GEAR TECHNOLOGY

JULY/AUGUST 2006

The Journal of Gear Manufacturing

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AUTOMOTIVE GEARS

- Auto News
- Transmission Trends

TECHNICAL ARTICLES

- Circular Fillets for Increased Capacity
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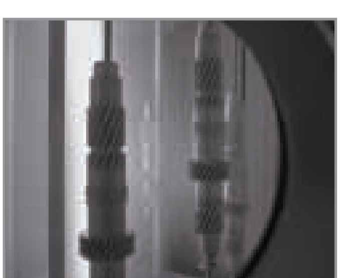

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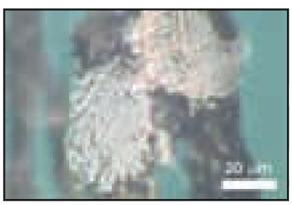
GEAR TECHNOLOGY

JULY/AUGUST 2006 *The Journal of Gear Manufacturing*

AUTOMOTIVE GEARS

18

TECHNICAL ARTICLES



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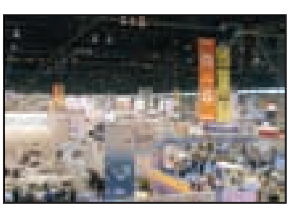

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On the cover: GM's 6L80 6-speed transmission (illustration by David Kimble); the 2007 Saturn Aura; and the 2007 Lincoln MKX.

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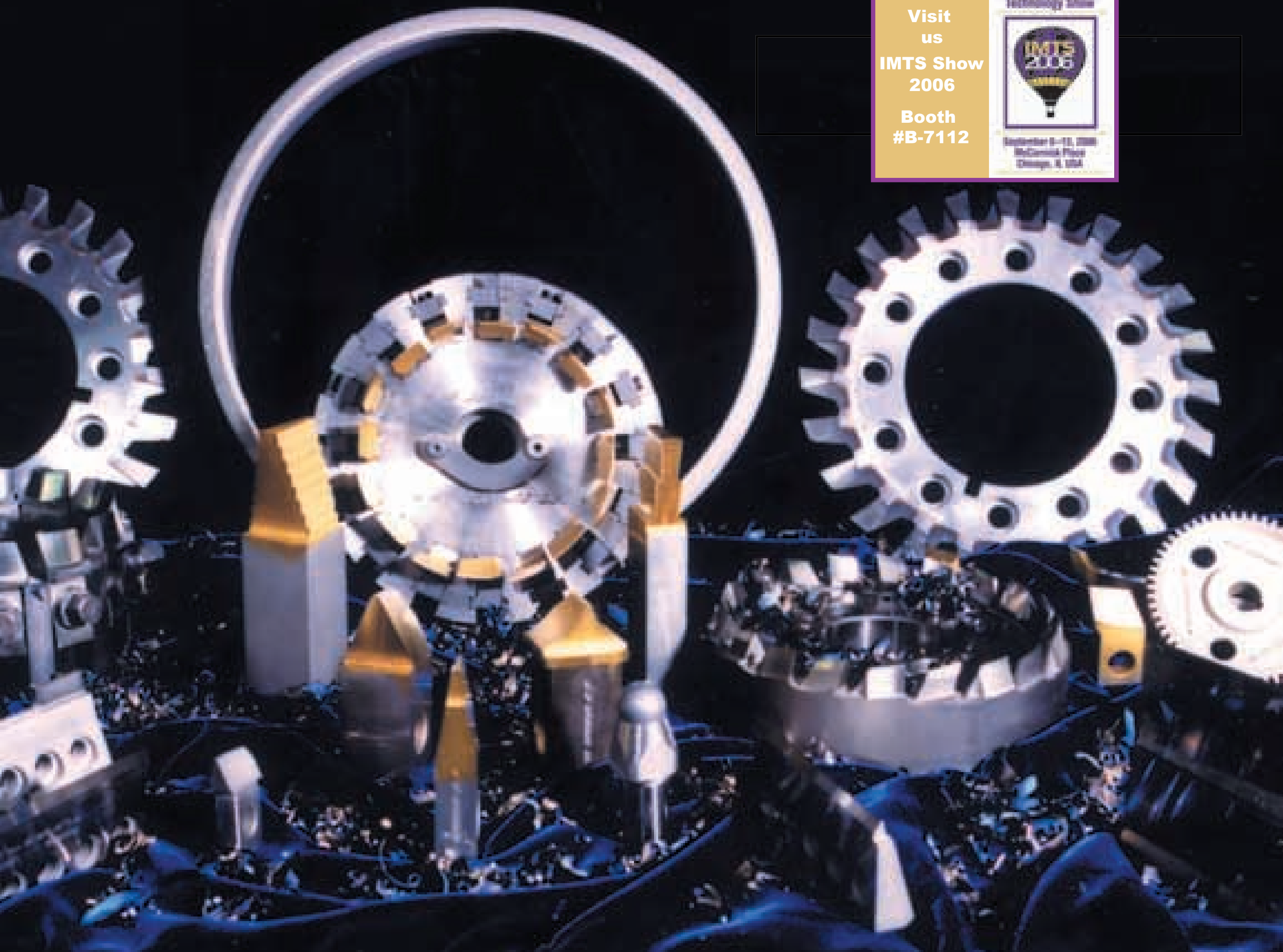

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Who Pays *Your* Salary?

I just finished conducting annual reviews with our employees. About this time, I meet with each employee to review progress and goals from the past year, set new goals for the coming year and review the financial performance of the entire company.

These reviews are a useful opportunity to get feedback about what is working and what isn't, from the employees' perspectives. We ask for suggestions about what we and they could and should do better. Some of the best decisions we've implemented have come out of these meetings.

Of course, the employees are always interested to know whether they'll be getting raises or bonuses—and if so, how much. But while conducting reviews this year, I found myself explaining the idea that their salaries, raises and bonuses really don't come from the company—or from a specific individual. They, we—all of us—get our paychecks from our customers.

Of course, customers don't come to us and buy our products and services because of their concern for our welfare or the welfare of our families. Customers come to us because they need our products or services to help them solve a problem or accomplish something they couldn't do themselves. If we don't provide these products and services to our customers, at prices they can afford, they will find suppliers that do.

In publishing, our first contacts are the advertisers. I need those companies to be successful using this product. To accomplish that, I spend a lot of time thinking about their customers (that would be you guys, the readers). If we didn't have information of interest to you, the readers, and if we didn't deliver the magazine to you, then our advertisers would have no economic reason to advertise.

Thinking about this concept, I started to wonder, Who really is the customer? The first reaction is that the customer is the company to whom we directly sell or ship our products. But in fact, the real customer may be farther down the line. Maybe it's your customer's customer who is the end user.

You can follow that line of reasoning until, eventually, you get back to individual consumers. You might not think *Gear Technology* has much to do with the average person, but in fact, it does. For example, our magazine sells advertising to a gear machine tool manufacturer. The machine tool manufacturer sells his products to an automobile transmission manufacturer. The transmission goes into an automobile that eventually ends up in a consumer's driveway. And the chain might even become a circle if that consumer is one of my employees, or even me.

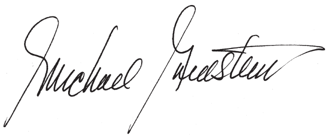
As a publisher, while my first concern is not individual automobile consumers, their wants and needs ultimately play a role in our decision-making process. For example, when we choose to run a technical article on gear noise, doesn't it ultimately benefit the end consumer?

Ultimately, a business has to pay attention to his complete chain of customers, from the one who pays him directly to the one who ultimately uses the end product. Of course, the ones closest to us on the chain are the ones we have to pay the most attention to, but often we can best serve them by thinking down the chain to the next level.

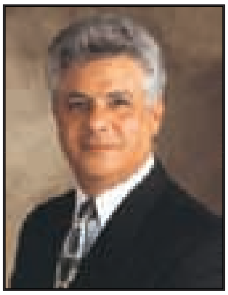
In our annual reviews, I encouraged each of my employees to think about our customers and how we can better serve them and to consider new ways to make our customers more successful with our products.

But this isn't an exercise just for publishers. The idea holds true in many businesses, including gear manufacturing.

So, who really is your customer? How can you make him more successful with your products? How far down the chain are you thinking? Have you thought about your customer's customer lately? If not, I encourage you to do so. After all, it's not your boss who pays your salary or who decides if you get a raise. Ultimately, it's the customer who does. Concentrate on making him successful with your products and services, and everyone benefits.



Michael Goldstein,
Publisher & Editor-in-Chief



Drake's Newest Thread Grinder Utilizes Robot Load/Unload System

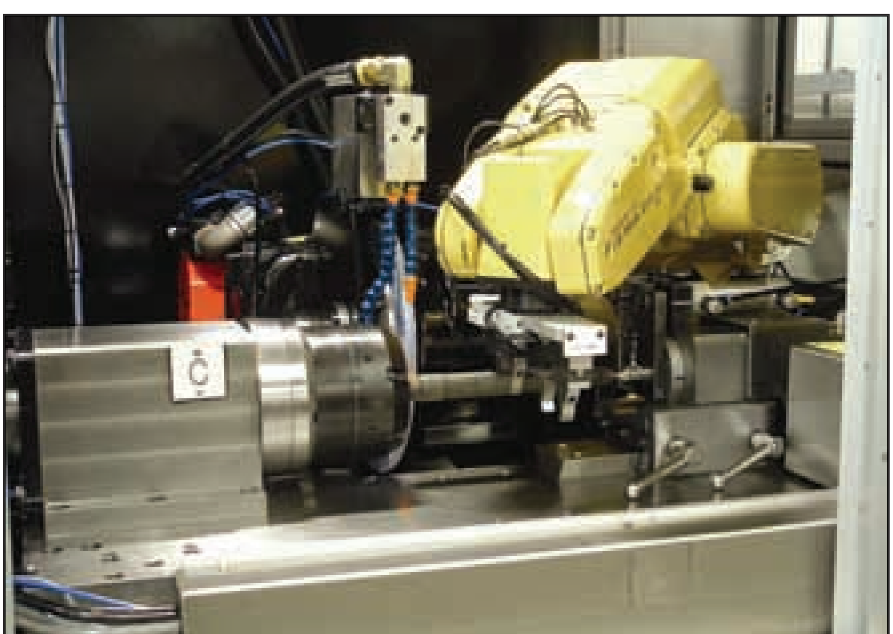
The GS: TE-LM thread grinder from Drake Manufacturing is fitted with a robot load/unload system that provides maximum throughput for high-volume production of ground threads.

The machine was first introduced at IMTS 2004, and Drake has been tweaking its features to unveil to the manufacturing community in time for IMTS 2006.

According to the company's press release, the machine will be tooled to grind the gear and worm profiles and bearing journals on a power steering pinion with vitrified CBN wheels. The machine is a full helix thread grinder equipped with a 180° power helix, linear motors and ways. It also will have the latest Fanuc controls programmed with customer parts, acoustic touch dressing, automatically generated wheel forms and a mineral-filled cast polymer base.

CEO John Drake says the company aims for cross-market appeal to both low-volume and high-volume manufacturers, from job shops to automotive customers. The grinder comes tooled for power steering of ball screws and worms. Drake adds the grinder is suitable for taps, multi-start worms, electronic steering components, fast lead ball screws, medical bone screws as well as gearbox, transmission and speed reducer components.

"The gear machine that's generally offered is the 350 mm. However, we've built up to one meter in overall length, and it's capable of going up to two or three meters now. The ballscrew manufacturers target a one, two, or three meter length product, and we can provide a longer bed length for them than for gear manufacturers," says Drake.



The auto load system consists of an infeed conveyor, robot with dual grippers and exit conveyor. The infeed conveyors allow faster changeovers on a family of parts. During the cycle, the robot loader lifts a new blank off the infeed conveyor and deposits a finished part on the exit conveyor at the same time.

Also, robot cycles are coordinated with Drake PartSmart programs for faster changeovers. The robot system is preprogrammed to automatically respond to part dimension changes as entered in the menu during changeover. The system can conform to customer restocking intervals and can be integrated in manufacturing cells. Drake has programmed the grinding and load/unload sequence into the control. Changes in part lengths are accommodated by adjusting the conveyor width and headstock position.

The company has integrated robot systems in its machines for years as part of its package for higher volume customers. The GS: TE-LM is the latest and most sophisticated of the robot-integrated machines.

"We have historically been a ball screw house," says Drake. "Today's speeds require competitive linear motors, so we're offering low maintenance and much lower moving parts. So far, we've experienced great acceptance in the screw and worm grinding market and are excited to offer the same technology for gear customers."

Drake also plans to bring to IMTS a "mini" thread grinder that targets smaller parts. With a workpiece envelope of 100 mm x 100 mm, this new grinder is designed for high volume, precision threaded parts manufacturers in the cutting tool, automotive, aerospace and medical equipment industries.

The company recently shipped a "mini" with a cycling autoloader configured to grind M0.8–0.25 taps for threading holes in cell phone circuit boards. The mini is suitable for grinding small taps, thread roles, thread gages, worms, ball screws, e-steering components, aerospace fasteners and surgical bone screws.

The mini is also equipped with the PartSmart software, so customers' parts are pre-programmed into its system. Operators follow the screen prompts and enter values.

With a footprint of 1.2 m x 1.8 m and weighing 3,000 kg, the grinder can be integrated into manufacturing cells and transported via forklift.

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Norton's New Gear Grinding Wheels Increase Life of Parallel Axis Spur Gears

The new BRGg VPHS high speed grinding wheels from Saint Gobain are designed to reduce cycle times by increasing metal removal rates. The increase in removal rates has been applied without metallurgical damage to the surface being ground and without loss of dimensional tolerance. Wheel life is lengthened, and the BRG high speed wheels can run up to 60 m/s.

Phil Plainte, applications engineer at Saint-Gobain, says the BRG wheels can be used on new and older machines.

"Machine builders are building smaller, more compact, versatile, faster-running machines. This is driven by the end-user who is looking for higher quality gears with much faster cycle times," he says. "The BRG product addresses the demand for higher quality gears and the new machines that run higher speed spindles and short grind cycles. The BRG can also be used on older machines producing high quality parts."

The BRG line includes both form grinding wheels and threaded grinding wheels, and they are available with a variety of abrasive types and grit sizes.

In addition, Plainte says up to six starts/leads can be put into the threaded grinding wheel form to enhance the machine/process time. The BRG abrasive and bond help generate gears with tight geometric tolerances and burn-free surfaces. In some cases, Plainte says, the grinding action can enhance the compressive residual stresses of the gear.

In one instance, the wheels were tested on a 7.3635"-diameter, 56-tooth helical gear and generated 28 good parts before needing to be dressed. Conversely, a competing grinding wheel produced 17 parts before it required dressing. In addition, when tested on a 2.8117" spur gear, the BRGg wheel produced more good parts between dressings and simultaneously reduced per part grinding time.

"Original cycle time on one particular part was approximately four minutes; now they average around 1.5 minutes. We can't say the reduction in cycle time is all BRG," says Plainte, "but the product definitely plays a big part in cycle reduction. The machine is a new Gleason 245TWG high speed grinder."

The gear grinding wheels will be displayed for the first time at IMTS and are offered with conventional aluminum oxide to the more advanced high performance ceramic grains. A complete line of dressing tools will also be displayed.

Norton Abrasives is optimistic about the products' reception at IMTS.

"There are three dynamics taking place in the gear industry right now. Every manufacturer is under pressure to improve production and efficiency. If you don't, you won't survive," Plainte says. "The other dynamic is drivetrain compression—more horsepower moving through smaller, lighter drivetrains. This can only be accomplished through better quality gears. Lastly, aerospace is in the middle of a seven-year up cycle, and gear manufacturers are struggling to keep up."

"The customers participating in this environment are investing in new equipment, such as high speed machines, and better tooling, like grinding wheels," he continues. "We're getting really positive feedback from customers in North America and Europe."

For more information:
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E-mail: *Barry.D.Cole@saint-gobain.com*
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Kinefac Introduces New Automated Production System

Kinefac is introducing a new Robo-Roller, which is a quick changeover, automated production system for rolling threads, knurls, worms and other forms on bolts, shafts and other parts. The system integrates a Fanuc robot with Kinefac's MC-15 FI CNC Kine-Roller.

Robo-Roller users perform automated rolling from a pallet system, bulk feed unit, hopper feed, manual load/unload station or in a conveyorized system without safety guarding. The Robo-Roller can handle parts weighing up to 3 lbs. if there is some grippable feature.

The Kine-Trol computer numerical control rolling and part handling system on the Kine-Roller provides operator support to specify the thread size, thread length and rolling die specifications to establish the rolling cycle and die match. Robot programs are set up using a teaching pendant and then stored.

Each die spindle has an independent servo drive system allowing numerically programmed rotational die match. The heavy-duty spindle and die actuation system has a maximum radial die load up to 70,000 lbs. and can accommodate dies up to 6.75" in diameter and faces of 4.5".

Kinefac's Kine-Spin/Barret Division is also introducing a new portable unit for effectively removing solid contaminants from coolant and lubricant fluids used in grinding, lapping, honing, wire drawing, deep drawing and other metal-working processes.

The unit uses the Barrett Clarifuge to remove the abrasive swarf, material coating residue, tramp metal particles and similar contaminants that are generated in metalworking.

The centrifuge and clari-clean centrifuge cowl cleaner are mounted on a cart and supplied with an 8 GPM circulation system. According to the company's press release, this allows the fluid cleaning operation to be brought to the machine and operated until the coolant or lubricant supply and machine reservoir

and sump are restored to normal operating conditions.

With the new system, removed contaminants are collected in a standard drum located on the cart, and therefore, the portable fluid recover unit can perform numerous machine operations before it must be returned to the location where contaminants are discarded in a hazard-controlled manner.

For more information:
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Worcester, MA 01603
Phone: (508) 754-6891
E-mail: *sales@kinefac.com*
Internet: *www.kinefac.com*

Boston Gear Publishes New Gearhead Brochure

Boston Gear has published a full-color, 36-page brochure to introduce new high precision helical planetary gearheads.

The gearheads are designed for direct attachment to popular servo and stepper motors.

According to the company's press release, planetary gearheads combine the load-sharing attributes of multiple-tooth contacts with balanced operation at high speeds. That makes them suitable for servo-driven solutions to packaging, medical, material handling, robotics, automotive, and other high-production applications. Features include a patented planet carrier design, patented input and output sealing designs, and special heat and surface treatments.

The brochure provides detailed information on the complete line of Boston gearhead products, including specifications, selection information, motor mounting instructions, and torque tightening recommendations.

For more information:
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Unveiled at the 2006 North American Auto Show in January, the 2007 Lexus 460 luxury sedan will be the world's first mass-produced car equipped with an 8-speed automatic transmission.

NEW TRANSMISSIONS

Drive Automotive Gear Industry

GM Ramps Up 6-Speed Production

By 2008, GM plans to sell one million vehicles annually with six-speed automatic transmissions. Recent announcements demonstrate the company's efforts to achieve that goal.

Already, GM sells a number of models with six-speed transmissions, including the GMC Yukon Denali; the Cadillac XLR-V, STS-V and Escalade lineup; and the Chevrolet Corvette. GM's full-size heavy-duty pickups can also be equipped with an Allison 1000 six-speed automatic.

Some new 2007 model year vehicles will be equipped with brand new six-speed automatic transmissions, according to a GM press release. New six-speed variants in the Hydra-Matic line were announced in May, including the 6T70 and 6T75 for front-wheel and all-wheel drive vehicles, and the 6L50 for rear-wheel and all-wheel drive vehicles.

Advantages of a six-speed automatic transmission include improved fuel economy, performance and smoother shifting. The six-speed automatic allows a reduced engine rpm at highway cruising speeds, which reduces engine wear, reduces noise and improves fuel economy.

"It's a best-of-both-worlds scenario with the six-speed automatic delivering great performance and enabling improved fuel economy," said Jim Lanzon, executive director of transmission engineering for GM Powertrain. "It is almost like having two transmissions in one—the high numerical first gear provides tremendous off-the-line acceleration, but the transmission is able to use the six gears to evenly distribute the torque and settle at an overdrive gear that helps deliver great fuel economy."

Lanzon also announced that the company would expand its goals to introduce 10 new variants of six-speed automatic transmissions by 2010, resulting in production of three million six-speeds annually.

The Hydra-Matic 6T70 will debut on the Saturn Aura and on one version of the Pontiac G6. The 6T75 will be offered on the Saturn Outlook.

The 6T70 and 6T75 were co-developed with Ford. According to the press release, this partnership allowed the new transmissions to reach production in less time and at about half the cost they would have required each company to develop on their own.

The 6L50, which debuts on the 2007 Cadillac STS performance sedan and SRX crossover SUVs, is based on the larger Hydra-Matic 6L80 featured in 2006 models.

In order to meet production levels for these new transmission programs, GM recently announced a number of plant expansions and investments.

In May, the company announced a $170 million expansion of transmission production at its Willow Run facility in Ypsilanti, MI. Production of the 6L80 transmission began there in 2005, and production of the 6L50 will be added.

"A new team concept arrangement was developed for six-speed production, which includes hourly members of UAW local 735 and salaried employees," said Kingsley P. Wootton, plant manager at Ypsilanti Transmission Operations. "The team concept includes cohesive teams of four to six members, with a designated leader that assists the team, ensuring an efficient operation. Team members work with engineering personnel to provide input on product design and its impact on manufacturing, with a goal of 'zero defects.'"

In June, GM announced a $332 million expansion of its Warren, MI, transmission manufacturing facility, in order to accommodate the production of the front-wheel-drive six-speeds.

With the latest announcement, investment by GM on six-speed transmissions since 2003 totals $1.7 billion, according to the company's release.

"Raising the bar on the execution of new products is a critical part of our turnaround plan, and the broad application of six-speed transmissions plays a key role in GM's strategy to be a leader in offering a range of fuel-efficient, fun-to-drive vehicles," said GM Chairman and CEO Rick Wagoner.

In February, GM announced a $500 million investment to produce six-speeds at its Toledo, OH, plant.

Honda Begins Production at Georgia Transmission Plant

Honda Precision Parts of Georgia LLC (HPPG) began mass production of transmissions in May. The company also announced plans to expand production capabilities and employment at the Tallapoosa, GA, plant.

HPPG manufactures five-speed automatic transmissions for the Honda Odyssey minivans and Pilot sport utility vehicles assembled at Honda's plant in Lincoln, AL, 60 miles from Tallapoosa. When the new facility is at full capacity, it is expected to produce 300,000 transmissions per year, according to Honda's press release.

Combined with Honda's Ohio transmission manufacturing locations, the company now has the capacity to produce one million automatic transmissions per year in the United States.

The announced expansion will add 100,000 square feet of factory space and 40 employees, bringing HPPG to 350,000 square feet and 440 employees. The additional $50 million investment will add machining and aluminum die-casting capabilities.

GM Chairman and CEO Rick Wagoner (left) and UAW VP Richard Shoemaker announcing an expansion of six-speed production in June.

GM employee Donna Ramsey on the Hydra-Matic assembly line of Willow Run.

Transmission production began in May at Honda Precision Parts of Georgia.

The 2007 Lincoln MKX will feature the 6F 6-speed transmission co-developed by Ford & GM.

Vacuum Carburizing Critical to Ford Transmission

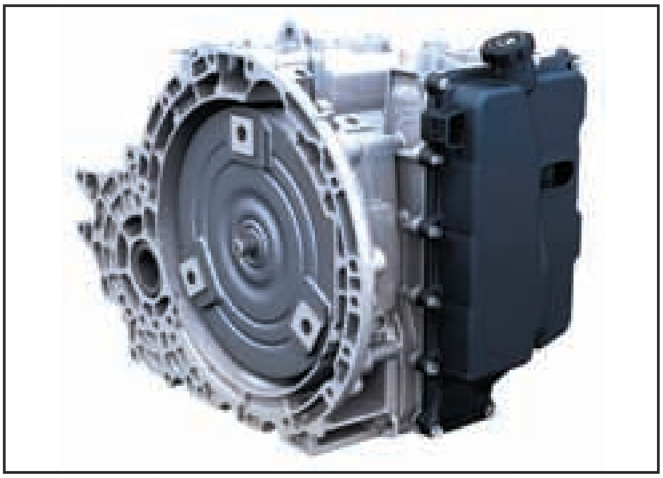
Ford Motor Co. has decided to develop a vacuum carburizing system with a high-pressure gas quench to improve dimensional control in distortion-prone components, including the 211 ring gear for its 6F transmission. This gear needed to be produced within tolerances that could not be achieved with traditional gas carburizing and quenching technology.

"We were confident that gas quenching would serve the purpose, but a lot of development work by the team was needed to prove out the heat-treat processes and meet dimensional requirements," said Bill Dowling, a 6-Sigma Black Belt from Ford's automatic transmission operations, who headed up the project.

The team developed a process that reduced distortion 50 percent and also improved fatigue properties compared with the traditional process. Vacuum carburizing also allows for the use of a lower-cost steel alloy while still meeting all the requirements for the parts.

"The savings on the material amount to several cents a pound," said Dowling. "Considering the volume of transmission gears produced at Sharonville on the 6F program, the savings are fairly substantial."

The 6F transmission is a six-speed, front-wheel drive automatic transmission that was co-developed with General Motors. It will be used first in the 2007 Ford Edge and 2007 Lincoln MKX crossover utility vehicles.



6F Transmission.

The 2007 Ford Edge will feature the 6F transmission.

Getrag and Bosch to Develop Hybrid Dual-Clutch Transmissions

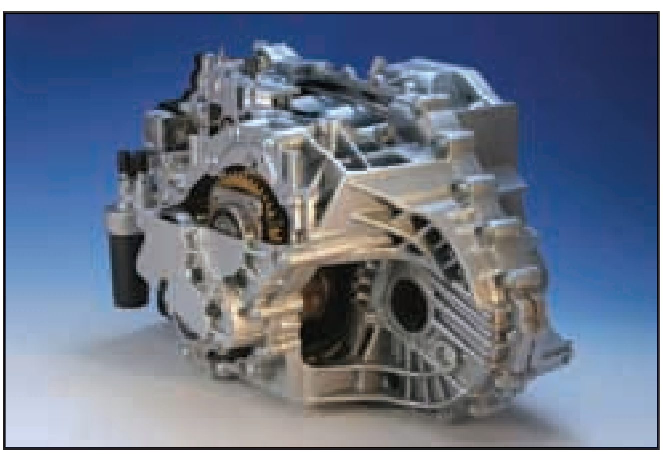
Getrag and Bosch have agreed to work together on hybrid transmission systems that use dual-clutch transmissions and electric final-drive units, according to a Getrag press release.

Key aspects of the collaboration include the integration of mechanical and electrical components and the development of appropriate software. Collaboration work also involves the development of final drive units with directly integrated electric motors.

Hybrid drive systems combine an internal combustion engine with an electric drive. In the case of parallel hybrids, the electric drive is fitted directly into the power flow of the drivetrain. The electric drive is used as a motor when accelerating and is used to convert kinetic energy into electrical energy when braking.

The dual-clutch transmission enables energy-efficient automatic shifting. Hybrid drive systems are designed to significantly reduce fuel consumption, especially in urban traffic. According to the release, the partners expect fuel savings of up to 20 percent compared with drive systems relying on conventional gasoline engines.

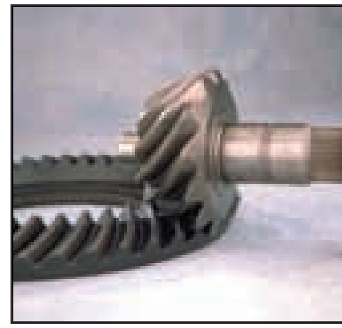
"The cooperation we have established with the world's largest supplier to the automotive industry is highly promising for the future development of hybrid drive technology," said Dieter Schlenkerman, chairman of the board of the Getrag Group. "The know-how of both companies will put us in the position to offer our customers throughout the world cutting-edge solutions for future drive concepts."



A dual-clutch transmission from Getrag.

Dana's New GenTech Gears Provide Quieter, Smoother, Stronger Operation

Dana Corp.'s Commercial Vehicle Systems group has announced that GenTech™, a new generation of heavy-duty drive axle gearing, is now available as a production shipment option.



According to the company's press release, GenTech hypoid gearing provides reductions in noise, vibration and harshness for medium- and heavy-duty vehicle applications that require increased torque capacity and maximum durability. GenTech gearing is also available for school buses, motor coaches and recreational vehicles.

"Dana's GenTech gearing is a proprietary system utilizing the latest computer-aided design methods, premium manufacturing processes, tightly controlled heat-treat and material specifications and advanced in-line testing," said Leo Wenstrup, Dana's senior product manager, drive axles. "We have optimized the tooth geometry and contact pattern by leveraging Dana's experience in automotive gear technology through the use of premium processes, advanced gear tooth cutting machinery and tools, and new manufacturing equipment."

GenTech gearing is initially available in Dana's newest Spicer S110 and S130 single drive axles. Additional axle models will follow later this year.

Creating an Alternative: Automakers Work to Offer Hydrogen as the New Gasoline

Concerned with the oil supply, automakers have been playing a double game with gasoline, trying to conserve it while they worked to replace it.

Their conservation efforts have been apparent for years. For better gas mileage, gear transmissions were replaced with continuously variable transmissions in some vehicles, and electric motors were added in a number of autos to alternate with their gas engines. Also, additional speeds—a sixth, a seventh, even an eighth speed—are being manufactured on many gear transmissions to increase fuel efficiency.

But, automakers were also developing ways to replace gas, and an alternative fuel now coming to the fore is hydrogen.

Based on remarks by Gary S. Vasilash, editor-in-chief of *Automotive Design and Production*, that alternative may not be far off. Vasilash, who has more than 25 years' experience in the auto industry, discussed hydrogen fuel cell technology in his keynote address at the official opening June 8 of LMT Group's new automotive support center in Auburn Hills, MI, near Detroit.

In his speech, Vasilash said DaimlerChrysler AG is convinced hydrogen fuel cells are the future, and the automaker is working on the technology now. He added that General Motors Corp. continues to invest in hydrogen fuel cell technology despite the company's current problems. "They are not deviating," he said.

In an interview later, he added that GM plans to have a cost-competitive fuel cell system completed by 2010.

If viable, hydrogen-powered vehicles would require major changes outside automakers' factories. Perhaps the most obvious change would involve the refueling of the cells. Hydrogen vehicles would need hydrogen stations just like regular vehicles need gas stations.

"The problem is infrastructure," Vasilash said.

In the United States, as in other countries, gas stations seem to be everywhere, so drivers can conveniently access them. A similarly convenient system would need to develop as the use of hydrogen-powered vehicles grew and became widespread.

Vasilash also mentioned DaimlerChrysler's development of BlueTec. This technology involves engine redevelopment to reduce diesel fuel consumption and emissions and an exhaust aftertreatment to further reduce emissions. This aftertreatment involves use of an aqueous solution stored in a reservoir that's part of the vehicle. After his speech, Vasilash said BlueTec's next milestone is its availability in more vehicles. He added that the issue with BlueTec is simply designing vehicles to accommodate the reservoir: "There is no hurdle that needs to be crossed."

Time will tell whether hydrogen becomes the new gasoline, whether it proves to be commercially viable and how widespread its use becomes, but automakers have been playing their double game for years now to create an alternative to gasoline, in case another option is needed.

Photo courtesy of Daniel Lippett.

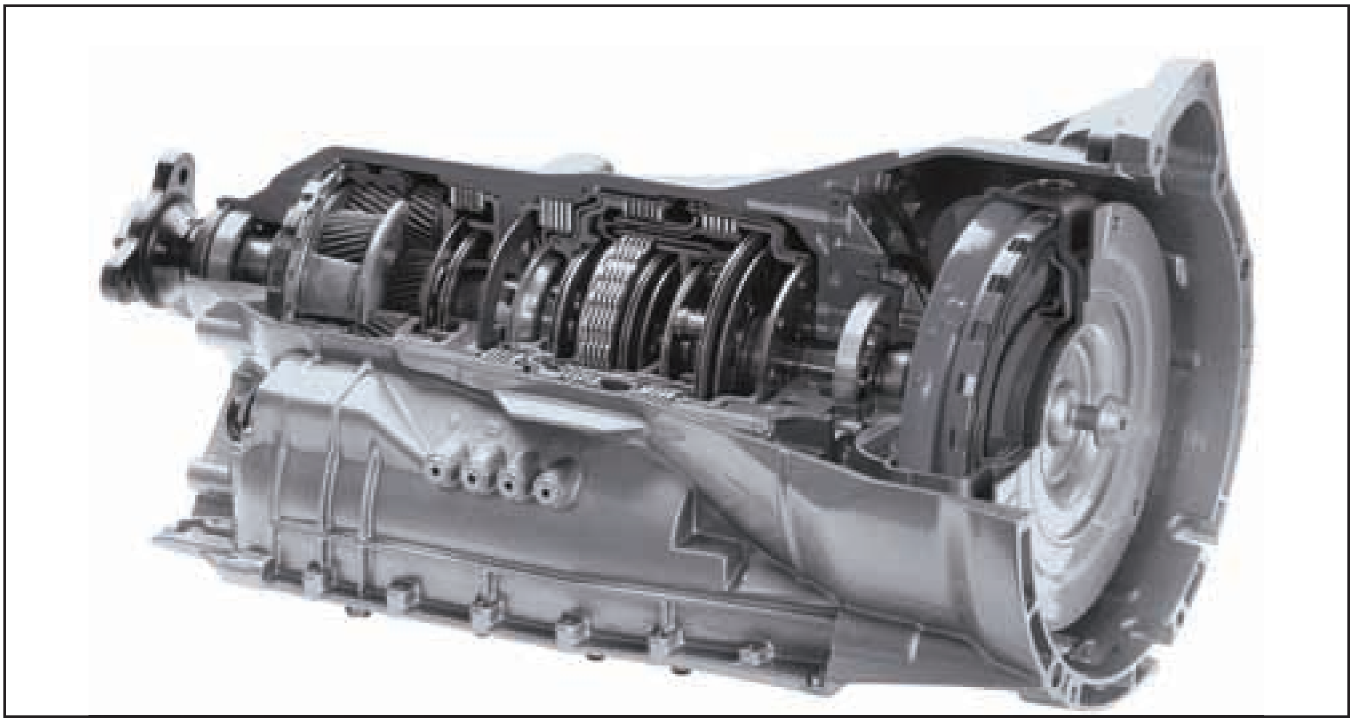
Gary S. Vasilash, editor-in-chief of *Automotive Design and Production*.

Comment on these articles by sending e-mail to *publisher@geartechnology.com*

Trends in Automobile Transmissions

Frank Buscemi, Manager—Public Affairs,

ZF Group—North American Operations



With all the work in transmission development these days, the demand for automobile transmission gears should remain strong for several years, but because of the great variety of projects and variations, transmission manufacturers and their suppliers will have to be as flexible as possible to keep up with the changes.

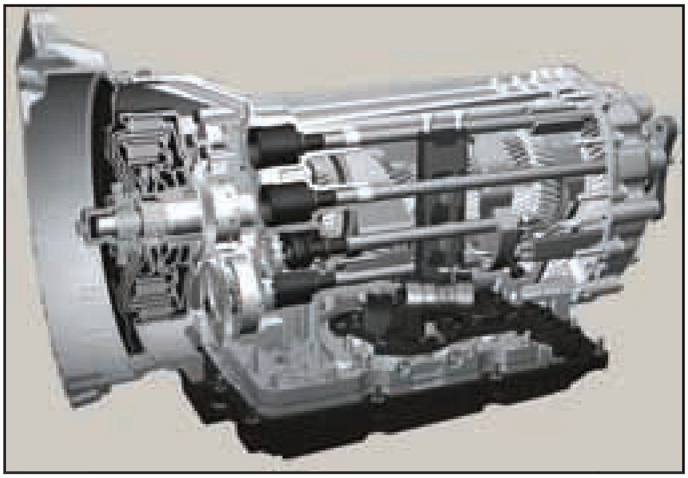
Automobile transmissions have come a long way since the days of simply choosing between automatic and manual. Today's drivers have more transmission options than ever before. Automatics and manuals are still there, but they are now accompanied by automated manuals (AMTs), dual clutch transmissions (DCTs), continuously variable transmissions (CVTs) and hybrid drives.

Within those categories, there are several options as well, including 4-, 5-, 6-, and 7-speed automatic transmissions, along with five- and six-speed manuals.

ZF, the world's third largest producer of automobile transmissions, has been providing gearboxes since 1915, when Graf von Zeppelin founded the company to provide gears and transmissions to his Zeppelin airships.

Today, ZF provides more than 1.2 million transmissions per year to automakers like Aston Martin, Audi, BMW, Ford, General Motors, Jaguar, Land Rover, Porsche, and Volkswagen and employs more than 6,300 in its car driveline technology division. In 2001, ZF introduced the world's first 6-speed stepped automatic transmission in the BMW 7-Series.

Dr. Harald Naunheimer, director of product development, car driveline technology at ZF, says that transmissions are highly individualized based on region and vehicle segment. "The models feature individual strengths, which, in turn, depend on the application conditions. Therefore, each case has to be individually assessed because general statements are not applicable."



The type of transmission plays a key role in defining a vehicle's character, image, segment and brand, making it a major factor in competitiveness. Each vehicle features individual strengths, depending on application conditions.

AMTs—Also known as sequential manual gearboxes (SMGs), AMTs have their roots in Formula 1 racing, using computer-controlled actuators that are prompted by paddle shifters mounted on the steering wheel. There is no clutch pedal. The system allows for both automatic and manual modes. SMGs are generally found on high-performance sports cars.

CVTs use a belt or chain to connect variable-diameter pulleys to provide an unlimited number of ratios. This allows for uninterrupted power to the wheels.

DCTs—Also called direct-shift gearboxes (DSGs), DCTs use two clutches instead of a single-sided clutch to transfer engine power through two sets of gears. The paths are set up similar to a manual transmission, with one path controlling gears 1, 3 and 5 and the other controlling 2, 4, and 6, but there is no clutch pedal. Also, there is constant power to the wheels.

Hybrids—With regard to hybrids, market share will depend on how the unit is used—mild or full-scale hybrid—and on regional legislation.

"North America and Japan are typically automatic transmission markets, with Japan using CVTs in the sub-compact market," Naunheimer says. "Europeans still prefer manuals—but even that has dropped slightly."

Since the introduction of the 6-speed automatic, several automakers have moved in this direction. At ZF's automatic transmission plant in Saarbruecken, Germany, much of the production has shifted from 5-speed automatics to 6-speeds. Nearly all elements of the transmission are manufactured in-house. Based on application requirements, the plant can hone, shave, grind or match grind its gears.

Dr. Ludger Reckmann, vice president of manufacturing for automatic transmissions at ZF's car driveline technology division, says the Saarbruecken facility will continue to increase capacity and finishing capability for its gear machining operations—a core technology of ZF. The company will also invest in quality-related processes, such as heat treating.

The average 6-speed automatic transmission contains 14 individual gears—ring, sun or planetary. These are integrated in several ways, depending on the vehicle.

Naunheimer says suppliers must be flexible, as the trend toward individual solutions will continue. "While the market share of manual transmissions in Europe is still around 80 percent, we expect that sub-compact cars (A/B segment) and light commercial vehicles will increase AMT use. Compact cars (C segment) and mid-size cars (D segment) can expect to see more DCTs. Mid-range and premium cars will continue to see automatic transmissions with hydrodynamic torque converters up to 7-speeds."

Reckmann says increasing fuel efficiency is the most important development objective for transmission engineers. "The main goal is to improve overall powertrain efficiency by reducing drag losses. However, we also look for shift-time reduction, cost improvements and noise reduction."

Reckmann also says that within the transmission industry, manufacturers will have to make adjustments in the near future based on limited floor space, corporate investment, cost and technology changes, adjustments that will lead to some outsourcing of non-critical components to low-cost countries.

"Suppliers will also have to create strategic partnerships for key components within the transmission—electronics and hybrids, for example—to develop new products," Reckmann adds.

The recent partnership formed by ZF and German-based supplier Continental-Teves to develop hybrid technology is one of several being formed within the industry. Continental-Teves brings electronics knowledge and expertise, while ZF provides the transmission, chassis and steering to create a complete hybrid driveline.

Gear manufacturing will see some changes as well. With more all-wheel drive vehicles on the road, demand for bevel and planetary gears—along with spur gear sets—will increase. And, with 8-speed automatic transmissions now hitting the market, the total number of gears necessary will also increase.

"With more gears needed, further improvements in gear quality, process costs and improvements will be required," Reckmann says. "Gear coating, heat treatment and finishing operations should remain the same, but more grinding, power honing and precision forging will be necessary."

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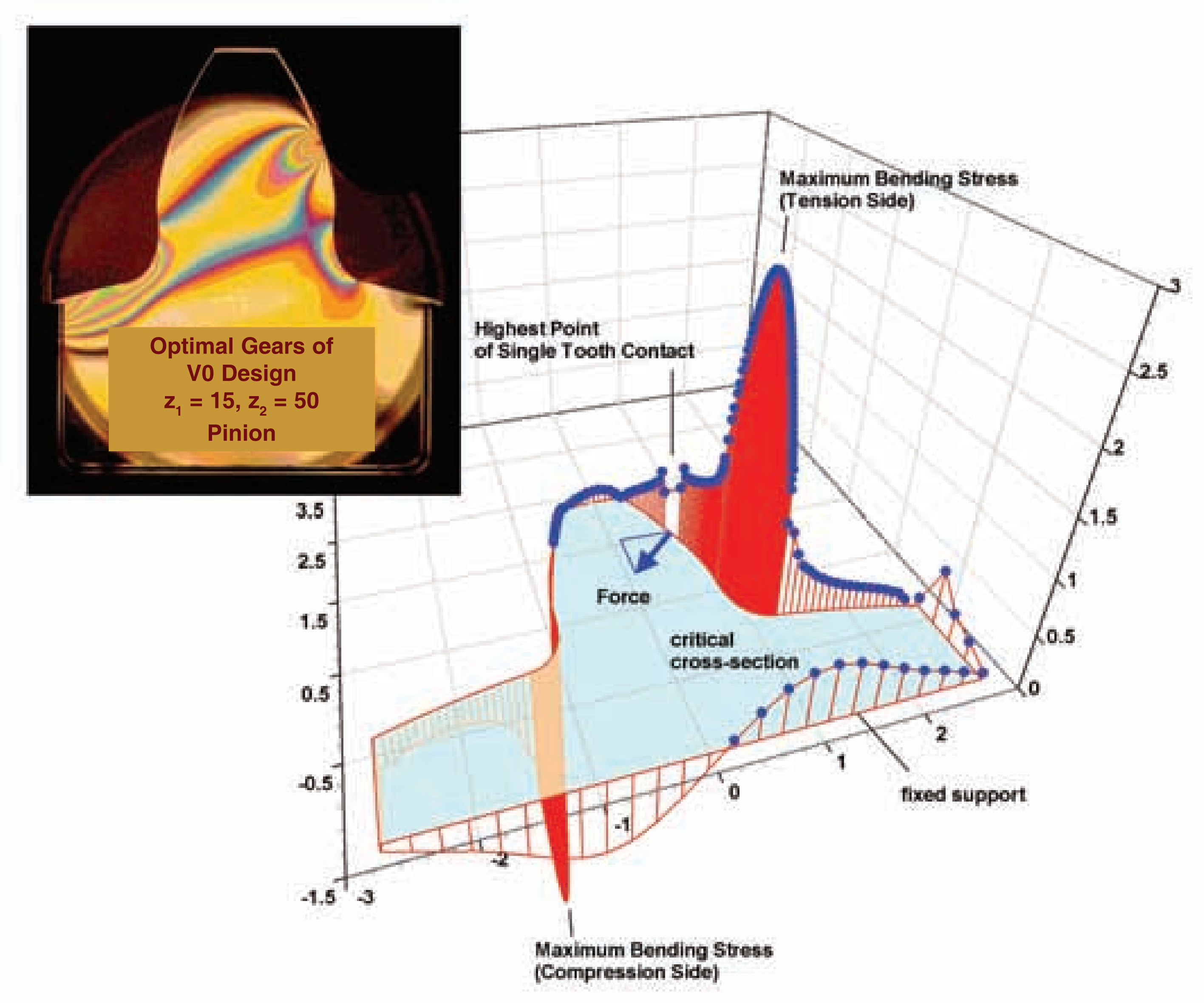
Generating Interchangeable 20° Spur Gear Sets with Circular Fillets to Increase Load Carrying Capacity

Christos A. Spitas and Vasilis A. Spitas



Management Summary

This article presents a new spur gear 20° design that works interchangeably with the standard 20° system and achieves increased tooth bending strength and hence load carrying capacity. In this design, a circular fillet replaces the normal trochoidal fillet, yielding larger cross-sections at the tooth root and lower stress concentration. The actual working tooth profiles remain unaffected. To verify this design, a series of numerical simulations was carried out using the boundary element method (BEM), and the results were confirmed with subsequent laboratory testing.

Introduction

The vast majority of gear applications use the standard 20° involute system because of its good combination of bending and surface pressure strength, involute insensitivity to errors in center distance and relative ease of manufacturing. Most gears are produced by hobbing or other generation-type processes, where a straight-tooth rack or equivalent tool produces the involute working gear tooth surface as well as a trochoidal root fillet.

Despite the benefits of this system, it is generally felt that a higher bending strength and hence load carrying capacity should be obtained. This is especially true with small numbers of teeth (less than 14 or 17 depending on the tip radius of the hob), where the standard involute teeth are susceptible to undercutting. This is a situation where the tip of the cutter removes material from the involute profile in a secondary cutting action. The resulting teeth have smaller thicknesses near their roots, where the critical section is usually located (Refs. 1–2), and this severely hampers load carrying capacity.

Up to now, a number of different gear tooth designs have been proposed aiming to address this problem (Refs. 3–4), but the most commonly employed method is that of positive profile shifting (Refs. 5–8), resulting in teeth with no undercutting, with enhanced fatigue characteristics and with changes in the nominal center distance of the gear pair. Other solutions of a hybrid-type have been proposed by Litvin et al. (Ref. 9) and Tsai and Tsai (Ref. 10). These designs typically rely on sturdier tooth forms, which often present unfavorable side effects, such as 1.) increase of average sliding velocity, 2.) non-interchangeability with the standard 20° tooth systems either due to shifting or due to non-involute shapes, 3.) lower pitting and scoring resistance, 4.) lower contact ratio resulting in more noise and vibration during operation (Ref. 2), and 5.) increased complexity and hence manufacturing costs. Clearly, any viable alternative to the standard system should, as a matter of course, address these issues effectively.

In this context, the present article takes into account a number of gear tooth design concepts developed by the authors (Ref. 11) in order to produce gear teeth with load carrying capacity increased up to 130% of the standard designs whilst keeping the rest of the gear pair working conditions unaffected. The resulting interchangeable spur gear tooth systems are generated by simple hobbing tool geometry, using standard cutting machines and procedures. The feature that increases the load carrying capacity in the gear teeth is the substitution of the standard trochoidal fillet produced with the standard circular-filleted hob teeth with a circular arc fillet generated with modified hob tip geometry.

The work reported in this article comprises the full generation, simulation and verification cycle of the new design encompassing: 1.) design of hob cutter fillet geometry, 2.) simulation of generation process, 3.) strength evaluation and comparison with standard gears using BEM, 4.) manufacture of gear tooth samples in the laboratory workshop, and 5.) static load carrying capacity testing of generated gears by measuring maximum fillet stress using photoelasticity.

The test results show agreement with the anticipated values, and the circular fillet design is proven to give consistently better results than the standard trochoidal fillet. The beneficial effect is more pronounced with tooth numbers fewer than 40. In addition, the usual undercutting restrictions are shown to no longer apply to circular fillet teeth cut with the present method, so that pinions of as few as nine teeth can be produced, possessing a load carrying capacity of more than three times that of standard undercut teeth.

Geometrical Analysis

As a first step, the gear model is non-dimensionalized with respect to the gear module (Ref. 11), so that a wider generality of the results can be achieved. The process follows.

Consider a pair of spur gears in mesh. Both gears should have the same nominal pressure angle α_o and the same module m in order to be able to mesh properly. It is also possible that these gears have addendum modifications x_1 and x_2, respectively, and therefore their pitch thickness is given by the following relationship:

$$s_{oi} = c_{si}\pi m + 2x_i \tan\alpha_o m = s_{oiu} m \quad (1)$$

where c_{si} is the thickness coefficient of gear i, (i = 1, 2), which in the general case is $c_{s1} \neq 0.5 \neq c_{s2}$, while s_{oiu} is the pitch thickness of the corresponding non-dimensional gear for which the module (m) and the face width (b) are both equal to unity.

In the absence of errors, the center distance and working pitch circle radius expressions are:

$$a_{12} = \frac{z_1 + z_2}{2} m + (x_1 + x_2) m = a_{12u} m \quad (2)$$

$$r_{bi} = \frac{z_i}{z_1 + z_2} a_{12u} m = r_{biu} m \quad (3)$$

Furthermore, Spitas (Ref. 12) has shown that the radius defining the position of the highest point of single tooth contact (HPSTC) on gear 1 can be expressed as:

$$r_{B'u} = \frac{r_{B'}}{m} = \sqrt{r_{k1u}^2 + (\varepsilon - 1)t_{gu}\left[(\varepsilon - 1)t_{gu} - 2\sqrt{r_{k1u}^2 - r_{g1u}^2}\right]} \quad (4)$$

The above equations show that it is possible to describe the governing relationships in the gear pair in a more generalized way, by non-dimensionalizing in terms of the module. This reduces the needed variables, as the interaction of a given

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gear with any mating gear need only consider the contact ratio instead of the full set of the mating gear attributes. Also, the use of non-dimensional teeth allows every geometrical feature f on the transverse section of a full-scale gear tooth to be connected with the corresponding feature f_u of the transverse section of the non-dimensional gear tooth through the equation:

$$f = f_u m \tag{5}$$

Stresses can also be calculated in non-dimensional teeth $\sigma_u(z, x, c_s, \varepsilon)$ with unit loading $P_{Nu} = 1$ and be related to the actual stress σ using the following equation:

$$\sigma = \sigma_u \frac{P_N}{bm} \tag{6}$$

as suggested by Rogers (Ref. 8) and Townsend (Ref. 1).

Having defined the non-dimensional form of the problem, the novel circular fillet tooth geometry is defined through the following procedure.

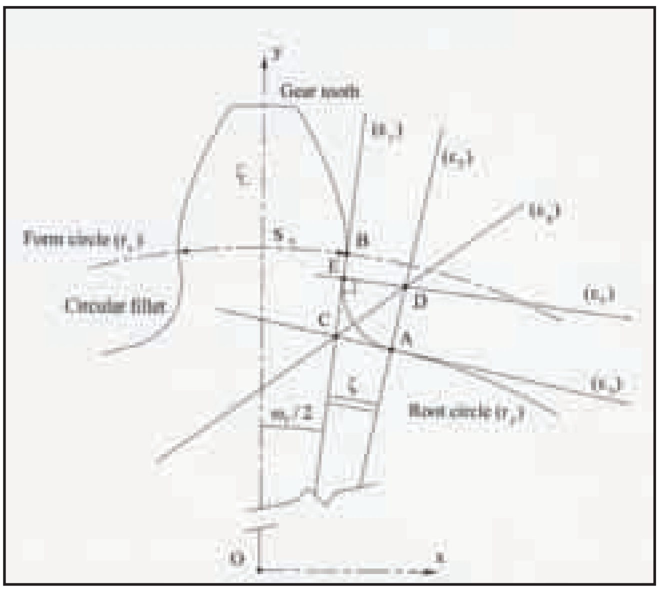

Figure 1—The geometry of the circular fillet.

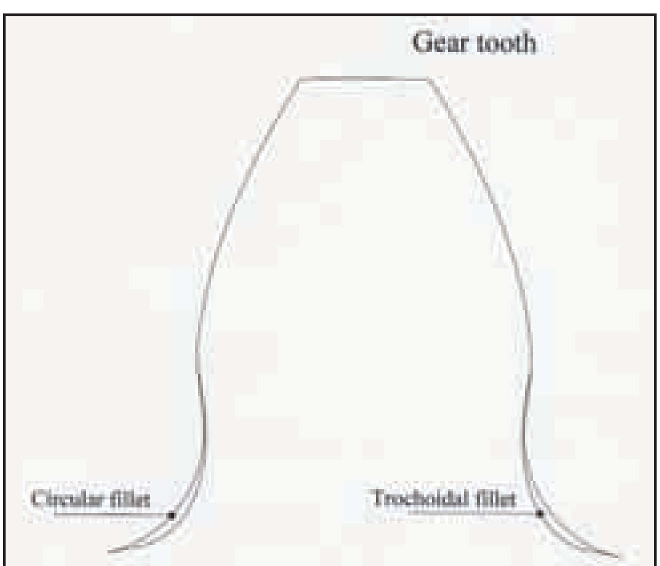

Figure 2—Superposition of circular fillet on a standard tooth.

Consider the involute spur gear tooth of circular fillet illustrated in Figure 1, where point 0 is the center of the gear, axis $0y$ is the axis of symmetry of the tooth and point B is the point where the involute profile starts.

Point A is the point of tangency of the circular fillet with the root circle r_f. Point D lying on $\varepsilon_2 \equiv$ OA represents the center of the circular fillet. Line ε_3 is tangent to the root circle at A and intersects with line ε_1 at C. The fillet is tangent to the line ε_1 at point E. Since it is always $r_s > r_f$ (Ref. 1), the proposed circular fillet can be implemented without exceptions on all spur gears regardless of the number of teeth or other manufacturing parameters. A comparison of the geometrical shapes of a tooth with a circular fillet and of one with a standard fillet is presented in Figure 2.

For the geometrical modelling, only non-dimensional teeth are examined, i.e. teeth with unit module (m) and face width (b). In non-dimensional teeth, the pitch radius and the pitch thickness are calculated:

$$r_o = \frac{z}{2} \text{ and } S_{ov} = \pi c_s + 2x \tan \alpha_o \tag{7}$$

$$S_s = r_s \left[\frac{S_{ov}}{r_o} + 2(\varphi_o - \varphi_s) \right] \tag{8}$$

where:
$\varphi_s = \tan \alpha_s - \alpha_s$ is the involute function on circle r_s and
$\alpha_s = \cos^{-1}(r_g/r_s)$ is the pressure angle on circle r_s

Angle $\omega_s/2$ that corresponds to the arc $S_s/2$ (Fig. 1) is given by the equation:

$$\omega_s/2 = \frac{S_s/2}{r_s} = \Omega_s \tag{9}$$

Angle ζ (Fig. 1) takes values between 0 and ζ_{max} so that:

$$\zeta_{max} = \frac{\pi}{z} - \Omega_s \tag{10}$$

The coordinates of points A and B are:

$$x_A = r_f \sin(\zeta + \Omega_s), \; y_A = r_f \cos(\zeta + \Omega_s) \tag{11}$$

$$x_B = r_f \sin \Omega_s, \; y_B = r_f \cos \Omega_s \tag{12}$$

The defining equations of lines ε_1 and ε_2 are, respectively:

$$(\varepsilon_1): y = \frac{1}{\tan \Omega_s} x, \; (\varepsilon_2): y = \frac{1}{\tan(\zeta + \Omega_s)} x \tag{13}$$

Since ε_3 is perpendicular to ε_2 and ε_3 passes through point $A(x_A, y_A)$ its defining equation is:

$$y = -\tan(\zeta + \Omega_s) x + \frac{r_f}{\cos(\zeta + \Omega_s)} \tag{14}$$

Point $C(x_C, y_C)$ is the intersection of ε_1 and ε_3, and therefore its coordinates should verify Equation 13 and hence:

$$x_C = r_f \frac{\tan \Omega_s}{\sin(\zeta + \Omega_s) \tan \Omega_s + \cos(\zeta + \Omega_s)}, \; y_C = \frac{x_C}{\tan \Omega_s} \tag{15}$$

Angle $B\hat{C}A$ is calculated as:

$$B\hat{C}A = \left(\frac{\pi}{2} - \Omega_s\right) + \zeta + \Omega_s = \frac{\pi}{2} + \zeta \tag{16}$$

Line ε_4 bisects the previous angle $B\hat{C}A$, so its inclination is:

$$\tan\left[\frac{B\hat{C}A}{2} - (\zeta + \Omega_s)\right] = \tan\left(\frac{\pi}{4} - \frac{\zeta}{2} - \Omega_s\right) \tag{17}$$

Point $C(x_C, y_C)$ belongs to ε_4 and thus the defining equation of line ε_4 is derived as:

$$y = \tan\left(\frac{\pi}{4} - \frac{\zeta}{2} - \Omega_s\right)x + r_f \frac{1 - \tan\Omega_s \tan\left(\frac{\pi}{4} - \frac{\zeta}{2} - \Omega_s\right)}{\sin(\zeta + \Omega_s)\tan\Omega_s + \cos(\zeta + \Omega_s)} \tag{18}$$

At this point, two distinct cases are considered. Point E coincides with point B (E ≡ B). In this case, ε_5 is perpendicular to ε_1 at point E ≡ B, so ε_5 must have an inclination equal to $-\tan\Omega_s$, and since point B belongs to ε_5, its defining equation is:

$$y = -\tan\Omega_s x + \frac{r_s}{\cos\Omega_s} \tag{19}$$

Point $D(x_D, y_D)$ should verify both Equations 13 and 19 and therefore has the following coordinates:

$$x_D = r_s \frac{\tan(\zeta + \Omega_s)}{\cos\Omega_s + \sin\Omega_s \tan(\zeta + \Omega_s)}, \quad y_D = r_s \frac{1}{\cos\Omega_s + \sin\Omega_s \tan(\zeta + \Omega_s)} \tag{20}$$

According to Figure 1, it is AC = BC, and after substitutions and calculations, we arrive at the equation:

$$\frac{r_f^2 + r_s^2}{2} = \frac{r_s r_f}{\cos\zeta}\left(\sin^2\Omega_s + \cos^2\Omega_s\right) \tag{21}$$

from which the value of ζ is derived as:

$$\zeta = \cos^{-1}\frac{2r_s r_f}{r_s^2 + r_f^2} \tag{22}$$

By defining the non-dimensional parameter $S = r_s/r_f > 1$, Equation 22 becomes:

$$\zeta = \cos^{-1}\left(\frac{2S}{1+S^2}\right) \tag{23}$$

Equation 23 is used for the determination of angle (ζ). Point E lies below point B.

In this case, it is $\zeta \leq \zeta_{max}$, and the center of the circular fillet of the tooth is calculated following the methodology described below:

$$CA = \sqrt{(x_A - x_C)^2 + (y_A - y_C)^2} = CE \tag{24}$$

$$AD = CA\tan\left(\frac{\pi}{4} + \frac{\zeta}{2}\right) \tag{25}$$

The coordinates of points $D(x_D, y_D)$ and $E(x_E, y_E)$ are, respectively:

$$x_D = (r_f + AD)\sin(\zeta + \Omega_s), \quad y_D = (r_f + AD)\cos(\zeta + \Omega_s) \tag{26}$$

$$x_E = (OC + CE)\sin\Omega_s, \quad y_E = (OC + CE)\cos\Omega_s \tag{27}$$

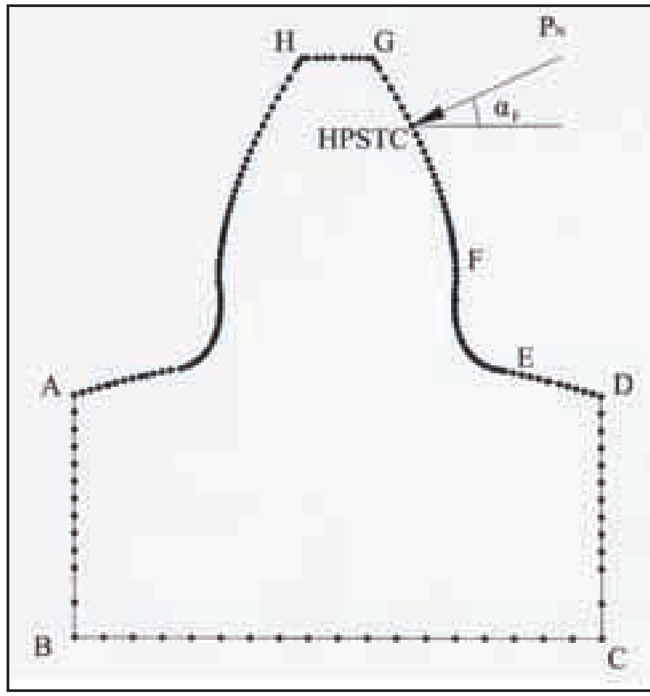

Figure 3—BEM mesh on a tooth model.

The remaining portion of the tooth profile between points B and E is a straight line.

Stress Analysis

The structural analysis of the spur gear tooth model is carried out using the boundary element method with quadratic isoparametric boundary elements. The calculation of the non-dimensional tooth profile and the generation of the mesh are done automatically using specially developed software. A unitary normal load is exerted at the highest point of single tooth contact (HPSTC). The resultant normalized stress is related to the actual stress of the full-scale tooth with Equation 6. The distance of the HPSTC from the center of the gear is calculated in terms of the contact ratio ε and the geometrical characteristics of the gear according to Equation 4.

A typical mesh is presented in Figure 3. The tooth base, ABCD, is discretized in 43 nodes or 21 elements and is considered to be fixed, i.e. the displacements of all nodes are zero.

Portion DE represents the root circle and is discretized in 11 nodes or five elements. Portion EF is the tooth fillet consisting of 72 elements or 144 nodes, portion FG is the involute part of the tooth consisting of 40 elements or 81 nodes and finally portion GH is the tip, discretized in five elements or 11 nodes. All nodes belonging to the tooth profile are considered to be unloaded (horizontal and vertical traction equal to zero) with the exception of the node marked HPSTC, which belongs to the involute part FG and on which the normal load P_N is acting.

The reason that the mesh is denser at the tooth fillet is that the maximum stress is expected to occur in this area, and therefore greater density will ensure increased accuracy of the results (both the position of the critical section and the magnitude of the maximum developed stress).

Manufacture of Test Gear Tooth Specimens

Several test gear tooth specimens with both standard trochoidal as well as circular fillets were produced by precision milling on a CNC machine tool in compliance with the geom-

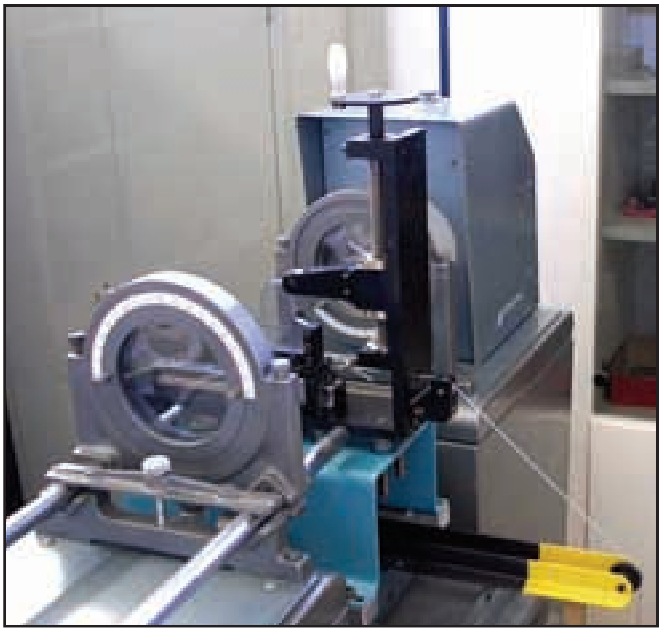

Figure 4—Test rig. Polariscope and loading apparatus with tooth specimen loaded at the tip.

etry of the BEM analysis layout to ensure full comparability of results. The specimens were produced from polycarbonate material suitable for photoelastic investigations. The other fixed parameters are: module = 5 mm, thickness coefficient = 0.5, and profile shift = 0.0

Specimens were produced to correspond to gears with tooth numbers 9, 17, 24, 32 and 40, covering the most basic range of teeth where the circular fillet is expected to yield the most noticeable benefits.

The cutting parameters were selected such that no distortion or residual stress would be present in the specimens. Indeed, all specimens underwent preliminary photoelastic testing in their unloaded state after machining. The residual stresses were negligible. Geometrical survey conducted on a TESA CMM yielded a maximum form deviation of 0.08 mm, which was deemed quite satisfactory given the overall large scale of the models and the required accuracy.

Experimental Verification

A comparative study was carried out between the structural properties of the standard trochoidal-filleted teeth generated by hobbing and the proposed circular-filleted involute teeth. Five distinct cases were examined for unshifted teeth with 9, 17, 24, 32 and 40 teeth, respectively. Results were general regardless of module because of the non-dimensional treatment of the governing equations as explained in the previous analysis.

The test rig used for mounting and loading the tooth specimens for photoelastic measurements is a customized design produced in the Machine Elements Laboratory of the National Technical University of Athens. The rig was mounted on a plane polariscope equipped with mercury and sodium vapor lamps in order to produce the various light wavelengths needed for the measurements. The test rig is presented in Figure 4.

Results and Discussion

In the case of trochoidal-filleted teeth, the N = 9 teeth case presented severe undercutting. The teeth were loaded at their highest point of single tooth contact (HPSTC) and since it has been established that the HPSTC for any given gear pair depends only on the geometrical characteristics of the gear and on the

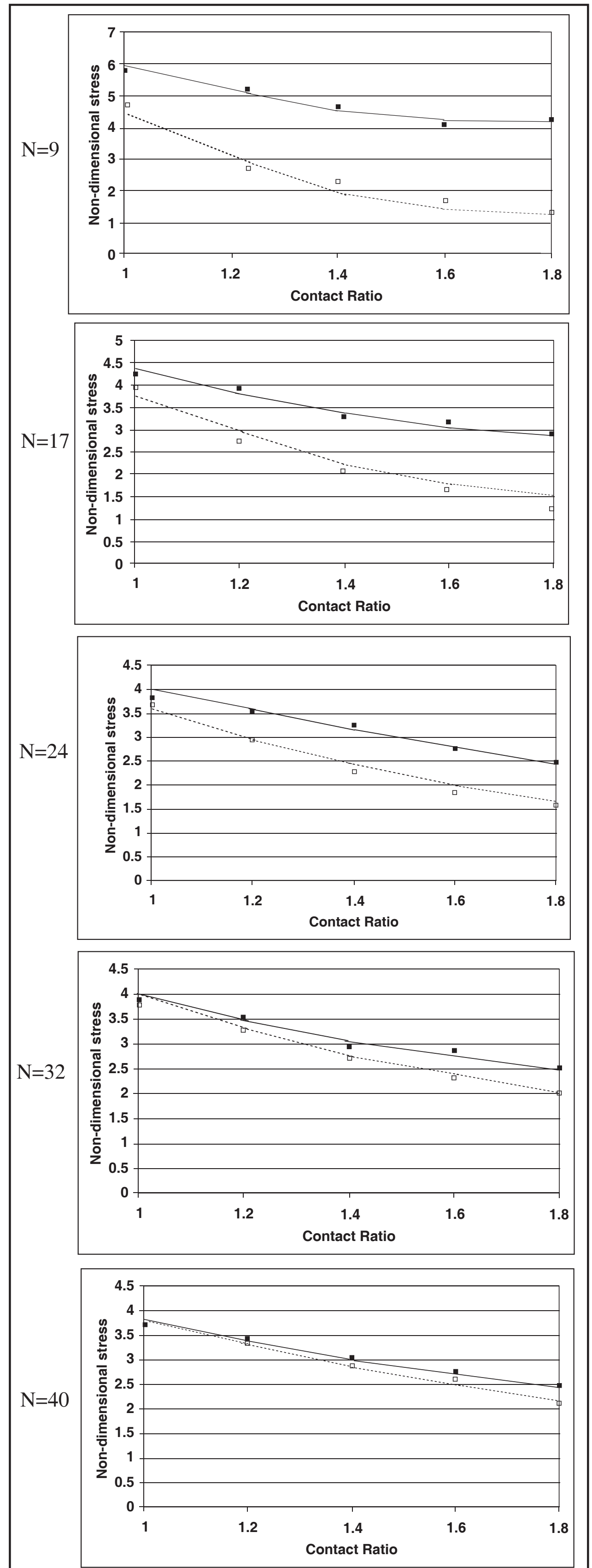

Figure 5—Non-dimensional stresses vs. contact ratio for gears with various numbers of teeth as predicted by BEM analysis and verified by photoelastic experiment. Solid line: Trochoidal fillet (BEM analysis), Solid rectangle: Trochoidal fillet (experiment), Dotted line: Circular fillet (BEM analysis), Hollow rectangle: Circular fillet (experiment).

contact ratio of the pair, the analysis results can be plotted in the form of non-dimensional stress versus contact ratio diagrams. Module- and loading level-specific results can be extracted for any given choice of parameters by use of Equation 6.

Photoelastic investigation confirmed the validity of the BEM results. Maximum deviation was less than 5%, which is deemed satisfactory. A superposition of the analytical and the experimental results is presented in Figure 5 and shows good agreement between the two. The effect of contact ratio was modelled in the experiments by displacing the position and angle of load application to correspond to the actual displacement of the HPSTC in each case.

An additional simulation was carried out where all gears were examined for interference during meshing. It was verified that no interference occurs when replacing the standard trochoidal fillet with the novel circular fillet, making the solutions geometrically and kinematically interchangeable without any restraint.

Figure 6 presents a summary of the obtained increase in load carrying capacity (decrease of maximum fillet stress) as calculated by the BEM calculations. The trend is quite clear and systematical. At N = 9 teeth, the trochoidal solution is at a severe disadvantage because of the undercutting and normally such gears are never produced. Use of the circular fillet allows for use of such otherwise prohibitively low tooth numbers in addition to its overall benefit of increasing load carrying capacity.

It is noted that the working tooth profiles remain identical in standard trochoidal and in circular fillet teeth, so no change in contact pressure takes place when switching to a circular fillet design. Circular fillets purely increase bending strength.

Overall, the new circular fillet design fares better than the standard trochoidal fillet design, especially in cases of pinions with small tooth numbers. With larger tooth numbers, the difference between the two designs becomes smaller and tends to be asymptotically zero when the number of teeth tends to infinity, as in the case of a rack, where even the standard design gives a circular fillet.

The proposed fillet geometry can be cut on conventional hobbing machines or rack-cutting machines without the need of special tooth number-specific milling tools. The generating rack need only deviate slightly from the standard rack, as its tip must be of an appropriate shape to generate the root fillet of the tooth. This generating process is simulated in Figure 7.

As it is usually pinions that undergo the higher bending stresses developed in the gear pair during meshing, the same generating rack can be used to cut the mating wheel teeth, although their root fillet geometry will not be strictly circular as for the pinion. Still, initial trials performed by the authors suggest that wheel teeth produced in this way will also exhibit increased strength compared to the conventional teeth generated with the standard circular tipped rack/hob.

Conclusion

A new circular fillet tooth design was proposed to replace the standard tooth designs currently produced by hobbing and other generation processes. The design and verification process comprised a full generation, simulation and verification cycle of the new design, including design of hob cutter fillet geometry, simulation of the generation process, calculation of fillet stress and strength evaluation and comparison with standard gears using BEM, manufacture of gear tooth samples and static load carrying capacity testing of generated gears by measuring maximum fillet stress using photoelasticity.

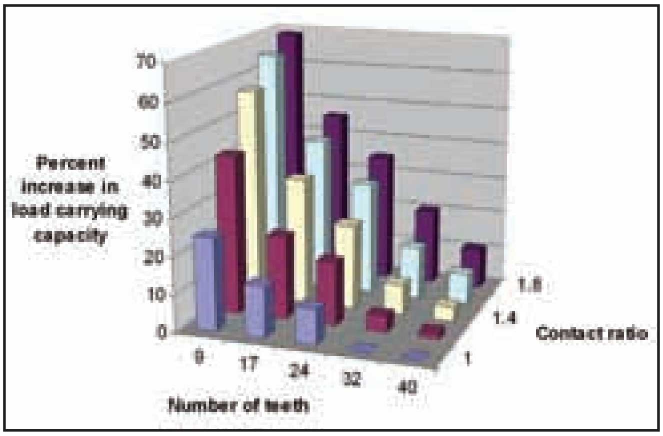

Figure 6—Calculated increase in load carrying capacity granted by circular fillet.

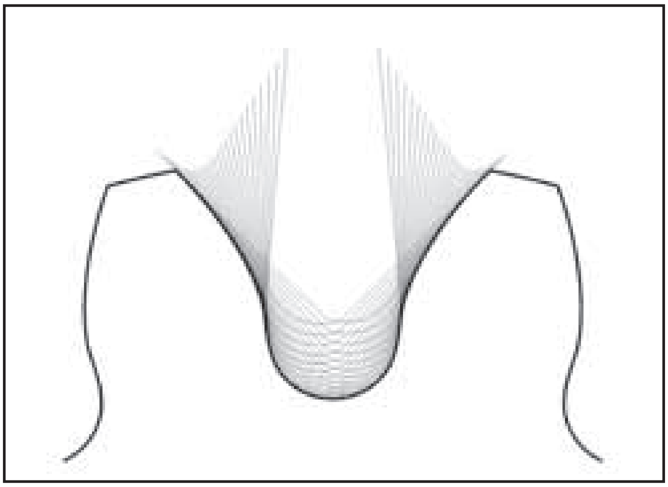

Figure 7—Schematic of the hobbing process for the production of a circular fillet.

The results show that the proposed solution achieves superior performance, especially at small tooth numbers, where undercutting otherwise becomes a serious issue, while maintaining full interchangeability with the current trochoidal fillet designs. In particular, no change is made on the involute working profile itself, so mesh kinematics and contact stresses (pitting resistance) are not affected at all.

The new design inherently dispenses with the undercutting problems that hinder the use of low tooth numbers (typically less than 17) and can be easily manufactured using standard involute hobbing tools with special tip design. The shape of the required modified hobbing tools was calculated and the generation process was shown to be no more complicated or demanding than that used currently for the production of standard gears.

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The Lubrication of DLC Coated Gears with Environmentally Adapted Ester-Based Oil

B. Kržan, M. Kalin and J. Vižintin

Management Summary

The development of new transmissions and gearboxes is characterized by increasing levels of torque and power, improved efficiency, increased life expectancy, prolonged service intervals, reduced amount of lubricant, and more stringent noise and environmental requirements. The environment, as a new factor in the design process, increases the focus on product improvements that are designed to avoid environmental problems before they occur. Surface coating is one of the future technologies for improving performance of case-hardened gears.

A main limiting factor in extending the use of hard coatings to machine component application is the lack of knowledge about how these inert coatings perform under lubricated conditions using today's lubricants, which were originally designed for steel/steel contact situations. The influence of ester-based lubricant on the scuffing capacity of WC-containing DLC coated spur gears was evaluated in a non-standard FZG test procedure. The properties of the formulated ester-based lubricant were investigated in comparison with conventional mineral gear oil. The results show that under present conditions, W-DLC coated gears could provide satisfactory wear resistance for moderate loads.

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Introduction

Generally, gears for power transmission drives use lubricants based on petroleum-derived base stocks. With the rapid advancement of gear design and manufacturing technology, gearboxes have become smaller, and output power has increased significantly. The net results are higher contact stresses, higher speeds and lower amounts of lubricant. With the decreased oil capacities, the lubricant must provide appropriate lubrication at higher operating temperatures, more effective cooling and suspension of contaminants. Therefore, selecting high performance lubricants becomes more and more important. Moreover, there is also a clear trend to use lubricants that cause less harm to the environment.

The current view is that depletion of scarce resources and increasing environmental pollution cannot continue for the next 50 years as they have in the past 50, without drastically affecting our quality of life. Use of environmentally adapted lubricants is one of the strategies to avoid environmental problems before they occur. Very good or even superior technical performance of some esters combined with very favorable ecological properties enable the formulation of high performance lubricants with extremely low evaporation rates, very high viscosity index and good boundary lubrication characteristics.

Diesters, polyol and complex esters are biodegradable in terms of one of the internationally recognized test methods, and they have low aquatic toxicity. Their advantage is also that they can be partly derived from renewable resources, including vegetable oils and animal fats. From an ecological point of view, the prospects for using renewable raw materials are favorable, provided the full potential of natural synthesis by means of energy from the sun is used. The production of vegetable oils constitutes a cycle in which no net release of carbon dioxide occurs (Refs. 1–2).

During recent years, significant progress has taken place in the development of advanced coatings used in tribology technology. The unique tribological properties of diamond-like carbon (DLC) films, such as low friction, high wear resistance and low deposition temperature, have made them very attractive for machine element applications. DLC films doped with metal (Me-C:H) have advantages over pure carbon coatings as internal stress is reduced and adhesion to steel substrates is improved. Beside tools and dies, diamond-like and related coatings are starting to find application in some mechanical component applications, including bearings and gears. They provide a great opportunity to improve durability and to reduce frictional losses of machine components (Refs. 3–4).

The present work attempts to combine the excellent friction properties of W-DLC coating with the established lubricating abilities of ester-based lubricant for improving gear performance. The modified FZG scuffing tests were carried out to investigate and compare the scuffing capacity of uncoated steel gears and W-DLC coated gears, lubricated with conventional mineral gear oil and an environmentally adapted ester-based formulation.

Experimental

Test equipment. The gear tests were performed on an FZG back-to-back test rig. Test conditions were similar to the standard procedure for load carrying capacity of lubricants according to ISO DIS 14635-1 (Ref. 5).

The test oils were subjected to the load, increased through 12 load stages, defined in the abovementioned standards. Duration of each load stage was 20 minutes (29,000 revolutions of the motor) at constant pinion shaft speed of 1,450 rpm. Starting bath oil temperature in each load stage was 50°C and was allowed to rise freely during the test. As the duration of load stages was prolonged with regard to the standard A/8.3/90 test procedure, the total work transmitted by the test gears up to the end of load stage was 25 percent higher. At the end of the last load stage, total work transmitted was 184 kWh.

The gear teeth flanks were visually examined after each load stage for cumulative damage, in particular scuffing marks and excessive wear. Also, test gears were weighed to the nearest milligram after every third load stage.

The method used for the quantitative evaluation of the wear particle concentration was direct reading (DR) ferrography. DR ferrography magnetically separates wear particles from lubricants and optically measures the relative concentration of particles present in the oil sample. The instrument is able to detect particles in the length range of 1–300 mm.

Test gears. Test gears were standard FZG type “A” spur gears. The test gears were designed with a large profile shift, which increases their sensitivity to adhesive wear modes of failure.

Uncoated test gears were made of DIN 20MnCr5 steel and were case carburized. The surface hardness after tempering was 60–62 HRC, with a case depth of 0.6–0.9 mm. The surface roughness was R_a = 0.35 μm for the pinion and R_a = 0.30 μm for the wheel.

The W-DLC coatings were deposited onto case-carburized type “A” spur gears by using a magnetron sputter deposition process, at a substrate temperature of about 200°C. The microhardness was about 1,200 HV. The primary coating constituents included W, C and H, with Cr used as a thin adhesion layer (150 nm). The coating thickness of the W-DLC layer was typically 1 μm at the root of the gear teeth and 2 μm at the tip.

Lubricants. The test lubricants were a complex ester formulation and a conventional mineral-based ISO VG 68 gear oil. Physical and chemical properties of the test lubricants are summarized in Table 1.

The saturated complex ester was composed of multifunctional synthetic alcohol, some petrochemical di-acids and some short chain (C8–C10) fatty acids from natural resources. The complex ester used as a base stock was nearly non-toxic for aquatic organisms and, according to the OECD 202 method, was classified as relatively harmless. Primary biodegradation in the CEC-L-33-A-93 test was 76.7% and ultimate biodegradation in the OECD 301F test was 62.2%. The degradation results identify a material that can be rapidly and extensively biodegraded in the environment.

Esters are inherently good boundary lubricants. However, some performance additives are still necessary. The additive system selected was based upon ashless components with mild EP being provided by an organic phosphorous-based chemistry. The EP additive was an amine

Table 1—Test Lubricants.

Property	Unit	Test method	Mineral oil formulation	Ester formulation
Density	kg/m³	ISO 12185	887	921
Viscosity @ 40°C	mm²/s	ISO 3104	68	48
Viscosity @ 100°C	mm²/s	ISO 3104	8.6	8.0
Viscosity index		ISO 2909	96	138
Pour point	°C	ISO 3016	–27	< –42
Elemental compositon	%m/m			
phosphorous		PML 07.18 (int.)	0.022	0.101
sulphur		ISO 14596	1.0	0.158

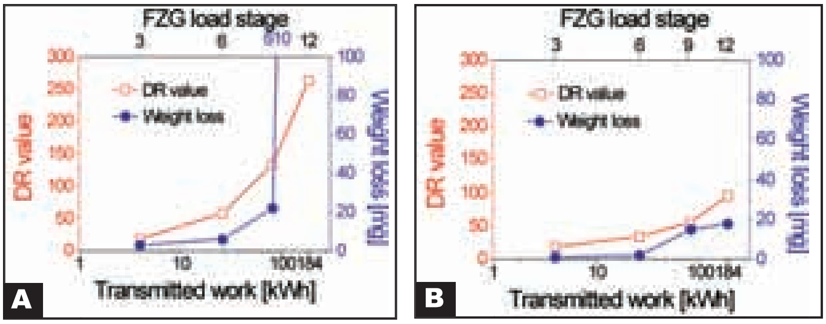

Figure 1—Wear measurement results for uncoated steel gears: a) lubricated with the mineral oil, b) lubricated with the ester formulation.

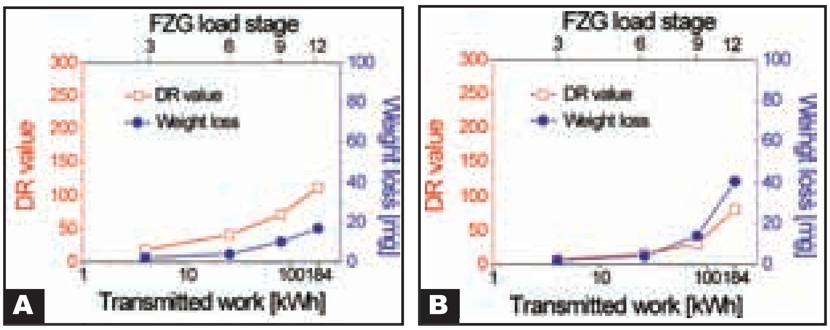

Figure 2—Wear measurement results for W-DLC coated gears: a) lubricated with the mineral oil, b) lubricated with the ester formulation.

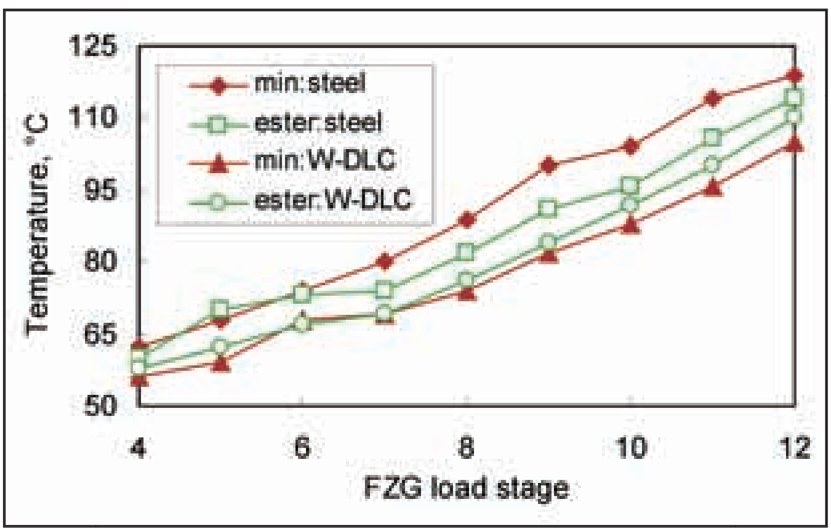

Figure 3—Bath oil temperature after the completion of load stage.

neutralized phosphoric acid ester, a common type of general purpose EP additive. The AW additive was a dialkyl dithiophosphate ester. Each additive was blended with the complex ester in the concentration of 1% (wt).

According to the gear manufacturer, the reference petroleum-based oil is recommended for heavily loaded gearboxes with surface hardened tool metallurgies. The mineral oil's viscosity was made one ISO viscosity grade higher than that of the ester-based oil to compensate for the effect of viscosity index difference, thus achieving about the same viscosity for the oils at working temperature (see Table 1), which was roughly around 100°C at higher loads.

Results

The most informative method for plotting wear results was found to be cumulative plots of the test gears' weight loss and wear particle concentration on the same graph with the reference to the total work transmitted and the FZG load stage.

The results of the scuffing investigations for the steel test gears are presented in Figure 1. It is evident that the scuffing load capacity of the ester formulation is higher compared to the mineral oil formulation. For the mineral oil, the weight loss of the test gears is within acceptable limits until 140 kWh of total work transmitted was reached. At 184 kWh, the cumulative weight loss of pinion and gear equals 610 mg, and all pinion flanks were damaged. With the ester formulation, the test gear's weight loss is much lower. After the test, the cumulative sum equals only 18 mg, and just a few scoring marks above the pitch line could be noted. The wear particle concentration results follow the gears' weight loss trend for both oils. The rate of wear particle concentration for the mineral oil gave rather high values, especially after the 6th load stage.

The wear results for W-DLC coated gears are presented in Figure 2 and show a steady, progressive increase in the test gears' weight loss and wear particle concentration for both oils.

Use of the ester formulation resulted in higher weight loss, while the wear particle concentration was lower compared to the mineral oil formulation. For ester formulation, scuffing marks became visible after 8 kWh of work transmitted and started at the root and later at the tip of the pinion teeth. After 15 kWh, most pinion tooth flanks were slightly polished. The first breakthrough of the W-DLC coating was observed at the root of the pinion after 26 kWh of work transmitted. For the mineral oil formulation, the coating breakthrough started at the same time after 26 kWh of work transmitted, but visible damage was more severe. Developing scoring damage was observed during the subsequent runs for both oils. After the test, the pinion flanks were polished, and the W-DLC coating was totally worn through at the root of the pinion.

Figure 3 shows the increase of bath oil temperature at the end of each scuffing load stage for both oils with uncoated steel and W-DLC coated test gears. The temperature increases steadily with the applied load for the oil and material combinations. Tests with the W-DLC coated gears resulted in lower temperatures, suggesting the surface tooth flank material is a stronger influence factor on temperature rise than lubricant used. The lowest oil bath temperature was found for the W-DLC coated gears lubricated with the mineral oil formulation.

Discussion

Wear results for the mineral oil formulation suggest that scuffing capacity is strongly influenced by the surface material of the test gears (see Figures 1a and 2a). The mineral oil and steel gear test combination exhibited the highest wear. Visual inspection of the pinion tooth flanks indicates the failure in the 12th load stage after 184 kWh of work transmitted. On the other hand, the mineral oil in combination with W-DLC coated gears resulted in significantly lower wear and passed the 12th load stage. In contrast, when using the ester formulation, wear results for steel and W-DLC coated gears are comparable and of lower magnitude.

Additional information about wear mode and mechanism could be obtained with the analytical wear particle analysis. Wear particles are the final product of surface damage, and their shape, morphology, size and concentration can give some information on the mode and the mechanism of wear. The wear particles are first fixed to a glass slide and then analyzed under an optical microscope.

Figure 4 displays the particles separated from the mineral oil formulation, and Figure 5 displays the particles separated from the ester formulation after 146 kWh total work transmitted, which is equivalent to the 11th test run in the modified FZG test procedure. Figures 4a, 4b, 5a, and 5b present typical wear particles from the entry region of the glass substrate. Particles at this location are typically the largest particles separated from the oil because the magnetic force, which attracts the particles, is proportional to the volume, whereas the viscous resistance of the particles to motion in the fluid is proportional to surface area. Comparing the photos, it is evident that wear particles obtained from the tests with W-DLC coated gears are larger than wear particles from the lubricants tested with uncoated steel gears. It can also be observed that wear particles from mineral oil are larger than particles from the ester formulation for both gear tooth flank materials.

The larger size of the wear particles separated from the mineral oil formulation for steel gears is expected because the coefficient of friction for ester-based lubricants and steel contact surfaces is typically lower (Refs. 6–7). The lower bath oil temperature (see Figure 3) also indicates the lower coefficient of friction for the ester formulation.

Even though the particles found in different gear material combinations discussed above are of different sizes and compositions, most of them are flat flakes having irregular shapes and generally featureless surfaces without characteristic

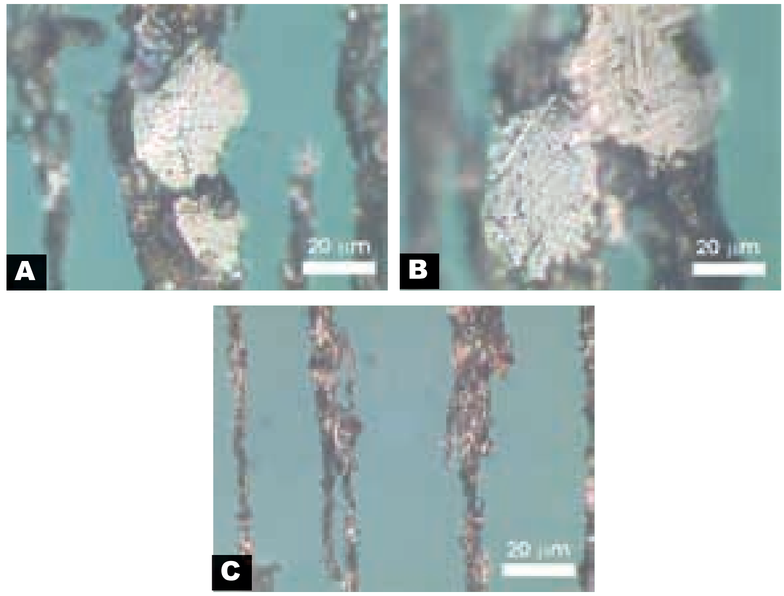

Figure 4—Entry region of the glass slide made from the mineral oil, magnified 500 times: a) uncoated steel gears, b) W-DLC coated gears, and c) W-DLC coated gears—the largest cutting wear particles.

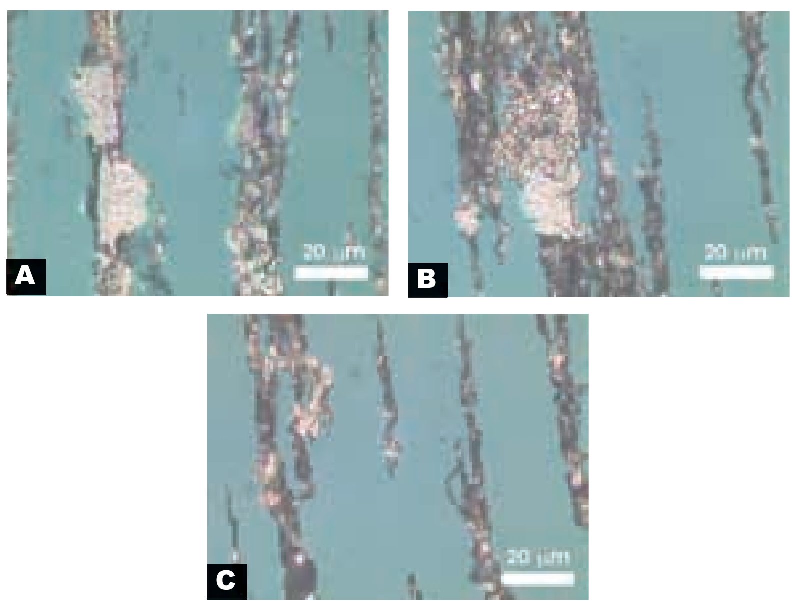

Figure 5—Entry region of the glass slide made from the ester formulation, magnified 500 times: a) uncoated steel gears, b) W-DLC coated gears, and c) W-DLC coated gears—the largest cutting wear particles.

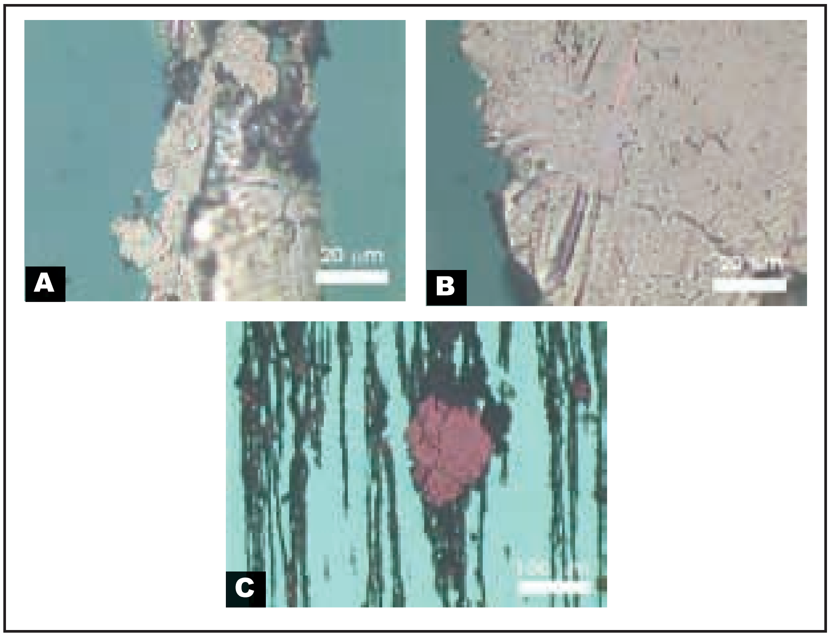

Figure 6—Glass slide made with the mineral oil from the test with the W-DLC coated gears, after 146 kWh work transmitted, magnified 500 times: a) large particles, b) the largest particle, and c) the largest particle, magnified 100 times.

striations indicating severe wear. In fact, this is the morphology observed for the majority of wear particles larger than 15 µm. This implies that the particles were all produced by the same wear mechanism. However, wear particles produced by the interaction of two component surfaces, such as gear teeth, are subjected to continuous high contact pressures and would therefore have a strong tendency to be flattened and smoothed by the forces acting on them. This process would account for the typical particle morphology observed and suggests considerable alteration to wear particle morphology occurs after the particles are produced.

Another characteristic group is cutting wear particles, presented in Figures 4c and 5c. Cutting or abrasive wear particles are produced by the penetration, ploughing or cutting of one surface by another. They take the form of miniature spirals, loops and bents. Their presence is abnormal. These types of particles are found only on the glass slides from the tests with the W-DLC coated gears. They are not found on slides made with lubricants obtained from the tests with steel gears. Also distinctive are very large flat particles obtained from the test with the mineral oil formulation and W-DLC coated gears. Figure 6 shows the largest particles separated from the oil. They ranged from 70–125 mm in major dimension, indicating a severe wear mode.

Very large wear particles and the presence of abrasive particles indicate the wear mechanism for uncoated and W-DLC coated gears is different. For the W-DLC coated gears, the wear probably started under the surface, while the prevailing wear mechanism for the uncoated gears is adhesive wear that started from the surface.

Conclusions

The following conclusions can be derived from this study:

1.) In tests with steel gears, the ester-based formulation resulted in higher scuffing load capacity than the mineral oil formulation.

2.) The scuffing performance of the mineral oil and W-DLC coated gears is significantly improved compared with the steel gears. However, some particles exceeding 100 µm in the major dimension indicate a severe wear mode. With the ester-based formulation, the wear rates for steel and W-DLC coated gears are similar.

3.) The surface tooth flank material is a stronger influence factor on temperature rise than the lubricant used.

4.) The wear mechanism for uncoated and W-DLC coated gears is different.

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How to Achieve a Successful Molded Gear Transmission

Rod Kleiss

Figure 1—A molding insert tool alongside the molded gear and the gear cavitiy.

Molded plastic gears have very little in common with machined gears other than the fact that both use the involute for conjugate action. The differences are quite fundamental. Machined gears are cut to size with specialized machinery designed specifically for the task. Molded gears are formed in gear cavities that are usually cut with wire electrical discharge machines (EDMs). A molded gear, its cavity and the molding insert tool are shown in Figure 1. These cavities are sized so that the molded gear will shrink to the proper size after molding. One cavity might be expected to form more than a million molded gears.

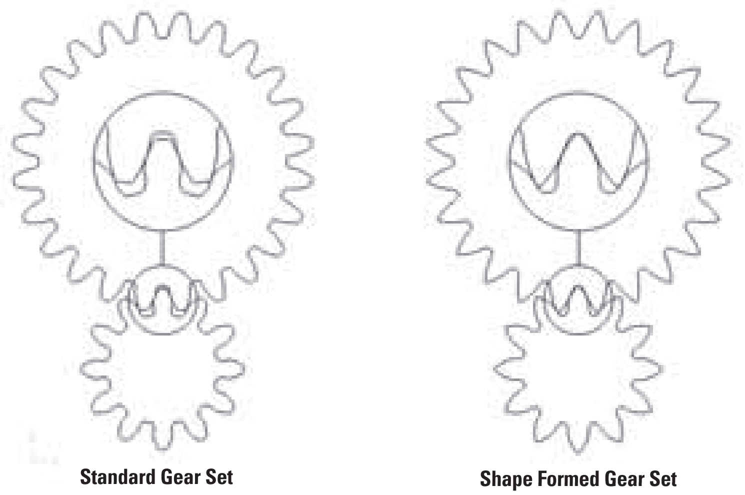
A gear cutting manufacturer is charged with the task of cutting gears within tolerance with every piece made. The gear mold manufacturer is charged with the task of making one nearly perfect gear cavity and then processing each gear from that cavity within tolerance for every piece made. This small but significant difference leads to many other variations. The differences begin as soon as the choice for molded gears is made.

Design

Molded gears invariably must operate in molded housings. This single fact has significant consequences. Molded housings and the shafts in them are rarely going to have the precision tolerances that a machined transmission can provide. The housings and gears will shrink and expand due to moisture and temperature, perhaps at different rates. The strength, hardness, and even efficiency of the plastic material will also vary due to local conditions. Surface tooth temperatures will rise under load, affecting plastic properties. These variables and others dictate a need for custom design of gear teeth.

The advantage the plastic gear designer has is in the application. Most plastic transmissions are unique. A gear mesh can be designed strictly for its intended function with a single mating gear (Fig. 2). Additionally, the molded gear can be optimized with very little regard for tooling (Fig. 3).

Wire EDMs can generate machined patterns with the precision of computer aided design. A gear cavity can be made with micron tolerances. Given the fact that traditional hobs are not required, diametral pitch or module are not important specifications. The involute base circle



Shared Attributes:	
Center Distance: 0.75" minimum	**Gear Ratio: 2 to 1 (24 to 12 teeth)**
Differences:	
Standard Gears:	**Shape Formed Gears:**
24 Diametral Pitch	0.11865 Base Pitch
20° pressure angle	25° operating pressure angle
0.06545 tooth thickness (both gears)	0.070 pinion tooth thk'ns/0.061 gear tooth thk'ns
Max contact ratio: 1.15	Max contact ratio: 1.59
Max center distance variation: + 0.022"	Max center distance variation: +0.032"

Figure 2—Comparison of standard gear mesh to custom shape formed gears.

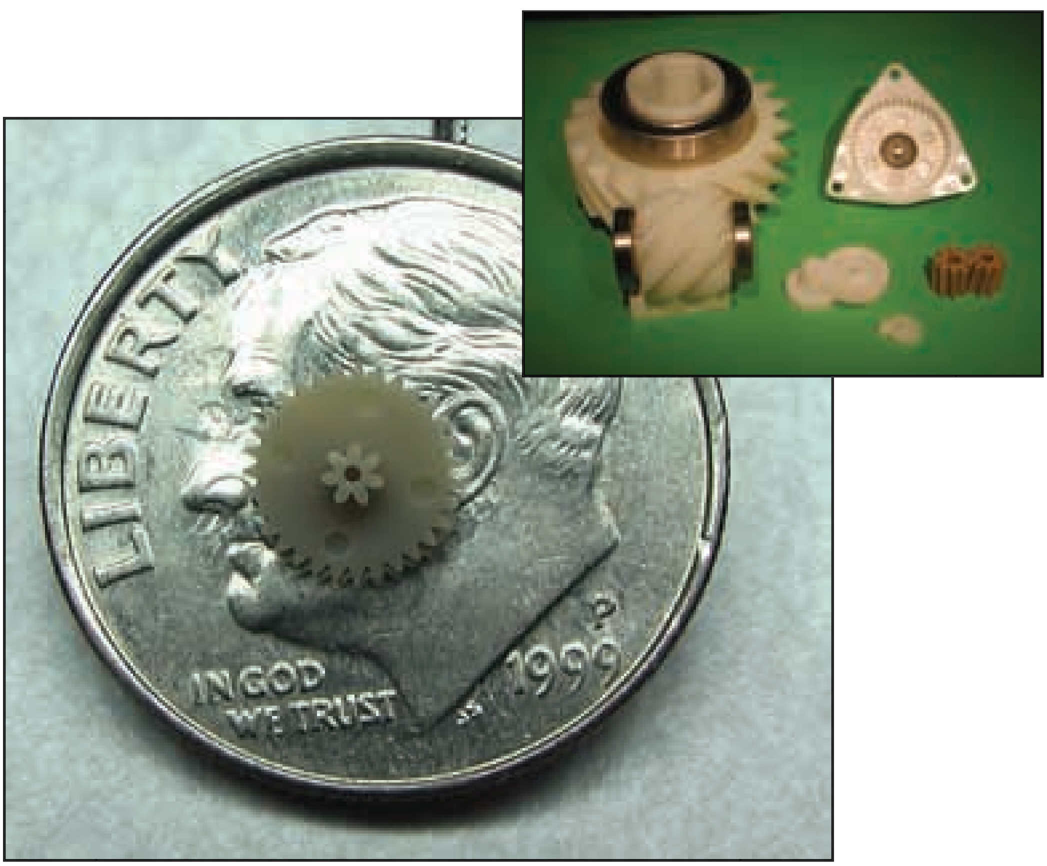


Figure 3—Molded gears can be made in many forms and varied sizes. A very fine pitch gear is shown in the larger image.

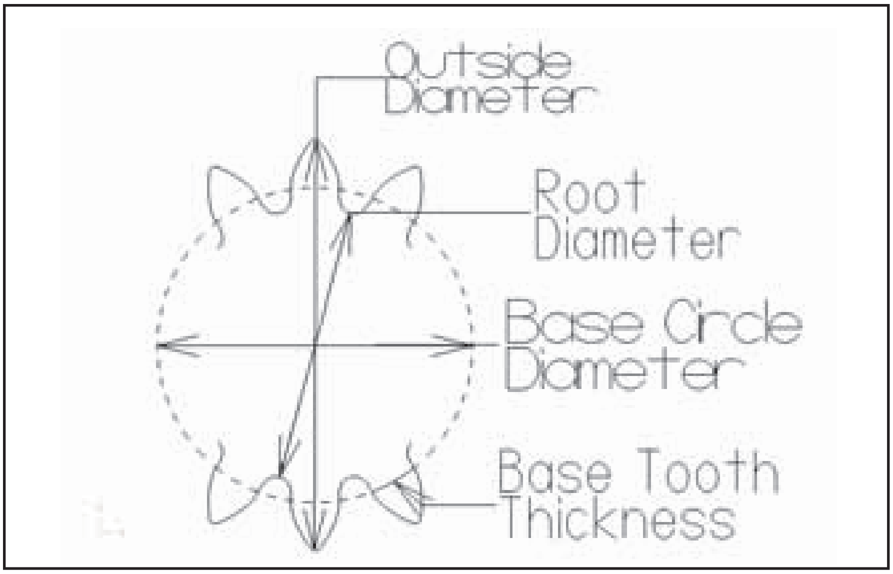


Figure 4—Distinct shrink rates for general plastic gears.

Figure 5—The involute shrinkage of a molded gear.

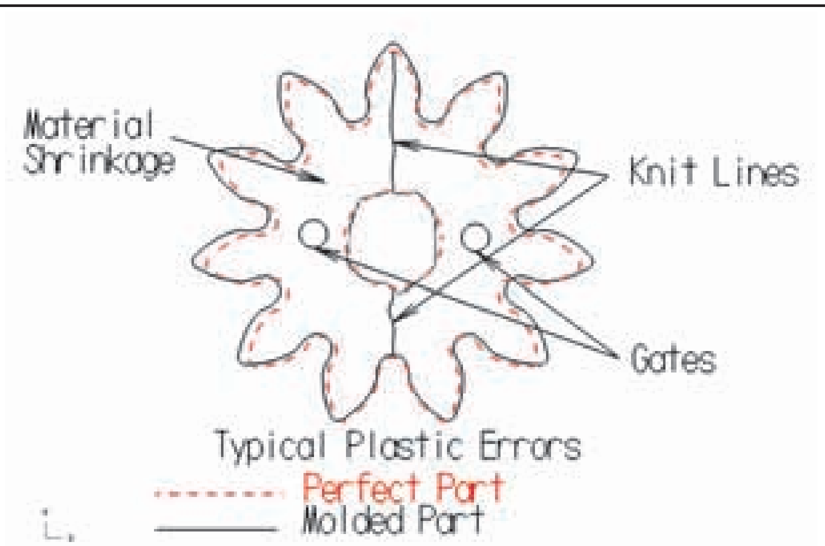


Figure 6—Typical errors in a molded part.

is the variable of importance. Pressure angles can be adjusted in an analog fashion to balance strength and depth of tooth engagement. Custom designed gears will offer a great improvement in performance, quietness, and allowable tolerances over standard gearing.

The Gear Molding Tool

With the gear mesh designed and toleranced, the next step is tool construction. Gear tooling must be precise, with excellent thermal stability, hardened sleeves and surfaces, exact gear cavity formation, and designed for high-pressure injection molding. The gear cavity itself must be specifically designed for the selected molding material.

There is no way to accurately predict the actual shrinkage for molded plastic gears in a specific application. This is due to a number of factors. Most importantly, plastic does not shrink from the cavity in an isotropic fashion (Fig. 4). The main body of the gear will shrink in a manner that may be similar to manufacturer's estimations, but the individual tooth is surrounded by steel and its cooling pattern will differ from the macroscopic pattern of the larger mass (Fig. 5).

A good method to determine shrink requires a two-step approach. Shrink factors are estimated for the gear in question. After the tool is made and the first gears are molded, they are then profile-inspected for exact involute geometry. The individual shrink rates are then determined, a new cavity is made to the measured shrink, and the final gear geometry is properly sized. Only profile inspection will be able to accurately determine involute shrinkage. Gear roll testing may give some idea of shrinkage anomalies, but it can also be misleading.

Sometimes heavily glass-filled material is selected for gears due to its low shrink rate. Shrinkage then becomes less of an issue in mold design. But this approach can cause its own problems. Unfilled engineering resins, such as nylon and acetal, mold into very precise shapes, albeit with shrinkage. Glass-filled materials will have knit lines where injection flow fronts merge. These knit lines can cause distortion at the tooth surface as well as localized weak spots on the gear (Fig. 6). Glass-filled gears will generally be much more abrasive during their life than equivalent unfilled gears. Generally, filler should only be used when a specific need has been established that outweighs potential problems.

Mold Processing

All molding is not equivalent. All molding machines are not equivalent. Gears require mold processing that is exact and repeatable. In general, virgin resin is used for high accuracy gears. Even with virgin resin, the material must be of correct dryness; its melt temperature must be controlled exactly and must be repeatable. Injection pressures must be established precisely. The interaction of the mold tool and process control must also be taken into account.

As plastic is injected at high temperature and pressure, the melt must displace air in the gear cavity. Vent paths must be created to allow air to escape, yet thin enough to stop the resin from venting as well. If the vents are too small, gas will be trapped and burning could result. If the vents are too big, plastic melt will flow through and cause flash on the part.

It is often advisable for the gearing customer to visit the molding facility before placing the final order. Just a cursory inspection of molding equipment, general plant cleanliness, inspection capabilities and personnel can help to evaluate the facility's potential for successful molding and control. For instance, it will be very difficult to mold precision gears in a non-temperature controlled environment. Molding precision gears in 90% humidity at 100°F is fraught with difficulty.

Inspection

Over the years, gear inspection has been refined to pinpoint most errors that create trouble in gear cutting. A profile scanning inspection of the involute profiles is usually done for only a few teeth around the gear. Metal gears are produced on turning machinery, and patterns can be expected from tooth to tooth. Plastic molded gears can have large solitary

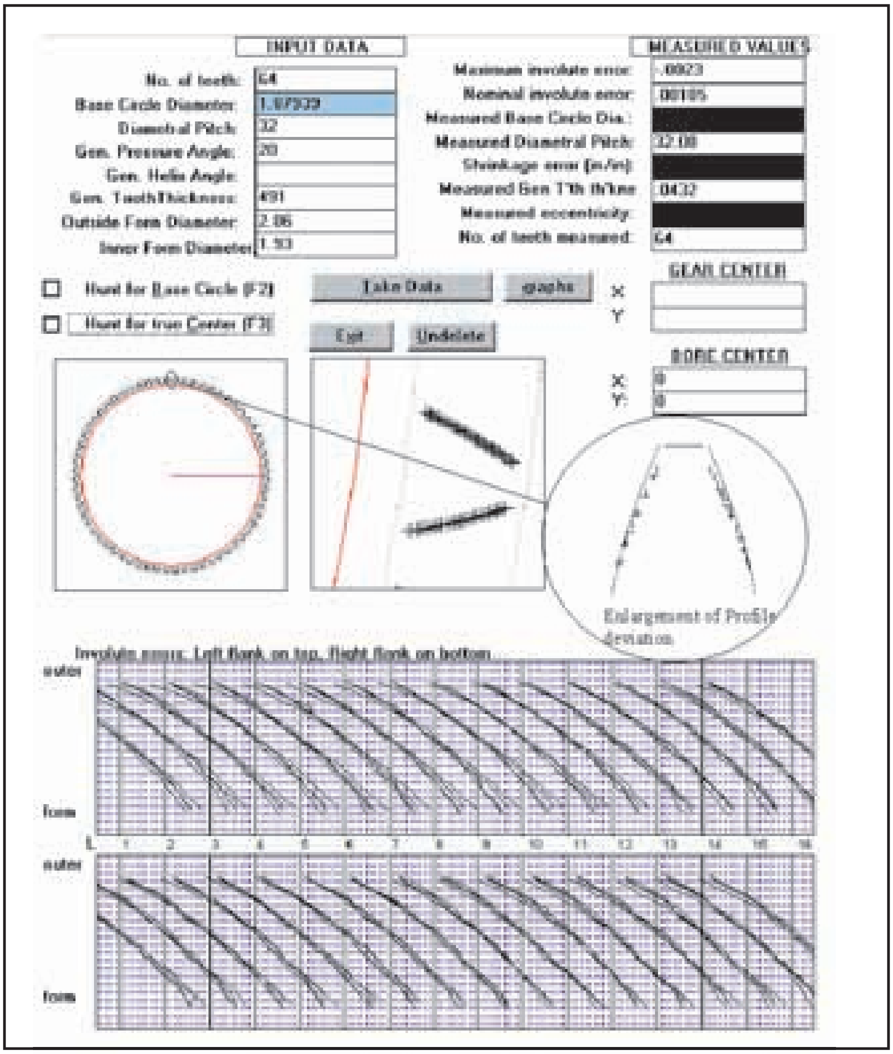


Figure 7—A poorly shrunken plastic molded gear.

errors anywhere on any surface of the gear. Furthermore, the molding process can introduce a much different kind of error than in traditional manufacture.

Since any molded gear will shrink, the involute profile is a target, not a given value. Whether one considers diametral pitch, module, base pitch, pressure angle or any other involute feature as the controlling geometry, this feature will be a variable in the actual part. It is necessary to set realistic tolerances for these truly variable features.

The only way to be certain that a plastic molded gear is within tolerance is by scanning the involute profile and determining the actual physical geometry of the gear. The molded part can be completely out of specification and still produce acceptable roll test results. Figure 7 shows a profile inspection of such a

Table 1—Suggested Gear Data Specification for Molded Plastic Gears.	
Number of Teeth	
Base Pitch (Basic Dimension)	
Base Circle Diameter	+/-
Base Circle Tooth Thickness	+/-
Root Diameter**	+/-
Outside Diameter	+/-
Involute Form Diameter	max
Tip Radius	max
Center Distance with Master Gear	TBD
Master Gear Specification	TBD
Tooth-to-Tooth Composite Error	max
Profile Form Tolerance (*fi*)	max

**Root trochoid must be directly generated. (Re: AGMA standard 1006-A97, Appendix F)

Operating Data	
Nominal Operating Diametral Pitch	
Nominal Operating Mesh Angle	
Nominal Operating Tooth Thickness	

gear. The involute base circle was very far off the defined value. The gear had 64 teeth, and a master used to measure the gear had 64 teeth. With such a large number of meshing teeth in roll testing, there was almost no tooth-to-tooth error. The gear simply appeared large, even though the base circle was small. The molder thinned the teeth, brought the gear into good specification with a roll test, and supplied parts to the customer. The parts immediately failed when meshed with a cut metal gear of correct size.

To prevent this type of error, the gear must be completely specified with each variable toleranced. One such method is recognized by the AGMA in the recently completed Information Guide for Inspection of Molded Plastic Gears. This specification layout is shown in Table 1.

Suggested Gear Data Specification for Molded Gears

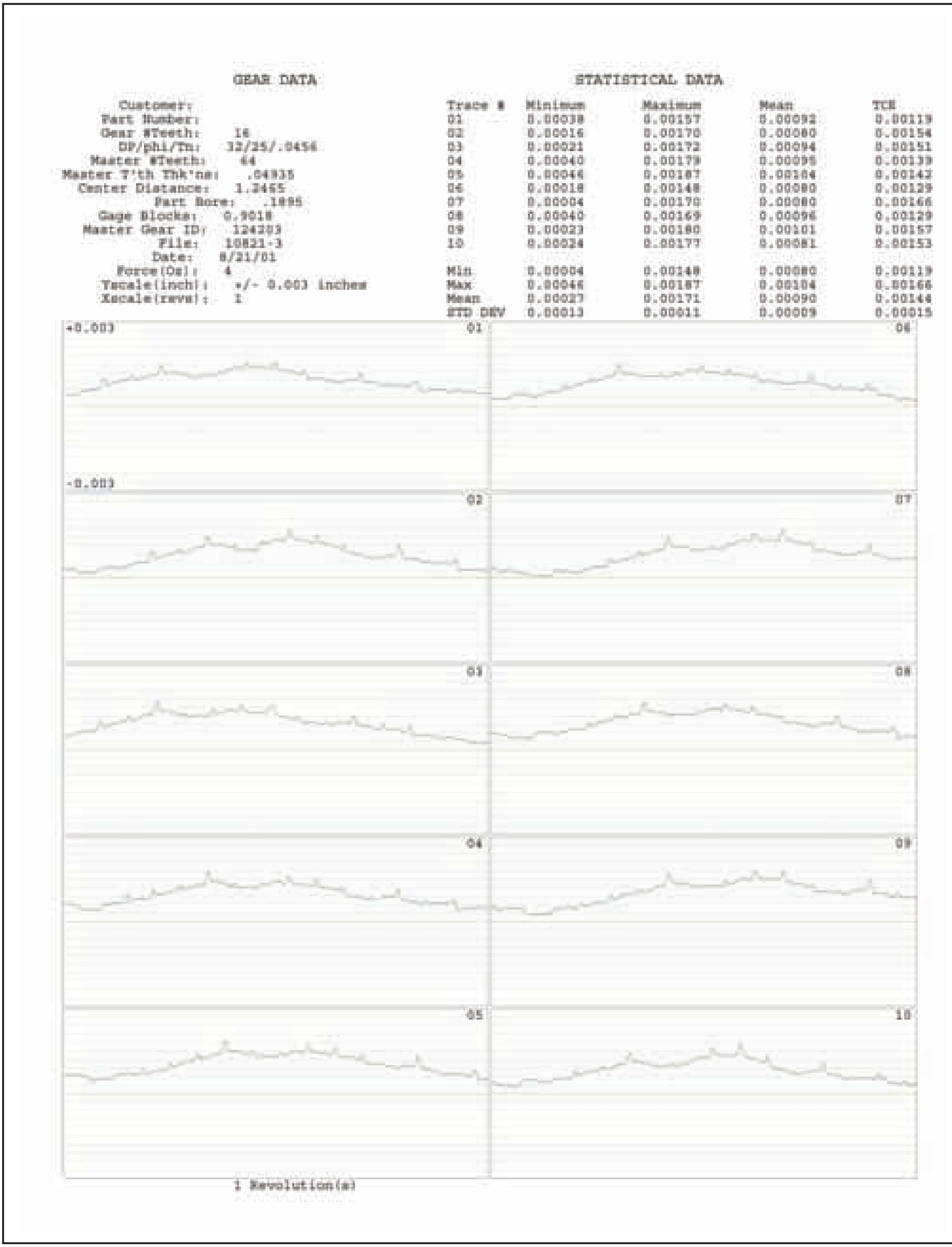
In the AGMA approach, the base circle geometry of the gear is used as the fundamental control. The indirect specification of diametral pitch and pressure angle are included in the operating data field as a reference for traditional analysis.

Gear roll testing is almost always the best way to assure consistency of the molded part in production. Rather than simply describe allowable total composite error (TCE) or tooth-to-tooth error (TTE), the actual center distance with a given master can be specified with indicated +/– tolerances (Fig. 8). This will provide an easy method to assure that the gears are molded consistently day after day. Roll tests of sample gears can be gathered to assure both the general form and the absolute size of the gears are within tolerance. Roll testing for plastic gears is more like establishing a roll test signature and confirming that the parts conform to that signature day after day.

The future for plastic molded gears is quite promising. Materials are improving greatly. Molding machinery is becoming more accurate. Inspection equipment is now capable of measuring these unique parts with great precision. In the future, plastic can be expected to replace metal gears in lighter duty applications. Companies continue to find uses for plastic gears in areas that cannot be served by metal gears.

In order to reach these new potentials, every step must be taken correctly and every advantage exploited. The result will be a remarkable new generation of power transmission products.

Rod Kleiss *spent the first five years of his engineering career working in the field of precision mechanics with Hewlett-Packard and Ball Aerospace. He has spent the remainder of his career focusing on plastic geared transmissions as a dynamically controlled precision mechanical system. Kleiss is president of Kleiss Gears Inc., a company that specializes in the design, inspection, and molding of high performance plastic gears.*



GEAR DATA

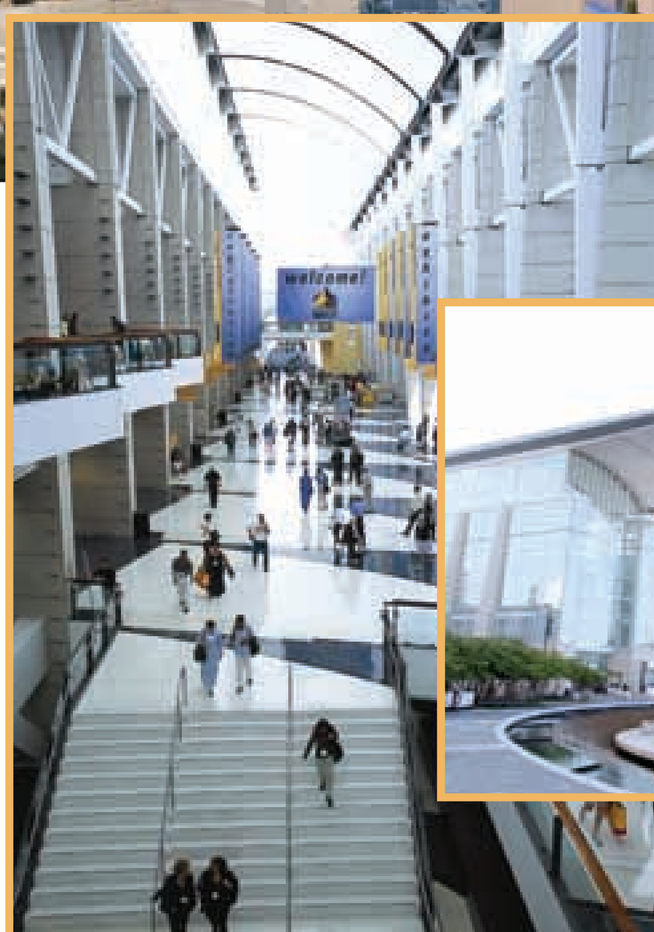
Customer:
Part Number:
Gear #Teeth: 16
DP/phi/Tn: 32/25/.0456
Master #Teeth: 64
Master T'th Thk'ns: .04935
Center Distance: 1.2465
Part Bore: .1895
Gage Blocks: 0.9018
Master Gear ID: 124203
File: 10821-3
Date: 8/21/01
Force(Oz): 4
Yscale(inch): +/- 0.003 inches
Xscale(revs): 1

STATISTICAL DATA

Trace #	Minimum	Maximum	Mean	TCE
01	0.00038	0.00157	0.00092	0.00119
02	0.00016	0.00170	0.00080	0.00154
03	0.00021	0.00172	0.00094	0.00151
04	0.00040	0.00179	0.00095	0.00139
05	0.00046	0.00187	0.00104	0.00142
06	0.00018	0.00148	0.00080	0.00129
07	0.00004	0.00170	0.00080	0.00166
08	0.00040	0.00169	0.00096	0.00129
09	0.00023	0.00180	0.00101	0.00157
10	0.00024	0.00177	0.00081	0.00153
Min	0.00004	0.00148	0.00080	0.00119
Max	0.00046	0.00187	0.00104	0.00166
Mean	0.00027	0.00171	0.00090	0.00144
STD DEV	0.00013	0.00011	0.00009	0.00015

Figure 8—Typical roll test signature of 10 molded gears.

AMT Expects Increase in Attendees Over 2004



IMTS 2004 brought just over 86,000 attendees to Chicago. Show organizers expect more than 90,000 this year.

IMTS 2006: Just the Facts

September 6–13, 2006
McCormick Place, Chicago, IL

Sponsored by

The Association for Manufacturing Techology

Pavilions:

- Abrasive Machining/Sawing/Finishing
- Controls & CAD/CAM
- EDMs
- Fluid Power
- Gear Generation
- Machine Components/Cleaning/Environmental
- Metal Cutting
- Metal Forming & Fabricating/Lasers
- Quality Assurance
- Tooling and Workholding Systems

Hours:

Lakeside Center: 9:00 a.m.–5:00 p.m.

North & South Buildings: 10:00 a.m.–6:00 p.m.

Sunday, September 10, hours for all buildings will be 10:00 a.m.–4:00 p.m.

Lines at McCormick Place's Starbucks concession stand will probably be a little longer at IMTS 2006, but the show's organizers won't be complaining.

John Krisko, exhibitions director at the Association for Manufacturing Technology, says this year's show should have some 90,000 attendees.

"Business conditions have improved, and we're basically sold out for exhibiting," says Krisko. "We've already surpassed our 2004 floor space and exhibitor numbers."

The association has been working for the past few years to ensure that show traffic reflects the growth of manufacturing in the Far East. To this end, the AMT has made presentations at the U.S. Department of State as well as the state departments of countries like China and India to familiarize the various foreign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bureaucracies with the show's mission.

"We've initiated presentations for the state department in India and China as well as our own in D.C.," Krisko says, "so that they know what our show is all about. Ultimately, that should make it easier when they encounter citizens in their own countries who would like to go."

Crash Course in Manufacturing for Students

Whether from the Far East or the Midwest, a portion of the show traffic will be younger this year. The AMT sponsors a student summit at the show, so high school and college students can view the exhibits and interact with manufacturing personnel. In 2004, more than 6,000 students and educators worldwide took part in the student summit, which opened with an orientation on careers in manufacturing.

Steve Glasder, a teacher at East Leyden High School in Franklin Park, IL, brought 93 students to the student summit in 2004 and commented that they saw more in one day at IMTS than he could hope to expose them to during the three-year machine tool program at the school.

Registration is free, and interested students and educators are encouraged to visit the show's website for additional information.

Emerging Technologies Help Manufacturers Stay Competitive

Even the IMTS visitor who has been in the industry for decades occasionally feels the need to "brush up." The Competitive Manufacturing Conference focuses on information designed to keep manufacturing companies current and competitive. Conference topics include smart machining, nanomanufacturing, micromanufacturing, additive manufacturing, lean manufacturing practices, case studies, outsourcing strategies and collaboration tools.

In addition, the Emerging Technology Center will showcase research in the field of manufacturing. When it debuted in 2004, the Emerging Technology Center aimed to create a link between research institutions and real-world manufacturers. Included in 2004's ETC were representatives from Penn State University's Machine Dynamics Lab, the Center for Intelligent Maintenance Systems at the University of Wisconsin-Milwaukee, the University of Toledo College of Engineering, the Y-12 National Security Complex, and the National Science Foundation Engineering Research Center for Reconfigurable Manufacturing Systems, to mention a few.

Registration Perks

Show organizers have arranged discounted rates at downtown hotels. Groups seeking 10 or fewer rooms are invited to book online on the IMTS website. Accommodations for groups requiring more than 10 rooms are handled by Connections Housing. More details are available by calling (800) 262-9974 or by e-mail at *monika@connectionshousing.com*.

Domestic travelers receive a 10% discount on United Airlines flights. International visitors are offered special discounts on any of the airlines in the Star Alliance. Both domestic and

international attendees can obtain the special prices using the same discount code: UA012S6.

Registration before August 4 is $25 per person. After August 4, registration is $50. Online registration closes Sept. 4.

Groups of five or more that register at the same time are charged $15 per person until August 4.

International registration is free for manufacturing industry personnel.

Metal-on-Metal Action

Comedy Central fans will be able to get their BattleBots fix while at IMTS this year. Robots will face off gladiator-style, and participants can handle the controls at the BattleBots IQ Booth D-4168.

A popular TV show, BattleBots first aired in 1999, featuring dueling homemade robots squaring off with help from rocket scientists, Hollywood special effects artists and die-hard fans.

BattleBots IQ, an educational program that includes curriculum, teacher training and a robotics competition, will be exhibiting at IMTS. BattleBots IQ was developed as a result of the TV series.

Comment on this article by sending e-mail to *publisher@geartechnology.com*

UTS Teaches Fundamentals to Optimization in its Metal Gear Course

Teaching gear designers about modifications for optimizing gear sets is the goal of "Metal Gear Design & Manufacturing" and is described by instructor Jim Marsch as the course's "most useful aspect."

Offered by Universal Technical Systems Inc., the class starts with a fundamental understanding of spur and helical gears and continues with discussion of topics like standard proportions, quality and gear design, but Marsch teaches them with a particular end in mind.

"What I'm leading up to is modifications," he says.

And what Marsch teaches is how to modify gears for greater load carrying capacity while still using standard forms. He also instructs students in the compromises that have to be made during modification, so they understand: "Not everything can be optimized at once."

But fundamentals first. The four-day course provides attendees with a basic understanding of involute spur and helical gear geometry, teaches them to apply gear design concepts in their work and helps them develop a working knowledge of gear software tools available from Universal Technical Systems. UTS provides software and consulting services for plastic and metal gears and holds the course in its headquarters in Rockford, IL.

Each student receives a 180-page course book with copies of slides from Marsch's PowerPoint presentations, as well as copies of problems and final reports on each problem. Also, students will be able to obtain older course material, from before UTS created Integrated Gear Software, its suite of gear design software. The instructor, Marsch, offers the material for its example problems. He also offers students a CD containing a presentation on planetary gear design.

Marsch himself is UTS' gear product manager. He's a gear engineer with more than 35 years of experience, including 12 years with Allis-Chalmers, where he designed agricultural tractor powertrains, and 22 years at Harnischfeger Corp., where he designed powertrains for cranes.

Besides classroom work, the course includes tours of two gear-industry companies in the Rockford area: Gleason Cutting Tools Corp. and Forest City Gear. Marsch takes students on a tour of Gleason Cutting Tools because: "I think it's very important for them to see the tools and how they're made." He takes them to Forest City Gear, a fine- to medium-pitch gear shop, so they can see how gears are cut, ground and inspected. "Most of the people who come to class have never seen gears cut," Marsch says.

At the end of the course, students are able to meet one-on-one with Marsch for an hour to discuss individual gear problems and questions. Attendees interested in one-on-one time should bring their prints and design problems.

Also, after registering for the course, attendees have free access to "Fundamentals of Gearing," an e-learning course from UTS. The online course is meant to teach basic gear design and manufacturing theory. The course provides explanations and formulas, graphics and animations for illustrating gear geometry, a glossary of gearing terms, and the ability to perform many calculations online using RuleMaster software, a Web version of UTS' TK Solver program, which is the calculation engine for Integrated Gear Software.

The Web material includes a training module, which UTS describes as "a handy platform for experiments," allowing users to enter different sets of data and view the results to get "a feel for the formulas." The course includes quizzes, and users can download course notes in PDF format and print them.

UTS will next hold the Rockford course Sept. 19–22. The class is limited to 15 attendees, who share eight workstations. As long as space is available, people can register for the course as late as Sept. 18. The class costs $1,250 per person.

Attendees reserve their hotel rooms themselves. However, UTS recommends the following hotels: Quality Suites of Rockford, Clock Tower Resort, and Candlewood Suites. Students reserving rooms at Candlewood can obtain a special room rate by mentioning UTS training.

For more information:
Kari Johnson
Universal Technical Systems Inc.
202 W. State St., Suite 700
Rockford, IL 61101
Phone: (815) 963-2220
Fax: (815) 963-8884
E-mail: ***sales@uts.com***
Internet: ***www.uts.com***

August 7–9—AGMA's Gear Manufacturing Technical Course. Liebherr facility, Saline, MI. Taught by Ron Greene and Geoff Ashcroft, the course covers gear theory, gear manufacturing, hobbing and shaping and their tools, production estimating, hard finishing, gear shaving and gear inspection. Tuition is $750. For more information, contact the Gear Consulting Group at (269) 623-4993.

September 11–15—AGMA Basics Course for Gear Manufacturing. Richard J. Daley College, Chicago, IL. Course features nomenclature, principles of inspection and gear manufacturing methods, including hobbing and shaping. $750 for AGMA members, $850 for non-members. For more information, contact the AGMA at (703) 684-0211 or by e-mail at *fentress@agma.org*.

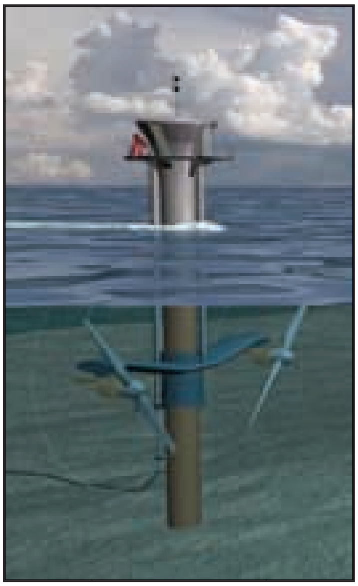
September 13–15—Basic Gear Noise Short Course. Department of Mechanical Engineering, Gear Dynamics and Gear Noise Research Laboratory, Ohio State University, Columbus, OH. Fundamentals of gearing, noise analysis and measurements are covered, and lectures are interspersed with demonstrations of the GearLab's measurement and computer software capabilities. $1,450 for the basic course, $2,200 for both the basic and advanced courses. For more information, contact the GearLab at (614) 688-3952 or via *www.gearlab.org*.

September 18–19—Advanced Gear Noise Short Course. Department of Mechanical Engineering, Gear Dynamics and Gear Noise Research Laboratory, Ohio State University, Columbus, OH. This is directed towards individuals who have already taken the basic course. The advanced course will consist of lectures and hands-on workshops. Based on student interest, course discussion may include subjects such as computer modeling, transmission error prediction, general system dynamics, bearing/casing dynamics and others. $950 if taken alone, $2,200 for both courses. For more information, contact the GearLab at (614) 688-3952 or via *www.gearlab.org*.

September 18–19—Gear Failure Analysis Seminar. Big Sky Resort, Big Sky, MT. Participants examine types of gear failure, such as macropitting, micropitting, scuffing, tooth wear and breakage. Possible solutions to these failures are presented. Early registration is recommended, as the June course was sold out. $645 for AGMA members and $820 for non-members. For more information, contact the AGMA at (703) 684-0211.

Gearboxes Go Underwater to Turn Tide's Energy into Electricity

Off the British coast, a U.K. company has been testing for three years a prototype 300-kilowatt, underwater turbine that converts tidal energy into electricity and is preparing to install a second prototype, an underwater turbine able to generate more than one megawatt of electricity.



Marine Current Turbines Ltd. is preparing to install its one-megawatt prototype for testing its ability to convert tidal energy into electricity.

Turbines fitted with rotors have been used for decades to convert wind energy into electricity, but they're now being adapted for offshore use underwater, to take advantage of another renewable energy source: tidal currents.

The U.K. company developing these prototypes is Marine Current Turbines Ltd., based in Bristol, England, but the company developing the latest prototype's gearboxes is Orbital2 Ltd. of Llangammarch Wells, Wales. Orbital2 is a gear consultancy for wind turbine, aerospace, automotive, industrial and marine applications. It specializes in epicyclic gear transmissions, as reflected in its one-megawatt turbine gearboxes.

Each of the new prototype's two gearboxes has a first-stage epicyclic transmission consisting of eight planetary spur gears and a second stage consisting of four planetary spur gears, followed by a parallel shaft output stage. All the gears are DIN 6 or better in quality.

Orbital2 created the gearboxes using a flexible pin design for ensuring equal load sharing between the transmission's planetary gears and across the tooth face widths and for providing uniform planet bearing loads. Frank Cunliffe, Orbital2's managing director, sees the flexible design as a virtue in marine turbines because: "You can't build infinitely stiff assemblies."

Also, input speed is 14.3 rpm, and output is 1,000 rpm. Cunliffe expects the turbine to run seven days a week, for about 20 hours a day: eight or nine hours in one direction and 11 or 12 in the opposite direction.

Each gearbox, generator and rotor combination will be mounted on a winglike extension of the turbine's monopile, which will be set in a hole drilled in the seafloor and will extend above sea level. Each generator will be able to create more than 500 kilowatts of electricity, and the axial-flow rotors will be able to turn via flow in either direction.

Though underwater, the extensions will be part of an assembly that can move up and down the monopile, allowing the rotors and generators to be lifted out of the water for maintenance. Marine Current Turbines has started shore-based testing of the new prototype and plans to install it in October in Northern Ireland's Strangford Lough. The turbine will be connected to an electricity network, and MCT expects it will be able to supply energy to about 800 houses.

MCT has designed its tidal turbines for erecting in water with depths of 20–30 meters, but the company has ideas for installations in deeper waters, in places with sufficiently fast, continuous ocean currents.

"We've stayed in 20–30 meters to prove the technology," says Joe Verdi, MCT's commercial director. Verdi is responsible for developing the company's technology for commercial use.

Proving the technology started in May '03, when MCT installed and began testing its 300-kilowatt prototype off the coast of Lynmouth, Devon. Verdi says the turbine's power coefficient (C_P) is on the order of 0.45.

A C_P is a measure of the fraction of power in the fluid flow that's extracted by the turbine. An amount of energy is lost due to natural forces, such as friction, so the turbine never extracts all the power. In the wind energy industry, for example, wind turbines rarely achieve a C_P of more than 0.50, meaning they rarely extract more than 50 percent of the wind's power and convert it into rotating energy to power their generators.

MCT's 300-kilowatt turbine is a simpler design than its one-megawatt turbine, though. The 300-kilowatt machine uses a single rotor, which operates with the tide in only one direction. The one-megawatt's two rotors will operate with tidal flow in either direction.

The twin-rotor system is the one MCT is bringing to the fore. The company plans for its turbines to be commercially available

Orbital2's gearboxes for the one-megawatt prototype feature eight planetary spur gears in the first-stage transmission, with four planetary spurs in the second stage, followed by a parallel shaft output stage.

in 2009 and plans to install twin-rotor turbines in locations around the world to develop the technology in other markets. Verdi says MCT is interested in France, North and South America, Southeast Asia, Australia and New Zealand.

Created in 2000, MCT was formed to develop cost-effective, reliable marine turbines to generate electricity in large-scale commercial developments: tidal farms. In line with that purpose, MCT is developing plans for a 10-megawatt tidal farm to generate electricity for about 5,500 homes.

MCT has received significant financial support from the U.K. Department of Trade and Industry, the European Commission and the German government.

However, the payoff could be significant, too: a new source of considerable energy, a source that's also reliable. "Tidal flows are consistent," Verdi says. "They are very predictable."

Bison CEO Appointed to National Education Commission

Ron Bullock, chairman and CEO of Bison Gear and Engineering, has been appointed a member of the Commission on 21st Century Education in Science, Technology, Engineering and Mathematics.

Ron Bullock

Testimony delivered before the National Science Board on K-16 STEM education led to an invitation to serve on this commission. The commission makes recommendations to the nation, via the National Science Board, for a new national action plan to address weaknesses in K-12 STEM education.

According to the company's press release, Bison has been recognized for its research to develop a high efficiency motor under a grant awarded by the National Science Foundation.

Bullock has held various positions with Bison since 1981, in R&D, engineering, marketing, operations and management. He has served on the Industry Advisory Council on Electronic Motors at Underwriters Laboratories, was a past director of the American Gear Manufacturers Association, led a research team for the Instrumented Factory for Gears at the Illinois Institute of Technology and served as president and trustee of the Gear Research Institute.

LMT Fette Opens Midwest Hob Facility

LMT-Fette established a new precision manufacturing facility in Libertyville, IL.

The first phase of the new plant will handle production of hobs from 8 DP to 30 DP. Hob re-sharpening and re-profiling services are available for hobs up to 9" in diameter.

Fette offers hobs in a full range of pitch sizes in premium powdered metal, solid carbide and indexable hobbing solutions.

Overton Gear and Klingelnberg Form Strategic Alliance

Klingelnberg GmbH and Overton Gear and Tool Corp. have formed a strategic alliance for manufacturing and supplying high quality, large spiral bevel gears for the worldwide market.

Customers should continue to contact their current sales representatives for ongoing and future business.

Overton Gear manufactures custom gears for the marine, off-shore, locomotive, mining, wind energy and construction industries. Located in Addison, IL, the company acquired Illinois Gear Corp. in May 2005.

The Klingelnberg Group develops, manufactures and sells gear production machinery and related equipment. Klingelnberg products are used in the automotive, truck, aircraft, agriculture, construction, power tool and marine industries. The company has operations in Zürich, Switzerland; Hückeswagen and Ettlingen, Germany; Györ, Hungary; and sales and service offices throughout Europe, North and South America and the Asia-Pacific region. Klingelnberg, along with Liebherr-Verzahntechnik, is part of the Sigma Pool international gearing partnership.

LMT Opens Automotive Support Center for North America in Detroit Area

The Leitz Metalworking Technology (LMT) Group officially opened its North American automotive support center June 8 in Auburn Hills, MI, a Detroit suburb.

"Our customers are looking for local support," said Dieter Brucklacher, executive chairman of the Leitz Group, who spoke at the center's opening ceremony. Brucklacher added that customers rated local engineering support as important as the product itself.

The new center offers engineering, service and tool management support. The facility was created to support the auto and machine tool industries with LMT's know-how in cutting materials and precision tools. To that end, the center is connected via an internal broadband network for real-time access to other LMT companies and to the LMT engineering center in Oberkochen, Germany. The network also allows for real-time video conferencing.

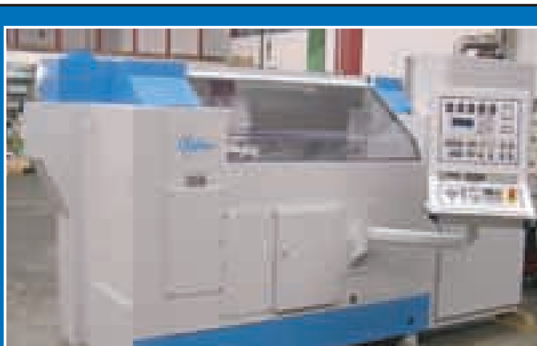
LMT will also provide training at the center for employees of auto, machine tool and die-and-mold manufacturers. Practical training will be carried out on the center's machine tools, which are also used for customer tests and tool demonstrations. The machine tools include a regrinding machine for refurbishing and modifying tools.

Dieter Brucklacher, Leitz Group executive chairman.

Photo courtesy of Daniel Lippitt.

In a press release, LMT described the facility as "a central element of the growth strategy with which the LMT Group aims to substantially increase sales in the North American market over the next few years." LMT has spent more than $2 million to enhance its service capabilities for the American market.

"LMT Fette already has a substantial presence in the



U.S.," Brucklacher said, referring to the LMT company that manufactures precision milling and gear cutting tools. LMT Fette's U.S. operation is located in Cleveland, OH, and includes customer service via a gear solution center.

Besides Fette, Brucklacher also mentioned another LMT company, Onsrud Cutter, which is based in Libertyville, IL, and produces end-milling cutters for the high-speed machining of aluminum, plastics and composite materials for the aerospace industry.

LMT plans to open other automotive support centers with comparable services in China, India and Brazil.

The LMT Group consists of six companies—Belin, Bilz, Boehlerit, Fette, Kieninger and Onsrud Cutter—and employs 3,000 people worldwide. The group manufactures precision tools for processes used to cut metal and plastics.

Getrag Group and Getrag Ford Transmissions Combine Under Single Brand

Fourteen individual companies in the Getrag Group and Getrag Ford Transmissions, all specializing in transmission systems, drivetrain components and engineering services, will be combined under the umbrella of the Getrag Corporate Group.

"Our aim in repositioning ourselves is to strengthen our market position in the global competitive environment and, in doing so, to create the conditions for further expansion," says Tobias Hagenmeyer, president of the Getrag Corporate Group. "We are planning a significant expansion of our activities across the world in the area of dual clutch transmissions. We are forecasting production volumes of up to 2.4 million dual clutch transmissions and total output of around 5.5 million transmissions by 2015."

Getrag also announced that Getrag Jianxi Transmission Co. Ltd., the joint venture between Getrag and the Jiangling Motor Co. Group in Nanchang, China, begins operations later this summer, as soon as the contract review process has been completed. Approximately 2,400 workers at the three production sites in Nanchang, Ganzhou and Yudu in southeast China will produce some 240,000 transmission. One million of the transmission components for the Chinese markets are anticipated as well.

Do you have news? Send it to: Robin Wright, Assistant Editor, *Gear Technology*. P.O. Box 1426, Elk Grove Village, IL 60007, Fax (847) 437-6618 or e-mail *robin@geartechnology.com*

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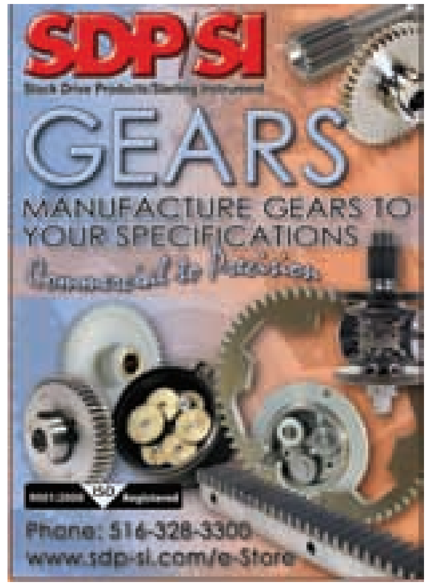

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A Movie for Gearheads and Other Mechanical People

Hollywood just doesn't cater to the gearheads of the world, but the truly devout mechanical enthusiast can find solace in Blockbuster's new release section this summer.

Burt Munro, played by Anthony Hopkins in "The World's Fastest Indian," wanted only one perfect run on the Indian Scout that he bought for $50 in 1920 and spent his whole life tinkering with in his New Zealand garage.

The Scout had a 37-cubic-inch, 42° V-twin engine with side valves. A helical gear primary drive was contained in an oil-tight, cast-alloy case, and a three-speed, hand-change gearbox with a foot clutch was fitted. A double-down tube cradle frame was used, rigid to the rear, and a leaf spring provided the forks with nearly 2" of movement at the front.

What did Munro do to prepare his bike for worldwide acclaim? Probably nothing the AGMA would recommend. He used an old spoke for a micrometer and cast parts in old tins. He built his own four-cam design to replace the two-cam system and converted to overhead valves.

He made his own barrels, flywheels, pistons, cams and followers and lubrication system. More or less, he hand-carved his con-rods from a Caterpillar tractor axle and hardened and tempered them to 143 tons tensile strength. He built a 17-plate, thousand-pound pressure clutch and used a triple chain drive. He experimented with streamlining, and in its final form, the bike was completely enclosed in a streamlined shell. The leaf-sprung fork was dispensed with and what appears to be a girder fork from a 1925–1928 Prince was substituted.

And the result of all this blood, sweat and tears? Munro took out a second mortgage to travel to Utah and race on the renowned Bonneville Salt Flats. As avid racing fans already know, he captured a land speed record at the flats in 1967.

Whether a sequel is in the works is not for The Addendum Team to reveal. Suffice it to say, we think the E! True Hollywood Stories of Gear Heroes could be a blockbuster franchise.

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