GEAR TECHNOLOGY
November/December 2010
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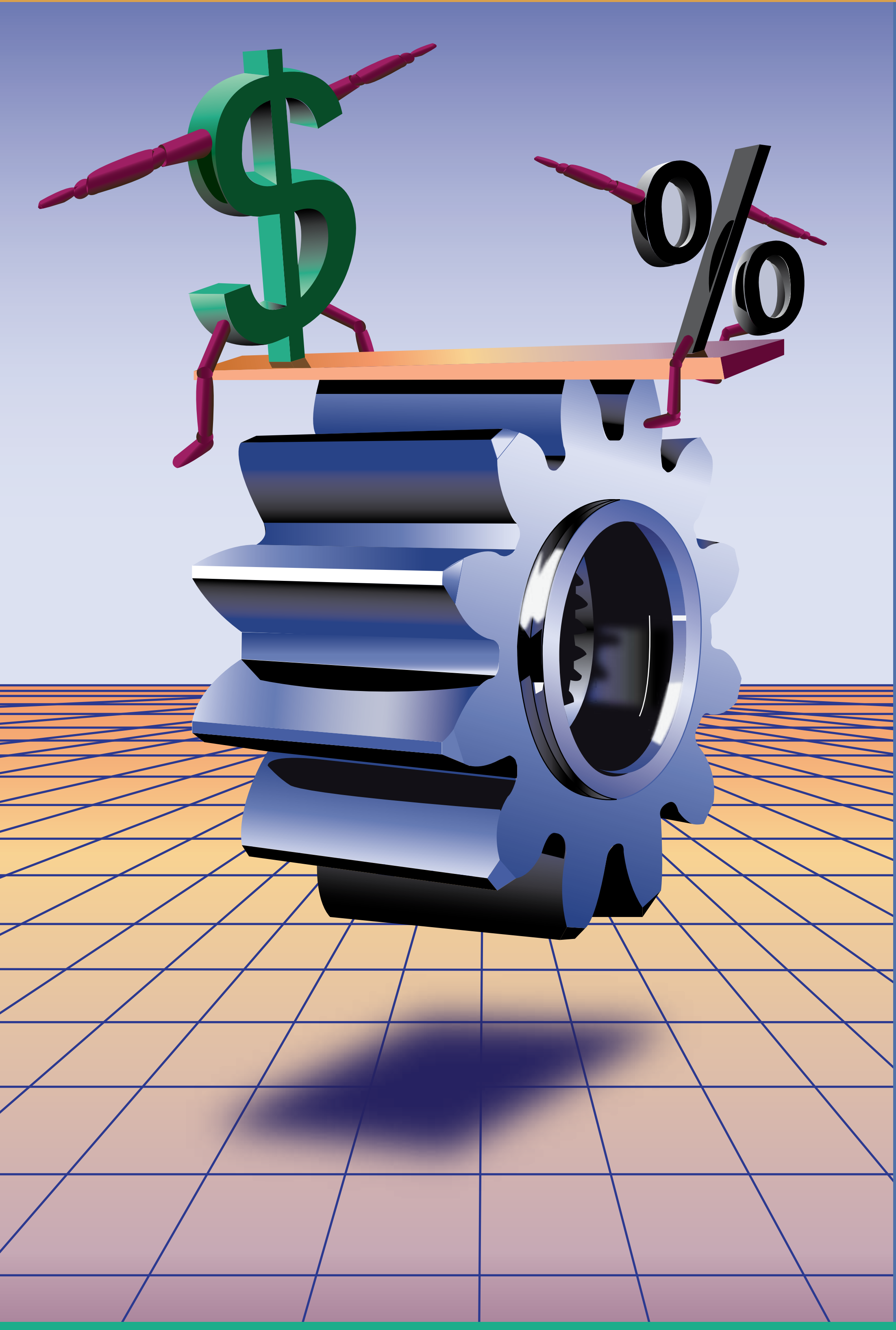
The Journal of Gear Manufacturing



State of the Gear Industry 2010

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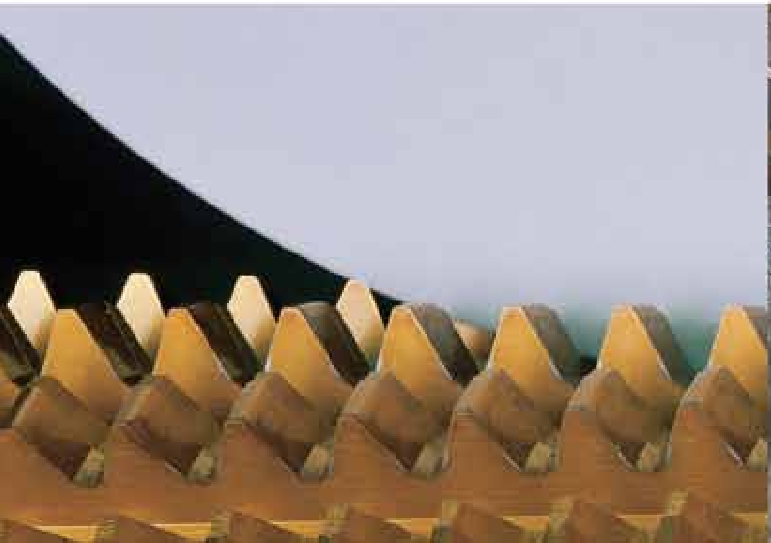

GEAR TECHNOLOGY

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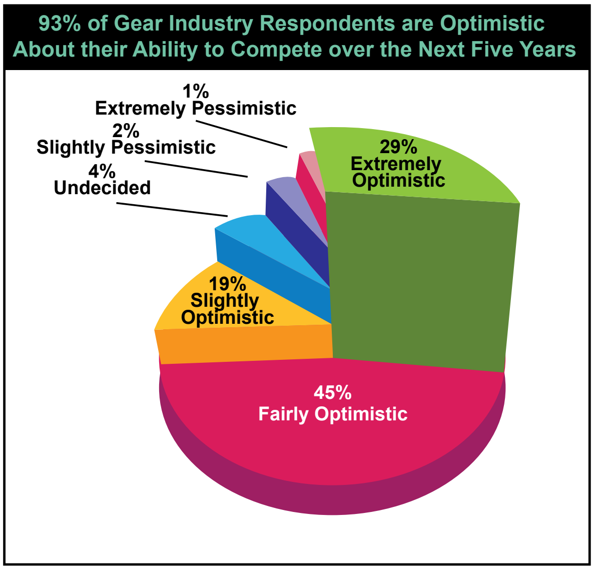

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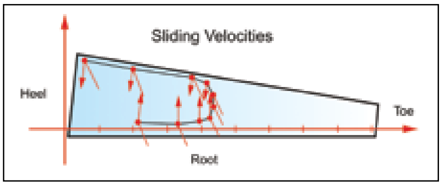

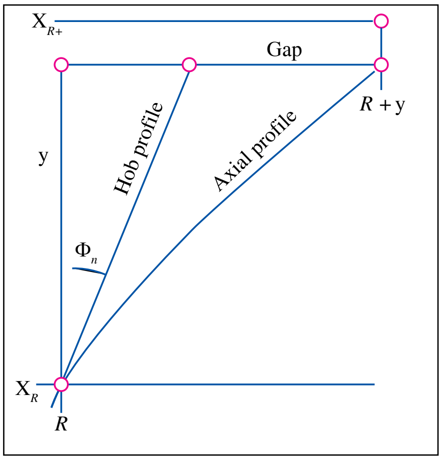

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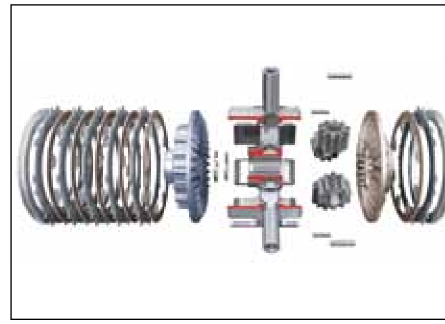

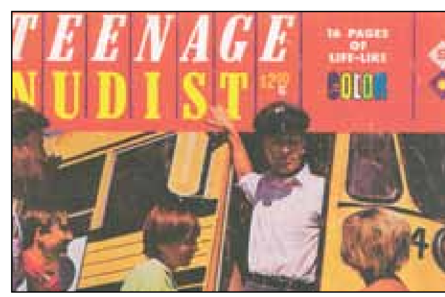

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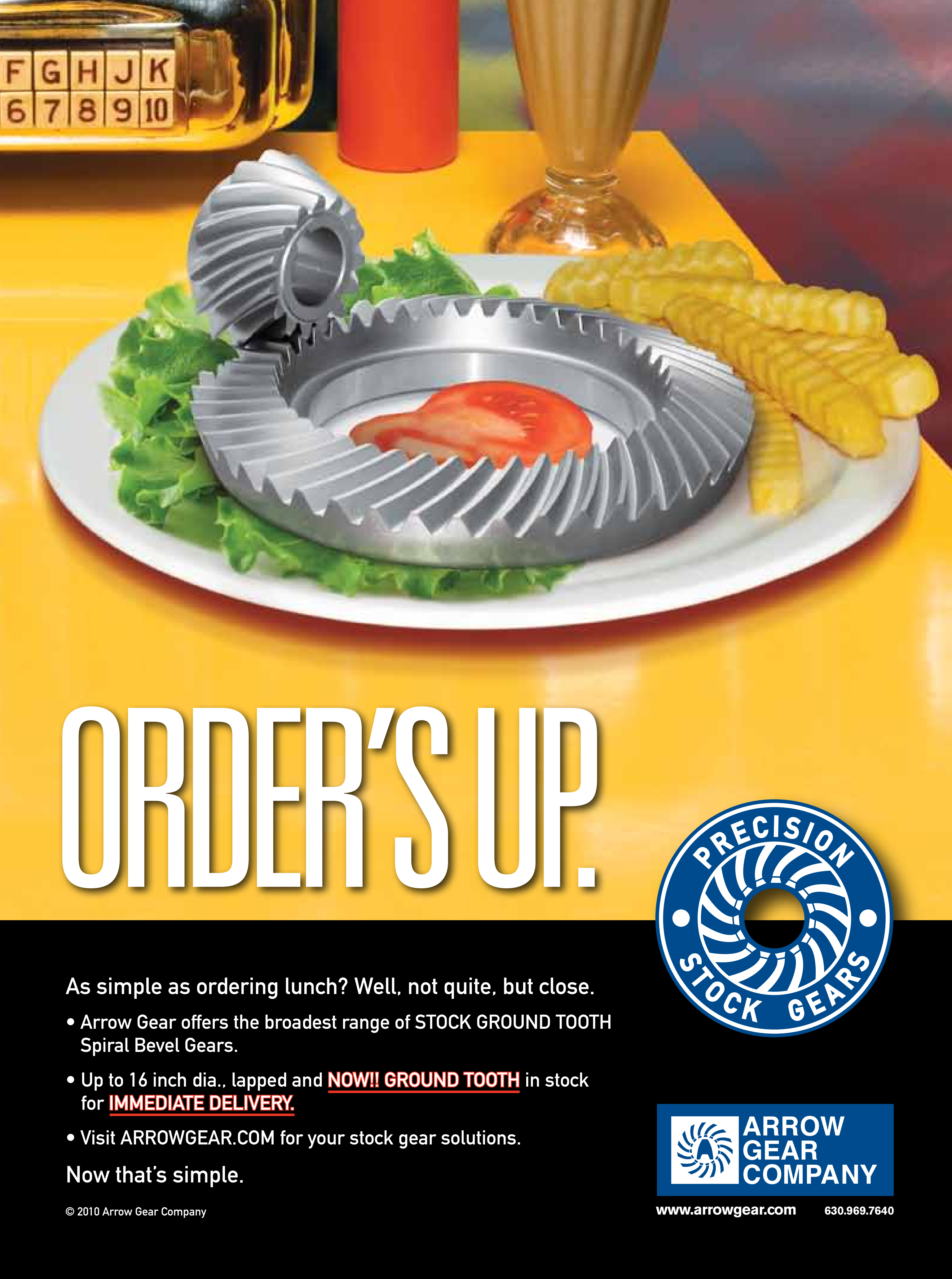

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The Slow Climb

Imagine a mountaineer, who had very nearly reached the summit of a great peak. He was confident, smiling and moving steadily upward—until the ground gave way beneath his feet and he tumbled back down the slope.

That was the manufacturing sector a year ago.

Now we're seeing signs that our climber is on the way back up. His feet are firmly planted, and he's once again headed in the direction of his quest.

Of course, it all started with the beginnings of recovery in the automobile industry. Both General Motors and Chrysler have emerged from bankruptcy and are back in the business of making cars again. During their struggles, the auto companies tightened their belts. They used up inventories because they had no money to spend. Now those inventories are diminished, and the automobile companies have money to spend again, and the trickle-down effect is helping many suppliers, including gear manufacturers.

At IMTS, we saw great evidence of gear industry activity across many sectors. There were times when the booths of all the major suppliers to the gear industry were overflowing with foot traffic. And it wasn't just one or two booths.

In fact, we found that the overall attitude at IMTS was extremely positive—to the point that more than a few attendees and exhibitors were somewhat embarrassed by the fact that they're doing so well. About a week ago, I asked a gear manufacturer how his business was. He, too, is extremely busy, but like those we met at IMTS, he was hesitant to admit it. His comment was that he's quietly wishing for an eighth day in the week in order to keep up with the demand.

It's almost as if these gear manufacturers don't want to jinx their success by boasting of it. It's almost as if they believe their surge in business could disappear as quickly as it came.

And maybe it could.

But still, there's more evidence that things seem to be going well in the gear industry. The results from our fifth annual State of the Gear Industry survey begin on page 29 of this issue. Overall, there has been a significant improvement in the attitude among gear manufacturers, with 93 percent of respondents optimistic about their ability to compete over the next five years (compared with only 85 percent last year). Also, whereas most gear industry respondents worked at locations where sales, production and employment went down in 2009, most have seen production and sales rebound significantly in 2010, with employment either holding steady or increasing.

Based on everything I've seen, heard and read, our mountaineer does seem to be on solid ground. Sure, he's hesitant. He's being careful because he doesn't want to fall down again.

One of the things about mountaineering is that it's important to have the right gear. You can't expect to climb without the right crampons, pitons, ropes and carabiners. The same is true in manufacturing. You have to have the right gear.

One of the benefits of the recession is that most companies have been hesitant to spend money, and so they've been accumulating cash. Now that business is picking up, many are coming to the realization that they need to start spending some of that cash to invest in the future.

I just hope they're not too late.

The amount of time between the day you decide to purchase a machine tool and the day it becomes a regular part of your production mix is often more than a year, and sometimes as much as two years. Not only do you have to place the order, you also have to wait for delivery, prove it out with sample parts, train your people how to use it and figure out how it fits in your production flow. That process just takes time. The part of this process over which you have the most control is choosing when to make the initial decision.

Don't forget that the busier the machine tool suppliers get, the longer the process can take. Machine tool manufacturers have limited capacity. So as their customers continue to get busier, the waiting lists and delays will grow, especially for certain types of machines and specific models.

So if you don't start that one- to two-year process today, you're in danger of being left behind. As companies are competing with other suppliers all over the world, the productivity advantages of new technologies are more and more relevant.

I know that with all that's going on in the rest of the economy, it's hard to believe that the manufacturing economy will continue to grow. But if you don't take advantage of the opportunity to invest in your business now, you may find yourself halfway up the slope without any extra rope.

Michael Goldstein
Michael Goldstein,
Publisher & Editor-in-Chief

The AGMA Brand

David Ballard, chairman, AGMA

I am well into the second year of my two-year term as chairman of the AGMA. Although our industry has slowed during the economic downturn, AGMA has been able to use the downturn in the economy to move ahead in many facets of our programs that will benefit our members for many years to come. Lately I am gaining a better understanding of how the "brand" of AGMA is viewed by our members and the industries we serve.

AGMA's brand begins with the standards that we have developed for the gear industry. Since its beginning in 1916, AGMA has been recognized by companies around the world for developing and maintaining the domestic gearing standards for the United States and for leadership in the development of ISO international standards. The AGMA logo that members display on their products, literature and websites represents the design and quality that is behind the gearing standards established by the association. Ultimately this should transcend to the end user, who, upon seeing the logo on products also recognizes the value behind the logo.

As for AGMA, our staff continues to work on educational programs that benefit manufacturers and designers with gearing design, application and failure analysis. We have added a new education manager staff position to develop and coordinate the needed education programs. We have in the works a new Learning Management System (LMS) for our education courses. The content will be indexed on the LMS in three categories: AGMA Online Workforce, AGMA Video Training, and AGMA Webinar Series. Find more on this education portal on our website (*www.agma.org*).

AGMA has also traveled abroad to conduct education training seminars. These courses are typically at the request of an international member, but it reflects how AGMA continues to step up to meet the needs of its members globally. Within our industry there is a vast knowledge base that can and should be tapped for educational purposes that benefit the industry and the industries served by the AGMA membership. Through these types of programs, we will increase the brand value of our association and be recognized as a technical resource to the industries served by our members.

There continues to be considerable news about energy efficiency. The United States is mandated to use premium efficient motors in December 2010. Canada will require motors to be premium efficient[1] as of January 2011, and Europe will be mandated to use energy efficient[2] (IE2) motors beginning June 2011. With so much information in the market focused on motor efficiency, AGMA recognizes the greater efficiency gains when the entire drive (gearing) system is optimized. In an effort to provide technical assistance and application direction, AGMA has formed a technical committee to develop guidance that will assist original equipment manufacturers and end users in evaluating the efficiency of their current power transmission system. This evaluation process will allow the engineer to optimize the gear drive system for maximum energy savings. The committee is working to have this tool available to the general market in the near future. This is a good example of how AGMA is meeting the needs of our industry for tomorrow as well as today.

The voice of the industry drives AGMA's agenda. I think this is one of the keys to AGMA's 94 years of service to the industry and one reason the association is respected the world over. The majority of AGMA programs are the result of direct action by members. It is not uncommon for a member to make a suggestion that results in a new or modified association program. Most of our education programs came directly from member suggestions.

Once identified and validated, the AGMA Board acts quickly to develop a solution and to charge the staff with implementing it.

AGMA continues to look ahead at the issues that continue to confront both our domestic and international members. We strive to anticipate members' needs and provide information that enables informed decision making. Visit our website, *www.agma.org*, for the latest on our members, industry news and global business trends. And, use your voice. If you know of an area where an industry solution is justified, please let me know.

[1] Premium efficient motors are currently the highest AC motor efficiency values for general-purpose AC motors. Reference NEMA MG1 table 12-12.

[2] Energy efficient motor values reference IEC 600034. These values are lower than premium efficient motor values.

Cylkro Face Gears:

DUTCH DESIGN AND SWISS INGENUITY CAUSE TRANSMISSION BREAKTHROUGH

When the Cylkro face gear transmission was first introduced by a Dutch company, it was met with years of skeptical disbelief and resistance from the traditional transmission establishment. Nevertheless, perseverance and a takeover in 2003 by Swiss-based ASSAG paved the way for growth and success in various industries and in many countries. This article describes the start and breakthrough of the Cylkro face gear transmission.

Origin. It was not until the early 1990s that face gears found some acceptance among the established transmission systems. Although face gears have been seen through history—for instance in the Chinese south-pointing chariot or in several Leonardo da Vinci designs—these examples more often ended up as museum pieces, not fit for industrial use. Then, some 20 years ago, the University of Eindhoven (Netherlands) began researching the possibilities of calculating and manufacturing face gears in such a way that these could be used in high-end, high-torque applications.

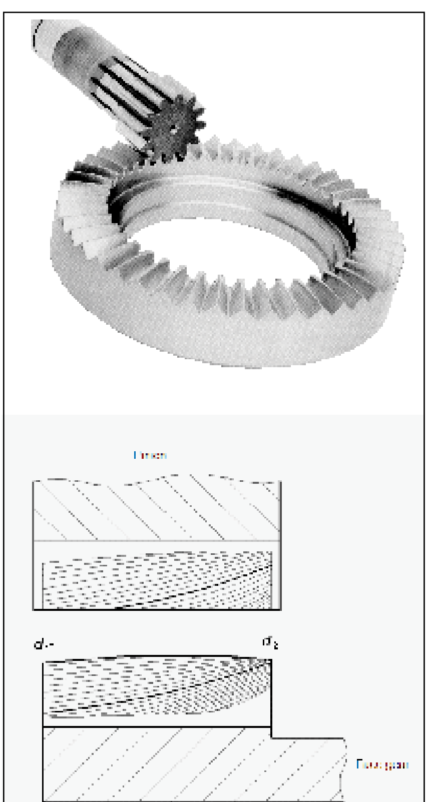

Figure 1—Face gear set (above) and helical contact lines on pinion and face gear.

Design. Derived from the Dutch words for cylindrical pinion and face gear, this new type of face gear transmission was named the Cylkro face gear. The first aim was to develop software to calculate the geometry and strength of the Cylkro transmission. A basic face gear set consists of one involute cylindrical pinion and one face gear, mostly at a 90° axis angle. It is the pinion's geometry, axial position and transmission ratio that determine Cylkro face gear geometry.

The shape of a Cylkro tooth or, more accurately, a tooth fillet, varies over its width. At the inner diameter, the fillet is relatively large as compared to the outer diameter. As a result, the point-of-contact of the Cylkro flank at the inner diameter is on a smaller radius of the pinion than at the outer diameter. Therefore, the lines of contact are inclined, even with a spur pinion. With a driving pinion, the meshing starts at the tip of the Cylkro tooth at the outer diameter (Fig. 1).

The pressure angle also varies over the tooth width, caused by higher velocity at the outer than at the inner diameter. The load capacity calculations for bending strength and pitting resistance are based upon the German standard DIN 3990 and ISO/DIS 6336, which apply to parallel gears. These include factors for geometry, meshing conditions, material properties, etc. The characteristics of the Cylkro transmission were translated into these factors with the help of FEM calculations. In order to avoid edge contact, the teeth of the pinion and/or Cylkro gear have to be crowned. Specific Cylkro software programs allow calculating the load distribution over meshing teeth and along the lines of contact, as well as tooth root bending stress and contact stress (pitting resistance) of a Cylkro face gear transmission.

Manufacturing. The Cylkro face gear production method was continuously improved and is described in a large number of patents. The processes include continuous hobbing, hard-cutting and several options for surface treatments (Fig. 2).

The geometry of the hob is based on the geometry of the pinion. Because one pinion can mesh with various face gears with different numbers of teeth and axis angles, it is possible to manufacture all these types of gears with one single hob.

Features. The wide range of gear ratios—from 1:1 up to 20:1, and more—is only one of the Cylkro face gear's specific characteristics. Other features are:

- Axial freedom of the pinion
- Free choice of axis angle from 0° to 135°
- The possibility of helical teeth or axis offset
- Multiple power transmissions; i.e.—two or more pinions mesh with one or between two face gears

Customer-specific applications. Almost all Cylkro face gear transmissions benefit from the advantage of axial freedom of the pinion at the mounting of the gear set (Fig. 3).

Compared to traditional angular transmissions such as bevel gear sets and worm gear sets, in which both gears have to be adjusted very precisely and even in pairs, the spur Cylkro face gear set only requires adjusting of the face gear. Thanks to the axial freedom of the cylindrical pinion, the axial position of the pinion does not affect the contact pattern.

Pinions can be exchanged easily and do not require meshing in pairs with the Cylkro face gear. This is of great benefit when there is expansion in the pinion axis due to heat generation. Or, the feature can also be utilized when the axial freedom becomes part of the application's function. For example, Saueressig embossing machines use the feature to slide one embossing cylinder closer to the second embossing cylinder. Another example is the starter gear in the Porsche Carrera GT, of which only an exclusive 1,200 cars were built and in which the pinion is axially pushed into the face gear at the moment of starting the engine.

It is possible to choose any axis angle between 0° and 135°, of which 90° is the most common. Smaller axis angles, such as 17°, are used in mixing equipment or driven tools with 45° angles from Sauter Feinmechanik GmbH (Fig. 4) and Benz-driven tools for the metal working industry.

Face gear sets with a helix angle are used, for instance, in automatic door systems. In this example (Fig. 5), the pinion only has three teeth and is shaped almost like a worm. However, the helical Cylkro transmission's efficiency remains very high as compared to the loss of efficiency in worm gear sets. Another advantage, specifically for the door system application, is the lack of self-braking factor. This means that in case of power failure and emergency, the doors can be opened easily by hand.

Gear ratios in the range of 1:1 to 1:5 are the typical choice for power applications. Larger gear ratios are more often used in hand-driven applications or in precision solutions such as printing machines or optical machinery from Zeiss. U.S.-based Danaher Motion has a full range of angle gear heads in which a total of 29 different Cylkro face gear sets are used. The gear head range is divided into five sizes, each size covering a gear ratio range of 1:1 to 1:5.

Finally, the multiple-power transmission—in which one or more pinions mesh with one wheel or between two Cylkro face gears—has been realized, for instance, in Hydrosta BV bow thrusters (Fig. 6) and Index Traub turning machines (Fig. 7).

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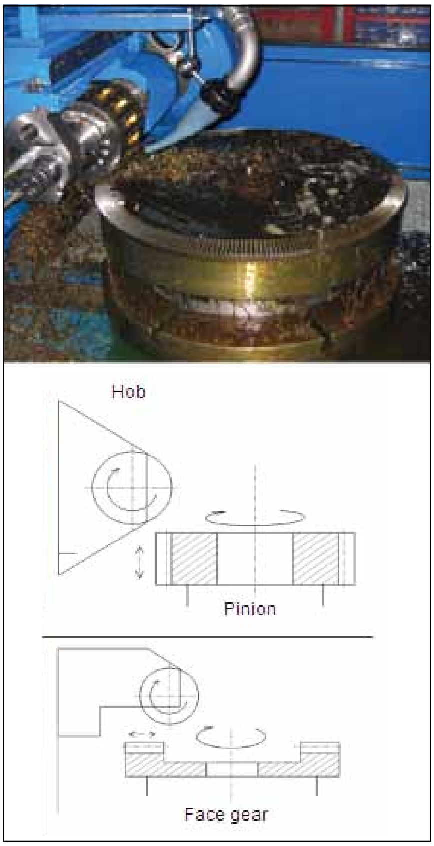

Figure 2—Manufacturing of Cylkro face gears by a six-axis CNC spur gear hobbing machine.

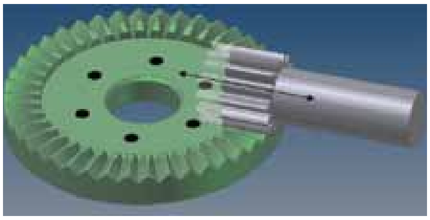

Figure 3—Axial freedom of the pinion in face gear transmissions.

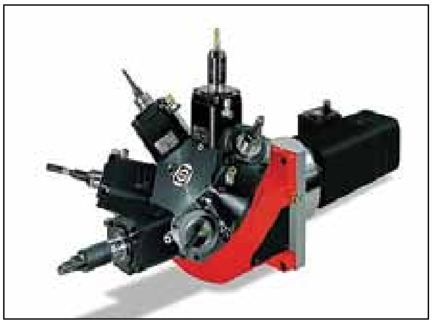

Figure 4—Compact tool exchanger based on face gear transmission.

Figure 5—Automatic door moving system with Cylkro transmission.

Figure 6—Cylkro face gears in counter-rotating bow thrusters achieve higher efficiency.

Figure 7—Multimodal Index Traub turning machine with inner and outer face gear rings.

Center Differential of the New Audi Quattro with Cylkro Face Gear Technology

An enormous breakthrough for the multiple-power Cylkro face gear transmission in the field of automotive applications was the introduction at the Geneva Autosalon earlier this year of the new Audi Quattro RS 5 with a self-locking crown gear differential in the Quattro drive train, which regulates the power distribution between the front and rear axles.

Two Cylkro face gears with different tooth geometries, resulting in a 40:60% torque split, are built into this lightweight differential (4.8 kg). ASSAG was given the responsibility of developing the tooth geometry of the face gears and pinions that are used in the heart of the Quattro drivetrain. Finally the successful cooperation resulted in a common patent application and ASSAG granted a license for serial production of the Cylkro face gears. Using the Cylkro face gear technology, Audi could realize a weight reduction of 2 kg compared to the conventional differential. Furthermore, the package of plates of the differential could be considerably reduced.

How it works. The Cylkro face gear with the largest number of teeth (Fig. 8, left side) is connected with the cardan shaft to the rear axle. The second face gear takes care of the power take-off to the front axle. In between the face gears, four planetary pinions are equally spaced at 90° in a planet carrier that is driven by the outgoing axis of the S-tronic 7-speed gearbox with double clutch.

The self-locking crown gear center differential attains a high efficiency ratio. This standard rear-biased configuration ensures sporty handling of the vehicle. In the basic situation, there is no difference in rotational speeds of the face gears and the planet carrier. If one of the axles starts to spin, for example, while it is on ice or snow, the self-locking face gear center differential will immediately engage. By a package of plates, the differential can widely vary the torque distribution between the front and rear axles. Up to 70% of the drive force can be fed to the front, and as much as 85% toward the tail-end (Fig. 9).

ASSAG could realize this wide variation by exactly locating and tolerating the contact patterns between the pinions and face gears. These contact patterns have been pre-defined by ASSAG within specified limitations. This leads to certain axial forces on the face gears and on the package of plates, finally resulting in a variation of the torque distribution in such a way that ASSAG could fulfill all Audi specifications.

In the crown gear differential, the gears are mounted without backlash. The result is a homogeneous conversion of the torque distribution without any delay. In conjunction with intelligent software in the braking system, the Quattro system assigns optimal torque to every driven wheel. Interventions of the ESP system will be reduced to a minimum. This increases the drivability of the Audi RS 5 in every situation. (After the release of the RS 5, Audi will equip future Quattro series with the face gear differential.)

Catalog products. The earlier mentioned Danaher's gear range was the instigator for ASSAG to look at its own standard range of catalog Cylkro face gear sets. This way, Cylkro face gear sets would also become available as a standard program allowing short delivery times and competitive prices. The program covers torques from 0.7 to 518 Nm at ratios up to 1:10. More information on the standard program is available in the Cylkro catalog or

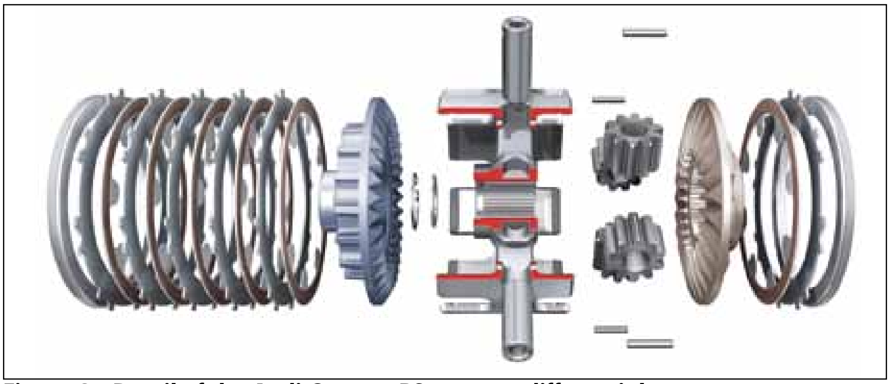

Figure 8—Detail of the Audi Quattro RS 5 center differential.

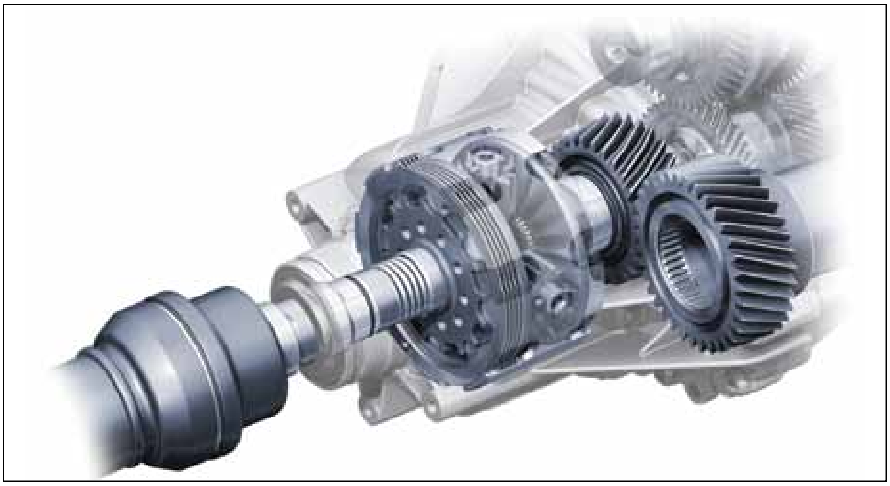

Figure 9—Embedded face gear center differential.

online on the ASSAG homepage.

Evolvere solutions. With the takeover by ASSAG Switzerland, new engineering knowledge and experience became available for the Cylkro technology. It found its way to the market not only as a face gear set, but, thanks to ASSAG's "Evolvere" concept, it is now also available as a complete angular gearbox. Evolvere is Latin for "to evolve" and so the Evolvere trademark stands for the optimal added value of Swiss transmission technology. It includes support in evaluating the best solution, considering cost-effective components and easy mounting and maintenance. ASSAG engineers construct transmissions of all types, for all kinds of industries and design animated 3-D models.

ASSAG provides three types of standard Evolvere gearboxes:

- Block-shaped gearboxes for 90° transmission ratios 1:1 to 1:4 (Fig. 10)
- Compact, flat gearbox for 90° transmissions with ratios 1:5 to 1:10 (Fig. 11)
- Octagonal gearboxes (Octodrive) for different angles and multiple inputs/outputs with ratios 1:3 and higher (Fig. 12).

All of them use the standard Cylkro face gear sets from the Cylkro catalog as described earlier.

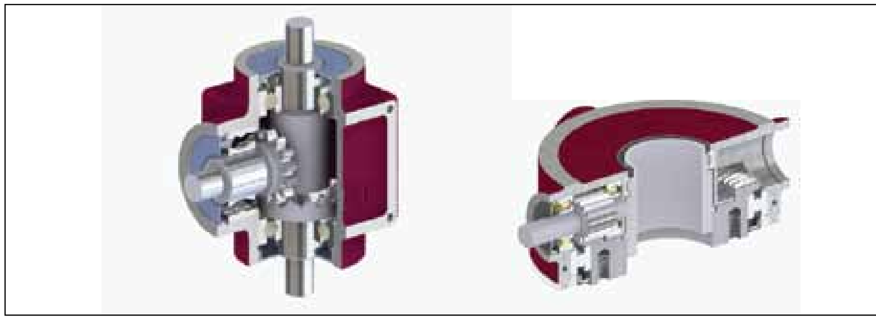

Figure 10—Block-shaped 1:1 and flat 1:5 versions of the Evolvere angular gearbox family.

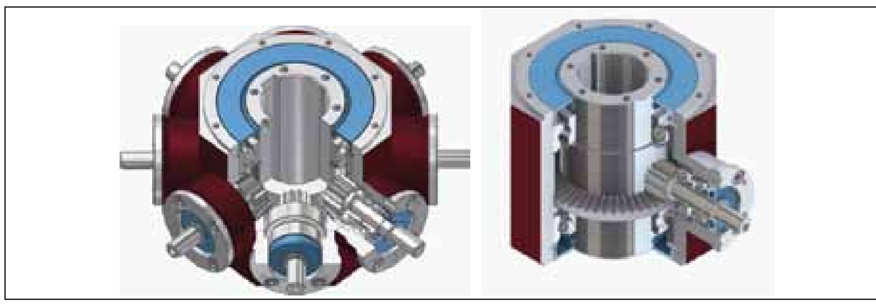

Figure 11—Octodrive gearbox (sectional view) in different configurations.

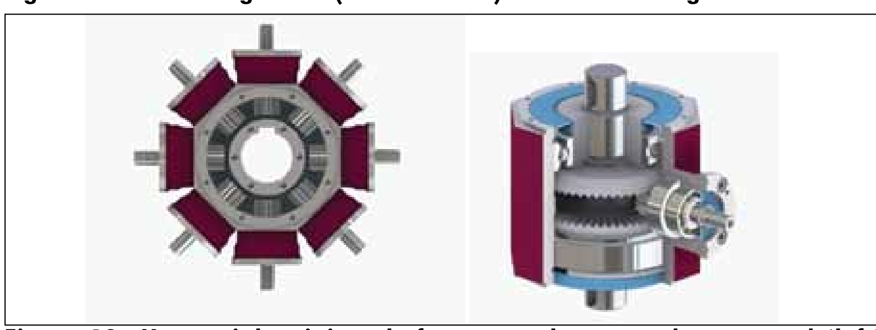

Figure 12—Up to eight pinion shafts on one layer can be mounted (left). Depending on the application, only one output may be needed. If required, it can be combined with a second face gear.

Octodrive Transmission Offers Customer-Driven Choices

ASSAG's angular gearbox program—Octodrive—affords customers the freedom of choosing the number of inputs, outputs, angles, ratios and other options. The customer has the possibility to design the gearbox according to his needs by choosing the relevant components in a dialog window. This allows for generation of multifunctional and high-quality angular gears quickly, with the resulting octagonal gearbox available from ASSAG partners or via the internet.

Figure 13—Application examples of Octodrive face gear transmissions: table-adjustment and multiple-lift drive combination.

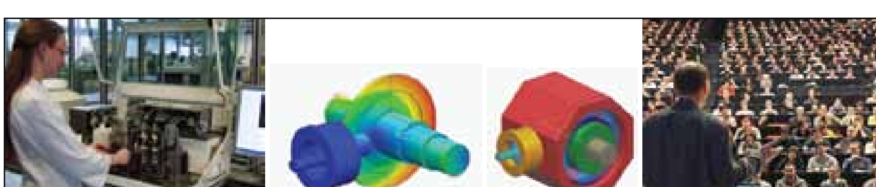

Figure 14—As a tool for researchers and educational purposes, Octodrive allows for the understanding, advancement and teaching of gear and transmission technologies under many aspects.

Octodrive face gear drives are

continued

delivered with output torques from 29 to 255 Nm and modules 0.7 to 3.5 with ratios 1:3 up to 1:10 (Table 1). This spectrum enables Octodrive to be applied in a large variety of applications. Hollow- or solid-shaft, as well as different options for motor adaptations, can be chosen (Fig. 12).

Clean technology. Friction-minimized angular ball bearings and optimized geometry and topology of the teeth out of hardened steel result in an efficiency factor of the gear transmission > 95%. High load capacity and long durability are realized despite a moderately light construction principle. It is grease-lubricated for life.

Based on self-oscillation analyses as well as optimization of tooth geometry and topology, the Octodrive transmission is designed for minimal noise generation with focus on the expected driving speed and load distributions.

Easy application. Octodrive is delivered along with a final testing certificate. Based on its octagonal form and self-centering of norm flanges, the gearbox fits practically anywhere and is implemented in a short time by the customer.

Large field of applications. Whether as a lifting unit, tool exchanger, in a robot, as part of a packaging line or as an angular gearbox of a robot, Octodrive fulfils the expected flexibility, bifurcation or inversion of the movement (Fig. 13). It enables the development of prototypes of complex machines in a timely fashion.

Synergies. The use of face gear sets based on the official Cylkro pro-

Table 1—Scale of the Octodrive program. M1 refers to the maximal constant torque at the pinion shaft.

Diameter (mm)	MI (Nm)	Ratio	Modul
95	18	3	1.25
95	12	4	1
95	9	5	0.9
95	5	6	0.7
115	30	3	1.5
115	22	4	1.25
115	14	5	1
115	10	6	0.9
115	5	8	0.7
140	50	3	1.75
140	39	4	1.5
140	27	5	1.25
140	15	6	1
140	10	8	0.9
140	5	10	0.7
160	64	4	1.75
160	47	5	1.5
160	29	6	1.25
160	16	8	1
160	10	10	0.9

gram allows the customer to order angular gearboxes with leading gearing technology and Swiss quality directly from the catalog at ASSAG's distribution partners or via Internet.

Summary and Forecast

During the past 20 years, the concept of a face gear transmission has developed into a well-defined, practice-proven and widely applied transmission, with the latest Cylkro success being the breakthrough in the automotive industry. Now available as a catalog product and as part of Evolvere and Octodrive gearboxes, the technology has become available to the standard gear market as well. ASSAG engineers continue to explore the possibilities of the Cylkro technology, both in the fields of application and in production techniques.

Cylkro and Evolvere are registered trademarks of ASSAG.

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www.assag.ch

Zeiss

OFFERS NEW MEASURING CAPABILITIES

With DuraMax, Carl Zeiss offers a compact 3-D coordinate measuring machine. DuraMax Gear transitions DuraMax into a shop floor gear measuring machine. "This enables us to fulfill the requests of many customers and introduce gear wheel measuring technology with small machines that can be used as close to production as possible," says Alexander Dollansky, product manager at Carl Zeiss Industrial Metrology.

The key features of DuraMax Gear are its suitability for a rough production environment and high permissible temperature fluctuations. DuraMax is suited for process control on the production floor, for quick in-between inspections of small workpieces and for testing volume parts directly in production. Because of its accuracy, DuraMax can also be utilized for many requirements in gear measuring tech-

continued

nology. DuraMax Gear comes with the required software and hardware, including stylus material for a broad range of applications. If the product being tested changes, standard inspection procedures often require new, expensive modifications. DuraMax Gear, however, when combined with CAD-based *Calypso* and *Gear Pro* involute measuring software, quickly, easily and reproducibly measures all changes. DuraMax Gear is available as a tabletop machine or with an optional base. Its design enables part loading from four sides. With *Calypso* measuring software and *Gear Pro* involute, which was specially developed to measure spur gears, it is now possible to complete all jobs in daily gear measurement for spur and helical gears, and splines in accordance with the applicable standards. With the *Calypso* Qs-Stat Out log output option included with delivery, customers are well-equipped to assess processes using a comprehensive, statistical evaluation of quality information relevant to production.

Carl Zeiss also recently introduced the new ACCURA CMM, a multi-sensor-capable measuring system that

permits fast, economical, precise and flexible measurements. As with any versatile modular system, customers can configure the ACCURA to fit their requirements. Based on their current tasks, they select the suitable configuration, i.e. sensors. Special software, such as *Gear Pro* for gears and *Holos Nt* for freeform surface measurements, is integrated along with *Calypso*, the standard CAD-based measuring software from Carl Zeiss. Subsequent modifications can be made very easily. If requirements change, different sensors and software can be easily added. Whether cut, shaped or molded parts, plastic or steel—all options of coordinate measuring technology are available. The ACCURA also permits the integration of Mass technology from Carl Zeiss. Combined with an RDS articulating probe holder, Mass permits the fast measuring program-guided change between contact sensors and the ViScan and LineScan optical sensors during a CNC run. The contact measuring sensors of the Vast family and the DT single-point sensor can also be used in various configurations.

For more information:
Carl Zeiss Industrial Metrology
6250 Sycamore Lane North
Maple Grove, MN 55369
Phone: (800) 327-9735
www.zeiss.com

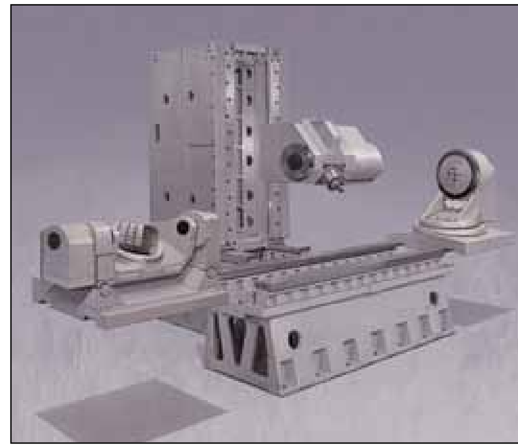
Mitsui Seiki

INTRODUCES MACHINING CENTERS TO NORTH AMERICA

Mitsui Seiki recently introduced its line of large-capacity, heavy-duty and configurable machining centers to North America. The new HU100 series is the first range of configurable machining centers aimed at a variety of manufacturing industries interested in versatile equipment for heavier applications. The concept of configurable machines has been a part of Mitsui Seiki philosophy for many years, but only recently has the concept been standardized for the marketplace. Configurable machines are those that are based on a set of standard modular components that can be arranged to suit specific customers' needs easily and affordably.

continued

Mitsui Seiki's customers can choose axis travels from 1,300 mm to 2,500 mm, four, five, or six axes of continuous motion with a choice of a rotary table, trunnion table, or table on table. There's also a variety of spindle choices, including fixed spindles from 6,000 rpm (50 hp, 2,000 ft./lbs. torque) for heavy milling up to 25,000 rpm for high-speed applications. Tilt or swivel spindle choices are available from 6,000 rpm (800 ft./lbs. torque) for heavy milling up to 12,000 rpm for high-speed machining. Mitsui Seiki also offers quill type spindles for precision boring with shorter tools. Mitsui Seiki applications engineers will assist customers in selecting the optimum component choices to meet their needs and objectives.



The HU100 series is suitable for larger aerospace and power generation parts with a work zone capacity of up to 2,500 mm dia. x 2,000 mm height. The machines accommodate weights from 4,400 lbs. (2,000 kg) to 17,500 lbs. (8,000 kg). These machines can be equipped with simple pallet changers to fully integrated FMS systems for work and raw material handling. Likewise, tool handling systems range from on-board magazines to central systems for more than 2,500 tools.

Launched earlier this year, the HU100 line has been sold to industries as diverse as aerospace, refrigeration compressor, mold & die, heavy equipment and energy. All of these industries have fundamental common requirements: machine rigidity/stiffness and high accuracy.

For more information:
Mitsui Seiki (U.S.A.), Inc.
563 Commerce Street
Franklin Lakes, NJ 07417
Phone: (201) 337-1300
Fax: (201) 337-3680
www.mitsuiseiki.com

Arrow Gear and Motoman Robotics

Offer Chamfering/ Deburring Solutions

PerfectEdge, a complete, full function robotic solution, meets the challenges of tight tolerance chamfering, deburring or finishing processes, while offering a non-touch teach method of programming. Designed in collaboration with Arrow Gear, it includes a six-axis Motoman DX1350D industrial robot and a precision indexer with hollow chuck controlled by a Yaskawa America Sigma-5 servo motor for fixturing shafted gears. The cell has an eight-slot protected tool holder, and an Alfred Jaeger spindle that provides the muscle, speed control and holding torque to perform the most demanding tasks.

PerfectEdge has the capacity to process 20 inch diameter by 10 inch cylindrical parts. Designed on a common base with fork pockets, PerfectEdge can easily be deployed or relocated. Programming is easy with menu-driven material removal software. Motoman Robotics' powerful *G-Code/Points Importer* off-line programming software drives the robot from your part data. *G-Code* converter translates the CAM file into programs that are downloaded to the robot controller. PerfectEdge then provides consistent, predictable results, part after part. The

continued

PerfectEdge cell features a polycarbonate and aluminum full-safety enclosure with electrically interlocked access doors. A particle and fume extraction system is included, making PerfectEdge easy to connect to an air handling system. Repetitive stress injuries and operator fatigue are eliminated, reducing scrap rates and improving safety. It is compliant with ANSI/RIA R15.06-1999 and other relevant ISO and CSA safety standards. A CE compliant version is available upon request.

For more information:
Motoman Robotics
805 Liberty Lane West
Carrollton, OH, 45449
Phone: (937) 847-6200
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www.motoman.com

Mazak's 430A

OFFERS OPERATOR EFFICIENCY

Mazak's new Vertical Center Smart (VCS) 430A offers value across a variety of applications, ranging from heavy-duty machining to high-speed machining. Able to utilize both EIA/ISO programs and conversational programming, the machine integrates Mazak SMART CNC to maximize ease-of-use and operator efficiency.

The VCS 430A features a 25 hp, 12,000 rpm No. 40 taper spindle that provides a maximum torque of 70.2 ft/

lbs. A tool-to-tool change time of just 1.3 seconds helps optimize productivity while the 35.43" x 16.93" table provides the largest machining area available for this machine class.

A variety of intelligent machine functions further simplify operation and boost reliability and productivity. Active Vibration Control (AVC) increases accuracy and tool life by minimizing machine vibration. Intelligent Thermal Shield (ITS) further ensures accuracy by actively managing heat displacement. Intelligent Maintenance Support (IMS) aids with preventive maintenance by tracking and reporting the status of the machine's perishable items, such as filters and cover wipers.

Ergonomics received special attention throughout the design of the VCS 430A. The machine provides convenient loading and unloading of parts by offering a large front-door opening of 36.22" and a tool clamp/unclamp switch located next to the spindle. Additionally, the operator door includes a top cover opening to facilitate use of a crane for materials handling.

The new Mazak MX Hybrid Roller Guide System is integrated into the VCS 430A to deliver levels of durability and reliability that result in long-term accuracy. The Mazak MX Hybrid Roller Guide System increases vibration damping to extend tool life, handles higher load capacities, accelerates and decelerates quicker to shorten cycle times, consumes less oil for "greener" operations and lasts longer with less required maintenance. The SMART CNC control keeps operation easy by offering conversational programming and simplified set-up, allowing for the fastest possible completion of a first part. The SMART CNC control also tracks and provides detailed tool information and offers the ability to perform time study analysis of operations.

For more information:
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8025 Production Drive
Florence, KY 41042
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www.mazakusa.com

ONLINE

Visit
www.geartechnology.com
for the latest Product News

Sandvik

RELEASES FREE APP FOR CUTTING DATA

Sandvik Coromant has released a free app designed to provide engineers and machinists with a convenient resource for calculating cutting data. Once downloaded and installed, the app helps users optimize performance of their turning, milling and drilling applications by calculating optimal settings based on a job's unique parameters. "We're always looking for exciting new ways to meet the ever-changing needs of our customers," says Lennart Lindgren, global vice president marketing and sales. "We developed this app to provide customers with a convenient resource that can be accessed anywhere they take their phone."

The Machining Calculator app features a help button that provides additional information on the calculation being executed and the input needed to generate results. Sandvik Coromant's app also contains a process cost comparison that determines how tool optimization can provide cost and time savings. The app works with both metric and inch measurements and is available for both iPhone and Android phones.

For more information:
Sandvik Coromant
1702 Nevins Rd.
Fair Lawn, NJ 07410
Phone: (201) 794-5000
www.coromant.sandvik.com

LMC

RELEASES NEW FACE DRIVERS

LMC Workholding introduces new Neidlein FFBHZ face drivers for gear grinding, gear hobbing and gear milling, suited for high run out result operations with zero backlash. The Neidlein FFBHZ face drivers design offers less downtime when changing drive diameter ranges within one face driver size. Hydraulic compensating drive pins guarantee shear force-free clamping of workpieces with badly machined surfaces at clamping areas. The built-in hydraulic cartridge is easy to change and maintain. Neidlein face drivers quickly grip and turn a workpiece from its end, allowing the entire part to be machined in just one operation, including cutting off the ends and eliminating the need to flip the part. Face drivers allow turning applications to have increased flexibility to lower cycle times and allow large interrupted and heavy cuts. Neidlein face drivers offer fast set-up, quick changeovers, improved part quality, reduced cost and lower maintenance.

For more information:
LMC Workholding
1200 West Linden Avenue
Logansport, IN 46947
Phone: (574) 735-0225
www.lmcworkholding.com

ADCOLE

RELEASES LED MICROMETER

A low-cost LED micrometer for their Model 1200 Crankshaft Gage that validates parts that have been rolled to the proper depth and stress-relieved is now available from Adcole Corporation of Marlborough, Massachusetts. The sensor provides ± 3 µm accuracy, ± 1 µm repeatability, and 0.25 µm resolution to validate that parts have been rolled to the proper depth. Capable of measuring fillet undercut depth (before and after rolling) at a 35 degree roll angle, journal straightness, and radial distance or journal shoulder to journal, this gage creates chart reports depicting any problem areas. Providing 3x higher accuracy and resolution than a laser micrometer, the Adcole LED Micrometer Sensor is attached to the follower carriage, and the Z-axis location is known by the standard gage for start/stop measurement. Supplied standard on the Model 1200 Crankshaft gage, it is also offered for retrofit. The Model 1200 features a menu-driven sequence builder utility for developing measurement sequences for new crankshafts and camshafts.

For more information:
Adcole Corporation
669 Forest Street
Marlborough, MA 10752
Phone: (508) 485-9100
www.adcole.com

CUT 1000

OFFERS ALTERNATIVE CUTTING PROCESS

GF AgieCharmilles' CUT 1000 OilTech is designed specifically for micromachining and ultra precision applications. The CUT 1000 OilTech

uses wires as small as .02 mm diameter to achieve surface finishes down to Ra 0.05 µm in continuous operation with suitable corner quality. CUT 1000 OilTech uses oil as an inert dielectric. The oil eliminates the effects of corrosion due to long periods of immersion in the workpiece. As a result, multiple workpieces can remain immersed in the dielectric for overnight or weekend machining needs. The oil also enables smaller distances between the wire and workpiece to produce smaller internal radii when compared to water-based machining. Equipped with a high-performance generator and the AC Duo wire system, the machine is able to work with two wire spools, each with a different wire type, and switch between wires of different diameters automatically. Programmable via the machine control, the forward feed of the wires, just like the threading, takes place automatically and is monitored by patented sensors. GF AgieCharmilles constructed the CUT 1000 OilTech with a table design that separates the X and Y-axes on a patented monobloc, eliminating mutual interference and tripping errors. A traveling table on the machine bed carries the work tank and moves the workpiece in the X direction exclusively. A second axis slide, also movable in the horizontal plane on the machine frame, accommodates the upper part of the C-frame and moves the wire in the Y direction. This configuration delivers ideal travels, positioning accu-

continued

racy and taper cutting capability for machining complex, high-precision parts. Cut 1000 OilTech accommodates workpieces up to 77 lbs. and has X-, Y- and Z-axis travels of 8.66" x 6.29" x 3.93"Additionally, CUT 1000 OilTech is equipped with Vision 5, a control system developed specifically for wire cut EDM that enables flexible data input in accordance with a workshop environment.

For more information:

GF AgieCharmilles
560 Bond St.
Lincolnshire, IL 60069
Phone: (800) 282-1336
www.gfac.com/us

Mahr

INTRODUCES NEW GENERATION OF DIGITAL CALIPERS

Mahr Federal has introduced a new generation of its popular line of MarCal digital calipers. Included are a number of innovations, such as lapped guideways, a new reference system that retains the zero position setting and an increased number of product options and accessories. MarCal digital calipers are available with protection against dust and immersion to class IP67, provide increased battery life, offer a range of data output options and are available in a wide range of sizes and blade and anvil configurations. The design recently won a German award for excellence in innovation and quality.

The new reference system available on MarCal R-designated digital calipers is designed to be a significant time-saver for operators. Unlike older models, which require the zero position to be

reset whenever the caliper is switched on, the MarCal reference system retains the zero setting, so that the unit is ready to measure whenever it is turned on or the jaws are moved. A convenient reference lock protects the setting from operator error, and the new reference system is also much more energy efficient, providing up to 50 percent longer battery life.

Mechanically, the new MarCal digital calipers are the only line to provide lapped guideway surfaces. Compared to ground guideways, this improvement not only smoothes slide operation and sensitivity, but significantly increases the service life of the instruments. Ergonomic improvements, including a thumb support and large LCD with 8.5 mm digits, facilitate operation. W-designated MarCal units include protection against dust, water, coolant, and lubricants to IP67, making them suitable for service in even the most difficult shop conditions.

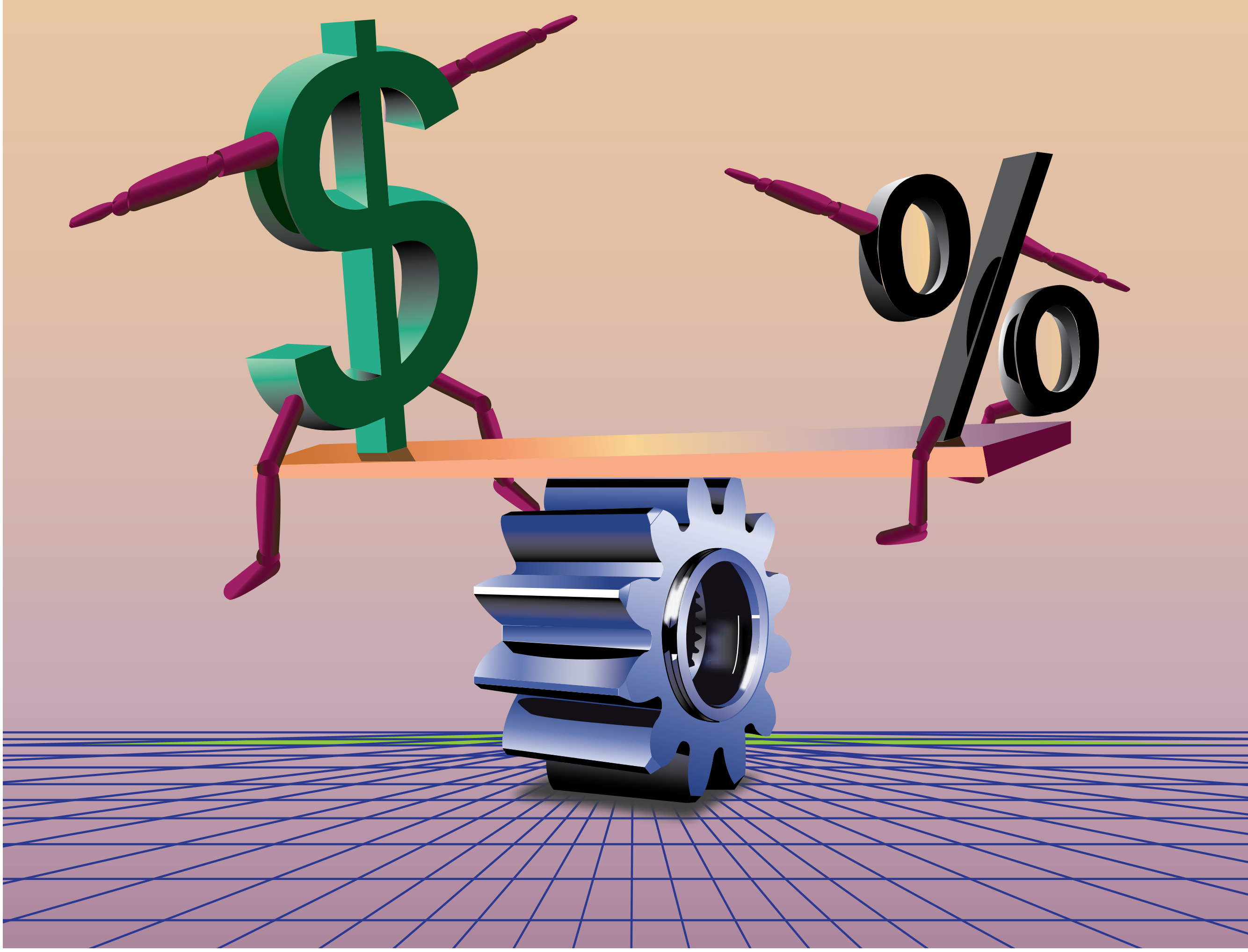
For more information:

Mahr Federal Inc.
1144 Eddy Street
Providence, RI 02905
Phone: (800) 343-2050
www.mahrfederal.com

State of the Gear Industry 2010

RESULTS OF RESEARCH

on trends in Employment, Outsourcing, Machine Tool Investment and Other Gear Industry Business Practices



In October, Gear Technology *conducted an anonymous survey of gear manufacturers. Invitations were sent by e-mail to thousands of individuals around the world. More than 300 individuals responded to the online survey, answering questions about their manufacturing operations and current challenges facing their businesses.*

The respondents considered here all work at locations where gears, splines, sprockets, worms and similar products are manufactured. They work for gear manufacturing job shops (39 percent), captive shops at OEMs (60 percent) and shops manufacturing gears for maintenance, spares and their own use (1 percent).

The survey covers gear manufacturing around the world, with 50 percent of respondents working in the United States, and 50 percent outside the United States.

A full breakdown of respondents can be found at the end of this article.

continued

What Factors Are Presenting Significant Challenges to Your Business?

"Capacity and capital."
—VP of sales at a U.S. manufacturer of aerospace gears

"Capital constraints."
—Corporate executive at an Indian manufacturer of automotive gears and shafts

"Changing political policy and related changes."
—Corporate executive at a U.S. manufacturer of buses

"Prices in China and India."
—Sales professional at a U.S. manufacturer of powder metal gears

"Corruption in the purchasing system of the mining industries."
—Manufacturing engineer at a gear job shop in Africa

"Customers shipping jobs and work overseas."
—Corporate executive at a U.S. gear manufacturing job shop

"Difficulty in obtaining sufficient quantities of steel for gear manufacturing."
—Engineer at a U.S. manufacturer of construction & off-road equipment

"Distance from our main customers."
—Production manager at an Indian manufacturer of transmission gears

"Documentation."
—Manufacturing engineer at a U.S. aerospace & defense OEM

"Because exports are so significant, sustained export markets are key to our sustenance."
—Corporate executive at an Indian manufacturer of automobile transmissions

"Fast delivery."
—Owner of a European manufacturer of pumps

"Federal tax policy, excessive corporate governance, lack of tort reform and unrealistic environmental compliance requirements."
—Corporate executive at a U.S. manufacturer of high-speed gear drives

"Finance."
—Design engineer at a European manufacturer of automobile transmissions

"Finding enough added value."
—Corporate executive at a European manufacturer of automotive actuators

"Frequently changing demand requirements."
—Manufacturing engineer at an Indian automobile OEM

"Fuel economy."
—Corporate executive at a U.S. manufacturer of truck axles

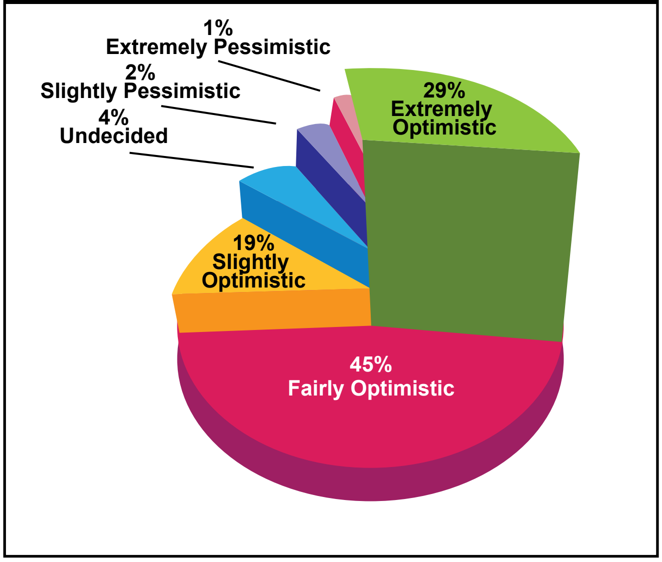
"General business confidence. As a pure jobbing shop, we depend on our customer base having the demand confidence to place orders."
—Corporate executive at a European gear manufacturing job shop

"Getting business."
—Corporate executive at a U.S. gear manufacturing job shop

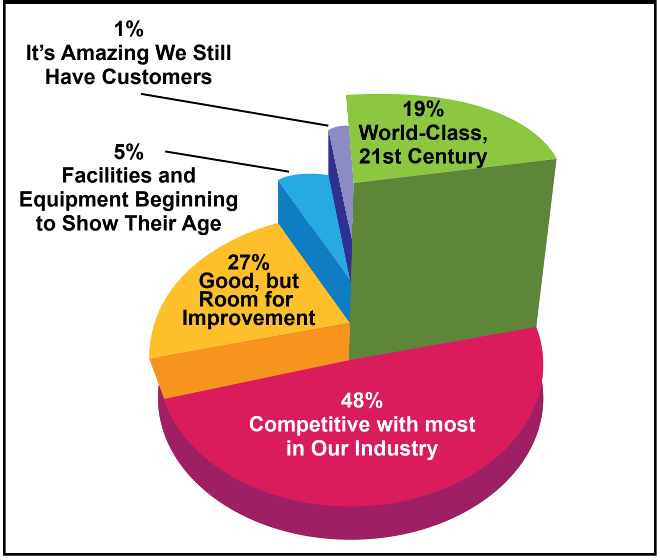
"Getting reverse engineering done and getting gears manufactured at a reasonable cost."
—Owner of a U.S. manufacturer of racing transmissions

"Increasing demand."
—Corporate executive at an Indian gear manufacturing job shop

93% of Gear Industry Respondents are Optimistic About their Ability to Compete over the Next Five Years



How Do Respondents Describe their Manufacturing Operations and Technology?



What Factors Are Presenting Significant Challenges to Your Business?

"International presence, Asia customs duties, availability of a new facility."
—Corporate executive at a European gear manufacturing job shop

"Lack of demand."
—Manufacturing engineer at a European gear manufacturing job shop

"Lack of skilled labor is an enormous problem!"
—Manufacturing engineer at a U.S. manufacturer of aerospace gearboxes

"Lead time for low-cost countries."
—Design engineer for a U.S. manufacturer of agricultural components

"Legacy information technology and business software stifles innovation."
—Manufacturing engineer at a U.S. manufacturer of enclosed gear drives

"Local and state regulations."
—Corporate executive at a U.S. aerospace gear manufacturer

"Machine availability."
—Corporate executive at a U.S. aerospace gear manufacturer

"Management decisions."
—Design engineering manager at a U.S. manufacturer of gears and drives

"Material cost."
—Corporate executive at a Far East manufacturer of automotive components

"The development of new technologies."
—Production worker at a European gearbox manufacturer

"OEM consolidation."
—Sales professional at a U.S. manufacturer of couplings

"Offshore quality issues."
—Corporate executive at a U.S. manufacturer of driveshafts

"President Obama."
—Design engineer at a U.S. manufacturer of satellites

"Rapid price changes of materials."
—Manufacturing engineer at a U.S. manufacturer of plastic film

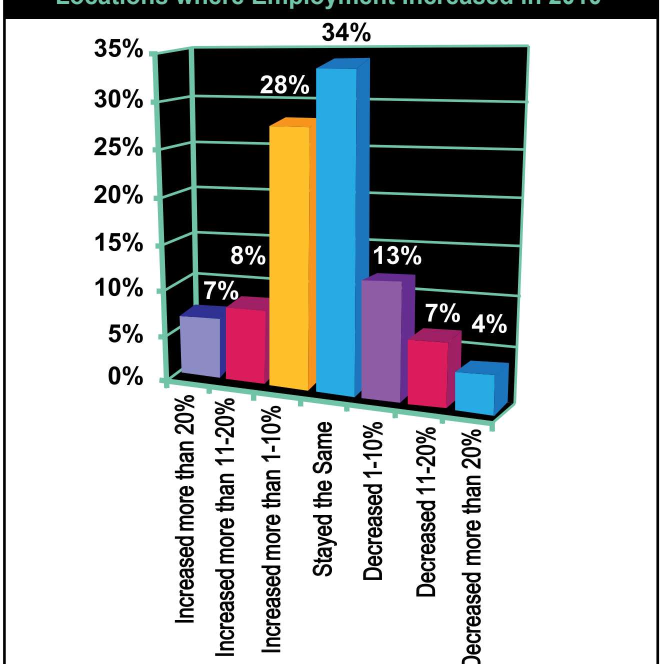
"Retention of skilled people as other industries' business improves and people are in demand."
—Manufacturing engineer at a U.S. manufacturer of truck axles

"Rising costs of all employee benefits in addition to health care, i.e. 401K plans, auto, etc. Also, corporate taxation rates discourage savings for offsetting future business climate weakness."
—Corporate executive at a U.S. manufacturer of enclosed gear drives

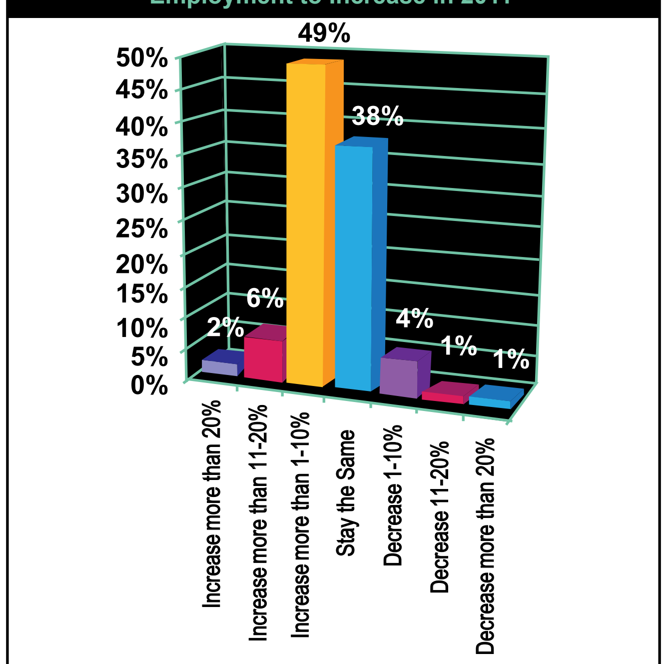
"Running out of factory space, where we are now in the process of looking for larger premises or additional space."
—Design engineer at a European manufacturer of gearboxes

continued

43% of Gear Industry Respondents Work at Locations where Employment Increased in 2010



Most Gear Industry Respondents Expect Employment to Increase in 2011



What Factors Are Presenting Significant Challenges to Your Business?

"Seasonal changes in order placement."
—Design engineer at a U.S. gearbox repair facility

"Shortage of steel and anti-friction bearings."
—Design engineer at a U.S. manufacturer of off-highway transmissions

"Steel prices."
—Technical manager at a European manufacturer of agricultural drives

"Still a buyers market. We need oil at $90 a barrel."
—Purchasing manager at a U.S. manufacturer of centrifugal pumps

"The Obama uncertainty factor makes our customers' industries hesitate, making it tough for us to gain projects that are right now non-existent."
—Sales professional at a U.S. engineering consultancy

"To sustain and improve the quality levels of our product."
—Design engineer at an Indian manufacturer of two- and three-wheel vehicles

"Tough competition, cost reduction, worker demand."
—Production worker at an Indian gear manufacturing job shop

"Understanding the ramp-up of our customers and their future inventory demands."
—Sales professional at a U.S. gear manufacturing job shop

What Are Your Company's Greatest Manufacturing/Engineering Challenges for 2011?

"1. Inconsistency in government policies. 2. Law and order situation 3. Fluctuation in currency exchange rate. 4. Rising energy costs. 5. Volatile political situation."
—Production manager at a Middle East manufacturer of motorcycle gears

"Ability to continue quality improvements."
—Corporate executive at a U.S. gear manufacturing job shop

"Ability to cut production costs on a continuous basis to retain the competitive edge."
—Corporate executive at an Indian manufacturer of agricultural transmissions

"Being competetive in Asia."
—Quality manager at a U.S. manufacturer of transmissions and axles

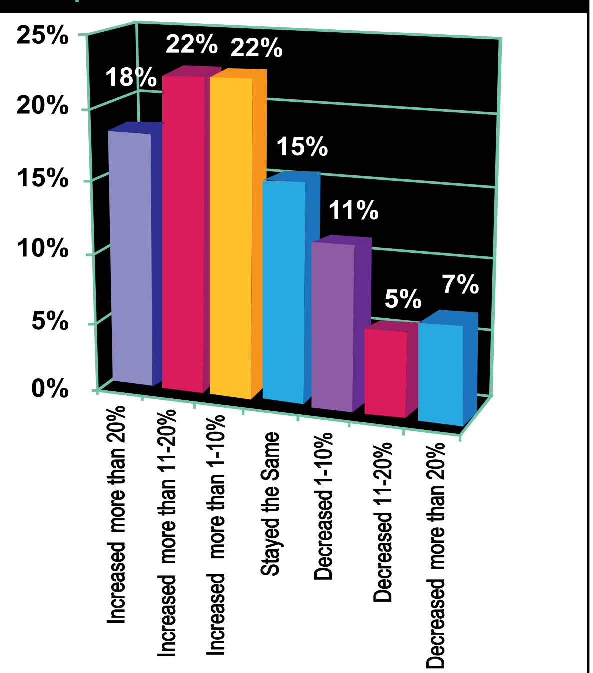
"Can't find skilled aerospace gear engineers. We have to train them, and that takes a lot of time."
—Corporate executive at a U.S. manufacturer of aerospace gears and gearboxes

"Capability of manufacturing at low cost."
—Manufacturing engineer at an Indian automobile OEM

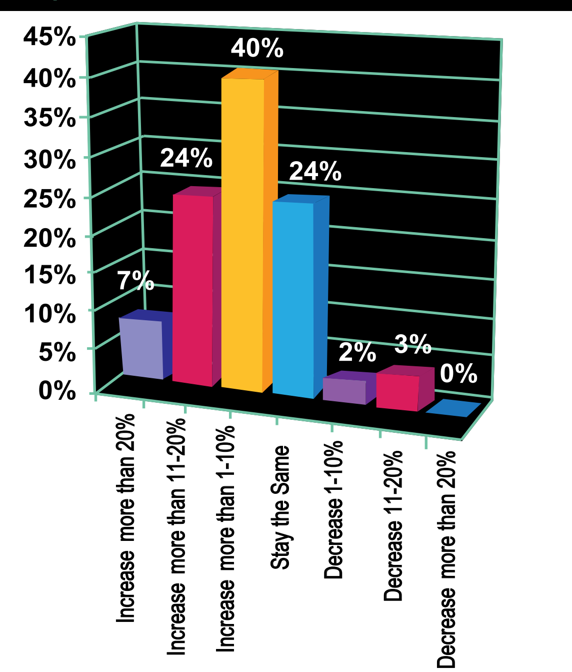
"Competing with Chinese product."
—R&D manager at a European manufacturer of wind turbines

"Climate Change and the following switch in technologies."
—Sales manager at a European manufacturer of turbo gearbox units

62% of Respondents Saw Production Volumes Increase



71% Expect Production Volume to Increase in 2011



What Are Your Company's Greatest Manufacturing/Engineering Challenges for 2011?

"Conclude site expansion toward 2011–2014 demands."
—Production manager at a European gearbox manufacturer

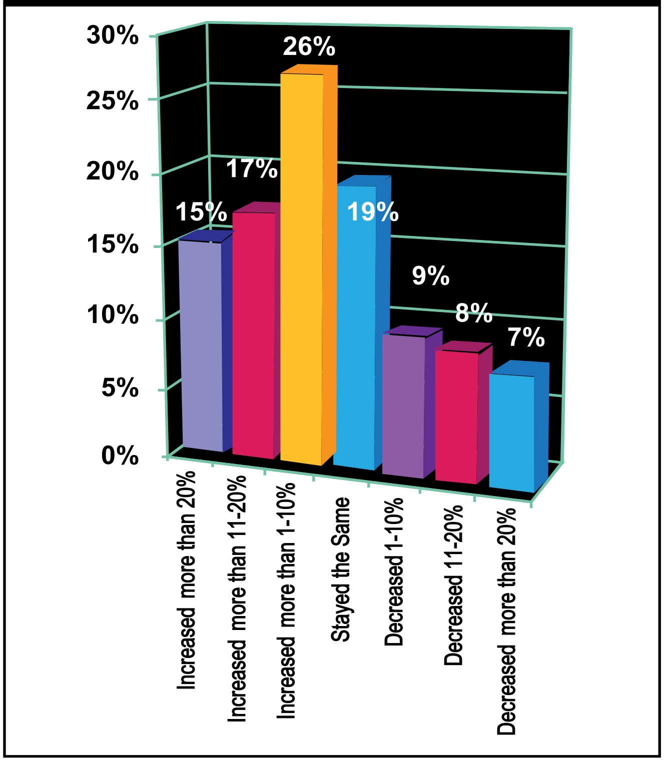
"Consolidations to control cost."
—Manufacturing engineer at a U.S. manufacturer of aerospace and defense components

"Developing new products."
—Design engineer at a U.S. manufacturer of bearings

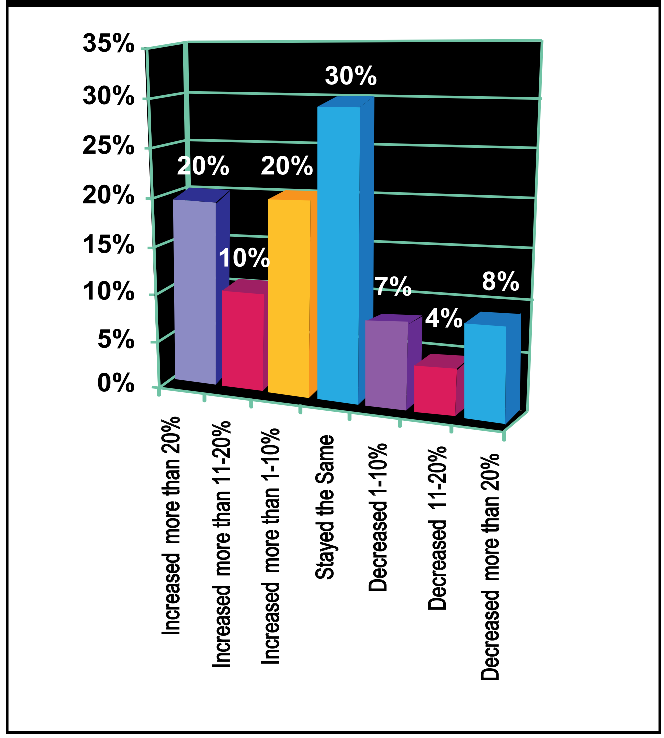
"Do more with less."
—Design engineer at a U.S. manufacturer of off-highway transmissions

continued

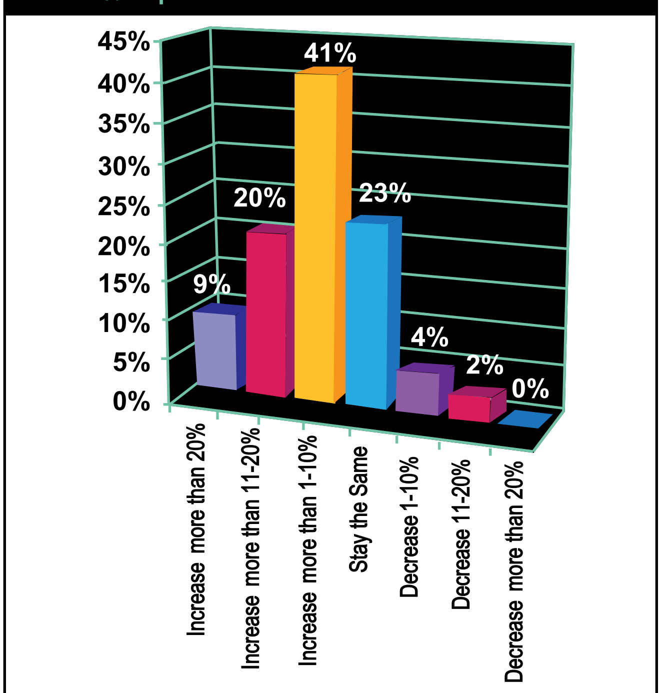
58% Saw Sales Volume Increase in 2010



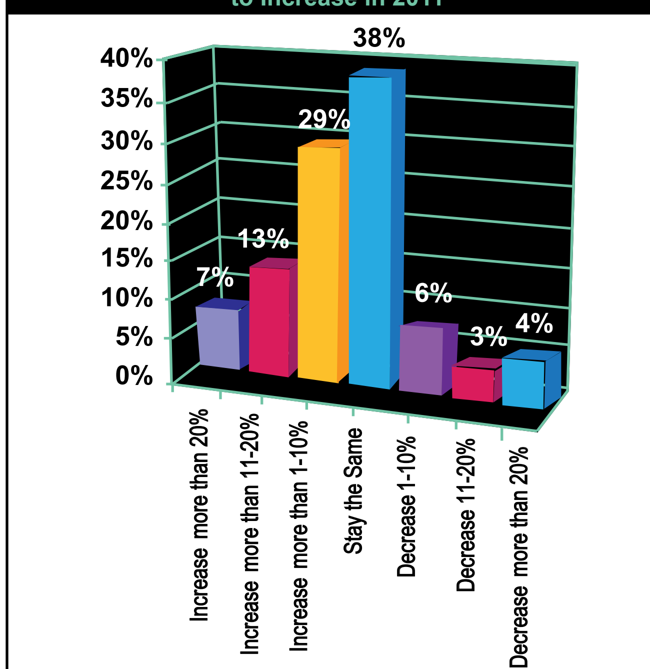
50% Work at Locations where Capital Spending Increased in 2010



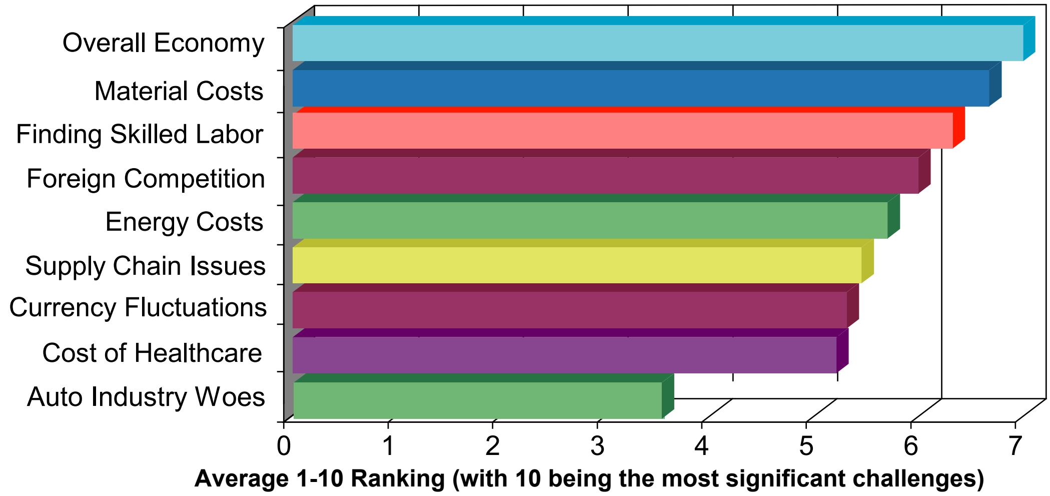
70% Expect Sales Volume to Increase in 2011



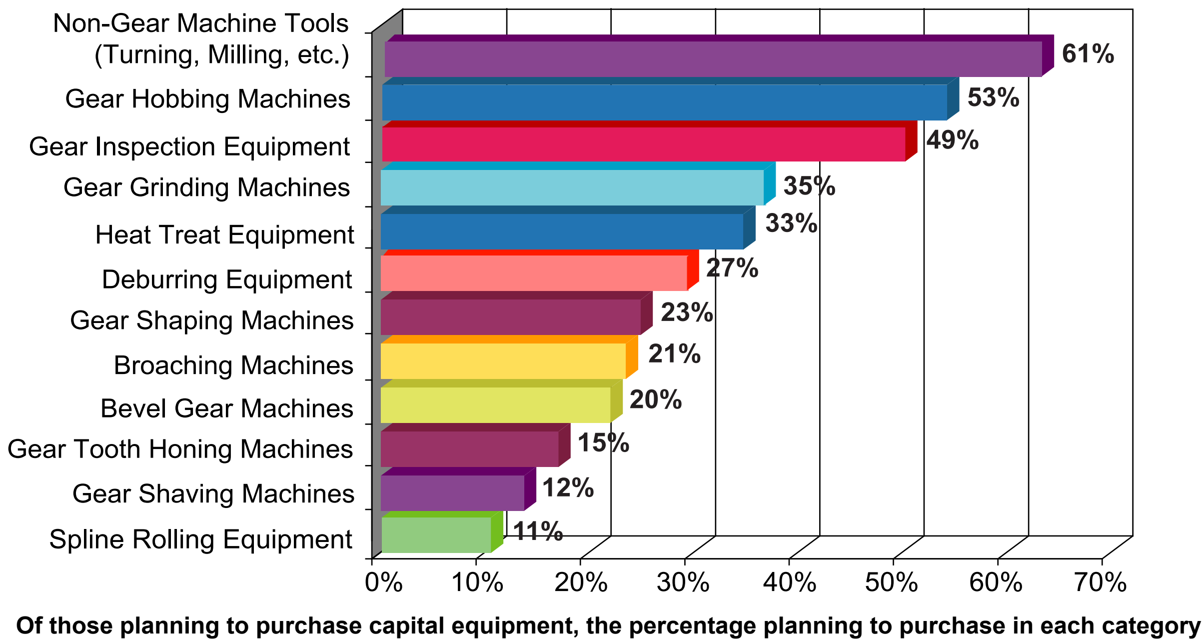
49% Expect Capital Spending at their Locations to Increase in 2011



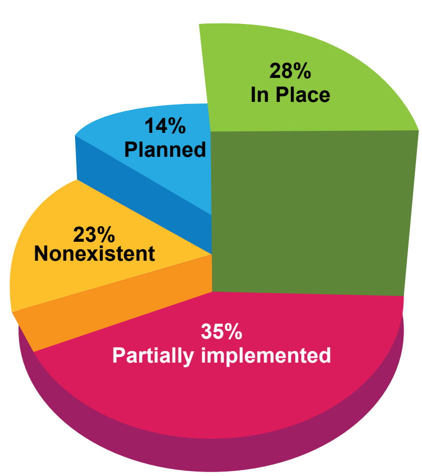
What are the Most Significant Challenges Facing Gear Industry Companies?



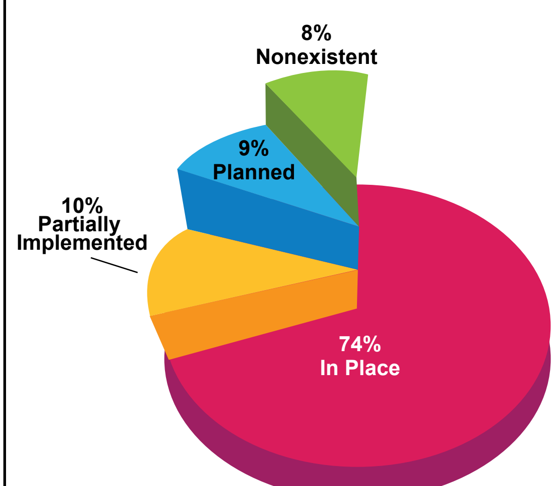
Machine Tool Purchase Plans 2011



Green or Sustainable Manufacturing Implementation



ISO 9000 Implementation



What Are Your Company's Greatest Manufacturing/Engineering Challenges for 2011?

"Enhancing capacity while reducing manpower."
—Corporate executive at an Indian gear manufacturing job shop

"Finding R&D support for solving gear distress associated with new technologies."
—Corporate executive at a U.S. manufacturer of enclosed gear drives

"Finding a solution to controlling price and reducing our lead times and order quantities."
—Design engineer at a U.S. manufacturer of agricultural components

"Finding qualified engineers."
—Marketing manager at a U.S. manufacturer of gears and gear drives

"Finding skilled labor."
—Engineering manager at a European manufacturer of agricultural transmissions

"Getting more involved in automotive industry."
—Purchasing manager at a European manufacturer of gearboxes

"Getting business."
—Corporate executive at a U.S. gear manufacturing job shop

"Getting new product started and funded."
—Corporate executive at a U.S. manufacturer of buses

"Government funding."
—Design engineer at a U.S. manufacturer of satellites

"Implementing a zero-defect mindset."
—Manufacturing engineer at a South American manufacturer of automotive transmission parts

"Implementation of newer technology."
—Sales manager at a U.S. gear manufacturing job shop

"Improving quality consistency."
—Corporate executive at a European manufacturer of speed reducers

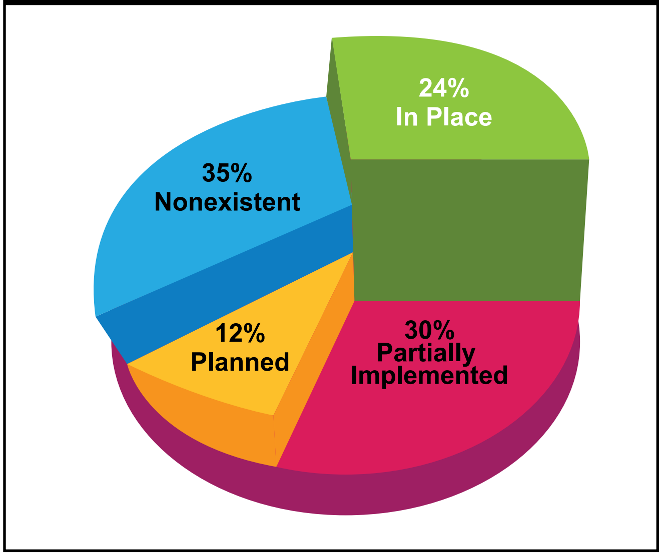
"Improved materials for lower product cost. Advanced analytic modeling."
—Corporate executive at a U.S. manufacturer of high speed gear drives

"Improving quality levels and going green."
—Corporate executive at a Far East manufacturer of sprockets

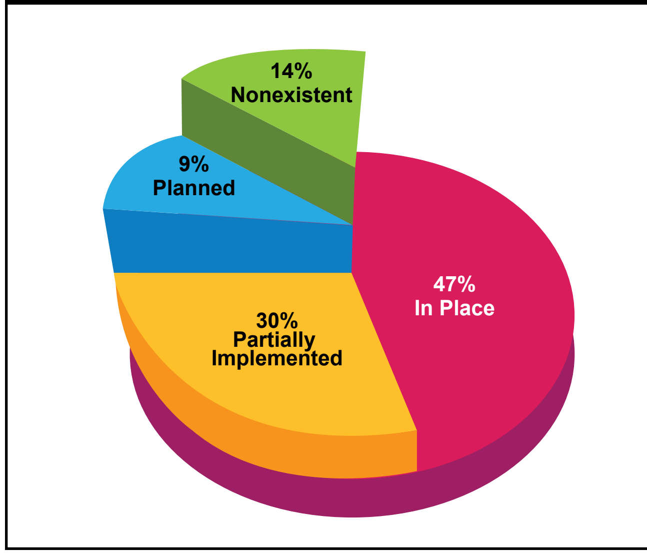
"Increasing car axle manufacturing business."
—*Corporate executive at a U.S. manufacturer of truck axles*

continued

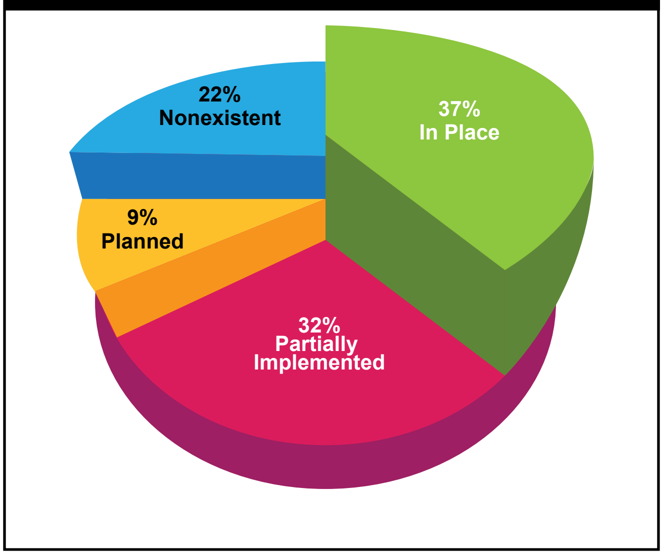
Six Sigma Implementation



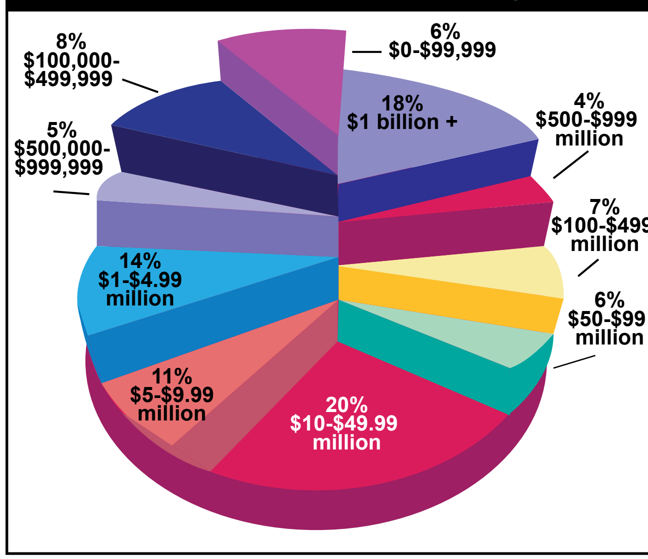
Lean Manufacturing Implementation



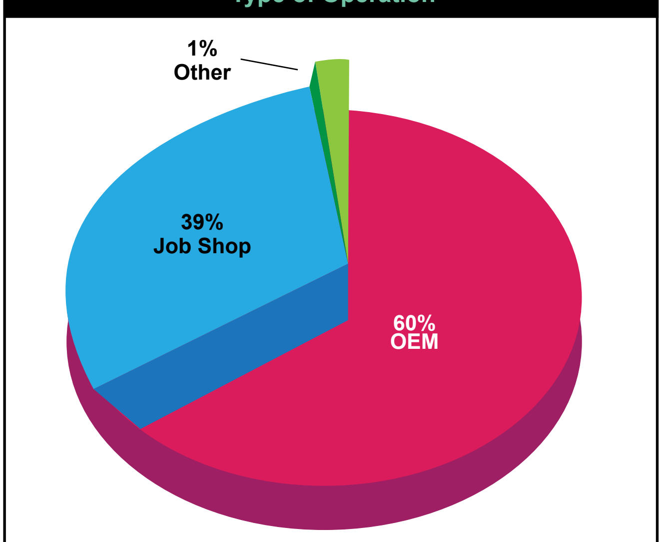
Statistical Process Control (SPC) Implementation



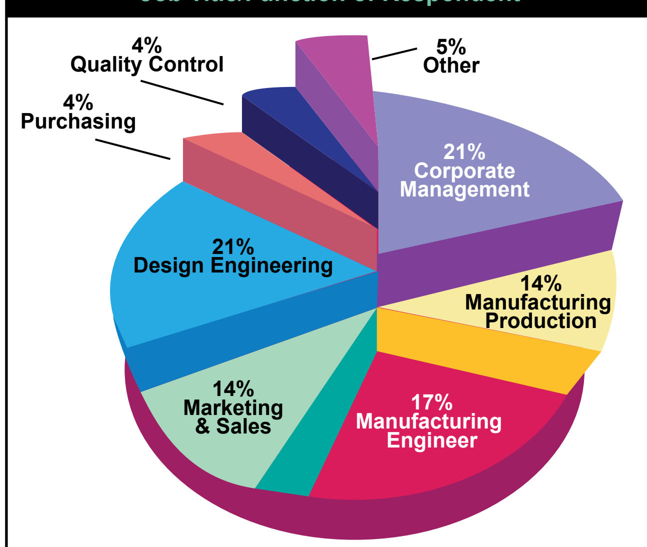
Annual Sales Volume of Company



Type of Operation



Job Title/Function of Respondent



What Are Your Company's Greatest Manufacturing/Engineering Challenges for 2011?

"Increasing quality of manufactured bevel gears."
—Manufacturing engineer at a European gear manufacturing job shop

"Increasing production levels to meet rapidly increasing sales opportunities."
—Manufacturing engineer at a U.S. manufacturer of construction and mining transmissions

"Increasing sales of inventory that I currently have been manufacturing."
—Owner of a U.S. manufacturer of racing transmissions

"Introduction of large-scale assembly processes."
—Corporate executive at a European manufacturer of actuators

"Keeping up with customer demand."
—Manufacturing engineer at a U.S. manufacturer of plastic products

"Keeping engineering and manufacturing costs in line with expectations."
—Design engineer at a U.S. gear and gearbox repair facility

"Maintaining correct staff abilities to meet stringent contract targets."
—Design engineer at a European automotive transmission consultancy

"Maintaining existing equipment with higher overall equipment effectiveness."
—Manufacturing engineer at a U.S. manufacturer of rack and pinion steering gears

"Maintaining our competitive advantage and increasing sales."
—Design engineer at a European design engineering firm

"Maintaining product cost with highest quality standards."
—Production worker at an Indian automobile OEM

"Manufacturing efficiencies."
—Sales manager at a U.S. gear manufacturing job shop

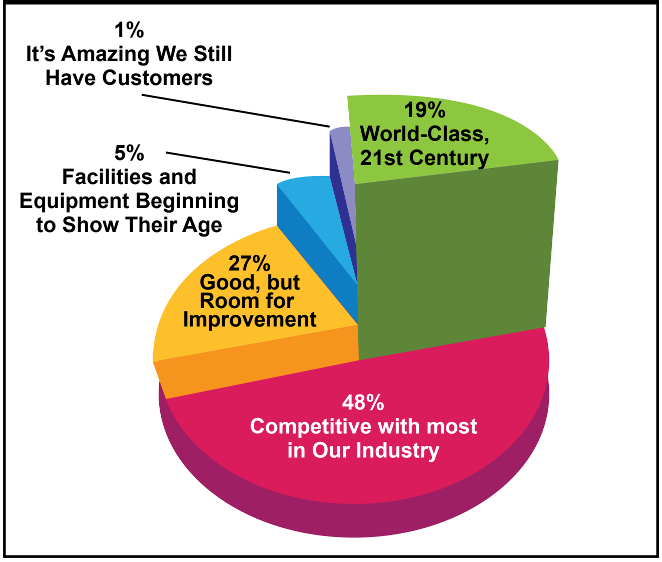
"Many new complex jobs in queue for processing."
—Corporate executive at a U.S. manufacturer of aerospace gears

"Material costs and equipment depreciation-obsolescence."
—Field service manager for a U.S. gear drive manufacturer

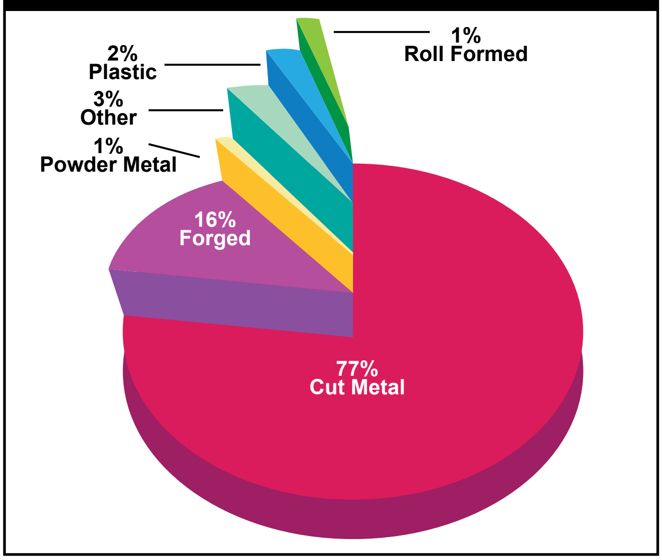
"Meeting foreign competition."
—Corporate executive for a U.S. manufacturer of gears and geared assemblies

continued

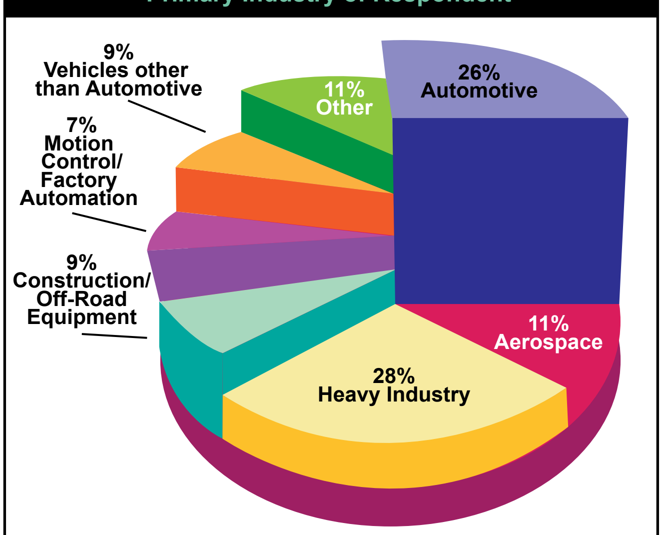
How Do Respondents Describe their Manufacturing Operations and Technology?



Primary Method of Manufacture



Primary Industry of Respondent



What Are Your Company's Greatest Manufacturing/Engineering Challenges for 2011?

"Need to upgrade our gear cutting technology with little capital."
—Corporate executive at a U.S. manufacturer of aerospace gears

"Implementing new processes."
—Sales manager at a U.S. manufacturer of powder metal components

"New production developments, multitasking."
—Corporate executive at a European manufacturer of ground gears

"No university-trained gear engineers available."
—Manufacturing engineer at a U.S. manufacturer of gears and drives

"Qualified and skilled personnel."
—Engineering manager at a Canadian manufacturer of gears and drives

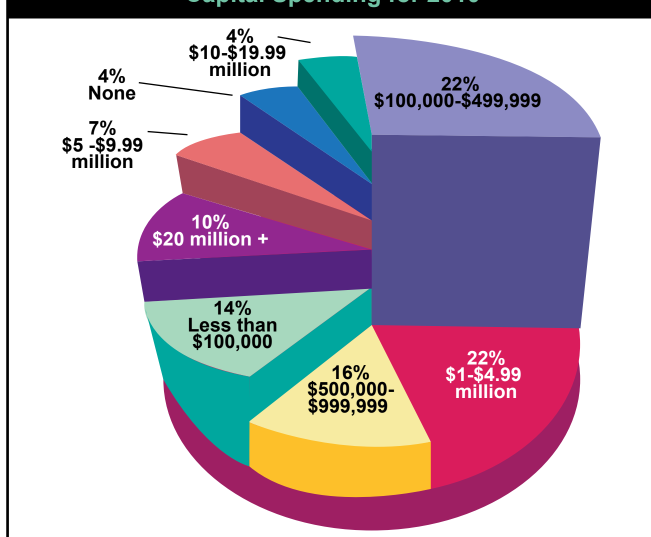
"Quality control and heat treatment."
—Manufacturing engineer at an African gear manufacturing job shop

"Quality employees."
—Corporate executive at a U.S. manufacturer of driveshafts

"Reducing costs."
—Manufacturing engineer at a U.S. manufacturer of custom bearings

"Resources and training."
—Design engineering manager at a U.S. manufacturer of industrial gears

Capital Spending for 2010



What Are Your Company's Greatest Manufacturing/Engineering Challenges for 2011?

"Reversing the economic climate made worse by the Obama regime. All sized private sector businesses are critical to the U.S. economy. We do not need more do-nothing, perpetually brain-dead government jobs! When it comes to what drives our country, Obama and his ideologues are immature, childish, ideological bozos."
—Application technician at a U.S. provider of technical services

"Shorter delivery time requirements."
—Sales manager at a U.S. manufacturer of couplings

"Skilled manpower."
—Heat treating manager at an Indian manufacturer of transmission gears

"Staying ahead of our competitors with innovative products. Trying to balance sales demand with manufacturing capacity."
—Design engineer at a European manufacturer of torque multipliers

"The economy. Will it continue to grow?"
—Design engineer at a U.S. manufacturer of automobile transmissions

"To manufacture various types of product with a minimum tolerance, or more precisely to supply product to a 0.005 accuracy. At present we are up to 0.015."
—Production manager at an Indian gear manufacturing job shop

"To meet the increase in demand without deterioration of quality."
—Design engineer at an Indian manufacturer of two- and three-wheel vehicles

"To supply the current increased demand as well as add new parts."
—Corporate executive at an Indian gear manufacturing job shop

"Winning orders in an increasingly competitive market."
—Corporate executive at a European gear manufacturing job shop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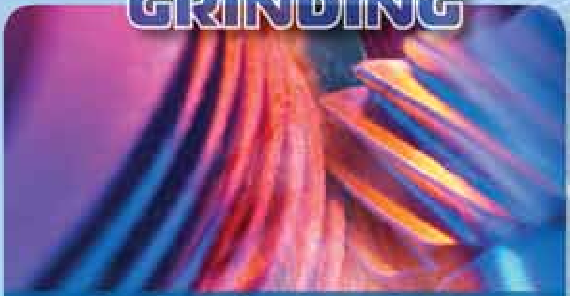

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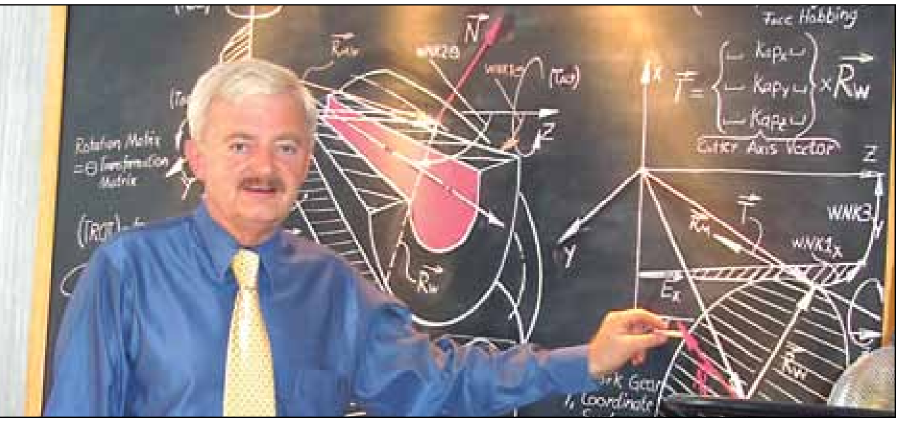
Tribology Aspects in Angular Transmission Systems

Part III: Zerol Bevel Gears

Dr. Hermann Stadtfeld

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



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

Design

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

- Axes are parallel → cylindrical gears (line contact)
- Axes intersect under an angle → bevel gears (line contact)
- Axis cross under an angle → crossed helical gears (point contact)
- Axes cross under an angle (mostly 90°) → worm gear drives (line contact)
- Axes cross under any angle → hypoid gears (line contact)

Zerol bevel gears are the special case of spiral bevel gears with a spiral angle of 0°. They are manufactured in a single-indexing face milling process with large cutter diameters, an extra deep tooth profile and tapered tooth depth. The axis of Zerol bevel gears in most cases intersect under an angle of 90°. This so-called shaft angle can be larger or smaller than 90°; however, the axes always intersect, which means they have, at their crossing point, no offset between them (*Author's note: see also previous chapter, "General Explanation of Theoretical Bevel Gear Analysis" on hypoid gears*). The pitch

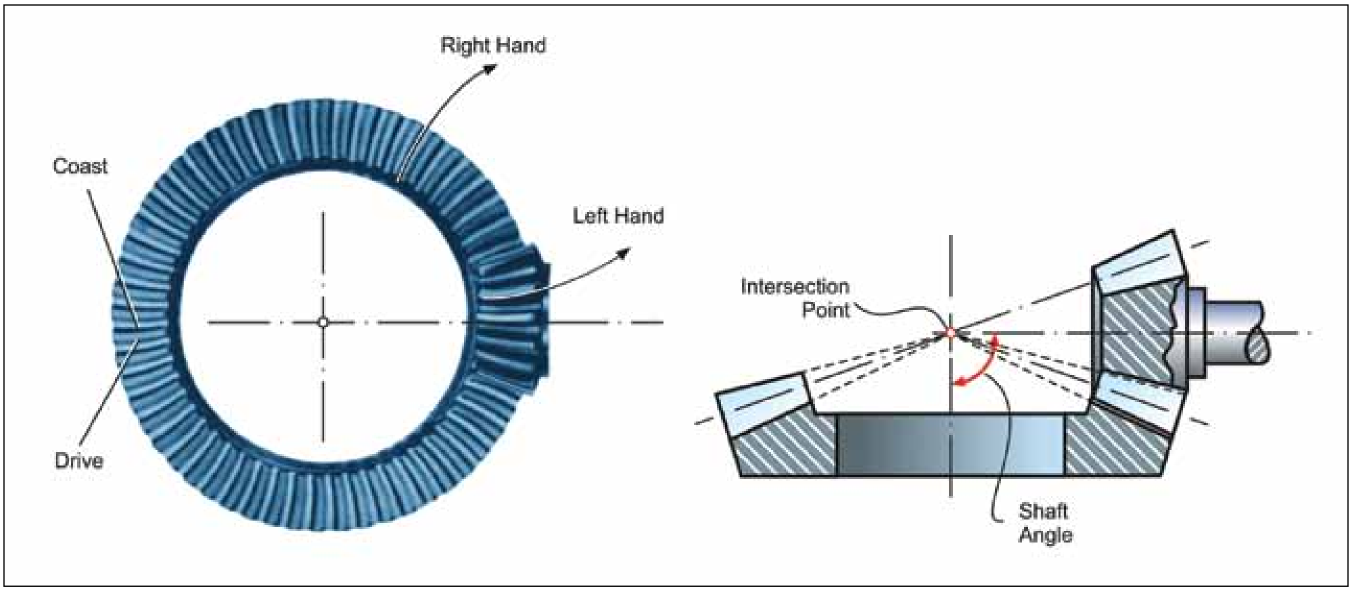


Figure 1—Zerol bevel gear geometry.

surfaces are cones, which are calculated with the following formula:

$$z_1/z_2 = \sin\gamma_1/\sin\gamma_2$$
$$\Sigma = \gamma_1 + \gamma_2$$
$$\text{in case of: } \Sigma = 90° \rightarrow \gamma_1 = \arctan(z_1/z_2)$$
$$\rightarrow \gamma_2 = 90° - \gamma_1$$

where:

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

The advantage of Zerol bevel gears is the low axial forces—like straight bevel gears—but their manufacturing process with completing face mill cutters is significantly faster, and grinding as a hard-finishing process with dressable grinding wheels leads to highly precise gear sets that are often used in aircraft applications.

Zerol bevel gear teeth follow in the face width direction a curve on the conical gear and pinion body that lies tangential to a cone element (zero spiral angle). The tooth lead function in the face width direction, if unrolled into a plane, is a circle. The tooth profile is an octoid. The tooth form with an octoid function will be associated with an initial "natural" profile crowning and, depending on the machining setup, some flank twist. Both effects are utilized together with certain corrective machine settings in order to generate the desired crowning (*see also "General Explanation of Theoretical Bevel Gear Analysis"*).

Figure 1 shows an illustration of a Zerol bevel gear set and a cross-sectional drawing. Per definition, Zerol gears are manufactured in a single-indexing process, applying a standard tooth taper, as shown in Figure 1. However, it is also possible to apply the face hobbing process with parallel-depth teeth in order to manufacture bevel gears with zero spiral angle. Those gears are not considered true Zerol gears, however, because the slot width taper—due to face hobbing—causes a cross-over and fins at the root bottom that are often not acceptable as a production result because of top interference with the opposite member and increased root bending stress.

Analysis

Since the mentioned distortions in tapered-depth tooth systems are detected through comparison to conjugate mating flanks, it is possible to define potential contact lines that would apply in case of no distortions and conjugate flank surfaces. In order to allow deflections of tooth surfaces, shafts, bearings and gearbox housing without unwanted edge contact, a crowning in face width and profile direction is applied. A theoretical tooth contact analysis (TCA) previous to the gear manufacturing can be performed in order to observe the effect of the crowning in connection with the basic characteristics of the particular gear set. This also affords the possibility of returning to the basic dimensions in order to optimize them

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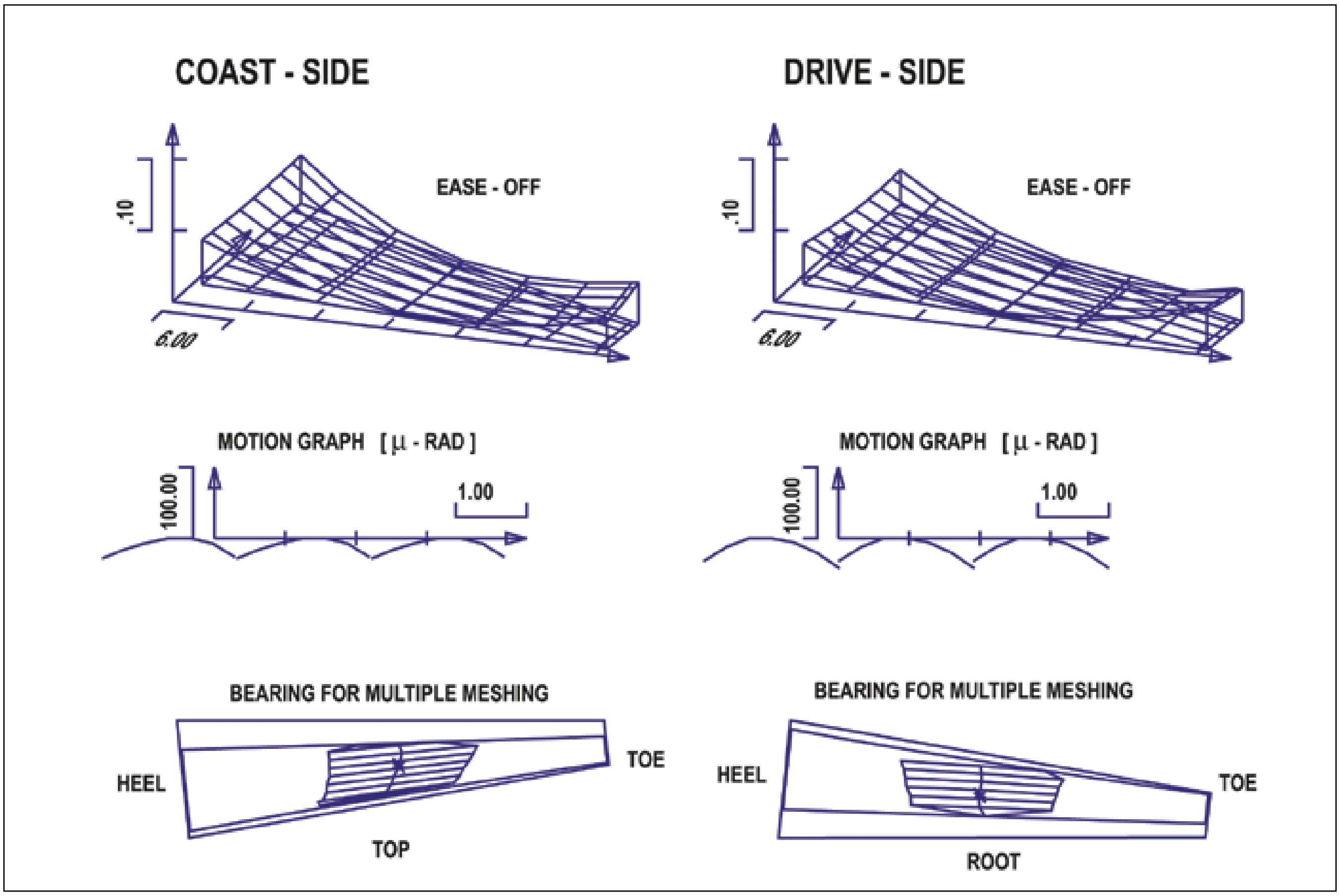

Figure 2—Tooth contact analysis of a Zerol bevel gear set.

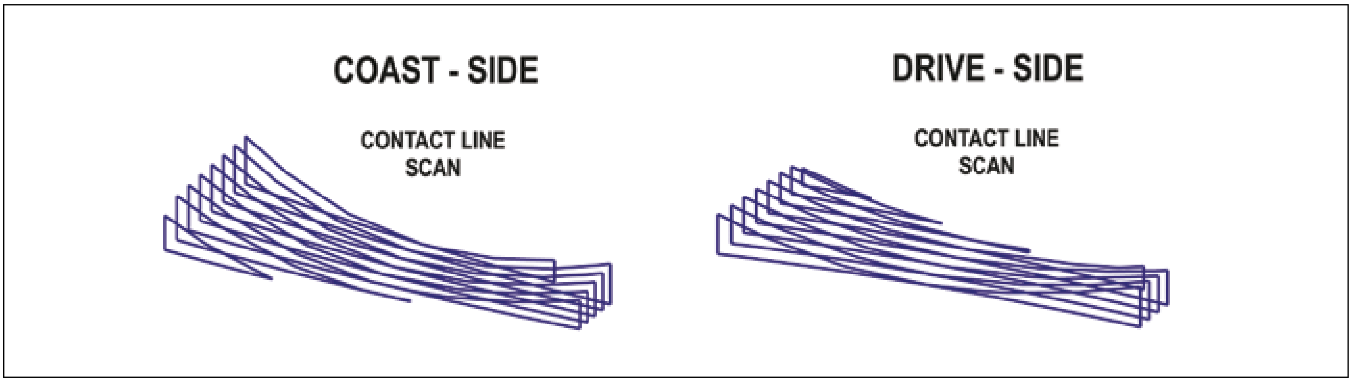

Figure 3—Contact line scan of a Zerol bevel gear set.

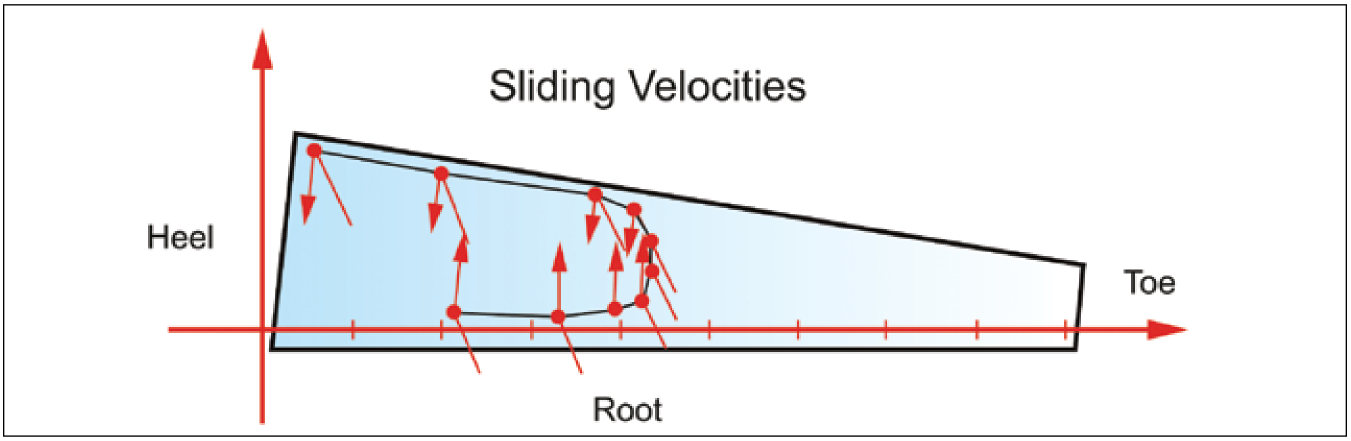

Figure 4—Rolling and sliding velocities of a Zerol bevel gear set along the path of contact.

if the analysis results show any deficiencies. Figure 2 shows the result of a TCA of a typical Zerol bevel gear set.

The two columns in Figure 2 represent the analysis results of the two mating flank combinations (*see also "General Explanation of Theoretical Bevel Gear Analysis"*). The top graphics show the ease-off topographies. The surface above the presentation grid shows the consolidation of the pinion and gear crowning. The ease-offs in Figure 2 have a combination of length and profile crowning, such that a clearance along the boundary of the teeth is established.

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

At the bottom of Figure 2, the tooth contact pattern is plotted inside of the gear tooth projection. These contact patterns are calculated for zero load, and a virtual marking compound film of 6 μm thickness. This basically duplicates the tooth contact one could observe by rolling the real version of the analyzed gear set under light load on a roll tester, while the gear member is coated with a marking-compound-layer of about 6 μm thickness. The contact lines are oriented in the face width direction, depending basically on the 0° spiral angle. The path of contact connects the beginning and end of meshing, and its orientation is nearly perpendicular to the contact lines.

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

Figure 3 shows 10 discrete, potential contact lines with their individual crowning amounts along their length (contact line scan). The gap geometry in contact line direction can be influenced by a change in ease-off topography and optimized regarding the gap kinematic cases (*see also "General Explanation of Theoretical Bevel Gear Analysis," Fig. 8*). The gap geometry perpendicular to the contact line direction (not exactly the same as the path-of-contact direction) does not significantly depend on the ease-off topography, but is mainly dominated by the geometry of the mating tooth profiles.

Figure 4 shows the sliding and rolling velocity vectors of a typical Zerol gear set for each path of contact point for the 10 discussed roll positions. Each vector is projected to the tangential plane at the point-of-origin of the vector. The velocity vectors are drawn inside the gear-tooth-projection plane. The points-of-origin of both rolling- and sliding-velocity vectors are grouped along the path of contact, which is found as the connection of the minima of the individual lines in the contact-line-scan graphic (Fig.4). The velocity vectors can be separated in a component in contact-line direction and a component perpendicular to that—in order to investigate the hydrodynamic lubrication properties—by utilizing the information from the contact line-scan (curvature and curvature change) and the tooth surface curvatures perpendicular to the contact line direction (*see also "General Explanation of Theoretical Bevel Gear Analysis," Fig. 8, cases 1–6*).

In the example of the discussed Zerol bevel gear set, the sliding-velocity vectors are basically profile-oriented. In the top area, the sliding vectors point to the root. Moving along the path of contact from top to bottom, the sliding-velocity reduces its magnitude while attaining a magnitude of zero at the pitch line. Below the pitch line, the sliding-velocity develops, growing positive magnitudes (towards the root of the gear tooth). The maximal magnitude of the sliding-velocities (top-versus-root) is a result of the distance from the pitch line. In the present case, the distance between the lowest active flank line to the pitch line is larger than the distance from the pitch line to the top. The rolling-velocity vectors point to the root and have basically all the same orientation. The orientation is a result of the spiral angle (zero spiral angle delivers profile-oriented rolling). The shrinking magnitude of the rolling-velocity (moving from top to bottom) is caused by the decreasing circumferential speed towards the outer diameter.

The freedoms for optimizing the lubrication gap geometry and kinematics in Zerol bevel

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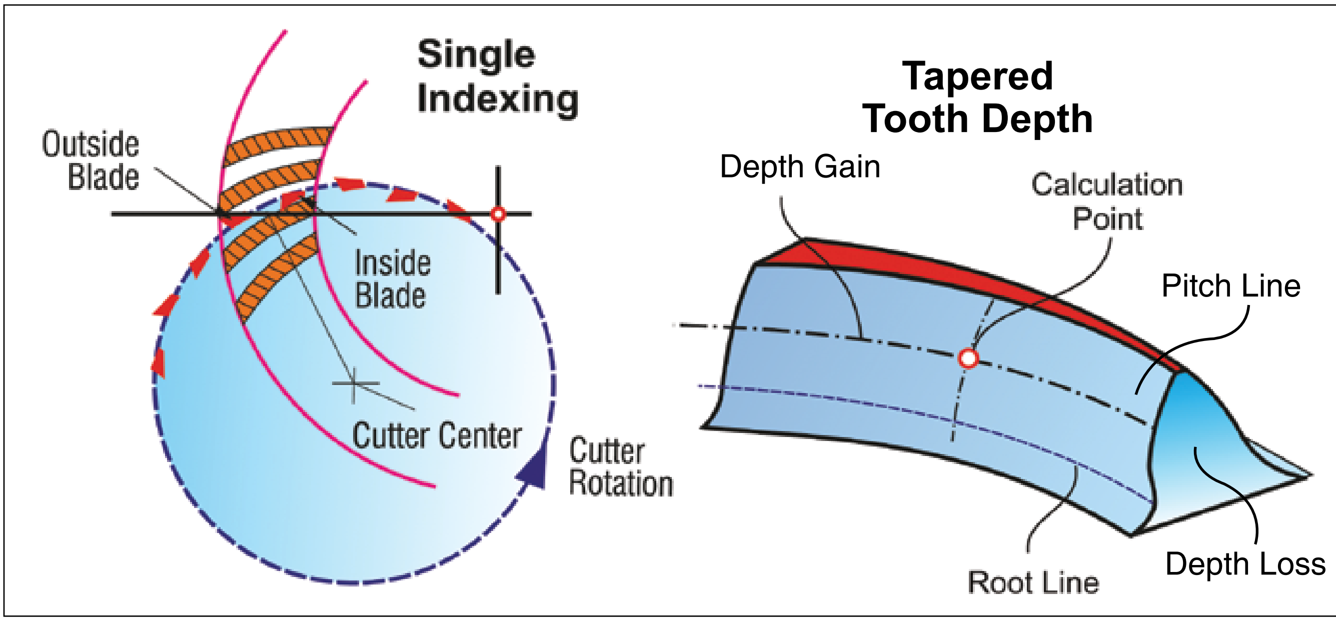

Figure 5—Face milling process and tapered tooth depth.

Figure 6—Zerol bevel gear cutting.

gears are limited to the relocation of the pitch line and a change in crowning in length direction.

Manufacturing

Zerol bevel gears are manufactured in a single-indexing face milling process. In the face milling process, the blades are oriented around a circle and pass through one slot (while they plunge or generate the flanks of that particular slot), as illustrated in Figure 5. The work is not performing any indexing rotation. At the blade tip and in equidistant planes (normal to the cutter head axis), the slot width produced has a constant width between toe and heel. In order to achieve a proportionally changing slot width (and tooth thickness), the root line of face milled bevel gears is inclined versus the pitch line (Fig. 5, right). This modification has to be implemented in both members, which is why the face angle requires the same modification as the root angle of the mating member.

Figure 6 is a photo of the view into the work chamber of a free-form bevel and hypoid gear cutting machine during the high-speed dry-cutting of a Zerol bevel gear. The face cutter head has coated carbide stick blades that are arranged in blade groups of one inside and one outside blade oriented around a circle.

Hard-finishing after heat treatment, if required by the particular application, is generally done by grinding. The grinding wheel resembles the cutter-head geometry, while the grinding machine uses the same set-up geometry and kinematics as the cutting machine for the previous soft machining.

Application

Most Zerol bevel gears to be used in power transmissions are manufactured by carburizing steel and undergo a case-hardening to a surface hardness of 60 Rockwell C (HRC) and a core hardness of 36 HRC. Because of the higher pinion revolutions, it is advisable to give the pinion a higher hardness than the ring gear (e.g., pinion 62 HRC, gear 59 HRC).

Regarding surface durability, Zerol bevel gears are very similar to straight bevel gears. At the pitch line, the sliding-velocity is zero, and the rolling-velocity under certain loads cannot maintain a surface-separating lubrication film. This might in certain cases of high load or speed result in pitting along the pitch line that can destroy the tooth surfaces and even result in tooth flank fracture. However, it is possible that the pitting can be stabilized if the damage-causing condition is not often represented in the duty cycle. Figure 7 is a photograph of typical pitch line pitting on a Zerol bevel ring gear flank surface.

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

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

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

where:

T	torque of observed member
A_m	mean cone distance
γ	pitch angle
α	pressure angle
F_x, F_y, F_z	bearing load force components

The bearing force calculation formulas are based on the assumption that one pair of teeth transmits the torque with one normal force vector in the mean point of the flank pair. The results are good approximations that reflect the real bearing loads for multiple-tooth meshing within an acceptable tolerance. A precise calculation is for example possible with the Gleason bevel and hypoid gear software.

Zerol bevel gears have lesser axial forces than spiral bevel gears. The axial-force component due to the spiral angle is zero. Zero spiral angle minimizes the face contact ratio to zero but results in the maximal tooth root thickness.

As a rule, bevel gears that are not ground or lapped after heat treatment show the highest root strength with the lowest spiral angles. This explains why in those cases Zerol and straight bevel gears are still the bevel gears of choice.

Zerol bevel gears can operate with regular transmission oil or, in the case of low RPMs, with a grease filling. With circumferential speeds above 10 m/min., a sump lubrication with regular transmission oil is recommended. The oil level has to cover the face width of the teeth that are the lowest in the sump. More oil causes foaming, cavitations and unnecessary energy loss. There is no requirement for any lubrication additives. The preferred operating direction of Zerol bevel gears is the drive side, where the convex gear flank and the concave pinion flank mesh together. In the drive direction (Fig. 8), the forces between the two mating members bend the pinion sideways and axially away from the gear, generating the most backlash. Coast-side operation reduces the backlash in extreme cases to zero, which interrupts any lubricant flank separation and leads to immediate surface damage—which is often followed by tooth fracture.

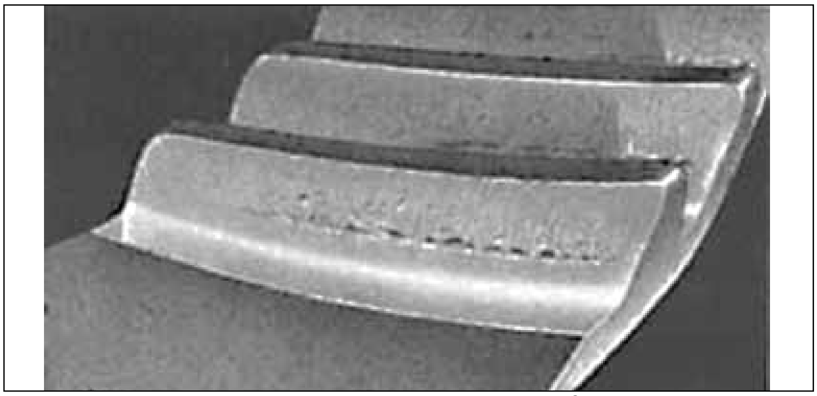

Figure 7—Pitch line pitting on a Zerol bevel gear surface.

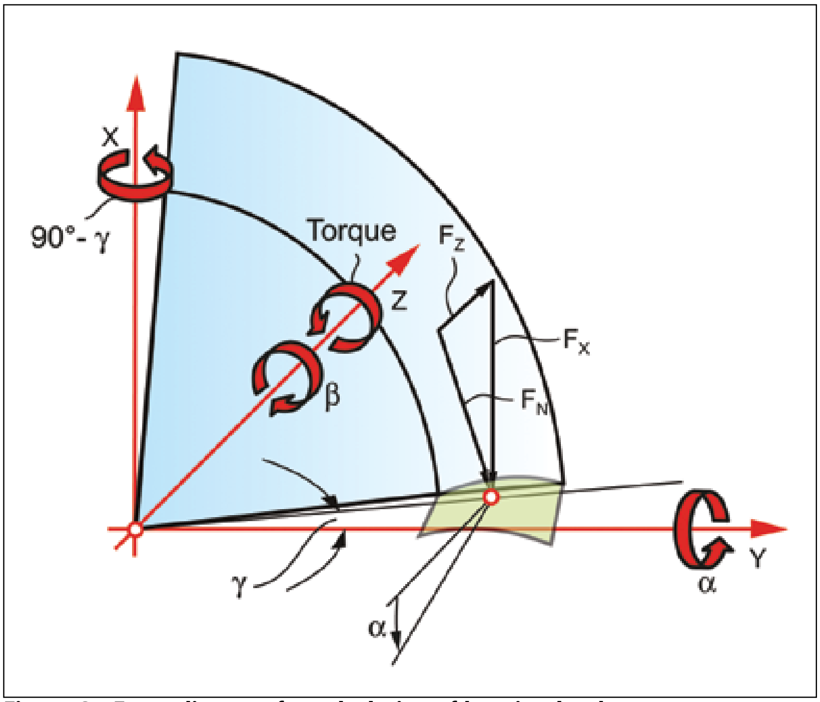

Figure 8—Force diagram for calculation of bearing loads.

Next issue: Spiral Bevel Gears.

Gear Fault Detection Effectiveness as Applied to Tooth Surface Pitting Fatigue Damage

David G. Lewicki, Paula J. Dempsey, Gregory F. Heath and Perumal Shanthakumaran

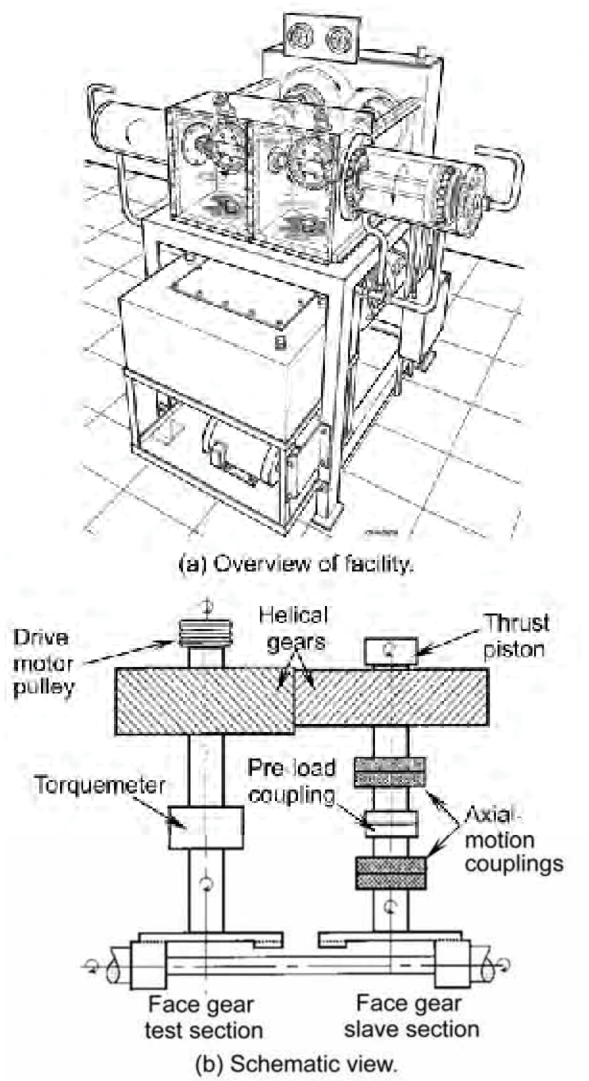

Figure 1—NASA Glenn spiral bevel gear, face gear test facility.

Management Summary

A study was performed to evaluate fault detection effectiveness as applied to gear-tooth-pitting-fatigue damage. Vibration and oil-debris monitoring (ODM) data were gathered from 24 sets of spur pinion and face gears run during a previous endurance evaluation study. Three common condition indicators (RMS, FM4, and NA4 [*Ed.'s note: See Appendix A—Definitions*]) were deduced from the time-averaged vibration data and used with the ODM to evaluate their performance for gear fault detection. The NA4 parameter showed to be a very good condition indicator for the detection of gear tooth surface pitting failures. The FM4 and RMS parameters performed average to below average in detection of gear tooth surface pitting failures. The ODM sensor was successful in detecting a significant amount of debris from all the gear tooth pitting fatigue failures. Excluding outliers, the average cumulative mass at the end of a test was 40 mg.

Introduction

Gears are used extensively in rotorcraft drive systems. Effective gear-fault detection is crucial to ensure flight safety. In addition, tremendous economic benefits can result from condition-based maintenance practices, for which gear-fault detection plays an important role.

Over the past 25 years, much research has been devoted to the development of health and usage monitoring systems for rotorcraft gearbox and drivetrain components. Three classic publications on gear diagnostics are by Stewart (Ref. 1), McFadden (Ref. 2) and Zakrajsek (Ref. 3). Samuel and Pines give a comprehensive review of the state of the art in vibration-based helicopter transmission diagnostics (Ref. 4). Dempsey, et al., presents a summary of current methods to identify gear health, with emphasis on FAA and U.S. Army rotorcraft applications (Ref. 5). Recent refinements to vibration-based, gear-fault detection have been made (Refs. 6–8) along with other methods such as vibro-acoustics (Ref. 9), acoustic emission (Ref. 10) and impact-velocity modeling (Ref. 11). A common theme noted is that experimental data verifying fault-detection algorithms and condition-indicator (CI) thresholds are sparse.

In a recent study on face gear endurance (Ref. 12), a number of test sets were instrumented with a gear-fault detection system and run until failure. The gears failed from tooth surface fatigue, and a large fault-detection database was populated. The objective of this study is to use this database and evaluate fault-detection effectiveness as applied to gear tooth pitting fatigue damage. A further objective is to evaluate the repeatability of the fault detection methods. Vibration and oil-debris monitoring data were gathered from 24 sets of gears run during the previous endurance evaluation study. The gears were tapered, involute spur pinions in mesh with face gears. Three common condition indicators (RMS, FM4 and NA4) were deduced from the vibration data and used to evaluate gear-fault detection. Receiver-operating characteristic curves were further used on the data to define threshold limits. Lastly, cumulative mass from oil-debris monitoring was used for fault detection.

Apparatus

Test Facility. The experiments reported in this report were tested at the NASA Glenn Research Center (GRC) spiral bevel gear/face gear test facility. An overview sketch of the facility is shown in Figure 1a, and a schematic of the power loop is shown in Figure 1b. The facility operates in a closed-loop arrangement. A spur pinion drives a face gear in the test (left) section. The face gear drives a set of helical gears, which in turn drive a face gear and spur pinion in the slave (right) section. The pinions of the slave and test sections are connected by a cross shaft, thereby closing the loop. Torque is supplied in the loop by physically twisting and locking a torque in the pre-load coupling on the slave section shaft.

Additional torque is applied through a thrust piston (supplied with high-pressure nitrogen gas), which exerts an axial force on one of the helical gears. The total desired level of torque is achieved by adjusting the nitrogen supply pressure to the piston. A 100–hp DC-drive motor, connected to the loop by V-belts and pulleys, controls the speed as well as provides power to overcome friction. The facility has the capability to operate at 750 hp and 20,000 rpm pinion speed. A torque meter in the loop on the test side measures torque and speed. The facility is also equipped with thermocouples, oil flow meters, pressure transducers, accelerometers, counters and shutdown instrumentation to allow 24-hour unattended operation.

Test gears. The design parameters for the pinions and face gears used in the tests are given in Table 1. A photograph of the test specimens is shown in Figure 2. The set was primarily designed to fail in surface-pitting-fatigue mode. The set had a

continued

TABLE 1—TEST GEAR DESIGN DATA

AGMA quality	12
Number of teeth; pinion, gear	19,73
Diametral pitch (teeth/in.)	10.6
Pressure angle (deg)	27.5
Shaft angle (deg)	90
Face width (in.); pinion, gear	0.8,0.6
Hardness (Rc); case, core	62,38
RMS surface finish (min)	16
Material	X53 steel

Figure 2—Test gears.

reduction ratio of 3.842:1. The pinions were slightly tapered, which allows the independent setting of backlash for the multiple pinions and idlers in the split-torque transmission application (Ref. 13). The pinions and face gears were made from carburized and ground vacuum induction melting vacuum arc re-melting (VIM–VAR) Pyrowear 53 steel per AMS 6308 using standard aerospace practices. At 6,000 lb-per-in. face gear torque, the calculated AGMA contact stress index was 250 ksi and the calculated AGMA bending stress index was 72 ksi, using approximate spur gear calculations per AGMA (Ref. 14).

Gear fault detection instrumentation. A schematic of the gear fault detection instrumentation is shown in Figure 3. Two high-frequency accelerometers and two photoelectric tachometers were used for vibration monitoring. One accelerometer was installed on the test (left)-side pinion housing and the other was installed on the slave (right)-side pinion housing and were used to monitor the left- and right-side meshes, respectively. The accelerometers had integral electronics with a nominal 10 mV/g sensitivity, 70 kHz resonant frequency and were linear within 10 percent up to 20 kHz. One tachometer was installed on the high-speed pinion shaft and the other was installed on the low-speed face gear shaft. Each produced once-per-shaft-revolution indications and was used for time averaging of the vibration data. The outputs of the accelerometers and tachometers were acquired and digitized by a PC.

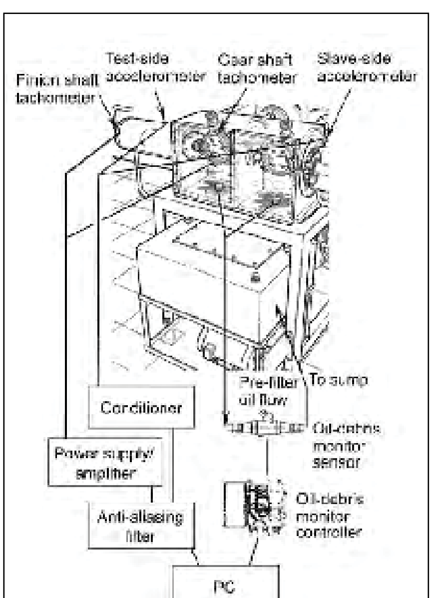

Figure 3—Gear fault detection instrumentation.

Vibration data were acquired once every minute during the tests. The accelerometers and tachometer signals were sampled at a 155 kHz sampling rate (each) for a 10-second duration by an in-house-developed computer program. The program performed linear-interpolation and time-synchronous averaging. This produced left- and right-vibration traces relative to the pinion and gear shafts. For the 10-second acquisition, approximately 380 averages were achieved for a gear trace and over 1,000 averages for a pinion trace. The traces represented the time-averaged vibration for a period of one revolution of the corresponding shaft, using 1,024 points for the pinion shaft trace and common condition indicators were calculated at each acquisition: RMS, FM4 and NA4. A commercially available in-line ODM was used to measure metallic content generated in the lubrication system due to mechanical component fatigue failures (Ref. 15). The ODM sensor element consisted of three coils that surrounded a nonconductive section of tubing. The two outside field coils were oppositely wound and driven by an AC current source. The center coil measured the disturbance to the magnetic fields caused by the passage of metallic particles through the sensor. The disturbance was measured as a sinusoidal voltage where the magnitude of the disturbance was proportional to the size of the particle. The ODM controller continuously monitored the sensor and stored values of the calculated cumulated mass of the debris as well as particle counts assembled in bins of particle sizes. The PC system from above polled the ODM controller through its COM port during each vibration acquisition, where it time-stamped and stored the accumulated mass along with the vibration CIs.

The ODM sensor was installed in the gravity-fed scavenge oil line coming from the test hardware (Fig. 3). This line contained oil from the left-side mesh, left-side pinion support bearing, right-side mesh and right-side pinion support bearing. Unfortunately, due to the test rig design, isolation of the oil lines for these components was not possible. However, the ODM data was still used as an indicator of the health of the gears as a whole.

Test procedure. For each set tested, detailed installation and break-in run procedures (Ref. 12) were followed to produce acceptable contact patterns and backlash. After acceptable installation, the pre-load coupling was adjusted to produce a face gear torque between 3,000 and 5,000 lb-in. The gears were then run at required speed and torque for the specific test (torque adjusted using load piston). Facility parameters (speed, torque, oil pressures and flows, temperatures) as well as the previously mentioned vibration and ODM data were collected. During the tests, the gears were inspected at routine intervals (5–10 million face gear cycles) or when an abnormal facility shutdown occurred. The gears were run until a surface-durability failure occurred or a suspension was defined. A surface-durability failure was defined as macropitting, or spalling, of at least 0.1 in. continuous length along the contact area on any tooth of a tested pinion or face gear. Once a test was completed, the failed gears were removed from the facility, cleaned and photographed for documentation purposes. A replacement set was installed per above and testing continued.

Twenty-four sets of gears were tested. Tests were performed at three load levels:

1. 7,200 lb-in. face gear torque (275 ksi calculated AGMA contact stress)
2. 8,185 lb-in. face gear torque (292 ksi contact stress)
3. 9,075 lb-in. face gear torque (307 ksi contact stress)

Test speeds were 2,190–3,280 rpm face gear speed, depending on the vibration levels of the test.

Results and Discussion

Endurance test results.A summary of the results from the endurance tests is given in Table 2. Twelve sets were run at 7,200 lb-in. Seven sets at 8,185 lb-in, and 5 sets at 9,075 lb-in. face gear torque. The test speeds were 2,190–3,280 rpm face gear speed. Initial tests were run at higher speeds to produce more cycles per time. However, due to wear of the specimens during test, excessive facility vibration levels were produced and the speeds were reduced to lessen vibration to acceptable levels. During pretest facility check-out runs, resonant speeds from around 2,500 to 3,000 rpm were discovered, and thus avoided during test.

Of the 24 sets of gears tested, 17 sets resulted in spalling/macropitting failures. The other 7 sets were suspended with moderate-to-heavy wear, but had no spalling. For all the 17 sets that failed, spalling occurred on the pinion. In some cases, spalling occurred on both the pinion and face gear. There were zero-instances of face gear spalling with no pinion spalling.

Thus, the remainder of this study will concentrate on pinion results only. The tests sets were classified into four groups: (1) pinion macropitting with single or few teeth pitted (this occurred for 5 sets); (2) pinion macropitting with multiple/all teeth pitted (this occurred for 12 sets); (3) moderate pinion wear but no macropitting (this occurred for 3 sets); and (4) heavy pinion wear but no macropitting (this occurred for 4 sets). An example of a pinion with single or few teeth pitted is given in Figure 4a. An example of a pinion with multiple teeth pitted is given in Figure 4b. The number of cycles tested per set ranged from 32.7 to 590.9 million pinion cycles.

Vibration and ODM data were continuously collected once every minute during all tests. Three gear-fault CIs (RMS, FM4 and NA4) were calculated from the time- averaged vibration signal for the pinions. The results for all the tests are given in Appendix B. Plotted are RMS, FM4 and NA4 versus data point, where each data point represents one minute of test. As previously mentioned in the Test Procedure section, test gears were replaced after failure or suspension with new sets and testing continued. The absolute start and end times for the 24 sets were intermixed. For each set shown in Appendix B, the data point number is relative to the specific set in question. Thus, as an example, data point 10,000 for set 1 (Fig. B1) does not correspond to the same point in time as data point 10,000 for set 2 (Fig. B2).

The plots in Appendix B are divided with two types of separators. The first separator is labeled "rig shutdown" (dotted lines), representing rig shutdowns either for routine inspection or abnormal facility parameter. In these cases, no changes were made to the test gear set setup or vibration monitoring system. The second separator is labeled "vib reset" and occurred when the vibration monitoring system was reset. This primarily occurred when the opposite side set was replaced due to failure or suspension. The major significance of a "vib reset" is the re-initialization of the running average of the

continued

(a) Single/few teeth macropitting failure.

(b) Multiple teeth macropitting failure.

Figure 4—Typical macropitting pinion tooth surface fatigue failures.

TABLE 2—RESULTS OF ENDURANCE TESTS

Set no.	Side	Face gear speed, rpm	Face gear torque lb-in.	M Pin cycles	Pinion condition
1	Right	2200 to 3280	7200	361.5	4
2	Left	2880 to 3280	7200	590.9	4
3	Right	2880 to 3280	7200	559.8	4
4	Left	2300	9075	577.2	2
5	Right	2300	9075	88.0	2
6	Left	2300	9075	38.4	2
7	Right	2300	9075	41.9	2
8	Left	2300	9075	32.7	2
9	Right	2300	8185	37.7	2
10	Left	2200 to 2300	7200	461.8	4
11	Right	2300	7200	65.7	2
12	Right	2300	7200	66.1	2
13	Left	2280	8185	126.0	1
14	Right	2300	8185	202.9	3
15	Left	2300	8185	102.6	1
16	Right	2300	8185	212.9	2
17	Left	2300	8185	42.6	1
18	Left	2300	8185	144.5	3
19	Left	2300	7200	35.7	1
20	Right	2190 to 2300	7200	45.3	2
21	Right	2190 to 2300	7200	99.1	2
22	Left	2300	7200	60.7	1
23	Left	2190 to 2300	7200	161.0	2
24	Right	2200	7200	113.0	3

Pinion condition:
1 = Macropitting, single/few teeth.
2 = Macropitting, multiple teeth.
3 = Moderate wear
4 = Heavy wear

variance for the NA4 parameter (see Equation 3, Appendix A). Lastly, portions of the data in Appendix B are also classified as "healthy" and "faulty," corresponding to a healthy or faulty pinion condition. This classification will be used for determining thresholds as described in a later section of this study.

The results from Appendix B will be used for analysis of gear-fault detection and described in detail in later sections of this study. For now, however, a few general comments can be made. Rig shutdowns and vib resets produced discontinuities in the CI responses. Some discontinuities were significant (the RMS response for data points 4,335 to 5,742 of Figure B5 as an example). For most cases, a failure of the opposite-side set was apparent in the CI responses of a given set. Figure B6 for set 6 is an example where set 5 failed at data point 1,128. In general, the magnitude of the RMS CI varied from set to set. FM4 was generally bounded within values of 2 to 5. NA4 was also generally bounded for healthy components, but showed a significant increase during failure. NA4, however, was usually more sensitive to inspections and shutdowns.

TABLE 3—MEAN AND STANDARD DEVIATION STATISTICS FOR ALL SETS, HEALTHY STATE CONDITION

		RMS		FM4		NA4	
Set no.	No. points	Mean	Std. dev.	Mean	Std. dev.	Mean	Std. dev.
1	8,782	7.82	0.61	2.85	.23	1.83	0.25
2	10,000	3.93	.52	2.87	.51	4.52	2.22
3	10,000	6.14	1.58	3.25	.55	5.32	2.85
4	2,000	2.90	0.10	3.16	.24	3.25	0.92
5	4,000	4.95	.65	2.26	.10	1.56	.50
6	873	4.21	.15	3.21	.25	4.38	1.43
7	647	7.92	.22	2.42	.05	2.46	0.49
8	532	3.00	.24	2.68	.11	3.21	.47
9	54	5.95	.27	2.55	.03	2.35	.12
10	15,440	4.74	1.30	2.81	.20	3.72	.69
11	6,510	5.04	1.06	2.57	.32	6.94	2.92
12	1,000	3.42	0.10	2.83	.09	2.29	0.16
13	13,155	3.64	.61	2.99	.26	4.08	1.30
14	13,155	9.34	1.85	2.31	.31	1.67	0.43
15	8,960	3.10	0.45	2.89	.12	3.11	.54
16	4,242	6.31	.11	2.14	.04	3.83	.67
17	4,000	2.53	.22	2.97	.18	2.73	1.04
18	12,918	3.07	.30	2.59	.14	4.57	1.04
19	2,000	5.24	.39	2.85	.24	2.85	0.38
20	237	5.03	.32	3.02	.09	3.43	.25
21	3,889	10.73	.68	2.08	.13	2.66	.60
22	2,000	2.98	.18	2.59	.13	3.25	.62
23	3,309	3.12	.15	2.40	.13	2.62	.36
24	8,768	6.13	1.03	3.13	.17	4.23	1.07
All	136,471	5.23	2.39	2.75	.42	3.65	1.91

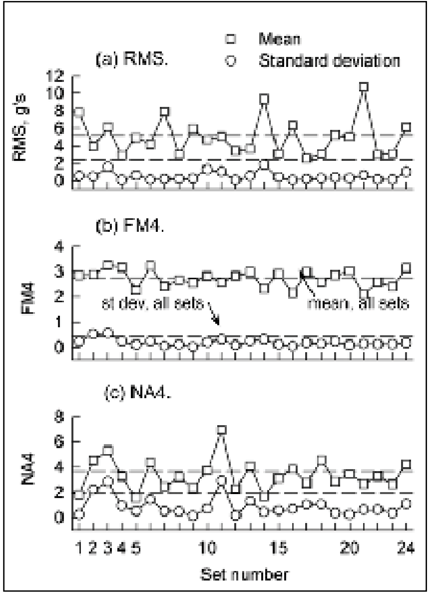

Figure 5—Mean and standard deviation statistics for all sets, healthy state condition.

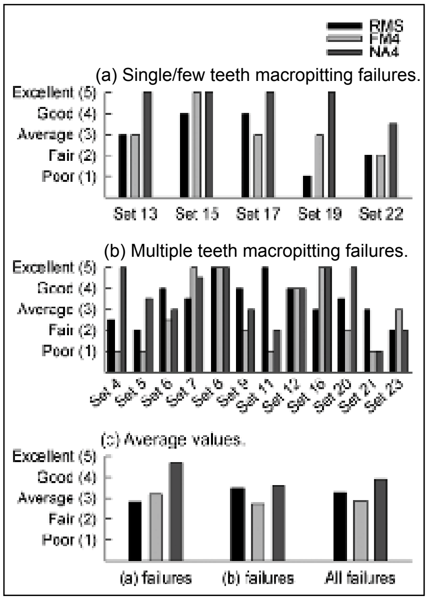

Figure 6—Qualitative analysis of condition indicator fault detection effectiveness.

Evaluation of data from healthy components. The objective of this section is to investigate the variability of the CIs for known healthy components. The data labeled "healthy" in Appendix B were assembled and the means and standard deviations of the CIs for these data were determined. For 15 of the 24 sets, the healthy data were selected at the start of the set installation. For the remaining sets, the healthy data were offset due to the influence of the opposite-side set failures on the CI results. The mean and standard deviation results are shown in Table 3 and Figure 5.

RMS had a large variation among sets, ranging in mean values from 2.53 to 10.73 g. FM4 had a fairly steady value of means, with a total average of 2.75 and a relatively low standard deviation. NA4 had a slightly higher mean than FM4 and significantly larger scatter.

Qualitative analysis of gear-fault detection. For the qualitative analysis, the gear-fault detection effectiveness was evaluated based on visual inspection of the CI plots from Appendix B. Each CI was rated for fault-detection effectiveness for each set with macropitting. Ratings varied from 1 to 5, where 5 was excellent effectiveness, 1 poor. A CI was given a 5 rating for a set if it showed an indisputable increase in value at the time of failure. An example of this is the NA4 response for set 13 (Fig. B13). In this case, NA4 increased by a factor of 50 at the end of the test. A CI was subjectively rated less- effective when it did not show a noticeable increase at time of failure; it decreased with increasing failure progression, exhibited extraneous jumps or spikes or was clouded with noise throughout the test. An example of a 3 rating is given for FM4 for set 17 (Fig. B17). Here, FM4 increased at the start of failure (data point 4,500) but decreased as the pitting failure propagated. An example of a 1 rating is given for FM4 for set 4 (Fig. B4). Here, FM4 showed no response to the failure at the end of the test.

Figure 6 depicts the results of the qualitative analysis. For the single/few teeth macropitting failures (Fig. 6a), NA4 showed an excellent fault-detection effectiveness. FM4 showed a slightly above-average effectiveness. NA4 and FM4 were primarily developed to detect isolated gear tooth faults, which explains the excellent performance of NA4. FM4 suffered in effectiveness due to noise and the decrease in values with increased fault progression. RMS showed a slightly below-average effectiveness, indicating that isolated gear-faults did not significantly increase the overall vibration signature.

For the multiple teeth macropitting failures (Fig. 6b), the fault-detection effectiveness of NA4 and FM4 decreased, compared to the single/few teeth failure modes. Again, this is not surprising since the parameters were developed to detect isolated tooth faults. The RMS fault-detection effectiveness increased due to the increased influence of the multiple teeth faults on the overall vibration signature. In general, and considering all failures (Fig. 6c), NA4 showed good fault-detection effectiveness, FM4 was slightly below average, and RMS was average.

Some general observations were noted. Again, CI discontinuities from the inspections and resets increased the difficulty for successful fault detection. This was especially true in the current test setup where opposite side set failures influenced CI performance. Another general observation was that the vibration spectrum was dominated by the gear meshes. This was deduced from analyzing gear orders in the time-averaged vibration as well as analyzing raw vibration signals (non-time-averaged) from facility accelerometers.

Quantitative analysis of gear-fault detection. Receiver operating characteristic (ROC) curves were used to validate the qualitative analysis. ROC curves are used in signal detection theory to identify tradeoffs between failure detection and false alarms. They have been used in the medical fields for making health decisions and for assessing the predictive accuracy of the tools used to make these decisions (Refs. 16–17). Interpretation of medical tests can vary between diagnosticians. ROC curves have been used as a tool to assess the performance of tests independent of the threshold, providing a common metric for comparison (Ref. 18).

The procedure in using ROC curves is as follows. First, CI data is extracted into healthy and faulty groups corresponding to healthy and faulty components. The means and standard deviations of the groups are then determined. Figure 7 shows probability density functions for sample data with a mean and standard deviation of 3.0 and 0.5, respectively, for the healthy set, and a mean and standard deviation of 5.0 and 1.0, respectively, for the faulty set. Note that normal distributions are used in this example and this assumption was used on all the data in this study. For a given CI value (CI = 3.5 in Figure 7 as an example), the false alarm rate and hit rate are the shaded areas in the figure, and can be determined from statistics using the CI value probability distribution to calculate the area under the curve. By sweeping through a range of CIs (usually from the mean of the healthy to the mean of the faulty set), one can tabulate and plot the hit rates versus false-alarm rates. This is known as the ROC curve. The ROC curve can be used to evaluate the CI fault detection effectiveness as well as to determine a threshold CI value. The threshold CI value with the best performance is the point corresponding to the upper-left-most point on the ROC curve. This maximizes the hit rate while minimizing the false-alarm rate. One method to determine the optimum numerical value of the threshold is to determine the CI value for the intersection of the tail edge of the healthy probability density function with the leading edge of the faulty probability density function.

ROC curves are given in Figure 8 for two examples. The first example has considerable overlap between the healthy and faulty groups. The threshold value is 3.62 for this example. The ROC curve is fairly smooth (Fig. 8a) and the threshold value has less significance due to poor separation of healthy and faulty data. If actual data performed in this manner, the CI would be a poor fault-detection indicator. The second example has a greater spread between the healthy and faulty groups. The ROC curve has a sharp edge (Fig. 8b) at the upper-left location and thus a tangible threshold. The threshold value with the optimum performance is 4.42 for this example. If actual data performed in this manner, the CI would be a good fault-detection indicator.

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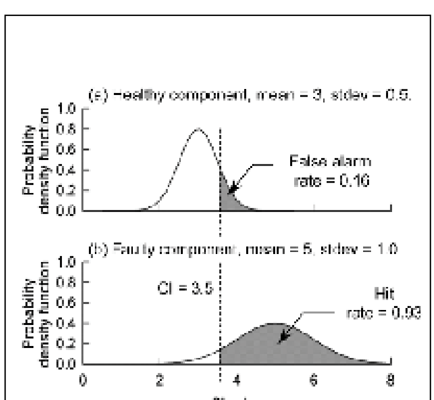

Figure 7—Sample probability density functions.

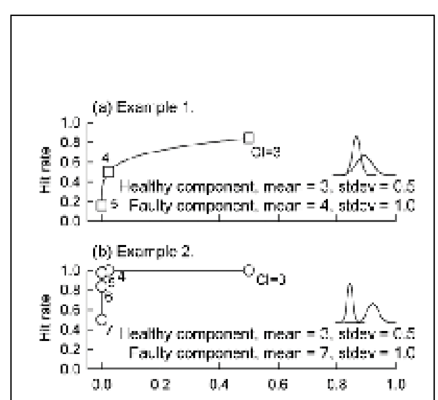

Figure 8—Sample receiver operating characteristic (ROC) curves.

ROC curves for RMS, FM4 and NA4 are given in Figures 9–11 for the macropitting single/few teeth failures (pinion condition 1). This was based on the healthy and faulty data of sets 13, 15, 17, 19 and 22. The means and standard deviations of the healthy and faulty data, along with the estimated thresholds from the ROC curve analysis, are given in Table 4. ROC curves for the macropitting, multiple-teeth failures (pinion condition 2) are given in Figures 12–14. The means, standard deviations and thresholds are given in Table 5. Note that analysis for the macropitting multiple-teeth failures only included 9 out of the 12 total sets for this failure mode (sets 4, 5, 7, 8,

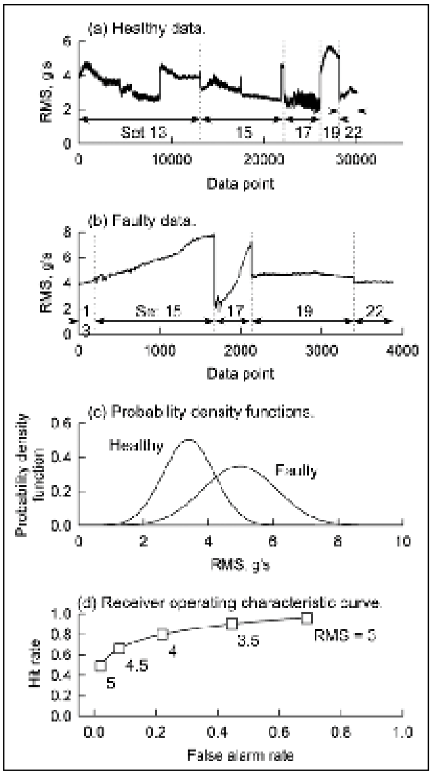

Figure 9—Summary results for RMS condition indicator for macropitting, single/few teeth failures.

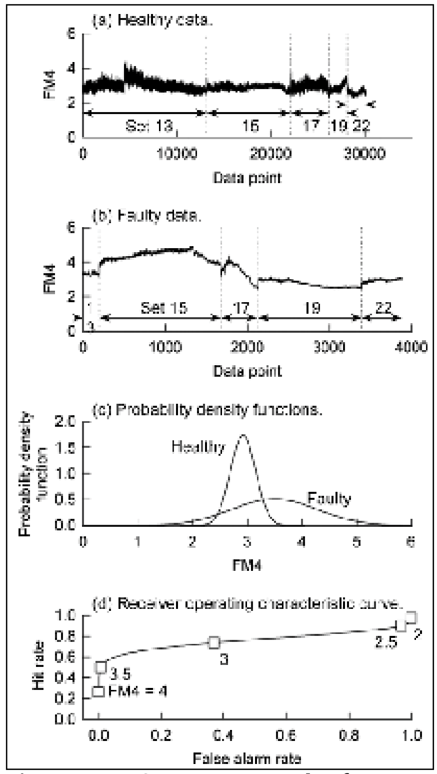

Figure 10—Summary results for FM4 condition indicator for macropitting, single/few teeth failures.

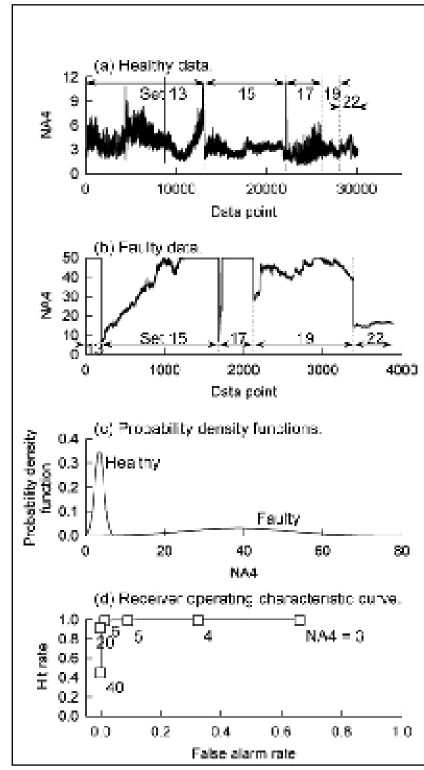

Figure 11—Summary results for NA4 condition indicator for macropitting, single/few teeth failures.

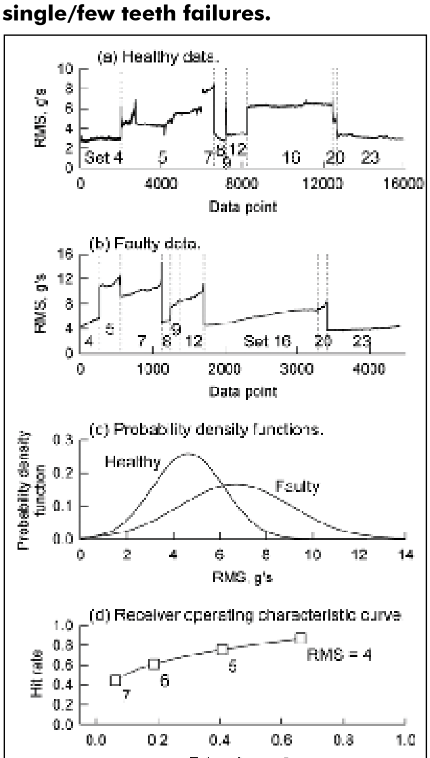

Figure 12—Summary results for RMS condition indicator for macropitting, multiple teeth failures.

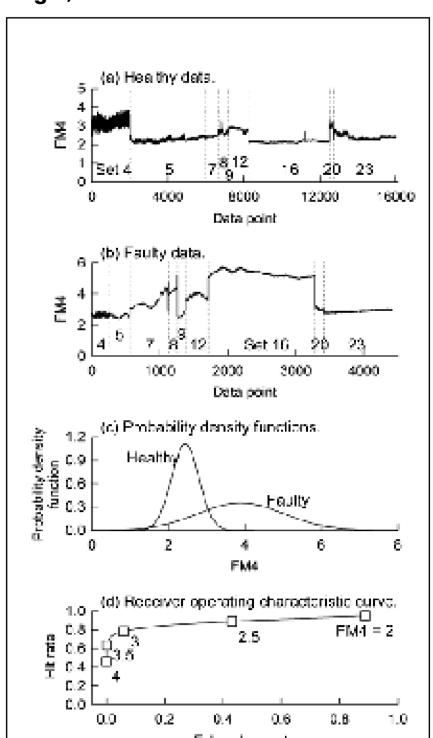

Figure 13—Summary results for FM4 condition indicator for macropitting, multiple teeth failures.

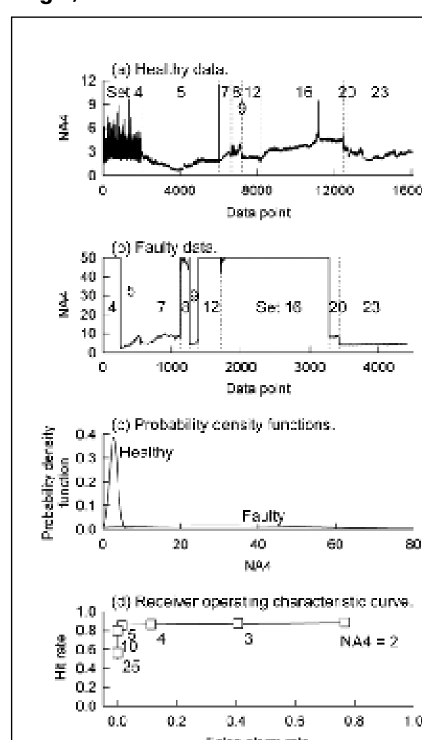

Figure 14—Summary results for NA4 condition indicator for macropitting, multiple teeth failures.

9, 12, 16, 20 and 23). This was due to difficulty in classifying the faulty data regimes for the excluded sets (6, 11 and 21). The CI plots of Appendix B show the groupings of healthy and faulty data that were used for the ROC curve analysis.

Results of the analysis showed that both RMS and FM4 did not show good separation between healthy and faulty data (Figs. 9, 10, 12 and 13). For RMS, significant variation in values from set to set occurred for both healthy and fault data. This increased the standard deviation of the data and thus caused poor separation. The RMS ROC curves were rather smooth, making the threshold less significant due to the poor separation between healthy and faulty data. For RMS from Tables 4 and 5, thresholds of 4.24 and 6.14 g gave hit rates of 0.74 and 0.59 and false-alarm rates of 0.14 and 0.16, indicating rather poor gear-fault detection effectiveness in itself.

For FM4, considerably less scatter occurred but the means between healthy and faulty data were relatively close together. One characteristic of FM4 is the decrease in value with increased fault progression. This lowers the mean for the faulty data and decreases the separation between healthy and faulty data. The FM4 ROC curves showed a slight inflection point at the upper-left portion of the curve. However, the hit rates were rather low. From Tables 4 and 5, FM4 thresholds of 3.29 and 3.04 gave hit rates of 0.61 and 0.77 and false-alarm rates of 0.06 and 0.05. Although the false alarm was low, the hit rate was also rather low, which hurt the gear-fault detection effectiveness of FM4.

The analysis showed that NA4 had very good separation between healthy and faulty data (Figs. 11 and 14). Even though NA4 had a significant amount of scatter (standard deviation), there was an extremely noticeable increase in mean for the faulty data, thus providing good separation. There was a problem, however, with the NA4 analysis. As stated, normal distributions were used in this study. This was a poor choice for the NA4 faulty data. NA4 values significantly increased with fault progression. Even though this increased the mean for the faulty data, it also significantly increased the standard deviation of the fault data. Since normal distributions were used, a symmetry scatter about the mean resulted. This caused artificially induced, lower hit rates. To help alleviate this problem, NA4 values were constrained to a maximum value of 50 in this study. Figure 13d shows hit rates of approximately 0.85 for NA4 values of 5 or less. In actuality, these hit rates approach 1.0. A better choice for the probability density distribution would have been a nonsymmetry distribution, such as a three-parameter Weibull distribution. From Tables 4 and 5, thresholds of 7.14 and 5.52 gave hit rates of 0.99 (correcting the value shown in Table 5) and false-alarm rates less than 0.01. Thus NA4 showed excellent gear-fault detection effectiveness.

Oil debris monitoring. The results from the oil-debris monitoring (ODM) system are given in Figure 15. Data from all 17 failed sets are included. Shown is the calculated cumulative mass per data point (one data point every minute). The ODM responded to all 17 failures. Some sets had definitive inflection points, indicating increased gear tooth pitting (Fig. 15a, set 22 at data point 4,900, as an example). Others had a steady increase in debris (Fig. 15a, set 13). Three sets were outliers with a larger amount of debris (sets 4, 5 and, to some degree, set 22). There did not appear to be significantly more tooth damage (or bearing failures) to correlate with the larger amount of debris, so its cause is unknown. Excluding the three outliers, the results were fairly consistent among sets with an average value of about 40 mg accumulative mass at the end of test.

As stated, there were difficulties in the facility setup with the ODM. A single sensor was used for both the left- and right-test sides. Thus, it was not possible to separate the results per side. This posed two problems. First, the measured results included the debris from both sides. Second, the failure of the opposite-side set during a test of a given set produced a significant

continued

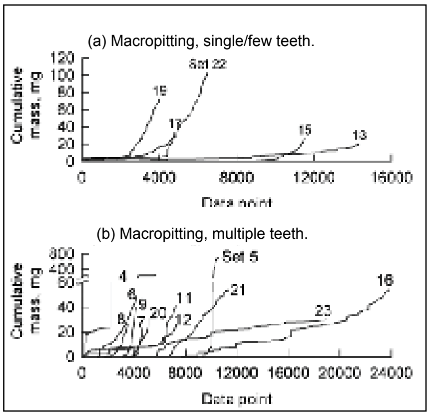

Figure 15—Oil debris monitor results.

TABLE 4—DATA SUMMARY FOR MACROPITTING, SINGLE/FEW TEETH FAILURE MODE (Pinion condition 1 of Table 2.)							
Condition indicator	Healthy		Faulty		Threshold		
	Mean	Std. dev.	Mean	Std. dev.	Value	Hit rate	False rate
RMS	3.39	0.79	4.97	1.14	4.24	0.74	0.14
FM4	2.92	.23	3.50	0.78	3.29	.61	.06
NA4	3.47	1.14	38.46	13.53	7.14	.99	.00

TABLE 5—DATA SUMMARY FOR MACROPITTING, MULTIPLE TEETH FAILURE MODE (Pinion condition 2 of Table 2.)							
Condition indicator	Healthy		Faulty		Threshold		
	Mean	Std. dev.	Mean	Std. dev.	Value	Hit rate	False rate
RMS	4.64	1.54	6.67	2.41	6.14	0.59	0.16
FM4	2.44	0.36	3.89	1.13	3.04	.77	.05
NA4	2.76	1.03	28.45	22.23	5.52	.85[a]	.00

[a] *Artificially low due to normal distribution*

Appendix B: CI Traces

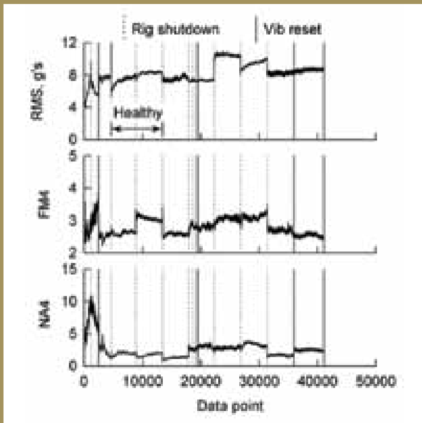

Figure B1—Set 1 vibration fault detection data.

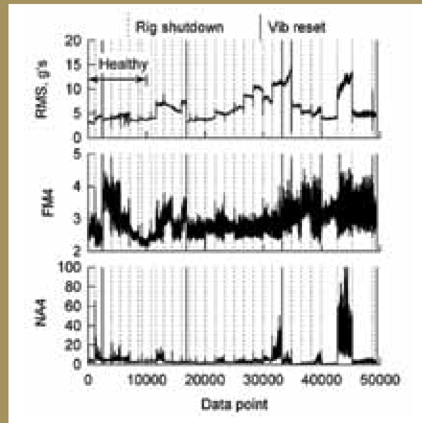

Figure B2—Set 2 vibration fault detection data.

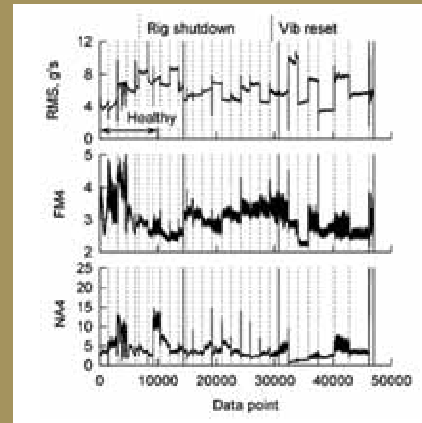

Figure B3—Set 3 vibration fault detection data.

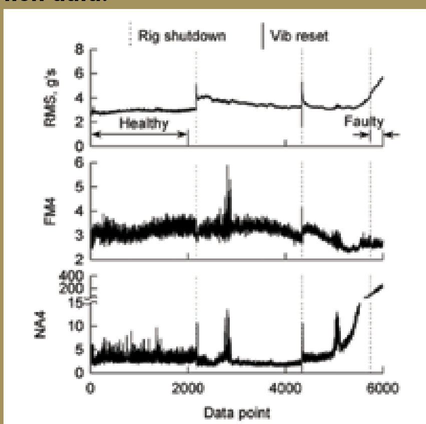

Figure B4—Set 4 vibration fault detection data.

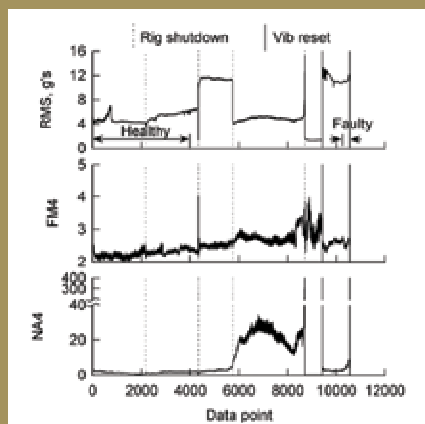

Figure B5—Set 5 vibration fault detection data.

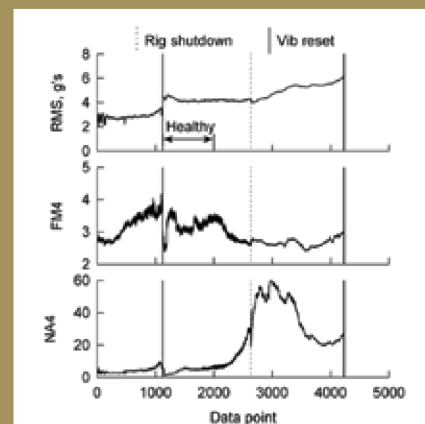

Figure B6—Set 6 vibration fault detection data.

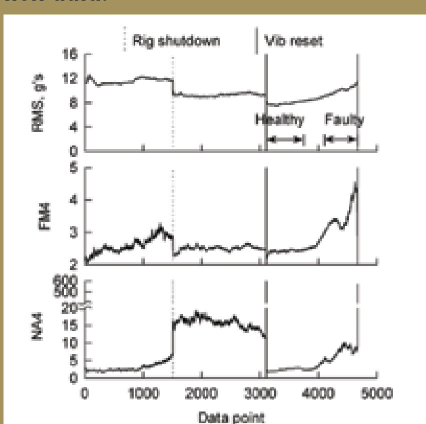

Figure B7—Set 7 vibration fault detection data.

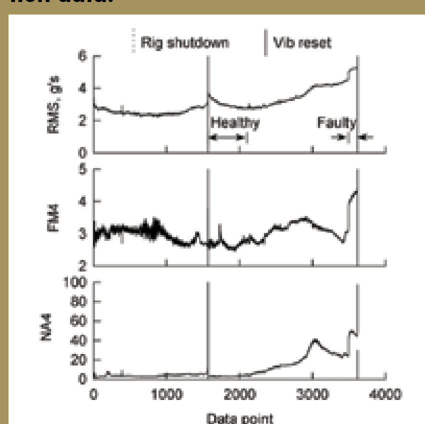

Figure B8—Set 8 vibration fault detection data.

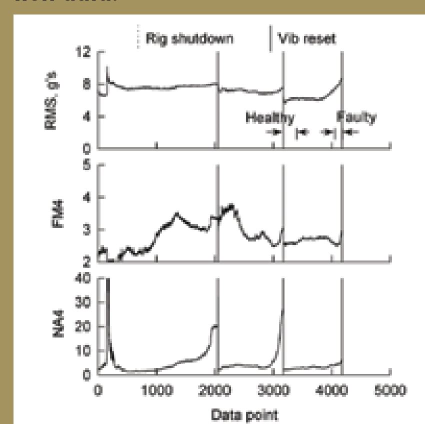

Figure B9—Set 9 vibration fault detection data.

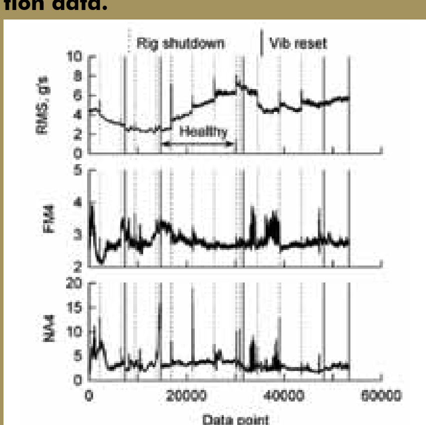

Figure B10—Set 10 vibration fault detection data.

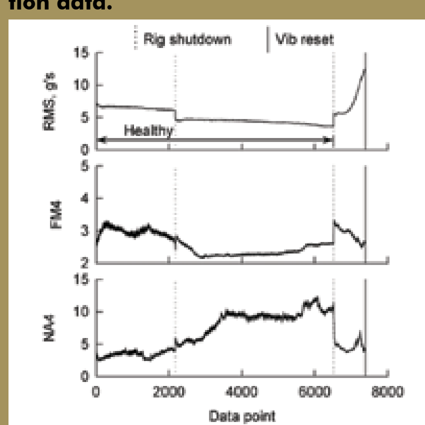

Figure B11—Set 11 vibration fault detection data.

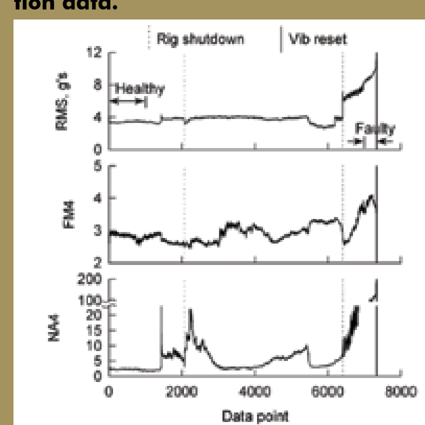

Figure B12—Set 12 vibration fault detection data.

Appendix B: CI Traces

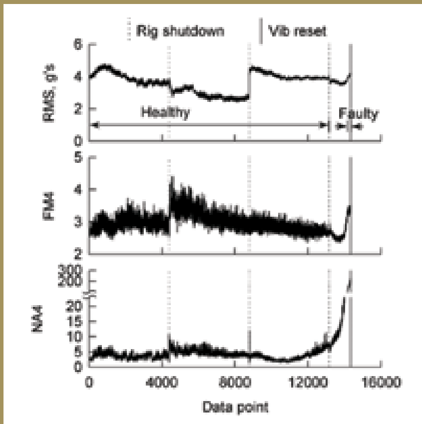

Figure B13—Set 13 vibration fault detection data.

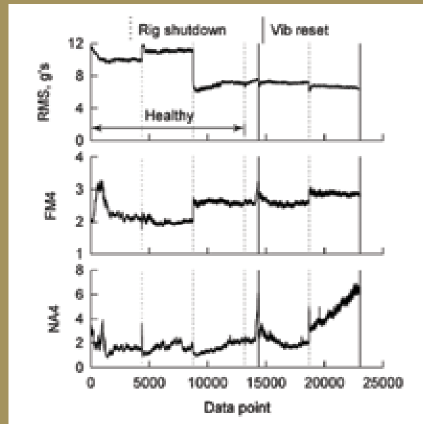

Figure B14—Set 14 vibration fault detection data.

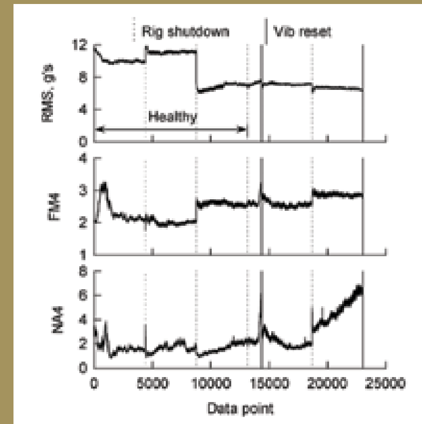

Figure B15—Set 3 vibration fault detection data.

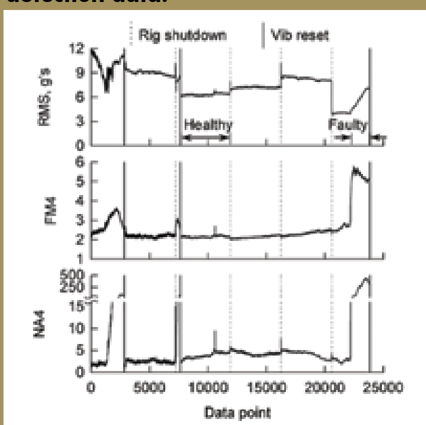

Figure B16—Set 16 vibration fault detection data.

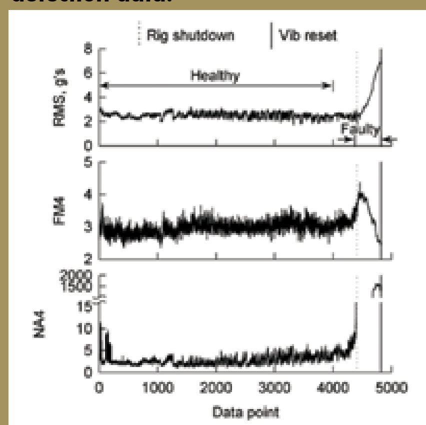

Figure B17—Set 17 vibration fault detection data.

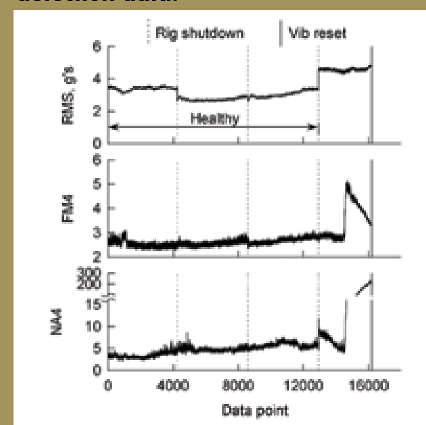

Figure B18—Set 18 vibration fault detection data.

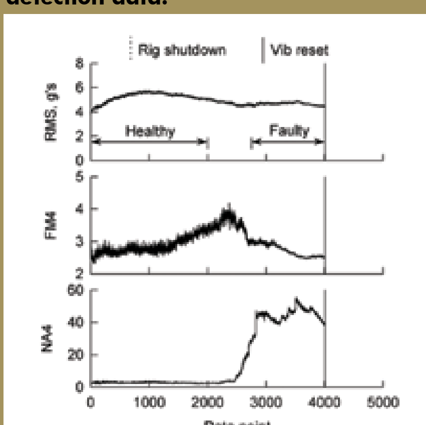

Figure B19—Set 19 vibration fault detection data.

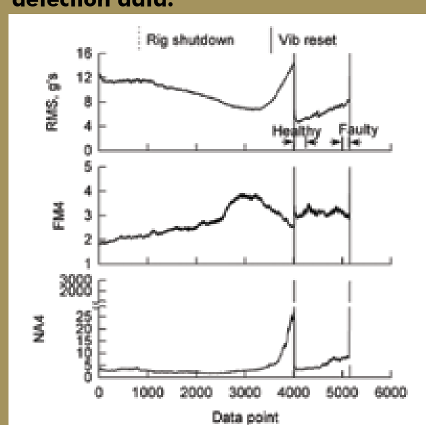

Figure B20—Set 20 vibration fault detection data.

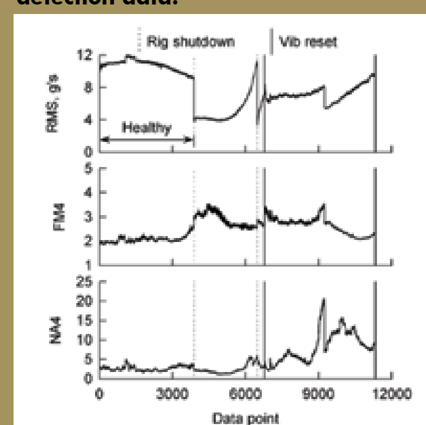

Figure B21—Set 21 vibration fault detection data.

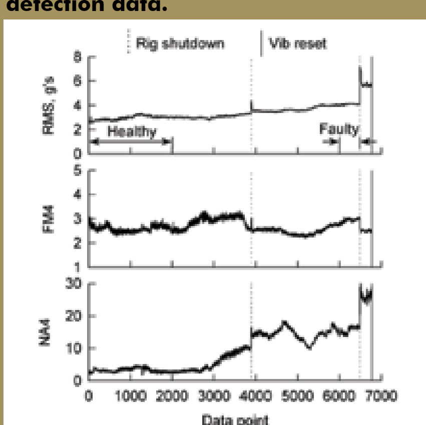

Figure B22—Set 22 vibration fault detection data.

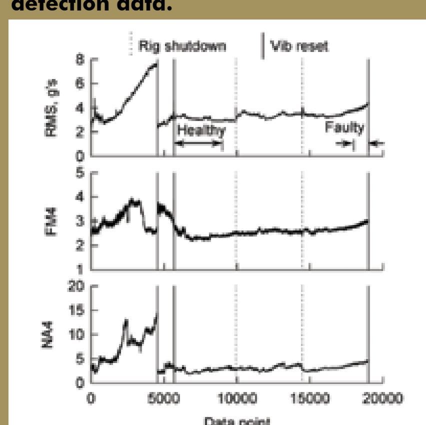

Figure B23—Set 23 vibration fault detection data.

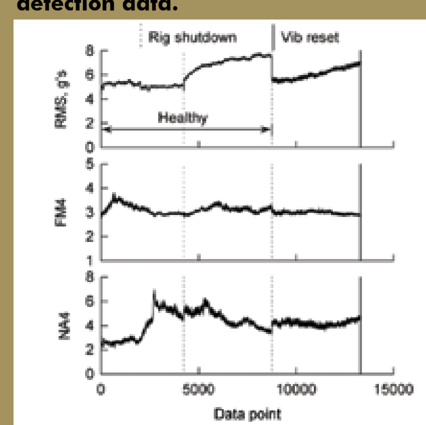

Figure B24—Set 24 vibration fault detection data.

amount of debris. Therefore, the ODM was reset to zero after each failure, thus producing an offset for some sets. Fortunately, no failures occurred at the same time for the left and right sides, leaving enough separation in the results to give meaningful data.

Conclusions

The objective of this study was to evaluate fault-detection effectiveness as applied to gear tooth pitting fatigue damage. Vibration and ODM data were gathered from 24 sets of gears run during an endurance evaluation study. Three common condition indicators (RMS, FM4 and NA4) were deduced from the time-averaged vibration data and used with the ODM to evaluate gear-fault detection. The following conclusions were obtained:

- The NA4 parameter showed to be a very good condition indicator for the detection of gear-tooth-surface pitting failures. Very good separation between healthy and faulty data occurred with NA4.
- The FM4 and RMS parameters performed average to below-average in detection of gear tooth surface pitting failures. FM4 had low scatter in results but had a relatively small separation in mean values of healthy and fault data. For RMS, significant variation in values from set to set occurred.
- The ODM sensor was successful in detecting a significant amount of debris from all gear tooth pitting fatigue failures. Excluding outliers, the average cumulative mass at the end of a test was 40 mg.

Acknowledgement

The U.S. Army Research Laboratory at NASA Glenn Research Center, in collaboration with the Boeing Company in Mesa, Arizona, has conducted this development demonstration in support of the Rotorcraft Drive System for the 21st Century (RDS-21) program. The RDS-21 Program was performed under a Technology Investment Agreement (TIA) between Boeing and the U.S. Army Aviation Applied Technology Directorate. The attention given by Mr. Jason Fetty, the contracting officer's technical representative, U.S. Army Aviation Applied Technology Directorate, and Mr. Stephen C. Slaughter, manager, Advanced Development Programs, the Boeing Company, are greatly appreciated.

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Appendix A: CI Definitions

Root mean square (RMS): The root mean square (RMS) is defined as the square root of the average of the sum of the squares of the time-averaged vibration trace (Eq. 1). For a simple sine wave, the RMS value is approximately 0.707 times the amplitude of the signal.

$$\mathrm{RMS} = \sqrt{\frac{1}{N}\left[\sum_{i=1}^{N} S_i^2\right]} \qquad (1)$$

where:

- S time-averaged vibration trace
- i data point number in vibration trace
- N total number of data points in vibration trace

FM4: The FM4 parameter (Eq. 2) was developed to detect changes in vibration pattern resulting from damage to a single gear tooth (Ref. 1). The metric is calculated by dividing the fourth statistical moment (kurtosis) of the difference signal by the square of the variance of the difference signal. The difference signal is defined as the time-averaged vibration trace—*S*, minus the gear mesh frequencies and shaft orders. The metric is non-dimensional with a nominal value of 3 for Gaussian noise (assumed for a healthy component):

$$\mathrm{FM4} = \frac{\sum_{i=1}^{N}(d_i - \bar{d})^4}{\left[\sum_{i=1}^{N}(d_i - \bar{d})^2\right]^2} \qquad (2)$$

where:

- d difference signal
- $\bar{d}$ mean value of difference signal
- i data point number in difference signal
- N total number of data points in difference signal

NA4: The NA4 metric (Eq. 3) was developed to overcome a shortcoming of the FM4 metric (Ref. 19). As the occurrences of damage progress in both number and severity, FM4 becomes less sensitive to the new damage. Two changes were made to the FM4 metric to develop the NA4 metric as one that is more sensitive to progressing damage. One change is that FM4 is calculated from the difference signal while NA4 is calculated from the residual signal. The residual signal includes the first-order sidebands that were removed from the difference signal. The second change is that trending was incorporated into the NA4 metric. While FM4 is calculated as the ratio of the kurtosis of the data record divided by the square of the variance of the same data record, NA4 is calculated as the ratio of the kurtosis of the data record divided by the square of the average variance. The average variance is the mean value of the variance of all previous data records in the run ensemble. These two changes make the NA4 metric a more sensitive and robust metric. The NA4 metric is calculated by:

$$\mathrm{NA4} = \frac{\sum_{i=1}^{N}(r_i - \bar{r})^4}{\frac{1}{M}\sum_{j=1}^{M}\left[\sum_{i=1}^{N}(r_{ij} - \bar{r}_j)^2\right]^2} \qquad (3)$$

where:

- r residual signal
- $\bar{r}$ mean value of residual signal
- i data point number in residual signal
- N total number of points in residual signal
- j time record number in run ensemble
- M current time record in run ensemble

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GEAR TECHNOLOGY

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The Effect of Straight-Sided Hob Teeth

Richard L. Thoen

Management Summary

It is well known that hobs with straight-sided teeth do not cut true involutes. In this paper, the difference between the straight side of a hob tooth and the axial profile of an involute worm is evaluated. It is shown that the difference increases as the diametral pitch increases, to the extent that for fine-pitch gearing, the difference is insignificant.

Introduction

The fact that hobs with straight-sided teeth do not cut true involutes has been known for a long time. Buckingham showed that the theoretical profile in the axial plane of an involute hob is not a straight line (Ref. 1). And Vogel portrayed an involute hob with curved—not straight-sided—teeth in the axial plane (Ref. 2).

The purpose of this article is to determine the space (gap) between the straight side of a hob tooth and the axial profile of an involute worm.

Basic Geometry

The hob is really a worm with axial cutting gashes. A relatively simple method of computing the axial profile of an involute worm is based on an imaginary, triangular sheet tangent to the base cylinder—such as shown in Figure 1 (and in Ref. 2).

In Figure 2, the vertical line tangent to the base circle is of the same length as a string unwound from the base circle.

The length of the line is $r_b(\mathrm{inv}\phi + \pi/2)$

where, from well-known equations:

$r_b = R\cos\Phi$

$\tan\Phi = \tan\Phi_n / \cos\psi$

$R = N / 2P$

$P = P_n\cos\psi$

And invϕ is obtained from $\cos\phi = r_b/r$, from which the $\mathrm{inv}\phi = \tan\phi - \phi$

where:

r is the radial distance to the involute and axial profiles (Figs. 1–2).

In Figure 1, the plane tangent to the base cylinder is an imaginary, triangular sheet that generates the helical tooth, in

continued

Nomenclature

r_b	Base radius
R	Reference radius (also known as pitch radius)
P	Transverse diametral pitch
P_n	Normal diametral pitch
r	Radial distance to involute and axial profiles
L	Lead
ϕ	Profile angle
Φ	Flank angle in transverse plane
Φ_n	Flank angle in normal plane
ψ	Helix angle at reference radius
ψ_b	Base helix angle
λ	Lead angle

the same way that a point on a string generates a spur tooth.

The base helix angle (ψ_b) of the triangular sheet is found from:

$$\tan\psi = 2\pi R/L \text{ and } \tan\psi_b = 2\pi r_b/L$$

where:

$$L = 2\pi R/\tan\psi = 2\pi r_b/\tan\psi_b$$

where:

$$\tan\psi_b = \tan\psi\ (r_b/R)$$

Figure 1—Involute and axial profiles of an involute worm.

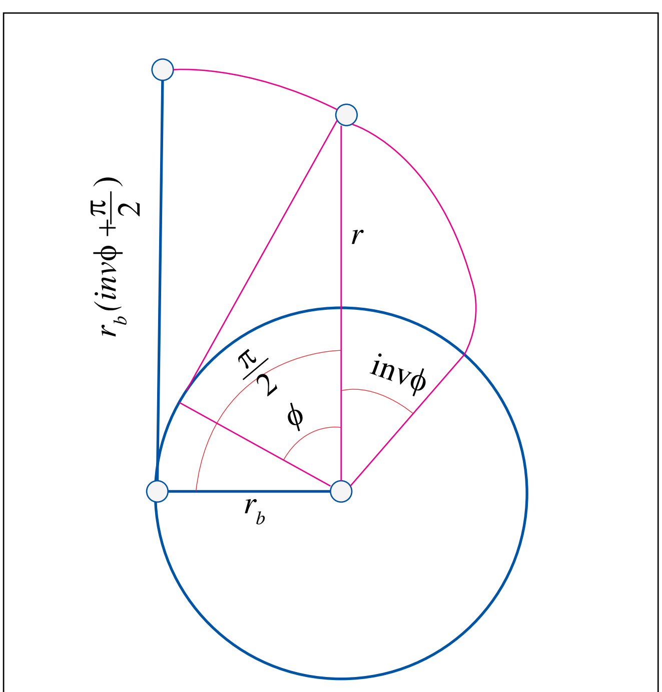

Figure 2—Transverse section of an involute worm.

where

$$r_b\ /\ R = R\cos\Phi\ /\ R = \cos\Phi$$

so,

$$\tan\psi_b = \tan\psi\cos\Phi$$

The horizontal distance (X) from the vertex of the base helix angle (ψ_b) to the transverse plane containing the r value is, from Figures 1 and 2:

$$\tan\psi_b = \frac{r_b\left(\text{inv}\phi + \frac{\pi}{2}\right)}{X} \tag{1}$$

$$X = \frac{r_b\left(\text{inv}\phi + \frac{\pi}{2}\right)}{\tan\psi_b} \tag{2}$$

This equation can be thought of as being that for X versus r in the axial plane.

To determine the gap between the hob and axial profiles for a specified value of r, the straight side of the hob tooth is superimposed on the axial profile at the reference radius ($r = R$), as shown in Figure 3. It should be noted that the hob profile is not tangent to the axial profile.

Numerical Example

It is not necessary to key in calculated values, because they can be stored in and recalled from a pocket calculator, such as the Hewlett Packard hp 33s, which can handle 12 digits.

Given $P_n = 20$, $\Phi_n = 20°$, $\lambda = 3°17'$:

Thus:

λ $= 17'/60 + 3° = 3.283°$

ψ $= 90° - \lambda = 86.71°$

R $= N/2P_n\cos\psi = 1/2•20\cos\psi = 0.4365$

$\tan\Phi$ $= \tan\Phi_n\ /\ \cos\psi$

Φ $= 81.05°$

$\cos\Phi$ $= 0.1554$

r_b $= R\cos\phi = 0.0678$

$\cos\phi$ $= r_b/R = R\cos\Phi/R = \cos\Phi$

$\text{inv}\phi$ $= \tan\Phi - \Phi = 4.940$

$\tan\psi_b$ $= \tan\psi\cos\Phi = 2.709$

$$X = \frac{r_b\left(\text{inv}\phi + \frac{\pi}{2}\right)}{\tan\psi_b} = 0.1630 \tag{3}$$

In Figure 3, for the hob profile:

$$y = 1/P_n = 0.05,$$

or $z = 0.05 \tan 20° = 0.0181$

so,

$$X_R + z = 0.184242$$

For the axial profile:

$r = R + y = 0.4865$

$\cos\phi = r_b/r,$

$\phi = 81.98°$

$\text{inv}\phi = \tan\phi - \phi = 5.669$

$$X_{R+y} = \frac{r_b\left(\text{inv}\phi + \frac{\pi}{2}\right)}{\tan\psi_b} = 0.181295 \qquad (4)$$

$$\begin{array}{rl} & 0.181295 \\ - & 0.181242 = X_R + z \\ \hline \text{Gap} = & 0.000053 \text{ inches} \end{array}$$

Conclusion

The gaps for various hobs are shown in Figure 4, wherein it is seen that there is an excess of material on the hob addendums (lack of material on gear addendums) and a lack of material on the hob dedendums (excess of material on gear addendums).

Figure 4 also shows that the gaps decrease as the diametral pitch increases. Indeed, for an 80-diametral pitch hob with 1°10′ lead angle, the largest gap is only one micro-inch. Conversely, for a 3.6-diametral pitch hob with 4°43′ lead angle, the largest gap is 0.0007 inches.

In the fine-pitch field, therefore, the effect of straight-sided hob teeth is negligible. As Louis Martin, chairman of the AMGA Fine-Pitch Committee from its inception in 1941 until 1953, stated (Ref. 3):

"The glaring mistake that has been made by the gear industry is to try to relate fine-pitch requirements with experience gathered from the coarse-pitch field."

References

1. Buckingham, Earle. *Spur Gears*. McGraw Hill, 1928, pp. 274–375.
2. Vogel, Werner F. *Involutometry and Trigonometry*, Michigan Tool Co., 1948, Figure 26, p. 314.
3. Martin, Louis. Discussion at an AGMA convention, May 30, 1960.

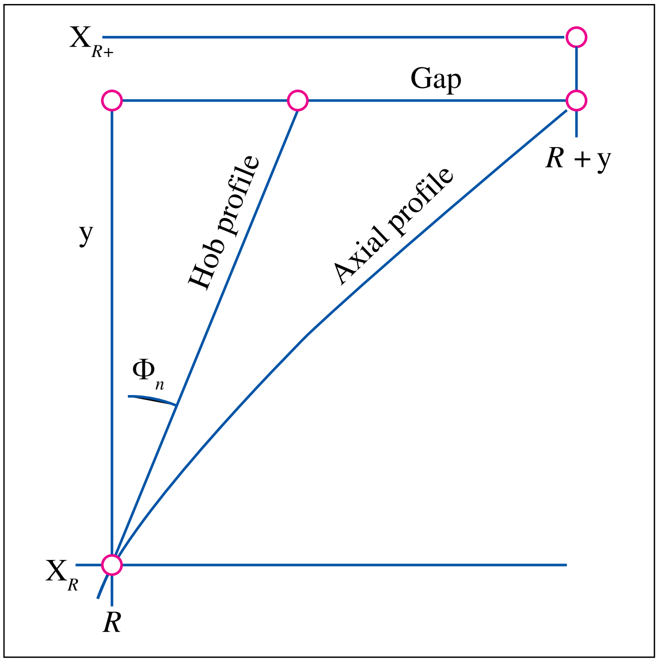

Figure 3—Determination of gap between hob and axial profiles.

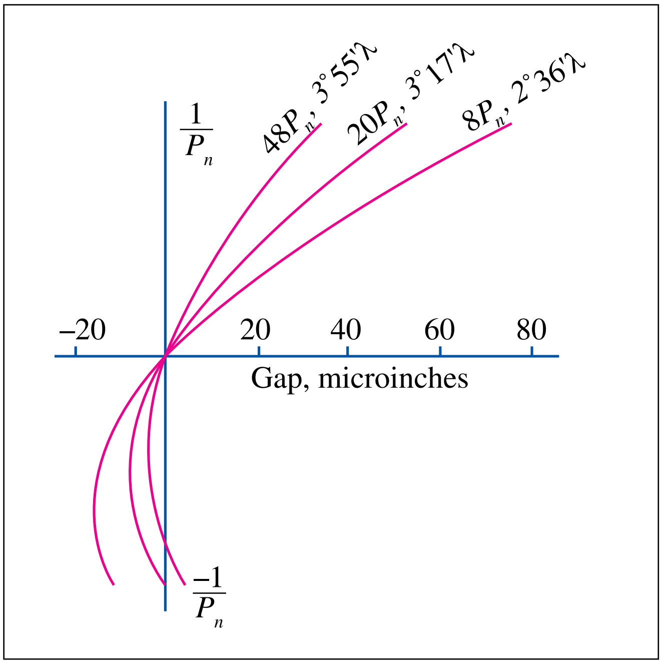

Figure 4—Gaps between hob and axial profiles for various hobs.

Richard L. Thoen *is a consultant specializing in medium- and fine-pitch gearing. He is the author of several articles and papers on measurement, involute mathematics, statistical tolerancing and other gearing subjects.*

11th ASME International

Power Transmission and Gearing Conference

August 28-31, 2011

Call for Papers

The Power Transmission and Gearing (PTG) Committee of the American Society of Mechanical Engineers, Design Engineering Division is pleased to invite you to participate in the 11th ASME International Power Transmission and Gearing Conference. This Conference will be held in conjunction with the ASME International Design Engineering Technical Conferences and Computers and Information in Engineering Conferences (IDETC/CIE 2011) in Washington, DC between August 28 - 31, 2011. Power transmission and gearing researchers and engineers from around the world attend this conference. This makes it an ideal forum for enhancing power transmission and gearing engineering by providing attendees an opportunity to become familiar with the latest research findings and applications that address critical engineering issues.

Authors and attendees are invited to participate in this premier international conference. This is a unique opportunity to disseminate your latest research findings to a global audience, and network with leading experts in the field. Papers are solicited on all aspects of gear and power transmission technology and range of applications (aerospace, automotive, wind turbine, and others).

Topics include but are not limited to:

- Gear System Design and Analysis
- Gear Strength and Durability
- Gear System Dynamics and Noise
- Gear Manufacturing
- Power Loss in Gear Systems
- Surface Engineering and Tribology
- Novel Transmission Concepts
- Components (Chains, belts, bearings, clutches, couplings)
- Plastic Gears
- Wind Turbine Gears
- Transmissions for Hybrid and Electric Vehicles

Deadlines:

Submission of Abstract	Feb. 11, 2011
Full-Length Draft Paper	Feb. 18, 2011
Submission of Final Paper	May 30, 2011

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www.asmeconferences.org/IDETC2011/Author/NewAbstract.cfm

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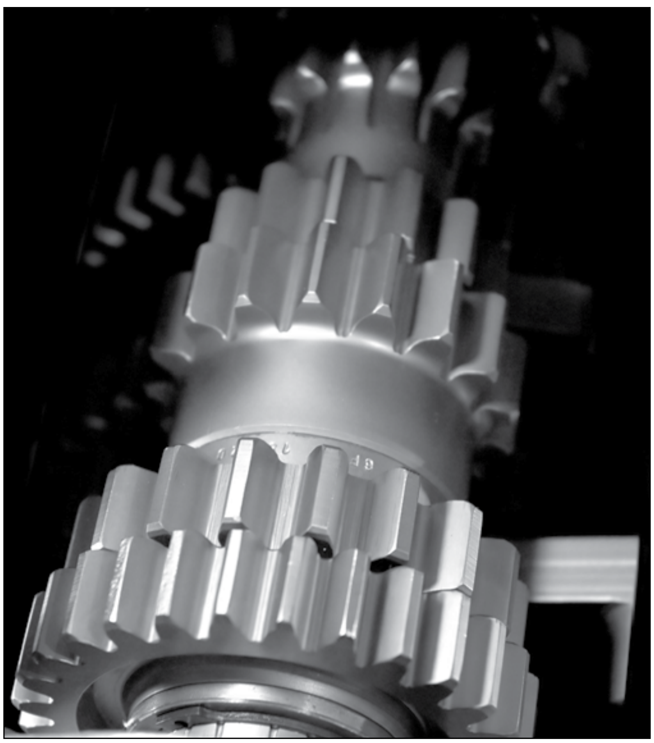
Technically Speaking, A Huge Success

2010 FALL TECHNICAL CONFERENCE



Yes, important and well-deserved awards were presented at AGMA's recent 2010 FTM Annual Awards Luncheon in Milwaukee. Vital committees—Aerospace and Cutting Tools—convened; there were continental breakfasts, networking-friendly receptions and excellent German cuisine. But here's the lead: for almost four days, some of the greatest minds in the gear and gear-related industry met beneath one roof to present and exchange their findings, theories and interpretations regarding the fine art of gearing. From dozens of submitted abstracts, 17 papers were ultimately chosen to represent to peers around the world the depth of knowledge and experience that AGMA members exemplify.

"I was impressed by the quality of presentations," says Dave Ballard, AGMA board chairman and corporate manager for SEW-Eurodrive, Inc. He adds, "Our industry has been known to change with the times; we are seeing some true innovations in design solutions."

Adds AGMA Technical Committee chair Phil Terry, "I was extremely pleased with the attendance; numbers we have not seen since the 1980s, plus the atmosphere of enthusiasm and the large number of new attendees."

Four sessions on topics of particular expertise and relevance were presented: (I) Manufacturing and Heat Treatment; (II) Load Capacity Analysis; (III) Gear Design Considerations; and (IV) Gear Applications.

Presentations addressed a wide variety of in-the-now gear applications—i.e., wind turbine gearbox components, high-contact-ratio gears, tooth engagement and load sharing, reverse engineering and self-locking gears. Of particular significance, the event drew a near-SRO crowd—seats were at a premium for the presentations and awards lunch. "The economy had an effect on our industry, but our registration numbers show us that companies are still very interested in advancing gear science. I was pleased to see almost half the attendees were first-time attendees and the audience was (representative) of a new generation of gear engineers," says Joe Franklin, AGMA president.

Also worth noting is that AGMA went "green" for the meeting, providing an electronic version of all paper presentations to attendees prior to their arrival via hand-held drives. Well-deserved kudos are due the AGMA Technical Committee—especially Phil Terry (Lufkin Industries), Charlie Fischer, Amir Aboutaleb and Abby Lane—for their efforts in working with the authors in guiding them through the process and in coordinating the event.

The 2011 FTM will be held in conjunction with Gear Expo, October 30–November 1, Cincinnati, Ohio. A call for papers for the event was issued in late October. Submission deadline is January 18, 2011.

***Following is a complete list of papers presented at the 2010 FTM, all of which are available electronically from AGMA*:**

Session I—Manufacturing and Heat Treatment

10FTM01—Complete Machining of Gear Blank and Gear Teeth

Author: Dr. Ing. Claus Kobialka (Gleason-Pfauter)

continued

10FTM02—Improving Heat Treating Flexibility for Wind Turbine Gear Systems through Carburizing, Quenching and Material Handling Alternatives
Author: Wallace (Jack) Titus (AFC-Holcroft)

10FTM03—A Novel Approach to the Refurbishment of Wind Turbine Gears
Authors: Mark Michaud and Gary J. Sroka (REM Surface Engineering) and Ronald E. Benson (REM Research Group)

10FTM04—Low-Distortion Heat Treatment of Transmission Components
Authors: Dr. Volker Heuer and Dr. Klaus Loeser (ALD); Donald R. Faron (General Motors); and David Bolton (ALD TT)

Session II – Load Capacity Analysis

10FTM05—Comparison of the AGMA and FEA Calculations of Gears and Gearbox Components Applied in the Environment of a Small Gear Company
Author: Vanyo Kirov (Bucyrus International, Inc.)

10FTM06—Finite Element Analysis of High-Contact-Ratio Gears
Authors: M. Rameshkumar, G. Venkatesan and P. Sivakumar (Combat Vehicles Research and Development Establishment, DRDO)

10FTM07—A New Statistical Model for Predicting Tooth Engagement and Load Sharing in Involute Splines
Authors: Janene Silvers, Carl D. Sorensen and Kenneth W. Chase (Brigham Young University)

10FTM08—Calculation of Load Distribution in Planetary Gears for an Effective Gear Design Process
Authors: Dr. Ing. Tobias Schulze, Dipl. Ing. and Christian Hartmann Gerlach (Drive Concepts GmBH); and Dr. Ing. Berthold Schlecht (Technical University of Dresden)

Session III – Gear Design Considerations

10FTM09—Recommendations for Reverse Engineering
Author: Charles D. Schultz (Beyta Engineering Service and Gear Technology Technical Editor)

10FTM10—Evaluation of Methods for Calculating Effects of Tip Relief on Transmission Error, Noise and Stress in Loaded Spur Gears
Authors: Dr. Mike Fish and D. Palmer (Dontyne Systems, Ltd.)

10FTM11—Point-Surface-Origin (PSO) Macropitting Caused by Geometric Stress Concentration (GSC)
Authors: R. Errichello (GEARTECH and Gear Technology Technical Editor); C. Hewette (Afton Chemical Corporation); and R. Eckert, (Northwest Laboratories, Inc.)

10FTM12—Flank Load-Carrying-Capacity and Power-Loss-Reduction by Minimized Lubrication
Authors: Dr. Bernd-Robert Höhn, Dr. Klaus Michaelis and Dr. Hans-Philipp Otto

10FTM13—Gear Design for Wind Turbine Gearboxes to Avoid Tonal Noise According to ISO/IEC 61400-11
Author: Dipl-Ing. Jörg Litzba (Hansen Transmissions International N.V.)

Session IV – Gear Applications

10FTM14—Analysis and Testing of Gears with Asymmetric Involute Tooth Form and Optimized Fillet Form for Potential Application In Helicopter Main Drives
Authors: Frederick W. Brown, Scott R. Davidson, David B. Hanes and Dale J. Weires (The Boeing Company); and Alex Kapelevich (AK Gears, LLC)

10FTM15—Driveline Analysis for Tooth-Contact-Optimization of High-Power Spiral Bevel Gears
Authors: Jesse Rontu, Gabor Szanti and Eero Mäsä (ATA Gears Ltd., Finland)

10FTM16—Analysis of Load Distribution in Planet-Gear Bearings
Authors: Louis Mignot, Loïc Bonnard and Vincent Abousleiman (Hispano-Suiza)

10FTM17—Self-Locking Gears: Design and Potential Applications
Authors: Alex Kapelevich (AKGears LLC) and Elias Taye (ET Analytical Engineering, LLC)

And the 2010 FTM Awards Go To….

2010 AGMA FTM awards were presented to the following for their contributions to AGMA and the worldwide gearing industry:

Chairman's Awards (for chairmen of technical committees who have released standards in the past year) were presented to:

Robert Wasilewski (Arrow Gear Company)

Todd Praneis (Cotta Transmission Company, LLC)

TDEC (Technical Division Executive Committee) Award:

Richard (Dick) Calvert (Chalmers & Kubeck)

November 29–December 2—Defense Manufacturing Conference. The Venetian, Las Vegas. The Defense Manufacturing Conference (DMC) 2010 is the nation's largest forum for scientists, technologists, engineers, managers, leaders and policy makers in the defense manufacturing industrial base. The meeting is the product of a partnership of the U.S. military departments and agencies of the Department of Defense. The 2010 DMC is hosted by the Office of the Secretary of Defense and the Missile Defense Agency and will address the conference theme of removing the barriers to "Achieving an Innovative, Responsive and Sustainable National Security Industrial Base." All of the military departments, defense agencies, many non-defense agencies, industry and academia will participate in this seminal national event for both defense and non-defense related manufacturing technology development and applications. For more information, visit *www.dmc2010.com.*

November 30–December 1—Innovative Automotive Transmissions and Drivetrains. Berlin. The International CTI Symposium and its specialist exhibition, Transmission Expo, is a European event for people seeking information on the latest technical developments in automotive transmissions and drivetrains. For more information, visit *www.getriebe-symposium.de*.

December 7–9—AWEA Small and Community Wind Conference. Portland, Oregon. More than 100 exhibitors will have the latest wind technology to show consumers, renewable energy professionals, dealers and installers how best to capitalize on wind technology for homes, farms and ranches, businesses and rural electric cooperatives. The small and community wind markets have matured rapidly, according to the AWEA, and have firmly arrived as major segments of the overall wind industry. Now getting wider use as a real alternative to typical energy sources, small wind technology is an option for the consumer that is eco-conscious. In addition, community wind development has proven itself to municipalities, schools, universities and other groups willing to band together to produce their own energy. The Small and Community Wind Conference & Exhibition will feature an exhibit hall designed to accommodate both business attendees and consumers, allowing you as an exhibiting company to get information to industry professionals, and potential customers about your latest wind solutions, products and services. For more information, visit *www.awea.org*.

December 7–9—Gearbox CSI–Forensic Analysis of Gear and Bearing Failures. This seminar helps gear designers gain a better understanding of various types of gears and bearings. Attendees learn about the limitations and capabilities of rolling element bearings and the gears that they support in order to properly apply the best gear-bearing combination to any gearbox, simple or complex. A certificate will be awarded upon completion of the seminar. For more information, visit *www.agma.org/events-training/detail/gearbox-csi1/.*

December 15–18—MDA India 2010. Mumbai, India. MDA is an international trade fair for motion, drives and automation, as well as hydraulics and pneumatics. MDA India is a showcase for the latest trends and technologies coming in from across the globe. The premier of MDA India in 2007 featured 222 exhibitors from 18 countries with a show size of 7,000 square meters. In 2009 the shows were moved to Mumbai, where they saw more than 400 exhibitors from 18 countries on a display area spanning over 18,000 square meters with official country pavilions from Germany, Italy, Spain, Taiwan, China and U.K. showcasing the latest technologies. The last show attracted 8,485 visitors. For more information, visit *www.mda-india.com.*

February 1–2—Offshore Wind Power 2011. Boston. Following the success of this year's Offshore Wind Power USA event in Philadelphia, Green Power Conferences would like to announce the 2nd annual event taking place in February 2011. Offshore wind energy is the new growth industry in North America, with wind resources off the U.S. coasts offering a vast, yet largely untapped energy potential. Offshore development promises to be a significant domestic renewable energy source, especially for coastal energy loads with limited access to interstate grid transmission. For more information, visit *www2.greenpowerconferences.co.uk*.

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AGMA

MOVES FORWARD WITH SPLINE TECHNICAL COMMITTEE

Following a long line of AGMA product and specialty standards, splines are finally getting an opportunity to join the discussion. In September 2010, AGMA hosted an organizational teleconference for the formation of a new technical committee on splines. AGMA staff members Charlie Fischer, Amy Lane and Amir Aboutaleb helped get the committee off the ground.

Aboutaleb, AGMA staff engineer, says more than 30 members attended the online meeting to discuss the proposal. The committee was formed "to address the existing lack of a comprehensive set of standards on design, rating, accuracy, application, inspection and maintenance of splines," Aboutaleb says.

The need for a spline committee was based on numerous inquiries received by AGMA headquarters. This need was echoed by some of AGMA's existing committees such as the Vehicle Gearing, Flexible Couplings, Cutting Tools and others through a request submitted to the Technical Division Executive Committee (TDEC). The request was presented to the TDEC at their last meeting in May and approved unanimously.

"In general, the committee will try to 'fill in the gap' so to speak," Aboutaleb says. "In their initial meeting, the participants decided to distribute a survey to determine what is being used in terms of spline standards and what is missing and remains to be developed.

Aboutaleb says the first teleconference was very encouraging based on the number of participants that gathered online to discuss the committee formation. "Due to the logistics of having 30 people on teleconference, there was no in-depth discussion in September. However, based on a quick survey of participants, we expect 20–25 people to attend the face-to-face meeting in Chicago."

The attendees of the September online meeting included representatives from automotive, aerospace, mining, couplings and other areas of the gear industry. The topics that struck the most interest included spline types, rating and applications. In addition, there was initial discussion of a possible information sheet on terms and nomenclature for splines.

At the November meeting, the Spline Technical Committee plans to iron out all the initial proposals and move forward with its agenda. Aboutaleb says that positions within the committee will be also be determined.

Those working with splines in the gear industry would benefit from getting involved in the committee and sharing experiences. As is the case with all AGMA committees, the full-time AGMA staff discusses committee work with potential new members and during visits with existing company members. Committee participation exposes members to new technologies, technical standards and education they may not receive on a day-to-day basis. Those that share an interest in specialty areas such as splines should consider becoming a member by filling out the online form at *www.agma.org*.

Brelie Gear

EXPANDS CAPACITY, INVESTS IN NEW MACHINERY

Brelie Gear Co, Inc. recently announced the acquisition of a new Mitsubishi GE-20 CNC gear hobbing machine. According to company president Steve Janke, this acquisition was made in part as a commitment to Brelie Gear's continual improvement in quality. Additionally the improvements in efficiency over the Mitsubishi machine it is replacing will help to meet the production requirements of their growing customer list.

Earlier this year Brelie Gear purchased the assets of

continued

Blahnik Manufacturing. The acquisition included purchasing multiple 10" chuck turning centers, a vertical machining center, as well as a horizontal machining center. Key personnel were also retained as a part of the acquisition. "The addition of this equipment and retaining key personnel will allow Brelie Gear to complete the gear manufacturing process for many of our customers from start to finish," says Janke. "Producing a high quality gear requires more than just cutting high quality teeth. Bringing these machining capabilities in-house helps us to control our costs and stay competitive in the marketplace, which helps our customers maintain their competitiveness."

Brelie Gear plant manager Tom Wagner discussed the equipment Brelie Gear has acquired since 2006:

- Koepfer Model 200 CNC gear hobber with automatic loading and unloading (purchased in 2006).
- Ty Miles MBLD -10-36-120 broaching machine (purchased in 2006).
- Koepfer MZ130 CNC 5000 RPM gear hobbing/worm milling machine with automatic loading and unloading (purchased in 2008).
- Koepfer Model 200 CNC gear hobber with skiving attachment and automatic loading and unloading (purchased in 2008).
- Mitsubishi GE-20 CNC gear hobber. This allows the company to run an 8" diameter gear as coarse as 6 DP (purchased in 2010).
- Koepfer Model 200 CNC gear hobber with skiving attachment, automatic loading and unloading, as

well as a triple distributor system that allows for additional part capacity for higher volume jobs (delivery due Spring of 2011).

Ronson Gear

ANNOUNCES STAFF MOVES

Stephen Bell (left) takes on a key role in internal sales/estimating while Darren Snow (right) has been promoted to operations manager.

In preparation for anticipated accelerated growth over the next few years, Ronson Gears has announced a change in management to better service new and existing customers both in Australia and overseas. Stephen Bell moves to a key role in internal sales/estimating after eight years as production manager. Darren Snow has been promoted to the new role of operations manager. Elwin Drummond is

continued

now the quality manager while Jason Graham has been promoted to technical manager. Gavin New will continue in his marketing and sales role and will also begin to undertake some administrative responsibilities. According to managing director Gordon New, "It's the company's people who are responsible for Ronson's long-standing reputation for outstanding quality and customer service. Our long-term relationship with customers in a range of industries including mining, aerospace, rail, agriculture, defense and automotive is important to us, and these positional changes will enhance those relationships."

Drake

SHIPS GAGE GRINDING MACHINE TO CHINA

Drake Manufacturing Services Co. recently shipped another in a series of high precision gage grinding machines to one of China's emerging API gage manufacturers. A precision tool manufacturer in central China is in the process of installing its second Drake internal gage grinding machine. The machine is a GS:TI 540 (Grinding System: Thread Internal) and will supplement and expand the customer's capacity to grind internal thread gages from 10 mm to 500 mm internal diameter threads. This company previously purchased a Drake GS:TI-LM 350 machine in 2005. It manufactures a full line of thread gages, taps, measurement equipment and other threading tools.

In the past two years, Baoshan Iron and Steel and Chengdu Chengliang Precision Tools have also expanded their gage grinding capabilities with the addition of Drake thread CNC internal and universal thread grinders.

Baoshan purchased a GS: TE/I-LM 650 universal grinder capable of grinding external threads up to 650 mm and internal threads up to 550 mm diameter. Baosteel produces American Petroleum Institute (API) thread gages used to check the integrity of threads on steel tubing and casing used in drilling for and transporting petroleum and natural gas.

Romax and Hansen

SIGN AGREEMENT FOR WIND TURBINE SERVICES

Romax Technology, the drivetrain and gearbox solutions specialist, and Hansen Transmissions International NV, the wind turbine gearbox designer and manufacturer, have signed an agreement to combine their expertise to wind turbine manufacturers, primarily focused on the Chinese and South Korean markets. Romax and Hansen will work together to deliver to certain Chinese and South Korean wind turbine manufacturers a complete and technically advanced service for the design, development and supply of state-of-the-art gearboxes for multi MW wind turbines. As quality, time-to-market and cost are the cornerstones of the wind turbine industry going forward, this collaboration between Romax and Hansen aims to meet the requirements of the wind turbine manufacturers. It will include the design, certification and field validation of their wind turbine drivetrains as well as serial supply of the main components

continued

through a one-stop service.

The cooperation between Romax and Hansen aims to combine a shorter time-to-market, with high-end standards, plus serial testing and supply. The serial supply can be provided from Hansen's existing production facilities in Asia, including its state-of-the-art assembly and test facility in Tianjin, China. The collaborative agreement is non-exclusive and allows both companies to continue to work independently.

"The wind power industry will continue to experience significant growth over the coming years," says Luc De Proost, chief business development officer at Hansen. "Our partnership with Romax is a clear commitment to strengthening our resources, increasing and enhancing our offering, and targeting best in class time-to-market in the Asian region."

Dr. Stefan Lammens, chief technology officer at Hansen, adds, "We are excited by the prospect of developing this complementary relationship with Romax, as it enables Hansen to strengthen our position as one of the leading designers and manufacturers of high quality gear units whilst at the same time bringing our development activity closer to our Asian customers."

Dr. Peter Poon, chief executive officer of Romax, commented: "It is our belief that combining the complementary capabilities and expertise of Hansen and Romax will drive the technical advancement and effective supply of gearboxes to the wind energy industry. At the same time, this is a step forward in the rapid global deployment of next generation wind turbines."

MTConnect Institute

PROPOSES LEGACY TOOL GROUP

The MTConnect Institute recently proposed the establishment of the Legacy Machine Tool Connectivity Working Group (WG). This group will be essential in addressing the very important issue of providing best practices and overall guidance for the physical connectivity of the thousands of legacy machine tools in manufacturing shops around the globe. The group will be led by David McPhail, president/CEO of Memex Automation Inc., and John Turner, director of technology for FA Consulting and Technology, as the co-chairs and consist of manufacturing equipment providers, ISVs, consultants and users.

Dave Edstrom, president and chairman of the board for the MTConnect Institute, stated, "This proposed working group is critical for enabling MTConnect on the countless number of manufacturing's legacy machine tools and I fully expect our MTConnect Technical Advisory Group to approve this new working group. We want to see an array of hardware options to enable MTConnect for both small and large installations."

"Bringing the capabilities of MTConnect to existing machine tools will allow manufacturers to increase productivity of their existing assets, driving down costs and increasing plant output. I am excited to co-chair the MTConnect Legacy Machine Tool Working Group to help bring the benefits of MTConnect to the existing installed base of machine tools," Turner says.

SME and Tooling University LLC

SIGN AGREEMENT

The Society of Manufacturing Engineers (SME), based in Dearborn, MI, and Tooling University LLC (Tooling U), based in Cleveland, recently announced they have signed a definitive agreement for SME to acquire all outstanding shares of Tooling U. The move is an important step in SME's strategy to address the global and growing need for skilled labor.

The substantial resources of SME's certification products, in-person training and webinar offerings will be combined with Tooling U's online training platform and more than 400 courses to provide the global manufacturing community with a comprehensive source of manufacturing knowledge. Educating the current and future manufacturing

continued

NEWS

workforce is critical for the health and growth of the manufacturing industry.

"By 2012, it is estimated the United States alone will be short three million skilled workers," says Barbara M. Fossum, Ph.D., FSME, and president of SME. "Acquiring Tooling U is part of a new initiative that will enable us to offer comprehensive learning and workforce development solutions to help companies combat this increasing talent shortage, and to provide a much broader span of continuing education opportunities for member career growth."

Tooling U's online training complements SME's highly regarded in-person training and certification programs to create a flexible, on-demand, customized and content-rich blended learning solution. "As the world of knowledge delivery continues to evolve, we will leverage the expertise and wealth of knowledge among our members to accelerate the development of new learning programs and meet the changing needs of the marketplace," Fossum says.

The full range of SME professional development resources is designed to benefit educational institutions such as high schools, community colleges, tech schools and universities, as well as practitioners and companies. SME will fill the gap for individuals seeking advancement in their careers in many areas of a manufacturing organization through continuing education, including in manufacturing management and leadership, according to Mark Tomlinson, SME executive director and general manager.

Following the acquisition, SME's Tooling U will continue to operate without interruption, providing online education with its existing staff from its base in Cleveland. Going forward, SME will invest in additional online courses and a more interactive interface to support all levels of employees from the machinist and shop-floor worker to manufacturing engineers and management personnel.

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GEAR MANUFACTURING

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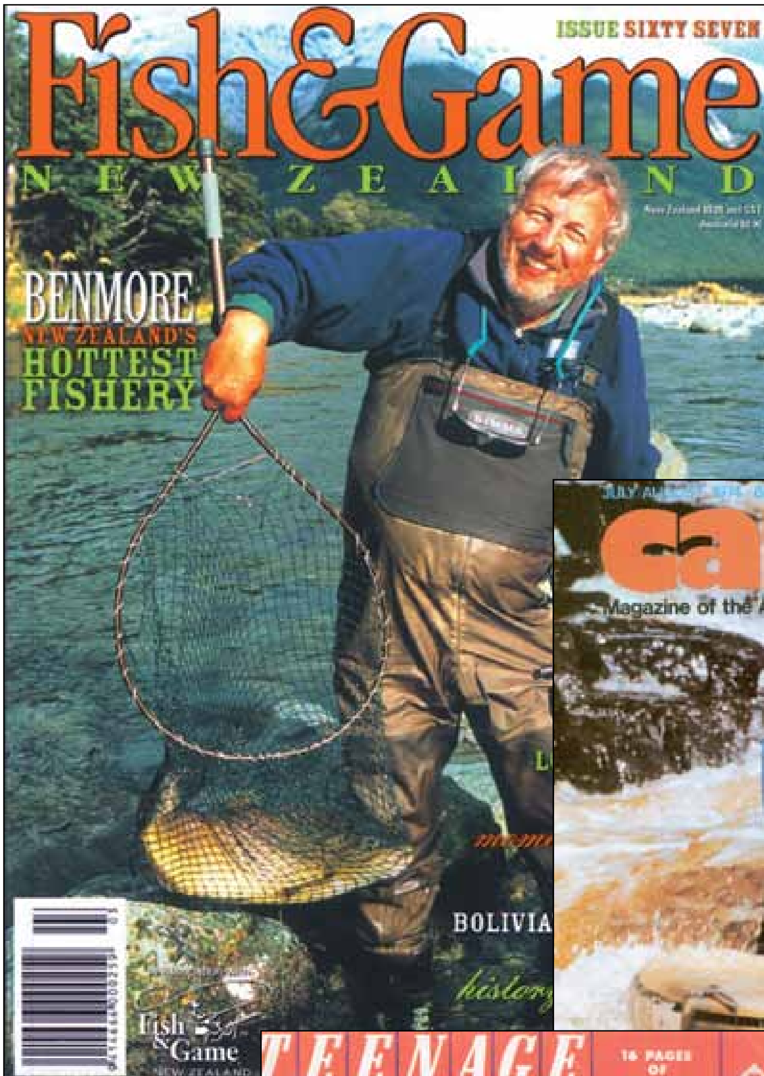

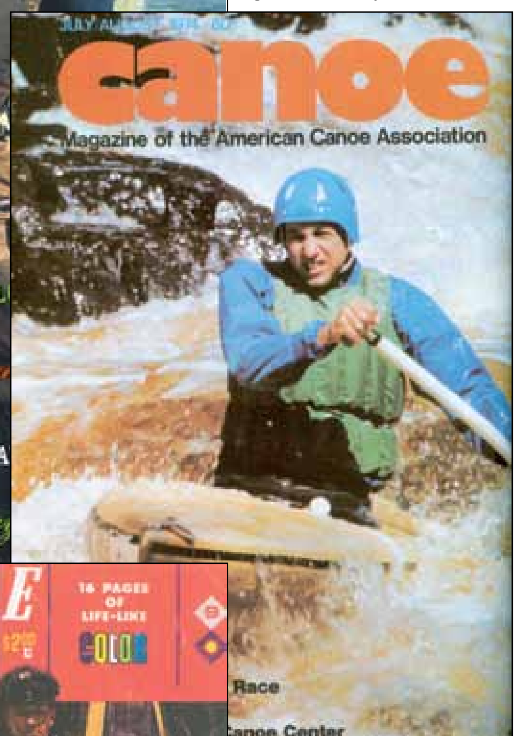

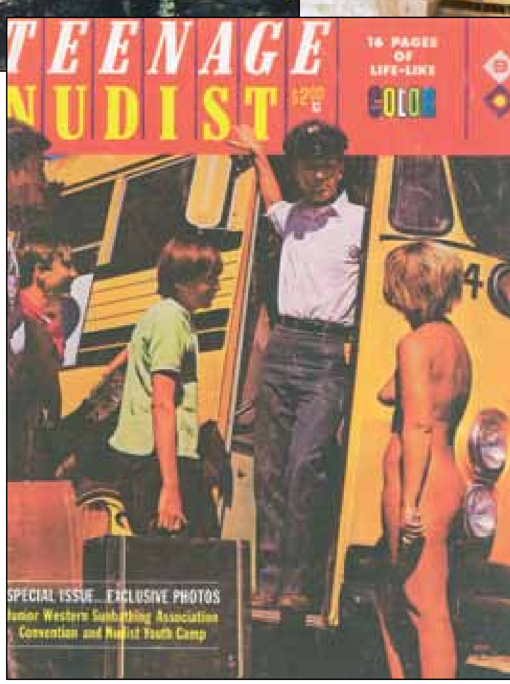

It has come to the attention of the Addendum staff that various people in our industry have been dabbling in pursuits outside of gearing. At first, we were quite upset by the thought that there are gear people out there who are involved in other activities. But our therapists have assured us that this is O.K. Not everyone can spend all of their free time pondering the beauty of the involute curve or contemplating the perfect gear ratio.

You can imagine our surprise when we learned that certain prominent members of the gear industry have not only been dabbling, but they've been caught on film. The papparazzi have captured these gear industry leaders and plastered their pictures on the covers of magazines—with nary a gear in sight. (And we thought Addendum staffers were the only ones who stalked gear industry leaders).

Fred Young of Forest City Gear was recently featured on the cover of *Fish & Game New Zealand* magazine. Looking into it further, we also learned this wasn't Fred's first cover shot. We dug up an old copy of *Canoe* magazine from July/August 1974, where Fred was also featured.

And then there's Herman Pfauter. You recently read in these pages about Herman's younger days, when he spent some time in California and worked as a bus driver. Here we present the cover of *Teenage Nudist* magazine, where Herman was caught in action.

We know it's tempting to condemn these men for their pursuits outside the gear industry. But we've given it a lot of thought, and there's a logical explanation. When Fred Young was captured on the cover of *Fish & Game*, we're now convinced that he was secretly researching the action for a new geared fly reel.

And after further examining the Herman Pfauter cover, we're now quite certain that Herman was, in fact, pondering the beauty of an involute curve.

If you or any other gear industry professional have been "caught in the act" and featured on another magazine cover, please scan it in and send us a copy at *addendum@geartechnology.com*. We're confident that we can come up with a logical explanation for your non-gear-industry activities.