems, so their gearboxes meet space and weight restrictions. Such complicated gearboxes are common among many larger marine vessels. Photo courtesy of Maag Gear AG.

An Unstable Foundation—A marine gearbox has to be designed and manufactured to compensate for the lack of stiffness in what is—in the ultimate sense—its true platform: water. Photo courtesy of Renk AG.

due to external forces.

Resistant to External Forces

Marine gearboxes must be able to resist external forces, like those introduced into the box's foundation from the surrounding structure or vibrational forces from the propeller and propeller shaft. To an extent, resisting external forces means being able to absorb them.

That extent can be considerable in high performance marine gears. Hoppe says gear assemblies for non-marine applications normally absorb a maximum of 20 percent of the amplitude of external forces, but marine assemblies absorb 30 percent of non-transient forces and up to factor 2 transient forces. The extra absorbency helps the marine gearbox avoid breakage.

Without the extra amount, the gearbox's bearing assembly could overload, disrupting the oil film, which would immediately damage the bearing, causing the gears to stop. At that point, the vessel wouldn't be propelling itself; the water would be.

Besides natural external forces, some vessels—naval vessels—have to be built to handle shock forces from man-made objects. Think large ammunition, like torpedoes. Hoppe says shock forces can be up to 50 g (500 meters/second^2). "These are very high acceleration forces."

Designed to Fit Space

Less unusual than being built for extreme external forces, marine gears, like other gears, have to be designed to fit in a limited space with a predetermined shape and still provide all their required performance.

"That is the challenge for the design engineer when it comes to marine gears," Thuswaldner says. He adds that the space can sometimes require making the gearbox more complicated than it would otherwise have to be: "But that's life in marine gearboxes."

As an example, Thuswaldner compares a marine gearbox for a gas turbine capable of generating 22 megawatts of power at 330 rpm and an industrial gearbox capable of the same output but built for a mill drive. The boxes' lengths and heights would be the same but the industrial gearbox would be more than 50 percent wider than the marine one.

The smaller width would be achieved with a different basic design. The marine gearbox would be a double-stage planetary gear system with as many planets as possible, while the industrial one would be a double-stage parallel-shaft gearbox.

The marine box would have 16 gears, an input shaft and an output one. The industrial box would consist of four gears and three shafts: an input, an intermediate, and an output. "That makes the gearbox very simple and very cheap, but heavy," Thuswaldner says.

The heaviness would be a main problem and a major reason for a planetary gear design. Weight matters, after all, in terms of a vessel's fuel consumption and handling characteristics. The planetary gear design's overall effect would be to allow the marine gearbox to be smaller and weigh less.

Built to Meet Multiple Standards

Whatever the gear arrangement, though, high performance marine gears often have to be manufactured to meet several standards simultaneously.

This situation can occur, for example, if a prospective owner isn't sure in what country he'll register the ship he's having built. The country matters because each one may have its own design and specification standards for marine vessels, and all ships registered with that country must meet those standards, which are administered by classification societies. In the United States, the society is the American Bureau of Shipping, known as ABS. In Italy, it's the Registro Italiano Navale, RINA.

"Each society has its own methods for calculating gears," says Rowe of ZF. He adds that calculation results can vary as much as 65 percent from society to society.

Knowing the various standards is also necessary when a company makes marine gears for customers from many different countries. In either case, when the marine gear manufacturer doesn't know which classification rules may be applied, it designs its transmissions to account for all possibilities.

Having to meet several standards at one time, to design for different power sources in one gearbox, and to compensate for the lack of a stiff foundation—these considerations increase the complexity of designing and manufacturing high performance marine gears and gearboxes, especially when they're used in vessels as diverse as coastal workboats, luxury yachts and oceangoing warships.

As Twin Disc's Wampler says:

"There's no such thing as a run-of-the-mill, suits-all-things marine gear."

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Correction 1

Gear Technology's January/February 2006 issue included a paragraph that required additional information to be understandable. The paragraph appeared on page 54, in the article "Investigation of the Noise and Vibration of Planetary Gear Drives." The paragraph was under the subsection heading "Measured vibration results for planet gears in vehicle tests."

The additional information, underlined here, should have appeared in the paragraph as follows: ". . . were made under a condition of gradual acceleration in first and fourth gears. <u>Another way to achieve a meshing phase difference is to use unequally spaced planet gears as shown for planetary gear set II in Table 2.</u> As seen in the figure, the vibration acceleration level of planetary gear set II with a meshing phase difference <u>achieved by the use of unequally spaced planet gears</u> was approximately 3–12 dB lower than that of planetary gear set I without a meshing phase difference. Better results . . . ."

Correction 2

In the March/April 2006 issue of *Gear Technology*'s article titled "Medical Device Manufacturing Keeps Gear Industry Healthy," we mistakenly identified Precipart Corp.'s location. Precipart Corp. is located in Farmingdale, NY, not Farmingdale, CT.

Correction 3

In the March/April issue of *Gear Technology*, the Gleason Genesis 130H CNC was featured with a misleading headline. The cycle time of the *loader mechanism* was one second and the machine's cycle time is two or two-and-a-half-seconds.

We apologize for the errors.

—The Editors

Rougher Tooth Surfaces Give Longer Gear Life?

Robert Errichello, Technical Editor

T.C. Jao and his co-authors reach the startling conclusion that gears with rough tooth surfaces have longer macropitting life. How can this be when all other research seems to show smoother surfaces give longer life? As it turns out, the devil is in the details.

They tested FZG PT-C macropitting gears and FZG GF-C micropitting gears, which are the same in all respects except PT-C gears have a tooth surface roughness of R_a = 0.3 μm, whereas GF-C gears have a tooth surface roughness of R_a = 0.5 μm. PT-C and GF-C gears are peculiar because they have no profile modification, such as tip or root relief. Consequently, under the high loads used in FZG tests, the tips of the gear collide with the flanks of the pinion. The collision creates a severe dent at the start of active profile (SAP) on the pinion. It is this dent, together with the difference in tooth surface roughness, that causes the difference in macropitting life between the PT-C and GF-C gears.

Practical experience shows that a dent caused by tip-to-root interference can be a root cause for macropitting. The authors' experiments show this is true. With PT-C gears, a narrow band of micropitting forms just above, and adjacent to, the dent at the SAP. Shortly after micropitting forms, macropitting initiates from within the micropitting band, and the macropits grow until the gear fails.

In contrast to PT-C gears, GF-C gears also form a band of micropitting at the SAP, but the band continues to grow toward the pitchline until it forms a wider band of more severe micropitting. Then, at a later time than the PT-C gears, macropits initiate at the top of the micropitting band.

The authors explain that the failure mechanism is different for GF-C gears because their rougher tooth surfaces cause more severe micropitting that removes the dent and prolongs initiation of macropitting. They ascribe the root cause of the macropitting to geometric stress concentration (GSC). In the case of PT-C gears, GSC is caused by the dent, whereas in the case of GF-C gears, GSC is caused by the step in the tooth profile at the upper edge of the micropitting crater.

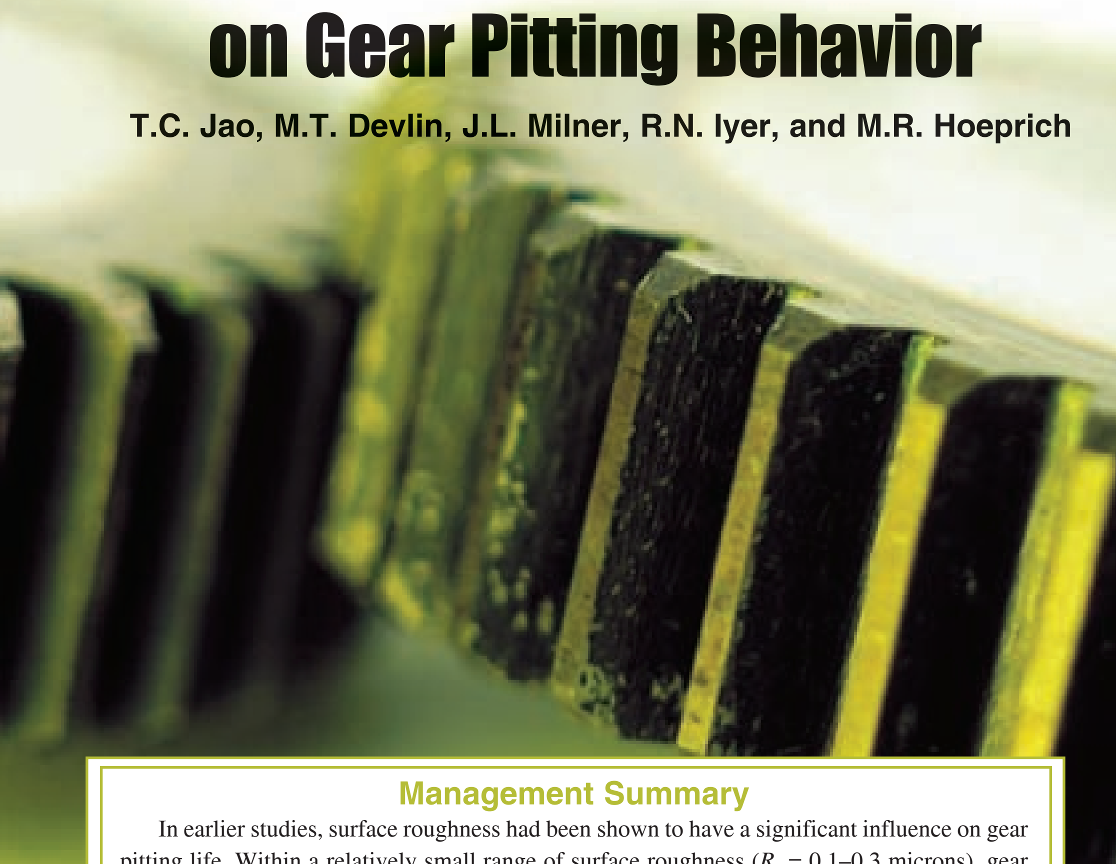
Considering the authors' findings, one has to question the validity of FZG gears for testing lubricants. Because FZG gears have no tip or root relief, they are not representative of industrial gears. Furthermore, PT-C gears produce point-surface-origin (PSO) macropits from the shoulder of the dent at the SAP. The mechanism and failure mode are similar to debris denting in rolling element bearings, where it is well known that macropits initiate from the shoulders of debris dents. Consequently, PT-C gears actually test PSO macropitting caused by tip-to-root interference, and do not measure the true macropitting resistance of lubricants. At best, they may measure differences in crack propagation rate, but GSC—not lubricant properties—control the macropit initiation.

With GF-C gears, severe micropitting destroys base pitch spacing, which obviously causes high dynamic tooth loads, and causes GSC at the top of the micropitting band near the pitch line. Therefore, GF-C gears are similar to PT-C gears in that they test PSO macropit initiation caused by GSC.

The authors should be congratulated for shedding light on the performance of FZG gears, for explaining the interactions between tip-to-root interference and surface roughness and showing the influence the interactions have on micropitting and macropitting.

Influence of Surface Roughness on Gear Pitting Behavior

T.C. Jao, M.T. Devlin, J.L. Milner, R.N. Iyer, and M.R. Hoeprich



Management Summary

In earlier studies, surface roughness had been shown to have a significant influence on gear pitting life. Within a relatively small range of surface roughness (R_a = 0.1–0.3 microns), gear pitting life as measured by the FZG pitting test decreases as gear surface roughness increases. This inverse relationship between gear surface roughness and pitting life is well understood in the field. To determine whether this inverse relationship is applicable to a wider range of surface roughness values, we have conducted a pitting study using gears whose surface roughness ranged from 0.1–0.6 microns. The results were not completely expected.

The study shows that the micropitting area is radically larger when the gear surface roughness is close to the upper limit of the range studied. Plasticity index, which approaches a value of around 3.7 for the rougher gear surface, appears to be responsible for the formation of such a large micropitting area. At the same time, the formation of a pit is also greatly delayed. Not only is the pitting life significantly longer, but the initiation of pits can occur near the pitch line. This paper discusses how high surface roughness introduces a wear mechanism that delays the formation of pits.

Introduction

Extended gear fatigue pitting life is not only an essential performance requirement for today's automotive and industrial gear oils, but also for automatic transmission fluid (ATF) or continuously variable speed transmission (CVT) fluid (Refs. 1–2). Past studies have shown that both gear surface roughness and chemical and physical properties significantly influence the fluid's pitting performance (Ref. 3). The fluid's chemical and physical properties affect oil film thickness, boundary frictional coefficient and corrosiveness. The effect of surface roughness on metal fatigue behavior has been studied extensively and is apparently quite well understood (Refs. 4–8). It has been well established that surface roughness is a major factor influencing the formation of micropitting (Refs. 7–9). It also has been shown that micropitting is the most common cause of pitting in modern clean steels since current steel processing technology essentially eliminates subsurface inclusions (Refs. 10–11).

Although it is generally accepted that micropitting can lead to pitting, the specific mechanism by which micropitting induces pitting is still poorly understood. The lack of in-depth understanding of this cause-and-effect relationship between micropitting and pitting hinders the advancement of gear oil, ATF and CVT fluid technology with respect to improvement in the

Dr. Tze-Chi Jao *is an R&D fellow and head of the fundamental research group of Afton Chemical Co., located in Richmond, VA. Afton was formerly Ethyl Petroleum Additive Co. Jao holds a doctorate in physical chemistry, and his research interests include fatigue behavior in gears, rheology, wear mechanisms in valvetrains, friction durability in transmissions, and nanotechnology. He has published more than 70 technical papers and has more than 13 patents in lubrication.*

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Table 1—Two-Variable Pitting Test Matrix Study.

Test Code	Fluid's 100°C Kinematic Viscosity [cSt]	Gear Type	Surface Roughness (R_a, micron)
02–06–02 (LH1)[a]	7.5	FZG Type C–M[b]	0.43
02–14–10 (LL1)[a]	7.5	FZG Type C–P[c]	0.20
02–12–08 (LH2)[a]	7.5	FZG Type C–M[b]	0.41
02–07–03 (LL2)[a]	7.5	FZG Type C–P[c]	0.23
02–09–05 (HH1)[a]	15.0	FZG Type C–M[b]	0.43
02–08–04 (HL1)[a]	15.0	FZG Type C–P[c]	0.20
02–10–06 (HH2)[a]	15.0	FZG Type C–M[b]	0.50
02–05–01 (HL2)[a]	15.0	FZG Type C–P[c]	0.20
02–11–07 (HL3)[a]	15.0	FZG Type C–P[c]	0.23

[a] The label given in parentheses is the abbreviated code for that particular test run; the first and second letters stand for the levels of viscosity and surface roughness, respectively. L and H mean low and high, respectively, while the numeric value indicates the order of the repeat runs.
[b] FZG Type C micropitting gear
[c] FZG Type C pitting gear

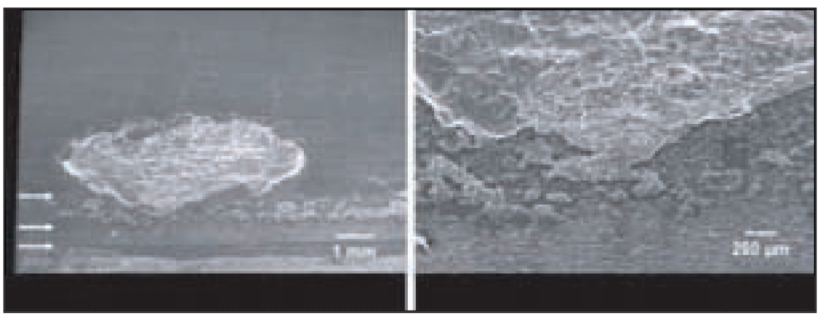

Figure 1—SEM images of the No. 12 tooth of the tested pinion gear of the HL3 run using the 15 cSt oil with an FZG Type PT-C pitting gear. The area between the two outer arrows is the micropitting band.

fatigue pitting life. To overcome this deficiency, we have devoted considerable effort to increase the fundamental understanding of how micropitting impacts pitting. Our earlier study indicated that, in addition to the effects of oil's physical properties, surface roughness has a large effect on the gear's fatigue pitting life (Ref. 3). Within a small variation of gear surface roughness, increasing the gear surface roughness decreases, almost linearly, its fatigue pitting life. To extend the model developed in the previous work to higher roughness values, we studied the effect of gear surface roughness on fatigue pitting life by doubling the surface roughness for test gears designed for the micropitting study. The results were not completely expected. This paper describes how a small increase in surface roughness decreases the fatigue pitting life, but a large increase can actually delay the formation of pits and thus significantly increase the gear's fatigue pitting life. In essence, a non-linear reversed effect of surface roughness on fatigue pitting life was observed.

Experimental

Gears tested. Both the FZG Type PT-C pitting gears and the Type GF-C micropitting gears tested were designed and made by ZF Friedrichshafen AG (Refs. 12–13). The supplier indicated that both types of gears were made from the same steel material and hardened by the same process. They had the same tooth profile geometry, pitch-line diameter, addendum and dedendum depths. However, the pinion and wheel gears of the same batch are usually not made from the same single melt, but they are hardened by the same process and at the same time. This could also be true for pinion gears that are provided from the same batch of gears.

The difference in surface roughness between the Type PT-C and GF-C gears was achieved by specially dressing the grinding wheels and the control of the surface roughness during grinding. The specification for the gears requires the surface hardness to be HRC 62 (Refs. 12–13); we did not independently verify the value.

In this study, before each pitting test was carried out, the surface roughness of the gear was measured by a one-dimensional profilometer. For the matched pinion/wheel set, three teeth of a pinion were chosen to measure the surface roughness of the contact side by profilometry along the center involute of each tooth profile. This was repeated for the gear. The arithmetic mean value of the six measurements was taken as the R_a value of the pair. The procedure for the measurements was described as part of the test procedure (Refs. 14–15). The surface roughness values of the pinion/wheel sets are shown in Table 1.

Oils. Two oils of different viscosities were prepared with the same additive package, which was developed for application in automatic transmissions. Even though both oils use polyalphaolefin (PAO) as the base oil, different combinations of PAOs were necessary to prepare the two oils at two 100°C kinematic viscosity levels—7.5 cSt and 15 cSt. The oils are shown in Table 1. The boundary frictional properties, film formation properties and anti-corrosion properties of the test oils were measured as described previously (Ref. 3).

Pitting test matrix. The two variables investigated in this study were the gear surface roughness and oil viscosity. Thus, a matrix of four different pitting tests was carried out. Each test was carried out twice.

Pitting test run conditions. The tests were conducted using the FZG pitting test PT C/9/90 procedure except the oil temperature was set at 120°C (Refs. 14–15). The Hertzian stress (P_c) for the load stage 9 used was 1,650 N/mm^2. The pitch-line velocity was 8.3 m/s. The expected tip deflection under this test condition was around 20–30 μm. During the test, inspection of the tested gear for micropitting and pitting was conducted every eight hours. The areas of micropitting and pitting were measured and recorded at each inspection.

Determination of fatigue pitting life. According to the FZG pitting test procedure, the fatigue pitting life was determined when any tooth or the sum of teeth in one gear accumulated a total pitting area of 5 mm^2. Such measurement for fatigue pitting life for a regular Type PT-C pitting gear was straightforward with no ambiguity.

However, the fatigue pitting life measurements for the runs involving Type GF-C micropitting gears were more complicated because before a total pitting area of 5 mm^2 was reached, the pinion dedendum surface already was covered with micropits. Thus, for the runs involved with Type GF-C micropitting gears, we determined the fatigue pitting life by two procedures. One procedure used the time when the teeth had accumulated a total micropitting area of 448 mm^2, which is approximately the sum of the micropitting band (about 2 mm deep and 14 mm wide) areas measured by unaided eyes from the individual teeth. The second procedure used the time when any tooth or sum of teeth had actually accumulated a total pitting area of 5 mm^2.

It is noteworthy that for the Type GF-C micropitting gears, the total micropitting area of 448 mm^2 is always reached before the total pitting area reaches 5 mm^2.

Surface analysis of the tested gears. SEM was used to analyze the wear and pits of the gear surfaces. A Form Talysurf was used to measure the deviation of the gear tooth profile from the original geometry. To find out how cracks propagate and if any subsurface nonmetallic inclusions could have initiated the cracks, tooth surfaces were sectioned where pits or spalls had occurred.

Results

SEM analysis of the tested gears. The SEM images of the tested pinion gear of each of the four matrix runs are shown in Figures 1–4. Figure 1 shows the SEM images of the No. 12 tooth of the tested pinion gear of the HL3 run at two different magnifications. The band between the upper and lower arrows is the micropitting band. As indicated in our previous paper (Ref. 3), pitting starts at the upper edge of the micropitting band.

Figure 2 shows the SEM images of the tested pinion gear of the LL2 run. Both HL3 and LL2 runs used the same FZG Type PT-C pitting gear but with two different oils. These two oils were formulated with the same additive chemistry but different viscosity grades of PAOs to achieve two different 100°C kinematic viscosities—15 cSt versus 7.5 cSt. The micropitting band widths shown in these two figures are narrow, as is typically seen in the tested gears of FZG Type PT-C pitting gears.

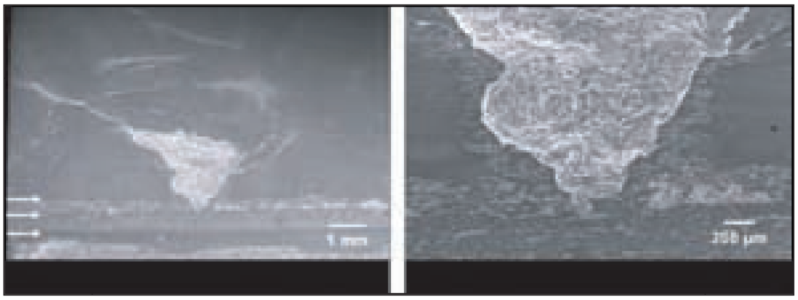

Figure 2—SEM images of the No. 5 tooth of tested pinion gear of the LL2 run using the 7.5 cSt oil with an FZG Type PT-C pitting gear. The area between the two outer arrows is the micropitting band.

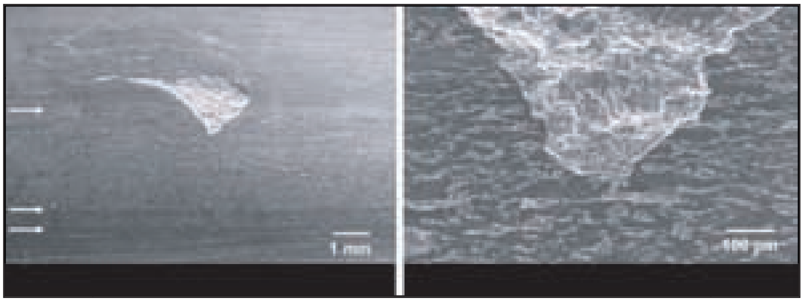

Figure 3—SEM images of the No. 2 tooth of the tested pinion gear of the HH2 run using 15 cSt oil with an FZG Type GF-C micropitting gear. The area between the two outer arrows is the micropitting band.

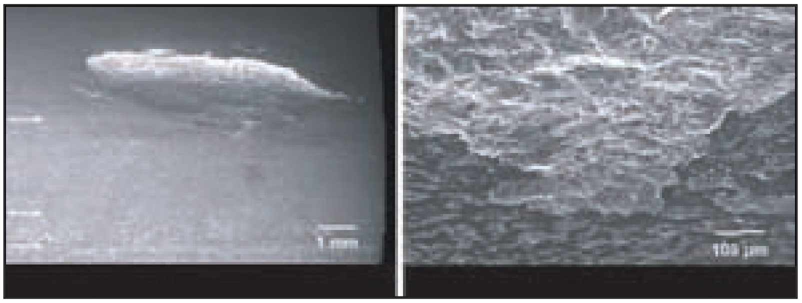

Figure 4—SEM images of the No. 10 tooth of the tested pinion gear of the LH2 run using 7.5 cSt oil with an FZG Type GF-C micropitting gear. The area between the two outer arrows is the micropitting band.

Figures 3 and 4 show the tested pinion gears of the two runs HH2 and LH2, both of which used FZG Type GF-C micropitting gears. Again, the same two oils of different viscosities were used. The noticeable common feature between these two figures is the relatively larger micropitting band width.

Overall, the micropitting band width appears to depend only on the type of gear used. The oil viscosity practically has no effect on the micropitting band width. However, all four sets of SEM images appear to have a common, polished wear band of approximately constant band width, which is shown between the lower two

Table 2—Pinion Dedendum Wear Band Comparison.	
FZG Pitting Test Run No.	Pinion Dedendum Wear Band (mm) Total[a] /Lower[b]
HL3 (No. 12 tooth)	1.11/0.48
LL2 (No. 5 tooth)	0.79/0.50
HH2 (No. 2 tooth)	3.12/0.48
LH2 (No. 10 tooth)	3.37/0.48

[a] Total width consisting of micropitting band and the polished wear band.
[b] Only the lower polished wear band.

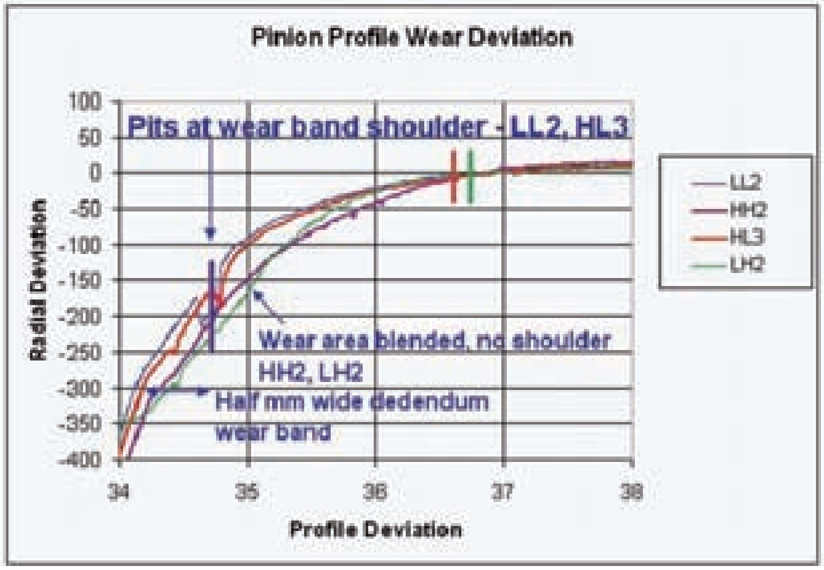

Figure 5—Wear-induced tooth profile deviation.

arrows. Table 2 summarizes the observations obtained from the SEM images shown in these four figures.

It is particularly interesting to compare the two sets of SEM images between Figures 1 and 3. The difference between the two SEM figures is that the former was obtained from a run that used an FZG Type PT-C pitting gear while the latter was taken from a run that used an FZG Type GF-C micropitting gear. Both runs used the same high viscosity oil; yet the impact on the micropitting band is quite dramatic. The run with the pitting gear had a fatigue pitting life of 42 hours, while the one with the micropitting gear had a fatigue pitting life of 170 hours.

Nevertheless, the micropitting gear accumulated a total micropitting area of 448 mm^2 in just 26 hours while the pitting gear accumulated a total micropitting area of only around 224 mm^2 by the time the test reached 42 hours. Similar results can be found comparing Figures 2 and 4. This time, the comparison is between two runs using the same low viscosity oil but different types of gears.

Tooth profile changes on the tested gears. To investigate why the two different types of gears produce such a large difference in the micropitting band width, we used a Form Talysurf to map the contours of the tooth profiles. Figure 5 shows the contours of the four tested pinion gear teeth, each representing one of the four matrix runs. The designation for the profile deviation of LL2's tooth No. 5 is LL2, of HH2's tooth No. 2 is HH2, of HL3's tooth No. 9 is HL3 and of LH2's tooth No. 10 is LH2.

There are two sources for the profile deviation: one is the deviation from the ideal involute geometry even when the gear is new and the other is due to wear. For LL2 and HL3, the profile measurements went over the small pits that formed on the shoulder at the upper edge of the approximately half-millimeter wide polished wear band. The measurements for HH2 and LH2 show a greater, but more continuous and smoother wear that decreases as it approaches the pitch diameter. Intentionally, all four traces do not go through any spall to prevent damaging the instrument.

The two almost overlapping vertical bars between the 34 and 35 mm grid marks indicate the locations where the spalls start to form on the LL2 and HL3 gear teeth while the two vertical bars between the 36 and 37 mm grid marks indicate the locations where the spalls start to form on the HH2 and LH2 gear teeth. LL2 and HL3 belong to FZG Type PT-C pitting gears while HH2 and LH2 belong to FZG Type GF-C micropitting gears.

For LL2 and HL3, it is clear that the spall forms at the shoulder, which appears immediately following the lower polished wear band and can serve as a stress raiser to initiate the formation of a pit. The contours of HH2 and LH2 show that a large deviation from the original tooth profile occurs due to wear, preventing the formation of a clear shoulder. Such a large profile deviation effectively delays the buildup of a stress raiser and removes surface material that might otherwise continue to fatigue and develop into a pit.

Tooth cross sections. Figures 6 and 7 show cross sections of pitting and micropitting gears of the HL3 and HH2 runs, respectively. Figure 6 shows the cross section of a pit and the associated cracks for the HL3 pitting pinion gear with the pit initiated below the pitch diameter and continuing past the pitch diameter and into the region where the traction changes its direction. Figure 7 shows the cross section of a pit and subsurface cracks for the HH2 micropitting pinion. The starting point for the initiation of the pit in the micropitting gear is quite different from that in the pitting gear. For the micropitting pinion, the starting point for the initiation of a pit is just below the pitch diameter, with the majority of the pit lying above the pitch diameter.

Fatigue pitting lives. The measured fatigue pitting lives obtained from the FZG pitting tests of the matrix described in Table 1 are shown in Table 3. For the runs using the micropitting gears, both values of the fatigue pitting life determined by the two different methods are listed. Table 3 also shows the EHD film thickness, boundary friction coefficients and anti-corrosion properties of the two oils used in the matrix study.

To quantify the effect of the EHD film formation, friction reduction and anti-corrosion properties of oils on the fatigue pitting life in the 120°C FZG pitting test, the physical properties of the two oils were correlated to the fatigue pitting lives obtained from the matrix study. Using multiple linear regressions with standard statistical techniques (Ref. 16), a model similar to the previous study (Ref. 3) has the following general form:

$$\text{Fatigue pitting life} = \text{A} + \text{B} * \textit{EHD FT} + \text{C} * \textit{BndFr} + \text{D} * \textit{AGV} + \text{E} * R_a \quad (1)$$

The final model for fatigue life contains only those terms with a statistical significance greater than 90 percent. After the values of the constants A, B, C, D, and E are known, the relative effects of the EHD film thickness (*EHD FT*), boundary friction (*Bnd Fr*), anti-corrosion properties (*AGV*) and surface roughness (R_a) on the FZG rig test results are determined.

Below is the model obtained using the fatigue pitting lives determined by the conventional procedure, which measures the time when the total pitting area reaches 5 mm².

$$\text{Hours to pitting} = -15.8 + 0.46 * \textit{EHD FT} + 203.8 * R_a \quad (2)$$

The two parameters, boundary friction coefficient and anti-corrosion properties, *AGV*, are dropped from this model because the variations in these two parameters are small compared with the variations in the EHD film thickness and surface roughness. The R-square for this correlation is found to be 0.552. Not only is the R-square low, the constant found for the surface roughness is a positive value, implying that the rougher the surface, the longer the fatigue pitting life. Such results are counterintuitive.

Using the fatigue pitting lives as determined by the conventional procedure gives a poor correlation, so a second model was developed using the fatigue pitting lives determined by the alternate procedure, which measures the time for the total micropitting area to reach 448 mm².

$$\text{Hours to pitting} = 69.0 + 0.49 * \textit{EHD FT} - 206.4 * R_a \quad (3)$$

Again, the two parameters, boundary friction coefficient and anti-corrosion properties, *AGV*, are dropped from the model for the same reason as above. The R-square for this model is 0.915, which is clearly far better than the previous

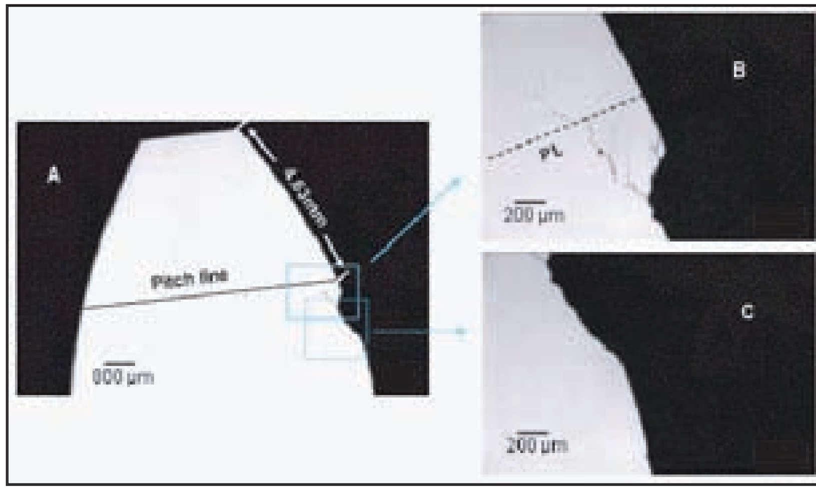

Figure 6—SEM image of a cross section through a spall on the 9th tooth of the HL3 run. This test involves a 15 cSt oil and a pitting gear.

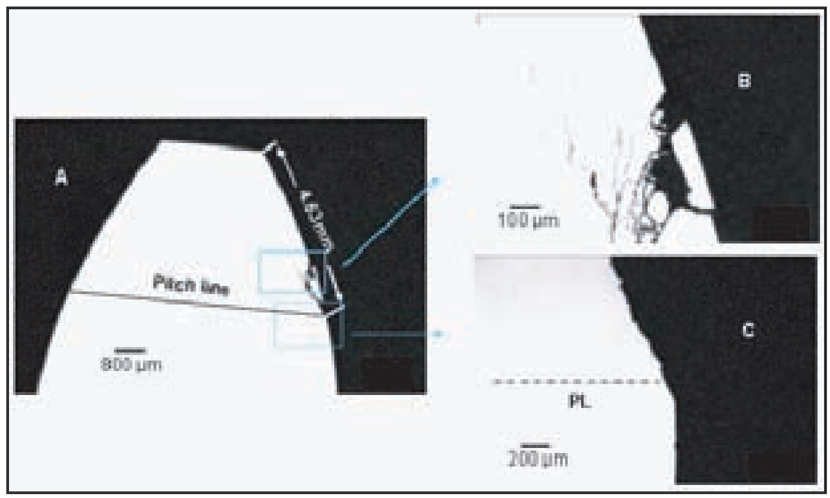

Figure 7—SEM image of a cross section through a spall on the 2nd tooth of the HH2 run. This test involves a 15 cSt oil and a micropitting gear.

Table 3—Physical Properties and Measured Fatigue Life Results.

Test Code	Gear R_a (nm)	EHD FT[a] (nm)	Boundary Friction Coefficient[b]	AGV[c]	Measured Fatigue Pitting Life
LH1	430	45	0.121	49	122 (10)[d]
LL1	200	45	0.121	49	58
LH2	410	45	0.121	49	66 (2)[d]
LL2	230	45	0.121	49	32
HH1	430	112	0.114	43	74 (26)[d]
HL1	200	112	0.114	43	74
HH2	500	112	0.114	43	170 (26)[d]
HL2	200	112	0.114	43	95
HL3	230	112	0.114	43	42

[a] Measured at 2 m/s and 120°C.
[b] Measured at 100°C.
[c] Measured by the modified Ball Rust Test as described in Reference 3.
[d] The value in parentheses is the fatigue life measured by the time when the total micropitting area has reached 448 mm².

Table 4—Summary of Surface Roughness and Plasticity Index Measured from the Coast Side of the Corresponding Driven Gears.				
Gear Code	R_a [nm]	R_z [nm]	R_q [nm]	PsiGW[a]
HH2	494	5,046	654	3.7
HL3	214	2,511	284	1.6
LH2	462	4,247	590	3.4
LL2	196	2,297	250	1.6

[a] PsiGW is a measure of plasticity index.

model. At the same time, the constant determined for the surface roughness parameter is a negative value, which implies that a rougher surface should produce a shorter fatigue pitting life. The results are in line with the expectation of the previously developed model by us using the fatigue pitting lives measured in the pitting tests employing exclusively pitting gears (Ref. 3).

Discussion

The findings of this study are not what we expected from the empirical model developed by correlating three physical properties of the oil and one physical property of the gear to FZG pitting test results carried out with the family of FZG Type PT-C pitting gears. The model predicts that gears with rougher surfaces should have shorter fatigue pitting lives (Ref. 3). Such results are in agreement with the simple theory of asperity contact in elastohydrodynamic lubrication developed by Johnson et al. (Ref. 17) based on the plasticity index concept of Greenwood and Williamson (Ref. 18).

In this study, the surface roughness of the micropitting gears (0.4–0.5 μm) is approximately two times that of the pitting gears (0.2–0.23 μm). Yet the fatigue pitting behaviors observed in these two types of gears in the FZG pitting test are dramatically different. The main differences are: 1) the micropitting band in the micropitting gears is about three times wider than that in the pitting gears, 2) the formation of pits in the micropitting gears is delayed considerably, and 3) the fatigue pitting lives determined by the standard FZG pitting test procedure do not correlate with the three physical properties of the oil and one physical property of the gear, but changing the method to measure the fatigue pitting life correlates well.

Noticing the large difference in the gear tooth profile deviation due to wear between the pitting gears and the micropitting gears, we examine the possible role of the plasticity index. Plasticity index is a measure of the probability that a material under load will undergo plastic deformation. Since plasticity indices of the driving pinions and driven gears were not measured before carrying out the pitting tests, we could only measure them after the tests. We also measured the surface roughness to compare with the values measured before the tests on the contacting sides of the teeth. The results are summarized in Table 4.

The R_a values in Table 4 closely agree with those shown in Table 1. This gives us confidence that surface roughness and plasticity index measurements from the coast side of the driven gears are reasonably reliable. This means that the plasticity index of the micropitting gears is larger than that of the pitting gears by a factor of greater than two. However, the measured plasticity indices are larger than theoretically expected from the definition given by Greenwood and Williamson (Ref. 18), since plasticity index is proportional to the square root of R_a or R_q and inversely proportional to the square root of summit radius of the asperity, β. R_a is the arithmetic mean of the gear tooth surfaces while R_q is the corresponding geometric mean value.

Nevertheless, plasticity indices of the micropitting gears are significantly larger than those of the pitting gears. Micropitting gears with a plasticity index at least twice as large as the pitting gears can more readily undergo micropitting wear resulting in greater gear tooth profile geometry modification and providing some stress relief.

As Figures 1–4 indicate, the micropitting wear band in the pitting gears is narrow and a shoulder is quickly built toward the edge of the micropitting wear band. With such a shoulder serving as a stress raiser, microcracks can be readily propagated to initiate a pit. This appears to be what was happening on the pitting gears. On the other hand, with a much higher plasticity index, micropitting gears can develop more extensive micropitting wear, extending the edge of the micropitting wear band all the way near the pitch line. Thus, the observed higher tooth profile deviation on the micropitting gears is understandable.

Let us then examine what happens beyond the micropitting band on the micropitting gears. Figure 7 shows that once the propagating cracks are formed, they will continue to grow in the same direction past the pitch line after which the traction direction reverses and the microcrack initiation direction also reverses. This also happens to some extent on the pitting gears, as shown in Figure 6. It is interesting to note that for the micropitting gears, once the crack for the pit is formed, it continues in the same direction beyond the pitch line, after which the traction direction is expected to reverse.

In the micropitting-initiated pitting mechanism, one expects that the direction of the crack for the formation of a pit on the dedendum should be pointing toward the pitch line while the corresponding direction of the crack on the addendum will reverse itself to point toward the pitch line again (Refs. 9 and 19). Figure 8 shows that the cracks for micropitting in the micropitting gears before and after the pitch line indeed reverse their direction as expected. In Figure 8, the solid arrows indicate the traction directions. It is possible that since the load on or near the pitch line is the highest because only a single tooth is carrying the load, the greater subsurface maximum Hertzian shear stress is responsible for continuing the pit-forming cracks.

Olver offered some observations indicating that a pitting crack does not necessarily follow the microcrack-initiation direction (Ref. 20). For smoother gears, microcracks will be shorter and more difficult to propagate into spalls. Near the pitch diameter, high contact loads and the often worn geometry that increases pitch diameter stresses will likely result in subsurface origin spalls.

The observed fatigue pitting lives obtained from the micropitting gears do not correlate well with the model developed for the pitting gears (Ref. 3) nor do they agree with the simple theory of Johnson et al. (Ref. 17) when the standard definition of fatigue pitting life of the micropitting gears is used. Still, the correlation could be made to conform better to the empirical model or the theory if fatigue pitting life is defined according to the alternate method. This is shown by the model of Equation 3. Thus, the study here confirms that within the family of pitting gears, which have a surface roughness value around 0.3 μm, the rougher surface will decrease the pitting fatigue life.

The simple rule does not appear to be applicable to the family of the micropitting gears, which have a surface roughness value of 0.5 μm, because of the much higher wear rate that effectively delays the formation of a geometric stress concentration site and thus lengthens the pitting fatigue life. Even though micropitting gears show higher apparent fatigue pitting life, the rapid formation of micropitting and the associated high wear rate change the gear tooth profile so significantly, it may have weakened the tooth strength significantly early on in the test to the level that is not desirable. In other words, a high level of micropitting fatigue itself may be undesirable as much as the formation of macropitting.

Conclusions

When the FZG pitting test is carried out with FZG Type GF-C micropitting gears instead of the standard FZG Type PT-C pitting gears, the fatigue pitting lives are unexpectedly much longer than those obtained with the standard FZG Type PT-C pitting gears. The delay in the pit formation is due to extensive micropitting wear changing the gear tooth profile to prevent the formation of a shoulder that can serve as a stress raiser for the pit formation, thus suggesting that a wear model should be considered. Higher plasticity index on the micropitting gear tooth surface is responsible for extensive micropitting wear.

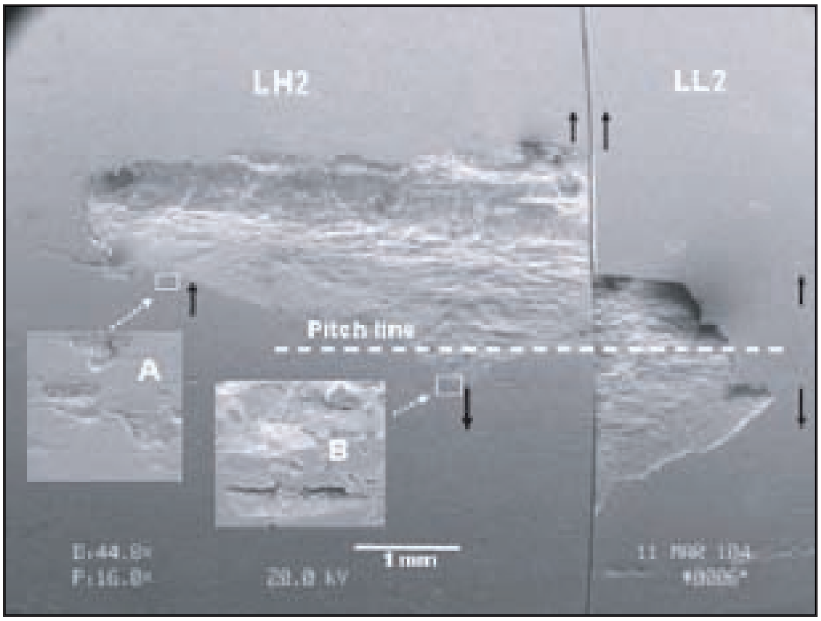

Figure 8—Micropitting crack directions before and after pitch line on the micropitting gear (LH2). The SEM was taken together from the two pinions. The location of the pitch line is defined with respect to the LH2 pinion; it is only approximate for the LL2 pinion. The magnification for the insert is 300X.

However, if an alternate method of defining the fatigue pitting life of micropitting gears is used, then the trend of rougher surface giving shorter pitting fatigue life is followed.

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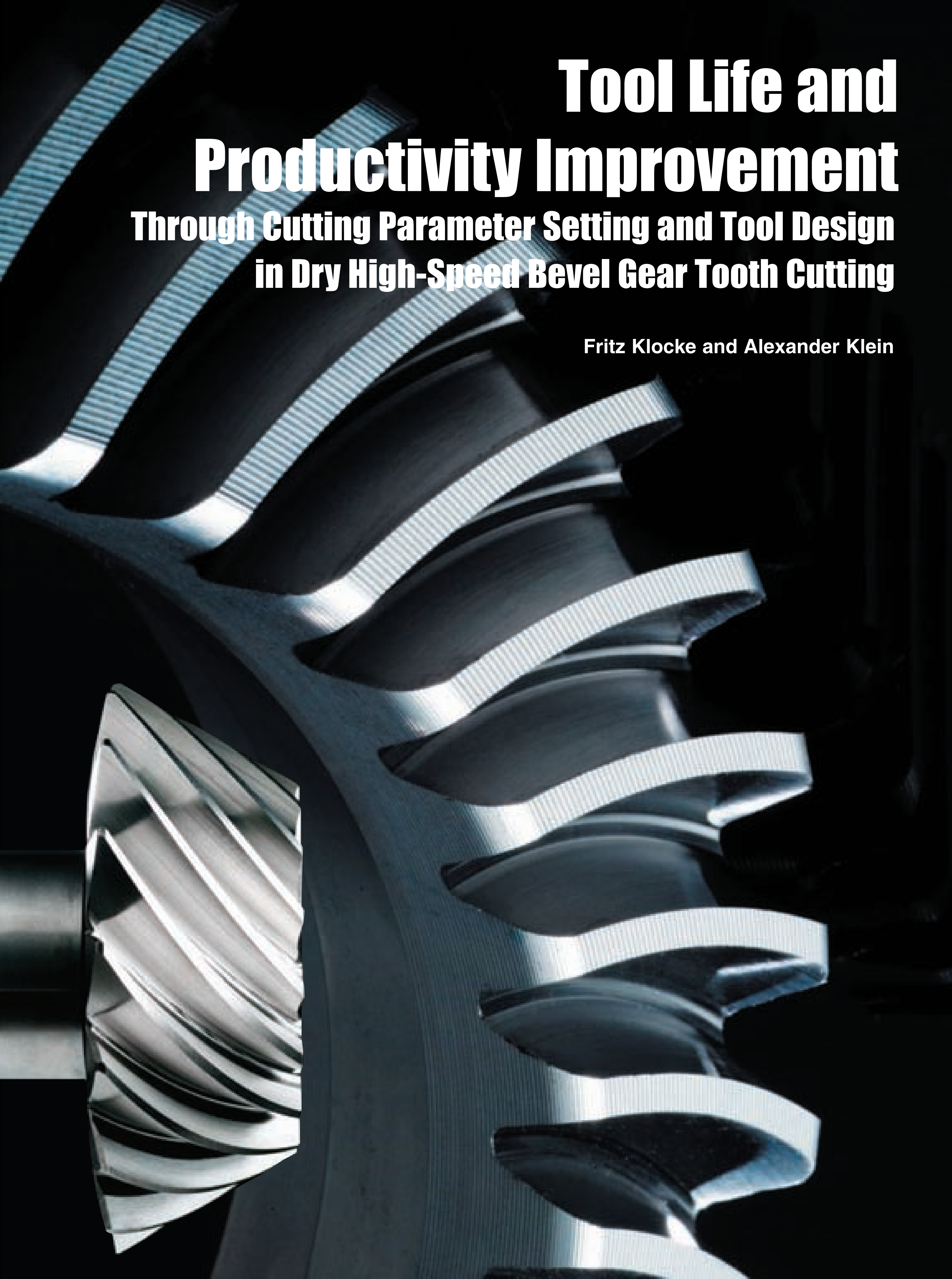
Tool Life and Productivity Improvement

Through Cutting Parameter Setting and Tool Design in Dry High-Speed Bevel Gear Tooth Cutting

Fritz Klocke and Alexander Klein



Management Summary

The introduction of carbide tools and hard coatings has led to a quantum leap in material removal rates and tool life in manufacturing automotive-sized bevel and hypoid gears, as the tools and coatings have in other areas of machining technology (Refs. 1–6). Further benefits of the new techology are the ability to abandon the use of coolant and the improvement of quality, which partly goes back also to advancements in the machine tool technology and the implementation of quality loops in manufacturing (Ref. 3–4 and 7–13).

However, the number of influential factors on manufacturing cost per piece is high, and an in-depth process analysis and understanding has not yet been established in many areas.

Yet this fundamental knowledge is required to show the right direction for further enhancements of economic efficiency and productivity, since thorough process development is difficult to conduct during and after the start of production in an industrial environment (Ref. 13). Therefore, thorough machining investigations are conducted at WZL within the scope of an AiF research project (13713 N/1) on high speed bevel gear machining.

This article presents some of the findings of cutting investigations at WZL in which the correlation of cutting parameters, cutting materials, tool geometry and tool life have been determined.

Finally, the idea for a new WZL face milling tool concept, which has some significant theoretical advantages over state-of-the-art tool systems, is presented.

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Dipl.-Ing. Alexander Klein *is team leader of the WZL research department Gear Manufacturing Technology. Also, Klein performs research in bevel gear manufacturing.*

Introduction

The value-added chain for bevel and hypoid gears in high-volume production usually contains the technologies which are shown in Figure 1. This article focuses on the soft cutting of teeth. It shall contribute to a deeper understanding of the correlation between the choice of the technological parameters and the consequences on tool life and cost per piece and shall also show some findings and approaches for an increase of manufacturing excellence.

Before attempting to decrease the cost per piece in bevel gear tooth cutting, it is suggested that an analysis be made of the technological system to be optimized, in order to identify the influential factors, the cost drivers and the possible levers for achieving relevant improvements. Therefore, a number of influence factors on the most important output variables are shown and structured in an Ishikawa chart (see Fig. 2). The manufacturing system can be subdivided into the tool system, the machine tool system (including work holding device), the workpiece and the cutting parameters.

The high number of influential factors, their interdependencies with regard to the output variables, and the intransparencies (areas where we lack knowledge) about all of the effects and correlations make the manufacturing system highly complex, according to Wöhe's definition (Ref. 14). Thus total understanding is nearly impossible. Despite this, the investigations can improve production, since selected parameters have been subjected to planned variations and full-factorial designs while other parameters (e.g. machine tool, workholding devices, workpiece, tool material and batch) have been kept constant to achieve a good signal-to-noise (S/N) ratio in laboratory trials.

Systematic Experimental Studies on Selected Parameters in Bevel Gear Machining and their Effect on Tool Wear

Experimental setup and design of experiments. Figure 3 shows the experimental setup for the wear studies at WZL. Instead of using a fully equipped cutter head, the parts are cut with one inside and one ouside blade each. Only this way has it been possible to conduct

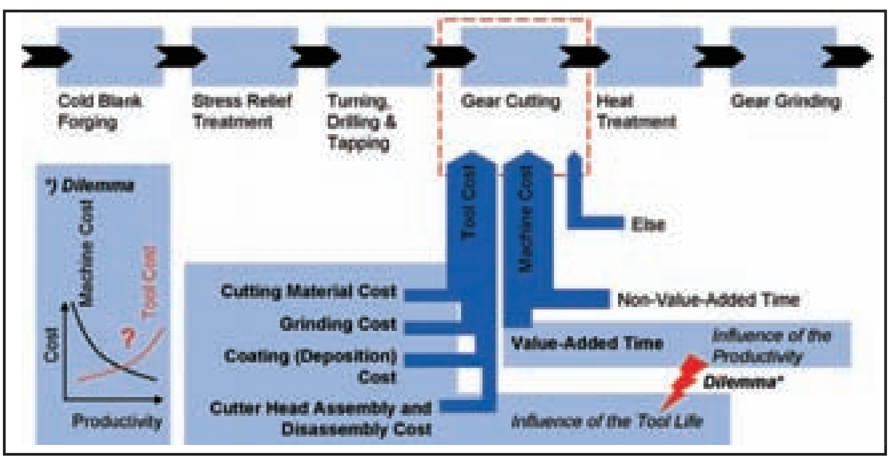

Figure 1—Manufacturing technology chain for high volume bevel gear production.

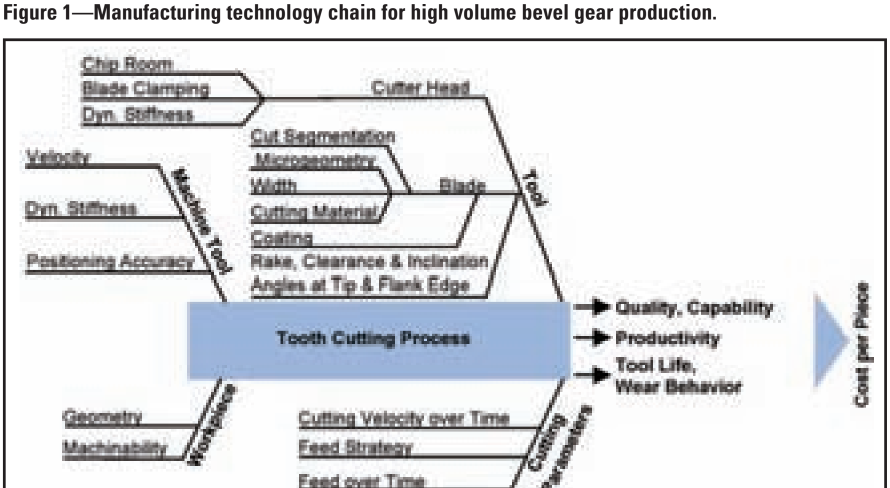

Figure 2—System analysis of bevel gear tooth cutting technology.

the amount of trials in the laboratory, since the number of required blanks is reduced by an order of magnitude. The blade wear is monitored over the tool life by measuring the wear land width or chipping size. Furthermore, all blades have been analyzed with a scanning electron microscope after the end of a trial. The end of tool life was defined as a maximum wear land width or chipping size of $VB_{max} = 0.15$ mm.

The presented findings were obtained by manufacturing a face-milling designed SUV ring gear, as shown in Figure 4. All blanks were manufactured in the same batch.

In most trials, fully coated carbide blades have been applied as specified in Figure 4.

Cutting parameter variation. In Figure 5, the obtained tool lives in an extended full-factorial cutting speed and feed variation are presented. Except the finishing operation trial ($f_{BG} = 0.03$ mm per blade group), all trials were intended to determine the tool wear behavior at high material removal rates (roughing).

The results show a significant negative effect of the cutting velocity on the tool life within the chosen set of other parameters. The cutting speed variation ($v_c = 280; 330; 380$ m/min) brings about a left-curved "tool life vs. cutting speed" curve. At very high cutting speeds and/or feeds, the wear mode changes from abrasive wear to small or larger chippings (see also Figs. 6 and 7).

In the feed variation, a strongly nonlinear behavior of the tool life, which would not have been revealed in the mere full-factorial matrix, was identified in the medium roughing feed trial. The tool life overview in Figure 5 and the wear charts and pictures in Figures 6 and 7 show that there appears to be an optimum feed at about $f_{BG} = 0.15$ mm (at the applied set of other parameters). It should be mentioned, however, that other effects, which might occur with a fully equipped cutter head (chip jam or dynamic effects) have been intentionally excluded in the single blade group laboratory trial.

The feed variation also shows that beyond tool life optimum feed, increases in feed rate have a negative effect on the tool life. However, the ratio of tool life drop and machining rate enhancements is notably better than the corresponding ratio in the cutting speed uprating.

The finishing feed trial

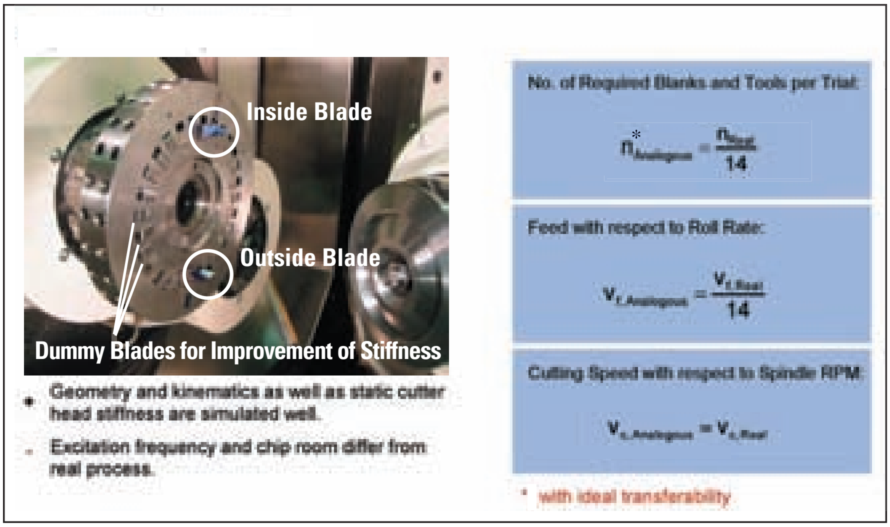

Figure 3—Setup for single blade group experimental trial at WZL.

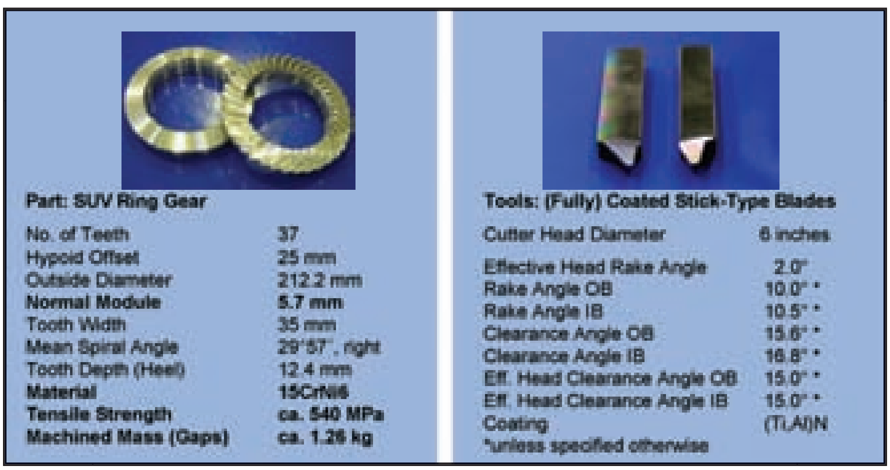

Figure 4—Trial workpiece and tool data.

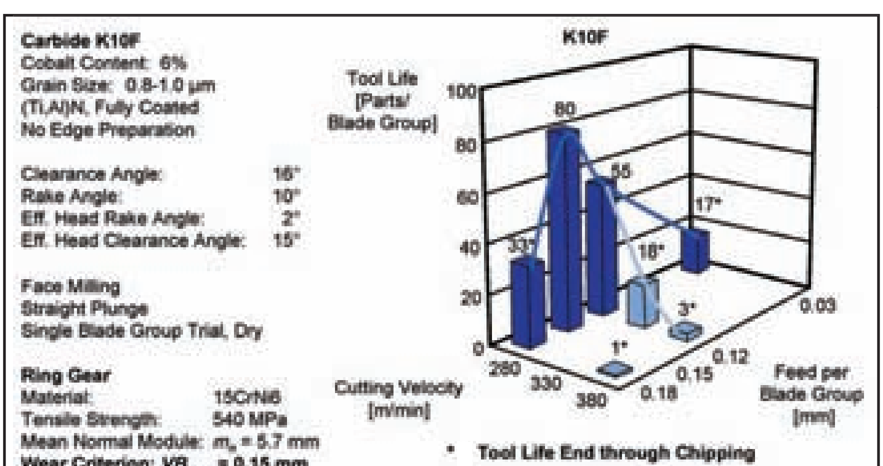

Figure 5—Tool life overview of the global cutting parameter variation.

(f_{BG} = 0.03 mm) supports the theory of the existence of only one maximum of the "tool life vs. feed" curve. The obtained tool life at the very low feed is, accordingly, much inferior to the tool life at higher feeds. Obviously, the ratio of load per cut and number of load cycles to machine a given volume of material has an optimum at undeformed chip thicknesses of around 0.15 mm. This complies with experience from other milling applications in which carbide tools are used. One problem in bevel gear manufacturing, however, is the fact that the cutting depth, and therefore the absolute cutting force, is exceptionally high, which can lead to problems with regard to the machine dynamics. Therefore, in the state of manufacturing, the feed rate is lowered with increasing plunge depth.

Tool substrate material variation. In order to determine the fitness for use of different cutting substrate materials, two very fine grain carbides and one ultrafine grain grade were tested at equal cutting parameters in a full-factorial variation. All tools were fully coated with (Ti,Al)N, unless specified otherwise.

The results show that the tools of the grade K10F (hardness of approximately 1,750 HV30, tensile rupture strength of approximately 3,100 MPa) and K30F (hardness of approximately 1,550 HV30, tensile rupture strength of approximately 3,240 MPa) achieved the best tool lives (see Fig. 8). The wear rates of the K10F tools are lower than the wear rates of the K30F blades (see Fig. 9), particularly at wear initiation. However, the wear of the K10F tool is more affected by chippings and microchippings (see Figs. 8 and 10). Hence, it seems possible that depending on the application (tool and workpiece geometry, workpiece material, etc.), either of the cutting materials can be identified as the best material in the individual case.

The tools of the ultrafine grain carbide K10UF, however, obtained only inferior tool lives and chipped early even at relatively low cutting parameters (see Figs. 8 and 11). At high cutting speeds and/or feeds, tool breakage occurred during the first cuts. Through tool examination, contamination as well as grinding cracks could be excluded as potential reasons for tool failure. Obviously, the high sensitivity towards high dynamic load is characteristic for this tool material. The often mentioned twofold ben-

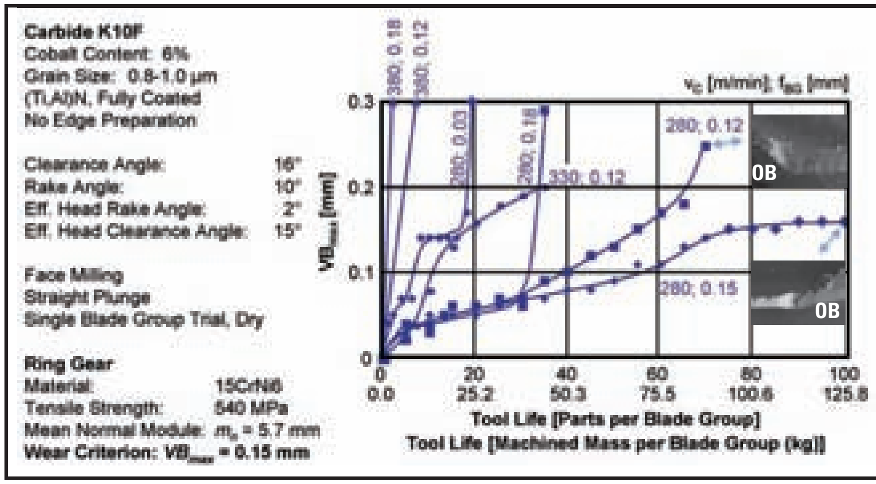

Figure 6—Wear charts of the cutting parameter variation.

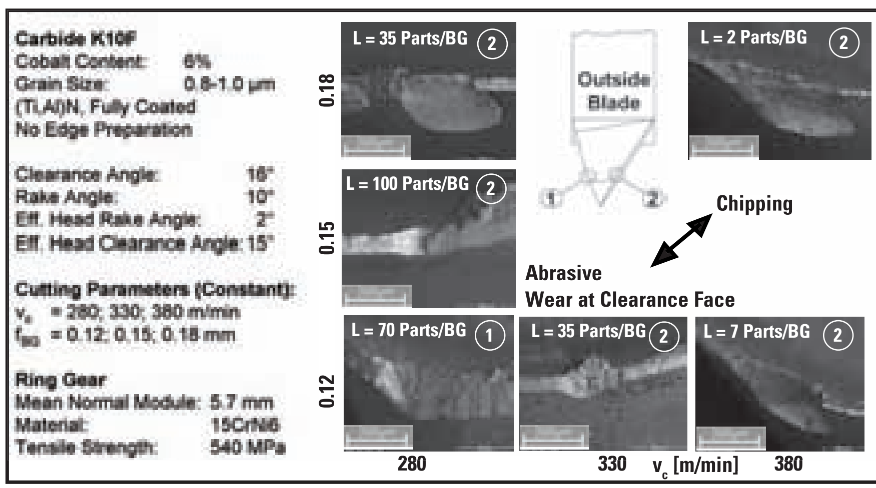

Figure 7—Wear mechanisms of carbide tool K10F.

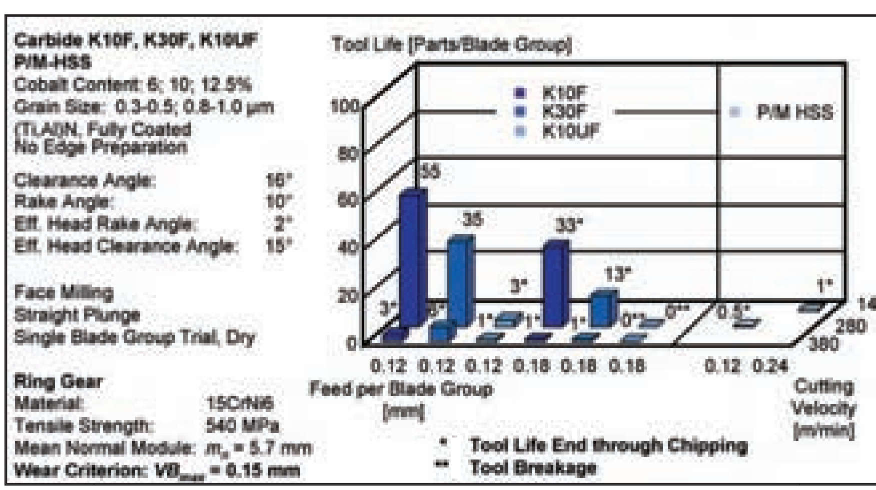

Figure 8—Tool life overview of the substrate variation.

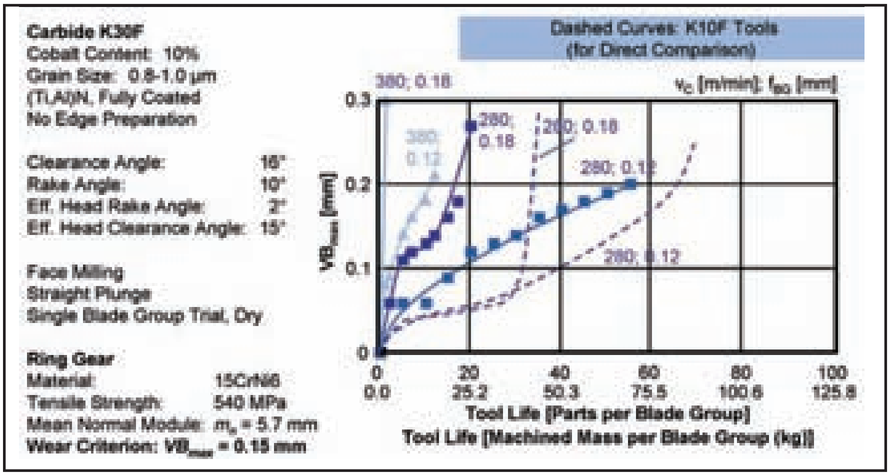

Figure 9—Wear charts for the K30F very fine grain carbide.

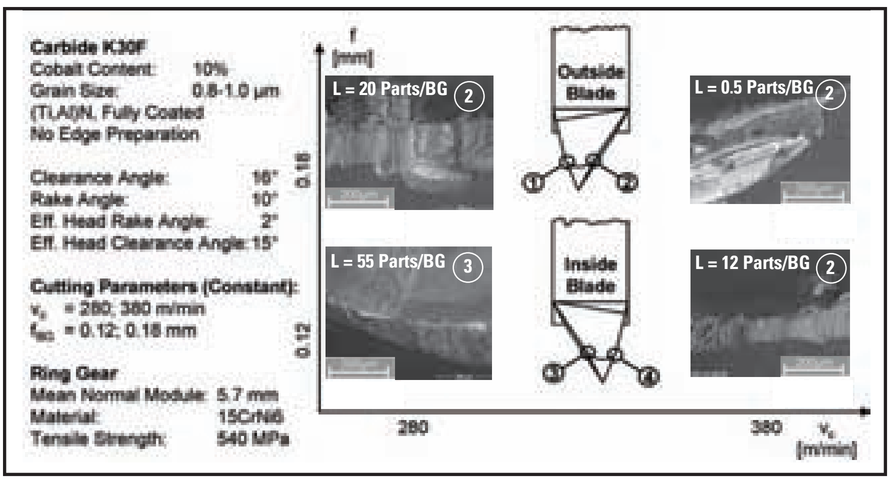

Figure 10—K30F tool wear mechanisms.

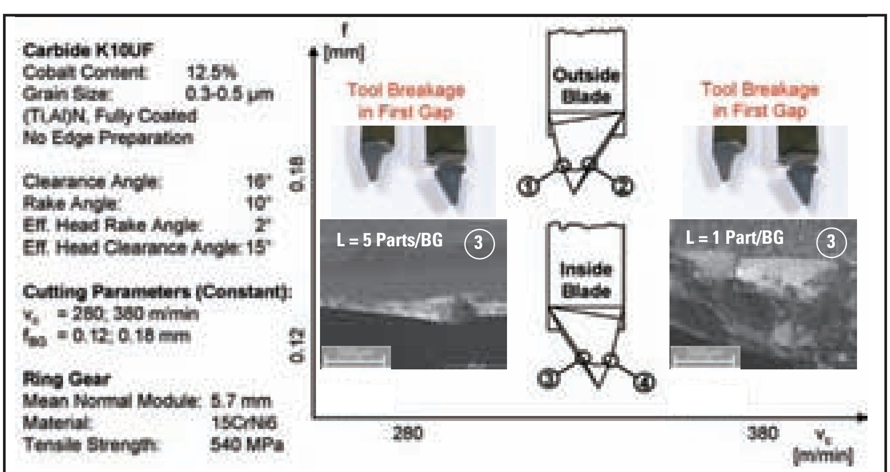

Figure 11—K10UF tool failure and wear.

eficial influence of the grain refinement for toughness and hardness might apply for the cutting edge stability at moderate dynamic or high static loads. High macroscopic loads on the tool body (not only the cutting edge area), however, can obviously lead to tool breakage earlier than they do for coarser grain grades. The key figure for toughness (tensile rupture strength) seems to show the opposite. The TRS value, however, is determined in a quasistatic bar-bending test and seems to have little significance for the material behavior at impact loads.

In two trials, fully coated blades of a state-of-the-art P/M grade that is applied successfully in parallel axis gear hobbing, have also been applied for bevel gear cutting. The results are presented in Figure 8. At the same material removal rate as the lowest cutting parameter set of the carbide trials, these tools could not achieve acceptable tool lives. Even a change of the ratio of cutting speed and feed to a value which is more suitable for HSS (while keeping the material removal rate constant) did not bring about tool lives which are competitive with carbide. This is obviously due to the significantly lower red hardness of the P/M HSS or even due to an overcritical heat load on the tool, which is related to the chip length. For details about tooth cutting with P/M HSS tools, please see Reference 15. It can be assumed that P/M HSS blades can achieve better tool lives at lower cutting parameters. Their potential for high productivity automotive bevel gear cutting, however, appears to be not as high as the machining rate potential of cemented carbide.

Coating concept variation. In dry, high productivity tooth cutting of spiral bevel and hypoid gears with stick-type blades, there are two competing technologies in terms of tool reconditioning. After the first use, the blades either are reground and thus mechanically decoated only from the clearance sides or are reground from the clearance sides and the rake. In the first case, the tool can be used again with coated rake and uncoated clearance side. This brings about disadvantages with regard to tool substrate load, but supersedes a recoating process. In the latter case, the tool has to be recoated after every regrinding, since its rake would be uncoated during cutting otherwise. The technological advantages are, however, coupled with the need for coating deposition after each application.

In order to provide laboratory data for an economical assessment of the alternative concepts, a full factorial cutting speed and feed variation has been conducted with blades that have been mechanically decoated by grinding at the clearance sides. Their tool lives can be compared to the tool lives of fully coated blades with all other parameters kept equal.

Figure 12 shows the comparison of the tool lives. It is obvious that the tool lives of the fully coated blades are remarkably higher than the tool lives of the only-rake-coated blades. The difference with regard to tool wear behavior becomes apparent when comparing the wear charts of the blades (see Fig. 13). While the wear rate of fully coated blades decreases after the initiation wear, it remains at a constantly high value compared with blades that are uncoated on the clearance sides. Although, in this case, the performance of the two tool systems differs strikingly and by a large amount, it should be stated that it is possible that the difference might be less distinct with tools of different clearance angles or at lower cutting speeds.

The cost effiency comparison of the tool systems is, naturally, highly influenced by the cost per coating deposition. The fact that coating prices differ significantly between the United States and Europe could be the main reason for the tendency that more fully coated tools are used in Europe than in the U.S.

Tool macrogeometry variation. Another important set of technological degrees of freedom besides substrate and coating specification and cutting parameters is the tool geometry.

In order to evaluate the influence of the blade macrogeometry on the wear behavior, tools with increased wedge angles (at tip and side) have been applied in a full-factorial variation of cutting speed and feed, and their tool wear progress has been compared to blades of standard geometry. The wedge angles have been increased by decreasing the clearance angle at the tool tip and side edges and by decreasing the side rake angle. On the one hand, the rake angle reduction results in a cutting force and power increase (Ref. 16). On the other hand, the shape stability and heat removal rate from the cutting edge are higher.

In Figure 14, the tool life comparison

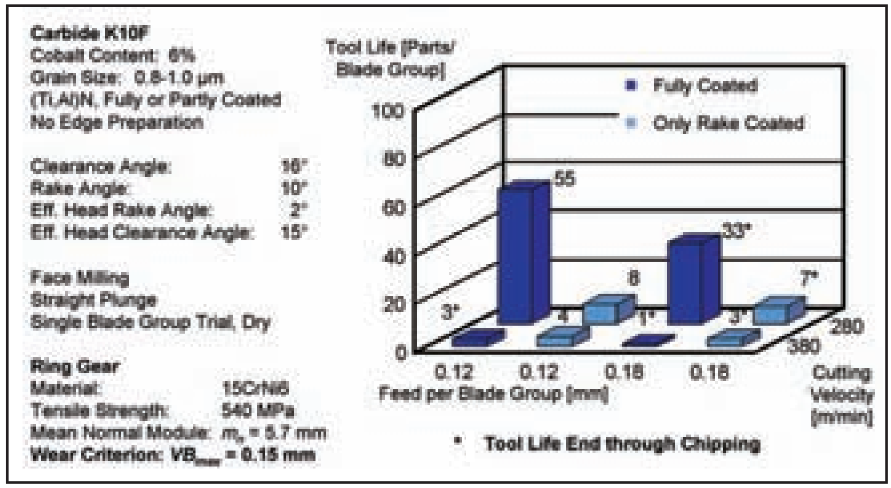

Figure 12—Tool life comparison between fully coated tools and only-rake-coated tools.

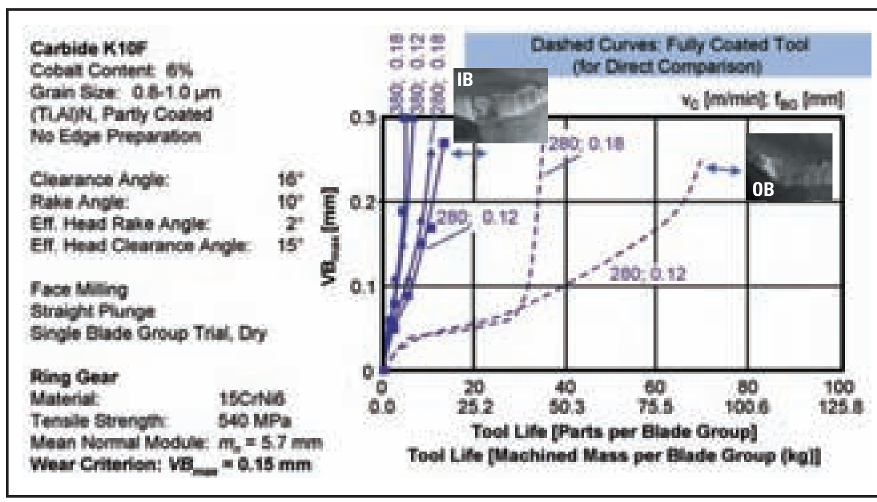

Figure 13—Wear charts of the only-rake-coated tools.

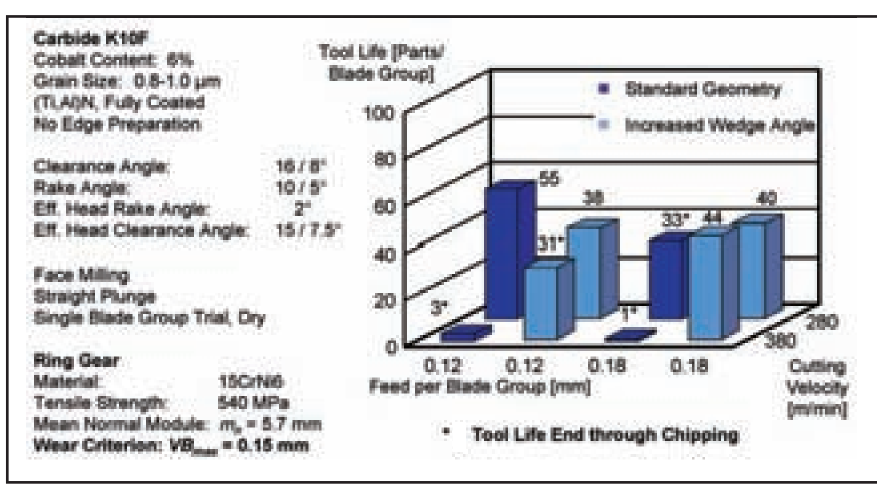

Figure 14—Comparison, standard geometry vs. increased wedge angle.

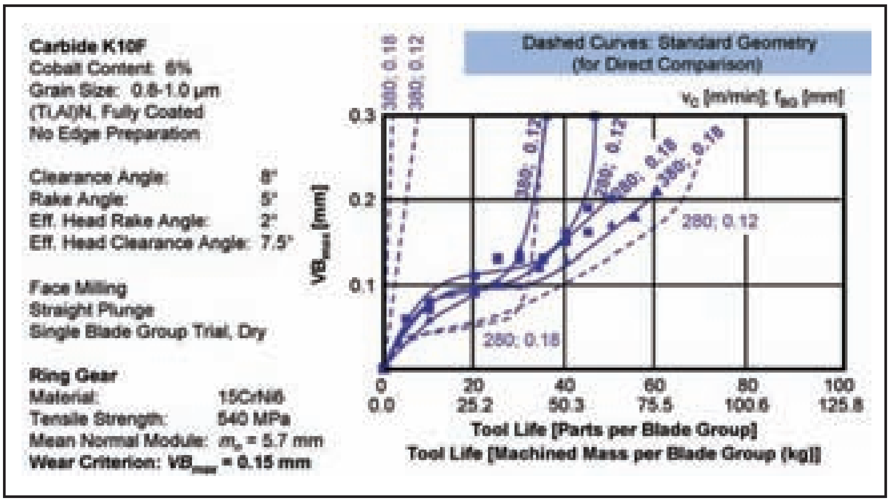

Figure 15—Wear charts of the geometry variation.

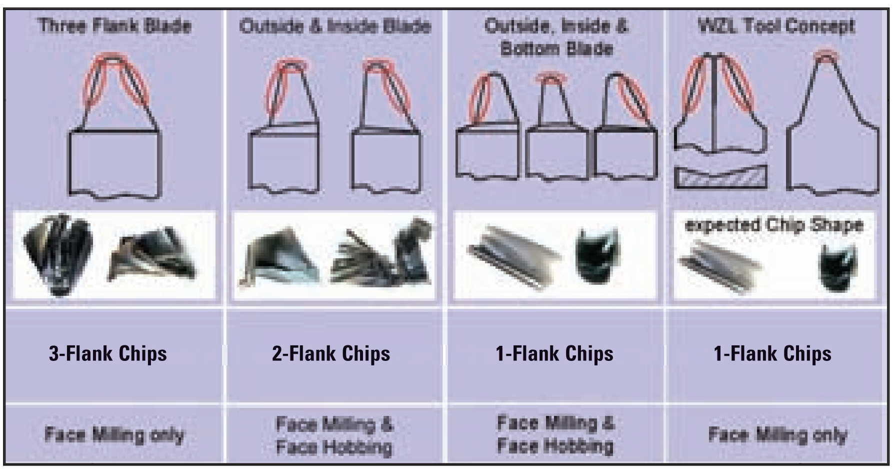

Figure 16—Cut segmentation of different tool concepts.

is presented. It is obvious that although the longest tool life was achieved with a conventional tool at the lowest machining rate, the average tool lives of the modified blades are notably higher than the average tool life of the conventional blades in this full-factorial trial.

Moreover, the sensitivity of the modified blades towards changes of applied cutting parameters and loads is clearly lower than the sensitivity of the standard-geometry blades. While the standard blades show a tool life drop at increased machining rates, the tools with higher shape stability reach a similar tool life, no matter which cutting parameter set in the wide trial range has been applied.

This difference between standard tools and modified blades can also be seen in the wear progress curves in Figure 15. While the wear curves of the standard geometry tools vary considerably, depending on which cutting parameters have been applied, the modified tools showed similar wear progress at all used cutting parameter settings.

Concept of a New Blade System

While small changes of technological parameters, such as rake and clearance angles, chip thickness or carbide cobalt content are intended to bring about step-like improvements, the choice of changing more basic settings provides the chance to establish a technological leap. Therefore, a tool concept, which has a fundamentally different cut segmentation than the state-of-the-art systems, is presented below. The statements below apply on the actual tool tip and are considered regardless of the tool shaft, since the idea is theoretically applicable to round, square, rectangular or pentagonal blade blanks or even cutting inserts which are mounted to a rigid cutter head body.

In Figure 16, three state-of-the-art blade systems for spiral bevel and hypoid gear manufacturing and the WZL tool concept are presented.

The fundamental advantage of the three-flank blade (left) is the fact that only one blade (per group) is required to cut the teeth. Therefore, it can theoretically achieve high material removal rates at a given chip thickness value. However, the number of degrees of freedom in terms of tool design is low since the side rake angle must be set to zero on both sides in order to avoid a negative rake angle on one side. Consequently, the tip tool cutting inclination λ_s is set to zero, too. This results in higher specific cutting forces (higher cutting edge load) and more agitated cutting. Additionally, the V-shaped chip cross-section (a three-flank chip) leads to a critical chip rolling behavior since the chip shape hardly allows for decent coiling of the chip.

For kinematic reasons, the three-flank blade can only be applied for face milling operations (Ref. 17).

The system of outside and inside blades avoids the disadvantages of V-shaped chips by separating the theoretical three-flank chips into two two-flank chips. Two-flank chips require less room for coiling up, but they have also turned out to tend to chip drag or buckle and to tear and collide with the opposite tooth flank. Due to the cut segmentation, a minimum number of two blades per blade group is required. The blades can, however, be manufactured with positive side rake angles and thus also with a low tip tool cutting inclination λ_s. Both can lower the cutting force and the impact load on the tool. The system of inside and outside blades can be used for face milling and face hobbing operations.

The third system, which consists of inside, outside and bottom blades can also be applied for face milling and face hobbing and has the highest number of degrees of design freedom. Through a consequent cut segmentation into mere one-flank chips, an advantageous chip collective is generated, and chip drag and buckling are avoided. Positive side rake angles can be established, as well as any tool cutting inclinations at the cutting edges. However, the extended chip

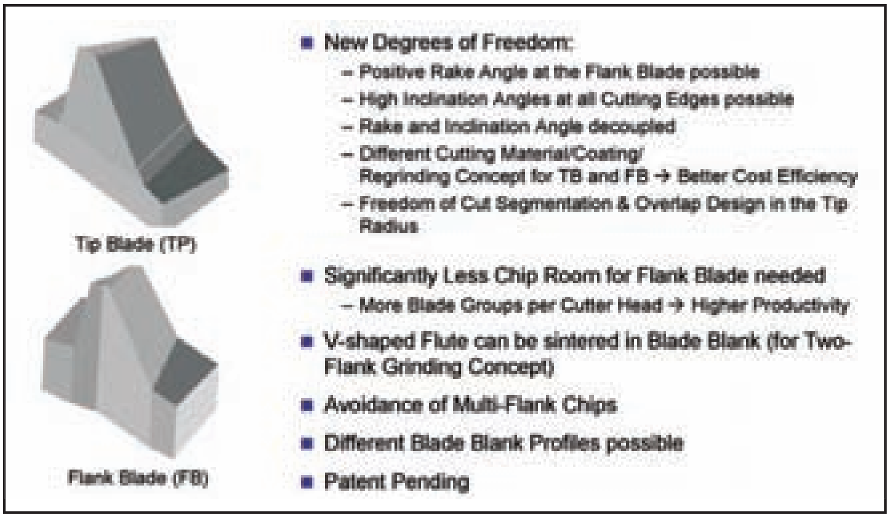

Figure 17—A possible design of the WZL tool concept and gained degrees of freedom.

segmentation into three chips requires a minimum of three blades per blade group. This main disadvantage has led to the fact that this tool concept is barely applied in high volume production nowadays.

The WZL tool concept combines the advantages of cutting only one-flank chips with a tool system that only requires two blades per group. The tip blade can be understood as a modified three-flank blade which is retracted at the side edges. The side blade can be seen as a modified three-flank blade which is retracted at the tip. Since no chip drag or buckling is expected, these tools are likely to achieve longer tool lives. The avoidance of multiflank chips, however, is not the only theoretical advantage of the new tool concept (see Fig 17).

Since the side cutting edges of the side blade do not touch each other, the rake can be featured with a groove that establishes a positive rake angle on the side edges, although the side blade cuts on both opposite sides. The different chip segmentation hence brings about some theoretical advantages of this tool system towards the system of outside and inside blades in face milling. Since the thicknesses of tip and side chips are very dissimilar, the design of outside, inside or even three-flank blades is always complicated by the fact that a trade-off between the different requirements has to be met. The new tool system, however, allows for adapting the side blade to the specific needs of this part of the tool and for adapting the tip blade accordingly. It could, for example, turn out to be the most cost efficient way to cut teeth with different cutting materials, coatings or coating concepts (full/rake only) on side and tip blades.

For these reasons, it seems promising to perform further research and development in order to make a fact-based assessment of the tool performance. To reduce the number of required machining trials, thorough design of experiments and a closed-loop tool development, as suggested in Reference 18, seems appropriate. In the near future, the influence of the tool shape and the cutting parameters on both the cutting forces and the chip coiling and flow behavior will be studied with high-speed photography and force measuring in a new test rig at WZL.

The presented basic concept as well as the subsidiary ideas have been documented in a patent application and are protected as main and subsidiary claims (Ref. 19).

Summary and Outlook

In a thorough cutting parameter variation, the effects of cutting speed and feed on the tool life of bevel gear cutting blades with conventional design were determined. While the cutting speed shows a negative effect on the tool life in the whole considered range, there appears to be an optimum feed for tool life around $f_{BG} = 0.15$ mm per blade group. Above and below this feed, the tool lives are significantly lower. Particularly for the very low feed of $f_{BG} = 0.03$ mm, which is characteristic for finishing operations (or the finishing part of a completing operation), the tool life drops considerably. These findings cannot be transferred ideally into industrial production since dynamic machine stiffness and chip room constraints can cause problems that lower or destroy process robustness. The studies, however, show the potential of present-day tools that has to be exploited for future enhancements of tooth cutting.

The negative effect of the cutting velocity in the range $v_c = 280–380$ m/min was observed on tools with conventional geometry design for each tested coating, each cutting material and on tools with a cutting edge preparation as well as on very sharp tools which were not rounded.

The very fine grain carbide grades K10F and K30F showed the best performance. In contrast to that, unacceptable tool lives were obtained with the ultrafine grain carbide K10UF and P/M-HSS blades.

The comparison of the wear behavior of fully coated tools versus only-rake-coated tools showed a distinct predominance of fully coated tools. The tool lives of the tools with uncoated clearance faces were lower by a very large amount.

One approach to improve the tool shape stability is through an increase of the wedge angle at the tip and/or flank edge. The modified tools were much less sensitive towards a high advancement of the cutting speed and/or feed. Since the trials showed a high potential for tool improvement but did not allow an assessment of the influence of the particular technological angles, further investigation appears to be promising.

Finally, a new face milling cutting tool concept which was developed at WZL has been presented. The concept tool has a number of theoretical advantages in relation to currently available tools. The consequent exploitation of these potentials might lead to a notable improvement of chip flow and tool life. A prerequisite for a successful implementation is, however, diligent development, followed by blade prototyping and testing in machining trials.

Acknowledgment

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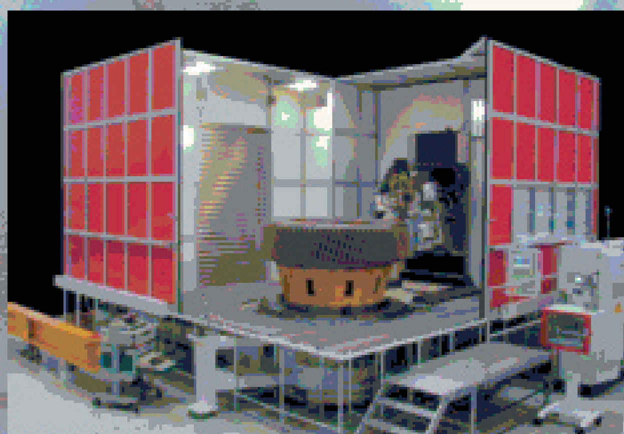
See the BIG PICTURE?



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Ondrives Manufactures "Showcase" Gearboxes

Ondrives Ltd. of Chesterfield, U.K., manufactured two gearboxes for a pharmaceutical company's teaching purposes.

According to the company's press release, the client presented gearboxes to its current students to show how internal gearing works in both an epicyclical arrangement and spur reduction box. To show how these mechanisms work, gearboxes were made with a clear Perspex casing that allowed a good view of the meshing gear teeth.

On the spur reduction box, a ratio of 500:1 was chosen so that a number of different gears would be moving at the same time, showing the possibilities of this type of box in a high ratio.

Both gearboxes have been mounted on mahogany plinths so they can be preserved for posterity.

Bourn & Koch Expands

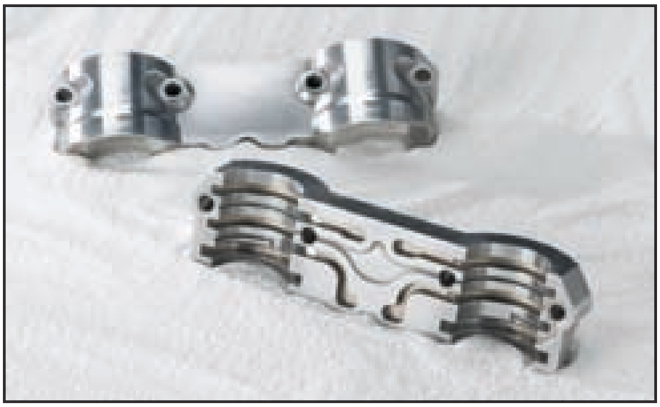
Bourn & Koch completed the addition of a new 20,000-square-foot warehouse and a new 20,000-square-foot, two-story engineering office building at its Kishwaukee Street site in Rockford, IL.

According to the company's press release, Bourn & Koch plans to improve the site through the fall of 2006. The company closed its Forest Hills Avenue site, which was acquired from Devlieg Bullard in 2004. The three million drawings of the portfolio of machine tools owned by Bourn & Koch are now secured on the 2nd floor of the new Kishwaukee Street engineering building.

Metal Powder Products Captures Design Excellence Award

The 2006 Powder Metallurgy Design Excellence Awards competition announced winners for outstanding new automobile engine and transmission parts made by the powder metallurgy process.

Sponsored by the Metal Powder Industries Federation, the competition showcased P/M's shape complexity, precision and special engineering properties. Awards were presented at the MPIF Annual Automotive Suppliers' luncheon at the SAE 2006 World Conference.



A P/M aluminum bearing cap made by a division of Metal Powder Products Co. won the grand prize in the automotive engine category. Designed originally for P/M, two caps are used in GM's new high feature V6 engine. It operates in the Cadillac CTS, SRX and CTX, Buick LaCrosse and Rendezvous and Saab 9-3 and is the first dual-overhead cam engine using a single cam cap across both camshafts, according to the company's press release.

Made to net shape, the multiple level part has a tensile strength of 17,000 psi and a hardness range of 85–90 HRH.

The Broachman Joins American Broach

Ken Nemec, a.k.a. "The Broachman," joined American Broach & Machine as the new marketing manager.

According to the company's press release, Nemec's primary responsibilities include building and coordinating a team of sales representatives, developing marketing plans, coordinating exhibit-related activities for IMTS and other regional shows as well as all other marketing activities.

Mahr Acquires Helios Messtechnik

The managing directors of Mahr Group signed an agreement for the company to acquire Helios Messtechnik GmbH & Co. KG.

The new corporation, Mahr Helios, will extend the Mahr Group's product portfolio in the field of shaft measurement. With the acquisition of Helios, the Mahr Group enhances its existing product range in horizontal length measurement with the brands SIP and Helios, as well as in the field of optical and tactile shaft measurement with the brands Helio-Pan and Helio-Scope.

A first combined exhibition occured at Control 2006, the international trade fair for quality assurance in Sinsheim, Germany.

Helios Messtechnik has a workforce of 75 employees and expects sales of about 10 million euros in 2006. The operating company is located in Dörzbach, Germany, and specializes in the fields of optical (non-contact) and tactile (contact) shaft measurement, as well as in precision gages.

The Helios Messtechnik GmbH & Co. KG acquisition complements Mahr GmbH Esslingen (precision gages) and Mahr GmbH Göttingen (precision length metrology and form and position tolerance testing). Thomas Keidel and Stephan Gais will act as managing directors of the newly formed corporation.

Schafer Gear Appoints General Manager

Randy Whitt

Schafer Gear Works appointed Randy Whitt as general manager of its South Bend production facility, with responsibility for the company's entire manufacturing operation, including engineering, quality assurance and scheduling.

Whitt was previously project manager for Federal Mogul's piston operations in South Bend, IN. Prior to that, he was plant manager of Federal Mogul's manufacturing facility in LaGrange, GA.

Emuge Opens New Manufacturing Center

Emuge Corp. moved to its new North American headquarters in West Boylston, MA.

According to the company's press release, the new 21,000-square-foot facility was custom designed with a technology center featuring a new Hermle 5-axis CNC machining center and an adjoining classroom/auditorium. The new facility will stock the complete line of Emuge products and house production manufacturing, engineering and application support. The building is strategically laid out for future manufacturing and expansion. The company plans to utilize the new facility to host future thread cutting, milling and workholding centers.

American Axles Breaks Ground for Chinese Facility

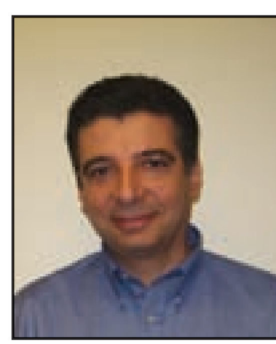
American Axle and Manufacturing broke ground for its Changshu Gear & Axle facility in Jiangsu Province, China.

According to its press release, the company acquired land use rights for 30 acres in an industrial park in the city of Changshu for the construction of a state-of-the-art, 180,000-square-foot, wholly owned manufacturing plant for its driveline systems. Products scheduled for manufacture include independent rear drive axles, power transfer units, integrated oil pan front axle module and driveshafts for Ssang Yong Motors, Beijing Benz DaimlerChrysler, General Motors and additional OEMs.

William A. Smith was appointed plant manager to oversee construction, build the organization and launch the product. Approximately 350 engineering and manufacturing positions will be created.

"We are extremely pleased to celebrate the ground breaking for AAM's first regional manufacturing operation here in Changshu, China," says Richard F. Dauch, AAM's executive vice president of worldwide manufacturing. "China and Asia represent a large and growing market that we intend to serve with AAM's latest technology products, processes and systems."

Wall Colmonoy Appoints Business Development Manager


Morris Warino

Morris Warino was appointed business development manager for Wall Colmonoy Corp. of Oklahoma City, OK.

According to the company's press release, Morris was national sales manager for a manufacturer of industrial process instrumentation for 17 years. Among his new responsibilities will be identifying new opportunities for WCC, Oklahoma City.

AGMA Announces New Board of Directors

Stan Blenke

The American Gear Manufacturers Association announced the 2006 board of directors at its annual meeting in March.

Stan Blenke, executive vice president of Schafer Gear Works, is the new AGMA chairman.

Rick Fullington, president of Milwaukee Gear Co., is the new vice chairman. Dennis Gimpert, president of Koepfer America, is treasurer. Dave Ballard, corporate manager of SEW-Eurodrive is chairman of the business management executive committee; Ed Lawson, director of metrology for Gleason/M&M Precision Systems Co., is chairman of the technical division executive committee; and Leslie Hennessy, vice president of strategic planning and business development for Lovejoy Inc., is chairman emeritus.

The complete 2006 board of directors are:
Dave Ballard—SEW-Eurodrive
Bryan Lammers—Caterpillar Inc.
Stan Blenke—Schafer Gear Works
Ed Lawson—Gleason/M&M Precision Systems Co.
Craig Danecki—Rexnord Geared Products
Jeff Lawton—Star Cutter Co.
Rick Fullington—Milwaukee Gear Co.
Lee Mason—Clarke Gear Co.
Dennis Gimpert—Koepfer America
Greg Porter—Standard Machine
Tom Grula—DuPont Engineering
Roland Ramberg—The Gear Works–Seattle Inc.
Sharon Haverstock—Scot Forge Co.
Andrew Sadanowicz—The Purdy Corp.
Leslie Hennessey—Lovejoy Inc.
Alan Seitz—Seitz Corp.
Martin Kapp—Kapp GmbH

Engineered Tool Opens Gear Cutting Division

Engineered Tools Corp. opened a division to manufacture and sell carbide and HSS stick blades and offer cutter head repair on bevel gear cutter bodies.

Cutter body replacement is currently taking place at the company's facility in Caro, MI, until the company moves to a new plant near Detroit later this year.

Ross Deneau, formerly vice president of A/W Systems, is manufacturing manager. John Ketterer is director of operations, and Ned Hoermann is sales manager.

Engineered Tools Corp. is a specialty manufacturer of precision cutting tools and wear parts in carbide, ceramic and HSS. According to the company's press release, ETC had been manufacturing stick blades for A/W from 2000–2005.

The new division has the equipment and capacity for 10,000 blades per month.

Mixing Solutions Names New Managing Director

John Brace was appointed managing director at Mixing Solutions. He will be responsible for its worldwide process agitation business in the Eastern Hemisphere.

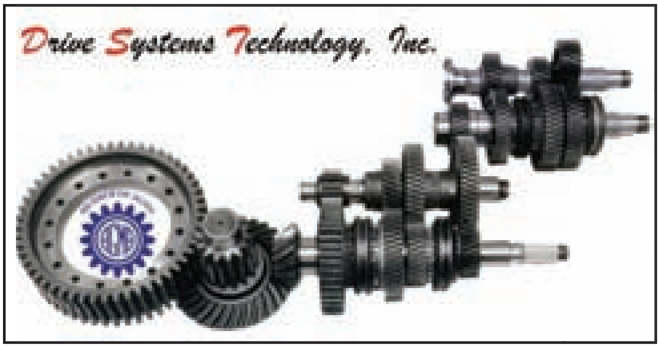
According to the company's press release, Brace will also be responsible for establishing a U.K.-based footprint for Mixing Solutions' American parent company, Philadelphia Gear.

Brace has worked in OEM and aftermarket operations and sales management in Europe with Dowty MECO, where he was manager of repair service. He was transferred to the U.S. as production manager and later general manager of Dowty MECO. His responsibilities included overseeing the Gulf Coast and Southeast regions, including sales, production, engineering and supply chain activities.

Carl Rapp, CEO of Philadelphia Gear, says, "John's vast experience in the industrial manufacturing sector puts him in an excellent position to help build both the MSL and PGC brands as we continue the worldwide expansion of both companies."

Mixing Solutions provides fluid mixing expertise with standard and custom-designed agitator drives and related equipment from 0.25–2,000 kW.

University of Wisconsin Offers Seminar for Gear Designers and Users



Gear buyers and users get a better understanding of gears, so they can make better decisions as customers. Gear manufacturers get a broader view of gear technology, which may make their companies more profitable.

Those are the practical benefits that Ray Drago, P.E., sees for the people who attend his course, "Advanced Gear Design & Theory." Held at the University of Wisconsin–Milwaukee, the three-day seminar for gear designers and users emphasizes selection, design, application and use of gears.

The benefit for gear buyers and users: "They understand the language better, the terminology better and make better consumers," says Drago, a gear technology consultant. "They're in less danger of getting the wrong product."

Drago says gear manufacturers may learn they missed gear manufacturing developments. He cites an attendee who first heard about high profile contact ratio spur gears in Drago's course. The gear manufacturer asked Drago about the gears and took Drago's answer back to his company, which improved its products as a result, making gears that ran smoother and quieter.

"They had a competitive advantage at very little cost to them," says Drago, who is founder and chief engineer of Drive Systems Technology Inc., a mechanical power transmission consultancy started in 1976.

Drago says attendees' gear experience can vary widely, with students ranging from college interns to chief engineers. Still, attendees must know elementary algebra, geometry and trigonometry. Knowledge of materials' basic strengths is helpful, but not essential.

The course will next be held June 28–30 by the UW–Milwaukee's School of Continuing Education.

In the seminar, students learn drawing data requirements, specifications, and formats, covering tolerancing and basic geometry data—both reference and required. They learn the basics of load capacity rating. Discussion of gear rating includes AGMA rating standards and explanation of models for assessing bending strength, durability and scoring hazard.

Attendees are taught about quality control, too, including AGMA quality recommendations. They're taught about composite and elemental inspection and about tests of tooth contact patterns, like rolling checks and single-flank tests.

Other topics cover types and selection of lubricant, types of additives and methods of applying lubricant. Gear failure modes are also discussed, and failed gears are presented to illustrate various modes.

Attendees learn about gear materials—ferrous and nonferrous—and heat treat processes for case hardening, including carburizing, nitriding, induction hardening and flame hardening.

Also, the course reviews gear generating processes—including hobbing, gear shaping, face milling, gear grinding—and gear forming processes—such as slotting, milling, broaching, precision forging and powder metallurgy.

The class has no defined size limit, but Drago says he prefers to keep the course to fewer than 40 students, so it can be interactive.

The class consists of PowerPoint presentations, with time allotted after each presentation for group questions and answers. Some presentations include videos. Two common videos cover gear manufacturing processes and the theory of gear tooth action.

Each attendee will receive a binder with 100+ pages of class materials. The materials will consist of color copies of the PowerPoint slides used in presentations and an extended supporting text for most covered topics.

Also, students may bring pictures, drawings and notes regarding their specific gear designs and applications to discuss with Drago after each day's class.

The course costs $1,095 per attendee and includes class materials, continental breakfasts and lunches. Students pay for their lodging and other meals themselves.

Attendees are responsible for getting to and from the airport and to and from the university's Center for Continuing Engineering Education. The center, however, is part of the Grand Avenue Mall; several hotels are part of the mall or are within a few blocks of it.

Students reserve their own hotel rooms, but hotel information is mailed with their enrollment confirmations. When reserving rooms, students should mention they're attending the UW–Milwaukee seminar to obtain the best rates.

UNIVERSITY of WISCONSIN
UWMILWAUKEE
School of Continuing Education

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Internet: ***www.sce-eng.uwm.edu***

Shorter Version of the MPIF/APMI International Conference

The MPIF/APMI International Conference on Powder Metallurgy & Particulate Materials will be held June 18–21 at the Manchester Grand Hyatt in San Diego and has been condensed into a three-day format.

Sponsored by the Metal Powder Industries Federation and APMI International, this year's conference is a 130-booth marketplace featuring 80 companies. The expo will be open Monday, June 19, and Tuesday, June 20. The exhibition will not conflict with major conference events, general sessions or program luncheons. Photomicrographs of unique P/M microstructures will be on display daily in the exhibit hall.

The session opens with a keynote presentation called "China and Your Bottom Line." Additional technical sessions covering topics as diverse as alloys, sintering, bearings, hard ceramics and titanium component technologies take place June 20–21.

New to this year's show is the poster program, featuring a display of international P/M-related posters. Authors will be available to discuss their posters from 2:15–3:45 on June 20. Awards for "Outstanding Poster" and "Poster of Merit" will be posted on June 20 prior to the designated discussion section.

For early arrivals, a golf excursion and a land-and-sea tour of San Diego are offered. Other highlights include "An Evening in Tijuana," which takes place on June 20. Registration and additional information about these events is available at the show's website: *www.mpif.org/meetings/2006*.

Full conference registration ranges from $1,150–$1,600 and includes two luncheons, a dinner, technical events, exhibits, preprints and proceedings. Additional pricing structures are offered. A block of hotel rooms has been reserved at the Manchester Grand Hyatt, starting at $195 per night.

EVENTS

June 5–7—AGMA Regional Gear School. Star SU facility, Hoffman Estates, IL. Concentrates on the relationship between the basic geometry of parallel axis gears and their inspection and manufacturing processes with an emphasis on logical troubleshooting. $750. Additional courses are planned for the West Coast and Southeast later in 2006. For more information, contact the Gear Consulting Group by telephone at (231) 829-3760.

June 18–21—International Conference on Powder Metallurgy and Particulate Materials. Manchester Grand Hyatt, San Diego, CA. Formerly known as PM²TEC, this conference is offered in a more condensed format for 2006. The exhibition features 130 booths showcasing PM equipment, powders, products and services. Special rates exist for MPIF and APMI members and speakers and session chairs. A variety of price packages from $45–$1,600 are available. More details are available on page 58. For more information, contact the Metal Powder Industries Federation by telephone at (609) 452-6692 or on the Internet at *www.mpif.org*.

June 19–22—Vibration Institute Symposium and Annual Meeting. Galt House Hotel & Suites, Louisville, KY. Technical papers in various vibration analysis disciplines including rolling element bearings, precision spindles, journal bearings, gearboxes, modal analysis/ODS, alarm settings and others. Numerous short courses and ISO certification tests will be offered as well as a vendor display area. Prices range from $100–$1,050. For more information, contact the Vibration Institute by phone at (630) 654-2254 or on the Internet at *www.vibinst.org*.

June 26–28—Gears in Vehicles 2006. Kultur-und Congress-Centrum, Graf-Zeppelin-Haus, Fredrichshafen, Germany. The largest gathering of the vehicle gears sector, the conference focuses on transmission requirements, gear shifting systems, transmission concepts, component optimization, dual-clutch transmission, hybrid CVT, commercial vehicle transmissions, converter automatic transmissions, 4WD systems and mechatronics. German and English are the official languages of the conference. For more information, including the price structure and registration details, visit the VDI Society Development, Design and Marketing website at *www.vdi.de/gif2006*.

June 28–30—Advanced Gear Design and Theory. University of Wisconsin-Milwaukee School of Continuing Education. This course is designed for the designer, user and beginning gear technologist. The main emphasis is on proper selection, design, application and use, rather than fabrication. A knowledge of geometry, trigonometry and elementary algebra is required. Course is taught by Ray Drago. $1,095. For more information, see page 57 and visit the school’s website at *www.sce-eng.uwm.edu*.

ADVERTISER INDEX

Use this index to contact any advertiser in this issue.

ADVERTISER	PAGE #	PHONE	E-MAIL	INTERNET
Aero Gear Inc.	58	(860) 688-0888	*buygears@aerogear.com*	*www.aerogear.com*
American Metal Treating Co.	63	(216) 431-4492	*bruce@americanmetaltreating.com*	*www.americanmetaltreating.com*
Applied Process	51	(734) 464-8000	*sales@appliedprocess.com*	*www.appliedprocess.com*
Arrow Gear Co.	4	(630) 969-7640		*www.arrowgear.com*
A/W Systems Co.	6	(248) 524-0778		
B&R Machine & Gear Corp.	IBC	(731) 456-2636, (800) 238-0651	*inquiry@brgear.com*	*www.brgear.com*
Becker GearMeisters Inc.	62	(734) 747-9670	*maagmachines@yahoo.com*	*www.maagmachines.com*
Clifford-Jacobs Forging Co.	60	(217) 352-5172	*sales@clifford-jacobs.com*	*www.clifford-jacobs.com*
Cole Manufacturing Systems	62	(248) 601-8145	*dsmith@colemfgsystems.com*	*www.colemfgsystems.com*
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De Ci Ma	23	+(39) (051) 611-7811		*www.decimaspa.it*
Deco Products Co.	55	(800) 327-9751, (563) 382-4264	*jraptes@decoprod.com*	*www.decoprod.com*
Donner + Pfister	24	(585) 586-5746	*gearmet@frontiernet.net*	*www.dpag.ch*
Dura-Bar	39	(800) BAR-MILL	*salesgt@dura-bar.com*	*www.steelalternative.com*
Eldec Induction USA Inc.	63	(248) 364-4750	*mail@eldec-usa.com*	*www.eldec.de*
Engineered Tools Corp.	8	(989) 673-8733		*www.engineeredtools.com*
Fässler Corp.	10	(414) 769-0072	*usa@faessler-ag.ch*	*www.faessler-ag.ch*
First Gear	53	(260) 490-3238	*greg@first-gear.com*	*www.first-gear.com*
Forest City Gear Co.	63	(815) 623-2168		*www.fcgear.com*
Gear Consulting Group	Inserted Card	(259) 523-4993	*gearconsulting@aol.com*	*www.gearconsultinggroup.com*
Gear Manufacturing Inc.	63	(800) 773-GEAR		*www.gearmanufacturing.com*
Gear Motions Inc.	63	(315) 488-0100	*ronwri@nixongear.com*	*www.gearmotions.com*
The Gear Works–Seattle Inc.	63	(206) 762-3333	*sales@thegearworks.com*	*www.thegearworks.com*
Gleason Corp.	OBC	(585) 473-1000	*dmelton@gleason.com*	*www.gleason.com*
Gleason Cutting Tools Corp.	62	(815) 877-8900	*dmelton@gleason.com*	*www.gleason.com*
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IMTS 2006	56			*www.imts.com*
Insco Corp.	25	(978) 448-6368	*sales@inscocorp.com*	*www.inscocorp.com*
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Proto Manufacturing	52	(519) 737-6330		*www.protoxrd.com*
The Purdy Corp.	12	(860) 649-0000		*www.purdytransmissions.com*
Quality Transmission Components	59	(516) 437-6700		*www.qtcgears.com*
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SETCO	25	(800) 543-0470	*sales@setcousa.com*	*www.setcousa.com*
The Shot Peener magazine	58	(800) 832-5653, (574) 256-5001		*www.shotpeener.com*
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Stock Drive Products/Sterling Instrument	63	(516) 328-3300		*www.sdp-si.com/e-store*
Sunnen Products Co.	16	(800) 325-3670		*www.sunnen.com*
Ticona	14	(800) 833-4882		*www.ticona.com*
TSA America LLC	4	(440) 614-0170	*sales@tsageartools.com*	*www.tsageartools.com*
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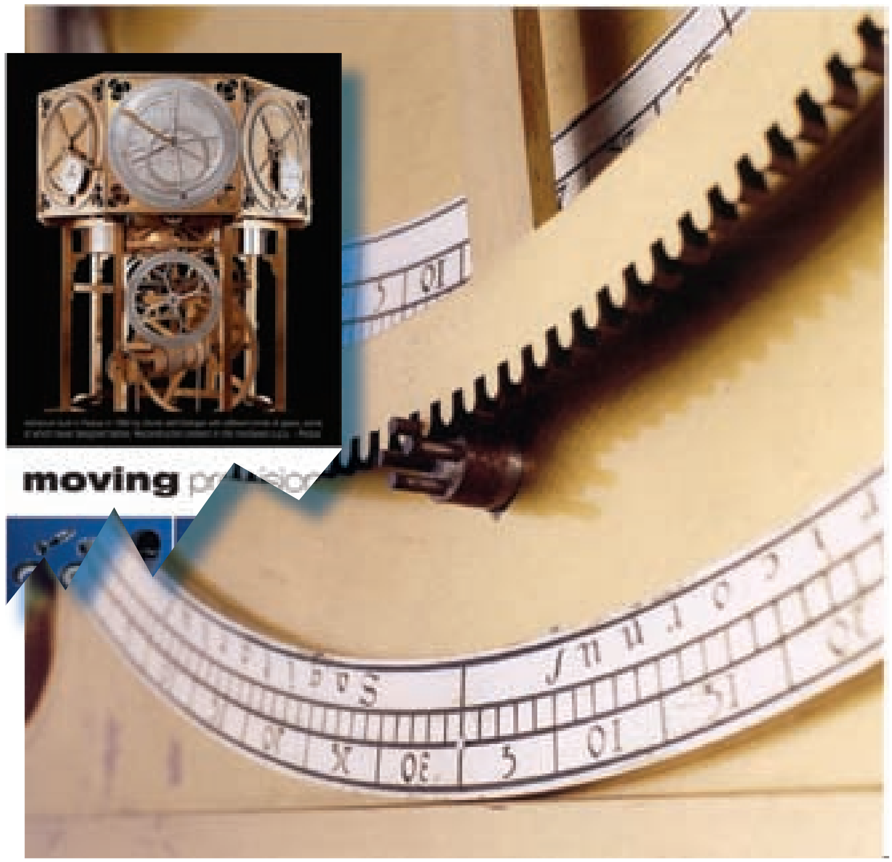
HEAT TREATING

HELP WANTED

Products of Padua: mG miniGears and an Ancient Geared Clock that Tracks Planets' Movements



Some things take time, but a magazine ad more than 600 years in the making? That's unusual, but it's one way of looking at the ads for mG miniGears that featured a complex, highly geared, planet-tracking clock called the Astrarium.

The Addendum team had noticed the clock in miniGears' ads in *Gear Technology*; the last one appeared in the Jan./Feb. '04 issue, and the top of it is at right. But we only recently learned the story behind the clock, why the ads featured it, and how a man's interest in the Astrarium led to the creation of two books and a CD about the ancient device.

The story starts with Giovanni Dondi, who lived in 14th-century Italy, in Padua. Although a doctor, Dondi was interested in astronomy and clockmaking, so much so he designed and built the original Astrarium in the 1360s. More than a regular clock, the Astrarium uses a year wheel and a geared assembly to track the movements of the sun, moon and five planets: Venus, Mercury, Saturn, Jupiter and Mars.

Dondi didn't include the other three planets because he didn't know about them. No one did. It's the 14th century, after all; Uranus, Neptune and Pluto hadn't been discovered yet.

Still, the Astrarium was a complicated device and needed 89 gears—including spurs, helicals, internals and ellipticals—to perform all its functions. Besides making it, Dondi wrote a manuscript, the *Tractatus Astrarii*, detailing how to build the clock; he left a way for the Astrarium to be brought out of the Middle Ages.

More than 600 years later, the Astrarium was brought out; it was re-created, built on commission in 1974. Two years later, an Italian engineer, Vincenzo de' Stefani, founded mG miniGears SpA. Already a resident of Padua, he decided to locate his gear-manufacturing company there.

The coincidence of Dondi and de' Stefani both being Paduans might have come to nothing if not for one other coincidence. "I am a lover and a collector of ancient clocks," de' Stefani says.

Aware of the Astrarium, de' Stefani was especially interested in it because of its gears and its connection with Padua. He decided to feature it in many of his company's advertisements.

His interest led him to a publishing project: two books and a CD. Creating them involved a lot of work and several de' Stefani friends also interested in clockmaking and Padua's medieval history. One friend, Aldo Bullo, was especially important. A Latinist, he could translate the ancient version of the *Tractatus Astrarii*. Work started in '01 and was finished two years later.

The thinner of the books is Dondi's manuscript, reproduced page by page, cover to cover, in full-sized color photographs. In the thick book, each left page is a black-and-white photo of a page from Dondi's manuscript. Each right page shows the manuscript's handwritten Latin as typed Latin and typed Italian. The CD includes color PDFs of the manuscript and its English translation.

Clearly, the books and CD are an extension of de' Stefani's local pride in the Astrarium: "It is a glory of Padua."

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