GEAR TECHNOLOGY

JULY/AUGUST 2003

The Journal of Gear Manufacturing

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GEAR EXPO PRE-SHOW ISSUE

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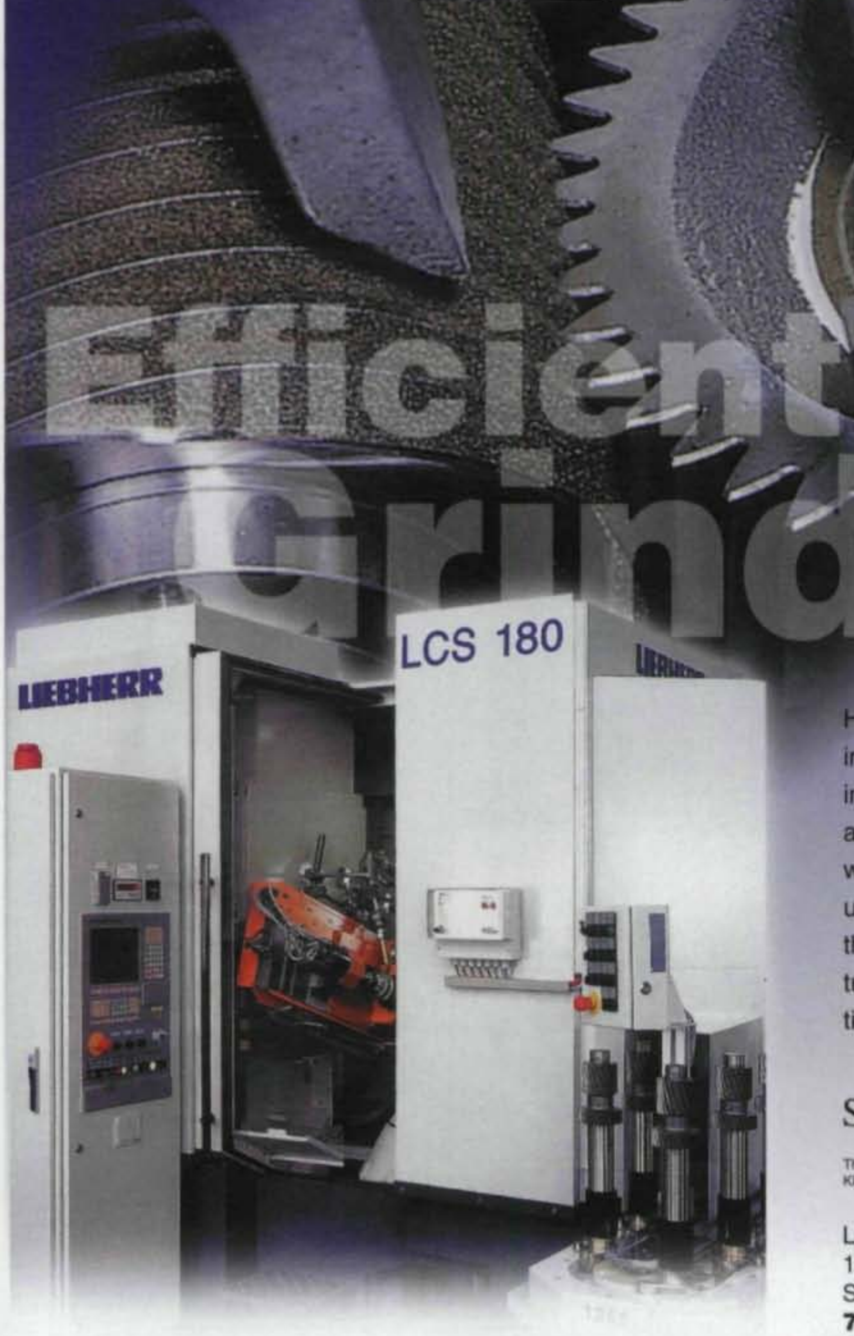
Cover photo courtesy of The Timken Co. Canton, OH

Generating and form grinding, with dressable or CBN tools, in one machine



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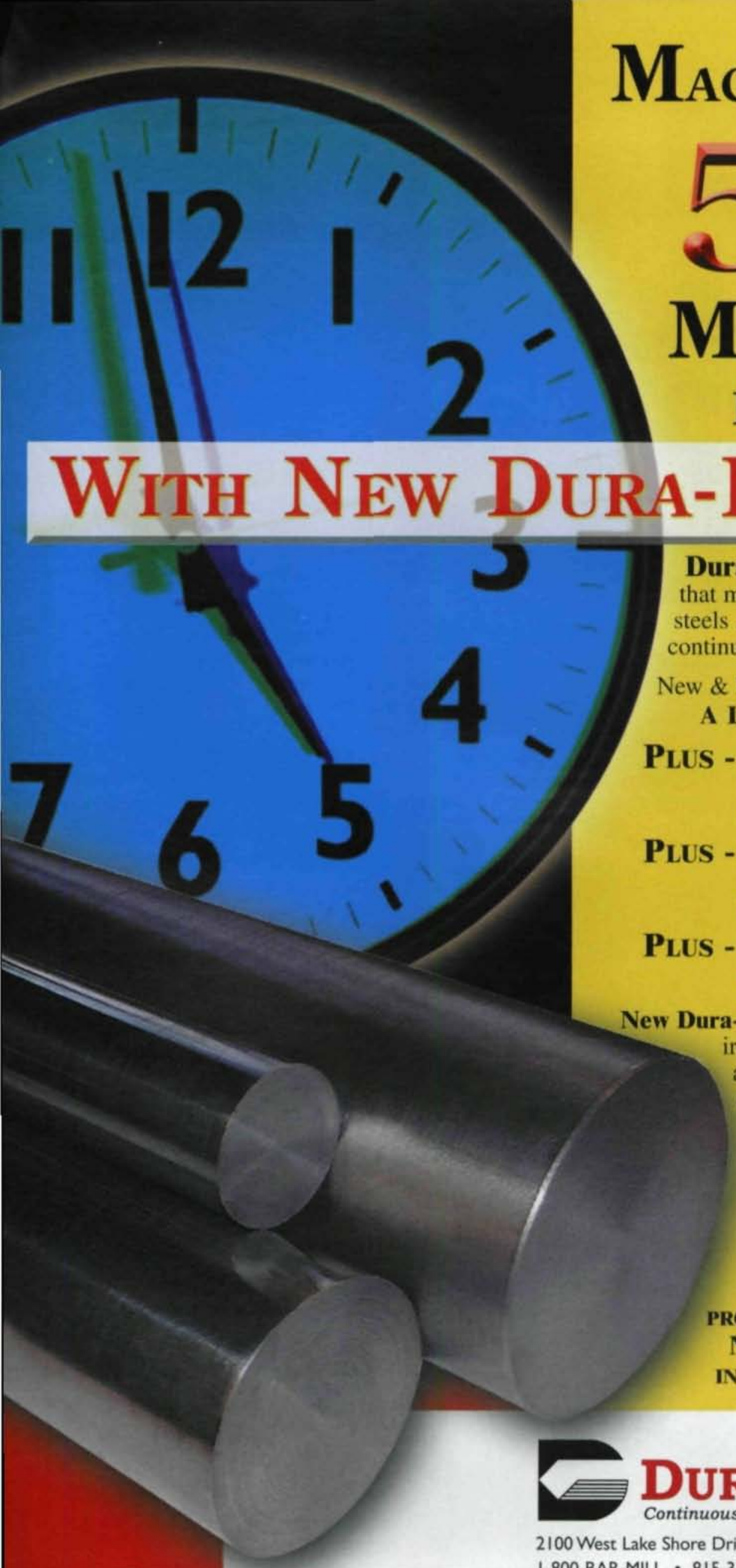

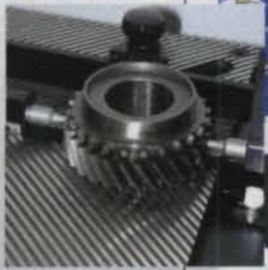

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Shades of Gray

In America and most parts of the world, people are looking for answers about what's going to happen next in the manufacturing economy. We're all looking for evidence that better times are ahead, or at least that the worst is over. We crave a clear indicator, something that shows us in black and white that the situation is going to get better.

Unfortunately, there don't seem to be any clear-cut signs. Instead of a black-and-white indication of where we're headed, all we get are many shades of gray.

In some cases, the news appears promising. For example, for most of this year, the Purchasing Managers Index has hovered around 50, the equilibrium point between growth and contraction. In January and February, the index was above 50, indicating growth in the economy.

But in other cases, the news is not so good. Industrial capacity utilization in the United States is at its lowest point in more than 20 years. Unemployment remains high. Machine tool consumption continues to decline.

Yet I continue to hear from people in industry that they're getting some activity in the form of quotes on jobs—not as good as sales, but activity nonetheless.

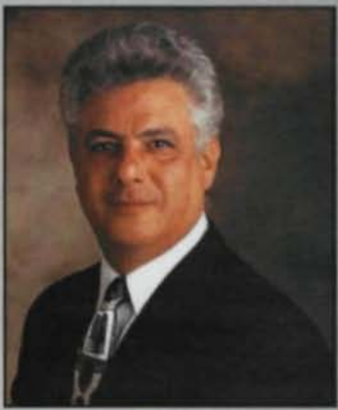
There's a lot of information out there, and sometimes it's hard to figure out which bits are important and which bits can be ignored. Often the statistics we see and hear—good news or bad—are completely irrelevant to our companies' business. Other bits of information may be extremely important.

This issue we're going to provide you with information we've been collecting over the past year that may shed some light on the state of the gear industry. For some time, we've been tracking the level of activity on our websites. I've always found that information to be both interesting and helpful. In this issue, we've prepared several charts detailing those statistics on page 27, and we offer that data for your interpretation and analysis.

The charts include traffic data to three websites, each with a slightly different audience, but all related to the gear industry.

powertransmission.com™ is home to our directory of power transmission components, including gears, bearings, motors and speed reducers. Activity on that website should be an indicator of interest in the industry's end products. *The Gear Industry Home Page*™ is home to our directory of suppliers to the gear industry, including machine tool and cutting tool manufacturers and suppliers of services, such as heat treating and consulting. Activity on that website indicates, in part, interest in planned capital equipment and consumable supplies and services. Finally, we've included statistics from the website of Cadillac Machinery Co., Inc., my own used gear machinery business. The used machinery market is more of a spot market, indicating the need for machines today or in the near future.

The most interesting thing about these statistics is that traffic has increased substantially on all three websites over the past six months. Each of them has moved to a new level compared with a year ago.



Traffic on *powertransmission.com*™ has risen, with page views steadily increasing over the last four months, reaching a high of 162,000 in April. Early in 2002, user sessions were about 30,000 per month. Now they are in the 35,000–40,000 range. More people than ever before are using the website to find suppliers of gears, bearings, motors, speed reducers and other power transmission products.

During most of 2002, traffic on *The Gear Industry Home Page*™ was flat, with user sessions in the 17,000–20,000 per month range and page views in the 65,000–70,000 range. Over the past four months, the numbers have gone up markedly, with highs of 116,000 page views in March and nearly 26,000 user sessions in April.

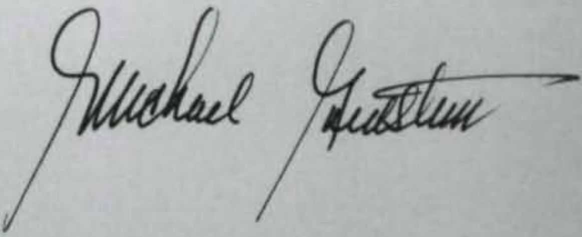
Cadillac's website has seen perhaps the most steady and significant growth. Although the overall numbers are much smaller, traffic has nearly doubled over the past year.

A part of the growth in traffic on these websites is due to the effectiveness of our marketing efforts. The monthly increase of visitors tells us that we're doing a good job getting the word out.

Also, part of the growth probably comes from increasing use of the Internet as a tool for researching and buying industrial equipment, components and services.

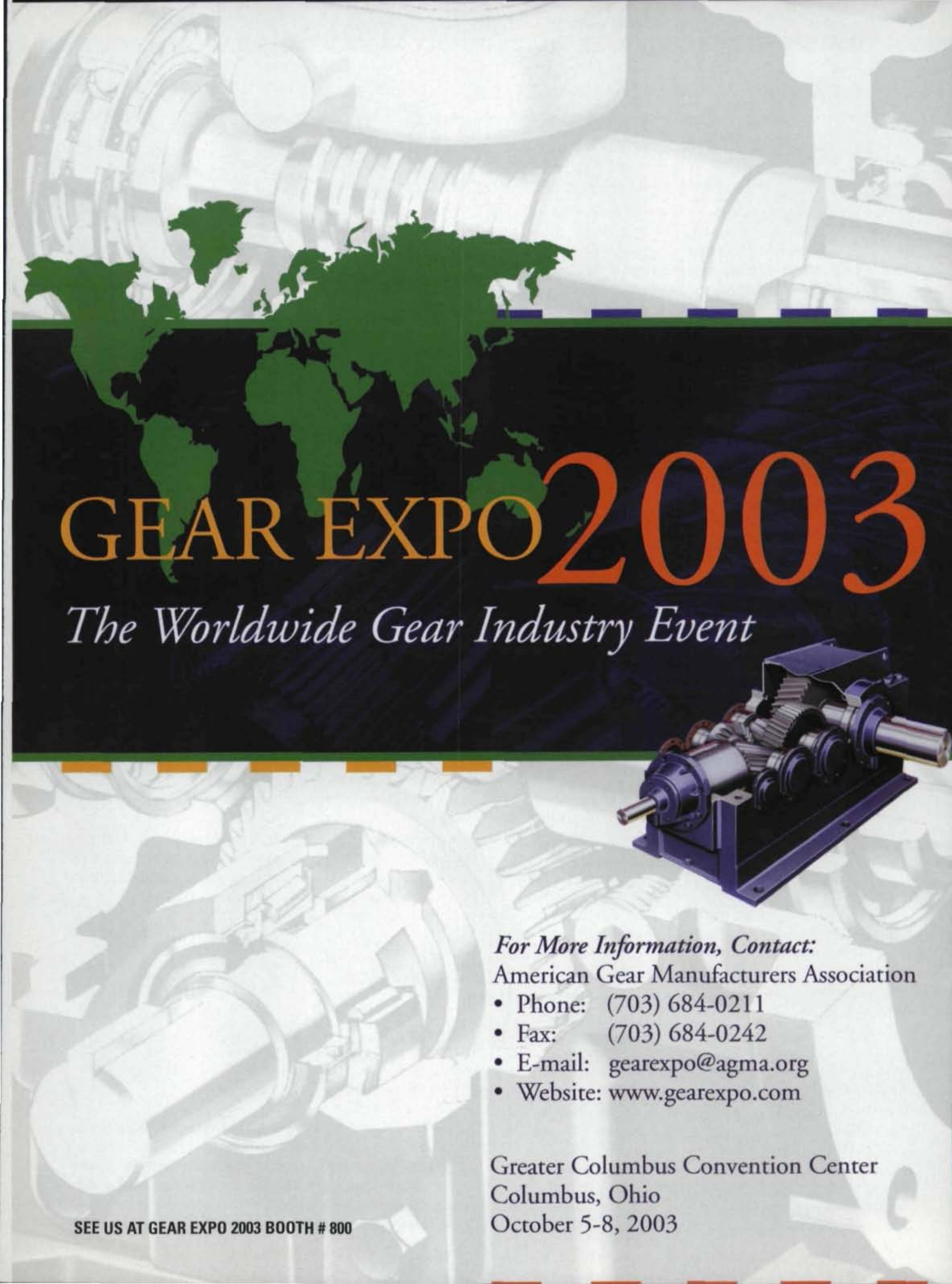
But part of the growth tells me that there is activity in our industry.

We offer no black-and-white conclusions about what lies ahead based on the information presented here. No matter how you interpret the trends, though, the growth on all three of these websites has to be taken as a good sign for our industry.



Michael Goldstein
Publisher & Editor-in-Chief

P.S.—Please tell us if you find the information in these charts interesting or useful. Also, please tell us if you have any suggestions about other information you'd like to see about the gear industry. Call (847) 437-6604 or send e-mail to *people@geartechnology.com.*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Welding Different Gear Materials

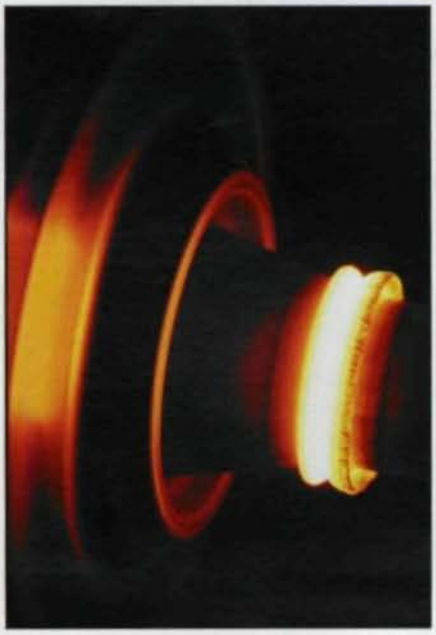
Friction welding has been thought of as a solution for customers with lots of volume to be welded and lots of capital investment. Industry opinions are changing, though, and it's becoming an economic option for small-volume batches.

The process has been around for awhile but has never been as popular as other conventional means of welding. Unlike traditional welding, friction welding can join both similar and dissimilar materials with different mechanical and physical properties through rotational forces and hydraulic pressure without flux or filler material. Good candidate materials for these combinations are carbon and alloy steels.

Among the gear industry applications are bi-metallic propeller shafts for the marine industry, splined axle shafts and flanged gear blanks.

In addition, friction welding takes place in a narrow, heat-affected zone, so it has minimal effect on adjacent machine or heat treat characteristics, says Joel Donohue, general manager of American Friction Welding.

The job shop, located in Brookfield, WI, has offered this service for years, mostly on hydraulics, pump shafts, electric motor shafts, drills, and hand tools, as well as automotive and construction equipment.



An axle is being friction welded to a standardized hub-like end. Photo courtesy of American Friction Welding, Inc.

Whatever the product is, once it arrives at American Friction Welding, the welding process starts with the product's axis of symmetry. It's easiest when the components have a natural axis of symmetry. If a part is not already equipped with this, then engineers develop the tooling to create one.

After the axis of symmetry is determined, weld parameters are developed which include rpm, axial load, time frame, and amount of axial shortening. After developing these parameters, parts are then tested in a situation that mimics its actual performance. Once those parameters are established and proven, they're loaded into the machine's controller. Then they begin the process of loading parameters and feeding components before commencing production. The machine controller monitors those set parameters and their limitations throughout the production run. If anything falls outside the parameters, the machine sends out an alert.

One workpiece is held in a rotating spindle and the other is held in a stationary clamp. Operators control the speed of the motor-driven workpiece. Then, an axial load is applied to the components being welded. Interfaces of the two components rub together, resulting in heat.

This is maintained until a predetermined amount of time or axial shortening occurs. At this point, a braking force is applied and the axial load is increased in the final forge phase. This force is held for a pre-set amount of time after the rotation stops.

Many times, a post-weld test is done on a sampling basis as the final step to check for torsional, bend or hardness properties of the heat affected zone. In addition, ultrasonic inspection is used to evaluate the weld integrity.

All of this, excluding the post run testing, takes from five seconds to several minutes. When factoring in the testing, the initial weld development process can last from a few hours to a few weeks, depending on part size, shape and configuration.

***Welcome to* Revolutions, *the column that brings you the latest, most up-to-date and easy-to-read information about the people and technology of the gear industry.* Revolutions *welcomes your submissions. Please send them to* Gear Technology, *P.O. Box 1426, Elk Grove Village, IL 60009, fax (847) 437-6618 or e-mail* people@geartechnology.com. *If you'd like more information about any of the articles that appear, please use Rapid Reader Response at* www.geartechnology.com/rrr.htm.**

As far as capabilities, American Friction Welding has eight direct-drive friction welding machines ranging from four to 125 tons, which can weld solid diameters from 1/8"–4 1/4".

"One machine has part orientation. Its spindle has an end coder on the spindle that keeps track of where the spindle starts and stops," says Donohue "The advantage of this feature is that the components can have uniquely machined features with a relationship to one another after welding within +/–1.5°." This kind of control is a major benefit to the direct-drive welding process.

However, there's another type of friction welding called inertia welding. Manufacturing Technology Inc. in South Bend, IN, does both and points out distinctive aspects of each approach.

Inertia welding uses flywheels bolted to a spindle chuck. The spindle accelerates to a pre-determined speed, its motor is disengaged and the workpieces are forced together under forge force. As the spindle speed slows, the kinetic energy stored in the rotating mass dissipates as heat through friction at the weld interface. The welding force continues to push the two components together for a short period of time after the rotation ceases.

Though it offers the operator less control, inertia welding generally has faster cycle times, a narrower heat-affected zone and fewer weld variables (rpm and pressure) and does not require clutches and brakes, says Kevin Grewe, sales engineer for MTI Welding.

"For gears, there's not a definite method that's better than the other. Sometimes there's the need to orientate one gear onto another and, in that case, the direct drive friction welding is best," he says. "We do slightly more inertia welding, mainly because automakers want such high cycle times."

Friction welding, whether done by inertia or direct drive, is not a solution for every company or application. Many times, the component configuration and material type are good indicators of whether the process is a good fit. Generally, free machining and resulfurized materials, which contain high levels of lead and sulfur, may have adverse effects on the joint, says Donohue.

For Kuhn Knight Manufacturing Inc. of Brodhead, WI, the friction welding process was a perfect fit. Hugh Hosely, a buyer for the company, hired American Friction Welding several years ago to friction weld Kuhn's plug assembly products.

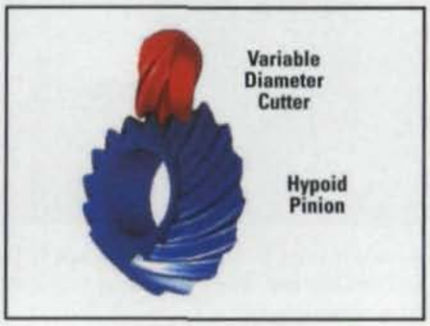
"We've been very happy with the service," he says, "it's inexpensive and produces a better product because two pieces become one. Also, with a traditional weld, they're only held together by a ring of weld, and this is far superior."

No Pitch Cones, No Face Cutters = Greater Gear Freedom

David Dooner wants more freedom for designers and manufacturers of spiral bevel and hypoid gears. He wants to give it to them by removing the gears' restrictions on face width, spiral angle and number of teeth.

He also wants them to be free to specify those gears with the same procedure they use to specify cylindrical gears.

And Dooner has a way to achieve this freedom: Eliminate pitch cones and



face cutters from the design and manufacture of spiral bevels and hypoids.

Dooner has been working on this way since the late 1980s, when he was a graduate student at the University of Florida, Gainesville. Studying for his doctorate, Dooner started to think about unifying the design and manufacture of different type of gears, using their kinematic structure to create a single framework.

He discussed his idea with his doctoral advisor, Ali Seireg, and it became part of his doctoral thesis. It also provided the foundation of his method for spiral bevels and hypoids.

After receiving his doctorate, Dooner continued to work on the method, collaborating with Seireg until Seireg's death in September. Today, Dooner is an associate professor in the mechanical engineering department at the University of Puerto Rico-Mayagüez.

To this point, his method remains theoretical—"It hasn't been developed."

Dooner says it hasn't been because: "First, the mathematical relationships are not immediate." and "Second, there is a well-established art, with an enormous base."

He adds that most people are focused on improving the existing method—"There's been little effort to developing new ways."

Dooner, however, has been focused on getting rid of pitch cones and face cutters to create a new way.

He explains: Pitch cones are a design tool for spiral bevel and hypoid gears. While helpful, the cones restrict the face widths of those gears.

With Dooner's way, the gears would be designed using mathematical formulas that compare the spiral bevels and hypoids with their theoretical ideals. That contrasts with today's practice of comparing them with the ideals that can be obtained through manufacture.

Dooner says using the theoretical ideals gets rid of the restrictions on spiral bevels and hypoids' spiral angle and number of teeth and may get rid of their limit pressure angles.

During manufacture, face cutters of spiral bevels and hypoids create teeth of increasing thickness as they move across a gear's face.

Dooner's cutters, though, would be like variable diameter gears with cutting teeth. They'd cut like variable diameter hobs, so they'd compensate for varying tooth thickness. The compensating would remove restrictions on face width and spiral angle. Without face cutters, number of teeth wouldn't be restricted, either.

Dooner says cutting spiral bevels and hypoids would become like hobbing cylindrical gears, so specifying them could become like specifying cylindrical gears. Tooth profile, pressure angle, spiral or helix angle, face width—they could all be used to design and manufacture spiral bevels and hypoids.

According to Dooner, the cutters could even be used to manufacture cylindrical gears. Different cutters would still be needed to manufacture different gears—coarser pitch cutters for coarser pitch gears, finer for finer, bigger cutters for bigger gears, smaller for smaller—but the cutters would be the same type and could be used in the same type of machine.

Dooner says his machine and cutters would streamline a gear manufacturer's machine tool facilities while making them more flexible in the types of gears they could create.

According to Dooner, his method would provide several other benefits. First, gear manufacturers would be able to create lead, crown and profile relief in their spiral bevel and hypoid gear sets at the same time they're cutting them.

Second, they might not need to lap, burnish or polish the sets. Dooner says if lapping, burnishing or polishing wasn't needed, gears and pinions would be interchangeable. If one broke, a gear manufacturer could replace just that member of a hypoid or spiral bevel gear set; he wouldn't have to make an entirely new set.

Third, his method would offer new alternatives in gears through its greater design flexibility. For example, spur hypoidal gears—not possible in today's manufacture—would be possible. Dooner describes spur hypoidals as non-intersecting, non-parallel gears whose teeth go into and come out of mesh all at once.

Moreover, Dooner simulated their manufacture and saw his method could even make spur hypoidals more efficient by reducing their sliding contact.

Spur hypoidal gears could be used in rear axles of automobiles.

Before benefits, there'd have to be development and testing. Both are possible, though. Dooner's work is at a point where a machine and set of cutting tools could be developed for testing. He doesn't know how much money it would take to create them, but he doubts it would be an unusual amount: "How much would it take to develop an existing machine?" ○

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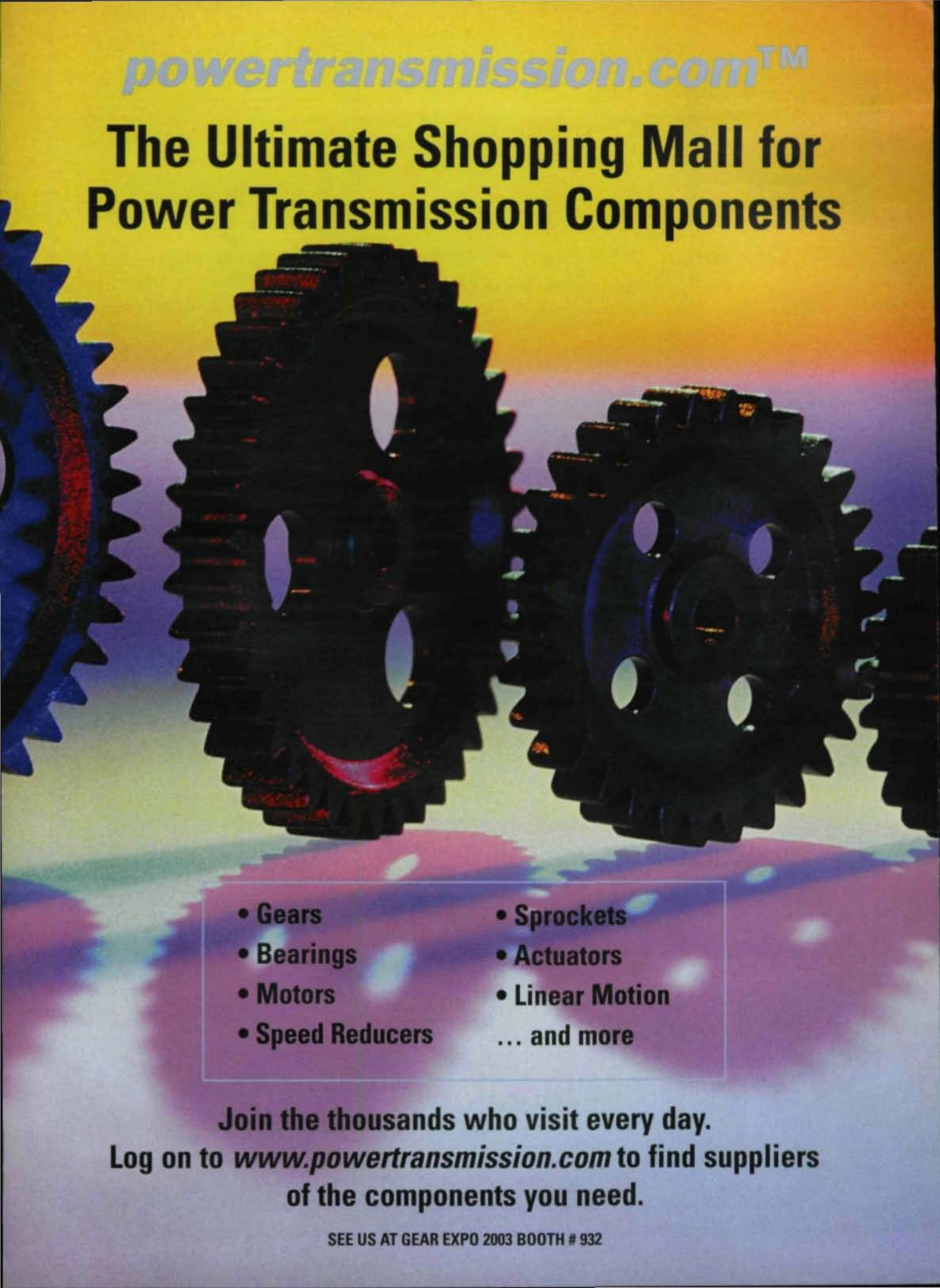
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HOTELS

Gear Expo exhibitors and attendees can take advantage of special AGMA rates, but only by booking before September 9, 2003. Published rates are for single/double occupancy. Mention reservation codes indicated below to take advantage of the discount!

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The full-service, 600-room hotel is directly connected to the convention center. For more information, visit *www.columbus.hyatt.com.*

Crowne Plaza Downtown—$129
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Telephone: (614) 461-4100
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Also connected to the convention center, it has 400 rooms and suites. For more info, visit *www.cmh-downtown.crowneplaza.com.*

Hampton Inn & Suites—$115
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Courtyard by Marriott Downtown—$110
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Code: AGMN
Located three blocks south of convention center, but within walking distance of several restaurants. For more info, visit *www.courtyard.com/cmhcy.*

Red Roof Inn Downtown—$100
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Telephone: (614) 224-6539, (800) 733-7663
Fax: (614) 228-4037
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Situated across the street from the convention center. This select-service hotel has 150 sleeping rooms and suites. Visit *www.redroof.com* for more information.

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Connected to the convention center, the hotel has 180 rooms and suites. For more information, visit the web site at *www.druryhotels.com.*

AGMA officials hope to sell companies on attending Gear Expo based on three factors—location, location, location.

The bi-annual trade show will be held Oct. 5–8, at the Greater Columbus Convention Center in Columbus, OH. Detroit, MI, has traditionally been the guaranteed venue for half of the Gear Expos because of its reputation as the automotive capital of the world. Alternating shows rotate between other Midwestern cities. Gear Expo '01 was held in Detroit and Gear Expo '99 in Nashville, TN.

According to AGMA vice president Kurt Medert, the Columbus location should prove to be an ideal choice because of its centralized location. While Nashville was too far south for many companies to send representatives, Columbus is within driving distance of the world's gear manufacturing centers.

"The city that constantly does well is Detroit," he says. "But with Columbus, 60% of our audience lives within 500 miles."

Numerous manufacturing companies are located within the Columbus metropolitan area itself and can make a day trip of Gear Expo. For example, Honda Motor

Columbus by night skyline, as viewed from the highway. Courtesy of the Greater Columbus Convention & Visitors Bureau. Photo by Rod Berry.

Co. Ltd. has several plants in the area, and Columbus is close enough for Kentucky companies to drive up for the day.

In the current economy, closeness can be an asset because it would enable attendees who live in the Midwestern or some of the Eastern states to drive to Gear Expo. Columbus is approximately seven hours driving distance from Chicago, three hours from Pittsburgh, and 12 from Des Moines. Show goers who are in a greater hurry can take advantage of the summer air fares or late booking rates. Cheaper airfares are expected closer to the show date, and AGMA has set aside a block of hotel rooms at special rates (see box). Hotels are all approximately a ten-minute drive from the airport and cab fare should run less than $15.

Regardless of how they get there, Medert expects Gear Expo attendees to be impressed with the show's facilities.

The Greater Columbus Convention Center was recently expanded with unlimited floor load restrictions, which is another reason it was the site selected. (Because of the heavy equipment normally brought to the show, Gear Expo cannot be held at facilities with restrictions of less than 350 pounds per square inch.)

Also, there are 30 restaurants within five to 10 minutes' distance of the show.

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Expo Exhibitors Look for Economic Upswing

Joseph L. Hazelton

Companies in the gear industry are looking for signs of an economic upswing as they prepare for Gear Expo 2003, and several are seeing such a sign.

The sign is quoting.

"Quoting activity has been heavier starting in the first quarter," says Dave Melton, expositions/communications manager for Gleason Corp. of Rochester, NY. "We are seeing products are being quoted, but because of the war [in Iraq], [purchases] are being pushed back."

Quoting is also up at Contour Hardening Inc. in Indianapolis, IN. Quoting has increased for its contract processing and heat treat systems.

"Which would be a pretty good indication that something's getting ready to bust loose," says Contour's vice president, Michael R. Chaplin.

In Alexandria, VA, Kurt Medert, vice president of the American Gear Manufacturers Association (AGMA), adds he's been hearing generally positive talk about business among gear manufacturers and suppliers through several recent association meetings.

At American Metal Treating Co., though, Bruce Devney is talking to gear manufacturers and isn't hearing encouraging news. "Nobody is talking right now about any rosy projections for the next six months," says Devney, president of the Cleveland, OH-based company.

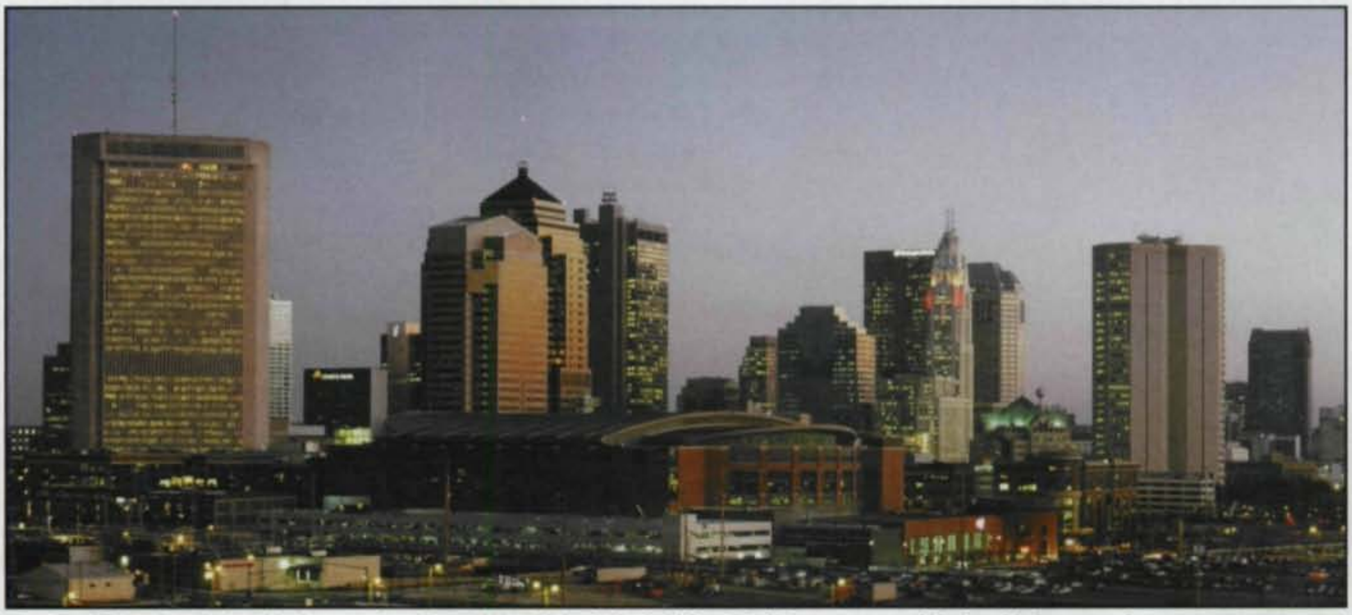
David Goodfellow, president of SU America Inc. of Hoffman Estates, IL, says, "We're hoping there will be some kind of resurgence in the gear industry by that time."

That time is Oct. 5–8, when the AGMA will hold Gear Expo 2003 in Ohio, at the Greater Columbus Convention Center.

But, with the economic slowdown, Melton expects expo attendance to be down. Devney agrees: "I expect fewer attendees, traffic should be slower—maybe even less exhibitors."

Goodfellow notes, though, the expo is in the heart of the Midwest—Illinois, Indiana, Iowa, Michigan, Ohio and Wisconsin—which is a "fairly substantial gear industry territory."

For a sense about attendance, *Gear Technology* surveyed people at 20 companies who were potential expo attendees. The survey was informal, so its results aren't representative. Still, of 20 companies, 11 replied they'd be sending



The city of Columbus, OH, will be home to Gear Expo 2003, which will be held Oct. 5–8. (Photo courtesy of Nationwide Insurance.)

In Columbus, more than 130 gear-industry organizations will exhibit at Gear Expo 2003. (Photo courtesy of Rod Berry.)

at least one person to the expo.

None of the people surveyed, though, replied that he'd be looking at machinery with the possibility of buying.

Still, Melton expects attendees to be a more focused, serious crowd. "They'll be looking hard at what new technologies are out there," the Gleason manager says. "They'll be more serious buyers than tire kickers."

Serious buyers are what A/W Systems Co. got at Gear Expo 2001 and are what it's hoping for again. "We probably got 12 to 15 good solid leads that turned into customers," says Ross Deneau, vice president of manufacturing for A/W, located in Royal Oak, MI.

A/W is one of what seems to be the industry's "fortunate few." The slowing economy hasn't touched the cutting-tool company—"Our particular case, it's never slowed down," Deneau says.

He explains A/W's repair work increased a lot—"When the economy slows down, people look to repair [more] than to replace." More recently, Deneau's seen increased buying of new tooling. After all, products eventually wear out.

And Gear Expo is a chance to look at new products.

Star Cutter Co. of Farmington Hills, MI, and SU America will feature their lines of gear-related products, consolidated through Star-SU Inc. and displayed in their shared expo space. The space will also feature Star-SU's line of Fellows gear shaper cutting tools, as well as a Fellows gear shaping machine from Bourn & Koch Machine Tool Co. of Rockford, IL.

In its space, Gleason will exhibit its 245 TWG, a threaded wheel grinding machine, and the GMX 275, an analytical gear tester. The GMX is first in a new series of analytical gear testers jointly developed by Gleason and Mahr GmbH of Gottingen, Germany.

A/W will display blades (both carbide and high speed steel), cutter bodies, hobs, shaper cutters and other products. A/W expects to feature its alliance with Klingelnberg Oerlikon Technology Center (KOTC) of Saline, MI. The two companies will keep separate spaces at the expo, though.

American Metal Treating Co. will display an induction hardening process for internal gears. The CNC process can treat internal gears with outside diameters up to 30 inches and diametral pitches from 2 to 6.

Contour Hardening will exhibit a dual-frequency hardening process that can send low and medium frequencies and radio frequencies through one coil.

New Business Allies Combine Expo Spaces

Like the companies of Sigma Pool, several newly allied businesses will have combined spaces at Gear Expo 2003.

The allied businesses are: Star Cutter Co. and SU America Inc., Bourn & Koch Machine Tool Co. and SU America, and Gleason Corp. and Mahr GmbH.

At Gear Expo 2001, Bourn & Koch of Rockford, IL; Star Cutter of Farmington Hills, MI; and SU America of Hoffman Estates, IL, occupied their own spaces. At Gear Expo 2003, they'll occupy one 3,200-square-foot space.

This combining comes from two alliances. The first consolidated the gear-related product lines of Star Cutter and SU America. Created in early 2002, the alliance resulted in a joint company, Star-SU Inc., for selling the consolidated lines.

The second alliance was between Bourn & Koch and SU America via the gear-related product lines of defunct Fellows Corp.

Founded in 1896, Fellows manufactured gear shaping machines and cutting tools from its headquarters in North Springfield, VT. On Feb. 13, 2002, Fellows ceased operations as its parent company, Goldman Industrial Group, filed for Chapter 11 bankruptcy.

By summer 2002, Fellows' manufacturing assets had been sold and split between Bourn & Koch, which acquired the assets related to gear shaper machines, and Star-SU, which acquired the assets related to gear shaper cutting tools.

The two companies also agreed Star-SU would be responsible for promoting, representing and selling Bourn & Koch-built Fellows gear shapers through Star-SU and Bourn & Koch distribution channels.

David Goodfellow, SU America's president, says this expo will be a chance for previous Fellows customers to be reintroduced to Fellows products, which have been partly redesigned and re-engineered.

Reflecting another alliance, Gleason of Rochester, NY, will share its 2,500-square-foot space with Mahr of Gottingen, Germany. In 2001, Mahr arranged with Gleason to use Gleason's G-AGE software on Mahr's PRIMAR form and gear measuring machine.

Their partnership soon became much more extensive. At the start of 2002, Gleason became the sales and support agent for Mahr's gear metrology products around the world. The products included the gear-specific version of Mahr's PRIMAR machine.

Gleason and Mahr also started to jointly develop new gear metrology products, which would be sold under the brand name "Gleason Mahr." Later in 2002, the two companies introduced the GMX 275, the first in a new series of analytical gear testers from the joint arrangement. ○

Besides new products, Gear Expo is a chance to talk.

Not surprisingly, exhibitors expect talk in the aisles to be about the economy.

Devney expects the "aisle talk" to include the continuing movement of gear manufacturing overseas to eastern Europe, mainland China and India.

Chaplin expects talk about the economy too, but he adds people will also be talking about quoting activity in their own companies and what it means.

Goodfellow says he expects talk to include gear manufacturers themselves having more chances to quote gear jobs for the automotive and truck industry as it continues to outsource more of its manufacturing.

As of June 3, the AGMA had 135 exhibitors set to occupy more than 38,000 square feet of space in Columbus' convention center. In 2001, Gear Expo had 192 exhibitors who occupied 57,000 square feet.

Medert, however, says many companies are waiting until the last minute before deciding whether to exhibit. He adds they're waiting in the hope that there'll be an economic upswing.

Despite them, Medert says: "Overall, I expect the same high-quality show we've always had."

Besides U.S. companies, the expo will host exhibitor companies from Australia, Canada, the Czech Republic, Finland, Germany, India, the People's Republic of China, Spain, Switzerland and Taiwan. Other non-U.S. companies will be represented through their American operations, such as Mitsubishi Gear Technology Center of Wixom, MI, and SU America.

In three months, these and other exhibitors will gather for the gear industry's trade show. By then, the possible upswing that several see may be an actual upswing.

As Goodfellow says: "We're hoping that there's some sense of recovery by then." ☼

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Russell, Holbrook & Henderson

Booth Number: 313

Product Line

As a leader in the fine-pitch gear tooling market, our line of TRU-VOLUTE hobs, shaper cutters and gear metrology products are recognized for the highest quality and precision in the industry.

Our products provide the superior quality and accuracy you demand. Our focus continues to be on the importance of quality tools, inspection methods and gear technologies, which are essential to first-class gear manufacturing.

Company Profile

Established in 1915, Russell, Holbrook & Henderson, Inc. has since been providing innovative products and solutions to the gear tooling industry.

Our parent company, Ogasawara Precision Laboratory Ltd., Japan, is a world leader in fine-pitch gear tool manufacturing and related products.

Contact

Russell, Holbrook & Henderson, Inc.

For the latest in our TRU-VOLUTE product updates, related articles and events calendar, please visit our online website at: http://www.tru-volute.com

Our sales and technical service can be reached at the following location:
Russell, Holbrook & Henderson, Inc.
A Member of Ogasawara Group
17-17 Route 208 North, Fair Lawn, NJ 07410
Telephone: (201) 796-5445
Fax: (201) 796-5664
http://www.tru-volute.com

"Gear Tool Solutions"

JRM International, Inc.

Booth Number: 538

Product Line

JRM International, Inc. has been a distributor of German and American machine tools and accessories since 1984. Burka-Kosmos supplies grinding wheels for all makes and models of thread-type, profile and generative gear grinders. James Engineering is a builder of gear deburring, chamfering and washing machines. Schrem hydraulic locknuts are designed for clamping all types of tools and workpieces. WEKO Gear Technology is a supplier of CNC inspection machines, machine service and rebuild/retrofit. You can also talk to the people from Kesel about their rack milling machines. Visit us at booth number 538.

JRM International, Inc.

JRM International, Inc.
5701 Industrial Avenue
Rockford, IL 61111
Phone: (815) 282-9330
Fax: (815) 282-9150
E-Mail: *jrmsales@jrminternational.com*
Web Site: *www.jrminternational.com*

Marposs Corp.

Booth Number: 1115

Need to Check Gears?

Looking for an easy, cost-effective way to check gears? The Marposs M62 bench gauge system with optional noise detection is accurate, easy to use and quickly re-toolable using ordinary hand tools. You'll save time and the cost of dedicated gauges. With the M62 you can measure:

- Radial composite deviation
- Radial runout
- Tooth to tooth radial composite deviation
- Tangential composite deviation
- Nicks
- Optional measurements include center distance, bore diameter and perpendicularity of bore to gear face.

All functional checks can be displayed on the rugged Marposs E9066 industrial PC system that offers a linear graphic display with full SPC functions.

Contact

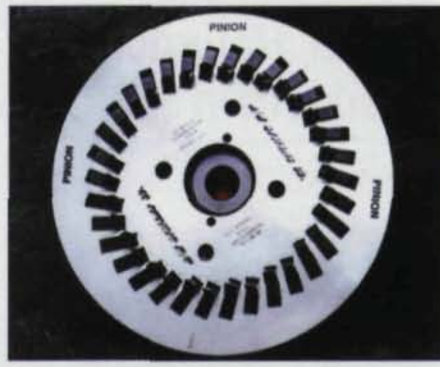
MARPOSS
3300 Cross Creek Parkway
Auburn Hills, MI 48326-2758
1-888-627-7677 • 1-248-370-0404
Fax: 248-370-0621
E-mail: marposs@us.marposs.com
www.marposs.com

A/W Systems

Booth Number: 111

Description:

We will have on display a selection of cutter bodies and blades. We will have straight bevel cutters, coniflex cutters and stick blades. (HSS & carbide) We will have bore gages and height gages. We will have hobs. We will have an assortment of diamond tools for hard turning and dressing. We will have keyseat cutters and dove tails on display. We will have CBN and diamond plated wheels. We will also have an assortment of hardware.



Contact:
A/W Systems
612 E. Harrison
Royal Oak, MI 48067
Phone: (248) 544-3852
Fax: (248) 544-3922

Dr. Kaiser **Diamond Dressing Tools**

Booth Number: 218

Product Line

Dr. Kaiser Diamantwerkzeuge, located in Celle, Germany, specializes in the design and manufacture of precision rotary diamond dressers for the gear industry. ***Dr. Kaiser*** precision diamond products are unsurpassed in quality, accuracy, and innovative design.

Company Profile

Dr. Kaiser provides dressing solutions for continuous generating, single tooth, spiral, bevel, gear, and plunge grinding applications in both natural and polycrystalline diamond versions. ***Dr. Kaiser*** PCD reinforced dressers significantly extend dresser life when compared with standard non-reinforced direct plated dressers, especially in fine-pitch applications. ***Dr. Kaiser*** offers quick delivery and the highest accuracy at competitive prices. Relapping and replating services are available.

S.L. Munson & Company is the exclusive distributor for ***Dr. Kaiser*** products in North America.

Contact
S.L. Munson & Company
401 Huger Street
Columbia, SC 29201
Phone: 800-775-1390
Fax: 803-929-0507
E-mail: *info@drkaiser.com*

Arrow Gear Company

Booth Number: 933

Arrow Gear to Exhibit at Gear Expo

Arrow Gear Company of Downers Grove, Illinois will be an exhibitor at this year's Gear Expo.

Arrow's booth will contain several points of interest for visitors to the show. A variety of gold plated gears will be on display to illustrate the depth of the company's machining capabilities. To communicate the full scope of Arrow's operation, an orientation video will be presented. In addition, printed materials and interactive CD-ROM programs will be distributed.

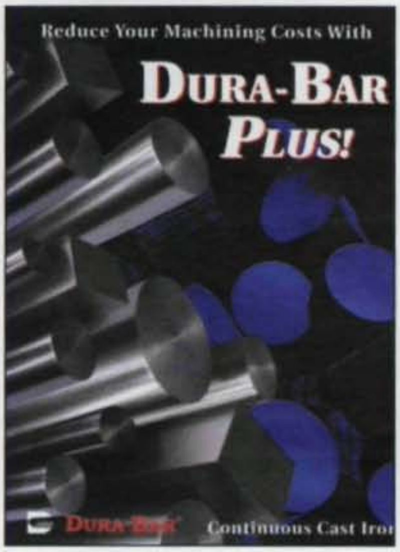
Arrow Gear produces high precision gears and gearboxes for aerospace and commercial markets. They have also introduced advanced computer techniques for the design and development of bevel gears.

Arrow Gear will be located at booth #933.

ARROW
GEAR
COMPANY

Dura-Bar

Booth Number: 927



Product Line

Dura-Bar is the only manufacturer of continuous cast iron bar stock in North America, and is the largest in the world. Rounds, squares, rectangles, tubes and special shapes are supplied through a nationwide distribution network. Inherent properties include noise and vibration damping, machinability, strength, and wear resistance, making our bar stock an alternative to steel and other metals in fluid power, gear and other applications. New Dura-Bar *Plus* is an engineered iron that machines better than carbon and alloy steels and other grades of continuous cast iron. A material of uncompromising consistency, it allows metal component manufacturers to machine more parts per hour, thereby decreasing cycle time, increasing productivity, and boosting profitability.

Dura-Bar
1-800-BAR-MILL (227-6455)
www.dura-bar.com
sales@dura-bar.com

Presrite Corporation

Booth Number: 820

Product Line

Presrite Corporation's eight-page color brochure provides an overview of the company's forging capabilities, and information on its equipment and its three forging plants. It also identifies industries served and customer benefits. Presrite is equipped with a total of 16 forging presses, ranging in size from 1,300 tons to 6,000 tons. The company also has a state-of-the-art gear lab and sophisticated machining equipment. Presrite produces minimum-draft, net- and near-net-shape forgings up to 3,000 lbs. and 18 inches in diameter, and provides induction heating capacity for ferrous materials of 28,000 pounds per hour.

Presrite Corporation
3665 East 78th Street
Cleveland, OH 44105-2048
216.441.5990 FAX: 216.441.2644
E-Mail: info@presrite.com
Web Site: www.presrite.com

Chamfermatic Inc.

Booth Number: 1125

Company Profile

Chamfermatic Inc. was started in 1996 with the goal of building the best, easy-to-set-up gear deburring machines on the market. We have, by listening to our customers' needs, accomplished this goal.

Product Line

Chamfermatic Inc. offers a complete line of gear deburring machines with capacities up to 36 inches in diameter, from manual to fully automated systems. We also incorporate parts washing along with deburring operations to remove oil, lapping compound and chips from your parts, while adding a rust preventative. All of our equipment has as standard: automatic air-operated door, operator interface, P.L.C. and a filtration system. Also, they are all portable.

Our customer service response time is second to none. We also offer contract deburring of your gears, along with rebuilding all makes and models of gear deburring machines, making Chamfermatic a full-service supplier to the industry.

Contact

Chamfermatic Inc.

7842 Burden Road
Machesney Park, IL 61115
Mike Magee, President

Phone: (815) 636-5082
Fax: (815) 636-0075
E-mail: chamfer96@aol.com

Star-SU

Booth Number: 1134

Star-SU offers a comprehensive line of gear cutting tools of all types, including services for:

Hobs
Shaper cutters
Milling cutters
Rack cutters
Band saw cutters
Profile grinding wheels, electroplated CBN and diamond-plated
Shaving cutters
Chamfering and deburring tools
Chamfer-roller tools
Rolling and deburring tools for sprocket gears
Master gears
Screw compressor rotors

Resharpening and recoating of hobs, milling cutters and shaper cutters
Resharpening of shaving cutters
Stripping and replating of electroplated CBN and diamond-plated grinding wheels
Complete Commodity Management Services (CMS)
Pickup and delivery services are available in certain areas

Visit our website to download our complete program of products and find the match for your particular need.
www.star-su.com
sales@star-su.com

Our partners SU America, Star Cutter and Bourn & Koch offer a complete line of machine tools for gear roughing, finishing and tool sharpening applications.
www.suamerica.com
www.starcutter.com
www.bourn-koch.com

See us at AGMA Gear Expo 2003, Columbus, OH, in Booth 1134.

Barit International Corporation

Booth Number: 917

Company:

Barit International Corporation is a one-stop solution provider for all gear cutting tool needs, supplying the world's industries with top quality gear cutting tools since 1989. We excel in cost efficiency, quality and precision.

Product Line:

- Gear hobs
- Worm gear hobs
- Involute and parallel spline hobs
- Shaper cutters
- Broaches
- Form relieved milling cutters
- Shaving cutters
- Rack type cutters
- Straight bevel cutters

Please visit our booth for more details!

Contact

BARIT INTERNATIONAL CORPORATION
3384 Commercial Ave.
Northbrook, IL 60062 USA
Tel: 847-272-8128
Fax: 847-272-8210
Website: *www.barit.com*
E-mail: *people@barit.com*

Nachi Machining Technology Co.

Booth Number: 500

Nachi Machining Technology Co. represents global technical innovation and leadership in the machine building and tooling industry. Nachi supplies high technology machining solutions through the Nachi-Red Ring brand of machines and tools.

Nachi Machining Technology Co.'s. goal is to facilitate processes which optimize productivity while maintaining and improving quality.

The exhibit includes a display of tools, process examples, and video presentations. Broaching, Roll Forming, Hobbing and Shaving Machines/Tools are employed to demonstrate manufacturing processes for typical work pieces. Work pieces and manufacturing equipment will be available for inspection at the show.

Contact

NACHI MACHINING TECHNOLOGY CO.
17500 Twenty-Three Mile Road, Macomb, MI 48044-1103
Phone: (586) 263-0100
Fax: (586) 263-4571
www.nachimtc.com

SU America

Booth Number: 1134

SU America offers a comprehensive line of gear roughing, finishing and sharpening equipment including machine tools for:

Hobbing for dry and wet machining
Spiral bevel gear cutting for dry and wet machining
Profile grinding of external and internal gears
Continuous generating gear grinding
Shave grinding/honing
Rotor, worm and broach grinding
Shaving cutter and master gear grinding
Chamfering and deburring

With a long history of manufacturing gear cutting tools Samputensili has gained a significant competency in applying its own machine tools to its manufacturing processes.
Options that include on-the-machine inspection, automatic dressing or robotic automation are available. We offer economical single purpose or multiple purpose package solutions depending on the application need.

SAMPUTENSILI

Visit our website to download our complete program of products and find the match for your particular need. We're closer than you may think.
www.samputensili.com
sales@suamerica.com

Our partner Star-SU Inc. offers a complete line of gear cutting and finishing tools complementing our range of machine tools.
www.star-su.com
sales@star-su.com

See us at AGMA Expo 2003, Columbus, OH, in Booth 1134.

Greg Allen Company

Booth Number: 732

Greg Allen Company is dedicated to providing gear tools to the finest gear manufacturers in the U.S., Mexico & Canada. Our product line includes HOBS, GEAR SHAPER CUTTERS, SHAVING CUTTERS, MAAG CUTTERS, & GEAR MILLING CUTTERS, and OUR SHARPENING FACILITY services & supports all the above. Every tool that we provide is 100% guaranteed to meet or surpass your specified requirements. Every tool we furnish bears our name, proving to you that we will stand behind our tools. We are featuring two of the finest gear tool producers in world at this year's AGMA Show in Columbus: DRAGON CUTTING TOOLS LTD. & TRANSMECANICA S.A.C.I. Both firms are recognized as World Class Producers of gear cutting tools, which are sold & serviced in Fairview Park, Ohio, by a staff averaging 25 years' experience in the gear tool field. If your firm has yet to realize the advantage of Dragon & Transmecanica...........we need to be introduced. Our booth is #732. Stop by and find out why our customers are happy and why they will agree that Greg Allen Gear Tools "GENERATE PROFIT."

Contact

Jan Thomas
Greg Allen Company / Index Technologies
21135 Lorain Road
Fairview Park, Ohio 44126
440-331-0038 , fax- 440-331-0516

AGMA

Booth Number: 800

GEAR EXPO is the worldwide gear industry event, and the only trade show devoted exclusively to gearing. This year the Show will be held October 5-8, 2003 at the Columbus Convention Center in Columbus, Ohio. The Show is international in scope and provides a biennial forum for the exchange of information on the broad range of machinery, supplies and services available for the gear manufacturing process. From software to hardware, from initial design through testing, exhibitors will be available to discuss solutions to problems associated with gear manufacturing.

For more information, visit *www.gearexpo.com* or e-mail *gearexpo@agma.org*.

News in the Gear Industry

Company Revitalizes Lees Bradner Gear Hobbers

Industrial Actuation Group of Limerick, ME, which recently purchased Fayscott Co., has revitalized two product lines: Lees Bradner gear hobbers and Reid precision grinders.

According to its press release, the company will focus on rebuilding, remanufacturing and CNC retrofitting the existing Lees Bradner hobbers. A line of CNC hobbers based on the 7-VH is in the works as well.

Among the product offerings are machines that have been rebuilt to specific standards and CNC hobbers with options to fit specific customer needs.

For more information on Fayscott, look online at *www.fayscott.com*. For more information on Lees Bradner hobbers or Reid precision grinders, e-mail *Fayscott@kynd.net*.

New Sales Manager at Lepel

Richard Detty was appointed sales manager of the induction heating division of Lepel Corp. Detty has more than 20 years' experience in induction heating, most recently at Pillar Industries as the northeast territory's regional manager. Prior to that, he worked for Ameritherm as a lab manager and senior applications engineer.

Lepel, headquartered in New York, NY, has been delivering induction heating solutions for 70 years.

BMPTA Tracks Industry's Production and Trade

A first-of-its-kind report tracking each European country's mechanical power transmission production and trade data on a comparable basis, intercountry and over time, was released by the British Mechanical Power Transmission Association (BMPTA).

The report lists which products different countries included or excluded under headings of chains, gears, gearboxes and transmissions.

Previously called the "Current Market Outlook," this report was updated to provide more consistency, according to the association's press release. Use of the euro provided an opportunity to generate data on the European market based on official EU sources, such as Eurostat.

The statistical profile is available on a CD-ROM for free to current BMPTA members. For ordering information, contact the BMPTA of Burton-on-Trent, U.K., by e-mail at *admin@bga.org.uk*.

New Department at Renold Gear

Renold Gear developed a service and repair department that accepts all types of worm, helical and bevel gear units, regardless of age or original manufacturer.

Based out of the company's facility at Milnrow, Lancashire, U.K., the department also provides opportunities for onsite maintenance and in-site repairs, plant surveys for planned maintenance and technical design expertise for all applications and design requirements.

According to the company's press release, service/repair gearboxes undergo exactly the same engineering processes as new units, and a comprehensive running test and leak check is carried out before pass off. A twelve-month warranty is standard for service/repair units.

Baldor Electric Launches Information Service

ProSpec, a service from Baldor Electric of Fort Smith, AR, provides up-to-date information and education about electric motors, drives, motion control and gear products. The web site can be accessed at *www.BaldorProSPEC.com*.

Six New Facilities Acquired by Metal Improvement Co.

Metal Improvement Co., a subsidiary of Curtiss-Wright Corp., acquired the assets of six USA E/M Engineered Coating Solutions facilities. They will operate under the name E/M Coating Services Division of the Metal Improvement Co., headquartered in Paramus, NJ.

Located throughout the country, the company provides solid film lubricant coatings in North America. Each facility can apply more than 1,100 different coatings to impart lubrication, corrosion resistance, and certain cosmetic and dielectric properties to certain components.

Industries served by these facilities include aerospace, automotive, electronics and the general industrial markets.

Philadelphia Gear's Technical Drawings Available Online

Customers of Philadelphia Gear can request free technical gearbox drawings through a new page on the company's Internet site.

Available requests include outline dimension, layout and mass-elastic drawings. According to the company's press release, all requests are processed within 48 hours.

Customers must provide the unit's serial number as well as the order number located on the nameplate of each enclosed drive.

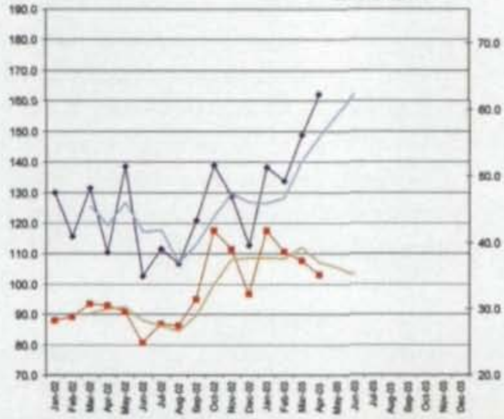
Available technical drawings contain information such as dimensions for input and output shafts and other technical information enabling engineers to ensure proper alignment and to resolve unit footprint issues on-site.

Send Us Your News
E-mail: *people@geartechnology.com*
Fax: 1-847-437-6618
Mail: P.O. Box 1426, Elk Grove, IL 60007 USA

Gear Industry Web Traffic

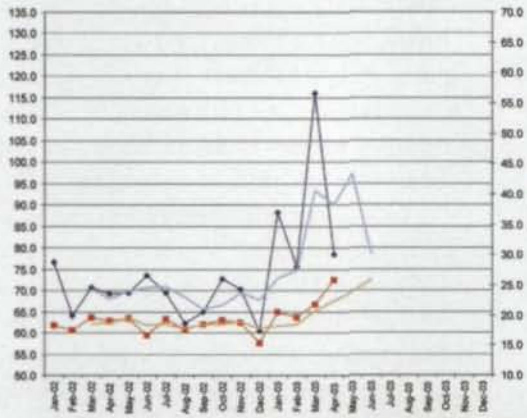
The following charts reflect traffic to three gear industry websites, each representing activity in a different part of the marketplace.

www.powertransmission.com in Thousands 2002-2003



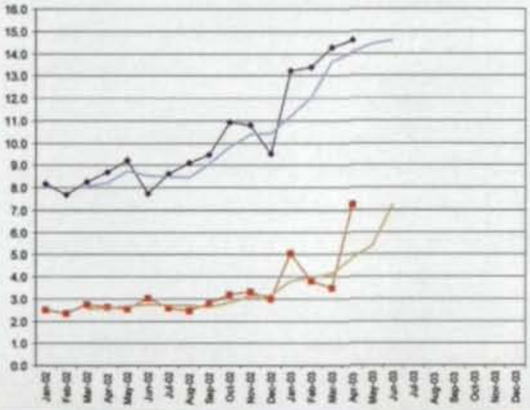
Visitors include those interested in buying gears, gear drives, and other power transmission components.

www.geartechnology.com in Thousands 2002-2003



Visitors include those involved in gear design, manufacturing, inspection and processing.

www.cadillacmachinery.com in Thousands 2002-2003



Visitors include buyers of used gear machinery.

Vegetable-Based Oil as a Gear Lubricant

Boris Kržan and Jožef Vižintin

Summary

Universal tractor transmission oil (UTTO) is multifunctional tractor oil formulated for use in transmissions, final drives, differentials, wet brakes, and hydraulic systems of farm tractors employing a common oil reservoir. In the present work, the gear protection properties of two formulated vegetable-based UTTO oils, one synthetic ester-based UTTO oil, one synthetic ester gear oil, and one mineral-based UTTO oil are investigated.

The data, presented in this paper, have demonstrated that the formulated vegetable-based UTTO oils have high lubricity, high viscosity indexes and provide equivalent or—in some aspects—superior gear protection performance compared with the mineral-based UTTO oil. The high-oleic sunflower oil formulation derived from the genetically modified plant has shown better results than the rapeseed-based (canola-based) oil formulation.

Introduction

It is generally recognized that mineral oil lubricants represent a potential danger in many applications because they are not readily biodegradable and are toxic. The need for biodegradable and non-toxic lubricants has been recognized especially in the areas where they can come in contact with soil, ground water, and crops.

Biodegradability is the ability of a substance to be decomposed by the action of bacteria into CO2, water, mineral compounds and bacterial bodies. Biodegradability is influenced by numerous factors, of which the main are the molecular structure and the chemical properties of organic compounds and the environmental conditions of biodegradability, such as the presence of oxygen, the possible level of nutrition and the pH (Ref. 1).

Vegetable oils and synthetic esters are the most common base stocks for biodegradable lubricants. Synthetic oils represent a fairly recent development in the lubrication market. They can be made by reacting alcohols with fatty acids. Synthetic oils offer improved performance compared with all other lubricants, but at a price. Both vegetable oils and synthetic esters are highly biodegradable and readily available, but vegetable oils occur naturally, have a "greener" image, and are, in general, three times cheaper. Properly balanced additives can compensate for low temperature performance and oxidative stability of the vegetable oils and favor them as the base stock of choice (Refs. 2–3).

A comparison of the simplified chemical structures of mineral and vegetable oils shows great similarities. The major difference is that vegetable oil is an ester, while mineral oil is a hydrocarbon. The presence of the polar ester group impacts several properties, making vegetable oil better than mineral oil in reducing friction and wear. The polar group also makes vegetable oil a better solvent for sludge and dirt, which would be otherwise deposited on the surfaces being lubricated. Because of these properties, it may be possible to reduce the amount of friction modifiers, antiwear agents, and dispersants required to formulate vegetable oil-based lubricants.

The agricultural equipment is ideally suited to use vegetable oil-based lubricants, because the equipment operates close to the environment. The opportunity exists to create a continuous cycle in which the agricultural equipment is lubricated by the oil from a plant growing in the field being cultivated by that same equipment (Ref. 2).

Universal tractor transmission oil (UTTO) is multipurpose oil widely used for agricultural, construction and other off-road vehicles. UTTO oil is

Table 1—Test Oils.

		Viscosity (mm2/s)			
Base stock	Oil type	$\nu_{40°C}$	$\nu_{100°C}$	VI	Oil code
Rapeseed oil	biodegradable UTTO	48.8	10.4	209	R
High-oleic sunflower oil	biodegradable UTTO	51.4	10.6	203	S
Synthetic ester	biodegradable gear oil	101	17.8	195	G
Synthetic ester	biodegradable UTTO	51.3	10.9	211	H
Mineral oil	UTTO	55.1	9.2	150	M

Table 2—Fatty Acid Content In Test Vegetable Base Stocks.

Base stock	Fatty acid content (%)					
	Palmitic C 16:0	Stearic C 18:0	Oleic C 18:1	Linoleic C 18:2	Linolenic C 18:3	Other
High-oleic sunflower oil	4.7	3.7	72.6	17.0	/	2.0
Rapeseed oil	6.1	2.5	49.1	32.2	6.9	3.2

C X:Y .... fatty acid chain of length X and containing Y double bonds; e.g. C 18:3 is an 18 carbon-chain fatty acid with three double bonds.

specially designed for lubricating the transmissions, final drives, wet brakes and hydraulic systems employing a common oil reservoir. UTTO oil has to meet some specific requirements to operate in agricultural and construction equipment. The oil must provide the correct frictional balance to prevent wet brake chatter and to allow smooth transmission clutch engagement.

At the same time, the oil must provide enough clutch capacity for efficient power transmission and enough brake capacity to stop the tractor in a reasonable time and distance. UTTO oil must also provide sufficient antiwear (AW) and extreme pressure (EP) properties for the whole transmission system, especially for the spiral bevel ring and pinion gears in the axles. The AW/EP additives must not be so active as to cause corrosion in the tractor's hydraulic system, where pumps containing alloys of copper can be present (Refs. 2 and 4).

Rapeseed and sunflower oils are currently used in Europe for the formulation of the biodegradable lubricants. In the present work, the antiwear and extreme pressure properties of formulated rapeseed and high-oleic sunflower-based UTTO oils, synthetic ester-based UTTO oil, synthetic ester gear oil, and mineral-based UTTO oil are investigated on FZG test equipment. The selected formulated vegetable-based UTTO oil was further tested on the helical gear test rig.

Sample Preparation

Oil samples. We have formulated two different vegetable-based UTTO oils for the investigations. The first formulation is based on the rapeseed base stock, while the second base stock is derived from a genetically modified sunflower plant with a high oleic content. The same additive system is used for both formulations. The properties of these two fully formulated vegetable-based UTTO oils were compared with a commercially available mineral-based UTTO oil, a fast biodegradable synthetic-based UTTO oil and a synthetic ester-based gear oil (see Table 1).

The test UTTO oils have a kinematic viscosity between 9 mm²/s and 11 mm²/s at 100°C. This viscosity offers sufficient thickness to promote good gear protection and is still suitable for the hydraulic system. The synthetic ester G has a viscosity two ISO grades higher than other test oils and is suitable as a gear oil only.

The main difference between the vegetable base stocks for R and S formulation lies in fatty acids content (see Table 2). The high-oleic sunflower base stock is derived from a genetically altered plant and possesses a significantly higher

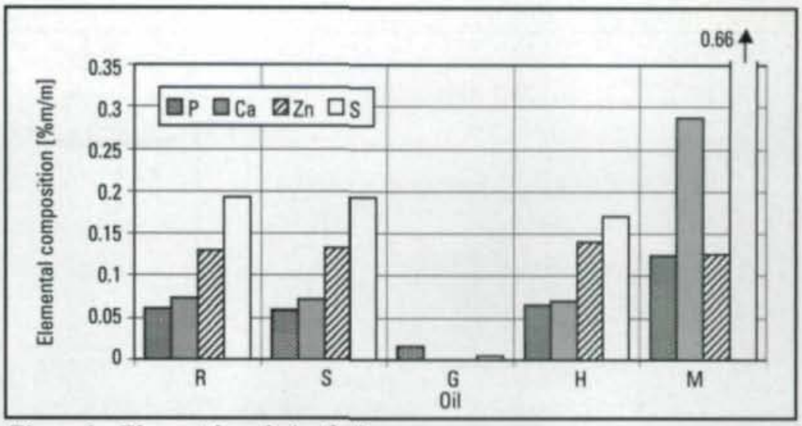
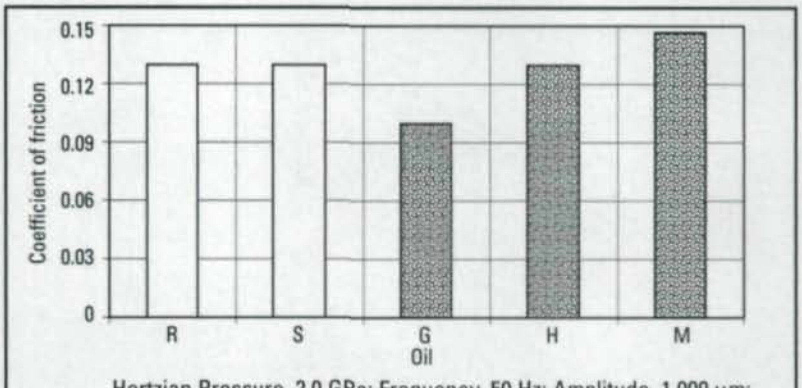


Figure 1—Elemental analysis of oils.



Hertzian Pressure, 2.0 GPa; Frequency, 50 Hz; Amplitude, 1,000 μm; Speed, 0.1 m/s; Temperature, 50°C; Duration, 120 min.

Figure 2—Friction coefficient mean values.

content of oleic acid than rapeseed oil. Due to the higher saturation, the high-oleic sunflower oil has better oxidation stability than the rapeseed oil.

Elemental analysis of additives. Spectrometry via ED-XRF (energy disperse X-ray fluorescence) has been used to obtain the elemental composition of additives for the test oils (see Figure 1). The elemental composition of additives is quite similar for the formulated vegetable-based UTTO oils R and S and the reference synthetic ester-based UTTO oil labeled H. The mineral UTTO oil labeled M contains a significantly higher level of calcium and sulfur than any other test oil. The synthetic ester-based gear oil labeled G shows a low amount of additive concentration compared with the UTTO test oils.

Preselection Experiments and Test Results

SRV test results. The coefficient of friction measurements have been performed on an SRV high frequency test device. SRV stands for the German words "Schwingung" (oscillation), "Reibung" (friction) and "Verschleiss" (wear). The device produces linear oscillating motion of a ball on a flat specimen under boundary lubricating conditions. A thin layer of lubricant is spread over the flat specimen before each test. On the SRV test rig, just the friction coefficient at

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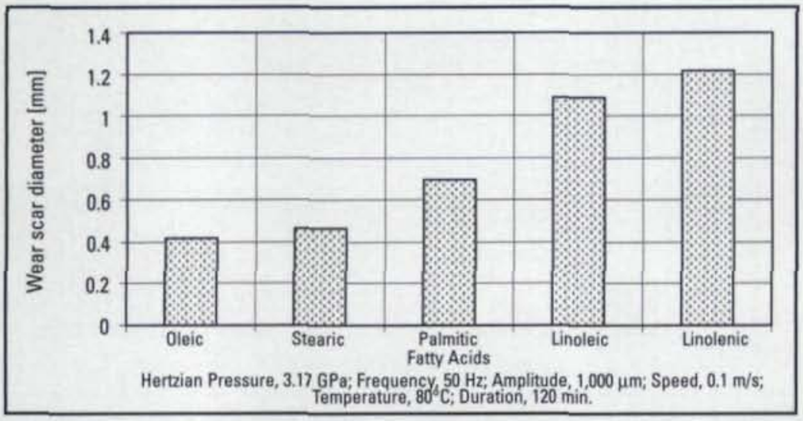

Figure 3—Antiwear properties of fatty acids.

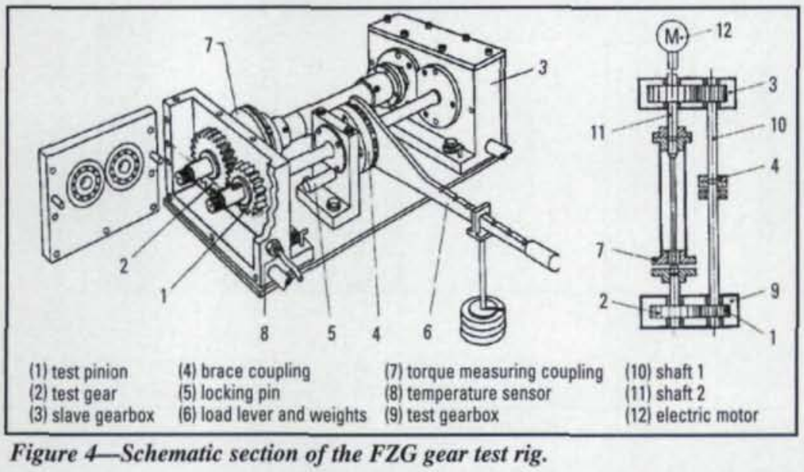

Figure 4—Schematic section of the FZG gear test rig.

sliding motion was measured. The test rig configuration and the test specimens are described in DIN 51 834 T2 (Ref. 5).

Figure 2 shows the results of friction coefficient measurement of the test oils. The mineral-based UTTO oil M shows the highest value of average friction coefficient. The biodegradable oils exhibit less friction, especially the test oil G. Biodegradable UTTO oils with similar elemental compositions of additives—R, S, and H—obtain almost the same value for friction coefficient.

Five different fatty acids contained in the vegetable oils were tested on the SRV test device to demonstrate their antiwear properties (see Fig. 3). The oleic fatty acid proved to have the best antiwear properties at the selected parameters. Antiwear properties and good oxidation stability make the oleic fatty acid the most desired fatty acid in the vegetable lubricant oil formulation.

FZG test results. The FZG gear test rig is commonly used to evaluate scuffing load capacity, pitting resistance and slow-speed, high-load wear resistance. Experiments are based on a failure of a standard gear set, lubricated with the test oil under specific test conditions, using the test rig illustrated in Figure 4 (Refs. 6–7).

The load-carrying capacity of lubricants was investigated by using the standard FZG A/8.3/90 test procedure. The test oil is subjected to a load, increasing by stages, until the scuffing failure criterion has been reached. Twenty millimeters of tooth scuffing indicate a test failure. The failure load stage is reported as a result (Ref. 6–7).

Investigations of the pitting resistance were performed on the FZG gear test rig in the standard pitting test C/8.3/90. After a two-hour run-in at load stage 6 (135.3 Nm), the test is run at load stage 9 (302 Nm) until the failure criterion is recorded. The number of pinion load cycles when the critical damage of the tooth flanks occur is reported as a result (Ref. 8).

UTTO oils are intended to lubricate transmissions and gearboxes of the tractors. In such systems, high temperatures, high loads and low speeds are very common conditions. The primary mode of failure observed with the spiral bevel gearing is scuffing, while the planetary units encounter normal abrasive wear. There are a number of methods to evaluate scuffing, but the primary concern of this investigation is to simulate normal rubbing wear of the planetary gears. In the slow-speed, high-load wear test, the C-type gears were used to reduce the sliding velocity and consequently the probability of scuffing. The test procedure is divided into two stages. The test gears are weighed before and after each stage and the weight loss associated with wear is recorded as a result which indicates the lubricant antiwear performance (Ref. 9–10).

The main FZG test conditions are summarized in Table 3.

The results of the FZG tests are summarized in Table 4. The best results on the FZG test rig were obtained with the synthetic ester-based oil G, which has a viscosity that's two ISO grades higher than other oils.

The results of scuffing load capacity for UTTO oils show better scuffing resistance for the formulated high-oleic sunflower-based oil S, which passed the 11th load stage while the other vegetable-based formulation R and reference UTTO oils passed the 10th load stage. Generally, UTTO oils exhibit a scuffing load stage between 9 and 11; therefore all tested oils meet these requirements (Ref. 11).

The results of pitting investigations show superior pitting resistance of the vegetable- and the synthetic esters-based oils, compared with the mineral-based UTTO oil. The high-oleic sunflower oil S showed very good pitting performance among the formulated UTTO oils.

The results of slow-speed, high-load wear investigations indicate no significant difference in wear rates among the UTTO oils. All test oils show low wear rates in a slow-speed, high-load FZG test.

On the basis of the FZG test results, we selected the formulated high-oleic sunflower-based formulation S for further investigations.

Helical Gear Test

Helical gear test rig. A helical gear test rig was used in order to demonstrate viscosity stability, antiwear properties, oxidation resistance and seal compatibility of the formulated high-oleic sunflower-based UTTO oil labeled S in controlled laboratory conditions. Through periodic sampling of the lubricant and used oil analysis, the condition of the test oil and parts of the gearbox were determined.

The helical gear test rig is schematically shown on Figure 5. The AC drive motor with frequency regulation runs a test gear-unit which is lubricated with the formulated high-oleic sunflower-based UTTO oil S. For load simulation, as a brake, the DC generator and electric heaters are used. The DIN CK60 pinion and DIN CK45 gear, case hardened to 60–62 HRC and not undercut, were used during the test. These helical gears had face widths of 30 mm, normal modules of 2.5 mm, and the drive pinion had 31 teeth meshing in a 1:1.5 ratio. The test rig was run continuously 12 hours per day at a constant load of approximately 60 Nm of torque. The oil temperature was maintained in the range of 78–82°C. A pair of helical gears was rotated until the lubricant deteriorated.

Oxidation of lubricants is normally measured by total acid number (TAN) and viscosity increase. New oils have an initial TAN, therefore the increase over the initial value measures oxidation. If the TAN exceeds 2.0 mg KOH/g over the original value, the oil should be changed. (The unit "mg KOH/g" is the quantity in milligrams of potassium hydroxide (KOH) needed to neutralize the acid constituents in one gram of lubrication oil.) A strong indicator of oil degradation is also its increase in viscosity. Normally a 20% increase in viscosity is a warning that the oil is reaching the end of its useful life.

Oil condition. The top line on the graph in Figure 6 represents kinematic viscosity of the high-oleic sunflower-based UTTO oil S, measured at 40°C. After initial shear-down, the viscosity is stable until 600 working hours, when a slight increase is observed. The bottom TAN line shows three distinct sections: initial increase is followed by a stable value until the sudden rise at 650 operating hours, which indicates the oil deterioration.

Gearbox condition. Oil in the gearbox could be a very useful condition monitoring media. If we can separate the debris from the oil, we can identify and track an abnormal wear condition without tearing down the equipment. Wear particles contained in the lubricating oil carry detailed and important information about the condition of the oil-wetted components in the gearbox. If no excessive wear is observed, then this indicates that the effective lubrication in the gearbox is maintained

Table 3—FZG Test Conditions.

Parameters	Unit	Scuffing	Pitting	Wear	
				Stage I	Stage II
Test gears type		A	C	C	
Load stage		*	9	10	
Gear torque	Nm	*	302	372.6	372.6
Circumferential speed	m/s	8.3	8.3	0.35	0.20
Pinion rotational speed	rpm	2,170	2,170	93	53
Running time	hour	1/4–one stage	until failure	20	30
Sump temperature	°C	90, at start	90	121	
* ... incrementally increased load.					

Table 4—FZG Test Results.

Oil	Scuffing A/8.3/90 (FZG load stage)	Pitting C/8.3/90 (cycles of pinion)	Slow speed wear C/0.35–0.25/120 (weight loss in mg)
R	10	$13.96\ 10^6$	11/14[1)]
S	11	$26.75\ 10^6$	19/22[1)]
G	>12	$30.00^*\ 10^6$	
H	10	$15.66\ 10^6$	
M	10	$7.70\ 10^6$	13/13[1)]
* ... pitting test was stopped, but critical failure did not occur. 1) ... Stage I/Stage II.			

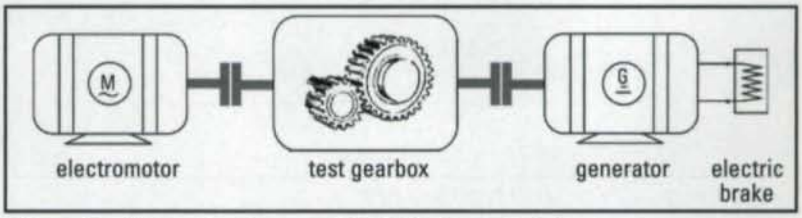

Figure 5—Schematic diagram of the helical gear test rig.

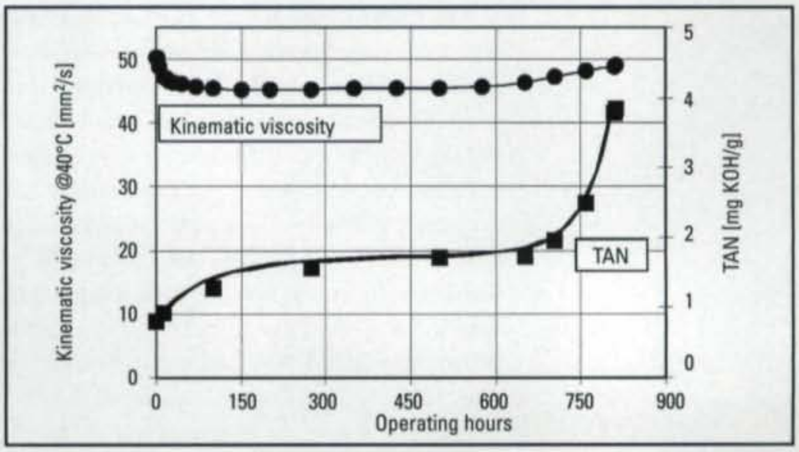

Figure 6—Trend values for viscosity and TAN.

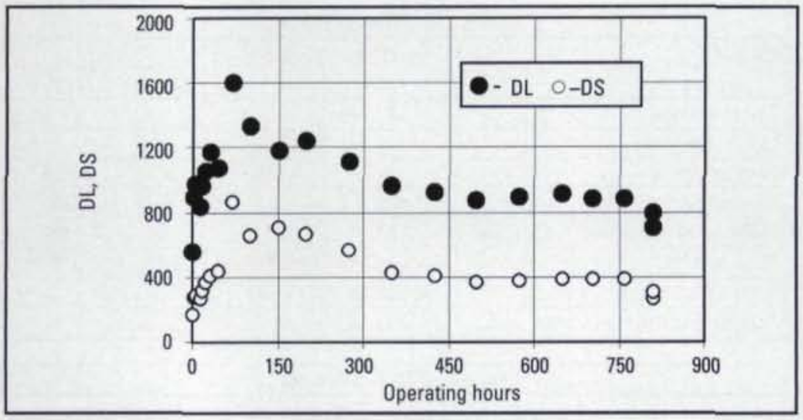

Figure 7—Quantitative ferrography readings.

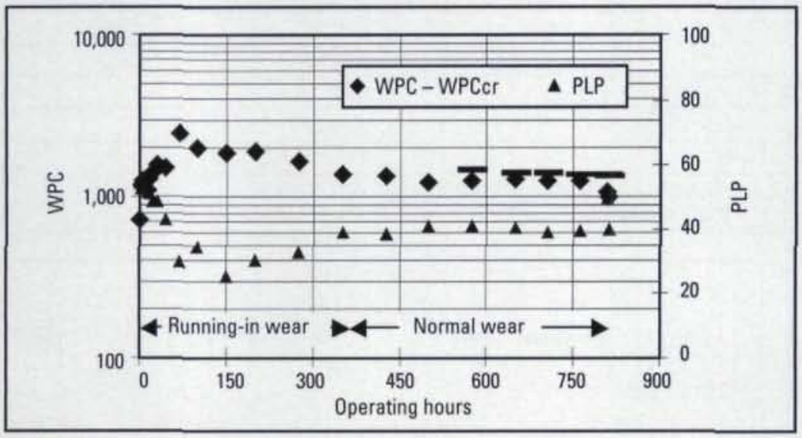

Figure 8—Trend values for WPC and PLP.

during the operation.

The method used for the quantitative evaluation of the wear particle concentration was direct reading (DR) ferrography. DR ferrography magnetically separates wear particles from lubricants and optically measures the relative concentration of ferrous particles present in the oil sample. The instrument is able to detect particles in the length range of 1–300 microns. The output of the DR ferrography consists of two digital readings, a DL (density large) for large particles (> 5 µm) and a DS (density small) for small particles (1–2 µm) (Ref. 12).

Figure 7 shows the measured values for DL and DS for high-oleic sunflower-based oil S as a function of operating time. These readings can be processed in several ways to allow easier identification of an abnormal wear mode. Two such ways are briefly described (Ref. 12):

Total Wear Particle Concentration (*WPC*)

$$WPC = DL + DS \qquad (1)$$

Percentage of Large Particles (*PLP*)

$$PLP = [(DL - DS)/WPC]*100 \qquad (2)$$

Although the magnitude of the *WPC* is important, the change from historical values is the indicator of machine wear condition. Quantitative ferrography is a trending tool and not a particle count. In normal operating conditions, a baseline of normal wear may be established and the average of *WPC* values can be calculated.

The average of Wear Particle Concentration value (WPC_{mean})

$$WPC_{mean} = 1/n\sum WPC^* \qquad (3)$$

where WPC^* means that only normal wear data are summed (outliers are excluded).

The *WPC* value should not exceed the value of an alarm limit—the critical wear particle concentration that is based on the WPC_{mean} value and standard deviation of the normal samples (outliers are excluded).

The critical Wear Particle Concentration (WPC_{cr})

$$WPC_{cr} = WPC_{mean} + 2\sigma \qquad (4)$$

where σ is a population's standard deviation.

The most informative method for DR results representation is a plot of *WPC* and *PLP* in the same graph, because an increase in both *WPC* and *PLP* is the best indication of an abnormal wear condition. Figure 8 shows the calculated values for *WPC* and *PLP*, which are plotted over time. The *WPC* value shows an initial sharp rise through a running-in process, during which the quantity of wear particles quickly increases and then settles to a lower value after 350 operating hours, when a normal wear period is beginning. The *WPC* and *PLP* values remain relatively constant in the normal running operation period, because the gearbox wear reaches a state of equilibrium in which the particle loss rate equals the particle production rate.

An alarm limit for severe wear WPC_{cr} can be calculated for the last six samples, because at least three data in the normal wear period are necessary to determine WPC_{mean} and a population's standard deviation σ before the WPC_{cr} for the first point can be calculated. All *WPC* values are beyond the alarm limit, therefore the gearbox operates in the normal wear mode. The test was ended when the test oil started to deteriorate and before the gear failures occurred.

To avoid leakage problems, seals used in a gearbox should be compatible with the test oil. Fluoroelastomer (Viton®) was used as a seal material for the test. The seals were inspected after the test and no change in the geometry was found.

Discussion

Two formulated vegetable-based UTTO oils, two synthetic esters, and a mineral UTTO oil were investigated with respect to their gear protection properties. The study has shown that the gear protection properties of the formulated vegetable-based oils are comparable with the corresponding mineral-based oil. However, the FZG investigations show significantly better results for pitting resistance for the vegetable-based oils and synthetic esters.

The vegetable-based oils and synthetic esters exhibit very good lubricity in boundary lubrication conditions, because the synthetic esters contain organic straight chain compounds with polar end groups—fatty acids (see Table 2). The polar nature of the biodegradable oils gives them a greater affinity for metal surfaces than nonpolar mineral oils. The need for antiwear additives is reduced. Therefore, with lower concentration of the additives, both the vegetable- and synthetic-based oils show lower friction coefficients than the mineral test oil (see Figs. 1–2).

The FZG test results indicate that oil viscosity has a strong influence on scuffing performance and pitting resistance (Ref. 13). The best results are obtained with the synthetic-based gear oil, which has a viscosity two ISO grades higher than the UTTO test oils. At the same time, this synthetic ester has the lowest concentration of additives. The gear oils are generally of higher viscosity than hydraulic oils, but UTTO oil is a multipurpose lubricant that has to meet both the gear protection and hydraulic system requirements.

Vegetable- and synthetic-based oils have excellent viscosity properties. Their viscosity indexes (VI) exceed 195, while the VI for mineral UTTO oil equals 150 (see Table 1). The higher VI allows for the formation of the thicker lubrication film and for better separation of the contact surfaces at working temperatures (Ref. 14–15). The UTTO oils are of the same ISO grade viscosity, but tests—especially for pitting resistance—show a great differentiation in the results. Besides the lubricant viscosity, the lubricant base stock has a great influence on pitting resistance, while the additive type and concentration have only a minor influence. The FZG pitting test conditions correspond to a Hertzian contact point pressure of 1.65 GPa, while contact pressure at the SRV test is 2.0 GPa. The pitting test results closely follow the SRV investigations. Measured friction coefficient (Figure 2), the fatty acid content (Table 2), and antiwear properties of fatty acids (Figure 3) determine the pitting performance. The higher number of cycles until failure is thus a function of the lower sliding friction at the point of contact and, consequently, lower tangential stresses on the surface, which can efficiently prevent fatigue failure associated with surface-initiated cracks (Ref. 16).

Summary

The vegetable-based UTTO oil formulations have the advantages of having a green source of oil and lower cost than a biodegradable synthetic UTTO oils. Agricultural equipment is ideally suited to use vegetable-based oil because the tractor is lubricated by oil derived from a plant growing in the field that has been cultivated by the same equipment.

The vegetable-based oil formulations exhibit low changes of viscosity with temperature. The viscosity index improvers can be used to enhance the viscosity index of mineral-based oils, but it is advantageous to have a high viscosity index "built into" the base oil molecule itself.

The tests show that the pitting resistance of vegetable- and synthetic ester-based UTTO oil formulations is significantly better compared with the mineral UTTO oil.

The investigations on the helical gear test rig have shown that the test high-oleic sunflower-based UTTO oil formulation derived from the genetically modified plant provides sufficient gearbox lubrication for 600 operation hours at a maintained oil temperature in the range of 78–82°C. ○

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Design of High Contact Ratio Spur Gears Cut With Standard Tools

Evgueni Podzharov and Almantas Mozuras

Abstract

In high precision and heavily loaded spur gears, the effect of gear error is negligible, so the periodic variation of tooth stiffness is the principal cause of noise and vibration. High contact ratio spur gears can be used to exclude or reduce the variation of tooth stiffness.

A simple method of designing high contact ratio spur gears cut with standard tools of 20° profile angle is presented in this paper. It consists of increasing the number of teeth on mating gears and simultaneously introducing negative profile shift in order to provide the same center distance.

Computer programs to calculate static and dynamic transmission error of gears under load have been developed to evaluate dynamic properties of gears. The analysis of gears using these programs showed that gears with high contact ratio of 1.96 have much less static and dynamic transmission error than standard gears.

Introduction

The periodic change of tooth stiffness, gear errors and friction force impulse at the pitch point are the principal causes of vibration and noise in gears. In high precision and heavily loaded gears, the effect of gear errors is insignificant, so the periodic variation of tooth stiffness and friction force impulse are the most significant causes of noise and vibration.

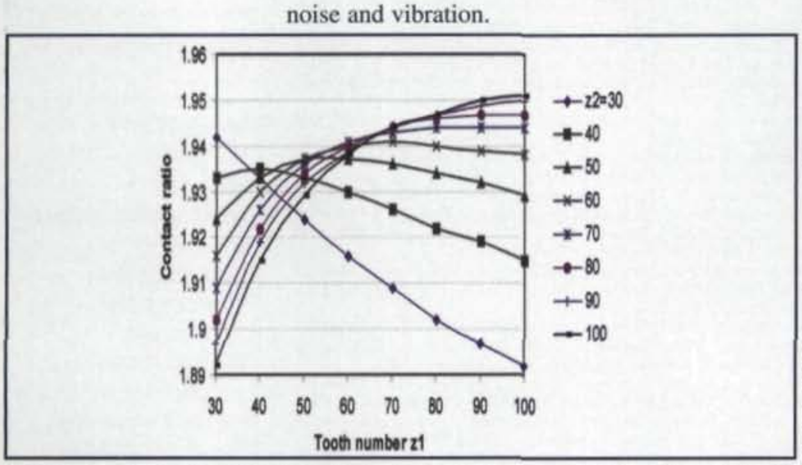

Figure 1—Contact ratios of HCR gears cut with a standard rack tool of 20° pressure angle.

High contact ratio spur gears can be used to exclude or reduce the variation of tooth stiffness. Kasuba (Ref. 1) established experimentally that the dynamic loads decrease with increasing contact ratio in spur gearing. Sato, Umezawa, and Ishikawa (Ref. 2) demonstrated experimentally that the minimum dynamic factor corresponds to gears with a contact ratio slightly less than 2.00 (1.95). The same result was found experimentally by Kahraman and Blankenship (Ref. 3) and theoretically by Lin, Wang, Oswald, and Coy (Ref. 4).

The increase in contact ratio can be implemented by decreasing pressure angle and increasing tooth height. In the previous works (Refs. 2–4), the increase in the tooth contact ratio was implemented by increasing tooth height. Vulgakov (Ref. 5) proposed a method of designing nonstandard gears in generalized parameters and found that spur gears with a contact ratio of more than 2.0 and a pressure angle more than 20° worked considerably quieter. Rouverol and Watanabe (Ref. 6) proposed maximum-conjugacy gearing, which has a low pressure angle at pitch point and increases slowly at the tip and at the root. The measurements also showed a considerable reduction in the noise level compared to standard gears.

Nevertheless, the use of standard pressure angle and standard tool is preferable. In the author's previous work (Ref. 7), a simple method of designing high contact ratio spur gears with a standard basic rack of 20° profile angle was presented. This method allows us to design gears with a contact ratio of nearly 1.95. In this paper, an analysis of static and dynamic transmission error in standard gears with 20° pressure angle and high contact ratio gears is given.

Method of Design of High Contact Ratio Spur Gears Cut with a Standard Tool

An increase of the contact ratio can be carried out by the method (Ref. 7) in the existent gears by incrementing the sum of numbers of teeth in the gears by two and then simultaneously introducing a negative displacement of the gear tooth profile in the following way. The operating transverse pressure angle is equal.

$$\alpha_{tw} = \arccos(a \cdot \cos\alpha_t / a_w) \qquad (1)$$

where α_{tw} is the operating transverse pressure angle;

α_t is the transverse pressure angle;

$a = 0.5m_t + (z_1 + z_2)$ is the standard center distance;

a_w is the operating center distance;

m is the module;

$z_{1(2)}$ is the tooth number of pinion (gear).

Then, the sum of profile shift coefficients is determined by

$$\Sigma x = (z_1 + z_2 + 2)(inv\alpha_{tw} - inv\alpha_t)/2tg\alpha_n \qquad (2)$$

where

inv() is the involute function of the angle;

α_n is the normal pressure angle.

The profile shift coefficients of the pinion $x_1$and gear x_2 can be selected, balancing specific sliding (Ref. 8). The conditions of interference absence must also be checked.

Contact ratio of a spur gear:

$$\varepsilon_\alpha = Z / p_b \qquad (3)$$

where Z is the active length of the line of contact;

$$p_b = \pi\, m \cos\alpha - \text{base pitch} \qquad (4)$$

The calculated values of contact ratio for the range of tooth numbers from 30–100 for both gears are presented in Figure 1. These values vary from 1.89 to 1.95.

The real value of contact ratio is slightly higher due to the consequence of the tooth deformation under load and edge contacts at the beginning and the end of contact. The absence of tooth undercut gives the range of values of tooth numbers (approximately $z > 26$) for these high contact ratio gears cut by a rack-type tool of 20° profile angle.

The pressure angle in HCR gears is lower than in standard gears. The lower pressure angle leads to greater sliding velocity and increases the risk of scuffing. The negative shift coefficient which must be introducted in this type of gear decreases the bending resistance of a pair of gear, but because of better load distribution between two pairs of teeth, this can be partly compensated. So, the HCR gears must be checked for bending and scuffing resistance of teeth.

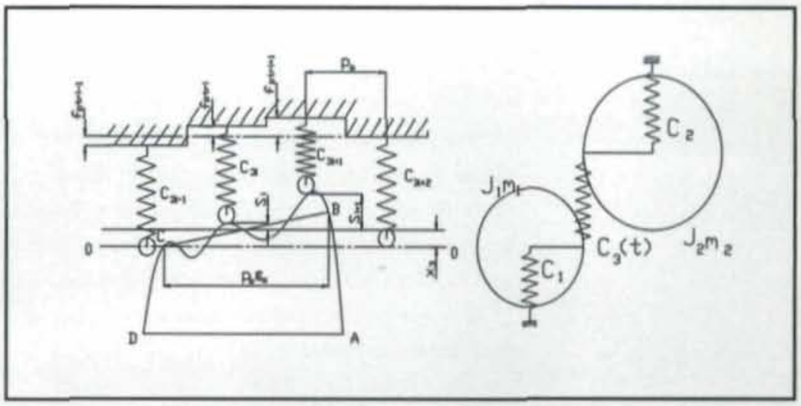

Figure 2—Gear dynamic model: (a) model of tooth engagement, (b) dynamic model.

Determination of Static Transmission Error in Spur Gears

The tooth engagement model presented in Figure 2a shows the influence of load distribution between teeth on effective gear errors. Following Yelle and Burns (Ref. 9) and Remmers (Ref. 10), the tooth profile is represented as a slide and the teeth as springs with rollers. The pitch error is modeled as a step base for spring and profile error, and base pitch error is displayed as an undulated inclined slide surface. From the analysis of this model, it is evident that having finish tooth errors, tooth edge contact designed as slopes AB and CD on the slide, the tooth load is not distributed uniformly between teeth.

Here C_{3i} is the stiffness of the *i-th* pair of teeth;

S_i is the kinematic error of the *i-th* pair, composed by base pitch error and profile error;

f_{ptri} is the circular pitch error difference of the *i-th* pair of teeth;

x_3 is the static transmission error under load;

f'_{iri} is the local kinematic error of the *i-th* pair of teeth;

f_{pb} is the base pitch error; and

ε_a is the contact ratio.

In Figure 2a, the positive error is directed outward from the slide and the negative error is directed into the slide. As a consequence, a positive error corresponds to spring compressions. With these definitions, the tooth deflection, which appears as the result of the action of positive tooth error, is also positive. Then, the transmission error x_3, can be expressed by current errors of several pairs of teeth and its deflections x_{3i} in the following way:

$$x_{3i} = x_3 + f'_{iri} \qquad (5)$$

where

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$$f'_{iri} = S_i + f_{ptri} \tag{6}$$

The kinematic error during tooth edge contact at the beginning and the end of tooth mesh (the sections AB and CD on the slide, see Fig. 2a) can be evaluated using the method exposed by Seireg and Houser (Ref. 11). The error in the section BC of the slide can be determined as a sum of cosine and linear functions.

The normal force between teeth is equal.

$$F_n = \sum_{i=1}^{n} C_{3i} x_{3i} = \sum_{i=1}^{n} C_{3i}(x_3 + f'_{iri}) \tag{7}$$

Transforming Equation 7, we have

$$x_3 = (F_n - \sum_{i=1}^{n} C_{3i} f'_{iri}) / \sum_{i=1}^{n} C_{3i} \tag{8}$$

Using formulas in Equations 5–8, one can find

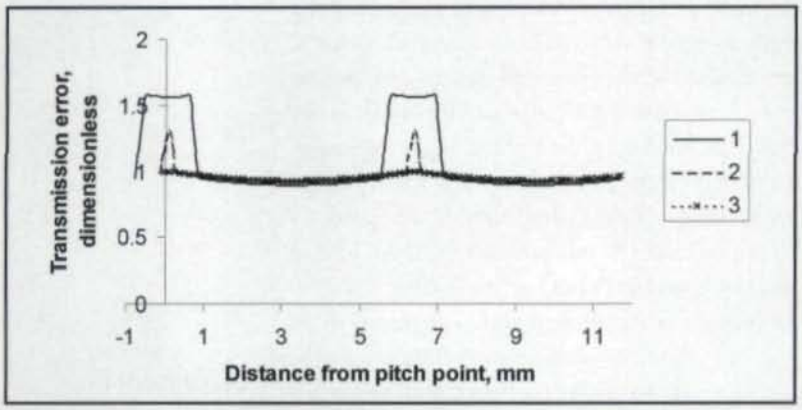

Figure 3—Static transmission error in gears: 1–standard gear, 2–high contact ratio gear with standard tooth height, 3–high contact ratio gear with increased tooth height.

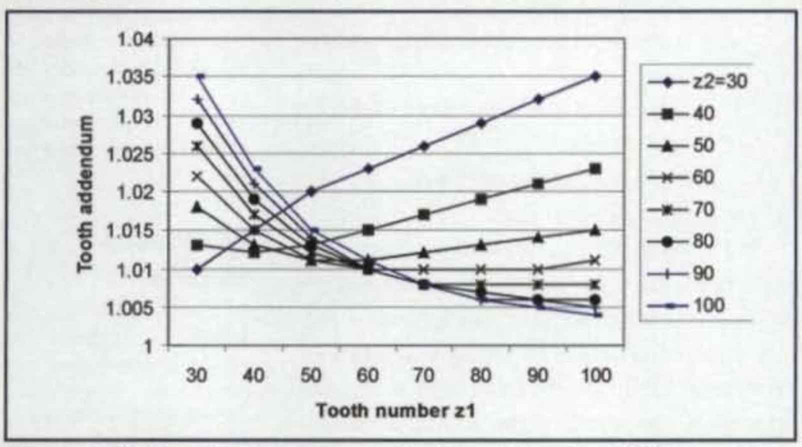

Figure 4—Tooth addendum corresponding to contact ratio of 1.96.

Table 1

No.	m, mm	a_w, mm	z_1	z_2	α_{tw}	x_1	x_2	h_a^*
1	3.0	135	40	52	20°	0.000	0.000	1.000
2	3.0	135	41	53	16.24°	–0.393	–0.520	1.000
3	3.0	135	41	53	16.24°	–0.393	–0.521	1.018

the kinematic error of tooth engagement under load at any moment in time, which is static transmission error. At first, the calculation is done with $n = 1$, assuming that $f'_{ir1} = f'_{ir\,max}$ is the maximum tooth error in the tooth engagement. Then, if $x_3 + f'_{ir2} > 0$, we accept $n = 2$ and continue the calculation. The method developed by Weber and Bonaschek (Ref. 12) is used to determine the tooth stiffness at any position. Calculation of static transmission error can be used in designing a gear with the purpose of selecting geometric and precision parameters, which assures a minimum excitation of vibration in gear engagement.

Analysis of Static Transmission Error of Standard and High Contact Ratio Spur Gears

The geometric parameters of gears analyzed here are shown in Table 1.

Here m is the module of gears;

a_w is the center distance;

$z_{1(2)}$ is the tooth number of pinion (gear);

α_{wt} is the operating transverse pressure angle;

$x_{1(2)}$ is the profile shift coefficient of pinion (gear);

$h_a^* = h_a/m$ is the addendum coefficient; and

h_a is the addendum, pinion (gear).

The results of the calculations of static transmission error for the gears with parameters shown in Table 1, without tooth errors, are presented in Figure 3. In the graphics of Figure 3, the values of transmission error are presented in dimensionless form, where $x_3^* = x_3/x_{30}$ where $x_{30} = F_n/C_{30}$ mean tooth deflection, and C_{30} = mean tooth stiffness of gear mesh.

It can be seen from the figure that the variation of transmission error of the gear with standard pressure angle and standard tooth height (without tooth errors) has a stepped form (curve 1). The stepped form of transmission error is due to a change in tooth stiffness between one-pair tooth contact zone and two-pair tooth contact zone. The static transmission error of this form can excite high-level vibration. The gear pair with high contact ratio and standard tooth height (Equation 2) has an increased tooth contact ratio (ε_a = 1.93). This contact ratio was obtained by increasing the number of teeth of the pinion and the gear each by one, and introducing negative profile tooth shifts (x_1 = –0.393 y x_2 = –0.52), according to the method proposed (Ref. 7).

The static transmission error here is not large, but it has a peak that can cause vibration excitation. To further reduce the transmission error, we must increase tooth addendum by 0.018 of its value (h_a^* = 1.018). The contact ratio of this gear is 1.96. In this case, the static transmission error almost completely disappears (curve 3 in Figure 3).

The increments of tooth addendum which correspond to a contact ratio of 1.96 were calculated by iteration and presented in Figure 4. For the range of tooth numbers from 30–100, the required increments of addendum vary from 1.005–1.035 of its value, and these increments can be made using standard tooth tools.

Dynamic Transmission Error

The dynamic model of a gear pair is presented in Figure 2b. Here, $C_{1(2)}$ is stiffness of support of a pinion (gear), $C_3(t)$ is stiffness of tooth mesh, $m_{1(2)}$ is mass of pinion (gear), $J_{1(2)}$ is moment of inertia of pinion (gear). These parameters have the following values $C_1 = 45.5$ MN/m, $C_2 = 81$ MN/m, $C_{30} = 36.3$ MN/m, $m_1 = 3.38$ kg, $m_2 = 4.5$ kg, $J_1 = 0.297 \cdot 10^{-3}\ kg\ m^2$, $J_2 = 0.835 \cdot 10^{-3}\ kg\ m^2$. The dissipative coefficient was expected to be 0.05. In this dynamic model, the tooth engagement is represented by the structural model shown in Figure 2a. The whole dynamic model is described by three differential equations with periodic functions that were solved by a program based on the use of the Runge-Kutta method (Ref. 13).

The results of the solution of the equations for three types of gears without errors are shown in Figure 5, for one period of stationary vibrations, and for tooth mesh frequencies from 0–3,000 Hz. The dynamic transmission error is also represented here in dimensionless form.

It can be concluded from the figures that:

1. Standard gears have very high amplitudes of vibrations (Figure 5a).
2. Amplitudes of vibrations diminish in high contact ratio gears with standard tooth height (Fig. 5b).
3. Vibrations completely disappear in the case of high contact ratio gears with slightly increased tooth height and contact ratio equal to 1.96 (Fig. 5c).
4. Curves at zero frequency are identical to the static transmission error curves in Figure 3.

Conclusions

Methods and programs have been developed to calculate static and dynamic transmission errors under load in spur gears. A tooth mesh of periodic structure, which takes into account deflection and errors of each pair of teeth in the enagement, is used.

The analysis of static and dynamic transmission errors in high-precision, heavy-loaded standard gears, high contact ratio gears of standard tooth height and high contact ratio gears with slightly increased tooth addendum showed that, in the last type of gears, the static and dynamic transmission errors can be almost completely excluded. ○

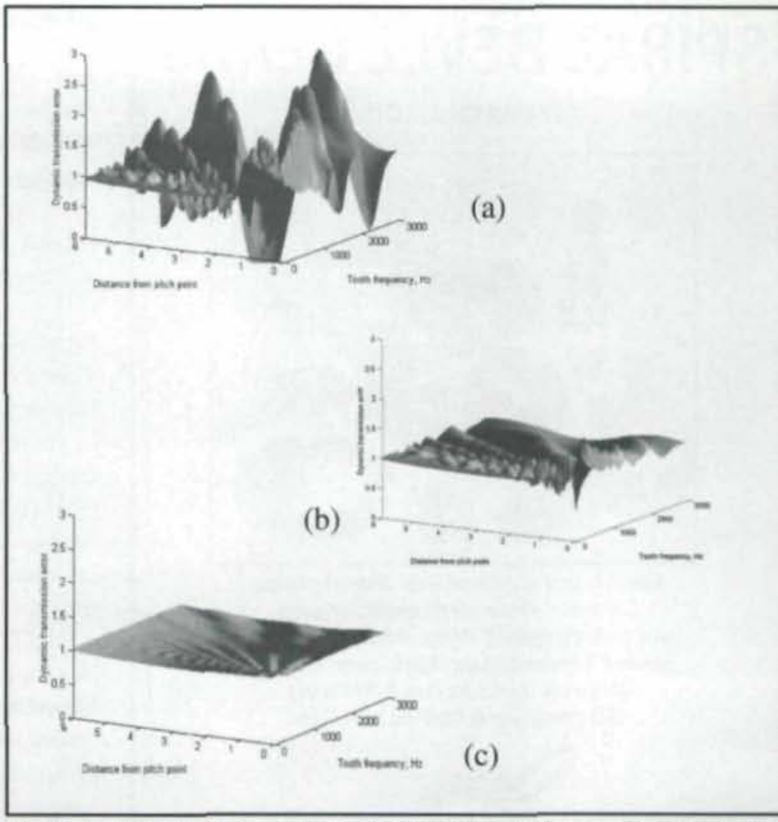

Figure 5—Dynamic transmission error: (a) Standard gear, (b) High contact ratio gear with standard tooth height, (c) High contact ratio gear with contact ratio of 1.96.

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TECHNICAL CALENDAR

July 17–18—12th Annual Gear Failure Analysis Seminar. Big Sky Resort, Big Sky, MT. Sponsored by the AGMA, this 30-participant seminar examines macropitting, micropitting, scuffing, tooth wear and breakage. $625 for AGMA members, $795 for non-members. For more information, contact the AGMA by telephone at (703) 684-0211 or by e-mail at *tech@agma.org*.

September 2–6—ASME/AGMA 2003 International Power Transmission and Gearing Conference. Marriott Chicago Downtown, Chicago, IL. Held once every four years, this event is sponsored in conjunction with the ASME International Design Engineering Technical Conferences, along with seven other conferences. $390–$680, depending on the level of membership and the number of conferences attended. For more information, contact ASME via the Internet at *www.asmegear.eng.ohio-state.edu/home.html*.

September 15–19—AGMA Training School for Gear Manufacturing. Richard J. Daley College, Chicago, IL. Offers classroom and hands-on training in gearing and nomenclature, principles of inspection, gear manufacturing methods, and hobbing and shaping. Designed for employees with six months' experience in setup or machine operation. $650 for AGMA members, $775 for non-members. For more information, contact the AGMA by telephone at (703) 684-0211 or by e-mail at *fentress@agma.org*.

September 15–17—ASM Heat Treat 2003. Indiana Convention Center, Indianapolis, IN. Showcasing the latest furnaces, controls and testing equipment, the show is co-located with the ASM International Surface Engineering Expo. Attendees are welcome to take part in the Heat Treat U seminars. Prices range from $125–$650, depending on ASM membership as well as the level of conference participation. For more information, contact ASM International on the Internet at *www.asminternational.org/heattreat/expo.htm*.

September 16–18—Accurate Diagnosis of Gear Problems. 49'er & Suites, Jackson Hole, WY. The course includes: How Each Cycle of Gearmesh Frequency is Generated; Gear Vibration Theory; Problems that Generate Harmonics of Gearmesh Frequency; Bent Shaft; Defective Teeth & Broken Gears, and more. $1,195 includes *The Gear Analysis Handbook* by James I. Taylor. For more information, contact Vibration Consultants Inc. by telephone at (813) 839-2826 or on the Internet at *www.vibcons.com*.

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Prominent Gear Engineer Darle Dudley: 1917–2003

Darle Dudley, an internationally known gear engineer, of San Diego, CA, died April 11 of heart problems and a serious infection. He was 86 years old.

Mr. Dudley was known for his two gear handbooks, which were translated into French, German and Spanish. He was also known for his 19-year chairmanship of the AGMA Aerospace Gearing Committee.

Born April 8, 1917, in Salem, OR, Mr. Dudley started his career in 1940. That year, he graduated from Oregon State University with a bachelor of science degree in mechanical engineering and joined General Electric Co. in Lynn, MA.

In 1952, Mr. Dudley met Martin Hartman, starting a 51-year friendship and association. Over the years, the two men worked on several gear projects together, including work on the Apollo space program.

"Darle worked on just about every kind of gear you can think of," Hartman says.

In 1964, Mr. Dudley left GE as manager of advanced gear engineering and went to Mechanical Technology Inc. of Schenectady, NY, as manager of mechanical transmissions. In 1967, he left MTI to serve as chief of gear technology at Solar Turbines Inc. of San Diego, CA.

In 1978, he retired early and founded the gear consultancy Dudley Engineering Co. Inc. Dudley Engineering consulted with companies involved in mining, turbine, aerospace and industrial production.

"We consulted in the gear business on everything from sewage plants to satellites," says Danny F. Smith, a 15-year associate and friend.

Smith met Mr. Dudley in 1988, through the Yellow Pages. Smith was finishing college and looking for a job—"It was a cold call." He worked for Mr. Dudley for 9 years.

"He was a brilliant man," Smith says, "but he was also kind and funny."

Today, Smith is senior principal design engineer at Solar Turbines Inc. Mr. Dudley recommended Solar Turbines, one of his former employers, to Smith.

In 1991, Mr. Dudley created a second consultancy, Dudley Technical Group Inc. of San Diego and transferred most of Dudley Engineering's assets to it. In 1997, he suffered a stroke, forcing his retirement from the gear industry.

Dudley Technical continues today in Doylestown, PA, under its new owner and president, Mr. Dudley's 25-year associate and friend, Mike Broglie.

Broglie credits Mr. Dudley with gearing knowledge as thorough as anyone he's ever met.

"It's extremely broad based in the different types of gearing that he worked on," Broglie says, "and he worked on all types of gears and all types of gear applications."

Broglie most remembers Mr. Dudley's speaking style, though. He says Mr. Dudley could convey complex ideas in simple, straightforward language.

As for Mr. Dudley's writing style: "It was identical." Broglie knows; he's read all of Mr. Dudley's books and contributed to *Dudley's Gear Handbook.*

Darle Dudley: 1917–2003

"As a human, he was a gentleman; as a gear engineer, there was no peer."
—Martin Hartman, 51-year friend & associate

Gear Books

Over 40 years, Mr. Dudley created four gear books.

His first book was *Practical Gear Design*. Published in 1954, the handbook was translated into French in 1958 and into German in 1961.

He then edited *Gear Handbook*, which was published in 1962. It was later translated into Spanish and published in Mexico in 1973. His next book was *The Evolution of the Gear Art*, published in 1969.

Mr. Dudley later expanded and revised his *Practical Gear Design*, which was published in 1984 with the title *Handbook of Practical Gear Design*. In 1994, it was republished.

In 1991, his *Gear Handbook* was published as a revised second edition and retitled *Dudley's Gear Handbook*. The 1991 edition was edited by Dennis P. Townsend, president of Townsend

Engineering of Westlake, OH.

Townsend views Mr. Dudley's writings as his greatest contribution to the gear industry, particularly Mr. Dudley's two handbooks: "They're used by just about everyone in the gear industry."

Awards

Mr. Dudley received six awards during his gear career, including three awards from European organizations.

In 1958, he received the AGMA's Edward P. Connell Award., given to people who have provided outstanding service to the gear industry. In 1966, he received the Golden Gear Award from *Power Transmission Design* for outstanding contributions to the gear art.

In 1977, Mr. Dudley received the Medaille D'Argent from the French Institute of Gears and Gear Transmissions for America's contributions to the global advancement of the gear art. In 1979, he was awarded the Worcester Reed Warner Medal from ASME for his engineering contributions to gears.

In 1986, he received a second award, the Medaille D'Or, from the French Institute of Gears and Gear Transmissions. In 1994, he received the Ernst Blickle Award from SEW-Eurodrive-Siftung of Bruchsal, Germany.

Chairmanships

Besides the aerospace gearing committee, Mr. Dudley was a founder and the first chairman of the AGMA Vehicle Gearing Committee. He founded and was first chairman of the Gear Technical Committee of the International Federation for the Theory of Machines and Mechanisms (IFToMM). He also was chairman of the ASME Research Needs Task Force for Gearing and was a member of the ASME Gear Research Institute.

Association Memberships

Mr. Dudley was a member of the American Gear Manufacturers Association (AGMA), the American Society of Mechanical Engineers (ASME, now ASME International), the ASME Gear Research Institute and of IFToMM.

He also was an honorary member of Verein Deutscher Ingenieure (VDI), an association of German engineers.

Mr. Dudley is survived by his wife, Dorothy; daughter, Cheryl Davis; son Curtis Dudley; stepson Robert Burson; sister, Dorine Wenger; nine grandchildren; and two great-grandchildren.

"As a human, he was gentleman; as a gear engineer, there was no peer," Hartman says. "I don't think we're going to replace him."

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Arrow Gear: Spiral Bevel Specialist

Joseph L. Hazelton

James J. Cervinka and Frank E. Pielsticker must've known the future when they named their new business Arrow Gear Co. in 1947. They started out to manufacture gears for hand tools and machine tools, but their business has taken off since then.

"Arrow" suggests flight, speed and weaponry. Today, Arrow Gear of Downers Grove, IL, is a leading manufacturer of gears for aerospace. The company specializes in high-tech, high-quality spiral bevel gears, and its products fly in commercial aircraft, military aircraft, helicopters, rockets and cruise missiles. About 70 percent of Arrow's sales are to aerospace customers.

Arrow's other gears transmit power in ground applications—in robotics, printing presses and off-road equipment, as examples. But, even on the ground, some Arrow gears are used for "flying"—like the ones they made for an Indy race car.

Arrow's course, though, started with a teenager and a summer job. Cervinka has had gears in his blood since high school, when he worked for Chicago gear manufacturer D.O. James Corp. "I'm a hands-on mechanic," he says.

Cervinka also studied engineering in college, but his education was first used in service to his country. During World War II, he served as an engineering officer in the Pacific Ocean, aboard the heavy cruiser U.S.S. Bremerton.

James J. Cervinka, Arrow's chairman and CEO, worked for a gear manufacturer in high school and served in the U.S. Navy in World War II. Both experiences were vital to starting Arrow.

Joseph Arvin joined Arrow in 1972 as manufacturing manager. Four promotions later, he became Arrow's president in 1987. After 31 years, his professional enthusiasm remains: "It's exciting to me to work with gears."

After the war, Cervinka went home to Riverside, IL, just west of Chicago. He was thinking about getting a master's degree, but he had to wait before he could get back into college. So the engineer took a job with American Gear in Clearing, IL, just outside Chicago.

Cervinka expected to be with American about six months, just 'til it was time to go back to college. On the job, he met fellow engineer Pielsticker, who was involved in calculations for spiral bevel gears for American.

Besides a master's degree, Cervinka was thinking about something his father encouraged him to do: Start your own business. "My father had a business before The Depression."

He and Pielsticker got along well, and they had complementing talents: Pielsticker was good at business administration and gear calculations, Cervinka at maintaining machinery and manufacturing gears. If they combined their talents, they could even run a business—so they did.

And they decided that business would manufacture spiral bevel gears. Cervinka explains that those gears were highly technical and: "We wanted to get paid for our brain work."

To equip Arrow, Cervinka and Pielsticker used a valuable currency—Cervinka's veteran status.

During World War II, the U.S. government used a lot of manufacturing machinery. Afterward, it no longer needed the machines, so it started to sell them.

Cervinka's veteran status meant great advantages: veterans were treated as first in line for the machines, which they could buy at very low prices, with just a down payment. They had five years to pay the balance, but they also could each receive a 10-year loan of $25,000.

Thus, Arrow acquired several Gleason Works machines and opened shop in a small industrial building in Worth, just southwest of Chicago.

Arrow then faced the problem of new businesses—finding customers.

"We knew where they were," Cervinka says candidly, "they

Arrow Gear Co.

Established: 1947
No. of Employees: 175
Size of Factory: More than 145,000 sq. feet
Industries Served: Aerospace, Robotics, Printing, Off-Road Equipment
Major Products: Spiral bevel gears, Straight bevel gears, Spur gears, Helical gears, Gear drives
Quality Registrations: ANSI-RAB QMS, AS9100, ISO9001-2000, and NADCAP
Industry Affiliations: AGMA, ASM International, ASME International
Website: ***www.arrowgear.com***

Arrow Gear Co. occupies more than 145,000 square feet in Downers Grove, a western suburb of Chicago, IL.

Spiral Bevel Contact Patterns—Right the First Time

What's the key piece of Arrow's highly accurate, closed-loop system for making spiral bevel gears?

"All the software," gear engineer Tom Mifflin says. "All that software is what makes everything work."

Mifflin should know; he manufactures spiral bevel gear teeth using the system, which can create aerospace-quality spiral bevel gears with contact patterns that are right the first time.

A 31-year employee, Mifflin is in charge of gear-tooth manufacturing and contact pattern design for Arrow.

The system itself consists of two Gleason Corp. machines, a Phoenix® gear cutter and a Phoenix gear grinder, and a Zeiss/Höfler coordinate measuring system. These blue and gray machines are scattered throughout the factory, sitting among green gear generators and gear checkers.

Also, the system uses Gleason's CAGE, T-900 finite element analysis (FEA), and G-AGE inspection analysis software.

The software brings the scattered machines together off the factory's cement floor, to computer screens in Mifflin's carpeted office.

Arrow bought the system's pieces over about 10 years. The last piece, the FEA software, was bought in 2001. Before then, the system had been operational for years, using loaded tooth contact analysis (TCA) software to design contact patterns.

Even without the FEA software, though, Arrow twice manufactured spiral bevel gears with first-time, right-on contact patterns for a jet engine project in the late 1990s.

The project started in 1998. Arrow had to manufacture two spiral bevel gear sets for the commercial aircraft jet engine. One set was for the tower shaft, the other for the accessory gearbox.

"The bevel gears manufactured by Arrow Gear performed excellently," Roger Levine says. "The contact pattern was excellent, right where it was supposed to be."

During the project, Levine was a staff engineer on mechanical systems at Pratt & Whitney Aircraft in East Hartford, CT. He was also leader of the team for the project's tower shaft and main gearbox and was mainly involved in designing the spiral bevel gears.

Levine says the contact pattern was "right on the money with the first set of gears." He adds that getting the pattern right the first time isn't a normal thing in his experience. Levine has helped design and build five spiral bevel gear sets for aircraft jet engine projects during his 30-year career.

Arrow also manufactured the spiral bevel gears for the accessory gearbox. That gearbox was being provided by Sundstrand Corp.

From his office computers, gear engineer Tom Mifflin manufactures spiral bevel gear teeth using Arrow's closed-loop system for manufacturing spiral bevel gears. Mifflin sees the software as the key to the system: "All that software is what makes everything work."

Mike Blewett, then a Sundstrand design engineer, was on the team that designed the accessory gearbox.

"We found that the patterns were pretty much exactly where Arrow predicted they would be," he says. "We never had to adjust the contact pattern ever since the first unit was shipped." o just weren't going to put their eggs in our basket."

At that point, Cervinka went looking for a gear order—at D.O. James. The company remembered him and was willing to take a chance on his new business.

Arrow started making spiral bevel gears for D.O. James' speed reducers. Cervinka recalls he and Pielsticker could provide their gears on-time and at a reasonable price—and they were better than what D.O. James had been getting.

The company stayed with Arrow for more than nine years. And Cervinka admits its gear orders really helped Arrow during its first years, giving it time to grow—"When you start, anything is blessed."

Today, Arrow occupies a building with more than 145,000 square feet, in Downers Grove, one of Chicago's western suburbs. A medium-sized gear manufacturer, Arrow employs 175 people; is equipped to design, manufacture, heat treat and inspect gears and to manufacture gearboxes; and has more than 500 customers.

At 56 years old, Arrow remains a Cervinka/Pielsticker business. Pielsticker died in 1987, but Cervinka, now 83, continues as Arrow's CEO and chairman of its board of directors.

Pielsticker's sons, James and Frank J., work at Arrow as executive vice president and materials director, respectively. Likewise, Cervinka has two sons, Mike and John, at Arrow. Mike is director of facilities and human resources, and John is assigned to special projects.

Arrow's president, however, is Joseph Arvin. Nearly 6 feet 3 inches tall, Arvin has the friendly reserve of a country gentleman. Raised in Indianapolis, IN, the industrial engineer has been with Arrow since 1972 and has been its president since 1987. He still retains the barest trace of a Southern accent—"You get that in Indianapolis," he says. "It was a lot worse before I came up here."

Today, Arrow is home to a highly accurate closed-loop system for manufacturing spiral bevel gears. The system can create aerospace-quality spiral bevel gears with contact patterns that are right the first time, saving development time for Arrow and its customers.

Also, the system has been on a streak. It's manufactured a spiral bevel gear set for five different projects: four aerospace and one automotive racing—the Indy car. In these consecutive projects, each of Arrow's gear sets had the right contact pattern the first time.

"That's what we're striving for," Cervinka says.

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On a Possible Way of Size and Weight Reduction of a Car Transmission

Stephen P. Radzevich

This study is focused on some features of the geometry and kinematics of gear hobbing operations. The principal goal is to determine the minimal hob idle distance that is required for complete generation of the gear teeth. This task is important in many aspects, especially for reducing the axial size of a hobbed shoulder-gear. Reduction of the axial size of a hobbed shoulder-gear leads to reduction of size and weight of the shoulder-gear itself and of the gear train housing. Developed by E. Buckingham, methods of analytical mechanics of gears are applied to determine an exact minimal allowed length of the gear hob idle distance. The results reported in the paper are applicable for manufacturing of spur and helical involute gears. Applying the results allows one to reduce the axial size and weight of a gear train and its housing. The consideration below is focused on hobbing of involute gears. Slightly modified results obtained are applicable for hobbing of splines, sprockets, ratchets and other form tooth profiles. The results obtained are important for the application of multi-start hobs of small diameter.

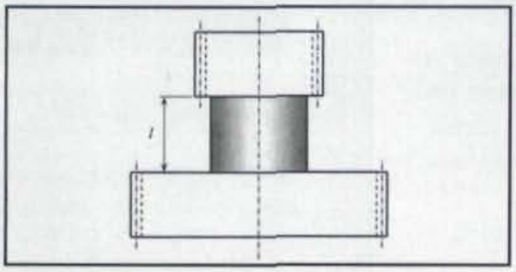

Figure 1—An example of a cluster gear.

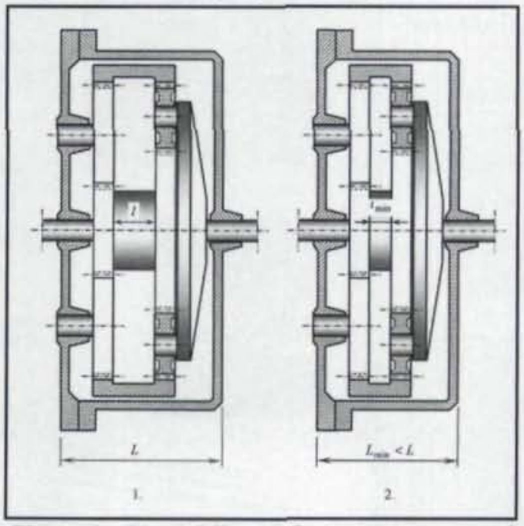

Figure 2—Fixed differential gear arrangement of Type D.

Introduction

Almost any external tooth form that is uniformly spaced around a center can be hobbed. Hobbing is recognized as an economical means of producing spur and helical gears with involute tooth profiles. Although the hobbing process is often associated with manufacturing of gears and splines, many other forms can be cut. Due to its versatility, hobbing is also recognized as an economic means of producing ratchets, sprockets and other special forms.

But a problem appears in mass production of cluster gears (Fig. 1), for example, in manufacturing transmissions in the automotive industry. For economic reasons, both gears in a cluster gear need to be hobbed and not machined with a gear shaper cutter. Productivity of gear hobbing operations significantly exceeds that of gear shaping operations. Moreover, hobbing of a cluster gear can be performed on the same hobbing machine and in the same set-up without work resetting. Therefore, hobbing of cluster gears does not waste time: a) for unloading the gear hobbing machine, b) for transporting gear blanks to a gear shaping machine, c) for installing gear blanks again on the gear shaper, and d) for solving a specific problem of gear facing. This last issue is important for manufacturing a multi-flow gear train, for example, in manufacturing planetary reducers, where torque to be transmitted is splitting on two or more flows.

However, the principal disadvantage of gear hobbing operations is that usually only cluster gears with long enough necks can be hobbed. Increasing length l of a neck increases a cluster gear's axial size, and thus increases the weight of the gear cluster itself. It also increases the weight of gear shafts and gear housings. In the case of $l=l_{min}$, size L could be significantly reduced to L_{min} (Fig. 2). This size reduction corresponds to a reduction of size and weight in a car transmission. The example above (Fig. 2.1) is taken from References 3, 5, and 11. The example discussed above clearly indicates the necessity of developing methods of hobbing cluster gears with the shortest neck, i.e., cluster gears with $l=l_{min}$.

In the paper, notation of parameters recommended by ANSI/AGMA (Ref. 1) is used.

Literary Survey

A few directions of hob feed are recognized. Downward feed direction (in conventional gear hobbing) and upward feed direction (in climb gear hobbing) are mostly used.

The reduction of total hob travel distance leads to a reduction of total gear hobbing time and is widely recognized. The parameters of a gear to be machined are given and a gear manufacturer cannot change them. Gear manufacturers cannot reduce gear face width. A total hob travel distance can be reduced if one gets to know the shortest allowed length of approach distance and the shortest allowed hob idle distance. In this paper, the shortest allowed hob idle distance is considered.

A hob rotates about its axis (Fig. 3), with a constant angular velocity ω_h. Simultaneously, the hob moves along the work axis in the feed direction (or in the opposite direction in a conventional gear hobbing operation).

Cutter total travel distance AD (Fig. 3) is equal to the sum of the hob approach distance AB, face width BC, and hob idle distance (or hob overrun) CD.

The hob approach is the distance that the hob has to travel parallel to the gear axis, from the point of initial contact between the hob and the point where center distance reaches the first gear face (at the point B). The hob approach distance can be computed by knowing the appropriate formula, which is available in Reference 4.

The gear face width is indicated on the part print. Its length cannot be changed by gear manufacturers.

The hob idle distance, a distance required to complete generating of the entire set of gear teeth, is the linear hob carriage travel beyond the second gear face C.

There is no complete understanding of what the hob idle distance exactly is and how to compute it minimally, allowing length for hobbing a given cluster gear. For example, in accordance with recommendations given by hob manufacturer FETTE GmbH: "No idle distance, except for a safety allowance, is required for straight teeth" (Ref. 4).

Impact of the Hob Idle Distance on Axial Size of a Hobbed Cluster Gear

The length of the hob idle distance is not equal to the length of the hob approach distance (Fig. 4).

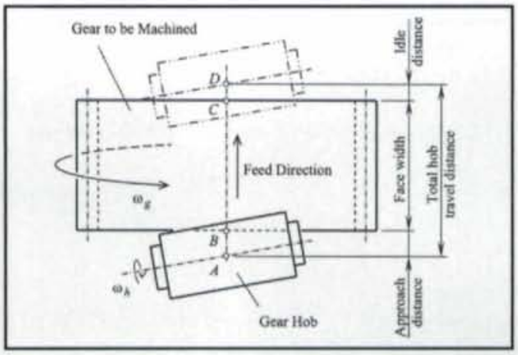

Figure 3—Kinematics of climb hobbing of an involute gear.

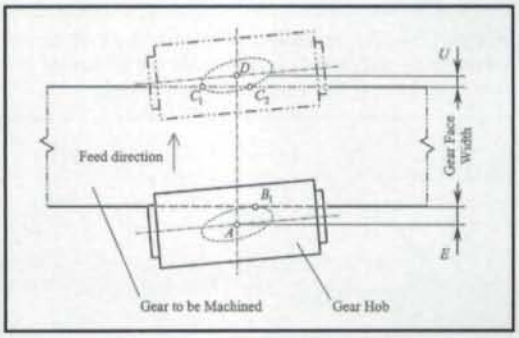

Figure 4—Comparison of length of the hob approach distance E and of the hob idle distance U.

Length of the hob approach distance is equal to the height of the highest point, B_1, on the penetration curve above the horizontal plane through the center distance—through the intersection of the cutter and gear axes (Fig. 4). When the hob is away from the upper face of the gear, the penetration curve intersects the upper face at points C_1 and C_2. This makes it evident that the length of the hob idle distance U is always shorter than the length of the hob approach distance E. In order to reduce axial size, hobbing cluster gears usually starts from the open end, where there is enough room for hob approach distance, and finishes at the opposite end, where there is a lack of space for the hob.

The hob idle distance U affects the axial size of a hobbed cluster gear and a hobbed shoulder-gear as well. When a longer idle distance U is required, a longer cluster gear neck l_{min} is necessary, and the axial size of the cluster gear becomes larger and vice versa (Fig. 4). As is evident in Figure 4, the minimal length of a cluster gear neck l_{min} depends on the hob idle distance U, i.e. $l_{min} = l_{min}(U)$.

To cut gear hobbing time and minimize the axial length of a cluster gear, the hob idle distance must be as short as possible. At the same time, the length of the hob idle distance must be sufficient for completing generation of the entire gear tooth surface. As mentioned above, in gear hobbing

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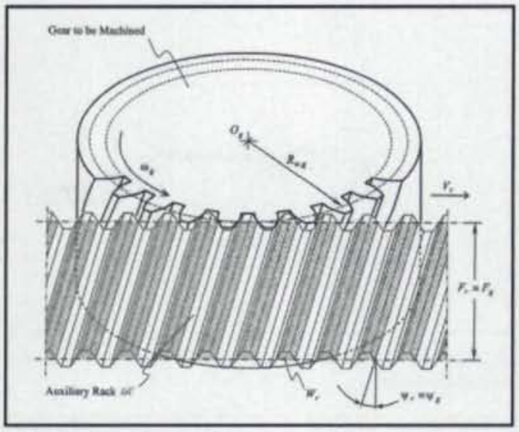

Figure 5—The auxiliary phantom rack R as an enveloping surface to consecutive positions of the tooth surface of the gear to be machined.

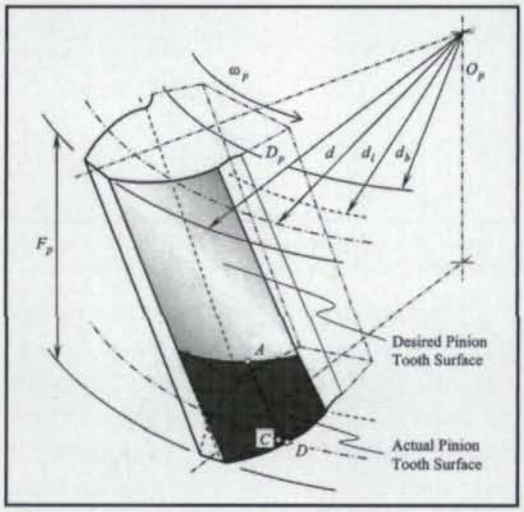

Figure 6—Deviation of an actual gear tooth surface from the desired gear tooth surface caused by lack of the hob idle distance.

operations, the gear blank is rotating around its axis O_g with a constant angular velocity ω_g. The auxiliary phantom rack R is traveling with a constant speed V_r. The velocities ω_g and V_r are synchronized in the manner that the auxiliary rack R pitch surface W_r is rolling without sliding over the gear pitch cylinder of radius $R_{w.g}$. For machining the entire length of the gear, it is required that the gear hob generates the auxiliary rack R of width F_r that is equal to or exceeds width F_g of the gear to be machined (Fig. 5).

A lack of length of the hob idle distance ΔU causes deviation of the actual tooth surface from its desirable shape (Fig. 6). The length of ΔU depends only on the relative location of the gear hob in the axial direction of the gear to be machined. It does not depend on hob diameter, while maximal deviation δ^* of the actual gear tooth surface from the nominal gear tooth surface does.

The initial solution to the problem is enhanced and developed here. A numerical example is also provided.

Impact of a Gear Hob-Setting Angle on the Hob Idle Distance

It is convenient to evaluate the impact of a gear hob-setting angle on the hob idle distance by considering a simple degenerated case, like when the hob-setting angle ζ_h is equal to zero ($\zeta_h = 0°$). In machining a spur gear, this occurs when $\zeta_h = 0°$, the crossed-axes angle Σ, i.e. the angle that makes axes of the gear O_g and of the hob O_h, is equal to a right angle ($\Sigma = 90°$).

For climb hobbing (Fig. 7.1), the hob idle distance U is measured from the upper face of the gear to be machined. Generating the gear tooth profile with the hob begins the instant that straight generatrix $E_1^{(r)}$ of the recessing flank of the hob tooth profile in its horizontal position reaches the lower face of the work (Fig. 7.2). When the hob travel distance exceeds the length $l_1 = d_{b.h}$ (here $d_{b.h}$ is the hob base diameter), the opposite flank of the hob tooth profile enters the gear profile. In this case, both flanks of the gear tooth are generated simultaneously. Generating the gear tooth profile accomplishes this in an instant, when the straight generatrix $E_1^{(a)}$ of the opposite flank (of the approaching flank) of the hob tooth in its horizontal position reaches the upper face of the work (Fig. 7.2). In the case under consideration, the length of the hob idle distance U is equal to half a base diameter of the hob, i.e $U = 0.5d_{b.h}$. This is the shortest allowed hob idle distance. To accomplish the generation of the gear tooth profile, it is required that the cutter travel distance is equal to or exceeds $L \geq F_g + d_{b.h}$. It is important to point out here that the shortest allowed hob idle distance could not be less than $0.5d_{b.h}$.

The above approach (Fig. 7.2) can be utilized when the gear hob-setting angle is not equal to zero and this angle is positive. In this case, the straight generatrices $E_2^{(a)}$ and $E_2^{(r)}$ (Fig. 7.3) are similar to the straight generatrices $E_1^{(a)}$ and $E_1^{(r)}$ in Figure 7.2, additionally shifted on the distance δ relative to each other along the gear axis. The value of the parameter δ depends on the length of the gear tooth profile-generating zone and on the gear hob-setting angle. This question is explained in more detail below. Distance δ increases the length of the hob idle distance U and the total hob travel distance, which is necessary for accomplishing the generation of the gear tooth profile.

It is evident that, in general, Figure 7.3 provides us only with qualitative results. It cannot be applied for computing the length of the hob idle distance in quantities.

When reducing the axial length of a cluster gear or hobbing an involute gear with any given value of pitch helix angle ψ_g it is preferable to design something that enables a crossed-axes angle $\Sigma = 90°$ with a gear hob. To continue the analysis, it is required to clearly understand the gear hob-setting angle ζ_h.

While generating the machining surface of a gear hob, the auxiliary rack *R* performs a screw motion with the hob axis as the axis of the screw motion. First of all, the axis of rotation of a cylindrical hob is parallel to the auxiliary rack pitch surface W_r. Secondly, orientation of the hob axis of rotation may vary relative to the pitch plane W_r (Fig. 8).

The idea of the gear hob-setting angle ζ_h can be traced back to a publication by E. Buckingham (Ref. 2).

The auxiliary rack tooth makes an angle ψ_r with the gear axis O_g. Very often, this angle is called the auxiliary rack pitch helix angle (Ref. 1). The gear hob axis of rotation intersects the center distance *C*. In Figure 8, the center distance is depicted as a point C of intersection of the center distance and of the auxiliary rack pitch surface W_r.

The straight line $^{\circ}O_h$ (parallel to the plane surface W_r) through the point C is orthogonal to the auxiliary rack tooth. For a spur gear, it is parallel to the gear face. The gear hob axis of rotation makes a certain angle with the straight line $^{\circ}O_h$. The gear hob axis of rotation $^{+}O_h$ makes a positive angle $+\zeta_h > 0°$ with the straight line $^{\circ}O_h$. The gear hob axis $^{-}O_h$ makes a negative angle $-\zeta_h < 0°$ with the same straight line $^{\circ}O_h$.

We suggest naming the angle ζ_h the gear hob-setting angle (Ref. 7). It is necessary to stress here that the gear hob-setting angle ζ_h is a parameter of a hob design. Its value does not have to equal the value of the gear hob lead angle $\lambda_h = 90° - \psi_h$. However, due to the restriction explained below, the difference between the angles ζ_h and λ_h cannot be significant. The equality $\zeta_h = \lambda_h$ may or may not take place. For example (Fig. 9), the same spur gear with a given parameter of its design can be machined with a few gear hobs of various hob-setting angles ζ_h, but with the same hand of hob lead angles.

Usually, the gear hob-setting angle is positive ($\zeta_h > 0°$) and is approximately equal to the hob lead angle λ_h. In this case, crossed-axis angle $\Sigma = 90° - \zeta_h$ is less than 90° (Fig. 9.1).

The method of spur gear hobbing with right crossed-axis angle $\Sigma = 90°$ is known. In this case, the gear hob-setting angle is equal to zero (Fig. 9.2). The same method can be applied to hobbing a helical gear. In the last case, the value of the crossed-axis angle can be computed by the formula $\Sigma = 90° - \psi_g$.

Finally, an involute gear can be machined with a hob with negative hob-setting angle: $\zeta_h < 0°$ (Fig. 9.3). In this case, crossed-axis angle $\Sigma = 90° + \zeta_h$ exceeds 90°. It is important to stress that such an opportunity for gear hobbing operations exists mostly theoretically. The top cutting edge of a gear hob tooth with negative hob-setting angle ζ_h becomes too short or has negative length. For this reason, the gear hob with $\zeta_h < 0°$ does not have a wide application in contemporary manufacturing

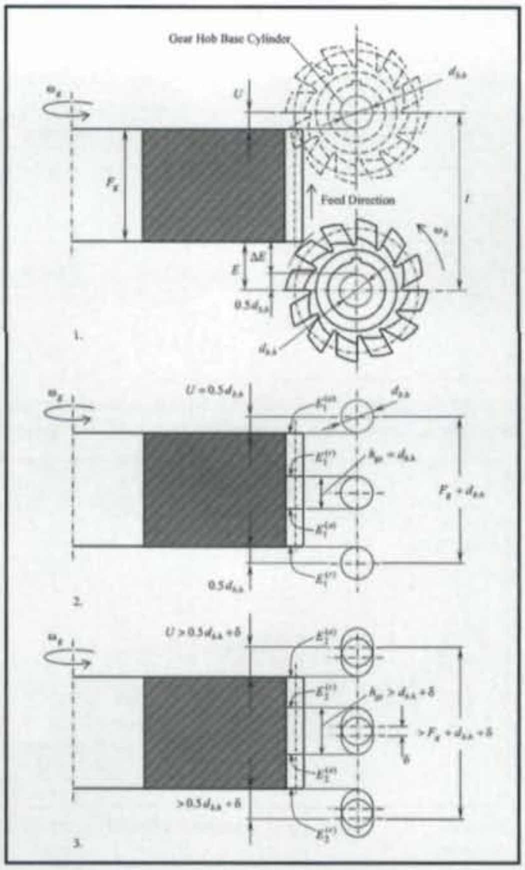

Figure 7—The impact of gear hob-setting angle on the length of the hob idle distance in climb hobbing of involute gears.

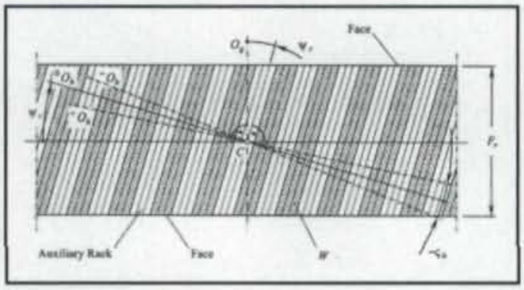

Figure 8—Front view on the auxiliary phantom rack* R *of an involute hob (see Fig. 5).

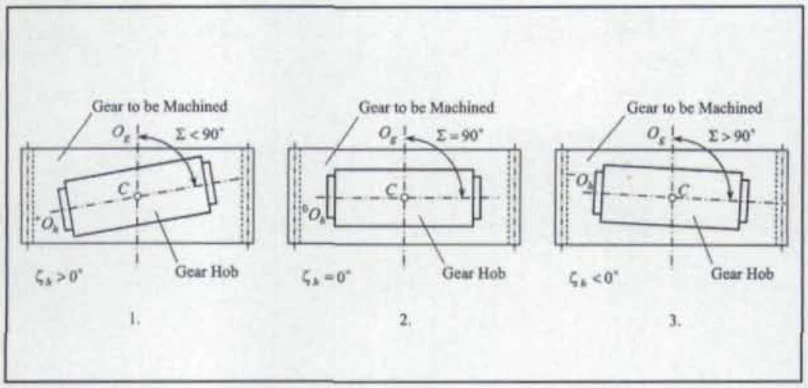

Figure 9—Relative orientation of the spur gear to be machined and of the gear hob with the gear hob-setting angle ζ_h 1-positive, 2-equal to zero, and 3-negative.

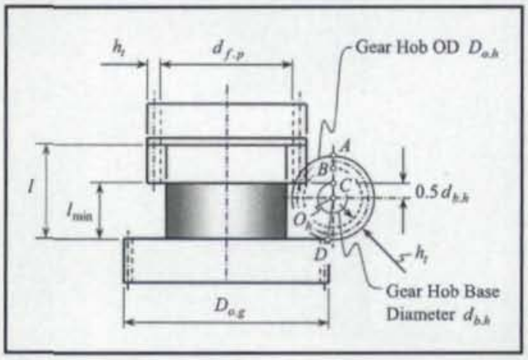

Figure 10—Computing minimal cluster gear axial length.

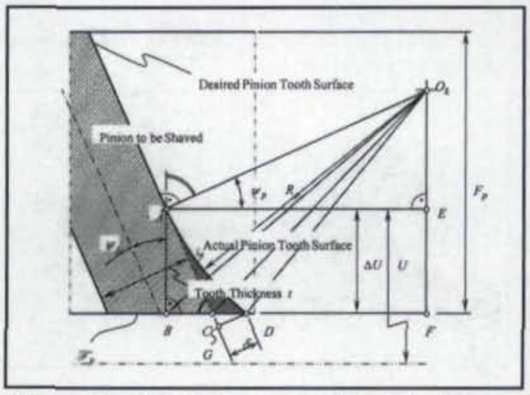

Figure 11—Unfolded section of the gear in Figure 6 by the pitch cylinder of diameter d that is co-axial to the gear. Letters "a", "c" and "d" in Figure 6 correspond to the same points in Figure 11.

of gears.

For machining a spur gear, it is required to maintain the gear hob-setting angle $\zeta_h = 0°$ (Fig. 9). In this case, the gear hob helix angle becomes very small, ψ_h approaching 0°. However, the angle ψ_h remains positive ($\psi_h > 0°$).

For machining a helical gear with a certain helix angle ψ_g, it is required to maintain the gear hob-setting angle $\zeta_h = -\psi_g$. In both cases, this yields the cross-axis angle $\Sigma = 90°$. The difference $(\psi_h - \psi_g)$ is equal to 0°. The hand of the gear hob start is opposite to the hand of the gear start.

Figure 9 clearly illustrates that the hob lead angle λ_h, and the gear hob-setting angle ζ_h have to be considered as two completely different parameters of the design of the gear hob. In some cases, the parameters of a gear hob design, λ_h, and ζ_h can be equal (or almost equal) to each other. In general, this requirement is not mandatory. In the example above (Fig. 9) the hand of the gear hob lead angle λ_h remains the same for all three cases, and for all of these cases it is positive ($\lambda_h > 0°$). However, the value of the lead angle λ_h for each of the three cases does not remain the same. The hob-setting angle ζ_h also can vary. It can be positive (Fig. 9.1), equal to zero (Fig. 9.2), or even negative (Fig. 9.3) as well. Under such a variation, the value of the lead angle λ_h also varies, but its hand remains the same. The sign of the lead angle λ_h does not change.

Determining Parameters of a Gear Hob for Machining the Shortest Cluster Gear

Gear hob outside diameter is the most critical issue in machining a cluster gear with the hob when the crossed-axes angle $\Sigma = 90°$. Usually, a gear hob outside diameter $D_{o.h}$ can be expressed as: $D_{o.h} = 2\,(AB + BC + CO_h)$ (Fig. 10). This formula yields an expression for $D_{o.h}$ which is explained further in Equation 1 in the box on page 49.

In Equation 1, whole tooth depth is designated as h_t (Ref. 1).

Aiming for a reduction of size and weight of a car transmission, the portion BC of the hob outside diameter has to be eliminated. It is required that $BC \equiv 0$. Under such an assumption, Equation 1 reduces to a formula shown as Equation 2 in the box.

Hob base diameter $d_{b.h}$ can be computed from the formula (Refs. 7, 9 and 10) shown in Equation 3 in the box; note that:

m is the gear hob modulus;

Z_h is the number of the gear hob starts;

ϕ_n is the gear hob normal pressure angle.

Further reduction of the hob outside diameter $D_{o.h}$ is restricted with the necessity of satisfying the first condition of part surface generation (Ref. 8). In cases, $D_{o.h} < 2h_t + d_{b.h}$, first necessary condition of proper part surface generation is not satisfied, and thus, the gear tooth cannot be machined in accordance to the gear drawing.

Figure 10 reveals that for the cases of right crossed-axis angle $\Sigma = 90°$, length l of the cluster gear neck can be calculated from the formula in Equation 4.

In Equation 4:

$D_{o.g}$ is the outside diameter of the cluster gear's larger gear, i.e. of the shoulder (see Fig. 10);

d_f is the form diameter of the cluster gear's smaller gear, i.e. of the pinion (see Fig. 10).

In all cases when $d_{f.p} + D_{o.h} \leq D_{o.g}$, length l of the cluster gear neck is equal to $l = 0.5(d_{b.h} + D_{o.h})$.

The results yield the resultant formula for l, shown in Equation 5.

To ensure right crossed-axes angle $\Sigma = 90°$, the gear hob-setting angle is assigned equal to the gear helix angle, but with the opposite sign, i.e. $\zeta_h = -\psi_g$. For computing the minimal length l_{min} of the gear cluster, it is required to resolve the set of two equations with two unknowns:

$$\frac{\partial l}{\partial Z_h} = 0 \text{ and } \frac{\partial l}{\partial \phi_n} = 0$$

Equation 5 yields both of these equations. The number of gear hob starts Z_h, calculated from the above set of two equations, usually is a fraction. It is required to approximate it to the nearest integer number, and afterwards to recalculate the final value of the corresponding hob normal pressure angle ϕ_n. In cases where the calculated value of Z_h is close to mid-interval of two consequent integer numbers, it is required to recalculate the value of the corresponding hob pressure angle for both cases. Afterwards, it is necessary to select that value which ensures a longer gear hob top cutting edge.

Another way for computing l_{min} could be developed. The number of starts Z_h of a gear hob could be assigned several consequent integer values: $Z_h = 1$, $Z_h = 2$, and so on. Furthermore, it is required to resolve the equation

$$\frac{\partial l}{\partial \phi_n} = 0$$

for each integer value of Z_h. Finally, it is required to select the smallest possible Z_h. It is also strongly preferred that the difference between the gear hob normal pressure angle ϕ_n and the hob normal pressure angle would be the smallest possible. This results in the longest top cutting edge of the hob.

Impact of Tolerance on Length of the Hob Idle Distance

Figure 11 shows the unfolded section of the gear (Fig. 6) by the pitch cylinder of diameter d, which is co-axial to the gear. Points A, C, and D in Figure 6 correspond to the same points A, C, and D as in Figure 11.

In order to complete the generation of a perfect gear tooth in its longitudinal direction, it is required that the gear hob overrun would be equal to or would exceed the hob idle distance U.

EQUATIONS

$$D_{o.h} = 2h_t + 2BC + d_{b.h} \tag{1}$$

$$D_{o.h} = 2h_t + d_{b.h} \tag{2}$$

$$d_{b.h} = \frac{m \cdot Z_h \cdot \cos\phi_n}{\sqrt{1 - \cos^2\phi_n \cdot \cos^2\zeta_h}} \tag{3}$$

$$l = 0.5\sqrt{2 \cdot D_{o.h} \cdot (D_{o.g} - d_{f.p}) - (D_{o.g} - d_{f.p})^2} \tag{4}$$

$$l = \sqrt{(D_{o.g} - d_{f.p}) \cdot \left(h_t + \frac{m \cdot Z_h \cdot \cos\phi_n}{2 \cdot \sqrt{1 - \cos^2\phi_n \cdot \cos^2\zeta_h}} - D_{o.g} + d_{f.p} \right)} \tag{5}$$

$$\Delta U(\delta) = \frac{-[\delta] \cdot \cos\phi_n \cdot \tan\psi_p + \sqrt{[\delta]^2 \cdot (\cos\phi_n)^2 \cdot (\tan\psi_p)^2 - \left[[\delta]^2 \cdot \left(\frac{\cos\phi_n}{\cos\psi_n} \right)^2 - 2 \cdot R_p \cdot [\delta] \right]}}{\cos\psi_p} \tag{6}$$

$$\Delta U(\delta) \cong \frac{\sqrt{2 \cdot R_p \cdot [\delta]}}{\cos\psi_p} \tag{7}$$

$$R_{1.h} = \frac{\sqrt{d^2 - d_{b.h}^2}}{2\sin\psi_h \sin\phi_n} \cot\lambda_{b.h} = \frac{\sqrt{d^2 - d_{b.h}^2}}{2\sin\psi_h \sin\phi_n} \tan\psi_{b.h} \tag{8}$$

$$\tan\psi_{b.h} = \frac{\sqrt{\sin^2\phi_n + \tan^2\zeta_h}}{\cos\phi_n} \tag{9}$$

$$d_{b.h} = \frac{3 \cdot 5 \cdot \cos 14.5°}{\sqrt{1 - \cos^2 14.5° \cos^2 12.4°}} = 44.623\ mm \tag{10}$$

$$\tan\psi_{b.h} = \frac{\sqrt{\sin^2 14.5° + \tan^2 12.4°}}{\cos 14.5°} = 0.344 \qquad \psi_{b.h} = \tan^{-1}(0.344) = 18.992° \tag{11}$$

$$\psi_h = \tan^{-1}\left(\frac{d}{d_{b.h}} \cdot \tan\psi_{b.h} \right) = \tan^{-1}\left(\frac{70}{44.623} \cdot 0.344 \right) = 28.365° \tag{12}$$

$$R_p \equiv R_{1.h} = \frac{\sqrt{70^2 - 44.623^2}}{2\sin 28.365° \sin 14.5°} 0.344 = 78.025\ mm \tag{13}$$

The hob overrun must not be less than the hob idle distance U. In cases where the actual hob idle distance is shorter than or equal to U, the actual longitudinal profile AD of the gear tooth deviates from its desirable longitudinal profile AC (Fig. 11). Maximal deviation occurs at the point D, at which deviation is equal to δ. It is required that the maximal deviation δ does not exceed tolerance $[\delta]$ on the gear tooth accuracy. The length ΔU depends on the relative location of the work and on the involute hob in axial direction of the work.

Incorporating a tolerance on gear tooth longitudinal profile yields a significant reduction of the length of the required gear hob idle distance. Analysis of Figure 11 yields a formula for computation of the allowed shortening of the hob idle distance that is shown in Equation 6.

For engineering computation, a simplified approximate formula is valid and presented in Equation 7.

In that equation, $R_p \equiv R_{1.h}$ is the first principle radius of curvature of the machining surface of the involute hob.

It is already proven (Refs. 9, 10) that the value of the first principle radius of the machining surface of the involute hob can be computed. The computation method is explained in Equation 8.

Base helix angle $\psi_{b.h}$ can be calculated from the formula laid out in Equation 9.

For example, for the left-hand involute gear hob of modulus $m = 3$ mm with pitch diameter $d = 70$ mm, normal profile angle $a_n = 14.30°$, number of starts $Z_h = 5$, and hob-setting angle $\zeta_h = 12.4°$, one can compute the hob base diameter by using the formula in Equation 10.

The hob base helix angle is equal to two formulas listed in Equation 11.

The hob helix angle (Ref. 6) can be found in Equation 12.

The above results yield a final calculation shown in Equation 13.

That result yields $U = 28.471$ mm. In the case, with the tolerance equal to $\delta = 0.1$ mm, one can compute that $\Delta U = 4.561$ mm, and therefore the actual hob idle distance required for perfect involute gear tooth profile generating is equal to $U^* = 28.471–3.949 = 23.910$ mm. Hob idle distance $U^* = 23.910$ mm is 19.1% shorter than the hob idle distance $U = 28.471$ mm. This reduces the length of the hob idle distance that leads to corresponding reduction of the size and weight of a cluster gear and of the gear train housing as well.

Conclusions

Methods of analytical mechanics of gears are applied to determine the exact minimal axial length of a cluster gear for conventional and climb hobbing of spur and helical gears. The results reported in the paper allow users to cut hobbing time and to reduce the size and weight of a gear train and its housing. The approach is especially important for applying the multi-start hob of small diameter. A similar approach can be utilized for hobbing of a non-involute profile, for instance, while machining splines, sprockets, ratchets, etc. The results presented might be incorporated as a part into software for CNC hobbing machines. ○

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Products for the Gear Industry

New Harmonic Drive Gearing Components from HD Systems

The CSD Series Size 40 from HD Systems delivers low backlash, 1 arc-minute positional accuracy and +/–5 arc second repeatability.

With outer diameters of 135 mm, the gearing components are 27 mm in length. The rated torque is 1,825 in.-lbs. and maximum torque is 6,160 in.-lbs. This series is available in gear ratios of 50, 100 and 160:1

For more information, contact HD Systems of Hauppauge, NY, by telephone at (800) 321-4374 or on the Internet at *www.HDSI.net*.

New Right Angle Gearmotors from Bison

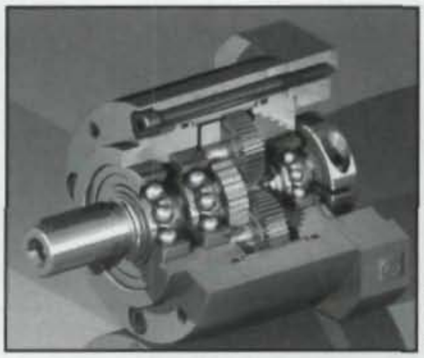
The right angle, hollow shaft gearmotor line from Bison Gear & Engineering has several mounting configurations, including base, end, side flange, shaft mount and pilot.

Shaft options include a 3–4" hollow, along with double and single extension 3/4", 5/8" and 1/2" diameter shafts with keys.

According to the company's press release, the gearmotors all have ball bearings and are lubricated with synthetic oil for maximum efficiency. Three AC motor options and four DC motor options span a horsepower range from 1/8–1/4 horsepower. Gear ratios range from 5:1–60:1.

The motors run at 1,800 rpm for low noise and low temperature rise.

For more information, contact Bison Gear & Engineering of St. Charles, IL, by telephone at (800) 282-4766 or on the Internet at *www.bisongear.com*.



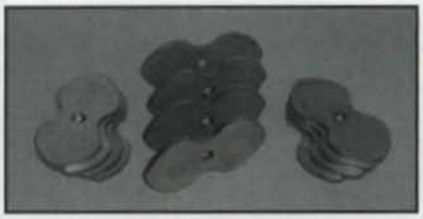
New Planetary Gearheads from ZF Great Britain

The PGE planetary gearheads from ZF Great Britain are compatible with any machine equipped with AC or DC servomotor applications, according to the company's press release.

The gearheads offer low weight and noise, a small size, and up to 97% efficiency. Positioning accuracy and reliability are a result of the 10 arc-minute backlash design and even load distribution between three planetary gears.

The design includes hardened and optimized gearing geometry and bearings. Nominal service life is typically 10,000 hours. Grease is used as a lubricant instead of oil, and therefore the gearheads can be mounted flexibly in any orientation on a machine.

For more information, contact ZF Great Britain of Nottingham, U.K., by e-mail at *rob.pearson@zf-group.co.uk* or online at *www.zf-group.co.uk*.



Spur Pinions from Trogetec

The minimal-sized epitrochoid (epicycloid) spur pinions from Trogetec are designed for partial conjugacy internal mesh with roller wheels.

Adaptable to many gearing applications, the pinions are used in parallel shaft drives, planetaries, star gearing, and cyclos.

According to the company's press release, blanks are CNC cut in seven-gage stock to +/–0.003" tolerances relative to the applicable nominal contour.

For more information, contact Trogetec Inc. of Riverton, WY, by telephone at (307) 856-0579 or by e-mail at *sales@trogetec.com*.

Servo Gearheads from Neugart USA

The PLS series of in-line planetary servo gearheads from Neugart USA is designed for low backlash and optimized with gearing machined to high precision, hardened and honed.

According to the company's press release, the low backlash will not increase during the unit's lifetime and the gearheads have the highest torque density in the industry. The company adds the gearheads are suitable for all mounting positions without modification. The gearheads' high stiffness reduces the chances of lost motion.

Additional options include a solid free output shaft, foot mounting plate, housing mount option and spline output shaft.

For more information, contact Neugart USA of Bethel Park, PA, by telephone at (412) 835-4154 or by e-mail at *sales@neugartusa.com.*

Surface Roughness Tester from Mitutoyo

The Surftest SJ-301 from Mitutoyo integrates features from bench-top surface roughness testers into a portable instrument.

The tester uses memory cards to store measuring conditions, measured data and statistical results. With a differential inductance detection, a dedicated processor including statistical process control (SPC) functions, plus a large display and built-in high speed thermal printer, the tester makes it possible to conduct comprehensive measurements anywhere in the plant.

With a detector/drive unit and display unit, the tester can measure 36 surface roughness parameters, according to the company's press release.

For more information, contact Mitutoyo America Corp. of Aurora, IL, by telephone at (630) 978-5385 or by e-mail at *info@mitutoyo.com.*

Updated CNC Hobbing Machines from Bourn & Koch

The 200 H and 400 H model CNC hobbing machines from Bourn & Koch Machine Tool Co. have been updated.

According to the company's press release, new features include a 7-axis CNC control, an added hob swivel and tailstock, a standard full enclosure, a flat-screen CNC control panel and a redesigned hob swivel and slide assembly. The standard version requires no hydraulics.

For more information, contact Bourn & Koch of Rockford, IL, by telephone at (815) 965-4013 or on the Internet at *www.bourn-koch.com.*

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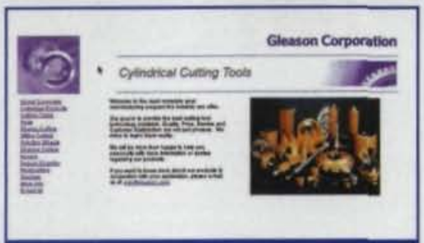

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To Climb A Mountain, A Railroad Needs Gears

Recently, the Addendum team has taken a keen interest in a Swiss mountain. Being the Addendum team, we haven't been interested in this rocky, fissured mountain for its natural majesty.

We've been interested in its geared railway, its very *steep* geared railway.

The mountain is Mount Pilatus, in central Switzerland, near Lake Lucerne. The railway is the Pilatusbahn. It climbs 1,689 vertical meters at an average grade of 42 percent and a maximum grade of 48 percent. At the higher grade, the railway's cars rise almost one meter for every two meters run.

For cars to make the climb, the railway uses a specially designed rack and cogwheel assembly. The design was so special ASME International designated the Pilatusbahn a historic mechanical engineering landmark in September 2001.

The design came from engineer Eduard Locher of Zürich in the late 19th century. His design consisted of a horizontal double rack.

The rack is centered between the rails, with a set of rack teeth facing each rail. Each set is engaged by a cogwheel—a gear. A pair of these wheels is mounted horizontally under each of the railway's cars.

Also, each cogwheel has a flange on its lower side. A horizontal disc, each flange extends under the rack itself, locking it to the rack.

Locher's design was special because it solved problems that had kept a railway from being built on Mount Pilatus. In honoring the Pilatusbahn as a landmark, ASME International researched the history of the railway.

In that history, a railway was proposed in 1873 using a rack and cogwheel design that had proven successful on other mountains. The design also located the rack between the rails, but its cogwheel was mounted vertically under the railway car. Driven by the car's locomotive, the cogwheel rotated, in effect climbing the rack.

The proposed railway wasn't possible. Its track gauge, maximum grade (25 percent), and passenger cars with separate locomotives would've resulted in a railway some 8,000 meters long, with curves too large for Pilatus' terrain and with rolling stock that was too heavy.

Also, Swiss authorities wouldn't permit a steeper grade because they worried high winds on Mount Pilatus would cause the cogwheel to "climb out" of engagement with the rack.

Locher's design solved those problems.

Mounted horizontally, his cogwheels wouldn't climb out of their rack, even on very steep grades. The cogwheel flanges also helped prevent climb out, even in high cross winds.

Other design differences lightened the railway's rolling stock and made the mountain route half as long as the 1873 route.

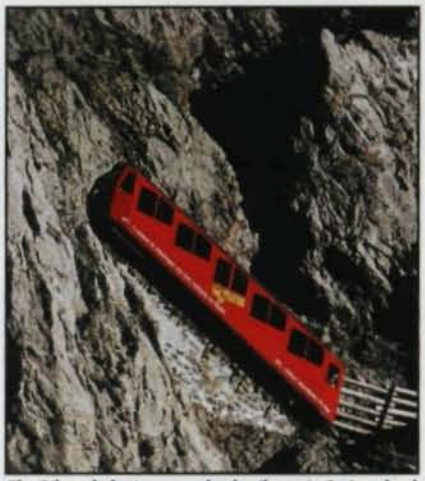
Construction of the railway started in April 1886. Operations started June 4, 1889.

More than 100 years later, the Pilatusbahn is still running. The railway is owned by PILATUS BAHNEN, a Swiss tourism company located in Kriens.

The railway transports an estimated 156,000 people every year up and down Pilatus' south side.

The people visit the top of Mount Pilatus, where two hotels are located, to view art exhibitions, take guided hiking tours, or sunbathe on the hotels' patios in summer and winter. They come to hold conferences and seminars in one of the hotels, the Hotel Kulm.

They also come for the experience of riding the railway.



The Pilatusbahn is a cogwheel railway in Switzerland and climbs 1,689 vertical meters to the top of Mount Pilatus. The steep climb has a maximum grade of 48 percent, or 21.6 degrees.

A segment of the Pilatusbahn's rack and cogwheel assembly is displayed as an outdoor sculpture at Alpnachstad, the railway's station at the foot of Mount Pilatus. (Photo courtesy of ASME International.)

To cover 1,689 vertical meters, the cars actually travel 4,594 meters of railway. The ride up takes 30 minutes; the ride down takes 40. Cars climb at 7–12 kilometers per hour and, for safety reasons, descend at a maximum of 9 kilometers per hour.

Besides the mountain's grade, Locher's design solved the problem of Pilatus' winds. Those winds don't always blow, but when they do: "They can go easily over 100 kilometers an hour," says André Zimmermann, PILATUS BAHNEN's chief executive officer. Even at such speeds, the railway can continue to run—"The system is that safe." ○

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